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DESIGN OF MAGNETO-RHEOLOGICAL BRAKE FOR AUTOMOTIVE APPLICATION

Bhau K. Kumbhar*, Vikas R. Khalkar, Pravin N. Jadhav, Satyajit R. Patil
[1]Mechanical Engineering Department, Gharda Institute of Technology, Khed (MH), India
[2]Automobile Engineering Department, Rajarambapu Institute of Technology, Islampur (MH), India

ABSTRACT

The Conventional Hydraulic Brakes (CHB) system is being used to provide the required braking torque to stop a vehicle. This system has number of drawbacks such as delayed response time to stop a vehicle, bulky size, brake pad wear etc. With the use of magneto-rheological (MR) brake, different limitations of CHB can be eliminated. MR brake is one of the possibilities to replace a CHB, which is the pure electronically controlled actuator and has a potential to further reduce the braking time. There lies a potential for investigation of MR brake for automotive application. Thus. in this paper, the efforts have been taken to design the MR brake for E-bicycle.

Keywords: Conventional Hydraulic Brake (CHB), MR fluid, MR brake, Design.

INTRODUCTION

The CHB has a number of limitations, including [1]:

- delayed response time (200–300 ms) due to pressure build up in the hydraulic lines,
- bulky size and heavy weight due to its auxiliary hydraulic components and brake actuators,
- brake pad wear due to its frictional braking mechanism, and
- low braking performance in high speed and high temperature situations.

Drawback of conventional hydraulic brake system of fading reduces the effectiveness. The improvement of the effectiveness of the braking system can be made by reducing the stopping time and distance. However, meanwhile there need to take care of vehicle stability. This can be possible by MR brake. Also to improve the performance of vehicle, weight of braking system is kept as less as possible. The MR brake as shown in Fig. 1 operates in a direct-shear mode, shearing the MR fluid filling the gap between the two surfaces [2].

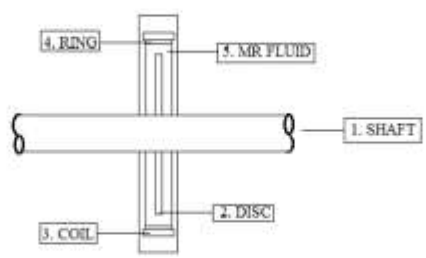

Fig. 1 MR Brake

The MR fluid brake is a device to transmit torque by the shear stress of MR fluid. MR rotary brake has the property that its braking torque changes quickly in response to an external magnetic field strength.

In this chapter, design process for an automotive MR brake has been discussed. In this work, practical design criteria such as material selection, sealing, working surface area, required braking torque, electromagnet design and MR fluid selection are considered to select a basic automotive MR brake configuration.

As per the past research it was found that for 1000 kg passenger vehicle 500 N-m torque is required for braking purpose. But previously made MRB prototype provided only 5% of required torque, that is only 23 N-m[1]. This torque is insufficient to apply brake for this passenger vehicle. For a two

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wheeler application braking torque requirement is very low as compared to the passenger vehicle. Also, it is light in weight hence, braking force will be less. Therefore it is decided to design MR brake for a two wheeler application. For this work, E-bicycle is selected, which is light in weight and has low braking torque requirement. The specifications of E-bicycle is given in Appendix-I. The E-bicycle is as shown in Fig. 2.

Fig. 2 E-bicycle

PROBLEM STATEMENT

The MR brake is to be designed for an E-bicycle. For this design, following data is available.

I.	Stopping time	: 2s, 3s, 4s, 5s
II.	Speed of a vehicle	: 30 km/h
III.	Mass (unladen condition)	: 50 kg
IV.	Mass (laden condition)	: 133 kg
V.	Mass on front wheel	: 50 kg
VI.	Mass on rear wheel	: 83 kg
VII.	Tyre size	: 18*2.125
VIII.	Wheelbase	: 1180 mm
IX.	Wheel Radius	: 228.72 mm

DESIGN OBJECTIVES

The design objectives for MR brake are listed below:

I. Calculation of Braking torque.
II. Selection of appropriate configuration for MR brake, like rotary/cylindrical, single disc/two disc, etc.
III. Selection of MR fluid, study of its properties, mainly shear stress, yield strength, viscosity, temperature range & determination of its volume.
IV. Selection of material for various elements of MR brake.
V. Determination of shaft size & maximum brake size
VI. Design of electromagnet.
VII. Selection of seals.

CALCULATION OF BRAKING TORQUE

Steps involved in calculation of braking torque:

Braking torque for the E-bicycle is calculated as per the following steps:

I. Determination of C.G. of selected vehicle.
II. Determination of deceleration values for a typical speed [25 km/hr & 30 km/hr] & for a typical road surface [having coefficient of friction 0.6 & 0.7].
   1) It involves the determination or assumption of stopping distance, stopping time.
   2) For this, various standards like FMVSS, IS, ISO, SAE, JSS are available, otherwise deceleration should be calculated for various stopping distance & stopping time values.
III. Based on the analytical formulae, determination of braking force for front & rear wheel.
IV. Determination of braking torque

Calculation of Braking Torque for E-bicycle:

The braking torque is calculated for the two wheeler (E-bicycle).

