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Modeling the machining stability of a vertical milling machine under the influence of the preloaded linear guide

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The prediction of machining stability is of great importance for the design of a machine tool capable of high-precision and high-speed machining. The machining performance is determined by the frequency characteristics of the machine tool structure and the dynamics of the cutting process, and can be expressed in terms of a stability lobe diagram. The aim of this study is to develop a finite element model to evaluate the dynamic characteristics and machining stability of a vertical milling system. Rolling interfaces with a contact stiffness defined by Hertz theory were used to couple the linear components and the machine structures in the finite element model. Using the model, the vibration mode that had a dominant influence on the dynamic stiffness and the machining stability was determined. The results of the finite element simulations reveal that linear guides with different preloads greatly affect the dynamic behavior and milling stability of the vertical column spindle head system. These results were validated by performing vibration and machining tests. We conclude that the proposed model can be used to accurately evaluate the dynamic performance of machine tool systems designed with various configurations and with different linear rolling components.

1. Introduction

Because of the demand for high-speed and high-precision machining, machine tools with excellent dynamic performance are being designed and manufactured. However, regenerative chatter vibration produced by the machine tool during machining has been a problem. The studies conducted by Tlusty [1–3] and Tobias [4] revealed that the chattering is caused by the dynamic interaction between the cutting tool and the workpiece during the chip generation process. They presented the basics of a dynamic cutting force model for the regenerative chatter vibration. Various analytical stability-analysis approaches were subsequently developed by Minis et al. [5,6] and Budak and Altintas [7,8] in order to quantitatively define the machining conditions for stable milling operation. The chatter stability analysis model accounts for the important geometrical parameters of the cutters, the cutting conditions, and the structural dynamic behavior of the machine tool. By using this model, the designer can recognize the importance of the structural dynamic characteristics of the machine tool and evaluate its machining performance at the design stage.

Modern machine tools are fabricated using a modular design concept in order to meet the requirements for reducing manufacturing costs and realizing multiple industrial applications. CNC milling machines are usually assembled with five modular components: a machine base, saddle, table, vertical column, and headstock with a spindle tool unit. In the spindle tool system, the feeding mechanism of the control axis is constructed in various configurations using linear guides, ball screws, and supporting bearings [9,10].

Before fabricating a prototype for performance evaluation, the designer should ensure that the design parameters yield the required machine performance. For this purpose, the finite element approach has generally been adopted for assessing the dynamic characteristics of machine tool structures because of the efficiency and reliability of the analysis based on this approach. While many researchers have concentrated on the development of individual models, the dynamic analysis model of a machine tool cannot be easily formulated as a combination of individual modules since the linear rolling components include rolling interfaces with nonlinear contact characteristics. It is well known that the ball bearing is a primary component dominating the dynamic behavior of the rotary-shaft system, and, in particular, of the spindle tool unit. Studies [11–15] have revealed that the dynamic behavior of a spindle tool system may deviate from the original behavior because the bearing stiffness may vary with changes in the preload under different operating conditions.

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At high speed operation conditions, the gyroscopic and centrifugal effect on spindle dynamic stability is profound [16–18]. Using a spindle-drawbar-bearing assembly model, Jiang and Zheng [18] found that the bearing stiffness decreased with the increasing rotational speed and hence lowered the dynamic stiffness of the spindle nose. Jiang and Mao [19] further developed a variable preload control system to suppress the gyroscopic and centrifugal effect and to stabilize the dynamic characteristics of spindle at wider speed ranges.

A linear rolling guide is commonly used in high-precision positioning systems. In the case of this component, the preload effect caused by the use of oversized rolling balls inside the ball groove affects not only the structural rigidity but also the vibration characteristics of the positioning stage [20–22]. The results of our previous study [23] indicate that feeding stages equipped with rolling guides at different preloads exhibited different vibration behavior, and this affected the dynamic stiffness of the vertical column spindle system. Jiang and Zhu [24] indicated that the joint interfacial characteristics could be affected by cutting load, which further affect the higher frequency vibration behavior of a vertical milling machine.

As revealed by the abovementioned studies, in addition to the structural stiffness of the main modularized parts, the stiffness of the interfaces between combined components is a predominant factor that influences the overall stiffness of a machine tool structure. Consequently, it is expected that the effects of preloading on the machining performance will vary as the specifications of the linear guide vary. However, to the author's knowledge, this effect has not been realized, and hence, it is worthwhile to evaluate the influence of the rolling interfaces on the machining stability of a milling machine.

The aim of this study was to investigate the machining stability of vertical milling under the influence of a linear guide modulus. A frequency–characteristic analysis is of great importance for gaining an understanding of the dynamic performance of the milling system. Therefore, a finite element model of the vertical column spindle system was first constructed with the integration of the modeling of linear rolling components. By using this model, a harmonic analysis under simulated cutting force conditions was performed in order to assess the dynamic behavior and subsequent cutting stability of the milling machine. Finally, the simulation results were validated by performing vibration tests and machining experiments on a prototype machine.

