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MAGNETO-RHEOLOGICAL FLUIDS AND THEIR USE IN DAMPERS TO LOWER VIBRATION EFFECTS

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ABSTRACT

Smart fluids and especially magneto-rheological (MR) fluids are growing in importance over the past few years for the important role that they play in the development of what is called, semi-active controlled devices. Such devices provide the advantages of both active and passive controlled devices, as they offer the adaptability of active control devices without requiring large power sources [3]. This is critical in structural applications where damping of vibrations resulting from earthquakes, tornadoes and other potential disasters is needed. In these cases providing a large power supply is not only very difficult but also extremely dangerous. The purpose of this work is to give an overview of the properties and structures of MR fluids and their usages in dampers for a wide variety of applications. Moreover a generalized mathematical model that can clearly describe the MR fluid behavior while the damper is in operation was obtained. This was done by incorporating the simple Bingham model into an axisymetrical model. The equations resulting from such a mathematical model were generalized using dimensionless analysis and then solved numerically using the finite difference method and the Gauss Seidel iterative technique. The resulting program, written using Matlab, is used to generate the characteristic curves of the MR damper which are needed to give an accurate simulation of how it will respond in the practical operating field. In addition, a study of the non-Newtonian behavior of the MR fluid which results in the plug flow region in the damper gap area during its operation was performed.

KEYWORDS:

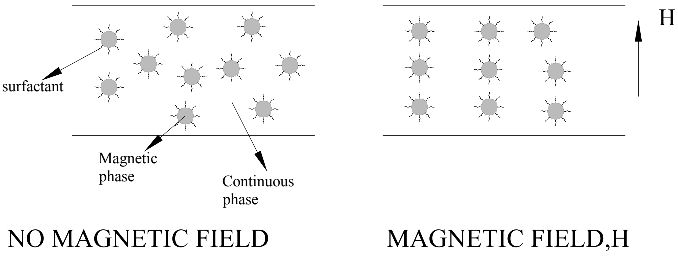
MR fluids, MR dampers, Axisymetrical model, Bingham model, Non-Newtonian behavior, Plug flow region.

INTRODUCTION

Rheology is the branch of science that deals with the flow and deformation of materials. Moreover Magneto-rheology is the branch of Rheology that deals with the flow and deformation of the materials under an applied magnetic field. MR fluid is composed of organic oil or aqueous liquid and varying percentages of iron particles. When inactivated the MR fluid behaves as ordinary oil [4]. When exposed to a magnetic field, micron-size iron particles that are dispersed throughout the fluid align themselves along magnetic flux lines. This reordering of iron particles can be visualized as a large number of microscopic spherical beads that are threaded onto a very thin string [1].

Many different ceramic metals and alloys can be used to prepare MR fluids as long as particles exhibit a low level of magnetic coercivity. Particle size, shape, density, particle size distribution, saturation magnetization and magnetic field are important characteristics of the magnetically active dispersed phase. In addition to the magnetic particles and the base fluids, anticorrosion additives are important factors affecting the rheological properties, stability and redispersibility of the MR fluid [14].

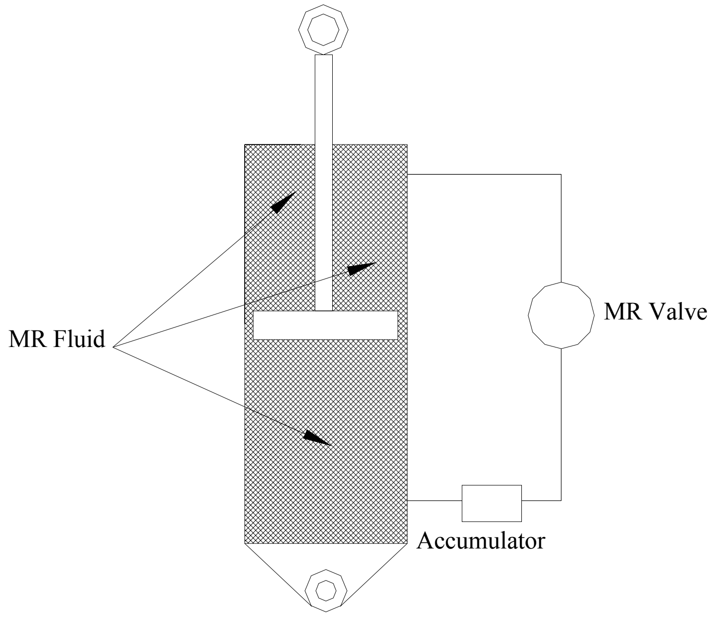
In terms of viscosity analysis, MR fluids appear similar to ordinary liquids and exhibit comparable levels of viscosity (0.1 to 1Pas at low shear rates) in the ‘off’ state. The fluids apparent viscosity changes significantly (10^5 – 10^6 times) within a few milliseconds when the magnetic field is applied. This change in viscosity is completely reversible when the magnetic field is removed and this is clearly shown on figure (1). That property is the most important one on applying the use of MR fluids in dampers. Once a magnetic field is applied, it induces a dipole in each of the magnetic particles. The life time of an MR fluid is defined by the application in which it is used. MR fluids are engineered and performed to meet the life requirements of the application, market or industry in which it is implemented or used. Life for a particular application is affected by the duty cycle or the amount of work (energy dissipated) by the device. For example, the damper product for truck seating applications is designed to handle millions of cycles over five years or more.



“Figure (1) Schematic of the formation of chain-like structures of magnetic particles in MR fluids in the direction of an applied magnetic field”

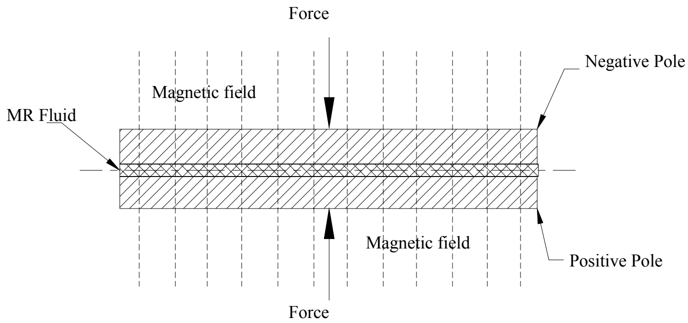
MR fluids are becoming increasingly important in applications concerning active control of vibrations or torque transfer. Shock absorbers, vibration dampers, seismic vibration dampers, clutches and seals are the most exciting applications of MR fluid. For these applications, rheological properties of the MR fluid, working mode of the device, design of the magnetic circuit and flux guide of coil configuration are crucial parameters for the operation of the actuators and devices [19].

