High Speed Backup Bearings for Outer-Rotor-Type Flywheels – Proposed Test Rig Design

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***Abstract* — Magnetic levitated rotors for high speed applications need backup bearing (BB) for safe spin down or relevitation after magnetic bearing failure or overload situations. Especially in flywheel energy storage systems long spin down times due to the high kinetic energy content and the vacuum environment complicate the use of conventional bearings as BBs. Depending on the geometric dimension, the rotational speed of the rotor and associated rotordynamic effects today's available bearings (usually rolling element bearings) cannot fulfill their task with the desired lifetime. This paper shows mechanical and thermal problems as well as approaches to mitigate them. For a built and tested Outer-Rotor-Type flywheel its BB system is discussed. For an increase of the surface velocity limit and the extension of the BB lifetime several design improvements are suggested. Properties of a test rig and its special rotor design for BB testing are described and proposed for further experimental investigation. A design of the test-BB is proposed and the data acquisition as well as the dynamic behavior of the rotor is described.**

I. Introduction

A. General problems

Rotors for high speed applications such as turbo compressors, grinding spindles and advanced flywheels are often contact-free suspended with magnetic bearings in order to reduce wear, friction losses and to provide rotordynamic properties. In order to increase availability and to spin down safely after bearing failure or overload, BBs need to be installed. At high operation speeds, current bearing systems like rolling element bearings are utilized highly and often have to be replaced after only a few drop-down events. Common reasons are mechanical bearing failures induced by overload situations. Mechanical seize of rolling elements as a result of transient thermal growth and high accelerations have also to be brought into concern. For BBs non-cage and full complement bearing design is often recommended.

B. Flywheel related problems

Considering the high kinetic energy content and the vacuum environment under which advanced flywheels are often operated, conventional bearings may not withstand a single spin down without mechanical failures. This is caused by the insufficient cooling condition of the bearing and the challenging lubrication. Both might be worsened by rotordynamic events caused by a nonlinear contact situation and be influenced by the gyroscopic properties of the rotor. It is assumed that thermal degradation of material properties including strength and variable friction coefficients are the major reasons for failure. Due to the long spin down of high inertia rotors the BB systems are exposed to these unfavorable conditions for a longer time than thinner rotors in other applications are.

C. Outer-Rotor-Type related problems

Outer-Rotor-Type flywheels are built as a hollow cylinder in order to place the rotor mass as far as possible from the rotational axis. The desired increase of inertia has the side effect of increased surface speed. This surface speed can easily exceed 100 m/s, which is very high compared to most of BB systems known from literature. One reason why ball bearings are difficult to use at such high speeds is explained at the following equation which calculates the tangential hoop stress σ_{hoop} in a thin walled ring, which might be the rotating ring of a bearing.

$$\sigma_{hoop} = \rho \cdot \Omega^2 \cdot r^2 = \rho \cdot v^2 \qquad (1)$$

In Equation (1) ρ is the material density in kg/m³, Ω is the rotational speed in rad/s, r is the radius of the ring in meters and v is the velocity of the ring. Bearings are often classified according to the DN-number representing surface speed in $mm \cdot min^{-1}$. The rotor described in the next passage has a DN-value of 7,2e6 mm min^{-1} which exceeds the known systems and leads to a hoop stress above 1000 MPa, which is in the range of material strengths of bearing steel. External bearing loads transferred through Hertzian stress and friction were not included in this calculation. Because of the high stresses inside the rotating bearing race and the expected high temperatures, the mechanical integrity of the bearing is obscure even without external loads.

D. Friction induced whirl movement of Outer-Rotor-Types

A typical behavior of Inner-Rotor-Types during a touchdown inside BBs is called the backward whirl. Thereby the rotors center of gravity performs an orbital movement inside the BB inner race against its own direction of rotation. This movement is induced by high friction forced between stator. Very high and destructive whirling frequencies can be reached. Vertical oriented rotors, like most flywheels, tend to slower and less destructive whirl movements in the direction of its rotation. [6] Because the rotor mass is not supported by radial BBs, the friction forces are lower than that of horizontally oriented rotors. Therefore the backward whirl was not reported in literature about vertical rotors.

