Comparison of magnetic bearings and hybrid roller bearings in a mobile flywheel energy storage

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Abstract

The comparison of active magnetic bearings and hybrid ceramic roller bearings regarding their application in a mobile flywheel energy storage is discussed in this paper. The consequences of using the two mentioned bearing types are presented concerning dynamic behavior, load capabilities, constructive parameters and reliability. To achieve a design guidance which bearing type to use two flywheel designs with the same power and capacity but different bearing concepts were simulated in a simplified drive cycle to show their different characteristics.

1 Introduction

Over the recent years there has been a noticeable increase of the demands to vehicle propulsion regarding the energy efficiency. Beside the further improvement of the existing combustion engines this caused a significant upturn of hybrid drive systems [1]. The goal in this concept is to combine combustion engines with electric motors in various proportions of power. With these additional degrees of freedom it is possible to use the operating ranges of both machines more efficiently and to recuperate breaking energy. In addition to hybrid drives also full electric drives are deployed.

Especially the recuperation offers a vast potential of energy conservation in the urban traffic towards the use of mechanical breaks. The amount of possible recuperation power is primarily dependent on the maximum charging current of the energy storage. For the currently used batteries such high currents lead to a remarkable reduction of the operational life span [1]. The use of additional buffering capacitors can provide relief nevertheless they need a lot of installation space due to their low energy density [1].

An opportunity to overcome these problems is the use of flywheel energy storages. They consist of an advanced flywheel combined with an electric motor which can operate both as motor and generator. With the application of high strength fiber composite materials it is possible to reach higher energy densities than with super capacitors. Especially the high charging and discharging power as well as the huge number of possible charging cycles make the flywheel storage a viable option as a short time storage for vehicles.

This paper deals with the dimensioning of a flywheel energy storage with special consideration of the bearing concept. The storable energy is linearly dependent on the polar moment of inertia Θ_p and quadratically of the rotational speed ω (equation 1), thus the rotational speed should be as high as possible to reach the highest energy density [2].

$$E_{kin} = \frac{1}{2} J \omega^2 \tag{1}$$

The most suitable bearing concepts for such applications are on the one hand hybrid ceramic roller bearings and on the other active magnetic bearings. These two bearing types have significant differences regarding the stiffness, installation space and friction losses which have great impact on the overall performance of the flywheel system.

The design of such a flywheel is a very complex task and needs therefore exact boundary conditions. This includes the amount of stored energy, the required charging/discharging power and the acceptable weight. To design the bearing correctly, it is inevitably to know the occurring loads over the whole operation range exactly. In a mobile application these are the acceleration forces resulting from the movement of the vehicle which can comprise the stability of the flywheel especially due to the gyroscopic effect

2 Sizing

The maximum energy content of the storage is dependent of the weight of the car in which the storage is deployed and of its requirements to the driving dynamics. The car model for the present design is a compact car with a weight of 1.3 tons and a full electric propulsion. The permanent drive power is assumed to $40\ kW$. The storage should be able to store the energy of a breaking process from the speed of $160\ km/h$ to a full stop. With an assumed charging efficiency of $\eta = 0.8$ this yields a necessary storage size of $300\ Wh$.

3 Flywheel Design

In the following 2 Flywheel designs with similar storage capabilities and charging/discharging power are presented. To reach a very compact overall design a switched reluctance machine with sloped stator poles and a rated power of $38\ kW$ has been developed, which allows a symmetric arrangement of the flywheel between the bearings and a very short shaft by optimally using the available installation space. The flywheel is built up of an aluminum hub with three press-fitted rings of wounded carbon fiber composites. By reducing the aerodynamic drag both structures have to run under vacuum and carry the same Flywheel to achieve a good comparability. Table 1 gives a review about the two designs and lists the most important parameters.

Design A carries hybrid super precision angular contact ball bearings. These bearings are equipped with ceramic rolling elements of silicon nitride and allow very high rotational speeds due to their low friction losses. The axial and radial stiffness is adjustable by preloading. To reach rotational speeds above $30\,000\ min^{-1}$ (lubrication: grease) the shaft diameter cannot exceed $40\ mm$. To achieve a sufficient stiffness the bearings are paired in a tandem arrangement (DT) with extra light preloading. The used bearings (NSK 40 BNR19X) have a bore diameter of $40\ mm$ and can handle rotational speeds up to $41000\ min^{-1}$ [3]. The preloading reduces this value but stays with adequate stiffness above $32000\ min^{-1}$.

