A Modeling Approach for Analysis and Improvement of Spindle-Holder-Tool Assembly Dynamics

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Abstract

The most important information required for chatter stability analysis is the dynamics of the involved structures, i.e. the frequency response functions (FRFs) which are usually determined experimentally. In this study, the tool point FRF of a spindle-holder-tool assembly is analytically determined by using the receptance coupling and structural modification techniques. Timoshenko's beam model is used for increased accuracy. The spindle is also modeled analytically with elastic supports representing the bearings. The mathematical model is used to determine the effects of different parameters on the tool point FRF and to identify contact dynamics from experimental measurements. The applications of the model are demonstrated and the predictions are verified experimentally.

Keywords:

Milling, Structural Analysis, Chatter

1 INTRODUCTION

Chatter vibrations result in poor surface quality and reduced productivity. Stability lobe diagrams can be used to determine the stable and more productive spindle speed – axial depth of cut combinations. The basics of chatter theory and stability lobe diagrams were introduced by Tobias [1] and Tlusty [2] for orthogonal cutting conditions and time invariant process dynamics. Minis and Yanushevsky [3] employed Floquet's theorem and Fourier series for the formulation of milling stability and used the Nyquist criterion for the numerical solution. Altintas and Budak [4] presented the analytical model for the stability limits in milling which was shown to be very fast for the generation of stability lobe diagrams [5].

These models require the tool point FRF, which is generally determined through experimental modal analysis. However, any change in the spindle-holder-tool assembly, such as tool and/or holder changes, will affect the system dynamics and measurements will have to be repeated. This can be very time consuming, and thus costly on production machines considering the number of different tool-holder combinations. In order to reduce experimentation, the receptance coupling theory of structural dynamics has been implemented for modeling the spindle-holder-tool dynamics semi-analytically [6-9]. It is suggested that the dynamics of the 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. Duncan and Schmitz [10, 11] improved the use of the receptance coupling approach to handle different holder types using a single experimental measurement. In a recent study, Ertürk et al. [12] presented an analytical model to predict the tool point FRF by modeling the spindle-holder-tool dynamics. They used Timoshenko beam theory [15], receptance coupling and structural modification techniques [12]. Due to the low length to diameter ratios of system components, the Euler-Bernoulli beam model may result in considerable errors in prediction of modal frequencies which is significantly improved by using the Timoshenko beam formulation.

One of the important requirements for accurate modeling of the machine tool dynamics is the knowledge of the connection dynamics, i.e. stiffness and damping at the interfaces. In previous studies that use a receptance coupling, the interface parameters are experimentally obtained by employing least square error minimization. This type of solution is time consuming, prone to numerical errors, and may not yield a unique solution due to high number of simultaneous and nonlinear sets of equations corresponding to a large frequency range covered in the FRF, and a high number of unknowns due to multiple number of interfaces. Furthermore, any modeling and measurement error would be compensated by the extracted inaccurate or incorrect interface dynamic parameters. Ertürk et al. [13] analyzed the effects of bearing supports and spindle-holder and tool-holder interfaces on the FRF, and suggested a fast and accurate approach for the identification of connection parameters.

In this paper, the analytical model developed for dynamic analysis of machine tools is summarized and the effects of bearing and interface dynamics on the tool point FRF are briefly discussed. The complete analytical modeling allows the model to be used in the spindle design phase as well as in the fast identification of the interface parameters, in addition to the prediction of the tool point FRF for a given assembly. The systematic approach suggested for the identification of bearing and interface dynamics is employed for a spindle-holder-tool assembly, and contact parameters are identified from experimental measurements. The model developed for spindle-holder-tool assembly is compared with a finite element model and is also experimentally verified.

2 MATHEMATICAL MODELING

2.1 Component and Assembly FRFs

Spindle, holder and tool are modeled as multi-segment beams by using Timoshenko beam theory. The individual multi-segment components 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 [12] in detail. The end point receptance matrix of a beam *A* can be represented as

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

where submatrices of the above matrix include the point and transfer receptance functions of the segment end points (1) and (2). For example, the point receptance matrix of node *A*1 in beam *A* is given as

$$[A_{11}]=\begin{bmatrix}H_{A1A1} & L_{A1A1}\\ N_{A1A1} & P_{A1A1}\end{bmatrix} \tag{2}$$

