ANALYSIS OF A CYLINDRICAL MAGNETORHEOLOGICAL FLUID BRAKE

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Abstract: *A magnetorheological (MR) fluid brake is a device that transmits torque by the shear force of an MR fluid. An MR rotary disk brake has the property that its braking torque changes quickly in response to an external magnetic field strength. This paper presents theoretical considerations of the design alternative for usual simple MR fluid brake. The equation of the torque transmitted by the MR fluid within the brake is derived to provide the theoretical foundation in the cylindrical design of the brake. Based on this equation, after mathematical manipulation, the calculations of the volume, thickness and width of the annular MR fluid within the cylindrical MR fluids brake are yielded.*

Keywords: Magnetorheological fluid, Brake, Construction, Torque, Volume.

INTRODUCTION

Early research of magnetorheological fluids (MRF) dates from 1948 [1], in early papers and later in the patent of Jacob Rabinow [2]. MRF are suspensions comprised out of tree main components, micro-sized magnetic particles, carrier fluid and anti-settling components. Nowadays MRF are usually comprised out of spherical carbonyl iron particles (1 ÷ 10 µm in diameter), mineral or synthetic oil and a variety of additives for inhibiting particle settling and agglomeration, reducing friction, preventing particle oxidation and wear.

Devices based on MRF include dampers, brakes, clutches, polishing devices, hydraulic valves, seals, composite structures, etc.

MRF are type of a smart materials that responds in a specific manner when they are exposed to external magnetic field. MRF rheological and tribological characteristics can be precisely set by means of magnetic field.

In the absence of magnetic field, MRF is indeed non-colloidal fluid, but after the magnetic field is introduced, MRF particles are being polarized and form chain-like structures, thereby restricting fluid flow. The change is rapid, reversible and controllable with the magnetic field strength [6]. The particle chains are oriented parallel to the magnetic field as presented in Figure 1.

Figure 1. Chain-like structure formation in controllable fluids [4]

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MRF can be only in one, out of two states, at the moment. If no magnetic field is applied, MRF is in "OFF-state". On the other hand, if magnetic field is applied, MRF is in "ON-state". Key MRF feature is the yield stress, both in the OFF-state and in ON-state. In OFF-state MRF is in a free-flow state and acts as a Newtonian fluid [5], thus its behavior is often represented as shown in (1)

$$\tau = \eta \cdot \dot{\gamma} \tag{1}$$

τ - yield stress (Pa),
η - dynamic viscosity (Pa s),
$\dot{\gamma}$ - shear rate ($\frac{1}{s}$).

MRF DEVICES OPERATIONAL PRINCIPLES

The operation modes of MRF devices can be divided as follows: flow mode (fixed plate mode, valve mode), shear mode (clutch mode), squeeze mode (compression mode) and any combination of these three [3].

The flow mode is characterized by the fact that MRF is forced to flow between two static plates by means of pressure drop. In this mode, flow resistance can be controlled by changing intensity of the magnetic field, whose main vector is perpendicular to MRF flow direction. Application examples can be found in servo valves, dampers and shock absorbers.
In the shear mode, MRF is located between two non-stationary plates. Plates can either have rotating or sliding movement and MRF flows because of this movement. As before, here as well, main magnetic vector is perpendicular to MRF flow direction. Shear stress and shear rate can be controlled by magnetic field intensity, like in flow mode. Examples of the shear mode include clutches, brakes, chucking and locking devices, dampers and structural composites [3].
Squeeze mode, flow of MRF is achieved by moving parallel plates away from or to each other. MRF devices are being used in small amplitude dumpers, for vibration isolation. Relatively high forces can be achieved in this mode. Characteristics of MRF can be controlled by magnetic field intensity, as well. Magnetic field direction is perpendicular to oscillating plates.

All cylindrical MRF brakes operate in the shear mode. As explained before there are two plates (surfaces) from which at least one has a shear movement. MRF brake system consists of rotor and stator, two surfaces. The rotor is usually inside the stator and is connected with shaft on which the braking torque is to be applied, Fig 4. The stator is linked to the rest of the system and can act as housing as well. MRF is located between rotor and stator and slides on their surfaces, causing small but noticeable viscous friction.

