AMBIENT-TEMPERATURE PASSIVE MAGNETIC BEARINGS FOR FLYWHEEL ENERGY STORAGE SYSTEMS

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ABSTRACT

Based on prior work at the Lawrence Livermore National Laboratory ambient-temperature passive magnetic bearings are being adapted for use in high-power flywheel energy storage systems developed at the Trinity Flywheel Power company. En route to this goal specialized test stands have been built and computer codes have been written to aid in the development of the component parts of these bearing systems. The Livermore passive magnetic bearing system involves three types of elements, as follows: (1) Axially symmetric levitation elements, energized by permanent magnets., (2) electrodynamic "stabilizers" employing axially symmetric arrays of permanent magnet bars ("Halbach arrays") on the rotating system, interacting with specially wound electrically shorted stator circuits, and, (3) eddy-current-type vibration dampers, employing axially symmetric rotating pole assemblies interacting with stationary metallic discs. The theory of the Livermore passive magnetic bearing concept describes specific quantitative stability criteria. The satisfaction of these criteria will insure that, when rotating above a low critical speed, a bearing system made up of the three elements described above will be dynamically stable. That is, it will not only be stable for small displacements from equilibrium ("Earnshaw-stable"), but will also be stable against whirl-type instabilities of the types that can arise from displacement-dependent drag forces. or from mechanical-hysteritic losses that may occur in the rotor. Our design problem thus becomes one of calculating and/or measuring the relevant stiffnesses and drag coefficients of the various elements and comparing our results with the theory so as to assure that the cited stability criteria are satisfied.

INTRODUCTION

Modern flywheel energy storage modules (electromechanical batteries) typically employ fiber-composite rotors spinning in vacuo at speeds of order 50,000 RPM. For important applications of the these EMBs it is required that the bearings employed should not only have low losses but also that they should be maintenance-free, their cost should be acceptably low, and their operating lifetime should be decades-long. Passive magnetic bearings could represent an ideal answer to meeting these severe requirements. With these requirements in mind a joint effort is underway at the Trinity Flywheel Power company and the Lawrence Livermore National Laboratory to adapt passive magnetic bearing concepts developed at the Laboratory [1,2] for application to flywheel modules of the types that have been developed at Trinity. These modules are typically high-power units (up to 350 kW per module) storing of order 1 kWh of electrically recoverable energy. The rotors of these units are in the form of hollow cylinders made of filament-wound carbon-fiber/epoxy composite material. The diameter of typical rotors is 25 cm., and their length is 30 cm. On their inner surfaces they carry permanent-magnet bars arranged to form a dipole Halbach array [3]. The Halbach array thus comprises the rotating field element in an ironless generator/motor whose three-phase stator is located within the rotor, isolated from the evacuated region by a vacuum barrier made of insulating material.
In the flywheel units now being considered for fitting with passive bearings, the bearing assembly (presently ball bearings) provides pendulous support for the rotor; no lower bearing is employed. The reason that this arrangement is a stable one in the presence of high torque loads on the rotor during the discharge cycle derives from the dipole nature of the generator/motor field. In such a case the generator itself imposes no displacement-dependent torques on the rotor that might produce whirl-type rotor-dynamic instabilities. Furthermore, the rotor itself is sufficiently stiff against flexural modes to guarantee that it behaves as a rigid body over its entire speed range. Under these circumstances, the primary requirement of the bearing

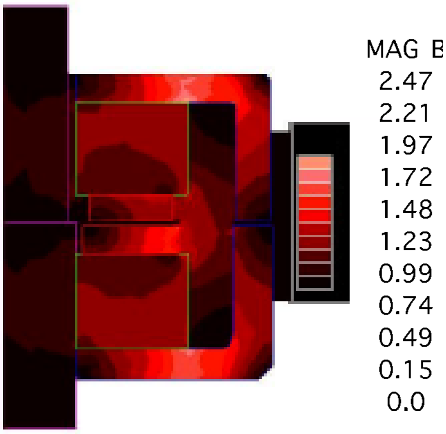

FIGURE 1: Section drawing of passive bearing element with calculated field strengths in kilogauss indicated by shading.

system is that it should stably levitate the rotor mass and should of itself be stable against whirl-type rotor-dynamic instabilities.

