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Endwinding Vibration Monitoring of Air Cooled Turbine Generators – The Case Study Continues

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SUMMARY

The endwinding region of large turbine generator stator windings is one of the most complex parts of a generator to design, manufacture and assemble. During normal operation, the endwindings are subject to high electrical forces at twice power frequency due to the currents in the stator bars, as well as mechanical forces transmitted via the core and bearings at rotational speed. During power system transients, the forces in the endwinding can be 100 times higher than in normal operation. The design of the endwinding must account for thermally induced axial expansion and contraction as the generator is loaded and unloaded. Due to the presence of high magnetic and electric fields, metallic components to restrain the movement of the stator bars caused by these forces are normally avoided.

Visual inspections are often considered the main tool to detect evidence of excessive movement of stator endwindings. The loosening of components results in dusting as they rub against each other causing the insulation to fret. If the excessive movement is related to mechanical resonance off-line “bump” testing can be performed to determine the natural frequency of the structure to identify the potential for resonance during operation. Resonance is when the natural frequencies of the structure coincide with the operational forcing frequencies of the generator.

The primary operational force is often referred to as 2f and is proportional to the square of the current. This force causes an oscillating response during generator operation that is at twice the power frequency or 120 Hz for a 60 Hz machine in North America. Additionally, if the forces from the once per turn revolution of the rotor due to residual unbalance can be transmitted to the stator windings those machines will experience endwinding vibration at this frequency as well, commonly referred to as 1x. For 2-pole machines this force is at 60 Hz in North America. If there is a natural frequency that influences either of these operational frequencies or harmonics then the vibration response on-line will amplify, sometimes catastrophically if the copper conductors vibrate enough and result in high cycle fatigue. Prior to that, components that are tied together will tend to loosen if vibrating excessively and if

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identified early, can be monitored for a period of time until repairs can be scheduled and performed.

In this paper off- line test results and on-line monitoring data from a 288 MVA, 21 kV air cooled generator will be described. Off-line impact test data led to the installation of fiber optic endwinding vibration sensors. Continuous on-line monitoring indicated an increase in vibration levels and visual inspection of the endwinding confirmed loosening of the endwinding support structure. This paper was originally presented at the CIGRE Study Committee A1 Colloquium in 2017 [1] and since then the repair efforts made to control the vibration have held tight but the system has loosened further in on the support structure. This emphasizes the importance of monitoring this condition so further repair efforts can be planned efficiently as this case study continues. This will serve as an update to the on-line vibration after the original repairs as well as the visual inspections made after the support structure loosened further in.

KEYWORDS

Rotating machines, stator winding, on-line monitoring, endwinding vibration

1. INTRODUCTION

Electrical connections between bars in series and to other parallels can be safely made in the stator endwinding. These connections must be made away from the stator core to prevent insulation failure at the connection points. The endwinding overhang can become quite long on higher voltage machines due to minimum creepage distance requirements and on higher speed machines for geometric reasons, e.g. 2 m or longer is not uncommon [2,3]. Long unsupported lengths of endwinding bars become susceptible to excessive motion.

There are two primary 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 [3] and its frequency is twice the system frequency, 100 Hz in 50 Hz systems, and 120 Hz in 60 Hz systems. The second force is occurring at the rotational speed of the machine which is at system frequency for two pole machines and half 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) [3]. An additional force in a stator endwinding is due to thermal expansion resulting in growth of the bars in the axial direction. Endwinding support design is a balance between stiffness to prevent movement from normal operating forces and flexibility to allow for growth due to thermal expansion. This need for opposite characteristics of stiffness and flexibility is a challenge when designing endwinding support systems.

To accommodate for these forces during operation each bar is often lashed to a support ring made of fiberglass and epoxy. The hoop strength of the support ring prevents movement in the radial direction. Insulating blocks placed between adjacent bars prevent movement in the circumferential direction. Depending on the length of the endwinding one or more rows of blocking may be present [2]. The organic materials that are predominately used to support the endwindings are susceptible to aging and can loosen over time.

