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**THE ELECTORRHEOLOGICAL EFFECT AND ITS APPLICATION TO
CONTROLLABLE DAMPER DESIGN**

by

Jamie J. Catania, B. Eng.

*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

Carleton University

Ottawa, Ontario

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Thesis Supervisor

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Abstract

Dampers which employ electrorheological (ER) fluid are examined theoretically and experimentally. Previous designs have shown decreased control near the second natural frequency of a quarter vehicle model test rig, possibly due to high shear rates. A new, adjustable flow rate damper was developed in an effort to overcome this problem. The new damper and a previous design were tested to evaluate damping characteristics. It was found that the new damper provided a wider range of force, but that this range was achieved at the expense of reduced force magnitude. Two configurations of the new design showed improved performance over the previous design, and were further tested using various control strategies in a quarter vehicle test rig. These strategies included: ride comfort control, wheel hop control, and a composite control strategy. Results from quarter vehicle testing, however, show little or no improvement in the high frequency performance of the new damper design. Evidence suggests that this may be due to transient behaviour of the fluid.

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Nomenclature

Symbol	Definition
c	damping coefficient
E	electric field strength
f_d	damper force
$f_{d,p}$	passive damper force
f_a	active actuator force
$f_{d,s-a}$	semi-active damper force
F	force
g	gravitational constant
h	lateral thickness from centre of electrode gap
h_o	half plug thickness
H	half electrode gap thickness
k_s	suspension spring stiffness
k_{tr}	tire stiffness
L	electrode length
m_s	sprung mass
m_{us}	unsprung mass
P	pressure
ΔP	pressure differential
ΔP_{zero}	zero electric field pressure differential
ΔP_{high}	high electric field pressure differential
Q	volumetric flow rate
Q_{zero}	zero electric field volumetric flow rate
Q_{high}	high electric field volumetric flow rate
v	velocity

v_o	velocity of plug
V	voltage
W	electrode width
$x_1, \dot{x}_1, \ddot{x}_1$	input displacement, velocity, and acceleration
$x_2, \dot{x}_2, \ddot{x}_2$	sprung mass (SDOF) or unsprung mass (2DOF) displacement, velocity, and acceleration
$x_3, \dot{x}_3, \ddot{x}_3$	sprung mass displacement, velocity, and acceleration (2DOF only)
α	dynamic/static tire deflection ratio
γ	shear
$\dot{\gamma}$	shear rate
ζ	damping ratio
κ	shear rate correction factor
μ_n	newtonian viscosity
μ_p	plastic viscosity
τ	shear stress
τ_y	yield stress
$\tau_{y, \text{stat}}$	static yield stress
$\tau_{y, \text{dyn}}$	dynamic yield stress
τ_e	excess shear stress
$\tau(0)$	ER fluid yield stress in response to zero electric field
$\tau(E)$	ER fluid yield stress in response to nonzero electric field
ω_n	undamped natural frequency

Chapter I

Introduction

Electrorheological (ER) fluids are suspensions of fine polarizable particles in an electrically insulating base oil. When subject to a transverse electric field in the order of kilovolts per millimetre, ER fluids demonstrate an instantaneous and reversible change in 'apparent' viscosity. Of the many uses proposed for these fluids, one is to provide variable damping characteristics for vibration control in mechanical systems. Dampers which exploit the variable nature of ER fluids are classified as providing *semi-active* performance.

This thesis examines the use of ER fluids in two semi-active dampers, hereafter referred to as Carleton University Transport Technology Research Laboratory (TTRL)

damper 'A' and damper 'B.' TTRL damper 'B', the successor to TTRL damper 'A', was designed by the author to improve upon what was considered a major drawback in the performance of damper TTRL 'A': it forced the ER fluid to operate at shear rates which are considered excessive for adequate damper performance.

Chapter II provides a brief introduction to ER fluids: modelling techniques, proposed causes for the ER effect, their drawbacks, and possible applications. Chapter III describes the performance of passive (conventional), active, and semi-active vibration control systems, with an emphasis on automotive application. The rheological performance of selected ER fluid is studied in Chapter IV with the aid of a Couette cylindrical rotary rheometer. Chapter V provides a summary of semi-active suspension design, along with a description of the methodology used in the design of TTRL damper 'B.' Chapter VI reports on the performance of both dampers when tested in a linear force versus velocity tester, and a quarter-vehicle simulator.

Chapter II

Electrorheological Fluids

2.1 The Electrorheological Effect

The electrorheological (ER) effect refers to the instantaneous and reversible modulation of the resistance to flow, or apparent viscosity, which can be induced in certain fluids through application of a transverse electric field. The effect is reversible and occurs in the order of milliseconds. According to Jordan and Shaw [1], in theory, "ER activity is predicted for any biphasic system wherein there is a substantial difference in permittivity between the dispersed phase and the medium." In general, most ER fluids consist of a polyelectrolyte particle suspension in an electrically insulating base oil.

2.2 History of the ER effect

The ER phenomenon was first termed the Winslow effect, after the American electrical engineer who first reported quantitative data on the subject in 1949 [2]. While work had previously been performed on evaluating the effect of an electric field on the viscosity of pure liquids, Winslow was the first investigator to describe experiments on fluids with suspended particles. Winslow patented many applications for his fluids [3,4,5,6], but interest waned in ER development, primarily because Winslow's fluids were highly abrasive and produced relatively little ER effect.

Research in ER was renewed by Klass and Martinek [7,8] who showed that the phenomenon is field rather than voltage dependent and that there was a strong shear stress/shear rate interdependence. Klass [9], also received patents for some ER fluid applications, including an 'electroviscous shock absorber'. In 1967, the UK Ministry of Defence began funding ER technology and as a result, Bullough [10,11,12,13,14,15,16], and Stangroom [17,18,19,20], have been significant contributors to ER fluid and application research. Since then, it has been recognized that substantial enhancement of the capabilities of ER fluids is required for more widespread application, and so research has focussed primarily on investigating the underlying mechanism or mechanisms, in an effort to enhance the ER effect.

2.3 Rheological Behaviour

2.3.1 Bingham Plastic Model

ER fluid behaviour is most commonly represented by a Bingham Plastic model.

The expression relating shear stress and shear rate for a Bingham plastic is:

$$\tau = \tau_y + \mu_p \dot{\gamma}. \quad (2.1)$$

In the absence of any imposed electric field, E , the bulk of ER fluids behave essentially like Newtonian fluids, demonstrating a zero or near zero yield stress (τ_y), constant plastic viscosity (μ_p), and a shear stress (τ) vs. shear rate ($\dot{\gamma}$) relation such as line 'A' in Figure 2.1. Upon excitation ($E > 0$), the fluid develops a static yield stress, τ_y , that is often proportional to the square of the electric field strength, while the plastic viscosity remains essentially unchanged. Flow occurs in the ER fluid only after the yield stress has been exceeded.

2.3.2 Yield Stress

As a result of the presence of yield stress in ER fluids, their response can be discussed in terms of two rheological regimes: pre-yield and post-yield (Figure 2.2.) In the pre-yield region (applied stress $< \tau_y$) the ER fluid can be said to behave essentially as a solid. As noted by Block and Kelly [21], it is not known at present whether the fluids undergo slow creep or are elastic in this region. Research has been performed by Conrad *et al* [22,23] amongst others on the evaluation of yield stress in the pre-yield regime. When the yield stress is obtained in this manner, it is termed the *static* yield stress. Another evaluation of yield, the *dynamic* yield, is found from the axial intercept of a τ vs. $\dot{\gamma}$ plot (a post-yield description.) While the Bingham model does not distinguish between the two, actual ER fluids generally produce a static yield stress which exceeds the value of the dynamic yield [21,22,23,24]. In addition, it is apparent that ER fluid

Figure 2.1: Shear Stress vs. Shear Rate for a Bingham Plastic

Figure 2.2: Shear Stress vs. Shear Strain for an ER Fluid. Adapted from Coulter [44].

behaviour is inconsistent in regions of low $\dot{\gamma}$ [21,22], and has even been called ‘complex, irreversible and often irreproducible...’ [21], thus, evaluation of either yield stress becomes difficult and ambiguous. Conrad [22] suggests a yield stress determination based on a fixed value of strain, but admits that such practices ‘may not give unique or unambiguous values...’, mainly because of the static yield dependence on shear rate [22,25], as well as other factors. For the application presented in this thesis, importance is placed only on the post-yield regime and any references to yield stress imply the dynamic definition of yield stress.

2.3.3 Viscosity

Viscosity in Newtonian fluids is based on steady, simple shear, and is described mathematically by the quotient of shear stress and shear rate:

$$\mu_n = \frac{\tau}{\dot{\gamma}} . \quad (2.2)$$

However, because of the structured behaviour of ER fluids, and the tendency of particles to coagulate on electrodes, the velocity gradient in a given shear flow is generally non-uniform. Therefore, the term ‘apparent viscosity’ (or ‘apparent plastic viscosity’) is preferred, and is represented by the slope of the shear stress vs. shear rate curve for an ER fluid.

2.3.4 Excess Shear & Shear Ratio

The ‘excess shear’ describes the amount of shear in a field excited ER fluid that

is not due to normal Newtonian effects, and is defined as the arithmetic difference between the field induced shear stress $\tau(E)$, and the zero field shear stress, $\tau(0)$:

$$\tau_e = \tau(E) - \tau(0) . \quad (2.3)$$

In the Bingham model, the excess shear stress is equivalent to the yield stress. Excess shear is sometimes referred to as the *strength* of the fluid.

Another parameter employed in describing ER fluid performance is the ‘shear ratio’, which is simply the value of the shear stress while under electric field excitation, $\tau(E)$, divided by the Newtonian shear stress, $\tau(0)$, at a given shear rate:

$$shear\ ratio = \frac{\tau(E)}{\tau(0)} .$$

This parameter is used to indicate the amount of controllability the fluid presents for a specific shear situation. The ‘shear ratio’ is sometimes misleading because it decreases steadily for increased shear rate. This drop is not completely attributed to diminishing excess shear stress, but simply a result of Newtonian shear stress, $\tau(0)$, becoming much larger than excess shear stress, τ_e , at increased shear rates.

2.4 Theories of Electrorheology

It has been noted by various researchers such as Bullough [10,11] and Hägele [26], that in order to utilize ER fluids to their promised potential, an order of magnitude improvement in ‘strength’ over the current level has to be achieved. Obtaining this order of magnitude necessitates a thorough understanding of the causes of the ER effect. However, this subject is still a contested issue among ER researchers. The problem

manifests itself in the combined fields of rheology and dielectrics; since modelling the behaviour of concentrated dispersions in either of these fields alone is a difficult task, it is not surprising that in combination the problem has yet to be solved. On the basis that the ER effect requires an electric field, and ER fluids are capacitive elements, it is agreed that some sort of induced polarization of the particle phase must be involved. However, there is little agreement as to what occurs subsequently to induce the rheological effects. Described below is a brief summary of the more common theories.

2.4.1 Fibrillation

The fibrillation theory was the original ER mechanism proposed by Winslow [2] and substantiated by others [1,17,21]. This theory attributes the 'chaining' of polarized particles to the induced rheological effects. In fact, microscopic observation of a stressed ER fluid shows that these chains actually bridge the gap between electrodes. The chains form a mechanical structure in the fluid, and the fibrillation theory states that this structure determines the magnitude of the ER effect. While it is accepted that particle chaining does indeed occur, researchers such as Klass and Martinek [7,8] and Block and Kelly [21] state that chaining cannot be a primary mechanism on the basis that the ER effect is much too rapid to be a result of such extensive structure formation. (Typical ER response to a step voltage is within one millisecond [10].) Also, according to Block and Kelly [21], it is unlikely that particle chaining can maintain the significant excess shear stress under high flow conditions.

2.4.2 Double Layer Polarization

Proposed initially by Klass and Martinek [8], this theory ascribes the changes in rheology of ER fluids to the interference of electric field distorted 'double layers'. Double layers include the collection of ions in the carrier oil immediately surrounding the dispersed particles, and undergo ionic migration or some form of dipole alignment upon excitation of the fluid by an electric field. This theory is supported by evidence that additives such as water or surfactants magnify the ER effect, possibly as a means for enhanced ionic migration.

2.4.3 Water 'Glue'

The role of water in ER fluids was first acknowledged by Winslow [2], who required that his fluids be moist to provide a substantial ER effect. Fluids have been developed since then that list water as a requirement for *any* ER effect [21]. However, water activation in ER fluids has proved to be a serious hindrance to application, mostly in terms of a limited temperature range. Recently, fluids have been developed [27,28] which claim to be 'substantially anhydrous', and can operate without need of added water. While this would indicate that water does not play a role in the ER effect, these 'substantially anhydrous' fluids may contain as much as 5% by volume of water, which is comparable to the amount in some water activated fluids. Thus it may be possible that the ER effect is not achievable without the presence of water.

The Water 'Glue' theory was first presented by Stangroom [17]. In an ER fluid, water molecules are assembled around the mobile ions of the particulate phase. When

an electric field is applied the ions migrate to one end of the particle, carrying the water molecules with them. One end of the particle becomes overtly wet, and water bridges can form between particles. Therefore, in order to shear the fluid, work must be done against the surface tension forces in the water bridges. The Water 'Glue' theory has been supported by other researchers [29].

2.4.4 Recent Explanations of the ER Effect

Another interesting theory is that of Atten and Honda [30] who have proposed that the ER effect is a result of electrically induced secondary motion in the fluid, a transfer of momentum similar to turbulence which opposes the flow.

Block [31] investigated the polarization of rotating dielectrics which have a high degree of asymmetry. It was found that dispersions showed maximum dielectric absorption at field excitation frequencies of the same order as the particle angular velocities induced by flow. Results also showed that if a spinning sphere of finite electrical conductivity or significant dielectric absorption was subjected to a constant electric field, the sphere would experience a mechanical torque opposing the motion. This phenomena was used to describe the resistance to flow experienced in field excited ER fluids.

2.5 Applications of ER Fluids

ER fluids are predicted to have widespread impact in a number of areas [15]. Part of the attraction of ER fluids is their fast response time and their suitability to use with

feedback control. The most notable applications of ER fluids are in the areas of power transmission, hydraulics, and vibration control.

In the area of power transmission, ER research has been performed on fluids primarily for use in clutches, brakes, and other forms of couplings such as drill chucks [1,5,6,10,12,15,21,32-37].

One of the simplest foreseeable hydraulic applications is the ER valve, which provides a nonmechanical means of controlling fluid flow. ER valves have also been proposed for use in creating more complex hydraulic elements, such as in the wheatstone bridge configuration to create a longitudinal motion controller, or as a means of creating wide bandwidth, high power vibrators [1,3,4,10,13,15,21,32,34,35,38,39].

Another popular area of application is in the vibration control of mechanical systems. This is perhaps the most broad area, as the utility of ER fluid technology for the purposes of providing variable damping has great relevance. One research topic is the creation of 'smart' composite materials, for structural or other purposes, which are essentially a sandwich of ER fluid and can exhibit variable vibration characteristics under different loading situations [1,10,15,21,24,32,40,41]. These materials have attracted interest in the field of robotics in particular. The most popular application, however, is probably in providing variable linear motion damping in the industrial sector and, specifically, in the area of automotive suspension systems [1,9,10,11,14,15,21,24-26,32,33,42-47]. This is the area of interest of the present study.

