Active magnetic bearing: a new step for Model-free control

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Abstract—Model-free control is applied to the stabilization of an active magnetic bearing, which is quite an important industrial device. Experimental results are compared to those obtained via other control techniques, showing at least on-par performance with this very straightforward approach.

I. INTRODUCTION

Most uses of active magnetic bearings (AMB) are found in industrial applications. In particular, they find their way into high-speed rotating equipment such as turbines, machine tools, vacuum pumps or compressors. Another significant use is flywheel-based energy-storage devices, in applications ranging from satellites to biomedical equipment [7]. Indeed, magnetic bearings have many advantages over their conventional counterparts:

• Thanks to contactless, mostly frictionless operation, they can support loads with very high rotational speeds.
• Since they do not require lubrication, they are suitable for environments where excluding contamination is key, such as clean rooms, or where efficient lubrication is a problem, such as deep vacuums.

The extension of EARNshaw’s theorem to magnetic forces shows it is impossible to design stable positioning systems by the mere use of permanent ferromagnetic magnets. While passive solutions based on diamanetic materials exist [23], they are uncommon in practice. This is why most applications implement active magnetic bearings (AMBs).

Active Magnetic Bearings are electromagnet-based and require an active control system to operate correctly [26]. They operate as follows. Each control axis (see Figure 1) features two electromagnets and a position sensor measuring rotor displacement. Each electromagnet generates a force which is proportional to the square of its coil current, and inversely proportional to the square of the air gap between its stator and the supported shaft. Through modulation of these forces, it is possible precisely position the shaft along the control axis. A centering device able to position a shaft along two degrees of freedom is obtained by combining two control axes, to completely maintain a shaft in levitation, two centering devices and a longitudinal AMB are necessary. Obviously, the nature of the forces involved introduces important nonlinearities in the physics of an axis. In addition, AMBs being very fast electromagnetic devices, major real-time constraints have to be considered when designing an appropriate control system.

Control of magnetic levitation systems, are the subject of numerous publications owing to their industrial importance (see e.g [2]–[6], [10], [17], [19]–[21], [25], [28], [31]), which rely on a wide array of modern control techniques. What makes this control problem hard stems mainly from its complex model.

The purpose of this paper is thus to apply the “model-free control” approach to that problem. Introduced by [11], [12], it has already been used successfully to solve numerous control problems spanning diverse application areas [14]. Moreover, for each real studied cases, local approximation model could be first order. Specificities of magnetic bearings – most importantly negligible friction – introduce novelty from the point of view of possible local model choice and allows us to contribute a new, simpler and more natural formulation of that technique.

This paper is organized as follows. First, section II describes the new formulation of model-free control. Then, its application to magnetic bearings, including lab experiments and a performance comparison with two different control techniques are discussed in section III. Finally, some insight into future developments is given in section IV.

II. MODEL-FREE CONTROL

A. General principles

Model-free control is based on continually estimating a numerical difference between a expect local model (can be seen as a virtual linear tangent system) and the real behavior of a system from the sole knowledge of its input-output signals. Let (1) be the unknown differential equation, possibly nonlinear, describing the input-output behavior with
input $u$ and output $y$

$$E(y, \dot{y}, \ldots, y^{(a)}, u, \dot{u}, \ldots, u^{(b)}) = 0$$  (1)

Associate to equation (1) the following equivalent equation representing the behavior of our closed loop system

$$F + \alpha u - K_P e - K_I \int e - K_D \dot{e} = 0$$  (2)

where

- $e = y - y^*$ is the difference between the output $y$ and the desired output $y^*$,
- $K_P$, $K_I$ and $K_D$ are tuning parameters,
- $F$, subsuming (1), must be numerically estimated in real time,
- $\alpha$, non-physical parameter must be chosen by the practitioner so that $F$ and $\alpha u$ have the same order of magnitude.

