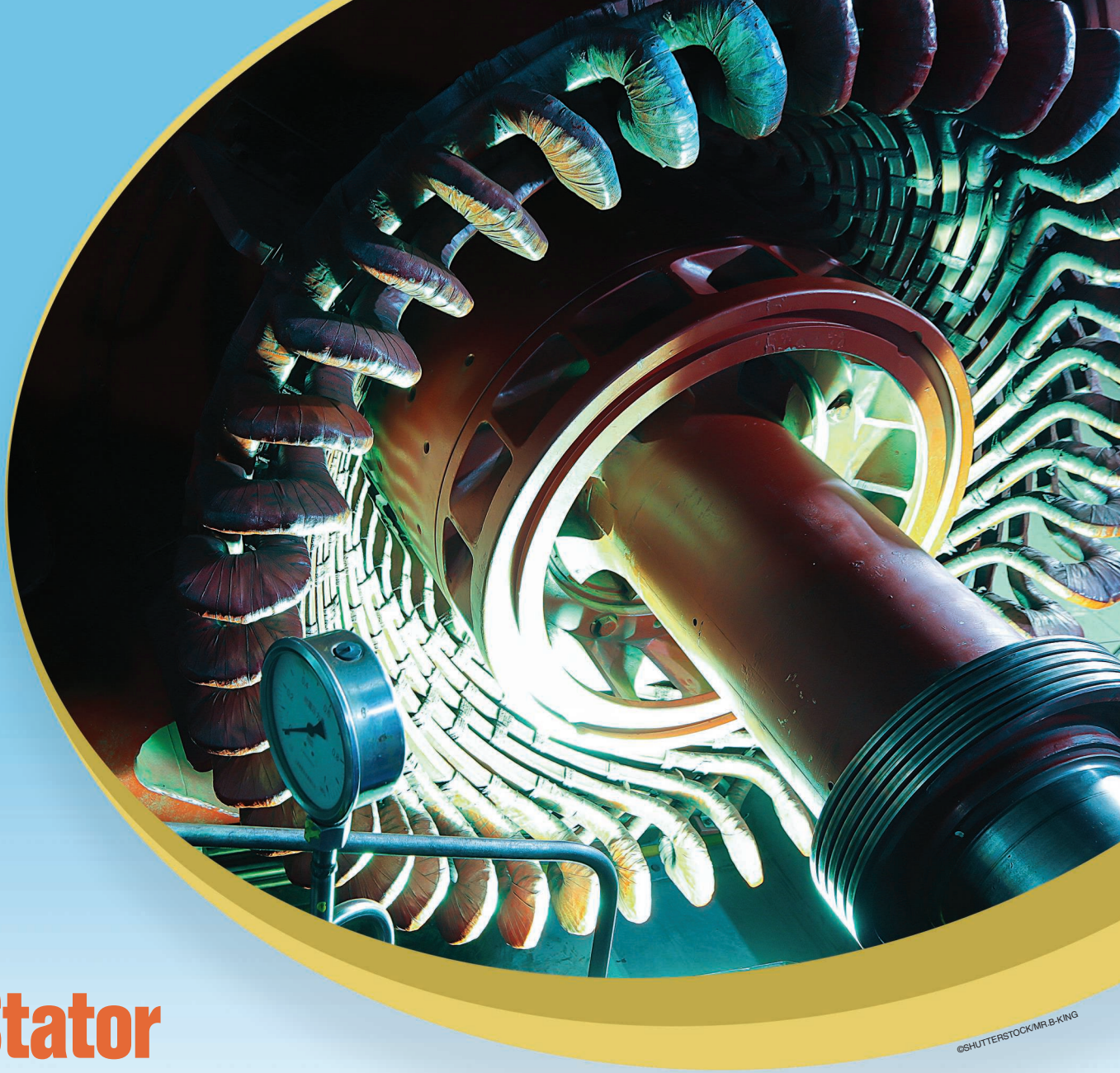

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Stator End-Winding Vibration in Two-Pole Machines

AVOIDING GENERATOR FAILURE

By John Letal, Bayu Satmoko, Nixon Manik, and Greg Stone

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IN THE PAST 15 YEARS, INSURANCE INDUSTRY DATA HAVE indicated that stator end-winding vibration has become the most important cause of generator failure. The source of vibration is the current creating magnetic forces between coils, and it may be amplified due to mechanical resonance. End-winding vibration leads to failure by insulation abrasion or copper fatigue cracking. The extent of the repair caused by the ensuing damage and the loss of production have proven to be very costly.

After 13 years of operation, one such failure occurred to a two-pole, 95-MW generator at a plant in Indonesia, which had an associated loss of US$40 million. This article describes the failure mechanism, possible reasons why end-winding vibration has become a more important issue in the past decade, and a new International Electrotechnical Commission (IEC) Technical Specification (TS) 60034-32 on methods to detect the problem well before in-service failure. A case study on the affected generator will show how a repeat stator winding failure was avoided using these tools.

Background

Stator end windings, that is, the portion of the stator winding that extends beyond the stator core at each end (Figure 1), are one of the most critical locations in two- and four-pole rotating machines. Although the stator windings produce electricity (in generators) and torque (in motors) in the portion of the stator winding within the laminated stator core, the end windings are essential to make connections between the parts of coils in the different stator slots. As the speed of the machine grows higher, electromagnetic forces and spacing constraints require the end windings to be longer, that is, extend beyond the core for a greater distance. As described later, this longer end winding makes it more difficult to ensure that the end windings can fulfill their function without reducing the reliability of the motor or generator.

To avoid in-service failures, the end-winding support structure must fulfill its function of connecting parts of coils in different stator slots with the following constraints:

1) Mechanically support the coils against movement resulting from the Lorentz magnetic force between adjacent coils. This force is proportional to the square of the stator current and causes an oscillating force that is twice the power frequency (2F) force or 120 Hz in a 60-Hz machine. The stator steady-state force is mainly in the radial and tangential directions and is caused by the current through the stator winding. In addition, the mechanical support structure of the end winding should also be able to survive a transient current up to 10 times the rated stator current if the motor or generator is exposed to a nearby phase-to-phase fault, and approximately six times the rated current that flows in the stator winding during starting of direct-on-line start motors.
2) Mechanically support the end winding against external vibrational forces that may be transmitted to the end winding from the stator frame. This can include the once-per-revolution force from the rotor bearings, referred to as the 1× force (for example, 60 Hz in a two-pole, 60-Hz machine).
3) In large motors and generators, allow for the axial expansion of the copper conductors due to the high operating temperature of the coils in the slot and end windings.
4) The support should not be affected by the high magnetic fields created by current flowing in the coils in the end winding. This effectively means that ferrous material (like steel) cannot be used in the end-windings support system since it will heat up in the magnetic field and may shift due to these magnetic forces.
5) The support should not be affected by high voltages (more correctly, high electric fields) present in the end winding since some of the coils are connected to the phase terminals. This effectively requires that the support structure not contain metallic components that are at 0 V since the potential difference between the metal and the surface of the stator coils will result in partial discharge (PD).

