USE OF MAGNETIC BEARINGS IN VIBRATION CONTROL OF A STEAM TURBINE WITH OIL-FILM BEARINGS

Rafael Pilotto, Rainer Nordmann, Heiko Atzrodt and Sven Herold

Fraunhofer Institute for Structural Durability and System Reliability – LBF, Darmstadt, Germany
email: rafael.pilotto@lbf.fraunhofer.de

It is well known that many problems in the field of noise and vibration occur in rotatory systems. To control such vibrations passive as well active measures are possible. The problems of using active control measures into rotating machinery are the energy transfer towards the rotating systems and also finding suited actuator concepts to control lateral vibrations. The most well-known passive supports in rotating machinery are the radial oil-film Bearings, widely used in turbomachinery as they allows the system to sustain high loads at high speeds. The availability and the reliability of this type of bearing is very high, since there is only fluid contact between parts with relative motion, ensuring long operational lifetime. Two of the most common bearing types are the widely used fixed-geometry bearings and the more expensive tilting-pad bearings. The tilting pad bearings are used in most extreme operational conditions due to the occurrence of rotor-bearing hydrodynamic instabilities, such as subsynchronous vibrations with high amplitude (oil-whirl and oil-whip). On the other side Active Magnetic Bearings (AMB) generate forces through magnetic fields, so there is no rotor-bearing contact (and no resulting instabilities). AMBs are used in compressors, turbines, aircrafts and other high-speed/high performance machinery. They allow an active vibration control of lateral vibrations. This paper contains the simulated vibration control of a Steam Turbine during a run-up up to the operational speed, supported in tilting- and fixed-pads bearings. The lateral vibrations are controlled either passively only by means of the RFBs or actively by substituting one of the bearings by an Active Magnetic Bearing. The results of both concepts – passive and active vibration control – are compared.
Keywords: Rotordynamics, Steam turbines, Vibration Control, Active Magnetic Bearings (AMB).

1. Introduction

With the emergence of new, and more complex, engineering problems, the development of new technologies becomes mandatory. The many knowledge fields of engineering, once independent nowadays are interrelated in order to develop new solutions and suitable methods to new patterns of testing and analysis.

Mechanical systems are always subjected to vibrations and therefore exposed to the temporal effects of fatigue and wear which can, in a short period of time, cause sudden losses in the properties (and therefore efficiency) of these systems. In this scenario it is often necessary to employ auxiliary devices to work in parallel to the main system, with the purpose of maintain the performance of the system as intended in the project specifications.

In the case of large turbines (both gas and steam models), the clearance gaps between rotating and non-rotating parts must be maintained in order to allow a safe transition throughout the resonance stage, where the amplitudes of vibration are the highest, during run-up/run-down without damaging any components. Using active vibration control to regulate these clearances can contribute to the reduction of undesirably high lateral vibrations, mainly bending oscillations of the rotors, and therefore increasing the efficiency of the machine.

The use of Active Magnetic Bearings (AMBs) arises as a possible solution to the vibration control problem, usually used as contactless suspension for rotating machinery parts; it also enables the application of compensating magnetic forces in the system, which have no mechanical interaction with the structure in analysis as seen in [1]. In this context, the expansion of the system by means of AMBs, sensors and controllers also provides the possibility acquiring the system's frequency responses during operation and thus continuously monitoring changes in its dynamic behavior (natural frequencies, critical speeds and damping).

This paper describes the comparison of the simulated vibration control of a Steam turbine system composed of seven shafts and supported by eight oil-film bearings. Two cases are studied: 1.The system with only oil-film bearings as supports. 2. The left-most bearing of the turbine is replaced by an active magnetic bearing. The results ,in harmonic and transient analysis, are compared.

The Methodology and theory section is divided in two parts. The first one gives an overview of the FEM model of the mechanical components of the system: Rotor shafts (Turbine) and supports (bearings and foundation), the second part comprises the electrical part of the system, specifically, the magnetic bearing model and the controller used. The Results section contains the comparison between the obtained simulated results from each case, followed by the Conclusions regarding the aforementioned results.

