

MODELING AND DEVELOPMENT OF MAGNETO RHEOLOGICAL DAMPER FOR CHANGING DAMPING SYSTEM

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Abstract

This study presents the design, modeling, fabrication, and experimental validation of a novel magnetorheological (MR) damper incorporating stationary electromagnetic coils wound around the outer surface of the damper cylinder - departing from the conventional piston-mounted configuration. This innovative arrangement overcomes the limitations of localized magnetism and variable wire lengths caused by piston motion, ensuring a uniform magnetic field along the entire piston stroke. The external coil placement also eliminates the need for a hollow piston rod and streamlines electrical routing. Computational fluid dynamics (CFD) simulations were performed to evaluate magnetic field distribution and fluid flow behavior. Experimental tests on a custom - designed rig demonstrated a tunable damping force ranging from 731.54 N (at 0 A) to 1591.47 N (at 0.2 A), corresponding to damping coefficients of 4.65 kNs/m and 10.13 kNs/m, respectively. Compared with conventional viscous dampers, the proposed MR damper offers superior tunability, a broader performance range, and simplified construction. These advantages make it a promising candidate for semi-active suspension systems in automotive, seismic mitigation, and defense applications.

Keywords: MR damper; Damping factor; Damping force; Damping coefficient; CFD

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Introduction

Magnetorheological (MR) dampers are semi-active vibration control devices that use fluids whose rheological properties change in response to an applied magnetic field. These devices offer advantages such as real-time tunability, low energy consumption, compact size, and rapid response - making them suitable for vehicle suspensions, seismic vibration mitigation, and defense systems. In conventional MR damper designs, electromagnetic coils are usually placed on the piston, introducing several limitations: Variation in wire length due to piston movement, complicated electrical connections and reliability issues, Non-uniform magnetic field intensity, and the need for a hollow piston to accommodate wiring.

This study proposes a novel MR damper design where the electromagnetic coils are wound externally around the cylinder, leaving the piston magnet - free. This configuration enhances magnetic field uniformity, improves simplicity of construction, and eliminates the Need for internal wiring.

Objectives

- Development and modeling of a MR damper considering the external magnetic field from outside of the cylinder.
- Fabrication and experimentation of the MR damper considering the external magnetic field from outside of the cylinder.
- Computational Fluid Dynamics Analysis of the fabricated MR damper.

Literature Review

The MR damper model was created by Yun-Juu Nam *et al.* (2008). There were upper and lower sections to the cylinder. A simple MR damper design is based on many assumptions, i.e., velocity distribution inside the cylinder is linear, the frictional force is minimal, and the MR ferromagnetic particles are uniformly dispersed throughout the field. The MR damper with an annular gap was mathematically modelled by Wang *et al.* (2003). The authors also studied the characteristics of the MR fluid as it passes through the annular gap and performed numerous computations to determine the ideal gap size. Yao and (2002) associates constructed a flow-mode MR damper in 2002. They concluded that as the strength of the magnetic field grows, so does the damping coefficient. Dong *et al.* (2009) state that magnetorheological elastomers, or MREs, were

used to test the MR damper's reactivity. The author developed the mathematical model for the MRE damper. The authors also mentioned the correlation between the shear modulus and the magnetic field density. Shuaishuai *et al.* (2017) developed an MR damper with tunable stiffness and damping. The authors plotted the results of the analysis on multiple graphs after several calculations. Real-time modeling calculations on non-parametric semi-active suspension models, which do not account for assumptions, have been provided by Metered *et al.* (2012). They used graphics to illustrate the semi-active suspensions' non-linear behavior. Metered *et al.* (2013) used the trial-and-error method to create a mathematical model representing the behavior of MR dampers. In order to estimate the damping force, the model responses must be quick and accurate. A cylindrical MR seat damper was created by Choi *et al.* (2000) using the Bingham model of the MR fluid. The seat damper's field-dependent damping force properties are experimentally assessed following production. A closed-loop control method was used to develop and implement a skyhook controller that lessens vibration at the driver's seat. The control responses-like acceleration transmissibility-are examined in the frequency and time domains. Ordinary differential equations (ODEs) of a physical MR damper were obtained by Guan *et al.* (2011) while considering the compressibility of the MR fluid. The relevant lumped parameter model is then created, and a quasi-static MR model that expresses the compression of MR fluid is coupled in series with a spring. An expression for hysteresis breadth is obtained by ignoring viscous components, and a dynamic design approach for MR dampers is suggested. Ahmadian and Norris (2000) conducted an experimental analysis of the MR damper when the MR damper was subjected to impact and shock loading. A drop tower is developed for this experimental analysis, which applies an impulse load. This drop tower guide mass was dropped from different heights, i.e., 12, 24, 48, 72, and 96 inches, to achieve different impact forces. The two MR damper configurations were tested, i.e., a single-stage, double-ended piston and a mono-tube damper with a two-stage piston. Also, in the case of a mono-tube damper with a two-stage piston, a theoretical model was derived and compared with the experimental data to study nitrogen accumulator behavior. The single-piston rod MR damper was created by Tu *et al.* (2012).

