A Model of Heat Dissipation for MR based Brake

A Wiehe, V Noack and J Maas
Hochschule Ostwestfalen-Lippe – University of Applied Sciences, Control Engineering and Mechatronics, Liebigstraße 87, D-32657 Lemgo, www.motion-ctrl.de

E-mail: ansgar.wiehe@hs-owl.de

Abstract. In contrast to conventional brakes actuators based on magnetorheological fluids (MRF) offer an advantage in short term, peak load decelerating. The dissipation of a high amount of energy in a short period of time results in a thermal destruction of conventional brakes. Due to the volume based energy dissipation of MR actuators, instead of the surface based energy dissipation of conventional brakes, the rise of temperature and the distribution of energy shows significant advantages. In this paper a design rule for special peak load MR actuators is derived. Furthermore the simplified model, which is the basis of the design rule, is compared to several simulation models, with different levels of detail.

1. Introduction
Electromagnetic brakes and clutches are used for decelerating inertias, controlling force flows and locking positions in applications of industry automation. Functionalities of brakes and clutches can also be realized by smart materials. Especially magneto rheological fluids offer a high potential to substitute conventional braking mechanisms by reducing typical disadvantages like noise and abrasion.

Beside normal operation modes in industrial applications, emergency braking in critical situations is sometimes necessary. Due to the partly high inertias, high braking powers can occur during deceleration. This can result in a thermal overcharging of conventional brakes and thermal destruction of the brake covering. In order to handle this extreme loading on one hand the conventional brakes are oversized and on the other hand an early breakdown is accepted. In the following is shown, that the usage of a MR actuator seems to be an adequate alternative. Therefore a model of energy conversion is presented, which can be used for dimensioning of such MR actuators.

2. Theoretical discussion
To explain the advantages of MR actuators in contrast to conventional brakes the energy balance of both systems is analysed. In braking situation the resulting thermal energy is partly transported to the ambience and partly rising the temperature of braking components. The derivation of the energy balance with respect to the time is given by:

\[ \frac{dW}{dt} = P_B = P_A + P_S, \]  \( (1) \)

where \( P_B \) is the braking power, \( P_A \) the power dissipated to the environment and \( P_S \) the power of the increasing temperature of the reference area. Because conventional and MR brakes have similar
materials covering the braking components, power \( P_a \) can be assumed as comparable for both devices. Furthermore for short time peak load braking the power \( P_a \) can be simplified as zero. The difference between a conventional brake and a MR actuator is caused by the energy assimilation of the braking components, given by

\[
P_a = \rho \cdot V \cdot c \cdot \frac{d\Delta \theta}{dt},
\]

where \( \rho \) is the density of the material, \( V \) the volume, \( c \) the specific heat and \( \Delta \theta \) the excess temperature with respect to the ambient temperature. The volume of energy converting parts in conventional brakes is quite small, that peak loads, which can not be dissipated to the ambience immediately, generate unacceptable high temperature rises. The braking material will be thermally destroyed. As described in previous this effect causes a decreasing durability of brakes stressed by emergency stops. In contrast, the energy converting volume of MR actuators can be arbitrary enlarged by choosing appropriated working gaps. Based on common definitions the energy conversion of braking energy is derived by the following approach. Considering an infinitesimal volume of the MR fluid, shown in figure 1, converted braking energy \( dW_b \) is given by:

\[
dW_b = dM \cdot d\epsilon = r^2 \cdot \tau \cdot d\varphi \cdot dr \cdot d\epsilon,
\]

where \( \tau \) is the yield strength and \( d\epsilon \) the relative movement.

![Figure 1. Infinitesimal Volume of MRF.](image)

Substituting the relative movement \( d\epsilon \):

\[
d\epsilon = \dot{\varphi} \cdot dh \cdot dt,
\]

by the local shear rate \( \dot{\varphi} \) gives the dissipated power \( P_b \) by:

\[
P_b = \int \frac{\partial W_b}{\partial t} = \iiint r^2 \cdot \dot{\varphi} \cdot d\varphi \cdot dr \cdot dh.
\]

Under consideration of a typical geometry of radial disc MR actuators, some simplifications can be made. By ignoring the viscosity of the MR fluid and assuming an uniformly distributed magnetic field, the yield strength \( \tau \) is not a function of the geometry \( \tau \neq f(\varphi, r, h) \). In case of discoidal geometries the radius does not depend on angle or height of the working gap \( r \neq f(\varphi, h) \). Assuming a laminar flow, the shear rate can be expressed by [1]:

\[
\dot{\varphi} = \frac{\omega \cdot r}{h_y},
\]

where \( \omega \) is the angular speed and \( h_y \) the high of the working gap. Using these simplifications the braking power of rotary MR actuators can be expressed by:
where \( r_i \) describes the inner and \( r_o \) the outer radial boundary of the working gap. Combining equation (2) with (7) gives:

\[
\frac{3 \cdot \rho \cdot c \cdot d\Delta \vartheta}{2 \cdot \tau \cdot \omega \cdot \frac{d}{dt}} = \frac{(r_o^3 - r_i^3)}{h_o \cdot (r_o^2 - r_i^2)}.
\]

Equation (8) can be used as a fundamental design rule for MR brakes specially designed for peak loads.

### 3. Simulation studies

Due to the assumptions made by introducing equation (2), the power \( P_A \), which is dissipated to the environment, was neglected. In a first step the resulting derivation is investigated by comparing simulation results of a model based on equation (2) with more detailed models of the heat transfer.

