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A New Stabilization Technique for Electrodynamic Bearings

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Abstract: Electrodynamic bearings (EDBs) are a kind of passive magnetic bearing based on the eddy currents that develop between a rotating conductor and a static magnetic field. This type of bearing is capable of providing radial stiffness compatible with high speed rotor requirements, but is intrinsically unstable. The stabilization of the rotor is the greatest challenge to the application of electrodynamic bearings. The present work presents a new stabilization strategy for EDBs. The proposed solution is studied by means of analytic models. Experimental results are shown to prove the validity of the modeling and of the stabilization solution.

Keywords: Electrodynamic Bearings, Rotordynamics, Stability

Introduction

Electrodynamic bearings (EDBs) constitute a type of magnetic bearing where contactless suspension is obtained by passive means as result from the interaction between a rotating conductor and a stationary magnetic field. The working principle relies on the relative motion between a conductor and magnetic field that induces eddy currents inside the conductor, generating magnetic forces. These forces can be exploited, for example, to support a rotating shaft.

Relative to active magnetic bearings, the passive nature of the suspension implies several advantages, such as reduced complexity, improved reliability and smaller size and cost. Nevertheless, electrodynamic bearings have drawbacks such as the difficulty in ensuring stable levitation in a wide speed range and the low achievable stiffness and damping in comparison to active magnetic bearings.

Previous studies on this kind of mechanical component are mainly focused on the application of EDBs to support flywheels [1-11] and turbomolecular pumps [12,13]. Latest researches on radial EDBs are concentrated in reducing eddy current losses by using homopolar configurations [9-13] and particular configurations of shunted coils and Halbach arrays [7,8]. These approaches allow to obtain systems that produce virtually no losses when the rotor is in the nominal position, generating losses only if an external disturbance introduces an eccentricity.

In the past decade, the research on electrodynamic bearings has been mainly focused on the performance in stationary conditions (fixed eccentricity and constant rotating speed).The results, obtained by means of finite element analysis [12-15] and experimental measurements [5,16], demonstrate values of radial stiffness compatible with industrial applications.

At present time the stabilization of the rotor constitutes the biggest challenge for the design and application of electrodynamic bearings in real world industrial applications. Stability issues are mainly linked to the intrinsic presence of rotating damping [17] and to the fact that, similarly to hydrodynamic supports, EDBs provide levitation only when the relative speed is above a threshold value. This implies that rotors supported exclusively by EDBs are unstable in all the speed rage if no non rotating damping is provided [18,19].

In the past few years analytic models to study the electromechanical dynamics of EDBs were presented [19]. A detailed study on the modeling of the electromechanical dynamics of EDB was offered by Amati, Tonoli and De Lépine [18]. In their work a study on the necessary conditions to obtain stable levitation was presented and discussed.

The investigation of stability of EDBs was firstly introduced by Filatov and Maslen [9]. In their work a solution for the stabilization was obtained by introducing non rotating damping between rotor and stator by means of an electromagnetic device. The system consisted in a permanent magnet attached to the rotating shaft and interacting with a conductor fixed to the stator of the flywheel. In this way it is possible to damp the whirling motion of the rotor by means of a contactless device, accordingly to the EDB issue. Similar strategies were used by Sandtner and Bleuler [7,8] and experimental results proved the feasibility of this design approach.

This approach is quite straightforward as damping is introduced directly between rotor and stator. Nevertheless it presents some weak points. The need of obtaining damping by contactless means discards most of the classic damping devices such as squeeze film dampers or elastomeric supports. Furthermore, the introduction of a magnet in the rotor increases the rotor mass and complexity, and the brittle nature of permanent magnet material can introduce difficulties when working at high rotational speeds.

The present paper presents a new strategy for the stabilization of EDBs. Instead of introducing damping between the rotor and the non rotating part of the bearing, the stabilizing action is introduced between the stator of the bearing and the outer case of the machine. In this way the damping is added between two non rotating parts and the application of common dampers becomes perfectly suitable. This configuration allows to use conventional and low cost devices as dampers, to avoid the addition of further components on the rotor, and minimize the construction issues of the whole system. The suggested configuration is studied by means of analytic models. Experimental results prove the validity of the modeling and of the proposed solution for the stabilization of electrodynamic bearings.

