TURBOGENERATOR DYNAMIC ANALYSIS TO IDENTIFY CRITICAL SPEED AND VIBRATION SEVERITY

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Vibration analysis is the main technique in diagnosing dynamic behavior of industrial machinery and equipment. This technique is applied for the vibration diagnosis 40MW turbo generator which is driven by a gas turbine. Due to high vibration amplitudes of bearings a numerical analysis of the generator rotor is used to highlight the dynamic phenomena and determination of critical speeds. So to validate the numerical model and identify critical speed a complex experimental protocol was conducted to measure both the coupling vibration monitoring system in real time and local measurement by fixing the sensor on the bearing housing.

The use of advanced digital signal processing led to highlighting the main faults and obtains the final solution to reduce the amplitude of vibration.

Keywords: vibration, critical speed, generator, mechanical looseness

1. Introduction

Power plants using gas turbines are one of the most effective solutions for producing electricity. Study of dynamic and thermal phenomena represented and represents topical energy industry around the globe. The importance of these machines required the development of new systems and techniques for monitoring the dynamic behavior of the entire chain of rotor bearings. These solutions are absolutely necessary in operating conditions from becoming more severe.

In reality, however, no rotor operation is vibration-free due to various excitations and so attempts of researchers are aimed at minimizing the vibration level [1]. The operating conditions of steam turbine rotor are severe, which include high temperature, high pressure [2] and also high vibration.

Turbo-generator rotor shaft may vibrations due to variable stiffness of bearings, a random exciting torque, the rotors misalignment, the whirling shaft, the disturbance of electricity, etc. In the resonance condition the vibration amplitudes may reach very quickly high unacceptable values [1]. Long term operation under such conditions may induce low cycle fatigue in rotor's material [2] and also in other turbine components. Because of the importance of the rotors

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in the power plant, the dynamic analysis and diagnosis attracts many researchers [3, 4].

Vibration diagnostics allow highlighting the various phenomena which occur during turbine operation. The experimental tests were performed on a 40MW gas turbo-generator. The gas turbo-generator is composed of a compressor, rotor turbine a gear box and the generator with the excitation side.

This paper considers the determination of critical speed on the generator and also analyzes vibration severity level.

2. Research context

In this paper the main focus is the determination of critical speed of generator rotor shaft and the analysis of the vibrations severity to highlight the most important effect of vibration and to identify the vibration cause. Gas turbine and generator vibrations have been recorded, with the main target to analyze and diagnose the vibration behavior and define the next maintenance activities. The main reason of the measurements was an observation from the control room operation personnel that it is a vibration variation with the reactive power at the generator bearing, load gear. The gas turbo-generator configuration and description of the components elements is show in the figure 1.

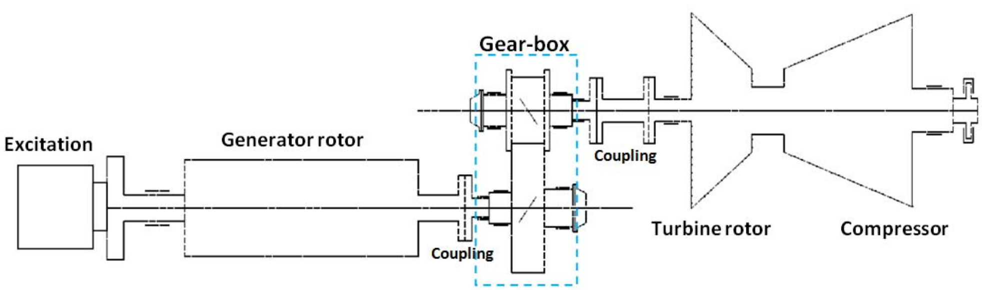

Fig.1. The configuration of the gas turbo-generator

3. Experimental setup

The vibration analysis was performed in two different ways, absolute vibration using the original speed sensors of the turbo-generator in control room and absolute vibration on the bearing housings of the gear-box and generator in local measurement. In order to analyze the vibration signature of the gear-box, accelerometers have been installed and vibration recorded continuously up to a frequency of 10 kHz.

