A 500-Wh POWER FLYWHEEL ON PERMANENT MAGNET BEARINGS

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SUMMARY

A 500-Wh disk-and-shaft power flywheel test prototype for possible application in uninterruptible power supply (UPS) systems is described. The system was set up in order to evaluate the stability of a passive radial permanent-magnet bearing system under static and dynamic power load of the integrated motor generator. Preliminary tests have been run under up to 15-kW transient discharge power load with no significant effect on the bearing stability.

INTRODUCTION

Flywheels have been in use for energy storage since the early history of mankind. The potter's wheel is a standard example for discontinuous-in, continuous-out energy conversion. Early steam engines as well as modern internal-combustion, piston-type engines use flywheels in a similar manner.

High-speed flywheels on the basis of carbon-fibre reinforced plastics material (CFRP) and magnetic suspensions have been developed for energy storage [1-3], in particular, for possible replacement of electro-chemical batteries in UPS systems, electric car drives and space applications. One specific advantage of a flywheel storage is that its total energy content can be discharged within less than a minute without injury to the system. Chemical batteries would not survive under similar conditions. When running on non-contacting, preferably magnetic bearings,a flywheel provides freedom of bearing friction and virtually unlimited life time.

For optimum reliability and power saving,*permanent-magnet* bearings (PMB) have been the preferred choice in space flight applications [4,5] as these are operable at *"zero-power"* stabilization conditions [6,7]. PMB technology has also been introduced to and commercialized in a variety of earth-born applications during the last two decades on the basis of some special bearing concepts developed at Forschungszentrum Jülich [8]. One of these [9] has been implemented in the flywheel system described in the present paper. The system was designed in the early 90's as a basic study for possible commercial applications in UPS and electric car drives.

A first unit was realized and tested [10] in order to investigate the stability of a passive radial PMB system under dynamic load conditions of the integrated power generator. We regard the outcome of this investigation as a milestone on the way of introducing the PMB technology into the field of heavy machine construction.

TECHNICAL DESCRIPTION

The flywheel system test prototype is shown in figure 1. The functional components are displayed in the sectional view of figure 2. The complete rotor system is encapsuled in a vacuum-tight aluminum housing that also serves as a safety containment. The system is designed for operation at a fixed vertical axis.

