Control system with a non-parametric predictive algorithm for a high-speed rotating machine with magnetic bearings

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Abstract. This paper deals with research studies of the magnetic bearing control systems for a high-speed rotating machine. Here, theoretical and experimental characteristics of the control systems with the Model Algorithmic Control (MAC) algorithm and the proportional-derivative (PD) algorithm are presented. The MAC algorithm is the non-parametric predictive control method that uses an impulse response model. A laboratory model of the rotor-bearing unit under study consists of two active radial magnetic bearings and one active axial (thrust) magnetic bearing. The control system of the rotor position in air gaps consists of the fast prototyping control unit with a signal processor, the input and output modules, power amplifiers, contactless eddy current sensors and the host PC with a dedicated software. Rotor displacement and control current signals were registered during investigations using a data acquisition (DAQ) system. In addition, measurements were performed for various rotor speeds, control algorithms and disturbance signals generated by the control system. Finally, the obtained time histories are presented, analysed and discussed in this paper.

Keywords: magnetic bearing; predictive algorithm; high speed rotating machine

1. INTRODUCTION

In recent years, innovative bearing technologies were developed and implemented in many industrial applications. Novel solutions, such as gas bearings, [1], and magnetic bearings solved many restrictions and disadvantages of classic rotor-bearing systems during an operational practice. Immense efforts were undertaken to implement the magnetic support technology in the high-speed rotating machinery. This technology has invaluable advantages in friction reduction in turbomachinery, compressors, generators, etc. [2,3,4,5,6]. Compared with classical mechanical bearings, the magnetic support technology provides benefits, such as a low amplitude level of rotor lateral vibrations, high durability, no mechanical contact between operation elements, such as a rotor and a stator, and a long-term high-speed running ability [7,8]. These features give the magnetic bearings a considerable potential to become a key element in rotating machines [9].

In classic rotating machines, some inefficiency occurring during their operation, e.g. the damages of the bearing elements without additional (external) diagnostic systems, is often almost undetectable. Therefore, a careful selection of the bearing system to support a high-speed rotor is carried out at a design stage. This is usually based on the load capacity, durability and operating condition parameters. Unfortunately, however, bearings are often damaged without previous symptoms arising during operation. Therefore, the significant problem is to develop modern bearing designs with a high diagnostic capacity. Such possibilities can be obtained by means of an active magnetic suspension system.

Applying the active magnetic bearing technology in the high-speed rotating machinery can overcome the physical limitations of the classic bearings. This technology enables us a decrease of stiffness and an increase of the damping ability of the radial bearings, which reduces critical rotor speed values. Active magnetic bearings allow precise control of the rotor position and enable monitoring “online”, diagnosing and identifying the high-speed rotating machines operation [10]. An effective control system with a proper controller should be designed to ensure strictly defined control quality indicators.

