International Journal of Machine Tools & Manufacture 50 (2010) 741–746

Contents lists available at ScienceDirect

International Journal of Machine Tools & Manufacture

journal homepage: www.elsevier.com/locate/ijmactool

Short Communication

Effect of preload of linear guides on dynamic characteristics of a vertical column–spindle system

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ARTICLE INFO

Article history:
Received 28 December 2009
Received in revised form
1 April 2010
Accepted 2 April 2010
Available online 21 April 2010

Keywords:
Vertical column–spindle system
Dynamic stiffness
Linear guide
Contact stiffness

ABSTRACT

Realization on the dynamic characteristics of the column–spindle system is of importance for enhancing the structural performance of a vertical milling machine. Generally, the spindle head is fed under linear guide mechanism through a ball-screw driver. To assess the dynamic characteristics of a vertical column–spindle system under the influence of a linear guide, this study developed a finite element model integrated with the modeling of linear components with the implementation of contact stiffness at the rolling interface. Both the finite element simulations and the vibration tests reveal that the preload of a linear guide greatly affects the vibration behavior associated with a spindle head, and the dynamic stiffness of the spindle head could be enhanced by increasing the preload of the linear guide. Current results clearly indicate that the simulations agree well with the experimental measurements. This also confirms that the proposed model can be successfully applied to evaluate the dynamics characteristics of machine tool systems of various configurations.

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1. Introduction

Currently, machine tool structures are designed with the modularity concept to satisfy the multipurpose or specific industrial applications. An innovative CNC machine tool was fabricated by assembling these modularized components by using feeding mechanisms of various structural configurations [1,2]. At the design stage, the designer could employ the finite element analysis to predict the static and dynamic behaviors of machine tool structures. However, the machine tool cannot be easily formulated as a composition of individual modules because there are various interfaces at connection joints such as bolted components, which form weak links between components and greatly affect the overall stiffness and mechanical characteristics of the assembled structure [3]. Consequently, the modeling of an interface with accurate interfacial characteristic is of importance in the dynamic analysis of a machine tool structure [4]. However, with high nonlinearity, the interface characteristics at joints could not be directly obtained using an analysis method unless experimental measurements were also made [5–7].

In linear components, rolling interfaces exist between the rolling balls and the grooves, which exhibit nonlinear contact characteristics of Hertzian type [8]. Owing to the influence of bearing stiffness, spindle-bearing systems have been a typical research topic explored by many researchers in investigating the variation of the dynamic characteristics during operation [9–12]. In general, the dynamic behavior of a spindle is mostly determined by the initial preload applied to the bearings [13]; however, it may deviate from the original behavior because of the variations in preload and bearing stiffness under different operating conditions [13,14]. In an individual study by Akturk et al. [9] and Harsha et al. [10], the bearing stiffness might increase with an increase in the number of balls in a ball bearing, which further increased bearing stiffness and reduced the vibration amplitude of the shaft. Lynagh et al. [14] proved that the bearing stiffness of angular contact bearings decreases with increase in rotational speed, which consequently affects the spindle dynamics. These studies clearly pointed out that the change in the bearing stiffness under different operating conditions mainly yielded the variation in dynamic characteristics of a spindle tool system.

For linear rolling guides, an oversized rolling ball was usually employed to produce adequate preload in order to increase the structural rigidity and the load-carrying ability of a rolling guide. While research studies [15–17] have shown that a rolling guide exhibits different vibration characteristics when the preload is set to different magnitudes. Hung [18] also verified that the external load acting on the positioning stage caused a variation in structure stiffness of linear guides, and hence, brought the stage to vibrate at different frequencies. Concluding from the above mentioned studies, we can realize the importance of effect coming from the rolling interfaces with linear components.

This study was aimed at investigating the influence of linear guides on the dynamic characteristics of a vertical

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doi:10.1016/j.ijmachtools.2010.04.002

column–spindle system. To this, a finite element model of the column–spindle system was proposed with the integration of the modeling of linear rolling components. The dynamic characteristics of the system such as the natural vibration modes and dynamic stiffness were predicted and validated with the experimental measurements performed on a prototype.