Determination of C.G. of a selected vehicle

Wheelbase of the vehicle is measured by measuring tape. Specification for the measuring tape is as follows:

- Make: plastic fiber glass tape measure.
- L.C.: 1 mm
- Range: 1 mm to 30 m

The measurements taken by measuring tape are as follows:

Wheelbase (L) of E-bicycle = 1180 mm
Wheel radius = 228.72 mm
Tyre size: 18 * 2.125

To find the position of C.G., the location of C.G. is determined from front axle & from rear axle. In addition to this, height of the C.G. from the ground surface has been measured. The detailed calculations are as follows:

The methods for finding the C.G. of vehicle are presented in [10], [11], [5].

Here, the method presented in [6] is adopted for the determination of C.G. which is as follows:

Determination of Longitudinal Position of C.G.:

To calculate position of C.G. from front axle & rear axle, the total weight of the vehicle, weight at front wheel, weight at rear wheel has been measured. For measurement of weights, the weight machine is used which is available in Civil Engineering Department laboratory, RIT.

The weight of a vehicle taken in unladden condition is as follows:

- Mass at front wheel : 20.18 kg

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- Mass at rear wheel : 29.81 kg
- Total mass of the vehicle : 49.99 kg

Then, the mass of a vehicle has been measured for the laden condition. This laden mass is taken by including one person mass.

Measurements of E-bicycle for laden condition:

- Mass at front wheel : 50 kg
- Mass at rear wheel : 83 kg
- Total mass of the vehicle : 133 kg

Calculations:

$b = W_{rs} \times L/W$ -----(1) [11]

where;

b = position of the C.G. from front axle

Wrs = weight on rear wheel

W = total weight

$b = 83 \times 1180/133$

$\boxed{b = 736.40 \text{ mm}}$

$c = W_{fs} \times L/W$ ----(2) [11]

where;

c = position of the C.G. from rear axle

Wfs = weight on front wheel

W = total weight

$c = 50 \times 1180/133$

$\boxed{C = 443.60 \text{ mm}}$

Determination of Height of C.G.:

To calculate the height of C.G., the method mentioned in [10] is adopted. In this process, front wheel is lifted up quite a small distance h_1 to find out the weight shifted on rear side.

Calculations:

According to the geometry in Fig. 3, taking moment about point A

$F(c_1 + c_2) = W \times c_1$ -----(3) [10]

$c_1 + c_2 = L\cos\theta$

where;

c_1 = position of C.G. from rear axle when vehicle is level ground.

c_2 = position of C.G. from front axle when vehicle is level ground.

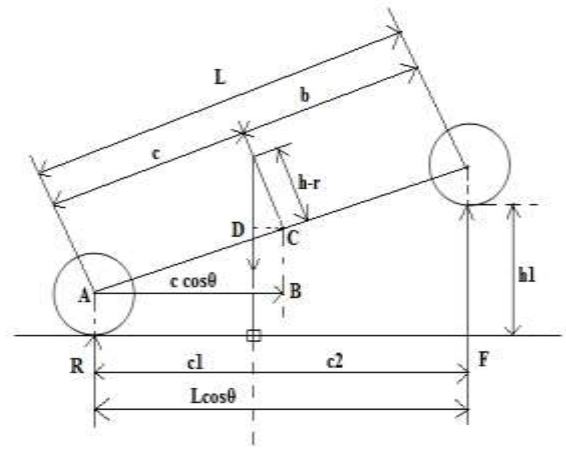

Fig. 3 Geometry for the method of finding C.G.[10]

But;

$\sin\theta = h_1/L$

h_1 = height of block = 110 mm

L = wheelbase = 1180 mm

$\sin\theta = 110/1180$

$\theta = 5.34°$

$c_1 + c_2 = L\cos\theta$

$\quad = 1180 \times \cos(5.34)$

$c_1 + c_2 = 1174.87mm$ ---------(4)

Now,

$c_1 = AB - CD$

$AB = c \times \cos\theta$ &

$CD = (h - r) \times \sin\theta$

$c_1 = c \times \cos\theta - (h - r) \times \sin\theta$

$c_1 = 443.60 \times \cos(5.34) - (h - 228.6) \times \sin(5.34)$

$c_1 = 441.67 - (0.093h - 21.27)$

$c_1 = 46.94 - 0.093h$ ---------(5)

Mass on rear wheel when front end lifted at height 110 mm = 90 Kg

Therefore,

mass shifted = 90 – 83 = 7 Kg

F = mass of front wheel – mass shifted

F=50–7 = 43 Kg ----------(6)

Putting values of Eq^n (2),(3),(4) in Eq^n (1)

$F(c_1 + c_2) = Wc_1$

$43 \times (1174.87) = 133 \times (46.94 - 0.093h)$

$\boxed{h = 893.49 \text{ mm}}$

From above calculations, the exact position of C.G. has been found. From front axle it is 736.40 mm, from rear axle it is 443.60 mm and height of the C.G. is 893.49 mm. The position of C.G. is shown in Fig. 4.

