ACCELERATION FEEDFORWARD FOR INCREASE OF BEARING STIFFNESS - APPLICATION FOR VERY SMALL AMBs

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ABSTRACT

The achievable dynamic stiffness of active magnetic bearings (AMBs) is typically quite low compared to ball bearings or hydrodynamic bearings. For many applications this is not a problem, or even a favorable characteristic. In the case of non-stationary bearings, however, base vibrations will cause large amplitude responses relative to the base due to the limited stiffness. *Acceleration feedforward control* is a method that helps to reduce these amplitudes considerably.

This paper presents the functional prototype of an AMB that meets the space requirements of commercial *hard disk drives* (HDDs). The work was carried out in the scope of a feasibility study, investigating a possible application of AMBs in HDDs. It is shown how acceleration feedforward improves the dynamic performance and that the dynamic specifications of modern disk drives can practically be satisfied with AMBs.

INTRODUCTION

In many applications of rotating machinery, operating forces or disturbance forces are acting directly upon the rotor. Cutting forces of milling spindles and pressure changes in pumps or compressors are examples. In that case, the resulting rotor displacements are directly related to the applied force and the bearing stiffness. If the source of disturbances are linear and angular accelerations of the base of the system, however, the forces acting upon the rotor are inertial forces, not external mechanical forces as in the first case. HDDs are a typical example of systems where base accelerations are the major disturbance source (eg. when a computer with its running hard disk is being moved). Therefore disk drive manufacturers are confronted with quite severe dynamic specifications, having to ensure that the spindle vibrations stay within certain amplitudes when accelerations are applied to the housing.

AMBs that have to deal with this type of disturbances have a major advantage over conventional bearings: Supposing that the base acceleration is measured, the controller can react already when an acceleration occurs, not having to wait until the acceleration results in a relative rotor displacement. The dynamic stiffness and therefore the robusteness against shocks from the outside can be increased significantly. This kind of acceleration feedforward has already been proposed for active magnetic bearings regarding aseismic evaluation for the use in nuclear power plants [4], or for improving the head positioning dynamics in magnetic disk drives [1].

THE SERVO LOOP

The dynamic specification for the spindle bearings in disk drives is mainly given by the bandwidth of the servo positiong system for the read- and write heads. When external excitations of the housing excite radial vibrations of the spindle, the heads may still be able to follow the tracks without causing any read- and write errors - as long as the vibration amplitudes stay within certain limits at a given frequency.

The standard test in disk drive industry for characterizing the bearings with respect to the servo loop is the *shaker test*. The bearing is mounted on a vibration table that sweeps through a defined frequency range, keeping the acceleration amplitude at a constant value. Then the frequency reponse of the radial bearing displacement relative to the base is measured.

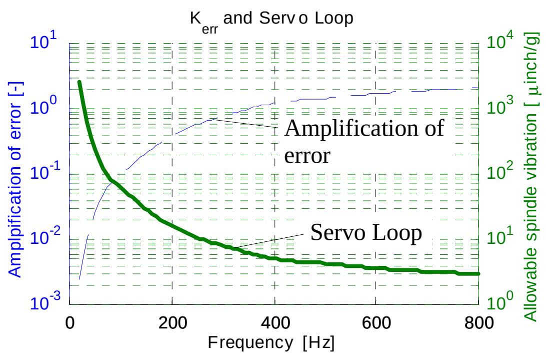

FIGURE 1: Typical servo loop of a hard disk drive

Ball bearings have stiffnesses in the order of several tens to the power of seven and stay under the servo loop even with very little damping. The stiffness of hydrodynamic oil bearings is typically already one order below ball bearings. But with their high damping, the amplitudes stay small enough in order to satisfy the servo specification. The stiffness of AMBs of that size usually lies yet another order below the one of hydros (some tens to the power of five). Since the damping can only be increased to a certain extend, it is extremely difficult to reach the specifications. Simulations and experiments with functional prototypes confirm this.