2. Vibration tests

2.1. Experimental configuration

A vertical column–spindle system was designed and constructed for vibration analysis, as shown in Fig. 1. The sliding carriage of the spindle head was mounted on the column with a pair of linear guides and was driven with a ball screw. The commercial linear guides (Hiwin EG series) have four ball grooves with a circular arc profile [19], and they are quantified as low preload (Z0, 0.02*C*) and high preload (ZB, 0.11*C*), where *C* denotes the dynamic load rating (11.38 kN). The driven ball screw has a diameter of 14 mm, and a basic dynamic load rating *C* of 4.07 kN. To lessen the axial backlash, the ball nut was slightly preloaded to a level of 0.06*C* [20]. The standardized ball screw support units coded EK12 and EF10 were used at both ends of the screw shaft [21].

The devices used for the vibration test consist of an impact hammer, accelerometers, signal amplifier, and Fourier transform analyzer. In the experiments, the prototype structure was bolted on a graphite bed and the sliding carriage with spindle head was positioned at top of the column. The accelerometers were mounted on the column and spindle tail to measure the vibration spectrum corresponding to different excitations, as shown in Fig. 2. The tests were performed on the prototype with linear guide modules adjusted at low and high amounts.

2.2. Vibration characteristics

The vibration spectrums measured for certain vibration modes are depicted in Fig. 2. The first bending vibrations of the vertical column were observed to occur at a frequency of 385 Hz, as shown in Fig. 2(a). Fig. 2(b) and (c) shows the vibration spectrum identified as the pitching and yawing motions of the spindle head, where local peaks occurred at 965 and 1010 Hz, respectively. For comparison, the natural frequencies of the column–spindle system with linear guides of different preloads are summarized in Table 1. The frequencies of yawing and pitching mode were observed to increase by 10% and 12%, respectively, when the linear guide preload was adjusted from low (Z0) to high (ZB). Current results clearly show that linear guides with different preloads enable the spindle head to exhibit vibration characteristics to different extents.

3. Finite element modeling of a vertical column–spindle system

3.1. Calculation of contact stiffness

As shown in Fig. 3(b), 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 [22]. For ball screw, it 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 [8]:

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

where Q denotes the contact force and α the elastic deformation at the contact point. K_h represents the Hertz constant, which is determined from the contact geometry and material properties of the linear components. Details are available in the literatures [23,24].

The normal stiffness at a specific preload can then be obtained as

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

As revealed in Eq. , the contact stiffness in the normal direction depends nonlinearly on the contact force, which is essentially determined by the initial preload set on the rolling balls.

