Optimization of a new radial electrodynamic damper

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***Abstract*—The study of electrodynamic dampers is an important point when working with electrodynamic bearings. These dampers may also be used in combination with reluctant passive magnetic bearings. This paper proposes a new design of a robust and inherently stable radial electrodynamic damper with a homopolar axial magnetic flux. An optimization, based on finite element models, of this new bearing topology is made. For this to be possible, a 2-D representation of the damper is developed equivalent to the 3-D damper. The results of the optimizations are presented and a discussion of the influence of some of the parameters is done.**

I. Introduction

High speed electrical drives are a promising technology [1]. Fast spinning and high efficiency electrical drives may be obtained using for instance permanent magnet drives, and thus avoiding excitation losses. These high speed electrical drives need adapted bearings. One remaining challenge is to design high speed bearings with a high efficiency, a long lifetime and simple to use. Developped solutions concern magnetic bearings and gas bearings. However, gas bearings do present poor dynamic stability and need additional damping [2]. On the other hand active magnetic bearings lead to more complex, less compact, and possibly more energy consuming designs. Passive magnetic bearings could be a solution but they also suffer from rotor dynamic stability issues: passive magnetic bearings based on the reluctant principle have almost no intrinsic damping [3] and passive magnetic bearings based on the electrodynamic principle may also present a completely unstable behavior when the damping present in the system is not well managed [4]. Another solution for stabilizing electrodynamic bearings has been investigated in [5] and consists in adding dissipative elements between the statoric part of the bearing and the ground. In this context, when wishing to work with passive or electrodynamic bearings, the study of dampers is an important part of the conception of the bearing.

When designing electrodynamic dampers, attention has to be paid to stability issues, just as for electrodynamic bearings. Indeed, [6] observes that electrodynamic dampers using conductors spinning with the rotor show an unstable behavior at high spin speeds. They offer a first analytical approach to understand those phenomena and make experimental evaluations of it. In [7], a general parameterized electromechanical model, based on a simple analogy between the resistive-inductive dynamics of eddy currents and the spring-damper in series dynamics, is described. [8] presents a general parameterized electromechanical model for the radial behavior of any kind of magnetic system subject to eddy currents. It is shown that for stability issues, when the intended effect is to produce damping, this damping should be generated by the rotor whirling motion and not the spinning motion.

Electrodynamic dampers are based on the interaction between a magnetic field generated by a permanent magnet or an electromagnet and currents induced by the variation of this magnetic field. When the variation of the magnetic field can be generated by a time-variation of the magnetic field, one talks about a transformer e.m.f. on the conductor, and when the variation of the magnetic field is generated by a space-variation of the latter, induced for instance by the motion of the conductor relatively to the magnetic field, one talks about a motional e.m.f. A theoretical comparison between dampers based on each type of e.m.f. is carried out in [9], and it is shown that for a given magnetic field, a given volume of conductor, and assuming that there is no inductive behavior, dampers based on a transformer e.m.f. may lead to a better damping efficiency. However, it is also said that in practical cases, dampers based on a motional e.m.f. are often better suited.

Electrodynamic dampers have been developped for damping axial, radial or torsional rotor motions. They can be based on a radial or an axial magnetic flux. [10] gives an overview of all these dampers, and gives analytical tools to predict the damping, but with a focus on the case of a radial damper with a radial magnetic flux. In [11], the electrodynamic bearing is combined with a radial electrodynamic damper with axial magnetic flux. Torsional electrodynamic dampers are investigated in [12] and [13].

An example of an inherently stable electrodynamic damper, acting between the rotor and the stator, combined with a reluctant passive magnetic bearing is given in [14].