2. Construction of vertical milling machine

To examine the influence of the linear guide preload on the machining characteristics, a small-scale vertical milling machine was designed and fabricated, as shown in Fig. 1. The vertical column and the feeding stage of the spindle head were constructed using carbon steel plates. Two linear rolling guides were secured on the front plate of the column at a span of 160 mm. The feeding stage of the spindle head was mounted on the column using pairs of linear guides and was driven with a high-precision ball screw. The commercial linear guides (HG series) have four ball grooves with a circular arc profile, forming a point contact at an angle of 45° [25]. To enhance their rigidity, the sliding blocks of the linear guide can be preloaded to different levels by setting oversized steel balls within the ball grooves. The levels are low preloading (Z0, 0.01*C*), medium preloading (ZA, 0.07*C*), and heavy preloading (ZB, 0.11*C*), where *C* denotes the dynamic load rating (11.38 kN). The ball screw has a diameter of 14 mm, a lead pitch of 4 mm, and a basic dynamic load rating *C* of 4.07 kN and it is slightly preloaded to a level of 0.06*C* so as to decrease the axial backlash [26]. Further, to ensure rigidity balance with the ball screw, standardized ball-screw support units coded EK12 and EF10 were used at both ends of the screw shaft [27]. The specifications of the linear components are listed in Table 1. In addition, a high-frequency spindle unit, with a weight of 4.1 kg and a maximum speed of 36,000 rpm, was installed on the feeding stage. Fig. 1 shows the global coordinate system used in the machine model, where the *X* and *Y* axes are defined as the two horizontal axes, and the *Z*-axis is the vertical axis in the column direction. A local coordinate system was defined at the spindle head to describe the vibration mode.

3. Finite element modeling

The key point for accurate modeling of a machine tool structure is the simulation of the rolling elements within the ball

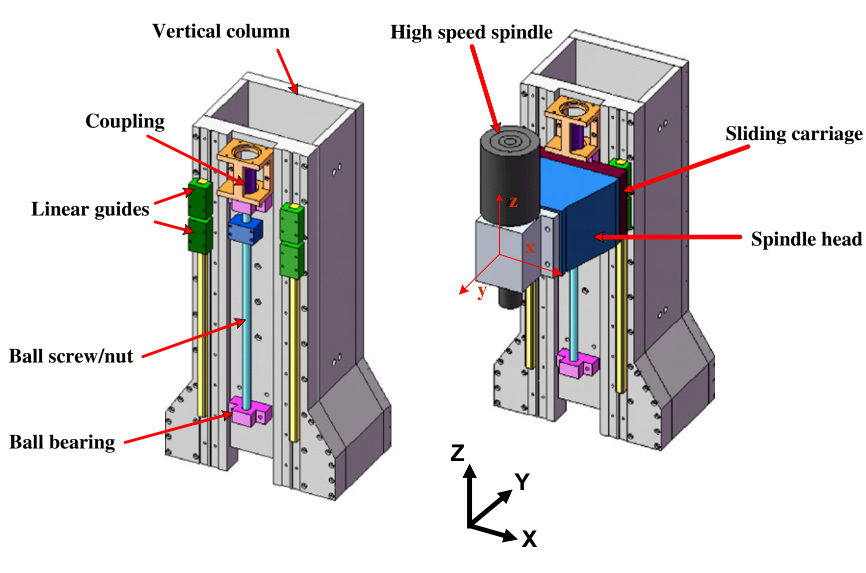

Fig. 1. Schematic construction of a vertical column milling structure with motorized spindle.

Table 1
Comparisons of the predicted natural frequencies of spindle head that was mounted on the linear guides with different preload.

Vibration mode	Finite element predicted natural frequency (Hz)		
	Low preload (Z0)	High preload (ZA)	High preload (ZB)
Pitching vibration of spindle head accompanied with bending vibration of vertical column	262	304	310
Pitching vibration of spindle head	332	379	407
Yawing vibration of spindle head	352	445	483

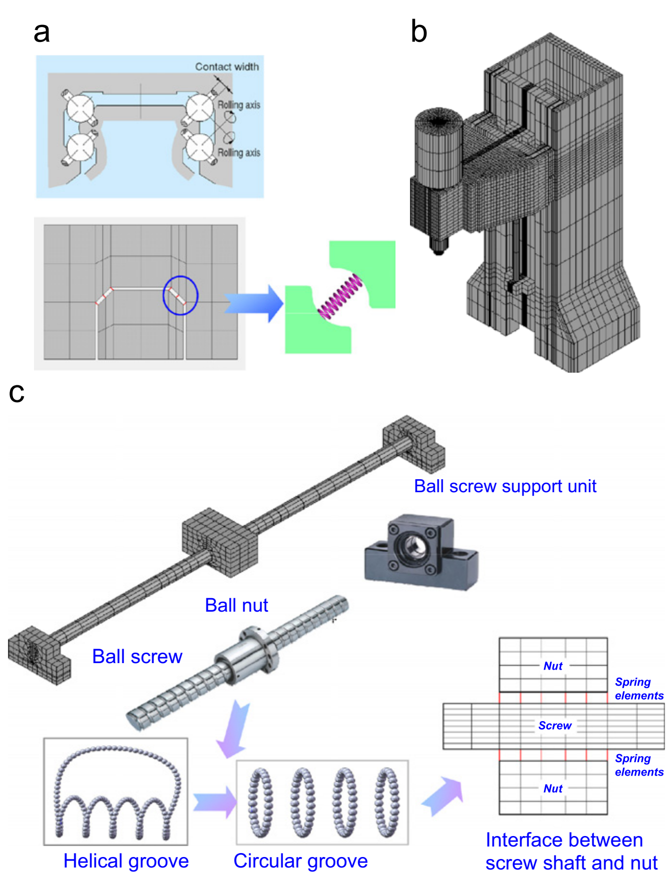

Fig. 2. Finite element modeling of the vertical column spindle system. (a) Modeling of rolling guide with spring elements at ball grooves, (b) finite element model of vertical column spindle structure and (c) simplified finite element model of ball screw-nut and bearings.

