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Design and Hysteresis Behavior of Magneto-Rheological Disc Brake for Hatchback Segment Commercial Car

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Abstract: ***In this paper, an optimized design for hatchback segment of commercial vehicle is considered to find the optimum dimensions for the MR brake, with the help of optimization tools and existing conventional dimensions. An axisymmetric model is generated using COMSOL based on the dimensions available in the existing literature and braking torque is calculated using Bingham plastic model. It is a coupled system i.e. Electric field – Magnetic field – Mechanical action. Initially the existing design from literature survey is taken as base model for the MR brake and the design is changed according to the dimensions found on the basis of the conventional dimensions of disc brake accessible in the existing hatch-back segment car. After finalizing the dimensions and MR fluid the four new design model for disc are considered for simulations. The brake are used several time during the dynamic condition so as to test the reliability of the Magneto-Rheological brake, the hysteresis analysis is also needed to be carried out by simulating it in a dynamic environment. For hysteresis analysis MATLAB Simulink is used for simulating the brake in dynamic condition with the help of Bouc-Wen Model. The main aim is to achieve a braking torque of MR brake near to the existing braking system of the vehicle.***

Index Terms: ***Bingham plastic Model, Bouc-Wen Model,Magneto-rheological disc Brake, Smart materials..***

I. INTRODUCTION

Automobile safety is one of the essential factor. When it comes to accelerating or controlling the vehicle at high speed, brake plays a major role in controlling the vehicle. The conventional braking system consists of many auxiliary assemblies to operate the braking system efficiently throughout the different braking conditions. Auxiliary systems such as master cylinder pressure differential switch proportioning valve, metering valve and vacuum brake booster unit etc. Conventional braking system used in cars have some shortcomings such as bulky system, brake noise due to metal-to-metal contact, brake pad need to be replaced periodically, problems with leakage in hydraulic line, high response time.

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In this piece of work, we have proposed a Magneto-rheological brake for hatchback segment of commercial vehicle. As the commercial vehicle are designed for all weather conditions and transportation of passenger, it should also ensure a safety to a passenger during high speed anddifferent road conditions. So much prominence is set for design of suitable braking system. MR fluids are recognized for its rapid time of response and stiffness which is openly relative to offered magnetic field.

A. Smart materials

It is one of the strongest material which is used to safeguard any type of system or a structure from failures. It properties can be changed by changing or altering the atmosphere. Commonly used smart material in automobile are shape memory alloy, magneto-rheological fluid, electro-rheological fluid etc. MR fluid are categorized as smart materials and are made up of combinationof different materials such as carrier fluids, small sized iron particles and to overcome the problem of sedimentation of iron particles, additives are used. The iron particle composition is around 20-40 percent and acquired by decomposition of iron carbonyl compounds. The MR fluid are generally water base or oil base i.e. a carrier fluid is either water or artificial oil etc. and are selected on the basis of viscosity as well as temperature. To prevent iron particles to sediment, wear and oxidize additives are added. MR fluid has three operational mode such as shear thinning, shear thickening, squeeze mode. The mode are selected based on the application. MR fluid are used widely in automobile for damper design[1], [2], fluid clutch [3] and brakes [4]~[6]. It is not only used for automobile application but also for super structure [7], actuators [8], bearing [9] and biomedical engineering [10], [11].In the fig1 the behaviour of MR fluid with and without magnetic field is shown. As Fe particles in MR fluid form a chain like structure during an external magnetic field. This Fe particle aligned along direction ofmagnetic field.

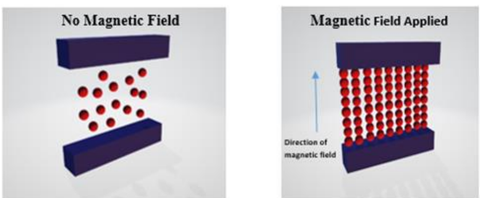

Fig 1. Effect of magnetic field

B. Parametric model.

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For the analysis of the MR brake the following two model has been used to capture the exact behaviour of MR fluid viz.

