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Practical Balancing of a MS3002 Twin-rotor Gas Turbine Based on Vibration Analysis

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Abstract

The impact of vibration phenomena on gas turbine operation and production quality can significantly reduce their service life. Balancing the rotating elements of these machines is a valuable method for reducing production losses and avoiding the need for total machine dismantling. In this study, a gas turbine type MS3002 with two rotors was examined on-site to minimize the phenomenon of unbalance and provide correction angles at the rotor level. This balancing process enabled the control and improvement of the mass distribution of the rotor to maintain efforts and vibrations caused by unbalance within acceptable limits while ensuring the optimal operation of the turbine. The results demonstrate the importance of balancing techniques in extending the service life of gas turbines and maintaining optimal performance.

Keywords

gas turbine, spectral analysis, vibrations, unbalance, balancing, rotor balancing

1 Introduction

Industrial systems have become more complex and subject to harsh operating conditions. These systems require maintenance strategies that ensure their optimal operation and improve their production quality. This paper presents a practical balancing method for a MS3002 twin-rotor gas turbine based on vibration analysis. The goal is to extend the lifespan of this machine and to estimate its characterization process and availability. Moreover, the method improves the stability of the predicted vibrations of the rotor of this machine.

Recent work on the reliability and maintenance of gas turbines highlights the growing importance of hybrid approaches combining advanced statistical methods and artificial intelligence. Thanks to recent technological developments, various studies in the industrial gas turbine literature have focused on improving their reliability and availability. This allows for optimal operation of these machines, preventing failures and predicting turbine degradation coefficients. Continuous monitoring of turbines is used to estimate their reliability and the duration of malfunctions, leveraging new prediction and diagnostic methods.

Indeed, the work of Ahmed Zohair et al. [1] has significantly contributed to the analysis of gas turbine reliability, notably through a comparative study between the two-parameter Weibull distribution and a modified Weibull distribution, aimed at improving failure risk estimation. Djeddi et al. [2] explored the contribution of LSTM neural networks and deep learning to optimize gas turbine availability through failure data analysis.

Furthermore, Charrak et al. [3] proposed an approach based on the Johnson SU distribution combined with the gray wolf optimization algorithm to improve the fit of reliability models to real-world data. Meanwhile, Brahimi et al. [4] studied advanced predictive maintenance by integrating techniques such as ANFIS, LSTM networks, and reliability analysis, with a view to intelligent monitoring of gas turbines.

To prevent turbine downtime and shutdowns due to the appearance of defects and anomalies in gas turbine

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components, numerous approaches to monitoring industrial systems have been developed. In particular, the work of Hadroug et al. [5] proposed the implementation of a vibration fault monitoring and detection system based on support vector machines (SVMs), enabling efficient classification of operating states. Also, Hadroug et al. [6] developed an intelligent diagnostic approach combining Fourier transform, wavelet analysis, and neuro-fuzzy systems to improve the detection and interpretation of vibrational faults. Hadroug et al. [7] exploited adaptive hybrid fuzzy inference systems for fault detection, taking into account the dynamic behavior of gas turbines. Finally, Hadroug et al. [8] also focused on implementing a fuzzy diagnostic strategy aimed at characterizing the correlations between observed symptoms and faults, thus contributing to a better understanding and identification of vibrational anomalies.

Other work has also been carried out on the development of various fault detection strategies for gas turbines and the improvement of their monitoring systems, such as: Djeddi et al. [9], Liu et al. [10], Zhou et al. [11], Xiong et al. [12], Bisker et al. [13, 14], Pennacchi and Vania [15], Kim et al. [16] and Liu et al. [17].

This work has made a significant contribution through the integration of artificial intelligence and signal processing techniques, enabling the proper functioning of these gas turbine machines.

Within the context of monitoring by vibration analysis and the diagnosis of turbine faults, bearings are highly stressed elements that require extensive study, as they are prone to faults that rapidly deteriorate, as noted in works by Bolaers et al. [18], Shan et al. [19], Ghosh et al. [20] and Hadroug et al. [21].

Fortunately, various control tools based on vibration, thermographic, acoustic analysis, and other methods have been developed to prevent failures and ensure the constant availability of these machines, as demonstrated by works such as those by Watban Khalid Fahmi et al. [22], Hakim Bagua et al. [23], Alaoui et al. [24], Alaoui et al. [25], Qi et al. [26], Aissat et al. [27], Aissat et al. [28] and Jia et al. [29].

