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Toolbox for control system analysis of machine tools

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Abstract—Machine tools have to reach high levels in accuracy and speed. For the detailed analysis of the machine dynamics with respect to these properties, a machine model has to include the main concepts of multibody dynamics and feedback control systems. With such a model available, dynamic investigations in frequency-and time-domain can efficiently be done. This work shows how to build a multibody system, and its integration in the feedback control loop. A control system model being coupled with high DOF structural model permits the simulation of interactions between machine tool manipulators and its spatial effects at the TCP (Tool Centre Point) including the position dependency of the properties. The available results and analysis functionalities of a proprietary implementation in a simulation environment are explained and illustrated.

I. INTRODUCTION

For the simulation of mechanical structures of machine-tools, FE-Methods and formulations of multibody dynamics (MB) are commonly used at universities and in industry. If a more complete simulation is required, like simulations in time domain or frequency responses of drives in a complex mechanical structure, a model for the control system has to be added. The FE-method is a popular way to describe machine tool structures. With the help of frequency responses and state-space representation, the coupling with a control system can efficiently be processed. But this way leads to increased computing time and decreased flexibility [1], especially for simulation of structure and control interaction.

The computing time with reduced FE-Models [2] can be significantly decreased. In addition, with the reduction the relevant mechanical mode-shapes can be considered. The formulation of MB in global coordinates [3, 4] delivers extensive modeling of a machine tool structure. The model structure allows direct coupling with a control system. With such a model, the machine tool can be evaluated and predicted due to its structure and control parameters already in a conception phase of a machine tool development [5].

The model allows efficient transient simulations like the positioning of the machine tool axis due to parameter dependent set points. With a global coordinate system for each body of the machine structure in three linear and three rotary directions, a suitable description of spatial cross-talk effects during positioning can be described, outlined in figure 1. In [6], the tilting of driven axes of a machine tool has been measured and simulated. The simulations were carried out by FEM- and MB-simulations. FE-method generally achieves low uncertainties. Using MB method and with applying reduction factors for the stiffness of the couplings between the bodies, whereby elastic body structures can be taken into account, the dynamic behavior especially for low and medium frequencies can also be reproduced qualitatively and quantitatively. The adjustment of MB-systems and FE-Models can easily be provided for compact structures. For slimmer structures with bigger offsets between load and gravity center points, p.m. for beams, a more detailed MB structure has to be created to cover elastic deformations. While keeping model limitations in mind, this paper shows the conceptions of a MB-model in global coordinates and its integration in a closed control loop system, presented in a proprietary user interface. The various available analysis functionalities in time and frequency domain are illustrated and explained.

Figure 1: Cross-talk due to acceleration

mass, M

X

Y

Z

TCP

guide-way system

ΔZ_TCP

ΔZ

ΔY

ΔX

Figure 1: Cross-talk due to acceleration
II. CLOSED LOOP SYSTEM

The following paragraph describes the components of the integrated machine model such as the mechanical structure and the closed loop control-elements concepts including the velocity and position control parameterization.

A. Mechanical Structure

The bodies of the MB-system are defined by their center of mass in the three direction linear and rotatory directions, which yields 6 degrees of freedom (DOF) for each body:

\[ q_i = (x, y, z, a, b, c) \] (1)

The inertias of the bodies define the mass-matrix of each body in 6 DOF. The couplings between the bodies are defined by physical stiffness resp. damping values in 6 DOF as well as offsets between the coupling points. In this step, the rotational matrices are linearized (“relative matrices”) to provide the complete special state-space representation (1). The mechanical structure as a MB-system provides the position dependent mass-, damping-, and stiffness-matrices M, D and K.

\[ M \cdot \ddot{q} + D \cdot \dot{q} + K \cdot q = f \] (2)
\[ q = (q_1, q_2, ..., q_{n-1}, q_n)^T \] (3)

Out of (3), the state space can be further described:

\[ x = A \cdot x \] (4)
\[ x = (q, \dot{q})^T \] (5)

The input matrix B resp. the output matrix C are calculated p.m. at the drive input points resp. at the measurement points in the same manner as K is calculated, thereby B is acting on acceleration level of the state space. The mechanical structure as state space for the closed looped system yields:

\[ \dot{x} = A \cdot x + B \cdot u \] (6)
\[ y = C \cdot x \] (7)

Also created by relative matrices, the C-matrix grips the states at measurement-, drive- and TCP-position.

B. Closed loop

For the closed loop, the cascaded PPI controller shown in figure 2 has been chosen. The PPI is commonly applied in machine tools and well known. The mechanical frequency responses of the state space equations (6) and (7) directly deliver the control parameters following [3]. The determination of the control parameters are briefly shown next:

1) Velocity-Loop PI

The time constant \( T_n \) of the velocity control loop is chosen by using:

\[ T_n = \begin{cases} \frac{1}{\kappa} & \text{for indirekt position measurement} \\ \frac{1}{\kappa} \left( 1 + \frac{1}{\omega_0} \right) & \text{for direct position measurement} \end{cases} \] (8)

\[ \kappa = \omega_0 \cdot \lambda^{0.8} \text{ and } \lambda = \left( \frac{\omega_0}{\omega_1} \right)^2 \] (9)

where \( \omega_0 \) is the resonance frequency and \( \omega_1 \) is the antiresonance frequency obtained from the mechanical drive frequency response out of the state-space equations (6) and (7). \( \kappa \) is the gain corresponding to the total mass of the driven axis of the velocity closed loop. Equation (9) describes a damping optimal value for a second order system. The proportional gain \( K_p \) can be obtained by using:

\[ K_p = J \cdot \kappa \] (10)

where \( J \) is the sum of the moved mass from the driven axis.

