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Design and practical Realization of an innovative Flywheel Concept for industrial Applications

L. Quurck, M. Richter, M. Schneider, D. Franz, S. Rinderknecht

The joint industry project 'ETA-Fabrik' at TU Darmstadt demonstrates different approaches to improve the energy efficiency of manufacturing processes. Within this project an innovative flywheel concept was designed and realized in order to provide energy storage and load smoothing services. The flywheel design is an outer-rotor setup. The rotor is a hubless hollow cylinder made of fiber reinforced plastic (FRP). All functional components are fully integrated into the rotor. For the radial suspension homopolar active magnetic bearings (AMBs) made of soft magnetic composite are used. A permanent magnetic bearing provides axial levitation. In order to increase the systems robustness a newly developed backup bearing system in a planetary arrangement with multiple independent bearing elements is integrated. The motor generator unit is a permanent magnet synchronous machine which is connected to the factory gird via a frequency inverter. The system is operated in high vacuum in order to reduce gaseous friction. Design challenges are the segmented sensor planes for the AMBs, the diametric enlargement of the rotor due to centrifugal forces, the anisotropic FRP as well as the thermal stability of the rotor in vacuum environment which leads to the demand of very low rotor losses. The paper describes the system and component design process and solutions which were incorporated in order to meet the design restrictions and challenges.

1. Background

The demand for energy storage systems in industrial applications has risen over the last years. They can fulfill different tasks such as peak shaving or load smoothing to improve the energy efficiency and stability of the supply system as well as to save power dependent cost. The joint industry project 'ETA-Fabrik' at TU Darmstadt demonstrates different approaches to improve the energy efficiency of manufacturing processes. Within this project an innovative kinetic energy storage system (KESS) was designed and realized in order to provide the mentioned services. Figure 1 shows the layout of the system as a CAD model and the KESS set up in the 'ETA-Fabrik'. The design is based on the first realized outer rotor type flywheel which was introduced by Schaede (2013) as a proof of concept. This first outer rotor flywheel was brought into operation in 2012 in the laboratory of the 'Institut für Mechatronsche Systeme im Maschinenbau' (IMS) in Darmstadt.
Magnetically suspended flywheels provide reliable high power density with maintenance- and wear-free operation and without calendric and cyclic degradation. High energy conversion efficiency compensates the occurring standby losses particularity when dynamic load profiles with frequent high power demands have to be smoothed or cut. Figure 2 shows a simulation of a representative industrial load profile with and without the used flywheel as well as the corresponding power and state of charge of the flywheel. The used flywheel is the 1.4 kWh, 60 kW flywheel ETA290 which will be described in this paper. Another application could be the reduction of diesel consumption in island grids with high shares of renewable energy production (Schaede et al. 2015).

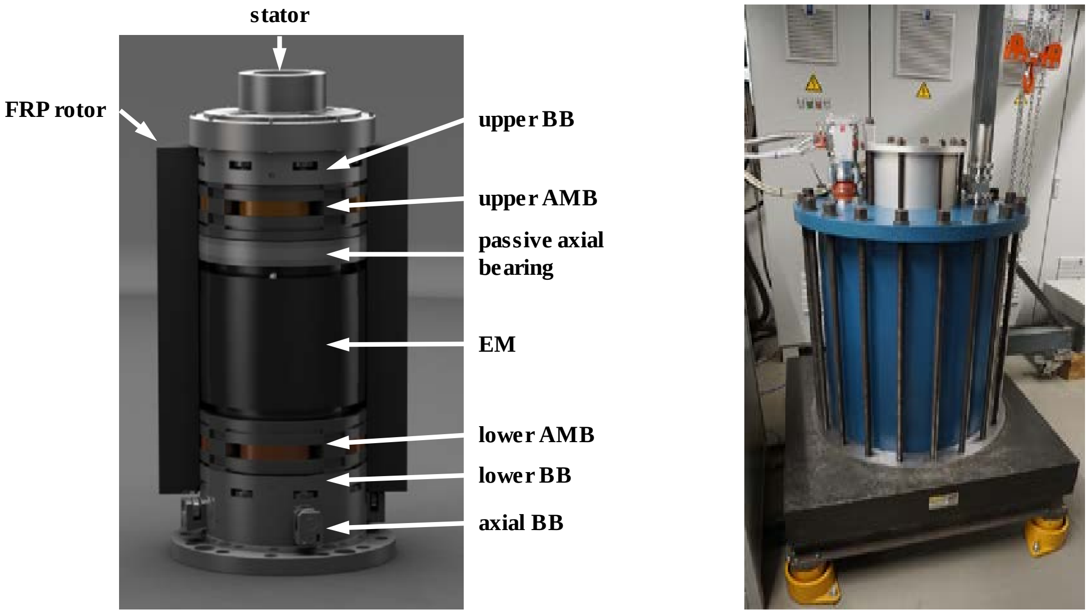

Figure 1. Left: CAD Cut-away model of the rotor mounted on the stator. Right: Photograph of the KESS in the factories utility compartment.

