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Adaptive Control of the 1-DOF Active Magnetic Bearing

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***Abstract.** active magnetic bearing is the electromechanical device allowing to suspend the rotor of the electric machine. Friction is eliminated and high speed of rotation can be achieved. However, the parameters of the electromechanical system may change during operation. Adaptive control system allows to maintain stability under varying parameters.*

***Keywords:** active magnate bearing, adaptive control system.*

I. INTRODUCTION

Active magnetic bearings (AMB) are increasingly used in various fields of industry [1],[2]. The absence of mechanical contact makes it possible to use them in ultra-high-speed electric drives (now it is much in demand) [3], [4]. The main trend of AMB development is the improvement of the control system: application of the modern elemental base of electronic components and improvement of control algorithms [5], [6]. The use of a traditional AMB control system with a PID controller is limited in some areas. For example, there are systems where the parameters can change during operation, while for the PID controller it is necessary to know exactly the parameters of the control object. In addition, the general trend of modern management systems is their intellectuality. In AMB, it is design engineering of self-tuning systems. One type of such systems is the adaptive control system [7]-[9].

II. MATERIALS AND METHODS

Let's consider the control system of 1 degree of freedom (1-DOF) active magnetic bearing. It includes rotor position sensor, a regulator, power amplifiers and electromagnets. (Fig. 1).

There are various AMB control systems, one of the most simple is a system based on a PD controller with current control. Let's assume that we want the free motion of a suspended body, described by the known equation [2]

$$m\ddot{y} - k_y y = k_i i \qquad (1)$$

under identical initial conditions exactly coincided with free motion of a mechanical linear oscillator with viscous friction described by the equation $m\ddot{y} + b\dot{y} + cy = 0$ or

$$\ddot{y} + 2\zeta\omega_0\dot{y} + \omega_0^2 y = 0\,. \qquad (2)$$

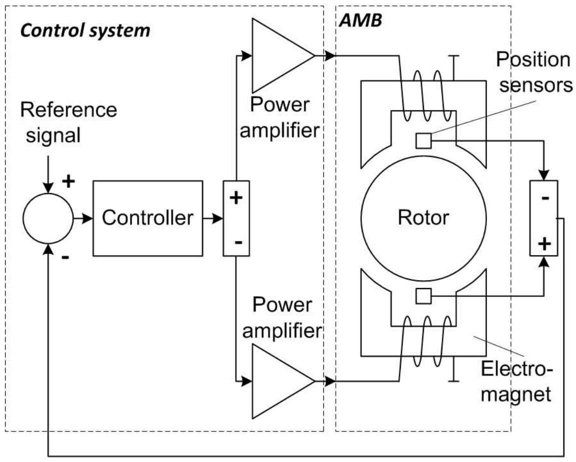

Fig. 1. Principle of operation of active magnetic bearing

Here m - is the rotor mass, y - is the position coordinate, k_y - is the coefficient "force-current", k_i - is the coefficient "force-displacement", b - is the coefficient of damper viscous friction, c - is the spring stiffness, $\zeta = b/(2m\omega_0)$ - is the dimensionless damping parameter, $\omega_0 = \sqrt{c/m}$ - is the frequency of the undamped free vibration. Obviously, for the identical systems' motions under identical initial conditions, it is necessary and sufficient that the current values of the accelerations of both systems to be identical. Expressing acceleration $\ddot{y}$ out of (2) and substituting it in (1), we obtain the current control action:

$$i = -(k_p y + k_d \dot{y})\,, \qquad (3)$$

where $k_p = (m\omega_0^2 + k_y)/k_i$ - is the feedback gain factor for the displacement and $k_d = 2\cdot\zeta\cdot m\cdot\omega_0/k_i$ - for the speed.

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As it is known, the controller that implements the control action (3) is called the PD controller. The block diagram of the 1-DOF AMB control system is shown in Fig. 2 [10].

