Study On the Thermal Characteristics of a Magnetically Suspended High Speed Machine Tool Spindle

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ABSTRACT

In this paper, the thermal characteristics of a magnetically suspended high-speed milling spindle are examined numerically and experimentally. The investigated spindle system includes built-in AC induction motor with power 5.5 kW and maximum speed 70,000rpm, HSK-32C tool holder and 5-axis active magnetic bearing system. The temperature distribution is analyzed numerically with finite element method considering heat generation due to power losses in magnetic bearings and built-in motor. The heat generation rates and heat transfer coefficients in spindle elements are calculated from some empirical equations. By experimental analysis, the temperature at the spindle system includes stator of magnetic bearings and motor and rotor are measured in various rotational speed conditions and cooling conditions.

INTRODUCTION

The demand of high speed machining is increasing because the high-speed cutting provides high efficiency of process, short process time, improved metal removal capacity and better surface finish. Active magnetic bearings (AMB) are very advantageous for high-speed spindle system because there is no contact between rotors and stators, so they allow much higher surface speed than conventional rolling bearings. So far, the magnetic bearings are applied in high speed milling and internal grinding spindles with the speed range 30,000 to 80,000 rpm and spindle diameter 25 to 100 mm.

The accuracy of spindle is a significant issue for the machine tool application because it directly affects the precision of machined shape and surface roughness of the work piece. The active control feature of magnetic bearing system provides many methods to increase accuracy such as adjusting dynamic properties, vibration rejection and rotational error control methods. For guaranteeing consistency of the accuracy not only the dynamic characteristics but also the thermal characteristics such as thermal displacement must be considered and examined.

Although the magnetic bearing system has very low friction loss in the bearing surface, the power losses caused from motor and magnetic bearing rotors and air friction cannot be neglected in high rotating speed. For the stator parts, dissipating heat is rather easier than shaft because some active cooling methods like circulating coolant are available. But for the shaft, because heat transfer rate is not high as conduction in stating parts, the generated heat will be accumulated in the shaft if the heat is not dissipated enough. This can cause thermal displacement of the shaft, which can be a source of machining error, and in worst case, elongate the shaft enough to touch the auxiliary bearing bringing system instability eventually.

There are many practical research works to model and analyze thermal behavior of high-speed machine tool spindle with rolling bearings. Bossmanns [2] characterized heat sources and transfer using empirical equations and built power flow model for high speed milling spindle with internal motor. Also other transfer function based model and FEM were used for thermal analysis of machine tool spindles. [3,4] Saari[5] analyzed high speed rotating systems with induction motor, including a compressor and a pump with magnetic bearings, by use of thermal network model. For the losses in magnetic bearings, many works including Kasarda's [6] have been done for analytic modeling and verified by run-down experiments. Stephens[7] used temperature measurement of test rotor system suspended by ball bearings to analyze core and windage losses in magnetic bearings and characterized with simplified model using data reduction algorithm. The results of these researches can be used for modeling magnetically suspended milling spindle system.

In this paper, the thermal characteristics of a magnetically suspended high-speed milling spindle are examined numerically and experimentally. The thermal model of the spindle is built as 2-D axisymmeric FEM model to investigate temperature distribution and analyzed for some operating conditions. Through the experimental analysis measuring temperature of manufactured spindle, the thermal model is examined its validity.

MACHINE TOOL SPINDLE WITH AMB

In FIGURE 1, the high-speed milling spindle with built-in induction motor (5.5kW, 70,000 rpm max.) and 5-axis magnetic bearing system is shown. This spindle has HSK-32C tooling system in front end, and the outer diameter of rotors of radial magnetic bearings and motor is 49.6 mm.