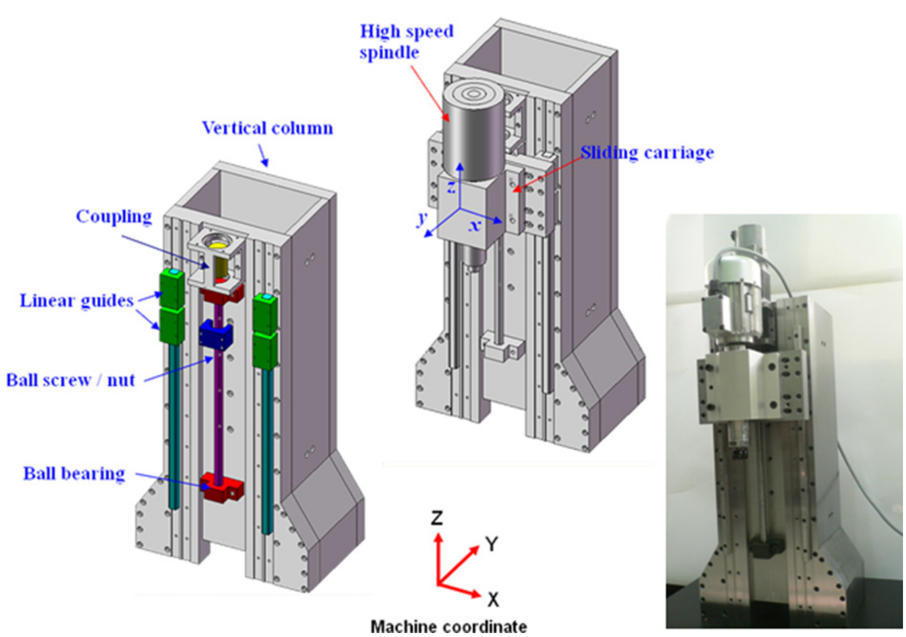

Fig. 1. The prototype of a vertical column–spindle system.

