Design and Testing of Magneto-Rheological Damper for Vehicle Suspension

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Abstract: Conventional suspension systems have been ruling the automobile industry since ages but with the advancement in material technology have led us to a new kind of suspension systems known as Active Suspension systems which involves smart materials like Magnetorheological Fluids (MR), Electrorheological Fluids (ER) etc. In present day conventional suspension system, we have a coiled spring setup in motorcycles which are passive in nature. But off late in 21st century the development of smart materials bought in new dimensions of research in the suspension technology which has led to the era of active suspension systems. Now we can see usage of Semi Active Suspension/Active suspension system in cars and heavy vehicles which enhances the life of automobile, ride comfort and drive control. Though these systems have been developed for high end cars and military vehicles, the implementation of this technology to motorcycles has been a topic of research interest to various motorcycle industries .We hence suggest implementing the MR fluid-based semi active suspension system to Motorcycles, this system senses the irregularities of the road with various sensors and tries to stiffen and soften the suspension system in motorcycles in turn providing safe and snuggly ride with increased control during rough terrain. Thereby with this suggested concept an alternative design consideration of suspension systems has been offered.

***Keywords* –Magnetorheological Fluid, Semi-Active Suspension, Passive suspension.**

I. INTRODUCTION

With the advancement in materials technology new class of material namely Magneto-Rheological fluids have taken over the automotive industry with respect to suspension systems. A magneto rheological (MR) fluid is a fluid whose rheological properties may be rapidly varied by applying a magnetic field. Owing to this feature of the fluid its applications is being considered in a large variety of devices, such as shock absorbers, brakes or clutches. The performance of the MR fluid device depends upon the magnetic field supplied to the MR fluid. The response time of the MR fluid is itself within several milliseconds which make the actuators relatively slow. Numerous studies conducted recently on design of MR dampers have shown the Bingham model of MR fluids under the presence of magnetic field and also the governing equation of the MR damper is derived based on Bingham plastic model. Therefore, the attempts to determine the principal design parameters based on Bingham model have been made. The main aim of this study is to design an effective MR damper with optimum performance for vehicle suspensions. A brief introduction about existing dampers and types of suspension systems used today is given below for better understanding of the MR damper.

II. MECHANICAL DAMPER

Dampers are necessary as the spring do not settle down fast enough. In other words, after a spring has been compressed and released it continues to shorten and lengthen or oscillate for a while.

If a wheel of a car hits a bump, the spring compresses, later the spring expands after the wheel passes the bump the expansion of the spring causes the car to be thrown upwards. This causes the wheel to momentarily leave the road and the car drops down, this action is repeated until the oscillations gradually die out. Such spring action on car would produce very bumpy and uncomfortable ride, it would also be dangerous as a bouncing wheel is would be impossible to control. There an element is needed to control the oscillation action of the spring, which is known as Shock Absorber or Damper.

There are many types of shock absorber available in which telescopic (piston-cylinder) shocks are commonly used. The telescopic shocks consist of outer cylinder, inner cylinder, piston and piston rod. At the bottom of the inner cylinder and piston a set of valves controls the flow of hydraulic fluids movement.

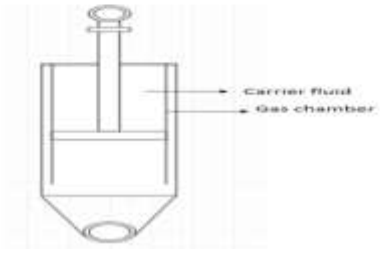

Fig 1: Conventional Automobile Damper

III. SUSPENSION SYSTEM

Suspension system of a vehicle connects the body of the vehicle to the tyre. Suspension system can be classified into three categories which is passive, semi active and active suspension system. This suspension system categorizing depends on the external power input or the control into the system.

