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A systematic approach for modeling and identification of eddy current dampers in rotordynamic applications

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Abstract

Eddy current dampers (ECDs) exploit Lorentz forces due to the induced eddy currents in a conductor subject to a time–varying magnetic field. ECDs can be used to introduce damping in rotordynamic applications without mechanical contact to the rotor, thus introducing negligible impact on the dynamic response of the whole system. They are suitable for applications where contactless support of a rotor is required, thus being a perfect match for passive magnetic bearings such as permanent magnet bearings and superconducting bearings. However, modeling and identification of the amount of damping induced by ECDs is a difficult task due to complicated geometry and working conditions. A novel and systematic approach for modeling and identification of the damping characteristics of ECDs in rotordynamic applications is proposed in the present paper. The proposed approach employs an analytical dynamic model of the ECD and curve fitting with results of finite element (FE) models to obtain the parameters characterizing the ECD's mechanical impedance. The damping coefficient can be obtained with great accuracy from a single FE simulation in quasi-static conditions. Finally, the accuracy of the identification approach is verified by comparing the results with experimental tests. The validity of this approach is in the cases where ECDs employ an axisymmetric conductor, thus covering most cases in rotordynamic applications.

Keywords : Eddy current dampers, Damping characterization, Modeling of eddy currents forces, Passive magnetic bearing, Finite element modeling, Damping in rotordynamic applications

1. Introduction

Eddy current dampers (ECDs) exploit forces due to eddy currents induced in a conductor by a time-varying magnetic field. The produced forces can be employed into different applications according to the frequency of motion due to the resistive and inductive effects in the conductor (Tonoli, 2007). At relative low frequencies, the resistive effect is dominant in the conductor, causing a dissipation of energy into heat. This effect is exploited to devise ECDs. Different from other passive damping mechanisms, ECDs are capable to provide damping without contact so that the damper introduces little or no impact on the dynamic response of the whole system. This damping has limited dependence on temperature and does not degrade in time (Sodano, et al., 2004). These advantages make ECDs ideal for applications where contactless support of a rotor is required, for instance, rotors supported by passive magnetic bearings.

In the context of magnetic bearings, it is well known that permanent magnet bearings (PMBs) and superconducting magnetic bearings (SMBs) are able to provide positive stiffness but have extremely low lateral damping (Cheah and Sodano, 2008). Therefore PMBs and SMBs require to be associated to damping devices such as ECDs that are able to provide non-rotating damping to allow safe and stable operation when crossing critical speeds and in super-critical regime (Le, et al., 2015), and to increase rotordynamic stability.

Figure 1 shows one common configuration of ECDs used in combination with passive magnetic bearings. The ECD consists of a cylindrical conductor placed in the air gap between permanent magnets of the PMB. As the magnets on the rotor move laterally, currents are induced in the static piece of conductor producing a damping force. In rotating systems, it is important that the ECD conductor is placed on the stator to avoid instability in super-critical regime due to rotating damping (Kligerman, et al., 1998).

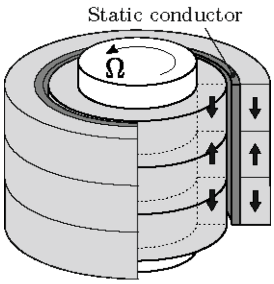

Fig. 1 ECD with passive magnetic bearing

Modeling of ECDs is a difficult task since the damping forces derive from a dynamic interaction between currents induced in a stationary conductor and a moving magnetic field. The calculation of the damping of ECDs has been conducted mainly using analytical equations or finite element (FE) simulations. There are different analytical equations of ECDs damping calculation (Kligerman and Gottlieb, 1998) (Cheah and Sodano, 2008). However, due to simplifications, the predictions of damping coefficients are affected by relatively big errors such as about 40% in (Cheah and Sodano, 2008), which is not admissible for high speed rotors. Another approach is to use 3D time-stepping finite element models to simulate in time the motion of the magnets relative to the conductor. This method is very accurate but extremely expensive in terms of computational time. For example, the analysis carried out by Yamaguchi, et al., (2006) required a total simulation time of 295 hours. Therefore, a better method with higher accuracy and efficiency for the design and systematic analysis of ECDs for rotating systems is required.