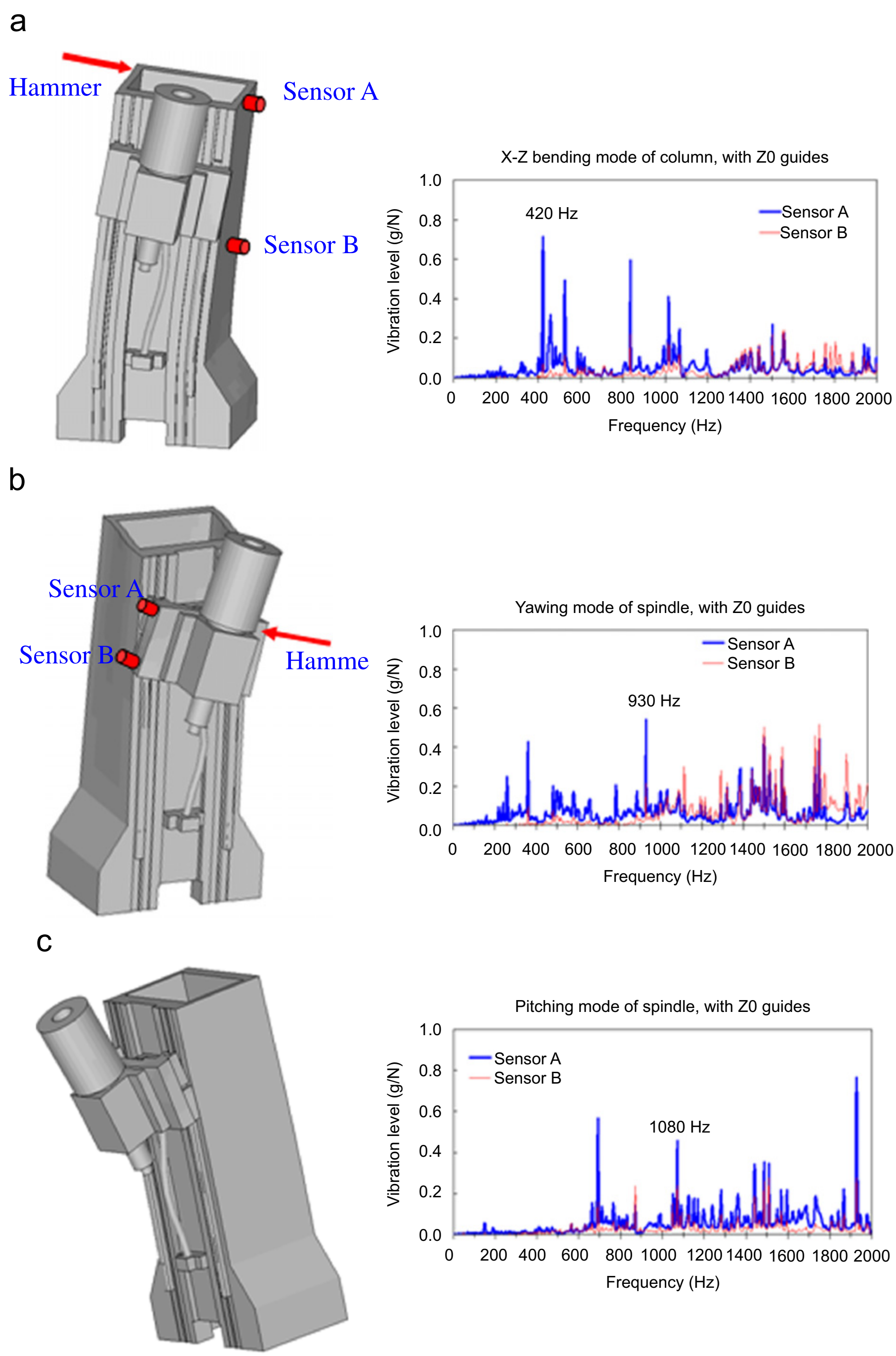

Fig. 2. The finite element predicted fundamental vibration motion of vertical column–spindle system, which also shows the configuration of the vibration test, including the locations of accelerometers (A, B) and impact hammer: (a) bending vibration along xz plane and vibration spectrum; (b) yawing vibration of spindle head and vibration spectrum and (c) patching vibration of spindle head and vibration spectrum.

3.2. Finite element modeling

Fig. 3 presents the finite element model of the vertical column–spindle structure, including a motorized spindle tool. Each structural component of the system was meshed with eight-node brick elements, with a total of 37,709 elements and 46,957 nodes, after a convergence test. The ball bearing, ball screw and ball nut, and linear guides were included in the model. For the model to be realistic, the main bodies of the linear components were modeled as solid elements and connected with spring elements at the rolling interfaces. To avoid the complexity in mesh generation of the rolling grooves, the four rolling grooves were simplified as two grooves, as shown in Fig. 3(b). The sliding block and guide rail were directly connected using a series of spring elements by intentionally ignoring the effect of the rolling balls [18]. But, the spring elements at each ball groove had an overall contact stiffness equivalent to that of the original guide model with four ball grooves. This two-point contact mode could greatly increase the efficiency of preparing the analysis model, without affecting the accuracy of the results. Based on the specifications of the rolling guides, the contact stiffness K_n was calculated as 75.9 N/μm for low preload and 129.7 N/μm for high preload.

The ball screw was modeled as a cylindrical shaft and meshed with 3D solid elements, rather than flexible beam elements [25].

Table 1
Vibration frequencies associated with mode shape of column–spindle system (spindle head height $Z=330$ mm; unit: Hz).

Mode shape	Preload					
	Z0			ZB		
	FEM	Vibration test	Difference (%)	FEM	Vibration test	Difference (%)
Bending vibration of column along X–Z plane	351	340	3.24	352	360	−2.22
Bending vibration of column along Y–Z plane	430	420	2.38	432	420	2.86
Torsion vibration of column about Z-axis	747	745	0.27	755	750	0.67
Yawing vibration of spindle head about local Z-axis	934	930	0.43	1036	1020	1.57
Pitching vibration of spindle head about local X-axis	1093	1080	1.20	1215	1205	0.83

