Advantage of superconducting bearing in a commercial flywheel system

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Abstract. The use of a superconducting magnetic bearing in an Urenco Power Technologies (UPT) 100kW flywheel is being studied. The dynamics of a conventional flywheel energy storage system have been studied at low frequencies. We show that the main design consideration is overcoming drag friction losses and parasitic resonances. We propose an original superconducting magnetic bearing design and improved cryogenic motor cooling to increase stability and decrease energy losses in the system.

1. Introduction
Today, high-Tc superconductors are employed in a number of large scale mechanical devices like magnets, motors and bearings supporting flywheels for energy storage. The latter have an evident advantage in using levitation phenomenon [1]: flywheels with the superconducting bearings have an idling energy losses about 0.001 % per hour, while the best commercial flywheels on mechanical bearings about 0.1 % per hour. But this reduced friction is accompanied by a reduction in stiffness, leading to more complicated dynamics. Being essentially nonlinear, this dynamic complexity often limits the efficiency of the high-end flywheel devices.

A lot of efforts in the last years were concentrated on complex study of different superconducting flywheel prototypes [2, 3, 4, 5, 6] and modelling of it’s dynamics [7, 8, 9]. In addition to these studies, it is very important to study the behaviour of the HTS bearing in a real commercial energy storage system to shorten the distance between applied science and practical usage. In this paper, we study the prospect of using a superconducting magnetic bearing in the UPT 100kW commercial flywheel.

2. Conventional UPT 100kWt flywheel energy storage system
2.1. Overall design
Urenco Power Technologies has developed a kinetic energy storage system based on a high-speed vertically placed rotor (see Figure 1). The rotor is a cylinder with an open central bore and is made from glass and carbon fibre composite material. It is about 90 cm in length, 33 cm in outer and 17 cm in inner diameter and has a mass of 110 kg. In the center of the bore is a steel shaft that carries the one half of top magnetic bearing and the stator of the motor (generator). The motor is used to speed-up the rotor during charging mode and to generate electric power during discharge mode. The stator has a conventional design of steel laminations used with

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a three-phase star connected winding. Heat is removed from the stator by circulating cooling water through the central shaft.

The inner side of the rotor is loaded with NdFeB magnets formed from powder embedded in the glass fibre matrix. At its upper end, magnetic poles are made circumferentially to form one part of a passive magnetic bearing. Magnetization pattern consists of 14 alternating north and south poles of 12 mm length. The maximum surface field of the bearing is 0.25 T, the radial stiffness is 230 N/mm and the rotor-stator gap as small as 1.5 mm. The central part of the rotor is magnetized axially with twelve poles that form the rotor of a permanent magnet motor.

![Diagram of the UPT flywheel energy storage system](image)

**Figure 1.** Overall design of the UPT flywheel energy storage system.

At its lower end, the rotor is supported by high-speed low-loss oil film bearing. The first part of the bearing consists of a steel rod attached to the centre of the rotor by a plane disk, while the other end of the rod is terminated with a ball. This ball is inserted into the metallic glass with a small cup in the centre, which matches the ball with high accuracy. The glass is filled with oil, so that ball and cup are separated by a film of oil during rotation. All this glass is connected to the bearing base by means of three steel struts, which support the mass of the rotor and provide the stiffness of the bottom bearing. The stiffness is inversely proportional to the cube of the strut’s length and needs to be adjusted with the stiffness of the top bearing for stable operation. The glass, struts and the base form the inner part of the bearing, which is placed inside a metallic housing filled with another oil. During vibrations the inner part of the bottom bearing is moving inside viscous oil and this damps its energy. The damper suppresses the amplitude of parasitic vibrations of the rotor while it passes through resonances. Due to its original design the bottom bearing has negligible loss and require no maintenance.

The above assembly is placed in a steel cylinder case which is pumped down to 1 mbar. The system has its own drive and control unit that provides full operational control. When the

**Figure 2.** Free spin-down dynamics of the UPT rotor. Arrows show resonance regions where deviations from linear dependency is observed.
flywheel is connected to an electrical power network it accumulates energy as the kinetic energy of the rotor, which spins with frequency up to 600 Hz. Then it can discharge this energy to the network in case of power failure.

