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REDUCTION OF THE ENERGY LOSSES BY APPLICATION OF CONTROLLABLE SQUEEZE FILM DAMPERS

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Abstract: *The unbalance induces lateral oscillations of rotors and forces transmitted to the rotor casing. The squeeze film dampers with integrated rolling element bearings represent a technological solution, which enables to reduce their magnitude, and consequently the resistance against the rotor rotation. To achieve optimum performance of the damping devices, their damping effect must be adapable to the current rotor speed. This paper reports a proposal of a controllable squeeze film damper, the damping effect of which is controlled mechanically by shifting its outer ring in the axial direction. The developed mathematical model of the damper is based on assumptions of the classical theory of lubrication and is completed with implementation of a gas cavitation. The results of the computational simulations show that an appropriate control of the damping force enables to reduce the energy losses in a wide range of operating speeds.*

Keywords: Squeeze film damper, Controllable damping, Friction in bearings, Energy losses reduction.

1. Introduction

The imbalance induces lateral vibration of rotors and increases the forces transmitted to the rotor stationary part. Their magnitude can be reduced by application of squeeze film dampers with integrated rolling element bearings. The resistance against the rotor rotation due to loading the bearings arrives in the energy losses. A simple dynamical analysis shows that adapting the damping effect to the current operating speed (Zapoměl et al., 2013) enables to reduce their magnitude.

The mathematical models of squeeze film dampers based on application of the classical theory of lubrication and the Reynolds equation can be found in a number of publications. The implementation of a cavitation caused by sucking the air from the ambient space is presented in Olsson (2004). Some approaches to control the damping force can be found e.g. in (Mu at al., 1991) and (El-Shafei et al., 2000). In this paper, a concept of a squeeze film damper with a cylindrical gap and the axially movable outer ring is presented. Changing the axial position of the ring changes the damper land length and thus the damping force. A gas cavitation is implemented in the damper mathematical model. The effeciency of the damping element was proved by the computational simulations.

2. The design configuration of the proposed controllable squeeze film damper

The main parts of the proposed controllable squeeze film damper operating with conventional oil (Fig. 1) are two concentric rings, between which there is a thin layer of a lubricating film. The inner ring is coupled with the damper housing by a squirrel spring and with the rotor journal by a rolling element bearing. The oil is supplied to the damper gap by a central groove. The outer ring is mounted with the damper body movably (Fig. 1). Its shifting in the axial direction changes the damper land length, which results in changing the pressure distribution in the damper gap and thus in controlling the damping force.

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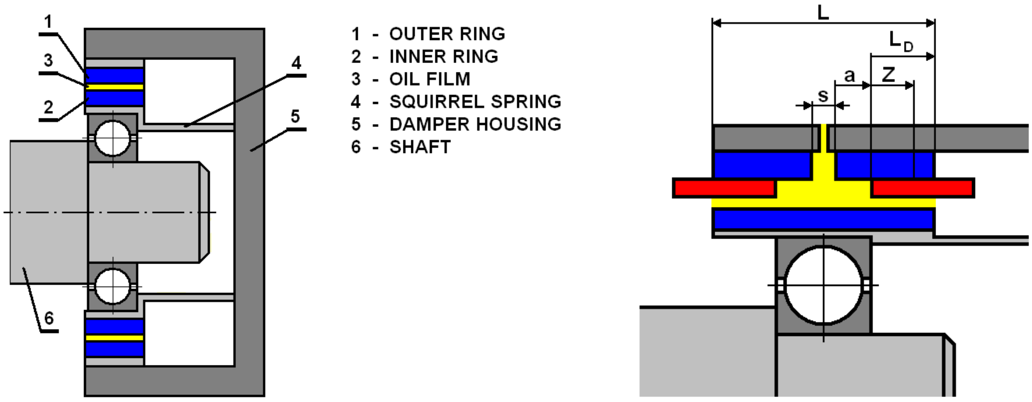

Fig. 1: The proposed controllable squeeze film damper.

3. Components of the controllable damping forces in cavitated squeeze film damper

The determination of the pressure distribution in the damper gap is based on application of the classical theory of lubrication, e.g. in Zapoměl (2007). In addition, it is assumed that both the geometric and design parametres enable to consider the damper as short, see e.g. in Zapoměl (2007).

