Peculiarities of Reactions Control for Rotor Positioning in an Active Journal Hybrid Bearing

Roman Polyakov, Denis Shutin, Leonid Savin, Alexander Babin

***Abstract*—** The most important part of rotor motion control using the active bearings is the regulation of the forces that act on the rotor from the bearing. In an active hybrid bearing it is implemented by means of changing the supply pressure in the area of friction. The mathematical model of such bearing based on the hydrodynamic lubrication theory allows studying the dependencies of the forces, which act on the rotor from the lubricant film, on the pressure distribution in the feeding chambers. The research results show the principal opportunity to control the rotor motion in a fluid-film bearing through a separate controlled supply of a lubricant to the friction area. An active journal hybrid bearing with four feeding chambers allows implementing a simple rotor position control scheme inside its radial displacement without a necessary consideration of the crosslinks between the regulated parameters in the control system. Moreover, the use of the control system allows increasing the load capacity of the fluid-film bearing, which is used to provide stability under the conditions of complex load schemes.

***Keywords*—**Active fluid-film bearing, active lubrication, hybrid bearing, pressure distribution, forces control, rotor motion control.

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.

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:

$$F_{rf} = \sum_{i=0}^{n} F_i = 0. \quad (1)$$

If initially the rotor was in an equilibrium state, 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:

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

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.

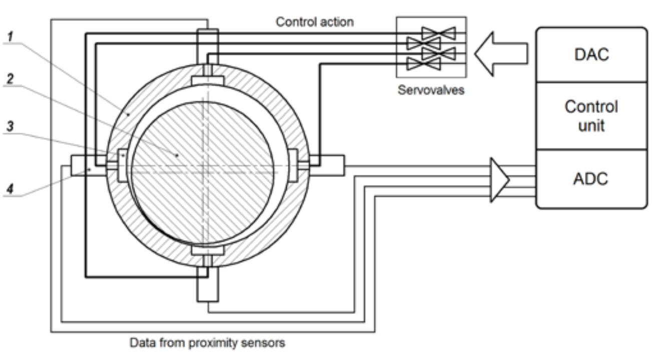


Fig. 1 Scheme of an active hybrid bearing

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 the 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:

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

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:

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

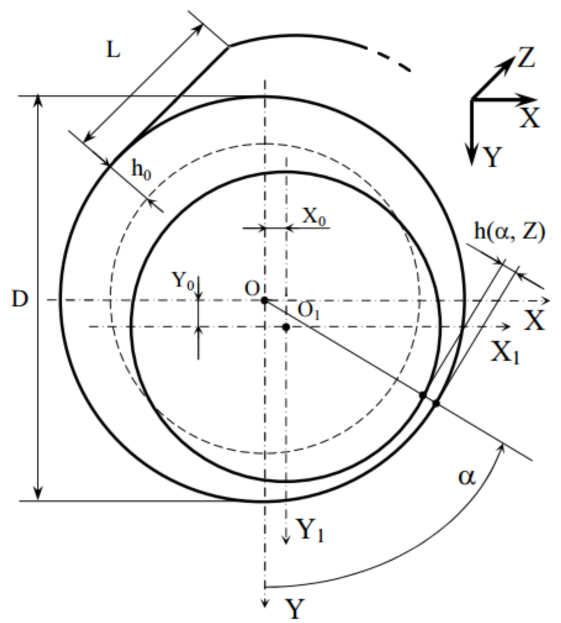


Fig. 2 The geometrical parameters of the modeled bearing

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. Different methods of numerical solution of fluid dynamics tasks are described in [6,7]. The finite differences method is often used for modeling fluid-film bearings. It provides quite accurate results and can be easily algotithmized. So, 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 [7]:

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

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) and using the Simpson's method of numerical integration.

IV. The Results of Mathematical Modeling

The model described above was realized using MatLAB software [9]. 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 as it can be implemented by the control system during the operation. Then we estimated 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 $[0,\ 3.5\cdot10^{-5}]$, 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%.

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 design advantage of the four-chamber hybrid active bearing is the simplicity of the control system which occurs due to the fact that the feeding chambers are located in pairs along the two axes, where control is implemented. This allows direct control along each of the axis by varying the pressure in the corresponding chambers.

