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Vibration analysis and measurement based on defect signal evaluation: Gas turbine investigation

Boulanouar Saadat [1*], Ahmed Hafaifa[2] and Mouloud Guemana[3]

[1] Applied Automation and Industrial Diagnostics Laboratory, Faculty of Science and Technology, University of Djelfa 17000 DZ, Algeria.

[2] Faculty of Science and Technology, University of Médéa, 26000, Algeria

Abstract. The vibration behavior of rotating machines is a very important industrial challenge, can cause aging and tiredness affect its components. In this work, we propose to study the gas turbine vibration behavior based on defect signal evaluation, an analysis made by the vibration signals measured. The tests results realized on the examined gas turbine shows that the vibration effects observed provide a very powerful tool for protected the mechanical condition of the examined gas turbine

Keywords: *Vibration behavior, rotating machines vibration analysis, gear defect evaluation, gas turbine, measures validation*

1. Introduction

Technological evolution of industrial equipment in recent years, allows to the vibration monitoring become the primary tool of conditional preventative maintenance [1, 2, 6, 17, 26 and 32]. There are many techniques that can be used in a maintenance program, such this techniques the vibration analysis [4, 8, 10, 15 and 21]. This vibration analysis technique allows detecting virtually all defects that may appear in rotating machines [16, 18, 23, 29 and 30]: Unbalance, alignment defects, worn or damaged bearing ... etc. This result in a variation of internal forces experienced by the machine, and thus to a change in its vibration behavior. The vibration analysis has become the main technique used in the management of the condition-based maintenance. The largest share of industrial equipment is mechanical in nature, this technique has many potential applications and provides the best benefits in a program intended to cover an entire factory [7, 14, 22 and 31]. It uses the noise or vibration created by mechanical equipment (and in some cases by industrial systems) to determine the actual operating mode [3, 5].

In this context, we propose in this work to examine and illustrate the ability of the application of the methods of vibration analysis based on gear defect evaluation to monitor the operating condition of a gas turbine. This is part of predictive maintenance policy tool in industrial production, taking the example of a gas turbine system type GE MS 3002. Indeed, the vibration analysis of rotating machinery is now widely used by manufacturers to diagnose faults on their machines before it will undergo a fortuitous is conditional maintenance. We show in this work, through various tests, the results clearly show the detection of malfunctions of rotating machinery in the examined gas turbine, and allowing for better performance during its operation for the maintenance strategy.

[*] Corresponding author.
E-mail: N_sadat@univ-djelfa.dz (Saadat B.).
AAIDL, Faculty of Science and Technology, University of Djelfa, 17000 , Algeria.

2. Vibration analysis

The exploitation of rotating machinery always produced effort (turning effort, turbulence, shocks, volatility) that will often cause subsequent failures [9, 12, 19 and 28]. To establish a diagnosis, it must rely on the fact that a machine emits outward many signals that are symptomatic of its operation, such as heat, power consumption, noise, vibration ... etc. Vibration analysis plays an important role not only in the diagnosis of the state machines for servicing, is the starting point for any diagnosis vibration [11, 20]. One can make the succeeding exchange suspicious parts: drive motor, rods, bearings and balancing all the rotating parts [13, 24 and 25]. Based on vibration analysis, we can make successive trading decisions suspicious parts: drive motor, coupling, bearings, and balance all rotating parts. It is efficient and economical separation with appropriate means according to different vibrations components, undertake targeted maintenance. The effective vibration amplitude is defined mathematically by the equation (1) and is expressed as a function of the peak amplitude.

$$A_{eff} = \sqrt{\frac{1}{T}\int_0^T a^2(t)\,dt} \tag{1}$$

Where , $a(t)$ is the instantaneous amplitude of the vibration signal, T is the length of the vibration signal and A_C the peak amplitude.