Design B is equipped with a fully active magnetic suspension. Both radial bearings are constructed in the heteropolar configuration with a pole area of $12\ cm^2$. With the utilization of Si-alloyed sheets a specific load capacity of $40\ N/cm^2$ is reachable [4][5]. This leads to a maximum bearing force of $480\ N$ for each bearing. The thrust bearing is designed for a higher stiffness because of the sloped airgap of the reluctance machines. They react very sensitive on a changing air gab which can cause problems in their control. The thrust bearing can carry $2500\ N$.

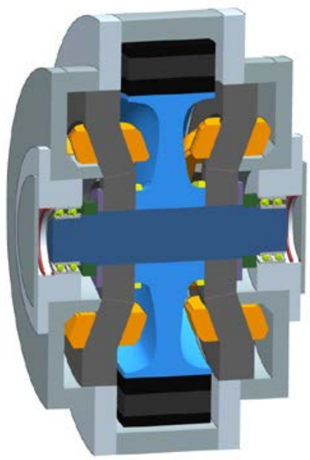
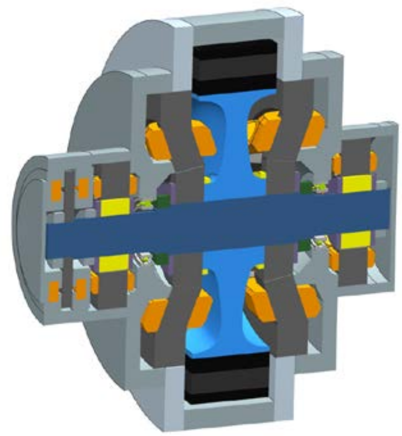
Design A		**Design B**	
			
Max. rotational speed	$32000\ min^{-1}$	Max. rotational speed	$32000\ min^{-1}$
Storeable energy	$374\ Wh$	Storeable energy	$383\ Wh$
Weight	$65\ kg$	Weight	$82\ kg$
Size (lenght/diameter)	$250mm\ x\ 400\ mm$	Size (lenght/diameter)	$370mm\ x\ 400\ mm$
Rigidity radial (each)	$200\ N/\mu m$	Bearing force radial (each)	$480\ N$
Rigidity axial (each)	$40\ N/\mu m$	Bearing force axial	$2500\ N$
Flywheel burst speed	$42000\ min^{-1}$	Flywheel burst speed	$42000\ min^{-1}$

Table 1: Design A (roller bearings), Design B (active magnetic bearings)

4 Rotordynamics

The determination of the critical speeds of a flywheel energy storage has great impact on its performance and stability. The passing of critical speeds is possible for stationary flywheels but it has to be done in a sufficiently short time. This can cause problems for mobile applications without a permanent energy supply. It may occur that the needed energy during the run up can suddenly not be provided or vice versa in case of discharging cannot be wasted. In both scenarios the rotor would stay too long at its critical speed which may cause bearing damages or even the full destruction of the storage. Additionally the uncontrolled self discharge implies such a risk. It is therefore strongly recommended to operate the rotor in the subcritical frequency range in case of mobile application. That leads to a rotor design with high rigidity and low rotor mass. The overcritical operation may also induce further problems arising from the structural damping of the material, especially from the laminated iron cores. This damping effect stimulates the vibration of the rotor beyond the critical speed whereby additional damping in the rotor suspension is needed to ensure a stable rotation [6]. Active magnetic bearings can provide sufficient damping by influencing their controller settings while roller bearings come up with very poor damping factors. It is then necessary to integrate damping elements in the suspension frame [6]. Further it is desirable to use disc-shaped rotors, i.e. with a higher moment of inertia about the rotational axis Θ_p than the moment of inertia about the orthogonal axis Θ_a . This Laval shaft or Jeffcott rotor has the advantage of suppressing the tilting eigenfrequencies due to the effect of gyroscopic moments. These frequencies are shifted to higher rotation speeds or disappear completely for a sufficiently high ratio of $\Theta_p/\Theta_\mathrm{a} > 2$ [6]. In case of a symmetric rotor suspension (design A) the tiling- and bending frequencies are decoupled and can be treated separately ($s_{12} = s_{21} = 0$) [6]. The bending eigenfrequency of a rotor ω_1 can be calculated from the bending stiffness of the shaft s_{11} and the mass of the rotor m. The tilting eigenfrequency can be determined on the same way with the stiffness of the shaft against tilting of the rotor and its moment of inertia Θ_a [6]. The rigidity of the shaft s_{11} is a combination of the bending stiffness c_w and the rigidity of the bearings c_b. The bending stiffness of the shaft c_w is difficult to calculate analytically, that's why a static finite element simulation was carried out which brings more accurate results.

