Defence Science Journal, Vol. 69, No. 3, May 2019, pp. 230-235, DOI : 10.14429/dsj.69.14424

Multiphysics Analysis of a Magnetorheological Damper

A.J.D. Nanthakumar[#,*], J. Jancirani[$], S.C. Rajasekaran[@], and K. Sarathkumar[&]

[#]*Department of Automobile Engineering, SRM Institute of Science & Technology, Kattankulathur - 603 203, India*
[$]*Department of Production Technology, Anna University, Chennai - 600 044, India*
[@]*Rambal Limited, Thiruporur - 603 110, India*
[&]*Center for Sponsored Research and Consultancy, Rajalakshmi Engineering College, Thandalam - 602 105, India*
[*]*E-mail: ajd.nanthakumar@gmail.com*

ABSTRACT

A Magnetorheological damping has evolved as a potential tool in vibration control. The design of magnetorheological damping involves analysis of fluid flow principles and electromagnetic flux analysis. This research paper involves design and analysis of a magnetorheological damper employed for vibration control. The analysis is carried over by considering the domain as an axisymmetric model. The damping force of the damper depends upon the shear stress due to fluid viscosity and yield stress induced due to magnetic flux applied. The damping force generated by the damper is calculated.

Keywords: Magnetorheological (MR) damper; CFD analyses; fluid flow; axisymmetric model, Magnetostatics

1. INTRODUCTION

Suspension systems or force isolation systems are employed to control the dynamic effects of vibration phenomenon. The ill-effects of vibration pose a definite threat to the functioning of the mechanical system. The vibration system essentially consists of a vibrating mass component representing the inertia of the system, spring component represented by its stiffness parameter and a shock absorber component represented by its damping co-efficient. Conventional passive vibration control systems have spring and damping equipment with fixed performance characteristics. The requirements of the present-day engineering systems have led the engineers to search for alternate strategies in which the vibration characteristics can be dynamically changed. Engineer's has only two parameters in his hand to regulate – stiffness co-efficient and the damping co-efficient. As stiffness co-efficient is arrived from the spring structural member, altering it would necessitate to replace the existing spring member from the system which is cumbersome and impractical. Thus, the engineer is allowed to act upon the damping co-efficient. This needs to involve a strategy starting from the design and conception of damping systems. As a result, active vibration control systems have been looked upon as an alternate option. Even though the active control systems have theoretically efficient performance, the practical implementation and installation issues make them practically infeasible.

This has led to the deployment of semi active vibration control systems which combines the advantages of active suspension with the low-cost advantage of conventional passive vibration control systems. A number of techniques have been experimented with partial success. Ghaffarzadeh[1], *et al.* have presented a variable orifice damper for building frames. The performance of the damper is varied by means of a small electric motor that adjusts the position of the servo valve spool, which in turn controls the size of a secondary orifice. Hafiz[2], *et al.* have developed an externally controllable orifice damper and verified that decreasing the ratio of orifice opening will produce higher damping force in both compression and extension stages. Asadi[3], *et al.* have proposed a hybrid electromagnetic damper. They have made the viscous medium for bias and fail-safe damping component while the electromagnetic component adds adaptability and variability. Such strategies still have practical difficulties related to installation issues in an existing system.

Magnetorheological damping seems to provide an alternate option with direct control and practically acceptable response characteristics. Winslow[4] have claimed a method and means for translating electrical impulses into mechanical force which had been demonstrated by installing a dielectric fluid mixture between two closely spaced discs. Rabinow[5] has developed a device for transmitting torque, comprising two closely spaced adjacent elements with a mass of contiguous relatively movable discrete paramagnetic particles. These initial works have led to the emanation of electrorheology and magnetorheology which for several decades had been in the domain of pure academic research due to the non-availability of magnetorheological (MR) fluids of acceptable quality. The advancements in material science engineering has made

Received : 31 August 2018, Revised : 27 February 2019
Accepted : 14 March 2019, Online published : 30 April 2019

possible the availability of relatively stable MR fluids. Many industrial agencies[6] have made significant advancements in preparing practical magnetorheological fluids for common use. Commercial deployment of MR dampers includes MagneRide, an automotive adaptive suspension system developed by the Delphi Automotive Corporation and was first installed in the Cadillac Seville STS model and at present used in many models for Cadillac, Buick, and Chevrolet. MR dampers have been retrofitted for the US Army Stryker and HMMWV vehicles known as magnetorheological fluid optimised active damper suspension (MROADS). Recent developments in vibration control of structures and structural components with magnetorheological fluids are revived by Raju[7], *et al.*.

