Model-based Feedback Control via Control Integrated Observers *

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Abstract

Today’s development of machine tools utilizes various simulation tools to design machines which are more cost and energy efficient, have higher dynamics, stiffness, and accuracies. In increasing extent, these simulation tools are currently interconnected and allow an interaction by defined interfaces. This leads to even more exact depictions of the reality and more complex machine models.

In the case of machine behavior the first natural frequency limits the performance of machine tool axes. In order to predict the machine behavior during linear movements or machining simple models are sufficient. In this paper a rigid body simulation is installed as Kalman-Filter on a machine tool control and runs in real time. The estimated position and velocities of the Kalman-Filter are used to control the axis. The paper will present the model generation, adaption, and verification via modal analysis. To allow linear movements with the rigid body simulation the model matrices need to be updated in each time step. The implementation of the model updating and the resulting linear motion of the model bodies during the run time of the control will be explained. The achieved results and improvements are evaluated by the identification of the dynamic test rig behavior at different positions.

Key words: control technology, rigid body simulation, real time simulation

1. Introduction

Through the connection of different simulation tools such as structural simulations via FEM, simulations of the feed drives and process forces etc., holistic machine tool simulations are achieved. In the scientific environment these efforts are summarized under the headline “virtual machine tool” 1). By using today’s computing power the calculation of the holistic simulations take several hours or even days. In order to predict the machine behavior in real time on the machine tool control several simulation tools can be replaced by real machine components such as the control loop of the feed drives. Furthermore, the machine control works with a cycle time of 1 to 2 ms. By considering the sampling theorem natural frequencies above 500 Hz cannot be controlled. Presuming that the machine components are designed towards high stiffness and damping properties, the coupling elements such as the carriages, linear guides and ball screws, etc. are the elements with the highest compliance. The compliance of the coupling elements can be depicted by using rigid body simulations.

The paper will focus on a rigid body simulation that is implemented in the machine tool control as Kalman Filter. In section 2 the state of the art will be presented. This is followed by the description of the test rig, the modelling and the model verification via experimental modal analysis in section 3. Finally the influences on the machine model, the control integration and validation will be described in sections 4 and 5.

2. Model-based Compensation of Machine Errors

The cascaded control is the most commonly used control structure in industrial feed drives of machine tools 2). Based on the cascaded control, structure extensions are made by different feed-forward, feedback or model based observers to reduce tracking errors and dynamic dislocations. Further sensors or actuators are in most cases used to compensate machine errors. Brecher et al. integrated adaptronic compensation modules on a Z-slide to compensate static and dynamic deformations of slider structures (e.g. in portal machines) 3), Zatarain et al. applied an acceleration sensor at the TCP 4). With the additional acceleration signal and the scale signal a model with 2 degrees of freedom is set up and integrated in a Kalman Filter. The Kalman filter reduces the noise of the feedback position signal and improves the accuracy and the dynamic behavior of the test bed.

Jonsson examines the dynamic behavior of the machine tool structure in different poses using the FE-method 5). In order to determine the system behavior in real time, he took the determined natural frequencies and modes and interpolated between the sampling points. Sekler utilizes the FE-bodies of the machine components and interconnected them by means of the Penalty-method, thus, the individual modes can be analyzed in the current machine position. By a modal reduction of the machine model the first 6 eigenvalues are identified using the QR algorithm in real time 6). Uhlmann et al. uses a Kalman-Bucy Filter containing a model of the mechanical structure which is built from FE-models and is reduced by the use of Krylov-subspace technique for model order reduction 7). The presented results on the compliant test-rig show that the first mechanical natural frequency is not limiting the controller bandwidth and that the dynamic dislocations can be reduced. By using Krylov-subspace or the component modal synthesis according to Craig and Bampton 8) the
machine model is time invariant and limited to oscillations. In 9) an approach is shown to represent a linear movement of machine axis by using Craig and Bampton model reduction. But until now, the calculation is not real time capable.

Kono et al. evaluated five different modeling approaches for machine tools 10). They compared rigid body simulations and simple elastic body simulations (where different machine components are modeled as elastic body) as well as FEM simulations with a modal analysis at a three axis machine tool. The results show that even the rigid body simulation with modeled coupling elements is sufficient to meet four out of the first six natural frequencies.

