MONITORING OF STATOR ENDWINDING VIBRATION ON MOTORS AND GENERATORS

Paper No. PCIC Europe (will be assigned)

John Letal
Iris Power
3110 American Drive,
Mississauga ON L4V 1T2
Canada

Mladen Sasic
Iris Power
3110 American Drive,
Mississauga ON L4V 1T2
Canada

Jason Elyes
Portlands Energy Centre
470 Unwin Ave,
Toronto, ON M4M 3M9
Canada

Abstract - High speed generator and motor failures due to stator endwinding vibration have become more prominent in recent years due to extensive repair costs as well as lost production that are often associated with such failures. The vibration is a response to the electromagnetic forces created by the current in the stator windings. Excessive vibration levels can be attributed to loosening of the support system or mechanical resonance where the structural natural frequencies coincide with the frequency of the electromagnetic force. Insulation abrading or failure by copper fatigue can result if left for an extended period of time. IEC 60034-32 and IEEE 1129 describe methods to monitor endwinding vibration on-line with fiber optic sensors in a way that the onset of looseness can be identified, and the support structure repaired in a timely manner. A 288 MVA, 21 kV steam turbine driven air-cooled generator with endwinding vibration data prior to and after a system event that led to the loosening of the support structure was trended on-line until suitable repairs could be made and the vibration levels returned to normal.

Index Terms — Stator Winding, Two-Pole Machines, Endwinding Vibration, Motor, Turbine Generator.

I. INTRODUCTION

The primary function of a stator endwinding is to allow for the safe electrical connection between bars in series and to other parallels. These connections must be made away from the stator core to prevent insulation failure at the connection points. On higher voltage machines the required creepage distance between the core and the connections can become quite long [1]. There are two forces contributing to endwinding vibration:

1) the electromagnetic force resulting from stator winding conductors carrying the current and

2) the force resulting from rotation of machine at nominal speed. The first force is proportional to the square of stator current and its frequency is two times higher than system frequency, 100 Hz in 50 Hz systems, and 120 Hz in 60 Hz systems. The second force is occurring at system frequency for two pole machines and half of the system frequency for four pole machines.

These forces can be measured in three directions. Considering the end view of stator these are normally specified as radial, tangential (or circumferential), and axial. For the electromagnetic force the directions of most concern are radial and tangential. This is because the force is generated by two parallel current carrying conductors, i.e. the force between the top and bottom bar (radial) and between two adjacent bars (tangential) [2]. This force in the axial direction is typically negligible. The turning speed force is present in the three directions, and typically more significant in the radial and tangential directions.

II. STATOR ENDWINDING OFF-LINE IMPACT TEST

There are two off-line impact test procedures that can be used to establish the natural structural characteristics of a stator endwinding structure: Driving Point and Modal Analysis [3].

A. Driving Point Impact Test

In a Driving Point impact test the force hammer and the accelerometer are at the same measurement location and the result is a measured response at the excitation point known as a frequency response function (FRF). This transfer function is expressed in the frequency domain with amplitude and phase. The phase will be between 0 and 180 degrees or 180 and 360 degrees and be ~90 degrees at a natural frequency. As a driving frequency approaches an undamped natural frequency, 1) the magnitude approaches a maximum and; 2) a phase shift crosses through 90° [4]. These two observations identify a natural frequency since the force response is at a maximum and the phase between the force and response is transitioning from in-phase to out-of-phase (or vice versa) at a natural frequency. These can be determined with impact testing to identify natural frequencies as shown in Fig. 1.

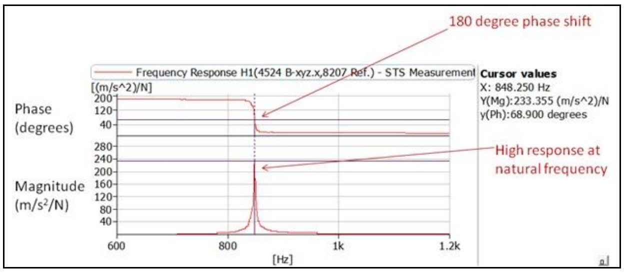

Fig. 1 – Natural Frequency Identified with Impact Test

The critical bands for a turbine generator are around rotational speed frequency and twice line frequency. Therefore, applied forces in the endwinding structure of a 2-pole, 60 Hz generator are at rotor rotational frequency, 60 Hz and at load current electromagnetic forces, 120 Hz. The concept of critical band refers to the risk of vibration amplification when the structural natural frequencies are close to the forcing frequencies. This condition is resonance. In service, the natural frequencies may drift in the bands due to temperature [5], aging and other variable factors. Thus, an acceptance band should be defined with these factors in mind. The acceptance criteria are based on the magnitude of the acceleration over force through the critical excitation bands of, for example -10 Hz and + 20 Hz of the fundamental excitation frequencies (60 and 120 Hz) [6]. If the response is greater than 0.44 (m/s2)/N the endwinding support structure may be loose [1].

