Compensation strategies for axis coupling effects

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Abstract

High dynamics of machine tool centers lead to high acceleration forces and, thus, may 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. On model based evaluation, compensation strategy has turned out to be crucial for effectiveness of dynamic compensation.

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1. 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 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.

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. Also straightness, pitch-error, friction and sag adjustments of a single axis relative to one other axis can be realized. 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. Especially in finishing processes high machine dynamics and excellent surfaces are requested whereby a high contour accuracy at high acceleration and jerk levels is needed. For an effective compensation, a detailed comprehension of sources for the dynamic effects for a particular machine structure and thus the occurring deviations is crucial. To predict the occurring dynamic deviations for complex geometries and to estimate the effects of compensation, a model of the machine structure including control needs to be created.

1.1. Measuring Deviations

Measurements are required to comprehend machine behavior in greater detail, and to validate the modeled structure and resulting errors. Machining tests included in ISO 230 [1] are time consuming and costly.

Several static measurement methods like 3D-ball plate [2] enable a precise analysis of straightness error but only for static positioning.

R-Test [3] and double ball bar offer a wide range of possibilities for 3D data acquisition for linear and rotary axes. These two measurement devices where used to gather reference data. The R-Test is useful to evaluate deviations during simultaneous axe movements and short travel movements. During the evaluation of the force coupling effects (see 2. Force Coupling Effects) the double ball bar has been used to detect one dimensional deviations of the tool centre point (TCP) position.
Alternatively, dynamic effects can be captured by gathering position data directly from the measurement system of the machine itself. Most NC-Systems offer interfaces for direct data logging.

2. Force Coupling Effects

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 located on a linear Z-Axis (see Fig 1).

\[
M_t = m_B r_B \cos(B) \ddot{Z} \quad (1)
\]

\[
F_{Z,B} = m_B r_B \cos(B) \dot{B} \quad (2)
\]

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

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 described by equations (1)-(3).

Capture of the values of internal measurement systems of the machine demonstrate that the resulting deviations are measured and influenced by numeric control. A Z-trajectory of 200mm and back to the initial point at high dynamic settings show strongly systematic deviations on the B-Axis measurement system (Fig 2).

B-Axis accelerations are causing similar displacements captured by the Z-Axis internal measurement system. Figure 3 demonstrates the effects of a B-Axis positioning movement of -20°, followed by a positioning movement of +40° and then -20° back to initial position.

### Nomenclature

- \(M_t\) Resulting torque on B-Axis
- \(m_B\) Mass of B-Axis
- \(r_B\) B-Axis centre of gravity position relative to the rotation centre of B-Axis
- \(B\) B-Axis position
- \(\dot{B}\) B-Axis velocity
- \(\ddot{B}\) B-Axis acceleration
- \(\dot{Z}\) Z-Axis acceleration
- \(F_{Z,B}\) Resulting force on Z-Axis due B-acceleration
- \(F_{Z,B}\) Resulting force on Z-Axis due B-velocity
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 structure cause characteristic errors, which have to be included in the model describing the machine. Due to restrictions in travel range of the probes for the extended measurements, only B-Axis movements have been performed, evaluated and used for calibration and verification of the model.

3. Modeling

A first modeling implemented in Matlab/Simulink based on the data of the internal measurement system, showed good accord with the internal NC-readings of various trajectories, but was not able to reproduce the effects measured with additional measurement setup directly on the machine structure. A more complex model for the 2- (B-Axis) + 3- (Z-Axis) mass spring damper system was needed to represent all effects considered important, for comprehension of machine behavior and thus for compensation.

The actual rigid body model consists of a state space model following the equations (4) and (5).

\[ x'(t) = A \cdot x(t) + b \cdot u(t) \]  
\[ y(t) = c^T \cdot x(t) + d \cdot u(t) \]

### Nomenclature

- \( x(t) \): State vector
- \( x'(t) \): Derivative of state vector
- \( u(t) \): Input signal
- \( y(t) \): Output
- \( A \): System matrix
- \( b \): Input matrix
- \( c \): Output matrix
- \( d \): Direct input-output matrix

The state-space model consist of 2 bodies 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 displacement of Z- and B-Axis, TCP position and displacement of force transmission point of the Z-Axis) and 24 states.

The modeled displacement of the TCP position, pictured in Fig 4, shows qualitatively good accordance with the measurements displayed in Fig 5. There are still moderate quantitative differences. Since parameters like stiffness and especially friction are difficult to estimate and are subject to changes due to temperature variation and wear it was not possible so far to reach more accurate results.

![Fig 4 modeled TCP displacement](image)

![Fig 5 measurement of TCP displacement](image)

4. Compensation strategies

The TCP path does not follow exactly the values of the set path. It is the reproduction of the set path through the machine structure influenced by the NC. So it is not obvious what kind of compensation will suit best for the case.

Either an offline or an online compensation can be performed. Offline means that the compensation will be based on measured or simulated values of the desired trajectory. Thus the trajectory needs to be simulated or followed on the machine first to gather the required
values [4]. This may be suitable for serial production, but not for flexible manufacturing. Offline compensation shall be applied on the set trajectory before the NC-Code is transferred to the numerical control.

An online compensation strategy will get its parameter directly from the actual values of the numerical control. 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 may be time critical as the required values need to be read and processed before the compensation parameters are available. The compensation can be applied either on the set path or at any other accessible point of the motion control. In this study, the compensation has been applied on the set path and the torque feed forward control (Fig 6).

Further there are several interfaces where compensation can be inserted. The most apparent way to compensate offline is to get the accruing deviations for a certain trajectory by measurement or simulation and superposition of the inverted deviation directly on the set path. A simulation of this compensation strategy shows (Fig 7) that a reduction of the initial deviation of about 20% can be reached (Fig 4). But following the model, a broader frequency band will be excited. Applying set point filtering could probably improve this result due to reduction of the bandwidth. For all the evaluations of compensation strategies the same B-Axis positioning movements as explained before have been used. The input values of the model have been collected directly from the NC set point generation during the measurement.

Since the deviations show strong dependence on the acceleration (see 2. Force Coupling Effects) it appears to be reasonable to use actual acceleration values in order to evaluate the compensation values. For all the following compensations the set values of acceleration will be used because they will change at an earlier stage 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. The acceleration values will be multiplied with an optimized gain and superposed directly on the set point value (Fig 8). The resulting simulation of the TCP position (Fig 9) shows a reduction of the deviations of about 70% of the initial deviations and less excitation of the machine structure compared to the offline compensation.
Since all measured dynamic deviations are due to occurring acceleration forces the variation of the acceleration and jerk value, will also have marginal influence on the machine behavior. The derivation of the set acceleration values will allow using jerk level as further input for the evaluation of the compensation value, which will still be superposed on the Z-position set point (Fig 10). The evaluation of the model shows resulting deviations of less than 20µm, which would be a reduction of about 75% of the initial deviations (Fig 11).

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

5. 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 or external measurements.

For effectiveness of dynamic compensation, compensation strategy is crucial. Offline compensation offers a possibility to compensate dynamic effects without deeper knowledge of accessing NC parameters. 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 (as deviation of acceleration). 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 effectiveness of dynamic compensation.
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