Modeling of Position-, Tool- and Workpiece-Dependent Milling Machine Dynamics

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Abstract: Depending on the machine design, milling machines can show a significant variation of their dynamic properties with respect to the axes configurations, in particular at high speed spindle rotations and high feedrates. Moreover, the workpiece and the milling tool are critical parts of the machine tool and can have a major effect on the dynamic properties. Certain combinations of milling tool, workpiece, tool engagement, process parameters and axes configurations can come along with undesired forced or self-excited vibrations. So far, planning of milling processes usually does not account for these unwanted vibrations. The focus of this paper is to present a modeling framework, which accounts for the abovementioned influences via simulation. The dynamic properties of various workpieces and tools as well as the dynamic properties for many different axes configurations are stored in databases. Based on these databases, the dynamics of any given machine tool configuration can be simulated efficiently based on a substructure coupling approach and an interpolation strategy.

1 Introduction

Although research has been focusing on vibrations during machining for more than sixty years, issues related to machine tool dynamics frequently occur in current production environments. The fundamental works of Tlusty, Tobias and Opitz [1–3] help understanding the occurrence of machine tool vibrations in general and self-excited chatter vibrations in particular. They have introduced stability charts that indicate, which process parameters (spindle speed and depth of cut) allow stable machining. Based on these pioneer works, further extensions of the simulation models are targeted in the following decades. More recently, a major contribution was made by Altintas, Budak and Merdol who introduced frequency domain approaches to efficiently determine stability limits for cutting processes [4, 5]. Time domain simulations of cutting processes do not directly yield stability limits, but they offer great flexibility [6, 7]. Such time domain approaches can comprise nonlinear cutting force models, model small immersions and intermittent cuts and are well suited for the simulation of the machined surface [8–11]. Moreover, due to their accuracy, time domain simulations frequently serve as references for frequency domain stability predictions.

Many more valuable research works related to machining dynamics have been conducted, so that only selected references are given in the previous paragraph. It can be noticed, that most of the works assume that the properties of the dynamic system remain constant. However, during the regular operation of a machine tool its structure undergoes natural modifications: As soon as the machine axes move, the dynamic properties of the machine change. Moreover, changes of workpieces, fixtures, tools and tool holders result in variations of the dynamic properties. Consequently a simulation framework is targeted, that allows for a modular assembling of the component dynamics to the dynamics of the complete machine tool. The general framework for dynamic substructuring proposed by De Klerk et al. [12] sums up large parts of the relevant coupling theory, but does not deal with practical applications. According to this framework analytical and experimental component models can be combined in different domains (physical, modal and frequency domain) to yield a description of the assembly dynamics. Further theoretical background can be found in the works of Allen [13] (modal coupling), Brechlin [14] (modal and frequency based coupling) and Voormeeren [15]. The last-mentioned reference gives a perfect review on the existing analytical component reduction and assembly techniques.
Only few practical adaptations of the substructure coupling approaches to machine tool structures can be found in literature: The modification of tools and tool holders is addressed by the receptance coupling technique. Schmitz [16] proposes to couple the dynamic compliance of the tool holder, expressed as frequency response functions (FRFs), with an analytical model of the tool. Later, this method was refined by using a tool dummy that is decoupled from the structure before the new tool is coupled [17–19]. The frequency based coupling of a machine tool slide to a frame was successfully realized in [20]. In this work measured frequency responses of the frame are coupled with analytically derived frequency responses of a slide. Law has investigated the effect of axes positions on the machine dynamics in physical and frequency domain based on reduced component models [21, 22]. In this work FRFs are efficiently predicted for different axes configurations and in [21] stability charts are calculated in different axes configurations as well.

A modular simulation system for machine tool dynamic would comprise methods to efficiently couple experimental or simulated models of machine structure, workpiece and tool in different axes configurations and allows efficient prediction of the dynamic properties (poles, mode shapes) of the assembly. Moreover, deflections and stability during different machining processes can be determined. This paper targets a first version of this modular simulation system while focusing on simulation. The idea of the modular simulation system is to efficiently predict the stability of the machining process and hence is similar to the idea of the comprehensive Machining Prediction Software developed by Jeppsson at Boeing [23, 24]. However, the targeted simulation system focuses on efficiently predicting the effects of structural modifications on the global dynamics of the machine assembly and thus can be seen as an extension to the system successfully realized by Jeppsson.

