A novel rolling element back-up bearing for a 9 t rotor application

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Abstract—**The paper outlines the development of a novel back-up bearing unit which complements Siemens' new magnetic bearings for an electric motor with a 9 t rotor. The back-up bearing utilizes rolling bearing elements. Principles for the rotor supported by the back-up bearings are derived and used to design the novel, so-called "Smiley" design. Dynamic simulations are presented predicting that the system fulfils the requirements. This is proven by comparison with experimental data.**

I. Introduction

A. Background

Active magnetic bearings are a favorable bearing concept for applications with high load, high speed and high efficiency requirements. Correspondingly, Siemens has developed new magnetic bearings based on standard drive technology for its new 23 MW electric motor [1] illustrated in figure 1. For safety reasons, active magnetic bearings require back-up bearings. During landing of a rotor on the back-up bearing regime, the rotor can develop undesired backward whirls, resulting in unacceptable high and hence destructive forces, if the back-up bearing design is not appropriate.

The development of whirls is a result of friction conditions in the system, and it is widely recognized, that a rolling element back-up bearing might provide advantageous friction conditions [2]. Schaeffler Technologies GmbH & Co. KG with its brands INA and FAG has a vast knowledge in rolling bearings for a huge variety of applications, so it was only natural that it would accept the challenge to design a rolling element back-up bearing.

B. Specific operation conditions

The operation of back-up bearings differs from that of normal rolling element bearings, e.g.

- After the active magnetic bearings are switched off, the rotor lands on the back-up bearings, this causes an impact on the back-up bearings. The impact leads to both, high forces and consequently high angular acceleration of the bearing inner rings. This back-up bearing acceleration is larger than normal for rolling element bearing applications.
- Back-up bearings operate for only a short period of time at high speed, therefore the dimensioning of the back-up bearings doesn't rely on the lifetime calculation of standardized rolling element bearings.

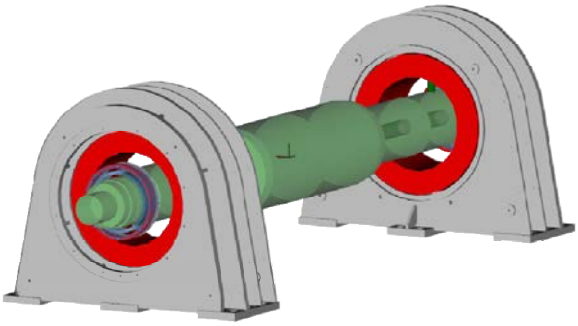

Figure 1. Simplified 3D model of an electric 23 MW motor with magnetic levitated shaft.

Other specific conditions are:

- The movement of the rotor in back-up bearings can be chaotic [3], which makes a prediction potentially difficult.
- The rotor might develop a backward whirl in the system, which must be avoided by the bearing design.
- The specific application restricts design possibilities.

This paper focuses on two situations the back-up bearing has to resist:

- impact with high forces and
- dynamics movements during run down of the rotor

by elementary and numerical investigations.

II. Design Principles

As a first stage in the design process, elementary investigations were performed to find principles fulfilling the above requirements, i.e. resistance against impact with high forces and dynamics during run down.

A. Initial landing impact

The impact caused by the landing rotor once it contacts the back-up bearing is the most relevant load condition the back-up bearing has to resist. For the draft design of the bearing, the system is simplified to a linear one. With the assumption of a constant back-up bearing stiffness, the contact forces dependent on the radial deflection are determined. The integral of the contact force over the deflection is the contact energy. The maximum deflection and maximum contact force are determined by the equilibrium of the initially potential energy, which the landing shaft converts to translational kinetic energy, and the contact energy. This is illustrated in figure 2.

