Integration of Control Design and System Operation of a High Performance Piezo-actuated Fast Steering Mirror

Ernst Csencsics¹, Benjamin Sitz² and Georg Schitter¹, Senior Member, IEEE

Abstract-This paper presents a novel piezo-actuated fast steering mirror (FSM) and focuses on the integration of control design and system operation in order to improve its tracking performance for high speed scanning. The FSM is centered around a membrane-like flexure design and two pairs of stack actuators operated in a push-pull configuration, employing an optical sensor for position measurement. With the flexure membrane the first fundamental resonance mode is placed as high as 6.7 kHz, while still enabling an angular range of \pm 2.4 mrad mechanical. This results in the highest performance metric given by the product of mechanical range times main resonance mode frequency reported for piezo FSMs so far. To improve the tracking performance under high speed scanning compared to a conventional raster scan operation, an integrated control and trajectory design yields a Lissajous trajectory together with tailored single tone (ST) and dual tone (DT) feedback controllers. It is demonstrated that in the Lissajous scan case the rms tracking error can be reduced by one order of magnitude, as compared to the conventional raster scan case with a PI+ controller providing a closed-loop bandwidth of 2.7 kHz.

Index Terms—Fast steering mirror, System analysis and design, Optimal control, Motion control, Lissajous trajectory

I. INTRODUCTION

Fast steering mirrors (FSMs) [1] or tip/tilt mirror systems are opto-mechatronic devices enabling tip and tilt motion of a mirror for optical scanning and pointing applications in various scientific and commercial systems. The first class of pointing applications includes tracking of objects and acquisition of optical signals [2], adaptive optics in telescopes [3], image stabilization in stare imaging systems [4] and beam stabilization in optical systems [5]. The second application class requiring good scanning properties includes scanning optical lithography [6], confocal microscopy [7], material processing [1] and scanning laser sensors [8].

FSM systems are mostly actuated by either voice coil [9] or piezoelectric actuators [10]. Due to their comparably large size, reluctance actuators are only rarely employed [11]. Voice coil actuators are typically employed when the system requirements include a large scan range and a moderate bandwidth, such as ± 87 mrad and 350 Hz [12], and larger dimensions of the mirror system are acceptable. Piezo-actuated FSM systems are mostly comparably compact, taking advantage of the small

actuator size and enable higher system bandwidths and scan speeds at a rather small scan range, e.g. ± 1.6 mrad and 5.3 kHz [13], making them well suited for the integration into high speed scanning systems [14]. The rather complex flexures for suspending the mover and pre-stressing the actuators [15], as well as mechanical amplification structures [16], [17] and specific suspension designs for space-qualified FSMs [18] are often highlighted key components in the reported piezo FSM systems. They typically require a considerable design effort, are most commonly EDM (electrical discharge machining) cut and in general rather challenging to optimize and manufacture [19].

In contrast to MEMS-mirrors with comparably small apertures, which are also used in several scanning applications and are reported with advanced open-loop [20] as well as feedback control schemes [21], FSMs are, independent of the actuation principle and the application class, typically reported with feedback controllers [5]. These controllers enable a high control bandwidth for good tracking as well as for good rejection of external disturbances. Considering scanning systems, these controllers are suitable for the conventionally used raster trajectories. Recently, Lissajous trajectories together with tailored dual tone (DT) feedback controllers have been shown [22] as alternative to raster trajectories for a scanning voice coil actuated low stiffness FSM, enabling a reduction of the tracking error by one order of magnitude. A similar approach using a single tone (ST) controller, which can, however, not compensate for inter-axis crosstalk, has been reported for tracking Lissajous trajectories on an atomic force microscope (AFM) [23]. An open question is the improvement in tracking performance when applying DT controllers with Lissajous trajectories to high stiffness FSM systems.

