Control of the Reaction Forces of the Lubricant Film in the Journal Hybrid Bearing

Leonid Savin, Denis Shutin, Roman Polyakov, Alexander Babin

Abstract— For the rotor systems which operate under complex conditions, it is quite topical to use the active bearings which allow to quickly adjust the displacement of the rotor. The most important problem here is control the reaction forces of the bearing, which determine to position of the rotor. In the active journal hybrid bearings this is achieved by means of controlling the lubricant supply pressure. On the basis of the mathematical model of the bearing the research was carried out on the relationship between the reaction forces and the pressure in the feeding chambers. The results of the study show the possibility to control the value and the direction of the reaction forces in the active journal hybrid bearing, and the linearity of the relationship between the reaction forces and the pressure distribution of the lubricant in the feeding chambers.

Keywords—Active bearing, active lubrication, hybrid bearing, pressure distribution.

I. Introduction

ONE of the problems of the traditional fluid-film bearings is the fact, that they are developed for a certain narrow range of operational parameters, and within this range they operate very well. But if the rotor system works under some complex conditions, including the undetermined, of a various load and frequent starts and stops, traditional bearings can not always cope with the negative factors which occur and shorten the life time of the bearing. These factors include the unwanted rotor displacement in the fluid-film bearing which occur during the vibration processes of a various nature, under heavy external loads and during the rotation in the area of high eccentricity in the start and stop phases in case of a hydrodynamic bearing. One of the possible solutions of the problem of compensating these negative factors is application of the bearings with an active control of the rotor displacement function.

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The most well known example of an active bearing are active magnetic bearings. The load capacity there is created with the electromagnetic field, and the problem of suspension of the rotor itself is a problem of controlling its displacement. Apart from the possibility to control the position of the rotor, they allow almost an unlimited rotational speed. However, extremely high complexity of the structure, the control algorithms and, respectively, the high cost and the difficulties with maintaining are the reasons why they are not widely used. So, the development of the active fluid-film bearings with the active control functions appears to be a more reasonable solution, since they have a more simple structure, close to the traditional bearings, and can be developed based on the less complicated control systems and algorithms, as the most important function of the load capacity provision is implemented by means of the properties of the lubricant film, and it requires no additional control. Rotor displacement control in the active fluid-film bearings is provided by means of varying the functioning parameters of the bearing.

II. The Methods of Rotor Displacement Control

The problem of the rotor displacement control is narrowed to the problem of controlling the value and the direction of the reaction of the bearing. The static rotor position condition is the zero value of the resultant force F_{rf} which is a sum of all the n forces acting on the rotor:

$$F_{rf} = \sum_{i=0}^{n} F_i = 0. \tag{1}$$

If initially the rotor was in a state of equilibrium, then under the changed reaction force it moves in a respective direction until the condition (1) is met again. So, the controllable value of the reaction force will be a control action for rotor positioning.

In the fluid-film bearings the reaction force control is implemented by means of changing the pressure distribution. The reaction of the fluid film is defined by the pressure distribution and can be determined by its integration over the bearing’s surface:

$$R_X = \int_0^L \int_0^{\pi D} p(x,z) \sin \alpha dxdz;$$
$$R_Y = \int_0^L \int_0^{\pi D} p(x,z) \cos \alpha dxdz, \quad (2)$$

where R_X, R_Y – reaction forces along the *X* and *Y* axes;
p(x,z) – lubricant pressure distribution function;
L – length of the bearing;
D – diameter of the bearing;
α – current angle of the surface of the bearing sweep in the process of integration.

