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Application of Magnetic Levitation in Active Mechanical Suspension Systems

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***Abstract*—Magnetic levitation has shown its potential in many engineering fields with promising future applications. This paper deals with the asymptotic tracking problem of desired reference position trajectories in an active mechanical suspension system using magnetic levitation foundations. A differential flatness-based output feedback controller is proposed for accomplishing this control objective using only position measurements. The electromagnetic circuit dynamics is considered for design of the control voltage to regulate the position of the mechanical system in accordance with the specified motion planning. A robust observer is also presented for real-time estimation of the unavailable signals of acceleration and velocity. The electric current is algebraically reconstructed through the estimated signals. The efficient performance of the proposed observer-control scheme is verified by computer simulation.**

***Keywords*—Active mechanical suspension, Magnetic levitation, Differential flatness, Observer.**

I. Introduction

The design and development of magnetic levitation-based devices and systems exhibit a growing trend due to their energy efficiency, high operation speeds, reliability, lifetime, reduction of pollution emissions and maintenance costs. Nowadays, technological innovations on practical applications of magnetic levitation can be found in high-speed passenger trains, magnetic bearings, vibration isolation systems, turbo machinery, machine tools, freight transportation, elevators, heart pumps, electric drives, toys and washing machines, with a promising future (see, e.g., [1-5] and references therein). Moreover, most of the available control schemes for magnetic levitation tasks are mainly based on simplified lineal mathematical models, which are only valid around certain nominal operation equilibrium points, and on the use of the electric current signal as control input variable but without considering the dynamics of the electromagnetic subsystem. In general, the research topic on magnetic levitation to robustly suspend a magnetic ball has been quite challenging over the years and a detailed survey seems to be out of the scope of this article. We thus refer the reader to some of the fundamental works in this area which have been helpful in the preparation of this paper [5-9].

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This paper deals with the magnetic levitation application to the problem of global stabilization and asymptotic tracking of an active linear mass-spring-damper mechanical suspension system. This configuration has the main advantage of saving electricity energy consumption when the system is located at a certain suspension position adjusted and established appropriately in the mechanical design of the system, as well as in the suitable smooth transference of the system to any desired position, taking clearly advantage of the spring force. The analysis is carried out for a mechanical system of one degree of freedom with a controllable electromagnet. Nevertheless, the employed design methodology could be extended to fully actuated or underactuated, differentially flat, mechanical systems with multiple degrees of freedom.

The differential flatness property [10] is used for the synthesis of an output feedback control scheme that efficiently accomplishes the pursued control objective. By means of differential flatness the analysis and design of a controller is greatly simplified, including tracking of reference trajectories in accordance with the desired motion planning for the system.

Our control design approach considers the electromagnetic circuit dynamics in the synthesis of a control voltage algorithm to regulate the position of the mechanical system in accordance with the specified motion planning. Since practical control implementations demand the use of a minimal number of sensors due to the goal of anyone of reducing costs, a lineal observation scheme is also presented for real-time estimation of the acceleration and velocity signals, which is robust with respect to state dependent disturbances, including possibly parametric uncertainty and external perturbations. In addition, the electric current signal is algebraically reconstructed through the estimated signals as a bonus thanks to the differential flatness property exhibited by the system. However, the electric current is not required by the controller proposed in this work. The efficient and robust performance of the proposed observation and control schemes is verified by computer simulation. The motion planning is specified to smoothly transfer the system from a rest position to another, but other reference trajectories can be implemented if required.

II. Active Mechanical Suspension System

Consider the mass-spring-damper suspension system shown in Fig. 1, where m, c and k are its mass, viscous damping and stiffness constant of the helical spring, respectively. Here, an electromagnet is used to induce an electromagnetic force f_{em} to control the position x of the mechanical system.