Let $\hat{F}$ be an estimate of $F$. The feedback control is chosen with the following form

$$u = -\frac{\hat{F}}{\alpha}$$  (3)

Together, (2) and (3) are called intelligent PID controller by the authors, or $iPID$ because of the form of (2) and by analogy with [11], [12], [14]. It leads to

$$K_P e + K_I \int e + K_D \dot{e} = 0,$$  (4)

which is stable for a suitable choice of gains.

**Remark 1:** This formulation does not require the discussion of the order of a local model, as in the previous propositions of the authors (see [11], [12], [14]). This is a substantial advantage.

**Remark 2:** If necessary (2) can be replaced by

$$F + \alpha u - K_P e - \Phi(e) = 0,$$

where $\Phi$ is a functional of $e$. Hence, model-free control allows a wide array of possible control laws.

**Remark 3:** Setting $K_D = 0$ in (2) yields an intelligent PI or $iPI$. This is by far the most common case in practice (see [14]).

**B. Estimation of $F$ and effective construction of $u$**

1) **General remark:** according to (2), estimating $F$ is straightforward (see the following paragraph). Even if it may be necessary to denoise $u$ and $y$, elementary filters have been sufficient until now. If $K_D \neq 0$ in (2), estimating $F$ requires an estimate of the derivative $\dot{y}$, i.e., the derivative of a noisy signal. Basic derivative filters are usually sufficient.

**Remark 4:** The case of higher noises has been successfully handled in practical settings [30] through algebraic techniques detailed in [13], [22].

2) **Used construction:** according to equation (2), authors choose to construct $\hat{F}$ as follow

$$\hat{F} = -\alpha u^* + K_P e + K_I \int e + K_D \dot{e}$$  (5)

with $u^*$ a denoised value of the control input. Combining with (3) the control function yields

$$u = -\frac{1}{\alpha} (-\alpha u^* + K_P e + K_I \int e + K_D \dot{e}),$$  (6)

Descritizing this equation yields

$$u_k = N(z).u_k - \frac{1}{\alpha}(K_P e_k + K_D D(z).e_k),$$  (7)

where $z$ is the delay operator, $N(z)$ a discrete filter applied to the control input and $D(z)$ discrete derivative operator. $N(z)$ function is twofold. It serves as a denoising filter for the control input and prevents the apparition of an algebraic loop in the control formulation. $K_i$ is zero as explained below. As already pointed out by the authors [1], model-free control leads to very similar forms of classical PID controllers but without having to use for setting a linear approximation of the system (at most the static gain).

**III. MAGNETIC BEARINGS**

**A. A simplified model**

The model used for simulations considers a single axis and is thus monovariable. Focusing on axis $y$ (Fig. 2), the radial acceleration of the rotor can be written as

$$m\ddot{y} = F_{yp} + F_{ym} + F_p,$$  (8)

where $F_{yp}$ and $F_{ym}$ are the coil-generated magnetic forces and $F_p$ an additive, constant disturbance such as gravity. Neglecting the effects of magnetic saturation and hysteresis, it follows

$$F_{yp} = -\frac{\lambda_1 i_{zp}^2}{2(e_0 - y)^2}$$ and

$$F_{ym} = -\frac{\lambda_2 i_{zm}^2}{2(e_0 + y)^2}$$  (9)

where $e_0$ is the nominal gap between the shaft and the coils and parameters $\lambda_1$ and $\lambda_2$ depend on the electromagnet and shaft geometries. Since each axis consists of two symmetrical actuators, the latter are both assumed equal to the single parameter $\lambda_y$. Combining equations (8) and (9), yields a model which is not linearizable at the origin – where the shaft
is centered and currents are zero (see [4]). However, a model suitable for linear analysis and control design can be obtained by applying a constant premagnetization bias current $I_0$ to both coils. The constant magnetic flux present in the two actuators eliminates the flux creation time, which leads to an almost linear response of the shaft for small current variations around $I_0$. Using a bias current has one major flaw though. Since the two coils are always active, their energy consumption is much higher. Nonlinear operation of an AMB is thus more efficient, as only one of the coils is active at any time.

