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Research article

Study on PID tuning strategy based on dynamic stiffness for radial active magnetic bearing

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ABSTRACT

The development of industry technology requires magnetic bearings to work in high speed conditions. However, the current stiffness and displacement stiffness of the magnetic bearing will decrease significantly due to the consequent eddy current effect, and that decrease will make the system unstable and even result in the rotor drop and instrument damage. Therefore, the traditional Proportional-Integral-Derivative (PID) method based on constant stiffness is not adaptable for high speed conditions. This paper proposes a PID parameters tuning strategy based on dynamic stiffness for the radial active magnetic bearing (RAMB). The dynamic stiffness model under eddy current effect is established by analyzing the equivalent magnetic circuit model in which parameters are frequency-dependent. The PID parameters tuning method for RAMB control system including dynamic stiffness model is put forward according to the characteristic equation and Routh-Hurwitz criterion. Different PID parameters are set in simulations and several corresponding experiments are conducted. Satisfactory control effects consistent with the theoretical analysis are obtained and thus the proposed PID tuning strategy is verified to be good. Simulations and experiments in this paper provide theoretical guidance for the design of controller parameters and have research significance for structural optimization of RAMB.

1. Introduction

Radial active magnetic bearings (RAMBs), composed of the electromagnetic actuator, position sensors, controllers and power amplifiers, have better controllability and damping characteristic as compared to passive magnetic bearings [1–3]. RAMBs are widely used in many industrial occasions, including flywheels, turbines, air compressors, molecular pumps, generators, and bearingless motors [4–6], for their obvious advantages of having no mechanical friction, wear and lubrication, but long life and high reliability.

PID algorithm is a simple, adaptable and mature control method. It has been widely used in the RAMB system and is usually used as the main controller of the system [7,8]. The challenging task is how to tune PID controller parameters to achieve high-frequency requirement of the rotor. For RAMB system, PID controller parameters tuning mainly relies on experience. Psonis et al. [9] studied the influence of PID controller parameters on system stability based on the linearization model of RAMB system, and the boundary conditions for the stability of system were found and presented according to the characteristic equation and Routh-Hurwitz criterion. Procedures of manual tuning were proposed in Ref. [10] which have been widely used in PID control research. The main proposed process is to overcome negative stiffness first by adjusting the value of K_p only, then add damping K_d into the system to reduce the rotor vibration, and finally increase the integral gain K_i gradually to eliminate the steady state error. This type of tuning requires the operation of a system and is thus sometimes called on-line tuning [11].

However, current researches for PID controller design consider the stiffness of RAMB as a constant, unaffected by rotational speed [12–15]. It is just applicable for the situation that rotor frequency is low and the changes of stiffness are unobvious. With the increase of rotor frequency, the stiffness of RAMB will decrease significantly, which will directly reduce the capacity of RAMB system, and even lead to instability when the impact is serious [16,17]. To ensure that the magnetic bearing system works stably with high rotational speed, the impact of rotor speed on RAMB stiffness cannot be ignored. Furthermore, PID controller parameters are difficult to set online in the condition of high rotation frequency and the decreased RAMB stiffness will also increase the difficulty of parameter tuning. In order to enhance system performance, improve experiment efficiency and reduce debugging risk, it is necessary to build an accurate control system model and establish a theoretical standard for PID design based on dynamic stiffness varying with rotor speed.

The induced eddy currents in high speed rotor can cause the

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amplitude decrease and the phase lag of the stiffness, which affects the dynamic performance and stability of the entire RAMB system [18–20]. For the solid core material of magnetic bearings, Sun Yanhua et al. [21] established an eddy current loss analysis model of radial magnetic bearing, then analyzed the relationship between dynamic stiffness of magnetic bearing and frequency. The conclusions were that the eddy current in the high speed rotor affected the magnetic field distribution in the stator poles, thus decreased the magnetic bearing stiffness. Though the solid structure was commonly replaced by laminated cores to reduce the eddy current loss, the magnetic bearings stiffness was still influenced significantly by eddy current when the rotation frequency was high enough [22–24]. In Ref. [25], the radial differential magnetic bearing with laminated structure was taken as the research object, and the influence of eddy current on the dynamic stiffness of magnetic bearing was studied by establishing the dynamic equivalent permeability model of laminated structure. The structural optimization scheme were then proposed and finally the recommendations to reduce the dynamic stiffness variation were provided. However, as an important factor, the dynamic stiffness is still not considered in the control system until now.

This paper proposes a PID parameter tuning strategy for RAMB system based on dynamic stiffness to improve the control performance of traditional PID under high speed conditions. The organization of the paper is as follows. Section 2 analyzes the dynamic equivalent magnetic circuit of RAMB, and the dynamic stiffness model is established considering the eddy current effect. In section 3, the dynamic stiffness is introduced into the transfer function of RAMB system, and the PID tuning strategy is put forward according to Routh-Hurwitz criterion. Section 4 presents the simulation of the whole PID control system for RAMB based on the proposed tuning strategy. The experiments of the RAMB in a molecular pump using proposed tuning strategy is performed and the experimental effects are compared with the simulation results in Section 5. Finally, the conclusion is presented in Section 6.

2. Model of dynamic stiffness for RAMB

Take the typical structure of RAMB for example, which is pictured in the Fig. 1 and the dimensions of this bearing are described in Table 1, an accurate dynamic stiffness model including eddy-current effects is established in this section.

