21, rue d'Artois, F-75008 PARIS
http : //www.cigre.org

A1-106

CIGRE 2018

Importance of Operating Parameters when Assessing the Condition of Machines On-line

J. Letal, H.G. Sedding, M. Sasic
Iris Power - Qualitrol
Canada

O. Husnjak
Veski
Croatia

SUMMARY

Condition based maintenance is a strategy that, if implemented effectively can extend the operational life of rotating machines. The philosophy originates from more traditional, time-based maintenance strategies where activities are scheduled based on the running hours of a machine. Instead, with condition based maintenance the scheduling is optimized such that these activities are only carried out when the condition of the machine indicates certain maintenance or repairs are required. The condition can be determined with a variety of monitoring technologies including those where the data is recorded when the machine is off-line (while not in operation) or on-line (when the machine is in operation). Generally, off-line testing is performed infrequently at larger time intervals compared to on-line monitoring because of the lost revenues when the machines are not in operation and the added expense of disassembly. For on-line measurements periodic or continuous monitoring can be employed and usually depends on the technology and resource availability. Ideally, multiple technologies are incorporated into a common software platform for easier analysis and data correlation. For condition assessment there are a variety of internationally accepted standards for absolute limits as well as acceptable changes to trend. Any trend analysis should only be considered when the machine is operating in similar conditions. This underlies the importance of trending operating parameters along with the monitored data. Additionally, some failure mechanisms can only be recognized with data collected at a variety of operating conditions (e.g. changing load) often done during troubleshooting activities.

This paper will focus on on-line monitoring technologies for large generators driven by steam/gas turbines and hydraulic turbines. Several case studies will be presented in a way to emphasize the importance of operating conditions for both trend analysis and when troubleshooting the machine condition. The technologies will include magnetic flux for rotor shorted turns, stator endwinding vibration, shaft and bearing vibration, and rotor to stator air gap (for hydraulic turbine driven generators). It will be seen that depending on the technology different operating parameters will affect the monitored data differently, some with a significant influence. If the data is collected periodically at predefined intervals there should be a mechanism to manually input these values for future reference. Conversely with continuous monitoring it is often more convenient to use modern communications protocols (e.g. Modbus over TCP/IP) due to the potentially large number of data sets to input the operating parameters at the time of data collection.

KEYWORDS

Rotating Machines, Condition Monitoring, Predictive Maintenance, Magnetic Flux, Endwinding Vibration, Shaft and Bearing Vibration, Air Gap

jletal@qualitrolcorp.com

INTRODUCTION

The life of a rotating machine follows a bathtub curve such that deterioration leading to failure is more likely during the initial stages of operation during a running-in period or much later in time when the machine starts to wear out. Consequently, there are several maintenance strategies for rotating equipment that can be employed [1-3]:

1. Breakdown maintenance where the machine is run to failure.
2. Periodic maintenance where at specific time intervals (e.g. hours of operation) routine maintenance and checks are performed.
3. Condition based maintenance where the maintenance of a machine is determined by the changing condition of the machine during it's normal operating period.

Depending on the situation there are advantages and disadvantages to each strategy. If the machine is low cost and there is no risk of consequential damage to other machines or processes as well as no safety concerns then a breakdown maintenance strategy can effectively be employed. Based on past experience it was eventually found that the normal operating period of the bathtub curve can be extended if certain maintenance activities are performed at specific intervals. Shutting down the machine not only results in lost production, but puts the machine at risk by introducing human error during machine disassembly/reassembly especially if the maintenance is not required at that time. This eventually led to trying to understand the conditions of the machine that are changing prior to failure. A major benefit of monitoring the condition of the machine to effectively plan maintenance periods is that the onset of faults can be identified which can significantly reduce the cost of repair with fewer catastrophic failures, the better utilization of maintenance activities, and the elimination of unnecessary periodic maintenance. It is for these reasons that condition based maintenance is being utilized on high value rotating machines. [1-3]

Condition based maintenance strategies require insight to the condition of the machine. This is accomplished through the installation of sensors to monitor various parameters related to the deterioration of the rotating machine. Depending on the parameter being monitored and the rate of deterioration the test period may be periodic or continuous [1-2]. For slower developing fault conditions, periodic testing can be performed with the machine on-line during normal operation and off-line when the machine is shut down and at a standstill. The major benefits of testing the condition of the machine on-line is that there is no interruption to production and that the test conditions are under normal operating stresses. This offers valuable insight to the condition of the machine which can be trended. These trends should only be compared at similar operating parameters to identify change related to the condition of the machine and not a change to the way the machine is being operated.

