A new mechanism design of electro-magnetic actuator for a micro-positioner

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Abstract

This paper proposes a new type of non-conventional electro-magnetic actuator (EMA) for micro-positioning. It is a repulsive magnetic system consisting of a motion pad and two active coils. The motion of the pad is caused by the repelling force between the coils and the permanent magnets on the pad. First, the dynamic model is derived and analyzed. Next, an adaptive sliding mode controller is developed to deal with the unknown parameters with the objective of precision positioning. The experimental results demonstrate satisfactory performances including regulation accuracy and control stiffness for the system.

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1. Introduction

High-precision positioning devices play an increasingly important role in a variety of high-tech applications, such as IC-photolithography (stepper and repeated positioning), materials science (tunneling microscopy), medicine and biology (cell biology). To meet these needs, more stringent manufacturing processes and advanced fabrication equipment should be developed. Recently, for a small range of motion, flexure systems seem to be able to satisfy many positioning requirements. An example is a piezoelectric actuator \cite{1–3} which can realize 10 nm resolution with 1 µm travel. On the other hand, larger range positioners, such as the steppers in IC-lithography, are based on devices with ball-screws or linear motors \cite{4,5}. However, ball-screws cause disturbances and backlash due to the roughness of the bearing elements, and linear motors have edge effects in a motion stroke.

These reasons have motivated the authors to design a new electro-magnetic actuator (EMA) to achieve the objectives of large-moving range and high-precision positioning. The EMA is a repulsive magnetic system with a passive motion pad and two active coils, the structure is shown in Fig. 1. The motion of the pad results from the repelling forces between the magnets fixed to the pad and the coils attached to the linear guide. The dual coils are devised to increase the stiffness of the system.

To accomplish the goal of positioning, a controller has to be designed to control the EMA system. However, in practice there will exist some uncertainty in the system, such as changing temperature within the coil, friction, etc., which will inevitably result in modeling errors. Therefore, the developed controller should be robust enough to tolerate these uncertainties and unmodeled dynamics. Besides these uncertainties, the controller also has to eliminate effects due to disturbances caused by operating.

There are two categories of methodologies available from the control point of view. One is robust control, such as $H_\infty$ or $LQG$ \cite{6,7}. These controller gains are fixed and designed to be robust in accordance with the nominal mathematical model. The other methodology is adaptive control \cite{8–11}, which can perform system identification implicitly or explicitly on-line while tuning the controller gains to guarantee the stability of the closed-loop system. In principle, if on-line system identification can be achieved, the performance of the adaptive controller should be better than that of the conservative robust controller, especially in the case of a definite model structure with unknown and variable parameters. Nowadays, high-speed PC-based controllers are more economical and reliable for intensive computations to realize a complex adaptive control.
In this paper, an EMA system is described which is a precision positioning system. Then, a dynamic model will be derived, and based on this model an adaptive sliding mode controller is designed to deal with unknown system parameters so as to achieve satisfactory system performance. To demonstrate the effectiveness of the entire system design, experimental results are provided for verification. The organization of this paper is as follows. The detailed description of the prototype system and its complete dynamics will be given in Sections 2 and 3, respectively. Section 4 concentrates on the system controller design and stability analysis, and Section 5 presents the experimental results as well as some discussions. Finally, conclusions and possible future research are given in Section 6.

2. Prototype design of the electro-magnetic actuator

This section describes the sequential design of the prototype EMA. The illustration about the materials that are used in this research, and the resulting geometric shapes, will be given at the end.

2.1. Motion pad design

This research constructs the motion pad embedded with two magnets, as illustrated in Fig. 1 which demonstrates various views of the entire mechanism. The pad is made of aluminum to keep the pad lightweight and prevent it from being magnetized. NdFeB seems at present to be the best choice of material for the magnetic actuator, since AlNiCos have low coercivity, ferrites have low remanence, and samarium magnets are still quite expensive. There are several useful characteristics of NdFeB, including a remanence of 1.29 T, coercivity of 990 kA/m, resistivity of 1.5 μΩ m, and a maximum energy product of 320 kJ/m³.

