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Towards high capacity HTS flywheel systems

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***Abstract* - ATZ and L-3 MM are currently mounting a compact designed flywheel energy storage system (FESS) with total magnetic bearing support. Final assembling and test operation is scheduled to be in autumn 2008. After calculations and experiments we decide to improve rotor stabilization by a stiffer geometry. In addition, two dynamical emergency bearings contribute to robust and safe flywheel operation in critical rpm situations. The planned energy capacity of 5 kWh is now obtained at about 8000 rpm while an increased capacity of 10 kWh will be stored at a speed of 10,000 rpm. The total weight of the flywheel unit is about 1200 kg plus power electronics and cooling system. The heavier 600 kg rotor causes new design and construction work in mechanical elements, magnetic support bearings, cooling and power electronics.**

Due to the here reported construction changes and increased rotor speed scaling even larger energy storage performance of 15 – 20 kWh seems achievable. ATZ and L-3 MM obtained a corresponding order to develop and deliver a 15 kWh / 400 kW HTS FESS for a Korean local grid UPS application.

Index Terms – **Flywheel, energy storage, rotor dynamics**

I. INTRODUCTION

Adelwitz Technologiezentrum (ATZ) and L-3 Communications Magnet – Motor (L-3 MM) have fabricated a 5 kWh / 250 kW flywheel energy storage system (FESS) using two magnetic bearings supporting the 0.6 t rotor. To date (August 2008) no HTS flywheel in continuous operation is available. Several projects are performed in a R&D status with total development costs of about some tens of Mio. US$ [1-5]. Most problems of development are associated with rotor dynamics and the rotor stabilization by the HTS magnetic bearings. Concluding the attempts and reports, it seems that magnetically supported flywheels are definitely not for those with short attentions spans. It requires a multidisciplinary team, having expertise not only in superconductivity but material science, magnetics, power electronics, vacuum techniques, and rotor dynamics. Our general flywheel concept and design follow conventional flywheels developed and applied to mobile applications by L-3 MM. Energy savings of 30 – 40 % could be obtained in the total energy balance even the rotor was supported by mechanical bearings (a PM iron configuration typically reduces the axial load). However, in long-term storage the total loss per day is 60% or more of the stored energy. Therefore, some groups started in the past the incorporation of HTS magnetic bearings making FESS concepts beneficial. From the developments and experiments the enormous complexity of a magnetically stabilized flywheel becomes evident. Rotor dynamics requires high load bearing performance, high stiffness and damping capabilities to operate the flywheel in all situations of charging / discharging. The present HTS flywheel project started in 2005. We developed and analyzed the radial HTS bearing with axial forces of 10 kN and 2 -4 kN/mm stiffness, the extremely low magnetic friction of the bearing and the damping behaviour [6]. While forces, stiffnesses and friction and their dependency on the HTS temperature satisfied our expectation, the intrinsic damping of a few percent was considered to be far from an effective rotor damping capability. Our assumptions and observations were consistent with the reports given by the other FESS developing groups.

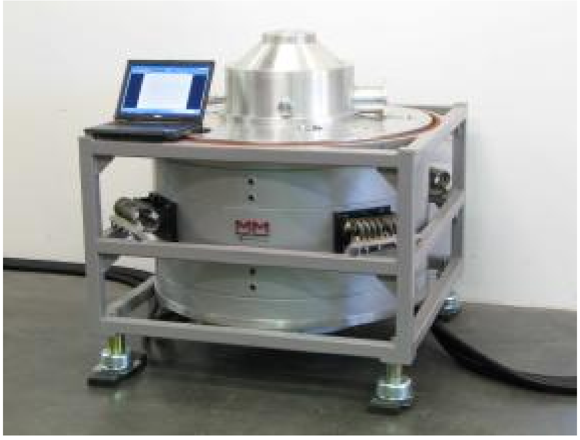

Fig. 1. ATZ / MM HTS Flywheel

Our technical concept of the HTS bearing on top and the PM bearing below the motor/generator was redesigned relative to the expected magnitude of the rotor vibration.

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Two dynamical damping systems are attached at both ends of the rotor shaft. The ATZ/MM HTS FESS in Fig. 1 with the abbreviation "MDS 12" (Magneto –Hydrodynamic Storage) is now ready assembled and will be tested for electric performance.

