Available online at www.sciencedirect.com
ScienceDirect
International Journal of Machine Tools & Manufacture 47 (2007) 23–32
INTERNATIONAL JOURNAL OF MACHINE TOOLS & MANUFACTURE DESIGN, RESEARCH AND APPLICATION
www.elsevier.com/locate/ijmactool

Effect analysis of bearing and interface dynamics on tool point FRF for chatter stability in machine tools by using a new analytical model for spindle–tool assemblies

A. Ertürk[a], H.N. Özgüven[a], E. Budak[b,*]

[a]Department of Mechanical Engineering, Middle East Technical University, 06531 Ankara, Turkey
[b]Faculty of Engineering and Natural Sciences, Sabanci University, Orhanli, Tuzla, 34956 Istanbul, Turkey

Received 3 January 2006; received in revised form 27 February 2006; accepted 1 March 2006
Available online 19 April 2006

Abstract

Self-excited vibration of the tool, regenerative chatter, can be predicted and eliminated if the stability lobe diagram of the spindle–holder–tool assembly is known. Regardless of the approach being used, analytically or numerically, forming the stability lobe diagram of an assembly implies knowing the point frequency response function (FRF) in receptance form at the tool tip. In this paper, it is aimed to study the effects of spindle–holder and holder–tool interface dynamics, as well as the effects of individual bearings on the tool point FRF by using an analytical model recently developed by the authors for predicting the tool point FRF of spindle–holder–tool assemblies. It is observed that bearing dynamics control the rigid body modes of the assembly, whereas, spindle–holder interface dynamics mainly affects the first elastic mode, while holder–tool interface dynamics alters the second elastic mode. Individual bearing and interface translational stiffness and damping values control the natural frequency and the peak of their relevant modes, respectively. It is also observed that variations in the values of rotational contact parameters do not affect the resulting FRF considerably, from which it is concluded that rotational contact parameters of both interfaces are not as crucial as the translational ones and therefore average values can successfully be used to represent their effects. These observations are obtained for the bearing and interface parameters taken from recent literature, and will be valid for similar assemblies. Based on the effect analysis carried out, a systematic approach is suggested for identifying bearing and interface contact parameters from experimental measurements.
© 2006 Elsevier Ltd. All rights reserved.

Keywords: Chatter stability; Bearing and interface dynamics; Parametric identification

1. Introduction

Regenerative chatter vibrations develop due to the dynamic interactions between the workpiece and the cutting tool, and result in unstable cutting, poor surface quality and permanent damage on the machine tool itself. The regeneration of waviness is known to be due to the phase between the subsequent cuts on the surface of the workpiece. Stability lobe diagrams supply the spindle speed—axial depth of cut combinations for which this phase is minimized and the cutting process is stable. The basics of chatter theory and stability lobe diagrams were introduced by Tobias [1,3] and Tlusty [2,4] for orthogonal cutting conditions and time-invariant process dynamics. Merrit [5] used the Nyquist stability criterion of feedback control theory for developing the stability lobe diagrams and obtained similar results, again for orthogonal cutting with time invariant process dynamics. However, the stability analysis of milling is complicated due to the rotation of the cutting tool which results in time varying directional factors and system dynamics. Tlusty [6–8] made time domain simulations for predicting chatter stability in milling. Minis and Yanushevsky [9,10] employed Floquet's theorem and Fourier series for the formulation of milling stability and used Nyquist criterion for the numerical solution. Altintas and Budak [11] presented the analytical model for the stability limits in milling which was shown to be very fast for the generation of stability lobe diagrams [12].

*Corresponding author. Tel.: +90 216 483 9519; fax: +90 216 483 9550.
E-mail address: ebudak@sabanciuniv.edu (E. Budak).