The magnetic circuit and structural design were considered when optimizing the structural damper parameters. With ANSYS finite element software, the magnetic circuit was analyzed using the finite element approach. Additionally, the formula for determining the MR damper damping force was developed. The MR damper fatigue test machine was used to evaluate the sample of the vehicle suspension. The fatigue testing equipment applied a frequency of 1 HZ, 1.5 HZ, and 2.0 HZ with a vibration amplitude of 32 mm. The data for the experiment were gathered at 0.1 Amp intervals. Kim *et al.* (2016) developed a magnetorheological (MR) damper with a bifold flow mode gap to improve damping performance. They also developed a mathematical model for the new damper. The investigation's conclusions showed that the MR damper with the bifold flow mode gap outperformed the conventional one in terms of effectiveness. The proposed MR damper's finite element analysis is performed using ANSYS software. The standard double-coil MR damper was proposed by Hu *et al.* (2016). The MR damper's double coil architecture allows for a wide range of damping force adjustments by controlling the quantities and orientations of the currents given (2008) to the double coil. A finite element analysis of the damper was conducted to determine how the damper piston's geometry impacts the MR damper's performance. Seven piston configurations were created for examination based on the quantity and form of chamfered ends. The impact of the current direction in both the same and reverse directions on the damper's performance was also investigated during experimentation. The magnetic design of the MR damper was proposed by Zhang *et al.* (2006), and it provides a way to analyze the potential saturation zone in the magnetic loop and apply the MR damper design. With the use of ANSYS finite element software, the saturation of the magnetic loop and magnetic flux density in the magnetorheological fluid gap are anticipated. The MR damper is experimentally verified on a test bed for electric hydraulic servos. The MR damper's ergogram was recorded at 0.1 A and 0.2 A intervals under various applied currents of 0-1 A and 0-2 A. Yang *et al.* (2007) looked at the damping that developed in an MR damper under different damping conditions. The structure and theoretical analysis of the design process of a novel MR fluid damper are presented. The damping equation by MR fluids within the damper is derived to provide the theoretical foundations for the damper design. The volume, thickness, and width of the annular MR fluid design are determined when the mechanical power level, piston speed, and damping

ratio are specified. Olabi *et al.* (2007) discuss the state of the art of an actuator with a control arrangement based on MRF technology. The study shows that excellent features like a fast response, the simple interface between electrical power input and the mechanical power output, and controllability make magnetorheological fluid the subsequent technology of choice for many applications. Simplicity and more intelligence in functionality are critical features of magnetorheological fluid technology. Direct shear mode (used in brakes and clutches) and valve mode (used in dampers) have been studied. Semi-active suspension systems employ advanced damping components (Guglielmino *et al.* 2008) that consume minimal energy and can adapt instantly to the vehicle's ride and handling characteristics. This study focuses on magnetorheological and controlled-friction dampers, demonstrating their effectiveness through a balanced approach of mathematical modelling and experimental validation. A wide range of applications is discussed, including seat suspensions, car suspensions, and heavy-vehicle tyre-load control, all of which highlight improvements in comfort, handling, and reduced road damage. The objective of this work is to evaluate the dynamic performance (Elattar *et al.*, 2016) of a quarter-vehicle active suspension system. A mathematical model and computer simulation were developed for an electrohydraulic servo system controlled using PID and PDF controllers. The controller parameters were tuned based on ISE and ITAE criteria, and the results showed that the PD controller achieved the shortest settling time, the PDF controller produced negligible overshoot, and the PID controller maintained overshoot within 5%. Chen *et al.* (2012) propose a polynomial function supervising fuzzy sliding mode control (PSF α SMC) with a skyhook surface method to enhance ride comfort in vehicle semi-active suspensions. The controller parameters are optimized offline using a multi-objective microgenetic algorithm and applied online for real-time control.