This papers proposes a radial electrodynamic damper with a homopolar axial magnetic flux, which could be used with a reluctant passive magnetic bearing, but also with an electrodynamic magnetic bearing. This damper is inherently stable thanks to its configuration, and is robust because it does not need to insert a permanent magnet on a fast spinning rotor. The effect of the insertion of teeth in the magnetic circuit of the damper is investigated. Parametric optimizations, based on finite element models, of this new bearing topology are made. For this to be possible, an original equivalent 2-D representation of the damper is developped equivalent to the

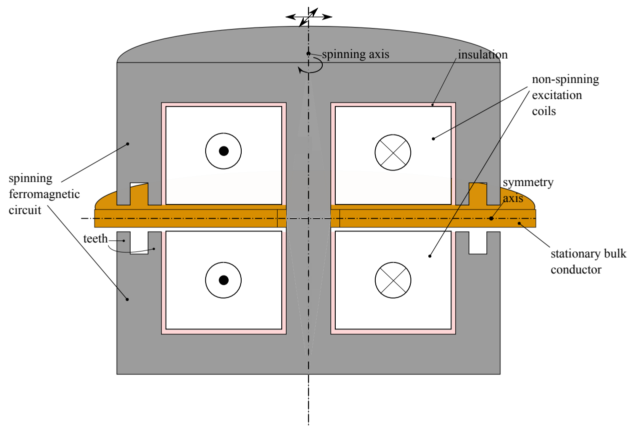


Figure 1. Schematic view of the studied electrodynamic damper.

3-D damper. The results of the optimizations are presented and discussed.

II. Structure and Principles of Proposed Electrodynamic Damper

The electrodynamic bearing studied in this paper is a radial damper with a homopolar axial magnetic flux, and is presented on Fig. 1. The conductor is a bulk copper plate linked to the stator, and is thus stationary. The axial homopolar magnetic flux is generated by an excitation winding, fixed in the airgap, and fed by a DC-current. This fixed coil magnetizes a spinning steel magnetic circuit, which allows on one side to have a robust design, as there are no brushes, and no permanent magnets to be inserted in potentially high spin speed pieces, and on another side, the possibility exists to feed the coil by a current adapted to the needed damping, so as not to generate damping, and losses, when unnecessary.
This damper topology, with a conductor fixed to the stator, is inherently stable. Indeed, when the rotor is spinning in a fixed centered or out-centered position, no forces are produced. However, when moving radially, damping will be produced.

The magnetic path is constituted of one central leg, an external leg, and an airgap. The conductor is a copper annular disk. The external leg of the magnetic circuit is terminated by teeth, uniformly distributed, each of equal width. The width of those teeth are characterized by a parameter called x_{teeth} which is worth 0 when the teeth are so large that there is no air anymore between the teeth, and worth 1 when the teeth are infinitely thin. As these teeth are only present on the rotor, they do not produce any reluctant force, but they allow for another magnetic flux distribution in the air gap, which could increase the damping.
The damper geometry is characterized by the parameters summarized in Table I, and the parameters to be optimized are represented on Fig.2.

Table I
Geometric parameters of the damper

Parameter	Description	Value
L_1	half of damper axial length	to be optimized
L_2	damper radius	to be optimized
l_1	external leg width	to be optimized
l_2	yoke width	to be optimized
l_3	internal leg diameter	to be optimized
h_{cu}	conductor thickness	to be optimized
n_{teeth}	number of teeth	fixed
x_{teeth}	teeth proportion	to be optimized
l_{teeth}	teeth width	$(1-x_{teeth})*l_1/n_{teeth}$
h_{teeth}	teeth height	$g+h_{cu}$
g	airgap between conductor and ferromagnetic circuit	$0.5\,\mathrm{mm}$
e_{insh1}	thickness of insulation between central leg and coil	$3.8\,\mathrm{mm}$
e_{insh2}	thickness of insulation between external leg and coil	$2\,\mathrm{mm}$
e_{insv}	thickness of insulation between yoke and coil	$3\,\mathrm{mm}$

