Automatic Initial Levitation with Active Magnetic Bearings

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***Abstract*— Active magnetic bearings (AMBs) require a feedback controller for stable operation. This paper describes a procedure which allows for fully automated initial levitation with no knowledge about the AMB system. Combined with some adaptive or self-tuning techniques, bearings could be setup automatically with minimal human interaction, without calibration or tuning.**

Based on the ideal and linearized model of a single mass one degree of freedom levitated body, the criterion for a stable suspension is reviewed. A step-by-step description outlines how the needed parameters are found which lead to a stabilizing controller. A more general, hybrid model including sensor and amplifier dynamics as well as a discrete-time PID controller is used to compare the regions of stabilizing control parameters to the ideal case. The presentation of experimental data and a discussion conclude this paper.

I. Introduction

Active Magnetic Bearings (AMBs) are capable of adjusting the force applied to the supported structure (typically a rotor) within a limited amplitude and bandwidth. Equipped with position sensors and a feedback controller, AMBs can imitate the behavior of physical systems such as a spring-damper suspension or more complex structures which are able to suspend flexible rotors. Control theory provides numerous tools to design such controllers with the desired properties and performance. However, most of these tools require a plant model and relatively precise knowledge of the AMBs, sensors and the rotor. So called robust controllers tolerate model inaccuracies but compromise performance. Adaptive Control features adaptation to unknown model parameters. However, typical AMB models do not account for events such as the rotor hitting the backup bearings. Therefore, a stable controller is required as a starting point for adaptation. Ironically, the properties of an AMB system can easily be found by experiment, once a stabilizing controller is in place.

In this paper, we propose a method to automatically establish initial levitation with only the knowledge of the maximum bearing current and the sampling rate or controller time interval. Sensor calibration and controller tuning can be automatically performed on-site, without any human interaction. This further helps establishing low-cost bearing series.

For the so-called *Auto Levitator*, parameters such as the bearing's negative stiffness k_x, force-per-current coefficient k_i, sensor gain and offset as well as rotor geometry, mass and moments of inertia are all unknown. Previously, Loesch et al. [2] proposed a way to acquire rotor parameters and a stabilizing controller by a simple experiment, which still requires knowledge of some bearing parameters. Methods for online tuning of a given, stabilizing controller to meet the required performance have been presented in [3], [4]. The entire start-up configuration and tuning could be automated when combined with this new method.

Digital control circuits, necessary to stabilize a levitated body, can in most cases easily accommodate some additional functionalities. Therefore, no extra hardware is required to implement the *Auto Levitator*. The block

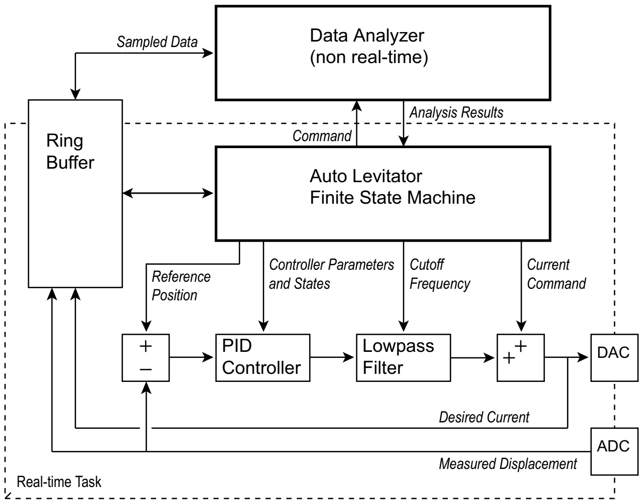

Fig. 1. Software block diagram

diagram in Fig. 1 shows how the *Auto Levitator* interacts with a typical AMB controller. The first part of the added code represents a finite state machine which performs the desired experiments by injecting perturbation currents and manipulating controller and filter parameters. The finite state machine is updated at the controller sampling rate and has to be scheduled for real-time execution. The second part of the software extension, labelled Analyzer, does not require real-time execution and is synchronized with the real-time control by a hand-shaking protocol. The analyzer performs time consuming signal processing tasks on buffered experimental data as requested by the finite state machine, for controller tuning. The following paragraphs discuss the automatic experiments performed by the finite state machine and the theoretical basis to establish initial levitation.