2.6 ER Fluid Research Goals

A number of researchers have offered requirements that ER fluids must meet before any considerable application in the areas mentioned above can be realized [1,10,11,21,26]. A list of some of the main research goals follows.

ER fluid requirements:

- 1. Provide as high a shear stress as is possible at low electric field and as low a shear stress as is possible at zero electric field. This is one of the most essential requirements for engineering applications like power transmission and hydraulic control valves. In other words, ER fluids should demonstrate high ER 'strength', high electrical resistance, high dielectric strength, and a low zero field viscosity. One of the most serious drawbacks of present ER fluids is the power penalty one must incur. Because ER fluids are non-ohmic, (current is often proportional to E^2), the power requirement becomes increasingly more onerous as higher voltages are applied to induce the desired ER effect.**
- 2. Properties should remain stable over as wide a temperature range as possible. A range from -40°C to 120° has been proposed [45], particularly for automotive application. It has been noted that water plays a crucial role in making this requirement difficult to meet.**
- 3. Fluids should not be prone to rapid sedimentation of the particulate phase, in storage or in an application such as a clutch, where the centrifugal effect can compound problems. One technique for inhibiting sedimentation is**

density matching of the particulate and the base oil, but this usually involves excessively viscous base oils and the density match typically only exists for a limited temperature range. Surfactants and other additives have been proposed to eliminate this problem.

4. The base oils should ideally be non-toxic, non-flammable, non-corrosive, of low volatility and environmentally stable.
5. The particles should not cause excessive wear or abrasion.

It is generally considered that the last three requirements are secondary and that the first two are of more importance in terms of practical applications. According to Jordan and Shaw [1]: "A fully anhydrous material with high ER activity and low conductivity is the goal."

Chapter III

Vibration Control

3.1 The Vibration Control Problem

The primary function of suspension systems in ground vehicles is to control body and wheel motions due to roadway inputs, payload changes, and forces from acceleration, braking, cornering, and aerodynamics. A conventional suspension system is composed of a spring and a damper connected in parallel between the sprung mass (vehicle body) and the unsprung mass (wheel, axle, braking components, etc.). The spring's purpose is to cushion the vehicle from shock and vibration over surface irregularities, and the damper's purpose is to attenuate resonant behaviour. These two elements are commonly passive, i.e., no energy is supplied to the system, it is only stored or dissipated.

A great deal of study has been devoted to fine tuning the conventional passive suspension system in terms of selecting the appropriate spring and damping parameters, for example, Sharp and Hassan [48,49]. However, because suspension parameters are often 'ideal' for only limited frequency bands, the conventional system still suffers from a tradeoff in providing both ride comfort and wheel hop control, as will be seen in the next section.

3.2 Vibration Control Systems

3.2.1 Passive or Conventional Suspension Systems

For illustrative purposes, the performance of the passive suspension system is examined when installed in a *single-degree-of-freedom* (SDOF) quarter vehicle model, shown schematically in Figure 3.1. The mass m_s simulates the mounted body, and the isolator is represented by the combination of a displacement sensitive spring of spring rate k_s , and a velocity sensitive damper of damping coefficient c . The equation of motion of the suspended mass can be written

$$\ddot{x}_2 + \frac{f_d}{m_s} + \frac{k_s}{m_s}(x_2 - x_1) = 0 \quad (3.1)$$

where f_d is the force generated on the attachment points from the damper. For a passive viscous damper,

$$f_{d,p} = (2\zeta\omega_n m_s)(\dot{x}_2 - \dot{x}_1) = c(\dot{x}_2 - \dot{x}_1) \quad (3.2)$$

where ζ is the damping ratio,

Figure 3.1: Passive SDOF Quarter Vehicle Schematic

$$\zeta = \frac{c}{2\omega_n m_s} \quad (3.3)$$

and ω_n is the undamped natural frequency of the system,

$$\omega_n = \sqrt{\frac{k_s}{m_s}} \quad (3.4)$$

These equations and the following discussion are based on the assumption of a linearized passive damper, for which the damping characteristic is shown in Figure 3.2(a). From the damping characteristic plot it can be seen that only the first and third quadrants are invoked, and therefore the power associated with $f_{d,p}$ is strictly dissipative, i.e.

$$f_{d,p}(\dot{x}_2 - \dot{x}_1) \geq 0. \quad (3.5)$$

Thus, if the relative displacement is increasing, $(x_2 - x_1) > 0$, $f_{d,p}$ must be tensile, and if $(x_2 - x_1) < 0$, $f_{d,p}$ must be compressive.

Actual dampers in automotive applications (Figure 3.2(b)) differ from the linearized model. For low velocities, the damping coefficient, c , is higher, providing a more firm setting to control motion of the body at its resonant frequency. At higher velocities, the damping coefficient is reduced to provide increased isolation of the sprung mass from higher frequency roadway inputs. Forces in rebound (extension) are higher than those in jounce (compression) in order to minimize the transmission of energy to the vehicle from the roadway, also, the force saturates at high levels of both jounce and rebound velocity to protect the car body and components from high velocity inputs such as potholes, speed bumps, etc.

Figure 3.2: Linearized and Actual Passive Damping Characteristic Plots.
Adapted from Redfield [70].

Upon examination of a transmissibility plot of acceleration for the system shown in Figure 3.1 and employing a linearized passive suspension, the tradeoff in performance attributed to passive suspensions is witnessed. The curves shown in Figure 3.3 are a measure of ride comfort and represent the ratio of mass acceleration, $\ddot{x}_2$, to input acceleration, $\ddot{x}_1$, for fixed damping ratios. For sinusoidal inputs well below ω_n , the motion of the mass closely follows that of the input and the transmissibility ≈ 1 . Near the natural frequency, where motion between the input and the mass increases, the damper forces become more predominant and ultimately limit the amount of resonance in the system. Ideally, the damper should be very stiff around the natural frequency, to contain the motion of the sprung mass and thus reduce the amplitude at resonance. However, at higher frequencies, larger values of ζ incur a corresponding penalty of reduced performance (decreased isolation) as the stiffer suspension transmits more of the harsh high frequency input. This is an inherent tradeoff with the passive conventional suspension system. Even though the damper is a dissipative element, it is still capable of accelerating the mass because of the nature of its configuration: it acts to reduce the *relative* velocity between its attachment points, and not the *absolute* velocity of the sprung mass.

3.2.2 Active Suspension Systems

In an active suspension system, a degree of control can be implemented by means of an externally powered force actuator, such as a hydraulic ram, and some prescribed control policy. The main advantage of the active element is that it can supply, as well

Figure 3.3: Passive SDOF Transmissibility Performance. Adapted from Krasnicki [61].

as absorb, energy in the system. An *ideal* force generator can supply a force of any magnitude and any sign, instantaneously. Actual force generators can approach the performance of the ideal force generator but only with much complexity and expense. Since the force produced by the active element is not dependent upon the relative velocity across its attachment points, the damping characteristic plot for an active isolator encompasses all four quadrants. Thus, in theory, attention can be given to both resonance and high frequency ranges.

One configuration for an active suspension system is to substitute the passive damper with the controllable force generator in parallel with a passive spring element (Figure 3.4.) The most popular control strategy for the force generator is 'skyhook' control, which was proposed by Krasnicki [50]. In 'skyhook' control the force generator is implemented to emulate a passive damper connected between the suspended mass and an inertial reference. This control policy has also been referred to as 'fictional' inertia damping because such a reference is not available in vehicular applications. In this situation, the force applied to the mass from the active element would be proportional to the absolute velocity of the mass, and not the relative velocity between the mass and the input:

$$f_a = (2\zeta\omega_n m_s)\dot{x}_2 = c\dot{x}_2 . \quad (3.6)$$

Examination of the transmissibility plot (Figure 3.5) for a system employing an active element such as that shown schematically in Figure 3.4 indicates the difference in performance when subjected to different damping ratios. In a passive suspension, increased damping provides superior performance at resonance but decreased performance

Figure 3.4: Active SDOF Quarter Vehicle Schematic

Figure 3.5: Active / Passive SDOF Transmissibility Comparison. Adapted from Krasnicki [61].

at higher frequencies. However, increasing the damping of our fictional damper in the active system increases performance throughout the entire frequency range.

Although active systems accomplish vibration control performance unattainable by passive means, the increased performance does not come without cost. Active systems can represent a significant power drain on the vehicle, and must modulate over a wide frequency range (0-14 Hz). The combination of the power and bandwidth requirements can make active systems in general quite costly and complex.

3.2.3 Semi-Active Suspension Systems

A semi-active suspension system is essentially an active system in which the active force generator is replaced by a *controllable passive* damper; this element can only dissipate energy, but with a damping coefficient that can be modulated at high bandwidth. In other words, an ideal semi-active element or semi-active damper is similar to an active element with the restriction that control over the sign of the force is lost - the damping force is controllable, but can only be dissipative. A semi-active suspension system is shown schematically in Figure 3.6.

The damping characteristic plot for an ideal semi-active damper encompasses any point in the first and third quadrants. Actual semi-active dampers follow curves such as those in Figure 3.7. One conceptualization of a semi-active damper is a conventional damper in which the oil flow path is controlled by a sort of servomechanism (Figure 3.7(a)), another is one which employs a fluid whose resistance to flow is ER controllable (Figure 3.7(b)). While it is true that both forms of semi-active damper require external

Figure 3.6: Semi-Active SDOF Quarter Vehicle Schematic

Figure 3.7: Semi-Active Damping Characteristic Plots

power to operate, the power is generally used for control purposes and not to directly supply damping force.

The skyhook strategy for control of the semi-active damper is employed again, i.e. the desired value of $f_{d,s-a}$ is $c\dot{x}_2$, but it is restricted in that the semi-active damper must obey the dissipative requirement, $f_{d,s-a}(\dot{x}_2 - \dot{x}_1) \geq 0$. Therefore, the damper will only be able to supply the required skyhook damping if the sign of the relative velocity is equal to the sign of the absolute velocity of the mass,

$$f_{d,s-a} = (2\zeta\omega_n m_s) \dot{x}_2 = c\dot{x}_2, \text{ if } \dot{x}_2(\dot{x}_2 - \dot{x}_1) > 0. \quad (3.7)$$

If $\dot{x}_2$ and $(\dot{x}_2 - \dot{x}_1)$ are of opposite sign, the semi-active isolator can only supply a force opposite in direction to the desired skyhook force. When this situation arises, the best the semi-active damper can do is to supply zero, or a minimum force, therefore

$$f_{d,s-a} = 0 \text{ (or min.)}, \text{ if } \dot{x}_2(\dot{x}_2 - \dot{x}_1) < 0. \quad (3.8)$$

The performance of the semi-active damper generally approaches that of the active damper. Figure 3.8 shows a comparison of the three vibration control schemes for a single-degree-of-freedom system, illustrating the performance gain of the semi-active system over passive components, without the price or power penalty that must be paid for active suspension implementation.

The 'switching' between different values of c , or between nonzero and zero values of c , that a semi-active damper must perform creates an inherently non-linear system, even for this single-degree-of-freedom model. Thus, strictly speaking, transmissibility as a function of frequency (or frequency transmissibility) is not defined for a semi-active

Figure 3.8: Active / Passive / Semi-Active SDOF Transmissibility Comparison
Adapted from Karnopp et al. [65].

damper. However, experience indicates [51] that the response of a semi-active vibration system is essentially sinusoidal when subject to harmonic base excitation. It has therefore become acceptable to simulate semi-active systems with sinusoidal excitation and interpret the response amplitude as frequency transmissibility [51,52]. While this use of 'transmissibility' is valid for both linear and non-linear systems, the normal implications associated with the transmissibility of a linear system do not hold for the semi-active damper. For example,

$$\left| \frac{x_2}{x_1} \right| = \left| \frac{\dot{x}_2}{\dot{x}_1} \right| = \left| \frac{\ddot{x}_2}{\ddot{x}_1} \right| \quad (3.9)$$

holds only for linear systems subject to harmonic excitation.

3.3 Two-Degree-of-Freedom-Systems

The preceding discussion employed a single-degree-of-freedom model of the ground vehicle suspension system. In fact, a more accurate model of a ground vehicle includes the effects of the unsprung mass (wheel, axle, braking components, etc.), and the stiffness and damping present in the pneumatic tire. In this section, the discussion will be enhanced through consideration of a *two*-degree-of-freedom (2DOF) model of the ground vehicle suspension system. A popular model of this sort is the quarter vehicle model, shown schematically in Figure 3.9. The quarter vehicle model has been used extensively in automotive suspension research and its effectiveness has been proven [53,54]. While the simple nature of the model prevents a comprehensive analysis of ride motion, insight into the dynamics of the vehicle control problem can be achieved without

Figure 3.9: 2DOF Quarter Vehicle Schematic

great complexity. In the figure, m_s is the sprung mass, m_u is the unsprung mass, k_s is the stiffness of the suspension, k_t is the stiffness of the tire, and c is the damping coefficient (whether fixed or controllable.) As is customary, the relatively small amount of damping in the pneumatic tire has been omitted.

In a typical passenger vehicle, the primary natural frequency of the system is the body or sprung mass natural frequency, which typically falls in the range of 0.8 to 2 Hz. Attenuation of the motion of the body is often called 'ride comfort control.' The goal of ride comfort control is to minimize the transmissibility (of acceleration) to the body from the roadway input over a range of input frequencies while maintaining relative displacement ($x_3 - x_2$) within acceptable limits.

The secondary natural frequency of a typical passenger vehicle is the unsprung mass natural frequency, which typically falls in the range of 8 to 14 Hz. Attenuation of the motion of the sprung mass is often called 'wheel hop control.' The goal of wheel hop control is to minimize the variation of the displacement of the wheel from the roadway, ($x_2 - x_1$), in order to maintain sufficient wheel-roadway contact force. Wheel hop control is important because it relates directly to the handling and stability of the vehicle, especially in applications involving large roadway inputs, such as in off-road or military vehicles [51].

The tradeoff in performance that exists when passive suspensions are employed can be redefined for the two-degree-of-freedom model. It is helpful to divide the frequency response analysis into three separate ranges: a primary natural frequency range, an intermediate frequency range, and a secondary natural frequency range. As a higher

value of damping is selected in a passive suspension, ride comfort and wheel hop performance are increased in the primary natural frequency range, but both are reduced in the intermediate frequency range. In the secondary natural frequency range, a low value of ζ provides good ride comfort but poor wheel hop control, with the tradeoff to poor ride comfort and better wheel hop control as ζ is increased. In simpler terms, the tradeoff can be argued to be one of ride comfort vs. wheel hop control for varying ζ , independent of frequency considerations. Many researchers have summarized the problem in the form of a plot showing body acceleration on the ordinate and tire load variation (representative of the amount of wheel hop) on the abscissa and the effects adjusting both the spring and damping rates (Figure 3.10.) Figure 3.10 illustrates not only the conflict in the two performance criteria, but the limit of performance that a passive system must incur, and that active and semi-active systems can overcome.