There are different ways of reconfiguring the pressure distribution conditioned by the specific design of the fluid-film bearing. E.g., in the tilting-pad bearings, described in [1], [2] and the foil gas-dynamic bearings the pressure distribution can be reconfigured by means of changing the shape of the radial gap, which is achieved by moving the pads or the foils with the actuators. In the bearings where magneto-rheological fluids are used as a lubricant or an additional element, e.g. [3], the reconfiguration of the pressure distribution can be achieved with the change in the viscosity features of such fluid under the effect of the magnetic field. Along with this, in the design of rotating machinery it is reasonable to use the hybrid bearings, where the load capacity is formed by both hydrostatic and hydrodynamic effects. Unlike hydrodynamic bearings, the presence of the hydrostatic part in the hybrid bearing allows it to operate effectively during the start and stop phases as it eliminates the mechanical contact of the rotor and the surface of the bearing, which is always a reason of an early break down. The hydrostatic part is provided by the supply of the lubricant to the friction area under pressure through the feeding chambers. A general scheme of an active hybrid bearing can be found in the Fig. 1.

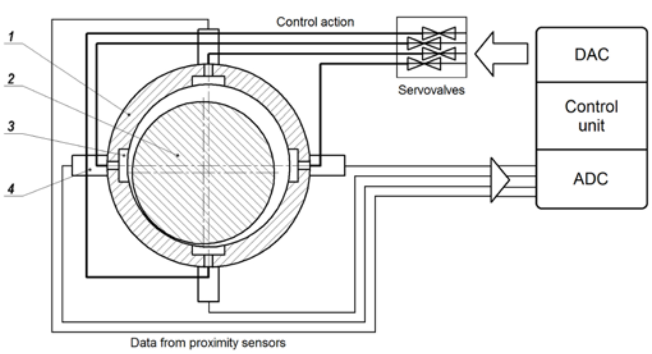


Fig. 1 Scheme of an active hybrid bearing

An active hybrid bearing operates as follows. The data on the present rotor 2 position in the bearing housing 1 are acquired by the sensors 4 and, by means of the ADC, are transferred to the control unit. The control unit acting as the regulator and the basis of the measurement system, produces the control action in accordance to the inbuilt control law. The control action is then converted from digital to analog form with the DAC and transferred to the controllable electro-hydraulic devices (servo valves) which perform the direct change to the pressure of the lubricant in the feeding chambers. The change in the pressure in the feeding chambers of the hybrid bearing results in a change in the configuration of the pressure distribution and, as a result, in a change of the reaction force which is a essential step of the rotor motion control.

To design an effective control system it is necessary to know the dependencies which connect the supply pressure of the lubricant and the reaction force of the bearing. The direct change of the reaction force of the hybrid bearing during the experiment is so far impossible because the force in question is an integral sum of the distributed forces which are applied on the rotor by every infinitesimal element of the lubricant film. So we carried out the research with the use of the mathematical model of the lubricant film. The aim of the study was to obtain the information on how the change in the supply pressure in one or in a number of chambers influences the value and the direction of the reaction force of the bearing.

III. Mathematical Model of a Hybrid Bearing

When developing a mathematical model of a hybrid bearing, the traditional for this class of bearings approach was used, which is the joint solution of the generalized Reynolds equation with the known function that describes the shape of the radial gap. The same technique is used, for instance, in [4].

We performed modeling with the following assumptions:

1) general assumptions of the hydrodynamic lubrication theory, which are the ideal wettability of the friction surfaces and a strong adhesion of the lubricant to the surfaces; laminar lubricant flow; all the friction processes in the system are of liquid friction type [5];

2) we considered the steady operational regime with a constant rotational speed, the peripheral speed *U=const*;

3) due to the steady operational regime we considered the isothermal task and was and the viscosity μ of the lubricant was considered to be constant;

4) due to the used incompressible model of the lubricant (water, oil) and considering the isothermal formulation the lubricant density $\rho = const$.

