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DESIGN DEFINITION OF THE PAD-SOLENOID SUBSYSTEM IN ACTIVE HYDRO-MAGNETIC BEARINGS

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Abstract. *Active hydro-magnetic bearings are hydrodynamic pad bearings with electromagnetic actuators, whose purpose is supporting the shaft and controlling the shaft lateral vibrations in rotating systems. In the design of the pad-solenoid subsystem of such active components, the pads must be sufficiently large to contain the solenoid, and the solenoid electromagnetic forces must be sufficiently large to overtake the oil film stiffness for a proper actuation condition. Considering that the oil film stiffness depends on the pad geometry itself, one has to find a compromise solution between pad geometry and solenoid size. In this work, a sensitivity analysis of the oil film stiffness is performed as a function of pad geometric parameters. For each pad geometric parameter, the equivalent force to overtake the oil film stiffness is calculated and compared to the electromagnetic forces provided for commercial solenoids. The results for each parameter are presented in charts that help finding the suitable solenoid, and respective pad geometry, for an effective actuation of the active bearing. The results show that the biggest solenoids are not necessarily the best choices. The proposed procedure represents a practical and straightforward method for the design of the pad-solenoid subsystem of hydro-magnetic bearings.*

Keywords: *active bearings, tilting-pad bearings, electromagnetism, parameter sensitivity, Reynolds equation*

1. INTRODUCTION

The advances in micro-computing and processing capacity are allowing engineers to investigate ways of expanding the performance of rotating systems. One of the ways of doing that is by controlling rotor vibration by adopting actuators and control system feedback. The control of rotor vibration represents an enlargement of machine operational range, since the machine is capable of adjusting to different operational conditions rather than those it was designed for (e.g. the increase of production flow demand). An additional advantage of controlling rotor vibration during machine operation is related to component failure (rubbing, blade loss, instability), whose consequences can be reduced until a machine stop can be programmed in appropriate time, and production flow is not strongly affected. As a consequence, active bearings began to be deeply investigated.

The first ideas of controlling vibration in rotating systems were based on active magnetic bearings (Keogh et al., 1995; Johnson et al., 2003; Desmidt et al., 2005). These bearings were firstly chosen because of their inherent electrical nature, which could easily endure on-line processing of sensor signals (control feedback). Additional advantages are the broad frequency band and contactless rotor actuation (Ulbrich,1993). However, magnetic bearings are prohibitively expensive when designed to large rotating systems, such as turbines and compressors. In such machines, the necessary forces for supporting the rotor are big, and the resulting magnetic system becomes complex and not compact. Besides, security systems such as auxiliary bearings must be inserted in the system to overcome electrical failures or system over load (Kasarda, 2000). Hence, the use of sole magnetic bearings is indicated to small/medium rotating system applications.

Active bearings based on hydrodynamic bearings were also investigated. The first ideas were proposed by Ulbrich and Althaus (1989), who applied piezoelectric actuators in the casing of a cylindrical journal bearing. Since then, many other ideas have been proposed for active actuation in hydrodynamic bearings, such as the hydraulic chambers (Althaus et al., 1993); the elastic casing geometry (Sun and Krodkiewski, 2000); the active squeeze film dampers (El-Shafei and Hathout, 1995); the active hydrostatic bearings (Bently et al., 2000; Santos and Watanabe, 2004); and the active lubrication (Nicoletti and Santos, 2005).

In this context, the concept of active hydro-magnetic bearings arises, aiming at taking advantage of the capabilities of both hydrodynamic and magnetic bearings. As one can see in Tab. 1, magnetic bearings have low load capacity when compared to hydrodynamic bearings. As a consequence, magnetic bearings must be robust (big, complex and expensive) to present similar load capacity to that presented by hydrodynamic bearings in a same application. If one adopts the hydrodynamic bearing as the supporting mechanism of the rotor, the size of the magnets can be reduced. In this case, the magnets become no longer a supporting + actuation mechanism (active magnetic bearing), but only an actuation mechanism (active magnetic actuator). Therefore, there is no need of a security system in the case of electrical failure, because the hydrodynamic bearing is the supporting element in the system, and it does not depend on electricity to work.