This paper presents the design and integration of a high performance piezo-actuated FSM and its tailored control system, built for high speed scanning operation and resulting in one of the fastest piezo FSMs, with the highest product of mechanical range times main resonance mode frequency (32.2 rad·Hz), reported so far. The main contributions of this work include (a) the mechanical FSM structure and particularly the flexure design and analysis for maximizing the performance metric of mechanical range times main resonance mode frequency, (b) the integration of DT controllers with high stiffness piezo actuated FSMs, tailored for tracking a desired Lissajous trajectory and (c) the performance comparison to the PI+ controlled raster scanning FSM [24], demonstrating the achievable improvement. The FSM is actuated by two

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¹E. Csencsics and G. Schitter are with the Christian Doppler Laboratory for Precision Engineering for Automated In-Line Metrology at the Automation and Control Institute (ACIN), Vienna University of Technology, 1040 Vienna, Austria. Corresponding author: csencsics@acin.tuwien.ac.at. ²B. Sitz is with the Automation and Control Institute (ACIN), Vienna University of Technology, 1040 Vienna, Austria.

pairs of piezo actuators in a push-pull configuration and employs an optical sensor system for position measurement. The mechanical FSM design is centered around a membranelike flexure, which is built to maximize the product of range times main resonance mode, simplifies the mechanical design challenges and is comparably easy to manufacture. To enable a high speed and high precision scanning motion an integrated design of feedback control and scan trajectory is done to obtain ST and DT feedback controllers, which are tailored to the targeted Lissajous trajectory and the system dynamics of the high stiffness FSM. The tracking performance is compared to the FSM controlled by a PI+ feedback controller tracking a raster trajectory with the same spatial and temporal resolution [24]. Section II describes the system design and elaborates on the various system components and design choices. The system identification in Section III is followed by the modelbased and trajectory-tailored controller design in Section IV. Section V evaluates and compares the system performance for the two scan cases, while Section VI concludes the paper.

II. FSM System Design

A. System Overview

Figure 1 shows an overview of the designed FSM system. To actuate the mover in both rotational directions the FSM is equipped with two pairs of stack actuators aligned along the two system axes and operated in a push-pull configuration. The mover with the mirror on its top side is suspended by a membrane-like flexure, which is mounted to the top of the system body and also provides the pre-stressing of the piezo stacks, required for dynamic operation. The actuators are pushing against specific contact points at the backside of the mover and are mounted to a piezo mounting plate on the system base. For reflecting a laser beam of the optical sensor, entering the system through the base plate and propagating through its center, the mover carries an additional reflective element on its backside.

The mover is designed to be sufficiently stiff for shifting structural modes of the moving parts to high frequencies and avoid deformations of the mirror surface at dynamic operation. It results to a solid \emptyset 5.5 x 20 mm aluminum cylinder, which is bonded to the backside of the flexure. Rotational motion of the mover in tip and tilt direction would cause shear and tensile forces on the piezo stacks, which have to be avoided as they could cause them to crack. A force interface between the rotary mover and each actuator stack, which is not able to transmit shear and tensile forces is implemented by steel balls bonded to the bottom of the mover and providing a single contact point to the steel plates bonded to the top of each actuator.

For actuation four PICMA stack actuators (Model P-885.91, Physik Instrumente GmbH & Co. KG, Germany) with a length of 36 mm and a maximum stroke of 38 μ m \pm 10% are used and positioned in a distance of 6 mm from the system center. To obtain a pure rotational motion of the mover, each actuator pair is operated in a push-pull configuration. The actuators are already elongated by half their range when the mirror is in the zero position, such that bi-directional motion of each



Fig. 1. Exploded view of the piezo FSM showing the system components. The mirror and the mover are attached to the suspending flexure membrane and are actuated with two stack actuators per axis. A cavity in the center of the FSM serves as path for the sensor laser beam, which is reflected by a small mirror on the backside of the mover.

actuator around the zero position [25] and thus a bi-directional rotational movement of the mirror is enabled. At the maximum rotational displacement of the mover one actuator thus has its minimal and the respectively other actuator has its maximum elongation. Piezo stack actuators are only able to exert pushing forces on a target, while pulling forces would damage them, such that they have to be pre-stressed by the flexure to enable dynamic operation and maintain contact to the mover when retracting. The actuator stacks are mounted to a thin piezo mounting plate (cf. Fig. 1), attached to the base plate, for restricting lateral movement. Vertical adjustment of the actuators for fine alignment with respect to the flexure is enabled by means of set screws in the base plate.

