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**THE RHEOLOGICAL PROPERTIES OF ELECTORRHEOLOGICAL FLUIDS AND
THEIR APPLICATION TO CONTROLLABLE DAMPERS FOR GROUND VEHICLES**

by

Christopher M. Bortolotto, BSME

A thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfillment 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

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paribus, proportionalem esse velocitati,
qua partes Fluidi separantur ab invicem.

Isaac S. Newton

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"THE RHEOLOGICAL PROPERTIES OF ELECTORRHEOLOGICAL FLUIDS AND
THEIR APPLICATION TO CONTROLLABLE DAMPERS FOR GROUND VEHICLES"

submitted by Christopher M. Bortolotto, BSME
in partial fulfillment of the requirements for
the degree of Master of Engineering.

Thesis Supervisor

Chair, Department of
Mechanical and Aerospace Engineering

Carleton University

September 7, 1995

ABSTRACT

The rheological properties of two electrorheological (ER) fluids are determined with a view to applying ER fluids to the design of controllable dampers for ground vehicles. A Couette coaxial, cylindrical electrorheometer is used to determine the rheological properties. The Bingham plastic is found to be a suitable rheological model. Axial and shear flow models are developed to employ ER fluid data in the design of multiple-annulus controllable dampers for ground vehicles. The axial flow model, coupled with the Bingham plastic model of ER fluids, is suitable for predicting damper behaviour. The shear mode damper develops much lower forces than the axial flow damper, but produces higher control ratios. A preliminary design is performed for an ER damper for a full-size vehicle.

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LIST OF SYMBOLS

A	area circumscribed by piston outside diameter
C	constant of integration
D	outside diameter of piston
E	electric field strength
F	damping force
k	dimensionless radius of inside wall of annulus
L	length of piston
N_{Re}	Reynolds number
ΔP	pressure drop across length of piston
Q	flow rate through piston
r	radius
r_r	ratio of total to zero-field shear stress (τ/τ_0)
R	radius of outside wall of annulus
S	dimensionless shear stress
S_y	dimensionless yield shear stress
T	temperature
u	flow velocity
v	dimensionless velocity
v_{rel}	relative velocity across damper
x	dimensionless radial coordinate
$\dot{\gamma}$	shear rate
λ	value of x at which $\tau = 0$
λ_-, λ_+	dimensionless radial limits of plug flow regime
μ_d	differential viscosity

μ_{pl}	plastic viscosity
μ_0	pseudoplastic differential viscosity at $\dot{\gamma} = 0$
μ_∞	pseudoplastic differential viscosity at high shear rates
τ	(total) shear stress
τ_0	shear stress with zero electric field strength
τ_{ER}	electric field-induced (ER) shear stress ($\tau - \tau_0$)
τ_w	shear stress at wall of gap
τ_y	dynamic yield shear stress

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I. INTRODUCTION

1.1 Introduction

Electrorheological (ER) fluids experience rheological changes when subjected to an external electric field. An ER fluid (ERF) responds to electrical excitation with an increase in yield shear stress and apparent viscosity. This phenomenon, known as the ER effect, has been acknowledged as having potential applications in vibration control, torque transmission, position control and fluid power. Potential devices include vehicle dampers, vibration mounts, clutches, brakes, hydraulic valves and robotics.

At the Carleton University Transport Technology Research Laboratory, there is interest in ER technology for use in semi-active ground vehicle suspensions. Such systems would include a passive spring element and a controllable damping element. The ability to accurately and reversibly modulate ERF behaviour with an electric field presents a promising means for the control of damping.

The main obstacles to overcome in such a project include knowledge of ERF behaviour (fluid modelling), flow modelling, damper design, electrical hardware design, control strategy and software design.

This thesis deals with the rheology (fluid behaviour), flow modelling and damper design. These are problems that must be overcome before system development can occur.

1.2 Background

Changes in the apparent viscosity of certain fluids under the influence of an electric field were reported as early as 1896 [1]. In 1949, Winslow reported these 'electroviscous' effects in suspensions, and attributed them to fibrillation of the particulate phase within its fluid carrier [1]. The notion of the electroviscous effect remained until the last few decades, when researchers started suggesting an electroplastic effect, or more generally, an electrorheological (ER) effect.

When subjected to a shear stress, a solid will deform with a corresponding strain. A fluid under shear stress will flow with a corresponding strain rate, or 'shear rate' - the time derivative of strain. All other factors being equal, the constant of proportionality between shear rate and shear stress is viscosity; the measure of a fluid's resistance to flow. Likewise, under an axial pressure gradient, the shear stresses developed in the flow field which resist pressure are due to viscosity.

Newtonian fluids such as air, oil, water and many homogenous fluids have linear shear stress (τ) vs. shear rate ($\dot{\gamma}$) characteristics which intersect the origin $(\dot{\gamma}, \tau) = (0,0)$ (Figure 1). The slopes of these characteristics are constant at given temperatures, i.e.; Newtonian fluids have constant viscosities. The field of rheology deals with the general case of fluids, plastics, etc. with τ vs. $\dot{\gamma}$ characteristics which are non-linear and/or do not intersect

the origin - non-Newtonian fluids. For the general case, the differential viscosity μ_d is the slope at a given point on the τ vs. $\dot{\gamma}$ curve [2](Figure 1).

The more common form of viscosity discussed in reference to non-Newtonian fluids is the apparent viscosity, μ_a . It is the slope of the line connecting a point $(\dot{\gamma}, \tau)$ on the curve with the origin [2](Figure 1).

Fluids may display three general types of behaviour. They may be A, pseudoplastic (shear-thinning), B, Newtonian, or C, dilatant (shear-thickening). Furthermore, they may have a yield stress (D, E, F). The case of a fluid with a yield value and a constant differential viscosity (E) is a Bingham plastic [2]. The slope of the line connecting the yield point to a point on the line is the plastic viscosity, μ_{pl} (Figure 1).

Some tendencies in rheology help illustrate these concepts. For instance, single-phase (homogeneous) liquids do not have yield stresses [2]. This suggests that in mixtures, it is the presence of another phase that creates yield properties. Chemical (ionic, organic) or physical (state, particle size) differences between phases may play parts in the rheology of a mixture. An ERF, like many mixtures, may have a natural yield value without electrical stimulation.

In fluids with natural yield values, the post-yield behaviour often emulates that of the continuous (liquid) phase [2]. Otherwise, the dispersed (particulate) phase is responsible for yield properties. Where shearing occurs, the

Figure 1 General rheological behaviour: shear stress vs. shear rate for various fluid types (from [2]).

continuous phase 'carries' the dispersed phase in the flow field with little effect upon the general shape of the fluid curve.

High polymers, as melts or moderately concentrated solutions, tend to be pseudoplastic. Concentrated suspensions and emulsions generally have a yield value [2].

1.3 ER Fluids

1.3.1 Definitions

It is generally agreed that an ERF (1) has a yield stress, (2) is composed of solid particles in a base liquid and that (3) its electrical conductivity changes when sheared [3]. Many solutions and mixtures have been tested by various researchers for ER behaviour. Useful ER behaviour has been found in mixtures of dielectric particles in insulator carrier fluids [4,5], for example, starch or polymer particles in silicon oil. Many combinations of carrier fluids, particulates and surfactants exist in the literature [6].

1.3.2 Fibrillation

The popular theory of the mechanism behind the ER effect is that in an ERF under electrical stimulation, fibrillation of the particulate phase manifests itself as yield stress and non-Newtonian behaviour - the ER effect [6]. It has been found that the ER effect is dependent upon a difference in conductivity and permittivity between the two phases of the

fluid [5]. This concurs with the theory of dielectrophoresis [7]. In many mixtures, moisture content is essential to the ER effect, but high moisture content is the major source of temperature sensitivity [8]. It is reported that moisture or mobile ions are needed to get distinct fibrillation; a difference in permittivity alone is insufficient [7]. However, for anhydrous copper phthalocyanine, fibrillation occurs. This suggests that conductivity or semi-conductivity is helpful if not necessary for ER effect, and that it must be a design parameter for ER fluids [7].

The particles in a stimulated ERF cling to their closest neighbours; with gentle agitation, they will form a more definite structure [3]. Using rheo-optical methods, it was determined that the process of fibrillation includes particle alignment, chaining, columnar thickening/chain fracture and finally, columnar fracture [7]. Many fluids only experience chaining and columnar thickening/chain fracture.

Observations of a shear zone at the break point of the fibrils have been made for water-based ERX-02, with fibrils grabbing at passing fibrils [7]. Two common themes seem to prevail: that a structure can reform when shearing stops, and that there is a steady state between destruction and reformation in combined shear and electric fields [7]. It appears that there is an intimate link between the ER effect and the fibrillation process.

The fact that field-induced yield stress extrapolates to

the flow regime, manifesting itself as enhanced apparent viscosity has led to other theories of particle interaction as the cause of the ER effect. The concept of Coulombic interactions between polarized particles in the flow field has been considered [6]. Universal agreement regarding the root cause of the ER effect is yet to be achieved.

