Scan performance of nanopositioning systems with large travel range

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ABSTRACT

This work presents two fine positioning stages with focus on the achieved positioning accuracy during motion. Both systems have a working range of ≥ 100 mm, are driven by linear motors and the position is measured by plane mirror laser interferometers. However the mechanical setting is quite different. The first system is a two axis fine positioning stage which is supported by ball bearing guides in a serial arrangement. Especially in the nanometer range this leads to problems caused by the highly nonlinear friction. In the second system a planar guiding with air bearings is applied. By dint of the model based control design the position accuracy in scanning mode is comparable up to a velocity of 1 mm/s.

Index Terms **–** nanopositioning, friction modeling, position accuracy, tracking error

1. INTRODUCTION

In today's high tech fabrication technologies, such as precision optics fabrication, micro- and nanotechnology and of course the semiconductor industry the measuring objects have dimensions in the millimeter range while measurable structures are in the range of atomic dimensions. Such precision measurements are performed with ultraprecise positioning systems that allow for multiaxial positioning of the specimen with respect to a sensing device. At the Ilmenau University of Technology scientists are working on providing the scientific basis for Nanopositioning and Nanomeasuring Machines (NPM/NMM) with large travel ranges of several hundred millimeters [1], [2], [3]. In order to perform measurements in such long travel ranges in an adequate time a scanning measurement mode is inevitable, while "scanning" in most applications means, moving along a straight line with constant velocity. Thus the scanning performance is determined by the lateral deviating from the ideal straight line one the one hand and by the deviation from the commanded constant velocity on the other hand.

In the course of the research work a number of different concepts for the realization of such positioning machines were developed and investigated. Therefore demonstration setups were built for individual functional subsystems as well as for entire multiaxial demonstrators. This paper investigates and compares the dynamic position accuracy of two nanopositioning machines with different concepts for the guidance and the actuation system. The differences in the mechanical setup of the two demonstrators amongst others lead to completely different requirements regarding the deployed control concept.

The following chapter introduces a two axis positioning demonstrator with ball bearing guides and in chapter three an air guided planar drive with two-dimensional travel range is presented.

2. TWO-AXIS DEMONSTRATOR WITH BALL BEARING GUIDES

2.1 Experimental setup

The first considered fine positioning stage has a working range of 200x200 mm (Figure 1). The mechanical design is a double H-structure, which means that the inner axis is placed inside the outer axis. Each axis is driven by two iron-free linear motors from the company IDAM. The engines are powered by internal developed analogue amplifiers. Hence, the necessary current can be provided with the required precision. The sinusoidal magnetic field of the permanent magnets along the moving direction is measured by integrated Hall sensors in the motor coils. These measurements are used to realize the commutation by the control system. Each axis is supported by two linear V-grooved high precision guideways. The position of the machine axes is measured by two plane mirror interferometers with a resolution of 80 pm [15]. The interferometers measure against the sides of a high precision corner mirror made of Zerodur. For data acquisition and control a modular dSpace real-time system in combination with Matlab/Simulink is utilized. The dSpace system consists of a DS1006 processor board and several analog as well as digital input and output boards. The position values are obtained from the interferometers in form of a 32-bit digital signal. The control algorithm works with a sampling rate of 10 kHz and actuates the analogue amplifiers for the drive coils with 16-bit resolution.

