Increasing bearing life-time of mobile flywheels by using bearing-less drive methods

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Abstract—**Fully electric and hybrid electric power trains enable reuse of braking energy. This decreases the amount of lost energy dramatically, especially for discontinuous operation. Good examples are urban traffic, public transportation or delivery services. Mobile flywheels provide an excellent potential for the emerging market of electrified power trains, but they are currently only used in a few applications such as motor sports (Formula 1, Le Mans, GT3 car racing). To enable a wider use in personal or public transportation, the life-span of the flywheel's bearings have to be increased significantly. This paper presents an alternative approach to extend the life-span of the rolling element bearings by using bearing-less drive methods.**

I. Introduction

Energy storage in combination with renewable energies is key to a low carbon society. For mobile power-trains energy storage also enables a more efficient operation by restoring recuperated kinetic and potential energy. This justifies the enormous effort put into the research and development related to batteries, capacitors and flywheels.

A more efficient and sustainable mobility requires energy storage that enables vehicles to take advantage of features such as load averaging and regenerative braking. The energy storage therefore underlies requirements like a high power density, a high cycle life-time and a long service life. Flywheels meet these requirements and provide further advantages such as no capacity fade over time and an extremely wide operating temperature range. Further no limited or environmentally harmful resources need to be used and the state of charge (SOC) is always known exactly.

II. Motivation

To enter the mobile energy storage market successfully, the weight, size and costs of flywheel systems have to be a minimum. With magnetic bearings, the flywheel system has virtually unlimited bearing life, but it would require back-up bearings and a more sophisticated and therefore more costly rotor design. Previous research conducted at Graz University of Technology has shown that the weight, size and costs of a mid-sized car's flywheel energy storage system (FESS) increases dramatically when magnetic bearings are used (project "PowerKERS", FFG 825.553). Additionally the rotor gets decoupled thermally from the stator, which can be cooled easily and therefore increases the risk of overheating due to the eddy current and hysteresis losses of the rotor lamination. Since the flywheel is operated in vacuum to reduce air friction loss, there is no convection cooling inside the vacuum housing. Therefore radiation would be the only way to dissipate power if there are no mechanical bearings used.

These facts are also represented by the current market situation too. All suppliers of mobile flywheel energy storage systems worldwide apply lighter, smaller and cheaper hybrid rolling element bearings. Unfortunately they suffer from a significant life-time shortening by loads which occur in driving operation. With an average car-life of 250.000 to 300.000 kilometers and an assumed average speed of 60 km/h the average car life-time is estimated with 4500 hours. The life-time of rolling element bearings can be estimated by the product of rotational speed and bearing diameter. To reach the 4500 hours of required car operation with an average flywheel rotor speed of about 40.000 RPM the maximum bearing diameter is 18 mm for steel ball bearings and 31 mm for hybrid ceramic ball bearings. The results are based on the extrapolated data given from Schaeffler's FAG product data sheet of deep groove ball bearings as depicted in Fig. 1 [1] and do not include occurring bearing loads.

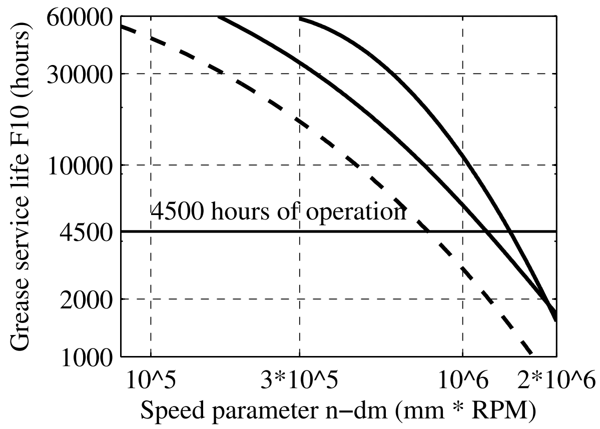

Figure 1. Estimated bearing life-time by grease service life for steel ball bearings (dashed) and hybrid ceramic ball bearings (solid) [1]

The graph shows that ceramic ball bearings can manage the high rotational speeds over the required period of time.