When no external magnetic field is applied, the MRF flows freely between two surfaces. But when applied, in every ferromagnetic particle a magnetic dipole is induced. When particles are allied with main magnetic vector direction, they start to attract each other. If ferromagnetic particles are perpendicular to main magnetic vector, they are repelling each other, while being forced to align with main magnetic field vector. Because of induced magnetic dipoles, particles attract and thus form chain-like structures, parallel with direction of the main magnetic vector, as presented in Figure 1.

CYLINDRICAL MRF BRAKES FUNDAMENTALS AND CONSTRUCTION

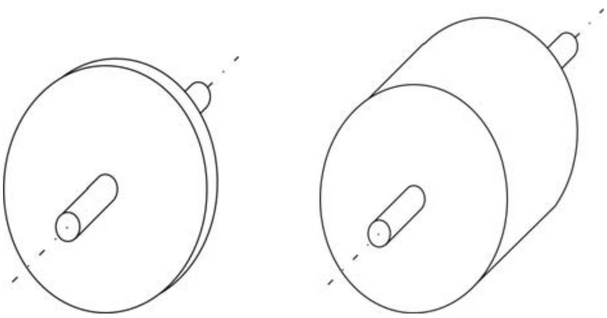
MRF brakes can be divided into five main subcategories:
i. drum,
ii. inverted drum,

iii. T-shape,
iv. disk and
v. multi-disk [4].

Each of the five types has the same parts,
i. rotor, stator,
ii. MRF,
iii. magnetic coil and
iv. housing, (depending on construction)

In this paper, only disk MRF brake and its main parts will be considered.
All five types have the same working principles and they all work in the shear mode. Figure 4 shows main parts of the MRF disk brake. Notice that MRF is located between rotor and stator and that the value of MRF width is much smaller than the widths of the stator and the rotor.

There are two basic forms of the disc type MRF brake. Both forms have disk (i.e. rotor), but the discs heights are different. Both forms are presented in Figure 2. The only difference between these two types is that the second type of disc has much larger mass and therefore much larger inertia, which can prove to be a drawback in applications such as this. Except this difference, construction and the rest of the parts are the same in both cases.



a) *b)*

Figure 2. Two types of discs for MRF brake.

To obtain relationship for braking torque it is necessary to adopt several restrictions
i. First, only disk with large diameter to height ratio will be considered.
ii. Second, braking torque relationship is divided into three parts: rim and two flanks.
iii. Third, main magnetic vector direction is parallel to brake's shaft and the intensity of the magnetic field increases as we move away from shaft to outer rim [7, 8].

The last part is especially important when taken into account the effect of centrifugal force on MRF and its concentration in the exploitation process.

THE RELATIONSHIP BETWEEN RHEOLOGY AND PARTICLE POLARIZATION

Figure3 shows two adjacent particles (dipoles) as a part of a chain along with the direction of applied magnetic field. The interaction energy of the two dipoles of equal strength $|m|$ and direction is [17]:

$$E_{12} = \frac{|m|^2 \left(1 - 3\cos^2\theta\right)}{4\pi\mu_r\mu_0 |r|^3} = \frac{|m|^2 \left(1 - 3\dfrac{r_0^2}{r_0^2 + x^2}\right)}{4\pi\mu_r\mu_0 \left(r_0^2 + x^2\right)^{\frac{3}{2}}} \tag{2}$$

where μ_r is the relative permeability of the medium i.e. dispersion.

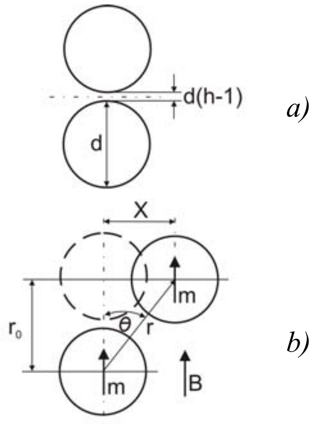


Figure 3. a) Geometry of two particles of diameter d within a particle chain, b) magnetic interaction of the two particles modeled as dipole moments $|m|$ and sheared with respect to one another [18].