2.2. Cylindrical solenoid analysis

A cross-section of the N-turn solenoid, with axial length h, inner radius r₁, and outside radius r₂, is shown in Fig. 2. Let a current of I amperes pass through the wire with direction φ. Referring to Fig. 2, the element of source volume can be expressed as dv' = r'dφ'dr'dz' in cylindrical coordinates. Since we assume that the turns of wire are wound uniformly on the cylindrical solenoid, by the Biot-Savart Law [12] the magnetic field intensity is

\[
H = \frac{1}{4\pi} \int_{-h/2}^{h/2} \int_{0}^{2\pi} \int_{r_{1}}^{r_{2}} \left( \frac{NI}{r_{2} - r_{1}} \right) \frac{\left( \phi' \times \vec{r}' \right) dp' \, d\phi' \, dz'}{p - p'} \, dz \, d\phi \, dr
\]

(1)

where \( \vec{r}' \) is the source-observer unit vector and \( \vec{r}' \phi' \) is the unit vector tangential to the cylindrical surface in the \( xy \)-plane. Expanding Eq. (1) and focusing on the axial field of the circular cylindrical solenoid, we can obtain an expression for the magnetic field intensity in the \( z \) direction as

\[
H_z = \frac{1}{4(-r_1 + r_2)h} \begin{vmatrix} NI \left\{ h \log \left[ r_1 + \sqrt{r_1^2 + \left( -\frac{h}{2} + z \right)^2} \right] \\
- h \log \left[ r_2 + \sqrt{r_2^2 + \left( -\frac{h}{2} + z \right)^2} \right] \\
- h \log \left[ r_1 + \sqrt{r_1^2 + \left( \frac{h}{2} + z \right)^2} \right] \\
+ h \log \left[ r_2 + \sqrt{r_2^2 + \left( \frac{h}{2} + z \right)^2} \right] \\
- 2z \log \left[ \frac{4 \left( r_1 + \sqrt{r_1^2 + \left( -\frac{h}{2} + z \right)^2} \right)}{r_1 z \left( -h + 2z \right)} \right] \\
- 2z \log \left[ \frac{4 \left( r_2 - \sqrt{r_2^2 + \left( \frac{h}{2} + z \right)^2} \right)}{r_2 z \left( -h + 2z \right)} \right] \\
+ 2z \log \left[ - \frac{4 \left( r_1 - \sqrt{r_1^2 + \left( \frac{h}{2} + z \right)^2} \right)}{r_1 z \left( h + 2z \right)} \right] \end{vmatrix}
\]

Fig. 1. Perspective of whole electro-magnetic actuator.
Therefore, once the specification of the electromagnet is determined, it is not hard to imagine that the magnetic field intensity in the $z$ direction can be simplified as the product of a nonlinear function of the observer’s position relative to the electromagnet. Thus, the current flowing into the electromagnet, i.e. Eq. (2), can be written

$$ H_z = G(z)I, \quad (3) $$

where $G(z)$ is a function of observer’s position relative to the electromagnet when the size of the electromagnet is specified.

In general, the relationship between the magnetic flux density $B$ and the magnetic field intensity $H$ in the $z$ direction can be defined as

$$ B(z, I) = \mu_0 H_z, \quad (4) $$

where $\mu_0$ denoting the permeability in a free space is equal to $4\pi \times 10^{-7}$ H/m. Without the presence of electrical field, the expression of the Lorentz force for an infinitesimal current loop can be simplified to

$$ F(z, I) = (m \cdot \nabla)B(z, I), \quad (5) $$

where $m$ is the dipole moment of this infinitesimal current loop.