The measurements have been performed during the start-up and loading process, to evaluate the critical speeds locations and magnitude, to identify possible changes with load variations. In order to evaluate the vibration amplitude sensitivity with MVAR variation, the unit was loaded gradually at zero MVAR up to a full load of 41 MW and at this active power the reactive was changed.

The experimental equipment used for measurement is divided in two groups: control room measurement and local measurement. For the control room monitoring system the equipment consist of a CRIO 9014 National Instrument and Fastview Digitline software, figure 2. The condition monitoring system using in control room is a Mark V cubical and the permanent vibration transmitters provide the signals from vertical direction of the existent bearing sensors with 150mV/IPS sensitivity. The vibrations are measured first as vibration velocity; peak values pk and r.m.s. values.

Speed sensors, installed temporary on site, on Turbine side on the extension shaft to high speed gear and on generator side at the exciter side (two laser sensors). The control room acquisition signal settings are performed with 5000 S/s/ch with 24 bits resolution.

Fig.2. The experimental setup for control room measurement data

On the local measurement the local signal source derived from temporary sensors, accelerometers with 100 mV/g sensitivity which have been installed on vertical, horizontal and axial direction on gearbox housing at high speed gear, load gear and exciter side of generator. The speed reference is the speed laser sensor installed on exciter side of generator.

For acquisition and signal processing in local measurement configuration was used two modules National Instruments USB 4432 with a speed of 25kS/s with 24 bits resolution, figure 3.

Fig.3. The experimental setup for control room measurement data

The exact position of the sensors and measurement protocol in local measurement configuration is shown in Figure 4.

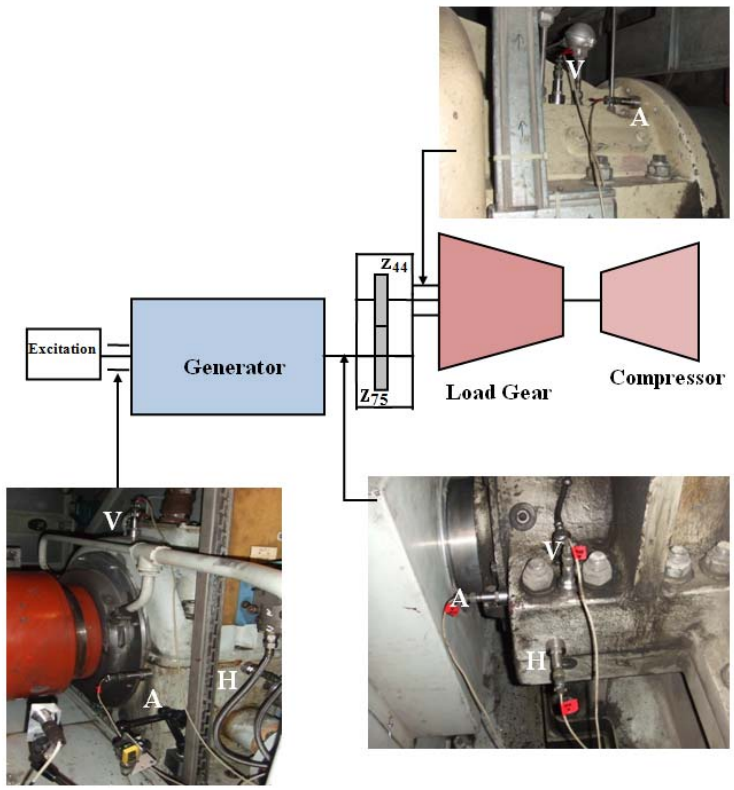

Fig.4. Sensor position on the bearing: horizontal (H), vertical (V) and axial direction (A)

5. Vibration issue

Vibration measurement is made in terms of gas turbo-generator startup and according with certain operating conditions. The first action of diagnosis is to evaluate the vibration effect and to identify the cause o vibration, the main purpose of the maintenance report for overhaul. During the start-up the bode diagram highlights a resonance of generator on gearbox bearing side at 1587 RPM and the second one at 2795 RPM which is not a resonance corresponding to the excitation bearing, figure 5.

