Numerical Assessment of a Polar Fuzzy Controller for Centrifugal Compressors

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Abstract

The aim of this study was to develop and to apply a new control approach dedicated to turbomachinery. The controller is fuzzy based using inputs expressed in polar coordinates. The advantage is that it manages two significant physical quantities, namely tangential and radial speeds that are related to steady state and transient behaviors respectively. A simple synchronous filter is associated to the controller in order to enhance the ratio command force / bearing dynamic capacity. The approach was previously applied experimentally with success for the control of an academic test rig. It is adapted here for the control of an industrial compressor whose flexible rotor is supported by Active Magnetic Bearings (AMB). At this stage, only numerical investigations are performed. The controller has to satisfy the standards and the end users requirements. In addition, it should be easy to implement. The behavior of the machine studied is assessed for several configurations of unbalances. Tests that correspond to usual industrial excitations (subsynchronous excitation, impulse response at nominal speed) are also carried out. Results obtained are satisfactory and give insight into the potential of the approach. In addition, and as the fuzzy controller parameters are independent from the rotor design, the approach is a first step for the standardization of magnetic bearing controller synthesis.

1 Introduction

Due to the progress made in electronics, AMBs are now widely used in different industrial applications and have been successfully implemented in the field of turbomachinery [1,2,3]. Their main advantages are that they provide a contact-less working environment, no sealing constraints, frictionless suspension, and that they constitute an active system [4]. On the other hand, AMBs produce only electromagnetic attraction and have nonlinear characteristics; therefore AMBs are inherently unstable and require feedback control to ensure stable operation.

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 [5,6,7,8]. The most successful of these controllers is the μ-synthesis controller whose parameters can be tuned automatically [4,9,10]. But, as the order of the controller is supposed to be at least that of the system plus the weighting functions, the optimization process becomes a delicate operation. The reason is that the number of specifications is greater than the number of available parameters in a digital signal processor. Consequently, the augmented PID controller is still widely used in industrial applications, since it permits designers to master the control process. However, the control of flexible rotors remains difficult due to unavoidable spillover effects. This problem is currently overcome by using phase lag and phase lead filters [11,12]. In this case also, tuning these controllers is a delicate operation and is time consuming.

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 [13,14,15,16]. 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. For a centrifugal compressor, inherent disturbances are generated by gas turbulences and lead to undesirable excitations. In addition, annular gas seals are utilized to prevent internal gas leakage. Whatever the technology used, these seals introduce a negative effective damping at low frequency [17]. Consequently, a given control strategy could not match all the particular requirements. Thus, it is interesting to develop a specific

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controller for a range product. The controllers have to be relatively standard in order to be easily adjusted to each machine. In this paper, this motivation was considered as a major requirement for the elaboration of the controller.

Generally, the controller is not sufficient to obtain acceptable unbalance responses. Thus, the specifications are satisfied by using general notch filters. These filters utilize the tachometer signal provided by rotational speed sensor. Synchronous filters are widespread in industrial applications. Different methods exist and can be classified in three different groups in function of the main goal [4]: the filters for the attenuation of forces, those for the attenuation of vibrations and finally for the generation of synchronous damping. However, the implementation of these filters requires additional hardware and obviously affects the global system reliability. Among these methods, just few do not need a rotor speed sensor [18,19].

Previous studies have shown that fuzzy controller approaches are well adapted for controlling flexible structures [20,21]. The advantages of fuzzy control are that it can be used in complex systems such as nonlinear, time-variant and systems including uncertainties [22]. 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 [5,6,23,24,25]. The nonlinearity of this type of controller offers several possibilities for controlling disturbances. Nevertheless, in the case of rotating machinery this nonlinearity can involve undesirable anisotropic behaviors, especially when applied in SISO controllers. However, fuzzy controllers can generally manage parameters in the time domain, so the approach used differs from the usual approaches utilized in industrial applications which are frequency domain based.

Previous works have shown the effectiveness of the controller developed on an academic test rig equipped with a flexible rotor supported by two radial AMBs. The controller considers each bearing as a single MIMO system with the displacements in the two orthogonal directions as inputs. This type of MIMO control has been used previously with cross control to compensate flywheel gyroscopic effects [26,27]. It was also applied to generate synchronal damping by using general notch filters associated with cross stiffness [28].

The work presented in this paper is a part of a research program to develop new control approaches dedicated to turbomachinery. The originality of the methodology developed is that it manages two significant physical quantities, namely tangential and radial speeds, 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. This controller enables tackling undesirable anisotropic behavior by using polar coordinates. A Sugeno algorithm with simple rules is used. In addition, a simple notch filter enables the reduction of synchronous stiffness when passing through the first bending critical speed. Thus, the control forces are reduced and the vibrations too.

The paper is divided into several sections. First, the model of the compressor used for numerical computation is described, and then the control strategies are shown. Only numerical results are presented. The controller is assessed for different configurations that correspond to the standard and the compressor final user specifications. In this work the unbalance response due to three unbalance repartitions is examined, then the effect of subsynchronous excitation is shown and the impulse response at nominal speed is analyzed. Finally, the advantages and disadvantages are highlighted and the conclusions are summarized in the final section.

