한국기계가공학회지, 제12권, 제6호, pp.167-174 (2013.12) ISSN 1598-6721 (Print)
Journal of the Korean Society of Manufacturing Process Engineers, Vol.12 No.6, pp.167-174(2013.12) ISSN2288-0771(Online)

A Study on the Optimal Design for a Magnetic Bearing-Rotor with Maximum Stiffness using a Genetic Algorithm

Chae-Sil Kim[*#], Hoon-Hyung Jung[**], Bong-Kwan Park[**]

유전자 알고리즘을 이용한 최대 강성을 갖는 자기베어링-회전체 최적설계에 관한 연구

김재실[#*], 정훈형[**], 박봉관[**]
(*,#,**창원대학교)

(Received 3 December 2013; received in revised form 18 December 2013; accepted 19 December 2013)

ABSTRACT

High speed rotor systems with magnetic bearings have been the subject of much research in recent years due to the potential for active vibration control. In this thesis, optimal design was conducted for an 8-pole heteropolar magnetic bearing used in the flexible rotor of a turbo blower. In connection with bearing stiffness, this optimal design process was conducted using a genetic algorithm(GA), which is based on natural selection and genetics. The maximum stiffness of the magnetic bearing-rotor was found by considering the critical speeds of the flexible rotor. As a result, the magnetic bearings were optimized to have maximum stiffness.

Key Words : Turbo blower; genetic algorithm(GA); magnetic bearing; bearing stiffness

1. Introduction

Modern technology in rotating machinery trends towards high-speed, high-precision and high-efficiency. A turbo blower used to produce highly compressed air in industries is typical of such rotating machinery. It is essential to have lots of studies of the technology elements that are related to achieving high turbo blower efficiency.

The high speed rotor system with magnetic bearings has been the subject for numerous researches in recent years due to its potential for active vibration control. Also, while relatively expensive compared to other bearings, the magnetic bearing has a semi-permanent lifespan because it makes no physical contact, making lubrication unnecessary, and operates over a wide range of temperature and automatic balancing[1-4].

*# C. S. Kim : Department of Mechanical Engineering, Changwon National University
E-mail : kimcs@changwon.ac.kr
** Dept. of Mechanical Engineering, Graduate School, Changwon National University

Typically, the nonlinearity represents the material properties of the electromagnet. The error that occurs in the linearization will be compensated at the level of the designed controller[5].

So, most magnetic bearings have been modeled by linearized governing equations. Optimization may be required for designing the magnetic bearing due to its many design variables and constraints[6-7].

Optimal designs for magnetic bearing have been focused on maximizing magnetic force until now. Although optimization for maximizing the stiffness of magnetic bearing are needed to be overcome lack of stiffness, one of disadvantages of magnetic bearing, no research has been reported for the optimal design of the stiffness.

This paper derives equations for the stiffness of magnetic bearing and establishes the optimization problem with defining object function and constraints for maximize the stiffness and then finds the optimal variables using a genetic algorithm(GA).

2. Theory

2.1 Critical speed analysis

Figure 1 shows a magnetic bearing-rotor system that includes the disk. For this model, a critical speed analysis of the rotor system was performed to determine how bearing stiffness was affecting the high speed stability. A critical speed analysis are cited from the method published by Jo et al.[8].

Figure 2 is a graph of the critical speed analysis results, showing the relation between bearing stiffness and natural frequency. As shown in the graph, at the rated operating speed of 25,000 rpm, it can be seen that the first mode and second mode do not overlap bearing stiffness ranges. Based on the results from the rated operating speed of 25,000 rpm, the bearing stiffness can be designed to avoid the critical speed range. We select the last value of magnetic bearing stiffness at a rated operating speed of 25,000 rpm.