With these assumptions the generalized Reynolds equation takes the following form:

$$\frac{\partial}{\partial x}\left[h^3 \cdot \frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial z}\left[h^3 \cdot \frac{\partial p}{\partial z}\right] = 12\mu V + 6\mu U \cdot \frac{\partial h}{\partial x}. \quad (3)$$

The geometrical parameters of the modeled system are shown in the Fig.2. XOY is the coordinate system with the origin on the bearing's longitudinal axis. X_1OY_1 is the coordinate system with the origin at the center on the rotor's axis. h_0 is the initial radial gap when the rotor's and the bearing's longitudinal axes coincides. Since we studied the relation between the lubricant pressure parameters and the bearing reaction forces, we considered only the aligned position of the rotor in the bearing when the rotor's axis is always parallel to the bearing's longitudinal axis. So the function determining the value of the radial gap over the surface of the fluid-friction bearing depends only on α coordinate and is as follows:

$$h(\alpha) = h_0 - X \cdot \sin\alpha - Y \cdot \cos\alpha, \qquad (4)$$

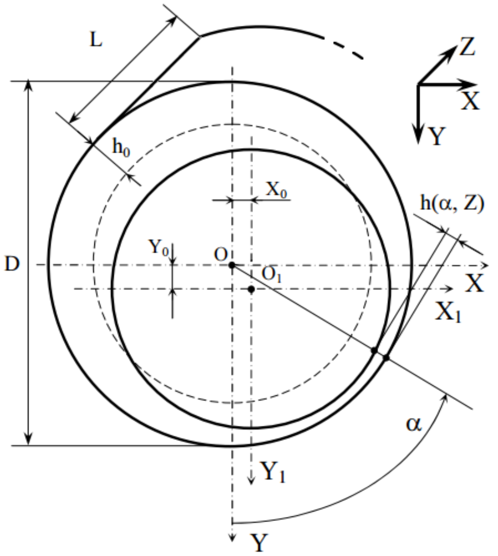


Fig. 2 The geometrical parameters of the modeled bearing

Since it is extremely difficult to find an analytical solution of the equation (3), it is a common practice to solve such problems using various numerical methods. We used the finite differences method to solve the equations (3) and (4) using the Sommerfeld hypothesis about the lubricant film continuity and the known pressure at the bearing edges as the boundary conditions. Besides that, since we calculated the pressure distribution for a hybrid bearing, the flow balance equation should be considered as well in order to calculate the pressure in the feeding chambers reduced on account of the throttling effect [6]:

$$Q_H = Q_x + Q_z + Q_y, \qquad (5)$$

where Q_H – general mass flow of the lubricant through the throttles;

Q_X, Q_Y, Q_Z, – mass flow of the lubricant through the throttles in separate directions.

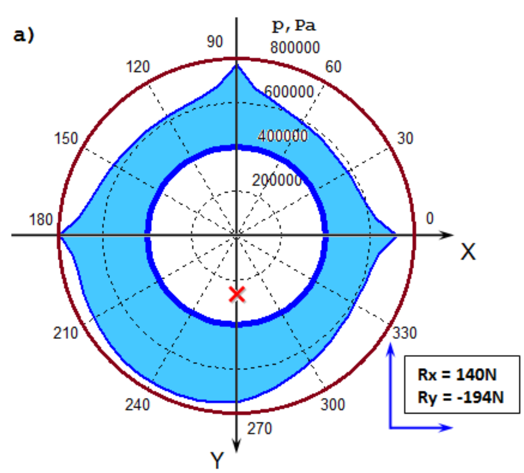
Solving the equation (5) for each of the feeding chamber we can vary the inlet lubricant pressure before its passing through the throttle as it occurs in an active hybrid bearing. Thus we get the border conditions for a numerical determination of the pressure distribution *p(x,z)* over the surface of the bearing.

Finally, knowing the geometrical parameters of the modeled bearing and the pressure distribution *p(x,z)* we can calculate the lubricant film reaction forces solving the equations (2) using any numerical integration method, e.g. in the present model we used the Simpson method.