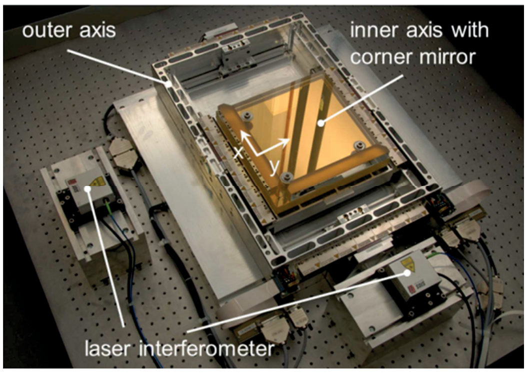

Figure 1 Two axis positioning stage with ball bearings

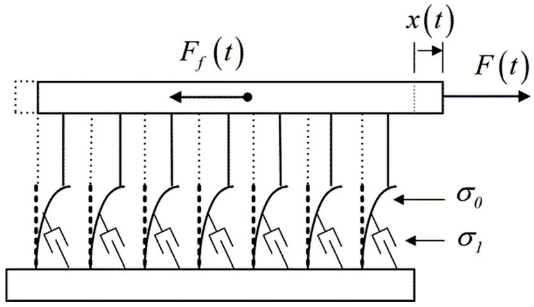

Figure 2 Bristle model

2.2 Control concept

The friction introduced by the ball bearings is the main challenge for a dynamic and precise motion control. Especially in the case of motions near zero velocity the highly nonlinear friction in the bearings dominates the system behavior [7]. Hence, linear approaches like PID control are not sufficient anymore. To achieve the required precision friction modeling and compensation is crucial to a dynamic positioning with accuracy on nanometer scale. In the last decades, dynamic friction modeling and compensation has made significant progress in the control community. Most of the major nonlinear friction phenomena, like presliding displacement, the Stribeck effect, frictional lag, stick-slip motion and other effects can be described by dynamic friction models. These models can be used for feedforward compensation of the inverse system behavior [6]. In the current work a friction model is integrated in a feedback controller. To represent the friction force a modified Lund-Grenoble model [5] is used. The model is based on the assumption that the surfaces of two rubbing objects consist of small bristles which interact with each other. For simplification only one side of the interacting surfaces has elastic bristles (see Figure 2). The deflection of all bristles is approximated by a mean Bristle deflection $z(t)$. The bristles are characterized by a stiffness σ_0 and a damping coefficient σ_1. Thus the friction force can be expressed as

$$F_f(t) = \sigma_0 z(t) + \sigma_1 \dot{z}(t) + \sigma_2 \dot{x}(t) \quad (1)$$

with

$$\dot{z} = \dot{x} - \frac{|\dot{x}|}{\tau} z \,. \quad (2)$$

The part $\sigma_2 \dot{x}(t)$ is the viscous friction. The maximum bristle deflection is in this modification of the model the constant value τ. Identification experiments have shown no significant correlation between F_f and $\dot{z}$. In this case $\sigma_1 = 0$. With this assumption the behavior of one axis can be described with

$$F(t) = m\ddot{x}(t) + \sigma_0 z(t) + \sigma_2 \dot{x}(t) \,. \quad (3)$$

Combining equation (2) and (3) and transforming in state space notation produces the nonlinear state space model of one axis of the positioning stage.

Based on this model a compensation controller with regard to [9] is designed. It is complemented by an integral output feedback. With this modification unmodeled effects do not cause static control errors.

The whole controller consists of the state feedback $r(\boldsymbol{x})$, the feedforward of the dynamic reference variables **M** and the integral output feedback K_I:

$$F = -r(\mathbf{x}) + \mathbf{M}\mathbf{w} + K_I \int (w - x)\, dt \quad (4)$$

$$r(\mathbf{x}) = -\sigma_0 z - \sigma_2 \dot{x} + m(q_0 x + q_1 \dot{x}) \quad (5)$$

The state feedback compensates the friction, reference-variable feedforward leads to an optimal trajectory tracking. The parameters q_0, q_1, the gain K_I and the **M** is calculated by pole placement.

Only the position of the corner mirror is measured by the interferometers. The bristle deflection and the velocity of the axis are unknown. To determine these signals an extended Kalman filter (EKF) is used [10]. The resulting control scheme is shown in Figure 3.