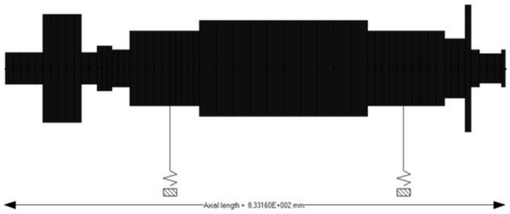


Fig. 1 A simple model for a magnetic bearing-rotor

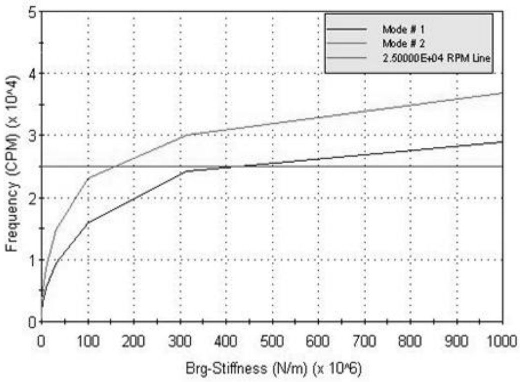


Fig. 2 Critical speed map

The procedure is explained in the next chapter.

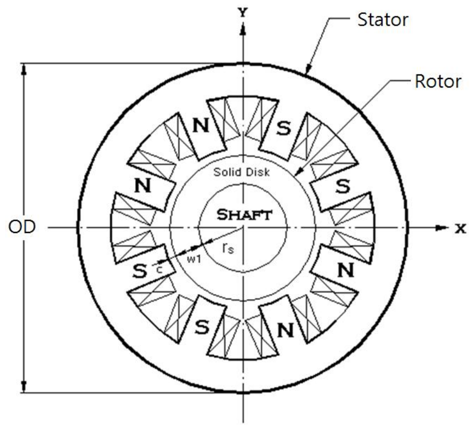
2.2 Equations for stiffness of magnetic bearing design

The basic equations (1) to (7) for designing a radial magnetic bearing are cited from the equations stated by Kim et al.[9].

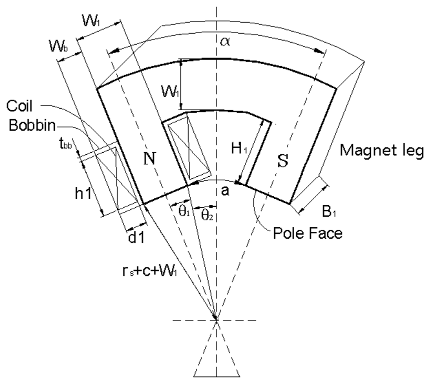
A typical 8-pole (N–S–S–N–N–S–S–N) heteropolar magnetic bearing, shown in Figure 3, is equipped for a shaft of radius, r_s. Design variables, which are the width of the pole leg, W_1, the height of the pole leg, H_1, and the axial length of the pole leg, B_1 are defined as numbers randomly generated within reasonable limits by applying GA.

The number of coil turns wrapped around each pole, N is defined as.

$$N = e_f \times nh_1 \times nd_1 \tag{1}$$



(a) 8 pole heteropolar magnetic bearing



(b) N-S pole

Fig. 3 Schematics of magnetic bearing

where the packing factor, e_f to be applied is 0.7. nh_1 and nd_1 which represent the number of coils along the height of the bobbin, and along the width of the bobbin, respectively, are calculated as

$$nh_1 = \frac{H_1 - 2\times t_{bb}}{2\times r_w} \quad (2)$$

$$nd_1 = \frac{1}{2\times r_w}\left\{(r_s + W_1 + c)\tan\left(\frac{\alpha}{2}\right) - t_{bb} - \frac{W_1}{2}\right\} \quad (3)$$

where the angle between two consecutive poles, α is 45°. r_w and t_{bb} represent the radius of wire with insulation and the thickness of the bobbin, respectively. The outer diameter of magnetic bearing, OD is defined as

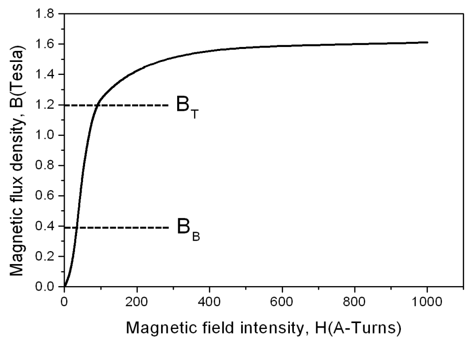


Fig. 4 B-H curve for silicon steel

$$OD = 2\times\{r_s + (2W_1) + c + H_1\} \quad (4)$$

The upper and lower bias current, I_{bT} and I_{bB}, are calculated from the magnetic flux densities at the B_T point and the B_B point in the region is assumed to be linear as shown in Figure 4.