Also, measurements of vibration signals emitted by rotating machines in operation, for its application to the diagnosis of gas turbine systems, have been the subject of several application works, such as: Hafaifa et al. [30], Nazokkar and Dezvareh [31], Zadeh Shirazi et al. [32], Song et al. [33], Benrahmoune et al. [34], Mahroug et al. [35], Nadji Hadroug et al. [36], Alili et al. [37], Ashraf et al. [38] and Chiker et al. [39].

Gas turbines need to be balanced in the industrial context to prevent failures and damages that affect their availability and productivity. This work uses the vibration analysis method to identify the signs of an unbalance problem and to decide the appropriate actions to solve it. This ensures the longevity of the equipment by avoiding unexpected failures, which helps to keep the production equipment running and to maintain a stable level of productivity. The main contribution of this work is the validation results of the proposed balancing method in relation to its real application on site. Moreover, this method allows to detect faults and to track their evolution over time, in order to prevent any deterioration of the system under study by scheduling preventive tasks.

2 Measurements and spectral analysis of turbine vibrations

Measurements of vibration data in the gas turbine under consideration are made from accelerometer sensors installed on the shaft, using a digital data acquisition system, as shown in Fig. 1. From peak to peak obtained the emergency stop of the turbine, which is triggered automatically. Practically, the peak factor can be chosen as a scalar indicator, which represents the ratio between the peak value and the effective value of the amplitude, which is determined as follows:

$$\text{Peak factor} = \frac{\sup\left|x(n)\right|}{\sqrt{\frac{1}{N_e}\sum_{n=1}^{N_e}\left[x(n)\right]^2}}, \quad (1)$$

where $x(n)$ is the measured time signal, N_e represents the number of samples taken from the signal.

Indeed, the system for measuring vibration data for the balancing of turbine rotors studied uses two measuring positions to be balanced, horizontal and vertical. To identify the value of the unbalance and its angle, as well as for the correction. Hence, the analysis of the dynamic

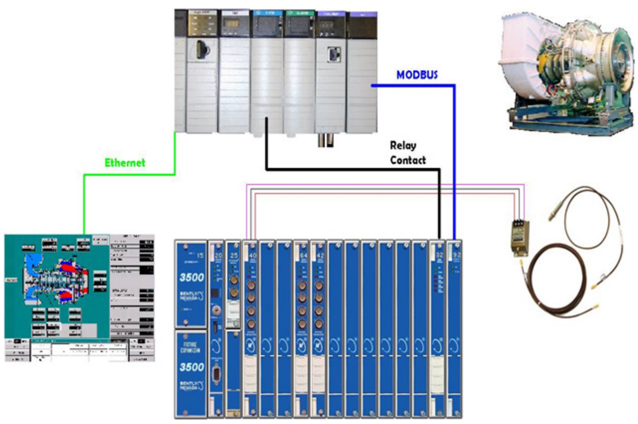

Fig. 1 Gas turbine data acquisition system

behavior of the turbine determines the excitation frequencies to limit the risks of resonance.

Vibration calculations and excitation tests must give excellent behavior at normal speed and behavior that remains acceptable at maximum speed. With vibrations in rotating equipment is written in the form:

$$x(t) = X\sin(2\pi ft + \varphi). \tag{2}$$

A fundamental property of vibration is given by the following relation:

$$D = \frac{V}{2\pi f} = \frac{a}{(2\pi f)^2}. \tag{3}$$

Like any movement, the vibrations recorded during the routes can be quantified by three fundamental quantities which are the displacement, the speed of this displacement and the acceleration undergone to carry out this displacement. From these quantities, we can choose the peak coefficient as a scalar parameter, that it represents the ratio between the peak and that of the amplitude; it is determined by Eq. (1), this peak coefficient, as shown in Fig. 2, gives information on the vibratory motion of the rotating part of the machine.

