Towards Practical Applications of Self-Sensing Magnetic Bearings

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ABSTRACT

The self-sensing bearing, i.e. an AMB using no specific sensing hardware at the rotor, has been developed based on controllability and observability of the AMB with voltage-input and current-output. It has the potential of widespread industrial application due to its unique advantages.

In this paper control techniques for the self-sensing magnetic bearing are investigated as a further step towards practical applications. We shall show how sinusoidal disturbances can be effectively compensated for this new bearing type. This means that unbalance compensation is realizable for self-sensing bearing systems. Experimental results have been obtained with a realistic rotor system using a floating point signal processor with high level programming and monitoring capabilities.

1. INTRODUCTION

Active magnetic bearings (AMBs) have many advantages over conventional bearings. Contact-free support of course effectively eliminates mechanical friction, lubrication and wear problems. At least as important is however the capability of AMBs to dynamically control the bearing force in just about any conceivable manner. Important examples are stiffness, damping and the treatment of harmonic disturbances like unbalance. Rotor and bearing vibrations can be substantially reduced or, in some cases, eliminated completly, through appropriate design of the controller [1-6].

The classical design concept of magnetic bearing control systems is based on feedback of rotor displacement signals. In earlier publications [7] we have presented an entirely new type of AMB, which does not need any sensing device at the rotor. The aim of this paper is to show, that this very unconventional AMB type has almost all the desirable features of the classical AMB. Specifically, we demonstrate that appropriate treatment of harmonic disturbances for unbalance compensation can be realized with much the same results as for the classical AMB.

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Additionally, the self-sensing bearing has some new and quite unusual features connected to what may be termed "negative static stiffness". This results in high static load (by a factor up to seven times the static load of the same bearing with classical PD control!) and in low power consumption of the controller (sometimes called "virtual zero power consumption [8]). It is needless to say that such features are of great advantage in various practical applications.

All of this has been demonstrated with an actual rotor system of realistic dimensions. In order to test many different controllers, a floating point signal processor system with high-level programming capability has been used. Experimental results are found in section 5.

2. PLANT MODEL

As the self sensing bearing has been presented in many recent publications [9, 10, 11] the main results needed for this paper and the basics of the self-sensing AMB are given here without many details.

The central element of every AMB is the electromagnet. The force-current-displacement relation of the electromagnets is linearized as usual in the following manner:

$$f = k_i i + k_s x \tag{1}$$

with the variables bearing force f, bearing current i and displacement x. The bearing is characterized by the two parameters k_i (force-current factor) and k_s (force-displacement factor). The conventional way to control the AMB is for the controller to provide a current i proportional to the desired force while the desired force itself is computed from the displacement signal. The self sensing bearing does not use the current as input variable to the plant. New input variable is the voltage to the electromagnet coil. The current is used as output variable of the plant. The current is also used as state variable for the simple 1-DOF system, bringing the minimum number of state variables per mechanical DOF to three. Since such a system is controllable and observable it is not necessary to measure the displacement in order to stabilize the bearing-rotor plant around an operating point. The state space model is given here:

$$\frac{d}{dt}\begin{bmatrix} x \\ v \\ i \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ \frac{k_s}{m} & 0 & \frac{k_i}{m} \\ 0 & -\frac{k_b}{L} & -\frac{R}{L} \end{bmatrix} \begin{bmatrix} x \\ v \\ i \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{1}{L} \end{bmatrix} u + \begin{bmatrix} 0 \\ \frac{1}{m} \\ 0 \end{bmatrix} w \tag{2}$$

with the variables x for displacement, v for velocity, i for current, u for voltage and w for disturbance force and with the parameters m for mass, R for coil resistance, L for inductance, and k_b for the voltage-velocity back-EMF. Theoretically, the voltage-velocity factor is equal to the force-current factor.

The controller can be designed in a straight forward manner as a state-feedback with an observer. As with any controller of an unstable system, the main difficulties are with accurate modelling and parameter measurement as well as with a sensible choice of the main parameters. These tasks are clearly more difficult for the self-sensing bearing than for the classical AMB. There is no way around some empirical knowledge and practical experience combined with theoretical understanding of the plant in order to achieve the desired result. Once the basic controller has been implemented and the AMB system operates with acceptable stiffness and damping, additional features are necessary for industrial applications.

It is not evident that important AMB control features like harmonic disturbance rejection can be realized for the self-sensing AMB. In the next section it is shown, that this is indeed possible.

3. HARMONIC DISTURBANCE REJECTION

Harmonic disturbance rejection (sometimes referred to as "unbalance compensation" is especially important for practical application of high-speed rotors with AMBs. This topic is treated here for the first time for self-sensing AMBs.