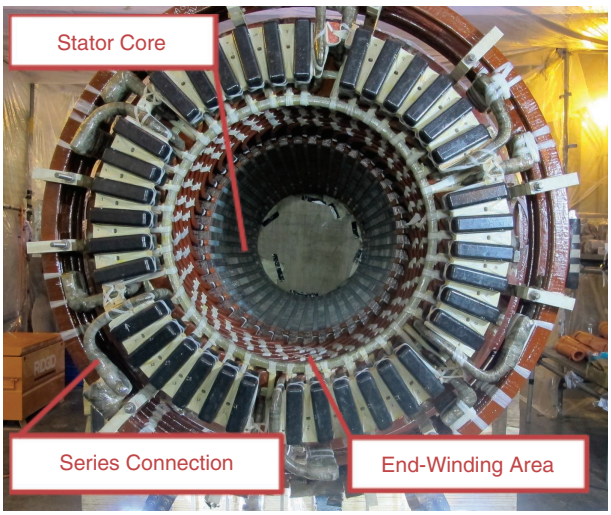

FIGURE 1. A turbine generator stator end winding.

These conflicting requirements have resulted in various solutions by different manufacturers over the years [1]–[3]. However, most modern end windings do not use metallic components; rather, they use fiberglass, epoxy, insulating blocking between adjacent coils, and fiberglass–epoxy composite surge rings or cones. All these materials are used to ensure the coils do not vibrate in the end winding, cause local overheating, or lead to PD and electrical tracking.

Unfortunately, if the end-winding support structure is not well designed and/or well made, or if the stator winding has aged, the coils in the end winding may start to vibrate under the 2F and 1× forces. This vibration can cause the following issues:

1) The vibration can lead to relative movement between end-winding components; for example, fiberglass rope affixing the coils to the support rings and braces abrading the coil insulation (Figure 2). Eventually, the abrasion can be severe enough to expose high-voltage copper conductors within the coil. If partly conductive contamination is present, it may result in a phase-to-ground fault.
2) End-winding vibration can loosen the stator wedges at the end of the stator core, which could eventually lead

to coil vibration within the stator core and to slot discharge. If the coils loosen in the slot, then this can cause the stator wedges to loosen as well. Also, loose coils in the slot may lead to abrasion of the PD suppression coatings and, hence, high PD.

3) In the most extreme case, if the copper conductors within the coil vibrate enough, they will fatigue crack. If all conductors in a coil fatigue crack, the stator winding current will continue to flow through the coil by forming a plasma at the severed copper, which will cause extreme heating of the conductors, melt the copper and nearby components, and often completely destroy the stator winding (Figure 3).

In the past, problems caused by end-winding vibration took many years, if not decades, to cause a failure. However, in the past 15 years, it seems that many two-pole machines, and air-cooled turbine generators in particular, have been failing in much shorter time periods. This article reviews the possible causes of these more recent issues. The article also describes ways owners of these machines can detect the issue at an early stage using both offline testing and online monitoring. A case study is presented and assesses a failure that occurred at a generating station unit where two of the authors work.

Recent End-Winding Vibration Issues

In the early 2000s, reports of catastrophic stator end-winding failures similar to that shown in Figure 3 were discussed at utility conferences by several operators of air-cooled turbine generators rated up to a few hundred megawatts. The end-winding failures were apparently occurring on several different brands of machines, often after only a few years of service. In addition, many cases of fretting (Figure 2) were being found in modern air-cooled, two-pole generators and greasing in hydrogen-cooled, two-pole generators. This led the Electric Power Research Institute (EPRI) to commission a report to educate end users on the design of two- and four-pole generator end windings as well as possible failure mechanisms [4]. Various researchers also published papers on the recent increase in end-winding vibration issues [5]–[10]. In 2010, FM Global, a large insurance company used by generating stations, presented data indicating that, for the period from 2000 to 2010, it paid out more in claims for generator failures caused by end-winding vibration than any other failure mechanism [11]. About the same time, machine manufacturers decided that an IEC standard was needed to educate end users about the end-winding vibration issue and ways to determine if a problem exists. This resulted in the 2016 publication of IEC TS 60034-32 [1].

In reviewing this information, it seems that many machine manufacturers changed the design or manufacturing of the stator end windings around the year 2000. The driver for the changes might have been a desire to simplify the design and/or manufacturing to reduce costs in the fiercely competitive gas turbine generator market. Unfortunately, it seems in many cases the mechanical natural frequencies sometimes moved in such a way that resonance (or amplification) of the vibration was occurring [4]–[6], [8], [9], [12]. That is, the natural frequency of some areas of the stator end winding were closer to the forcing frequency (usually 120 Hz in 60-Hz machines or 100 Hz in 50-Hz machines). In other cases, the lead from a stator coil to the circuit ring bus was often not supported in as many locations as tended to occur in previous designs, or the method of bonding the ropes and cords to the coils and support members were not as robust.

The publicity surrounding the premature failures motivated many manufacturers to alter their designs, and thus most machines made since approximately 2012 have end windings that are less prone to end-winding vibration. However, many two-pole (and a few four-pole machines) are still operating with suboptimal end-winding designs.

FIGURE 2. An example of insulation abrasion (the white powder caused by fretting) created by the relative movement between the coil, support ring, and support brace.

FIGURE 3. Collateral damage to the stator winding and core caused by fatigue cracking of the copper conductors due to severe end-winding vibration.

Offline Methods to Assess the Looseness of a Stator Coil Support System

Visual inspections of the end winding and surrounding support structure are generally considered the main way to detect evidence of excessive movement [12]. The dusting that results between components, which are supposed to be held tightly together, is an indication that they have loosened, are rubbing against each other, and causing the insulation to fret (Figure 2).

Bump testing can be used to identify the natural frequencies characteristic of the end winding and support system structure. In the bump test, various areas in the end winding are hit with a hammer and the response to this impact is measured with temporarily installed piezoelectric accelerometers. These response frequencies are proportional to (the square root of) stiffness. Resonance is a condition where the natural frequencies of the windings and support system coincide with the normal operating forces at turning speed (1×) and twice line frequency (2F), which amplifies normal vibration levels at these frequencies and increases the copper fatigue rate. As the end-winding looseness increases, the natural frequencies decrease. This makes the bump test a trendable offline test if the data are collected at the same locations. Having the natural frequency information of a structure can be used to avoid resonance.

