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A1-201

CIGRE 2014

Recent endwinding vibration problems in air-cooled turbine generators

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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 and fabricate. During normal operation, the endwindings are subject to high mechanical 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. 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. This constraint has resulted in a wide variety of endwinding support structures from the various manufacturers. If a component of the endwinding or the endwinding basket as a whole has a natural frequency close to the forcing frequencies, the vibration response will be in a resonance condition and the result can be catastrophic. Off-line impact testing has long been a tool to identify these natural frequencies and help determine if a resonant condition may exist. Not only can this testing be used to assess the condition of a stator endwinding, but it can also be used to identify the locations that are most likely to vibrate.

To avoid premature failure, excessive motion in the endwinding during operation can be monitored. To effectively do so, not only should the sensors be installed at locations most likely to vibrate, but they should cover a wide enough range to capture all of the vibrating frequency components. There is little guidance on acceptable vibration levels, so an increasing trend is of concern.

Although it is clear that the endwinding support system of even the largest generators can be designed to achieve 30 or more years service without excessive loosening, in the past decade a large number of in-service faults have occurred due to endwinding vibration. Even more machines have been discovered during visual inspections to have premature deterioration of the endwindings due to looseness. These problems have been primarily associated with air-cooled machines typically installed in gas turbine or combined cycle plants. To greater or lesser degrees, most large generator manufacturers have been affected. It is suspected that competitive issues (and in particular cost) may be forcing manufacturers to compromise on proven design and manufacturing methods.

KEYWORDS

Endwinding vibration, turbine generators, bump testing, vibration monitoring

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INTRODUCTION

The design of stator endwindings of large turbine generators is one of the most challenging aspects of machine design. The endwindings are subject to large vibrating forces especially from the magnetic fields created by the power frequency current in the endwindings [1-5]. Yet the endwinding must have a support structure that contains few or no metal components but resists these forces for 30 or more years.

Manufacturers have designed several different ways of constraining endwinding movement which, for the most part, have been successful in providing long life with minimal maintenance. However, in the 1980s, it was apparent that some designs of large machines with direct hydrogen inner cooling suffered from premature failure due to insulation fretting and cracking of the copper strands due to high cycle fatigue [4]. Modified endwinding support structures were successful in controlling this movement [5]. In the past decade anecdotal information suggests that many different stator windings in air-cooled 2-pole generators rated 100 MVA and above seem to be experiencing premature deterioration and sometimes catastrophic failure [6-8]. This apparent increase in problems has stimulated a CIGRE questionnaire on the rate of incidents, although the results are not yet known. Recent insurance company data seems to give quantitative confirmation that stator endwinding failure has become one of the more prominent causes of insurance payouts for generator failures [9]. The reasons for this increase in endwinding vibration problems, which seems to occur with many of the major brands, are not clear. However, it may be because manufacturers are being forced to reduce cost to remain competitive, and the endwinding is one area where major cost savings may be realized.

This paper summarizes the root causes of endwinding vibration that can lead to premature aging and failure, and presents several recent examples of problems. Measures for new and existing generators to warn of possible problems via off-line or on-line testing, together with suggested warning limits, are given.

CAUSES OF ENDWINDING PROBLEMS

An extensive, modern discussion of the causes of stator endwinding vibration is presented in [1]. In short, the main force in the endwinding that can lead to vibration is usually magnetically induced via the power frequency current. Since the magnetic force is proportional to the AC current squared, 50/60 Hz (f) current creates a 100/120 Hz (2f) force. These magnetic forces are primarily in the radial and tangential directions. In addition, if there are strong higher frequency current harmonics flowing in the generator, for example due to design and/or due to connected loads such as inverter drives and induction furnaces, even higher magnetic force frequencies may be present. During a phase-to-phase short close to the generator, the stator currents can be as much as 10 times higher than rated, creating a transient magnetic force up to 100 greater than normal [1,2].

In addition to the magnetic force, there can be a force induced on the endwinding that is at the rotational frequency – 50 Hz (or 60 Hz) in a 2-pole machine. This rotational mechanical force may be caused by rotor unbalance, rotor turn insulation shorts, bearing problems, etc which lead to bearing vibration that couples via the generator frame and stator core to the stator endwinding.