The concept of the modular simulation system is explained in sect. 2. The preparation of the machine structure models is presented in sect. 2.1, the preparation of fixture-workpiece and tool models is presented in sect. 2.2. In sect. 2.3 it is postulated that the three databases are readily prepared as the section deals with the details of the assembly procedure. As soon as assembly has been performed an eigenvalue problem is stated and solved. The modal parameters stemming from the solution of the eigenvalue problem are used for simulation studies in sect. 3. Within sect. 3, the study related to the variation of the axes configurations is correlated with experimental results. Finally, the paper is concluded in sect. 4.

2 Efficient modeling of structural modifications

The innovative approach followed here is to perform extensive calculations on component level in advance and to store the component specific data, ready for future access, in databases. The components considered here are machine structures, fixture-workpiece assemblies and tools. As indicated in Fig. 1, for analysis of a specific machine configuration the necessary models are taken from the prepared databases. For the axes configuration under study, the machine structure matrices are selected. Moreover, the matrices describing the mounted tool and the mounted fixture-workpiece assembly are selected from the databases. Besides the mass, damping and stiffness matrices (M, K, C), the databases contain the degrees of freedom matrices, the coordinates and elements matrices and the nodes vectors. The last mentioned properties are used for the definition of constraints and for visualization purposes.

2.1 Position dependent machine structure

The machine structure model does not comprise workpiece and tool. The machine structure considered here is depicted in Fig. 2, right. The structure consists of machine bed (1), machine table (2), column (3), slide (4), main spindle (5) and ball screw drives. The structural components are connected to each other via three dimensional spring damper elements (SDEs). These SDEs represent coupling elements as linear guides, foundation elements and bearings. The nodes of the SDEs are connected to the structural components via multi point constraints (MPCs). For a more detailed description of the mathematical formulation of these inter-component connections the reader is referred to [25]. The correct choice of the SDE-parameters is crucial for gaining a realistic machine model. Usually experience or correlation with measurements is needed for the correct choice of the stiffness-, and damping-parameters. Approximate values can be found in the suppliers’ catalogues or in literature [26, 27]. For the connection to the tool and the workpiece interface degrees of freedom (DOFs) are defined on the spindle and the table, respectively, Fig. 2, left. The tool is connected to the last node of the main spindle, this node has six DOFs. The workpiece is connected to four nodes that are located in the corners of the machine table. Each of these four nodes has six DOFs. The interface nodes of the workpiece are constraint to the table via interpolation MPCs, see [25].
2.1.1 Reduced and damped component models.

In order to gain small matrix sizes and to account for material damping, model reduction is applied according to the description in [28] for all structural components except the spindles. Having applied the MACNEAL-reduction [29], the differential equation for one of the structural components takes the form

\[ \begin{align*}
\dot{\eta} + \gamma \eta &= R_{MacNeal} \left( \begin{bmatrix} 0 \\ -u_b \end{bmatrix} \right) \\
\end{align*} \]

where \( M_c = \begin{bmatrix} 1 & 0 \\ 0 & 0 \end{bmatrix} \), \( K_c = \begin{bmatrix} \Omega & -\phi^T b^T \\ -b\phi & -b\psi \end{bmatrix} \)

and \( R_{MacNeal} = \begin{bmatrix} \phi & \psi \\ 0 & 1 \end{bmatrix} \) is the reduction basis. (2)

The reduction basis is built from the mode shapes of the free floating component \( \phi \) and the residual flexibility attachment modes \( \psi \). Formulations of these reduction basis ingredients can be found in [15, 30], for example. Modal damping \( (D_1, D_2, \ldots, D_n) \) is defined for the \( n \) modes (un-
damped angular eigenfrequencies \( \omega_1, \omega_2, \ldots, \omega_n \) of the free floating component to build the damping matrix

\[
C_c = \begin{bmatrix}
\Sigma & 0 \\
0 & 0
\end{bmatrix}
\]

(3)

where \( \Sigma = \text{diag} ([2\omega_1 D_1, 2\omega_2 D_2 \ldots 2\omega_n D_n]) \).

The movement of the reduced component is described by the modal weights \( \eta \) and the coupling forces \( g_b \). A single component, not affected by external forces, is excited only by the deflections \( u_b \) of the coupling nodes, which are selected from all deflection via \( u_b = bu \) with the Boolean Matrix \( b \).

