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Prediction of the Dynamic Characteristics of a Milling Machine Using the Integrated Model of Machine Frame and Spindle Unit

Jui P. Hung*, Yuan L. Lai, Tzuo L. Luo and Hsi H. Hsiao

Abstract**—**The machining performance is determined by the frequency characteristics of the machine-tool structure and the dynamics of the cutting process. Therefore, the prediction of dynamic vibration behavior of spindle tool system is of great importance for the design of a machine tool capable of high-precision and high-speed machining. The aim of this study is to develop a finite element model to predict the dynamic characteristics of milling machine tool and hence evaluate the influence of the preload of the spindle bearings. To this purpose, a three dimensional spindle bearing model of a high speed engraving spindle tool was created. In this model, the rolling interfaces with contact stiffness defined by Harris model were used to simulate the spindle bearing components. Then a full finite element model of a vertical milling machine was established by coupling the spindle tool unit with the machine frame structure. Using this model, the vibration mode that had a dominant influence on the dynamic stiffness was determined. The results of the finite element simulations reveal that spindle bearing with different preloads greatly affect the dynamic behavior of the spindle tool unit and hence the dynamic responses of the vertical column milling system. These results were validated by performing vibration on the individual spindle tool unit and the milling machine prototype, respectively. We conclude that preload of the spindle bearings is an important component affecting the dynamic characteristics and machining performance of the entire vertical column structure of the milling machine.

Keywords**—**Dynamic compliance, Milling machine, Spindle unit, Bearing preload.

I. INTRODUCTION

BECAUSE of the demand for high-speed and high-precision machining, machine tools with excellent dynamic performance are being designed and manufactured to meet the requirements for reducing manufacturing costs and realizing multiple industrial applications. At present, CNC milling machines are usually assembled with main modular components: a machine base, saddle, table, vertical column, and headstock with a spindle tool unit. The feeding mechanism of the control axis is constructed in various configurations using linear guides, ball screws, and supporting bearings [1,2], which has been verified to greatly affect the structural dynamic characteristics and machining stability. Apart from the feeding mechanism, spindles, which are usually subjected to dynamic excitation during machining, also plays an important role in machining since they affect surface finish, chatter, tool life, bearing life and noise when excessive vibration was induced.

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In machining practice, the regenerative chatter vibration produced during machining has been recognized as a fatal problem for a machine tool toward high performance. According to studies conducted by Tlusty [3-5] and Tobias [6], the chattering is caused by the dynamic interaction between the cutting tool and the workpiece during the chip generation process. To achieve very high speed machining with high stability, motorized spindles have been developed. Generally, the mechanical part of the spindle assembly consists of hollow spindle shaft mounted to the housing with at least two sets of mainly ball bearing systems, which have greatest influence on the dynamics and lifetime of a spindle. For the bearing system, angular contact ball bearings are most commonly used due to their low-friction properties and ability to withstand external loads in both axial and radial directions.

The dynamic behavior of an existing spindle is most quickly obtained by measuring its frequency response function (FRF) between force and displacement at the tool tip for further identify chatter free cutting conditions in process planning of part machining operations. But for the spindle under developed, it is not possible to assess its dynamic behavior. Therefore, spindle simulation models may a convenient and effective way allowing for the optimization of spindle design parameters either to achieve maximum dynamic stiffness at all speeds for general operation, or to reach maximum axial depth of cut at the specified speed with a designated cutter for a specific machining application. For example, studies [7-11] have revealed that the dynamic behavior of a spindle tool system is significantly affected by the preload state of the supporting bearings and may deviate from the original behavior because the bearing stiffness may vary with changes in the preload under different operating conditions. At high speed operation conditions, the gyroscopic and centrifugal effect on spindle dynamic stability is profound [12-14].