II. Parameter Search for Initial Levitation

A stabilizing controller is needed to establish levitation. Consider a single mass levitated body with one degree of freedom such as a balanced beam or a thrust bearing (see section IV). Let's further assume that there are two opposing electromagnets operating with the currents $i_{bias} + i_c$ and $i_{bias} - i_c$ with i_c being the control current. (If the bias flux is provided by permanent magnets, then i_{bias} is zero, and the opposing coils can be connected in series.) In this case, the commonly used linearized model for active magnetic bearing systems [1] describes the plant adequately and the dynamic equation can be written as

$$m\ddot{x} = i_c k_i + x k_x + F_d \quad (1)$$

with the mass m, the displacement x and a disturbance force F_d, e.g. resulting from the weight of the suspended body. (For the balanced beam test rig, replace the mass by the beam's moment of inertia, and the displacement x by the tilting angle; the equation, however, doesn't change for small angles.) The coefficient k_i is the *force-per-current* constant and k_x is the *force-per-displacement* constant. Applying a state feedback law as shown in Fig 2, it becomes intuitively clear how the open-loop unstable AMB can be stabilized. (Note that this state feedback corresponds to a PD controller (11) with $K_P = -k_1/k_{ss}$ and $K_D = -k_2 f_{sp}/k_{ss}$; sensor gain k_{ss} [V/m], sampling frequency f_{sp} [Hz].) With $F_d = 0$ and $k_2 = 0$, the forces acting on the floating mass are the destabilizing positive feedback xk_x and the proportional control action xk_1k_i. By selecting k_1 such that the inequality

$$k_x + k_1 k_i < 0 \quad (2)$$

is satisfied, a *restoring* force proportional to the displacement x will be applied to the floating body. By choosing any negative number for k_2, damping (a force opposing the motion) is added to the closed loop system. Thus the floating body will come to a rest at the origin $x = 0, \dot{x} = 0$ or, if $F_d \neq 0$ at $x = -F_d/(k_1k_i + k_x), \dot{x} = 0$. Damping and stiffness of the suspension can be chosen as desired.

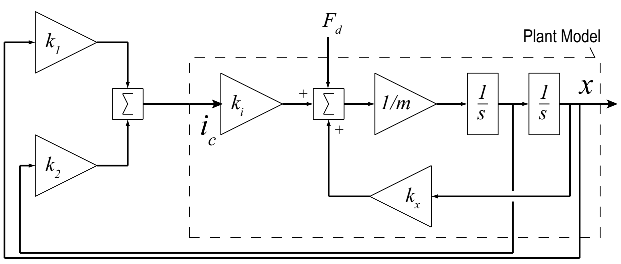

Fig. 2. AMB block diagram with state feedback

Alternatively, the closed-loop system from Fig 2 can also be written in state space form:

$$\begin{bmatrix} \dot{x} \\ \ddot{x} \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ \frac{k_x + k_1 k_i}{m} & \frac{k_2 k_i}{m} \end{bmatrix} \begin{bmatrix} x \\ \dot{x} \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{m} \end{bmatrix} F_d. \quad (3)$$

The controller parameters k_1 and k_2 appear in the second line of the system matrix, presented in control canonical form. Thus one can choose the coefficients of the characteristic polynomial of the system matrix which in turn control the location of the eigenvalues in the complex plain or the poles in the transfer function F_d to x [6].

For initial levitation, a proportional feedback just large enough to overcome the negative stiffness and some damping is desired. The automatically performed experiments acquire the needed information about the AMB system and include the following steps:

1) The control current i_c is driven from zero to i_{max}, which causes the body (if it is not already from the beginning) to lean all the way to the one side where the maximum positive current increases the flux density. The displacement is measured and stored as $\hat{x}_1$.
2) Now, the most negative current is applied and the body is jerked to the other side and pulled against the backup bearing. The displacement is measured again and kept as $\hat{x}_2$. A negative difference $\hat{x}_1 - \hat{x}_2$, indicates that the force axis and the measured position axis point in opposite directions, in other words, a positive current causes a force pulling in negative direction. The position signal is to be inverted, as well as $\hat{x}_1$ and $\hat{x}_2$. Next, the geometric center between the bearing poles is calculated: $\hat{x}_0 = \hat{x}_1 + \frac{\hat{x}_2 - \hat{x}_1}{2}$. Note that all the *hat*-parameters are measured and quantified in the control device's specific scale, not in meters or Newtons.
3) The current is slowly driven from $-i_{max}$ to i_{max} while monitoring the position signal. As soon as it crosses the center $\hat{x}_0$, the lift-off current i_{c2} is recorded. Note that the actual lift-off occurs earlier, however, due to the flat slope of the current signal, the error remains very small.
4) The previous step is reversed to measure the lift-off current i_{c1} from the other backup bearing. Furthermore, a velocity measurement $\hat{v}[k] = \hat{x}[k] - \hat{x}[k-1]$ is derived from the position signal, and the maximum stored as $\hat{v}_{max}$.
5) Finally, the closed-loop system is tested with the controller parameters K_P, K_D and K_I, which are based on the findings from the steps above. The position signal is monitored and the controller is deactivated immediately if stability is not yet achieved.

