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Effect of the Machine Frame Structures on the Frequency Responses of Spindle Tool

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Abstract**—**Chatter vibration has been a troublesome problem for a machine tool toward the high precision and high speed machining. Essentially, the machining performance is determined by the dynamic characteristics of the machine tool structure and dynamics of cutting process. Therefore the dynamic vibration behavior of spindle tool system greatly determines the performance of machine tool. The purpose of this study is to investigate the influences of the machine frame structure on the dynamic frequency of spindle tool unit through finite element modeling approach. To this end, a realistic finite element model of the vertical milling system was created by incorporated the spindle-bearing model into the spindle head stock of the machine frame. Using this model, the dynamic characteristics of the milling machines with different structural designs of spindle head stock and identical spindle tool unit were demonstrated. The results of the finite element modeling reveal that the spindle tool unit behaves more compliant when the excited frequency approaches the natural mode of the spindle tool; while the spindle tool show a higher dynamic stiffness at lower frequency that may be initiated by the structural mode of milling head. Under this condition, it is concluded that the structural configuration of spindle head stock associated with the vertical column of milling machine plays an important role in determining the machining dynamics of the spindle unit.

Keywords**—**Machine tools, Compliance, Frequency response function, Machine frame structure, Spindle unit

I. Introduction

IN machining practice, the regenerative chatter vibration produced during machining has been recognized as a fatal problem for a machine tool toward high performance. Such chattering phenomenon has been shown to be caused by the dynamic interaction between the cutting tool and the workpiece during the chip generation process according to studies conducted by Tlusty [1-3] and Tobias [4]. Following the dynamic cutting force model developed by Budak et al., [5,6], stability lobes diagrams regulating the machining conditions with or without chattering can be predicted. It is noticed that the chatter-free axial cutting depths is determined by the real part of the frequency response function of the spindle tool tip [5,6]. Therefore, evaluation of the machining performance of a milling machine can be easily achieved in terms of the prediction of the dynamic frequency response of the spindle tool point. On the other hand, currently, machine tool structures are designed with the modularity concept for satisfying the multipurpose or specific industrial applications. A vertical milling machine was constructed with five main modules, including the machine base, saddle, table, vertical column, and headstock with a spindle.

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With the modular design concept, an innovative machine tool can be fabricated by assembling these modularized components by using feeding mechanisms of various structural configurations [7,8]. Before fabricating a prototype for performance identification, the designer should evaluate whether all the design requirements are satisfied in order to achieve the required performance.

However, structural stiffness of the main modularized parts and the stiffness of the joints or interfaces between combined components are predominant factors influencing the overall stiffness of a machine-tool structure. In addition, 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]. 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.

This study was therefore aimed to analyze the machining stability of a vertical milling system using an integrated model of the spindle unit and the machine frame structure. For the creation of the analysis model, the finite element approach is considered as an effective tool. Most of the studies [10-13] employed the reduced finite element model to model the machine frame, in which the linear motion components were neglected or simulated with combination of mass, beam and spring elements, and the spindle unit was modeled as the beam element with spring element as the supported bearings [9,11,14, 15]. In this study, a finite element model of the vertical spindle tooling system was constructed, in which the bearing stiffness of the rolling elements within ball bearings, ball screw and linear guides were taken into consideration. This solid modeling approach can clearly illustrates the real three dimensional structural vibration behaviors of a machine tool system, enabling to get insight into the coupled phenomenon among the frame structure and linear components [16]. The predicted modal frequencies of the dominated modes were validated by performing vibration tests on the machine prototype, respectively. The frequency responses at spindle tip were then evaluated at different resonant modes which could be related to the influences of machine frame and spindle unit.

II. Finite Element Modeling of Milling Machine

A. Construction of vertical milling machine

A small scaled 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 pairs of linear rolling guides were secured on the front plate of the column at a span of 160 mm. The feeding stage of spindle head stock was mounted on the column using two pairs of linear guide modulus, which were secured on the front plate of the column at a span of 160 mm.

In addition, the spindle head stock can be configured as a short or long head with the compact design, as shown in Fig. 1, which was considered to enable the of the spindle head to have different structure rigidity. With these different models, the influence of structural configuration of spindle head stock on the spindle dynamic behavior under of machining loadings was investigated.

The linear components used in feeding mechanism include linear guides and ball screw, which have some features prescribing the contact status as follows. The ball grooves of linear guide modulus are designed with circular arc profile forming a point contact at an angle of 45° [17]. 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 [18]. Moreover, two standardized ball-screw support units coded EK12 and EF10 were used at both ends of the screw shaft to ensure its rigidity [19]. In addition, a commercial high speed engraving spindle unit installed on the spindle feeding stocks for further tests.