The relations between the modes of detection, for a harmonic vibration, excluding any other periodic vibration, the following conditions are considered as follows:

$$\begin{cases} X_{eff} = \dfrac{X_c}{\sqrt{2}} = \dfrac{X_{cc}}{2\sqrt{2}} \\ X_c = X_{eff}\cdot\sqrt{2} = \dfrac{X_{cc}}{2} \\ X_{cc} = X_{eff}\cdot 2\sqrt{2} = 2\cdot X_c \end{cases}. \tag{4}$$

The proposed approach for balancing the turbine rotor studied is based on vibration analysis using the Fourier transform, provides access to vibration-related parameters and

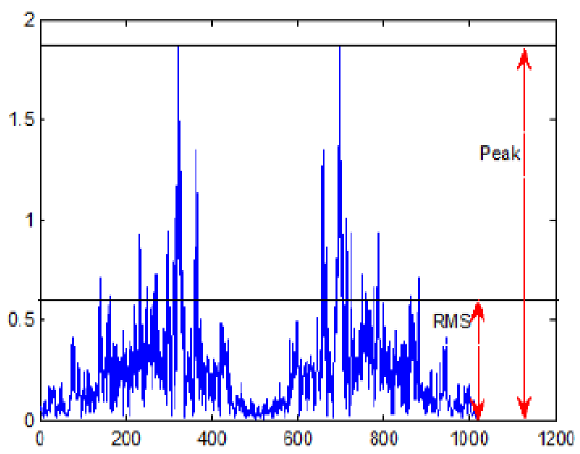

Fig. 2 Amplitudes of a vibration signal

represents the state of the vibrational motion of this machine. However, we define a real signal $x(t)$ at finite energy, the Fourier transform of the signal $X(t)$ is defined by Eq. (5):

$$\begin{aligned} TF_f\{x(t)\} &= X(f) = \langle x(t), e^{2\pi jft}\rangle \\ &= \int_{-\infty}^{+\infty} x(t)e^{-2\pi jft}dt. \end{aligned} \tag{5}$$

For a discrete signal $x(k)$ of length N given by Eq. (6):

$$X(m) = \sum_{k=0}^{N-1} x(k)e^{-\frac{2\pi jmk}{N}}. \tag{6}$$

The spectrum of an aperiodic signal is given by its TF:

$$X(\omega) = TF[x(t)] \quad \text{Spectrum of } x(t). \tag{7}$$

The energy spectral density (DSE) and in the case of discrete signals are defined *via* the discrete Fourier transform, Eq. (5) respectively by:

$$\begin{aligned} P_x(m) &= |x(m)|^2 \\ S_x(m) &= |X(m)| \end{aligned}. \tag{8}$$

It can be shown that the energy of the signal, which is written by definition as the integral of the energy transported at each instant $x(t)$, can also be expressed as the integral of the energy contributions transported by each pulse or frequency:

$$\begin{aligned} E &= \int_{-\infty}^{+\infty} |x(t)|^2 dt = \int_{-\infty}^{+\infty} |x(\omega)|^2 \frac{d\omega}{2\pi} \\ &= \int_{-\infty}^{+\infty} |x(f)|^2 df. \end{aligned} \tag{9}$$

If the square modulus of the FT is replaced by the square modulus of c_n, the energy spectrum of the signal has become:

$$|x(\omega)|^2 = |x(f)|^2, \tag{10}$$

what is represents the energy spectrum of $x(t)$ with:

$$\lim_{T\to\infty} Tc_n = \left|\int_{-\infty}^{+\infty} x(t)e^{-i\omega t}dt\right|. \tag{11}$$

Based on the analysis of vibration tests, the use of on-site dynamic balance to reduce vibration can give good results and is conducive to the reduction of rotor deformation studied.

3 MS3002 gas turbine

The MS3002 gas turbine is a mechanically driven twin shaft machine used to drive a centrifugal compressor.

It mainly consists of two mechanically independent shafts; the first contains the HP high pressure turbine and the axial compressor and the second shaft contains the LP low pressure impeller. The examined MS3002 turbine is composed of three main sections:

1. Gas generator section: this section includes the rotor and the body of the compressor which has 15 compression stages;
2. Stator section: this section is composed of suction side compressor body, front compressor body, rear compressor body and discharge side compressor body;
3. Combustion section: the combustion section of the MS3002 gas turbine engine consists of the combustion housing which is composed of six combustion bodies, six sets of combustion cap and sleeve, six sets of transition pieces and six fuel injectors, two spark plugs, two flame detectors, six smoke tubes.

