Soumis le : *12/12/2013*
Forme révisée acceptée le : *18 12.2015*
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Nature & Technology

Monitoring system based on real data acquisition for vibrations control in gas turbine system

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Abstract

This work deals with the problem of vibration control in rotating machinery, through the implementation of a data acquisition system, for monitoring vibrations in examined gas turbine. The vibrations are measured and processed for supervised the mechanical condition of the examined gas turbine. The tested results obtained are very satisfactory using the monitoring system to control vibrations in the studied gas turbine.

Keywords— Gas turbine; data acquisition system; vibrations control; vibration alarms

1. Introduction

Gas turbines are rotating machinery driving continuous flow, fitted with an axial compressor and combustion chambers and is able to produce itself a fluid under pressure and at high temperatures, which in undergoing its expansion phase deferent in turbine stages, mechanical energy supplied to the outside. It is a true stand-alone unit that is sufficient to itself [1, 3, 9, 14 and 18]. Understanding of the vibration behavior in turbine is an industrial issue whose importance has steadily increased in recent years [5, 6, 8, 10 and 19]. The dynamic deformation uncontrolled structure (eg shields and blades of a turbine), can cause premature aging of components, or even unacceptable noise and vibration. Current studies on the vibration behavior of gas turbine and treatments as well as analyzes of the measured signals can subsequently assembled vibration effects observed to material causes that generate and provide a very powerful tool for maintenance purposes, especially in the gas and oil industry or the competition is expressed by the quality and costs [13, 16].

The objective of this paper is the implementation of a data acquisition system for monitoring vibrations in the gas turbine TITAN 130. Certain operating conditions for gas turbines cause abnormalities in their operations. The appearance of strong vibration causes damage in organs of the turbine. So, taking into account malfunctions to determine the appropriate requirements for maintenance or repair. All turbines are all prone to vibration phenomena under certain conditions. The results obtained show how, in a maintenance policy, the vibrations are measured, processed and used for diagnosis, to assess the mechanical condition of the turbine.

2. Gas compression station SC3

The gas compressor station SC3 / Sonatrach , Djelfa , Algeria , which proposes to exhaust our case studies , has 03 Turbochargers TITAN 130 power literally every 14.5 MW , to increase the gas pressure pipeline GG1 of 42 bar to 70 bar. Gas compressor station SC3 - MOUDJBARA has 03 Turbochargers mounted in parallel to increase the pressure of the gas at 72 bar max in gas pipeline GG1. A vibration monitoring system is installed in order to control the vibration in the turbine. Our study was devoted to the vibrations in the turbine TC2. The procedure of data acquisition parameters and variables of the turbine is made by a computer-assisted, installed at the station operation and engineering. The supervision system of the SC3 facility has two types of labels (Analog and all or nothing). This system is shown in the following figure 1.