Table I. EVALUATED AND WEIGHTED DESIGN METHODS FOR STANDARD BEARING SYSTEMS AND THE PROPOSED BEARING SYSTEM

Main FESS application		stationary FESS & racing	stationary FESS	proposed method for flywheel hybrid cars
	Weight	Mechanical bearings	Active magnetic bearings	Mechanical bearings with additional brush-less load relief
Bearing life	5	- - -	+++	+
Construction space	3	+++	- - -	-
Variable damping	3	- - -	+++	+++
Rotor dynamics	2	+++	- - -	++
Thermal conductivity	2	+	- - -	+
Costs	2	+++	- - -	- -
Weight	1	+++	- - -	-
Sum of points		2	-6	10

Unfortunately they do not last long enough when the occurring bearing loads are considered. The estimated bearing life time for the ceramic rollers is higher than for steel ball bearings, since they feature a lower density, which reduces centrifugal forces on the outer bearing ring. They also have a much harder surface, which reduces wear. With the hybrid ceramic rolling element bearings it is possible to meet the requirement of 4500 hours of operation. Unfortunately the bearing life also strongly depends on occurring bearing loads, the operating temperature and contamination of lubricant. With the estimated occurring bearing load spectrum that results from the conducted research project ”PowerKERS’ (FFG 825.553) at Graz University of technology, the bearing life is not sufficient though. The new proposed method combines the advantages of both, simple mechanic bearings and active magnetic suspension without the need of backup bearings. The results are listed in Table I.

III. TECHNICAL OUTLINE

The motivation is clearly to reduce the bearing loads due to driving operation. The bearing loads are compromised of static and dynamic rotor imbalance, linear vehicle acceleration and gyroscopic torque. The share of rotor imbalance depends of course on the balancing quality. Acceleration and deceleration of the vehicle and gyroscopic reaction torque due to vehicle deflection are the main contributors to bearing’s loads.

In stationary flywheel energy storage systems with ball bearings a resilient bearing mount is used. The resilient bearing mount is basically a soft mechanical bearing mount with a progressive stiffness, as can be found in elastomers and springs. This bearing mount reduces imbalance loads drastically, especially in overcritical operation. To avoid excessive rotor oscillation caused by gyroscopic reaction, rotor bending or vehicle movement, a certain rate of damping is required. In overcritical operation damping of rotor oscillation leads directly to loss of rotational energy. Therefore a variable rate of damping is desired.

A constant rate of damping can be applied by squeeze film dampers, which are used in turbo machinery. In this case the damping forces are transferred to the rotor through the rolling element bearings, and that results in constant bearing loads in overcritical operation. To avoid this constant bearing loads the damping forces have to be applied to the rotor contact-less. Applying damping to a rotor with no contact is possible by active and passive type magnetic bearings [2].

A. *Which magnetic bearing type to choose?*

Superconducting magnetic bearings feature good stiffness and damping properties but both characteristics are not variable and they require cryogenic cooling, which is not possible with reasonable effort for vehicles. Especially not if the are occasionally in use. Therefore cryogenic cooled superconducting bearings are applied successfully in large scale stationary FESSs (e.g. Beacon Power LLC, Boeing R&T). Passive magnetic bearings, e.g. ”Null-Flux” homopolar induction bearings or permanent magnet bearings feature a certain rate of stiffness and damping. They can be used to compensate the axial bearing loads due to the rotor’s weight. Their power density is lower than active bearings though and a variable rate of damping is missing.

That leaves active control as best option. A previous investigation at Graz University of Technology has shown, that even a magnetic bearing, that produces just a fraction of the estimated maximum bearing force would increase the rolling element bearing’s life-time dramatically [3]. The reason for this is that smaller bearing loads are much more likely in the bearing’s load spectrum and therefore a small load relief has still a remarkable effect. The bearing load spectrum is basically an estimation of the probability density function of occurring bearing forces.

Additional active magnetic bearings (AMBs) are equipped with sophisticated design efforts for the flywheel’s rotor structure. They also increase the rotor length, reducing bending critical speeds. Even though smaller AMB are able to extend the FESS’s service interval, it still requires additional construction space and a more sophisticated rotor design. To just gain bearing life, a much simpler approach is to adapt the flywheel’s stator to produce radial force and relief the mechanical bearings. The proposed method is in general used in bearing-less drives and there are three types of electric motors for flywheels. Induction motors, permanent magnet motors, and reluctance motors [4].