New Stabilization Technique for Electrodynamic Bearings

The stabilization of the rotor constitutes the biggest challenge for the design and application of electrodynamic bearings in real world industrial applications. The dynamic behavior of a rotor on electrodynamic bearings can be studied using a Jeffcott rotor model. The equation of motion that rules its dynamics, written in complex coordinates is

$$m_r \ddot{z}_c + F_z = F_{ext}, \tag{1}$$

where $z_c = x_c + jy_c$ [17]. F_{ext} is a generic disturbance force while F_z is the force generated by the EDB [18]. Equation 1 is the base for studying the stability of a rotor

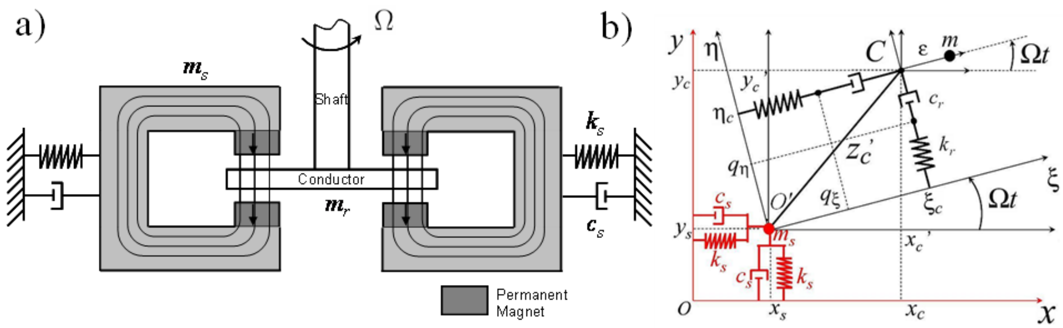

Figure 1: a) scheme of an EDB stabilized by introducing damping between the non rotating part of the bearing and the casing. b) schematization of the equivalent Jeffcott rotor.

supported by EDBs and to understand the necessity of additional non rotating damping to achieve stable levitation.

Stabilization by Stator-Casing Interaction. Instead of introducing damping directly on the rotor by using contactless devices, the Authors propose to introduce damping between the non rotating part of the bearing and the casing of the machine. Figure 1a shows a schematic representation of this solution.

The underlying idea is to dissipate the energy through an element placed between two non rotating parts. For this purpose, classical elements such as elastomeric rings would be perfectly suitable. Besides, all the critical aspects characterizing the solution proposed by Filatov and Maslen in [9], such as additional elements on the rotor, are avoided, especially for the weight, complexity and cost issues. This principle is far from being new, in fact it is used on different kinds of high speed rotors where damping is added between the non rotating part of the bearings and the casing of the machine. For example in the form of squeeze films or elastomeric mounts. The innovative aspect of the proposed solution consists in exploiting this effect to achieve the stable levitation of a rotor on radial electrodynamic bearings.

Although this approach is cheap and relatively easy to realize, the question of the choice of the damping elements remain. The effect of the stiffness and damping onto the minimum speed at which the rotor reaches stability is not obvious.

From a modeling point of view, the damper responsible for the introduction of the non rotating damping c_s (Fig. 1b) shares the same displacement with the mass of the statoric part m_s. In order to define the parameters m_s, c_s, k_s properly, it is necessary to include these features into the model of a Jeffcott rotor running on electrodynamic bearings.

The model of this system can be built making reference to Fig. 1b and Eq. 1. It can then be used to find optimal values for the stiffness k_s and damping coefficient c_s in order to obtain the minimum stabilizing rotation speed.

The equation of motion at the base of such system is obtained considering that points O' and C of Fig. 1b are the geometrical centers of the magnetic field and of the rotating conductor, respectively. O is the center of the inertial reference frame (O, x, y). O', x', y' is a translational reference frame fixed to the magnetic field, which moves with the stator remaining parallel to (O, x, y). O', ξ, η is a rotational reference frame fixed to the center of the magnetic field and rotating with the rotor with angular speed Ω.