The common approach to chatter prediction is based on calculating only the frequency transfer functions of the spindle and tooling system [15, 16]. However, structural stiffness of the main modularized parts and the stiffness of the interfaces between combined components are predominant factors influencing the overall stiffness of a machine-tool structure. In addition, as mentioned above, the behavior of a spindle tool unit may be in some cases influenced by its interaction with the machine tool frame. Under such conditions, relevant predictions of the spindle dynamic properties can be made using a model of the whole spindle-machine frame system [9]. The coupling of the spindle and machine frame models, possibly complemented with a tool model, also represents an important contribution to the tasks of advanced tool tip movement and workpiece surface quality.

Consequently, it is expected that the effects of bearing preloading on the spindle dynamics and on the dynamic response at the tool tip that is associated with the machine machining performance will be very apparent. However, to the author's knowledge, this effect has not been realized, and hence, it is worthwhile to evaluate the influence of spindle bearing on the dynamic characteristics of a milling machine. The aim of this study was to investigate the machining stability of vertical milling under the influence of a spindle bearing. 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 spindle and vertical column spindle system were constructed respectively to examine the bearing effect. And finally, with the integration of the modeling of spindle unit and machine tool structure, the dynamic harmonic analysis under simulated cutting force conditions was performed in order to assess the dynamic behavior. The simulation results were validated by performing vibration tests on a spindle and machine prototype.

II. Finite Element Modeling of Spindle Unit

A. Modeling of the rolling interface

For a high speed spindle unit and machine tool structure, there are composed of the linear components with rolling balls or roller, other than the spindle housing or machine frame structure. In these components, the interface between rolling balls and ball grooves are critical sites affecting the mechanical characteristics such as the static or dynamic stiffness. The key point for accurate modeling of dynamic characteristics of a high speed spindle unit and machine tool structure is the simulation of the rolling elements within the ball grooves of the linear components. The rolling interface primarily determines whether the simulation results approach the real characteristics of the system [17,18]. Therefore, these linear components should be appropriately introduced into analysis model of milling machine.

Fig. 1 shows the angular contact ball bearing and linear guide, which are all designed with contact profile of a circular arc so as to provide the capabilities to withstand external loads from both different directions when they are properly preloaded. In these components, the contact status between ball groove and rolling ball is a two-point or four-point contact mode, depending on the contact geometry of these linear components. Such contact status actually can be regarded as at Hertzian contact mode and hence 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 [19]. The contact stiffness at the contact point can further be determined using the Harris model in [20]. This model describes the dependence of the bearing internal condition (contact angles, contact forces, contact deformation) on its external condition (external load, external deformation). A brief description as follows:

The relationship between contact force and contact deformation is given as (Fig. 2)

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

where Q denotes the contact force and α is the elastic deformation at the contact point. K_h represents the Hertz constant and can be determined from the following parameters [21, 22]:

$$k = 1.0339\left(\frac{C_y}{C_x}\right)^{0.6360} \tag{2}$$

$$K_h = \frac{\pi k E'}{3f}\sqrt{\frac{2C\varepsilon}{f}} \tag{3}$$

$$f = 1.5277 + 0.6032\,ln\left(\frac{C_y}{C_x}\right) \tag{4}$$

$$\varepsilon = 1.0003 + 0.5968\left(\frac{C_x}{C_y}\right) \tag{5}$$

$$\frac{2}{E'} = \frac{1-\mu_a^2}{E_a} + \frac{1-\mu_b^2}{E_b} \tag{6}$$

In the above equations, E represents Young's modulus; μ denotes Poisson's ratio of the material; and constants C, C_x, C_y are related to the diameter D of the rolling ball and the radius R_i of the raceway, which can be expressed as shown in the following equation, in terms of the form factor f_i of the contact geometry of the raceway.

$$C_x = \frac{D}{2},\; C_y = D\frac{f_i}{2f_i - 1},\; C = \frac{C_x C_y}{C_y - C_x} \tag{7}$$

From eq. (1), the normal stiffness can then be obtained as

$$K_n = \frac{dQ}{d\delta} = \frac{3}{2}K_h\delta^{1/2} = \frac{3}{2}K_h^{2/3}Q^{1/3}\,. \tag{8}$$

As revealed in Eq. (8), the contact stiffness in the normal direction depends nonlinearly on the contact force Q, which is essentially determined by the initial preload set on the rolling balls.

