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A closed-form approach for identification of dynamical contact parameters in spindle–holder–tool assemblies

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ABSTRACT

Accurate identification of contact dynamics is very crucial in predicting the dynamic behavior and chatter stability of spindle–tool assemblies in machining centers. It is well known that the stability lobe diagrams used for predicting regenerative chatter vibrations can be obtained from the tool point frequency response function (FRF) of the system. As previously shown by the authors, contact dynamics at the spindle–holder and holder–tool interfaces as well as the dynamics of bearings affect the tool point FRF considerably. Contact stiffness and damping values alter the frequencies and peak values of dominant vibration modes, respectively. Fast and accurate identification of contact dynamics in spindle–tool assemblies has become an important issue in the recent years. In this paper, a new method for identifying contact dynamics in spindle–holder–tool assemblies from experimental measurements is presented. The elastic receptance coupling equations are employed in a simple manner and closed-form expressions are obtained for the stiffness and damping parameters of the joint of interest. Although this study focuses on the contact dynamics at the spindle–holder and holder–tool interfaces of the assembly, the identification approach proposed in this paper might as well be used for identifying the dynamical parameters of bearings, spindle–holder interface and as well as other critical joints. After presenting the mathematical theory, an analytical case study is given for demonstration of the identification approach. Experimental verification is provided for identification of the dynamical contact parameters at the holder–tool interface of a spindle–holder–tool assembly.

1. Introduction

Self-excited vibration of machine tools during the cutting process (the so-called *regenerative chatter*) is caused by the cutting tool–work piece dynamic interaction and results in process instability, poor surface finish and reduced material removal rate. Modeling of chatter mechanism for minimizing its catastrophic consequences has been studied in detail for the last 50 years [1–5]. It is well known that the regeneration effect is due to the phase between two vibration waves during the subsequent cuts on a surface [6], and this phase is minimized for certain cutting speeds. Stability lobe diagrams provide stable depth of cut–spindle speed combinations and they have been used for predicting chatter stability for decades. The literature includes both numerical [3] and analytical [4,5] approaches for generating stability lobe diagrams of spindle–tool assemblies. Regardless of the approach used, a common point of the models used for generation of stability lobe diagrams is the requirement of the tool point frequency response function (FRF) of the assembly. Although experimental modal analysis by simple impact testing is the typical technique employed for obtaining the tool point FRF [6], recently, researchers have attempted to obtain the tool point FRF semi-analytically to minimize experimentation and save time in practical applications. Schmitz et al. [7–9] implemented the well-known receptance coupling theory of structural dynamics [10–12] in order to couple the experimentally obtained dynamics of spindle–holder assembly and the analytically obtained tool dynamics by using the contact dynamics at the holder–tool interface. The aim was to make only one experiment at the holder tip and then to obtain the tool point FRF of the assembly for different tool overhang lengths through the receptance coupling equations [7–9]. Provided that the dynamical contact parameters at the holder–tool interface are known accurately, this semi-analytical approach can provide accurate results and save considerable time.

Several improvements were made to the approach proposed by Schmitz et al. [7–9] in the last 5 years. Park et al. [13] included the

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rotational degree-of-freedom at the tool holder–tool joint. Kivanc and Budak [14] modeled the tool as a two-segment beam considering the changing area moment of inertia for more accurate results. They [14] also studied the effects of the contact length and the clamping torque on the holder–tool contact stiffness and damping properties. Duncan and Schmitz [15] improved the use of receptance coupling approach to handle different holder types by extending it to the coupling of holder segments. It should be underlined again that the accuracy of these models strongly depends on the accurate identification of dynamical contact parameters at the holder–tool interface. Accurate modeling and identification of contact mechanics has been an important problem in several engineering applications, and its nature has been investigated by scientists and engineers for decades [16]. Expectedly, it has also been subject to research in machine tool engineering where some researchers investigated the spindle–holder interface dynamics for analyzing and improving structural stability [17–21]. Recently, Schmitz et al. [22] introduced off-diagonal elements to the diagonal joint stiffness matrix used in their early work [7–9] to account for the translations imposed by moments and rotations caused by forces. More recently, Ahmadi and Ahmadian [23] considered the holder–tool interface as a distributed elastic layer to model the change in the normal contact pressure along the joint interface.