2. MAGNETORHEOLOGICAL DAMPER

2.1 Magnetorheological Fluids

Electrorheological (ER) and Magnetorheological (MR) fluids are a sort of smart fluids whose rheological properties can be controlled by varying the electric filed or magnetic field across the fluid flow direction. Magnetorheological fluids undergo a phase change under the influence of externally applied magnetic flux as depicted in Fig. 1. They are composed of ferro/para magnetic particles (Carbonyl iron or cobalt material) of size of the order of few microns dispersed in a non-magnetic carrier fluid (usually silicone based mineral oil).

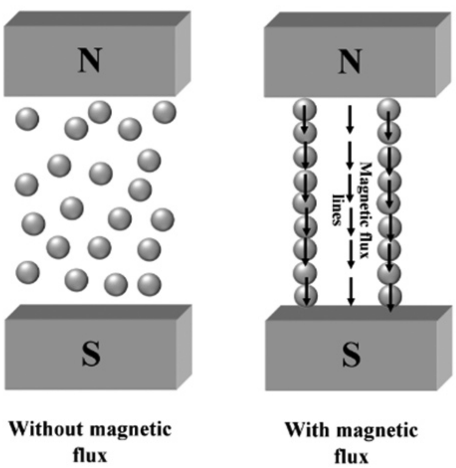

Figure 1. Chain formation in magnetorheological fluid under magnetic flux.

2.2 Modes of Operation of Devices based on MR Fluids

The classification of magnetorheological devices is based on their mode of operation. There are primarily three modes of operation[8] viz., Flow (valve) mode (Fig. 2 (a)), Shear mode, (Fig. 2 (b)), Squeeze mode, (Fig. 2 (c)). Magnetorheological dampers are a sort of flow mode devices, Magnetorheological clutches and brakes work under shear mode of operation, High force small amplitude dampers and mounts come under squeeze mode. Pinch mode[9,10] (Fig. 2 (d)) is comparatively a later discovery which has promising implementation in controllable orifice-like valving systems. Many of the present-day devices combine at least two modes of operation for effective control and hence called as mixed mode devices [11].

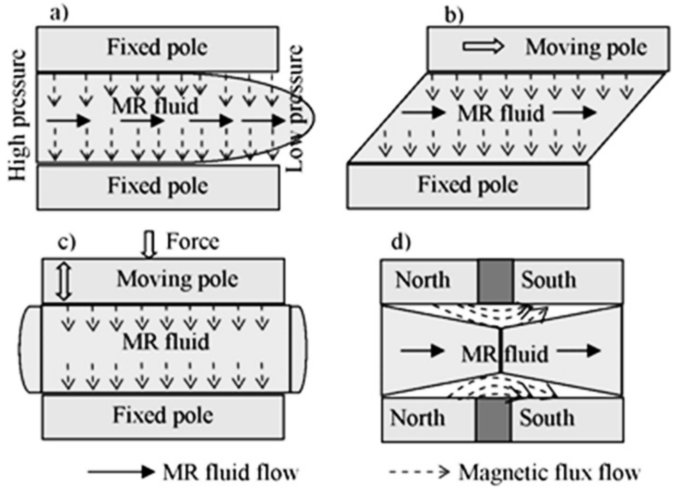

Figure 2. Modes of operation of magnetorheological fluid-based devices.

