DESIGN OF A STATOR-CONTROLLED MAGNET BEARING

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ABSTRACT

The subject bearing has a laterally movable stator which consists of a permanent magnetic ring axially polarized and two disks made of magnetic material. The radial magnetic flux at air gaps between the stator and rotor form an unstable bearing. To make it stable, the stator is mounted on mechanical springs and motion-controlled by feeding back the rotor displacement. The motion control actuators are likely the stationary electromagnetic types. A design methodology for the new bearing concept is presented herein with emphasis on sizing the system parameters for stability. The bearing is best suited for supporting vertical rotors, such as those of energy or momentum storage flywheels. It has a simple rotor structure, minimal eddy-current loss, and electronically maneuverable stiffness and damping.

INTRODUCTION

A conventional active magnetic bearing (AMB) has stationary electromagnetic poles around its rotor. In rotation, the rotor surface material comes in and out of the magnetic flux of the protruding poles. The changing flux in the surface material generates heat due to magnetic hysteresis and eddy current. The latter not only causes heat loss but also delays the control response of the electromagnets. To reduce the eddy current effect, the conventional AMB cores are usually made of silicon steel laminations. The eddy current heat loss on high speed rotors can be a serious problem, because it is difficult to dissipate in vacuum. It may generate high rotor temperature causing stress and other thermal related problems. Using a homopolar AMB with extended pole edges in the circumferential direction may reduce the eddy current heat, but can not totally eliminate it. Note that the heat is proportional to the square of rotor speed times number of poles. This has led to the use of continuous ring pole permanent magnet (PM) bearings. Since the magnetic flux of the ring-shaped poles is not interrupted during rotation, the eddy current and hysteresis core losses can be kept to a minimum. As an example, two radial PM ring bearings have been designed for a flywheel energy storage power quality application [1]. These bearings have stationary and rotating disks packed with many axially polarized PM rings. They are expensive to fabricate and have centrifugal stress concern at high speeds. Also, they are soft and have no damping; their large axial negative stiffnesses require oversized active thrust magnetic bearings.

Herein a new PM bearing concept called stator-controlled magnetic bearing (SCMB) is presented. As shown in Figure 1a, the stator consists of two disks made of magnetic material such as silicon steel, and each disk has a center hole served as a magnetic pole. A PM ring with axial polarization is sandwiched between the two disks . The rotor is simply a circular cylinder made of magnetic material with an outer diameter slightly smaller than the stator disk holes. The magnetic flux circulating through the annular air gaps form an unstable magnetic bearing with a negative stiffness. To make a stable bearing as demonstrated by Oka and Higuchi [2], the stator should be mounted on mechanical springs and its motion be controlled with feedback of the rotor motions. The latter are measured by using two displacement sensors. The actuators for controlling the stator motions are likely a stationary type of AMB as shown by the concept variation in Figure 1b. The actuators are not the main interest of this paper. The focus here will be on how to design a stabilizing controller and how to size the bearing proper and the actuator capacity. Note that the rotor leans on a backup bearing when the stator is not under control.

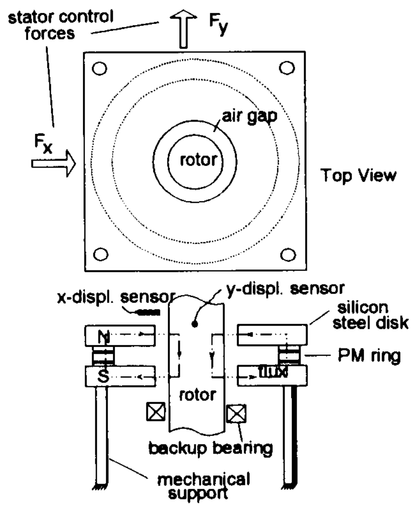


Figure 1a. Stator-controlled magnetic bearing concept

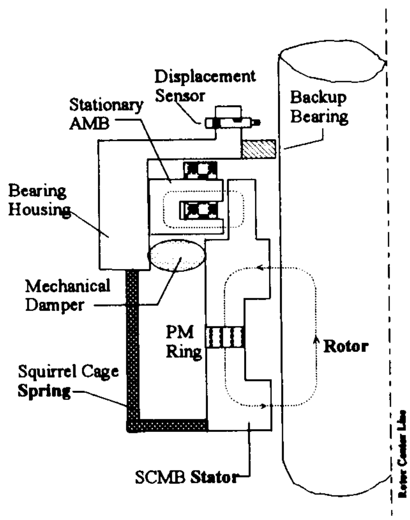


Figure 1b. SCMB practical implementation

FORMULATION OF STATOR-CONTROLLED MAGNETIC BEARING

There are two stator motion-control axes which are assumed to be independent of each other. The dynamics of each control axis can be represented by Figure 2. The equations of motions are the following:

$$M_s X_s'' = K_m(X_s - X_b) - F_s \tag{1}$$

$$M_b X_b'' = -K_m(X_s - X_b) - KX_b - CX_b' + F \tag{2}$$

where

M_s = rotor mass at bearing
M_b = stator mass
X_s = rotor displacement

X_b = stator displacement
' = differentiate once with respect to time
" = differentiate twice with respect to time
K_m = stiffness coefficient of magnetic field in air gaps
K = stiffness coefficient of stator mechanical support
C = damping coefficient of stator mechanical support
F_s = static load on rotor
F = stator control force.

For the stator feedback control, the rotor displacements relative to ground are measured in two orthogonal directions as shown in Figure 1a. A PID (proportional, integral, derivative) control scheme is appropriate for the application and the stator control force is represented by (3).

$$F = C_p X_s + C_d X_s' + C_i \int X_s dt \tag{3}$$

where

C_p = proportional constant
C_i = integral constant
C_d = derivative constant
t = time.

The first priority of the bearing design is to make a stable control system by choosing a proper set of PID constants. For evaluating stability, the static force F_s in equation (1) may be ignored. Taking Laplace transform of (1), (2) and (3), and combining the three transformed equations, the following normalized system characteristic equation is obtained:

$$S^5 + CS^4 + (K-\mu-1)S^3 + (C_d-C)S^2 + (C_p-K)S + C_i = 0 \tag{4}$$

where $\mu = M_b/M_s$. All the parameters in (4) are normalized or dimensionless quantities as defined below. The sign "==>" means "imply".

S ==> S/B_s (S=Laplace variable, $B_s=\sqrt{K_m/M_s}$)
C ==> $C/\sqrt{K_m M_s}$
C_d ==> $C_d/\sqrt{K_m M_s}$
K ==> K/K_m
C_p ==> C_p/K_m
C_i ==> $C_i/K_m B_s$.

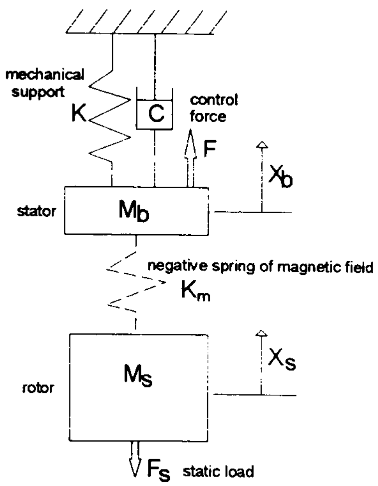


Figure 2. Control axis representation

The above normalization is done with respect to the magnetic field stiffness (or negative spring rate) K_m and the rotor mass M_s which are the basic given quantities of the bearing system. The artificial parameter B_s provides a calibration of the frequency location of the lowest system mode. The bearing design work is to choose a set of values for six parameters, i.e., μ, K, C, C_p, C_d and C_i, so that equation (4) has stable roots which all lie in the left half of the S-plane.

SYSTEM SIZING METHOD

Out of the six parameters, only the mass ratio μ may be independently chosen. The stator mass relative to the rotor mass which includes two disks, one PM ring and a part of the mechanical support, can be estimated. The remaining five normalized parameters can be determined by using the pole-placement method. A desirable set of five roots of the normalized equation (4) may include a pair of reasonably damped complex conjugate roots and three negative real roots. To demonstrate the method, let's consider the following "desirable" five roots:

$$S = -0.3 \pm 0.5j \,;\quad -0.6 \,; -0.6 \,;\quad -1.0$$

Then the system characteristic equation can be re-created below:

$$(S+0.3+0.5j)(S+0.3-0.5j)(S+0.6)^2(S+1.0) = 0$$

or

$$S^5 + 2.8S^4 + 3.22S^3 + 2.144S^2 + 0.8464S + 0.1224 = 0 \tag{5}$$

Comparing (5) to (4), we obtain:

$$C/\mu = 2.8 \,;\ (K-\mu-1)/\mu = 3.22 \,;\ (C_d-C)/\mu = 2.144 \,;$$

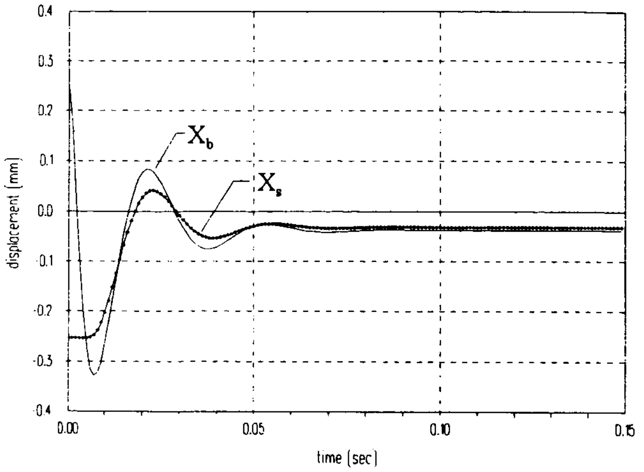
$$(C_p-K)/\mu = 0.8464 \,;\ C_i/\mu = 0.1224$$

Let's choose the stator mass to be 1/8 of the rotor mass M_s, i.e., $\mu = 0.125$. Then the five normalized system parameters are:

$$C = 0.35 \,;\ K = 1.5275 \,;\ C_d = 0.618 \,;\ C_p = 1.6333 \,;\ C_i = 0.0153$$

NUMERICAL EXAMPLE AND TRANSIENT SIMULATION

To test the performance of the system with the above parameters, a transient simulation of the rotor lifting off a backup bearing in a SCPM bearing has been performed. The transient results as presented in Figure 3 to Figure 5 show the rotor and stator displacements and the associated forces in one of the two orthogonal axes. The force exerted on rotor by stator is defined as $F_m = K_m(X_s-X_b)$. The system has parameters as chosen above and it has a rotor mass M_s = 10 Kg (22 lb) and a negative stiffness $-K_m$ = -7×10^5 N/m (-4000 lb/in) in the PM-created magnetic field. A static force, i.e. F_s=4.45 N (1 lb) is applied to the rotor to show the integral control effect. Before lift-off, the rotor leaned on a backup



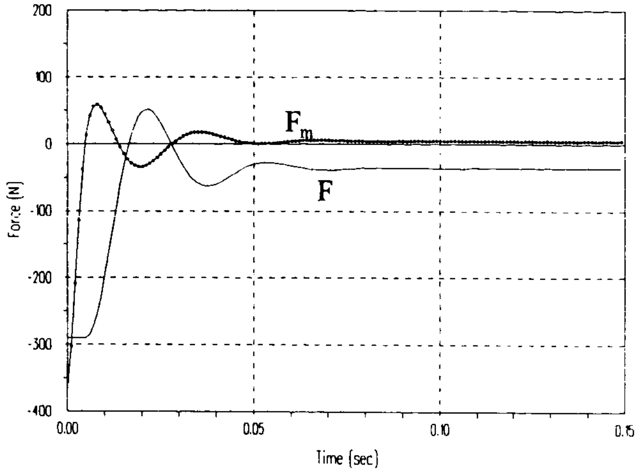


Figure 3. Lift-off transient without integral control

bearing 0.25 mm (0.010 inch) away from the center, while the stator leaned on an opposite side stop, also 0.25 mm away. The stator moved over toward the shaft side to create lifting force when the control began. The over-shooting rotor displacements in Figure 3 are the result of the chosen complex conjugate root pair being not well damped. The figure also shows the importance of the integral control. Without it, a large static force can make the rotor so eccentric that the rotor may not be able to lift off the backup bearing. The static displacement offset also causes a static control force (F). The rotor eccentricity inside the stator is opposite to the static load direction. This phenomenon is also a common feature in sensor-less magnetic bearings [3].

Figure 4 shows that with integral control, the steady-state shaft displacement and control force are eliminated.