The rotor of the HP turbine of the axial compressor reaches 20% of its speed by means of a launching device, the air drawn in from the atmosphere is compressed in the axial compressor and then discharged to the combustion chambers where it mixes with the fuel. Ignition is provided by spark plugs. The hot gases exiting the combustion chamber at high pressure are directed to the impeller (HP) and then to the impeller (LP) so that they are released into the atmosphere. The turbine becomes independent when the speed of the axial compressor reaches 60% of its rated speed, where there will be a disconnect between the launch device and the turbine.

For the MS3002 turbine rotor balance, two measurement points are chosen, the first on bearing No. 1 of the HP high pressure turbine, and the second on bearing No. 2 of the same high pressure HP turbine with an accelerometer fixed on the drainage pipe of landing No. 2, in order to calculate the mass and the angle of correction.

4 Results of investigations

Practically, the operation of turbine in conditions of limited stability causes vibration amplitudes often become too high for the structure to support them. For this, the amplitude of deformation of the shaft must be controlled and its resonance frequencies known, in order to avoid major vibrations and ensure optimum machine performance. Indeed, the vibrations resulting, under the effect of the forces developed by the rotating elements of the turbine, such as the bearings which are opposed to the forces of the imbalances, prove to be minimized with balancing solutions. For this, measurements of the acceleration characteristics are made, in order to judge whether the rotor has vibration problems, such as mass imbalance and center deviation. In the acceleration process, 1v, 2v, 3v, 4v, 5v vibration and 1x, 1y, 3x, 4x, 4y, 5x and 5y shaft vibration are measured on compressor side and turbine side bearing pads at high pressure. Therefore, several balancing tests were carried out on the MS3002 gas turbine, the objective of the balancing operation being the reduction of vibration levels at the rotational frequency of this rotating machine.

Firstly, the initial state of the machine has been prepared taking into account the vibration values before balancing. Therefore, when starting the machine, the vibration level reaches 8.42 mm/s peak at bearing No. 1 and 10.72 mm/s peak for bearing No. 2 of the HP high pressure turbine with a rotation speed equal to 6450 RPM. Based on the analysis of the vibration signatures of this start-up, a balancing operation on the turbine shaft to reduce the vibration levels on these two bearings is programmed. Hence, this start was considered as a first launch of the balancing operation, for the correction of the unbalance with the installation or removal of known masses on determined places of the rotor, in order to counter the effect of the initial unbalance.

For the case of HP high pressure turbine rotation speed equal to 6450 RPM, the values of the fundamental vibrations and their phases are obtained in amplitude in [mm/s eff], as shown in Table 1, which is confirmed by the results of Fig. 3, which represents the amplitude of the FFT spectrum of the vertical position of the bearing on bearing No. 1 HP turbine and Fig. 4 of the same amplitude on bearing No. 2 HP turbine.

After analysis of the spectra of Figs. 3 and 4, the dominant peak on the two planes is found at the same time on the rotation frequency, so it is an unbalance phenomenon. This requires the fixing of a test mass of 90 g on the coupling of the HP rotor. This is confirmed by the results of the second test on bearings No. 1 and 2 of the HP turbine given

Table 1 Results of the fundamental vibrations and their phases obtained for bearings No. 1 and 2 of the HP turbine

Bearing No. 1 HP vertical position	Bearing No. 2 HP vertical position	Frequency
8.42 mm/s Root Mean Square (RMS)	10.72 mm/s RMS	NGV
0.91 g	4.95 mm/s RMS	NGA
6.0 mm/s	8.72 mm/s RMS	1F0
0.76 g	3.79 mm/s RMS	2F0
2.51 g	3.22 g on the last axial compressor stage	54F0