$$\omega_{1,0} = \sqrt{S_{11}/m} \tag{2}$$

$$\omega_{2,0} = \sqrt{S_{22}/\theta_a} \tag{3}$$

$$S_{11} = \frac{c_w + c_b}{c_w \cdot c_b} \tag{4}$$

$$S_{22} = S_{11}(a + l/2)^2 \tag{5}$$

The determined frequencies describe the vibration behavior of the rotor A (roller bearings) at standstill ($\Omega = 0$) $\omega_{1,0}$ and $\omega_{2,0}$, $692\ Hz$ and $652\ Hz$ which show good compliance with the results of the dynamic FEM-simulation. The already mentioned influence of the gyroscopic effect shifts the tilting eigenfrequencies to higher values, where ω_2 is the frequency of the forward whirl (excited by unbalance) and w_4 is the frequency of the backward whirl (excited by foundation vibrations) [4]. Only ω_2 is interesting for the flywheel because an external harmonic excitation can be neglected. The dependency of ω_2 on the rotor speed Ω is shown in equation 9 based on the standstill-frequency $\omega_{2,0}$.

$$\frac{\omega_{2,4}}{\omega_{2,0}} = \frac{J_p\Omega}{2J_a\sqrt{S_{22}/J_a}} \pm \sqrt{\left(\frac{J_p\Omega}{2J_a\sqrt{S_{22}/J_a}}\right)^2 + 1} \tag{6}$$

The forward bending eigenfrequency ω_1 is also shiftet up by the gyroscopic forces (equation 10), but this effect is much smaller than at the tilting eigenfrequencies, the value changes to $693\ Hz$.

$$\omega_1 = \sqrt{\frac{1}{2}\left(\frac{s_{11}}{m} - \frac{s_{22}}{J_p - J_a} \pm \sqrt{\left(\frac{s_{11}}{m} - \frac{s_{22}}{J_p - J_a}\right)^2 + \frac{4(s_{11}s_{22} - s_{12}{}^2)}{m(J_p - J_a)}}\right)} \tag{7}$$

The low stiffness of magnetic bearings leads to a completely different behavior of the rotor. Due to the relatively low bearing forces and the high rigidity of the shaft additional rigid body modes occur at very low frequencies and can be described as translational and angular vibrations of the shaft [4].

Their frequencies depend on the controller settings of the bearings. For a specific point of operation the bearing stiffness can be determined and then the natural frequencies of the elastically supported shaft can be calculated [4]. For the present design B the bearing stiffness can be assumed to $c_b = 500\ N/mm$. The translational frequency is then $34\ Hz$ and the angular one at $42\ Hz$ .They can easily be passed during the run up and the bearing system can provide sufficient damping, however the flywheel cannot operate constantly in the near of these modes. Further eigenfrequencies occur in the range of the natural eigenvalues of the unsupported free-free system [4]. They can adequately be calculated by a finite element simulation and have much higher values in the present case than the rigid body modes (figure 4). The influence of the bearing forces on these modes is pretty small and leads to a slight increase of their frequencies [4], so the free eigenvalues of $1643\ Hz$ (bending of the shaft) and $1050\ Hz$ (tilting of the Flywheel) can be regarded as minimum values. Between these rigid body modes and the free eigenvalues there is no other critical frequency that can be excited by rotor unbalance. By considering the maximum rotational speed of the flywheel no higher eigenvalues have to be calculated.

