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Identification of bearing dynamics under operational conditions for chatter stability prediction in high speed machining operations

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ABSTRACT

Chatter is a major problem causing poor surface finish, low material removal rate, machine tool failure, increased tool wear, excessive noise and thus increased cost for machining applications. Chatter vibrations can be avoided using stability diagrams for which tool point frequency response function (FRF) must be determined accurately. During cutting operations, due to gyroscopic moments, centrifugal forces and thermal expansions bearing dynamics change resulting in tool point FRF variations. In addition, gyroscopic moments on spindle–holder–tool assembly cause separation of modes in tool point FRF into backward and forward modes which will lead to variations in tool point FRF. Therefore, for accurate stability predictions of machining operations, effects of operational conditions on machine tool dynamics should be considered in calculations. In this study, spindle bearing dynamics are identified for various spindle rotational speeds and cutting forces. Then, for a real machining center, tool point FRFs under operating conditions are determined using the identified speed dependent bearing dynamics and the mathematical model proposed. Moreover, effects of gyroscopic moments and bearing dynamics variations on tool point FRF are examined separately. Finally, computationally determined tool point FRFs using revised bearing parameters are verified through chatter tests.

1. Introduction

Chatter is a major problem causing poor surface finish, low material removal rate, machine tool failure, increased tool wear, excessive noise and increased cost for machining applications. In order to avoid chatter, mechanisms of dynamic cutting process have been examined in detail for decades [1–5], and stability diagrams which provide stable regions in terms of the depth of cut and spindle speed combinations, have been developed [3–6]. Implementation of stability theory to production causes considerable increase in productivity. In order to obtain stability diagrams, frequency response functions (FRF) at the tool tip are needed. In general, tool point FRF is obtained experimentally using impact testing which requires additional time before machining operations. In order to obtain stability diagrams, modal analysis must be performed and tool point FRF should be determined for every combination of spindle, holder and tool, which is time consuming and may be costly, especially for production machines. These experimental limitations have lead researchers to investigate analytical methods to model machining centers consisting of spindle, holder and tool subassemblies which would eliminate or reduce the need for experiments. In order to obtain tool point FRF analytically; Schmitz et al. [7–9] proposed a semi-analytical method which applies the receptance coupling technique to couple the experimentally obtained spindle–holder subassembly receptances with the analytically obtained tool receptances using the contact parameters at the holder tool interface. Schmitz's semi-analytical method in determining tool point FRF has been followed by several studies based on receptance coupling [10–12]. In addition to semi-analytical models, Ertürk and coworkers [13] proposed an experimentally verified [13,14] analytical model for predicting tool point FRF by combining receptance coupling and structural modification techniques. In this mathematical model, all components of a spindle–holder–tool assembly were modeled analytically with the Timoshenko beam theory, and combined with the contact parameters at the spindle–holder and holder–tool interfaces.

However, discrepancies between calculated stability diagrams and actual stability of the process are frequently observed. One major contributor to these deviations is the changes in machine tool dynamics under cutting conditions. Because, during high speed cutting operations gyroscopic moments, centrifugal forces and temperature increase cause variations in bearing dynamics. In

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addition to variations in bearing dynamics, dynamics of the subassemblies (spindle, holder and tool) may also be affected from gyroscopic moments at high speeds. Thus, tool point FRFs measured at idle state or calculated computationally for idle state of the machining center may lead to incorrect chatter stability predictions. Therefore, for high speed cutting operations, development of a mathematical model including rotational effects plays a crucial role in accurate prediction of the chatter stability.

Due to the gyroscopic effects, natural frequencies and corresponding modes of the system separate into backward and forward modes [15–17]. Also due to the gyroscopic effects, unlike non-rotating systems, dynamic responses of the rotating system in two orthogonal planes are coupled. Thus, cross-coupling effects should be considered in analytical modeling of rotating systems. In addition, bearing characteristics are also affected by the rotational effects. During high rotational speeds, centrifugal forces and gyroscopic moments act on the balls of the bearings pressing the balls toward the outer race. This effect causes changes in the contact angles and kinematics of the balls as well as redistributing the contact loads in the bearing which leads to decreased stiffness [18]. On the contrary, damping of the ball bearings increases under the rotational effects [18]. In order to model bearing behavior at high rotational speeds, Stone [19] proposed a general theory which includes gyroscopic and centrifugal effects. Harris [20] also proposed a method that requires solution of nonlinear equations numerically, and showed the variation of bearing stiffness under high spindle speeds with different preloads.

