EXPERIMENTAL COMPENSATION OF RUNOUT AND HIGH ORDER HARMONIC EFFECTS IN LORENTZ FORCE TYPE INTEGRATED MOTOR-BEARING SYSTEM

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ABSTRACT
An integrated motor-bearing system integrates the functions of an active magnetic bearing and an electric motor into a single unit. Because this system requires the simultaneous generation of torque and radial control force, there will inevitably be undesired high order harmonic components in the rotating magnetic field, which will result in torque ripple and radial force distortion. Sensor target runout is a severe excitation source and, as such, causes a lot of vibration. In this paper, we propose an experimental compensation procedure for the oscillation and coupling of torque and radial force using a digital controller of the Lorentz force type integrated motor-bearing system in a dual disk rotor configuration. After the compensation of high order harmonics, the runout profile and rotor unbalance are identified by the extended influence coefficient method. The proposed scheme does not require complicated analysis or modeling of high order harmonic effects, and it can also compensate for manufacturing errors. The experimental results confirm that this compensation method effectively attenuates the rotor vibration throughout the operating range of rotational speeds.

NOMENCLATURE

- $A$: Magnitude of motoring current
- $B$: Magnitude of fundamental wave in air gap flux density
- $C$: Magnitude of radial force current
- $k$: Number of turns per concentrated winding
- $l$: Effective length of winding part crossing the flux
- $r$: Effective radius of stator windings
- $t$: Time
- $2\alpha$: Winding pitch angle
- $\theta$: Angular coordinate based on $x$ axis
- $\phi$: Phase shift of radial force control current
- $\psi$: Phase shift of motoring current, torque angle
- $\omega$: AC current frequency [rad/sec]
- $\Omega$: Rotational speed

INTRODUCTION
An integrated motor-bearing system integrates the functions of an active magnetic bearing (AMB) and an electric motor into a single unit. It is capable of simultaneously generating torque for motoring as well as radial or axial forces for levitating the rotor through the control of rotating magnetic flux distribution. Although it may have a simple and compact mechanical structure, it preserves the inherent advantages of AMB, such as no contact and lubrication, high peripheral speed and precision operation, and adjustability of the bearing stiffness and damping [1]. However, the design and analysis of its magnetic field and of its controller do become somewhat complex because the motoring and levitation of the rotor must be simultaneously controlled. Since the end of the 1980s, extensive research has been carried out, resulting in the development of various kinds of integrated motor-bearing systems [2]. The new integrated motor-bearing systems that have been proposed in recent years deserve much attention, as they employ the Lorentz force for rotor position regulation as well as motoring [3-7]. Unlike the Maxwell force, the bi-directionality and good linear properties of the Lorentz force can simplify the analysis and design of the system when it is applied to stabilize the rotor position. In addition, use of the Lorentz force allows the rotor permanent magnets to have sufficient thickness so that the demagnetization problem can be prevented.

Although the integrated motor-bearing systems have numerous merits, they are limited by the conventional structure of electric motors that needs for the motoring torque, so it is
inevitable that there will be undesirable high order harmonics in the rotating magnetic field. Even a small amount of these harmonics can cause high frequency oscillation as well as coupling of the torque and radial force. In addition, for the magnetic force, the integrated motor-bearing system requires a feedback controller like an AMB to use the measured displacement of the rotor. The positioning accuracy of the system controller essentially depends on the quality of measured signals, which is strongly affected by the resolution of the sensors in use and by the presence of the sensing target runout that, as a severe excitation source, contributes much to the rotor vibration. The mechanical runout is caused by the non-concentricity and surface irregularity of the sensor target, while the electrical runout is due to residual magnetic fields, metallurgical microscopic segregation, or localized stress concentration [8].

This paper proposes an experimental compensation procedure for torque ripple, radial force oscillation, and coupled torque and radial force that uses a digital controller in the Lorentz force type of integrated motor-bearing system in a dual disk rotor configuration. After compensating for high order harmonic effects, the runout profile and rotor unbalance are identified by the extended influence coefficient method [1]. In previous work, a dual rotor disk configured experimental setup was developed in order to verify the authors’ proposed general design formula in the integrated motor-bearing system using Lorentz force [7]. The basic operational principle is, here, briefly reviewed with the parameters applied to the experimental setup.