3.4 Semi-Active Control Strategies

3.4.1 Ride Comfort Control

Many researchers have performed simulations utilizing forms of optimal control which attempt to simultaneously optimize the conflicting performance goals of ride comfort and wheel hop control, while maintaining a suitable suspension travel. The advantage of optimal control theory is that it results in an optimal system for the selected weighted cost function. However, in the case of suspensions, the weighted cost function is hard to define [54,56]. Furthermore, according to Karnopp [56], 'if active damping is of value,... a force component proportional to body velocity is required...' and goes on

Figure 3.10: 2DOF Passive Tradeoff. Adapted from Decker [73].

to state that 'The generation of a body absolute velocity signal from accelerometer data or by other means is not a trivial task.' This fact has also been acknowledged by Miller [57], who agrees that any measurement of absolute velocity is at best difficult. Integrating acceleration leads to problems with instability, infinite d.c. gain, and low frequency temperature drift. Attempts at approximating an integrator with a high order filter result in hardware implementation difficulties, especially in the case of analog filters. It is these same reasons which make Karnopp's skyhook control troublesome, as it relies solely on absolute velocity for control purposes.

Jolly and Miller [58] and Alanoly and Sankar [59] have investigated a method of control entitled 'relative control', which employs only the relative displacement, $(x_2 - x_1)$, and relative velocity, $(\dot{x}_2 - \dot{x}_1)$, signals measured across the suspension. One of the main reasons for the investigation of Jolly and Miller is that one is able to implement relative control without electronics, and they include in their study a proposal for an all-mechanical, semi-active damper device. While relative control does not necessarily result in ideal performance, its ease of practical implementation makes it quite attractive. It is for these reasons that relative control is the basis for the ride comfort control schemes used by researchers in the Transport Technology Research Laboratory (TTRL) at Carleton University.

The basic premise behind relative control is to minimize the net instantaneous suspension force on the sprung mass. If the spring force and the damping force acting on the sprung mass are in opposite directions (eg. one upwards and one downwards) then the damping force is set to completely offset the spring force. If the forces are in the

same direction, then the damping force is set to its minimum possible value. Restating equation 3.1,

$$\ddot{x}_2 + \frac{f_d}{m_s} + \frac{k_s}{m_s}(x_2 - x_1) = 0 \quad (3.10)$$

we can derive the desired damper force directly:

$$f_{d,s-a} = -k_s(x_2 - x_1) \text{ if } (x_2 - x_1)(\dot{x}_2 - \dot{x}_1) < 0 \quad (3.11)$$

and

$$f_{d,s-a} = 0 \text{ (or min.)}, \text{ if } (x_2 - x_1)(\dot{x}_2 - \dot{x}_1) > 0. \quad (3.12)$$

When a comparison is made between skyhook control and relative control, it can be argued that the difference lies essentially in the switching condition. Skyhook control uses the product of absolute velocity and relative velocity to determine damping, and relative control uses the product of relative displacement and relative velocity. Jolly and Miller [58] examined the difference in magnitude and phase between absolute velocity and relative displacement to determine whether relative control was not unlike skyhook control. Their findings show that for a broad frequency range, the phase difference is essentially zero and the magnitude ratio is ≈ 0.7 . However, there is very poor correlation at frequencies below approximately 0.8 Hz. Alanoly and Sankar [59] made a direct comparison of the two strategies through simulation and found that the performance was 'very similar.'

In another effort to simplify the hardware implementation, Roley [60] and Krasinski [61], investigated the performance of the 'On-Off' control scheme. In 'On-Off'

control, instead of continuous modulation of damping forces to match the desired force, the damper has only two settings: one with a high damping coefficient, and the other with a near zero damping coefficient. In the case of relative control, the scheme is modified to switch to high damping if the spring and damper forces are in opposite directions, and supply as little force as possible when suspension forces are in the same direction. A number of researchers [45,58,61] have determined that an On-Off scheme degrades performance of the semi-active system slightly, but still offers significant improvements over passive suspension systems. Relative On-Off control is the control scheme used to evaluate ride comfort performance of the TTRL dampers.

3.4.2 Wheel Hop Control

As with ride comfort control, a popular approach to control of the unsprung mass is to use a variation of Karnopp's skyhook strategy, and provide damping according to a fictional inertia policy. This too results in the same problems of implementation as with ride comfort control. Wu *et al* [62] has proposed two strategies for wheel hop control which avoid this problem. The first simply changes the state of the ER damper between minimum and maximum damping, depending on the sign of the relative velocity across the suspension. The objective is to maintain an acceptable load between the unsprung mass and the roadway input. If the relative velocity indicates the damper is exerting an upward force on the unsprung mass, the voltage is set at minimum, and if the force is downwards on the unsprung mass, the voltage is set at maximum. The second strategy approaches the problem in the same manner as relative control, but in this case the

objective is zero variation of tire deflection, and so the equation of motion is solved for the damping coefficient in order to make $(x_2 - x_1) = 0$. This scheme however would require significantly more hardware implementation, including an additional accelerometer for the unsprung mass and so the first scheme was the one selected for use in testing the ER dampers.

3.4.3 Composite Control

It has also been noted by Wu *et al* [62], that the strategies for improving ride comfort do not necessarily improve road holding and vice-versa. For this reason, three 'composite' control strategies were proposed, which switch between ride comfort and wheel hop control. Switching between strategies is based on the comparison of dynamic tire deflection to the static tire deflection:

$$\alpha = \frac{|x_2 - x_1|}{\left[\frac{(m_s + m_w)g}{k_{tr}} \right]} \quad (3.13)$$

If the calculated value of α is below a threshold value, normal ride comfort control is activated, otherwise a wheel hop strategy is activated. Most of the work presented by Wu *et al* deals with a composite control strategy that uses maximum damping as its wheel hop control. This strategy was also used to compare the performance of the two TTRL dampers.

Chapter IV

Rheometric ER Fluid Testing

4.1 Introduction

ER Fluid testing at Carleton is performed primarily on a Couette cylindrical rotary rheometer, known to be a proven rheometric technique [63]. The purpose of the testing is to develop an understanding of the rheological behaviour of the fluids that are to be utilized in the semi-active dampers. The term 'Couette' is applied to rheological devices employing shear mode, where the fluid is sheared between one moving and one stationary electrode. Another ER fluid mode is Poiseuille flow (sometimes termed 'flow mode') where the fluid is forced through two electrodes that are stationary relative to each other. A discussion of the two fluid modes is given in Chapter V.

4.2 Structure of Apparatus

The ER rheometer consists of two concentric cylinders separated by a gap of active ER fluid and entirely contained by a plexiglass shell (Figure 4.1). The inner cylinder, or rotor, is driven by a variable speed DC motor, while the outer cylinder, or stator, is restrained by a reaction torque sensor. The stator is connected to the high tension side of a variable voltage supply to facilitate electrical loading of the fluid, and the rotor is connected to ground.

4.3 Experimental Arrangement

The purpose of the testing was to determine the shear stress provided by ER fluids under variable shear rates, temperatures and electric field strengths. Figure 4.2 illustrates schematically the experimental arrangement. A description of the components and their functions are specified as below:

1. Rheometer: The basic structure of the rheometer is described above. Specifically, the rotor is connected to a shaft which is held in position laterally and allowed rotational movement by bearings at each end of the rheometer. The electrodes, both rotor and stator, are manufactured out of aluminum. The working surfaces have been sandblasted to provide roughness that is intended to prevent slippage of the ER fluid on the electrode wall when it assumes a gel-like state at high electric field excitation. A study of different surfaces was performed by Bortolotto [63], who determined different surfaces had no conclusive effect on shear stress, meaning that either slippage is unaffected by surface type or that no slippage is present at all. The electrodes are sized to provide a fluid gap

Figure 4.1: Couette Rheometer Cross Section

Figure 4.2: Rheometer Experimental Arrangement

of 3 mm, with the inner radius = 0.0695 and the outer radius = 0.0725 m. This provides a ratio of outer to inner gap radius of less than 1.05, a design requirement specified by Van Wazer *et al* [64], in order to assume negligible radial effects and a linear velocity profile. Thus the shear rate across the gap is assumed constant and equal to the 'gross' shear rate: rotor tangential velocity divided by gap width.

2. DC Motor: The rheometer rotor was driven by a Boston Gear 1 hp DC motor through a flexible coupling. The motor is capable of providing a variable speed from 0 to 1750 rpm, translating into shear rates for the rheometer of 0 to 4000 s^{-1} .

3. High Voltage Generator: Voltage is supplied by a Universal Voltronics high voltage DC power supply with manual voltage control, integral panel voltmeter (range: 16kV, increment: 500V), and integral panel milliammeter (range: 10mA, increment: 0.1 mA.)

4. Temperature Control: Temperature control of the ER Fluid was provided by the cold room facilities at Carleton University.

5. Speed Measurement: A Boston Gear Digital Tachometer, compatible with the DC motor was used for speed measurement in rpm.

6. Torque Measurement: A calibrated Transducer Techniques TRS-2K reaction torque sensor is connected between the rheometer base and the support stand. The signal from the sensor is sent to a Vishay/Ellis 20 strain indicator/amplifier, and then further amplified and filtered by a Krohn-Hite filter before being recorded on a PC data acquisition program via a Computer Boards A/D card.

7. Temperature Measurement: A thermocouple is attached to the upper end of the rheometer and fitted such that it is immersed in the ER fluid. Display of temperature is

provided by a digital readout.

4.4 Experimental Procedure

Three fluid compositions were found to be suitable for rheometer testing after performing preliminary evaluations. These evaluations consisted of visual inspection of ER behaviour when subject to an electric field via two electrodes immersed in a glass beaker of fluid. The three fluids included two different mass ratios of Sigma Z3125 Zeolite in 10 centistoke silicon oil (2:5 dispersant to base oil and 4:5 dispersant to base oil), and a 2:5 mass ratio of BDH 'AnalaR' Aluminum Oxide (calcined) in 10 centistoke silicon oil. No other additives were included in the fluids, and the amount of water present is that which occurs naturally - it was neither added nor removed. The fluids were tested at three temperatures: -6, 33, and 57 °C, and nine discrete shear rates from 50 to 1400 s⁻¹. The target for electric field strength variation was to include values of 0, 1, 2, and 3 kV/mm, meaning voltages of 0, 3, 6, and 9 kV were required for the 3mm gap. However, voltage breakdown, especially at higher temperatures and shear rates, often prevented data collection for the 3 kV/mm field strength.

Before each fluid test, the rheometer was disassembled, cleaned thoroughly, reassembled and run through a preliminary dry friction test before being filled with ER fluid. Dry friction tests provided consistent, level curves of torque representing an equivalent shear stress of approximately 10-15 Pa. Fluid shear effects on the upper and lower surfaces of the rotor were omitted from the investigation.

Once the system had been calibrated and friction-tested, and the rheometer filled

with an ER fluid, the apparatus was ready for fluid testing. For low temperature testing, the rheometer was placed in a cold room in order to achieve the desired fluid temperature. Then, for each electric field strength, the following steps were taken. The fluid was agitated to ensure thorough dispersion of the particle additive in the base oil, and if necessary, additional cooling of the fluid was performed due to heating upon agitation. The rheometer was driven to the first shear rate, the high voltage supply set for the specified field strength, and a speed correction made due to slowing of the rotor under ER shear stress load. Three samples, averaging hundreds of readings over 2-8 seconds, were made of the torque with the data acquisition system. The current was noted and then the voltage switched off. The motor speed was reversed, the voltage switched back on, a speed correction was made, and three more samples were taken in the opposite direction for verification of a proper zero in the data. The voltage was switched off and the torque was evaluated at the next shear rate. Once the data for all shear rates was generated, the next electric field strength was evaluated. This process was repeated for all electric field strengths until a complete set of data for the fluid was achieved for that temperature, and then data for the next temperature was collected.

Data in the form of torque vs. rpm was converted to equivalent shear stress and equivalent shear rate. Plots of shear stress vs. shear rate vs. E for each fluid at each temperature were generated. After the data for each fluid is presented, a comparison of the results is made between the three fluids investigated herein and previous fluids studied at Carleton, namely a 2:5 mass ratio mixture of BDH 'AnaiaR' starch in 10 centistoke silicon oil, and a fluid of a proprietary composition, termed ERF 'B.'

4.5 Results and Discussion

Figure 4.3 describes the performance of the 2:5 mass ratio zeolite mixture at 33°C. It is apparent from this figure that the Bingham plastic representation of ER fluid behaviour is at best a first approximation. While the curves indicate the presence of a yield stress, the ER fluid displays a non-linear dependency on shear rate, in some cases dilatant (shear thickening, increasing μ_p) and others pseudoplastic (shear thinning, decreasing μ_p .) This non-linear behaviour is common for all fluids tested in this study.

As the electric field strength, E , is increased for the 2:5 zeolite fluid, the curves become increasingly more pseudoplastic from dilatant behaviour at zero E . However, after a shear rate of 2000 s^{-1} is surpassed, the curves all tend to remain near linear or dilatant behaviour. When this loss of pseudoplasticity at high E is coupled with the natural increase in zero field shear stress at high shear rate (due to normal viscous effects) the controllability of the fluid, or shear ratio, suffers, producing a range of superior ER effect from 250 s^{-1} to 2000 s^{-1} . This feature of the fluid is of considerable importance in ER damper design, as maximizing controllability over as wide a range as possible is a primary design goal.

The overall ER effect of the 2:5 zeolite mixture is comparable to fluids previously tested in the lab: a yield stress of approximately 200 Pa at $E = 3 \text{ kV/mm}$, an excess shear stress of approximately 200 Pa, and an apparent viscosity, μ_p , of approximately 0.1 Pa-s. A table of average apparent viscosities for the fluids tested is given below. The values were obtained from the average slope of a linear least squares approximation to the data (for each electric field strength).

Figure 4.3: Shear Stress (Pa) vs. Shear Rate (1/s) for 2:5 Zeolite/Silicon Oil at 33 °C.

Table 4.1: Average Apparent Viscosity, μ_p

Temperature	2:5 Zeolite	4:5 Zeolite	2:5 Aluminum Oxide
- 6 °C	0.17	0.40	0.12
33 °C	0.11	0.19	0.08
57 °C	0.10	0.16	0.06

Figure 4.4 illustrates the performance of the zeolite fluid when the amount of dispersant is doubled. At high values of E , the fluid allowed very high amounts of current flow, illustrating a decrease in dielectric strength, most likely due to the increased number of charge carriers. This loss of dielectric strength prevented testing at 3 kV/mm, and curves for 0, 1 and 2 kV/mm are shown. For the zeolite mixtures in particular, it was found that current increased (or dielectric strength decreased) with shear rate, a characteristic not common with the aluminum oxide mixture, which displayed a increasing dielectric strength with shear rate (Figure 4.5.) This presented problems for data collection at high shear rates for the zeolite mixtures, and at low shear rates for the aluminum oxide mixtures. While the author cannot provide the reasons for this difference in behaviour, this is an important result for fluid designers, and researchers interested in the mechanism(s) which cause the ER effect.

In general, the 4:5 zeolite mixture provided superior strength characteristics over the 2:5 mixture, demonstrating a maximum yield stress and excess shear of approximately 750 Pa. However, this does not come without a penalty of two times the zero field viscosity. The mixture also showed the trend from dilatancy to moderate pseudoplasticity, exhibiting a sharply decreasing apparent viscosity from 0.23 Pa-s at 0 kV/mm, to approximately 0.15 Pa-s at 2 kV/mm.

Figure 4.4: Shear Stress (Pa) vs. Shear Rate (1/s) for 4:5 Zeolite/Silicon Oil at 33 °C.

Figure 4.5: Current Density ($\mu\text{A}/\text{cm}^2$) vs. Shear Rate (1/s) at 33 °C.

