

# Comparison of compensation strategies of structural effects

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**Abstract**— High dynamics of machine tool centers lead to high acceleration forces, which deform the machine structure and result in force coupling effects. In order to achieve high productivity combined with high path accuracy at existing machine tools, a numerical compensation of these dynamic effects is desirable. A model of an existing machine tool system allowed the evaluation of various compensation algorithms. The expected deviations of the uncompensated system can be gathered by simulation or measurement and further be used as set trajectory for the NC (offline compensation). Since this compensation values are subjected to the influence of the set point generation and the machine structure, only a marginal reduction of the deviations can be observed. Better results can be achieved by real time calculation of the compensation values based on actual velocity and jerk values (online compensation). Online compensation opens a broad range of possibilities for access points and compensation models. Good results on a model based evaluation of compensation strategies have been reached with compensation superposed directly on the torque feed forward input of the drives evaluated from set acceleration and jerk

## I. INTRODUCTION

The requirements of machine tools concerning their precision and productivity are rising continuously. Especially in precision and ultra precision manufacturing, high path accuracy and thus stiff machine structures are required. Due to material restrictions and high manufacturing cost, the possibilities to improve the machine's properties through structural modification are limited. Other influencing factors on high accuracy are the parameter settings of the numerical control, where limitations of acceleration and jerk will result in decreased machine dynamics and, ultimately, reduced productivity. According to [9] most of the commonly used numerical controls for machine tools offer a wide range of compensation options. Pitch-error, backlash- and friction-compensations are nowadays commonly used, as described in [5]. Also straightness, pitch-error, and deviations due to friction of a single axis relative to one other axis can be

realized according to [9]. Usually these compensations are implemented by using a chart that correlates the deviations in direction of one axis to the position of another axis.

Dynamic effects are usually not principal objects of compensation, despite the fact, that inertial forces dominate in high performant finishing processes. Especially hereby high machine dynamics and excellent surfaces are requested whereby a high contour accuracy at high acceleration and jerk levels is needed. In [10] is described, that for effective compensation, a detailed comprehension of sources for the dynamic effects for a particular machine structure and thus the occurring deviations are crucial. According to [11] and [12], the prediction of actual dynamic deviations for complex geometries and estimation of compensation effects, a model of the machine structure including control needs to be created.

## II. MEASURING DEVIATIONS

Measurements are required to comprehend machine behavior in greater detail, and to validate the modeled structure and resulting errors. According to [6] [7] and [8], machining tests included in ISO 230 [1] are time consuming and costly. Several static measurement methods like the 3D-ball plate, explained in [2], enable a precise analysis of straightness errors e.g. but only for static positioning. R-Test, described in [3], and double ball bar offer a wide range of possibilities for 3D data acquisition for linear and rotary axes. These two measurement devices were used to gather reference data. The R-Test is useful to evaluate deviations during simultaneous and interpolated movements of several axes and for a special class of motions. During the evaluation of the force coupling effects the double ball bar has been used to detect one dimensional deviations of the tool centre point (TCP) position. Additionally, dynamic effects have been captured by gathering position data directly from the measurement system of the machine itself. Most NC-Systems offer interfaces for such direct data logging.

### III. FORCE COUPLING EFFECTS

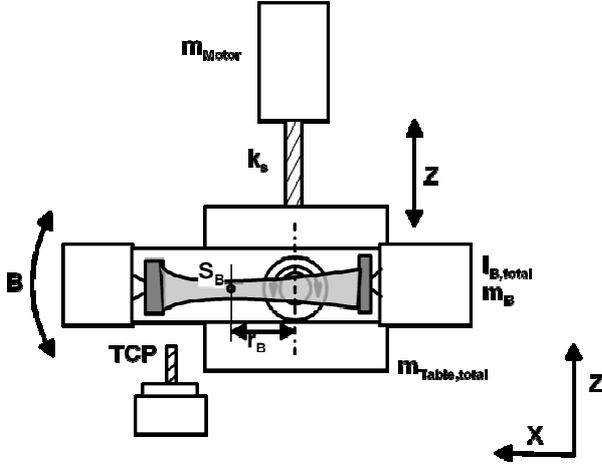


Figure 1. Axis configuration for a blade milling machine

A five-axis blade milling machine, showing significant force coupling effects has been used as base of the investigation and for the verification of the model. The most obvious effects appear between the rotational B-axis which is mounted on a linear Z-axis, shown in Fig. 1.