- Bingham model.
- Bouc-Wen model.

a. Bingham model.

It is a simple plastic model used to represent ideal plastic fluids.The MR fluid follows the attributes of certain class of Non-Newtonian fluid called Bingham fluid which does not show any change in shear rate until a considerable stress has been attained.The Bingham model behaviour is shown in the fig 2

$$Shear\ stress\ (\tau) = \tau_1 + \eta\dot{\gamma}(1)$$

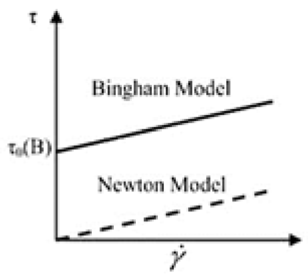

Fig 2. Bingham model

b. Bouc-Wen model.

When it comes to dynamic behaviour and reliability of the system it is essential to test the system under dynamic environment by considering each parameter which affect the performance of system directly or indirectly As the MR brake make utilization of magnetic field for application of brake, so as to access the performance of the brake Bouc-Wen model is used for hysteresis analysis of the system.

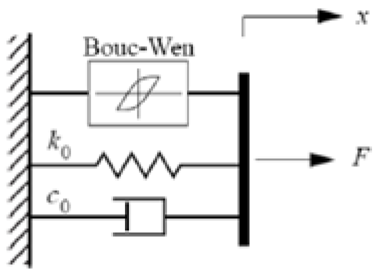

Fig 3. Bingham model

The braking torque T generated by the brake is equal to[12].

$$T = k_0(\theta - \theta_0) + c_0\dot{\theta} + \alpha z(2)$$

Where the evolutionary parameter z is governed by the following expression.

$$z = -\gamma|\dot{\theta}||z||z|^{n-1} - \beta\dot{\theta}|z|^n + A\dot{\theta}(3)$$

here θ denotes the angular displacement of the disc, c_0 is the damping coefficient which is taken as a constant, α denotes the coefficient predicted by the control system and the dynamics of magneto-rheological fluid. The variable z can develop through sinusoidal to quasi-rectangular function of time depends on the parameter β, γ and A. These constants are utilize for controlling the linearity during unloading and the smoothness of the transition from the pre-yield to post-yield region. In the model c_0 and α maintain a linear relationship with the control voltage u.

II. METHODOLOGY

.For the design of MR brake, initially, the design dimensions are based on measurement takenfrom conventional vehicles. To obtain a suitable design parameter it is essential to optimize the parameters for upgraded design. The conventional dimension of traditional brake is adopted from the available disc brake model measurement (i.e. disc brake for hatchback segment). By relating the MR brake dimension with the conventional system the design dimension is considered. The conventional brake dimension is considered as the maximum dimension range for design and the available MR brake dimension are compared to a conventional design for ideal dimensions of commercial car MR brake is considered. For optimizing the dimension Taguchi's method is used using Minitab software. The proposed design dimension is mentioned for commercial vehicle MR brake are optimized.

A. Dimensions

For the initial design of MR brake, a single disc is considered. The 2D model of single disc MR brake is shown in fig 4. For designing MR brake for commercial vehicle initially we have to consider physical boundaries of the conventional system and the base dimension are considered from the literature survey. The dimension of the copper coil is selected from American wire gauge standards catalogue on the basis of current supplied. By taking into account the various dimensions of the elements of the brake, the major factor to be considered is the physical area available for its installation. The final diameter must be such that it should be accommodated within the space available. Therefore a clearance of minimum 4 mm is recommended within the total brake assembly and wheel rim. Hence, the prescribed diameter for the model for a 15" wheel diameter, 15inch $\times$ 25.4mm/inch - 2 x 4mm = 373 mm i.e. 37.3 cm amounting to a maximum of 18.65 cm radius. Note that this dimension is valid for hatchback segment car with a 15inch diameter of the wheel [13].