The magnetic and rotational mechanical forces lead to some vibration of the entire endwinding as a unit. The purpose of the endwinding support system is to limit this vibration to a sustainable level and to ensure that these forces do not lead to vibration of individual elements (relative movement between endwinding components). The endwinding support system almost always has one or more "support rings" or a cone of some sort. In turbine generators the rings or cone are usually made from a polyester or epoxy fibreglass laminate, or fibreglass rope (which is impregnated with B-stage epoxy or is dry and then impregnated during a global VPI process). The rings are placed either inside the stator bar layers (i.e. closer to the rotor than the bars), or radially outside of the bars, or both. Cones are placed radially ouside of the layer of bars. Each bar is lashed to the ring or cone. Insulating blocks a few centimetres in length, placed between adjacent bars, provide circumferential support. One or more rows of such blocking may be present in each endwinding. There are also blocks between the top and bottom layers of bars. The hoop strength of the support rings/cone help to ensure that the coils/bars do not move radially. Most of the endwinding blocking materials, as well as any cords/ropes that may be used to bind bars to one another and to the support rings/cone, are made from insulating materials. It is not uncommon for the endwinding support system to be different between the connection end and the turbine end of the machine, resulting in different behaviours.

Another consideration in endwinding design, especially for large 2- and 4-pole generators, is the growth of the coils in the slot and the endwinding as a result of operating temperatures. As a stator goes from no load to full load, the copper conductor's temperature will increase and, due to the coefficient of thermal expansion, the bars grow in length. The endwinding support system must be able to accommodate this growth, otherwise the support system and even the bars can become distorted. Accomplishing this tends to be somewhat of an art [1,2], although analytical methods are often helpful [3].

The most likely cause of endwinding vibration problems is when the endwinding structure, either globally or locally, has a natural frequency (when the winding is at operating temperature) that is close to the rotational or magnetic forcing frequencies, e.g. rated speed and 2f. These natural frequencies may be present in a new stator due to poor design or manufacturing (e.g. misplaced blocking). The natural frequencies may also change over time due to shrinkage of the insulating materials, loss of bonding, stretching of cords and lashing materials, etc due to long term thermal aging. Finally the frequencies may change due to large current transients from the power system that stretches cords and lashing and/or breaks the bond between components.

EXAMPLES OF RECENT ENDWINDING PROBLEMS

There are many possible causes of endwinding vibration. Those associated with design are:

- Insufficient number or inadequate support rings or bracing members.
- Insufficient fiberglass in the stator bar insulation system, which imparts less mechanical strength to the bars.
- Inadequate or inappropriate application of bonding resins to fibreglass ties and roving.
- Insufficient allowance for workmanship variations that can lead to changes in structural natural frequencies
- Operation at higher temperatures than used in the past, which can move natural frequencies closer to the rated speed and 2f forcing frequencies and result in resonance, as well as reduce the strength of the support and bonding components.
- Operating generators designed for 50 Hz in a 60 Hz power system.

All of these may have resulted from the need for manufacturers to reduce cost in response to competitive pressures. Figures 1 to 4 show examples of endwindings that have developed specific local damage to insulation fretting due to local natural frequencies. All are from 2-pole air-cooled turbine generators rated <200 MVA. Note that often such problems occur on the connections between the line end bar and the circuit ring buses (Figure 2). Figure 5 shows more global endwinding vibration. Fretting is when one component moves relative to another and results in a white powder due to abrasion of the insulation. When mixed with oil, the white powder becomes black or brown, and is often called "greasing" (Figure 3).

In addition to insulation fretting, copper conductors can crack from vibration fatigue, resulting in more and more of the copper strands breaking. Eventually most of the strands are broken and the current in the bar is interrupted, resulting in considerable collateral damage from the arcing that occurs (Figure 6). These types of faults give rise to very large insurance claims referred to in [9]. Unfortunately, copper is a material that will fatigue crack at even low vibration amplitudes with a sufficient number of vibration cycles. This is why current harmonics can be dangerous if there are also endwinding natural frequencies above 2f.

Figure 1: Winding bar insulation abrasion at support ring

Figure 2: Insulation abrasion (dusting) at circuit rings

Figure 3: Greasing, due to a mix of insulation dust and oil at insulation abrasion locations

Figure 4: Loose block leading to bare copper due to abrasion

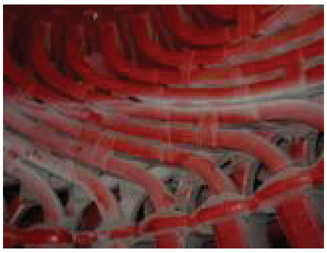

Figure 5: Widespread endwinding vibration and insulation abrasion

Figure 6: Endwinding vibration leading to copper fatigue cracking and current interruption at full load.

