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A Convenient Vibration Compensation and Control Method of Bearingless Induction Motor

Wenshao Bu[1], Ziyuan Li[2], Chunxiao Lu[1], Xiaohong Wang[1] and Juanya Xiao[1]

[1]*College of Information Engineering,*
Henan University of Science and Technology,
[2]*College of Electrical Engineering,*
Henan University of Science and Technology,
Luoyang 471023, China
wsbu@163.com

Abstract

To achieve the rotor's high precision suspension control of bearingless induction motor, a kind of convenient vibration compensation and control method is presented. The inverse model of magnetic suspension system is established, and by inverse system method, the decoupling control between two radial displacement components are achieved firstly; after then, through the compensation of unbalanced vibration control current, the compensation control force of unbalanced vibration that is used to counteract the unbalanced exciting force of bearingless rotor, is generated. Simulation results have shown that when adopting the proposed control strategy, the unbalanced vibration of bearingless rotor can be inhibited effectively either in the start stage or in the steady operation state.

Keywords*: Bearingless Induction Motor, Unbalanced Rotor, Unbalanced Vibration, Inverse System, Compensation Current*

1. Introduction

Bearingless motor is proposed based on the similarity in structure between magnetic bearing and usual motor's stator. In bearingless motor, a suspension windings is embedded in the stator slots along with the motor windings, the intervention of suspension control magnetic field breaks the balance distribution of conventional motor magnetic field, thus a resultant radial electromagnetic force that points to the direction of magnetic field enhancement is generated, witch can be used to achieve the suspension control of rotor [1-4]. Compared with AC motor supported by magnetic bearing, bearingless motor not only owns the advantages of no friction, no lubrication and no mechanical noise, but also owns some special characteristics, such as shorter shaft, higher critical speed, lower magnetic suspension cost, *etc.* For bearingless motor is a newly type motor that is fit for high speed operation, urgent requirements for the control technology of bearingless motor have been put forwarded in some application field, such as aerospace, sealed transmission of chemical materials, and advanced manufacturing, *etc.* and then it has a broad application prospect[2]. Bearingless control technology can be used to all kinds of AC motors; among them, bearingless induction motor owns the merits of robust structure, larger ratio of force to current, etc, then it has become a research hotspot [5-9].
For a rotary drive motor, the mass eccentricity of rotor is unavoidable that is caused by uneven material, machining accuracy, assembly error, *etc.*, which would generate centrifugal force that has the same frequency with the angular speed of shaft, and would lead to the unbalanced vibration of rotor. Then the suspension control precision of bearingless rotor would be effected inevitably and even lead to the collision accident between stator and rotor cores [10]. Therefore, the unbalanced vibration of bearingless

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motor is one of the key problems that should be solved in its practical application [10-11].

About the unbalanced vibration control of magnetic suspension rotor, there have been relevant researches. But the existing references are mainly concentrated in the magnetic bearing motor [9, 10], researches on unbalanced vibration control of bearingless motor, are still in its infancy. To simplify their control system structure, the existing references usually adopt the static decoupling control strategy which based on classic field-oriented method [11-12]. Although there are many literatures about the dynamic decoupling control strategy of bearingless motor [3-8], the unbalanced vibration problem caused by rotor mass eccentricity are rarely taken into account. For the structure complexity of bearingless motor, and for the indefinability of rotor mass eccentric direction and eccentricity, the high-performance control of rotor unbalanced vibration is very difficult.
In this paper, aiming at above problems, the unbalanced vibration control strategy of bearingless induction motor based on inverse system decoupling control of magnetic suspension system is studied. Simulation results show that when adopting proposed control method, the unbalanced vibration of bearingless rotor can be effectively suppressed, and the suspension control precision of rotor can be greatly improved.

2. Inverse Decoupling Control of Magnetic Suspension System

When the rotor mass eccentricity is ignored, the radial suspension motion equation of bearingless rotor can be expressed as follows:

$$\left.\begin{aligned} \ddot{\alpha} &= (F_\alpha - f_\alpha - f_{L\alpha})/m \\ \ddot{\beta} &= (F_\beta - f_\beta - f_{L\beta})/m \end{aligned}\right\} \tag{1}$$

Where: m is the rotor mass; α and β are the radial displacement components along horizontal and vertical directions; F_α and F_β are the controllable magnetic suspension force components along horizontal α and vertical β directions respectively; f_α and f_β are the unilateral electromagnetic force along horizontal α and vertical β directions respectively; $f_{L\alpha}$ and $f_{L\beta}$ are the radial load force along horizontal α and vertical β directions respectively.