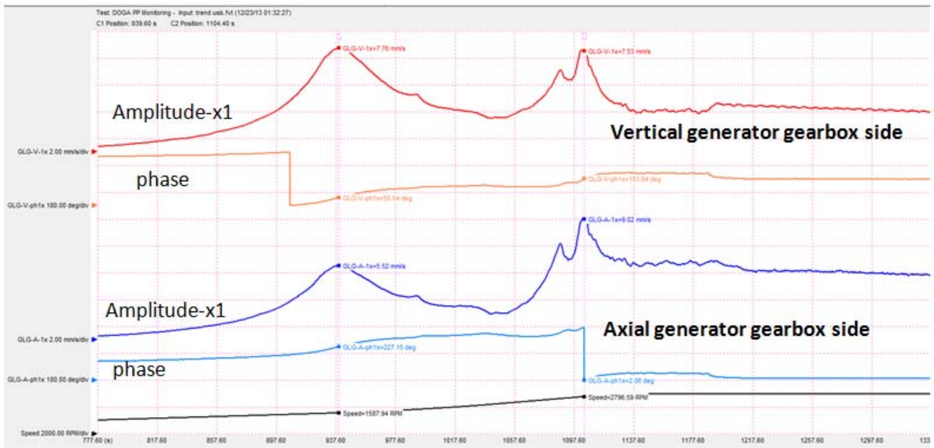

Fig.5. Bode diagram for generator bearing on gear-box side

The axial vibration of the gear-box housing is large amplitude and has a resonance at the same speed, figure 6. Local resonance is suspected on the gearbox casing due to the mechanical looseness of the foundation.

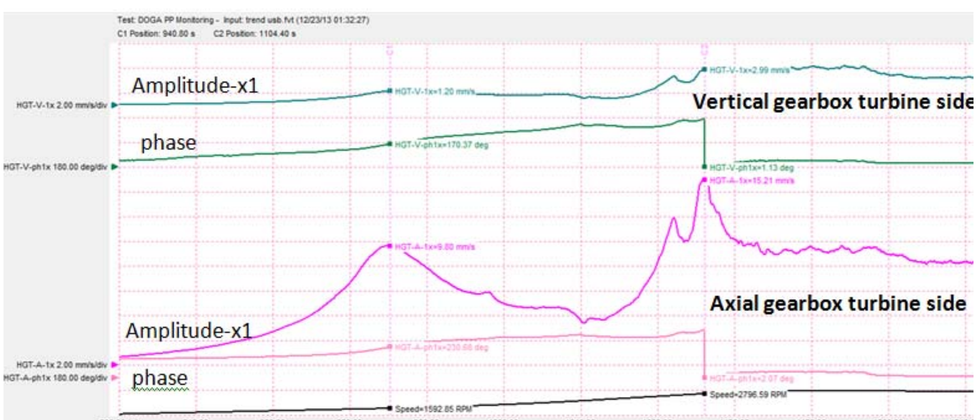

Fig.6. Bode diagram for gear-box bearing on turbine side

The vibration amplitude for generator bearing on excitation side is presented in table 1.

Table 1

Generator	Horizontal	Vertical	Axial
	mm/s rms		
Excitation bearing	2.5	4.5	1.2
Gear-box bearing	2.4	5.5	4.8

According with the ISO standard 10816 and gas turbo-generator specific data the vibration levels exceed limits. In this case the dynamic rotor theory is used to calculate the critical behavior and to identify critical speed.

6. Theoretical model of generator rotor

To characterize the dynamics of generator rotor was made a numerical model to emphasize the critical speed. A typical dynamic behavior in rotating machinery is the critical frequencies where in most cases the flexible rotors are two critical speeds, figure 7 [3, 4 , 5].

