MAGNETIC BEARING WITH UNIAXIAL CONTROL USING MAGNETIC, ELECTRODYNAMIC AND ELECTROMAGNETIC LEVITATION

Rogério Issamu Yamamoto
Oswaldo Horikawa
ESCOLA POLITÉCNICA DA UNIVERSIDADE DE SÃO PAULO, SÃO PAULO - SP - Brazil
rogerio.yama@usp.br
ohorikaw@usp.br

Isaias da Silva
UNIVERSIDADE FEDERAL DE SÃO PAULO, DIADEMA - SP - BRASIL
isaias.sv@uol.com.br

Abstract. *Magnetic bearings use magnetic force to sustain loads without any contact between the rotor and the bearing, therewith there is no friction or wear. Due to these advantages, these bearings are applied in wide range of applications, from flywheel energy storage systems to artificial hearts. In the first, the magnetic bearing reduces energy loss by friction, raising the performance, and in the second, the magnetic bearing minimizes the damage to blood cells. Many types of magnetic bearings are known, each one based on different levitation techniques, e.g. based on the use of superconductive materials. However, the superconductivity is obtained only at temperatures around 100 K, imposing limitations for practical applications at room temperature. A more promising technique is the electrodynamic levitation at room temperature. The repulsion force for the levitation is generated by the relative movement between a magnetic field and a conductor. The inconvenient of this technique is that high heat rates are dissipated in the conductor. In order to reduce the heat generation a well-designed electrodynamic suspension must be accomplished. Another levitation technology is based on electromagnetic forces, i.e. the levitation force is obtained by electromagnetic coils. Based on readings of position sensors the current to the coils is regulated keeping the object in a fixed position. The main disadvantage of the electromagnetic levitation is its construction complexity because it requires a sensor, a controller, a power amplifier and an actuator for each degree off freedom of the system. In this context, focusing applications like flywheel energy storage system in which the rotor starts from zero and reaches 50.000 rpm order speeds or even higher, this work presents a magnetic bearing that combines three levitation techniques. The first one is based on permanent magnets. This provides stable levitation forces in the radial direction of the rotor. However this technique does not assure stability of the rotor position along its axial direction. To solve this problem, a second technique, the electrodynamic levitation at room temperature based on permanents magnets is applied which supplies damping for the bearing system, contributes to the rotor stability along its axial direction and reduces the electromagnetic levitation energy consumption and the third is an active electromagnetic bearing which levitate the rotor at low speed and assist the electrodynamic bearing in axial stability. The proposed hybrid bearing is simpler than a bearing whose rotor is sustained only by electromagnetic bearings in 5 degrees of freedom, therefore reducing the energy consumption when using only electromagnetic bearings and can work at room temperature without a refrigeration system. This work presents the principle of the new hybrid bearing, the development steps of this bearing besides showing both computer simulated numerical and experimental results.*

Keywords: *Magnetic Bearing; Flywheel; high speed rotors; electrodynamic levitation*

1. INTRODUCTION

Magnetic bearings are devices which use the magnetism to sustain a rotor in space, so there is no contact between the rotor and the bearing. Such bearings has been applied on system like flywheels or artificial hearts in which it is necessary or even imperative the reduction or elimination of the friction, at the first case the objective is to increase the device efficiency and in the second case the bearing must avoid the damage caused on blood cells (HAMLER, et al., 2004) e (HORIKAWA, et al., 2008).

Magnetic bearings can be classified in two large groups, i.e. passive and active bearings. Passive bearings do not require controllers or actuators to stabilize a rotor. On the other hand active type bearings need controllers, sensors and actuators to keep the rotor stability (SCHWEITZER, 2009).

Among the passive bearings there are the following types: permanent magnet bearings, superconductor bearings and electrodynamic bearings, and among the active bearings the best known is the active electromagnetic bearing type.

Passive bearings are only based on static and constant magnetic field, i.e. there isn't any control system to control the magnetic force, therefore its feasibility to simultaneously stabilize the rotor in all degree of freedom are limited and they are generally used for very specific applications, e.g. when the required bearing stiffness and damping are low. Passive magnetic bearings can use different concepts to generate the levitation force which sustain the rotor weight, among this concepts three types can be highlighted: permanent magnet, superconductor and electrodynamic levitation (SCHWEITZER, 2009).

Permanent magnet bearings only use permanent magnets to generate levitation force, the concept applied in these type of bearings is the simple repulsion or attraction forces, which appear when two or more magnets are placed in the field of each other. These kinds of bearings present low cost and a long life cycle, basically they can be made by two magnetic rings, generally made of NdFeB alloy, due the magnetic characteristics of this compound, and they can be arranged in several ways, like those present in Fig. 1 (YONNET, 1978).