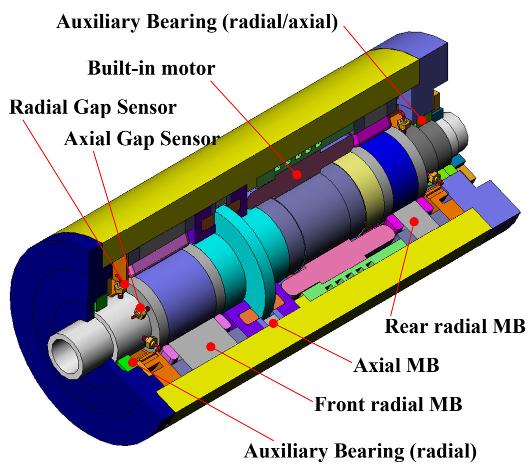


FIGURE 1: Magnetically suspended milling spindle

The radial magnetic bearings have 8-pole heteropolar configuration and controlled by digital controller based on direct feedback PID controller with conventional bias-current linearization. The bias currents are 3.5A for both front and rear radial magnetic bearing generating 0.75T of bias flux density. 4 eddy current gap sensors (0.1 μm resolution, AEC5706) are used for detecting radial rotor position, and a feedforward controller with LMS algorithm is applied to eliminate electrical runouts of radial sensors during rotation. For axial direction, two gap sensors (0.2 μm resolution, AEC5503) are located front side for minimizing displacement at spindle front end, and the axial magnetic bearing is between front radial bearing and motor.

Ten PWM amplifiers (8 A, 20kHz) drive all magnetic bearing coils with bandwidth higher than 2 kHz. The natural frequencies of magnetic bearing systems are adjusted around 250Hz for both front and rear radial directions. The manufactured shaft has its first bending mode at 1650 Hz, so with notch filter to suppress resonance of flexible mode, this spindle can run up to 70,000 rpm without critical speed.

THERMAL MODELLNG OF A SPINDLE WITH MANGETIC BEATINGS

Heat Sources

The spindle system with magnetic bearings has heat sources in both stating and rotating parts. As other motorized spindles for machine tools, the internal motor is biggest source of heat, but other power losses in stators and rotors of magnetic bearings and from friction of air become significant in very high speed.

1. Losses in magnetic bearings

In magnetic bearings, there are ohmic 'I^2R' winding losses in coils of electromagnet caused by static bias current. For the spindle in this paper, total ohmic losses in electromagnets are calculated as 39W and 30W for front and rear magnetic bearing and 36W for axial bearing coil. And for the laminated cores of magnetic bearings, iron losses due to eddy current and hysteresis exist in rotors and stators. The formulas from Kasarda [6] are used to calculate for these iron losses in this paper. FIGURE 2 shows calculated core losses in the magnetic bearing rotors and stators. The alternating hystresis loss with 10% of bias flux density is assumed for stator cores.

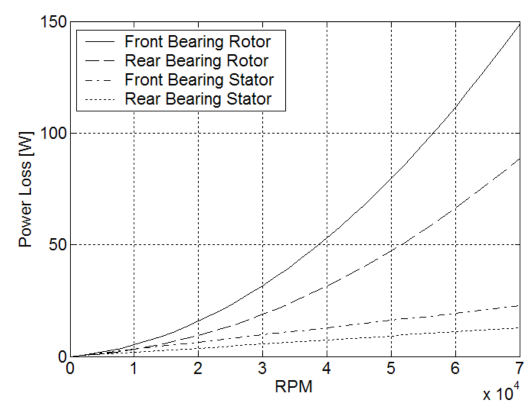


FIGURE 2: Power losses in magnetic bearing cores

2. Losses in induction motor

The internal induction motor has its I^2R winding loss and iron loss caused by eddy current and hysteresis in stator along with the core loss and winding loss in rotor. [5, 8] These losses are function of the load and rotational speed. However In this study, losses in the motor are estimated base on data from the manufacturer.