B. *Which motor type to choose?*

There are many possible types, which will be reviewed in the following paragraphs from the perspective of vehicular flywheel systems.

1) Induction machines: They can be constructed to produce radial force by means of 2 separate sets of windings as proposed by [5]. IMs with a squirrel cage structure feature for good driving torque performance. Without the squirrel

Table II. EVALUATED AND WEIGHTED BEARING-LESS DRIVE TYPES FOR THE PROPOSED FLYWHEEL BEARING SYSTEM

Motor type	Weight	**Induction motor**	**Permanent magnet motor**	**Switched reluctance motor**	**Synchroneous reluctance motor**
Rotor losses	5	- - -	+++	- - -	+++
Power density	3		+++	+	+
High rotor Temp.	3	- -	- - -	+++	+++
Costs	3		- -	+++	++
Mechanical safety	2	- -	- - -	++	++
No-load torque	2	+++	- - -	+++	+++
Round-trip efficiency	2		+++	-	+
Noise	1	+	+++	- - -	
Reactive Power	1	-	+++	- -	- -
Points		-20	9	9	43

cage, a laminated or massive rotor is coated with a plated copper layer. The copper coated type induction machines are used for high speed applications. The massive rotor design is not suitable for flywheels due to the classical magnetic losses in the solid rotor. Additionally in case of a rotor failure, the released energy of solid rotor fragments is dangerous. In contrast, sheets of soft magnetic lamination do not break simultaneously and therefore less energy is released at the same time. Further the thin sheets are expected to dissipate the kinetic energy faster into deformation.

2) Permanent Magnet Motors: They benefit from a high power density and there are bearing-less types available in a large variety of topologies. Cylindrical PM rotors, inset type PM rotors, buried PM rotors and slice rotors [6]. The permanent magnets no load flux flows through the stator even when the flywheel is in standby and that results in magnetization drag. This affects flywheels that have a relatively high power compared to their stored energy. That applies for vehicular systems. An example is a short term flywheel energy storage for hybrid vehicles. An automotive flywheel would therefore be empty in estimated 15 minutes. Such a high self-discharge rate is still not a problem for mobile FESSs. The rotor drag increase the fuel consumption theoretically by estimated 0.1 liters or less of gasoline per 100 driven kilometers.

The limited and costly rare earth elements of the permanent magnets are a serious drawback for a possible market entry of mobile flywheels. They also limit the rotor temperature to about 120 to 140 °C. High temperature samarium cobalt magnets extend the possible rotor temperature to 200 °C and more. Unfortunately they are more brittle than the neodymium iron boron magnets. To hold the rotor's permanent magnets in place a thin layer of high tensile strength material is needed to preserve a small magnetic gap. Carbon fibre composites feature the actual highest ratio of tensile strength to density and are therefore the best choice. Nevertheless this facts increase the costs drastically and the standard epoxy resin matrix is degraded by rotor temperatures above 110 °C. In terms of safety massive magnets are also critical. In the event of a bursting rotor smaller fractions release less energy at the same time. With circumferential speeds of several hundred meters per second the rotor fractions collide with the crash ring with the same speed as a rifle bullet.

3) Reluctance motors: They are in general robust, cost effective and sustain high rotor temperatures theoretically up to 300 °C. The rotor temperature is actually limited by the rolling element bearings. Equally as induction machines they have no no-load drag and they also require a higher amount of reactive power.
Switched reluctance motors (SRMs) are particularly cheap and robust, since the have at least 3 concentrated windings. As long as the motor keeps spinning, they even produce torque with just one remaining active phase. That is very valuable in case of a single or multiple phase-fault. SRMs suffer from high magnetic losses in the rotor and the stator due to the repeated current excitation of all phases for a single revolution. With an alternating bi-directional current pattern, they magnetization losses can even be reduced by about 20 % [7]. Previous work has shown that the rotor losses of SRMs can be estimated by simulation based on the soft magnetic material properties [8]. For an example lets assume an average power dissipation of 300 W at a rated power of 20 kW with a duty cycle of 50 %. Under these circumstances the rotor of a car flywheel reaches estimated temperature of 200 °C.