FEEDFORWARD CONTROL

Illustration For 1 DOF

The principle of the acceleration feedforward can be illustrated using a simplified model of an AMB with one degree of freedom. The equation of motion for the rotor is:

$$m \cdot (\ddot{z}_{Base} + \ddot{z}_{Rotor}) = k_s \cdot z_{Rotor} + k_i \cdot i_x \qquad (1)$$

Using a simple PD-controller, adding a feedforward term and closing the control loop, one gets:

$$m\ddot{z} + k_i D_c \dot{z}_{Rotor} + k_i P_c - k_s z_{Rotor} = -m\ddot{z}_{Base} + k_i i_{forward} \qquad (2)$$

The base acceleration, $\ddot{z}_{Base}$, acts as inertial disturbance force which can be compensated by means of the forward current, $i_{forward}$. As a result, the amplitude response can be reduced, in the ideal case down to zero, without having to increase the bearing stiffness, thus moving the eigenvalues of the system to higher frequencies.

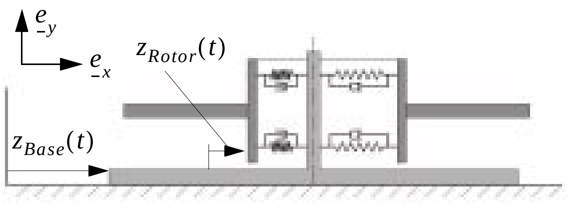

FIGURE 2: Principle for one DOF

Multi DOF Rotor

Placement of accelerometers. Acceleration feedforward as it was proposed in [4] uses seperate accelerometers placed close to the electromagnets of the bearings. This way a decentralized control structure can be applied. For a short and compact rotor as in the present case (see next chapter), however, this cannot be realized due to the limited volume inside of the rotor. Therefore the accelerometers are mounted directly onto the base, and the feedforward currents in the actuators have to be calculated through the rotor's mass matrix. Therefore, a precise identification of the system is necessary for achieving good results, or the feedforward matrix has to be tuned by a trial and error method.

Equations For Multi DOF System. The equation of motion for the rigid body rotor in its center of gravity (COG)-coordinates is given by:

$$[M]\ddot{\underline{z}} + [G]\dot{\underline{z}} - [{}^fT_{b\to c}][K_s][{}^zT_{c\to b}]\underline{z} = [{}^fT_{b\to c}][K_i](\underline{i}_x + \underline{i}_f) - [M]\ddot{\underline{z}}_{base} \qquad (3)$$

$[M]$ and $[G]$ are the mass-, respectively the gyroscopic matrix, $[K_i]$ is the matrix of the current- to force constants, and $[K_s]$ contains the negative bearing stiffnesses. $[{}^fT_{b\to c}]$ defines the transformation for the forces from bearing- to COG coordinates, and $[{}^zT_{c\to b}]$ finally gives the transformation for displacements from COG- to bearing coordinates.
As in the one DOF case, the excitation from the base has to be compensated by the feedforward current, $\underline{i}_f$. From equation (3) it can be seen that the feedforward matrix for a model based implementation is given by:

$$\underline{i}_f = ([{}^fT_{b\to c}][K_i])^{-1}[M]\cdot\ddot{\underline{z}}_{base} \qquad (4)$$

THE AMB PROTOTYPE

Several prototypes of AMBs which meet the space requirements of conventional HDDs have been built at EPFL. Among these are bearings using using motor-bearing combinations [2],[7], minimum actuator configurations which couple the radial- and axial directions [8], [3] and bearings with rather conventional designs. The prototype which is presented here, and in which the feedforward control is implemented, is of the last type.

Bearing Configuration

The prototype has a conventional bearing configuration (see **FIGURE 3** and **FIGURE 4**). Two independant radial bearings are placed on an upper and on a lower plane. They are both located on the inside of the rotor, mounted on a central shaft. A nine pole, three phase brushless DC-motor, as it is used in a typical HDD of today, is placed between the two radial bearings. The thrust bearing on the bottom of the rotor controls the axial degree of freedom. It is directly indegrated into the base plate of the system. The disk which is mounted onto the outside of the hub is not shown in the drawing.

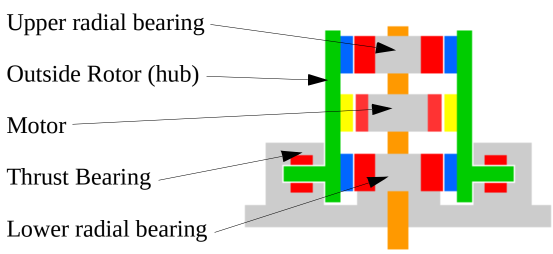

FIGURE 3: Bearing Configuration

Position Sensors

Eddy current positon sensors are integrated into the setup. The sensing coils are made of 30µm copper wire and have an outer diameter of 1.6 mm. They are glued into sensor carriers of plexiglass. The driving electronics for the coils is presently an a printed circuit board, located outside of the bearing. But the feasibility of integrating this part into the bearing as well has already been shown ([5]).

