Energy Cost assessment of the Active Control of a the flexible rotor supported by AMB

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***Abstract*— From a designer point of view, optimizing the energy necessary for the control is an important element that could lead to downsize the control cabinet and to an increase of margins according to power amplifier capabilities. Also, reducing the energy used, generates a less significant environmental impact. Two fuzzy logic based controllers were developed and assessed. First, the "Mechanical" performances are compared with respect to an augmented PID controller. Then, the energy necessary for the control is compared.**

I. INTRODUCTION

Active Magnetic Bearings (AMB) are widely utilized in different industrial applications. Several applications have been successfully implemented in the field of Turbomachinery [1,2]. Several studies have been devoted to the elaboration of new controllers that enable better design and performance in operating situations, with acceptable levels of stability and robustness [3-7]. The most successful, for industrial applications, is the augmented PID controller, since it permits designers to master the control process. However, the control of flexible rotors remains difficult due to the unavoidable spillover effects.

Industrial requirements involve several constraints regarding performance, robustness, easiness of implementation and final tuning. These requirements are dictated by international standards such as ISO 14839 and API 617 [8-11]. The specifications of the final users must also be taken into account. The spectrum of applications covered by AMBs is nowadays largely diversified. Each application presents its own operating conditions and excitations.

Fuzzy controllers are well adapted for controlling flexible structures. The advantages of fuzzy control are that it can be used in complex systems such as nonlinear, time-variant and systems including uncertainties [12]. They are less sensitive to variations of system parameters and they allow the utilization of membership functions adapted to the dynamic behavior of the system considered. They also enable the calculation of nuanced actions and take into account several data variations, thereby ensuring robust behavior. Defoy et al. [13] developed a new control strategy that utilize polar coordinate. The originality of the methodology developed is that it manages two significant physical quantities, namely tangential and radial velocities, which are associated to steady state and transient behaviors respectively. The outputs are the forces computed in the polar coordinates and are converted into currents that drive the action lines at the same time.

In this paper the mechanical performances and also the energy necessary for the control of an academic test rig were assessed experimentally. The aim is to evaluate the possibility of optimizing the energy consumption that could lead to downsize of the control cabinet and to an increase of margins according to power amplifier capabilities. In addition, reducing the energy used will be transformed in a less significant environmental impact, which is a current concern nowadays.

First the test rig will be presented, then the different control strategies will be described and finally the results will be discussed.

II. TEST RIG

The experiments were performed using an academic test rig (Fig. 1). It is a commercial product manufactured by SKF® and was delivered with a dedicated PID controller. The test rig is equipped with two identical AMB called NDE (Non Drive End) and DE (Drive End) bearings. Each bearing has a maximum static capability of 280N. The action lines are positioned in the configuration load between axes. They are powered in differential driving mode with a bias current of 1A. Current are provided in the range of 0-3A using PWM amplifiers. Two displacement sensors (variable reluctance probes) are integrated in the housing of each bearing and are non-colocalised with actuators. The Input/Output panel gives access to the displacements measured and enables entering current settings for the amplifiers. Each AMB has one back-up bearing with a clearance radius of 0.1mm.

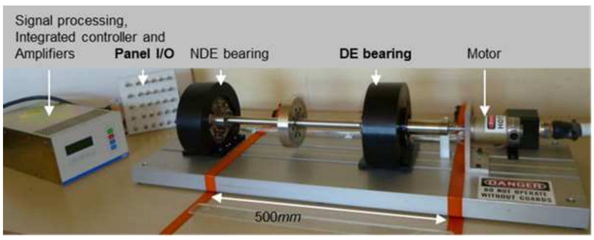

Figure 1. Test rig

The shaft is composed of three parts bolted together. A central part (diameter: 19.05mm; length 344mm) with a disc 120mm in diameter and 25mm long placed at mid-span, together with two shaft ends (35mm of main diameter). The stack of laminated steel sheets is shrunk on each of these two shafts. Due to the different lengths of the latter (190.5mm at DE and 110.5mm at NDE), the rotor is not symmetric in relation to the central disk. The total rotor length is 645mm. The rotor mass is 5.89kg. Three balancing planes are available: one at the central disc and one at each bolted plane. The rotor is driven by a 500W electric motor with a maximum speed of 12,600rpm. Power transmission is provided by a flexible coupling. The operating speed range used in this work is 0 to 10,000rpm, which includes three critical speeds (two forward rigid modes and the first forward flexible mode). The speed of the rotor is monitored by using a speed sensor placed close to the motor.