In the case of fault conditions that are fast developing or in situations where the machine is at the later stages of wearing out, there are advantages to continuously monitoring the condition on-line to prevent catastrophic failure. Continuous monitoring requires the sensors to be permanently installed and connected to a data acquisition unit. Signal processing algorithms can extract summary numbers so change can be identified in real time. Modern monitoring systems have established communication protocols that can transfer a large quantity of summary numbers using Modbus over TCP/IP that were previously limited for economic and infrastructure reasons, for example one 4-20mA analog signal per summary number.

There seems to be an increasing trend in leading edge monitoring systems that are utilizing the availability of internet connected devices to transfer and store data. The adoption of Industrial Internet of Things, or IIoT continues to rise, but there are difficulties that need to be overcome before this technology can begin to be widely accepted [4]. The main challenge is that installation and maintenance cycles in power plants are on a relatively much larger cycle (in years or decades) than the information technology required to support IIoT that seem to be constantly changing with software and hardware upgrades. As well, monitoring systems generally being used in power plants are purposefully designed to not be capable of data transfer data outside of the power plant network as there is often a major concern in data security [4]. If these obstacles can be overcome there is significant potential with incorporating IIoT technologies in condition monitoring of rotating machines from a data management and diagnostics point of view.

ON-LINE CONDITION MONITORING TECHNOLOGIES

Machines can wear out in a variety of ways and depending on the failure mechanism different conditions of the machine will change representing this degrading condition. As a result, to employ a comprehensive condition based maintenance program a variety of monitoring technologies need to be considered. Depending on the most likely failure mechanism for a particular machine there are several to choose from. For high valued gas/steam turbine driven generators and hydraulic turbine driven generators these can include, among others, several of the following technologies.

Rotor Flux

The integrity of rotor winding turn insulation can be assessed with rotor flux monitoring. Problems with rotors result from exposure of winding copper and insulation to high centrifugal loads and thermal expansion forces, leading to breaks in the winding insulation and to copper cracking and contamination. A magnetic flux sensor installed on a stator tooth in the air gap is sensitive to radial flux density as the rotor passes by the probe. Shorted rotor winding turns results in reduced slot leakage flux and is an indication of insulation failure in the rotor winding.

In conventional flux monitoring, distortion of the radial flux signal is minimal where the air gap flux density curve crosses through zero which is a function of generator load. Because of this, it is required to take multiple readings at various generator load points for maximum sensitivity to shorted turns, but recently it has been demonstrated that accurate detection of rotor winding shorted turns can be obtained with a reduced need to vary the load on the generator with suitable instrumentation and algorithms [5]. Alternatively, smart triggers can be used to automatically capture the flux information during normally occurring load variations. IEEE 1129 [6] further discusses the instrumentation to effectively monitor rotor magnetic flux.

Endwinding Vibration

Stator endwinding vibration has become an important parameter to monitor as copper inside the winding can fatigue and eventually open under load. The failure mechanism is common on large air cooled 2-pole turbo generators since the endwindings, or the section of the winding that extends beyond the core can be quite long due to design constraints. The cantilever effect of this overhang will vibrate due to operational forces between the top and bottom bars of the winding related to the current as well as the interaction between the rotor and the stator at twice line frequency (100/120 Hz) [2, 7]. The vibration levels will be amplified if a natural frequency of the support structure is close to this forcing frequency resulting in resonance. This can be the case on machines built since the year 2000 as there seemed to have been a trend to reduce the endwinding support stiffness, probably to reduce manufacturing cost in the competitive air cooled turbo generator market. This reduction in stiffness has led to more machines with insufficient support of the endwindings causing them to vibrate at much higher amplitudes than normal and leading to a reduced normal lifetime of the machine. A recent insurance study has shown that over a 10 year period in the early 2000s approximately 5% of claims could have been prevented with endwinding vibration monitoring, but what was most concerning was the extent of damage due to this failure process in that roughly 50% of the dollars paid per claim was associated with endwinding vibration [8].

Off-line testing to assess the condition of endwindings includes a bump test which is an attempt to identify the natural frequencies of the support system. These are related to stiffness and mass of the structure and as the stiffness decreases (with age) so do the natural frequencies. It is important to consider that increasing the temperature of the winding will also change the stiffness of the endwindings since the test is done at much lower temperatures than during machine operation [7, 9]. It is for these reasons that IEEE 1665 [10] emphasizes a band of frequencies around twice line frequency for acceptance criteria of endwinding bump test data.