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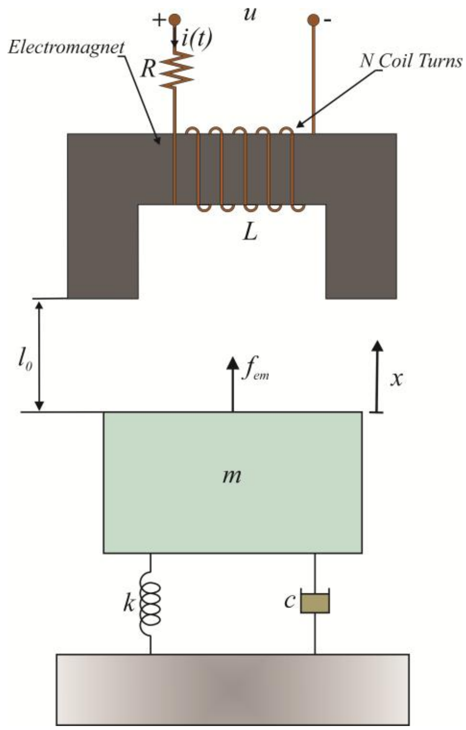

Figure 1. Active mechanical suspension system.

The mathematical model of the mechanical suspension system is described by

$$m\ddot{x} = -c\dot{x} - ky + f_{em}(z,i) \tag{1}$$

with [8]

$$f_{em} = \frac{k_m i^2}{2(z+a)^2}, \quad z \geq 0 \tag{2}$$

where the displacement x is measured from its static equilibrium position, in which the upward spring force exactly balances the downward gravitational force on the mass [11], z is the distance between the field source and the system's mass, a is a constant, which is commonly determined by experimentation, and k_m is the electromagnetic force constant.

The differential equation governing the electric current i is obtained by applying the Kirchoff's voltage law as [8]

$$L\frac{di}{dt} = -Ri + u \tag{3}$$

where R is the winding resistance plus any additional series resistance in the control circuit, L is the coil inductance, and u denotes the control voltage.

The mathematical model describing the dynamics of the mechanical system with an electromagnet is then given by equations (1)-(3)

$$\begin{aligned} m\ddot{x} &= -c\dot{x} - kx + \frac{k_m i^2}{2(l_0 - z_1 + a)^2} \\ L\frac{d}{dt}i &= -Ri + u \end{aligned} \tag{4}$$

where l_0 is the initial length between the core and the static equilibrium position ($u \equiv 0$) of the system.

Defining the state variables as $z_1 = x$, $z_2 = \dot{x}$ and $z_3 = i$, one obtains from (4) the state space description

$$\begin{aligned} \dot{z}_1 &= z_2 \\ \dot{z}_2 &= -\frac{k}{m}z_1 - \frac{c}{m}z_2 + \frac{k_m z_3^2}{2m(\delta - z_1)^2} \\ \dot{z}_3 &= -\frac{R}{L}z_3 + \frac{1}{L}u \\ y &= z_1 \end{aligned} \tag{5}$$

where $\delta = l_0 + a$.

The electromechanical system (5) is differentially flat, with flat output given by the position of the system $y=z_1$. Then, all the system variables can be differentially parameterized in terms of the flat output and a finite number of its time derivatives [10]. For this, the time derivatives up to third order for y are obtained as

$$\begin{aligned} y &= z_1 \\ \dot{y} &= z_2 \\ \ddot{y} &= -\frac{k}{m}y - \frac{c}{m}\dot{y} + \frac{k_m z_3^2}{2m(\delta - y)^2} \\ y^{(3)} &= -\frac{k}{m}\dot{y} - \frac{c}{m}\ddot{y} + \frac{k_m}{m(\delta - y)^2}\left[\frac{\dot{y}}{(\delta - y)} - \frac{R}{L}\right]z_3^2 \\ &\quad + \frac{k_m}{mL(\delta - y)^2}z_3 u \end{aligned} \tag{6}$$