In Figure3, θ represents the angle of shear strain, *r* is particles center-to-center distance and *x* represents actual value of shear strain. By defining the scalar shear strain of the particle chain as $\in=\frac{x}{r_0}$, the interaction energy E_{12} between two particles can be written as:

$$E_{12}=\frac{\left(\in^2-2\right)|m|}{4\pi\mu_r\mu_0 r_0^{\ 3}\left(\in^2+1\right)^{\frac{5}{2}}} \tag{3}$$

Now it is assumed that the particles are aligned in long chains and that there is only magnetic interaction between adjacent particles within the chain, i.e., there are no multipole interactions. Then the total energy density (energy per unit volume) associated with the one dimensional shear strain can be calculated by multiplying the particle-to-particle energy, given by equation (III), by the total number of particles and dividing by the total volume:

$$U=\frac{3\phi\left(\in^2-2\right)|m|^2}{2\pi^2\mu_r\mu_0 d^3 r_0^3\left(1+\in^2\right)^{\frac{5}{2}}} \tag{4}$$

where ϕ is the volume fraction of particles in the dispersion and *d* is the particle diameter.

TORQUE

The OFF state of MRF is the state when no magnetic field is present. Absence of magnetic field makes MRF to acts as a Newtonian fluid [13, 14] which means that fluid has no yield stress and can freely flow between rotor and stator. From here the share stress can be presented as (5):

$$\tau=\eta\cdot\dot{\gamma} \tag{5}$$

τ- tangential stress, *η* – dynamic viscosity and $\dot{\gamma}$ - tangential velocity vector i.e. shear rate. But if this state is changed by introducing a magnetic field, MRF can not be described by Newtonian law any more but instead the Bingham's expression can be used [10, 11, 12]:

$$\tau = \tau_B + \eta \cdot \dot{\gamma} \tag{6}$$

As mentioned earlier disk MRF brakes have a three part torque equation. First two parts of braking torque equation are obtained from flank sections of MRF brake. There are two identical flank sections of MRF in MRF brake, as presented in Figure 4. Because of its symmetry, we will concentrate at only one flank of MRF. Second section that is to be described is the rim section of MRF disc brake. This section position is between the outer rim of the rotor and the inner rim of the stator, Figure 4.

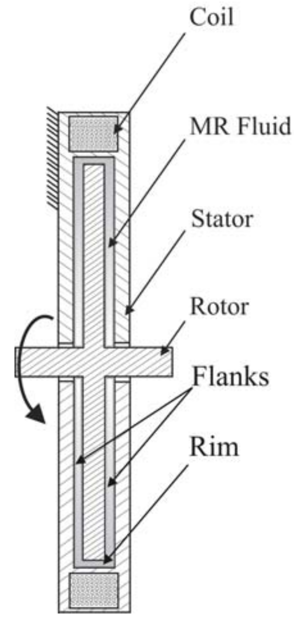


Figure 4. Main components of a MRF disc brake.

MRF flank section is situated between rotor and stator, but this time direction of the main magnetic flux is perpendicular to the surfaces of the rotor and the stator, thus creating ideal setting for MRF movement in shear mode. Simplified shape of MRF flank section is presented in Figure 5.

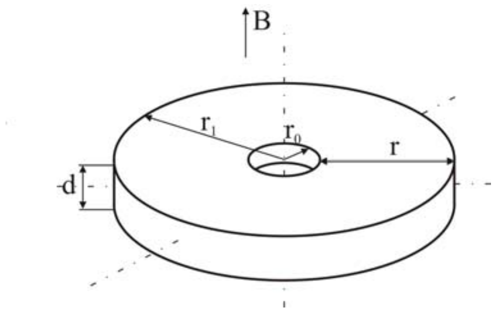


Figure 5. Simplified shape of the flank section of a MRF in a MRF disc brake.