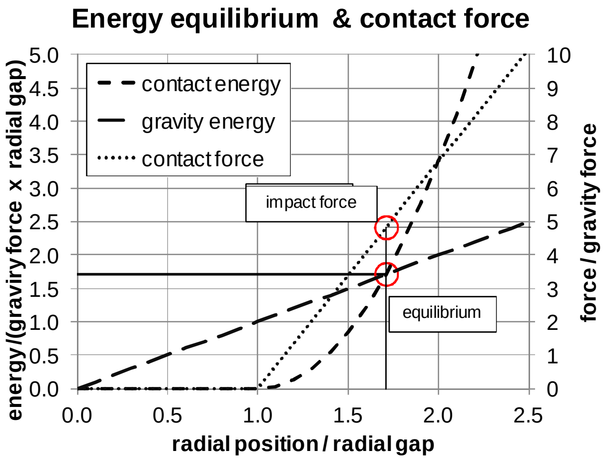

Figure 2. Illustration of the energy equilibrium and the resulting contact force as shock factor. When the gap is closed, the contact force rises linearly with radial position. The contact energy is the integral of the contact force over the radial position. Values normalized by the radial gap between rotor and bearing and by the weight.

So the contact force depends on the initial potential energy of the shaft, and the contact stiffness.

A variation of the contact stiffness influences the contact force: the lower the contact stiffness, the lower the contact force, hence a low stiffness is desirable. On the other hand, the lower the contact stiffness, the higher the deflection. So the specified maximum deflection describes a lower limit to the contact stiffness.

The assumption of a constant stiffness is not valid for back-up bearings with rolling elements, but the procedure can also be used for nonlinear systems.

B. *Run down the back-up bearings*

During run down the desired position of the shaft is as close as possible to the static equilibrium position. A linearization of the stiffness at this equilibrium position can be used to describe the system. The linear contact force in radial direction is described by

$$\left|F(r)\right| = \begin{cases} 0 & r < 0 \\ c(r - r_0) & r \geq r_0 \end{cases} \tag{1}$$

c *radial stiffness*
F *radial force*
r *radial position*
r_0 *radial gap.*

The linearization of the system at the equilibrium is

$$F(x,y) \approx \begin{pmatrix} 0 \\ F_G \end{pmatrix} + \begin{pmatrix} \left.\dfrac{\partial F(r)}{\partial x}\right|_{\substack{x=0 \\ y=-r_0-\frac{F_G}{c}}} & \left.\dfrac{\partial F(r)}{\partial y}\right|_{\substack{x=0 \\ y=-r_0-\frac{F_G}{c}}} \end{pmatrix} \begin{pmatrix} x \\ y \end{pmatrix} \tag{2}$$

x *horizontal coordinate direction*
y *vertical coordinate direction*
F_G *gravity force*
others as above.

This leads to a system with an orthotropic support with the equations for the stiffness in horizontal direction

$$c_x = \frac{c\,F_G}{F_G + c\,r_0} \tag{3}$$

c_x *horizontal stiffness*
others as above

and in vertical direction

$$c_y = c \tag{4}$$

c_y *vertical stiffness*
others as above.

With this linearization the critical frequencies of the rotor lying in the back-up bearing can be predicted. Figure 3 shows the natural frequencies of the shaft depending on the stiffness of the back-up bearing.

Depending on the obtained natural frequencies and the operating conditions, the contact stiffness can now be adopted to the application.

Finally, the stability of an overcritical, orthotropic system can be considered as a design criterion according to Gasch [3]. The author introduces an orthotropic parameter μ_L which is the ratio of radial stiffness difference over radial stiffness sum

$$\mu_L = \frac{c^2\,r_0}{2\,c\,F_G + c^2\,r_0} \tag{5}$$

μ_L *orthotropic parameter*
others as above.

The orthotropic parameter showed to be not critical for this system.

III. Novel Back-up Bearing Design

The Schaeffler back-up bearing design is a further development of the published design from Schmied and Pradetto [8], consisting of two angular contact rolling element bearings combined as face-to-face arrangement. The new bearings are fully complemented with ceramic rolling elements.