Magneto Rheological Damper Design

Design Calculations

The design parameters were determined based on the required damping force range for a passenger car application (700-1600 N). Key governing equations included those for force, magnetic field intensity, damping coefficient, and Reynolds number. The primary variables used in this study are summarized in Table 1.

Table 1. The variables used in the research study

Variable	Description	Unit
Fd	Damping force	N
C	Damping coefficient	KNs/m
v	Piston velocity	m/s
μ	Dynamic viscosity	$Pa.S$
H	Magnetic field intensity	A/m
B	Magnetic flux density	T
I	Current supplied to coil	A

Assumptions

- The MR fluid is assumed to exhibit uniform, steady, and laminar flow.
- A radial magnetic field distribution is considered across the piston gap.
- MR fluid properties are assumed constant, based on the manufacturer's provided data.

Selection of MR Fluid - The selection of MR fluid was based on key factors such as stability, durability, operating temperature range, and compatibility with damper materials.

MR fluid dampers are typically designed to ensure magnetic saturation of the fluid under normal operating conditions. For this study, the AMT-DAMPER fluid from Falcon MR Tech (Chennai, Tamil Nadu) was selected, as it achieves maximum yield stress under these conditions. Its technical specifications relevant to the MR damper are presented in Table 2. From the provided data, the maximum yield strength of the MR fluid is 60 kPa, and the magnetic field intensity is $H = 140$ kA/m.

Table 2. Technical Specifications

Sr. No.	Parameter	Value
1	Outer Dia. of Cylinder	70
2	Inner Dia. of Cylinder	58
3	Rod Diameter	16
4	Piston Diameter	52
5	Thickness of Piston	20
6	Length of Cylinder	170
7	Length of Rod	560

From the provided data, the maximum yield strength of the MR fluid is 60 kPa, and the magnetic field intensity is $H = 140$ kA/m.

Magnetic Circuit Design - In this design, the pole length refers to the distance between the two poles of the electromagnet, which is 50 mm, corresponding to the piston stroke.

The magnetic circuit sizing process involves the following steps:

1. Selection of the Optimum Magnetic Field - The target magnetic field intensity is set at $H =$

140 kA/m, to achieve the maximum electromagnetic effect.

2. Material Selection - Steel was chosen for the cylinder and piston due to its high magnetic permeability. Specifically, low-carbon steel with a carbon content below 0.15% was used for enhanced magnetic performance.

3. Pole Length and Coil Winding - The prototype MR damper pole length is 0.050 m. An initial design using AWG 18 wire (diameter per standard wire gauge) would require $N = 3500$ turns of copper wire. However, accommodating 3500 turns within a 50 mm pole length proved impractical.

4. Wire Gauge Selection - Based on practical considerations and expert input, AWG 28 wire was selected. This gauge can carry up to 3 A of current and allows the required number of turns to be wound within the available space.

The final calculated dimensions of the MR damper are presented in Table 3.

Table 3. Dimensions of a Designed MR Damper

Sr. No.	Parameter	Value
1	Appearance	Dark Gray
2	Liquid Density g/cm ³	2.45 – 2.55
3	Operating Temperature Range 0°C(0°F)	-20 to +150 (-4 to +302)
4	Max. Yield Stress @ 140 kA/m	60 kPa
5	Power Requirements	2-24 V at 0.5 – 2 A

CFD Simulation and Analysis**Magnetic Field Simulation**

A 2D axisymmetric model of the MR damper was developed and simulated in *ANSYS Maxwell* to evaluate the magnetic flux distribution. Simulations were performed at coil current levels of 0.10 A, 0.15 A, and 0.20 A. The results showed that:

- The magnetic flux density reached approximately 0.18 T near the fluid flow gap.
- The magnetic field remained uniform across the gap due to the fixed coil geometry, ensuring consistent activation of the MR fluid.