3. Windage losses in rotating parts

The losses due to air friction in rotating surface can be calculated from empirical equations. The ambient rotating surfaces, the drag loss can be calculated using expanded formula of moving flat plate as (1) using rotating speed ω, density of air ρ_g, viscosity of air ν, radius R and length L. [6]

$$P_{wa} = 0.074\rho_g L\pi R^4\omega^{2.8}\left(\frac{\nu}{2\pi R^2}\right)^{0.2} \quad \text{[W] (1)}$$

For the surfaces faced stator through narrow air gap, Saari [5] applied drag coefficients for concentric cylinder and enclosed disk. For enclosed rotating cylinder with zero axial flow, the friction loss due to rotation is,

$$P_{wc} = 0.515\frac{\rho_g L\pi R^4\omega^3}{\mathrm{Re}_\delta^{\,0.5}}\left(\frac{\delta}{R}\right)^{0.3} \quad \text{[W] (2)}$$

This formula is valid for $500 < \mathrm{Re}_\delta = \dfrac{\omega R\delta}{\nu} < 10^4$ when R is radius of rotating surface and is radial gap. For the enclosed rotating disk the power loss by drag force is calculated as (3),

$$P_{wd} = \frac{\pi\rho_g\omega^3 r_1\left(r_2^{\,5} - r_1^{\,5}\right)}{s\,\mathrm{Re}_r} \quad \text{[W] (3)}$$

for $\mathrm{Re}_r = \dfrac{\omega r^2}{\nu} < 10^4$, r_2, r_1: outer / inner radius.

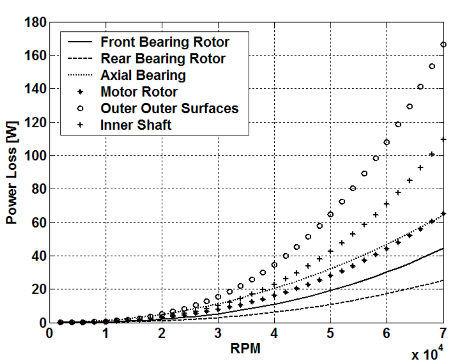


FIGURE 3: Windage losses in the rotating parts

Equation (1) is used for free rotating surface such as shaft end and inner surface, (2) for radial magnetic bearing and motor rotor, and (3) for axial magnetic bearing rotor. FIGURE 3 shows calculated value of windage losses in each rotating part. Total windage loss in the shaft can be obtained by summing all power losses, which is 470W at 70,000 rpm.

Heat Transfer Coefficients

The heat transfer rate for unit area is described as equation (4) in interfacing surfaces. For spindle system modeled here, every heat transfer is assumed occurring in surfaces. By using empirical formulas, heat transfer coefficients, h_i are calculated.

$$q_i = h_i A_i \Delta T_i \quad (4)$$

The some of heat generated in the stators is transferred to spindle housing by conduction, and other part is transferred to surrounding air by convection. The heat transferred by conduction can be removed by active cooling of spindle housing by circulating coolant. In this manner, this spindle has a cooling jacket in the middle of housing to dissipate heat from the motor stator. Cooling oil is circulating through the narrow spiral duct, which has cross section of 5 x 4 mm^2. The convection coefficients can be calculated by empirical equation for turbulent flow [2, 3]. In this paper, h_c= 2,000 W/m^{2}K was used.

For rotating surfaces, such as front end and inner surface of shaft, equation (5) from [2] is used, and this was also applied for rotating surfaces of magnetic bearings.

$$h_r \approx 9.7 + 5.33(\omega r)^{0.8} \quad \text{[W/m}^2\text{K]} \quad (5)$$

For outer surfaces of spindle and inner static surfaces such as coils, free convection condition is assumed with free convection coefficient, 9.5 W/m^{2}K.

It should be noted that these coefficients of heat transfer are calculated from results of based on experiments done before 1970s, so they are need to be corrected for exact solution by comparing with experimental results.

If the generated heat is not removed enough, the heat in the rotor will be accumulated and cause resultant elongation of the shaft. For the spindle in this study, thermal displacement is expected 0.1 mm if mean temperature of the shaft is 29℃ higher than surrounding temperature. This thermal elongation is very important because it causes machining error and system instability if it is big enough to touch the auxiliary bearing.