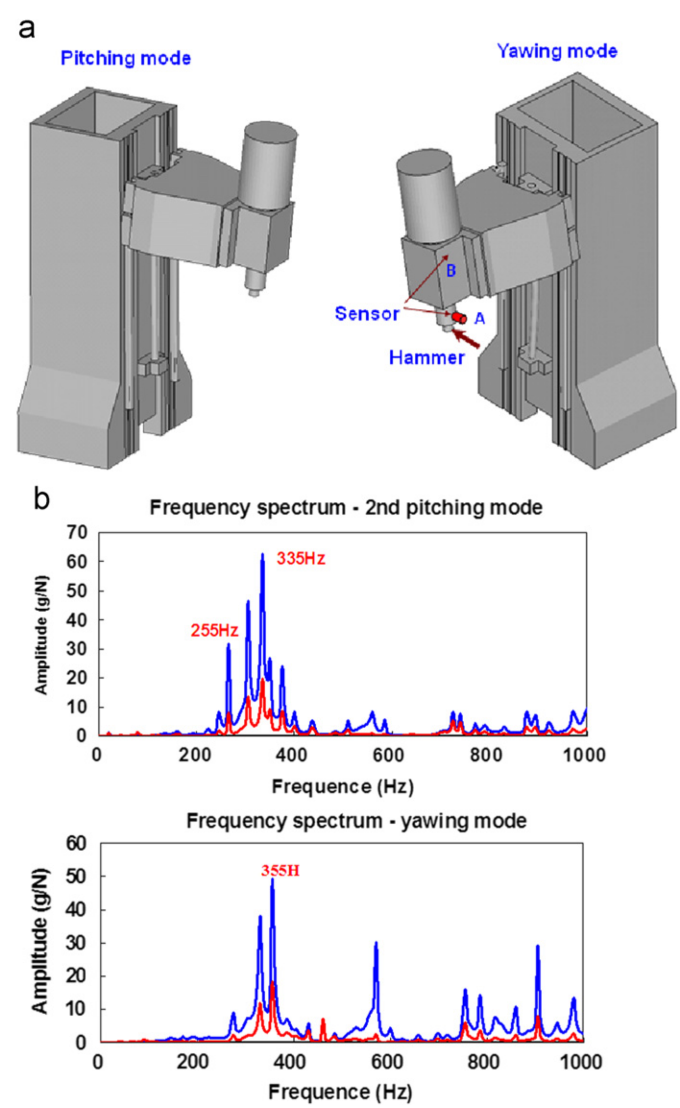

Fig. 3. (a) The finite element predicted vibration motion of vertical column spindle system, including pitching and yawing vibrations. The configuration of the vibration test is presented in this figure, indicating the locations of accelerometers (A, B) and impact hammer is presented. (b) The measured vibration spectrum of pitching and yawing modes of spindle head.

grooves of the linear components. The rolling interface primarily determines whether the simulation results approach the real characteristics of the system [22,23]. Therefore, the linear components were simplified and introduced into the finite element model of the vertical milling machine.

3.1. Modeling of the rolling interface

As shown in Fig. 2(a), the linear guide was designed with two or four rows of rolling balls, where the ball groove had the contact profile of a circular arc, forming a two-point contact mode [28]. The ball screw was designed with a Gothic groove, forming a four-point contact mode. In general, the contact force between a rolling ball and the raceway can be related to the local deformation at the contact point by the Hertzian expression [29]

$$Q = K_h \alpha^{3/2}, \tag{1}$$

where Q denotes the contact force and α is the elastic deformation at the contact point. K_h represents the Hertz constant, which is determined by the contact geometry of the ball groove or raceway and the material properties of the contacting components. The details are available in the literature [30,31]. The normal stiffness at a specific preload can then be obtained as

$$K_n = \frac{dQ}{d\alpha} = \frac{3}{2} K_h \alpha^{1/2} = \frac{3}{2} K_h^{2/3} Q^{1/3}. \tag{2}$$

As shown in Eq. (2), the contact stiffness in the normal direction depends nonlinearly on the contact force, which is essentially determined by the initial preload on the rolling balls.

3.2. Creation of finite element model

Fig. 2 presents the finite element model of a vertical column spindle structure including a motorized spindle tool. After a convergence test, each structural component of the system was meshed with eight-node brick elements with a total of 38,121 elements and 46,597 nodes. The components of the feeding mechanism, including the ball bearing, ball screw, and ball nut, and the linear guides were included in the model. To make the model realistic, the main bodies of the linear components were modeled as solid elements and connected to spring elements at the rolling interfaces. A rolling guide has four ball grooves with circular profiles, forming a two-point contact state between the rolling ball and the groove. To reduce the complexity of the mesh generation of the rolling grooves in the model, the four rolling grooves were represented as two grooves, as shown in Fig. 2(b). The sliding block and guide rail were directly connected using a series of spring elements by ignoring the effect of the rolling balls. The spring elements at each ball groove had an overall contact stiffness equivalent to that of the original guide model. This two-point contact mode greatly increased the efficiency of the analysis model without affecting the accuracy of the results. Given the specifications of the rolling guides (ball diameter $d=3.175$ mm, diameter of circular groove $D=3.297$ mm, dynamic loading capacity $C=11{,}380$ N), the contact stiffness K_n was calculated as 9.76 N/μm for low preloading, 18.7 N/μm for medium preloading, and 21.7 N/μm for high preloading.