The approach here consists of including the harmonic disturbance in the state vector as two new state variables w_1 and w_2 of unknown amplitudes, but known frequency ω (the rotational frequency). The extended state vector and system matrix become

$$\boldsymbol{x} = \begin{bmatrix} x \\ v \\ i \\ w_1 \\ w_2 \end{bmatrix} \qquad \boldsymbol{A}_{ex} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ \frac{k_s}{m} & 0 & \frac{k_i}{m} & \frac{1}{m} & 0 \\ 0 & -\frac{k_b}{L} & -\frac{R}{L} & 0 & 0 \\ 0 & 0 & 0 & 0 & -\omega_e \\ 0 & 0 & 0 & \omega_e & 0 \end{bmatrix} \tag{3}$$

The unknown amplitude and phase of the unbalance determines the initial conditions for the two disturbance variables, which then move like sine and cosine.

This complete system is observable from the current measurement alone. It is therefore possible to design a feedback control, which eliminates the sinusoidal vibration on either one of the three variables current, force or displacement. All three cases are of considerable practical relevance. The force and current amplitudes required for displacement control of course depend on the magnitude of the unbalance and on the rotational speed. Measurement results for current control are given in section five. The control layout method needed is described in textbooks on control theory, e.g. [12].

One limit case is of special interest: If the rotational velocity approaches zero, the observability of the system (3) is lost. This means in practice, that a step-disturbance force cannot be corrected. The next section is devoted to this phenomenon and its interesting consequences.

4. NEGATIVE STIFFNESS AND ZERO POWER

For this control plant, it can be shown that the most simple controllers as they are proposed in section two result in steady-state values for voltage u and current i always equal to zero and *independent of a static load-force* $w=F_d$=constant, whereas the steady-state value of the deflection x is constant and *negative*, i.e. opposed to the load direction. This can be seen by from the second equation of system (2) at steady state:

$$m\mathrm{d}x/\mathrm{d}t = k_s x + k_i i + F_d \quad \text{hence} \quad x(t \to \infty) = -F_d/k_s \tag{4}$$

In practice, this means simply that the self-sensing bearing always seeks the operating point with zero control current, i.e. with the same bias current, *independently of the static load.* Therefore, the airgap changes according to the static load: It becomes smaller at the magnet, from which the rotor is pulled away. In this way, the force is increased according to (1) without increasing the control current. This deflection is opposite to the direction of the applied force. This is the reason, why the static load capability of the self-sensing bearing is dramatically increased over the same set-up equipped with a classical PD-control. In the classical case, the active air-gap gets larger as the load increases. Over this larger air-gap, the maximum magnetic force is smaller than in the nominal (unloaded) case, the decrease in force is roughly proportional to the square of the air-gap. For this reason, classical AMBs with high static load are often equipped with an integrating feedback which keeps the rotor in nominal position under static load change. This can already drastically increase static load capacity, as anybody with some AMB experience can tell. The self-sensing bearing does even much better than that, since it not only keeps the air gap from getting larger, but since it actually decreases the active air-gap. Therefore, surprisingly high static load forces can be reached with even the most simple self-sensing bearing. It could be said that a sort of "super integrator" is inherently built into every self-sensing bearing.

5. EXPERIMENTAL RESULTS

All which has been described up to now would only be of limited value had it not been tested in actual experiments. As expected, it is more difficult to fine tune the controller of a self-sensing bearing than a classical one. The first reason for this fact is, of course, that the minimum number of free parameters of the simplest stabilizing controller is only two for the classical AMB, but it is five for the self-sensing bearing (three state feedback coefficients and two for the reduced observer. In practice, a full order observer is preferred, bringing the theoretical number of free parameters to six). This simple difference means that it is not possible for a first trial operation to rely on the manual parameter tuning so familiar in conventional AMBs. However, our experiments prove clearly that, once tuned, self-sensing AMBs can be as robust and reliable, as stiff and as well damped as classical bearings. The experiments prove also that very low costs are indeed realizable and that the static load is much higher than that of a comparable classical AMB.

The simple system with extremely few components (analog control) has been demonstrated twice in the USA. The measurement results of this paper are from a rotor system which has not been designed for low costs, but which is used for development of sophisticated rotor control including harmonic disturbance rejection. This system is equipped with a floating point DSP.

The first figure (1) shows the step response of the rotor displacement. Two curves are shown, measured and observed displacement. Of course, the measurement is not used in the on-line control, it is strictly a monitoring measurement. The step applied corresponds to a change in the static load. The initial displacement is in the direction of the applied step, as with every bearing. The "negative stiffness" of the self-sensing bearing is clearly visible in the steady-state value of the step response which is opposite to the initial displacement.

As mentioned before, the displacement is not observable from the current measurement alone. Therefore, the static response of the observer always converges to zero. Velocity however is observable, therefore the dynamic behaviour of the displacement is accurately reproduced by the observer.

The second figure shows the disturbance rejection of unbalance harmonic vibration. The signal displayed is the current. The rotor with its unbalance rotates at 3000 rpm. Obviously, the harmonic signal dominates in the bearing current. In the next picture, all conditions are the same, except that harmonic disturbance rejection with respect to the current is now implemented according to section (3). As seen on the picture, the harmonic signal vanishes in the background noise of the current signal.

