Modeling and Control of a Piezo-Actuated High-Dynamic Compensation Mechanism for Industrial Robots

Björn Olofsson, Olof Sörnmo, Ulrich Schneider, Anders Robertsson, Arnold Puzik and Rolf Johansson

***Abstract*— This paper presents a method for modeling and control of a piezo-actuated high-dynamic compensation mechanism for usage together with an industrial robot during a machining operation, such as milling in aluminium. The machining spindle is attached to the compensation mechanism and the robot holds the workpiece. Due to the inherent resonant character of mechanical constructions of this type, and the nonlinear phenomena appearing in piezo actuators, control of the compensation mechanism is a challenging problem. This paper presents models of the construction, experimentally identified using subspace-based identification methods. A subsequent control scheme, based on the identified models, utilizing state feedback for controlling the position of the spindle is outlined. Experimental results performed on a prototype of the compensation mechanism are also provided.**

I. Introduction

Due to the limited positioning accuracy and stiffness of industrial robots, machining has traditionally been performed using dedicated CNC machines, when accuracy higher than 0.1 mm is required. However, since industrial robots may offer more flexible and cost-efficient machining solutions than CNC machines, it is desirable to use industrial robots for machining tasks, such as, *e.g.*, milling in aluminium.

Within the EU/FP7-project COMET [1], the aim is to develop solutions for machining with industrial robots with accuracy greater than 50 μm. For high-precision machining, a high-dynamic mechanism for real-time compensation of the remaining position errors of the robot is developed. This unit is called a High-Dynamic Compensation Mechanism (HDCM).

This paper presents modeling and control of a prototype of the HDCM-unit. The construction of the unit has been discussed in several earlier papers, see, *e.g.*, [2], [3], and therefore, the mechanical design is only briefly described below. The focus of this paper will be on the dynamic properties and how the subsequent control design should be optimized for satisfactory milling results. It will be shown how nonlinear effects in the HDCM can be handled and how the mechanical vibrations in the construction can be reduced by using appropriate control design methods.

This paper is organized as follows. Section II describes the experimental setup of the industrial robot and the HDCM-unit, as well as the corresponding environment for simulation and testing of the control design. Section III describes initial experiments performed on the HDCM for dynamic characterization. Based on the dynamic characterization, modeling of the construction is discussed in Section IV and the subsequent model-based control design is described in Section V. Experimental results are presented in Section VI and finally conclusions and future work are given in Section VII.

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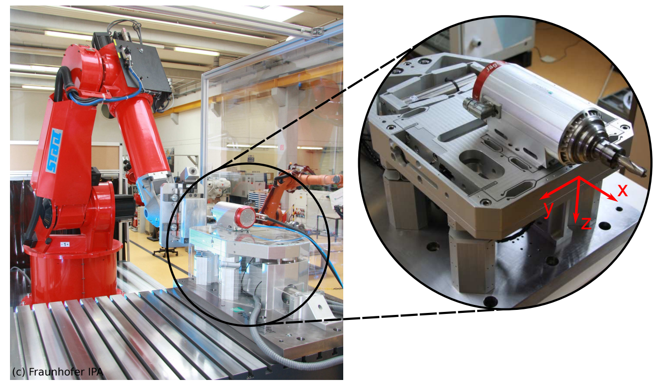

Fig. 1. The experimental setup for real-time compensation of positioning errors during machining operations, where the robot holds the workpiece and the spindle holding the milling tool is mounted on the HDCM-unit. A detail of the HDCM and the Cartesian axes, along which compensation is possible, are also displayed.

II. Experimental setup

A prototype of the HDCM has earlier been developed and reported in several publications, see, *e.g.*, [2]. In the experiments in this paper, the HDCM is to be used together with a REIS RV40 industrial robot. The robot holds the workpiece and the spindle is consequently attached to the HDCM, see Fig. 1.