The synchronous reluctance machine is the best overall choice for a simple, cost effective, robust and efficient flywheel for hybrid electric or fully electric vehicles as summarized in Tabluar II. The radial forces are produced by combining 2 sets of windings. One 3-phase rotating magnetic flux field for the torque production (flux oriented d-q control) and one set of windings for radial forces [6]. The superposition of both magnetic fluxes enables to relief bearing loads and to gain bearing life.

IV. METHOD

A. Rotor Design

To design an internal motor flywheel (simplest and easiest method) with a desired energy content, the rotor geometry can be simplified by a cylindrical shape. With the known density ρ of the soft magnetic rotor laminations the energy content can be formulated directly by the rotor's diamter d and height h. That leaves the diameter to height ratios as one degree of freedom for the rotor design. A ratio smaller than one ($d/h < 1$) results in a larger area to produce radial force and also in a longer lever to counteract gyroscopic torque. Long and slim rotors ($d/h < 1$) tend to be softer in axial bending than short and wide rotors ($d/h > 1$). The slim rotors also enable to use soft magnetic steel sheets with less tensile strength. To increase tensile strength cobalt alloys are used. With added cobalt costs increase dramatically as well as hysteresis losses due to the increasing grain size. On the contrary the cobalt alloys feature saturation flux densities B_s of about 2.2 Tesla whereas Si-Fe sheets usually saturate at 1.8

to 2.0 Tesla.
Figure 2 shows that a rotor with 90 Wh of stored energy can be built out of standard steel sheets, but it is not recommended due to the resulting long and thin rotor geometry which reduces the axial bending stiffness of the rotor.

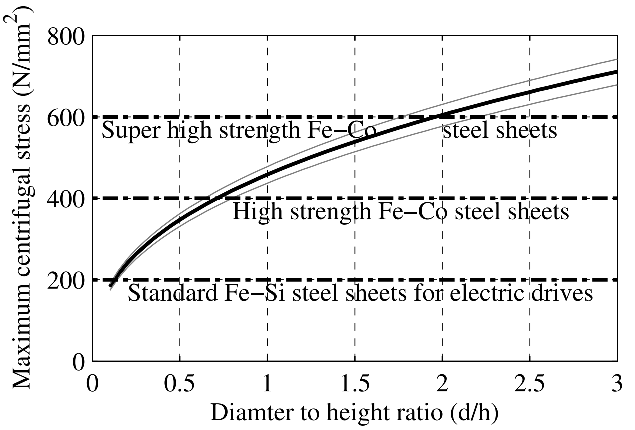

Figure 2. Maximum tensile strength for 90 Wh of stored energy (uncertainty boundaries are ± 10 Wh) with a maximum rotational speed of 60.000 RPM and a density of $\rho = 8.2\ g/cm^3$. The material limits (dashed-dotted lines) include a safety factor of 2.

With high strength cobalt alloys d/h ratios of 0.5 to 0.7 are possible resulting in strong radial forces and good torque capability. A good method would be to produce driving torque with flux densities up to 1.8 Tesla and leave a flux density reserve of 1.8 to 2.0 Tesla for radial force production and counteracting gyroscopic torque. The ratio in Figure 3 is calculated with this proposed flux density reserve of 0.2 Tesla and the principle of virtual displacement.

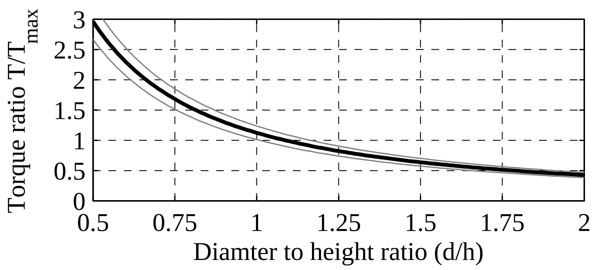

Figure 3. Ratio of highest counteracting torque T to the defined maximum gyroscopic torque ($T_{max} \approx 130$ Nm) occurring at an angular vehicle deflection speed of 1 rad/s which is equivalent to driving over a speed bump at 50 km/h. The uncertainty boundaries are ± 10 % of the radial force capability.