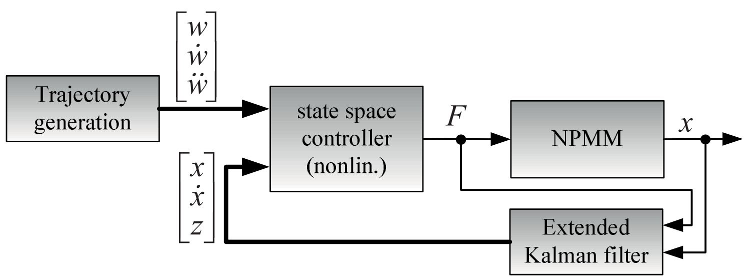

Figure 3 Control scheme of the two axis precision stage with ball bearing guides

3. PLANAR DIRECT DRIVE WITH AIR BEARINGS

3.1 Experimental setup

The basic idea for both of the investigated drive systems is to move a Zerodur-reflector very precisely while its position is measured in six degrees of freedom with high resolution laser interferometers [3], [4]. Following this idea our approach is to apply a planar direct drive system with aerostatic guiding for the lateral positioning of the Zerodur-reflector in a large travel range [14]. Figure 4 shows the principal setup and the main components of such an integrated planar drive. The moving slider is supported on three air bearing pads, providing virtually frictionless planar guiding with respect to the granite base plate. Thus the slider is free to move in x, y and φ_z. In regulated operation however, the movement in x, y and φ_z is actively controlled while the movement in z, φ_x, φ_y is mainly determined by the flatness of the granite base. The direct drive system comprises of three linear actuators each consisting of a pair of frame fixed flat coils and a permanent magnet array on the sliders underside. The driving force acts as Lorentz force perpendicular to the coil path. To achieve a position independent and constant horizontal driving force we commutate the phase currents depending on the actual slider position. The forces of the three individual drives act simultaneously on the slider and by dint of their 120° arrangement a resulting driving force in any direction within the xy-plane as well as a torque around the z-axis can be generated. By this means, the slider is driven in x-, y- and φ_z-direction while its displacement is measured with high-resolution single beam (x) and double beam (y, φ_z) plane mirror interferometers [15].

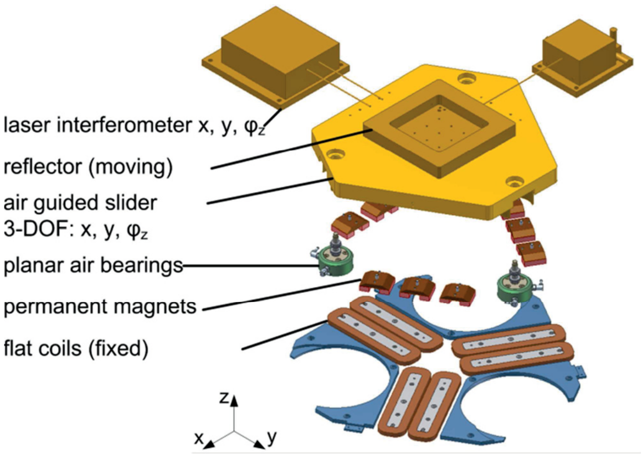

Figure 4 Scheme of the planar direct drive

The main benefit of this setup is the simple kinematic structure with the directly driven slider as the only moving part. The absence of any other transmission elements and the frictionless guiding allow for outstanding positioning characteristics. However, for the interferometers a reflector is necessary in the same size as the intended travel range. An established solution is to apply a corner cube made of Zerodur glass ceramics with coated mirror planes and a supporting structure for the specimen. In this case the corner cube is mounted on top of the slider as an individual component. At large travel ranges >100 mm this inevitably leads to a large mass and a high center of gravity for the moving part of the system. Moreover, in this way it is hard to achieve a satisfactory stiffness between the point of the force application (permanent magnet) and the point of the position measurement (laser spot on the mirror surface). In closed loop operation this compliance limits the achievable bandwidth and has thereby direct impact on the achievable servo error. These considerations led to the development of a planar drive system where the slider itself is made of Zerodur and has the reflectors directly bonded to it on the upper side [16]. Figure 5 shows this positioning system (PMS100) which provides a travel range of Ø100 mm and which is operated in an air-conditioned and vibration isolated environment. The aerostatic guiding comprises three vacuum-preloaded porous media air bearings. As with the roller guided system SIOS SP2000 plane mirror interferometers are used for the position measurement. Once the slider is floating, it has no mechanical fixture. Therefore open loop operation of the positioning system is not possible, but it is necessary to operate the planar drive as a closed loop control system. Again the control algorithms are implemented on a rapid control prototyping hardware dSpace DS1006 which in terms of the hardware parameters (resolution, sample frequency, etc.) is similar to the system described in chapter 2. Table 1 shows the main parameters of the PMS100 nanopositioning system.