Modal analysis is a bump test procedure that provides a comprehensive evaluation of the end-winding basket to ensure any natural frequencies near 2F are likely to lead to failure [1], [4]. Examples include an oval mode shape for two-pole machines (as the stator tends to deflect naturally in an oval shape due to the two-pole rotating magnetic field) and a square shape for four-pole machines. To capture modal data, usually the force is recorded from a calibrated hammer at a fixed location, and the accelerometer is moved at equidistant positions around the end-winding basket on the same plane.

Bump testing (driving point and modal) should be carried out after a major repair or rewind to characterize the effect on the natural frequencies from the structural changes. If similar data are collected periodically, it is possible to trend these frequencies, particularly if a baseline test is performed during installation of the machine. It is only possible to trend these frequencies if the data are collected using the same methods and at the same locations. This requires good reporting practices since the period between tests can be several years due to limited machine access. Driving point tests, in which the accelerometer and hammer are at the same location, can also be carried out to assist in the location of sensors to monitor the vibration online for local resonances.

When performing a bump test, the effect of temperature must be considered. The natural frequencies identified at room temperature during offline testing will shift downward in frequency during machine operation by as much as 10 Hz or more [1], [4], [8]. It is for this reason that typical acceptance criteria for a stator end-winding bump test avoid natural frequencies above a certain magnitude within a frequency band around 1× and 2F favoring the upper frequency band [1], [4], [12], [13]; for example, 54 to 72 Hz and 108 to 144 Hz (–10%/+20% of 1× and 2F) for a two-pole, 60-Hz machine.

If a resonance condition on the stator end windings occurs during machine operation, the stator wedges and the coils in the slot may loosen as well. There are several offline methods that can be used to assess coil looseness in the slot. A manual tap will make a distinct thud if the wedge is hollow or loose [2], [3], [12], which is a distinctly different sound than a tap on a wedge that is tight. The qualitative analysis of listening to the sound of a tapped wedge during the test is subjective and relies on operator experience. Alternatively, a robotic, calibrated hammer can be used to strike the wedge beside an accelerometer to measure the vibration response of the wedge; this type of hammer provides repeatable and more easily compared results. A tight wedge is used as a reference to which the remaining wedges are compared.

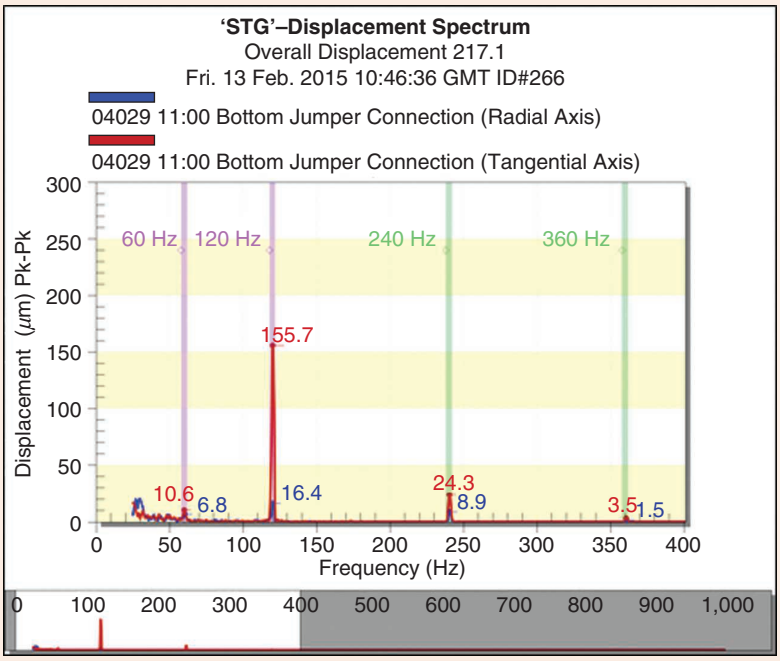

FIGURE 4. Displacement spectra at the onset of looseness. Pk: peak.

Online Methods to Assess the Looseness of a Stator Coil Support System

Permanently installed accelerometers are considered the best way to understand the end-winding behavior during operation [1], [4], [12]. Unlike offline testing, online monitoring evaluates conditions while the machine is at operating temperature and load. Also, online monitoring can assess the effect of a transient current on the end-winding structure without having to take the machine out of service or disassembling it for inspection, both of which add potentially unneeded cost and risk. Modern stator end-winding monitoring systems use fiber-optic accelerometers to avoid introducing metal into the high-voltage and high magnetic fields present in the end windings, which could potentially cause overheating or lead to PD and electrical tracking.

Historically, the vibration quantity used to assess end-winding vibration is displacement, which is the total distance of the component being monitored as it moves around its rest position. The main benefits of continuous monitoring of stator end-winding vibration levels online are the ability to trend the displacement over time and whether the trend is related to machine operation (increased load results in higher forces and, thus, higher vibration amplitudes). However, it is sometimes necessary to evaluate the end-winding vibration against recognized acceptance criteria. Unfortunately, progress in the development of such an absolute criterion is slow, but it is generally recognized that 250 μm (or 10 mil) peak to peak at 2F (120 Hz for a 60-Hz machine) is cause for some concern [3], [4], [14]. A recent EPRI study, which used a modeled approach, suggested much higher theoretical limits for end-winding vibration, ranging from 500 to 1,200 μm peak to peak [10], which might cause the copper conductors to fatigue crack. The hypothesized levels were separated into four categories based on the stiffness of the end-winding support, with higher acceptable levels on more flexible designs, which is a reasonable approach. A so-called knockdown number of four [10] would reduce the limits to 125–300 μm peak to peak, making the limits more consistent with the practical approach used to develop IEEE Standard 1129, which is largely based on the North American experience from past