The receptance functions, which are denoted by letters *H*, *N*, *L* and *P*, are defined as follows:

$$\begin{aligned} y_j &= H_{jk}\cdot f_k \qquad \theta_j = N_{jk}\cdot f_k\\ y_j &= L_{jk}\cdot m_k \qquad \theta_j = P_{jk}\cdot m_k \end{aligned} \tag{3}$$

where y and θ represent the linear and angular displacements, respectively, and f and m are the forces and the moments, respectively, at the points *i* and *j*. Two beams, *A* and *B*, can be coupled dynamically using rigid receptance coupling and the receptance matrix of resulting two-segment beam *C* can be obtained as follows:

$$[C]=\begin{bmatrix}[C_{11}] & [C_{12}]\\ [C_{21}] & [C_{22}]\end{bmatrix} \tag{4}$$

where

$$\begin{aligned} [C_{11}]&=[A_{11}]-[A_{12}]\big[[A_{22}]+[B_{11}]\big]^{-1}[A_{21}]\\ [C_{12}]&=[A_{12}]\big[[A_{22}]+[B_{11}]\big]^{-1}[B_{12}]\\ [C_{21}]&=[B_{21}]\big[[A_{22}]+[B_{11}]\big]^{-1}\cdot[A_{21}]\\ [C_{22}]&=[B_{22}]-[B_{21}]\big[[A_{22}]+[B_{11}]\big]^{-1}[B_{12}] \end{aligned} \tag{5}$$

By following the same formulation, one might continue coupling more segments like a chain to form an *n*-segment beam. In order to include the dynamics of bearings, the structural modification technique presented by Özgüven [14] can be used as shown in detail in [12]. In this case, the dynamic structural modification matrix represents the translational and rotational, stiffness and damping information of the bearings. The final step is to couple the main system components to obtain the tool point FRF. However, these components should be coupled elastically due to the flexibility and damping at the contacts. When the end point receptances of the spindle on bearings (*S*) are coupled with those of the holder (*H*), the end point receptance matrices of the spindle-holder assembly (*SH*) can be obtained from:

$$[SH_{11}]=[H_{11}]-[H_{12}]\Big[[H_{22}]+[K_{sh}]^{-1}+[S_{11}]\Big]^{-1}[H_{21}] \tag{6}$$

$[K_{sh}]$ is the complex stiffness matrix representing spindle-holder interface dynamics. Note that the receptance matrix $[SH_{11}]$ is very similar to $[C_{11}]$ given in equation (5) with the addition of $[K_{sh}]^{-1}$ only. Finally, the tool (*T*) can be added to the spindle-holder (*SH*) system to obtain the end point FRFs of spindle-holder-tool (*SHT*) assembly. The FRF required for the stability lobe diagram of a given spindle-holder-tool assembly is the one that gives the relation between the transverse displacement and force at the tool tip, which is the first element of the following matrix:

$$[SHT_{11}]=[T_{11}]-[T_{12}]\Big[[T_{22}]+[K_{ht}]^{-1}+[SH_{11}]\Big]^{-1}[T_{21}] \tag{7}$$

2.2 Application of the Model

The spindle-holder-tool combination, geometry of which is shown in Figure 1, is used for demonstrating the application of the model. Each of the system components, i.e. spindle, holder and tool are composed of several sections with different diameter and lengths which are modeled as multi-segment beams. The dimensions of the components, bearing and interface dynamical properties are given in [12]. In order to verify the results of the model, the vibration modes of this assembly were calculated using the finite element method using ANSYS® 9.0. The beam element BEAM188, which is based on Timoshenko beam theory, is used by restricting the degrees of freedom other than motion in one transverse direction and flexural rotation so that the finite element model is consistent with the model proposed. In order to represent the dynamics of bearings and spindle-holder and holder-tool interfaces, combination element COMBIN14 (Spring-Damper) of ANSYS® 9.0 is used. The natural frequencies obtained by the analytical model and the finite element solution are tabulated in Table 1. As can be seen from the table, the natural frequencies of the assembly obtained by the model presented in this paper and those obtained by using the finite element software are in good agreement and the maximum difference observed for the first seven modes is about 5 %.

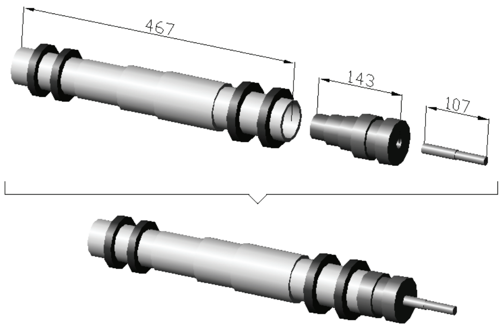

Figure 1: Components of the assembly used in the example.