Figure 4.6 depicts the performance of the 2:5 aluminum oxide mixture. As mentioned above, the mixture displayed increasing dielectric strength with shear rate (in contrast with the zeolite mixture) preventing adequate data collection at lower shear rates and high E. Furthermore, the current flow in the fluid did not show a strong temperature dependency (albeit over a small temperature range) and did allow data collection for 3 kV/mm at 57°C, whereas most ER fluids tend to lose dielectric strength with increased temperature. In general however, the fluid did not display adequate performance in terms of ER effect, and did not have the characteristic reduction of apparent viscosity demonstrated by the zeolite mixtures, resulting in relatively low shear ratios.

The effect of temperature on the ER fluids was studied for a limited temperature range from -6 °C to 57 °C. Shear ratios (controllabilities) of the fluids were compiled for each temperature and compared with data supplied by Bortolotto [63] for fluids which have previously been studied. Of the three temperatures studied, data at 33 °C is most useful as this is considered closest to the operating temperature of the damper under normal conditions. Care should be taken when interpreting shear ratios at very low shear rates ($\leq 400 \text{ s}^{-1}$) as ratios are magnified due to low zero field shear stress.

For the zeolite and aluminum oxide mixtures, low temperature (-6°C) increased the viscosity of the fluid and reduced the amount of excess shear, resulting in generally poor performance of the fluids in terms of shear ratio (Figure 4.7) and general ER effect, despite the fact that the fluids demonstrated higher dielectric strength and thus allowed higher field strength testing. The zeolite mixtures still provide relatively adequate ER effect, and this is reflected in the shear ratio performance. The starch and proprietary

Figure 4.6: Shear Stress (Pa) vs. Shear Rate (1/s) for 2:5 Aluminum Oxide/Silicon Oil at 33°C.

Figure 4.7: Shear Ratio vs. Shear Rate (1/s) at $-6^{\circ}C$.

mixtures offer very poor ER effect at this temperature.

High temperature (57°C) testing of the zeolite mixtures indicated increased levels of current flow and sharply reduced dielectric strength, preventing data collection at 2 kV/mm for the 4:5 mixture and at 3 kV/mm for the 2:5 mixture. Even though the viscosity of the fluid was reduced, performance of the fluid at high temperature (Figure 4.8) suffered compared to that at 33 °C (Figure 4.9), mainly due to the reduced allowable electric field strengths. A rather surprising reaction to high temperature is produced by the aluminum oxide mixture: reduced current flow and higher dielectric strength, allowing testing at 3 kV/mm, and therefore better performance. This is surprising because it indicates that the dielectric strength of the fluid has an extremely nonlinear dependence on temperature, increasing for low and high temperatures but relatively low for median temperatures. The reduced apparent viscosity combined with the higher field excitation produced better performance for the aluminum oxide mixture at the elevated temperature. However, this performance still does not exceed that of the zeolite fluids at realistic shear rates. The starch mixture shows increased shear ratio in the median shear rate range, comparable to the aluminum oxide and zeolite 2:5 mixtures, while the proprietary fluid, ERF 'B', still does not provide adequate performance.

Independent of dielectric strength, ER fluids will likely show the best performance when the apparent viscosity is at a minimum. This is the case in the present study for the aluminum oxide and starch mixtures and ERF 'B', where high temperatures did not prevent data collection and the 57 °C data produces the highest shear ratios. For the zeolite mixtures, the dielectric strength was much more a function of temperature and thus

Figure 4.8: Shear Ratio vs. Shear Rate (1/s) at 57 °C.

Figure 4.9: Shear Ratio vs. Shear Rate (1/s) at 33°C.

performance at high temperature suffered, resulting in optimum performance at 33 °C (Figure 4.9.) Here the zeolite mixtures display the highest shear ratios recorded in the testing, approximately 18 at low shear rate.

The preferred mechanical properties of the fluids for the intended application are similar to those described in Chapter II. Since no significant difference was apparent between fluids in terms of sedimentation or abrasive qualities, the shear ratio and apparent viscosity become the basis of comparison, and thus the zeolite/silicon oil mixtures show superiority. Results show similar shear ratio performance for the two different mass ratios of zeolite/silicon oil at normal operating temperatures, and so the 2:5 mixture was selected for damper testing, as it demonstrates nearly half the apparent viscosity.

Chapter V

Semi-Active Damper Design

5.1 Introduction

In this chapter a discussion is presented on the design of semi-active suspension dampers. The use of an electromechanical variable orifice to alter the flow resistance of a conventional hydraulic fluid is one means of obtaining variable damping force [26,50,51,65-70], and is included in the discussion. An alternate approach is the use of the ER effect: implementation of an electrically charged fixed orifice to change the flow resistance of a field responsive (ER) fluid. A description of the original Carleton semi-active ER damper (TTRL 'A') and its performance is presented, followed by a description of the design methodology behind the author's semi-active ER damper, TTRL 'B'.

5.2 Electromechanical Variable Orifice (EVO) Dampers

The method of obtaining variable rate dampers used most often is through implementation of a variable orifice valve in the flow path of the damping fluid. The actual design configuration of an EVO damper can take many forms, one of which is pictured in Figure 5.1. The damper shown is of the single tube type, incorporating the magnetically controlled variable orifice inside the piston of the damper. Control of the orifice, possibly combined with check valves, can provide the appropriate jounce and rebound characteristics, resulting in an adjustable damping coefficient, c . A typical damping characteristic plot for an EVO type damper is shown in Figure 5.2.

It has been noted by Hägele *et al.* [26], that EVO dampers of this type are capable of continuous adjustment from a very soft to a very hard curve. The EVO approach has other advantages such as sturdiness, compact design, and 'limp home' characteristics, meaning that, upon failure of the control system or power supply, the damper provides a degree of performance which still allows operation of the vehicle. One criticism however is that the valves do not act fast enough to respond effectively to feedback signals in vehicular application. A basic requirement is that the valve be adjustable between the limits of operation within less than approximately 5 ms [26], to provide the necessary bandwidth control. Also, the valves have to withstand significant hydraulic power (3 kW, [26]), and this further complicates their design. In order to meet these requirements, a very small moving mass is necessary, and thus careful analysis of the hydraulic force, prediction of the deformation in the orifice, and stationary and transient electromagnetic field prediction for the magnetic drive [26].

Figure 5.1: EVO Damper Piston Cross Section. Adapted from Hägele [26].

Figure 5.2: EVO Damping Characteristic Plot

5.3 Electrorheological Dampers

Another solution to the problem of implementing semi-active control is through exploitation of the variable shear stress characteristics of ER fluids. Since the ER effect can provide the variability required for semi-active damping within the required time constraints, the mechanical design of the device is much simplified and provides a more elegant solution to the semi-active damper problem, potentially decreasing manufacturing costs and increasing reliability. A typical damping characteristic plot for an ER damper is shown in Figure 5.3.

5.3.1 Flow Modes

Operation of an ER damper is based on one of two post yield ER electrode interaction configurations as shown in Figure 5.4: Poiseuille flow, shown in Figure 5.4(a), in which a pressure gradient induced flow occurs between two parallel electrodes that are stationary relative to each other, and Couette flow, shown in Figure 5.4(b), in which the fluid is sheared between two parallel electrodes moving relative to each other. In the Couette flow arrangement, the magnitude of the electric field across the gap controls the amount of force, F , required to move one electrode at a constant speed, v . Restating equation (2.1),

$$\tau = \tau_y + \mu_p \dot{\gamma} \quad (5.1)$$

where $\dot{\gamma}$ is the shear rate, therefore:

Figure 5.3: ER Semi-Active Damping Characteristic Plot

Figure 5.4: Post Yield Electrode Interaction Configurations. Adapted from Coulter [44].

$$\gamma = \frac{\partial v}{\partial y} = \frac{v}{2H} \quad (5.2)$$

we can then write

$$F = WL\tau_y + WL\mu_p \frac{v}{2H} \quad (5.3)$$

where W is the width of the electrodes, v is the relative velocity between electrodes and $2H$ is the distance between electrodes.

In the Poiseuille flow arrangement, the ER fluid is forced to flow through the electrodes due to the presence of a pressure difference over the length of the gap. Depending on the arrangement of the damper, the pressure difference can be used to determine the damper piston force. This pressure difference can be stated explicitly if one omits higher order plug flow effects:

$$\Delta P = \frac{3}{2} \left(\frac{\mu_p L Q}{WH^3} + \frac{\tau_y L}{H} \right) \quad (5.4)$$

where Q is the volumetric flow rate through the electrodes.

Plug flow (in Poiseuille flow) is a phenomenon that occurs in all fluids that exhibit a yield stress, such as Bingham plastics. The region(s) of the fluid that do not demonstrate shear stresses in excess of the yield stress, such as the centres of pipes and channels, are occupied by a solid plug, creating a flat peak on the velocity profile. Plug flow effects are discussed further in section 5.5, where they are accounted for in the author's damper force prediction program.

It has been verified [24,26,45] that a Poiseuille flow damper is approximately 40%

less efficient than a Couette flow damper. However, Couette flow dampers are more difficult to design and thus Poiseuille flow damper design and testing is more popular among researchers [26,33,45,46,62]. A discussion of different Poiseuille flow, Couette flow, and mixed (Poiseuille and Couette) damper designs follows.

5.3.2 Damper Designs

The basic premise behind ER damper design is to utilize the resistance to flow generated by the ER effect to provide a controllable axial damping force on the piston rod. Figure 5.5 details some ER damper design possibilities.

Figure 5.5(a) is a basic External Valve damper, and was the subject of analysis by Bullough [10,11], Lou *et al* [42], and Jolly and Miller [58]. The pressure differential across the valve is controlled by the excitation voltage, and is magnified by the piston area ratio to give the damping force. Figure 5.5(b), the Fixed Electrode design, studied by Coulter [44] and Petek [46], is an extension of the external valve damper, but instead incorporating the valve inside the primary tube in the form of one or more annular gaps. The Electrode Piston damper, shown in Figure 5.5(c), in which fluid is forced through annular gaps in the piston, was studied by Hägele *et al* [26], Lou *et al* [42], and by researchers at Carleton [33,45,62]. The damper piston comprises a number of channels that permit ER fluid flow upon motion of the piston. The pressure drop across the piston is determined by the resistance to flow provided by electric field excitation across these gaps. Dampers (a), (b), and (c) are based on Poiseuille flow in the active gaps.

Figure 5.5(d) and (e) are dampers based on Couette flow, but are actually

Figure 5.5: ER Damper Design Possibilities

considered mixed-flow dampers, because Poiseuille flow is present as well as Couette flow in the gaps due to the displacement of fluid by the piston. Analysis of these dampers has been performed by Lou *et al* [42] and Coulter [44], and satisfactory performance has been reported by Stanway [47], Wu *et al* [33,62], and Wong [45].

Pictured in Figure 5.5(f) is a damper studied by Coulter [44], which converts the linear motion of the piston into rotational motion of an electrode. It has the significant advantage of a reduced ER material requirement, pure Couette flow operation, and the capability of piston travel in excess of damper casing length. However, efficiency of the ball screw mechanism is a major concern, as it limits performance due to the increased minimum, or zero field, damping force.

5.4 Previous Work Performed at Carleton

5.4.1 Transport Technology Research Laboratory (TTRL) Damper 'A'

Researchers at Carleton have designed and manufactured a working ER semi-active damper, hereafter referred to as Transport Technology Research Laboratory (TTRL) damper 'A.' The damper is of the piston electrode type, utilizing Poiseuille flow post-yield ER characteristics. A sectioned view of the damper is shown in Figure 5.6.

TTRL 'A' consists of a simple stainless steel exterior bucket, or damper casing, and an electrode piston comprising five concentric aluminum cylinders acting as the electrodes. A hollow brass piston rod connects the piston to the sprung mass, acts as the ground for the two grounded electrodes, and houses the insulated voltage feed for the other three electrodes. In operation, the entire piston-electrode assembly moves relative

Figure 5.6: TTRL Damper 'A'

to the damper casing, forcing fluid through the annular gaps formed by the electrodes. The damping force produced by TTRL 'A' is thus controlled by adjusting the voltage across the electrodes.

5.4.2 Performance of TTRL Damper 'A'

Part of the contribution of the author is in the design and testing of a successor to TTRL damper 'A', referred to as TTRL damper 'B.' In order to facilitate a discussion on the design goals of TTRL 'B', it is necessary to provide a brief summary of the performance of TTRL 'A', from testing with the quarter vehicle model test facility at Carleton University. A complete description of the apparatus and test procedures is presented in Chapter VI; a brief description follows.

The quarter vehicle model test facility is designed to emulate a two-degree-of-freedom quarter vehicle model. It consists of a sprung mass of 44.2 kg, an unsprung mass of 6.2 kg, a suspension spring rate of 2864 N/m, a tire spring rate of 24640 N/m, is driven through an eccentric cam by a variable speed DC motor, and constrained to move in the vertical direction only. The experimental damper is fitted into the apparatus in parallel with the suspension spring, between the sprung and unsprung masses. The driving mechanism is designed to provide the system with sinusoidal excitation at frequencies from 0.8 to 14 Hz. Control of the voltage to the electrodes and data acquisition is performed by a PC via D/A and A/D cards respectively.

A general conclusion on the performance of TTRL damper 'A' is that it provided the expected advantages of semi-active control over passive dampers at low frequencies

(i.e. the sprung mass frequency), but performance at higher frequencies is not significantly better than that of a passive damper. A comparison was made by Wu *et al* [62], of the performance of damper TTRL 'A' and that of passive dampers of varying damping ratio. The results of that comparison are shown graphically in Figure 5.7. As can be seen from the figure, the equivalent variability in damping ratio between zero and high electric field excitation is sharply reduced at high frequencies. A conclusion of Wu *et al* was that the loss of variability was due to a decrease in the shear ratio of the fluid because of the increased shear rate at high frequency. The high shear rates (approximately 10000 s^{-1}) were a result of the electrode piston forcing a large volume of fluid through a small total gap area. This conclusion is supported in the present study by the results of rheometric ER fluid testing discussed in Chapter III. Also stated in the study was that the duration of the control voltage at high frequencies may be too short to have an adequate effect on the state of the ER fluid, or stated more generally, the transient performance of the fluid may include effects not accounted for in the steady state rheometer testing. For example, Lou *et al.* [38] have shown that fluid inertia can limit the bandwidth of an ER valve to a level far below the rheological variation of an ER fluid. However, if the difference in damping force supplied at zero and high electric field strength is not significant, then any transient effects can be considered a secondary complication.

5.5 Design of TTRL Damper 'B'

5.5.1 Design Goals

The primary design goal of TTRL damper 'B' is to investigate the performance

Figure 5.7: Damping Ratio vs. Frequency for TTRL Damper 'A'. Courtesy of Wu et al. [62].

of a reduced shear rate damper, in accordance with reasons provided in the discussion above. Specifically, the evidence provided in Chapter III supports the use of lower shear rates primarily because of the corresponding increase in shear ratio of the fluid. This increase in shear ratio at lower shear rates was due to two effects: the reduction in zero field shear stress, and the added effect of a change from dilatant to pseudoplastic behaviour with an increase in electric field strength. Additionally, there is a beneficial reduction in apparent viscosity at lower shear rates, regardless of electric field strength, which helps to reduce the normal (non-ER) component of shear stress.