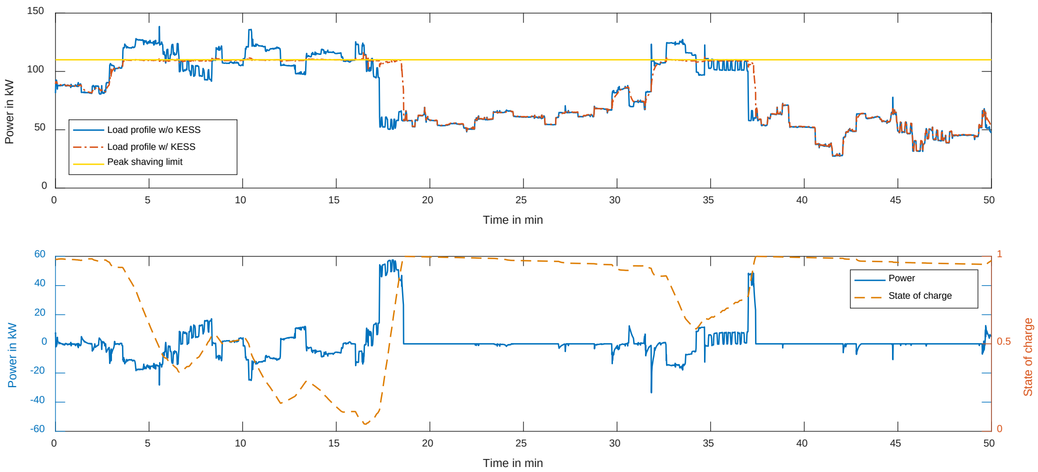

Figure 2. Simulation results of the peak shaving operation in the factory (top: factory load profiles, bottom: storage power and state of charge)

2. Design Goals and Design Process

In order to be able to design a suitable energy storage system for a specific application the individual requirements have to be analyzed. For load smoothing and peak shaving applications in an industrial environment the storage system needs medium to high electric power and moderate energy. As can be seen in Figure 2, more than 1 cycle per hour can be expected. Therefore, modern flywheel systems with low conversion losses, low self-discharge losses and a high cycle life can be an interesting alternative to other technologies. General design goals are minimal maintenance and wear even during continuous service, leading to the requirement of thermal stability. In order to be able to adapt to different industrial processes the system should be independently scalable in terms of output power and stored energy. The design of the flywheel should also allow for an efficient serial manufacturing. Due to the industrial environment liquid cooling and a connection to the 400 V AC grid can be assumed.
The outer-rotor design has potentially high energy densities because every active rotor component rotates with the maximum radius leading to a highly integrated system. Due to the large inner rotor radius high electric output power is possible. A general design conflict lies in the simultaneous optimization of energy density and energetic

losses. High energy densities can be reached at high rotational speeds and large radii. For the electric machine and the magnetic bearings, high speeds lead to high remagnetization frequencies resulting in high hysteresis and eddy current losses. The optimization of energy density and energetic losses can also be motivated from an economic point of view. The energy density can be interpreted as an indicator for the material dependent initial costs of the storage system. The losses that are generated during operation have to be compensated by electricity from the grid and can thus be interpreted as operating costs. An optimal system is characterized by the minimal sum of both dimensions over its lifetime.

In order to minimize the drag losses due to gaseous friction the system is operated at high vacuum levels ($<10^{-4}$ mbar). This operation condition in combination with the magnetic levitation of the rotor, however, leads to a poor thermal connection between the rotor and stator. Heat exchange can only be obtained by radiation. During the design of the magnetic bearings and the electric machine, losses on the rotor thus have to be minimized in order to prevent overheating and enable high energy efficiency.

From a methodologic point of view, a serial or independent design of the relevant system components (rotor, active magnetic bearings (AMB), electric drive (ED) and stator) is not promising due to the system's high degree of integration. Therefore, a simultaneous initial design process is carried out by Schneider (2014) using detailed component models in combination with optimization algorithms.

3. System Design

The following subsections describe the challenges within the design process of the main components. This includes rotor design, electric machine design, magnetic bearing design and back-up bearing (BB) design. The chapter concludes with an overview of the final parameters and the integration concept into the factory.

3.1. Rotor Design

The flywheel design is an outer-rotor setup. The rotor is a thick walled, hubless hollow cylinder made out of fiber reinforced plastic (FRP). The FRP rotor is produced in an industrial filament winding process with mainly circumferential fiber orientation to encounter the tangential stress. The state of stress can be assumed as plane due to the axially almost constant density distribution. This circumstance generally meets the orthotropic material strengths and reduces shear stresses in the composite. The maximum rotational speed, and therefore the kinetic energy content of the rotor, is restricted by the transverse strength in radial direction which is dominated by the resin matrix, while the utilization of longitudinal fiber strengths is comparably low. Another restriction is the diametric enlargement of the rotor by the centrifugal forces. At high speed, the diametric enlargement can be up to 1% of the inner diameter, what is the ultimate strain of common conventional FRPs. The ETA290 flywheel utilized about 0,5% of strain. The resulting significant enlargement of the air gap has to be taken into account at the design of the magnetic bearings, the electric machine and the backup bearings.

