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Principles, Characteristics and Applications of Magneto Rheological Fluid Damper in Flow and Shear Mode

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Abstract

Magneto rheological (MR) fluid has been attracting great research attention because it can change its characteristics very rapidly and controlled easily in the presence of an applied magnetic field. The devices using MR fluid like dampers, clutches, polishing machines, hydraulic valves, etc., have a great promising future. Magneto-rheological (MR) dampers are semi active control devices that use MR fluids to suppress the vibrations. In this paper, the various modes of usage and characteristics are discussed. Mathematical modelingof the MR fluid dampers based on Bingham plastic model and Herschel Bulkley model are presented.

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Key words - MR fluid, MR damper, smart materials and devices

1. Introduction

Magneto rheological (MR) fluids have been investigated by many researchers as their material properties can be modulated through an applied electro- magnetic field. Specially, they are capable of reversibly changing from a linear Newtonian fluid to a semi solid with in a fraction of the milli seconds and the yield strength of this semi-solid is controllable. The fluid causes the maximum yield stress of about 50 to 100kPa. The magnetic field dependent shear strength of MR fluid depends on several factors including the size, composition, volume fraction of the particles and the strength of the applied magnetic field. Systems that take advantage of MR fluids are potentially simpler and more reliable than conventional electromechanical devices. Sodeyama and Suzuki et.al have developed and tested an MR damper which provided maximum damping force of 300kN. Maher YahyaSalloom and Samad designed an MR valve for which simulation was carried out by magnetic finite element software (FEMMR). H.yoshioka, J.C. Ramalloet.al.constructed and tested MR fluid based damper on a base isolated two-degree freedom building model subjected to simulated ground motion which is effective for both far- field and near- field earthquake excitations. Jansen and Dyke evaluated the performance of number of semi active control algorithms that are used with multiple MR dampers. Spancer and Dyke et.al.Proposed a new model to effectively use as semi-active control device for producing a controllable damping force portraying the nonlinear behavior of MR fluid damper.N.Seetaramaaiah and Sadaket.al.have designed a small capacity MR fluid damper which achieved the requirements of dynamic range and controllable force. Lai and W.H Liao have found that MR fluids can be designed to be very effective vibration control actuators. Q.H.Nguyanet.Al.Optimized the dimension of the damper based on minimization of objective function which is

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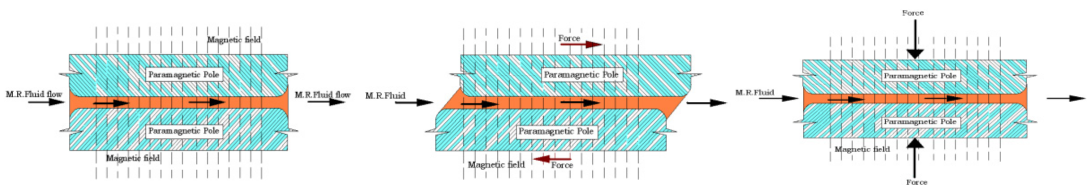
obtained by using Golden-section algorithm and local quadratic fitting technique. Laura M, Jansen and Shirley J. Dykenpresented the results of a study to evaluate the performance of a semi active control algorithms for use with MR dampers.Henri Gavin Jesse Hoagg et.al have made a comparison between ER and MR devices in the context of electrical power requirements.ZekeriyaparlakTahsinet.Al.have carried out the design optimization of MR damper to achieve target damper force by combining finite element methods and C.F.D. analyses of flow simultaneously.The main objective of the paper is to mathematically model the MR damper based on Bingham plastic model and Herschel Bulkleymodel.The quasi-static analysis is performed for the MR fluid flow in the annular duct and the various parameters are evaluated.