2 Detailed model

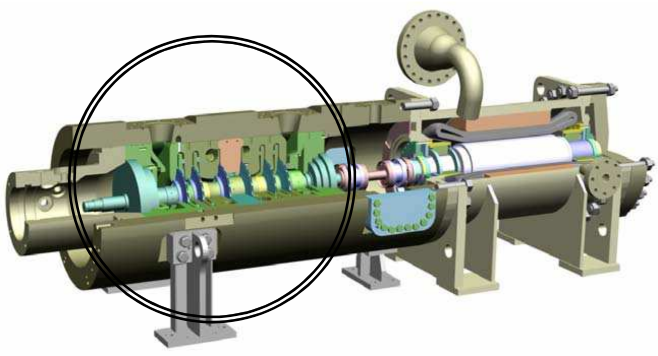

Figure 1: Integrated Compressor Line

Numerical computation was performed on a centrifugal compressor equipped with a magnetic thrust bearing and two identical AMBs called NDE (Non Drive End) and DE (Drive End) bearings (Figure 1). The action lines are positioned in the configuration load between axes. They are powered in differential driving mode. Two displacement sensors are integrated in the housing of each bearing and are non-colocalised with actuators, as shown in Figure 2. A flexible coupling is positioned between the compressor rotor and the electrical motor. Thus, the lateral analysis of the two rotors is uncoupled and the study is focused on the compressor.

The rotor is composed of a shaft of 2m length, six impellers and two stacks of laminated steel sheets shrunken on each bearing location. The total rotor mass is 370kg. The compressor delivers 5.5MW. The operating speed range used in this work includes three critical speeds (two forward rigid modes and the first forward flexible mode), that makes the machine supercritical.

As suggested by the third part of ISO 14839 [15], the frequency bandwidth considered in this study is 0-2kHz.

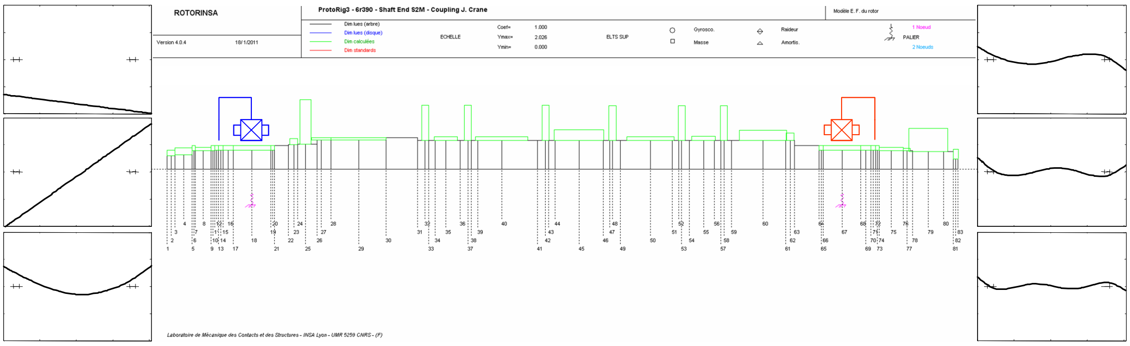

Figure 2: Finite Element model of the rotor (Red DE, Blue NDE)

The numerical model (Figure 2) was formulated with the finite element method. The radial AMBs are presented as suggested in the standard ISO 14839-1 [13]. For the sake of clarity, only the shapes of the first six modes are plotted in Figure 2. The rotor model is composed of the following elements: rigid discs (for the impellers and the two stacks of laminations) and the flexible shaft. Regarding the shaft, it is modeled with 82 Timoshenko beam elements with two nodes and 4 degrees of freedom, namely two displacements and two rotations per node. The calculations of the natural frequencies and the mode shapes were not performed with AMBs but only with direct stiffness (no cross stiffness and no damping). The stiffness value (2.10^7N.m^{-1}) was chosen in the range where the critical speed of the three first modes varies as a function of the bearings stiffness value [4]. The final effective stiffness is chosen in this range since this value enables significant action on the system strain energy and damping. This stiffness is then removed and the AMBs are considered as restoring forces. The ROTORINSA® software is used to compute the modal matrixes. The pseudo modal method is used to keep only the first 20 modes (to conform to the bandwidth frequency) [29]. Table 1 presents the natural frequencies of the rotor computed with the constant stiffness for the frequency bandwidth studied.