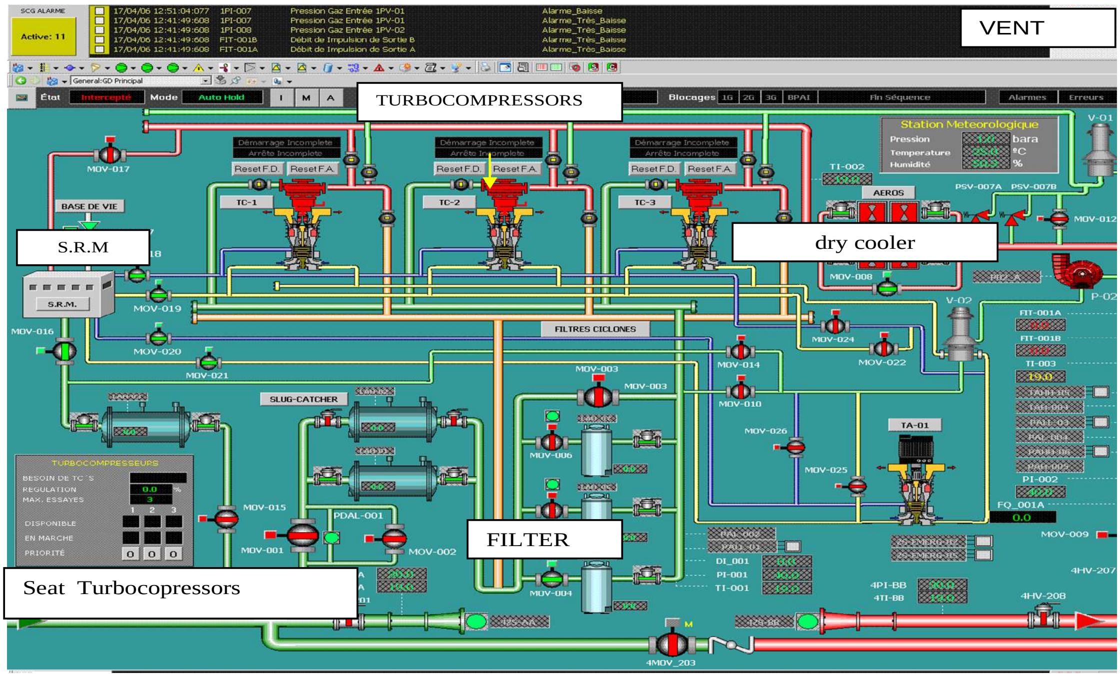

Fig. 1. Synoptic monitoring SC3 / station Sonatrach, Djelfa, Algeria

This synoptic monitoring shown in figure 1 offers the user a window into the operation of the equipment. A view of navigation appears on the left side of the display. This view allows you to quickly navigate to all project views, all reports or additional application.

2.1. Vibrations data analysis

Data analysis of vibrations in the turbine TITAN 130 is used to collect data and to make different treatments and scaling of this data. For our application we considered the radial positions of the vibration sensors in the turbine TITAN 130, shown in the figure 2.

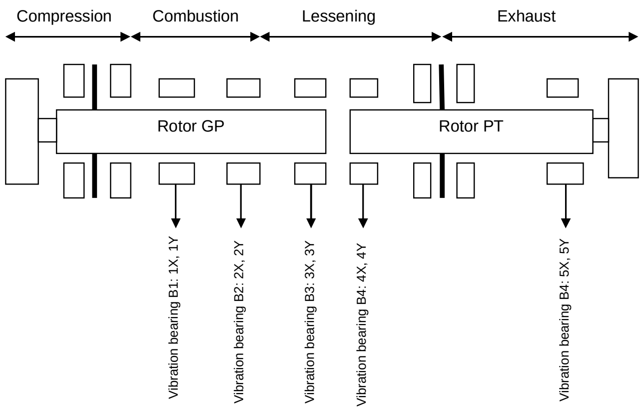

Fig. 2. Sensors positions for radial vibration in gas turbine TITAN 130

Like any movement, vibration recorded during the routes can be quantified by three fundamental quantities [6, 12, 17, 22 and 23]. The displacement and the speed of this movement and acceleration experienced to make this move.

The model considered the turbine has two inputs: T_1 ambient temperature and the mass flow rate $\dot{m}_c$ of the fuel, given by the following equations:

$$\dot{m}_c = \sqrt{\Delta P}.\frac{1}{2}\rho\mu^2 = \frac{\sqrt{\Delta P}}{k} \Rightarrow \dot{m}_c = \frac{1}{k}\sqrt{\Delta P} \qquad (1)$$

The air mass flow $\dot{m}_a$ is given by:

$$\begin{cases} \dot{m}_a = k'(P_2 - P_1) \\ \dot{m}_a = k'(PCD - P_{atm}) \end{cases} \qquad (2)$$

We must control the combustion of the gas turbine by controlling the ratio of the yield f given by:

$$f = \frac{(T_3 - T_2)}{\eta_b CV + T_3} \qquad (3)$$

Sensors provided from the outputs of the engine include: high speed HP pressure (NGP) tree, the low pressure shaft speed (NPT) and the blade temperature. These measurements can be used to provide different pairs of input-output for the vibration control in the closed loop. For the movement of a harmonic and written by the vibration equation:

$$x(t) = A\sin(wt + \varphi) \qquad (4)$$

With w is the pulse, φ is the phase and A is the amplitude and the unit used is micrometers (µm).