For the creation of the finite element model with linear components, such as bearing and linear guide, in order to reduce the complexity in mesh generation of the ball grooves with rolling balls, , the outer and inner ring was directly connected using a series of spring elements by ignoring the effect of the rolling balls, as shown in Fig.(2).This two-point contact mode greatly increased the efficiency of the analysis model without affecting the accuracy of the results [23].

B. FE model of spindle unit

Fig. 3 shows the high speed engraving spindle considered in this study, in which the spindle shaft with uniform cylindrical cross section is directly coupled with driven motor rotor at a maximum speed of 36000 rpm. The spindle shaft is supported in housing by two bearings BC71903 in front group and two bearings BC7002 at rear end. These angular contact bearings are all mounted in back to back configuration and they can be preloaded to different amount by spring preloaded unit. To investigate the dynamics of the spindle, a solid model of the spindle bearing system was prepared using the finite element analysis software. The FE computational mesh is created as solid hexahedron element for both the shaft and the spindle housing (see Fig. 3).

Regarding the supporting bearings, the outer and inner rings were respectively simplified as a part of the spindle housing and shaft in geometry. A series of spring elements were employed to directly

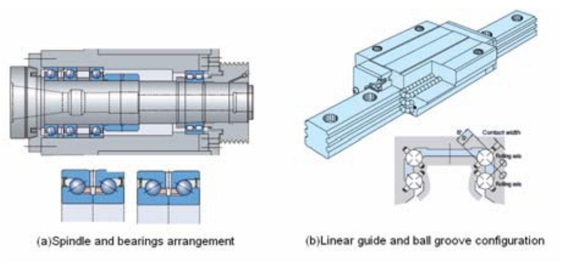

Fig. 1 Configuration spindle bearings and rolling ball grooves within linear guide

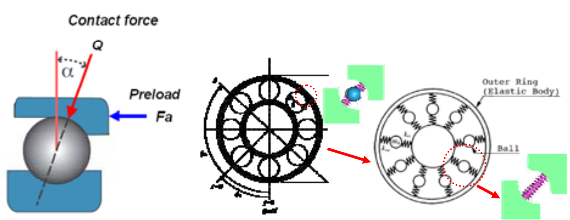

Fig. 2 Contact status between ball groove and rolling ball, and the modeling of the rolling interface.

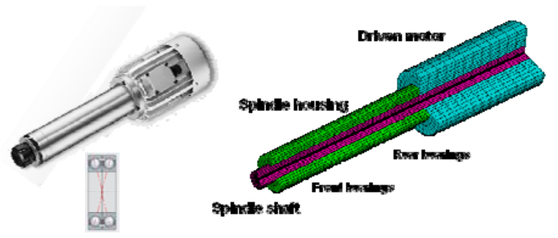

Fig. 3 High speed engraving spindle unit and finite element model, including the housing, shaft and bearings.

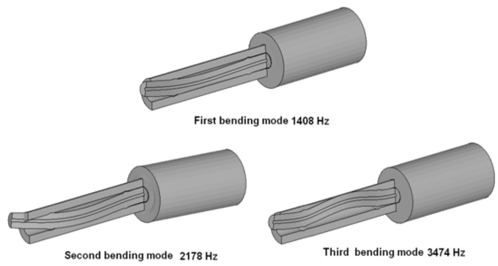

Fig. 4 Fundamental vibration modes and natural frequencies of spindle unit.

connect inner and outer rings, which described the rolling contact characteristics of rolling balls and ball grooves. The driven motor was modeled with cylinder having equivalent weight as the original spindle unit. For the finite element analysis, all the metal components have the following material properties of carbon steel: elastic modulus E = 200GPa, Poisson's ratio $\mu = 0.3$, and density $\rho = 7800 Kg/m^3$.