IV. Relationship Between the Supply Pressure and the Reaction of the Bearing.

During the modeling of the lubricant film of the hybrid bearing we changed the supply pressure of the lubricant in one or several feeding chambers in order to estimate the change of the reaction forces parameters. The modeled rotor-bearing system has the following parameters: type of the bearing – hybrid with 4 point feeding chambers; length of the bearing – 63 mm, diameter – 40 mm, radial gap – 100 µm; length of the lubricant supply channel (throttle) – 8 mm, diameter – 1 mm; lubricant – water; rotational speed of the rotor – 10000 rev/min. A quasistatic state was modeled, where the rotor is located in the bearing with a relative eccentricity 0.35, in the coordinate system with its center connected with the center of the cross-sectional area of the bearing, the coordinates of the tip of the rotor [0, $3.5 \cdot 10^{-5}$], momentary velocities V_x=0 m/s, V_y=0.1 m/s. The feeding chambers are named X^+, X^-, Y^+ и Y^- in accordance to their location in relation to the axis of the XOY coordinate system. The set parameters of the rotor-bearing condition complies to the possible context of the rotor position control problem, when it moves towards the chamber Y^+ under the influence of some force factors. The goal was to provide an additional counterforce in the opposite direction (towards Y^-). This effect can be achieved by means of increasing the pressure in the chamber Y^+ and/or decreasing the pressure in the chamber Y^-.

In the Fig. 3 the diagrams of the pressure distribution in the lubricant film are shown in the cross-sectional area along the centers of the feeding chambers. In the bearing in the Fig. 3a the supply pressure is 0.4 MPa. This regime is equal to the operation of a traditional hybrid bearing with a common feeding collector for all the feeding chambers. A 50% increase in the supply pressure in the chamber Y^+ (Fig. 3b) resulted in the 46% increase of the resulting force R_Y that counteracted the displacement of the rotor, whereas the change of the R_X is less than 1%. The same value increase of the pressure in the Y^+ chamber together with the 50% decrease of the pressure in the Y^- (Fig. 3c) led to a further increase of the R_Y by 79% in comparison to the initial value, whereas the increase of the reaction R_X was still less than 1%.



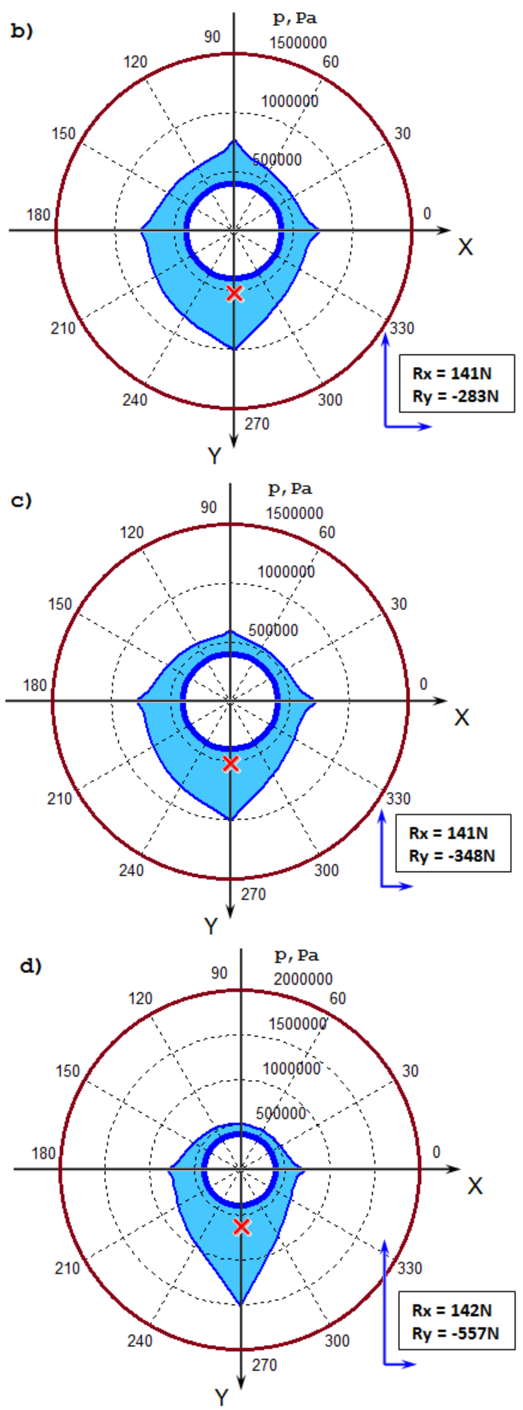


Fig. 3 The diagrams of the pressure distribution

An additional change in the supply pressure to the chambers Y^+ and Y^- in the same direction by another 100% and 50% accordingly led to a further gain in the effect (Fig. 3d). An absolute value of the R_Y had changed by 363N or 187% whereas the change of the R_X force absolute value is 2N or 1.5%. Such force reaction change is comparable to the value of accuracy of the numerical methods and can be considered as insignificant.