2) Position Loop P

The proportional gain parameter \( k_p \) of the position loop can be given as followed:

\[ k_p \leq \begin{cases} \frac{\kappa}{4} & \text{for indirekt position measurement} \\ \frac{\kappa \cdot \omega_0}{4 \cdot (\kappa \cdot \omega_0)} & \text{for direct position measurement} \end{cases} \] (11)
3) Drive dynamics

The implantation of the current control loop of the drive can be approximated with a first order low pass behavior with a time constant between 0.5ms and 10ms e.g.

III. SIMULATION

Based on the modeling concepts introduced in chapter II, a closed loop model can be built automatically by defining the mechanical structure of a machine tool. This chapter shows how to create a mechanical structure and how to simulate in frequency and time domain with a developed proprietary graphical user interface (GUI): Toolbox for control system analysis of machine tools, figure 3.

A. Input of a mechanical structure

The buildup of a mechanical structure requires the definition of the bodies (see figure 4), their couplings (see figure 5), and the drive- and the measurement-positions resp. directions. In the following, an example machine (see figure 8) has been chosen to show the functionalities of the GUI. With the definition of the bodies and their couplings the kinematics of a machine tool is generated automatically. A body is defined by its center of gravity, its mass and moment of inertia according to the global coordinate system. These definitions provide the M-matrix. A coupling point is defined by the connected bodies, position, direction, stiffness- and damping-values in 6 DOF. Therewith the D- and K-matrix are defined. For the B- and the C-matrices, Definitions about position and directions are sufficient. With the definitions above, the control-parameters are calculated automatically following (8) - (11) based on [3]. The derived control parameters in the toolbox can manually be changed and adapted any time. The effects of changing a control parameter (see figure 6) are automatically calculated and shown in every simulation carried out.
B. Frequency domain

The following frequency responses at predefined output points can be plotted for individual axis directions and drives:

- Current-control-loop
- Mechanical open loop V(s)/F(s): velocity output to force input (both at drive position)
- Mechanical open loop V(s)/Pos(s): position output at position measurement to force input at drive position
- Velocity open loop: output at drive measurement ($K_p = 0$) to velocity set point.
- Velocity closed loop: output at drive measurement to velocity set point.
- Position open loop: output at position measurement ($K_p = 0$) to position set point.
- Position closed loop: output at measurement position to position set point.

The location of the internal position measurement system for the simulation is defined accordingly to the location of the axis measurement location (reader head) of the machine or at the TCP.

The simulation in the frequency domain allows analyses, like the estimation of the reachable bandwidth of a control system in the concept phase of a machine or the parameter adjustment for an existing machine.

The main advantage of global coordinates (1) offers the estimation of cross-talk-effects as shown in figure 7, where a cross-talk mode of the position loop in Z direction at the TCP due to the X-Drive shows peaks at 22 and 36Hz. In figure 8, the mode shapes of these relevant resonance frequencies are shown.

C. Time Domain

In the time domain, transient simulation can be carried out. Set-points can be generated for given jerk, acceleration, and velocity values. The position length can freely be chosen in the working envelope of the machine.

Figure 9 shows the set points generated with the following values: jerk 25m/s$^3$, acceleration m/s$^2$, velocity 6000 mm/min and positioning distance 10mm, as well as the response of the closed loop system on position- and acceleration-level. The contouring error due to the typical first order low pass behavior of the mechanical system can clearly be seen.

Figure 10 shows the maximum cross-talk at TCP in Z-direction due to positioning in X-direction at 0.12 sec. and 0.24 sec. The acceleration in X-direction has also their maximal values at these times: So the acceleration dependency of the cross-talk at the TCP can be stated.
The graphical illustration at these times is shown in figure 11, where the cross-talk effect can be seen as tilting of the base compared with the X-axis, which causes the deviation at the TCP in Z-direction.

IV. CONCLUSION

The most compliant mechanical component in a stiff machine tool is usually given by the couplings between the mechanical machine components like linear guide way or rotary bearings. The deflection behavior in the low and medium frequency range is actually given by the deflection of these coupling elements. Based on that, the approach with MB dynamics leads to quite comparable results with those obtained by FEM methods for low and medium frequencies. For the creation of MB structures, several proprietary user interfaces have been developed, where kinematic position dependencies for static behavior and mode-shapes can be analyzed. These interfaces deliver the necessary MB system matrices of the mechanical machine structure for the control system analysis.

In the control toolbox presented here, a control model is implemented, where the system matrices of the mechanical structure are integrated in a feedback control system, in which first control parameters are generated automatically following [3]. Afterwards control parameters can be changed easily, so the frequency responses from drives to other drives and the TCP can interactively be analyzed. In addition to control parameters, transient responses can efficiently be analyzed due to their positioning parameters, like jerk, acceleration, velocity and path lengths.

With these functionalities, the interaction of multiple controlled drives within the machine structure resulting in a relative displacement of the TCP can be analyzed in dependence of several parameters and with respect to speed and accuracy in arbitrary direction. Systematic lateral deviations of the TCP such as inertial cross-talk reflect the chosen configuration of drives and structure. Further functionalities of the presented tool cover the influence of arbitrary process forces and gravity.

Future steps include further enhanced modeling, like the integration of FE-reduced models, measurement and simulation of systematic structural effects and the efficient compensation of these deviations, like cross-talk.

V. REFERENCES