Dynamic Simulation of Active Magnetic Bearing Supporting Rotor
Centrifugal Compressor During Drop On Touchdown Bearings

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Summary
The active magnetic bearing (AMB) has many advantages compared with the conventional bearing. It is the most widely used bearing in many machine applications; such as centrifugal compressor, air blower, and turbo molecular pump. AMB applications must be equipped with auxiliary bearings system to prevent the damage to the impellers and motor in case of system failure. The auxiliary bearing is referred to as the touchdown bearing in this study. When a failure is occurred, the active magnetic bearing cannot support the rotor stably. Touchdown bearings serve as a backup for the active magnetic bearings to support the rotor during a drop-down failure event. Therefore, a properly designed touchdown bearing system is necessary to protect the active magnetic bearing assembly and other critical machine components from direct contact with the rotor during AMB loss in power outage events. This system uses a ball bearing as a type of touchdown bearing. A finite element based 2-DOF (two-degree of freedom) flexible rotor model is used to indicate the rotor behavior. The rotor model also considers the contact force between the shaft-inner race and ball bearing force based on Un-lubricated Hertzian contact models. This study presents a dynamic rotor simulation during a drop event applied in a magnetic centrifugal compressor. This simulation is built using MATLAB with rotor speed effect and touchdown bearing design to predict the rotor behavior based on the rotor orbit and response.

Keywords: Active magnetic bearing, Rotor drop event simulation, Touchdown bearing, Centrifugal compressor, Finite element method

1. Introduction
Advanced technology development has brought significant improvement to the active magnetic bearing (AMB). The AMB is widely used in dynamic rotors. For example, it is commonly used in the centrifugal compressor. Active magnetic bearings have many advantages compared with conventional bearings. The advantages of magnetic bearings are minimal friction and wear, no need for a lubrication system and there is no physical contact between the rotating and stationary machine parts. The auxiliary bearing is an important component that must be utilized in active magnetic bearing applications to prevent damage to impellers and motors when magnetic bearings fail to operate. Some researchers call the auxiliary bearing as a catcher bearing or retainer bearing. In this study, the auxiliary bearing is called the touchdown bearing.

Touchdown bearings will be used when a drop event occurs. Drop events occur when the active magnetic bearing cannot support the rotor stably. Touchdown bearings serve as a backup for active magnetic bearings to support the rotor during a drop down event. Cao et al. \(^1\) presented the detailed nonlinear transient analysis formulation for a rotor drop event. In the dropdown situation, the rotor drops from the magnetic bearings down to the touchdown bearings. The touchdown bearing design parameters have a significant influence on the rotor behavior. Dynamic rotor simulation can be used to simulate the rotor response and behavior. Many researchers studied touchdown bearing design parameters to determine the rotor response and behavior. Various touchdown bearing design parameters were studied using simulation models such as the friction coefficient, mass of unbalance, stiffness and damping support coefficient \(^2-3\). The rotor response or rotor motion indicates the rotor behavior, so that the rotor needs to be modeled in the simulation. Some researchers modeled the rotor as a rigid rotor in their simulation \(^4-5\). However, in this study the rotor is modeled as a flexible rotor with 2-DOF (degree of freedom) and solved using the finite element method using MATLAB software. Finally, from the rotor orbit and response, the rotor speed and touchdown bearing design (with or without damping and stiffness support) are

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examined and analyzed.

Nomenclature

\[ M \]: shaft mass matrix, kg
\[ C \]: shaft damping matrix, Ns/m
\[ G \]: shaft gyroscopic matrix, kg.m²
\[ K \]: shaft stiffness matrix, N/m
\[ \psi \]: radial contact angular position, rad
\[ X,Y \]: shaft center location, m
\[ \omega_r \]: rotor angular velocity, rad/s
\[ g \]: gravitational acceleration, m/s²
\[ L \]: contact length, m
\[ m \]: rotor mass, kg
\[ u \]: rotor eccentricity, m
\[ t \]: time, s
\[ k_i \]: inner race stiffness, N/m
\[ k_o \]: outer race stiffness, N/m
\[ \delta_r \]: shaft deflection, m
\[ \theta_j \]: attitude angle of ball \( j^{th} \)
\[ \mu_r \]: friction coefficient

2. Simulation Method and Rotor Model

2.1 Simulation method

The rotor drop event is simulated by deriving a dynamic model. The dynamic model design simulates the rotor motion when the rotor is dropped onto the touchdown bearing using the Finite Element method with MATLAB software. Figure 1 shows the rotor model schematic diagram.

