Effect of Drawbar Force on the Dynamic Characteristics of a Spindle-Tool Holder System

Jui-Pui Hung, Yu-Sheng Lai, Tzuo-Liang Luo, Kung-Da Wu, Yun-Ji Zhan

Abstract**—**This study presented the investigation of the influence of the tool holder interface stiffness on the dynamic characteristics of a spindle tool system. The interface stiffness was produced by drawbar force on the tool holder, which tends to affect the spindle dynamics. In order to assess the influence of interface stiffness on the vibration characteristic of spindle unit, we first created a three dimensional finite element model of a high speed spindle system integrated with tool holder. The key point for the creation of FEM model is the modeling of the rolling interface within the angular contact bearings and the tool holder interface. The former can be simulated by a introducing a series of spring elements between inner and outer rings. The contact stiffness was calculated according to Hertz contact theory and the preload applied on the bearings. The interface stiffness of the tool holder was identified through the experimental measurement and finite element modal analysis. Current results show that the dynamic stiffness was greatly influenced by the tool holder system. In addition, variations of modal damping, static stiffness and dynamic stiffness of the spindle tool system were greatly determined by the interface stiffness of the tool holder which was in turn dependent on the draw bar force applied on the tool holder. Overall, this study demonstrates that identification of the interface characteristics of spindle tool holder is of very importance for the refinement of the spindle tooling system to achieve the optimum machining performance.

Keywords**—**Dynamic stiffness, Drawbar force, Interface stiffness, Spindle-tool holder.

I. Introduction

RECENTLY there has been rapid increase in need for high speed, high precision, and high efficiency machining. This also prompts the development of high speed spindle tool system to be applied on machine tool with high speed machining ability [1], [2]. In practice, the machining performance is determined by the interaction of the dynamic characteristics of the machine-tool structure and the dynamics of the cutting process [3], [4]. Also the machining stability is greatly dominated by the dynamic characteristics of the spindle tool system. Therefore, understanding the factors affecting dynamic behavior of a spindle tooling system and the affecting factors is a perquisite in dominating the final machining performance of machine tool system [5], [6]. The influencing factors consist of bearing preload, interface property between spindle and tool holder associated with drawing bar system. However, the more important thing is the realizations on these effects caused by these factors should be established on the machining system constructed by the machining frame structure and spindle tooling system.

In essential, the dynamic characteristics at the tip of the tool are affected strongly by the characteristics of the connection between the tool and the tool holder, and between the tool and the spindle [7]. Considerable efforts have been made recently to the measurement and computation of the interfacial characteristics of various tool holder-spindle interfaces [7]-[11]. Levina [7] studied the effects of angular deformations in the spindle–tool holder interface on deflection at the tool tip. Generally, the too holder is fixed inspindle nose through the drawbar mechanism, which generated a contact bonding status determined by the drawbar force. Levina [10] investigated the effect of drawbar force and taper tolerance on the static stiffness of the tool holder–spindle connection. Smith et al. [12] also showed that increased drawbar force increases the static stiffness of the tool holder–spindle interface, at the expense of reduced damping.

On the other hand, for the design of a milling tooling system with better performance and efficiency, the dynamic behavior of a spindle tool unit was concerned and should be adjusted by means of the modeling technology on the assemblage model of a spindle-tool holder-tool unit. Basically, this requires a fully modeling of the coupling interface between the cutter and the tool holder and between the tool holder and spindle, apart from the bearing groups. In general, the required tool holder–spindle machine dynamics can be obtained by modal testing, but time- and cost-consuming associated with this complicated task should be overcome for a large number of tool–holder combinations in typical production facilities. A number of significant developments have been completed to improve the tool and holder modeling techniques and understand the connection stiffness and damping behavior [13]-[18]. For example, Schmitz and Donaldson [13], [14] implemented the receptance coupling theory of structural dynamics by using experimentally obtained spindle–holder dynamics and analytically obtained tool dynamics for the prediction of the FRF. Ertürk et al. [17], [18] analyzed the effects of bearing supports and spindle-holder and tool-holder interfaces on the FRF, and suggested a fast and accurate approach for the identification of connection parameters.

Although the effects of tool holder interface properties on the dynamic behavior are demonstrated previously in the literatures [13]-[15], quantifications of the effects with the adjustment of the drawbar force on the modal parameters associated with spindle frequency responses is worthy of further study. This

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study was aimed to identify how the changing of drawbar force affects the dynamic characteristics of a spindle tool. The results are expected to provide a reference for setting the tool holder-spindle dynamic characteristics. In addition, we employed the finite element method to modeling the dynamic behavior of the spindle-tool holder unit. With experimental measurements as a validation, the proposed model can be applied for further development of the whole machine tool integrated with the spindle tool and the machine frame structure model.