However, for real-time control of this system, it is difficult to establish a compact analytical model of the magnetic force from the derivation, Eqs. (2)–(5). Thus, to solve this problem, the alternative method is to derive an empirical model based on practical measurements of $F(z, I)$, at different values of current $I$ and position $z$ as shown in Fig. 3, where we set $z = 0$ at the top surface of coil and let the total measuring range $z$ from the center through the top surface to the outside of the coil be $-28 \text{ mm} \leq z \leq 24 \text{ mm}$. There is one interesting situation in this figure that is worth pointing out. Even though the force measurement is made at some offset from the central axis of the electromagnet, the measured values are still quite close to those on the central axis. The largest mismatched values are within 10% of the central axis values. Based on the relation between the magnetic force vs. the desired current $I$ and position $z$, it can be found that the force/current ratio is approximately proportional to position $z$ in the range $-28 \text{ mm} \leq z \leq 0 \text{ mm}$, inside the coil. So, it is justified to simplify $F(z, I)$ as a function which depends only on the position variable $z$ and driving current $I$, i.e., Eq. (5) can be approximated as

$$ F(z, I) = K \cdot z \cdot I, \quad (6) $$

where $K$ is a constant relating the force/current ratio and the position $z$.

3. Modeling of the electro-magnetic actuator

The mechanical design of an EMA was described in the previous section. Its analytical model will be derived and analyzed in this section.

Before proceeding with the modeling, several assumptions are necessary in order to make the problem more tractable.

- The $N$ turns of wire distributed across their cross-sections can be viewed as lumped into a single wire located at their cross-sectional geometric center. This allows close approximation of the true forces in a more practical manner.

- Each magnet is considered one magnetic dipole carrying the same magnetic dipole moment and is located at the center of each magnet.

- The moving range of the motion pad is set to $-28 \text{ mm} \leq z \leq 0 \text{ mm}$ inside the coil, i.e., the force/current ratio and the position $z$ is linearized to this range.

The motion pad is considered to be a rigid body with the center of mass coincident with the center of gravity. The linear momentum depicted in Fig. 1 is given by

$$ M \ddot{z} = F(z, I_1) - F(z, I_2) + F_{\text{fr}}, $$

where $M$ is the mass of the pad, $F(z, I_1)$ and $F(z, I_2)$ are the forces induced by the left and right stator electro-magnetic
circuit in the z position, respectively, $F_{0i}$ is the Coulomb friction [13] of the linear guide, and $h$ is the axial length of the coil. Then, the dynamic equation (7) can be rewritten

$$M \ddot{z} = K(I_1 + I_2)z + \frac{Kh}{4}(I_1 - I_2) + F_{\text{int}}.$$  \quad (8)

### 4. Controller design and stability analysis

The complete dynamic model was derived in the previous section based on several assumptions. However, in practice there exist some modeling errors, such as changing temperature within the coil, friction, etc. Therefore, the controller to be developed should be sufficiently robust to these uncertainties and unmodeled dynamics. Besides these uncertainties, the controller also has to eliminate the effects due to disturbances caused by operating. Based on experimental evidence [14–17], an adaptive sliding mode control is designed for this system, which can perform system identification on-line implicitly or explicitly while tuning the controller gains so as to maintain the stability of the closed-loop system, especially in the case of a definite model structure with unknown and variable parameters.

#### 4.1. Controller design

In order to simplify the notation in the following derivations, the control inputs are redefined as

$$\begin{bmatrix} u_1 \\ u_2 \end{bmatrix} = \begin{bmatrix} 1 & 1 \\ 1 & -1 \end{bmatrix} \begin{bmatrix} I_1 \\ I_2 \end{bmatrix}.$$  \quad (9)

Since we have two control variables $u_1$ and $u_2$ for the controller specification and one output variable to be driven to zero, we choose to set $u_1$ to some constant. In other words, $u_1$ and $u_2$ (hence $I_1$ and $I_2$) are dependent control variables. Thus, with $u_1$ being set to a constant $C$ to limit the total input currents and $u_2$ being set as a differential input current of the two coils, we can completely specify the currents ($I_1$, $I_2$). Then, using Eq. (9), Eq. (8) can be rewritten into the state-space form

$$M \ddot{E} = AE + BU + W,$$  \quad (10)

where $E = z$ is the error state variable, $U = u_2$ is the control input, and $A = K(I_1 + I_2) = KC$, and $B = Kh/4$. Note that the external friction and high order terms left out due to linearization are all aggregated into the term $W$.