Table 1. Advantages and disadvantages of hydrodynamic and magnetic bearings (Ulbrich, 1994).

	advantages	disadvantages
tilting-pad hydrodynamic bearing	• high load capacity • decoupling between orthogonal directions • easy assembling	• additional peripherals (oil supply system) • dirty system • possible dynamic instability
magnetic bearing	• easy change of dynamic characteristics • null contact to rotor • clean system • broad frequency band	• low load capacity • complexity proportional to size • mechanical security system against electrical failure

The idea of associating the load capacity of hydrodynamic bearings with the actuating capacity of magnetic actuators has been investigated in literature in different ways. Most of the studied applications are related to electric motors whose rotors are supported by hydrodynamic bearings and driven by magnetic bearings (Eastman Kodak, 1996; Mechanical Tech, 1997; Chen and Zhang, 2002). Although these applications are very compact, the bearings are mounted as separated elements in the rotating system, being the hydrodynamic and the magnetic bearings distinct components. Besides, these design solutions are suitable for small machine applications. A more robust solution is proposed by Mechanical Tech (1991), where the hydrodynamic bearing casing is supported by a magnetic bearing. This solution presents both bearings mounted as a single system. However, the magnetic bearing works as a supporting + actuation system, being the hydrodynamic bearing merely used as a security system in the case of electrical failure. The load capacity of the hydrodynamic bearing is neglected by this design solution.

1.1 The Active Hydro-Magnetic Bearing

The active hydro-magnetic bearing in study is schematically shown in Fig.1. The rotor is supported by four tilting pads disposed in pairs in the orthogonal directions Y and Z. Oil is injected in the bearing through connections in the bearing casing, thus providing the necessary lubrication for the supporting mechanism (hydrodynamic lubrication). Commercial electromagnetic actuators (solenoids) are mounted in each of the pads, thus forming the actuation mechanism of the active bearing.

(1) rotor
(2) tilting pads
(3) solenoids
(4) bearing casing
(5) oil supply connections

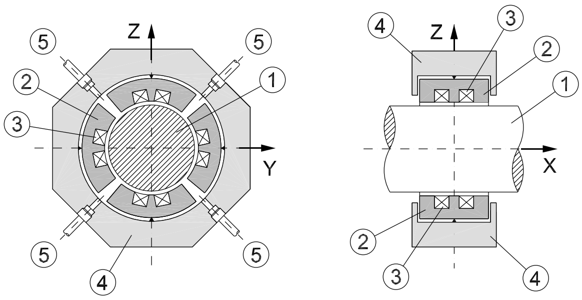

Figure 1. Schematic view of the active hydro-magnetic bearing.

One can implement a control system feedback by measuring the rotor movements (displacement and velocity) through proximity probes. The signals are filtered and on-line processed, multiplied by the control gains, and sent to the electromagnetic actuators, whose electromagnetic forces are used as control forces. The rotor movements are then controlled by the system.

1.2 The Compromise Problem Between Hydrodynamic and Electromagnetic Forces

In the design of active hydro-magnetic bearings, one has to guarantee that the electromagnetic forces, generated by the solenoids, can overcome the oil film equivalent stiffness in the bearing gap. If this does not occur, the solenoids will not be efficient enough to control the rotor movements.

The oil film equivalent stiffness depends on geometric and operational parameters, such as the bearing width (L_z), the pad aperture angle (α_0), the rotating frequency (Ω), the assembling clearance (h_0), the distance from pivot to sliding surface (Δs), the rotor radius (R), and the pad inner radius (R_s) – Fig.2. In general, the bigger the rotor-pad interface area is, the bigger the oil film equivalent stiffness will be (Hamrock et al., 1994).

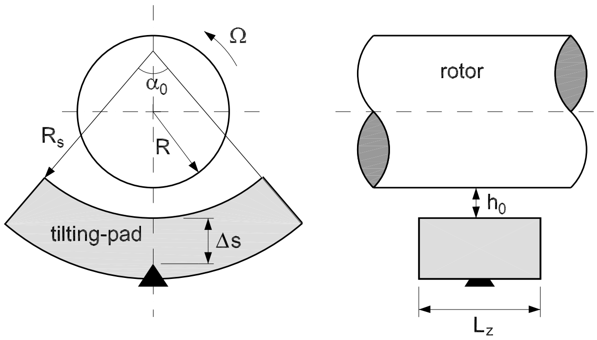

Figure 2. Geometric parameters that affect the oil film equivalent stiffness.

