Active hybrid bearings as mean for improving stability and diagnostics of heavy rotors of power generating machinery

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Abstract. The article describes a theoretical study of basic features of active hybrid bearings used as supports for heavy energy-generating machines equipment, primarily turbine generators and their turbines. Due to the adjustable oil supply pressure and using a controller of a rotor position AHB reduces its vibration level. Vibration velocity can be reduced from several times for smaller rotors to tens of percent for heavier ones. The eccentricity of the rotor position in the bearing also affects the efficiency of vibration reduction. In this case also some nonlinear effects are observed. In addition, AHB can possibly be used as a diagnostic tool for monitoring the health of a heavy power generating machinery due to its ability to generate power test impulses for rotor excitation and further analysis of its response.

1. Introduction
Hydrodynamic fluid-film bearings are widely and successfully used in rotating machinery of power generating equipment (turbo-generators, turbine engines, etc.) in many operation modes. With that, the behavior of critical subsystems and units in emergency situations define the reliability of such machinery. Its failure often leads to very expensive repair and even catastrophic results. Reliability becomes even more critical with increasing machine power.

Bearings are critical units of such systems that define the reliable and safe behavior of the machine. A number of defects and undesirable processes in power-generating rotating machinery lead to increase rotor vibration level. In relation to bearings, vibrations can be caused both by external (e.g. unbalanced rotating mass) of internal (self-excited oil whip) reasons. In emergency cases the machine must be stopped, but the shutdown regime may last many hours for heavy machinery with power of 30-50 MW and more. Introducing special means for reducing vibrations into bearings can help avoiding further damage for bearings and the whole machine.

Many authors offer different solutions for this problem, mostly different types of active bearings. Some systems adjust bearing characteristics, such as stiffness and damping, in general, for long period of time. The frequency of impact on rotor is significantly lower than perturbation frequencies. Authors offer adjusting bearings characteristics by changing position of tilting pads [1], changing the elliptical ratio of bearings [2]. Other solutions provide active vibrations suppression with impact frequencies equal to perturbation frequencies, e.g. by applying additional electromagnetic forces [3], by high-speed moving fixed pads [4], the whole bearing bushes [5], by adjusting oil supply pressure [6, 7]. The advantage of the last type of bearings is that they are very similar to conventional bearings used in power
generating rotating machinery, controlling devices are compact and can be set outside bearing units. The basic characteristics of active hybrid bearings (AHB) with adjustable oil supply pressure have been previously investigated in [8].

The present article shows the theoretical study of main features of AHB as bearings of heavy power-generating rotating machinery. The main features considered here are vibration suppression and using such bearings as a mean for diagnostics of such machinery, including active diagnostic and prediction as important parts of modern conception of digital power-generating machinery [9, 10].

2. Structure and mathematical model of active hybrid bearings
The considered AHB for heavy power generating machinery has combined principle of creating bearing capacity, by both hydrostatic and hydrodynamic effects. In the AHB 4 pockets with independent oil supply are evenly located in the middle of the cylindrical bearing surface in circumferential direction, by 2 for each vertical and horizontal direction (X and Y axes). The pockets area does not exceed 2.5% of all the bearing surface, pockets width is 10% of the bearing width. The angular length of each pocket is 22 degrees. Such configuration provides mostly hydrodynamic operation of the rotor-bearing system which is usual for such machines. The pressure in each pocket is controlled by a separate hydraulic servo valve.

The theoretical study in this work was carried out with a mathematical model of a rigid rotor in the described AHB [7]. For the purposes of this study, the rotor was modeled by a point mass; its dynamics is described by the Euler equations and solved by standard ODE45 method. The fluid film model is based on joint solution of the Reynolds equation and the flow balance equation. The solution is implemented using the finite differences method. The problem was solved in an isothermal formulation. P-regulators were used for controlling the rotor position at X and Y coordinates. The output controller signal is proportional to the control error:

\[
\begin{pmatrix}
U_x \\
U_y
\end{pmatrix} = \begin{pmatrix}
K_x \\
K_y
\end{pmatrix} \cdot \begin{pmatrix}
(X - X_0)/h_0 \\
(Y - Y_0)/h_0
\end{pmatrix},
\]

where \( U \) is the control signal at the corresponding axis, \( K \) is the proportional gain, \( X \) and \( Y \) are current coordinate of the journal center in the bearing, \( X_0 \) and \( Y_0 \) are its desired coordinates (setpoint), \( h_0 \) is the radial clearance of the bearing. Setpoints for each set of bearing parameters are chosen as coordinates of the journal center at its equilibrium position in a non-controlled bearing.