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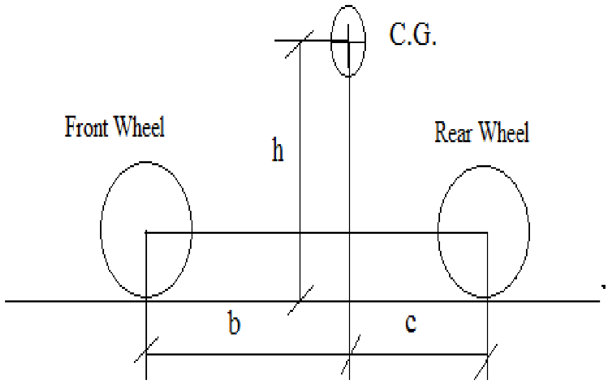

b=734.40mm, c=443.60mm, h=893.50mm

Fig. 4 Position of C.G.

Calculation of Deceleration values:

The vehicle selected for design having a highest speed of 25 km/hr, therefore for design two maximum speeds have been selected i.e. 25 km/hr & 30 km/hr. For finding deceleration values for above mentioned speed, different values of stopping time like 2s, 3s, 4s, 5s are assumed. Calculations for deceleration values are as follows:

$t_s = V/D_x$ ------ (7) [11]

where,

t_s = stopping time of the vehicle (sec.)

V = velocity of the vehicle (m/s)

D_x = deceleration of the vehicle (m/s^2)

1) For velocity V = 25 km/hr :

V = 25 km/hr = 6.94 m/s

Stopping time (t_s) = 2s

Deceleration $D_x = V/t_s$

= 6.94/2

D_x = 3.47 m/s^2 = 0.35g

2) For velocity V = 30 km/hr :

V = 30 km/hr = 8.33 m/s

Stopping time (t_s) = 2s

Deceleration $D_x = V/t_s$

= 8.33/2

D_x = 4.17 m/s^2 = 0.43g

For different values of stopping time, deceleration values are calculated and presented in Table 1.

Table 1. Deceleration values for various stopping time

Sr. No.	Stopping Time (sec.)	Deceleration Values (m/s^2)	
		V = 25 km/hr (6.94 m/s)	V = 30 km/hr (8.33 m/s)
1	2	3.47	4.17
2	3	2.31	2.78
3	4	1.73	2.08
4	5	1.39	1.67

Calculation of Maximum Braking Force:

From above table, various deceleration values are obtained. For these deceleration values, the maximum braking force has been calculated. To calculate the maximum braking force, two values of coefficient of friction that is 0.6 & 0.7 is considered. These values are taken for two road surfaces. Normal earth road (dry) surface & concrete road surface (wet) have coefficient of friction as 0.6 & 0.7 respectively.

Maximum braking force at front wheel:

$F_{xmf} = \mu_p \times W_f$ -------- (8) [6]

Where,

F_{xmf} : maximum braking force at front wheel.(N)

μ_p : peak coefficient of friction.

W_f : load on front axle. (kg)

Dynamic load on the front axle is to be calculated by formula:

$W_f = W_{fs} + (h/L \times W/g \times D_x)$ --------(9) [11]

$W_{fs} = W \times c/L$

Where;

W_{fs} : front axle static load.(kg)

h : height of C.G. from ground surface.(mm)

L : length of wheelbase.(mm)

W : total mass of the vehicle.(kg)

g : gravitational acceleration.(m/s^2)

D_x : deceleration.(m/s^2)

c : position of C.G. from rear axle.(mm)

1) Maximum braking force at rear wheel:

$F_{xmr} = \mu_p \times W_r$ ------------ (10) [11]

Where;

F_{xmr} : maximum braking force at rear wheel.(N)

μ_p : peak coefficient of friction.

W_r : load on rear axle. (kg)

Load on rear axle is to be calculated by formula:

$W_r = W_{rs} - (h/L \times W/g \times D_x)$ -------- (11) [11]

$W_{rs} = W \times b/L$

where,

W_{rs} : rear axle static load.(kg)

h : height of C.G. from ground surface. (mm)

L : length of wheelbase.(mm)

W : total mass of the vehicle. (kg)

g : gravitational acceleration. (m/s^2)

D_x : deceleration. (m/s^2)

b : position of C.G. from front axle. (mm)

From above formulae the maximum braking force at front and rear wheel has been calculated. The calculation is done for one sample value of deceleration & coefficient of friction.

Case I) In this case, speed is assumed to be 25 km/hr, deceleration is 3.47 m/s^2 which is calculated for stopping time 2s and value of coefficient of friction is 0.6. Calculations for these values are as follows:

1) At front wheel:

W_{fs} = 133 *[443.60/1180] = 49.99 kg.

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F_{xmf}= 0.6 * [49.99 + (893.49/1180 * 133/9.81 * 3.47)] = 51.37 N

2) At rear wheel:

W_{rs} = 133 *[736.40/1180] = 83 kg.

F_{xmr}= 0.6 * [83 – (893.49/1180 * 133/9.81 * 3.47)] = 28.43 N

Above calculations represent the maximum braking force at front & rear wheel. For the coefficient of friction 0.6, the braking force calculated for different deceleration values are presented in Table 2.

Table 2. Braking force for various deceleration values at mu = 0.6 (case I)

Sr. No.	Stopping Time (sec)	Deceleration (m/s^2)	Braking Force (N)	
			Front wheel	Rear wheel
1	2	3.47	51.37	28.43
2	3	2.31	44.22	35.57
3	4	1.75	40.78	39.02
4	5	1.39	38.56	41.24

For coefficient of friction 0.7, the braking force calculated for different deceleration values are presented in Table 3.