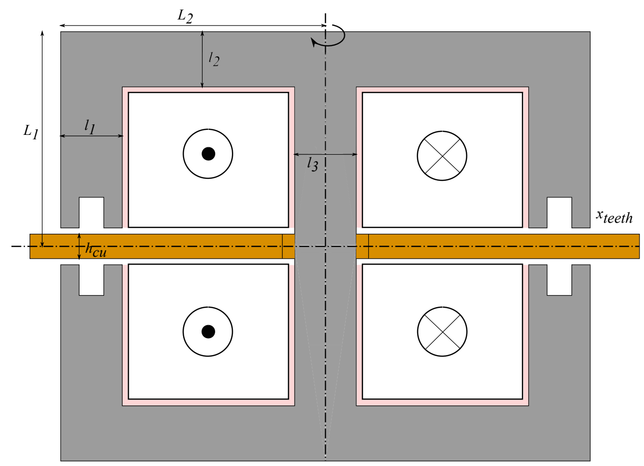


Figure 2. Geometric parameters to be optimized.

III. Optimization Approach

A. Objective function and optimization tool

The optimization is based on a genetic algorithm. The evaluation of each individual of the population is made by a finite element model, as explained below. The objective function is to optimize the damping coefficient for a given excitation speed, per damper volume unit. This can be expressed as:

$$\max_{\{L_1,L_2,l_1,l_2,l_3,h_{cu},x_{teeth}\}} \frac{\iiint \bar{J}_{induced} \times \bar{B}\ dV_{conductor}}{\bar{v}\ V_{damper}}, \quad (1)$$

where $\bar{J}_{induced}$ is the current induced in the conductor, $\bar{B}$ is the magnetic flux density, $\bar{v}$ is the excitation speed, and V_{damper} is the damper volume.

B. Equivalent representation for 2-D FE model

A way to model the damper is to make a 3-D FE model of the damper, but this is not realistic to be used for an

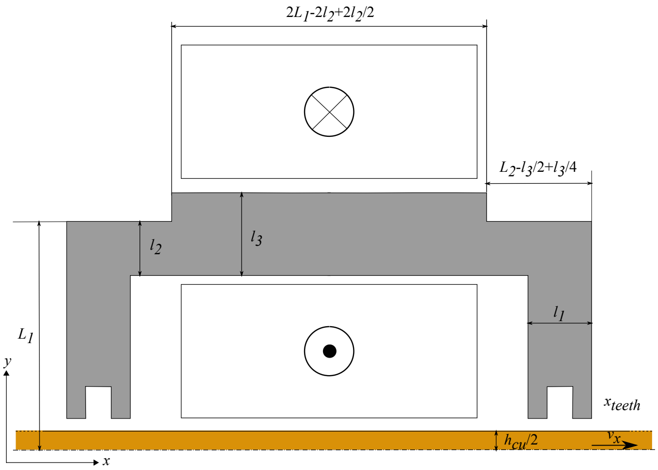


Figure 3. Equivalent 2D representation of the damper.

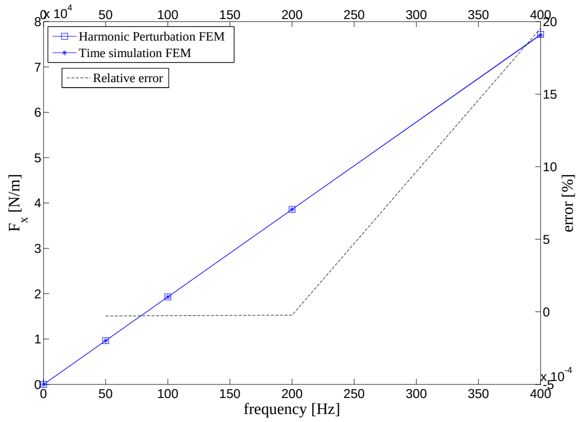


Figure 4. Evolution of the damping force as a function of the excitation frequency, for quasi-static time-simulations and for harmonic perturbation simulations.

optimization as it is too much time consuming. The damper has an axisymmetric geometry, however the excitation speed is not axisymmetric. This is why an equivalent 2-D model of the damper was made. It is obtained by unfolding the parts of the magnetic circuit above and underneath the conductor, as illustrated in Fig. 3. The geometric dimensions of this equivalent model depend on the initial geometry, and the corner reluctances are taken into account by a lengthening of the adjacent legs, based on the same idea as explained in [15]. The excitation speed is applied on the conductor.