Flow Regime and Reynolds Number

The Reynolds number was calculated to confirm the flow regime within the MR damper. For piston velocities up to 0.3 m/s, the Reynolds number remained below 1200, indicating laminar flow conditions. This validated the applicability of linear damping models for subsequent performance analysis.

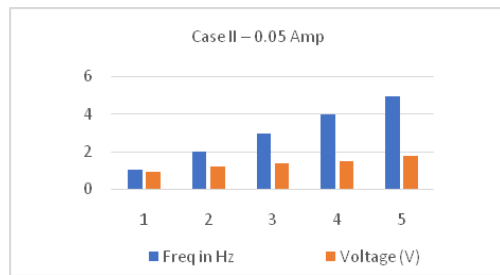


Figure 5. Voltage Reading at 0.05 Amp Supply Current

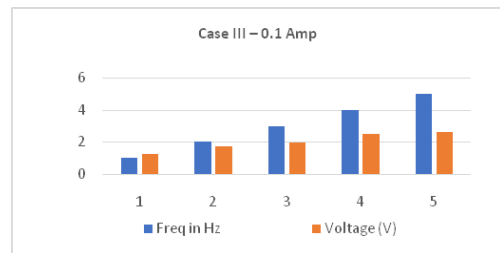


Figure 6. Voltage Reading at 0.1 Amp Supply Current

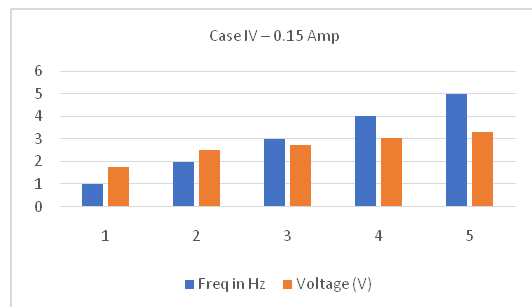


Figure 7. Voltage Reading at 0.15 Amp Supply Current

Experimental Setup and Testing

Prototype and Materials

The MR damper prototype consisted of the following specifications:

Cylinder outer diameter: 60 mm
 Piston diameter: 25 mm
 Stroke: 80 mm
 Coil turns: 1900
 Wire gauge: 34 SWG

The coil was wound externally around the cylinder, properly insulated, and energized using a variable power supply. This design eliminated the need for internal wiring.

Test Rig and Procedure

A custom-built test rig was used to simulate a single-degree-of-freedom suspension system (Jugulkar *et al.* 2016). The test rig is shown in Figure 3. Experiments were carried out at an excitation frequency of 5 Hz, with the input current to the MR damper varied from 0 A to 0.2 A. The damping force and damping coefficient obtained for each current level are presented in Table 4.

Table 4. Experimental Results of Damping Force and Damping Coefficient at Various Current Levels

Current (A)	Min Force (N)	Max Force (N)	Damping coefficient (KNs/m)
0.00	731.54	731.54	4.65
0.05	854.36	1002.17	5.47
0.10	1067.45	1214.83	6.63
0.15	1286.57	1419.85	8.26
0.20	1412.41	1591.47	10.13

Experimental Outcomes

Figure 8 illustrates the characteristic curve of the designed MR damper. By using sensors and an electronic control unit to predict the vibration frequency and the required damping force, the current supplied to the electromagnet can be adjusted accordingly. This enables the damper to achieve the desired damping performance in real time.

Acceleration Testing

The readings are presented in Figure 9, which illustrates the variation of acceleration with frequency for different current values. For a given current, the acceleration increases steadily as the frequency rises up to 4 Hz. Beyond this point, at 5 Hz and 6 Hz, the acceleration shows a sudden rise followed by a drop at frequencies above 6 Hz. The frequency range between 5 Hz and 6 Hz corresponds to the natural frequency of vibration for the current prototype.

Computational Fluid Dynamic Analysis of the fabricated MR damper

Applicability in Military and Seismic Systems

The proposed damper's robustness, fail-safe operation, and simplicity make it suitable for:

Military vehicles: Operates even without active current. Seismic base isolators: Offers real-time response and long-term reliability. High-durability environments: Minimal moving parts reduce failure points.

