Design and Development of Magneto Rheological Fluid Base Damper

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ABSTRACT

Magnetorheological dampers, or as they are more commonly called, MR dampers, are being developed for a wide variety of applications where controllable damping is desired. MR fluid dampers have the capability of changing their effective damping force depending on the current input to the damper. These applications include dampers for automobiles, heavy trucks, prosthetic limbs, gun recoil systems, bicycles, and possibly others related to mechanical discipline like brake, clutch etc.

This synopsis first introduces MR technology through a discussion of MR fluid and then by giving overview of MR device that is being developed. Number of factor in the construction of the damper, as well as the properties of the fluid and the electromagnet, create a dynamic response of the damper that cannot be fully described with a static model dependent on current and velocity. This study will compare different techniques for modelling the force response of the damper in the current-velocity space. All the dynamic response characteristics of the damper are captured in data collection, random input signals were used for velocity and current inputs.

This work issues the design and analysis of the linear magnetorheological damper. Basic information concerning the characteristics of the typical magnetorheological fluid and the damper incorporating it, were presented with the detail description of the applied fluid developed in our premises. With reference to the computations, the prototype damper was designed, manufactured and tested under different operating conditions. Performed calculations were verified with the experimental results and their accuracy was evaluated. The conclusions and observations from the research were compiled in the summary.

1. Brief Description on State of Art

Vibration suppression is considered as a key research field in engineering to ensure the safety and comfort of their occupants and users of mechanical structures. To reduce the system vibration, an effective vibration control with isolation is necessary. Vibration control techniques have classically been categorized into two areas, namely passive and active controls. For a long time, efforts were made to improve the effectiveness of the suspension system by optimizing its parameters, but due to the intrinsic limitations of a passive suspension system, improvements were effective only in a certain frequency range. Compared with passive suspensions, active suspensions can improve the performance of the suspension system over a wide range of frequencies. Semi-active suspensions were proposed in the early 1970s [1], and can be nearly as effective as active suspensions. When the control system fails, the semi-active suspension can still work under passive conditions. Compared with active and passive suspension systems, the semi-active suspension system combines the advantages of both active and passive suspensions because it provides better performance when compared with passive suspensions and is economical, safe and does not require either higher-power actuators or a large power supply as active suspensions do [2].

An exhaustive **literature review** is carried out to understand the present practices and theories in shock absorber design. It also helps to obtain a better understanding of how individual internal components and internal flows had been designed and modelled in the past. It is background information on damper technology.

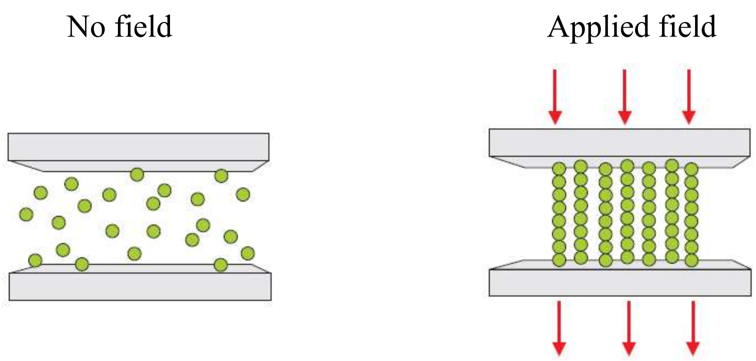
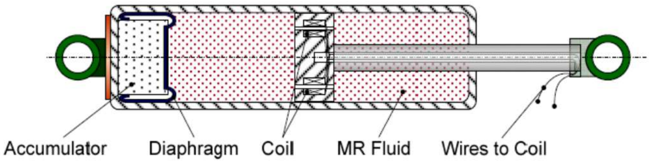


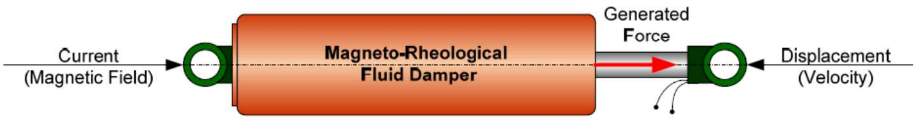
Figure 1: Chain-like structure formation in controllable fluids

The initial discovery and development of MR fluid can be credited to Jacob Rainbow at the US National Bureau of Standards in the late 1940s [6, 7]. These fluids are suspensions of micron-sized, magnetisable particles in an appropriate carrier liquid [8-12]. Normally, MR fluids are free flowing liquids having consistency similar to that of motor oil. However, in the presence of applied magnetic field, the iron particles acquire a dipole moment aligned with the external field which causes particles to form linear chains parallel to the field, as shown in Fig. 1. This phenomenon can solidify the suspended iron particles and restrict the fluid movement. Consequently, yield strength is developed within the fluid. The degree of change is related to the magnitude of the applied magnetic field, and can occur only in a few milliseconds. A typical MR fluid contains 20-40% [5] by volume of relatively pure, soft iron particles, e.g., carbonyl iron. These particles are suspended in mineral oil, synthetic oil, castor oil, water or glycol. A variety of proprietary additives similar to those found in commercial lubricant are commonly added to discourage gravitational settling and promote suspension, enhance lubricity, modify viscosity, and inhibit wear. Recently developed MR fluids appear to be attractive alternative for designing of semi active system (controllable fluid dampers) compared to other smart fluids. [6-12]. Magneto rheological (MR) fluids possess rheological properties, which can be changed in a controlled way. These rheological changes are reversible and dependent on the strength of excited magnetic field. MR fluids have potential beneficial applications when placed in various applied loading (shear, valve and squeeze) modes. The squeeze mode is a geometric arrangement where an MR fluid is sandwiched between two flat parallel solid surfaces facing each other. The distance between these two parallel surfaces is called the gap size. These surfaces are either pushed towards or pulled apart from each other by orthogonal magnetic-induced force.