A. Construction

The construction of the HDCM is such that motion of the spindle is possible in three Cartesian directions, hereafter called x, y and z, respectively, see Fig. 1. The three axes are designed to be decoupled. The motion in each direction is achieved by the forces generated by three individual piezo actuators. The extensions of the piezo actuators are translated to a corresponding translational movement of the spindle *via* a flexure mechanism. The flexure mechanism is constructed such that the gear ratio of the displacement of the spindle and the extension of the piezo actuator is approximately five in each direction. This realizes a compensation range for the machining spindle of approximately 0.5–1 mm in each Cartesian direction.

B. Actuation and sensors

The extension of the piezo actuators is changed by applying a voltage, and the extensions are measured using strain gauges, attached to the actuators. The Cartesian displacement of the spindle is measured with capacitive sensors, one in each direction.

C. Interface to the HDCM

In order to develop the control structure for the HDCM-unit, all sensors and actuators are integrated using a dSPACE system of model DS1103 [4]. Using the software ControlDesk, the user can implement new control strategies in a simple manner as well as develop graphical user interfaces. The control design described in this paper has been implemented in MATLAB Simulink, then generated to C–code using the *Real-Time Workshop* toolbox [5]. The compiled C-code is installed in the dSPACE system and executed at a sampling frequency of 10 kHz.

III. Dynamic characterization of the construction

Due to the inherent resonant character of mechanical systems and the nonlinear effects that appear in piezo actuators, accurate positioning control of the HDCM without vibrations is a challenging control problem. A model-based solution is here pursued in order to control the tool position.

A. Nonlinear phenomena in the piezo actuators

Experiments have been performed on the HDCM in order to determine the effect of the nonlinear phenomena in the piezo actuators. The experiments indicated that the main nonlinearities that need to be handled are hysteresis and the creep phenomenon. Results from experiments where the voltage to the piezo actuators are alternatingly increasing and decreasing are shown in Fig. 2. It is obvious that the hysteresis needs to be handled actively for accurate positioning. It is also noted that the hysteresis is both rate and amplitude dependent. On the other hand, experiments showed that the nonlinear creep phenomenon in the actuator is a much slower process, and thus easier to handle.

Although different in nature, both of these nonlinear effects can be handled using high-gain feedback. The control design will be described in Section V.

B. Frequency characterization of the mechanical construction

In order to characterize the frequency properties of the mechanical construction, several frequency response experiments have been performed. The frequency spectra in the different directions, displayed in Fig. 3, were estimated using the periodogram method. An important property of the system is the location of the first natural eigenfrequency. It is noted that the characteristics are quite different in the three Cartesian directions. In particular, two natural eigenfrequencies are visible in the x- and z-axes, whereas only one is visible in the y-direction. The first eigenfrequency appears in the frequency range 33–47 Hz in all of the three axes.

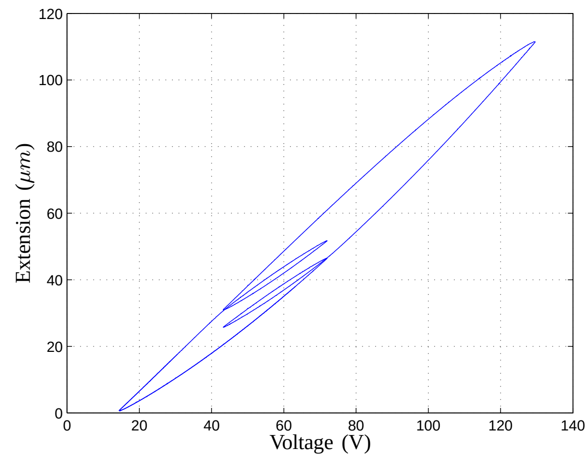

Fig. 2. Extension of the piezo actuator as function of the input for a triangular wave with varying amplitude as input.