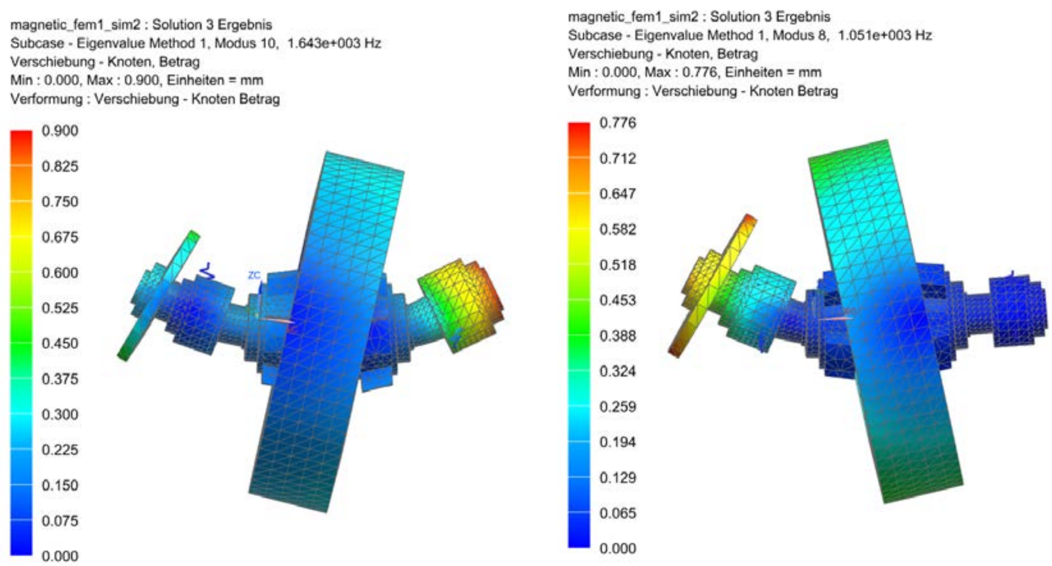


Figure 1: Mode shapes of the free eigenvalues of design B

The calculation of the gyroscopic moments under high rotation speeds requires the consideration of the rotor with magnetic bearings as a continuum. The Laval-model can no longer be used because of the distributed masses of the bearing elements. This is very time-consuming and exact solutions are difficult to find. The gyroscopic effect has a significant impact on the rotor parts with high moments of inertia [4], which are the flywheel and the thrust bearing. These parts have disk-like shapes, where the eigenfrequencies are shifted up under high rotation speeds. The radial bearings just have little influence. Therefore no distinct reduction of the critical speeds can be expected.

5 Dynamic Loads

For a good bearing design it is inevitably to know the acting forces on the rotor during its operation. For a stationary flywheel these are the centrifugal forces resulting from the rotor unbalance.

They are well controllable because they are directly dependent on the rotor speed and can be reduced technically to small values by rotor balancing [6].

However, if the flywheel is mounted in a moving car there are acting additional forces which are induced by the vehicle movement in different driving situations. At first these are acceleration forces due to accelerating, breaking and turning of the car. Beyond this, rotations about the vehicle axes appear which are dependent on the vehicle speed, the profile of the track, the vehicle geometry and the elasticity and damping of the wheel suspension in a complex way. An efficient method to obtain these loads is to simulate the vehicle in different driving situations with a multi body model. Figure 2 shows the applied model and the coordinate directions of the car and the road contact. The car model represents the mass distribution and geometry of a real car by rigid bodies connected with different joints and the appropriate mass density.

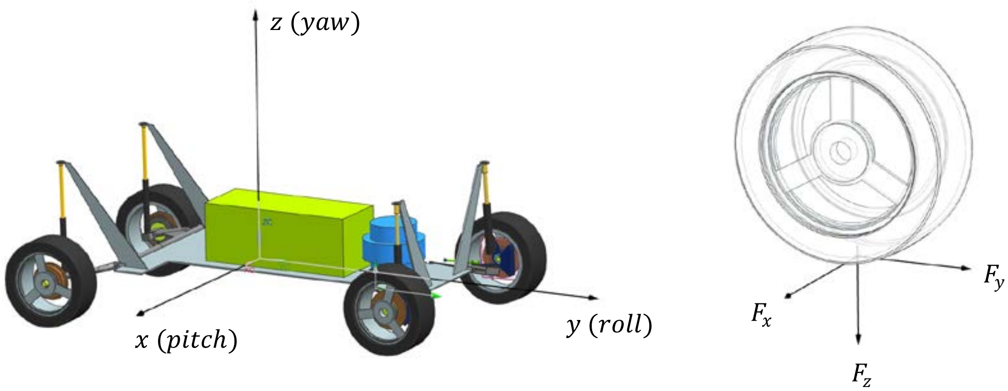


Figure 2: vehicle coordinate system, wheel contact

Data of the vehicle model:

Mass	$1290\ kg$
Gauge	$1400\ mm$
Wheelbase	$2500\ mm$

The acceleration forces F_x and F_y in case of planar vehicle movement are dependent on the friction coefficient (static friction) which is $\mu_r = 0{,}85$ for dry asphalt [7].

$$F_x = F_y = \mu_r \cdot F_z = \mu_r \cdot m \cdot g \quad (8)$$

with

$$F_x = F_y = m \cdot a \quad (9)$$

Equations (8-9) show that horizontal acceleration does not exceed the constant of gravity even in case of $\mu_r = 1$, therefore the maximum horizontal acceleration acting on the flywheel is assumed to $a_h = 10\,m/s^2$. The consideration of the vehicle rotation about the coordinate axes show that the rotation about the z-axis (yaw) is the most frequently rotation with the largest angles due to the steering of the car. Further rotations are enforced by driving on ramps like in parking garage (pitch) or on one-sided obstacles (roll). The angular velocity of the resulting rotation is then dependent of the weight distribution in the car and the vehicle speed while overrunning the obstacles. The occurring accelerations are thereby limited due to the soft wheel suspension and damping of the car structure. A far higher load for the bearings arises from the rotations about the x- and y- axis of the car caused by the gyroscopic moments, which can reach huge values for rotors with high moments of inertia and high rotation speeds. Figure 3 shows the direction of the gyroscopic moment M_y as a result of the tilting of the whole rotor about the x-axis. If the angular velocity $\dot{\varphi}_x$ is known, M_y can be calculated with equation 10 [6].

$$M_y = J_p \Omega \dot{\varphi}_x - J_a \ddot{\varphi}_y \tag{10}$$

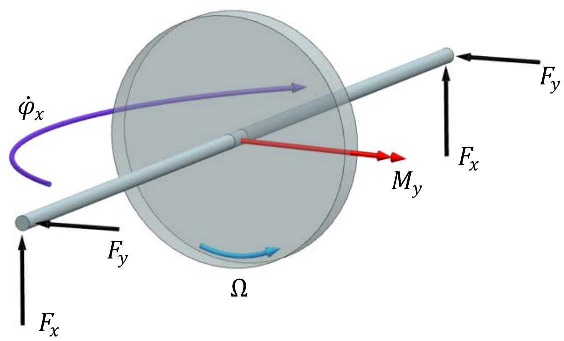


Figure 3: Direction of the gyroscopic moment

To reduce these gyroscopic loads as much as possible the rotor axis of the flywheel can be arranged vertically in the car to at least eliminate the influence of the steering movements. Numerous simulations have been made with different track geometries and speeds to find the most incriminating driving situation for the flywheel. There have been considered just cases which exclude the misuse of the car as well as collisions with other cars or objects. The drive on a ramp like in a parking garage showed up as the most demanding situation for the bearings. Figure 4 shows the characteristics of the the vertical acceleration and the angular velocity of a car on a run over a ramp with a pitch angle of 15° with a speed of 25 km/h.