The ultimate strength of an MR fluid depends on the square of the saturation magnetization of the suspended particles. The key to a strong MR fluid is to choose a particle with a large saturation magnetization. The best practical particles are simply pure iron, as they have saturation magnetization of 2.15 Tesla. Typically, the diameter of the magnetisable particles is 3 to 10 microns. Functional MR fluids may be made with larger particles; however, particle suspension becomes increasingly more difficult as the size increases. Smaller particles which are easier to suspend could be used, but the manufacturing of such particles is difficult. Due to the special behaviour of MR fluid, it is used for vast applications such as: dampers (also called shock absorbers), clutches, rotary brakes, prosthetic devices, polishing and grinding devices, etc. Among them, MR fluid dampers are widely used because of their mechanical simplicity, high dynamic range (Ratio of Controllable force to uncontrollable force), low power requirements, large force capacity and robustness. This class of device has shown to match well with application demands and constraints to offer an attractive means of protecting various engineering systems against interrupted force. MR dampers are being developed for a wide variety of applications where controllable damping is desired. MR damper which utilize the advantages of MR fluids, are semi-active control devices and is popular topic for researchers. A typical MR damper includes MR fluid, a pair of wires, housing, a piston, a magnetic coil and an accumulator as displayed in Fig. 2a. Here, the MR fluid is housed within the cylinder and flows through a small gap between cylinder and piston. The magnetic coil is built in the piston or on the housing. When a current is supplied to the coil, the particles are aligned and the fluid changes from the liquid state to the semi-solid state within milliseconds. Consequently, the controllable damping force is produced. The force produced by a MR damper depends on magnetic field induced by the current in the damper coil and the piston velocity as in Fig. 2b.



(a) Hardware structure



(b) Working principle

Figure 2. General configuration of a MR fluid damper.

It is capable of generating the force with magnitude sufficient for rapid response in large scale applications [13-15], with requirement only a battery for power [10]. Additionally, these devices offer highly reliable operations and their performance is relatively insensitive to temperature fluctuations or impurities in the fluid [9]. As a result, there has been active research and development of MR fluid dampers and their applications [6-18, 20].

2. Definition of the Problem

- To understand basics of the nonlinear force- displacement, force-velocity and force-time behaviour of a new magneto-rheological (MR) fluid damper.
- To develop a theoretical study to predict nonlinear behaviour of new MRF dampers; and
- To conduct a comprehensive experimental study on the proposed MRF damper to validate all theoretical results.

This study aims design, fabrication and characterization of an MRF damper with design and fabrication of test rig for characterization of developed damper.

<u>Theoretical Calculations</u>

A theoretical prediction as well as experimental measurement of the MR damper performance for harmonic input with 0.75, 1, 1.5 and 2 Hz frequency and 0.5, 1, 1.5 and 2 cm amplitude are taken at different current values of 0, 0.25, 0.5, 0.75 and 1A. Damping force, controllable force and dynamic range will be calculated for each combination of mentioned parameters.

3. Objective and Scope of work

The primary objectives of this research are:

1. To develop the MR fluid for application of damper
2. To conceptually design an MR damper
3. To develop the prototype of the MR damper
4. To evaluate the performance of the MR damper experimentally.

In summarized form, following are the scope for present research project.

I. Understand basic theory of vibration.

II. Understand MR fluid basics, behavior, production and applications in various fields as a smart fluid.

III. Understand various designs which are commonly used for MR dampers.

IV. Develop design alternatives from existing MR damper designs.

V. Design of the prototype damper

VI. Prototype fabrication of improved design MR damper

VII. Design and fabrication of test rig for testing damper.

VIII. Testing of the newly developed damper for optimization.

IX. Comparison of MR damper with conventional damper.

4. Original Contribution by the Thesis.

The entire work in this synopsis, as well as thesis is the original work with the research papers presented/published in international journals and conference as the back bone. The proposed model has been visualized as a collection of various modules, each of which with relevant publications. The details of the associated papers are as follows:

1) Magneto rheological fluid (MRF) – Model, Operation mode and Application (Proceedings of International Conference on Advances in materials and Product Design, AMPD-2015, ISBN: 978-93-5196-956-3, p:398-405)
2) Test Rig Design for Measurement of Shock Absorber Characteristics (Proceeding of 3rd Afro - Asian International Conference on Science, Engineering & Technology AAICSET-2015, ISBN: 9-780993-909238, p: 126-130)
3) A Study of MR Fluid Based Damper by Mathematical Model (International Journal of Engineering Science and Futuristic Technology, A Peer-reviewed journal, ISSN : 2454-1338(O), ISSN : 2454-1125(P), Volume 02 Issue 02, February 2016)

4) Instrumentation of Shock Absorber Test Rig (International Journal of Engineering Science and Futuristic Technology, A Peer-reviewed journal, ISSN: 2454-1338(O), ISSN: 2454-1125(P), Volume 03 Issue 01, March 2016.

Furthermore, the MR fluid used in this project is designed and developed as our original product. Test rig which is also designed, fabricated and instrumented according to requirement is our original product.

5. Methodologies of Research, Results and Discussion

This study focuses on the theoretical analysis, design, fabrication and characterization of a small Magneto-Rheological (MR) fluid damper. This study first introduces MR technology through a discussion of MR fluid and then by giving a broad overview of MR devices that will be developed. After giving the broad overview of MR technology and devices, MR damper basics will be presented. This section includes a discussion of MR damper types, mathematical fundamentals and an approach to magnetic circuit design.

With the necessary background information covered, a prototype MR damper, for automobile suspension system is designed, developed, tested and then discussed. These test results were presented and compared with the conventional hydraulic dampers. In conclusion, recommendations were made for materials as well as for seal selection and other design aspects.