The use of preloaded bearings is a standard practice to maintain rotational accuracy and sufficient stiffness of spindle shaft in both the radial and axial directions, simultaneously supporting the basic operational requirements for machine tool applications. In this study, in order to examine the effect of the bearing preload on spindle dynamic characteristics, the preload force was set at slight (12N), light (25N), medium (59N) and high (120N) amount for front bearings, and slight (15N), light (34N), medium (69N) and high (150N) amount for rear bearings following the bearing technical guidelines. Bearing stiffness values for each spindle bearing groups under different preload amount were then determined using the above presented Harris's model. The value of the bearing total stiffness is distributed among the spring elements created in the FE model circumferentially surrounding the spindle shaft. This spindle bearing model can also able to take account of variation in the bearing stiffness due to the change in operation conditions. Catalogue data on bearing geometric dimensions were considered in the calculation of the contact stiffness. The dynamic characteristics of the spindle under different bearing preload were evaluated by modal and harmonic analysis.

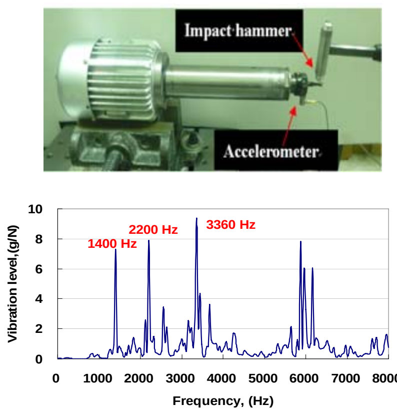

Fig. 5 Vibration testing of spindle unit and the measured vibration spectrum.

C. Dynamic characteristics of the spindle unit

The fundamental vibration modes for spindle shaft with slight bearing preload (12 and 15N for front and rear bearings) are assessed from modal analysis and depicted in Fig.4, which show that the dominant modes are associated with the bending vibrations of the spindle shaft. For each mode, the maximum displacement is found to occur at the spindle tip. The vibration frequencies corresponding to the first, second and third modes are 1408, 2178 and 3474 Hz respectively. The finite element predicted modal frequency was validated by vibration test on spindle unit. Fig. 5 shows the testing configuration of a commercially spindle unit, in which the accelerometer was mounted on the spindle collet to measure the vibration signal excited by the impact hammer at the tool tip. The natural frequencies of the spindle unit were extracted from the recorded FFT diagram, as shown in Fig. 5. It is found from this figure that the spindle unit vibrates at resonant frequencies of 1400, 2200 and 3360HZ, agreeing well with the results of finite element modal analysis. This comparison also validates the accuracy of spindle model created for subsequent dynamic analysis.

The predicted dynamic frequency response of the spindle unit in the radial direction under frequency varying load is illustrated in Fig. 6. It is observed that the second bending mode is the vital mode that may make the spindle more compliant than the others. The computed FRFs of the spindle system with different bearing preload are also illustrated in Fig. 6. The resonant frequency corresponding to this critical mode increased from 2178 Hz to 2280 Hz as the preload of bearings were adjusted from slight to medium amount. In addition, the maximum dynamic compliance of the spindle also decreases from 16.27 N/µm to 13.5 N/µm with the increase of front bearing preload.

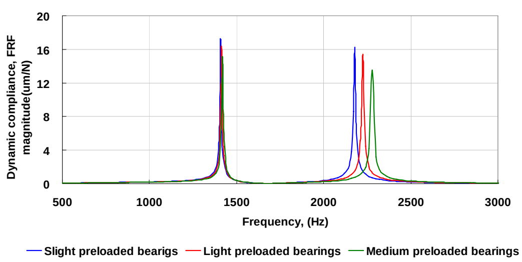

Fig. 6 Computed FRFs of the spindle system equipped with bearings at different preload amounts.