On-line monitoring of endwinding vibration is much more effective because it is a direct measure of how much the windings are vibrating during operation. Unfortunately, there is little guidance into how much vibration is tolerable, but IEEE 1129 does offer some based on historical levels of 2-pole 60 Hz generators in North America, indicating that less than 125 microns peak-to-peak are considered adequate while 250 microns peak-to-peak is cause for concern [6]. Of more importance would be the trend. A significant increase over a short period of time would be more concerning than a relatively high vibration amplitude that is not changing. IEC 60034-32 [11] offers some guidance indicating that an increase of more than 25% in the displacement under the same operating conditions is considered significant. This lends itself to continuous monitoring of the endwinding vibration levels with operating conditions.

Shaft and Bearing Vibration

Vibration monitoring with sensors installed on the bearing housings and shaft journals will monitor the forces generated from a rotating machine. These forces cause vibration and may change in direction with time, change in amplitude with time, result in friction between rotating and stationary components, cause impacts, or cause randomly generated vibration. The vibration amplitudes are proportional to dynamic forces meaning that increased forces will reduce the normal operating period of the machine. In general, problems that cause high vibration amplitudes are rotor unbalance, shaft misalignment, looseness, bearing wear/misalignment, rubbing, and electrical problems. On high value machines or those critical to the process, vibration monitoring is often part of the protection system since high vibration should automatically trip and shut down the machine to prevent catastrophic failure. [3]

Acceptable vibration amplitudes, both on the bearing and shaft journals are very well defined both in terms of absolute criteria and tolerable change in ISO 10816 (for bearing) [12] and 7919 (for shaft journals) [13]. Each series of standard has a part 1 general requirements and subsequent parts for machine type. There is a significant amount of work to combine these standards into a new ISO 20816 [14] series for evaluation of both bearing and shaft journal vibration in the same document.

Air Gap

A convenient extension to shaft and bearing vibration on hydraulic driven turbine generators is to monitor the air gap between the generator stator and rotor, especially since the stator and rotor can become quite flexible due to centrifugal forces, thermal effects, magnetic forces, and mechanical systems failure. Several sensors (at least 4) should be installed equidistant around the stator bore such that the rotor shape can be accurately calculated and the stator shape can be interpolated between the sensors. In general, the larger the stator the more sensors are required for these calculations.

IEEE 1129 [6] introduces this technology for salient pole machines, but to date there is very little guidance for acceptable minimum air gaps and rotor/stator circularity other than erection tolerances [15] and OEM (both of machines and monitoring systems) experience. An effort is being made in a recent working draft standard of IEC 60034-33 [16] to provide air gap and geometry tolerances during operation.

CASE STUDIES

A challenge for an effective condition based maintenance strategy is to combine all the information from a variety technologies of different failure mechanisms that are wearing the machine at different rates to make a maintenance decision that is effective. The following case studies will show how one or more of these technologies can be used to identify a degrading machine condition. It will be apparent that the operating parameters are paramount in assessing change to the condition of the machine.

Endwinding Vibration Correlation with Stator Core Temperature

An interesting endwinding vibrational behaviour was identified on a hydro pump-generating unit (155 MVA), in which three endwinding accelerometers were mounted on the stator endwindings mutually at 120° (sensors marked with EV1-EV3). Figure 1 shows trends of vibrational velocities (mm/s) on all three measurement positions in generating mode for 5 hours in duration and, immediately after that, in pump mode for 7 hours in duration. During operation, the endwindings are subjected to the force proportional to the product of currents through neighbouring windings. If the frequency of the current is 50 Hz, this means that the force frequency between endwindings is 100 Hz. An amplitude of 125 microns peak-peak at 100 Hz is equivalent to 27.7 mm/s rms.

The upper diagram on Figure 1 shows vibration trends of all three measurement positions (EV1 in red, EV2 in blue, and EV3 in green) at the 10x harmonic of rotational frequency which is 100 Hz in steady state operation in generator mode. The lower diagram shows: rotational speed (red), active power (blue), generator stator current – phase R (green) and stator core temperature (magenta). Besides the expected change in vibration amplitudes when the regime changes causing the generator stator current to change which changes the amplitudes of forces between the windings there were three rather large, additional changes to the endwinding vibration amplitudes. Those were the rather fast change in EV1 from 20 mm/s to 5 mm/s immediately after the generator start mode and the abrupt change in EV2 and EV3 vibrations on unit run-down. The EV1 vibration reduction from 20 to 5 mm/s occurred at constant generator current, that is, at constant force between end-windings. The only significant variable that changed was a rapid increase in the stator core temperature.

