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Increase of Energy Efficiency by Means of Application of Active Bearings

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Abstract**—**In the present paper, increasing of energy efficiency of a thrust hybrid bearing with a central feeding chamber is considered. The mathematical model was developed to determine the pressure distribution and the reaction forces, based on the Reynolds equation and static characteristics' equations. The boundary problem of pressure distribution calculation was solved using the method of finite differences. For various types of lubricants, geometry and operational characteristics, axial gaps can be determined, where the minimal friction coefficient is provided. The next part of the study considers the application of servovalves in order to maintain the desired position of the rotor. The report features the calculation results and the analysis of the influence of the operational and geometric parameters on the energy efficiency of mechatronic fluid-film bearings.

Keywords**—**Active bearings, energy efficiency, mathematical model, mechatronics, thrust multipad bearing.

I. Introduction

TRADITIONAL fluid-film bearings are developed to work in a narrow range of operational parameters. However, to use this type of bearings under complex conditions, e.g. frequent launches and stops, changing load, etc., it is necessary to introduce new elements to the rotor system. Caused by the development both in software and in hardware, the application of active fluid-film bearings presently is an object of research in the directions of optimization and design on such bearings, study on new control algorithms and new control devices.

To estimate the feasibility of using the active bearings in the rotor machines, one has to consider the developed bearings and the results of their application. It is also necessary to remember that the use of hydro- or aerostatic bearings is not always dictated by the necessity of minimizing the rotor vibrations. The character of friction in such bearings allows to obtain a long life-time expectancy, as the mechanical contact between surfaces is eliminated. However, the density of the lubricant often sets severe limitations on the area of applications of such bearings, due to the possibility of unacceptable amplitudes of vibrations. Nevertheless, to a certain extent active fluid-film bearings appear to be more effective than traditional ones, and with an adequate control algorithm, they can significantly decrease the unwanted vibrations.

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In 1989, Ulbrich and Althaus [1] carried out the experimental study on the active bearing with so-called tilting pads, which allowed to control the wedge effect, which significantly increased the load capacity of the bearing. The research was continued by Fürst [2]. The idea went basic for the articles [3] and [4], and the use of actuators for such bearings was considered in [5] and [6]. The separation of the actuators to control each element independently was theoretically studied in [6], where it is shown that with such approach the damping and the stiffness of the bearing could be adjusted, which was experimentally studied in [7].

The idea of radial lubricant supply was introduced in 1994 at IUTAM conference and was published in [8]. It showed the method of pressure distribution calculation for hydrostatic and hydrodynamic bearings, which was a beginning of an active lubrication research. The combination of hydrostatics and hydrodynamics was considered in [9], [10], where the pressure is controlled with the use of servovalves. In the majority of cases, the behavior of the rotor is significantly increased when the active bearings are used. In [10] the thrust bearings are studied, where the lubricant is supplied in the axial direction which matches the direction of the load. The control is implemented with servovalves and the modeling was based on Reynolds equation and the flow balance.

The undeniable fact is that the development in the field of mechatronic bearings is caused partly by the development of the control systems, namely the DAQ devices, and partly by the development of the control algorithms. The development of software makes it possible to develop complex algorithms, which causes the more complex hardware development. Artificial intelligence, fuzzy logic methods and neural networks, shape recognition, etc. – all these can be used to design the mechatronic bearings for the high-speed rotors. Application of such technologies has already begun in the field of magnetic bearings as shown in [11].

With the developments in the field of manufacturing of such bearings and in the areas, where such bearings are used, come greater requirements. As well as parts of other machines and machines themselves, fluid-film bearings are always restricted in terms of size and weight; moreover, the developers need to account one of the most crucial characteristics of a machine, namely, its energy efficiency.

II. Conceptual Model

Application of active fluid-film bearings provides not only accurate diagnostics using modern hardware tools, but also generally extends the functionality of the machine part. Fig. 1 illustrates the conceptual model of a test rig designed to study the possibility of active control applied to hybrid thrust fluid-film bearings, i.e. combining the hydrodynamic 'lifting effect' due to the inclined surfaces of the pads and high velocity, and the hydrostatic effect due to constant supply of a lubricant under pressure.