In the simulation, the rotor is modeled as flexible with 72 elements and 73 nodes as shown in Fig. 2. All the elements in the finite element model are modeled as two degrees of freedom (2-DOF) per node, i.e. the calculations consider only two translational degrees of freedom (x and y displacement). Therefore, the total number degrees of freedom are 146. In the rotor design, two magnetic bearings are set into nodes 22 and 61, and two touchdown bearings into nodes 13 and 70.

2.2 Rotor drop modelling

The active magnetic bearing system has two different motions; the motion before the drop event and motion of drop event. The motion before the drop event means that the rotor is supported by active magnetic bearing (AMB). However, when a system failure is occurred, the rotor will drop onto the touchdown bearing because the active magnetic bearing cannot support the rotor stably. That event is called the motion drop event. The rotor motion when it drop down can be described by the displacements x and y. Considering the external forces acting on the rotor include contact force \( F_c \), touchdown bearing force \( F_b \), unbalance force \( F_u \) and gravity force \( F_g \), the general equations of motion are expressed as:

\[
M\ddot{x} + (C + \omega_r)\dot{x} + Kx = F_{cx} + F_{bx} + F_{cx}
\]

\[
M\ddot{y} + (C + \omega_r)\dot{y} + Ky = F_{cy} + F_{by} + F_{cy} - F_g
\]

where:

\[
F_{cx} = m\omega_r^2 \cos \omega_r \cdot t
\]

\[
F_{cy} = m\omega_r^2 \sin \omega_r \cdot t
\]

\[
F_g = m \cdot g
\]

2.3 Contact force model

The clearance between the shaft and touchdown bearing is called as the air gap. The touchdown bearing air gap is typically half of the air gap of the AMB system. Therefore, when the rotor is dropped there will be a contact between the shaft and touchdown bearing (inner race), instead of AMB system. Figure 3 illustrates the contact forces model.
which refers to Liu et al.\(^6\). Based on Fig. 3, the touchdown bearing contact forces in the X and Y direction are given in Eqs. (6)–(7).

\[
F_{ex12} = F_{n12} \cos \psi_{1,2} - F_{t12} \sin \psi_{1,2} \quad (6)
\]

\[
F_{ey12} = F_{n12} \sin \psi_{1,2} + F_{t12} \cos \psi_{1,2} \quad (7)
\]

2.4 Touchdown bearing model

Ball bearings are commonly used as touchdown bearings. However, in some applications, sleeve bearings can be used as touchdown bearings. In rotor drop simulations, ball bearing behavior is important since the bearing behavior is affected by the geometric, material properties etc. Equations (12)–(15) describe the equations of motion (EOMs) for the ball bearings:

\[
m_{b12} \ddot{x}_{12} = F_{ex12} - C_{b12} (\dot{x}_{12} - \dot{x}_{012}) - K_{b12} (x_{12} - x_{012}) \quad (12)
\]

\[
m_{b12} \ddot{y}_{12} = F_{ey12} - C_{b12} (\dot{y}_{12} - \dot{y}_{012}) - K_{b12} (y_{12} - y_{012}) \quad (13)
\]

\[
m_{b12} \ddot{z}_{12} = C_{s12} (z_{012} - z_{12}) + K_{s12} (z_{012} - z_{12}) - C_{b12} (z_{12} - z_{012}) - K_{b12} z_{12} \quad (14)
\]

\[
m_{b12} \ddot{\omega}_{12} = C_{s12} (\dot{\omega}_{012} - \dot{\omega}_{12}) + K_{s12} (\dot{\omega}_{012} - \dot{\omega}_{12}) - C_{b12} (\omega_{12} - \omega_{012}) - K_{b12} \omega_{12} \quad (15)
\]

where \(m_{b12}\) and \(m_{b21}\) are the mass of the touchdown bearing inner and outer race, then \(C_{b}\) and \(K_{b}\) are the ball bearing damping and stiffness. Whereas \(C_{s}\) and \(K_{s}\) are the damping support and stiffness support if they exist.