At the time when the body lifts off and with $\ddot{x}$ still being zero, the body is no longer leaning on the backup bearings and the following force balance is given at the startup from both backup rails:

$$i_{c1}k_i + (\hat{x}_1 - \hat{x}_0)k_x + F_d = 0, \quad (4)$$
$$i_{c2}k_i + (\hat{x}_2 - \hat{x}_0)k_x + F_d = 0. \quad (5)$$

Solving for F_d reveals

$$F_d = -\frac{i_{c1} + i_{c2}}{2} k_i \quad (6)$$

which is substituted into (4) and (5) which leads to the following:

$$\frac{k_x}{k_i} = \frac{i_{c2} - i_{c1}}{2(\hat{x}_1 - \hat{x}_0)} = \frac{i_{c1} - i_{c2}}{2(\hat{x}_2 - \hat{x}_0)}. \quad (7)$$

Rearranging (2) shows that a measurement for the minimum proportional gain K_P to overcome the negative stiffness is given by the right hand side of (7). A small safety factor is included to get a stable closed-loop system with low stiffness:

$$K_P := 1.2\frac{k_x}{k_i}. \quad (8)$$

The velocity feedback gain K_D is chosen such that maximum control action is opposing the motion when the body moves at maximum recorded speed:

$$K_D := \frac{i_{max}}{\hat{v}_{max}}. \quad (9)$$

The parameter K_I is appointed such that the integration is very slow and does not significantly influence the dynamic behavior of the suspension, for example

$$K_I := \frac{K_P}{10 f_{sp}}. \quad (10)$$

Nevertheless, the integral action is able to compensate for the constant disturbance force F_d and removes any steady-state position error. The suspended body could be left significantly off-centered if applying a soft or low-stiffness PD controller only. A discrete-time version of such a PID controller is given below,

$$i_c[k] := K_P e[k] + K_D(e[k] – e[k-1]) + K_I int[k] \quad (11)$$
$$int[k] := int[k-1] + e[k] \quad (12)$$

with discrete time step k, the position error $e[k] = \hat{x}_0 - \hat{x}[k]$ and a small K_I. In order to run this procedure, the allowable current range $i_{min} - i_{max}$ has to be known to prevent damage to the AMB, and the controller's sampling rate f_{sp} is used to setup the current ramps as well as for the calculation of K_I, both of which are not very critical.

III. Practical Considerations

The validity of the 2nd order linearized model is limited since it ignores the dynamics of the power amplifier, position sensor and time-delay of the digital controller. A more complete model as shown in Fig. 3 is used to determine stabilizing K_P and K_D through simulations.

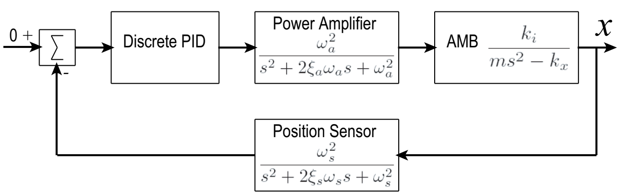

Fig. 3. A more detailed model, including power amplifier, sensor and controller

The AMB linear model is based on the thrust bearing of the compressor test rig with $k_x = 6800$kN/m, $k_i = 1150$N/A and the rotor mass $m = 37$kg. Both the power amplifier as well as the position sensor are modelled as 2nd order systems with undamped natural frequencies at 500Hz and 10kHz and damping ratios of 0.6 and 1.5. The digital PID controller operates at a 5kHz sampling rate and has a 40% cycle-time delay (80μs) from reading the sensor signal until the current command is updated. The parameter K_I is chosen very small such that the dynamics are basically not affected. Since there is no sensor to measure the velocity of the rotor, an approximation is made by multiplying the difference between the currently and previously measured position with the sampling frequency.

