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Modelling of magnetorheological squeeze film dampers for vibration suppression of rigid rotors

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ABSTRACT

The magnetorheological squeeze film damping devices for vibration suppression of rigid rotors are studied in this article. The development of their mathematical model is based on assumptions of the classical theory of lubrication with the exception of lubricant. Because the magnetorheological fluids affected by a magnetic field belong to the class of liquids with a yielding shear stress, the lubricant is represented by bilinear theoretical material. The pressure distribution in the full oil film is then described by a modified Reynolds equation. In addition, the influence of cavitation and of the magnetic forces, by which the damping device acts on the rotor journal, were taken into account. The advantage of the developed mathematical model is that, unlike the Bingham or Herschel-Bulkley materials, the flow curve of the bilinear liquid is continuous. It reduces the nonlinear character of the damping forces and thus raises the numerical stability of the computational procedures. The solution convergence is reached also in cases when the procedures based on modelling the magnetorheological fluid by Bingham or Herschel-Bulkley materials fail. Application of bilinear material provides a better description of physical behavior of magnetorheological oils affected by a magnetic field during the damping process. The simulations show that changing magnetic induction in the lubricating film makes it possible to achieve optimum performance of the damping device in a wide range of the rotor operating speeds and confirms increased numerical stability of the computational procedures.

1. Introduction

To reduce amplitude of the lateral vibrations of rotors excited by unbalanced rotating parts, damping devices added to the rotor supports are frequently used. To achieve their optimum performance in a wide range of rotational speeds their damping effect must be controllable, as widely discussed in [1].

Several mechanical, hydraulic and electromagnetic principles applicable to control the damping forces are reported in the literature. Mu et al. proposed a normal squeeze film damper with a conical gap [2]. The control of the damping effect is achieved by changing the geometric parameters of the damper gap which is accomplished by shifting the outer damper ring in the axial direction. El-Shafei and El-Hakim [3] developed a design solution that controls the damping forces by changing the axial position of the damper end sealings. Application of a reluctance type of electromagnetic damper in combination with a rolling element bearing is reported in [4,5].

A completely new concept of controllable damping devices is based on the utilization of sensitivity of some lubricating fluids to electric or magnetic fields (electrorheological, magnetorheological fluids). The recent results of the research carried out in the field of rheology, structure, composition, physical properties and flow of electrorheological and magnetorheological fluids are reported in [6–11]. The study of behavior of magnetorheological fluids at high shear rates is presented in [12]. A micromechanical model for magnetorheological fluids is introduced in [13].

The principle of electrorheological squeeze film dampers applicable in the field of rotordynamics and their mathematical model are presented in [14]. Many journal articles and conference papers such as [15–19] deal with the design, function and experimental investigations of magnetorheological squeeze film dampers and with their application for the vibration attenuation of rotating machines. The investigation of their efficiency and effect in reducing vibration of various rotor systems is reported e.g. in [20–25]. Piccirillo et al. [26] dealt with the application of magnetorheological damping devices to suppress the impacts and control the chaotic oscillations of a rotor-bearing system.

Zapoměl et al. [27,28] developed mathematical models of a short

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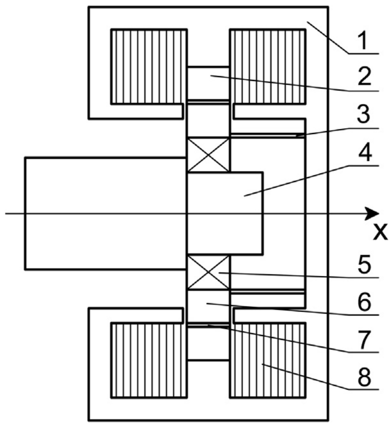

Fig. 1. A magnetorheological squeeze film damper: 1 – damper housing; 2 – outer ring; 3 – squirrel spring; 4 – shaft; 5 – ball bearing; 6 – inner ring; 7 – magnetorheological fluid; 8 – electric coil.

and long squeeze film magnetorheological damper which are based on modelling the magnetorheological oil by Bingham material and intended for 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 [29].

An alternative to magnetorheological dampers are elastomeric damping devices which utilize magnetorheological properties of specially prepared elastomeric materials for vibration attenuation. More details on performance of these damping devices can be found in [30].

The magnetorheological oils when effected by a magnetic field behave as fluids with a yielding shear stress. This means their flow occurs only if the shear stress between two adjacent layers exceeds a limit value - a yielding shear stress. In areas, called a core, where the yielding shear stress is not reached, the magnetorheological liquids exhibit properties close to solid materials.