1.3.3 ER Fluid Models

ER fluids have been suggested to behave approximately like Bingham plastics, with electric field-dependent yield stresses and rather constant plastic viscosities [3,9,10,11,12,13,14]. However, the Bingham plastic model may not necessarily be accurate near yielding [4]. It is possible that the mechanisms of the ER effect are different, or behave differently in the static and dynamic (flow) regimes. This would make the post-yield region one of transition.

The Bingham plastic model is the simplest rheological model which includes a yield stress. More refined models are complex and difficult to use. Although ER fluids do not behave exactly like Bingham plastics, the Bingham plastic model is a useful concept for guiding experiments and designing devices. The use of this model has been suitable for the comparison of results obtained from different methods and has been extremely helpful in understanding the basic phenomena of the ER effect [3].

Evidence of pseudoplasticity in ERFs has been reported

[6,13]. A general model for pseudoplastic curves has been proposed [2]. The model allows μ_s data to be defined in terms of shear rate $\dot{\gamma}$, viscosities μ_0 and μ_∞ (in which the subscripts reflect the value of $\dot{\gamma}$), with two empirical coefficients and one empirical exponent. Drawbacks regarding the application of this model to ERFs are its complexity and its disclusion of a yield value. In any rheological modelling that involves yield behaviour, it is imperative to work in the τ -domain as opposed to the μ -domain, which can lead to false conclusions regarding $\dot{\gamma}$ -dependence [6].

1.3.4 Yield Shear Stress

The determination of the yield shear stress of ER fluids can be difficult. It is significant to distinguish between static and dynamic yield stress. The best way to illustrate this distinction is as follows: the static yield stress is the value which is interpreted from the τ vs. $\dot{\gamma}$ characteristic, while the dynamic yield stress is that which is found on the τ vs. $\dot{\gamma}$ characteristic.

Research comparing 'shear trolley' and Couette electrorheometer methods for testing static yielding [4] revealed that for the shear trolley, τ_y was proportional to E . The shear trolley apparatus consists of a flat, horizontal plate on wheels and a base which is electrically grounded. Shear is applied to the trolley with a cable, which is strung over a pulley and loaded with weights. The wheeled plate is

the positive electrode and the wheels provide electrical insulation, such that the electric field is applied to the gap between the plate and the base. This research suggested differences in the τ_y vs. E relationship depending upon the method of loading τ and applying E . For the Couette electrorheometer, τ was proportional to E^2 , but the results were inaccurate near the origin. Shock and vibration caused premature failure, and residual (time) effects complicated the measurements. The dynamic shear yield stress, τ_y , was found to be less than the static value. This research was inconclusive regarding the dependence of τ_y on E or E^2 , or whether both apply over different ranges or for different fluids. The conclusion was that one cannot test in static mode and expect the data to be useful in a flow situation.

The dynamic shear yield stress, τ_y , has been defined as the least-squares intercept of τ over the tested range of $\dot{\gamma}$ [3]. Another way to determine τ_y is to measure it experimentally by reducing the shear rate in a rheometer to zero [4]. Time-dependent effects such as rheopexy and thixotropy make yield stress measurement difficult for ER fluids [3,4].

Some research [3] has claimed that the static yield stress, found by increasing the shear stress on an ERF until shearing occurs, corresponds with the dynamic yield stress, if rheopexy is avoided in the static test. A common model regarding the results of different test methods is illustrated

in Figure 2 [4,6]. An ERF under $E=0$ is represented by curve A (it is simplified as a Newtonian fluid, though this is not necessarily true). Curve B is the Bingham plastic model corresponding to the static yield stress. Curve C shows the typical decrease of shear stress associated with post-yield flow, while D is the typical result of unloading the shear rate to zero from a flow condition. Curve E shows a reduction in the field-induced shear stress at higher shear rates.

The relationship between the dynamic yield shear stress (τ_y) and electric field strength is dependent upon temperature and differs among fluids. Some researchers claim that τ_y increases in proportion to E^2 [1,11], but the reported power of E ranges from 1 to 2.4 [4,15].

1.3.5 Shear Rate

The premise of Couette rheometry is that the measured shear stress at the boundary (wall) is the result of uniform, laminar shear flow in the flow field (gap). Such a premise requires the assumption of a linear velocity profile with a "no-slip" boundary condition. In almost all cases, the velocity profile and boundary conditions cannot be observed. The validity of these assumptions, and therefore the accuracy of the rheometric results can be facilitated with proper rheometer design, as explained in sections 2.2 and 2.5.

For results obtained from a Couette rheometer, use of the gross shear rate (relative tangential velocity of rotor to

Figure 2 Electrorheological behaviour: shear stress vs. shear rate (from [6]).

stator divided by width of gap) can produce errors at low shear rates [4], because the assumption of a linear velocity profile may not be valid. Otherwise stated, non-uniform shear rates are possible in such a device, especially in the post-yield regime. Wall slippage is another phenomenon that complicates the shear rate assumptions. Slippage is common in concentrated suspensions, making the no-slip assumption questionable. Phase separation can cause a carrier fluid film at the walls [3]. With an applied field, the columns of fibrils that form tend to attach to one electrode and drag along other - apparently, the strength of these columns can be greater than their adhesion to an electrode. Therefore, electrode surfaces may be significant in rheometer design [7].

Radial effects, such as velocity profile curvature (deviation from a linear velocity profile), pose another complication to shear rate assumptions. These can be alleviated with proper design. Simultaneous observations of sheared and stationary regions in a rheometer gap have been made, and were explained to be the result of curvature in the Couette device [3].

1.3.6 Temperature

Research has raised doubts about the effective temperature range of ER fluids; this is one of the most limiting factors in their practical application. The required temperature range of operation has been cited as -30 to 110°C

[14], though this will depend upon the application, flow rates and heating/cooling.

Many mixtures rely on moisture to create the ER effect. Below 0°C, hydrous ER fluids lose their effectiveness, possibly due to the restriction of charge mobility [6]. At high temperatures (above 70°C), evaporation (water loss or dissolution) may be the limiting factor [6]. High temperature performance is also complicated by high currents and electrical breakdown. Electrolytic decomposition of the water in the mixture has been cited [8] as another possible complication. Overall, moisture control presents a problem in terms of the reliability and repeatability of the ER effect.

The apparent viscosities of ER fluids typically have temperature coefficients which are negative at 0 kV/mm but positive with the application of an electric field, as reported in [13]. τ_y tends to increase significantly with T [16]. However, there is evidence that τ_y actually peaks with respect to T [6,16]. In [15] it is reported that τ_y peaks between 25 and 40°C - this behaviour is due to a complex interaction between moisture and dielectric properties.

1.3.7 Axial Flow Modelling

The existence of a yield value presents complications for flow modelling. In axial flow, fluids with yield properties, such as Bingham plastics, display a phenomenon known as plug flow. The centre of the flow field in any tube or channel,

where the stress is lowest, is occupied by a solid (unsheared) plug. This creates a velocity profile with a flat peak; the maximum velocity exists across the plug (Figure 3). Despite the plug flow velocity profile, the shear stress is linear across the gap, as in laminar flow [2,9,11]. The width of the plug depends upon the yield stress of the fluid and the slope of the shear stress profile - as governed by the magnitude of the shear stress at the wall, τ_w and therefore u . The plug forms in the region where the yield stress is not exceeded.

1.3.8 Apparatus

Researchers in this field typically utilize rotating-cylinder Couette viscometers. Differences in design include rotor and stator diameter and length, gap ratios, methods of driving, etc. To the author's knowledge, there is no evidence in the literature of the use of different surface profiles, although it has been suggested [7]. Another apparatus that has been used is the shear trolley [4], described previously.

A Couette-type concentric cylinder electrorheometer is used to determine rheological properties in this research. This form of rheometer is considered to be useful and versatile. The ER fluids are tested at shear rates between 0 and 3000 s^{-1} , to cover the range foreseeable in vehicle damper flows. Electric field strengths of 0, 1, 2 and 3 kV/mm are employed at temperatures of -13, 0, 15, 30, 40 and 50°C . Different rotor and stator (electrode) surfaces are used to

kR	Inside Radius of Annulus
$\lambda_{+}, \lambda_{-}R$	Limits of Plug Flow Regime
λR	Geometric Mean of λ_{+}, λ_{-}
R	Outside Radius of Annulus
τ_y	Yield Shear Stress

Figure 3 Axial Bingham (plug) flow through an annulus such as in the controllable damper.

compare results for possible phase separation or wall slippage. It is desired to measure fluid shear, and thus a no-slip boundary condition must be facilitated.

Our electrorheometer has certain features to facilitate accurate, repeatable measurement. Large end-gaps reduce end effects. A small radius ratio (i.e.; small gap compared to radius) reduces radial effects. A thermocouple is employed in the fluid close to the gap to ensure consistency with respect to temperature. Filtering of the torque signal reduces mechanical and electrical noise. A flexible coupling reduces mechanical noise and provides electrical insulation from the motor. The torque cell is mounted on PVC adaptors for electrical insulation. Furthermore, the motor is insulated with rubber sheeting on the mounting feet.