The eccentric mass distribution of the B-axis (due to A-axis mounting and drive) causes interactions between B-acceleration and Z-position and between Z-acceleration and B-angle position. The reacting forces can be computed by equations (1)-(3).

$$T_B = m_B r_B \cos(B) \ddot{Z} \quad (1)$$

$$F_{Z,\ddot{B}} = m_B r_B \cos(B) \ddot{B} \quad (2)$$

$$F_{Z,\dot{B}} = m_B r_B \sin(B) \dot{B}^2 \quad (3)$$

#### Nomenclature

$T_B$	Resulting torque on B-axis
$m_B$	Mass of B-axis
$r_B$	Offset of B-axis centre of gravity position relative to the rotation centre of B-axis
$B$	B-axis angle
$\dot{B}$	B-axis angular velocity
$\ddot{B}$	B-axis angular acceleration
$\ddot{Z}$	Z-axis acceleration
$F_{Z,\ddot{B}}$	Resulting force on Z-Axis due B- acceleration
$F_{Z,\dot{B}}$	Resulting force on Z-Axis due B- velocity
$I_B$	Inertia of B-Axis
$S_1/S_2$	Center of gravity of body1 respectively body 2
MS	Measurement system
$\Delta x$	Distance between TCP and the MS in X direction
$\Delta y$	Distance between TCP and the MS in Y direction
$k_s$	Stiffness of ball screw
$k_{XY}$	Translatory stiffness of guidance in X-/Y-direction
$k_{AB}$	Rotary stiffness of guidance in A-/B-direction
$x(t)$	State vector [24 × 1]
$x'(t)$	Derivative of state vector [24 × 1]
$u(t)$	Input signal [2 × 1]
$y(t)$	Output [10 × 1]
$A$	System matrix [24 × 24]
$B$	Input matrix [2 × 24]
$C$	Output matrix [10 × 24]
$D$	Direct input-output matrix [10 × 2]

$$I_B = m_B \cdot r^2 \quad (4)$$

$$F_{Z,\ddot{B}} = I_B \ddot{B} \quad (5)$$

$$EZ\ddot{B} = \frac{\Delta z \cdot F_{Z,\ddot{B}}}{\sum k_A + k_Y \Delta z^2} \Delta y + \frac{I_B \cdot \ddot{B}}{\sum k_B + k_X \Delta z^2} \Delta x \quad (6)$$

The occurring deviation of the TCP for the quasistatic case can be described in relation to occurring acceleration forces and structural stiffnesses. The resulting force in Z-direction due to acceleration of the B-axis can be calculated as shown in equation (4) & (5). The deviation of the TCP position in relation to the acting forces and the structural stiffness can be calculated as shown in equation (6). In case of indirect position control, the deformation of the ball screw drive has additionally to be taken into account. This deformation can be simplified as a drive stiffness  $k_{bs}$ . With an indirect measurement system, the offsets  $\Delta x$  and  $\Delta y$  have to be calculated between TCP and ball screw.

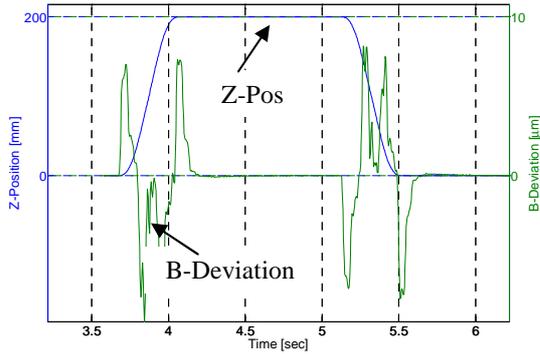


Figure 2. Z-Positioning, measured B-Deviation

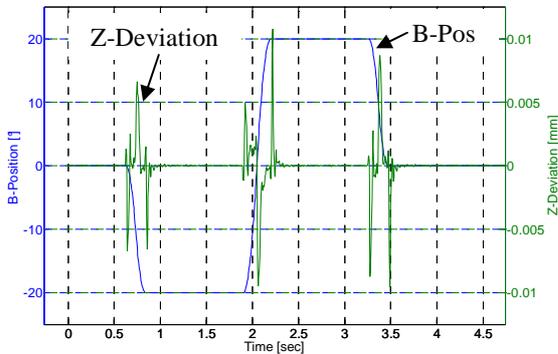


Figure 3. B-Positioning, measured Z-Deviation

A pure structural deformation would shift only the TCP position and thus it would not be mapped by the internal measurement system. The deviation of the TCP position due to structural deformation would further be strongly proportional to the acting force, which is not the case with force coupling effects. Sampling of the values of internal measurement systems of the machine demonstrate that the resulting deviations are measured and influenced by numerical control. A Z-trajectory of 200mm and back to the initial point at high dynamic settings show strong systematic deviations on the B-axis measurement system, illustrated in Fig. 2. B-axis accelerations are causing similar displacements detected by the Z-axis internal measurement system. Fig. 3 demonstrates the effects of a B-axis positioning movement of  $-20^\circ$ , followed by a positioning movement of  $+40^\circ$  and then  $-20^\circ$  back to initial position.

