Damping Strategies on a Horizontal Rotor Supported by Electrodynamic Bearings

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***Abstract*—Electrodynamic Bearings (EDBs) exploit Lorentz type of magnetic force due to the eddy currents induced in a rotating conductor, which has relative motion in a magnetic field, to levitate the rotor. The character of passive levitation without negative stiffness that can be realized using permanent magnets in room temperature with relatively simple configuration makes EDBs an appealing alternative to Active Magnetic Bearings (AMBs). However, the instability of rotors supported by EDBs is the key issue that prevents the application of EDBs. In this paper, the damping strategies for EDB systems are investigated using additional AMBs implemented. The hybrid bearing system has been built to stabilize EDB system using damping from AMBs which work as Active Magnetic Dampers (AMDs). It can be further exploited to investigate the damping strategies with active control of the AMBs. The results are presented in this paper showing different possible solutions of EDB stabilization.**

I. INTRODUCTION

A. *Electrodynamic Bearings*

Electrodynamic suspension (EDS) exploit repulsive forces (Lorentz forces) due to eddy currents to produce levitation. Different configurations of EDS have been developed since the eddy currents can be induced either by using alternating current driven electromagnets or by the relative motion between a conductor and a constant magnetic field. The same theory can be applied in rotating systems to realize electrodynamic magnetic bearings. Electrodynamic bearings (EDBs) levitate rotors using Lorentz forces mainly due to the relative motion between a conductor and a magnetic field, thus they are passive type of MBs and able to produce positive stiffness without introducing negative stiffness in any direction [1]. The characters of EDBs make them a promising alternative to AMBs, provided that stable operation can be obtained.

B. *Stabilization of EDBs*

In fact, the main issue that prevents the application of EDBs is the instability due to rotating damping arising in the conductor. It is well known that rotating damping causes instability in supercritical speed range. Different methods have been proposed to solve this problem, mainly with passive dampers. Eddy current damper was introduced in [2] to compensate the rotating damping in EDB system implemented in a flywheel energy storage system. This method requires conductor on the stator and attaching permanent magnet (PM) on the rotor. The brittleness of most PMs limits the application subject to strong centrifugal stress at high rotating speed. Another passive solution proposed in [3] is to introduce damper between the statoric part of the bearing and the case of the machine instead of introducing damping between the rotor and the statoric part of the bearing. This method allows a reduction of the rotor mass and complexity compared to the previous one. However, the effectiveness of these two solutions are limited according to the rotordynamic analysis in [4].

Alternatively, an active solution has been proposed by the authors [5] [6] to stabilize the rotor supported by EDBs. This solution combines AMDs with EDBs to exploit the rotordynamic control capacity of AMBs to provide proper non-rotating damping to the rotor. The functionality of the EDB-AMD hybrid bearing system has been demonstrated in [5]. In the present paper, by tuning the control of AMDs, the same system is exploited to investigate the damping strategies of EDB systems. Alternative solutions like passive and semi-active magnetic damping are discussed.

II. SYSTEM CONFIGURATION AND MODELING

The configuration of the horizontal rotor EDB–AMD system has been described in detail in [5], as illustrated in Fig. 1. It consists of two radial homopolar EDB units (a), two radial AMDs (b) and one electric motor (c) at the center of the rotor.

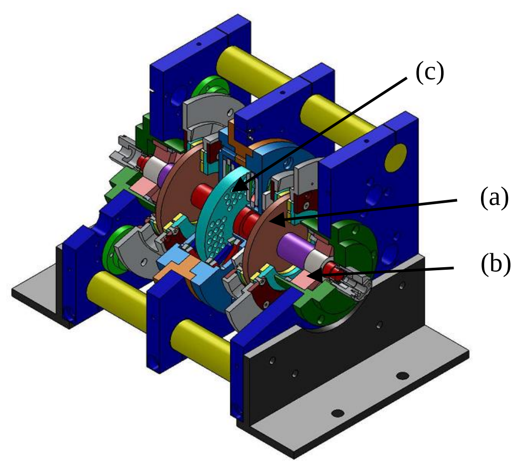

Figure 1. Configuration of the EDB-AMD horizontal rotating system.

Based on the double flux EDB system [6], in conjunction with classic radial AMBs (Fig. 3), a full-size test rig (Fig. 2) has been built to demonstrate the performance of the hybrid bearing system and to investigate the damping strategies for

EDBs. The parameters description of the test rig has been presented in [6]. Some main parameters are listed in Table I.

