Vibration attenuation of rotating machines by application of magnetorheological dampers to minimize energy losses in the rotor support

J Zapoměl¹,², P Ferfecki²,³
¹Department of Dynamics and Vibration, Institute of Thermomechanics, Dolejškova 5, Prague, 18200, Czech Republic
²Department of Applied Mechanics, VSB-Technical University of Ostrava, 17. listopadu 15, Ostrava-Poruba, 70833, Czech Republic
³IT4Innovations National Supercomputing Center, VSB-Technical University of Ostrava, 17. listopadu 15, Ostrava-Poruba, 70833, Czech Republic
jaroslav.zapomel@vsb.cz

Abstract. A frequently used technological solution for minimization of undesirable effects caused by vibration of rotating machines consists in placing damping devices in the rotor supports. The application of magnetorheological squeeze film dampers enables their optimum performance to be achieved in a wide range of rotating speeds by adapting their damping effect to the current operating conditions. The damping force, which is produced by squeezing the layer of magnetorheological oil, can be controlled by changing magnetic flux passing through the lubricant. The force acting between the rotor and its frame is transmitted through the rolling element bearing, the lubricating layer and the squirrel spring. The loading of the bearing produces a time variable friction moment, energy losses, uneven rotor running, and has an influence on the rotor service life and the current fluctuation in electric circuits. The carried out research consisted in the development of a mathematical model of a magnetorheological squeeze film damper, its implementation into the computational models of rotor systems, and in performing the study on the dependence of the energy losses and variation of the friction moment on the damping force and its control. The new and computationally stable mathematical model of a magnetorheological squeeze film damper, its implementation in the computational models of rigid rotors and learning more on the energy losses generated in the rotor supports in dependence on the damping effect are the principal contributions of this paper. The results of the computational simulations prove that a suitable control of the damping force enables the energy losses to be reduced in a wide velocity range.

1. Introduction

Imbalance of the rotating parts belongs to the principal causes of excitation of the lateral vibration of rotors. A frequently used engineering solution for suppression of its undesirable effect on the operation of rotating machines consists in placing damping devices between the rotor and its casing. To achieve their optimum performance, their damping effect must be adaptable to the operating conditions [1]. This is offered by magnetorheological damping devices. Their application is intended especially for rotors the working conditions of which are variable such as a wide extent of running speeds, rotors with changeable configuration adaptable to the needs of the working process (variable
rotor length, tools of different mass attached to the rotor shaft, etc.), which changes their resonance frequencies, or for rotors working in different environments (air, gaseous, liquidy with variable depth of submerging the working tools), which changes the additional inertia effects and consequently the critical speeds.

Many journal articles and conference papers deal with the design, function and experimental investigation of magnetorheological squeeze film dampers, and with their application for vibration attenuation of rotating machines [2], [3], [4]. The study of their efficiency and effect on the vibration reduction of various rotor systems is reported in [5], [6], [7]. Piccirillo et al. [8] dealt with the application of magnetorheological damping devices to suppress impacts and to control the chaotic oscillations in a rotor-bearing system.

Zapoměl et al. [9], [10] developed mathematical models of a short and long squeeze film magnetorheological damper which are based on modelling the magnetorheological oil by Bingham material and are intended for the analysis of both the steady state and transient rotor vibrations. These models were used to study the vibration attenuation of simple rotor systems running at different operating regimes [11]. The extension of this work is represented by the development of a new and enhanced model of a squeeze film magnetorheological damper based on application of a bilinear material to describe the pressure distribution in a thin lubricating layer of magnetorheological oil [12], [13].

In this article, the influence of the controllable damping effect on the energy losses arising in rolling element bearings that are the part of squeeze film dampers is investigated. These losses depend mostly on the force transmission through the support elements. The computational simulations proved that the proper setting of magnitudes of the damping forces enables to minimize the energy losses and to satisfy the requirements put on the maximum allowable amplitude of the rotor vibrations. The extension of the computational algorithms and procedures, their stable numerical behaviour, learning more on applicability of the controllable damping force and extension of the possibilities to achieve the optimum performance of magnetorheological damping devices belong to the principal contributions of the presented work.

