Modeling the Orientation-Dependent Dynamics of Machine Tools with Gimbal Heads

Law, M. (a)[1]; Grossi, N. (b); Scippa, A. (b); Phani, A. S. (a); Altintas, Y. (a)*

a) – Department of Mechanical Engineering, The University of British Columbia, Vancouver, Canada

b) – Department of Industrial Engineering, University of Firenze, Firenze, Italy

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Abstract

Machine tools with gimbal heads continuously rotate and/or pivot their spindle heads to maintain the required relationship between the tool and the surface being cut. This results in orientation-dependent dynamic behavior at the tool-center-point, which further results in orientation-dependent machining stability of the system, potentially limiting productivity during continuous cutting. A dynamic substructuring procedure is proposed to model this behavior by orienting machine tool substructures to their proper configuration prior to synthesis to obtain the synthesized orientation-dependent response. Influence of the changing dynamics on chatter stability is further investigated; and, a strong dependence of machining stability on orientation is observed.

1 Introduction

Recent growth in transport and energy sectors that require large part manufacturing to machine complex free-form surfaces has spurred demand for large high-performance multi-axis machine tools [1]. To maintain the required relationship between the tool and the complex free-form surface being cut in such multi-axis machine tools, the tool's lead and tilt angles need be continuously

[1] Presently at the Fraunhofer IWU, Chemnitz, Germany.
* Corresponding author. Tel.: +49 371 5397-1518; E-mail address: mohit.law@iwu.fraunhofer.de

modified. This relationship is maintained by a gimbal head type kinematic configuration with a continuously rotating and/or swiveling spindle head. By swiveling the spindle head, these machines offer additional functionality and versatility by making it possible to approach the part from underneath; as may be required for machining strategies involved with large turbine/impeller manufacturing.

The continuously rotating and/or pivoting spindle heads in such machine tools results in its dynamic behavior (stiffness) at the tool-center-point (TCP) varying as a function of the spindle's orientation. A lack of this dynamic stiffness may lead to unstable regenerative chatter vibrations which are detrimental to the performance and integrity of the entire machine tool system; resulting in: poor surface quality, damage of work piece and machine structural elements, and, ultimately limiting productivity [2, 3]. Since the performance of these machine tools is directly influenced by the machine's dynamic stiffness; it becomes necessary to assess the changing dynamic behavior (stiffness) early in the design cycle such as to provide valuable guidelines for the design of a multi-axis machining system. Moreover, the changing dynamic behavior at the TCP in turn causes the chatter stability and machining productivity to vary as a function of orientation, potentially limiting the productive cutting conditions during continuous cutting.

Evaluation of these varying dynamics at the design stage is a complex process, involving the use large order finite element (FE) models for the entire machine tool [4] that is to be solved for each discrete position and/or orientation by adopting cumbersome re-meshing strategies; or by adopting co-simulation multibody dynamic modeling schemes, which pose their own set of challenges [5]. Earlier investigations by the authors in [6] presented a computationally efficient alternative to evaluate position-dependent dynamic behavior of machine tools based on a reduced model dynamic substructuring strategy. The approach presented was able to treat varying dynamics caused due to translational motion, but did not cover machine tools with gimbal heads which suffer from orientation-dependent dynamics due to the swivelling and/or rotational motion of the spindle head. This orientation-dependency, though important has been less investigated in the literature. Recent investigations by Hung et. al. in [7] also confirmed the importance and influence of the spindle orientation on tool point dynamics and machining stability. Their [7] preliminary investigations however did not include the effects of spindle-bearing assembly dynamics on the overall response.

To overcome the above mentioned issues and to present a comprehensive modeling scheme to model and evaluate the orientation-dependent dynamics in machine tools, a generalized modeling scheme is offered in this paper. At first, as a

representative example of a large machine tool, a virtual machine tool model of a gantry mill with three linear axes is considered in Section 2. As a separate substructure to be synthesized with the machine tool model, a gimbal head with two rotary axes including a swiveling spindle head is also modeled. To investigate the effects of the dynamics of the spindle head on the overall response, a detailed spindle head assembly is modeled using Timoshenko beam elements. A dynamic substructuring approach based on receptance coupling is formulated in Section 3 that facilitates response analysis of the entire structure, i.e. a complete 5 axis machine tool by combining the dynamic models of the individual substructures for a given configuration (orientation) of the spindle head. Simulation driven investigations of the influence of spindle orientation on the tool point dynamics and chatter stability for representative machining operations are discussed in Section 4 and 5 respectively; followed by the main conclusions in Section 6.

2 Modeling the orientation dependent dynamics for a gimbal head machine tool

A virtual machine tool model of a gantry mill with a gimbal head similar to that shown in Figure 1 is considered in this study. The 5 axes of the machine are decomposed into 3 linear axes (X, Y, and Z) on the gantry and 2 rotary/swivel (A and C) axes on the gimbal head as shown schematically in Figure 1.