Modes	At rest (Hz)	125% of trip speed	
		Backward (Hz)	Forward (Hz)
Rigid 1	48	48	48
Rigid 2	74	72	77
1	152	144	160
2	324	306	341
3	554	528	579
4	835	799	871
5	1171	1114	1231
6	1564	1478	1658
7	1920	1840	2012
8	2382	2313	2462

Table 1: Rotor natural frequencies

The new controller is then implemented under Simulink® environments. The equations of motion are expressed in the state space form (Equation (1)) by using the reduced modal matrixes (mass M_Ψ, stiffness K_Ψ, damping C_Ψ, and gyroscopic G_Ψ). F_{ext} is the external forces that include disturbance forces $F_{disturbances}$ and unbalance forces F_b. Ω is the rotational speed and q the generalized modal displacements. Ψ is the eigenvector matrix that enables the transfer from physical to modal coordinates. The bearings and the gyroscopic matrix are considered as restoring forces and are taken into account on the right hand side of the equation of motion as presented in Figure 3. The physical displacement δ appears only as an input of the AMBs. The shaft is oriented along the y axe and the direction of rotation is defined from the axe z to x.

$$\begin{cases} \begin{pmatrix} \dot{q} \\ \ddot{q} \end{pmatrix} = \begin{bmatrix} [0] & I \\ -M_\Psi^{-1}K_\Psi & -M_\Psi^{-1}C_\Psi \end{bmatrix} \begin{pmatrix} q \\ \dot{q} \end{pmatrix} + \begin{bmatrix} [0] \\ M_\Psi^{-1} \end{bmatrix} \left(-\Omega G_\Psi \dot{q} + \Psi^t F_{ext} \right) \\ Y = \begin{pmatrix} q \\ \dot{q} \end{pmatrix} \end{cases} \tag{1}$$

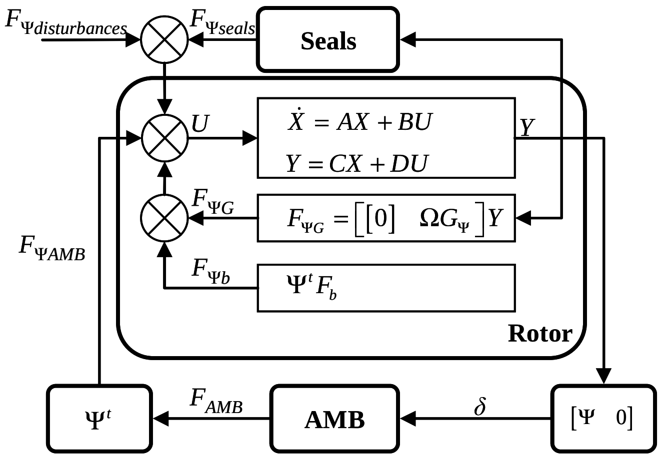

Figure 3: Diagram of the numerical model

As the AMBs are powered in differential driving mode, they can be considered as linear in the range of nominal displacements. Thus, they are taken into account with a current stiffness and a negative stiffness of $6.3 N.\mu m^{-1}$. These values were estimated with respect to the bearing geometry and load. The electronic part of each AMB is taken into account with a low pass filter with a cut off frequency below 2kHz.

3 Control Strategy

The fuzzy approach is well adapted for controlling complex systems and is less sensitive to parameter variations. Unfortunately, this kind of controller cannot prevent spillover effects. Two solutions can be considered: either the application of modal control or implementing a fuzzy controller based on a Simple PID (SPID). The SPID control enables managing the stability of high frequencies by introducing phase lag compensators (Figure 4). These filters guarantee stiffness and damping for high frequency modes [12]. In this study, the latter solution was chosen. The fuzzy process is used to modulate the gains of the SPID determined previously. The controller is then enhanced by using a synchronous filter. In the following, the fuzzy and the polar approaches are introduced, then the SPID is presented and finally, the synchronous filter is described.

3.1 Fuzzy approach

The main principles of the fuzzy approach are first the fuzzification of the inputs into linguistic quantities, e.g. high and low. Each input state is associated with a mathematic membership function. Second, the inference engine implements a set of linguistic rules based on the behavior of the system. These rules are conditional and must describe all the possible events that can occur. This evaluation requires solid knowledge of the system. A few rules are sufficient to describe the system's behavior, and can lead to the simplification of the design process and the optimization of the performances. Finally, the control is obtained after defuzzification. This consists in aggregating the conclusion of each rule. In this paper, the Takagi-Sugeno method is used for its effectiveness in real-time computations.

The fuzzy logic controllers developed consider each bearing as a single MIMO system (Figure 4). The inputs are the radial position and speed and the tangential speed stemming from the displacements measured in the two directions. The outputs of the fuzzy controller are then used to modulate the gain of the SPID. The outputs are the forces computed in the polar coordinates and then converted into command currents.