As mentioned earlier MR fluids consist of iron particles suspended in a carrier fluid. A wide variety of carrier fluids such as silicon oil, kerosene and synthetic oil can be used for MR fluids. Such a fluid must be chosen carefully to accommodate the high temperature to which it can encounter during operation. The carrier fluid must also be compatible with the specific application without suffering irreversible and unwanted property changes. The ferromagnetic particles are often carbonyl particles, since they are relatively inexpensive. Other particles, such as iron-cobalt or iron-nickel alloys, have been used to achieve higher yield stresses from the fluid. But the high cost of the cobalt or nickel alloys makes the fluids containing these alloys impractical for most applications due. The MR fluid must also contain additives to prevent the sedimentation and promote the dispersion of the ferromagnetic (iron) particles. The fluid that is transferred from above the piston to below (and vice versa) must pass through the MR valve which is a fixed-size orifice upon which a magnetic field can be applied using an electromagnet to the orifice volume. This magnetic field results in a change in viscosity of the MR fluid, causing a pressure differential for the flow of fluid in the orifice volume. The pressure differential is directly proportional to the force required to move the damper rod. As such, the damping characteristic of the MR damper is a function of the electrical current flowing into the electromagnet. This allows the damping of an MR damper to be easily controlled in real time. As for the accumulator, it is a pressurized volume of gas that is physically separated from the MR fluid by a floating piston or bladder. The accumulator serves two purposes. The first is to provide a volume for the MR fluid to occupy when the shaft is inserted into the damper cylinder [14]. The second is to provide a pressure offset so that the pressure in the low pressure side of the MR valve does not induce cavitations in the MR fluid by reducing the pressure below the vapor pressure of the MR fluid. All of the external components have been incorporated internally, providing a compact design that is very similar in size and shape to existing passive vehicle dampers. The only external parts are the two electrical leads for the electromagnet, which are connected to the current source. A functional representation of the MR dampers is shown on figure 2.



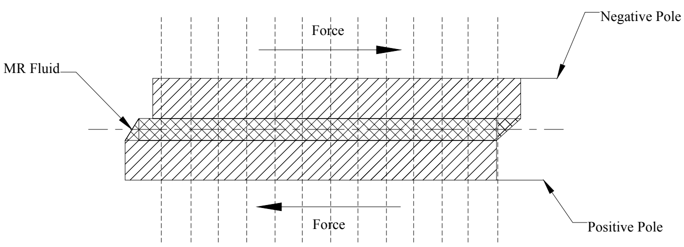
“Figure (2) a functional representation of an MR damper [19]”

The modes of operation of MR dampers and usages are different. MR fluid can be used in three different ways, all of which can be applied to MR damper design depending on the damper’s intended use. These modes of operation are referred to as squeeze mode, valve mode and shear mode. A device that uses squeeze mode has a thin film (on the order of 0.5 mm) of MR fluid that is sandwiched between paramagnetic pole surfaces as shown in figure 3.



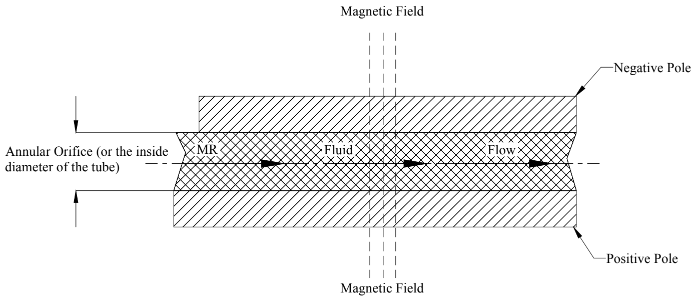
“Figure (3) MR fluid used in squeeze mode”

An MR fluid device is said to operate in shear mode when a thin layer (≈0.1 to 0.3 mm) of MR fluid is sandwiched between paramagnetic moving surfaces. This mode is clearly demonstrated in figure 4.



‘Figure (4) MR fluid used in shear mode”

The last and mostly important mode of MR damper operation is the valve mode, which is clearly demonstrated in figure 5. It is the most widely used of the three modes. An MR device is said to operate in valve mode when the MR fluid is capable of flowing from one reservoir to another by itself.



“Figure (5) MR fluid used in valve mode”

The biggest challenge of MR fluid is to have high temperature stability and durability. The second biggest challenge of MR fluids is the materials science oriented studies such as chemistry, polymer physics, in synthesizing stable and re-dispersible MR fluids. In addition more studies are required to be conducted on MR fluids exposure to high and low temperatures and high shearing stresses [14].

LITERATURE REVIEW

Hu et al[1] studied the quasi-static and dynamic behavior of a magneto-rheological damper by implementing a rate dependent elastoslide (RDES) model that established the damping mechanism for the magneto-rheological(MR) damper using the Bingham model. The used model was characterized by being time dependent or able to describe the unsteady behavior of the MR fluid used. Proposed model parameters and damper mechanism relations were considered and relations between them were set. Moreover, a careful choice of a numerical method that correctly showed the model response under various loadings was made. The nonlinear amplitude-dependent behavior of the MR damper was described using six constant parameters in the model. Results prove that predicted forced response correlated very well with the single and dual frequency experimental results which captures the nonlinearity of the MR damper using a simple model structures. Finally the RDES model showed to be physically motivated in the time domain and it can be used in a variety of applications such as initial or forced- response analysis of MR damper systems and a wide range of proposed control strategies dealing with the MR damper.

Dogruer et al [2] illustrated the fact that vibrations exceeding certain limits can cause poor ride quality and stability resulting in a rollover or severe damage to vehicle elements and passengers. They added that a variable damper in the semi-active system can provide a controllable damping force and active system performance without requiring high power consumption and with very low design variations from conventional passive systems. Hence they presented a full damper design that can be used for such a purpose. Theoretical simulation and experimental results were presented and shown to be close to each other. They also proposed a Bingham plastic fluid model for the used magneto-rheological fluid (MRF) damper. Their results showed that a high dynamic force range is obtained while keeping the damper size(i.e. geometry) nearly the same. In addition, power requirements of proposed damper are significantly less than previous dampers. Moreover, the theoretical results based on the fluid mechanics and electromagnetic finite element analyses are seen to be in good agreement with the experimental results obtained.

Xiaojie et al [3] proposed a study, based upon the Herschel-bulkley constitutive equation, to develop a theoretical model for predicting the behavior of magneto-rheological (MR) and electro-rheological (ER) dampers while being used in the field of operation. During their study they managed to extend the constitutive equation to include the compressibility effects that mainly accounts for the nonlinearity in the dynamic behavior of MR/ER fluid dampers. A steady one dimensional laminar flow was assumed during the analysis where a closed form solution for the pressure drop, especially in the case of the MR damper, was obtained. The main advantage of this model is that it only depends on geometric and material properties of the MR/ER material and the device that encloses them. It was clearly demonstrated that the proposed model can clearly resemble the response made by both types of dampers during their operation. This was validated by comparing theoretical results with experimentally obtained data.

Olabi et al [4] gave a brief introduction concerning the growing usages of magneto-rheological fluid (MRF) applications such as in the automotive and the construction industries. Advantages, such as simplicity and intelligent functionality were used to account for this growth. An enhancement to the fact that MRF technology can be used for any system that desires controllable motion by changing the fluid's viscosity was made. They basically presented an actuator that depends upon the MRF technology for its control operation. For such a presentation a description of the rheological basis of MRF was illustrated together with the MRF mode of operation and MRF damper components. The study clarified the excellent characteristics obtained by using MRF technology. These include fast response, simple interface between electrical power input requirements and mechanical power output and finally controllability.

