Performance evaluation of touchdown bearing using model-based approach

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Abstract In the active magnetic bearings (AMBs) supported rotating machinery, touchdown bearings are considered as safety devices to support the rotor in the deficiency of electromagnetic field. Generally, the industrial AMB machines do not have force sensors for touchdown bearings and the system is only equipped with the position sensors to monitor the rotor displacement that disables the opportunity to measure the forces during dropdown events that might be destructive for the safe operation of the rotor system. This study explores the relative severity of dropdowns that are evaluated from a computational rotor model using the rotor displacement data recorded

from the position sensors installed in the machine as an input. The model for double-row angular contact ball bearing type touchdown bearings is integrated with the rotor model. The simulation model is verified by comparing the simulated rotor orbits against measured orbits at four different dropdowns. The Fast Fourier transform (FFT) is used to observe the studied dropdown events in frequency domain revealing from the rotor response the following details: harmonics of the operation speed, rub-impact frequencies, pendulum motion of the rotor and the first two bending frequencies of the rotor settled on the touchdown bearings. The critical speed map also verifies the bending frequencies and identified support properties. The model-based approach can be used to evaluate and compare a single dropdown event with respect to previous events providing an insight for decision making whether touchdown bearing should be replaced.

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1 Introduction

At present, there is a high demand for applying active magnetic bearings (AMBs) in high speed machines. AMB systems are contactless, wear-free, and without lubrication. In such a machine, the electromagnetic

field levitates the rotor, the control system is integrated with the rotor and enables controlling the position of the rotor. The application of AMBs in rotating machinery facilitates running the machine at a higher rotational speed compared to a conventional rotor-bearing system. Simultaneously, the dynamic characteristics and performance of the machine become complicated. Furthermore, these systems demand the installation of touchdown bearings as a safety tool to prevent the rotor failure due to the insufficiency of the electromagnetic field or faults in the control system.

For both the AMB supported rotor and conventional rotor-bearing, research teams and industry become interested in the development of a dynamic model enabling the simulation of the behavior of the system during the operation. Operational data has been applied to develop and update the model of rotating machinery and identifying unbalance in the rotor, bending, misalignment, cracks in the rotor as well as estimating bearing parameters [19]. The recorded behavior of the rotor during run up and coast down can be also used to diagnose the fault in the system. Chandra and Sekhar [3] used the run up signal for the diagnostic of misalignment, crack detection, and rotor-stator rub. For the AMB supported rotor the following aspects need to be studied: (i) normal operation, where the electromagnetic field levitates the rotor, and (ii) failure of the electromagnetic field and rotor dropdown.

For many cases, predicting the dynamic parameters of AMBs is required for the rotor dynamic analysis of the system, stability analysis and control system. One solution to identify the equivalent stiffness and damping in AMBs is utilizing the unbalance response [39]. Even though the unbalance excitation is a simple way to excite the rotor, it has some drawbacks, as it cannot be applied to very low or very high speeds. At very low speeds, the unbalance response may be too small to be distinguished. At very high speeds, excessive unbalance may lead to problems in the control system [39]. In addition, the measurement noise and model error may influence the accuracy of the model. Tiwari and Chougale [36] have developed an identification method for the estimation of the dynamic parameter of the AMB of a flexible rotor that is tested against the above issue. The method for the identification of a speed-dependent dynamic parameter has also been introduced by Tiwari and Talatam [37]. For this purpose, at different speeds, the

controlling current and unbalance response have been measured. In the identification of AMB systems, the model can be decomposed into submodels. The advantage of this approach is that the physical interpretation of the results for each submodel can be investigated and can improve the entire model [32].

The above-mentioned studies have mainly demonstrated the behavior of the rotor when the AMB is active. The electromagnetic failure or overload may cause the rotor drops. In this situation, the touchdown bearings have to endure the rub-impact in the dropdown. Several studies have been dealt with the rotor dropdown. Kärkkäinen et al. [17] have introduced a dynamic model for rotor dropdown and investigated the effect of various friction models in the behavior of the rotor during delevitation. The simplified model of the bearing can be extended to develop the model for cageless touchdown bearings as shown by Helfert [11] and for the cageless touchdown bearing with misalignment as proposed by Halminen et al. [9]. Sun et al. [33] have demonstrated a detailed model for the touchdown bearings, where the friction coefficient, support characteristics and side loads from AMB have been addressed as the main parameters in the design of touchdown bearing. The implementation of the active actuator to reduce the rub and sever impact has been developed by Ginzinger and Ulbrich [7]. They applied the feedback control with the magnetic actuator enabling the softening of the transition from the condition of impact bounce to full contact and reducing the impact severity. Liu et al. [20] studied the rotor orbit during the bounce and rub of the rotor in the touchdown bearing. They utilized the Fourier transform for identifying the pendulum movement and the full rub of rotor. An experimental study by Lahriri and Santos [18] has shown that the touchdown bearing with a pin enables recentring the rotor after impact with the bearing and decreases the impact severity. They measured the contact force and presented the hysteresis loop in the dropdown.

The comparison of the dropdown simulation against recorded behavior of the rotor in the dropdown can be considered as an assisting tool to develop more accurate model and has attracted several research teams. Still further research on this field is going to improve the existing dropdown model. A comprehensive study by Schmied and Pradetto [28] is one of the earliest studies on touchdown bearings. They investigated the dropdown of a one ton rotor both

numerically and experimentally, and stated that the whirling in the dropdown can be reduced by means of a corrugated ribbon, low inertia of the inner ring, and a low friction coefficient. Identifying the dynamic properties of the corrugated ribbon damper between the touchdown bearing and housing has been presented in the experimental study by Jarroux et al. [13]. Schmied and Pradetto [28] also counted the time lag in the controller and possible evidence of the electromagnetic force at the start of dropdown as a cause of deviation in the results of numerical model and experiment. In the study by Siegl et al. [29] the comparison of the rotor orbit recorded in the dropdown of a nine ton rotor has been used for verification of the simulation. A similar approach has been used by Janse van Rensburg [27]. Fonseca et al. [4] have investigated the nonlinear behavior in dropdown due to rotor unbalance both numerically and experimentally and presented the rotor orbit for three level of unbalances. They found that for the low value of unbalance, the rotor oscillates at the lower half of the touchdown bearing while the high value of unbalance causes the rotor to bounce from contact and thus impacts the bearing, severely. The comparison of the touchdown bearing force and measured force can be found in few publications. Fumagalli [5] has investigated the force in the touchdown bearing utilizing the measured acceleration in the vertical direction. The capability of using different contact model in the simulation of dropdown is discussed by Jarroux et al. [13]. They measured the touchdown bearing force and presented the normalized bearing force and orbit versus measured results. Recently, Kang and Palazzolo [14] studied the 1/2X forward whirling during the dropdown of the rotor by numerical simulation and experiments. In the study of Sun et al. [35, 34] the Kalman filtering technique have been used to estimate rotor's displacements, velocities and accelerations from noisy observations.