The Reynolds equation governing the pressure profile reads

$$\frac{\partial^2 p}{\partial Z^2} = \frac{12\eta}{h^3}\frac{\partial h}{\partial t}. \tag{1}$$

p is the pressure, Z is the axial coordinate, η is the oil dynamical viscosity, h is the oil film thickness, and t is the time.

Double integration of (1) arrives in the relation for the pressure distribution

$$p = \frac{6\eta}{h^3}\frac{\partial h}{\partial t} Z^2 + C_1 Z + C_2. \tag{2}$$

C_1 and C_2 are the integration constants.

In the first step, the integration constants are determined for the boundary conditions

$$p = p_S \ \text{ for } \ Z = 0 \tag{3}$$

$$p = p_A \ \text{ for } \ Z = L_D \tag{4}$$

p_s is the inlet oil pressure in the central groove, p_A is the atmospheric pressure and L_D is the land length of one half of the damper (Fig. 1).

If the pressure gradient at the end of the damper ($Z = L_D$) is positive, a gas cavitation occurs. In cavitated area the Reynolds equation does not hold and it is assumed that the pressure remains constant there and equal to the pressure in the ambient space. In this case, the integration constants C_1 and C_2 are determined by application of the following conditions

$$p = p_S \ \text{ for } \ Z = 0 \tag{5}$$

$$p = p_A \ \text{ for } \ Z = Z_C \tag{6}$$

$$\left[\frac{\partial p}{\partial Z}\right]_{Z=Z_C} = 0 \ \text{ for } \ Z = Z_C \tag{7}$$

Z_C is the axial coordinate of the border between the non-cavitated and cavitated regions. Because the pressure of the medium in cavitated area is constant, the pressure gradient is zero. As a result, no flow occurs between the non-cavitated and cavitated regions. This is expressed by the boundary condition (7).

The hydraulic force components are obtained by integration of the pressure distribution around the circumference and along the length of the damper. The land length of the half of the damper L_D reads

$$L_D = \frac{L}{2} - \frac{s}{2} - a\,. \tag{8}$$

L is the total length of the damper, s is the width of the central feeding groove and a is the axial shift of the outer ring. It is evident that the change of a changes L_D and thus it controls the damping force.

4. Energy losses in the rolling element bearings

Several approaches have been developed to determine the friction moment in rolling element bearings. If the loading is not extereme, the friction moment can be expressed

$$M_{RB} = F_{BE}\, f_B \frac{d_H}{2} \tag{9}$$

where F_{BE} is the magnitude of the equivalent force transmitted through the bearing, f_B is the friction coefficient and d_H is the diametre of the bearing hole (diametre of the shaft journal). f_B depends on the type of the bearing and its value ranges approximately from 0.0010 to 0.0045.

The equivalent force for radial bearings is calculated as

$$F_{BE} = X\, F_R + Y\, F_A\,. \tag{10}$$

F_R, F_A are the radial and axial component of the force transmitted through the bearing and X, Y are the radial and axial loading coefficients, the value of which depends on the bearing type (Bolek et al., 1989).

The radial bearing force F_R consists of three components, of the hydraulic damping force that is transmitted through the oil film, the elastic force produced by bending of the squirrel spring and of a prestress force that acts on the squirrel spring to centralize the equilibrium position of the rotor in the damper. The loss power P_{RB} in the rolling element bearing is given as the product of the friction moment M_{RB} and angular speed of the rotor rotation ω

$$P_{RB} = M_{RB}\omega\,. \tag{11}$$

5. The investigated rotor system

The investigated rotor is rigid. It consists of a shaft and of one disc. With the stationary part it is coupled by squeeze film dampers with integrated rolling element bearings. The rotor turns at constant angular speed, is loaded by its weight and excited by the rotor unbalance. The whole system can be considered as symmetric relative to the disc middle plane. The squirrel springs are prestressed to eliminate their deflection caused by the rotor weight.