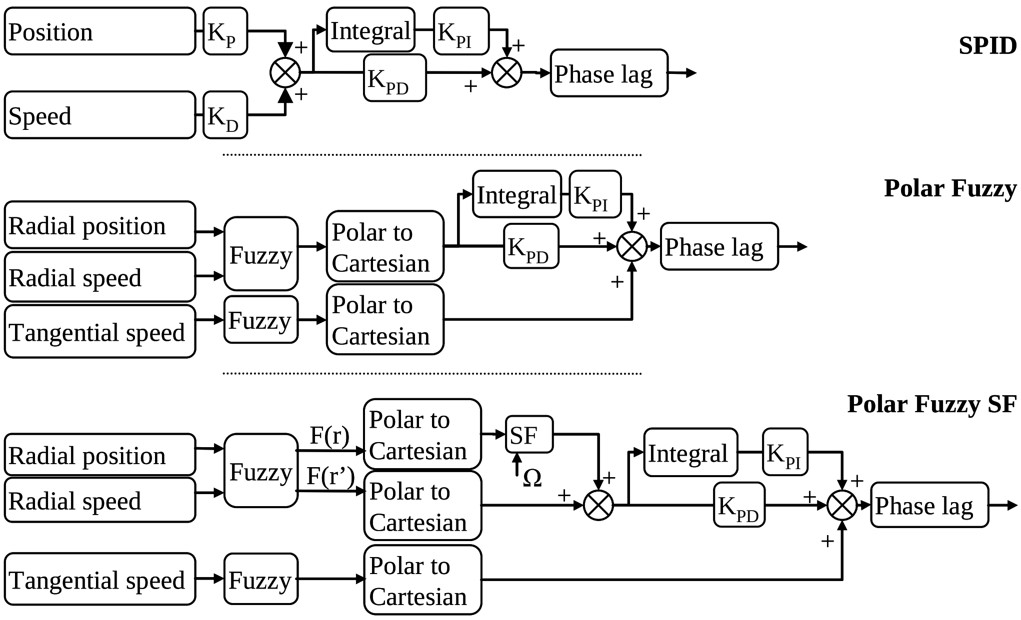

Figure 4: Schemes of Controllers

AMBs are normally symmetric bearings. With this property, the trajectory of the rotor inside the bearing is almost circular. The orbital properties of the rotor in steady state are defined by a constant radius, a constant tangential speed and a nil radial speed. These parameters describe closely the dynamic behaviour of the system, and lead to a better interpretation of the information by the controllers. Indeed, these physical quantities enable the controller to distinguish the difference between the disturbance produced by the unbalance and the transient disturbance that excites the rotor radially. Trapezoidal membership functions were utilized. The rules are:

- For permanent behaviour, when tangential speed is high tangential damping increases by 50%.
- For transient behaviour, when radial speed is positive, radial stiffness increases by 50%.

3.2 SPID

Usually, the characteristics of an augmented PID are determined as a function of the dynamic behavior and the number of modes included in the operating conditions. Also, the stiffness is chosen low and the damping is concentrated around system natural frequencies.

The first step of the fuzzy approach was the definition of the specifications of the SPID. The controller bandwidth is divided into low and high frequency domains with a transition around the second flexible mode. The idea is to standardized the design in the low frequency bandwidth, which corresponds to the operating speeds range that includes three vibration modes (cylindrical, conical and the first flexible). The consequence of this standardization is the reduction of the performance of the first flexible mode response. The high frequency stability and robustness depend on the mode shapes and frequencies. The parameters, that provide a suitable behavior are specific to each bearing. These properties are managed by lead-lag filters tuned for each bearing.

The controller must overcome the principal sources of instability, such as the negative stiffness and the seals dynamic coefficients. A high stiffness enables to counteract AMB and seals negative stiffness. SPID have naturally its minimum stiffness around the first rigid body mode. Consequently, if the designed stiffness is "sufficient" to control the first mode, this will be the case for the higher ones. Since the stiffness is high, the possibilities to introduce damping are limited. In addition, the seals effects introduce negative effective damping especially in low frequency, then the damping of this domain becomes a priority. Finally, the damping is made constant for the remaining operating speeds (Figure 5). Consequently, the controller became less efficient but more multifunctional. Moreover, the bearing characteristics are almost independent of rotor geometry in the operating frequency range. It can be supposed that the modification of the global gain can be sufficient to adapt the controller to different sizes of bearings and rotor for a given application.

