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Axis position dependent dynamics of multi-axis milling machines

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Abstract

Numerous analytical methods are available to predict the stability of milling processes. Most of these methods base on the assumption, that the dynamics of the machine tool are time invariant. This assumption seems to be valid in many cases. However, in case of huge translational or rotatory axes movements or process-induced changes in the work piece's mass and elasticity a time variant dynamic model might be needed. This paper presents a method to model the axis position dependent dynamics of a multi-axis milling machine. According to this method, the modal parameters of the machine tool are predetermined in different discrete axis positions. An interpolation strategy allows calculating the modal parameters in arbitrary resolution along arbitrary tool paths. Here, an exemplary 2.5-dimensional milling process serves as an example. The conventional step-by-step time domain simulation procedure is complemented by the modal interpolation strategy to account for changing machine dynamics. The effect of changing dynamics on the process is determined and a comparison to a cutting test is performed.

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1. Introduction

Vibrations during machining can have different reasons. One of the reasons is the regenerative effect that can cause excessive chatter vibrations [1]. This kind of vibration has already been described by Tlusty, Tobias and Opitz [2, 3, 4]. They have introduced the so called stability chart that separates all possible combinations of revolution speed and depth of cut into stable and instable combinations. In the recent decades research has focused on the development of algorithms to efficiently calculate such stability charts for turning and milling processes. Altintas and Budak [5] have presented the so-called zero order approximation, where the time-periodic unperturbed milling force is averaged out and the stability chart is efficiently approximated in the frequency domain. This method is particularly useful for modelling high immersion milling processes. For low immersion milling with only few teeth in cut Merdol and Altintas [6] presented the multi frequency solution which was later extended by Bachrathy [7] to cope with more complex tool geometries. Besides the methods in frequency domain several time domain approaches are available. In time domain, the vibrational perturbation of the cutting process can be described by a system of delay differential equations (DDE). These DDE can be solved approximately with help of discretization techniques. Bayly and his colleagues have presented the temporal finite element method [8]. For each temporal element they parameterize polynomial functions to approximate the vibration of the tool during the cut. Insperger and Stépán have introduced the semi-discretization method [9]. According to this method the delay term is kept constant for a short time. For this small time the resulting ordinary differential equation (ODE) is solved under the restriction, that its solution is compatible to the solution of the preceding and subsequent ODEs. Besides solving the DDEs approximately, a numerical step-by-step time-domain approach is possible. For each time step the force acting on the tool and work piece is calculated based on the present

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cutter-work piece engagement. The deflection response of the machine structure to this force is determined. This deflection influences the engagement and thus the force changes. The new force is used to repeat the calculations in the next time step. This step-by-step approach has been adopted by several authors, e.g. [10, 11, 12].

The abovementioned research works are announced here to represent the large amount of work carried out on modelling regenerative chatter in cutting processes. Most of the documented models are built for time invariant structural dynamics. This approximation seems to be valid in many cases. However, the dynamic properties of machine tool structures can vary for various reasons. Tool and work piece exchanges alter the dynamic system. Moreover the removal of material from the work piece can have a noticeable effect. As soon as a machine tool undergoes excessive axis movements or synchronous movements of several axes, the changing stiffness and mass distributions lead to changing dynamics which might influence the process vibrations.

According to [13], modal parameters can be interpolated to describe the dynamics of a work piece, whose dynamic properties change remarkably, when material is removed. Law and his colleagues [14] set up a reduced three axis milling machine model that can be moved to different positions. Based on this model Frequency Response Functions (FRFs) are determined for different axes positions and are fed to frequency domain stability simulations. Moreover [15] has developed a method to interpolate FRFs, that can represent crossing eigenmodes.

As far as we now, time domain process simulations that account for changing dynamics due to axes movements have not been studied excessively so far. However, time domain approaches can handle more aspects (e.g. nonlinearities, changing cutter-work piece engagements) than frequency domain approaches. As a consequence, this paper deals with a method that allows incorporating the time variant structural dynamics in the step-by-step time domain simulation of process vibrations. A three axis milling machine serves as an application example and an exemplary 2.5-dimensional milling process is simulated.

2. Time and positon variant dynamics of machine tool

Changes of the machine axes positions can affect the dynamic properties. Here, a method to account for the resulting time variant machine dynamics is presented. The proposed method interpolates the modal parameters (poles and eigenvectors) between a set of spatial sampling points. The interpolation strategy can be used to calculate FRFs or time domain forced responses for complex tool paths accounting for changing dynamics. The following paragraph describes how FRFs have been determined experimentally for a three axis milling machine. The FRFs determined for different axis positions are compared. The subsequent paragraph presents the basic idea of modelling time or axis position dependent dynamics. The issue of changing mode orders is addressed as well as the extension of a time domain simulation approach of forced responses.

2.1. FRFs measured in different axis positions

The dynamic compliance of a three axis milling machine is determined experimentally by measuring frequency response functions in two axis positions. In each of the axes positions the 3 by 3 FRF-matrix

$$\boldsymbol{H} = \begin{bmatrix} G_{xx} & G_{xy} & G_{xz} \\ G_{yx} & G_{yy} & G_{yz} \\ G_{zx} & G_{zy} & G_{zz} \end{bmatrix} \tag{1}$$

is determined. This frequency dependent matrix describes the relative dynamic compliance between tool and work piece. Figure 1 shows a picture of the measurement setup that is used to determine G_{xx}. The force excitation is done with a hydraulic exciter that is positioned between a dummy tool and a dummy work piece. The generated force $F = F_{stat} + F_{dyn}$ is measured as well as the accelerations on the work piece and the tool side ($\ddot{x}_{wp}$, $\ddot{x}_{tool}$). Moreover the relative displacement x_{rel} between work piece and tool is directly captured by an inductive sensor.

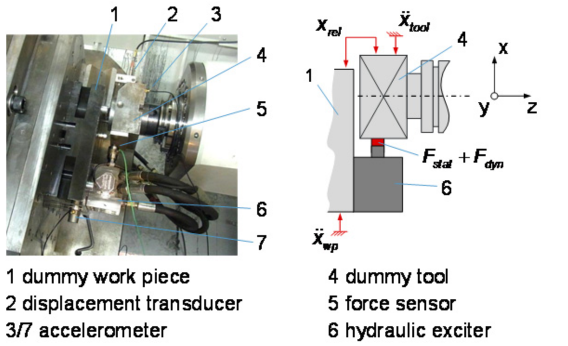

Figure 1: Measurement setup for determination of G_{xx}.

The sketch in Figure 2 shows the two different y-positions that have been considered. The absolute values of the direct FRFs G_{xx}, G_{yy} and G_{zz} are plotted with a linearly scaled ordinate for the two different y-positions. Naturally the dynamic compliance differs between the Cartesian directions, but noticeable differences appear between the tested y-positions as well. The filled areas give a good impression of the differences due to the y-position-variation. The differences in the static compliances are easy to explain by considering the changing leverages. The changes in the dynamic properties cannot be explained just as intuitively. Especially in the frequency range between 60 and 90 Hz the dynamic compliance in x-directions seems to be sensitive to modifications of the y-position. The z-direction seems to show a major sensitivity in the range between 120 and 150 Hz. The dynamic properties between the discrete set of measured positions is unknown. Although the filled areas illustrate the range between the measured compliances, it is theoretically possible, that positions in between show compliances that exceed the filled areas. This issue can be resolved by considering the modal parameters instead of the frequency dependent compliance as is explained in the following paragraph.