The displacement $x(t)$ of a harmonic vibration and written by the flowing equation:

$$x(t) = A\sin(\omega t + \varphi) \qquad A_{eff} = A_C\,\frac{\sqrt{2}}{2} \tag{2}$$

Where ω is the pulsation, φ is the phase and A is the amplitude.

The speed and acceleration of the vibration is obtained by the displacement differentiating:

$$X(t) = A\sin(2\pi ft) \Rightarrow V(t) = \frac{dX}{dt} \Rightarrow V(t) = 2\pi fA\sin(2\pi ft + \pi/2)$$

$$\gamma(t) = \frac{d^2X}{dt^2} \Rightarrow \gamma(t) = (2\pi f)^2 A\sin(2\pi ft \pm \pi)$$

$$w = 2\pi f = \frac{2\pi}{T} \tag{3}$$

Where T is the period express (s) and f is the frequency in hertz (Hz)

The vibration signal delivered by a sensor may be represented in different ways [6, 27 and 32]; the first that comes to mind is the representation function of time. This representation is also used to monitor the vibration behavior of a machine, based on its operating parameters. For it to be interpreted, the signal must be decomposed into different elementary sinusoidal components. If this decomposition is possible, theoretically, its representation in the time domain becomes quickly usable, as shown in Figure 1.

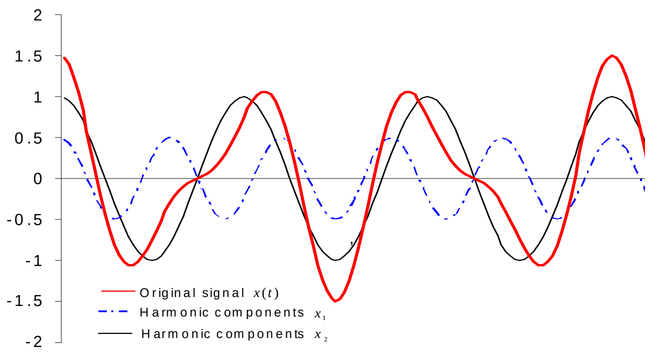

Figure 1: Vibration signal decomposition into two harmonic components

To represent the vibration signal in a usable form, we tried to represent in a diagram of amplitude-frequency spectrum [26, 30]. Like any movement, vibration is characterized by relations between the three variables, involving frequency.

$$x = \frac{v}{2\pi f} = \frac{\gamma}{(2\pi f)^2} \tag{4}$$

$$x = \frac{\gamma}{2\pi f} = 2\pi f . x \tag{5}$$

$$\gamma = 2\pi f v = (2\pi f)^2 . x \tag{6}$$

It is apparent from equations (4), (5) and (6) the following relations among the vibration modules:

$$|X| = \frac{|V|}{\omega} = \frac{|A|}{\omega^2} \tag{7}$$

$$|V| = |X| \cdot \omega = \frac{|A|}{\omega} \tag{8}$$

$$|A| = |V| \cdot \omega = |X| \cdot \omega^{2} \tag{9}$$

Equations (7), (8) and (9) show the importance of the choice of the physical quantity to be measured by monitoring a rotating machine, as is the case of a gas turbine have generally consisted a rotor, and bonds of a structure, as shown in Figure 2.