DESIGN PRINCIPLE

Let us consider 6 concentrated windings spaced equidistantly along the periphery of the stator as shown in Fig. 1. The positive sign is assigned to the magnetic flux coming out of the page and to the effective current running from the center of the stator to the outer radial direction. Under that sign convention, the stator will deliver a positive torque in a counter-clockwise direction. For simplicity of analysis, assume that the magnetic flux density generated by the 8 pole rotor permanent magnets has only a sinusoidal waveform of

$$B_R = B \cos(\omega t - 4\theta)$$

where $B$ is the amplitude of the flux density, $\omega$ is the AC circular frequency, and $\theta$ is the angular coordinate. Now consider the currents for motoring torque given in the form of

$$I_i^M = A \cos(\omega t + \psi + \frac{2\pi}{3} i) \quad i = 0, 1, \ldots, 5$$

where $A$ is the magnitude of the current and $\psi$ is the phase shift on the basis of the rotating magnetic flux of the rotor. The stator winding distribution can be simply represented by using Dirac’s delta function as

$$D_i = k\delta(\theta - \frac{\pi}{3} i + \alpha) - k\delta(\theta - \frac{\pi}{3} i - \alpha)$$

where $k$ is the number of turns in a concentrated winding and $2\alpha$ is the winding pitch angle. Then the motoring torque produced by these currents can be obtained, using Lorentz law, as

$$T = \sum_{i=0}^{5} r l \int_{0}^{\frac{2\pi}{3}} B_R I_i^M D_i d\theta$$

$$= 6ABkr l \sin(4\alpha) \sin \psi$$

where $r$ and $l$ are the effective radius and length of stator windings respectively. This suggests that we can control the system like a standard synchronous AC motor by adjusting the phase shift $\psi$ as its torque angle. In addition, the torque attains its maximum value when the winding pitch angle $2\alpha$ is tuned to the pitch angle of the rotor permanent magnets.

As in the previous procedure for motoring torque generation, here the currents for the radial control force are considered in the form of

$$I_i^C = C \cos(\omega t + \phi + \frac{\pi}{3} i) \quad i = 0, 1, \ldots, 5$$

where $C$ is the magnitude of the current and $\phi$ is the phase shift. Note here that the phase difference of radial control current in adjacent windings is half that of the motoring current. Applying these currents to the windings represented by equation (3), we obtain the expressions for the radial control forces in the x and y directions as

$$F_x = -\sum_{i=0}^{5} I_i^C B_R D_i \sin \theta d\theta$$

$$= 3BCkl \sin(5\alpha) \cos \phi$$

The basic operational principle is, here, briefly reviewed with the parameters applied to the experimental setup.

**Figure 1** Coordinates and winding distribution on the stator
Equation (6) and (7) prove that the strength and the two dimensional direction of the radial force can be linearly controlled by the magnitude \( C \) and the phase shift \( \phi \) of the current, respectively. They also reveal that the winding pitch angle to get the maximum radial force is slightly different from that for the maximum torque. The radial control force is also independent of the rotational speed of the rotor, which makes it easy to design the radial position controller. The radial control current represented by equation (5) can be rewritten by substituting the following for equations (6) and (7) as

\[
I^C_i = \frac{1}{3Bkl\sin(5\alpha)} \left( F_x \cos(\alpha t + \frac{\pi}{3} i) - F_y \sin(\alpha t + \frac{\pi}{3} i) \right)
\]

\[
I^C_i = \sum_{i=0}^{5} I^C_t^2 \cdot B_k l_i C_i D_i \cos \theta d\theta
\]

\[
F_y = 3Bkl \sin(5\alpha) \sin \phi
\]

Hence, the radial control current of each winding, superposed upon the corresponding motoring current, can be calculated from the regulation force signals that are independently determined by the PD controller using measured displacement signals in the \( x \) and \( y \) direction.

EXPERIMENTAL SETUP

The experimental setup was constructed in an upright configuration in order to be free of gravitational force in the radial direction, as shown in Fig. 2. Two eddy-current type proximity probes (PU-05A, AEC) were mounted at 90° from each other, around the lower rotor disk for non-contact measurement of the rotor position. Figure 3 shows one of the stator disks whose windings are molded and fixed into a disk shaped plastic skeleton. Each rotor disk consists of an iron back yoke, thick permanent magnets and a rim-shaped sensor plane attached around the rotor disk as shown in Fig. 4. The specifications of the experimental setup are listed in Table 1. The design of the power amplifier is based on linear power operational amplifier chipsets. It includes a current feedback loop that enables a quick response to the input signal in spite of the phase delay caused by the inductance and large back EMF of the stator windings. The digital control scheme has a simple PD action and modulation with reference to the rotating frequency, implemented on the dSPACE motion control board.