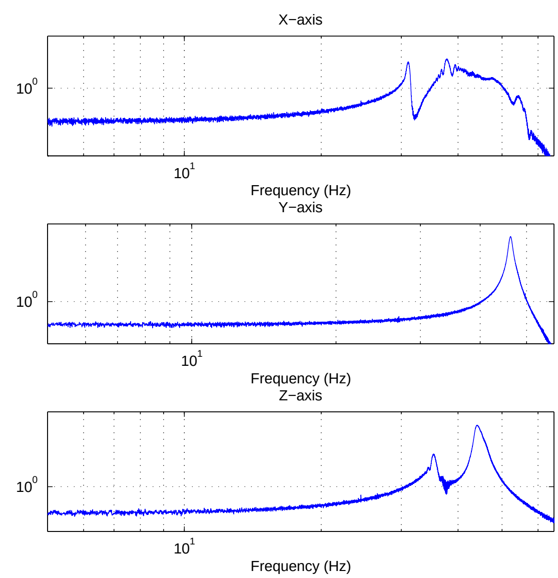

Fig. 3. Estimated frequency spectra in the Cartesian directions.

The locations of the eigenfrequencies are important since they limit the achievable bandwidth, *i.e.*, the velocity of the control loop, in the final closed-loop control system. Increasing the bandwidth beyond the resonance frequency requires a lot of control actuation and the sensitivity to model errors becomes significant.

IV. Modeling of the mechanical construction

In order to design control algorithms, it is advantageous to perform modeling of the HDCM prior to the design. Two different methods for modeling can be chosen. Firstly, modeling based on mechanical relations can be established, where the construction specific parameters are either analytically calculated or experimentally determined.

The other approach is to consider black-box input-output models without investigating the internal mechanical construction. This is a common approach in model-based control, which results in satisfactory control performance given that the model captures the essential dynamics of the system. This approach is investigated in this paper for modeling of the HDCM.

A. Identification based on black-box models

Using system identification methods [6], mathematical models describing the HDCM can be determined. The axes can be assumed to be decoupled, conditioned that the mechanical design is made such that the motions of the different directions are independent. This assumption is made in this paper. Consequently, each axis is considered as a system with one input and one output. Identification of the models was done in the System Identification Toolbox [7] in MATLAB and the State space Model Identification (SMI) toolbox [8] for identification of state-space models.

Accordingly, consider discrete-time state-space models of the innovation form

$$x_{k+1} = \Phi x_k + \Gamma u_k + v_k \tag{1}$$

$$y_k = C x_k + D u_k + e_k \tag{2}$$

where $u_k \in \mathbb{R}^m$ is the input, $x_k \in \mathbb{R}^n$ is the state vector, $y_k \in \mathbb{R}^p$ is the output and v_k and e_k are noise sequences. The matrices $\{\Phi, \Gamma, C, D\}$ in the state-space representation are identified using one of the available implementations of subspace-based identification methods, such as the N4SID-method [9] and the MOESP algorithm [10]. During the identification of the models, a Kalman gain vector for a minimum variance estimate of the states in the model is also determined, based on the noise properties.

B. Collection of input-output data

The collection of experimental input-output data is performed in such a way that the input u_k is considered to be a scaled version of the input voltage to the actuator, whereas the output y_k is defined to be the position of the spindle as measured by the capacitive sensor.

When performing system identification, an appropriate input signal has to be chosen, such that the system is excited properly. In this paper a chirp-signal is chosen—*i.e.*, a sinusoid with constant amplitude and linearly increasing frequency—as input, since this signal gives excitation in a well-defined frequency range. Consequently, the start and end frequencies in the chirp-signal have to be chosen based on the frequency range of interest. Given the frequency spectra displayed in Fig. 3, a reasonable range of excitation is 10–60 Hz.

C. Model-order selection and preprocessing of the data

When performing identification of the state-space models, a model order has to be chosen. To this purpose, the singular values calculated during the identification procedure using the N4SID or MOESP algorithms are utilized. By plotting these singular values, the gap between the model and the noise level is identified. Based on this information, a suitable model order can be chosen.

Prior to the identification, the input-output data is processed, such that the mean and the linear trend are removed. Also, the data, which is acquired at 1 kHz, is decimated to a sample rate of 1000/6 ≈ 167 Hz, which is suitable given the location of the eigenfrequencies in the different axes.