2.2. Dynamics: energy loss and resonances

The speed-down dynamics of the machine can be studied by looking at the dependency of rotational frequency on time, Figure 2. The speed of rotation decays by a linear law

\[ \omega(t) = \omega_0 - \left(\frac{M_{fr}}{I}\right) \cdot t, \]

where \(M_{fr}\) - the torque of friction forces, \(I = 1.82 \text{ kgm}^2\) is the moment of inertia of the rotor. From the experiment fit we have \(M_{fr} = 0.053 \text{ Nm}\) and it is equal to the energy loss per period of rotation (multiplied by \(2\pi\)) - 0.33 Joule. These losses mainly originate from magnetic hysteresis in the steel parts of the motor’s stator which pass through the magnetic poles on the rotor.

Linear dependency is only disturbed near regions of rotor vibration. There are two such regions around 24 and 10 hertz. Figure 3 shows detailed resonances structure for top and bottom bearings and middle part of the rotor. The first peak at 9.7 Hz is a cylindrical mode, where all of the rotor shifts in a radial direction. The damping of this oscillation is very small, about 1.7 Hz and Q-factor is about 6. It means that the most part of energy of oscillation could be returned back to the rotation energy when the frequency is changed, however due to low damping the amplitude of oscillation is quite high and could result in damage to the machine.

The second peak at 24.7 Hz is smaller, with damping of 7 Hz and Q-factor 3.5 and represents a pseudo-conical mode, where rotor tilts or turns around its mass centre or point near the bottom bearing. One can see, that vibration in the bottom bearing is stopped at 18 Hz, while the top bearing has an unclear peak, which looks like long tail of the first peak at 10 Hz. The damping here is much stronger as well as the stiffness.

The machine has only a damper in the bottom bearing, where the ball and cup bearing assembly moves in viscous oil. This requires similar stiffness of top and bottom bearings for stability. In the limit, with very stiff bottom struts, the bottom damper does not move and so there is no damping. The top magnetic bearing in a working position has theoretical radial stiffness of 230 N/mm and very small axial stiffness. The estimated stiffness from experimental resonance frequencies is 208 - 313 N/mm. It should be accurately positioned to align magnetic poles and to allow the bottom bearing to support the weight of the rotor.

![Figure 3](image-url)  
**Figure 3.** Rotor’s resonance peaks. Amplitude of radial vibrations in top, middle and bottom parts of the rotor.

![Figure 4](image-url)  
**Figure 4.** Cross section of superconducting magnetic bearing for commercial flywheel system.
2.3. Superconducting bearing construction

It is well known about the advantages of superconducting bearings in high-speed rotational machines [1, 7]. In this case, we plan to install a superconducting journal bearing instead of the passive magnetic one. The superconducting bearing has good stiffness in both radial and axial direction, has very low AC energy loss for rotation and provides additional damping during vibration.

The new superconducting bearing consists of a steel cryostat for liquid nitrogen which is fixed to the central shaft and a set of YBCO high temperature superconductors glued to the outer part of the cryostat, Figure 4. The outer diameter of the HTS bearing is 167 mm, inner diameter is 96 mm and length is 175 mm. It will be filled with 1.8 L of liquid nitrogen via two tubes guided through the top flange of the main cylinder case. As the bearing works in vacuum chamber, it will have rather good thermal insulation. To improve it further, tufnol spacers will be used to avoid direct contact to metallic parts of the stator. Thermocouple based temperature monitoring will be used to watch the temperature of the superconductors. Seventy five superconducting samples with thickness 5 mm and diameter 31 mm are needed to build this bearing. All construction should be done with high accuracy as the gap between superconductors and rotor should be kept as small as 1.5 mm to achieve high bearing stiffness. In order to stabilize the rotor in a central position while the superconductors are above their critical temperature we left one ring of permanent magnet cut out from the original magnetic bearing.

