Dynamics of an Electromechanical Touchdown Bearing Mechanism

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Abstract **─** Safety bearings are used as a back-up system for the rotor-systems that use active magnet bearings for supporting the rotor. When the control fails due to unexpected events, the rotor drops on the safety bearings that prevent further damages of the system. This paper presents dynamic studies of a novel electromechanical safety bearing solution. The parts of the studied system are modeled as rigid bodies and the purpose is the study the dynamics of a real system in means of simulation. The safety bearing mechanism is manufactured and measured in the laboratory environment. The study shows that the dynamic behavior of the system can be repeated using the simulation tools that combines mechanical description of the system, electromagnetics and control system. Also, the electromechanical safety bearing solution seems to be feasible from the dynamic perspective.

Index Terms **─** Electromechanical system, mechanism, multibody dynamics, multidisciplinary simulation, safety bearings.

I. INTRODUCTION

Active magnetic bearings are widely used for promoting higher rotational speeds. Higher rotational speeds brings multiple advantages such as higher power density and savings on material costs. Actively controlled magnetic bearings are able to maintain high performance level through the full operational speed range contradicting the difficulties of other non-contacting bearings types such as air foil bearings. Contactless operation enables higher system efficiency and the regular maintenance of the bearing components is unnecessary. Active control requires a position feedback of the system that requires position sensors, but that brings advantage of monitoring the performance of the system without additional on-line sensors or regular on-site monitoring intervals.

The downside of the magnetic bearings is the unavoidable need for using an auxiliary systems that ensures the safe operation of the rotor system in emergency cases when control of the rotor-bearing system fails due to sudden event such as exceeding load limits or disturbances in the electric circuit. The regularly used approach is to use regular types of bearings that catch the rotor in case of dropdown. Regular bearing types such as deep groove ball bearings or angular contact ball bearings are used for this purpose. The inner bearing diameter is few hundred micrometers larger than the rotor diameter enabling the contactless regular operation. The effects of the most important parameters of touchdown bearings were studied by Ishii and Kirk in 1996 [1]. In case of dropdown, the rotor drops the distance of the airgap and the very first moments of the contact event between the rotor and the bearing inner bore are dramatic due to extreme peak stresses in the bearing components and high angular acceleration of the bearing inner ring and individual rollers in the bearings as shown by Neisi et al. [2].

The paper introduces a novel electromechanical actuator that can be used for safety mechanism of AMB supported rotor system. Previous studies has presented other types of active auxiliary systems such as piezo-hydraulic actuators [3] and controllable electromagnetic actuators [4]. Jiang et al. [5] studied two different control strategies of radially active auxiliary bearings in means of simulation. Our paper presents a study investigating the dynamics of an active touchdown bearing mechanism using a multibody simulation approach. Multibody simulation is regularly used tool for studying mechanisms

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and machines having multiple parts interacting through joints and forces [6]. It allows combining different disciplines in the system as long as the physical phenomena to be investigated can be modeled either analytically or numerically.

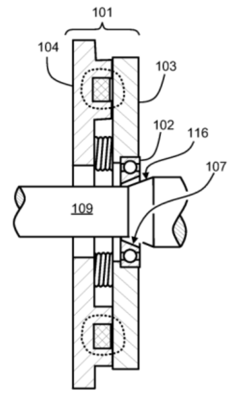
The main objective of the study is to explore the dynamical characteristics of the mechanism. It is vital for the operation that the launch is fast enough to catch the rotor before it hits on the bottom of the bearing ring. The launch can be accelerated by controlling the voltage of the electromagnetic actuator, pre-loading and selecting the mechanical springs appropriately and designing the mechanism properly to ensure smooth and direct translational movement of the touchdown bearing. The mechanism is manufactured and tested in a laboratory environment where three position probes were assembled circumferentially to measure the displacement of each point with respect to the main body.

The study contributes to the existing knowledge by introducing an interesting electromechanical device for increased safety that is studied in means of simulation and laboratory experiments for confirming the functionality of the mechanism. The simulation results and the experiments confirm that the electromechanical touchdown bearing mechanism is able to capture the rotor before it completely drops on the bearings.

