Vibration Analysis For Reducing Excessive Vibration Level on Gas Turbine Generator (GTG) 100 MW in Cogeneration Power Plant

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Abstract. The goal of this research was to gather knowledge and to analyze the problem of excessive vibration level of Gas Turbine Generator (GTG) 100 MW system installed in cogeneration power plant in Sumatra Island, Indonesia which supports one of biggest oil and gas industries in Indonesia. The case research related to vibration problem were presented to diagnosis the main causes of excessive vibration that occur in the gas turbine generator during operation. Vibration analysis is one of the most important activities in predictive maintenance. Vibration monitoring system and machinery diagnostic technical specification are presented. The vibration data of this research were collected using online vibration monitoring system Bently Nevada 3500 series and system 1® display software at different bearing locations during transient (shutdown & start-up and steady state (on-line) condition. Assessment on overall vibration levels shall refer to Original Equipment Manufacturer (OEM) alert & danger set points, as well as relevant ISO 20816-2 standard. Finally, recommendation of reducing excessive vibration level is provided to ensure safe and reliable operation of the GTG unit.

1. Introduction
Currently, the gas turbine & generator is the most versatile item of turbomachinery system. The gas turbine is a power plant that produces a great amount of energy depending on its size and weight. Industrial gas turbines can be used in several different modes in critical industries such as oil and gas industries, process plants, aviation, marine, industrial mechanical drives and electrical power generation [1].

The condition of a gas turbine generator (GTG) engine can be estimated by measuring the vibration levels. As a system, GTG vibration is normally monitored by the plant’s condition monitoring system that serves as back up to its machine protection system. The vibration [2] is defined as any motion that repeats itself after an interval of time. This motion can theoretically continue endlessly if there is no damping in the system and no external effects (such as friction). The physical motion of rotating machines generates vibration, which gives a physical indication of the health of equipment and the generated vibration frequencies and magnitudes represent the machine vibration signature. Vibration analysis is one of the most important in predictive maintenance, which is a powerful tool which allows early detection of faults in rotating machinery. The malfunction of machines like unbalance, bent shaft, misalignment, mechanical looseness, resonance, rotor rubs, journal bearings faults, electrical faults, etc. can be determined in detail using vibration analysis [3].
Machinery diagnostic application is an important concept required for effective machinery malfunction diagnosis and determines root cause for machinery vibration problem. Analysis of phase, vibration vector, fast Fourier transform (FFT), data plots like time base, average shaft centerline, polar, bode, spectrum, etc. are discussed in detail [4].

In the field machinery vibration monitoring and analysis practices, a variety of relevant measurement and standards for rotating equipment are developed and published by International Organization for Standardization (ISO). Generally, assessment on overall vibration levels of gas turbine generator shall refer to OEM alert & danger set points, as well as relevant ISO 20816-2 standard [5].

In this research, vibration analysis is carried out on one of three GTG machine trains, which consist of SIEMENS SGT6-3000E gas turbine and BRUSH DAX air cooled generator with 100MW load capacity and rotational speed 3600 RPM. This machine train has been experienced an excessive vibration level reading. This excessive vibration values have been recorded at bearing #2 and bearing #3 as shown in the vibration monitoring system. Analysis of this vibration behavior are required to find out the remedial action to be done during next maintenance program.

The concept of vibration measurement in this machine is permanent monitoring, which is a system whereby a set of instruments is continuously checking machine condition at a limited number of measuring point [6]. Gas turbine generator vibration monitoring system, like other rotating machineries, are usually equipped with some contact proximity probes, as vibration indicator. These indicators are usually installed in main vibration monitoring tools. Both vibration data and trends are captured and presented [7]. The vibration data were collected using online vibration monitoring system Bently Nevada 3500 series and system 1® display software at different bearing locations during transient (shutdown / start-up) and steady state (on-line / full speed full load) condition. The shaft relative vibration (XY non-contact proximity probes) and absolute vibration (seismic contact probes) were installed on each of the GTG train’s fluid-film bearings. Both vibration monitoring systems equipped with alert and danger set point. The GTG is tripped in danger condition. The XY pairs of non-contact proximity probes are mounted at 45-degrees left (Y-probes) and 45-degrees right (X-probe), shown in figure 1.

![Figure 1. Relative and absolute vibration probes installation diagram](image)

2. Theory Approach
In general, in most power plants industries, rotating parts are key components to generate electric power. The faults of rotating machinery may cause its machine performance degradation and entire system failures or break downs. These conditions are directly related to plant maintenance cost and even the level of safety. As part of condition – based maintenance, implementation of vibration signal monitoring is one of important way to avoid and prevent plant system failures. The following discussion explains about machinery fault types detected using vibration analysis as technical reference of this research.
Unbalance. Unbalance is the most common source of vibration in rotating equipment. However, rotor with a vibration problem should not automatically be assumed to be out of balance. Vibration spectrum can diagnose true machine unbalance condition. Vibration due to unbalance occurs at a frequency of 1X shaft running speed of the unbalance element, and its amplitude is proportional to the amount of unbalance. As explained and documented by [8], the unbalance of rotating equipment has unique characteristic in vibration spectrum or behavior of machine and can be characterized primarily by one times (1X) shaft running speed.

