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Analysis of an ultra-precision positioning system and parametrization of its structural model for error compensation

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Abstract

Conventional compensation of position errors of machine tools relies only on measured values. Due to this principle it is not always possible to compensate the errors in time, especially dynamic ones. Moreover, the relevant control variables cannot always be measured directly. Thus, this approach proves to be insufficient for high precision applications. In this context, a model-based error prediction allows for minimal position errors. However, ultra-precision applications set high demands for the models' accuracy. This paper presents the design of an accurate and real time-capable structural model of an ultra-precision positioning system. The modeling method for the developed ultra-precision demonstrator is shown and the initial parameter identification is presented.

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Keywords: Structural modelling; parametrization; model-based compensation; ultra-precision

1. Introduction

Ultra-precision machining is considered as a key technology for the manufacturing of reflective optical components with high precision complex surfaces. Presently the performance of ultra-precision machining is still limited by small feed rates and small spindle rotation speeds compared with conventional machining [1]. This is due to the high precision and stability requirements on the process.

In order to improve the performance of ultra-precision machining without affecting the demanded precision in the nanometer range new machine and control concepts have to be developed. In particular, errors caused by increased dynamics such as unbalances of rotating components must be compensated reliably.

In this context conventional compensation of errors relying only on measured values proofs to be insufficient to maintain the precision in the nanometer range. Due to accessibility problems it is not always possible to place sensors at the positions of interest like the tool center point (TCP) during machining. Therefore a model of the machine is needed to predict errors at such positions. In the last years much research work has been conducted in the model-based compensation of dynamic errors for conventional systems [2-3]. For ultra-precision systems model-based methods for the compensation of geometric errors of precision and ultra-precision machine tools have been proposed in literature [4]. However predictive compensation of errors due to the dynamic compliance of ultra-precision machine tools has not been investigated so far. For this an accurate and real-time capable structural model of the ultra-precision system has to be built.

This paper begins with a description of an ultra-precision two axis workpiece table. Afterwards the system's model and the experimental analysis results are presented. Finally the offline adjustment of the model based on measurement data is shown. The parametrized model aims to compensate errors due to increased dynamics.

2. Ultra-precision demonstrator

The ultra-precision demonstrator shown in Figure 1 is a two-axis positioning system consisting of a linear axis with aerostatic bearings (Z-axis) and a novel linear axis with

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electromagnetic bearings (X-axis) [5]. The electromagnetic guide provides the capability to compensate errors in five degrees of freedom in the range of ± 25µm. Furthermore it allows for damping of dynamic disturbance forces actively. On the contrary, the aerostatic guide is not actively controlled and consists of nine flat rectangular air bearing pads which are made up of porous media. The air gap is about 5µm for an input pressure of 4 bar.

Most of the system's components are made up of granite because of its good damping and thermal properties. In order to avoid tilting moments a box-in-box concept has been chosen. The cross table is mounted onto a support plate made up of granite which is installed on a steel frame. The TCP is located on the top of the electromagnetic slide. The travel of both axes is 100 mm.

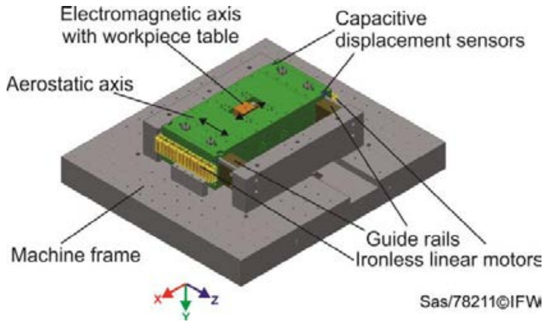

Figure 1: Ultra-precision demonstrator

Both axes are driven by two ironless linear motors in gantry configuration. The position feedback in the direction of motion is provided by linear encoders with a resolution of 1 nm. The deviations in the other two translational directions and the three angular deviations for the aerostatic axis are recorded by five capacitive sensors. For the electromagnetic axis twelve capacitive sensors are used. The installed displacement sensors allow for the online adjustment of the model by the use of an observer (kalman filter).

