Crossing Borders within the ABC

Automation, Biomedical Engineering and Computer Science

Faculty of Computer Science and Automation

www.tu-ilmenau.de

Home / Index: http://www.db-thueringen.de/servlets/DocumentServlet?id=16739
1. INTRODUCTION

In order to meet the increasing requirements in high performance and high dimension accuracy for positioning systems, there has been active research on hovering, high-precision planar positioning systems in the past few years [1], [2], [3], [4], [5]. While latest technology provides precise positioning capabilities down to the Nanometer scale, further improvements in high-precision dynamic positioning will only be possible by applying modern nonlinear control methods. This contribution constitutes the first step towards this ambitious goal by providing a detailed mathematical model of Tetra’s state-of-the-art high-precision planar positioning system “PPS200”.

The “PPS200” system consists of a stator base and a slider which is levitated over the stator plate by means of air bearings. The slider is suitable to carry a maximal pay load of 14kg which is to be positioned in the plane with a specified yaw-angle. The travel range of the slider is 200x200mm² and the maximal yaw angle is ±0.25°. A number of permanent magnets mounted at the bottom of the slider such that their magnetic field interacts with the current through the coils of the stator plate resulting in a force (Lorenz-force) which consequently propels the slider. The position driven is measured by an optical, incremental measurement system consisting of three optical sensors located at the center of the stator plate [5]. One sensor is used to measure motion in the direction of z and the rest of them being y1 and y2 are used in order to compensate or even actively control the yaw ϕz along the vertical z axis as well as detecting the y-position driven. Unsurprisingly other positioning systems share similar assemblies [4].

In order to capture the slider’s motion an earth-fixed coordinate system (x, y, z) located at the center of the stator plate is used. The coordinates x and y describe the position of the slider’s center of gravity, while ϕz denotes the slider’s orientation or yaw.

The block-diagram in Fig. 2 shows the basic structure of the closed loop system. The aim is to control the slider coordinates (x, y, ϕz) measured by the sensor system described, via the current through the coils considered as control input. Therefore the variables (x_d, y_d, ϕ_{zd}) define the reference trajectory for the slider to travel.
The driving coil generates an actuator signal being the desired current vector \( \mathbf{i}_d \). The actuator signal \( \mathbf{i}_d \) is then fed into a commutation stage composed of power transistors in switching mode amplifying the PWM-signal. The load is connected in an H-bridge, while the current through the load is measured through the voltage across the measuring resistance \( R_M \) and fed back to the controller in order to track the desired value.

The desired current through the coils \( i_d \) is represented by the voltage \( u_{af} \) across the input filter which is a simple RC-low-pass filter. It can be described by

\[
\dot{u}_{af}(t) = \frac{1}{T_f} (u_{af}(t) - u(t)),
\]

where \( T_f = RC \) is the lag time constant. After filtering, the desired set point for the current through the coil is denoted by the voltage \( u_{af} \) being the input for the PI-controller which generates the actuator signal \( u_R \), also measured in volts. The control law for the PI-controller is given by

\[
u_R(t) = K_R \left( e(t) + \frac{1}{T_{nR}} \int_0^t e(\tau) d\tau \right).
\]

\( K_R \) and \( T_{nR} \) denote the controller gain and the time constant of the integrator, respectively. The control error is

\[
e(t) = u_{af}(t) - K_{RR} u_{aL}(t).
\]

In the comparator unit (denoted “comp” in Fig. 3) the signal \( u_R \) is compared to a triangle wave form \( u_{\Delta} \) with a base frequency of roughly 22 kHz. Thus the output of the comparator switches to high, if \( u_R > u_{\Delta} \) or to low, if \( u_R < u_{\Delta} \) generating the pulse width modulated signal with respect to the desired input signal. This signal is amplified by power MOSFETs operating in switching mode (see switching stage in Fig. 3). The amplifiers output is given by the piecwise constant function

\[
u_{aL}(t) = \begin{cases} +u_{aL,max} & \text{for } u_R(t) > u_{\Delta}(t) \\ -u_{aL,max} & \text{for } u_R(t) < u_{\Delta}(t) \end{cases},
\]

where \( u_{aL,max} = 150V \).