Milecki et al [5] described a magneto-rheological fluid shock absorber that is used mainly in vehicle applications and in fully automated production lines in which different elements with different energy will be moved at the same time and where their kinetic energy is known. Theoretical relationships, which allow for the design and modeling of MR fluid shock absorbers, were shown. Their investigations concerning the usage of MRF fluids in shock absorbers proved to improve braking characteristics. This was done by a comparison that has been made with shock absorbers without control, and the characteristics obtained when using force control has shown a significant improvement in the MRF case. The braking system of a controlled shock absorber proved that it can be quiet easily adapted to the requirements by a simple electronic circuit.

Zhang et al [6] strengthened on the fact that magneto-rheological fluids yield stresses might not be strong enough to meet some industrial applications' requirements. Their work aimed to investigate such a feature of magneto-rheological fluids through experimental and theoretical approaches. This is done by examining the squeeze- strengthen effect of MR fluids experimentally by proposing an apparatus specifically designed for such an aim and theoretically by implementing a semi-empirical model that is used to simulate such an effect. The comparison between theoretical and experimental data showed that the proposed model can accurately model the squeeze strengthens effect of the magneto-rheological fluid. This paper was intended to give an additional guidance to develop high-efficiency testing devices especially for the case of measuring high yield stresses of magneto-rheological fluids. Moreover, the proposed theoretical model showed the ability to be used for the explanation of the squeeze-strengthen effect of Electro-rheological fluids as well.

Sims et al [7] investigated the performance of semi-active vehicle suspensions, which are based upon smart fluid dampers, by using numerical analysis. Advantages of linearising the force – velocity response are established after utilizing a smart damper in an open loop system. Such analysis was carried out by implementing electro- rheological (ER) damper models. In general, the study verified that the usage of smart dampers is not suited for vehicle applications when implemented in an open loop condition. However, a controllable viscous damper results from applying force feedback control. These results were demonstrated to hold for both sinusoidal and non sinusoidal excitations. The authors emphasized on the importance of using feedback control strategy especially for non-sinusoidal excitations systems to hold under the on/off control of a smart fluid damper.

Breese et al [8] presented the development and evaluation of field-controllable, semi-active fail-safe magneto-rheological fluid (MRF) damper for mountain bicycle. It is to be noted that a fail-safe damper is referred to as a damper that retains a minimum required capacity in the event of a power supply or electronic system failures. They designed two MRF dampers and tested them for the front and rear suspension of a mountain bicycle. The two dampers were considered to be fail-safe and light weight. Theoretical and experimental studies were conducted to evaluate the performance and dynamic force range of these dampers under road input conditions. The first damper, performed similar to the original equipment manufacturer (OEM) shock absorber or damper in its passive off mode (i.e. no electric input current). It also showed a significant increase in the force generated, with increase in the applied electric current. The damper was shown to meet and exceed the performance of the OEM damper which was the original aim of their study. For the second damper, they demonstrated that it produced a slightly lower force than the OEM damper in the passive off mode case. This was because this damper used a slightly softer coil over spring when placed on a bicycle. On activation by an electric current substantial gain in damping forces were generated. Again this damper was seen to meet and exceed the abilities of an OEM damper. However, all experiments were conducted while the dampers were cold, providing experimental results neglecting temperature effects which could be vital during road trips especially long ones, affecting the sensitivity and reliability of the results to a great extent.

Giua et al [9] showed a procedure for the design of a semi-active suspension, to be applied, to road vehicles with MR-dampers. As an available alternative to a purely active suspension system, the use of a semi-active system is considered to overcome the high cost and large power requirements of any active control system. A two-phase design technique for MR semi-active suspensions was presented. The first phase of the work consisted of the design of a target control law which was used to describe the active system and this was achieved. In the second phase, this target law is approximated by controlling the damper coefficient of the semi-active suspension in particular in the time delay required for the updating of coefficient. They assumed that the new value of the coefficient was chosen to minimize the difference between the target and the semi-active control law at the following time .But this was not satisfied at all chosen values. The non linear behavior of both the damper and the spring was taken into account to approximate the target active control law. In addition, the dynamic model of the suspension system was chosen to have only two degrees of freedom and this result in misleading data.

Yang et al [10] made an overview of the essential features and advantages of MR materials and devices. A short and very brief comparison between active and passive control devices together with considerable attention to the increased usefulness of semi-active device especially MR fluid dampers was given. It was shown that the MR devices overcome many of the expenses and technical difficulties associated with semi-active devices. Two main types of MR fluid devices were fully described, one was the full-scale seismic MR fluid damper and the second was the Quasi-static analysis of MR fluid dampers. In addition, a quasi-static axi-symmetric model was derived based on the Navier-Strokes equation of motion to predict the dampers force-velocity behavior. And in certain cases, an approximation was made for a substantially simpler parallel-plate model, which was shown to be useful to investigate the damper's behavior. These models comparison yielded that an error of less than 0.5%, might be found in certain cases. Furthermore, a basic geometry design consideration was made based on the parallel plate model for simplicity but this led to insensitivity of results in some conditions. And this is shown especially when trying to describe the dynamic behavior of MR dampers using quasi-static models. On testing the dynamic performance considerations of MR dampers, a current driver has been shown to be effective in reducing the response time of MR dampers. Experimental results showed that a parallel connection of the damper coil results in faster response time than a series connection. Only 0.2 s was needed to achieve 158 kN force and 0.1s needed to reduce the damper force from 159 kN to 8 kN. However, it was seen that more than a second is required to drop the force from 159 kN to off-state force 8 kN for both connections. The reason for this phenomenon is due to the residual magnetic field remaining inside the damper after the removal of the current . To overcome this problem, the coils were back-driven at the maximum allowable value of 6A (in parallel) until damper force reaches off-state value. And it was only then when the 0.1s is considered an enough time for the off-state force of 8 kN to be reached. This of course affected the simplicity of the proposed design.

Calderón et al [11] studied theoretically the dynamics of a system of two magnetically active particles suspended in a non-magnetic fluid subjected to a rotating magnetic field, when a modulation on the Mason number Ma (i.e. the ratio of viscous to magnetic forces) is applied. They found that there are two different dynamic regimes depending on the value of Mason number in the system. At low Mason number, a rigid doublet rotates phase locked to the field, while at high Mason numbers an asynchronous rotation of the pair of magnetically active particles and an additional radical oscillations appear. They observed a resonant like phenomenon between the periodic modulation of Manson number and the intrinsic radial oscillation of the system without the occurrence of modulation. Evidence of stochastic resonance (SR) in MR suspensions under rotating magnetic fields, when a weak periodic modulation and a noise source are added to the Mason number, was also presented in their work.

