INVESTIGATIONS OF THE BEHAVIOUR OF A MAGNETICALLY SUSPENDED ROTOR DURING CONTACT WITH RETAINER BEARINGS

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ABSTRACT
Retainer bearings play an importand role in equipping machines with Active Magnetic Bearings (AMBs).
They shall guarantee the safety of a machine under possibly all loading conditions, a demand, which may especially become apparent in the field of aircraft applications.
Situations of AMB power loss were already main object of research work, but investigations on AMBs and retainer bearings operating in a load sharing mode are widely missing.
The concept of a test rig is presented, which shall provide the possibility to experimentally study various operating conditions for retainer bearings in AMBs, including the mentioned load sharing mode.
Results gained from experiments with this rig, shall be used as a basis for a software tool, being develloped to analyze a variety of different system designs.
A first approach of modelling dynamics during rotor drop due to AMB power loss is presented and discussed.

INTRODUCTION
Active Magnetic Bearings (AMBs) are of increasing relevancy in many rotor dynamic applications.
Due to the principle of operation they have several advantages compared to traditional support mechanisms. There is no need of lubrication, frictional losses are neglectable, stiffness and damping are controlable.
With increasing performance and reliability of AMBs due to technological improvements (materials used, strategies of controller, electronic components) they are more and more accepted.
Ensured reliability of the system especially under critical conditions of operation will be of essential influence on the successful transfer of this technology to new fields of application.
Integral parts of AMBs are catcher bearings, which shall inhibit a contact between rotor and non rotating parts (stator of AMB, housing, etc) generally by means of a mechanical stop.
The catcher bearings regularly take over this task during all offline time of the machine, in case of AMB power loss or failure, or even if the load capacity of the AMB is exceeded.
Depending on the application, individually different conditions may determine the applicability of a specific design of a catcher bearing in conjunction with its AMB. These conditions can be for instance the demanded momentary or long-time high reliability, intervals of maintenance, the range of operational temperature or no available lubrication.
In stationary machinery a magnetically suspended spinning rotor may eventually be brought to standstill within short time. In this case catcher bearings provide for a safe coast down.
In some applications, e.g. aircraft turbines, the rotor may continue spinnning due to the windmilling effect, even after deenergizing the machine on AMB failure. In those cases AMBs have to comply with stringent requirements regarding their saftey of operation, whereas the capabilities of their catcher bearings might be of specific interest.
In this regard it must also be examined how AMB and catcher bearing behave in load sharing operation, a condition that is expected to occur in aircraft engines during manoeuvre, landings or unbalance due to blade loss of a turbine.
Cases of rotor to bearing contact may in general be characterized by type and duration of the forces acting on rotor, AMB and catcher bearing, as well as by the condition given by the AMB (e.g. operating or deactivated) .

Frequently quoted work on this topic, like [4], investigates the dynamic behaviour of the rotor with regard to different designs (lubricated/unlubricated journal bearings, rolling bearings, different materials) of the catcher bearing in the case the AMB is deenergized. A specific behaviour, the whirl motion for instance, and the causes for its development is analysed in [1].
To our knowledge, investigations on the (medium-term) interaction of AMB and catcher bearing during overload situations is widely missing. Only [5] presented a design (ZCAB) of a catcher bearing, which was, besides common drop-tests, tested in load sharing function for short term, shock like overloads of the AMB.
The aim of our work is to gain knowledge on the dynamic behaviour of rotors in case of contacts with its catcher bearings especially regarding those situations where the operating AMB is supported by the catcher bearing in a load sharing mode.
For this purpose a numerical tool is developed, which allows the rotor-dynamic analysis of the interesting load cases. The tool is complemented by building up a test rig, which will enable to validate the results from the numerical calculations.
Whereas the tool is applicable widley independent from a specific design of a system (e.g. single/multishaft machines, geometrie of rotor, number of AMBs, catcher bearings, etc.), the test rig was conceived to simulate load conditions typically expected to occur in AMBs used in aircraft engines.
The loads in these cases will lead to

1. permanently acting forces resulting from unbalance of the rotor due to blade-loss of the turbine,
2. transient forces of short-term and midium-term duration (some tens of seconds), resulting from base excitation due to landings or manoeuvres [6],
3. permanent acting forces via base excitation due to a neighbour engine running or windmilling with unbalance.

All cases mentioned above are possible with both intact or partly or completely deactivated AMB respectively, and therefore subject of our planned experiments besides basic drop-tests.