B. Choice of material

The appropriate choice of material is an essential step for sound design. As in this paper, we are dealing with the magnetic field so the selection should be based on magnetic properties along with structural and heat transfer properties. On the basis of structural properties, the material for disc and shaft are chosen. As the disc inside the casing is rotating at high RPM, during braking the heat is generated due to friction between MR fluid with the disc and inner surface of the casing. So to remove the generated heat from the disc and casing, the phenomena of heat transfer is used.As thenon-magnetic material possessing higher conductivity and convection coefficient have good heat transfer properties. So the material for the shaft is chosen as AISI 4340 and for disc and casing is chosen as AISI 1010. The selection between smart fluid i.e. ER fluids And MR fluids are carried out on the basis of the comparison between the properties of the smart-fluids.

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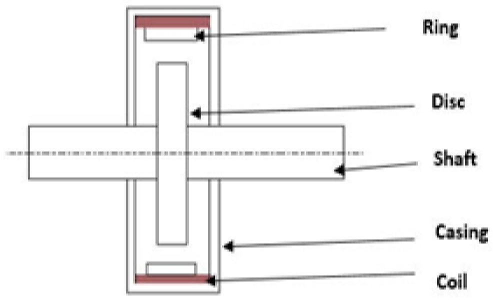

Fig 4. 2D model of MR brake

The material is selected on the basis on availability, cost, suitable for the application, yield stress and magnetic properties of the material. Material for the coil is chosen as copper. MRF 140CG [14] is selected for simulation purpose. Table I,shows the selected material for the application.

Table II. Selected material for MR brake

COMPONENTS	MATERIAL
Shaft	AISI 4340
Disc	AISI 1010
Casing	AISI 1010
MR fluid	MRF 140 CG
Coil	Copper

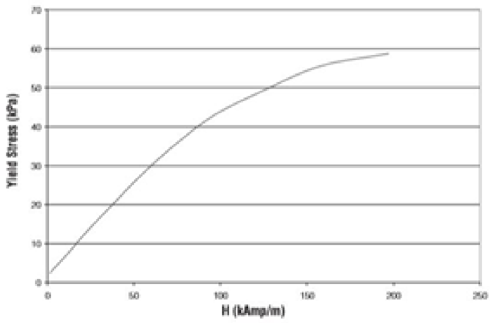

Fig 5. Magnetic field versus Yield stress (MRF 140 CG)[14].

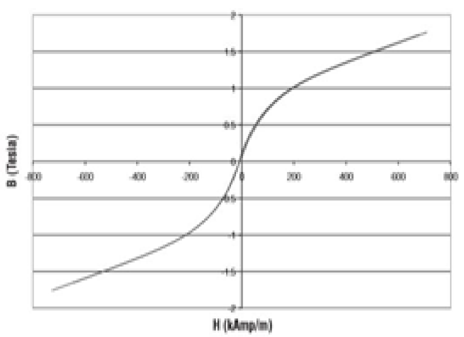

Fig 6. B-H Curve [14].

C. Critical parameter optimization

Optimization is the procedure of obtaining the most feasible parameters from a given set of parameters. For maximum performance of MR brake, design based on optimized dimensions should be considered. The optimization process is carried out with Minitab 17 software by implementing Taguchi's optimization method.