TABLE I. MAIN PARAMETERS OF THE TEST RIG

	Parameter	Value
ω	Nominal Speed	20000 rpm
m_r	Rotor mass	4.35 kg
l_r	Rotor length	305 mm
J_p	Polar moment of inertia	0.00572 kgm^2
J_t	Transversal moment of inertia	0.01995 kgm^2
a_1	EDB$_1$ arm	67 mm
a_2	EDB$_2$ arm	67 mm
b_1	AMB$_1$ arm	125 mm
b_2	AMB$_2$ arm	125 mm

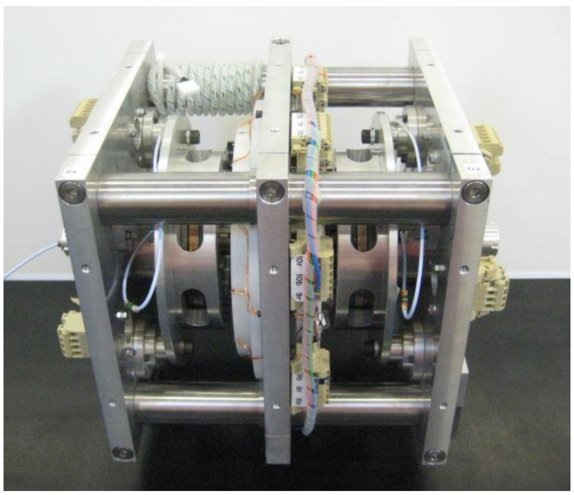

Figure 2. The full-size test rig.

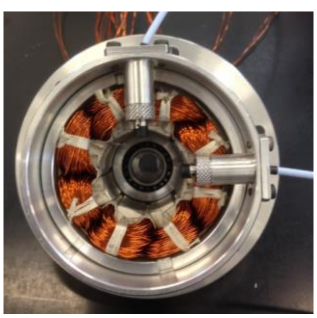

Figure 3. Radial AMB actuator.

The structure of the radial double flux homopolar EDB is depicted in Fig 4. It consists of a conductor disc mounted on the rotor and placed in an axisymmetric magnetic field. The working principle of such kind of radial EDB is that if the rotor is moving off the center while rotating, eddy currents will be induced due to the relative motion between the rotor and the magnetic field, thus restoring forces are produced because of the interaction between the eddy currents and the magnetic field. The geometric parameters of the EDB are shown in Table Ⅱ.

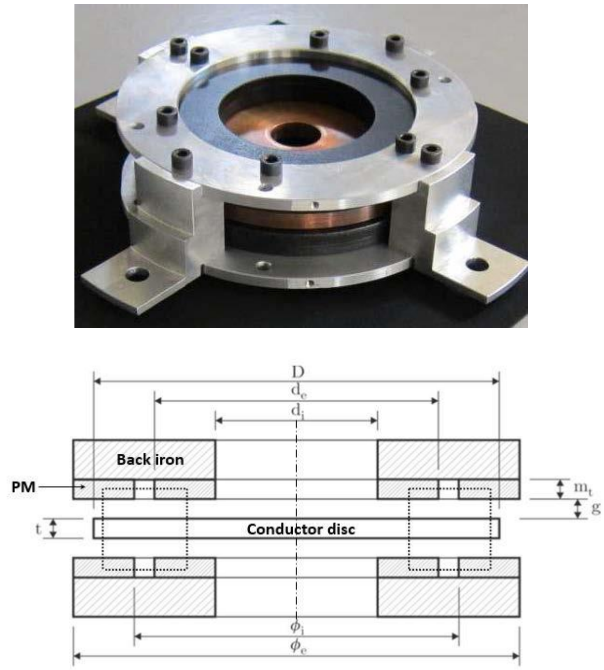

Figure 4. The double flux radial EDB and its structure.