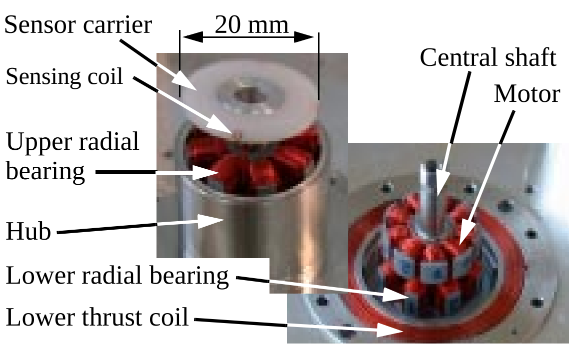

FIGURE 4: Realization

The radial displacements are measured differentially, the axial displacement is measured only from one side in order to simplify the construction.

Bearing Characteristics

TABLE 1 gives an overview of some important characteristic bearing sizes.

TABLE 1: Characteristic bearing sizes

Variable	Value
Displacement-to-force constant	35 [kN/m]
Current-to force constant	10 [N/A]
Proportional feedback gain	$1.6 \cdot 10^4$ [A/m]
Sensor gain	37.5 [kV/m]
Rotor mass	0.1 [kg]
Moment of inertia: I_x and I_y	$3.5 \cdot 10^{-5}$ [kgm^2]
Moment of inertia: I_z	$6.3 \cdot 10^{-5}$ [kgm^2]

The given current-to-force- and the displacement-to force constants are based on the results of a system identification. They lie within about 20% of the calculated values. Together with the proportional feedback gain it can be seen that the machanical stiffness of the bearings is about $1.6 \cdot 10^5$ [N/m].

IMPLEMENTATION OF THE FEEDFORWARD CONTROL

At the present state of the project, the acceleration feedforward is done only for one translational DOF. The feedforward matrix is then reduced to two elements.

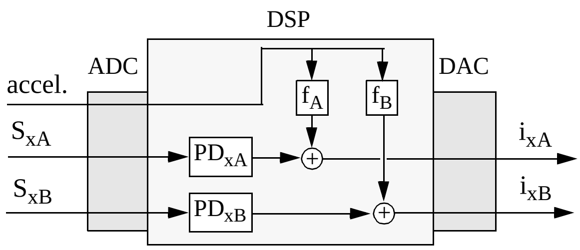

FIGURE 5: Control scheme for compensation of one DOF

The signal of the accelerometer is read into the DSP by the AD converter and then added to the control signals with different weights, f_A and f_B. The order of the weighting constant was first calculated using the mass matrix, and then the fine tuning was done by hand.

EXPERIMENTAL RESULTS

"Hammer" Test

First, a simple experimental labaratory setup for first tests was used. It consists of a linear slide which guides the bearing base along one linear degree of freedom. A hammer swinging from a constant height hits the base, creating a repetitive acceleration. The amplitude of the impact can be set in a range from about one to five g.

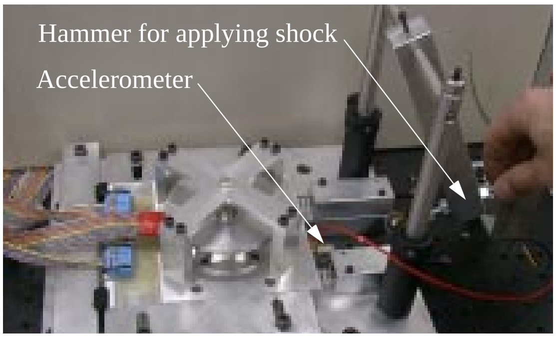

FIGURE 6: Simple test setup for laboratory use

The xy-plots in **FIGURE 7** show the measured displacements of the rotor in the upper and the lower bearing plane after an impact, with- and without feedforward control (the tangle of lines in the center is the response with feedforward control).