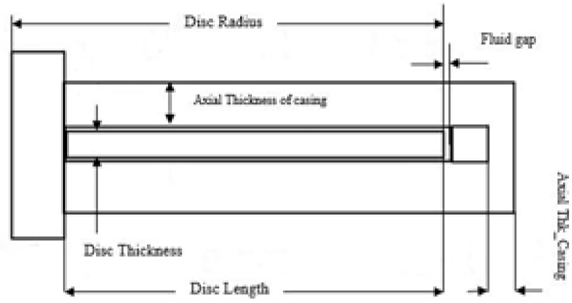

Fig 7. Critical parameters for MR brake

Table III. L27 Orthogonal array for critical parameters

Length of Disc	Thickness of Disc	Axial thickness-casing	Radial thickness-casing	Radius of disc	Fluid gap	Braking torque (N-m)
135	8	8	10	146	0.8	430.31
135	8	8	10	148	0.9	424.35
135	8	8	10	150	1	420.11
135	9	9	11	146	0.8	423.45
135	9	9	11	148	0.9	423.75
135	9	9	11	150	1	418.25
135	10	10	12	146	0.8	418.30
135	10	10	12	148	0.9	419.58
135	10	10	12	150	1	417.01
136	8	9	12	146	0.9	405.60
136	8	9	12	148	1	395.73
136	8	9	12	150	0.8	460.73
136	9	10	10	146	0.9	424.47
136	9	10	10	148	1	415.38
136	9	10	10	150	0.8	470.02
136	10	8	11	146	0.9	399.89
136	10	8	11	148	1	401.09
136	10	8	11	150	0.8	453.94
137	8	10	11	146	1	390.64
137	8	10	11	148	0.8	453.20
137	8	10	11	150	0.9	418.42
137	9	8	12	146	1	397.50
137	9	8	12	148	0.8	429.46
137	9	8	12	150	0.9	432.06
137	10	9	10	146	1	392.93
137	10	9	10	148	0.8	445.54
137	10	9	10	150	0.9	450.04

NOTE:*All input parameters are in mm*

The critical parameter taken into account for improving braking torque. The primary aim is to attain the acceptable value of braking torque within the possible dimensional limit. From the array generated, it is observed that the maximum braking torque is obtained when the following dimensional parameters are taken into consideration.

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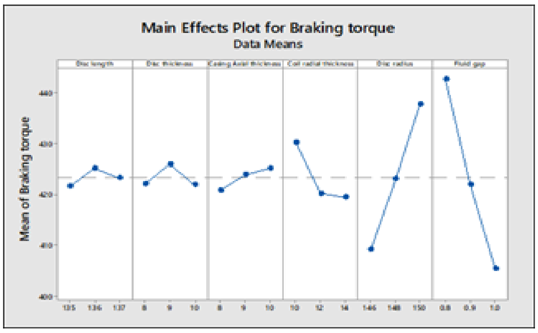


Fig 8. Means for the critical parameters

Table IV. Optimized design parameter

Length of disc	136
Thickness of disc	9
Axial thickness- casing	10
Radial thickness- casing	10
Radius of disc	150
Fluid gap	0.8
Braking Torque (N-m)	470.02

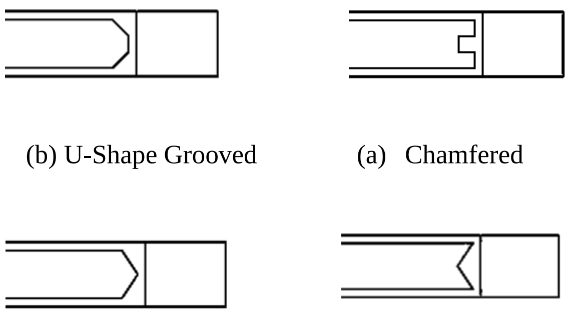
NOTE:*All input parameters are in mm*

The main effect plot for braking torque is shown in fig 8. Hence the final optimized parameters are shown in tableV.

III. STATIC ANALYSIS

The design simulation is carried on COMSOL multi-physics which uses the finite element method for analysis of the design. In this simulation, the 2D axisymmetric model is created in the design peripheral. For reduction of computational time, this model is selected. The optimized dimension based on a commercial vehicle is implemented in the model and four disc designs are considered for the simulation are shown below in fig 9

After creating the 2D axisymmetric model shown in fig 10, materials for each and every part is defined along with the properties of MR fluid, the BH curve is also defined in the material properties. The AC/DC module is used for simulation purpose.