A ball bearing consists of a number of moving parts. For each ball, there are normal compressive forces, centrifugal force and ball gyroscopic moments. This study models the touchdown bearing by neglecting the ball centrifugal forces and gyroscopic moments. Some other researchers also neglected those forces to reduce the simulation time. Neglecting both of these forces, the direct method to calculate normal compressive force \((Q)\) uses the following equation:

\[
Q_j = k_b \delta_{j}^{3/2} \quad (16)
\]

The total deformation \(\delta_{j}\) and also the total stiffness \(k_{b}\) of the inner and outer ball-raceway can be determined using Hertzian contact stiffness \(\gamma\), while the values of \(k_{s}\) and \(k_{b}\) depend upon the geometric and material properties of ball and raceway which can be determined as follows:

\[
k_{b} = \left[ \frac{1}{\left(\frac{1}{K_s}\right)^{2/3} + \left(\frac{1}{K_b}\right)^{2/3}} \right]^{3/2} \quad (17)
\]
Referring to Karkkainen [8], the resultant touchdown bearing forces applied to the shaft in X and Y directions are as follows:

\[
F_{bx} = - \sum_{j=1}^{n} Q_j \cos \theta_j \tag{18}
\]

\[
F_{by} = - \sum_{j=1}^{n} Q_j \sin \theta_j \tag{19}
\]

3. Simulation Results

This section presents the rotor drop event simulation results, with different rotor speeds and touchdown bearing designs (with or without stiffness and damping support). Figure 5 describes the simulation flowchart. These results will be discussed for both touchdown bearings; TDB1 (front touchdown bearing) and TDB2 (rear touchdown bearing) with the initial conditions x=0 and y=0. Based on the simulation method, the rotor motion is solved using the Finite Element Method and only considers two translational degrees of freedom (x and y displacements).

Table 1 describes the touchdown bearing specifications as shown in Fig. 6. Whereas Table 2 shows the compressor and rotor specifications.

![Flowchart simulation](image)

**Fig.5 Flowchart simulation.**

![Touchdown bearing](image)

**Fig.6 Touchdown bearing.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner &amp; outer radius ((R_i) &amp; (R_o))</td>
<td>0.03 m &amp; 0.0425 m</td>
</tr>
<tr>
<td>Number of balls (Z)</td>
<td>27</td>
</tr>
<tr>
<td>Ball diameter ((d_b))</td>
<td>7.938 \times 10^{-3} m</td>
</tr>
<tr>
<td>Touchdown bearing stiffness ((K_b))</td>
<td>1.46x10^8 N/m</td>
</tr>
<tr>
<td>Raceway Young’s modulus ((E_r))</td>
<td>208 GPa</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor type</td>
<td>Magnetic/Centrifugal</td>
</tr>
<tr>
<td>Cooling capacity</td>
<td>1054 kW</td>
</tr>
<tr>
<td>Max rpm</td>
<td>20000</td>
</tr>
<tr>
<td>Motor type</td>
<td>PM motor / 4 poles</td>
</tr>
<tr>
<td>Motor and shaft casing pressure</td>
<td>6.5 bar abs / 22°C</td>
</tr>
<tr>
<td>Mass of rotor (m)</td>
<td>32 kg</td>
</tr>
<tr>
<td>Shaft radius ((R_s))</td>
<td>29.9x10^{-3} m</td>
</tr>
<tr>
<td>Rotor eccentricity ((u))</td>
<td>1.25x10^{-6} m</td>
</tr>
<tr>
<td>Air gap</td>
<td>1x10^{-4} m</td>
</tr>
<tr>
<td>Shaft young’s modulus ((E_s))</td>
<td>200 GPa</td>
</tr>
<tr>
<td>Contact length (L)</td>
<td>11 x10^{-4} m</td>
</tr>
</tbody>
</table>

3.1 Rotor speeds effect

Figures 7 and 8 show the rotor orbit of both touchdown bearings with different rotor speeds; 4500 RPM and 5000 RPM. All rotor orbits illustrate that the rotor will drop to the bottom of the touchdown bearings and then the jump motion occurs. The speed increment causes the bounce motion become higher and wider. Based on the rotor orbit can also be shown that the rotor will touch the touchdown bearings inner race since the orbits are greater than the touchdown bearing gap of 1x10^{-4} m.