In general, magnetic bearing control systems are mostly limited to apply the proportional-integral-derivative (PID) controllers [11,12]. In monograph [11], overviews of magnetic bearings and bearingless electric drives are presented. The principle of operation and mathematical models of active magnetic suspensions (AMS) and the controller design based on the PID algorithm are also described. Additionally, in that monograph, synchronous and asynchronous bearingless electric motors are presented in details. Finally, monograph [12] discusses the magnetic bearing design procedure, advantages of magnetic bearing systems in rotating machines, the PID controller design...
methods for the magnetic suspension control systems and their hardware implementation. Furthermore, paper [13] presents a computer model of an active magnetic bearing used to determine a distribution of the magnetic field. As a result, values of the electromagnetic force and the magnetic flux density were applied for the AMS nonlinearity analysis. On the other hand, in [14], an analysis of the magnetic field distribution of an active magnetic bearing was carried out employing the finite element method (FEM). The obtained computational results were experimentally verified and analysed. Additionally, paper [10] presents a procedure for an experimental identification of dynamic parameters of an active magnetic bearing. Other types of control systems were also investigated in the literature. Namely, studies of magnetic bearing control systems with robust controllers are presented in papers [15,16]. In paper [15], the flywheel control system with an active magnetic bearing (AMB) and the robust controller were shown. Additionally, a design methodology of the optimal controller \( H_{opt} \) was characterised, taking into account uncertainties and nonlinearities of the model. The procedure for selecting weighting functions and reducing control laws was described in that paper as well. Paper [16] presents the control system of a homopolar magnetic bearing with permanent magnets and the control system based on \( H_2, H_{opt}, H_\text{rpm} \) robust algorithms. The obtained system’s registered time responses and frequency characteristics were demonstrated in this study and compared with similar results obtained using the PID controller. Additionally, in [17], a comparison between two control algorithms dedicated to the control system of radial active magnetic bearings was presented. In this monograph, the pole-placement control and sliding control methods were introduced and applied. That paper also presents a collation of experimental results. However, an application of robust control methods may lead to overestimations of parametric and non-parametric noise levels in the models, which are difficult to measure and cause incorrect control signals. For that reason, in that paper, the predictive control method was proposed. In this approach, a predictive control process with a strictly defined reference trajectory results in a more minor variation of the control signal. The developed algorithm enables stable and undisturbed operation of even non-linear and unstable objects. The predictive control methods are based on a structural identification of the control plant [19]. They are divided into two groups: the algorithms with parametric identification and the algorithms with non-parametric identification. The first algorithm group includes the Extended Horizon Adaptive Control (EHAC) algorithm, the Extended Prediction Self-Adaptive Control (EPSAC) algorithm and the Generalised Predictive Control (GPC) algorithm. The second group contains the Model Algorithmic Control (MAC) algorithm, the Model Predictive Control (MPC) algorithm and the Dynamic Matrix Control (DMC) algorithm. The MAC algorithm assumes a simple adaptive control with an impulse response model. In contrast, the MPC algorithm uses a differential control with an impulse response model. However, using the DMC method, an algorithm for predictive control with a step response model is obtained.

The contents of this paper are organised as follows. In the first section, a laboratory model of the magnetically suspended rotor is introduced. Next, theoretical studies of control systems with the predictive and PD algorithms are presented. Then, optimal parameters of the PD controller were found using the pole placement method described in [18,20] and then experimentally tuned on a test rig. The third section presents a laboratory stand with a control system dedicated to the rotor-bearing system. Afterwards, some experimental results are presented along with their analysis. Finally, concluding remarks based on the achieved results have been formulated.

2. MATHEMATICAL MODEL OF THE LABORATORY ROTOR SHAFT

Taking into account a geometric shape of the laboratory rotor shaft under study and the frequency range of dynamic processes to which it will be subjected, with a sufficient accuracy for research purposes this rotor can be represented by a rigid body with 4 degrees of freedom. These degrees of freedom correspond to four generalised coordinates, namely: two translational displacements of the rigid body gravity centre in two mutually perpendicular directions to the axis of rotation, and two rotational displacements about the axes mutually perpendicular to the axis of rotation. This rotor shaft is

![Fig.1. Rigid-body model of the rotor-shaft of high speed rotating machine suspended by the active magnetic bearings](image-url)
supported by means of two active magnetic bearings, as shown in Fig. 1. A rotor shaft motion is described in the ortho-
Cartesian stationary coordinate system Oxyz, the origin of which is attached to the rigid body gravity centre. The axis Oz
coincides with the bearing line, and the axes Oy and Ox are perpendicular to Oz in the vertical and horizontal direction,
respectively. In Fig. 1, the symbols a and b denote the distances from the gravity centre O to the left-hand bearing #1 and
the right-hand bearing #2, factors kOy, kOx and cOy, cOx, i=1,2, are the stiffness and damping coefficients of both bearing’s
susensions in the vertical and horizontal directions, respectively. The symbols kOxy, kOz and cOxy, cOz denote the cross-
coupling components of the stiffness and damping coefficients. Viscoelastic properties of the active magnetic bearings
generally depend on current stiffness coefficient, displacement stiffness coefficient and control law parameters [8]. In the case
under study, when the PD control algorithm is applied, the coefficients of stiffness kOy, kOx and damping cOy, cOx of both
magnetic bearings are equal to 2.88·10^7 N/m and 900.96 Ns/m, respectively, and the all cross-coupling coefficients kOxy,
kOz and cOxy, cOz are equal to zero, i=1,2.
The equations of motion of this model derived using
Lagrange’s equations of the second kind, and taking into
consideration gyroscopic effects related to the shaft rotation, have the following form:

\[ \mathbf{M} \cdot \ddot{\mathbf{r}}(t) + (\mathbf{C} + \Omega \mathbf{G}) \cdot \dot{\mathbf{r}}(t) + \mathbf{K} \cdot \mathbf{r}(t) = \mathbf{F}(t, \Omega^2), \]  