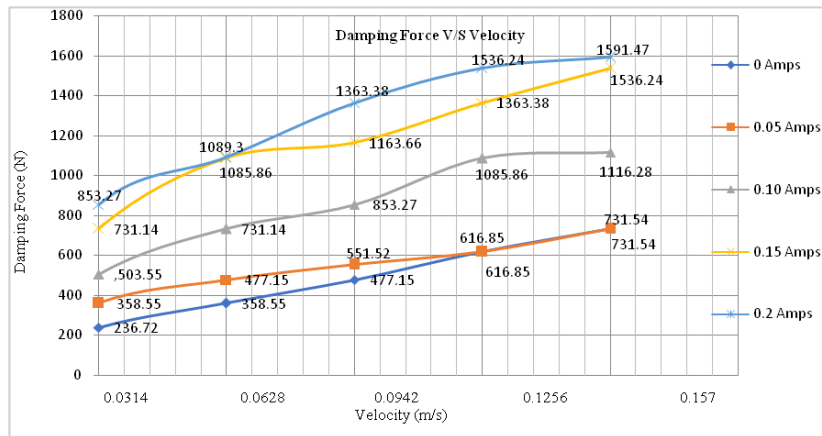


Figure 8. The designed MR damper's charecteristics curves

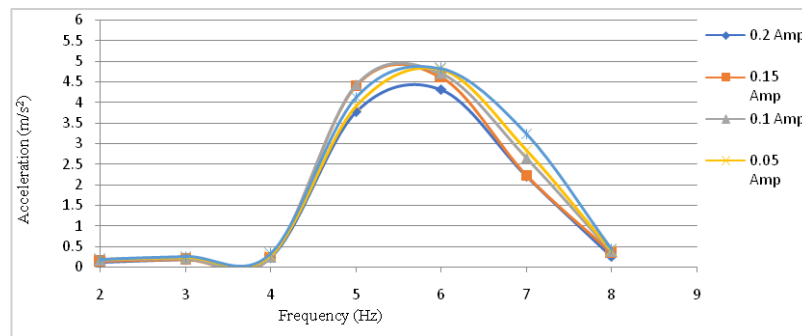


Figure 9. The variation in the accelration with respect to excitation frequency

Calculation of Viscosity

The viscosity values, which vary with changes in current, are critical (Dixon, 2008) inputs for the CFD analysis. However, the viscosity of MR fluid cannot be directly measured using a standard viscometer, as the fluid must be energized with current to alter its viscosity. This posed a challenge in obtaining accurate viscosity data. To address this, a custom setup based on the basic principle of a viscometer was developed, and the values were determined experimentally.

$$\mu = \frac{2}{9} \times \frac{V^2 g (\rho_p - \rho_f)}{\mu}$$

Where,

ρ_g = density of ball material i. e. steel in kg/m³

ρ_f = density of fluid in kg/m³

V = velocity of the ball (m/s)

g = gravitational acceleration (m/s²)

μ = dynamic viscosity of fluid (Pa.s)

Principle of the Viscometer

A viscometer typically operates on a simple principle, illustrated in Figure 10. It consists of a container filled with oil and a steel ball. When the steel ball is released from the top surface of the oil, it falls through the fluid at a certain velocity before settling at the bottom. This velocity is inversely proportional

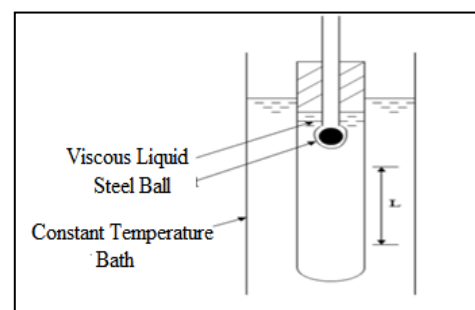


Figure 10. Principle of viscometer

to the oil's viscosity - meaning the more viscous the oil, the longer the ball takes to reach the bottom. The viscosity of the oil is then calculated using the following formula, which represents the fundamental principle on which the viscometer operates.

Experimental Set Up for Viscosity Measurement

The experimental setup consists of a steel ball of diameter 11mm, MR fluid, a damper cylinder, and a power supply. The MR fluid is poured into the cylinder. The steel ball is tied with a string and left in a fluid. A marking was done on the string so that one could know when the steel ball reached the bottom. The viscosity measurement setup is shown in Figure 11.