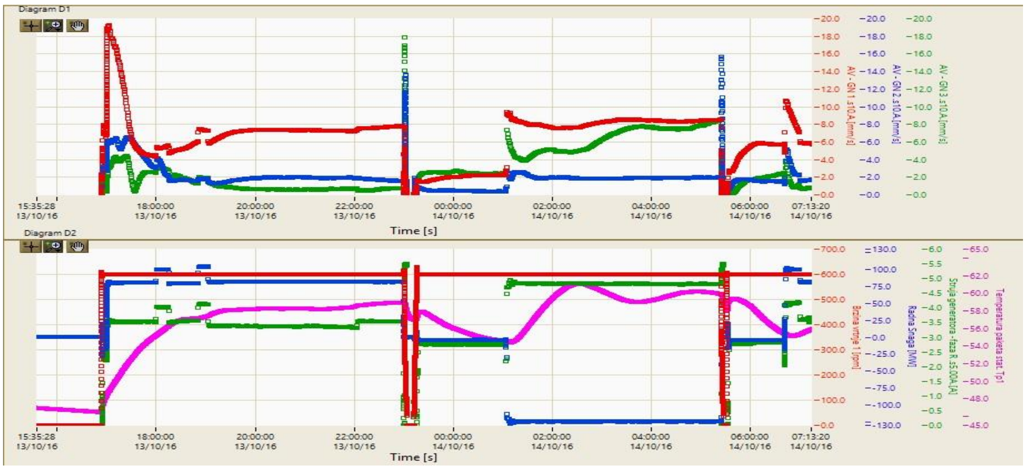

Figure 1 – Top Plot: Endwinding Vibration 10x Rotational Speed Amplitude (red, blue, green)
Bottom Plot: Rotational Speed (red), Active Power (blue), Stator Current (green), Stator Core Temperature (magenta)

Such sudden changes in the vibration amplitudes in steady state conditions (unchanged voltage and current) were a consequence of changes in the stiffness of the endwinding support structure when the core was experiencing thermal dilatation during the warm-up process. Figure 2 shows trends of the vibrational velocity at EV1 during pump start when the core was relatively cold (45°C – top plot) and again when it was warmed up (65°C – bottom plot). The pump start was conducted as a motor with constant stator current of 0.3 kA and uniformly changing frequency from 0 to 52.5 Hz. At that frequency (5x harmonic of rotational frequency) the unit runs at 630 rpm and the frequency of the force between windings is 105 Hz.

For rotational speed between 300 and 630 rpm there were many endwinding resonant frequencies. In the cold condition (top plot) there was a prominent resonant frequency near 600 rpm which was the main contributor to the relatively high vibration amplitudes at EV1 when the unit starts from a cold state. In the warm condition (bottom plot) this resonant frequency moved to higher frequencies due to thermally induced changes in stiffness and consequently the vibrations reduced rapidly when the core temperature increased.

The diagrams in Figure 2 show that for rotational speeds above 300 rpm (forces between windings with frequencies above 50 Hz) there were several resonant frequencies whose values change with stator core temperatures. This resulted in relatively large vibrations when the unit runs down and when the electrical braking system was applied.

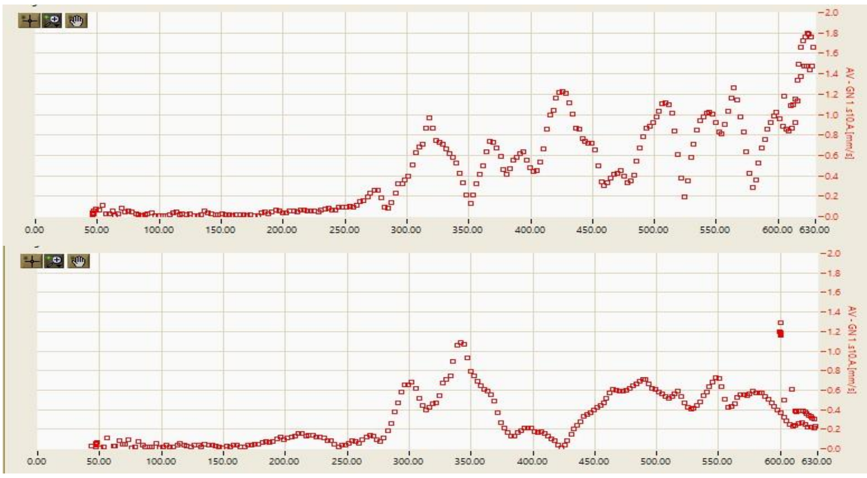

Figure 2 – Endwinding Vibration 10x Amplitude versus Rotational Speed
Top Plot: Stator Core Temperature 45ºC
Bottom Plot: Stator Core Temperature 65ºC