Given the requirements just described, our task is define a mutually compatible set of bearing elements that, together, will comprise a passive magnetic bearing system capable of levitating rotors of the weights given above. To this end a combination of computer-based simulations, theoretical formulae, and test-bench measurements is being employed in order to arrive at a viable design. In this report we will describe the progress to date in the development

The Bearing Elements

As described in previous publications [1,2] the principle elements of our passive bearing system are three in number. They are: (1) a set of axially symmetric permanent-magnet energized, levitating elements, (2) "Halbach stabilizers" comprised of cylindrical (or planar) arrays of permanent-magnet bars that interact inductively with shorted windings to produce centering forces (positive stiffness) as required to overcome the inherent Earnshaw-instability of the levitating elements, and (3), a centrifugally disengaging mechanical bearing system that performs the dual role of, first, maintaining stable levitation until the rotor is spinning above a low "critical speed," and, second, acting as a "touchdown" bearing in the case of shock or seismic loads on the system that exceed the load-carrying capability of the magnetic passive bearing elements.

For the levitating elements we are [presently considering various possibilities. The first of these is a repelling-type element in the form of a soft-iron "pot" within which is embedded permanent-magnet material that energizes the poles of this assembly. The outline form of one of these elements is shown in Figure 1, which depicts the output of a computer program analyzing the magnetic fields produced by this element.

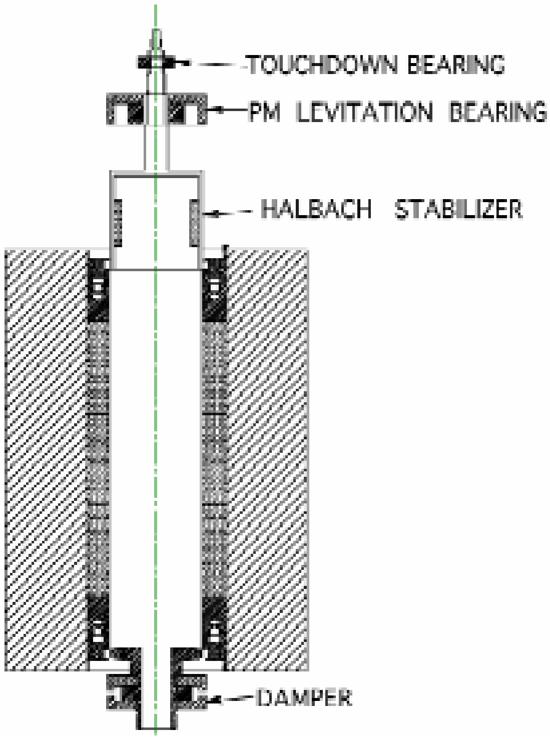

FIGURE 2: Section drawing of pendulous passive magnetic bearing system supporting the flywheel rotor

The type of Halbach stabilizer that we are presently considered is of the cylindrical type, i.e., one in which the rotating part consists of a high-order Halbach array surrounding a special "stator" consisting of a close-packed array of shorted circuits. In its simplest form the stator consists of an array of elongated window-frame-shaped shorted turns. In another form, the "window frames" are of the full width of the stator, with stacked closing conductors on the top and bottom. This latter configuration, one described in a previous publication [1], has the potentiality for very low losses since in the centered position the magnetic fluxes through each circuit cancel by symmetry. Only when the rotating element is displaced do currents flow and restoring forces are generated.

For the third element, the mechanical "touchdown" bearing, we are planning to use a modified version of the presently employed ball bearings, using appropriate means for disengagement.