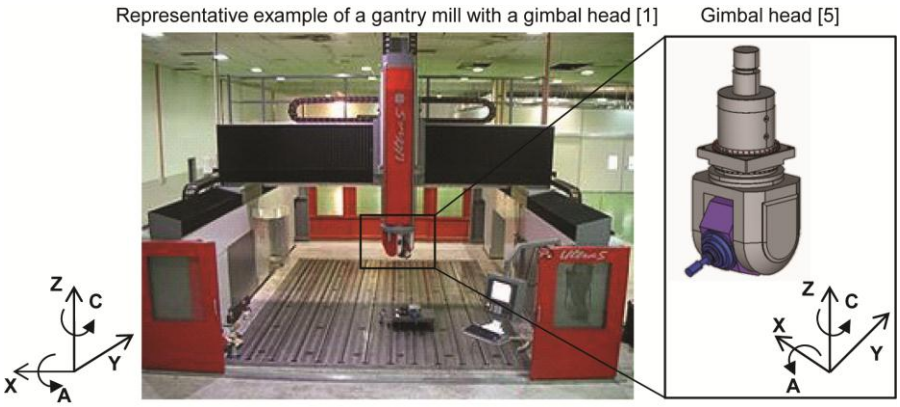

Figure 1: *Representative example of a gantry mill [1], with gimbal head - details shown on right [8]*

2.1 Virtual machine tool model

At first, a detailed FE model of a virtual machine tool without the spindle assembly is constructed within the finite element environment from its available CAD model. This model is based on a large gantry type machine in [8]. The dynamic response at the connection end of the gimbal head, i.e. at location 3 within Figure 2 is extracted and is as shown in Figure 3. Figure 2 includes a schematic of the spindle assembly connected to the gimbal head of the machine tool.

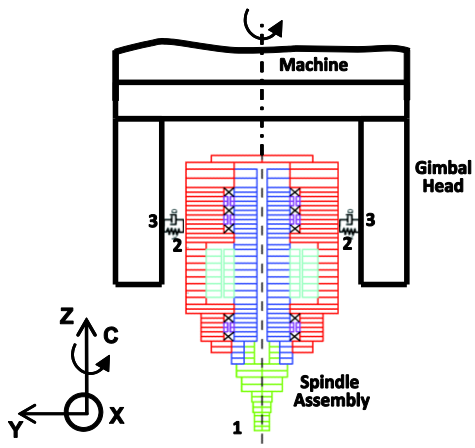

Figure 2: *Schematic of the spindle assembly connected to the gimbal head of the virtual machine*

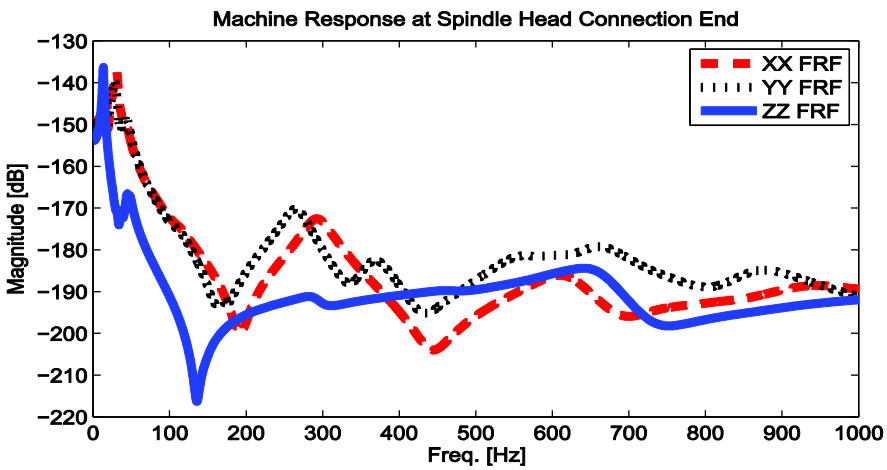

Figure 3: *Frequency response functions at the gimbal head connection head on the machine*

The frequency response functions (FRFs) in Figure 3 correspond to the response due to the structural modes of the large machine tool. A uniform damping of the level of $\zeta = 0.06$ for all structural modes is assumed. The response in the machine Z direction, i.e. along the length of the Z-slide is stiffer than in the machine X and Y directions. The low-frequency modes in the X and Y directions correspond to the bending modes of the Z-slide of the machine.