Based on these results and the aim of a real time computation that allows also a linear movement of machine axes the paper focuses on rigid body simulations implemented in the control as observers. Zirn 11) and Weikert 12) presented a procedure to build the rigid simulation models which will be used in the paper.

3. Model of the Two-Axis Test Rig

The following chapter will introduce the test rig and the used control model to estimate the machine behavior.

3.1 Two-Axis Test Rig

The model based observer is implemented on a two axis test rig. A maximum velocity of 1 m/s can be reached in both directions with an acceleration of 7 m/s². The linear movement is realized by ball screw drives and provides a working area of 600 x 550 mm. As machine tool control Beckhoff TwinCAT 3 is used. Figure 1 shows the test rig with the used global coordinate system and the designation of the test rig bodies.

![Fig. 1 Test rig](image)

The cross slide body J is thereby designed as a modular body. The stiffness of the structure of body J can be adjusted for further research by a replacement or removal of support elements.

3.2 Modelling

The rigid body simulation is used to build the machine model and bases on the equation of motion.

\[
M\ddot{q} + D\dot{q} + Kq = F
\]  

(1)

The respective matrices \( M, D \) and \( K \) are built by an approach which is mentioned by Weikert and Zirn 11). Fig. 3 shows the formalism to create the global stiffness matrix \( K \) and the rigid body model of the test rig. The global stiffness matrix \( K \) is calculated by using the local stiffness matrix of each spring-damping-unit (SDU) between two bodies. The local stiffness matrices \( K_{im} \) between body I and body J for an individual SDU \( m \) can be described with equation (2).

\[
K_{im} = \begin{bmatrix}
N_{im} \cdot K_{m} \cdot N_{im} & -N_{im} \cdot K_{m} \cdot N_{jm} \\
-N_{jm} \cdot K_{m} \cdot N_{im} & N_{jm} \cdot K_{m} \cdot N_{jm}
\end{bmatrix}
\]  

(2)

Here \( K_{m} \) is the diagonal stiffness matrix of each SDU. The SDUs represent the stiffness of each carriage and linear guide way. \( N_{im} \) and \( N_{jm} \) are the referential matrices between the center of gravity of body I or J to the SDU \( m \) with the following format

\[
N_{im} = \begin{bmatrix}
1 & 0 & 0 & 0 & \Delta x_{im} & \Delta y_{im} \\
0 & 1 & 0 & 0 & 0 & 0 \\
0 & 0 & 1 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 \\
0 & 0 & 0 & 0 & 0 & 1
\end{bmatrix}
\]  

(3)

where \( \Delta x_{im}, \Delta y_{im} \) and \( \Delta z_{im} \) represent the distance between the center of gravity of body I and the SDU \( m \) in the respective direction X, Y and Z.

The four local stiffness matrices \( K_{im} \) between body I and J (represented by the carriages) are summarized to the coupling element SDJ. SDI represents the coupling between body I and the ground and SDK represents the respective coupling of body K to the underlying body J. Subsequently the coupling elements are summarized and arranged in the global stiffness matrix \( K \) as shown in Fig. 2.

![Fig. 2 Arrangement of coupling elements in the global stiffness matrix](image)

Additionally two extra lines and columns are added to the matrix to couple the drives in X- and Y-direction. The mass matrix \( M \) is a diagonal matrix where the inertia of the respective body is used in the respective direction. The inertia as well as the distances between the carriages and the bodies’ center of gravity are determined from CAD data. The stiffness parameters are obtained from catalogues and in feed direction the stiffness of the ball screw is measured. The damping matrix is built by transferring the damping ratio from the modal analysis via the mode matrix. The determination of the damping matrix will be explained in the following section.
3.3 Model Verification

For the modal analysis a LMS Scadas III front-end and LMS Testlab 11.0 software is used. Nine tri-axial acceleration sensors are mounted on the test rig where three sensors are installed at each body. Body J and K are excited next to the spindle nuts by impulse testing. Each tri-axial sensor delivers three acceleration transfer functions in the respective direction. The signals of the three sensors of each body are accumulated in X-, Y-, and Z-direction to transfer functions of the three bodies in the respective direction.