The used model also takes into account dynamic properties of servo valves as previous studies show its sufficient influence on control efficiency [11]. Servo valves are considered as an aperiodic first-order element. The time constant for high-speed valves is estimated to be about 0.004 s.

Joint numerical solution of the equations of the described components as a quasi-stationary task allows calculating trajectories the journal center and studying the rotor behavior. The stiffness values of the modeled bearings are in good agreement with the values from the standards [12, 13], which gives the reason to consider that the results obtained with the model are adequate enough to evaluate the operation of studied bearings.

3. Design of research and results
Since the dynamic properties of servo valves limit using AHB in high-speed machinery, the study was implemented for low-speed power generating turbo machines with rotation speed of 1500 rpm, mainly used at nuclear power plants. A four-pole 320 MW turbo-generator [14] was used as the main object for studying. The parameters of its rotor-bearing system are given in table 1. Previously operation of AHB was theoretically and experimentally studied with small rotors, so one of objectives of this study is evaluation of applicability of AHB to heavy machinery.
3.1 Influence of size of rotor-bearing system on efficiency of vibrations suppression

Applying AHB to heavy machinery is connected some restrictions on the magnitude of controlling force acting on rotors due to their weight. For AHB this limitation is the oil supply pressure. The operating pressure of many hydraulic elements, including servo valves, is limited to 20-30 MPa. In addition, at significant pressures, the requirements for bearing seals, pumps and related equipment increase. So, the ability of AHB to reduce values of vibration parameters (displacement, velocity) at different maximum oil supply pressure $P_S$ was studied.

**Table 1.** Main parameters of the modeled rotor-bearing system.

| Rotor & Lubricant | Bearing |
|-------------------|---------|
| Rotor diameter $d$ | Bearing length $L$ |
| Rotor mass $m$ | L/D ratio $L/D$ |
| Rotation frequency $N$ | Radial clearance $h_0$ |
| Imbalance value $\delta$ | Clearance ratio $h_0/D$ |
| Oil dynamic viscosity $\mu$ | Oil pockets width $B_p$ |
| Oil density $\rho$ | Oil pocket length $\alpha$, $L_p$ |

| 561 mm | 670 mm |
| 106490 kg | 1.2 |
| 1500 rev/min | 560 µm |
| 1 mm | 0.001 |
| 0.03 Pa·s | 47 mm |
| 886 kg/m$^3$ | 110 mm |

On the basis of initial set of parameters of a rotor-bearing system in table 1, 8 different sets of such parameters were generated. All 8 rotor-bearing systems, including the one listed in table 1, are characterized by equal value of Sommerfeld number $So=\mu NL D(D/4h_0)^2/mg$ of 0.135, where $D$ is a bearing diameter. Various rotor weights and bearing dimensions were obtained by proportionally scaling the values $L$, $D$, $d$, $m$, $h_0$, $B_p$, $L_p$ by the scaling coefficient $K$, varying from 0.2 to 1.6 in increments of 0.2. This gives values of rotor diameters from 112 to 896 mm and masses from 4260 to 272620 kg. The relative eccentricity of the journal center $\varepsilon$ is 0.57. The P-controllers gains were $K_X=K_Y=20$. The root mean square values (RMS) of vibration velocity along the X and Y axes were evaluated with and without control. The oil supply pressure without control was of 0.2 MPa. The tested $P_S$ values were 7.5, 15, and 21 MPa. The simulation results are shown in figure 1.

![Figure 1](image1.png)

**Figure 1.** Journal center RMS velocities with and without P-controller on (a) X axis and (b) Y axis at different values of maximal oil pressure $P_S$.

Decreasing of absolute and RMS values of vibration velocity at one axis in time domain is illustrated in figure 2. The modeled system has parameters shown in table 1. The initial rotor position in the bearing was at the setpoint. P-controller was turned on at time of 0.25 s, and after that the RMS vibration velocity decreases from 7.6 to 3.7 mm/s.
The intermediate conclusions based on the obtained results are as follows. Using P-controlled AHB allows reducing vibrations within the described conditions. Maximal vibration velocity reduction is achieved for smaller rotors and reached up to 83% from the initial level (from 6.8 to 1.1 mm/s along X-axis and from 4.7 to 1.0 mm/s along Y-axis). Increasing rotor’s mass leads to decrease of vibration reduction rate. Reduction vibration velocities for the rotor with \( m = 272620 \) kg and \( d = 896 \) mm along X and Y axes is 31.1% and 31.6% correspondingly.