Therefore, the differential parameterization results in

$$\begin{aligned} z_1 &= y \\ z_2 &= \dot{y} \\ z_3 &= \sqrt{\frac{2m}{k_m}\left(\frac{k}{m}y + \frac{c}{m}\dot{y} + \ddot{y}\right)}(\delta - y) \\ u &= \frac{L(\delta - y)}{\sqrt{\frac{2k_m}{m}\left(\ddot{y} + \frac{k}{m}y + \frac{c}{m}\dot{y}\right)}}\left[y^{(3)} + \frac{c}{m}\ddot{y} + \frac{k}{m}\dot{y}\right. \\ &\quad \left. -2\left(\frac{\dot{y}}{\delta - y} - \frac{R}{L}\right)\left(\frac{k}{m}y + \frac{c}{m}\dot{y} + \ddot{y}\right)\right] \end{aligned} \tag{7}$$

The flat output y then satisfies the following input-output differential equation:

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$$y^{(3)} = -\frac{k}{m}\dot{y} - \frac{c}{m}\ddot{y} + 2\left[\frac{\dot{y}}{\delta - y} - \frac{R}{L}\right]\left(\frac{k}{m}y + \frac{c}{m}\dot{y} + \ddot{y}\right) + \frac{1}{L(\delta - y)}\sqrt{\frac{2k_m}{m}\left(\ddot{y} + \frac{k}{m}y + \frac{c}{m}\dot{y}\right)}u \quad (8)$$

In the next section, the structural property of differential flatness will be used to design a controller to perform closed-loop trajectory tracking tasks to smoothly transfer the mechanical system from an operation position to another.

III. Differential Flatness Control

From the nonlinear differential equation (8), we propose the following differential flatness controller for asymptotic tracking tasks of some desired reference position trajectory $y^*(t)$:

$$u = \frac{1}{b}(v - \phi) \quad (9)$$

with

$$\begin{aligned} v &= \left(y^*\right)^{(3)}(t) - \alpha_2\left[\ddot{y} - \ddot{y}^*(t)\right] - \alpha_1\left[\dot{y} - \dot{y}^*(t)\right] - \alpha_0\left[y - y^*(t)\right] \\ \phi &= -\frac{k}{m}\dot{y} - \frac{c}{m}\ddot{y} + 2\left[\frac{\dot{y}}{\delta - y} - \frac{R}{L}\right]\left(\frac{k}{m}y + \frac{c}{m}\dot{y} + \ddot{y}\right) \\ b &= \frac{1}{L(\delta - y)}\sqrt{\frac{2k_m}{m}\left(\ddot{y} + \frac{c}{m}\dot{y} + \frac{k}{m}y\right)} \end{aligned}$$

The use of this controller yields the closed loop dynamics for the tracking error, $e(t) = y - y^*(t)$,

$$e^{(3)} + \alpha_2\ddot{e} + \alpha_1\dot{e} + \alpha_0 e = 0 \quad (10)$$

Therefore, selecting the design parameters α_i, i = 1, 2, 3, such that the characteristic polynomial associated with (10) be Hurwitz, one can guarantee that the error dynamics is globally asymptotically stable.

A possible inconvenience that could present the controller (9) is its request of on-line measurements of the acceleration and velocity signals. Fortunately, the electromechanical system (5) is completely observable from the flat output y. Thus, we will also propose a robust lineal observation scheme to estimate to those unavailable signals, which can be used in case of the lack of acceleration and velocity sensors. Note that the acceleration signal can also be reconstructed from measurements of electric current, position and velocity by using the third equation of (6).

IV. Design of a robust linear observer

In the design process of the observation scheme, consider the perturbed mathematical model

$$y^{(3)} = b^*(t)u + \xi(t) \quad (11)$$

with

$$b^*(t) = \frac{1}{L(\delta - y)}\sqrt{\frac{2k_m}{m}\left[\ddot{y}^*(t) + \frac{c}{m}\dot{y}^*(t) + \frac{k}{m}y\right]}$$

where $\xi(t)$ is considered as an unknown state-dependent disturbance input signal, which includes $\phi(t)$ and deviations of $b^*(t)$ with respect to the actual gain $b(t)$. Note that $\xi(t)$ could also include small perturbations due to parametric uncertainty and unknown external forces.

