Optimization based AMB Controller Design and Verification for Flexible Rotors

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***Abstract*—Engineering costs especially for controller design are substantial and obstruct active magnetic bearings (AMBs) for broader industrial applications. An optimization based AMB controller design method is developed to solve this problem. The optimization criteria are selected to describe AMB practical performance. Controller components are chosen considering that parameters can be manually interpreted and modified on-site for commissioning. A multi-objective genetic algorithm (MOGA) toolbox is used to tune these controller component parameters to minimize the design criteria automatically. The method has been verified in a controller design process for an AMB levitated machine. With this method, the engineering cost for controller design can be reduced significantly.**

I. Introduction

Compared to traditional mechanical bearings, AMB offers many benefits for machines, for instance, lubrication free, contamination free, and long life operation, because there is no mechanical contact between the rotor and the stator[1]. In addition, AMB has inherent monitoring, adjustable stiffness and damping features which are really important for high speed and high reliability applications. There is a trend to equip AMBs in high speed machines to meet performance requirements[2]. One of the main barriers today for broader industrial AMB applications is the cost. AMBs can double the price of a large machine. Engineering costs especially controller design are substantial for AMB systems. Another barrier is that AMB requires specific knowledge. Tuning of controller parameters requires engineering experience in AMB area, especially for flexible rotors applications.

Genetic algorithms can solve these problems which have been widely used to industrial design and optimization applications[3]. Skalak introduced genetic algorithm to choose AMB actuator parameters[4]. Wei introduced multi-objective optimization to tune AMB position controller parameters[5][6]. However, the optimization criteria in [5] are not intuitively related to AMB performances which increases commissioning difficulty. A more practical method is proposed in this paper: optimization criteria are chosen to describe AMB system performances, and controller components are selected considering that parameters can be manually interpreted and modified on-site for commissioning. The verification shows that the method works well in a test machine.

This paper is organized as follows: In Section 2, the AMB-rotor system modeling is introduced. In Section 3, the optimization criteria are discussed and controller design is given and analyzed. In Section 4, the experimental verification is given. Finally, a conclusion is given in Section 5.

Figure 1. Test rotor with finite element model.

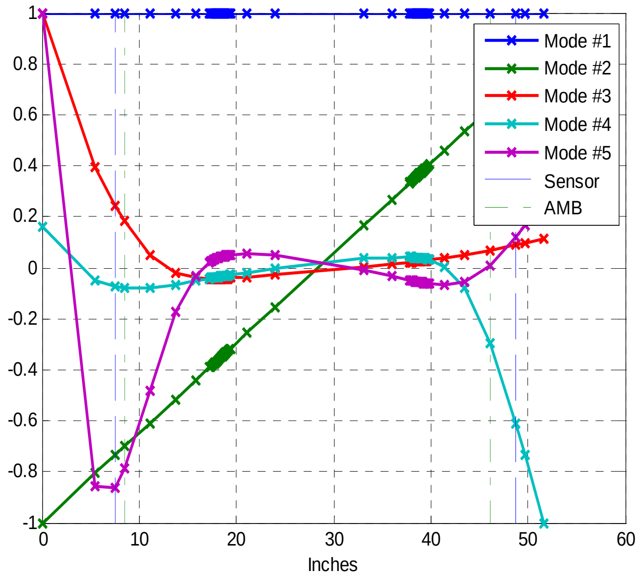

Figure 2. Undamped rotor modal shapes with sensor and actuator locations.

II. AMB-Rotor System Modeling

A. Rotor model

In order to describe rotor flexible modes precisely, a test rotor is modeled in finite element method with 75 nodes, as shown in Figure 1. Figure 2 shows the analyzed undamped rotor modal shapes from the FE model. The equation of motion can be described as follow, in which the matrices are calculated from the FE model:

$$
\begin{cases}
\dot{X}_r = [A_r + \Omega G_r]X_r + B_{r\_a}F_a + B_{r\_d}F_d \\
Y_a = C_{r\_a}X_r \\
Y_s = C_{r\_s}X_r
\end{cases}
\tag{1}
$$

where X_r are the rotor displacements in modal coordinate, G_r is the gyroscopic matrix, and Ω is the rotational speed. Y_a and Y_s are the rotor displacements at actuator and sensor position, respectively. B_{r_a} and B_{r_u} are the input matrices for actuator force and unbalance disturbance, respectively. F_a and F_d are the actuator force and disturbance force, respectively.