B. Finite element model

Fig. 2 presents two finite element models of the vertical column spindle structure with short and long spindle head, respectively. All the structural components of the milling system were meshed with eight-node brick elements, with a total of 39801 elements and 51823 nodes for short spindle head and total of 40343 elements and 51397 nodes for long spindle head. The components of the feeding mechanism, including the ball bearing, ball screw, and ball nut, and the linear guides were included in the models.

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. The contact stiffness at rolling interfaces was defined by Hertz contact theory. Such model was also used to simulate the spindle bearing components. Generally, 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 proposed by Hung et.al., [16], the four rolling grooves in linear guide were simplified as two grooves. 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 [17].

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. 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 [19] 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 [20].

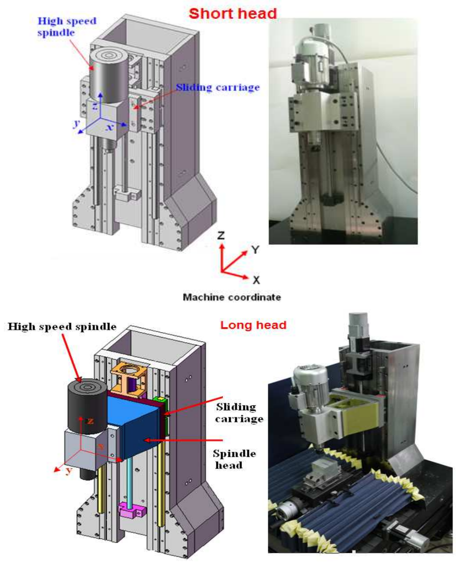

Fig. 1 Schematic of the vertical milling machine, including column frame, feeding mechanism and spindle head associated with the spindle unit. The spindle head can be constructed into short stock or long stock

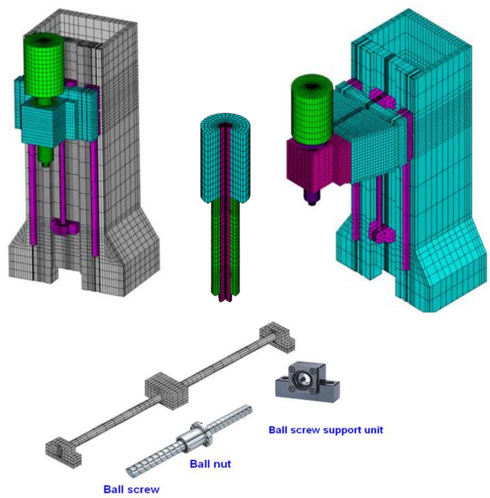

Fig. 2 Finite element models of the milling machine with different spindle head stock, illustrating the machine frame integrated with the spindle-bearing model

C. Model validation with vibration test

To validate the finite element models, we first performed modal analysis to obtain the natural vibration characteristics of the vertical milling systems and then compared the results with experimental measurements. The models for verification tests are implemented with the same specifications in linear feeding mechanism and spindle unit. The fundamental vibration modes of the vertical milling systems with different heads are illustrated in Fig. 3. The natural frequencies of the vertical milling systems with different heads are listed in Table 1 for comparison. As was indicated in the finite element simulation [16], 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. In the case of milling machine with short spindle head, 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. In the case of milling machine with long spindle head, the first and second pitching motions of the spindle head occur at 257 and 334 Hz, and the firs and second yawing frequency is 256 and 357 Hz. respectively. It is obvious that these lower modes are governed by the linear guides of the feeding mechanism of spindle head [16]. Although the two models have the same linear components in feeding mechanism components, we can find that the models with long spindle head stock show lower frequencies as compared to the model with short spindle head stock. Clearly, the structure design in spindle head plays an important role affecting dynamic behavior of the milling head.

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.

III. Frequency Response Function

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, 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. According to the vibration tests, the damping ratio was found to vary with the modal mode, which inherently is determined by the machine structure associated with interfacial joints existing in various components. To get an oversight on the frequency responses of the machine models, in this analysis, the damping ratio employed in harmonic analysis within the frequency ranges (0-4000Hz) was assumed as 0.30% for both short and long head stocks. The harmonic analysis was performed on the finite element model of milling machine by applying a unit force at the spindle tip.

