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ANALYSIS OF INFLUENCE OF SHORT MAGNETORHEOLOGICAL DAMPING ELEMENTS ON VIBRATIONS ATTENUATION OF RIGID ROTORS

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Abstract. *A frequently used technological solution for reducing lateral vibrations of rotating machines as a result of their unbalance consists in adding damping devices to the rotor supports. To achieve their optimum performance their damping effect must be adaptable to the current operating conditions. This is offered by magnetorheological squeeze film dampers. Magnetorheological oils belong to the class of liquids with a yielding shear stress. In the developed mathematical model of a short magnetorheological squeeze film damper the lubricant is represented by Bingham material. Then the pressure distribution in the lubricating layer is governed by the Reynolds equation adapted for Bingham fluid. The yielding shear stress depends on magnetic induction. Its stationary value can be approximated by a power function and its dependence on its time history in the past (characterized by a time constant) by a convolution integral that is consequently transformed to a linear differential equation of the first order. Efficiency of the studied magnetorheological damper was investigated by means of computational simulations. The obtained results show that the rising magnetic flux passing through the layer of the magnetorheological oil arrives at increasing attenuation of the rotor vibrations. On the other hand the damping effect is going down with increasing magnitude of the delayed yielding time constant.*

1 INTRODUCTION

A technological solution frequently used to reduce lateral vibrations of rotors induced by their unbalance consists in adding damping devices to the rotor supports. To achieve their optimum performance their damping effect must be adaptable to the current operating conditions. There are several possibilities based on different physical phenomena how to control the damping force. Some mechanical and hydraulic approaches are discussed in [1-2]. A new concept is represented by application of squeeze film dampers lubricated by magnetorheological oils. The derivation of the relations governing the pressure distribution in the lubricating layer can be found in [3-4]. Some results of the experimental research of the vibration attenuation of a small test rotor damped by magnetorheological dampers are reported in [5].

Magnitude of the damping forces developed by magnetorheological damping devices depends on formation of a chain structure of ferromagnetic particles dispersed in carrying oil by acting of a magnetic field. This paper deals with the study of effect of the time history of magnetic induction on the formation process and thus on efficiency of attenuation of the rotor lateral vibrations.

2 DETERMINATION OF THE DAMPING FORCES

The main parts of magnetorheological squeeze film dampers (Fig. 1) are two concentric rings. The inner one is firmly fixed to the damper housing while the outer one is coupled with the rotor journal by a rolling element bearing and with the damper body by a squirrel spring. The gap between the rings is filled with magnetorheological oil. The rotor journal lateral vibration arrives at squeezing the lubricating layer which produces the damping effect. An important part of the damper is an electric coil generating magnetic flux passing through the lubricating layer. As resistance against the flow of magnetorheological fluids depends on magnetic induction, the damping effect can be controlled by the change of electric current feeding the coil.

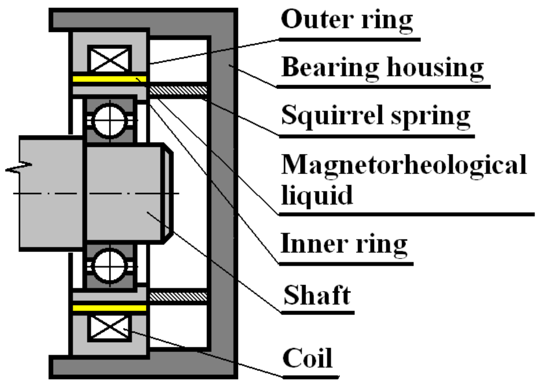

Figure 1: The arrangement of a magnetorheological squeeze film damper.

The developed mathematical model of a magnetorheological squeeze film damper is based on assumptions of the classical theory of lubrication [6-7] except those for the lubricant. As the magnetorheological oil behaves as liquid with a yielding shear stress, it is modelled by Bingham material. In addition, it is assumed that the damper is symmetric relative to its middle plane perpendicular to the journal axis and that its design and geometry make it possible to consider it as short [6-7].