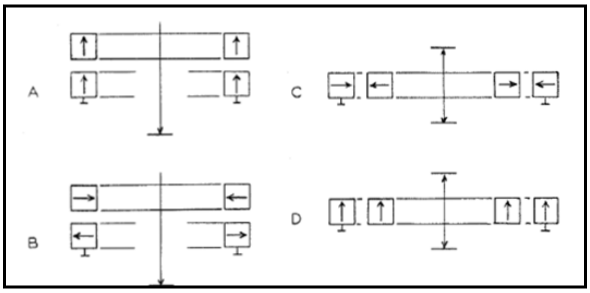


Figure 1: Four basic arrangements of passive magnetic bearings – (a) and (b) attraction type (c) and (d) repulsion type (YONNET, 1978)

However, according to Earnshaw's theorem (EARSHAW, 1848), the stability in all degrees of freedom cannot be achieved using only permanents magnets, nonetheless these bearings can be applied to control some degrees of freedom of a body or used as auxiliary bearings reducing passive loads on the main bearing (MOSER, et al., 2006) and (YONNET, 1978).

Superconductors bearings use special materials called superconductors which present a unique feature called superconductivity, i.e. such materials virtually have no electrical resistance and their relative magnetic permeability becomes null when they are subjected to temperatures around 100 K, called critical temperature, T_C. According to the Meissner-Oschenfeld effect these materials, when submitted to an external magnetic field and when the temperature is below the T_C, tend to eliminate the field inside the material, as shown in Fig. 2, so generating a repulsion force, (SCHWEITZER, 2009) and (CALLISTER, 2002).

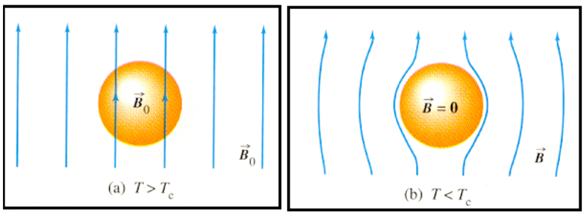


Figure 2: (a) superconductor material above the critical temperature and (b) below the critical temperature (T_c) (RODGERS, et al., 2004)

The main disadvantage of these bearings is the condition to the Meissner-Oschsenfeld effect works, i.e. the material must be below the critical temperature (T_C) so that the material becomes superconductor. Therefore, the application of these types of bearings requires the use of an efficient cooling system which must be linked to the superconductor material, (HAMLER, et al., 2004) and (NICOLSHY, et al., 2000).

Electrodynamic bearings include a class of devices based on the principle of electrodynamic repulsion, which in turn is based on the Lenz Law, i.e. according to this law "an electric current induced in a conductor always appears to oppose the variation of the phenomenon that caused it". In fact this means that any variable magnetic flux will be reflected when focused on a conductor surface generating repulsion and drag forces which are normal and tangential to the conductive surface, respectively (FURLANI, 2001). Figure 3 shows an example of a simple electrodynamic levitation system that uses an aluminum disc and a permanent magnet piece. When the disc rotates with an angular speed that causes a tangential velocity to be higher than a certain critical velocity, the magnetic fields interaction, i.e. the magnet field and the disc induced field, generates two forces, a normal force (F_1) and a tangential force (F_2). The normal force is responsible for the levitation and the tangential force called the drag force causes heat dissipation in the disc.

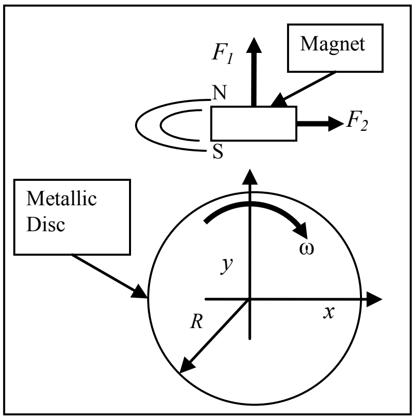


Figure 3: An electromagnetic levitation simple scheme.

The levitation and the drag force are dependent mainly on the frequency of the magnetic flux, i.e. on high frequencies levitation force dominates and in low frequencies the drag force predominates, as can be seen in Fig 4 (STEPHAN, et al., 2009).

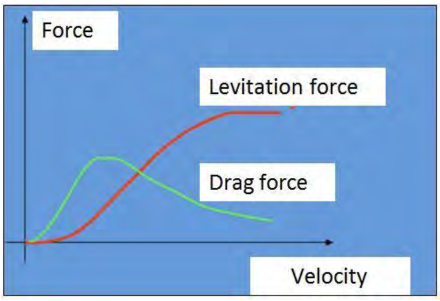


Figure 4: Levitation and drag forces characteristics in an electrodynamic suspension (AMANTI, et al., 2008)

The electrodynamic bearings have relatively simple construction architecture when compared to electromagnetic bearings, however they have both low stiffness and damping compared to the active bearings and their stability is ensured only at high speeds (AMANTI, et al., 2008).

