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Variation of the Dynamic Characteristics of a Spindle with the Change of Bearing Preload

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***Abstract*—**This paper presents the variation of the dynamic characteristics of a spindle with the change of bearing preload. The correlations between the variation of bearing preload and fundamental modal parameters were first examined by conducting vibration tests on physical spindle units. Experimental measurements show that the dynamic compliance and damping ratio associated with the dominating modes were affected to vary with variation of the bearing preload. When the bearing preload was slightly deviated from a standard value, the modal frequency and damping ability also vary to different extent, which further enable the spindle to perform with different compliance. For the spindle used in this study, a standard preload value set on bearings would enable the spindle to behave a higher stiffness as compared with others with a preload variation. This characteristic can be served as a reference to examine the variation of bearing preload of spindle in assemblage or operation.

***Keywords*—**Dynamic compliance, Bearing preload, Modal damping.

I. Introduction

THE spindle tool system is the most important component for a machine tool executing the machining operation. For various application areas, a wide variety of spindles of different designs have been developed [1], [2]. In machine tool spindle 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. In addition, the bearing groups could be adequately preloaded to increase the rigidity of spindle unit. For the machining of large frames in auto industry, spindles were designed with high power and high rigidity, which requires a heavy preload applied to the bearings. However, several studies [3]-[6], Gunduz et al. [7] have revealed that the dynamic behavior of a spindle tool system is significantly affected by the preload state of the supporting bearings.

Recently, in order to meet requirements for machining precision in airplane and semiconductor industries, spindles with high speed and high precision are developed. But occurrence of chatter vibration during machining was a fatal problem for a machine tool toward high performance. Studies [8], [9] reported that chattering is caused by the dynamic interaction between the cutting tool and the workpiece during the chip generation process. Also the machining stability is greatly determined by the dynamic characteristics of the spindle tool system. Therefore, for the design of spindle with high machining performance, the machining stability over a wide operation range should be taken into consideration. Under this condition, the setting amount of preload to the bearings is of importance to ensure the spindle with enough dynamic stiffness and damping capability.

On the other hand, for different machining application with better efficiency, the dynamic performance should be adjusted by applying appropriate preload to the bearing groups during assemblage of a spindle unit. Basically, two preloading methods are commonly adopted, that is, constant pressure preload (spring preload) and position preload (fixed preload). The latter is the most common preload arrangement, which can be easily arranged using spacers with proper size between two bearing sets. As shown in Fig. 1, the axial preload is determined by the clearances (a) between the inner spacer and inner ring of bearing and clearance (b) between the outer spacer and outer ring of bearing. Reducing the inner ring spacer can increase preload, while reducing the outer ring spacer decreases the preload. For example, as suggested in the bearing user guidelines [10], if the preload of the 7914 angular contact bearing needed to be changed from light to medium, the inner ring spacer length would need to be reduced by adequate size, about several micrometers. However, as discussed in studies [11]-[13], increasing the bearing preload can increase the bearing stiffness, but also reduces the damping ratio of the bearings. This would be unfavorable for suppression of the chattering in high speed machining.

On other respect, the bearing preload could be deviated from initial value with little fluctuation because of the size variation of the rolling balls due to the temperature rise in high speed operation or surface worn of ball grooves after long term operation. This may cause unexpected influence on the dynamic characteristic of spindle. Although the effect of preload variation on natural frequencies and dynamic stiffness are demonstrated previously in the literatures, quantifications of the influence of the preload variations on the modal parameters associated with spindle frequency responses is worthy of further study.

This study was aimed to identify how the changing of spacer clearance affects the dynamic characteristics of a spindle tool. In particular, we intended to examine the correlations between the spacer clearance and modal parameters by experimental approaches. The results are expected to provide a reference for tuning or examine the variation of the dynamic characteristics of the spindle.