Generally, industrial AMB machines do not have force sensors for touchdown bearings, and the system is only equipped with the position sensors to record the rotor displacement. Besides that determining the severity of dropdowns is not straightforward. The touchdown bearing should successfully withstand number of safe dropdown as should be agreed between the vendor and customer as stated in ISO 14839-4 [22] or in API 617 [1] (According to API 617-8 edition number of dropdowns from full speed should not be

less than two). Determining the severity of dropdowns is ongoing research topic and it is not easy to find the precise analysis requirements for touchdown bearings. According to study of Gouws [8] the delavitation severity indicators, safe envelope method and the linear extrapolation method can be considered as methods for evaluating severity of dropdown. The authors of the current study have applied the simulation model to study the force and stress in the touchdown bearing [24]. However, the previous study did not introduce methods to evaluate the severity of the dropdown. Current study aims to apply the model-based method for estimating the touchdown bearing forces. This study gives an intuition to relatively compare the dropdowns with respect to each other without a need to modify the original configuration of industrial machine for force measurement. Relative comparison of the dropdowns with respect to each other can assist the condition monitoring teams to investigate the condition of machine and do further analysis for determining the safe dropdown. To relatively compare dropdowns, the maximum Hertzian stress at different dropdowns is selected as a criteria to compare different dropdowns and analyzed via statistical method. In this study, the measured displacement of rotor and angular velocity of rotor are used to obtain initial condition of rotor and also updating the dropdown simulation. The support properties are estimated using the equivalent single mass model. The simulation model is tested for the dropdown test at four different speeds. Apart from this, the Fast Fourier transform (FFT) is used to study the speed dependent and the constant frequencies found in analyzing measured data. The critical speed map is used to confirm the bending frequencies of the rotor settled in the touchdown bearing and the estimated support properties.

2 Model of rotor and touchdown bearing

The FE-model of the flexible rotor is developed by applying the Timoshenko beam element theory where shear deformation is taken into consideration [25]. The present study concentrates on the lateral vibration of the rotor, and in the FE-model of the rotor the axial and torsional degrees of freedom are constrained. The equation of motion of the rotor-touchdown bearing can be written as:

$$\mathbf{M}\ddot{\mathbf{q}} + (\mathbf{C} + \omega\mathbf{G})\dot{\mathbf{q}} + (\mathbf{K} + \dot{\omega}\mathbf{G})\mathbf{q} = \omega^2\mathbf{F}_1 + \mathbf{F}_2 \quad (1)$$

where $\mathbf{M}$ is the mass matrix, $\mathbf{C}$ is the damping matrix, $\mathbf{K}$ is the stiffness matrix, $\mathbf{G}$ represents the gyroscopic matrix, and $\mathbf{q}$ is the vector of the generalized coordinates. $\mathbf{F}_1$ is the vector of nodal unbalance, $\mathbf{F}_2$ denotes the vector of externally applied forces, and ω is the angular velocity of the rotor.

In this study, the touchdown bearings are double row angular contact ball bearings, modeled based on the ball bearing model described in [30]. Figure 1 depicts the cross section of the bearing, where e_x , e_y and e_z show the displacements of the inner race along the main axes. The relative displacement between the bearing races can be obtained as follows:

$$\begin{aligned} e_j^r &= e_x \cos \beta_j + e_y \sin \beta_j \\ e_j^t &= e_z - (\psi_x \sin \beta_j + \psi_y \cos \beta_j)(R_{in} + r_{in}) \end{aligned} \quad (2)$$

where ψ_x and ψ_y are the misalignment of the inner race on the x and y -axes, β_j is the attitude angle of the j th ball, R_{in} is inner race radius and r_{in} is the inner race groove radius. When the bearing is under the load, the contact angle can be calculated as follows:

$$\phi_j = \tan^{-1} \left(\frac{e_j^t}{R_{in} + r_{in} + e_j^r - R_{out} + r_{out}} \right) \quad (3)$$

Then, the distance between the races can be defined as follows:

$$d_j = r_{out} + r_{in} - \frac{R_{in} + r_{in} + e_j^r - R_{out} + r_{out}}{\cos \phi_j} \quad (4)$$

Fig. 1 Cross section of the ball bearing

The total elastic compression of the bearing races is:

$$\delta_j^{\text{tot}} = 2r_b - d_j \quad (5)$$

where r_b is the radius of ball. The force between the j th ball and the inner race can be expressed as [10]:

$$F_j = K_c^{\text{tot}} (\delta_j^{\text{tot}})^{3/2} \quad (6)$$

where K_c^{tot} is the total contact stiffness.

The maximum Hertzian contact stress in the bearing can be calculated as [10]:

$$\sigma_{\text{max}} = \frac{3F_j}{2\pi a^* b^*} \quad (7)$$

where a^* and b^* are semi-major/minor axes of the elliptic contact, respectively.

2.1 Model of the contact between the rotor and touchdown bearing

The Hunt and Crossley contact model [12] is used for the contact between the rotor and touchdown bearing. This model has been widely used in the dropdown simulation. Alternative models for the rotor-touchdown bearing can be found in the study of Jarroux et al. [13]. In the nonlinear contact model of Hunt and Crossley, the contact force is expressed in terms of the elastic force and the dissipation force as follows:

$$F_r = \begin{cases} K\delta^n + b\delta^n \dot{\delta}^q & e_r > c_r \text{ and } F_r > 0 \\ 0 & e_r \leq c_r \text{ and } F_r \leq 0 \end{cases}, \quad (8)$$

where K is the contact stiffness between the rotor and touchdown bearing that is obtained with the Hertzian model, $q = 1$ and the power exponent n is determined by contact type, $n = 10/9$ for the line contact between the rotor and inner ring, $\dot{\delta}$ represents the rate of indentation of the rotor on the inner race, and b is a damping parameter connected to the coefficient of restitution which can be expressed as $b = 3/2\lambda K$. The range for the contact parameter λ and its effect on the hysteresis behavior have been discussed in detail by Hunt and Crossley [12]. In this study the coefficient λ equals to 0.08, is selected for the contact of the rotor and touchdown bearing [31].