Nomenclature

d	spool length	MS	mean squares
d_{cyl}, ,	diameters of the cylinder	V_p	velocity of the piston
e	house thickness	x	Longitudinal coordinator
g	gap of the flow channel between the fixed poles	τ_0	Yield stress due to applied magnetic field
w	width of the flow channel between the fixed poles	τ	shear stress,
m,c,k	fluid parameters m,c, k >0	τ_y	field-dependent yield stress,
A_p	piston cross section area	ΔP_η	viscous component of pressure drop
B_f	Magnetic Flux density	ΔP_τ.	yield stress component of pressure drop
D_1	A constant which depends on boundary Conditions	ΔP	Total pressure drop
F	damper force	$\dot{\gamma}$	shear strain rate
H	magnitude of the applied field	$u_x(r)$	Flow velocity
I	Damper input current	$\tau_{xr}(r)$	shear stress
L	length, of the flow channel between	ρ	fluid density
N	number of turns	η	fluid viscosity
Q	volumetric flow rate	λ	contact ratio
r, r_1, r_2, R, R_1, R_2.	radial coordinates	V	fluid volume
P	pressure of the fluid	F_τ F_{uc} F_f F	controllable, uncontrollable, viscous and friction damper output force respectively

2 .Modes of operation for MR fluid devices

MR fluid devices are utilized in flow mode (valve mode), shear mode (clutch mode), squeeze mode (compression mode) and any combination of these three. Flow mode is the most widely used mode of operation in which the viscosity of the MR fluid flowing between stationary plates is changed by varying the applied magnetic field. In this mode, the activation regions are defined as those areas where MR fluid is exposed to the magnetic flux lines. Examples of devices working in this mode include servo valves, dampers, shock absorbers and actuators. In case of shear mode, MR fluid is present between sliding plates and related with magnetic field which is perpendicular to the direction of motion of these shear surfaces. Examples of shear mode devices include brakes, clutches, dampers, locking devices and structural composites. The distance between parallel plates is changed when they are operated in squeeze mode. In this mode relatively very high forces can be achieved. It is especially suitable for the vibrating dampers with low amplitudes and high dynamic forces. The various modes of operations of MR devices are shown in fig. 1.



(a) valve mode(b) shear mode(c) squeeze mode

Fig.1. Various modes of operation of MR Devices

3. MR Fluid Dampers:

These dampers provide large damping force with low power requirements which can be further classified into following categories.

3.1 Dampers in flow mode:

These dampers generally consist of a cylinder and a piston. The magnetic field is applied to control the flow of MR fluid. These dampers may be either of single end or double end piston rod. These have been applied to the semi active vibration control system for heavy duty vehicle seat suspensions.

3.2 Dampers in the shear mode:

Researchers havedeveloped a MR fluid rotary shock absorber, which can be used for a passenger car suspension system. The damper mainly consists of housing, electromagnet, shaft, blade MR fluid, etc. When the shaft is subjected to torsional vibration then the shear in the MR fluid dissipates the vibrating energy. The system was installed on to the vehicle chassis and tested through real –road travelling; a drastic reduction in body response, namely bounce, pitch and roll was improved. This system model is integrated by Bingham model of the shear mode.

3.3 Dampers in the squeeze mode:

Lord Corporationdeveloped a MR fluid damper working on the principle of the squeeze mode . This damper is being used for active control of damping in numerous industrial applications. The damper functions by moving a disc or baffle in a chamber of MR fluid in which the initial motion is axial and then secondary motion is lateral. When the magnetic field is increased, a transition from a viscous to a viscoelasticbehaviour was observed which a strong influence on the energy had dissipated by the damper and on the damping force.

3.4 Dampers in flow and shear combined mode

In these dampers, an annular gap is provided between cylinder walls and piston. The movement of the piston causes fluid to flow and shear stress exists through the whole annular space. The magnetic coil is wound on the piston or on the inner surface of the cylinder. A damper with a single end piston rod requires volume compensation where as a damper with double ended piston rod does not require volume compensation. Further, the piston is supported by a shaft on both ends which gives good stability of the piston.

4. Mathematical modeling of MR Dampers:

The damper design is done based on the following facts. The mechanical energy required for yielding increases with increase in applied magnetic field intensity which in turn increases yield shear stress. In order to calculate the change in pressure on either side of the piston within the cylinder, yield stress is required which is obtained from the graph of yield stress vs magnetic field intensity provided by Lord corporation for M.R. fluid -132 DG Initially, a quasi static axisymmetric model is developed based on the Navier-Stokes equation for the MR damper. The Herschel-BulkleyVisco-Plasticity model is applied to understand the shear thickening and thinning behavior of MR fluid. Simple equations based on this plasticity model are used during the initial design phase of MR damper. To prove the adaptability of the MR fluid technology for various requirements, a small capacity MR damper has been designed and manufactured. The following assumptions are made for the quasi-static analysis of the damper.The piston moves at a constant velocity and the fluid flow is fully developed.The Herschel-Bulkleyvisco-plasticity model describes the field dependent characteristics and shear thickening/thinning effects.The total shear stress in the Herschel-Bulkley model is given by equation

$$\tau = \left[\tau_0(H) + K\left|\dot{\gamma}\right|^{\frac{1}{m}}\right]\mathrm{sgn}\left(\dot{\gamma}\right) \quad (1)$$