HIGH ORDER HARMONIC COMPENSATION

In order to enable the torque and radial control force to be simultaneously generated, the integrated motor-bearing system should have a rotating magnetic flux field in the air-gap, where the unwanted high order harmonic components are inevitable. Figure 5 shows the flux density distribution measured by a Gauss meter in the midst of the air gap without the stator disk. Due to the large air-gap length and the absence of stator core, the distribution curve looks very smooth and close to a...
Table 1 Specifications of experimental setup

<table>
<thead>
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<th>Dimensions</th>
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<tr>
<td>Outer radius of magnet</td>
<td>50 mm</td>
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<tr>
<td>Inner radius of magnet</td>
<td>25 mm</td>
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<td>Magnet thickness</td>
<td>9 mm</td>
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<tr>
<td>Space between magnets</td>
<td>2 mm</td>
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<tr>
<td>Air gap length</td>
<td>9 mm</td>
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<tr>
<td>Number of turns</td>
<td>216</td>
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<td>Nominal coil diameter</td>
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<tr>
<td>Winding width</td>
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<tr>
<td>Winding thickness</td>
<td>7.8 mm</td>
</tr>
<tr>
<td>Pitch angle</td>
<td>36°</td>
</tr>
<tr>
<td>Back iron thickness</td>
<td>9 mm</td>
</tr>
<tr>
<td>Rotor disk mass</td>
<td>964 g</td>
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<table>
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<tr>
<th>Materials</th>
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<tr>
<td>Magnet</td>
<td>Sintered NdFeB (N-40H)</td>
</tr>
<tr>
<td>Rotor disk frame</td>
<td>AL7075</td>
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<tr>
<td>Stator disk frame</td>
<td>Fiber-reinforced bakelite</td>
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<tr>
<td>Rotor back iron</td>
<td>SS400</td>
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<table>
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<tr>
<th>System parameters</th>
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<tr>
<td>Current stiffness</td>
<td>8.39 N/A</td>
</tr>
<tr>
<td>Proximity probe sensitivity</td>
<td>5 mV/μm</td>
</tr>
<tr>
<td>Maximum supply voltage</td>
<td>±70 V</td>
</tr>
<tr>
<td>Sampling frequency</td>
<td>5 kHz</td>
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</table>

sinusoidal wave; however, it contains odd orders of harmonic components, as shown in Fig. 6. In addition, there are mechanical manufacturing errors in the rotor and stator, such as uneven rotor permanent magnets, misalignment of stator windings, and tilting of the rotor disks. Similar to the high order harmonic effects, it is reported that these errors can distort torque and radial force and cause coupling. The effects of high order harmonics and mechanical tolerance can be classified into 4 cases:

- Torque ripple
- Radial force oscillation
- Coupled torque
- Coupled radial force

All of these phenomena are periodic to the rotational angle and the system uses a digital controller that can generate complicated waveforms. Moreover, the Lorentz force for both the motoring and the radial position control is linearly proportional to the magnitude of the current, so each effect can be identified by the variation of controller command at different current magnitudes. The basic scheme of compensation is thus that the radial position of the rotor and the current command signals are acquired at the specific conditions. The compensation is performed at the very low rotational speed of 1 rpm, where the runout excitation and unbalance response become negligible. In order to improve the accuracy, all measured data are averaged and filtered by the FFT algorithm. Then the processed data are loaded as current modulation references.

Torque ripple can be easily identified by comparing the rotor angle measured by an encoder with the AC frequency signal of the motoring current in the controller. In order to separate the effects of the radial control current from the motoring one, supply only the motoring current to the system and support the lower part of the rotor shaft with a ball bearing that replaces the touchdown bearing. Figure 7 shows that the difference between the encoder signal and the angle of motoring command can be well compensated by the modified motoring current waveform that is shown in Fig. 8. In fact, the
The torque ripple effect in the system is quite small compared to that in the cored motor, and it is spontaneously canceled out in high speed rotation due to the large inertia of the rotor disk. However, at a very low speed operation, it causes great trouble for other compensation schemes.

