Measurement and simulation of acceleration correlated position errors in machine tools

M. H. Nguyen, S. Weikert, K. Wegener
Institute of machine tools and manufacturing (IWF), Department of mechanical and process engineering, Swiss Federal Institute of Technology (ETH), Switzerland

Abstract

Machine tools have to reach high levels in accuracy and speed. For the detailed analysis of the machine dynamics in respect to these properties, this paper presents a measurement method and a simulation model to analyse dynamic effects, especially on acceleration level. The proposed measurement procedure consists of measuring circular movements of the TCP (tool centre point) and comparing these movements with measurements at the machine axes. Due to this combination, structural deformations of the machine tool can be derived and correlated to distinct acceleration states of the machine axes. The evaluation of this measurement procedure directly provides values of acceleration correlated position errors. Furthermore, the simulation proves that these systematic position and straightness errors are valid for arbitrary movements, which shows the potential the measurement method. The simulation model includes the main concepts of multi-body description and feedback control systems. A control system model being coupled with the structural model permits the simulation of interactions between machine tool structure and its spatial effects at the TCP.

1 Introduction

The offset between the centre of gravity and drive force input location causes a momentum acting on the machine tool structure also stated in [1] and [2]. Figure 1 schematically shows the drive force inducing a momentum, which leads to tilt motions and causes straightness deviations (EXZ e.g.) linked to inertial cross-talk and positioning deviations (EZZ e.g.) linked to in-talk. The corresponding equations are given by (1) and (2). Due to process conditions of machine tools at the TCP and therefore missing measurements at this point, dynamic displacements at the TCP can’t directly be compensated by common drive closed loop control systems. This paper proposes a measurement method and a simulation model to qualify and to quantify the mentioned effects.

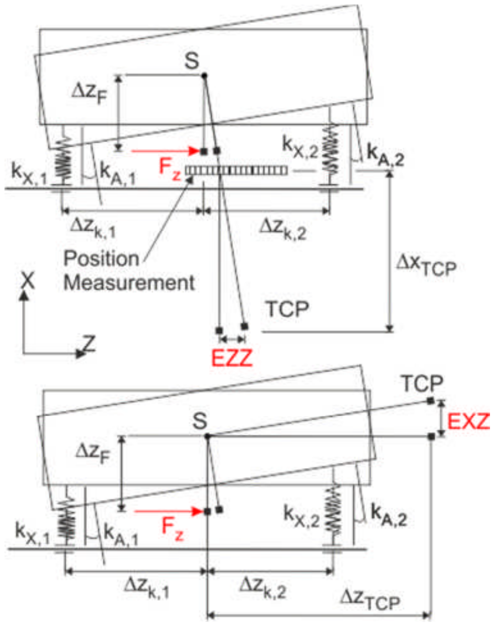

Figure 1: Inertial effects; top: in-talk (EZZ), bottom cross-talk (EXZ)

(1)

(2)

2 Measurement method

This section describes the measurement method following [3] and is briefly shown next.

2.1 Measurement motion

The sometimes step-wise like dynamic input due to drive forces leads to an excitation of the machine. To distinguish the cause of dynamic effects due to jerk and acceleration, circular motions provide low frequencies of excitation. For example, a circular motion with a radius of 2mm and an acceleration of $5m/s^2$ yields 8Hz as excitation frequency. The measurement of small circles also allows small measurement ranges which provide moderate measurement uncertainty. A further advantage of circles is the simplified synchronization of internal (p.m. direct measurement of the axes) and external measurements (at the TCP) due normalisation at each circle point which is shown in equations (5) and

(6) in section 2.2. In this way, internal measured states, like acceleration, can directly be allocated to external measured displacements at the TCP.
The circular motions are carried out by interpolating movements of the linear axes. To get complete information about the axes, the circles are performed in the three perpendicular planes of machine coordinate system, clockwise and counter-clockwise and with different parameter settings, set acceleration or feed rate.

2.2 Measurement system

The difference between direct measurement systems of the axes and the external measurement (at the TCP) describes the structural errors in the ZX-plane as position errors EZZ and EXX as shown in figure 2.

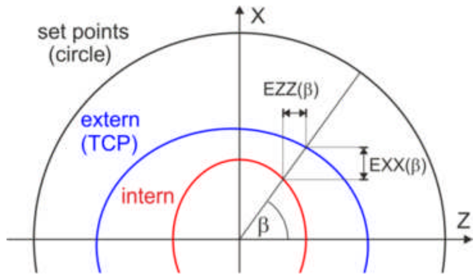

Figure 2: Definition of position errors EXX and EZZ in a circular motion

The R-Test [4] is a measurement setup, where the external measurement at the TCP carried out using a precision sphere and three incremental probes perpendicularly mounted to each other (see figure 3).

