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High Precision Hybrid Reluctance Actuator with Integrated Orientation Independent Zero Power Gravity Compensation

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Abstract-This paper presents a novel method for enabling energy-efficient and orientation-independent gravity compensation in active vibration isolation systems by taking advantage of the inherent negative stiffness of a hybrid reluctance actuator (HRA). Counteracting the gravitational force acting on the mover of an HRA with the position dependent force of the actuator's integrated permanent magnet, the gravitational force can be compensated without the need for additional actuation. An HRA-system with two translational degrees of freedom (DOF) is developed, comprising a permanent magnet for generating a constant biasing flux for both system axes and two actuator coils per axis for actively controlling the position of the mover. The system prototype has an actuation range of \pm 0.7 mm in both DOF, while enabling energy efficient gravity compensation of payloads up to 500 g by an additional current feedback control loop. Experiments demonstrate that the current consumption for compensation of a 500 g payload is reduced from 1.58 A to 10 mA, which corresponds to a reduction of the power consumption by a factor of 25000.

Index Terms—Energy efficiency, Mechatronics, Nanopositioning, Precision engineering, Electromagnetics.

I. INTRODUCTION

G RAVITY compensation is required in a variety of applications, ranging from conventional magnetic bearings [1] over vibration isolation equipment [2] to active sample tracking vibration isolation concepts used in inline measurement systems [3], [4]. There, a robotic measurement platform levitates and actuates a high precision measurement tool to compensate vibrations and maintain a constant distance between tool and sample via feedback control. Constant actuator currents for compensation of the gravitational force decrease the efficiency of the system and cause heat dissipation, which impairs the resulting positioning precision and quality of measurements [5]. This urges the need for energy efficient gravity compensation methods.

A fundamental design approach which is frequently used for gravity compensation [6]–[8], is proposed in [9]. There, a levitation-force is generated by making use of repulsive and attractive forces between the magnets of the stator and the mover. By applying current to the coils of the stator, the flux and therefore the resulting force can be manipulated. By replacing the stator magnets with Halbach-arrays [10], the resulting force can be increased by a factor of four

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[11]. It is worth noting, that this principle works only for a certain orientation of the actuator and that gravity can not be compensated, if the actuator would be aligned upside down. Another drawback of the principle is a relatively small motor constant due to voice coil actuation.

Compared to voice coil actuators, hybrid reluctance actuators (HRAs) provide higher force-to-volume ratios [12], which enable more compact actuator designs. Due to several challenges such as the nonlinear force-to-current relation, they have hardly been implemented in high precision positioning and scanning systems in the past [13], [14]. Nevertheless, several reluctance actuated systems have been developed, including fast tool servos [15], fast steering mirrors [16], [17] and nano-positioners [18]. A high force linear hybrid actuator with a range of $\pm 1 \mathrm{\,mm}$ and low stiffness has also been reported for the active vibration isolation in precision machines [19]. Next to their nonlinearity, their negative stiffness is in general a challenge for system design and is therefore often compensated by mechanical flexures [20]. With their higher motor constants and the already integrated permanent magnet, HRAs are a promising alternative to replace conventional combinations of voice coil actuators and permanent magnets in active vibration isolation systems in order to improve energy efficiency and versatility.

The contribution of this paper is (i) a novel method for energy efficient gravity compensation, employing the negative stiffness of HRAs to compensate external forces on the mover and (ii) the integration of linear ball bearings to deliberately establish different global and local system dynamics for the sake of system control. The proposed gravity compensation concept is demonstrated and evaluated by integration into a system with 2 degrees of freedom (DOFs), enabling orientation independent gravity compensation of the mover for 360° rotational motions of the entire system.

II. SYSTEM DESIGN

A. HRA for Energy Efficient Gravity Compensation

The basic configuration of an HRA with one DOF is shown in Fig. 1. It consists of a ferromagnetic yoke and moving part, a permanent magnet and a pair of coils which are wound around the yoke parts. The output force of the actuator is dependent on the flux through the working air gaps which can be controlled by the current applied to the serially connected coils. The integrated permanent magnet generates a biasing flux which linearizes the force to current ratio and enables

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Fig. 1. Principle of energy efficient gravity compensation using an HRA with one degree of freedom. (a) depicts the principle of a single-degree of freedom HRA. (b) shows the magnetic force F_m which increases with larger displacements of the mover from the center position, causing negative stiffness of the system. By placing the mover at the point where the magnetic force and the external force are in equilibrium, powerless gravity compensation is enabled (I=0).

high motor constants compared to voice coil actuators. The biasing flux is responsible for a position-dependent force F_m , pulling the mover towards the yoke, as illustrated in Fig. 1a. This force is zero if the mover is in the center position (x = 0) and increases for larger displacements of the mover, resulting in a negative stiffness behavior of the system [18]. While this negative stiffness states a challenge for control design, it can be exploited for zero power gravity compensation.