3.1 Passive Suspension

The passive suspension system has been used in the automotive industry since 1906. It has a spring and a damper mounted on each wheel of the vehicle in parallel. A passive suspension system is one in which the characteristics of components (springs and dampers) are fixed. Passive suspension is a compromise between vehicle handling and ride comfort as shown in fig. 1

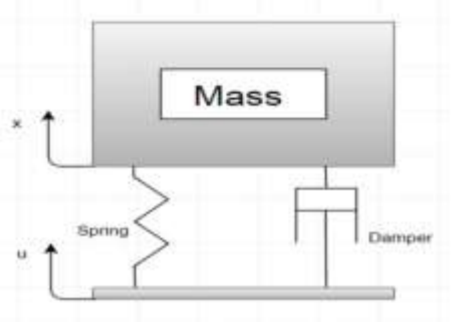

Fig 2: Passive suspension and semi-active suspension system

A heavily damped suspension will yield good vehicle handling, but also transfer much of the road input to the vehicle body. A lightly damped vehicle will yield a more comfortable ride, but can significantly reduce the stability of the vehicle in turns. Good design of passive suspension can to some extent optimize ride and stability, but cannot eliminate this compromise. Due to lack of altitude control of the vehicle and demand for better ride comfort and stability, it inspires and motivates the automotive industry to consider the use of active suspensions.

3.2 Semi-active Suspension

In this type of system, the conventional spring element is retained, but the damper is replaced by controllable damper to dissipate the energy. A semi-active system uses external power to adjust the damping levels, and to operate an embedded controller and set of sensors. The controller determines the level of damping. The controllable damper usually acts like with limited capability to produce a controlled force when dissipating energy. Some of the semi-active suspension systems use a passive damper and a controllable spring. Semi-active suspension system is shown in fig 1.The advantage of using semi-active suspension is the operational cost is less than active suspension and only small amount of energy is consumed.

3.3Active Suspension

In active suspension system, the passive dampers or both the passive dampers and springs are replaced with a force actuators as shown in fig 2.

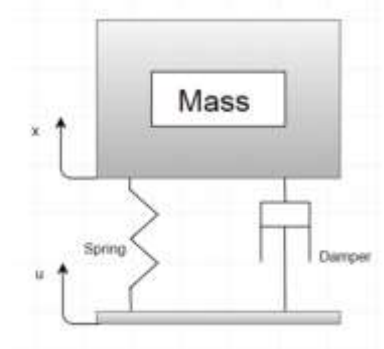

Fig 3: Active Suspension System.

The force actuators are able to both add and dissipate energy from the system, unlike a passive damper, which only dissipates energy. With an active suspension, the force actuators can apply force independent of the relative displacement or velocity across the suspension. This results in better compromise between the ride comfort and vehicle stability as compared to passive system.

IV. MECHANICAL DESIGN OF MR DAMPER

Fig 4 depicts the schematic configuration of the MR damper considered in this work. This MR damper consists of a MR fluid, a casing, a gas accumulator, and a piston head in which the magnetic circuit is included. The casing is divided into upper and lower chambers by the piston head where the upper chamber is fully filled with the MR fluid. When the piston head moves, the MR fluid is moved through the annular gap between the casing and the piston head from one chamber to the other, where the MR fluid is exposed to the applied magnetic field. In order to compensate for changing the fluid volume occupied by the piston rod in the upper chamber, the gas accumulator is located in the lower chamber. The magnetic circuit is composed of the MR fluid, the electromagnetic coil, and the ferromagnetic yoke for forming the magnetic flux path. When an external current is supplied to the electromagnetic coil, a magneto motive force is generated in the magnetic circuit and the corresponding magnetic field is applied to the MR fluid. Then, the dynamic yield stress of the MR fluid is changed depending on the magnetic field intensity, and the resultant damping force is activated in the opposite direction to the motion of the piston head. In the absence of a magnetic field, the damping force occurs only due to the viscosity of the MR fluid itself. However, in the presence of a magnetic field, an additional damping force occurs due to the MR effect of the MR fluid. Therefore, the damping force of the MR damper can be continuously and reversibly controlled by adjusting the coil current.

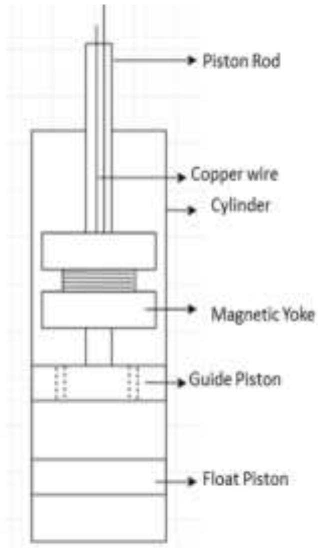

Fig 4: Schematic diagram of MR damper.