D. Identified models

Experimentally identified discrete-time state-space models on the form (1)–(2) of the x, y and z-directions in the open-loop system were estimated. All models are of the same format. However, the model-orders vary in different directions, reflecting the number of natural eigenfrequencies, *cf.* the frequency spectra in Fig. 3. The model orders are 4, 2 and 5 in the x-, y- and z-directions, respectively. The model order selection was based on the singular values analysis during the identification procedure. As an example, the model obtained using the SMI toolbox in the y-direction is

$$x_{k+1} = \begin{bmatrix} -0.1846 & 1.071 \\ -0.8762 & -0.1588 \end{bmatrix} x_k + \begin{bmatrix} -1.029 \\ -0.06196 \end{bmatrix} u_k \tag{3}$$

$$y_k = \begin{bmatrix} -0.4567 & -0.03502 \end{bmatrix} x_k + 0.3321 u_k \tag{4}$$

The frequency spectra of the identified models are shown in Fig. 4. It is noted that there is good correspondence with the estimated periodograms in Fig. 3. A measure of the fit of the models to the data, are the *variance accounted for* (VAF) values. These numbers are 92.5, 99.5 and 97.1 for the identified models in the x-, y- and z-directions, respectively. This indicates that the models capture the essential dynamics of the system.

V. POSITION CONTROL OF THE HDCM

The control problem of the HDCM can be divided into two parts. Firstly, the nonlinear effects of the piezo actuators need to be reduced. Secondly, the oscillatory mechanical construction needs to be accurately position controlled. The control structure chosen in this paper is described below.

A. Inner piezo actuator control loop

Nonlinear systems can be controlled by both model-based feedforward control and with feedback control. Several approaches to modeling the hysteresis and subsequent model-based control design have been presented in literature, such as the Prandtl-Ishlinskii model and the Preisach model, *e.g.*, [11], [12], [13]. However, as the extensions of the piezo actuators in the HDCM are available for measurement with the strain gauges, a more straightforward solution is chosen, where an inner feedback loop is closed around the nonlinear actuator. The controller is a linear PID controller, whose continuous time transfer function $G_C(s)$ can be written as

$$G_C(s) = K_p + \frac{K_i}{s} + \frac{sK_d}{1 + sK_d/N} \tag{5}$$

where K_p, K_i and K_d are controller parameters. It is noted that the derivative part in the controller is lowpass filtered,

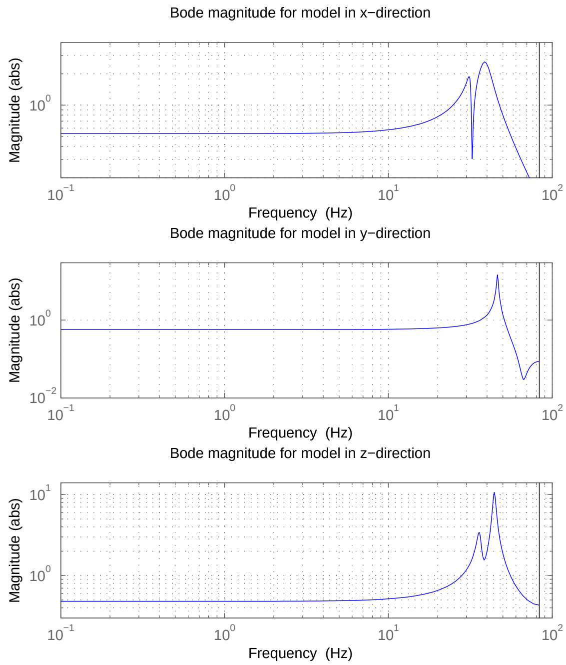

Fig. 4. Bode magnitude of the discrete-time state-space models identified using subspace identification, in the x-, y- and z-directions, respectively.

in order to reduce the amplification of high-frequency noise contaminating the measured signal from the strain gauge. The cutoff frequency in the lowpass filter is determined by the parameter N. The PID controller also has to be accompanied by an anti-windup scheme, to handle the case when the controller saturate the actuators. Discretization of this continuous time controller for implementation in the dSPACE system is straightforward, see, *e.g.*, [14].