B. *Modal Analysis Impact Test*

Modal Analysis consists of measuring motion at various locations of a structure when it is excited by some driving force at the same location. That is, the response is measured at several locations on a structure for the same force location. The pattern of motion usually takes certain shapes which are related to the natural frequencies or natural motion tendencies of the structure. This provides a description of the structural characteristics through curve fitting techniques to generate a shape table that closely represents the dynamics of a structure.

Prior to collecting data for Modal Analysis, Reciprocity has to be checked to establish if the structure is homogenous so useful modal data can be collected. If the force hammer is at location 1 and the accelerometer response is collected at location 2, the profile of the FRF must be the same as when the hammer is at location 2 and the response is collected at location 1 as shown in Fig. 2.

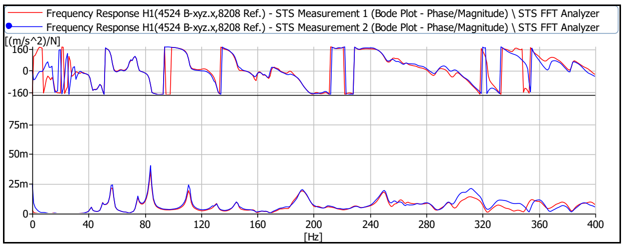

Fig. 2 – Reciprocity Data to Validate Modal Data

An endwinding structure can be modeled with a circular ring. When the structure takes certain shapes at similar frequencies to a force, the resonant condition amplifies the vibration on an endwinding. For a 2-pole machine the shape for twice supply frequency deflection is oval. This shape is not the only mode that can be excited by forces within the rotating machine. Other modes such as cantilever modes where the whole structure is moving up and down or breathing modes where the structure is expanding and shrinking could also become resonant if forces act on the winding in the critical directions and at the critical frequencies [2]. However, the oval mode shape in Fig. 3, for 2-pole machines is the most critical for vibration analysis of the stator because it naturally gets driven by the electromagnetic forces. The acceptance criteria can use similar bands as the Driving Point test, for example -10 Hz and + 20 Hz of the fundamental excitation frequencies (60 and 120 Hz) [6].

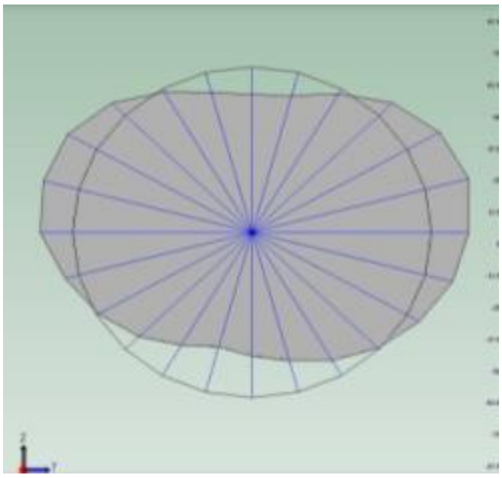
Fig. 3 – Critical Mode Shape for 2-pole Machine

III. STATOR ENDWINDING ON-LINE VIBRATION MONITORING

Vibration sensors can be used to detect movement directly and are often mounted on the shafts and bearing housings of rotating machines to assess the condition of the machine related to its rotor dynamics. The same sensors cannot be used on stator endwindings, particularly near the coil ends where the vibration amplitudes are the highest because metallic materials near high currents will heat up due to losses induced by magnetic fields. Metallic sensors may also compromise the electrical clearances of the endwinding to ground and can result in partial discharge. However, it is possible to monitor the movement of endwinding coil ends with the non-metallic fiber optic accelerometers though.

Since the operating forces of a generator stator endwinding are primarily at rotor rotational and electromagnetic frequencies it is useful to view the on-line endwinding vibration data in the frequency domain and trending the amplitudes of these frequencies. As with any on-line monitoring system it is important to correlate the data with the operating conditions of the machine. For stator endwinding vibration this includes: 1) stator current since the electromagnetic forces are proportional to current squared, 2) stator winding temperature since the natural frequencies of the structure may drift due to temperature sometimes causing the vibration response to become resonant as the windings heat or cool, 3) active power and, 4) reactive power [3,5]. Although there is no consensus regarding endwinding vibration acceptance limits, any displacement higher than 250 µm p-p is considered concerning [1-2,7-8].