This plasticity model reduces to Bingham visco-plastic model when the fluid parameter ‘m’ approaches unity.Several researchers have tried to develop quasi-static model for controllable fluid dampers by assuming MR fluid as Bingham fluid. Phillips (1969) developed a set of non-dimensional variables and corresponding quintic equation to determine the pressure gradient of flow through a parallel duct. Gavin [10] assumed that the yield stress for MR fluid satisfied an inverse power law which explained the radial field distribution. However, other researchers assumed a constant yield stress in the annular gap. In the following segment, an axi-symmetric model is developed based on Navier-stokes equation for the MR flow through an annular duct assuming Herschel-Bulkleyvisco-Plasticity model. The pressure drop can be obtained numerically from the developed equations and then the damping force can also be evaluated.

4.1 MR Fluid Flow in an Annular Duct

The Navier–strokes equation can be applied to evaluate the pressure gradient existing in the annular region. The fluid shear stress opposes the flow due to pressure gradient which is given by

$$\rho\frac{\partial}{\partial t}u_x(r)+\frac{\partial}{\partial r}\tau_{xr}(r)+\frac{\tau_{xr}(r)}{r}=\frac{\partial p}{\partial x} \tag{2}$$

As quasi-static analysis is being performed, the fluid inertial term vanishes from the above equation and it reduces to

$$\frac{d}{dr}\tau_{xr}(r)+\frac{\tau_{xr}(r)}{r}=\frac{dp}{dx} \tag{3}$$

The above approximation does not hold good for oscillatory or unsteady flow. The solution of eq. 3 is obtained as

$$\tau_{xr}(r)=\frac{1}{2}\frac{dp(x)}{dx}r+\frac{D_1}{r} \tag{4}$$

The variation of shear stress and velocity for the MR fluid flow in the annular gap is shown in figure 2. Theflow field can be divided into three regions. It is observed that shear stress exceeds the yield stress and hence the fluid flows in regions A & B. There is no shear flow of the fluid in region C as the shear stress is less than the yield stress. The region C is frequently referred to as the plug flow region.

4.2Modelling of M R damper based on the Herschel-Bulkley model:

To understand the fluid shear thickening or thinning effect, the Herschel- Bulkleyvisco-plasticity model is employed. In the region A, the shear strain rate $\dot{\gamma}=du_x/dr\geq 0$ .Therefore, the shear stress given by eq. 1 becomes

$$\tau_{xr}(r)=\tau_0(r)+K\left[\frac{du_x(r)}{dr}\right]^{\frac{1}{m}} \tag{5}$$

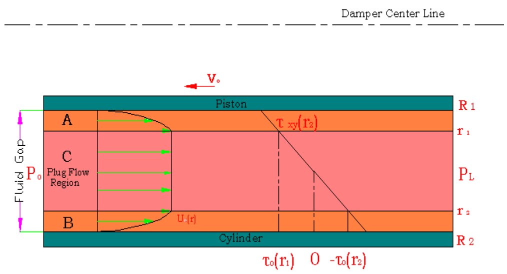


Fig. 2. Velocity and shear stress profiles of MR fluid flow through an annular gap

The above eq.5 is substituted into eq. 4 and integrated with respect to ‘r’ by imposing the boundary condition that the flow velocity at r= R_1 is $u_x(R_1)=-v_p$

$$u_x(r_1)=\int_{R_1}^{r_1}\frac{1}{K}\left[\frac{1}{2}\frac{dp(x)}{dx}r+\frac{D_1}{r}-\tau_0(r)\right]^m dr-v_0 \qquad (R_1\leq r\leq r_1) \tag{6}$$

region C, the shear strain rate $\dot{\gamma}=\frac{du_x}{dr}\leq 0$ thus, the shear stress is given by

$$\tau_{rx}(r)=-\tau_0(r)+K\left[\frac{-du_x(r)}{dr}\right]^{\frac{1}{m}} \tag{7}$$