In our experience none of these problems will occur if there are no natural frequencies (at operating temperatures) near a forcing frequency on a relatively new machine. However, with long-term thermal aging, the support system (or its components) may loosen and fretting may occur. Similarly, as mentioned above, a high current transient may also loosen the support structure and result in vibration.

IDENTIFYING WHICH MACHINES ARE AT RISK

Bump Testing

"Bump" (or impact) testing is the best way to ensure that damaging endwinding vibration does not occur on a new machine. Preferably, every 2- and 4-pole machine should be given a bump test after manufacture to find the global and local natural frequencies and mode shapes of the endwinding, at both ends. As a minimum, each model/design of generator should be bump tested to find design related problems. However, such "type" tests will not find issues associated with workmanship – for example, misplaced blocking, poor impregnation with bonding resins, missed roving, etc.

The bump test involves striking the endwinding and measuring the response of the endwinding with piezoelectric accelerometers at several locations. The equipment required includes:

- A "calibrated hammer" with a mass of about 0.5 kg that can impact the endwinding and measure the magnitude of the impact force with a transducer mounted in the hammer.
- Detection accelerometers that are temporarily bonded to the coils/bars (usually with beeswax). At least two accelerometers or one dual-axis accelerometer are needed to measure the vibration in the circumferential and radial directions. A triaxial accelerometer can provide response information in the axial direction as well.
- A Fast Fourier Transform (FFT) type of spectrum analyzer that can respond to frequencies up to about 10 kHz to simultaneously capture the force input and the three accelerometer responses producing frequency response transfer functions for analysis. Figure 7 shows a plot of the normalized acceleration as a function of frequency.
- For advanced structural analysis, software to compute the vibration mode shape tables.

Such instrumentation and software is now widely available, and compared to bump tests performed in the 1980s, the current technology is relatively easy to use. Our experience now is that the bump test, including modal analysis, can usually be done by 2 people in less than a day.

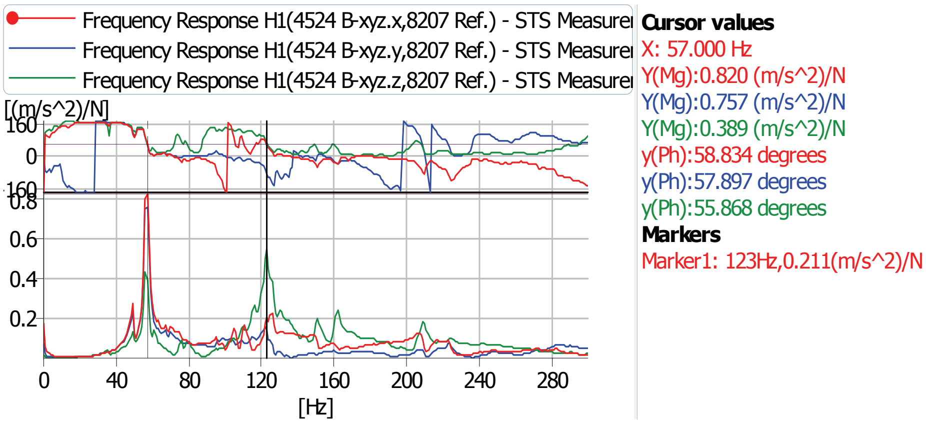

Figure 7: Bump test result from the endwinding of a 60 Hz 2-pole turbine generator. The upper plot is the phase angle of the response while the lower plot is the normalized amplitude of the response in terms of acceleration per Newton of impact force. Unfortunately, there are significant resonant peaks at 60 and 120 Hz on this machine, and significant fretting was found. The three lines represent vibration in the radial, circumferential and axial directions.

IEEE 1665 is the only consensus standard that appears to give numerical advice for acceptable bump test results, although an IEC working group is working on a guide for performing the test (IEC TC2, WG32). If the test reveals that there is a local natural frequency or a global vibration mode within about -10 Hz and +20 Hz of twice the power frequency (the "exclusion band"), and -5 Hz and +15 Hz of rated speed frequency, then it is likely that severe endwinding vibration may eventually occur in service [10]. The upper limit is higher since it allows for the decrease in natural frequency that will occur as the stator winding temperature in the endwinding increases, since the bump test itself is usually performed at about room temperature [6]. If the endwinding is expected to operate above 100°C, something that is more and more common in air-cooled generators, then an even higher upper limit may be needed [6]. Also, if the vibration response (accelerance) is greater than 0.45 m/s^2 per Newton of applied force, and near rated speed or 2f, our experience shows the endwinding may already be loose.