II. Experimental Measurements

In this study, the high speed spindle (DDS BT-30) was employed for tests, in which the front and rear bearing sets coded NSK-7008C were mounted in DBB arrangement. Fig. 1 illustrates the experimental configuration of the spindle unit without tool holder. The spindle was assembled with specific spacer between the front and rear bearing groups for higher preloaded stiffness. It was then suspended by two wire ropes for test. An accelerometer mounted on the spindle nose was used to measure the vibration signal excited by the impact hammer at opposite side. The dynamic response was then extracted from the recorded FFT spectrum.

For realizing the contribution of the tool holder to the dynamic behavior of a spindle unit, this spindle was then assembled with a tool holder (BT30) under standard drawbar force of 260 Kg. Following the same procedures, the dynamic frequency response functions of the spindle with tool holder were measured.

The measured vibration responses of the spindle units with and without tool holder are illustrated in Fig. 2 for comparison, which are expressed in terms of the compliance varying with frequency. It is found from Fig. 2 that the spindle unit without tool holder behaves three fundamental modes at about 825, 1707 and 3336 Hz. For the spindle with a tool holder, the first four vibration frequencies for the spindle tool with standard drawbar force are 814, 1314, 2404 and 4008 Hz, respectively. The fundamental modes could be identified as the bending vibrations of the spindle shaft according to finite element modeling of the spindle system in next section. It is noticed that spindle without tool holder shows a lower compliance of 0.015 um/N at first vibration mode. For spindle with tool holder, the compliance is about 0.75 um/N, which is greatly higher than that of the spindle without tool holder, indicating the influence of the tool holder.

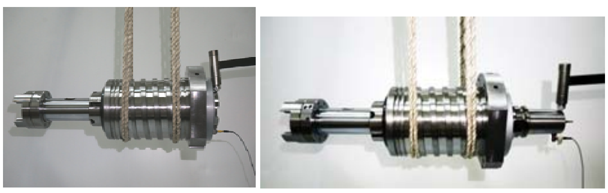


Fig. 1 Experimental configuration of vibration tests

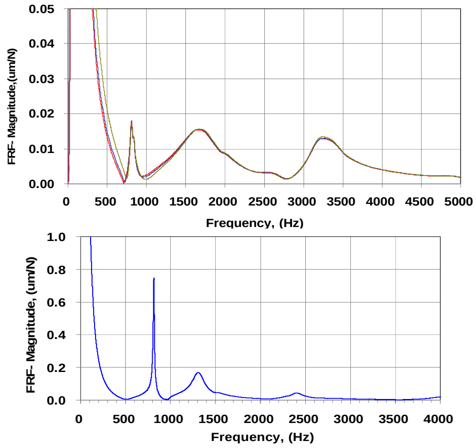


Fig. 2 Frequency response functions measured for spindles without and with tool holder

III. Modeling of the Spindle-Tool Holder System

To investigate the dynamic behavior of the spindle, a finite element model of the spindle unit including the rotating shaft, the spindle housing and supporting bearings was created. There exits rolling interfaces in bearing between rolling balls and raceways, which primarily contributes to the dynamic characteristics of spindle tool. Therefore, modeling of the rolling interface is the prerequisite for the creation of the analysis model and accurate evaluation of the dynamic characteristics of the spindle tool.

A. Modeling of Angular Contact Bearing

As shown in Fig. 3, the contact force between the rolling ball and the raceway can be related to the local deformation of contact point by the Hertzian expression [16] as follows:

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

where Q denotes the contact force and δ 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. Details are available in the literature [17]. The normal stiffness at a specific preload can then be obtained as

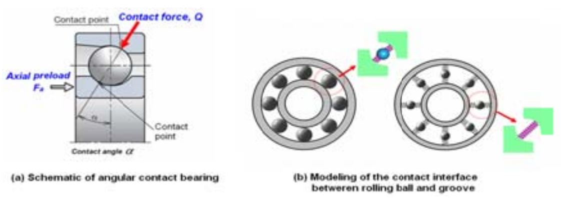


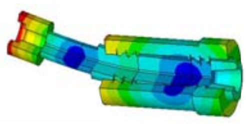
Fig. 3 Modeling of the angular contact bearing

$$K_n = \frac{dQ}{d\delta} = \frac{3}{2} K_h \delta^{1/2} = \frac{3}{2} K_h^{2/3} Q^{1/3} \tag{2}$$
$$Q = F_a / N \cdot sin\alpha$$