In the present paper we present a method for the calculation of the mechanical properties of ECDs for rotordynamic applications. The proposed method uses an analytical model of the Lorentz forces arising in the conductor in combination with a 3D FE model to obtain the radial damping properties of the ECD. The solution of the FE model is obtained under quasi–static conditions, thus being simple to implement, and fast to solve, in a variety of commercial FE software. Finally, the proposed method is validated experimentally showing very good accuracy for the calculation of electromechanical parameters and modeling of dynamic behavior of an ECD.

2. Modeling of ECD and parameters identification

This section describes the lumped parameters approach used to model the ECD and its interaction with the rotor. The model parameters are determined with a procedure based on curve fitting between an analytical model of the Lorentz forces and a Finite-Element (FE) analysis in quasi-static conditions.

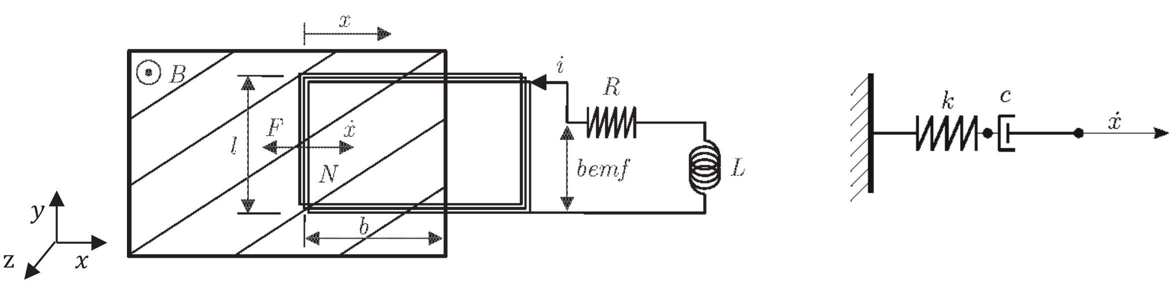

Fig. 2 Electromechanical equivalence between (on the left) a short-circuited rigid coil in motion in a constant magnetic field and (on the right) a mechanical system made of a linear spring and a viscous damper connected in series.

2.1 Analytical model

The analytical model of an ECD is based on the electromechanical equivalence between the behavior of a conductor in motion in a constant magnetic field and the behavior of a mechanical system made of a linear spring and a viscous damper in series. In its simplest form, a motional eddy current damper is constituted by a linear voice coil whose electric terminals are shunted by a resistive and inductive load (Fig. 2 a), where R and L are respectively the resistance and the self-inductance of the coil. The forces generated by this device are the same as for a purely mechanical system made of a linear spring and a viscous damper in series (Fig. 2 b). The stiffness and damping coefficients k and c of this mechanical analogue can be defined as (Tonoli, 2007):

$$\begin{aligned} k &= (NBl)^2 / L \\ c &= (NBl)^2 / R \\ \omega_{RL} &= \frac{R}{L} = \frac{k}{c} \end{aligned} \tag{1}$$

where ω_{RL} (rad/s) is the electrical pole of the system, B (T) is the constant magnetic field on the positive z-axis in Fig.2, while N and l (m) are the number of turns of the voice coil and the active length of the rigid coil, respectively.