5.5.2 Flow Analysis

It must be noted that the results of Chapter III are derived from Couette rheometric testing of the fluid, whereas it was the intention of the author to design a damper based on the Poiseuille mode of ER fluid flow, due to the documented difficulties one can expect to encounter in designing a Couette flow damper. This requires that some preliminary analysis of Poiseuille flow must be performed in order to investigate the nature of shear rates and their effect on Poiseuille flow as compared to Couette flow. In particular, the tendency of ER fluids to exhibit plug flow must be considered, as well as its effect on shear rate. It should be kept in mind, however, that the following discussion has its basis in the assumption that the shear stress relation to shear rate in Poiseuille flow is similar to that in Couette flow, which cannot be validated based only on data from Couette rheometry alone. It should also be kept in mind that ultimately the concern is with maximizing the difference between the zero and high field damping forces, whether

or not reduction of shear rates is a means to that goal.

5.5.2.1 Couette Flow

In Couette flow dampers (Figure 5.8(a)), the flow is shear driven: one of the electrodes moves with respect to the other, causing, an (assumed) constant shear stress and constant shear rate across the gap of ER fluid. The presence of an electric field does not affect the value of the shear rate in the fluid, as it is a function of the difference in velocity between electrodes and the thickness of the fluid gap. The resulting damping forces are predicted based on results from Couette testing of the ER fluid at a single shear rate, as in Chapter III.

5.5.2.2 Poiseuille Flow

A different situation arises in Poiseuille damper design (Figure 5.8(b)). Here, the flow of ER fluid in the gap is driven by a pressure difference over the length, L . A standard momentum/force balance gives us the following expression for shear stress:

$$\tau(h) = \left(\frac{\Delta P}{L}\right)h \quad (5.5)$$

producing the shear stress distribution as in Figure 5.8(b), where ΔP is the pressure difference over the length, and h is the distance laterally from the centre of the gap. Here the shear stress is not constant across the gap, but varies linearly from the centre to each electrode. As a result of this, difficulties arise in interpreting Poiseuille damper performance through use of Couette rheometric data. It is helpful at this point to divide

(a)

Ideal Couette Flow

(b)

Poiseuille Flow (Zero Electric Field)

(c)

Poiseuille Flow (High Electric Field)

Figure 5.8: Velocity, Shear Stress and Shear Rate Distributions

the discussion into zero electric field flow and high electric field flow.

5.5.2.3 Poiseuille Flow - Zero Electric Field

When there is no electric field present, a first approximation the fluid behaviour is Newtonian. (All of the theoretical analysis presented will assume constant apparent viscosity, μ_p) In the absence of any yield stress, we can substitute the following Newtonian relationship between shear stress and shear rate:

$$\tau = \mu_p \left(\frac{\partial v}{\partial h} \right) \quad (5.6)$$

into equation (5.5) and arrive at an equation for the velocity profile in the gap:

$$v_{zero} = \frac{\Delta P_{zero} H^2}{2\mu_p L} \left[1 - \left(\frac{h}{H} \right)^2 \right] . \quad (5.7)$$

Integration of this velocity over the area of the gap produces the volumetric flow rate:

$$Q_{zero} = \frac{2W\Delta P_{zero} H^3}{3\mu_p L} . \quad (5.8)$$

The shear rate distribution for this flow can be found from equations (5.5) and (5.6), and is similar to the shear stress, as it also varies linearly from zero at $h=0$ to a maximum at the electrode wall:

$$\frac{\partial v}{\partial h}_{zero} = \frac{\Delta P_{zero} h}{\mu_p L} . \quad (5.9)$$

The average shear rate can then be obtained by evaluating the shear rate at midpoint

between the centre of flow and the electrode wall:

$$\frac{\partial v}{\partial h_{zero, avg}} = \frac{\Delta P_{zero} H}{2\mu_p L} . \quad (5.10)$$

5.5.2.4 Poiseuille Flow - High Electric Field

At high electric field, the ER fluid develops a yield stress, and those sections of the fluid which do not exhibit a shear stress higher than the yield stress are entrained into a 'solid' plug. In Poiseuille flow, this plug exists in the centre of the gap (Figure 5.8(c)), and a flat velocity profile results across the plug. While the shear stress distribution is of the same form as the zero field case, the shear rate profile is not. Therefore, an analysis of the two fluid states at the same shear rate is not possible in Poiseuille flow, as it is in Couette flow. The approach taken by the author is to compare the zero and high field average shear rates for the same volumetric flow rate, Q . This requires evaluation of the velocity profile in the high field strength situation: use of equation (5.5) for the shear stress distribution and equations (5.1) and (5.2) for the Bingham behaviour of the ER fluid gives

$$v_{high} = \left(\frac{\Delta P_{high}}{2\mu_p L} \right) [H^2 - h_o^2 - 2h_o(H - h)] \quad (5.11)$$

where h_o is the distance from the centre of the gap to the edge of the plug. However, in the plugged region, the fluid velocity is constant and equal to the velocity at h_o :

$$v_{high} = v_o = \left(\frac{\Delta P_{high}}{2\mu_p L} \right) (H - h_o)^2. \quad (5.12)$$

Differentiation of the velocity with respect to h gives the shear rate distribution:

$$\begin{aligned} \frac{\partial v}{\partial h_{high}} &= \frac{\Delta P_{high}}{\mu_p L} (h - h_o), \text{ if } h > h_o \\ \frac{\partial v}{\partial h_{high}} &= 0, \text{ if } h \leq h_o \end{aligned} \quad (5.13)$$

and an average shear rate can be calculated for the active section of fluid:

$$\frac{\partial v}{\partial h_{high, avg}} = \frac{\Delta P_{high} H}{2\mu_p L} \left(1 - \frac{h_o}{H} \right). \quad (5.14)$$

The same result is obtained by simply dividing the velocity of the plug by the distance $(H - h_o)$, imitating a Couette situation generated by the moving plug. These results indicate that unlike Couette flow, the shear rate cannot be considered the same for both field strength situations in Poiseuille flow. As before, integration of the velocity profile over the gap area gives the volumetric flow rate:

$$Q_{high} = \frac{2W\Delta P_{high} H^3}{3\mu_p L} \left[1 - \frac{3h_o}{2H} + \frac{1}{2} \left(\frac{h_o}{H} \right)^3 \right]. \quad (5.15)$$

5.5.2.5 Shear Rate Comparison

The results of equations (5.8) and (5.15) show that if the same pressure difference existed for both flows, the volumetric flow rate would be reduced by the factor in the square brackets in equation (5.15) at high electric field, due to the presence of the plug.

In our situation, the interest is in comparisons of equal volumetric flow rates, in order to evaluate the difference in zero and high electric field force for a given piston velocity. Therefore we require the magnitude of ΔP_{zero} , the zero field pressure difference, that ensures this. Setting equations (5.15) and (5.8) equal gives us the expression

$$\Delta P_{zero} = \Delta P_{high} \left[1 - \frac{3}{2} \left(\frac{h_o}{H} \right) + \frac{1}{2} \left(\frac{h_o}{H} \right)^3 \right]. \quad (5.16)$$

Now we are in a position to compare shear rates in each of the field strength situations, zero and high, at a fixed volumetric flow rate. Substituting the corrected pressure difference from (5.16) into the equation for shear rate at zero field strength (5.10) to ensure similar volumetric flow rates gives us a corrected average shear rate equation:

$$\frac{\partial v}{\partial h}_{zero, avg} = \frac{\Delta P_{high} H}{2\mu_p L} \left[1 - \frac{3}{2} \left(\frac{h_o}{H} \right) + \frac{1}{2} \left(\frac{h_o}{H} \right)^3 \right]. \quad (5.17)$$

Comparison with the average shear rate at high field (equation (5.14)) shows that the shear rates differ only by the values in the brackets. Dividing the expression for average shear rate at high field excitation by the expression for average shear rate at zero field excitation gives us a shear rate correction factor, called κ :

$$\kappa = \frac{\left[1 - \frac{h_o}{H} \right]}{\left[1 - \frac{3}{2} \left(\frac{h_o}{H} \right) + \frac{1}{2} \left(\frac{h_o}{H} \right)^3 \right]}. \quad (5.18)$$

Figure 5.9 shows the variation of κ with fraction of plugged flow, h_o/H . For the purposes of this argument, the shear rates can be considered equal up to a fraction of plug flow of approximately 0.5. However, the increase in κ tends to be exponential; for higher

Figure 5.9: Shear Rate Correction Factor (κ) vs. Plug Fraction (h_0/H)

fractions of plug flow the difference in shear rate becomes very significant. In fact, the analysis predicts that if the fraction of plugged flow reaches a value of 0.9, the shear rate at high field excitation becomes almost seven times the amount at zero field excitation. Thus, unlike Couette flow, the two fluid states, zero and high field, can not be evaluated at the same shear rate. In predicting performance through use of Couette rheometric data, an increased shear rate must be used when changing from zero to high electric field strength. In the context of our objective, which is to maximize the difference between the zero field damping force and the high field damping force, this result can only be beneficial, as shear stress (usually) increases with shear rate.

Therefore, the design goal of damper TTRL 'B' must be redefined. While it is still true that the damper is to take advantage of higher values of shear ratio, it is as a result of decreased volumetric flow rate, Q , in the fluid gap, and not directly due to a reduced shear rate. We can say with certainty that at zero field strength, a decrease in Q results in lower shear rates and lower damping forces, which is desirable, and the same objective as with reduced shear rates in Couette flow. At higher field strengths, the issue is complicated by plug flow, but the prediction is made that plug flow has the effect of increasing the effective shear rate, resulting in similar or higher forces for high field excitation, depending on the fraction of plugged flow. This implies even further enhancement of the shear ratio, surpassing the increase demonstrated by the Couette data. In summary, the performance due to a reduction in volumetric flow rate through the active fluid gaps in an ER damper are expected to be at least equivalent to those predicted by the Couette rheometer results in Chapter III.

5.5.3 Design Conceptualization

In order to reduce the volumetric flow rate of ER fluid in the active gaps, a departure from the piston electrode design of TTRL damper 'A' must be made, since the relative velocity across the attachment points of the damper cannot be changed. The design selected is that of the Fixed Electrode type, in which the fluid gaps are annuli, created by concentric cylindrical electrodes that are fixed relative to the damper casing as in Figure 5.5(b). The damper piston moves inside the innermost cylinder at a fine tolerance, indirectly forcing fluid through the gaps when it moves relative to the damper casing. Furthermore, the damper can be designed for adjustability, by allowing additional cylinders to be added and the piston diameter reduced, so that a number of fluid gaps of the same thickness are created. The volume of flow generated by the piston is divided over the number of gaps, and the speed of the fluid through the active gaps is thus adjustable - more gaps and a smaller piston obviously leading to reduced flow rates at the same relative velocity across the damper attachment points.

5.5.4 Detailed Design

Once the basic arrangement of the damper was established, a number of design issues still remained. Among these issues are the mechanical sealing of the ER fluid, arranging insulation between live and grounded electrodes and insulation between live electrodes and external components, supply of voltage to the electrodes, provisions for agitation of the fluid, and ease of addition/removal of electrodes, pistons, and ER fluid. A primary issue however is the selection of the size of the fluid gap to be used between

electrodes. There are a number of implications associated with this parameter, the most important being: voltage requirement, damper dielectric strength, damper capacitance, and range of damping forces.

The voltage requirement the ER damper makes of the high voltage supply is a direct result of the size of the fluid gap in the damper. The fluids tested in this study exhibited voltage breakdown at an approximate field strength of 2-3 kV/mm. The high voltage supply used in quarter vehicle testing at Carleton can supply a maximum of 10 kV, resulting a maximum gap width of 3-5 mm, in order to maximize the field strength.

Dielectric strength of the damper is another concern, as voltage breakdown becomes more likely with smaller gap sizes [11]. While the dielectric strength of the fluid does not necessarily change, the nature of fluid flow behaviour can sometimes lead to weaknesses in its insulating quality.

Capacitance of the damper/fluid combination is also of concern as a large capacitance can lead to significant delays in switching between different voltage levels. As mentioned previously, in order for the damper to perform effectively over the required frequency range, switching between states must be accomplished in under 5 ms. Capacitance increases considerably with smaller gap sizes, and this effect must be considered in the design.

The most significant effect of gap size on damper performance however is in the magnitude of the zero and high field damper forces, and the resulting controllability of the damper. Investigations of the effect of gap size on damper forces were made by many researchers including Bullough [11], and Hägele [26]. Their conclusions are quite

similar: smaller gaps, while producing significant levels of force, suffer in terms of controllability - that is the high field force is only marginally larger than the zero field force. Larger gaps may supply adequate controllability, but the magnitude of the maximum force may be insignificant for the damping purposes of the application. This tradeoff is an important design issue that must be evaluated in the context of the specific application.

5.5.4.1 Damping Force Estimation Program (FOREST)

In order to assess the variance in controllability and force for the proposed damper design, a force estimation program, FOREST, was coded in FORTRAN, using the fluid flow analysis presented above. The program uses an 'unrolled' version of the fixed electrode damper; it calculates the performance based on parallel plate flow, as opposed to annular flow. The configuration and geometry of the damper, in terms of the number of cylinders, cylinder length, cylinder thickness, etc, are input along with the Bingham models of the fluids to be used. FOREST then assumes a relative velocity across the damper attachment points, and calculates a volumetric flow rate based on the damper piston area. The pressure difference over the length of the gap is then assumed to be equivalent to that across the damper piston, giving the projected damping force.

Tables of maximum and minimum force output were generated as a result of variation in gap width, relative velocity, and number of electrodes, based on maximum and minimum yield strengths of the EK fluids. Additional modifications to FOREST allowed for computation of volumetric flow rate reduction, and fraction of plugged flow.

Because of the assumptions made by FOREST, the data it generated was used primarily to achieve an understanding of the nature of tradeoffs presented by varying gap thickness, and not for accurate predictions of damping force. These assumptions include: the 'unrolled' approximation of the electrodes, the equivalence of pressure difference across the gap length and pressure difference across the piston, ideal fluid flow behaviour (e.g. no end effects), and the assumption of Bingham behaviour of the ER fluids, which underestimated the potential gains in performance at lowered volumetric flow rates.

A final design gap thickness of 3 mm was selected for the following reasons.

1. Controllability of the damping force is the primary design goal. Predictions of controllability made by FOREST did not reach adequate levels until a thickness of 3 mm was achieved. This is likely a result of design configuration, as smaller gaps do not effectively reduce the flow rate, Q .

2. Adjustability of the damper performance through use of additional cylinders was considered important, both as a means of investigating the effects in an experimental situation, and also as a failsafe measure in the event of unforeseen difficulties with a single configuration. In order to accommodate this feature, the gap thickness would need to be large to offset increases in capacitance that may inhibit performance when the damper contained a number of extra electrodes.

3. TTRL damper 'A' and the Couette rheometer, both equipped with gap thicknesses of 3 mm, demonstrated voltage breakdown when field strengths were in the 2-3 kV/mm range. In order to avoid potential problems of voltage breakdown due to smaller gap sizes, and reduce the probability of complications in interpreting the

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PM-1 3½"x4" PHOTOGRAPHIC MICROCOPY TARGET
NBS 1010a ANSI/ISO #2 EQUIVALENT

PRECISIONSM RESOLUTION TARGETS

performance, a strong argument for keeping the gap size at 3 mm was present.

4. The main goal is to determine the effect of reduced flow rates. To ensure this reduction, a gap width in excess of 2 mm is required, especially with a low number of gaps in the damper.