2.2 Modeling the spindle head

The spindle assembly, shown in Figure 4, includes the tool-tool-holder, spindle shaft, spindle cartridge, bearings, spacers, rotor, stator, housing and other accessories such as nuts and caps. An end-mill cutter of 30 mm diameter with an overhang of 145 mm from the spindle nose with a HSK100 type tool-holder is modeled as a representative tool-tool-holder combination required for machining on this large machine tool. All components of the spindle assembly are modeled with Timoshenko beam elements based on design guidelines in [9]. Bearings are modeled as radial-axial springs, stiffness for which is obtained from the manufacturer's catalogue. Modeling the spindle with beam elements offers the advantage of rotating the spindle assembly to the appropriate configuration (as necessary) using transformation matrices prior to assembly with the virtual machine tool model.

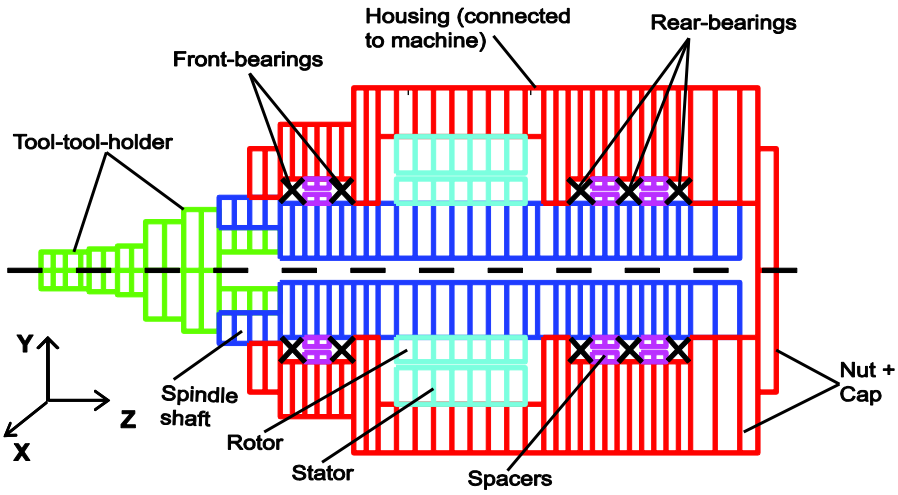

Figure 4: *Detailed finite element model of the spindle assembly including the tool-tool-holder*

The free-free response at the TCP for the spindle head assembly shown in Figure 4 is given in Figure 5. A uniform damping of the level of $\zeta = 0.02$ is assumed in constructing the FRFs in Figure 5. The spindle assembly response is symmetric in the X-Y plane. The response along the spindle Z direction, i.e. along the axis of the spindle shaft is also stiffer than in the spindle X and Y directions. The mode at ~350 Hz corresponds to the spindle bending mode, and the higher frequency response, i.e. >1000 Hz corresponds to the spindle shaft and tool-tool-holder bending modes.

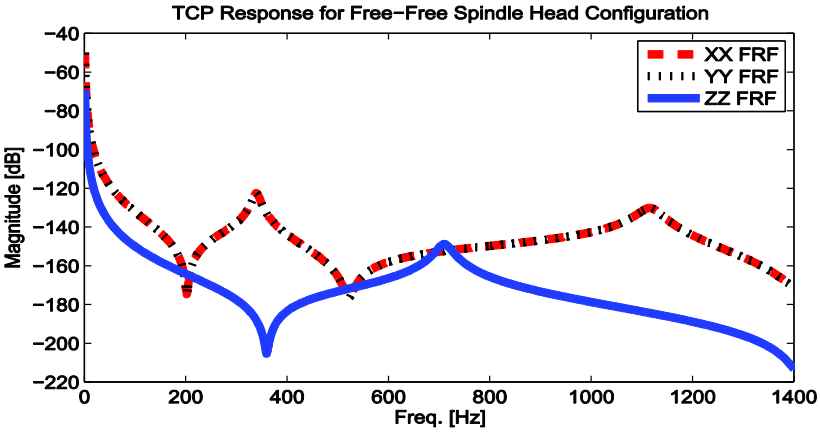

Figure 5: *Frequency response functions at the TCP for free-free spindle assembly configuration*

2.3 Modeling the orientation-dependence

To model the orientation-dependent dynamics, the free-free spindle assembly may be rotated to the proper orientation using beam transformation matrices at the elemental level (as in [8]) prior to synthesis with the machine model. Alternatively, the machine response may be rotated to the proper configuration (using rotational operators) prior to assembly with the free-free spindle head response.