If the bump test results taken on the new machine are available, later bump tests can give objective information if the winding is loosening due to aging or power system transients, since the natural frequencies are proportional to stiffness. A decrease in stiffness as a result of the winding loosening will cause the natural frequencies to decrease and possibly move into an exclusion band. In this condition, the vibration levels during operation will be amplified (due to resonance), sometimes quite severely. With this in mind, bump test data can provide an indication of the resulting frequency content and relative amplitudes of the online vibration data.

On-Line Endwinding Vibration Monitoring

On generators where visual inspection has identified endwinding vibration problems, or the bump test reveals that there may be natural frequencies close to rated speed or 2f, or if other generators of the same make or model have been shown to have vibration issues, it may be worthwhile to monitor the vibration directly during operation. Up to the 1980s this involved the use of piezoelectric accelerometers that were permanently installed on the endwindings. In general, plant operators were hesitant to use such monitoring since the sensors are metallic and operating at ground potential in a high electric and magnetic field region. Furthermore, there was fear that the sensor may become loose and damage the machine, or may initiate electrical breakdown due to electrical tracking. In the late 1980s, non-metallic fiberoptic accelerometers were developed which alleviated these concerns, facilitating more widespread endwinding vibration monitoring [11,12].

There are now many types of fibreoptic accelerometers available. For endwinding vibration applications, the sensors should have the following characteristics:

- Frequency Range: 10 to 1000 Hz (since current harmonics may result in higher frequencies than 2f, and if the endwinding is loose, the impact between components will generate higher frequencies)
- Dynamic Range: 0 to 400 m/s^2
- Resolution: less than 0.2 m/s^2
- Resonant Frequency: greater than 2000 Hz
- Temperature Range: -20°C to +130°C or higher if the endwindings are expected to operate at higher temperatures (note that some fibreoptic accelerometers are very sensitive to temperature).

Early designs of fibreoptic accelerometers were not very reliable. For long term reliability, it is useful to ensure that the sensors can withstand an aging test of 150 m/s^2 at higher than expected operating temperature for many hours without their output changing.

In our experience with installing such endwinding vibration monitoring since the 1980s, it is best to install at least six pairs (radial and tangential direction) of accelerometers at the connection end. Although local endwinding vibration is less likely to be an issue at the non-connection end (since there are no long leads connecting to the circuit ring bus), three or more pairs of accelerometers are sometimes installed. It is also prudent to install a fibreoptic or conventional accelerometer on the stator core to help determine another possible source of any endwinding vibration. Choosing the optimum locations (i.e., the locations most likely to vibrate) for each pair of accelerometers needs careful thought. The best way to select locations is to perform the bump test to directly measure the locations most likely to vibrate. Some of the disputes about what the Alert level should be for high vibration may have been caused by the inconsistent installation locations of the sensors [6]. For example, installing a sensor where fretting damage is evident may result in a lower than expected vibration level, if such locations are nodes resulting in no motion at certain frequencies.

Most instrumentation outputs the peak acceleration, peak velocity, and/or peak-to-peak displacement. To date, however, most of the published data are in terms of displacement (μm peak-to-peak). Displacement is a measure of the distance moved from the initial position and emphasizes low frequencies (Figure 8). The only frequencies that should normally be detected are at rated speed and 2f. It would be unusual for any other frequency to be detected unless there are high power frequency harmonics or the windings are very loose resulting in impacts between components.

Most continuous systems output an overall vibration displacement over a wide frequency range (say 20 Hz to 1000 Hz). There is little guidance on when generator owners should become concerned at the level of endwinding vibration. About 10 years ago a recommendation of 250 μm peak-to-peak was published [11]. More recently a new draft of IEEE 1129 also suggested 250 μm as the alert level for when further investigation is needed [13]. The document developed by IEC TC2 WG32, referred to above, will also be a guide for on-line monitoring, but guidance on alert levels will not be included in the published technical specification.

The trend in maximum displacement over the years is also meaningful. If the displacement is gradually increasing over the years, then this is an indication that the endwinding support system is loosening. As mentioned previously, the trend over time is only meaningful if the data is collected at the same load and winding temperature.