Excessive motion in the stator endwinding structure is most likely to occur in form wound two-pole, high voltage machines since the endwindings are relatively long. The windings pivot at the slot exit because they are held tight in the slot which creates a cantilever effect in the endwinding area. Proper support is required to prevent movement from this effect where the vibration forces can lead to fatigue cracking of the insulation and even the conductors at the core end or at the connections. Stator winding deterioration may be the result of poor design or aging of endwinding support components. Both can be contributing factors to resonant conditions which is when the endwinding natural frequency is similar to the frequency of electromagnetic forces (twice the system frequency). In the case of resonant conditions occurring in operation, normally occurring vibration will be significantly amplified. Frequent stops, system disturbances, and aging can also affect the condition of endwinding and result in a change of winding natural frequency.

Different methods can be used to assess the condition of the endwinding, such as impact (bump) testing and visual inspection, both requiring partial disassembly of machine [4]. The purpose of the impact test is to find the natural frequency of the stator winding and identify the loosest components in the endwinding. These tests are periodic, and cannot provide timely information about how fast the insulation is degrading. With this in mind and given that there is not always access to the stator windings for inspection, on-line monitoring of stator windings has become the best way to detect changes in the endwinding support structure.

Electromechanical vibration monitoring is well developed and it is possible to monitor machine condition with vibration data collected on the shaft and bearing housings. This is most effective for identifying issues related to the rotor dynamics of the machine (unbalance, misalignment, bearing looseness) with metallic piezoelectric accelerometers and non-contact probes measuring eddy currents proportional to shaft displacement. These same methods cannot be applied on stator endwindings in particular near the coil ends where the vibration amplitudes are the highest because metallic materials in close proximity to 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. With the development of non-metallic fiber optic accelerometers though, it is possible to monitor the movement of endwinding coil ends directly [5].

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° [6]. 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.

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 [6], 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) [8]. If the response is greater than 0.44 (m/s2)/N the endwinding support structure may be loose [2].

2. 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 Figure 1. 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.

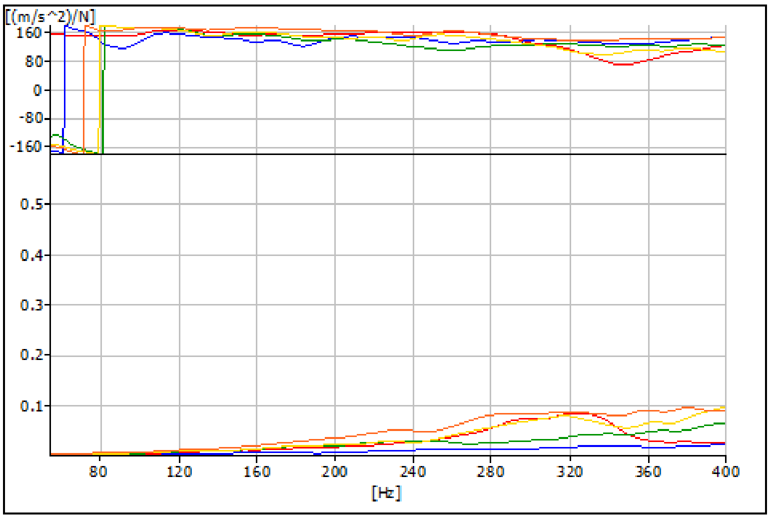

Figure 1 – 2012 Off-line Impact Test at 11:00 Jumper Connections

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 Figure 2.