B. Actuaor model

Actuator model can be described as follows:

$$
\begin{cases}
\dot{X}_a = A_a X_a + B_{a\_u}u + B_{a\_y}Y_a \\
F_a = C_a X_a
\end{cases}
\tag{2}
$$

where X_a are the internal states of actuators and power amplifiers, and u are the control signals. B_{a_u} and B_{a_y} are input matrices for control signals and rotor displacement at actuator position.

C. Complete model

The complete model can be assembled by combing the rotor model and the actuator as follows:

$$
\begin{cases}
\begin{bmatrix} \dot{X}_r \\ \dot{X}_a \end{bmatrix} = \begin{bmatrix} A_r + \Omega G_r & B_{r\_a}C_a \\ B_{a\_y}C_{r\_a} & A_a \end{bmatrix} \cdot \begin{bmatrix} X_r \\ X_a \end{bmatrix} + \\
\qquad \begin{bmatrix} B_{r\_d} \\ 0 \end{bmatrix} \cdot F_d + \begin{bmatrix} 0 \\ B_{a\_u} \end{bmatrix} \cdot u \\
Y_s = \begin{bmatrix} C_{r\_s} & 0 \end{bmatrix} \cdot \begin{bmatrix} X_r \\ X_a \end{bmatrix}
\end{cases}
\tag{3}
$$

III. OPTIMIZATION BASED CONTROLLER DESIGN

A. Optimization Criteria

Guoxin had an insightful description about AMB controller objectives in his PhD thesis[7]:

1) In the low frequency range below 1.0 Hz, the controller was an integrator with a flat static gain.

2) In the rigid body frequency range from 1 Hz to 100 Hz, the controller was essentially a PD controller with stiffness and damping required to stabilize the rigid body modes.

3) In the frequency range above 100 Hz, the controller placed a series of complex poles and zeros (generalized second order filters) to stabilize the flexible rotor and substructure modes. The controller gain rolls off above 500 Hz.

Combined with ISO standards[8][9], the optimization criteria for controller design are given as follows, and the expressions are given to calculate the criteria values during the optimization process.

a) closed loop system is stable

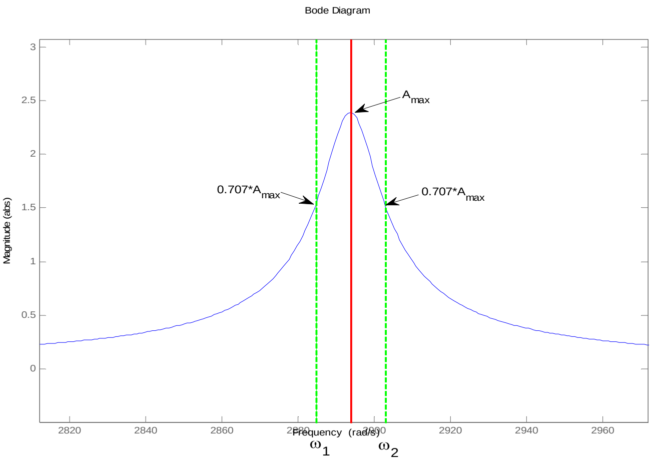

Figure 3. Rotor benidng mode Q factor calculation.

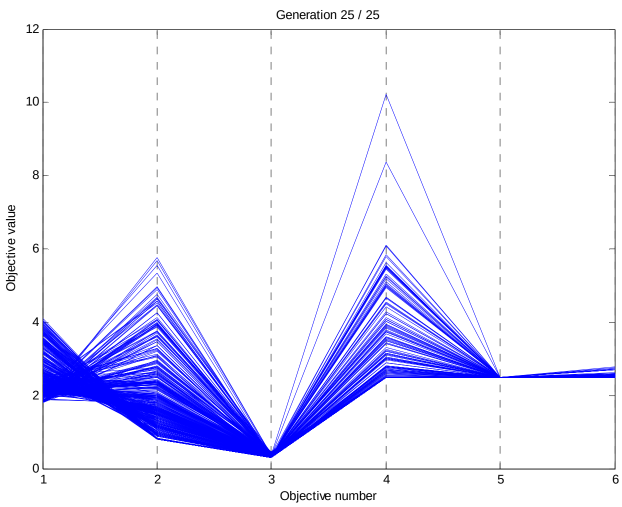

Figure 4. Optimization results for controller design.