Figure 11. Viscosity measurement setup

Procedure for viscosity measurement:

- MR fluid was poured into the damper cylinder to a height of 120 mm.
- The power supply was switched off to record readings for 0 A current.
- A string, to which the steel ball was tied, was marked at a length of 98 mm (calculated as 120 mm – 22 mm, where 22 mm is the diameter of the steel ball). This ensured that when the marking aligned with the fluid surface, the ball would be exactly at the bottom of the fluid.
- The ball was released from the surface of the fluid, and the time taken to reach the bottom was measured.
- Using the known travel distance and measured time, the velocity of the ball was calculated.
- Applying the viscometer principle for the first case (0 A):
- The power supply was then switched on, and a current of 0.05 A was applied. The same procedure was repeated to calculate the viscosity. This process was carried out for current values ranging from 0 A to 0.2 A in

increments of 0.05 A. The resulting viscosity values are presented in Table 5.

- Radius of ball: 11 mm = 0.011 m
- Time (t): 2.71 s
- Height (h): 120 mm = 0.12 m

Therefore,

$$V = \frac{0.12}{2.71} = 0.044 \text{ m/sec}$$

Now,

Using the principle of viscometer,

$$V = \frac{2}{9} \times \frac{R^2 g (\rho_p - \rho_f)}{\mu}$$

$$0.044 = \frac{2}{9} \times \frac{0.011^2 \times 9.81 \times (8050 - 2500)}{\mu}$$

$$\mu = 33.56 \text{ Pa.s}$$

Table 5. Viscosity values

Sr. No.	Current (Ampere)	Time (Second)	Dynamic Viscosity (Pa.s)
1.	0.00	2.71	33.56
2.	0.05	2.99	36.59
3.	0.10	3.55	43.31
4.	0.15	4.04	49.29
5.	0.20	4.35	53.23

CFD Analysis and results

The primary objective of the CFD analysis was to study the fluid flow characteristics within the MR damper. Since the design equations for the MR damper assume linear behavior, it was essential for the flow to remain laminar. If the flow became turbulent, nonlinear equations would be required for accurate design calculations.

The CFD process was carried out in the following steps:

- Defining the Modeling Goals Before starting the analysis, the objectives and requirements of the CFD study were clearly defined. This included specifying the modeling goals, identifying physical models, determining boundary conditions, defining the computational domain, and establishing the required accuracy. Assumptions and simplifications were also listed to make the model practical for computation.
- Creating the Model Geometry Based on the identified assumptions and simplifications, the geometry of the MR damper was developed. The CAD model was either created using external CAD software and imported into the

CFD environment or directly modeled within the CFD package's geometry builder.

- Defining and Creating the Mesh Meshing was performed to discretize the model for simulation. Since mesh quality greatly affects simulation accuracy, particular attention was given to regions with steep velocity or pressure gradients. The swept mesh approach was used, balancing accuracy with the available computational resources.
- Setting up the Solver and Physics The solver type (density-based, pressure-based, steady, or unsteady) was selected according to the problem requirements. Appropriate physical models - such as turbulence modeling or multiphase flow - were chosen as needed. Material properties, operating conditions, and boundary conditions were defined. Convergence criteria were also set, and the flow field was initialized before computation.
- Monitoring the Solution During the simulation, key variables were monitored to track convergence. The solution was considered converged when variable fluctuations became negligible over successive iterations.
- Evaluating the Results Post-processing tools were used to extract meaningful results, including velocity fields, pressure distribution, flow patterns, eddy formations, and other relevant parameters. Visualization aided in understanding the flow characteristics within the damper.

- Revising the Model The results were reviewed to check the suitability of the selected solver, physical models, and mesh quality. If discrepancies were found, the mesh density, boundary conditions, or physical models were adjusted, and the analysis was repeated.

Results of the CFD

The analysis was conducted for maximum acceleration across five current settings: 0 A, 0.05 A, 0.1 A, 0.15 A, and 0.2 A. The CFD provided the key outputs, i.e., Velocity distribution at various sections of the cylinder, Pressure distribution along the cylinder, Flow behavior (laminar/turbulent nature), Eddy formations and regions of recirculation.

The section at the exit of the orifice was identified as the most critical for velocity measurements. Velocity values for maximum acceleration were recorded for all five cases, with currents varied from 0 A to 0.2 A in 0.05 A increments. These values are presented in Table 6, while Figures 12-16 show the corresponding CFD plots of velocity at the orifice exit.