Table 3. Braking force for various deceleration values at mu = 0.7 (case I)

Sr. No.	Stopping Time (sec)	Deceleration (m/s^2)	Braking Force (N)	
			Front wheel	Rear wheel
1	2	3.47	59.94	33.16
2	3	2.31	51.60	41.50
3	4	1.75	47.58	45.52
4	5	1.39	44.99	48.11

Case II) In this case, speed is assumed to be 30 km/hr, deceleration is 4.17 m/s^2 which is calculated for stopping time 2s and value of coefficient of friction is 0.6. Calculations for these values are as follows:

1) At front wheel:

W_{fs} = 133 [443.60/1180] = 49.99 kg.

F_{xmf} = 0.6 * [49.99 + (893.49/1180* 133/9.81*4.17)] = 55.68 N

2) At rear wheel:

W_{rs} = 133 *[736.40/1180] = 83 kg.

F_{xmr}=0.6*[83–(887.04/1180*133/9.81*4.17)] = 24.12 N

Above calculations represent the maximum braking force at front & rear wheel. For coefficient of friction 0.6, the braking force calculated for different deceleration values are presented in Table 4.

Table 4. Braking force for various deceleration values at mu = 0.6 (case II)

Sr. No.	Stopping Time (sec)	Deceleration (m/s^2)	Braking Force (N)	
			Front wheel	Rear wheel
1	2	4.17	55.68	24.12
2	3	2.78	47.12	32.68
3	4	2.08	42.81	36.99
4	5	1.67	40.29	39.51

For coefficient of friction 0.7, the braking force calculated for different deceleration values are presented in Table 5.

Table 5. Braking force for various deceleration values at mu = 0.7 (case II)

Sr. No.	Stopping Time (sec)	Deceleration (m/s^2)	Braking Force (N)	
			Front wheel	Rear wheel
1	2	4.17	64.96	28.13
2	3	2.78	54.98	38.12
3	4	2.08	49.95	43.15
4	5	1.67	47.00	46.10

As per the values shown in Table 5, it is found that maximum deceleration for the speed of 30 km/hr is 4.17 m/s^2. Therefore assuming maximum deceleration 5 m/s^2 to calculate maximum braking force for two road surfaces.

For coefficient of friction 0.6, the maximum braking force for the maximum deceleration value at front & rear wheel is calculated as follows:

1) Maximum braking force at front wheel:

W_{fs} = 133 *[443.60/1180] = 49.99 kg

F_{xmf}= 0.6 * [49.99 +(893.49/1180 * 133/9.81 * 5)] = 60.80 N

2) Maximum braking force at rear wheel:

W_{rs} = 133 *[736.40/1180] = 83 kg

F_{xmr}= 0.6 * [83 – (887.04/1180 * 133/9.81 * 5)] = 19.00 N

For the values of coefficient of friction 0.6 and 0.7, the maximum braking force for the maximum deceleration value at front & rear wheel is calculated, presented in Table 6.

Table 6. Maximum braking force for maximum deceleration

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Sr.No.	Coefficient of Friction (μ_p)	Deceleration Value (m/s^2)	Braking Force (N)	
			Front	Rear
1	0.6	5	60.80	19.00
2	0.7	5	70.93	22.47

From all above results, it has been concluded that the highest maximum force is obtained at deceleration 5 m/s^2 with the coefficient of friction 0.7 is 70.93 N for front wheel. Therefore the braking torque is calculated for highest value of maximum braking force.

Calculation of braking torque:

Braking torque at the front wheel is calculated by using value of wheel radius & braking force.

$$T_{bf} = F_{xmf} \times r \qquad \text{------ (12) [11]}$$

Where;

T_{bf} = Braking torque at front wheel (N-m)

F_{xmf} = Maximum braking force at front wheel (N)

r = Wheel radius (m)

Calculations:

T_{bf} = 70.93 * 0.228

T_{bf} = 16.172 N-m

Therefore maximum braking torque for E-bicycle is 16.172 N-m at front wheel.

SELECTION OF APPROPRIATE CONFIGURATION FOR MR BRAKE

There are different configurations which are used by various researchers such as rotary/cylindrical, single disc/ two disc etc. Huang et al.[7] used cylindrical geometry with annular space for MR fluid between rotor and stator. Li and Du[14] used the disc shape geometry for the design of MR brake. Karakoc et al.[1] selected the rotory configuration of brake having two disc construction. From this design, they got the torque near about the 23 N-m which is near about calculated braking torque, therefore this configuration is adopted with some modifications in it.

Regarding the decision of the MR fluid gap, Sukhwani and Hirani[9] developed the MR brake having 1 mm gap & 2 mm gap. They suggested that the torque produced by MR fluid reduces with increment of gap from 1 to 2 mm at almost all the speeds and field combinations except at very high speeds. Therefore as per the requirement of braking torque, it is decided to keep MRF gap of 1mm. From above study, rotary configuration is selected to this application, because it is suitable for automotive application as requires less space. As surface area is more in case of two disc type configuration, as compared to one disc configuration, it is decided to use two disc configuration so as to increase working surface area.

SELECTION OF MR FLUID

There are various MR Fluids in the market which are used for different purpose. In today's date, Lord Corporation is far ahead of other commercial manufacturers of MR fluid. The important properties of MR fluids from the view point of automotive brake applications are yield stress, no field viscosity and operating temperature range.