For the practical design of MR damper few assumptions are made (i) the polar particles are uniformly distributed in the MR fluid without any concentrations, (ii) the flow velocity of the MR fluid is obtained from no slip and linear velocity distribution conditions, (iii) the frictional force is negligible, and (iv) the thermodynamic and inertial effects of the MR fluid can be also disregarded.

The magnetic field dependent characteristics of the MR fluid can be determined by Bingham plastic model. it has been proved by numerous researchers that this simple model is very useful for the design and modeling of MR fluid-based devices (Jolly et al., 1999; Stelzer et al., 2003). Therefore the equation for proposed MR damper can be written as (Han et al., 2002; Choi and Wereley, 2003).

$F_d = F_{MR} + F_\eta + F_g$ (1)

In the above equation,

$F_{MR} = (A_p - A_r)(2L/h)\,\tau_p(H)\,\mathrm{sgn}(\dot{x})$ (2)……………… Force due to magnetic field

$F_\eta = (12\eta L_p/bh^3)(A_p - A_r)^2\dot{x}$ (3)………………….. Force due to viscosity

$F_g = A_r P_{g0}\,(V_{g0}/(V_{g0} - A_r x))^n$ (4)…………………… Force due to gas pressure

In the common case, the viscous damping force and the gas force can be considered to be relatively small, compared with the MR effective force. Therefore, the principal design parameters of the MR damper can be effectively determined only by using force due to magnetic field. The maximum magnetic field intensity H_{max} is constrained by the magnetization property of the MR fluid, and then the maximum yield stress $\tau_{y,max}$ is determined in the relation of $\tau_y(H_{max})$. For the convenient fabrication of the MR damper, the gap size h is generally specified in the range of 0.5–2mm. In addition, the cross-sectional areas A_p and A_r can be sensibly determined considering the required size and capacity of the MR damper. Then, with the aid of these design parameters, the length of the magnetic pole required for the desired damping force $F_{d,des}$ can be found by,

$L = F_{d,des} * h/(2(A_p - A_r)\tau_{y.max})$ (5)

The next challenge is to design the electro-magnetic coil to provide adequate magnetic field intensity to the MR fluid as much as possible. The increased magnetic field intensity leads to the improvement in the static performance of the MR damper. The

relationship between the magneto motive force generated in the magnetic circuit by applying the coil current and the corresponding magnetic field intensity is given by
Ampere's law:

$$NI = \Phi_c H.dl = H_f\, l_f + H_s l_s \qquad (6)$$

Another constraint is added to prevent magnetic saturation in the yoke and is it based on Gauss law:

$$\emptyset = \int \boldsymbol{B}.\mathbf{n}dS = B_f S_f = B_s S_s \qquad (7)$$

Where B_f and B_s represent the magnetic flux densities of the MR fluid and yoke, respectively. And, S_f and S_s represent the cross-sectional areas of the MR fluid and yoke, respectively. The value of B_f is obtained according to value of max yield stress of the fluid and $B_{s,sat}$ from magnetization curve , both of which are presented in the properties table below. In addition S_f is found by bL/2 with help of equation (5). In order to prevent yoke from getting saturated magnetically S_s should be determined in the following range:

$$S_s \geq S_{s,\,min} \equiv \frac{Bfmax\ Sf}{Bs,sat} \qquad (8)$$

So, with the aid of minimum cross sectional area of the yoke $S_{s,min}$ determined in equation 8, the outer radius of casing r_o and the yoke core r_c can be respectively obtained as:

$$r_o = \sqrt{\frac{Ssmin}{\pi} + ri^2} \qquad (9)$$

$$r_c = \sqrt{\frac{Ssmin}{\pi} + rb^2} \qquad (10)$$

With aid of all the equations above the design parameters of the damper is calculated and tabularized as follows:

Table 1:- Design specifications of the MR damper

Parameters	Symbol	Value
Desired damping force	F_{des}	2000N
Available maximum current	I_{max}	5.0A
Coil turn number	N	80
Gap size	h	1.0mm
Inner radius of the casing	r_i	22.0mm
Outer radius of the casing	r_o	23.0mm
Outer radius of piston	r_p	21.0mm
Radius of piston rod	r_s	5.0mm
Radius of yoke core	r_c	16.2mm
Length of magnetic pole	l_p	10.0mm
Length of yoke core	l_c	10.0mm
Coil diameter	-	0.5mm
Electric resistance of the coil	-	1.24Ω
Initial gas pressure	P_{g0}	10 bar
Initial gas volume	V_{g0}	11634 mm^3

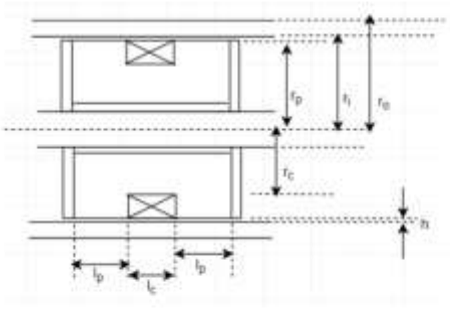

Fig 5: Design parameters of MR damper.

Table 2:- The properties of the MR fluid used and the magnetic yoke are tabularized as follows:-

Property	Value	Material
$\tau_{y,max}$	38.28kpa	A novel MR fluid by BMSCE lab
$B_{f,max}$	0.676T	SS41 steel or EN 8 grade steel for yoke (An and Kwon 2003)
η_f	150A/mm	
$H_{s,sat}$	1.989A/mm	
$B_{s,sat}$	1.326T	

V. MODELLING OF MR DAMPER

After completing the crucial part of designing we arrived at the suitable dimensions using the formulas on MAT lab software. Using those dimensions we modeled the design using CATIA V5 tool and created a virtual damper for easy understandability.

Fig 5: CATIA models.

VI. MANUFACTURING OF MR DAMPER

In order to manufacture the MR damper we performed various machining processes to attain accuracy in dimensions.
The following are the list of important operations that were performed,

- Turning–
- Drilling
- Grinding
- Milling
- Threading

- Honing
- Milling

Fig 6: Manufactured MR Damper

VII. WORKING

Once the suspension is locked in position using the attachment points, the input and output points of the coil are connected to a constant current source. Once the system is turned on, a constant current flow through the coil, creating magnetic field lines around the core (the inner cylinder). Now these magnetic field lines will cause the iron particles in the liquid to align with the field lines, thereby increasing the concentration of the particles along these lines. This will move the liquid from a liquid state to a semi-solid state, and the viscosity will eventually increase. The increase in viscosity leads to an increase in the resistance force on the piston during its movement and therefore to an increase in the damping effect. The operating mode would be the compression valve mode. The valve mode is used around the port in the piston, while the compression mode takes place between the top of the piston and the cylinder head.

VIII. EXPERIMENTAL TEST SETUP

The test setup consisted of a Compression testing machine with a capacity of 10 tonne, a 5V DC battery to supply the MR damper current and a dial gauge to record the displacement of the piston. The damper has a stroke of ±25mm and was supplied various currents from 0amps to 1amp. The damper was placed between the jaws of the machine in its extended position and due to limitation of the dial gauge displacement of the piston was constrained to a maximum of 3 mm. The hydraulic upper arm applied load on the piston rod and corresponding displacements were recorded with help of dial gauge. The test setup used is as shown below

Fig 7: Experiment Test Setup

IX. RESULTS AND DISCUSSIONS

The results of the above test can be tabulized as follows:-

For 0 Amp current,

Sl. No.	Displacement (mm)	Force (Kg-F)
1	0.1	100
2	0.2	115
3	0.3	120
4	0.4	120
5	0.5	120
6	0.75	120
7	1	121
8	1.5	121
9	2	122
10	2.5	122
11	3	125

For 0.5 Amp current,

Sl. No.	Displacement (mm)	Force (Kg-F)
1	0.1	120
2	0.2	120
3	0.3	120
4	0.4	120
5	0.5	121
6	0.75	125
7	1	125
8	1.5	125
9	2	125
10	2.5	125
11	3	130