The allowable strain of most functional components is considerably lower than 0.5%, especially for the soft- and hard-magnetic components of the active and passive bearing. To avoid high stresses and failure in these components they are segmented circumferentially. This terminates the tangential hoop stresses under rotation. Further effects of this segmentation are discussed later in this paper. Despite the speed restriction due to maximum strain the ETA290 rotor reaches a specific energy density of 8.8 Wh/kg, while further design improvements will allow 14 to 20 Wh/kg by decreasing the weight of the functional components and simultaneously increase the maximum speed without changing the main diameters or mechanical strain.

The lack of a hub and an inner shaft simplifies rotordynamics and control. Due to the hollow and thick walled design the elastic eigenfrequencies are comparably high. Figure 3 shows the Campbell diagram of a 3D FE modal analysis of the rotor. The first three elastic eigenfrequencies are axially symmetric modeshapes with comparably low gyroscopic influence. The first bending eigenfrequency is above 1011 Hz. This simplifies the control of the AMB and gives the opportunity to enlarge the rotors length and adapt the energy content to the specific applications preferences without a significant influencing on the control design. For example, increasing the rotor length and thereby the kinetic energy content to 140%, lowers the first elastic eigenfrequency (1^{st} bending mode) to 502 Hz which is still two times the maximum rotational frequency. The rigid body eigenfrequencies are defined by the magnetic bearing properties. The passive magnetic bearing, which is stabilizing the axial degree of freedom, creates an 11 Hz eigenmode which does only change slightly due to thermal influences and due to diametric enlargement. Its negative stiffness in radial direction is compensated by the AMB. The four parallel and conical mode shapes are crossed below 50 Hz while design of the bearing and controller contains the rotor position within a 100 µm radius.

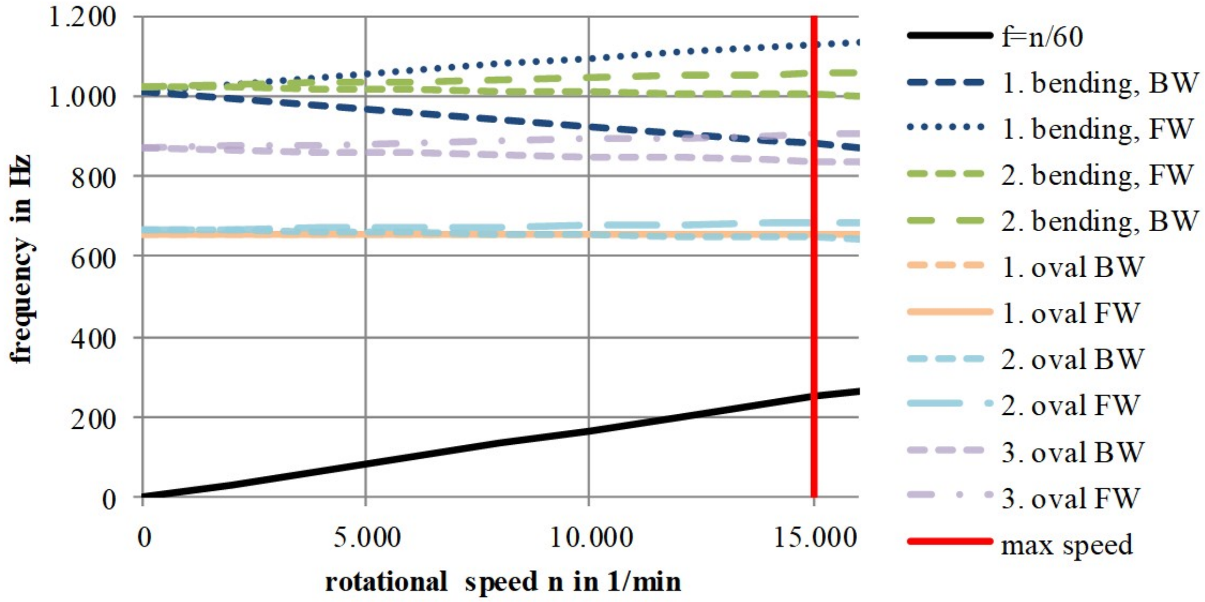

Figure 3. Campbell diagram of the 3D-FE-modalanalysys of the ETA290 flywheel rotor in free-free configuration