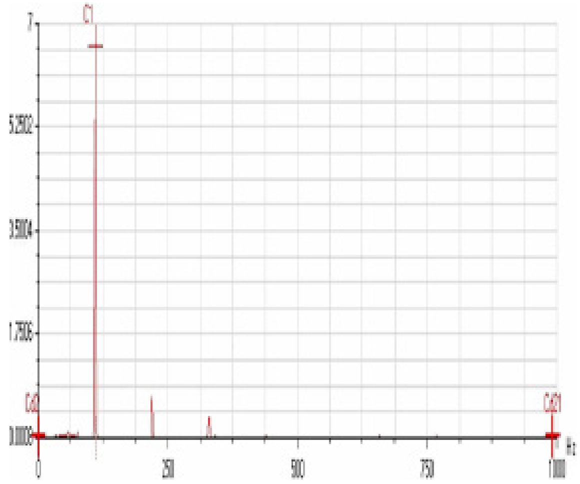

Fig. 3 Amplitude of the FFT spectrum of the vertical position of the bearing No. 1 HP turbine

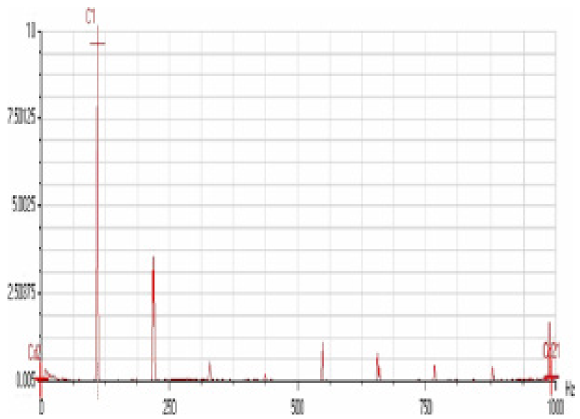

Fig. 4 Amplitude of the FFT spectrum of the vertical position of the bearing No. 2 HP turbine

in Table 2. As well as the results given in Fig. 5 of the bode diagrams of relative vibrations of bearing No. 1 of the high pressure turbine and the bode diagrams obtained for bearing No. 2 vibrations shown in Fig. 6. Hence, the vibration level of bearing No. 1 has decreased by half, not against the vibrations of bearing No. 2 only to be reduced by 20%.

Other tests have been made, to see the influence of the test mass during the balancing procedure, which consists of carrying out mass influence tests on balancing planes using calculation code made. The results obtained, shown in Table 3 and Figs. 7 and 8, confirm that the vibration levels on bearings No. 1 and No. 2 are unchanged

Table 2 Results of vibration amplitudes obtained for bearings No. 1 and 2 of the HP turbine

Bearing No. 1 HP vertical position	Bearing No. 2 HP vertical position	Frequency
7.40 mm/s RMS	11.4 mm/s RMS	NGV
1 g	4.8 g RMS	NGA

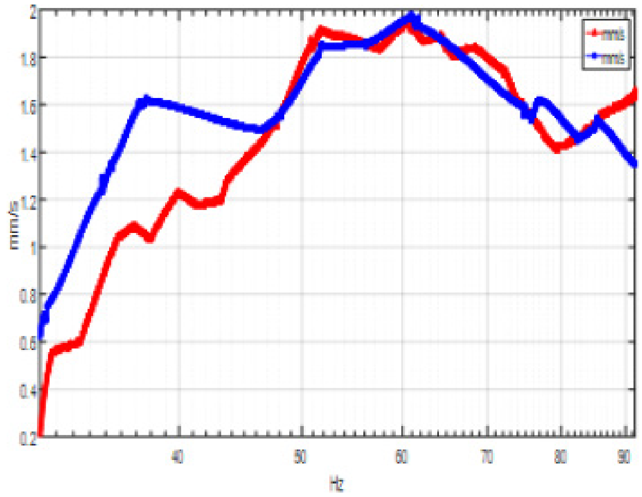

Fig. 5 Bode diagrams of relative vibrations of bearing No. 1 of the HP high pressure turbine

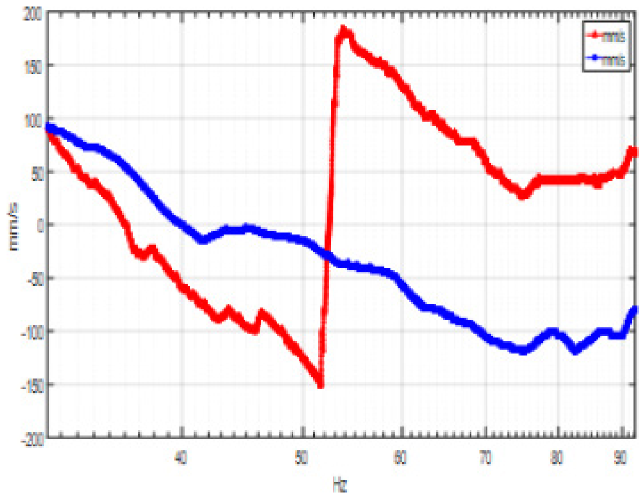

Fig. 6 Bode diagrams of relative vibrations of bearing No. 2 of the HP high-pressure turbine

