Multi-physical Analysis of the Forces in the Flexible Rotor Supported by the Magnetorheological Squeeze Film Dampers

P. Ferfecki, J. Zapoměl and M. Marek

Abstract To reduce lateral vibrations of rotating machines, the damping devices are placed between the rotor and its frame. This is enabled by magnetorheological squeeze film dampers. To achieve their optimum performance their damping effect must be adaptable to the current working conditions. In this paper, modelling of the magnetorheological squeeze film damper is based on the assumptions of the classical theory of lubrication except that for the lubricant. The model is completed with the magnetic force acting between the damper rings. Therefore, the magnetostatic approaches to determining the magnetic field in the damper gap at different distinguishing levels were proposed and compared. The developed mathematical model was applied for analysis of a rotating machine with a flexible shaft. The carried out computational simulations confirmed that the developed magnetorheological damper arrives at significant suppression of the rotor vibration in a wide range of running speeds.

Keywords Magnetorheological squeeze film damper · Hydraulic forces · Magnetic forces · Magnetostatic analysis · Flexible rotor · Vibration attenuation

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1 Introduction

Lateral vibration of flexible rotors can be significantly reduced if damping elements are inserted into its support. For this purpose, magnetorheological (MR) squeeze film dampers can be applied. Their damping effect is controlled by changing induction of the magnetic field passing through the film of the MR fluid.

The mathematical model of a short squeeze film MR damper intended for the analysis of both the steady state and transient rotor vibrations is presented in [1]. Modelling of the MR damper that takes into account both the hydraulic fluid film and magnetic forces is reported in [2, 3].

The development of the enhanced mathematical model of the MR squeeze film damper for rotordynamic applications and learning more on the main forces acting in between the rotor and the MR damping devices are the principal contributions of this article. For this purpose, two magnetostatic modelling approaches have been developed, and the semi-analytical relations defining the magnetic forces acting in the damper were determined.

2 The Hydraulic and Magnetic Forces Between the Damper Rings

The main parts of MR squeeze film dampers (Fig. 1a) for rotordynamic applications are two concentric rings between which there is a layer of MR oil. The outer ring is mounted with the stationary part directly, and the inner one is connected with the squirrel spring carrying the ball bearing. The spring enables the oscillation of the inner damper ring in the radial direction and prevents its rotation together with the shaft. In the damper body the electric coil generating magnetic flux passing through the lubricating oil is imbedded. As its resistance against the flow depends on magnetic induction, the change of the electric current makes it possible to control the hydraulic damping force.

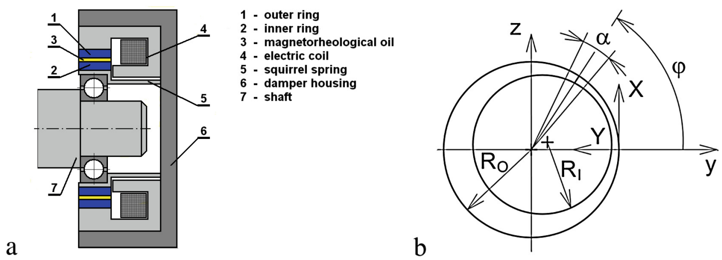

Fig. 1 The scheme of a MR squeeze film damper (**a**) and the damper (xyz) and the fluid film (XYZ) coordinate systems (**b**)

The mathematical model of the studied damping element is based on utilization of the classical theory of lubrication with the exception for the MR oil that is represented by Bingham material, the yielding shear stress of which depends on magnetic induction. Then the pressure distribution in the full oil film referred to a short [1] damper is governed by a Reynolds equation modified for Bingham fluid

$$h^3 p'^3 \pm 3(h^2 \tau_y \mp 4\eta \dot{h} Z) p'^2 \mp 4\tau_y^3 = 0 \quad \text{for } Z > 0. \tag{1}$$

The upper and lower signs in (1) hold for the pressure gradients $p' < 0$, $p' > 0$, respectively. Z is the axial coordinate (Fig. 1b), h is the oil film thickness [4], τ_y, η are the yield shear stress and viscosity of the Bingham liquid, and $(\cdot)$ denotes the first derivative with respect to time. The details on derivation and solution of the modified Reynolds equation (1) can be found in [1].