$$I_{bT} = \frac{C_f c B_T}{\mu_0 N},\ I_{bB} = \frac{C_f c B_B}{\mu_0 N} \quad (5)$$

where c and C_f represent the nominal radial clearance and the correction factor (=1.2), respectively, and μ_0 is the magnetic permeability of free space.

Total rotor power loss in the magnetic bearing suggested by Kasarda[10], P_T occurs due to both winding effects and iron power losses which are expressed as

$$P_T = P_e + P_{ha} + P_{hr} + P_W \quad (6)$$

where P_e is eddy current power loss, P_{ha} is alternating hysteresis power loss per cycle for one rotor lamination stack, P_{hr} is rotational hysteresis loss, and P_W is the windage losses due to air friction.

The heat generated in the coil is the same as the heat produced by convection in the coil surface[11]. Therefore the maximum temperature in the coil, T_s

is

$$T_s = \frac{(I_{bT} + I_{bB})^2 R}{2hA_{coil}} + T_\infty \quad (7)$$

The where h is the convection factor of air. A_{coil} is the coil cross-section area, R is the coil resistance, and T_∞ is the temperature, in the atmosphere.

The position stiffness and the current stiffness equations (8) to (9) for a radial magnetic bearing are stated by Kelm et al.[12]. The K_m, position stiffness of the magnetic bearing is the change of the vertical force by small disturbance in the vertical displacement, and is defined as shown in equation (8).

$$K_m = -\frac{4\mu_0 A_p N^2 (\cos 22.5)^2 (I_{bT}^2 + I_{bB}^2)}{c^3 C_f^2} \quad (8)$$

The β, current stiffness, of the magnetic bearing is as shown in equation (9).

$$\beta = \frac{4\mu_0 A_{pole} N^2 (\cos 22.5)^2 (I_{bT} + I_{bB})}{c^2 C_f^2} \quad (9)$$

As shown in Fig. 5, the positions of the sensor and the magnetic bearing. are different. It should be compensated to exactly calculate the stiffness of a magnetic bearing. A proportional-derivative controller is used to get the good performance. The magnetic bearing stiffness, considering the position difference and controller's gains, K_{Bri} is calculated as:

$$K_{Bri} = \frac{F}{X} = \beta \cdot K_{pa} \cdot K_p (1 + \tau_d) \xi \cdot \left(\frac{l_2}{l_1}\right) \quad (10)$$

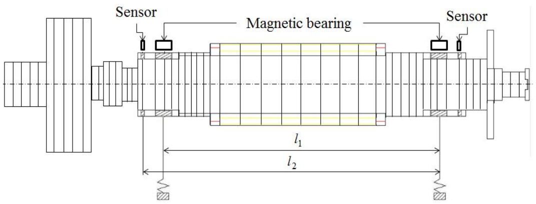


Fig. 5 A magnetic bearing-rotor system

where,

K_{pa} : gain of the power amplifier
K_a : proportional gain of the controller
τ_d : time-constant of derivative gain
ξ : sensitivity of sensor
l_1 : distance between centers of bearings
l_2 : distance between sensor and the center of the back bearing

Each controller has applied the values provided by the manufacturer, $K_p = 1.2$, $K_{pa} = 1$, $\tau_d = 0.0012$, $\xi = 10000\, V/m$ and we calculated the magnetic bearings stiffness to levitate the rotor. The distance ratio between the sensor and the magnetic bearings is approximate 1.045. By adding position stiffness K_m using linear magnetic field analysis, magnetic bearing stiffness values are derived as Eq. (11)

$$K_{mag} = K_{Bri} + K_m \quad (11)$$

3. Optimal design for the magnetic bearing using a GA

3.1 Optimization problem establishment

The optimization problem established in this study is to find the design variables which maximize the magnetic bearing-rotor stiffness, K_{mag}, while satisfying the performance limits and the geometric constraints.