In order to transfer the measured damping ratio to specify the damping matrix $D$ of the control model, the natural frequencies and the mode vectors of the undamped systems are determined by equation (4).

$$\det(-\omega_0^2 M + K) = 0 \quad \omega_0 = \frac{1}{2\pi}$$

The mode vectors $\tilde{q}$ are determined by

$$(-\omega_0^2 M + K)\tilde{q} = 0$$

and combined to the modal matrix $\Phi = \tilde{x}_1, \tilde{x}_2, ..., \tilde{x}_n$. The natural frequencies and the mode shapes of the control model are compared with the results of the experimental modal analysis. Each mode of the model up to a frequency of 350 Hz is matched with one of the measured modes and mapped with the corresponding damping ratio. With the modal mass matrix, the damping ratio $D_m$ and the natural angular frequency $\omega_0$, the modal damping matrix can be determined with

$$D_{mij} = 2 \cdot \Phi^T M \Phi \cdot \omega_0 D_{lj}.$$  

Finally, the damping matrix can be determined by back transformation

$$D = \Phi^{-T} D_{mij} \Phi^{-1}.$$  

The damping matrix is implemented in the control model and the acceleration transfer functions in X- and Y-direction are identified. In both cases the structure is excited via impulse hammer. The transfer function $A_y(\omega)$ (Eq. (8)) is identified by exciting the X-slide (body K) next to the spindle nut (see Fig. 1) in X-direction and measuring its acceleration.

$$A_y(\omega) = \frac{x_{body X}(\omega)}{F_{body X}(\omega)}$$

The transfer function $A_y(\omega)$ is identified by exciting the cross table (body J) in y-direction (next to the spindle nut; see Fig. 1) and measuring the acceleration of the X-slide (body K) in Y-direction by using the following equation

$$A_y(\omega) = \frac{x_{body Y}(\omega)}{F_{body Y}(\omega)}$$

In both cases the control model is excited with a Dirac-impulse. These transfer functions are compared with the summarized acceleration transfer function of the three acceleration sensors mounted on body K. Fig. 4 shows the transfer behavior from the model and the experimental modal analysis.

The first natural frequencies in X-direction at 46 Hz and 76 Hz are oscillations of the complete test rig in X-direction and a rotation around the Z-axis. The two frequencies are mainly caused by the stiffness of the mounting feet. The detected frequency at 226 Hz shown in the model is a movement of body K in X-direction. By considering the mode shapes in Y-direction a movement of body J and K in the model as well as in the modal analysis is visible at a frequency of approximately 50 Hz. The second natural frequency in the control model is approximately at 82 Hz and presents a movement of the cross table (body K) as well as the X-slide (body K) in Y-direction. The amplitudes are adjusted by the mentioned procedure to determine the damping matrix in a satisfactory way. A deviation less than 15 % for the amplitudes of the first natural frequencies in both directions is achieved. With empirical adjustment even better results can be realized. Summarized it can be seen that the control model is sufficient to determine the dynamic behavior of the test rig and will be integrated in the machine tool control as model based observer.

3.4 Comparison with FE-analysis

The used rigid body model do not consider the dynamic behavior of the flexible bodies such as the cross table (body J) and the X-slide (body K). In order to evaluate the influences of the dynamic behavior of the machine components a finite element analysis is realized. The cross table body J is thereby the machine component with the highest compliance. Fig. 5 shows the simulation results. Thereby the ball screw of the X-axis and the X-slide (body K) are simulated simultaneously with the cross slide. The first natural frequencies are generated by the ball screw at 356 Hz in Z-direction and 363 Hz in
Y-direction.

The first three natural frequencies generated by the cross table body J are at 462 Hz, 489 Hz and 601 Hz. The machine control and the control integrated machine model are working at a cycle time of 2 ms. By considering the sampling theorem the natural frequencies generated by the cross table structure do not influence the control model prediction negatively.

3.5 Model Influences

The dynamic behavior of machine tools is mainly influenced during machining by process forces or changes in the machine set up. Set up variations are caused by movements of machine slides and changes of the load masses such as work piece or fixture changes. To allow a prediction of the dynamic behavior of a machine tool via model based observers the changes of the machine set up must be known in the model. Fig. 6 illustrates the position influences of body K during the movement from the middle position towards X=300 mm. To enhance the visibility of the dynamic behavior a further load mass of 150 kg is mounted on the X-slide.