As indicated in Fig. 1a, in order to compensate the gravitational force acting on the mover, it has to be displaced towards the opposite direction of the external force until the amplitude of the magnetic force F_m equals the amplitude of the external force F_{ext} and both forces compensate each other. As a consequence, the mover can be positioned around this new energy optimal operating point, without the need for a constant current required for gravity compensation.

In order to enable orientation independent operability for rotational motions of the system, the proposed system design is based on two linear-motion HRAs (according to Fig. 1a), which are aligned perpendicular to each other. By sharing the biasing flux of a center-mounted permanent magnet for both axes, a compact system design is realized. With the intention of realizing a working range of ± 0.7 mm in both DOFs, the length of the working air gaps between mover and yoke l_g has to be larger in order to prevent the mover from snapping into the yoke. To balance the required offset force for the targeted load mass of 500 g and the maximum flux density in the yoke parts, it the working air gap length results to 1.2 mm.

B. Magnetic Circuit Analysis

For evaluation of the actuator's hybrid reluctance force, an analytical model of the magnetic circuit is derived. In the interest of simplicity, the yoke is assumed to be ideal magnetic permeable. Non-linearities such as saturation and hysteresis within the magnetic flux paths are not considered. Consequently, the derived model is only valid for flux densities smaller than the saturation flux density of the yoke material.



Fig. 2. Model of the magnetic circuit. Depending on the position of the mover, the permanent magnets' flux Φ_m gets distributed to all working air gaps where a position dependent force is generated.

The simplified magnetic circuit is shown in Fig. 2, where the reluctances of the working air gaps between mover and yoke are denoted as R_{x1} , R_{x2} , R_{y1} and R_{y2} . The reluctance of the permanent magnet and the non working air gap is modeled by R_m and R_f respectively. The reluctances of the working air gaps are dependent on the mover's position

$$R_{x1} = \frac{l_g - x}{\mu_0 A} \qquad \qquad R_{x2} = \frac{l_g + x}{\mu_0 A} \tag{1}$$

$$R_{y1} = \frac{l_g - y}{\mu_0 A} \qquad \qquad R_{y2} = \frac{l_g + y}{\mu_0 A}, \tag{2}$$

with μ_0 the magnetic permeability in vacuum, x and y the displacement of the mover from the center position, A the pole face area of the yoke and l_g the length of the air gap between mover and the yoke when the mover is in the centre position.

The magnetomotive force (MMF) of the permanent magnet is modeled by its length l_m , its coercivity field strength H_c and a flux-leakage-factor λ as $\Psi_m = H_c l_m \lambda$, while the MMF of one coil is denoted as $\Psi_{c,x} = NI_x$, where N represents the number of windings and I_x the current applied to both coils of a single axis, which are connected in series. By considering Hopkinson's law of magnetics, the biasing flux generated by the permanent magnet is calculated to

$$\Phi_m = \frac{\Psi_m}{R_m + R_f + R_p} \tag{3}$$

where R_p denotes the resulting reluctance of all four reluctances of the working air gaps connected in parallel. By utilizing the network equations of the circuit

$$\Phi_m - \Phi_{x1} - \Phi_{x2} - \Phi_{y1} - \Phi_{y2} = 0 \qquad (4)$$

$$\Phi_{x1}R_{x1} + \Psi_{c,x} - \Psi_m + \Phi_m(R_m + R_f) = 0 \qquad (5)$$

$$\Phi_{x2}R_{x2} - \Psi_{c,x} - \Psi_m + \Phi_m(R_m + R_f) = 0$$
(6)
$$\Phi_{x2}R_{x2} - \Psi_{c,x} - \Psi_m + \Phi_m(R_m + R_f) = 0$$
(7)



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(b)

0

x (mm)

0.2

0.4

0.6

-0.2

-0.6

-0.4

Fig. 3. Analytically obtained and simulated motor constant and actuator stiffness of the system x-axis in dependence of the mover displacement within the two dimensional actuation range. (a) shows the motor constant $K_{m,x}(x, y)$ and (b) the actuator stiffness $k_{a,x}(x, y)$, parametrized for various uni-directional displacements in y-direction with respect to the center position. Simulated data points are obtained at a current of 1 A and marked by 'o'.

the magnetic flux in the working air-gaps and therewith the resulting total force for a single degree of freedom (DOF) is calculated by applying Maxwell's stress tensor [20]:

$$F_x = \frac{\Phi_{x1}^2 - \Phi_{x2}^2}{2\mu_0 A} = K_{m,x}(x,y)I_x + k_{a,x}(x,y)x.$$
 (9)

As indicated in (9), the resulting force can be split into two components: one component including the motor constant $K_{m,x}(x, y)$ and a second component including the actuator stiffness $k_{a,x}(x, y)$. It is noticeable that both components are dependent on the mover's displacement in x and y direction as can also be seen in Fig. 3.

