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Deflection compensation on a force sensing mobile machine tool

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Abstract

Within this paper, a model based deflection compensation for a mobile machine tool prototype is presented. The model compensates the deflection of the tool center point (TCP) based on strain gauge measurements at the machines foot modules. The measured strains at each foot are used to calculate pose dependent gravitational forces as well as disturbance forces. These forces are then passed to the model. Hence, the model reconstructs the deflection of the TCP based on the measured forces at the foot modules.

Keywords: *mobile machine tool; displacement compensation; strain gauges; mobile machining; sensory foot modules*

1. Introduction

Machining of large parts with a mobile machine tool results in significant savings of time, energy and transportation costs caused by moving a small and lightweight machine instead of the heavy part. The mobile machine tool, which is described in this paper, can therefore be easily transported to the larger work-piece and is able to crawl discontinuously along the surface of the workpiece. This allows flexible machining of large parts at reduced costs [1]. Due to the smaller and lightweight structure of mobile machines, stiffness and accuracy are reduced in comparsion to machine tools. Especially, displacements due to the machines own weight regarding different poses of the axis as well as occurring process forces have to be taken into account [2].

Machine integrated concepts to measure forces for a displacement compensation have been successfully demonstrated as a force sensing spindle holder in [3] and as a sensory machine structure in [4]. Findings from these projects have not been transferred to a mobile machine tool, yet.

This paper aims for presenting a design approach of a sensor system to compensate displacements of the tool center point (TCP) of a mobile machine tool prototype considering its own weight, process forces and a detection of overload. The system consists of strain gauges at the machines vacuum feet, a model to reconstruct the resulting forces from strain measurements as well as an analytical model to predict the displacement from the calculated forces.

Nomenclature

W	Mass matrix
M	Torque vector
G	Damping matrix
C	Stiffness matrix
K	Calibration matrix
V	Relation matrix
J	Mass moment of inertia
F	Force vector
q	Coordinate vector

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2. The mobile machine tool prototype

The mobile machine tool prototype was designed by the "Korea Institute of Machinery & Materials" (KIMM) and was presented in [5]. It consists of two linear axis (X and Y) to move the TCP in a plane. A Z-axis is implemented at the TCP to guide the spindle towards the workpiece. While machining, the mobile machine tool uses three vacuum feet to fixate itself on the workpiece. For crawling on the workpiece, a separate set of three vacuum feet is used while the X- and Y-axis as well as a rotating C-axis relocate the main body of the machine tool.

The spatial location of the mobile machine on the workpiece is captured by using a separate 3D coordinate measuring system. This system is based on multilateration and absolute distance measuring. More details about the systems laser tracker design can be found in [6].

A simplified test bench including the X-, Y-axis and a set of three vacuum feet was installed at the "Institute for production engineering and machine tools" (IFW) at the Leibniz University Hannover in Germany to collaborate with the KIMM and to be able to investigate the system in parallel. Offering a proof of concept beside the simulation based approach on developing the sensor system to compensate for pose dependent TCP displacements is the main aim of the test bench. It consists of two stacked linear axis and attachments for the connection of the vacuum feet. This test bench is mounted on a slot table, which can be tilted up to 90° to simulate different orientations of the mobile machine tool on the workpiece. A small control cabinet is mounted below the slot table including the power electronics and a Beckhoff IPC for implementation of the developed algorithms. A CAD model of the test bench can be seen in figure 1 as well as a picture in figure 4.

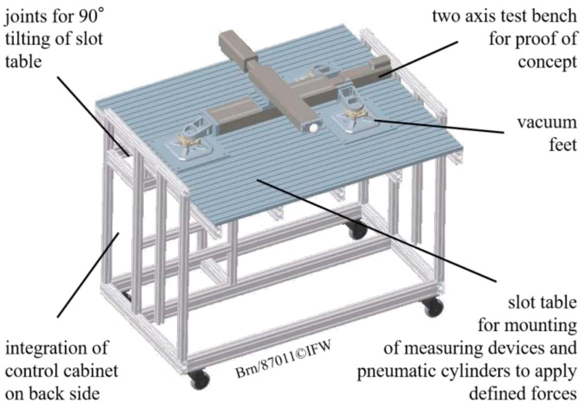

Fig. 1. Test bench for validation of the sensor system.