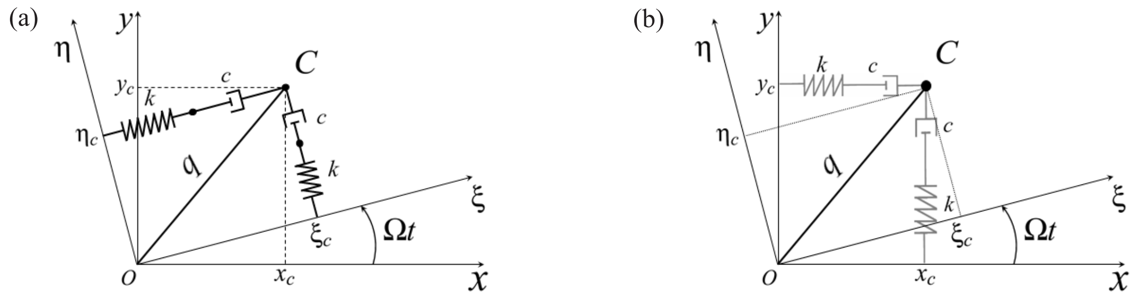

Fig. 3 Equivalent mechanical representation of the eddy current force model in a conductor in relative motion about an axisymmetric magnetic field. (a) The general model for a spinning conductor, usually used for electrodynamic bearings, (b) model for ECDs, where the magnetic field rotates whereas the conductor is static.

This model can be adapted to the case when the conductor and magnetic field are in relative rotation. In this case the interaction can be studied with two reference frames, one fixed to the magnets, the other to the conductor. The schematic representation of this physical system is shown in Fig. 3a. Points O and C give, respectively, the position of the axes of magnetic field and conductor. The reference frame (O, x, y) is fixed whereas (O, ξ, η) is a rotating reference frame that rotates with the same spin speed Ω as the conductor. The pairs (x_c, y_c) and (ξ_c, η_c) are the coordinates of point C in the fixed and rotating reference frames, respectively. In Fig. 3b, the magnetic field rotates whereas the conductor is fixed to the stator, which shows the specific model of an ECD.

Applying the method introduced by Amati, et al. (2008), the dynamic behavior of the eddy current forces generated by the conductor's spin and relative motion can be expressed as:

$$\dot{F} = k(\dot{q} - j\Omega q) - \mathrm{F}(\omega_{RL} - \mathrm{j}\Omega) \tag{2}$$

where Ω is the spin speed of the conductor and q represents its complex displacement. Equation (2) is linear and has constant parameters. It allows modeling dynamically the resistive and inductive effects on the eddy currents. Therefore the forces can be calculated provided that the equivalent electromechanical parameters k and c are known. For dynamic modeling of the forces produced by the ECD, Eq. (3) must be rewritten for the non-spinning conductor (Ω=0):

$$\dot{F} = k\dot{q} - F\omega_{RL} \tag{3}$$

For quasi-static condition, which means constant relative displacement (q=q_0=constant) and constant spin speed Ω, the force produced by eddy currents becomes:

$$F = \frac{k}{1 + (\frac{\omega_{RL}}{\Omega})^2} q_0 - j\frac{c\Omega}{1 + (\frac{\Omega}{\omega_{RL}})^2} q_0 \tag{4}$$

This force is proportional to the displacement and depends on the rotor spin speed. It can be observed that for values of Ω much lower than the frequency of the electric pole the real component of the force vanishes, and the imaginary component assumes the value:

$$F = -jc\Omega q_0 \tag{5}$$

Figure 4 shows how the forces evolve with increasing spin speeds and how the characteristics of their real and imaginary components can be exploited for the purpose of identification. The frequency of the electric pole ω_{RL} is clearly shown and can be used to identify the bandwidth of the damper. It shall be noted that the real component has an asymptote equal to kq_0 that represents a perfectly elastic force. This behavior can only be exploited at values of spin speed much higher than the electric pole frequency, and is of particular interest in the field of electrodynamic bearings. At low speeds, the slope of the imaginary component near zero is given by Eq. (5). This equation can be used for the identification of the damping coefficient of the ECD, which behaves as a simple viscous damper for speeds much lower than the electric pole frequency.