2. Methodology and Theory

2.1 Mechanical System

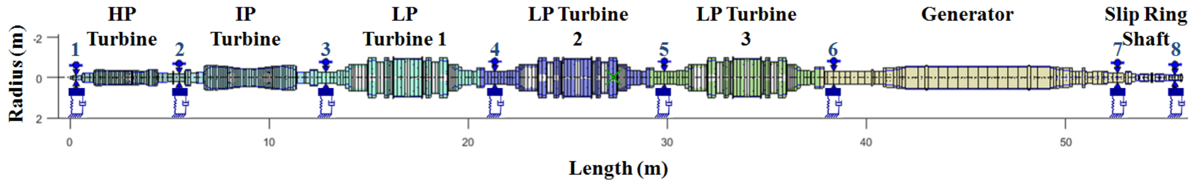
The model of the complete turbine system, which is composed of seven different shafts which are connected via rigid couplings, built using FEM in the software MADYN 2000 is shown in fig. 1(a). The whole turbine system has a length of 57m and weight of 398 tons.

The system is supported using radial oil-film bearings and each bearing (numbered in fig 1(a)) has an added mass-springer-damper system that simulates the foundation in which the turbine is assembled. The mass, stiffness and damping coefficients are located in Table 1.

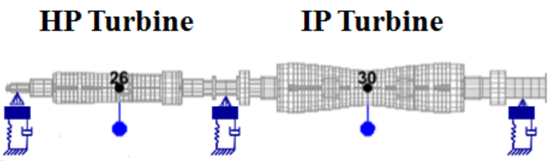
Table 1: Coefficients for the system foundation.

Location	Mass (kg)	Stiffness (N/m)	Damping(Ns/m)
1	x: 9950; y: 9950	x: 6.67e+9; y: 5e+9	x: 489000; y: 705000
2	x: 14090; y: 14090	x: 6.67e+9; y: 5e+9	x: 582000; y: 839000
3	x: 9000; y: 9000	x: 6.67e+9; y: 5e+9	x: 465000; y: 671000
4	x: 9000; y: 9000	x: 6.67e+9; y: 5e+9	x: 465000; y: 671000
5	x: 9000; y: 9000	x: 6.67e+9; y: 5e+9	x: 465000; y: 671000
6	x: 11620; y: 11620	x: 6.67e+9; y: 5e+9	x: 528000; y: 762000
7	x: 9370; y: 9370	x: 6.67e+9; y: 5e+9	x: 474000; y: 684000
8	x: 1850; y: 1850	x: 2e+9; y: 1.25e+9	x: 115000; y: 152000

There are two unbalances in the system, located at the middle part of the high pressure turbine and at the middle of the intermediate pressure turbine. The unbalances have a value of $8{,}12.10^{-2}$ kg.m and $2{,}51.10^{-1}$ kg, respectively. Each unbalance is going to be simulated individually. The unbalance locations can be seen in fig. 1(b).



(a)



(b)

Figure 1 – (a) Turbine FEM Model with the bearings and foundation locations numbered. (b) Turbine detail with the unbalance locations.

The equation of the FEM model can be seen in Eq. 1, where *[M]* is the global mass matrix, *[G]* the gyroscopic effect matrix, *[K]* the stiffness matrix and *[D]* is the structural damping matrix and Ω is the rotational speed of the shaft. The Damping matrix is assumed to be proportional to *[K]* by a factor of 2e-4, as estimated for steel shafts in [2].

The foundation is allocated in the complete system using the notations $[M_f]$, $[D_f]$ and $[K_f]$, meaning, respectively, the mass, damping and stiffness of each component of the foundation. The *[B]* matrix correspond to the coupling between the rotor and the foundation, they are null matrices with non-zero values only on the associated degrees of freedom.