II. Technical Flywheel Concept

A. Overall flywheel structure

The assembled HTS Flywheel body without the power electronics is presented in Fig. 1 and the FESS parameter are given in Table I. The HTS flywheel cylindrical housing is fixed in a steel frame with four flexible carrying elements at the corners. Inside the container the components with the carbon / glass fiber rotor, the concentrical motor/generator and bearings are assembled on a central shaft. Two sets of dynamical emergency bearings and the two magnetic bearings on the both ends of the shaft have to guarantee the precise mechanical adjustment during all rotor revolution speeds. The flywheel housing is evacuated by a rough / turbo-molecular pump combination to a basic vacuum of 10^{-2} -10^{-3} mbar. Friction experiments show the influence of air friction on the fast rotating surface of the rotor up to pressure of about 5 x 10^{-3} mbar. Outgassing is expected at the beginning of the flywheel tests and will be decreased with operation time.

TABLE I: HTS FLYWHEEL PARAMETER

Parameter	value
Footprint [m²]	1.2 x 1.2
Total height [m]	1.0
Housing diameter [m]	1.1
Total weight [t]	1.2
Rotor weight [t]	0.6
Rotor diameter [m]	~ 1
Rotor speed [rpm]	10 000 max.
Storage capacity [kWh]	10 max.
Power [kW]	250

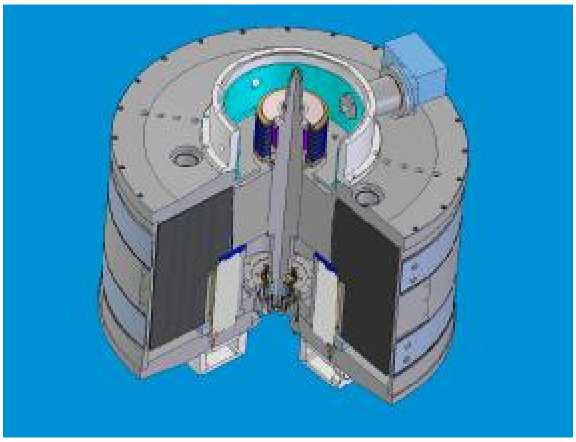

Fig. 2. HTS flywheel inner structure

The inner structure and architecture of the HTS FESS is given in Fig. 2. In the top position the upper dynamical emergency bearing is located as a mechanical touch – down bearing (not shown in Fig. 2). The HTS heavy- load bearing in cylindrical symmetry is mounted in the FESS head. It provides axial and radial stiffness to the shaft and supports magnetically the total load of the rotor. Going down in Fig. 2 it follows the composite rotor with concentrically motor/generator. Below M/G a radial PM bearing stabilizes the rotor radially but has a negative axial stiffness. This stiffness has to be compensated by the positive axial stiffness of the HTS bearing. Finally, the lower dynamical touchdown bearing with appropriate means for disengagement is assembled. It is multi -functional and supports the rotor in cases when the HTS bearing is not in operation (warm support) and in emergency situation. A 35 Watt Gifford McMahon cryo-cooler is attached to the head of the housing. The generator /motor is controlled by a power electronic unit.

B. Glass / Carbon fiber rotor

The flywheel energy storage system has a vertical rotational axis and all components are aligned to the central shaft (Fig. 2). The rotor in Fig. 3 is a hollow cylinder wounded of glass and carbon fiber rings. It has a diameter of 1 m and 0.5 m height. The rings are arranged concentrically with the stronger material towards the outer periphery where the stresses are the highest. Either some of the rings are press fitted together in pairs to impose some pre-compression in the radial direction. The ring-pairs are separated from each other by compliant elastomeric interlayer's preventing the transmission of radial stresses from one ring-pair to the next. Alternatively, all composite rings can be press-fitted or shrink-fitted to the neighbours, again with the carbon fiber rings in the outermost layer. The multi-rim rotor structure shows a more evenly stress behaviour. At a rotor speed of 10 000 rpm the maximum rim speed is about 550 m/s which corresponds to 40 Wh/kg specific energy. These values are 50% lower than the theoretical usable rotor parameters but are considered industrial "fail-safe" standard. Both the kinetic energy as well as the hoop stress of the cylindrical rotor shell is proportional to the square of the angular velocity ω^2. In consequence of the non – linear dependency most of the FESSS energy (75%) is stored between the maximum rotor speed and half of the top speed.