The constraint conditions are to: (a) restrict the outer diameter of the magnetic bearing to 0.214 m considering the total outer diameter of the high speed machine tools; (b) restrict the energy loss to 500 W at maximum rotating speed, 25,000 rpm; (c) limit the maximum temperature in the coil to 85℃ to avoid damaging the film of the coil; (d) wrap the wire over 10 floors at the bottom of the bobbin so that one pole does not interfere with the adjacent pole and the number of coil turns is sufficiently assured.

The optimization problem is summarized as:

$$Maximize \quad K_{mag} \tag{11}$$

$$OD \leq 0.214\,[m],\ P_T \leq 500\,[W],$$

$$T_s \leq 85\,[^\circ C],\ nd_1 \geq 10 \tag{12}$$

3.2 Design variables

Three design variables are selected with geometric variables including the axial length of the magnetic bearing, B_1, the height of the pole leg, H_1, and the width of the pole leg, W_1. In an optimal design by GA, each design variable is given by a set of selectable values and then each value is decoded to a binary code. It is, therefore, necessary to determine the selectable values in the range of design for each variable. The axial length of pole may be determined to be about 16.0 - 28.0 mm due to the restriction of the installing space considering the axial lengths of both shaft and built-in motor. The height of the pole may be about 20.0 - 32.0 mm considering the limit of the outside diameter of the bearing. The width of the pole may be given to be around 14.0 - 26.0 mm because of the geometric constraint for the bearing.

The design variables in this investigation are summarized as:

$$B_1: 0.016 \leq B_1 \ \leq 0.028\,[mm]$$

$$H_1: 0.020 \leq H_1 \ \leq 0.032\,[mm] \tag{13}$$

$$W_1: 0.014 \leq W_1 \leq 0.026\,[mm]$$

Since the limits of the design variables have been determined, the chromosome lengths having information for the design variables should be determined as a number of bits. The chromosome lengths of five design variables, i.e. B_1, H_1 and W_1 are assigned in 7, 7, 7 bits, respectively. So the total length of a chromosome is 21 bits.

3.3 Fitness evaluation

The optimization problem established in this article as equation (12) is a typical constrained optimization problem. Since genetic operators used to manipulate the chromosomes often yield infeasible offspring, the major concern for applying GA to the constrained optimization is how to handle constraints.

The penalty technique is often used to handle infeasible solutions. In essence, this technique transforms the constrained problem into an unconstrained problem by penalizing infeasible solutions by constructing an evaluation function called the fitness, which is the objective function added to a penalty term for any violation of the constraints.

Therefore the optimization problem is transformed to an unconstrained optimization problem by defining the fitness as:

$$Fitness = K_{mag} - P_1 - P_2 - P_3 - P_4 \tag{14}$$

$$P_1 = \left(\frac{OD - 0.214}{0.214}\right) \times pratio_1\ ,\ OD > 0.214 \tag{15}$$

$$P_2 = \left(\frac{P_T - 500}{500}\right) \times pratio_2\ ,\ P_T > 500 \tag{16}$$

$$P_3 = \left(\frac{T_s - 85}{85}\right) \times pratio_3\ ,\ T_s > 85 \tag{17}$$

$$P_4 = \left(\frac{9 - nd_1}{9}\right) \times pratio_4\ ,\ nd_1 < 10 \tag{18}$$

where K_{mag} is the object function and P_i is the penalty term for ith constraint. The $pratio_i$ is a non-dimensional weighting factor added to handle infeasible solutions. In this article, $pratio_i$ in all of the constraints are identical and variable $pratio_i$ are applied as generations proceed: 20% for 1 through 15 generation, 50% for 16 through 30 generation, 70% for 31 through 45 generation, and 80% over 46 generation.

3.4 Optimal design procedure

The optimal design procedure of the magnetic bearing with GA is shown in the Figure 6. Optimal values are obtained when the fitness values are saturated, i.e. converged through iteration with zero penalty value.