2. The magnetorheological squeeze film damper

The principal parts of a squeeze film magnetorheological damper are the inner and outer damper rings between which there is a thin layer of magnetorheological oil (figure 1). The inner ring is coupled with the rotor journal by a rolling element bearing and with the damper housing by a cylindrical squirrel spring. The lateral vibration of the rotor arrives at squeezing the oil film which produces the damping effect. The damper is equipped with electric coils generating magnetic flux, which passes through the lubricating layer. As the resistance against the flow of magnetorheological fluids depends on magnetic induction, the change of the current can be used to control the damping force.

![Figure 1. The magnetorheological squeeze film damper.](image-url)
Magnetorheological oils belong to the class of fluids with a yielding shear stress. It implies their flow occurs only at that part of the lubricating film in which the shear stress between two adjacent layers exceeds a limit value - the yielding shear stress. In areas called core where the shear stress is lower than the limit value, the magnetorheological fluid behaves as a solid material.

Results of the experiments carried out at various working places show that the yielding shear stress of magnetorheological oils depends on magnetic induction and that this dependence can be approximated by a power function

$$\tau_y = k_y B^n, \quad (1)$$

where $\tau_y$ is the yielding shear stress, $B$ is the magnetic induction, and $k_y$, $n_y$ are the proportional and exponential material constants of the magnetorheological oil.

The magnetic flux density in the damper gap depends on the current feeding the coils and on the design arrangement of the damping device. From the engineering distinguishing level point of view, the inner and outer damper rings can be considered as a divided core of an electromagnet with the gap filled with magnetorheological oil. This enables to express a simple relation between the magnetic induction, electric current $I$ and the thickness $h$ of the oil film at the investigated location

$$B = k_B \mu_0 \mu_r \frac{I}{h}. \quad (2)$$

$\mu_0$, $\mu_r$ are the vacuum and relative permeabilities of the magnetorheological oil, respectively. $k_B$ is the design parameter that is defined as a product of the number of the coil turns and magnetic efficiency introduced as a ratio of the magnetic flux passing through the lubricating layer to the total one generated by the electric coils. Its value depends on the damper design and can be determined by solving a magnetic problem referred to the analysed device utilizing, e.g. the finite element method. More details can be found in [14], [15].

The relation for the thickness of the oil film reads [16], [17]

$$h = c - e_H \cos(\varphi - \gamma), \quad (3)$$

where $c$ is the width of the damper clearance (when the inner and outer damper rings take a centric position), $e_H$ is the rotor journal centre eccentricity, $\varphi$ is the circumferential coordinate (figure 2), and $\gamma$ is the position angle of the line of centres (figure 2).

**Figure 2.** The damper coordinate systems.
3. Determination of the magnetorheological damping forces

The developed mathematical model of the magnetorheological damping device is referred to the dampers that are symmetric with respect to the plane perpendicular to the journal axis and whose geometric and design parameters enable to consider them as short [16], [17]. The pressure distribution in the full oil film is determined on the assumptions of the classical theory of lubrication [16], [17], except for the lubricant. As the magnetorheological oil belongs to the class of fluids with a yielding shear stress, in the mathematical model it is represented by a bilinear material the yielding shear stress of which depends on magnetic induction.

![Figure 3. The cross section of the oil film with the constituted areas.](image_url)

Two areas are constituted in the cross section of the lubricating film (figure 3). In the one closer to the damper centre, the core extends across the whole thickness of the oil film and touches the rings surfaces. In the areas closer to the damper ends, a region where the oil flows occurs, too. A series of derivations starting from the equation of equilibrium of an infinitesimal element specified in the oil film, the equation of continuity and the constitutive relationship referred to bilinear material described in more details in [12] arrives at the Reynolds equation adapted for bilinear material which governs the pressure distribution in the full oil film (4) - (6), the relation for the axial coordinate $Z_c$, which defines the border between the two regions (7) and at the relation for the pressure gradient in the axial direction at that location (8). The coordinate systems defining displacement of the rotor journal and positions in the oil film are depicted in figures 2 and 3.