Figure 3 shows endwinding vibration amplitudes at all three positions (top plot) during a unit coast down. The bottom plot shows the phase R current and magnetic field. The electrical braking is automatically turned on at 450 rpm increasing the generator current to 5.5 kA and magnetic field to 0.18T.

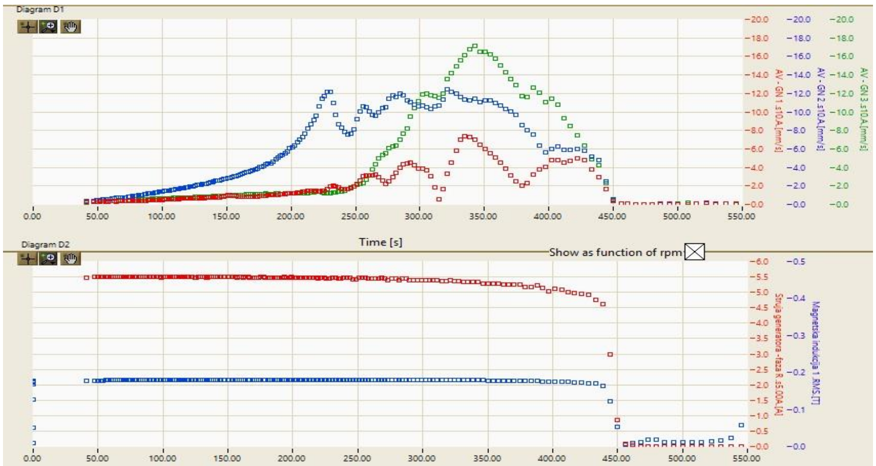

Figure 3 – Top Plot: Endwinding Vibration 10x Amplitude (red, blue, green) versus Rotational Speed
Bottom Plot: Stator Current 5x Amplitude (red) and Magnetic Flux (blue) versus Rotational Speed

With this vibration amplitude correlation to operating conditions it was recommended to turn on electrical braking at lower speeds, e.g. 250 – 300 rpm to avoid resonant frequency vibrations on coast-down which can occur multiple times a day during pumping operation. This effort will help to extend the normal operating period of the generator.

Air Gap Changes with Speed and Comparison with a Similar Unit

Air Gap data was recorded during coast downs of 2 similar hydro generators (158 MVA). The difference between the maximum and minimum air gap should not change during speed changes if the rotor is extending / shrinking uniformly due to the changing centrifugal and magnetic forces (no loose rim sections or similar). This ideal condition can be seen during the Unit A coast down, maximum (green) and minimum (red) in Figure 4. The vibration was extracted from the signals so only the changes in rotor shape can be observed. The values for each pole were obtained so the minimum, average, and maximum trended values can be obtained for each air gap sensor. These values for one sensor are displayed in Figure 4.

Figure 4 - Unit A Coast Down
Top Plot: Air Gap Minimum (red), Air Gap Average (blue), Air Gap (green)
Bottom Plot: Rotational Speed

Comparing this to a coast down recorded on Unit B in Figure 5 it can be seen that the difference between the maximum and minmum air gap is changing with speed indicating that the geometry of the rotor is changing.

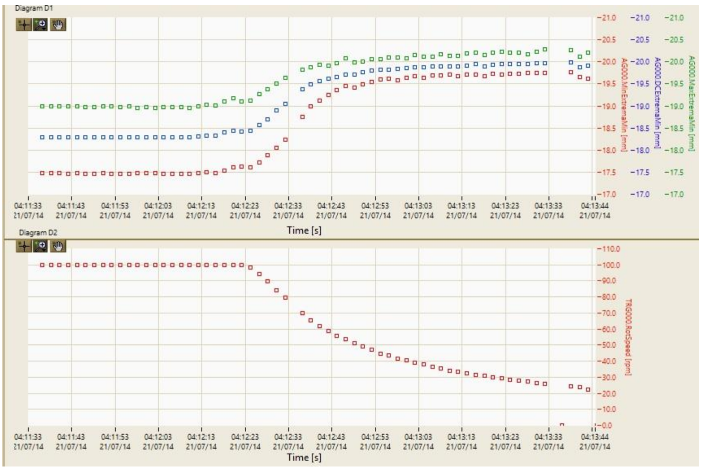

Figure 5 - Unit B Coast Down
Top Plot: Air Gap Minimum (red), Air Gap Average (blue), Air Gap (green)
Bottom Plot: Rotational Speed