Electromagnetic bearings use the force produced by electromagnetic fields to actively control the dynamics of an electromagnet, so the control system acts on the electromagnet current generating an electromagnetic force that keeps the rotor in a specific position. Figure 5 illustrates a radial electromagnetic bearing schema, in this figure is indicated the key components of an active electromagnetic bearing (SCHWEITZER, 2009). Referring to the figure can be noted that to control one degree of freedom it is necessary a sensor to measure the displacement of the rotor and a microprocessor, which has the control logic. The input signal of the microprocessor is provided by the sensor and the output signal feeds a power amplifier which is used like an interface between the microprocessor and the electromagnet, i.e. converts electric current to electromagnetic force applied to levitate the rotor. The control logic ensures the stability and damping for the rotor. These characteristics can be adjusted by the controller parameters (SCHWEITZER, 2009).

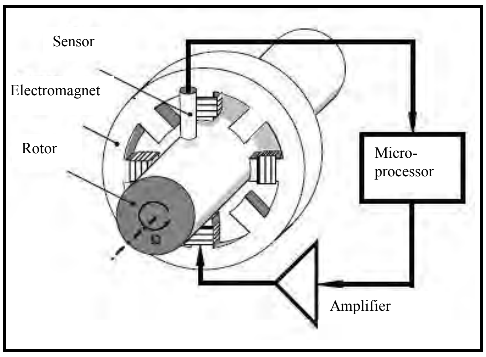


Figure 5: Radial electromagnetic bearing scheme (SCHWEITZER, 2009)

2. PROPOSED HYBRID TYPE BEARING

The proposed bearing is a magnetic bearing of hybrid type; therefore it is a magnetic bearing which uses more than one levitation concept. The rotor levitation along its radial direction is achieved by two passive radial permanent magnet bearings placed at the rotor's end and that provide, in the adopted magnets arrangement, high positive radial stiffness, however the axial stiffness is negative. So, in order to partially overcome this negative axial stiffness and also to introduce some radial damping to the rotor an electrodynamic bearing is used. However, as stated before, electrodynamic bearings only work when the relative speed between the magnetic field and the conductor is higher than a critical speed. So, in order to provide the rotor stabilization at low speeds, allowing the application of the electrodynamic bearing, an axial type electromagnetic bearing is used, comprising the hybrid type proposed bearing. Figure 6 depicts the proposed hybrid magnetic bearing scheme.

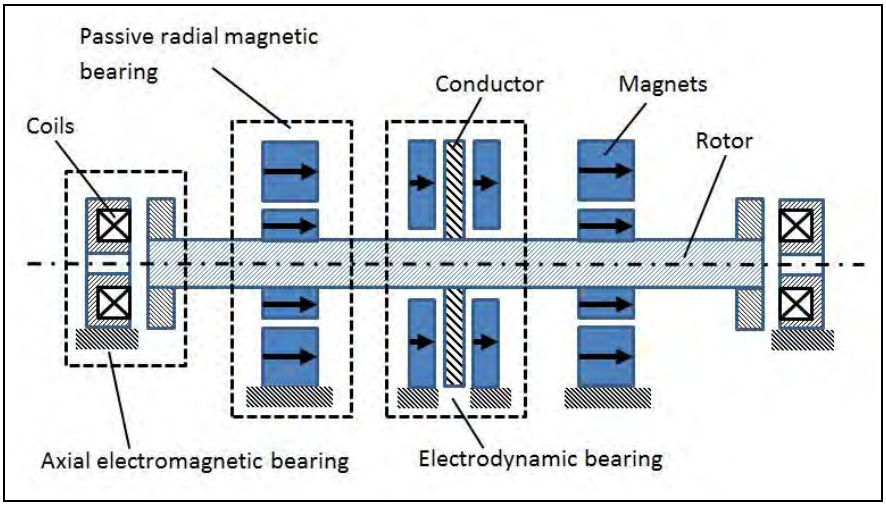


Figure 6. Proposed hybrid magnetic bearing scheme

2.1 RADIAL PASSIVE MAGNET BEARING

The adopted radial passive magnet bearing is depicted in Fig. 7. This configuration was chosen after computer simulation of some magnet ring arrangements using finite element method. This radial bearing has a negative axial stiffness, k_a, and a positive radial stiffness, k_r, and the stiffness are related according the following equation (YONNET, 1978)