The frequency bandwidth considered in this study is 0-2kHz. The test rig can be controlled either by using the dedicated device or externally via the input/output panel. The new controller is implemented under Simulink® and dSpace® environments, with a sampling frequency of 20kHz.

The AMB is powered in differential driving mode with a bias current I0 and a control current i (1).

$$F_B = 4\,\xi_B\,\mu_0\,S\,N^2 \left[\frac{(I_0 - i)^2}{\left(2\,g_0 - 2\,x + \frac{L}{\mu_r}\right)^2} - \frac{(I_0 + i)^2}{\left(2\,g_0 + 2\,x + \frac{L}{\mu_r}\right)^2} \right] \quad (1)$$

Where ξ_B is a coefficient depending on the bearing geometry (0.92 in our case), μ_0 the permeability of vacuum space, *S* the pole area, *N* the number of turn in one coil, g_0 the nominal air gap, *x* the rotor position, μ_r the relative permeability of the iron core and *L* the average length of magnetic flux lines.

III. CONTROL STRATEGIES

Three control strategies were assessed. The first is the one developed by the test rig constructor, a fuzzy controller and a polar fuzzy controller (Fig. 2).

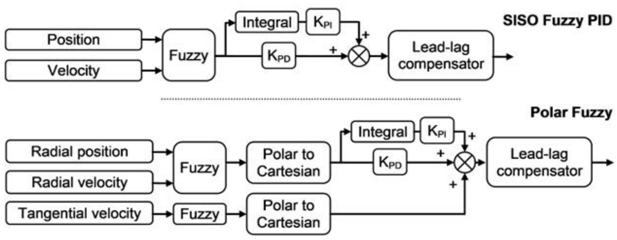

Figure 2. Schemes of the controllers

A. *Fuzzy Controller*

It consist of the sum of a fuzzy PI with a fuzzy PD controller. The displacement was measured and the velocity calculated by numerical derivation. The two data (displacement and velocity) were used as inputs for the fuzzy controller. Four generalized bell shape membership functions were utilized [14]. These membership functions are associated with four fuzzy sets: positive/negative displacements and positive/negative velocities

The design of a membership function take into account the noise of measurement. The displacements and velocities are measured, and there respective amplitude gives directly the measurement uncertainty. These amplitudes of displacements and velocities should be insufficient to change the prevalent membership function. On the other hand, in order to be efficient small amplitude of displacement and velocity should fully belong to one of the two fuzzy sets (positive or negative). Finally, as nominal behavior is considered and, the displacements and velocities are relatively small compared to noise of measurement, only two membership functions were utilized for each input of the fuzzy controller.

The rules are chosen in order to optimize the energy dissipation and to minimize the kinetic energy of the rotor (the force is always opposite to the velocity). The rules selected were:

- When position and velocity have the same sign, then the proportional gain is activated. Note that the other gains are always activated.
- When the displacement and velocity have opposite signs, the damping gain is conserved but the stiffness is set to zero.

This study considers only the case of nominal behavior with low displacements. Consequently, if the membership functions are tuned too smoothly, a linear controller can approximate the non-linear one. And, if the membership functions are tuned with too sharp transitions, disturbances can appear, and can lead to system instability. In our case, where the ratio of tolerated displacements over the uncertainties due to measurements is small, there is a limited range of variation for the membership functions that enables an interesting benefit of the fuzzy controllers without disturbing the system.

B. *Polar Controller*

Generally, the response of the system, as acceleration or displacement, is measured by using sensors placed along the structure. Regarding the lateral behavior, two sensors oriented in two perpendicular axes are sufficient to describe the dynamic behavior. Then, the measured signals are processed and analyzed either in time, frequency or time-frequency domains.