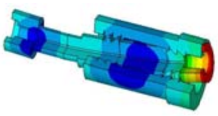
B. Finite Element Model of Spindle without Tool Holder

To investigate the dynamic behavior of the spindle with finite element approach, a solid model of the spindle bearing system including the rotating shaft and the spindle housing was created and meshed with hexahedron elements (see Fig. 4). This model has 43224 elements and 49564 nodes. To reduce the complexity in model creation and mesh generation of the motion components, such as the ball bearing, we modeled the contact configuration between rolling balls and the raceway as a two-point contact mode. For the supporting bearings, the outer and inner rings were respectively simplified as a part of the spindle shaft and housing in geometry, respectively. The inner and outer rings are directly connected using a series of spring elements by neglecting the effect of rolling balls, as shown in Fig. 3.

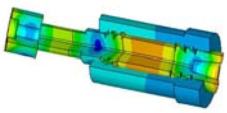
(a) Finite element model of spindle without tool holder



(b) First mode



(c) Second mode



(d) Third mode

Fig. 4 Finite element model of the spindle without tool holder and the first three fundamental vibration modes

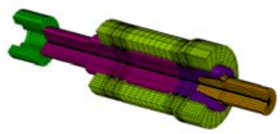
For the tested spindle, the overall contact stiffness of each bearing was calculated as 264N/μm according to the equations (1) and (2). The stiffness of each bearing is distributed on the spring elements circumferentially surrounding the spindle shaft created in the FE model. For finite element analysis, all the metal components have the following material properties of carbon steel: elastic modulus E = 200GPa, Poisson's ratio μ = 0.3, and density ρ = 7800Kg/m3.

For validation of the finite element model, the modal analysis and harmonic analysis were performed, respectively. The fundamental vibration modes of the spindle without tool holder were assessed under free-free boundary conditions, as depicted in Fig. 4. It is found that the fundamental modes are associated with the bending vibrations of the spindle shaft. The frequency response function at spindle nose was also illustrated in Fig. 2 for comparison with experimental measurement. The natural frequencies associated with the first three vibration modes are 812, 1743 and 3441 Hz, respectively, agreeing well with the experimental measurements (825, 1707 and 3336 Hz.). The proposed analysis model of the spindle can appropriately predict the dynamic behavior of the spindle tool system.

C. Finite Element Model of Spindle with Tool Holder

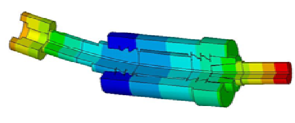
Again, a finite element model of the spindle with a tool holder was generated by introducing the model of tool holder into the previous spindle model, totally 47832 elements and 54850 nodes, as shown in Fig. 5. In this case, the interface with the associated mechanical properties between the spindle nose and tool holder were mainly concerned, and was simulated by using surface to surface interface elements with the adequate contact stiffness, which was believed to contributes to the dynamic characteristics of spindle tool. Basically, the magnitude of contact stiffness was determined by the interface contact force that was generated by the drawbar force.

The modal shapes of the spindle-tool holder unit are depicted in Fig. 5, which are associated with bending vibrations of the spindle shaft and the tool holder. The maximum displacement of each mode occurs at the end of tool holder. Especially, the mode shape of the second mode clearly show the relative separation motion occurs at the interface between the tool holder and spindle nose, which also indicates the dominance of the interface stiffness on the vibration frequency. The actual interface contact stiffness was identified through the parameter identification method by comparing the vibration frequencies obtained from vibration tests and finite elements simulations. With this identified value of the interface stiffness of 15 N/um, the first four vibration frequencies for the spindle tool with standard drawbar force are 807, 1243, 2373and 3821 Hz, respectively.

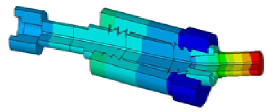


(a) Finite element model

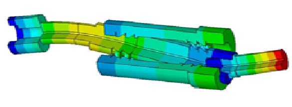
(b) Standard tool holder



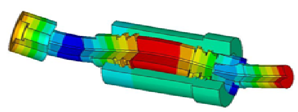
(c) Mode 1 (807Hz)



(d) Mode 2 (1243Hz)



(e) Mode 3 (2373Hz)



(f) Mode 4 (3821Hz)

Fig. 5 Finite element model of the spindle with tool holder and the first four fundamental vibration modes

IV. Effect of Draw Bar Force

As demonstrated in previous section, the addition of tool holder in spindle nose alter the dynamic behavior of the spindle unit, in which the tool holder-spindle interface plays an important role on the compliance of the tool; while the interface stiffness was determined by the drawing force applied on tool holder for fixation. To investigate the influence, we further conducted the vibration tests on the spindle tool holder system with different drawbar force. The drawbar force was set as 160, 210, 260, 340 and 380 Kg, respectively. For each case of spindle–tool holder unit, the dynamic frequency response functions measured at the tool tip were measured following the previous procedure.