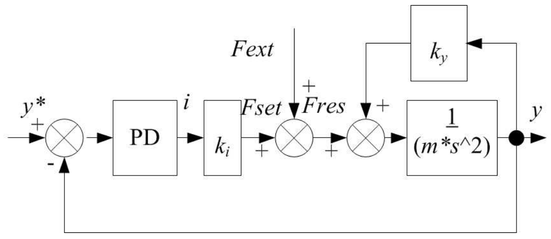

Fig. 2. Block diagram of the 1-DOF AMB with PD controller

Where y *- is the reference position signal, i - is the reference current, *Fset* - is the reference force, *Fext* - the external disturbance. *Fres* = *Fset* + *Fext*.

Let us set the following real values for the parameters: m = 10 kg, k_i = 500 N/A, k_y = 2,5 *10^6 N/m, the frequency of undamped oscillation ω_0 = 350 rad/s and the dimensionless damping parameter ζ = 0,5. Then k_p = 7450 A/m, k_d = 7 Ac/m.

The Simulink model is shown in Figure 3.

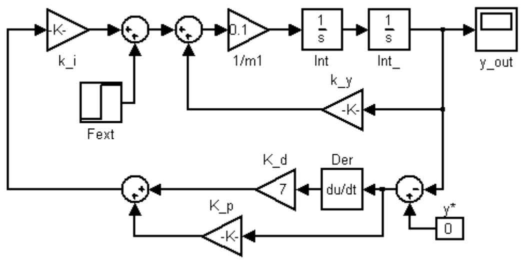

Fig. 3. Simplified Simulink model of the 1-DOF AMB with PD controller

As a result, with an external stepped action of the force *Fext*= 200 N at time point 0,01 s, we obtain the transient process shown in Fig. 4.

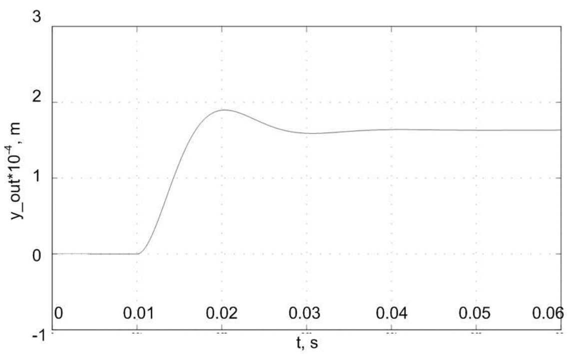

Fig. 4. Transient process under the influence of an Fext= 200 N

However, if the “negative” stiffness increases, k_y = 4*10^6 N/m, the system becomes unstable (Fig.5).

It is possible to recalculate the values of the coefficients for the new regulator parameters, then the view of the transient process will become again as in Fig. 4. However, the problem is that the “negative” stiffness parameter can change during operation. For example, when the electric motor is turn on, the “negative” stiffness of the suspension is added to the “negative” stiffness of the electric motor, caused by the radial magnetic forces of stator, which will depend on the currents in the motor windings. In addition, the rotor parameters can change, for example, change of the spindle mass in the process of filament winding. In these cases, adaptive control is needed (maintenance of the stability of the suspension dynamic qualities in a wide range of changes in its parameters).

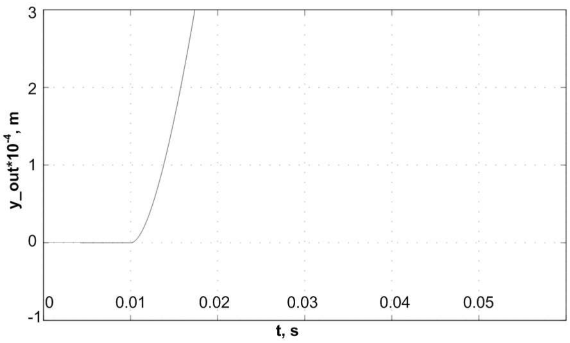

Fig. 5. Transient process under the influence of an external force 200 N (k_y = 4*10^6 N/m)

III. RESULTS AND DISCUSSION

Let's consider a system with adaptive control.

The current-controlled suspension equation can be written as:

$$\ddot{y} = f(y, \dot{y}, i) . \tag{4}$$

Condition of incompleteness of information about the control object: for all possible values of the arguments of the function (4), the partial derivative with respect to the control current is positive $\partial f / \partial i > 0$.