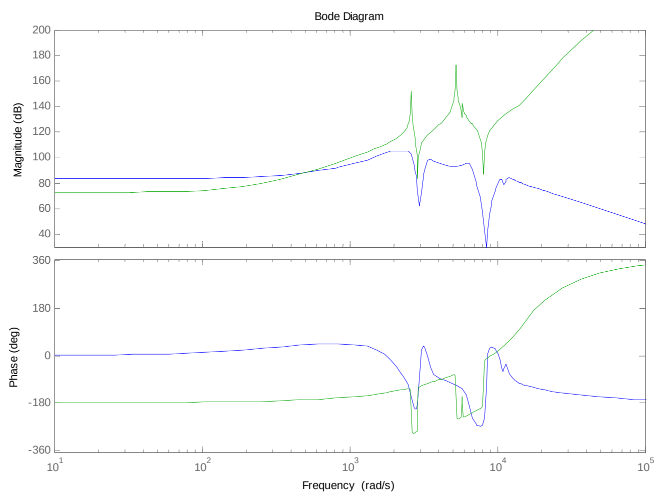

Figure 5. Designed controller(blue) vs inversed plant model(green).

$$G_s(s) = \frac{1}{1 + P(s) \cdot C(s)} \tag{4}$$

b) closed loop system has robust performance (sensitivity function is less than 3)

$$\max\{S(s)\} = \max\left\{\frac{1}{1 + P(s) \cdot C(s)}\right\} < 3 \tag{5}$$

c) closed loop system has stable margin at critical speeds

$$G_Q(s) = \frac{P(s)}{1 + P(s) \cdot C(s)} = P(s) \cdot S(s) \tag{6}$$

$$Q = \frac{A_{\max}(G_Q)}{\omega_2 - \omega_1} \tag{7}$$

d) controller has low gain at high frequency for lower noise sensitivity

$$J = \sum_{f_h}^{f_B} A_c(f) \tag{8}$$

B. *Controller components*

A PID controller is used to stabilize AMB-rotor system rigid modes. A roll-off filter is used to eliminate high frequency noises. Lead-lag compensators and notch filters are added to deal with flexible modes. The equations are shown as follows respectively:

$$G_{PID}(s) = K_p + \frac{K_i}{s} + \frac{K_d s}{T_f s + 1} \tag{9}$$

$$G_R(s) = \frac{\omega_r^2}{s^2 + 2\zeta_r \omega_r s + \omega_r^2} \tag{10}$$

$$G_L(s) = \frac{s^2 + 2\zeta_2 \omega_2 s + \omega_2^2}{s^2 + 2\zeta_1 \omega_1 s + \omega_1^2} \tag{11}$$

$$G_N(s) = \frac{\alpha^2 s^2 + 2\alpha\xi\omega_n s + \omega_n^2}{s^2 + 2 \cdot 0.05 \cdot \omega_n s + \omega_n^2} \tag{12}$$

A multi-objective genetic algorithm(MOGA) toolbox, e.g. Matlab global optimization toolbox, could be used to tune controller parameters automatically. Figure 4 shows the optimization results after 25 generations.

IV. Verification Results

In order to verify the design method, an experimental verification was executed in a test machine as shown in Figure 6. The test machine has radial bearings in both drive and non-drive end. The levitation process went on really smooth. In the first try the rotor was levitated but with small vibration. By changing only one parameter (controller gain), the test rotor achieved stable levitation.

Figure 6. Photo of the test machine.

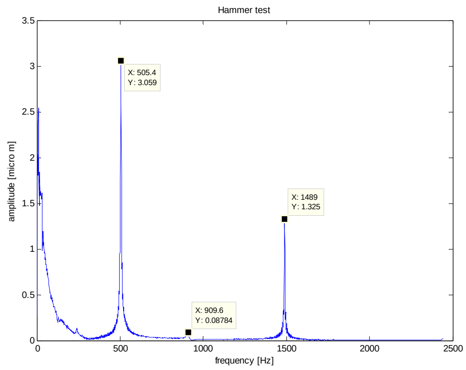

Figure 7. Hammer test result when the rotor was leviated.

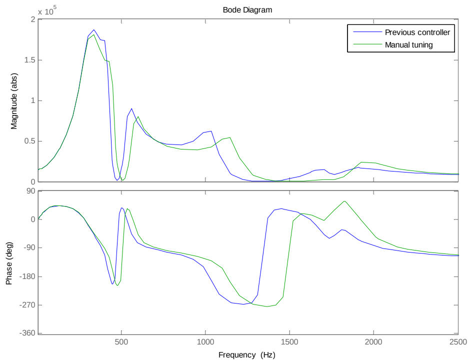

Figure 8. Manually tuned controller result.