The displacement of point C in the inertial reference frame is

$$z_c = z_s + z_c' \tag{2}$$

where $z_s = x_s + jy_s$ is the position of point O' while $z_c' = x_c' + jy_c'$ represents the position of point C relative to the frame O', x', y'. It must be noticed that the displacement ζ_c relative to the frame O', x', y' is:

$$\zeta_c = z_c' e^{-j\Omega t}. \tag{3}$$

The equation of motion of the stator mass m_s is then

$$m_s \ddot{z}_s + c_s \dot{z}_s + k_s z_s = F_z + F_{ext}. \tag{4}$$

The equations of motion of the stator and rotor can be solved together with the state equation of the bearing by taking into account of Eq. 2 and Eq. 3 and that the electromechanical interaction occurring in the EDB can be modeled by springs k_r in series to viscous dampers c_r. As shown in Fig. 1b, they are rotating with the rotor speed Ω. References [16, 18] present a deeper discussion about modeling and design of rotors supported by EDBs.

These equations describe the dynamics of a Jeffcott rotor supported by EDBs and stabilized by the introduction of non rotating damping between the bearing stator and casing of the machine. Solving them in the Laplace domain it is possible to find out the transfer function z_c / F_{ext} between the external force disturbance and the rotor position. The denominator of the transfer function is

$$s^5 + a_4(\Omega)s^4 + a_3(\Omega)s^3 + a_2(\Omega)s^2 + a_1(\Omega)s + a_0(\Omega) = 0, \tag{5}$$

with

$$\begin{aligned}
a_4(\Omega) &= c_s / m_s + k / c - j\Omega \\
a_3(\Omega) &= k / m + k / m_s + k_s / m_s + c_s / m_s (k / c - j\Omega) \\
a_2(\Omega) &= kc_s / (m_s m) + k_s k / (m_s c) - (k / m + k / m_s + k_s / m_s)\, j\Omega \\
a_1(\Omega) &= k / m(k_s / m_s - j\Omega c_s / m_s) \\
a_0(\Omega) &= -j\Omega k_s k / (m_s m).
\end{aligned}$$

Equation 5 has been considered to analyze the stability of the system. The stable levitation is achieved when all roots of Eq. 5 are in the left half of the complex plane. The roots move according to the rotation speed Ω, and the system is stable as soon as the rotation speed is above a threshold value $\Omega > \Omega_s$. The value of Ω_s depends on the various parameters of the system, and is very difficult to obtain analytically. A numerical procedure in which the rotation speed is increased until the system becomes stable has been adopted to study the stability.

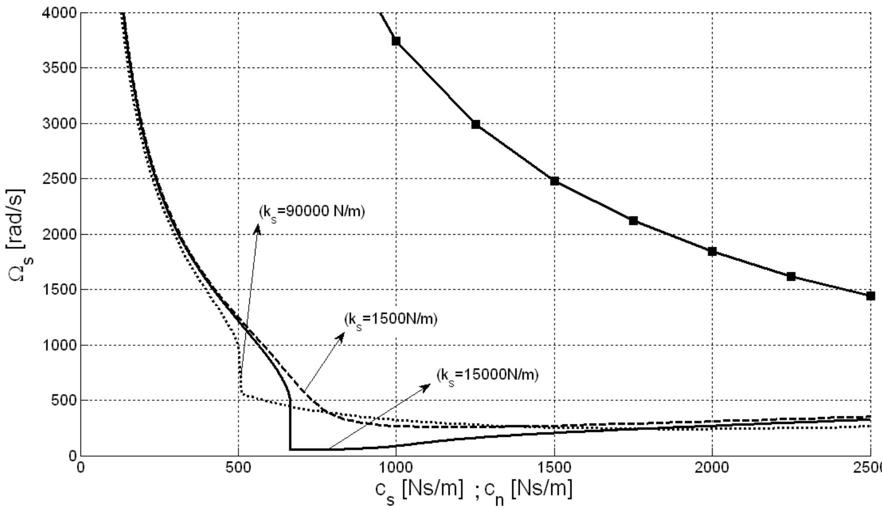

Figure 2: Comparison between two stabilization methods considering the same value of added damping in both cases. Non rotating damping between stator and rotor (solid line with square markers); damping between stator and case (three other lines on the lower left part).