In a similar manner, the ball screw was modeled as a cylindrical shaft and meshed with 3D solid elements rather than flexible beam elements [32]. The overall stiffness in the radial and axial directions could be directly determined based on the dimensions of the screw spindle and ball nut. In addition, to reduce the complexity and

Table 2
Experimentally measured vibration frequencies and damping ratios of the vertical column spindle systems.

Vibration mode	Low preload (Z0)		High preload (ZB)	
	Natural frequency (Hz)	Damping ratio, ξ	Natural frequency (Hz)	Damping ratio, ξ
Pitching vibration of spindle head, accompanied with bending vibration of vertical column	255	0.009	310	0.006
Pitching vibration of spindle head	335	0.013	390	0.008
Yawing vibration of spindle head	355	0.008	465	0.005

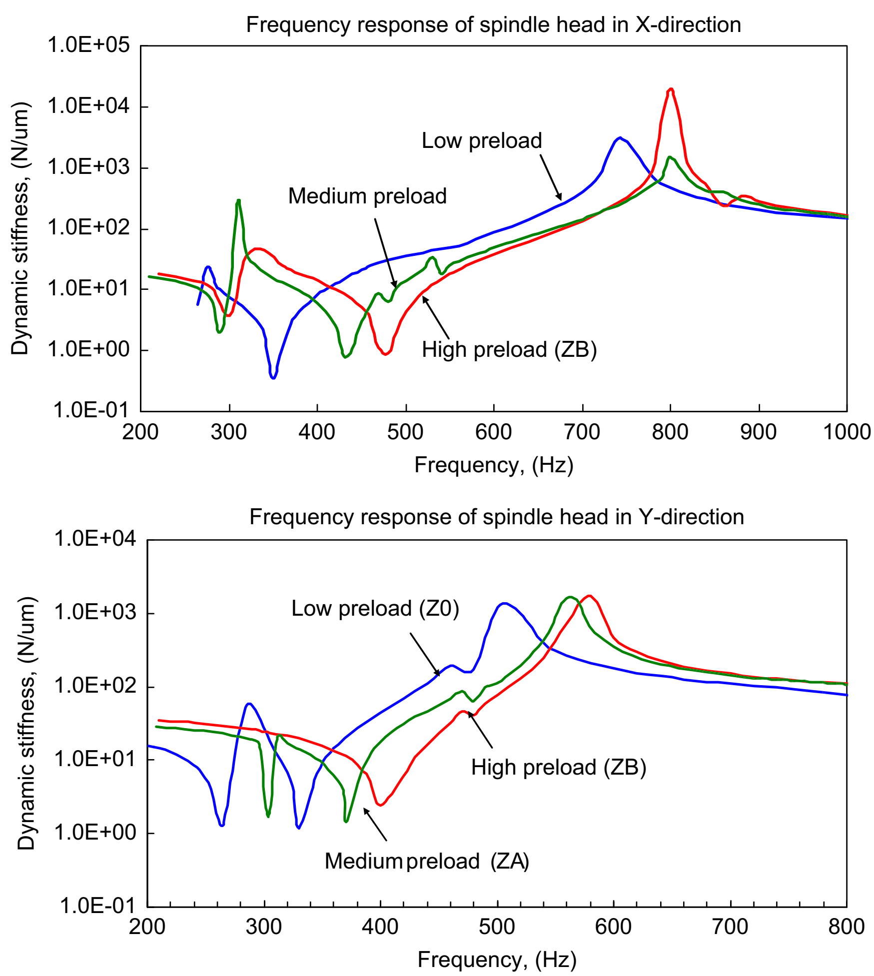

Fig. 4. The frequency responses of the spindle tip in the X and Y directions, respectively, comparing the difference in dynamic stiffness between the head equipped with low and high preloaded guides, respectively.

inconvenience of meshing the helical groove around the screw shaft, the contact between the screw and the nut was simplified to a circular contact mode, as shown in Fig. 2(c). With this simplified mode, the rolling interface at the ball groove was simulated with elastic spring elements connecting the ball nut and the screw shaft. Given the specifications of the ball screw, the associated contact stiffness at the screw groove was estimated as 152 N/μm, which is slightly less than that obtained from technical information [26] for the ball nut (approximately 168 N/μm). The inner and outer rings of the bearings were connected by spring elements distributed around the ring raceway, which provided stiffness in both the axial and radial directions to sustain the ball screw. The bearing stiffness for the angular contact bearing, obtained from the bearing manufacturer, was 88 N/μm [27].

The motorized spindle unit consists of a driven motor, the spindle shaft, and the bearing assemblies. The spindle dynamics are therefore more complicated than those of any other linear component and are important in machine tool design, but they are not included in the current study. For simplification, the spindle unit was modeled as a cylindrical body with various sections and a weight equivalent to that of the original spindle unit. For the finite element analysis, all the components have the following material properties of carbon steel: elastic modulus $E=200$ GPa, Poisson's ratio $\mu=0.3$, and density $\rho=7800$ kg/m^3.