Oh et al [12] showed presumably that the vibrations of a truss structure can be suppressed by an (MR) fluid damper, which belonged to a class of smart materials. They compared their results with previous results obtained for similar (ER) electro-rheological dampers. A variable MR fluid damper was designed and fabricated for this study. The principal characteristics of an (MR) damper were then measured in dynamic tests in order to experimentally demonstrate the effectiveness of the semi-active vibration suppression using MR damper. They also proposed a mathematical model to simulate the MR damper. On comparison between proposed model and experimental results, they found that the predicted results are in agreement with the measured data for all input magnetic fields. They carried out an additional vibration suppression experiment, in order to investigate whether the MR damper is actually effective for semi-active vibration system and they were able to show that the performance of MR damper is even better than that of ER dampers especially when vibration is suppressed to a small level.

Barroso et al [13] presented a quasi-controller for a non-linear structure and displayed its efficiency in simulation against suites of several earthquakes of near and far field origin and intensity. The control law applied employed two-stage logic to the decentralized control law. The structure is actuated semi-actively using MR dampers when it is moving away from its vibration equilibrium position and acted as a passive damper when moving towards it. In other words, the MR damper is powered to apply a restoring force, whenever the structure is moving away from its equilibrium position and un-powered to act as a passive viscous damper, when structure is moving towards its equilibrium. Therefore, it actively resisted outward motion and damped inward motion. They discussed importance of tracking permanent deflections in controlling non-linear structures. However, on using the performance matrices they found that while peak values provide a good performance indication, the resulting information is incomplete, as it did not take into account the cumulative damage to the structure. Their experimental results showed that the structural damage was a function of both peak as well as cumulative values. In addition, when they used a linear structural model, similar to those used in previous semi-active controller investigations, the equilibrium point was clearly found to be the original building position as assumed in their work. However, the permanent deformations that resulted from strong motions led to an equilibrium position that changed with time. And they found that only through the usage of very large actuator forces that it may be possible to re-yield the structure back to its original position, resulting in a considerable improvement in the performance by decreasing residual drift matrices.

Kelso [15] in his presentation of MR damper technology illustrated the development of a fast response, low power and mostly important cost or commercially effective solution to active damping. His work included a characterization of a new prototype MR damper involving the description of the device technology and characterization of the test results. The aim of this technology was to address a wide range of research applications and potentially diverse damping environments. In addition the design challenges were to find a simple, cost effective package that holds commercial development appeal. The design included adjustable zero field damping which describes the passive damping in the absence of an electric field. Design components such as commercially available seals are used for design with an optimized electromagnetic circuit. The design can also be easily disassembled and reassembled for fluid on MR gap height changes. The design also assures that the majority of wear resulting from piston-cylinder reciprocation occurs at the seal and is easily and inexpensively replaced. In addition, all seals, O-rings and guide bearings are commercially available in commonly found size. Furthermore, design challenges exist in creating a DC electromagnetic circuit that can generate sufficient flux across the MR-gap in a minimum amount of time. His device design enhances the fact that the response time, required to change the apparent viscosity of MR fluid, is far less than the time a mechanical system requires to vary orifice size. As indicated in his work the force versus displacement relationship proves that proposed MR damper provides consistent controllability of damping throughout the current range. The electro-magnetic circuit within the piston is capable of a time response that supports high band width, semi active control. The coil configuration that delivers a desirable time response, zero-field damping adjustability works for a broad range of damping needs, and the dynamic force range due to the MR effect is diverse and linearly proportional throughout its range.

Godaninejad et al [16] presented experimental and theoretical studies on heat generation and dissipation of field-controllable magneto-rheological (MR) fluid shock absorbers. They considered three different MR fluid dampers and carried out experiments to study the effect of temperature on each. Each damper represented a specific ground-based vehicle application, such as bicycle, automotive and heavy vehicle applications. Experiments were performed on a vibration table using specific testing conditions and data were collected and analyzed using data acquisition equipment. Data such as, temperature, force, velocity and position were acquired from the prototype dampers. The temperature increase of an MR fluid damper subjected to a sinusoidal input motion, and different power inputs to the electromagnet were recorded. The model was then numerically solved and then enhanced to account for MR fluid damper large temperature changes. A comparison was made between theoretical and experimentally determined data and this comparison validated satisfactorily their proposed model, as a useful prediction tool. Isolation of different input and output parameters was possible using non-dimensional terms. These non-dimensional parameters showed that the temperature increase was fairly insensitive to changes in the heat transfer coefficient at the damper surface, but it was sensitive to the work increases caused by the electromagnet, and a nearly linear relation was found. The effect of damping force decrease due to temperature increase was noticed and recorded for the three different dampers. They demonstrated that for large forces applied to the MR fluid damper the temperature change is significantly high, even for a short period of time. However, all tests and results were based on velocity analysis and feedback, although it is known that accurate measurement of displacements and velocities are difficult to achieve directly in full scale applications. Therefore, it is worth noting in this respect that, since accelerometers can readily provide reliable and inexpensive measurement of accelerations at arbitrary points, as pointed by many papers, more accurate results would have been obtained if the development method was based on acceleration feedback rather than velocity feedback.

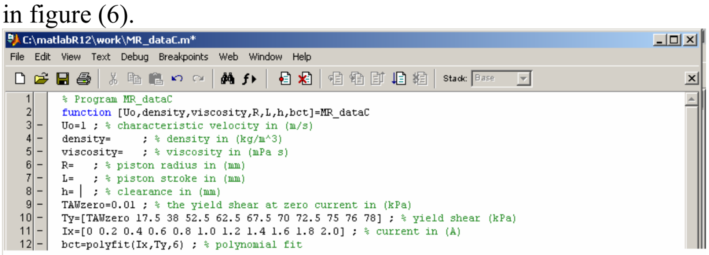
Spencer et al [17] showed the importance of smart base isolation for protecting structures against severe seismic events. Results of an experimental study of a particular adaptable, or smart, base isolation system that employs the MR dampers were shown. A passive base isolation system employing the MR damper subjected to a constant current was experimentally examined as the baseline to evaluate the effectiveness of the Smart base isolation system. Results show that in the passive mode, the MR damper behaved as a yielding device and approximates the behavior of lead rubber bearings. An optimization was performed experimentally to obtain the optimal passive damper configuration. As compared to this optimal hysteretic passive system, the smart isolation system achieved significant acceleration reductions over the entire range of earthquake intensities considered indicating that the smart damping isolation system can be effective over a wide range of ground motion intensities and characteristics. However, he carried out experiments and analytical work on a two degree of freedom model of the structure employed for design control. They assumed a linear behavior to take place for both the structure and the isolation bearings. These assumptions, however are not accurate enough and are approximated, knowing that a smart damping device such as MR dampers have highly non linear characteristics.

Spencer et al [18] examined the efficiency of various models used for the prediction of the response of full-scale MR dampers. The prediction ability of both axi-symmetric model and a simpler approximated parallel plate model is studied. A brief comparison between the active and passive control systems, together with special attention given to MR damper advantages when used in a semi-active control system. It was shown that the MR damper gathers the advantages of both the active and passive control systems which were summarized as their low power requirements, high force capacitance, mechanical simplicity and very fast response. A comparison between several kinds of control systems was presented. The simple parallel plate model was shown to be satisfactory accurate for approximating the force-velocity behavior in the axi-symmetric flow field and to provide an important tool for design of MR dampers. However, the resemblance between the measured and predicted force-velocity relationship is shown to be inadequate at small currents.