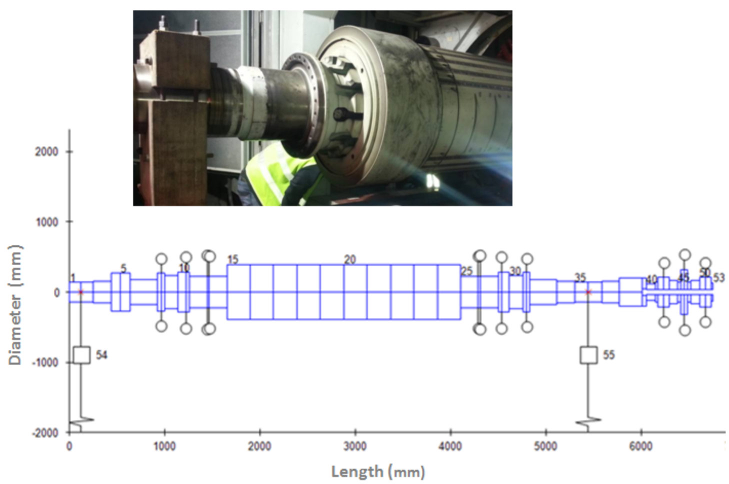

Fig.7. Theoretical meshing of the generator rotor

For critical speed determination for the numerical model require the radial stiffness for the bearings. The stiffness bearings are provided by the manufacturer of the rotor generator, figure 8.

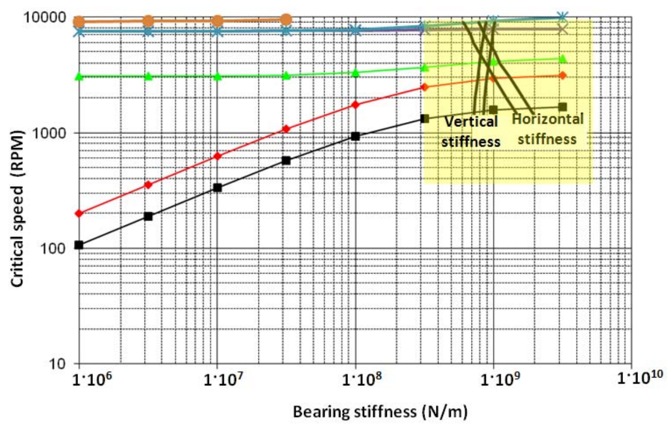

Fig.8. Bearings stiffness on horizontal and vertical direction

The model results are shows in figure 9 and 10 for the rotor bearings: excitation side (drive-end DE) and for gearbox side (non drive-end NDE).

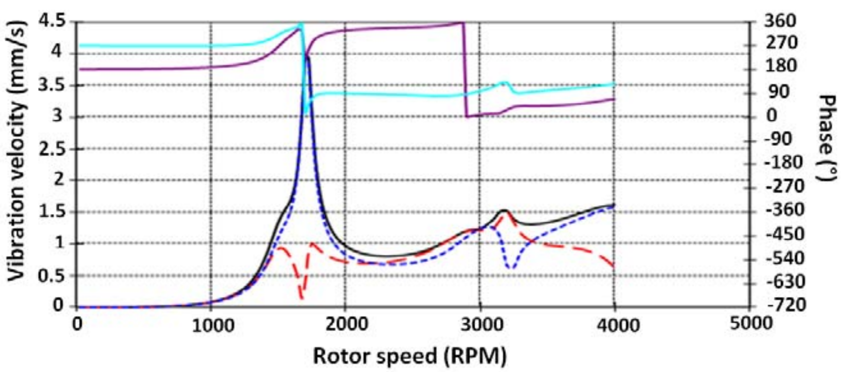

Fig.9. Critical speed for excitation bearing side (drive-end DE)