$$k_a = -2k_r \tag{1}$$

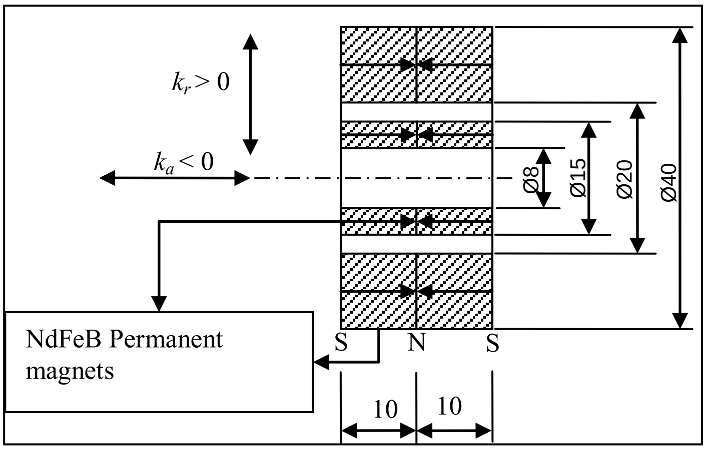
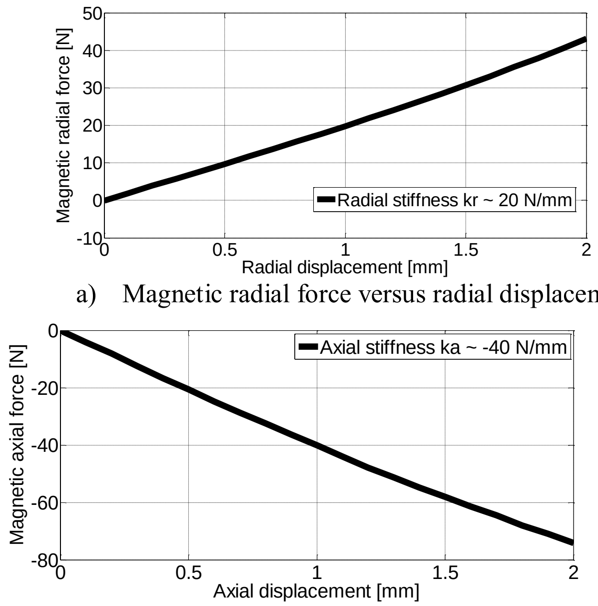


Figure 7: Magnetic ring arrangements for the passive radial bearings used through this work (dimensions in mm)

Using the finite element method and considering the magnets dimensions and material shown in Fig. 7, some numerical results were obtained. Figure 8 shows the magnetic forces as function of the inner magnets displacement along the radial and axial directions. Through these figures one can see that the axial and the radial stiffness are $k_a \cong -40\ {}^{N}/_{mm}$ and $k_r \cong 20\ {}^{N}/_{mm}$, respectively. As predicted in Eq.1, radial stiffness is twice in magnitude the axial stiffness. In the proposed bearing two pair of ring type magnets are used, one at each rotor extremities as shown in Fig. 6. Thus, the rotor total stiffness are $k_a \cong -80\ {}^{N}/_{mm}$ and $k_r \cong 40\ {}^{N}/_{mm}$.



a) Magnetic radial force versus radial displacement

b) Magnetic axial force versus axial displacement

Figure 8 – Radial (a) and axial (b) forces generated by the passive radial magnetic bearing

2.2 ELECTRODYNAMIC BEARING

The electrodynamic bearing is used to improve positive axial and radial stiffness as well as damping as the rotor angular speed increases. The main problem associated with these bearings is the high heat generation as consequence of the eddy currents induced in the bearing conductor part. There are many works about electromagnetic levitation that propose solutions to the heat losses (FILATOV, et al., 2002), (FILATOV, et al., 2002) e (LEMBKE, 2003). Among these works, the so called homopolar bearing architecture has the simplest configuration and presents the lowest heat generation. This is due to the uniform distribution of the magnetic field around the conductor, i.e. no eddy current is induced in the conductor when it is rotating without radial or axial oscillations relatively to the uniform magnetic field center. Therefore, the forces and damping are generated only when the rotor displaces from the field center. So, following the homopolar bearing concept, a radial and axial electrodynamic bearing was designed. Figure 9 shows four coaxial magnet rings working in attraction mode assembled in two steel discs. An aluminum disc fixed to a rotor is placed between the magnets. This disc is the electrodynamic suspension conductor.

The steel discs are used in the electrodynamic bearing to close the magnetic circuit, increasing the magnetic flux that cross through the aluminum disc. So, when the rotor's angular speed is higher than a critical angular speed and, when it is displaced away from its centered position, an electric current is induced in the aluminum disc, generating axial, radial and tangential forces and also some damping. The axial force contributes to compensate the axial negative stiffness effect, the radial force and the damping increase the passive radial stiffness and reduce rotor's radial oscillations and the tangential or drag force causes a torque that tries to reduce rotor's angular speed. Figures 10, 11 and 12 depict the electrodynamic radial and axial forces, the drag torque and the solid losses, respectively, obtained by computer simulation using the configuration shown in Fig. 8 and considering a rotor radial and axial displacement of 0.2 mm and a rotor angular speed of 2500rpm. Figure 10 shows that the electrodynamic stiffness for these operating conditions are approximately $6.5\ {}^{N}/_{mm}$, $9\ {}^{N}/_{mm}$ and $15\ {}^{N}/_{mm}$ for axial direction and for radial directions 1 and 2, respectively. The reason that the radial stiffness values are different from each other is because the rotor radial displacement was applied only along radial direction 1. Figure 11 depicts the drag torque that acts on the bearing conductor disc for the mentioned operating conditions. Observing this figure, one can see that the drag torque is very small, this agrees with the results presented in Fig. 12 which shows that the solid losses in the conductor disc is about 180mW. These losses are considered satisfactory for the present operating conditions and bearing dimensions.