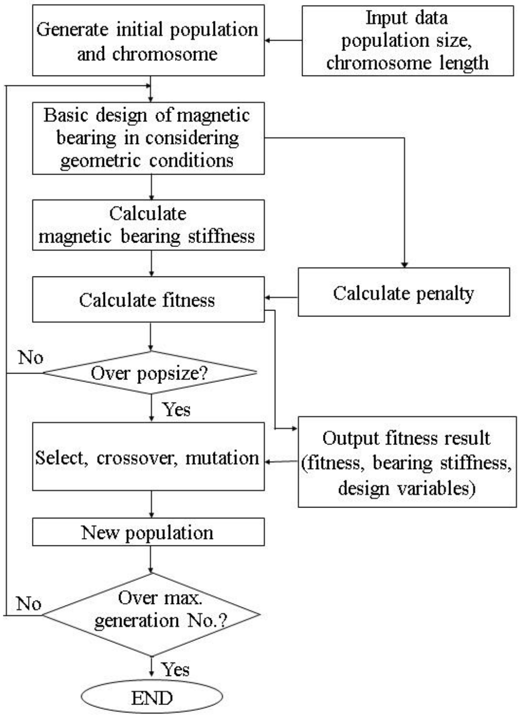


Fig. 6 Flowchart for optimal design of the magnetic bearing with GA

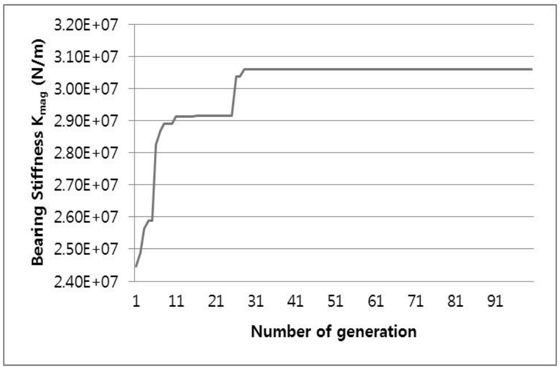
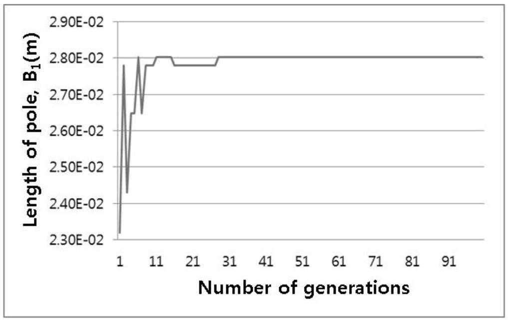


Fig. 7 Bearing stiffness in each generation

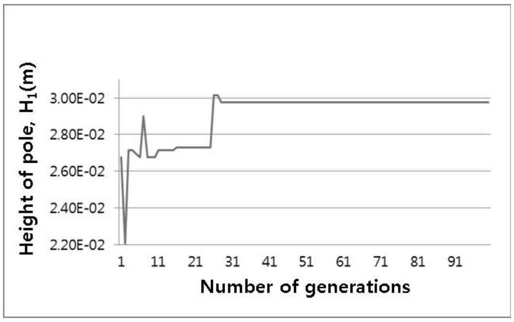
4. Results and review

4.1 Optimization of bearing stiffness

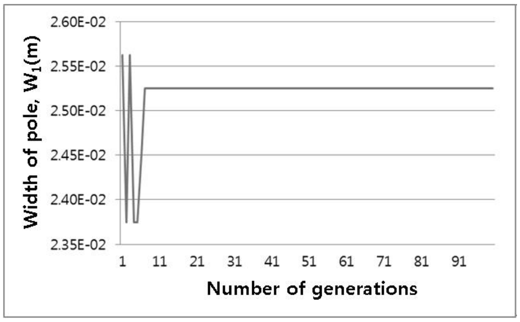
Figure 7 shows the optimization results using GA, which represent the maximum bearing stiffness. When



(a) Length of pole



(b) Height of pole



(c) Width of pole

Fig. 8 Optimal design value in each generation

Table 1 Optimal configurations

Design Variables	B_1	H_1	W_1
Optimal design values	0.0280 m	0.0298 m	0.0253 m
Max. bearing stiffness	30.6 MN/m		

the generation is increased, the bearing stiffness is increased in the direction of evolution. The optimal bearing stiffness by linear analysis, 30.6 MN/m, is obtained. The design variables search globally in the infeasible area as well as the feasible area.