In order to reduce the nonlinear effects in the piezo actuators, the proportional gain K_p and the integral gain K_i should be increased as much as possible, without causing instability. It will be shown by experimental results in Section VI, that this approach—*i.e.*, using a linear controller for reducing the nonlinear effects in the piezo actuator—results in satisfactory performance of the control of the piezo actuators.

B. Model-based feedback control of the HDCM

By utilizing the identified state-space models, a state feedback control loop can be designed for each of the three Cartesian directions of the HDCM. However, new models need to be identified after closing the inner feedback loop around the piezo actuators, where the reference signal to the inner PID control loop is considered as the input instead. Since the difference compared to the open loop models presented in the previous section is small, the models with the closed inner loop are not presented here.

State feedback is an appropriate structure, since damping can be introduced in the construction by suitable control design. The control law for state feedback control of the system (1)–(2) can be stated as follows

$$u_k = -Lx_k + u_{ff} \tag{6}$$

where the parameter vector L is to be chosen and u_{ff} is the feedforward control signal. The design procedure is to determine the L-vector by linear-quadratic (LQ) optimal control [14], *i.e.*, such that the cost function

$$J(u) = \sum_{k=1}^{\infty} x_k^T Q x_k + u_k^T R u_k \tag{7}$$

where the matrices Q and R are user defined weights in the optimization, is minimized.

Since all states in the state-space model of the HDCM are not available for measurement, a Kalman filter is introduced for estimation of the states, based on the measured position signal and the identified model. The Kalman filter is organized as [14]

$$\hat{x}_{k+1} = \Phi\hat{x}_k + \Gamma u_k + K(y_k - C\hat{x}_k - Du_k) \tag{8}$$
$$\hat{y}_k = C\hat{x}_k + Du_k \tag{9}$$

where the estimated states $\hat{x}_k$ and the estimated output $\hat{y}_k$ have been introduced. Since the identified model is based on experimental data, where the mean are subtracted from the real data, a *disturbance state* is added to the observer, *i.e.*, a new, constant state

$$\hat{x}^e_{k+1} = \hat{x}^e_k \tag{10}$$

is introduced. By adding this state, the correct static gain for the estimation is achieved [14]. The Kalman gain K is determined by pole placement, *i.e.*, such that the eigenvalues of the matrix $(\Phi - KC)$ are appropriate. The model identification procedure provides the Kalman gain vector for estimation of the states in the model. The corresponding pole placement is used also in the Kalman filter with the disturbance state, but with one additional pole corresponding to the extra disturbance state.

The control law for the state feedback control is then based on the estimated states, *i.e.*, $u_k = -L\hat{x}_k + u_{ff}$. In order to remove stationary errors in the position control loop, integral action is introduced in the state feedback. This is done by extending the state vector with the integral state

$$x_i(t) = \int_0^T (r(t) - y(t))\,\mathrm{d}t \tag{11}$$

where the reference signal $r(t)$ has been introduced. Introducing this extra state also requires that the state feedback vector L is augmented with one element, *i.e.*, $L_e = \begin{bmatrix} L & l_i \end{bmatrix}$, where l_i is the integral gain. Also, note that the integral state needs to be discretized prior to implementation in the dSPACE system.

Different approaches can be chosen to handle the feed forward control signal. In the scheme presented in this paper, the feed forward control u_{ff} is chosen as a direct term from the reference signal, $u_{ff} = l_r r$. The parameter l_r determines the gain of the closed-loop system and is experimentally tuned by the user. The final control structure is summarized in the block scheme in Fig. 5.

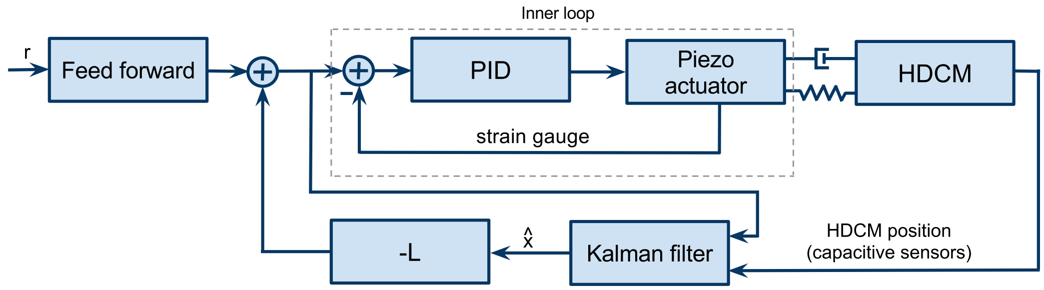

Fig. 5. Control structure for model-based control of the HDCM, where each Cartesian axis is considered separately.