(b) U-Shape Grooved (a) Chamfered

(c) V-Shape Grooved (d) Projected V-Shaped

Fig 9. Different MR disc brake design

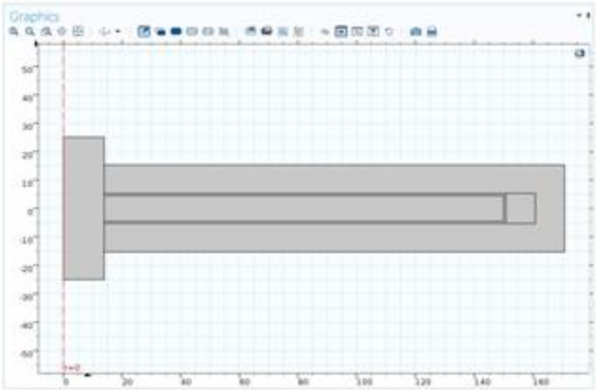


Fig 10. 2D axisymmetric model of MR Brake

IV. DYNAMIC ANALYSIS

The MR disc brake model is integrated with the Bouc-Wen model for the analysis to be performed out numerically by virtue of modeling in MATLAB®/Simulink®. Although the recognition of the non-linear hysteretic parameters is quite complicated and may necessitate for extensive experimental and numerical analyses for the purpose of acquiring pronounced gain of the MR brake attributes. Using the equations (1), (2) & (8), the model [15]created in the MATLAB®/Simulink environment was modified with integration of Bouc-Wen model. The model created has two input variables viz. z_s and$\dot{z}_s$ one output parameter which is the force generated by the brake. Although the Bouc-Wen parameter β and γ pose no physical meaning, it greatly influences the shape and size of the hysteresis loop. It is quite cumbersome to determine the exact values in order to determine the appropriate trajectory of the loop. For theabove model, the values are taken as β = 0.25 and γ = 0.75. Where n is taken as 2 as it decide the slope of the curve between loading and unloading of the system. The model has been designed based on the experimental setup to be performed. The input to the induction motor is taken as the sinusoidal voltage of 230 volts, 50 Hz. The inductance and resistance parameters of the induction motor are taken as 0.003 H and 3.94Ω[15]. The other criterion of the Bouc-Wen model are taken through literature survey [16].where the fluid gap is taken from the optimized dimensions i.e. 0.8 mm.

V. RESULTS AND DISCUSSION

The magnetic analysis has been done in COMSOL using AC-DC nodule is used, the magnetic field available when 1.2A of current is supplied to a coil. The proposed design provides the magnetic field of 59100 A/m with a flux density of 0.43 T. These two results are presented below in fig 11 and 12 respectively

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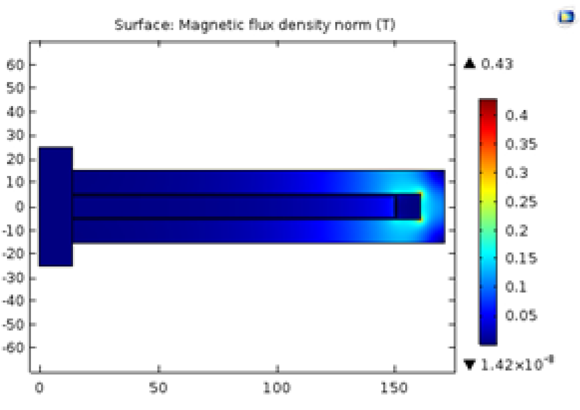

Fig 11. Magnetic field intensity norm

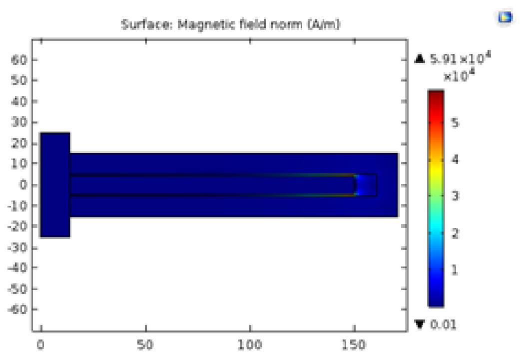

Fig 12. Magnetic flux density norm

The braking torque is calculated from equation (8). The yield stress value is taken from the MR fluid magnetic field versus yield stress graph given in fig5. The magnetic field value obtained from the simulation result is compared with the graph and the corresponding yield stress value is obtained as 29717.83 kPa.