For preliminary investigations in this study, the machine response will be rotated prior to assembly with the spindle assembly to obtain the orientation-dependent dynamics at the TCP. Swivel motion (A axis) as well as gimbal head rotation (C axis) is accounted for by introducing rotational operators such that the oriented machine tool transfer function matrix, $\boldsymbol{\Phi}_{OR_{MC}}$ becomes:

$$\boldsymbol{\Phi}_{OR_{MC}} = \boldsymbol{R}_{x/z}{}^{T} \boldsymbol{\Phi}_{MC} \boldsymbol{R}_{x/z} \tag{1}$$

where $\boldsymbol{\Phi}_{MC}$ is the machine tool FRF matrix at the connection point, i.e. at location 3 within Figure 2; and includes machine FRFs in machine principal directions:

$$\boldsymbol{\Phi}_{MC} = \begin{bmatrix} \Phi_{xx} & \Phi_{xy} & \Phi_{xz} \\ \Phi_{yx} & \Phi_{yy} & \Phi_{yz} \\ \Phi_{zx} & \Phi_{zy} & \Phi_{zz} \end{bmatrix} \tag{2}$$

$\boldsymbol{R}_{x/z}$ within Eq. (1) is the transformation matrix necessary for rotations about the X or Z axis; and may be expressed as:

$$\boldsymbol{R}_x = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\alpha_x & -\sin\alpha_x \\ 0 & \sin\alpha_x & \cos\alpha_x \end{bmatrix}, \text{ and } \boldsymbol{R}_z = \begin{bmatrix} \cos\gamma_z & -\sin\gamma_z & 0 \\ \sin\gamma_z & \cos\gamma_z & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{3}$$

wherein α_x and γ_z are the rotations to be carried out about the A and C axis respectively.

As demonstration of the effect of rotation on the response of the machine at the connection point, transformed machine tool FRFs in machine X and Y directions are compared in Figure 6 for rotation only about the X axis. As evident in Figure 6, there remains no change in the dynamics (FRFs) about the axis of rotation; and only the Y and Z axis (not shown) response will change due to transformations.

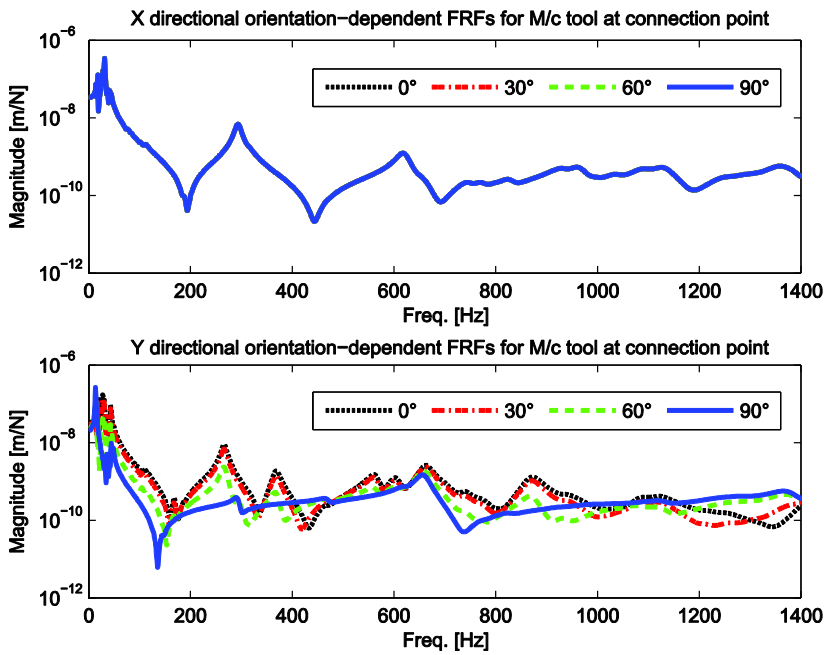

Figure 6: *X and Y direction orientation-dependent FRFs for the machine tool at the connection pt.*

3 Dynamic substructuring

The rotated (oriented) response of the machine tool at the connection point, i.e. location 3 within Figure 2 is combined with the free-free response of the spindle head assembly at location 2 (within Figure 2) to obtain the synthesized orientation-dependent TCP dynamic behavior.

3.1 Receptance coupling substructure analysis

The component receptances for each of the substructures, i.e. the machine tool and the spindle assembly can be represented in a compact matrix generalized form as:

$$\boldsymbol{x}_i = \boldsymbol{R}_{ij}\boldsymbol{q}_j \tag{4}$$

where $\boldsymbol{R}_{ij}$ is the generalized receptance matrix that describes translational component behavior, and i and j are the respective measurement and excitation locations and, $\boldsymbol{x}_i$ and $\boldsymbol{q}_j$ are the corresponding generalized displacement and force vectors. For additional details about the structure of $\boldsymbol{R}_{ij}$, the reader is directed to [10].

