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Generalized method for the analysis of bending, torsional and axial receptances of tool–holder–spindle assembly

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ABSTRACT

Existing researches for the prediction of tool point receptances have focused on developing dedicated methods for cutting tools used in each single cutting operation such as milling and drilling processes. This paper presents a generalized method for the analysis of the tool point receptances of cutting tools suitable for being mounted on a rotating spindle. Translational and rotational dynamic responses related to all axes (X, Y and Z) are simultaneously modeled in a unified way to predict the tool point bending, torsional and axial receptances of all kinds of rotating tools, such as milling, drilling and boring cutters. To facilitate modeling, the tool–holder–spindle assembly is divided into four substructures, i.e., spindle–holder subassembly, shank of tool, fluted part of tool and tool–holder joint interface. The fluted part of tool is modeled as a three-dimensional Timoshenko beam with varying cross-section, while the tool–holder joint interface is regarded as a zero-thickness distributed layer and modeled as a joint substructure composed of a set of independent spring–damper elements. Assembling criterion is derived to couple the dynamic responses of all substructures to calculate the tool point receptances. Meanwhile, compared with past experimental means, a measurement procedure to eliminate the adapter's mass effect on torsional and axial receptances is designed. The proposed method is experimentally proven for two kinds of rotating tools, i.e., mills and drills.

1. Introduction

Self-excited vibrations caused by regeneration of waviness [1], also known as regenerative chatter, result in unstable cutting process, poor surface finish, reduced productivity and damage on machine tools and cutters. In the past several decades, many efforts [2–8] were made by predicting stability lobe diagrams (SLDs) to select the chatter-free cutting process parameters. Reliable SLDs always require accurate frequency response function (FRF) of tool point. Usually, FRF of tool point is experimentally obtained by standard impact testing. This kind of means has two typical characteristics that may limit its wide applications in machining industries. First, it is strongly dependent on the combinations of tool, holder and spindle; hence, repeated and time-consuming measurements should be conducted once the sizes or materials of holder and tool change. Second, measurement of FRF is almost impractical for small-scale cutters widely used in micro- and nanomachining field. To this end, development of generalized computing methods becomes a vital alternative to predict the tool point receptance of the tool–holder–spindle assembly. The practical procedure to realize this can be described as follows:

(i) Divide the tool–holder–spindle system into some substructures.
(ii) Measure the receptance of the substructures that are difficult to be modeled theoretically, such as the spindle.
(iii) Model the dynamic response of the remaining substructures that are easy to be analyzed, such as tool and holder.
(iv) Establish criterion to couple the results obtained from Steps (ii) and (iii).

Regarding this technique, research kernel is how to divide the substructures (Step i) and how to establish the theoretical model of the substructures that could be analyzed (Step iii). Schmitz et al. [9] are the earlier researchers to study this topic and treated the tool–holder–

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Nomenclature

Matrix and vector

$\mathbf{H}_{\mathrm{H}}(\omega)$ the frequency response function (FRF) matrix of the spindle–holder subassembly with ω being the angular frequency

$\mathbf{M}_c$ the mass matrix associated with the N_c nodes of the component c, where c=S or F

$\mathbf{C}_c$ the damping and stiffness matrices associated with the N_c nodes of the component c, where c=S, F or TH

$\mathbf{K}_c$ the stiffness matrices associated with the N_c nodes of the component c, where c=S, F or TH

$\mathbf{Q}_c = [\mathbf{q}_{c,1}^{\mathrm{T}}, \mathbf{q}_{c,2}^{\mathrm{T}}, \ldots, \mathbf{q}_{c,N_c}^{\mathrm{T}}]^{\mathrm{T}}$ denotes the displacements vector of the component c related to N_c nodes with c=H, S or F

$\mathbf{q}_{c,i} = [u_{c,i}, v_{c,i}, w_{c,i}, \theta_{c,i}, \phi_{c,i}, \psi_{c,i}]^{\mathrm{T}}$ denotes the displacements vector of ith node of the component c, where $u_{c,i}$, $v_{c,i}$, $w_{c,i}$, $\theta_{c,i}$, $\phi_{c,i}$ and $\psi_{c,i}$ (c=H, S, or F) designate translational and angular displacements of ith node of the component c related to X, Y and Z axes, respectively

$\mathbf{F}_{c-\tilde{c}} = [\mathbf{f}_{c-\tilde{c},1}^{\mathrm{T}}, \mathbf{f}_{c-\tilde{c},2}^{\mathrm{T}}, \ldots, \mathbf{f}_{c-\tilde{c},N_c}^{\mathrm{T}}]^{\mathrm{T}}$ denotes the loads vector corresponding to the N_c nodes at the component c, and they are applied by the component $\tilde{c}$ (c=H, S or F; $\tilde{c}$ = H, S, F or TH and $\tilde{c} \neq c$)