VI. Experimental results

Several experiments were performed in order to evaluate the performance of the control design. Firstly, the PID controllers for the nonlinear piezo actuators are tuned, in order to achieve as high performance as possible. In experiments it is observed that the control is working satisfactory, despite the nonlinearities in the piezo actuators. When applying a triangular wave with a frequency of 3 Hz as input, the control error is within approximately $\pm 1\ \mu m$. This provides experimental evidence that the PID controller is sufficient for controlling the positions of the piezo actuators.

A. Tuning of controller parameters

In order to determine the state feedback vector L, the weight matrices Q and R in the LQ design need to be determined. Based on the identified model for the HDCM in the y-direction, the characteristics of the closed loop system was investigated for different weight matrices. Especially, the choice of the R-matrix determines the aggressiveness of the controller. Bode plots of the closed loop system for different choices of R, where the Q-matrix has been chosen as the identity matrix, can be seen in Fig. 6. The direct term l_r in the control law has been chosen such that the static gain is one in all cases. It is noted that a lower weight results in a more aggressive controller, where the resonance in the system is well damped, at the cost of reduced bandwidth. Hence, the controller needs to be tuned as a trade-off between the attenuation of the poorly damped resonance in the system and the aggressiveness of the controller. A too aggressive controller may result in unsatisfactory control performance or even instability when applied to the experimental setup, due to noise in the measurement signals.

B. Evaluation of the control design

In order to evaluate the model-based control design on the experimental setup, a reference signal was recorded as the deflection of the robot during a milling operation in one dimension, measured with a laser sensor. The recorded signal corresponds to the deflection of the industrial robot in the milling direction, which equals the y-direction of the HDCM.

Experiments were performed on the real setup with varying weight matrices. Also, the integral state was added to the state feedback, whose influence is determined by the parameter l_i. The weights $Q = I$ and $R = 2.5$, where I is the identity matrix, turned out to result in satisfactory control performance.

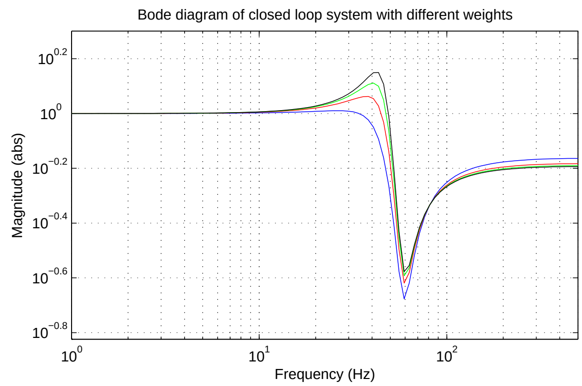

Fig. 6. Bode diagram for the closed loop system for $Q = I$ and different choices of the R-matrix in the LQ-design. The choices of R are 1.0, 2.0, 3.0 and 4.0 for the blue, red, green and black line, respectively.

The reference signal is filtered using a notch filter, where the notch is located at the eigenfrequency of the HDCM in the y-direction, as observed in the frequency spectrum in Fig. 3. This is done in order not to excite the mechanical resonance in the construction. Another option is to lowpass-filter the reference signal. However, since the construction itself is of lowpass-character, frequencies above the natural eigenfrequency in the reference signal will be attenuated.

The recorded signal was applied as reference signal in the y-direction of the HDCM. Fig. 7 shows the control performance of the inner PID controller loop. It is noted that the control error with this reference signal, which contains high frequencies, is within approximately $\pm 3\ \mu m$. This shows that the PID controller is a satisfactory control structure, as the precision required in the inner control loop is achieved. Fig. 8 shows the spindle position, as measured by the capacitive sensor. Also this figure indicates good control performance, with a control error of within approximately $\pm 15\ \mu m$.