Now, Eq. (8) can be rewritten

$$D_B \ddot{E} = D_A E + U + v,$$  \quad (11)

where $D_A = 4C/h$, $D_B = 4M/K h$, and $v = 4W/K h$. However, due to the lack of knowledge of the system parameters, we assume $D_A$, $D_B$, and $v$ are unknown. Assume a sliding surface variable $S$ with the form

$$S = G_D \dot{E} + G_P E,$$  \quad (12)

where $G_D$, $G_P > 0$. In this application we try to regulate state error $E$ to zero, which simultaneously also regulates the derivative of $E$ to zero. To do this, if it can be proved that the sliding surface goes to zero in finite time, then $E$ and $\dot{E}$ are forced to zero exponentially. To relate the sliding surface to the motion dynamics, we find the time derivative of the sliding surface

$$\dot{S} = G_D \dot{E} + G_P E.$$  \quad (13)

As described in the previous section, an adaptive controller is used here, which is capable of estimating parameters of the system on-line while simultaneously controlling the system. After system parameters are estimated, Eq. (11) in the control command can be used. So, we substitute the controller design estimates acquired from the on-line estimator to derive the control input:

$$U = \hat{D}_B G_D^{-1}(-LS - G_P \dot{E}) + \hat{D}_A E - \hat{v} - N \text{sat}(S),$$  \quad (14)

where $L$, $N > 0$, $\hat{D}_B$, $\hat{D}_A$, and $\hat{v}$ are the estimates of $D_B$, $D_A$, and $v$, respectively, and sat(.) is the saturation function defined as

$$\text{sat}(S) = \begin{cases} 1 & S > \varepsilon \\ \frac{S}{|\varepsilon|} & \varepsilon \geq S \geq -\varepsilon \\ -1 & S < -\varepsilon. \end{cases}$$  \quad (15)

Thus, substituting Eq. (14) into Eq. (11), the system dynamic equation can be obtained

$$D_B G_D^{-1} \ddot{S} = -\hat{D}_A E + \hat{D}_B(-L G_D^{-1}S - G_P^{-1}G_P \dot{E})$$

$$+ \hat{v} - LD_B G_D^{-1}S - N \text{sat}(S),$$  \quad (16)

where the estimation errors are defined as $\hat{D}_B = D_B - \hat{D}_B$, $\hat{D}_A = D_A - \hat{D}_A$ and $\hat{v} = v - \hat{v}$. By applying appropriate gains $L$, $G_D$ and $G_P$, the state variable can quickly converge, forcing the errors to zero in a shorter period of time.

#### 4.2. Stability analysis

In the previous section, we have derived the closed-loop function in Eq. (16), which involves estimation errors. Now, with the help of an estimator based on adaptive control theory, we can calculate the estimates so that appropriate control commands are also derivable.

Define a candidate Lyapunov function $V$, which is positive definite:

$$V = \frac{1}{2} \left( D_B G_D^{-1}S^2 + I_1^{-1} \hat{D}_A + I_2^{-1} \hat{D}_B + I_3^{-1} \hat{v}^2 \right),$$  \quad (17)

where $I_1^{-1}$, $I_2^{-1}$ and $I_3^{-1}$ are all positive. The time derivative of the Lyapunov candidate function $V$ is:

$$\dot{V} = S(D_B G_D^{-1} \dot{S}) + \hat{D}_A I_1^{-1} \hat{D}_A + \hat{D}_B I_2^{-1} \hat{D}_B + \hat{v} I_3^{-1} \hat{v},$$  \quad (18)

along the solution trajectory of $S$ in Eq. (12). Then it can be rearranged by means of Eq. (20) into

$$\dot{V} = \hat{D}_A I_1^{-1} \hat{D}_A - SE + D_B (I_2^{-1} \hat{D}_B - L G_D^{-1} S^2)$$

$$- G_D^{-1} G_P \dot{E} + \hat{v} I_3^{-1} \hat{v} + N \text{sat}(S).$$  \quad (19)
Table 1
The system design constants