6. CONCLUSION

The research results presented in this paper prove that the self-sensing bearing has the capability required for practical applications. Beside the fact that it has the potential of very high cost effectiveness, it is robust and well damped. The disturbance rejection capabilities of such a system are investigated in this paper. Theory and experiment show, that harmonic disturbance rejection schemes for current, force or displacement can be implemented as with the classical AMB in spite the fact that no sensing hardware is used at the rotor.

Steady-state disturbances cannot be suppressed for the displacement, but the steady-state response is in such a direction as to drastically increase the static load capacity of the self-sensing bearing vs. a classical AMB of similar layout ("negative stiffness", the air gap is small where the load is large). At the same time, power-consumption is minimized. It is therefore safe to predict that the self-sensing bearing will find practical applications.

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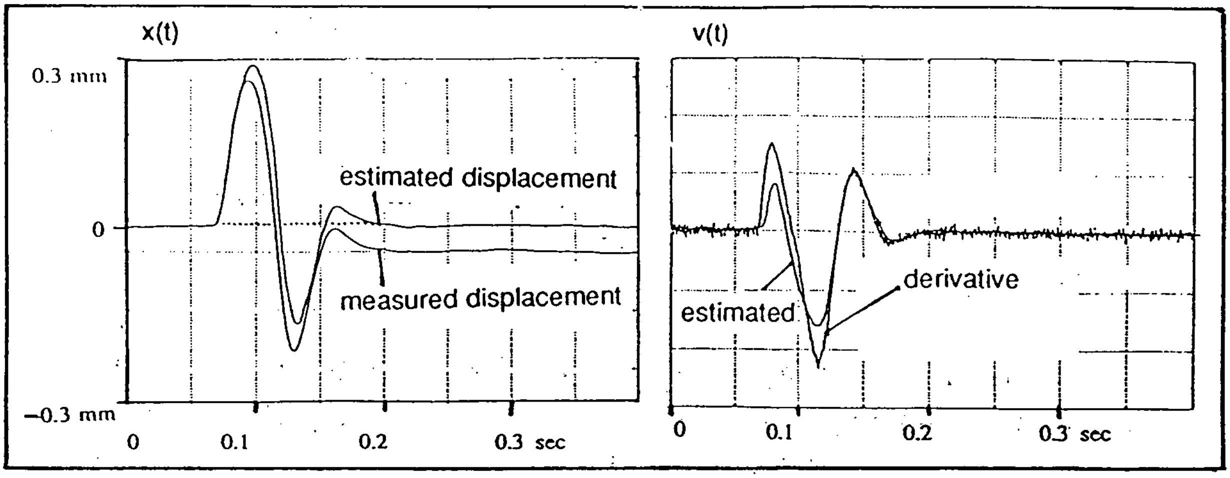

Fig. 1 Displacement (left) and velocity (right) response to a step in load force. One value is measured (resp. differentiated) and the other is computed by the Luenberger observer from current measurement only. Note the steady-state difference in displacement.

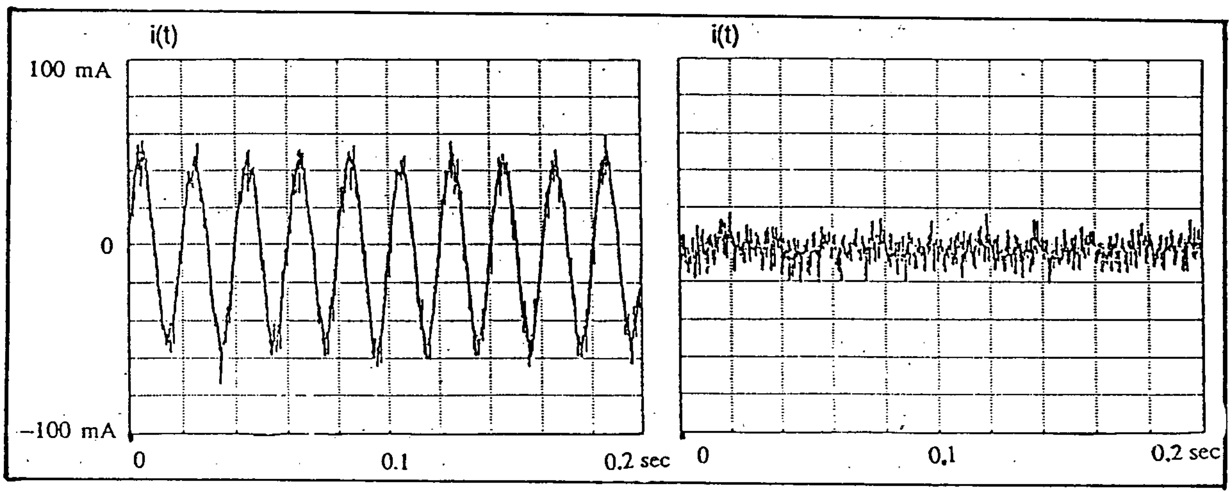

Fig. 2 Unbalance response at 3000 rpm . Control current of self-sensing AMB before and after unbalance disturbance rejection.