Mode	Model [Hz]	FEA [Hz]	Diff. [%]
1	71.7	71.6	0.14
2	195	193.8	0.62
3	877.8	867.5	1.19
4	1438.3	1424.3	0.98
5	1819.5	1752.6	3.82
6	3639.3	3442.5	5.72
7	3812.5	3634.8	4.89

Table 1: Natural frequencies of the assembly used in the case study.

At this point, it is worthwhile to mention the importance of using Timoshenko beam theory rather than the Euler-Bernoulli beam model. It is well known that rotary inertia and especially shear deformation [15] are very important for non-slender components and at high frequencies. In the previous studies that use the Euler-Bernoulli model with receptance coupling, good agreements between experimental and predicted FRFs were obtained. There are two main reasons for this. First of all, in these studies, the connection parameters at the holder-tool interface were obtained by fitting the model to the experiment tool point FRFs. In such an approach, depending on the component geometries, using Euler-Bernoulli beam theory may result in modeling errors which can be compensated by the incorrect connection parameters. Secondly, at lower frequencies, the FRFs are primarily controlled by the elasticity of the interfaces between spindle-holder and tool-holder, rather than the flexural rigidities of the holder and tool themselves [12]. In the frequency range of interest, the part of the holder outside the spindle behaves almost as a mass with no elastic contribution, and the elastic contribution of the tool is at most from its first mode. In the case of stiffer connection dynamics (so that component structural behaviors become more important) and/or when the frequency range of interest is wider, it becomes a must to use the Timoshenko model for accurate results. Deficiency of the Euler-Bernoulli model in such a case is illustrated in Figure 2 which shows the tool point FRF of the same assembly when a much stiffer connection is assumed between the components and the frequency range is extended to 10 000 Hz. Based on these results, it can be concluded that the Euler-Bernoulli model may yield inaccurate results, especially at high frequencies and/or for stiffer connection dynamics. In addition, if the individual component FRFs are of interest, such as spindle tip or free tool FRFs, it is necessary to use the Timoshenko model since the structural dynamics will be the main source of the vibrations for the case when there is no interface.

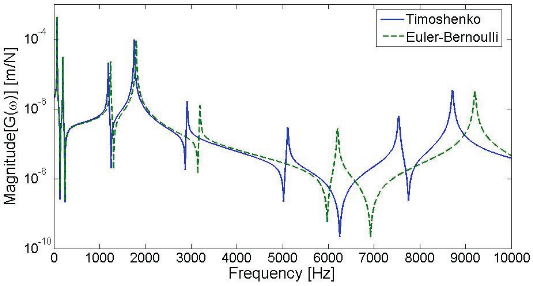

Figure 2: The tool point FRF for highly stiff connection at spindle-holder and holder-tool interfaces.

2.3 Effect Analysis for Tool Point FRF

In order to study the effects of bearing and interface parameters, their values are varied in a wide range [13] about their nominal values which are obtained from the recent literature. For example Figure 3 shows that the bearing stiffness values have a considerable effect on the first two modes of the system which are the rigid body modes, whereas, they have almost no effect on the remaining (elastic) modes. It was observed [13] that the dynamics of the softer bearing pair (front bearings in this case) primarily controls the first rigid body mode whereas the stiffer rear bearings mainly affect the second rigid body mode. Therefore, for the system used, the spindle geometry and bearing properties have the most important effect on the first two modes. This also implies that if chatter develops in one of the first two modes, changing the holder or the tool may not help.

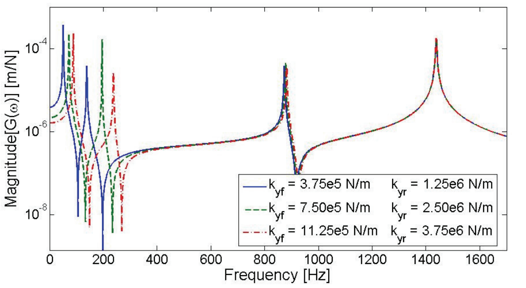

Figure 3: The combined effect of bearing stiffness values on the tool point FRF.