D. Construction of vertical milling machine

A small scaled vertical milling machine was designed and fabricated, as shown in Fig. 7. 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 a pair of linear guide modulus. The ball grooves of linear guide modulus are designed with circular arc profile forming a point contact at an angle of 45° [24]. The sliding blocks of the linear guide was preloaded at slight amount, which is rated at 1.5% of the dynamic load rating (1.138kN). The ball screw driving the stage 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 [25]. Moreover, two standardized ball-screw support units coded EK12 and EF10 were used at both ends of the screw shaft to ensure its rigidity [26]. In addition, the spindle unit analyzed in above section was installed on the spindle feeding stage.

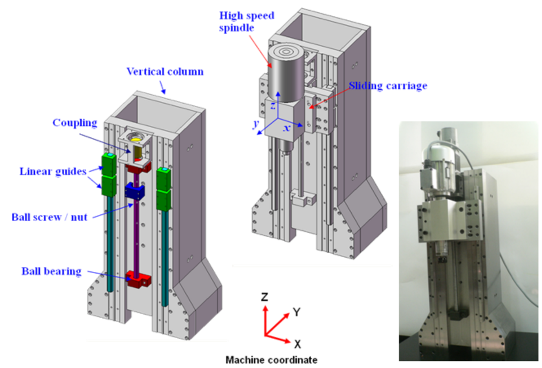

Fig.7 Schematic of the vertical milling machine, including column frame, feeding mechanism and spindle head associated with the spindle unit.

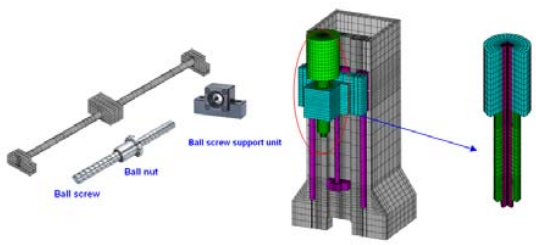

Fig.8 Finite element mode of the machine frame of milling machine integrated with the spindle-bearing model.

E. Finite element model of machine frame

Fig.8 presents the finite element model of a vertical column spindle structure including a motorized spindle tool. Each structural component of the milling system was meshed with eight-node brick elements, with a total of 48757 elements and 58127 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. Following the modeling method for spindle bearings, the four rolling grooves in linear guide were simplified as two grooves, as shown in Fig. 2. The sliding block and guide rail were directly connected using a series of spring elements by ignoring the effect of the rolling balls. The overall spring elements at each ball groove had a contact stiffness equivalent to that of the original guide model, which was calculated as 9.76 N/μm for the specifications of the rolling guides available in reference [24].

In a similar manner, the ball screw was modeled as a cylindrical shaft and meshed with 3D solid elements rather than flexible beam elements. 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 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.9. 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. The contact stiffness at the screw groove was estimated as 152 N/μm based on the specifications of the ball screw, 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 spindle dynamics are more complicated as compared with other linear components and are considered to have an important influence on machining performance of machine tool. To obtain a whole analysis model of a milling machine, the spindle-bearing system as shown in above section was implemented in feeding stock mounted in machine frame, as shown in Fig. 8, The connection between the spindle unit and the spindle head socket on machine frame was assumed in well fixation state.

F. Model validation with vibration test

To validate the finite element model, we first performed modal analysis to obtain the natural vibration characteristics of the vertical milling system and then compared the results with experimental measurements. The natural frequencies of the vertical milling system predicted by modal analysis are listed in Table 1.The fundamental vibration modes were illustrated in Fig. 9.