After the levitation, a hammer test was carried out to verify the rotordynamic features. The hammer test result is given in Figure 7. The result shows that the first bending mode is 505Hz and the third bending mode is 1489Hz, while in the rotordynamic model they are 473Hz and 1339Hz respectively. With the hammer test result, the notch filter frequencies for the first and third bending modes have been changed manually, as shown in Figure 8.

Without any more parameter changing, the designed controllers levitated the rotor and rotated to the maximum continuous speed 200Hz(12000rpm). Figure 9 shows the maximum displacements of drive and non-drive end during the speed ramp up and down test. Figure 10 shows the rotor orbits measured by the position sensor at 200Hz in 0.5 seconds. The orbits in both ends are less than 3µm, which shows the levitation is in a really stable status.

V. CONCLUSION

An optimization based AMB controller design method has been described in this paper. The optimization criteria are chosen to describe AMB performances, and controller components are selected considering manually tuning on-site. A multi-objective genetic algorithm toolbox can be used to tune the parameters automatically. The design results have been verified in experiments. During the experimental verification, only 3 of the total 34 parameters have been changed and the machine rotates to the maximum continuous speed (12000rpm) with performances much better than the ISO standard requirement.

For a newly built AMB system, if the objectives and variables are set up properly, the AMB position controller design process can be done in half an hour. Even including the setup of the rotordynamic features and other system parameters such as actuators, sensors, and amplifiers, the AMB controller design process can be finished in several hours. The engineering cost for controller design can be reduced significantly.

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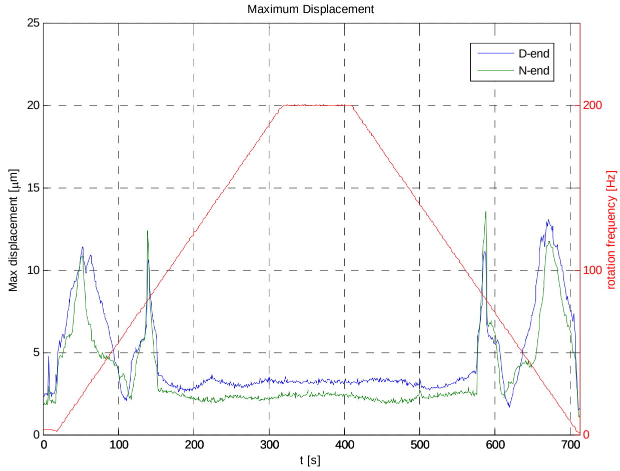

Figure 9. D-end and N-end maximum rotor displacement during a ramp up and down test from 0 to 200Hz.

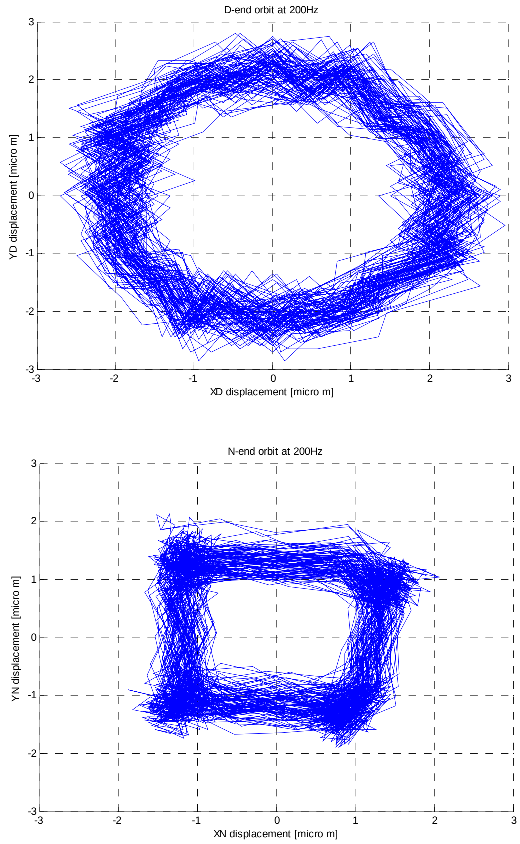

Figure 10. D-end and N-end rotor orbit at 200Hz.