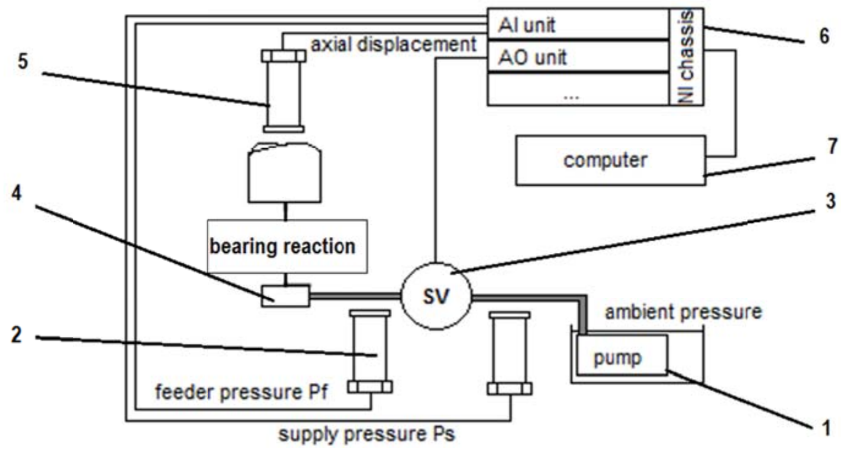

Fig. 1 Conceptual model of a hybrid thrust fluid-film bearing

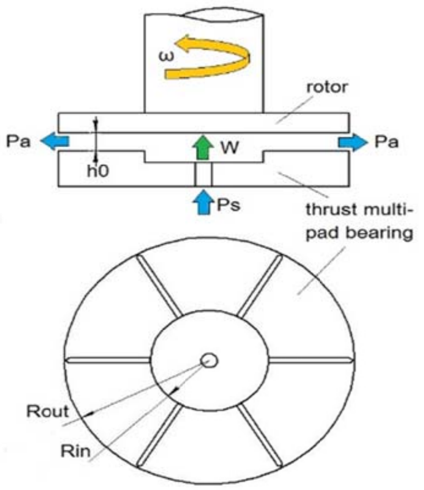

Fig. 2 Conceptual model of a hybrid thrust fluid-film bearing

The operational principle of an active hybrid multi-pad thrust bearing is as follows: The pump 1, located in the tank with the fluid (water, oil, etc.), supplies the media into the housing of the bearing 4. The pressure data is acquired with the sensor 2. The data on the present position of the rotor is acquired with the displacement sensor 5. To process the sensor data and to generate the control influence, the Data Acquisition (DAQ) module 6 is used, which is connected to the PC. Software PID-controller generates the signal, according to which the pressure is changed by means of a servovalve 3.

III. Mathematical Model of a Hybrid Thrust Multi-Pad Bearing

In order to evaluate the effect of controlled lubrication, a computational model of a thrust bearing, shown in Fig. 2, was developed, which allows calculation of static characteristics during a steady state regime and estimation of power losses due to friction with various types of lubricants.

Due to the fact that a lubricant is externally supplied under pressure *Ps* higher than the atmospheric pressure *Pa*, and that the rotor is rotating at some speed ω, two effects combined create some load capacity *W*, and the surfaces of a bearing and a rotor become separated by an axial gap *h0*. While the hydrodynamic effect depends significantly not only on geometry of a pad and the properties of a lubricant, but also on the velocity of a rotor, the hydrostatic effect depends mostly on the supplied pressure, thus is a lot easier to control. The size of a pad disk (*Rin* and *Rout* being inner and outer radiuses accordingly) is however chosen considering the overall design of a particular machine.

The mathematical model of a hybrid thrust multi-pad bearing is based on the Reynold's equation, which allows numerical calculation of the pressure distribution across the surface of the disk. For a thrust bearing shown in Fig. 2, the following equation for a steady regime was used:

$$\frac{\partial}{\partial\varphi}\left(\frac{h^3}{\mu}\frac{\partial p}{r\partial\varphi}\right)+\frac{\partial}{\partial r}\left(r\frac{h^3}{\mu}\frac{\partial p}{\partial r}\right)=6\omega r\frac{\partial h}{\partial\varphi}, \quad (1)$$

where r and φ are polar coordinates, h – axial gap function, μ - lubricant's viscosity, ω – angular speed.

