Flywheel Challenge: HTS Magnetic Bearing

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**Abstract.** A 200 mm cylindrical engineering prototype high temperature superconducting (HTS) was designed and fabricated. Measurements show that the 17 kg PM rotor can suspend safely 1000 kg in axial direction and 470 kg radially. The rationale for the bearing performance is to stabilize a 400 kg rotor of a new compact 5 kWh/280 kW flywheel energy storage system (COM – FESS). Measurements of the magnetic bearing force, stiffness and drag-torque are presented indicated the successful targeting a milestone in the HTS bearing technology. The influence of the PM configuration and the YBCO temperature on the bearing performance was experimentally studied, providing high-force or high-stiffness behaviour. The axial stiffness 5 kN/mm at 0.5 mm displacement is the highest value of a HTS bearing we know.

1. **Introduction**
Flywheel Energy Storage Systems (FESS) and turbo-machinery would have the largest benefit of the low-drag torque and the unlubricated wear-free operation of high temperature superconducting (HTS) magnetic bearings. Most of the HTS bearings have been developed in connection with magnetically suspended flywheel projects [1-3]. Melt textured superconducting material with high Jc at 77 K is now available in larger units to built even larger bearings.

Size and geometry of a cylindrical bearing seem commensurate at best with the design of compact FESS. Axial or thrust bearings have been tested as well for flywheel application [2]. At displacement the restoring forces of those bearing types can be high, especially the repulsive forces. However, due to the zfc part in the forces the stiffness is typically low which may be impractical for rotor stabilization.

In this paper, we report the successful test of a compact Ø 200 mm HTS bearing capable to suspend one ton in axial direction. The bearing has been investigated for application in a new very compact 5 kWh / 250 kW HTS flywheel project (COM – FESS) starting in 2005. Toward an understanding of high load behaviour we conducted a number of experiments of force, stiffness in axial and radial direction, investigated cross loads and rotational losses.

2. **Compact flywheel rotor design**
A key design of a compact flywheel offering the potential of higher energy and power density in comparison with conventional energy storage devices such as chemical batteries is an advanced composite rotor. High material strength is needed to achieve maximum rotational speed. In comparison with other materials sophisticated graphite fiber reinforced composite rotors enable to store greater amounts of energy per unit weight or volume basis. Table 1 compares some flywheel materials with their maximum energy density. Most of the flywheel rotors are produced in the shape of a hollow cylinder. Fig. 1 shows the calculated achievable energy storage as a function of the inner to outer rotor diameter ratio r_i/r_o. Evidently, at the same rotor diameter and mass a thin ring close to the outer diameter can store almost twice of the energy of a rotating bulk cylinder. The energy per weight increases with the r_i/r_o ratio while the corresponding energy per volume decreases.
Practically, $r_i/r_a = 0.75$ is good compromise between the energy per weight efficiency and the absolute storable energy.

Table 1: Max. rim speed and energy density of flywheel materials

| material   | density $\rho$ [kg/m³] | tensile strength $\sigma$ [MPa] | rim speed $v$ [m/s] | energy density $E/m$ [Ws/kg] |
|------------|------------------------|---------------------------------|---------------------|------------------------------|
| steel      | 7830                   | 1300                            | 415                 | 106                          |
| titanium   | 5100                   | 1200                            | 575                 | 143                          |
| fib. glass | 1900                   | 1300                            | 680                 | 335                          |
| Graphite fiber | 1546               | 6300                            | 1570                | 1570                         |

In addition, the rotor design should follow basic rotor dynamics of a rigid body keeping the moment of inertia around the rotational axes, e.g. $J_z$ about 3 times larger or smaller than the moments of inertia of the perpendicular axes $J_{x,y}$. The reason for this is that the flywheel's rotor motion in all operating situations, e.g. accelerating and decelerating, must be a stable rotation about the axis with the most moment of inertia or the axis with the least moment of inertia. In practice, the designed rotors are cylinders 3 (or more) times longer than wide or vice versa.

With the above considerations about material and geometrical factors the applied rotational bearing determines the flywheel storage efficiency and the lifetime of the system.

3. Bearing design and hardware description

The flywheel test bearing was constructed in ATZ's laboratory. The cylindrical bearing design is comparable to the encapsulated bearing published recently [5]. The journal bearing consists of axially stapled PM rings of the size OD 200 mm x ID 150 mm x 8 mm with Fe shims between giving the well known high magnetic flux gradient periodic structure. Co-centrically at a gap distance of 2 mm the HTS ring assembled from 28 pieces melt textured YBCO bulks of the total size OD 230 mm x ID 204 mm x 120 mm is located. The YBCO ring is glued in an outer double wall Cu hollow cylinder storing about 1.5 liter LN$_2$ and is cooling the YBCO indirectly by thermal conduction. Fig. 2 shows a schematic and a photograph of the magnetic bearing (top thermal shields are removed).