Measurement Concept

The experimental setup shown in Figure 1 will provide as a basis for the planned investigations.
A short description of its components is given later on.
Transient or permanent forces with typical amplitudes and temporal progression can be generated by an electromagnetic actuator (4) and unbalance weights mounted on a disc (5) forcing the rotor (1) to contacts with the instrumented catcher bearing (7).
To judge a specific catcher bearing design regarding its operational functionality in conjunction with that of the AMBs, reactional forces of the AMBs (2), (6) and of the catcher bearing (7) are measured besides recording the orbit of the rotor.
Results from these experiments for various constellations of design parameters like type and geometrie of catcher bearings used, or stiffenss and damping of their support, shall enable to verify the function and the underlying models of the simulation tool.

Components of Test Rig

The rotor (1), also refer to Figure 2, has an overall length of approx. 700 mm and a mass of approx. 30 kg.
The frequency of its first bending mode (free-free) is calculated to be above 350 Hz. which is 1.5 times the exciting frequency of approx. 230 Hz for the planned maximum operational speed of 14 000 rpm.
Mounted on the rotor-shaft are the rotor of an asynchronuous motor and sleeves (8), (11) und (10), carrying the lamination sheets of the electromagnetic actuator and the two AMBs.
The shaft ends can be furnished with running sleeves of different diameter to match various catcher bearing geometries.

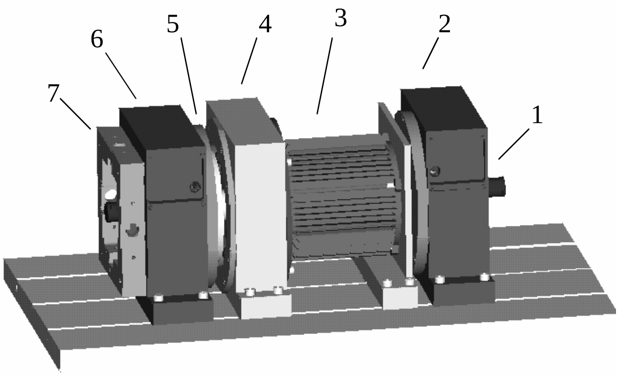

FIGURE 1: Components of the Test Rig

Two radial AMBs (2) and (6), each with a static load capacity of approx. 800 N at 1.0 mm air gap, provide for approx. 5 times the static force due to gravity acting on the rotor.
The position of the rotor will at first be controlled by a PID type digital controller, which evaluates the signals from the displacement sensors.
Measuring the magnetic flux by means of Hall-sensors will ease the precise acquisition of the interesting force information for the accompanying numerical analysis.

An integrated asynchronuous drive (3) controlled by a frequency converter is used for realizing rotational speeds up to 14000 rpm. Rotational speed and angle of rotation are measured optically (referring to [4], the angular location of unbalance is of strong influence on the post drop motion of the rotor).

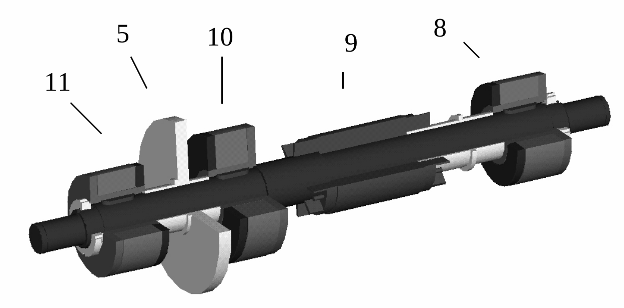

FIGURE 2: Rotor

An electromagnetic actuator (4) will be used to pull the rotor against the reactional force of the AMB in an excentrical position provoking a contact with the retainer bearing (7).
The static force producible by the actuator will reach approx. 1500N, which is nearly twice that of each AMB and approximately 10 times the reactional force of each AMB due to gravity.
By individually limiting the maximum operating current of the AMBs, the ratio between the force generated by the actuator and AMB respectively, can be adjusted in support of that of the actuator.
Short, impulse like as well as permanent forces can be generated with predefined amplitude and temporal progression in two degrees of freedom, whereas flux information from Hall-sensors is used to control the accurate dosage of the initiated force.
A disc (5) equipable with excentrically mounted masses will be firstly used to generate unbalance in one plane.
The electromagnetic actuator may be used in parallel to introduce an additional transient or permanent force on the rotor (e.g. manoeuvreing aircraft with unbalanced running engine).
In a possible upgrade of the experimental setup it is planned to even simulate unbalance forces with this actuator.
The catcher bearing (7) consists of a mechanism to incorporate rolling bearings of various types and sizes.
Bearings with an outer diameter in the range of approx. 60-100 mm and inner diameter of 45-70 mm may be fit directly or using an adapter.