Proceeding in region C in a similar manner with the boundary condition $u_x(R_2)=0 \; at \; r=R_2$, the velocity inthis region is given by

$$u_x(r_2)=\int_{r_2}^{R_2}\frac{1}{K}\left[\frac{1}{2}\frac{dp(x)}{dx}r+\frac{D_1}{r}+\tau_0(r)\right]^m dr \qquad (r_2\le r\le R_2) \tag{8}$$

It is clear that the flow velocity is constant in the plug flow region because the shear stress is less than the yield stress. Thus, the flow velocities at the boundaries of the plug flow region satisfy the condition, $u_x(r_1)=u_x(r_2)$. Combining eq.6and 8 yields

Also the shear stresses τ_{rx} satisfy $\tau_{rx}(r_1)=\tau_0(r_1)$ and $\tau_{rx}(r_2)=-\tau_0(r_2)$ (9)

therefore D_1 can be determined by using eq. 4as

$$D_1=\frac{r_1r_2(\tau_0(r_2)r_1+\tau_0(r_1)r_2)}{r_2^2-r_1^2} \tag{10}$$

The expression for the volume flow rate Q given by

$$Q=2\pi\int_{R_1}^{R_2} r u_x(r)dr \tag{11}$$

Because shear strain rate, $\left[\frac{du_x(r)}{dr}\right]$ is zero in the plug flow region $r_1<r<r_2$. The above equation modified to

$$Q=v_p A_p=\pi R_1^2 v_p-\pi\int_{R_1}^{r_1} r^2\left[\frac{1}{K}\left(\frac{1}{2}\frac{dp(x)}{dx}r+\frac{D_1}{r}-\tau_0(r)\right)\right]^m dr+\frac{du_x(r)}{dr}dr-\pi\int_{r_2}^{R_2} r^2\left[\frac{1}{K}\left(\frac{1}{2}\frac{dp(x)}{dx}r+\frac{D_1}{r}+\tau_0(r)\right)\right]^m dr \tag{12}$$

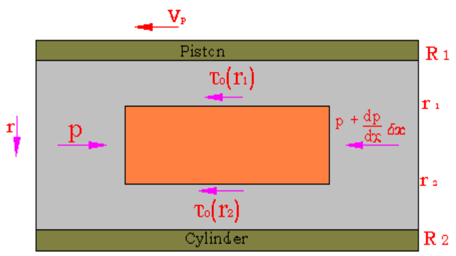


Fig. 3. Free body diagram of MR fluidsthrough an annular gap

The equation of motion of fluid material enclosed by the boundaries $r=r_1$ and $r=r_2$ as shown in the above figure is given by

$$\frac{dp}{dx}\pi(r_2^2-r_1^2)dx+2\pi r_2\tau_0(r_2)dx+2\pi r_1\tau_0(r_1)dx=0 \tag{13}$$

To summarize the discussion, the resulting above equations can be solved numerically to determine r_1, r_2 and the pressure gradient dp/dx between the two ends of the cylinder using the Herschel-Bulkley model. From eq.13 the thickness of the plug flow region can be obtained as

$$(r_2 - r_1) = -\frac{2\left[\tau_0(r_1)r_1 + \tau_0(r_2)r_2\right]}{\frac{dp(x)}{dx}(r_1 + r_2)} \tag{14}$$

which varies with the fluid yield stress τ_0. It can be observed that the flow can only be established when $r_2 - r_1 < R_2 - R_1$,which implies that the plug flow needs to be within the gap. Otherwise there is no flow.The damper force is then computed as