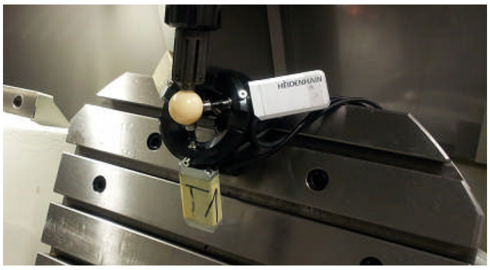

Figure 3: R-Test setup

The spatial measurement range is ±3mm, the measurement uncertainty of the R-Test measurement setup is U(k = 2) = 2.8µm. The internal measurement is provided by closed-loop position measurement of the machine tool.

2.3 Evaluation Method

The evaluation is exemplarily shown next for a circle in the ZX-plane. The set points (z_{set},x_{set}) and the set acceleration ($\ddot{z}_{set}$,$\ddot{x}_{set}$) of circular motion are given by:

$$z_{set} = r\,cos(\omega t) \Rightarrow \ddot{z}_{set} = -r\omega^2\,cos(\omega t) = -a_{set}cos(\omega t) \quad (3)$$
$$x_{set} = r\,sin(\omega t) \Rightarrow \ddot{x}_{set} = -r\omega^2\,sin(\omega t) = -a_{set}\,sin(\omega t) \quad (4)$$

The set points (z_{set},x_{set}) can also be obtained by normalising the internal measured values (z_{int},x_{int}):

$$z_{set} = r\frac{z_{int}}{\sqrt{{z_{int}}^2+{x_{int}}^2}} \Rightarrow \ddot{z}_{set} \quad (5)$$
$$x_{set} = r\frac{x_{int}}{\sqrt{{z_{int}}^2+{x_{int}}^2}} \Rightarrow \ddot{x}_{set} \quad (6)$$

with r: radius, ω: angular velocity and a_{set}: set acceleration. The actual points (z_{act}, x_{act}, y_{act}) of the internal and external measurements are assumed as:

$$z_{act} = z_{set} + EZ\ddot{Z}\ddot{z}_{set} + EZ\ddot{X}\ddot{x}_{set} \quad (7)$$
$$x_{act} = x_{set} + EX\ddot{X}\ddot{x}_{set} + EX\ddot{Z}\ddot{z}_{set} \quad (8)$$
$$y_{act} = EY\ddot{Z}\ddot{z}_{set} + EY\ddot{X}\ddot{x}_{set} \quad (9)$$

with:

- $EZ\ddot{Z}$, $EX\ddot{X}$: In-talk parameter of Z- and X-axis
- $EX\ddot{Z}$, $EY\ddot{Z}$, $EZ\ddot{X}$, $EY\ddot{X}$: Cross-talk parameter of Z- and X-axis
- The unit these parameters $EI\ddot{J}$ is [µm/m/s^2] resp. [µm s^2/m]

Assuming small displacements of the acting axes of the machine tool, equations (7) – (9) denote the functions to fit the internal and external values. The cross-talk parameters are calculated with “least square plane fit”. The measurement of a single plane delivers two cross-talk parameters: for circle in the ZX-plane the values for $EY\ddot{Z}$ and $EY\ddot{X}$ are obtained, see equation (9). To get the complete set of cross-talk parameters, the circles have to be measured in the three perpendicular main planes of the machine coordinate system.
The in-talk parameters are calculated using “least square ellipse fit”. Since the in-talk cause even ellipses and the cross-talk cause tilted ellipses, the actual points have to be adjusted for the even ellipse fit. With the help of these parameters, external measured displacements can be allocated to internal measured states, in this paper the acceleration state. The fitting of the internal and external measured displacements values presented in the following section is based on the internally measured acceleration.

2.4 Measurements

As presented in [3], the evaluation shows higher dispersion of parameters at low accelerations for the circles, which were carried out with low dynamics. With low dynamics, the errors due to static and geometric effects are of the same magnitude and so influence the acceleration correlated position errors. Lower dispersion can be obtained with higher acceleration. The measurements shown next illustrate exemplarily the evaluation procedure and the fitting results.