The second part of the actuator consists of the driving coils and two further coils in series acting as chokes for improving EMC. The resistance \( R_G \) in Fig. 3 represents the sum of

\(^1\)Institut für Mikroelektronik- und Mechatronik-Systeme gGmbH, Ilmenau
of the coil winding resistances of those three coils and \( L_G \) denotes their overall inductance. Finally \( R_M \) is a resistance to ground used to measure the current through the coils. The amplified pulse width modulated signal \( u_{aL} \) acts as the input of this network. The dynamics of this circuit is given by:

\[
\frac{di(t)}{dt} = -\frac{1}{L_S} \left( (R_M + R_S)i(t) - u_{aL}(t) \right). \tag{5}
\]

The output voltage \( u_{aL} = R_Mi \) is fed back via the gain \( K_{RK} \) in the feedback loop.

For verification of the derived model we conducted several experiments at the test rig. At the input of the PWM-amplifier we applied rectangular pulse signals with amplitudes varying between 1 and 4 V while the slider was re-distributed. Besides the ripples that are characteristic for a PWM-amplifier at constant currents, we can observe a significant increase of the slope when the current amplitudes vary. The parameters \( K_1, \ldots, K_4 \) were obtained by standard nonlinear least-squares optimization.

In Fig. 4 the measured data is compared to the simulation using the nonlinear inductance model (6). The overall performance of the model is very satisfying and even that characteristic increase of the slope due to core saturation (c.f. Fig 4 (a)) is very well matched.

4. Electromechanical Coupling

The forces acting on the slider are generated by the interaction between the electromagnetic field induced by the current through the driving coils and the magnetic field of the permanent magnets at the bottom of the slider. According to Lorentz’ law the electromagnetic force is given by

\[
F_L = i l \times B, \tag{7}
\]

where \( i \) is the current through a wire with constant length \( l \) and \( B \) is the flux density of the homogeneous magnetic field of a permanent magnet [7].

In order to derive a sufficiently precise description of the electromagnetic coupling we make the following (mild) assumptions:

1. The vertical distance between the magnets and the coil-surface is sufficiently small so that the magnetic field is homogeneous (in the vertical direction) and perpendicular with respect to the coil surface.

2. The sag (subsidence) of the air bearings (roughly 5 μm) is coupled with the quantities \( z, \theta_s \) and \( \varphi_e \) of the slider. The influence of these quantities on the absolute value as well as the direction of the Lorentz-force is negligibly small. Hence, we assume that the magnetic field is always perpendicular to every single coil-winding such that the forces for the \( x \)- and \( y \)-direction are decoupled from each other. The forces act always perpendicular with respect to the corresponding driving coil regardless of the orientation of the slider.

3. The rotation about the \( z \)-axis is sufficiently small such that the electromagnetically relevant area between the magnets and the coils’ surfaces is independent of the slider’s orientation \( \phi_z \). In fact the maximum rotation is restricted to \( \phi_{z, \text{max}} = \pm 0.25^\circ \).

Fig. 5. Shape of the magnetic field of the permanent magnets along the \( x \)-axis

![Diagram of the magnetic field of the permanent magnets along the x-axis](image-url)
With the above assumptions the electromagnetic force generated by the \( j \)-th coil and the slider’s magnetic field is given by

\[
F_j = i_j l B, \tag{8}
\]

where \( i_j \) is the current through the respective coil, \( l \) is the effective length of the conductor and \( B \) is the magnetic flux density of the magnetic field effective for the coil.

The magnetic field generated by the magnets is not homogeneous along the \( x \)-axis (bottom and top coils) and the \( y \)-axis (left and right coils), see Fig. 5 for an illustration. Thus the flux density \( B \) of the magnetic field interacting with the coils depends on the slider position above the stator plate. Fig. 6 shows the measurements of the magnetic flux over the \( x \)-position of the slider. Approximating this function by

\[
B_x = B \cos \left( \frac{2 \pi x}{T_K} \right), \tag{9}
\]

where \( T_K \) is the distance between the centers of adjacent magnets, yields satisfying results as shown also in Fig. 6.

![Fig. 6. Magnetic flux density along the \( x \)-direction.](image)

Fig. 7. Feedforward-structure to compensate the spacial dependence of the magnetic field.