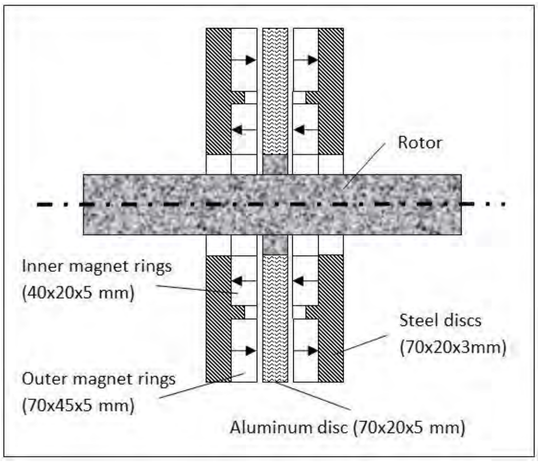


Figure 9: Radial and axial electrodynamic bearing scheme

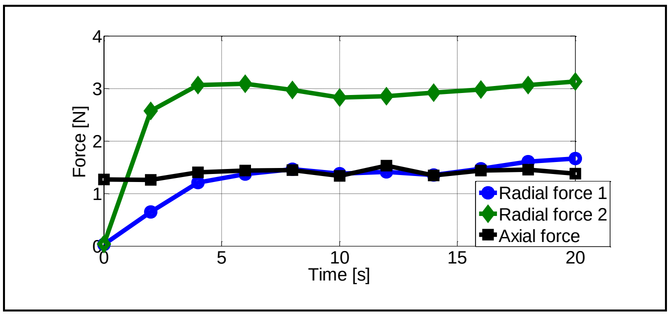


Figure 10: Electrodynamic bearing forces as function of time (rotor's angular speed of 2500 rpm and rotor axial and radial displacement of 0.2 mm)

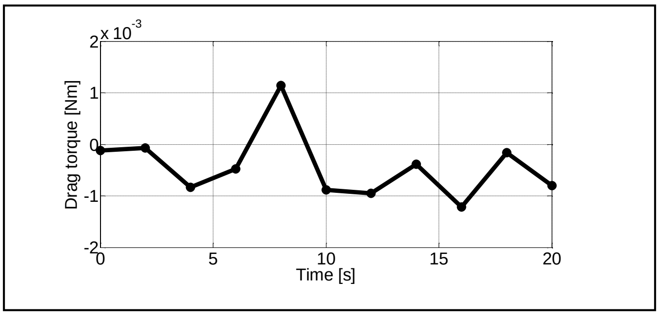


Figure 11: Electrodynamic bearing drag torque as function of time (rotor's angular speed of 2500 rpm and rotor axial and radial displacement of 0.2 mm)

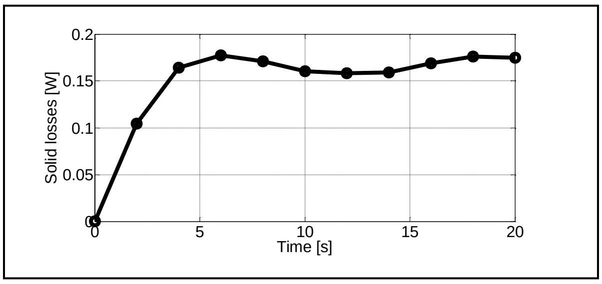


Figure 12: Electrodynamic bearing solid losses as function of time (rotor's angular speed of 2500 rpm and rotor axial and radial displacement of 0.2 mm)

2.3 AXIAL ELECTROMAGNETIC BEARING

The active electromagnetic bearing is responsible to compensate for the negative axial stiffness of the passive magnet bearing, therefore allowing the hybrid bearing operating at low and high rotation speeds. This bearing is formed by a gap sensor, a controller, an amplifier and an electromagnet actuator, as illustrated in Fig. 13. The actuator shown in Fig. 13 were simulated and the results presented in Fig 14. The bearing is at its nominal operating point when the gap between the actuator face and the steel disc fixed to the rotor's ends is 0.3 mm. This value was chosen based on the value of the passive axial stiffness for the whole bearing $k_a = -80\ N/m$. To overcome this negative stiffness, the actuator force/current and also the force/displacement factors must be positive and higher in modulus than this value. Adopting the values depicted in Fig. 14, the designed actuator is capable of stabilize the rotor axially.