The Viscosity of the fluid without magnetic field application is one of those properties. Since the gap between the stator and rotor is filled with the fluid, viscous torque is present in the MRB. Thus in order to keep the viscous torque low, a fluid with a low viscosity has to be selected. The shear stress gradient with respect to the applied magnetic field intensity is another important property of the MR fluid. By keeping the shear stress gradient high, the amount of braking torque that could be generated by the brake will also increase. Another important property of MR fluid is the operating temperature range. By considering the brake actuator, the kinetic energy will be transferred as heat in the MRB, and it will heat up while braking. Since the magnetic properties of the materials in the MRB and the MR fluid viscosity depends on the temperature, the fluid with a broader operating temperature range has to be selected in order to maintain the braking performance at higher operation temperatures [1].

Three candidate MR fluids of Lord Corp. have compared on the basis of these properties. MRF 132 DG is most suitable MR fluid available among those offered by Lord Corp. MRF 132 DG is selected for this work due to its broad operating temperature range from -40°C to +130°C. Table 7 shows the properties of MRF-132DG.

Table 7. Properties of MRF – 132DG[1]

Property	Value/Limits
Base fluid	Hydrocarbon
Operating temperature	-40 to 130°C
Density	3090 kg/m^3
Color	Dark gray
Yield stress	45kPa
Weight percent solid	81.64%
Specific heat at 25°C	800 J/kgK
Thermal conductivity at 25°C	0.25-1.06 W/mK
Flash point	>150°C
Viscosity at 40°C	0.09(±

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	0.02)Pa s
K (MRF dependent constant parameter)	0.269 Pa m/A
β (MRF dependent constant parameter)	1

The curve of yield stress Vs magnetic field intensity shows the value of yield stress as shown in Fig.5. The curve shows the saturation occurs at magnetic field intensity 250 kA/m & having yield stress of 45 kPa.

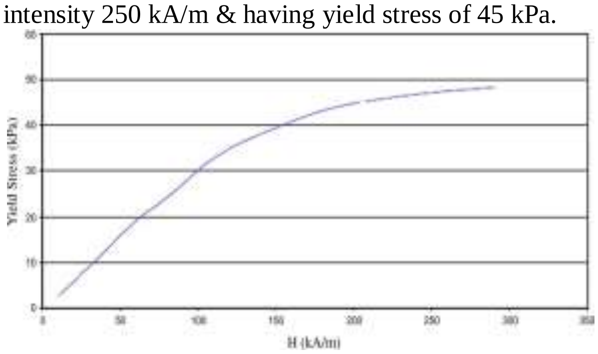

Fig. 5 Shear stress Vs Magnetic field intensity for MRF 132DG [10]

The selected configuration for MR brake is summerized in Table 8.

Table 8. Selected Configuration of MR brake

Parameter	Configuration Selected
Geometry	Rotary
No. of Discs	Two
MR Fluid Gap	1 mm
MR Fluid	132 DG

SELECTION OF MATERIAL FOR VARIOUS COMPONENTS OF MR BRAKE

The material selection is another critical part of the MRB design process. Materials used in the MRB have crucial influence on the magnetic circuit as well as the structural and thermal characteristics. Here, the material selection issue is discussed in terms of the (i) magnetic properties and (ii) structural and thermal properties. The property that defines a material's magnetic characteristic is the permeability (μ_r). Magnetic materials are characterized by absolute magnetic permeability μ as function of applied magnetic field. In case of magnetostatics, magnetic permeability relates the magnetic flux density (B) and magnetic field intensity ($B = \mu H$). However, permeability of ferromagnetic materials is highly non-linear. It varies with temperature and applied magnetic field (e.g. saturation and hysteresis)[1]. In Table 9, some candidate examples of ferromagnetic and non-ferromagnetic materials are listed.

Table 9. Examples of ferromagnetic and non-ferromagnetic materials [11]

Ferromagnetic materials ($\mu_r > 1.1$)	**Non-ferromagnetic materials ($\mu_r < 1.1$)**
Alloy 225/405/426	Aluminum
Iron	Copper
Low carbon steel	Molybdenum
Nickel	Platinum
42% Nickel	Rhodium
52% Nickel	302–304 stainless steel
430 stainless steel	Tantalum Titanium

As ferromagnetic material, there is a wide range of alloy options that are undesirably costly for the automotive brake application. Therefore, a more cost-effective material with required permeability should be selected. Considering the cost, permeability and availability, low carbon steel, AISI 1018 is selected as a magnetic material in magnetic circuit (i.e. the core and disks). Corresponding B–H curve of steel 1018 with the saturation effect is shown in Fig.6[1].