$$F = \Delta P A_p \tag{15}$$

Where

$$\Delta P = P_L - P_O = -L\frac{dp(x)}{dp} \tag{16}$$

The velocity profile can be determined from eq.6 to 8 as follows

$$\left\{\begin{array}{ll} u_x(r) = \int_{r_2}^{R_2} \frac{1}{K}\left[\frac{1}{2}\frac{dp(x)}{dx}r + \frac{D_1}{r} + \tau_0(r)\right]^m dr & (r_2 \le r \le R_2) \\ u_x(r) = \int_{r_1}^{r_2} -\frac{1}{K}\left[\frac{1}{2}\frac{dp(x)}{dx}r + \frac{D_1}{r} + \tau_0(r)\right]^m dr & (r_1 \le r \le r_2) \\ u_x(r) = \int_{r_2}^{R_2} -\frac{1}{K}\left[\frac{1}{2}\frac{dp(x)}{dx}r + \frac{D_1}{r} + \tau_0(r)\right]^m dr & (r_2 \le r \le R_2) \end{array}\right\} \tag{17}$$

Moreover, the shear stress diagram can be obtained from eq.4 ,It can be seen that when the yield stress $\tau_0 = 0$, there is no plug flow region which implies that $r_1 = r_2$ and hence eq.10and 13 are no longer valid due to the singularity . However, in this case, the velocity achieves its maximum at $r = r_1$ where the shear stress is zero. By using eq.4 the following equations can be employed to obtain pressure gradient where yield stress $\tau_0 = 0$

$$Q = v_p A_p = \pi R_1^2 v_p - \pi \int_{R_1}^{r_1} r^2 \left[\frac{1}{K}\left(\frac{1}{2}\frac{dp(x)}{dx}r + \frac{D_1}{r}\right)\right]^m dr + \pi \int_{r_1}^{R_2} r^2 \left[-\frac{1}{K}\left(\frac{1}{2}\frac{dp(x)}{dx}r + \frac{D_1}{r}\right)\right]^m dr \tag{18}$$

$$D_1 = \frac{1}{2}\frac{dp}{dx}r_1^2 \tag{19}$$

4.3. Modelling of M R damper based on the Bingham model:

The Herschel-Bulkley model is converted to the Bingham model when the MR fluid parameter $m = 1$.using eq.9 and 12, the resulting equations for the Bingham model are

$$\frac{dp(x)}{dx}\left[\frac{(R_2^2 - r_2^2 - R_1^2 + r_1^2)}{4}\right] + D_1 In\left(\frac{R_2 r_1}{r_2 R_1}\right) + D_2 - \eta v_p = 0 \tag{20}$$

$$Q = v_p A_p = \pi R_1^2 v_p - \frac{\pi}{8\eta}\left[\left(\frac{dp(x)}{dx}\left(R_2^4 - R_1^4 - r_2^4 + r_1^4\right)4D_1\left(R_2^2 - R_1^2 - r_2^2 + r_1^2\right) + 8D_3\right)\right] \tag{21}$$

$$\frac{dp(x)}{dx}(r_2^2 - r_1^2) + 2\left[\tau_0(r_2)r_2 + \tau_0(r_1)r_1\right] = 0 \tag{22}$$

$$\begin{cases} D_1 = \dfrac{r_1 r_2 \left[\tau_0(r_2) r_1 + \tau_0(r_1) r_2 \right]}{r_2^2 - r_1^2} \\ D_2 = \int_{r_2}^{R_2} \tau_0(r_2) dr + \int_{r_1}^{R_1} \tau_0(r_2) dr \\ D_3 = \int_{r_2}^{R_2} \tau_0(r) r^2 dr + \int_{r_1}^{R_1} \tau_0(r) r^2 dr \end{cases} \tag{23}$$

and the velocity profile is given by

$$\begin{cases} u_x(r) = -\dfrac{1}{4\eta}\dfrac{dp}{dx}\left(R_1^2 - r^2\right) + \dfrac{D_1}{\eta} In \dfrac{r}{R_1} - \dfrac{1}{\eta}\int_{R_1}^{r} \tau_0(r) dr - v_p & R_1 < r < r_1 \\ u_x(r) = -\dfrac{1}{4\eta}\dfrac{dp}{dx}\left(R_2^2 - r_2^2\right) - \dfrac{D_1}{\eta} In \dfrac{R_2}{r_2} - \dfrac{1}{\eta}\int_{r_2}^{R_2} \tau_0(r) dr & r_1 < r < r_2 \\ u_x(r) = -\dfrac{1}{4\eta}\dfrac{dp}{dx}\left(R_2^2 - r^2\right) - \dfrac{D_1}{\eta} In \dfrac{R_2}{r} - \dfrac{1}{\eta}\int_{r}^{R_2} \tau_0(r) dr & r_2 < r < R_2 \end{cases} \tag{24}$$