B. Suspension Membrane

A metallic membrane is used for the suspension, carrying the mirror and the mover centrically bonded to its front and backside, respectively. It connects the static and the moving components of the system and is mounted to the body structure of the FSM by two clamping rings. The flexure membrane is also used for pre-stressing the piezo actuators in order to enable dynamic operation. The range of the actuators in the pre-stressed case is decreased when the stiffness of the flexures is increased (e.g. by increasing the thickness) according to [26]

$$\Delta L_A \approx L_0 \left(1 - \frac{k_A}{k_A + k_S} \right),\tag{1}$$

with k_A the stiffness of the actuator, k_S the stiffness of the flexure, and L_0 being the range of the stack actuator in the stress-free case. A stiffer flexure design on the other also increases the frequency of its structural modes, which is clearly beneficial from a controls perspective. As a result there are two competing aspects for the flexure design: (i) the flexure needs to be sufficiently stiff for pushing the main resonance mode

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to a high frequency and (ii) the flexure has to be compliant enough to not significantly diminish the range of the piezo stacks.

For the flexure design a simulation based parameter study with several available materials and thickness configurations was conducted to investigate the tradeoff between range loss and the main resonance mode. To enable a large system bandwidth a main resonance mode higher than 6 kHz was targeted, while the range loss due to the pre-stressing should not exceed 5%. With values of $k_A = 25 \text{ N}/\mu\text{m}$ for the stiffness and L_0 = 38 μ m \pm 10% for the maximum range of the PICMA stack actuators this results to a maximum stiffness of 1.25 N/ μ m according to (1). To determine the stiffness k_S of each flexure configuration a static mechanical analysis is performed in ANSYS Workbench (Ansys Inc., PA, USA). For obtaining the main resonance mode of each configuration, which is essentially the tilt mode, a modal analysis, which reveals the structural modes of each flexure, is further conducted.

Among the various combinations a 0.8 mm brass flexure provided the best trade-off between range loss and main resonance mode, yielding a stiffness of $k_S = 1.27 \text{ N}/\mu\text{m}$ (5.1% range loss according to (1)) and a main resonance mode at 6.5 kHz. The first three modes of the obtained flexure are shown in Fig. 2. A piston mode appears at 3.8 kHz, the main resonance mode (tip/tilt motion) at 6.5 kHz and a first non-desired structural mode at 28.4 kHz. Comparing the results of e.g. a tested steel flexure to the obtained 0.8 mm brass flexure, shows for example a 3.5 times larger range loss at an only 2.4 times higher first resonance frequency.



Fig. 2. Modal analysis of the flexure membrane. The first three modes are shown with the first being the piston mode, the second being the desired main resonance (tilt) mode and the third being the first undesired higher structural mode.

C. Position Sensor

For closed-loop control of the FSM the mirror position needs to be measured in the two rotational degrees of freedom (DoF). To obtain a direct measure of the actual mirror angle, instead of measuring the strain of the piezo actuators e.g. by means of strain gauges [15], an optical position sensor is employed. In the FSM prototype the optical position sensor is located behind the mirror unit and comprises a laser source, a beam splitter and a position sensitive device (PSD; S5990-01, Hamamatsu Photonics, Japan) for detecting displacements in x- and y-direction (see Fig. 3). The laser beam from the source is redirected towards the back of the mirror unit via a beam splitter. The mirror unit has a cavity in the center, such that the beam can propagate to the backside of the mover where



Fig. 3. Cross section of FSM with optical position sensor. Via the beam splitter the laser beam is redirected to the sensor mirror on the back of the mover and reflected back to the PSD. The PSD measures the laser spot position in both lateral directions to determine the tip/tilt position.

an additional small sensor mirror is mounted, which reflects the laser back to the beam splitter. The beam passes the beam splitter again onto a PSD that, together with custom made readout electronics, measures the position of the laser beam in two dimensions. When the mover is tilted the beam on the detector is displaced enabling a measurement of the rotational position around the x- and y-axis. The PSD has a linear lateral detector range of about 3.5 mm. To maximize the signal-tonoise ratio the entire linear detector range should be traversed when the mirror is displaced with the maximum angular range, such that the PSD board is placed in a distance of 260 mm behind the mirror mount. The entire sensor is placed into an opaque housing with a small aperture for the incoming laser beam in order to minimize the influence of ambient light on the measurement. To reduce the sensor noise level a first order lowpass filter with a bandwidth of 22 kHz is inserted into each readout channel after the detector.