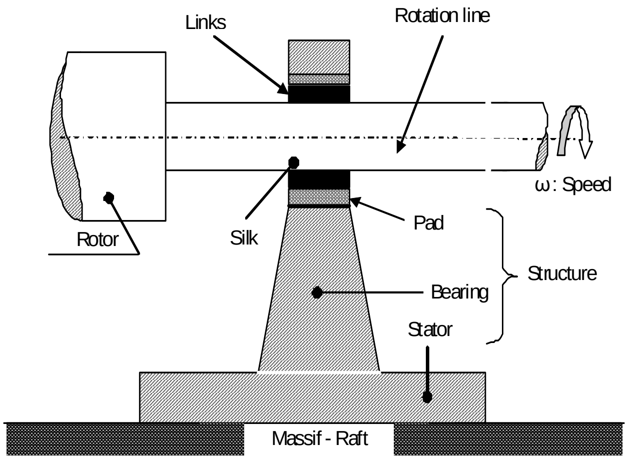

Figure 2: Elements of a rotating machine

Any abnormality affecting a rotating machine (unbalance, imbalance phenomenon oil swirls, tree deformation, release bearing, bearing fault, electromagnetic anomaly in the stator or rotor of a motor, defective mesh, ...), translates to vibrations signals. Whose frequencies correspond to the occurrence of forces that induce and their harmonic frequencies (multiples of frequencies of occurrence). The overall measure used to quantify the default spectral analysis allows qualifying. The forces applied to the system are the inertia forces; they operate at the center of gravity.

$$\vec{F} = m \cdot \frac{d^2\overrightarrow{OG}}{dt^2} \quad (10)$$

With , it is derived by: $\overrightarrow{OG} = \begin{cases} X = x + e\cos(\omega t) \\ Y = y + e\cos(\omega t) \end{cases}$

$$\frac{d^2\overrightarrow{OG}}{dt^2} = \begin{cases} X'' = x'' - e\omega^2\cos(\omega t) \\ Y'' = y'' - e\omega^2\sin(\omega t) \end{cases} \quad (11)$$

$$\overrightarrow{F_i} = \begin{cases} mx'' - me\omega^2\cos(\omega t) \\ my'' - me\omega^2\sin(\omega t) \end{cases} \quad (12)$$

Stiffness forces it depends on Young's modulus, length of the shaft, the inertia of the shaft for conservative case, given by:

$$\vec{F} = -K\overrightarrow{OA} \Rightarrow \overrightarrow{OA} = \begin{cases} x \\ y \end{cases} \Rightarrow \vec{F}_k = \begin{cases} -kx \\ -ky \end{cases} \quad (13)$$

The damping forces are generally weak, the external depreciation is taken into consideration, it applies to the speed:

$$\overrightarrow{F_c} = -c.\frac{d\overrightarrow{OA}}{dt} \qquad (14)$$

Where . $\frac{d\,\overrightarrow{OA}}{dt} = \begin{cases} x' \\ y' \end{cases} \Rightarrow \overrightarrow{F_C} = \begin{cases} -cx' \\ -cy' \end{cases}$

The application of the principle of dynamics gives the equation of motion gives us:

$$\begin{cases} m\ddot{x} + c\dot{x} + kx = me\omega^2 \cos(\omega t) \\ m\ddot{y} + c\dot{y} + ky = me\omega^2 \sin(\omega t) \end{cases} \qquad (15)$$

The spectrum can be obtained by applying the Fourier transform, which has the property of decomposing a signal into its complex basic components defined by their amplitude and frequency. This allows passing from one time representation to a spectral representation [22, 26]. The signal obtained by a vibration sensor in the time domain, by definition:

$$F(f) = \int_{-\infty}^{+\infty} f(t)\exp(-j2\pi.ft)dt \qquad (16)$$

Where $F(f)$ is the Fourier transform (FT)

Typically, the function $f(t)$ is representative of a non-defined mathematical function by a simple signal, it should be sampled into discrete dots, and thereafter its spectrum can be calculated by substituting the Fourier integral of the algorithm by the Fast Fourier Transform (FFT), we use the following matches:

$$F(k\Delta f) = \frac{1}{N}\sum_{n=0}^{n=N-1} X(nt_e).\exp(-j2.\pi\frac{k.n}{N}) \qquad (17)$$

$$t \rightarrow nt_e \; f \rightarrow m\Delta f \quad dt \rightarrow t_e \qquad \int_{-\infty}^{+\infty} \rightarrow \sum_{-\infty}^{+\infty}$$

$$F(n-k) = \frac{1}{N}\sum_{n=0}^{n=N-1} X(n.t_e).\exp(-j.2\pi.(N-k).\frac{n}{N}$$
$$= F(-k)\exp(-j.2\pi.n) = F(-k)$$

With $\exp(-j.2\pi.n) = 1$ and $|F(-k)| = |F(k)|$.