In this model, the objective function (1) simplifies to:

$$\max_{\{L_1,L_2,l_1,l_2,l_3,h_{cu},x_{teeth}\}} \frac{\iint J_{induced,z} B_y \; dS_{conductor}}{v_x \; V_{damper}}, \qquad (2)$$

As in the real system, the moving part of the damper will have a magnitude-limited motion, we considered a sine signal of amplitude v_x for the excitation speed in the optimization. The maximum displacement is worth $0.5\,\mathrm{mm}$, and the highest considered frequency is $400\,\mathrm{Hz}$. This gives a maximum amplitude of speed worth $v_x = 1.26\,\mathrm{m/s}$.

Therefore, in the first instance, quasi-static time simulations with a sinusoidal excitation speed have been conducted. For these simulations, it is assumed that the electric time constant is much smaller than the mechanical time constant. These quasi-static time simulations are then compared to harmonic perturbation simulations, where the problem is linearized around the stationary solution with no speed. The results for both simulations are shown on Fig. 4. The difference for the highest considered frequencies comes from the fact that for those speeds amplitudes, we are not in the linear part of the force vs speed characteristics anymore, as can be observed on Fig. 5. However, for the smaller frequencies, we are effectively working in the linear part of this characteristic, and even for the higher frequencies, the error remains small: less than 0.2%. This very small difference validates the harmonic perturbation approach.

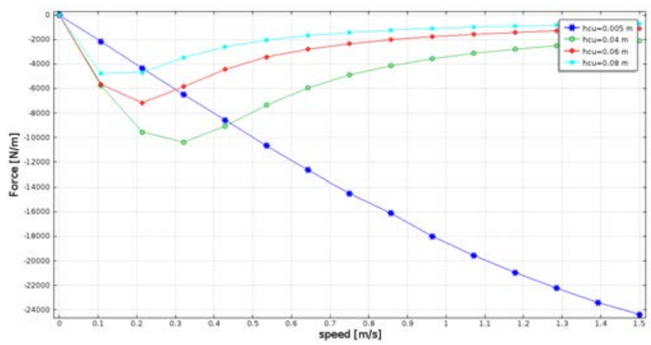


Figure 5. Evolution of the damping force as a function of the excitation speed for the optimized damper for $f = 50\,\mathrm{Hz}$, i.e. an maximum amplitude of speed worth $v_x = 0.16\,\mathrm{m/s}$.

IV. Optimization Results

The value of the optimized parameters for an excitation frequency of $100\,\mathrm{Hz}$, and 2 teeth are given in Table II, and the equivalent 2-D representation of the damper is illustrated on Fig. 6. The objective function Eq. (2) is then worth $2.66\;10^6\,\mathrm{Ns/m^5}$ for an excitation current density of $5\,\mathrm{A/mm^2}$, and the geometric dimensions (Table II) give a damper volume (see Fig. 2) of $0.021\,\mathrm{m^3}$.

Tracing the evolution of the objective function with respect to the teeth proportion, we can see on Fig. 7(a) that the optimal damper does not have teeth. Indeed, given that the damping force expression is $\bar{F} = q(\bar{v} \times \bar{B})$, there is no gain in having teeth which give a lower magnetic flux density inside the conductor.