(1)

where \( \mathbf{r}(t) = [\mathbf{y}(t), \mathbf{x}(t), \mathbf{ψ}(t), \mathbf{θ}(t)] \) denotes the vector of
the generalised coordinates corresponding respectively to
translational displacements of the rotor-shaft gravity centre
along the axes Ox and Oy and to angular displacements around
these axes, and \( \Omega \) is the rotor constant angular velocity. The symbols \( \mathbf{M}, \mathbf{G}, \mathbf{C} \) and \( \mathbf{K} \) denote respectively the matrices of inertia and gyroscopic effects, the matrices of damping and stiffness of
the bearing suspension. All forcing terms are
determined, and then matrices \( \mathbf{Q} \) and \( \mathbf{q} \) are obtained from (2) in the form
of the product of the discrete object transfer function and the
Z-transform of the discrete Dirac impulse. Assuming the
discrete delay time equal to 1 and the object model with the
autoregressive part, the polynomial \( V \) can be determined from
the following relationship:

\[ V = \frac{B}{A}. \]  

(3)
The primary goal of the MAC algorithm is to minimise the
divergence between the output signal prediction \( \hat{y} \) and the reference \( w \) according to the following cost function:

\[ J = \sum_{j=1}^{H} (\| \hat{y}(i+j) - w(i+j) \|^2 + \rho u^2(i+j)) \]  

(4)

To design the MAC control algorithm first, the coefficients of
polynomial \( V \) described by Equation (3) should be determined, and then matrices \( \mathbf{Q} \) and \( \mathbf{q} \) ought to be composed:

\[ \mathbf{Q} = \begin{bmatrix} v_0 & 0 & \cdots & 0 \\ v_1 & v_0 & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ v_{L-1} & v_{L-2} & \cdots & v_0 \\ v_H & v_{H-1} & \cdots & v_{H-L} \end{bmatrix} \]  

(5)

\[ \mathbf{q} = [q_1, q_2, \ldots, q_H] = \mathbf{v}^T [\mathbf{Q}^T \mathbf{Q} + \rho \mathbf{I}]^{-1} \mathbf{Q}^T \]  

(6)

The polynomials \( R, S \) and \( T \) of the controllers take the following form:

\[ R = 1 + z^{-1} \sum_{j=1}^{H} q_j (V^2 - V), \]  

(7)

\[ T = K_m \sum_{j=1}^{H} q_j, \]  

(8)

\[ S = \sum_{j=1}^{H} q_j, \]  

(9)
while the control algorithm is described by the equation:

\[ R_u(i) = T_w(i) - S_y(i), \quad (10) \]

In turn, the coefficient \( K_m \) standing in Equation (8) defines the reference trajectory. It is determined by the ratio of the polynomials \( B_m \) and \( A_m \). Usually, the first order trajectory can be calculated from the following relation:

\[ K_m = \frac{B_m}{A_m} \frac{(1-\rho)z^{-1}}{1-\rho z^{-1}}, \quad (11) \]

where \( \rho \) is the controller tuning parameter that determines its speed, i.e. \( 0 < \rho < 1 \).

Based on Equations (2) – (11), the MAC controller polynomials \( R, S, T \) were calculated for the rotor-bearing system shown in Fig. 1. The theoretically analysed algorithms of the designed control system were implemented in MATLAB and Simulink software. Firstly, the single-degree-of-freedom system was analysed, by means of which rotor motion was investigated in the vertical direction only. A scheme of the corresponding control system is presented in Fig. 2.