3. Development of the sensor system

3.1. Requirements

The main task of the sensor system is to capture a measurand that can be directly or indirectly related to the TCPs spatial displacement in X, Y and Z direction. In case of an indirect measurement, an additional calibration to reconstruct the displacement from a different physical value is needed. The intended properties of the system can be found in table 1.

Table 1. Target properties of sensor system.

Description	Value	Unit
Degrees of freedom to be measured	3	-
Accuracy of displacement measurement	0.05	mm
Measurement range	2	mm
Measurement frequency	100	Hz
Cost of measuring system	2000	€

Considered sensors were eddy current sensors, capacitive sensors, laser triangulation sensors, confocal sensors, tactile sensors, strain gauges, draw-wire sensors and a 360° lidar scanner.

None of mentioned sensors could provide a solution within the defined budget and the target accuracy except strain gauges. Therefore, they were the primary choice to measure the strain, then reconstruct the forces and finally to calculate the deflection due to the forces.

3.2. Designing the sensory modules

For convenient integration into the mobile machine tool and separate calibration, the sensor system should be detachable. The machines flanges of the vacuum feet were chosen due to their exchangeability and their placement within the flow of forces. Furthermore, the design of the flanges can be customized to incorporate the strain gauges as well as further electronics. Hereby, no modifications have to be done to the machine tool and the measuring system can be calibrated separately. All effective forces acting on the machines structure can be observed. In contrast to a central measuring system (as can be found in [4]), it is additionally possible to measure the forces of each flange separately to monitor an eventual overload of the vacuum feet.

In a first step, the flange was optimized by using FEM to reduce its compliance below 50 µm at 500 N of force in every direction. Secondly, three spots with high strains in the measurement directions X, Y and Z were chosen and analyzed regarding the measurable strain amplitudes. Each position was chosen to produce differing strain maxima in X, Y and Z to be able to reconstruct the direction of force within the strain gauge measuring system (figure 2).

Based on the simulations three flanges were manufactured. Aluminum was chosen as the material to achieve a light weight design for the given space. Afterwards, the strain gauges were integrated and connected to a measuring electronic for testing and calibration.

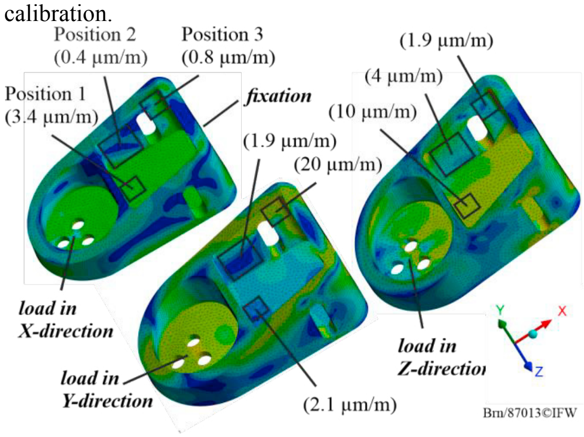

Fig. 2. Chosen positions for strain gauges at a load of 500 N.

3.3. Signal processing

One signal processing electronic is used for each foot module to connect the three strain gauges per module. This guarantees short analog transmission distances with the lowest risk of electromagnetic distortions. Within the electronic, the signal is digitized and then transmitted via CAN-Bus to the Beckhoff IPC. The sensory flange module with its three strain gauges and the electronic can be seen in figure 3.

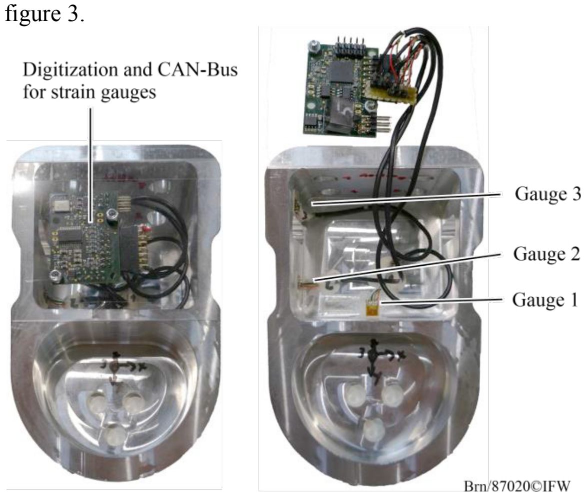

Fig. 3. The sensory flange module with electronic.