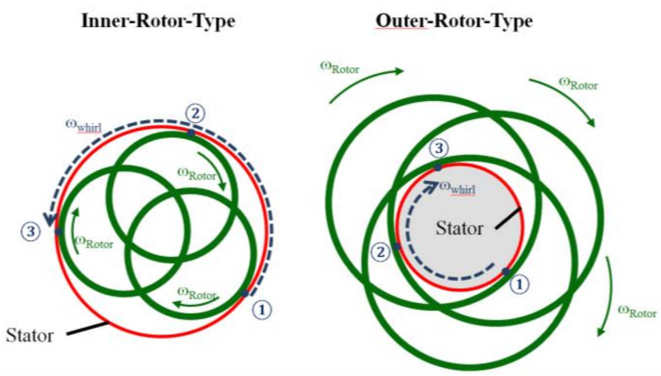

Figure 1. Friction driven whirl movements of Inner- and Outer-Rotor-Types. Numbers represent chronology of the rotor movement inside the BB.

With Outer-Rotor-Types, the kinematic description of the whirl movements "forward" and "backward" needs to be adapted. The directions of friction induced whirls are inverted at Outer-Rotor-Types. Friction driven whirl movements appear here as forward whirls due to the different geometric situation. Therefore the rotors center of gravity rotates in the same direction as the rotor. Fig. 1 shows a simplified 2-dimensional illustration of the differences between Inner- and Outer-Rotor-Types when performing friction driven whirls.

The forward whirl frequencies here tend to very high frequencies and to have a high destructive potential to the whole system like the "backward whirl" of Inner-Rotor-Types. BB failures or rigid contact between rotor and stator may cause high friction forces and drive these whirl movements. However, backward whirls are still expected to happen as long as the remaining friction forces stay under a certain level. This non-friction-driven whirls happen at a certain frequency which could be identified as the coupled elastic eigenfrequency of the rotor contacting the stator. [6]

II. First Demonsrtator rotor and BB system

A. *Description of the existing flywheel rotor*

An Outer-Rotor-Type flywheel has been built at the IMS for feasibility proof and component testing. The Flywheel is built as a hollow cylinder from fiber reinforced plastics (FRP). The components of the electric drive and the magnetic bearings as well as the contact area of the BB are fully integrated at the inner diameter and are mechanically supported by the FRP. Some technical data of the rotor is given in Table I. Fig. 2 and 3 show some views and details of the system. The two radial active magnetic bearing plains are marked as well as the axial permanent magnetic bearing (PMB) can be seen there.