To simulate various shapes of the axial gap between the rotor and the bearing, Bezier curves were used to approximate the surface in order to avoid calculation errors and inaccuracies. Fig. 3 shows various shapes, which could be modelled using three key points. While the shapes (a) and (b) can be used in traditional hydrodynamic fluid-film bearings, the shape (c) is most commonly referred to as a hydrostatic thrust bearing, so in this paper this type will be merely used in order to compare the influence of geometry on bearing's operation.

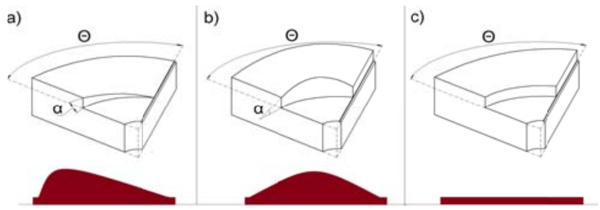

Fig. 3 Pad shapes: (a) plane slider pad, (b) curved slider allowing the operation in both directions, (c) flat pad; (Θ – pad angle, α – inclination angle of the surface)

In Fig. 4, the approximation curves are shown with three key points, which allow approximation of the surface using four 3rd order Bezier curves. Fig. 5 depicts the pressure field of a thrust hybrid bearing with six fixed pads and the following geometric and operational parameters: *Rin* = 20 mm, *Rout* = 40 mm, ω = 1000 rad/s, α = 0.5° (see Fig. 3 (a) was modelled). Guaranteed axial gap *h0* = 50 μ, supply pressure *Ps* = $1.5 \cdot 10^{-5}$ Pa. Lubricant – water with corresponding viscosity.

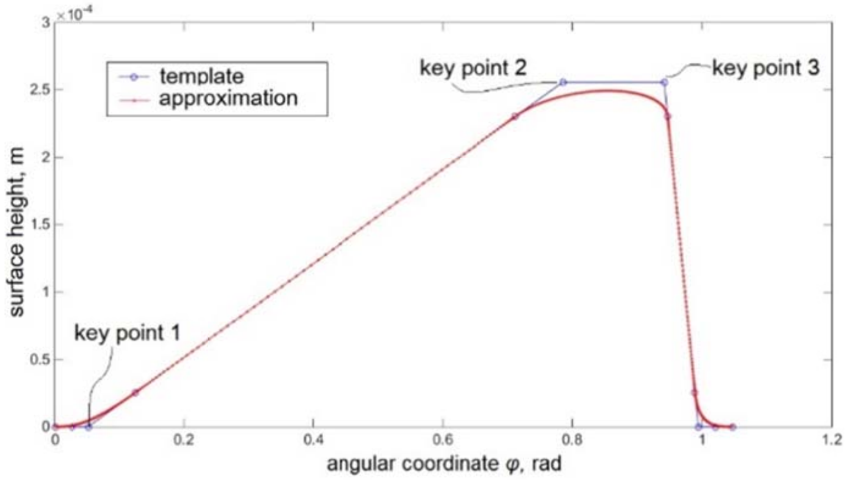

Fig. 4 Approximation of the pad's surface

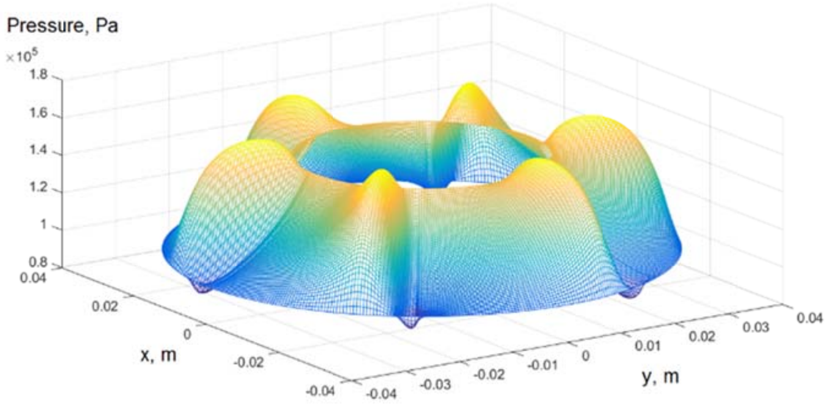

Fig. 5 Pressure distribution on the surface of a bearing

IV. Power Loss and Friction

The obtained pressure distribution over the surface of a pad disk allows estimation of a number of integral characteristics, including friction torque of the lubricant film as of particular interest in this case. The power loss due to friction can then be estimated as:

$$N_{fr} = z\omega \int_{Rin}^{Rout} \int_0^{\varphi} r \left[\frac{h}{2}\frac{\partial p}{r\partial \varphi} + \frac{\mu\omega r}{h}\right] r d\varphi dr, \quad (2)$$

where z is a number of pads, and p – pressure distribution, obtained from numerically solving (1).