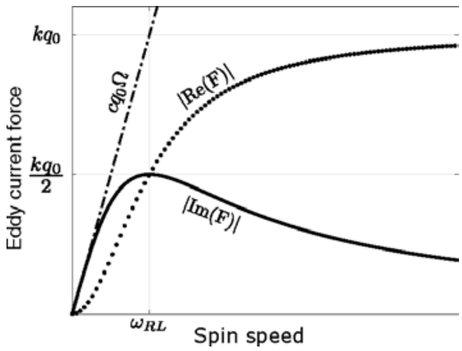

Fig. 4 Eddy current forces with increasing the spin speed. The solid line is the modulus of the force along x (real) and the dotted line is the modulus of the force along y (imaginary). The dash-dot line grows with the same slope as the imaginary component for $\Omega \ll \omega_{RL}$.

2.2 Quasi-static FE simulation

For estimation of ECDs, FE method provides the capacity of calculating the distribution of flux density B for all domains and the current density distribution J inside the conductor. Thus the produced Lorentz forces can be obtained with volume integration over $J \times B$ within the conductor domain. For the present analysis, COMSOL Multiphysics is used.

The Quasi-static simulation was performed for a given Ω and by introducing a small eccentricity ε between the ECD magnets domain Γ_m and conductor domain Γ_c on the x direction such that $q_0 = \varepsilon$. q_0 is a real number, thus the Lorentz forces calculated by the FE can be directly compared to the forces of Eq. (4). The real component gives the force along x whereas the imaginary component gives the force along y.

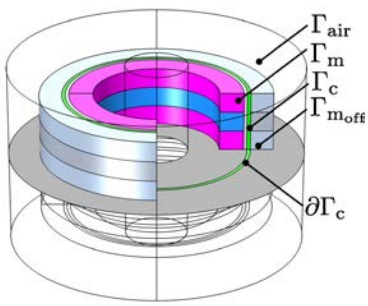

Fig. 5 Three dimension model of a permanent magnet bearing equipped with an ECD. The arrows indicate the different domains considered for simulation: Γ_{air} is the air surrounding the system, Γ_m are the rotor permanent magnets, Γ_c is the copper sleeve that acts as ECD, $\Gamma_{m_{off}}$ are the stator permanent magnets that are turned off for simulation, $\partial\Gamma_c$ boundary of the sleeve domain.

2.3 Identification of parameters from FE results

For the identification of the ECD electromechanical parameters, it is necessary to calculate the Lorentz forces for *n* different values of conductor spin speed *Ω*. The identification of parameters *k* and *c* is performed using a curve fitting procedure. The errors between values calculated with Eq. (4) and the FE model are minimized using the least squares approach.

If the frequencies of the lateral vibrations of the rotor are expected to be much lower than the electric pole frequency, it is reasonable to model the ECD as a simple viscous damper. Thus the damping coefficient *c* can be obtained from the results of the FE model using Eq. (5) as:

$$c_s = -\frac{F_{Ltzy}}{\varepsilon\Omega} \tag{6}$$

Using this equation, it is possible to calculate the damping coefficient from a single FE solution obtained for a low value of *Ω*. It is a very convenient method to obtain the damping coefficient.

3. Experimental validation

Experimental tests are exploited to validate the proposed model and to prove its reliability for design purposes. Hammer impact tests are performed on a dedicated test rig. The frequency response function (FRF) between the input forces on the moving element and the output acceleration is measured. The FRF is also calculated with bode function in MATLAB using the state space model of the ECD test rig. In the state space model, the estimated mechanical parameters of the ECD are used. Experimental and analytical results of the FRF are compared to validate the proposed model.

3.1 Description of the test rig

A test rig is designed to study the dynamic behavior of an ECD used to introduce nonrotating damping in a rotor supported by PMBs. Figure 6 shows the test rig components and table 1 lists the parameters characterizing the configuration. It is composed of a very stiff central column (1) made in nonmagnetic stainless steel. The inner set of magnets (2) of the PMB is fixed to the column. The ECD copper sleeve (3) is press fit onto the magnets in (2), thus on the static part of the test rig. The outer set of magnets (4) is fixed to a moving element (5), which is connected to the central column with four compliant beams that guarantee a correct relative positioning between the two sets of magnets. The magnets' magnetization polarities are shown in Fig. 6b. The test rig is used to measure the FRF between an input force applied to the moving element using an instrumented hammer and the output acceleration measured with a piezoelectric accelerometer. The FRF was obtained using a LMS Scadas digital signal analyzer.