So we have:

$$|F(N-k)| = |F(-k)| \qquad (18)$$

Where t_e is the time sample signal, n is the number of the sample $0 \to N$, k is the number of the frequency line, Δf is the interval between two frequency lines (the time sample $\theta = N.t_e$),

$f_e = \frac{1}{t_e}$ is the sampling frequency of the signal and $f_e = 2.f_{max}$ is the maximum frequency of analysis. Generally, the decibel level L of vibration is giving by:

$$Lv = 10 \log_{10} \frac{V}{V_{ref}} \qquad (19)$$

3. Industrial application

In this work, we have examined a gas turbine installed in the DMLL management service SONATRACH, LAGHOUAT, Algeria. Measurements were performed on the gas turbine GE MS 3002, from our mobile accelerometers, as shown in Figure 3, for both levels (1 and 4) install us three sensors accelerometers positions (Horizontal, Vertical and Axial) at each level.

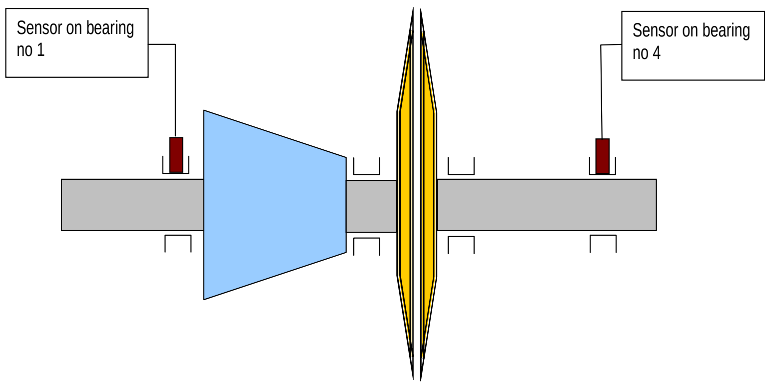

Figure 3: Sensors position installed for gas turbine monitoring

After several tests conducted on the site, the obtained spectra shown in Figures 4, 5, 6, 7, 8 and 9, its analysis shows that the energy of the fundamental signal which implies an effect of the "unbalance" and in that the axial first and second angular misalignment which manifest. An overview of GDE (Gear Defect Evaluation) values calculated from the measured bearing number 1, shows a GDA relatively high (about 0.54). This is synonymous with a problem of guiding the shaft in its bearing. It is noted that the phase difference between the vertical and horizontal position is 51.14 ° in the case of unbalance of phase around 90 °, because the error of our sensor location with translational 39.86 ° from the horizontal, this operation due to the congestion prevents the base level, so the spectrum manifest in the effect of an unbalance. The recommendations proposed by this work, are to check the alignment between the turbine and gearbox and after checking the alignment balancing on site will be programmed to reduce vibration.

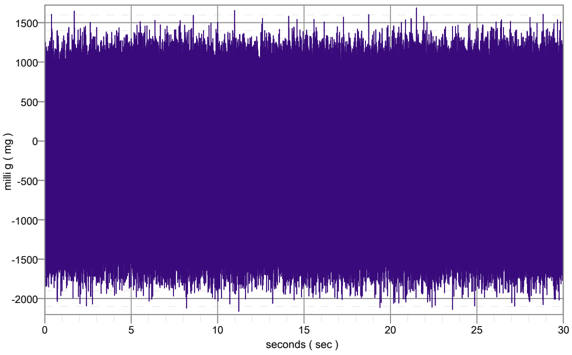

Figure 4: Signal tests RMS = 583.6 mg / Peak to Peak = 3849.2 mg / Crest Factor = 3.7 / kurtosis = 2.9 / Velocity = 5.3 mm