Comparison Between Stabilization Strategies. The behavior of the threshold speed is studied considering the data of a real case study presented in literature by Filatov *et al* [11]. The numerical values for the coefficients in the model are $m_r = 68$ kg of rotor mass, $k = 545$ kN/m and $c = 900$ Ns/m of EDB stiffness and damping coefficients.

The model of a rotor stabilized by non rotating damping between the rotor and the stator is obtained under the same assumptions presented in the previous section, except for a non rotating damping force $F_{cn} = c_n \dot{z}_c$ that has been added in Eq. 1. The behavior of the stabilization speed Ω_s is studied for different values of non rotating damping c_n.

The effect on the stability threshold Ω_s of the spring stiffness k_s and the viscous damping factor c_s is investigated considering a fixed value for the mass of the bearing stator ($m_s = 1$ kg). The simultaneous influence of k_s and c_s is analyzed in order to identify the values that minimize the threshold speed Ω_s.

Figure 2 show the results for both configurations. The solid line with square markers refer to the classical configuration (non rotating damping c_n between the rotor and the stator) and the three other lines refer to the new configuration for three representative values of stiffness k_s. The graph underlines that the proposed configuration allows to obtain threshold speeds Ω_s much lower than those obtained using the classical method for the same value of non rotating damping. It also evidences the small effect of the spring stiffness k_s for low and high values of damping c_s.

Experimental Validation

In this section the experimental work carried out to validate the model of a rotor supported by EDBs and the new stabilization system proposed in the paper is described.

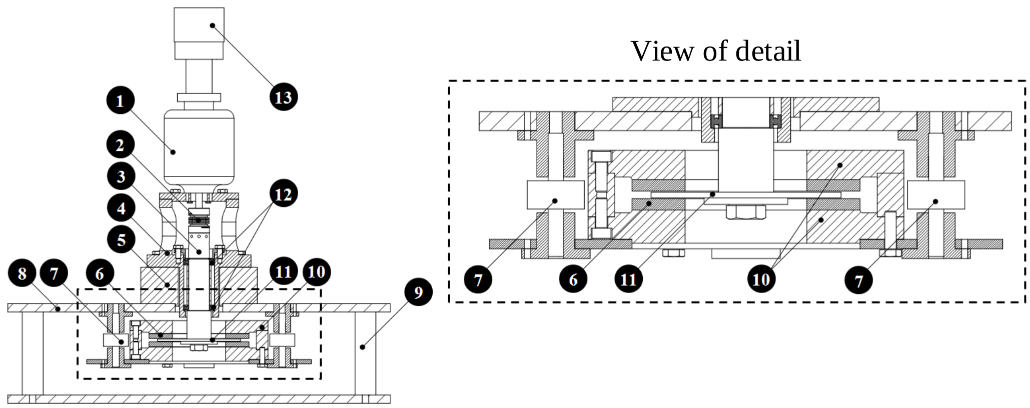

Figure 3: Description of the test rig for dynamic testing of electrodynamic bearings.

The analytic modeling of the test rig is presented first and the comparison between analytic and experimental results is used to validate the analytic model of the EDB in dynamic conditions.

The tests are addressed to measure the vibratory motion of the statoric part of the magnetic circuit connected to the ground by a spring/damper element when the axis of the rotor is held fixed.

Description of the Test Rig. The test rig of Fig. 3 has been devised to validate the dynamic performance of the EDB.

It is composed by three main parts: a) base plates (casing of the machine), b) statoric part of the EDB and c) rotating part including the conducting disc of the EDB.

The rig structure is composed by parts 4, 5, 8 and 9. The rotating part is composed by parts 2, 3 and 11 (conducting disc of the EDB). The rotating part is put in rotation by a brushed electric motor (1) and supported by two ball bearings (12). Both the rig structure and the rotating part are very stiff elements and do not introduce relevant effects in the electrodynamic bearing system. The statoric part of the EDB is composed by parts 6 (permanent magnets) and 10 (back iron). These parts are connected to the casing of the machine by means of four elastomeric mounts (7). These supports introduce damping and stiffness between parts a) and b) listed above.