A method which may be considered more direct for determining damper performance would employ a variable velocity ram. However, the data obtained with a Couette rheometer describes the fundamental behaviour of the fluid, i.e.; shear stress vs. shear rate. Such data can be applied to the design of any ER damper or device, if the flow in that device is modelled properly. A ram or axial flow rig produces data in a form such as pressure drop vs. flow rate which is unique to that device. Knowledge of the actual fluid behaviour is most directly deduced from a shear mode device.

1.4 Thesis Organization

This thesis presents the behaviour of two ER fluids over a range of temperatures. This research also presents axial (damper) flow modelling, validation with a quarter-vehicle two-degree-of-freedom rig (steady and transient state) and the analysis and design of an ER damper for a full-scale vehicle.

The rheological properties of two ER fluids are determined in the fundamental form of shear stress vs. shear rate with respect to electric field strength and temperature. The two fluids tested are termed ERF 'A' and ERF 'B'. ERF 'A' is a mixture of BDH 'AnalaR' starch and 10 centiStoke silicon oil. A mass ratio of 2:5 starch to oil is used. ERF 'B' is a grease-like proprietary mixture obtained from a U.S.-based company. The methods and apparatus used in this effort are described in chapter II. The rheological properties are modelled such that shear stress can be expressed as a function of shear rate (fluid modelling). These results are presented in chapter III.

To make the acquired rheological data useful, it is modelled in the flow geometry of its intended use. The shear flow in a concentric cylinder Couette electrorheometer is often called "shear mode" by people in the field of ER technology. Carleton University's two-degree-of-freedom quarter vehicle test rig employs axial flow through the piston of an active damper. This application is termed "flow mode."

It is desired to develop a flow model (force vs. velocity) which allows the employment of fluid models (shear stress vs. shear rate) in the design and analysis of any damper with any number of annular gaps. The piston is analyzed as a group of concentric cylinders. The development of flow models is found in chapter IV.

It is also desired to compare the results of shear mode and flow mode designs for similar geometries. A shear mode flow model is also developed in chapter IV. The implications of comparing non-similar flows become apparent. The comparison is of practical use, however, for a designer attempting to achieve a desired mechanical effort with spatial constraints.

The models developed are presented in forms which include the two Bingham plastic parameters (yield stress and plastic viscosity), the damping force and the relative velocity of the piston to the damper shell. Any of these variables can be isolated, such that a damper can be designed based on desired force, a control system can continuously modulate the yield stress based on the required damping force, or fluid requirements can be determined for a potential design.

The axial flow model is ultimately used to design an active ER damper for a full-size vehicle. The model is employed in damper design and the determination of fluid performance requirements. These applications are found in chapter V.

Chapter VI presents the conclusions made as a result of

this research. Recommendations for further study or improvements in technique are also made.

II. METHOD AND APPARATUS

2.1 Introduction

This chapter describes the procedures and equipment used in the acquisition of data for this project.

The optimal design and performance estimation of ER fluid devices require the use of accurate rheological data. Depending on the application, this data may be most useful in the form of shear stress, apparent viscosity or yield stress vs. shear rate, temperature or electric field strength.

2.2 Electrorheometer

Data obtained with a Couette (e.g.; concentric-cylinder) electrorheometer represents the fundamental behaviour of an ER fluid. It can be applied universally to the design of any ER damper or device, if the flow in that device is modeled satisfactorily.

The rheological data was acquired with a Couette-type concentric cylinder electrorheometer which was designed by X.M. Wu (Post-Doctoral Fellow, Transport Technology Research Laboratory, Carleton University), built in-house and modified by the author. It is pictured in Plate 1 and Figure 4.

The aluminum rotors and stators are interchangeable so that the three surface sets (described in section 2.5) can be used in one assembly. The rotors mount on an internal hub and the stators mount inside the shell. The rotor hub is in turn

Plate 1 Electrorheometer without Motor

Figure 4 Schematic diagram of Couette-type electrorheometer.

mounted and keyed to a shaft, which rides on bearings in the end plates. The steel end plates are assembled to the shell, which is plexiglass. This allows viewing of the contents of the rheometer, and electrically isolates the stator from the rotor assembly. The stator is used as the positive electrode. The shaft is driven by a 1 hp reversible variable-speed motor via flexible elastomeric coupling to alleviate the effect of misalignment. The fluid shear subjected to the stator is transmitted to the shell and bottom end plate, which is restrained by a torque load cell.

The L-frame provides a base for the motor and torque cell, which is isolated from the frame and bottom end plate by PVC spacers. The torque cell (Lebow & Assoc., Inc. Reaction Torque Sensor 2110-2k) has a capacity of 2000 in.-lb. The cell has a suitably high natural frequency (ca. 20 kHz), such that errors due to dynamic effects during testing are minimized.

The torque signal was calibrated and amplified with a "Vishay-Ellis 20" conditioner/amplifier. An op-amp circuit provided further amplification. The signal was filtered at a cut-off of 50Hz, which allowed all foreseeable mechanical signals to be recorded while attenuating electrical noise. The signal was recorded on a 386-33 MHz PC via A/D card.

The rotor/stator sets are designed such that the gap ratio r_o/r_i is less than 1.05. This enables us to assume [2] that there are negligible radial effects and that a linear velocity profile is achieved, i.e.; the shear rate is constant

across the gap and equal to the "gross" shear rate, which is calculated by dividing the rotor tangential velocity by the gap width.

The width of the end gaps is very large compared to the gap between the rotor and stator. Furthermore, the greatest fluid velocities and shear rates are developed in the rotor/stator gap. An analysis of fluid motion in the rheometer shows that the shear stress applied to the shell and end plates is negligible compared to that upon the stator. With an applied electric field, the additional shear stress is applied only to the stator. For these reasons, the assumption of negligible end effects is considered to be reasonable.

High voltage was supplied to the rheometer by a Universal Voltronics High Voltage DC Power Supply, with a capacity of 16kV and 10mA.

2.3 Temperature Control

Environmental control was achieved with the use of an environmental chamber - the cold room. This chamber was capable of maintaining temperatures down to -30°C . Since shearing elevated the temperature of the ERF being tested, the environmental temperature was kept colder than the desired test (ERF) temperature so that the ERF could cool down for the next voltage setting. The ER fluid temperature was monitored in the rheometer with a thermocouple and digital thermometer unit with a resolution of 0.1°C . The thermocouple could not be

placed in the gap due to the application of the electric field. Thus there was some difficulty in monitoring the precise gap temperature, where most of the heat was being produced. However, the thermocouple was placed just above the gap to provide the best possible temperature measurement. Tests were performed quickly such that a temperature tolerance of 1°C was maintained.

2.4 Schedule of Tests

One ER fluid tested with one set of electrode surfaces comprised a set of tests. Each set produced six plots; one for each temperature. Between each set of six tests, the rheometer had to be disassembled, drained and checked for bearing condition. A new rotor/stator set and/or new ER fluid was put in and the rheometer was reassembled. The respective radial locations of the end plates, shell and spacers were marked for consistent alignment.

2.5 Calibration

Before each set of tests, the torque cell was calibrated with known weights applied to a hanger (torque arm), which was mounted to the cell, with the other end of the cell fixed. The calibration also included a procedure in which the weights were hung from a point on the centreline of the cell to ensure that lateral (transverse) forces would not affect the torque readings in the rheometer.

The rheometer as a system is designed to have negligible end and radial effects. The other factor to consider is laminar flow. According to [2], the turbulence limitations regarding Couette rheometers are defined by the following relationship:

$$(N_{Re})_{crit} = \{[v_i (r_o - r_i) \rho] / \eta\}_{crit} = 41.3 \sqrt{r_o / (r_o - r_i)} \quad (1)$$

in which values above $(N_{Re})_{crit}$ indicate transitional or turbulent flow.

Using this relationship, a turbulence transition line was calculated and plotted (Figure 5), indicating that anything under it may be prone to turbulence. The data acquired tends to be above or near this line, and it was determined that laminar flow was a reasonable assumption.

Before every measurement, the system was 'zeroed' to ensure proper repeatability. Bearing friction was accounted for by "zeroing out" the bearing friction torque.

The three different rotor and stator surfaces were used to compare results for possible wall slippage, i.e., to ensure that measurements represented ER fluid shear mechanisms (no-slip boundary condition), rather than carrier fluid shear due to phase separation. The rotor/stator sets were: smooth (machined finish), grooved (2.5mm deep), and grousers (raised 0.25mm). These are shown on Plate 2 and in Figures 6, 7 and 8. It was hoped that the different surface profiles would be significant in assessing the interaction of the ER effect with electrode surfaces, such that more insight into design

Figure 5 Shear stress vs. shear rate at various electric field strengths for ERF 'A' and ERF 'B' at 30°C.

Plate 2 Rotors

Figure 6 Smooth rotor and stator.

Figure 7 Grooved rotor and stator.

GROUSERED ROTOR

ELEVATION SECTION VIEW

ADHERE .25 mm DIA.
NYLON STRING, 100 mm
LONG - 12 PLCS.