All these matrices were obtained from the works of [3]. *{F}* is the external forces vector, in this case the unbalance located in each turbine, and the magnetic bearing force; *{q}* is the generalized coordinate vector, and $\{q_f\}$ is the associated generalized coordinate vector for the foundation of the system, each with two translations and two rotations for each node of the FEM model.

$$\begin{bmatrix} [M] & [B] \\ [B]^T & [M_f] \end{bmatrix} \begin{Bmatrix} \{\ddot{q}\} \\ \{\ddot{q_f}\} \end{Bmatrix} + \left(\begin{bmatrix} [D] & [B] \\ [B]^T & [D_f] \end{bmatrix} + \Omega \begin{bmatrix} [G] & 0 \\ 0 & 0 \end{bmatrix} \right) \begin{Bmatrix} \{\dot{q}\} \\ \{\dot{q_f}\} \end{Bmatrix} + \begin{bmatrix} [K] & [B] \\ [B]^T & [K_f] \end{bmatrix} \begin{Bmatrix} \{q\} \\ \{q_f\} \end{Bmatrix} = \begin{Bmatrix} \{F(t)\} \\ \{0\} \end{Bmatrix} \quad (1)$$

2.2 Electrical System

2.2.1 Active Magnetic Bearings (AMBs)

Considering a typical 8-pole radial magnetic bearing, the eight poles are divided into four electromagnets (fig. 2). Magnet 1 generates radial force in the x-axis direction whereas magnet 3 generates radial force in the opposite (−x-axis) direction. Therefore magnets 1 and 3 are working in differential mode; the same is true for magnets 2 and 4.

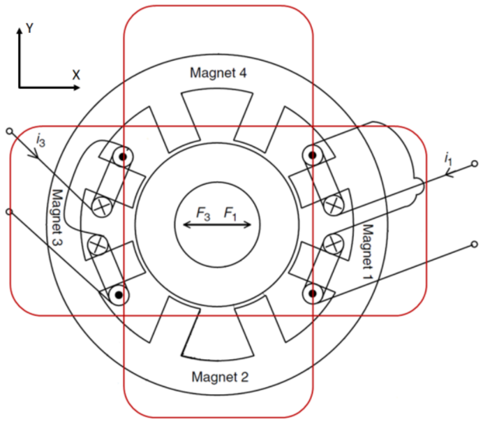


Figure 2 – Radial Magnetic Bearing model (Adapted form [1]) .

Considering only the x-axis for the mathematical deduction since the same assumptions are valid for the y-direction, we have the forces in each magnet:

$$F_1 = \frac{k'_i}{4} i_1^2 + \frac{k'_x}{2} x_1, \qquad F_3 = \frac{k'_i}{4} i_3^2 + \frac{k'_x}{2} x_3 \tag{2}$$

$$k'_i = \frac{N^2 \mu_0 S \cos(\pi/8)}{g^2}, \qquad k'_x = -\frac{N^2 \mu_0 S \cos\left(\frac{\pi}{8}\right) I_b^2}{g^3} \tag{3}$$

$$i_1 = I_b + i_x, \qquad i_3 = I_b - i_x \tag{4}$$

$$x_1 = g - x, \qquad x_3 = g + x \tag{5}$$

Where N is the sum of the number of winding turns of two short-pitched coils, $\mu 0$ is the permeability factor of the mean where the AMB is immersed (air), S is the area of one stator pole, g is the airgap between the pole and the turbine shaft, I_b is the bias (permanent) current and i_x is the regulating current (control) current and x is the instant separation between rotor and the AMB at any given time as seem in [1]. It can be seen that k_x has a negative sign once the greater the distance x the smaller the resulting force on the bearing [4]. Since $i_1 > i_3$, $x_1 < x_3$, as the greater the current, greater the attraction force and therefore smaller the distance between rotor and bearing is. Therefore for the resulting force in the x-axis and similarly for the y axis, we have:

$$F_{m_x} = F_1 - F_3 = k_i' I_b i_x - k'_x x = k_i i_x + k_x x \tag{6}$$

$$F_{m_y} = F_4 - F_2 = k_i' I_b i_y - k'_x y = k_i i_y + k_x y \tag{7}$$

Where k_i is known as the force-current factor and k_x is the force-displacement factor. Equations 6 and 7 show that the force of the magnetic bearings in the x-axis and y-axis are exclusively dependant of the regulating current i_x and the distance x between rotor and bearing. For this simulated AMB, the values of k_i and k_x are 5,03e+4 N/m and 1,01e+7 N/A.

