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Evaluation of dynamic stiffness of machine tool spindle by non-contact excitation tests

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ABSTRACT

This paper presents a non-contact excitation method for evaluating the dynamic stiffness of a rotating spindle. The spindle response to an excitation force is measured, and frequency response functions (FRFs) are obtained. Based on the measured FRFs, dynamic uncertainty and its effect on cutting stability are investigated. Regenerative forces are generated using displacement feedback with a time lag element, and a closed-loop excitation test is executed automatically. The stability map obtained from the closed-loop test and the stability charts calculated from the FRFs are compared, and the uncertainty of the spindle dynamics during operations is clarified.

1. Introduction

Machine tool spindles must rotate cutting tools precisely and provide sufficient energy for material removal [1]. For high-productivity machining, spindles should be designed to have sufficient stiffness to resist cutting forces. The basic components of a spindle, such as its rotating shaft, support bearings, and contact parts, have elasticity, and their contribution to stiffness at the tool point is complex. State-of-art design methods consider the nonlinearity of preloaded bearings [2], interface dynamics [3], and the thermal distribution of the components [4,5]. Measurement methods are important in the evaluation of the designed stiffness and its achievement in the manufacturing process.

Rantatalo et al. investigated the vibration characteristics of rotating spindles, including the rotor dynamics, using non-contact magnetic excitation [6]. Abele et al. investigated the speed dependency of the natural frequency and damping of a hybrid spindle using the active magnetic bearing (AMB) component [7]. Ozturk et al. demonstrated the effect of spindle bearing preload on dynamic stiffness and chatter stability [8]. Abele and Fiedler proposed the state-space identification of the frequency response functions (FRFs) of spindle compliance during milling operations [9]. Suzuki et al. investigated a different identification method based on chatter information [10]. Tobias discussed the nonlinearity of spindle stiffness and its effect on chatter [11]. The Author investigated the influence of spindle temperature on stiffness using a non-contact measurement method and a magnet loading device [12].

The influence of machine dynamics on the occurrence of vibration has been analyzed using measured and calculated machine dynamics. Regenerative chatter analysis in the frequency domain has been conducted using FRFs and cutting models [13]. In general, the analytical predictions are confirmed by the results of cutting tests. However, cutting processes have various effects, such as the multiple regenerative effect [14,15], process damping [16], multiple frequencies, and multiple delay [17]. These effects make it difficult to isolate the dynamic behavior of the spindle for analysis.

In this research, the uncertainty of the dynamic stiffness of a rotating spindle was investigated using a non-contact excitation method. Two excitation tests were combined. One test used a conventional open-loop excitation, and the other used a closed-loop excitation. In the closed-loop test, the regenerative force was generated on the basis of the monitored spindle displacement. Chatter behavior was investigated in conjunction with the uncertainty of the FRFs measured in the open-loop test.

2. Measurement system

2.1. Configuration of measurement setup

Fig. 1 shows an overview of the test device used. The device consists of a magnet loader, a dummy tool (a load target), measurement sensors, and a personal computer (PC) to control the process. The loader has coils that generate magnetic forces, which are controlled by coil currents. In this research, one coil set was used to emulate the cutting forces of side milling or boring. The coil current was supplied by an amplifier, to which a current command was sent from the PC through a digital-to-analog (D/A) converter. Displacement sensors were installed in the head stock of the spindle to measure the spindle displacement at the holder. The displacement signal was filtered using a high-pass filter and captured by an analog-to-digital (A/D) converter. This high-pass filter filters out the machine vibration other than the spindle vibrations. A tool dynamometer was installed between the machine table and the magnet loader to measure the attractive force. The force signal was also captured by the A/D converter. Using

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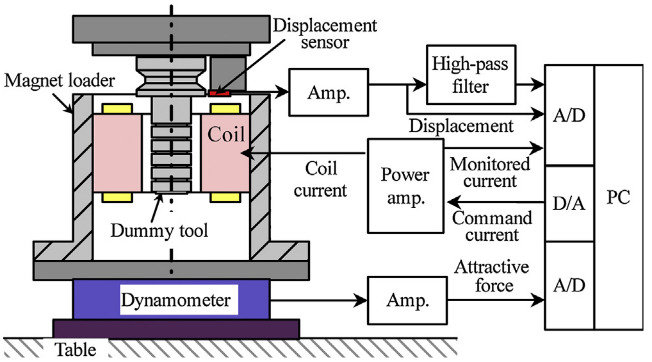

Fig. 1. Configuration of measurement setup.

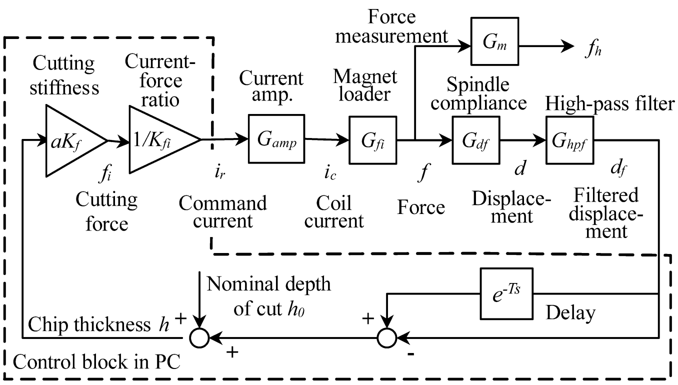

Fig. 2. Block diagram of excitation system.

this device, the spindle stiffness of a multitasking machine was measured. The spindle's key specifications were as follows: the maximum speed was 12,000 min^{-1}, the preload type was constant-position preload, the front bearing employed angular contact bearing, the rear bearing employed cylindrical roller bearing, and the diameter of the front bearing was 60 mm.