Flank section of braking torque

Flank section of MRF in a MRF disc brake can be also approximated as presented in Figure 6 [15]. This is mainly because the magnetic field has highest intensity close to its coil and the most of the MRF should be in this region of the magnetic field highest intensity. Torque equation is obtained by dividing annular form into two surfaces. First surface of flank section is perpendicular to main magnetic flux direction and the latter one is parallel to it, Figure 7. Infinitesimal torque dT in infinitesimal part of flank section, dA, equals:

$$dT = r\tau dA \tag{7}$$

r – inner radius of the ring, τdA - infinitesimal force dF.

$$dT = r\tau r\theta dr \tag{8}$$

θ - shearing angle (2π), therefore, former equation can be expressed as follows:

$$dT = r\tau r 2\pi dr \tag{9}$$

$$dT = 2\pi r^2 \tau dr \tag{10}$$

$$T = 2\pi \int_{r_0}^{r_1} r^2 \tau dr \tag{11}$$

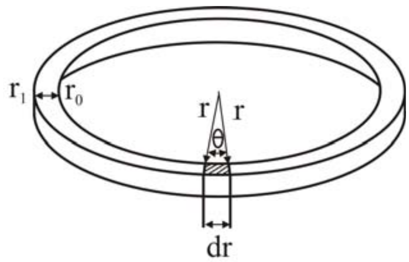


Figure 6. Approximation of the flank section of a MRF with emphasis on side surface.

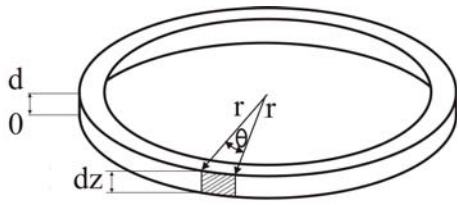


Figure 7. Approximation of the flank section of a MRF with emphasis on rim surface.

Notation (7) to (10) are the same for the second surface of flank section as well, but the difference is visible only at the final step with in the confines i.e. intervals of integration, which are now from 0 to d. Equivalent equation to equation (11) but for second surface is (12).

$$T = 2\pi r^2 \tau \int_{0}^{d} dz \tag{12}$$

Taking into account that the value between *0* and *d* is very small, practically negligible compared to value of radius *r,* this part of equation it will not be consider in this paper.

So the final torque equation of the flank section can be presented as (13) and because there are two flanks it will be multiplied by 2, (14).

$$T = 2\pi \int_{r_0}^{r_1} r^2 \tau dr \tag{13}$$

$$T = 4\pi\int_{r_0}^{r_1} r^2 \tau dr \tag{14}$$

Bingham plastic can not be applied at this point into this equation because the shear rate is too large. Instead, the Herschel-Bulkley model is used [15]:

$$\tau = \tau_B + K\left(\dot{\gamma}\right)^n \tag{15}$$

K – consistency, n – flow index

$$\dot{\gamma} = \frac{r \cdot \omega}{d} \tag{16}$$

ω - angular velocity. To simplify the calculation a few assumptions will be introduced:

i. MRF is incompressible and has laminar flow,
ii. gravitational and centrifugal influence in ON-state on MRF is negligible,
iii. MRF is in direct contact with stator and rotor,
iv. because of the small clearances it is conceder that the MRF has filled total volume intended for it and that there are no "air gaps".

From this point on the final braking torque equation, for flank section, can be formulated, (17). It can be seen that the torque is the radius r and the shear stress τ function.

$$T = 2\pi\int_{r_0}^{r_1} r^2\left(\tau_B + K\left(\frac{r\omega}{d}\right)^n\right)dr \tag{17}$$

$$T = 2\pi\int_{r_0}^{r_1} r^2\tau_B dr + 2\pi\int_{r_0}^{r_1} r^2 K\left(\frac{r\omega}{d}\right)^n dr \tag{18}$$

τ_B is a function of the magnetic field [7].