II. TOUCHDOWN BEARING MECHANISM

The novel electromechanical touchdown bearing mechanism is launched, when a sudden undesired event of the system is identified. Figure 1 shows the schematics of the touchdown bearing mechanism including the list of parts [7]. The mechanism comprises electromagnetic actuators that are used for loading the mechanical springs and holding the mechanism in place. Once the mechanism is launched, the voltage of the electromagnetic actuators are driven to negative value resulting zero electromagnetic force. The parameters were selected based on fastest experimentally obtained launch event. The mechanical springs push the lid where the touchdown bearing sits to catch the rotor before it drops on the bottom of the touchdown bearings. The inner sleeve of the bearing is conical which angle match with the conical sleeve angle of the rotor that ensures the centering of the rotor when it is captured.

The conventional touchdown bearings with regular cylindrical inner bore may result in whirling motions of the rotor in a dropdown where the rotor orbits the centerline on the inner ring of the bearing. The inner ring should accelerate as fast as possible to meet the rotational speed at which the surface speed of the rotor and the inner ring bore would be equal. The proposed mechanism would completely close the airgap of the touchdown bearing disabling completely the whirling motion of the rotor.



101 = mechanism
102 = touchdown bearing
103 = moving lid
104 = stator of the mechanism
107 = conical bearing surface
109 = rotating shaft
116 = conical shaft shoulder

Fig. 1. Schematics and a part list of the studied touchdown bearing mechanism [7].

The studied mechanism is a simplified version of an actual system comprising a stationary part including the windings of the electromagnetic actuator that would be mounted to the end-plate of the machine, six mechanical springs (type Sodemann C04800810620M), and the moving lid that would include the touchdown bearing in the actual application. In the proof-of-concept, the stationary part is attached to a plate that is held in a screw bench and the lid does not have the touchdown bearings since the main objective is to study the release mechanism. The mechanism is loaded by increasing the current in the windings that generates a force field that is larger than the mechanical spring force. As the released lid is in the maximum distance from the stator and the electromagnetic attractive force is proportional to distance square, loading the lid requires much higher current than the required current for holding the airgap closed in the regular operation. The laboratory prototype and the 3D model of the studied system is illustrated in Fig. 2. The locations of displacement measurements are highlighted with yellow dots including the number of the proximity probe that will be used later to specify the observation point.

Fig. 2. Proof-of-concept electromechanical touchdown bearing mechanism in left and 3D model in right. Yellow points illustrate the location of eddy-current sensors.

III. SIMULATION MODEL

The actuator was modeled in MSC Adams 2017.1 and the simulation results were compared against the laboratory measurements. Simulation model was parametric enabling system testing with different initial values such as air gap length and varying spring coefficient of mechanical springs. The parts are modeled as rigid bodies comprising mass properties. The simulation model included the following parts:

- 2 bodies: frame and lid;
- 2 frictional contacts between bodies, radial and axial;
- 6 springs with individual spring coefficients;
- Electromagnetic pulling force distributed to points of mechanical springs.

A. Electrical modeling in simulation

Electrical modeling point of view the system is seen as variable inductor with breaking resistor making series RL-circuit. Resistance value and initial current are given as input parameter. Circuit inductance and force depend on current and position of lid thus look-up tables have been created for force and inductance based on electrical simulation.

Considering variable inductance, RL-circuit current can be calculated:

$$U_L + U_R = 0, \tag{1}$$

where U_L is voltage over the inductance and U_R voltage over the resistance. The sum of voltages can be also described as an equation:

$$L\frac{di}{dt} + i\frac{dL}{dt} + iR = 0, \tag{2}$$

where $L\frac{di}{dt}$ corresponds voltage generated over the inductance related to change of current, $i\frac{dL}{dt}$ voltage over the inductance generated by the changing inductance (mechanical movement of the lid) and iR the voltage loss related to resistive losses.

The system is studied in means of multibody dynamics simulation in time domain. The electromagnetic and mechanical forces are located in the same six points. Springs are assumed to have a linear spring coefficients whereas the electromagnetic forces and inductances are modeled as splines according to the look-up tables that are presented in Fig. 3. The state of the system is solved using time integration using a step size of $2.5 \cdot 10^{-6}$ s. Displacements are recorded in the locations of eddy current sensors that were used to measure the displacement in the experiments.