It is assumed that the disturbance signal $\xi(t)$ can be locally described by a family of Taylor polynomials of (r-1)th degree as [12]

$$\xi(t) \approx \sum_{i=0}^{r-1} p_i t^i \quad (12)$$

where all the coefficients p_i are completely unknown.

An approximated extended state space local model for the perturbed dynamics (11) is then given by

$$\begin{aligned} \dot{\eta}_1 &= \eta_2 \\ \dot{\eta}_2 &= \eta_3 \\ \dot{\eta}_3 &= b^*(t)u + \xi_1 \\ \dot{\xi}_1 &= \xi_2 \\ \dot{\xi}_2 &= \xi_3 \\ &\vdots \\ \dot{\xi}_{r-1} &= \xi_r \\ \dot{\xi}_r &= 0 \end{aligned} \quad (13)$$

where $\eta_1 = y$, $\eta_2 = \dot{y}$, $\eta_3 = \ddot{y}$, $\xi_1 = \xi$, $\xi_2 = \dot{\xi}$, $\xi_3 = \ddot{\xi}, \cdots,$ $\xi_r = \xi_i^{(r-1)}$.

Based on the extended model (13), we propose the Luenberger-like observer

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$$
\begin{aligned}
\dot{\eta}_1 &= \eta_2 + \beta_{r+2}\left(\eta_1 - \eta_1\right) \\
\dot{\eta}_2 &= \dot{\eta}_3 + \beta_{r+1}\left(\eta_1 - \eta_1\right) \\
\dot{\eta}_3 &= b^*(t)u + \xi_1 + \beta_r\left(\eta_1 - \eta_1\right) \\
\dot{\xi}_1 &= \xi_2 + \beta_{r-1}\left(\eta_1 - \eta_1\right) \\
\dot{\xi}_2 &= \xi_3 + \beta_{r-2}\left(\eta_1 - \eta_1\right) \\
&\vdots \\
\dot{\xi}_{r-1} &= \xi_r + \beta_1\left(\eta_1 - \eta_1\right) \\
\dot{\xi}_r &= \beta_0\left(\eta_1 - \eta_1\right)
\end{aligned}
\tag{14}
$$

The estimation error dynamics is then obtained as

$$
\begin{aligned}
\dot{e}_1 &= -\beta_{r+2}e_1 + e_2 \\
\dot{e}_2 &= -\beta_{r+1}e_1 + e_3 \\
\dot{e}_3 &= -\beta_r e_1 + e_{p_1} \\
\dot{e}_{p_1} &= -\beta_{r-1}e_1 + e_{p_2} \\
\dot{e}_{p_2} &= -\beta_{r-2}e_1 + e_{p_3} \\
&\vdots \\
\dot{e}_{p_{r-1}} &= -\beta_1 e_1 + e_{p_r} \\
\dot{e}_{p_r} &= -\beta_0 e_1
\end{aligned}
\tag{15}
$$

where $e_1 = \eta_1 - \eta_1$, $e_2 = \eta_2 - \eta_2$, $e_3 = \eta_3 - \eta_3$, $e_{p_k} = \xi_k - \xi_k$, $k = 1, 2, \cdots, r$.