TABLE II. GEOMETRIC PARAMETERS OF THE EDB

	Parameter	Value
D	Conductor outer diameter	120 mm
de	Inner permanent magnet outer diameter	89 mm
di	Inner permanent magnet inner diameter	64 mm
mt	Permanent magnet thickness	5.5 mm
t	Conductor disc thickness	8 mm
g	Axial air gap	0.75 mm
Φe	Outer permanent magnet outer diameter	120 mm
Φi	Outer permanent magnet outer diameter	95 mm

A. Working theory

In such a hybrid system, the AMDs can be used as AMBs to guarantee suspension for speeds below the EDB's stability threshold or for speeds where EDBs are not able to provide sufficient levitation forces. They can also act as dampers for higher speeds ensuring a stable levitation. PID architecture (Fig. 5) is implemented in the position control of AMBs/AMDs to provide proper stiffness and damping. With PID control, the value of the rotor eccentricity can be controlled, thus the performance of the EDBs that work on the same rotor can be controlled consequently. It gives the hybrid system capacity to be exploited to characterize the EDBs. Because the EDBs provide restoring force only when the rotor rotates with an eccentricity or under another disturbance force, if the rotor is levitated by the AMBs and perfectly centered in the magnetic field of the EDBs, EDBs will not contribute to levitation. Therefore, to allow the EDBs work properly, the integral term in the controller should be deactivated. PD control is used for the operation of the hybrid bearing system. By tuning the controller of the AMBs, the damping strategies of EDBs can be analyzed.

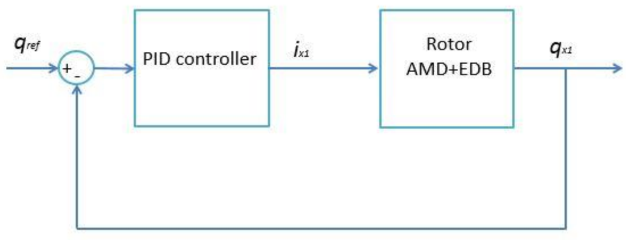

Figure 5. Close loop position control of the EDB-AMD rotating system.

B. Modeling

The analytical model of the EDB-AMD rotating system consists of a 4 degrees of freedom (DOF) model of the rotor supported by the combination of electrodynamic force and AMBs. The dynamic equations of a 4 DOF rotor can be represented as:

$$M\begin{Bmatrix}\ddot{q}\\ \ddot{\phi}\end{Bmatrix} - j\Omega G\begin{Bmatrix}\dot{q}\\ \dot{\phi}\end{Bmatrix} + \begin{Bmatrix}F_b\\ M_b\end{Bmatrix} = \begin{Bmatrix}F_{ext}\\ M_{ext}\end{Bmatrix} \quad (1)$$

where M and G are the mass and gyroscopic matrices:

$$M = \begin{bmatrix} m & 0\\ 0 & J_t\end{bmatrix}, G = \begin{bmatrix} 0 & 0\\ 0 & J_p\end{bmatrix} \quad (2)$$

The equations are written in two complex coordinates: q and ϕ, which represent the transverse and angular displacements respectively:

$$\begin{aligned} q &= x + iy\\ \phi &= \phi_y - j\phi_x \end{aligned} \quad (3)$$

F_b and M_b are the forces and moments acting on the rotor from bearings. In the hybrid bearing system, contributions from both EDBs and AMDs should be considered. Fig. 6 is the scheme of the bearing forces acting on the rotor. It shows the 4 DOF rotor model and the relative positions of EDBs and AMDs.

The EDB force generated by the double flux configuration can be represented with two parallel sets of spring and damper, as shown in Fig. 7. Thus, the double flux EDB is characterized by the parameters k_1, c_1, k_2 and c_2. The parameters have been characterized in [7]. The EDB force in quasi-static conditions can be written as:

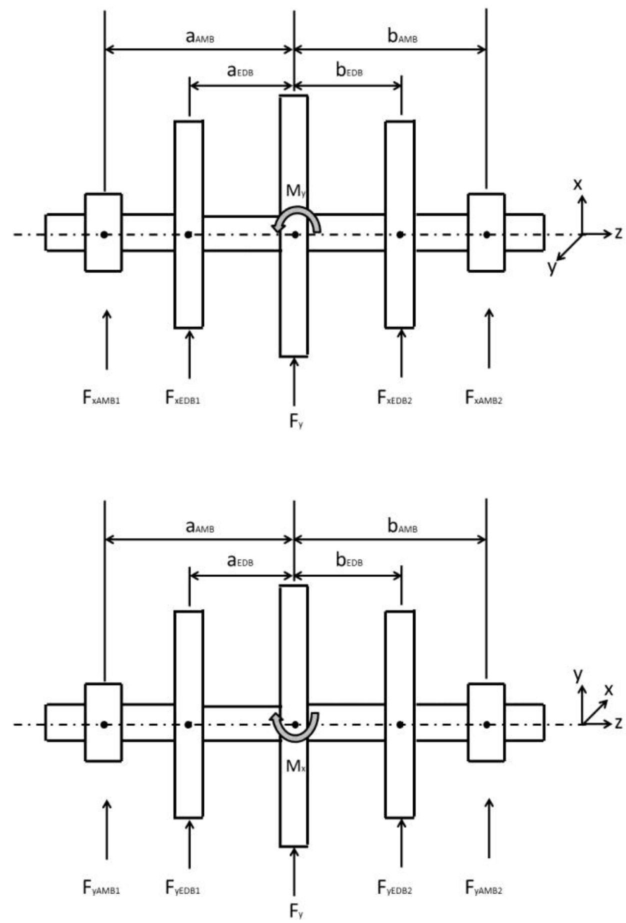

Figure 6. Schematic of the rotating system with hybrid bearings. The mechanical equivalence of both EDBs and AMDs are illustrated.

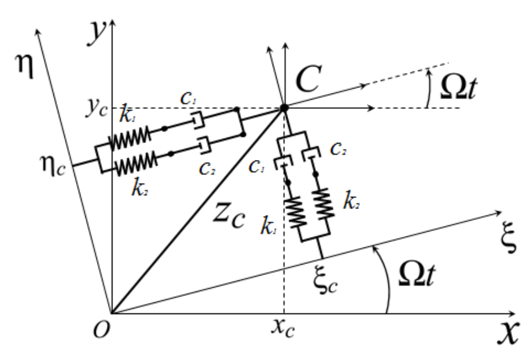

Figure 7. Mechanical equivalent of the double flux EDB.

$$F = \left(\frac{k_1}{1+(\frac{\omega_{RL1}}{\Omega})^2} + \frac{k_2}{1+(\frac{\omega_{RL2}}{\Omega})^2}\right)z_{c0} - j\left(\frac{c_1\Omega}{1+(\frac{\Omega}{\omega_{RL1}})^2} + \frac{c_2\Omega}{1+(\frac{\Omega}{\omega_{RL2}})^2}\right)z_{c0} \quad (4)$$

$$= F_{\parallel} + F_{\perp}$$

The linearized equations of the AMD actuators in the fixed x and y directions can be represented as:

$$\begin{Bmatrix} F_{xAMD} \\ F_{yAMD} \end{Bmatrix} = \begin{bmatrix} k_s & 0 \\ 0 & k_s \end{bmatrix} \begin{Bmatrix} x_{AMD} \\ y_{AMD} \end{Bmatrix} + \begin{bmatrix} k_i & 0 \\ 0 & k_i \end{bmatrix} \begin{Bmatrix} i_{Cx} \\ i_{Cy} \end{Bmatrix} \quad (5)$$

where the variables x_{AMD}, i_{Cx}, y_{AMD} and i_{Cy} indicate respectively the displacements and the control currents in the two directions x and y of the actuator action plane, k_s and k_i are respectively the displacement stiffness and current stiffness of the AMDs.

The complete numerical model of the EDB-AMD rotating system has been built and validated in [6]. A corresponding Simulink model has been built for simulations of the rotating system's performance.

III. Stability Analysis and Damping Strategies

The EDB-AMD configuration is an ideal platform for dynamic analysis of the rotor supported by EDBs, since the rotordynamics can be monitored and controlled. The system performance analysis using PD control of AMDs has been performed for different control parameters. Stability analysis has been made in [6] for the control with constant derivative parameter tuning the proportional gain, which did not delicately split the effects of damping and stiffness from AMDs. In this section, the effects of both non-rotating damping and stiffness from AMDs are investigated.

A. Stability analysis

The validated numerical model of the complete system can be exploited to determine and to demonstrate the dynamical behavior and performances of the system. Root locus analysis is an efficient method to investigate the rotordynamic stability when tuning AMD control parameters. The real and imaginary parts of the calculated eigenvalues are plotted in the complex plane, where the sign of the real part decides stability. Negative real part confirms asymptotic stability whereas a non-negative value indicates instability.

The transfer function of the PD position controller in Laplace form is (6), where K_p is the proportional gain, K_d and T_d are the derivative gain and the derivative time constant, respectively.

$$\frac{i_{cx\_AMD}}{\varepsilon_{x\_AMD}} = K_p + \frac{K_d \cdot s}{1 + \frac{T_d}{N} \cdot s} \quad (6)$$

The closed-loop stiffness k_d and non-rotating damping c_n introduced by each AMD can be expressed as (7).

$$\begin{aligned} k_d &= K_p \cdot k_i + k_x \\ c_n &= K_d \cdot k_i \end{aligned} \quad (7)$$

Thus, a simplified mechanical model of the hybrid system can be built, as shown in Fig. 8, where the AMDs are represented by a spring (k_d) and damper (c_n) connected in parallel. Stability analysis of the simplified mechanical model will be performed to see the amount of stiffness and non-rotating damping required in the system for stable operation.

