ANALYSIS AND MEASUREMENT OF ROTATING ERROR OF AMB

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ABSTRACT
This paper first suggests the use of Fourier frequency analysis method of two dimension function to analyze radial rotating errors occurred in the front and rear magnetic bearings of main shaft and points out that the main cause to affect radial rotating accuracy of rotating shaft at a high speed is the dynamic imbalance of shaft itself, whose conclusion provides the theoretical basis for the improvement of rotating accuracy of the high speed shaft.

INTRODUCTION

As a rotating shaft at a high speed, there should be a static non-motion rotating axle line on it. Nevertheless in the real case, this rotating axle line can not avoid doing minor error motion due to its manufacturing accuracy and dynamics. In the authors' lab, there is a magnetic bearing system with the five degrees of freedom. The main cause affecting the rotating accuracy of main shaft is found via the actual measurement and analysis of this system. As a result, a method to improve the rotating accuracy of main shaft is presented in this paper.

1. Mathematical Description of Main Shaft

In engineering practice, when the shaft rotates, the rotating shaft is inevitable to do the minor error motion due to its manufacturing accuracy and dynamic principle. Any point in any cross section of the shaft with its position in the plane can be described using 2D vector quantity so that the rotating motion by the point can be expressed by the complex function $M(t)$ [1] of time, whose formula of rectangular coordinates is as follows:

$$M(t)=x(t)+j\cdot y(t) \qquad (1)$$

When the shaft has the rotating error, the rotating motion of the point can be considered as the overlapped results of both vector quantities of the circumference motion $M_r(t)$ of the observing and measuring point rotating along the shaft and the radial error motion $M_e(t)$ of the shaft:

$$M(t)=M_r(t)+M_e(t) \qquad (2)$$

Since the rotating shaft center can not rotate along the shaft, $M_e(t)$ is sure not to contain the component of circumference motion rotating synchronically with the shaft.

2. Frequency Analysis of Rotating Motion of Rotating Main Shaft

If the motion $M(t)$ of a certain point on the rotating shaft is supposed to be periodical motions, it can be extended in terms of Fourier series. Usually, when the shaft rotates a whole circle, the motion single on that point can be considered as a cycle. Supposing that the rotating angle velocity is ω, $M(t)$ can be extended as the following Fourier series [2],

$$M(t)=\sum_{n=-\infty}^{\infty} C_n e^{jnwt} \quad (n=0,\pm1,\pm2....) \quad (3)$$

$$C_n = \frac{1}{T}\int_0^T M(t) e^{-jnwt}\,dt \quad (4)$$

The above Eq (3) and Eq.(4) indicate that periodical rotating motions can be decomposed into many frequency components doing circumference motions, whose corresponding motion angle velocity is $n\omega$, of which the zero-order component C_0 is one point on the fixing frame of reference, representing the average position of observing and measuring point in space, i.e. the people have had a common understanding of so-called average rotating center[2]. The positive order component $C_1 e^{j\omega t}$ is the circumference motion of the observing and measuring point rotating along the shaft. i.e. $M_r(t) = c_1 e^{j\omega t}$. The negative order frequency component $c_{-1}e^{-j\omega t}$ is a circumference motion rotation towards the reverse direction to the rotating shaft with same frequency so that it is completely attributed to the error motion, i.e. the negative order component in rotating errors of the rotating shaft.

In calculating frequency component of rotating motion, the 2D function FFT can be used. The single of rotating motion is a complete function so that its amplitude-frequency spectrum is non-symmetry. In theory, all the negative components just like all the positive components are likely to exist, whose actual existence has been proved by the actual measurements[2].

3. Analysis of Multi-Circumference Rotating Motion

The rotating motion of rotating shaft may, in fact, have no strict periodicity to follow, whose motion tracks may have slight variations or great differences per revolution along with the shaft. In this case, the rotating motions with more circles in succession can be taken to analyze the frequencies.

It is just here that we can understand the rotating center as a point within the whole cross section of rotating shaft; and so long as this point can't do the synchronous motion along with rotating shaft, the first positive frequency component of this point in the plane motion is bound to be the zero. It is here that the rotating center with the definition given is the result of averaged calculations of rotating motion of rotating shaft within a certain time section. Therefore, the locus of rotating center on the shaft can be variable, being closely correlated with sample length of the observed and measured rotating motion single. Accordingly, the stronger the periodicity of rotating shaft error motion is, the smaller the variation in the locus of rotating center on the shaft is.

4. Analysis of Dip Angle Or Incidence Error Motion

If the rotating accuracy of the main shaft of machine tool is analyzed completely, the inspection and measurement of radial error is incomplete, so that the inspection and measurement of dip angle is an important aspect. The magnetic bearings developed by Xi'an University of Technology have the necessary conditions for the overall inspection and measurement, that is, the front and rear bearings have two base rings and four radial displacement sensors respectively and one axial displacement sensors so that the radial, axle and incidence inspections and measurements can be achieved completely.