2.3 Applications of Magnetorheological Damper

Carlson[12], *et al.* have discussed about various devices operating with magnetorheological fluids as working fluids. The automotive industry has been the main beneficiary of the development of magnetorheological fluids-based damper. Apart from dampers fitted in the main suspension system, magnetorheological dampers have also been employed in driver seat suspension systems[13]. Magnetorheological dampers have also found their place in structural vibration mitigation of civil structures[14]. Magnetorheological dampers have also been employed in prosthetic knees[15]. They have also been employed for ride comfort improvement in railway applications. Sharma & Kumar[16] have included the magnetorheological damper as a secondary vertical suspension system to improve ride comfort of rail vehicles. MR dampers have also been employed for the control of cable vibration of cable-stayed bridges[17]. The characteristics of high performance and low energy consumption makes MR dampers suitable for cable stay applications. MR dampers also have made their way into vibration control of landing gears of helicopters and airplanes. Choi[18], *et al.* have developed an adaptive magnetorheological landing gear damper for a helicopter.

3. DESIGN AND ANALYSIS OF MAGNETORHEOLOGICAL DAMPERS

The design of magnetorheological dampers is based on load resistance requirements, size constraints and work configurations. The mode of operation of the magnetorheological fluid in a magnetorheological damper is generally pressure driven flow mode[19]. Mixed mode magnetorheological dampers have also been in vogue[11]. Based on the construction of the hydraulic housing component (cylinder), magnetorheological dampers can be designed as single tube – mono – dampers, twin tube – dual - dampers – and tri tube devices. The first two designs have been adapted widely for effective damping. Mono tube dampers are compact in size and has its own inherent advantages of simplicity and relatively few internal components. Mono tube dampers require a separate chamber for inert gas enclosed by a floating chamber for piston rod compensation and other thermal compensation issues. Twin tube dampers possess two concentric cylinders which have a continuous fluid domain between the inner cylinder and outer cylinder. Twin tube dampers have the distinct advantage of neglecting a separate inert gas chamber which needs very accurate machining with very close tolerance. The twin tube damper can also maintain symmetry in the damping force characteristic than mono tube damper[20].

This paper focusses on the design and analysis of a magnetorheological damper. More emphasis is laid on understanding the multiphysics involved in the working of the magnetorheological damper. For simplicity in design and ease of manufacturability concerns, a twin tube damper is taken into consideration. A single annular orifice is to be embodied in the piston of the magnetorheological damper. The flow of the magnetorheological fluid through the annular flow under the influence of magnetic flux is studied.

3.1 Magnetorheological Damper Structure

A magnetorheological damper essentially consists of a main piston, separating the working fluid into two chambers – compression chamber and rebound chamber. An annular gap is present in between the piston and cylinder. Annular gap is generally preferred for a magnetorheological damper. Fig. 3 represents the piston assembly portion of one such design. Few designs incorporate the annular gap in the piston assembly itself which is followed in the present research work. In such cases, the whole magnetic circuit is incorporated in the piston assembly itself. The elements of the magnetic circuit and the flux path flow in both the cases are similar. The magnetic flux path is shown only for the upper half of the piston. It is understood that magnetic flux flow is also present in the lower half. The dimensions of the proposed damper are as given in Table 2.

3.2 Magnetorheological Damper Design

The configuration as shown in Fig. 3 represents the magnetic circuit for the MR damper. When the current flows in the coil, a magnetic flux is generated which depends upon the

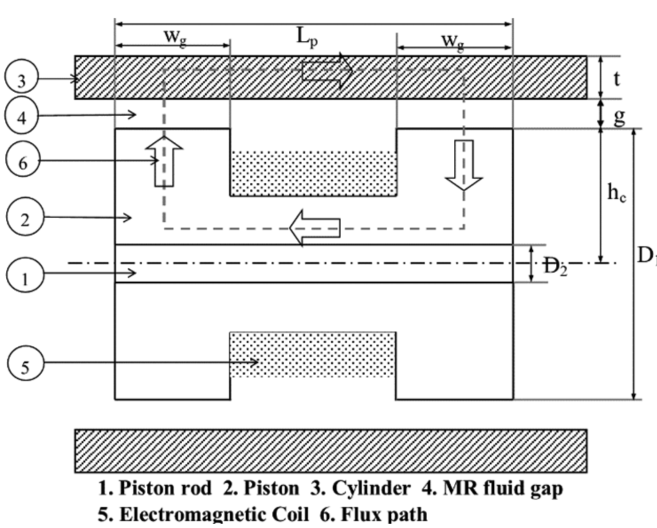

Figure 3. Magnetorheological damper piston geometry.

number of turns of the coil and the magnitude of current flow. The flux thus generated then flows along the axial dimension of the piston. Then it follows the piston profile where piston is made of material with good magnetic permeability. Then it flows through the outer cylinder which is also made of material with high magnetic permeability. While jumping from the inner piston to the outer cylinder, the magnetic flux crosses the MR fluid domain wherein, the MR fluid in the vicinity of the flux flow is magnetised. Due to the magnetisation of the MR fluid in the gap between piston and cylinder, the MR fluid undergoes a phase rheological change.