2.2.2 PID Controller

The PID controller is composed by three components: proportional, integral and derivative [5]. The proportional term (K_p) is often related to the stiffness of the system, the integral term (K_i) is related to the duration of the error and its magnitude, eradicating the residual stationary error derived from the pure proportional controller. Finally, the derivative term (K_d) is associated with the damping of the system and helps to predict its behavior, improving the settling time and stability. The derivative term has a drawback once it amplifies high frequency measurements, which compromise the controller performance. This limitation can be overcome by adding a filter coefficient

(N_f) that acts as a low-pass filter in order to remove high-frequency noise components from the measurements. The controller with the added filter is given by Eq. (8).

$$PID(s) = K_p + \frac{K_i}{s} + \frac{K_d N_f s}{s + N_f} = \frac{(K_p + K_d N_f)s^2 + (K_p N_f + K_i)s + K_i N_f}{s^2 + N_f s} \tag{8}$$

The Figure 3 shows the PID model with the filter in the derivative part:

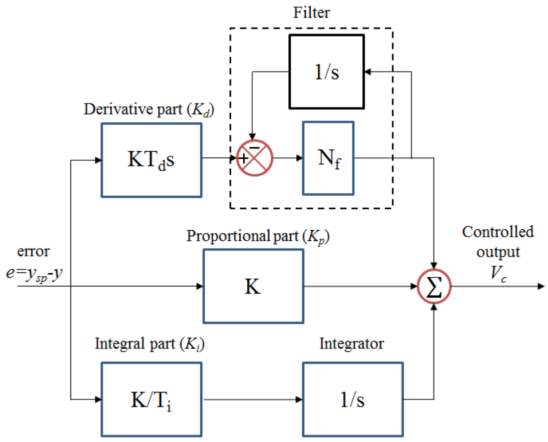


Figure 3 – PID model with derivative filter.

Where the error signal is the difference between the desired output signal (y_{sp}) and the real output signal (y). The parameters of the PID were obtained using the Ziegler-Nichols technique [6].

The flowchart seen in fig. 4 shows how the mechanical system and the electromagnetic system work in tandem. The displacements (in both directions x and y) resulting of the unbalance are measured by the sensor whose signal is compared to the reference signal (since the desired displacement is the minimum, the reference is set as zero), the error signal is sent to the controller, the output voltage of the controller is transformed into electric current by a power amplifier and then sent to the coils of the magnetic bearing, generating the magnetic force F_m that is applied to the mechanical system. The sensibility of the simulated displacement sensor is set to 1 V/mm and the power amplifier gain is set to 1.5 A/V.

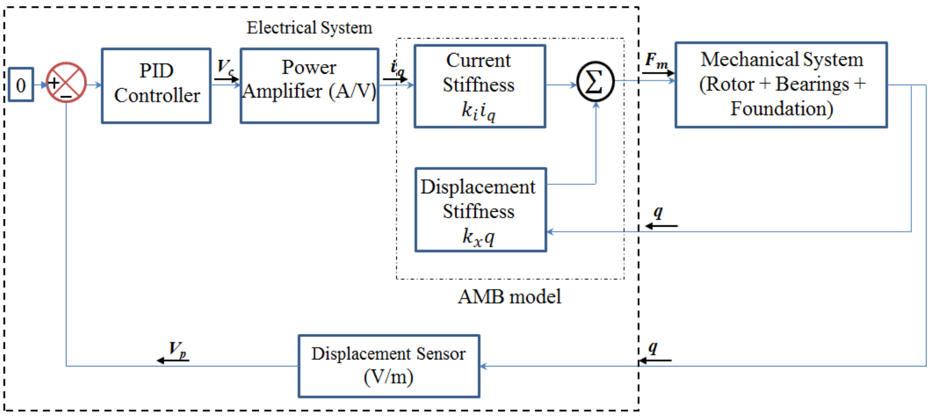


Figure 4 – Flowchart describing the system behavior.