A maximum of four strain gauges per foot module and electronic can be transmitted at 600 Hz. The transmitted data is already smoothed by a mean value filter at the analog to digital converter (ADC). Each channel of the ADC measures at 10 kHz. Therefore, the specification of a sample rate of 100 Hz for the compensation of the deflection due to the process forces can be achieved.

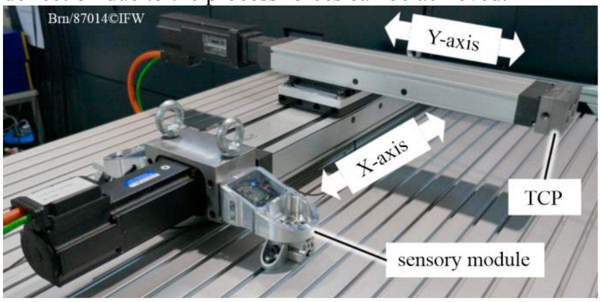

Fig. 4. Two-axis test bench with mounted sensory modules.

3.4. Calibration of strain to force

Based on the applied force at the TCP and the strains at the nine sensor positions a correlation can be established. This relationship is expressed by the calibration matrix. As a first proof of concept, all sensors are used to directly relate to the TCP. The relation applies as follows:

$$F = K \cdot u_{DMS}$$

with the force vector $\boldsymbol{F}$, the calibration matrix $\boldsymbol{K}$ and the signal vector $\boldsymbol{u_{DMS}}$,

$$F = \begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix}, \qquad u_{DMS} = \begin{bmatrix} DMS_{11} \\ DMS_{12} \\ DMS_{13} \\ DMS_{21} \\ DMS_{22} \\ DMS_{23} \\ DMS_{31} \\ DMS_{32} \\ DMS_{33} \end{bmatrix} \qquad and$$

$$K = \begin{bmatrix} k_{11} & k_{12} & k_{13} & k_{14} & k_{15} & k_{16} & k_{17} & k_{18} & k_{19} \\ k_{21} & k_{22} & k_{23} & k_{24} & k_{25} & k_{26} & k_{27} & k_{28} & k_{29} \\ k_{31} & k_{32} & k_{33} & k_{34} & k_{35} & k_{36} & k_{37} & k_{38} & k_{39} \end{bmatrix}.$$

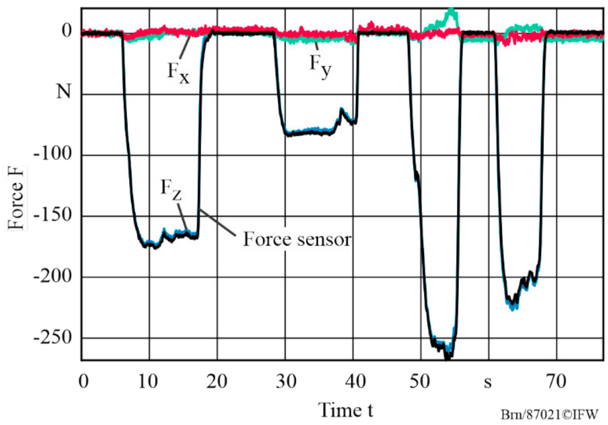

Fig. 5. Comparison of applied force and predicted force at the TCP in Z-Direction

To determine the calibration matrix, known forces have to be applied at the TCP. The single axis force sensor HBM U9B was used to measure X, Y and Z forces separately. While measuring in one direction the other directions were assumed with zero force. Next, the overdetermined system of equations can then be solved with the method of least squares. Figure 5 shows exemplary the results for a calibration in Z. As can be seen, a match with only minor amplitudes in X and Y direction could be achieved. Hence, the calibration matrix can be used as an input for the deflection model in the following.

4. Model to estimate the machines displacement

In the next step, the deflection model was designed. The model estimates translational deflections caused by the measured forces at the TCP. In order to achieve the simplest and at the same time most easy-to-understand modelling possible, the mobile machine tool was initially regarded as a rigid body. This first iteration is shown in figure 6.