The rotating elements of the flywheel assembly are shown in Figure 2. Exclusive of bearing components it weighs 40 kg. and has a nominal operating speed range of 20,000 RPM to 40,000 RPM. It comprises a filament wound cylinder of carbon fiber and epoxy lined with permanent magnets used for the rotating element of the flywheel's motor generator [4]. The motor generator magnet array is designed specifically such that the interaction between the flywheel magnet array and the flywheel stator is one of pure torsion. Hence the motor generator has a zero value radial derivative and does not influence bearing performance.

Rotor-Dynamic Stability Issues: Eddy-Current Dampers

In addition to the levitating elements and the touchdown bearing, we are planning to incorporate a vibration damper of the eddy current type, similar to types that have been employed by Fremery [5]. The type of eddy-current damper that we have considered here is shown schematically in Figure 3. As shown in the figure the damper consists of two annular permanent magnet discs oriented in the attractive mode. These magnet discs are to be attached to and coaxial with the rotating system. Located midway between the two discs is a metal (copper or aluminum) sheet. When the magnet discs are rotating, and their axis of rotation is fixed in space, the magnetic field is constant at the disc and no eddy currents are induced. However, if the axis of rotation moves transversely the disc will be exposed to a time-varying magnetic field and eddy currents will be generated in such a way as to oppose and damp the motion in a way similar to viscous damping. It is important to note that the damper plate is stationary, while the damping magnets are rotating. If the alternative arrangement, i.e., stationary magnets and rotating disc, were to be used, the "damper" itself would become a driving source for whirl instability.

In ref.[2] a derivation of the damping coefficient for such a damper was given, with the resulting equation for the damping coefficient:

$$\beta = \frac{\pi t}{\rho}\int B_z^2 r dr \qquad (1)$$

Here B_z is the z-component of the magnetic field on a plane midway between the two disc magnets, t (meters) is the thickness of the conducting sheet of metal between the magnets, and ρ (ohm-meters) is its resistivity. The derivation was made under the assumption that the inequality $t(b-a) << \delta^2$ is satisfied, where b (m.) is the outer radius of the annular magnets, a (m.) is the inner radius, and δ (m.) is the electrical skin depth at the frequency of the motion of the conducting sheet. Satisfaction of this criterion assures that the magnetic field will remain approximately constant within the conducting disc during its motion, i.e., that skin effects will be small. In the measurements to be reported the thicknesses and velocities tested were such as to span the regime (at low velocities) where this criterion was satisfied, up to higher velocities and/or greater thicknesess of the metal, where it was not satisfied, with a consequent major reduction in the value of the damping coefficient. In the design of systems using this type of eddy-current damper to control whirl instabilities it is important to take this source of non-linearity in the damping coefficient into account in order to insure that stability will be maintained over the range of operating speeds and displacements.

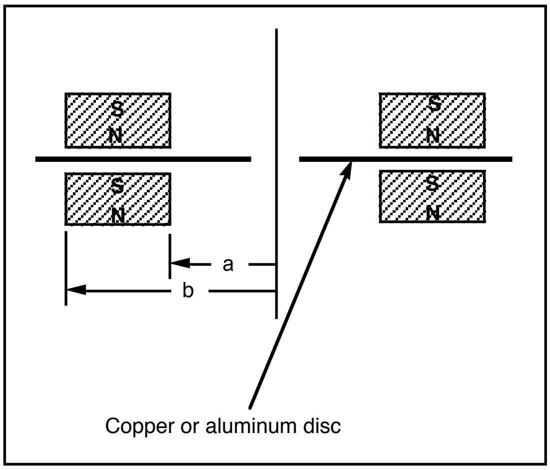

FIGURE 3: Schematic drawing of eddy-current damper

Three independent means for determining the damping coefficient of eddy-dampers of the type described here were investigated and cross-checked with each other. In a given situation whichever one of these methods is easiest to employ can be used for the design. Before describing the results from the three methods we note a relationship, one that can be derived from equation 1, that can be used to perform an indirect evaluation of the damping coefficient through a simple force measurement. We note that the integral in equation 1 is same integral that gives the attractive force between the two magnets, as it is proportional to the integral of the Maxwell stress tensor across a plane midway between the two magnets. It follows that the damping coefficient can be deduced directly from a force measurement through the following relationship:

$$\beta = \left[\frac{\mu_0 t}{\rho}\right] F \qquad (2)$$

The second method for determining the damping coefficient is to calculate the z-component of the magnetic field between the discs and use this result to evaluate the integral in equation 1. Such a calculation is also useful when evaluating the levitating force between annular magnets in designing other elements of the passive magnetic bearing system. The method we employed in evaluating the field was to integrate the expression for the vector potential of a loop current to create surface currents that correspond to the Amperian currents representing our permanent magnet material, and then to integrate this expression to determine the z-component of the magnetic field.

Eddy-Current Dampers: Experiment

A direct measurement of the damping coefficient was made by constructing a test rig that consisted of two annular disc magnets mounted on the end of a pivoted lever arm. The length of the lever arm between the pivot point and the center of the disc magnets was 61.

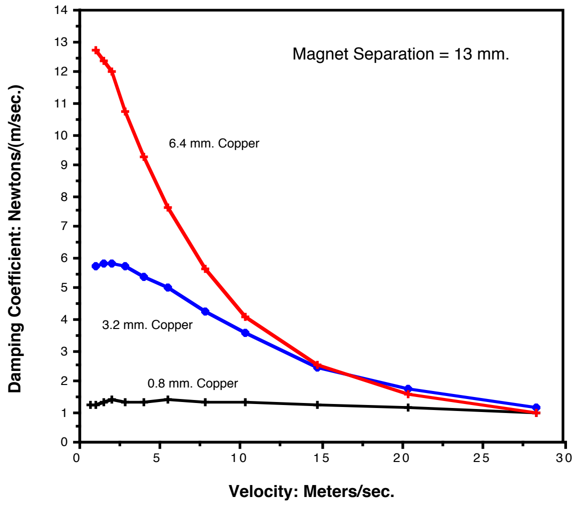

FIGURE 4: Plotted values of measured damping coefficients as a function of velocity.

cm. This lever arm rested on a load-cell (located at 30.5 cm. from the pivot point) to measure the forces arising from eddy currents induced in a rotating copper disc positioned between the magnet discs. The copper disc was much larger in diameter than the magnets, which were positioned near the outer edge of the disc. In this way the field between the magnets passed through a moving conductor whose velocity was approximately constant within the gap between the magnets, thus closely simulating the situation that would arise in a damper using the same magnets and the same thickness of damper plate. The lever assembly was mounted on the tool post of a lathe, while the copper disc was attached to a mounting plate that was held in the lathe's chuck. The velocity of the copper surface between the two magnets was then determined from the radial position of the magnets and the rotation speed of the chuck. The latter could be varied over a sufficient range to simulate the range of velocities that might be encountered when the damper was incorporated into a passive bearing system.

The magnet discs were commercial NdFeB magnets with a nominal remanent field of 1.23 Tesla. They were 0.635 cm. in thickness, with an outer diameter of 4.45 cm. and an inner hole diameter of 1.27 cm. Several copper discs were employed, with thicknesses of 0.8 mm., 1.6 mm., 3.2 mm., and 6.4 mm. The distance between the center of each disc and the center line of the magnet discs was 15.25 cm.. To determine the velocity of the copper surface between the magnet discs. this distance was multiplied by the rotation speed of the lathe chuck in radians per second. This conversion factor (between lathe RPM and conductor velocity in meters/sec.) is: Velocity (m/sec.) = .0157 x (rotation speed in RPM). The rotation speeds varied between 45 RPM and 1800 RPM, corresponding to velocities between 0.71 m/sec. and 28.2 m/sec.