3. Results

In this section, two analysis are studied, an unbalance response (Harmonic Analysis), and a run up simulation (Transient Analysis), to determine the influence of the AMB (Active Magnetic Bearing) substitution of the RFB (Radial Film Bearing) on the High Pressure Turbine inlet on the dynamic behavior of the system, as seen in Figure 4. It is noted, however that the foundation comp

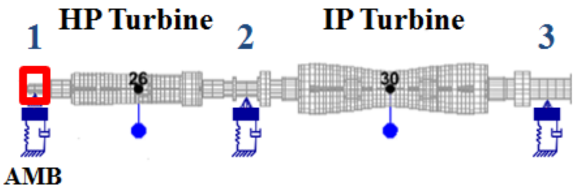


Figure 4- Detail of the location of the AMB, and the studied sections of the system

3.1 Harmonic Analysis

The figure 5 shows the unbalance response of the system on the location of the unbalances, i.e. on the center section of the High Pressure Turbine (HPT) and Intermediate Pressure Turbine (IPT), , respectively. It can be seen that the substitution of the Bearing 1 (Fig 4) for an AMB results in a reduction of the peak amplitude of vibration in the region of the critical speeds.

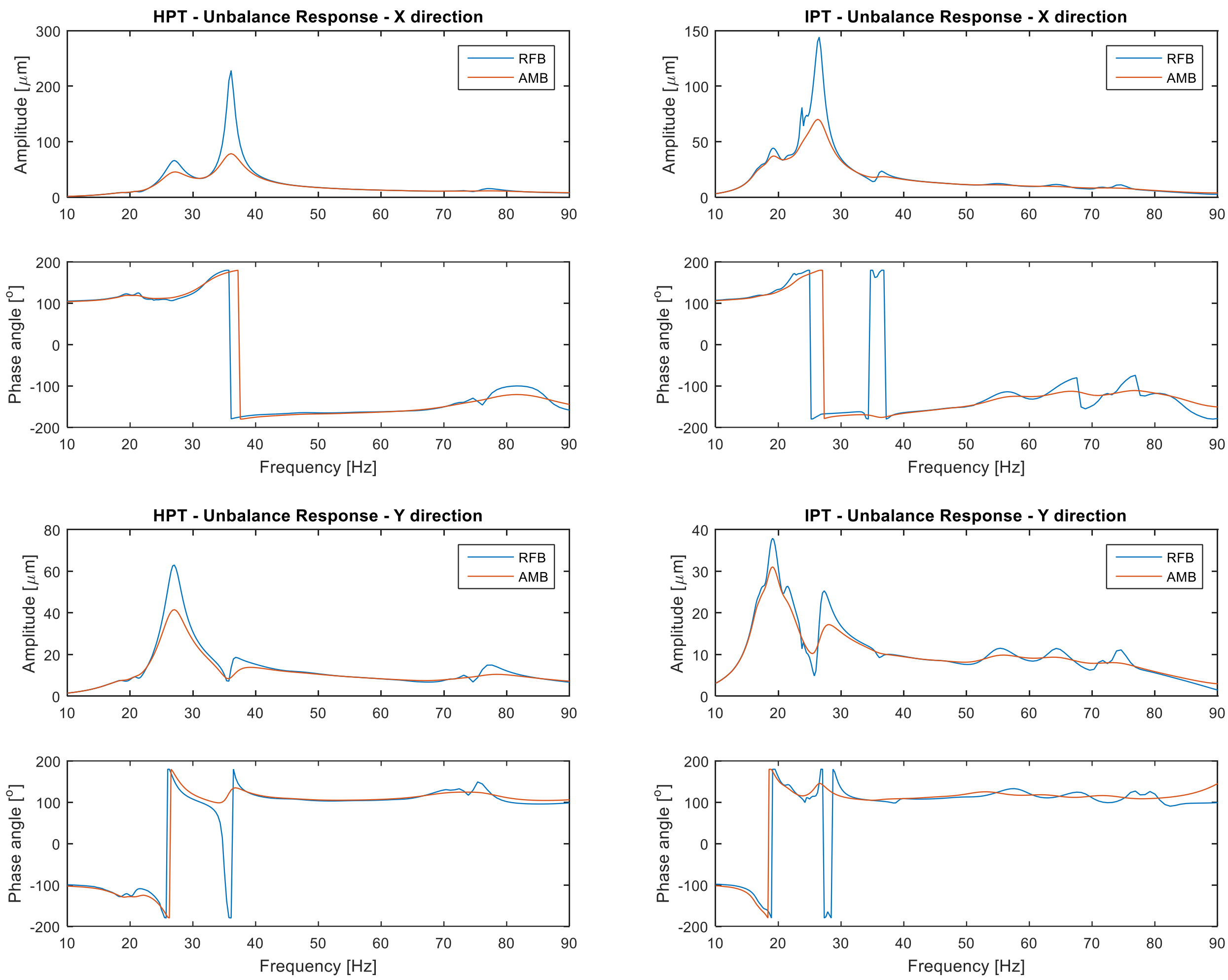


Figure 5– (a) Unbalance Response for the High Pressure Turbine. (b) Unbalance Response for the Intermediate Pressure Turbine.