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad mass $M$</td>
<td>52 g</td>
</tr>
<tr>
<td>Pad length</td>
<td>200 mm</td>
</tr>
<tr>
<td>Total stroke</td>
<td>50 mm</td>
</tr>
<tr>
<td>NdFeB size</td>
<td>28<em>48</em>3 mm(W<em>L</em>H)</td>
</tr>
<tr>
<td>NdFeB Br</td>
<td>12 000 G</td>
</tr>
<tr>
<td>Turns of coils</td>
<td>500 turns</td>
</tr>
<tr>
<td>Resistance of coils</td>
<td>25 Ω</td>
</tr>
<tr>
<td>$C$</td>
<td>1 A</td>
</tr>
<tr>
<td>$K$</td>
<td>0.086</td>
</tr>
<tr>
<td>$h$</td>
<td>50 mm</td>
</tr>
<tr>
<td>Outside diameter of coils</td>
<td>50<em>70</em>50 mm(W<em>L</em>H)</td>
</tr>
<tr>
<td>Inside diameter of coils</td>
<td>30<em>50</em>50 mm(W<em>L</em>H)</td>
</tr>
</tbody>
</table>

Now, from Eq. (19), the adaptive laws can be derived as

$$\begin{align*}
\hat{D}_A &= -\hat{D}_A = -\Gamma_1 SE, \\
\hat{D}_B &= -\hat{D}_B = -\Gamma_2 (LG_D^{-1}S^2 + G_D^{-1}G_P \hat{S} \hat{E}), \\
\hat{v} &= -\hat{v} = \Gamma_3 S.
\end{align*}$$

If these equations hold, Eq. (19) will simplify to

$$\begin{align*}
\dot{V} &= -L G_D^{-1} S^2 - N \text{sat}(S) \\
&= \begin{cases} 
-L G_D^{-1} S^2 - N |S| \leq 0 & \text{if } |S| > \varepsilon \\
-L G_D^{-1} S^2 - N \frac{S^2}{|S|} \leq 0 & \text{if } |S| < \varepsilon
\end{cases}
\end{align*}$$

where we have used the fact that $G_D$, $N$ and $L$ are all positive constants. By the Lyapunov stability criteria [18], one can immediately conclude that all the signals, $S$, $D_A$, $D_B$, and $\hat{v}$, are bounded. Using arguments of Barbalat’s Lemma, we establish that $S \in L_2$, $\dot{S} \in L_\infty$, and the most important consequence, $|S(t)| \to 0$ as $t \to \infty$.

Due to convergence of $S$ to zero, it can be readily verified that the error state variable $E$ and its time derivative converge to zero asymptotically, which is the design goal for the EMA system controller.

Remark. From Eq. (21), it should be clear that the convergence of $|S(t)|$ does not depend on the saturation function, sat($S$). Instead, sat($S$) enhances the system’s robustness to some residual disturbance if its gain is sufficiently high. An example is when the term $v$ can be separated into $v + \tilde{v}$ where $v$ is a constant, and $\tilde{v}$ is some bounded term.

5. Experimental results

The experimental hardware, including the main body, sensor system, driver system and controller hardware will be described here. Fig. 4 shows the physical set-up. The constants in the system design are listed in Table 1. A number of experimental results, including the transient and the steady-state responses in different situations, are also provided in this section to demonstrate the performance of this system with the controller presented in Section 4. Based on these results, some important aspects of future research will be presented.