Misalignment. The second major concern on malfunction of rotating equipment is misalignment. One of study on rotor unbalance and shaft misalignment in rotating machinery has been conducted. In order to understand the dynamic characteristics of these machinery faults, a model of a complete motor flexible-coupling rotor system capable of describing these failures was developed. Generalized system equations of motion for a rotor under misalignment and unbalance conditions were derived using the finite element method [9]. In general condition, excessive misalignment typically produces a large twice (2X) harmonic component of vibration and a high level of axial vibration [10].

Bent or bow rotor. The phenomenon of bends in rotor may be caused in several ways, i.e. due to thermal distortion, creep or a previous large unbalance force. In general, when a bent rotor is encountered, the vibration in the radial as well as in the axial will be high and the vibration spectrum will normally have 1X and 2X component at slow roll speed. During thermal bow, rubbing will occur between rotor and stator, causing a local hot spot and thermal expansion, there may be specific symptoms will assist in the vibration diagnosis [11].

Rotor to Stator Rubs. Rotor to stator rub, can be one of the most damaging malfunctions of rotating machinery. Rotor to stator rub is the event where rotating parts contact with stationary parts. Rotor to stator rubs produce a vibration spectrum that similar with mechanical looseness. The rubbing may be either partial or continues. Rubbing excites one or more natural frequencies of the shaft and generates a series of frequencies to the spectrum that are integer fractions of sub-harmonics of the running speed, for example 1/2X, 1/3X, etc. [12].

Cracked Rotor. Fatigue cracking is one important fault in rotating machines. The cracked rotor vibration symptoms and the early diagnosis of cracked rotor can be detected mainly on two symptoms, i.e. changes in 1X and/or 2X vibration vectors. Unexplained changes in the synchronous (1X) shaft relative lateral vibration amplitude and/or phase at the operating speed and changes of the slow roll vector on start-up and or shutdown. While the occurrence of twice the rotative speed (2X) vibration component – occasionally at the operating speed, but especially on start-up and shutdown [13,14].

Mechanical looseness. Mechanical looseness in rotating machine, can normally occur at internal assembly, machine to base plate interface, and machine structure. A looseness between the rotor – supporting pedestal and the foundation is a common malfunction in rotating machines and usually caused by the poor quality of installation or long period of impact vibration of the machine. High harmonics are usually associated with mechanical looseness. The system with mechanical looseness generally exhibits changes in the synchronous responses and an appearance of the 1/2X fractional harmonic component and multiple harmonic component such as 2X, 3X, etc. [15].

Oil Whirl, oil whip and dry whip. Oil whirl, oil whip and dry whip condition are several operational problems with vibration of machines supported on journal bearings. These kinds of instabilities are serious malfunctions in rotating machinery and may cause a machine catastrophic failure if they occur simultaneously. Vibration due to journal bearings are complicated and have various characteristic. The instability appears at sub-synchronous frequency of about slightly less than 1/2X, and close to 0.47X. [16].

Electrical Faults. Vibration problem in generator, as part of gas turbine generator system, was normally induced by electromagnetic forces in addition to the usual forces from mechanical effects such as unbalance, misalignment, etc. This fault can be extremely frustrating and may lead to greatly reduced reliability. However, this electrical malfunction on generator can be detected thru vibration spectrum. In practice, the vibration spectrum or pattern emerging due to electrical problem on generator will be at 1X shaft running speed and will thus appear similar unbalance. To do this,
understanding the nature of vibration spectrum can assist in identifying the exact malfunction in electrical machine [17,18].

Resonance. Resonance means a phenomenon that occurs when a periodic external force is applied to a system having a natural frequency equal to the driving frequency. Resonance is also related to critical speed. The excessive or high vibration amplitude at critical speed of the machine can be catastrophic for any system and must be avoided at all costs [19].

Several studies related to excessive vibration on gas turbine phenomenon have been conducted, i.e. study on dynamic behavior characteristic related to resonance and critical speed on gas turbine GE MS3002 [20] and the phenomenon of high vibration in gas turbine 17.5 MW load capacity installed in power and desalination plant [21].

The correction of common faults caused by vibration is required to ensure a safe and reliable operation on machine. Field balancing, correct alignment and bearing inspection or repair are several methods as action recommendation to solve vibration problems in rotating equipment [22,23].

3. Research Methodology
The research methods are explained and presented in this section. First, the main problem is defined by determining and explaining in detail the problem occurred in cogeneration plant related to excessive vibration level. The components that cause vibration within the machine must be identified. The running speed of the machine, operation condition, and type of measurement that produce the FFT spectrum are also included in this stage.