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II. Experimental Measurements

A. Adjustment of Bearing Preload

In this study, a commercial high speed spindle (DDS BT-30) was employed for tests, in which the front and rear bearing sets coded NSK-7008C were mounted in DT arrangement for higher stiffness. Each spindle unit was respectively preloaded at different amounts by using the spacers with different clearance, as shown in Fig. 2. The clearance is determined by the difference of axial length between the outer and inner spacers, irrespective to the clearance in radial direction. The spacer clearance can be set at different amount to preload the bearings in axial direction at different amounts. Here, according to the manufacturer, a standard medium preload was set on bearings in spindle by using a spacer clearance of 17 μm. For the other two spindles, the bearing preload were individually slightly increased and decreased by using two different spacers of different clearances, respectively, which can also be employed to simulate the variation of bearing preload due to unexpected tolerance in assemblage.

B. Experimental Configuration

Fig. 2 illustrates the experimental configuration of the spindle unit without tool holder (BT30). Each spindle was assembled with specific spacer between the front and rear bearing groups. It was then suspended by two wire ropes for test. The accelerometers were mounted on the spindle nose, spindle tail and spindle housing, respectively to measure the vibration signals excited by the impact hammer at the opposite side of spindle nose. The dynamic responses were then extracted from the recorded FFT spectrum. The modal parameters such as damping ratio and dynamic stiffness associated with the dominant vibration modes were extracted from the measured FRFs.

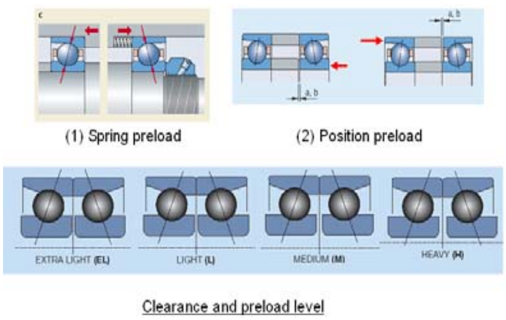

Fig. 1 Preloading method of bearings [10]

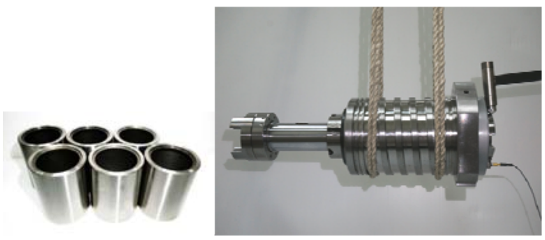

Fig. 2 Bearing spacers and configuration of vibration test

III. Results and Discussions

A. Frequency Response Functions

The measured vibration responses of the spindle units with different preload are illustrated in Figs. 3-5 for comparison, which are expressed in terms of the compliance varying with frequency. The frequency responses shown in Fig. 3 were measured at the spindle nose. It can be found that the spindles with different preload amounts show similar vibration behaviors. For spindle with standard preload, it shows four fundamental modes at about 319, 790, 2875 and 3346 Hz. For the other two spindles, the natural frequencies are found to shift slightly to a higher or lower value due to the preload variation in bearings. Especially, variation of the natural frequency and vibration amplitude associated with the third and fourth modes were more apparent than the first and second modes. According to the study of Hung et al., [14] the vibrations modes were mainly caused by the bending deformation of the spindle shaft, as shown in Fig. 6.

The frequency responses shown in Fig. 4 were measured on the housing near the spindle nose, which also show the four vibration modes as was found in Fig. 3. It is clearly found that the second mode have an apparent peak as compared with others, which indicates this mode is mainly caused by the vibration of housing with respect to the spindle shaft. In addition, the frequency and the compliance of the second mode are almost not affected by the variation of the bearing preload.