The main parts of a magnetorheological squeeze film damper are two concentric rings between which there is a layer of magnetorheological oil (Fig. 1). The inner ring is coupled with the rotor journal by a rolling element bearing and with the damper housing by a squirrel spring. The lateral vibration of the rotor squeezes the oil layer which produces the damping effect. The damping device is equipped with an electric coil generating magnetic flux passing through the lubricating oil. As resistance against its flow depends on magnetic induction, change of the applied current can be used to control the damping force.

In mathematical models the magnetorheological liquids are usually represented by Bingham or Herschel-Bulkley materials. In this paper the development of a new and enhanced mathematical model of a short magnetorheological squeeze film damper based on representing the magnetorheological oil by bilinear material is reported. Unlike the Bingham fluid, its flow curve is continuous which contributes to reducing the nonlinear character of the damping forces. In addition, the influence of magnetic forces by which the damping device acts on the rotor journal and thus influences the damping effect is taken into account. The performed computational simulations show that the new mathematical model increases the computational stability of the procedures for determination of the steady state vibrations of rotors compared to those based on modelling the magnetorheological oil with Bingham material. Another advantage is that the flow curve referred to a bilinear material describes better behavior of real magnetorheological oils than the flow curve related to Bingham material.

The development of a new, enhanced mathematical model of a short magnetorheological squeeze film damper; increase of computational stability of the procedures in which it is implemented; considering the influence of magnetic forces on the damping process; and learning more on the effect of magnetorheological damping devices on the vibration attenuation of rotating machines, are the main contributions of the presented article.

This paper is organized as follows. In Section 2, the basic equations governing the hydraulic pressure and the specific magnetic force are derived. In Section 3, the relations for the hydraulic and magnetic forces acting on the rotor journal are determined. In Section 4, the motion equations of the investigated rigid rotor are presented. Section 5 provides results of the computational simulations and their evaluation. Some concluding remarks are drawn in Section 6.

2. Hydraulic pressure in the magnetorheological oil film

The newly developed mathematical model of a magnetorheological squeeze film damper is based on assumptions of the classical theory of lubrication [31,32], except for the lubricant. The magnetorheological fluid is represented by bilinear material whose yielding shear stress depends on magnetic induction. In addition, it is assumed that the damper is symmetric relative to the plane perpendicular to the shaft axis and that its geometric and design parameters allow it to be considered as short [31,32]. This means the prevailing flow in the lubricating layer occurs in the axial direction, and therefore its circumferential component can be neglected.

The lubricating layer is thin. It enables deriving the relations for the pressure distribution in the full oil film from the equation of continuity (1), the equation of equilibrium of an infinitesimal element specified in the oil film (2) and from the constitutive relationship with reference to bilinear material (3)–(5):

$$\frac{\partial w}{\partial Z} + \frac{\partial v}{\partial Y} = 0, \tag{1}$$

$$\frac{\partial p}{\partial Z} = \frac{\partial \tau}{\partial Y}, \tag{2}$$

$$\tau = +\tau_y + \eta \frac{\mathrm{d}w}{\mathrm{d}Y} \quad \text{for} \quad \tau_C < \tau, \tag{3}$$

$$\tau = \eta_C \frac{\mathrm{d}w}{\mathrm{d}Y} \quad \text{for} \quad -\tau_C \le \tau \le \tau_C, \tag{4}$$

$$\tau = -\tau_y + \eta \frac{\mathrm{d}w}{\mathrm{d}Y} \quad \text{for} \quad \tau < -\tau_C, \tag{5}$$

where p is the pressure, τ is the shear stress, Y, Z are the radial and axial coordinates defining positions in the oil film (Fig. 2), v, w are the radial and axial velocity components of the oil flow, τ_y is the yielding shear stress, τ_C is the shear stress at the core border and η_C, η are the dynamic viscosities of the oil in and outside the core area respectively.

The velocity boundary conditions correspond with the assumption that the oil perfectly adheres to the surface of the inner and outer damper ring

$$w = 0, \ \ v = 0, \ \text{for} \quad Y = 0, \tag{6}$$

$$w = 0, \ \ v = \dot{h}, \ \text{for} \quad Y = h, \tag{7}$$

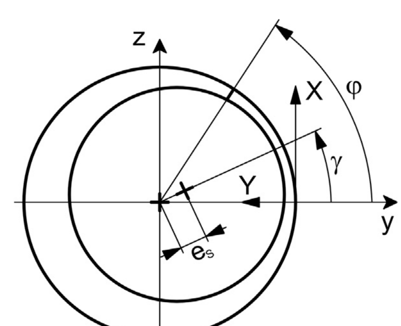

Fig. 2. The damper (xyz) and the fluid film (XYZ) coordinate systems: φ is the circumferential coordinate, γ is the angle of the line of centers, and e_S is the journal center eccentricity.