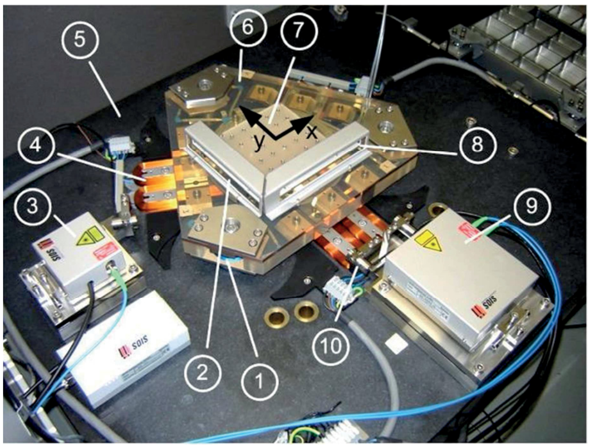

Figure 5 Experimental setup PMS100

1	air bearing	6	Zerodur slider
2	x-reflector	7	object table
3	x-interferometer	8	y, φ_z-reflector
4	drive coils	9	y, φ_z-interferometer
5	granite base	10	capacitive probes

travel range	Ø100 mm
measurement resolution x, y	0.02 nm
acceleration in x, y	250 mm/s^2
velocity in x, y	30 mm/s
measurement resolution φ_z	0.001 µrad
moving mass	9.6 kg

Table 1 Parameters of the PMS100

3.2 Control concept

The basic structure of the control concept is shown in Figure 6. System feedback is given by the three measured values of the laser interferometers. These are used to calculate the three coordinates in x, y and φ_z (input transformation). The three individual controllers for these coordinates represent the core of the control system. Each controller is built up as a cascade of a velocity- and a position controller with PID characteristics. The required machine states are reconstructed from the measured values and the controller outputs with the help of an observer. The observer is based on the rather simple system model of a double integrator. However this proved to be sufficient, as there are only marginal disturbances and disturbing forces and as the system can be considered as a rigid body in the relevant frequency range. Output values of the three axis controllers are the commanded accelerations in x, y and φ_z, which are then transformed into the required phase currents for the six drive coils (output transformation). To do so, at first the required drive forces for the three actuators are determined and then the corresponding phase currents for the drive coils (commutation) are calculated with respect to the actual slider position.

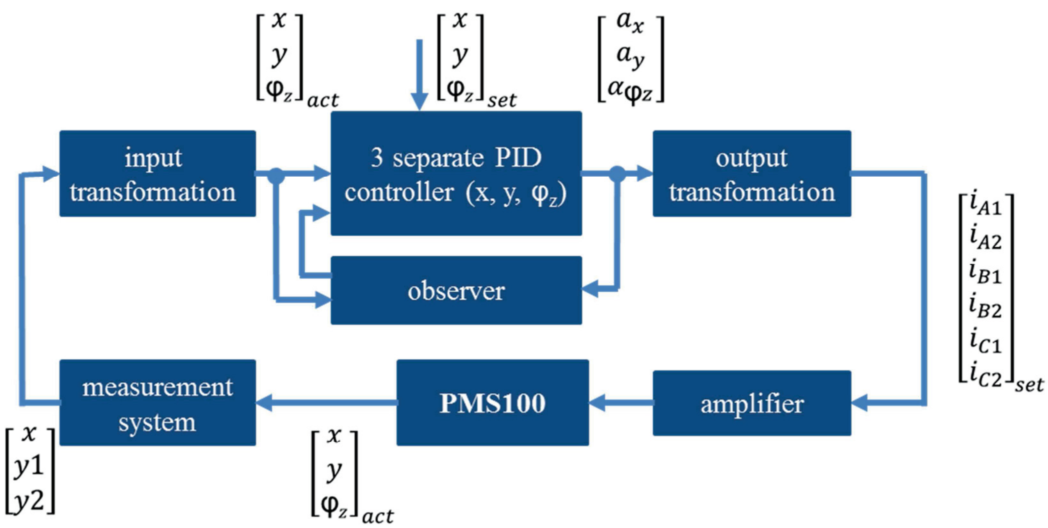

Figure 6 Control structure of the PMS100 positioning system

4. COMPARISON OF THE SCAN PERFORMANCE

4.1 Design of experiment

The two described positioning systems are to be applied within an ultraprecision measuring machine. Thus, for scanning operation the achievable tracking error while moving at a commanded constant velocity is critical. To obtain a quantitative measure of the tracking accuracy different experiments are carried out where straight lines are to be driven at a given constant velocity in x- as well as in y-direction. The path length is set according to the particular velocity to gain a similar measurement time in each run. Figure 7 exemplarily shows such a commanded trajectory as profile of the slider velocity and its position over time. The individual path lengths and velocities are given in Table 2.