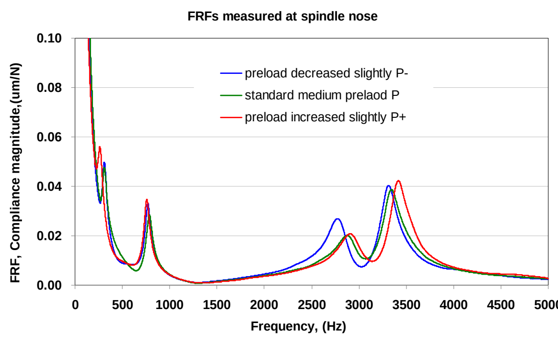

Fig. 3 Frequency response functions measured at spindle nose

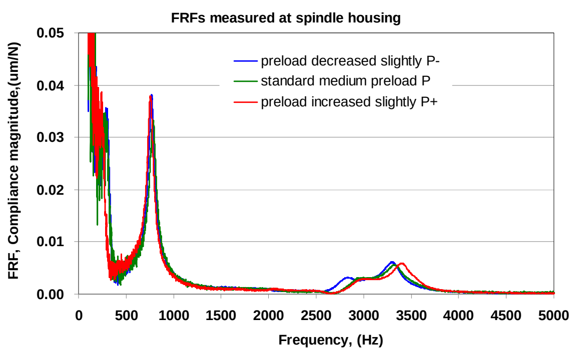

Fig. 4 Frequency response functions measured at the spindle housing

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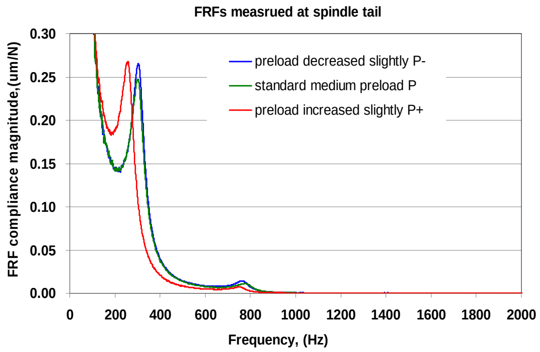
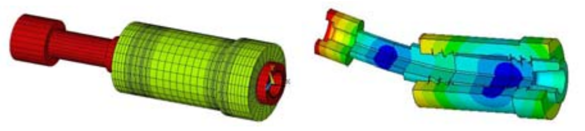
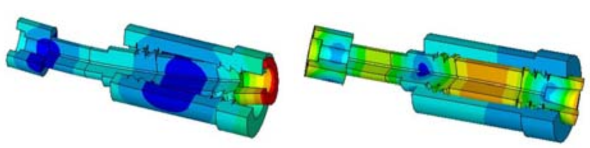


Fig. 5 Frequency response functions measured at the spindle tail



(a) Finite element model of spindle (b) First mode



(c) Second mode (d) Third mode

Fig. 6 Fundamental vibration modes of a spindle [14]

The frequency responses shown in Fig. 5 were measured at the coupling at spindle tail. As was found in Fig. 5, the first mode shows apparent vibrations peak with compliance of 0.25 μm/N approximately. This is due to the bending deformation of the spindle shaft caused by the impact at the spindle nose. The frequency of this mode is also affected to shift when the spindle is assembled with different preload.

B. Effect of Preload Variation

To realize the variation of the dynamic characteristics of spindle unit due to the variation of bearing preload, we extract the modal parameters, such as the modal frequency, modal compliance and modal damping from the frequency responses functions for each tested spindle.

Fig. 7 compares the modal frequency of spindles with different variation in bearing preload. As seen in this figure, the modal frequencies are slightly affected to increase or decrease with the variation in bearing preload. Generally, a decrease of bearing preload induces a decrease in frequency by about 4% less; however, an increase of bearing preload will cause an increase in the frequency by about 3% less, in addition to the first mode. This phenomena clearly show that an increased preload on bearings increased the bearing stiffness and hence a higher frequency. To the contrary, when the bearing preload is reduced, the bearing stiffness and hence the natural frequency are also affected to decreased. The extent of the variation in bearing stiffness is determined by bearing preload which was changed in assemblage of spindle unit.

Fig. 8 compares the modal compliance of spindles with different variation in bearing preload. It can found from this figure that the compliance of each mode is affected to increase or decrease with the variation in bearing preload. For each mode, the compliance of the spindle with standard preload is lower than the others with preload variation. The compliance at spindle nose is about 0.2–0.4 μm/N approximately. But as the preload increase or decrease, the maximum compliance at spindle nose is increased by 5% approximately. This indicates that a variation in preload enable the spindle to be more compliant. However, the decrease or increase in bearing preload shows a different influence trend on the spindle compliance. There exit differences in compliance ranging from 5% to 33%, depending on the vibration modes. As seen, a slight decrease of the bearing preload indeed reduces the stiffness of the spindle. Whereas, a slight increase of the bearing preload does not bring a positive effect on the stiffness of the spindle.