The direct receptances at, and cross receptances between the free-end (i.e. location 1 in Figure 2) and the coupling end (location 2 in Figure 2) of the free-free spindle assembly are:

$$\boldsymbol{x}_1 = \boldsymbol{R}_{11}\boldsymbol{q}_1;\ \boldsymbol{x}_2 = \boldsymbol{R}_{22}\boldsymbol{q}_2; and,\ \boldsymbol{x}_1 = \boldsymbol{R}_{12}\boldsymbol{q}_2 \tag{5}$$

Due to symmetry and Maxwell's reciprocity, $\boldsymbol{R}_{12} = \boldsymbol{R}_{21}$. Similarly, the direct receptances at the connection point on the gimbal head attached to the virtual machine tool model (location 3 in Figure 2) are represented as:

$$\boldsymbol{x}_3 = \boldsymbol{R}_{33}\boldsymbol{q}_3 \tag{6}$$

Ensuring equilibrium conditions at the joint part for an assumed rigid joint at the connection between the spindle assembly and the gimbal head attached to the machine tool, the assembled receptances, $\boldsymbol{G}_{11}$, at the TCP in the generalized form are given as [10]:

$$\boldsymbol{G}_{11} = \boldsymbol{R}_{11} - \boldsymbol{R}_{12}(\boldsymbol{R}_{22} + \boldsymbol{R}_{33})^{-1}\boldsymbol{R}_{21} \tag{7}$$

where $\boldsymbol{G}_{11}$ has the same structure as $\boldsymbol{R}_{ij}$, and its constituent receptances are obtained from solutions in Eqs. (5-6).

Each time the TCP response is desired at a different orientation, the machine tool receptances are first transformed to the desired configuration using Eq. (1), and subsequently substituted into Eq. (7) wherein they are combined with the spindle head response.

4 Orientation-dependent dynamic behavior

For the machine under consideration, the spindle-assembly can tilt (A axis) up to ± 95˚ about its neutral position, i.e. the position when the tool axis is aligned with the Z-slide of the machine along the Z axis. Additionally, the entire gimbal head can rotate a full 360˚ about the Z axis, i.e. the C axis motion. Orientation-dependent dynamic behavior is investigated for two cases as shown schematically in Figure 7: for swivel motion of the tool; and for rotation about the Z axis (C axis motion) for when the tool is inclined at 30˚ about the X axis.

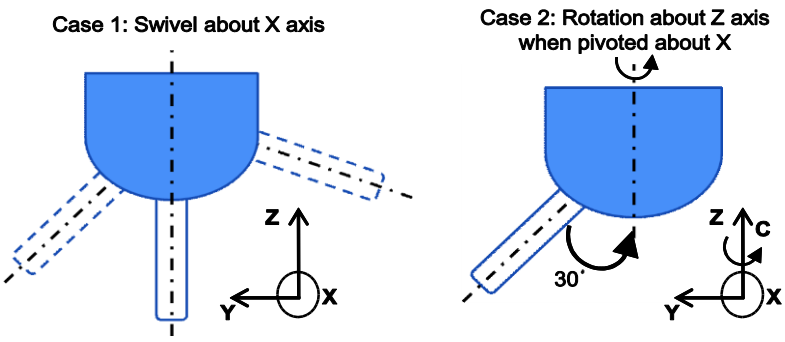

Figure 7: *Investigation of orientation-dependent dynamics for two cases: (1) for swivel motion about X axis; and, (2) for pivot about X axis + rotation about Z axis*

Orientation-dependent response for swivel motion of the tool, i.e. for Case 1 in Figure 7 is as shown in Figure 8. The TCP response is symmetric about the neutral position of the tool, hence comparisons in Figure 8 are limited to the case of swivel motion in the +ve direction. Since swivel motion is about the machine X axis, the X directional response does not change, and as such, only the changing Y

directional response is compared in Figure 8. Furthermore, since the Z directional response is stiffer than the machine X and Y directions, as observed earlier in Figures 3 and 5, Z direction response is also neglected for comparisons.

As evident in Figure 8, the response changes more significantly for higher swivel motions and is found to be more flexible at the 90˚ orientation than between 0-60˚. The dynamic stiffness for the mode at ~260 Hz at the 90˚ orientation is ~100% less than at the at the 0˚ orientation. A slight change in frequency of this dominant mode is also observed due to the effect of orientation. The dominant mode at ~260 Hz corresponds to the global bending mode of the Z-slide of the machine tool along with the spindle head assembly; whereas the dominant mode at ~340 Hz corresponds to the first local bending mode of the spindle head assembly. The oriented-response at the TCP in Figure 8 for the assembled configuration is significantly different than the oriented-response of the virtual machine tool model in Figure 6. The difference is primarily due to the effect of mode interactions between the free-free spindle head modes in Figure 5, and the oriented machine modes; thus also highlighting the importance of including a detailed model of the spindle.