Responses in time of the rotor displacement and the control currents were obtained using the system presented in Fig. 2. There was determined the response of the rotor observed along the vertical \( O_y \) axis shown in Fig. 1. The rotor has been loaded by the gravity force and the unit step signal. This signal was a reference trajectory and appeared after 100 ms in the form of a kinematic excitation with the value of \( 2.5 \times 10^{-6} \) m. Time histories obtained using the analysed control system are presented in Fig. 3 and Fig. 4.

In Fig. 3, the time histories of the vertical rotor displacement obtained using the control systems with the MAC algorithm and the control system with the PD controller are presented. The control system with the predictive algorithm is characterised by almost ten times smaller control error than the control system equipped with the PD algorithm. The proposed approaches have similar transient processes, and the settling time of the predictive algorithm is shorter than in the control system with the PD algorithm.

Additionally, in Fig. 4 there are shown time histories of the electric currents responsible for control of the vertical motion determined using the control system with the MAC algorithm and the control system with the PD controller. More severe oscillations of the current have been registered in the case of the control system with the predictive algorithm than in the case of the standard system with the PD controller. As a result, the predictive system is saturated for a longer time, but this difference is slight. Additionally, a steady level of control current is lower than in the case of the PD controller. Summarizing, it turned out that the predictive control system worked better than the standard one. In the next part of this paper experimental studies of the non-parametric algorithm will be performed.
4. LABORATORY TEST-RIG AND EXPERIMENTAL RESULTS

The magnetic suspension system of the laboratory rotor-shaft was used for studies of predictive control algorithms. This system is presented in Fig. 5, and it consists of two active radial magnetic bearings and one axial (thrust) bearing. Each of the active radial magnetic bearings can generate a maximum electromagnetic force of 400 N. Air gaps in the radial and axial magnetic bearings are equal to 0.25 mm and 0.3 mm. Thus, the operating point current for each bearing is equal to 4 A at the maximum current value of 8 A. The radial magnetic bearings have 30 windings, whereas the axial one has 60 windings. Eddy current sensors are located in the electromagnet covers. Signals from these sensors provide information about the rotor position for the closed-loop control system, which is fundamental data for a correct operation of a diagnostic system. The principal parameters of the bearing system and the control algorithm are presented in the Appendix.

![Fig.5. The magnetic suspension system of the laboratory rotor-shaft.](image)

A scheme of the control system of the magnetic bearing suspension of the laboratory rotor-shaft is presented in Fig. 6. It consists of the control unit, amplifiers, electromechanical actuators of the supported rotor-shaft and the eddy current sensors. The control unit bases on a PC and dSpace platform with input/output cards with 16-bits A/D and D/A converters. The control algorithms were designed in MATLAB and Simulink software and then they were implemented in the

dSPACE platform employing Matlab Real-Time Workshop (RTW) toolbox.

Within experimental studies, there were analysed step responses using the predictive control system with the MAC algorithm. Initially, responses in time due to a unit step signal were obtained at zero rotational speed of the rotor. This signal created a reference trajectory in the form of a square wave with an amplitude of $2.5 \times 10^{-6}$ m and frequency of 1 Hz. Time responses due to an impact of the input (reference) signal acting in the air gap in the vertical direction have been registered at the left- and right-hand bearings, where the MAC predictive algorithm was applied. These results are presented in Fig. 7. Here, the reference signal is marked using the green line. In this time-history diagram the analogous response obtained by means of the PD controller is marked using the blue line to compare a control quality of the standard and the predictive control system. A time-history of the vertical displacement of the left-hand bearing journal is presented in Fig. 7. The MAC algorithm is characterised by a settling time

![Fig.7. Vertical displacements of the left-hand bearing journals obtained using the control algorithms being tested](image)
of approximately 25 ms. In turn, the value of the maximum overshoot $A$ is equal to 27%. In the case of the control system with the PD controller, the settling time and overshoot $A$ have values of 100 ms and 70% values. The time-history of the vertical displacement of the right-hand rotor-shaft bearing journal is presented in Fig. 8. The predictive control systems with the MAC algorithm are characterised by the settling time equal to 50 ms. The value of the maximum overshoot $A$ is equal to 29%. Here, the use of the control system with the PD controller resulted in the control time and overshoot $A$ have of about 250 ms and 95% values, respectively.