IV. CASE STUDY – 288 MVA, 21 KV, 2-POLE, AIR-COOLED GENERATOR

In June 2012 off-line impact testing was performed after a modification to the endwinding support structure on the exciter end of a 288 MVA, 21 kV, 2-pole air cooled generator. The results showed no critical natural frequencies near the critical bands. The FRF results of the 11:00 top and bottom jumper connections to the circuit ring bus are shown in Fig. 4. The 5 lines represent all 3 directions on the top jumper and in the radial and axial directions on the bottom jumper. The tangential direction on the bottom jumper was not accessible.

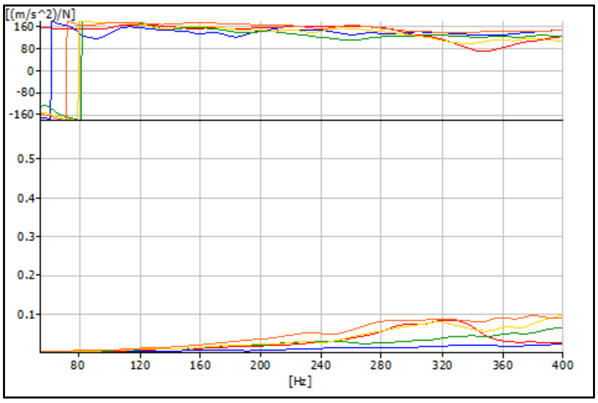

Fig. 4 – 2012 Off-line Impact Test at 11:00 Jumper Connections

Fiber optic Endwinding Vibration Accelerometers (EVAs) were installed to monitor the vibration of the endwinding support structure and windings during operation. The sensors were installed at the winding location where the greatest response was measured with an impact test. After 2 years of operation, there was a significant increase in vibration amplitudes resulting in overall displacement levels as high as 350 µm p-p on the sensor installed near the 11:00 jumper connections, see Fig. 5.

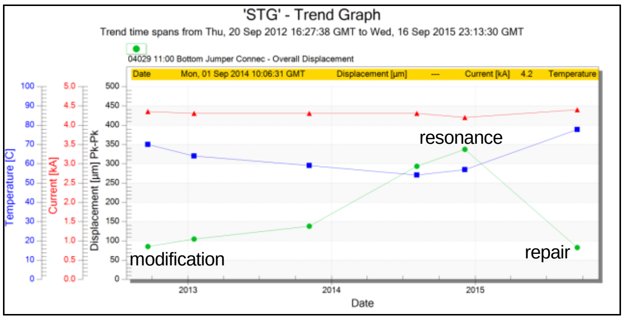

Fig. 5 – On-line Endwinding Vibration Trend of 11:00 Bottom Jumper Connection

The dominant vibration peak was at 120Hz and the highest amplitude recorded was 315 µm p-p, see Fig. 6. Although there is no consensus regarding endwinding vibration acceptance limits, any displacement higher than 250 µm p-p is considered concerning [1-2,7-8]. In this case study, this amplitude is particularly concerning because it increased from 66 µm p-p measured in September 2012 since a significant increase is more than 25% under the same operating conditions [3].

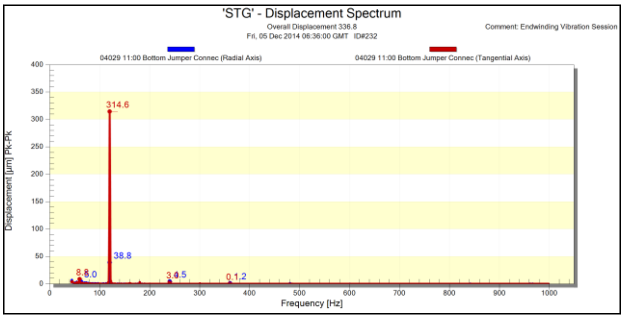

Fig. 6 – Displacement Spectrum of 11:00 Bottom Jumper Connection

The vibratory condition prompted a visual inspection of the endwinding support structure during the next outage that revealed loose support blocking and dusting between the jumper connection and radial support ring in 5 locations, see Fig. 7.


Fig. 7 – Loose blocking near 11:00 Jumper Connection [8]

The original outage plan did not include inspection of this generator, meaning the loose blocking would not have been identified without the endwinding vibration monitoring system. In addition to the visual inspection, an impact test showed that the on-line vibration levels increased due to resonance near 120 Hz, Fig. 5. The significant reduction in stiffness from the loose support block reduced and separated the natural frequencies resulting in a resonance condition during operation. Since the support structure was loose (non-homogeneous) the reciprocity of the structure was assumed to be poor so a Modal Analysis was not performed. Regardless of the mode shape being excited during operation there was clear and significant amplification of the on-line vibration data due to resonance at 120 Hz.