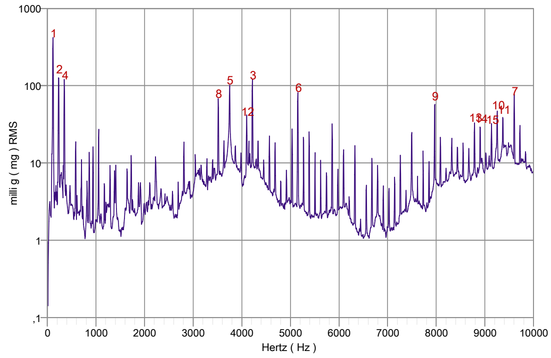

Figure 5: Spectrum obtained with FFT 0-10 000 Hz and acceleration RMS: 580.1 mg

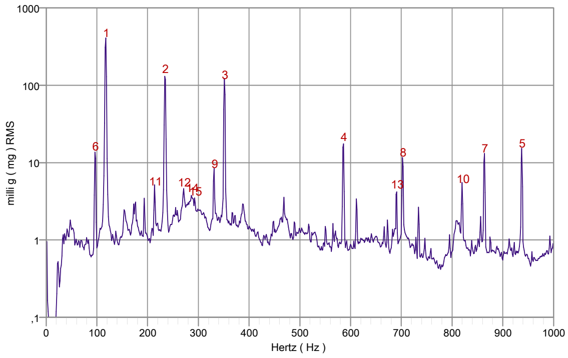

Figure 6: Spectrum obtained with FFT 0- 10 000 Hz and Acceleration RMS: 477.6 mg

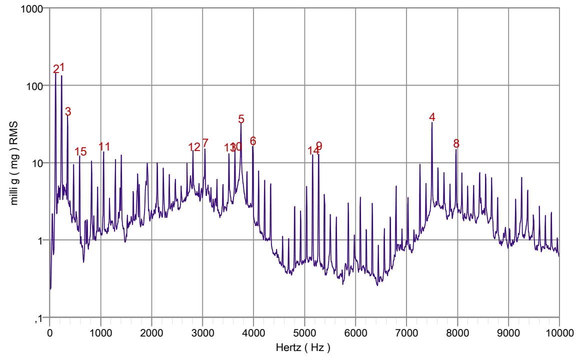

Figure 7: Spectrum obtained with FFT 0-10 000 Hz and acceleration RMS: 222.6 mg

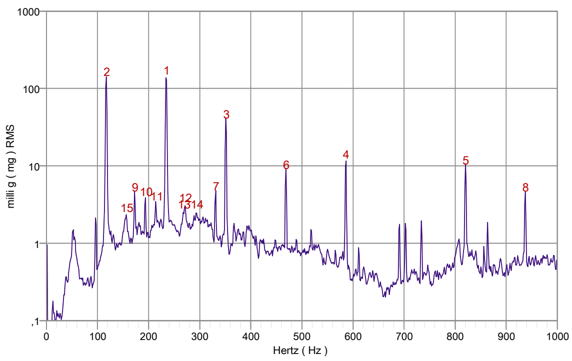

Figure 8: Spectrum obtained with FFT 0-10 000 Hz and acceleration RMS: 222.6 mg

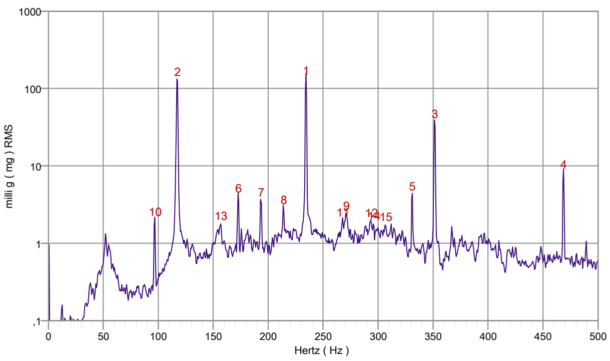

Figure 9: Spectrum obtained with FFT 0-10 000 Hz and acceleration RMS: 223.3 mg

A default of unbalance is indicated by a high amplitude component in the rotational frequency of the rotor in the radial direction, axially sometimes in the case of rotor cantilever. Also, an amplitude which can vary greatly with the speed of rotation with a phase shift close to 90 ° between the two components corresponding to orthogonal radial measurement points on the same bearing of the rotor.