As the spindles (main spindle and ball screw spindles) do not comprise of many nodes, there is no need to reduce the corresponding full finite element (FE) matrices \( M \) and \( K \). However, modal damping is defined and a transformation to physical domain as described in [30] yields the damping matrix

\[
C_c = \begin{bmatrix}
C & 0 \\
0 & 0
\end{bmatrix}, \quad \text{where} \quad C = (M\phi) \Sigma (\phi^T M).
\]

(4)

Contrary to the reduced structural components, the deflections of the spindles remain present in the analytical component description.

### 2.1.2 Assembly of structural components

The assembly approach followed is demonstrated by considering only two reduced structural components. Some of their interface DOFs (\( u_{b1} \) and \( u_{b2} \)) are constrained according to \( R_1 u_{b1} - R_2 u_{b2} = 0 \). First, the component matrices are arranged in block diagonal form and the uncoupled equation of motions is

\[
\begin{bmatrix}
\Omega_1 & -\phi_1^T b_1^T & 0 & 0 \\
-b_1 \phi_1 & -b_1 \psi_1 & 0 & 0 \\
0 & 0 & \Omega_2 & -\phi_2^T b_2^T \\
0 & 0 & -b_2 \phi_2 & -b_2 \psi_2
\end{bmatrix}
\begin{bmatrix}
\eta_1 \\
\eta_{b1} \\
\eta_2 \\
\eta_{b2}
\end{bmatrix}
= \begin{bmatrix}
0 \\
u_{b1} \\
0 \\
u_{b2}
\end{bmatrix},
\]

(5)

if external forces, mass and damping terms are neglected. If Boolean matrices \( B_1 \) and \( B_2 \) are defined such that \( u_{b1} = B_1 u_{b1} \) and \( u_{b2} = B_2 u_{b2} \) the transformation

\[
\begin{bmatrix}
\eta_1 \\
\eta_{b1} \\
\eta_2 \\
\eta_{b2}
\end{bmatrix}
= L
\begin{bmatrix}
\eta_1 \\
\eta_{b1} \\
\eta_2 \\
\eta_{b2}
\end{bmatrix}, \quad \text{where} \quad L = \begin{bmatrix}
1 & 0 & 0 \\
0 & 0 & -\phi_2^T R_1^T \\
0 & 1 & 0 \\
0 & 0 & \phi_1^T R_2^T
\end{bmatrix}
\]

is the assembly matrix,

realizes the connection of the substructures. Hence, the coupled form of (5) is

\[
\begin{bmatrix}
\Omega_1 & 0 & B^T \\
0 & \Omega_2 & R \\
B & R
\end{bmatrix}
\begin{bmatrix}
\eta_1 \\
\eta_{b1} \\
\eta_2 \\
\eta_{b2}
\end{bmatrix}
= \begin{bmatrix}
0 \\
u_{b1} \\
0 \\
u_{b2}
\end{bmatrix},
\]

(7)

where \( B = [R_1 B_1 b_1 \phi_1 - R_1 B_2 b_2 \phi_2] \)

and \( R = -R_1 B_1 b_1 \psi_1 B_1^T R_1^T - R_2 B_2 b_2 \psi_2 B_2^T R_2^T \).

The LAGRANGE multipliers \( \lambda \) express the coupling forces between the coupled structures. Depending on the type of the MPC between the structural components, the matrices \( R_1 \) and \( R_2 \) take different forms. If coupling of mass and damping matrices is done in the same way, it can be noticed, that the constraints do not affect the assembled mass and damping matrices. As machine axes are moved, different DOFs become involved in the constraints, and only the stiffness matrix has to be updated.

By means of the model reduction the number of degrees of freedom is greatly decreased. For the machine presented in Fig. 2 the full model comprises approx. 800,000 DOFs whereas the reduced model comprises only approx. 5,000 DOFs. In a preparatory calculation the assembled stiffness matrix is determined and stored for different axes positions. The sampled positions form a grid all over the working volume of the machine tool as indicated in Fig. 2, right.