2.4.1 Cross-Talk

Figure 4 shows the evaluation of the cross-talk parameter $EX\ddot{Z}$ of a ZX-circle based on the planar inclination of the circle in YZ-plane. The reached acceleration was 4.5 m/s². The evaluation therefore yields to $EX\ddot{Z} = -1.55\ \mu m\ s^2/m$.

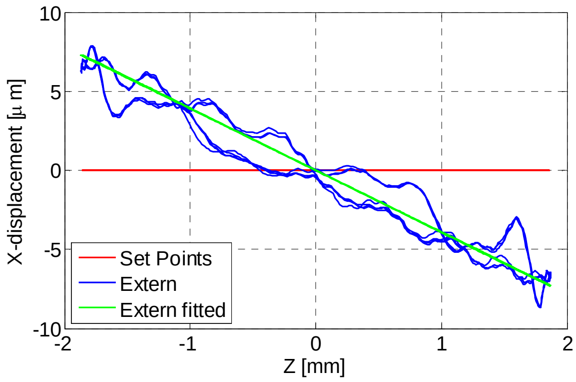

Figure 4: View on ZX-Plane of set points and external measurement of circular YZ-Motion

2.4.2 In-talk and Cross-talk

The evaluation of the in-talk parameters $EX\ddot{X}$and $EZ\ddot{Z}$were carried out after the evaluation of the cross-talk parameters, which are causing the tilting in the ellipse of the measured ZX-circle. Figure 5 shows the fitting effects of in-talk and cross-talk combination. The assumption of (7) and (8) is visible and distinct for higher accelerations, where the dynamic displacements are higher in amplitude.

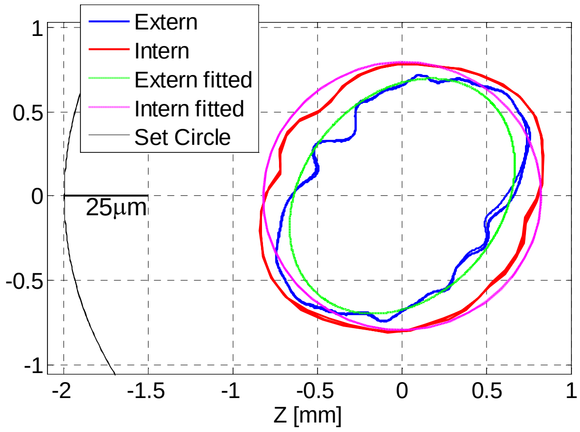

Figure 5: Magnified external and internal measurements and fitted ellipses in reference to the set circle (r = 2mm)

3 Simulation Model

The measurements presented in the previous sections premise transient simulations. A control system model being coupled to a structural model permits the simulation of interactions between machine tool and the spatial effects at the TCP in the time domain. The use of multi-body description for the machine structure provides fast computations and analyses, especially to achieve the effects of drive input forces on the machine structure and though the dynamic displacements at the TCP. This section describes the simulation model following [5] and is briefly shown next.

3.1 Mechanical Model

The mechanical structure of a machine tool is commonly modelled with FE-methods and with formulations of multi-body dynamics (MB). Assuming that the most compliant stiffness in a machine tool are given by the guidance and bearing between the machine axes, the MB approach can reproduce dynamic behaviour in low frequencies qualitatively and quantitatively, which is stated in [6]. Further advantages are short computation time and direct control interaction. For the simulation of cross-and in-talk effects due to tilt motions, a body i of the MB-system requires for 3 degrees of freedom (DOFs) for linear and rotary directions, which yields to 6 DOFs:

$$\boldsymbol{q_i} = (x, y, z, a, b, c) \tag{10}$$

The complete set of DOF of a MB-system are defined by all bodies n with

$$\boldsymbol{q} = (\boldsymbol{q_1}, \boldsymbol{q_2}, \dots, \boldsymbol{q_{n-1}}, \boldsymbol{q_n})^{\boldsymbol{T}} \tag{11}$$

The mechanical coupling of the bodies is provided by position-dependent mass-, damping- and spring-matrices , and with the equation of motion:

(12)

The efficient construction of the system matrices is explained in [5].