$$T = 2\pi\tau_B\int_{r_0}^{r_1} r^2 dr + 2\pi K\left(\frac{\omega}{d}\right)^n\int_{r_0}^{r_1} r^{2+n} dr \tag{19}$$

$$T = 2\pi\tau_B\int_{r_0}^{r_1} r^2 dr + 2\pi K\left(\frac{\omega}{d}\right)^n\int_{r_0}^{r_1} r^{2+n} dr \tag{20}$$

After sorting, the final form of the flank section torque equation can emerge in the for of (21).

$$T = \frac{2\pi\tau_B}{3}\left(r_1^3 - r_0^3\right) + \frac{2\pi K\left(r_1\frac{\omega}{d}\right)^{n-1}}{d(n-1)} r_1^4\left(1 - \left(\frac{r_0}{r_1}\right)^{n+3}\right)\omega \tag{21}$$

Rim section of braking torque

Second section that is to be described in torque equation is the rim section of MRF disc brake. The position of this section is between the outer rim of the rotor and the inner rim of the stator, Figure 4. Rim section of MRF can be presented in manner shown in Figure 8. Here main dimensions are presented as well as the main magnetic flux density direction. This part of MRF has poor braking possibilities mainly because of the parallelism between magnetic flux density direction and surfaces of the rotor and the stator. Main magnetic flux density direction is, in this situation, parallel to the outer rotor rim surface and also with inner stator rim surface. Nevertheless, this section can contribute to overall braking torque. For this purpose it is best to reduce the thickness of the MRF rim i.e. dimension S, Figure 8. By doing so, it is possible to increase the chain braking effect and increase the overall braking torque. To get the greatest braking torque possible, rim section of MRF should have dimension ratios similar to the ones presented in Figure 9.

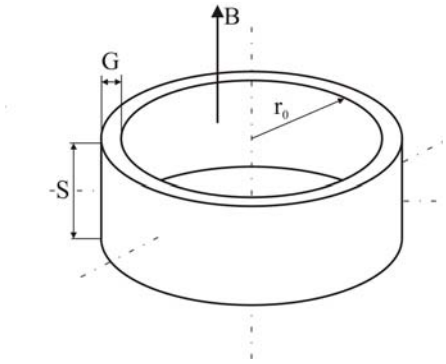


Figure 8. General shape of the rim section of a MRF in a MRF disc brake.

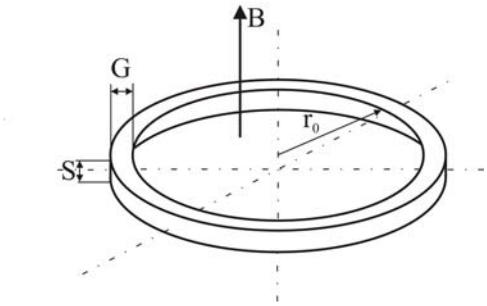


Figure 9. Thinner shape of rim section of a MRF in a MRF disc brake

Principle of expressing the final torque equation for the rim section of MRF disc brake is practically the same as for the flank section. For this part dimensions S and G are used instead of previously used d and r, respectively, see Figure 8. With this in mind, the approximations will be made accordingly. First few steps in developing rim braking torque equation are the same as for the flank section equation, specifically from (7) to (13). In previous section equation (12) was neglected because of the small value of dimension d, this will not be the case this time. Instead we will alter equation (11) by same principles as (12) in earlier section on the count of small difference between r_0 and r_1. Form of MRF brake rim section equation now can be formulated as

$$T = 2\pi r_1^{\,2} \tau \int_0^S ds \tag{22}$$

Further:

$$T = 2\pi r_1^{\,2} \tau S \tag{23}$$

By combining the (23) and the (15) we get:

$$T = 2\pi r_1^{\,2} S\left(\tau_B + K \dot{\gamma}^n \right) \tag{24}$$

τ_B is quite small or equal to zero because the main magnetic flux density direction is parallel with rim section in “S” direction, only influence that remains is the one of shear rate.