In fact, the estimation errors can be parameterized in terms of the output error e_1 and a finite number of its time derivatives as follows

$$
\begin{aligned}
e_2 &= \dot{e}_1 + \beta_{r+2}e_1 \\
e_3 &= \ddot{e}_1 + \beta_{r+2}\dot{e}_1 + \beta_{r+1}e_1 \\
e_{p_1} &= e_1^{(3)} + \beta_{r+2}\ddot{e}_1 + \beta_{r+1}\dot{e}_1 + \beta_r e_1 \\
e_{p_2} &= e_1^{(4)} + \beta_{r+2}e_1^{(3)} + \beta_{r+1}\ddot{e}_1 + \beta_r\dot{e}_1 + \beta_{r-1}e_1 \\
e_{p_3} &= e_1^{(5)} + \beta_{r+2}e_1^{(4)} + \beta_{r+1}e_1^{(3)} + \beta_r\ddot{e}_1 + \beta_{r-1}\dot{e}_1 + \beta_{r-2}e_1 \\
&\vdots \\
e_{p_r} &= e_1^{(r+2)} + \beta_{r+2}e_1^{(r+1)} + \beta_{r+1}e_1^{(r)} + \beta_r e_1^{(r-1)} + \beta_{r-1}e_1^{(r-2)} \\
&\quad + \beta_{r-2}e_1^{(r-3)} + \cdots + \beta_1 e_1
\end{aligned}
\tag{16}
$$

The characteristic polynomial associated with (15) is given by

$$
\begin{aligned}
p(s) &= s^{r+3} + \beta_{r+2}s^{r+2} + \beta_{r+1}s^{r+1} + \beta_r s^r \\
&\quad + \beta_{r-1}s^{r-1} + \cdots + \beta_1 s + \beta_0
\end{aligned}
\tag{17}
$$

which is completely independent of any coefficients p_i of the Taylor polynomial expansion of the disturbance signal $\xi(t)$.

The design parameters β_i are then chosen so that the polynomial (17) is a Hurwitz polynomial. Thus, estimates of the acceleration and velocity signals are obtained by using the observer (14). Moreover, the electric current signal can be computed from (7) as

$$
\hat{z}_3 = \sqrt{\frac{2m}{k_m}\left(\frac{k}{m}y + \frac{c}{m}\eta_2 + \eta_3\right)}\left(\delta - y\right) \tag{18}
$$

Hence, we propose the following output feedback controller for desired reference position trajectory tracking tasks in terms of the estimated signals:

$$
u = \frac{1}{b^*}\left(\hat{v} - \phi\right) \tag{19}
$$

with

$$
\begin{aligned}
\hat{v} &= \left(y^*\right)^{(3)}(t) - \alpha_2\left[\eta_3 - \ddot{y}^*(t)\right] \\
&\quad - \alpha_1\left[\eta_2 - \dot{y}^*(t)\right] - \alpha_0\left[y - y^*(t)\right] \\
\phi &= -\frac{k}{m}\eta_2 - \frac{c}{m}\eta_3 + 2\left[\frac{\eta_2}{\delta - y} - \frac{R}{L}\right]\left(\frac{k}{m}y + \frac{c}{m}\eta_2 + \eta_3\right) \\
b^*(t) &= \frac{1}{L(\delta - y)}\sqrt{\frac{2k_m}{m}\left[\ddot{y}^*(t) + \frac{c}{m}\dot{y}^*(t) + \frac{k}{m}y\right]}
\end{aligned}
$$

I. Simulation Results

Some computer simulations were performed in an active mechanical suspension system with an electromagnet characterized by the parameters described in Table 1 [9]. The values of the viscous damping, stiffness and initial length between the core and the static equilibrium position were chosen as: k =100 N/m, c =1 Ns/m and l_0 = 0.02 m, respectively.

TABLE I. SYSTEM PARAMETERS

m = 0.54 kg	L = 0.8052 H
R = 11.88 Ω	k_m = 0.0015 N m^2/A^2
a = 0.008114 m	

The design parameters of the controller were selected to have the following third order characteristic polynomial for the closed-loop tracking error dynamics:

$$
p_c(s) = \left(s + p_1\right)\left(s^2 + 2\zeta\omega_n s + \omega_n^2\right)
$$

with $p_1 = \omega_n = 20$ rad/s and $\zeta = 0.7071$.