For vibration speeds $v(t)$ and vibration acceleration $a(t)$ is obtained by derivation give by:

$$v(t) = \frac{dx(t)}{dt} Aw\cos(wt + \varphi)$$
$$\Rightarrow a(t) = \frac{dv(t)}{dt} \qquad (5)$$

Commonly used units are millimeters per second (mm / s).

It follows from equations (4) and (5) the following interlink among the modules:

$$\begin{aligned} |X| &= \frac{|v|}{w} = \frac{|A|}{w^2} \\ |v| &= |X|.w = \frac{|A|}{w} \\ |A| &= |v|.w = |X|.w^2 \end{aligned} \tag{6}$$

Equation (6) highlights the importance of choice of the physical quantity to be measured by monitoring the gas turbine. Compared with a measurement speed mode, the measurement in displacement mode will effectively mitigate all medium and high frequency components and amplify the low frequency components. However, the measure acceleration mode will have the effect of reducing the low frequency components and amplify the high frequency components.

In this work; we study a radial vibration in gas turbine, by the twist in the simple case of a circular cross rotor section. The dominant torsional displacement is the rotation of the cross sections given by the angular displacement α , the simplified displacement is used as follows [24, 25]:

$$\begin{cases} u_1(x_1, x_2, x_3, t) \approx 0 \\ u_2(x_1, x_2, x_3, t) \approx -x_3\alpha(x_1, t) \\ u_3(x_1, x_2, x_3, t) \approx x_2\alpha(x_1, t) \end{cases} \tag{7}$$

For the deformations calculations flowing the Hamilton functional, is give as following:

$$\begin{cases} \varepsilon_{11} = \varepsilon_{22} = \varepsilon_{33} = \varepsilon_{23} \\ \varepsilon_{12} = -\frac{1}{2} x_3 \frac{\partial \alpha}{\partial x_1} \\ \varepsilon_{13} = -\frac{1}{2} x_2 \frac{\partial \alpha}{\partial x_1} \end{cases} \tag{8}$$

The Hamilton functional construction is given by the following equation:

$$\begin{aligned} & H(\alpha) = \int_{t_0}^{t_1}\int_0^l \left[\frac{1}{2}\rho I_1 \left(\frac{\partial \alpha}{\partial t}\right)^2 - \frac{1}{2} G I_1 \left(\frac{\partial \alpha}{\partial x_2}\right)^2 + M_1 \alpha \right] dx_1 dt \\ & \textit{with} \\ & I_1 = \int_S (x_2^2 + x_3^2) dx_2 dx_3 \end{aligned} \tag{9}$$

The rotor motion is given by:

$$\begin{aligned} & \rho I_1 \frac{\partial^2 \alpha}{\partial t^2} - \frac{\partial}{\partial x}\left(G I_1 \frac{\partial \alpha}{\partial x} \right) = M_1 \\ & \forall x_1 \in \left]0, l\right[, \forall t \in \Re \end{aligned} \tag{10}$$

Where $G = \dfrac{E}{2(1+\nu)}$ is the shear modulus of the rotor in the limitations conditions $x_1 = 0$ and $x_1 = l$.

$$\begin{cases} \alpha(0,t) = 0 \\ G I_1 \frac{\partial \alpha}{\partial x_1}(0,t) = 0 \end{cases} \quad \textit{and} \quad \begin{cases} \alpha(l,t) = 0 \\ G I_1 \frac{\partial \alpha}{\partial x_1}(l,t) = 0 \end{cases} \tag{11}$$

The condition $\alpha = 0$ is a condition of installation and the condition $G I_1 \frac{\partial \alpha}{\partial x_1} = 0$ when no torsion is a free end provided. The stress on the rotor surface is determined by the using of the following equation:

$$\begin{pmatrix} \sigma_{11} \\ \sigma_{22} \\ \sigma_{33} \\ \sigma_{23} \\ \sigma_{13} \\ \sigma_{12} \end{pmatrix} = \begin{pmatrix} E & \nu E & \nu E & 0 & 0 & 0 \\ \nu E & E & \nu E & 0 & 0 & 0 \\ \nu E & \nu E & E & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{E}{2(1+\nu)} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{E}{2(1+\nu)} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{E}{2(1+\nu)} \end{pmatrix} \begin{pmatrix} 0 \\ 0 \\ 0 \\ -x_3\alpha_{x_1} \\ -x_2\alpha_{x_1} \\ 0 \end{pmatrix}$$