Spencer Et al [19] presented a good and brief introduction to the semi-active devices and their importance. A detailed explanation of MR damper mechanical models that are most widely used to describe and resemble its performance was shown. In addition to proposing an improved mechanical model for the MR damper, they were to overcome some drawbacks and inadequacy of the other models, especially in the cases were the velocity and acceleration have opposite signs and the magnitude of the velocities were very small. Furthermore, their proposed model was not numerically stiff and was numerically tractable which enabled them to improve this model numerically by having the lowest errors with respect to time over all other models. It was suggested that the proposed model can portray the behavior of a typical MR damper while in the field of operation.

THEORETICAL ANALYSIS

The MR fluid chosen for the application of our theoretical model is the MRX-145-2BD. This type of MR fluid is mainly used to work inside the large scale MR dampers for its large yield stress created upon the input of small current intensities to the damper. However any other type of MR fluid can be used by the program user, under the condition that he or she knows and inputs all the fluid required data into the specified file. Data such as viscosity, density and the yield stress are all needed for the specification of the MR fluid being utilized. An example of such data used in the program is shown in figure (6).


C:\matlabR12\work\MR_dataC.m*

File Edit View Text Debug Breakpoints Web Window Help

```
% Program MR_dataC
function [Uo,density,viscosity,R,L,h,bct]=MR_dataC
Uo=1 ; % characteristic velocity in (m/s)
density=     ; % density in (kg/m^3)
viscosity=   ; % viscosity in (mPa s)
R=   ; % piston radius in (mm)
L=   ; % piston stroke in (mm)
h=   ; % clearance in (mm)
TAWzero=0.01 ; % the yield shear at zero current in (kPa)
Ty=[TAWzero 17.5 38 52.5 62.5 67.5 70 72.5 75 76 78] ; % yield shear (kPa)
Ix=[0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0] ; % current in (A)
bct=polyfit(Ix,Ty,6) ; % polynomial fit
```

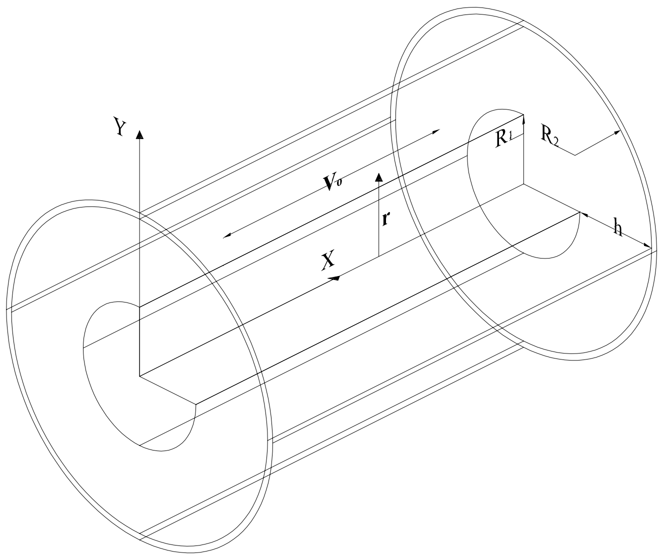
“Figure (6) the function file opened by the user to input the MR damper configuration and MR fluid data”

A dynamic model, the parallel plate model, based on the Navier-Stokes equation is developed to predict and analyze the MR damper behavior. The Bingham model is employed to describe the MR fluid field dependent characteristics. Unlike previous quasi-static models where the flow field in the damper is assumed to be constant and steady. The present model deals with the unsteady flow nature of the damper especially those subjected to sinusoidal excitation.

Concerning the damper design, a typical MR damper consists of an outer cylinder that houses the damper's piston, the magnetic circuit, an accumulator and a chosen volume of MR fluid. During the motion of MR damper piston, the MR fluid flows in the annular gap between the piston and the cylinder. For the dynamic analysis of the damper we assumed that the MR damper has an initial input in the form a sinusoidal exciting force signal. This is transferred to a pressure force generating a pressure gradient. The velocity field should be solved to find the velocity distribution and hence the force-velocity relationship has to be obtained. Figure7 describes the MR damper design geometry and variables that will be used for the analysis. R_i represents the piston radius, y is the variable in the radial direction normal to the solid boundary, and r is the radial variable measured from the center line of the piston. And that annular gap is represented by h on the figure. The flow is assumed to be fully developed, incompressible, isoviscous and laminar. According to the previous assumptions the flow is in the damper is governed by the momentum equation,

$$\rho \frac{\partial}{\partial t} u\,(r) - \frac{\partial}{\partial r} \tau_{rx}(r) - \frac{\tau_{rx}(r)}{r} = -\frac{\partial p}{\partial x} \qquad 1$$

where $u\,(r)$ represents the flow velocity, $\tau_{rx}(r)$ is the shear stress, “*r” is* the radial coordinate, “*x”* is the longitudinal coordinate and $\dfrac{\partial p}{\partial x}$ the pressure gradient.



“Figure (7) Schematic diagram of the MR damper showing the chosen Coordinate system”

A simple Bingham model was used to describe the essential field dependent fluid characteristic, which is the shear stress. The relationship between the stress and the rate of strain was then taken as

$$\tau = \tau_o \,\mathrm{sgn}(\partial u/\partial r) + \mu(\partial u/\partial r)\,,\ |\,\tau\,| > |\,\tau_o|$$

$$\text{and } \dot{\gamma} = \partial u\,(r)/\partial r = 0\ ,|\,\tau\,| < |\,\tau_o|\,. \qquad 2$$

Neglecting the curvature effect of the damper surfaces and assuming that $u(y,t) = U_o \Phi(y) e^{int} = U_o\, \phi\, e^{int} + i U_o\, \theta\, e^{int}$ and $\partial p / \partial x = P e^{int}$ 3

where, *"int"* represents an integer value.

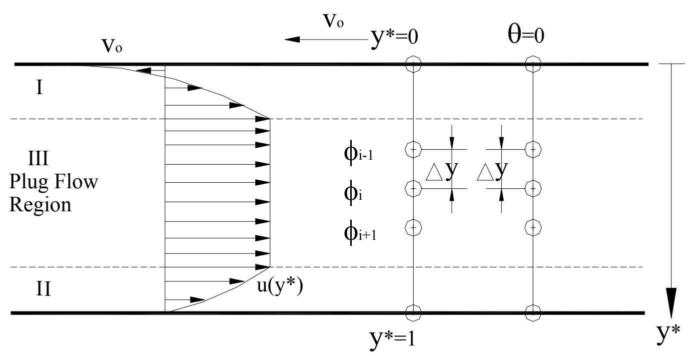
we also obtain that $\partial^2 \phi / \partial y^2 = P/\mu U_o - (n/\nu)\theta$ and $\partial^2 \theta / \partial y^2 = (n/\nu)\phi$. 4

And in dimensionless form the previous ϕ and θ equations respectively are written as

$$\frac{d^2\varphi}{dy^{*2}} = \left(\frac{h}{L}\right) P^* - \mathrm{Re}\, S\theta \qquad 5$$

$$\frac{d^2\theta}{dy^{*2}} = \mathrm{Re}\, S\varphi \qquad 6$$

The finite differencing technique is to be applied to MR damper problem considered here and the domain of solution is shown in figure8.