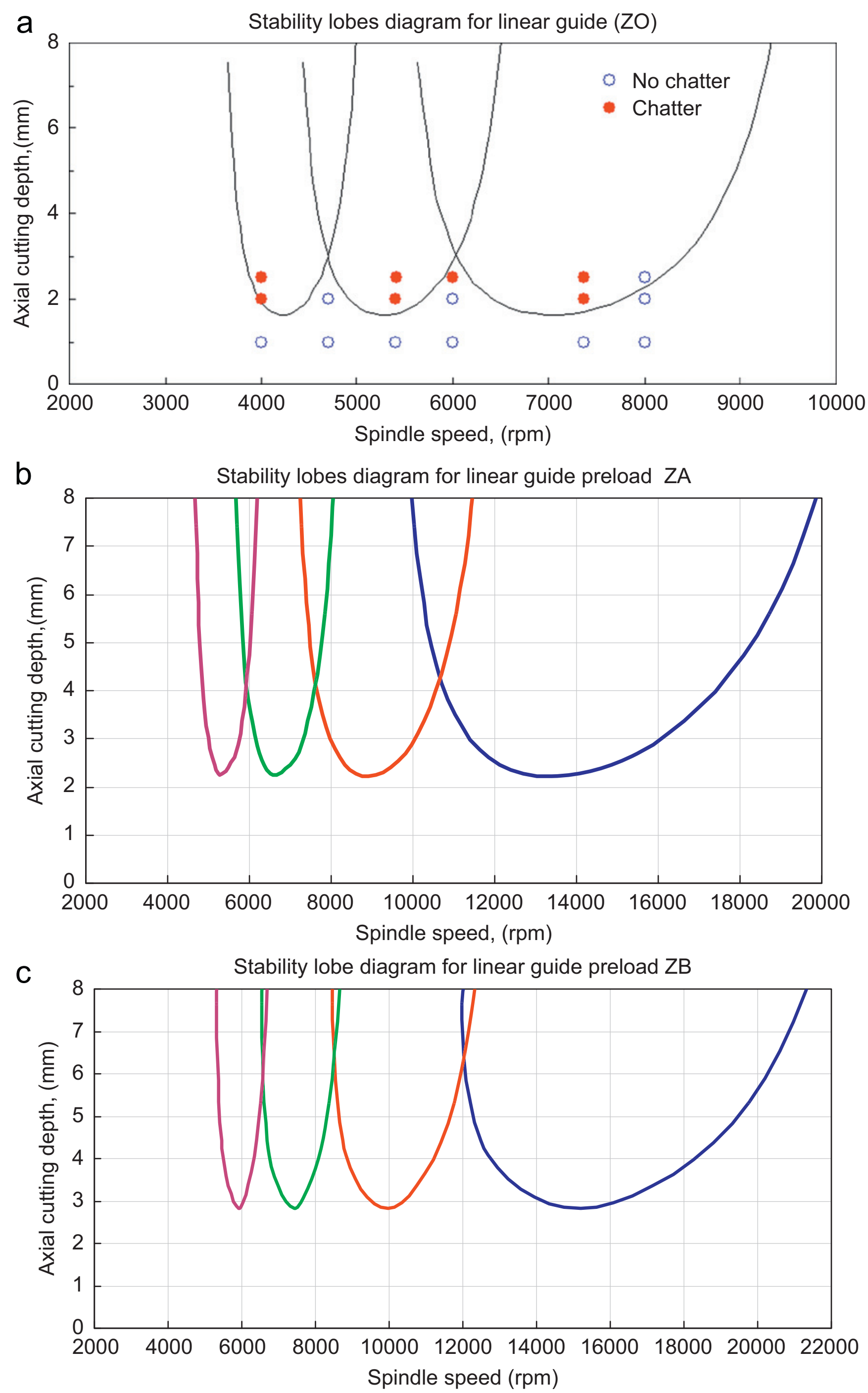

Fig. 5. Stability lobes diagram of the vertical milling machine. The spindle head was mounted on the feeding stage with low-, medium- and high-preloaded linear guide modulus, respectively.

3.3. Model identification with vibration test

To validate the finite element model, we first performed modal analysis to obtain the natural vibration characteristics of the vertical column spindle head. We compared the results with experimental measurements. According to the finite element simulation [19], the vibration motions associated with the vertical column spindle head are critical modes that lead to a lower dynamic stiffness in the spindle. They are illustrated in Fig. 3(a) as the pitching and yawing vibration modes, respectively, of the spindle head.

The natural frequencies of the spindle head predicted by modal analysis are listed in Table 1. When the feeding stage of the spindle head is equipped with linear guides with a small preload (Z0), the frequencies associated with the first and second pitching motions and the yawing motion of the spindle head are 262, 332, and 325 Hz, respectively. For a feeding stage with linear guides with a high preload (ZB), the corresponding natural frequencies of the spindle head are 310, 407, and 483 Hz. Comparisons of the results in Table 1 indicate that the linear guide preload has a large influence on the vibration frequency of the spindle head. The frequency increases by 18–37% when the preload of the linear guide at the feeding stage is adjusted from Z0 to ZB.