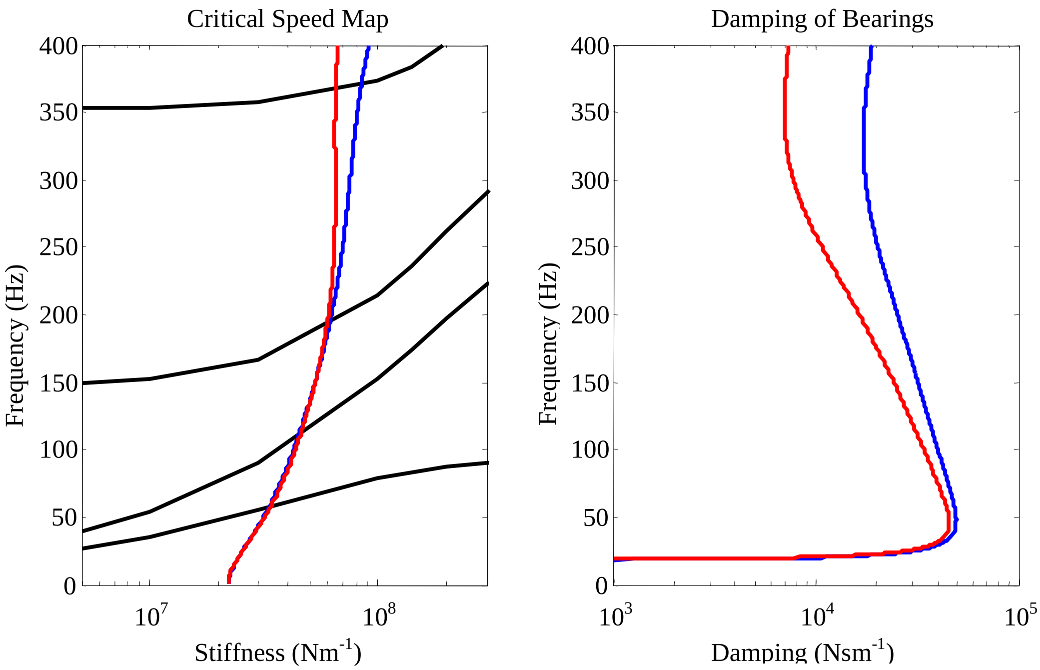

Figure 5: Standardized stiffness and damping (Red DE, Blue NDE)

3.3 Synchronous Filter

The fuzzy controller developed provides performances close to the requested specifications. Synchronous filters can be necessary to fulfill these specifications. A simple notch filter enables the reduction of vibration and control forces when passing through critical speeds. Among the different existing methods, the notch filter implemented in this study rejects the synchronous bearing stiffness only. This choice was motivated by the fact that generally, filters used to cross bending mode critical speeds reduce the synchronous stiffness and add synchronous damping. As the fuzzy controller generates damping, only the first effect was necessary. The filter has a gain of -6dB at the synchronous frequency. It was set on beyond the rigid modes and before the flexible one and it is maintained until the maximum speed. Finally, this filter has the advantage to be independent of rotor characteristics.

4 Numerical Results

The approach was successfully applied for the control of an academic test rig in a previous study, In this work only the numerical simulations of an industrial compressor are discussed.

First, the sensitivity of the system, which enables quantifying stability and robustness, was qualified. The fuzzy influence, as introduced in this work, essentially affects the low frequency domain. Consequently, the stability of the controller could be considered as the stability of the SPID. The sensitivity of the SPID was measured with a maximum value of 2.4. It is 0.6 below the zone A limit specified by the part 3 of ISO 14839 [15].

In the following, results obtained with the fuzzy controllers are compared with standard specifications and the results obtained with SPID alone for three study cases:

- Unbalance responses during run-up and run-down from 0 to 125% of trip speed.
- Behavior in the presence of subsynchronous excitation when operating at nominal speed.
- Response to an impact when operating at nominal speed.

4.1 Unbalance responses

Three different unbalance repartitions were studied which correspond to the standard specifications: one unbalance at middle, two opposite unbalances at rotor ends and finally two unbalances at rotor ends coupled with an opposite unbalance at middle. The envelope of the radial displacement was chosen as a representative value of the system response. The unbalance responses exhibited three critical speeds: the cylindrical mode, the conical mode, and the last, was identified as the first flexible critical speed. Figure 6 illustrates the system response and the control force for unbalance at middle span. This configuration was chosen for its relative simplicity of interpretation and demonstration. It could be noticed that the conical mode does not appear in this unbalance response. In addition, the Synchronous Filters (SF) introduce a slight softening effect (the bending critical speed is shifted toward a lower value) and reduce the bearing dynamic forces.