Then the pressure distribution in the thin film of the lubricating oil is described by the Reynolds equation modified for Bingham material [3]

$$h^3 p'^3 + 3\left(h^2\,\tau_y - 4\eta_B\,\dot{h}\,Z\right)p'^2 - 4\tau_y^3 = 0 \text{ for } Z > 0 \tag{1}$$

where

$$p' = \frac{\partial p}{\partial Z} \tag{2}$$

p denotes the pressure, p' is the pressure gradient in the axial direction, h is the thickness of the oil film, τ_y, η_B are the yielding shear stress and the Bingham viscosity, Z is the axial coordinate referred to the frame of reference describing positions in the oil film (Fig. 2) and ($\dot{}$) denotes the first derivative with respect to time. The details on calculation of the film thickness h can be found in [6-7].

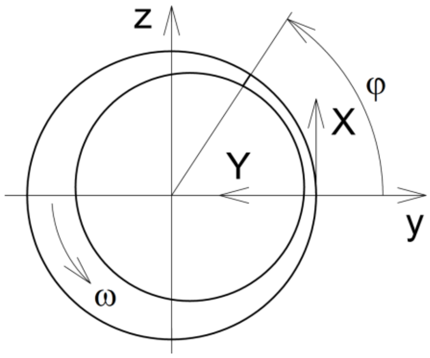

Figure 2: The damper coordinate system.

The Reynolds equation (1) holds only at locations where the beeing squeezed lubricant flows in the direction outside from the damper gap (where the pressure gradient is negative). In areas where the clearance between the inner and outer rings rises with time the lubricant is sucked into the gap from the ambient space and therefore, occurrence of a cavitation is assumed there. In cavitated regions a constant pressure equal to the pressure in the outer space is supposed.

With regard to the assumed damper symmetry, the damping force components are calculated by integration of the pressure distribution p_d in the oil film taking into account a different pressure determination in non cavitated and cavitated regions

$$F_{dy} = -2R\int_0^{2\pi}\int_0^{\frac{L}{2}} p_d \cos\varphi \,\mathrm{d}Z\,\mathrm{d}\varphi\,, \tag{3}$$

$$F_{dz} = -2R\int_0^{2\pi}\int_0^{\frac{L}{2}} p_d \sin\varphi \,\mathrm{d}Z\,\mathrm{d}\varphi\,. \tag{4}$$

F_{dy}, F_{dz} are the y and z components of the damping force, R is the middle radius of the damper gap, L denotes the length of the damping device and φ is the circumferential coordinate.

3 DETERMINATION OF THE YIELDIG SHEAR STRESS

Magnetorheological fluids are formed by carrying liquid in which tiny ferromagnetic particles of micrometre size are randomly dispersed. After application of a magnetic field they start to form a chain structure which causes that the liquid starts to flow only after the shear stress between two adjacent layers exceeds a limit magnitude (the yielding shear stress).

The steady state value of the yielding shear stress τ_{ys} can be expressed by a power function of magnetic induction B

$$\tau_{ys} = k_y B^{n_y} . \tag{5}$$

k_y, n_y are the proportional and exponential material constants of the magnetorheological oil.

The formation of the chain structure of ferromagnetic particles is a process that is rapid but not instantneous. This implies the yielding shear stress depends on the whole time history of magnetic induction in the past which can be suitably described by a convolution integral

$$\tau_y = k_y \int_0^t \frac{1}{T_y} e^{-\frac{1}{T_y}(t-\vartheta)} B^{n_y} \,\mathrm{d}\vartheta . \tag{6}$$

Relation (6) is consequently transformed to the differential form

$$T_y \, \dot{\tau}_y + \tau_y = \tau_{ys} . \tag{7}$$

ϑ denotes the time in the past history before the moment of time t and T_y is the time constant of the delayed yielding process whose value is usually of order of units of miliseconds.