FIGURE 5: Magnetic bearing element test stand

Torques on the disc caused by eddy currents were measured by taking the difference of the load-cell readings obtained when the disc was at rest and when it was spinning at a chosen speed. The measured force in Newtons was then divided by a factor of 2 to take into account the 2:1 lever ratio of the test rig. Dividing this number by the velocity of the conductor between the magnets in meters/sec. yields the damping coefficient β (Newton-sec./meter). In the data taken several different spacings between the magnets were used, to investigate the variation of the damping coefficient with magnetic field intensity.

The essential content of the measurements taken can be represented by the graphs, shown in Figure 4. As can be seen at low velocities the damping coefficient is directly proportional to the thickness. However, as the velocity is increased the damping coefficient for the thicker discs drops to values approaching that of the thin disc, showing the emerging dominance of the skin effect in determining the damping.

Bearing Component Testing Apparatus

An apparatus has been designed and built to measure all the relevant forces and torques of our magnetic bearing components. A photograph of this apparatus is shown in Figure 5. In the tests two bearing components are held in proximity to each other. The upper bearing component is rotated at up to 50,000 RPM. The lower bearing component is translated in x, y, and z in increments of 0.025 mm.. A total displacement of up to 25. mm. is available but the clearances of the components are much less than this and the full range of travel is not used. The non-rotating components are mounted on a force measuring stage that measures x, y, and z force and x, y, and z torque. The entire

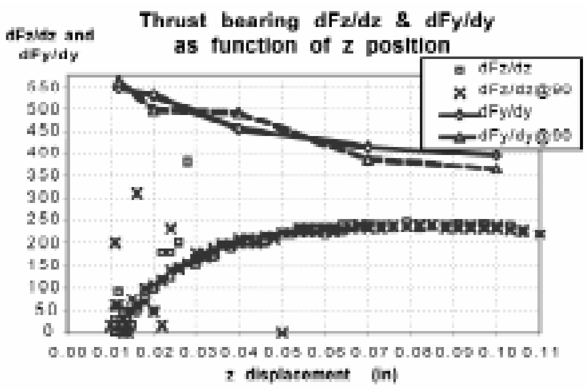

FIGURE 6: Measured lateral and axial force derivatives for passive bearing element depicted in Figure 1

mechanism is operated in rough vacuum in order to eliminate the influence of aerodynamic drag on the measurements.
An example of some of the measurements that have been performed using the component tester is shown in Figure 6. In the figure plots various components of the axial and transverse force derivatives as a function of separation for the levitation element shown in Figure 1. In the plot the force derivative (in English units of lbf/inch) is plotted against the separation of the discs in inches. In these measurements, the components are displaced with respect to one another while not spinning. Two sets of data are overlaid. The sets differ only in that the components are rotated 90 degrees with respect to one another between the two sets. These data show that the components are axisymmetric. Note that the ratio of transverse stiffness to axial stiffness is not a constant factor of -2, as would be the case for simple axisymmetric disc magnets. This deviation from theory is due to the use of material in the magnetic circuit with a flux-dependent permeability.

Measurement on a Halbach-Array Stabilizer
Measurements were performed that mocked-up aspects of a Halbach array stabilizer. A high-order Halbach array was constructed on the inner surface of a short cylinder. This assembly was mounted on the rotating element of the test apparatus. A test stator winding was placed inside this array on which measurements were made. The stator comprises diametrally wound loops of Litz wire, configured in the flux-canceling geometry as described in Reference 1 Litz wire is used in the windings to suppress parasitic eddy currents. When centered, the EMF created by passage of the Halbach poles cancel out and no current flows. When the stator is displaced, a non-zero net EMF causes current to flow in the loop. At operating speed, self inductance of the loop causes a phase delay of 90 degrees between the current flowing in the loop and the magnetic field of the array. The force developed from the current flowing in the stabilizer stator and the field of the magnet array is a repulsion force and therefore a positive radial stiffness. The stator winding used in the tests shown in Figure 7.