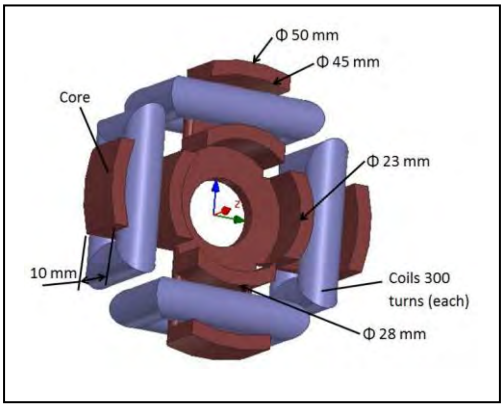
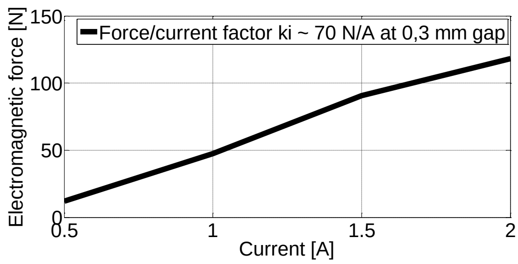
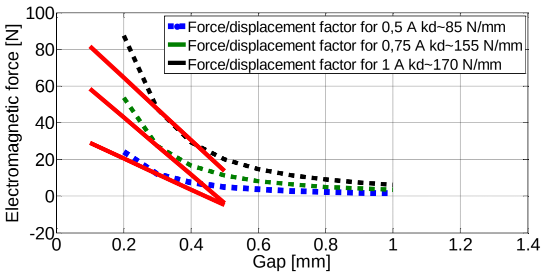


Figure 13: Axial electromagnetic actuator



a) Electromagnetic force versus current determined for 0.3 mm gap



b) Electromagnetic forces versus gap

Figure 14: Electromagnetic forces generated by the axial electromagnetic actuator

2.4 BEARING CONTROL SYSTEM

As stated before, the rotor is unstable in the axial direction, i.e. the magnetic axial stiffness k_a is negative. In order to stabilize the rotor along the axial direction, a closed loop control system must be implemented. Figure 15 shows the closed loop system block diagram. Considering the bearing dynamic model shown in Fig. 16, the transfer function that describes the bearing and the rotor axial dynamics are derived. Using Newton second law, assuming that the rotor axial displacement is enough small, and assuming null initial conditions, the axially controlled magnetic bearing transfer function is given by the following equation:

$$G(s) = \frac{\Delta X(s)}{\Delta I(s)} = \frac{k_i/m}{s^2 - \frac{(2|k_a| + |k_d|)}{m}} \tag{2}$$

Where $\Delta X(s)$, $\Delta I(s)$, k_i, k_a, k_d and m are the rotor axial displacement from its nominal position, the instantaneous current in the electromagnetic actuator, the force/current factor, the magnetic bearing passive axial stiffness, the force/displacement factor and the rotor mass, respectively. Eq. (2) denominator is the characteristic equation of $G(s)$ and is used to determine both the system fundamental frequency and its stability. So, finding the roots of Eq. (2) yields:

$$s = \pm\sqrt{\frac{(2|k_a| + |k_d|)}{m}} \tag{3}$$

This equation shows that the system has one root located in the right side of the complex plane, i.e. the system is open loop unstable. Also de system fundamental frequency is $\omega_n = \sqrt{\frac{(2|k_a| + |k_d|)}{m}}$, $rad/_s$. A classical digital PID controller algorithm is used to stabilize the rotor axially.

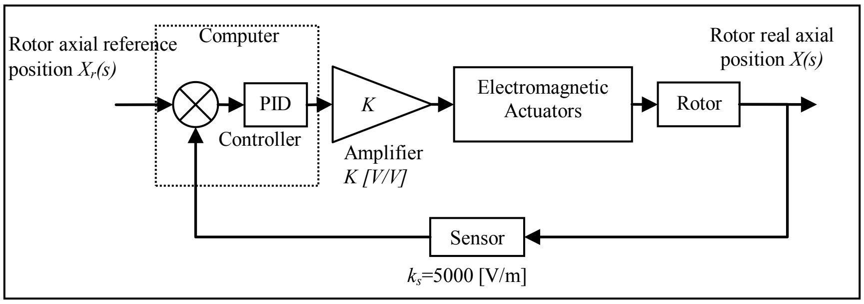


Figure 15: Block diagram for the axially controlled magnetic bearing

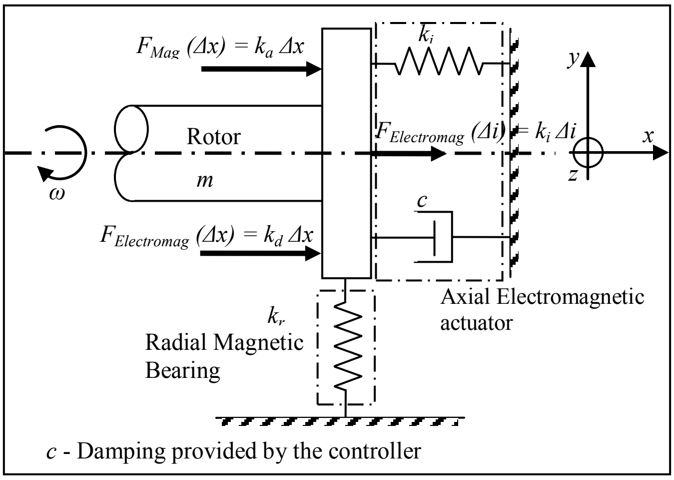


Figure 16: Axially controlled magnetic bearing dynamic diagram (half bearing only)