The indention of the rotor in the bearing can be expressed as follows [17]:

$$\delta = e_r - c_r \quad (9)$$

where c_r represents the radius of the air gap, and the radial displacement of the rotor found to be (See Fig. 3):

$$e_r = \sqrt{e_{x,r}^2 + e_{y,r}^2} \quad (10)$$

where $e_{x,r}$ and $e_{y,r}$ are the radial displacement between the rotor and the inner race on the x - and y -axes, respectively.

The friction force between the rotor and touchdown bearing is:

$$F_\mu = \mu F_r \quad (11)$$

where μ is the friction coefficient between the rotor and the inner race. The friction is modeled by combining the Coulomb, Stribeck and static frictions, the friction force is formulated as the function of the relative speed between the rotor and touchdown bearing (Fig. 2), where the discontinuity of the friction force due to rapid sign change near zero velocity is avoided by defining the polynomial expression near zero speed. The friction can be modeled with several models, further information about different friction models can be found in the paper by Olsson et al. [26]. The friction force creates torque against the direction of rotation that can be calculated as follows:

$$M_v = F_\mu r_r \quad (12)$$

In Fig. 3, the support stiffness and damping are shown by K_{sup} and C_{sup} , and will be described in

Fig. 2 Friction force as a function of relative velocity

Fig. 3 Contact model between the rotor and touchdown bearing

Sect. 3.2. The bearing damping C_b is about $(0.25-2.5) \cdot 10^{-5}$ bearing stiffness K_b [15]. The angular velocity of the inner race and rotor are denoted as ω_i and ω_r , respectively.

In this work, the thermal expansion of the touchdown bearing due to the friction heat is neglected. This simplification has been made as the dropdown will be studied for short period of time (0.2 s) and within this period no significant rise in the temperature of touchdown bearing is expected [23, 38].

2.2 Aerodynamic torque

The recorded behavior of the rotor under investigation shows that the rotor speed in the dropdown follows the exponential profile. In the model characteristics of the exponential profile for rotor speed are taken from the measurement data. Then, the aerodynamic torque in the deceleration of the rotor can be expressed as follows:

$$M_{ad} = I_p \dot{n} \quad (13)$$

where $\dot{n}$ is a derivative of the speed of the rotor during deceleration and I_p is the polar moment of inertia of rotor.

3 Estimation of touchdown bearing force

The test prototype studied and most industrial AMBs do not have a force sensor for touchdown bearings. In this study, the force in the touchdown bearing is estimated by applying the measured displacements of the rotor. To accomplish this, the measurement data is integrated with the dropdown simulation. This process may involve several challenges since there are numerous parameters that are not measured and needed to be determined. The earlier study on the dropdown simulation of the rotor under investigation [24], considers that at the start of dropdown, the rotor is at the center of the coordinate system, while the measurement data reveals that due to the effect of the electromagnetic field, the rotor might not be located at the center. In this study, the initial conditions of the rotor are determined using measured signals and modal expansion. In addition, the measurement results show a dissimilarity between the measured air gap and nominal air gap, and the air gap for the touchdown bearings at both ends is found to be different (10 μm difference). This difference may be resulted from several reasons, such as a manufacturing error or the sensitivity of the sensor. In the model, the air gap is determined from the measurement data and the difference in the air gap at both ends is taken into account. The angular velocity of the rotor recorded in the test is used for the calculation of the gyroscopic effect and the aerodynamic torque. The estimation of the force in the touchdown bearing is carried out by

developing sub-models for the initial position of the rotor, support properties, and friction coefficient.

The touchdown bearing force can be calculated by means of the bearing model presented in Sect. 2. For verification purposes, the rotor orbit obtained from the simulation is compared with the measurement orbit, which helps to justify that with the level of force estimated, the rotor displacement is in the same order as the measured data. The estimation has been based on the flowchart shown in Fig. 4.

3.1 Initial conditions of the rotor

AMB supported machines are equipped with a limited number of vibration sensors, while the numerical model used in the transient analysis of rotor dropdown requires considering the initial displacement and velocity along all degrees of freedom. Implementing the modal expansion enables estimating the initial position of the rotor at all degrees of freedom. To

Fig. 4 Flowchart of estimation of touchdown bearing force

proceed with this method, the measurable part of the system can be related to the deformation at all degrees of freedom as follows [21]:

$$\mathbf{r}_m = \mathbf{Q}\mathbf{r}_t \quad (14)$$

where $\mathbf{r}_m$ is the vector of measured data, $\mathbf{Q}$ is the measurement matrix and $\mathbf{r}_t$ represents the deformation at all degrees of freedom. In this approach, the deformation at all degrees of freedom can be estimated with help of a reduced mode shape matrix as follows:

$$\Phi = [\Phi_1 \quad \Phi_2 \quad \Phi_3 \quad \cdots \quad \Phi_k] \quad (15)$$

where k is the number of mode shapes used for reduction and can be equal to the maximum number of measurement points. All degrees of freedom can be solved using the following equation:

$$\mathbf{r}_t = \{\Phi[(\mathbf{Q}\Phi)^T(\mathbf{Q}\Phi)]^{-1}[\mathbf{Q}\Phi]^T\}\mathbf{r}_m \quad (16)$$

In this work, four measurement points (horizontal and vertical displacement of the rotor at the non-drive end (NDE) and drive end (DE)) are used to monitor the lateral vibration of the rotor. Hence, the first four modes from the mode shape matrix, confined to rigid body modes, are applied to feature the initial condition of the rotor. This assumption may be valid for the cases where the operating speed of the rotor is below the first critical speed. In the matter of the AMB supported rotor operating above its first critical speed, applying the first four modes from the mode shape matrix can still provide reasonable results since the deformation of the rotor due to its flexibility is small compared to the displacement of the rotor with respect to the supports. The initial condition for the velocity is obtained by differentiation of the displacement signal. The initial velocity of the rotor has a remarkable effect on determining the path of the rotor in the dropdown. At the initial stage of the dropdown, the calculated velocity signal might be noisy. Accordingly, noise smoothing and taking an average of the velocity signal just after the fall of the rotor might be beneficial to find the four measurement points for the velocity of the rotor at sensor locations. Then, by employing the modal expansion, the initial velocity of the rotor along all degrees of freedom is known and can be used for transient analysis.