The modal-analysis result was verified via a vibration test conducted on the vertical column spindle structure. Fig. 3(a) presents a schematic for the vibration testing, in which accelerometers were mounted on the spindle head to measure the vibration signal induced by an instrumented hammer. In this manner, the yawing mode of the spindle head about the X-axis in the local coordinate system was measured. Fig. 3(b) depicts the frequency spectra corresponding to the yawing and pitching vibrations of the spindle head that was mounted on the feeding stage with low preload linear guides. The natural frequencies corresponding to the first and second pitching motions and the yawing vibration were measured as 255, 335, and 355 Hz, respectively. Comparisons with the data in Tables 1 and 2 show that the finite element predictions agree well with the experimental result. This further confirms the accuracy of the modeling of the vertical spindle column structure, which consists of the rolling interfaces in the various linear components.

The frequency response functions of the machine structure play important roles in the prediction of the system dynamic behavior and the machining stabilities with or without regenerative chatter. In general, the dynamic response of the machining system can be obtained through harmonic analysis. The structural damping parameter required for the harmonic analysis was calculated from the experimentally measured vibration signal using the Half-power method. Table 2 presents the damping ratios of the vertical column spindle structure in the X and Y directions. We found that the damping characteristics are affected by the preload of the linear guides. A vertical spindle head system installed with a high preloaded guide modulus has a lower damping characteristic than that with a low preloaded linear guide. In addition, the vertical column spindle system has different structural damping characteristics in different vibration directions. This is caused in part by the inherent damping capability of the linear guides that was also shown to vary with the preload of the rolling ball [33].

In finite element governing equation for harmonic analysis, the damping matrix was assumed to be proportional to the structural stiffness matrix, $[K]$ according to the relationship $[C]=\beta[K]$. The value β_{mr}, representing the structural damping factor, is calculated from $2\xi_{mr}/w_r$, where ξ_{mr} is the modal damping ratio for a particular vibration mode. This assumption indicates that the damping characteristics is frequency-dependent, governed by the dominant vibration mode of the machine tool system.

3.4. Prediction of tool-point frequency response

To predict the frequency response function of the vertical column spindle structure, a harmonic analysis was performed on the validated finite element model by applying a unit force at the spindle tip. Fig. 4 illustrates the frequency response of the spindle tip in the X and Y directions, respectively, and also demonstrates the influence of the preload of the linear guide modulus. The lowest dynamic stiffness in the X direction was calculated as 0.352, 0.81, and 0.903 N/μm, respectively, for spindle heads with low-, medium-, and high-preloaded guides. In addition, the lowest dynamic stiffness occurs at the frequency of the yawing vibration of the spindle head. For the dynamic response in the Y direction, the spindle head has a minimum stiffness ranging from 1.17 to 2.41 N/μm depending on the preload of the linear guides. Comparing with the results in Fig. 4, we see that the stiffness in the X direction is weaker than that in the Y direction since the linear guide is more rigid in the normal direction of the sliding blocks than in the lateral directions.

The frequency-induced minimum stiffness in the Y direction was found to approach the pitching vibration of the spindle head. The results of the harmonic analysis indicate that the yawing and pitching modes of the spindle head dominate the dynamic characteristics of the vertical milling machine. It can be expected that the spindle head will behave more compliantly when the yawing or pitching vibration is induced during machining. This consequently causes the spindle tool to deform greatly or to vibrate unstably and is likely to degrade the machining performance.

Based on these results, we can verify that the linear guide supporting the feeding stage of the spindle head is an important component that affects the dynamic characteristics of the entire vertical column structure of the milling machine

4. Prediction of machining stability and experimental verification

4.1. Stability model and stability lobe diagram

The machining stability of the vertical milling machine can be predicted based on the analytical model developed by Alintas and Budak [7,8]. In their approach, the time-varying force coefficient of the dynamic milling process model was approximated by Fourier-series components. Following this, the stability relationship between

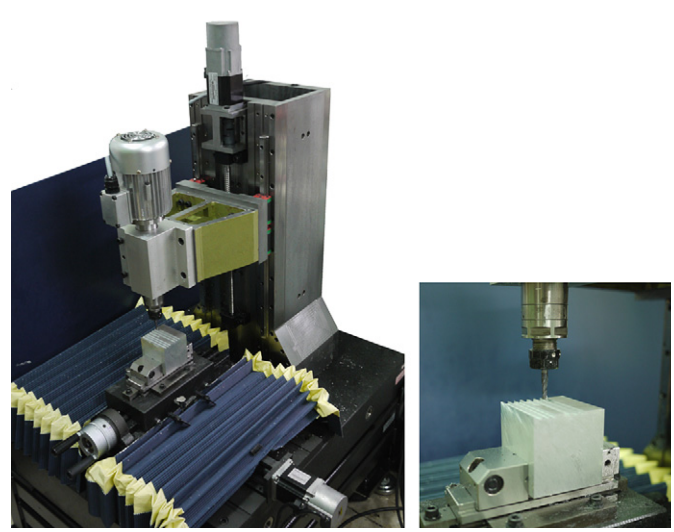

Fig. 6. Configuration of machining tests on the vertical milling machine.

the chatter-free axial cutting depths (Z_{min}) and the spindle speed (n) in end-mill operation were derived by Gagnol et al. [34] as follows.