Osama Ashour et al. in [21] used a non-linear model for configuration and control for on-line monitoring of gas turbines. Also, the rapid shutdown of the turbine may cause inconvenience at rotating turbine components [2, 4, 7, 11, 15 and 20]. In this work, we propose a organization chart for the control vibration in gas turbine TITAN 130, shown in Figure 4, with changes in the control program. To preserve the bodies of the turbine and increase the response time of the operator to find the source of vibrations.

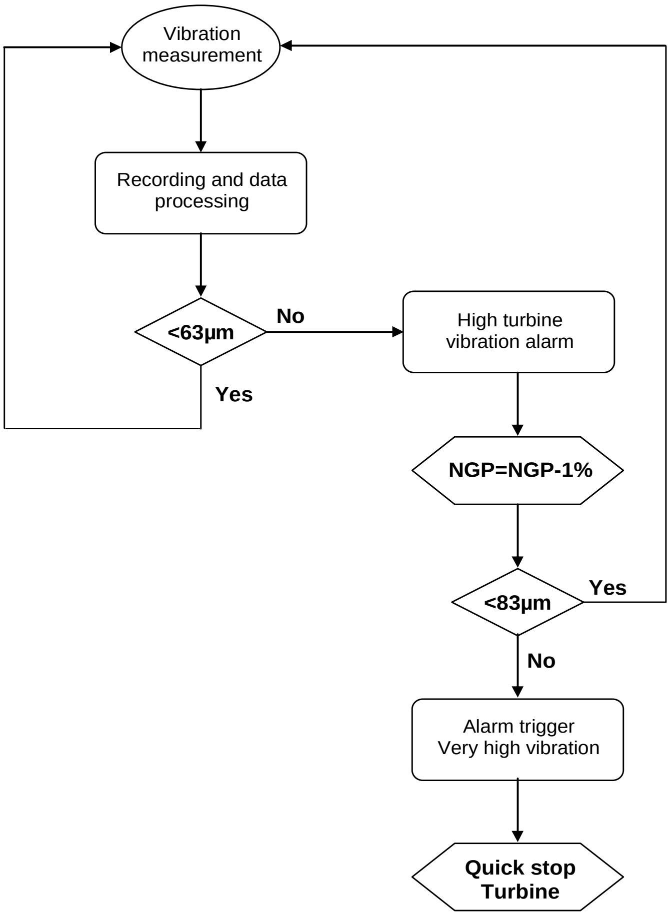

Fig. 3. Proposed organization chart for vibration control in gas turbine TITAN 130

3. Applications results

The presented results in this section are obtained with the using of a real data at the examined gas turbine TITAN 130.

