DESIGN OF A MAGNETICALLY SUSPENDED FLYWHEEL ENERGY STORAGE DEVICE

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ABSTRACT

Magnetic bearings are ideally suited for kinetic energy storage devices due to their low frictional losses and their long expected lifetime. In order to minimize the aerodynamic drag of the energy storage system the flywheel is placed in a vacuum housing. Magnetic bearings operate without physical contact and require no lubricants. Therefore they are an ideal suspension for this vacuum–application.
In this paper a design carried out at the ETH Zurich of a kinetic energy storage device with a useable 1 kWh of energy is described. The design of the rotor was strongly influenced by a new concept in the design of electrical machines. The motor consists of an inner and an outer rotor. In order to optimize the electrical machine the rotor diameter had to be maximized. This leads to high stress in the rotor. Therefore a detailed Finite Element analysis of the rotor was necessary.

INTRODUCTION

During the last decade, developments in the field of composite materials and magnetic bearings made it possible to build kinetic energy storage devices more efficient (i.e. smaller, higher rotational speed and less weight). There are two main applications for energy storage systems. The first is to store energy for a long time. Here the time between loading and unloading energy is long and the goal has to be minimum losses during standby. Typical applications are uninterruptible power supplies. The second type of energy storage applications are for short time energy storage. The time between loading and unloading is small und the goal is to minimize the losses during loading/unloading and less during standby.
In applications with extreme peaks in power consumption, like seam-welding machines or island-type electrical networks, a short time energy storage device can level these peaks and additionally recover energy from the application.
We found a demand for a system with a capacity of a useable 1 kWh of energy and high power (250 kW) of the motor/generator. This leads to a short time for loading/unloading of 15 seconds.
Compared with kinetic energy storage devices, static energy storage devices like batteries or capacitors have limited cycles lifetime and low power, respectively low capacity.
For this reason a research project 'Kinetic Energy Storage (KIS)' was startet at the ETH two years ago. The goal was to develop a kinetic short time energy storage system for stationary applications within the Mechatronics Group. The project partners of the ICMB are the Institute of Electrical Machines and the Chair of Power Electronic and Electrometrology.

The system consists of the following key components.

- a composite fibre–reinforced high–speed flywheel to store the energy
- magnetic bearings to suspend the rotor
- a high speed motor/generator to provide power to and from the flywheel
- high efficiency power electronics for the motor/generator
- vacuum housing to reduce the losses due to friction

The requirements of high power combined with low losses leads to new concepts for the electrical machine and the power electronics.

SURVEY OF THE KIS SYSTEM

TABLE 1: Data of the KIS system

diameter of the housing	: 870 mm
height of the housing	: 810 mm
diameter of the flywheel	: 680 mm
length of the rotor	: 365 mm
total mass of the rotor	: 175 kg
nominal speed	: 15000 rpm
no–load speed	: 7500 rpm
test speed	: 17250 rpm
usable energy	: 1 kWh
nominal power	: 250 kW

A cross section of the energy storage device (without housing) is shown in figure 1. The vertical rotor axis allows to separate the forces due to disturbance/unbalance from the gravity load and to use only one axial magnetic bearing on the top of the rotor.

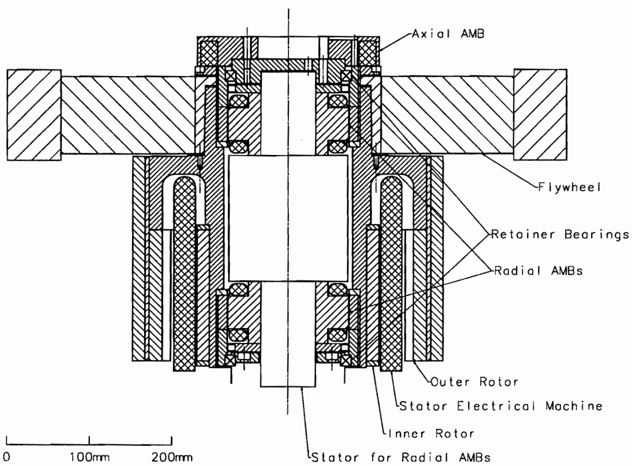

FIGURE 1: Cross section of the kinetic energy storage system [1]

FLYWHEEL DESIGN

The highest circumferential velocity of the flywheel in this application is about 600 m/s. When failing the usually used isotropic materials break into few bigger parts, with the disadvantage, that only little energy will be dissipated in breaking energy. The energy stored in the flywheel can cause damage to the whole system. The flywheel for this project was wound with orthotropic composite materials. It is designed so, that a crack will grow in tangential direction. This maximizes the breaking energy and leads to many small pieces and a much better failure behavior.