Extended measurements with probes exposed further deviations of the entire machine structure which are not visible for the measurement systems of the machine. These tilt movements of the whole moving structure cause characteristic errors at the TCP, which have to be included in the model describing the machine. Due to restrictions in the travel range of the probes, only B-axis movements have been performed, evaluated and used for calibration and verification of the model.

#### IV. MODELING

The effects can be investigated on a reduced system consisting of two rigid bodies, illustrated in Fig. 4 (rotary stiffness omitted in the drawing). The bodies possess six degrees of freedom and are coupled with a spring-damper system between each other. A model for the mass-spring-damper system was needed to represent all effects considered important, for description of machine behavior and thus for compensation. The rigid body model is represented by state space equations (7) and (8).

$$x'(t) = A \cdot x(t) + B \cdot u(t) \quad (7)$$

$$y(t) = C^T \cdot x(t) + D \cdot u(t) \quad (8)$$

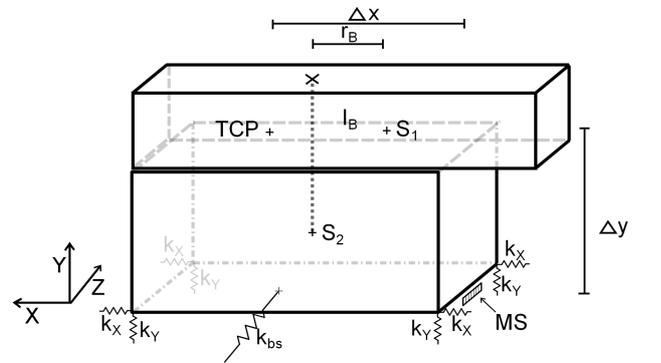


Figure 4. Reduced system

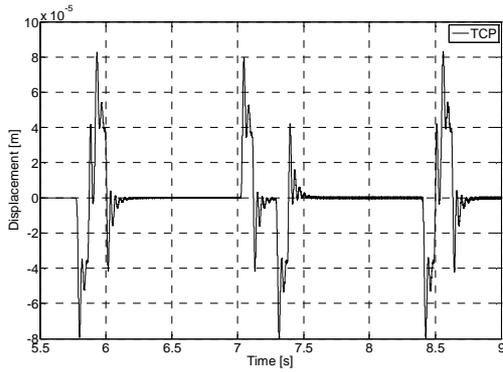


Figure 5. Modeled TCP displacement

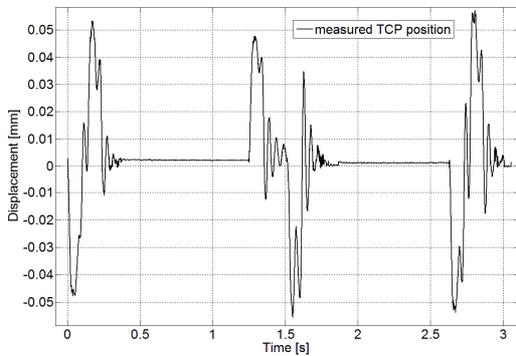


Figure 6. Measurement of TCP displacement

The state-space model consist of 2 bodies (12 DOFs  $\rightarrow$  24 states) with 2 input signals (force on Z-axis and torque on B-axis), 10 output signals (Z-displacement of upper and lower body, A-rotation of upper and lower body, B-rotation of upper and lower body, measurement system feed back of Z- and B-axis, TCP position and displacement of force application point of the Z-drive) and 24 state variables.

The modeled displacements of the TCP position, shown in Fig. 5, are qualitatively in good accordance with the measurements displayed in Fig. 6. There are still moderate quantitative differences. Since actual parameters like stiffness and especially friction are difficult to estimate and are subject to changes due to external influences like e.g. temperature variation or wear, it was impossible to reach more accurate results.