Fig. 3. Multi- rim flywheel

C. Motor / Generator M/G

With the motor generator (M/G) unit integrated concentrically in the hollow cylindrical rotor a highly compact structure of the flywheel is obtained. The flywheel stores energy when the M/G unit works as a motor by increasing the rotor speed. Electric energy is released when the M/G is switched to the generator mode by reducing the rotor rpm. The maximum storable energy capacity is determined by the rotor's mass moment of inertia and the maximum permissible rotational speed

D. HTS Bearing

The HTS part of the top magnetic bearing, not shown in Fig. 2, is schematically described in Fig. 4. The bearing

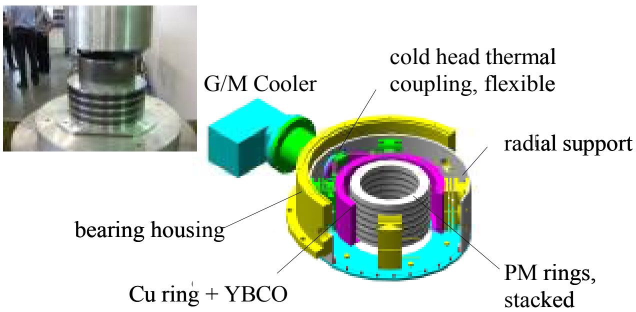

Fig. 4. Schematics of the HTS bearing structure with cold head; the insert shows part of the assembled PM rotor

itself has been developed, analyzed and tested separately [6]. The bearing is located in the head of the flywheel. The HTS part consists of multi-seeded YBCO bulks, machined into the segmented shape and glued into a thick wall copper ring. The copper ring is cooled by 35 W/80 K G/M cryo-cooler. The bearing is mechanically supported by a G-10 structure giving high stability but having low thermal conductivity. In axial direction the support structure was tested up to a load of 2 tons with a heat loss of less than 0.3 Watt.

Concentrically to the Cu/YBCO cylinder the PM ring is connected with flywheel shaft. The PM rings in the geometry OD 200 mm x ID150 mm x 8 mm are stacked together with Fe flux collectors between (Fig. 4, insert). The PM rings are sensitive to centrifugal forces and therefore are armed with a carbon fiber bandage. The bandage provides a pre-stress on PM rings to prevent cracks or damages during high – speed operation.

E. Rotor dynamics

Compared to mechanical bearings the magnetic bearings are some decade's lower stiff. Therefore, utilizing magnetic bearings for rotor stabilization the vibration should be limited to a maximum value of 0.5 mm. First calculations and experiments based on the stiffness values of the HTS (k_{ax} = 2 kN/mm, k_{rad} = 1 kN/mm) and PM bearing (k_{ax} = -1.3 kN/mm, k_{rad} = 0.7 kN/mm) showed that the original rotor design would cause critical rotor vibrations in millimetre range. Therefore, we decided to re-design the total rotor. In a process of calculation, measurements and rotor re- work an optimum design with all the rotor masses were found. The amplitudes of the rotor shaft caused by possible unbalances and small tilting of the main axis of inertia could be limit to less than the critical amplitude of

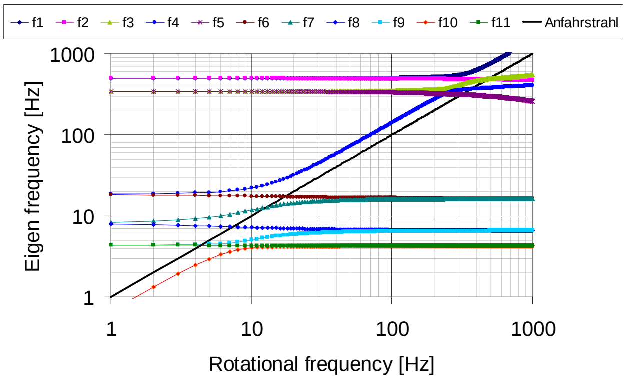

Fig. 5. Calculated diagram of flywheel rotor Eigen frequencies

0.5 mm. Due to the new construction the rotor Eigen frequency (rigid-body critical frequency) could be shifted. In the consequence, the rotor mass increases by 100 kg to more than 600 kg.