D. System Setup

In Fig. 4a the experimental setup is depicted. It shows the entire FSM with the mirror mounting plate and the PSD board together with the custom made readout electronics and the components of the optical position sensor. A front view of the system with the 10 mm mirror bonded to the center of the flexure membrane is shown in Fig. 4b. The backside of the flexure with the mounted mover is depicted in Fig. 4c, showing also the sensor mirror for the optical position sensor, which reflects the incoming laser beam, as well as the 1 mm steel balls of the force interface. The piezo actuators are driven by a piezo amplifier (Techproject EMC GmbH, Austria; output voltage 0-150V) with a measured bandwidth of 10 kHz and a second order lowpass characteristic at higher frequencies and are operated in a push-pull configuration [27]. The comparably low amplifier bandwidth is deliberately chosen such that it can be directly used to significantly reduce the excitation of higher modes of the mechanical structure (cf. Fig. 2). A rapid prototyping system (Type: DS1202, dSPACE GmbH, Germany) is used for acquiring the position signals, implementing the controllers and for deriving the input signals for the piezo amplifier. For calibration of the optical position sensor an external reference measurement system (Confocal sensor confocalDT 2451, Micro-Epsilon Messtechnik GmbH

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& Co. KG, Germany) is used. The external sensor measures the mirror position from the front side and uses a small angle approximation to obtain the angular displacement.

III. SYSTEM IDENTIFICATION

For identifying the dynamics of the FSM prototype a system analyzer (3562A, Hewlett-Packard, Palo Alto, CA, USA) is used. The inputs of the piezo amplifier and the output signals of the optical position sensor are considered as system inputs and outputs, respectively.

The measured frequency response data of the x- (blue) and y-axis (red) of the system are shown in Fig. 5. Both system axes show comparable dynamics with slight deviations, most likely resulting from mounting tolerances. The first fundamental resonance mode of the system occurs at 6.7 kHz, showing good agreement with the simulation values from the flexure design in Section II-B. At 3 kHz and 3.6 kHz two more modes can be observed below the main resonance mode. While the second mode at 3.6 kHz can be matched to the piston mode of the modal analysis, which may be excited due to unsynchronized piezo motion caused by hysteresis or alignment issues, the first mode at 3.3 kHz does not show up in the modal analysis but might be explained by the finite stiffness of the upright supporting structure [25] The system shows fourth order lowpass characteristics above the main resonance mode at 6.7 kHz. This is caused by the piezo amplifier bandwidth and reduces the excitation of higher system modes. To model the frequency response of both system axes the 8th order plant model

$$G(s) = K \cdot \frac{\left(\prod_{i=1}^{2} s^{2} + 2\zeta_{z_{i}}\omega_{z_{i}}s + \omega_{z_{i}}^{2}\right)}{\left(\prod_{i=1}^{4} s^{2} + 2\zeta_{p_{i}}\omega_{p_{i}}s + \omega_{p_{i}}^{2}\right)} \cdot e^{-sT}, \quad (2)$$

with gain K = 5.82e16, time delay $T = 16.7 \ \mu s$ (sampling delay of the rapid prototyping system) and parameters according to Table I is employed. The gain of the system axes

TABLE I FSM system dynamics model coefficients.

Index	ω_{Index} [rad/s]	ζ_{Index}
z_1	2.05e4	0.03
z_2	2.41e4	0.02
p_1	1.93e4	0.05
p_2	2.29e4	0.02
p_3	4.26e4	0.05
p_4	5.03e4	0.44

at DC is more than 30 dB larger than the crosstalk between the system axes, which is revealed by crosstalk measurements (data not shown). This justifies the application of one singleinput-single-output (SISO) controller per axis.