3.2 Control System

Based on (9) and (10), the state space can be described as

(13)
(14)

With the input-matrix external forces as gravitational and drive-forces are added on the corresponding states. The output-matrix delivers output values for p.m. the measurement systems of a control system e.g.. The state space system delivers:

(15)
(16)

In this paper, the closed loop control system is a cascaded PPI controller as shown in figure 6**.**

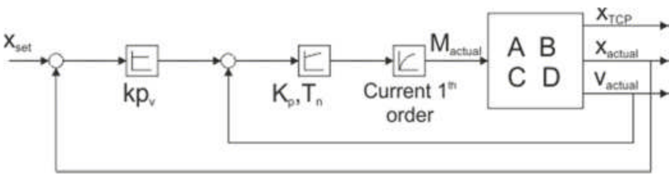

Figure 6: Cascaded PPI controller

The PPI controller contains a closed loop for velocity measurement and a superposed position closed loop. This control concept is commonly applied in machine tools. The parameters of the PPI controller can directly be derived based on the physical parameters of the drives. The analytical derivation of these first assumptions of theses control parameters is described in [6].

3.3 Simulation of the measurement motion

Based on the modelling presented in section 3.2 and 3.3, this section shows the simulation of the measurement movements and the analogy evaluation procedure as described in section 2. The physical parameters as mass values for the bodies or stiffness for guideway are taken from manufacturer's data, so there was no parameter identification done for the following simulations.

3.3.1 Cross-talk

Figure 4 shows the evaluations of the cross-talk parameter $EX\ddot{Z}$ of a ZX-circle out of the planar inclination of a circle in YZ-plane. The reached acceleration was 4.7 m/s². The evaluation yields to $EX\ddot{Z} \ = -0.55 \ \mu m \ s^2/m$.

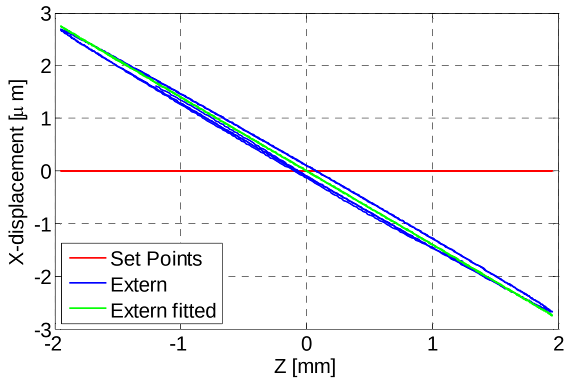

Figure 7: View on ZX-Plane of internal and external simulation of circular YZ-Motion

3.3.2 In-talk

Figure 8 shows the fitting effects of in-talk and cross-talk combination. The assumption of (7) and (8) is noticeable.

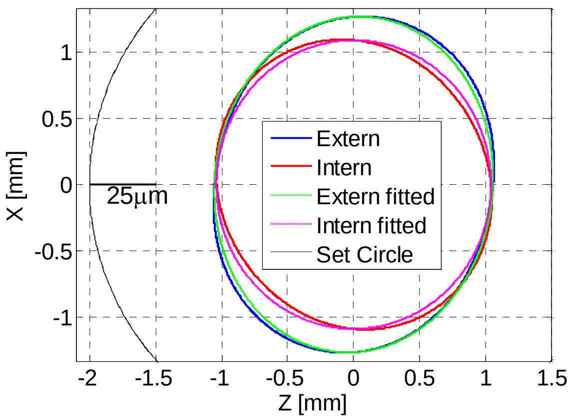

Figure 8: Magnified external and internal measurements and fitted ellipses in reference to the set circle (r = 2mm)

3.4 Simulation of larger movement

Figure 9 illustrates a positioning movement over 20mm with a set velocity of 30'000 mm/min, a set acceleration, 10 m/s^2 and a set jerk 250 m/s^3. Due to the jerk and velocity settings, the set acceleration could not be reached. The first order behaviour of the mechanical system leads to a contouring error which can be seen by the difference between set positioning and actual positioning. The proportional dependency of the cross-talk at the TCP in X-direction in dependency of the actual acceleration in Z-direction is visible. The cross-talk displacement at the maximal reached actual acceleration of 7.9 m/s^2 is 3.7 µm which yields $EX\ddot{Z} \;=\; -0.48\ \mu m\ s^2/m$ at these points.