Table 3 Vibration results obtained for bearings No. 1 and 2 of the HP turbine

Bearing No. 1 HP vertical position	Bearing No. 2 HP vertical position	Frequency
7.27 mm/s RMS	9.2 mm/s RMS	NGV
0.87 g RMS	2.94 g RMS	NGA

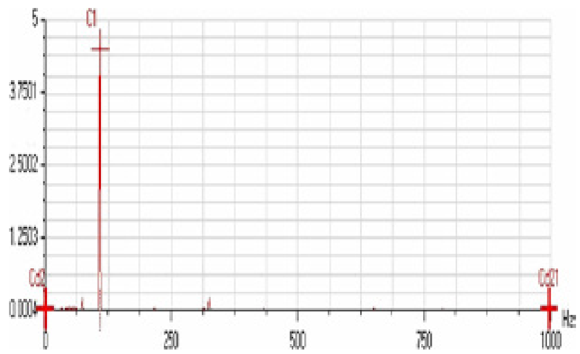

Fig. 7 Amplitude of the FFT spectrum of the vertical position of the bearing on stage No. 2 HP turbine after loading

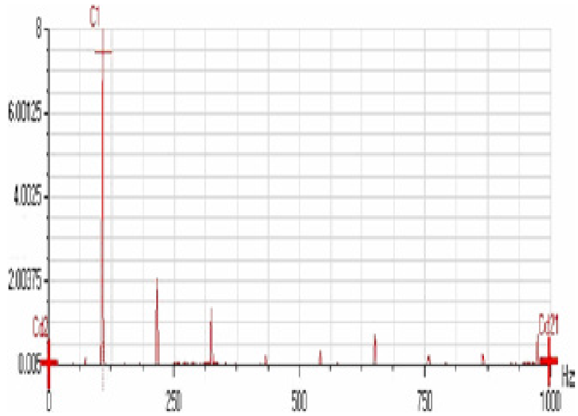

Fig. 8 Amplitude of the FFT spectrum of the vertical position of the bearing No. 2 HP turbine after loading

(no influence of the test mass). This requires an increase in the test mass with change of location.

After calculating the correction mass, the calculated mass is 260 g at 176° degree and the placed mass is 174 g at 180° degree, confirmed by the results of the 4 test in Table 4.

Following this balancing operation, an improvement in the reduction of vibrations in the axial and vertical directions of bearings No. 1 and No. 2 is clearly observed. This is confirmed by the bode diagrams of relative vibrations of bearings No. 1 and No. 2 of the HP high pressure turbine after loading, shown in Figs. 9 and 10. With the chosen correction mass is 260 g at the 176° position. With the

Table 4 Results of vibration amplitudes obtained for bearings No. 1 and 2 of the HP turbine

Bearing No. 1 HP vertical position	Bearing No. 2 HP vertical position	Frequency
4.91 mm/s rms	8.55 mm/s rms	NGV
0.92 g rms	2.67 g rms	NGA

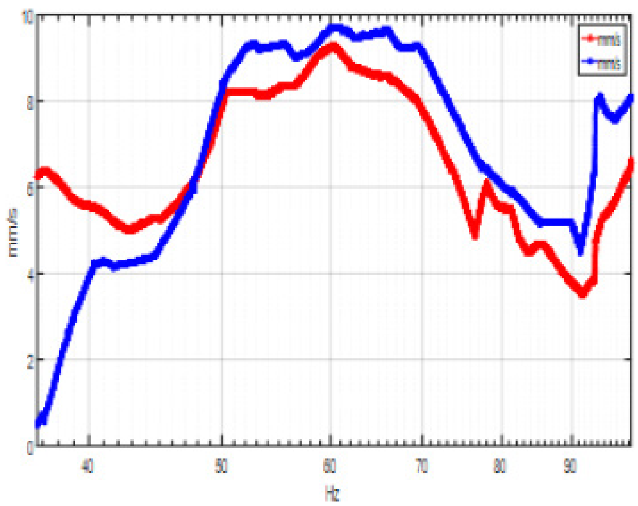

Fig. 9 Bode diagrams of relative vibrations of bearing No. 1 of the HP high pressure turbine after loading