### 2.2 Workpiece and tool models

Fig. 3, left, depicts the machine table and the fixture-workpiece assembly mounted to the table. The fixture-workpiece assembly is modeled in a similar way as the other structural components. It is only that additional nodes are introduced on the surface that is machined. These nodes are constrained to the workpiece surface and their deflections can be read out after model reduction. Later, the deflections of these nodes are interpolated to determine the deflection of the workpiece at the tip of the tool, see sect. 2.3. Moreover, four SDEs are part of the workpiece-fixture assembly. These SDEs offer the possibility to model damping and stiffness properties of the face-to-face contact between fixture and machine table. The lower nodes of the SDEs are the interface nodes to the machine table and comprise six DOFs each.

The interface plane between main spindle and tool is chosen just before the standard shape of the standard tool interface as indicated in Fig. 3, right. Consequently, the standard tool interface is not a part of the analytical tool...
model. The tool is modeled via one dimensional beam elements. Different sections are defined for the elements, to capture the tool geometry. The tool matrices are not reduced. Instead, modal damping is defined and the damping matrices are constructed as for the main spindle and the ball screw spindles, see sect. 2.1.1. The interface of the tool is formed by one node comprising six DOFs.

2.3 Assembly of substructure models

Evaluation of the dynamic machine properties for a certain axes configuration is done in two steps. The procedure is illustrated in Fig. 4. First, those eight points of the grid of axes positions, Fig. 2, right, that form the smallest possible cuboid around the point to be evaluated, are determined. For each of these points the component matrices of machine structure, workpiece and tool are assembled. This assembly is done based on the compatibility matrix B. This Matrix expresses the compatibility of the interface DOFs of table and workpiece on the one hand, and the compatibility of the interface DOFs of spindle and tool on the other hand. Assembly of the matrices is realized by multiplication by the assembly matrix L, which is the nullspace of B [12]. A generalized eigenvalue problem is formulated and its solution yields mode shapes \( V \) and poles \( p \) for the assembled system. Moreover, the upper residual flexibility \( UR \) is determined: The limited number of mode shapes already models parts of the static compliance. This part is called spectral compliance \( G_{sp} \) and is subtracted from the total static compliance \( G \), which is determined as the inverse of the assembled stiffness matrix [15]. Now, eight sets of modal parameters \( V, p \) and \( UR \) are available for the eight nodes of the identified cuboid. If we take the examples presented in Fig. 2 and in Fig. 3 and only the x- and y-directions are evaluated, the mode shape matrix \( V \) has the size 34 by number of modes. Two rows of \( V \) correspond to the deflections at the tool tip in x- and y-direction and 32 rows of \( V \) correspond to deflections of the evaluation nodes on the workpiece in x- and y-direction.

Second, the modal parameters are weighted according to the current axes configuration. Shape functions as known for linear hexahedron and linear quadrilateral finite elements [31] are incorporated to interpolate the modal parameters Fig. 4, right. Poles and upper residuals are directly weighted by multiplication with the shape function values. As the sign of the modes shapes arbitrarily varies, each mode shape is scaled by the sign of the first entry before weighting, to make sure that the shapes have the same phase. Following up on this interpolation in the
grid of axes positions, the position of the tool relative to the machined plane on the surface is evaluated. The evaluation nodes forming the smallest rectangle around the tool tip are identified. The mode shape entries related to these nodes are weighed by values of the shape functions for a quadrilateral element to yield the deflection of the workpiece at the tool tip. The interpolation between the evaluation nodes on the workpiece causes the dimension of the mode shape matrix to shrink down to 4 by number of modes, if only x- and y-direction are considered.

As soon as the two described interpolations are completed, the modal parameters (poles, eigenvectors and the residual flexibility) completely describe the relative compliance between workpiece and tool. These parameters are used to synthesize the relative FRFs between workpiece and tool, which in turn are used for the frequency domain calculation of chatter stability limits. Alternatively the modal parameters are directly used for time domain simulations of the machining process as explained in [28, 32].

3 Investigation of different machine configurations

Section 2 describes the theory behind the modular simulation system. In this section, the simulation approach is adopted for the calculation of the dynamic properties and chatter stability limits of the sample machine structure presented in Fig. 2. Modifications of axes positions, fixture-workpiece assembly and tool are made and the effect on the relative dynamic compliance between workpiece and tool and the chatter stability limit is predicted.