The speed-dependent transfer function representing the ratio of the Fourier transform of the displacement $X(j\omega)$ at the tool tip over the dynamic cutting force $F(j\omega)$ can be expressed as

$$H(j\omega)=\frac{X(j\omega)}{F(j\omega)}=R_e(\omega)+jI_m(\omega) \tag{3}$$

$$Z_{min}=\frac{-1}{NK_tK_rR_e(\omega)} \tag{4}$$

$$\varphi=\pi-2\tan^{-1}\frac{I_m(\omega)}{R_e(\omega)} \tag{5}$$

$$n=\frac{60\omega_c}{N(2k\pi+\varphi)},\quad k=lobes(0,1,2\ldots). \tag{6}$$

In the above equations, $H(jw)=R_e(w)+jI_m(w)$, R_e and I_m are, respectively, the real and imaginary part of the transfer function of the spindle tool tip. K_t and K_r are the cutting resistance coefficients in the tangential and radial directions to the cutter. N is the number of cutter teeth and k is the lobe number. To predict the machining stability, a two-tooth carbide cutter was employed to machine the stock material of Al7075. The cutting resistance coefficients were calibrated as K_t=796 N/mm^2 and K_r=0.21 [35]. According to the harmonic analysis, the yawing vibration mode of the spindle head is the dominant factor influencing the dynamic behavior of the vertical column

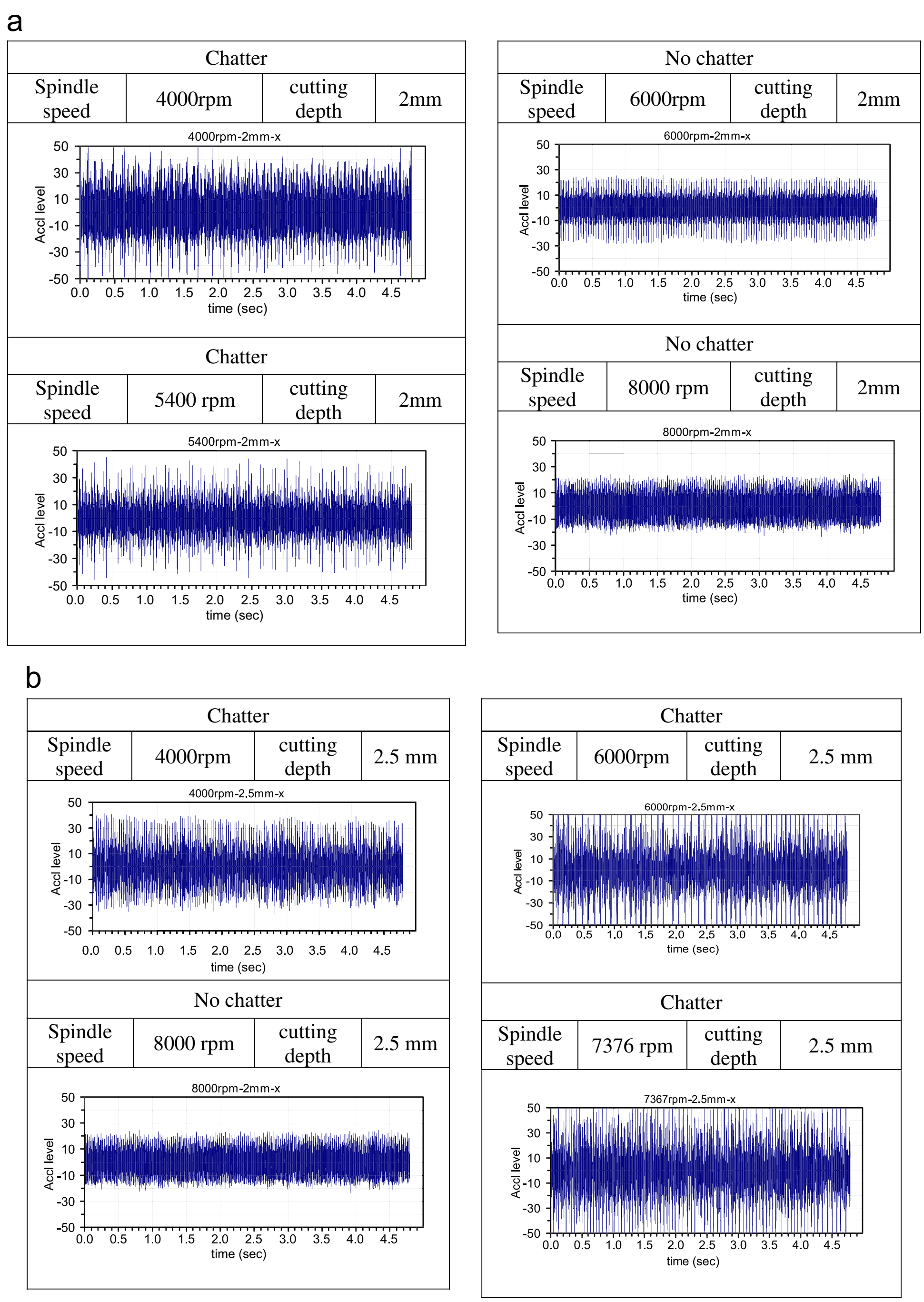

Fig. 7. Vibration spectrums measured in machining tests, indicating the occurrence of regenerative chattering: (a) cutting depth of 2.0 mm and (b) cutting depth of 2.5 mm.

structure; as a result, the structure exhibits the lowest dynamic stiffness. Therefore, the tool-point frequency response functions at the central point of the spindle tool were predicted to obtain the stability lobes. The stability lobe diagram was obtained by integrating the finite-element-predicted tool-tip frequency response in the X direction into the chatter stability approaches.