Similarly, the bigger the solenoids are, the bigger the generated electromagnetic forces will be. Considering that the solenoids are mounted in the bearing pads, if the solenoids are big, big will be the pads, and consequently the oil film equivalent stiffness will also be big. Therefore, one arrives at a compromise problem: the solenoids must be sufficiently large to generate the necessary electromagnetic forces to overcome the oil film stiffness, but sufficiently small to fit inside the bearing pads.

In this work, one performs a sensitivity analysis of the oil film equivalent stiffness as a function of the geometric parameters of the bearing pad. With this information, one can estimate the maximum displacement that the rotor will present under different electromagnetic actuation forces from the solenoids. Based on data provided by solenoid manufacturers, one can build design charts that will help choosing the appropriated solenoid for a given configuration of the bearing pad geometry, thus solving the compromise problem stated above.

2. MATHEMATICAL MODEL OF THE BEARING (HYDRODYNAMIC FORCES)

Given the bearing geometric properties and operational conditions, the hydrodynamic pressure distribution in the bearing gap is obtained by solving the bi-dimensional Reynolds equation over each i-th pad sliding surface:

$$\frac{\partial}{\partial \bar{x}}\left(\frac{h_i^3}{\mu}\frac{\partial p_i}{\partial \bar{x}}\right)+\frac{\partial}{\partial \bar{y}}\left(\frac{h_i^3}{\mu}\frac{\partial p_i}{\partial \bar{y}}\right)=6U\frac{\partial h_i}{\partial \bar{y}}+12\frac{\partial h_i}{\partial t} \tag{1}$$

where $p_i(\bar{x},\bar{y})$ is the hydrodynamic pressure in the i-th pad; $h_i(\bar{y})$ is the oil film thickness in the i-th pad; μ is the oil dynamic viscosity; U is the rotor surface velocity; ($\bar{x}$,$\bar{y}$) are the axial and tangential coordinates of a reference frame fixed on the pad sliding surface; and t is time.

The pressure distribution is integrated over each i-th pad surface area, resulting in the net forces F_{n_i} and F_{t_i}, which are the normal and the tangential hydrodynamic forces acting on the i-th pad surface. The resultant forces acting on the rotor and moments acting on the pads can be calculated as follows:

$$\begin{cases} F_{Y_h} = -\displaystyle\sum_{i=1}^{4}\left[F_{n_i}\cos\left(\phi_i+\alpha_i\right)-F_{t_i}\sin\left(\phi_i+\alpha_i\right)\right] \\ F_{Z_h} = -\displaystyle\sum_{i=1}^{4}\left[F_{n_i}\sin\left(\phi_i+\alpha_i\right)+F_{t_i}\cos\left(\phi_i+\alpha_i\right)\right] \end{cases} \quad \text{(hydrodynamic forces acting on the rotor)} \tag{2}$$

$$M_i = F_{t_i}\Delta s \quad \text{(hydrodynamic moments acting on the } i\text{-th pad)} \tag{3}$$

where ϕ_i is the angular arrangement position of the i-th pad in the bearing (they are arranged as 0^o, 90^o, 180^o and 270^o – Fig.1); and α_i is the angular position of the i-th pad in relation to its pivot (rotation around pivot).

Given the hydrodynamic forces and the external static loading applied to the rotor, one can numerically calculate the rotor-pads equilibrium position by the Newton-Raphson Method (Fig.3). Once the equilibrium position is known, the oil film equivalent stiffness is obtained by using the perturbation method proposed by Allaire et al. (1981).