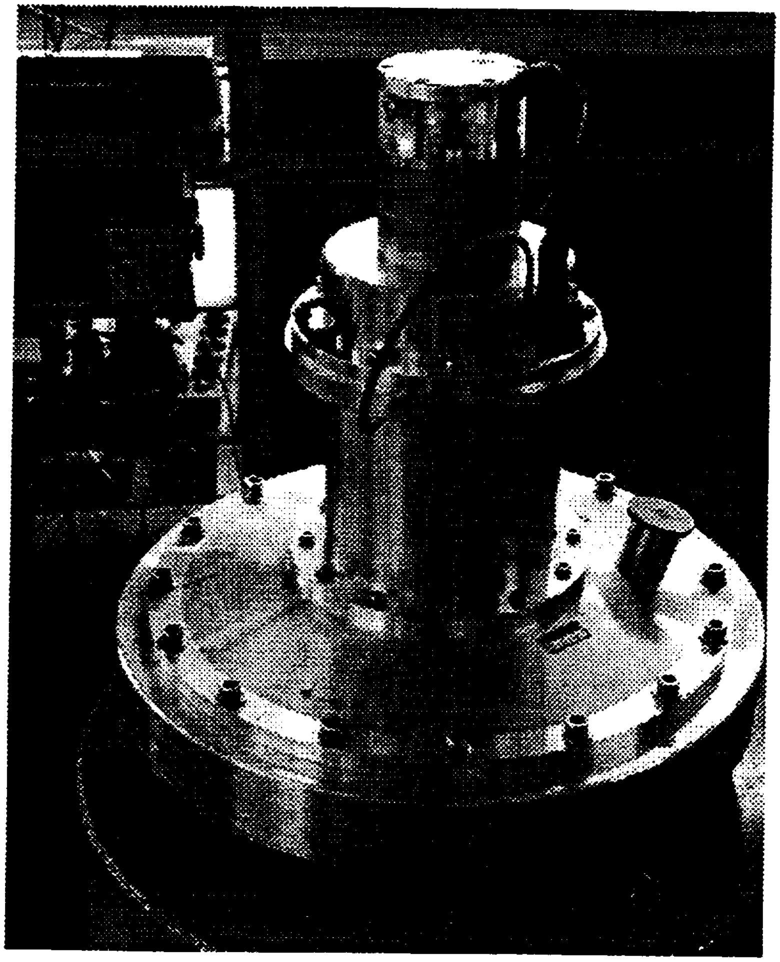

Figure 1. Flywheel system

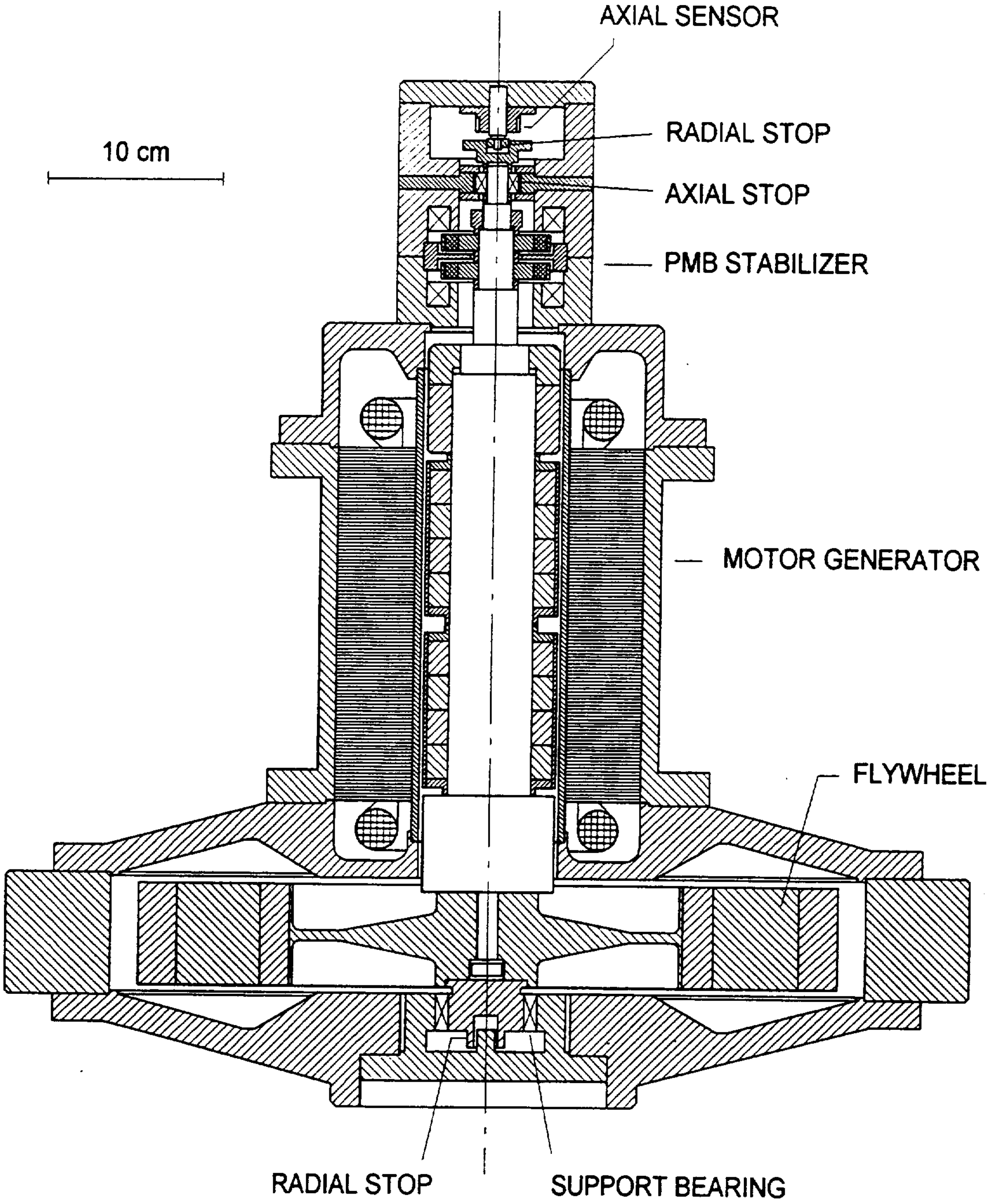

Figure 2. Sectional view of the flywheel system

The flywheel itself consists of three press fit annular components. The outermost ring is made of high-modulus M46J CFRP, the next one is standard T300 CFRP and the inner one is aluminum. An aluminum hub with elastic collars provides tight mechanical coupling between the ring package and the driving shaft where the hub is fixed by a central high-strength bolt.