A. Experimental Evaluations

Fig. 6 presents the dynamic frequency response functions measured for spindle-tool holder with different drawbar forces. As shown in the figure, the spindles behavior similar dynamic responses, in which the compliance at first modal is greatly affected by the drawbar force on tool holder, but modal frequency remains unchanged. The second mode was found to occur at different frequency that is affected to change with the drawbar force on tool holder.

For each spindle samples, the modal characteristics associated with the first two modes including the modal frequency, modal damping and compliances are extracted. The relationship between the drawbar force and the modal parameters are presented in Fig. 7. As shown in Fig. 7 (a), the second modal frequency is positively affected by the drawbar force (R^2=0.6597). This implies that the interface stiffness between tool holder and spindle nodes increases with the increasing drawbar force, about by 6% with respect to increment of the drawbar force by 52%. As shown in Fig. 7 (b), it is noticed that the first modal damping ratio is much less than the second mode and is slightly affected by the drawbar force. The second modal damping ratio is negatively related to the drawbar force (R^2=0.7818). With the increasing drawbar force from 160 to 380Kg, the damping ratio associated with the second mode decreases from 5.64 to 4.68.

As shown in Fig. 7 (c) the drawbar force has a negatively influence on the dynamic stiffness of the first and second, (R^2=0.5104, 0.7951), with variations of23% and 17% with respect to increment of the drawbar force by 52%. The negative effect of the drawbar force on the dynamic stiffness of spindle tool holder system can be ascribed to the fact that the decrement of damping ratio is much more than that increment of the interface stiffness caused by the increasing drawbar force. In other words, increasing the drawbar force on the tool holder may not enhance the dynamic stiffness of the spindle tool unit. Since the second mode is governed by the interface properties between the tool holder and spindle nose, the effect of the drawbar force on the second modal damping ratio seem to be more important than on the first mode. This experimental investigation also implies that the variation of damping ratio with the varying drawbar force dominate the dynamic behavior of the spindle tool system.

B. Finite Element Modeling

The effect of drawbar force on the dynamic characteristics of spindle unit has been investigated the through the vibration tests. Also, the dynamic behavior of a spindle- tool holder unit was also successfully simulated, in which the modeling of the ball bearings and the interface characteristics between the tool holder and spindle nose is of importance. While, the tool holder interface characteristics are identified through the validation with the experimental measured dynamic characteristics of physical spindle. Generally, such interface characteristics is greatly determined by the force applied on the tool holder. In this section, the effect of the drawbar force associated with its variation can also be investigated through the introduction of the tool holder interface characteristics into the analysis model.

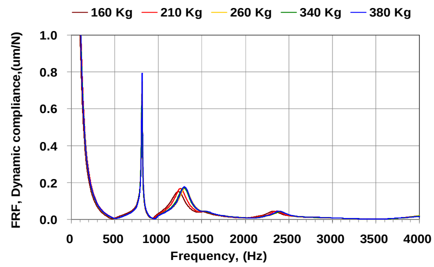


Fig. 6 The dynamic frequency response functions measured for spindle-tool holder with different drawbar forces

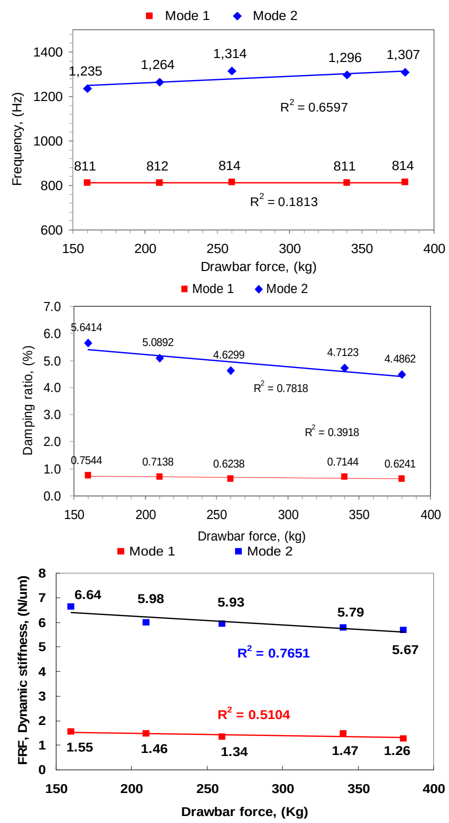


Fig. 7 Variations of the modal parameters of the spindle-tool holder system with different drawbar forces, (a) modal frequency (b) modal damping ratio (c) dynamic stiffness