In a very similar way, sensitivity of tool point FRF to the spindle-holder interface dynamics is studied. It is observed that the translational stiffness at the spindle-holder interface dominantly affects the first elastic mode of the FRF. It is also observed that the variations in the rotational stiffness at the same interface have almost negligible effect on the FRF [13]. A similar analysis is performed in order to study the sensitivity of FRF to the tool-holder interface dynamics [13]. It is observed that the translational stiffness strongly controls the second elastic mode. Similar to the spindle-holder interface, the variations in the rotational stiffness at this connection have negligible effect on FRF. Therefore, the observations made so far indicate that, for the first elastic mode, spindle-holder interface is the most important link in the chain, whereas the same is true for holder-tool interface for the second elastic mode in this case study. The connection damping values have similar effects, but on the FRFs peak amplitudes instead of the frequencies. For example, it is observed that the front bearing damping affects the FRF values at the first rigid body mode, whereas translational contact damping at the spindle-holder interface mainly alters the peak value of the first elastic mode [13]. The above conclusions can be used in parametric identification of connection dynamics of a given spindle-holder-tool assembly from experimental measurement of tool point receptance much more easily and accurately compared to previous approaches used. Having the information of which connection parameters affect which mode, identification should be performed by extracting the parameters of interest from their relevant modes.

3 EXPERIMENTAL RESULTS

The measured tip point FRF of the suspended spindle shown in Figure 4 and the model predictions of the same FRF using both beam theories are given in Figure 5. Note that the inaccuracy associated with using Euler-Bernoulli theory increases at higher frequencies.

Figure 4: Spindle suspended for free-free measurements.

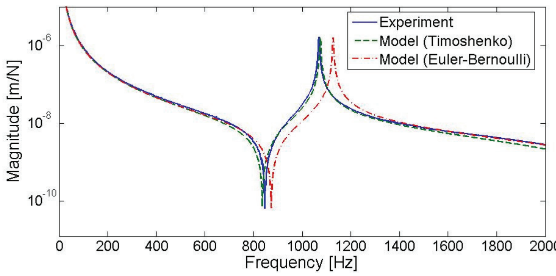

Figure 5: Measured and predicted FRF for the spindle.

A BT40 type holder, in which a carbide tool of 12.7 mm diameter and 175 mm length is inserted with an overhang length of 74 mm, is assembled to the free spindle shown in Figure 4. Then, the tool point FRF of the free assembly is measured by the impact test. The measured FRF and the model simulation of the tool point FRF are given in Figure 6. Note that the interface parameters are obtained by making use of the effect analysis. Performing the effect analysis, it is observed that the spindle-holder interface controls the first mode and holder-tool interface controls the second mode. The interface parameters identified by using the FRFs at the relevant modes only are given in Table 2. Note that, in the parametric identification process, mainly the translational parameters are tuned and average values are used for the rotational interface dynamics.

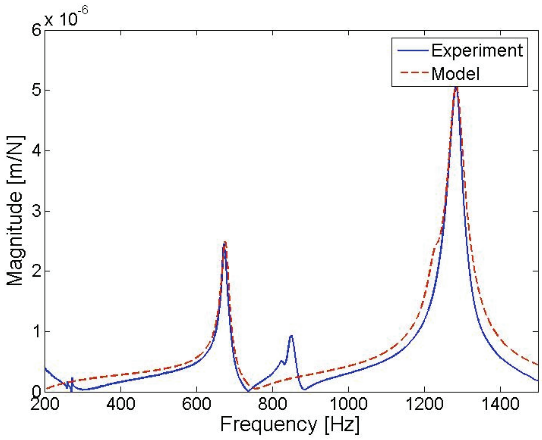

Figure 6: Measured and predicted tool point FRF of the assembly.

	Spindle-Holder Interface	Holder-Tool Interface
Translational stiffness [N/m]	3.05×10^{7}	1.14×10^{7}
Translational damping [N.s/m]	365	34
Rotational stiffness [N.m/rad]	1.5×10^{6}	1.5×10^{6}
Rotational damping [N.m.s/rad]	40	40

Table 2: Connection parameters identified from the tool tip measurement.

4 CONCLUSIONS

In this study, an analytical method that uses Timoshenko beam theory, receptance coupling and structural modification techniques is presented for modeling spindle-holder-tool assemblies in machining centers in order to obtain the tool point FRF which is required for prediction of chatter stability. Effects of bearing and interface parameters on the tool point FRFs are analyzed and important conclusions regarding the mode-interface relations are derived. These results are used for developing a fast and accurate parameter identification approach. The applications of the model are presented and the model predictions are verified experimentally.

The model in its presented form can be used by the spindle or machine tool builders in the design of spindles to optimize the spindle geometry and/or bearing locations for maximum dynamic stiffness at a desired frequency or frequency range. The method can also be used for improving the chatter stability by selecting better tooling and clamping conditions. By using the model proposed, the changes in the system dynamics due to possible variations in the tool/holder types can be followed easily in practical applications. In case where the spindle data (geometry and bearings) is not available, the method can be modified to combine the analytically predicted tool-holder dynamics with the measured spindle FRF.

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