The shaft carries two sets of permanent magnets made of plastic bound NdFeB material with diametrical magnetization. The magnets are part of a motor generator with a nominal power of 50 kW at 40,000 rpm of rotor speed. The annular magnets fit into centrifugal belts made of T300 CFRP. The magnet packages are fixed to the shaft by an axial thrust nut. The motor generator stator is mounted outside the vacuum enclosure. The latter passes through the magnetic gap of the motor generator.

The PMB system comprises a support bearing located just below the flywheel and a stabilizer assembly located on top of the motor generator.

The support bearing is shown in more detail in figure 3. This bearing contains four double-ring permanent magnets with attractive polarization [11] giving the rotor a net upward pull thus balancing its total mass of 23 kg. The mass of the permanent magnet material contained in the support bearing amounts to 184 g which is 0.8 % of the rotor mass.

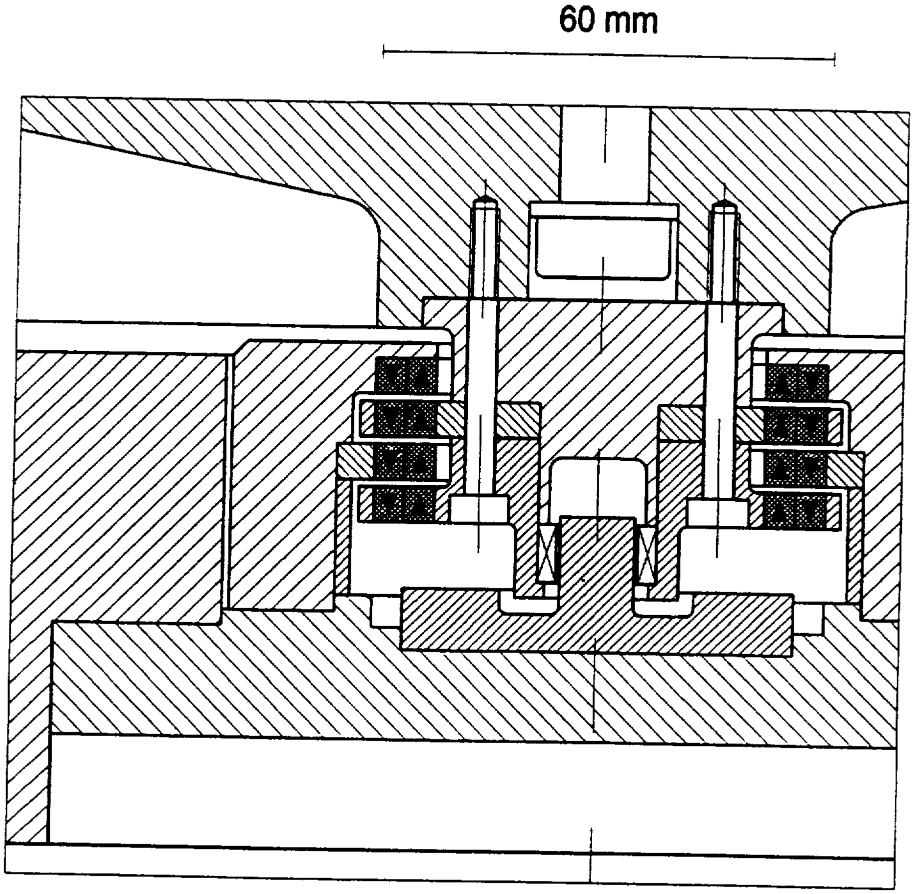

Figure 3. Close-up view of support bearing and lower radial stop assembly

Each double ring is made of two rings with opposite polarity. The forces between the magnets are attributed to surface currents as visualized in figure 4. These currents are built up from elementary currents associated with the electron spin at an atomic scale. All elementary currents within the magnet material cancel, as schematically indicated by loops at the axial surfaces, except those at the cylindrical surfaces where counterrotating neighbours are missing. The magnitude of the surface current of a given magnet equals the coercive force of the magnet material hence can be read from the corresponding data sheet. The NdFeB material presently used provides a coercive force of about 850 kA/m. With a magnet height of 4.7 mm we have about 4 kA of surface current on each of the cylindrical surfaces. The resulting currents are surprisingly high and could never be established in electric conductors of similar size. By fitting the double rings with opposite magnetization the surface currents sum up to 8 kA at their contact surface instead of cancelling each other. As the magnetic pull between parallel currents grows according to the product of their magnitudes, the 8-kA contact surfaces of the double rings provide four times as much pull across the axial gaps of the support bearing as the 4-kA bare surfaces on the in- and outsides. Thus, the four double-rings magnets provide 50% more pull than eight single rings of similar volume.