The friction coefficient can also be obtained as follows:

$$k_{fr} = \frac{F_{fr}}{W}, \quad (3)$$

where F_{fr} and W are friction force and total load on a bearing respectively.

For the numerical results the case of a thrust bearing with the same operational and geometric parameters is considered, but the tests will focus on the effect of geometry of the pads on power loss due to friction.

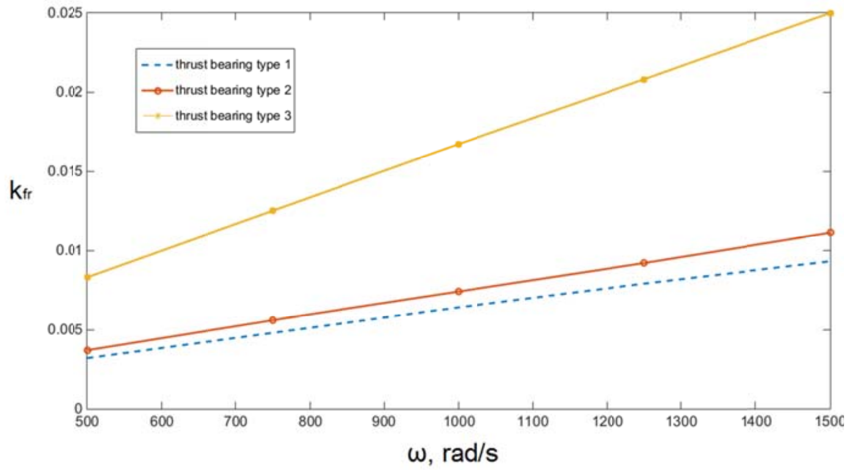

Fig. 6 Friction coefficient with respect to ω

As seen from Fig. 6, friction coefficient increases with the increase of angular speed for all three types of pad geometry. However, there is an obvious decrease of friction when particular geometry is introduced. The inclination of the pads creates additional lifting force and thus decreases the friction coefficient.

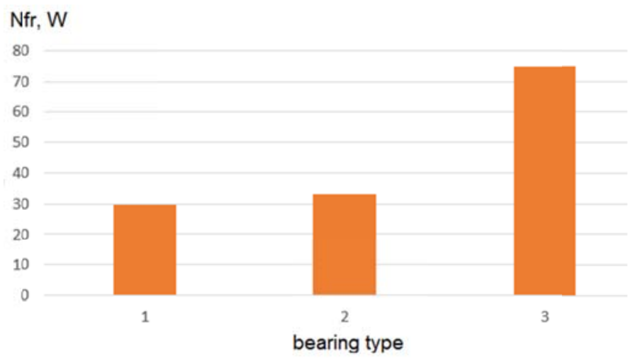

Fig. 7 Pressure distribution on the surface of a bearing

As well as the friction coefficient, the same effect can be seen comparing power loss due to friction for the given bearing types. This shows the importance of wise choice of thrust pad geometry, for decreasing friction between machine parts is not only a matter of providing sufficient lubrication. Instead, the combination of hydrodynamic or EHD (elastohydrodynamic) models with a principle of rotor displacement control shown above allows determining the optimal control parameters based on the analysis of character Stribeck curves for every particular case.