Table 1. Magnetorheological damper design parameters

Design parameters	Notation
Number of turns of the coil	N
Magnitude of current flow in coil	I
Particle volume fraction of MR fluid	Φ
Pole length	w_g
Gap between piston and cylinder	g

3.3 Design Parameters

The parameters which influence the MR damper capacity is summarised in Table1. The number of turns in the coil and the current flowing through the coil determines the magnetomotive force and is as given by Eqn. (1)[21,22].

$$H = nI \tag{1}$$

where I is current in amperes flowing in the winding of a solenoid having n turns per meter.

Table 2. Magnetorheological damper dimensions

Design parameters	Symbol	Value
Diameter of piston assembly	D_1	0.046 m
Diameter of piston rod	D_2	0.010 m
MR fluid gap	g	0.001 m
Thickness of outer pole / cylinder	t	0.005 m
Height of coil portion	h_c	0.009
Total width of inner poles	w_g	0.030 m
Overall length of MR fluid domain	L_p	0.042 m
Off state viscosity of MR fluid	η	0.042 Pa-s
Area of cross section of piston	A_p	$1.661\text{x}10^{-3}$ m^2
Average circumference of fluid gap	w	0.1445 m

The magnetorheological fluid MRF 122EG supplied by Lord Corporation USA is used for the experimental purpose. The volume fraction of particles in MRF 122EG[23] is 0.22. The volume fraction of the particles in the MR fluid influence the yield stress characteristic under the action of magnetic flux. The relationship between the yield stress (τ_y) and the magnetic flux density (B) for Lord MRF-122EG[24] is as given by Eqn. 2.

$$\tau_y = (6.9\times10^2) + (4\times10^4)B - (1\times10^5)B^2 + (9.1\times10^4)B^3 \tag{2}$$

The MR damper is designed as a two-dimensional axisymmetric model which is as shown in Fig. 4. The following dimensions are taken for the initial prototype design. The length of the coil portion is 0.012 m. The piston diameter is taken as 0.04 m. The gap between the piston and the cylinder (i.e.) inner poles and outer pole is kept at 0.001 m. This research work studies the effect of current flow in the coil in achieving the damping force of the damper. For constructing the electromagnetic coil SWG 35 Copper wire is utilised. The current in the coil is varied from 0.25 A to 2 A in step of 0.25 A. The number of turns in the coil is kept at 300 turns. The pole length of the inner pole is kept at 0.010 m. A finite element analysis of the working domain is initially performed.

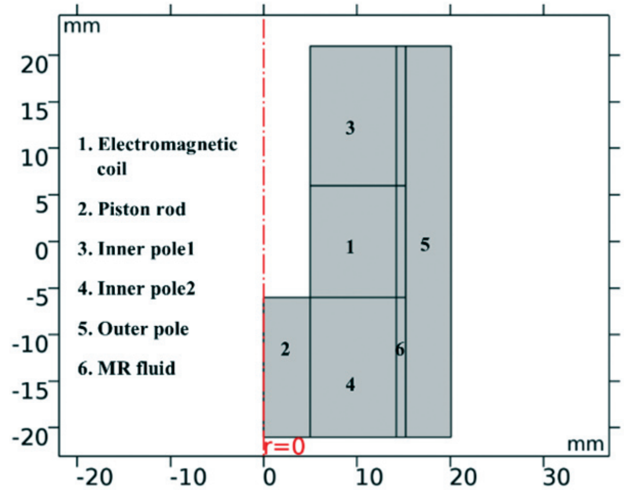

Figure 4. Axisymmetric model of MR damper working domain.