[1] AMB: Active Magnetic Bearing

A new method to consider all manufacturing related infuences for the stress calculation of the composite flywheel had to be found. The most significant influences beside rotation are prestressing of the fibres, shrinkage due to the curing of the resin and temperature strains.

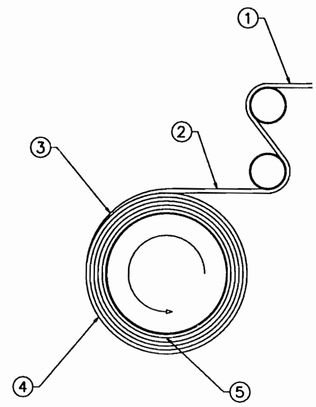

FIGURE 2: Flywheel manufacturing process

Figure 2 describes the winding process of the flywheel. With this procedure developped at the ETH it is possible to bring internal stress into the flywheel. At phase 1 the band is without stress. At phase 2 the band gets prestressed. At phase 3 the band is laid down and the prestressing leads to a radial pressure. At phase 4 the hardening and temperature cycle begins. The strain of the matrix remains plastic. At phase 5 the matrix begins to get hard, but the hardening and temperature cycle is not finished. The strain of the matrix gets elastic.

The manufacturing process of the flywheel will minimize stress at nominal speed and achieve a good failure mode.

MAGNETIC BEARINGS

It is necessary to place the system in a vacuum housing in order to minimize aerodynamic drag.
Magnetic Bearings operate without physical contact and require no lubricants. Therefore they are ideally suited for vacuum–applications.

The system is suspended by one axial and two radial current–controlled AMBs. The axial bearing is on the top of the system and carries the weight of the rotor. The operating point (static force) is the weight of the rotor. The disturbing forces for the axial bearing are small. The radial bearings are inside of the (inner) rotor. They don't have to hold static forces, but the radial disturbing forces, mainly due to unbalance.

TABLE 2: Data of the magnetic bearings

diameter of radial bearings	: 148 mm
maximum current	: 8 A
maximum voltage	: 120 V
nominal air gap	: 0.6 mm
maximum force (axial)	: 2000 N
maximum force (radial)	: 400 N

ROTOR DESIGN

The mechanical design of the rotor was strongly influenced from the concept of the electrical machine. It is a synchronous motor with lamination sheets on the inner rotor and permanent magnets mounted on lamination sheets on the outer rotor.

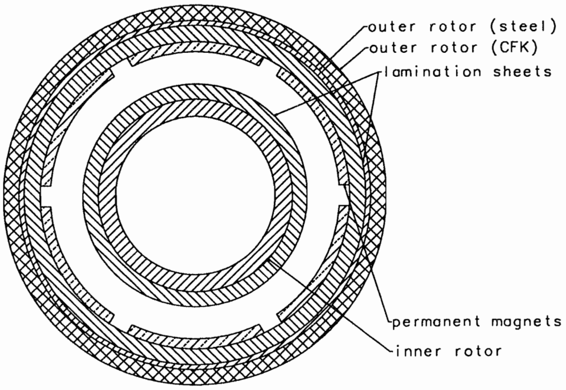

FIGURE 3: Cross section of the rotating part of the electrical machine [2]

The electrical machine was optimized to reduce the energy losses in the rotating part during loading/unloading and standby. The permanent magnets and the lamination sheets of the outer rotor are segmented and cause high stress due to centrifugal load in the outer rotor. The data of the electrical machine is shown in the following table.

TABLE 3: Data of the electrical machine

number of pole pairs	: 3
number of phases	: 3
nominal power	: 250 kW
rated phase voltage	: 900 V
rated phase current	: 100 A

In order reduce the energy losses of the rotating part and to minimize the unilateral magnetic pull it is necessary to maximize the rotor diameter and minimize the motor length. Therefore it was necessary to design the inner and the outer rotor to nearly reach the material strength of the soft magnetic iron sheets respectively the rotor.

[2] CFK : carbon fibre

The stress calculation due to centrifugal load for concentric rings can be done with continua mechanics. This allows twodimensional analysis for centrifugal loads and shrink fit and gives good results for most applications.

The equations for the radial and tangetial stress and the radial strain for one ring (index i) are given below. The constants a_i and b_i can be calculated with the boundary conditions (e.g. the radial stress of a free face is 0, the radial stress and the radial strain of two rings at a contact face are equal). This leads to a linear equation system for a_i and b_i.