Bearing loads due to gyroscopic reaction torque can be counteracted with d/h ratios of up to 1.08. Since the defined maximum gyroscopic torque is mainly depending on the stored energy the ratio gets better for thinner rotors. Another important bearing load source is linear acceleration forces. The analysed measurement data from the research project "PowerKERS' (FFG 825.553) conducted at Graz University of Technology has shown that the highest amplitudes of linear acceleration that do not lead to a complete technical failure of the vehicle occur at fender benders and small crashes. The g-rates are in the range of 10 g ($98.1\ m/s^2$) as long as the impact is absorbed by plastic and thin metal components. That leaves the vehicle by definition in an operational state and such g-rates should therefore be covered by the proposed system. Figure 4 shows that for d/h ratios below 0.84 g-rates can be counteracted for the given energy content of 90 Wh and the maximum rotational speed of 60.000 RPM.

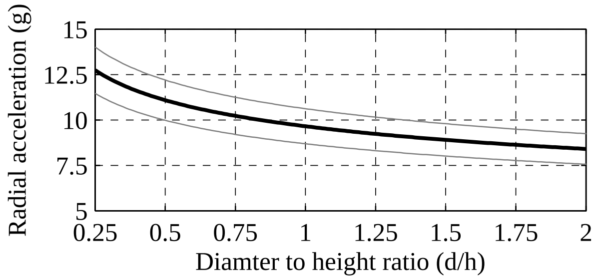

Figure 4. Maximum radial acceleration as a function of the rotor geometry to counteract linear acceleration.

B. Estimating rolling element bearing life-time

As depicted by Figure 5 a test car from the FH Joanneum University of Applied Sciences was equipped with linear acceleration sensors and angular speed sensors in 3 spatial axes. From the recorded measurement the bearing load shares are estimated on the basis of the Euler Equations, the flywheel geometry and the flywheel's suspension. It should be noted, that the test car was not equipped with a flywheel. The feedback from the flywheel's reaction torque to the car is negligibly small which was proven by simulation in [9]. With the estimated bearing force statistics basic life-time estimation formulas from rolling element bearing suppliers can be fed. This step is explained in more details in [3].

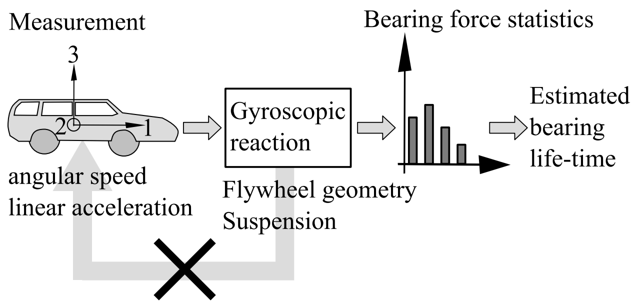

Figure 5. Overview of the rolling element bearing force statistics and life-time estimation.

The following results are calculated with a d/h ratio of 0.7, a stored energy of 90 Wh and a maximum rotational speed of 60.000 RPM. The rotor's diameter results in 0.11 m and the height is 0.157 m. The measurement data was collected on a course with a length of 60 km comprised of single-lane and dual-lane city traffic, stop-and-go traffic, inter-urban roads and a motorway segment. Driving over a speed bump is included as well. To get a one-dimensional bearing force probability density function as depicted in Figure 6 the resulting bearing force components are added geometrically.

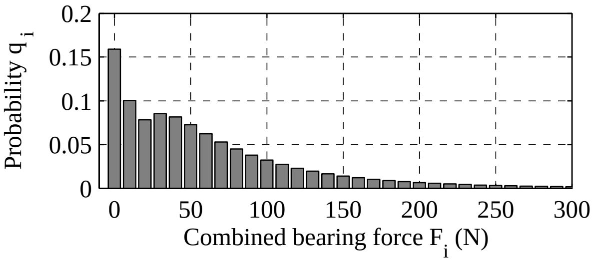

Figure 6. Estimated probability density function of the occurring bearing loads based on measurement data from 4 test-cars (ML320, Opel Movano, VW Passat and Smart Roadster). The bearing forces include linear acceleration loads and gyroscopic torque loads.