\[
\frac{\partial}{\partial Z} \left( \frac{1}{\eta_c} h^3 p' \right) = 12h, \quad (4)
\]
\[
\frac{\partial}{\partial Z} \left[ \frac{1}{\eta} \left( h^3 p' + 3h^2 \tau_c + 8 \frac{\tau_c^2}{p'^2} - 12 \frac{\tau_c \tau_c}{p'^3} \right) - 8 \frac{\tau_c \tau_c}{\eta_c p'^3} \right] = 12h \quad \text{for} \quad w > 0 \quad (h < 0), \quad (5)
\]
\[
\frac{\partial}{\partial Z} \left[ \frac{1}{\eta} \left( h^3 p' - 3h^2 \tau_c - 8 \frac{\tau_c^2}{p'^2} + 12 \frac{\tau_c \tau_c}{p'^3} \right) + 8 \frac{\tau_c \tau_c}{\eta_c p'^3} \right] = 12h \quad \text{for} \quad w < 0 \quad (h > 0). \quad (6)
\]
\[
Z_c = -\frac{\tau_c h^3}{6 \eta_c h}, \quad (7)
\]
\[
p_c' = -\frac{2\tau_c}{h}. \quad (8)
\]

$p$ is the pressure, $p'$ stands for the pressure gradient in the axial direction, $Z$ is the axial coordinate defining position in the oil film (figures 2, 3), $w$ is the flow velocity in the axial direction, $\tau_c$ is the
shear stress at the core border and \( \eta_c, \eta \) are the dynamic viscosities of the oil in and outside the core area, respectively, \( p_c' \) denotes the pressure gradient in the axial direction at a location given by the axial coordinate \( Z_c \) and \( (.) \) notates the first derivative with respect to time.

The Reynolds equation (4) and equations (5) and (6) hold for the extents of the axial coordinates \( 0 \leq Z \leq Z_c \) and \( (Z_c < Z \leq L/2) \), respectively. \( L \) denotes the damper length. More information on solving the Reynolds equations modified for bilinear material can be found in [12].

In that part of the damper gap in which the thickness of the oil film rises with time \( (\dot{h} > 0) \) a cavitation is assumed. In cavitated areas, the Reynolds equation does not hold. In accordance with the results of the observations carried out at different working places, the pressure of the medium in cavitated regions is supposed to be constant and equal to the pressure in the ambient space.

Components of the magnetorheological damping forces are calculated by the integration of the pressure around the damper circumference and along its length in the axial direction taking into account a different pressure distribution in noncavitated and cavitated areas:

\[
F_{mry} = -2K \int_0^{2\pi} p_d \cos \varphi \, d\varphi \, dZ,
\]

\[
F_{mrz} = -2K \int_0^{2\pi} p_d \sin \varphi \, d\varphi \, dZ.
\]

\( F_{mry}, F_{mrz} \) are the \( y \) and \( z \) components of the hydraulic damping force, \( p_d \) is the pressure distribution in the lubricating layer and \( R \) is the middle radius of the damper gap.

4. A friction moment in the rolling element bearing in a squeeze film damper

A friction moment arising in rolling element bearings depends on their type, material of the races, and rolling elements, on the transmitted force, kind and material parameters of the lubricant (oil, grease, oil mist, etc.), and on angular speed of the rotor rotation. With reference to [18] it can be expressed as a sum of two components

\[
M_{BF} = M_{BFR} + M_{BFS},
\]

where

\[
M_{BFR} = 0.001 G_{BR} \left( \frac{3}{\pi} \times 10^7 \nu \omega \right)^{0.6},
\]

\[
M_{BFS} = 0.001 G_{BS} f_{BS}.
\]

\( M_{BF} \) is the resulting friction moment in the bearing, \( M_{BFR} \) is the friction moment caused by the resistance against the rolling, \( M_{BFS} \) stands for the moment produced by the slide friction caused by the rolling elements slips, \( \nu \) is the kinematic viscosity of the oil lubricating the bearing, \( \omega \) is the angular speed of the journal (inner bearing race) rotation, \( G_{BR}, G_{BS} \) are the bearing coefficients and \( f_{BS} \) is the friction coefficient the value of which depends on the lubricant and type of the bearing [18] (for ball bearings \( f_{BS} = 0.4 \div 1.0 \)).