On Fig. 7(b), the evolution of the objective function with the conductor thickness is shown. There is an optimal thickness, and this thickness is quite high. The fact that the iron reluctance is not negligible probably plays a role in this optimal

Table II
OPTIMIZED GEOMETRIC PARAMETERS OF THE DAMPER, FOR $f = 100\,\mathrm{Hz}$

Parameter	Optimized Value
L_1	173 mm
L_2	138 mm
l_1	29 mm
l_2	43 mm
l_3	95 mm
h_{cu}	51 mm
n_{teeth}	2
x_5	5%

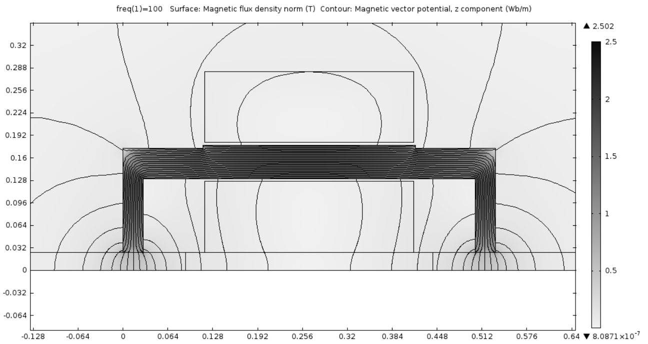
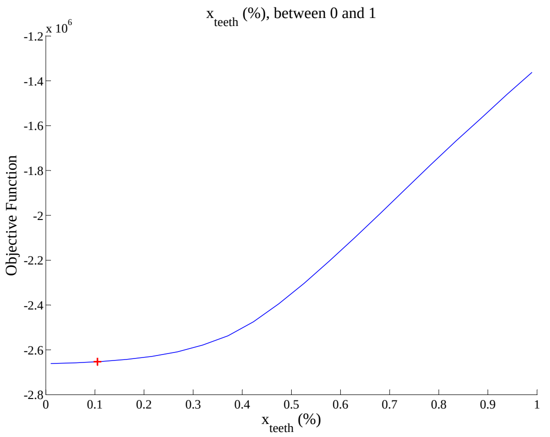
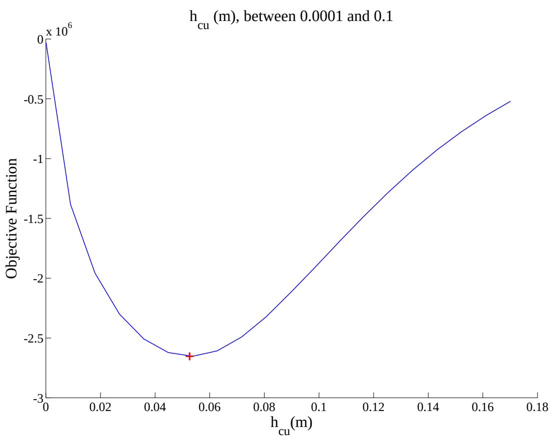


Figure 6. Equivalent 2D representation of the optimized damper, for 2 teeth and an excitation frequency of 100 Hz.

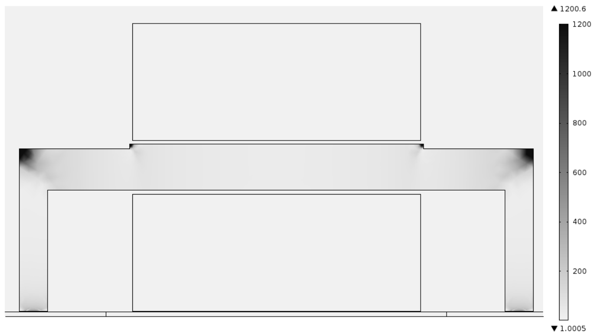


(a)

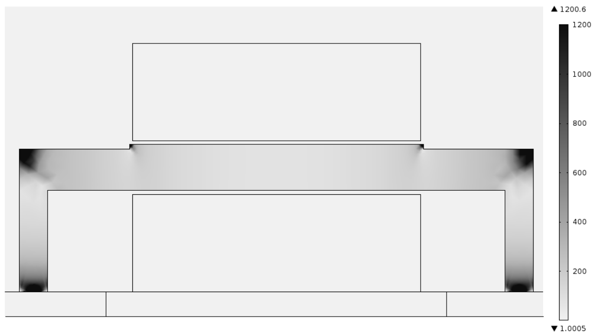


(b)

Figure 7. Evolution of the objective function as a function of the optimization parameters (a) x_{teeth} the teeth proportion (b) h_{cu} the thickness of the conductor .