In the presence of magnetic field, the yield stress $\tau_0 = 0$. The pressure gradient can be obtained directly from

$$\frac{dp}{dx} = \frac{8\eta v_p}{\pi} \frac{\dfrac{\pi}{2}\left[2R_1^2 - \left\{\dfrac{R_2^1 - R_1^2}{In\left(\dfrac{R_2}{R_1}\right)}\right\}\right] - A_p}{\left[R_2^4 - R_1^4 - \left\{\dfrac{R_2^2 - R_1^2}{In\left(\dfrac{R_2}{R_1}\right)}\right\}\right]} \tag{25}$$

In general, the yield stress τ_0 in the axisymmetric model will be a function of r . But when $R_2 - R_1 << R$, the variation of the yield stress in the gap can be ignored, eq.23 can be further simplified substantially as follows:

$$\begin{cases} D_1 = \dfrac{r_1 r_2 \tau_0}{r_2 - r_1} \\ D_2 = \tau_0\left(R_2 + R_1 - r_1 - r_2\right) \\ D_3 = \dfrac{1}{3}\tau_0\left(R_1^3 + R_2^3 - r_1^3 - r_2^3\right) \end{cases} \tag{26}$$

In this case, the thickness of the plug flow can be calculated by using eq.14

$$r_2 - r_1 = -\frac{2\tau_0}{\dfrac{dp(x)}{dx}} \tag{27}$$

Which is constant and depends upon the yield stress and pressure gradient of the flow.

5. Design of MR Damper:

The proposed damper involves both the direct shear mode and pressure driven Valve mode. The fluid flows through the annular gap between the cylinder housing and the piston during the motion of the piston. The above quasi-static analysis of the MR damper is performed by assuming the Bingham's Visco Plastic model.

The flow of the MR fluid is governed by the following Bingham's equations:

$$\tau = G\dot{\gamma} \qquad \tau < \tau_0 \tag{28}$$

$$\tau = \tau_0(H) + \eta\dot{\gamma} \qquad \tau > \tau_0 \tag{29}$$

In the absence of the magnetic field, $\tau \approx \eta\dot{\gamma}$ The pressure drop (ΔP) in an MR fluid device which operates in the flow mode is assumed to be the sum of a viscous component (ΔP_τ) and a field dependent yield stress component (ΔP_η)

$$\Delta P = \Delta P_\eta + \Delta P_\tau = \frac{12\eta Q L}{g^3 W} + \frac{C\tau_y L}{g} \tag{30}$$

The constant c, varies from 2 to 3 depending on the ratio $\left[\frac{\Delta P_\tau}{\Delta P_\eta}\right]$ is present in the device under consideration. When the above ratio is approximately less than or equal to 1, the value for c is chosen to be 2. For theratios of approximately 100 or larger, the value for c is chosen to be 3. The volume of MR fluid exposed to the magnetic field is responsible in providing the desired MR effect and is thus named the minimum active volume V.

$$V = k\left(\frac{\eta}{\tau^2}\right)\lambda W_m \tag{31}$$

where $k = 12/c^2$ is a constant and $V = Lwg$ can be regarded as minimum active fluid volume in order to achieve the desired control ratio 'λ' at a required controllable mechanical power level W_m $(W_m = Q\,\Delta P_\tau)$

eq. 31 can be further manipulated to give

$$wg^2 = \frac{12}{c}\left(\frac{\eta}{\tau_0}\right)\lambda Q \tag{32}$$

This equation provides geometric constraints and the necessary aspect ratios for MR devices based on MR fluid properties, the desired control ratio or dynamic range, and the device flow or speed.MR fluid devices are usually designed such that MR fluid can be nearly magnetically saturated. It is under this condition that the fluid will generate its maximum yield stress, τ_0 . However, the value 'τ_0' that is used in the above equations should be chosen from the MR fluid specifications to reflect the anticipated operating condition.Based on the operational model for pressure driven flow, plotted in figure1, itis possible to determine the effect of geometry on MR damper performance, controllable force and dynamic range 'D' .The damper resisting force comprises of controllable force 'F_τ' due to controllable yield stress 'τ_0' and un controllable force 'F_{uc}' .The uncontrollable force includes a viscous force 'F_η' and a friction force 'F_f' . By definition, the dynamic range is the ratio of the total damper output force 'F' and uncontrollable force 'F_{uc}'.