To measure the mechanical range and hysteresis of the system a 1 Hz sinusoidal signal with 10 V amplitude is applied to the to the piezo amplifier input of one system axis at a time. The measurement results of both axes are shown in Fig. 6, demonstrating a mechanical range of ± 2.4 mrad in tip and tilt direction. The maximum hysteresis is about 15% for both system axes. With the obtained main resonance mode at 6.7 kHz and the mechanical range of ± 2.4 mrad the product of mechanical range times main resonance mode frequency, as performance measure for the system design, results to 32.2 rad·Hz.

IV. CONTROLLER AND TRAJECTORY DESIGN

Given the obtained system dynamics with the main resonance mode at 6.7 kHz a PI controller is designed with the aim of maximizing the system bandwidth for tracking of a high speed raster trajectory [24]. Based on the raster trajectory a Lissajous trajectory with equal spatial and temporal resolution is designed together with integrated ST and DT controllers.

A. PI Controller with Notch Filters (PI+ Controller)

As most commercial piezo actuated FSMs use feedback controllers for position control of the mirror, a PI based controller design is taken as benchmark for both axes. This approach is reasonable as proportional-integral (PI) controllers are also applied in many other piezo actuated high stiffness systems, such as AFMs [25], with a crossover frequency placed below the main resonance mode of the system. As given in [24] the PI+ controller

$$C(s) = \underbrace{K_{PI} \cdot \frac{(s + \omega_4)}{s}}_{PI \ controller} \cdot \underbrace{\left(\prod_{i=1}^3 s^2 + 2d_n w_n \omega_n s + \omega_n^2\right)}_{Notch \ filters} \quad (3)$$

with $K_{PI} = 50$, three notch filters and parameters according to Table II is obtained in a loop shaping approach, resulting in a crossover frequency of 1.4 kHz and a predicted bandwidth of 3 kHz. As the dynamics of both system axes are similar, the same controller can be applied to both system axes.

TABLE II
COEFFICIENTS OF THE DESIGNED PI+ CONTROLLER.

Index	ω_{Index} [rad/s]	d_{Index}	w_{Index}
1	1.94e4	0.5	0.2
2	2.29e4	0.5	0.1
3	4.26e4	0.025	2
4	1e4	-	-

B. Trajectory Design

A raster trajectory with high spatial and temporal resolution, which has its first 11 harmonics covered by the closedloop bandwidth, is obtained by selecting $f_{1R} = 250$ Hz and $f_{2R} = 0.5$ Hz for the fast and slow scan axis, respectively. This trajectory has a spatial resolution of 4 μ rad/mrad and a frame rate of 1 frame/s. For a fair comparison, a Lissajous trajectory with the same spatial and temporal resolution is obtained by selecting $f_{1L} = 386$ Hz and $f_{2L} = 397$ Hz for the x- and y-axis, respectively [23].

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Fig. 4. Experimental FSM setup. (a) shows an overview of the experimental setup with the FSM, the mounting plate and the optical position sensor with laser source, beam splitter and PSD board. (b) shows a front view of the FSM, with the mirror, the flexure membrane and the BNC connectors for the actuators observable. (c) shows the backside of the mover with the mounted mover, the steel balls and the sensor mirror.



Fig. 5. Frequency response of the FSM x- (solid blue) and y-axis (solid red) measured with the optical position sensor. The fitted system model (dashed black) models the first three system resonances at 3, 3.6 and 6.7 kHz.



Fig. 6. Measured hysteresis for x- (solid blue) and y-axis (dashed red). The mechanical system range is ± 2.4 mrad with a maximum hysteresis of 15% for both axes.