The final design configuration of TTRL damper 'B' is shown in Figure 5.10. The damper incorporates six removable electrodes, with the outer casing of the damper serving as a final grounded electrode, making a total of six active fluid gaps. The electrodes are held fixed relative to the damper casing by two similar PVC (insulating) end caps which maintain the spacing between electrodes. The length of the electrodes is reduced as the electrodes near the centre of the damper, to reduce the restriction to flow in the upper and lower regions. These reductions in length also provide a reservoir effect at the ends of the active gaps in an attempt to ensure even pressure distributions across each gap. High voltage supply enters through the upper lid of the damper and is distributed alternately to electrodes through an insulated distribution bar and small machine screws that tap into the top of each electrode. In the same manner, the grounded electrodes are connected to the outer casing of the damper.

The damper piston rod enters through a sleeve bearing in the upper lid, as experiments with TTRL damper 'A' showed that o-ring sealing provided excessive frictional force. Air is allowed to enter and leave the damper to offset the volume of fluid displaced by the piston rod.

Figure 5.10: TTRL Damper 'B'

Chapter VI

Damper Testing

6.1 Introduction

This chapter comprises the testing of the new design, TTRL damper 'B', with an emphasis on comparison with the original Carleton ER damper, TTRL damper 'A'. The chapter is divided into two sections. The first section describes the testing performed on the dampers with the MTS (Mechanical Test System) facilities at Carleton University. The focus of the testing in this section is on the damper alone, and provides specific performance independent of other factors, in the form of damping characteristic plots. The second section describes the testing of the dampers when incorporated into the

quarter vehicle vibration testing facility in the Transport Technology Research Laboratory. While these tests do not isolate the performance of the damper as well as tests with the MTS machine, they are important as they illustrate the performance of the dampers in their specified role as semi-active vibration isolators. All tests performed on the dampers incorporated the 2:5 mass ratio zeolite/silicon oil ER fluid.

6.2 MTS Testing

6.2.1 Introduction

The MTS apparatus comprises a single-degree-of-freedom hydraulic actuator, load cell, and control system, primarily intended for cyclical loading of specimens in fatigue and fracture investigations. The apparatus is particularly suitable for testing of the ER dampers developed at Carleton as it demonstrates a great deal of flexibility in its control system. For the purposes of damper testing, the control system was used to specify a position function to be followed by the actuator, and report on the real time position of the actuator and the axial reaction force. When the damper is tested in the apparatus in this manner, the damping force is determined as a function of constant damper velocity, a useful means of evaluating performance at different values of field excitation.

6.2.2 Description of Apparatus

Figure 6.1 illustrates schematically the experimental arrangement. A load cell is connected between the controllable hydraulic actuator and the ER damper. The damper and actuator are fixed securely to a sturdy frame. A microconsole controls the hydraulic

Figure 6.1: MTS Experimental Arrangement

pressure for the actuator in order to follow the prescribed position function, and provides amplification/conditioning of the force and displacement transducer signals from the load cell and linear variable differential transformer (LVDT). Both force and displacement were recorded via outputs from the control panel of the MTS microconsole on a 2 channel digital oscilloscope. In order to generate data for constant velocity, a sawtooth position generation function was used, allowing velocity to be calculated from the slope of the recorded position output, and controlled by specifying the frequency and amplitude of the sawtooth. Sections of constant velocity were isolated in the recorded output, and the corresponding force obtained directly.

A description of the components and their functions are specified as below.

1. ER Damper: The basic structure of the ER damper is described in Chapter V. Care was taken in mounting of the damper to ensure pure axial motion, and rigidity in the mounting frame.
2. Hydraulic Actuator: The ER damper piston positioning was performed with a MTS Model 244.12 hydraulic actuator and servovalve, incorporating a linear displacement sensor.
3. Hydraulic Actuation System: Hydraulic pressure for the actuator was supplied by an MTS distribution system, incorporating a reservoir, pump, motor and heat exchanger. Hydraulic fluid flowrate and direction was controlled by the microconsole control system based on the specified position function, via a hydraulic servovalve.
4. MTS Microconsole: The microconsole uses the position function provided by a MTS 458.90 Function Generator as a reference for positioning of the actuator, with feedback

control supplied by a linear position sensor (LVDT). A MTS 661.19 Force Transducer provided measurement of damping forces to the console for signal conditioning and amplification.

5. Data Collection: Data collection was performed by a Nicolet Pro 30 2 channel digital oscilloscope. Position and force signals from the MTS microconsole were provided via ± 10 V outputs. Data was sampled at 100 μ s intervals, with 12 bit accuracy.

6. High Voltage Generation: Voltage is supplied by a Universal Voltronics high voltage DC power supply with manual voltage control and integral panel voltmeter (range: 16 kV, increment 500 V.)

6.2.3 Experimental Procedure

Before the MTS system was judged to be suitable for testing of the ER dampers, an investigation into the accuracy of the MTS load cell was conducted. The reason for this is that the load cell was designed for use in high stress metal specimen testing, and thus capable of indicating loads up to 25 kN. The dampers tested were not thought to supply forces in excess of 500 N, and therefore a calibration was performed to ensure suitability of the load cell with the proposed experiment. It was decided that the load cell must provide repeatable accuracy within a maximum of ± 5 N over the predicted damping force range specified. This performance goal was achieved. As a cautionary measure, the amplifier gain was recorded and corrected if necessary, and the output was rezeroed, before each test. The displacement transducer, while not as much of a concern because the full range of displacement was being utilized, was also calibrated at the output

voltage.

After the apparatus was judged to be ready for damper testing, a configuration of the damper (in terms of the number of gaps) was selected and ER fluid was added before sealing and mounting in the apparatus. For each configuration of TTRL damper 'B' and for TTRL damper 'A', the following steps were taken. The amplifier gain for both position and force transducers were shunt calibrated and the outputs were rezeroed. The fluid was then agitated with the hydraulic actuator to ensure an even dispersal of particles in the base oil. The frequency and amplitude of the position function generator waveform were set for the first of seven constant velocities to be evaluated: 0.01, 0.02, 0.05, 0.1, 0.2, 0.3, and 0.4 m/s. For reasons unknown to the author, the hydraulic actuator provided higher velocities in compression than in tension, and results for tensile performance are only presented up to 0.35 m/s. These velocities are representative of velocities incurred at frequencies of up to 8 Hz in the quarter vehicle testing facility. The high voltage control was then switched to the first voltage, and the output of the force transducer was examined for consistent data. Samples of at least three cycles of the trace for position and force outputs were captured and recorded on the oscilloscope, representing both tensile (rebound) and compressive (jounce) characteristics of the damper. This procedure was repeated for each electric field strength, and then the position function generator was altered to provide data for the next velocity. After all combinations of the seven velocities and three voltages, 0, 1, and 2 kV/mm were investigated, the damper was removed and the next configuration was prepared and tested. Problems with voltage breakdown at high velocities prevented data collection at 3 kV/mm for the MTS testing

as well as the quarter vehicle testing.

Data from the oscilloscope traces was converted into velocity and force. In the case of velocity, the slope of the sawtooth position function was evaluated to obtain the corresponding constant velocity. Plots of force vs. velocity vs. electric field strength for TTRL damper 'A' and five configurations of TTRL damper 'B' were produced. Positive forces and velocities are tensile (rebound), and negative are compressive (jounce). Also included with the test results are the predictions for maximum (2 kV/mm) and minimum (0 kV/mm) damping force made by the program FOREST.

6.2.4 Results and Discussion

Figures 6.2 to 6.7 describe the MTS performance of TTRL damper 'B' and 'A'. In general the curves resemble the shear stress vs. shear rate curves described in Chapter IV. The slope of the force vs. velocity curves, or the damping coefficient, c , generally increases with velocity for the zero field curve, and varies with velocity for the high field curves. The curves also tend to demonstrate non-zero forces at high field and zero velocity, a function of the yield stress feature of ER fluids not common with traditional oil dampers or semi-active EVO dampers.

As predicted for TTRL damper 'B', the configurations that offer less volumetric flow rate through the active gaps provide significantly more controllability. However, this controllability does not come without the expense of reduced forces. For the single gap configuration, considerable force is produced, approximately 600 N (compressive) at 0.45 m/s and 2 kV/mm. The controllability at this velocity is approximately 3.2. With the

Figure 6.2: Force (N) vs. Velocity (m/s) for 1 Gap TTRL 'B'

Figure 6.3: Force (N) vs. Velocity (m/s) for 2 Gap TTRL 'B'

addition of another gap, the force is reduced to 170 N, and the controllability rises to 4.5. This same tradeoff of force for controllability is evident with reduced velocity. The two gap configuration displays controllabilities of 4.5 at 0.35 m/s and near 40 at 0.05 m/s but with a 33% loss of high field force.

As more gaps are implemented, rather erratic performance results. This appears to be a result of the complicated flow path the damper provides, a possible function of varying differential pressures over the gap lengths. Also, the pressure distribution may change from gap to gap, resulting in potential zero flow situations for gaps with lower pressure differences, and 'selective' flow patterns. For the 3, 4, and 5 gap configurations, a continuation of the trend of less force and more controllability is observed, as a result of reduced volumetric flow rate per active gap. The controllability performance for the 5 gap configuration is surprisingly poor, however the forces produced by this configuration are reduced enough to become susceptible to effects of coulomb friction on the damper piston and piston rod, and this may be a factor in reduced performance. The results of the damper with 6 gaps is not presented as the small forces produced introduce uncertainty due to the load cell used. For the purposes of the quarter vehicle testing rig, the most suitable damper configurations were chosen to be the 1 gap and 2 gap configurations, both with similar or better controllability and both with better force production over the range of velocities than TTRL damper 'A'. A table of damping force controllabilities is given below. The values were calculated from division of high field MTS data by zero field MTS data for specific velocities.

Figure 6.4: Force (N) vs. Velocity (m/s) for 3 Gap TTRL 'B'

Figure 6.5: Force (N) vs. Velocity (m/s) for 4 Gap TTRL 'B'

Figure 6.6: Force (N) vs. Velocity (m/s) for 5 Gap TTRL 'B'

Figure 6.7: Force (N) vs. Velocity (m/s) for TTRL 'A'

Table 6.1: Damping Force Controllabilities

Velocity	TTRL A	TTRL B 1 gap	TTRL B 2 gap	TTRL B 3 gap	TTRL B 4 gap	TTRL B 5 gap
0.05	13	12	18	30	38	10
0.10	10	7	10	10	12	4
0.20	5	4	8	9	9	4
0.35	3	3	5	6	4	2

The results indicate that a lower volumetric flow rate damper does result in better controllability performance over the range of velocities tested. While the MTS apparatus prevented testing at velocities expected at the second natural frequency of the quarter vehicle model, and thus a complete picture of comparative performance, there is evidence to support improved controllability at higher velocities, based simply on extrapolation of performance at lower velocities.

Plug flow effects were discussed in Chapter V and predicted to have positive effects on controllability. Plug flow is most prevalent in high field strength, low volumetric flow rate situations and this corresponds to data obtained with configurations of a large number of gaps. The 3 and 4 gap dampers demonstrate controllabilities greater than 30 at low velocity, indicating the likelihood that the average shear rate for high field flow exceeds the average shear rate for zero field flow, since these shear ratios are not possible at a constant shear rate as evidenced by the rheometry results of Chapter IV.

Another feature of the MTS data is the slightly higher force production in compression than in tension. This can be attributed to a problem in the design of the damper. Because the volume occupied by the piston rod must be displaced as the piston moves from its highest to its lowest position, a quantity of air is present at the top of the

damper. On the downward (compressive) stroke, this air follows the piston into a non active region as fluid is pushed through the bottom of the damper and up the active annuli. However, on the upward (tensile) stroke, this air follows the fluid down through the active annuli, thus reducing the capability of the damper to produce force. This phenomenon was verified through manual operation of the damper with the lid removed, and is considered to be a significant deficiency in the performance of TTRL damper 'B'. It not only hinders force capabilities, but also invalidates predictions made by any rudimentary modelling and thus understanding of the ER damper behaviour. The presence of air in the gaps may also account for some of the irregularities found in the MTS data, especially as an increased number of gaps are used. (It should also be remembered the inconsistencies and irreproducibility common in ER fluids, as mentioned in Chapter V.)

FOREST predictions of damping force are shown in the figures. In general, the predictions are quite accurate at relatively low volumetric flow rates. For each configuration, predictions are very good up to approximately 0.05 m/s. At higher velocities, predictions suffer due to the variability in apparent viscosity of the ER fluid at increased shear rates (FOREST assumes constant apparent viscosity.) The linear assumption is also to blame for the overprediction of forces at zero electric field for most of the data. The actual fluid displays lower apparent viscosity increasing to higher values with shear rate. Overall, FOREST is a useful tool for prediction of the general performance of the damper configurations, but it does show limitations, particularly in the way it models ER fluid behaviour.

6.3 Quarter Vehicle Model Testing

6.3.1 Introduction

The quarter vehicle testing rig is designed to investigate the performance of the ER semi-active damper in its role as a suspension element in a two-degree-of-freedom quarter vehicle model. The rig employs sensing equipment and a controllable high voltage supply to invoke a control strategy that provides open loop control of the voltage to the damper in real time. Tests are performed at discrete input frequencies from 0.8 to 14 Hz, for five different control strategies: ride comfort control, wheel hop control, a composite ride comfort/wheel hop strategy, and both minimum and maximum passive damping. Performance of the damper is interpreted through three performance criteria: transmissibility of acceleration to the sprung mass, dynamic tire deflection ratio (wheel hop), and suspension travel ratio, which is a measure of the rattlespace requirement.

While single discrete frequency input testing is not an exact simulation of actual road conditions, its relevance in representing the dominant features of a real vehicle has been argued convincingly [53,54]. The simple assumptions involved with the quarter car model also provide insight into the performance of the system without greatly complicating the analysis.

6.3.2 Description of Apparatus

The apparatus is represented schematically in Figure 6.8. A support stand for the actual testing rig consists of two vertical shafts fixed firmly to a base plate. The sprung and unsprung masses are constrained to move vertically by ball bushings running along

Figure 6.8: Quarter Vehicle Model Experimental Arrangement

the shafts. A cam driven by a variable speed DC electric motor provides the base excitation for the quarter vehicle model. The sprung mass, $m_s = 44.2$ kg, and the unsprung mass, $m_{us} = 6.2$ kg, with slight variances in the unsprung mass for TTRL damper 'B' due to the addition of extra electrodes, when changing configurations. The suspension spring rate, $k_s = 2864$ N/m, and the tire spring rate, $k_{tr} = 26460$ N/m. The components represent a 1/7th scale model of the masses involved with an actual quarter vehicle.

Potentiometers on the testing rig measure the relative displacement between the unsprung mass and the input ($x_2 - x_1$), and the sprung and unsprung mass ($x_3 - x_2$). A calibrated accelerometer is fixed to the sprung mass, and these three signals are used for performance evaluation of the ER damper. In order to implement the control strategies, the relative displacement between the unsprung and sprung mass is sent to an operational-amplifier differentiator to provide the relative velocity between the two masses.

A description of the components and their functions is described below.

1. ER Damper: The basic structure of the ER damper is described in Chapter V. The damper is mounted in the standard vehicle suspension configuration, in parallel with the suspension spring between the sprung and unsprung mass.
2. High Voltage Control System: The high voltage supply system, designed for use with the quarter vehicle testing rig, comprises a high voltage controller, a discharge circuit, and a standard automotive ignition coil and 12 V battery. The system accepts a reference voltage from the computer control/data collection system and uses it to step up the 12 V battery voltage to the desired high voltage with the ignition coil. The discharge circuit

allows for quick discharge of the high voltage to ground, as is necessary for the application. Also implemented is a Beckman digital multimeter for measurement of current in the 12 V circuit, and thus power draw of the damper.