3.1 Modification of axes positions

For the evaluation of the axes position influences, the machine is equipped with workpiece no. 1 and tool no. 1, see Fig. 7. In a first study, Fig. 5, left, the slide is positioned in two different y-positions. The direct FRFs $G_{xx}$ and $G_{yy}$ express the relative dynamic compliance between tool and workpiece. They are synthesized for both axes configurations. Moreover, the chatter stability limits are determined for both positions according to the frequency domain approach presented in [4] and for the process illustrated in Fig. 5, right. The dashed and solid lines in the bode plot and the gray and black lines in the stability diagram below are determined for fixed axes configurations. If the dynamic compliances are compared, we can see a major effect of the y-position on the mode at approx. 113 Hz, which is excited in x-direction. At the higher y-position this mode leads to a greater peak in $G_{xx}$ than it does in the lower y-position. Analysis of the corresponding mode shape reveals that this mode is characterized by torsion of the col-
modification of axis position

**Figure 5:** Effect of axes positions on FRF and chatter stability.

umn about the y-axis. Naturally, this torsion is more easily excited by a force in x-direction that acts at higher y-positions. Furthermore, the predicted phase responses indicate less phase shifts for the lower y-position. This correlates well with the observation, that higher stability limits are predicted for the lower y-position. Particular differences are remarkable for revolutions in the range from 1800 to 2200 min\(^{-1}\). In this range the stability limit for the higher y-position lies in between 8 and 20 mm whereas the limit for the lower stability is much higher than 20 mm in the complete interval.

As described in [33] modal parameters can easily be transformed to coefficients of digital filters. These digital filters allow a step-by-step time domain simulation. In [32] this approach is complemented to account for the residual flexibility and is used for the time domain simulation of milling processes. The filter coefficients are predetermined for every time step. During simulation the filter coefficients are continuously updated, to cope for the changing dynamic properties. This procedure is followed here for the time domain simulation of the slot milling process presented in Fig. 5, right. The simulation starts at the lower y-position of the slide and finishes at the upper y-position. The relative deflection between tool and workpiece is plotted and shows an increasing mean value, which correlates with the increase in static compliance. Moreover, the vibration amplitudes seem to increase along the tool path. The spectrogram plotted bellow the deflection time signal gives an impression of the frequency content of the signal. The outstanding frequency is the tool passing frequency.
but the chatter frequency at 118 Hz seems to gain importance at higher y-positions. All in all the spectrogram indicates a stable machining process, which is close to the stability limit. If the variance of the dynamic properties is neglected and instead chosen to be constantly equal to the properties in position no. 2, the simulation predicts a clearly unstable machining process. This is indicated by both, the time signal, showing high vibration amplitudes, and the frequency spectrum, showing dominant peaks at the chatter frequency.

Although the stability chart and the time domain simulation with constant machine dynamics indicate instability for position no. 2, the time domain simulation with variant machine dynamics predicts a stable machining process. Apparently, it is not sufficient time for the process to become unstable. This observation is seen as a remarkable result.

FRFs are measured for the machine configurations discussed above by means of an electrodynamic exciter. The exciter is elastically supported in the crane using rubber bands and its stinger is connected to an impedance head being glued to tool no. 1. The tool tip FRFs, measured for the x-directions in positions no. 1 and no. 2, are depicted in Fig. 6, left (gray lines). These compliances are compared to the corresponding simulated FRFs (red lines). As opposed to the compliances presented in Fig. 5, the absolute dynamic compliances of the tool are focused in this case. Moreover, the FRFs presented in Fig. 6, are determined when the machine table is positioned at a higher distance from the tool. In the case of position no. 1 measured and simulated compliance show a good agreement. In the case of position no. 2 a remarkable disagreement is visible in the frequency band 110 to 135 Hz. The eigenfrequencies predicted and measured for the mode present in this band differ almost about 20 Hz.

The process presented in Fig. 5, right, is conducted experimentally. The accelerations of workpiece and tool are captured by accelerometers and the spindle revolutions and the axial depth of cut $a_p$ are varied to identify the stability limit. The dots in Fig. 6, right, represent the conducted experiments. Depending on the color of the dots, the corresponding processes are assessed as stable, unstable or as processes close to the stability border. The experimentally identified stability limits show good agreement with the predicted stability limits in some parts of the stability chart. At revolutions higher than 2000 min$^{-1}$ the predicted stability limits are higher than the ones identified from cutting tests. Possibly, the machine tool model could be improved in a refined model updating process. Uncertainty persists about the joint properties and the cutting force model. Nonlinear effects are not captured by the machine model, up to now. As mentioned related to Fig. 5, the