Defining dq is the reference frame oriented by the rotor flux-linkage of four-pole torque system. Then, the radial electromagnetic force models can be derived as follows [6]:

$$\left.\begin{aligned} F_\alpha &= K_m(i_{s2d}\psi_{1d} + i_{s2q}\psi_{1q}) \\ F_\beta &= K_m(i_{s2d}\psi_{1q} - i_{s2q}\psi_{1d}) \end{aligned}\right\} \tag{2}$$

The unilateral magnetic pull components along horizontal α and vertical β directions can be expressed as follows:

$$\left.\begin{aligned} f_\alpha &= k_s\alpha \\ f_\beta &= k_s\beta \end{aligned}\right\} \tag{3}$$

In upper equations, k_m is constant determined by the structure of bearingless induction motor; k_s is displacement stiffness coefficient determined by the structure of bearingless induction motor; ψ_{1d} and ψ_{1q} are air gap flux-linkage components along d and q reference axis respectively; i_{s2d} and i_{s2d} are suspension control current components along d and q reference axis respectively.

Under dq reference frame, the air gap flux-linkage components along d and q reference axis can be expressed as following equations [8]:

$$\left.\begin{aligned} \psi_{1d} &= \frac{L_{m1}}{L_r}(\psi_{r1} + L_{r1l} i_{s1d}) \\ \psi_{1q} &= \frac{L_{m1} L_{r1l}}{L_r} i_{s1q} \end{aligned}\right\} \tag{4}$$

Now, choose relevant state variables of the magnetic suspension system firstly as follows. Input variables:

$$u = (u_1, u_2)^T = (i_{s2d}, i_{s2q})^T \tag{5}$$

State variables:

$$x = (x_1, x_2, x_3, x_4)^T = (\alpha, \beta, \dot{\alpha}, \dot{\beta})^T \tag{6}$$

Output variables:

$$y = (y_1, y_2)^T = (\alpha, \beta)^T = (x_1, x_2)^T \tag{7}$$

By arranging the upper equations, the state equations of two-pole magnetic suspension system can be derived as follows:

$$\left.\begin{aligned} \dot{x}_1 &= x_3 \\ \dot{x}_2 &= x_4 \\ \dot{x}_3 &= \left\{ K_m[\psi_{1d} u_1 + \psi_{1q} u_2] - k_s x_1 - f_{L\alpha} \right\} / m \\ \dot{x}_4 &= \left\{ K_m[\psi_{1q} u_1 - \psi_{1d} u_2] - k_s x_2 - f_{L\beta} \right\} / m \end{aligned}\right\} \tag{8}$$

According to the interactor algorithm, the reversibility of magnetic suspension system can be analyzed. For the paper length, the detail process of reversibility analysis would not be explained here.
Selecting the input variables of the inverse system as follow:

$$\nu = (\nu_1, \nu_2)^T = (\ddot{y}_1, \ddot{y}_2)^T \tag{9}$$

Then substituting equation (9) into equation (8), the inverse system mathematical model of magnetic suspension system can be derived as follows:

$$\left.\begin{aligned} u_1 &= K[\psi_{1q}(m\nu_2 + k_s x_2 + F_{L\beta}) + \psi_{1d}(m\nu_1 + k_s x_1 + F_{L\alpha})] \\ u_2 &= K[\psi_{1q}(m\nu_1 + k_s x_1 + F_{L\alpha}) - \psi_{1d}(m\nu_2 + k_s x_2 + F_{L\beta})] \end{aligned}\right\} \tag{10}$$

Where, the expression of parameter K is $K = 1/[K_m(\psi_{1d}^2 + \psi_{1q}^2)]$.

Connecting the inverse system in front of the magnetic suspension system in series, then the two-pole magnetic suspension system can be decoupled into two second-order linear integral subsystems; then by adopting appropriate controllers, the dynamic decoupling control of radial displacement components can be achieved. On this basis, the unbalanced vibration compensation of bearingless rotor would be made in the next step.

