Eigen frequency and damping in a passive magnetic bearing system

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Abstract
A complete passive magnetic bearing system, consisting in a Permanent Magnet radial bearing and a Superconductive Magnetic Bearing has been simulated and constructed. The forces in both types of magnetic bearings are not linear. The non-linear behavior has been implemented in a FEM model and compared with linear models. A spin down test has been conducted to the built system and the Eigen frequencies have been recorded. The Eigen frequencies calculated with and without the non-linear behavior implemented in the model are very similar.

Keywords : Flywheel, Permanent Magnet Bearing, Superconductive Magnetic Bearing, Eigen frequencies, Passive Bearings, Numerical Analysis, FEM

1. Introduction

The field of study of rotor dynamic is of great importance in electrical machines designs. The stiffness in the rotor bearings define the behavior of the rotor. Magnetic bearings are used to support devices with moving parts contactless, increasing efficiency and reducing wear in a variety of applications such as wind turbines and flywheels, for example.

Magnetic Bearings can be classified into two main types: Active Magnetic Bearing (AMB) and Passive Magnetic Bearing. AMB requires position sensing, sophisticated control system and power electronics to drive the current in the electromagnets (Maslen and Schweitzer, 2009). AMB offers the possibility to identify parameters in real time and to adjust its operation to the working conditions. Its implementation is costly because of the number of components. On the other hand, Passive Magnetic Bearings (Lijesh and Hirani 2015, Enemark and Santos, 2015, Jungmayr et al. 2014, Bachovchin el al. 2013) operate without the necessity of any control system. They can be based on superconducting levitation or in permanent magnets (Samanta and Hirani, 2008, Sotelo et al. 2005). Permanent Magnet Bearings (PMB) are intrinsically unstable and cannot be employed in all degrees of freedom.

In the present work, a complete passive magnetic bearing system is tested experimentally and some dynamic properties are presented (Rodriguez et al. 2014). The magnetic bearing system consists of a vertical shaft with a Superconductive Magnetic Bearing (SMB) in its base and a radial PMB on the top. The upper bearing is a PMB, and it consists of two rings of axially oriented permanent magnets. The lower bearing is a SMB, and it is composed by melt textured $YBa_2Cu_3O_{7-\&}$ superconductor (YBCO) in the stator and a rotor made with permanent magnets and steel arranged in a flux collector configuration.

The Eigen frequencies of a rotating shaft depend on the stiffness and damping of the bearings. The shaft is accelerated with an external machine and the vibrations are measured for the speed interval from 0 rpm to the first Eigen frequency. The dynamic study of the rotor is of great importance as the Eigen frequencies determine the speed range at which machines can operate.

2. System Description

The system under study consists of a shaft with a passive magnetic bearing system. The upper bearing is a PMB, and it consists of two rings of axially oriented permanent magnets (Rodriguez and Stephan, 2012). The lower bearing is SMB, and it is composed by melt textured $YBa_2Cu_3O_{7-\&}$ superconductor (YBCO) in the stator and a rotor made with permanent magnets and steel arranged in a flux collector configuration (Rodriguez et al. 2014). A picture and a schematic of the complete system can be seen in Fig. 1.

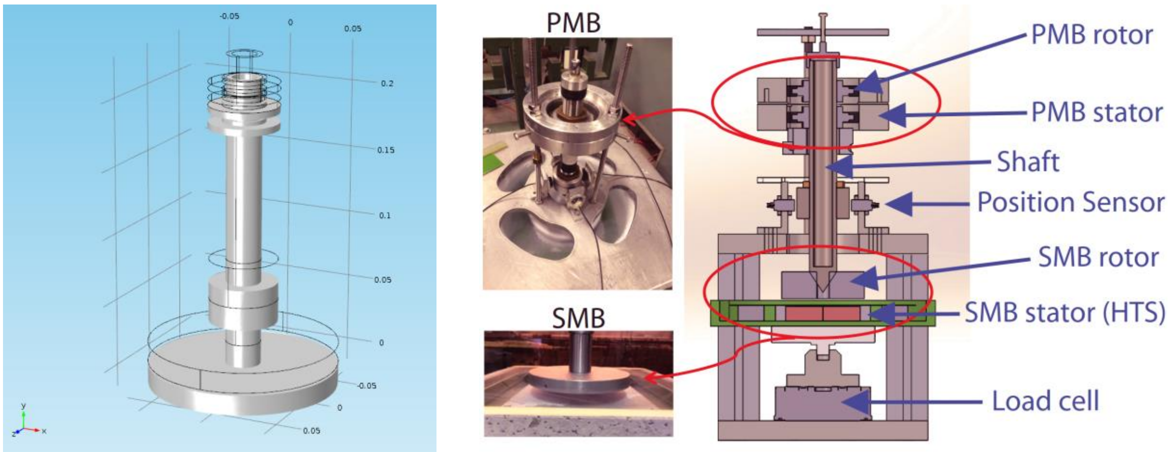


Fig. 1 CAD representation and picture of the experimental set up of the passive magnetic bearing system.

2.1. Rotor dynamic model

The rotor dynamic equation of the motion for a rigid body is:

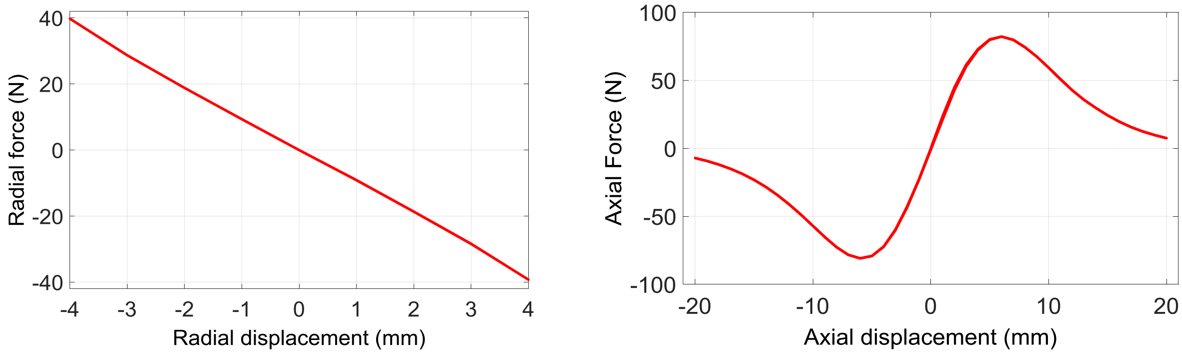
$$M\ddot{q} + D\dot{q} + Kq = f(t) \qquad (1)$$

With M being the rotor mass, D the damping matrix, K the stiffness matrix of the rotor bearings, q the vector of displacements and *f(t)* the vector of forces. An analytic solution of the equation is only found under some simplifications, such as: the rotor is a perfect rigid body, it has symmetry, axial and radial force components are decoupled and the bearing’s force is linear with displacement. The system under study does not meet these conditions as the bearings at each side of the rotor are different and the forces in the PM bearing are not decoupled. FEM simulations are the standard method to calculate Eigen frequencies, and they will be used in this article (Bleuler el al. 2009).

The dynamic model of a rotor with passive PM bearings requires a somehow advance FEM model. Standard commercial software only allows decoupled and linear forces in the bearings. Several FEM simulations will be presented to evaluate the Eigen frequency under different assumptions.

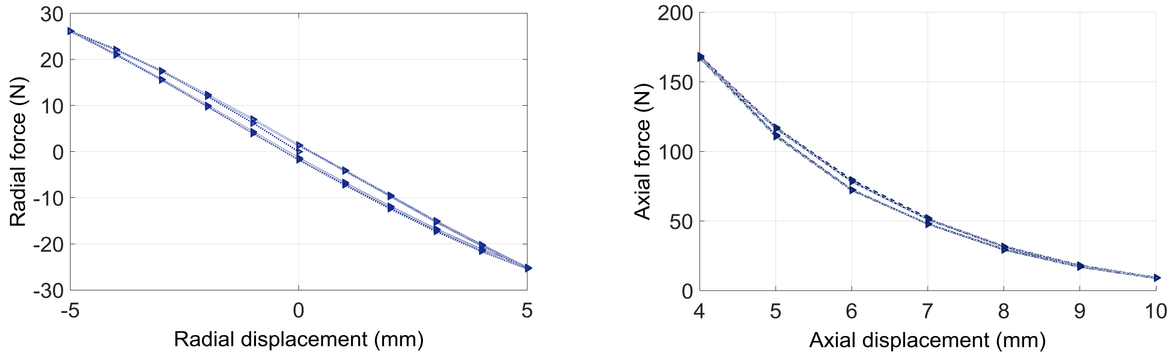
2.2. Static forces

The PMB and SMB have been tested individually. The axial and radial forces for both magnetic bearings have been described in static measurements presented in (Rodriguez et al. 2014). Figures 2 and 3 show these forces.



(a) (b)

Figure 2 (a) Radial and (b) axial forces in the PMB for the radial and axial displacement, respectively.



(a) (b)

Fig. 3 (a) Radial and (b) axial forces in the SMB for the radial and axial displacement, respectively.

COMSOL is the FEM program used for the simulation of the magnetic bearing set up. The bearings are usually simulated following Hook's law, which is linear. The forces measured in the magnetic bearings provides the information for the FEM program, as presented in Fig. 2 and 3. The PMB placed on the top presents a behavior close to linear in the radial direction and an instable and non-linear behavior in the axial behavior. Therefore the radial and axial forces may be approximated by a linear and exponential function respectively. The SMB placed at the bottom of the shaft follows Hook's law in the axial direction quite well and presents a stable non-linear behavior in the radial direction. The equations used in the simulations are presented in Table I.

Table I. Forces modeling the magnetic bearing behavior

	PMB	**SMB**
Radial	-15 [N/mm]	- 40 [N/mm]
Axial	44 * h * exp(-0.0139*h^2) [N/mm]	60 * exp(-0.3*h) [N/mm]
Axial linearized at h=0 [mm]	0 + 44*h [N/mm]	60 – 18*h [N/mm]

The axial force in the PMB is intrinsically unstable. A small deviation in the *h* direction will create a positive force that will increase this deviation. It is modeled so that the maximum is 160 N at a 6 mm displacement. The axial force in the SMB is modeled so that the force at origin is 60 N corresponding to a distance between the bearing and the superconductive material of about 8 mm, and 150 N with a 3 mm displacement. The system stability is obtained in combination with the PMB and the SMB. The equilibrium point in the axial direction may be set with a small offset from the center of the PMB. PMB and SMB are coupled in this point. A small perturbation in the middle point would produce a fast change in the force direction. Several offset points have been simulated. The geometry studied is presented in Fig. 4. The maximum displacement of the rotor is the distance between the center of the inner ring on the shaft and the static outer ring in the PMB.

The axial forces in the SMB and PMB follow non-linear functions. The graphs intersect in two points, meaning that there are two equilibrium points, one stable and one instable. This creates a numerical problem to the FEM solver. A small force is introduced in the form of a spring with a constant of 10 N/m to help the solver converging to the stable solution. Note that the force is three orders of magnitude lower than the forces from the bearings, and the contribution of this balancing force is therefore negligible.

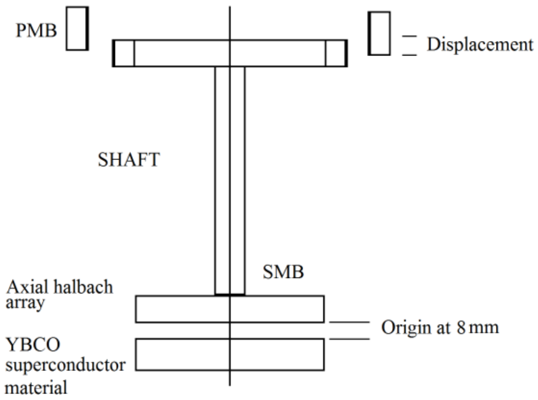


Fig. 4 Schematic representation of the simulated geometry.

3. Results

3.1. Simulation Results

The shaft with the magnetic bearing system has been simulated. The forces that define the magnetic bearings have been presented in Table I. The Eigen frequency has been calculated for three different offsets to study the influence of this parameter in the system stiffness.

In a set of simulations, the shaft has been simulated linearizing the force from the bearings. Linear forces reduce the simulation time, and the results are not affected by this simplification. The Eigen frequencies obtained are presented in Table II.

Table II. Eigen frequencies for different displacements.

Displacement	**Non – linear model**	**Equilibrium point**
5 mm		- 1.93 mm
4 mm	Same for the 3 cases: 0.43, 8.63, 11.8, 27.1, 27.3 and 37.5 Hz 25.8, 518, 708, 1626, 1638 and 2250 rpm	- 1.48 mm
3 mm		- 0.79 mm

The model was simulated again with a linearization in the SMB force and a non-linear model in the PMB. Then, the force in both SMB and PMB was linearized according to the equations presented in Table I. The Eigen frequencies were exactly the same as for the model with non-linear bearings. The equilibrium point changed. The equilibrium point for the linear system becomes directly proportional to the displacements, and the divergence from the non-linear system increases more as we study points away from the linearization point. The displacement values are presented in Table III.

Table III. Eigen frequencies for different displacements.

Displacement	Linear model in SMB – Equilibrium point	Linear model in both SMB and PMB - Equilibrium point
5 mm	- 1.97 mm	- 3.3 mm
4 mm	- 1.52 mm	- 2.2 mm
3 mm	- 0.83 mm	- 1.1 mm

3.2. Experimental Results

A rotor has been built and accelerated with an external flexible shaft in order to verify the model and dynamic behavior of the complete PMB. The rotor was left to spin down freely, slowing down due to friction losses and forces in the superconductive magnetic bearing have been measured. Figure 5 presents the axial forces measured at a spin down test of the rotor. The axial force measurement shows vibrations at 850 rpm.

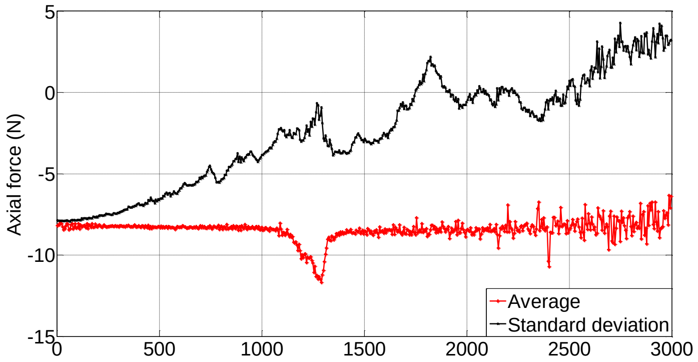


Fig. 5 Axial forces measured at the lower bearing at the spin down test.

The rotor decelerates slowly at the spin down test. The dynamic behavior of the system passing through a resonant frequency is also of interest. A new test was performed in which an axial flux motor was mounted in the shaft and accelerated the system. Figure 6 shows the acceleration rate and the forces in the lower SMB. The motor added mass to the system, so the parameters in the presented simulations are not applicable to this results. It is observed that the acceleration contributes to the rotor stability.

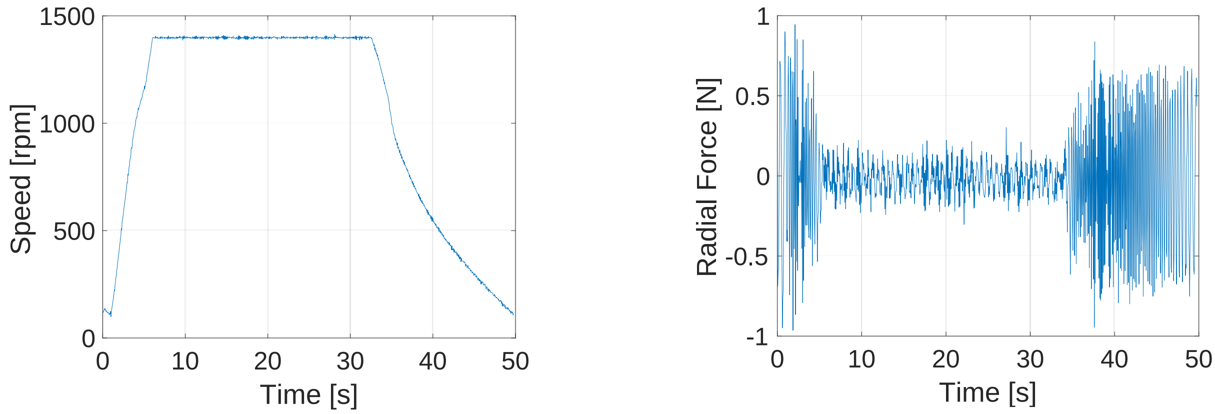


Fig. 6 Acceleration rate with an axial motor mounted on the shaft and the radial forces measured during the test at the lower bearing. No significant forces are observed when the motor passes the Eigen frequency.

4. Discussion

A rotor with a passive magnetic bearing system was simulated, build and tested. The system has a combination of SMB and PMB to achieve complete passive levitation. Magnetic bearings present a non-linear behavior that was modeled in FEM simulations. The Eigen frequencies were obtained both experimentally through a spin down test and by FEM analysis.

The paper studied the linearization of the forces. Three FEM models were simulated; first both bearings where modeled with non-linear behavior; a second model represented the SMB with linear forces and the PMB with a non-linear model; a third model represented both bearings with linear forces. It was found that the three models gave very different equilibrium points but the same Eigen frequencies. Static simulations of Eigen frequencies are calculated at the equilibrium point and to not benefit from more advanced models. The discrepancy of test and simulation results in Eigen frequency values may be explained by the stiffness in the PMB. The system has been simulated with higher values then the test set up.

In future work, dynamic simulations of the non-linear models will be conducted. An axial flux motor has been mounted in the test set up to accelerate the shaft and study the dynamic behavior. Some preliminary results have been presented showing that a fast acceleration through the resonance frequencies contributes to the system stability.

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References

Bachovchin, K. D., Hoburg, J. F. and Post, R.F., Stable levitation of a passive magnetic bearing, IEEE Transactions on Magnetics, 49 (1), art. no. 6242412, pp. 609-617, (2013). DOI: 10.1109/TMAG.2012.2209123.

Bleuler, H., et al. Magnetic bearings: theory, design, and application to rotating machinery. Eds. Gerhard Schweitzer, and Eric H. Maslen. Springer Science & Business Media, 2009.

Enemark S.,and Santos, I. F., Nonlinear dynamic behavior of a rotor-foundation system coupled through passive magnetic bearings with magnetic anisotropy - Theory and experiment, Journal of Sound and Vibration, Article in Press., (2015). DOI: 10.1016/j.jsv.2015.10.007

Jungmayr, G., Marth, E., Amrhein, W., Berroth, H.-J., and Jeske, F., Analytical stiffness calculation for permanent magnetic bearings with soft magnetic materials, IEEE Transactions on Magnetics, Vol, 50 (8), art. no. 6762946, (2014), DOI: 10.1109/TMAG.2014.2310437.

Lijesh K.P, and Hirani, H., Magnetic bearing using rotation magnetized direction configuration, Journal of Tribology, Vol. 137 (4), art. no. 042201, (2015). DOI: 10.1115/1.4030344.

Rodriguez E.F., Stephan R.M., Radial and Axial Forces of a Passive Magnetic Bearing. In Proceedings of the First Brazilian Workshop on Magnectic Bearings, 2013, Oct. 25.

Rodriguez E, Santiago JD, Pérez-Loya JJ, Costa FS, Sotelo GG, Goncalves de Oliveira J, Stephan RM. Analysis of passive magnetic bearings for kinetic energy storage systems. In14th International Symposium on Magnetic Bearings 2014.

Samanta, P., Hirani, H., Magnetic Bearing Configurations: Theoretical and Experimental Studies, IEEE Trans. on Magnetics, Vol. 44, No. 2, Feb. 2008.

Schweitzer, G., and Maslen, E. H., Magnetic Bearings: Theory, Design, and Application to Rotating Machinery. New York, NY, USA: Springer-Verlag, 2009.

Sotelo GG, Ferreira AC, and de Andrade Jr R., Halbach array superconducting magnetic bearing for a flywheel energy storage system. Applied Superconductivity, IEEE Transactions on. 2005 Jun;15(2):2253-6.