Non-contact measurement of dynamic stiffness of rotating spindle

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Abstract

To investigate the dynamic stiffness of the rotating spindle, a non-contact loading method is applied. A magnetic loading device is used to provide swept-sine-wave load, while the spindle displacement is measured with eddy-current-type sensors. Frequency responses of the compliance are then measured, while spindle speeds and warm-up conditions are designed to regulate the temperature status in the spindle. The spindle temperature effects on the natural frequency and peak value are discussed based on the measurement results.

1. Introduction

Dynamic stiffness of the machine tool spindle is one of the important factors for high performance machining as it influences critical cutting conditions. Although there have been great advances in chatter suppression technology [1-2], the chatter prediction still suffers from the uncertainty in dynamic characteristics of the machining system. Regarding the uncertainty issues, the speed effect on spindle stiffness is investigated in ref. [3]. In the spindle system, heat generation and thermal expansion cause the preload changes in support bearings [4], which affect the load-displacement relationships and natural frequency. Moreover, there are other factors affecting the natural frequency such as centrifugal and/or gyroscopic forces on the components. Impact hammer test has been widely used to measure the dynamic stiffness [5]. However, it is difficult to impose constant impact force on a running spindle. For this reason, non-contact measurement methods have been applied using active magnetic bearings [6-9].

In this research, the uncertainty of the dynamic stiffness of the rotating spindle is investigated by a non-contact loading method. A magnetic loading device [8] is used to provide swept-sine-wave load. The spindle displacements are measured with eddy-current-type sensors. The frequency response functions (FRFs) of the compliance are obtained from the measured load and displacement. The spindle temperature condition is changed to investigate the effects on the natural frequency and the peak value in the compliance.

2. Measurement method

The magnetic loading device is used to apply dynamic force to the spindle. Figure 1 shows the configuration of the measurement system. The magnetic loading device is located on the machine table, which attract a dummy tool attached to the spindle head. The dummy tool is integrated with a tool holder. The loading device has two coil sets to attract both sides of the dummy tool. However, currently, only one set is used to provide the force. This is because it is difficult to keep proper gaps at both sides between the coil and the dummy tool. The coil current is supplied with an amplifier. Either voltage or current command can be sent from a PC to the amplifier through a Digital to Analog (D/A) converter. To measure the attractive force, a tool dynamometer is installed between the machine table and the magnet loader. The attractive force measured by the dynamometer and the coil current monitored in the current amplifier are captured by the PC through an Analog to Digital (A/D) converter. The relationship between the coil current and the attractive force is obtained in a static
loading test. The current command is decided by referring this relationship.

When the spindle rotates and/or the alternating current is given to coil windings, eddy currents are generated mainly on the targeted surface of the dummy tool. In order to reduce the eddy currents, grooves are engraved on the dummy tool surface. The geometry of groove is optimally designed by using electromagnetic finite element simulation (JMAG by JSOL corporation) [9].

The spindle displacement is measured in two directions by using two eddy current displacement sensors. One is in the major force direction, and the other in its orthogonal direction. The major force direction is defined by the static loading test.

3. Estimation of frequency response function

To evaluate the generated force and the spindle's dynamic stiffness, a multitasking machine is used for experiments. The specifications of the measured spindle are shown in Table 1.

| Specification            | Value                  |
|--------------------------|------------------------|
| Maximum speed            | 12000 (min⁻¹)          |
| Tool interface           | HSK A-63               |
| Preload type             | Constant position preload |
| Front bearing            | Angular contact ball bearing DB |
| Rear bearing             | Cylindrical roller bearing N type |

The loading test for non-rotating spindle is carried out to investigate the relationship between the coil current and attractive force. Alternating current with a DC offset is supplied to the coils with an amplifier. Figure 2 shows an example of measured coil current. The amplitude of the coil current is changed with its frequency due to the inductance. The command voltage is modified so that the amplitude would not decrease with the increase of frequency. The frequency is changed from 0 Hz to 1000 Hz. The entire frequency range is divided into ten ranges. In the each range, the frequency is continuously swept for six seconds. Figure 3 shows measured forces in several sweep intervals with the DC components. \( F_x \) represents the component in the major direction, and \( F_y \) in the orthogonal direction. From the ten sets of the measured data, the FRF from the measured force to the spindle displacement in the major-force direction is estimated. As can be seen in Figs. 3 (a)-(d), the change in the attractive force is significantly dependent on the frequency range. The variation in the force's amplitude is large especially in the lower frequency ranges.