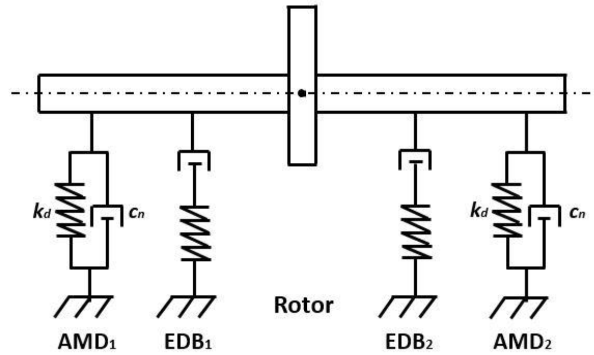

Figure 8. Schematic of the rotating system with hybrid bearings. The mechanical equivalence of both EDBs and AMDs are illustrated.

A stability map is plotted in Fig. 9 in terms of k_d and c_n for the speed range of 0 to 6000 rpm. The region above the stability line (solid) represents safe zone to keep the rotor stable in the testing speed range. This map offers possibility to choose the suitable values of k_d and c_n in the AMDs. It can be exploited to define the control strategies of the EDB-AMD hybrid bearing system. Instead of setting constant control parameters, an optimized controller with varying control parameters can be obtained to reduce the unnecessary contribution of AMDs.

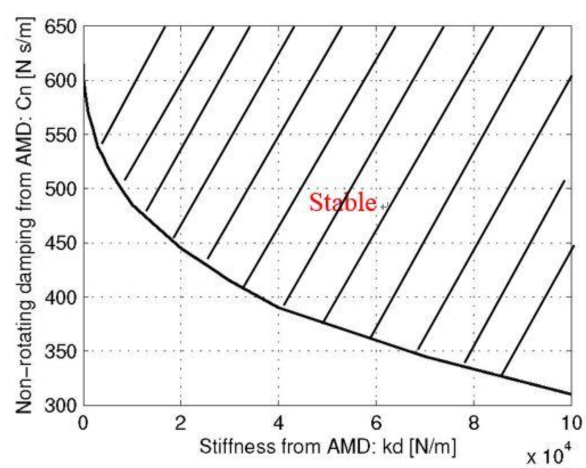

Figure 9. Stability map in terms of k_d and c_n.

B. Simulating/Realizing viscous dampers

If the closed-loop stiffness k_d equals to 0, which means AMDs do not introduce positive stiffness and their inherent negative position stiffness are compensated, AMDs provide only the non-rotating damping c_n thus behave as pure dampers. In this case, AMDs can be exploited to realize or "simulate" general viscous dampers. Thus, the AMDs in the model are replaced with simple viscous dampers, as shown in Fig. 10.

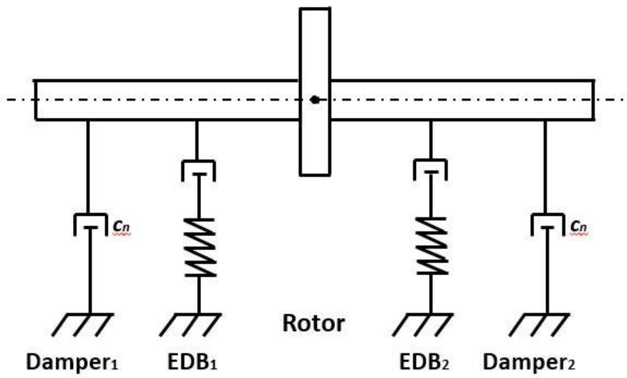

Figure 10. Schematic of the rotating system with EDBs amd dampers.