3. SENSITIVITY ANALYSIS OF THE OIL FILM EQUIVALENT STIFFNESS

The sensitivity analysis of the oil film equivalent stiffness is performed as a function of the parameters shown in Fig.2, whose nominal values are presented in Tab.2. Each parameter is varied individually, keeping the remaining ones constant

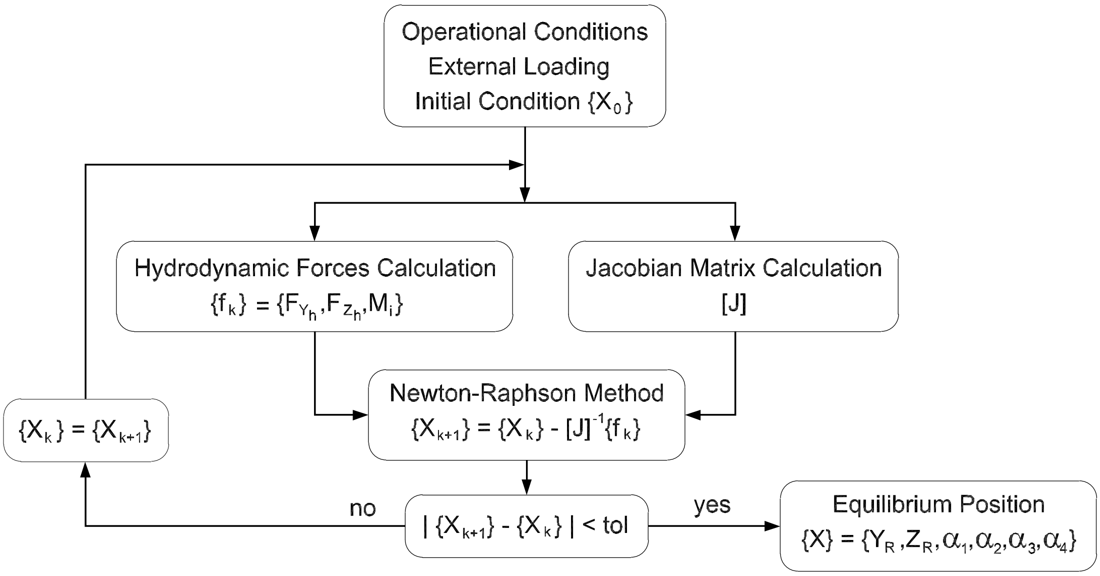

Figure 3. Flowchart of the algorithm for calculating the rotor-pads equilibrium position.

at their respective nominal values. Attention must be paid to parameters h_0 and R, because they are related to the bearing pre-load factor. In order to keep the pre-load factor constant all over the analysis, the following relationship is applied:

$$c = 1 - \frac{h_0}{R_s - R} = 0.9 = \text{constant} \tag{4}$$

where c is the pre-load factor; and R_s is the pad inner radius. Hence, by varying R and keeping h_0, one shall change R_s to keep the pre-load factor constant. This similarly occurs when R is constant and h_0 is varied.

Table 2. Nominal values of the design variables.

bearing width (L_z)	80 mm	assembling clearance (h_0)	200 μm
distance between pivot and pad surface (Δs)	30 mm	rotor radius (R)	40 mm
pad aperture angle (α_0)	60^o	pad inner radius (R_s)	42 mm
rotating frequency (Ω)	30 Hz		

In the analysis, the rotor remains centered in the bearing (no external loading), and one will focus on the direct stiffness. Because of the centered position of the rotor in the bearing, the equivalent stiffness in orthogonal directions will be equal ($K_{yy} = K_{zz} = K$).

Figure 4 presents the results for the oil film equivalent stiffness obtained by varying the parameters. As one can see, the oil film equivalent stiffness is linearly dependent on the bearing width (L_z – Fig.4(a)) and on the rotating frequency (Ω – Fig.4(d)), and it depends on the pad aperture angle in a near linear way (α_0 – Fig.4(c)). The variation of rotor radius (R – Fig.4(f)) increases the equivalent stiffness in a quadratic way. The variation of assembling clearance (h_0 – Fig.4(e)) decreases the equivalent stiffness in a near exponential way, resulting in the highest stiffness variation among the design parameters. Hence, the assembling clearance is the most relevant design variable. The distance between the pad pivot and the pad sliding surface has no effect on the oil film equivalent stiffness (Δs – Fig.4(b)), showing that it is the least relevant parameter and it can be excluded from the design variables.