The design arrangement of the damper makes it possible to consider the inner and outer rings as a divided core of an electromagnet with the gap filled with magnetorheological oil. Then the magnetic induction in the lubricating layer can be expressed

$$B = k_B \mu_0 \mu_r \frac{I}{h} . \tag{8}$$

μ_0 is the vacuum permeability, μ_r is the magnetorheological fluid relative permeability, I is the applied current and k_B is the design parametre whose value depends on the number of the coil turns and on the design arrangement of the damping device. After inserting (8) into (5) and introducing the damper design parametre k_{DP}

$$k_{DP} = k_y \left(k_B \mu_0 \mu_r \right)^{n_y} \tag{9}$$

the steady state value of the yielding shear stress reads

$$\tau_{ys} = k_{DP} \left(\frac{I}{h} \right)^{n_y} . \tag{10}$$

4 THE INVESTIGATED ROTOR SYSTEM

The analyzed rotor is rigid. It consists of a shaft and of one disc (Fig. 3). At both its ends it is supported by rolling element bearings and squeeze film magnetorheological dampers. The rotor turns at constant angular speed and is loaded by its weight and unbalance. The squirrel springs of the damping elements are prestressed to be eliminated their deflection resulting from the rotor dead loading.

The task was to investigate the rotor lateral vibrations after the current feeding the magnetorheological dampers is switched on and to analyze influence of the delayed yielding process on amplitude of the resulting transient and steady state oscillations.

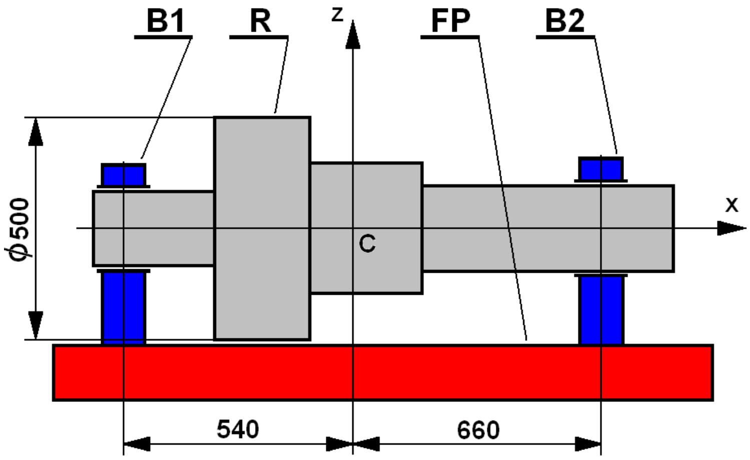

Figure 3: Scheme of the investigated rotor.

In the computational model the rotor is represented by an absolutely rigid body and the dampers by spring elements and nonlinear force couplings. Axis x of the introduced coordinate system is identical with the shaft centre line in the rotor undeformed position and its origin is placed in its intersection with the plane perpendicular to the shaft axis going through the rotor centre of gravity. The rotor can perform four independent motions, displacements in two mutually perpendicular directions (in the directions of axes y and z) and two rotations about y and z axes. Therefore, its oscillations are governed by a set of four equations of motion

$$m\ddot{y} = -b_{TP}\dot{y} + F_{S1y} + F_{S2y} + me_T\omega^2 \cos(\omega t + \psi_T)\,, \tag{15}$$

$$m\ddot{z} = -b_{TP}\dot{z} + F_{S1z} + F_{S2z} + F_{PS1} + F_{PS2} - mg + me_T\omega^2 \sin(\omega t + \psi_T)\,, \tag{16}$$

$$J_D\ddot{\varphi}_y + \omega J_A\dot{\varphi}_z = -b_{RP}\dot{\varphi}_y + F_{S1y}x_{TS1} - F_{S2y}x_{TS2}\,, \tag{17}$$

$$J_D\ddot{\varphi}_z - \omega J_A\dot{\varphi}_y = -b_{RP}\dot{\varphi}_z - (F_{S1z} + F_{PS1})x_{TS1} + (F_{S2z} + F_{PS2})x_{TS2}\,. \tag{18}$$