The effects of the low-pass filters (power amplifier and sensor model) and the delay result in phase lag which has to be compensated for by higher damping (phase lead). Simulation results (Figs. 4 and 5) underline the necessity of damping for the system to be operational. In fact, the higher the proportional gain is chosen, the more damping is required to keep the closed-loop system stable. Furthermore, a decrease in amplifier bandwidth has the same effect as an increase in controller delay (or computation time), both are compensated for by an increase in damping. Fast integral action (Fig. 6)requires high damping if the K_P is low at the same time. A typical selection for K_P is $2k_x/k_i$ which reverses the bearing's natural negative stiffness.

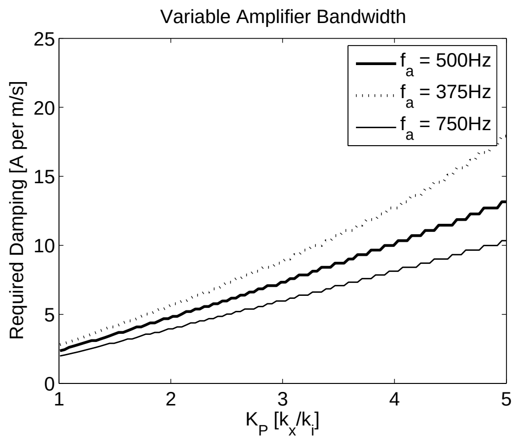

Fig. 4. Damping requirement for different amplifier bandwidths

The signal of the position sensor contains noise, limiting the proportional feedback gain and constraining even worse the feedback of the approximated velocity signal. Therefore, the initial guess for K_P is set just slightly above the stability threshold k_x/k_i (8). Still, there is no guarantee that setting K_D according to the measured maximum velocity and maximum control current i_c results in a stable close-loop system. As an alternative, one could measure the noise level in the position signal while the body is stationary, pressed against the backup bearings and a large current is running through the coils. Now K_D can be set to the highest practical level, in particular, to a level such that the amplified noise does

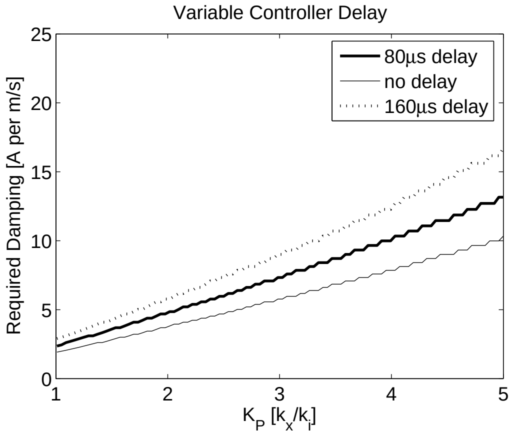

Fig. 5. Damping requirement for different controller delays

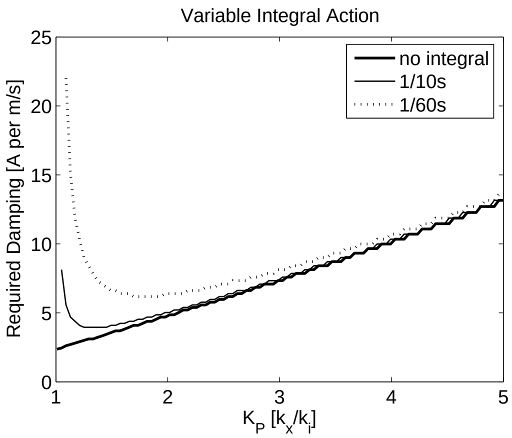

Fig. 6. Damping requirements for different integral actions

not saturate the power amplifier.

Another possible obstacle for the *Auto Levitator* might be the fact that the AMB system is identified while the floating body or rotor is leaning on the backup bearings and is not centered. Due to the nonlinearity of the actuator, k_x/k_i could get over estimated. In theory, this would not cause a problem and simply result in a rather stiff initial controller, however, the damping requirements will rise unnecessarily. Furthermore, position sensors could become nonlinear or might saturate when the displacement is very large. Consequences could result in levitation at an off-centered position (asymmetric saturation), and again in an over estimate of k_x/k_i (saturation).

IV. Active Magnetic Bearing Test Rigs

Two AMB test rigs were used to evaluate the *Auto Levitator*. The first one (shown in Fig 7) consists of a beam free to rotate on a pivot and two electromagnets serving as actuators, located at each end of the beam. These two coils are connected in series and apply a torque to the beam by adjusting the bias flux provided by permanent magnets. This setup serves as test platform for different types of control algorithms and has been used for self-sensing.