3.2. Electric Machine Design

In general, different electric machine types are suitable for flywheel systems. Due to the specific requirements from the field of application a permanent magnet synchronous machine (PMSM) is selected. As mentioned before, the rotor losses have to be minimized which excludes asynchronous machines due to their power dependent high copper losses. Synchronous reluctance machines are suited for high frequency applications due to the very low drag losses during idling. But due to the high power demand of the factory with many charging and discharging cycles, higher conversion efficiency is desired. Even though the PMSM creates drag losses when no power is requested from the flywheel it is the best option considering all factors of this industrial application.
As mentioned before, the all functional components have to be integrated into the rotor by segmentation considering the resulting stresses due to centrifugal forces. This circumstance leads to the requirement of a very thin PMSM rotor in radial direction in order to reduce the centrifugal forces applied to the FRP rotor. The goal can be achieved by increasing the number of pole pairs of the machine because the magnetic flux in the rotor in circumferential direction per pole is reduced. At the same time the active axial length of the machine decreases leading to shorter copper windings and thus lower copper losses. The EM rotor design uses permanent magnets which are embedded in electrical sheets. An ironless Halbach array would also be possible but it is much more expensive due to a higher magnetic volume and complex magnetization directions.
A high number of pole pairs, however, leads to high remagnetization frequencies and thus to high core losses. Typically, a high number of pole pairs can be found in so-called torque motors that run at low to medium frequencies. A high speed motor typically has a lower number of pole pairs and can thus be operated at higher frequencies. In this case this results in a design of the EM with four pole pairs and a rotor speed of 250 Hz. Therefore, the output frequency of the inverter has to be 1000 Hz which demands switching frequencies of at least 8 kHz. Furthermore, the control of the inverter operates without a rotor position signal because no commercial encoder system is available for outer-rotor designs with time variant air gap in high vacuum environment. The sensor-less drive mode results in reduced dynamics of the power control compared to encoder drive mode.
In order to design the EM in detail, a transient magnetic finite element analysis is carried out. The influence of the switching of the inverter is neglected, therefore electric currents in sine wave form are assumed. Figure 4 shows the exemplary results for a low speed operating point of 7500 rpm and a positive (charging) electric power of 60 kW. From this analysis the machine losses can be estimated using material-specific loss coefficients for the electrical sheets and the copper windings for different operating point (specified by electrical power and rotational speed).

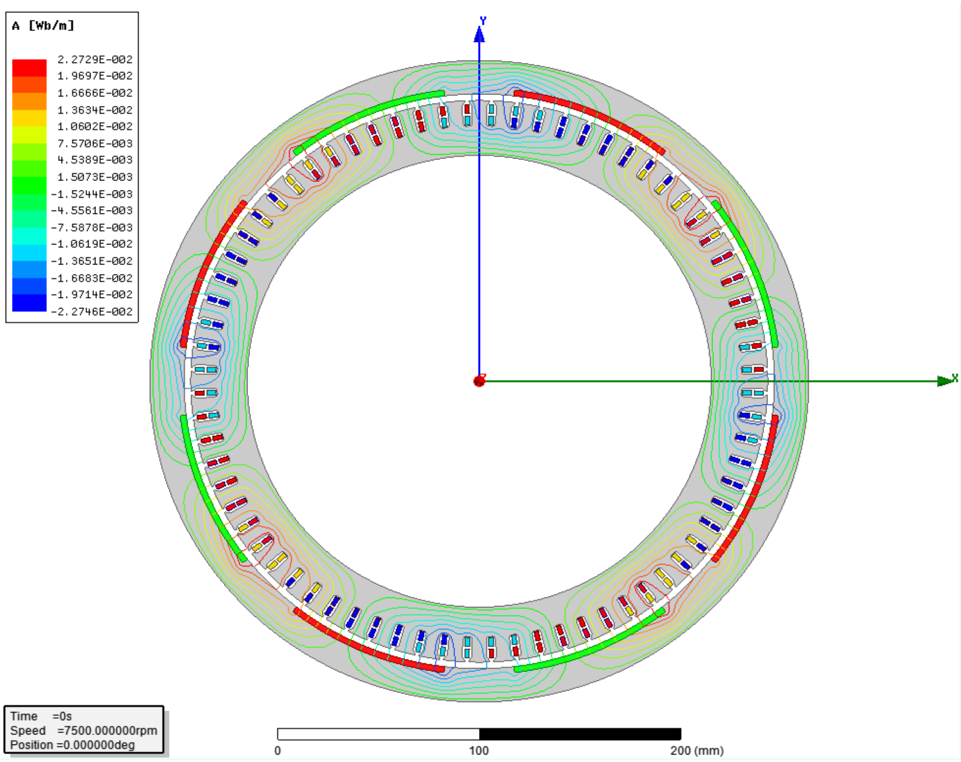

Figure 4. Results (2D magnetic flux) of the finite element simulation of the EM at 7500 rpm and +60 kW

By calculating the steady state losses, given as root mean square values, for 6 rotational speeds and 9 electrical powers, a loss map of the system can be defined. Figure 5 shows the loss map for the electric machine including rotor and stator losses. Due to the poor thermal coupling of rotor and stator, the rotor losses of all components have to be reduced. Rotor losses of the EM lie below 50 W. For the analysis sinusoidal currents and voltages are assumed. In a practical application pulse width modulated inverters are used that create current and voltage harmonics which again lead to core losses in the rotor. These losses can be reduced by installing inductors but have to be taken into account for practical applications.