$$T = 2\pi r_1^{\,2} S K \dot{\gamma}^n \tag{25}$$

Now the final form of the rim section braking torque equation can be formulated as:

$$T = 2\pi r_1^2 SK\left(\frac{r\omega}{G}\right)^n \tag{26}$$

Overall braking torque

It is now easy to put together overall MRF braking torque, by combining formerly developed parts of equation. Final equation of the overall braking torque is presented as:

$$T = \frac{4\pi\tau_B}{3}\left(r_1^3 - r_0^3\right) + \frac{4\pi K\left(r_1\frac{\omega}{d}\right)^{n-1}}{d(n+3)} r_1^4\left(1-\left(\frac{r_0}{r_1}\right)^{n+3}\right)\omega + 2\pi r_1^2 SK\left(\frac{r_0\omega}{G}\right)^n \tag{27}$$

From here main parts of equation can easily be seen. As mentioned before τ_B is function of magnetic field [16] and can be expressed as follows:

$$\tau_B = kH_{MR}^{\beta} \tag{28}$$

Coefficients k and β are provided by MRF manufacturers. H_{MR}^{β} is magnetic field intensity. It is now possible to provide torque equation as a function of magnetic field, MRF characteristics and geometrical parameters of brake. These three input parameters are in correlation to surrounding conditions of application itself, for example maximum allowed dimensions at mounting place or the price of MRF. Firstly mentioned example can give us a guidelines on MRF brake radii and widths and from there on it is possible to see how much room is there left for the coil.

MRF VOLUME

Minimum of MRF required for effective shear mode device functioning can be represented as follows [12, 19]

$$V = k\left(\frac{\eta}{\tau^2}\right)\lambda W_m \tag{29}$$

where k is a constant. $V = Lwg$, can be regarded as the necessary active fluid volume in order to achieve the desired control ratio λ at a required controllable mechanical power level W_m. Letters in Figure 10 correspond to subscripts used in preceding text. For direct shear: k=1, $\lambda = \frac{F_\tau}{F_\eta}$ and $W_m = F_\tau S$. It is important to note that in this case the minimum active fluid volume is proportional to the product of three terms: a term that is a function of fluid material properties $\frac{\eta}{\tau^2}$, the desired control ratio or dynamic range λ, and the controlled mechanical power dissipation W_m sought. W_m is mechanical power, $W_m = F_\tau\omega$.

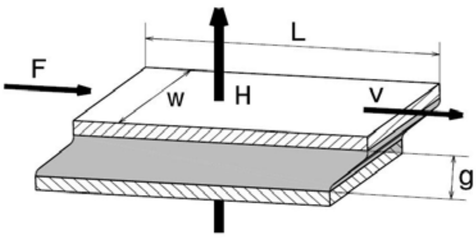


Figure 10. MRF in the shear mode [9]

Form Figure 10, it can be notified that the MRF volume can be calculated from width and height of the two surfaces in contact. If taken into account all previous notations and Figure 10, minimal MRFs height can be presented as (30)

$$d = \left(\frac{\eta}{\tau} \right) \lambda \omega \qquad (30)$$

CONCLUSION

MRF disk brakes are showing a great deal of potential when it comes to modern braking systems. Vehicle brakes have been intensively developed in a past few decades. Geometrical space available for wheel brake on vehicle, is however greatly limited by wheel dimensions. Also, applications such as synchromesh gearbox, clutch, radiator fan etc, have their geometrical limitations as well. Because of this, new types of braking systems are beginning to gain on importance. One of these systems is MRF disk brake.

The most significant MRF brake, the cylindrical MRF disk brake, characteristics are discussed in this paper. A three part overall momentum equation has been developed. Equation consists out of two flanks parts and rim part. Final formulation is a function of shear stress, magnetic field intensity and geometrical parameters.

Basic construction guidelines regarded shape and height to radii ratios have been lay down. At the end a MRF volume and height formulation was derived.

Stated parameters and equation, in this paper, are foundation for future finite element analysis and optimization of MRF disk brakes. Optimization should be conducted in odder to obtain a maximum braking torque that can meet the required braking torques for various automotive applications. Future MRF disc brakes development should be directed toward novel rotor and stator shapes that can utilize magnetic field more effectively.

ACKNOWLEDGMENT

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