In the case of Figure 5, the shaft backup bearing clearance was reduced by a factor of two, i.e., from ±0.25 mm to ± 0.125 mm. Comparing Figure 3 with Figure 5, it is interesting to learn that the maximum control force was also reduced by a factor of two.

Every bearing has its stiffness and damping properties, and so does a SCMB. Obviously, when the stator is not under control, a SCMB bearing has a negative stiffness coefficient ($-K_m$) and no damping.

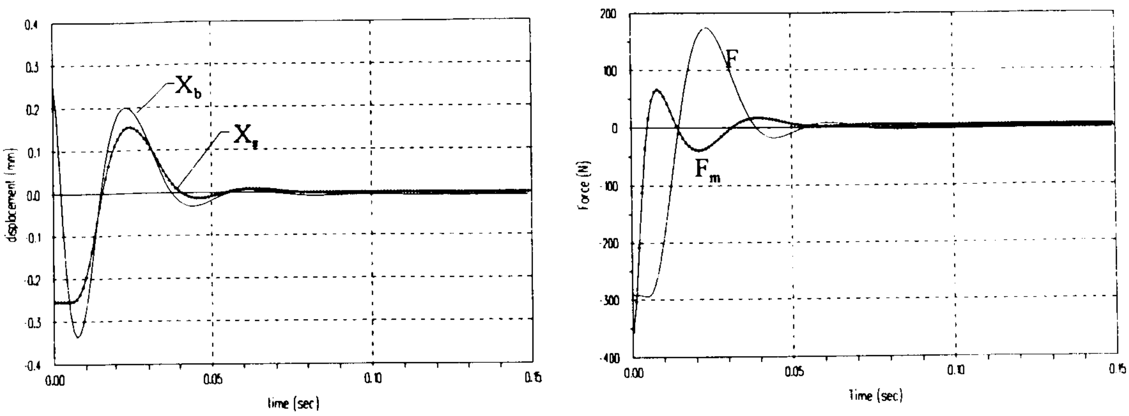


Figure 4. Lift-off transient with integral control

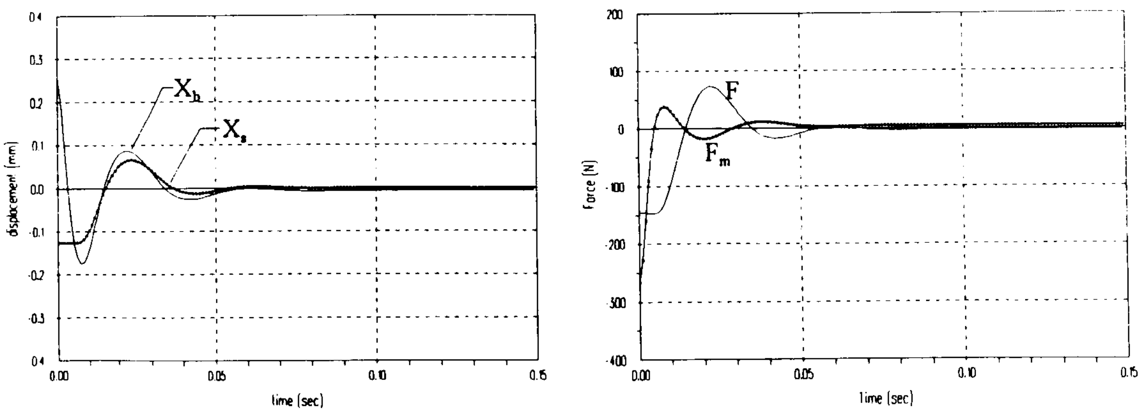


Figure 5. Lift-off transient with less shaft excursion

When the stator is under control, it can be readily shown that the effective SCMB dynamic stiffness is:

$$K_{dyn} = K_m (X_b - X_s)/X_s$$

$$= K_m[(C_p-K)+(C_d-C)S+C_i /S -M_bS^2]/[M_bS^2+CS+(K-K_m)] \quad (6)$$

Apparently, the dynamic stiffness is a function of the stator mass M_b among other parameters. For the above example, the normalized dynamic stiffness (K_{dyn}/K_m) is plotted in Figure 6. The phase plot in this figure shows that positive damping only occurs at the system natural frequency, i.e., in a normalized frequency range between 0.3 to 1.2. One can extend this range by choosing other sets of system roots to cover other system natural modes, if needed.