The influences of eddy current in high speed occasions are from two aspects. The one is that the eddy currents caused by the time-varying control currents will affect the magnetic resistance of both rotor and stator. The other is that the eddy currents deduced due to the change of magnetic field on rotor surface will also affect the magnetic resistance of rotor. The following three assumptions are made in this paper for analyzing convenient:

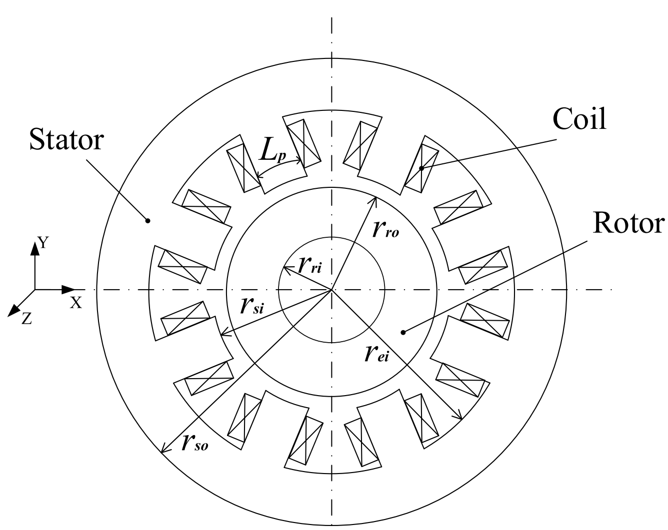

Fig. 1. Structure of the RAMB

Table 1
Main parameters of the RAMB.

Parameter	Value	Parameter	Value
Stator pole axial length (l_a)	13.5 mm	Pole arc length (l_p)	5.248 mm
Stator outer radius (r_{so})	36 mm	Number of turns (N)	300 turns/pole
Stator yoke inner radius (r_{ei})	32 mm	Air gap length (g)	0.3 mm
Stator inner radius (r_{si})	17.3 mm	Laminated thickness (d)	0.35 mm
Rotor outer radius (r_{ro})	17 mm	Conductivity (σ)	7.46×10^6 S/m
Rotor inner radius (r_{ri})	13 mm	Relative permeability (μ_r)	5000

(a) Suppose the BH characteristics of laminated material is linear and the effects of saturation and hysteresis can be ignored.
(b) Rotor laminations can be “unrolled” into periodic flat plates.
(c) No flux across the laminated insulating layer and the magnetic flux is the spatial function varying with time in Z direction.

The connection of relative permeability and rotation frequency is determined by the Maxwell equation and boundary conditions based on the above three assumptions. With the control current frequency increasing, eddy currents inhibit flux toward the center of the lamination, which resulting in less total flux. The relative permeabilities of stator and rotor considering eddy-current effects are deduced respectively in Ref. [18]. The stator's relative permeability has been analyzed by

$$\mu_{rs} = \mu_r \frac{\tanh\left(\sqrt{j\omega\sigma\mu_0\mu_r}\,\frac{d}{2}\right)}{\sqrt{j\omega\sigma\mu_0\mu_r}\,\frac{d}{2}} \tag{1}$$

where μ_r is the static relative magnetic permeability, ω is the rotation frequency, 2σ is the lamination resistivity, μ_0 is the magnetic permeability of vacuum, and d is the lamination thickness. The relative permeability for rotor is determined by Fourier transform and the *n-th* relative permeability [18] can be expressed by

$$\mu_{nr} = \mu_r \frac{\tanh\left(\sqrt{j\omega\sigma\mu_0\mu_r k_n r_{ro} + j\omega\sigma\mu_0\mu_r}\,\frac{d}{2}\right)}{\sqrt{j\omega\sigma\mu_0\mu_r k_n r_{ro} + j\omega\sigma\mu_0\mu_r}\,\frac{d}{2}} \tag{2}$$

where k_n is a coefficient related to the stator structure.

The dynamic equivalent magnetic circuit of the RAMB which is pictured in Fig. 1 with a relative permeability considering rotor frequency is shown in Fig. 2. In the circuit model, R_{x1+}, R_{x2+}, R_{x1-}, R_{x2-}, R_{y1+}, R_{y2+}, R_{y1-} and R_{y2-} are the magnetic resistances of air gap in different directions, R_{rn} is the dynamic resistance of the *n-th* harmonic of rotor, R_{s1} and R_{s2} are equivalent magnetic resistances of stator.

When the rotor is in the center of RAMB, according to the distribution of magnetic flux density, the resistances of air gap are expressed as

$$R_{x1+} = R_{x2+} = R_{x1-} = R_{x2-} = R_{y1+} = R_{y2+} = R_{y1-} = R_{y2-} = R_g = \frac{g}{\mu_0 l_a l_p} \tag{3}$$

R_{s1} and R_{s2} will be derived based on (1) and the results are respectively given by

$$R_{s1} = \frac{\pi(r_{so} + r_{ei})\sqrt{i\omega\sigma\mu_0\mu_r}\,\frac{d}{2}}{8\mu_0\mu_r l_a (r_{so} - r_{ei})\tanh\left(\frac{d}{2}\sqrt{i\omega\sigma\mu_0\mu_r}\right)} \tag{4}$$