Ertürk et al. [24] proposed an experimentally verified [25,26] analytical model for predicting the tool point FRF by combining the receptance coupling and structural modification techniques where all components of the spindle–holder–tool assembly were modeled analytically with the Timoshenko beam theory. They [24] formed the individual system components (spindle, holder and tool) by *rigid receptance coupling* of free–free Timoshenko beams and included the dynamics of bearings to the spindle by structural modification with an efficient dynamic coupling algorithm. Then, these three main components of the system were combined by *elastic receptance coupling* with the information of contact dynamics at the spindle–holder and holder–tool. The analytical model proposed for the prediction of tool point FRF [24] was shown to be very efficient in predicting chatter stability [26] when it is combined with the analytical stability lobe diagram model presented by Budak and Altintas [4,5]. The influence of bearing and interface dynamics on the tool point FRF was also studied [27] by using the analytical model proposed. It was observed that the variations in the dynamical contact parameters at the spindle–holder and holder–tool interfaces as well as bearing dynamics have a strong effect on the resulting tool point FRF of the system. For a typical system, it was identified that the dynamical contact parameters (stiffness and damping) at the spindle–holder and holder–tool interfaces affect the dominant elastic modes of the tool point FRF [27]. Variations of the translational contact stiffness were found to be affecting the frequencies of the elastic modes, whereas the variations in the translational contact damping altered the peak values of these respective modes. Furthermore, for a typical assembly, an uncoupled trend was observed between the effects of the contact dynamics at the spindle–holder and holder–tool interfaces such that the dynamics of the former interface controlled the spindle bending mode of the assembly, whereas that of the latter interface controlled the tool mode of the assembly (which were the first and the second elastic modes of the assembly used by Ertürk et al. [27], respectively). From this observation, the important suggestion made was identifying the dynamical contact parameters of an interface from the respective vibration mode(s) they control. Considering the fact that the earlier work [7–9] used the nonlinear least square error minimization for identifying the contact parameters, the approach suggested by Ertürk et al. [27] was very practical to implement and time saving for two reasons. First, there is no need to use the entire frequency band of the experimental FRF, since a simple effect analysis made by perturbation of the contact parameters in the model yields the frequency range(s) where one should identify those parameters. Indeed, theoretically, it is meaningless to identify a damping parameter by minimizing the error in the analytical (or semi-analytical) FRF at the structural stiffness/mass controlled *off-resonance* frequencies. Secondly, due to the nonlinearity of the least square error minimization approach, it is not uncommon to obtain more than one solution set of the contact parameters since the numerical solution may converge to the results for a local minimum. The approach suggested [27] avoids both of these problems and reduces the time required for identifying the dynamical contact parameters not only at the holder–tool interface but also at the spindle–holder interface as well as the dynamical parameters of bearings.

In this paper, a new approach for identification of dynamical contact parameters in spindle–holder–tool assemblies is presented. The elastic receptance coupling equations [24] used for coupling the system components are rearranged to give the complex stiffness matrix of the joint (interface) of interest (e.g., spindle–holder or holder–tool joint). After expressing the fully populated complex stiffness matrix at the joint of interest in terms of the analytical and experimental receptance matrices, the contact stiffness and damping parameters are extracted by utilizing the conclusions of our previous work [27] summarized in the previous paragraph. The identification approach suggested is first used in a case study for analytical demonstration. Then, it is verified experimentally for a spindle–holder–tool assembly with a focus on the holder–tool interface. Although the approach presented in this paper concentrates on the contact dynamics at the spindle–holder and holder–tool interfaces, it can also be used to identify the bearing dynamics and the dynamics of the other critical joints of a machine tool assembly.

2. Theory

2.1. Mathematical background

A typical spindle–holder–tool assembly and its components are shown in Fig. 1. In the analytical model proposed by Ertürk et al. [24],

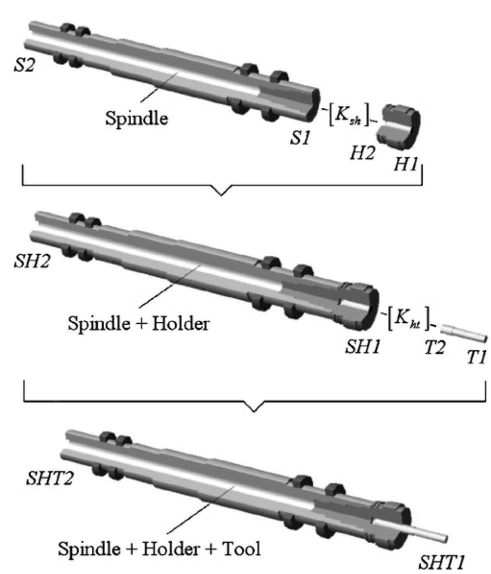

Fig. 1. Components of spindle–holder–tool assembly and the complex stiffness matrices of spindle–holder and holder–tool interfaces.