The stationary value of the yielding shear stress of the MR fluid needed for solving the modified Reynolds equation (1) can be approximated with sufficient accuracy by a power function

$$\tau_y = k_y B^{n_y}. \tag{2}$$

B is the magnetic induction in the damper gap, and k_y, n_y are the material constants of the MR oil.

The mutual interaction between the damper rings is accomplished by the hydraulic and magnetic forces. The hydraulic force is produced due to squeezing the MR oil film and pushes the rings one from another. Its components are obtained by integration of the hydraulic pressure distribution around the circumference and along the length L of the damper

$$F_{MRy} = -2R_I \int_0^{\frac{L}{2}} \int_0^{2\pi} p_d \cos\varphi \, d\varphi \, dZ, \; F_{MRz} = -2R_I \int_0^{\frac{L}{2}} \int_0^{2\pi} p_d \sin\varphi \, d\varphi \, dZ. \tag{3}$$

F_{MRy}, F_{MRz} are the y and z components of the hydraulic force, R_I is the inner ring radius, p_d is the pressure distribution in the MR oil film, and φ denotes the circumferential coordinate (Fig. 1b). In cavitated areas, the pressure of the medium is assumed to be constant and equal to the pressure in the ambient space.

Contrary to the hydraulic damping force, the magnetic force is attracting and is induced by the magnetic flux generated in the electric coils.

The 3D approach uses a spatial distribution of the magnetic flux to determine magnetic induction in the damper gap and the force acting between the rings. As the damper permanent magnetization is avoided, the saturation of the magnetic flux and hysteresis are not taken into account. The magnetostatic problem is assumed to be linear and the finite element method was applied for its solving. The discretization of the damper body and the surrounding air environment is evident from Fig. 2a. The total magnitude of the magnetic force acting between the rings in the direction

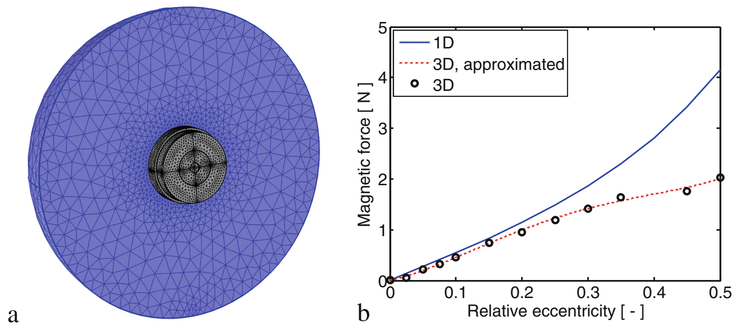

Fig. 2 Discretization of the damper model (**a**) and the magnetic force in dependence on the relative eccentricity (**b**)

of the line of the center was calculated by means of the Maxwell stress tensor and is drawn in Fig. 2b.

At the lowest distinguishing level (1D approach), the inner and outer damper rings can be considered as a divided core of an electromagnet. The semi-analytical relations for the magnetic induction and the magnetic force are based on magnetic reluctance [5] of the MR fluid film of a sufficiently small angular segment between two coaxial cylinders

$$B_i = k_B \mu_0 \mu_r \frac{I}{h_i}, \quad F_{MA,i} = \frac{B_i^2 \alpha L (R_I + R_O)}{4 \mu_0 \mu_r}, \quad i = 1, 2, \ldots, \frac{2\pi}{\alpha} \tag{4}$$

B_i is the magnetic induction in the center of the damper gap and in the center of the angular segment, $F_{MA,i}$ is the magnetic force between the angular segment of the inner and the outer damper rings, I is the electric current, R_O is the outer damper ring radius, h_i is the oil film thickness in the center of the angular segment of the damper gap, k_B is the design parameter, α is the angular segment of the damper gap (Fig. 2b), μ_0 is the vacuum permeability ($4\pi \times 10^{-7}$ H m^{-1}), and μ_r is the magnetic permeability of the MR fluid.

The design parameter was determined from the value of magnetic induction in the middle of the oil film obtained by application of the 3D approach to solve the magnetic problem for the case when the damper rings take a concentric position.

The dependence of the total magnetic force acting between the centers of the inner and outer damper ring computed by application of the 1D approach on the relative eccentricity is drawn in Fig. 2b. It is evident that the difference between the values (Fig. 2b) provided by the 3D and 1D approaches is less than 15 % in the relative eccentricity range 0.0–0.2 in which the MR damper operates.