With the equations for resulting actuator force and stiffness available, the targeted actuation range of ± 0.7 mm, load mass and compact system dimensions, an iterative design process was used to obtain the parameters for the magnetic circuit design. The design started out at available magnets with high coercivity and the pole face area, followed by determining the required magnet length and air gaps to ensure manufacturability and to limit the flux density in the yoke parts to about 1 T. Finally, the number of coil turns was chosen, such that the flux generated at the maximum current of the available power electronics, matches the biasing flux in a respective axis [17]. In order to also accurately consider non-linearities, such as

TABLE I PROPERTIES OF MAGNETIC CIRCUIT.

Property	Symbol	Value
Length of working air gap	l_g	$1.2\mathrm{mm}$
Coercivity of magnet	H_c	$\approx 900 \mathrm{kA} \mathrm{m}^{-1}$
Length of magnet	l_m	$35\mathrm{mm}$
Length non working air gap	l_f	$1\mathrm{mm}$
Number of windings per coil	Ň	135
Pole face area of yoke	A	$144\mathrm{mm^2}$
Flux leakage factor	λ	0.78

hysteresis and saturation, and for estimation of the previously introduced flux-leakage-factor λ , a 3D-model of the actuator is created and simulated by using FEA-software (Maxwell, ANSYS Inc., Canonsburg, USA). The design resulted in the magnetic circuit parameters in Tab. I, a motor constant $K_{m,x}(0,0)$ of 7.8 N A^{-1} and an actuator stiffness $k_{a,x}(0,0)$ of 11.6 kN m^{-1} for the mover in the center position. By tuning the flux-leakage-factor to $\lambda = 0.78$, the analytically calculated results of $K_{m,x}(x,y)$ and $k_{a,x}(x,y)$ show good agreement with the results of the FEA-simulation (see Fig. 3).

For energy efficient gravity compensation, the maximum force solely generated by the permanent magnet is of high interest as it determines the maximum passively compensable payload. Using (9) and assuming the current I_x to be zero, the permanent magnet's force dependent on the displacement of the mover can be evaluated. This force is largest at the points of maximum displacement of the respective axis, where the distance between mover and yoke is minimized, and zero displacement of the orthogonal axis. A maximum force of $1.68 \times 10^4 \,\mathrm{N/m}$ · $7 \times 10^{-4} \,\mathrm{m} = 11.8 \,\mathrm{N}$ (evaluate curve for y = 0 mm in Fig. 3b at x = 0.7 mm) is generated, which enables to carry the weight of the mover (213g) and the targeted additional payload of 500 g without the need for additional coil currents. With the maximum coil current of 4 A, determined by the power electronics, a maximum force of 31.2 N can be generated, resulting in a maximum admissible net payload of almost 3 kg.

C. Mechanical System Design

The main purpose of the stator is guidance of magnetic flux generated by the coils and the permanent magnet. Therefore, the stator must be made out of ferromagnetic material. The sensors, which are needed for measuring the relative displacement between mover and stator, are mounted on the sides of the stators legs. Vibration of the legs may introduce structural modes to the system transfer function, hence a stiff design of the stator is mandatory. With the intention of reducing the diffusion of eddy currents and therefore the resulting phase lag in the system, a layered design of the ferromagnetic yoke is targeted [21]. In order to ensure an even distribution of the biasing flux to all legs of the actuator, the permanent magnet is placed at the intersection of the yokes of both axes.

To mechanically constrain the mover's motion to two DOFs, a guiding mechanism is required. In recent works, the mover of HRAs is guided by mechanical flexures [3], [17], [18] which enable guided motion without friction. However, the positive stiffness of the guiding flexures reduces the resulting overall



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Fig. 4. Exploded view of the system with all parts of the prototype visible. Recognizable is the perpendicular alignment of both system axes' bearings.

negative stiffness of the HRA. Linear ball bearings are subject to friction but add no positive stiffness to the global system dynamics for large stroke motions, such as for changes of the operating point. In their pre-slide phase at small velocities, however, they show spring-damper like properties [22], which can be deliberately used to obtain a local plant behavior that is open loop stable. Despite the introduced friction, they enable a positioning accuracy in the range of the sensor resolution in closed-loop control [23]. Combining ball bearings with the negative actuator stiffness thus enables a system design with a globally dominant negative stiffness, used for zero power gravity compensation, but a locally open-loop stable behavior, beneficial for open-loop identification and feedback control design.

