Design of a Maganeto – Rheological Damper for Automotive Application

Yogesh S. Pawar, Dr. Lalitkumar M. Jugulkar

Abstract—In the past decades Magneto – Rheological (MR) fluid technology has gained significant development. The application of MR fluids has grown rapidly in civil engineering, transportation with the development of MR fluid based devices, especially MR fluid dampers. The MR fluid dampers could offer very good semi active vibration control, because of its features such as fast response, environmentally robust characteristics, large force capacity, and low power consumption. In this paper, design of a MR fluid damper model is done, that can be used for the semi active control of automotive suspensions. After designing MRF damper it will be integrated to variable stiffness and damping suspension system to get better performance at varying load condition

Keywords—magneto – rheological fluids, design, dampers.

1 INTRODUCTION

Vibration suppression is considered as an important research field in automotive engineering to ensure the safety and comfort of the occupants or the passengers and vehicle itself. An effective vibration control with isolation is necessary, to reduce the system vibration. Compared with the passive suspensions, active suspensions can improve the performance of the suspension system over a range of frequencies. Semi-active suspensions were proposed in the early 1970s, and can be nearly as effective as active suspensions. When control system of semi active suspension fails, still it can work as a passive suspension. Compared with active and passive suspension system, the semi-active suspension system incorporates the advantages of both active and passive suspensions because it provides better performance when compared to passive suspensions and is safe, economical and does not require either a large power supply or higher power actuators as active suspensions do[8]. Semi-active suspensions can be made up of variable orifice dampers or it uses controllable fluids. Generally two types of fluids are uses for development of controllable dampers are: electro-rheological (ER) fluids and magneto-rheological (MR) fluids. Small electro-rheological effects have long been known, but large effects with possible practical applications were first studied by Winslow (1947) [7].

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Passive suspension systems consist of a constant stiffness element and constant damping element. This stiffness and damping element are designed at optimum load condition acting on vehicle. But as the load acting on vehicle varies repeatedly, performance of suspension gets affected. If the load conditions are less than designed load conditions then parameters like comfort and road holding capacity gets affected. So in order to improve the performance with changing load conditions, stiffness and damping coefficient must be vary during their operation. Jugulkaret al.[6] has suggested that the variable stiffness and variable damping as shown in figure 1. It comprises of two variable dampers, one is controlled by varying number of opening in orifice. Another damper can be replaced by MR damper which will give fine tuning to the step described by author.

As shown in figure 1 k1 and k2 are stiffness of springs, c is the coefficient of damper used in variable stiffness, m1 is sprung mass or load on vehicle then equivalent stiffness is given by [6],

$$k_{eq} = \frac{F_0}{z_0} = (k_1 + k_2) - \left(\frac{4\pi^2 f^2 m_1 k_2}{k_2 + 2\pi f c}\right) - \left(\frac{k_2}{k_2 + 2\pi f c}\right)$$

Equation shows that last two terms will be zero if value of c is infinite and of some value of damping coefficient is inserted then this will be significant change in equation stiffness.

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2 Magneto-rheological fluid

A. Maganeto-rheological (MR) techonology

Rheology is the science of the deformation of solids and the flow under stress. Magneto-rheological (MR) liquids have properties dependent on the magnetic field respectively, which is simply a response. The change of viscosity with external magnetic field is the basic feature of MR technology.

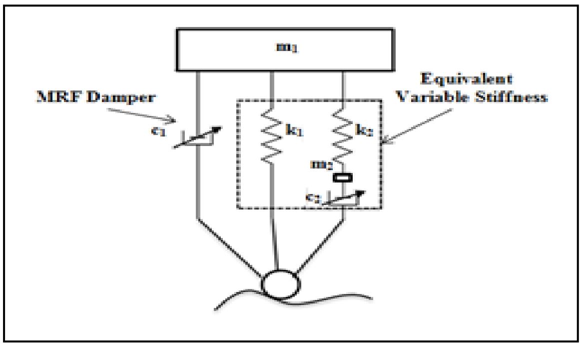

Figure 1: Shock absorber with variable stiffness and damping [6]

The ferromagnetic particles are forced to form a chain like structure by the external magnetic field, which resists the free fluid motion, and with the external magnetic field, the fluid behavior becomes controllable. The MR effect is due to the difference between the rheological properties of the fluid with and without a magnetic field [5].