The frequency responses of the spindle tip in the X and Y directions for model with short and long spindle head stock are depicted Fig. 4 and Fig. 5 in terms of the amplitude in logarithmic scale and dynamic compliance of real part, respectively. To get insight into the coupled effect of the machine frame and spindle bearing on tool tip dynamic responses, the result predicted for spindle model without machine frame is included comparison. It is found from these figures that the two machine models behave similar dynamic characteristics within the frequency ranges from 0 to 4000 Hz, which are different from that of the spindle model. In essential, the dynamic behavior of machine model can be characterized by combination of structural modes and spindle modes. At the lower frequency ranges less than 1500Hz, the vibration modes associated with the yawing and pitching motions of the spindle head govern the dynamic behavior of the milling machine, which are features as the structural modes. These modes are mainly affected by structural stiffness of spindle head stock, which are further determined by the interface properties of linear components in feeding mechanism of the head stock. Such modes do not appear in the spindle model. The dynamic response characteristic at high frequency ranges (1500-4000Hz) is apparently governed by the bending stiffness of spindle shaft which is dependent on the bearing stiffness of the supporting bearings and rigidity of spindle shaft. It is also noticed that the machine models show slight difference in the dynamic characteristic of the spindle modes even if they are equipped with the same spindle unit. This clearly shows the contribution of the machine frame to the tool point frequency response.

Regarding to the dynamic compliance, it can be found from the predicted FRF that the spindle tool becomes more compliant when the excitation is close to the spindle modes. The dynamic stiffness induced at lower frequency range, relating to yawing and pitching mode of spindle head stock, is higher than that induced at high frequency, relating to the bending mode of spindle shaft.

For the milling tool with short spindle head stock, the maximum dynamic compliance at yawing mode about X axis is about 3.5 μm/N (460Hz), which is lower than that of the milling tool with long spindle head stock, about 5.9 μm/N (360Hz). For the response in Y direction, the maximum dynamic compliance for milling tool with short spindle head is about 3.01 m/N (400Hz), which is higher than that of the milling tool with long spindle head stock, about 1.6 μm/N (360Hz). This indicates that the milling tool with a short or long spindle head stock substantially behave a different capability in resisting the lower frequency vibration, depending on the mode excited in operation. Considering the dynamic compliance in X direction at high frequency range, the maximum value induced by bending vibration of spindle shaft is calculated as 18.34 μm/N (2074Hz) for short head model and 17.17 μm/N (2156 Hz) for long head stock, respectively.

Regarding to the high frequency compliance in Y direction, the maximum value is predicted as 16.55 μm/N (2201Hz) for model with short head stock and 15.47 μm/N (2152 Hz) for model with long head stock, respectively. These results indicate that the milling tool with a short or long spindle head stock do not show a apparent difference in high frequency response around spindle modes. 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 indeed 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. In addition, 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 spindle tool to deform greatly or to vibrate unstably and is likely to degrade the machining performance. On the other hand, according to the analytical machining stability model developed by Alintas and Budak [5, 6], the chatter-free axial cutting depths is determined by the negative real part of the frequency response function of the spindle tool tip. As is noticed in Fig. 4 and 5, the spindle tool unit demonstrates a smaller value of the real part when it is mounted on the long spindle head stock. This means that it has a greater cutting depth for stable machining; to the contrary, the spindle tool unit mounted on short spindle head stock shows a larger real value in frequency response, leading to smaller cutting depth for stable machining. Basing on these results, we can verify that the structural configuration of spindle head stock associated with the vertical column of milling machine plays an important role in determining the machining dynamics of the spindle unit.

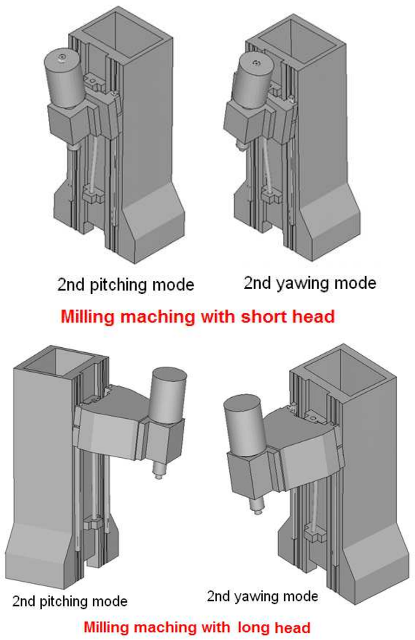

Fig. 3 Fundamental vibration modes associated with the spindle head of vertical milling system with short and long head stock