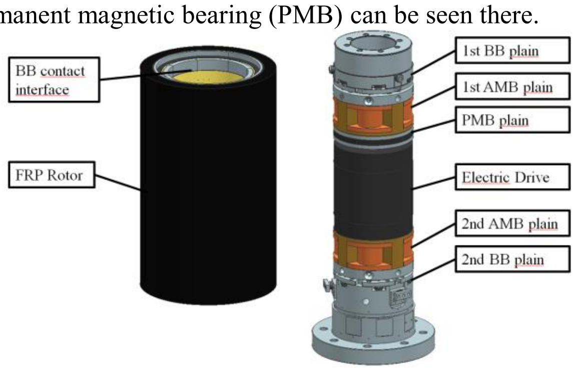

Figure 2. Rotor and stator model of the Outer-Rotor-Type IMS-Flywheel

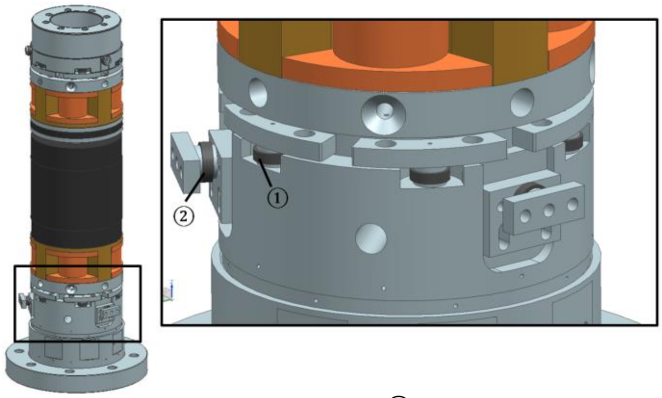

Figure 3. Detail view of the lower BB plain. ①: Radial rolling element bearing; ②: Axial rolling element bearing

The rotor can be described as rigid as numerical calculation and experimental modal analysis showed. First measured elastic eigenfrequency is above 1160 Hz, while the rotational speed of 40.000 results in a harmonic frequency of 667 Hz. It can be assumed that elastic eigenmodes are not heavily exited by the AMB which has a cut off frequency at around 1 kHz. Experimental tests confirm this assumption.

TABLE I. Technical data of the IMS Flywheel with its mechanical design limits

	Parameter	Value
d_o	outer-rotor diameter	300 mm
d_i	inner-rotor diameter	180 mm
h	rotor height	500 mm
N_{rotor}	max. revolutions per minute	40.000 rpm
v_{surf}	max. surface speed @ BB	376 m/s
I	rotor inertia	0,7 kg m²

The system was put into operation in Dec. 2013 and reached its preliminary max. speed of 21.000 rpm. For several technical reasons higher speed was not achieved yet, exemplary the first BB system was not completely supposed to withstand a highspeed drop safe without mechanical failures. The reached energy content at 21.000 rpm is 0,48 kWh. Mechanical challenges and uncertainties which lead to the reduced max. speed of the first BB system are discussed in the next paragaph II.B. An improved BB System with higher load capacity and higher predicted speed limitations is described and discussed at paragaph II.C and II.D.

B. *Description of the first BB System*

As no manufacturer can provide centrally mounted bearings with an outer diameter of 180 mm, that can withstand the occurring rotational speeds and accelerations, alternative concepts are considered. Instead of one single BB per BB plain, six smaller bearings are uniformly distributed around the circumference of the stator. Similar designs were proposed and tested in [1] and [2]. Clearance elimination at BB contact mentioned in [1] and [2] was not implemented. While being levitated the rotor does not contact the bearings. After being dropped into the bearing the rotor does contact one or two bearings in each plain at the same time. Because of the play between the inner diameter of the rotor and the bearings the rotor is able to change its position inside the BB system. Because of the different sizes of the rotor inner diameter and the outer bearing race diameter, a kinematic translation leads to different angular speeds which can be calculated using Equation (2). The surface velocity still stays the same as it is the boundary condition at contact area when slipping is disregarded.

$$N_{BB} = N_{rotor} \cdot d_i / d_{o,BB} \qquad (2)$$

Data of the chosen bearings are listed in Table II.

TABLE II. TECHNICAL DATA OF THE IMS FLYWHEEL

	Parameter	Value
$d_{o,BB}$	Outer bearing diameter	19 mm
$d_{i,BB}$	Inner bearing diameter	10 mm
$N_{rotor,prel.}$	Preliminary revolutions per minute	21.000 rpm
N_{BB}	Max. bearing revolutions per minute	189.000 rpm
n_{BB}	Number of bearings per plain	6

Because of the high mechanical stress in the outer races of the bearing the maximum angular speed of the rotor was reduced to the half of the projected speed of 40.000 rpm until the dynamic system behavior in BB is better investigated or design improvements mentioned in the next section are implemented.