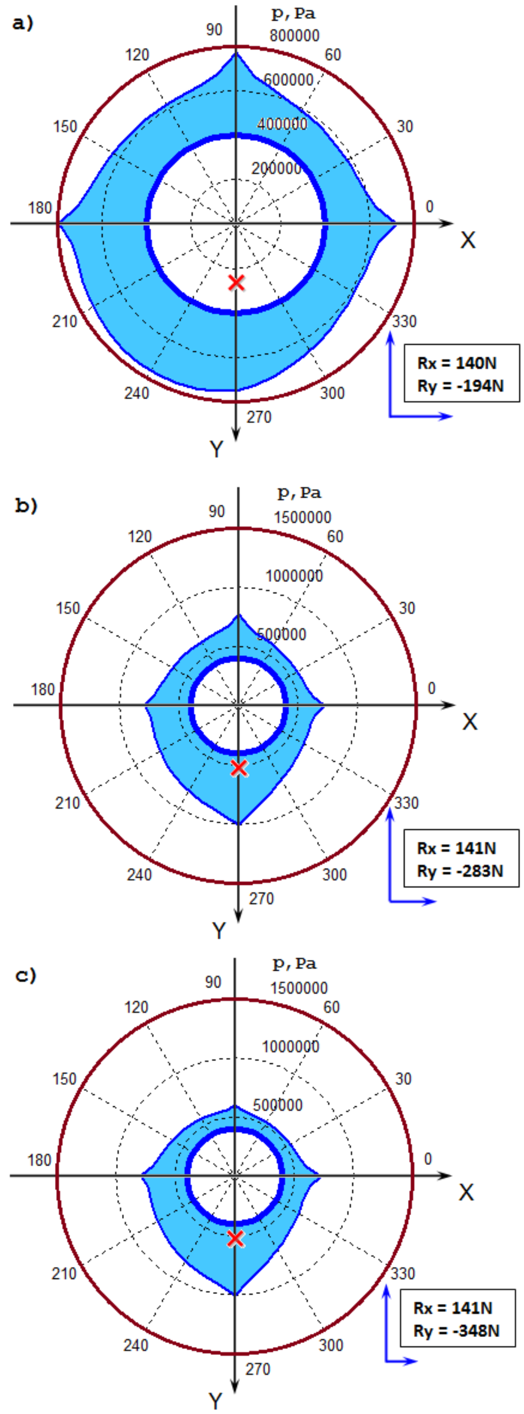


Fig. 3 The diagrams of the pressure distribution

At the same time, the feeding chambers of the active hybrid bearing work in one area of lubricant. The change in pressure in a chamber along one of the axis results in a change in the pressure distribution and also results in a change of reaction force acting along another axis. So, one of the aims of the research was to determine how significant such influence is, and is there a necessity in additional L_{XY} and L_{YX} crosslinks in a control algorithm in order to increase the accuracy of control (fig. 4).

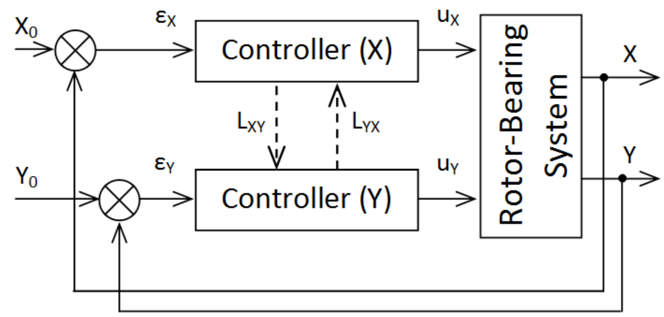


Fig. 4. The structure of the control system of the active journal hybrid bearing

To solve this problem, the change in the reaction along two axes was studied by changing the supply pressure in one chamber. Due to the fact, that the shape of the gap between the rotor and the bearing has a significant influence on the pressure distribution, the change in the reaction was studied with different eccentricities. Herewith, the change of eccentricity could be achieved by means of displacing the rotor in any direction in a XOY plane. It should be taken into account that the value of the radial gap is significantly less than the bearing diameter D. So even the maximum displacement of the rotor along one axis (e.g. axis X, so $\Delta X = h_0$) leads to the minor change of the gap at the point of the feeding chamber at the other axis, i.e. $\Delta Y << \Delta X$ (fig. 5). Accordingly, the hydraulic resistance of the areas and pressure distribution around the feeding chambers at the Y axis also change insignificantly. It means that while the rotor coordinate X and the R_X reaction change under some reason the Y coordinate the R_Y reaction stay almost constant, so the control action along the one axis almost does not influence on the state of the control loop relating to the other axis.