Further evidence of rotor rim deformation can be seen when displaying the rotor shape calculated by simultaneously recording air gap data from 4 air gap sensors installed 90º apart. Figure 6 compares the rotor shape at different speeds/centrifugal forces. If the geometry of the rotor is good, the shape should be circular because centrifugal forces are radially equal.

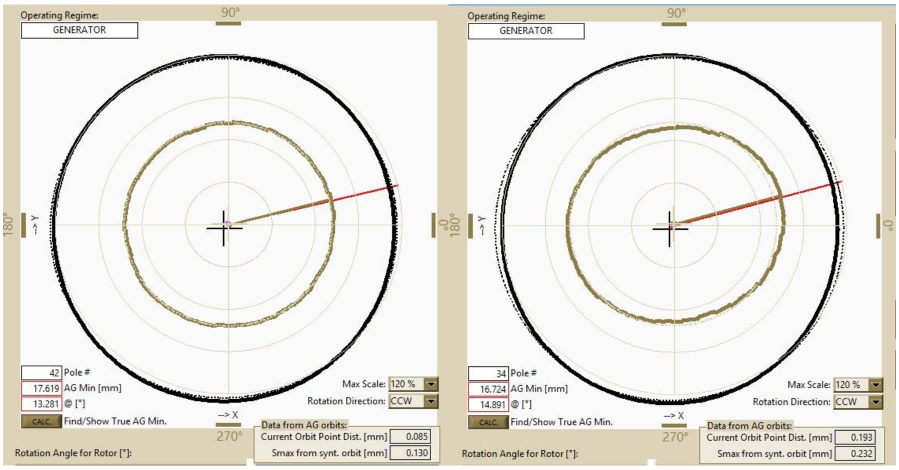

Figure 6 – Unit B Stator and Rotor Geometry
Left side: 60 rpm
Right side: 100 rpm

This oval shape at higher speeds/centrifugal forces on Unit B was the result of uneven stiffness in two directions and likely due to the loss of a significant portion of the shrink fit. The ability to record this information during coast downs and comparing with similar machines is valuable when planning maintenance activities.

Increasing 1x Vibration with Load

Shaft and bearing vibration measurements were performed on a newly refurbished hydraulic driven turbine generator unit (333 rpm, 144 MW, 160 MVA) using a portable vibration system. Vibration measurements were recorded in the following operating regimes:

1. run-up to nominal speed followed by automatic excitation (field flash)
2. excitation turned off and rotation on nominal rotational speed in mechanical rotation (free run) for about 10 minutes
3. excitation turned on and rotation without load for about 10 minutes
4. synchronization and load increase to 25 MW; rotation for about 10 minutes
5. load increase to 50 MW; rotation for about 10 minutes
6. load increase to 75 MW; rotation for about 10 minutes
7. load increase to 100 MW; rotation for about 10 minutes
8. load increase to 120 MW; rotation for about 10 minutes
9. load increase to 144 MW; rotation for about 10 minutes
10. load decrease to 120 MW; rotation for about 5.5 hours on constant load (120 MW)
11. load decrease and excitation turned off;
12. overspeed to 485 rpm and free run-down (without braking)

Figure 7 shows amplitude trends of the first harmonic of the rotational speed for shaft relative vibrations through these operating regimes.