NUMERICAL ANALYSIS

Many methods are used for thermal model of spindle system, such as thermal network, power flow, transfer function and finite element model.

In this study, the finite element model was used because it's easy to visualize results such as temperature distribution. The commercial FEM package, ANSYS 7.1 is used. To reduce computational efforts, the spindle model was modeled as an axisymmetric 2-D model as FIGURE 4. Because the magnetic bearing and motor stators are not axisymmetric shapes, the modified thermal conductivities and specific heats are used for these parts. [6]

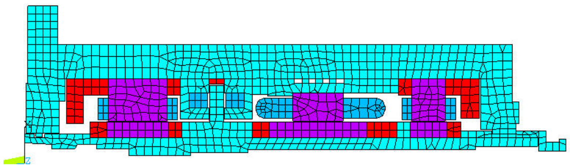
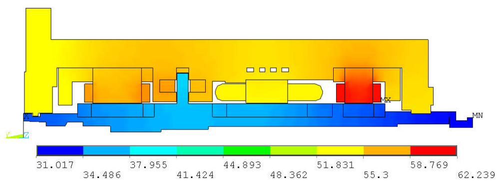
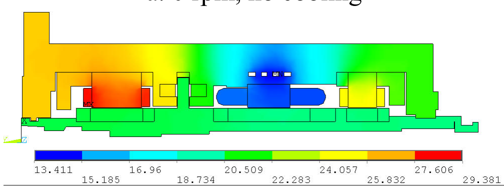


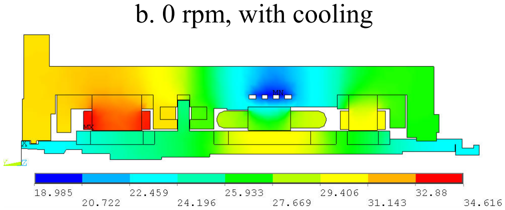
FIGURE 4: Numerical analysis model (FE)



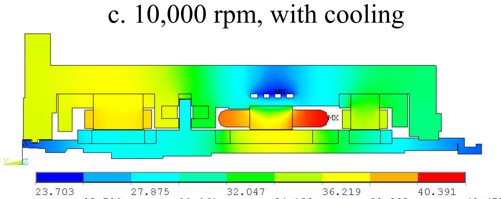
a. 0 rpm, no cooling



b. 0 rpm, with cooling



c. 10,000 rpm, with cooling



d. 20,000 rpm, with cooling

FIGURE 5: Temperature distribution at 0 rpm

The heat sources and heat transfer coefficients were applied referring FIGURE 2 ~ 3 and motor data. Four cases of heating and cooling as bellows were analyzed to examine validity of this model.

1. no rotation without cooling: only heat generations in bearing coils
2. no rotation with cooling: heat generations in bearing coils and cooling at cooling jacket with 10℃ oil
3. 10,000rpm with cooling: heat generations in magnetic bearings, motor and rotating surfaces and cooling with 10℃ oil
4. 20,000rpm with cooling; loss is lager than 3rd case and cooling with 10℃ oil

In the FIGURE 5, the temperature distributions at case 1 to 4 are shown. As the cooling of motor stator removed heat from bearing coils, temperature of rotor dropped to 21℃ from 31℃. The front magnetic bearing core was shown to be hottest in 10,000rpm because the cooling jacket located far after axial magnetic bearings. When speed was increased to 20,000 rpm, the temperature at stator of motor became higher due to increase of motor loss.

EXPERIMENTAL RESULTS

The temperature of spindle was measured in 4 points in stator with thermocouples (T-type), and rotor at front end by infrared temperature sensor (OS36, Omega).

First, the spindle was tested in stationary condition without cooling. In this case, it can be regarded as the heat is generated only in the coils of magnetic bearings, and the iron loss of stators rotors of magnetic bearings can be neglected as case 1 of numerical analysis. As the result shown in FIGRE 7, the temperature of magnetic bearings was not saturated until 1 hour, and temperature of rotor was also increasing same as motor stator. It is seemed that the heating of air trapped in the spindle made this rise of temperature. When oil of 10℃ was flowed at cooling jacket, the temperature was stabilized within 15 minutes at 30℃ at front bearing, and the temperature of rotor was also dropped to 21℃.