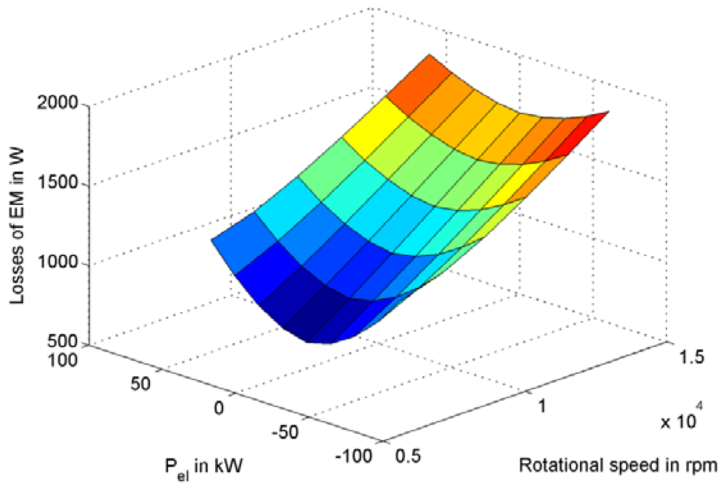

Figure 5: Loss map of the EM (rotor and stator losses) derived by a FE simulation

Analytical and numerical loss models of the AMBs and the gaseous friction losses of the rotor show that the flywheel's loss structure is mainly dominated by the losses of the EM. Losses of magnetic bearings and air friction losses lie in the range of 140-180 W and below 10 W respectively. A loss optimization should thus focus on an efficient EM design.

3.3. Magnetic Bearing Design

The goals during the design of the active magnet bearing system are stable levitation over the entire operation range with as low losses as possible especially with respect to rotor losses which lead to rotor heating. As a special design feature, the speed dependent diametric enlargement of the air gap has to be faced. In order to keep hysteresis and eddy current losses low, a homopolar AMB configuration is used. In Figure 6 the two axis radial AMB design, air gaps (magnified) magnetic flux path and the position sensors in differential arrangement are visualized.

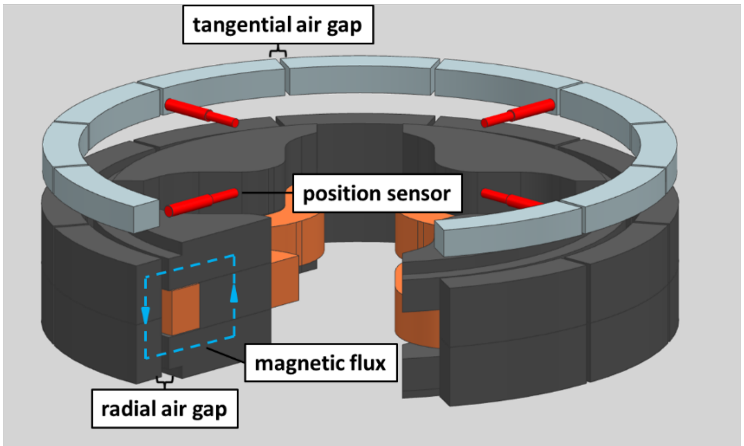

Figure 6: Homopolar AMB configuration with differential sensor configuration and corresponding air gaps

Despite a homopolar configuration a near constant, non changing pre-magnetization flux on the rotor is obtained keeping both losses low. Unfortunately, the homopolar configuration makes it hard to use thin electric sheet as the flux conducting material to further reduce eddy current losses. Thus soft magnetic composite is used as an alternative which allows isotropic 3-dimensional flux. As a result of the diametric enlargement of the rotor, the segmented sensor and actor planes of the rotor create tangential air gaps which increase with the rotor speed. When looking at the tangential air gaps one has to distinguish between those on the bearing side and those on the position sensor side. The first can be neglected due to the homopolar configuration as the tangential air gap does not lie in the path of the flux and thus does not further influence the behavior of the bearing. The latter is more crucial to the system. The tangential air gap has significant influence on the measured position as the air gap is being misinterpreted as a change in the rotor position whenever it passes the position sensor head. In Figure 7 exemplary position signals are shown. Looking at the single sensor signals (SP and SN) high peaks can be seen.