We believe that this approach is quite suitable for non-rotating structures, but when dealing with rotating machinery there is a lack of information regarding the rotational speed. This lack could be made up by describing the response measured in polar coordinates. The observation of the measurements in the polar domain can lead to an easier interpretation of the dynamic behavior, where the steady state and the transient behaviors could be directly distinguished, particularly in the case of real time controlled systems.

The transformation from Cartesian to polar representation is obtained classically as indicated in (2). Where x and y are the measured displacements along the x and y direction in the

Cartesian representation, while r and θ are the corresponding polar quantities.

$$\begin{cases} r = \sqrt{x^2 + y^2} \\ \tan\theta = \dfrac{y}{x} \end{cases} \tag{2}$$

Then the radial and the tangential velocities, Vr and Vt respectively, are calculated as follow:

$$\begin{aligned} V_r &= \dot{r} \\ V_t &= r\,\dot{\theta} \end{aligned} \tag{3}$$

Where $\dot{r}$ is obtained by numerical differentiation of the radial position.

In a steady state case and in the presence of synchronous excitation as unbalance, the orbit is circular (symmetric rotor), the radial displacement is constant consequently the radial velocity is nil while the tangential velocity is constant. It is worth mentioning that in the case of system dissymmetry, the displacement orbit is elliptical and the radial speed and position as well as the tangential speed have harmonic variation that corresponds to the second harmonic of the rotating speed.

The actuating in the polar domain has also several advantages by introducing targeted action on stiffness and damping. In this work, the controller was developed in the polar coordinate system, measurements and actuating are still realized in the Cartesian coordinate system. This is due to the fact that technology used is designed to operate in one direction.

The physical quantities introduced by polar transformation enable the controller to distinguish the disturbance produced by the unbalance from the transient disturbance that excites the rotor radially. Indeed, when the rotor is submitted only to unbalance excitations, the rotor orbit is circular and the radius of the whirl orbit is constant or change slowly during run up. The radial velocity is almost nil, the variations are only due to the noise of measurement.

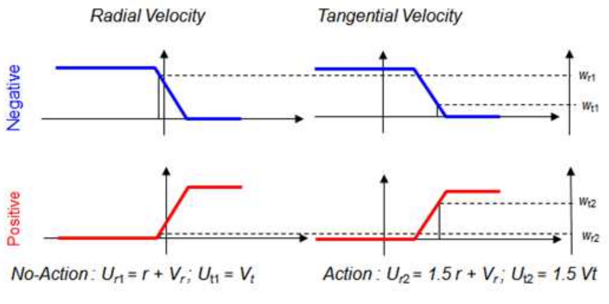

Figure 3. Membership functions of the polar fuzzy controller

Four trapezoidal membership functions were utilized (Fig. 3). The rules selected are:

1. For steady-state behavior, when tangential velocity is high tangential damping is increased by 50%.
2. For transient behavior, when radial velocity is positive, radial stiffness is increased by 50%.
3. If the velocities variation is small or negative, there is no action.

Then the fuzzy controller commands are expressed as actions along x and y direction:

$$\begin{cases} F_x = F_r\cos\theta - F_t\sin\theta \\ F_y = F_r\sin\theta + F_t\cos\theta \end{cases} \tag{4}$$

IV. EXPERIMENTAL RESULTS

The fuzzy controllers were compared with the initial augmented PID controller delivered with the test rig. The unbalance responses during run-up from 0 to 10,000rpm in 100 seconds was assessed.

Results are presented among the two direction of the drive end bearing (V13 & W13) and the non-drive end (V24 & W24).