Here the parameter consider for the calculation of braking torque at 700 rpm are:
Radius (r) = 136 mm
Fluid gap (G) = 0.8 mm
Radius (outer), b_2 = 150mm
Radius (inner), b_1 = 150mm – 136mm = 14mm
MR-fluid viscosity (η) = 0.35 Pa-s
Equation 1. Can be written as:
Total shear stress (τ) = $\tau_1(H) + \eta\omega\frac{r}{G}$(4) (4)
Shear stress (τ) = 30714.574 N/m^2
Substituting the value of shear stress in braking torque equation:
Braking torque,$T_B = F_B \times b = \tau.b.dA$(5)
Where, $F_B$is Braking force and b is Radius of disc
For MR brake area of the disc is given by:
$A = \pi b^2$
$dA = 2\pi b.db$(6)
Substitute the equation (6) in equation (5),
$T_B = 2\pi.n.\tau.b^2.db$(7)
This is the equation for braking torque, for a single disc design the contact is made on both sides of the disc, therefore, the value of $n = 2$.
The equation is for small area for the entire surface we need to integrate the equation (7) from limit $b_1$to b_2,

$$T_B = \int_{b1}^{b2} 4\pi\tau b^2.dr \quad (8)$$

This is the equation for the entire surface for calculating braking torque, T_B = 479.94 Nm (For Cylindrical disc)

Table VI. Comparison of Braking Torque

Disc Shape	Magnetic Field Intensity (H) [A/m]	Magnetic Flux Density (B) [T]	Braking Torque (T) [N-m]
Cylindrical Shape	5.91 x 10^4	0.43	479.94
Chamfered	5.24 x 10^4	0.42	438.41
U-Shape grooved	5.69 x 10^4	0.43	466.30
V-Shape grooved	5.75 x 10^4	0.34	470.02
Projected V-Shaped	5.36 x 10^4	0.42	445.85

NOTE: *The braking torque calculated might be affected due to losses during practical application*

The obtained hysteresis plot of the MR brake model based on the parameter obtained from the literature survey [15].

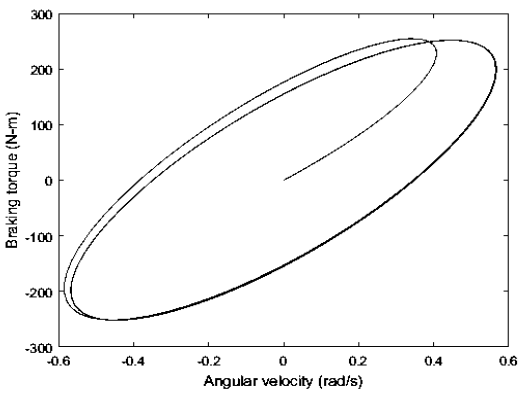

Fig 13. Braking torque versus Angular velocity

Table VII. Conventional vehicle data

Vehicle type	Speed (In Km/h)	Stopping distance (in m)
Hatchback	80-0	25.97

Table VIII. Comparison of model with Conventional system

Model	Maximum braking torque (N-m)
Static	479.94
Dynamic	264.67
Conventional	1149.90

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As deduced from TableIX, the proposed model is able to achieve approx. 41.73% of the conventional braking torque. Where as in dynamic model the braking torque is lower than the static model due to hysteretic losses occurs in the MR disc brake.

VI. CONCLUSION

The proposed model is able to achieve moderate amount of braking torque as compared to the conventional braking system for the hatchback segment. While changing the shape of disc, it was found that there is a decrease in the magnetic field intensity due to the deflection of magnetic field lines. The braking torque obtained from the Bouc-Wen model is less than that of the Bingham model due to the effect of hysteresis. It is still able to capture the exact non-linear behavior of MR fluid more effectively. The torque achieved in Bouc-Wen model is not accurate as the parameters are not appropriate. To capture the actual behavior, the model need to be fabricated and the parameters obtained from the experimental result need to be simulated for precise braking torque and hysteresis losses in dynamic conditions.

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