3. Model of the ultra-precision axis

The model of the positioning system is in state space representation (1). Here A is the system matrix, B the input matrix, C the output matrix, D the feedforward matrix, x the state vector and y the output vector.

$$\dot{x} = Ax + B\,u \qquad (1)$$
$$y = Cx + Du$$

The model predicts the structural deviations at the tool center point based on the actuating forces as input. First the initial model's parameters have to be adjusted. The offline parametrized model calculates the deviations at the positions of the 17 capacitive sensors by the use of the output matrix C_{mes}. The determined values are compared with the measured values at the same positions. The discrepancy is used by the model-based kalman filter in order to adjust the model online. The adjusted model calculates the deviations at the TCP by the use of the second output matrix C_{TCP}. The measured geometric errors at the TCP are stored in a look up table and added to the calculated values. Figure 2 shows the principle of the model based observer. The resulting error at the TCP is then assigned to the appropriate system's actuators according to their direction, amplitude and frequency.

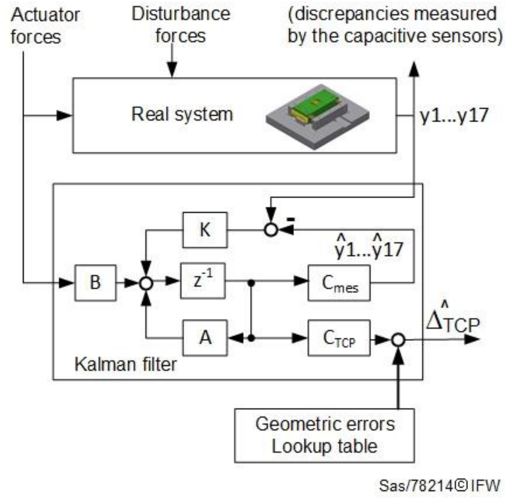

Figure 2: Model based Observer

A structural model of the 2-axis positioning system is built. The electromagnetic axis is not yet ready for operation regarding its control system. Thus, in this paper, the positioning system is modeled and analyzed assuming the magnetic guide to be rigid. Therefore the slide has been fixed.

3.1. Modeling method

In [6] two approaches for modeling the ultra-precision positioning system are presented and discussed. A top down method based on the detailed FE model has been chosen. A model order reduction is applied on the detailed structural model based on the modal superposition method. The reduction method describes the structure's response in terms of n eigenmodes. Here n is the number of the modes to be extracted. The result is a state space model with the system matrix A expressed as follows, where ω_i is the angular frequency of mode i, d_i is the effective modal damping of mode i and E is the unit matrix (2).

$$\mathbf{A} = \begin{bmatrix} \mathbf{0} & \mathbf{E} \\ \boldsymbol{\tau_1} & \boldsymbol{\tau_2} \end{bmatrix} \text{ with } \tau_1 = \begin{bmatrix} -\omega_1{}^2 & 0 & 0 \\ 0 & \dots & 0 \\ 0 & 0 & -\omega_n{}^2 \end{bmatrix} \text{ and} \qquad (2)$$
$$\tau_2 = \begin{bmatrix} -2d_1\omega_1 & 0 & 0 \\ 0 & \dots & 0 \\ 0 & 0 & -2d_n\omega_n \end{bmatrix}$$

Before applying the model order reduction method inputs and outputs have to be defined as nodes. Inputs are the application points of the driving forces. Outputs are the TCP and the centres of the measuring surfaces of the capacitive sensors. The next section describes the FE model and the parameters used.

3.2. FE model

Figure 3 shows the mesh of the simplified design of the positioning system. The model aims to predict the dynamic errors at the TCP. Only the relative motion between workpiece and tool is interesting. Structural dynamic vibration that comes from the steel frame is not considered. Thus the frame is excluded from the FE model which reduces the computing time.

The aerostatic bearings are modeled as spring-damper elements that act between the appropriate guiding surfaces of the aerostatic slide and the guide rails. Initially the manufacturer's stiffness parameters are applied. Furthermore a local damping factor of 0,01 is assumed for the air pads. For the global damping factor an initial value of 0,02 has been chosen.