**Figure 6:** Experimental verification of the prediction of $G_{xx}$ and the stability limits.
3.2 Modification of workpiece & fixture and tool

The simulation studies presented in Fig. 7 predict the effects of modifications of the fixture-workpiece assembly and the tool on the dynamic properties and the stability limits. Again, the stability charts are calculated for the process presented in Fig. 5, the y-position corresponds to position no. 2. The other axes positions are stated in the figure. The fixture-workpiece assembly known from Fig. 3 is exchanged by another assembly. The new fixture-workpiece assembly (workpiece no. 2) is statically slightly stiffer in y-direction but weighs only a bit more than half of the old assembly (workpiece no. 1). A major effect of the modification of the workpiece-fixture assembly is remarkable related to the mode at approx. 130 Hz (with workpiece no. 1). This mode shifts to approx. 200 Hz if the new assembly is mounted and leads to a lower resonance. Initially the mode at 130 Hz limits machining productivity at revolutions between 1100 and 1200 min\(^{-1}\). In this range the stability limit is predicted to be in the range \(a_p = 10\) to 13 mm. If workpiece no. 2 is mounted, this stability limit at these revolutions is moved to much higher values of \(a_p\).

The tool can have a major effect on the stability limits. This is proven by the study presented in Fig. 7, right. The initial tool no. 1 is a stiff and short indexable cutter, whose dynamic flexibility can be neglected compared to the flexibility of the spindle and the remaining machine structure.
The new tool no. 2 is a long and slender tapered tool. The compliance of the tool has a remarkable effect on the total compliance of the machine assembly. The amplitude and phase responses do clearly indicate differences in both direct FRFs $G_{xx}$ and $G_{yy}$. The great compliance of the tool effects that the complex dynamic compliances are moved in the direction of the positive realpart. This “gooseneck-effect” [34] comes along with a reduction of the phase shift in wide ranges of the frequency spectrum. In the example in Fig. 7 only a sharp phase drop remains at approx. 155 Hz. The stability limits are affected as drastically as the dynamic compliances. After modification of the tool, the machine assembly behaves almost like a single degree of freedom system and consequently one set of stability lobes is relevant for the stability chart. The predicted stability lobes (black lines) are narrow and leave broad gaps where, theoretically, high values of $\alpha_p$ should be possible.

4 Summary and Outlook

Undesired forced and self-excited vibrations can limit the productivity of machining processes. Consequently one strives to design machine tools and processes that are not likely to show this kind of unwanted behavior. More and more machine tool manufacturers and machine tool users use simulation to predict which the effect of modifications of machine and processes is. Common evaluation criteria are the mode shapes of the machine assembly, the relative dynamic compliance between work piece and tool or chatter stability charts.

Until now, holistic modeling of machine assemblies, comprising machine structure, tool fixture and workpiece, is time expensive, needs expert knowledge and comes along with severe uncertainties regarding the model accuracy. In order to face these modeling issues it is seen as a long term research goal to provide industry a modular simulation system, which allows efficient and reliable prediction of machine and process dynamics. A particular benefit of this simulation system is the ability to efficiently predict the effect of structural modifications, which are faced regularly. The basic idea of the simulation system presented here is to store the dynamic properties of machine structures and their attachment components (fixture-workpiece assemblies and tools) in databases. For the evaluation of a new machine configuration the prepared component models are taken from the databases and an efficient prediction of the dynamic properties of the configuration is possible.

The paper deals with a sample machine structure, two different fixture-workpiece assemblies and two different tools and focusses on simulation. For a grid of axes configurations the machine structure properties are stored in a database. In order to evaluate machine configuration, the three substructures are coupled and the modal parameters are interpolated in the grid of axes configurations. Following this approach, the relative dynamic compliance between workpiece and tool and the chatter stability limits are predicted within seconds. Moreover, the predicted modal parameters are used for a time domain simulation of the milling process whereby the time variant machine dynamics are taken into account.

Future research activities will target extensions of the modular simulation system. One important feature is to combine experimental and analytical models. The interfaces of the machine structure (machine spindle and table) could be characterized by frequency response functions and analytical models of tool and workpiece could be coupled to this experimental model. This way, uncertainties related to the analytical model of the machine structure could be excluded and the expensive modeling of the machine structure is no longer necessary.

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