The coupled radial force is defined as the radial force produced by the torque current command. In an upright rotor configuration, it is assumed that the feedback control loop of the radial position can automatically estimate the radial control current only for the coupled radial force. The profiles of the radial control current were measured for several different levels of torque current, as shown in Fig. 9. In order to improve the data definition captured by the proximity probe, the gains of the PD controller are brought down to their minimum values in the stable range, i.e., \( K_P = 10 \) and \( K_D = 0.04 \). Since the coupled radial force is proportional to the magnitude of the torque current and the differences between adjacent data are almost equal, the reference current signals for the compensation were calculated using the profiles of \( A = 2.5 \) A to 1.5 A. It is found, however, that the radial control current still remains when the motoring current becomes zero, probably due to the positioning errors of stator windings and rotor permanent magnets and the trivial discrepancy of ground level among

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**Figure 7 Torque ripple compensation**

**Figure 8 Modified profiles for motoring current reference**

**Figure 9 Measurement of coupled radial force currents for (a) 0th, (b) 1st, and (c) 2nd winding**
Figure 10 shows the profiles of residual control currents with magnitude of less than 20 mA.

The radial force oscillation is the sway of the radial control force in magnitude and direction, caused by the radial control current, with respect to the rotor angle. As an idea to compensate for this oscillation, the radial position controller tries to maintain the force equilibrium even if a certain amount of standing constant radial force is loaded at the rotor. The experimental setup is thus inclined to a 30° angle in the x or y direction so that gravity can be used as a reference radial load. Since the coupled torque have not yet been compensated, it is important to set the magnitude of torque current to the maximum limit of 2.5 A to guarantee a constant rotational speed. In order to compensate for the static deflection of the rotor position, a constant force command of 7.3 N is added to the feedback controller in the opposite direction of the rotor weight. The whirling motions of the rotor during the compensation are shown in Fig. 11, where the 3rd harmonic component swings the rotor dominantly.

Figure 10 Estimation of residual control current

Figure 11 Whirling orbits generated by uncompensated radial control current in (a) x or (b) y direction

Figure 12 Normalized radial control current references for (a) x or (b) y directional force
derivative gain action, the radial position controller is intended to overestimate the radial control current for the large motion of the rotor, which then makes the iteration process necessary. Figure 12 represents the normalized reference current profiles for the radial control obtained after 3 iterations.

Finally, the coupled torque is defined as the torque generated by the radial control current. It can disturb the rotational angle and cause the direction of the radial control force to yaw. Keep the inclination of the experimental setup and reduce only the torque current to the magnitude of 0.2 A, then the yawing motions of the rotor position can be observed as shown in Fig. 13. In order to identify the coupled torque, a PID controller for the torque regulation was added to the original one, where the rotor position signal in the orthogonal direction was used as the feedback command. The supplementary controller gains were determined experimentally as $K_p = 1.0$, $K_i = 1.0$, and $K_D = 0.01$, and digital low-pass filters that had a cut-off frequency of 100 Hz were applied to the derivation and integration process. As in the previous compensation, the normalized reference current profiles for the coupled torque were sufficiently converged after 3 iterations, as shown in Fig. 14.

**RUNOUT AND UNBALANCE COMPENSATION**

The runout of the displacement sensor target in an integrated motor-bearing system acts as an excitation source via the feedback control loop, and often leads to serious rotor vibration. Kim and Lee proposed an in-situ runout identification for AMB systems using the extended influence coefficient method [1]. This method estimates the influence coefficients from the frequency response function obtained by a comparison of the harmonic components in the original and in the trial run at the same rotational speed. Then the runout profile can be identified from the coefficients and compensates for the runout input by a feedforward control scheme. The compensation of high order harmonic effects mentioned so far is a prerequisite for applying this method to the integrated motor-bearing system. This method has the advantage that
target runout is identified and compensated under a given operating condition of the integrated motor-bearing system, and does not require any extra sensors or devices for the measurement and compensation of runout. The proposed scheme is, however, very similar to the open loop controller for suppressing unbalance response, except in terms of considering higher harmonic components. Hence, the runout excitation synchronous to the rotation and the unbalance response cannot be distinguished from the fundamental harmonic component of the measured signal. Fortunately, the proximity probes mounted at the experimental setup directly detect the displacement of the rotor, as shown in Fig. 2, from the surface around the lower rotor disk. Moreover, the effects of tilt and misalignment between the upper and lower disks have been already compensated through the previous compensation. Therefore, the fundamental component in the runout signal can be negligible, and the 1X component of the measured signal was regarded as only the unbalance response in the compensation process. The runout signals measured from \( x \) and \( y \) directional proximity sensors are identical except for a 90\(^\circ\) phase difference, so there is no need to consider the \(-1X\) component in the directional runout spectrum, which has to be zero in a complex notation.