σ_r : radial stress
σ_φ : tangential stress
u_r : radial strain
E : Young's modulus
ν : Poisson's ratio
ρ : mass density
α : thermal expansion coefficient
θ : temperature
ω : rotational speed
r : radius
a, b : constants

$$\sigma_{r_i} = \frac{a_i\,E_i}{1-\nu_1} - \frac{b_i\,E_i}{(1+\nu_i)\,r^2} - \tfrac{1}{8}(3+\nu_i)\,\omega^2\,r^2\,\rho_i - \tfrac{1}{2}\alpha_i\,E_i\,\theta$$

$$\sigma_{\varphi_i} = \frac{a_i\,E_i}{1-\nu_i} + \frac{b_i\,E_i}{(1+\nu_i)\,r^2} - \tfrac{1}{8}(1+3\,\nu_i)\,\omega^2\,r^2\,\rho_i - \tfrac{1}{2}\alpha_i\,E_i\,\theta$$

$$u_{r_i} = \frac{b_i}{r} + a_i\,r - \frac{(1-{\nu_i}^2)\,\omega^2\,r^3\,\rho_i}{8\,E_i} + \alpha_i\,(1+\nu_i)\,r\,\theta$$

The circumferential velocity of the lamination sheets of the electrical machine at the inner rotor at test speed is 198 m/s. A shrink fit is necessary to guarantee contact (i.e. negative radial stress at the contact zone). This leads to a positive prestress in the lamination sheets. The prestress and the stress due to centrifugal load must not exceed the yield stress of the soft magnetic iron (400 MPa). The lamination sheets of the bearings are less critical, because the shrink fit causes negative prestress.

The stress in the lamination sheets of the outer rotor due to centrifugal load would be higher than the yield stress of the soft magnetic iron. Therefore a high prestress and a big shrinkage allowance would be necessary. For the KIS rotor this is not possible, because either the geometry ratio between the lamination sheets and the steel ring would give only insufficient prestress or using the retainer ring (steel and carbon fibre) the achievable temperature difference for the shrinking process would be to small. In this case the shrinkage is limited because carbon fibre materials have no (very little) thermal expansion. An unsufficient prestress would lead to cracking in the lamination sheets. Compared to cracked lamination sheets segmented lamination sheets have lower eddy–current losses of the electrical machine. In order to maximize the rotor diameter not only the permanent magnets but also the lamination sheets were segmented. This leads to high stress in the retainer ring.

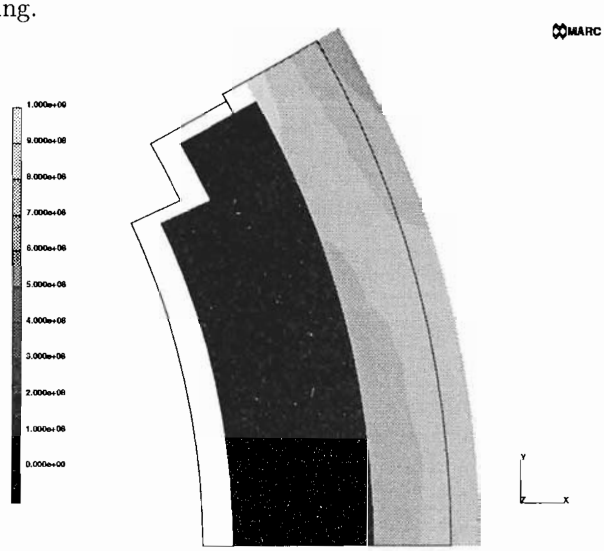

FIGURE 4: Equivalent stress (in Pa) and deformation of the outer rotor due to centrifugal load (rotational speed 250 Hz);
shown is 1/12 of the radial cross section (plain stress respectively plain strain analysis)

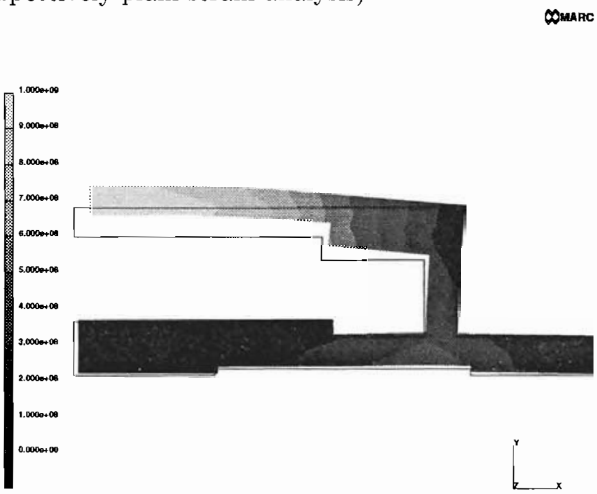

FIGURE 5: Equivalent stress (in Pa) and deformation of the outer rotor due to centrifugal load (rotational speed 250 Hz);
shown is the axial cross section (axisymmetric analysis)

The segmented permanent magnets and lamination sheets can't be simulated as rings but as an additional radial pressure at the inside of the steel ring. A Finite Element analysis of a cross section of the outer rotor gives more detailed results for stress and strain.