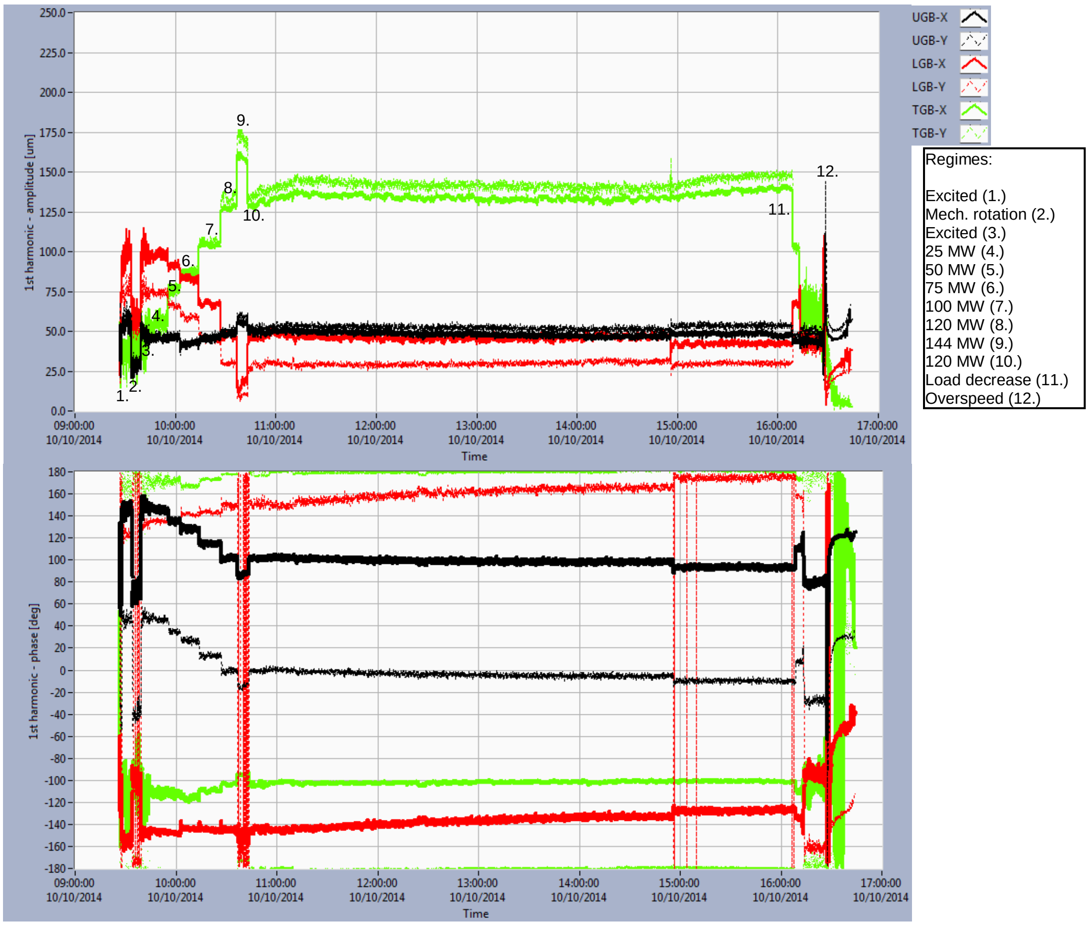

Figure 7 – Trends 1x Rotational Speed of Shaft Relative Vibration on Upper Guide Bearing (black), Lower Guide Bearing (red), Turbine Guide Bearing (green) Amplitude and Phase

The relative vibration amplitudes at 1x at the turbine and lower guide bearings depended significantly on load while not as significantly at the upper guide bearing. The peak amplitude increased from ~50 µm in mechanical rotation (nominal speed) to ~175 µm at 144 MW load. There was practically no influence on the turbine guide bearing vibrations when the excitation was turned on. The TGB phases changed very little during the load increase. The maximum change from the mechanical rotation to maximum load was ~10-15°. Since the first harmonic vibration amplitude was changing and there were almost no changes in the vibration phases it can be concluded that the forces on the turbine rotor changed significantly during the load increase and that these forces changed in the same direction during the whole process.

On the lower guide bearing the vibrations changed significantly when the excitation was turned on. The vibration amplitudes decreased upon load increase. In mechanical rotation, the amplitudes were ~50 µm, on excitation ~100 µm and on maximum load they were very small values (below 25 µm). The phases changed sharply when the excitation was turned on (~70° when compared to the mechanical rotation) and changed very little during the load increase. The maximum change from rotation with excitation to maximum load was ~10-15° (~10° in the X, and ~15° in the Y direction). From the significant changes in peak amplitudes on load increase, it can be concluded that at the lower guide bearing the forces changed significantly with load, but that the direction of the force was practically the same in all working regimes.

On the upper guide bearing the situation was the opposite. Peak vibration amplitudes were practically unchanged during the experiment (around ~50 µm) but the phases changed significantly – from rotation with excitation (no load) to maximum load by ~70°. So, in order to draw an adequate conclusion about the machine behaviour, one should always check amplitudes and phases as this example shows that the invariance of amplitudes does not necessarily mean that the forces acting on the bearings do not change.