As shown in Fig 1., let $M2$ point be the front support point of the shaft, and $M3$ point be the rear support point, $M1$ point be random point outside the shaft extending line. $M1(T)$, $M2(T)$, and $M3(T)$ represent the radial rotating motion tracks to the corresponding points (2D plane motion), and then, we can derive the following equation:

$$M1(T) = M2(T) + a/b[M2(T) - M3(T)] \quad (5)$$

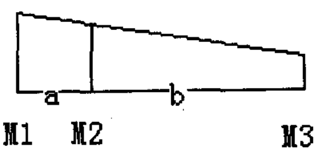

FIGURE 1: The Sketch for Analysis of Incidence Error Motion

Excluding + 1 order frequency component in $M1(T)$, the incidence error motion can be obtained. Because many kinds of manufacture tools are always fixed in the position outside the shaft extending line, it is more important to analyze the motion $M1(T)$.

5. Measurement and Analysis of Rotating Error Motion of Magnetic Rotating Shaft

In this paper, the prototype with magnetic main shaft developed by Xi'an University of Technology is served as the measuring system, whose main parameters are: the main shaft length, L = 110 mm, Shaft diameter, D = 30 mm, and one step inherent frequency is 2.4KHZ. The sensitivity of displacement-type sensor, S = 14 micro-volt/ micron; and the test instrument is NE1200 frequency spectrum analyzer made by Japanese SEEI Firm. Stable rotation speed of magnetic rotator is 18750 rpm, i.e. 312.5HZ. Sampling cycle is 50 micro second; and sampling data can be 64 circles. The 4 displacement types of vortex sensors are arranged at X and Y directions being vertical with each other in the front and rear supporting points of the shaft, being used to measure the radial rotating error motion of the rotators. The 2D frequency is made of time-regional signals on X and Y directions measured in order to obtain the radial error motion frequency spectrum at the front and rear supports.

We can also obtain the tracks of both radial motion $M2(T)$ of shaft at the front support and radial motion $M3(T)$ of rotor at the rear support of rotating shaft. The calculation equation of the incidence error motion can be used to draw the tracks of rotor external terminal $M1(T)$ as shown in Fig.2.

In this experiment, the structure of magnetic bearings can decide

$$M1(T) = M2(T) + \frac{1}{10}[M2(T) - M3(T) .$$

After the first positive order frequency in $M1(T)$ is eliminated, $M1(T)$ is recalculated with error motion track at that point obtained, as expressed in Fig 3.

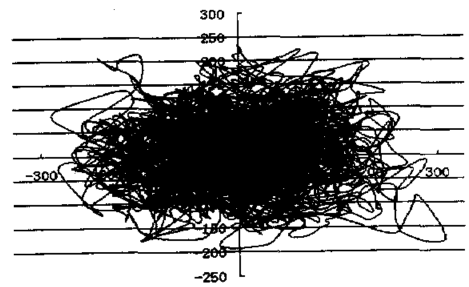

FIGURE 2: M1(T) Radial Motion Track (Coordinates Unit: Micro-Volt)

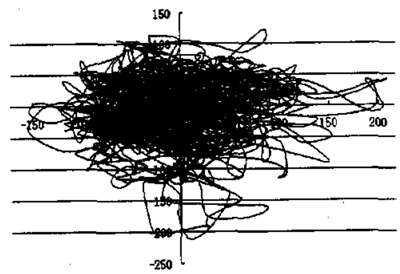

FIGURE 3: After the first positive order frequency in $M1(T)$ eliminated (Coordinates Unit: Micro-Volt).

As viewed from the variations between the Fig 2 and Fig 3, when the first positive order frequency component in radial rotating motion is eliminated, the rotating scope expressed by $M1(T)$ becomes apparently small. Therefore, the first positive order frequency component in the real shaft rotating motion (shaft deviation motion) can have a great impact upon radial motion.

6. Reasons to Cause Rotor Deviation Motion

The main reasons to cause the rotating machinery vibration are the imbalance of main shaft itself. The degrees of uneven distribution of rotor qualities as well as the balance patterns are of great randomization. However, as viewed from the vibration patterns, the shaft imbalance can be divided into two groups:

(1) Static Imbalance

Centrifugal inertial force system caused by rotor with imbalance qualities can be simplified into a resultant force through the mass center, but the resultant force moment for the mass center is the zero, i.e. there is only imbalance inertial force but no inertial torque for the mass center. It is just at this time that the dynamic pressure to which the rotating bearings are subject is in the same direction. As far as the magnetic shaft developed by us are concerned, they are tested and monitored in their manufacturing process so that the inertial force is basically eliminated, as a result of which they have no more effects upon rotating accuracy.