Figure 4 shows the FRF from the measured force to spindle displacement. In Fig. 4, a significant peak is observed at 780 Hz, which has been observed to be the natural frequency of the spindle. Other local peaks and phase shifts are also observed, which implies that the machine dynamics is involved in the force measurement.

Fig. 1. Measurement setup with magnet loader.

Fig. 2. (a) Monitored coil current; (b) Enlargement.

Fig. 3. Attractive force in frequency range of (a) 100-200 Hz; (b) 300-400 Hz; (c) 600-700 Hz; (d) 900-1000 Hz

Fig. 4. Measured FRF from force to spindle displacement
To clarify the influence, an impact hammer test was carried out. The housing of the loader, near the loading point, is hit by an impact hammer, and the FRF from the impact force to the measured force by the dynamometer is evaluated as shown in Fig. 5. The FRF in Fig. 5 shows several natural frequencies which are caused by the machine dynamics. To estimate the real attractive force, the FRF in Fig. 4 is filtered with the FRF in Fig. 5, which results in the FRF shown in Fig. 6. The impact hammer test was carried out on the spindle independently and the measured FRF is also shown in the figure. Although the FRF in the magnet load test includes several local resonances, the entire profile indicates that the 1st mode natural frequency of the spindle is dominant. This agrees with the impact test result. It was also found that damping and frequency are different between the two tests. The difference may come from the force pattern (impulse or harmonic) and the energy scale.

4. Uncertainties in spindle dynamics

4.1. Test procedure

The FRFs of the rotating spindle are measured to investigate the thermal effect on the spindle dynamics. The temperature was measured with a thermocouple installed near the outer race of the front bearing. The test procedure is as follows: after short warm up for safety, the spindle was rotated at 12000 min\(^{-1}\) to increase the temperature. When the temperature was increased by 1 K, a FRF was measured by the same way shown in the section 3. In the FRF measurement, the spindle speed in the FRF measurement was set at 60 min\(^{-1}\). The frequency range was set from 700-900 Hz, where the 1st mode natural frequency was included. After the saturation of the temperature, the spindle rotation was stopped. Then, when the temperature was decreased by 1 K, the FRF measurement was carried out until the temperature becomes stable again. Note that the spindle speed in the FRF measurement was also set at 60 min\(^{-1}\).

4.2. Test result

Figures 7(a)-(d) show the FRFs measured while the spindle temperature was increasing and saturated. It can be seen that the natural frequency increases with the increase of temperature. The peak value increases first in Figs. (b) and (c), and then decreases at the saturated temperature in Fig (d).

Figure 8 shows the relationship between the spindle temperature and the natural frequency. The entire profile shows hysteresis characteristics. This is because the natural frequency still increases at the beginning of the temperature decrease. This implies that the bearing preload is increased even after the spindle stops. The inner race must be subject to higher temperature while the outer race is cooled down by the cooling circuit, which may cause the preload increase.

Figure 9 shows the relationship between the temperature and the peak value. The peak value starts to increase with the temperature increase, and then decreases as the temperature reaches saturation. The peak value decreases slightly with the temperature decrease, and then increases again. The hysteresis at the reversal of the temperature implies the preload increase. The decrease of the peak value indicates the increase of the damping. The preload caused by the temperature difference in the bearing system may also influence the damping.
Fig. 8. Relation between temperature and natural frequency

Fig. 9. Relation between temperature and peak value

5. Conclusions

To investigate uncertain behavior in the dynamic stiffness of a rotating spindle, a non-contact measurement method was applied. The natural frequency of the first mode of the spindle and the peak value were analyzed with the temperature measured at the spindle housing. The measurement test shows that the natural frequency and the damping increase as the spindle temperature increases to the saturation condition.

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