The relations for the determination of the bearing parameters \( G_{BR}, G_{BS} \) can be found in [18]. For ball bearings not axially loaded they have the form

\[
G_{BR} = R \left( 1000d_m \right)^{0.96} F_{BR}^{0.54},
\]

\[
G_{BS} = R \left( 1000d_m \right)^{0.96} F_{BS}^{0.54}.
\]
\[ G_{RS} = S_1 \left(1000 \ d_m^5 \right)^{0.26} F_{BR}. \]  \quad (15)

\( F_{BR} \) is the radial force transmitted through the bearing, \( d_m \) is the mean bearing diameter and \( R_1, S_1 \) are the bearing geometric constants. Their values depend on the type and size of the rolling bearing and can be found in [18] (for ball bearings \( R_1 = 3.5 \times 10^{-7} \div 5.5 \times 10^{-7}, S_1 = 2.0 \times 10^{-3} \div 6.5 \times 10^{-3} \)).

The power loss \( P_{BF} \) in the bearing is then equal to the product of a friction moment and angular speed of the rotor rotation

\[ P_{BF} = M_{BF} \omega. \]  \quad (16)

5. The investigated rotor system

The investigated rotor is rigid (figure 4). It is composed of a shaft and of one disc. The rotor journals are mounted on rolling element bearings that are pressed in the inner rings of the magnetorheological squeeze film dampers. The rotor rotates at constant angular speed and is loaded by its weight and excited by the disc imbalance. The whole system can be considered as symmetric relative to the disc middle plane perpendicular to the shaft axis. The squirrel springs of the damping elements are prestressed to have their deflection caused by the rotor weight eliminated. The rotor is driven by an electric motor which is connected to one of its ends by a coupling, which allows small radial displacements of the rotor journal caused by its lateral vibration.

![Figure 4. The investigated rotor system.](image)

The task was to analyse the rotor steady state vibrations, the force transmission between the rotor and its casing, the energy losses, and the friction moment caused by the rolling element bearings in dependence on the damping force produced by the magnetorheological dampers in a specified range of the rotational speeds.

In the computational model the rotor, its frame, the rolling elements bearings, and the damper rings are considered as absolutely rigid. The magnetorheological dampers are implemented by linear springs and nonlinear force couplings which represent acting of the oil layers on the rotor journals.

Because of the system symmetry, lateral vibration of the rotor is governed by two nonlinear motion equations

\[ m\ddot{y} + b_p \dot{y} + 2k_p y = me_r \omega^2 \cos \alpha t + 2F_{my}, \]  \quad (17)

\[ m\ddot{z} + b_p \dot{z} + 2k_p z = me_r \omega^2 \sin \alpha t - mg + 2F_{mz} + 2F_{PS}. \]  \quad (18)

\( m \) is the rotor mass, \( k_p \) is the stiffness of one squirrel spring, \( b_p \) is the coefficient of the rotor damping produced by the environment, \( g \) is the gravity acceleration, \( e_r \) is the eccentricity of the rotor centre of gravity, \( y, z \) are the horizontal and vertical displacements of the rotor centre, \( t \) is the time, \( F_{PS} \) is the
force prestressing one squirrel spring in the vertical direction, and (\cdot) notates the second derivative with respect to time.

The trigonometric collocation method was applied to obtain the steady state solution of the equations of motion.

The radial force transmitted through the rolling element bearing is equal to the sum of the hydraulic damping force by which the oil layer acts on the rotor journal and the damper housing, the elastic force caused by the deformation of the squirrel spring and of the force prestressing the squirrel springs to eliminate their stationary deflection caused by the rotor weight

\[
F_{BRy} = -F_{sw} + k_D y, \quad (19)
\]
\[
F_{BRe} = -F_{sw} + k_D z - F_{PS}, \quad (20)
\]

where \(F_{BRy}, F_{BRe}\) are the y and z components of the radial force loading the rolling element bearing.

6. Results of the simulations

The technological and operation parameters of the studied rotor system are: the rotor mass 426 kg, the squirrel spring stiffness 2.0 MN/m, the coefficient of the rotor external damping 5 Ns/m, the rotor imbalance 21.3 kg.mm, the magnetorheological squeeze film damper length/diameter 40/150 mm, the width of the damper gap 0.9 mm, the oil dynamic viscosity not effected by a magnetic field 0.3 Pa-s, the oil dynamic viscosity in the core 300 Pa-s, the magnetorheological oil proportional and exponential constants 2000, 1.1, respectively, its relative permeability 5, the damper design parameter 60, the mean diameter of the rolling element bearing 120 mm, dynamic viscosity and density of the oil lubricating the rolling element bearing 0.002 Pa-s, 860 kg.m\(^{-3}\), the friction coefficient referred to the slip between the balls and the bearing races 0.6 and the bearing rolling and sliding geometric constants \(R_1, S_1\) \(4 \times 10^{-7}, 3 \times 10^{-3}\), respectively.