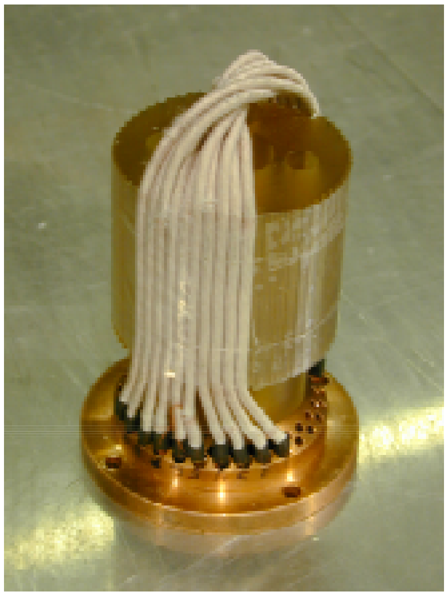

FIGURE 7: Photograph of test flux-cancelled stator winding for Halbach stabilizer

Halbach Stabilizer Stator
The first step in validating this architecture is to measure the net EMF developed in a diametral loop of wire. A sample measurement is shown here. Instead of using the dedicated test stand, these measurements were obtained by supporting the array in a precision milling machine for extremely precise control of relative rotor-stator displacement. The qualitative behavior of the voltage follows the theoretical prediction. That is, no voltage is generated in the centered position, and the voltage generated rises linearly with displacement from the centered position. However, in the measurements the net EMF corresponded in terms of voltage per unit frequency per unit displacement exceeded the theoretically predicted values by as much as 40%. We do not at present know the origin of this discrepancy. One possible origin of the discrepancy: errors in the measurement of the parameters of the Halbach array used to excite the test windings.
The voltage and current measurements which have been performed are consistent with a radial stiffness of about 260,000 Newtons/meter (1500 lbf/in), which would adequately stabilize the repelling bearing element described earlier.

Tests of a Pendulous Rotor
Extensive tests have been conducted using a flywheel rotor supported via a compliant suspension with parameters comparable to those planned for the magnetic levitation system. In these tests the flywheel rotor is hung from a single ball bearing. The bearing is mounted in a cooled jacket suspended by low-stiffness flexural mounts. No damping was used other than the hysteretic damping of the flexure spring material. The operating experience with this test flywheel shows that it passes through one or more low-speed pendulum and

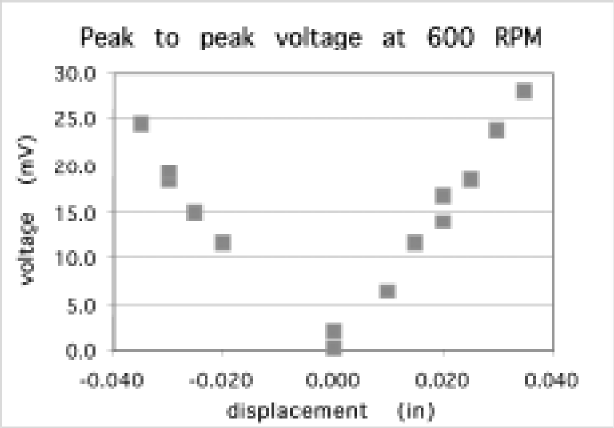

FIGURE 8: Measured circuit voltage output as a function of lateral displacement for flux-cancelled Halbach stabilizer

suspension modes (in the speed range of several hundred to several thousand RPM), Once above these suspension-related modes the flywheel is stable and nearly silent in operation.

Status and Future Directions

The results that have been obtained to date en route to a final design for a passive bearing system for flywheel energy storage units confirm the general validity of the analytical models given in References 1 and 2. Components with design values sufficient to attain passive levitation of current flywheel rotors have been built and are being characterized. A number of practical matters pertaining to the design and implementation of the levitating bearing component and the Halbach-array stabilizer have been identified. No fundamental obstacles to the eventual implementation of a fully passive magnetic bearing system are evident at this time.

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