m is the rotor mass, J_D and J_A are the diametre and axial moments of inertia the rotor, b_{TP} , b_{RP} are the external damping coefficients referred to the rotor translational and rotational motions, x_{TS1}, x_{TS2} are the distances between the rotor supports and the rotor centre (intersection of the shaft centreline and the perpendicular plane going through the rotor centre of gravity), ω is angular speed of the rotor rotation, e_T is eccentricity of the rotor unbalance, g is the gravity acceleration, F_{S1y} is F_{S1z} are the y and z components of the support force (hydraulic and elastic) acting on the shaft at location of support B1, F_{S2y} is F_{S2z} are the y and z components of the support force (hydraulic and elastic) acting on the shaft at location of support B2, F_{PS1}, F_{PS2} are the prestress forces at supports B1, B2, ψ_T is the phase shift of the unbalance force, y, z are the y and z displacements of the rotor centre, φ_y, φ_z are the rotor rotations about axes y and z and $(\ddot{\ })$ denotes the second derivative with respect to time.

To solve the motion equations the Adams-Moulton direct integration method was applied.

5 THE SIMULATIONS RESULTS

The technological and geometric parameters of the studied rotor are: the rotor mass 425.9 kg, the rotor diametre and axial moments of inertia 18.9 kgm^2, 9.9 kgm^2 respectively, the external damping coefficients referred to the rotor translational and rotational motions 10 Ns/m, 10 Nms/rad respectively, the squirrel spring stiffness 30 MN/m, the damper diametre/length 150/60 mm, the damper clearance 1 mm, the oil dynamical viscosity (no influence of a magnetic field) 0.3 Pas, the exponential coefficient of the yielding shear stress in its dependence on magnetic induction 2, the damper design parameter 0.001 N/A^2 and the rated speed of the rotor rotation 400 rad/s. At the moment of time of 0.1 s the electric current feeding the coils of both magnetorheological dampers is switched on and reaches its final value of 2.5 A during the time period of 10 ms.

Fig. 4 shows the steady state orbits of the rotor centre (R) and of the journal centres in supports B1 and B2 before switching the current on. The orbits are concentric as the squirrel springs of both dampers are prestressed to be eliminated their deflection caused by the rotor weight.

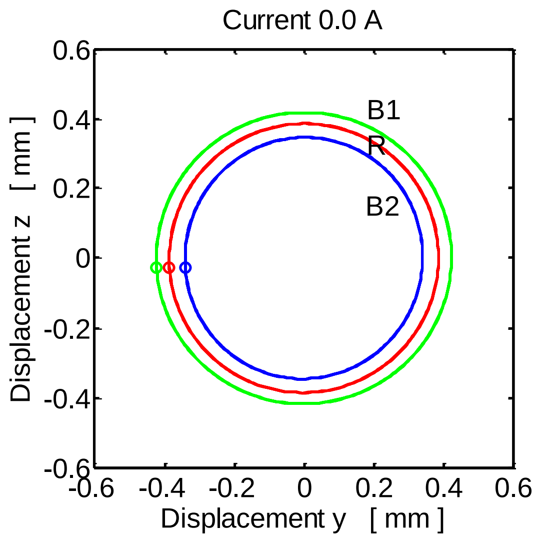

Figure 4: Orbits of the rotor (R) and journal centres (B1, B2) before switching on the current.

Time histories of the horizontal displacement of the rotor centre for three values of the delayed yielding time constant (1 ms, 5 ms, 9 ms) are drawn in Fig. 5-7. It is evident that application of the current increases the damping effect which leads to reducing amplitude of the vibrations. The results also show that the oscillations amplitude after switching the current on strongly depends on the delayed yielding phenomenon. The rising time constant considerably decreases the damping effect which leads to lower suppresion of the rotor oscillations.