$$D = 1 + \frac{F_\tau}{F_\eta + F_f} \tag{33}$$

$$F_\eta = \left[A_p v_p + \frac{wgv_p}{2}\right]\frac{12\eta L A_p}{wg^3} \tag{34}$$

$$F_\tau = \frac{c\,\tau_0\,L\,A_p}{g}\,\mathrm{sgn}(v_p) \tag{35}$$

In eq.34 parameter c is bounded in the interval of [2.07,3.07].The controllable force [Eq.6.7] can be rewritten as

$$F_{\tau} = \left[2.07 + \frac{12Q\eta}{12Q\eta + 0.4\,wg^2\tau_0} \right] \frac{\tau_y L A_p}{g} \operatorname{sgn}(v_p) \tag{36}$$

which indicates that the controllable force range is inversely related to the gap size g. To maximize the effectiveness of MR damper, the controllable force should be as long as possible Therefore a small gap size is required .However, a small gap size decreases the dynamic range.The expression of dynamic range 'D' eq.32 can be rewritten using eq.33 and 34

$$D = 1 + \frac{c\,\tau_0 L\, A_p}{\left(A_p + \frac{wg}{2}\right)\frac{12\eta L A_p\, v_p}{wg^2} + gF_r} \tag{37}$$

The quasi-static model developed is useful for designing the MR fluid damper for experimental study.The prototype MR damper is designed for the following requirements:

Table 1: Specifications of the MR Damper

Maximum piston speed	0.2 m/sec
Length of the piston	0.042 m
Stroke	0.024 m
Bore diameter of cylinder	0.04 m
Piston rod diameter	0.012 m
Maximum operating temperature	70°c

The MR fluid used for the designed damper is MRF-132 DG.From figure 6.4 one can obtain the following parameters values:

Table 2: MRF 132DG Parameters

Shear rate '$\dot{\gamma}$'	1000 s^{-1}
Maximum magnetic field H	250 kA/ m
Off-state plastic viscosity 'η'	0.107 Pa-s
Yield stress 'τ_0'	46.5 kPa

The design of MR fluid dampers assumes two main stages 1) Hydraulic circuit design 2) Magnetic circuit design. For the hydraulic circuit design, the gap dimensions 'g' are derived from condition of maximum dynamic range 'D'

The relationship between the various parameters in eq. 36 are :

$$A_p = \frac{\pi\left[\left(d_{cyl} - 2g\right)^2 - d_{sh}^2\right]}{4} \tag{38}$$

$$w = \pi\left(d_{cyl} - g\right) \tag{39}$$

$$C = 2.07 + \frac{12Q\eta}{12Q\eta + 0.4wg^2\tau_y} \tag{40}$$

$$Q = A_p V_p \tag{41}$$

The plot of dynamic range D Vs gap dimension ,g as provided by the fluid manufacturer gives the value of gas 0.0004 m for $D = 65$,

Table 3: Parameters of hydraulic circuit

W	$0.1256m$
Q	$0.2256\times10^{-3}\, m^3/\mathrm{sec}$
ΔP_τ	$14390\times10^3\, Pa$
ΔP	$15903\times10^3\, Pa$
λ	9.51
A_P	$1.128\times10^{-3}\, m^3$
ΔP_η	$1513\times10^3\, Pa$
c	2.948
F	$17943\, N$
v	$0.00211\times10^{-3}\, m^3$

6. Practical design considerations

Commonly, the MR damper piston does not remain centered during operation .This may due to either manufacturer error or side loads due to inappropriate installation (which may result in non uniform temperature increases and local overheating, bearing malfunction and leakage or scratching of the insulation and causing a short in the magnetic coil. To overcome this problem, two end collars made up of bronze are installed on either side of prototype MR damper

Fig. 4.Photograph of the developed MR damper

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