## V. COMPENSATION STRATEGIES

The TCP path generally does not match the set path exactly but is the reproduction of the set path by the machine structure influenced by the NC. According to [13] the parameter where the compensation is applied on and also the base values for the calculation of the compensation have significant influence on the effectiveness of the compensation algorithm. It has to be kept in mind, that commercial numerical control systems use cascaded PID control only, and also offer only very limited access to influence or manipulate the set point generation. Well known decoupling techniques can not be applied here.

A demonstrative compensation application is described in [4], where the measured deviations of the TCP position have been used as correction values on the position set path. This measuring or simulation of the expected deviation can not be performed during manufacturing process and will therefore be called offline compensation. Any trajectory needs to be simulated or followed on the machine first to gather the required values. This may be suitable for serial production, but not for flexible manufacturing. Offline compensation is applied on the set trajectory before the NC-Code is transferred to the numerical control.

In contrast to the described offline compensation, an online compensation strategy will get its parameter directly from the actual values of the numerical control during manufacturing process. From these input values, a real time application derives the actual compensation values and writes them directly on the current position set point or feed forward value. Thus the NC needs to be accessible for reading and writing commands. Online compensation is time critical as the required values need to be read and processed before the compensation parameters are available. It would be obvious to use the look ahead values of the NC command to calculate the values for compensation. But available NC interfaces do not offer access to the look ahead values, only actual values can be readout and altered. The compensation can be applied either on the set path or at any other accessible point of the motion control like feed forward, torque control or current values. In this study, the compensation has been applied on the set path and the torque feed forward control as shown in Fig. 7.

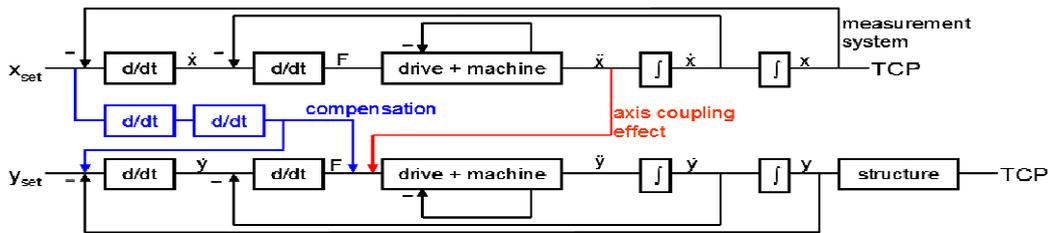


Figure 7. Schematic of compensation appliance

Fig. 8 shows the simulated TCP displacement under application of the described offline compensation, where a reduction of the initial deviation (shown in Fig. 5) of about 20% can be reached. Under the assumption, that the model represents also the frequency response accurately, a broader frequency band will be excited under application of offline compensation. Applying set point filtering could probably improve this result due to reduction of the bandwidth. For all the model based evaluations of different compensation algorithms the same B-axis positioning movements as explained before have been used as model input. The input values of the model have been collected directly from the NC set point generation.

For all following compensations the actual set values of acceleration have been used because they are available earlier than the actual values captured by the measurement system. Since the difference between the magnitude of set value and actual value is comparatively small, earlier availability has been considered as more important than accuracy in magnitude. The acceleration values will be multiplied with an optimized gain and superposed directly on the set point value of the compensated axis. The set position input after compensation can be seen in Fig. 9. The resulting simulation of the TCP position in Fig. 10 shows a reduction of the initial, uncompensated deviations of about 70% and less excitation of the machine structure compared to the offline compensation.