4.2 Optimal configuration for a magnetic bearing-rotor system

As the generations increase, the design values are changed to the fitness. Figure 8 shows that the given three design variables have good convergences for linear optimization. In Figure 8, the axial length of the pole leg, B_1, is converged to the maximum value in its design limits because of enlarging the area of the pole to obtain more magnetic force. The height of the pole leg, H_1, has a tendency to converge to the value in the design limit for satisfying the constraint condition of the outer diameter. The width of the pole leg, W_1, is converged to the value of the design interval due to the restrictions of outer diameter and the width of bobbin.

Table 1 shows the value of the optimal design variables at the maximum stiffness which can be recognized as the optimal design. When the stiffness converges to the maximum, i.e., the optimization is completed.

5. Conclusion

An optimization problem for a general 8-pole magnetic bearing was established for maximizing its stiffness. A genetic algorithm was applied to the optimal design. The designed stiffness of the magnetic bearing was investigated to avoid critical speeds of flexible rotor. The convergence of fitness and no match of critical speeds might show that the magnetic bearing was well optimized with its maximum stiffness.

Acknowledge

Following are results of a study on the "Leaders INdustry-university Cooperation" Project, supported by the Ministry of Education(MOE).

REFERENCES

1. Bleuler, H., “A Survey of Magnetic Levitation and Magnetic Bearing Types”, JSME International Journal Series Ⅲ, Vol. 35 No. 3, pp. 335-342. 1992.
2. Ha Youngho, Lee Jongwon, "Intelligent type magnetic bearing", Journal of KSME, 35(10): 742-755, 1994.
3. Brunt M., "Practical applications of the active magnetic bearings to the industrial world", Proc. of the 1st Int. Symp. on Magnetic Bearings, pp. 225-244, 1988.
4. Dussaux M., "Industrial applications of the active magnetic bearings technology", Proc. of the 2nd Int. Symp. on Magnetic Bearings, pp. 33-38, 1990.
5. Kim, C. S., "Magnetic bearing eddy current effects on rotordynamic system response", Doctor of Philosophy Thesis, Mechanical Engineering, Texas A&M University, 1995.
6. Pang D., Krik, An J. A., D. K., Huang C., "Design optimization for magnetic bearing", Proc. of the 26th IECEC, USA, 35(4), pp 186-191, 1991.
7. Imlach J., Allaire P. E., Humphris R. R., Barett L. E., "Magnetic bearing design optimization", IMechE Journals, pp. 53-59, 1988.

8. Jo H. M., Jung H. H., Kim C. S., Cho S. Y., "A Review on the Design of a Tilting Pad Journal Bearing for a 1.2MW Turbo-blower", Journal of KSMPE, Vol. 10, No. 2, pp. 26-31, 2011.
9. Kim C. S., Lee J. W., Park J. K., "Optimal design for a heteropolar magnetic bearing considering nonlinearities", International Journal of KSPE, 3(1), pp. 13-19, 2002.
10. Mary E. F., Kasarda., "The measurement and characterization of power losses in high speed magnetic bearing", Ph. D. Thesis, Mechanical and Aerospace Engineering, Virginia University, pp. 21-26, 1997.
11. Shankar Jagannathan, "Analysis and Design of Magnetic Bearing," Master Thesis, Mechanical Engineering, Virginia University, pp. 27-30, 1991.
12. Kelm R. D., Humphris R. R., Allaire P. E., "Analysis and testing of a magnetic bearing for flexible rotors", Dep. of Mechanical and Aerospace Engineering, University of Virginia, Report No. UVA/643092/MAE86/344, 1986.
13. Jung H. H., Jo H. M., Kim C. S., Cho S. Y., “Multi-Body Dynamic Modeling for a Flexible Rotor and Vibration Control using a Novel Phase Adjusting Technique”, Journal of KSMPE, Vol. 10, No. 1, pp. 87-92, 2011.