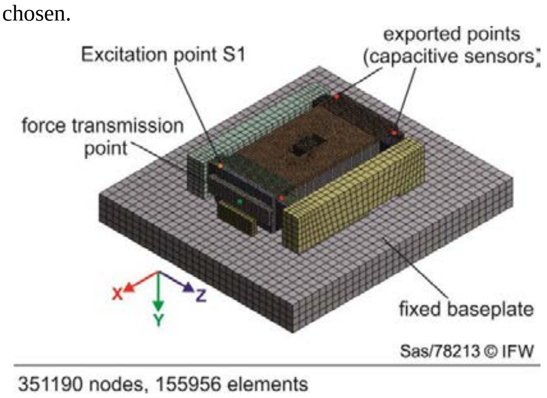

Figure 3: FE model

4. Metrological analysis of the system

In this section geometric and dynamic measurements are carried out for the ultra-precision demonstrator. The results are used to identify the initial model's parameters.

4.1. Geometric errors

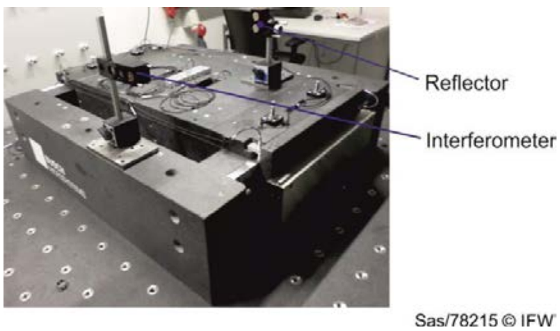

Figure 4: Setup for the measurement of the position error

The following geometric errors of the positioning system have been measured by means of a laser interferometer: The positioning error in Z-direction, the straightness errors in X and Y direction and the orientation errors (pitch and yaw). Figure 4 shows the measurement setup for the determination of the positioning error in Z-direction. The positioning error was recorded 10 times at 5mm-intervals from both directions along the travel of 100 mm. Figure 5 shows the mean values of the positioning error along the travel in positive and in negative direction. The maximum positioning error is -2,5 µm. The maximal reversal range of 3,4 µm has been recorded at the position Z= -50 mm.

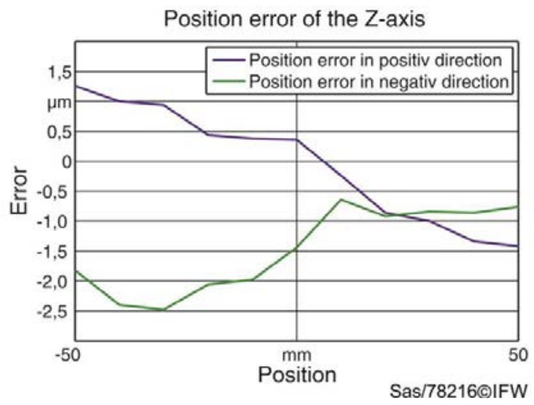

Figure 5: Positioning error in positive and negative Z-direction

The quasistatic measurements of the orientation and straightness errors were recorded 5 times for a speed of 1mm/s. Figure 6 shows the straightness error in the Y direction. The maximal deviation is -0,215 µm.

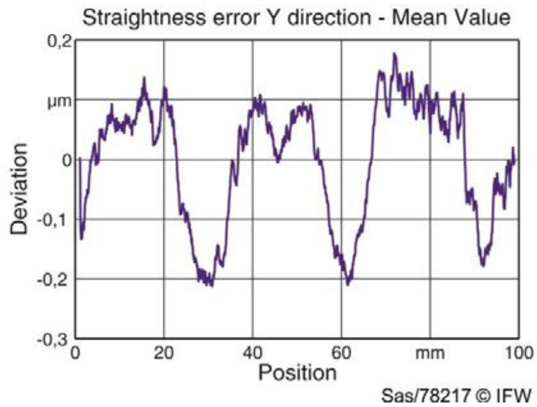

Figure 6: Straightness error in Y-direction

The geometric errors are evaluated and saved in look-up tables. By interpolating between the values for the defined positions the geometric errors are added to the errors predicted by the model as Figure 2 shows.