Fig. 4: Curve of the discharge pressure

Fig. 5: Vibration radial of the bearing B1

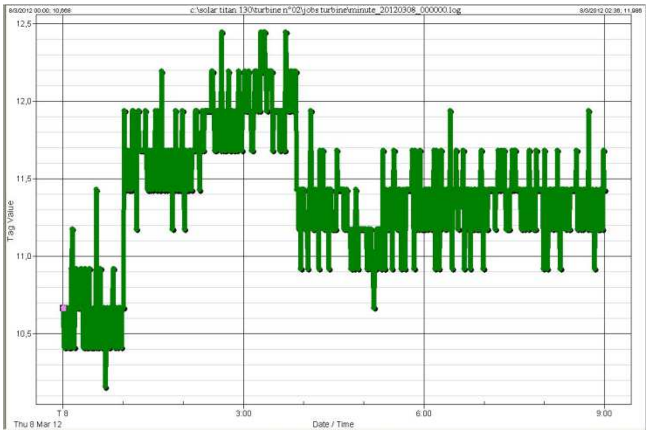

Fig. 6: Vibration radial of the bearing B2

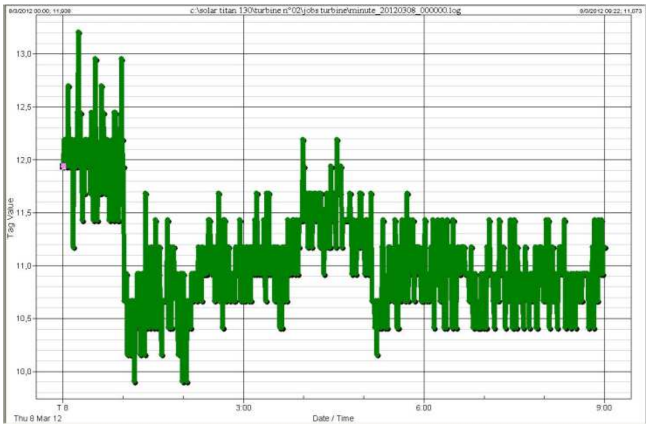

Fig. 7: Vibration radial of the bearing B3

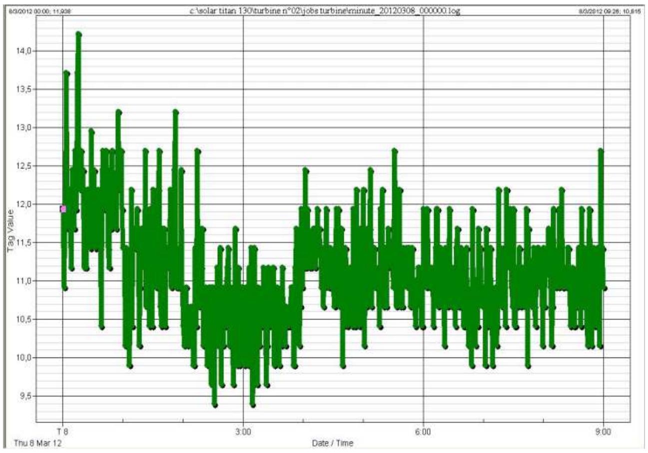

Fig. 8: Vibration radial of the bearing B4

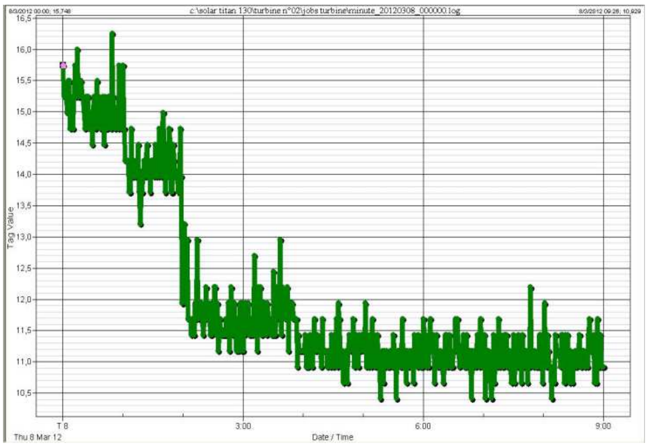

Fig. 9: Vibration radial of the bearing B5

The shown results, obtained by the proposed vibration monitoring system, this results shown that all the vibrations of the turbine TITAN 130 has the following features:

- Nature of random vibration pulse type,
- Amplitude variable in the lower allowable value time,
- Variable frequency period of the order of minutes.

It follows that the turbine is operating in its normal diet and these vibrations are produced by events due to the normal force generated by the rotation of the turbine. From the results of the vibrations curves, we have found that the amplitudes of the vibrations are proportional to the speed of the turbine, the gradual decrease of the speed decreases the amplitudes of the vibrations.

4. Conclusion

Through measures and analyzes signals of the vibration behavior of the examined gas turbines, we can confirm that the proposed control algorithm is basic tools for supervision of the mechanical state of the turbine and providing supervision system. The proposed control algorithm gives best results on vibration control of industrial gas turbine. The validation of the developed algorithm was tested on real time at the gas compressor station SC3 / Sonatrach, Djelfa, Algeria, by direct implementation in the examined gas turbine.

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