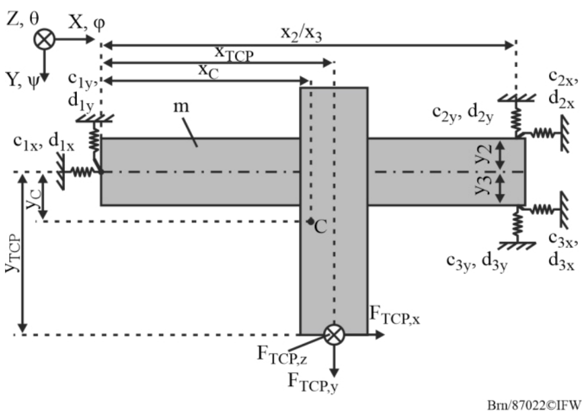

Fig. 6. Rigid-body model for mobile machine tool.

Each foot module is represented by three-dimensional stiffness and damping parameters. Furthermore, the machine can adopt different poses. The model must therefore adapt the center of gravity C according to the machines pose. This is taken into account via geometry parameters to represent the pose of the machine.

Every axis performs individual eigenmodes, which aren't described by only one rigid body. Therefore, the model was extended to a multi-body model to include the dynamic behavior. In order to map the first respective eigenmodes, the two axes are each divided into two "virtual" bodies, so that the machine structure consists of four bodies. As an example, figure 7 shows the resulting free body diagram for the Y-axis. Point P1 splits this axis and is always at the outer edge of the X-axis. Both axes are connected at point P2. The two bodies are created with the respective centers of gravity S regarding the pose and their own body coordinate systems.

The equations of motion can be set up for each body. Therefore, the interface between two bodies is modelled as a ball joint with three torsional stiffnesses and dampings. One body is then described by three translational (force) and three rotational (moment) equations. Using four bodies results in 24 equations in total. By mathematical description of the relative movement of the bodies to one another, this number can be reduced. Exemplary for body K1, the relation of movement considering only small angular movements is as follows:

$$x_1 \approx x_{TCP} \cdot \cos(\theta_{pos}) + y_{TCP} \cdot \sin(\theta_{pos}) + a \cdot \theta_1 - z_{S1} \cdot \psi_1$$

$$y_1 \approx y_{TCP} \cdot \cos(\theta_{pos}) - x_{TCP} \cdot \sin(\theta_{pos}) + z_{S1} \cdot \varphi_1$$

$$z_1 \approx z_{TCP} - a \cdot \varphi_1$$

This leads to a coordinate vector $\boldsymbol{q}$ with only 12 entries:

$$q = [x_{TCP}\ y_{TCP}\ z_{TCP}\ \ \varphi_1\ \psi_1\ \theta_1\ \ \varphi_2\ \psi_2\ \theta_2\ \ \varphi_4\ \psi_4\ \theta_4]^T$$

The equations of motion then have to be resolved according to the coordinate vector or its temporal derivatives. The elimination of the internal forces and the solution of the linear equation system can be done via the Symbolic Toolbox in Matlab. Then the equation of motion is available in matrix notation as follows:

$$W \cdot \ddot{q} + G \cdot \dot{q} + C \cdot q = V \cdot F$$

with the mass matrix $\boldsymbol{W}$(12x12), the damping matrix $\boldsymbol{G}$ (12x12), stiffness matrix $\boldsymbol{C}$(12x12), the coordinate vector $\boldsymbol{q}$ (12x1), the relation matrix $\boldsymbol{V}$(12x6) and the force vector $\boldsymbol{F}$ (6x1). The relation matrix represents the combination of model parameters.

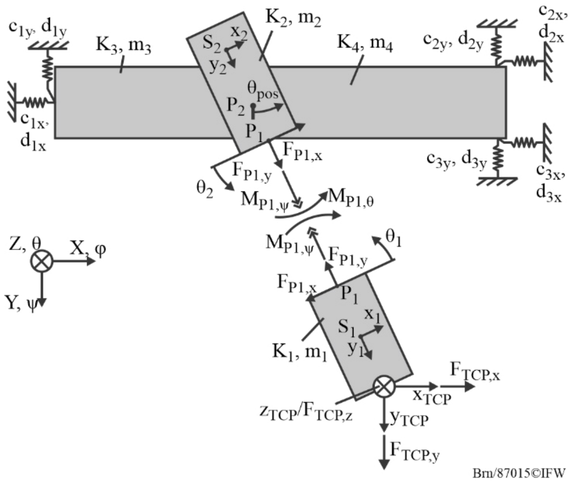

Fig. 7. Multi-body model of Y-axis for mobile machine tool.