3.4 Finite Element Analysis of Working Domain

The working domain of the MR damper is shown in Fig. 4. Comsol[25] Multiphysics® is used for carrying out FEA analysis. The following steps are used for magnetostatic analysis of the MR damper in COMSOL suite:

(a) *Physics environment creation*: The space dimension is selected as 2D axisymmetric. The problem involves the determination of magnetic field quantities due to the current flow in the coil. Hence the AC/DC interface in COMSOL suite is chosen. Then the physics related to magnetic fields (MF) module is added to the problem study. As the analysis is done under magnetostatic conditions, a stationary study is carried over.

(b) *Domain and material selection*: The whole axisymmetric model is selected as a domain for magnetic fields study. The piston rod is made of Stainless steel and the piston rod portion domain is attributed with Stainless steel 405 material. The electromagnetic coil portion is attributed with Copper solid material. The piston material comprising the inner poles and outer poles is attributed with AISI 1040 steel. The MR fluid gap domain is attributed with the properties of MRF 122EG fluid supplied by Lord Corporation, USA.

(c) *Meshing the domain*: The domain is meshed using the meshing tool with physics-controlled mesh. For the whole domain the maximum element size is 0.00223 m and the minimum mesh size is 0.000126 m. The MR fluid gap is meshed with a maximum element size of 0.00906 m. The interface between the MR fluid gap and the poles is meshed with elements of maximum size of 0.00564 m and minimum size of 0.0000805 m.

(d) *Apply loads and boundary conditions*: The axisymmetric boundary condition is attributed to the center axis line. The outer boundaries of the whole domain is applied with magnetic insulation boundary condition based on the assumption that the flux does not leak out of the working domain. The coil domain is applied with the coil boundary condition of the COMSOL interface with the coil wire cross sectional area of 3.57667 x 10^{-8} m^2 corresponding to Copper wire SWG 35. The coil current is varied from 0.25 - 2 A in steps of 0.25 A. The ampere law domain condition is attributed to the whole domain except coil portion.

(e) *Obtaining the solution*: The computational analysis is done in a computing machine with Intel i7 processor with four physical cores and eight virtual cores with a base processor frequency of 2.3 GHz. The ram capacity of the computing machine is 8 GigaBytes. The COMSOL multiphysics solves the computation with MUMPS solver with a tolerance factor of 1 and residual factor of 1000. The problem is solved for varying the current flowing through the coil 0.25 A to a maximum value of 2 A in steps of 0.25 A.

(f) *Postprocessing the results*: The COMSOL multiphysics suite is capable of plotting the variation of flux density, flux flow lines for the domain as two dimensional surface and three dimensional revolved surface. Figure 5 depicts the flow of magnetic flux lines across the working domain. It is clearly found that the flux lines jump from inner pole to the outer pole through the MR fluid gap. More amount of flux lines is present in the MR fluid gap between inner poles and outer pole. The flux lines are almost absent in the MR fluid gap closer to the electromagnetic coil. Fig. 6 shows the distribution of magnetic flux density in two-dimensional axisymmetric surface and three dimensional rotational volume. The values in the plot correspond to a coil current of 1 A and 300 turns of coil. Similar plots can be obtained for values of coil curent from 0.25 A - 2 A. The magnetic flux density values in the MR fluid domain plotted against the corresponding coil current is as depicted in Fig. 6.

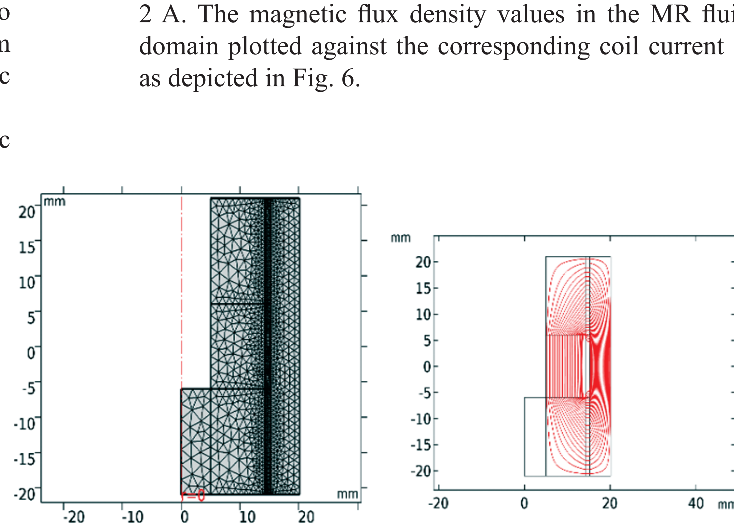

Figure 5. Meshing and magnetic flux lines distribution in MR damper working domain.