Figure 7 presents the changes in the acceleration transfer function by varying the load mass from 0 kg, 100 kg to 200 kg. The first natural frequency is decreasing from approximately 50 Hz towards 43 Hz as well as the amplitude is reduced by 20 per cent by an increase of the load mass. Furthermore, the amplitude of the rotational mode around the Z-axis is increasing and shifted towards lower frequencies.

Changes of the X-slide position and the additional load masses influence the machine and model behavior in a not negligible extend. Therefore an automatic load mass identification and a position update is integrated in the machine model.

4. Control Integration

The following section will focus on the integration of the machine model into the machine tool control. Therefore the machine model is transformed into a state space model

\[
q = \begin{bmatrix}
0 & 1 \\
-M^{-1}K & -M^{-1}D
\end{bmatrix} \dot{q} + \begin{bmatrix} 0 \\ -M^{-1}B \end{bmatrix} \begin{bmatrix} \dot{F}_x \\ \dot{F}_y \end{bmatrix}
\]

(10)

\[
y = C \cdot q
\]

(11)

Where A is the system matrix, B the input matrix and C the output matrix. The feedthrough matrix is not used and omitted in equation (11). The matrix A has a dimension of 40 x 40. Fig. 8 illustrates the integration of the machine model (control model) in the machine tool control. The control model is integrated in the feedback of the cascaded control structure of the feed drives. The actual position and velocity signals measured in the encoder and linear scales are fed into the Kalman Filter. The Kalman filter estimates the position and velocities of the table position (TCP) in X- and Y-direction and returns the estimated values into the closed loop of the cascaded control structure within the cycle time of 2 ms. The matrix A is updated in an extra task with a cycle time of 20 ms. Thereby the actual position of each slide is used. The position changes the center of gravity of each body. In consequence the stiffness matrix K is changed. The damping ratio is assumed to be constant. Via scaling and squaring method the matrix A is discretized.
The observer (Kalman Filter) is implemented on a Beckhoff TwinCAT 3 CNC, that is used as machine control. The Kalman-Filter is integrated in the control loop by a conversion of the created matlab simulink model into a C++ program that can be integrated in the Beckhoff development environment. The positioning controller as well as the Kalman-Filter runs at a cycle time of 2 ms.

5. Experimental Validation

To analyze the functionality and behavior of the Kalman-Filter on the machine control the system is subjected to a sinusoidal torque with an increasing frequency (sweep). The sweep is applied to the drive of the Y-axis as well as to the control model and increases 8 Hz/s starting from 60 to 300 Hz. To enhance the visibility of the dynamic behavior a further load mass of 150 kg is mounted on the X-slide. The velocity of the X-slide (body K) is measured with an external Laservibrometer in Y-direction. Fig. 9 shows the experimental setup. The machine control uses the motor encoder and linear scale which are far away from the X-slide. Via the proposed model based control algorithm the X-slide position and velocity is estimated. On the right hand side of Fig. 9 the velocity transfer functions at middle position of X-slide and at an eccentric position (X-slide = 250 mm off-center) is shown. In both cases the transmission characteristics of the encoder, which is used in the velocity control loop, shows nearly the same behavior. However, the X-slide leads to another interpretation. At the middle position a resonance frequency is shown at approximately 250 Hz. Via a further operational vibration analysis the resonance frequency can be identified mainly as a rotation of the X-slide around the X-axis. At the eccentric position a further resonance frequency at approximately 200 Hz appeared which is a rotation of the X-slide and less intensive rotation of the cross table around the Z-axis. By comparing the results of the vibrometer with the encoder, the rotation around the Z-axis is not visible. The linear scale recognizes the rotation but only with a small amplitude. The control integrated Kalman Filter based on the machine model described above is able to detect the oscillation in both X-slide positions.

6. Conclusion

This paper presents the implementation of a machine model into the machine tool control which is able to run in real time. By a position and load mass update the model is adjusted to the respective slide positions. Still existing deviations are mainly caused by the linear guides and carriages. Therefore, catalogue values are used and empirically adjusted. In the next steps the lateral and normal stiffness will be determined in experimental analysis.
References