3. EXPERIMENTAL RESULTS

In order to demonstrate the effectiveness of the proposed hybrid magnetic bearing, a prototype was built, Fig. 17. The prototype main parts were made of aluminum alloy. The rotor was made of stainless steel and the electromagnetic actuator core and the rotor ends discs of SAE 1020 steel. All permanent magnets are NdFeB alloy. The rotor was accelerated using two air nozzles that applies compressed air against grooves machined in its central position.

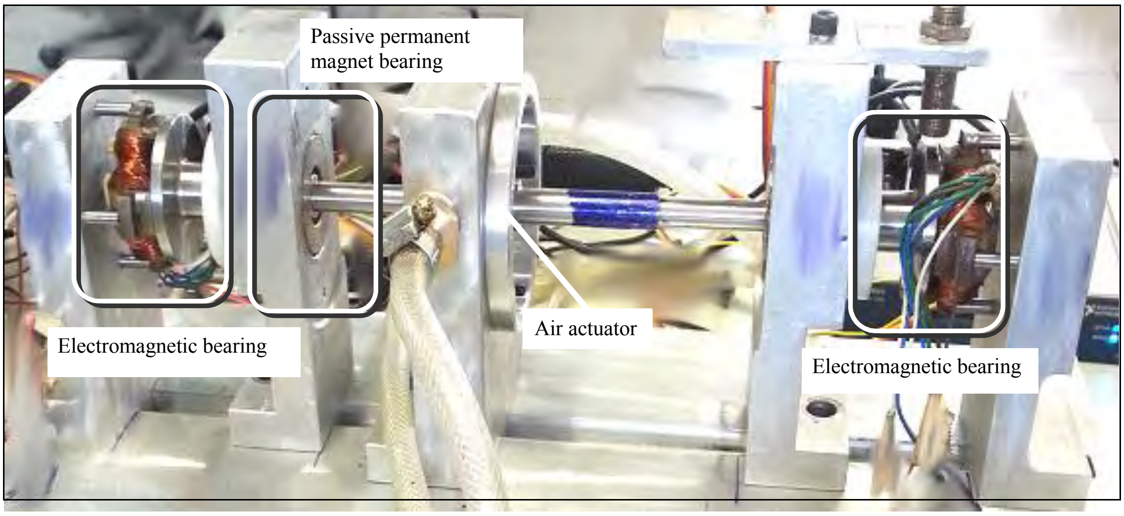


Figure 17: The prototype without the electrodynamic bearing

The first experimental analyses were done without the electrodynamic bearing, i.e. only with the radial passive magnetic bearings and the axial electromagnetic bearings. Afterwards, the electrodynamic bearing was incorporated to the prototype. Figure 18 presents the prototype with the three types of bearings. The rotor axial and radial displacements were measured by noncontact gap sensors and its angular speed by a noncontact capacitive sensor which reads two teeth machined in the steel disc used as the electromagnetic actuator target. Due to practical reasons, two rotor angular speeds were adopted for the experiments: 1500 and 1900 rpm.

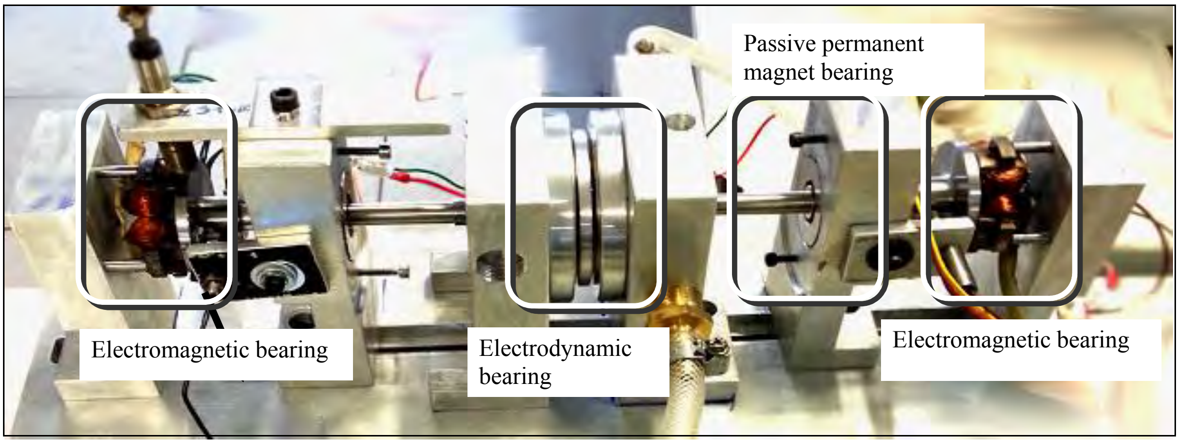


Figure 18: The prototype with three bearings

Figure 19 shows the rotor radial displacements at 1500 and 1900 rpm with and without the electrodynamic bearing. Observing this figure, one can see that the rotor radial oscillations are reduced by almost 50% and 28% when the rotor angular speeds are 1500 and 1900 rpm, respectively. Figure 20 depicts the rotor axial displacements. Also here, the amplitude of oscillations is attenuated when electrodynamic bearing is used. These results demonstrate the effectiveness of the electrodynamic bearing to improve both the bearing passive stiffness and the rotor damping.