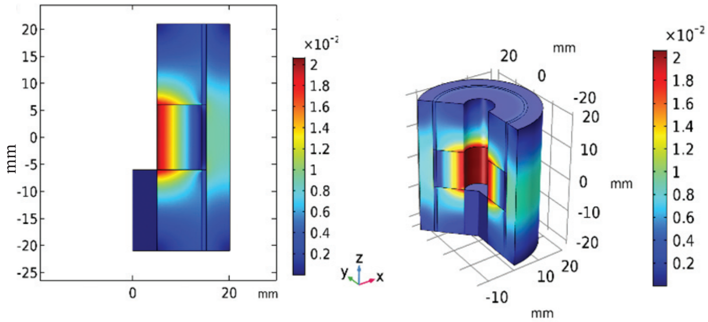

Figure 6. Variation of magnetic flux density in 2D and 3D.

3.5 Application of MR Damper in Combat Vehicles

Ha[28], *et al.* have discussed the application of MR damper in combat vehicles. They have calculated the damping force capacity of a MR damper deployed for military vehicles derived from yield stress of MR fluid influenced by magnetic flux. Thus, the yield stress obtained from Eqn. 2 can be used for calculating the damping capacity of a MR damper. The damping capacity can be varied by changing the number of turns of the coil and the magnitude of current flowing through the coil.

4. CONCLUSIONS

The generation of damping force by a magnetorheological damper depends upon the dimensions of the piston, dimensions of the annular orifice, rheological behaviour of the damper fluid, number of turns of the electromagnetic coil and the magnitude of current flowing through the coil. This research work has determined the magnitude of flux due to current flow in the coil and thus calculated the increase in yield stress due to the change in magnetic flux. Further research work is proposed to be carried out in which the dimensions of the magnetic circuit such as the width of the inner poles can be suitably varied to achieve the required damping force.

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ACKNOWLEDGEMENTS

This research work was supported by University Grants Commission, South Eastern Regional Office, Hyderabad, India under Minor Research Projects scheme: MRP-6103/15 (SERO/UGC). The authors wish to thank M/s. Rambal Limited, Thiruporur division, Thiruporur – 603 110, India for their generous help in testing the MR damper developed.

CONTRIBUTORS

Mr A.J.D. Nanthakumar, has completed BE (Production Engg.), in 2001 and ME (Automobile Engineering), in 2004. He is currently pursuing his PhD in the Department of Production Technology, Anna University, MIT Campus, Chennai, India. His current areas of research are magnetorheological damping, vehicle dynamics and semi-active suspension systems.
His contributions to the present work include parametric study of magnetorheological damper design and multiphysics analysis of the working domain.

Dr J. Jancirani, received her BE (Mechanical Engg.) in 1995 and ME (CAD) in 1997. She completed her PhD from Anna University (MIT Campus) in 2004. Currently she is serving as Professor in the Department of Production Technology, Anna University, MIT Campus, Chennai, India. Her research areas include : Vehicle dynamics, automotive braking systems and CAD.
Her contributions to the present work include design of the magnetorheological damper and analysis of the damper.

Mr S.C. Rajasekaran, received DMIT (Automobile Engineering) from Madras Institute of Technology, in 1977. Currently he is serving as Technical Advisor for Rambal India Ltd., Thiruporur, India. He is having industrial experience in the product design, development of automotive shock absorbers and air suspension systems for vehicles.
His contribution to the present work includes providing technical inputs for the design of magnetorheological damper and testing of the developed prototype damper.

Mr K. Sarathkumar, received his BE (Automobile Engineering) in 2017. He is serving as Junior Research Fellow in the Center for Sponsored Research and Consultancy, Rajalakshmi Engineering College, Thandalam, Chennai, India.
His contributions to the present work include the analysis of the fluid domain of the designed magnetorheological damper.