3. Excitation Control: A Boston Gear 1 hp variable speed DC motor drives a cam of 7.5 mm eccentricity through two different gearing ratios, and is capable of speeds from 50 to 1750 rpm, translating into excitations from 0.8 to 14 Hz. A Boston Gear digital tachometer provides speed measurement of the motor in rpm.

4. Computer Control System: A turbo Pascal program monitors inputs through a Computer Boards A/D converter and sends outputs through a Computer Boards D/A converter. Inputs include: the relative displacements $(x_2 - x_1)$ and $(x_3 - x_2)$, a filtered relative velocity $(\dot{x}_3 - \dot{x}_2)$, and a filtered sprung mass acceleration $\ddot{x}_3$. Outputs include the reference signals for the high voltage control and the discharge circuit control. The program loops at a rate faster (300-700x) than the excitation frequency, each loop updating the sampled inputs and correcting, depending on the control strategy, the outputs for voltage control. The program also collects the data from the inputs in order to calculate the performance parameters, as specified below.

6.3.3 Control Strategies

Five control strategies were implemented in the testing of the two ER dampers. They comprised a ride comfort control strategy, a wheel hop control strategy, a composite ride comfort/wheel hop control strategy, and minimum and maximum passive damping strategies. The latter two are used primarily as benchmarks for performance of the

dampers.

The ride comfort control strategy is based upon the ON/OFF relative control strategy discussed in Chapter III, selected predominately for its ease of implementation. The relative displacement and relative velocity across the suspension are multiplied together, a positive result indicating that the forces from the suspension spring and damper are in the same direction, and a negative result indicating the forces are in the opposite direction. The voltage in the damper is set at either maximum or zero, so that the total suspension force on the sprung mass is kept at a minimum. This in turn results in reduced acceleration of the sprung mass.

The wheel hop control strategy is one designed by Wu *et al.* [62], and changes the state of the ER damper between minimum and maximum damping, depending on the sign of the relative velocity across the suspension. The objective is to maintain an acceptable load between the unsprung mass and the input. If the relative velocity indicates the damper is exerting an upward force on the unsprung mass, the voltage is set at zero, and if the force is downwards on the unsprung mass, the voltage is set at maximum.

The composite control strategy, also designed by Wu *et al.* [62], attempts to correct the tradeoff between ride comfort and wheel hop. The strategy is similar to the ride comfort strategy, but also compares the amount of dynamic deflection to the given static deflection between the unsprung mass and the input. If this comparison indicates that the dynamic deflection is low compared to the static deflection, the strategy concentrates on ride comfort. If the dynamic deflection is high compared to the static deflection, the voltage to the ER damper is set for maximum damping, in order to reduce

the amplitude of the unsprung mass vibration.

In the minimum damping passive mode, the voltage is set at zero for the duration of the test. Likewise in maximum damping passive mode, the voltage is set to maximum for the duration of the test.

6.3.4 Performance Criteria

Three performance criteria are evaluated at the end of each test. They are the transmissibility of acceleration to the sprung mass, the dynamic tire deflection ratio, and the suspension travel ratio. The suspension travel ratio is an indication of the rattlespace requirement for the semi-active suspension, and is of secondary importance in evaluating the performance of the ER damper.

After the testing rig is operating at the desired frequency and the control strategy is invoked, the program waits for transient effects to dissipate and then collects data for three complete cycles of the input. The rms value of the acceleration signal, and the two relative displacement signals are then calculated for the three cycles. The transmissibility is then the ratio of the rms value of acceleration of the sprung mass to that of the input excitation. The dynamic tire deflection is the ratio of the rms value of deflection of the unsprung mass to that of the input, and the suspension travel ratio is the ratio of the rms value of deflection between the sprung and unsprung mass to that of the input.

6.3.5 Experimental Procedure

Before testing, the damper is filled with ER fluid and sealed. The damper is then

installed in the quarter vehicle rig, and a series of pre-test verifications are performed to ensure calibrated operation of the four input signals. The potentiometers are also zeroed before each test, due to slight differences in mass between TTRL dampers 'A' and 'B' causing different static deflections. Also, the voltage supply and discharge circuit are tested, and the control voltage correlated to the specified control strategy.

For each control strategy the following steps are taken. The input excitation is set for the first frequency to be tested, and the specified control strategy is invoked on the damper with the computer control system. When consistent data is achieved (variance under approximately 5%), the values of the three performance criteria at the specified frequency are recorded. The input excitation is set to the next frequency, and the process is repeated. After the complete frequency range is investigated, a new control strategy is selected for testing. This procedure is performed for TTRL damper 'A', and two configurations (1 gap and 2 gap) of TTRL damper 'B', based on performance witnessed in MTS results. Plots are created of performance criterion vs. frequency vs. control strategy for each damper configuration tested.

6.3.6 Results and Discussion

Results of the quarter vehicle testing are shown in Figures (6.9) to (6.17). Interpretation of the results of data from the quarter vehicle test rig is somewhat difficult, due to combined effects on system performance from each of the control strategy, the high voltage supply unit, and the ER damper. Test results are shown for TTRL damper 'A' and two configurations of TTRL damper 'B', tested with the same 2:5 mass ratio

mixture of zeolite/silicon oil ER fluid. Configurations with more than two gaps resulted in low damping forces and poor performance of the damper in the quarter vehicle test rig.

For the 1 gap configuration, transmissibility performance (Figure 6.9) shows a tendency towards overdamping compared to performance of TTRL damper 'A' (Figure 6.11), due to the significantly higher damping forces produced by this configuration. The resonance of the sprung mass is significantly reduced, and the peak frequency has shifted to 2 Hz, from 1 Hz with TTRL damper 'A'. The midrange response displays early converging of the two passive modes, which can be attributed to the loss of controllability in the configuration, due to its strong dependence on normal viscous effects at high frequencies. It appears that the flow rates are simply too high in the single active gap. Here, the semi active performance does not illustrate the advantages that it does in TTRL damper 'A' (Figure 6.11) and the 2 gap configuration of TTRL damper 'B' (Figure 6.10), which show reduced resonance at the first natural frequency and low transmissibility for the remaining frequencies, beating the tradeoff one endures with a passive arrangement.

As with all transmissibility results, the different semi-active control strategies have little effect on performance. The composite control is expected to perform similarly to ride comfort control as it is essentially the same strategy for low dynamic deflection situations. The wheel hop control similarity may be attributable to the switching strategies. The wheel hop control strategy is based on the sign of relative suspension velocity, and switches every half cycle. The ride comfort control strategy is based on the sign of relative suspension velocity times relative suspension displacement, and switches every quarter cycle. Assuming sinusoidal response, this means that the two strategies

Figure 6.9: Transmissibility vs. Frequency (Hz) for 1 Gap TTRL 'B'

coincide for half of the duration of the test.

Figure 6.10 shows the transmissibility performance of the 2 gap configuration. Comparison of the MTS data between this configuration (Figure 6.3) and TTRL 'A' (Figure 6.7) shows slightly lower zero field force production and higher high field force production for TTRL 'B'. This is evident in the transmissibility data presented. The passive zero field curve demonstrates a higher peak at the body natural frequency and similar performance at all other frequencies. The passive high field curve displays almost identical performance for both dampers, but this is expected at low frequencies due to the nature of the apparatus. Both dampers have the capability of 'locking up' the suspension at high field and low frequencies; since the 'lockup' force is less than the maximum force provided by either damper, the apparatus does not distinguish between the dampers.

The missing data point in the plots for 2 gap performance is a result of resonance that exceeded the limits of operation of the unsprung mass rattlespace, making it impossible to record data at the 8 Hz frequency. This is a problem that existed for any configuration that implemented more than one gap and is a result of low damping force at zero field. While the MTS data shows only slightly less force for 2 gaps than TTRL 'A', it is believed that this further reduction was enough to cause the problem. Thus it seems that the original goal of increased controllability is achieved (reduced zero field force), but that this goal has had detrimental effects when implemented in the quarter vehicle apparatus.

Figures 6.12 to 6.14 show dynamic tire deflection ratio performance for the ER dampers. In general, higher damping is preferable for wheel hop control, with the

Figure 6.10: Transmissibility vs. Frequency (Hz) for 2 Gap TTRL 'B'

Figure 6.11: Transmissibility vs. Frequency (Hz) for TTRL 'A'

exception of the intermediate frequencies. The results presented in the three figures reinforce this, as TTRL damper 'A' (Figure 6.14) and the 2 gap configuration of TTRL damper 'B' (Figure 6.13) provide very similar results, while the increased force production of the 1 gap configuration (Figure 6.12) shows better dynamic tire deflection control at low and high frequencies, but reduced control at intermediate frequencies. In terms of semi-active performance, the wheel hop control strategy displays slightly superior performance over the other semi-active control strategies in the median frequency range for all dampers. While this difference is not significant, all semi-active strategies illustrate the advantage over passive strategies, which suffer from either high sprung mass transmissibility or high midrange transmissibility.

Figures 6.15 to 6.17 indicate the amount of suspension travel required for each control strategy and each damper. While this data is merely an indication of the suspension travel requirement and not as indicative of 'performance' as the other two criteria, the data is useful for comparison of the two dampers.

Figure 6.15 shows the performance of TTRL damper 'A'. At low frequencies, there is a significant difference between the two passive strategies, the zero field passive indicating large oscillations at the first natural frequency, and the high field passive locking the suspension for frequencies below approximately 2 Hz. Above 2 Hz, there exists virtually no difference between any of the five control strategies.

Figure 6.16 shows the performance of the one gap configuration of TTRL damper 'B'. Again, due to the excessive forces involved, the apparatus does not significantly distinguish between any of the control strategies - the suspension is overdamped for all

Figure 6.12: Dynamic Tire Deflection vs. Frequency (Hz) for 1 Gap TTRL 'B'

Figure 6.13: Dynamic Tire Deflection vs. Frequency (Hz) for 2 Gap TTRL 'B'

Figure 6.14: Dynamic Tire Deflection vs. Frequency (Hz) for TTRL 'A'

Figure 6.15: Suspension Travel Ratio vs. Frequency (Hz) for TTRL 'A'

Figure 6.16: Suspension Travel Ratio vs. Frequency (Hz) for 1 Gap TTRL 'B'

cases. However, at higher frequencies, the different control strategies do not converge together as they do in TTRL damper 'A', indicating that the improvement in force production has allowed a degree of control at high frequencies that is not present for TTRL damper 'A'.

Investigation of Figure 6.17 demonstrates the same result. While the suspension travel ratio has not been reduced, separation of the control schemes at these frequencies implies an improved suspension control range.

It should be remembered that these differences in performance at high frequency are only appreciable in the suspension travel ratio and to an almost insignificant extent in the dynamic tire deflection ratio performance. Therefore, while the redesigned damper indicates a degree of improved controllability of the suspension at higher frequencies, the improvements are not substantial enough to overcome the effects of extraneous causes, whether they be due to the nature of the apparatus, or to transient performance of the fluid.

Figure 6.17: Suspension Travel Ratio vs. Frequency (Hz) for 2 Gap TTRL 'B'

Chapter VII

Conclusions and Recommendations

7.1 Conclusions

7.1.1 ER Fluid Mechanical Properties

1. The Bingham plastic representation of ER fluid behaviour is at best a first approximation. The apparent viscosity of the fluid appears to be a function of shear rate, especially at high electric field excitation. For the 2:5 mass ratio zeolite/silicon oil mixture at 3 kV/mm, the apparent viscosity varies from 0.31 at a shear rate of 250 s^{-1} , to 0.03 at 1500 s^{-1} , to 0.17 at 3000 s^{-1} .

2. Doubling of the amount of zeolite dispersant results in similar controllabilities, and approximately twice the zero field viscosity: 0.11 for the 2:5 mass ratio fluid and

0.19 for the 4:5 mass ratio fluid at 33 °C.

3. The zeolite mixtures display a range of superior shear ratios, from 18.5 to 2.5 between shear rates of 250 s^{-1} and 2000 s^{-1} . Above 2000 s^{-1} , the shear ratios drop from 2.5 to 2.0. This has a significant effect on the applications of ER fluids.

4. While an improvement over the starch based and proprietary fluids, the zeolite/silicon oil mixtures provide only adequate ER effect in terms of controllability. It has been reported that there are fluids available, such as Lord VersaFlo ER-201, with significantly increased ER activity over the fluids tested.

5. The aluminum oxide/silicon oil mixture did not provide adequate ER effect. In contrast to common ER behaviour, the current flow allowed by the mixture displayed an inverse dependency on shear rate. This may be of interest to ER researchers pursuing the causes of the ER effect, as behaviour of the ER effect and electric current are seen to be closely linked [11].

6. For the zeolite/silicon oil fluids, the ER effect is still prevalent at temperatures of -6 °C, however controllability is reduced by increased zero field viscosity. At 57 °C, the ER effect is present, but increased current and reduced dielectric strength limit high field excitation, thus reducing controllability.

7.1.2 TTRL Dampers 'A' and 'B'

1. Shear rates differ for each field strength in constant flowrate Poiseuille flow, preventing the direct use of Couette shear stress data for evaluation of force and controllability. Shear rates are increased for high electric field Poiseuille flow, due to

plug flow effects.

2. Reduction of volumetric flow rate in TTRL damper 'B' provides increased force controllability. While a strong trade off between controllability and force magnitude exists for TTRL damper 'B', the 1 gap and 2 gap configurations yield similar or improved controllability and improved force production over TTRL damper 'A'.

3. A large number of gaps leads to erratic behaviour of TTRL damper 'B', possibly due to variations in pressure distribution for each gap.

4. The presence of air in TTRL damper 'B' causes problems with force production, and may also contribute to erratic behaviour.

5. The computer program, FOREST, provides good predictions of damper performance for low volumetric flow rates. At higher flow rates, predictions suffer due to approximations made by FOREST, in particular, the constant apparent viscosity in the Bingham representation. FOREST adequately predicts overall performance of the ER damper, in terms of force magnitude and the effect of changing geometric parameters.

6. Improvements in the force production and controllability of TTRL damper 'B' do not necessarily result in improved quarter vehicle performance in terms of the three criteria: transmissibility, dynamic tire deflection, and suspension travel ratio. This is possible due to the fact that the mechanical properties of the ER fluid cannot respond adequately to the rapidly varying electric field. Excessive force production for the 1 gap configuration of TTRL damper 'B' resulted in overdamping, and reduced zero field force production for the 2 gap configuration prevented complete data collection near the natural frequency of the unsprung mass.

7. While both semi-active dampers provide little or no improvement over passive dampers at high frequencies, possibly due to the aforementioned transient behaviour of the fluid, TTRL damper 'B' indicated increased controllability when compared to TTRL damper 'A' in terms of the suspension travel ratio.

7.2 Recommendations

7.2.1 ER Fluid Mechanical Properties

1. For modelling and simulation purposes, an improved representation of fluid behaviour, incorporating variable apparent viscosities, would be beneficial.

2. Based on the experimental data of the ER fluids tested, it appears that in order to develop commercially viable ER dampers, there is a need for new ER fluids which demonstrate lower apparent viscosities, higher shear strengths (of the order of 1 kPa), and lower temperature dependence over the range from -40 to 120 °C.