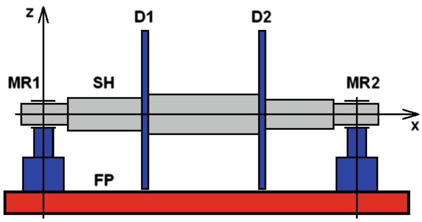

Fig. 3 The scheme of the investigated rotor system

3 The Motion Equation of the Investigated Rotor System

The investigated rotor system (Fig. 3) consists of a flexible shaft (SH) and two discs (D1, D2). The rotor is mounted on a rigid frame (FP), and the squeeze film MR dampers (MR1, MR2) are inserted into its supports.

The task was to investigate the influence of the magnetic attractive and carrying hydraulic damping forces on the vibration amplitude of the rotor.

The rotor turns at constant angular speed, is loaded by its weight, and excited by imbalance of the discs. The mass, diameter, and axial moments of inertia of the discs are considered. The squirrel springs of both dampers are prestressed to be eliminated their deflection caused by the rotor weight.

In the computational model the shaft is considered as flexible and is represented by a beam-like body discretized into finite elements. The discs are assumed to be absolutely rigid and the MR dampers are represented by springs and nonlinear force couplings.

Lateral vibration of the investigated rotor system is described by the equation of motion that takes the form

$$\mathbf{M}\,\ddot{\mathbf{q}} + (\mathbf{B} + \eta_{\mathbf{V}}\,\mathbf{K}_{\mathbf{SH}} + \omega\,\mathbf{G})\,\dot{\mathbf{q}} + (\mathbf{K} + \omega\,\mathbf{K}_{\mathbf{C}})\,\mathbf{q} = \mathbf{f}_{\mathbf{A}} + \mathbf{f}_{\mathbf{PS}} + \mathbf{f}_{\mathbf{MR}} + \mathbf{f}_{\mathbf{MA}}. \tag{5}$$

$\mathbf{M}$, $\mathbf{B}$, $\mathbf{K}$, $\mathbf{G}$, $\mathbf{K}_{\mathrm{C}}$, $\mathbf{K}_{\mathrm{SH}}$ are the mass, damping, stiffness, gyroscopic, circulation matrices of the rotor, and the stiffness matrix of the shaft, respectively, $\mathbf{f}_{\mathrm{A}}$ is the vector of applied and unbalance forces, $\mathbf{f}_{\mathrm{PS}}$ is the vector of prestressed forces, $\mathbf{f}_{\mathrm{MR}}$ is the vector of hydraulic MR forces, $\mathbf{f}_{\mathrm{MA}}$ is the vector of magnetic forces, $\mathbf{q}$ is the vector of generalized displacements, η_{V} is the coefficient of viscous damping of the shaft material, ω is the angular speed of the rotor rotation, and $(\cdot)$ denotes the second derivative with respect to time.

A trigonometric collocation method was applied to obtain the steady state solution of the equation of motion (5).

4 Computational Results of the Rotor Dynamics

The frequency responses depicted in Fig. 4a is referred to the center of D1 disc. The responses were determined for the cases when no oil was supplied to the dampers (no damping), and when the MR oil was solidified (over damped).

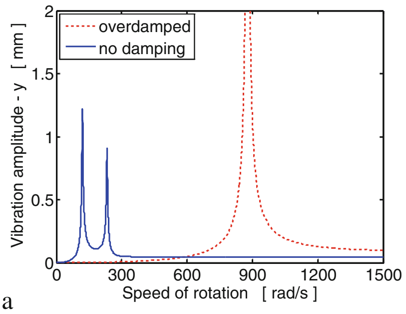

Fig. 4 The frequency responses (**a**) and the vibration amplitude (**b**) in the horizontal direction

The dependence of the vibration amplitude at location of D1 disc referred to the angular speed of the rotor rotation of 120 rad/s on the applied current is drawn in Fig. 4b. The results show that the rising magnitude of the current increases reduction of the vibration amplitude. The computation of the magnetic force by the 3D approach gives almost the same vibration amplitude, and the difference between the 1D and 3D approaches is less than 10 %.

5 Conclusions

It was shown that the magnetic attractive force in the MR damper is much smaller than the hydraulic damping one. The main contribution of this article is the development of a more accurate mathematical model of a short MR squeeze film damper and the procedures for determining the magnetic force based on solving a spatial magnetostatic problem utilizing the finite element method and on application of a semi-analytical relation.

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