Consequently, the mover is guided by two pairs of linear ball bearings, mounted perpendicular to each other, as can be seen in Fig. 4. The serial kinematic design – axis y is mounted on top of axis x – leads to axis x having a higher moving mass than axis y. This configuration allows movement in x- and y-direction, while rotation in any DOF as well as motion in z-direction is only possible due to play in the bearings and is thus almost completely suppressed. Unlike the parts of the statoryoke, the mover – which is mounted on top of the guiding-mechanism – is made from solid ferromagnetic material in order to ensure high rigidity and stiffness of the moving mass.

III. REALIZATION OF ACTUATOR PROTOTYPE

A. Mechanical Components

Since the applied current is equal for both coils of one axis, each pair of coils is connected in series. For a proper



Fig. 5. Photograph of the assembled prototype, mounted with the x-y plane in vertical orientation. Due to a swivel mount, the entire actuator can be rotated around the z-axis to demonstrate the systems orientation independent functionality.

tuning of the current amplifier's PI-gains, the serial-resistance $R_s=0.85\,\Omega$ as well as the serial-inductance $L_s=4.95\,\mathrm{mH}$ of both axes' coils are measured using a precision RLC-meter (E4980AL, Keysight Technologies, Santa Rosa, USA). The material chosen for the magnetic yoke is non-grain-oriented steel (1.0330) with a saturation flux density of $B_{sat}\approx1.5\,\mathrm{T}$. The used sheets have a thickness of $0.5\,\mathrm{mm}$ and are coated by a layer of spray paint for electrical isolation. In order to enable a perpendicular alignment of both axes, the yokes are partially cut-out at the point of intersection.

For rapid prototyping, all parts of the support structure are 3D-printed using the fused deposition modeling (FDM)method. Polyethylene Terephthalate Glucol-modified (PET-G) is the chosen printing material due to its good printability and high durability of the printed object.

B. Electrical Components

The displacement of the mover relative to the stator is measured in both DOF by interferometric sensors (IDS3010, attocube systems AG, Germany), which are aligned to small mirrors on the sides of the mover. The measurement resolution of both sensors is 2 nm, the sensors are operating at a sample time of 120 ns.

The system is controlled by a rapid prototyping platform (DS1005, dSPACE GmbH, Germany) which operates at a sample rate of 20 MHz and is programmed using Matlab-Simulink (The MathWorks, Inc., Nattick, USA). Custom made analog current amplifiers are employed for providing the current required for actuation. The gain of the current amplifiers is 400 mA V^{-1} . In order to neglect the dynamics of the current amplifiers at small frequencies, a -3 dB-bandwidth of the current amplifiers of 10 kHz is desired. To realize these bandwidths, the PI-gains of the controllers are adjusted with respect to the previously measured values of resistance and inductance of the coils. The resulting prototype is depicted in Fig. 5.

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IV. System Dynamics & Control Design

A. System Identification

The system dynamics are identified with the x-y plane of the actuator in a horizontal orientation. By employing a system analyzer (3562A, Hewlett-Packard, Palo Alto, USA), the frequency responses of both axes $G_x(s)$ and $G_y(s)$ from the reference input of the current amplifier to the measured positions x_m, y_m of the mover are measured. A swept-sine reference-signal with an amplitude of 100 mV and a frequency range of 1 Hz-10 kHz is applied to the reference input of the current controller. The application of the aforementioned reference signal leads to current amplitudes of 40 mA. The digital measurement signal of the interferometer is converted by the dSPACE system in order to provide an analog signal to the system analyzer.

When the mover is placed at the exact center position, the resulting force on it would be zero (see Fig. 1b). Due to the positive stiffness introduced by the ball bearings in their preslide phase, which adds to the negative stiffness of the actuator, the resulting local dynamics for small velocities are openloop stable, as discussed in Sec. II-C. This enables open-loop system identification with small signal excitation, regardless of the plant's global negative stiffness behavior.

B. Local System Dynamics

The results of the measurements of the local system dynamics are shown in Fig. 6. Due to the discussed influence of friction, they do not show the characteristics of the global negative stiffness system, such as a constant phase of -180° and the absence of a resonance peak [24], but rather the characteristics of a damped mass-spring system with overall positive stiffness. While the resonance frequency of axis x occurs at approximately 205 Hz, the resonance frequency of axis y is at approximately 305 Hz. This is explained by the fact that axis x needs to move the mass of the mover as well as the entire ball bearing of axis y, whereas axis y needs to move only the mass of the mover itself (see Fig. 4). Due to the influence of eddy currents, diffusing within the ferromagnetic yoke, a phase lag of approximately -200° at frequencies just beyond the resonance frequency is visible. The phase further drops by 360° at frequencies of approximately 500 Hz and 640 Hz for axis x and y, respectively, which is most likely due to parasitic rotational modes of the ball bearings.