3.2 Support properties

The support properties are estimated by using the equivalent single mass model shown in Figure 5 [28]. The method initiates by applying the time history of the vertical displacement of the rotor measured by sensors. The first contact of the rotor and the touchdown bearing is studied to estimate support properties. The equivalent stiffness can be calculated as follows:

$$\omega_k = \sqrt{K_{eq}/M} \quad (17)$$

where M is the share of the rotor mass at the location of each support and ω_k is the natural frequency of the single mass system. The damping ratio is 0.025 which is selected based on the literature [16].

The equivalent stiffness in (17) represents the combination of K_{sup} and K_b introduced in Fig. 3. To simplify problem, it is assumed that the support stiffness, K_{sup} , in horizontal and vertical are equal. Further analysis is required to separate these stiffness components. The least-square optimization function in the Matlab software has been used to determine the parameters of the supports. The equivalent stiffness properties are used for an initial guess of the support stiffness, whereas the bearings are modeled using the proposed method in Sect. 2. A dropdown event is simulated until the bounce after the first contact (0.015 s). The simulated response is compared against the measured data and the optimization algorithm is used to define the optimum values for support stiffness, damping, and mass.

Fig. 5 Single mass model for support

3.3 Friction coefficient

Rotor dropdown is simulated using static friction coefficients ranging between 0.05 and 0.20, and the simulated orbits are compared with the measured ones. The friction coefficient that imitates the behavior of the measuring system most closely is selected based on visual observation for the parameter of friction coefficient in all of the performed studies.

4 Prototype and measured data

The studied prototype is a part of the gas turbine shown in Fig. 6. The unit produces 400 kW electricity and includes two-stage high-pressure and low-pressure turbines. The dropdown test was carried out at 60 Hz, 100 Hz, 150 Hz and 210 Hz rotation speed for the low-pressure turbine. For safety consideration the dropdown test was conducted at lower speed than nominal speed (550 Hz). The dropdown test was performed as follows. In the beginning of the test, the active magnetic bearings supported the rotor. Then, the magnetic bearing was powered off. In the test prototype, two non-contact displacement sensors near each touchdown bearing are used to measure the displacement of the rotor. The system also has a non-contact displacement sensor for measuring the axial displacement of the rotor. During the dropdown test, an axial AMB was kept on service to study the radial movement without disturbance from the axial displacement of the rotor. The angular velocity of the rotor was recorded.

The experimental condition enables us to have more control over the test condition, and it can be

different from the actual working condition. For instance, in this study, the axial AMB was kept on service to focus on studying the horizontal and vertical movement of the rotor, while in the failure of the electromagnetic field in actual working conditions, the axial AMB is most probably also out of service. In that case, the rotor experiences coupled radial and axial movement that is even more unpredictable and difficult to repeat by means of simulation. An example of the axial displacement of the rotor during the dropdown can be found in the study of Neisi et al. [24]. Furthermore, it is expected that the dropdown at the operation speed will be considerably influenced by the unbalance force and, as a result, the touchdown bearing experiences more chaotic force. Apart from this, in most industrial machines it is not possible to acquire information about the condition of touchdown bearing mainly due to the limitation in the measurement of desired parameters. A model-based approach can be used to overcome this problem. In the first stage, using the test condition can be considered as an assisting tool for verification of the dynamic model. Secondly, the study introduces a method presenting how to utilize the measurement data for the dropdown simulation.

4.1 Measurement data

The relative displacement of the rotor and touchdown bearing was measured via non-contact displacement sensors. As the displacement sensors were oriented at 45-degree to the main axis of the rotor, the measurement data is mapped to the coordinate the system used in the simulation. Figures 7 and 8 show the measured displacement of rotor for the dropdown at 60 Hz. Figure 7a depicts the vertical displacement in the NDE touchdown bearing. After the first contact, the rotor bounces back in the air up to 50 μm , then establishes a new contact. Afterward, the vertical displacement of the rotor decreases and the rotor stabilizes gradually within the air gap length. As Fig. 7 shows, the rotor position at the beginning of the simulation is not zero and it should be considered in the simulation. From the beginning of the test until the first contact, the rotor moves slightly in the horizontal direction, which is due to the initial velocity of the rotor before dropdown. After the first contact, when the rotor is bouncing back into the air, simultaneously

Fig. 6 The prototype for gas turbine [24]

Fig. 7 Displacement of rotor at NDE touchdown bearing, dropdown test at 60 Hz, **a** y-direction **b** z-direction

Fig. 8 Displacement of rotor at DE touchdown bearing, dropdown test at 60 Hz, **a** y-direction **b** z-direction

moves horizontally and hits the inner ring which is caused by friction force during the contact. Then, the rotor moves back toward center. The rotor has continuous contact with inner ring around 0.07 s, and starts to swing at the bottom of the bearing, which can be observed from the z -direction as an oscillation around the zero displacement. The swinging motion decreases gradually and almost dies down within the introduced time range.

Figure 8 depicts the rotor displacement at the DE touchdown bearing. After the first contact, the height of the bounces of the rotor from the NDE bearing is considerably higher than DE bearing. The rotor under

investigation is unsymmetrical, and the mass center of the rotor is closer to DE bearing. Therefore, first the rotor contacts DE bearing, and at the same time the rotor bounces at NDE side and the vertical displacement of the rotor at NDE reaches 50 μm. The bounce of the rotor from the DE bearing is moderate, and in the next bounce the rotor does not come into contact with the touchdown bearing in z -direction. In addition, the rotor hits the bearings at different times, and it has a higher number of contacts in the DE rather than the NDE, which is characterized by the conical movement of the rotor at low frequencies. Furthermore, the mass center of the rotor is close to the DE bearing, which

results in a higher indentation in the DE touchdown than in the NDE touchdown.