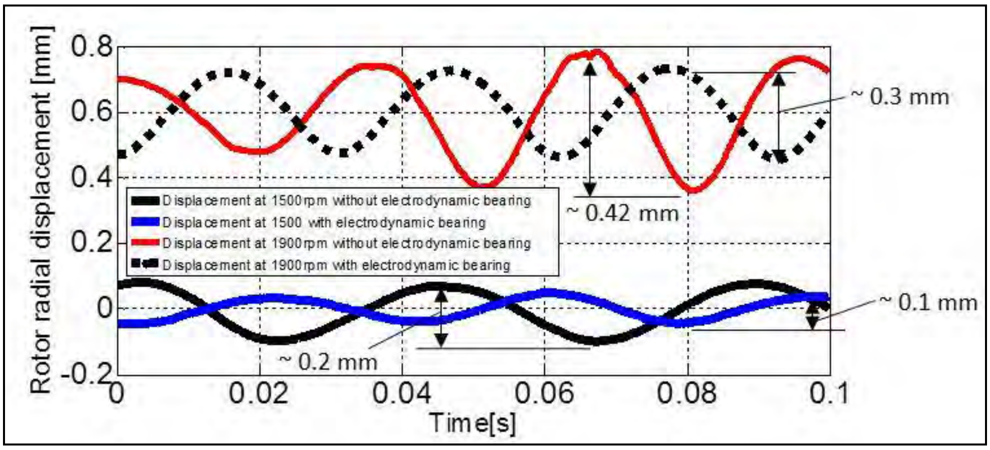


Figure 19: Rotor radial displacements at 1500 rpm and 1900 rpm with and without electrodynamic bearing

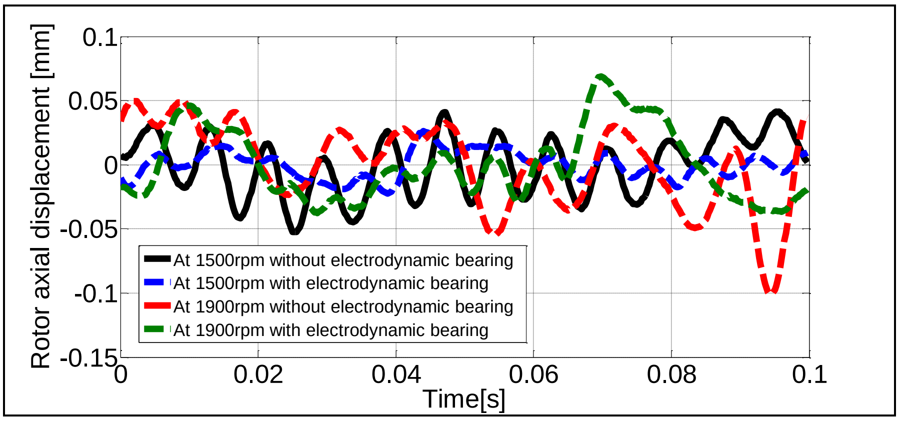


Figure 20: Rotor axial displacements at 1500 rpm and 1900 rpm with and without electrodynamic bearing

4. CONCLUSION

This work presented a new concept of hybrid magnetic bearing which can be applied in several mechatronic systems, e.g. electromechanical batteries called flywheels, artificial hearts pumps, turbomolecular pumps etc. The proposed bearing architecture was based on three magnetic bearings concept: i) passive magnet bearing, ii) electromagnetic active bearing and iii) electrodynamic bearing. The radial passive magnetic bearing assures a high radial stiffness. The negative axial stiffness of the passive bearing is compensated by the active axial electromagnetic bearing. Finally, the electrodynamic bearing contributes to increase the rotor passive axial and radial stiffness and also improving radial damping. Numerical simulations provided several data, which assisted the development of the proposed bearing and the experiments demonstrated, as expected, that the electrodynamic bearing improves the rotor radial stability and also reduces its radial and axial oscillations. Despite these improvements efforts will be continued to achieve more intensive reduction in the rotor oscillation. Also, the dynamic behavior of the rotor as well as the axial direction control analyses using the proposed hybrid magnetic bearing under higher rotor angular speeds will be analyzed in a future work.

5. ACKNOWLEDGMENT

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