"Figure (8) Structural drawing of the full solution domain"

Using the finite difference expression, equation5 can be rewritten in a numerical form in order to be introduced to the numerical algorithm as

$$\left.\frac{\partial^2\varphi}{\partial y^{*2}}\right|_i = \frac{\varphi_{i+1_i} + \varphi_{i-1} - 2\varphi_i}{(\Delta y^*)^2} \qquad 7$$

and by substitution into the above equation we get,

$$\frac{\varphi_{i+1_i} + \varphi_{i-1} - 2\varphi_i}{(\Delta y^*)^2} = \left(\frac{h}{L}\right) P^* - \mathrm{Re}\, S\theta_i \qquad 8$$

rearranging terms we have,

$$\varphi_i = \frac{1}{2} \left\{ \varphi_{i+1_i} + \varphi_{i-1} - (\Delta y^*)^2 \left(\frac{h}{L}\right) P^* + (\Delta y^*)^2\, \mathrm{Re}\, S\theta_i \right\} \qquad 9$$

$$\text{Let } F_i = \frac{1}{2} \left\{ -\left(\frac{h}{L}\right) P^* - (\Delta y^*)^2\, \mathrm{Re}\, S\theta_i \right\} (\Delta y^*)^2 \qquad 10$$

$$\varphi_i = \frac{1}{2} \left\{ \varphi_{i+1_i} + \varphi_{i-1} \right\} + F_i \qquad 11$$

similarly for θ (equation7) can be put in finite difference form using the approximation

$$\left.\frac{\partial^2\theta}{\partial y^{*2}}\right|_i = \frac{\theta_{i+1} + \theta_{i-1} - 2\theta_i}{(\Delta y^*)^2} \qquad 12$$

We thus have, $\dfrac{\theta_{i+1} + \theta_{i-1} - 2\theta_i}{(\Delta y^*)^2} = \mathrm{Re}\, S\varphi_i$ 13

That is,

$$\theta_i = \frac{1}{2} \left\{ \theta_{i+1} + \theta_{i-1} - (\Delta y^*)^2\, \mathrm{Re}\, S\varphi \right\} \qquad 14$$

$$\text{Let } E_i = -\frac{1}{2} (\Delta y^*)^2\, \mathrm{Re}\, S\varphi_i \qquad 15$$

$$\theta_i = \frac{1}{2} \left\{ \theta_{i+1} + \theta_{i-1} \right\} + E_i \qquad 16$$

Equation 10 and 15 are tractable to numerical solution using the Gauss Seidel iterative technique. Moreover in dimensionless form, the relation between the plug region boundaries was derived as

$$y_2^* - y_1^* = 2\tau_o^* \Big/ \left(C^* \cos t^* - D^* \sin t^* \right) \qquad 17$$

Where $D^* = R_e S\phi(y_1^*)$ and

$$C^* = (h/L) P^* - R_e\, S\, \theta(y_1^*). \qquad 18$$

The MR damper piston velocity is found, using the continuity assumption, in dimensionless form to be

$$V_0^* = \frac{V_0}{U_0} = 2\left(\frac{h}{R_1}\right) \frac{1}{\cos t^*} \int_0^1 \left[1 + y^* \left(\frac{h}{R_1}\right) \right] \left[\varphi \cos t^* - \theta \sin t^* \right] dy^* \qquad 19$$

And the boundaries of the plug region could then be expressed in terms of the dimensionless piston velocity V_0^* as

$$y_1^* = y_2^* - 2\tau_o^* \Big/ \left((h/L) P^* \left| \cos t^* \right| \right) \qquad 20$$

$$y_2^* = \left[\tau_o^* \Big/ \left((h/L) P^* \left| \cos t^* \right| \right) \right] + \left[V_o^* \left| \cos t^* \right| \Big/ \left((h/L) P^* \left| \cos t^* \right| - 2\tau_o^* \right) \right] + 0.5 \qquad 21$$

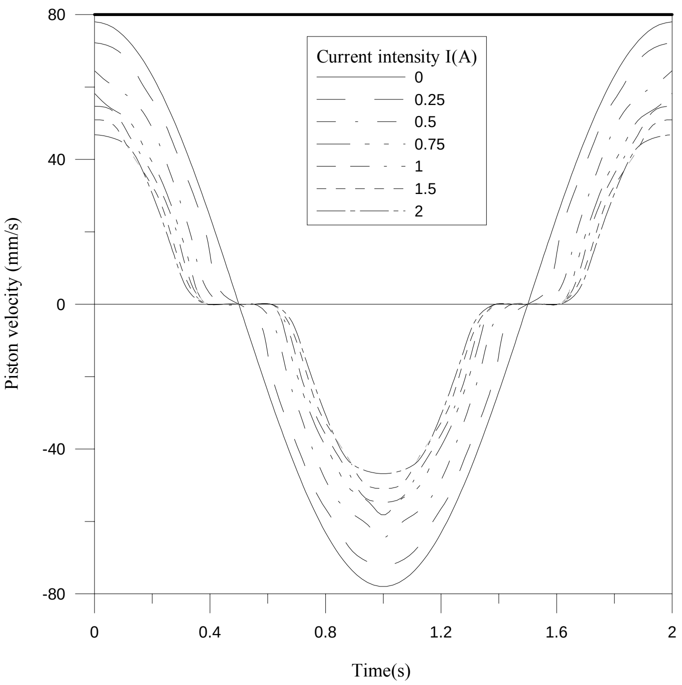
In addition the dimensionless shear stress was found to be

$$\tau_0^* < \frac{1}{2} \left(\frac{h}{L}\right) P^* \left| \cos t^* \right| \qquad 22$$