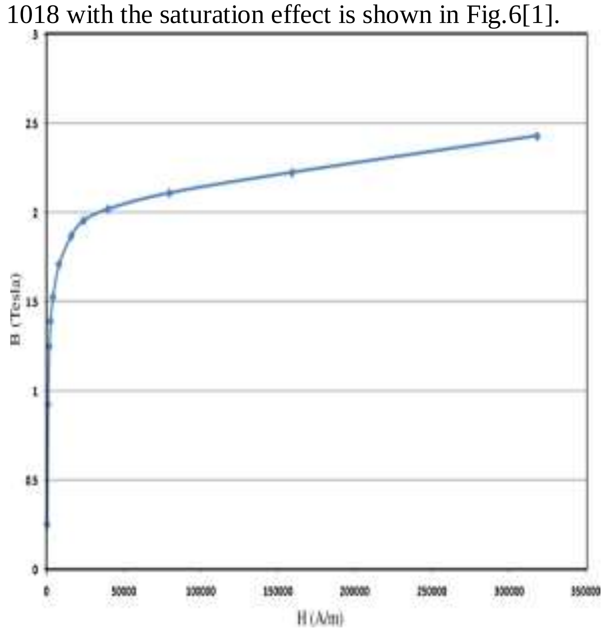

Fig. 6 B–H curve of steel 1018 [1]

In terms of structural considerations, there are two critical parts: the shaft and the shear disk. The shaft should be non-ferromagnetic in order to keep the flux

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far away from the seals that enclose the MR fluid[1]. A suitable material for the shaft is 304 stainless steel due to its high yield stress and availability. For the shear disk material, already chosen AISI 1018 has a high yield stress. The remaining parts are not under any considerable structural loading.

Thermal properties of the materials are another important factor. Due to the temperature dependent permeability values of ferromagnetic materials and MR fluid viscosity, heat generated in a brake should be removed as quickly as possible. In terms of material properties, in order to increase the heat flow from the brake, a material with high conductivity and high convection coefficient has to be selected as materials for the non-magnetic brake components. Aluminum is a good candidate material for the thermal considerations[1]. Table 10 shows the material selected for different parts of MRB:

Table 10. Material selected for various components of MRB

Parts	Material
Shaft	304 stainless steel
Shear disc	AISI 1018
Casing	Aluminum
Magnetic circuit (core)	AISI 1018

DETERMINATION OF SHAFT SIZE & MAXIMUM BRAKE SIZE

Shaft Size:

Shaft design on the basis of Torsion:

Determination of shaft size is made by deciding its maximum diameter. As we know the maximum braking torque required to stop the vehicle, therefore shaft size is calculated by considering the equation:

$$T = \frac{\pi}{16} \times \tau \times D^3 \qquad \text{--------(13) [12]}$$

Where;

T = maximum braking torque (N-m)

τ = permissible shear stress (N/mm^2)

D = shaft diameter (mm)

The maximum braking torque is 16.172 N-m for E-bicycle, therefore taking maximum torque as 25 N-m to calculate the maximum diameter of the shaft. Material for the shaft is 304 stainless steel; therefore yield strength for the stainless steel (S_{yt}) is 386 N/mm^2 [12].

To calculate the permissible shear stress using equation:

$$\tau = 0.5\ S_{yt}/\text{factor of safety} \qquad \text{-----------(14)}$$

Assuming factor of safety 5, permissible shear stress is calculated as follows:

$\tau = 0.5 * 386/5$

$\tau = 38.6$ N/mm^2

Therefore to calculate the diameter of the shaft

$$T = \frac{\pi}{16} \times \tau \times D^3$$

$25 * 10^3 = \pi/16 * 38.6 * D^3$

D = 14.88 = 15 mm

Shaft design on the basis of combined twisting and bending moment:

The combined twisting and bending moment has been considered & checked size of the shaft. Therefore to calculate shaft diameter, the simple geometry for bending moment shown in Fig 7.

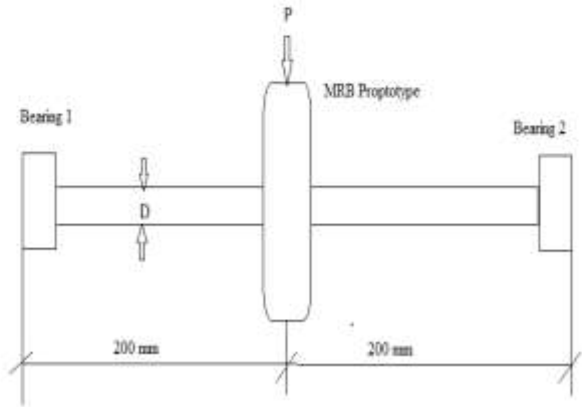

Fig. 7 Bending & Torsion Moment

Calculations:

Considering the bearing is kept at a distance 200 mm from prototype and mass acting on shaft due to brake prototype is 30 kg then;

P = 30 kg

P = 30 * 9.81 = 294.3 N

Therefore, considering the permissible yield stress for the stainless steel material is

$S_{yt} = 386$ N/mm^2

$S_{ut} = 634$ N/mm^2

$0.18\ S_{ut} = 0.18 * 634 = 114.12$ N/mm^2

$0.30\ S_{yt} = 0.30 * 386 = 115.8$ N/mm^2

From these two values, minimum value is selected for calculation of permissible shear stress.

$\tau = 0.75 * 114.12$

$\tau = 85.59$ N/mm^2

Bending & torsional moment calculations are as follows:

M_b = Bending moment

M_t = Torsional moment

Therefore,

$M_b = 294.3 * 200$

$M_b = 58866$ N-mm

Then,

$M_t = 25000$N-mm

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Assuming combined shock & fatigue factor applied to bending (K_b) & torsional (K_t) moments.
K_b = 1.5 & K_b= 1.0 [13].

$$D^3 = \left[\frac{16}{(\pi \times \tau)}\right] \times \sqrt{[(K_b \times M_b)^2 + (K_t \times M_t)^2]} \quad \text{-----(15)[12]}$$

D = 25.79 mm

For different values of load & different values of combined shock & fatigue factor applied to bending (K_b) & torsional (K_t) moments, calculated diameter of the shaft is as shown in Table 11.