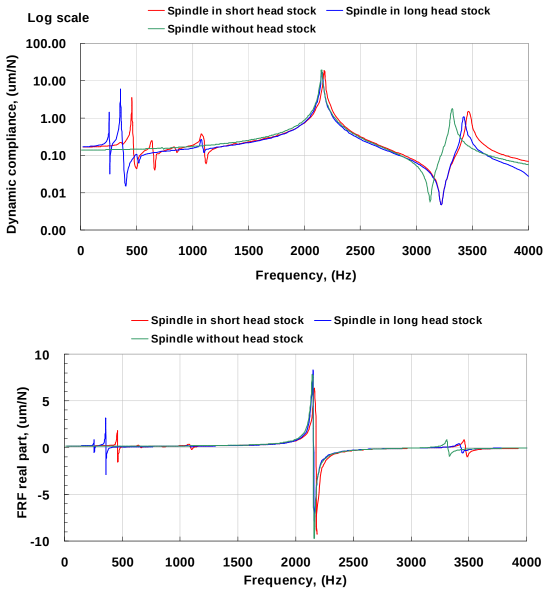

Fig. 4 Frequency responses of the spindle tip in X-direction in terms of the amplitude and real part of dynamic compliance, respectively, which the difference between the machine models with different spindle head stocks

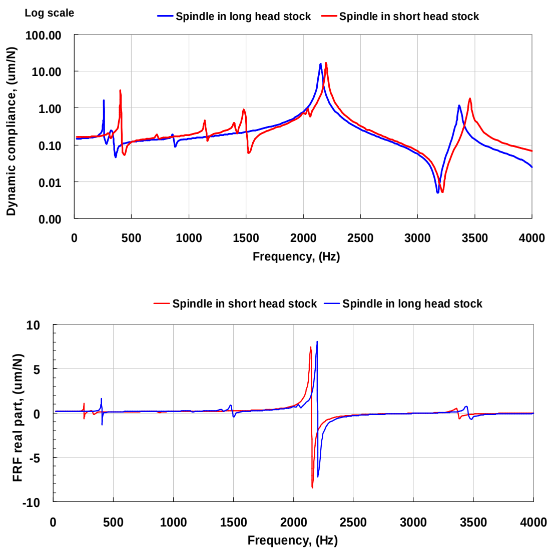

Fig. 5 Frequency responses of the spindle tip in Y-direction in terms of the amplitude and real part of dynamic compliance, respectively, which the difference between the machine models with different spindle head stocks

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

		VT with short spindle head stock			VT with long spindle head		
No	Vibration mode	FE modal analysis	Vibration test	Diff.(%)	FE modal analysis	Vibration test	Diff.(%)
1	1st pitching of spindle head	315	318	-0.95	257	255	0.78
2	1st yawing of spindle head	373	359	3.75	256	250	2.34
3	2nd pitching of spindle head	401	415	-3.49	334	335	-0.30
4	2nd yawing of spindle head	457	458	-0.22	357	355	0.56
6	Bending vibration of spindle shaft coupled with yawing mode of spindle head	1134	1145	-0.97	--	--	--
7	Bending vibration of spindle shaft coupled with pitching mode of spindle head	1410	1370	2.84	--	--	--
8	Bending modes of spindle shaft	2177	2170	0.32	2156	2175	-0.88

IV. CONCLUSIONS

This paper was aimed to investigate the influences of the spindle bearing preload and the machine frame structure on the dynamic frequency of spindle tool unit through finite element modeling approach. To this purpose, we first created a realistic finite element model of the vertical milling system, in which the spindle-bearing model was incorporated into the spindle head stock of the machine frame. 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.

Some key points are drawn from current results as follows:

- At the lower frequency, the yawing and pitching modes of the spindle head govern the vibration behavior of the milling machine, which is mainly governed by linear guides in feeding mechanism of the headstock. The dynamic compliance induced by yawing mode is slightly affected by the structural geometry of spindle head stock, irrespective to the spindle too unit and hence the preload amount set on the spindle bearings.
- Under low frequency machining conditions, the spindle tool unit mounted on the short spindle head stock can behaves better dynamic characteristics, which is favorable to the heavy machining operation, as compared with that mounted on long spindle head stock.
- At high frequency ranges near 2200Hz, the vertical milling machine behaves more compliant when the excited frequency approaches the natural mode of the spindle tool regardless the structural geometry of spindle head stock.

Finally, the proposed analysis model appropriately addresses the importance of the structural design in developing a milling machine with required machining performance.

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