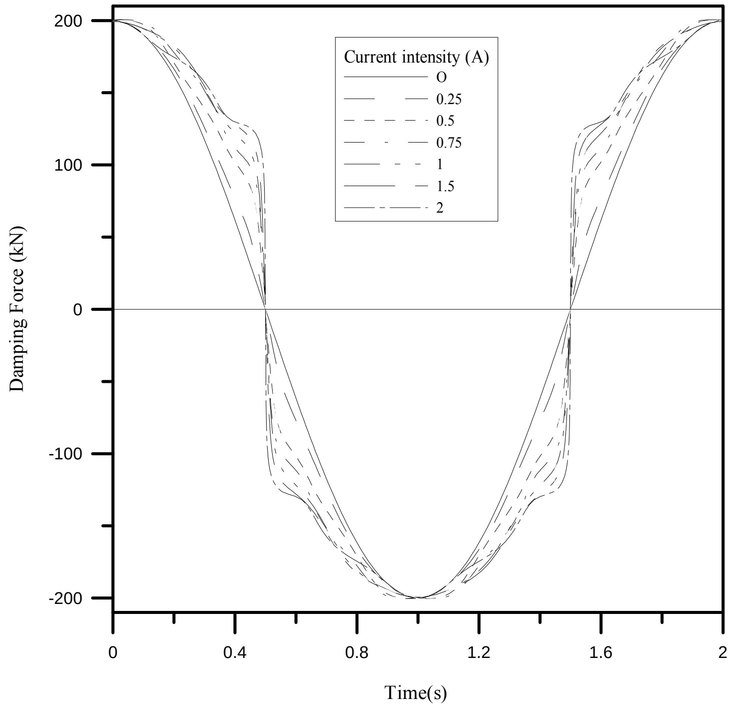
RESULTS AND DISCUSSIONS

Using equation (19), at any chosen time't', the MR damper piston's velocity V_o^* is calculated during the iteration process. And at each iteration a newer corrected value of V_o^* is found. This is continuously done until the values of φ_{m+1}, φ_m, θ_{m+1} and θ_m respectively reach the specified relative tolerance of $\leq$ 0.01, for both θ and φ. After θ and φ convergence, θ, φ and the obtained V_o^* are considered the correct values at that specific time and the whole procedure is then repeated for another value of time't' with a time interval of Δt and so on. This allows us to obtain the entire velocity profile through the whole time domain. Moreover the displacement at any specific time is found by simple integration

of equation (19) with respect to the domain. The result of this integration is the piston displacement at each time‘t’ and this can be plotted together to get the displacement-time.
Using the previously mentioned analysis and by using a dedicated Matlab program that is designed from scratch specifically for the purpose of testing our model, several figures could be plotted to describe the behavior of the MR damper. Figures 9 and 10 show the relationship between the piston velocity against time for sinusoidal force excitation and the relationship between the damping force and the time for sinusoidal displacement or velocity excitation respectively.. It is seen that at zero ampere the plotted curves on both figures have a smooth sinusoidal shape with no plastic, Bingham, behavior shown. But as the current increases the viscous plastic behavior is seen to increase by showing stiffer curves especially at small values of forces and velocities. This is because at these cases the piston velocity is nearly zero, as it changes its direction of motion.

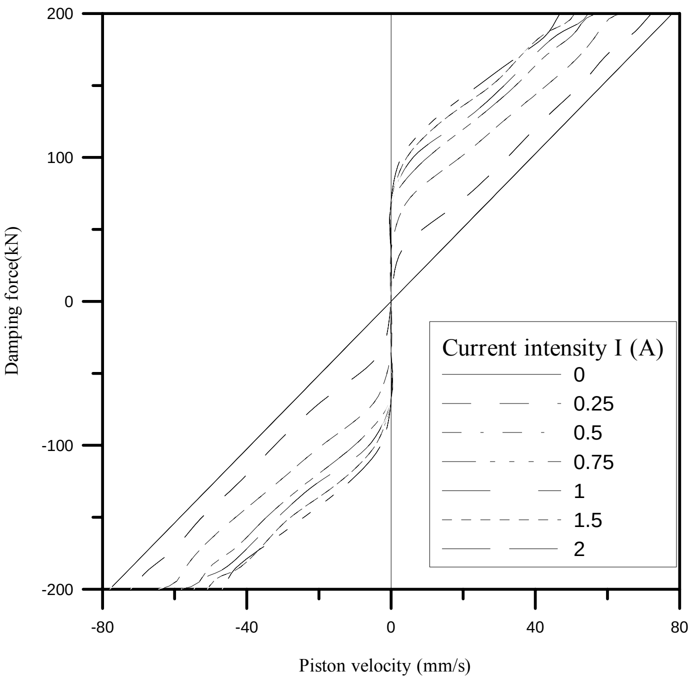


“Figure (9) The Damper piston velocity vs. Time at different input current intensities”



“Figure (10) The Damping Force in (ken) vs. the Time in (sec) At different input current intensities”

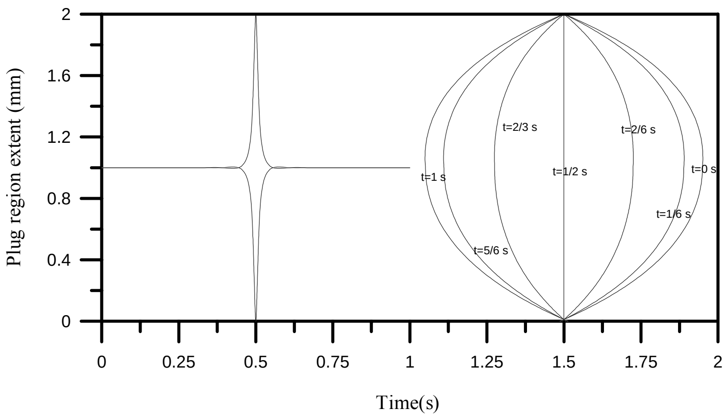
In addition figure 11, shows the relationship between the damping forces against the piston velocity. At zero current intensity the relation- ship is almost linear as if a Newtonian fluid is being used for damping but as the current increases the Bingham model plastic effect is shown to affect the characteristics considerably



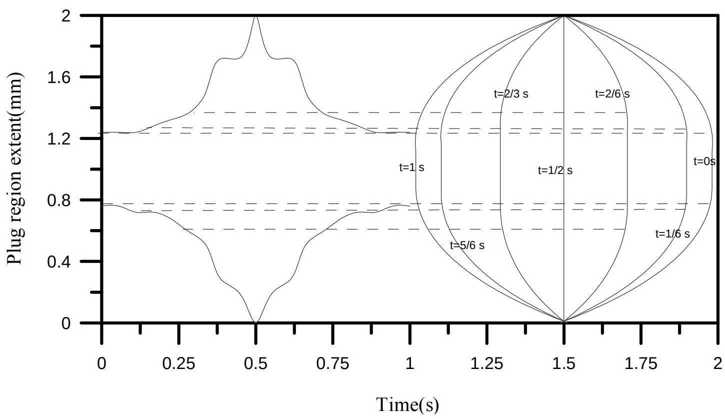
“Figure (11) Damping force vs. the damper Piston velocity At different input current intensities”

Moreover figures 12a through 12f show the effect of increasing the current intensity on the plug flow region thickness, which is created through the MR fluid gap, against time. Again it is seen that as the current increases the extent plug flow region increases until the whole region almost turns into a plug flow region, showing complete solid like effect at the highest allowable current intensity of 2 A, where almost the whole fluid is magnetized into chain solid like structures.