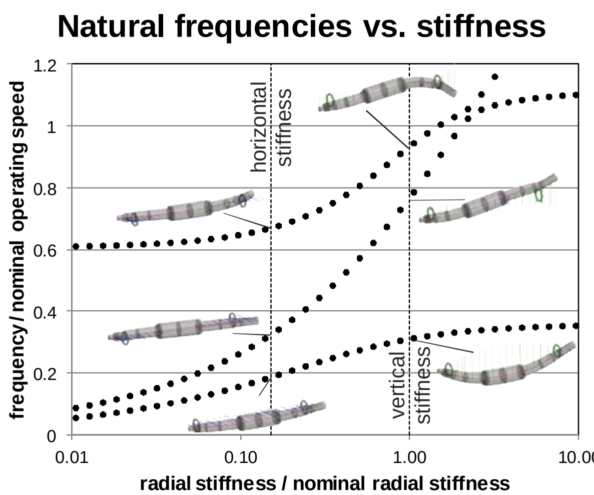

Figure 3. Natural frequencies of the shaft depending on the radial stiffness of the back-up bearing. The horizontal und vertical stiffness calculated is marked. Stiffness normalized by nominal stiffness and frequency normalised by nominal operating speed.

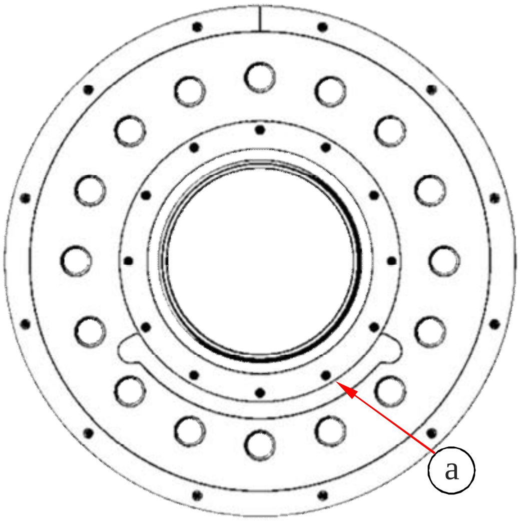

Figure 4. Back-up bearing housing with “Smiley” design at position a) to reduce the stiffness in the direction of the main load and to disturb the symmetry of the back-up bearing.

To resist the higher impact loads the basic design was improved. As shown for the linear system, a stiffness reduction leads to lower impact forces. Additionally, Simon showed [9] for a pendulum type centrifuge that a polygonal back-up bearing reduces maximum contact forces at the back-up bearing during run up. Helfert [6] tried to adopt this effect with two misaligned rolling element bearings. To combine both effects, the reduced stiffness in the main load direction and the non symmetric properties, the housing was modified by introducing a cut-out, which is due to its shap called “Smiley” design (figure 4) which reduces the stiffness in the direction of the gravity.

The new design reduces the impact and overcomes the rotational symmetry of the back-up bearing thus reducing the probability of whirl.

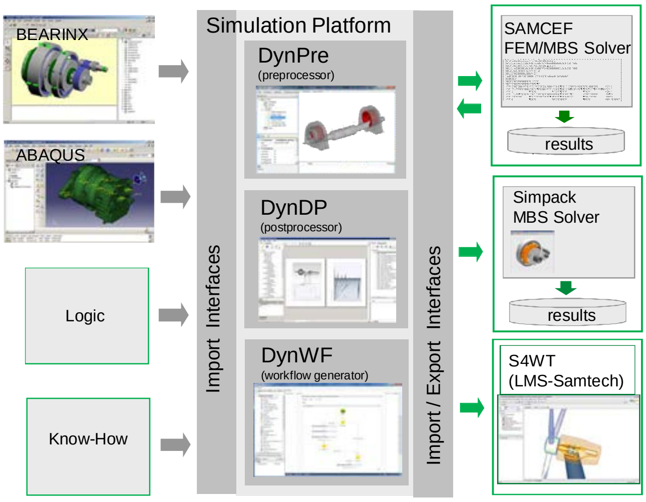

Figure 5. Scheme of the Schaeffler Simulation Platform.

IV. Simulation

A. Motivation

The above principles explain the smiley design for back-up bearing systems but are too coarse to dimension it. For this task the wide range of simulation methods present at Schaeffler for roller bearing development were utilized. These simulation methods enable investigations from the microscopic level of the roller raceway contact to complex system simulations of full drive trains including control units [7].