For the outer rotor a retainer ring with high material strength is necessary. In designing the retaining ring the aim must be to achieve an optimal relationship between strength and wall thickness. The operational forces include, in addition to the centrifugal forces acting on the lamination sheets, the centrifugal forces of the retaining rings themselves.

There are several possibilities for a retainer ring. The outer and the inner rotor could be one part. The main disadvantages are that it is more difficult to machine this workpiece and that a steel with the desired strength is not available.
Another possibility is to use retainer rings analog to turbogenerators. The retaining ring is fixed on the rotor body ends by shrink fit.

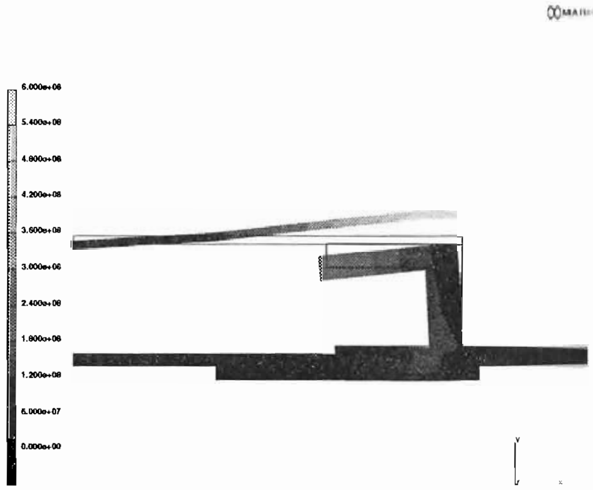

FIGURE 6: Equivalent stress (in Pa) and deformation after the shrinking of the steel ring (shrinkage allowance 3/1000)

The ring is solution heat treated and then work hardened so as to obtain the desired 0.2 % proof stress. The nitrogen alloyd austenitic steel used for retainer rings for turbogenerators is non–magnetizable, has a high fracture toughness and can attain a 0.2 % proof stress of more than 1500 N/mm^2 at a stretching rate of approxmately 50 %. Even a 0.2 % proof stresses of about 2000 N/mm^2 is possible. The problem is to work harden a ring with the diameter of the KIS rotor. Retainer rings for turbogenerators have diameters of more than 1 m up to 2 m. For these rings a special 6000 t–press is necessary. To work hard the ring for the KIS rotor would be possible but leads to an expensive solution. Progress and new solutions for nitrogen alloyd steels and the work hardening process will make it possible to use these kind of steels for applications like the KIS system. Carbon fibre rings can achieve high strength in circumferential direction. The axial strength and the maximal shear stress are limited to the material data of the matrix (epoxy resin). In the zone between the shrink fit area and the lamination sheets there is high shear stress due to the different centrifugal loads. Therefore a retainer ring consisting of an inner steel ring and an outer carbon fibre ring was chosen. For the retainer ring an alloyd treated steel is used. After hardening the desired strength can be obtained. This steel is better available, less expensive, but magnetizable and will therefore cause eddy current losses.

ROTORDYNAMICS

In order to design the controller for magnetic bearings it is necessary to have an accurate mathematical description of the mechanical plant. The equation of motion is given below.

M: mass matrix
D: damping matrix
G: gyroscopic matrix
K: stiffness matrix
h: outer forces
q: vector of coordinates
ω: rotational speed

$$M\ddot{q} + (D + \omega G)\dot{q} + Kq = h$$

The KIS rotor was modelled with madyn. This Finite Element program was made specially for rotordynamic problems. General purpose Finite Element programs only compute the mass and the stiffness matrices. For an energy storage system, which behaves like a gyroscope, it is essential to use a model with the gyroscopic matrix. The inner and the outer rotor were modelled with two sets of beam elements connected with a stiff element. This is a sufficient approximation, because modelling the connection with an element representing the material data of steel leads to comparable results. The lamination sheets and the flywheel are modelled mainly as additional masses and additional moments of inertia. The magnetic bearings are approximated as spring–damper–elements.