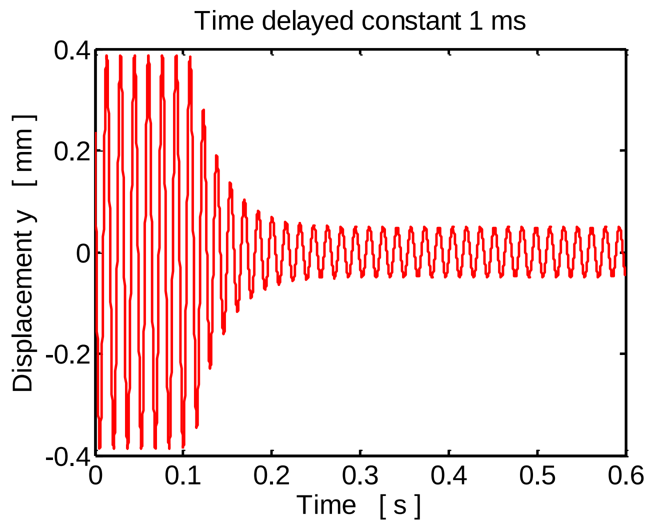

Figure 5: The horizontal displacement of the rotor centre of gravity - 1 ms delayed yielding time constant.

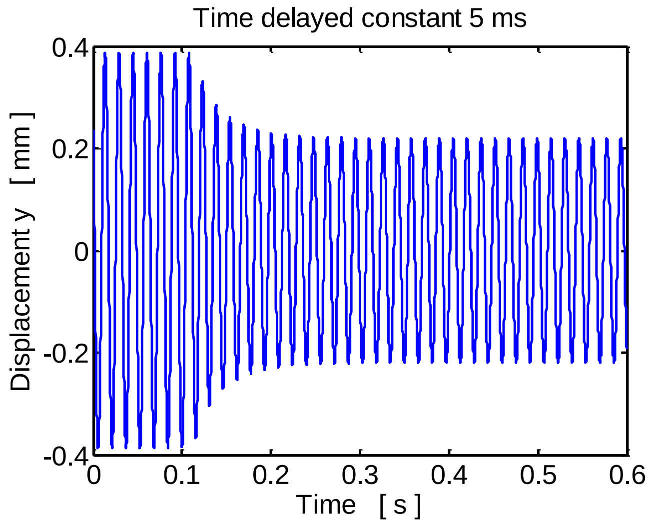

Figure 6: The horizontal displacement of the rotor centre of gravity - 5 ms delayed yielding time constant.

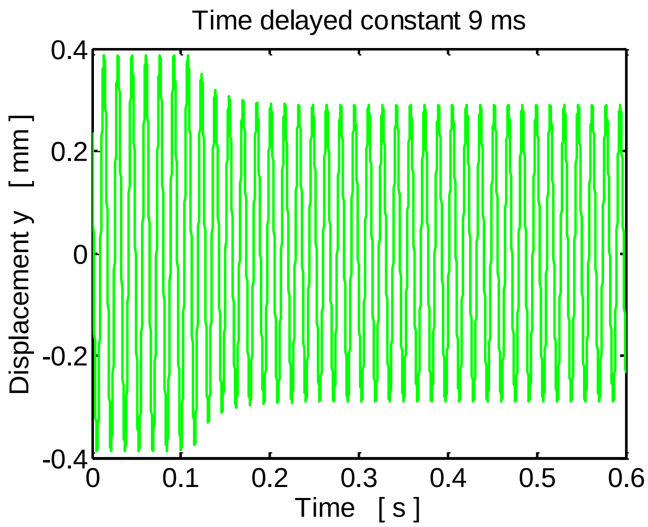

Figure 7: The horizontal displacement of the rotor centre of gravity - 9 ms delayed yielding time constant.

6 CONCLUSIONS

The principal aim of the carried out study was to investigate influence of the delayed yielding process occuring in thin lubricating films of magnetorheological squeeze film dampers on the vibration attenuation of asymmetric rigid rotors performing a complicated spatial motion. The magnetorheological oil was modelled by Bingham material and influence of the time history of magnetic induction on the yielding shear stress was described by a convolution integral. The carried out computational simulations proved that the rising applied current reduces amplitude of the rotor vibrations. On the contrary, the delayed yielding decreases the damping force and the decrease rises with increasing value of the delayed yielding time constant. Such behaviour can reduce efficiency of magnetorheological squeeze film damping devices during the working regimes when fast responses on varying operation conditions or control manipulations are needed.

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