4.2. Modal analysis

For the modal analysis five positions of the Z-axis are investigated along the travel of 100 mm (-50, -25, 0, 25 and 50 mm). Figure 7 shows the measurement setup.

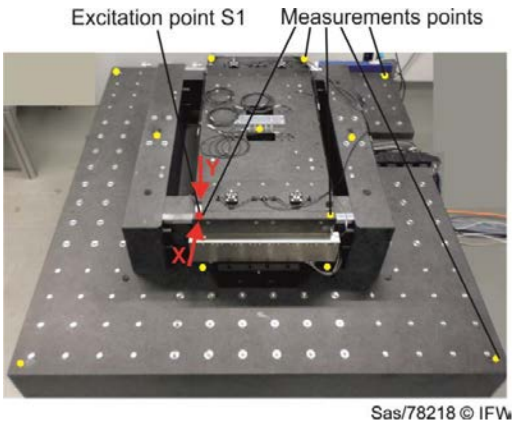

Figure 7: Setup of the modal analysis

At every Z-position an impulse is applied 10 times on a defined point. The system's dynamic behavior is recorded by acceleration sensors. These have been placed on the excitation point S1, on the Z-axis's slide and on the granite baseplate to record the eigenmodes of the complete system. The magnet axis is not yet ready for operation. Thus the investigation of the vibration behavior on the TCP is not yet possible. The point S1 on the aerostatic slide has been chosen to determine the vibration behavior in the X- and Y-direction. Here, the linear drives are powered on to hold the slide's position.

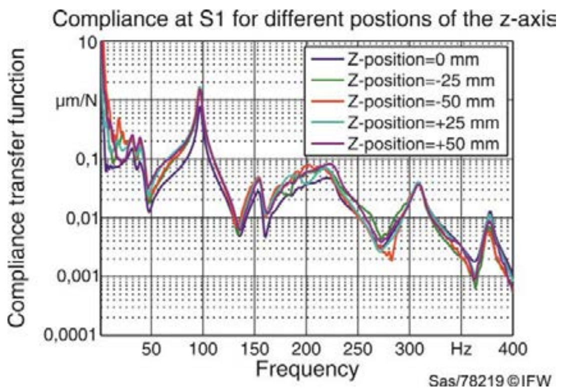

Figure 8: Dynamic compliance in Y direction for several axis-positions

Figure 8 shows the compliance transfer function on the excitation point in Y direction for several Z-axis positions. The most eigenfrequencies are the same along the travel of 100 mm. Only at a frequency of 18 Hz a resonance for the positions z=-50 mm, z=-25 mm and z=+25 mm can be detected. However discrepancies in the resonance amplitudes can be detected along the travel. Regarding the required positioning accuracy for the ultra-precision system the position of the Z-axis could have an impact on the system's dynamic compliance.

For the middle Z-axis-position (z=0) the eigenmodes have been calculated. For an eigenfreqency of 32 Hz and 39 Hz the aerostatic axis and the granite baseplate are moving together in the same way. This means that these vibrations come from the steel frame. Thus they don't have to be built in the FE model. Up to a frequency of 97 Hz the eigenmodes show a movement of the aerostatic slide while the baseplate is not moving. This means that the eigenmodes up to 97 Hz come from the compliance of the slide's structure or from the compliance of the air bearings. Figure 9 shows the system's compliance transfer function at the excitation point S1 in X- and Y-direction.