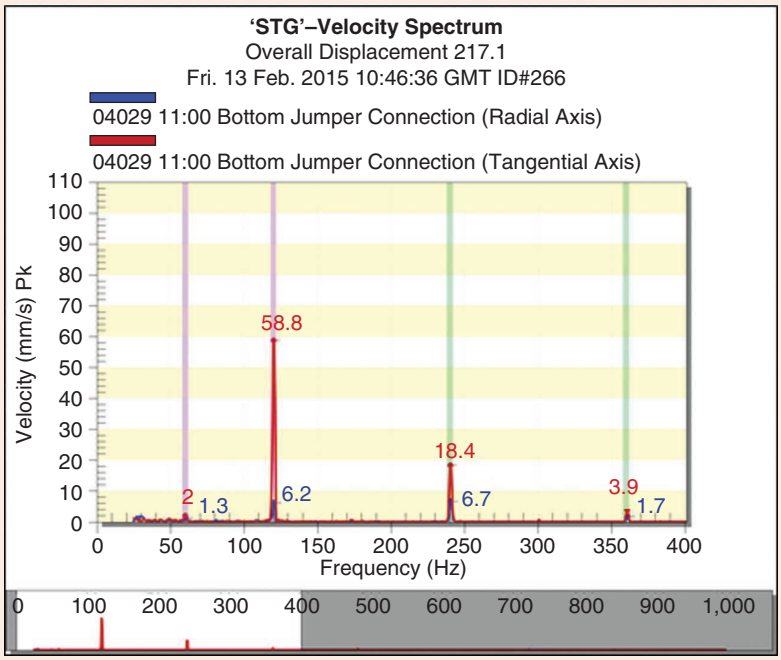

FIGURE 5. Velocity spectra at the onset of looseness.

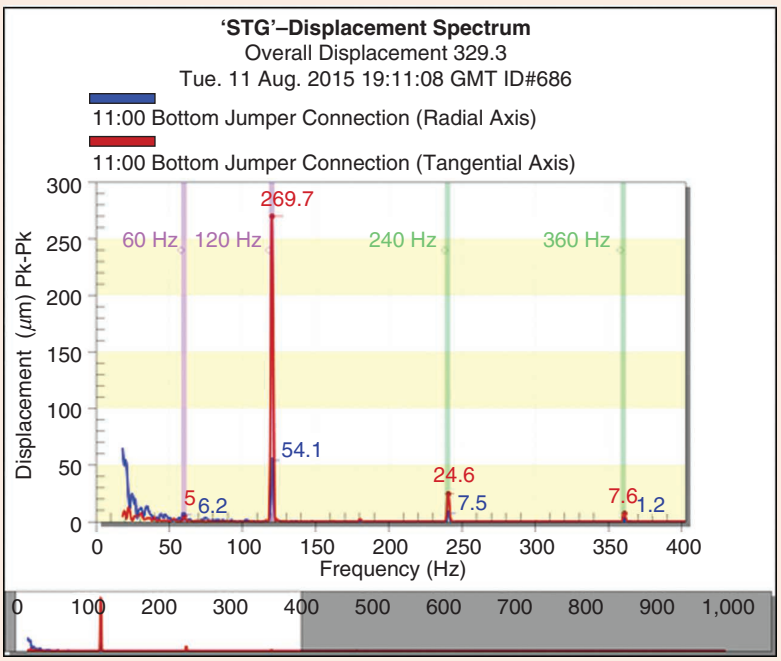

FIGURE 6. Displacement spectra just prior to visual inspection confirming looseness.

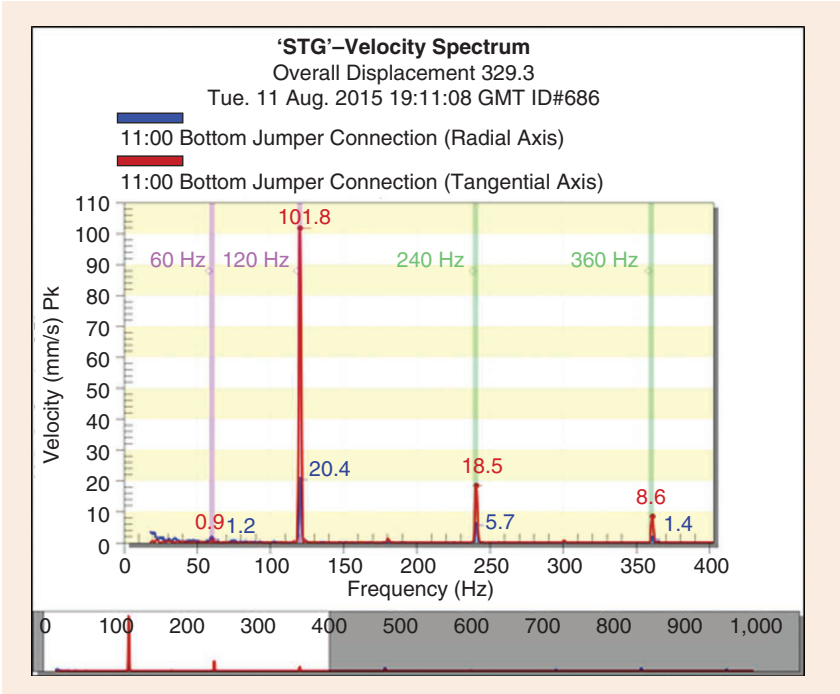

FIGURE 7. Velocity spectra just prior to visual inspection confirming looseness.

years with 60-Hz generators [14]. Regardless of the source, of most concern is whether there is a rapid increase in the vibration levels at similar operating conditions, which can be irreversible [1], [4], [10], [14]. An increase of more than 25% in the displacement under the same operating conditions is considered significant [1].

Vibration collected from an accelerometer requires double integration of the signal to get displacement. Integrating the signal twice with respect to time puts emphasis on amplitudes at lower frequencies. Thus, if the focus of vibration assessment is lower than a few hundred hertz, as is the case for monitoring stator end-winding at 1× and 2F fundamental frequencies, displacement is suitable. However, mechanical looseness can present itself in a vibration spectrum with a series of harmonics (multiples of some fundamental frequency) [1] often amplified by one or more resonances, so it can sometimes be advantageous to view the data as acceleration or integrated just once to velocity. Velocity has a smoothing effect over a wider range of frequencies to more readily identify harmonics possibly associated with mechanical looseness. The excitation force is still 1× or 2F, and the looseness allows the component to vibrate more freely than if there was a tight bond due to a reduction of stiffness in the support structure. It is this condition that can result in rattling and the production of harmonics in the frequency spectrum. For comparison, we provide displacement and velocity spectra at the onset (Figures 4 and 5) and leading up to (Figures 6 and 7) end-winding support structure looseness identified during a visual inspection (Figure 8) [9]. Not only is the dominant frequency at 2F (120 Hz for 60-Hz machine), which is considered a typical end-winding vibration, but the harmonics present are indicative of mechanical looseness. When comparing the relative amplitudes at 240 and 360 Hz to the fundamental 120 Hz, they are more easily identified when viewing the velocity.