Analytic Model. The rotating shaft and the ball bearings that support it have been designed to be stiff enough to avoid any flexural critical speed in the speed range of interest. The rotating part has been modeled as a rotating rigid body on rigid supports.

The statoric part of the bearing is modeled as a mass m_s connected to the ground by a mechanical spring k_s and a viscous damper c_s, that model the behavior of the rubber element supporting the stator. The interaction between the conductor and the magnetic circuit has been modeled with the series of rotating stiffness and damping k_r and c_r according to what presented in [18]. Figure 4 shows a scheme of the model.

The resulting equation of motion is the same as Eq. 4. Considering that point C is now constrained by the two ball bearings ($z_c' = 0$), the transfer function between the external force F_{ext} and the stator acceleration $\ddot{z}_s$ is

$$\frac{\ddot{z}_s}{F_{ext}} = \frac{s^3 + b_2(\Omega)s^2}{a_3 s^3 + a_2(\Omega)s^2 + a_1(\Omega)s + a_0(\Omega)} \tag{6}$$

where

$$\begin{aligned}
&a_3 = m_s \\
&a_2(\Omega) = m_s \omega_{RL} - m_s j\Omega + c_s \\
&a_1(\Omega) = c_s \omega_{RL} + k_s + k_r - c_s j\Omega \\
&a_0(\Omega) = k_s \omega_{RL} - k_s j\Omega - k_r j\Omega \\
&b_2(\Omega) = \omega_{RL} - j\Omega\,.
\end{aligned}$$

This transfer function can be used to analyze the dynamic behavior of the system. The validation is carried out by measuring the frequency response function between the same input and output during impact tests.

Testing Procedure. During the experiments the test rig was equipped with one accelerometer (PCB piezotronics 353B18 frequency range up to 10 kHz) positioned on the statoric part of the EDB in radial direction. The rotation speed is measured by an optical encoder (part 13 in Fig. 4) connected to the shaft of the motor. Impact tests were realized by hammering the statoric part of the EDB with an instrumented hammer (PCB piezotronics 086C03 frequency range up to 8 kHz). For this type of tests the rig structure has been locked to a 300 kg seismic mass supported by air springs in order to insulate the test rig from external vibrations.

Accelerations on radial direction and the hammer impact force are measured and the frequency response function between them is calculated by the LMS Scadas III digital signal analyzer. Time responses were also acquired for various rotation speeds to evaluate experimental *root loci* points. In this way the stability of the system can be studied for different values of rotation speed.

Results. The calibration of the analytic model was performed using the experimental data at rotation speed of 0 rpm. The parameters k_s and c_s were identified as the values that give the best fit between experimental and analytic frequency response functions. These values were kept constant for the study at higher rotation speeds. The values of the parameters of the model are $k_s = 342000$ N/m, $c_s = 100$ Nm/s, $m_s = 7$ kg, $k = 38400$ N/m and $c = 63.75$ Nm/s.

Figure 5a shows the plots of the FRFs obtained experimentally (solid lines) and those obtained with Eq. 6 (dashed lines). In the figure, the plots of the FRF for three representative values of rotation speed (0, 900 and 2100 rpm) are almost the same. The analytic response for the three values of speed has also been introduced in the graph.

All the curves are almost superimposed due to the amount of damping introduced by the stabilization system. If the damping introduced was lower, the curves at different rotating speed would tend to present peaks of different amplitudes.

Figure 5b presents the root loci plot. Four experimental roots, represented by the circular markers, were obtained by analyzing the time response of the system at rotation speeds of 600, 1200, 1800 and 2400 rpm.