B. Schaeffler Tools

Finite Element Analysis was performed to analyze the stress and strain due to the impact to the back-up bearings in its pedestal. Thereby, the simulation gained accuracy from the available ABAQUS roller element developed by Schaeffler, enabling exact consideration of each individual roller [4] [5].

To investigate the behavior of the full rotor due to the bearings during run down, dynamic simulation had to be carried out. For this class of problem Schaeffler developed its “Schaeffler Simulation Platform”. The platform focuses on modeling quality and efficiency by automation of frequently used process sequences for drive trains. As figure 5 illustrates, the simulation platform collects different model definitions e.g. from BEARINX®, Schaeffler’s tool for bearing analysis, or from FEA codes. It even allows the import of customer models in collaborative projects. The collected data are then used to create and run the dynamic simulation models in various commercial solvers, such as the MBS code Simpack and the MBS/FEA code SAMCEF Mecano. The Schaeffler Simulation Platform is completed by features allowing efficient and consistent post-processing and workflow control. Due to all the features it became apparent that the Schaeffler Simulation Platform enables application engineers to access the complex field of dynamic simulation.

C. Run down simulation

For the run down simulation of the rotor in the system as displayed in figure 6, a co-simulation combining BEARINX® with an elastic multibody simulation was performed, with all subsystems verified in the Schaeffler Simulation Platform

The simulation of the back-up bearing behavior was performed in BEARINX® using directly its files.

The results of the FEA for the housing including the smiley design were condensed to a non-linear spring damper

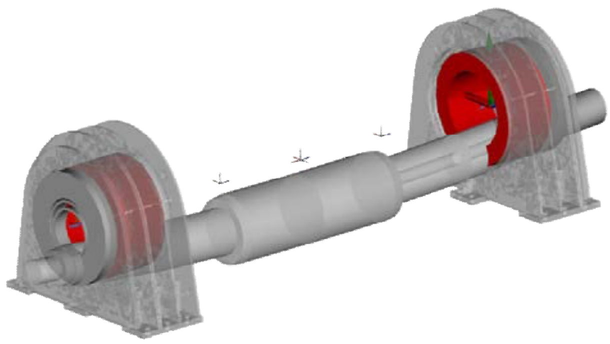

Figure 6. Model of the electric motor consisting of the rotor, back-up bearings and elastic pedestals.The deformation of the shaft and the bearings are scaled 1000 times to visualize the deformations.

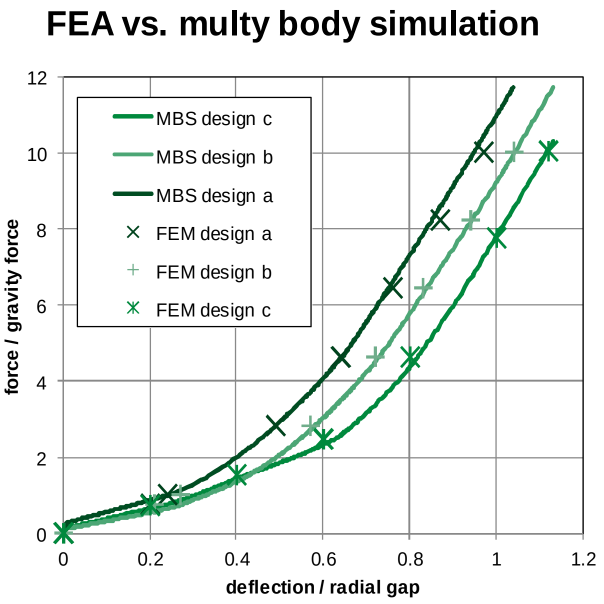

Figure 7. Comparison of defelection in smiley direction between FEM and multi body simulation for different designs with same parameters. The force is normalized by gravity, and the deflection by the radial air gap.

system. The validity of the approach was checked by another investigation in the Schaeffler Simulation Platform: a submodel containing the FEA model of the housing, the back-up bearing and their non-linear linkage was compared to back-up bearings and the spring damper system. Figure 7 illustrates for the vertical direction, that both approaches show identical properties which enable parameter studies with the spring damper model.

The model of the shaft utilizes Siemens' experience in producing and simulating rotors for electric motors. Geometry and material properties of the rotor were imported using the Schaeffler Simulation Platform. The model was validated by a comparison between natural frequencies of the free shaft determined with the Schaeffler Simulation Platform and those determined by Siemens.

