DESIGN OF A RADIAL ACTIVE MAGNETIC BEARING

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***Abstract* – A design procedure of a radial active magnetic bearing is discussed in the paper. The initial design is improved by a numerical optimization, using the differential evolution strategy in combination with the 2D finite element method. The objective function is found empirically as a bearing mass-force ratio. The force is calculated by the Maxwell's stress tensor. Furthermore, a data fitting is performed in order to confirm the calculated maximal bearing force trough the comparison with the measured force.**

I. INTRODUCTION

Active Magnetic Bearings (AMB's) represent a highly non-linear and coupled electro-magneto-mechanical system. Two radial bearings and one axial bearing are used to control five degrees of freedom of a rotor [1]. In this way a contact-less suspension, i.e. a levitation of the rotor is achieved. Higher speed, no friction, no lubrication, precise position control and active vibration damping make AMB's particularly appropriate for applications in the machine–tool industry.

The design of AMB's is expected to satisfy demanding requirements in the best possible way. To consider the influence of non-linear iron properties, local saturation and magnetic flux leakage, Finite Element Method (FEM) based calculations are very helpful. The design can be found either by experiences and trials or by applying numerical optimization methods. Because of the unknown dependency of the objective function and its gradients from optimization parameters, stochastic search methods are recommendable – particularly the Differential Evolution (DE) strategy, described in [2].

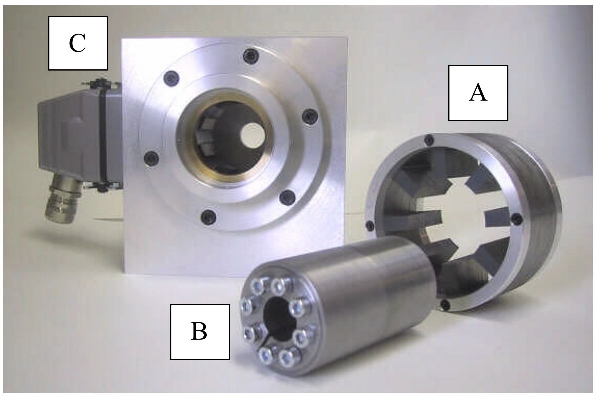

Fig. 1. Radial AMB: A – stator, B – rotor, C – housing

In this paper a design procedure for an eight-pole radial AMB is presented. The initial design is found by an analytical approach. Next, the numerical optimization of the bearing geometry using DE – a direct stochastic search algorithm is applied. An optimization task is to find such a design, that the maximal bearing force is reached along with the minimum possible mass of the entire construction. The objective function is evaluated by FEM-based 2D calculations applying the Maxwell's stress tensor. The optimization procedure proposed in [3] is used. It is performed in a special environment tuned for FEM-based numerical optimizations, described in [4]. Obtained parameters are compared for the initial and the optimized design. Furthermore, the calculated maximal bearing force is confirmed by the comparison of the measured maximal force of the radial AMB shown in Figure 1. The data fitting is considered due to manufacturing problems as well as due to the magnetic air gap enlargement according to [5]. The agreement between calculated and measured maximal bearing force is quite good.

II. BEARING FORCE

The stator of the radial AMB is treated as four "horse–shoe" electromagnets, as shown in Figure 2. Windings of all electromagnets are assumed to be ideal and identical. The rotation of the rotor and the non-linear iron properties are not considered. The resultant electromagnetic force of two diametrical electromagnets F is described by Equation (1). The bias current I_0 is operating the winding of both electromagnets, while the force control is performed by superposing a control current $i_{y\Delta}$ to the winding of one electromagnet and subtracting it in the winding of other one, where $i_{y\Delta} \leq I_0$. The rotor position is denoted as y, while δ_0 denotes the nominal air gap. N is the total number of turns for one electromagnet, A is the area of one pole, while υ and μ_0 denote the leakage factor and the permeability of vacuum respectively. Geometric factor of an eight-pole radial bearing is considered by an angle of $\pi/8$.

$$F = \tfrac{1}{4}\upsilon\mu_0 N^2 A\cos(\pi/8)\left(\left(\frac{I_0 + i_{y\Delta}}{\delta_0 - y}\right)^2 - \left(\frac{I_0 - i_{y\Delta}}{\delta_0 + y}\right)^2\right) \quad (1)$$