The spin whirl map diagram in Fig. 5 shows the critical speed of different modes against the rotational speed. Clearly, the low critical rpm's are caused by the magnetic bearing support. After re-designing the rotor geometry the frequency map is calculated using an FEM rotor dynamic program tool. Now because of the magnetic bearing support, interferences occur at frequencies below 20 Hz (~ 1000 rpm) which is the rigid body critical frequency of the rotor. With respect to this the flywheel speed is operating over critical.

The low frequency amplitudes are damped by dynamical emergency bearing devices located at both ends of the rotor shaft. Possible other excitations are located above 250 Hz which is higher than the operating range of the flywheel (0-10 000 rpm).

F. Power electric unit

In the industrial test the HTS FESS is dedicated to support a 220 V safety line in an E.ON power plant in Ingolstadt. The electric electronic part consist of a sensor system controlling various temperatures (HTS bearing, M/G, housing), vacuum, rotor position, and vibrations. The power electric system is connected with electric feeding, the control engineering and power regulation. Typically the FESS is operating with a DC voltage of 500 – 750 V. For safety reasons fast power switches can separate the power electric system from the grid. Electric key element is a power management controller (PMC) connected with a microprocessor interface.

Fig. 6. Power distribution unit of the 5 kWh HTS FESS

In the UPS function the FESS controls the power of a 220 V DC safety line. In case of a grid power fail the flywheel can release the energy within a few milliseconds stabilizing the grid.

III. Scaling the Flywheel Energy Storage Capacity

A. Energy losses

Clearly, the first real HTS flywheel will not be the most efficient one. Basically, energy loss of a conventional flywheel generally holds for 1/3 Generator/ Motor G/M, 1/3 air friction, and 1/3 mechanical bearing. It is expected that the losses of the present HTS flywheel are substantially reduced: The 1/3 G/M electric losses during operation will be almost zero. The air friction is strongly reduced due to improved vacuum conditions of about 10^{-3} mbar and the direct HTS bearing loss is a few Watt. The present GM cooler operates rather inefficiently (COP~2%). Future machine cooler of the Stirling – type will improve the cold power production by a factor (theoretically 10% efficiency). In addition, HTS flywheel losses increase less powerful with scaling up the storage capacity.

B. Scaling of storage capacity

To get a higher storage capability one has two basic scaling factors: The size and mass of the rotor and the speed. The present MDS rotor has a geometrical shape of a hollow cylinder which gives two fundamental benefits. The volume -saving concentrically motor /generator design inside the hollow rotor structure favours simultaneously the more efficient storable energy per rotor mass. Fig. 7 demonstrates the calculated achievable energy storage as a function of the inner to outer rotor diameter ratio r_i /r_a. 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_a 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.

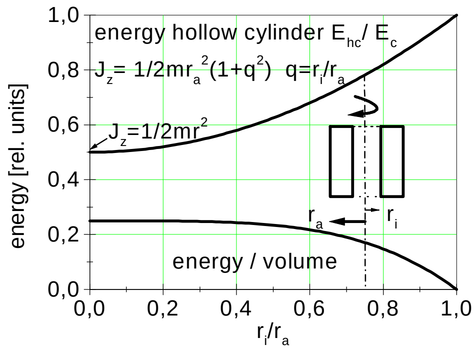

Fig. 7. Geometry and energy factors of a rotating hollow cylinder

The here provided concept of the rotor stabilization allows substantial larger energy storage of 15 – 20 kWh. A larger HTS flywheel is already designed and dedicated to operate in an electric utility in Korea. Thereby, the simultaneous scaling of the electric power to 400 kW meets requirements of industrial use.

IV. Conclusion

A 5 kWh / 250 kW HTS flywheel is tested and assembled in all components. Calculations and component tests showed the necessity to incorporate two additional dynamical emergency bearings and a stiffer rotor design to stabilize the rotor dynamic in passing critical rpm's. With the flywheel improvements larger energy storage at greater speeds can be achieved.

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