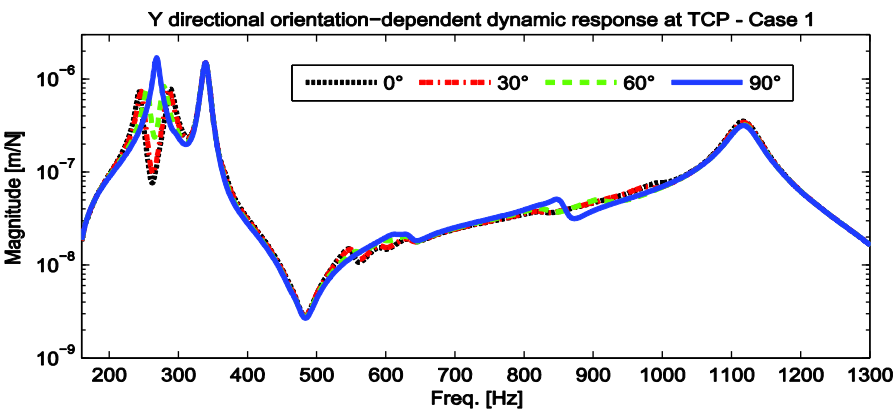

Figure 8: *Orientation-dependent dynamics for the case of swivel motion about the X axis*

Orientation-dependent response for rotation about the Z axis (i.e. C axis motion) for when the tool is pivoted at 30˚ about the X axis, i.e. for Case 2 in Figure 7 is shown in Figure 9. When the spindle head rotates 90˚ about the Z axis, the local X and Y TCP directions, i.e. TCP coordinates get switched; i.e. TCP response in X direction at 0˚ is the same as TCP response in the Y direction at 90˚, and vice-a-versa. This behavior, which repeats for every 90˚ rotation about the Z axis, is

captured by the model – as is evident from comparisons in Figure 9. Though response continuously changes for a full 360˚ rotation about the Z axis, it is symmetric about every 90˚ change in rotation about the Z axis; hence, response comparisons up to only 90˚ are made in Figure 9. Since the spindle head rotates about the Z axis after pivoting 30˚ about the X axis, the Z directional response does not change as a function of rotational angle, and hence is not compared in Figure 9.

The dynamic behavior for rotation about the Z axis is also observed to change, causing both a shift in the frequencies of the dominant modes as well as a change in the dynamic stiffness. Dynamic stiffness for the mode at ~260 Hz is observed to change by as much as 60% for rotation from 0-90˚. The higher frequency response corresponds to the local tool-tool-holder bending modes and is not observed to be a strong function of orientation for either of the cases.

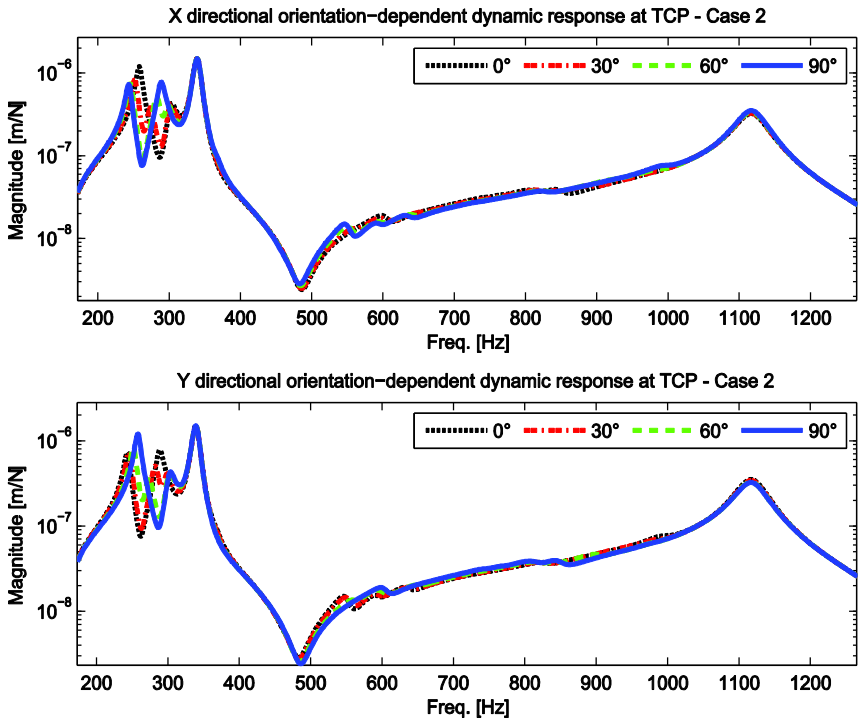

Figure 9: *Orientation-dependent dynamics for the case of rotational motion about the Z axis when pivoted at 30˚ about the X axis*

These gimbal head machine tools are mostly deployed for machining steels; which are cut at lower cutting speeds and tends to generate excitation frequencies in the range of 150-1000 Hz - the range corresponding to machine tool structural modes and the spindle modes [1]. Hence, the TCP response comparisons in Figures 8-9 are limited up to 1300 Hz, since the stability of the process will be dominated by the modal parameters corresponding to the structure and the spindle – as discussed in the next Section; which includes discussions on the influence of orientation of machining stability