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B. Magneto-rheological fluids (MRF)

A magneto-rheological fluid is a fluid with rheological behavior which depends on the strength of a magnetic field. The rheological status changes reversibly from liquid to the solid. A conventional damper oil is considered to be a Newtonian liquid, I that it has a simple viscosity, albeit temperature dependent. MR fluid has a yield stress and a post yield marginal viscosity, both dependent on the applied field. Hence it is basically Bingham plastics, characterized by two parameters, the yield shear stress and the subsequent marginal viscosity [8].

C. MRF components

Basically three components are present in the MRF: basic fluid, metal particles and stabilizing additives.

Base Fluid: The base fluid has the function of the carrier and naturally combines lubrication and damping features. There are different types of liquid which can be used as the carrier fluid i.e. mineral oils, silicon oils or hydrocarbon oils. In the off-state i.e., without any magnetic field, the MR fluids behave like the base fluid in accordance with their chemical composition. Because of every type of particle suspended in a MR fluid, the base fluid will be having high viscosity.

Metal Particles: In the on state i.e., with a magnetic field in to form a chainlike structure. The chain like structure restricts the motion of the fluid. Because of this resistance to flow caused by the chain like structure, the MR effect is produced. The metal particles are usually made of carbonyl iron, or powder iron, or iron/cobalt alloys to achieve a high magnetic saturation. The key to a strong MR fluid is to choose a particle with a large saturation magnetization. The amount of metal powder in MRF can be up to 50% by volume [5]. Permeability and size of particle of a particle is important factor for controlling the MR effect.

Stabilizing Additives: The additives contain stabilizer and surfactant. Additives are suspending agents, thixotrops, friction modifiers and anti-corrosion/wear components. To improve settling stability, highly viscous materials such as grease or other thixotropic additives are used. Ferrous naphthanate or ferrous oleate may be used as dispersants [5]. Additives are required to control the viscosity of the liquid, the friction between the particles, the settling rate of the particles, and to avoid the in-use thickening for defined number of off duty cycles.

D. Modes of operation

MRF operate at three different modes depending on the fluid flow and the rheological stresses are: valve, direct shear and squeeze mode.

1) Valve mode

The valve mode as an operational mode is used in dampers, shock absorbers. It is schematically as shown in Figure 2.

In this mode of MRF operation, fluid flow through the two fixed surfaces and magnetic field is applied in a direction perpendicular to the direction of flow. The pressure drop is created in this mode and can be given as, in a damper it is a sum of the viscous (pure rheological) component and the magnetic field dependent (magneto rheological) component [4].

2) Direct shear mode

Direct shear mode is used in clutches and brakes. It is schematically shown in Figure 3.

In this mode, the fluid flows between surfaces having a relative motion between them and a magnetic field is applied in a direction perpendicular to the direction of low. The total force in the shear mode can be separated into a viscous (pure rheological) component and a magnetic field dependent (magneto-rheological) component [4].

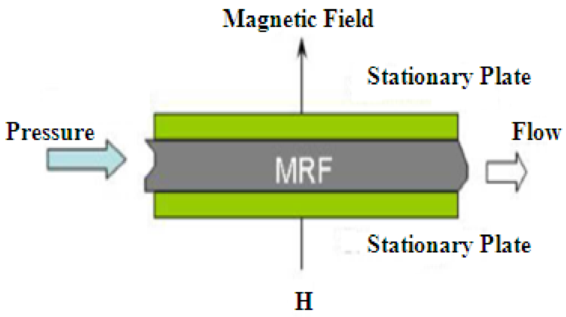

Figure 2: Valve mode [4]

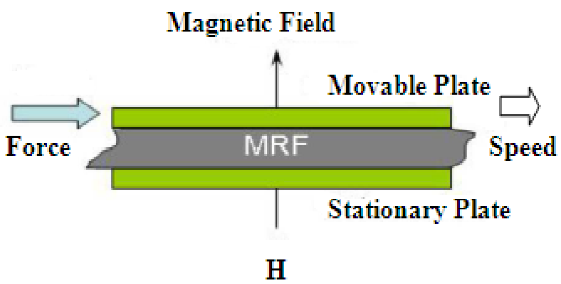

Figure 3: Direct shear mode [4]

3) Squeeze mode

This mode has not been studied so thoroughly comparing with the direct shear mode and the vale mode, but some small amplitude vibration dampers use this mode. It is schematically shown in Figure 4.