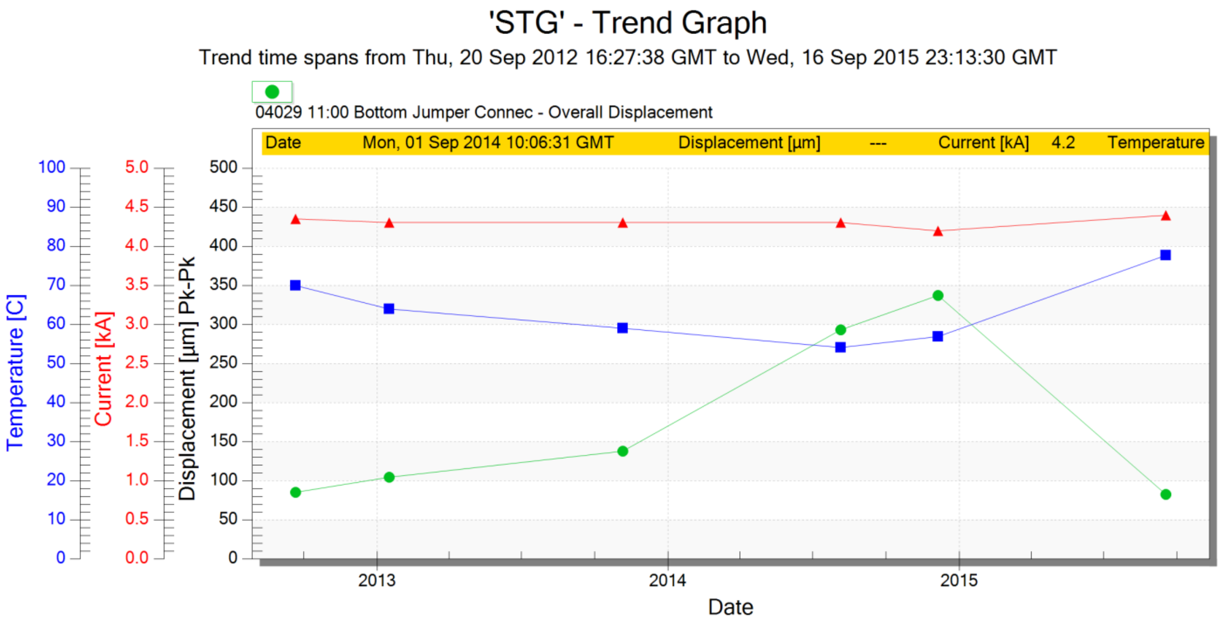

Figure 2 – On-line Endwinding Vibration Trend of 11:00 Bottom Jumper Connection

The dominant vibration peak was at 120Hz and the highest amplitude recorded was 315 µm p-p, see Figure 3. Although there is no consensus regarding endwinding vibration acceptance limits, any displacement higher than 250 µm p-p is considered concerning [2-3, 9]. In this case study, this amplitude is particularly concerning because it increased from 66 µm p-p measured in September 2012.

A recent EPRI study using a modeled approach suggested much higher action levels for endwinding vibration ranging from 500 to 1200 µm peak to peak [10]. The hypothesized levels were separated into 4 categories based on the stiffness of the endwinding support with higher acceptable levels on more flexible designs which is a reasonable approach. The resulting action levels though, are not consistent with this case study nor the practical approach used to develop IEEE 1129 which is largely based on experience from past years with 60 Hz generators in North America [9].