For the High pressure turbine, a reduction of over 70% can be observed in the X direction and approximately 33% in the Y direction, and for the Intermediate pressure turbine, a reduction of 50% can be seen in the X direction and a small reduction of 20% in the 1st critical speed and 35% for the 2nd. Along with this, subtle changes on the critical speeds themselves are noted due to the changes in the mechanical system that the AMB provides.

3.2 Transient Analysis

For the Transient analysis, the simulated run-up covers from 10 Hz to 90 Hz in 5 seconds, and the amplitude of vibration was compared when the crossing of the critical speeds occurred. The results of these simulations regarding the displacement of the node where the unbalances are located (highest vibration amplitudes) can be seen in figure 6 (time domain response) and 7 (orbits).

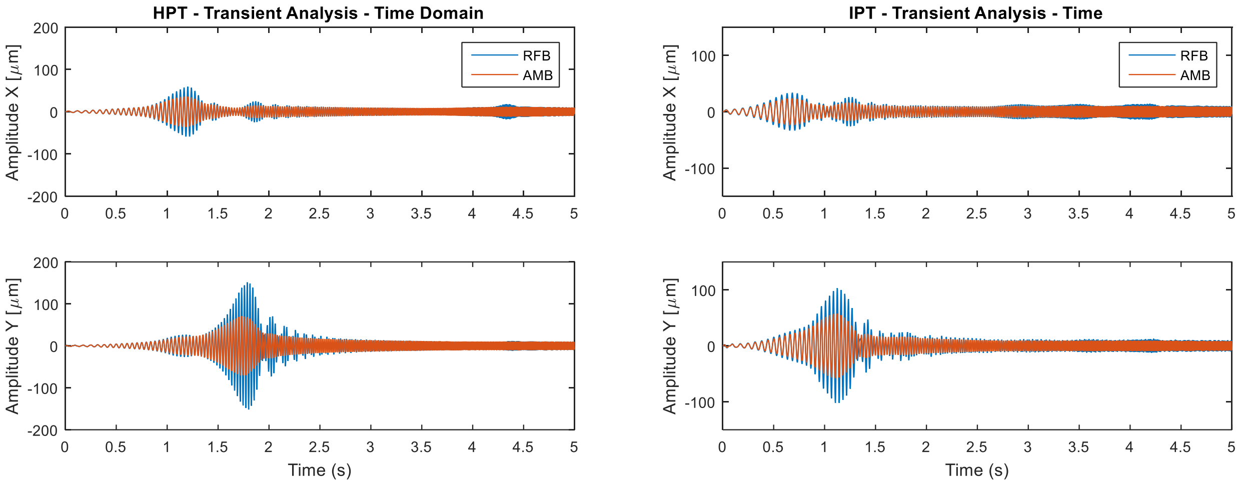


Figure 6 – (a) Transient analysis for the High Pressure Turbine. (b) Transient analysis for the Intermediate Pressure Turbine.