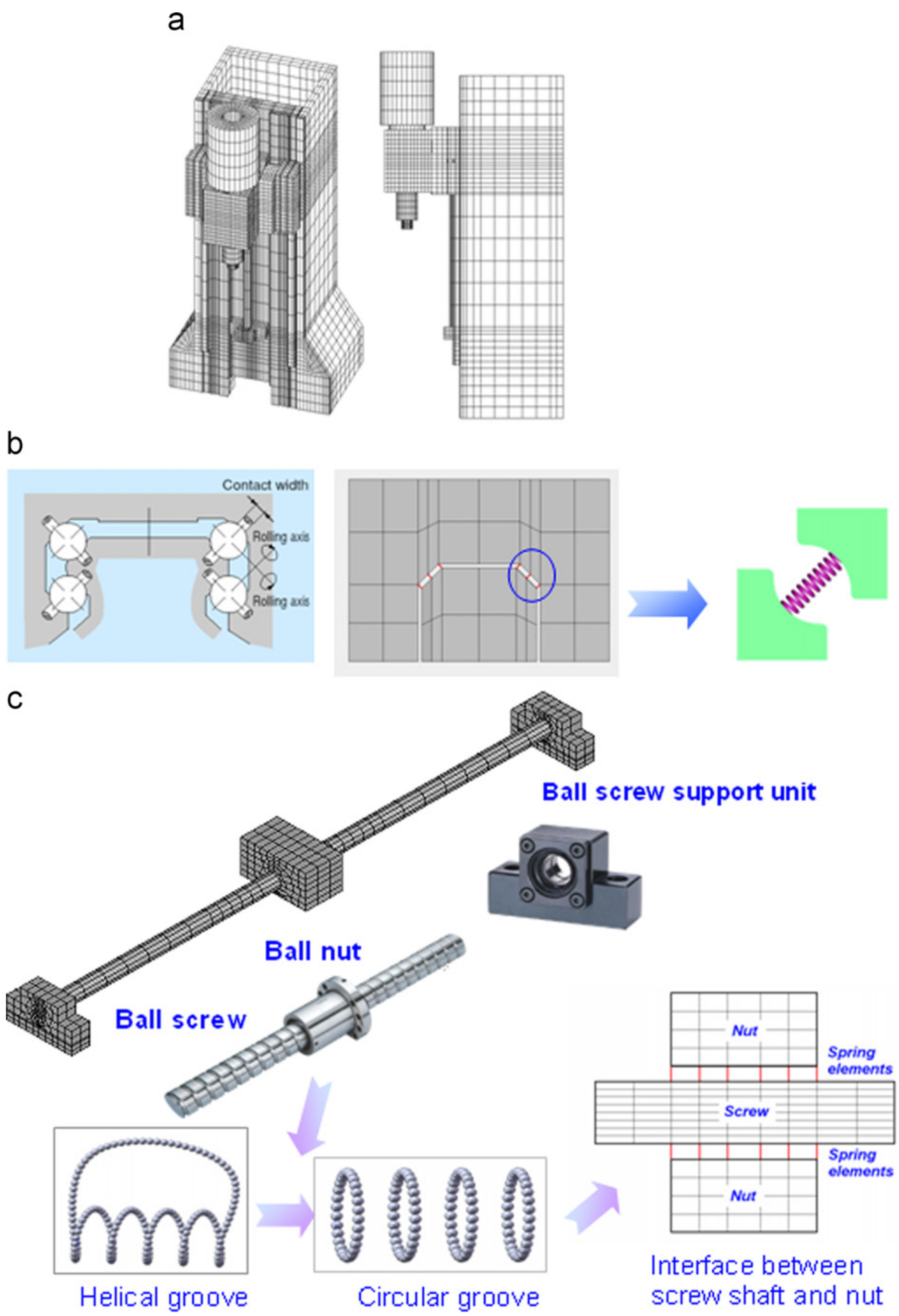

Fig. 3. (a) Finite element model of vertical column–spindle system, the spindle head positioned at a height of 330 and 165 mm. (b) Modeling of linear rolling guide, and (c) modeling of ball screw and support units with ball bearings.

In addition, considering the complexity and inconvenience in meshing the helical groove around the screw shaft, the contact between the screw and the nut was simplified as a circular contact mode, as shown in Fig. 3(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. According to 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 of the ball nut (approximately 168 N/μm) obtained from technical information [20]. For a bearing, the inner and outer rings were also connected by spring elements distributed around the ring raceway, which provided the stiffness in both axial and radial directions to sustain the ball screw. The bearing stiffness for the angular contact bearing was obtained from the bearing manufacturer, which was reported as 88 N/μm [21].

The spindle dynamics of the motorized spindle unit are more complicated than any other single linear component and has been widely discussed in other researches [9–14], but not included in the current study. For simplification, the spindle unit was modeled into a cylindrical body of various sections, but with an equivalent weight of the original spindle unit.

For finite element analysis, all the materials have the properties: Young's modulus $E=200$ GPa, Poisson's ratio $\mu=0.3$, and density $\rho=7800\,\mathrm{kg/m^3}$. The vibration mode shapes associated with the frequencies of the vertical column–spindle system were obtained from modal analysis. The harmonic analysis was performed to assess the dynamic response of the vertical column–spindle structure when the spindle tail subject to an external force.