Time histories of the control current responsible for stabilising vertical motion of the left- and right-hand rotor-shaft bearing journal are appropriately presented in Fig. 9 and in Fig.10, respectively. Measurements of this control current were made using the standard control systems with the PD controller and the control system with the predictive algorithm. Here, an operation of the standard control system is characterised by the highest value of current changes. The smallest amplitudes of current changes characterise an operation of the control system with the MAC controller.

In the case of time-history of the control current in the right-hand bearing presented in Fig. 10, an operation of the control...
system with the PD controller is also characterised by the largest value of current changes. Applying the control system with the MAC controller leads similarly to the largest value of the current value and the smallest amplitudes of current changes.

In the following stages of the experimental research, time histories of the left-hand rotor-shaft bearing journal displacements in the vertical direction at constant rotational speeds were recorded. Consequently, in Figs. 11 and 12, there are presented registered time histories, FFT amplitudes and power spectrum characteristics of the rotor-shaft journal vertical displacement registered at the rotational speeds of 300 rpm and 2000 rpm, respectively.

For each of these time histories maximal amplitude values of rotor vibrations were noted. Here, the smallest displacement of the left-hand bearing journal has been observed at 300 rpm, when the system with the PD controller was used. In turn, in the control system with the predictive algorithm, an amplitude of ca. 10% larger value has been registered. However, the smallest vibration amplitude of the right-hand bearing journal vertical displacement was observed at 2000 rpm due to the predictive control algorithm MAC application, and the biggest one has been registered when the control system with PD algorithm was used.

5. CONCLUSIONS

The paper presents results of theoretical and experimental studies of predictive control systems dedicated to the high speed rotating machine with a rotor suspended by means of the active magnetic bearings. For this purpose, a laboratory test-rig equipped with proper control systems has been built. Furthermore, measurements were performed for an excitations in the form of a unit step signal at rotor-shaft zero rotational speed, and without excitation at a constant rotational speeds, when various advanced control algorithms were used. The results of experimental research confirm qualitatively the findings obtained using the theoretical studies. Namely, in both cases, the control system with the PD controller caused a greater control error than the predictive control system. Moreover, peak values of the control currents were also more significant than those obtained using the MAC algorithm.

In the first stage of the predictive control system application, time histories of vertical displacements of the rotor bearing journals caused by the unit step signal were registered. The preliminary experiments have been obtained at zero rotational speed under an excitation in the square waveform with an amplitude of $2.5 \cdot 10^{-6}$ m and frequency of 1 Hz. The parameters determining a quality of the PD algorithm and advanced control algorithms are contained in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>PD</th>
<th>MAC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Settling time [ms]</td>
<td>100</td>
<td>25</td>
</tr>
<tr>
<td>Overshoot [%]</td>
<td>70</td>
<td>27</td>
</tr>
<tr>
<td>Control error in steady-state [m]</td>
<td>$4.6 \cdot 10^{-5}$</td>
<td>$7.5 \cdot 10^{-6}$</td>
</tr>
<tr>
<td>Settling time [ms]</td>
<td>250</td>
<td>50</td>
</tr>
<tr>
<td>Overshoot [%]</td>
<td>95</td>
<td>29</td>
</tr>
<tr>
<td>Control error in steady-state [m]</td>
<td>$4.1 \cdot 10^{-5}$</td>
<td>$1.6 \cdot 10^{-5}$</td>
</tr>
</tbody>
</table>

In Table 1, there are compared values of the settling time, overshoot and the control error in steady-state operational conditions obtained using the different control algorithms, i.e. PD and MAC controllers. Here, in each experimental test, the standard system resulted in worse dynamic properties than the prediction one. The most significant difference was observed for the steady control error value, which was greater by one
magnitude order in the case of the standard system with the PD controller than when the predictive control system was used. On the other hand, the predictive control system with the MAC algorithm resulted in the smallest overshoot and the shortest settling time.

Next, time histories of the vertical displacement of the left-hand bearing journal at constant rotational speeds were recorded. These tests were carried out at rotational speeds of 300, 600, 900, 1500 and 2000 rpm. For each of the registered response maximal values of the vibration peaks were registered and included in Table 2.