Table 6. Flow velocity at orifice exit for maximum acceleration

Sr. No.	Current (Ampere)	Velocity (m/s)
1	0.00	8.82
2	0.05	8.53
3	0.10	8.50
4	0.15	8.48
5	0.20	8.40

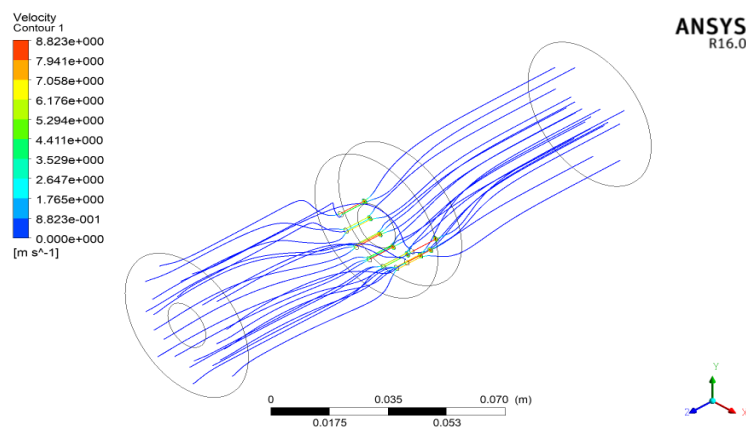


Figure 12. The flow velocity at the orifice exit when the current is 0 amperes and the velocity is 8.82 m/s

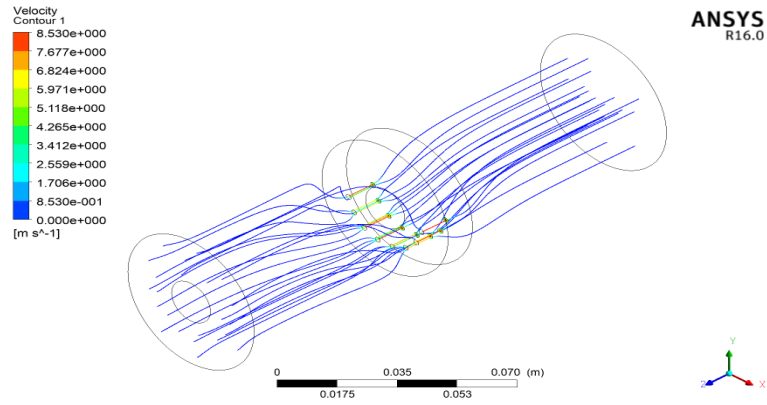


Figure 13. The flow velocity at the orifice exit when the current is 0.05 amperes and the velocity is 8.53 m/s

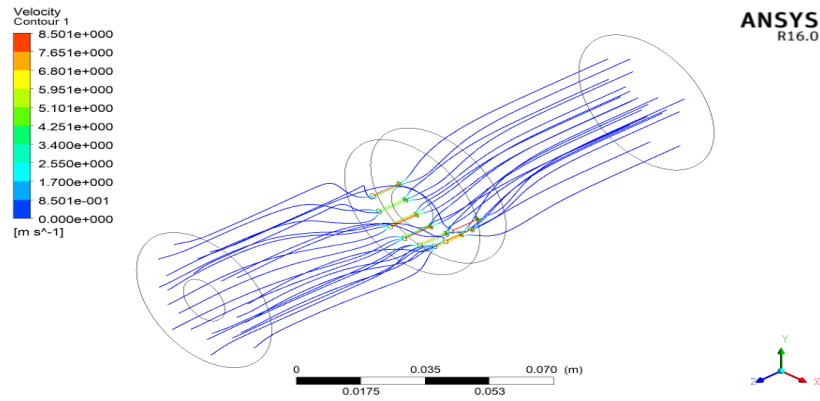


Figure 14. The flow velocity at the orifice exit when the current is 0.1 amperes and the velocity is 8.50 m/s