For 0.8 Amp current,

Sl. No.	Displacement (mm)	Force (Kg-F)
1	0.1	125
2	0.2	125
3	0.3	127
4	0.4	127
5	0.5	130
6	0.75	130
7	1	130
8	1.5	130

For 1 Amp current,

9	2	132
10	2.5	135
11	3	135

Sl. No.	Displacement (mm)	Force (Kg-F)
1	0.1	130
2	0.2	130
3	0.3	131
4	0.4	131
5	0.5	132
6	0.75	132
7	1	132
8	1.5	132
9	2	132
10	2.5	134
11	3	140

Using the above data graphs of load v/s displacement and max load v/s max current were plotted, which suggested that the damper under consideration showed variable viscosity and thus the approach used for designed the MR damper was a successful one. As evident from the above tables and below graphs there is an increase in the load for the same amount of piston displacement with increase in current , suggesting that there is an increase in the viscosity of the MR fluid inside the damper. Between 0 amperes and 0.5 ampere there is a 20kgf difference in the load and from 0.5 to 1 ampere there is 10kgf increase in load.

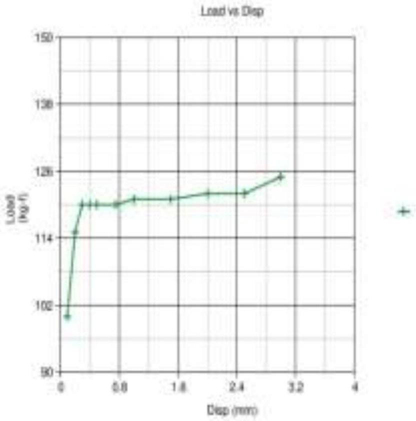

Fig 8: Force v/s Displacement graph at 0 Amp current.

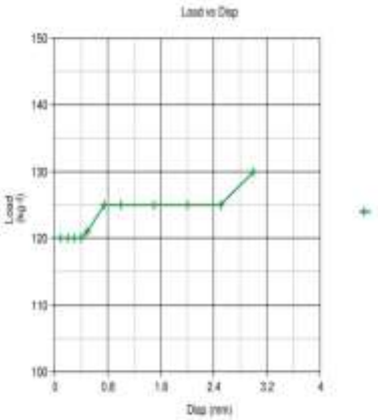

Fig 9: Force v/s Displacement graph at 0.5 Amp current.

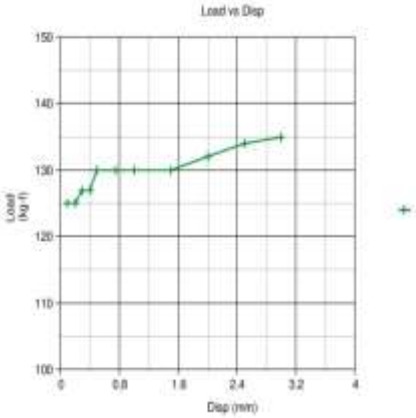

Fig 10: Force v/s Displacement graph at 0.8 Amp current.

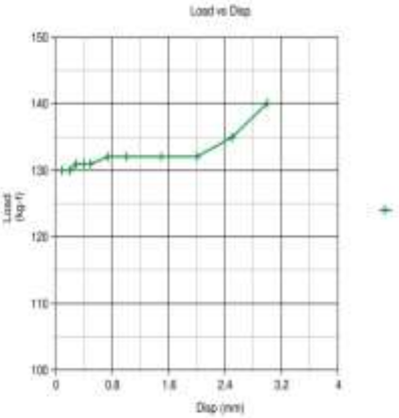

Fig 11: Force v/s Displacement graph at 1 Amp current.

X. CONCLUSION

In this article, a design methodology for MR dampers is proposed so as to improve the performance of the damper when compared to conventional mechanical dampers. The results obtained in this work can be summarized as follows:

1. With the experimental results obtained it can be verified that the performance of MR damper is superior to those of conventional type dampers.
2. By further reducing the cross sectional area of the yoke through which the magnetic flux passes the characteristics of the damper can be improved
3. It is expected that the proposed design methodology can be used as basic material for expanding application fields of the MR damper.

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