IV. RESULTS

Release was tested with differences in spring coefficients that were fine-tuned according the measured results. The used spring coefficients are presented in Table 1. The variation in spring coefficients is unrealistically large, but the unrealistic coefficients were used for compensating the effect of sliding friction and other non-linear effects that causes uneven motion of the lid. Initial current was set to 1.5 A, and 20 Ohm breaking resistor was used. Obtained release result can be seen from Fig. 4. The curves show the displacement of the lid that are observed from the same locations where the displacement probes are located in the prototype mechanism. The studied locations are referred as Place 1, 2 and 3, respectively. The result curve starts from the time step, where the system is launched. The lid hits in hard stop approximately 5 ms from the release. The first 3 ms is spent for waiting the electromagnetic force drop. Once the mechanical force exceeds the electromagnetic force and the static friction, the lid starts to move and reach the destination in 2 ms.

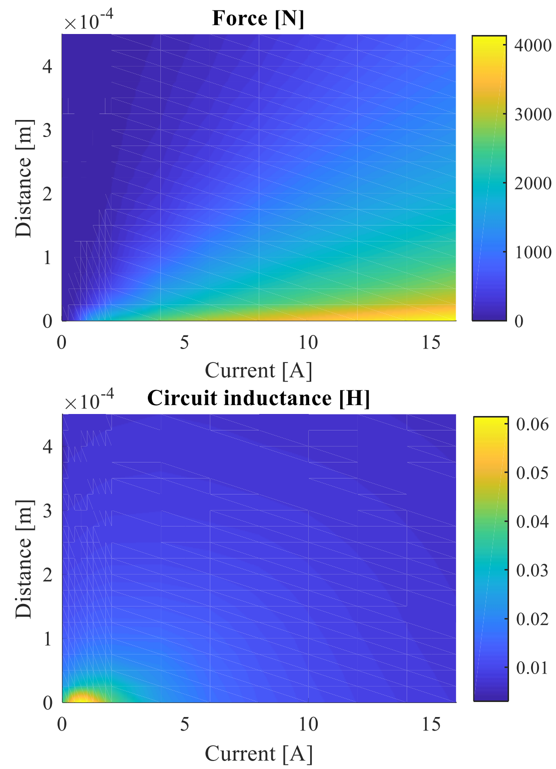


Fig. 3. Look-up tables for electro-mechanical pulling force and circuit inductance.

Table 1: Simulation spring coefficients

Spring Number	Angle	Spring Rate
1	30°	17.2 N/mm
2	90°	14.2 N/mm
3	150°	17.2 N/mm
4	210°	23.2 N/mm
5	270°	23.2 N/mm
6	330°	12.2 N/mm

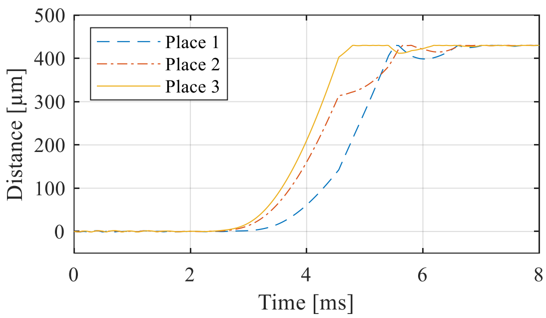


Fig. 4. Displacement in the simulated release.

A. Laboratory tests

The proposed electromechanical touchdown bearing mechanism was studied in the laboratory in order to study the performance and key parameters of the system in an adjustable and well-known environment. The system was adjusted by current in the electromagnetic actuator. The system was loaded using current of 8 A that was enough to produce force that exceeded the mechanical spring force. The obtained release current was 1.9 A and the current level of 4.5 A was selected for holding the lid in place replicating the typical operational conditions. The system was released by actively adjusting the current to a negative value enabling the fastest possible drop in electromagnetic force. Figure 5 shows the current profile during the launch, whereas Fig. 6 shows the corresponding voltage profile.

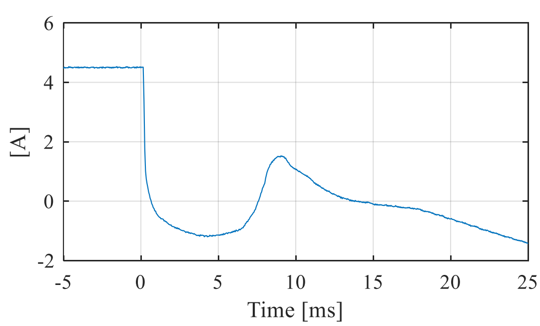


Fig. 5. Current profile of the system during a launch.