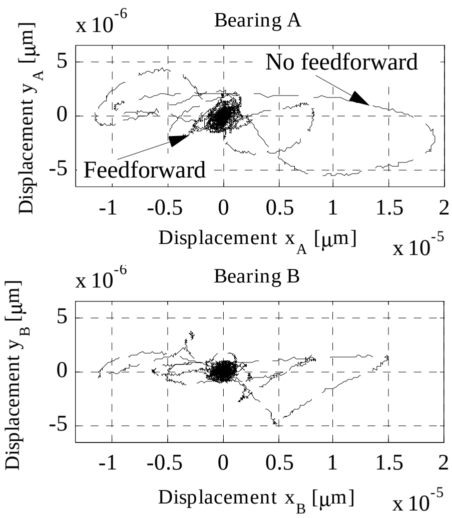

FIGURE 7: Bearing displacements with- and without acceleration feedforward control

The rotor is spinning at 6000 rpm and the amplitude of the impact is about 5 g. In the plots the response is displayed for the situation with, respectively without feedforward control. The amplitudes are reduced by a factor of about ten.

Shaker Test

Since with the described setup only a limited frequency range can be excited, the shaker test on a controlled vibration table was carried out as well. The tests were carried out from 20-1000 Hz with an acceleration amplitude of 1g. The radial displacements are measured at the front side of the disk. **FIGURE 8** shows the measured and the simulated result for both controller types. The simulations coincide quite well with the measurements except for the low frequency range with the feedforward control. The reason lies in the frequency response of the accelerometer which shows considerable phase loss for frequencies below 50 Hz. This phase loss isn't included in the model.

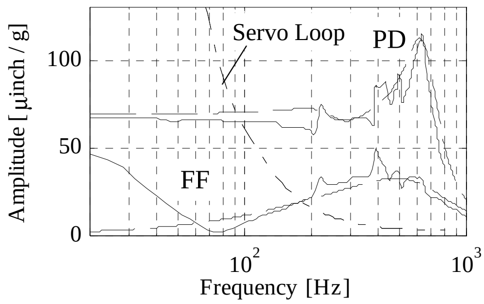

FIGURE 8: Radial response with shaker test. Comparison between controller with- and without acceleration feedforward.

Generally it can be stated that for low frequencies (not regarding the problem with the phase loss of the accelerometer) the feedforward compensation works nearly perfectly. The amplitudes are reduces by a factor of up to 30. For higher frequencies, however, the amplitudes of the compensated response rise, and the remaining reduction at some frequency points is only a factor of two. With respect to the servo loop, the response is a factor of 5 to 10 too high there.

THEORETICAL DISCUSSION

The reason for the rise of the amplitudes at higher frequencies lies in the sensitivity of the system to phase loss of the acceleration signal. One main source is the zero order hold (ZOH) and the delay of the AD- and DA-conversion which is necessary for adding the acceleration signal to the controller output on a digital level.

Of course one could imagine to add the acceleration signal on an analog level after the controller. The simulation shows that the potential of improvement by doing this is in the range of 100%.

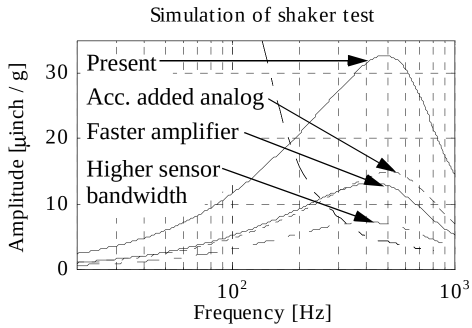

FIGURE 9: Simulation of system with less phase loss

CONCLUSION

The prototype of an AMB in the size of a commercial disk drive has been presented. The work was carried out within the scope of a feasibility study, investigating a possible application of AMBs in HDDs from a technical point of view.

The dynamic stiffness of AMBs is a critical issue. By using acceleration feedforward control, the dynamic stiffness can be increased considerably. Experimental work and predictions using simulation suggest that the major dynamic specification, the *servo loop*, can be satisfied.

Acceleration feedforward represents a powerful method for applications of non-stationary magnetic bearings for keeping relative amplitudes at a minimum. With the continuous development in the field of microelectronics and today's possibilities in system integration, the disadvantage of adding additional sensing devices will lose its importance when regarding the benefits in performance that are gained.

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