Since there exists various effects that cause tool point FRF vary under operational conditions, for accurate prediction of tool point FRF, researchers developed models including rotational effects for the machine tool analysis [21–24]. In addition to gyroscopic and centrifugal effects, thermal expansions may also cause variations in the contact conditions of bearings, and thus affect dynamic properties of the bearings. In order to include thermal effects, thermo-mechanical models have also been proposed [25–27]. Therefore, for accurate modeling of the spindle–holder–tool assemblies in machining centers, these effects should also be considered.

In this study, a procedure for bearing parameter identification both for idle state and under rotating conditions is presented. First, bearing parameters are identified for the idle state, and by employing effect analysis, bearing parameters that cause variations in the elastic mode of the tool point FRF are determined. Then, using chatter test results and the proposed mathematical model, variations of bearing stiffness and damping with respect to spindle speed and cutting force are identified. Moreover, proposed modeling approach is verified by using experimentally identified spindle–holder interface parameters and speed dependent bearing dynamics in the modeling of a different holder–tool combination. Thus, tool point FRF of the spindle–holder–tool assembly which is not used in the identification procedure is predicted without performing any further experiments, and verified through chatter tests. In addition to verification of the proposed identification procedure, effects of the gyroscopic moments and bearing stiffness variations on tool point FRFs and stability are examined through some case studies.

2. Mathematical theory

2.1. Inverse stability solution for in process tool point FRF identification

Variations in tool point FRF under operating conditions can be experimentally identified using the method proposed by Özşahin et al. [28]. In this method, first, analytical expressions of chatter frequency and corresponding axial depth of cut are written in terms of process conditions, cutting force coefficients and structural dynamics of spindle–holder–tool assembly. In the experimental part of the identification procedure, stability tests are carried out to identify chatter frequencies and corresponding stability limits at various spindle speeds in the range of interest. Then, the analytical chatter frequency and axial depth of cut expressions together with the experimentally identified stability limit and chatter frequency pairs at different spindle speeds can be used to identify the structural dynamics. Therefore, as opposed to the standard prediction approach where stability limit and chatter frequency are predicted for a given process and FRF's, in this case the inverse is done by using the measured chatter frequencies and stability limits to identify the unknown FRF's at different spindle speeds. In order to simplify the solution procedure, only the modal parameters of the dominant mode in tool point FRF is treated as unknown (natural frequency and damping ratio) and remaining modes in the tool point FRF are kept as the same as the ones for the idle state. Main advantage of this method is that the necessity of expensive equipments, complicated experimental setups and signal processing are eliminated. Method suggested [28] requires impact testing for the idle FRF measurement and a simple microphone for the determination of chatter frequency and stable axial depth limit.

2.2. Analytical modeling of the spindle–holder–tool assembly dynamics

In determining tool point FRF theoretically, the receptance coupling approach proposed in previous studies [12,13] is combined with a recent method developed by Orkun et al. [29]. In this study, spindle–holder–tool assembly is divided into subcomponents (spindle, holder and tool) and each subcomponent is modeled using Timoshenko beam model including gyroscopic moments. In Timoshenko beam model, end point and cross FRFs of the spindle (S), holder (H) and tool (T) are calculated. In the spindle modeling front and rear bearing dynamics are added by using the structural modification technique [30]. Then, spindle (S) and holder (H) receptance matrices are coupled using contact parameters at spindle–holder interface as follows:

$$[SH_{11}] = [H_{11}] - [H_{12}]\left[[H_{22}] + [K_{sh}]^{-1} + [S_{11}]\right]^{-1}[H_{21}] \tag{1}$$

where SH and $[K_{sh}]$ represent spindle–holder assembly and contact parameters at spindle–holder interface, respectively.

Then spindle–holder assembly receptance is coupled with tool receptance matrices using contact parameters at holder–tool interface as

$$[SHT_{11}] = [T_{11}] - [T_{12}]\left[[T_{22}] + [K_{ht}]^{-1} + [SH_{11}]\right]^{-1}[T_{21}] \tag{2}$$

where SHT and $[K_{ht}]$ represent spindle–holder–tool assembly and contact parameters at holder–tool interface, respectively.

Modeling approach of the spindle–holder–tool assembly is shown in Fig. 1. In this approach, it is important to use accurate values for bearing stiffness and damping. Since bearing properties change under operating conditions, in order to obtain accurate tool point FRF, thus accurate stability predictions for the high speed machining operations, speed and force dependent bearing parameters should be determined and used in the model.

3. Identification of bearing dynamics

In this section, the procedure for identification of bearing properties that affect machining stability is presented. First, bearing parameters at idle state are identified. Then effects of bearing parameters on tool point FRF are investigated. Using the effect analysis, bearing parameters that affect both tool point FRFs and stability of the machining operations are determined using