2. Unbalanced Vibration Compensation of Bearingless Rotor

Defining *uv* as the reference frame that rotates synchronously with rotor.
In rotary motion, rotor mass's eccentricity would produce exciting force *F*a that acts on the bearingless rotor, the two components of exciting force along α and β directions can be expressed as follows[11, 12]:

$$\left.\begin{aligned} F_{a\alpha} &= m\xi\omega^2\cos(\omega t+\theta) \\ F_{a\beta} &= m\xi\omega^2\sin(\omega t+\theta) \end{aligned}\right\} \tag{11}$$

Where: ξ is the rotor mass's eccentricity; ω is the rotation angular speed; θ is the mass eccentricity direction angle of bearingless rotor in the stationary αβ reference frame.
For the symmetry of bearingless induction motor structure, the displacement stiffness along α and β directions are the same [11]. Therefore, under the action of periodic unbalanced exciting force, the periodic unbalanced radial displacement components along α and β directions can be derived as follows [12]:

$$\left.\begin{aligned} \alpha &= A\cos(\omega t+\theta-\gamma) \\ \beta &= A\sin(\omega t+\theta-\gamma) \end{aligned}\right\} \tag{12}$$

In equation (12): A is the amplitude of unbalanced radial displacement, and its value is related to the rotor speed. To improve the suspension control precision of bearingless rotor, it is necessary to suppress the unbalanced exciting force. First of all, the unbalanced radial displacement components should be extracted from the measured radial displacement components. Here, according to the characteristics of unbalanced radial displacement, *i.e.* varying with the angular speed, the unbalanced radial displacement components would be extracted frame the measured radial displacement components; after then, the unbalanced vibration compensation control force components can be derived. The detail process can be summarized as following steps:
1) By synchronous rotating coordinate transformation, the measured radial displacement components in stationary αβ reference frame are converted to those in *uv* synchronous rotating reference frame firstly. Hereinto, the unbalanced radial displacement components are transformed to DC signals; but after coordinate transformation, other radial displacement components are still alternating signals.
2) Through low pass filter, the unbalanced radial displacement components under *uv* reference frame, *i.e.* u_m and v_m can be extracted; then closed loop control would be made for u_m and v_m, and the reference signals of unbalanced vibration compensation force components under *uv* reference frame, *i.e.* F_{cu}^* and F_{cv}^*, can be derived. The unbalanced vibration compensation force components F_{cu}^* and F_{cv}^* can be used to overcome the influence of unbalanced exciting force components;
3) Furthermore, by anti-rotation transformation for F_{cu}^* and F_{cv}^*, the unbalanced vibration compensation force components under stationary αβ reference frame, *i.e.* $F_{c\alpha}^*$ and $F_{c\beta}^*$ can be derived.
4) At the end, replacing F_α and F_β in equation (2) with $F_{c\alpha}^*$ and $F_{c\beta}^*$, and by reorganization, the compensation control current components of unbalanced vibration under *dq* reference frame can be derived as follows:

$$\left.\begin{aligned} i_{c2d}^* &= K(\psi_{1d}F_{c\alpha}^*+\psi_{1q}F_{c\beta}^*) \\ i_{c2q}^* &= K(\psi_{1q}F_{c\alpha}^*-\psi_{1d}F_{c\beta}^*) \end{aligned}\right\} \tag{13}$$

5) Superposing i_{c2d}^* and i_{c2q}^* signals on the output of magnetic suspension inverse system, *i.e.* superposing i_{c2d}^* and i_{c2q}^* on $i_{s2d}\,(u_1)$ and $i_{s2q}\,(u_2)$ signals respectively, then based on the decoupling control of radial displacement, the compensation control force of unbalanced vibration can be produced, witch can be used to overcome the influence of unbalanced exciting force that acts on bearingless rotor, and the unbalanced radial

displacement components can be suppressed or eliminated.
Figure 1 shows the unbalanced vibration compensation control principle of two-pole magnetic suspension system.

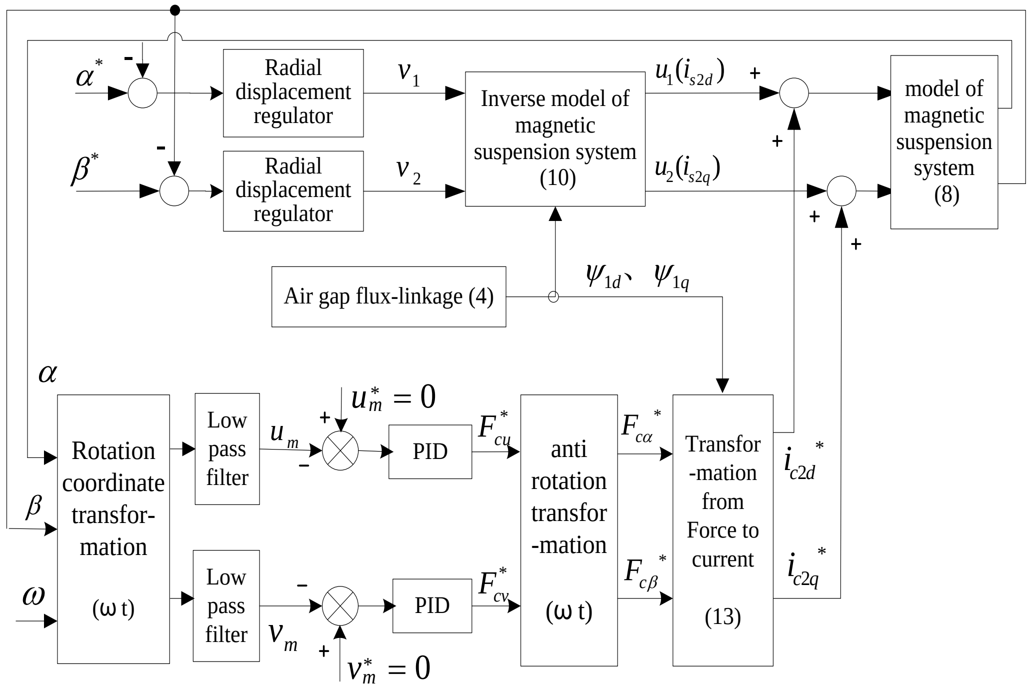
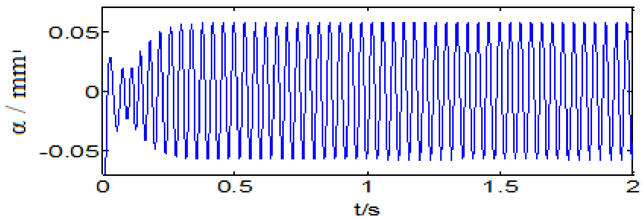


Figure 1. Unbalanced Vibration Compensation Control Principle of Magnetic Suspension System

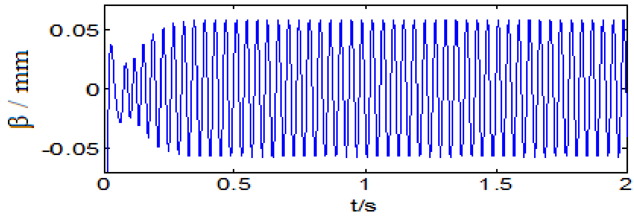
3. Simulation and Analyses

To verify the presented control strategy, aiming at the bearingless induction prototype motor, simulation has been made. Simulation parameters and conditions: inner diameter of stator r=62mm; effective length of iron core l=82mm; air gap length of auxiliary bearing δ_1=0.2mm; rotor inertia J=0.189kg/m^2; mass offset of rotor ξ=0.5mm; rotor mass m=10kg. Torque system parameters: P=2.0Kw, stator resistance R_s=1.6Ω, stator leakage inductance L_{s1l}= 0.0043H, rotor resistance R_r=1.423Ω, rotor leakage inductance L_{s1l}= 0.0043H, single-phase magnetizing inductance is 0.0859H. Suspension system parameters: stator resistance R_{s2}=2.7Ω, stator leakage inductance L_{s2l}=0.00398H, rotor leakage inductance L_{r2l}=0.00398H, single- phase magnetizing inductance is 0.230H. Setting simulation conditions as follows: initial values of two radial displacements α_0=-0.12mm, β_0=-0.16mm; given value of rotor speed is 1500r/min, given value of rotor flux linkage is ψ_r^*=0.6Wb; and the given values of radial displacements $\alpha^*=\beta^*$=0.0mm.

Figure 2 shows the simulation waveforms of radial displacement components when the unbalanced vibrations are not controlled, Figure 3 shows the relevant trajectory of rotor's geometric center. From Figure 2 and Figure 3, it can be known that although closed-loop controls are adopted, the radial displacement components of bearingless rotor along α and β directions fluctuate periodically; and in steady state, the trajectory of rotor's geometric center is a circle with relatively large amplitude. *i.e.* the unbalanced vibration of bearingless rotor is displayed obviously.



(a) Radial displacement component along α direction



(b) Radial displacement component along β direction

Figure 2. Waveforms of Rotor Displacement without Vibration Compensation

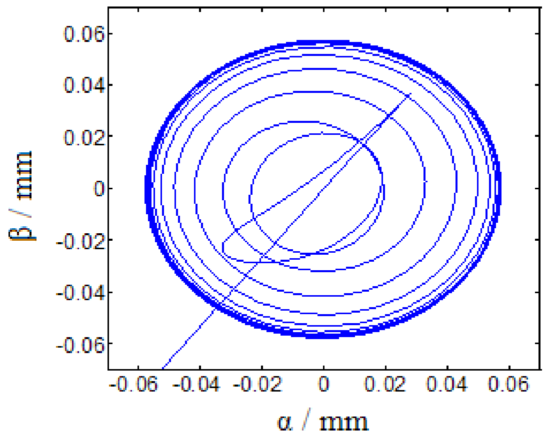


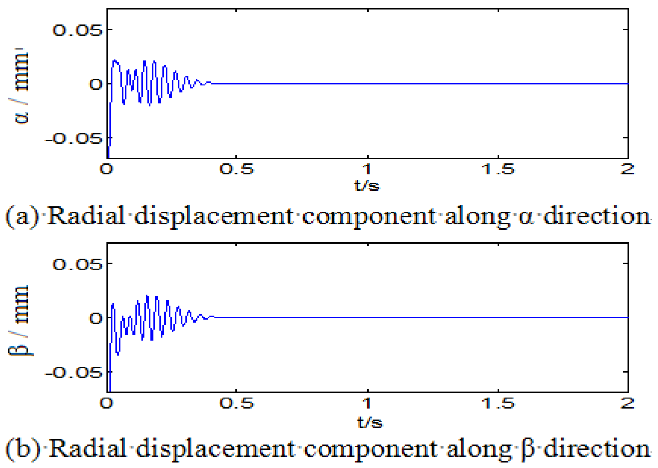
Figure 3. Trajectory of Rotor's Geometric Center without Vibration Compensation

According to the control method presented in Figure 1, the unbalanced vibration can be controlled effectively. Figure 4 shows the simulation waveforms of bearingless rotor's radial displacement components when the unbalanced vibration control method is adopted and Figure 3 shows the relevant trajectory of rotor's geometric center. From Figure 2 and Figure 3, there are following conclusions:

1) At the moment of starting, because the compensation control force has not been fully established, a small amplitude fluctuation of bearingless rotor's radial displacement occurs.

But the fluctuation of radial displacement components can be quickly suppressed.

2) After entering the steady state, under the combined effect of magnetic suspension control system and the unbalanced vibration control system, the radial displacement components of bearingless rotor along α and β directions are gradually reduced to zero, and the trajectory of rotor's geometric center is almost reduced to a point. Then the unbalanced vibration of bearingless rotor is effectively suppressed, and the magnetic suspension control precision of bearingless motor can be greatly improved.



(a) Radial displacement component along α direction

(b) Radial displacement component along β direction

Figure 4. Waveforms of Rotor Displacement after Vibration Compensation

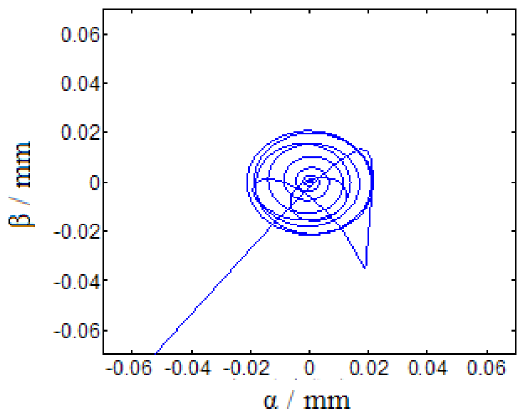


Figure 5. Trajectory of Rotor's Geometric Center after Vibration Compensation

4. Conclusion

To solve the unbalanced vibration problem of bearingless rotor, the mathematical model of magnetic suspension system and that of its inverse system are established firstly, by inverse system method, decoupling control between two radial displacement components is achieved. On this basis, by way of the compensation controller of unbalanced vibration, the unbalanced displacement components are extracted, the compensation control force commends of unbalanced vibration are produced; then in *dq* reference frame, the compensation of unbalanced vibration control current components are carried out; at the end, simulation and verification of the proposed compensation control method is made.

When the proposed control method is adopted, there are following conclusions according to the simulation results:

1) The unbalanced vibration can be suppressed effectively in the start stage;

2) The unbalanced vibration can be eliminated in steady state, and the suspension control precision of bearingless rotor can be improved greatly.

Acknowledgments

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