The resulting force on the slider in \( x \)-direction is given by

\[
F_x = F_{Top} + F_{Top2} + F_{Btm1} + F_{Btm2}. \tag{10}
\]

By choosing the currents \( i_{d_x} \) as sinusoidal functions of the slider position \( x \) with appropriate phase-shift accounting for the offsets due to the coil position on the stator, the spacial dependency can be compensated by the commutation. Assuming an ideal compensation yields for the force on the slider:

\[
F_x = 2l \dot{B} i_{d_x}, \tag{11}
\]

where \( l \) in \([m]\) is the effective length of the conductor and \( \dot{B} \) is the magnitude of the magnetic flux density in \([T]\). Note, that \( l \dot{B} \) is constant such that the force \( F_x \) only depends on the current \( i_{d_x} \), but is independent of the slider’s position. Similar considerations yield according results for \( F_y \) and \( M_z \).

5. MECHANICAL SLIDER MODEL

![Fig. 8. Planar (a) and vertical (b),(c) forces as well as torque along the \( x \) (a), \( y \) (b) and \( z \) (a) axes (red:forces and torque, blue:angles, green:distances)](image)

The slider is essentially a mass with six degrees of freedom suspended by air bearings. Deriving the equations of motion we shall distinguish between the planar movements \((x, y, \phi_x)\) and the additional spacial movements \((z, \phi_z, \psi)\) considering the third dimension. In the former case the prominent forces on the slider are the propelling electromagnetic forces and some viscous friction forces due to the air bearings. In the latter case, the slider motion is dominantly influenced by the suspending forces, but we shall also consider the fact that the propelling forces attack outside the center of gravity and thus cause for roll and pitch motion.

To begin with we derive the planar equations of motion, see Fig. 8 (a). The black square in the figure denotes the points of attack of the electromagnetic forces along the rows of permanent magnets at the bottom of the slider. Assuming that the planar forces are decoupled (see Section 4) we obtain the following balance of forces by applying the principle of linear momentum (Newton’s second law):

\[
m \ddot{x} = F_{Btm} + F_{Top} - F_{Rg}, \tag{12}
\]

\[
m \ddot{y} = F_{Btm} + F_{Top} - F_{Rg}, \tag{13}
\]

where \( m \) is the slider mass and \( F_{Btm}, F_{Top}, F_{Rg}, F_{Ls} \) are the resulting forces of each respective pair of coils. The linear damping terms \( F_{Rg} = k_x \dot{x} \) and \( F_{Rg} = k_y \dot{y} \) denote the so-called drag- or viscous friction forces (Stokes law) evoked by the air bearings and the viscosity of air (see [2]).
For the motion about the z-axis we apply the principle of conservation of angular momentum:

\[ J_{CG} \ddot{\phi}_z = M_{Btm} + M_{Rgt} - M_{Top} - M_{Lft}, \tag{14} \]

where \( J_{CG} \) denotes the slider’s moment of inertia about the z-axis and the respective torques are given by

\[
M_{Btm} = \left( \frac{d}{\cos(\phi_z)} + x \tan(\phi_z) \right) F_{Btm}, \tag{15}
\]
\[
M_{Top} = \left( \frac{d}{\cos(\phi_z)} - x \tan(\phi_z) \right) F_{Top}, \tag{16}
\]
\[
M_{Rgt} = \left( \frac{d}{\cos(\phi_z)} + y \tan(\phi_z) \right) F_{Rgt}, \tag{17}
\]
\[
M_{Lft} = \left( \frac{d}{\cos(\phi_z)} - y \tan(\phi_z) \right) F_{Lft}. \tag{18}
\]

For the dynamics in z-direction we are investigating the influence of the air bearings. The air bearings are vacuum preloaded meaning that they provide vacuum air as well as pressurisation (see [8]). We approximate their dynamical behaviour with a simple parallel spring-damper model similar to the approach in [3]. The suspension force \( F_{ci} \) of the i-th air bearing is then given by

\[
F_{ci} = c_{di} z_i - d_{di} \dot{z}_i \tag{19}
\]

where \( d_{di} \) is the damping-constant, \( z_i \) is the distance between the stator surface and the i-th air bearing, and \( c_{di} \) is the spring constant which again depends on the air gap \( z_i \). Elementary geometric considerations (c.f. Fig. 8 (b),(c)) yield for the air gaps at each bearing:

\[
z_1 = z + \frac{b}{2} \left( \sin(\theta_x) + \sin(\psi_y) \right) \tag{20}
\]
\[
z_2 = z + \frac{b}{2} \left( \sin(\theta_x) - \sin(\psi_y) \right) \tag{21}
\]
\[
z_3 = z - \frac{b}{2} \left( \sin(\theta_x) - \sin(\psi_y) \right) \tag{22}
\]
\[
z_4 = z - \frac{b}{2} \left( \sin(\theta_x) + \sin(\psi_y) \right) \tag{23}
\]

For identification of the spring constant \( c_{di} \) we resort to extensive experiments conducted at the IMMS who established a typical load deflection curve of an air bearing depicted in Fig. 9. Note however, that for a particular air bearing this curve may differ significantly.