FIGURE 8. Visual inspection showing looseness between the support block, circumferential support ring, and connection between the winding and circuit ring [9].

Operating deflection shape analysis can also be used for global vibration and comparisons with modal analysis. If the relative locations of the accelerometers are known and signals are collected at the same time using a multichannel instrument, the relative amplitudes and phase timing of the vibratory signals can be compared and even animated to represent the global behavior of the end winding. This can help correlate the operational movement online with the mode shape from the bump test data recorded offline, to help assess the potential for exciting a global resonance of the end-winding basket. Global resonance is an indication of an impending fault, which often requires thorough investigation and structural modifications to reduce the amplification. On the other hand, local resonance can often be controlled by periodically tightening the structural components and continuing to analyze the trend. In most cases, a permanent fix is difficult, if even possible, when an end-winding vibration issue is present, so online monitoring can be utilized to effectively plan and schedule maintenance outages.

In extreme situations where the stator wedge and/or coil in the slot have loosened, fiber-optic accelerometers can be used to directly monitor the vibration at the slot exit. In general, the maximum movement

in the slot is near the exit. This is based on the physical evidence of the location of the surface abrasion during winding inspections, and it is for this reason that attention is paid to the end wedges after stator winding faults because they are generally the loosest. Alternatively, capacitive sensors can be used to measure the relative vibration between the wedge and the stator coil, but this approach requires the modification of the wedges to properly install the sensors. In addition, the results of the capacitance change can be misleading if the wedges loosen and the coils remains tight. Generally, loose wedges can be tolerated, but loose coils require tightening to avoid damage to the surface coating.

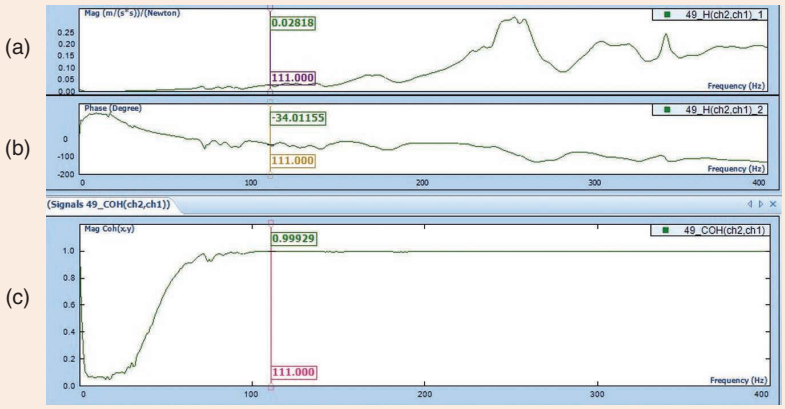

FIGURE 9. A typical frequency response function depicting 0.028 (m/s^2)/N (or 0.013 g/lb-F) at (a) 111 Hz with (b) phase and (c) coherence. g/lb-F: acceleration (gravity) per unit of force (pounds).

A Case Study: A Two-Pole, 95-MW Generator

In 2013, after 13 years in operation, unit 2 (Figure 1) tripped when the generator relay (differential-phase current) was activated due to a phase B-to-phase C fault. This resulted in generator damage and the loss of production of more than US$40 million. An incident investigation concluded that the probable cause was vibration occurring on the generator end winding, which abraded the insulation and led to cracking of the coil insulation just outside of the stator slot (Figure 3). As is known from similar failures elsewhere, in severe cases, this vibration may cause the copper conductors to fatigue crack. Once all the copper conductors in a coil had cracked, a high-arcing current occurred between the severed ends. The plasma melted the insulation in adjacent coils, leading to the phase-to-phase fault. The recommendation of the investigators was to install a generator stator end-winding vibration monitoring system.

Bump testing was used to determine the most responsive areas where the potential for vibration was greatest to locate where to install the accelerometers. The maximum accelerance responses [in (m/s^2)/N] were recorded in the critical frequency band of 90–120 Hz for this 50-Hz machine (Figure 9).

An effort was made to distribute the accelerometers sufficiently to maximize coverage for monitoring the generator stator end-winding vibration. Dual-axis sensors were installed to measure the radial and tangential directions at six locations on the connection end of the generator. Various components were selected to generalize the vibration behavior of the stator end winding, including two sensors' online end connections from the windings to the circuit rings, one on a jumper connection, one on a winding near the series connection end cap (Figure 10), and two on the circuit rings. Additionally, a single-axis sensor was installed on the stator core as a reference. The instrumentation used to continuously monitor the end-winding maximum overall displacement vibration data online was commissioned to store the raw maximum overall displacement vibration levels once daily on all sensors for the data set.

Vibration levels greater than 250 μm (or 10 mil) peak-to-peak (broadband) on the connection end led to a shut down in October 2016 (Figure 11). The predominant frequency was subsynchronous (below turning speed). This peak was present on all sensors at varying amplitudes, including the core. It is likely that the source was external to the end windings (Figure 12). Viewing the data as velocity demonstrated some higher frequency response unrelated to the 2F (Figure 13), which was totally suppressed when viewing the displacement spectra. It is likely that

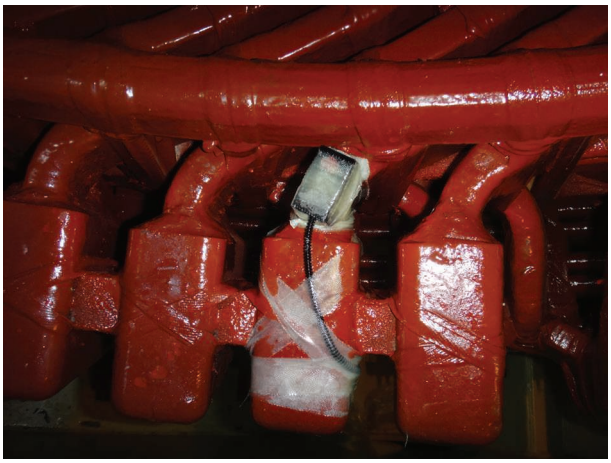

FIGURE 10. An example of a sensor installed on a winding near the end cap, which insulates the connection between two coils.