$\mathbf{f}_{c-\tilde{c},i} = [f_{\mathrm{x},c-\tilde{c},i}, f_{\mathrm{y},c-\tilde{c},i}, f_{\mathrm{z},c-\tilde{c},i}, M_{\mathrm{x},c-\tilde{c},i}, M_{\mathrm{y},c-\tilde{c},i}, M_{\mathrm{z},c-\tilde{c},i}]^{\mathrm{T}}$ denotes the loads vector corresponding to ith node at the component c, and they are applied by the component $\tilde{c}$, where $f_{\mathrm{x},c-\tilde{c},i}$, $f_{\mathrm{y},c-\tilde{c},i}$, $f_{\mathrm{z},c-\tilde{c},i}$, $M_{\mathrm{x},c-\tilde{c},i}$, $M_{\mathrm{y},c-\tilde{c},i}$ and $M_{\mathrm{z},c-\tilde{c},i}$ (c=H, S or F; $\tilde{c}$ = H, S, F or TH and $\tilde{c} \neq c$) designate the forces and moments applied on the ith node of the component c related to X, Y and Z) axes, respectively, and they are applied by the component $\tilde{c}$

Subscript

H the spindle–holder subassembly
S the shank of tool component
F the fluted part of tool component
TH the tool–holder joint interface component

spindle assembly as two substructures consisting of a spindle–holder subassembly, whose FRFs were experimentally measured, and an overhung part of tool, whose receptance was modeled by analytically considering its translational response. They proposed a receptance coupling substructure analysis (RCSA) procedure by which the analytical dynamic response of the overhung part of tool was coupled with the experimentally determined receptance of the spindle–holder subassembly to predict the tool point receptances.

Since RCSA was successfully applied to machining process, there are also some follow-up studies that divided tool–holder–spindle assembly into two substructures [10–14] or three substructures [15–21]. Actually, no matter how to divide, the key issue lies in reliably modeling the dynamic response of the substructures that could be analyzed. On the improvement of modeling accuracy, the past literature has mainly been on including the effect of rotational dynamic responses related to bending vibration [10,19,13], considering the shear deformation of tool and holder [14,16,17,21], and determining stiffness and damping properties of joint interfaces [11,12,18–20,22].

Park et al. [10] pointed out that besides the translational response used by Schmitz et al. [9], rotational degree of freedom also has important influence on the receptance of the tool point, and derived theoretical equations to analytically calculate tool point FRFs by considering translational and rotational dynamic responses related to X- and Y-axes. Albertelli et al. [13] used finite difference method to identify the translational and rotational receptances of the spindle–holder subassembly. Some other researchers [14,16,17,21] used Timoshenko beam theory to study the influences of shear deformation on the tool point FRFs, and found that the prediction accuracy at high frequency can be improved if shear deformations of tool and holder are well included. Specifically, Ozsahin and Altintas [21] studied the influence of tools' asymmetry on tool point receptances by considering the second moment of inertia of fluted tools. Mancisidor et al. [14] predicted the tool point dynamics using Timoshenko beam with fixed boundaries and pointed out that the cut-off frequency problem could be overcome by this approach in the receptance coupling procedure.

On the other hand, since the tool–holder–spindle system has been divided into many substructures, how to model the interfaces between different substructures is another important issue needed to be solved in FRFs prediction procedure. Schmitz et al. [18] developed multiple connection models for the tool–holder interface, and employed finite element method to determine the position-dependent stiffness and equivalent viscous damping values for thermal shrink fit holders. Ahmadi and Ahmadian [12] combined the measured FRFs of holder–spindle and analytical models of the tool via a distributed damped-elastic spindle–holder interface. Ahmadian and Nourmohammadi [20] separately analyzed tool, holder and spindle and synthesized dynamics of these three substructures. Park and Chae [22] utilized Euler–Bernoulli beam to calculate the tool point receptances of modular cutting tools including the dynamic effect of fastener joint, which was identified by using both analytical and experimental dynamic response.

It should be highlighted that the above work is mostly focused upon the prediction of the tool point bending receptances that are only associated with translational and rotational dynamic responses related to X- and Y-axes. This is generally sufficient for peripheral milling since its dynamic chatter problem is mainly bending instability. However, both the bending and axial receptances of tool–holder–spindle assemblies become important in ball end milling, bull-nose end milling and plunge milling applications [23,24]. What's more, the combined torsional–axial dynamics, as well as bending responses, are essential for drilling dynamics analysis [25,26]. That is, besides bending receptances, translational and rotational dynamic responses related to Z-axis has also aroused interests among researchers [27–31].

Schmitz [27] is the first scholar who took into account the translational and rotational dynamic responses related to all axes (X, Y and Z) in an extended RCSA method for the prediction of tool point FRFs in a tool–holder–spindle system. In this work [27], both the tool fluted geometry and the physical properties of joint interfaces, i.e., stiffness and damping, were not considered in the simulation model. Filiz and Ozdoganlar [28–30] investigated the effect of actual fluted geometry on free–free bending, torsional and axial vibrations of cutting tools, which were not amounted in the spindle. This theoretical model [28–30] was further extended by Bediz et al. [31] to predict the bending receptances of end milling tool in a tool–holder–spindle assembly. On the aspect of experimental study, only Schmitz [27] measured the tool point torsional and axial receptances to verify his theoretical model using a homemade additional adapter, which provided surfaces to