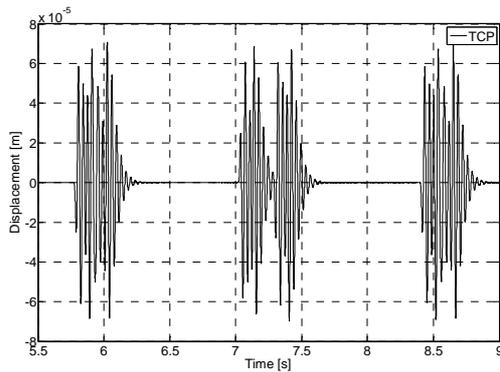


Figure 8. Simulation of TCP displacements with offline compensation

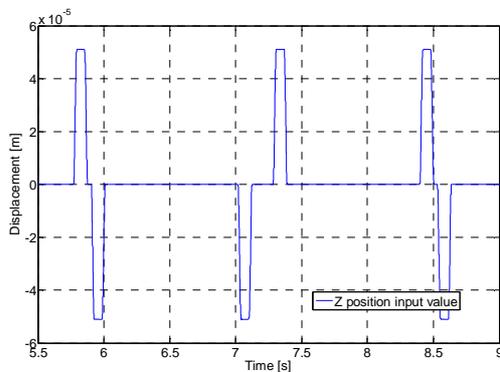


Figure 9. Set point input based on set acceleration

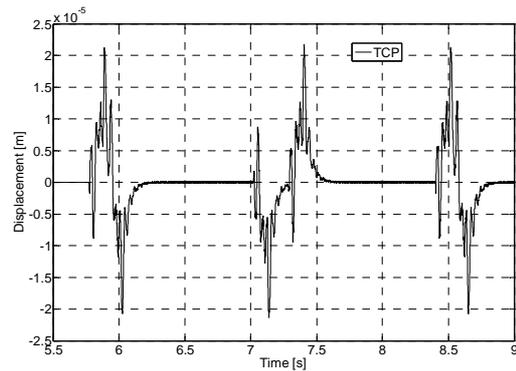


Figure 10. Simulation of TCP displacements with compensation based on set acceleration

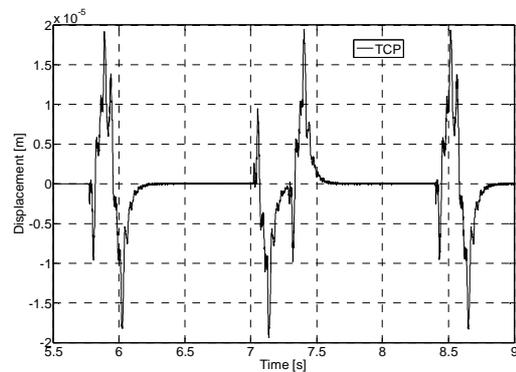


Figure 11. Simulation of TCP displacements with compensation based on set acceleration and jerk

The derivative of the set acceleration values will allow using jerk level as further input for the evaluation of the compensation value. As already mentioned, the force coupling effects are dependent of the acting forces due to acceleration. Since the occurring deviations will be compensated by the position control as soon as the acceleration remains constant, the periods of rising or decreasing acceleration are crucial for dynamic compensation. To reach an over proportional compensation of the acceleration level during increasing and decreasing acceleration, the jerk can either be added to the acceleration or be multiplied with the actual acceleration level for compensation input. The evaluation of the model, plotted in Fig. 11, shows resulting TCP deviations of less than  $20\mu\text{m}$ , which would be a reduction of about 75% compared to the initial, uncompensated deviations under application of a compensation level calculated from actual jerk added to the acceleration level.

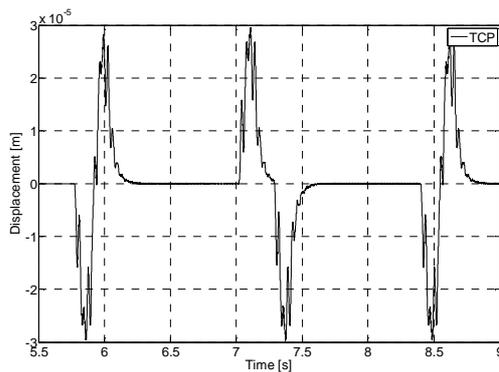


Figure 12. Simulation of TCP displacements with torque feed forward compensation

The position value of a given trajectory is exposed to severe influence of the NC such as interpolation, filtering and limitations. Thus it would be desired to apply any online compensation at the latest accessible point of data processing, which would be the torque feed forward input of the drives. Compensation based on acceleration and jerk led to a modeled reduction of the TCP deviations in Z-direction of about 60% (Fig. 12). Although the modeled compensation based on acceleration and jerk values applied on the set path showed best results, the compensation applied on the torque feed forward control is less affected by the set geometry and controller settings than the position set point value.

## VI. SUMMARY AND OUTLOOK

Adequate modeling of dynamic deviations of machine tool structures need to be based on precise measurements either from the internal reading of the Numerical Control and/or external measurements. For effectiveness of dynamic compensation, the compensation strategy is crucial. Measured or simulated TCP deviation offers a possibility to compensate dynamic effects offline without deeper knowledge of the interfaces for accessing the NC. But offline compensation needs in advance execution or modeling of the desired trajectory to gather occurring deviation values. Online compensation opens a broad range of possibilities for access points and compensation models. Best result on a model based evaluation of compensation strategies has been reached with compensation superposed on the position set point evaluated from set acceleration and jerk. Due to less influence from filtering etc. a compensation appliance directly on the torque feed forward input of the drives would be desired. A test bench consisting of a linear axis combined with two rotary axes will serve for validation of these results and prove the effectiveness of dynamic compensation.

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