The unbalance is called "static" or "dynamic" according to the levels of order 1 of the rotation frequency will vibrate for a given phase or in phase opposition radial direction. A speed: 7100 r / min speed vibration reach Vib value = 10.8mm / s and a phase shift φ = 59.49. We perform a verification with a launch mass fixation chosen (M = 170g) and start the machine. A speed V = 7113 rev / min, vibration velocity V = 4.48 mm / s phase φ = 20.78 °, as shown in Figures 10 and 11.

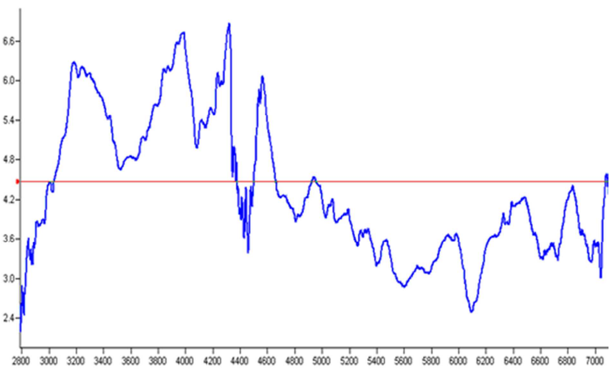

Figure 10: Vibration amplitude depending on the speed

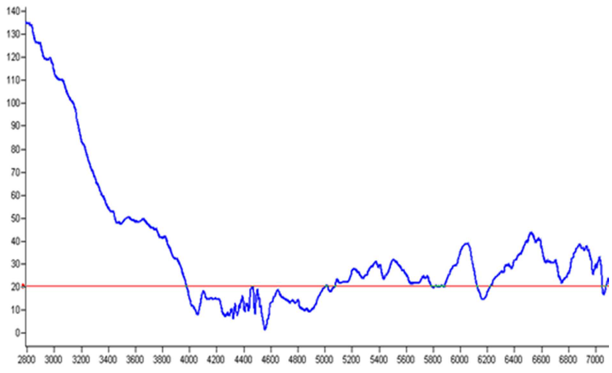

Figure 11: Vibration phase depending on the speed

Vibration is decreased to the value 4.48 mm / s, and can not decrease to a value less than the obtained value (4.48mm / s) because the fixing screws do not support the larger than the selected mass weight (M = 170g). The proposed recommendations are the vibration monitoring of gas turbine which is observed during my internship some shortcomings in the present trend of defects after the intervention of the maintenance group (cell vibration) despite its expert performance management vibration analysis and processes.

4. Conclusion

The objective of this work is to provide the elements needed to monitor the vibration behavior in gas turbine and to defining a diagnosis approach to this examined system. The defects detection, such as flaws unbalance and misalignment and other alters the structure of the signals, the amplitude can increase and modulation amplitude and phase occur, the signals collected were found in different areas of analysis, spectral domain, domain transfer function analysis and the factors. The vibrations analysis proposed in this paper, based on defect signal evaluation, contains all information concerning the state of the mechanical parts of the studied gas turbine. The difficulty lies in the analysis of vibration signals and the identification of related components to be monitored. Our investigation in this practice study, with the presence of unbalance defects (major problem in rotating machines), consist to detect and locate such defects using spectral analysis in a low frequency range. The proposed tests based on defect signal evaluation gave good and efficient results.

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