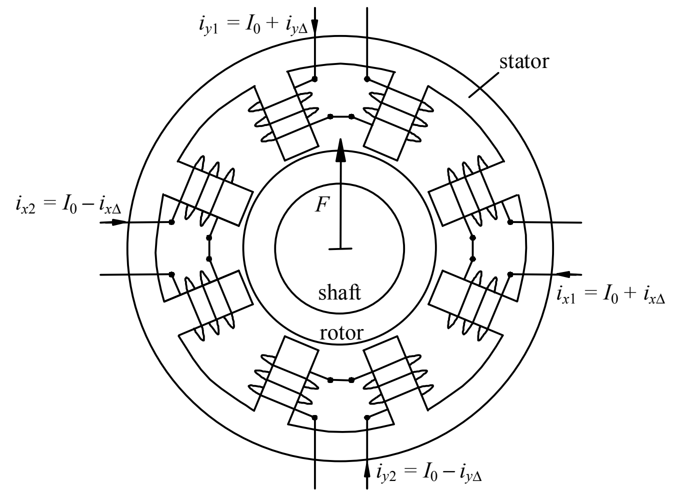

Fig. 2. The bearing geometry and winding connections

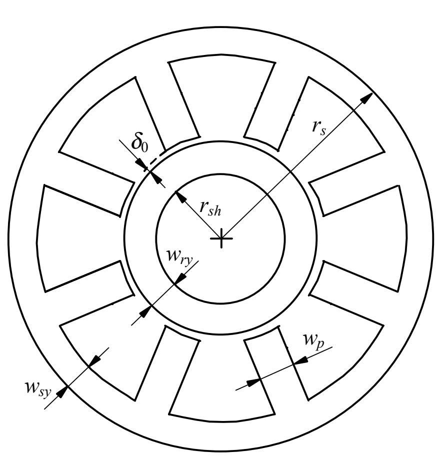

Fig. 3. Design constrains and optimization parameters

III. DESIGN

A. Initial Design

The design of the radial AMB should satisfy required performances and design constrains shown in Table I and in Figure 3. It is found by an analytical approach employing Equation (1), and several empirically determined geometrical parameters. The analytically obtained result are used as initial design for the numerical optimization of the bearing geometry.

TABLE I
REQUIRED PERFORMANCES AND DESIGN CONSTRAINS

parameter	value
maximal bearing force F_{max} [N]	580
rms load force F_{rms} [N]	200
nominal air gap δ_0 [mm]	0.4
stator radius r_s [mm]	52.5
shaft radius r_{sh} [mm]	17.5
bias current I_0 [A]	5

B. Optimization

The optimization of the radial AMB is carried out in a special environment tuned for FEM-based numerical optimizations, decribed in [4]. The procedure proposed in [3] is described in the following six steps:

- **Step 1:** The geometry of the bearing is described parametrically and the initial parameter values are determined by an initial design. Optimization parameters are: the stator yoke width w_{sy}, the rotor yoke width w_{ry}, the pole width w_p (all shown in Figure 3) and the bearing axial length l.

- **Step 2:** The new parameter values are determined by the DE, described in [2]. The rotor position is placed in the center ($x = 0$, $y = 0$). The electromagnets in the x axis are supplied by the bias current I_0, while the electromagnets in the y axis are supplied in such a way that the maximal force is reached, i.e. by currents $i_{y1} = 2I_0$ and $i_{y2} = 0$.

- **Step 3:** The bearing geometry, the material, the current densities and the boundary conditions are defined. The procedure continues with **Step 2** if the parameters of the bearing are outside the design constraints.

- **Step 4:** First, the mesh is generated. Then, the non-linear solution of the magnetic vector potential A is obtained by means of 2D computations based on the FEM. The problem is formulated by Poisson's Equation (2), where ν represents the magnetic reluctance, J is the applied current density, and ∇ is Hamilton's differential operator (Nabla operator). Furthermore, the analysis of errors, mesh refinement, and the repeated potential calculation are performed.

$$\nabla \cdot (\nu \nabla \mathrm{A}) = -J \qquad (2)$$

- **Step 5:** The force is determined by the Maxwell's stress tensor, following Equation (3). B_y and B_x are the normal and the tangential component of the air gap flux density, μ_0 is the permeability of vacuum, and l is the axial bearing length. The integration is performed over the contour C placed along the middle layer of the five-layer air gap mesh.

$$F = \frac{l}{2\mu_0} \oint_C \left(B_y^2 - B_x^2\right) ds \qquad (3)$$

- **Step 6:** The objective function (4) and penalties (5) are found empirically. m and F are the mass and the force for actual parameter values, while m_0 and F_0 are its initial values. The value of the objective function is determined from **Step 3** through **Step 5**. It is minimized from **Step 2** through **Step 6** until a minimal optimization parameter variation step or a maximal number of evolutionary iterations are reached.

$$q = \frac{m}{F}\frac{F_0}{m_0} + p_1 + p_2 \qquad (4)$$

$$p_1 = \frac{F_0}{F} \quad \text{if} \quad F < F_0$$
$$p_2 = \frac{m}{m_0} \quad \text{if} \quad m > m_0 \qquad (5)$$

C. Results

The optimization data are shown in Table II. The results, given in Table III, show that the optimization has increased the maximal force of the radial AMB for more than 8.6%, while the mass has remained unchanged.