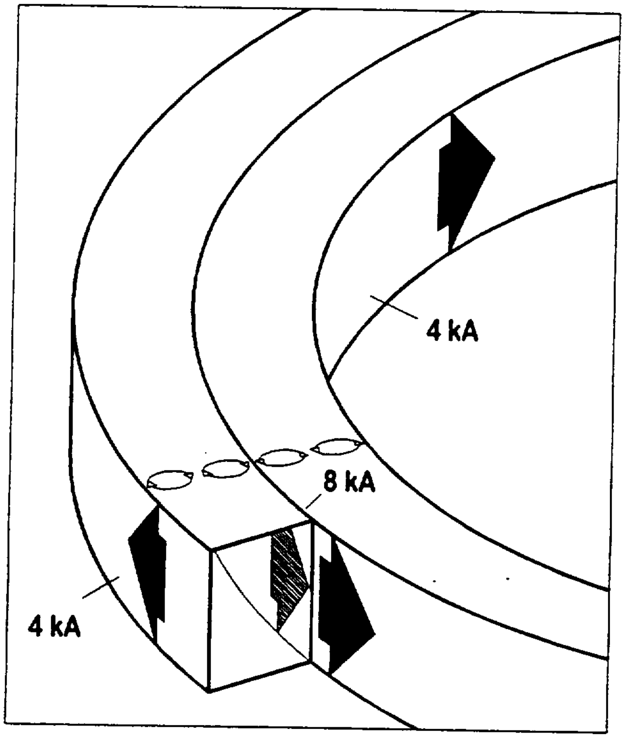

Figure 4. Double-ring permanent magnet with associated surface currents

The total surface current contained in a given PMB system may be regarded as a key parameter for characterization of its load capacity. The support bearing described here carries about 64 kA of surface current. This remarkably high current and the corresponding forces are steadily available without energy loss and without any obvious time limitation. For this reason it is proposed to attribute a permanent magnet the quality of a superconductor, not even a "high-temperature" superconductor but rather an "ambient-" or "room-temperature" superconductor. In fact, the practicable temperature range of permanent magnets extends up to several hundred degrees C for certain magnet materials. At higher temperatures the magnetization weakens due to thermally induced disorder at an atomic scale. The demagnetization is enhanced in PMB applications based on a repulsive magnet configuration.

In the present attractive PMB configuration the currents not only provide the axial mass support of the rotor but also tend to align the rotor and stator magnets in the radial directions thus giving the rotor a radial stability. The radial stiffness is about 220 N/mm.

Along the axial direction, the currents tend to pull the gaps tight. The system is intrinsically unstable along the axial direction. The instability is specified by a negative stiffness which for geometrical reasons is about twice the magnitude of the radial stiffness. Axial stabilization is achieved by the stabilizer located at the upper end of the driving shaft (see figure 2). This type of stabilizer was developed for both, axial and radial stabilization functions [12]. The stabilizer is contained in an iron case. Two electric coils fixed inside the stabilizer case and a pair of permanent magnets on the rotor enable axial force actuation of the rotor. The coils are fed from a bidirectional DC power source (not shown) which is controlled by an axial position sensor located opposite to the upper end face of the driving shaft. The sensing circuit is automatically adjusted such as to drive the rotor to the axial position where the magnetostatic and gravitational forces acting on the rotor cancel. At this point the coil current approaches zero. Axial stabilization of the system is achieved at *"zero-current"* conditions. In fact, a small stand-by current (typically on the order of one watt of electric power) is required to keep the electronics alive for immediate action in case of axial force disturbances of the rotor. The power capability of the axial actuator system is adequately dimensioned to supply the lift-off force required on system start for pulling the rotor out of its axial rest position.

The stabilizer further provides radial centering and damping forces for the rotor. Centering is achieved by annular extensions of the iron case which penetrate the axial coils ending up at a close distance in front of the rotating magnets. As with the support bearing radial stability is provided by attraction forces. The radial stiffness of the present system at the stabilizer site is about 20 N/mm. Radial damping is achieved by eddy currents induced by radial rotor vibration in a copper disk extending into the gap between the rotating magnets. The damping force is about 27 Ns/m.