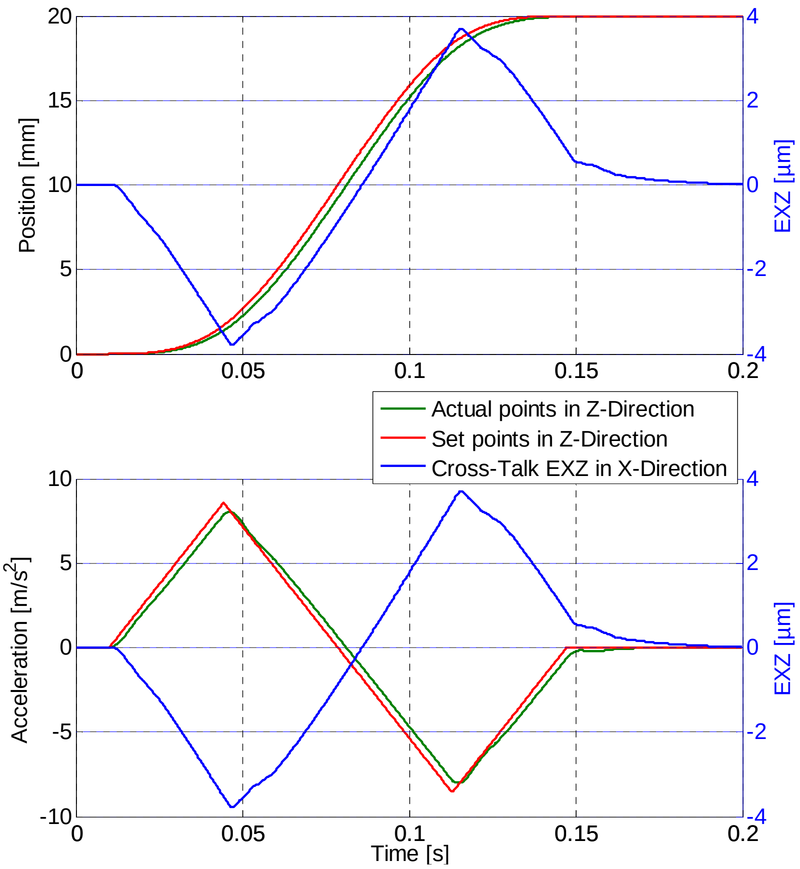

Figure 9: Positioning movement of 20mm in Z direction. Position and acceleration set points compared with actual values and resulting cross-talk deviation in X direction at the TCP

4 Conclusion

The systematic behaviour of cross-talk and in-talk due to the acceleration state of a machine tool axis is distinct in measurements and simulations. Since no parameter identification has been done in this paper, the quantitative values of the acceleration dependent position errors of simulations and measurements differ about the factor of 2 to 3. The cause for this difference is the absence of any parameter identification for the MB-formulation of the machine tool structure. Taking into account structural elasticity by applying appropriate reductions factors for the catalogue data of the guideway stiffnesses as explained in [7] and [8], quite good accordance with measurement data can be obtained. A further model approach is the integration of reduced FE-models in MB-dynamics.

The proportionality values $EI\ddot{J}$ obtained by the evaluation method of the circles are quite similar to the ratio of acceleration and straightness deviations during a large dynamic positioning movement. So, systematic machine tool behaviour can be assessed even is only a small measurement range is available which is the case for the R-Test. Future work is the use of the obtained acceleration correlated position errors for analysis and compensation.

References

1. S. Weikert, S. Bossoni, K. Wegener. "Evaluation of Machine Tool Concepts Under Friction Influences", ASPE Technical Symposium Proceedings, Atlanta, 2007
2. B. Bringmann, P. Maglie, "A method for direct evaluation of the dynamic 3D path accuracy of NC machine tools". CIRP Annals – Manufacturing Technology, Vol. 58, pp. 343-346, 2009.
3. M. H. Nguyen, S. Weikert, K. Wegener "Evaluation method for direct evaluations of acceleration correlated position errors in machine tools", ASPE 2012 Proceedings, 506 – 509
4. S. Weikert, W. Knapp, "R-Test, A New Device for Accuracy Measurements on Five Axis Machine Tools". CIRP Annals – Manufacturing Technology, Vol. 48, pp. 433-436, 2009.
5. M. H. Nguyen, S. Weikert, K. Wegener "Toolbox for control system analysis of machine tools", Mechatronics 2012 Proceedings, Vol. 1/3
6. O. Zirn, S. Weikert, „Modellbildung und Simulation hochdynamischer Fertigungssysteme," Springer Verlag, 2006
7. D. Kono, S. Weikert, A. Matsubara ,K. Yamazaki. "Estimation of Dynamic Mechanical Error for Evaluation of Machine Tool Structures". Int. J. of Automation Technology, Vol. 6, N. 2, pp. 147–153, 2012.
8. P. Maglie, "Virtual Machine Tools: from Simulation-driven Structure Design to Practice-oriented Software Development", Diss. ETH No. 19801, 2011