The bearing pedestals are modeled with superelements. This modeling incorporates the experience of the influence of elastic structures on the load on bearings.

D. *Sample Results*

Figure 8 and figure 10 show results of the simulation in waterfall diagrams. The frequencies that are excited at run down correspond to the frequencies predicted with the linearization illustrated in figure 3. The comparison between the horizontal and the vertical displacements shows that the resonance frequencies differ e.g. line "A" in figure 8 (approx. 0.2) and figure 10 (0.3). This is due the orthotropic effects of the design and shaft position.

E. *Experiment*

To qualify the application Siemens and Schaeffler performed tests of the motor landings above nominal speed, see [2] for a description of the procedure of the tests.

Figures 9 and 11 show results of the measurements as waterfall diagrams. The frequencies excited during the run down correspond to the frequencies in the simulation. The difference between resonance frequencies in the horizontal and vertical directions differs as predicted in the simulations.

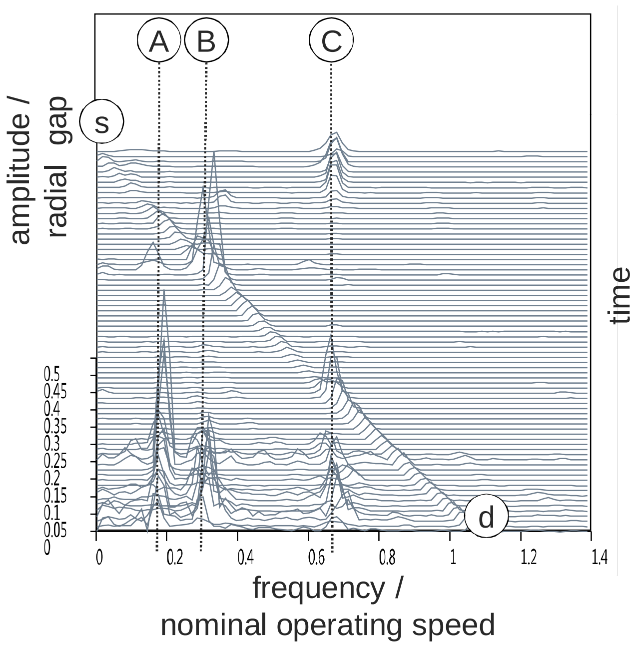

Figure 8. Simulation results: Waterfall diagram of the horizontal rotor displacement in the magnetic bearing. The rotor is dropped at d and stopped at s. A, B and C are the predicted natural frequencies (cf. fig 3). The frequency is normalized by the nominal operating speed, the amplitude by the radial gap.

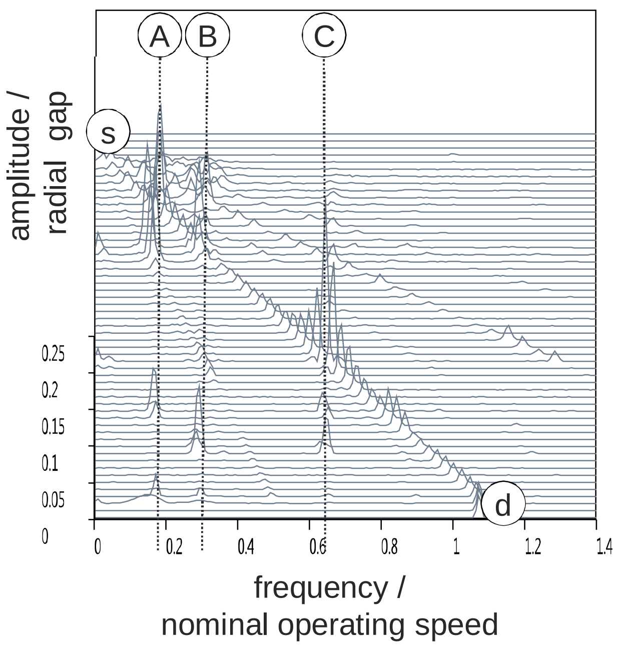

Figure 9. Experimental results: Waterfall diagram of the horizontal rotor displacement in the magnetic bearing. The rotor is dropped at d and stopped at s. A, B and C are the predicted natural frequencies (cf. fig 3). The frequency is normalized by the nominal operating speed, the amplitude by the radial gap.