CUT-AWAY PLAN VIEW

GROUSERED STATOR

ELEVATION SECTION VIEW

ADHERE .25 mm DIA.
NYLON STRING, 100 mm
LONG - 13 PLCS.

CUT-AWAY PLAN VIEW

Figure 8 Grousered rotor and stator.

techniques could be attained. The surface profiles were designed to provide roughness without changing the effective gap width. It was estimated that the circumferential distance between grousers would have to be on the order of particle size to cause an effective gap reduction.

2.6 Method

Shear stress vs. shear rate curves were obtained for the two ER fluids at electric field strengths of 0, 1, 2 and 3 kV/mm. E was controlled manually on the power supply and monitored on the built-in voltage meter. A voltage divider was used periodically to verify the supply voltage between tests. The tests were run at ERF temperatures of -13, 0, 15, 30, 40 and 50°C. Temperature was controlled manually by shearing the fluid until it heated to the proper temperature. Cooling was achieved by maintaining a cooler external environment. Temperature was monitored on the digital thermometer.

For each curve, data was acquired by setting an excitation voltage and taking rheometer torque readings at predetermined motor speeds, corresponding to approximately 50, 125, 250, 500, 1000, 2000 and 3000 s^{-1} . The actual rotor speed (and therefore shear rate) was measured and input to the data acquisition system. The only control exercised over motor speed was to ensure clustering of the data. A data point was established by averaging thousands of torque readings over 2-10 seconds, depending upon the motor speed. This was done to

include several revolutions of the rotor without unnecessarily heating the fluid. At least three such data points were produced for each shear rate, field strength and temperature. After producing a data curve, the excitation voltage and motor were turned off and the ERF was reagitated and allowed to cool. The voltage was then set to the next field strength, and data was recorded point-by-point over the shear rate scale again.

2.7 Quarter Vehicle Rig

A two degree-of-freedom quarter vehicle rig was designed by X.M. Wu, built in-house and modified by M. Stark (Graduate Research Assistant, Transport Technology Research Laboratory, Carleton University) to investigate the possible applications of ER fluids to controllable dampers, to develop control strategies and to perform simulations. The rig is scaled down approximately eight times and consists of a chain-driven eccentric cam, a solid follower, a primary spring (representing the suspension spring), a secondary spring (representing the stiffness of the tire) with the semi-active damper in parallel, and weights for the quarter vehicle (sprung) mass. The sprung and unsprung masses mount on yokes which are fitted with linear ball bearings, and ride on vertical posts. The damper has a quadruple-annulus piston with alternating positive and negative electrodes. High voltage is supplied with the aforementioned Universal Voltronics power

supply.

Instrumentation for the rig includes a B+K 4370 Uni-gain piezoelectric accelerometer for the sprung mass, complete with B+K 2635 charge amplifier and B+K 2511 vibration meter. There is a Duncan Electronics, Inc. 441SL10k Slideline potentiometer (10k Ω capacity) to measure the relative displacement of the secondary suspension. This signal is differentiated via analog means to produce a relative velocity signal. A KISTLER 9212 force transducer (5000-lb. capacity) is mounted in the damper rod for damping force measurement. The force transducer is accompanied by a KISTLER 5014 charge amplifier.

The rig was used in this research to validate the axial flow model (Chapter V). The rig is pictured in Plate 3 and is schematically shown in Figure 9.

Plate 3 Instrumented Quarter-Vehicle Test Rig

Figure 9 Schematic diagram of a quarter vehicle test rig (from [17]).

III. RESULTS

3.1 Introduction

Plots of shear stress vs. shear rate vs. electric field strength were produced for each of the six temperatures and three rotor/stator sets for ERF 'A' and ERF 'B'. Appendices 1 through 33 contain all of the plots of this fundamental rheological data.

3.2 Post-yield Behaviour

In the post-yield region (below 50 s^{-1}), there is a significant drop from the measured value of post-yield stress to values at low shear rates. This is similar to behaviour reported previously [4,6]. It is important to note that the data points in the post-yield region as seen in the Appendices were measured by running the rheometer at a low speed. These stresses appear high, and seem to contradict the trend of the curves. These points were prone to viscoplastic behaviour, with coagulation in the fluid gap. In ERF 'A', there was significant separation of the particulate phase from the carrier fluid with obvious coagulation at the electrodes.

3.3 Low Shear Rates

From 100 to 1000 s^{-1} , an increase in E produces a transition from linear (Bingham plastic) to pseudoplastic behaviour. Pseudoplastic behaviour is characterized at low

shear rates by a relatively large differential viscosity, μ_0 , which is the slope of the τ vs. $\dot{\gamma}$ curve. The slope decreases with increasing shear rate. With further increase in the shear rate, the slope approaches a positive constant value, μ_∞ .

3.4 High Shear Rates

Above 1000 s^{-1} , the curves tend to straighten out and become somewhat parallel, indicating that the viscous shear component is ultimately similar over the range of electric field strengths.

3.5 Rheology with Zero Electric Field Strength

Figure 10 is a plot of τ_0 vs. $\dot{\gamma}$ vs. T (zero electric field). Note that shear stress decreases as temperature increases. This is typical for liquids, but not all fluids; gases become more viscous with increasing temperature. In the case of ERF 'B', substantial shear stress is produced with no electric field, especially at low temperatures. This figure can be used as a reference for figures in this report which present the shear ratio r_γ , or the field-induced shear stress τ_{ER} against $\dot{\gamma}$, E or T . The shear ratio is the ratio of the total shear stress with an electric field applied to the zero-field shear stress at the same shear rate. It is an indicator of the range of mechanical control that can be expected of an ER fluid device. τ_{ER} is the difference between the total and zero-field shear stress at the same shear rate. These two

Figure 14 Shear ratio vs. electric field strength at various shear rates for ERF 'B' at 40°C.

variables are defined as follows:

$$\begin{aligned} r_r &= \frac{\tau}{\tau_o} \\ \tau_{ER} &= \tau - \tau_o \end{aligned} \quad (2)$$

3.6 Field-induced Shear Stress

Figure 11 shows that for ERF 'B', the field-induced shear stress τ_{ER} decreases from 250 to 2800 s^{-1} . ERF 'A' experiences a peak in τ_{ER} , which occurs at greater shear rates as E is increased. The points making up the τ_{ER} vs. $\dot{\gamma}$ curves in Figure 11 were calculated by subtracting τ_o from the total shear stress at each value of E and $\dot{\gamma}$.

3.7 Shear Ratio

From the data acquired, the dependence of ER fluid shear stress upon various factors can be analyzed. For instance, a plot of r_r vs. T vs. $\dot{\gamma}$ is presented for a field strength of 3 kV/mm (Figure 12). The shear ratio increases with temperature because τ_o decreases (Figure 10) and τ_{ER} increases. This increase is most pronounced at low shear rates, which are more conducive to fibrillation. Similar results are reported in [18]. The dependence of r_r upon temperature is much greater for ERF 'A' than ERF 'B'. There is generally little ER effect below 0°C in either fluid tested.

Figures 13 and 14 (r_r vs. E vs. $\dot{\gamma}$) show that r_r increases with E at all shear rates. This increase is also most pronounced at lower shear rates. Figures 13 and 14 reveal the

Figure 11 Electric field-induced shear stress vs. shear rate at various electric field strengths for ERF 'A' and ERF 'B' at 30°C.

Figure 12 Shear ratio vs. temperature at various shear rates for ERF 'A' and ERF 'B' at 3 kV/mm.

extent of mechanical control available in a device employing these fluids.

Figures 15, 16 and 17 reveal the nature of the shear ratio r_s against shear rate at various temperatures. Obviously, the greatest range would be realized by designing the device for low shear rates (low flow velocity), other factors permitting.

3.8 Rotor/Stator (Electrode) Surfaces

The results of tests with the three rotor/stator sets were compared for differences in shear stress under the same conditions of $\dot{\gamma}$, E and T . The surface conditions appear to have no substantial effect upon the shear stress. This either suggests that wall slippage is being avoided and thorough fluid shear is occurring, or that the extent of wall slippage is similar for the three surfaces. The former is more likely, due to the significant roughness profiles of the grooved and grousured electrode sets.

3.9 Yield Shear Stress

Dynamic yield stress, τ_y , has been defined as the least-squares intercept of shear stress over the range of tested shear rates [3], i.e.; extrapolation of the shear stress vs. shear rate curve to zero shear rate. Using this definition, Table I displays τ_y as a function of E for the six temperatures tested.

Figure 13 Shear ratio vs. electric field strength at various shear rates for ERF 'A' at 40°C.

Figure 14 Shear ratio vs. electric field strength at various shear rates for ERF 'B' at 40°C.

Figure 15 Shear ratio vs. shear rate for ERF 'A' and ERF 'B' at 0°C and 3 kV/mm.

Figure 16 Shear ratio vs. shear rate at various electric field strengths for ERF 'A' and ERF 'B' at 30°C.

Figure 17 Shear ratio vs. shear rate at various electric field strengths for ERF 'A' and ERF 'B' at 50°C.