As was indicated in the finite element simulation [23], the vibration motions associated with the vertical column spindle head are critical modes that lead to a lower dynamic stiffness in the spindle, which are termed as the pitching and yawing vibration modes of the spindle head, respectively. These vibration motions with lower frequencies are also found in current analysis mode. The first and second pitching motions of the spindle head occur at 315 and 401 Hz. The first and second yawing motion occur 373 and 457 Hz, respectively. It is obvious that these lower modes are governed by the linear guides of the feeding mechanism of spindle head [23]. While the higher modes around 1410 and 2075 Hz are mainly associated with the bending vibration of the spindle shaft and tool, which are almost irrespective to the spindle head and slightly affected by the vertical column structure. It was worthy to notice that these higher modes did not appear in model presented in previous study [19] where the spindle unit was simply simulated as a cylindrical structural part, without considering the bearing induced dynamic characteristic which has been demonstrated for the spindle model discussed in this study. But through the use of a 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. 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] = β[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 spindle dominant vibration mode, about 0.30%.

To predict the frequency response function at tool tip, harmonic

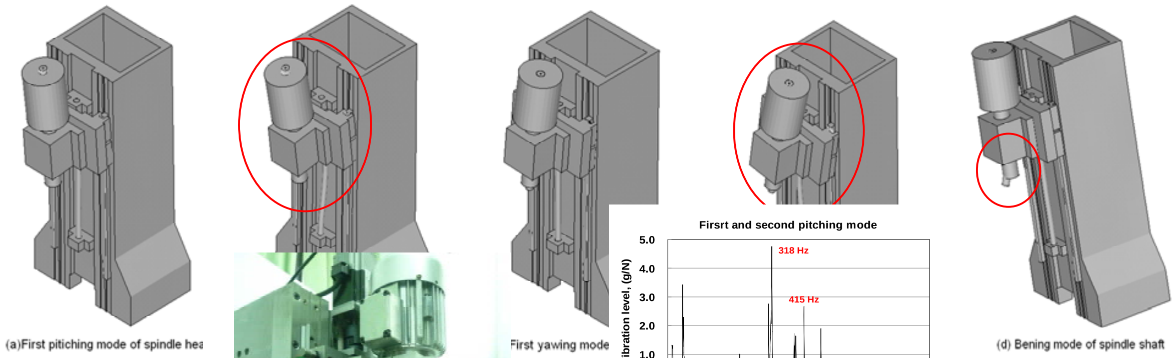

Fig. 9 Fundament em in spindle unit.

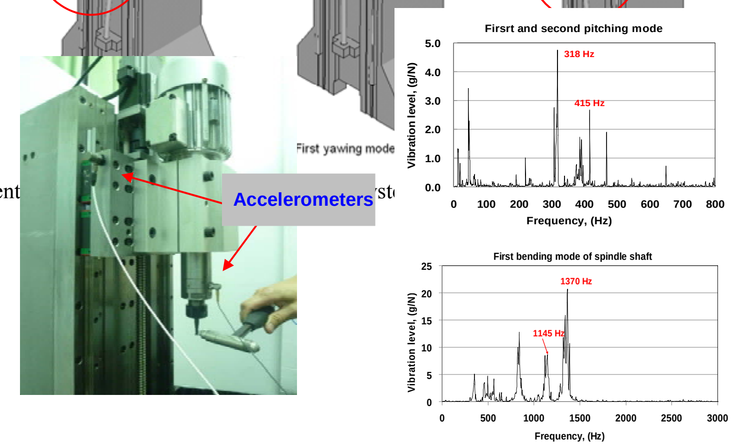

Fig. 10 Vibration test conducted on the vertical column spindle structure, showing the location of accelerometer and impact hammer, with attempt to measure the pitching modes of spindle stage and bending mode of spindle tool.

fully model integrated with machine tool structure and spindle unit, the realistic dynamic characteristics of vertical milling machine system were predicted.