5 Simulation result

The main parameters of the rotor are shown in Table 1. The touchdown bearings are pair of angular contact ball bearings of the type XCB71914-E-2RSD-T-P4S-U which are installed in X-configuration. The FE-model of the rotor presented in Fig. 9 has 36 nodes, and the touchdown bearings are connected to the nodes 4 and 26. The axial disc of the rotor is modeled as a mass point at the node 1.

The compressor impeller and turbine impeller are modeled as mass points (nodes 30 and 33). Table 2 shows the material properties of the impellers. In the FE-model, the generator band and AMB lamination are also taken into account. The rotor is balanced according to ISO 1940 G1.0 and the residual unbalance in the rotor is included in the analysis, however, it does not have a significant effect on this study. The numerical simulation utilizes Matlab integrator *ode45*, which implements the fourth-order Runge–Kutta numerical integration. The simulation time step is $1 \cdot 10^{-5}$ s, the maximum integration step size is $5 \cdot 10^{-6}$ s, the maximum allowable integration error is $1 \cdot 10^{-5}$, and the dropdown is simulated for 0.2 s.

The nominal air gap of the machine is 250 μm , while the dropdown test indicated that the air gap is different than the nominal air gap. Possible reasons for the discrepancy between the nominal air gap and measured values may be related to manufacturing and assembly tolerances, some deflections in the system due to previous dropdown, the differences in the locations of the sensors and bearings, and inaccuracies of the measurements. It should be noted that in an

industrial machine the support structure is generally complicated and the simulation usually requires simplifications in the support definition. For instance, the machine might be equipped with the corrugated ribbon damper between the touchdown bearing and the housing that might cause a nonlinear system response during the dropdown. The corrugated ribbon may experience plastic deformation, or the touchdown bearing may not return in the original position after the dropdown due to dry friction. However, in this prototype, the corrugated ribbon damper does not exist in the system, and the former reasons could be an explanation for the difference between the nominal air gap and measurement. To determine the air gap from the measurement data, the end of the test where the rotor completely settles in the bearing is defined as the air gap limit (later on, Fig. 17 will illustrate the dropdown until the rotor fully stops). In Fig. 17a, average of the rotor displacement from 48 to 50 s has been selected as the lower limit for the air gap (270 μm at NDE), and the time interval of 51–53 s is selected for defining the upper limit of air gap (275 μm at the NDE). Similarly, the lower and upper limits of the air gap at DE were found to be 280 μm and 285 μm , respectively.

5.1 Support stiffness and damping

The estimation of the support properties is initiated by studying the vertical displacement of the rotor measured in the test. The measurement data confined to the first contact between the rotor and the touchdown bearing has been selected to study the support properties because it shows how much the supports are compressed under the rotor weight without disturbance from other phenomena involved in the dropdown. Figure 10 zooms in the vertical displacement of

Table 1 The Parameters of rotor used in the simulation

Parameter	Value
Modulus of elasticity	$2.05 \cdot 10^{11}$ Pa
Mass of rotor	59.5 kg
Poisson's ratio	0.3
Contact stiffness between rotor and touchdown bearings	$2.02 \cdot 10^9$ N/m ^{1.11}
Lower limit for air gap at the NDE, DE	270, 280 μm
Upper limit for air gap at the NDE, DE	275, 285 μm
Rotor length	1082 mm
Maximum diameter of rotor at electrical core	116 mm

Fig. 9 Model of rotor,
a Drawing and **b** FE-model **Table 2** Mass properties of the impellers [24]

	Compressor	Turbine
Material	Aluminum alloy 7075-T6	Heat resistant alloy steel
Center of mass x, y, z [m]	0, 0, 0.0648	0, 0, 0.03749
Moment of inertia I_d, I_p [kgm ²]	0.0197, 0.0126	0.0436, 0.0449

the rotor for 0–0.04 s (Figs. 7 and 8 present the measurement for 0.2 s). The time t_1 shows the moment when the rotor comes into contact with the touchdown bearing for the first time, and the rotor displacement starts to exceed the air gap. During the time span of t_1 – t_2 , the rotor is in contact with the bearing. Then, the rotor lifts from the bearing's inner ring. The impact duration T represents half of the frequency of the equivalent system shown in Fig. 5 [28]. The static equilibrium where the impact period is calculated based on it, has been determined from the measured displacement of the rotor at the end of the dropdown test where the angular velocity of rotor reaches to zero (275 μm and 285 μm for the NDE and DE, respectively). By performing the static balance with respect to the mass center of the rotor, the equivalent mass at each support is evaluated. Then, by applying (17), the equivalent stiffness of the system is calculated and the results are shown in Table 3.

The stiffness value presented in Table 3 is a combination of the support and bearing stiffness. For the next step, the mechanical model of the touchdown bearing presented in Fig. 3 requires to determining the support stiffness from the equivalent model. In the conventional rotor-bearing system, the support model

can be simplified as a series springs, and knowing the bearing stiffness (from the bearing manufacturer), the support stiffness can be calculated. However, in the touchdown bearing application, due to the rapid change in the rotation speed of the rotor and the nonlinear dynamic loads experienced by the bearing, the constant stiffness of the bearing might be questionable.

To clarify this, understanding the equivalent stiffness calculated from the measurement data, an initial guess for the unknown parameters as support stiffness and mass is selected (stiffness of $1 \cdot 10^8$ [N/m] at the NDE, $4 \cdot 10^8$ [N/m] at DE and support mass of 40 kg). From the literature, the damping ratio of 0.025 for steel structure support damping is selected. Then, applying the detailed model for the touchdown bearing presented in Sect. 2, the load and deformation of the bearing at different speeds are taken into account. The simulation is performed and the vertical displacement of the rotor is compared with the measurement data. Then, the least square optimization method is used to tune the support properties and minimize the difference between the vertical displacement of the rotor obtained from the simulation limited to the first contact of rotor and bearing. The optimization applies

Fig. 10 Equivalent stiffness of the support from time history, dropdown test at 60 Hz, **a** NDE **b** DE