Thus, the change of the pressure in the feeding chambers allows to form a direct force action necessary to provide the rotor position control in an active hybrid bearing. The graph in the Fig. 4a shows the change of the absolute values of the lubricant film reaction forces in case of the change in the supply pressure to the chamber Y^+. The Fig. 4b shows the graphs that show the corresponding changes in the reaction forces in comparison to the basic values which the results for the configuration in the Fig. 3a were set as. These changes can be considered as additionally generated forces which are the control actions from the control system of an active hybrid bearing on the controlled object.

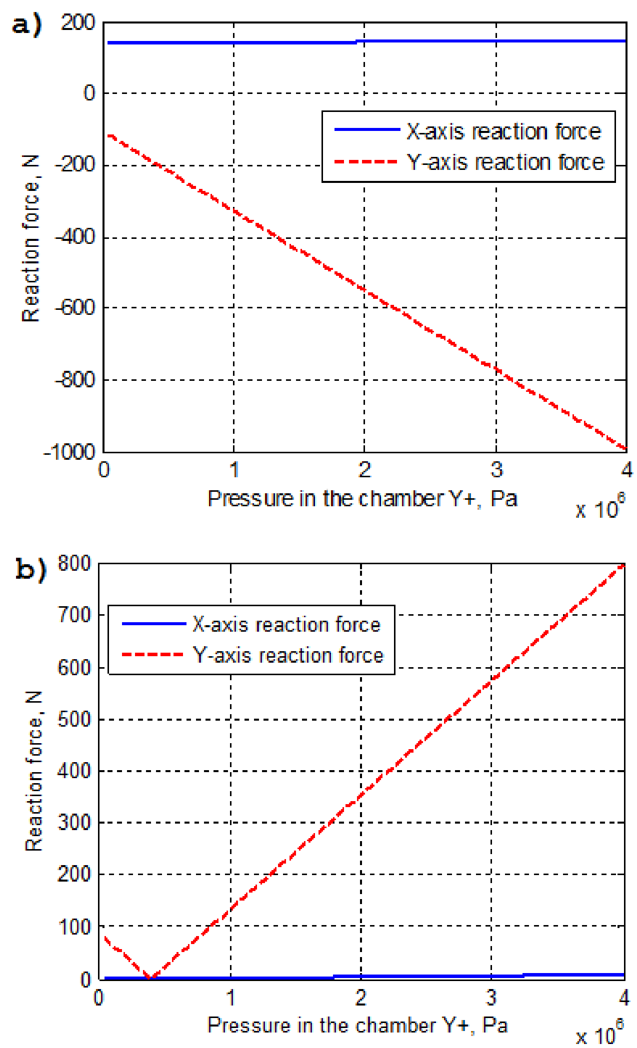


Fig. 4 The relationship between the lubricant film reaction forces and the pressure in the feeding chambers of an active hybrid bearing

V. Conclusion

The results of the research show the presence of a fundamental possibility of a reaction forces control in a hybrid bearing by means of separate change in the pressure of the lubricant supply in one or in a number of chambers. The relationship between the value of the control action and the lubricant pressure in one feeding chamber is close to linear.

This allows to describe these relations with simple, including linear, relations when modeling the functioning of the active hybrid bearings. The direction of the additional control action also corresponds to the location of the feeding chambers, where the change of the pressure takes place. This allows to use the linear dependencies when calculating the necessary relationships between the change in the pressure in the feeding chamber to form a required direction of the resulting control action in the active hybrid bearing.

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