This mode is used for low motion and high force application. This operational mode is recently developed mode as compared to the other two. In this mode of MRF technology externally applied force is absorbed with the help of MR fluid. The yield stress developed through this mode is approximately ten times of the stress developed in either valve or shear mode [4].

3 Magneto-Rheological Dampers

When exceeding certain limits, vibrations can cause poor ride quality and stability. This results in severe damage to vehicle elements and/or passengers. Continuous exposure to shocks increases the risk of spinal injury and lower back pain for passengers and drivers. Conventional suspension system designs containing passive dampers, have reached their practical limits towards achieving the optimum compromise between ride comfort and road handling and their characteristics cannot be adapted to varying road profiles. Semi-active MRF dampers, consist of a conventional spring and an adaptive damper. Because of this MRF damper suspension are capable of changing their response on demand. Also provides same performance as active suspensions without high power consumption [8].

E. MRF damper working

MRF dampers have been most widely used for commercial applications among MR devices. Unlike hydraulic dampers, MRF dampers do not require mechanical valves to restrict flow. Instead such a device has an electromagnetic coil incorporated into the piston while the reservoir is filled with the MR fluid. A magnetic field is developed in the angular orifice (as shown in Figure 5) when suitable current is applied. As a result, a yield stress is developed in the fluid as it passes through the flux path. This leads to an apparent increase in fluid viscosity. When MR fluid is used in valve mode, the areas where the MR fluid is exposed to magnetic flux lines are usually referred to as "choking points". The fluid flow of the MR fluid is restricted from one side of the piston to other when the fluid is in the vicinity of the choking points. Varying the magnetic field strength has the effect of changing the

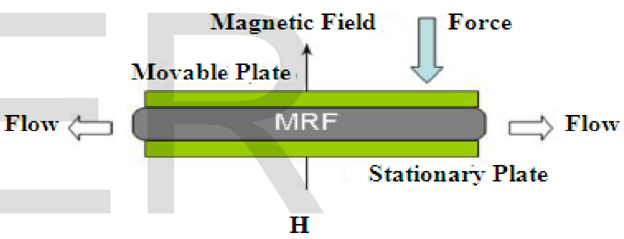

Figure 4: Squeeze mode[4]

apparent viscosity of the MR fluid. Upon exposure to a magnetic field, MR fluid as a whole will appear to have undergone a change in viscosity. As the magnetic field strength increases, the resistance to fluid flow at the choking points increase until the saturation point has been reached. The saturation point is the point where any increase in magnetic field strength fails to yield an increase in damper resistance [9].

4 Design of MRF Damper

Now the focus is on the design of the automotive MRF damper. Similar to conventional viscous damper, it is composed of a hydraulic cylinder. But instead of fork oil it is filled with MR fluid. In this cylinder actuating piston is available to absorb energy, it is an electromagnetic piston. The piston is wrapped in copper wire and forms electromagnetic coil. When the current flows in the coil, a magnetic flux in the piston cylinder is generated and the yield stress value of the MR fluid increases.

F. MRF damper structure parameters

MRF damper's magnetic circuit mainly consist of piston, MR fluid and cylinder body. Hence structural design of piston and cylinder plays an important role in design of MRF damper. Following considerations must be considered while designing the MRF damper.

1) MRF damper piston diameter

MRF damper piston diameter determines the effective area size of the piston. Larger cylinder size means more effective area of the piston; the damper provides greater damping force. Viscous damping force and piston area are in quadratic relationship. So as the piston size increases, viscosity damping force increases more than controlled magnetic force. Hence piston diameter should be selected in the coordination of adjustable damping force between the total damping force.

2) Piston rod diameter

Piston rod diameter also determines the effective area of the piston, the smaller effective area of the piston, the damping force provide smaller. Generally piston rod diameter is determined on the basis of bulking load. It is in the range of 0.3 to 0.35 of cylinder diameter.