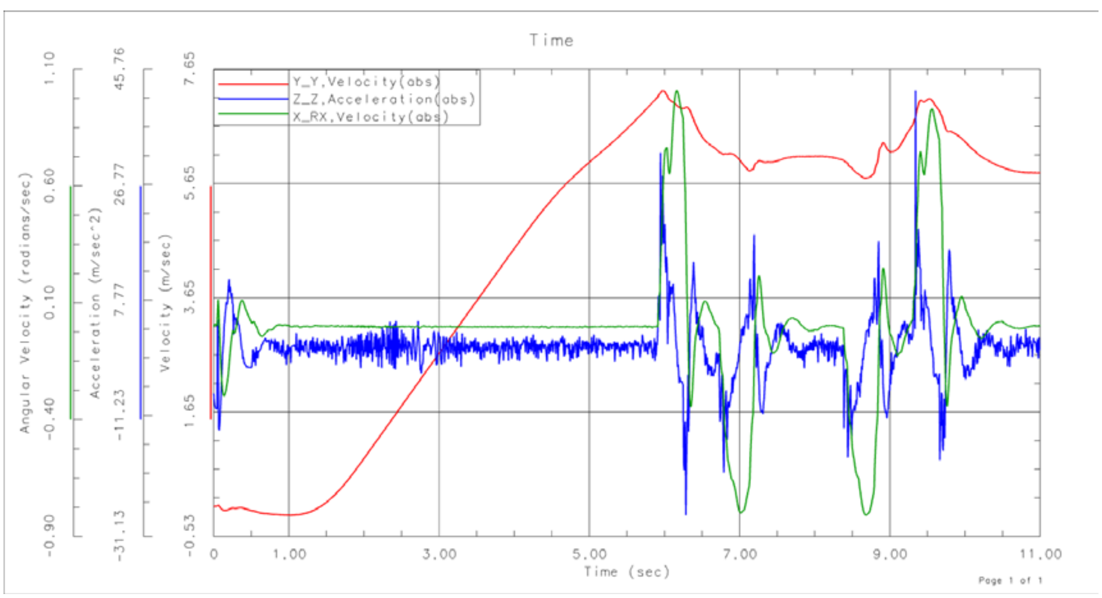


Figure 4: Speed, vertical acceleration and angular velocity about the x-axis

It can be seen that the vertical acceleration reaches values of $30\ m/s^2$ which can be tolerated by the axial bearings of both designs. The rotation about the x-axis of the car reaches values of $0{,}7 rad/s$. The consequence of this for design A is a radial bearing force of $2000\ N$ which is far smaller than the maximum bearing force of the paired angular contact bearings ($> 15\ kN$) therefore design A is able to resist these forces. The magnetic bearings of design B get a radial force of $1700\ N$ that leads to a massive overloading. The consequence here would be the contact of the rotor and the touch-down bearings which must be prevented.

A possibility to reduce the impact of the gyroscopic moments is the use of a suspension frame that enables the rotation of the whole flywheel storage about the x- and y-axis of the car. The rotation angles stay small and the car always returns to the previous position, that's why such a gyroscopic suspension frame can be realized with limited effort.

6 Conclusion

The analysis shows that a flywheel energy storage for mobile applications up to a storage size of $400\ Wh$ can be realized both with roller bearings and active magnetic bearings. The two bearing types have very different effects on the performance of the whole system. Active magnetic bearings show a better behavior regarding the rotor dynamics due to their variable stiffness and damping parameters and allow very high rotation speeds. Hybrid roller bearings instead can only be applied for smaller shaft diameters because of their limited rolling element velocity, whereby the mentioned subcritical operation range is limited due to the lack of rigidity.

However the sensitivity to external disturbance forces is a problem for magnetic bearings based on their relatively low bearing forces. This requires a massive oversizing or the installation of an additional gyroscopic suspension frame whereas roller bearings provide more stability to the system.

The losses are in both cases dependent of many influencing parameters which can just be determined exactly in the operation of the systems. Roller bearings have a strong dependency of their friction losses on the rotational speed. Magnetic bearings instead have more constant losses over the whole speed range of the rotor, that's why especially at high rotor speeds magnetic bearings can be expected as more efficiently. A significant disadvantage of magnetic bearings is that they have to be active at all times when the vehicle is moved, i.e. also in driving situations where the flywheel is not used (e.g. at long distance highway drives), since a missing bearing force would lead to uncontrolled movements of the shaft in the touch-down bearings. Additionally the standby behavior is worse because the low rigid body modes have to be passed at every run-up. Hence hybrid roller bearings show a clearly more robust behavior for a flywheel energy storage in road vehicles.

Finally the high constructional effort and the linked increase of weight have to be considered when using magnetic bearings. Also the costs of the bearing itself, the required power electronics and the position sensors are significantly higher than that of roller bearings.

Due to the different features of both bearing types also different fields of application can be mentioned. While active magnetic bearings with their superior dynamic behavior, better endurance and lower friction losses are suitable for heavy duty applications with predictable driving cycles like busses or delivery vans. Hybrid ceramic roller bearings instead are a good choice for the mass production of passenger cars based on their low production costs, more compact design, lower weight and their high robustness.

7 Acknowledgements

This work was supported by the European Social Fund, the Sächsische Aufbank SAB and the research network CITE of the Chemnitz University of Technology.

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