The 200 mm magnetic bearing is built in a G-10 vacuum cryostat with an outer diameter of about 400 mm. The upper and lower cryostat cover is transparent for ease of observation. The narrow gap with a magnetic distance of 2 mm prevents any encapsulation of the bearing cold part. When warm, the 17 kg PM rotor is held in the field cooled place by 8 radial and 2 axial shafts going from the outside by vacuum feedthrough’s to the inner vacuum part. After field cooling the shafts are pulled back. Under magnetic forces rotation is achieved by means of an axial gear that is driven by an external motor. The total weight of the test bearing is 55 kg with 5 kg melt texture YBCO built in.
The LN$_2$ cooling was simplified by easy cooling down and LN$_2$ storage. The bearing was tested in the self-constructed G-10 vacuum chamber. Vacuum conditions are typically between $10^{-4}$ mbar and lower vacuum for measuring the increased thermal and rotational losses. The temperature of the YBCO ring was controlled by 2 thermo-couples; one in the centre of the YBCO ring (7 mm depth from inner surface), and one on the Cu container. When the HTS is encapsulated, we measured typically a temperature gradient of about one Kelvin over the HTS thickness. Because of the open PM-HTS structure (the YBCO ring sees the warm PM rotor in 2 mm distance) the typical cool-down temperature is 78 – 79 K. Experiments have been performed under LN$_2$ and sub-cooling temperatures of 70 – 72 K to study forces and stiffnesses as a function of the HTS temperature.

### 4. Bearing forces and stiffnesses

Originally, we tried to build a 500 kg bearing already proposed in prior HTS bearing developments [4, 5]. This target coincides with new flywheel project to stabilize a 400 kg flywheel rotor fully magnetic. Surprisingly, as a milestone in the high Tc bearing development we reached a maximal axial load of one ton at mild sub-cooling. The force measurements are shown in Figure 3 and Figure 4.

In Figure 3 the axial force vs. displacement curves show a strong dependence on the superconducting temperature as expected. The maximum axial forces at 3-4 mm displacement are 6000 N, 8000 N and 10000 N at 82.3 K, 78.5 K and 72.2 K, respectively. From Figure 3 it is observed that at larger displacements (higher forces) the curves changes from linear to gradual lower function with a curve bending. Only at the first mm we observe a linear displacement-force behaviour. The non-linear curve shape is a result of hysteretic behaviour where the flux lines are dislocated from their original pinning centre positions to new ones. After removing the axial forces the rotor shows an offset with respect to its original position. In case of the 1000 kg load the offset was 1.2 mm. After few load cycles the offset vanishes.

The hysteretic behaviour and rotor offset influence the performance of a superconducting bearing. Especially, the decreasing stiffness with increased rotor displacement can restrict the application. Therefore, in Figure 4 we compare the results of Figure 3 by a variation in the magnetic excitation. Again, we obtain at about 3.5 mm displacement a maximum force of almost 10000 N. In order to obtain the elastic property with a constant stiffness in Figure 4 the magnetic excitation system has been changed to lower flux density and narrowing the pole pitch (at the same PM mass).

In Figure 4 the measured radial forces are given, too. For this measurement the rotor was field cooled out of the center and moved parallel. The maximum radial force is 4700 N at 3.2 mm displacement and at a HTS temperature of sub-cooled 72 K. The linear curve shapes suggest almost no hysteretic effects and a minor influence of the PM excitation system. Correspondingly, very small or no radial...
offset have been measured. Under the influence of hysteretic effects the radial to axial force ratio is about 0.4 while for force measurements with less hysteresis the radial to axial factor is close to 0.6. To simulate realistic flywheel conditions we have measured so called radial / axial cross forces. The obtained results give evidence that under constant radial forces / displacement 1.2 mm the axial values decrease by 25 \% with respect to the load in the rotor central position. If the rotor has a constant axial load of 500 kg / 2 mm displacement, the radial forces increase by about 20%.

5. Rotational loss
Compared to the results for encapsulated bearings we expect higher rotational losses due to azimuthal inhomogeneities in the magnetic flux distribution of the rotor. These inhomogeneities produce hysteretic losses and simultaneously eddy current losses in the rotor and the bearing housing. In our bearing we have the copper container for LN\(_2\) and non-metallic (G-10) vacuum cryostat suggesting rotor eddy current losses only. In Figure 5 the free rotor decay is shown for two runs. The rotor was accelerated to 1740 and 1900 rpm and the speed degradation was observed. From the almost linear shape of the curves we exclude any contribution from air friction. By differentiating the curves we obtain the speed degradation per time (14.5 rpm/min) and calculate an averaged AC and hysteresis friction moment of the rotor of \(M_{AC + hys} = 5 \times 10^{-4} \text{Nm}\).

6. Conclusions
We have tested a 200 mm compact HTS magnetic bearing for flywheel application up to a maximal load of 1000 kg axially and 470 kg radially at 72 K. The maximal axial stiffness is 4.5 kN/mm and the radial 1.8 kN/mm. The influence of temperature and hysteretic depinning on the bearing force is investigated. The magnetic force density is 6 N/cm\(^2\) in radial direction and 13 N/cm\(^2\) with respect to axial forces. Axial – radial cross forces influence the bearing performance. The measured rotational friction \(M = -5 \times 10^{-4} \text{Nm}\) of the bearing depend on hysteretic and eddy current losses due to inhomogeneities in the azimuthal flux distribution.

7. References
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