TABLE II
THE OPTIMIZATION DATA

parameter	value
number of optimization parameters	4
population size	20
reached number of iterations	60

TABLE III
OPTIMIZATION PARAMETERS AND OBJECTIVE FUNCTION

parameter	initial value	optimal value
stator yoke width w_{sy} [mm]	8.5	7.2
rotor yoke width w_{ry} [mm]	9.0	7.8
pole width w_p [mm]	10.0	9.0
axial length l [mm]	53.0	56.3
mass m [kg]	2.691	2.688
maximal force F [N]	580	630
objective function q	1	0.92

IV. ACTUAL MAXIMAL BEARING FORCE

In the manufacturing process several changes appeared:

- Slightly exchanged dimensions due to the laser cut, as shown in Figure 4.
- Additional holes drilled into the stator package, as shown in Figure 4 – screws are used to fix firmly the stator package.
- Reduction of the bearing axial length to 40.0mm due to the redesign of requirements.
- Choice of a thinner lamination in order to reduce the eddy current losses in the rotor – the material M36 is used instead of the material M43 (AISI standard).

Therefore, a much lower maximal force of the radial AMB is expected.

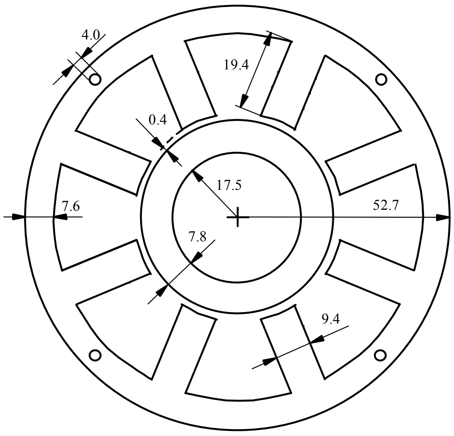

Fig. 4. An actual bearing geometry [mm]

B. Force Measurement

The measurement setup is shown in Figure 5. The maximal bearing force is measured via a handle–lever that connects the shaft and the force sensor – a HBM Z6FC3 bending beam is used. The rotor position is measured with an induction proximity probe Vibro-meter TQ401, while a LEM HY 10-P sensor provides the current measurement. All four electromagnets are supplied by a 20kHz, 60V PWM current source inverter. For the real time feedback algorithm execution, and data acquisition a dSPACE 1103 board is used.

The bearing force is quite difficult to measure due to the inherent instability of the rotor position. Therefore, two independent PID controllers were implemented to set the rotor center position ($x = 0$ and $y = 0$). Furthermore, an additional external force being measured was applied to the force sensor, while the control current $i_{y\Delta}$ required to stabilize the rotor position was measured. The maximal bearing force was reached when the maximal value of the control current $i_{y\Delta} = I_0$ was measured, i.e. $i_{y1} = 2I_0$ and $i_{y2} = 0$. The gravity force was compensated afterwards.

C. Results

The bearing axial length was reduced for almost 30% due to the redesign of requirements. In order to compare measurements and calculations the calculated bearing maximal force should be reduced linearly, since a 2D FEM-based calculation was used. Furthermore, the data fitting of the calculated maximal bearing force was considered. According to [5] and our own experiences a fictive air gap enlargement of approximately 0.05mm is expected due to the manufacturing procedure of the rotor. The comparison of calculated and measured maximal bearing force is given in Table IV. The agreement between calculated and measured maximal bearing force is quite good.

TABLE IV
CALCULATED AND MEASURED MAXIMAL BEARING FORCE

maximal bearing force	value [N]
calculated value for reduced length	447
calculated value for reduced length and fitted air gap	353
measured value	340

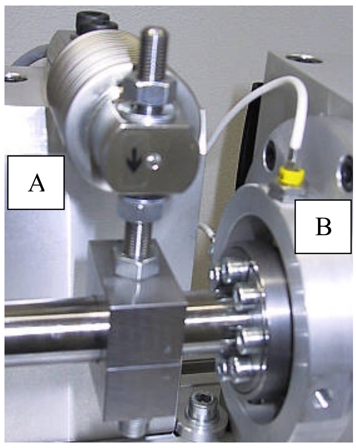

Fig. 5. Measuremet setup: A – beanding beam, B – proximity probe

V. CONCLUSION

The paper describes a design procedure for an eight-pole radial AMB. It has been shown that the use of the DE-based optimization in combination with the FEM can increase the maximal bearing force at an unchanged mass. The maximal force of an actual radial AMB is rapidly reduced due to several manufacturing changes and redesign of requirements. Furthermore, the calculated maximal bearing force has been confirmed trough the data fitting and measurement. The agreement between calculated and measured force is quite good. In the next step of this development, FEM-based force calculations considering the data fitting will be performed in the entire operating range along with force measurements.

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