Increasing maximal oil injection pressure \( P_S \) increases vibrations reduction level. Herewith, increase the \( P_S \) from 7.5 to 15 MPa results in much more significant gain in efficiency than a further increase to 21 MPa. So, the rational value of \( P_S \) can be determined within the typical limits of operational pressures of hydraulic elements.

Despite the decrease of vibration displacements and velocities magnitudes due to AHB operation, the non-linear properties of the oil film in some cases lead to distortion of rotor orbits. The effect is more noticeable on light rotors with less inertia. It can be seen in figure 3, unevenness of rotor motion increases from the heaviest rotor (figure 3c, \( m = 208720 \) kg) to the lightest (figure 3a, \( m = 17040 \) kg). Increasing stability of rotor motion can be improved by using more advanced regulators than the P-controller, taking into account the nonlinear oil film properties and the presence of cross-links along the X and Y coordinates.

3.2 Influence of rotor eccentricity on efficiency of vibrations suppression
The characteristics of the rotor-bearing system, in particular described by the \( S_0 \) number, can change during the system operation. A typical situation is increase of oil the temperature, which leads to decrease of its viscosity \( \mu \) and increase of rotor eccentricity \( \varepsilon \), the parameter \( S_0 \) also decreases. Nonlinear properties of oil film its nonlinear reactions can then modify the rotor motion control process. So, another study similar to that described in the section 3.1 was implemented, however, now the oil viscosity \( \mu \) was a variable parameter. For the used oil MS-20, its values for typical oil temperatures in bearings of 38, 45 and 55°C are 0.03, 0.02 and 0.01 Pa·s, respectively. \( P_S \) pressure was 21 MPa, other parameters were same to those in the section 3.1. The simulation results are shown in figure 4.

It can be seen that the dependences are qualitatively different along the two axes. On the Y axis, the degree of reducing vibration velocity monotonously decreases with decreasing \( \mu \) and \( \varepsilon \). In a similar situation for the X axis, the greatest reduction in vibration is achieved with medium value of \( \mu \) of 0.02.
The phenomenon can be explained by influence of cross-coupling reactions in the oil film and the preload along the $Y$ axis created by the rotor’s weight. Possibly there is need in additional study to confirm the presence and to give more detailed explanation to this phenomenon.

**Figure 4.** Dependence of vibrational velocities reduction on oil viscosity $\mu$ and rotor eccentricity $\varepsilon$.

### 3.3 Implementation of diagnostic impacts

Since AHB allows creating controllable force impact on rotor, such impacts can not only be used for direct positioning the rotor, but the possible use is also as test impacts generator for checking the rotor-bearing system health. Some diagnostics methods use the rotor mechanical response as a source of diagnostics data, e.g. for determining dangerous transverse cracks [15, 16]. Though the response of the rotor in bearings differs from the single rotor response that requires additional research, the advantage of such approach is ability to test the system during the operation of a machine.

Figure 5 shows the results simulation of the pulse applied to the rotor by increased pressure in the lower $Y$-axis pocket (figure 5c). The rotor response along this axis and its orbit are shown in figures 5b and 5a, correspondingly. Impulse duration is 1/10 of the rotor revolution, and it allows transfer an excitation in a given direction. Testing the rotor in different directions can be implemented by applying a series of similar impulses at different rotation phases. The obtained rotor response can be processed using different methods, e.g. machine learning methods, in order to classify its features connected with different processes in a machine, including early failure diagnostics.

**Figure 5.** Test impact on the rotor by a pressure impulse in AHB: (a) orbit of the journal center; (b) rotor response at $Y$ axis; (c) a pressure impulse in the lower pocket at time of 0.2 s.
4. Conclusions
The study shows that AHB can effectively reduce rotor vibrations caused by imbalance in heavy power generating machinery from several times for smaller rotors to tens of percent for heavier ones. The obtained vibrations reduction rates allow using such bearings for providing more stable operation of such machinery, especially in emergency cases. If due to some reasons vibration velocity exceeds 11 mm/s, it must be stopped immediately according to standards in Russian Federation [17], the values for other countries may vary. Reducing it by mean of AHB to a value within the range of 7..11 mm/s allows continuing operation for more several days. This time can be used for making or preparing to the repair and avoid sudden finance losses. AHB can also be a protection from additional damage to a machine during its emergency shutdown. The uneven rotor motion during control, especially for smaller rotors, can possibly be eliminated by improving regulators comparing to the shown simple P-controller. Also, hydraulic controlling components for AHB can be chosen from serial products and still provide satisfactory vibration reduction level. In addition, AHB can possibly be used as a diagnostic tool for monitoring the health of a heavy power generating machinery due to its ability to generate sufficient power test impulses for rotor excitation and further analysis of its response.

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