4. Results and discussions

4.1. Vibration modes

Five fundamental vibration motions of the vertical column–spindle structure were extracted from finite element modal analysis, in which three interesting modes are depicted in Fig. 2 for further illustrations. The first and second modes are the bending vibrations of the column structure along the *YZ* and *XZ* planes at frequencies of 351 and 429 Hz, respectively. The third mode is the twisting mode at frequency of 747 Hz about the *Z*-axis. The fourth and fifth modes are mainly related to the vibrations of the spindle head. The spindle sways about the local *y*- and *x*-axis at the frequencies of 930 and 1080 Hz, respectively, which are also termed as the yawing and pitching vibrations of the spindle head. As observed in the mode shape, the linear guide with the rolling interface plays an important role in determining the vibration behavior of the vertical spindle head where the spring elements connecting the sliding blocks and guides actually support the head.

4.2. Preload effect of linear guide

Table 1 compares the vibration frequencies of a vertical column–spindle system equipped with linear guides of different preloads. For a column structure equipped with linear guides of low and high preloads, the maximum difference of the vibration frequency between them is less than 1%. However, for higher modes associated with spindle head, the frequencies are greatly affected by the preload of the linear guides. The spindle head vibrates in the yawing mode at the frequency of 930 Hz when it was equipped with low preloaded guides (Z0), while it vibrates at a higher frequency of 1027 Hz when equipped with guides of high preload. There is an increase of 10% in the frequency of yawing mode when the preload of the linear guide changed from a low to a high value. A similar effect was also found to appear in the pitching vibration. There is an increase of 12.4% in the vibration frequency when the preload of the guide modules was adjusted from low to high.

Comparisons of the results in clearly show that the finite element predictions agree well with the vibration measurements. The maximum difference between them is less than 3%, regardless of the preload amount of the linear guides. This also demonstrate that the analysis model developed in this study can accurately reflect the dynamic characteristics of a vertical column–spindle structure, while the success of predictions greatly depends on the modeling of linear guides with correct bearing stiffness implemented in the finite element model.

4.3. Dynamic stiffness

As presented in the modal analysis, the linear guides dominate the vibration characteristics of the column–spindle structure and spindle tool tips. The harmonic analysis was therefore performed to measure the frequency response at the spindle tail. The dynamic frequency response of the spindle tip subjected to a dynamic force in the *X* and *Y* directions was assessed and presented in Fig. 4.

As observed in Fig. 4, the column–spindle system with low preload linear guides exhibited a low dynamic stiffness of 0.60 N/μm in the *X* direction at 932 Hz frequency. When the high preload guide modules were used to support the sliding carriage, the spindle head exhibited a minimum dynamic stiffness of 1.41 N/μm at a higher frequency of 1040 Hz.

A similar frequency response was also found in the *Y* direction. It can be seen that the transition of the preload from low to high resulted in an increase in the resonant frequency from 1090 to 1210 Hz and in the minimum dynamic stiffness from 1.16 to 2.56 N/μm.

It is obvious that high preload guide modules indeed enhance the dynamic stiffness of spindle head as compared to light preload. As noticed in the figures, the frequency for the occurrence of the minimum stiffness corresponds to the yawing and pitching modes, which are associated with the vibration of the spindle head in different directions. When one of the two modes was excited during machining, the spindle head behaved more compliantly than at another frequency; this consequently causes the spindle tool to deform greatly or vibrate unstably and thereby degrade the machining performance. From this analysis, we can again identify that a linear guide could be a key component that affects the dynamic characteristics of the entire column–spindle structure.