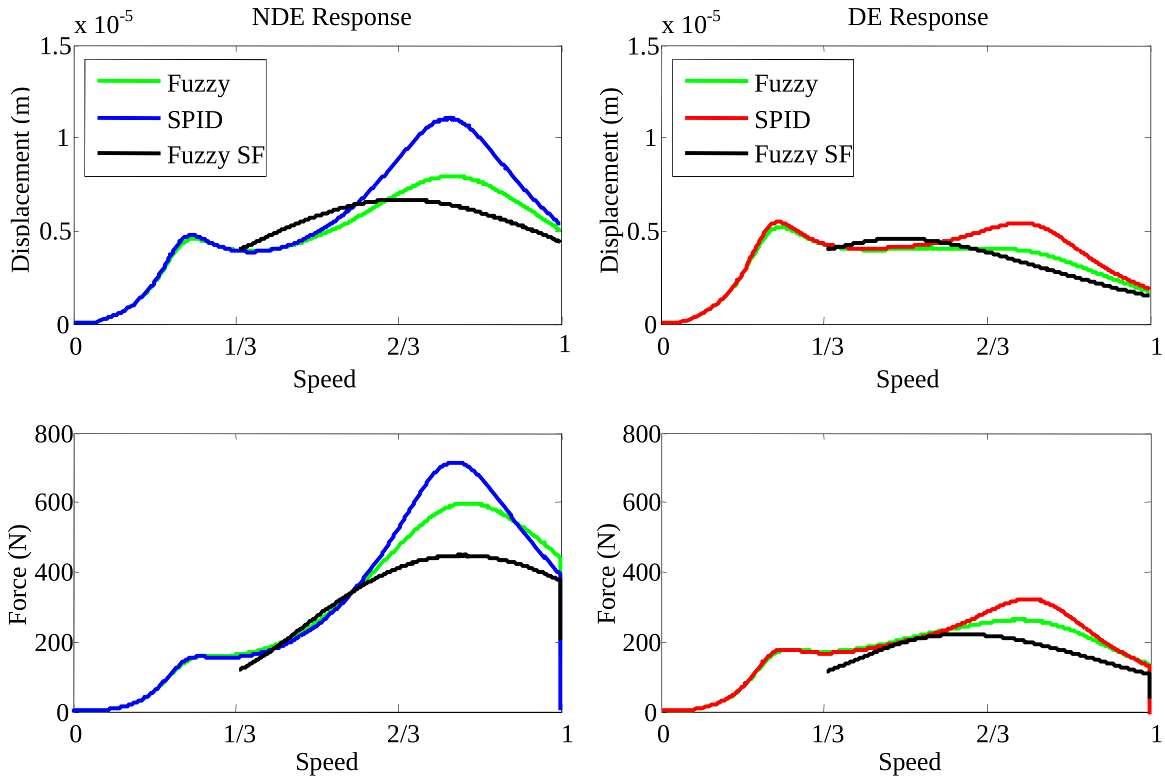

Figure 6: System response due to unbalance at middle span

As shown in Table 2, the unbalance responses demonstrate the better efficiency of the fuzzy controller. The maximum displacement decreased by 27% and by more than 40% with the activation of the synchronous filters. The amplification factors (AF) were also assessed. As expected, the use of fuzzy approaches enhanced the dynamic behaviour. The initial maximum AF (2.9) was reduce by 20% and almost halved for the bending critical speed when using synchronous filters. Finally, the maximum force necessary was reduced by 12% and by 28% with synchronous filters.

	Modes	Unbalance at middle			Opposite unbalances at ends			Unbalances at ends with opposite in middle		
		AF	Disp. pp µm	Force N	AF	Disp. pp µm	Force N	AF	Disp. pp µm	Force N
SPID	Cylindrical DE	2.5	11	179	-	-	-	-	-	-
	Cylindrical NDE	2.4	9.5	157	-	-	-	-	-	-
	Conical DE	-	-	-	1.8	**27.4**	-	-	-	-
	Conical NDE	-	-	-	1.6	16.4	446	-	-	-
	Bending DE	**2.9**	10.8	323	-	**26**	801	2.4	20.6	632
	Bending NDE	**2.8**	22.1	715	-	-	-	**2.6**	**36.2**	1190
Fuzzy	Cylindrical DE	2.2	10.4	177	-	-	-	-	-	-
	Cylindrical NDE	2.2	9.1	-	-	-	-	-	-	-
	Conical DE	-	-	-	-	-	-	-	-	-
	Conical NDE	-	-	-	1.5	12.2	371	-	-	-
	Bending DE	-	8.2	263	1.4	**25.5**	847	1.8	16	569
	Bending NDE	2	15.8	598	-	-	-	2.3	**26.5**	1049
Fuzzy SF	Cylindrical DE	2.2	10.4	177	-	-	-	-	-	-
	Cylindrical NDE	2.2	9.1	-	-	-	-	-	-	-
	Conical DE	-	-	-	-	-	-	-	-	-
	Conical NDE	-	-	-	1.3	11.3	271	-	-	-
	Bending DE	1	9.2	221	1.4	21.5	732	1.5	13.6	455
	Bending NDE	1.2	13.4	448	-	-	-	1.5	20.8	855

Table 2: Unbalance responses

The SPID enables to fulfil the requirement of the annexe (4F) of API 617 [16] in term of maximum displacement (90µmpp), but not the specifications for compressors on classic bearings (25µmpp). In addition, the AF is superior to 2.5 for the third critical speed. Consequently, the machine could not operate at this speed and separation margins should be considered [16]. With the fuzzy controller, the AF is always below the limit, and the maximum displacement is closed to the specifications for classic bearings. Finally, the association of the fuzzy controller and the synchronous filter enables to satisfy all the standard requirements.

4.2 Subsynchronous excitations

This configuration corresponded to the aerodynamic disturbances applied on the coupling between compressor and motor. This study was carried out with dynamic coefficients of seals. There is no standard specification. The main source is the gas inside the coupling cavity. These disturbances are visible around the first natural frequency because AMBs have a low stiffness at this frequency and a negative stiffness. In addition, dynamic coefficients of seals, shown in Table 3, create a negative effective damping and a negative stiffness [30].