C. *Discussion of the first BB system*

Fortunately the planetary arrangement of multiple smaller bearings brings some advantages. First of all the selected ball bearings provide dry lubrication for long lifetime and low friction coefficients in high speed and high temperature operation. Dry lubrication also does not suffer from evaporation under high vacuum conditions. According to the manufacturer's information these bearings are used in turbo molecular pumps as BB at around 90.000 rpm withstanding several dropping events without failure. A second application of this bearing type is in micro turbines as thrust bearings. They withstand more than 200.000 rpm at high temperature for several minutes. Because of these operational experiences the selected system was assumed to withstand at least one rotor drop from 20.000 rpm to safe standstill. Hoop stress stays under 280 MPa so that additional bearing loads can be tolerated.

The second advantage of the chosen concept is the polygon shaped orbit of the whole BB system. It is assumed that whirling movements of the rotor might be suppressed, which was tested in former works [3]. In addition, low BB inertia is assessed to be good during touchdown events because synchronization time between rotor and BB can be shortened. The shorter these synchronization times are, the is lower the probability of a friction driven whirl movement. The first BB system has several magnitude orders lower inertia compared to a theoretical possible central bearing configuration. Furthermore specific compliancy of the bearing can be achieved by design parameters in order to support rotordynamic behavior in BB. Some disadvantages still have to be taken into account with this design approach. The hoop stress stays the same following (1) and (2), as angular speed of the BB is inversely proportional to its outer diameter $d_{o,BB}$. Additionally extra load from the balls being pressed against the outer bearing ring occurs. In general outer ring rotation is not the best load case for ball bearings. Additional axial forces and tilt moments are possible loads during dropping events. When bearing tilting exceeds a certain level, the ball filling grooves of the bearing can cause seizing when roller elements roll over these grooves.

Regarding all drawbacks and uncertainties of rotordynamics and thermodynamics no high speed BB testing was made with the first BB system. More investigation and design improvements mentioned in next section are necessary to increase lifetime and to enhance performance.

D. *Load case optimization*

To reduce mechanical and thermal loads of the BB system some design changes are implemented in an improved stator design of the flywheel. The main aim for this improvement is to reduce angular BB speed without reducing rotor speed. This can be done by increasing the outer diameter of bearing interface. The bearing speed is reduced linearly with the increasing bearing interface shown in Equation (2). Adding roller elements at the bearings outer diameter is the easiest way to achieve this as shown in Fig. 4 on the left side. Major drawback of this configuration is the different strain of the roller element and the bearing which can result in a lift off at high speed. Depending on geometric parameters, high preload and elastic elements between the outer ring and the roller element can solve this problem. Calculations were made for the introduced flywheel and this solution was abandoned because lift of could not be prevented without seizing the bearing due to the necessary high preloads. Elastic interlayers were also discussed but calculations failed because of unknown thermal behavior of the BB.

A promising configuration can be found when the inner race of the bearing is chosen for rotation. Forces are transmitted with roller elements called "rpm-reduction-disc" (RRD). RRD are mounted at the inner bore by one or more ball bearings. By adding a second bearing the load capacity can be increased simultaneously and tilting of the rotor as well as axial forces can be supported. Thermal junction to the stator and general load case are improved at the same time. The RRD design is illustrated in Fig. 4 on the right side. In the next paragraph the improved BB system for the IMS Flywheel is described in more detail.

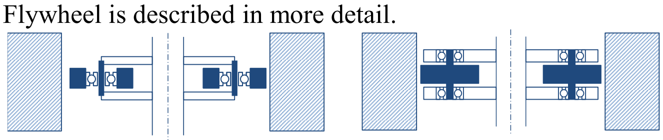

Figure 4. Left: Roller elements mounted on the outer bearing race; right: RRD mounted inbetween a set of two ball bearings.

E. *Improved BB data*

RRD are made with an outer diameter of 25 mm and turned out of tempering steel. Due to the full cylinder design and the reduced bearing speed, the mechanical stress induced by centrifugal forces stays under 460 MPa when the RRDs are accelerated to the surface speed of the flywheel rotating at 40.000 rpm. The maximum stress in the BB system is reduced to less than half compared to the first BB System without the RRDs at full speed. The bearing size stays the same as in the first system but a second bearing was added to heighten load capacity. Fig. 5 shows a cross-section of the lower BB plain for the IMS Flywheel. Experimental testing remains to be done. For efficient testing another experimental environment has to be set up. A special test rig is assumed to be the best environment for investigation as it is described in the following chapter.