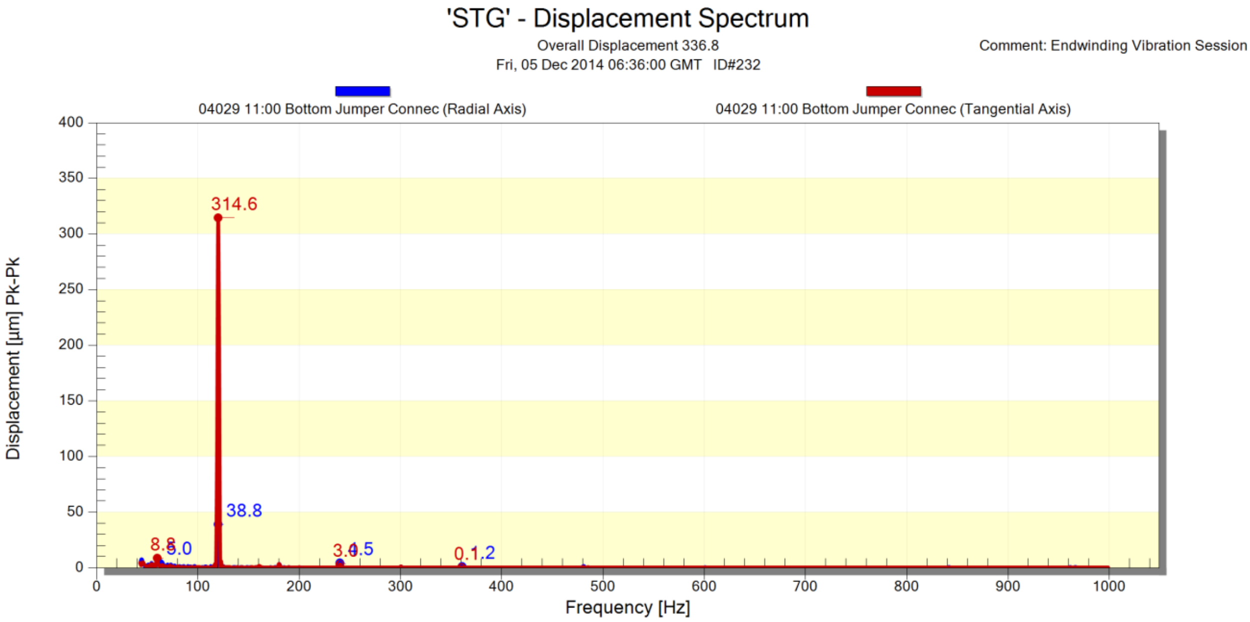

Figure 3 – Displacement Spectrum of 11:00 Bottom Jumper Connection

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 Figure 4.

Figure 4 – Loose blocking near 11:00 Jumper Connection in 2015

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, Figure 5. The significant reduction in stiffness from the loose support block reduced and separated the natural frequencies resulting in a resonance condition during operation.

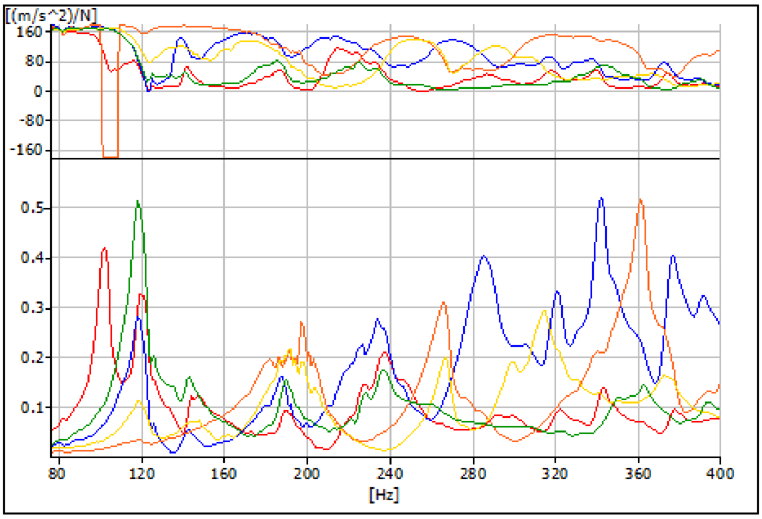

Figure 5 - 2014 Off-line Impact Test at 11:00 Jumper Connections before repair

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, Figure 6 and on-line vibration data, Figure 2 show that after the blocks were replaced and tightened the condition was similar to that of 2012.

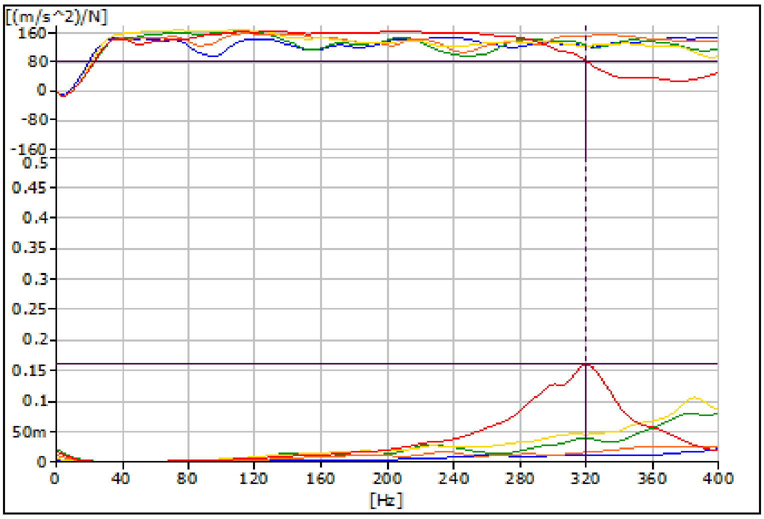

Figure 6 - 2015 Off-line Impact Test at 11:00 Jumper Connections after repairs

The vibration levels and thus the repairs remained stable for about 4 years. In April 2019 the 120 Hz displacement amplitude started to increase in the tangential direction of a sensor installed on the jumper connection at 1:00, see Figure 7.