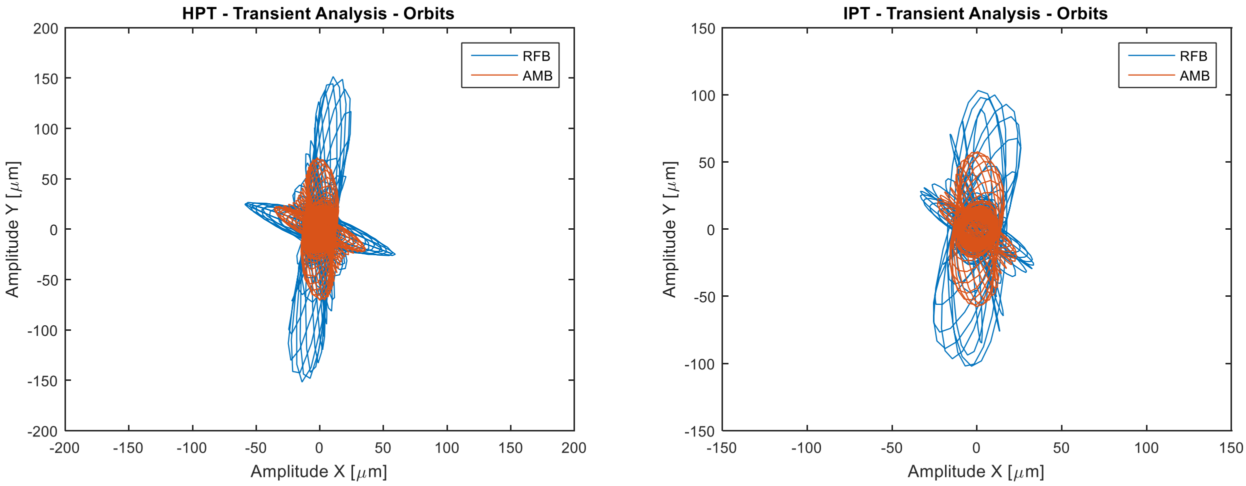


Figure 7 – (a) Orbits for High Pressure Turbine. (b) Orbits for the Intermediate Pressure Turbine.

It can be observed that the PID-Filter controller is capable of reducing the vibration amplitude in these two shafts. It can also be seen that the AMB was more effective in the HPT than in the IPT, this occurs since the AMB is positioned adjacent to the HPT and the IPT is still supported only by regular bearings. The peak reduction for the High pressure turbine reached 55% while in the Intermediate pressure turbine reached 40%. It can also be noted that after the critical speeds are through, the amplitude of vibration is practically the same with or without the AMB.

4. Conclusions

This paper presents the possibility of use of an Active Magnetic Bearing as a replacement for the regular Radial Film Bearing in the inlet of the High Pressure Turbine of a 3-Stage-Steam turbine system. The obtained results shown that the electrical system (AMB and PID-filter) system is able to reduce the vibration amplitudes during the passage of the critical speeds (highest amplitudes), where the system is most vulnerable. This shows the advantage of the active vibration control over its passive counterpart.

In spite of that, there are still improvements that must be done. Since the system is extremely large and complex (27 critical speeds up to 90Hz), only one AMB in the inlet of the HPT is not going to help when the far right shafts (Generator and Slip-ring shaft) are passing their critical speeds. In this view, it is important to verify the feasibility of the installation of more AMBs into the turbine and to verify which critical speeds are the most hazardous to the complete system. Other important improvement is to optimize the controller utilized; one option is the LPV (Linear Parameter varying) technique [7] to transform the AMB-PID system into an adaptive controller with optimized parameters for several rotational speeds, so it can be set for several critical speeds of a complex system, such as this one.

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