Position (mm)	Kzz=Kxx (N/µm)	Kzx=-Kxz (N/µm)	Czz=Cxx (Ns/m)	Czx=-Cxz (Ns/m)
635.1	0.89	0.84	550	-265
744.1	0.87	0.95	629	-281
943.1	1.05	1.06	711	-295
1049.1	-8.23	-0.24	2717	5252
1155.1	1.28	1.38	963	-332
1332.1	1.19	1.27	875	-319
1440.1	1.19	1.27	875	-319
1520.3	-6.08	0.64	2037	3955

Table 3: Seals dynamic coefficients at 100% speed

The SPID was designed to provide stability and robustness in the low frequency range, and particularly to improve the dynamic behavior of the rotor excited by turbulences. Figure 7 presents the response of the rotor with the SPID alone and with the fuzzy controllers. Calculations were performed with the third unbalance distribution at operating speed (2/3 of the maximum speed). Due to confidentiality constraints, the synchronous response was dissociated (the vertical scale was conserved). The excitation was a white noise limited to 0-30Hz, with an amplitude of 15N in frequency domain. This value can seem low but, as it was applied on a given frequency range, in temporal domain this signal exhibit peaks close to 750N.

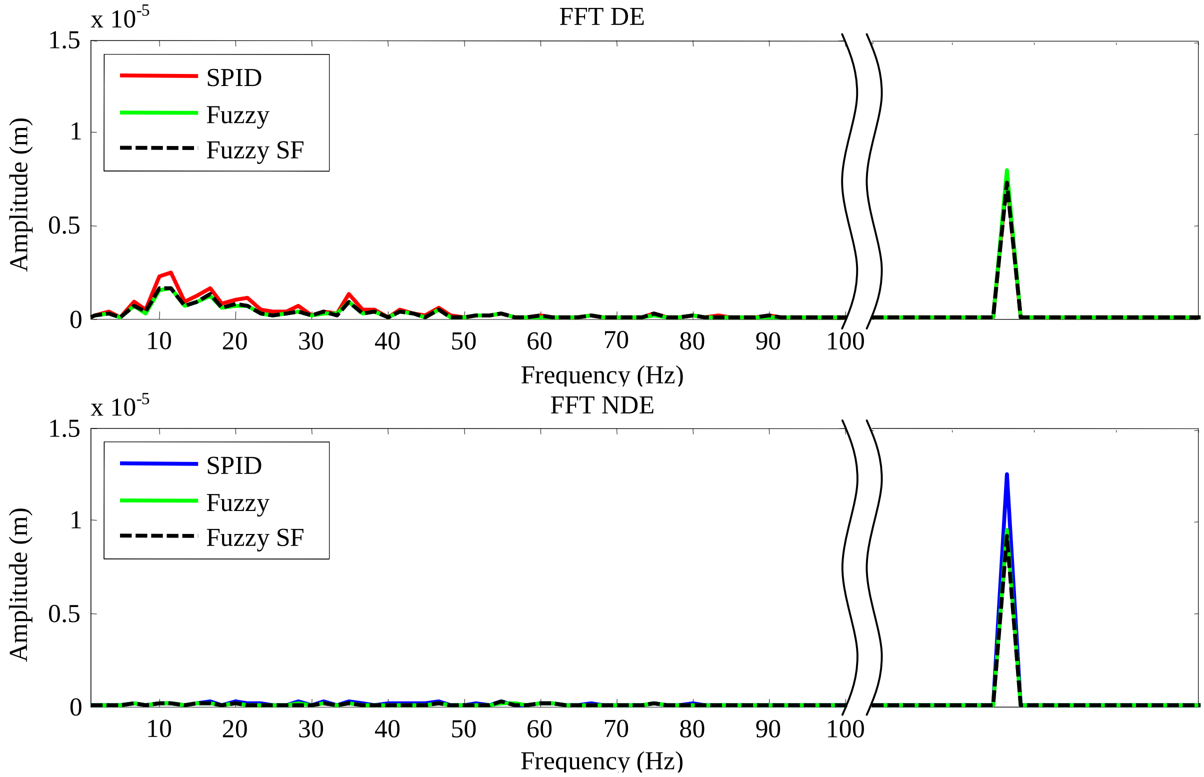

Figure 7: Unbalance at middle span

As expected, the results are very similar with or without synchronous filter. The fuzzy controller permits a reduction of 25% of subsynchronous vibration levels (low frequency domain). On the other hand, and as the synchronous responses on DE side are almost identical, it is possible to compare also the displacement obtained in time domain. Whereas, the SPID exhibits a maximum displacement of 58µm, the fuzzy controller limits the displacement to 40µm, leading to an attenuation of more than 30% of the vibration level. This difference is important since the maximum of displacement when using the fuzzy controller falls inside class A as defined by ISO 14839 [14] while it was between the class B and C with the SPID. It is worth mentioning, that class C is considered unsatisfactory for long term continual operation.

4.3 Impact response at operating speed

Figure 8 presents the rotor trajectory at the NDE bearing for the response to an impact at a stabilized speed (operating speed) with the third unbalance distribution. The impact is defined by a triangular signal of amplitude 500N during 0.01 second applied at the NDE bearing. The red arrow in Figure 8 shows the direction of the impact and the black arrow the direction of rotation. It can be seen that the displacements, stemming from the fuzzy controller, are lower than those obtain for the SPID alone. The non-dimensional vectors on Figure 8 show the additional radial forces applied by fuzzy controller relatively to SPID. The time is represented by the curve colour, the oldest position in dark blue and the newest in red.