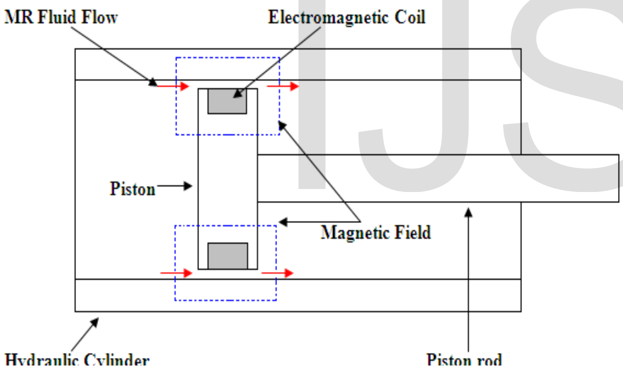

Figure 5: Typical MRF damper

3) Gap size

The structure and size of the damping gap directly affects the damping characteristics of MRF damper. We know that the more far away from the coil, the value of the magnetic field is smaller. Controlled magnetic force will be less. But as gap size is smaller, maximum value of damping force is greater. Also if gap size is reduced, the viscosity damper force of MRF damper increases rapidly, in order to meet the requirements of damper tuning range, damping inner space value should be increased as much as possible. It is generally kept between 0.5 to 3 mm.

4) Piston effective length

More the piston effective length, the damping force also is larger. But as the piston effective length increases, cross sectional area of magnetic circuit also increases; this results in less reluctance also magnetic leakage increases. Therefore, to compensate these effects, gap size also needs to be increases with increase in piston effective length[9].

G. Design of MRF damper

The design procedure of the MR damper involves the following steps.

- Determination of input data and choice of design solution
- Selection of the working MR fluid
- Determination of the optimal gap size
- Magnetic circuit design

1) Input data and choice of the design solution

The input data are defined by a specification and design solutions are chosen from operating and limiting solution.

- Maximum damper controllable force $F\tau$ = 3237 N
- Maximum displacement of MR damper = 0.45m
- Maximum working frequency = 5Hz
- The design conditions are
- Internal diameter of damper cylinder = 0.055m
- Piston rod diameter = 0.016m
- Working temperature = -20 to 150 ˚C

2) Selection of the working MR fluid

In design of MR damper is a magneto-rheological fluid is firstly selected. To ensure the safety of personnel, MRF must be non-toxic, harmless, non-corrosive, high stability, high reliability, less liquid precipitation; the zero viscosity should be low. Most important parameter is price. Price of MR fluid must be lower so that the production of MR damper can be commercialized, market oriented, practical. Here MR fluid of Falcon MR Tech is chosen. It has following specification.

- Product code – AMT-DAMPRO
- Manufacturer - Falcon MR Tech, Thiruvanmiyur, Chennai, Tamil Nadu
- Technical specification –

TABLE I. SPECIFICATION OF MR FLUID

Parameter	*Value*
Appearance	Dark Gray
Liquid Density g/cm^3	2.45 – 2.55

Parameter	Value
Operating Temperature Range 0°C	-20 to +150
Max. Yield Stress @140 kA/m	60 kPa
Viscosity at 0°C Pa-S	0.650
Flash Point˚C	>180
Power Requirements	2-24V at 0.5--2 A

Values from given information can be obtained as

- Magnetic field intensity H = 140 kA/m
- Maximum yield strength τ_y = 60kPa

3) Determination of the optimal gap size

At the time of designing MR damper two key parameter must be considered via Dynamic range and maximum value of controllable force. The controllable force Fτ is the force due to field induced yield stress τy. The dynamic range D is defined as the ratio between the total damper output force F and the uncontrollable force Fuc, where uncontrollable forces include the fluid viscosity force Fη and the friction force Ff.

$$D = \frac{F}{F_{uc}} = \frac{F_{uc} + F_\tau}{F_{uc}} = 1 + \frac{F_\tau}{F_\eta + F_f}$$

Value for forces given by [8],

$$F_\tau = \frac{c\tau_y(H)A_p}{h}L$$

and

$$F_\eta = \frac{12\eta A_{R_m} v_p}{A_g h^2} A_p L$$

So dynamic range becomes

$$D = 1 + \frac{c\tau_y A_p L A_g h}{12\eta A_{Rm} V_p A_p L + F_f A_g h^2}$$

Where,

h – Gap thickness, m

R_m – Gap average radius $=\left(R_p + \left(\frac{h}{2}\right)\right)$, m

R_p – Piston radius, m

A_g – Gap area $=2\pi R_m h$, m^2

A_{Rm} – Area of mean radius $=\pi {R_m}^2$, m^2

L – Length of piston, m

τ_y – Maximum yield stress at applied magnetic field,

□ – Constant, 0.25Pa.s [8]

V_p – Piston velocity, $2\times10^{-2} m/s$[7]

c – Function of the flow velocity profile.