Second, several technical literatures related to this research are reviewed and used as technical references. Third, perform a complete vibration data acquisition and processing. This includes vibration data collection, process them in the vibration monitoring system, and record the results in a form suitable storage system. Fourth, vibration data trending. In this stage, the trending and filtering of vibration data during transient and steady state condition were carried out. Fifth, vibration data is analysed to find out the main root cause of excessive vibration level. In this stage, analysis usually follows a process of elimination which the components or issues that do not contribute to the system are eliminated. The other remaining component which contributes in affecting the machine health shall be identified. Then finally, provide a complete recommendation as remedial action to be conducted to solve the problem of excessive vibration. The execution of remedial action shall follow the scheduled site maintenance program.

4. Result and Discussion
This section presents research execution and full spectrum analysis of the vibration response. The vibration data were retrieved on mid of June 2019 during GTG maintenance program (compressor wash program) by covering two operational condition, i.e. transient (shutdown/start up) and steady state (online 100 MW load capacity). From this data, an excessive vibration case history related to GTG unit is analysed and discussed.

The excessive vibration level on GTG was found during full speed no load (FSNL) on bearing #2 gas turbine with 6.8 mils and bearing no#3 generator with value 7.4 mils. While vibration level during steady state / full speed no load (FSFL) was detected at 6.0 mils on bearing #2 gas turbine and at 6.10 mils on bearing #3 generator. Both alert and danger set point on GTG unit are 5.7 mils and 8.6 mils, respectively. These overall shaft vibration amplitudes values on both conditions were above alert limit and zone C of ISO 20816-2. Hence, machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. In general, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.

Vibration level in transient condition should be focused because of the higher amplitudes. Figure 2 and figure 3 show bode plot of bearing #2 and bearing #3 on transient condition, i.e. from start-up to full speed no load (FSNL). During this period, at the vibration value reached about 0.69 mils (bearing #2) & 0.38 mils (bearing #3) at slow roll speed and going to 6.0 mils (bearing #2) and 6.10 mils
(bearing #3) at operating speed (3600 rpm). The overall amplitude value is predominated by 1X filtered amplitude and indicates that the rotor on unbalance condition.

![Figure 2. Bode plot of bearing #2 on machine start up](image)

![Figure 3. Bode plot of bearing #3 on machine start up](image)

In the same period as above, figure 4 and figure 5 illustrate the shaft relative vibration trends of bearing #2 and bearing #3 during transient condition. The direct and 1x component vibration amplitude trends of both bearing #2 and #3 indicated significant decrease and significant phase angle changed.

![Figure 4. Direct and 1X shaft relative vibration trend plot bearing #2](image)
Figure 5. Direct and 1X shaft relative vibration trend plot bearing #3

Figure 6, figure 7, figure 8, and figure 9 describe the direct (compensated) and 1X filtered orbit plots on bearing #1 and bearing #4, within FSNL and FSFL condition, respectively. All plots explained that significant orbit shape changed captured on bearing #1 and #4, large movement shaft centreline on bearing #4 at FSNL as compared with FSFL, which were suspected as bearing and seals clearance problem.

Figure 6. Orbit plot of bearing #1 (FSNL)  Figure 7. Orbit plot of bearing #1 (FSFL)  
Figure 8. Orbit plot of bearing #4 (FSNL)  Figure 9. Orbit plot of bearing #4 (FSFL)  

Figure 10, figure 11, figure 12, and figure 13 illustrate vibration amplitude changes of shaft relative full spectrum on bearing #2 and bearing #3, respectively. On bearing #2 and bearing #3, vibration amplitude changes of shaft relative full spectrum, and 1X vibration amplitude captured significant which lead the bearing on unbalance condition.
Figure 10. Full spectrum plot of bearing #2 (FSNL)

Figure 11. Full spectrum plot of bearing #2 (FSFL)

Figure 12. Full spectrum plot of bearing #3 (FSNL)

Figure 13. Full spectrum plot of bearing #3 (FSFL)

Figure 14 and figure 15 demonstrate the average (AVG) shaft centreline during start up to base load period on both bearing #3 and bearing #4. From this plot, it suspected that rotor has experienced an abnormal behaviour such as preload, probably rubs and also jacking oil system problem during this period.

Figure 14. AVG shaft centreline plot bearing #3

Figure 15. AVG shaft centreline plot bearing #4

5. Conclusion
In this research, the main cause of excessive vibration on GTG unit is investigated in detail using vibration analysis. Based on the result of analysis and a detailed evaluation of the acquired information from the research, the following will be recommended solution in reducing excessive vibration with an implementation plant for corrective action in the GTG unit machine.

(1) Perform lube oil analysis on bearing lube oil system to observe wear particle since clearance event suspicious was detected on the bearing system.

(2) Propose to check clearance on bearing and seal #2, #3 and #4 and ensure that the bearing and seal clearance are within tolerance prior running the GTG unit.

(3) In situ GTG rotor balancing as part of unbalance resolution on bearing #2 and bearing #3.
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