Fig. 5 illustrates the stability diagrams of the vertical milling machine, showing the differences in the stability boundary resulting from linear guides with different preloads. As seen in Fig. 5, when high preloaded guides were installed on the spindle feeding stage, the limiting axial depth with free chatter is about 2.9 mm for spindle speeds ranging from 4000 to 8000 rpm. For the spindle feeding stage equipped with guides with medium or low preloads the minimum axial depth with free chatter is reduced to approximately 2.3 and 1.6 mm, respectively, for the same spindle-speed range. The results of the stability analysis verify that the linear guide preload significantly affects the vertical milling machine behavior.

4.2. Experimental verification: machining test

The cutting stability boundaries predicted in the above sections were experimentally verified by a machining test. As shown in Fig. 6, the machining tests were conducted using a prototype of the vertical milling machine in which a linear guide modulus with low preload was installed in the feeding stage of the spindle head. A Tungsten carbide cutter (two teeth with a diameter of 5 mm and length of 50 mm) was used to machine the stock material of Al7075. Full-immersion slots were milled at varying axial depths and spindle speeds in the X direction at a constant feeding rate of 75 mm/min. The evaluation of the machining stability was based on the vibration signal measured at the spindle head during the machining and a visual examination of the machined surface. Moreover, for a reliable reference without chatter, an initial chatter-free cutting depth was chosen for a specified spindle speed according to the predicted stability lobe diagram.

The experimental results are compared with the predicted stability lobes in Fig. 5(a). Each cutting with chatter is indicated by a solid circle and each cutting without chatter by an open circle. For example, for a cutting with an axial depth of 2 mm, chatter was detected at spindle speeds of 4000, 5400, and 7376 rpm, but no chatter occurred when the spindle speed was 8000 rpm. At a spindle speed of 4000 rpm, unstable cutting was observed when the cutting depth was set to 2 mm, but no chatter occurred when the depth was adjusted to 1 mm. The chatter was observed and verified from the vibration spectrum presented in Fig. 7. It is obvious that at the same cutting depth (2.0 mm), the vibration amplitudes induced at spindle speeds of 4000 and 5400 rpm are greater than those induced at speeds of 6000 and 8000 rpm. A series of tests for a cutting depth of 2.5 mm were also performed. As shown in Fig. 5(a), the cutting states for spindle speeds of 4000, 6000, and 8000 rpm were located at the margin of the stability lobe diagram. However, significant chattering vibrations were measured in machining tests when the spindle was operated at 4000, 6000, and 7376 rpm and no chatter was observed for a spindle speed of 8000 rpm. A visual examination of the machined surface also helps to identify the results of stable and unstable machining, as shown in Fig. 8. In this figure, a series of slot cuttings with an axial depth of 2.5 mm were machined into different surface finishes at different spindle speeds. Poor surfaces were produced at spindle speeds of 6000 and 7376 rpm because of chatter, but when the spindle was operated at 8000 rpm, chatter-free cutting resulted in a better surface. The results of the machining tests confirm the influence of the cutting parameters such as the spindle speed and the cutting axial depth on the surface quality, and this is also demonstrated by the stability lobe diagram. This result also shows that the stability regimes for machining operations performed on certain workpieces with a specific milling cutter can be predicted based on a finite element simulation of the dynamic behavior of the machine tool.

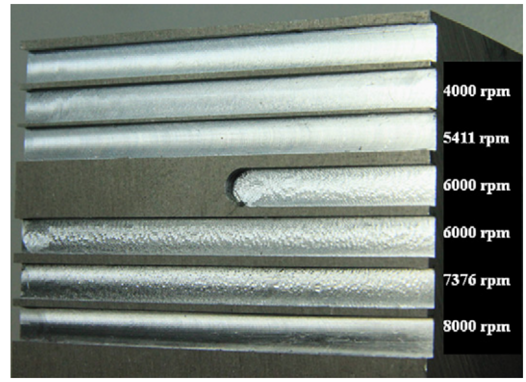

Fig. 8. Machined surface finish under different spindle speeds (slot cutting depth = 2.5 mm).

5. Conclusions

In this paper, a finite element modeling technique involving simulation of linear components with rolling interfaces was presented. This technique can be applied to predict the machining stability of a vertical milling machine, and the effectiveness of this technique was successfully verified by conducting machining tests on a prototype. According to the finite element analysis, the yawing vibration mode associated with the spindle head was the dominant factor influencing the dynamic stiffness and machining stability of the milling machine, which were governed by linear guides installed on the feeding stage. The results of the stability analysis clearly demonstrate the dependence of the machining stability of the vertical milling machine on the preload of the linear guide. The guides with a high preload that were installed in the spindle feeding mechanism are more effective than the linear guides with low preloads in enhancing the dynamic stiffness of the system. This further improves the machining performance of the milling machine for a wider range of spindle speeds and cutting depths. The experimental results obtained in this study demonstrate the accuracy of the finite element simulation model. The model can be used to optimize the limit on the machining stability of a milling system by adopting various design configurations that determine the geometrical structure of the linear components.

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