The time histories of the forces transmitted between the rotor and its frame in the horizontal and vertical directions referred to angular speed of the rotor rotation 200 rad/s and applied current 0.4 A are depicted in figure 5. The mean value of the vertical component is not zero because the rotor is loaded by its weight. The corresponding time histories of the friction moment and the mean power loss in one rolling element bearing can be seen in figures 6 and 7.

![Figure 5](image-url)  
**Figure 5.** The forces transmitted to the rotor frame (\(\omega = 200 \text{ rad/s}, I = 0.4 \text{ A})

The frequency response of the rotor on excitation caused by the imbalance in the range of operating speeds 0 - 1000 rad/s and for the applied currents 0.0, 0.1, 0.4 and 1.0 A are drawn in figure 8. The
corresponding dependences of the maximum friction moment and the mean power loss in one rolling element bearing on angular speed of the rotor rotation are depicted in figures 9 and 10. It is evident that the rising current enables to attenuate significantly the vibration amplitude in the whole extent of the operating speeds. On the other hand, the transmitted force and the energy losses are reduced only in the speed interval approximately lower than the critical velocity. In the range of the angular speeds higher than the critical one rising current leads to a considerable increase of the transmitted forces and the energy losses.

![Figure 6](image1.png)

**Figure 6.** Time history of the friction moment \((\omega = 200 \text{ rad/s}, I = 0.4 \text{ A})\).

![Figure 7](image2.png)

**Figure 7.** Time history of the power loss \((\omega = 200 \text{ rad/s}, I = 0.4 \text{ A})\).

As evident from the frequency responses drawn in figure 8, the current of 1.2 A would have to be applied to keep the vibration amplitude below the allowable value of 45 \(\mu\text{m}\) in the whole extent of the operating speeds 0 - 1000 rad/s. This would overload the bearings in the interval of lower angular velocities and would arrive in increase of the friction moment. A suitable control of the applied current in dependence on the rotational speed makes it possible to minimize the force transmitted through the rolling element bearings and thus to reduce the energy losses. The proposed speed - electric current relationship is depicted in figure 11. The corresponding dependences of the vibration amplitude and of the minimum and maximum friction moment during one rotor revolution on the speed of the rotor rotation in one bearing are drawn in figures 12 and 13. Figure 14 shows a comparison of dependences of the power loss on the rotor angular velocity related to the operating regimes when the current remains constant having the value of 1.2 A and when magnitude of the current is controlled. The
reduction of the power loss is evident from figure 15. The results show that efficiency of this operation manipulation depends on the speed of the rotor rotation. In the investigated case, its maximum is reached for the angular velocity of about 770 rad/s.

**Figure 8.** The frequency response ($I = 0.0, 0.1, 0.4, 1.0$ A).

**Figure 9.** Relation of the maximum friction moment on the speed of rotation ($I = 0.0, 0.1, 0.4, 1.0$ A).

**Figure 10.** Relation of the mean power loss on the speed of rotation ($I = 0.0, 0.1, 0.4, 1.0$ A).
Figure 11. The current control: the proposed speed - electric current relation.

Figure 12. Dependence of the vibration amplitude on the rotor angular speed (current controlled).

Figure 13. Dependence of the friction moment on the rotor angular speed (current controlled).
7. Conclusions
The presented research work offers a novel study on the application of magnetorheological squeeze film dampers. Its goal is to perform the investigation on minimizing the energy losses in the rotor support elements, which arise due to the forces transmitted between the rotor and its casing. The developed computational procedures use a bilinear material with the yielding shear stress depending on magnetic flux density to model the magnetorheological oil. The simulations proved that their numerical behaviour is considerably more stable than of those based on application of Bingham material. The extension of the efficient computational procedures, the evidence that a proper control of magnitudes of the damping forces makes it possible to reach the optimum compromise between minimizing the energy losses in the bearing elements, and satisfying the requirements put on the allowable amplitude of the rotor oscillations in a wide range of operating speeds and learning more on the influence of the magnetorheological squeeze film dampers on the oscillatory behaviour of rotors are the principle contributions of the work presented in this paper.