Table 11. Various values of shaft diameter

Sr. No.	Load (P) (N)	K_b	K_t	D (mm)
1	294.3	1.5	1	25.79
2	294.3	2	1.5	28.48
3	294.3	3	3	32.98

But for manufacturing suitability & considering the maximum brake size, it has to be decided to keep the shaft size 30 mm.

Maximum brake size :

On the basis of past researches[1],[14], the brake size must be restricted so that brake can be fitted inside a wheel rim like the typical CHB. Based on the available space envelope for mounting of the brake on the wheels of the E bicycle, the overall size was estimated to be around 152- 202 mm (6 to 8 inches). Hence the overall size of the proposed MR brake is to be restricted to 6-8 inches.

For deciding the size of the disc, the overall layout of MRB is considered. Also the structural details of vehicle specifically wheel & wheel spindle is to be considered. The radius of disc was proposed to be 77 mm approximately. For the determination of thickness of the disc, past researches showed that disc thickness ranges from of 6 mm to 12 mm. The thickness of the disc from past researches is presented in Table 12.

Table 12. Disc thickness from past researches

Research	Disc Thickness (mm)
Sukhwani & Hirani (2007)	6
Marannano et al.(2011)	6
Nguyen and Choi (2010)	5
Park et al.(2006)	12

DESIGN OF ELECTROMAGNET

The main goal of the magnetic circuit analysis is to direct the maximum amount of the magnetic flux generated by the electromagnet onto the MR fluid located in the gap. This will allow the maximum braking torque to be generated. As shown in fig 8, the magnetic circuit in the MRB consists of the coil winding in the electromagnet, which is the magnetic flux generating ‘‘source” (i.e. by generating magnetomotive force or mmf), and the flux carrying path.[1]

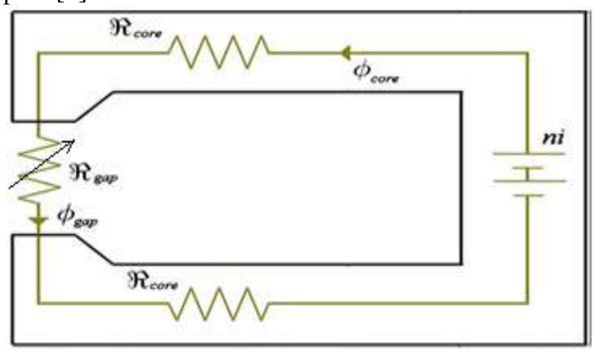

Fig. 8 Magnetic circuit representation of the MRB[1]

As shown in Fig. 3.7, the magnetic circuit in the MRB consists of the coil winding in the electromagnet, which is the magnetic flux generating ”source” (i.e.by generating magnetomotive force or mmf), and the flux carrying path. The path provides resistance over the flux flow, and such resistance is called as reluctance (R). Thus, in Fig. 8, the total reluctance of the magnetic circuit is the sum of the reluctances of the core and the gap, which consists of the MR fluid and the shear disk. Then, the flux generated (Φ) in a member of the magnetic circuit can be defined as[1]:

$$\varphi = ni/R = mmf/R \quad \text{----- (a)}$$

Where;

$$R = 1/\mu A \quad \text{----- (b)}$$

In Eq.(a), n is the number of turns in the coil winding and i is the current applied; in Eq.(b), μ is the permeability of the member, A is its cross-sectional area, and l is its length. Recall that in order to increase the braking torque, the flux flow over the MR fluid needs to be maximized. This implies that the reluctance of each member in the flux path of the flux flow has to be minimized according to Eq.(a), which in turn implies that l can be decreased or/and μ and A can be increased according to Eq.(b)[1].

Electromagnet design mainly consists of two parameters taking into account that is magneto motive force (mmf) & number of turns of wires. Coil is the source (i.e. mmf) in the magnetic circuit. From literature[10], the efficient wire size is the AWG21. More current density can be applied to the MRB using this wire size, which means that the magnetic field intensity generated within the brake will increase. Therefore, the amount of the torque generated will be increased with the increasing

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magnetic field intensity. Therefore, in this work, the AWG 21 size copper wire is used. The specifications of AWG21 is given below[15].

Selected wire : AWG21

diameter of wire = 0.77 mm

maximum current density for AWG21= 2.653A/ mm^2

Therefore, c/s area = 4.66e-7 m^2

Calculated value of braking torque for E-bicycle is 16.172 N-m. Therefore maximum value considered for braking torque(T) = 25 N-m.