In the next step, the mass, damping and stiffness parameters of the model have to be identified. As mentioned before, the mobility of the machine was taken into account insofar as on the one hand the relative position of the centers of mass are defined by adaptable geometric parameters. On the other hand, the virtual division of the structure into the individual bodies is carried out depending on the machine position. In consequence, the masses and moments of inertia as well as the damping and stiffness parameters are depending on the pose. First, it must be examined whether stiffness and damping parameters can be found for the model that map the movement of the TCP for a certain pose. The masses and moments of inertia as well as the geometry parameters are thus defined via the pose. Therefore, the stiffness and damping values of the individual coupling points have to be fitted based on a deflection measurement. The deflection measurement setup included a Renishaw XM-60 6-DOF laser interferometer. Random forces were applied using a force sensor at the TCP to investigate the models reaction.

The result of the model parameterization is shown in the following two figures. It can be seen that the analytical model is particularly capable of mapping the displacement caused by a force in Z-direction (figure 8). Forces in Y-direction (figure 9) are not correctly calculated in their absolute value although the silhouette of the prediction is correct. The load in X behaves analogously to the Y-axis and is therefore not shown. Thus, a model extension by adapting the axis coupling is necessary to reproduce the deflection in X and Y direction correctly.

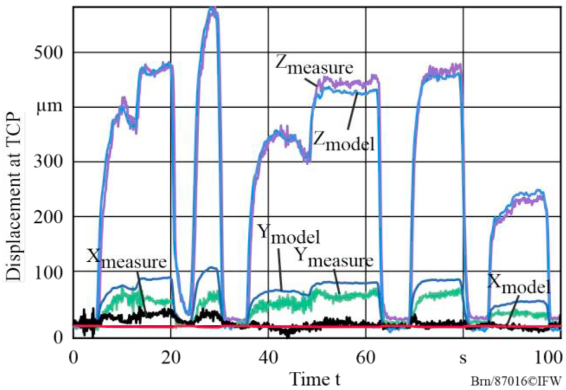

Fig. 8. Measured and simulated displacement at the TCP for forces in Z.

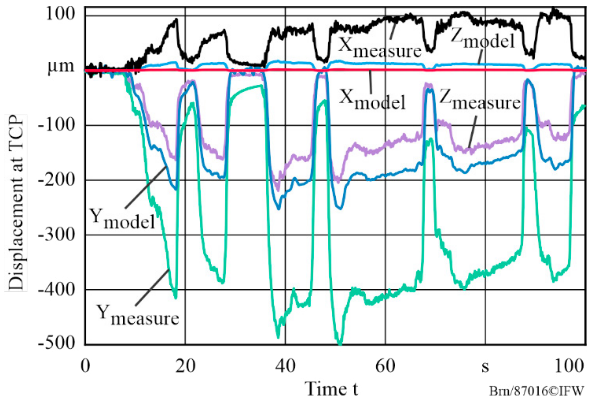

Fig. 9. Measured and simulated displacement at the TCP for forces in Y.

The final comparison of the measured and predicted displacement showed high correlation in Z-direction for a load in the same direction. The deflection error could be reduced below 50 µm by predicting 430 µm of a total 450 µm. On the downside, only 250 µm of the 500 µm displacement for a load in Y-direction could be predicted. Hence, the model based compensation combined with the sensory foot modules is capable of reducing a major portion of the robots position error under load.

5. Conclusion

This paper gave insights on the design of the sensory foot modules as well as genuine measurements on a test bench to validate a deflection compensation for a mobile machine tool. First, the design and calibration of the sensory foot modules was show. Forces could be reconstructed by using a calibration matrix and the strain measurements. Second, the deflection measurements compared the calculated displacement to the actual displacement while varying forces were applied to the TCP. Predictions by the model were congruent within the target 50 µm for the Z direction. In contrast, only 50 % of the deflection could be predicted in X and Y direction. Therefore, the target of 50 µm could not be achieved. Missing degrees of freedom and linkages within the model were identified as the root cause for the deviations in X and Y direction. In future, further detailing of the models degrees of freedom should enable a reduction of 50 µm in every direction.

Acknowledgements

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