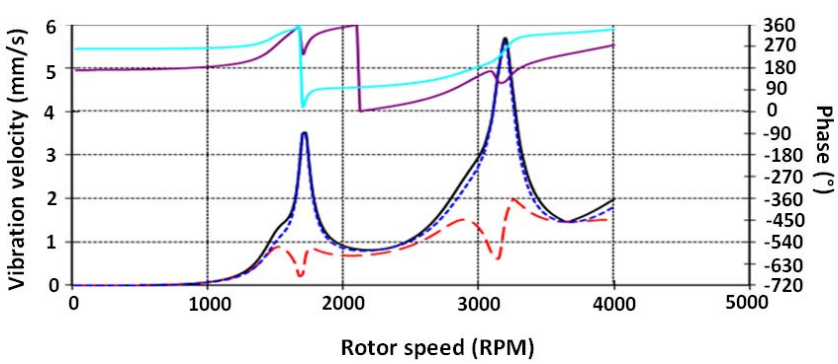

Fig.10. Critical speed for gearbox bearing side (non drive-end NDE)

The first critical speed calculate is around 1700 RPM and the second one is 3200 RPM. The two critical speeds are the result of dynamic rotor model and are also the two critical areas called static and dynamic respectively [5]. The two modes of vibration are confirmed by measurements but do not highlight any critical speed around 2800 RPM which means that this is a local resonance reducer caused by mechanical looseness of the foundation.

6. Vibration analysis and results

The proposal repair after vibration diagnosis report consists in: inspection and gear repair; inspection and repair of gearbox anchor foundation system; alignment of the generator and balancing. The vibration trend during the gas turbo-generator after de overhaul is show in the figure 11, 12 and 13.

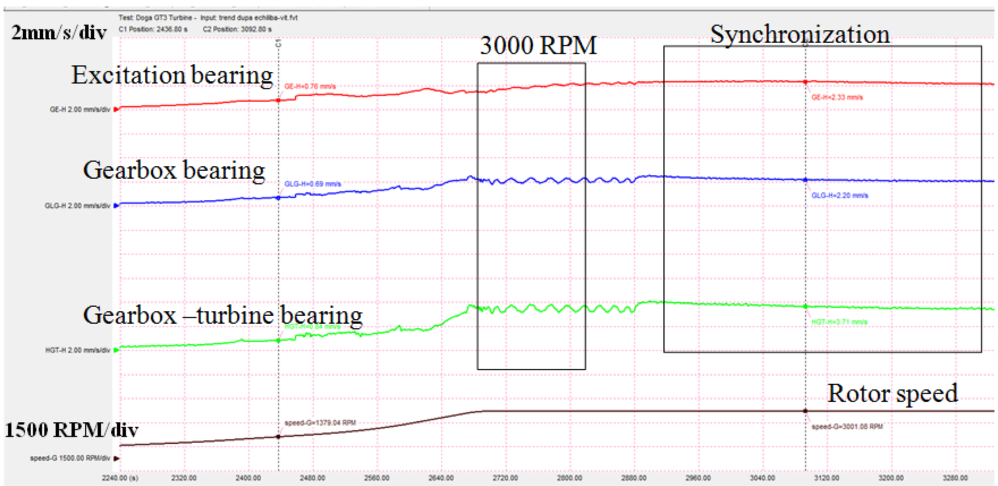

Fig.11. Vibration trend after overhaul – horizontal direction

The vibration analysis before the outage was indicating a resonance peak on generator bearing gearbox side, on vertical and axial direction at around 2800 rpm. This resonance was a result of the improper loading of the frame and now it is not present (was shifted over 3000 rpm). After the overhaul is observed vibrations shows a much lower level than before repair. The evolution of vibration parameters presents constant amplitude on the synchronization state. These results show that the foundation problems in terms of anchoring can lead to severe dynamic effects. In this configuration, the gas turbo-generator dynamic stability can be strongly influenced by the lack of structural rigidity.