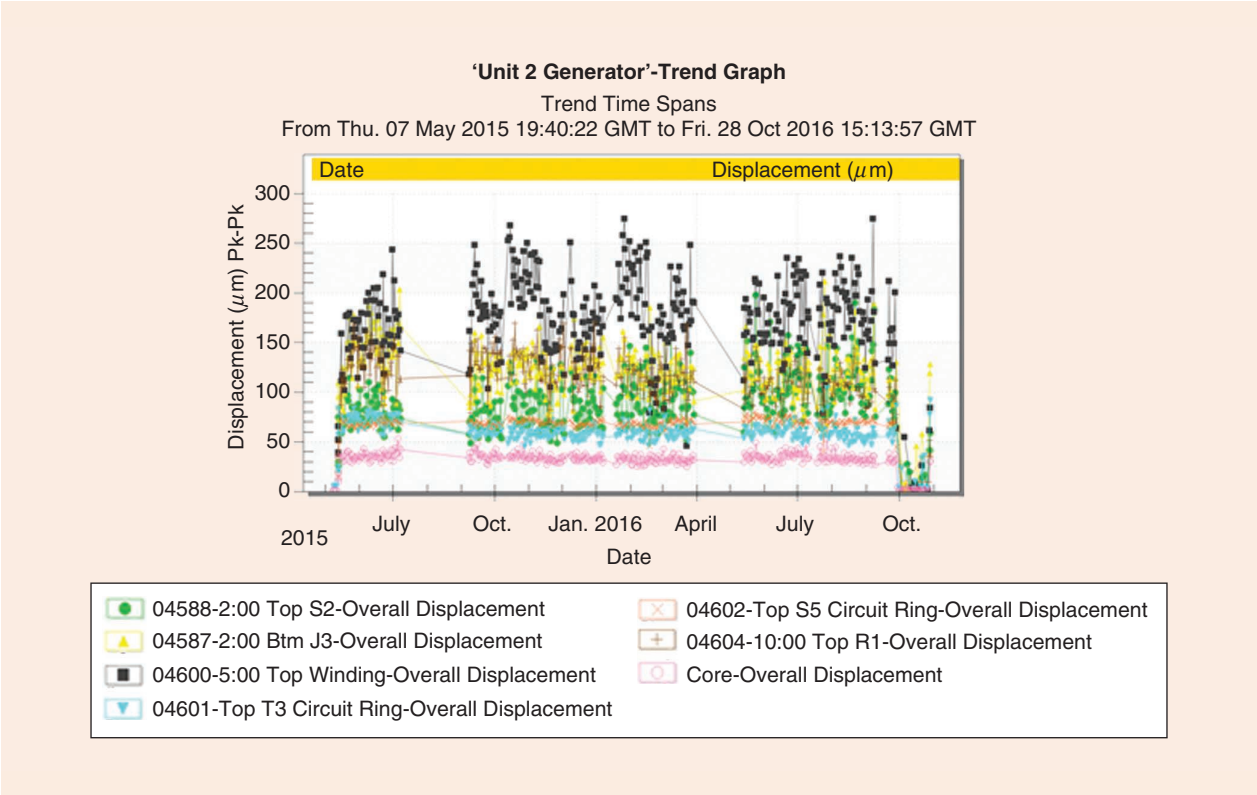

FIGURE 11. The overall broadband (25–1,000 Hz) displacement trend before and after the October 2016 repair.

the 333-Hz vibration amplitude was being amplified by a natural frequency. The vibration amplitude from the electromagnetic force at 100 Hz was acceptable at fewer than 75 μm (or 3 mil) peak to peak (Figures 12 and 14).

During the outage, a manual tap test of the stator slot wedges showed that many were loose and needed to be replaced. A bump test on the end windings was also performed, and there seemed to be some correlation between five of the loose wedges with end turns that had high responses in the bump test on the turbine end. The loose wedges numbered 1, 19, 22, 25, and 28 correspond with T1, T25, T22, T19, and T16 locations that experienced high bump test levels (Figure 15).

The failure criteria used for the bump test were an accelerance response of greater than 0.055 $(m/s^2)/N$ (or 0.025 g/lb-F) within 5 Hz of 50 and 100 Hz and greater than 0.110 $(m/s^2)/N$ (or 0.050 g/lb-F)

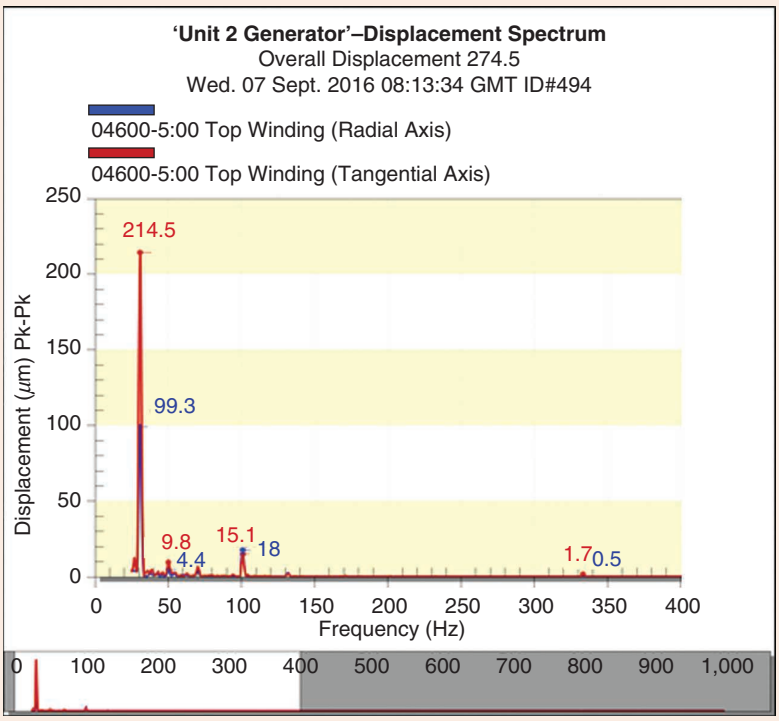

FIGURE 12. Displacement spectra before repair.

within 10 Hz. The results from the bump test also led to the decision to install additional sensors on the turbine end of the generator stator end winding. The stator slot wedges that were loose were also replaced.

Wedge replacement on the connection end appears to have affected the end-winding vibration response. After the outage, the unfiltered vibration levels were reduced to 200 μm (or 8 mil) peak to peak for the remainder of the year (Figures 11 and 16). Tightening the coils in the slot seems to have limited the end-winding vibration response from the subsynchronous external source. The 100-Hz vibration remained unchanged.