Due to the increasing vibrations with load, the load limit was set to 120 MW since there is a high risk of bearing damage (bearing segments and seals etc). At 120 MW, vibration peak amplitudes reached ~170 µm. Due to these vibrations the oil film thickness between the shaft and bearing segments was reduced to such a value that threatened the dynamic stability of the unit. The vibration response of another unit of the same design was much lower, further indicating that there was an issue, particularly at the turbine guide bearing. The turbine rotor was asymmetrical possibly due to:

- blade manufacturing faults (geometry) – that is, blade curvature for one (or more) blades
- position faults (welding) for one (or more) blades

CONCLUSION

Condition monitoring has been effectively employed in condition based maintenance strategies to assist with planning and scheduling maintenance activities. Maintenance and repair costs are generally less if degrading machine conditions are identified and corrected earlier. Ideal monitoring systems employ a common hardware and software platform to combine multiple technologies. This allows for the ability to identify many different degrading condition and can also increase confidence in the decision if multiple technologies are indicating the same condition. A challenge as these monitoring systems continue to evolve is keeping up with the rapidly changing Information Technology infrastructure. Conversely the potential in this ever-evolving infrastructure is very appealing from a data management and diagnostic point of view. Regardless, these systems must also contextualize the trend data with the operating conditions of the machine. In some cases, this correlation helps to troubleshoot specific machine conditions that limit the normal operating period as well as identify whether an increasing, long term trend is related to a changing machine condition or the way the machine is operating.

BIBLIOGRAPHY

[1] P. Tavner, L. Ran, J. Penman, H. Sedding, *Condition Monitoring of Rotating Electrical Machines*, London, United Kingdom: The Institution of Engineering and Technology, 2008.

[2] G.C. Stone, I. Culbert, E. A. Boulter and H. Dhirani, *Electrical Insulation for Rotating Machines*, 2nd ed. Hoboken, NY, USA: Wiley-IEEE Press, 2014.

[3] S.S. Rao, *Mechanical Vibrations*, 5th ed. Upper Saddle River, NJ, USA: Prentice Hall, 2011.

[4] B. Hougland, “The State of IIoT in 2017,” *Design Engineering*, vol. 63, no. 5, pp. 34-36, October 2017.

[5] M. Sasic, B. Lloyd, A. Elez, *Finite Element Analysis of Turbine Generator Rotor Winding Shorted Turns*, IEEE Transactions on Energy Conversion Vol. 27, No. 4, December 2012.

[6] *IEEE Guide for Online Monitoring of Large Synchronous Generators (10 MVA and Above)*, IEEE Standard 1129, 2012.

[7] H.O. Ponce, B. Gott and G. Stone, “Generator Stator Endwinding Vibration Guide: Tutorial,” EPRI, Palo Alto, CA, USA, Report 1021774, 2011.

[8] S. Purushothaman, “Optimum Condition Monitoring Based on Loss Data History”, presented at EPRI On-Line Monitoring Workshop, Chicago, IL, USA, 2013.

[9] M. Sasic, H. Jiang and G. C. Stone, "Requirements for fiber optic sensors for stator endwinding vibration monitoring," *2012 IEEE International Conference on Condition Monitoring and Diagnosis*, Bali, 2012, pp. 118-121.

[10] *IEEE Guide for the Rewind of Synchronous Generators, 50 Hz and 60 Hz, Rated 1 MVA and Above*, IEEE Standard 1665, 2009.

[11] *Measurement of stator end-winding vibration at form-wound windings*, IEC TS 60034-32, 2016.

[12] *Mechanical vibration – Evaluation of machine vibration by measurements on non-rotating parts – Part 5: Machine sets in hydraulic power generating and pumping plants*, ISO 10816-5, 2000.

[13] *Mechanical vibration – Evaluation of machine vibration by measurements on rotating shafts – Part 5: Machine sets in hydraulic power generating and pumping plants*, ISO 7919-5, 2005.
[14] *Mechanical vibration* -- *Measurement and evaluation of machine vibration* -- *Part 1: General guidelines*, ISO 20816-1, 2016.
[15] *Hydroelectric Turbine Generator Units* - *Guide for Erection Tolerances and Shaft System Alignment, Part V*, Canadian Electrical Association (CEA), 1989, rev. 1998
[16] *Specific technical requirements for hydro generators*, IEC 60034-32/WD Edition 4.0, 2018