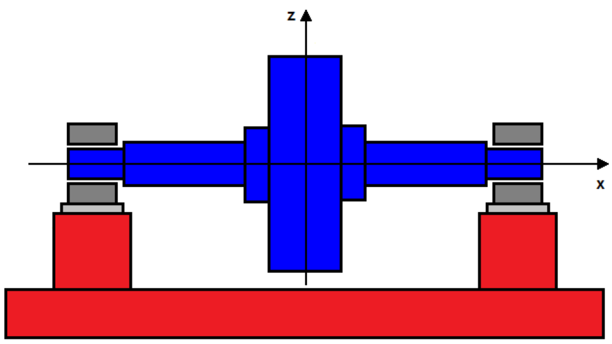

Fig. 2: The investigated rotor system.

The technological and operating parametres of the investigated rotor system are: the rotor mass 430 kg, the stiffness of one squirrel spring 2 MN/m, the rotor imbalance 21.3 kg.mm, the squeeze film damper length/diameter 50/150 mm, the width of the central feeding groove 5 mm, the width of the damper gap 0.2 mm, the oil dynamic viscosity 0.02 Pas, the inlet pressure 200 kPa, the diameter of the bearing hole 110 mm, the radial and axial loading coefficients X and Y are 1.0 and 0.0, respectively, and the bearing friction coefficient 0.0015 (one row ball bearing).

A simple dynamical analysis shows that the critical speed of the rotor rotation is approximately 96 rad/s.

Fig. 3 provides the time histories of the loss power related to one rolling element bearing for two angular speeds of the rotor rotation of 100 and 500 rad/s and two shifts of the outer damper ring of 0 and 10 mm. Larger shift means lower damping effect. It is evident that higher damping arrives at reduction of the loss power for the velocity close to the critical speed. When the rotor turns at a speed much higher than the critical one, lower damping reduces the loss power.

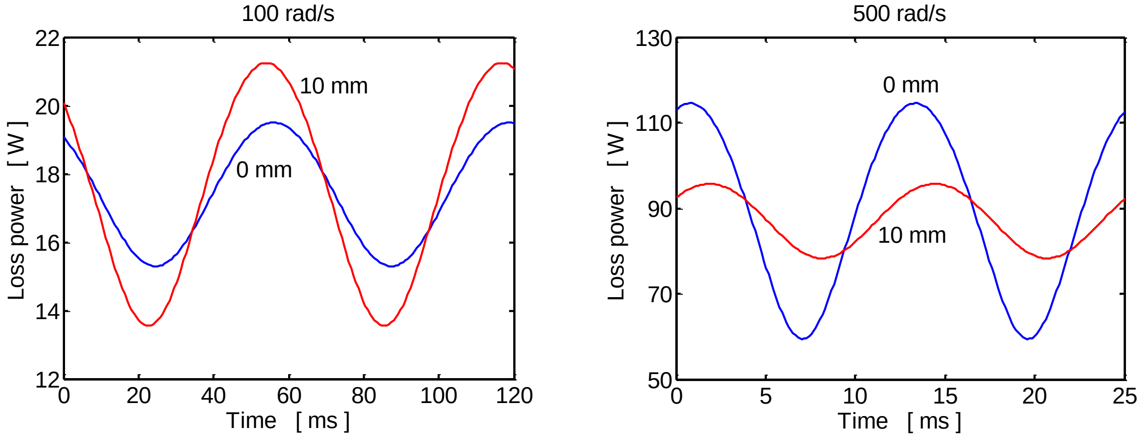

Fig. 3: The power loss in one bearing for speeds of 100 rad/s and 500 rad/s.

6. Conclusions

The carried out research work presents a study on the effect of controllable squeeze film dampers on reducing the energy losses in the rotor supports, which offers a new field of their application. The damping effect is controlled by shifting the outer damper ring in the axial direction, which reduces the damper land length. The developed mathematical model of the squeeze film damper is based on the classical theory of lubrication, the cavitation caused by sucking the air from the ambient space to the damper gap is implemented. The extension of the computational procedures, the evidence that a proper control of magnitudes of the damping forces makes it possible to minimize the energy losses in the bearing elements, and learning more on the influence of the controllable squeeze film dampers on the oscillatory motion of rigid rotors are the principle contributions of the work presented in this paper.

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