The draft in Figure 3 shows the arrangement of the catcher bearing and its principle of operation in case of a contact.
Normal and frictional component of the force generate a displacement of the elastically suspended inner module carrying a rolling bearing.
A specifc attachment of the four leaf springs d1-d4 causes them to be loaded almost only in their bending line. The force acting on the inner module can therefore be determined by separately measuring the vertical and horicontal displacement.

Peak levels of initial contact forces durig rotor drop may reach values of e.g. 12.5 times the bearing static load, as shown in [3].

The leaf springs used in our arrangement were firstly layed out to provide for an overall stiffness of $\cdot\ 10^7$N/m. This will lead to a maximum displacement of 100 um of the inner module, assuming a static force of 3000N (20 times the bearing location static load).
The overall eccentricity of the rotor is limited by hard mounted safety bearings to a maximum of 200 um, including 100 um clearance of the soft mounted rolling bearing.

Optical sensors at locations s1-s8 measure the displacement of the inner module with a resolution of 1 um, according to differences of 30 N in force levels for the chosen dimension of the stiffness.

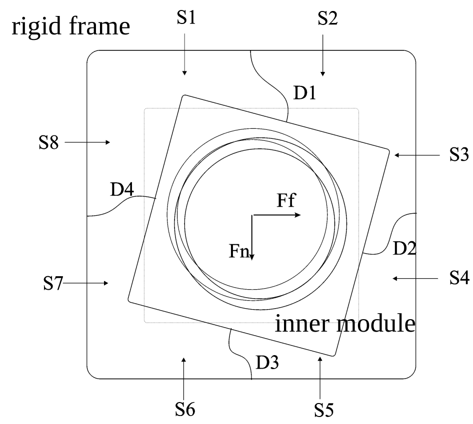

FIGURE 3: Catcher Bearing -

Conditional upon the method used, contact forces between rotor and bearing inner race can not be seized in their entire dynamic range.
The frequence content of the sensed displacement signals is altered due to the dynamic behaviour of the rolling bearing as well as to the low pass characteristic of the mass of the inner module.
Additional acceleration sensors may thus be appropriate to increase the accuracy of measuring fast transients.

Besides the stiffness of the inner module support, the influence of damping of its movements will be investigated regarding the bearing forces and the behaviour of the rotor. The availability of (external) damping may be of great importance to face e.g. the potentionally destructive whirl motion [2].

There are several possible ways to introduce damping in our arrangement. Firstly, instead of using solid leaf springs D1-D4, laminated versions consisting of up to 10 sheets (at unchanged stiffness) can provide for some

inner damping. Indeed, it is difficult to determine the degree of damping in advance in this case.
Additional damping by e.g. corrugated ribbon (between bearing outer race and inner module) or by external dampers, mounted between inner module and rigid frame may be used if required.
In all cases the damping characteristics should remain unchanged regarding e.g. temperature during charging time.

Modelling
To make a transfer of the results received from the investigation explained above on systems unscalable to the presented test-rig possible, it is necessary to have a software tool capable to model those systems and predict their performance. This software tool is develloped in parallel to building up the test rig, which will be used to validate the simulation results.

In the following a first approach of predicting the behaviour of a rotor falling into retainer bearings without AMB-suspension will be discussed on the basis of experimental results from Fumagalli [7].
For his experiments Fumagalli used a rotor with a weight of 3.36 kg, a length of 326 mm and a first elastic eigenfrequency of approximatley 1600 Hz for free-free constellation. The finite element model of the rotor consists of 88 degree of freedoms and was transformed into modal coordinates applying an external modal damping of 1%. The model was than reduced to 16 modes, whereby the decision wether a mode is being kept or deleted, is based on the Controllability and Observability Gramians of the rotor, see [8] for a detailed description. Both retainer bearings, ball bearings and positioned at both ends of the rotor, are modeled with one degree of freedom representing the rotational movement of the inner ring, and based on the assumption that there is no sliding between the inner ring, bearing ball and outer ring, only pure rolling. After Fumagalli this assumption is not valid for the begin of a touchdown event.