0890-6955/$ - see front matter © 2006 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijmachtools.2006.03.001

The common point of the numerical and analytical milling stability models is the fact that they all require the tool point FRF, which is generally denoted by $G(\omega)$. Performing experimental modal analysis by using a low mass accelerometer, instrumented hammer and a spectrum analyzer is a common way of obtaining the tool point FRF. However, any change in the spindle–holder–tool assembly, such as tool and/or holder changes, will affect the system dynamics and a new measurement will be required. This approach may consume considerable amount of time which is costly on production machines since the machine should be stopped for every new measurement, even for the simplest changes in the assembly. In order to reduce experimentation, the receptance coupling theory of structural dynamics has been implemented for modeling the spindle–holder–tool dynamics semi-analytically [13–18]. It is suggested that the dynamics of spindle–holder subassembly can be obtained experimentally at the holder tip for once, then, it can be coupled with the dynamics of the tool, which is obtained analytically by considering the tool as a beam with free end conditions. This semi-analytical approach can give accurate results and save considerable time in practical applications as long as the holder–tool interface dynamics is modeled or experimentally identified accurately. Duncan and Schmitz [19,20] improved the use of receptance coupling approach to handle different holder types using a single experimental measurement. In a recent study, Ertürk et al. [21] presented an analytical model for predicting the tool point FRF, in which they use Timoshenko beam theory, receptance coupling and structural modification techniques for modeling the spindle–-holder–tool dynamics. In all these recent studies, the common objective is to minimize experimentation for the prediction of tool point FRF and the common crucial requirement of them is the accurate knowledge of the connection dynamics. By using the new model they suggested, Ertürk et al. [21] have shown that the relative motion between the components (spindle, holder and tool) can be more important than their individual structural motions since the components are non-slender. This result, implicitly points out the importance of the accurate knowledge of interface dynamics for obtaining the tool point FRF correctly in the existing analytical and semi-analytical models.

In classical tool receptance coupling, the interface parameters are iteratively obtained by employing least square error minimization until the model and experimental FRFs are fitted for all the modes appearing in the existing frequency range. In such an approach, any modeling and measurement error will be compensated by the extracted inaccurate or incorrect interface dynamic parameters, and therefore they can successfully be used only for the configuration for which the experimental study is carried out. Consequently, identification of correct contact parameters is an important issue in employing mathematical models. Knowing how spindle–holder and holder–tool interface dynamics affect the resulting tool point FRF will help to identify contact parameters more accurately, and will also make it possible to observe how possible errors in the extracted interface dynamics are reflected on the tool point FRF of the assembly.

In this paper, by using the analytical model developed [21], the effects of bearing and interface dynamics on the tool point FRF are analyzed. From the results of the effect analysis, suggestions are made for developing systematic approaches for the identification of bearing and interface dynamics from experimental measurements. The analytical model developed in the previous study [21] is briefly summarized in the next section.

2. Mathematical modeling

In the model developed, spindle, holder and tool, which are the main system components, are modeled as multi-segment beams by using Timoshenko beam theory. The individual multi-segment components (spindle, holder and tool) are formed by coupling the end point receptances of uniform beams rigidly. Determination of the end point receptances of a uniform Timoshenko beam with free end conditions is given in [21] in detail.

Consider the rigid coupling of two uniform beams as shown in Fig. 1. Using the formulation presented in [21], the end point receptance matrices of beams A and B can be obtained as

$$[A]=\begin{bmatrix}[A_{11}] & [A_{12}]\\ [A_{21}] & [A_{22}]\end{bmatrix}, \tag{1}$$

$$[B]=\begin{bmatrix}[B_{11}] & [B_{12}]\\ [B_{21}] & [B_{22}]\end{bmatrix}, \tag{2}$$

where submatrices of the above matrices include the point and transfer receptance functions of the segment end points. For example, the point receptance matrix of node $A1$ in beam A is given as

$$[A_{11}]=\begin{bmatrix}H_{A1A1} & L_{A1A1}\\ N_{A1A1} & P_{A1A1}\end{bmatrix}. \tag{3}$$

Note that $[A_{11}]$ actually represents $[A_{A1A1}]$, and just for simplicity it is written in the following formulation as $[A_{11}]$ (the same is true for other receptance matrices). The receptance functions, which are denoted by letters H, N, L

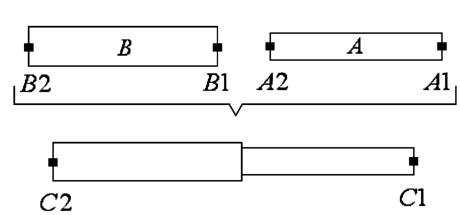

Fig. 1. Rigid coupling of two uniform beams with free end conditions.

Download English Version:

https://daneshyari.com/en/article/784686

Download Persian Version:

https://daneshyari.com/article/784686

Daneshyari.com