Table 3 Equivalent stiffness based on measurement data

Parameter	NDE	DE
M [kg]	17.8	41.7
t_1 [ms]	7.40	7.95
t_2 [ms]	9.99	9.75
T [ms]	2.59	1.8
ω_k [rad/s]	1212	1745
K_{eq} [N/m]	$2.61 \cdot 10^7$	$1.27 \cdot 10^8$

a tolerance function of 10^{-6} , and the maximum iteration of 10^3 . Table 4 presents the final values for the support parameters. To evaluate the accuracy of the support model, the support properties calculated

Table 4 Estimated support properties

Parameter	NDE	DE
Support mass [kg]	51.2	51.2
Support stiffness [N/m]	$1.305 \cdot 10^8$	$4.026 \cdot 10^8$
Support damping [Ns/m]	$3.27 \cdot 10^3$	$5.74 \cdot 10^3$

from the dropdown at 60 Hz are used for the simulation of dropdown at 100 Hz, 150 Hz and 210 Hz, and the displacement of the rotor is compared with the corresponding data from the measurement (Fig. 11). The results show that the model provides quite reasonable results at different dropdown speed,

Fig. 11 Vertical displacement of the rotor, first contact, dropdown at 60 Hz, 100 Hz, 150 Hz and 210 Hz (solid line: simulation, dashed line: measurement), **a** NDE **b** DE

mainly for the contact time and the indentation of the rotor in the bearing. In the dropdown at 60 Hz, a small difference is observed in the bounce height in the simulation data and measurement data. In the dropdown test at higher speed, the difference in the simulation results and the measurement is found to be greater that can be attributed to the effect of the unbalance force. In this case, the evaluation of the unbalance phase angle might be difficult to extract from this specific figure for two reasons. First, the prototype does not have the angular displacement encoder, the accurate phase angle of the rotor remains unclear. Secondly, the simulation model encounters the recorded position of the rotor in the sensor locations and the velocity and the direction of the motion is derived from the displacement data that consequently takes into consideration in some extend yet not completely the effect of the unbalance force and its phase angle (The method of including the initial conditions of the rotor is presented in Sect. 3.1). This can cause the behavior of the rotor immediately after the shutdown of the AMBs has a deviation with respect to the measurement. Also, the rotor experiences a small lift right after the AMB shutdown in all measured cases (Fig. 11, the very earliest moments 0...0.001 s). This indicates that there are other contributing factors affecting this behavior than the phase of unbalance since it is highly unlikely that phase would be in the same direction in all test cases and it should behave more unpredictable. Some of the potential candidates for this behavior are, for example, certain unidentified aerodynamic forces or higher magnetic remanence after the AMB shutdown in the electromagnet that is acting against gravity. This can create a force in the upward direction that is then seen as an instantaneous lift of the rotor.

5.2 Friction coefficient on rotor orbit

The friction coefficient for the contact between the rotor and bearing is an unknown that is estimated by studying the rotor orbit for the static friction coefficient in the range of 0.05–0.20. The comparison of the orbit in the simulation and measurement shows that with the static friction coefficient of 0.13 and the dynamic friction coefficient of 0.1, the path where the rotor drops, the maximum indentation of the rotor in the bearing and the pattern of the bounce height are in good agreement with the measurement. Then, using

these parameters, the model is tested for the rotor dropdown at 60 Hz, 100 Hz, 150 Hz and 210 Hz.

A primary evaluation of the rotor orbits showed that when the dropdown occurs at lower speed, after first few contacts of the rotor and the inner ring, the rotor tends to settle in the touchdown bearing, whereas for the dropdown at higher speed the number of the contacts of the rotor with inner ring and also the rotor bounce are increased and the orbit becomes chaotic. For instance the rotor orbits for the dropdown at 150 Hz are shown in Fig. 12. The horizontal and vertical displacement of the rotor corresponding to these orbit are shown in Fig. 13, and might reveal more information about the rotor behavior. Figure 13a shows that after the first few contacts of the rotor and NDE touchdown bearing (after 0.05 s), there is a phase difference between the horizontal displacement of the rotor in the simulation and measurement. The simulation results also show that the horizontal and vertical displacement of the rotor in DE touchdown bearing have a better match with the measurement data rather than corresponding data for NDE touchdown bearing.

The phase difference between the simulation and measured signal for the horizontal displacement of the rotor at NDE touchdown bearing can be partially explained as results of the possible difference between the friction model used in the simulation and the friction and rub occurred between the rotor and touchdown bearing in the test prototype. Nevertheless, the friction is not the only item that is causing this difference. The rotational dynamics of the touchdown bearing is an important factor that might cause some inaccuracies in the simulation model. The polar moment of inertia of the bearing inner ring is obtained from the 3D model of the bearing. However, the material density has always some variation that might cause some variance for the inertia details. The inertia of the bearing inner ring is very essential information since the acceleration of the inner ring is extremely high. Another important factor that might influence the dynamics of the bearing inner ring is the preload of the bearing pair. In this study, the preload is defined according to the bearing catalog that suggests the classification for bearing preload. However, actual preload in the prototype can be different due to manufacturing tolerances and thermal effects.

In this case study the touchdown bearings at both sides of the rotor are the same type and interestingly the phase shift in the horizontal displacement of the

Fig. 13 Rotor displacement dropdown at 150 Hz **a** horizontal at NDE **b** vertical at NDE **c** horizontal at DE **d** vertical at DE

rotor in the NDE touchdown bearing is higher than DE touchdown bearing, that can be also due to the difference in the acceleration rate of the inner ring of the touchdown bearing at both ends. As the mass center of the rotor is close to the DE touchdown

Fig. 12 Rotor orbit dropdown at 150 Hz **a** NDE **b** DE

bearing it accelerates faster. The difference in the acceleration rate of the inner ring affects the friction torque and therefore the phase difference in the NDE is high. The similar pattern is observed for the dropdown of the rotor at 60 Hz, 100 Hz and 210 Hz.

The radial displacement of the rotor in the simulation and measurement is used to evaluate the capability of model. As can be seen in Fig. 14 the simulation model for both limits of air gap enables to provide reasonable results with respect to the measurement data. This figure shows that applying upper limit of air gap the radial displacement of the rotor is more close to the actual displacement of rotor whereas using the lower limit of air gap might result in some unexpected jump (Fig. 14c after 0.1 s). The radial displacement of the rotor also shows a phase difference between the simulation and measured signal (after first few contacts) that might be resulted from friction model. In case of 210 Hz rotational speed, the measured displacement curves show some single higher peaks that are not captured in the simulation model. However, it remains unclear if these unexpected peaks are related to higher rotational speed in general or this specific rotational speed. The orbit shown in Fig. 12 presented the simulation with upper limit of air gap.