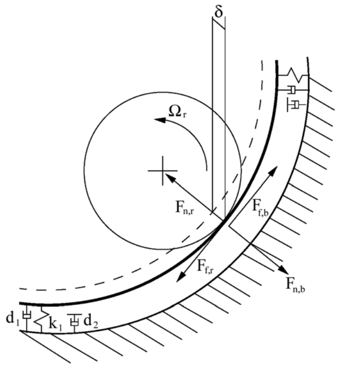

FIGURE 4: Contact Model, inspired by [9]

The contact force was determined with a method inspired by Kirk [9], Figure 5 illustrades this method.
The normal contact force is a function of the deformation of the bearing δ and the nonlinear force-deformation-behaviour of the bearing provided by the bearing manufacturer, this is illustrated by k_1.
The deformation is obtained by the following equation in case of a contact:

$$\delta = |x_r - x_b| - \varepsilon\,, \qquad \textbf{(1)}$$

with the displacement of the rotor x_r, the translatorical displacement of the bearing x_b, and the airgap ε between the bearing inner ring and the rotor when centered in the retainer bearing. x_b is eqal to the displacement of the bearing housing to which it is attached at the connecting degree of freedoms, because the model of the bearing only considers one rotational degree of freedom but no translatorical one.
But of course there will be some damping mechanism acting during the contact represented by the nonlinear damping d_1.
Because of the increased energy dissipation during impacts with higher impact velocities compared to that regarding lower velocities, a second nonlinear damping d_2 was necessary, acting when the penetration transcends a value δ_c.
This results in the following law for the normal contact force:

$$N = \begin{cases} k\delta^n + d_1\delta^n\dot{\delta}, & \text{for } \delta_c > \delta \geq 0 \\ k\delta^n + d_1\delta^n\dot{\delta} + d_2(\delta - \delta_c)^n\dot{\delta}, & \text{for } \delta \geq \delta_c \end{cases} \qquad \textbf{(2)}$$

$$F_n = \begin{cases} N, & \text{for } \delta > 0 \text{ and } N > 0 \\ 0, & \text{for } \delta < 0 \text{ or } N < 0 \end{cases} \qquad \textbf{(3)}$$

where $k\delta^n$ was fitted to the force deformation curve of the bearing, while d_1 and d_2 were determined by a parameter-study regarding rebound heigths after a rotor-drop. The normal force is acting on the rotor as well as on the bearing, but because the bearing model considers only a rotational degree of freedom this force is normally passed through to the housing, which has been neglected in this case.
Beside the normal Force, the friction force acts on the rotor and the inner ring of the bearing, determined by

$$F_r = \mu F_n \qquad \textbf{(4)}$$

μ was set to such a value, that the acceleration of the inner ring of the bearing is equal to the measured rate by Fumagalli.
According to [10], the frictional moment in the bearing itself, can be split into a part proportional to the bearing

load and a part proportional to the rotational speed of the bearing.
A further assumption is, that this also holds during the whole acceleration-phase of the bearing. Appropriate values for both parts have therefore been underlain.
Figure 5 shows the experimental results of Fumagalli

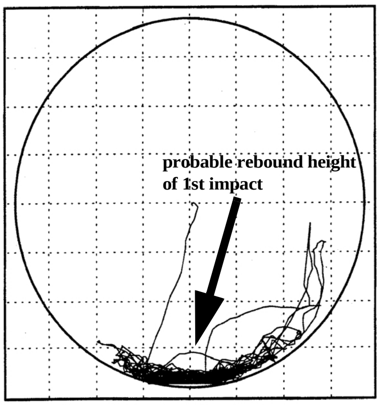

FIGURE 5: Measurement of a rotor-drop, [7]

after shutting down both AMB´s at 16 000 rpm. Fumagalli did not state if his results show the rotor-motion measured by the displacement sensors at the left or the right side of the rotor, so a comparison with both sensor-positions is necessary. Figure 6 shows the calculated rotor orbits at the displacement sensors.
For the simulation no AMB´s were modeled; it was also assumed that the rotor has no unbalance and the initial condition of the simulation was a centered rotor with no translatorical velocity. The dashed circles in Figure 6 illustrade the air-gap ε of the rotor, which was 0.3 mm at both retainer bearings. If the position of the rotor center (x_r) at the retainer bearing is inside the circle, than the rotor has no contact with that retainer bearing.

There are three phases of the rotor-motion to discuss comparing the measurements and the simulation results.
The first one is the phase of the rotor drop due to his weight, the next one is that of the first impacts, which is especially noticeable in the simulations, followed by the last phase, where the rotor performs an oscillating motion along the inner ring of the retainer bearing. At least in the simulation this oscillating motion is non-symetric, that means for example, while the rotor has a negative displacement in the horizontal direction at the left end, it has a possitive one at the right end.
Comparing the first phase of the simulation with the experiment, it is visible that the rotor does not fall straightly down in the experiment as it does in the simulation. The reason for this behavour could be the collapsing magnetic field of the AMB´s, still producing a decreasing magnetic force. That situation could be represented in the simulation by an adequate horizontal,