In analysis, each spindle tool holder system with different drawbar force was modeled using the same finite element model established in Section III, but the interface characteristics were estimated by adequately adjusting the interface stiffness based on the values for standard drawbar force. The natural frequencies predicted under a specific drawbar force were compared with experimental measurements. Following this, the tool holder interface characteristics for each drawbar force were identified, as summarized in Table I, which shows the predicted dynamic frequencies of a spindle tool unit under different interface stiffness associated with draw bar force.

TABLE I
NATURAL FREQUENCIES OF A SPINDLE-TOOL HOLDER UNDER DIFFERENT DRAWBAR FORCE AND THE CORRESPONDING VARIATION OF INTERFACE STIFFNESS

Mode	Drawbar force (Kg) Variation of interface stiffness				
	160	210	260	340	380
	-28 %	-14 %	0	14 %	28 %
1	804	806	807	809	810
2	1208	1227	1243	1261	1274
3	2337	2356	2373	2391	2403
4	3792	3807	3821	3840	3855

Variation of interface stiffness was estimated based on the standard drawbar force (260Kg).

To assess the dynamic frequency response of the spindle unit with different drawbar forces, we performed the harmonic analysis. The results are presented in Fig. 8, which are expressed in terms of the dynamic compliance as a function of the frequency. Comparison between Figs. 6 and 8 shows that the dynamic frequency response predicted from finite element model of the spindle tool system agree well with those obtained from measurements on physical spindle.

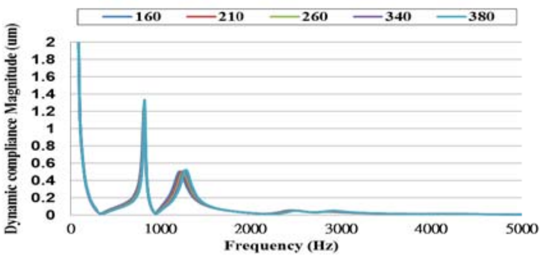


Fig. 8 Predicted frequency response functions of the spindle-tool holder system with different drawbar forces

Tables II and III present the modal parameters associated with the first and second modes for spindle unit under different interface characteristics and drawbar force. The modal frequency of the first mode was not affected by the interface stiffness, while the dynamic stiffness decreases with increasing interface stiffness. For the second mode, the modal frequency and dynamic stiffness are affected to decrease with the increasing interface stiffness. As observed in experiments, the effect of the drawbar force on the dynamic behavior of a spindle tool holder system also successfully investigated.

TABLE II
MODAL PROPERTIES OF THE FIRST MODE

Draw bar force (Kg)	Modal Frequency (Hz)	Dynamic stiffness (N/μm)
160	804	0.809
210	806	0.796
260	807	0.785
340	809	0.779
380	810	0.750

TABLE III
MODAL PROPERTIES OF THE SECOND MODE

Draw bar force (Kg)	Modal Frequency (Hz)	Dynamic stiffness (N/μm)
160	1208	1.9844
210	1227	1.9787
260	1243	1.9779
340	1261	1.9406
380	1274	1.9156

V. CONCLUSION

The study has investigated the influence of the tool holder interface stiffness on the dynamic characteristics of a spindle tool system. The interface stiffness was produced by drawbar force on the tool holder. Based on the results, the following conclusions are presented:

1. The adjustment of drawbar force from a smaller to larger value produce a negative influence on the dynamic stiffness and damping ratio associated with the dominated modes of a spindle tool.
2. A lower drawbar force increases the interface stiffness, but decreases the damping ability and the dynamic stiffness of the dominated vibration mode, which may enable the spindle tool to be favorable for high speed and precision machining with stability. To the contrary, a higher drawbar force increases the interface stiffness and the rigidity of the spindle tool, which is appropriate to application in heavy machining.
3. The influence of the drawbar force on the dynamic characteristics of the spindle can also be investigated through the finite element model proposed in this study. The prediction accuracy is determined by the modeling of the rolling interface of ball bearings in spindles and tool holder interface, in which the bearing preload and contact stiffness can be calculated based on the Hertz contact theory, but the tool holder interface stiffness can be obtained through identifications method with the results of the vibration tests of a physical spindle.

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