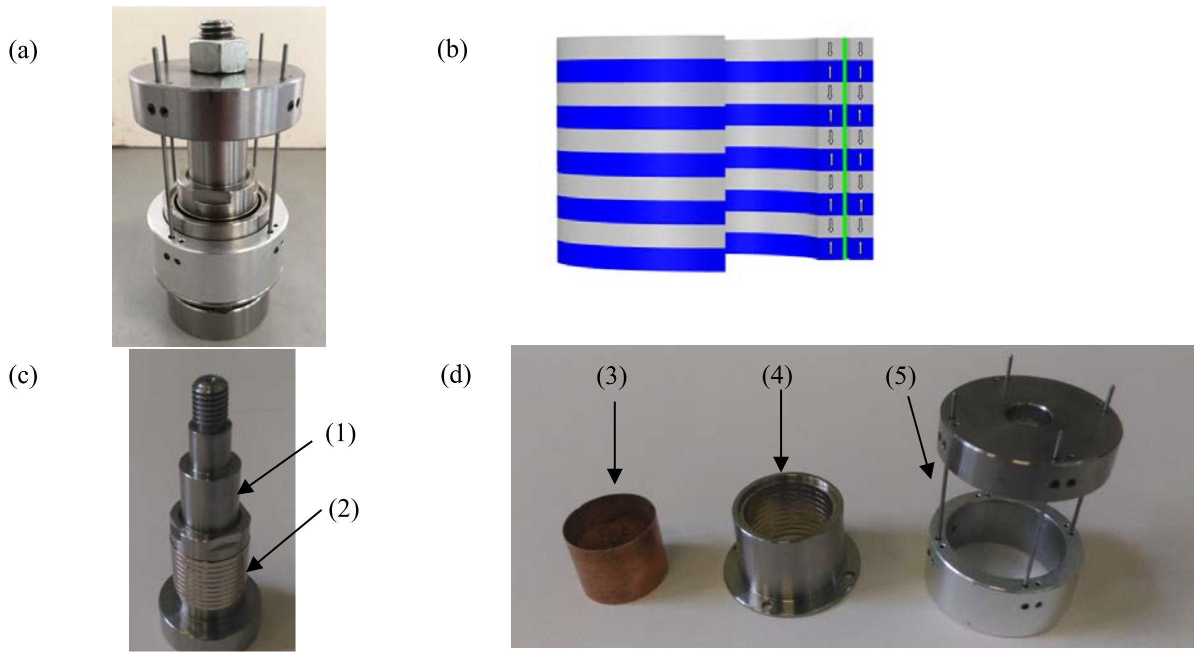

Fig. 6 Test rig used for validation of modeling and identification methods. (a) Complete view, (b) cross –section view of

Table 1 Test rig parameters.

Parameter	Symbol	Value	Unit
Inner magnets inner diameter	di	27.3	mm
Inner magnets outer diameter	do	33.85	mm
Outer magnets inner diameter	Di	35.85	mm
Outer magnets outer diameter	Do	42	mm
Radial air gap	g	1	mm
Eccentricity in quasi-static simulation	ε	0.1	mm
Thickness of magnet ring	t_m	2.95	mm
Number of stacked magnets	n	10	-
ECD inner diameter	d_{iECD}	33.85	mm
ECD outer diameter	d_{oECD}	34.85	mm
ECD height	h_{ECD}	23.6	mm
Conductivity of copper	σ	5.8e7	S m^{-1}
Mass of the accelerometer	m_{acc}	10	g
Mass of the moving structure	m_{mov}	237.5	g
Mass of outer magnets	m_{mov1}	155	g

3.2 Modeling of the test rig

The lateral dynamics of the moving mass in the test rig can be modeled as a single degree of freedom mass-spring-ECD system (Fig. 7). The equation of motion of the mass coupled to the dynamic equation of the eddy current forces (Eq. 3) can be written in the state space form as:

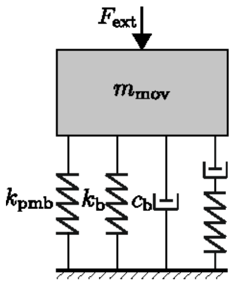

Fig. 7 Single degree model of the lateral dynamics of the moving mass.

$$\begin{Bmatrix} \ddot{x} \\ \dot{x} \\ \dot{F} \end{Bmatrix} = \begin{bmatrix} -\dfrac{c_b}{m_{mov}} & -\dfrac{(\mathrm{k}_{pmb}+\mathrm{k}_b)}{m_{mov}} & -\dfrac{1}{m_{mov}} \\ 1 & 0 & 0 \\ k & 0 & -\omega_{RL} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ x \\ F \end{Bmatrix} + \begin{bmatrix} \dfrac{1}{m_{mov}} \\ 0 \\ 0 \end{bmatrix} \{F_{ext}\} \tag{7}$$

$$\{\ddot{x}\} = \begin{bmatrix} -\dfrac{c_b}{m_{mov}} & -\dfrac{(\mathrm{k}_{pmb}+\mathrm{k}_b)}{m_{mov}} & -\dfrac{1}{m_{mov}} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ x \\ F \end{Bmatrix} + \begin{bmatrix} \dfrac{1}{m_{mov}} \end{bmatrix} \{F_{ext}\}$$

where $\boldsymbol{k_{pmb}}$ is the radial stiffness of the PMB, $\boldsymbol{k_b}$ is the stiffness of the four steel beams, $\boldsymbol{c_{st}}$ is the equivalent structural damping of the beams, and $\boldsymbol{m_{mov}}$ is the mass of the moving part. This model was implemented in MATLAB, and the FRF was obtained using the Bode function.

3.3 Parameters Identification of the ECD

In order to obtain the best possible representation of the mechanical impedance of the ECD, the quasi-static FE simulations have been run for spin speeds of 0-80 krev/min with steps of 10 krev/min. The complete FE model contains 247 851 domain elements and 1 646 357 degrees of freedom. Each simulation costs relatively small amount of time. For example, it lasts 2 minutes 52 seconds for the simulation with the spin speed 60 krev/min using a notebook PC equipped with an Intel i7-2.7 GHz processor.

The procedure of parameters identification can be divided into two steps: first, the damping coefficient of the simple damper model c_s was identified with Eq. (6) and using the FE results for Ω = 1 krev/min; second, the parameters of the complete model k and c were obtained with curve fitting using Eq. (4) and the whole set of FE results.

The results of identification are plotted in Fig. 8, where the markers are the results from FE simulations and the solid lines are the results of Eq. (4) with the identified coefficients. The values of identified coefficients are given in Table 2. It can be noticed that the identified values of c and c_s are very close to each other.

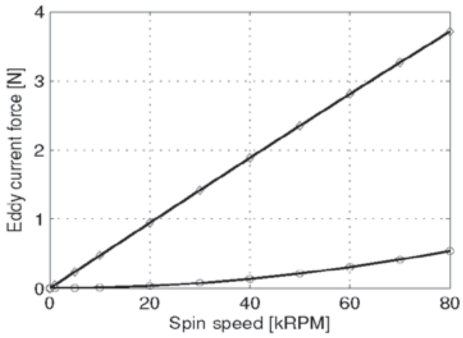

Fig. 8 Eddy current force identification, comparison between identified model (solid lines) and finite element results (circular and diamond markers).

Table 2 Identified parameters of the test rig.

Parameter	Symbol	Value	Unit
Stiffness of beams	k_b	16200	N m^{-1}
Equivalent structural damping	c_{st}	0.227	N s m^{-1}
Stiffness of PMB	k_{pmb}	181500	N m^{-1}
Remanent magnetic flux density	B_r	1.18	T
ECD stiffness complete model	k	130370	N m^{-1}
ECD damping complete model	c	2.2646	N s m^{-1}
ECD simple damper model	c_s	2.2632	N s m^{-1}

3.4 Validation

The aim of the experimental tests is to validate the modeling and identification methods for ECDs. Using the equipment described in Section 3.1, the FRF for each set was obtained by averaging the results of five impacts. The acquisition was made in the range of 0-512 Hz with 0.125 Hz of resolution.

The first set of impact tests is to isolate the stiffness and damping contribution of the steel beams. For this purpose, the outer magnets in the test rig were removed from the moving element. The coefficients k_b and c_{st} were identified by fitting the model of Eq. (7) to the measured FRF. The measured points and fitted model are shown in Fig.9a.

Once the coefficients of the steel beams were identified, a set of impact tests with the complete system were performed. The measured FRF is used to compare to the model. In the model, k_{pmb} was obtained with FE simulation and could also be obtained experimentally.

This comparison is shown in Fig.9b. It can be seen that the model follows almost perfectly the experimental points. It can also be observed that the model slightly underestimates the damping. This can be attributed to the value of conductivity of the copper sleeve. The value used for simulations was identified as the minimum nominal value guaranteed by the producer. A small increase in the value would increase the damping thus improving the correspondence with the tests.

The good agreement between tests and model is considered as a proof of validity of the proposed modeling and identification methods. For the present case, where the electric pole frequency of the ECD is much higher than the frequency of vibration, the use of the complete damper model is not necessary, and the results yielded by simple damper model are equally accurate. The simple damper model reduces significantly the simulation time required for identification. However, it should be used only when the frequencies of vibration are known to be much lower than the electric pole frequency. The bandwidth of the damper can also be identified with this method.

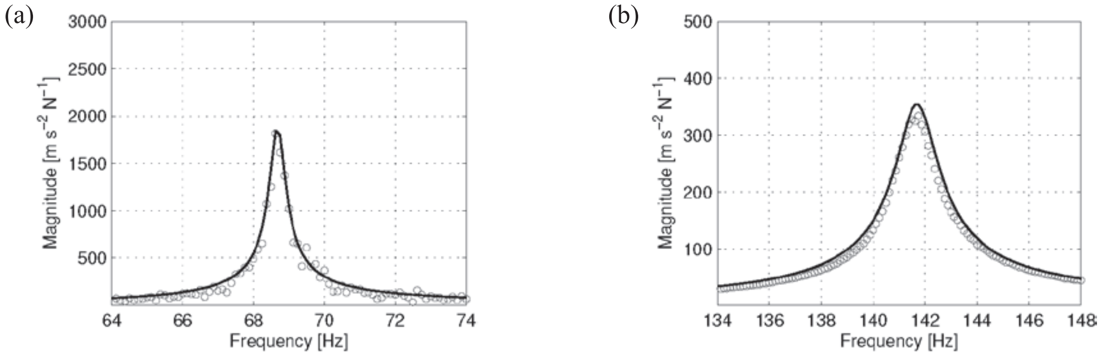

Fig. 9 Comparison between experimental and analytical FRF's for (a) static magnets removed, (b) complete model.

4. Conclusions

A novel and systematic approach for modeling and identification of eddy current dampers has been presented. The scope of the research was dedicated to ECDs for rotordynamic applications, especially for rotors supported by passive magnetic bearings. The approach is based on an analytical model of ECD. Identification of ECD parameters is realized with curve fitting of the model to results obtained from finite element simulations. The FE simulation for this purpose is developed in quasi-static condition thus is easily and fast to be solved. The validity of the model has been demonstrated by experimental tests on a dedicated test rig. This approach presents several advantages compared to available methods: it is efficient in terms of computational time, easily repeatable for different configurations and accurate. The effectiveness of this approach is valid for ECDs that employ axisymmetric conductors, thus covering most cases in rotordynamic applications. The potential applications are in the fields of permanent magnet bearings, superconducting magnetic bearings and passive suspension of bearingless rotating machines.

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