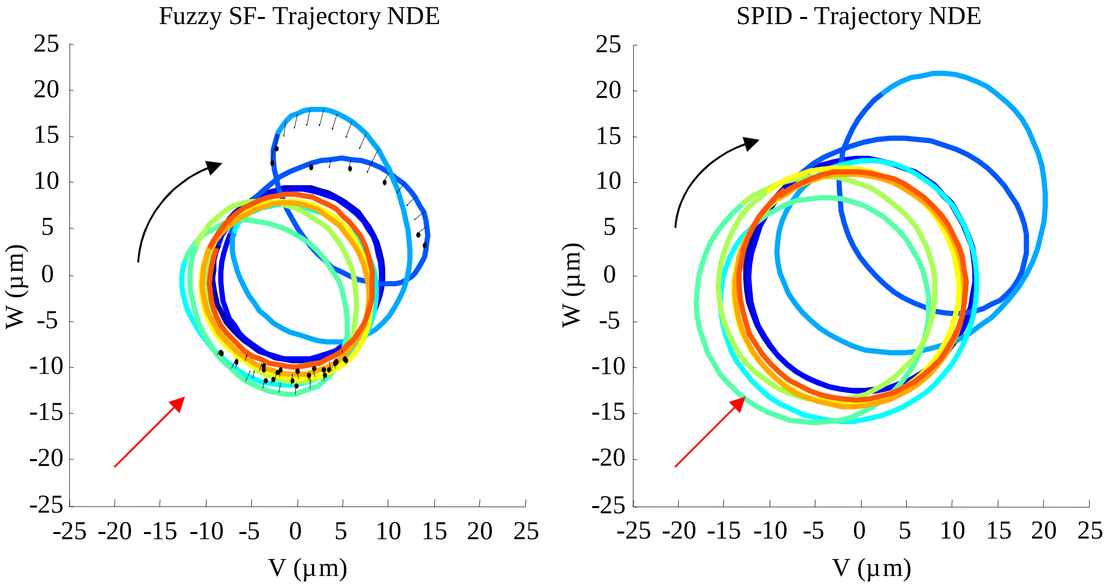

Figure 8: Rotor trajectories

The fuzzy controller exhibits a more interesting behavior with a maximum displacement of 19.1µm and 18.1µm when synchronous filters are activated, when SPID exhibits 24.7µm. Obviously, a part of this attenuation stems from synchronous responses which have an amplitude of 9.8µm without SF, 9.2µm with SF and 12.7µm when using SPID.

5 Conclusions

A new control approach is developed and assessed numerically for the control of the dynamic behavior of an industrial compressor. The controller is fuzzy based using inputs expressed in polar coordinates.

The design procedure was presented. First, a simple PID was tuned (SPID) and then the fuzzy controllers modulated the SPID gains. The SPID is designed in order to have low frequency characteristics independent of rotor geometry. The fuzzy blocks are easy to tune as the fuzzy rules benefited from their formulation as a function of the dynamic behavior of the rotating machinery. This behavior is almost the same for a given range of products, the geometrical variations could affect the amplitudes of the response but not the phenomena considered. Thus, the rotordynamic analysis can be done easily during the machine design process. As a centrifugal compressor is produced in a very limited number and as the bearing characteristics can impact the rotor design, the approach developed becomes an important advantage.

Polar coordinates permit to analyze the dynamic behavior of a rotating machine making steady state and transient behaviors easily differentiated. Besides, the polar physical quantities are interesting to apply targeted actions. The fuzzy logic, which is a decision support system, is a powerful tool in order to use this information for control issues.

The dynamic behavior is compared for several configurations of excitations. Results obtained when using the polar fuzzy controller were compared to standard specifications. Results were also compared to those obtained when using a SPID that is similar to an augmented PID. The controller developed exhibits stable and robust behavior. All standard specifications were respected in associating the fuzzy controller to a SF. With respect to the SPID behavior, better performances were obtained for the same level of robustness. Regarding the subsynchronous excitation, the fuzzy controller enables a reduction of 25% of vibrations level in low frequency.

It is worth mentioning, that the results obtained with the fuzzy controller without synchronous filter are very close of the standards requirements. A study regrouping several machines could certainly permit to determine the adapted fuzzy controller. In this way, two advantages would be obtained: the standardization of the controller and the suppression of the speed sensor. Thus, the hardware of the bearing would be simpler and more robust.

On the other hand, the polar fuzzy controller adds several gains that have to be mastered in order to manage the performance, the stability and the robustness of the system. The fuzzy controllers were designed to control relatively small disturbances which concern nominal behavior (class A ISO 14839) while the fuzzy rules were not adapted for large displacements. New rules are therefore necessary to improve performances in this case.

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