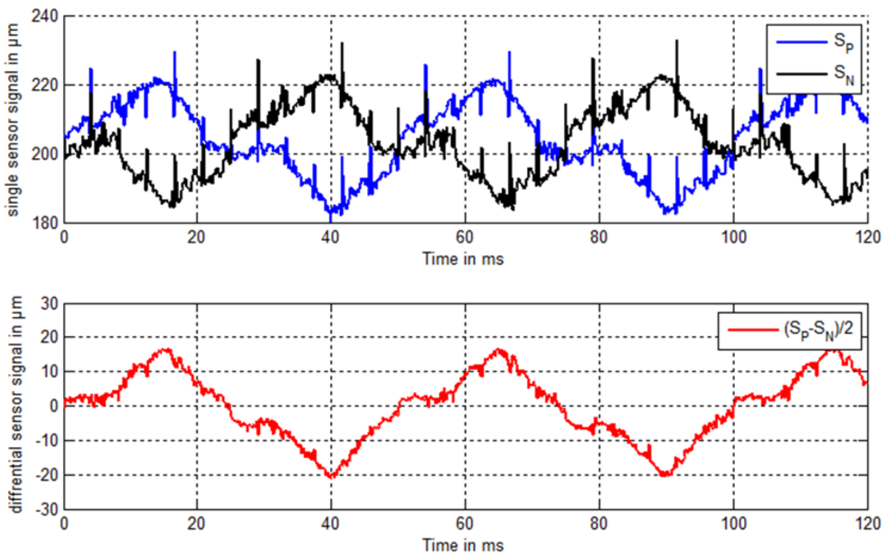

Figure 7: Single sensor signals (top) and differential sensor signal (bottom)

To overcome this undesired behavior, the mathematical difference (S_P-S_N) between two counter positioned sensors is used. Ensuring a symmetrical construction of the rotor segments, tangential gaps occur at the same time at both sensors and are thus not present in the difference of their signals (shown in Figure 7 at the bottom). This further holds for elastic axial symmetric eigenmodes of the rotor. Possible sensor errors like temperature drift and the speed dependent enlargement of the radial air gap between the sensor head and rotor are not present to this position signal anymore.
Unfortunately, the influences of the radial air gap enlargement on the actuator part of the magnet bearing cannot be circumvented in a same manner. With an increasing radial air gap the magnetic force drops by the power of two with the air gap, resulting in a highly nonlinear magnetic force over the rotational speed range. One way to overcome this undesired behavior from a control point of view is to increase the initial air gap. As the absolute change of the air gap remains the same, the relative change of the air gap as well as the linked nonlinear magnetic force characteristics decrease. The enlarged initial air gap on the other hand has the drawback of an increased magnetic bearing size due to the higher number of coil turns which is needed to overcome the additional air gap reluctance.
The rotor itself can be seen rigid from a control point of view since its first bending mode is more than 3.5 times higher than the highest rotational frequency, as shown in section 3.1. Due to its high moment of inertia the rigid body eigenfrequencies of the levitated suspended split up. Thus the highest rigid body eigenfrequency to be stabilized by the AMB are below 50 Hz and gives an upper limit of the desired frequency range of the AMB system. Despite these specific rotordynamic characteristics and the previously mentioned influences of the diametric enlargement of the rotor, a decentralized PID position control with a subordinated PI current control is sufficient to stabilize the system.

3.4. Backup Bearing Design

As most magnetically suspended rotors, the ETA290 flywheel uses rolling element bearings as backup bearings (BB). During regular operation the BB rests on the stator without mechanical contact to the rotor. When overload, malfunction or power loss of the AMB arise, the BB should prevent touching and damaging of other parts. Mechanical failures of the BB system can be classified as critical to the systems integrity because destructive whirl movements and very high bearing loads can occur as well as rapid momentum transfer from rotor to the stator when friction excessively rises. For high energy flywheels the preferences towards reliability and service times are notably challenging because of the kinetic energy density and the long spin down times even if high electric power braking is applied. For emergency braking during grid black out an additional passive brake resistor is added, spin down times are still above 2 minutes from full speed.
The size of inner diameter of the ETA290 flywheel together with the high rotational speed complicates the use of conventional bearing components. The bearing speed expressed in the DN number is higher than 4e6 mm/min, which classifies the bearing situation as high speed application. This is aggravated by the vacuum environment resulting in an unfavorable lubrication and a lack of cooling. Therefore, an alternative BB design was chosen. First introduced by Chen et al. (1997) and similarly published by Jansen et al. (2014) the basic idea of this new approach is to use multiple small bearing units circumferentially distributed around the stator in each BB plain. This planetary arrangement of independent bearings leads to better high speed capability, lower BB inertia and lower friction. Further information is given by Quurck et al. (2014). In Figure 8 an illustration of the ETA290 BB design is given as well as a photograph of the components during built up.
For the ETA290 flywheel, bearing units with full complement hybrid spindle bearings, mounted on a steel roller element are chosen. The roller element contacts the rotor's BB plain, while the outer bearing races are placed into the stator. In each of the two BB plains, eight bearing units form a polygon shaped free orbit in which the rotor can move contact-free.
Beside the higher speed capability of these bearing units, another advantage is given by the polygon shaped orbit. Due to the noncircular orbit, the rotor does not perform forward or backward whirls with high frequencies, what is often the case with circular BBs which are comparably stiff. Bearing loads of the polygon shaped BB are comparably low and plastic deformation can be prevented as long, as friction inside the bearing elements is low. The global behavior of a vertical and gyroscopic rotor in this kind of planetary bearing is described by Quurck et al. (2016). It can be described as a jumping character with a low speed, friction driven backward whirl component. Exemplary simulation data of a rotor drop event with a 150 kg flywheel from 12000 rpm in a planetary BB with six bearing elements is given in Figure 9.