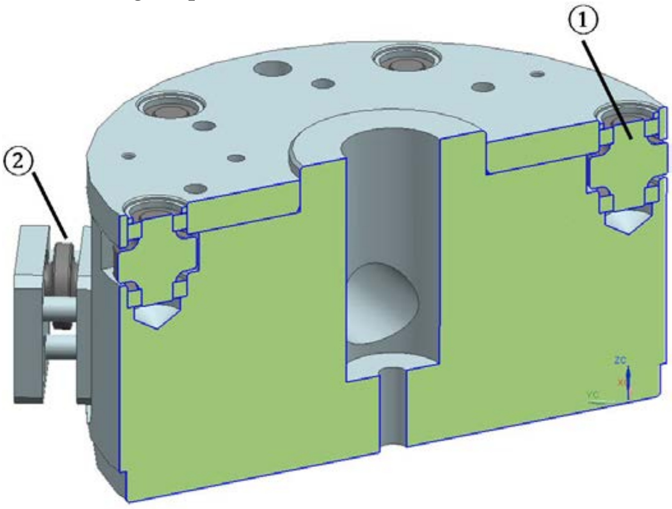

Figure 5. Cross-section of the lower BB plain of improved bearing BB system for the IMS Flywheel. ①: RRD with rolling element bearing for radial support; ②: RRD with rolling element bearing for axial support

III. Proposed Test Rig

A. Nesessity of experimental testing

Because of complex dynamics, unknown thermal junctions and a lack of knowledge about mechanical limits like thermal induced degradation of material properties or lubrication, the experimental investigation of the BB system behavior is reasonable. Experimental investigation on the described flywheel would be desirable but brings the high risk of irreparable damages. Therefore an alternative test rig for intense and safe experiments is considered as the best way for observation and testing of design improvements. Furthermore the experimental investigations may deliver important information about parameters for numerical investigation and optimization. In the following section a design of an active magnetic levitated substitute rotor and its components are discussed.

B. Properties of test rig

The suggested substitute rotor needs to be magneticly levitated and accelerated to comparable surface speeds. After reaching the desired speed, the rotor has to be dropped into the BB for example by cutting the power supply of the AMB. To generate similar operation environment as for the flywheel the housing needs to be evacuateable.

In addition to the required mechanical strength and robustness of stator and rotor, modularity is one main feature. The test rig has to withstand numerous dropping tests. Increasing robustness and thermal stability can be achieved by using steel instead of FRP. For high speed testing full disc design leads to lower material stresses compared to hollow disc rotors. Inner-Rotor-Type design is proposed therefore. Except of the different direction of the friction driven whirl the Inner-Rotor-Type is supposed to behave like Outer-Rotor-Type. Moreover standard components and machinery can be used. Planetary bearing arrangement can still be provided and high surface speeds are achieved by the diameters at the BB plains. The AMBs and the drive system are restricted to lower diameters and therefore surface speeds due to the mechanical integrity of the interference fitted sheets and magnets. Vertical orientation of the rotational axis is important for the investigation especially because the majority of prior works deal with horizontal rotors. In [5] and [6] two different vertical oriented flywheel rotors were dropped into BB and only forward whirl movement was observed. The observed whirl frequencies seem to be limited by the rigid body eigenfrequency of the rotor contacting the BB. The backward whirl movement was not observed, which can be explained by the absence of a strong radial force like weight force. Such strong radial forces induce tangential friction forces and drive whirling. For a rotordynamic comparability of the test rig, the gyroscopic properties must match to that of the original flywheel rotor. This can be achieved by setting the ratio of axial and polar moments of inertia as described by Equation (5) near to the original rotors ratio.

