EFFECTS OF VARIABLE BEARING STIFFNESS ON THE STABILITY OF MILLING PROCESSES

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Summary In this effort, considering the effects of spindle speed on modal parameters of tool tip, a set of differential equations variable mass, stiffness, and damping is set up to descript the dynamics of milling processes. Semi-discretization method is used to determine the stability boundaries related to chatter. Results indicate the effects of variable modal parameters and higher order modes of spindle cannot be ignored to predict the stability of the milling processes.

INTRODUCTION

High speed milling is a technology used to improve the productivity and quality of finished surface. The machined quality is often associated with the relative vibration between tool and workpiece, especially, important to avoid chatter. In order to predict the stability boundaries related to chatter, one needs the information of transfer functions of tool tip and workpiece [1-4]. Usually, high speed spindle is supported by the angular contact ball bearing. The stiffness of the ball bearing decreases as rotational speed increase. This consequently causes the decreases of the stiffness and the natural frequencies of the spindle with the increase of spindle speed due to the bearing softening and gyroscopic effect [5-6]. In addition, in the former works [1-4], only the first structural mode is considered. However, the second or higher order mode may be dominantly to the overall system. Hence, in the present paper, the dynamics of spindle in the work of Shin [5] is integrated into the dynamics of milling processes. Semi-discretization method [7] is used to determine the stability boundaries of high speed milling processes. The impacts of different modes on the stability of the milling processes are also studied in the paper to predict the stability more precisely.

SPINDLE DYNAMICS CHANGE AT HIGH SPEED

The contact angles between the ball and inner and outer raceways are identical in the static state such as shown in the Fig. 1a. When the spindle starts rotating, the inner raceway contact angles increase and the outer raceway contact angles decrease due to the centrifugal force developed (Fig.1b).The contact angles have a large influence on the radial and axial stiffness of bearings. This consequently causes the change of the effective stiffness and natural frequencies of the spindle-holder-tool assembly system. Fig 2 Shown the change of the first third natural frequencies and effective stiffness of the tool tip with the increase of spindle speed. The dramatic loss of radial stiffness results into the decrease of effective stiffness and natural frequencies, especially for the first mode.

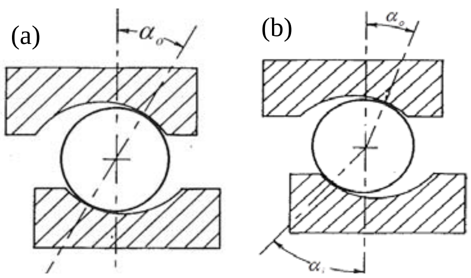

Fig. 1. Angular contact ball bearing without/with rotation.

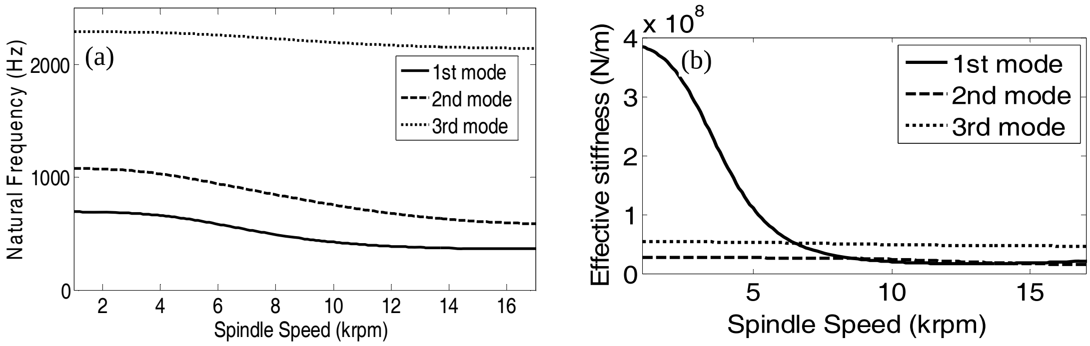

Fig. 2. The effects of spindle speed on: a) natural frequencies, b) effective stiffness at the tip of tool.