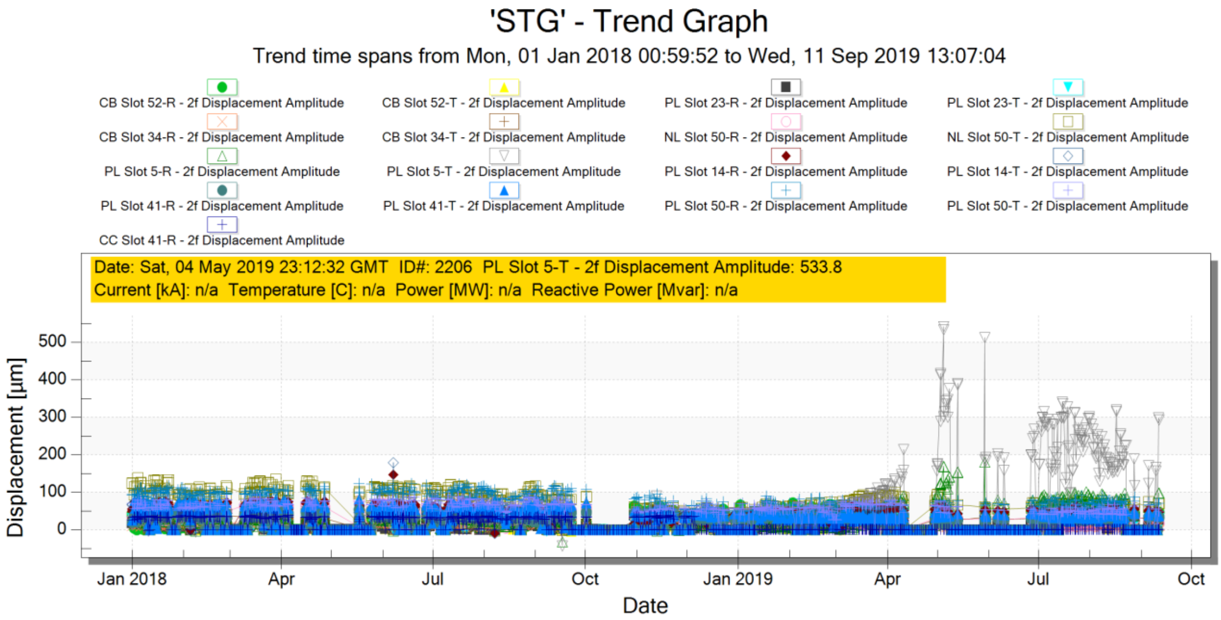

Figure 7 - On-line Endwinding Vibration Trend of 2f Displacement Amplitude

Trending the 2f displacement with operating conditions (MW, Mvar, winding temperature, stator current) showed no obvious correlation and of significance is comparing with stator current, see Figure 8. This trend indicates the vibration was independent from the forces generated by the stator current and may be an early indication of structural looseness due to resonance.