Let there be given a program function $f = f^0$ that implements the required action of motion of the body. Given a differential program of the form $\ddot{y} + 2\zeta\omega_0\dot{y} + \omega_0^2 y = 0$, we have

$$f^0(y, \dot{y}) = -(2\zeta\omega_0\dot{y} + \omega_0^2 y) \tag{5}$$

Now the problem arises: for each time moment t and its corresponding state $y(t), \dot{y}(t)$ it is required to find a value $i^0(t)$ for which

$$f\ (y, \dot{y}, i^0) = f^0(y, \dot{y}) \tag{6}$$

If there were complete information about the suspension, i.e. for a known function $f\ (y, \dot{y}, i)$, the required value i^0 could be found on the basis of an analytical solution of equation (6). In the case under consideration, equation (6) can only be solved algorithmically:

$$di/dt = \rho_1 \cdot \Delta f \ , \Delta f = f^0(y,\dot{y}) - f \ (y,\dot{y},i\ ) \,. \quad (7)$$

Indeed, with the assumption made about the function $f\ (y,\dot{y},i)$ for each instant of time, the following limit is valid:

$$\lim_{\tau\to\infty} i(\tau) = i^0(t)\,, \quad f(y,\dot{y},i^0) = f^0(y,\dot{y})\,, \quad (8)$$

where τ - is fast time.

This is a consequence of the fact that the tracking system (7) is stable.

Substituting the expressions from (6) and (4) into (7), we obtain

$$di/dt = \rho_1 \cdot \Delta f \ , \ \Delta f = -(\ddot{y} + 2\zeta\omega_0\dot{y} + \omega_0^2 y) \,. \quad (9)$$

The block diagram of the control loop is shown in Figure 6.

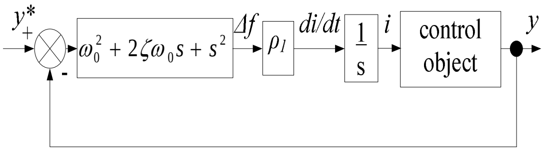

Fig. 6. Block diagram of the control loop with adaptive control

The peculiarity of the algorithm (9) is that it is not necessary to know the explicit functional dependence.

This functional dependence may be unknown. The structure of the control algorithm does not explicitly contain the parameters of the control object. Therefore the algorithm (9) is in the full sense adaptive. The practical implementation of the algorithm is based on the measurement of acceleration.

Tracking accuracy is substantially determined by the value of *ρ1*. For a finite value of *ρ1*, the required action of body motion will be approximate. The degree of approximation increases with increasing *ρ1*. The value of ρ1 should be selected according to the following condition: the current values must be processed substantially faster than in the program system (2).

The Simulink model of the 1-DOF active magnetic bearing control system is presented in Figure 7

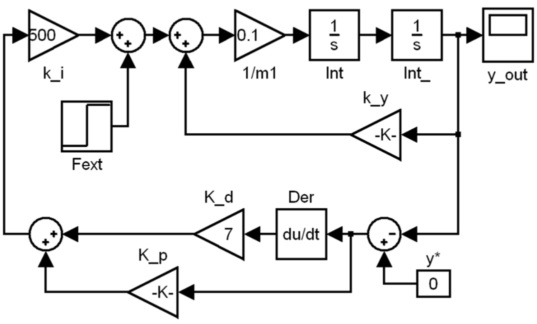

Fig. 7. Simplified Simulink model of the 1-DOF AMB with adaptive control

The parameters of the adaptive PD controller are adjusted to create a control law (2) (*ω₀* = 350 rad/s and *ζ* = 0,5.)

Figures 8 and 9 show the transient processes in the system with the adaptive regulator under the external action of a force of 200 N with coefficients of “negative” stiffness k_y = 2,5*10[6] N/m and k_y = 4*10[6] N / m.