In the latter operating mode, currents $i_{yp}$ and $i_{ym}$ are mutually exclusive and can be expressed as a function of a single virtual current $i_y$:

$$i_{ym} = \begin{cases} -i_y & \text{if } i_y < 0 \\ 0 & \text{otherwise} \end{cases} \quad \text{and} \quad i_{yp} = \begin{cases} i_y & \text{if } i_y > 0 \\ 0 & \text{otherwise} \end{cases}.$$  \hspace{1cm} (10)

Equation (10) implies that either $F_{yp}$ or $F_{ym}$ is zero at any given time. Equation (9) then yields

$$F_{yp} + F_{ym} = \frac{\lambda_y \text{sign}(i_y)i_y^2}{2(e_0 - \text{sign}(i_y)y)^2},$$  \hspace{1cm} (11)

which gives the simulation model through substitution into (8).

**B. Model parameters**

Values of the physical parameters $m$, $\lambda_y$ and $e_0$ are given in table I. The setpoint follows a low-pass-filtered, 10-Hertz square signal. As magnetic bearings are subject to minimal damping, the chosen control law is an $iPD$ controller which, from [1], matches a classical $PID$ controller. Its parameters are $K_I = 0$, $\alpha = .9$, $K_P = 14692$ and $K_D = 266$. These values are the same as those of the nonlinear PID use for comparison in real experiment.

In the simulation, the constant disturbance $F_p$ changes sign at $t = .25$ s. A low amplitude noise (less than $2 \times 10^{-6}$) is added to the output. The simulation results, obtain with the simplified model and the control law (7) given on Figures 3 and 4, show the efficacy of the control law without any tuning after the choose of the parameter values.
C. Experimentation

In contrast to the simulations presented above, the experiment deals with a complete bearing where all degrees of freedoms are driven simultaneously by model-free control laws.

The test-bench used for these experiments is a laboratory AMB supplied by MECOS-TRAXLER AG, model miniVS (Figure 5). It features a magnetic suspension unit comprising a rotor, two active centering devices and an active longitudinal bearing, whose parameters are summarized in Table I.

<table>
<thead>
<tr>
<th>TABLE I</th>
<th>EXPERIMENTAL AMB PARAMETERS</th>
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<tbody>
<tr>
<td>Parameter</td>
<td>Variable</td>
</tr>
<tr>
<td>Rotor mass</td>
<td>m</td>
</tr>
<tr>
<td>Maximum coil current</td>
<td>$I_{max}$</td>
</tr>
<tr>
<td>Nominal gap</td>
<td>$e_0$</td>
</tr>
<tr>
<td>Coil parameter</td>
<td>$\lambda_{g,e}$</td>
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<tr>
<td>Acquisition period</td>
<td>$T_m$</td>
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<td>except imMod control</td>
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<td>ADC resolution</td>
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<td>Input numeric filter</td>
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<tr>
<td>time constant</td>
<td>$t_o$</td>
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<tr>
<td>Control laws period</td>
<td>$T_e$</td>
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</tbody>
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All five control axes are driven by a single PC running real-time Simulink code. Inputs are sampled at a frequency higher than that of the control law to allow for efficient filtering. Two series low-pass filters with a time constant $t_o$ are used to this end.

In order to assess the performance of the model-free approach, a total of three control laws have been implemented:

- The model-free control described above. All axes are assumed independent.
- A global, Euler-Lagrange model-based nonlinear control law [9]. PID controllers are tuned to output desired values for the second derivatives of the generalized coordinates of the model according to chosen closed-loop dynamics. The third order of this closed loop is form as a product of a first order (time constant: 0.0045 s) and a second order (angular frequency: 180 $rd.s^{-1}$ and damping factor: 1.1). Full model equations are then used to compute the matching currents to apply to the plant.
- A discrete nonlinear controller [3] where desired currents are obtained through a table-based numerical inversion of the behavior of an axis as a function of the desired shaft position at the next time step.