4. SOLENOID LOAD CAPACITY CHARTS

The results presented in Fig.4 show the relationship between the geometric parameters of the bearing pads and the oil film equivalent stiffness. In order to choose the appropriated solenoid to be inserted in the pad, one has to find the relationship between the solenoid load capacity and the geometric parameters of the pad. Restating the compromise problem: the solenoids must be sufficiently large to generate the necessary electromagnetic forces to overcome the oil film stiffness, but sufficiently small to fit inside the bearing pads. For that, one calculates the resultant static displacement of the rotor when subjected to the maximum electromagnetic force generated by the solenoids, as follows:

$$\delta_{st} = \frac{F_{s_{max}}}{K} \tag{5}$$

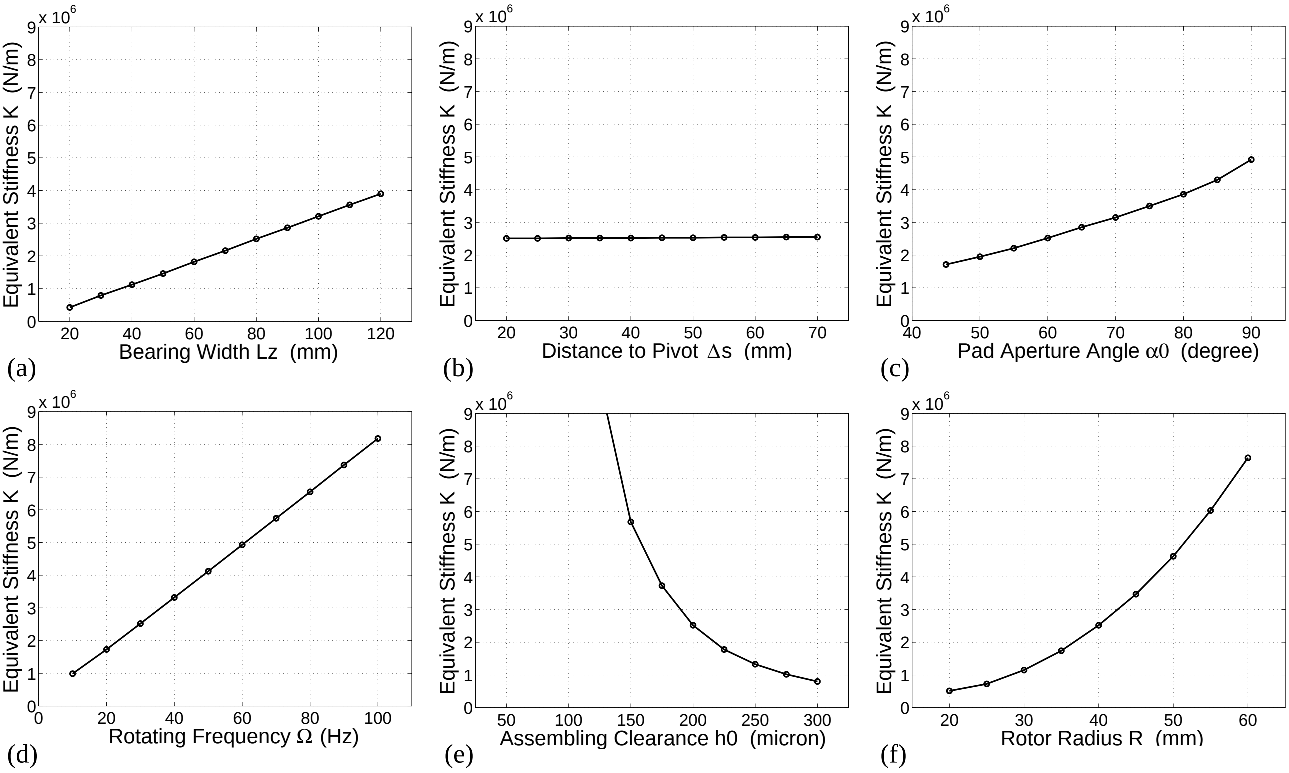

Figure 4. Oil film equivalent stiffness as a function of the design parameters (sensitivity analysis).

where δ_{st} is the rotor static displacement due to the maximum electromagnetic force; $F_{s_{max}}$ is the maximum electromagnetic force generated by the solenoid at maximum current supply (provided by the manufacturer); and K is the oil film equivalent stiffness.