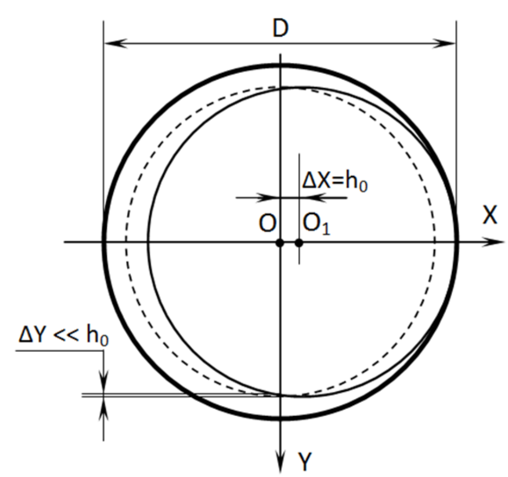


Fig. 5. The geometrical aspects of the radial gap change

In order to test the hypothesis we simulated the operation of the control system by changing the pressure in one feeding chamber (Y^+). The border cases of eccentricity change were studied with rotor displaced from the center position ($e=0$) in the directions: 1) only positive direction of X axis (fig. 5a); 2) only positive direction of Y axis (fig. 5b); 3) equal displacement along both X and Y axis in positive direction (fig. 5c).

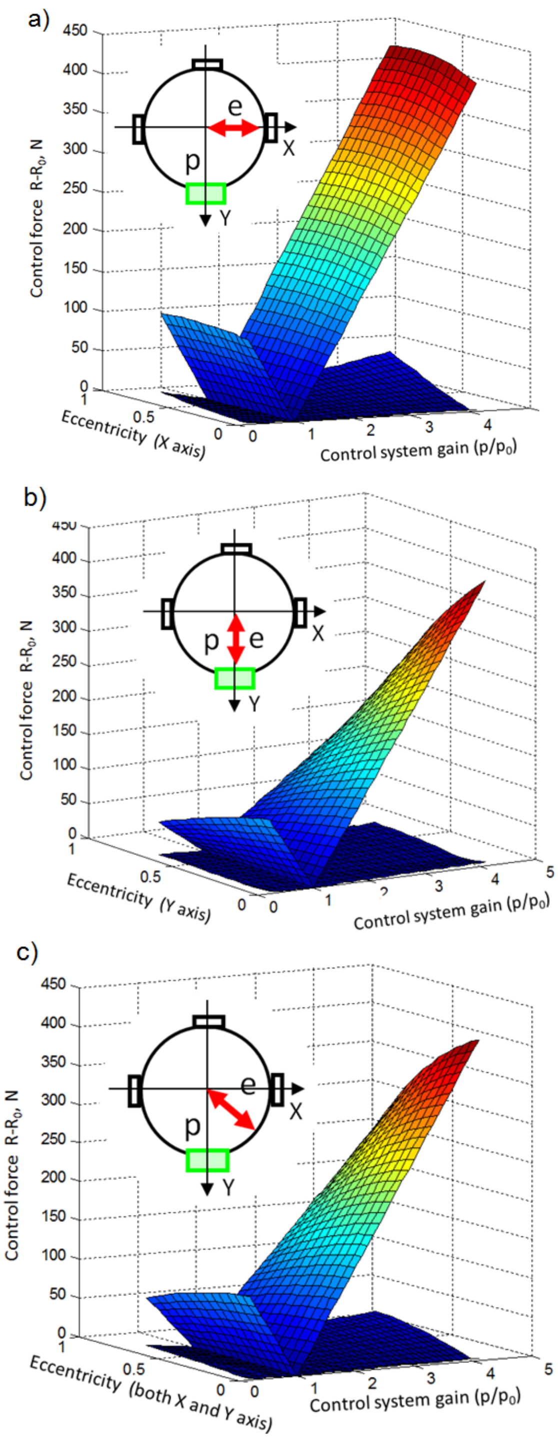


Fig. 5 The relationship between the lubricant film reaction forces and the pressure in the feeding chambers of an active hybrid bearing