C. Dual Tone and Single Tone Controller

To design DT controllers [22] for tracking the desired Lissajous trajectory, an H_{∞} -approach is used for systematically considering the properties of the target trajectory in the controller design. The approach shapes the controller by minimizing the H_{∞} -norm of the system model, extended by weighting functions for the sensitivity function S(s) = 1/[1 + C(s)P(s)] and the input sensitivity function U(s) = P(s)/[1 + C(s)P(s)]). The weighting function

$$W_S(s) = \prod_{i=1}^{2} \frac{s^2 + 2\omega_{r_i}\zeta s + \omega_{r_i}^2}{s^2 + 2\omega_{r_i}\frac{\zeta}{a}s + \omega_{r_i}^2}$$
(4)

represents the requirements on S(s) and is composed of two inverse notch filters. The notches enforce good tracking at both drive frequencies and reduce the control effort at all other frequencies, by rolling off steeply before and after. The inverse notch frequencies $\omega_{r_1} = 2.425e3$ rad/s and $\omega_{r_2} = 2.494e3$ rad/s are derived from the drive frequencies f_{1L} and f_{2L} , respectively. Height and width of the inverse notches are tuned to $\zeta = 1e-4$ and d = 90. In order to limit the controller effort at low and high frequencies, the weighting function

$$W_U(s) = 8e3 \cdot \prod_{i=1}^2 \frac{s + \omega_{z_i}}{s + \omega_{p_i}},\tag{5}$$

with ω_{z_1} = 2.2e3 rad/s, ω_{z_2} = 2.63e3 rad/s, ω_{p_1} = 0.063 rad/s, and ω_{p_2} = 2.2e7 rad/s is used for U(s). Using the derived system model from Section III and the weighting functions the dual tone controller

$$C_{DT}(s) = K_{DT} \cdot \frac{\left(s^2 + 2\zeta_{z_1}\omega_{z_1}s + \omega_{z_1}^2\right) \left(\prod_{i=2}^4 s + \omega_{z_i}\right)}{\left(\prod_{i=1}^3 s^2 + 2\zeta_{p_i}\omega_{p_i}s + \omega_{p_i}^2\right)},$$
(6)

with $K_{DT} = 50$ and parameters according to Table III, is obtained after a controller order reduction. As the system axes have similar dynamics and the controller is tuned for both drive frequencies, the same controller can be applied for both system axes.

To enable a comparison between DT controllers and ST controllers [23], that are solely tuned to the drive frequency of the respective axis, two ST controllers are additionally

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Index

 z_1

 z_2

 z_3

 z_4

 p_1

 p_2

 p_3

designed. The adapted weighting functions

100 Magnitude [dB] 50 0 -50 10 10² 10³ 10⁴ 100 Phase [°] 0 PI+ controlle DT controller ST controller x-axi -100 - ST controller y-ax

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 $W_{S1}(s) = \frac{s^2 + 2\omega_{r1}\zeta s + \omega_{r1}^2}{s^2 + 2\omega_{r1}\frac{\zeta}{\delta}s + \omega_{r1}^2}$ $W_{S2}(s) = \frac{s^2 + 2\omega_{r2}\zeta s + \omega_{r2}^2}{s^2 + 2\omega_{r2}\frac{\zeta}{\delta}s + \omega_{r2}^2}$ (8)

 ζ_{Index}

4e-3

0.8

1e-4

1e-4

(7)

with single inverse notch filters at ω_{r1} and ω_{r2} are used together with $W_U(s)$ (see (5)) for the ST controller design for the x- and y-axis, respectively.

TABLE III COEFFICIENTS OF THE DESIGNED DT CONTROLLER

 ω_{Index} [rad/s]

2.46e3

0.059

0.062

181

27.6

2.425e3

2.494e3

D. Controller Implementation

For implementation on the rapid prototyping system the controllers are discretized using Pole-Zero-Matching [28] for a sampling frequency of $f_s = 60$ kHz. The poles and zeros are directly transformed to the discrete time domain by using the relation $z = e^{s/f_s}$ to guarantee that the controller peaks are located exactly at the drive frequencies.