5.1 Methodology

The better market access to various manufacturers' fluids as well as their improved parameters and quality stimulates and encourages to create and modify new shock absorbers, clutches, brakes and other devices. Therefore there is a need to develop a comprehensive algorithm for designing such devices utilizing the unique properties of the magnetorheological fluids. The presented work is a proposition of such algorithm with its experimental verification.

The magnetorheological fluid developed in house was used in the prototype. The fluid is a suspension of a 4 to10 micron diameter sized magnetically susceptible particles, in Castror oil carrier fluid. According to the data available by testing this MR fluid on rheometer at this laboratory, the density of the liquid is around 3 g/cm^3 and off state viscosity of a 3.5 Pas. The maximum yield stress value is 15 kPa and it is achieved with the magnetic induction of 0.7 T. When exposed to a magnetic field, the rheology of the fluid reversibly and instantaneously changes from a free-flowing liquid to a semi-solid state with the controllable yield strength as a consequence of the sudden change in the particles arrangement. Figure 3 shows the detail relations of magnetic flux density, viscosity and shear stress available from rheometer.

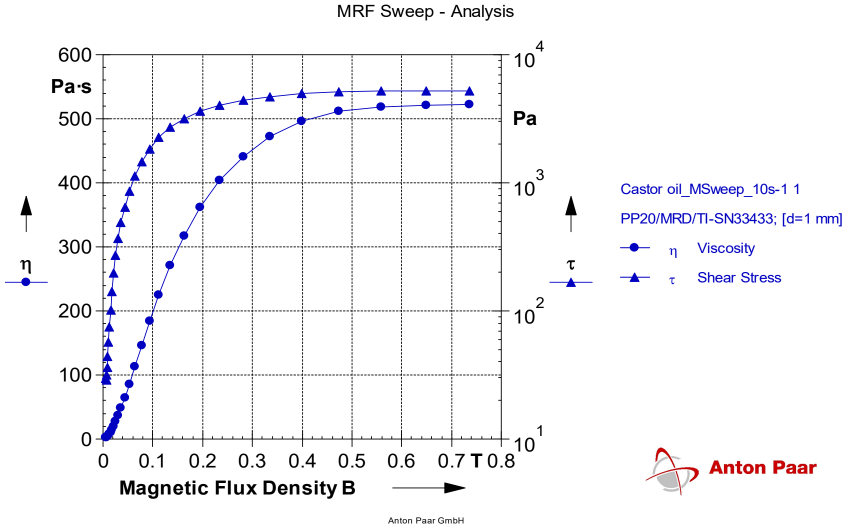


Figure 3. Relation between Magnetic flux density, Viscosity and Shear stress

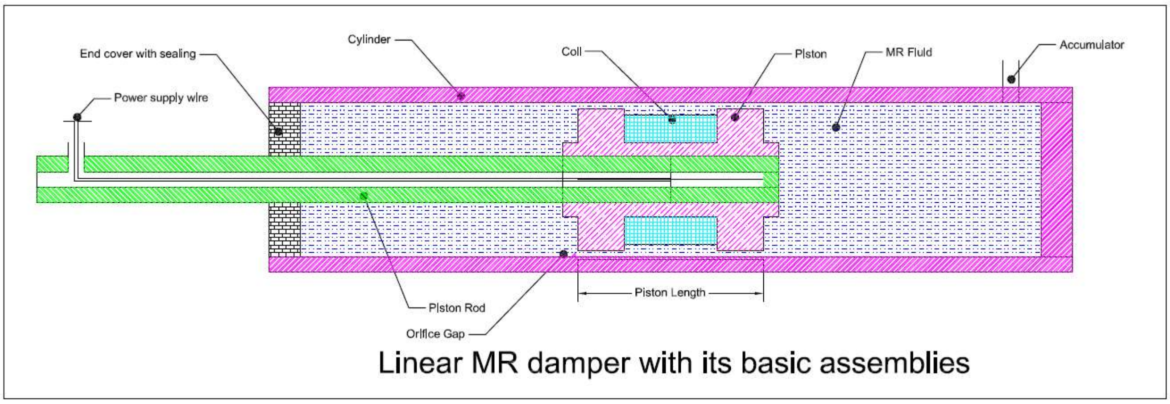


Figure 4. Basic assembly of proposed MR Damper

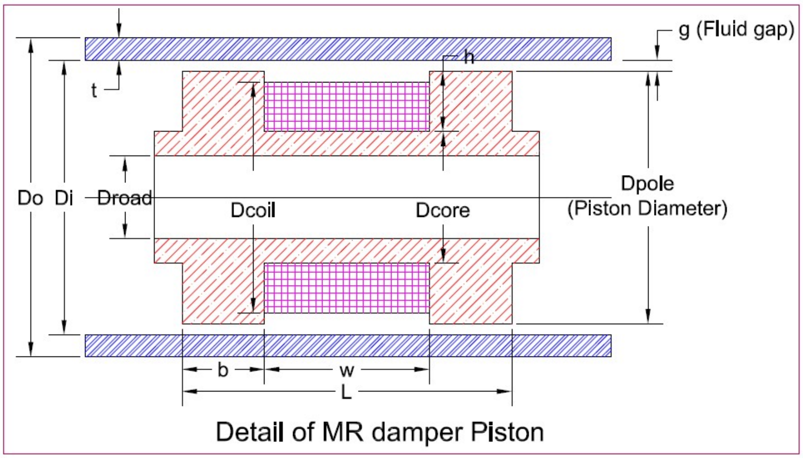


Figure 5. Details of proposed MR Damper Piston

The linear damper prototype is presented as a blueprint in Figure 4. Details of MR damper piston is shown in Figure 5. The photo of the device assembly with its individual components is presented in Figure 6.