Partial discharge levels were recorded prior to the August 2016 outage. The maximum peak PD magnitude (Qm) value was 59 mV (Figure 17). For air-cooled stators equipped with 80-pF sensors on their terminals, these magnitudes are considered low when compared to

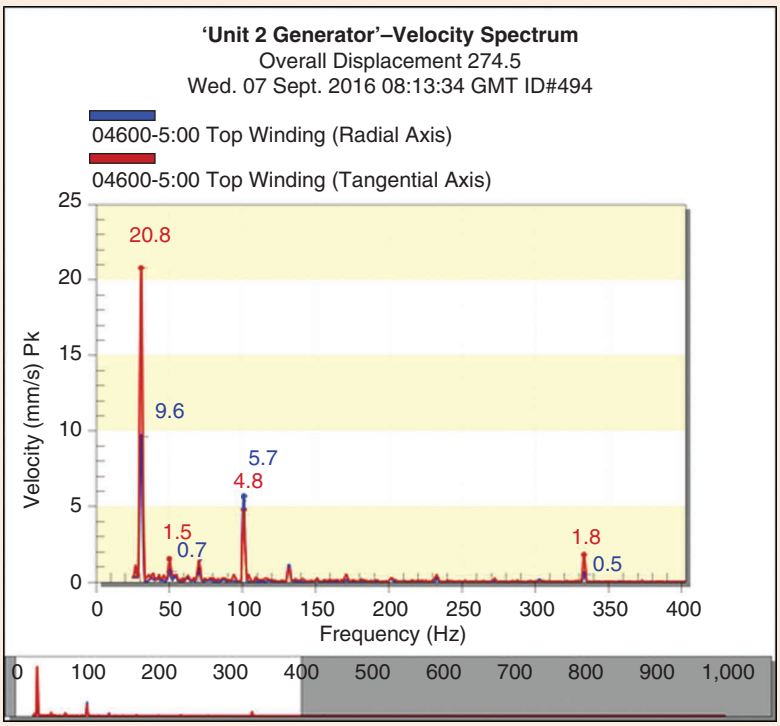

FIGURE 13. Velocity spectra before repair.

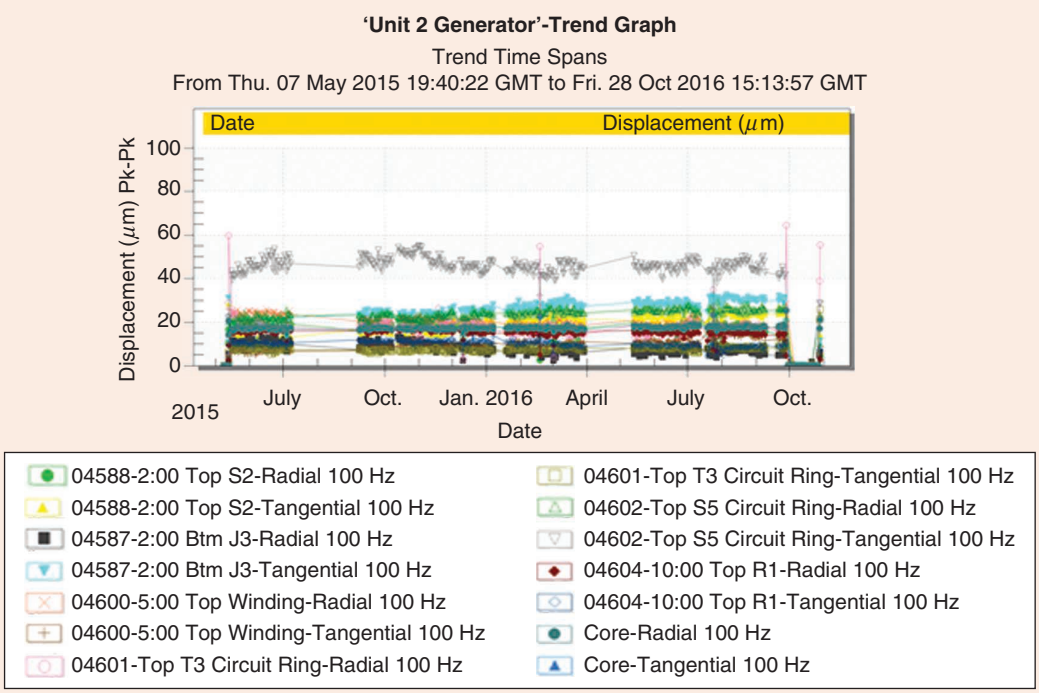

FIGURE 14. The 100-Hz displacement trend before and after repair.

other similar machines [15]. Even though the wedges were loose, the coils were not vibrating sufficiently to degrade the semiconductive surface coating.

Conclusions

Problems in two- and four-pole motors and generators can be avoided with careful specifications prior to the purchase or rewind of these machines. A bump test with appropriate natural frequency exclusion zones has a lower potential for resonance resulting in lower vibration amplitudes during operation.

For those machines with high end-winding vibration amplitudes, the increased rate of copper fatigue can lead to stator winding failure. The degrading looseness conditions can extend from the end turn and support system to the coils inside the slot, including the wedge tightness. Periodic testing offline can assess the degree of looseness and help to determine the potential for resonance during operation. If there are local resonance issues or when a generator is being fitted for online monitoring, driving point testing is sufficient. Conversely, if there are concerns about global resonance related to design or if there has been a significant structural change to the end winding and support system, then additional modal analyses should be performed.

The best way to assess degrading looseness conditions for machines with end-winding vibration issues is to monitor the response to operational forces online. Rapid deterioration of the support system can be identified by monitoring for increases in vibration trends in similar operating conditions; doing so can assist the planning of maintenance activities. Any change to the structural support system should be validated with a bump test to limit the potential for resonance, and the repairs can be deemed successful when testing demonstrates a reduction in vibration amplitudes online.

FIGURE 15. The unit 2 generator with wedges and bump test locations identified.