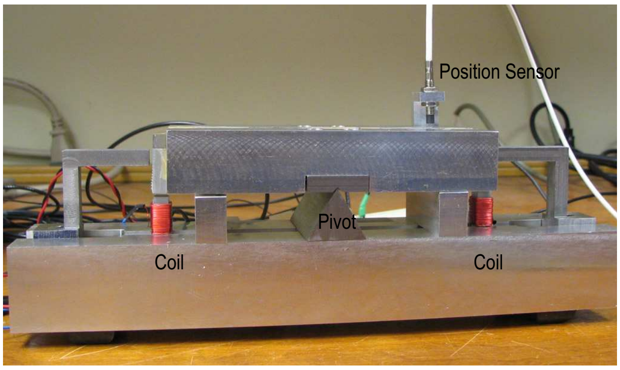

Fig. 7. Balanced Beam AMB test rig

The second machine (see Fig 8) is a compressor test rig which has been build to measure fluid forces acting on the impeller and to actively control surge [5]. The impeller sits on a shaft which is kept in place by two radial AMBs and one AMB in thrust direction with load capacities of 1400N and 6600N, respevtively.

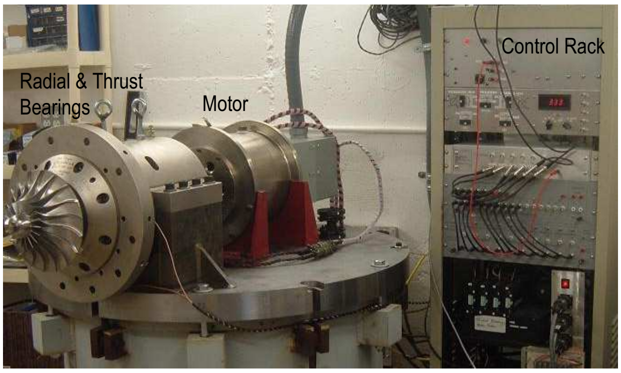

Fig. 8. High speed centrifugal compressor AMB test rig

V. Experimental Results and Conclusion

The Auto Levitator was implemented in C on a standard PC with Real-time Linux OS, and the presented data was obtained from the balanced beam test rig. The entire identification procedure as described before is shown in Fig 9. The upper plot displays the current command signal determined by the *Auto Levitator* state machine and after 22 seconds by the PID controller. The measured position signal is shown in the bottom plot. The first rising and falling current ramps measure the position sensor range, during the second set of ramps (between seconds 12-22) the bearing characteristics are detected and finally, the controller with the identified parameters is activated. The beam evidently reaches the desired position $\hat{x}_0$ at about 4V, and the command signal exhibits the noise amplification. At this stage, the bearing system is ready for tuning of the control parameters until given performance requirements are met or some optimal performace is reached.

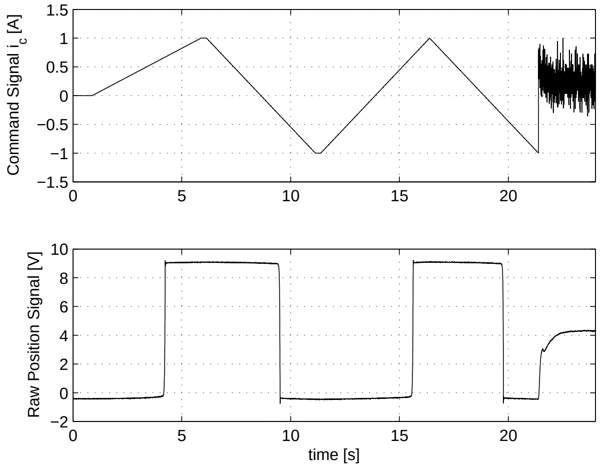

Fig. 9. Successful initial levitation of the balanced beam

The automatic levitation procedure has successfully been tested on a single degree of freedom systems, and led to stable levitation right away. A simplified AMB model was used to characterize a stabilizing state feedback or PD controller, and simulations were used to establish regions for desired control parameters.
An extension to full rotor suspension by consecutively levitating one axis after another seems feasible. Some difficulties could arise from bearings using permanent magnet bias flux (which can not be turned off) that might interfere with the automatic experiment of a neighboring bearing.

VI. Acknowledgements

The authors would like to thank Dr. Paul Allaire, Dr. Eric Maslen and Dr. Zongli Lin for their advise and support on this project.

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