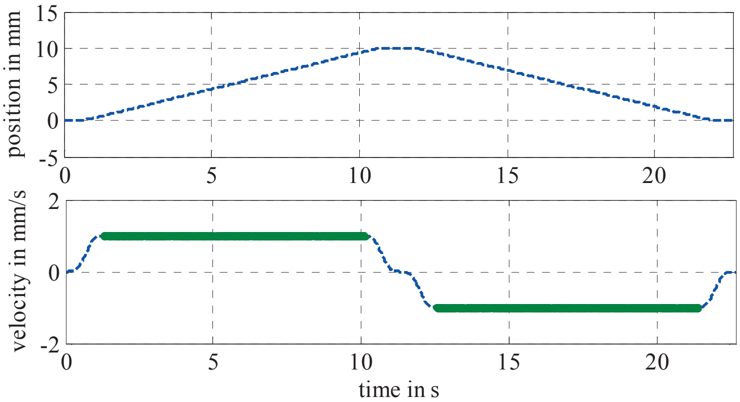

Figure 7 Position and velocity signals for a two way experiment (distance 10 mm, max. velocity 1 mm/s, highlighted: v=const)

Velocity in mm/s	Distance in mm
0.001	0.01
0.005	0.05
0.01	0.1
0.05	0.5
0.1	1
0.5	5
1	10
5	50
10	100

Table 2 Overview of all distances and velocities

The tracking error is given as the lateral deviation from the ideal straight line i.e. the control deviation perpendicular to the direction of movement. As quantitative measure the root mean square error (RSME) is used. The error e_i is defined as the difference between the required and the measured position. At this point the orthogonal deviation from the direction of movement is considered. Consequently, the required position always equals zero and e_i is given as the measured value in the stationary axis at each time step.

Besides keeping the slider on the required path it is also essential to stick exactly to the required velocity. Thus, the quantitative measure for the velocity error is calculated similarly.

$$RMSE = \sqrt{\frac{\sum e_i^2}{N-1}} \qquad (6) \qquad\qquad RMSEv = \sqrt{\frac{\sum e_{v,i}^2}{N-1}} \qquad (7)$$

The velocity error $e_{v,i}$ is defined as the difference between the required and the measured velocity at each time step. The required velocity originates from the trajectory plan while the actual velocity is calculated from two consecutive position readings. A first order lowpass filter with 500 Hz corner frequency is used for signal noise reduction. Only the path sections with constant velocity are taken into account for the determination of the tracking error.

4.2 Experimental results

Figure 8 and Figure 9 show the experimental results of the tests at the two-axis demonstrator with ball bearing guides. Up to a velocity of 1 mm/s the lateral tracking error RMSE is well below 1 nm. But at higher speeds the tracking error significantly increases. A similar characteristic can be observed concerning the velocity error. Up to 1 mm/s the velocity error RMSEv can be kept below 1 µm/s. At 5 mm/s it steps up to 5 µm/s and it stays increasing at higher velocities.