The weight support and attitude control functions are provided by two separate assemblies (support bearing and stabilizer) with quite different features at two distant locations. The system stabilization mechanism is thus comparable to the typical wing-and-tail-plane air craft design concept [9].

Axial and radial excursions of the rotor assembly are limited by mechanical stops (often termed touch-down or catcher bearings). The bidirectional axial stop is located on top of the stabilizer. It consists of a double ball bearing fixed to the shaft and a pair of axial disks on the stator giving the rotor a free axial play of ± 0.2 mm with respect to the axial equilibrium position. Radial stops on either end of the rotating system are designed for tangentially contacting the rotor at an inner cylindrical surface. In case of mechanical contact due to backward conical motion (precession) the tangential friction associated with the mechanical contact brings the rotor axis back to the central position [13,10]. The upper radial stop consists of a ball bearing fixed to the housing that fits into an axial bore at the rotor with a radial play of ± 0.2 mm. The lower radial stop is a simple pivot-and-sleeve type friction bearing with similar radial play.

RESULTS AND DISCUSSION

The present flywheel system was designed as a test prototype for operation up to 40,000 rpm of rotor speed. The preliminary test program, however, was stopped in view of safety aspects at 33,000 rpm when we observed a sudden alteration of the rotor balance. At this maximum test speed the energy content of the wheel was about 430 Wh.

The main goal of the investigation has been achieved by demonstrating that a *"passive"* radial PMB system is capable of matching the forces associated with a power motor generator under static as well as under dynamic load conditions. The maximum pulsed power extracted from the system before the test stop was 15 kW at 30,000 rpm of rotor speed. At this and all preceding points of observation the rotor excursions excited by pulse power load were very small compared to the free play of the rotor shaft within the limits given by the mechanical stops.

The discharge power was drained into a power resistor network. A discharge efficiency of 97.4% was measured during 15-kW pulsed power extraction at 30,000 rpm.

System testing was originally planned for a maximum power load of 50 kW at 40,000 rpm. Under these target conditions the current and associated force load on the rotor would have increased by a factor of 2.5 with respect to the maximum realized load. We were able to extract currents of this magnitude by short circuiting the motor generator leads by a high current switch. This was done at 4,200 rpm maximum in view of the switch ratings. By this action excitation of the lowest precessional motion of the rotor system became significant. The radial amplitudes, however, were still well within the free radial play of the rotor shaft.

The design of the present flywheel system has been made in view of application in an UPS system. In order to ensure undelayed power supply from the system the motor generator was equipped with a permanent-magnet rotor. Idling losses have been recorded up to the maximum tested speed of 33,000 rpm where the losses reached about 200 W. The almost linear frequency dependence identifies the losses as being mainly of hysteresis type. Extrapolating to 40,000 rpm we expect about 250 W of idling losses which is 0.5% of the nominal 50-kW output power.

Evaluation of the magnetic bearing related losses is difficult as these are very small compared to the motor losses. An upper limit of the bearing losses was determined after replacing the motor permanent magnets with a steel tube and running the motor in an asynchronous mode. About 0.2 W of idling losses were determined at 4,200 rpm in this configuration. In an earlier investigation of a 12-kg neutron beam chopper suspended by a similar bearing system but with the rotation axis horizontal [8] we determined about 3 W of idling losses at 20,000 rpm under high-vacuum conditions (no air friction). We further found a linear increase of the idling losses with rotor speed. This implies the predominance of hysteresis losses. By linear extrapolation of our present results we expect about 2 W of bearing related losses at the nominal flywheel speed of 40,000 rpm. This figure is in agreement also with theoretical considerations [10].

After fixing the rotor balance failure that terminated the preliminary test phase the system is intended for power tests up to full speed. The system is also considered as a basic test unit in view of day-and-night cycle energy storage application with the present motor generator replaced by a 5-kW machine and up to three more stacked flywheel assemblies on the rotor. Even on maintaining the present motor generator design with a permanent-magnet rotor, the self-discharge time constant of that energy storage system would be several days. A non-iron or externally excited type of motor generator would contribute to a further reduction of the idling losses with a corresponding further increase of the run-down time constant.

By modification and investigation of the present system towards more compact dimensions and higher speeds we expect to extend our knowhow also in view of possible car-drive and other mobile applications.

ACKNOWLEDGEMENT

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