$$K_{inertia}=I_p/I_a \tag{5}$$

The ratio of the introduced flywheel is near 0.5. The higher the ratio of a rotor is, the bigger the gyroscopic behavior of the system is. Thin and long rotors have low ratios while flywheels tend to have bigger ratios. Formulas for axial and polar moment of inertia of a full cylinder are given in Equations (6 & 7).

$$I_p=1/2\ m\ r_o^2 \tag{6}$$

$$I_a=1/4\ m\ r_o^2 + 1/12\ m\ l^2 \tag{7}$$

m represents the cylinder mass, r_o is the outer radius and l is the length. $K_{inertia}$ is a major feature for a stable control of the rotordynamics in active magnetic bearings [4], as it represents the gyroscopic properties of the rotor. Another important dynamic characteristic is the elasticity of the substitute rotor. The first elastic bending eigenfrequency should be widely above the rotation frequency of the rotor as it is given for the original rotor made from FRP.

C. Substitute rotor for BB testing

Regarding the dynamic behavior of the substitute rotor a design has been done. Two discs act as the contact area to the stator bound planetary BB sets and provide high gyroscopic behavior. For assembly at least one disc has to be removable from the rest of the rotor. The rotor shown in Fig. 6 has an inertia ratio of $K_{inertia}$ = 0,5 and its first bending eigenfrequency is 1806 Hz. Turning at 20.000 rpm, FE-Analysis shows maximum mechanical equivalent stress around 300 MPa at the BB discs. The maximum surface velocity is at 209 m/s, which is assumed to be sufficient for BB testing. Polar moment of inertia is around 0.05 kg m² leading to a kinetic energy content of max. 0.03 kWh. The weight of the rotor is 17 kg. For fast breaking capability the max. power of the electric drive is around 15 kW. Electric emergency braking takes under 50 seconds from full speed to standstill, disregarding friction inside the BB system. One important feature of the rotor is the possibility of using a secondary BB plain which can be mounted at both ends of the rotor. The clearance at the secondary BB should be slightly larger compared to the primary BB. Thereby it can be used as a emergency BB when the primary BB has a severe mechanical damage. Moreover investigations on dynamic behavior of the rotor being dropped inside the conventional central mounted BB without the planetary BB can be done and may deliver important information about general behavior of both approaches.

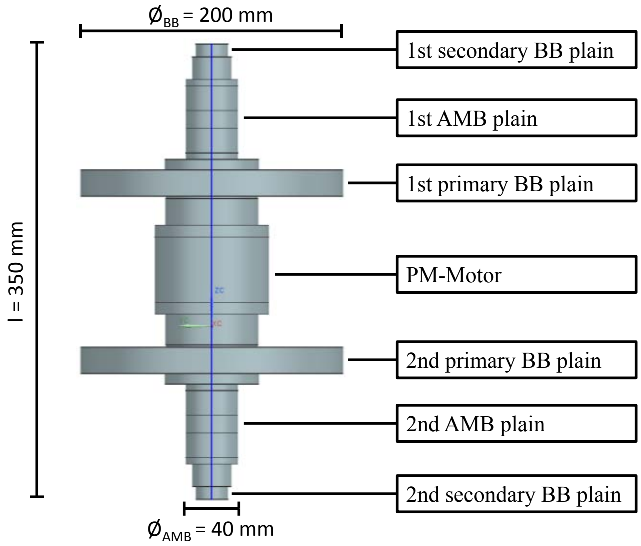

Figure 6: Model of the substitute rotor for BB testing

D. *Test Rig Equipment*

The substitute rotor is completely levitated by AMBs. The active axial bearing compensates the weight of the rotor and is capable to center the rotor during radial touchdown events in order to prevent axial contact. This is necessary if the influence of an axial BB contact is unwished. The positioning sensors of the AMB are used to record the orbits during testing with high sampling rates up to 20 kHz. Because of the high stiffness of the rotor no additional range sensor is needed in the BB system. Optical temperature sensing by pyrometers is used to calculate heat generation at the BB contact zones. Temperature probes can record the temperature at the non-rotating parts of the BB system. Thereby heat transfer is observable.