By measuring the transfer function $G_{xy}(s) = \frac{y_m(s)}{u_x(s)}$ and $G_{yx}(s) = \frac{x_m(s)}{u_y(s)}$ of both axes, the crosstalk between both axes is evaluated. Due to the serial kinematic design, the crosstalk $G_{xy}(s)$ from axis x to axis y is significantly larger than vice versa. However, due to a magnitude margin $(G_x(s) - G_{yx}(s), G_y(s) - G_{xy}(s))$ of 60 dB and 34 dB at small frequencies, the crosstalk between both system axes is negligible (data not shown). This enables the use of a single-input single-output (SISO) design approach, with each axis being controlled by an individual feedback controller.

C. Position Control

As the system targets a quasi-static operation for zero power gravity compensation of the mover with no or only slow



Fig. 6. Measured frequency responses of both axes' plants and complementary sensitivity functions of both system axes' position control loop. Due to friction of the linear ball bearings in their pre-slide phase, the system shows damped mass-spring characteristics.

changes of the operating point, the position control design is based on the measured local system dynamics with the ball bearings in the pre-slide phase. The control structure of a single system axis is shown in Fig. 7. The position of the mover is stabilized by the position control loop. The quasistatic setpoint for the position control loop, i.e. the operating point, is determined by the zero current control loop, which aims to minimize the quasi-static current consumption. Due to structural modes of the plant at frequencies only slightly higher than the respective resonance frequency, the plant is controlled at the spring line using integral control. For suppressing the respective resonance peak, the integral controller is combined with a notch filter with a transfer function given by

$$R_n(s) = \frac{1 + 2(\frac{\zeta}{a})(\frac{s}{\omega_n}) + (\frac{s}{\omega_n})^2}{1 + 2\zeta a(\frac{s}{\omega_n}) + (\frac{s}{\omega_n})^2}$$
(10)

and ω_n denoting the notch frequency. The depth of the notch is adjusted by the parameter *a*, while the width of the notch is set by parameter ζ . The resulting controller is given by

$$C_p(s) = C_i(s)R_n(s) = \frac{K_i}{s} \frac{1 + 2(\frac{\zeta}{a})(\frac{s}{\omega_n}) + (\frac{s}{\omega_n})^2}{1 + 2\zeta a(\frac{s}{\omega_n}) + (\frac{s}{\omega_n})^2},$$
 (11)

where K_i denotes the integral gain of the controller. The closed-loop bandwidth of the position control loop has to be at least one order of magnitude larger than the bandwidth of the zero current control loop in order to neglect its dynamics in the design of the zero current loop. Considering a trade-off between gain- and phase margins, the open loop crossover frequencies of the position control loops are tuned to be at 16 Hz and 20 Hz for axis x and y, respectively, by adjusting the parameters of the controllers, as listed in Tab. II. At these frequencies, gain margins of 23 dB and 24 dB with corresponding phase margins of 57° and 76° are obtained. The

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TABLE II				
PARAMETERS OF POSITION CONTROLLERS				

Parameter	Symbol	Axis X	Axis Y
Integral gain	K_i	2000	4000
Notch frequency	ω_n	$1457 {\rm rad s^{-1}}$	$1885 {\rm rad} {\rm s}^{-1}$
Width of notch	ζ	1	0.5
Height of notch	a	4	3.5

resulting complementary sensitivity functions of both axes are shown in Fig. 6, indicating $-3\,\mathrm{dB}\text{-bandwidths}$ of $28\,\mathrm{Hz}$ and $47\,\mathrm{Hz}$ for axis x and y, respectively.

With the designed position controllers additional closedloop measurements of the local system dynamics are acquired at various operating points in order to investigate their validity. This is necessary as for operating points of the mover at significant offsets from the center position and no closed-loop position control, the negative stiffness due to the permanent magnet, determining the global dynamics, would pull the mover into the stator yoke. The dynamics are again measured using small signal excitation and are, as expected, all found to resemble the local dynamics shown in Fig. 6. This confirms the design assumption that by using ball bearings local dynamics can be established in the entire actuation range.

D. Global System Dynamics

In order to validate the global system dynamics and to investigate the transition criteria from the local to the global dynamics, closed-loop experiments with large signal excitation and hysteresis measurements are conducted. The feedback controllers are only capable of stabilizing the position of the mover as long as local dynamics are valid, with the bearings in their pre-slide phase. Independent of the global operating point, the bearings remain in their pre-slide phase as long as the mover maintains quasi-static conditions with the velocity not exceeding a critical value. By using a larger signal amplitude and slowly increasing the frequency f_d of a sinusoidal position reference until the system becomes unstable, the critical velocity $v_{crit} = 2\pi \hat{x_d} f_d$ is found to be approximately $30 \,\mathrm{mm \, s^{-1}}$ for both axes. Exceeding this velocity, the bearings transition to their sliding phase, violating the condition for local dynamics and letting the negative stiffness dominated global system dynamics become effective. Thus, to maintain a stable system, the critical velocity must not be exceeded, which can be guaranteed by rate-limiting the admissible setpoint changes \dot{x}_d to the position control loop.