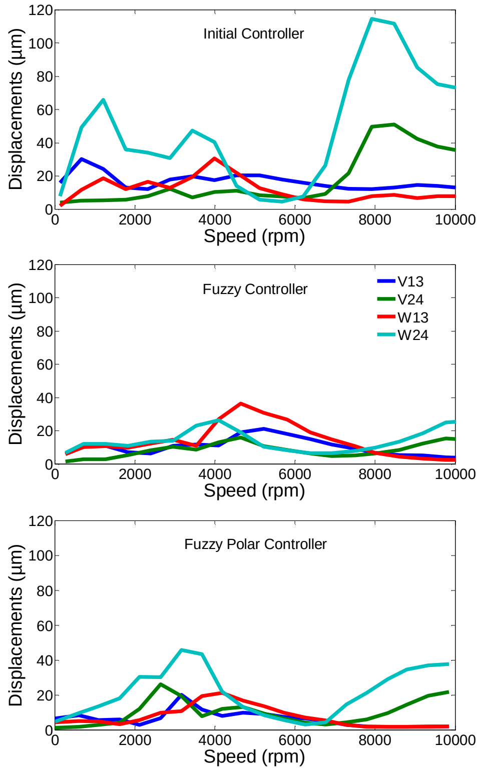

Figure 4. Unbalance responses during run-up

First, the displacement levels resulting from the three controllers assessed are compared in figure 4. The rms value of the displacement was chosen. The unbalance responses exhibited three critical speeds: the first around 1000rpm, that occurred only when using the initial PID controller was identified as foundation mode; the second in the vicinity of 4000rpm, corresponds to the cylindrical mode; and the last, near 9000rpm, was identified as the first flexible critical speed. The conical mode was damped too strongly to appear in the unbalance response.

The three controllers provided performances that match the requested specifications. Form a performance point of view, the fuzzy controller exhibits the best results. Nevertheless, this controller exhibited anisotropic behavior that could be undesirable. This anisotropy is due to the fact that the membership functions were expressed in the Cartesian coordinate system. Consequently, the control forces generated, when reaching displacement thresholds, lead to square orbit shape.

The rms value of the current was calculated (Fig. 5). It is representative of the energy necessary for the control in each case. No calculations, at this stage, were performed to quantify the energy. We estimate that the rms value is properly representative of the energy consumed.

The aim is not to minimize the energy necessary for the control, which is negligible with respect to the energy necessary to perform the run-up of the system. But as the AMB are designed (even partially) as a function of the max current, we believe that the controller with minimum energy could lead to a downsizing of the control cabinet and to an increase of margins according to power amplifier capabilities.

The results obtained show that the fuzzy polar controller represent the optimal behavior performance / consumption.

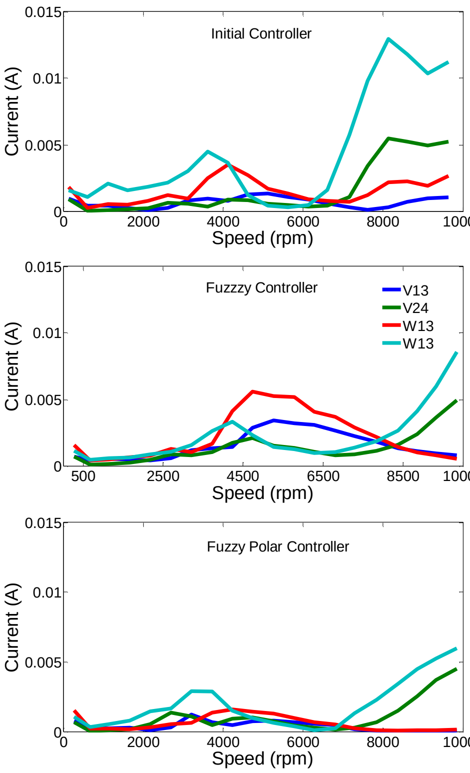

Figure 5. Current consumption during run-up

V. Conclusion

The mechanical performances and the energy necessary for the control of an academic test rig were assessed experimentally. The aim is to evaluate the possibility of optimizing the energy consumption that could lead to downsize of the control cabinet and to an increase of margins according to power amplifier capabilities.

Two fuzzy controllers were presented and compared experimentally to a PID controller. Better performances with fuzzy controllers were obtained. The energy consumption was compared also. It could be noticed that the Fuzzy controllers are significantly less energy consuming.

The idea is to select a controller, with an optimized energy consumption / mechanical behavior, aiming at optimizing the design of AMB. The results obtained are encouraging for the ongoing development.

Only those three controllers were assessed. Effectively, for a completeness of the study more controllers have to be assessed, but at this stage, the aim was to evaluate the fuzzy logic bases strategies for the control of turbomachines.

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