2.2. Control block and excitation test

Fig. 2 shows the block diagram of the excitation force generation. The blocks from i_r to d_f correspond to the analog components, where i_r is the command current sent to the current amplifier, i_c is the coil current of the magnet loader, f is the magnet force, f_h is the measured force, d is the spindle displacement measured at the toll holder, and d_f is the filtered displacement. The transfer functions of the current amplifier, magnet loader, spindle compliance, high-pass filter, and force measurement are G_{amp}, G_{fi}, G_{df}, G_{hpf}, and G_m, respectively. The cut-off frequency of the high-pass filter was set at 300 Hz because structural vibrations of the machine occur at frequencies below this threshold.

The blocks from d_f to i_r are in the digital control system, which represents the cutting force feedback. To simulate regenerative vibration, a time lag element with a spindle rotation period T was inserted in parallel to the feedback loop. In Fig. 2, h_0 is the nominal cut depth, K_f is the specific cutting force, and a is the cut width. Multiplication of a and K_f yields the cutting stiffness. The term f_i refers to the emulated cutting force, and the term K_{ft} refers to the transferring gain from force to coil current.

In the open-loop test, the feedback pass was cut, and a swept-sine signal was given directly to the command current i_r. From the measured i_r and d_f, an FRF was estimated. This FRF is the serial product of G_{amp}, G_{fi}, G_{df}, and G_{hpf} and is referred to as the open-loop FRF ϕ. The force signal was also measured, and the FRF from f_h to d_f was estimated to check the open-loop FRF.

In the closed-loop test, T was set to the real rotation period of the spindle, and aK_f was set. The feedback loop was then closed to check the oscillation. All tests were performed automatically under spindle rotating conditions.

The relationship between the coil current and magnetic force is as follows:

$$f = K_i i_c^2 \tag{1}$$

where K_i is the current force coefficient, which is inversely proportional the square of the gap distance. A sine wave with an offset is given to the current as follows:

$$i_c = I_0 + I_a \sin(\omega t) \tag{2}$$

where I_0 is the current offset, I_a is the current amplitude, and ω is the excitation frequency. Substituting (2) into (1) and referring to the fundamental harmonic term, the relationship between the force and command current can be estimated as follows:

$$f_r = K_{ft} I_a \sin(\omega t) = K_{ft} i_r \tag{3}$$

where $K_{ft} = 2K_i I_0$. The nominal depth of cut determines the current offset.

3. Nonlinearity in spindle stiffness

To evaluate nonlinearity in the spindle stiffness, the current offset I_0 and amplitude I_a were changed, and open-loop FRFs were measured. The spindle speed was set at 3000 min^{-1}. Using the measured data, the FRFs from f_i to d_f (G_{di}) and the FRFs from f_h to d_f (G_{df}) were estimated. The coherence in each measurement was over 0.9 except for the vicinity of the natural frequency, where the minimal coherences are 0.75 for G_{di} and 0.45 for G_{df}. This reason is that the vibration components other than the force-excitation response also increase at the resonance frequency.

Fig. 3 shows the estimated FRFs for the variable offset and constant amplitude. The natural frequencies decreased as the current (force) offset increased, which reflects the soft spring characteristic in the force–displacement relationship. G_{df} exhibits a sharper change in gain and a greater lag in phase than G_{di}. Note that the force measurement unit has a natural frequency near 300 Hz, which provides uncertainty to both the gain and the phase.

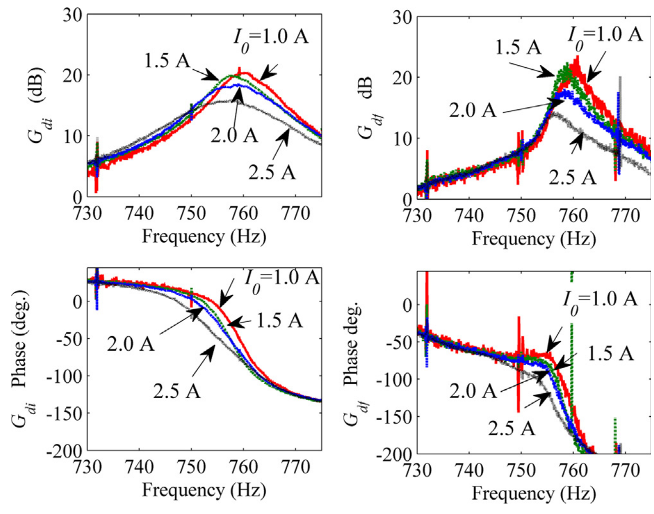

Fig. 3. Measured FRFs for variable offset and constant amplitude: I_a = 0.3 A, G_{di} is FRF from f_i to d_f, G_{df} is FRF from FRF from f_h to d_f.

Fig. 4 shows the estimated FRFs for the constant offset and variable amplitude. The change of the natural frequency could be seen in G_{df} where I_a is less than 0.6 A. However, it could not be seen for larger amplitudes, where the gain decrease could be seen in G_{di}. This indicates that the soft spring effect is weak at this force offset and the gain in the current–force relationship is decreased with the increase of the current amplitude.