As per the calculations presented in the [16], magneto motive force & number of turns are calculated.

$$\tau(H) = \frac{3T}{8\pi(r_z^3 - r_w^3)\lambda} = \frac{3\times 25}{8\pi(0.077^2 - 0.015^3)0.9} = 7.32kPa \quad \text{---(16)}$$

For this value of shear stress, from Fig. 5, referring curve of shear stress vs magnetic field intensity for MRF- 132DG, the value of magnetic field intensity H is 30 kA/m that determines the maximum magnetic flux through MR fluid from B-H Curve of MRF-132DG (Fig. 9), B = 0.3Tesla. The permeability of MR fluid is considered as constant and equal to μf = 5μ0 = 20πe-7 H/m.

$$NI = B_f \times \zeta / \mu_f \quad \text{---------(17)}$$

Where,

NI = Magneto motive force (Amp-turns)

B_f = Magnetic flux density of MR fluid (Tesla)

ζ = Total Gap of fluid (m)

μ_f = Permeability of fluid = (20 π * 10^{-7} H/m) [17]

MR fluid magnetic flux density is taken from B-H curve shown in Fig. 9.

NI = 0.3 * 4 * 10^{-3}/ (20 π * 10^{-7})

NI = 190.98 ≈ 200 Amp-turns.

The product NI is named magneto motive force (mmf) of the magnetic circuit. Then number of turns (N) is calculated for maximum value of current(I). The maximum value considered for current is 1.5 Amp.

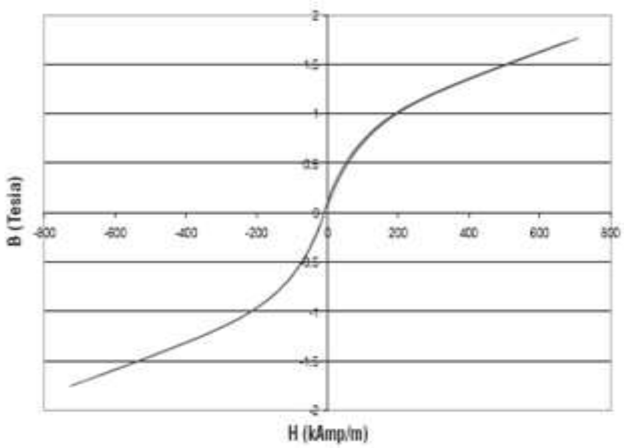

Fig. 9 B-H curve for MR fluid 132 DG[1]

NI = 200

N = 200/1.5 = 133.33 ≈ 135 turns

Then for various values of current, values of current density are calculated, shown in Table 13.

Table 13. Current density for various values of current

I (A)	Current density (A/mm^2)
0.2	0.43
0.4	0.86
0.6	1.29
0.8	1.72
1.0	2.15
1.2	2.58
1.4	3

Calculation of MRF Volume:

The analytical expressions for the calculation of volume of MR fluid have presented in [7].

$$V = \pi \times R^2 \times H \quad \text{---(18)}$$

Where;

V = Volume of MRF (mm^3)

R = Radius of the disc (mm)

H = Effective width of MR effect developed by MRF (mm)

Effective width of MR effect developed by MRF is calculated from following expression:

$$H/R = \frac{3}{4} \times \frac{T_B}{T_\eta} \times \eta \frac{\omega}{\tau} \quad \text{---- (19)}$$

Where;

T_B / $T\eta$ = Control Torque Ratio (15 to 20)

η = Viscosity of fluid (0.112 ± 0.02 Pa-s)

ω = Angular Speed of the wheel (rad/s)

τ = Permissible yield stress (45 kPa)

Where;

$\omega = V/r$

V = velocity of the the vehicle (m/s) [V= 6.94 m/s for 25 km/hr]

r = radius of the wheel (m)

ω = 6.94 / 0.228 = 30.43 rad/s

Calculations:

H/66 = 3/4 * 15∗ 0.112 ∗ 30.43 /45 ∗ 10^3

H = 0.0527 mm.

Therefore,

V = π * 66^2 * 0.0527

V = 721.18 mm^3

Above calculation is done for the speed of 25 km/hr & having control torque ratio is 15.

For control torque ratio 20, results are tabulated as below & also for the speed of 30 km/hr same calculations have done & results are as tabulated below:

Table 14. Volume of MRF

Sr. No.	Speed (km/hr)	Angular Velocity(ω) (rad/s)	Control torque Ratio (T_B / T_η)	Volume of MRF (mm^3)
1	25	30.43	15	721.18
			20	962.03
2	30	36.54	15	866.32
			20	1155

Selection of Seals:

Sealing of the MRB is another important design criterion. MR fluid is highly contaminated due to the iron particles in it, the risk of sealing failure is increased. MR fluid leakage would occur if the fluid was repetitively solidified (due to the repetitive braking) around the vicinity of the seals[1]. Therefore to make the brake leakage proof, sealing is very essential. So during the manufacturing of brake, it would be necessary to choose the appropriate seals. In this work, Viton 'O' rings are found most suitable.

CONCLUSION

This work is mainly concerned with the design and analysis of MR brake for two wheeler application. A theoretical study on MR fluid and MR brake have been carried out. The MR brake has been designed for E-bicycle, which is light in weight and has low braking torque requirement. The required braking torque has been calculated for E-bicycle. From analytical calculations, the value of braking torque has been determined which is equal to 16.172 Nm. The selected configuration of MR brake is acceptable in terms of braking torque, but it would be required to reduce the weight of brake. Therefore it would be required to optimize this design in order to reduce the weight of brake while obtain the sufficient braking torque

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Author Biblography

	**Bhau K. Kumbhar** The author has completed his M.Tech in Automobile and he has two and half years of Teaching Experience while one year of Industrial Experience Email: