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Compensation and Control of Bearingless Induction Motor's Unbalanced Exciting Force Based on LMS Filter

Wenshao Bu, Chunxiao Lu, Haitao Zhang, Juanya Xiao and Xiaoqiang Li

College of Information Engineering, Henan University of Science and Technology, Luoyang, 471023, China
wsbu@163.com

Abstract

To solve the problem of bearingless induction motor' unbalance vibration that is caused by rotor mass eccentricity, the generation mechanisms of unbalanced exciting force and unbalanced vibration displacement are analyzed firstly. Then, the extraction method of unbalanced radial displacement based on LMS algorithm is presented, and the compensation strategy of unbalanced exciting force is researched. By real-time regulation and calculation, the compensation forces of unbalanced vibration and unilateral magnetic pull are achieved, and the impacts of unbalanced exciting force and unilateral magnetic pull are compensated. Simulation results of the unbalance vibration control system have shown that the amplitude of unbalance vibration can be greatly inhibited, and the suspension control precision of bearingless rotor can be improved greatly.

***Keywords**: Three-phase Bearingless Induction Motor; Unbalanced Exciting Force; Signal Extraction based on LMS Filter; Unilateral Magnetic Pull; Compensation and Control*

1. Introduction

Conventional AC motor owns the characteristics of simple structure and low cost [1-3], but the conventional motor cannot meet the need of long time high-speed operation. Motor with magnetic bearings has been widely used in high-speed drive kingdoms [3-5], but it has a series of disadvantages, such as: higher magnetic suspensions cost, difficulty to over speed, [6-8]. Based on the comparability of stator structure between magnetic bearing and conventional AC motor, the bearingless motor is proposed [9-12]. Bearingless motor is a new type of electric machine applicable for high-speed operation. In the stator slots of bearingless motor, there are two sets of windings that have a difference in pole-pair numbers, i.e. motor windings (with pole pair p_1 and angular frequency ω_1), and suspension control windings (with pole pair p_2 and angular frequency ω_2); under the action of suspension control magnetic field, the equilibrium distribution of motor magnetic field is broken, then a resulting radial electromagnetic force comes into being, witch acts on the rotor and points to the direction of the magnetic field enhancement. When the qualification of "$p_2=p_1\pm1$, $\omega_1=\omega_2$" is met, the resulting radial electromagnetic force is controllable either in amplitude or in direction. The controllable radial force is so-called magnetic suspension control force; witch can be used to control the suspension of rotor [9-12].

Bearingless control technology can be used to all kinds of AC motor. Because of the advantages of no friction, no lubrication, no mechanical noise, bearingless motor has become a research focus at home and abroad. But, because of the machining accuracy and assembling precisions, there inevitably exists mass eccentricity in rotor to a certain extent, which will lead to the noncoincidence between the geometric central axis and the inertial central axis of rotor[3,13-15], and then the rotary

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exciting force that acts on the rotor will come into being. The exciting force is periodic, and owns the same rotational angular frequency with the mechanical rotor [13-15], which will lead to the periodic unbalance vibration of rotor. The unbalanced exciting force is proportional to the square of the rotor's speed, when the motor speed is enough high, the unbalanced vibration displacement caused by exciting force may exceed the air-gap length of motor, and lead to the collision between the stator and rotor. Then, to find an effective way to overcome the influence of exciting force, and to improve the control precision of rotor's suspension operation, the research on the unbalance vibration problem of bearingless motor is one of the key problems on the practical road of bearingless motor.

About the unbalance vibration problem of magnetic bearing motor, a variety of control methods are proposed, such as notch filter, adaptive feedforward and feedback compensation, robust control, and sliding mode control [3-8]. But, about the unbalance vibration control of bearingless motor, research literatures are very rare, the existing researchs are limited to bearingless synchronous motor, and mainly concentrated in the compensation method of unbalanced displacement, and the compensation purpose is to force the rotor to rotate around its inertial axis [13-15], and then, still cannot meet the requirements of suspension control precision in some application fields, such as high speed grinder drive field. About the unbalance vibration control of bearingless induction motor, no studies have been reported.

In this paper, a bearingless induction motor is regarded as the control object, to overcome the vibration of rotor's radial displacement caused by unbalanced exciting force, the mechanism of unbalance vibration is analyzed firstly, then the compensation control method for unbalanced exciting force is researched; simulation analysis is made based on Matlab/Simulink. Simulation results have shown that the proposed method can overcome the influence of unbalanced exciting force; the suspension control precision of bearingless rotor can be improved greatly.

2. Compensation Control of Unbalance Vibration

2.1 Generation of Unbalance Vibration

Definition: 'αβ' is the stationary reference frame; '*uv*' is the reference frame rotating synchronously with bearingless rotor.

When rotor's mass eccentricity exists, its geometric center G (α_m, β_m) don't coincide with its inertial center *M* (α_c, β_c), then the coordinate relationship between *G* and *M* can be expressed as following:

$$\alpha_c = \alpha_m + \xi \cos(\omega t + \theta) \tag{1}$$

$$\beta_c = \beta_m + \xi \sin(\omega t + \theta) \tag{2}$$

In equations (1) and (2): ξ is the mass eccentricity distance of rotor; ω is the mechanical angular velocity of bearingless rotor; θ is direction angle of mass eccentricity.

When the rotor rotates around its geometric center, the unbalance exciting force $\vec{F}_a$ along the direction of mass eccentricity will be produced. When the rotation speed is constant, $\vec{F}_a$ is a periodic rotation force with constant amplitude, the components of unbalance exciting force along stationary α and β directions can be expressed as following:

$$F_{a\alpha} = \left|\vec{F}_a\right| \cos(\omega t + \theta) = m\xi\omega^2 \cos(\omega t + \theta) \tag{3}$$

$$F_{a\beta} = \left|\vec{F}_a\right| \sin(\omega t + \theta) = m\xi\omega^2 \sin(\omega t + \theta) \tag{4}$$

The random radial displacement control of bearingless motor can be achieved by existing decoupling control method; here, only the periodic unbalanced radial displacement caused by periodic exciting force and its decoupling control method be researched. For simplicity, ignoring the influence of rotary effect and the random displacement of bearingless rotor; according to Newton's law, the motion equation of the rotor's geometric center along α and β directions can be expressed as following[9,12].

$$m\frac{d^2\alpha_m}{d^2t}+c_x\frac{d\alpha_m}{dt}+k_x\alpha_m=m\xi\omega^2\cos(\omega t+\theta) \tag{5}$$

$$m\frac{d^2\beta_m}{d^2t}+c_y\frac{d\beta_m}{dt}+k_y\beta_m=m\xi\omega^2\sin(\omega t+\theta) \tag{6}$$

Solving equations (5) and (6), the periodic displacement components of rotor's geometric center, i.e. the unbalanced radial displacement components can be derived as following:

$$\alpha_m=A\cos(\omega t+\theta-\gamma) \tag{7}$$

$$\beta_m=B\sin(\omega t+\theta-\gamma) \tag{8}$$

Where: A and B are the amplitudes of unbalanced radial displacement along α and β directions respectively. For the structural symmetry of bearingelss induction motor, A equals to B under normal circumstances.

From equations (7) and (8): under the action of rotary unbalanced exciting force, the unbalanced radial displacement components of rotor's geometric center along α and β directions are sinusoidal signals whose angular frequencies are the same with the angular speed of rotor, and the motion trajectory of rotor's geometric center is a circle. I.e. under the action of unbalanced exciting force, periodic unbalance vibration of rotor occurs. So, if no compensation and control measures are adopted to counteract the unbalance exciting force, the suspension control precision of bearingless rotor would be greatly affected. Furthermore, for the unilateral air gap of emergency auxiliary bearing is small in general, the unbalance vibration amplitude of rotor may reach the air-gap length of emergency auxiliary bearing under high rotation speed, and may lead to the collision between rotor and auxiliary bearing.

To overcome the influence of periodic unbalanced exciting force, and to improve the quality of rotor's suspension operation, it is necessary to generate a compensation force F_c that rotates synchronously with the rotor, so as to counteract the unbalanced exciting force F_a, i.e. the condition in equation (9) should be satisfied.

$$F_{cu}=-F_{au},\quad F_{cv}=-F_{av} \tag{9}$$

In equation (9): F_{cu} and F_{cv} are the compensation force components along rotary 'u' and 'v' reference axis respectively; F_{au} and F_{av} are the unbalance exciting force components along rotary 'u' and 'v' reference axis respectively.

Because of the uncertainty of rotor's mass eccentricity angle θ and the difficulty to determine the rotor's mass eccentricity distance ζ, the amplitude and direction of unbalance exciting force are difficult to determine, then the compensation control force of unbalance vibration should be achieved by real time estimation and regulation.

2.2 Extraction of Unbalanced Radial Displacement

The rotor's radial displacement can be divided into random radial displacement and unbalanced radial displacement. The random radial displacement component owns randomicity; it can be controlled by the magnetic suspension decoupling control system of bearingless induction motor, and will not be introduced here.

The unbalanced radial displacement components are sinusoidal signals whose angular frequency is the same with the rotary angular frequency of rotor in 'αβ' reference frame, as shown in equations (7) and (8). The adaptive LMS filter has following characteristics: when the reference input is correlative with the disturbance in main signal, the relative disturbance can be filtered out from the main signal. The unbalanced radial displacement components are sinusoidal signals whose angular frequency are the same with the rotary angular frequency of rotor, own obvious characteristics, the design of relative reference input is easy, then the adaptive LMS filter can be used to extract the unbalanced radial displacement. I.e. the radial displacement can be regarded as the main signal, the unbalanced radial displacement can be regarded as the disturbance signal, and then the adaptive LMS filter can be used to extract the unbalanced radial displacement of bearingless rotor, as shown in Figure.1.

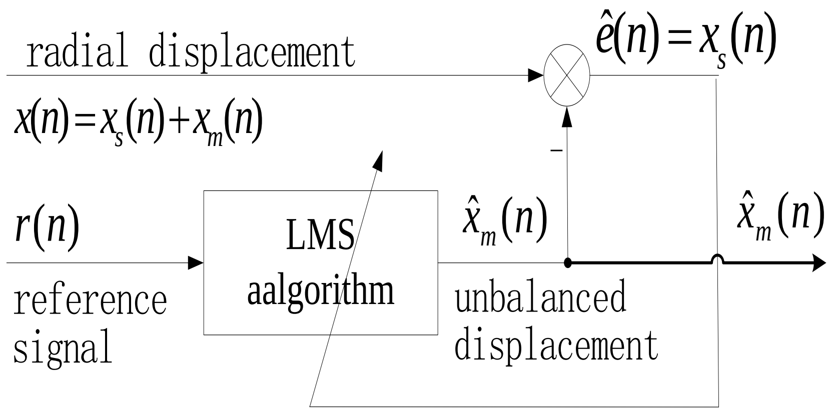


Figure 1. Extracting the Rotor's Unbalanced Radial Displacement

In Figure.1: $x(n)$ is the radial displacement of rotor; it can denote $\alpha(n)$ or $\beta(n)$; $x(n)$ signal is combined with random radial displacement $x_s(n)$ and unbalanced radial displacement $x_m(n)$; the input signal $r(n)=\sin(\omega n+\varphi)$, is a reference one relative with unbalanced displacement $x_m(n)$ of rotor. For simplicity, setting "φ=0". Then, adopting widrow-Hoff steepest descent method to update the weight value of LMS filter, the algorithm is following:

$$W_m(n+1)=W_m(n)+2\mu e(n)r(n-m),\ 0\le m\le M \tag{10}$$

In equation (10): W_m is the weight value of LMS filter; M is the rank number of LMS filter; μ is the step length factor, can be used to adjust the stability and constringency velocity of LMS adaptive filter; $r(n-m)$ can be derived by delaying $r(n)$ signal; the initialization value of W_m can be set to zero.

The current error:

$$e(n)=x(n)-\hat{x}_m(n) \tag{11}$$

The estimation value of unbalanced radial displacement can be expressed as following:

$$\hat{x}_m(n) = \sum_{m=0}^{M} W_m(n) r(n-m) \tag{12}$$

2.3 Estimation and Regulation of Unbalanced Vibration Compensation Force

The unbalanced displacement components of rotor in stationary reference frame are sinusoidal signals that have the same with rotor's rotary angular frequency. When they are transformed to '*uv*' reference frame, the unbalanced displacement components of rotor will become DC variables, as shown in equation (13).

$$\begin{bmatrix} u_m \\ v_m \end{bmatrix} = \begin{bmatrix} \cos(\omega t) & \sin(\omega t) \\ -\sin(\omega t) & \cos(\omega t) \end{bmatrix} \begin{bmatrix} \alpha_m \\ \beta_m \end{bmatrix} \tag{13}$$

For *A* equals to *B* under normal circumstances, then equation (13) can be simplified to equation (14).

$$\begin{bmatrix} u_m \\ v_m \end{bmatrix} = A \begin{bmatrix} \cos(\theta-\gamma) \\ \sin(\theta-\gamma) \end{bmatrix} \tag{14}$$

In equation (13) and (14): u_m and v_m are unbalanced radial displacements of rotor along '*u*' and '*v*' reference axis.

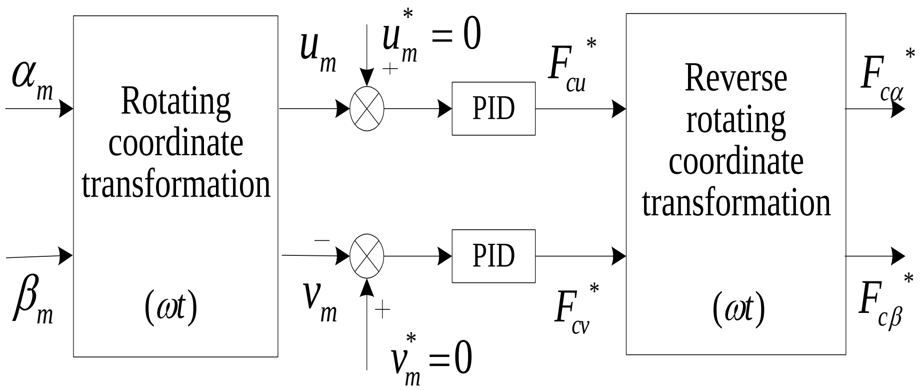


Figure 2. Schematic Diagram of Unbalanced Vibration Compensation Force

The ideal status of unbalanced exciting force compensation is that the amplitude of unbalanced radial displacement be controlled to be zero. Then, taking the given signals of unbalanced radial displacements as zero, and by closed loop controls of u_m and v_m, the compensation control force components of unbalance vibration along '*u*' and '*v*' directions, i.e. F_{cu}^* and F_{cv}^*, can be derived; then by anti-rotation transformation, the compensation control force components of unbalance vibration in stationary αβ reference frame, i.e. $F_{c\alpha}^*$ and $F_{c\beta}^*$, can be derived, as shown in Figure.1.

If only superimposing $F_{c\alpha}^*$ and $F_{c\beta}^*$ on the output of radial displacement regulators of bearingless induction motor, cam the feedforward compensation of unbalanced exciting force be achieved.

2.4 Control System of Unbalance Vibration

Figure.3 presents the schematic diagram of the control system of bearingless induction motor with the function of unbalance vibration control. Hereinto: Inverse system control strategy based on rotor flux orientation is adopted for four-pole torque system; in two-pole suspension system, the magnetic suspension forces are controlled based on the air-gap flux orientation of torque system. ψ_{1m} and φ_m are the amplitude and phase angle of air-gap flux-linkage of torque system, they can be online calculated according to the relationship between air-gap flux-linkage and rotor flux-linkage of torque system. For the paper length limit, the control algorithm of torque system and random radial displacement control system will not be introduced here.

The suspension motion equations of rotor are expressed as following:

$$m\ddot{\alpha} = F_{\alpha} + F_{a\alpha} + F_{L\alpha} + f_{s\alpha} \tag{15}$$

$$m\ddot{\beta} = F_{\beta} + F_{a\beta} + F_{L\beta} + f_{s\beta} \tag{16}$$

In equations (15) and (16): F_{α} and F_{β} are control force of rotor radial suspension motion along α and β reference axis; $F_{a\alpha}$ and $F_{a\beta}$ are unbalanced exciting force along α and β directions; $F_{L\alpha}$ and $F_{L\beta}$ are radial force load along α and β directions; $f_{s\alpha}$ and $f_{s\beta}$ are unilateral magnetic pull along α and β directions, their expressions are as following [13]:

$$f_{\alpha} = k_s \alpha\,, \quad f_{\beta} = k_s \beta \tag{17}$$

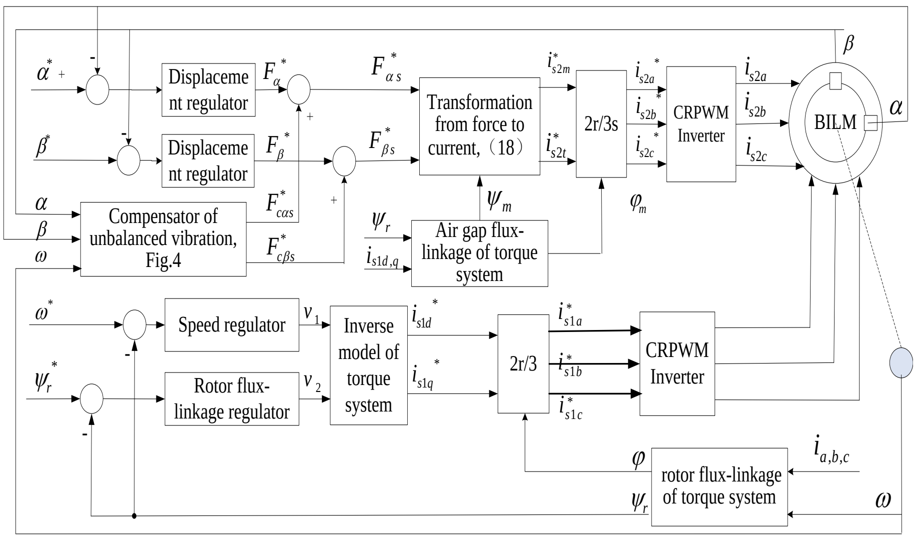


Figure 3. Control System of Bearingless Induction Motor with the Function of Unbalance Vibration Control

To improve the suspension control precision of rotor, it is necessary to compensate the periodic unbalanced exciting force and unilateral magnetic pull. The compensator of unbalance vibration is shown in Figure.4.

In Figure.4: $F_{c\alpha}$ and $F_{c\beta}$ are compensation force components of unbalanced vibration along α and β directions, they are reverse with the unbalanced exciting force components; $F_{c\alpha s}$ and $F_{c\beta s}$ are the comprehensive compensation force components of unbalanced exciting force and unilateral magnetic pull.

Superposing the compensation force components on the control force components of rotor radial suspension motion, the needed controllable magnetic suspension force components $F_{\alpha s}^{*}$ and $F_{\beta s}^{*}$ can be achieved. Then by transformation from force to current, the control current components of magnetic suspension force in MT reference frame, i.e. air-gap flux orientation reference frame of torque system, can be achieved. The transformation equations are as following:

$$i_{s2m}^{*} = F_{\alpha s}^{*}/(K_m\psi_{1m}), \quad i_{s2t}^{*} = -F_{\beta s}^{*}/(K_m\psi_{1m}) \tag{18}$$

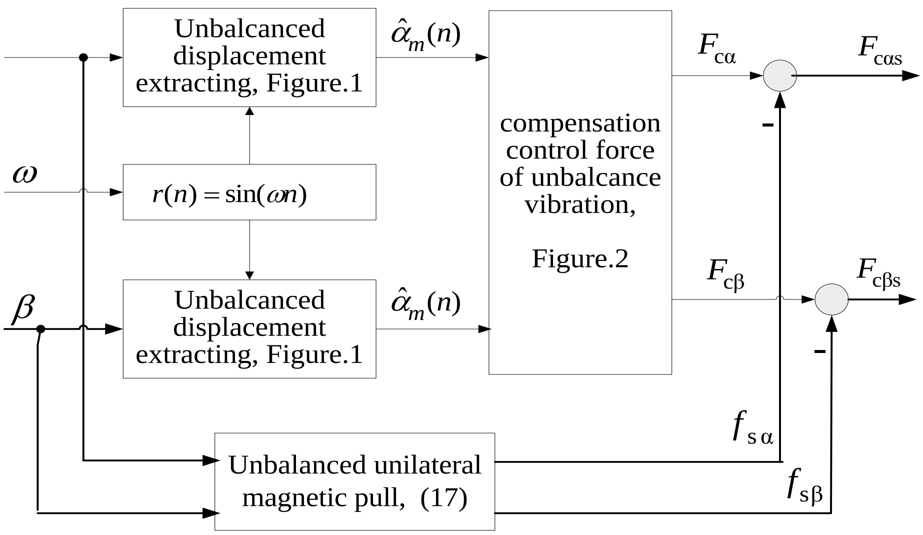


Figure 4. Compensator of Unbalance Vibration

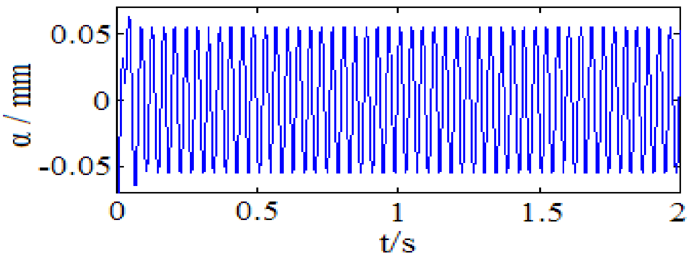
3. Simulation and Analysis of Control System

To verify the presented control method, simulation model based on Matlab is established. In the simulation model of magnetic suspension system, the action of periodic unbalanced exciting force caused by the rotor's mass eccentricity has been considered.

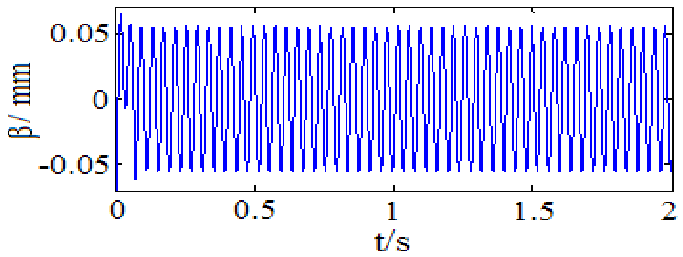
Motor parameters: 1) internal diameter of stator r=62mm, Core length l=82mm, air gap length of auxiliary bearing δ_1=0.2mm, rotary inertia J=0.189kg/m^2, mass eccentricity distance ξ=0.3mm; 2) four-pole torque system: 2.2kw, R_s=1.6Ω, stator and rotor leakage inductances L_{s1l}=0.0043H, L_{s1l}=0.0043H; R_r=1.423Ω, single- phase excitation inductance equals to 0.0859H; 3) two-pole suspension system: R_{s2}=2.7Ω, stator and rotor leakage inductances L_{s2l}=0.00398H, L_{r2l}= 0.00398H, single-phase excitation inductance equals to 0.230H.

Setting the simulation conditions: Initial displacement α_0=-0.12mm, β_0=-0.16mm, given displacement $\alpha^*=\beta^*$=0, m=10kg, initial given speed ω^*=314rad/s, i.e. 1500r/min, initial given rotor flux-linkage ψ_r^*=0.95Wb; the LMS adaptive filter module in MATLAB toolbox is adopted, its order number is 63, step length factor μ=0.001.

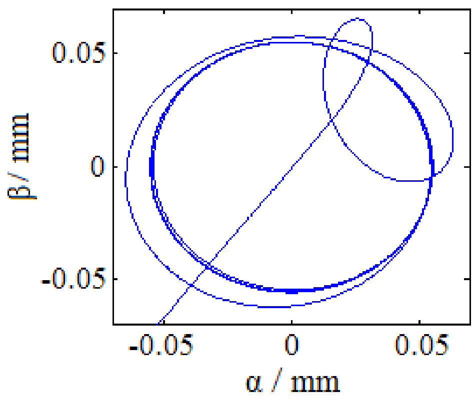
Figure.5 gives the waveforms without load and compensation control force of unbalance vibration. From Figure.5: under the action of periodic unbalanced exciting force, obvious unbalance vibration of rotor occurs, i.e. there are period unbalanced radial displacements along α and β reference axis, the amplitude of unbalanced radial displacement is about 55um; in steady state, the motion trajectory of rotor's geometric center is a approximate circle, the suspension control precision of bearingless rotor is lower.



(A) Radial Displacement Along A Reference Axis



(B) Radial Displacement along B Reference Axis

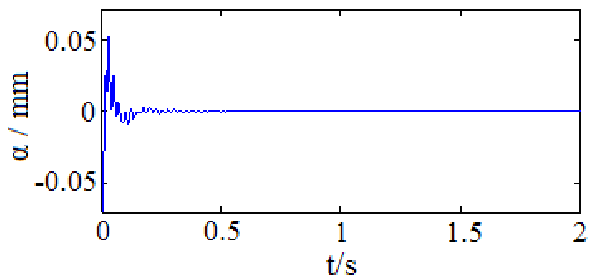


(C) Motion Trajectory of Rotor's Geometric Center

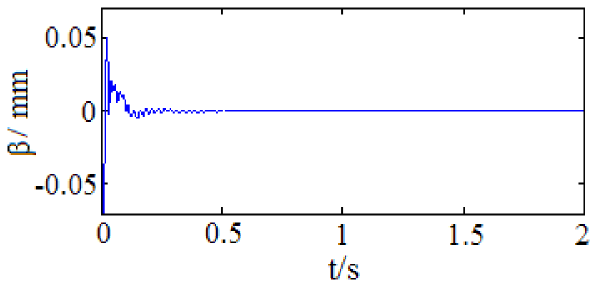
Figure 5. Simulation Waveforms without Unbalance Vibration Compensation Control

Figure.6 gives the waveforms without load when the compensation control force of unbalance vibration is added. From Figure.6: After a small amplitude vibration of radial displacement, with the action of compensation control force, the unbalanced radial displacement is rapidly inhibited. In steady state, the motion trajectory of rotor's geometric center is almost contracted to a point, the unbalanced radial displacement is completely eliminated, and the suspension control precision of bearingless rotor is greatly improved.

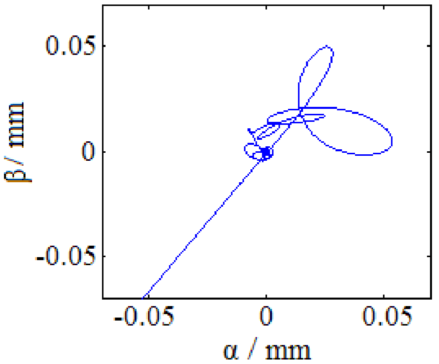
In addition, to research the operation performance with torque load, at the moment of 2.0s, torque load T_L is changed from 0 to 12.7N.m, the simulation results with torque load in shown in Figure.7. From Figure.7: when torque load is add, there are small fluctuations of radial displacement on the moment of adding torque load, but under the control action of system, the components along α and β reference axis are all rapidly reduced; In steady state, the components of unbalanced radial displacement along α and β reference axis are basically maintained at zero. The simulation results have shown the better ability of anti-load disturbance.



(A) Radial Displacement along A Reference Axis

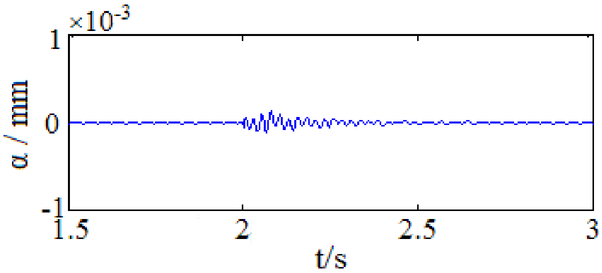


(B) Radial Displacement along B Reference Axis

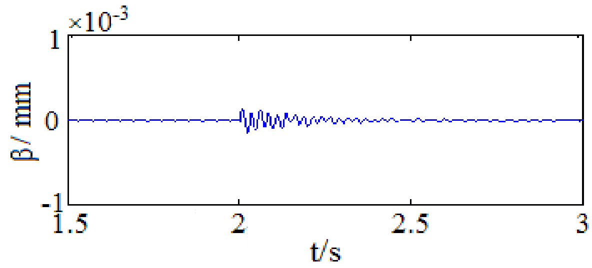


(c) Motion trajectory of rotor's geometric center

Figure 6. Simulation Waveforms with Unbalance Vibration Compensation Control



(A) Radial Displacement along A Reference Axis



(B) Radial Displacement along β Reference Axis

Figure 7. Simulation Waveforms When Load Is Suddenly Added

In the operation process of bearingless induction motor, the speed may need to change, to verify the working performance of bearingless induction motor when speed changes, at the moment of 2.0s, the rotor fulx-lingkage is changed from 0.95Wb to 0.5Wb, at the moment of 3.0s, the motor speed is changed from 1500r/min to 3000r/min, Figure.8 gives the displacement waveform when motor speed changes. From Figure.8: in the speed change process, the radial displacement components of rotor are not more than 10μm, the simulation results have shown the higher control performance of bearingless induction motor.

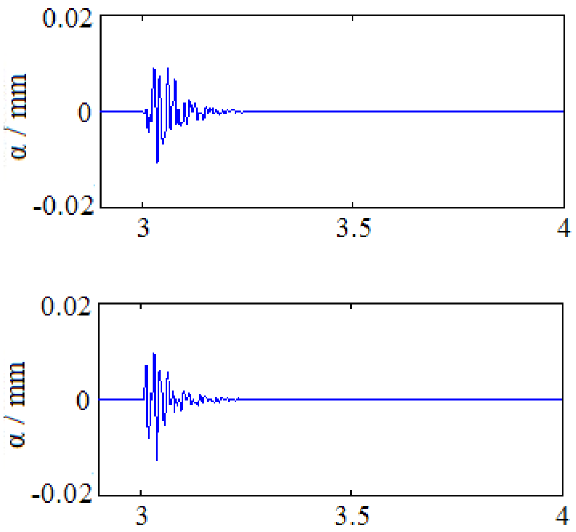


Figure 8. The Radial Displacement Waveforms When Motor Speed Changes Suddenly

4. Conclusion

Aiming at the unbalance vibration problem of bearingless induction motor caused by rotor's mass eccentricity, the generation mechanisms of unbalanced exciting force and unbalanced displacement are introduced firstly. Then, based on LMS adaptive filter, the extraction algorithm of unbalanced displacement is researched; to eliminate the unbalanced radial displacement of rotor, adopting compensation control force of unbalance vibration to counteract the unbalanced exciting force, the real time estimation and adjustment method of compensation control force is proposed. At the end, based on Matlab/ simulink toolbox, Simulation and analysis of the unbalance vibration control system of bearingless induction motor is made.

From the simulation results:

1) When the proposed compensation control strategy of unbalance vibration is adopted, the unbalanced radial displacement of bearingless induction motor can be greatly

suppressed in the dynamic process; in steady state, the unbalanced radial displacement can be completely eliminated; the suspension control precision of bearingless rotor can be improved greatly.

2) Either adding torque load, or changing the motor speed, the fluctuation of radial displacements are all very small, the control system of bearingless induction motor owns higher ability of anti-disturbance.

3) The proposed compensation control strategy of unbalance vibration is effective and feasible.

Acknowledgements

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