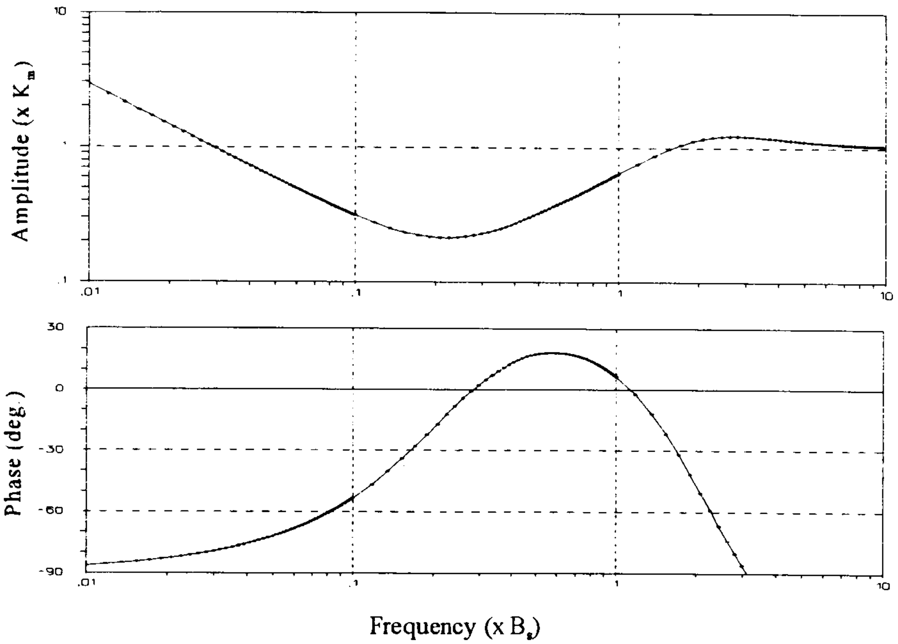


Figure 6. SCMB dynamic property

BEARING COMPONENT DESIGN

The key parameter of a SCMB is the magnetic stiffness K_m which dictates the size of the bearing. The bearing design work starts with a given value of K_m which relates to a given rotor mass. The magnetic stiffness is a function of the magnetic flux density (B in Tesla), the nominal concentric air gap (g in meter), and the circular pole area (A in meter squared). It has a closed-form solution as presented in [4] which may be simplified as:

$$K_m = (AB^2/2\mu_o)/g \qquad \text{N/m} \tag{7}$$

where

$A = \pi DL$ = circular pole area of one disk, m^2
D = rotor diameter, m
L = axial thickness of one stator disk, m
μ_o = permeability of free space = $4\pi x 10^{-7}$ Tesla/A-T

Iterative calculations are involved in using (7) and choosing the proper values of rotor diameter, disk thickness, air gap and achievable flux density. Once the flux density B is determined, the sizes of a PM ring, i.e., its thickness and axial area, may be estimated using a conventional method where flux leakage factors are considered.

The clearance (g_b) between the rotor and a backup bearing is smaller than the magnetic air gap g. For a given stator support stiffness (K), the required actuator force capacity is directly proportional to the clearance or approximately equal to "Kg_b". Therefore, the backup bearing clearance should be kept to a minimum as practically possible.

A SCMB can be made with axial bearing stiffness if small tooth pairs are machined on both the rotor and the stator disk inner diameters. This reluctance centering type of passive bearing has been studied and documented in [5]. Therefore, using radial SCMBs, an active thrust magnetic bearing may be spared.

CONCLUSIONS

The stator-controlled magnetic bearing concept has a laterally moveable stator without protruding poles to face the rotor. The annularly distributed radial magnetic flux provided by a permanent magnet ring in the air gaps are uniform circumferentially. There is no concern of eddy current or magnetic hysteresis losses. Since it is an actively controlled magnetic bearing, its stiffness and damping properties can be electronically manipulated. Therefore it is ideal for supporting high speed rotors, such as those of momentum and energy storage flywheels.

In this paper, a method has been presented on how to design the new bearing concept. The presentation included a concise formulation of the rotor and stator control dynamics and a procedure for determining the system parameters to achieve stability. Through numerical simulations, it has been shown that the inherently unstable bearing would work under proper stator motion control. Also presented were how to size the bearing proper and the required actuator force capacity.

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