(2) Dynamic Imbalance of Rigid Rotors

It refers to the fact that the resultant force torque of centrifugal inertial force system caused by the rigid rotor with imbalance qualities is unequal to the zero, i.e. the pure torque is imbalance. When the rigid rotors are in the state of dynamic imbalance, the bearings on the both ends in the rotating process are subject to the dynamic pressure with the opposite directions or to unequal pressures[3]. The first step inherent frequency of magnetic main shaft in the experiment is 2.4KHZ, but working frequency of magnetic main shaft in practice is far lower than 2.4KHZ so that it can be considered that the magnetic main shaft can be converted into the rigid rotor. Accordingly, the effect of dynamic imbalance of rigid rotor itself upon the laws of rotating accuracy can be applied to the magnetic main shaft in this experiment. At present, the experimental accuracy concerning dynamic balance of rigid rotors can not be unlimitedly improved because of being subject to constraints of conditions of various kinds so that the effect of dynamic balance of magnetic rotors can not reach the ideal requirements. And hence, the typical frequency spectrum data from the front and rear measuring points are sampled to make the table.

TABLE 1: Frequency Spectrum Data

A	B	C	D
0	38.31	30.49	27.11
50	32.73	21.00	49.71
100	34.31	32.42	106.95
313	38.58	40.00	190.88
-50	0.00	0.00	52.95
-100	30	31.0	211.57

A: Frequency: (HZ)
B: Amplitude of front measuring point: (DB)
C: Amplitude of rear measuring point: (DB)
D: Difference Values of phase angle of the front and rear measuring points:(Degree)

It can be known from the data analysis that:

(1) OHZ represents the direct current component of error motion, which can reflect the deflection degrees of circle center of enveloping circle of error motion tracks.

(2) Through monitoring the sensors, the amplitudes in the case of 50HZ, 100HZ, -50HZ, and −100HZ are mainly caused by electric resources interference introduced by sensors.

(3) At +313HZ,the frequency is the same as the rotating frequency of the shaft so that the amplitude at this point represents the synchronous deflect center motion of the shaft.

(4) At +313HZ, the difference between the phases (angles) of the front and rear supporting points are 190.88 degrees. If in the case of pure dynamic imbalance, that angle should be 180 degrees, but there are other affecting factors so that it is impossible to remain in the agreement

with theoretical values. The differences of phase angles obtained from the experiments can illustrate that the amplitude at 313Hz is caused by the dynamic imbalance of the shaft itself.

7. Conclusions

To summarize the above analysis, we hold that the first positive order harmonics in the rotating shaft is the main reason affecting the rotating accuracy of rotating main shaft , while the main cause of the first positive order harmonics is induced the dynamic imbalance of rigid main shaft itself. Also, there exist the uneven distributions in qualities of the rotors. When the rotor deflection center switchovers in the radial cross sections at the front and rear supporting, the deflection center position is 180 degrees in the difference, which can cause a great radial deflection center in position M1. After the analysis of data of multi-groups measured under the rotating shaft working at a high speed, it can be found that the dynamic imbalance of the rotating shaft is the main factors affecting the rotating accuracy of the rotating shaft.

8. Looking forward to the Method of Improving Rotating Accuracy

Through the above analysis, we suggest the control scheme of feed-forward. The feed-forward control is the control method to carry out the correction in accordance with the direct disturbance momentum. Once the interference affecting the controlled parameters adverts, the feed-forward controller can control over the variations in the disturbance momentum directly in terms of the magnitudes and directions of measured disturbances whereby offsetting the effect of disturbance momentum upon the controlled and measured parameters.

The disturbances we want to control are the +1 first frequency component caused by dynamic imbalance of the main shaft, representing the deflection motion (to be detectable) with stronger periodicity. After the previous several +1 first components are detected, the periodical motion with deflection center can be re-controlled.

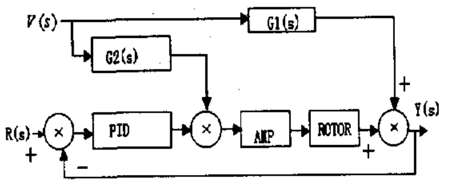

FIGURE 4: The Feed-Forward Control Scheme

G1 (s) ---- Interference Signal Transmission Functions

G2(s)-----Feed-Forward Controller Transmission Functions

V(s) ------ Interference quantity

REFERENCES

1. Zhang HuaRong , Research on Rotating Accuracy Theory , The . publication of ICMMA 1989 (3)

2. Yang shizhong et al., Testing and Evaluation of Rotating Accuracy, The Journal of Shaanxi Institute of Mechanical Engineering 1986 (3)

3. Cu yuhu: << Mechanical Principle>> Beijing, High Education Publishing House ,1979.5.