In order to estimate the influence coefficients, a trial runout signal was composed of the harmonic components from \( \pm 2X \) to \( \pm 30X \) and then loaded into the memory of the digital controller. The higher components in the measured response are independent of the rotor unbalance, so a trial unbalance force signal was also added to the controller. The control gains were adjusted to \( K_P = 40 \) and \( K_D = 0.157 \) for the standard operating condition. The resulting responses were captured for the original and test runs, increasing the rotational speed \( \Omega \) from 250 to 2000 rpm because the unbalance force is proportional to \( \Omega^2 \). Since the sampled data are essentially periodic, with a period of one revolution, a rotational angle-based discrete complex Fourier transformation of the measured signals was performed, using only rectangular windows to avoid spectral leakage. Then the complex Fourier coefficients of the runout and unbalance were calculated to estimate the

![Graph 1](image1.png)

**Figure 15** Estimated runout profile and its directional spectrum

![Graph 2](image2.png)

**Figure 16** Estimated radial force vector for the unbalance compensation
compensation signals in the time domain, which were finally substituted for the test data in order to investigate the runout and unbalance responses under the compensation.

Figure 15 shows the estimated runout profile and corresponding directional spectrum. Every data were calculated according to the rotational speed and were then averaged because, as expected, there was little difference between the results. Note that the 4th, 6th, and 8th harmonic components are dominant, which has been solved by the fact that the eddy current type proximity probes interfere with the fringing flux of the rotor disk alternative according to the rotational angle. Figure 16 represents the estimated force vector for the unbalance compensation. The magnitude agrees well with the theoretical expectation, and the unbalance of the rotor is identified as 5.2 g-cm. However, the phase angle based on the x axis increases along with the rotational speed. The phase delay as shown in Fig. 16 can be regarded as a reflection of the electromagnetic actuator dynamics, which is affected by flux leakage, fringing effect, air-gap discrepency, and eddy current loss, which includes the dynamics of power amplifiers with current feedback [9]. Consequently, the phase angle of the unbalance compensation signal was evaluated in the form of a 1st order exponential decayed function. Figure 17 compares the measured whirl responses of the experimental setup at 1500 rpm, before and after the compensation using the estimated runout and unbalance signal shown in Figs. 15 and 16. The complicated rotor vibration with amplitude of about 20 µm has been attenuated to less than 2 µm. The amplitude variations of radial vibration in accordance with the rotational speed are plotted in Fig. 18. Whirling motions were observed at every 50 rpm interval. The upper limit of rotational speed, 2300 rpm, results mainly from the voltage saturation of the back electromotive force, which is directly proportional to the rotational speed. The compensation scheme suppresses residual vibration at a level of about 2 µm overall, but the responses at a rotational speed of less than 500 rpm are somewhat larger. From the aspect of bearing noise, the reason for this seems to

Figure 17 Comparison of whirl responses for runout and unbalance compensation at 1500 rpm

Figure 18 Amplitude attenuation of radial vibration under the runout and unbalance compensation
be related to the resonant mode of the self-aligning ball bearing installed at the upper part of the rotor shaft. Considering the tolerance of a general ball bearing, these results are sufficient to suggest that the runout and unbalance are effectively identified and compensated, even though the nominal sensor resolution is 0.5 µm.

CONCLUSION

This paper has described an experimental approach to the calibration of torque and radial force to avoid the effects of high order harmonics and to the compensation of runout and unbalance in a Lorentz force type integrated motor-bearing system. The experimental results imply that the compensated digital controller attenuates well the rotor vibration at all operating speeds. The proposed scheme does not require complicated analysis or modeling of high order harmonic effects. Moreover, it can also compensate for manufacturing errors, such as an uneven rotor flux distribution and misalignment of stator windings. Therefore, this method is quite practical and may be applied to the identification or the development of a precision controller for various kinds of integrated motor-bearing systems.

REFERENCES