Let $y_1$, $z_1$ and $y_2$, $z_2$ be the positions of the shaft ends. $y_1$, $z_1$ and the $x$ axis are kept at the nominal gap by an IPD controller. $y_2$ is made to follow a square reference signal varying from zero to $e_0/8$ at a frequency of 2 Hz. Likewise, $z_2$ is made to follow a sinusoidal reference signal varying from $-e_0/8$ to $e_0/8$ at the same frequency. Authors chose this reference signal as good indicators to interpret the performance of control laws. Both $y$ and $z$ axes are subject to a perturbation due to gravity, while the $x$ axis is to be stabilized close to the nonlinearizable origin of the model.\(^1\)

As shown on Figures 13 and 14, keeping the $x$-axis at the origin is hard as it is the point where the coil currents are zero. The time needed to establish a current in each coil induces a slight delay that prevents instantaneous reaction from the controller. In contrast, this phenomenon does not occur on the $y$ and $z$ axes since a nonzero current is always flowing through the coils to oppose gravity.

Figure 7 shows the value of the $y_2$-axis position, featuring both the reference square signal and a desired output signal obtained through low-pass filtering of the former. The matching control signal is shown on Figure 8. A significant noise level can be observed as the input filter does not completely cancel measurement noises and the derivative term $D$ of the controller is a rough approximation.

The control signal itself is shaped by the combined influence of three elements:

- Since both $y$ and $z$ axes are directed towards the ground, negatives currents are needed to compensate for gravity.
- The value of the current necessary to compensate for gravity is $-2.07 A$ for the nominal gap.

\(^1\)As the table on which the test-bench resides is not perfectly horizontal, this axis still experiences some gravity. This explains the non-zero average of the control signal shown on Figure 14.
The offset relative to the nominal gap. In the experiment, the rotors move away from the electromagnet which maintains its position. Hence, the current has to increase with the gap to keep the magnitude of the force opposing gravity. The model error estimator $\hat{F}$ perfectly fulfills its function and compensates for this nonlinearity.

Figure 9 details a single step response of the $y_2$-axis. The system response is slightly ahead of the reference signal $y^*$ it should be following. Indeed, $\hat{F}$ compensates for errors between the real system and that on which the control law is based with a slight delay. This phenomenon also depends on the value of the $\alpha$ parameter, here 2.0$^2$ for both the $y$ and $z$ axes.

Figure 10 details a single step response of the $y_2$-axis of all three control laws. They all have been tuned to feature the same response time.

Compared to the nonlinear global PID control, the model-free controller also eliminates the steady state error due to gravity but without any overshoot. Its behavior is also almost indistinguishable from that of the model inversion-based controller. Moreover, it achieves this result with a much lighter computing cost, keeping in mind the model-based controller. Moreover, it achieves this result with a much lighter computing cost, keeping in mind the model inversion-based controller.

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The model-free controller yields as good results than the nonlinear PID (The difference is that the PID has not filtered reference of the step signal of the input) and matches the laboratory benchmark model inversion-based controller with a much lower computing cost than the two other control laws. Still, the level of noise obtained when trying to stabilize the shaft at a point where the current is zero and the model is nonlinearizable shows a tight coupling of the quality of the results and the $\alpha$ tuning parameter. This shows a possible path towards improvements of this control approach. As the proposed control scheme is not directly connected to the AMB model, it remains valid as well for this type of devices at different scales than other systems.

IV. CONCLUSION AND FURTHER WORK

The model-free controller yields as good results than the nonlinear PID (The difference is that the PID has not filtered reference of the step signal of the input) and matches the laboratory benchmark model inversion-based controller with a much lower computing cost than the two other control laws. Still, the level of noise obtained when trying to stabilize the shaft at a point where the current is zero and the model is nonlinearizable shows a tight coupling of the quality of the results and the $\alpha$ tuning parameter. This shows a possible path towards improvements of this control approach. As the proposed control scheme is not directly connected to the AMB model, it remains valid as well for this type of devices at different scales than other systems.

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