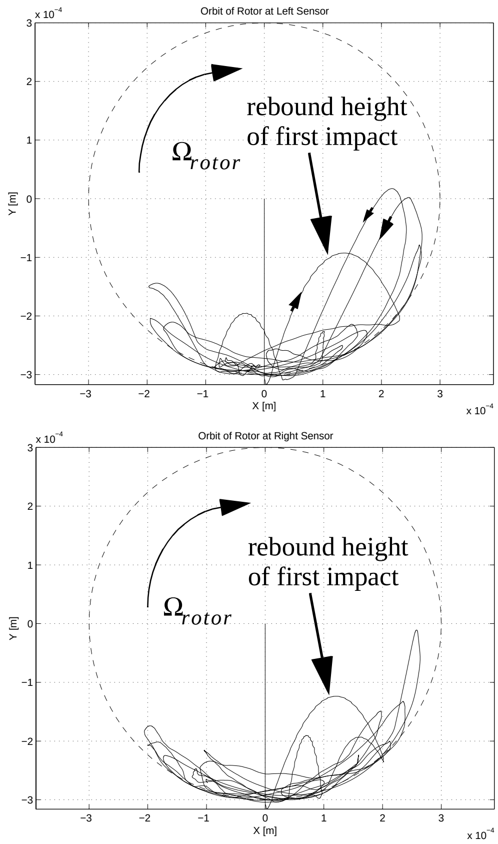

FIGURE 6: Simulation of the Rotor-Orbits during Touchdown

initial velocity. But Fumagalli presented only the experimental results for one point of the rotor and not at least at a secound one, therefor the first phase of his experimental results is not unique. A consequence of that would be a variation of initial conditions, corresponding to Fumagalli´s measurements. But this is not practical; even if the experiment is represented by this variation, the result of the simulation with the initial condition corresponding to the experiments is not fully comparable to the experimental results due to many assumptions made in the model of the system.
The comparison that can also be done with the initial condition shown in Figure 6.
During the secound phase there is one major discrepancy: The height of the rebounds, predicted by the simulations, is to high, particularly for the first impact. That means that the energy, dissipated during the impact, is to small.
Considering only the first impact, the height of its rebound in the simulation depends on the flexibility and damping properties of the rotor and the bearing but also on the selected initial condition.
Parameter-studies showed that also for other initial conditions, the rebound height is unresonable high, so the discrepancy in the rebound heigth between the simula-

tion and the experiment seems to be independent from the initial condition.
To get a better performance regarding the rebounds, the energy-dissipation has to be raised.
As a result of the parameter-study the damping parameter of the retainer bearing reached already values which force the retainer bearings to dissipate almost the whole energy they receive during the impact.
That means that almost the whole energy for the rebound heigth of the rotor is stored in the rotor itself during the impact
A reason for the discrepancy of the rebound height between experiment and simulation could therefore be an underestimation of the rotordamping. But there is also another possible reason for that. A smaller stiffness value of the retainer bearing provides the possibility for the retainer bearings to receive more energy and therefore to dissipate a greater energy-amount.
That makes clear that it is important to know the flexibility of the rotor and the retainer bearings exactly, and also the flexibility of elements surrounding the bearing.
The fact that the retainer bearings dissipate almost the whole energy they receive during the impact disagrees with the rather low damping coefficient of ball bearings that can be found in literature. But the aim of this first approach to model the touchdown a rotor in it´s retainer bearing was to predict the behaviour of the rotor. To make exact statements about the bearing loads and behaviour on basis of the simulation, a more accurate model of the retainer bearings is needed.

A comparison of the third phase of the rotor motion between Figure 5 and Figure 6 show that the oscillating motion of of the simulation is similar to that of the experimental results in Figure 5, especially the range in circumferential direction for the right displacement sensor. However, this motion of the rotor is influenced by the dynamic behaviour of the rotor, the initial condition of the simulation and by the contact force.
One parameter that has a major influence on the oscillating motion is the friction coefficient, which itself depends by the method of it´s determination on the inner frictional moment of the bearing. But the inner frictional moment is a result of a roughly assumption.
Thus the result of the simulation during phase three are based on uncertain approximation and on initial conditions which do not correspond exactly to the experiment. Therefor phase three of the rotor motioncan not be used to judge the quality of the model.

Conlusion

According to the presented first approach of predicting the dynamic behaviour of a rotor following an AMB-power loss, it became apparent, that a precise prediction of the rotor behaviour needs to validate the underlain models e.g. for the retainer bearing and the rotor by appropriate experiments. These can be performed by means of the presented concept of a test rig. Furthermore experiments have to be performed to enable statments concerning the validity of the modelling in general.

Acknowledgement

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