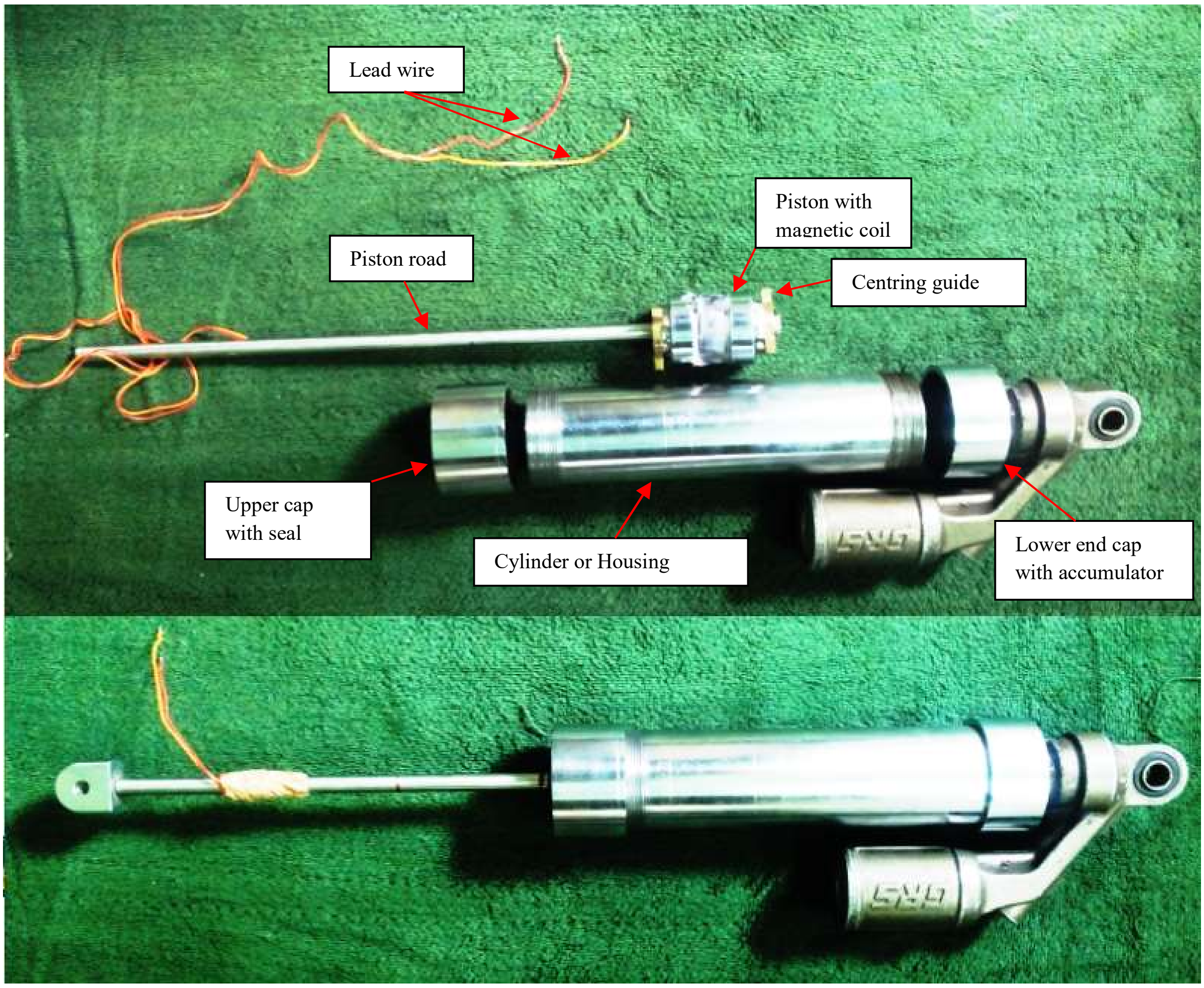


Figure 6. Real detail and assembly photo of predicted MR Damper

MR Fluid Based damper

The damper is made up of two principal components: housing and piston. Housing contains a volume of magnetorheological (MR) fluid. One fluid which has shown itself to be particularly well-suited for this application consists of carbonyl iron particles suspended in castor oil. Housing is a cylindrical tube with a first closed end with an accumulator and attachment eye associated therewith. A second or open end of the cylinder is closed by upper end cap. A seal is provided to prevent fluid leakage from housing. Accumulator is necessary to accommodate fluid displaced by piston rod as well as to allow for thermal expansion of the fluid. It also prevent cavitation effect. Piston head is spool shaped having an upper and a lower outwardly extending flange. Coil is wound upon spool-shaped piston head between upper flange and lower flange. Piston head is made of a magnetically permeable material, in this case, low carbon steel. Guide rails are attached above and below side of piston to keep the piston in centring position to housing during operation. Piston head is formed with a smaller maximum diameter (in this case, D_{pole}) than the inner diameter, D of housing. The external surfaces of guides are contoured to engage the inner diameter D of housing. Guides are made of non-magnetic material, in this case, bronze, and it maintains piston centred within gap ‘g’. In this model, gap g (in conjunction with coil) functions as a valve to control the flow of MR fluid past piston. Electrical connection is made to coil through piston rod by lead wires. A first wire is connected to a first end of an electrically conductive rod which extends through piston rod to outside of damper. The second end of the windings of coil is attached to a “ground” connection on the outside of damper. The upper end of piston rod has threads formed thereon to permit attachment of damper, as depicted in figure. An external power supply, which provides a current in the range of 0-4 amps at a voltage of 12-24 volts, is connected to the leads. The outer surface of coil is coated with epoxy paint as a protective measure. The damper of this experiment functions as a Bingham type damper, i.e., this configuration approximates an ideal damper in which the force generated is independent of piston velocity and large forces can be generated with low or zero velocity. This independence improves controllability of the damper making the force a function of the magnetic field strength, which is a function of current flow in the circuit

Shock Absorber Test Rig

The shock absorber is characterized by its instantaneous value of position, velocity, acceleration, force, pressure, temperature etc and various plots among these parameters. For the measurement of listed parameters of the shock absorber a test rig is designed and developed. An experiment on the test rig is carried out at different speeds and loads which lead to the output in terms of sinusoidal waveform on attached oscilloscope. The waveform is used to find out the characteristics at different load-speed combination. The results obtained are used to find out the behaviour of shock absorber at different speed and loads.

Shock absorber test rig is a machine to test shocks and generate graphs for the shock characteristics. These graphs could be printed or stored for the shocks so user could develop database of how each shock works under the test conditions. This machine replaces the trial and error approach into a reliable and efficient method to determine the characteristic of damper.

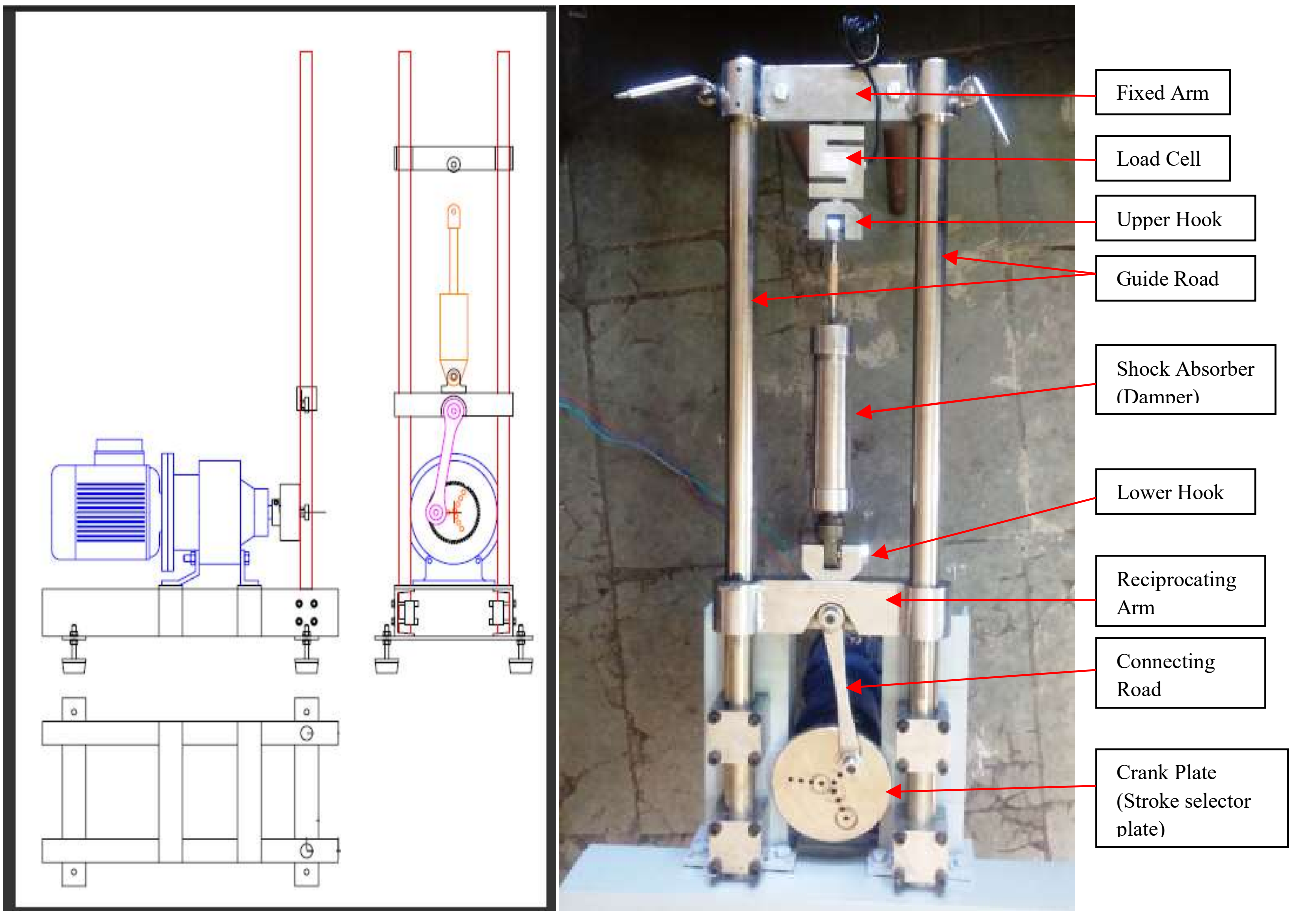


Figure 7. Shock absorber test rig

Figure 7 indicates details of shock absorber test rig setup. The setup consist of Piston and crank (Single Slider Crank Mechanism). This mechanism consists of a flywheel (crank), connecting rod and piston (reciprocating arm). The crank plate has holes drilled to achieve different stroke lengths shown in figure 7. The advantage of this mechanism is its cost effectiveness because there is less high tolerance machining. The frequency is adjusted by using variable gear drive system based electric motor. The output from the motor is geared down using gearbox. The maximum output shaft speed is in the range of 350 to 400 RPM at full speed of motor having 1440 RPM. Variation of stroke is possible by fixing the connecting road in appropriate hole made in crank plate, so the stroke is set to give the desired maximum speed within the limits of the damper and test apparatus. There are twelve screwed holes located over spiral shape; connecting rod can be fixed in suitable hole to select stroke length. The longer the stroke, the greater the power needed on motor to move the shock absorber

<u>Test Procedure</u>

To obtain the data used to characterize the designed MR fluid damper behaviour, a series of experiments on the test rig were conducted under various sinusoidal displacement excitations while simultaneously altering the magnetic coil in a varying current range. The output of each test was the force generated by the damper. During all the experiment, the damping force response measured together with the variation of piston displacement and supplied current for the damper. Fig. 8 depicts an example of relationship between the damping force, piston velocities, and applied current and dynamic response of the damper corresponding to a sinusoidal excitation with 1.0 Hz of frequency at 0.15 cm of amplitude applied to the damper.

Table 1. Setting parameters for experiment

Amplitude	Frequency In Hz	Current Value in Amp.					
0.5 cm	0.75	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	1.0	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	1.5	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	2.0	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
1.0 cm	0.75	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	1.0	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	1.5	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	2.0	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
1.5 cm	0.75	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	1.0	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	1.5	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	2.0	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
2.0 cm	0.75	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	1.0	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	1.5	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	
	2.0	I=0 A	I=0.25A	I=0.5A	I=0.75A	I=1.0A	

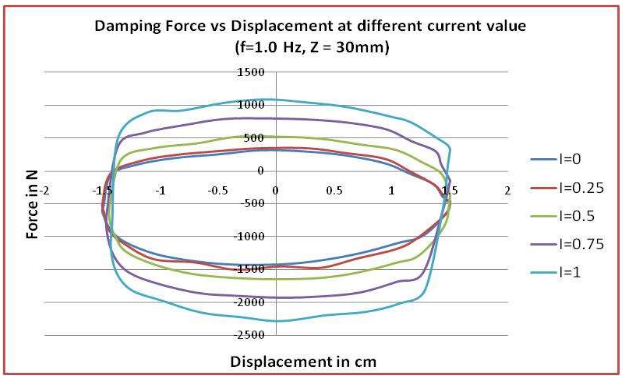
Results of various experimental tests under sinusoidal displacement excitations are presented. These tests include: variable input current tests, frequency dependent test and amplitude-dependent tests.

The setting parameters for experiments are listed in Table 1. Instantaneous values of time, displacement and force for complete cycle is stored in computer via oscilloscope in voltage form. Finally the data are converted in the form of physical unit by C program developed for this special application. Instantaneous value of velocity is calculated by differentiation of displacement with respect to time. Observations are taken according to this plane and represented in graphical form.

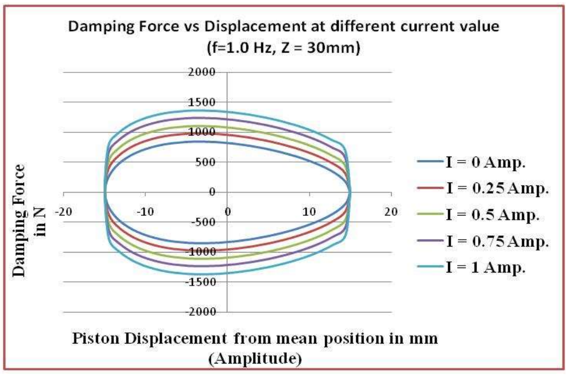
Here graphical representation for only highlighted data in table is shown due to space limitation. In thesis, all details are presented.

5.2 Results

Force-displacement tests under sinusoidal displacement excitation were conducted to investigate the fundamental behaviour of the MR damper. In this experiment, different sinusoidal displacement excitations at frequencies of 0.45, 1, 1.5 and 2 Hz were employed. The input current to damper coil was constant at 0, 0.25, 0.5, 0.75 and 1 A respectively. All tests were conducted at the temperature of 30±3 °C to reduce temperature effects.

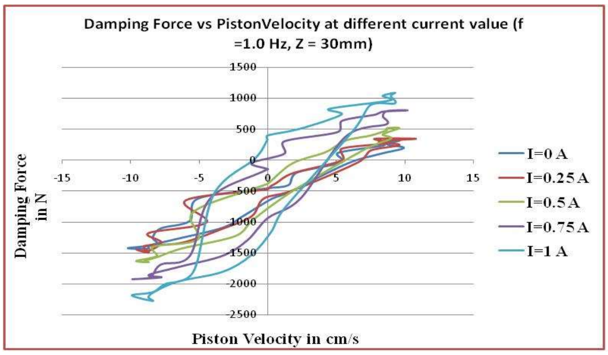


Measured value

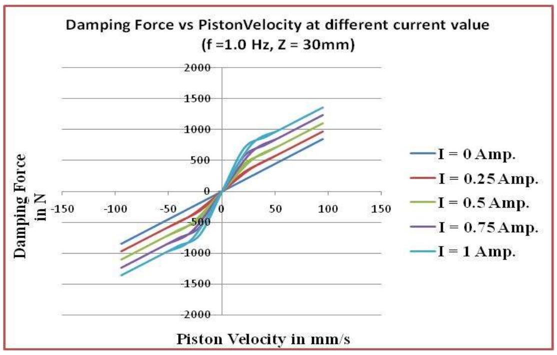


Predicted value

Figure 8-A: Force-displacement relationships under 1.5 cm amplitude for different current values at f = 1 Hz