Table I - Yield Shear Stress as a Function of Electric Field Strength at Various Temperatures

T	ERF 'A'
-13°C	$\tau_y = 3.19 E + 2.22 \text{ [Pa]}$
0°C	$7.3 E + 6.8$
15°C	$6.20 E^{1.43} + 9.9$
30°C	$17.4 E^{1.61} + 8.9$
40°C	$20.6 E^{1.65} + 14.0$
50°C	$26.0 E^{1.71} + 6.0$
=====	
	ERF 'B'
-13°C	$\tau_y = -3.17E + 197$
0°C	$-0.10E + 161$
15°C	$27.6 E + 160$
30°C	$26.2 E^{1.71} + 124.9$
40°C	$20.9 E^{1.67} + 115.8$
50°C	$16.8 E^{1.62} + 113.3$
=====	

Note in Table I that for ERF 'A', the power of E increases as temperature increases. For ERF 'B', the power of E decreases slightly with increasing T. This is possibly due to a peak of ER behaviour at 30°C or the counteractive effects of the natural ($E=0$) yield stress, which decreases with increasing temperature.

3.10 Settling

Recalling the phase separation noted in section 3.2, it was also observed that ERF 'A' showed a pronounced tendency to settle out of suspension. Agitation alone was insufficient to restore the fluid to a suspended state after lengthy settling. ERF 'B' tended to separate only slightly; a small amount of base oil would collect on top of a sample after sitting several days.

3.11 Fluid Modelling

The change of character of ER fluids with increasing electric field strength poses difficulties in terms of fluid modelling, i.e.; quantifying the relationship of τ vs. $\dot{\gamma}$. This is so because the very nature of the shear stress as a function of shear rate changes with the controlling variable, E. As seen in Figure 5, there is a change from linearity to pseudoplasticity with increasing field strength, characterized by a "hump" in the shear stress at low shear rates. The curves do not fit a "family" of functions. A practical approach is to

fit a simple model to the rheological data; one that is reasonably consistent and representative of the ER fluid behaviour. Simplicity and generality are important in capturing the essence of ER fluid behaviour, considering the variety of ER fluids under development. For instance, compare ERF 'B', which resists settling and has the consistency of grease, to ERF 'A', which is quite fluid, settles rapidly and is more pseudoplastic under the influence of an electric field.

The Bingham plastic model is fitted to the τ vs. $\dot{\gamma}$ curves. A Bingham plastic has a yield value (τ -axis intercept, concurring with the definition in section 3.9) and a constant plastic viscosity (slope), and is defined mathematically as;

$$\tau = \tau_y + \mu_{pl} \dot{\gamma} \quad (3)$$

No flow occurs in such a material until τ exceeds the dynamic yield stress τ_y . Beyond yielding, the shear stress is proportional to the flow velocity gradient (shear rate) $\dot{\gamma}$ by a coefficient μ_{pl} , called the plastic viscosity, which in this case is equal to the differential viscosity.

The results of this modelling produce a yield stress range of 2-139 Pa and plastic viscosities ranging from 34-98 mPa-s for ERF 'A'. ERF 'B' has a yield stress range of 113-289 Pa and a plastic viscosity range of 155-243 mPa-s. Like the yield stress, the plastic viscosity can be expressed in terms of electric field strength and temperature. Table II lists plastic viscosity as a function of E at various temperatures.

The plastic viscosities of both fluids decrease with increasing T , but decrease only slightly with increasing E (note small coefficients of E in Table II). This dependence is so slight that for practical purposes, the terms in E can be neglected, reducing the functions to a dependence upon T only.

By plotting the coefficients, exponents and constants in Tables I and II against temperature and by applying least-squares curve-fitting, a single equation for each ERF fluid can be produced. The results of this are, for ERF 'A';

$$\tau = \frac{[(.553 T - 1.11) E^{(.00785 T + 1.335)} - .0508 T + 11.42]}{[-.000666 T + .0840] \dot{\gamma}} \quad (4)$$

and for ERF 'B';

$$\tau = \frac{[(-.47 T + 40.1) E^{(-.0045 T + 1.847)} - .58 T + 141.2]}{[-.00122 T + .213] \dot{\gamma}} \quad (5)$$

In both equations, τ is in Pa, T is in $^{\circ}\text{C}$, E is in kV/mm and $\dot{\gamma}$ is in s^{-1} .

Our observations reveal deviations from Bingham plastic behaviour, particularly in the yield and low shear rate region. Figure 5 exemplifies this. In comparison to a Bingham plastic line fitted to the $10\text{--}3000 \text{ s}^{-1}$ data at 3 kV/mm, the measured data is high at yielding, dips low from $100\text{--}250 \text{ s}^{-1}$, then rises and converges on the Bingham line. For high field strengths (3 kV/mm), where pseudoplastic behaviour is obvious, a power curve with a positive exponent less than 1 (i.e.; a concave-down curve) corresponds very closely with the measured data beyond the post-yield region. However, this type of curve

Table II - Plastic Viscosity as a Function of Electric Field Strength at Various Temperatures

T	ERF 'A'
-13°C	$\mu_{pl} = -.000140 \text{ E} + .0979 \text{ [Pa} \cdot \text{s]}$
0°C	$-.00206 \text{ E} + .0824$
15°C	$-.00267 \text{ E} + .0705$
30°C	$.00232 \text{ E} + .0579$
40°C	$-.00326 \text{ E} + .0571$
50°C	$-.0110 \text{ E} + .0570$
=====	
	ERF 'B'
-13°C	$\mu_{pl} = .0022 \text{ E} + .236$
0°C	$-.0014 \text{ E} + .211$
15°C	$-.00357 \text{ E} + .179$
30°C	$-.00652 \text{ E} + .184$
40°C	$.00127 \text{ E} + .163$
50°C	$.00210 \text{ E} + .154$
=====	

does not provide an accurate representation of the yield properties of ER fluids, especially considering the high shear stresses in the post-yield region.

Indeed, many researchers have published models or data claiming that their subject ER fluid exhibits Bingham plastic behaviour (see Chapter I), with an electric field-induced yield stress, and a plastic viscosity which is either independent of E , or which decreases slightly with increasing E .

In [4], it is reported that "field-induced snatching of the restrained outer cylinder" occurred at low rotational speeds. This has been described as slip-stick behaviour. Similar behaviour was observed in the current research, and it was often necessary to run the rheometer at shear rates above 30 s^{-1} to avoid this behaviour and its inherent irrepeatability. Even so, the data presented can only represent the gross shear rate, with no clear indication of the velocity profile in the fluid gap. This made the yield stress a difficult value to extrapolate.

The report in [4] also describes the difficulty of defining the yield stress. In that research, the static yield stress determined by shearing an energized ER gap produced a concave-up post-yield curve, while reduction of the shear rate to zero produced a concave-down curve. Similar observations were also made in the current research.

IV. FLOW MODELLING

4.1 Introduction

The purpose of this chapter is to provide force vs. relative velocity information for a designer employing ER fluids in a vehicle damper. Since this is extremely dependent upon damper geometry, a multiple-annulus flow mode design similar to that used in the aforementioned quarter-vehicle rig is chosen as a model (Figure 18). A mathematical model is produced, based on equations for flow through an annulus.

Electrorheological (ER) fluids have been shown to behave like Bingham plastics, with an electric field-dependent yield stress and a roughly constant plastic viscosity. To make electrorheological fluid data (i.e.; τ vs. $\dot{\gamma}$) useful, it has to be modelled in the flow geometry in which it will be used. In this study, the fluid data was obtained in a concentric cylinder Couette electrorheometer. This type of flow is often called "shear mode" by people in the field of ER technology. The quarter-vehicle test rig in this study employs axial flow through the piston of an active damper. This application is termed "flow mode."

It is desired to develop a flow model which would allow us to employ rheological data in the design and analysis of any multiple-annulus damper. The piston is analyzed as a group of concentric cylinders, creating axial flow through multiple annuli.

Inside Diameters of Annuli

1st Annulus:	33 mm
2nd Annulus:	45 mm
3rd Annulus:	57 mm
4th Annulus:	69 mm

Figure 18 Diagram of the multiple-annulus ER damper used in the quarter vehicle rig (from [17]).

kR	Inside Radius of Annulus
λ_+R	Limits of Plug Flow Regime
λ	Geometric Mean of λ_+ and λ_-
R	Outside Radius of Annulus
τ_y	Yield Shear Stress

Figure 19 Axial Bingham (plug) flow through an annulus such as in the ER damper (from [20 & 21]).

Fluids with yield properties, such as Bingham plastics, display a phenomenon known as plug flow; the centre of the flow field in any tube or channel, where the stress is lowest, is occupied by a solid plug. This creates a velocity profile with a flat peak; the maximum velocity exists across the plug. The width of the plug depends on the yield stress of the fluid and the shear stress profile. The plug forms in the region where the yield stress is not exceeded.

It is also desired to compare the results of shear mode and flow mode designs for similar geometries. A shear mode flow model is developed. The implications of comparing different flows become apparent, in light of velocity profile and shear rate at the device walls. The comparison is of practical use, however, for one attempting to achieve a desired mechanical effort with spatial constraints.