Dynamic stiffness was usually used as a performance index for evaluating the mechanical characteristics of a machine tool structure. However, it is influenced not only by the structural rigidity of the vertical column and the headstock, but also by the dynamic performance of the spindle-tool unit. It is well recognized that the dynamic behavior of a spindle tool is more complicated due to the variation in the bearing stiffness under changeable operation conditions. But in most researches on spindle dynamics, the effect of the machine tool structure was not taken into consideration. A recent study by Cao and Altintas [12] further verified that the modeling of a spindle alone could not accurately predict its dynamics when it is used along with a machine tool. They also demonstrated that the virtual cutting performance of a machine tool could be well simulated with an integrated model of the spindle-tool unit and the machine tool headstock.

In this study, we mainly focused on investigating the influence of a linear guide on the vibration characteristics of a vertical

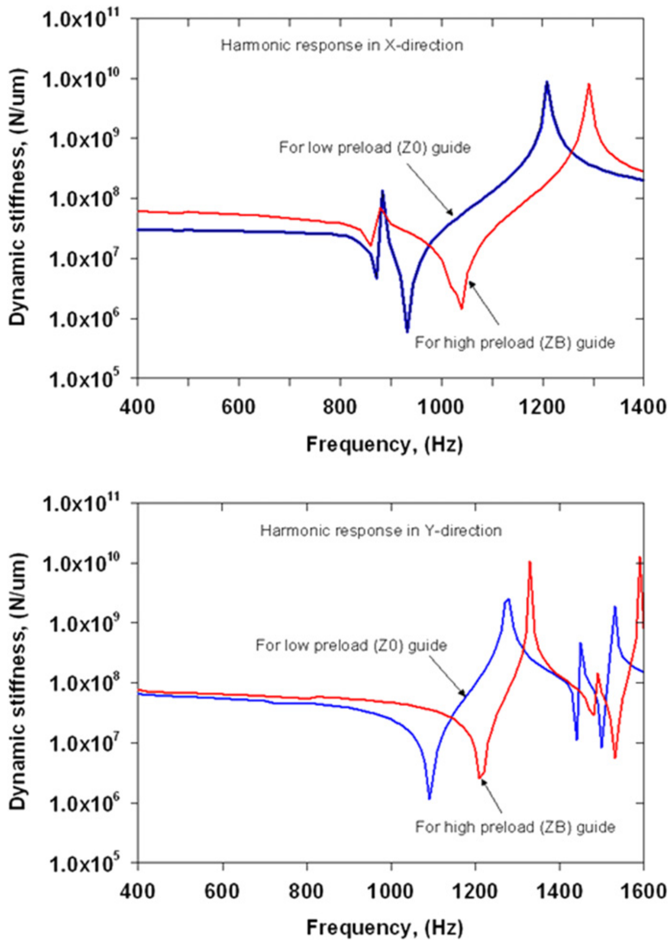

Fig. 4. Harmonic response of spindle head in X and Y directions, which compares the difference between the heads equipped with low and high preloaded guides, respectively.

column–spindle structure. The spindle tool unit was therefore modeled into a simplified cylindrical component of various sections, without considering the spindle dynamic effects induced by the interaction of the bearing stiffness with the rotary inertia of the spindle shaft. As suggested in a study [12], a good model of spindle dynamics greatly depends on the completeness of the analysis model, which should include a machine tool model. Therefore, in order to obtain a more practical and accurate prediction of the dynamic performance of a machine tool, in our future work, a spindle-bearing system will be implemented in the proposed model of a vertical column–spindle system. The results obtained from the integrated model merit further discussions.

5. Conclusions

This study investigated the influence of linear rolling guides on the vibration characteristics of a vertical column structure through finite element analysis and vibration tests. Both the finite element simulations and vibration tests reveal that the linear guide preload greatly affects the vibration behavior associated with a spindle head. The minimum dynamic stiffness in the X and Y directions were found to occur at the yawing and pitching frequencies of the spindle head, respectively, which were governed by the stiffness of linear guides. There is an increase of 10% in the vibration frequency and 120% in the dynamic stiffness when the preload of the rolling guide is adjusted from low to high. As a conclusion, the proposed model has been verified to quantify the influences of the preload of linear guide on the dynamics characteristics of machine tool systems of various configurations.

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