Figure 7 shows the measured frequency responses of the implemented controllers. The integrating behavior at low frequencies and the three notch filters of the PI+ controller between 3 and 8 kHz are clearly observable. The controller gain of the DT controller is significantly below the gain of the PI+ controller over the entire frequency range, except at the drive frequencies f_{1L} and f_{2L} , where it shows sharp 50 dB gain peaks. The ST controllers for x- and y-axis show a single sharp 50 dB gain peak at the axis-related drive frequency f_{1L} and f_{2L} , respectively, while the controller gain is already around 0 dB at the respectively other drive frequency. At low and high frequencies the ST controllers show slightly lower gains than the DT controllers.

V. EVALUATION OF SYSTEM PERFORMANCE

For evaluation of the system performance the closed-loop dynamics, the tracking performance and the positioning uncertainty are investigated.

The measured complimentary sensitivity functions of the yaxis of the FSM with the PI+, the DT and the ST controllers are shown in Fig. 8. The system with the PI+ controller achieves a control bandwidth of 2.7 kHz with a small gain peaking of 2.5 dB at 1.9 kHz, which is due to the large stability margins. The frequency response of the system with DT controllers reaches the 0 dB line exactly at both drive frequencies f_{1L} and f_{2L} , while staying well below -40 dB

Fig. 7. Measured frequency response of the implemented PI+ (dotted blue), DT (solid red) and ST controllers (dashed green, dash-dotted black). The PI+ controller is designed for a crossover frequency of 1.4 kHz. The DT controller shows localized high control effort at both drive frequencies of 386 and 397 Hz. The ST controllers of each axis show high controller gain only at the related drive frequency

Frequency [Hz]

10³

10⁴

 10^{2}

at high and low frequencies. At the drive frequencies the phase is exactly at -360°, indicating a perfect phase match for tracking the sinusoidal motion. The frequency response with the ST controller reaches the 0 dB line only at the related drive frequency ($f_{L2} = 397$ Hz).



Fig. 8. Measured complimentary sensitivity function of the FSM y-axis with the designed controllers. The control bandwidth of the PI+ controlled FSM (dotted blue) is 2.7 kHz. With the DT controller (solid red) the frequency response reaches the 0 dB line exactly at both drive frequencies. With the ST controller (dashed green) the frequency response reaches the 0 dB at 397 Hz.

To compare the tracking performance, the desired raster and

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Lissajous trajectories are applied with a scan amplitude of 25% of the full scale range (0.6 mrad) and tracked with the related controllers. The tracking performance is evaluated in terms of the rms tracking error and the rms controller output with the results given in Table IV. All trajectory-controller-combinations result in comparable values of the controller output but the rms tracking error from both Lissajous cases with the ST and the DT controller is a factor 3.6 (18 μ rad rms) and a factor of 11.2 (5.8 μ rad rms) smaller as compared to the raster case (64.9 μ rad rms). This clearly shows the improvement caused by the localization of the control effort around the drive frequencies, entailing a better phase match, in contrast to the widely distributed control action of the PI+ controller.

TABLE IV Comparison of tracking performance

Trajectory /	Frequency	RMS Error	Ctrl. Output
Controller	[Hz]	[%]	$[V_{rms}]$
Raster / PI+	0.5/250	10.8	71.9
Lissajous / ST	386/397	3	74.3
Lissajous / DT	386/397	1	74.1

The tracking error when using the DT controller, as shown in Fig. 9a, is another factor 3.1 smaller than for the ST controlled case, which can be explained by the error spectra. Figure 9b and 9c shows the power spectral density of the



Fig. 9. Tracking error of the FSM x-axis in time and frequency domain. (a) depicts the tracking response in the time domain with both axes being controlled by the DT controller. (b) shows the case of both axes being controlled with the ST controller. A large error component at 397 Hz (yaxis drive frequency) and a beat component at 11 Hz are observable. (b) shows the case of both axes being controlled with the DT controller. A small error component at 386 Hz (x-axis drive frequency) is observable.

measured error signal of the x-axis (386 Hz), while the FSM

is tracking the desired Lissajous trajectory using ST and DT controllers, respectively. In the ST controller case a large component of the tracking error at the drive frequency of the y-axis (397 Hz), resulting from inter-axis crosstalk, as well as a resulting beat component at 11 Hz can be obtained. The DT controller has high control gains at both drive frequencies and is thus able to also compensate the crosstalk component.