5.3 Bearing force

Figure 15 depicts the magnitude of the force experienced by the touchdown bearing as a function of time. The detailed model for the touchdown bearing is used and the contact force between the individual ball and the bearing race is calculated. Then, utilizing the bearing model the magnitude of bearing force is obtained [30]. Figures 15a and b indicate that for the dropdown at 60 Hz, the highest bearing force in the touchdown bearing occurs in the early contacts of the rotor and bearing. The maximum force for the NDE and DE are found to be 2907 N and 4889 N, respectively. Under the gravity force the rotor gradually tends to stabilize in the touchdown bearing. The average of the bearing force from 0.15 to 0.2 s at the NDE and DE are found to be 218 N and 440 N, at the NDE and DE, respectively. As mentioned earlier, the static equilibrium with respect to the mass center of the rotor results in 17.8 kg and 41.7 kg at the location of each touchdown bearing (correspond to 177 N and 409 N at the NDE and DE, respectively). Even though

the simulation is performed for a short period of time, it is clear that the magnitude of the bearing force gradually tends to hover around the static force on each bearing. This can justify the force estimation of the touchdown bearing. For the dropdown at 100 Hz, the bearing forces are found to have a similar pattern as at 60 Hz. However, for the dropdown test at 150 Hz, the bearing force is more arbitrary, and interestingly some peak in the maximum force for the NDE occurs at 0.07 and 0.1 s, respectively (Fig. 15c). The dropdown at 210 Hz also has arbitrary pattern.

It can be restated that the highest force occurs in the DE bearing. The maximum bearing forces for the NDE and DE are about 3000 N and 4800 N, respectively. Depending on the dropdown case, the peaks could be observed at different moments. The results indicate that by increasing the dropdown speed the rotor experiences chaotic motion and the bearing force will be more arbitrary. In the dropdown at low speeds, the dropdown is mainly affected by the height of drop and the gravity force, whereas in the dropdown at higher speed the unbalance force becomes increasingly important and can lead to chaotic motions that consequently results in higher touchdown bearing forces. In addition, the result shows that using the upper limit for the air gap (i.e., larger air gap) yields to higher bearing forces.

The validation of the magnitude of the bearing force requires establishing experimental setup for force measurement that needs to be done in future. Furthermore, evaluating the severity of dropdown and further decision about replacement of touchdown bearings is not straightforward. The touchdown bearing should successfully withstand number of safe dropdown as should be agreed between the vendor and customer as stated in ISO 14839-4 [22] or in API 617 [1]. The vague statement from the standards implies that determining number of safe dropdown and severity of dropdown is beyond the current state of art and therefore needs to be agreed between vendor and enduser. The outcome of this work gives an insight to compare the behavior of the rotor in the different dropdowns, this can assist the condition monitoring team to investigate which dropdown was most severe and take the required action or replacing the bearing.

In this study, the maximum Hertzian stresses are used as a basis to compare different dropdowns. The distribution of the maximum Hertzian stress is studied for a period of 0.2 s. The box plot shown in Fig. 16 is

(a)

(b)

(c)

(d)

(e)

(f)

(g)

(h)

◀ **Fig. 14** Radial displacement of the rotor in the simulation and measurement **a** 60 Hz at NDE **b** 60 Hz at DE **c** 100 Hz at NDE **d** 100 Hz at DE **e** 150 Hz at NDE **f** 150 Hz at DE **g** 210 Hz at NDE **h** 210 Hz at DE

used to graphically compare the stresses in the dropdown at 60 Hz, 100 Hz, 150 Hz and 210 Hz. The red line in the middle shows the median. The red+ in the plot shows the outlier, where there the stresses are more than three times of standard deviations from the mean of the stress. The high number of outliers indicating the high variation of the stress with respect to mean.

The main data of the statistical analysis of stress are shown in Table 5, and the following characteristics can be found from the relative comparison of dropdowns:

- The overall feature shows that the boxes are skewed and the stress in the dropdown is not distributed equally.
- In all cases the maximum Hertzian stress at DE touchdown bearing is higher than NDE touchdown bearing which can be as a results of unsymmetrical rotor and location of mass center.
- The average of the stress in the DE touchdown bearing for the dropdown at 210 Hz is 1159 MPa, whereas for the NDE touchdown bearing the highest average of stress found to be at 100 Hz. This shows that depend on the dropdown case either the DE or NDE touchdown bearing can experience highest average stress.

Table 5 shows that the maximum stress at DE touchdown bearing is observed at 60 Hz, and for NDE touchdown bearing at 150 Hz, while it is expected that the maximum stress occurs at dropdown at higher speed (210 Hz). First, a more comprehensive testing procedure including several dropdowns at the same rotational speed would give a better insight for the currently observed results in terms of repeatability. It is worth mentioning that the maximum stress in the dropdown is not dependent only on the angular velocity of the rotor at the start of the dropdown, and it could be affected by other factors. For instance, at the beginning of dropdown, the rotor might not be necessary located in the center and the initial location and velocity of the rotor (angular velocity as well as horizontal and vertical velocity) can be different in

each test. Apart from this, due to limitations in evaluating the unbalance phase in the measurement, the initial condition of the rotor might not be fully in according to the measurement. Therefore, the component of the force acting in the bearing might be affected by the unbalance phase and consequently, it can affect the estimated bearing force and affects the stress value. This can be an explanation that why the maximum stress does not necessarily appear at a higher rotational speed (210 Hz).

5.4 Further analysis of experimental results

When the dropdown test is conducted at 60 Hz, after 48 s the angular velocity of the rotor is reduced to zero and the rotor is completely settled in the touchdown bearing (Fig. 17c). In practice, the simulation of the entire dropdown incident is computationally expensive and might not be necessary. The bearing force presented in the previous section was calculated for a short time: 0.2 s. During this time, the angular velocity of the rotor decreased from 60 Hz to approximately 58 Hz. The experimental results were further analyzed to ensure that sever contacts occur at the early stage of the dropdown, and the simulation time of 0.2 s enables studying the highest bearing force and investigating the overall feature of the touchdown bearing force.