Using the controlled hybrid bearing system, the system dynamic behavior can be analyzed to estimate possible damping solutions. The analytical model of this system can be obtained with modifications on the existing EDB-AMD system model. The state-space model of the rotor coupled with dampers is:

$$\begin{Bmatrix} \ddot{x} \\ \ddot{\phi}_y \\ \ddot{y} \\ \ddot{\phi}_x \\ \dot{x} \\ \dot{\phi}_y \\ \dot{y} \\ \dot{\phi}_x \end{Bmatrix} = \begin{bmatrix} -M^{-1}\Omega G - M^{-1} c_n T_{AMD} R_{AMD} & 0_{4\times4} \\ I_{4\times4} & 0_{4\times4} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{\phi}_y \\ \dot{y} \\ \dot{\phi}_x \\ x \\ \phi_y \\ y \\ \phi_x \end{Bmatrix} + \begin{bmatrix} -M^{-1} \\ 0_{4\times4} \end{bmatrix} \begin{Bmatrix} F_x \\ M_y \\ F_y \\ M_x \end{Bmatrix}$$

$$\begin{Bmatrix} \dot{x} \\ \dot{\phi}_y \\ \dot{y} \\ \dot{\phi}_x \\ x \\ \phi_y \\ y \\ \phi_x \end{Bmatrix} = \left[I_{8\times8} \right] \begin{Bmatrix} \dot{x} \\ \dot{\phi}_y \\ \dot{y} \\ \dot{\phi}_x \\ x \\ \phi_y \\ y \\ \phi_x \end{Bmatrix} + \left[0_{8\times4} \right] \begin{Bmatrix} F_x \\ M_y \\ F_y \\ M_x \end{Bmatrix} \tag{8}$$

Where

$$\begin{Bmatrix} F_x \\ M_y \\ F_y \\ M_x \end{Bmatrix} = \begin{Bmatrix} F_{xEDB} + F_{x\_ext} \\ M_{yEDB} + M_{y\_ext} \\ F_{yEDB} + F_{y\_ext} \\ M_{xEDB} + M_{x\_ext} \end{Bmatrix} \tag{9}$$

The minimum amount of non-rotating damping c_n required for system stability at different speeds have been calculated and plotted in Fig. 11. It shows the relationship between the amount of non-rotating damping from each damper and the corresponding stability threshold of rotational speed. With this reference, it is possible to design a proper damper in either active or passive way to keep the rotor in stable operation within the desired speed range.

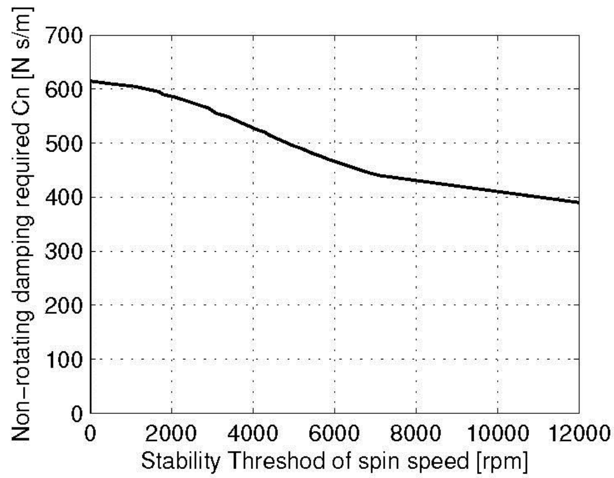

Figure 11. The amount of non-rotating damping introduced by each damper and the corresponding stability threshold of rotational speed.

This result is remarkable because, the EDB-AMD test rig allows realizing experimental tests synthesizing not only different amount of non-rotating damping but also to modify it according to speed and frequency. It means that the EDB-AMD system can allow investigating, in a controlled laboratory environment, a damping strategy that could then find also a passive implementation.

Alternative solutions that can provide non-contact and non-rotating damping can be determined accordingly. The suitable candidate could be electromagnetic dampers either passive, active, or semi-active according to the damping requirements and other relevant design considerations.

IV. Conclusion

The EDB-AMD hybrid bearing system has been exploited in this paper to investigate the damping strategies for EDB systems. By tuning the control parameters in the AMDs, the rotordynamic stability analysis subject to different amount of stiffness and non-rotating damping has been performed. This work is meaningful due to the following two aspects. First, the stability map including the effects of both stiffness and non-rotating damping from AMDs can lead to an optimized control strategy with varying control parameters depending on rotating speeds of the EDB-AMD hybrid bearing system. The second possibility it offers is to find alternative damping solutions for the stabilization of EDB systems. Numerical simulations have been performed utilizing AMDs with only damping contribution. Relevant results have been achieved showing the minimum amount of non-rotating damping required for stable operation in the desired speed range. The future work is to realize the damping strategy in the controlled test rig and determine alternative damping solutions accordingly.

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