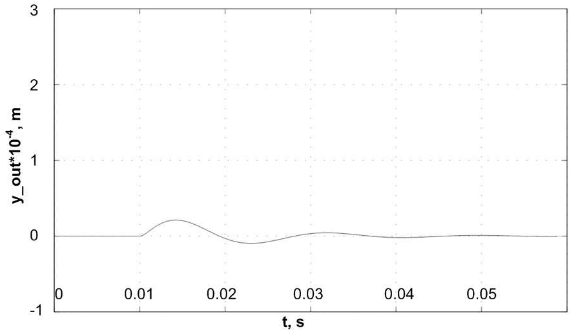

Fig. 8 Transient process in the adaptive control system at k_y = 2,5*10[6] N/m

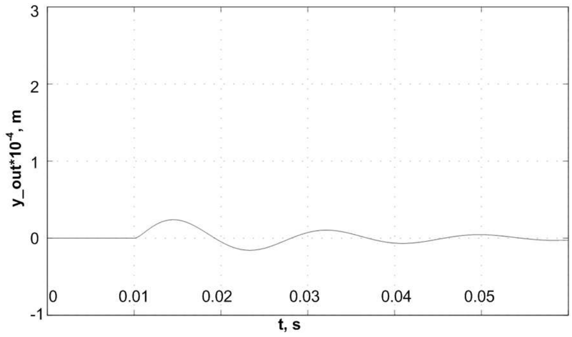

Fig. 9 Transient process in the adaptive control system at k_y = 4*10[6] N/m

As the graphs show, the transient processes are almost identical. The results are similar when the rotor mass is changed.

IV. CONCLUSION

The article deals with the synthesis of the adaptive regulator system for a 1-DOF active magnetic bearing.

The method of constructing a regulator based on the desired parameters of the system transient processes is presented. Unlike other papers, the results of computer Simulink simulation 1-DOF active magnetic bearing are presented. A quantitative comparison of the transient processes in the active magnetic bearing with a PID controller and with an adaptive control system is given. The advantage of the adaptive control system is shown. The system remains stable even with significantly varying of “negative” stiffness coefficients.

It should be noted that the adaptive controller can be applied not only to a system in which the parameters of a control object are changed over time, but also in systems in which the parameters of the control object are unknown. Application of such systems for AMB enables to considerably simplify system design engineering and setting.

V. ACKNOWLEDGMENTS

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REFERENCES

[1] G. Schweitzer, H. Bleuler, and A..Traxler, *Active Magnetic Bearings.* Zurich, VDF Hochschulverlag AG, pp. 31-77 1994

[2] Y. Zhuravlyov *Active Magnetic Bearings: Theort, Design, Aplication*. St. Petersburg: Politechnica, 2003, pp. 12-25

[3] G. Schweitzer, and E. Maslen, *Magnetic Bearings - Theory, Design and Application to Rotating Machinery*. Springer-Verlag Berlin Heidelberg , pp. 33-78 2009.

[4] A.Chiba, T.Fykao, O.Ichikawa, M.Oshima, M.Takemoto and D.G.Dorrell, *Magnetic Bearings and Bearingsless Drives.* ELSEVIER, pp. 127-135.

[5] V.S. Polamraju, G.V. Sobhan, K. Nagesh, J. Amarnath and M. Subbarao. "Stabilization of active magnetic bearing system using single neuron PID controller", ARPN Journal of Engineering and Applied Sciences vol. 9, no. 7, july 2014

[6] Ming-Mao Hsu, Seng-Chi Chen, Van-Sum Nguyen and Ta-Hsiang Hu "Fuzzy and online trained adaptive neural network controller for an AMB system", Journal of Applied Science and Engineering, vol. 18, No. 1, pp. 47 -58. 2015

[7] T.A.Izosimova and Yu. K. Evdokimov "Adaptive control of the dynamic behavior of the rotor in active magnetic bearings", Dynamics of complex systems - XXI century No. 3, pp. 37–42, 2014.

[8] Silu You "Adaptive backstepping control of active magnetic bearings", Bachelor of Electrical Engineering Huazhong University of Science and Technology July, 2007.

[9] Kai-Yew Lum, Vincent T.Coppola and Dennis S. Bernstein "Adaptive autocentering control for an active magnetic bearing supporting a rotor with unknown mass imbalance", IEEE transactions on control systems technology, vol.4, No.5, september 1996.

[10] S. Loginov "Suspension control in a bearingless reluctance motor without current feedback", Electrotechnical complexes and control systems No.4, 2010 pp. 33-37.