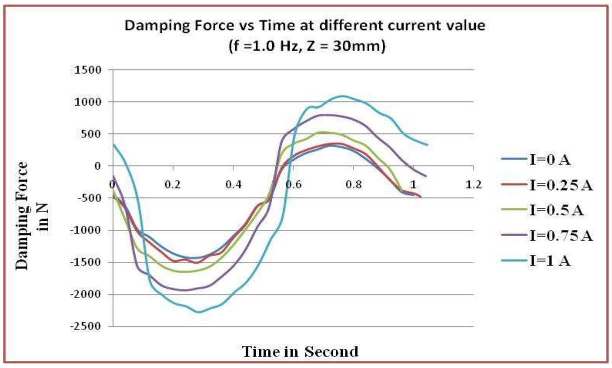


Measured value

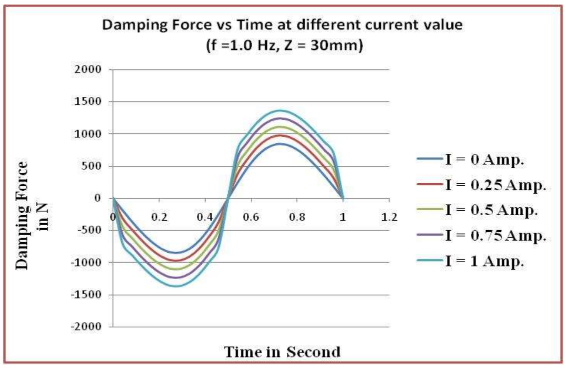


Predicted value

Figure 8-B: Force-velocity relationships under 1.5 cm amplitude for different current values at f =1 Hz



Measured value



Predicted value

Figure 8-C: Force-Time relationships under 1.5 cm amplitude for different current values at f = 1 Hz

Table 2: Measured Maximum Force, Controllable Force And Dynamic Range And Their Comparison With Analytical Results.

			For I=0 A	For I=0.25 A	For I=0.5 A	For I=0.75 A	For I=1 A
N=45 RPM, f=0.75 Hz	Max. Force in N at 15 cm amp. (Span/2)	Measured	566.65	694.00	823.50	925.00	1075.00
		predicted	631.97	761.41	890.84	1020.28	1149.71
		Error in %	-10.34	-8.85	-7.56	-9.34	-6.50
	Controllable Force in N	Measured	0.00	127.35	256.85	358.35	508.35
		predicted	0.00	129.43	258.87	388.30	517.74
		Error in %	0.00	-1.61	-0.78	-7.71	-1.81
	Dynamic Range	Measured	0.00	0.22	0.45	0.63	0.90
		predicted	0.00	0.20	0.41	0.61	0.82
		Error in %	0.00	9.73	10.66	2.92	9.51
N=60 RPM, f=1 Hz	Max. Force in N at 15 cm amp. (Span/2)	Measured	875.00	1000.00	1124.50	1272.00	1434.50
		predicted	842.60	972.04	1101.47	1230.91	1360.34
		Error in %	3.85	2.88	2.09	3.34	5.45
	Controllable Force in N	Measured	0.00	125.00	249.50	397.00	559.50
		predicted	0.00	129.43	258.87	388.30	517.74
		Error in %	0.00	-3.43	-3.62	2.24	8.07
	Dynamic Range	Measured	0.00	0.14	0.29	0.45	0.64
		predicted	0.00	0.15	0.31	0.46	0.61
		Error in %	0.00	-7.00	-7.19	-1.55	4.06
N=90 RPM, f=1.5 Hz	Max. Force in N at 15 cm amp. (Span/2)	Measured	1155.00	1285.00	1397.50	1542.00	1669.50
		predicted	1263.86	1393.29	1522.73	1652.16	1781.60
		Error in %	-8.61	-7.77	-8.22	-6.67	-6.29
	Controllable Force in N	Measured	0.00	130.00	242.50	387.00	514.50
		predicted	0.00	129.43	258.87	388.30	517.74
		Error in %	0.00	0.44	-6.32	-0.34	-0.63
	Dynamic Range	Measured	0.00	0.11	0.21	0.34	0.45
		predicted	0.00	0.10	0.20	0.31	0.41
		Error in %	0.00	9.90	2.51	9.06	8.74
N=120 RPM, f=2 Hz	Max. Force in N at 15 cm amp. (Span/2)	Measured	1545.00	1673.00	1803.00	1900.00	2070.00
		predicted	1685.20	1814.64	1944.07	2073.51	2202.94
		Error in %	-8.32	-7.81	-7.26	-8.37	-6.03
	Controllable Force in N	Measured	0.00	128.00	258.00	355.00	525.00
		predicted	0.00	129.43	258.87	388.30	517.74
		Error in %	0.00	-1.11	-0.34	-8.58	1.40
	Dynamic Range	Measured	0.00	0.08	0.17	0.23	0.34
		predicted	0.00	0.08	0.15	0.23	0.31
		Error in %	0.00	7.87	8.71	-0.28	10.60

MR damper force-displacement, force-velocity and force-time behaviour

Figures 8 - A show the measured and predicted force-displacement loops with different input current levels for predicted damper configuration. The displacement excitation is a sinusoidal waveform with amplitude of 1.5 cm. Other experimental results with amplitude of 0.5, 1.0 and 2.0 cm with frequency of 0.45, 1.5 and 2 Hz are presented in final thesis due to space limitation in synopsis.

As can be seen, the MR damper resisting force increases as the applied current increases. Moreover, the area enclosed by the force-displacement loop also enlarges, and more energy is dissipated. Figures 8 - B provide the measured MR damper force-velocity behaviours and comparisons with theoretical results. Due to the plastic viscous force, a larger damping force is seen at high velocity. Figures 8 - C provide the measured MR damper force-time behaviours and comparisons with theoretical results. Table 2 provides the measured maximum damping force, controllable force and dynamic range with their comparison to analytical results. Again, close agreement is observed with maximum errors of about 10%.