7.2.2 TTRL Dampers 'A' and 'B'

1. An investigation into transient behaviour of the ER fluid in an ER damper should be made in order to identify possible hindrances to performance.

2. Experimental evaluation of the effect of changing the active fluid gap size on damper performance should be studied.

3. The problem of air in the ER damper can be reconciled through the use of a reservoir. This will reduce problems with low force production, erratic data, and allow easier prediction of fluid behaviour.

4. High velocity measurement of ER damper performance with the MTS facilities would provide a direct indication of force controllability at higher frequencies.

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Appendix A

Damper Force Estimation Program (FOREST)

PROGRAM FOREST

C-----

C Written by: Jamie Catania, March 21, 1993

C IRDA - inner radius

C ORDA - outer radius (of damper)

C NC - number of cylinders

C NG - number of gaps

C CT - cylinder thickness

C GT - gap thickness

C IRG(I) - inner radius of gap 1,2,3...

C ORG(I) - outer radius of gap 1,2,3...

C CIRC(I) - average circumference of gap 1,2,3...

C TA - total effective area of fluid contact

C PISA - piston area (maximum possible)

C GAPA - total gap area

C MAGF - magnification factor of speed through annuli

C DIAMP - diameter of the piston

C DIAMD - diameter of the damper

C BREADT - total circumference of all gaps

C MUPLSL - Bingham plastic viscosity at low electric field

C MUPLSH - Bingham plastic viscosity at high electric field

C VREL - relative velocity of the piston

C MAXYS - Maximum yield stress of the ER fluid (Bingham Plas.)

C MINYS - Minimum yield stress of the ER fluid (Bingham Plas.)

C FILNM1 - output file name for analysis

C FILNM2 - output file name for grapher software

C CNTRAT - force control ratio of the damper

C PLGRT1 - ratio of plug size to gap size at low electric field

C PLGRT2 - ratio of plug size to gap size at high electric field

C F1,F2 - range limits used in rootfinding for force

C F3 - recursive estimate of force used in rootfinding

C FCF1,2,3 - polynomial function in force evaluated at F1, F2, F3

C SFCF1,2,3 - sign of polynomial function in force evaluated at F1, F2, F3

C RANGE - range of force estimation (F2-F1)

C ROOT1 - final value of force estimation at low field excitation

C ROOT2 - final value of force estimation at high field excitation

C NIGS - number of iterations of the gap size

C SGSI - starting gap size
 C GSIN - gap size iteration increment
 C SHRAT1 - shear rate estimation at low electric field strength
 C SHRAT2 - shear rate estimation at high electric field strength
 C A1,2,3 - low field force polynomial parameters
 C B1,2,3 - high field force polynomial parameters

C-----

C234567

```

INTEGER NC,NG,I,J,K,M,N,NIGS
REAL IRDA,CT,GT,L,IRG(30),ORG(30),ORDA,TA,PI,PISA,GAPA,MAGF
REAL DIAMP,DIAMD,BREADT,MUPLSL,MUPLSH,VREL,CIR(30),CNTRAT
REAL MAXYS,MINYS,PLGRT1,PLGRT2
REAL F1,F2,F3,FCF1,FCF2,FCF3,SFCF1,SFCF2,SFCF3,RANGE,ROOT1,ROOT2
REAL NIGS,GSIN,SGSI,SHRAT1,SHRAT2
CHARACTER *15 FILNM1,FILNM2
DOUBLE PRECISION FCTN1,FCTN2,A1,A2,A3,B1,B2,B3,CAPAC,PERM
  
```

C-----

C Input data collection

C-----

```

WRITE(6,*) 'Enter the output filename (including extension)...'
READ(6,*(A)) FILNM1
OPEN(1,FILE=FILNM1,STATUS='UNKNOWN')
WRITE(6,*) 'Enter the output grapher filename (inc. extension)...'
READ(6,*(A)) FILNM2
OPEN(2,FILE=FILNM2,STATUS='UNKNOWN')
WRITE(6,*) 'Enter all geometric data in metres.'
WRITE(6,*) '(The piston radius will be 1mm smaller'
WRITE(6,*) ' than the SMALLEST cylinder.)'
  
```

C WRITE(6,*) 'Enter the number of cylinders...'

C READ(6,*) NC

C NC = NC + 1

C Note: the program counts the outer damper casing as the first cylinder

```

WRITE(6,*) 'Enter the number of iterations of the gap size...'
  
```

```

READ(6,*) NIGS
  
```

```

WRITE(6,*) 'Enter the starting gap size...'
  
```

```

READ(6,*) SGSI
  
```

```

WRITE(6,*) 'Enter the gap size increment (0 for one iteration)...'
  
```

```

READ(6,*) GSIN
  
```

C WRITE(6,*) 'Enter the cylinder thickness...'

C READ(6,*) CT

```

WRITE(6,*) 'Enter the fluid approximations.'
  
```

```

WRITE(6,*) 'Enter the zero voltage yield stress (in Pa)...'
  
```

```

READ(6,*) MINYS
  
```

```

WRITE(6,*) 'Enter the high voltage yield stress (in Pa)...'
  
```

```

READ(6,*) MAXYS
WRITE(6,*)'Enter plastic viscosity for zero voltage (in Pa-s)...'
READ(6,*) MUPLSL
WRITE(6,*)'Enter plastic viscosity for high voltage (in Pa-s)...'
READ(6,*) MUPLSH
C  WRITE(6,*)'Enter the relative velocity in (m/s)...'
C  READ(6,*) VREL
  WRITE(1,3) MINYS,MUPLSL
  WRITE(1,4) MAXYS,MUPLSH
3  FORMAT(2X,'Zero voltage yield stress :',F7.1,' Pa',
1  ', Plastic viscosity :',F6.2,' Pa-s')
4  FORMAT(2X,'High Voltage yield stress :',F7.1,' Pa',
1  ', Plastic viscosity :',F6.2,' Pa-s')
C-----
C  Main Program
C-----
  CT = 0.003
  ORDA=0.048
  L=0.180
  PI = 3.141592654
  NC = 1
C-----
C  K is the number of cylinders to compute (1 to 6) (2 to 7 for program)
C  Main loop
C-----
  DO 500 K=1,6
    NC = NC + 1
    VREL = 0.0
C-----
C  M is the number of velocities to compute (1 to ?)
C  First nested loop
C-----
  DO 500 M=1,4
    C  velocity iteration increment is added here
    VREL = VREL + 0.01
    GT = SGSI - GSIN
C-----
C  N is the number of gap sizes to compute (1 to NIGS)
C  Second nested loop
C-----
  DO 500 N=1,NIGS
    GT = GT + GSIN
    WRITE(1,20) NC,CT,VREL,GT
C-----

```

```

20  FORMAT(/2X,'There are',I2,' cylinders (including the outer casing)
      1 of thickness ',F5.3,' m',/2X,'Relative velocity :',F4.2,' m/s',
      2 10X,'Gap thickness :',F8.5,' m')

```

C-----

C Geometric Calculations

C-----

```

      NG=NC-1
      IRDA=ORDA-NG*(CT+GT)
      I=1
      IRG(I) = IRDA + CT
      ORG(I) = IRDA + CT + GT
      CIRC(I) = PI*(IRG(I)+ORG(I))
      WRITE(1,9)
9    FORMAT(/4X,'gap#   Inner radius   Outer radius   Circum.']/)
      WRITE(1,11) I,IRG(I),ORG(I),CIRC(I)
      DO 10 I=2,NG
          IRG(I) = IRG(I-1) + CT + GT
          ORG(I) = ORG(I-1) + CT + GT
          CIRC(I) = PI*(IRG(I)+ORG(I))
10   WRITE(1,11) I,IRG(I),ORG(I),CIRC(I)
11   FORMAT(3X,I3,F15.4,F16.4,F16.4)
      PISR = IRDA - 0.001
      PISA = PISR*PISR*PI
      BREADT = 0.0
      GAPA = 0.0
      CAPAC = 0.0
      PERM = 3.8*8.84E-12
      DO 30 I=1,NG
          CAPAC = CAPAC + (PI*L*2.0*PERM)/(LOG(ORG(I)/IRG(I)))
          BREADT = BREADT + CIRC(I)
30   GAPA = GAPA + PI * (ORG(I)*ORG(I) - IRG(I)*IRG(I))
      CAPAC = CAPAC * 1E12
      MAGF = PISA/GAPA
      DIAMP = PISR*2
      DIAMD = ORG(NG)*2
      H=GT/2
      WRITE(1,35) BREADT
      WRITE(1,36) DIAMP
      WRITE(1,37) DIAMD
      WRITE(1,39) PISA
      WRITE(1,45) GAPA
      WRITE(1,46) MAGF
      WRITE(1,47) CAPAC

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C-----

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35  FORMAT(4X,'Total circumference of the gaps :',F10.5,' m')
36  FORMAT(4X,'Diameter of the piston :',F19.3,' m')
37  FORMAT(4X,'Diameter of the damper :',F19.3,' m')
39  FORMAT(4X,'Piston area :',F30.8,' m^2')
45  FORMAT(4X,'Total gap area :',F27.8,' m^2')
46  FORMAT(4X,'Speed magnification factor :',F15.2)
47  FORMAT(4X,'Capacitance of damper :',F20.1,' pF'/)
C-----
C    Calculation of force polynomial parameters (low and high field)
C-----
      A1 = (8*BREADT*H*H*H)/(3*MUPLSL*L*PI*DIAMP*DIAMP)
      B1 = (8*BREADT*H*H*H)/(3*MUPLSH*L*PI*DIAMP*DIAMP)
      A2 = (PI*DIAMP*DIAMP*VREL*0.25) + (BREADT*H*H*MINYS/MUPLSL)
      B2 = (PI*DIAMP*DIAMP*VREL*0.25) + (BREADT*H*H*MAXYS/MUPLSH)
      A3 = (BREADT*PI*PI*L*L*MINYS**3*DIAMP**4)/(48*MUPLSL)
      B3 = (BREADT*PI*PI*L*L*MAXYS**3*DIAMP**4)/(48*MUPLSH)
C-----
C    WRITE(1,65) A1,A2,A3
C    WRITE(1,65) B1,B2,B3
C65  FORMAT(/1X,F15.8,F15.8,F15.8)
C    DO 70 J=-70,700
C      FCTN1 = (A1*J*J*J)-(A2*J*J)+A3
C      FCTN2 = (B1*J*J*J)-(B2*J*J)+B3
C70  WRITE(1,75) J,FCTN1,FCTN2
C75  FORMAT(I10,F20.8,F20.8)
C-----
C    Rootfinding routine for low field force polynomial
C-----
      F1= (2*A2)/(3*A1)
      F2= 200000.0
      RANGE = ABS(F2-F1)
90  IF (RANGE .GE. 1.0) THEN
      FCF1 = (A1*F1*F1*F1)-(A2*F1*F1)+A3
      FCF2 = (A1*F2*F2*F2)-(A2*F2*F2)+A3
      SFCF1 = SIGN(1.0,FCF1)
      SFCF2 = SIGN(1.0,FCF2)
      F3 = (F1+F2)/2.0
      FCF3 = (A1*F3*F3*F3)-(A2*F3*F3)+A3
      SFCF3 = SIGN(1.0,FCF3)
      IF (SFCF1*SFCF2 .GT. 0.0) THEN
        WRITE(6,*) 'There is an error in rootfinding.'
        GOTO 100
      ENDIF
C    IF (ABS(FCF3)0.LT. 0.05) THEN

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C          WRITE(6,*) 'ROOT FOUND WITH BISECTOR'
C          GOTO 100
C      ENDIF
C      IF(SFCF1*SFCF3 .GT. 0.0) THEN
C          THE ROOT IS BETWEEN F3 AND F2
C          F1 = F3
C          RANGE = ABS(F2-F1)
C      ELSE
C          THE ROOT IS BETWEEN F1 AND F3
C          F2 = F3
C          RANGE = ABS(F2-F1)
C      ENDIF
C      GOTO 90
C  ENDIF
100  ROOT1 = F3
C-----
C      Rootfinding routine for high field force polynomial
C-----
      F1= (2*B2)/(3*B1)
      F2= 200000.0
      RANGE = ABS(F2-F1)
190  IF (RANGE .GE. 1.0) THEN
      FCF1 = (B1*F1*F1*F1)-(B2*F1*F1)+B3
      FCF2 = (B1*F2*F2*F2)-(B2*F2*F2)+B3
      SFCF1 = SIGN(1.0,FCF1)
      SFCF2 = SIGN(1.0,FCF2)
      F3 = (F1+F2)/2.0
      FCF3 = (B1*F3*F3*F3)-(B2*F3*F3)+B3
      SFCF3 = SIGN(1.0,FCF3)
      IF (SFCF1*SFCF2 .GT. 0.0) THEN
          WRITE(6,*) 'There is an error in rootfinding.'
          GOTO 200
      ENDIF
C      IF (ABS(FCF3) .LT. 0.05) THEN
C          WRITE(6,*) 'ROOT FOUND WITH BISECTOR'
C          GOTO 200
C      ENDIF
C      IF(SFCF1*SFCF3 .GT. 0.0) THEN
C          THE ROOT IS BETWEEN F3 AND F2
C          F1 = F3
C          RANGE = ABS(F2-F1)
C      ELSE
C          THE ROOT IS BETWEEN F1 AND F3
C          F2 = F3

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        RANGE = ABS(F2-F1)
    ENDIF
    GOTO 190
ENDIF
200 ROOT2 = F3
C-----
    WRITE(1,202)
    WRITE(1,203)
202  FORMAT(44X,'zero voltage      high voltage')
203  FORMAT(44X,'-----')
    WRITE(1,205) ROOT1,ROOT2
205  FORMAT(4X,'Predicted force :',21X,F10.2,' N',F16.2,' N')
C-----
C   Plug ratio calculation
C-----
    PLGRT1 = 100*(PI*DIAMP*DIAMP*L*MINYS)/(4*ROOT1*H)
    PLGRT2 = 100*(PI*DIAMP*DIAMP*L*MAXYS)/(4*ROOT2*H)
    WRITE(1,215) PLGRT1,PLGRT2
215  FORMAT(4X,'Percentage of plugged flow :',12X,F8.1,' %',F16.1,' %')
C-----
C   Shear rate calculation
C-----
    SHRAT1 = (ROOT1*H*(1-(PLGRT1/100.0)))/(PISA*2.0*MUPLSL*L)
    SHRAT2 = (ROOT2*H*(1-(PLGRT2/100.0)))/(PISA*2.0*MUPLSH*L)
    WRITE(1,220) SHRAT1,SHRAT2
220  FORMAT(4X,'Approximate equivalent shear rate :',6X,F7.1,' s^-1',
1 F13.1,' s^-1')
C-----
C   Force control ratio calculation
C-----
    CNTRAT = ROOT2 / ROOT1
    WRITE(1,225) CNTRAT
225  FORMAT(4X,'Force control ratio :',F10.4//)
    WRITE(2,230) NC,VREL,GT,MAGF,ROOT1,ROOT2,PLGRT1,PLGRT2,SHRAT1,
1 SHRAT2,CNTRAT,CAPAC
230  FORMAT(I2,F5.2,F7.4,F6.1,F8.1,F8.1,F5.1,F5.1,F8.0,F8.0,F6.2,F7.0)
500  CONTINUE
C-----
    CLOSE(1)
    CLOSE(2)
    STOP
    END

```


END

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FIN