From this estimated probability density function the estimated fatigue life L10h can be calculated by Equation 1. L10h is the estimated amount of operational hours that a large number of bearings will sustain with a probability of 90 %. The rotational speed n in RPM is assumed with 44.750 RPM, which corresponds to the flywheel's state of charge (SOC) of 50 %. The corresponding rotational speed at the moment of the occurring bearing load could be considered. This relatively big step in complexity would additionally require to simulate the driving operation with the hybrid drive concept and was therefore left out.

$$L_{10h} = \frac{16666 \cdot C^3}{n \ \cdot \sum_{i=1}^{N} q_i \cdot F_i^3} \tag{1}$$

C. Lubrication method

The hybrid ceramic bearings can be operated grease-lubricated and oil-lubricated. Oil-lubrication requires a closed oil circuit with a filter system and a small oil pump. The oil-flow has to be controlled depending on the rotational speed to reduce unnecessary viscous losses. Oil-lubrication increases the design effort and therefore mainly the costs. On the contrary it features the possibility of direct cooling and higher rotational speeds for rolling element bearings. Therefore higher dynamic load ratings can be achieved for the same speed range and that enables a longer bearing life. Grease-lubrication is just less sophisticated and less costly. Tabular III compares the estimated fatigue life of a grease-lubricated and an oil-lubricated hybrid ceramic bearing.

D. Results

The data presented in Tabular III shows that even with oil-lubrication 10 % of all flywheels would become inoperative within half of the required life-time of 4500 hours. The average estimated car-life of 4500 hours is reached in public transportation applications within 1 year.

Table III. ESTIMATED FATIGUE LIFE FOR 2 HYBRID CERAMIC BEARINGS FROM UKF [10]

Bearing Type	719 UHC 30 A15	719 UHC 20 A15
Lubrication	oil	grease
Diamter	30 mm	20 mm
Dynamic load rating C	2470 N	1820 N
Axial pretensioning	60 N	40 N
L10h (90 %)	2106 h	894 h
L5h (95 %)	1306 h	554 h
L2h (98 %)	695 h	295 h
Required life-time factor	2.1 - 6.5	5 - 15.2

E. Assessing the bearing-life extension

The chosen geometry enables enough area to produce the required radial forces. Depending on the power electronics intermediate circuit voltage the number of windings of the radial force windings have to be adjusted. The aim is to be able to apply the required control current fast enough. The length of the maximum time-lag depends on the way the flywheel's is allowed to travel. Due to the progressive stiffness of the resilient bearing mount the rotor can move in a small orbit around the center of rotation without producing significant bearing loads. The goal is to keep the rotor inside the no-load orbit. With a given air gap of 600 μm and the diameter to height ratio of 0.7 about 40 % of the air gap (240 μm) have to be reserved for centrifugal growth of the rotor and for temperature dependent growth of the rotor (maximum Temperature assumed T_{max} = 200 $°C$). This leaves about 300 μm for the resilient bearing mount to get from very soft to hard. This should leave an orbit of about ±100 μm for the proposed brush-less method to counteract the rotor-movement. In the range of ±100 μm to ±300 μm the resilient bearing mount acts as retainer bearing.

With a linear acceleration of 98.1 m/s^2 (10 g), which equals a fender bender, the flywheel rotor leaves the 100 μm orbit in 1.4 ms. Driving over a speed-bump at 50 km/h results in gyroscopic torque in the range of 100 to 130 Nm. With this torque applied to the vertical flywheel axis the rotor leaves the 100 μm orbit in 0.82 to 0.94 ms. The dead time due to unfortunate rotor position lies in between 0.25 and 0.75 ms depending on the flywheel's state of charge (SOC). Therefore the force rise-time has to be below 1 ms.

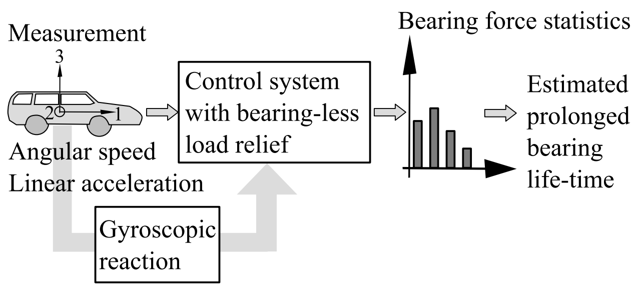

Figure 7. Overview of the system to prolong the rolling elements bearing life

V. RESULTS

By comparing the statistics of the bearing loads an estimated life-time expansion factor of about 10 to 12 is feasible with manageable efforts in power electronics and control. Higher factors are difficult to achieve with just measuring the rotor displacement. One way to improve the system is to measure the vehicles linear acceleration and angular velocity in driving operation and use the measurement signals to calculate a feed forward control as depicted in Fig.7. This possible improvement was not evaluated here and is part of ongoing research activities.