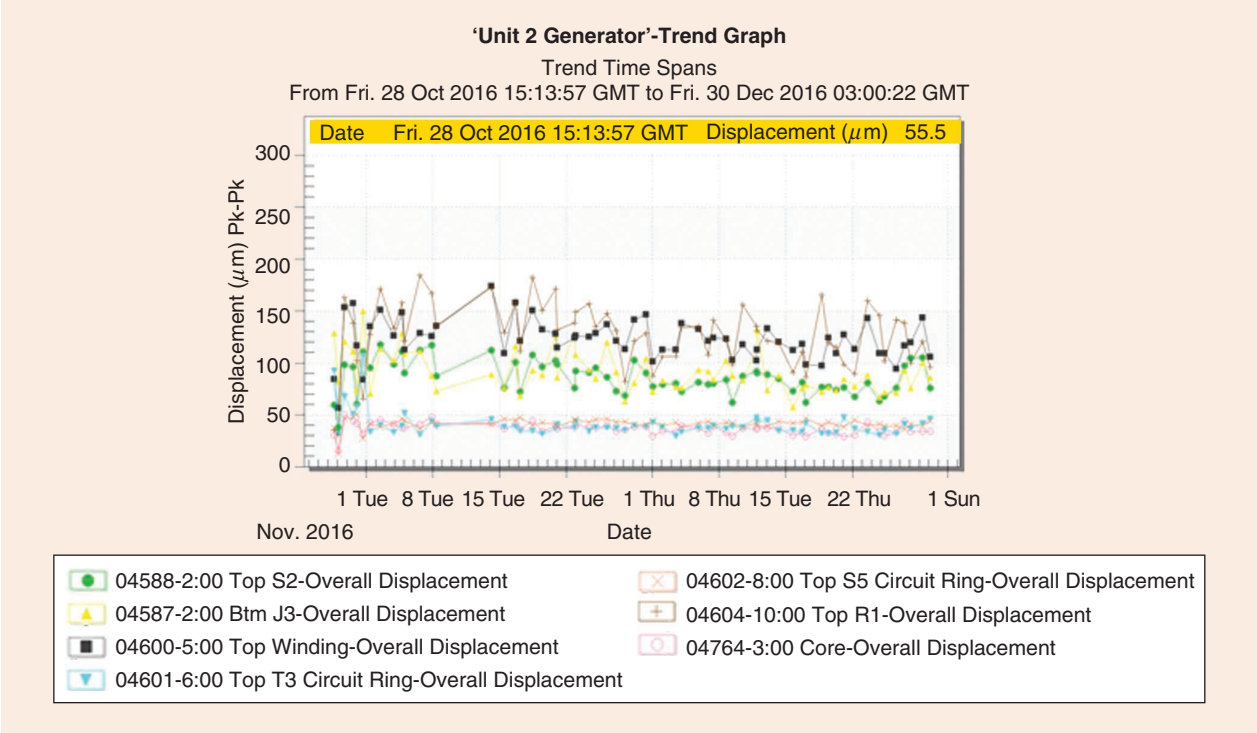

FIGURE 16. The overall broadband (25–1,000 Hz) displacement trend after repair.

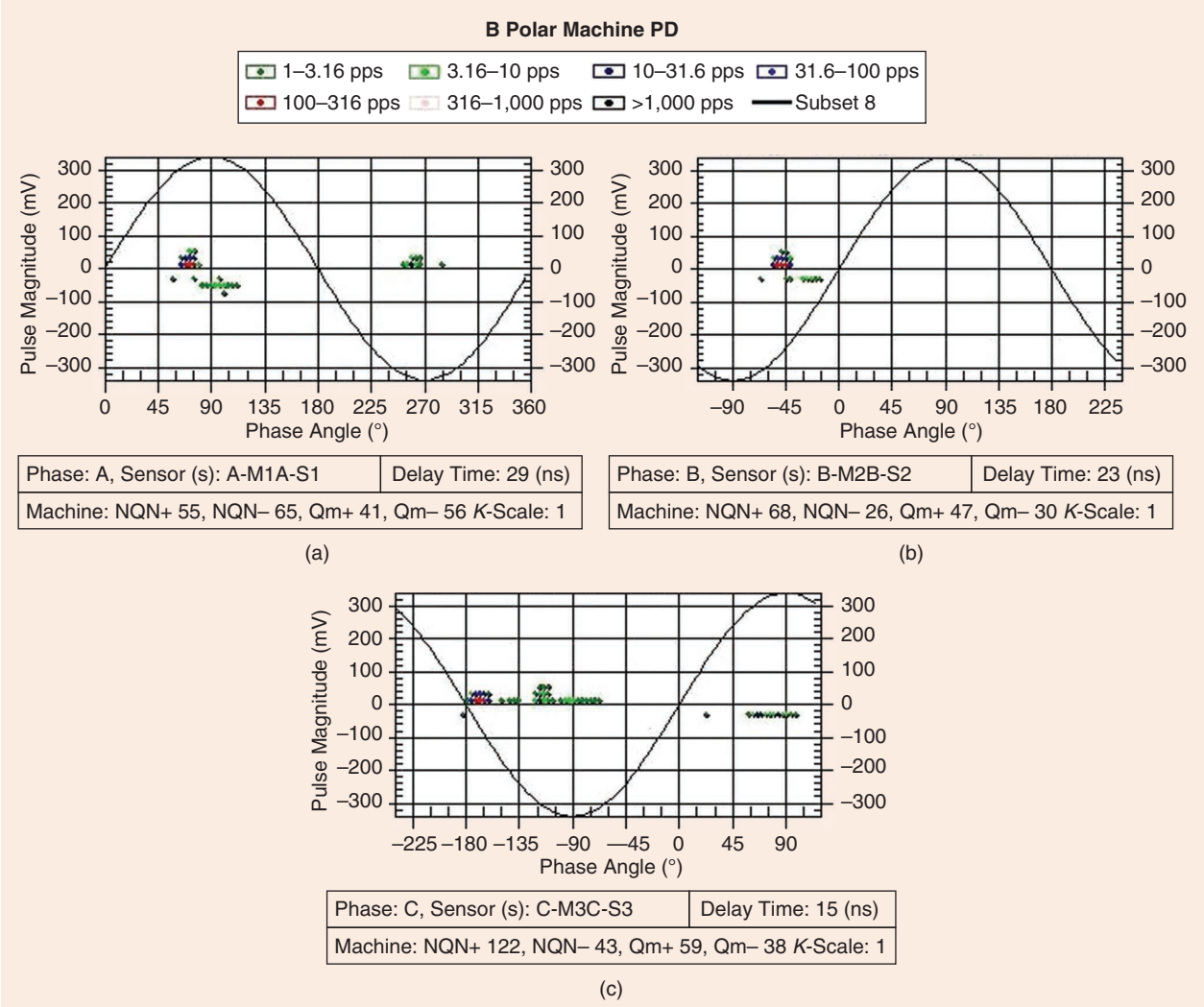

FIGURE 17. The machine PD at a sensitivity range of 20–340 mV. (a) Phase A; (b) phase B; (c) phase C.

Author Information

John Letal (jletal@qualitrolcorp.com) and ***Greg Stone*** are with Iris Power, Mississauga, Canada. ***Bayu Satmoko*** and ***Nixon Manik*** are with Barito Pacific Petrochemical, West Java, Indonesia. Letal is a Senior Member of IEEE. Stone is a Fellow of IEEE. This article first appeared as "Stator End-winding Vibration in Two-Pole Machines" at the 2018 IEEE IAS Petroleum and Chemical Industry Technical Conference. This article was reviewed by the IAS Petroleum and Chemical Industry Committee.

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