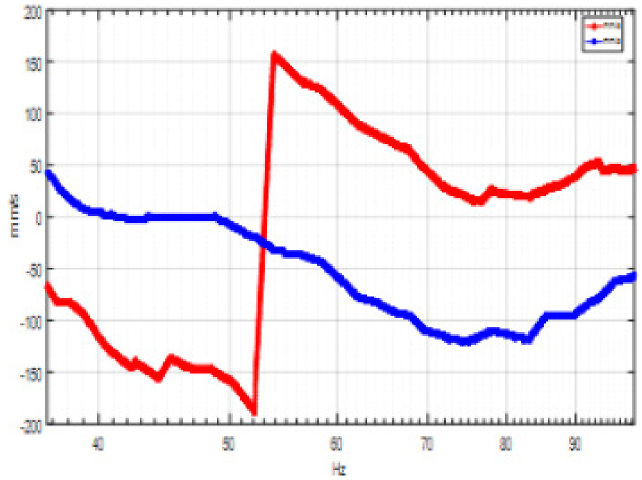

Fig. 10 Bode diagrams of relative vibrations of bearing No. 2 of the HP high-pressure turbine after loading

removal of the mass of 180 g from the 0° position in front is made to ensure the results of the balancing operation.

During the acceleration process, the 1v, 2v, and 3v vibrations fluctuate considerably and peaks appear several times. When the speed is about 6000 rpm, the peaks of 2v and 3v appear at the same time, and the maximum value of 2v is 78 µm.

When it reaches 6600 rpm, the peaks of 1v, 2v, and 3v appear at the same time. When it reaches 2150 rpm, the 1v and 2v peaks appear at the same time, and the maximum value of 1v is 83 µm at approximately 2800 rpm. Similarly to the phase change point of view, the phase changes faster when the peak occurs. When the speed is 3220 rpm, the 1v and 3v of the slab vibrations are in-phase vibrations, and when the speed is 2150 rpm, the 1v and 3v of the slab vibrations are an anti-vibration phase. However, when the rated speed is 3000 rpm, the 1v, 2v, and 3v tile vibrations are all very small, all less than 15 µm.

The results obtained show that the shaft speeds influence the vibration behavior and stability of the rotating shafts. As the proof mass (correction) decreases the amplitude of the shaft and disk as a function of frequency, it is found that as the frequency increases; it is noted that the balance is maintained.

5 Conclusion

The experimental tests carried out within the framework of this work, through the tests carried out on-site on the gas turbine studied, made it possible to optimize the distribution of the blades of a stage of the rotors of the turbines, in order to determine the quantity and the location of the masses of correction (unbalance). This operation has the advantage of reducing production losses on site by avoiding the turbine shutdown for overhaul. The results obtained also show the presence of high levels of vibration in bearing 2 of the high-pressure (HP) turbine, which confirms the presence of a problem of deformation or loosening on this rotor (on the anchor bolts). If necessary, it must be repaired at the next overhaul, i.e., after 16,000 h of operation. This is validated by the program developed and confirmed by practical results, using the data recorded on the turbine rotor and the analysis of vibration tests on the dynamic behavior of the structure and shaft, such as resonance frequencies, fouling, and vibration tolerance thresholds. This has led to an improved rotor mass distribution to reduce the eccentricity of the center of gravity and the resulting centrifugal forces. Practically, the proposed approach using real-time vibration analysis allows reducing the costs related to maintenance, with precision on the type of defects (unbalance, misalignment, defective fasteners, shaft deviation, friction, play, resonance). As well as the limitation of the production stops with good reliability and productivity of the studied turbines.

Acknowledgement

The goal of this work is rotor balancing of an MS 3002 turbine using vibration analysis data, allow to increase the life of this equipment with stable operation. Hence, the various investigations were carried out by the joint research team on gas turbines in collaboration with the Applied Automation and Industrial Diagnostics Laboratory of the University of Djelfa, Algeria.

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