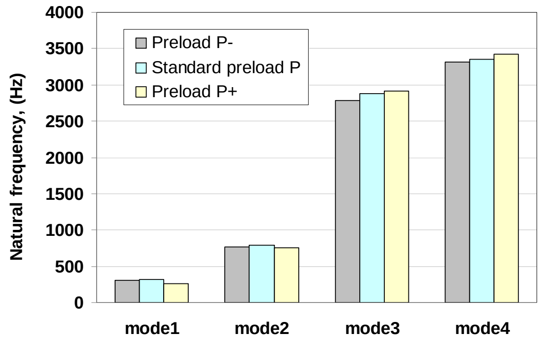


Fig. 7 Comparison of the modal frequency of spindles with different variation in bearing preload

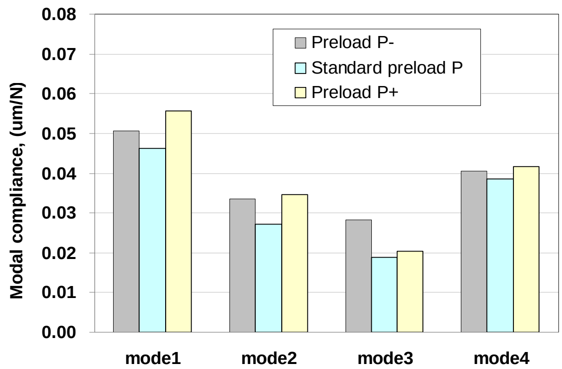


Fig. 8 Comparison of the modal compliance of spindles with different variation in bearing preload

Fig. 9 shows the variation in modal damping caused by change of bearing preload in spindle. As shown in this figure, the damping ability associated with the first and second modes of the spindle structure with standard bearing preload is better than others with lower or higher preload. The decrease percentage in damping ratio ranges from 9 to 42%, when the bearing preload is decreased from standard value. However, the damping ratios vary by 2 to 6% when the bearing preload is increased. It is known from this comparison that the influence of the preload variation on damping ability is varying with vibration modes.

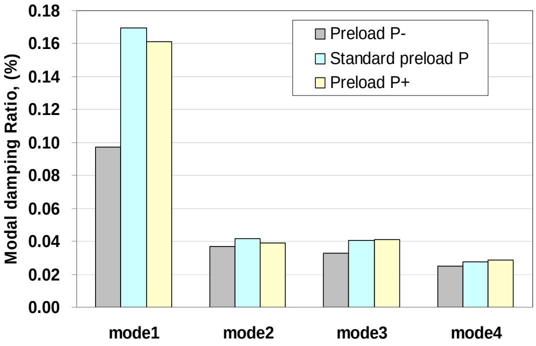


Fig. 9 Comparison of the modal damping of spindles with different variation in bearing preload

IV. CONCLUSION

The study presents an investigation on the variation of dynamic characteristics of a spindle under the influence of the bearing preload. Based on the results, the following conclusions are drawn:

1. Basically, the vibration behavior of the spindle is dominated by the bending deformation of the spindle shaft in different mode, which is further determined by the stiffness of the bearings in spindle housing. Besides, the spindles with different bearing preload behave a similar dynamic behavior.
2. A variation in bearing preload will affect the dynamic characteristics of the spindle to vary with different extent. As found in vibration tests, the spindle with a lower bearing preload vibrates at lower frequency when compared to the one with standard preload. Such influence can be quantified by examining the variation of the modal parameters such as modal compliance and modal damping.
3. In addition, the modal compliance measured at spindle nose is affected to vary when the bearing preload changes. For the standard spindle used in this study, the modal compliance is lower than the others with a positive or negative variation in bearing preload. This can be ascribed to the fact that a variation in bearing preload may change the bearing stiffness and damping ability at the rolling interface, which are dependent on the vibration mode.

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