Based on the modification of a lumped system approach [2] a simple model is proposed. It is assumed that one circular surface of the working gap remains at ambient temperature \( \vartheta_a \) and the other surfaces are under adiabatic condition. Furthermore assuming the generation of the braking heat in the centre of the working gap the heat flow \( Q_\Delta \) through the circular surface is given by:

\[
Q_\Delta = \frac{2 \cdot \pi \cdot \lambda_{MRF} \cdot (r_o^3 - r_i^3)}{h_i} \cdot (\Delta \vartheta_{MRF}),
\]

where \( \lambda_{MRF} \) is the heat conductance value of the MRF and \( \Delta \vartheta_{MRF} \) is the excess temperature in the centre of the working gap. With equation (1) the thermal behavior is given by a first-order differential equation:

\[
P_B = \rho_{MRF} \cdot V_{MRF} \cdot c_{MRF} \cdot \Delta \dot{\vartheta}_{MRF} + Q_\Delta.
\]

A further, more detailed model includes the iron parts around the working gap and the convective heat transfer on the actuators surface. The heat transfer in the real actuator is a multidimensional problem. To simplify the heat transfer to a one-dimensional problem, the convective heat transfer of the circular surface is neglected. The way of the mean heat flux through the iron cylinder is assumed at the centre radius in axial direction, on the half height \( h_{iron} \) of the iron cylinder the heat flux faces to the outer surface in radial direction. Using the abbreviations:

\[
C_{MRF} = \rho_{MRF} \cdot V_{MRF} \cdot c_{MRF}, \quad C_{iron} = \rho_{iron} \cdot V_{iron} \cdot c_{iron}, \quad R_{MRF} = \frac{h_i}{2 \cdot \pi \cdot \lambda_{MRF} \cdot (r_o^2 - r_i^2)}, \quad R_{iron,a} = \frac{h_{iron}}{2 \cdot \pi \cdot \lambda_{iron} \cdot (r_o^2 - r_i^2)}, \quad R_{iron,r} = \frac{r_o}{\pi \cdot (r_o + r_i) \cdot \lambda_{iron} \cdot h_{iron}}, \quad R_A = \frac{1}{2 \cdot \pi \cdot \alpha_{conv} \cdot r_o \cdot h_{iron}},
\]

the thermal behavior can be described by a second degree differential equation system [3]:

\[
\begin{bmatrix}
\Delta \dot{\vartheta}_{MRF} \\
\Delta \dot{\vartheta}_{iron}
\end{bmatrix} = \begin{bmatrix}
\frac{1}{C_{MRF} \cdot (R_{MRF} + R_{iron,a})} & \frac{1}{C_{MRF} \cdot (R_{MRF} + R_{iron,a})} \\
\frac{1}{C_{iron} \cdot (R_{MRF} + R_{iron,a})} & -\frac{1}{C_{iron} \cdot (R_{MRF} + R_{iron,a})} + \frac{1}{R_A}
\end{bmatrix} \begin{bmatrix}
(\Delta \vartheta_{MRF}) \\
(\Delta \vartheta_{iron})
\end{bmatrix} + \begin{bmatrix}
1/C_{MRF} \\
0
\end{bmatrix} \cdot P_B.
\]


Using FE methods instead of the discrete modeling method described above is an alternative way for the simulation of the thermal behavior. To compare adequately the results of the FE method and the discrete modeling method an according FE model is generated. The braking power $P_B$ is in general a function of the radius, given by equation (5). For the simplified FE model the braking power is assumed to be constant relating to the volume. The convective heat transfer of the circular surfaces is neglected. The resulting heat flow of the simplified FE model is shown in figure 2.

To investigate the thermal behavior in a more realistic way a second FE model is generated, where the power dissipation given by equation (5) is taken into consideration. Furthermore the geometry of the working gaps is modified to a double gap geometry, which is typical for radial disk brakes. In figure 4 the transient response of the described models is shown. The braking power is assumed as a falling ramp, which is typical for decelerating inertias to zero velocity. The maximum temperatures of the simulation models are lower than the maximum temperature predicted by the design rule model ensuring safety of operation. In comparison to the simulation models the prediction of the model for the design rule is sufficient accurate to use it of the design procedure.

**Figure 2.** Heat flux of simplified FE model.  
**Figure 3.** Heat flux of complex FE model.

**Figure 4.** Loading Case for simulation studies.  
**Figure 5.** Time response of thermal models.

## 4. Conclusion

The theoretical discussion of the energy dissipation of conventional and MR based brakes shows an advantage in rise of temperature and distribution of energy. The thermal destruction of braking components in peak load applications can be avoided by the usage of MR brakes. A design rule, derived with a simplified model, can be used for the design of this kind of MR actuators. A first investigation of accuracy of the simplified model gives a sufficient correlation with more complex simulation models. In a further step the simulation models will be validated by measurements.

## References

[1] Yamaguchi, H.: *Engineering Fluid Mechanics*. Springer Verlag, 2008.
[2] Kavlícoglu, B.M. et al.: *Heating of a High-torque Magnetorheological Fluid Limited Slip Differential Clutch*. Journal of Intelligent Systems and Structures, Vol. 19 – 2008, pp 235.
[3] Wiehe, A.; Kern, S.; Maas, J.: *Rotatorischer MRF-Aktor für einen Türassistenten*. at - Automatisierungstechnik 56 (2008) 3, S. 155-164, Oldenbourg-Wissenschaftsverlag.