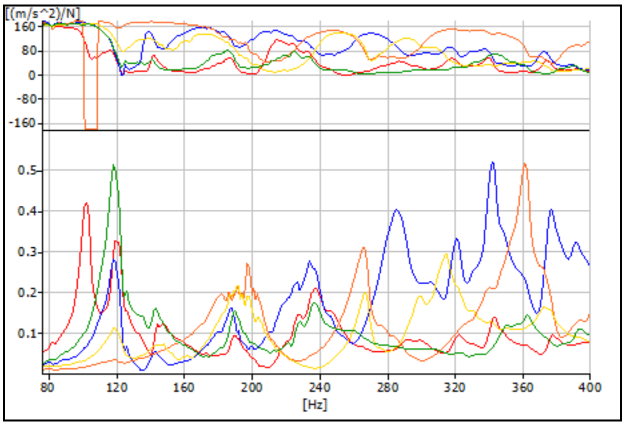

Fig. 8 - 2014 Off-line Impact Test at 11:00 Jumper Connections before repair

The 2014 outage plan did not allow for enough time for a permanent repair so the endwinding vibration levels were monitored closely until the loose blocking could be addressed in 2015. Impact testing, Fig. 9 and on-line vibration data, Fig. 5 show that after the blocks were replaced and tightened the condition was similar to that of 2012.

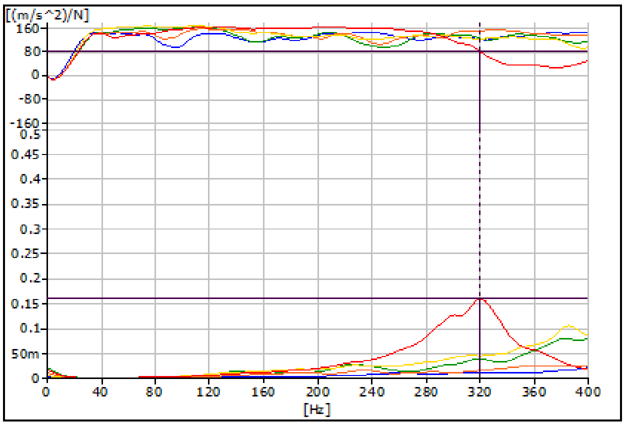

Fig. 9 - 2015 Off-line Impact Test at 11:00 Jumper Connections after repairs

In the fall turnaround outage of 2018 additional bump testing was performed and no issues were found. This confirms that the repair was still in good condition and that the online monitoring system was performing well.

V. CONCLUSIONS

Stator endwinding vibration is leading to premature failure of large 2-pole rotating machines, but if a degrading condition can be identified ahead of time then corrective actions can be considered to extend the lifetime of the windings. Visual inspections at regular maintenance intervals is the most reliable way to identify loose endwindings and/or support structures. Off-line bump testing will identify natural frequencies to determine the potential for resonance during operation. If the testing is done with thorough record keeping it becomes trendable which can help identify a degrading condition to the support structure. Unfortunately, visual inspections and bump testing can only be done during a turnaround outage with some disassembly to the machine. As a result, on-line monitoring of the vibration directly is the most effective way to identify the onset of looseness so that countermeasures can be taken ahead of machine failure [9]. Without this tool the structural looseness identified in this case study would not have been identified with the same advance notice to plan effective repairs.

VI. REFERENCES

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VI. VITA

John Letal is a Rotating Machines Engineer at Iris Power responsible for supporting mechanical monitoring initiatives. Prior to Iris, he spent 7 years with Siemens TurboCare in field service troubleshooting large rotating equipment using vibration analysis. John holds a Master’s degree in Mechanical Engineering from the University of Toronto and is registered as a Professional Engineer in Canada. jletal@qualitrolcorp.com

Mladen Sasic graduated from the University of Sarajevo, Yugoslavia in 1987 with a BSc degree in Electrical Engineering. He has been a rotating machines engineer since 1995 and has contributed to many technical papers and standardization work groups.
msasic@irispower.com

Jason Eyles graduated from Carleton University in 2008 with a Bacherlor’s Degree in Mechanical Engineering. Joined Ontario Power Generation in 2009 and worked at Lambton Generating Station as a System Engineer on the Steam Turbines, Condenser and Feedwater systems. Moved to a central head office group supporting machine dynamics and vibration monitoring in late 2012 until 2014. He joined TransCanada in 2014 as a Mechanical Engineer providing support to the Canadian power facilities specializing in Steam Turbines. Seconded to Portlands Energy Centre as the Plant Engineer in the Spring of 2016 supporting all facets of the plant and transitioned to his current role as Maintenance Manager at Portlands Energy Centre in the fall of 2018. Jason@portlandsenergycentre.com