(a)



(b)

Figure 8. Representation of the iron relative permeability (a) for $h_{cu} = 10\,\mathrm{mm}$ and (b) for $h_{cu} = 52\,\mathrm{mm}$

value: when the copper thickness decreases, the iron becomes highly saturated, as can be seen on Fig. 8.

The evolution of the normalized parameters with the excitation frequency is represented on Fig. 9. We can observe that their values are almost constant with frequency. This could be expected, as the force amplitude increases linearly with the frequency (see Fig. 4): the damping coefficient is constant and remains optimal for all the considered frequencies. The small variations of the optimal parameters are also due to the

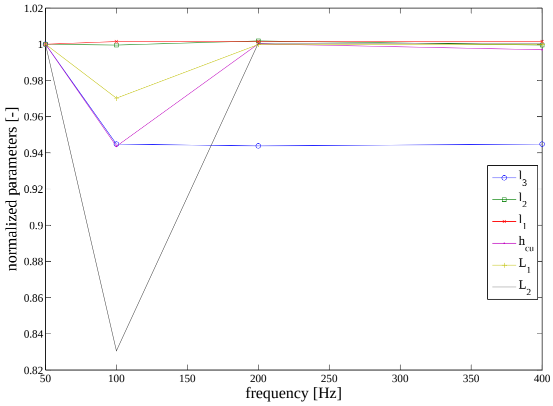


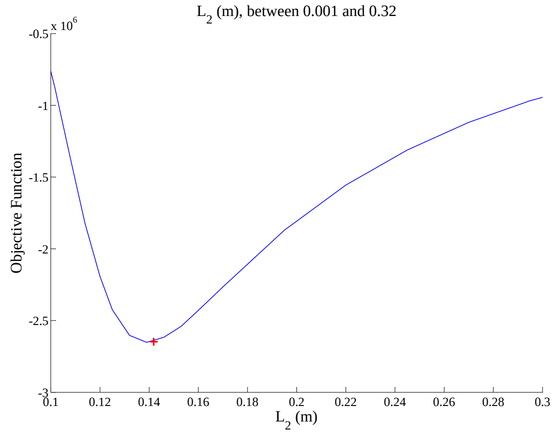
Figure 9. Evolution of the normalized parameters with the excitation frequency.

sensitivity of the objective function which becomes low when approaching the optimum, and they would dissapear if more generations were taken in the genetic algorithm. This can be observed on Fig. 7 and Fig. 10, where we are not exactly on the optimum.

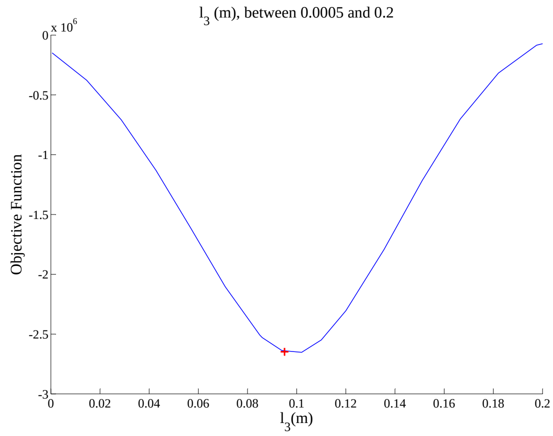
V. Conclusion

Previous works on electrodynamic bearings have shown the importance of adding stabilizing damping to the system. In this paper we have proposed an inherently stable radial electrodynamic damper with a robust design. This damper can easily be used with an electrodynamic bearing or with passive magnetic bearings.

By an equivalent plane 2-D representation and harmonic perturbation FE simulations, an optimization of the geometric dimensions of the damper has been conducted, the objective function being the volumetric damping coefficient.



(a)



(b)

Figure 10. Evolution of the objective function as a function of the optimization parameters (a) L_2 the damper external radius (b) l_3 the magnetic circuit central leg diameter .

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