From this comparison, it follows that the predictive algorithm indicated better properties than the PD algorithm. At small rotational speeds, i.e. up to 300 rpm, the time histories of bearing journal displacements obtained using the standard algorithm were characterised by at least similar or smaller maximal amplitudes than those when the predictive system was applied. But with an increase of the rotational speed of the rotor, the analogous time histories of journal displacements recorded when using the prediction system were characterised by smaller vibration amplitudes. It is to remember that magnetic suspensions are structurally unstable system. Therefore, they need a control system, which assures a stable suspension operation around an operational point. Determining controller settings is a complex process for the unstable system. Thus, in the framework of performed research the different controllers were tested, and optimal controllers for the active magnetic suspensions were looked for. In this study, the predictive controller was compared to the PD controller. The PD algorithm is a standard type of controller in the automatic system applied for active magnetic bearings. Therefore, the comparison of predictive controllers with the PD controller is justified.

The presented results of tests analyses of predictive control systems confirm a legitimacy of their use and development in the magnetic suspension systems of the high-speed rotating machines. The constructed laboratory test-rig for the magnetically suspended rotor-shaft, together with the positive effects of preliminary studies, have opened new perspectives for development of research devoted to magnetic bearing suspension systems of the high-speed rotating machines. The results obtained during these studies are the basis for identifying and monitoring systems of rotors supported by active magnetic bearings. In further research in this field, dynamic behaviours at greater rotor speeds will be investigated when using other types of advanced algorithms, e.g. robust and slide controllers.

TABLE 2. Maximum values of the vibration amplitudes at constant rotational speeds of the rotor

<table>
<thead>
<tr>
<th>Rotational speed [rpm]</th>
<th>PD</th>
<th>MAC</th>
</tr>
</thead>
<tbody>
<tr>
<td>300</td>
<td>7.984e-06</td>
<td>8.651e-06</td>
</tr>
<tr>
<td>600</td>
<td>9.837e-06</td>
<td>7.310e-06</td>
</tr>
<tr>
<td>900</td>
<td>6.062e-06</td>
<td>5.426e-06</td>
</tr>
<tr>
<td>1500</td>
<td>6.085e-06</td>
<td>6.143e-06</td>
</tr>
<tr>
<td>2000</td>
<td>1.052e-05</td>
<td>6.434e-06</td>
</tr>
</tbody>
</table>

REFERENCES


APPENDIX

TABLE 3. Numerical parameters of the laboratory rotor-shaft

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of rotor</td>
<td>m</td>
<td>5.6</td>
<td>kg</td>
</tr>
<tr>
<td>Polar moment of inertia</td>
<td>I</td>
<td>0.0246</td>
<td>kg m²</td>
</tr>
<tr>
<td>Diametral moment of inertia</td>
<td>I₀</td>
<td>2.763·10^-7</td>
<td>kg m²</td>
</tr>
<tr>
<td>Bearing stifeiness coeffient</td>
<td>k</td>
<td>2.88·10⁹</td>
<td>[N/m]</td>
</tr>
<tr>
<td>Bearing damping coeffient</td>
<td>c</td>
<td>900.96</td>
<td>[Ns/m]</td>
</tr>
<tr>
<td>Prediction horizon</td>
<td>H</td>
<td>10</td>
<td>[s]</td>
</tr>
<tr>
<td>AMB air gap</td>
<td>x₀</td>
<td>0.25</td>
<td>[mm]</td>
</tr>
<tr>
<td>Current work point</td>
<td>i₀</td>
<td>1.5</td>
<td>[A]</td>
</tr>
<tr>
<td>Rotor total length</td>
<td>l</td>
<td>0.373</td>
<td>[m]</td>
</tr>
<tr>
<td>Bearing span</td>
<td>a+b</td>
<td>0.238</td>
<td>[m]</td>
</tr>
</tbody>
</table>

This work was co-financed by Military University of Technology under research project UGB 899.

ACKNOWLEDGEMENTS

This research was co-funded by the National Center of Research and Development as the grant titled “The use of surface engineering new technologies and magnetic bearings in the construction of a miniature turbine jet engine”. This work was co-financed by Military University of Technology under research project UGB 899.


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