A synchronous drive system is used to accelerate the rotor to the desired dropping speed. After dropping the rotor into BB the drive is able to brake with up to 12 Nm. Idle coasting of the dropped rotor is another opportunity for simulating an unassisted spin down e.g. when the drive system has a serious failure and cannot actively decelerate rotation. For drive control and dynamic observation of the rotor speed, a rotational encoder for angular position is necessary test rig equipment.

E. *Design of the advanced BB system*

In Fig. 7 a design of a BB system with planetary arrangement of multiple RRDs is shown. The primary BB design for the test rig differs from the original flywheel system in the geometric dimensions and the inverse design of the Inner-Type-Rotor. A further design difference is the variable air gap between the rotor and the RRDs. Every single RRD is beared inside a radial slideable mount. Therefore the influence of the air gap size can be investigated. The air gap is not varied during a test in this design approach. Advanced approaches with variable actuated air gap control may be investigated in later works as well as predefined compliancy. Positive influences on bearing loads and lifetime is expected from certain dimensioning.

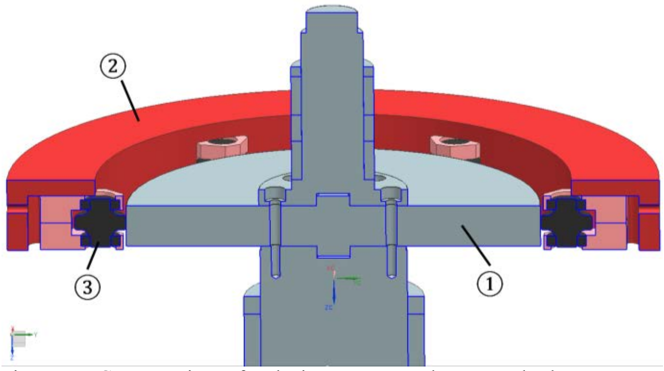

Figure 7: Cross-section of substitute rotor and proposed planetary BB. ①: Rotor; ②: Stator of planetary BB set; ③: RRD with rolling element Bearing inside a slidable mount

I. SUMMARY

This paper introduces a novel high speed Outer-Rotor-Type Flywheel with an experimentally achieved energy content of 0.47 kWh at 21.000 rpm. Full speed of 40.000 rpm was not achieved yet. One reason are the uncertainties of the first BBs load capacity. The risk of a BB failure at high speed operation was assumed to be too high. For heightening load capacity and speed range of the BB, a design improvement was made and introduced. For efficient testing a special test rig with a substitute rotor is suggested. The substitute rotor for this test rig is designed as Inner-Rotor-Type with comparable dynamic properties as the flywheel rotor. The modular steel design of the fully active levitated substitute rotor provides the needed thermal and mechanical robustness for multiple BB dropping tests and reduces risk and cost of future investigation. Properties of the data recording of the test rig are described and an exemplary BB system with planetary arrangement of multiple single bearing sets with adjustable air gaps is introduced.

II. OUTLOOK

Primary aim of the proposed test rig is to investigate a high speed BB system for Outer-Rotor-Type flywheels. The current configuration with 6 planetary bearings in each plain is not assessed to withstand multiple dropping events from high speed without failure. Because of difficulties in mechanical and thermal calculation and unknown thermal degradation of material properties inside the bearings, experimental testing is obligatory to learn more about limiting loads, temperatures and wear of the bearing. Influences of ambient pressure on friction, thermal behavior and wear are expected to deliver important information for further BB designs. When important parameters are identified and determined numerically simulation can achieve better results and further design improvements will be achieved more efficiently.

The use of roller elements called “rpm-reduction-disc” (RRD) as an improvement of the current flywheel BB may improve performance and lifetime of the system. Increasing the limiting speeds of the BB enables the possibility to increase the kinetic energy content of the energy storage under safe and economic circumstances. This is an important step towards industrial utilization and commercial operation of Outer-Rotor-Type flywheels.

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