![Fig. 9. Load deflection curve of a single air bearing established by the IMMS](image)

The dynamics in the z-direction are then given by

\[
m \ddot{z} = \sum_{i=1}^{4} F_{ci} - mg. \tag{24}
\]

Finally we consider the balances of torques about the x- and y-axis:

\[
J_{CG} \ddot{\theta}_x = z_{cg} \left( F_{Btm} + F_{Lft} \right) + \frac{b}{2} \left( F_{c1} + F_{c2} - F_{c3} - F_{c4} \right) \tag{25}
\]
\[
J_{CG} \ddot{\psi}_y = z_{cg} \left( F_{Rgt} + F_{Top} \right) + \frac{b}{2} \left( F_{c1} + F_{c4} - F_{c2} - F_{c3} \right), \tag{26}
\]

where \( z_{cg} \) denotes the lever arm of the driving forces with respect to the center of gravity, c.f. Fig. 8 (b),(c).

The moments of inertia in (14),(25),(26) are given by

\[
J_{CG} = \frac{1}{12} m (a^2 + c^2) \tag{27}
\]
\[
J_{CG} = \frac{1}{12} m (a^2 + c^2) \tag{28}
\]
\[
J_{CG} = \frac{1}{6} ma^2 \tag{29}
\]

where \( a \) is the width of the slider, \( c \) its hight and \( m \) its mass.

The complete model of the slider is given by the differential equations (12), (13), (14), (24), (25) and (26) representing the dynamics of the slider in an adequate and simple way.

6. EXPERIMENTAL VERIFICATION

The model derived above was implemented in Simulink and verified using measurements of the test rig. For those experiments the test rig was run in open loop and square-wave input signals with various amplitudes were applied demanding a movement in x-direction. During those experiments the slider was fixed in the y-direction to avoid measurement errors due to disturbances.

Fig. 10 shows the results for \( i_{d1} = 0.1A, 1A \) and 2A, respectively. Note that only the position (figures (a1)) was measured at the test rig. The velocities (\( b_{i1} \)) as well as accelerations (\( c_{i1} \)) were obtained by numerical differentiation. Moreover, the air gaps in Fig. 10 (d1) are simulative results only as no sensors for the air gaps are available at the test rig.

By examination of the Figures (a2),(3), (b2),(3), (c2),(3) of the second and the third measurement one can observe that for currents \( i_{d1} > 1A \) the overall mathematic model approximates the real system quite good. However, Fig. 10 (a1)-(c1) depicting the results of the first measurement illustrates that there is some kind of damping effect present at low currents \( i_{d1} < 1A \). The velocity (\( a_2 \)) increases rapidly at the beginning (0-0.5s) but decreases around 0.5s and settles in a steady-state. Those damping effects might be caused by eddy-current and magnetization effects as observed in [6]. Another possible reason for this behaviour may be due to errors in the commutation or the geometrical assembly of the permanent magnets. Our derived model does not account for any of these effects and thus is not capable to reflect this damping behaviour. Extending the model in this regard will be subject of future investigations.

Fig. 10 (d1,2,3) show the simulation results of the air gaps at two of the bearings for an input amplitude of 3.4. The variations of the air gap due to the acceleration in x-direction are very small (in the range of 0.07\( \mu \)m), which is deemed to have no impact on the electromagnetic coupling. The resulting pitch and roll motion are also small such that their influence on the dynamics of the slider is indeed arguable.
7. CONCLUDING REMARKS

In this paper we present a study on the dynamic behaviour of Tetra’s high-precision planar positioning system “PPS200”. We derived physical models of the main components of the system and analysed various nonlinear effects. The resulting system description proves suitable to simulate the dynamics of the positioning system to a large degree of precision. Significant deviations of the simulation and experimental results are only observable at very low input currents. Further investigations are needed to analyse the cause of the damping effect observed for this input range. The derived model is suitable to serve as a basis for analysis and application of nonlinear control algorithms.

8. REFERENCES