It is observed in Fig. 8 that the control error signal exhibits periodic behavior of different frequencies. The frequency spectrum of this signal is displayed in Fig. 9. Three peaks at 9, 140 and 250 Hz are clearly visible. The first peak corresponds to the eigenfrequency of the robot and the higher frequencies are related to eigenfrequencies of the piezo actuator, the rotation of the spindle and the impact

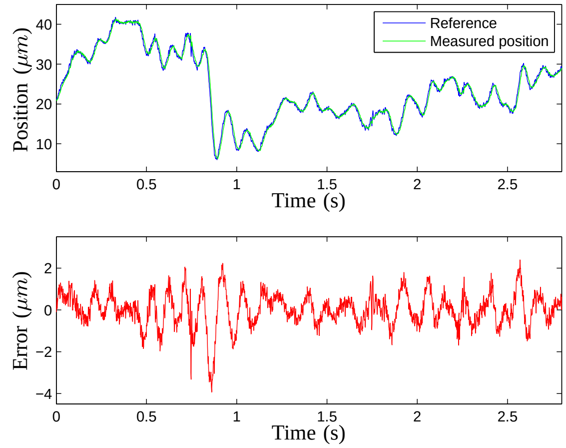

Fig. 7. Control performance in the inner PID control loop in the y-direction.

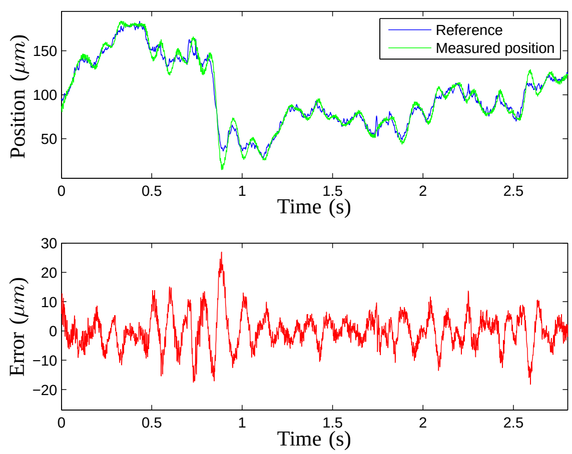

Fig. 8. Performance of the model-based spindle positioning control in the y-direction.

of the milling tool on the workpiece. Further, it is noted that the first significant eigenfrequency of the HDCM in the y-direction at 47 Hz is well damped as a result of the control design.

VII. Conclusions and Future Work

This paper has investigated modeling and control of a piezo-actuated high-dynamic compensation mechanism. The developed control structure was realized in a discrete-time implementation and experimentally verified on the prototype of the HDCM. By tuning the state feedback controller appropriately, damping was introduced in the mechanical construction by control design. The resulting control error for the reference signal recorded during a milling operation was within approximately ± 15 μm, which by far achieves the desired accuracy of 50 μm for the complete milling task.

Based on the experimental results presented in this paper, the model-based approach for control of the HDCM is promising. However, the control scheme needs to be tested

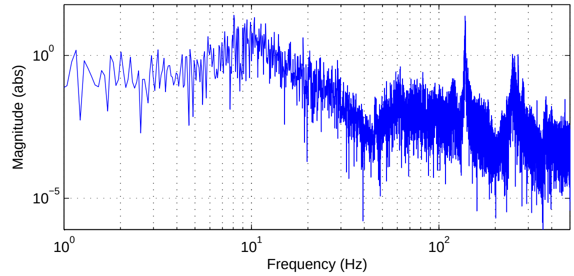

Fig. 9. Spectrum of the control error signal.

further during milling operations in order to take the process-specific disturbances into account in the control scheme. Disturbances are for example the spindle rotation and the process-forces. To this purpose, the HDCM-unit and the robot will be equipped with a 3D-accelerometer.

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