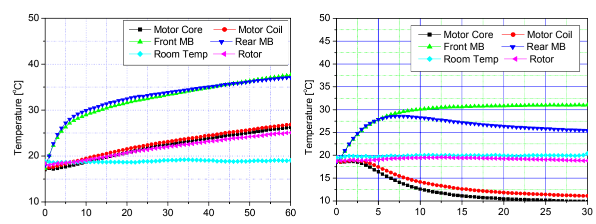
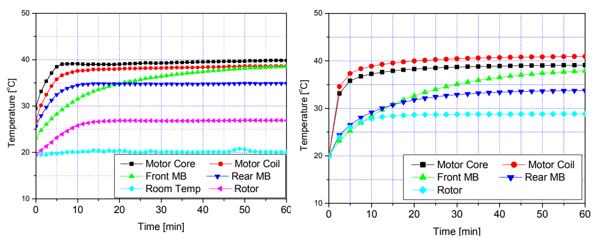


FIGURE 7: Temperature measured at no rotation

The measured temperature at 20,000 rpm with cooling is compared with calculated result using FEM in FIGURE 8. With rotation, the temperature was raised at higher rotational speed especially in motor. Transient response of experiment was faster than FEM result because the applied convection coefficient and specific heat was not exactly modeled.



a. measured b. FEM

FIGURE 8: Temperature at 20,000 rpm

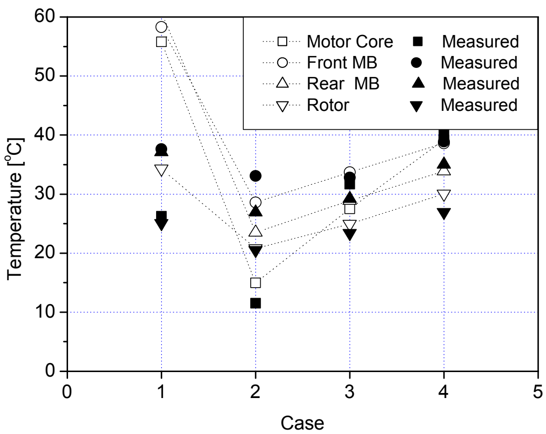


FIGURE 9: Temperature measured and calculated by numerical analysis
(measured values of case 2 is at 60 minutes after)

From the FIGURE 9, which shows measured and calculated temperatures of 4 cases, it can be noted that the FEM model is valid to estimate thermal characteristics of the spindle. The rotor temperature at 20,000rpm was measured 27℃ increased 3.5℃ from that of 10,000rpm..

Using this FEM model, the saturated shaft temperature at 70,000 is expected about 70℃ assuming 210W of power loss at rotor and 640W at stator of motor. This temperature is high enough to elongate the shaft 0.1 mm.

DISCUSSION AND FUTURE WORKS

In this paper, the thermal model of a milling spindle system with 5-axis active magnetic bearing was developed considering heat sources and heat transfers. Using numerical model including heat generation in the rotors and stators of magnetic bearings and induction motor, temperature distribution was estimated in some operating conditions and compared with measured from experiment. The experimental result showed that the 2-D axisymmetric finite element model built in this study was valid to demonstrate thermal characterizes of the spindle system.

From the results of temperature rises in the shaft, it is noted that extra cooling of rotor is necessary to minimize thermal displacement of rotor, or to prevent heating up the rotor in high speed up to 70,000 rpm. Other cooling methods such as cooling of whole housing and blowing cold air to inside spindle and various ways to reduce losses in magnetic bearing system in control algorithm and deign will be considered to apply, and their effects will be verified comparing with thermal model. Along with the efforts reducing heat in the shaft, more experimental works of machine tools spindle with magnetic bearing system will be performed to build more exact thermal model.

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