Acknowledgement: This work has been supported by the Czech Science Foundation 15-06621S and the National Programme of Sustainability project „IT4Innovations excellence in science - LQ1602“.
References
[1] Zapoměl J, Ferfecki P and Kozánek J 2013 Determination of the transient vibrations of a rigid rotor attenuated by a semiactive magnetorheological damping device by means of computational modelling Applied and Computational Mechanics vol 7(2) pp 223-234
[2] Gong X, Ruan X, Xuan S, Yan Q and Deng H 2014 Magnetorheological Damper Working in Squeeze Model Advances in Mechanical Engineering, vol 6 410158
[3] Forte P, Paterno M and Rustighi E 2004 A Magnetorheological Fluid Damper for Rotor Applications International Journal of Rotating Machinery vol 10(3) pp 175-182
[4] Kim K J, Lee C W and J.H. Koo J H 2008 Design and modeling of semi-active squeeze film dampers using magneto-rheological fluids Smart Materials and Structures, vol 17(3) 035006
[5] Jian X, Li-dong H, Kai W and Xiu-jin H 2015 Optimizing control of a two-span rotor system with magnetorheological fluid dampers Journal of Beijing Institute of Technology vol 24(4) pp 558-565
[6] Carmignani C, Forte P and Rustighi E 2006 Design of a novel magneto-rheological squeeze-film damper Smart Materials and Structures vol 15(1) pp 164-170
[7] Aravindhan T S and Gupta K 2006 Application of Magnetorheological Fluid Dampers to Rotor Vibration Control Advances in Vibration Engineering vol 5(4) pp 369-380
[8] Piccirillo V, Balthazar J M and Tuset A M 2015 Chaos control and impact suppression in rotor-bearing system using magnetorheological fluid The European Physical Journal Special Topics vol 224(14) pp 3023-3040
[9] Zapoměl J, Ferfecki P and Forte P 2012 A computational investigation of the transient response of an unbalanced rigid rotor flexibly supported and damped by short magnetorheological squeeze film dampers Smart Material and Structures vol 21 105011
[10] Zapoměl J and Ferfecki P 2010 Mathematical modelling of a long squeeze film magnetorheological damper for rotor systems Modelling and Optimization of Physical Systems vol 9 pp 97-10
[11] Zapoměl J, Ferfecki P. and Forte P 2013 A Computational Investigation of the Steady State Vibrations of Unbalanced Flexibly Supported Rigid Rotors Damped by Short Magnetorheological Squeeze Film Dampers ASME Journal of Vibration and Acoustics vol 135 064505
[12] Zapoměl J and Ferfecki P 2015 A 2D Mathematical Model of a Short Magnetorheological Squeeze Film Damper Based on Representing the Lubricating Oil by Bilinear Theoretical Material Proceedings of the 2015 IFToMM World Congress (Taipei)
[13] Zapoměl J, Ferfecki P and Kozánek J 2015 Modelling of magnetorheological oils in rotordynamic damping devices by bilinear material Proceedings of the ICoEV 2015 International Conference on Engineering Vibration (Ljubljana) pp 563-570
[14] Zapoměl J and Ferfecki P 2010 A numerical procedure for investigation of efficiency of short magnetorheological dampers used for attenuation of lateral vibration of rotors passing the critical speeds Proceedings of the 8th IFToMM International Conference on rotodynamics (Seoul) pp 1-8
[15] Zapoměl J and Ferfecki P 2015 Analysis of the vibrations attenuation of rotors supported by magnetorheological squeeze film dampers as a multiphysical finite element problem Proceedings of the conference Výpočty konstrukcí metodou konečných prvků (Prague) pp 79-80
[16] Zapoměl J 2007 Computer modelling of lateral vibration of rotors supported by hydrodynamical bearings and squeeze film dampers (Ostrava: VSB-Technical University of Ostrava) in Czech
[17] Szeri A Z 1980 Tribology: Friction, Lubrication, and Wear (Washington, New York, London: Hemisphere, Publishing Corporation)
[18] SKF Rolling bearings catalogue, http://www.skf.com/binary/56-121486/SKF-rolling-bearings-catalogue.pdf