Let us assume, that the superconductors is field cooled in magnetic field $H_0$ at a distance $z_0$ from the rotor magnetic wall. In this case, field $H_0$ is trapped inside the sample. If the distance is changed by $\Delta z$, then the field acting on the superconductors will be $H$. Now, the superconductor will act as a diamagnet in respect to any changes of external magnetic field. For the ideally diamagnetic superconductors cooled at field $H_0$ we have $B/\mu_0 = H + \chi(H - H_0)$, where $\mu_0$ is magnetic permeability of vacuum, $\chi \to -1$ for perfect diamagnetism. Assuming the above $B - H$ relationship we estimated the field distribution, force and stiffness of the bearing using the finite element package software Comsol multiphysics. We obtained a stiffness value of 140 - 180 N/mm for the top superconducting bearing.

The model does not account for a small equilibrium magnetization of the field cooled superconductor, finite $J_c$ and possible force hysteresis. Also in the real case we do not have a continuous superconducting wall. This factor can be treated by calculating the stiffness per unit area of superconducting surface and then multiplying by the appropriate fill factor. In conclusion, it seems that the model overestimates stiffness, while data from the experiment gives the combined stiffness of the whole rotor.

To estimate damping in superconducting bearing one can use a formula for hysteresis AC loss in hard HTS bulks from [10]:

$$w(\rho) = (c/24\pi^2)h^3(\rho)/J_c.$$  

Here $w(\rho)$ is surface density of energy loss per period, $\rho$ is polar space coordinate, $c$ is the velocity of light, $h(\rho)$ is AC magnetic field at superconducting surface, which in case of ideal hard superconductors is twice that without the superconductor due to a local shielding current. In our case $h(\rho) > 0.28$ T, $J_c = 10^4$ A/cm$^2$, the area of superconducting surface is 876 cm$^2$. This gives the value of energy loss per period 0.08 Joule, in comparison with total energy of oscillation of 0.22 Joule for 1 mm amplitude and resonant frequency 10 Hz.

Another approach to calculate AC losses is to model the superconductor using a finite element method. Here the $E - J$ constitutive law together with an $H$-formulation is used to calculate the current distribution and electromagnetic fields in HTS, then the forces of the interaction between the magnet and the superconductor and the AC loss inside the superconductor can be obtained. This numerical method is based on solving the time dependent partial differential equations and is adapted to the commercial finite element software Comsol Multiphysics [11, 12].

In our case we define the superconductor by electric field law $E(J) = E_0(J/J_c)^{21}$ and place it in the AC field of a permanent magnet, Figure 5. The result shows, that critical currents...
Flow only in a thin undersurface layer of the superconductor and screen the superconducting volume from external AC magnetic field. The force between magnet and superconductor can be readily calculated, knowing current and field distribution, Figure 6. This figure also shows the force dependence if we model the superconductor as a diamagnet with an effective magnetic permeability $\mu_{\text{eff}} = 0.06$. Finally, the AC energy losses in the superconductor can be evaluated as ohmic losses by the formula $Q_{\text{loss}} = \int V_{\text{SC}} J \cdot E \, ds$. This yields $8.3 \times 10^{-4}$ Joule for one sample and $0.06$ Joule for the total bearing which consists of 75 samples. So, our estimation shows that the superconducting bearing could damp 27-36% of the total energy of oscillation.

3. Conclusion
We have studied the dynamics of a conventional UPT 100 kW flywheel system supported by passive magnetic and hydrodynamic pivoted bearings. We have found, that the energy losses in the system are mainly due to magnetic hysteresis in the steel part of motor’s stator. There are two low-frequency vibration modes: cylindrical and pseudo conical, which lead to instability and additional loss of kinetic energy. New superconducting bearing design has been proposed to replace the magnetic passive bearing. Using the superconducting bearing will provide good stiffness and add additional damping to the rotor.

Acknowledgments
The authors would like to thank N. Hari Babu and K. Iida for providing superconducting samples and technical assistance. This work has been supported by the EPSRC grant No. GR/S70364/01

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Figure 5. Critical state model of superconductor in AC field of permanent magnet. The surface plot is the critical current density, the streamline is the magnetic field. An arrow indicates the direction of permanent magnet magnetization.

Figure 6. Calculated dependency of levitation force on the distance between magnet and superconductor. Comparison of solutions by critical state and diamagnetic models of superconductor.
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