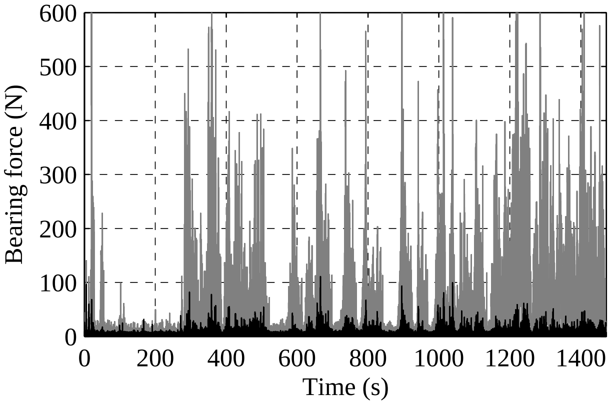

Figure 8. Time dependent combined bearing force (grey) and reduced bearing force (black) from a slice of the test drives (ML320).

With the estimated life-time extension factor of 10 to 12 it could be possible to even use the grease-lubricated bearing and achieve a 5 % drop-out rate in 80 % of the mean car life-time. With the oil-lubricated bearing the life-time could be extended to 15.000 hours which would be suitable for public transportation as well (3 years service interval). Test drive measurements of public transportation systems have not been evaluated yet to estimate the bearing life-time of such a flywheel system.

VI. CONCLUSION AND OUTLOOK

The introduction and motivation brings the reader closer to the special topic of flywheel energy storage systems. The technical outline explains why a brush-less design in combination with a synchronous reluctance machine is the best overall choice for a simple, cost effective and easy to design flywheel system that has the best chances to enter the market of mobile high power energy storage systems. Section IV explains how the simple internal motor topology can be designed to fulfil certain requirements. Further the method of estimating the rolling element bearings fatigue life on test drive measurement data is introduced. Results of the measurement data evaluation is presented by 2 example bearings. The results are shown for two different lubrication methods. The limitations of assessing the bearing life-time is described as well and an estimated achievable bearing life-time extension factor is presented. A possible improvement of the overall system is also mentioned.

REFERENCES

[1] "Schaeffler Group: FAG - Deep grove ball bearings data sheet," 2014, available online at www.schaeffler.com, last visit June 2014.

[2] G. Schweitzer and E. Maslen, Eds., *Magnetic Bearings*. Springer, 2009.

[3] M. Recheis *et al.*, "Improving kinetic energy storage for vehicles through the combination of rolling element and active magnetic bearings," in *39th Annual Conference of IEEE Industrial Electronics Society*, 2013.

[4] A. Rahman *et al.*, "Super high speed electrical machines - summary," in *IEEE Power Engineering Society General Meeting*, 2004.

[5] A. Chiba *et al.*, "Characteristics of a bearingless induction motor," *IEEE Transactions on Magnetics*, vol. 27, no. 6, pp. 5199–5201, 1991.

[6] *Magnetic Bearings and Bearingless Drives*. Elsevier, 2005.

[7] B. Schweighofer, M. Recheis *et al.*, "Rotor losses in a switched reluctance motor - analysis and reduction method," *EPJ Web of Conferences*, vol. 40, no. 17008, 2013.

[8] M. Recheis *et al.*, "Selection of magnetic materials for bearing-less high speed mobile flywheel energy storage systems," in *Proceedings of the 21st Soft Magnetic Material Conference*, 2013.

[9] G. Bischof *et al.*, "Investigation of a passenger car's dynamic response due to a flywheel-based kinetic energy recovery system," *International Journal of Vehicle Mechanics and Mobility: Vehicle System Dynamics*, vol. 52, no. 2, pp. 201–217, 2014.

[10] "UKF hybrid cermaic bearing product datasheets," 2014, available online at www.ukf.de, last visit June 2014.