The modal analysis result was verified via a vibration test conducted on the vertical column spindle structure, as shown in Fig.10, 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.10 also depicts the frequency spectra corresponding to the yawing and pitching vibrations of the spindle head and spindle tool tail. The natural frequencies corresponding to the first, second pitching and the yawing vibrations were measured as 318, 415 and 359, 458 Hz, respectively, and the vibration frequency associated with the bending of spindle shaft were about 1370 and 2177 Hz, which are also close to the predicted natural mode of spindle unit. Comparisons with the data in Table 1 show that the finite element predictions agree well with the experimental result. This further confirms the accuracy of the modeling of the vertical milling system integrated with the machine frame and spindle unit.

Prediction of tool-point frequency response

The frequency response functions at spindle tool tip play important roles in the prediction of the structural dynamic behavior and the machining stabilities with or without regenerative chatter. In general, analysis was performed on the finite element model of milling machine by applying a unit force at the spindle tip. Fig.11 illustrates the frequency responses of the spindle tip in the X directions, in terms of the amplitude in logarithmic scale and real part of dynamic compliance respectively, which also demonstrate the preload effect of the spindle bearing. In addition, to illustrate the coupled effect of the machine frame and spindle bearing on tool tip dynamic responses, the result predicted from the model with faked spindle unit was included for comparison. In the faked spindle unit, the spindle-bearing dynamics was not taken into consideration.

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TABLE I COMPARISON OF THE VIBRATION FREQUENCY OF MILLING MACHINE OBTAINED FROM FINITE ELEMENT MODAL ANALYSIS AND VIBRATION TEST (UNIT: HZ)

		VT with spindle-bearing model			VT with simplified cylindrical spindle model
No	Vibration mode	FE modal analysis	Vibration test	Difference (%)	FE modal analysis
1	1st pitching of spindle head	315	318	0.94	319
2	1st yawing of spindle head	373	359	3.90	381
3	2nd pitching of spindle head	401	415	3.37	413
4	2nd yawing of spindle head	457	458	0.22	476
5	Torsion of vertical column structure	648	625	3.68	662
6	Bending of spindle shaft coupled with yawing mode of spindle head	1134	1145	0.96	----
7	Bending of spindle shaft coupled with pitching mode of spindle head	1410	1370	2.92	----
8	Bending modes of spindle shaft	2177	2170	0.3	----

It was found that at the frequency below 1500Hz, both the milling machine models with real spindle unit and faked spindle unit show a similar vibration characteristic. At the lower frequency ranges, the yawing and pitching modes of the spindle head govern the vibration behavior of the milling machine. It is noted that these modes are mainly affected linear guides in feeding mechanism of the headstock, irrespective to the spindle too unit. But, at high frequency ranges (1800-2400Hz), the vertical milling machine with real spindle-bearing model shows a different dynamic response characteristic when compared with the milling machine with faked spindle unit. It can be found from the predicted FRF that the spindle tool unit behaves more compliant when the excited frequency approaches the second natural mode of the spindle tool.

From these figures, we also find that the resonance frequency vertical milling machine will increase with increase of the preload amount of the spindle-bearings; at the same time, the minimum dynamic stiffness of the vertical column -spindle tool system will increase with the preload amount of spindle bearing preload. According to the predicted FRF, the maximum dynamic compliance in X direction was calculated as 18.56, 17.83 and 17.10 μm/N respectively for spindle tool unit with slight, light and medium preloaded bearings. On other respect, despite of the preload amount of spindle bearing, the dynamic compliance induced at the yawing mode are almost identical for different FE model of milling system, about 1.56 μm/N, which is greatly higher than these induced at the bending mode of spindle shaft. This means that when the spindle operates at lower frequency, the linear guide preload will have more influence on the milling system stiffness, to the contrary, when the spindle operates at higher frequency about 2000Hz, the preload of spindle bearing will mainly determine the spindle stiffness as compared with the machine frame structure.