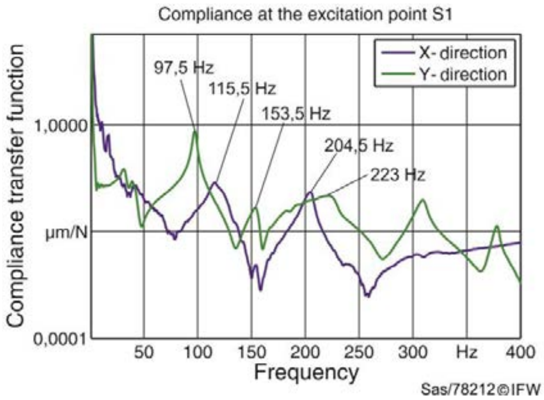

Figure 9: Dynamic compliance in the X- and Y-direction (Z=0 mm)

5. Parametrization of the model

Based on the knowledge about the system's dynamic behavior from the experimental modal analysis the model's stiffness and damping parameters are adjusted.

A modal analysis is carried out for the complete system. The magnetic guiding system is expected to have a bandwidth of about 400 Hz. The expected controlling bandwidth is about 250 Hz. Thus the eigenfrequencies of interest are set up to 400 Hz. The excitation point that has been chosen for the experimental modal analysis is defined as a further input and output node for the model order reduction to compare the measured and calculated compliance transfer functions.

First of all the calculated eigenfrequencies are adjusted to the measured ones by identification of the stiffness values of the air pads. For this purpose the calculated eigenmodes that match the measured ones are considered.

In the next step, the damping values are modified in order to adjust the resonance amplitudes to the measured ones. The damping values are the system's global damping ratio and the local damping values of the mode shapes.

Figure 10 shows the calculated and the measured compliance transfer functions at the excitation point S1 in the X- and Y-direction and the appropriate mode shapes. Three calculated mode shapes for the frequencies 97 Hz, 115 Hz and 222 Hz agree with the measured ones. However the calculated mode shape for the frequency 153 Hz differs from the measured one. Here, the model calculates only the compliance in Y-direction.

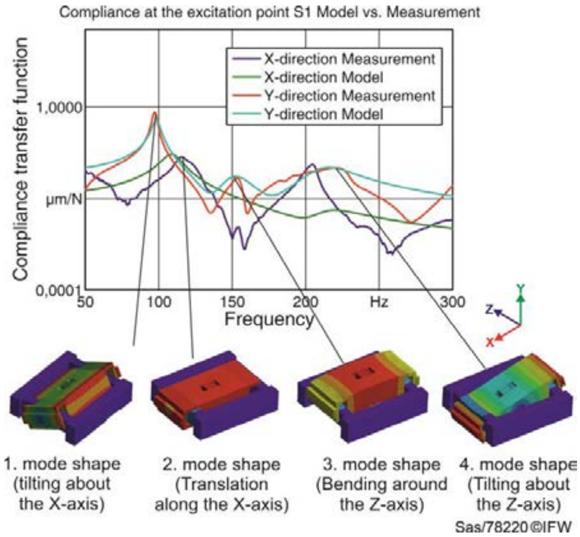

Figure 10: Comparison Model vs. Measurements

6. Conclusion and outlook

The productivity of ultra-precision machining is limited by low feed rates and spindle speeds. To increase the dynamic without affecting the position accuracy the discrepancies caused by higher velocities and accelerations have to be compensated. Conventional compensation relying only on measured values is insufficient for the accuracy demanded because it relies only on measured values. Since there are restrictions in measuring the errors caused by dynamic compliance at the TCP a system's model is needed to predict the position errors at the TCP.

In this paper a model of an ultra-precision positioning system is presented. The model is the result of the order reduction of an FE model.

Geometric errors have been measured along the travel. These are compensated by the use of lookup tables. In order to parametrize the dynamic model an experimental modal analysis has been carried out for the Z-axis in x- and y-direction. The FE model has been adjusted according to the measured mode shapes by identifying the stiffness and damping parameters. The results show that the calculated compliance transfer functions match with the measured ones for three mode shapes. After that a model order reduction has been applied to obtain the system's state space model.

This model is considered as an initial model. In the next step the model will be optimized online by the use of the integrated sensors which are the capacitive distance sensors. Based on an observer the system's states are adjusted. Furthermore an online parameter identification, as described in [7], is carried out to increase the model's accuracy.

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