The resultant rotor eccentricity can be calculated by dividing the static displacement by the assembling clearance:

$$\varepsilon = \frac{\delta_{st}}{h_0} = \frac{F_{s_{max}}}{K\, h_0} \tag{6}$$

where ε is the rotor eccentricity due to the maximum electromagnetic force.

Hence, depending on the geometry of the pad and solenoid size, the rotor will present the eccentricity ε when subjected to the maximum electromagnetic force that the solenoid can provide. An eccentricity of 1, or greater than 1, means that the solenoid, at maximum force, is able to pull the rotor towards the pad, zeroing the gap. An eccentricity smaller than 1 means that the solenoid is not able to zero the gap between rotor and pad. The bigger the rotor eccentricity is, the larger the solenoid load capacity in the active bearing will be. Although not physically feasible, an eccentricity greater than 1 means that the solenoid has a load capacity more than sufficient to close the gap between the rotor and the pad.

The solenoids in study are provided by Metalmag Produtos Magnéticos Ltda. A picture of the solenoid is shown in Fig.5, and the characteristics for the different solenoid models are presented in Tab.3. Figure 6 presents the rotor eccentricity as a function of the design parameters and the solenoid models.

Figure 5. Metalmag comercial solenoid.

Table 3. Characteristics of Metalmag solenoids.

model	diameter (mm)	thickness (mm)	maximum force (N)
30/40	30	40	100
40/40	40	40	210
50/45	50	45	380
75/50	75	50	930
100/60	100	60	1680

As one can see in Fig.6, in the case that the parameters have the nominal values (Tab.2), the solenoid models 30/40, 40/40, and 50/45 provide maximum forces that result in rotor eccentricities smaller than 1 (Tab.4, ε_0 column). This

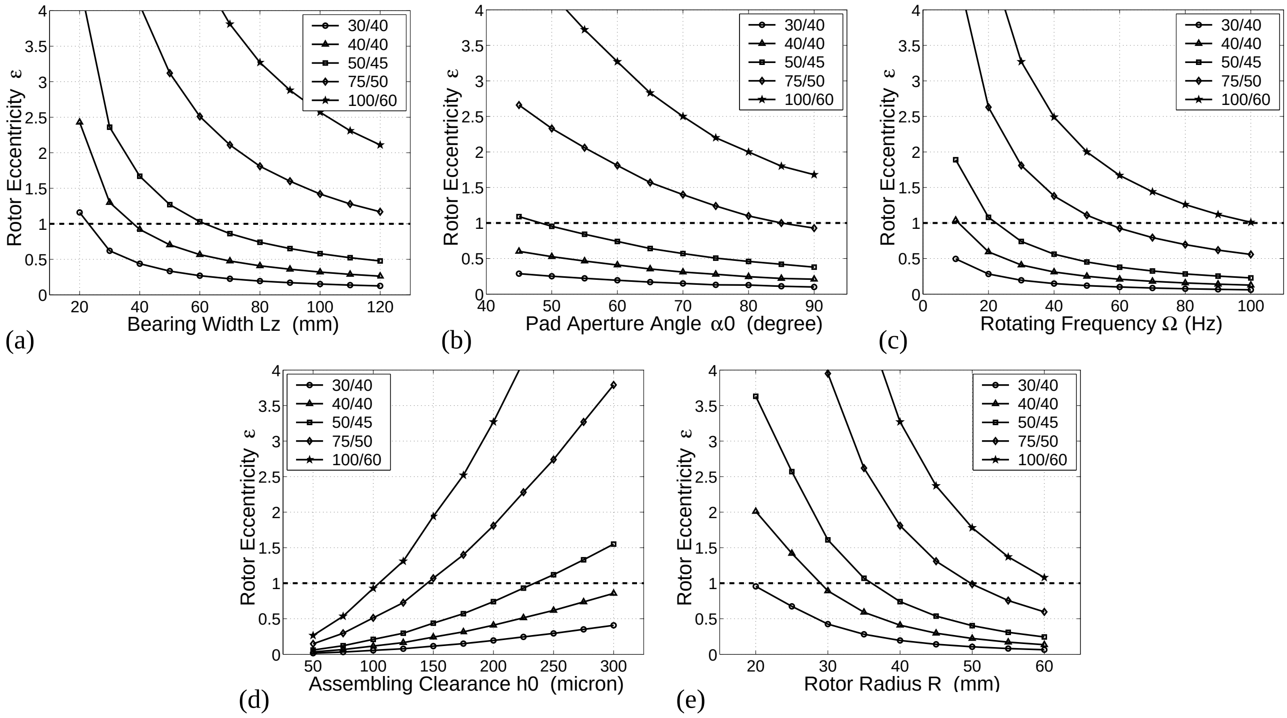

Figure 6. Rotor eccentricity as a function of the design parameters and solenoid models (solenoid load capacity charts).