![Fig. 4. The physical system.](image)

![Fig. 5. Transient responses of the motion pad with the largest translation displacement.](image)

5.1. Hardware implementation

This sensor device is ILD 1400-50 manufactured by Micro-Optronic Technology (in Germany). With its rise time as low as 200 ps and an active range up to 50 mm (ROM) with a resolution of 0.1% of ROM, this sensor is fast enough and can cover all the traveling range needed for this application. The drivers are C0-502-001Q torque amplifiers manufactured by CMC Inc. (in the USA). They are linear drivers designed to be servo drivers for DC motors. The power is 250 W (5 A at ±50 V or 10 A at ±24 V). The microcomputer is a 733 MHz Pentium PC which accommodates real-time control.

5.2. System performance

For comparison, the experimental performances of a conventional PID controller for the EMA system are also shown here. The PID controller gains are defined by Ziegler-Nichols frequency response analysis [19]. The control parameters for the adaptive sliding mode controller and PID controller are set to $L = 2.5$, $G_P = 20$, $G_D = 4.7$, $\varepsilon = 0.0015$, $K_P = 20$, $K_I = 0.5$, and $K_D = 4.7$. Fig. 5 shows the transient responses to the largest initial transitional displacement range, which is 25 mm for this system. Although the PID controller (dotted line) and adaptive sliding mode controller (solid line) can both successfully control the EMA system up to some degree of precision following a trend similar to that of the transient
Fig. 6. Steady-state responses of the motion pad.

(a) Desired (dotted line) and actual (solid line) position.
(b) Position tracking error.

Fig. 7. Response to a 5 mm per step input control.

(a) Desired (dotted line) and actual (solid line) position.
(b) Position tracking error.

Fig. 8. Harmonic motion along the linear guide in the $z$ direction.

In order to further validate the robustness of this system with the adaptive sliding mode controller, a sudden pulse disturbance is applied to the motion pad in the $z$ direction. The plot in Fig. 10(a) shows that convergence to the steady state occurs in about 0.7 s for the adaptive sliding mode controller, while that for the PID controller takes about 1.5 s (Fig. 10(b)), or twice as long as the adaptive sliding mode controller.

The above experimental results, large-moving range, step-train response, sinusoidal motion, and disturbance test, have achieved the desired goals. In general, the adaptive sliding mode controller performs much better than the traditional PID controller. For the PID controller, there are significant larger settling time and steady-state errors for the PID controller. This is the salient feature of an adaptive sliding mode controller, which can adjust its controller on-line to adapt to changes in the environment. One can see that the error signals converge to their steady state values within about 1 s. Then, in Fig. 6, the final precision is within 5 µm in translation, which is about the limitation of the sensor device.

Fig. 7 shows a 5 mm per step input control test. The dotted line is the desired reference trajectory and the solid line is the actual output position. The resolving ability, i.e., repeatable accuracy, of the measuring device is about 5 µm. In Figs. 8 and 9, the motion pad performs a harmonic motion along the linear guide in the $z$ direction. The plot indicates that the motion pad moves well in the desired trajectory.
controller in several aspects of, for instance, short transient response, precision positioning, and high robustness.

6. Conclusions

As reported in this paper, a prototype of a precision positioning EMA system for a larger moving range has been built. The dynamics of the EMA system has been thoroughly analyzed and a complete model derived. The system is treated as a multi-input signal-output system, and an adaptive sliding mode controller has been designed and implemented using a microcomputer. The experimental results demonstrate that precision motion control is achieved.

The main objective of this work was to illustrate how an adaptive sliding mode controller can be applied to an EMA system. To be suitable for high-tech industries, the machine must be used in a high-quality clean room or in a vacuum environment as, for example, in the semiconductor industry. In order to meet this requirement, we want to use the magnetic levitation techniques instead of the linear guide in this EMA system. It is believed that more complex modeling and control problems will therefore be induced in the maglev EMA system.

Future work will concentrate on developing an improved nonlinear dynamic model with a coupled force (levitating force and propulsive force) and developing a new nonlinear controller design for the maglev EMA system.

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References


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