Figure 17 features the magnitude of the rotor displacement of the DE touchdown bearing recorded in the experiment until the rotor fully stops. The measurement results confirm that the maximum indentation of the rotor in the touchdown bearing occurs at the early stage of the dropdown. Therefore, 0.2 s is enough to identify the maximum force applied on the bearings. However, the measurement results reveal that during the deceleration of the rotor, the system might experience resonance. The similar behavior has been observed for the NDE. The main reason for the source of such excitation at low frequencies and its severity requires further analysis.

In the next step, the Fast Fourier transform (FFT) is applied to analyze the measurement signal, the following items were considered for signal processing:

- From the time domain signal, $1 \cdot 10^6$, $1.3 \cdot 10^6$, $1.4 \cdot 10^6$ and $1.5 \cdot 10^6$ points (corresponding to 50 s, 65 s, 70 s and 75 s) from the dropdown test at 60 Hz, 100 Hz, 150 Hz, and 210 Hz, respectively is extracted.

Fig. 15 Magnitude of bearing force for dropdown **a** 60 Hz at NDE **b** 60 Hz at DE **c** 150 Hz at NDE **d** 150 Hz at DE

- The sampling frequency is $2 \cdot 10^4$ Hz, the number of FFT points is 32768, and the resolution of frequency is 0.6104 Hz.
- The overlapping of 50 percent and the windowing Hanning are implemented, and at the end the amplitude is multiplied by 2 (Hanning correction).

The waterfall plots for the horizontal and vertical displacement of the rotor reveal two main characteristics: (i) The harmonics of the rotation speed at 1X, 2X and 3X are clearly seen in all dropdown tests. The frequencies at 1.5X, 2.5X, 3.5X and 4.5X appeared in the dropdown test at 100 Hz, and 150 Hz feature the rub impact of the rotor in the bearing. See Fig. 18 for the dropdown test at 100 Hz. For clarity, the low frequencies representing the rigid body mode are excluded, and the y-axis of waterfall (rotor speed) up to 95 Hz is shown. (ii) Resonance at a constant frequencies of 29 Hz, 134 Hz, and 235 Hz were appeared in all tests. The dominant frequency of 29 Hz represents the frequency of the rotor due to the pendulum movement that can be calculated according to Beléndez et al. [2]. The frequencies of 134 Hz and 235 Hz can be recognized by following the critical speed map depicted in Fig. 19. The vertical lines plotted at the equivalent stiffness (calculated support stiffness and bearing stiffness from simple method by [6]) of the NDE and DE intersect the first and second bending modes in the vicinity of the identified frequencies of 134 Hz and 235 Hz, respectively. In the other words, 134 Hz and 235 Hz represent the first and second bending modes of the rotor when it is settled in the bearing. In addition, there is no evidence of a bearing fault in the waterfalls implying that the touchdown bearings were able to withstand the tests.

To summarize this section, it can be restated that the support properties are unknown, and the critical speed map and the waterfalls are used to verify the identified support stiffness as follows: (i) From the waterfall, the first and second bending frequencies of the rotor laying on the bearing inner races are identified. (ii) The critical speed map is used to evaluate the effect of support stiffness on the natural frequencies (critical speeds). As shown in Fig. 19, the

(a)

(b)

(c)

(d)

Fig. 17 Magnitude of rotor displacement in the DE touchdown bearing, dropdown test at **a** 60 Hz **b** 100 Hz **c** 150 Hz **d** 210 Hz

Fig. 16 Box plot of the stress distribution within 0.2 s (MPa)

Table 5 Stress in the bearing within 0.2 s (MPa)

Parameter	Maximum stress	Average	Median	ST ^a
DE (60 Hz)	3577	1005	906	795
NDE (60 Hz)	3014	970	708	623
DE (100 Hz)	3576	1142	824	794
NDE (100 Hz)	3049	1038	808	600
DE (150 Hz)	3546	1103	655	818
NDE (150 Hz)	3089	1024	784	622
DE (210 Hz)	3559	1159	830	793
NDE (210 Hz)	3061	1025	748	634

^a ST=Standard deviation

equivalent stiffness intersects the first and second bending mode within the frequencies obtained from a waterfall confirming the used method.

6 Conclusions

This study employed the model-based approach to estimate the touchdown bearing force. The measurement data recorded in the dropdown of the rotor has been united with the simulation of dropdown. The modal expansion method was employed to obtain the initial condition of the rotor. The support properties were estimated with a single mass model, representing the equivalent stiffness of the system. The support model was found to be robust for the dropdown of the rotor at different speeds.

(a)

(b)

(c)

(d)

Fig. 18 FFT waterfall for DE, dropdown test at 100 Hz **a** y-direction **b** z-direction

The rotor orbits obtained by simulation are seen to correspond to the test results, mainly in terms of the rotor path, the indentation of the rotor in the bearing, and

the bounce of the rotor after contact with the bearing. The results showed that when the dropdown test was conducted at a higher speed, the rotor exhibits more

Fig. 19 Critical speed map for the rotor when supported by a touchdown bearing

chaotic movement. The numerical value of the bearing force can be validated in the future by establishing the setup for force, measurement. At this stage, current model can be used to relatively compare the drop-downs, and understand how the drop-downs are differ from each other. The statistical analysis of the maximum Hertzian stress in the different dropdown tests revealed that dependent on the dropdown speed, the highest average of stress can be observed in either NDE or DE bearing, and dropdown at higher speed found to be more sever.

In addition, the FFT analysis on the measured displacement of the rotor reveals the harmonics of rotation frequencies ($1X, \dots, nX$). When the dropdown is conducted at a higher speed, the fractional harmonics at $1.5X, 2.5X, \dots$ are observed, which can be related to the rub-impact frequencies. Furthermore, in the waterfall, the constant frequencies because of the pendulum movement of the rotor as well as the first and second bending modes of the rotor settled in the bearing have been observed and have been verified by using the calculated support properties and critical speed map. In addition, the test results reveal that the system will pass some resonance at low frequencies until the rotor completely settles in the bearing.

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Compliance with ethical standards

Conflict of interest The authors declare that they have no conflict of interest.

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