The non-linear spring module of the bearings in their preslide phase introduces a global hysteresis between the applied force and the position of the mover [22]. As indicated in (9), the generated force and the applied current are directly proportional, such that a hysteresis is also obtained between current and mover position. Experimentally, the current consumption of a single system axis is recorded for low pass filtered (cut-off frequency 1 Hz) step trajectories of the mover over the entire actuation range, with the x-y plane of the actuator in horizontal orientation. The time signal for position and current of axis x is shown in Fig. 8a. Current peaks are noticeable at the trajectory's points of inflection. By plotting



Fig. 7. Cascaded control structure of the overall system. The position of the mover is controlled by the feedback controller $C_p(s)$. By feeding back the coil current and adjusting the operating point x_d of the mover by the zero current controller $C_{zc}(s)$, the energy consumption of the system for compensation of quasi-static external forces can be reduced.



Fig. 8. Current consumption of axis x depending on its position. (a) shows the current consumption and position of a low pass filtered step trajectory. (b) shows the data of graph (a), plotted in a x-y graph. Due to hysteresis between current and position, the position of zero current consumption is not unambiguous.

the position over the current, as in Fig. 8b, the resulting hysteresis is obtained. Due to the hysteresis there are two solutions for an energy optimal operating point (zero current), which are approximately 0.2 mm apart and are depending on the direction of approach. This aspect is revisited in the zero current control design.

E. Zero Current Control

By adjusting the position of the mover relative to the stator according to the measured coil currents, zero current consumption can be realized [25]. Corresponding to the SISO-design approach, each axis is controlled independently. As discussed, the critical velocity must not be exceeded in order to maintain the local dynamics of the bearings in their preslide phase. Consequently, only slow changes of the effective load mass, e.g. due to a rotation of the system, are admissible. By applying an integral gain of $K_i = 1.5$, the transfer function of the zero-current controller results to

$$C_{zc}(s) = \frac{K_i}{s} = \frac{1.5}{s},$$
 (12)

leading to open-loop crossover frequencies of 0.1 Hz of axis x and 0.3 Hz for axis y. Consequently, only quasi-static currents, which are required for compensation of gravitational forces will be compensated. Considering the maximum current of the current amplifiers of 4 A and the controller dynamics, the maximum time derivative of x_d , i.e. the velocity of a setpoint

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change, is found to be 6 mm s^{-1} , which is five times smaller than the critical value v_{crit} identified in Section IV-D.

As discussed in the analysis of the hysteresis, there are two solutions for an operating point with zero current (cf. Fig. 8b). When applying the designed zero-current controllers to the system, the overshoot of the mover approaching the respectively closer operating point leads to a continuous oscillation of the system between both operating points. In order to prevent these oscillations, the integration of the error e_{zc} in the controller C_{zc} is disabled, as soon as the absolute value of the current decreases below a empirically determined threshold of $I_{th} = 75$ mA. This leads to a constant controller output and reference for the position control loop x_d . When the error exceeds the threshold I_{th} , the integration of the error re-enabled.

V. EXPERIMENTAL RESULTS

For assessing the functionality and performance of the system, the positioning uncertainty as well as the effect of zero current control for (i) the static single axis case and (ii) the slow load transition from one axis to the other are investigated.

In order to evaluate the uncertainty of the position control, the positioning resolution of the system is evaluated by applying steps to the reference inputs x_d and y_d of the position control loops. As depicted in Fig. 9, a step height of 4 nm can be clearly resolved. The position error of axis x has a peak-to-peak value of 12 nm and an RMS value of 1.94 nm, while the position error of axis y has a peak-to-peak value of 14 nm and an RMS value of 2.10 nm, clearly demonstrating the nano-metre resolution of the system.



Fig. 9. Evaluation of positioning uncertainty. The reference signal includes steps of 4nm applied to the system with the mover being initially at the center position. (a) shows the position signal of the x-axis with an rms error of 1.94 nm. (b) shows the response of the y-axis with an rms error of 2.10 nm.

The functionality of the zero-current control with one DOF is demonstrated in Fig. 10a. The actuator is mounted with the x-y plane in vertical orientation, as in Fig. 5. Starting at = 0 s with zero-current control switched off and active position control, a current of 0.47 A is required to keep the mass of the mover at y = 0 mm. At $t \approx 9$ s, an additional payload of 500 g is applied to the mover, increasing the current to 1.58 A in order to keep the position of the mover at zero. At $t = 17 \,\mathrm{s}$, the zero-current control loop is activated, forcing the position control loop to lift the mover to y = 0.55 mm. With the biasing flux of the permanent magnet now compensating almost the entire gravitational force on the mover, the coil current is reduced to 10 mA. Compared to the reference state of the system with the mover in the center position, activation of zero current control reduces the required electrical power, determined via the ohmic losses of the coils (serial resistance of $R_s = 0.85 \Omega$ per axis), from 2.12 W to 85μ W, yielding a reduction by a factor of almost 25,000.