The resulting positioning uncertainty of the FSM controlled by the PI+ controller is determined at zero reference applied to both axes. Adding the lowpass filter, mentioned in Section II-C, at the position sensor output, the noise level can be reduced down to 3.8 μ rad rms, which is 0.08% of the full scale range. Dual axis operation for the DT control case is demonstrated in Fig. 10 which shows a subset (10% of the entire trajectory) of the desired Lissajous trajectory with a scan amplitude of 2 mrad. The measured positions show good accordance with the reference trajectory.



Fig. 10. Measured Lissajous scan trajectory (solid red) with DT controllers applied to both axis. A 10% subset of the entire trajectory is shown together with the related reference trajectory (dashed blue).

In summary it is shown that the proposed piezo actuated FSM with the flexure membrane results in a range of ± 2.4 mrad and a first fundamental resonance mode at 6.7 kHz and that an integrated design of Lissajous trajectory and DT controller enables a significant performance improvement by reducing the tracking error by one order of magnitude as compared to the conventional operation.

VI. CONCLUSION

This paper presents a novel piezo-actuated FSM design with a membrane-like metallic flexure and an optical position sensor, which relies on two pairs of stack actuators operated in a push-pull configuration. The FSM provides an angular range of ± 2.4 mrad mechanical, while achieving a first fundamental resonance mode of the actuator as high as 6.7 kHz. This results in a product of mechanical range times main resonance mode frequency of 32.2 rad-Hz, which is, according to a thorough literature and product review, the highest value reported so far. By employing an integrated design of control system

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and trajectory, a Lissajous trajectory with equal spatial and temporal resolution as a 250 Hz raster reference trajectory is designed together with tailored ST and DT controllers to improve the tracking performance compared to the conventional raster scan case. It is demonstrated that the rms tracking error can be reduced from 10.8% in the raster case with a PI+ controller, having a bandwidth of 2.7 kHz, to 1% in the Lissajous case with the tailored DT controller and that both system configurations result in comparable controller outputs. Future work is concerned with the integration of the optical position sensor into the mirror unit and adapting the prototype to reduce system dimensions for obtaining a more compact FSM design at the same performance level.

ACKNOWLEDGMENT

The financial support by the Christian Doppler Research Association, the Austrian Federal Ministry for Digital and Economic Affairs, and the National Foundation for Research, Technology and Development, as well as MICRO-EPSILON MESSTECHNIK GmbH & Co. KG and ATENSOR Engineering and Technology Systems GmbH is gratefully acknowledged.

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Ernst Csencsics is a postdoctoral researcher at the Automation and Control Institute (ACIN) at Vienna University of Technology. He received a MSc. and PhD degree in Electrical Engineering from TU Vienna, Austria in 2014 and 2017, respectively. His primary research interests are on high performance mechatronic systems design, control of opto-mechatronic systems, precision engineering, and flexible in-line metrology systems.

Post-print version of the article: E. Csencsics, B. Sitz and G. Schitter, "Integration of Control Design and System Operation of a High Performance Piezo-actuated Fast Steering Mirror," *IEEE Transactions on Mechatronics*, vol. 25, no. 1, p. 239-247, 2020. DOI: 10.1109/TMECH.2019.2959087

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Benjamin Sitz was a diploma student at Automation and Control Institute (ACIN) at Vienna University of Technology. He received his MSc. degree in Electrical Engineering fromTU Vienna, Austria in 2016. He is currently working on control systems for manufacturing machines at an industrial company.



Georg Schitter is Professor at the Automation and Control Institute (ACIN) of the Vienna University of Technology. He received a MSc. in Electrical Engineering from TU Graz, Austria (2000) and his PhD degree from ETH Zurich, Switzerland (2004). His primary research interests are on highperformance mechatronic systems and multidisciplinary systems integration, particularly for precision engineering applications in the high-tech industry, scientific instrumentation, and mechatronic imaging systems, such as scanning probe microscopy, adapbut custome for amiconductor industry.

tive optics, and lithography systems for semiconductor industry.

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