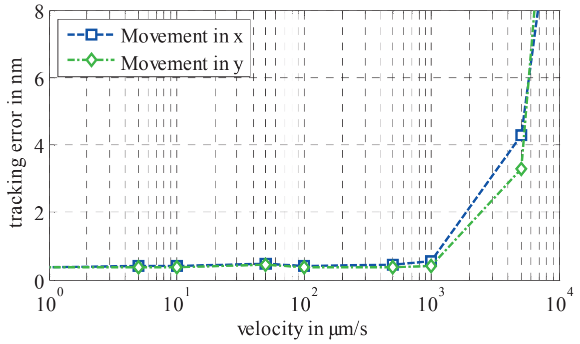

Figure 8 Lateral tracking error, system with ball bearings

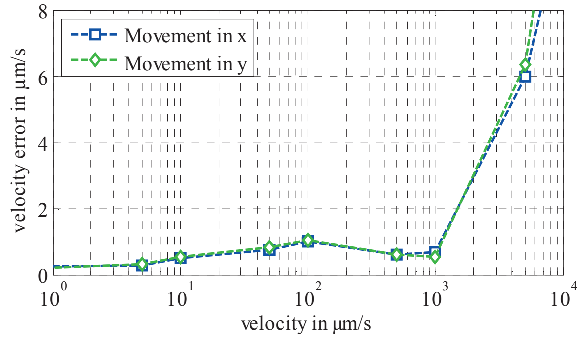

Figure 9 Velocity error, system with ball bearings

The control system is able to compensate for almost all disturbances including the friction forces up to a scanning speed of 1 mm/s. This is possible because of the high quality of the estimated bristle deflection by the EKF. Further investigations [12] have shown a significant influence of the sampling rate on the performance of the Kalman filter. In case of a constant time sampling, an increasing velocity corresponds with a decreasing regional sampling. The same effect occurs with constant velocity and a decreasing time sampling. At 10 kHz sampling rate a velocity of 1 mm/s leads to a regional step size of 100 nm, which means that only every 100 nm the driving force is updated by the control system. This limitation is a possible explanation for higher tracking errors at higher velocities. In this case a sufficient compensation of nonlinear disturbances is not possible anymore.

Figure 10 and Figure 11 show the results in case of the air guided positioning system. Basically we observed no significant difference between the movements in x- or y-direction. The lateral tracking error RSME in the tested velocity range lies below 1 nm and no variation was observed at higher scanning speeds. The results for the velocity error are likewise and also at higher speed a velocity error of less than 1 µm/s is achievable. However the temporary raise of RSMEv at 50 µm/s and 100 µm/s is remarkable, together with the subsequent decline at higher speeds. This comes from the amplitude- and offset errors in the interferometer signal demodulation and the impact of the 500 Hz lowpass filter. Overall the extremely low values for the tracking error and the velocity error stem from the almost complete absence of disturbances and the high bandwidth of the system. This, in turn evolves from the characteristics of the mechanical setup namely the aerostatic guiding of the slider and the high stiffness in the actuation chain.

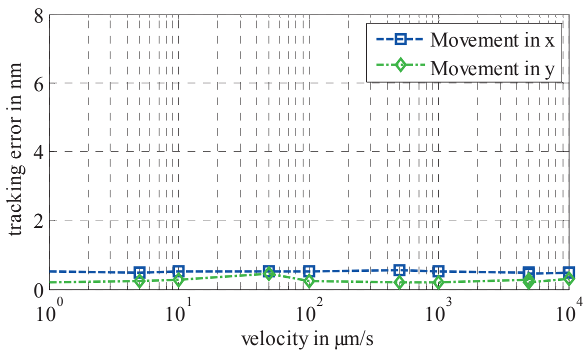

Figure 10 Lateral tracking error, system with air bearings

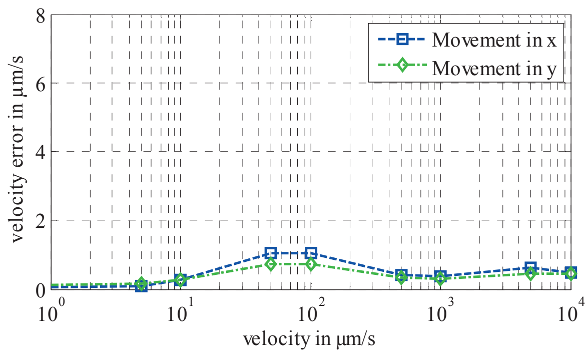

Figure 11 Velocity error, system with air bearings

5. CONCLUSION

The results of these investigations show that both systems can provide scanning motions with nanometer precision. For scanning speeds up to 1 mm/s both systems achieve tracking errors of less than 1 nm together with velocity errors of less than 1 µm/s. Significant differences can be seen at velocities from 1 mm/s on. The air guided positioning system shows excellent dynamic behavior which arises from the elimination of disturbances (especially friction effects), the high system stiffness and bandwidth and a corresponding control design. At the roller guided system on the other hand a sophisticated friction modeling and the implementation of the friction model in the advanced control algorithm provide the basis for the superb scanning accuracy.

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