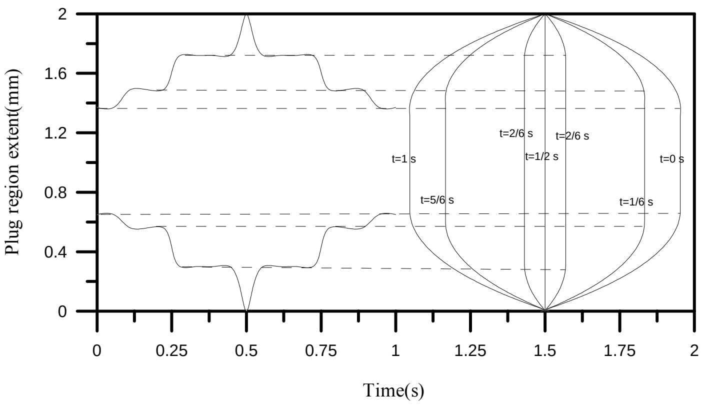
For completion, the velocity distribution, at different current intensities are also plotted on the same figures at several time intervals starting from zero and ending at 2 s with a plot being made every $\frac{1}{6}$ s in time. It should be noted that the extent of the plug flow region at each time interval corresponds to the flat part of the velocity profile. This however is not clear in figure12a as at the described current intensity, zero current; no plug flow region exists except instantaneously when the piston comes to rest upon changing its direction of movement after 0.5 s of starting its motion.



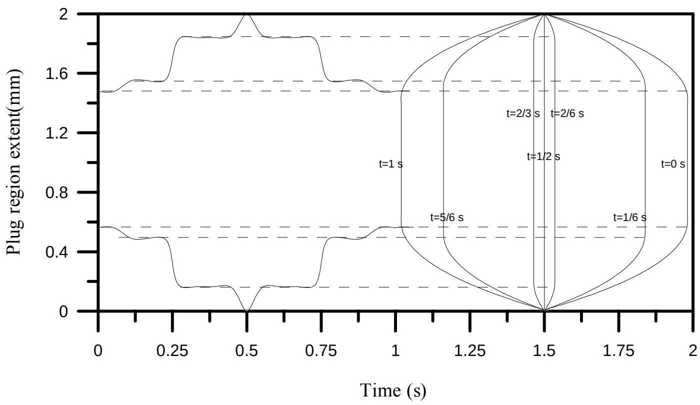
“Figure(12a) The Plug region extent and the velocity distribution against Time in (s) for zero input current intensity”



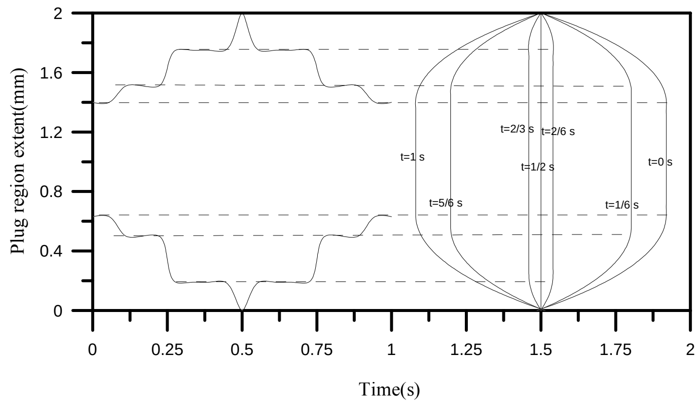
“Figure(12b) The Plug region extent and the velocity distribution against Time in (s) for input current intensity of 0.25A”



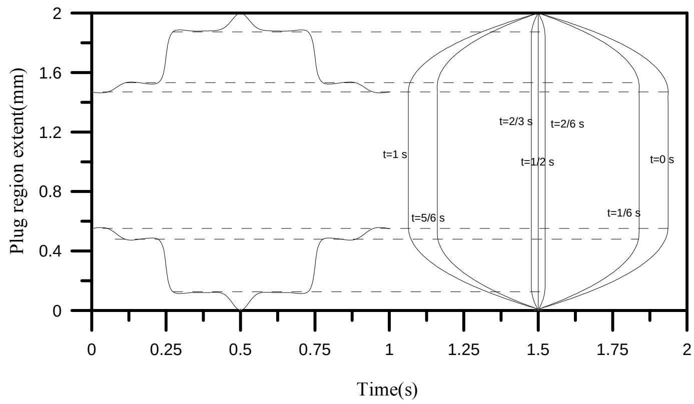
"Figure(12c) The Plug region extent and the velocity distribution against Time in (s) for input current intensity of 0.75A"



"Figure(12e) The Plug region extent and the velocity distribution against Time in (s) for input current intensity of 1.5A"



'Figure(12d) The Plug region extent and the velocity distribution against Time in (s) for input current intensity of 1A"



"Figure (12f) The Plug region extent and the velocity distribution against Time in (s) for input current intensity of 2A"

CONCLUSSION

In this paper, a generalized and accurate time dependent mathematical model was successfully utilized to obtain MR damper characteristics for simulation. Moreover, a special concern was made to focus on the rheological behavior of the MR fluid. Such rheological characteristic was presented by the careful monitoring of the plug flow region, which is considered as a solid like behavior, created by the MR fluid while being used for damping purposes inside the MR damper. In general, an MR fluid was seen to behave like a normal Newtonian fluid when no current was applied on the damper. However as soon as a current intensity, even a small amount, is applied on the MR damper, a plastic Bingham behavior was seen to appear and increase proportionally with the current applied. Our model solution was based on the Bingham mechanical model because of its simplicity and it proved to give good results upon comparison with previously obtained theoretical models and satisfactory results upon comparing it with experimentally obtained results for the MR damper. However, for more accurate results that can highly resemble the experimentally obtained data, it is recommended that future models are to be based upon more complicated mechanical models for their solution such as the Gamota and Filisko model or the Bouc-Wen model

Moreover as previously discussed, it was assumed that the input velocity signal has a sinusoidal like form so that the simple harmonic motion is applied. However, it is recommended that other forms of input signals should be studied and their effect on the MR damper behavior be noted. In Addition during the theoretical model derivation, the curvature effect of the MR damper was neglected for simplicity purposes. But this should be taken into account in the future to monitor if this curvature has any effect on the results obtained. Furthermore, during the course of this study, the yield shear stress (t_y) was assumed to be constant all through the gap (h) and does not depend on the location in (y). Therefore a future detailed study of the effect of the location (y) on the yield shear stress is to be considered.

NOMENCLATURE

Magnetic system description

B the flux per area expressed Wb/m^2 or (Г)

Co the damping coefficient (Ns/m)

n Flow index indicating the degree of deviation from Newtonian behavior

Forces and amplitudes

f_c the frictional force (N)

f_o the offset in the measured force due to the presence of the accumulator (N)

u flow velocity (m/s)

p pressure amplitude (Pa)

q Volume flow rate (m^3/s)

MR damper constants

R_1 the piston radius (mm)

R_2 the cylinder inner housing radius (mm)

Dimensionless numbers used

Re the Reynolds number $\mathrm{Re} = \frac{U_0 h}{\nu}$

S the Strouhal number $S = \frac{nh}{U_0}$

Coordinate system specifications

x longitudinal coordinate(mm)

y coordinate in the direction normal to the solid boundary(mm)

r radial coordinate (radial distance measured from the center line of the piston)(mm)

Greek symbols

$\dot{\gamma}$ shear strain rate (1/s)

μ the viscosity of the MR fluid (found in the gap) (kg/ms)

ν the kinematic plastic viscosity $\nu = \frac{\mu}{\rho}$ (m^2/s)

ρ fluid density (kg/m^3)

τ the total shear stress (Pa)

τ_0 the yield stress caused by the applied field (Pa)

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