5 Orientation-dependent machining stability

When the structural dynamics of the machine vary within the machine's work space due to the tool's constantly changing orientation during continuous cutting, the chatter stability and the resulting limits on the material removal rates vary as well. The changing stability of the milling system is determined using a modal model of the machine and the following characteristic equation [2]:

$$det([\boldsymbol{I}] + \Lambda[\boldsymbol{\Phi}_{OR}(i\omega_c)]) = 0 \tag{8}$$

where $$\Lambda = \Lambda_R + i\Lambda_I = -\frac{1}{4\pi} N_t K_t a\left(1 - e^{-i\omega_c T}\right) \tag{9}$$

is the eigenvalue of the characteristic equation, Λ_R and Λ_I are its real and imaginary parts; N_t is the number of teeth on the cutter; K_t is the cutting force coefficient of the material being cut; a is the axial depth of cut; ω_c is the chatter frequency; and, T is the tooth passing period. $\boldsymbol{\Phi}_{OR}$ within Eq. (8) corresponds to the orientation-dependent directional matrix [2]. Since the response in the Z direction was observed to be sufficienly stiffer than the X-Y directions, chatter stability investigations are limited to the two-dimensional analytical model [2].

The limiting depth of cut, described by the parameters in Eq. (8) and Eq. (9), may be analytically determined as [2]:

$$a_{lim} = -\frac{2\pi\Lambda_R}{N_t K_t}\left[1 + \left(\frac{\Lambda_I}{\Lambda_R}\right)^2\right]. \tag{10}$$

The details of evaluating stability charts can be found in reference [2]. Stability was simulated for continuous cutting for both cases in Figure 7 for full immersion milling (slotting) of AISI 1045 steel with $N_t = 4$; $K_t = 2{,}362$ MPa; and the radial coefficient, $K_r = 0.5$. End-milling of steel is treated as the representative

machining operation to be carried out on this machine. Orientation-dependent machining stability, i.e. stability lobes along with the corresponding chatter frequencies are shown in Figures 10-11 for Cases 1 and 2 respectively.

The change in machining stability for Case 1, i.e. for continuous swivel motion of the tool about the X axis during cutting is given in Figure 10. The stability lobes are observed to be a strong function of orientation; with the stability boundary being lower for higher swivel motions; due to the dynamic stiffness being lower at the 90˚ orientation than between 0-60˚. The absolute minimum stability limit is found to be independent of the orientation and occurs around the dominant spindle bending mode of ~340 Hz; however, the stability limits at some speeds, for example at 2500 RPM and 5000 RPM is found to vary by as must as ~80-100% - depending on the orientation angle. This may be explained by the change in the chatter frequencies that is observed at these two speeds. Chatter at these two speeds occurs at ~260 Hz which corresponds to the global bending mode of the Z-slide and the spindle head, which is also observed to become dominant at certain orientations – as evident from earlier comparisons in Figure 8.

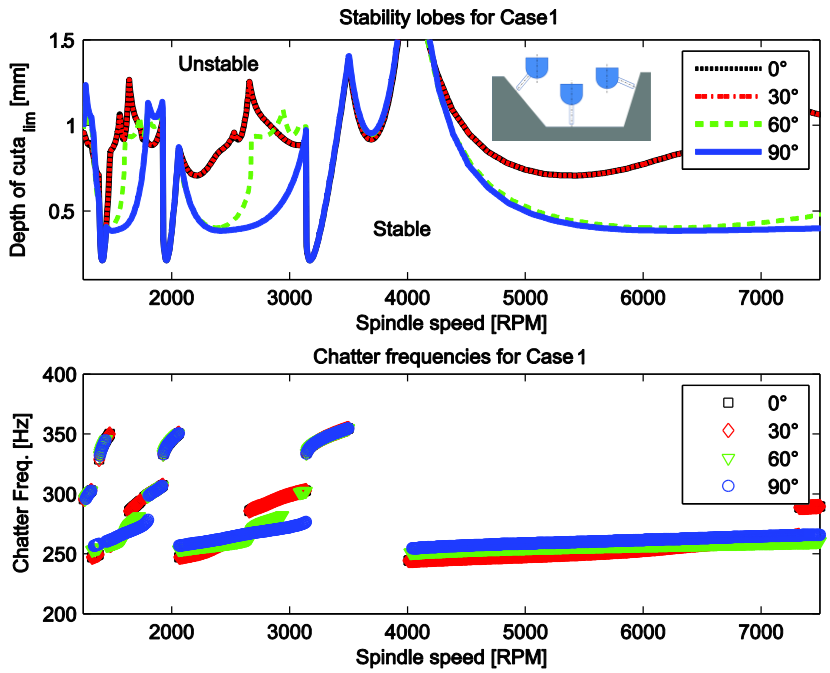

Figure 10: *Orientation-dependent machining stability for Case 1, i.e. for swivel motion about X axis. Stability lobes (top), with the corresponding chatter frequencies (bottom)*