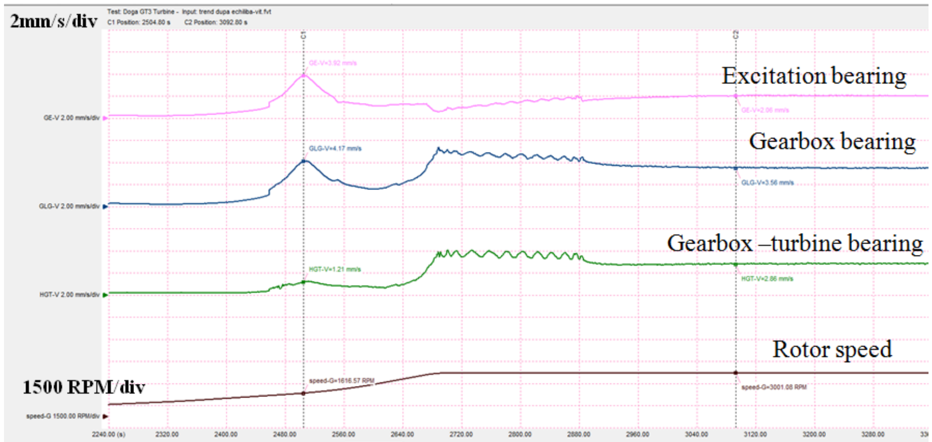

Fig.12. Vibration trend after overhaul - vertical direction

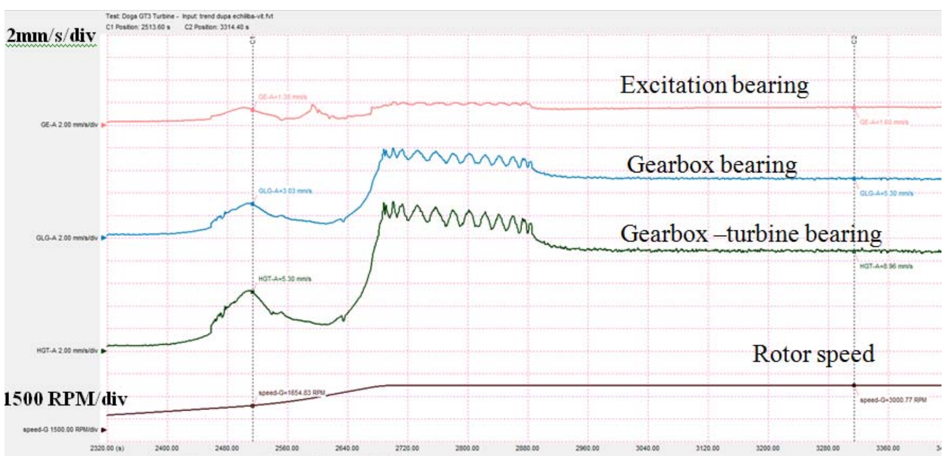

Fig.13. Vibration trend after overhaul - axial direction

The vibration level after repair status is presented in the table 1. The vibration amplitude on the excitation bearing of the generator is lower than before repair and the same evolution we have for gear-box bearing were on vertical direction the vibration level is quite low.

Table 2

Generator	Horizontal	Vertical	Axial
	mm/s rms		
Excitation bearing	1.8	1.9	1.2
Gear-box bearing	2.9	3.2	4.7

6. Conclusions

Vibration diagnostics allow highlighting the various dynamic phenomena which occur during gas turbo-generator operation. A vibration issue has been detected and it required a complex dynamic analysis. The vibration analysis was performed in two different ways, absolute vibration using the original speed sensors of the turbo-generator in control room and absolute vibration on the bearing housings of the gear-box and generator in local measurement. All tests are performed on three-dimensional configuration.

The first action of diagnosis is to evaluate the vibration effect and to identify the cause o vibration. During the start-up the vibration analysis highlights a resonance of generator on gearbox bearing side at 1587 RPM and the second one at 2795 RPM which is not a resonance corresponding to the excitation bearing.

For this reason a numerical analysis of the rotor dynamic was achieved by developing a theoretical model that allowed the determination of critical speeds.

The vibration analysis show a local resonance reducer caused by mechanical looseness of the foundation. This prediction was confirmed after the repair and reinforcement of the frame structure.

The model validation and prediction confirmation was done by testing the gas turbo-generator after repair when the vibration level decreases and the critical speed of 2800 RPM does not exist.

A complete analysis is proposed in the future to consider the entire diagnosis such as acceleration, angular position of vibrations vector, unbalance, misalignment, bending shaft, electric parameters etc..

R E F E R E N C E S

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