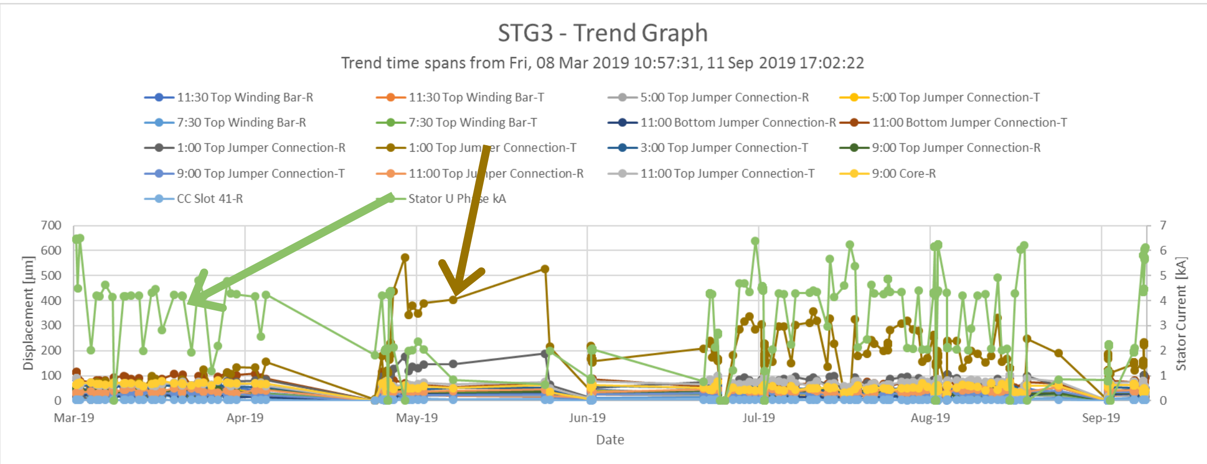

Figure 8 - On-line Endwinding Vibration Trend of 2f Displacement Amplitude and Stator Current

Operational Deflection Shape from the data sets were recorded before and after the increase in vibration in May 2019 at similar operating conditions show that the sensors at the 11:00 Top Jumper Connection and the 1:00 Top Jumper Connection were vibrating opposite of each other (out of phase) before the change in vibration and then rocking together (in phase) after the change. This relative phase change is an indication of structural change and therefore possible shift of resonance frequency towards 120 Hz. While animated Operational Deflection Shapes are more impressive to view, this concept of phase change can be seen in Figure 9.

Figure 9 – Operational Deflection Shapes May and July 2019

An outage in October 2019 allowed for a visual inspection, see Figure 10. The circuit rings on this unit continued to show signs of movement in the form of dusting and fortunately this movement was not yet being transferred to the stator winding (line and neutral bars) which have remained tight.

Figure 10 – Dusting Between the Rope Lash and Circuit Ring near 1:00 Top Jumper in 2019

A bump test was also performed and showed a significant shift from >200 Hz in 2015 to around 120 Hz, see Figure 11. The peak of the natural frequency measured at this location was at 116 Hz and along with the significant increase in online vibration levels >400 µm pk-pk and then relative decrease to >300 µm pk-pk it is reasonable to suggest that the natural frequency went through 120 Hz especially considering the bump test was performed at room temperature.

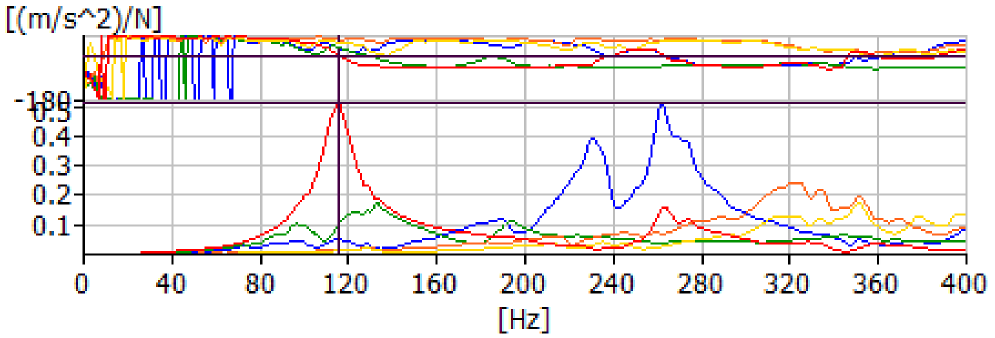

Figure 11 – 2019 Off-line Impact Test at 1:00 Jumper Connections

3. CONCLUSIONS

Advancements in the design of sensors and data processing enabled more effective monitoring of the stator endwinding condition. This is even more important now, when machines are designed less conservatively than before, and operated with more cycling leading to an increased number of problems. The case study presented shows the importance on-line vibration monitoring to detect the onset of looseness in the support structure in a timely manner. A visual inspection and impact test confirmed this condition.

In general, once excessive vibration in the stator endwinding and support structure has been detected the following remedies should be considered [2]:

1. Complete endwinding support replacement
2. Reinstall blocking and lashing material
3. Installation of additional blocking and bracing

The above should be done in conjunction with impact testing to ensure any changes made to the support system does not have a negative effect on the resulting vibration amplitudes during operation. Continuing to monitor the on-line stator endwinding vibration will indicate if the results of any structural changes are positive and when additional remedies are required from aging of the support structure and/or excessive forces from high current faults.

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