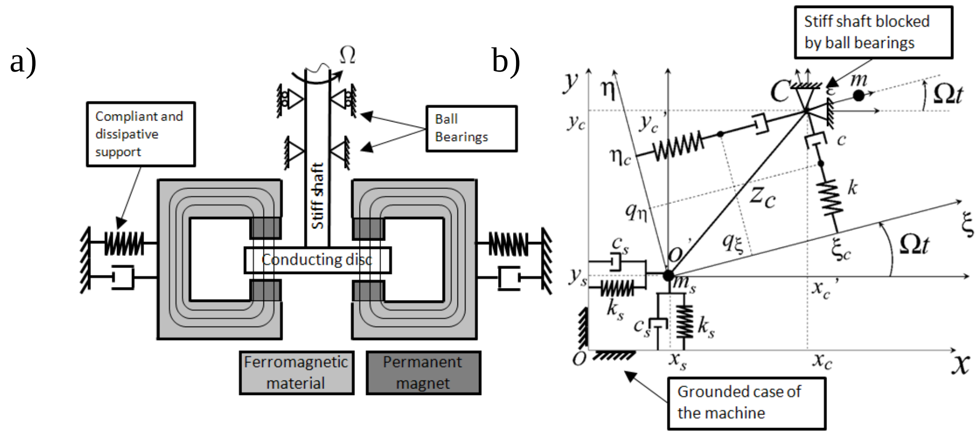

Figure 4: a) Scheme of the test rig, rotating shaft rigidly connected to the ground and magnetic circuit supported by compliant and dissipative elements between. b) Representation of parameters of the analytic model of the test rig.

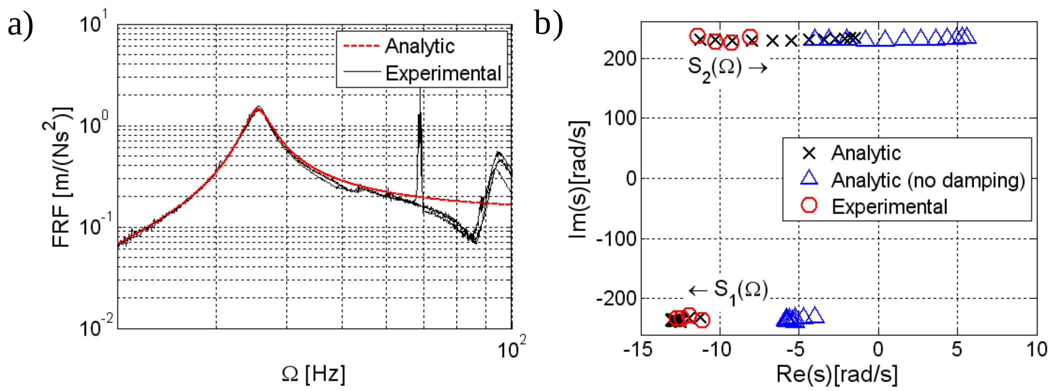

Figure 5: a) Comparison between experimental and analytic FRF for the new stabilization technique. b) Root loci plot of the main poles of the test bench.

The analytic results obtained using the same values of k_s and c_s identified previously are represented by the 'x' markers.

It is possible to note the good correlation between experimental and analytical points in both cases (FRF and root loci) and this is a proof of the validity of the model. The analytic model is used to predict the behavior at higher rotational speeds.

It is possible to predict the behavior of a system characterized by the same value of k_s and with $c_s = 0$. In this case the system becomes unstable for a rotating speed which coincides with the natural frequency of the stator. This behavior is represented in Fig. 5b by the triangular markers.

The comparison between the damped and undamped *root loci* highlights the stabilizing role played by the non rotating damping c_s. It's worth noticing that for the damped case the roots of the system never enter in the right side of the complex plane, so that the stability of the system is guaranteed.

Conclusions

The paper is dedicated to the study of passive stabilization of electrodynamic bearings. A new stabilization technique that consists in the addition of a dissipative element in parallel to a compliant one placed between the non rotating part of the EDB and the case of the machine is presented and studied. A comparison between the classical stabilization method (introduction of damping between rotor and stator by electromagnetic means) and the new technique is presented. The comparison shows that the second option leads to a relevant reduction of the stability threshold for the same value of added damping. Furthermore it allows a reduction on the rotor mass and complexity, leading to simpler and more effective systems.

The effectiveness of the new stabilization technique is validated by means of experimental tests. The very good correspondence between experimental results and the analytic model shows the validity of the solution and of the underlying models. The results show also that the new strategy is capable of effectively introducing non rotating damping into the system allowing to solve the stability problem of EDBs.

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