This parameter can be expressed as [8]

$$c = 2.07 + \frac{12\eta A_p V_p}{12\eta A_p V_p + 0.4 A_g h \tau_y}$$

To maximize the effectiveness of the MRF damper, the controllable range of the force should be as large as possible. A small gap size will increase the controllable force range but, when the gap size h is very small the viscous force Fη increases much faster than the controllable force Fτ and dynamic range D decreases. If the gap size h is large, both Fη and Fτ decreases. Hence there exists an optimal gap size which maximizes the dynamic range D. in Figure 6 graph of dynamic range vs. gap size is plotted. From figure it is clear that dynamic range is maximum for gap size 3mm.

4) Magnetic circuit design

The flux in the magnetic circuit flows axially through the piston steel core beneath the windings, radially through the piston pole, through a gap of thickness where the MR fluid flows, and axially through the cylinder wall. The design of a magnetic circuit requires the determination of length of coil, inner diameter of coil and the total number of turns N in the coil.

The design process for sizing the magnetic circuit involves the following steps

- Selection of the operating point of the MR fluid from consideration of given information, H = 140kA/m
- Choice of the steel for the piston and cylinder. Lowcarbon steel (the carbon content of the steel should be less than 0.15%) is selected because of a high magnetic permeability.
- Choice of magnetic pole length. For the MRF damper pole has a length of 1.5×103 m.

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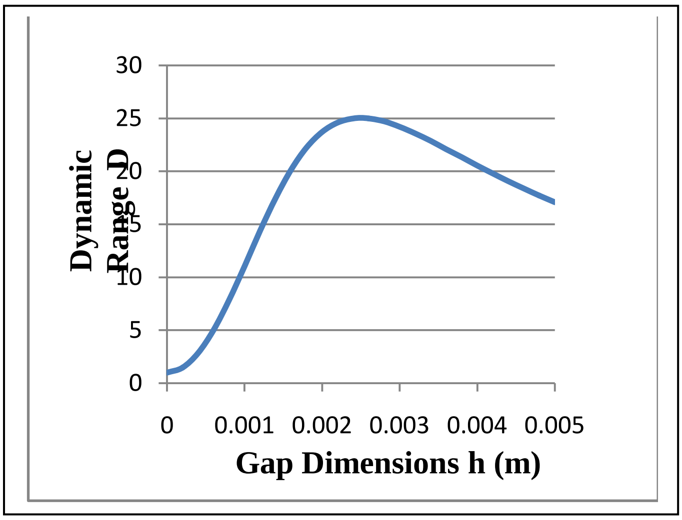

Figure 6: Dynamic range vs. gap dimensions

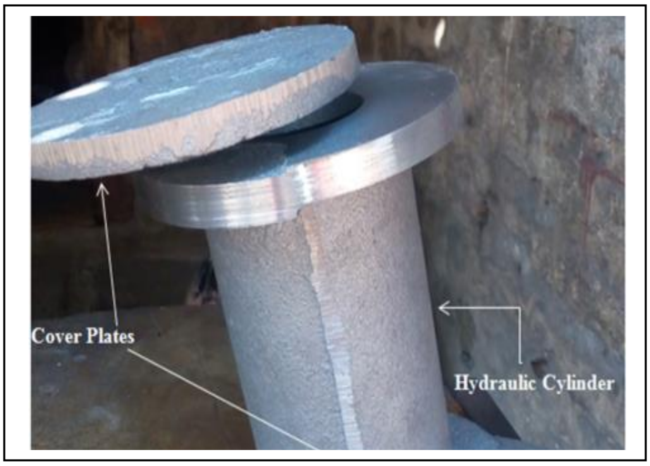

Figure 7: Manufactured MRF damper hydraulic cylinder

- Now

$$H = \frac{I \times N}{L}$$

Where,

I=current in Ampere, 0.5 to 2 A,

N=No. of turns

L = length in m

- This gives N = 40 no of turns of Cu wire with diameter AWG 23 gauge.

In this paper structural dimensions are as follows

TABLE II. STRUCTURAL DIMENSIONS

Parameter	***Value***
Outer Dia. of Cylinder	70mm
Inner Dia. of Cylinder	53mm
Rod Diameter	16mm
Piston Diameter	52mm
Length of Piston	20mm
Length of Cylinder	170mm
Length of Rod	400mm
Gap Between Piston & Cylinder Wall	3mm

As per the dimensions given in the table II, manufacturing of MRF damper is started; some pictures are as follows (Figure 7). Cross section view of the proposed system is shown in figure 8.