The models developed are presented in forms which include the two Bingham plastic parameters (yield shear stress and plastic viscosity), the damping force and the relative velocity of the piston to the damper shell. Any of these variables can be isolated, such that a damper can be designed based on desired force, a control system can continuously modulate the damping force based on the required yield stress, or fluid requirements can be determined based on damper performance requirements.

In a practical application, all of the aforementioned parameter dependencies can be accommodated in a control sub-

routine, however, the only significantly controllable parameter is the yield stress.

4.2 Development

This section describes the development of flow models for ER fluids in multiple-annulus dampers.

Consider an axial flow damper with a relative velocity v_{rel} across it. Using the multiple-annulus piston as a control volume, the damper will displace fluid at a volumetric flow rate;

$$Q = v_{rel} A = \frac{\pi}{4} D^2 v_{rel} \quad (6)$$

regardless of "blockage" area. Assuming negligible extraneous Coulomb friction, the damping force is proportional to the pressure drop across the piston;

$$F = \Delta P A = \Delta P \frac{\pi}{4} D^2 \quad (7)$$

which is entirely manifested as shear stress along the walls of the piston annular gaps.

Simplifying the piston geometry, it is analyzed as a group of concentric cylindrical shells. The length, L , remains the same.

Now the plug and laminar flow between concentric cylinders can be modelled, for application to any multiple-annulus damper employing a Bingham plastic or a Newtonian fluid [20,21,22]. Referring to Figure 19, the following

equation describes the behaviour of a Bingham plastic;

$$\tau_{rz} = \pm \tau_y - \mu_{pl} \frac{du_z}{dr} \quad (8)$$

in which r is the radius. The minus sign applies to a radial location less than λR and the positive sign applies to a radial location greater than λR , which is the geometric centre of the annulus. Figure 19 shows that $\lambda_1 R$ and $\lambda_2 R$ represent the inner and outer boundaries of the plug. Radial location to either side of the plug will create a positive or negative velocity gradient (du/dr). Equation 8 states that the shear stress in excess of the yield stress equals the product of the shear rate and the plastic viscosity.

Assuming steady flow, the axial force due to axial pressure difference (ΔP) balances with the shear force, which resists the flow. For an infinitesimal annular fluid element (L by Δr), the limit of the force balance equation [20] as dimension Δr approaches 0 produces;

$$\frac{d}{dr} (r \tau_{rz}) = \frac{\Delta P}{L} r \quad (9)$$

Introducing dimensionless parameters:

$$S = \frac{2 \tau_{rz} L}{R \Delta P} \quad (10)$$

$$S_y = \frac{2\tau_y L}{R\Delta P} \quad (11)$$

$$x = \frac{r}{R} \quad (12)$$

$$v = \frac{2\mu_{pl} L}{R^2 \Delta P} u_z \quad (13)$$

Using the boundary conditions $v(k)=0$, $v(1)=0$ and integrating Eq.9 over r ;

$$\Delta\tau_{rz} = \frac{\Delta P}{2L} \left[r - \frac{\lambda_- \lambda_+ R^2}{r} \right] \quad (14)$$

Common factoring R ;

$$\tau_{rz} = \frac{\Delta PR}{2L} \left[\frac{r}{R} - \frac{\lambda_- \lambda_+ R}{r} \right] \quad (15)$$

Likewise;

$$\pm\tau_y = \frac{\Delta PR}{2L} \left[\frac{\lambda_{\pm}}{R} - \frac{\lambda_- \lambda_+ R}{\lambda_{\pm}} \right] \quad (16)$$

Substituting (15) and (16) into (8);

$$\frac{du_z}{dr} = \frac{\Delta PR}{2\mu_{pl} L} \left[\frac{\lambda_{\pm}}{R} - \frac{\lambda_- \lambda_+ R}{\lambda_{\pm}} - \frac{r}{R} + \frac{\lambda_- \lambda_+ R}{r} \right] \quad (17)$$

Common factoring R , non-dimensionalization and integration yield the following;

$$v = \pm S_y x - \frac{x^2}{2} + \lambda_- \lambda_+ \ln(\sqrt{R}) + C \quad (18)$$

Boundary conditions: $v = 0$ at $x = k$;

$$\therefore v_{inner} = \lambda_+ \lambda_- \ln\left(\frac{x}{k}\right) - S_y (x - k) - \frac{1}{2} (x^2 - k^2) \quad (19)$$

Likewise;

$$v_{outer} = v(\lambda_+) + S_y (x - \lambda_+) - \frac{1}{2} (x^2 - \lambda_+^2) + \lambda_- \lambda_+ \ln\left(\frac{x}{\lambda_+}\right) \quad (20)$$

Also known:

$$\begin{aligned} v(\lambda_-) &= v(\lambda_+) \\ S_y &= \lambda_+ - \lambda_- \\ v_{inner}(k) &= v_{outer}(1) = 0 \end{aligned} \quad (21)$$

Using Eq.19, 20 and 21, it can be shown that;

$$(1 - k^2) = 2S_y \left[1 + k - \frac{1}{2} (\lambda_- + \lambda_+)\right] + 2\lambda_+ \lambda_- \ln\left(\frac{\lambda_-}{\lambda_+ k}\right) \quad (22)$$

The plug region can have a width from zero to the full annular gap. Thus, S_y is constrained as follows;

$$0 \leq S_y \leq 1-k \quad (23)$$

For $S_y = 0$, $\lambda_- = \lambda_+ = \lambda$. Substitution into Eq.22 reveals the value of λ for any value of S_y , keeping in mind that λ is the geometric mean of λ_+ and λ_- ;

$$\lambda = \left[\frac{(1-k^2)}{2 \ln(\frac{1}{k})} \right]^{\frac{1}{2}} \quad (24)$$

Integration of the velocity profile reveals that;

$$\frac{F}{V_{rel}} = \frac{8A^2 \mu_{pl} L}{\pi \Sigma \left[(1-k_i^4) - 2\lambda_i^2 (1-k_i^2) - \frac{4}{3} (1+k_i^3) S_{yi} + \frac{1}{3} (4\lambda_i^2 + S_{yi}^2)^{\frac{3}{2}} S_{yi} \right] R_i^4} \quad (25)$$

The dilemma at this point is that Eq.25 is implicit. S_y is different for each annulus and it cannot be solved directly.

Presumably, dP/L is identical for all annuli, as is τ_y , which is strictly a material property dependent upon E . Thus, based on Eq.11, S_y varies from annulus to annulus in proportion to $1/R_i$. With this in mind, an approach to the solution of F/V_{rel} may be to substitute Eq.7 and Eq.11 into

Eq.25, producing an equation in P and τ_y . For given values of τ_y and v_{rel} , P and thus F could be solved numerically.

For the purpose of validating this model against data acquired in this project, it is worthy to note that reasonable approximations can be made to Eq.25. Noting that $k > .84$ for all annuli in our piston and recalling the constraints on S_y from Eq.23, S_y^2 is negligible. The last term in the denominator of Eq.25 can thus be approximated by;

$$\frac{8}{3} \lambda_i^3 S_{yi} \quad (26)$$

The approximation for Eq.25 becomes;

$$\frac{F}{v_{rel}} = \frac{8A^2 \mu_{pl} L}{\pi \sum [(1-k_i^4) - 2\lambda_i^2(1-k_i^2) - \frac{4}{3}(1+k_i^3 - 2\lambda_i^3) S_{yi}] R_i^4} \quad (27)$$

By substituting Eq.11 and separating the summation term, τ_y can be isolated;

$$F = \frac{8AL}{\pi} \frac{A \mu_{pl} v_{rel} + \frac{\pi}{3} \tau_y \sum (1+k_i^3 - 2\lambda_i^3) R_i^3}{\sum [(1-k_i^4) - 2\lambda_i^2(1-k_i^2)] R_i^4} \quad (28)$$

This model will be validated against acquired data in Chapter V.

The following shear flow model is developed in a manner that makes the damper geometry consistent with the axial flow

model. A rotary shear mode damper can be designed with a shear area that is equal to that of the flow mode ER damper. However, to achieve a direct comparison and to avoid the complexity of taking into account the effect of the linkage between the rotary damper and the wheel, this development employs a hypothetical linear shear mode damper that has a constant shear area equal to that of the flow mode damper.

Assuming that pure shear flow is created, i.e., that the open annular area comprises almost all of the piston area A , the shear rate developed in such a damper is simply;

$$\frac{du}{dr} = \frac{V_{rel}}{R(1-k)} \quad (29)$$

Here Eq.8 is repeated in the form pertinent to this flow, describing the behaviour of Bingham plastics;

$$\tau = \mu_{pl} \left(\frac{du}{dr} \right) + \tau_y \quad (30)$$

The damping force generated is the product of the shear stress and the shear area;

$$F = \tau L 2\pi \Sigma R_i \quad (31)$$

Substitution of Eq.30 into Eq.31 gives;

$$F = \left[\frac{V_{rel}}{R(1-k)} \mu_{pl} + \tau_y \right] L 2\pi \Sigma R_i \quad (32)$$

It must be emphasized that this model is simplified, and is only useful for the theoretical comparison of design approaches, or "modes."

reaching its steady (locked up) state. Data measured from the rig indicate a response that resembles hysteresis - this result increases with E . The nature of these curves makes it very difficult to verify the accuracy of the model.