Table 4. Rotor eccentricities due to geometric parameter values.

model	ε_0 (nominal values)[1]	ε_1 (minimum L_z)[2]	ε_2 (minimum R)[3]	ε_{12} (minimum L_z and R)[4]
30/40	0.2	0.6	1.4	1.8
40/40	0.4	0.9	2.0	2.5
50/45	0.75	1.25	2.6	3.1
75/50	1.8	1.95	3.4	3.55
100/60	3.25	2.6	1.8	0.9

[1] Design variables with nominal values presented in Tab.2.
[2] Bearing pad width with minimum allowable value (physical restriction) and remaining parameters with nominal values.
[3] Rotor radius with minimum allowable value (physical restriction) and remaining parameters with nominal values.
[4] Bearing pad width and rotor radius with minimum allowable values (combined physical restriction) and remaining parameters with nominal values.

means that these solenoid models are less suitable for actuation purposes in the active hydro-magnetic bearing, because their maximum force cannot overcome the hydrodynamic forces (oil film equivalent stiffness) resulted from the bearing geometry with nominal values. In this case, the use of such solenoid models in the active bearing could result to a poorer performance of the control system due to their smaller load capacity. Therefore, the choice of a solenoid model to be used in the active hydro-magnetic bearing with nominal geometric values lies between the 75/50 and 100/60 models.

One must still solve the compromise problem of fitting the solenoid inside the bearing pads, and fitting the pads inside the bearing. The first physical restriction related to this compromise problem is in the bearing pad width (L_z). The bearing pad width (L_z) must be larger than the solenoid diameter. Consequently, for the solenoid 75/50, the pad width must be greater than 75 mm, and for the solenoid 100/60, the pad width must be greater than 100 mm. If the parameter values are nominal, then the pad is 80 mm wide and the solenoid 100/60 does not fit in the pad. Hence, the solenoid 75/50 should be adopted in the active hydro-magnetic bearing with nominal values because it not only fits in the pad bearing, but also presents a reasonable load capacity (rotor eccentricity at maximum load of 1.8).

If one changes the bearing pad width to the minimum allowable values of each solenoid model ($L_z = 30\ mm$ for 30/40 model, $L_z = 40\ mm$ for 40/40 model, $L_z = 50\ mm$ for 50/45 model, $L_z = 75\ mm$ for 75/50 model, $L_z = 100\ mm$ for 100/60 model) and keeps the remaining parameters at nominal values, one arrives at the eccentricities shown in Tab.4, ε_1 column (data obtained from Fig.6). As one can see, there is an improvement in load capacity, indirectly represented by the increasing of rotor eccentricities at maximum load. However, the rotor eccentricities are still below 1 for the models

30/40 and 40/40. Besides, there is a reduction in load capacity of the 100/60 model because the pad bearing width had to be enlarged to fit the solenoid, which increased the oil film equivalent stiffness.

The second physical restriction of the system is related to the pad arrangement in the bearing. Considering that there will be four pads mounted in the active bearing, the following design restriction must be obeyed:

$$2\,(R + h_0) > D_s \tag{7}$$

where D_s is the solenoid diameter. This expression comes from the arrangement of four pads in the bearing, where one pad cannot overlap the neighboring pads (physical restriction – Fig.7).

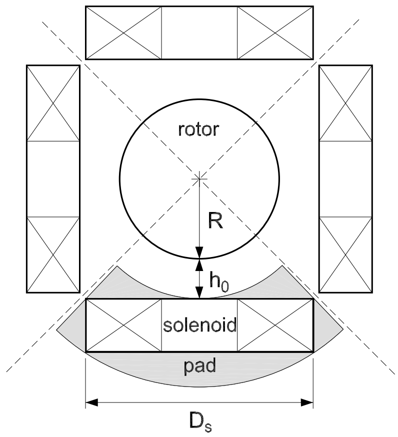

Figure 7. Physical restriction of the four pad configuration in the bearing.