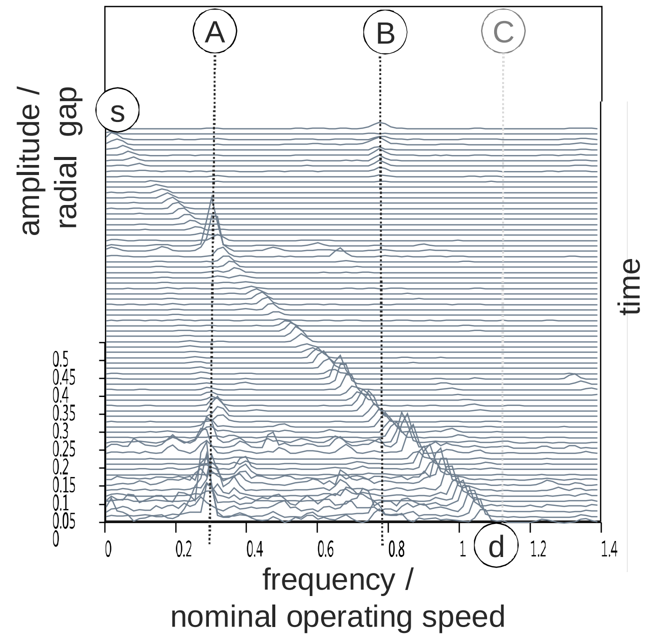

Figure 10. Simulation results: Waterfall diagram of the vertical rotor displacement in the magnetic bearing. The rotor is dropped at d and stopped at s. A, B and C are the predicted natural frequencies (cf. fig 3). The frequency is normalized by the nominal operating speed, the amplitude by the radial gap.

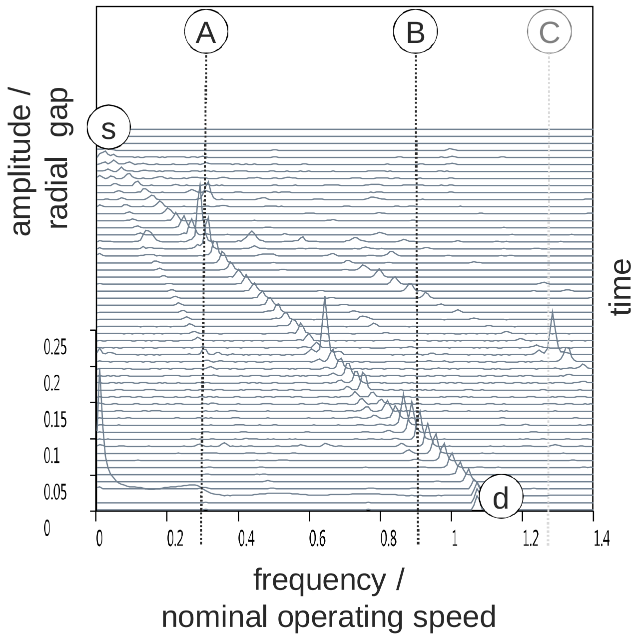

Figure 11. Experimental results: Waterfall diagram of the vertical rotor displacement in the magnetic bearing. The rotor is dropped at d and stopped at s. A, B and C are the predicted natural frequencies (cf. fig 3). The frequency is normalized by the nominal operating speed, the amplitude by the radial gap.

Hence the orthotropic effects are confirmed.

V. Summary and Conclusion

For a novel rolling element back-up bearing used for a rotor in a magnetic bearing application, design principles were presented: Firstly, a method to estimate the required vertical stiffness of the system was derived. Secondly a method to calculate the horizontal stiffness, which affects the natural frequencies for the system of the back-up bearings supporting the shaft during run down was presented.

Furthermore a detailed numerical approach was shown. Comparisons with experimental data showed good agreement, and proved the theoretical approach and the simulations.

VI. Acknowledgement

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