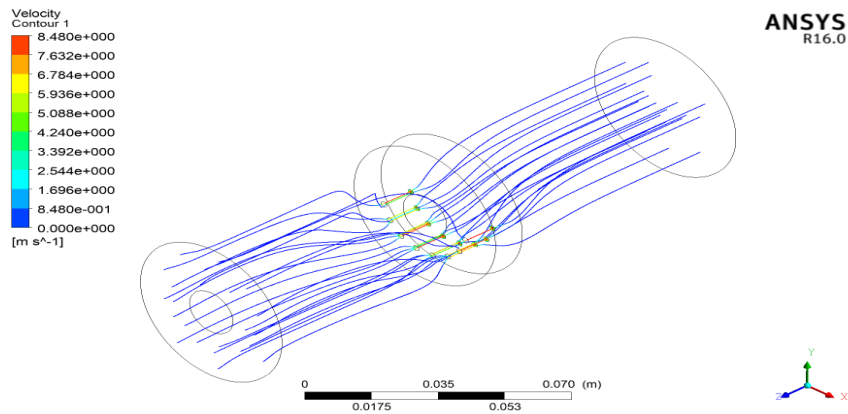


Figure 15. The flow velocity at the orifice exit when the current is 0.15 amperes and the velocity is 8.48 m/s

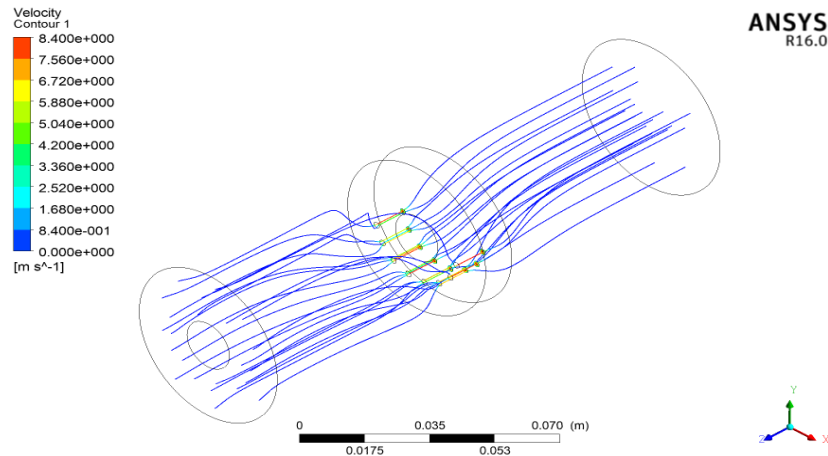


Figure 16. The flow velocity at the orifice exit when the current is 0.2 amperes and the velocity is 8.40 m/s

Calculation of Reynolds's Number

The Reynolds's number is the main criteria for getting the nature of the flow i.e. laminar or turbulent. If the flow is laminar, the design calculations for carrying out the design procedure must be done by linear equation and if it is turbulent flow, the equations used must be non linear.

Let us consider case 1, i.e., Reynold's number of flow when the viscosity is 33.56 Pa.s at 0 amp current.

The Reynold number,

$$Re\ no. = \frac{\rho V D}{\mu}$$

where,

D is the diameter of orifice = 2 mm = 0.002 m

V is the velocity of exit of orifice (from CFD)

ρ is the density of MR fluid = 2500 kg/m³

μ is the Dynamic Viscosity of MR fluid at 0 Amp 33.56 Pa.s

$$Re\ no. = \frac{2500 \times 8.82 \times 0.002}{33.56} = 1.31$$

Hence, it is a laminar flow.

Similarly, the Reynolds's number for each case is calculated and tabulated in following Table 7.

Table 7. Reynolds's Number for All Cases

Sr. No.	Current (Ampere)	Reynolds number
1	0.00	1.31
2	0.05	1.18
3	0.10	0.99
4	0.15	0.86
5	0.20	0.85

Based on the readings and observations, the Reynolds number was found to be well below 2000, indicating laminar flow. Therefore, the use of linear equations in the design calculations is justified.

Conclusions

This research introduces an innovative MR damper featuring externally mounted electromagnetic coils, offering an effective solution to key limitations in traditional MR damper designs. Simulations verified uniform magnetic field distribution, and experimental results demonstrated strong damping tunability - from 731.54 N to 1591.47 N, with damping coefficients ranging from 4.65 to 10.13 kNs/m. This configuration holds promise for a wide range of applications, particularly in adaptive suspension systems, defense, and civil infrastructure requiring reliable and adjustable damping. Future work should explore long-term fatigue performance, temperature effects, and scaling for heavy-duty use.

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