The F vs. v_{rel} curves produced by the model mimic the idealized (Bingham) shear stress vs. shear rate characteristics. Deviations from Bingham plastic behaviour, such as pseudoplasticity, would take on the same shape in the F vs. v_{rel} model.

The axial flow model is reasonably accurate and predictive of damping force, especially considering that the flow through the piston is unsteady, reversing and possibly not uniform. An increase in electric field strength produces a larger yield force, which appears as a step when plotted over the entire (+/-) velocity range. This step is reflected in the distorted curves of medium field strengths, such as Figures 21 and 22. This behaviour is analogous to a damper with combined Coulomb and viscous damping. Coulomb damping force is independent of relative velocity, as is the yield force.

The ostensibly hysteretic behaviour seen in Figures 21 and 22 seems to be due to a dissipation of energy in the damper. Using the $+F, +v_{rel}$ quadrant in Figure 21 as an example, the "top" portion of the curve is formed in the first half of the compression stroke of the damper, with relative velocity and damping force increasing. As the damper passes through

stress dependence on electric field and a low zero-field yield stress. It seems therefore that a starch/mineral oil mixture such as ERF 'A' would be a better candidate than a naturally thick fluid, such as ERF 'B' (disregarding other problems such as settling, propensity for re-agitation, etc.). Indeed, this agrees with the findings in this study that ERF 'A' has greater shear ratios (total shear stress over zero-field shear stress) than ERF 'B'.

5.3 Comparison of Model to Test Rig Data

Values for the geometry of the damper in the quarter-vehicle test rig were substituted into the axial flow model. Yield stress and plastic viscosity data from ERF 'A' at 30°C were employed for consistency with the rig test conditions. Damping force vs. relative velocity data from the model were plotted against measurements from the rig (Figures 20, 21 and 22), with a steady-state sinusoidal "road" input and fixed DC excitation voltage. The relative displacement signal was differentiated by analog means for the relative velocity measurements. The instantaneous damping force was measured with a load cell in the damper rod.

A combination of system response and measurement error cause the damping force and relative velocity signals to be approximately sinusoidal, with some noise and distortion (Figure 23). Possible sources of deviation include spring nonlinearity, signal offsets, imperfect differentiation of the

Figure 20 Force vs. relative velocity obtained using the axial flow model and quarter vehicle rig, with ERF 'A' at 30°C and $E = 0$ kV/mm.

Figure 21 Force vs. relative velocity obtained using the axial flow model and quarter vehicle rig, with ERF 'A' at 30°C and $E = 1$ kV/mm.

Figure 22 Force vs. relative velocity obtained using the axial flow model and quarter vehicle rig, with ERF 'A' at 30°C and $E = 2$ kV/mm.

Figure 23 Force and relative velocity signals from the quarter vehicle rig test, plotted vs. time (ms) at $E = 1 \text{ kV/mm}$.

relative displacement signal, Coulomb damping and system nonlinearity. The measured damping force vs. relative velocity curves are distorted due to the aforementioned errors. Figure 20 shows the data produced at 0 kV/mm. The model predicts greater damping forces than those measured. This discrepancy could be the result of turbulence in the rheometer tests at 0 kV/mm (see Figure 5). The predicted slope of the F vs. v_{rel} curve (the damping coefficient) is a function of plastic viscosity, which is the slope of the best-fit Bingham plastic line on the τ vs. $\dot{\gamma}$ curve. Turbulence in the rheometer tests would manifest itself as a higher plastic viscosity and therefore a greater damping coefficient.

Figure 21 shows that at 1 kV/mm, the curve is distorted about the origin, indicating the effect of yield properties. The model agrees with the laboratory data. Peculiar behaviour was observed at this voltage. The relative velocity did not appear to change smoothly like a sinusoid. It tended to jerk at zero velocity, locking up momentarily, then releasing. Lock-up is a condition in which the damper yield force is greater than the force exerted on the damper, allowing no relative motion between the piston and the shell, producing a suspension transmissibility of 1. Predictably, as seen in Figure 21, the force exerted on the damper had to exceed the yield force before relative velocity developed.

Figure 22 shows that at higher field strengths such as 2 kV/mm, the F vs. v_{rel} behaviour gets rather complex before

reaching its steady (locked up) state. Data measured from the rig indicate a response that resembles hysteresis - this result increases with E . The nature of these curves makes it very difficult to verify the accuracy of the model.

The F vs. v_{rel} curves produced by the model mimic the idealized (Bingham) shear stress vs. shear rate characteristics. Deviations from Bingham plastic behaviour, such as pseudoplasticity, would take on the same shape in the F vs. v_{rel} model.

The axial flow model is reasonably accurate and predictive of damping force, especially considering that the flow through the piston is unsteady, reversing and possibly not uniform. An increase in electric field strength produces a larger yield force, which appears as a step when plotted over the entire (+/-) velocity range. This step is reflected in the distorted curves of medium field strengths, such as Figures 21 and 22. This behaviour is analogous to a damper with combined Coulomb and viscous damping. Coulomb damping force is independent of relative velocity, as is the yield force.

The ostensibly hysteretic behaviour seen in Figures 21 and 22 seems to be due to a dissipation of energy in the damper. Using the $+F, +v_{rel}$ quadrant in Figure 21 as an example, the "top" portion of the curve is formed in the first half of the compression stroke of the damper, with relative velocity and damping force increasing. As the damper passes through

mid-stroke and continues compressing (still $+F, +v_{rel}$), it slows down - both F and v_{rel} decrease - but it does not trace the curve of the first half of the compression stroke. Having dissipated system energy, it finishes that portion of the stroke with less dissipation (the "bottom" force curve). The same behaviour repeats itself in the $-F, -v_{rel}$ quadrant, for the extension stroke.

It must be emphasized that ER damping is not limited to this behaviour. The tests plotted in Figures 21 and 22 were performed with fixed electric field strengths to validate the flow model. It has been shown [19] that with an appropriate control strategy, the electric field strength can be modulated according to v_{rel} and relative displacement to control vehicle suspension transmissibility. Electric field control enables full use of the τ vs. $\dot{\gamma}$ (and thus F vs. v_{rel}) capabilities of the ER fluid. The entire region between the "top" curve ($E = 3$ kV/mm) and the "bottom" curve ($E = 0$) can be utilized, for instance, to make a smooth transition through $v_{rel} = 0$ without locking up.

Above 2 kV/mm, the damper readily locks up, because the yield force is greater than the force exerted upon the damper by the test rig. The yield force depends upon damper geometry and the yield stress of the fluid, and the applied forces depend upon the state of the system. Figures 20, 21 and 22 were produced with the test rig in a near-resonant state.

5.4 Comparison of Shear and Flow Modes

The shear mode is compared analytically to the flow mode. The force generated in shear mode is much smaller than that in flow mode, as is seen by comparing Figure 24 to Figures 20, 21 and 22. Figure 25 shows that the control ratios are higher in shear mode. The control ratios plotted are of damper force at 3 kV/mm to that at 0 kV/mm. As expected, the shear rates at the gap walls are much higher in the flow mode, due to the shape of the velocity profile and the excess flow caused by blockage area. This leads to a greater dominance of the viscous term in the flow mode, and therefore lower control ratios.

From a fluid dynamics point of view, it is not pertinent to compare the shear and flow modes at face value (force, control ratio). Flows are usually compared in the context of non-dimensional velocity profiles. The shear and flow modes analyzed in this paper are extremely different in that respect. Possibly the only common ground on which these flows may be compared is upon the shear rate at the gap wall. Such an analysis reveals that for the same geometry, the shear rates are much lower in the shear mode.

An attempt was made to achieve the same forces in shear mode as in flow mode, by manipulating the geometry of the shear model. The first iteration with individual variables produced impossible designs. The practical limits of a compromise between a decrease in gap size (from 3 to 1mm) and

2 OF 2

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Figure 24 Force vs. relative velocity at various electric field strengths for the shear mode damper model, using ERF 'A' at 30°C.

Figure 25 Control ratio vs. relative velocity for the axial flow mode and shear mode models, using ERF 'A' at 30°C and $E = 3$ kV/mm.

an increase in gap radii (without increasing D) were made. The yield force achievable in shear mode was 25% of that in flow mode. However, the viscous force was only 4% of that in flow mode. Thus, even at the limit of practical design, the shear mode barely performs within an order of magnitude of the flow mode, in terms of damping force magnitude. This view is supported in [4], which states that only positive displacement ER devices can achieve the pressure level and power density required of typical hydraulic machines.

5.5 Preliminary Design Exercise

The flow mode was thus chosen for an example of a preliminary design for a full-size damper. Performance requirements were suggested by Coulter and Duclos [10]; maximum and minimum forces of 1000 N and 100 N respectively at a relative velocity of 0.5 m/s. Realistic spatial constraints were estimated at 0.075m for L and 0.090m for D , based on full-size vehicle rattlespace and spring diameter.