Fig. 10. Experiments for evaluation of the zero current control loops. (a) shows zero current control with single axis operation: By activation of zero current control at 17 s, the current required for gravity compensation reduces to 10 mA. (b) depicts zero current control with two DOF and a rotational movement of the actuator by 90° .

In order to demonstrate the orientation independence of the developed system, the slow load transition of the gravity compensation from one axis to the other is demonstrated, while rotating the actuator by 90° around the z-axis, as shown in Fig. 10b. The gravitational force of the mover and the payload is initially compensated by axis y only ($t \approx 7$ s). During the rotation, the gravitational force is dynamically compensated by both axes (dashed green circle in Fig. 10b). After completing the rotation, the gravitational force is compensated solely by axis x, demonstrating the seamless transition of compensation of the gravitational force from axis y to axis x. With the power consumption determined by the ohmic losses of the coils, $i_P = \sqrt{i_x^2 + i_y^2}$ represents the current leading to an

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equivalent total power consumption as in both actuator axes. When applying a payload of 500 g at $t \approx 6 \text{ s} i_P$ increases to 1.56 A. By activation of zero current control (axis x at $t \approx 14 \text{ s}$, axis y at $t \approx 16 \text{ s}$), the current required for gravity compensation reduces to 20 mA. After the rotation by 90°, which takes place between $t \approx 23 \text{ s}$ and $t \approx 30 \text{ s}$, a current of 53 mA is required to hold the additional payload.

Finally considering the effects of vibrations on the stator part, two spectral ranges have to be distinguished in the face of the operating principle. Vibration components within the bandwidth of the position control loop but beyond the bandwidth of the zero current control loop will be rejected by the position control loop but the resulting current variations will not affect the current controller output, such that the operating point remains unaffected. Within the bandwidth of the current control loop, external vibrations will additionally lead to a gradual adaptation of the operating point.

In summary, it is successfully shown that the proposed method for operating an HRA enables actively controlled zero power gravity compensation of the mover and that it can be used for the design of orientation independent levitation mechanisms.

VI. CONCLUSION

This paper contributes a novel method for energy efficient gravity compensation by making use of the inherent negative stiffness of HRAs and demonstrates that it can be used to enable orientation independent operation for a system with one rotational degree of freedom. By balancing gravitational forces on the mover with the position dependent force of the integrated permanent magnet of an HRA, gravitational forces can be compensated powerless. For evaluation of the proposed method, an HRA system with two degrees of freedom is developed and modeled to determine the system parameters and the compensable force for the design process. The paper further contributes the integration of linear ball bearings to deliberately establish different global and local system dynamics for the sake of system control. Position controllers for both axes are designed based on the local openloop stable system dynamics, originating from the pre-slide stiffness of the ball bearings. The positioning resolution of the system is evaluated by measuring the response of the system to steps in the reference, showing an RMS-error of $1.94\,\mathrm{nm}$ for axis x and $2.10\,\mathrm{nm}$ for axis y. Zero current control is realized by feeding back the measured coil current and accordingly adjusting the position of the mover by the position controllers. In conducted experiments, the power consumption for compensation of a payload of $500\,\mathrm{g}$ is reduced by a factor of 25000. It is also shown that gravity is energy efficient compensated during a rotational motion of the system by 90°, successfully demonstrating the proposed method's orientation independent functionality.

REFERENCES

- J. Yonnet, "Permanent magnet bearings and couplings," *IEEE Transac*tions on Magnetics, vol. 17, no. 1, pp. 1169–1173, Jan. 1981.
- [2] J. Janssen, J. Paulides, E. Lomonova, B. Delinchant, and J.-P. Yonnet, "Design study on a magnetic gravity compensator with unequal magnet arrays," *Mechatronics*, vol. 23, pp. 197–203, 2013.