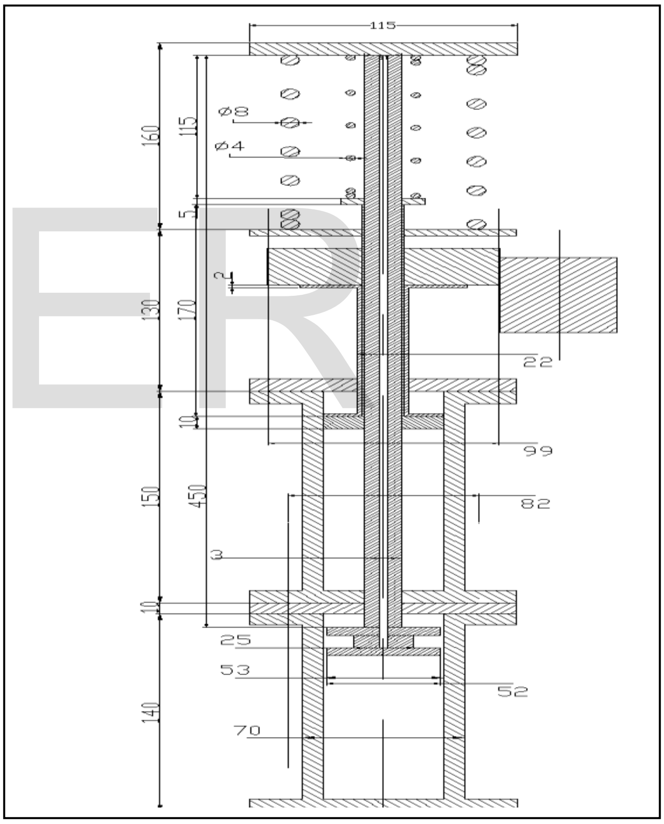

Figure 8: Cross section view of proposed system

5 Conclusion

Future development include an increase inthe acceptable lifespan of MR fluid devices, in terms of the total

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energy dissipated from theequipment throughout its working lifetime and fluids cost must be decrease. This paper introduces a design for vehicle suspension damping system of MR damper. Through discussion on influence of the main parameters on damping force and theoretical calculations provides theoretical basis for design in the future.With the help of this type of suspension arrangement we can achieve better performance of variable stiffness and damping suspension. Total dimensions of variable stiffness and damping suspension are; length 0.6m diameter 0.11m. The components have been manufactured by casting and machining operations like turning boring and honing operations.

REFERENCES

[1] Nishiyama, H., Fushimi, S. and Nakano, M., 2002. Numerical simulation of MR fluid damping characteristics using a modified Bingham model. Journal of intelligent material systems and structures, 13(10), pp.647-653.

[2] Zhang, H.H., Liao, C.R., Chen, W.M. and Huang, S.L., 2006. A magnetic design method of MR fluid dampers and FEM analysis on magnetic saturation. Journal of intelligent material systems and structures, 17(8-9), pp.813-818.

[3] Yang, Y., 2007. Analysis of magnetorheological fluid damper. Journal of Central South University of Technology, 14(1), pp.263-265.

[4] Olabi, A.G. and Grunwald, A., 2007. Design and application of magneto-rheological fluid. Materials & design, 28(10), pp.2658-2664.

[5] Gołdasz, J. and Sapiński, B., 2015. Insight into magnetorheological shock absorbers. Springer.

[6] Jugulkar, Lalitkumar Maikulal, Shankar Singh, and Suresh Maruti Sawant, “Analysis of suspension with variable stiffness and variable damping force for automotive applications”. Advances in Mechanical Engineering 8, no. 5: 1687814016648638, 2016, pp. 1-19.

[7] Dixon, J., 2008. The shock absorber handbook. John Wiley & Sons.

[8] Guglielmino, E., Sireteanu, T., Stammers, C. W., Ghita, G., &Giuclea, M. (2008). Semi-active suspension control: improved vehicle ride and road friendliness. Springer Science & Business Media.

[9] Yang, Z., Wang, H., Han, X. and Fang, W., 2011, August. Damping force of MR damper analysis and experimental. In Electronic and Mechanical Engineering and Information Technology (EMEIT), 2011 International Conference on (Vol. 5, pp. 2528-2531). IEEE.

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