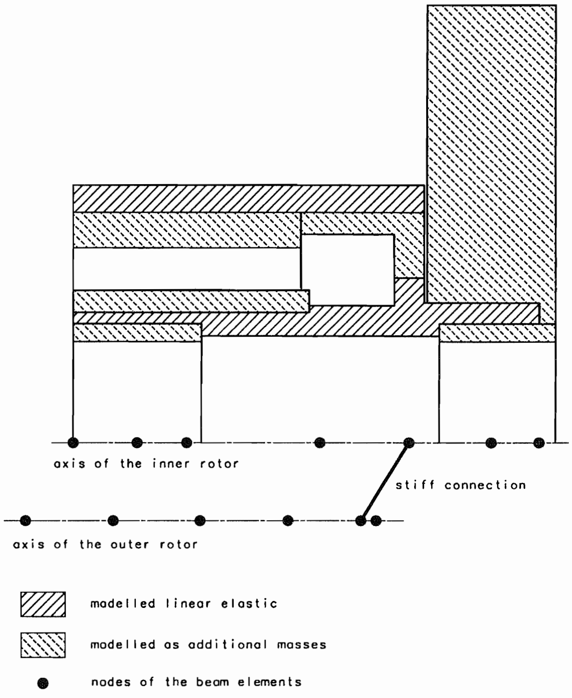

FIGURE 7: Model of the KIS rotor for rotordynamic analysis;
the axis of the inner and the outer rotor in the model are identical, but they are drawn separated for a better visualisation

Eigenfrequencies and mode shapes depending on the rotational speed were computed.

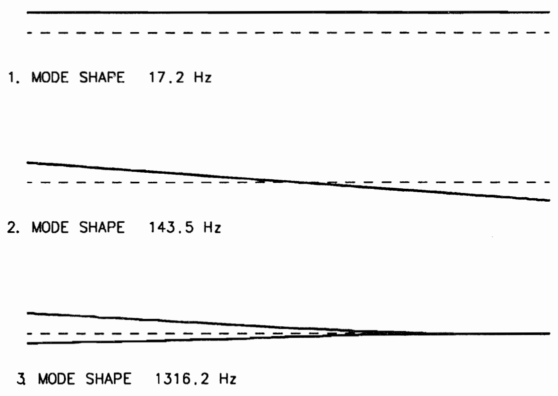

FIGURE 8: The first three mode shapes (synchronous rotation) of the KIS rotor (rotational speed 125 Hz)

Simulations with outer forces due to unbalance of the flywheel and the rotor were made. The displacement and the reaction forces of the bearing of the well balancend rotor are computed in order to dimension the AMB.

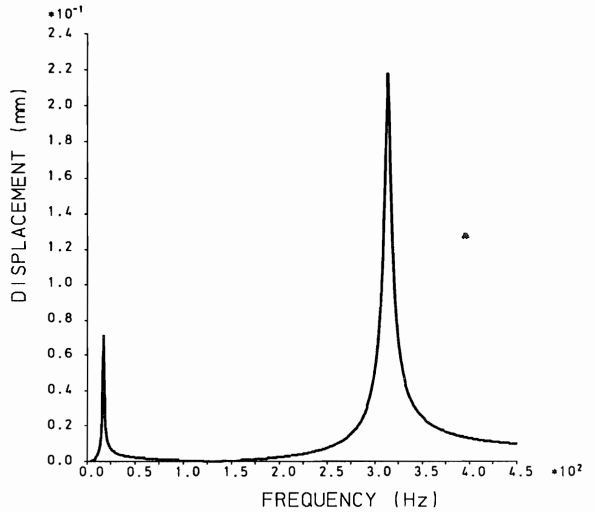

FIGURE 9 : Displacement of the rotor in a radial magnetic bearing due to unbalance versus the rotational speed of the rotor;
$Q = \varepsilon\omega = 6.3\,mms$, Q : unbalance quality, ε : excentricity, ω : rotational speed

SUMMARY

In order to optimize the electrical machine it was necessary to design the rotor to reach nearly the material strength of the lamination sheets and the rotor. For the design of the rotor several possibilities for the retainer ring were examined. The shrinking process and the stress due to centrifugal load were analyzed with continua mechanics and Finite Element methods.

OUTLOOK

The kinetic energy storage system will be realized and tested by autumn 1994. Afterwards the system will be slightly modified and used for seam-welding machines in industry. These tests will give experiences under realistic conditions and should lead from a prototype status to an industrial application.

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[10] VSG. *Retaining Rings.* VSG Vereinigte Schmiedewerke.