- [3] S. Ito, B. Lindner, and G. Schitter, "Sample-tracking vibration isolation with hybrid reluctance actuators for inline metrology," *IFAC-PapersOnLine*, vol. 52, no. 15, pp. 537 542, 2019.
 [4] D. Wertjanz, E. Csencsics, and G. Schitter, "3 dof vibration compen-
- [4] D. Wertjanz, E. Csencsics, and G. Schitter, "3 dof vibration compensation platform for robot-based precision inline measurements on freeform surfaces," in *IEEE Transactions on Industrial Electronics*, vol. 69, no. 1, pp. 613–621, 2021.
- [5] D. B. Hiemstra, G. Parmar, and S. Awtar, "Performance tradeoffs posed by moving magnet actuators in flexure-based nanopositioning," *IEEE/ASME Transactions on Mechatronics*, vol. 19, no. 1, pp. 201–212, 2014.
- [6] R. Saathof, M. Thier, R. Hainisch, and G. Schitter, "Integrated system and control design of a one dof nano-metrology platform," *Mechatronics*, vol. 47, pp. 88–96, 2017.
- [7] R. Deng, R. Saathof, J. Spronck, S. Hol, and R. Schmidt, "Integrated 6dof lorentz actuator with gravity compensator for precision positioning," in 22nd International Conference on Magnetically Levitated Systems and Linear Drives, 2014.
- [8] H. Zhang, B. Kou, Y. Jin, and H. Zhang, "Modeling and analysis of a new cylindrical magnetic levitation gravity compensator with low stiffness for the 6-dof fine stage," *IEEE Transactions on Industrial Electronics*, vol. 62, no. 6, pp. 3629–3639, 2015.
- [9] S. Hol, E. Lomonova, and A. Vandenput, "Design of a magnetic gravity compensation system," *Precision Engineering*, vol. 30, no. 3, pp. 265 – 273, 2006.
- [10] K. Halbach, "Application of permanent magnets in accelerators and electron storage rings," *Journal of Applied Physics*, vol. 57, no. 8, pp. 3605–3608, 1985.
- [11] Y.-M. Choi, M. G. Lee, D.-G. Gweon, and J. Jeong, "A new magnetic bearing using halbach magnet arrays for a magnetic levitation stage," *Review of scientific instruments*, vol. 80, no. 4, p. 045106, 2009.
- [12] F. Cigarini, S. Ito, S. Troppmair, and G. Schitter, "Comparative finite element analysis of a voice coil actuator and a hybrid reluctance actuator," *IEEJ Journal of Industry Applications*, vol. 8, pp. 192–199, 2019.
- [13] N. H. Vrijsen, J. W. Jansen, and E. A. Lomonova, "Comparison of linear voice coil and reluctance actuators for high-precision applications," *Power Electronics and Motion Control Conference (EPE/PEMC)*, 2010.
- [14] A. van Lievenoogen, A. Toma, and U. Ummethala, "Challenges in the application of hybrid reluctance actuators in scanning positioning stages in vacuum with nanometer accuracy and mgauss magnetic stray field," *American Control Conference*, 2013.
- [15] X.-D. Lu and D. Trumper, "Ultrafast tool servos for diamond turning," *CIRP Annals*, vol. 54, no. 1, pp. 383 – 388, 2005.
- [16] D. J. Kluk, M. T. Boulet, and D. L. Trumper, "A high-bandwidth, high-precision, two-axis steering mirror with moving iron actuator," *Mechatronics*, vol. 22, no. 3, pp. 257 – 270, 2012.
- [17] E. Csencsics, J. Schlarp, and G. Schitter, "High-performance hybridreluctance-force-based tip/tilt system: Design, control, and evaluation," *IEEE/ASME Transactions on Mechatronics*, vol. 23, no. 5, pp. 2494– 2502, 2018.
- [18] S. Ito, S. Troppmair, B. Lindner, F. Cigarini, and G. Schitter, "Longrange fast nanopositioner using nonlinearities of hybrid reluctance actuator for energy efficiency," *IEEE Transactions on Industrial Electronics*, vol. 66, no. 4, pp. 3051–3059, 2019.
- [19] D. Laro, R. Boshuisen, J. Dams, and J. van Eijk, "Linear hybrid actuator for active force cancellation," *International Symposium on Linear Drives* for Industry Applications, vol. 8, 2011.
- [20] D. Wu, X. Xie, and S. Zhou, "Design of a normal stress electromagnetic fast linear actuator," *IEEE Trans. Magn.*, vol. 46, pp. 1007 – 1014, 2010.
- [21] E. Csencsics, J. Schlarp, and G. Schitter, "Bandwidth extension of hybrid-reluctance-force-based tip/tilt system by reduction of eddy currents," in 2017 IEEE International Conference on Advanced Intelligent Mechatronics (AIM), 2017.
- [22] C. Hsieh and K. Yang, "On the role the friction plays in an ultra precision positioning system," in 2009 IEEE International Conference on Control and Automation, 2009.
- [23] T.-Y. Lin, Y.-C. Pan, and C. Hsieh, "Precision-limit positioning of direct drive systems with the existence of friction," *Control Engineering Practice*, vol. 11, no. 3, pp. 233 – 244, 2003.
- [24] R. Munnig Schmidt, G. Schitter, A. Rankers, and J. van Eijk, *The Design of High Performance Mechatronics*, 2nd ed. Delft University Press, 2014.
- [25] T. Mizuno and Y. Takemori, "A transfer-function approach to the analysis and design of zero power controllers for magnetic suspension system," *Electrical Engineering in Japan*, vol. 141, pp. 67 – 75, 2001.

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