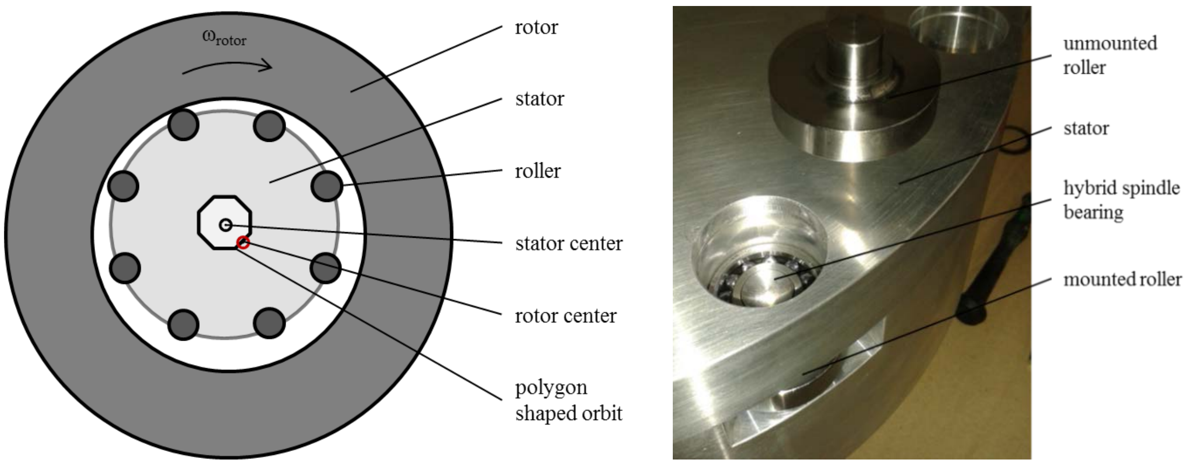

Figure 8: Planetary BB design in schematic drawing (left), photograph of the BB system during build up (right)

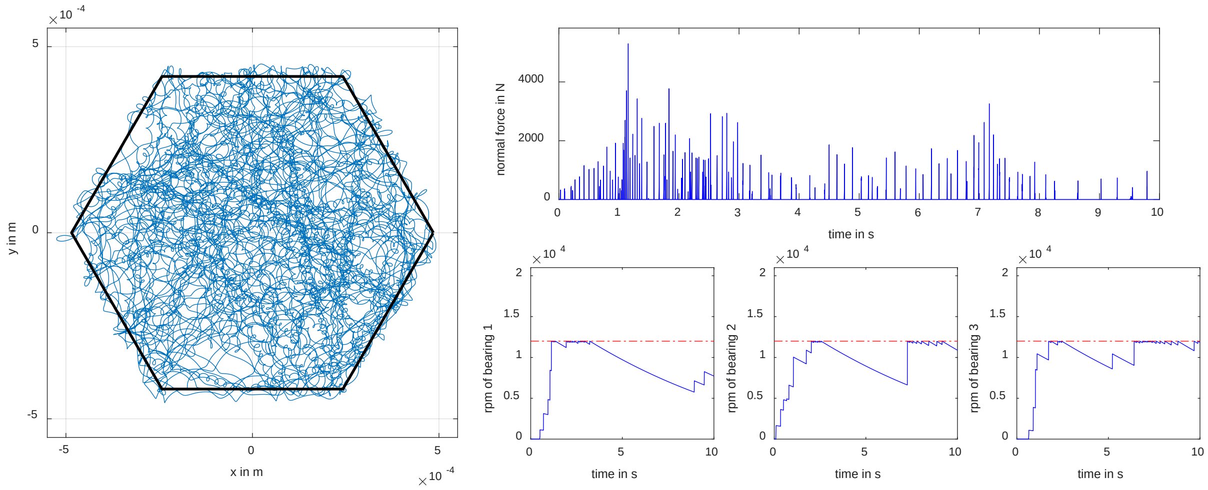

Figure 9: Rotor drop simulation in a planetary BB with six bearing units. Simulation time is 10 seconds, displayed is data of the lower BB plain. Left: Rotor position data. Right top: Normal force applied to the rotor. Right bottom: Rotational speed of bearing unit 1, 2 and 3 in solid, rotor speed as chain dotted line.