Orientation-dependent machining stability comparisons for Case 2, i.e. for a complete rotation about the Z axis (C axis motion) while the tool is pivoted at 30˚ about the X axis for continuous cutting is shown in Figure 11. The machining operation would be akin to a circular milling operation. Stability comparisons are limited from 0-90˚, since the lobes would be symmetric for every 90˚ rotation about the Z axis. As evident in Figure 11, the stability lobes for the 0˚ and the 90˚ are the same – since the dynamics between 0˚ and 90˚ for the X and Y directions get switched – as observed in earlier comparisons in Figure 9. As was observed in Figure 10 for Case 1, the absolute minimum stability limit for Case 2 is also independent of orientation. However, as before the stability limits are different at certain speeds, and chatter is observed to occur at different frequencies for these speeds. For example, the absolute minimum stable depth of cut is limited by chatter occurring at the ~340 Hz bending mode, whereas at a spindle speed of 6000 RPM, chatter occurs between ~240-260 Hz – depending on orientation.

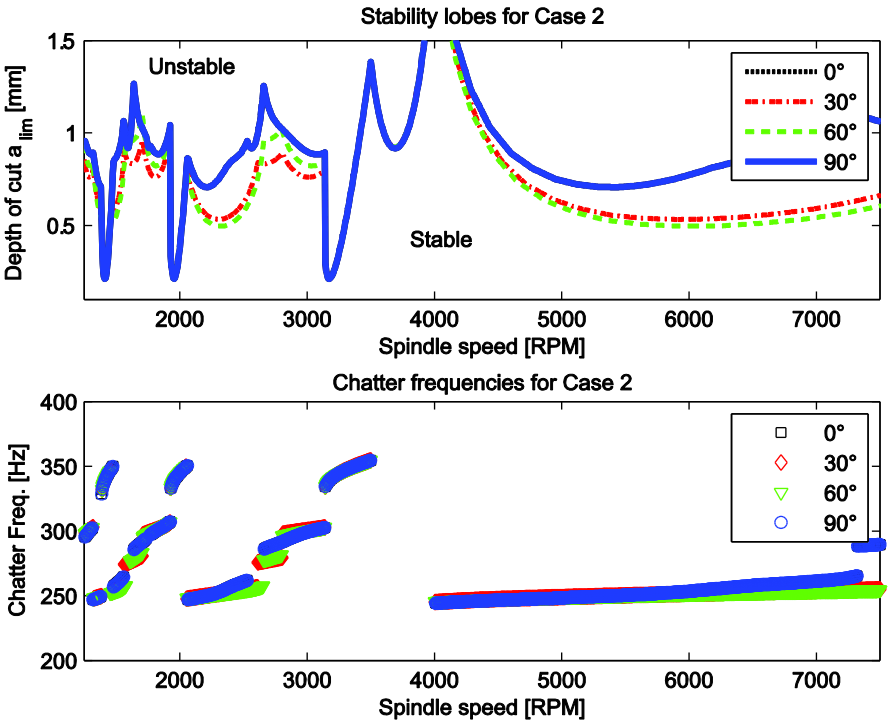

Figure 11: *Orientation-dependent machining stability for Case 2, i.e. for rotary motion about Z axis when the tool is pivoted at 30˚ about the X axis. Stability lobes (top), with the corresponding chatter frequencies (bottom)*

The changing machining stability observed in Figures 10-11 for both cases of continuous cutting may need the planning of dynamically changing machining trajectories such as to ensure stable cutting – which pose its own set of challenges; or, alternatively and more conservatively, it may result in selection of cutting parameters below the lowest of all possible stability thresholds, thereby resulting in a slower material removal process – which is undesirable.

6 Conclusions and future work

For machine tools with gimbal heads which have spindle heads that may rotate and/or swivel during continuous cutting operations, the changing dynamics due to change in orientation and its influence on machining stability of the system was successfully demonstrated in this study. The spindle head assembly modeled as a separate substructure was coupled to a virtual machine tool model using a receptance coupling approach, as part of a dynamic substructuring strategy.

Changing dynamics and stability were investigated for two different cases for swivel and rotational motion of the machine tool gimbal head. Simulation driven investigations show a strong dependence of the TCP dynamics and stability on orientation – varying by as much as 100% in dynamic stiffness and limiting stable depth of cuts for change in orientation within the 0-90˚ rotation and swivel motion. Experimental investigations for model validation are necessary and form part of the planned future work.

The proposed modular approach facilitates evaluation of the influence of different spindle/gimbal heads design concepts on the overall orientation-dependent dynamics at the TCP. Assessing this changing dynamic behavior early in the design cycle is expected to provide valuable guidelines for the design of high-performance multi-axis machining solutions.

Acknowledgments

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