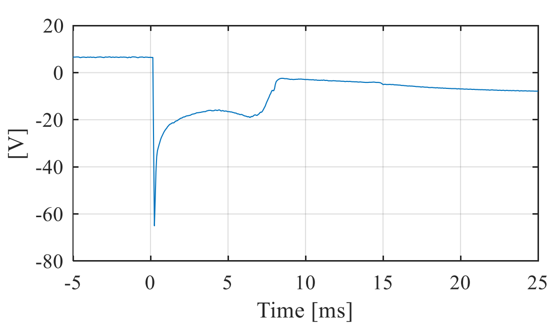


Fig. 6. Voltage profile of the system during a launch.

The displacement of the lid is observed in three positions. The displacement curves are presented in Fig. 7. The shape of the displacement curves are very close to the displacement curves of the simulated system. One of the reasons for such is the unrealistic spring coefficients, but still the overall results indicate clearly that the simulation model is able to predict the force interactions and the dynamics of the system well. The main differences in the curves are the time between the launch and the first motions and the time span that is required for closing the gap. In simulation model, the time from the launch to the first motions is 2.5 ms whereas in the measurement the lid experiences some motion instantly after the release. The simulation model is able to close the gap in 2 ms whilst the measurement indicates approximately 3-4 ms time span depending on the start and end points of observation. Table 2 tabulates the different phases of the release in both simulated and measured case. It can be also observed that the measured curves show more flexibility in the system when compared to simple rigid multibody simulation model. The system is attached in the laboratory in a screw bench from one side enabling some motion due to the flexibility of the system structure.

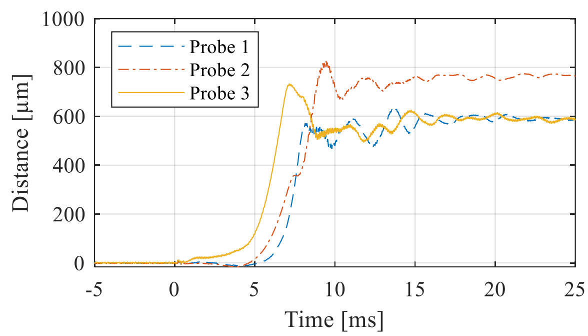


Fig. 7. Measured displacements during a launch.

Table 2: Phases of the release

Release Time	**Simulated**	**Measured**
Current drops to release point	< 1 ms	< 1 ms
Waiting lid to start moving	2.5 ms	1 ms
First side reaches 430 µm	4.5 ms	7 ms
Other side reaches 430 µm	5.5 ms	8 ms

In the actual rotating application with the introduced specifications, the rotor would drop approximately 50% of the airgap distance before the designed electro-mechanical system would catch the rotor. Naturally, the time and distance of the drop would depend on the combinations of touchdown bearing airgap, initial position, orientation and velocity of the rotor, and the rotor unbalance.

V. CONCLUSION

This paper studies a novel electromechanical system that is used for a safety system of an AMB supported rotor system. The mechanism comprises mechanical spring that will be loaded by an electromagnetic actuator. Once the current of the electromagnetic circuit is actively driven to negative value, the electromagnetic force is removed and the mechanical springs push the moving part of the system for catching the dropping rotor. The system is studied in means of simulation and experiments. The simulation uses rigid multibody dynamics approach where the electromagnetic forces are modeled as simple analytical equations. The spring coefficients of the mechanical springs were varied due to measured results showing that the moving lid tilts during the launch. The simulation results are close to the measured results in terms of the time span that is required for launching the system. The launch is that fast that it is able to capture the rotor from the air and smooth the drop enabling safe run down.

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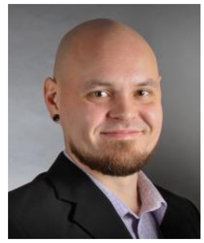

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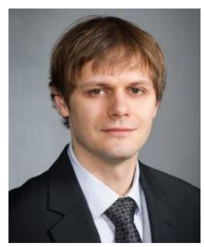

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