The perturbation signal $\xi(t)$ was modeled as a second order time polynomial. Therefore, the characteristic polynomial for the sixth order resulting observation error dynamics was set to be of the following form:

$$
p_o(s) = \left(s^2 + 2\zeta_o\omega_o s + \omega_o^2\right)^3
$$

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with $\omega_o = 500$ rad/s and $\zeta_o = 5$.

Fig. 2. depicts the efficient performance of the controller (19) using estimates of the velocity and acceleration signals provide by the implementation of the observer (14). The effective tracking of the reference position trajectory $y^*(t)$ can be clearly observed. This trajectory was specified to smoothly transfer the mechanical system from the rest position to the desired position of 0.01 m in approximately 2 seconds.

On the other hand, Figs. 3 and 4 show the closed loop signals of the electric current and the control voltage required to perform the motion planning.

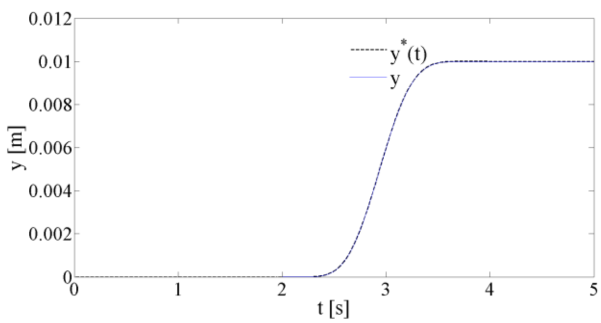

Figure 2. Closed loop position trajectory tracking response.

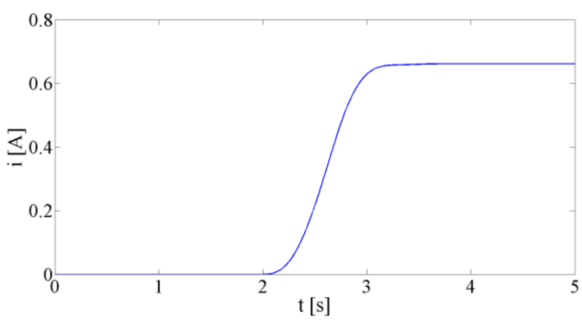

Figure 3. Closed loop response of the electric current signal.

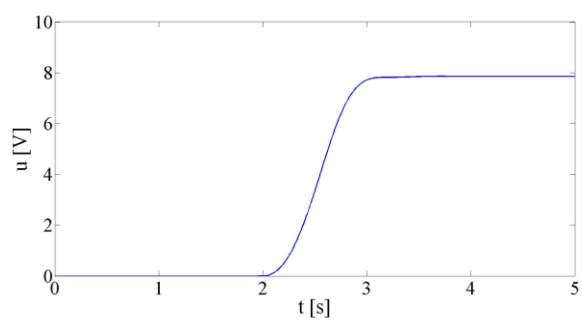

Figure 4. Control voltage signal applied to the electromagnet.

II. Conclusions

In this work we have proposed an output feedback controller for global stabilization and asymptotic tracking tasks of some position desired reference trajectory for an active linear mass-spring-damper mechanical suspension system. The dynamics of the electromagnetic circuit was included for the synthesis of a control voltage algorithm to efficiently regulate the mechanical system toward the desired nominal operation reference trajectories. In addition, a robust linear observation scheme was proposed to estimate in real-time the acceleration and velocity signal in order to avoid the use of more than one sensor for the control implementation. An expression to algebraically reconstruct the electric current signal was also presented. Some computer simulation results were provided to show the efficient and robust performance of the observer-control scheme. Future work will be oriented to verify experimentally the robustness of the controller and the observer proposed in this paper with respect to parametric uncertainty and external perturbation forces. Moreover, a comparative analysis on energetic efficiency will be performed between the configuration of the presented mechanical suspension system and the traditional problem of suspending a magnetic mass.

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