Considering that $R \gg h_0$, the main limiting factor in this physical restriction is the rotor radius (R). Hence, the rotor radius must be greater than 15 mm for 30/40 model, 20 mm for 40/40 model, 25 mm for 50/45 model, 32.5 mm for 75/50 model, and 50 mm for 100/60 model, approximately. If one changes the rotor radius to the minimum allowable values of each solenoid model and keeps the remaining parameters at nominal values, one arrives at the eccentricities shown in Tab.4, ε_2 column (data obtained from Fig.6). As one can see, there is another improvement in load capacity, and all the solenoid models present rotor eccentricities at maximum load greater than 1. Again, there is a reduction in load capacity of the 100/60 model because the adjustment in rotor radius to fit the solenoid increased the oil film equivalent stiffness.

The combination of the two physical restrictions can be done by combining the rotor eccentricities as follows:

$$\varepsilon_{12} = \varepsilon_0 + (\varepsilon_1 - \varepsilon_0) + (\varepsilon_2 - \varepsilon_0) = \varepsilon_0 + \Delta\varepsilon_1 + \Delta\varepsilon_2 \tag{8}$$

By applying Eq.(8), one arrives at the values presented in Tab.4, ε_{12} column. As one can see, when the two physical restrictions are obeyed (combined minimum L_z and R with remaining parameters at nominal values) all solenoid models result in rotor eccentricities above 1.8, with exception of solenoid 100/60. The models 30/40, 40/40 and 50/45, which were not suitable for the application with nominal values, present reasonable load capacities when the values of L_z and R are the minimum allowable ones. In this case, the best choice would be the 75/50 model, because it results in a rotor eccentricity of 3.55 (maximum load capacity among the solenoids).

It is interesting to note that the 100/60 model presents the worst load capacity among all solenoids when the two physical restrictions are obeyed, although this model can generate the highest electromagnetic forces. The problem lies on the solenoid size. Because of the solenoid size, the bearing pad and the rotor radius have to be increased and, as a result, the oil film equivalent stiffness increases as well. This increase of oil film stiffness is not compensated by the increase in maximum actuation forces by choosing a bigger solenoid. In this case, the choice of the biggest solenoid (largest electromagnetic forces) is not advantageous.

5. CONCLUSIONS

This work presents a methodology for defining the pad-solenoid subsystem in active hydro-magnetic bearings, i.e. for solving the compromise problem between hydrodynamic and electromagnetic forces (geometry of the rotor and pads against the size and actuation forces of the solenoids). First, a sensitivity analysis of the oil film equivalent stiffness is done as a function of the bearing parameters (design variables). Second, the rotor eccentricity at maximum solenoid load is calculated for the commercial solenoids in study, thus forming the load capacity charts as a function of the design variables. Third, the two physical restrictions of the bearing are applied: $L_z > D_s$ and $2(R + h_0) > D_s$.

In the case that the design variables have nominal values, the solenoid 75/50 represents the best choice regarding load capacity. The smaller solenoids (30/40, 40/40, 50/45) present low load capacities, and the larger solenoid (100/60) does not fit in the bearing pad (it does not obey the first physical restriction).

In the case that the two physical restriction are obeyed, and the minimum allowable values of L_z and R are adopted with the remaining parameters at nominal values, the solenoid models 30/40, 40/40, 50/45, and 75/50 present reasonably high load capacities, and could be chosen. The best load capacity is obtained with the 75/50 solenoid. Curiously, the largest solenoid (100/60) presents low load capacity, although it is the one that generates the maximum electromagnetic force among all solenoids. The increase of actuation forces due to the solenoid size does not compensate the increase in oil film stiffness due to the increase of pad and bearing sizes.

The proposed methodology helped finding the best bearing geometry and solenoid size configuration regarding actuation efficiency, where the biggest solenoid was not necessarily the best choice.

6. ACKNOWLEDGEMENTS

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