These values were substituted into Eq.28. Trials based on ERF 'A' suggested that with annular gaps of 1.25mm, a maximum force of 1014 N would be attainable, but the minimum force would be 785 N.

Based on this failure, the model was used to determine the fluid properties required to meet the damper performance requirements. Based on 3mm gaps, and plastic viscosities similar to ERF 'A', a fluid with a yield shear stress range of

76-1693 Pa would suffice. It was noticed in this run, however, that S_y only varied from .026 to .057; 39% of its allowable range. When adjusting gap size, S_y was found to be an important parameter to monitor. As mentioned in Chapter IV, S_y is constrained to 0,1-k. One goal of a robust design should be to utilize this range fully - this reflects maximum control range.

A few more trials were run. Results using 2.5mm gaps suggested a yield stress range of 2-1349 Pa would satisfy the damper performance requirements, while utilizing 66% of the S_y range.

With so many variables, such trials can be quite iterative. The significant point to be made is that the model is versatile and can be used to choose damper geometry, determine required ER fluid properties or assess the usefulness of an ER fluid.

A fluid developed in the U.K. [23] has a yield stress range of 0-4000 Pa. The Newtonian viscosity of the fluid is 350 mPa-s, and the plastic viscosity decreases with field strength, such that it is negative above 2 kV/mm.

These values were used in Eq.28. The solution revealed that 4mm gaps would produce a minimum damping force of 149 N and a maximum force of 1670 N. Although these results do not satisfy the performance requirements suggested by Coulter and Duclos, they provide the author with enough information to confidently claim that this ER fluid can be used to satisfy

the requirements. It is important to keep in mind that the model presented in this paper is based on a piston with four equally charged annular gaps. An optimized design, however, may involve any number of gaps, not necessarily the same size and not necessarily all electrically charged. The various possibilities of this damper concept will have to be the subject of further research or development.

VI. CONCLUSIONS AND RECOMMENDATIONS

6.1 Rheological Properties

The shear stress created in an ER flow field increases with electric field strength. This increase is attributable to an electric field-dependent yield stress, which increases by a factor of $E^{1.43}$ to $E^{1.71}$ for ERF 'A' and a factor between $E^{1.62}$ and $E^{1.71}$ for ERF 'B' over the range of temperatures tested.

Temperature generally increases the shear ratio, which is defined as the ratio of total shear stress with an electric field applied to that without electrical excitation. The total shear stress does not necessarily increase with temperature, because the plastic viscosity of an ER fluid decreases with increased temperature. The lower limit of useful ER behaviour appears to be 15°C for the fluids tested in this study.

An electrical field increases the yield stress of an ER fluid, while the post-yield behaviour is roughly similar for all field strengths, that is, the ER contribution to the total shear stress decreases only slightly over the range of shear rates tested. These traits lend themselves to a Bingham plastic model.

The Bingham plastic is the simplest rheological model which incorporates an electric field-dependent yield stress value [3]. It is the most widely used model for representing and analyzing ER fluid behaviour. There is a pseudoplastic model [2] which would accommodate the reported post-yield

deviations from Bingham plasticity. However, its complexity and lack of a yield term outweigh its potential accuracy.

6.2 Flow Modelling

The presence of a yield stress poses difficulties in flow measurements or modelling in which the shear rate is not uniform. The flow field may include "solid" material in areas of low shear stress, while the remainder of the field exhibits laminar flow. In axial flow, this effect manifests itself as plug flow in the centre of the flow field.

With the rheological data acquired and the fluid modelling presented in this thesis, flow models for a multiple-annulus ER damper are developed and presented. The models are, by necessity, highly dependent upon the geometry of the flow field and hence the device. These models are essential to the design and analysis of ER fluid devices.

Although it affords higher control ratios, the shear mode of operation is essentially impractical for ground vehicle dampers, due to the low achievable damping forces.

For the fluids that have been tested in this study, the Bingham plastic model is suitable for application to multiple-annulus damper flows. The axial flow model provides engineers with the means to design and analyze ER dampers.

6.3 Feasibility of Current Technology

The ER fluids used in this study cannot meet both

requirements of damping force magnitude and control range in a full-size vehicle damper. However, there are fluids with high yield stress ranges which satisfy these requirements. If ER fluids with high yield stress ranges and low zero-field viscosities prove useful with respect to temperature range, abrasion, life span, cost and reliability, the developers of ER fluids will hold the key to a very promising technology.

6.4 Recommendations for Improved Apparatus

The type of research presented in this thesis can be performed with a smaller rheometer and drive. The resulting apparatus would use less ER fluid, be easier to clean and set up between test sets and be more portable for use in the environmental chamber.

Much like the motor, the torque load cell used was far larger than required for this research. This was done for economic reasons, such that the motor and cell could be used in other research projects. A smaller rheometer and cell would facilitate more precise measurement.

The ability to gear down the drive would be beneficial. The high shear stress at low shear rates (and thus low rheometer speeds) proved to be difficult in terms of drive stability (speed variations) and speed measurement. Gearing (or belting) would allow the drive motor to operate at a more stable speed and torque while taking measurements at extremely low shear rates. This would allow a better investigation into

yield and post-yield behaviour.

Along with a speed reducer, improvements to the configuration of the rheometer should include a bearing layout that would ensure concentricity between the drive and torque cell. The current configuration has bearings mounted in the rheometer end caps, allowing no means to align the motor and torque cell. Any eccentricity in the set-up produces a wobbling motion in the rheometer, with transverse loads being transmitted to the torque cell. Great care was taken to measure the effects of such loads in calibration and to achieve proper alignment, but these were time-consuming.

Another source of error from one test set to the other was in the bearing friction. The lower bearing was prone to penetration, particularly with ERF 'A'. More precise bearings with better seals should be employed.

Finally, another possible source of error was moisture variation. Although the environmental chamber typically had a steady humidity at steady state, entering the chamber between tests and warming the chamber from test to test (to accommodate temperature range) seemed to create changes in moisture and conductivity. Perhaps the ER fluids themselves have moisture affinities that change with temperature.

6.5 Recommendations for Future Research

In this research, the ER fluid particulate concentration was not investigated. However, informal tests seemed to

indicate that some ER fluids may have a 'saturation' concentration, beyond which the ER effect does not increase. An investigation that includes concentration as a variable should yield interesting results.

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APPENDICES

A1-A33: Shear Stress vs. Shear Rate Plots

The fundamental rheological behaviour of a fluid is described by shear stress as a function of shear rate.

This section contains all of the rheological data acquired in this study. It is presented in graphic form. The relationships reported in this paper were determined from this data.

Due to the quantity of data, it is presented in 33 graphs. Each graph reflects the data collected for one ER fluid, using one type of rotor/stator surface, at one or two designated temperatures, depending upon visual clarity. Each graph presents the data collected over the full number of designated electric field strengths.

Figure A1 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using smooth rotor and stator at -13°C.

Figure A2 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using smooth rotor and stator at 0°C.

Figure A3 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using smooth rotor and stator at 15°C.

Figure A4 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using smooth rotor and stator at 30°C.

Figure A5 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using smooth rotor and stator at 40°C.

Figure A6 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using smooth rotor and stator at 50°C.

Figure A7 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grooved rotor and stator at -13°C.

Figure A8 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grooved rotor and stator at 0°C.

Figure A9 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grooved rotor and stator at 15°C.

Figure A10 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grooved rotor and stator at 30°C.

Figure A11 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grooved rotor and stator at 40°C.

Figure A12 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grooved rotor and stator at 50°C.

Figure A13 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grousers rotor and stator at -13°C.

Figure A14 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grousured rotor and stator at 0°C.

Figure A15 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grousered rotor and stator at 15°C.

Figure A16 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grousered rotor and stator at 30°C.

Figure A17 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grousured rotor and stator at 40°C.

Figure A18 Shear stress vs. shear rate at various electric field strengths for ERF 'A' using grousured rotor and stator at 50°C.

Figure A19 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using smooth rotor and stator at -13°C and 0°C.

Figure A20 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using smooth rotor and stator at 15°C.

Figure A21 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using smooth rotor and stator at 30°C.

Figure A22 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using smooth rotor and stator at 40°C.

Figure A23 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using smooth rotor and stator at 50°C.

Figure A24 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grooved rotor and stator at -13°C and 0°C.

Figure A25 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grooved rotor and stator at 15°C.

Figure A26 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grooved rotor and stator at 30°C.

Figure A27 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grooved rotor and stator at 40°C.

Figure A28 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grooved rotor and stator at 50°C.

Figure A29 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grousered rotor and stator at -13°C and 0°C.

Figure A30 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grousered rotor and stator at 15°C.

Figure A31 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grousured rotor and stator at 30°C.

Figure A32 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grousured rotor and stator at 40°C.

Figure A33 Shear stress vs. shear rate at various electric field strengths for ERF 'B' using grousured rotor and stator at 50°C.

END

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