CHATTER ANALYSIS

Allowing for variable stiffness effects, for a workpiece-tool system involved in a milling operation, the governing equations of motion can be shown to be of the form

$$\mathbf{M}(\Omega)\ddot{\mathbf{q}}(t)+\mathbf{C}(\Omega)\dot{\mathbf{q}}(t)+\mathbf{K}(\Omega)\mathbf{q}(t)=\hat{\mathbf{K}}(t)\mathbf{q}(t)-\sum_{i=1}^{N}\int_{z_1(t,i)}^{z_2(t,i)}\hat{\mathbf{K}}^{i}(t,z)\mathbf{q}(t-\tau)dz+\bar{\mathbf{K}}(t)f \tag{1}$$

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where, the modal parameters of the tool are functions of the spindle speed. The cutting-force components, which appear on the right-hand side of the equations, are time periodic functions.

$$\dot{\mathbf{Q}}(t)=\mathbf{W}_0(\Omega,t)\mathbf{Q}(t)+\sum_{i=1}^{N}\mathbf{W}_i(\Omega,t)\mathbf{Q}(t-\tau)+\begin{bmatrix}\mathbf{0}\\ \bar{\mathbf{K}}(t)f\end{bmatrix} \tag{2}$$

It is assumed that the nominal periodic orbit of system (2) is represented by $\mathbf{Q}_0(t)$. Then, a perturbation X(t) is provided to this nominal orbit resulting in

$$\mathbf{Q}(t)=\mathbf{Q}_0(t)+\mathbf{X}(t) \tag{3}$$

After substituting equation (3) into equation (2), the equation governing the perturbation is determined as

$$\dot{\mathbf{X}}(t)=\mathbf{W}_0(\Omega,t)\mathbf{X}(t)+\sum\nolimits_{i=1}^{N}\mathbf{W}_i(\Omega,t)\mathbf{X}(t-\tau) \tag{4}$$

Through a semi-discretization procedure [7], the problem of stability of continuous delay differential equations (4) can be transferred as to determine the stability of the linear discrete poincaré map

$$\mathbf{Y}_k=\mathbf{\Phi}\mathbf{Y}_0 \tag{5}$$

This finite-dimensional matrix $\mathbf{\Phi}$ represents an approximation of the infinite-dimensional monodromy matrix associated with the trivial solution $\mathbf{X}(t)=\mathbf{0}$ of (4). One can determine the stability of system (1) by the eigenvalues of the transition matrix $\mathbf{\Phi}$. As two representative results, in Fig. 3a and 3b, the predicted stability lobes constructed for dynamic modal parameters and static modal parameters are presented. It is clear that the minimum of the axial critical depth of cut(ADOC) obtained for the dynamic modal parameters is less than that obtained for the static modal parameters, especially, in the case of only the first mode is considered. By comparing the stable lobe obtained for first mode and second mode, one can find the second mode is dominant to determine the stability boundaries at the low rotational speed region (0-8000rpm). These indicate both the effects of variable modal parameters and higher order modes cannot be ignored to predict the stability of the milling processes.

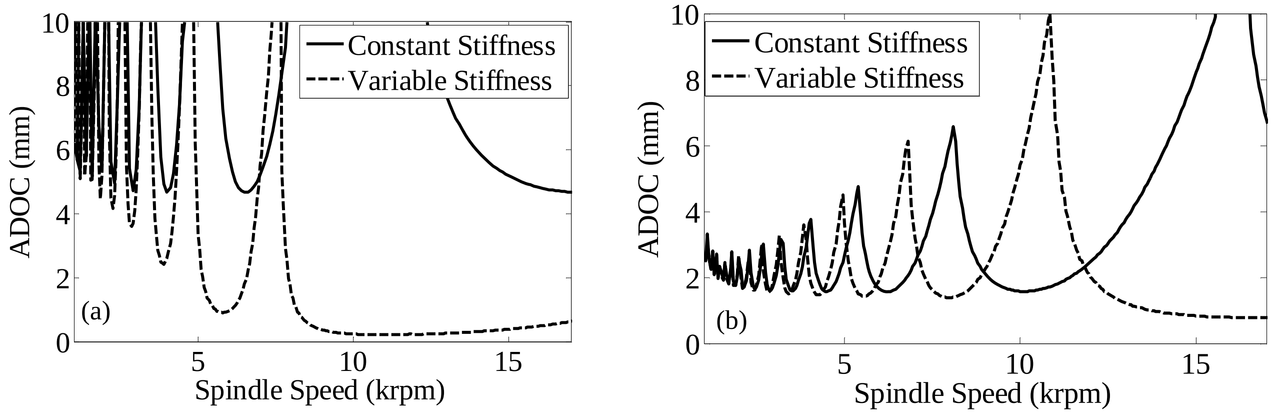

Fig. 3. Stability lobes obtained for: a) the first mode, b) the second mode.

CONCLUSIONS

In this study, the speed depending characteristics of high speed spindle with angular contact bearing have been presented. Results show the stiffness and the natural frequencies of spindle decreases as spindle speed increases. The effects of this characteristic on the stability of milling processes are illustrated. To predict the stability boundaries accuracy, the effects of speed dependent system dynamics and higher order modes should be considered properly.

ACKNOWLEDGEMENTS

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