V. Active Control Principle

Rotor machines, and especially bearings, both radial and thrust, are exposed to high friction losses during the start-up and shut-down and experience friction losses during operation due to the viscosity of a lubricant. The regimes, in which the rotor-bearing system functions in terms of lubrication is shown in Fig. 8 and is analyzed in detail in [12]. The blue dotted line represents the Stribeck curve for the model, described in the present study. However, in reality the curve takes the shape of the black solid line in Fig. 8, during the start-up and shut-down, when the Hersey's number ($\mu U/W$) is low the asperities of the surface touch each other, influencing the friction coefficient as well as the geometry of the pad due to elastic deformation. So, by means of introducing elastic contact between the surfaces, other regimes can be accounted for. As one can see from the Stribeck curve, there is a range of Hersey's numbers, where the friction coefficient is minimal. This happens during the transition period from mixed lubrication to full hydrodynamic. Given that, the idea of control is in generating such control signal, which would act on the rotor so, that the rotor's displacement due to this action results in rotor taking a more efficient trajectory in terms of friction. The control system would evaluate the real position of the rotor given that the desired position can be determined numerically. The ADC/DAC would then generate the signal, i.e. electric signal to a servovalve, closing or opening it to change the pressure in the feeder and thus changing the position of a rotor along the vertical axis.

The system was checked during the work on [13] and showed good compliance to the numerical results. Fig. 9 shows the real and modeled rotor position in a journal bearing with one active feeding chamber, the problem was modeled under stationary conditions, i.e. $\omega = 0$.

In Fig. 9, blue line represents simulation results, red line shows experimental data. The load applied to the rotor was approximately 15 N, the control system was first off, and then turned on. One can notice a significant decrease of the amplitude of rotor's displacement once the control system was turned on. However, this approach has its drawbacks, since ordinary servovalves appear to be hardly applicable for vibration minimization, as their frequency response is only linear up to a few Hz. Rotor displacement caused by static load on the rotor can be, however, reduced and if rotor is operating under stable conditions, its trajectory can be adjusted to match the energy efficient trajectories.

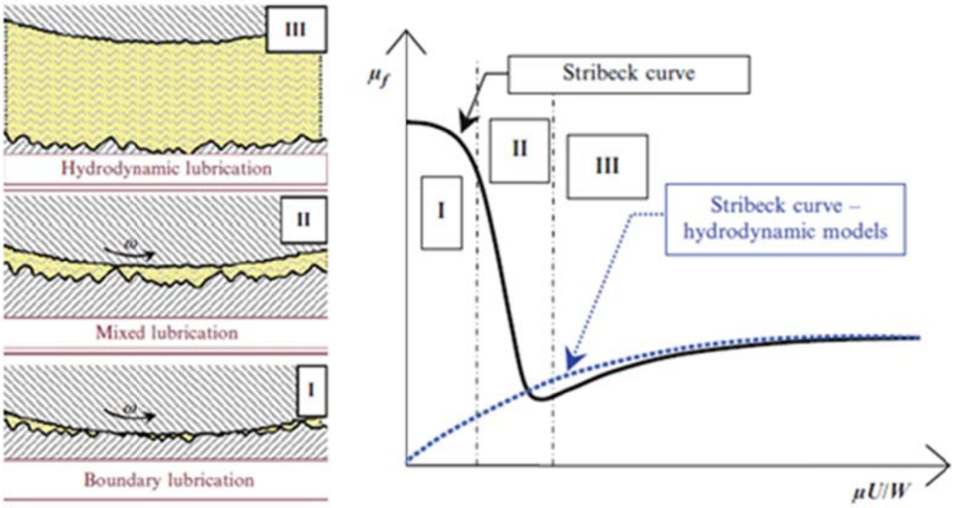

Fig. 8 Stribeck curve and the lubrication regimes

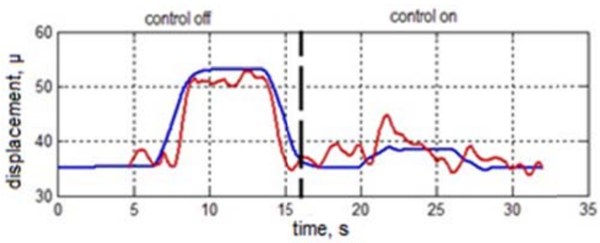

Fig. 9 Effect of active control on rotor's displacement

VI. Conclusion

The paper covered a mathematical model of a thrust hybrid fixed-pad bearing, developed to study the possibility of application of controlled lubrication to increase energy efficiency of a rotor machine by adjusting the rotor's position. The latter can be done using electromechanical servovalves, which increase or decrease the supply pressure in the bearing's feeding channel, thus changing the value of the reaction force of a thrust pad and displacing one surface relatively to another. Such displacement can be based on a Stribeck curve for a particular thrust bearing, from which a range of gap heights can be determined, where the friction coefficient has minimal value.

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