4. Final Design and Integration into the Factory

Using synthetic load profiles from measured single machines of the ‘ETA-Fabrik’ the dimensioning of electric power and capacity was finished before the factory was built. To enable significant peak shaving and load smoothing capability the storage demand was defined by 1.4 kWh and 60 kW. The factories electric protection is rated by 170 kW. Minimum 60 kW are available from 7500 rpm. While the electric power is easy to achieve, the kinetic energy is more challenging. Derived by the first prototype the inner diameter was enlarged to 290 mm and the outer diameter rose to 430 mm to meet the demands. The upper speed limit was set to 15000 rpm for secure long time operation with low risk of rotor fatigue. The resulting surface speed is 227 m/s at the inner diameter, and 337 m/s at the outer dimeter of the FRP rotor. The total rotor weight is 153 kg, while the non-FRP components take two thirds of the weight. This ratio is responsible for the comparably low energy density of 8.9 Wh/kg and shows a high potential for improvement.

The nominal air gap of the BB system is 0.25 mm, the air gap of the AMB was set to 1 mm at zero rpm. Both air gaps enlarge radially up to 0.6 mm because of the centrifugal forces. The static load capacity of one BB unit with two spindle bearings is 4200 N. The lubricant of the bearing is MoS2, as it is resistant to evaporation and its dry friction is calculable by the theory of Birkhofer and Kümmerle (2012). The physical dry friction model provides information for a thermal and a wear model of the BB which enables lifetime prognosis of these components. The system’s first run up was performed in middle of 2015 in the laboratory of the IMS. In early 2016 the whole KESS was moved into the utility compartment in the basement of the ‘ETA-Fabrik’ where it is connected to the factories point of common coupling in order to influence the residual load of the factory. The factories load is measured in the electrical main cabinet and transmitted via PROFIBUS to the programmable logic control (PLC) where the control strategy is implemented. A control loop running at 1 kHz can be used to control the electric power of the KESS.

5. Conclusion and Outlook

In the paper a novel type of flywheel was introduced for industrial applications. The high degree of integration of active magnetic bearings, permanent magnet synchronous machine and composite rotor together with the outer-rotor topology lead to many challenges during practical realization. Development, construction, and assembly of the prototype are completed so far. Open tasks are the implementation of the KESS controller on a PLC and its integration into the factory's energy management communication bus. This is going to be completed within the mentioned research project.

Based on research questions that rose from this research project, follow-up projects were started. One project focuses on an intensive modelling and testing on component level. Here a special test rig for the experimental validation of back-up bearings and novel failure tolerant magnetic bearing concepts is set up. A second test rig focuses on the long life fatigue strength of the composite rotor. Due to the uncommon stress state in the outer-rotor configuration induced by centrifugal forces, special test configurations have to be developed on material level. On component level flywheel shaped probes have to be cycled until they disintegrate to evaluate fatigue and failure modes of high thick walled, high speed FRP rotors.

By implementing design improvements concerning the radial thickness of the AMB and EM the centrifugal loads can be reduced and the rotational speed heightened. A prototype with a maximum speed of 17500 rpm and 2.4 kWh is currently implemented at the IMS. The specific energy of this system is 13.9 Wh/kg. Heightening the maximum allowable stress in the FRP to around 50% of the static ultimate strength would further raise these values to 4.8 kWh and 27.8 Wh/kg respectively, but the fatigue tests have to be done first and results have to proof these less conservative design restrictions.

A further research project called 'SWIVT' focuses on a different field of application. Smart micro grids with decentralized generation can be stabilized using energy storage systems. For such a task an electric hybrid energy storage system consisting of a lithium-ion battery and a flywheel is developed. The flywheel design is based on the described prototype and tries to improve it further on a system level.

In general, the main future challenges lie in the further integration of the component design on a system level. Also the power electronics (frequency inverter and magnetic bearing amplifiers) have to be adapted to the flywheel concept. The detailed knowledge about the reliability in terms of rotor fatigue and back-up bearing failure play a major role in order to increase the technology readiness of the design.

Acknowledgments

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"ETA-Fabrik" Project homepage: http://www.eta-fabrik.de

Addresses:

M.Sc. Lukas Quurck[1]
quurck@ims.tu-darmstadt.de

M.Sc. Michael Richter[1]
richter@ims.tu-darmstadt.de

M.Sc. Maximilian Schneider[1]
schneider@ims.tu-darmstadt.de

M.Sc. Daniel Franz[1]
franz@ims.tu-darmstadt.de

Prof. Dr.-Ing. Stephan Rinderknecht[1]
rinderknecht@ims.tu-darmstadt.de

[1] Institut für Mechatronische Systeme im Maschinenbau (IMS), TU Darmstadt, Otto-Berndt-Str. 2, 64287, Darmstadt, Germany