Besides, it is worth to notice that the spindle tool unit installed on the machine frame shows a dynamic behavior different from that of the spindle tool unit at free-free status. This clearly indicates that the machine frame structure and head stock will affect the stiffness of spindle tool unit when it was mounted on the headstock; while appropriate design of machine frame structure with enough structural stiffness can help to increase the overall stiffness of the spindle tool unit. The results of the harmonic analysis indicate that compared with the yawing and pitching modes of spindle head, the bending mode of the spindle tool unit dominates the dynamic characteristics of the vertical milling machine. It can be expected that the spindle tool unit will behave more compliantly when the high frequency bending vibration is induced during machining. This consequently causes the

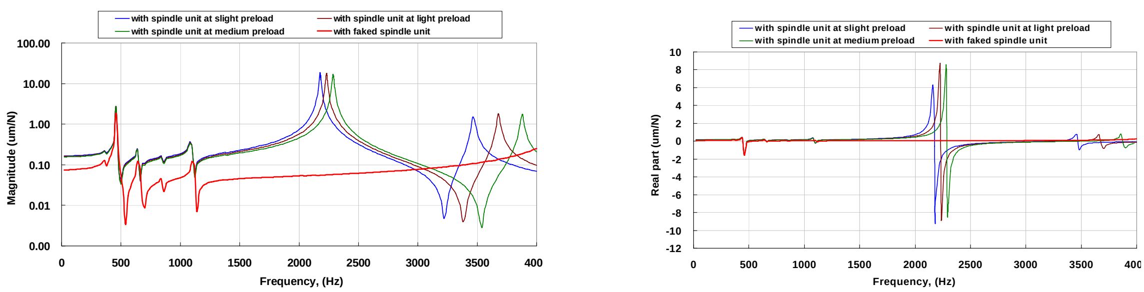

Fig.11 Frequency responses of the spindle tip in terms of the amplitude and real part of dynamic compliance, respectively. The preload of spindle bearings were set at different amount in the finite element model of milling machine.

spindle tool to deform greatly or to vibrate unstably and is likely to degrade the machining performance. According to the analytical machining stability model developed by Altintas and Budak [28,29], the chatter-free axial cutting depths (Z_{min}) is determined by the real part of the frequency response function of the spindle tool tip. As was noticed in Fig. 11(b), the spindle unit with higher preloaded bearings demonstrated a lower value of the real part of the FRF than with lightly preloaded bearings, which clearly implies that a larger cutting depth without causing the chattering can be achieved when the spindle unit was equipped with high preloaded bearings. This can effectively increase the production rate with better machining quality. Based on these results, we can verify that the preload of the spindle bearings is an important component that affects the dynamic characteristics and machining performance of the entire vertical column structure of the milling machine.

III. Conclusions

The purpose of this paper is to develop a dynamic modeling and analysis model of spindle unit integrated with machine frame. With the model, the dynamic characteristics of a spindle tool unit with different bearing preload were analyzed. Current results indicate the bending vibration of the spindle shaft is the main mode affecting the dynamic compliance of the spindle tool unit. Also, the bearing preload was found have a certain influence on the dynamic characteristics of the spindle tool unit. For the investigation of the dynamic characteristics of a vertical milling machine, the spindle-bearing model was incorporated with the machine frame model. This full finite element of a milling machine was successfully verified by the vibration test performed on the prototype machine. Using this model, the realistic dynamic characteristics of the milling machine was demonstrated. Current results show that at the lower frequency, the yawing and pitching modes of the spindle head govern the vibration behavior of the milling machine. It is noted that these modes are mainly affected by linear guides in feeding mechanism of the headstock, irrespective to the spindle too unit. But at high frequency ranges near 2200Hz, the vertical milling machine with real spindle-bearing model show a different dynamic response characteristic when compared with the milling machine with faked spindle unit. It can be found from the predicted FRF that the spindle tool unit behaves more compliant when the excited frequency approaches the second natural mode of the spindle tool. Overall, the model developed in this study can be used to optimize the limit on the machining stability of a milling system by adopting various design configurations of spindle tool unit and further design in bearing arrangement with specification of the spindle unit system.

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