Figures 9 and 10 illustrate the rotor response during simulation time from 0-0.001s. The dots line indicates the touchdown bearing gap. All rotor responses show that the rotor orbit will drop and then touch the inner race bearing. The displacement exceeds the bearing gap (1x10^{-4} m). The Y-displacement shows that the rotor will drop at 0.0004 seconds. An increase in speed leads to higher rotor displacements. The rotor motion is unstable to the end of the simulation because the rotor is assumed to rotate at constant speed (without deceleration speed).
The finite element method is commonly used for AMB flexible rotor modeling. Flexible rotor modeling is involved in rotor shaft deflection. However, this study only focused on the deflection that occurred at rotor drop motion in TDB1 and TDB2. Figure 11 shows the rotor shaft deflection of TDB1 and TDB2 results at 4500 RPM. The shaft deflection value affects the rotor displacement. As mentioned in Eq. 9, when the shaft deflection is equal or less than zero, the normal force and friction force magnitude also become zero. This means that there is no contact between the shaft and inner race touchdown bearing.

As shown in Fig. 11 the TDB1 shaft deflection values at 0.003-0.004 seconds are less than zero. Therefore, the TDB1 displacements at 0.003-0.004 seconds are still in the touchdown-bearing gap as
shown in Fig. 9(a). That condition means the shaft did not touch the TDB1 inner race during that time. Also for TDB2, based on Fig. 11 the TDB2 shaft deflection value at 0.0066 seconds is greater than zero. As illustrated in Fig. 9(b) the TDB2 y-displacement at 0.0066 seconds is greater than the TDB2 gap. This indicates that contact exists between the TDB2 shaft and inner race.

In addition, Fig. 12 shows the rotor orbit preliminary results at 20000 RPM. However, further feasibility study is needed for high speed simulations. From those figures can be seen that the rotors will be dropped to the bottom of touchdown bearing. As shown from the rotor orbit, the rotor motion tends to oscillate at the bottom and jumping motion also occurred.

3.2 Effect of damping support and stiffness support

Touchdown bearing design parameters have a significant effect on the rotor dynamic behavior during the drop down event. The most examined touchdown bearing design parameters are the stiffness, damping and friction coefficient. The influences of those designs are widely known, as pointed out by some researchers [9-10]. They mentioned that those design parameters bring non-linear effects to the rotor behavior. The simulation must consequently be performed to know the damping and stiffness support effects, so that the appropriate touchdown bearing design can be selected.

As mentioned in the above paragraph, this section will examine the effect of damping support and stiffness support as an improvement to the original touchdown bearings design. Since the original design did not have those support components. Tolerance rings will be used in this study as an improvement to the original touchdown bearings. The tolerance ring model seems as stiffness support ($K_s$) and damping support ($C_s$). Figure 13 illustrates the additional damping and stiffness support design to the touchdown bearing original design. The damping and stiffness support values are 87600 Ns/m and 3.5x10^6 N/m.

Figure 14 shows that with a rotor speed of 4500 RPM, the rotor orbit still touch the inner race. However, the bouncing motion can be reduced by using damping and stiffness support. It also can be seen in Fig. 15 the rotor y-displacement decreases in comparison with touchdown bearing without damping and stiffness support as shown in Fig.9.
Touchdown bearing design takes a very important role in rotor behavior. The simulation results show that the rotor motion is affected by the rotor speed. As the increasing of the rotor speed, the rotor displacement can be higher. In addition, using damping and stiffness support can be considered as an improvement to the original touchdown bearing design. Based on the simulation results, the addition of damping and stiffness support can reduce the rotor bouncing motion.

Therefore, conducting dynamic simulations of the active magnetic bearing during a drop event is very useful as an evaluation method or a consideration before designing the compressor with active magnetic bearing system. Considering another touchdown bearing design and applying the rotor deceleration in this study is necessary for further improvement, in order to make the simulation more complex and detail.

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References