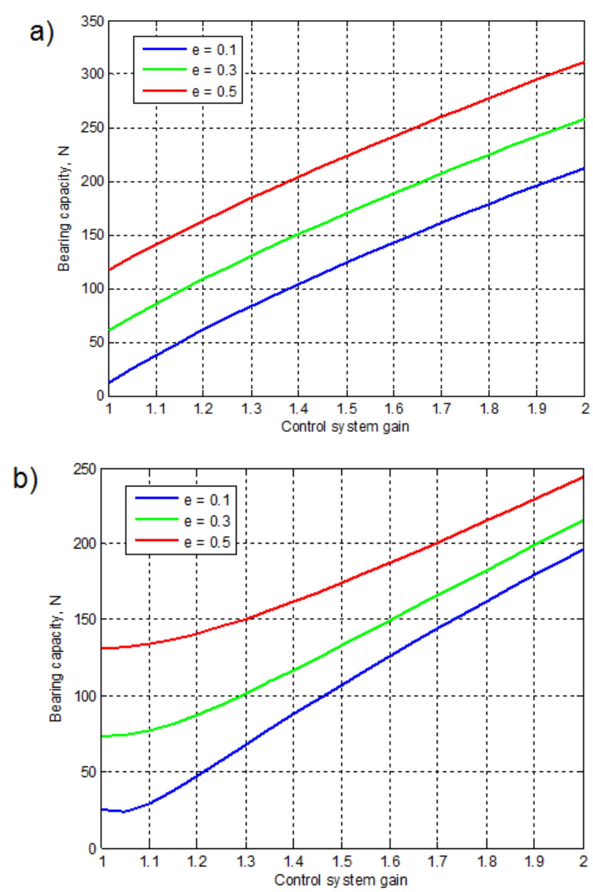
The modeling results show that with any level of control signal and with any direction of rotor displacement with the change of the lubricant supply pressure in the chamber occurs a significant change in the reaction force along the axis of this chamber, whereas the change in reaction force along another axis is no higher than 8% for any situation. Given that only border situations of rotor displacement direction were studied, any other displacement will not result in a higher change in the force reactions.

The obtained result allows making a following conclusion: in a general case the necessity of involving the additional feedback loops in the control system, which would consider the crosslinks, is absent. At the same time, the obtained value of 8% allows making the presented approach a potential field of control accuracy increase when developing the precise rotor positioning systems.

The change of the force factors acting in a fluid-film bearing as a result of the control system functioning unavoidably leads to the change in a load capacity parameter of the bearing, because it defines the workability of the rotor-bearing system at a whole. The full load capacity R of the bearing is determined as follows:

$$R = \sqrt{R_X^2 + R_Y^2}. \qquad (6)$$

During the research the influence of the control signal on the value of full load capacity of the hybrid bearing was studied. Eccentricity there was varied the same way as it was described previously (fig. 5a, b and c), however, to keep unnecessary data the modeling procedure was implemented only for fixed eccentricities 0.1, 0.3 and 0.5.



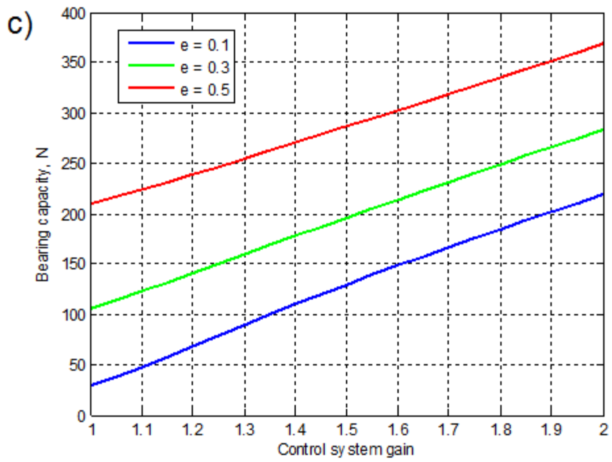


Fig. 5 The relationship between the control system gain and the load capacity of an active hybrid bearing

The modeling results show that the increase in the supply pressure over one of the axis results in a significant increase of the bearings load capacity. Here by means of increasing the load capacity the effect of the direct force influence on the rotor is achieved, if the reaction force ratio is set by the control system so that the resulting force acts in the direction of preferred rotor displacement.

V. Conclusion

The results of the implemented study show the principal opportunity to implement the rotor position control in an active hybrid bearing by means of separate lubricant supply control in the feeding chambers of the bearing. Here for the four-chamber hybrid bearing it is possible to implement a quite simple control scheme, due to the fact that it was determined, that control of the reaction along one of the axis does not significantly influence the change in the reaction force along another axis. This allows implementing a 2-channel system of control with a fully independent control of each of the channels (coordinates X and Y for the rotor position in a journal bearing) with no crosslinks needed between the control parameters. Moreover, the presence of active control in a fluid-film bearing allows to increase the full load capacity of such bearing, which results in a better performance under complex loads.

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