5.3 Discussion (MR fluid damper characteristic analysis)

In order to design the MR fluid damper models, an investigation into factors which affect the dynamic responses of the damping system has been done. The first affecting factor is the applied displacement/velocity on the piston rod of the damper. Figures 8 displays a comparison between damping results under various sine excitations with 1.5 cm amplitude and frequency 0f 1.0 Hz, while the supplied current level was in range from 0 to 1A. The results show that at fixed current level applied to the damper, the damping force varies due to the piston rod velocity which is caused by the simultaneous change of frequency and/or amplitude of the applied excitation. The second factor affecting the damper behaviour is the change in current applied to the damper coil. Figures 8 shows an example of measurement results in plots of force-displacement, force-velocity and force-time relations with respect to sinusoidal excitation of 1.5 cm amplitude while the current supplied to the damper was in range between 0 and 1A. From these figures, it is readily apparent that:

(1). The force produced by the damper is not centered at zero. This effect is due to the effect of an accumulator containing high pressure nitrogen gas in the damper. The accumulator helps to prevent cavitation in the fluid during normal operation and accounts for the volume of fluid displaced by the piston rod as well as thermal expansion of the fluid.

(2). Greater the current level, greater in the damping force. Increasing the current in the device's coil, enlarges the magnetic field flux and thus, increases the yield stress denoted as $\tau_0(B)$, figure 3, increases the controllable force, increases also the dynamic range.

(3). The change rate of force is faster at lower current levels because of the effect of magnetic field saturation. Based on the above analyses, it is clear that the damping force of the MR fluid damper depends on the displacement/velocity of the damper piston and the current supplied for the coil inside the damper.

(4). Referring to Table 2, MR damper with low frequency and high current value have higher dynamic ranges. Dynamic range is increased by increasing current value witch increase controlled force up to saturation state. By increasing value of frequency, instantaneous value of velocity is increased witch increases viscous force or uncontrolled force, reduces dynamic range.

(5). As shown in Fig. 3, the magnetic field is almost saturated at the input current level of 1 A for MR fluid used for this investigation of MR damper; only very small increase in yield stress is observed when the input current increases from 0.9A to 1A. However, the yield stress increase is more noticeable in the current range of 0.2A to 0.8A, which is also effect the material of cylinder housing; in this case it is low carbon steel.

(6). With the help of mathematical model, if gap size for damper is increased, large gap size reduces the magnetic field due to its larger magnetic resistance. Consequently, it reduces the yield stress of the MR fluid; it assumes that the materials used in the magnetic loop are the same. This also implies that the use of proper material, in this investigation, it is low carbon steel, which has a high conductive permeability, increases the magnetic field in the gap at a high current level. This results in an increased yield stress. Moreover, these configurations also exhibit reduced damping forces due to their geometry.

(7). From figures. 8-A (measured), force overshoots are clearly seen at the displacement extremes, where the velocity changes its direction. These overshoots appear to be primarily due to the stiction phenomenon found in MR fluids. Because large acceleration occurs at these points due to the velocity discontinuity of the sinusoidal displacement excitation, other effects, such as fluid inertial force may also contributed to these overshoot.

(8). The predicted and measured vales of damping force, controllable force and dynamic range are differing about $\pm 10\%$. This is due to the assumed simplifications of the phenomena connected with the operation of the device, revealed a major influence on the computational inaccuracies. This effect may also be related to the simplifications in calculation of the pressure drop.

(9). Precise computations of the magnetic induction are complicated, while the accurate experimental research is time and effort consuming. The $\tau_0(B)$ function provided by the manufacturer of the MR fluid is usually approximate and imprecise which negatively influence the accuracy of the computations. It is experimentally concluded that the maximum damping force varies linearly as the value of the yield stress varies. Difficulties in determining the magnetic induction value explain the high variation of the error. The most accurate calculations can be obtained for the smallest gap height, due to the possibility of the precise determination of the magnetic field and the small error of the simplified model of the flow between parallel plates.

6. Achievements with respect to objectives

All factors related to objectives of present research project define in section 3 of this report are achieved positively. MR fluid is prepared and tested with rheometer available in the laboratory of physics department, M. K. Bhavnagar University and the output is shown in figure 3. All the parameters are studied carefully related with MR damper design and finally design and make a prototype MR damper. Investigation is carried out by mathematical modelling and experimentally on proto type MR damper with design and developed test rig at this laboratory

7. Conclusions

In this work, physical phenomena of a MR fluid base damper have been carefully investigated through both experimental data and modelling methodologies. The test rig using the MR fluid damper has been fabricated in order to design the model as well as to evaluate the proposed damper.

The assumed simplifications of the phenomena connected with the operation of the device, revealed a major influence on the computational inaccuracies. It can be concluded that the most accurate calculations can be obtained for the smallest gap height, due to the possibility of the precise determination of the magnetic field and the small error of the simplified model of the flow between parallel plates. The analysis suggests the need to develop more precise tools supporting the design process of the devices with MR fluids.

It seems reasonable to create a reverse algorithm that will allow estimation of the geometry of the device based on the desired value of the dissipated energy. In addition it is necessary to determine more accurately the value of the magnetic induction in the flow gap of the MR device. It would be also interesting to define the influence of the temperature on the viscosity and the yield stress, as well as to take this influence into account for the theoretical calculations.

Various tests have been carried out using a test rig to verify the damping characteristics of developed MRF dampers. The following test results were obtained:

(1) The damper functioned by using a unit of the electromagnet under an appropriate electrical current control.

(2) The magnitude of the damping force depends on the input magnetic field, but it has an upper limit.

(3) In the absence of an applied magnetic field, an MRF damper exhibits viscous like behaviour, while it shows friction-like behaviour in a magnetic field.

Through a series of experiment, it is confirmed that the behaviour of the MRF dampers is fairly predicted by the velocity-displacement and velocity-force relationship over a wide range of applied current, amplitude, and frequency. It is clarified that the MRF dampers provide a technology that enables effective semi-active control in real development of various structures.

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