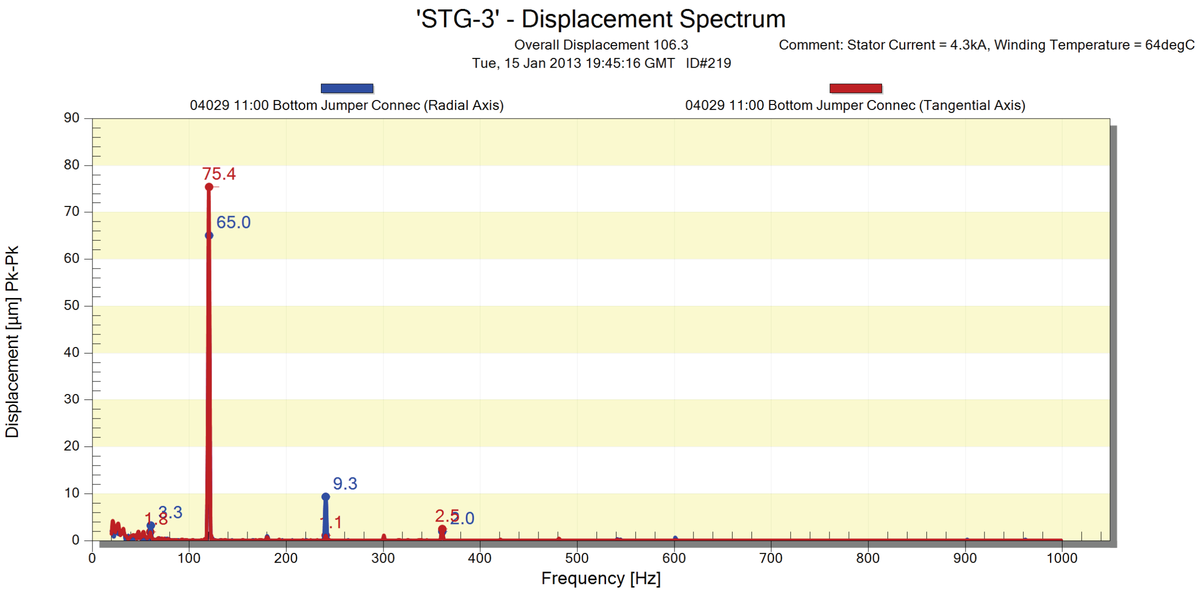

Figure 8: Displacement (0 to 90 µm) vs. frequency (0 to 1000 Hz) for a two pole, 60 Hz turbine generator. Note that the 120 Hz response (due to magnetic forces) is higher than the rotational speed forces at 60 Hz caused by stator frame vibration.

In addition to displacement, velocity and acceleration can also be displayed in frequency plots and trends. Velocity (mm/s peak) is a measure of the rate of change in displacement or the speed (and direction) of vibration and provides a smoothing effect over a wide range of frequencies. As structures loosen the response becomes non-linear and results in harmonics of the fundamental frequencies to many multiples. Harmonics also result in vibration at multiples of 100/120 Hz. The smoothing effect of velocity will provide equal weighting to the fundamental frequencies (at 50/60 and 100/120 Hz) and the corresponding harmonics. This characteristic of velocity should be considered when assessing the health of the stator endwinding.

Acceleration (g peak or rms) is a measure of the rate of change in velocity. It is the raw signal from an accelerometer. With this in mind, acceleration should not be ignored, especially at higher frequency harmonics. These may be excited by natural frequencies resulting in a resonant response that may not be present when considering the double integrated signal in displacement. The same displacement at 10 times the frequency results in 100 times the acceleration.

The increased cycle rate of vibration at higher frequencies will result in faster copper fatigue, even though the displacements may be lower. As discussed, all three measures of vibration (displacement, velocity, and acceleration) have their place when assessing the health of a stator endwinding and should be available from the vibration monitoring system.

CONCLUSION

Stator endwinding vibration has developed into an important failure mechanism, possibly due to the efforts by manufacturers and endusers to reduce costs. Additionally, load cycling machines with demand fluctuations cause additional stress on the endwinding support structure. As a result, some machines are being operated with insufficient support leading to excessive motion between parts. In order to avoid premature failure due to high cycle copper fatigue, off-line bump testing should be performed on critical machines, and/or the vibration during operation should be monitored. The sensors used to monitor endwinding vibration should be installed at locations most likely to vibrate, identified with the aid of bump testing.

Once excessive vibration in the stator endwinding area has been detected, the following remedies should be considered [11]:

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

The above should be done in conjunction with bump testing to ensure any changes made to the support system do not have a negative effect on the resulting vibration amplitudes during operation. By continuing to monitor the online vibration, an indication of whether the structural changes were positive and when additional remedies are required can be determined.

ACKNOWLEDGEMENT

The photos in Figures 4 to 6 were provided by Mr. Clyde Maughan.

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