EXPERIENCE WITH TORSIONAL VIBRATION AT A 600 MW TG UNIT

Jurek Ferleger, Turbine Specialist/Consultant
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ABSTRACT

Turbine-generator power trains employ multiple large rotors. The rotors are connected through relatively flexible shafts and couplings. The turbine rotors feature a number of bladed discs that have substantial torsional inertia. The generator rotor also contributes substantially to torsional inertia and stiffness of the power train. Consequently, a large turbine-generator shaftline exhibits a substantial number of relatively low natural frequencies. The transverse rotor vibrations can be minimized with precision balancing, and can be measured with standard plant equipment. However, the torsional oscillations, characterized by the angular vibratory twisting of the rotor about its centerline, which are superimposed on its angular velocity, cannot be readily detected. Furthermore, damping in the torsional vibrations is usually very light. Thus, substantial undetected torsional response may be present in some units. This response is of special concern if a particular torsional mode is close to the double line excitation (120 Hz in a 60 Hz electrical grid). This is because the transmission line phase imbalances and/or load imbalances may cause significant torsional excitation at twice the speed of the rotor. Equipment manufacturers attempt to avoid resonance at 120 Hz through complex analyses and testing of their units. Nevertheless, some older operating units may have a natural torsional mode close to the double line excitation. While a few units have experienced distress or even catastrophic failure due to such coincidence, other ones have long successful operating history in spite of such coincidence. This paper discusses an experience with torsional vibration issues at a fossil plant in South Central United States. The unit exhibited a torsional resonance in the vicinity of 120 Hz but has not experienced distress associated with torsional vibrations.

INTRODUCTION

The aforementioned 600 MW unit was designed for 1000 °F and 2415 psiA inlet temperature and inlet pressure, respectively. The station started commercial operation in the early 1990s. The power train consists of two Double Flow Low Pressure (DFLP) sections and a High Pressure – Intermediate Pressure (HPIP) section. The axisymmetric rotating components are joined through couplings and tightly fitted coupling bolts. The DFLP rotors are also symmetric about the inlet planes, shown in red in Figure 1. The unit is equipped with a 629,000 KVA, 24,000 V hydrogen/water cooled generator.

The subject unit is currently undergoing a retrofit of LP elements. During the process of composing a turbine procurement specification and bid evaluation, it was realized that the unit may have a torsional mode in the vicinity of the double line excitation. However, no distress associated with this condition has been identified during regular inspections. This paper outlines the experience of the subject unit with a torsional mode coincident with 120 Hz excitation and offers an explanation for the absence of any distress associated with this response.

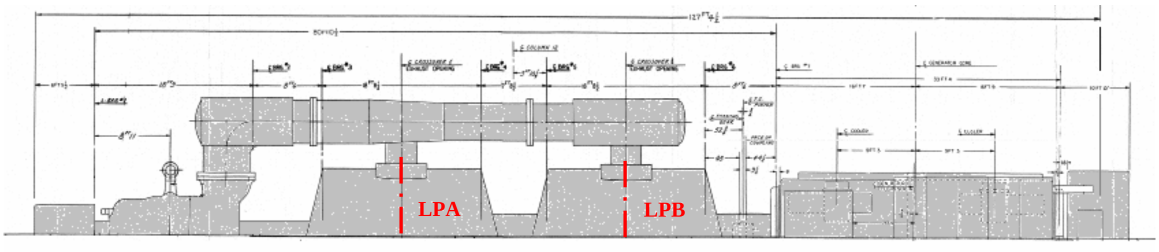


Figure 1: Turbine Generator Mechanical Outline

OUTLINE OF TG SHAFTLINE VIBRATORY CHARACTERISTICS

During start up and under steady state operation, turbine rotors are subject to many complex excitations which may arise due to steam flow irregularities, rotor imbalance, electrical grid perturbances and other phenomena. The response to these excitations depends on the available damping, energy cancellation effects (further explained in the following sections), etc. These effects are different for each particular mode. Vibration of turbine rotating components can be broadly divided into the following classes:

- **Axial or longitudinal vibrations (tension-compression of the shaft):** This type of vibration is not usually of major concern to operators of large Turbine Generator (TG) sets. In an unlikely situation when longitudinal vibrations are present, these are identifiable by the thrust bearing proximity probes.

- **Flexural vibrations:** This type of response is characterized by the bending of the shaft in the transverse direction. The transverse vibrations can usually be detected by bearing vibration sensors. Large vibration must be prevented at the operating speed of the TG unit, as outlined in ISO 7919-2 [1] and ISO 10816-2 [2]. Equipment manufacturers “tune” the first (and second, if applicable) lateral speed at least 10% away from the operating speed of the turbine.

- **Torsional vibrations**: These vibrations are characterized by the twisting of the rotor about its centerline, superimposed on its angular velocity. Large turbine generator rotor trains exhibit many torsional vibration modes with natural frequencies below or near double frequency of the grid (120 Hz). These types of shaftline vibrations cannot be detected with standard plant monitoring equipment. The mode shapes vary from a simple twist with a node in the middle of the rotor to a complex motion with substantial last stage or penultimate stage blade participation. (For a unique blade frequency, there are a number of similar system torsional modes because of differences in blade deflection patterns at the opposite ends of the rotor, i.e. in phase, out of phase, etc.) Other shaftline torsional modes exhibit substantial generator and/or exciter twist. A measurable twist at the turbine couplings or the stubshaft may also be present. Furthermore, damping in torsional vibration modes is very low. Thus, if excitations are present, high levels of stress may be reached (however low damping also results in sharp peaks which means that small difference between an excitation and a natural frequency is sufficient to avoid resonance). Consequently, prudent design practice requires sufficient separation between the torsional frequency and excitation frequency, as outlined in ISO 22266 [3] and Nuclear Industry Insurance ("NEIL") Guidelines [4]. Figure 2 outlines conceptually typical torsional modes of a power train composed of HPIP rotor + 2 DFLP rotors + Generator. Similar to the flexural modes, the fundamental mode has a single node (point of zero angular displacement), the second mode has two such nodes and so on. A higher mode, likely to incorporate large participation of the last or penultimate row of blades is also shown in Figure 2.

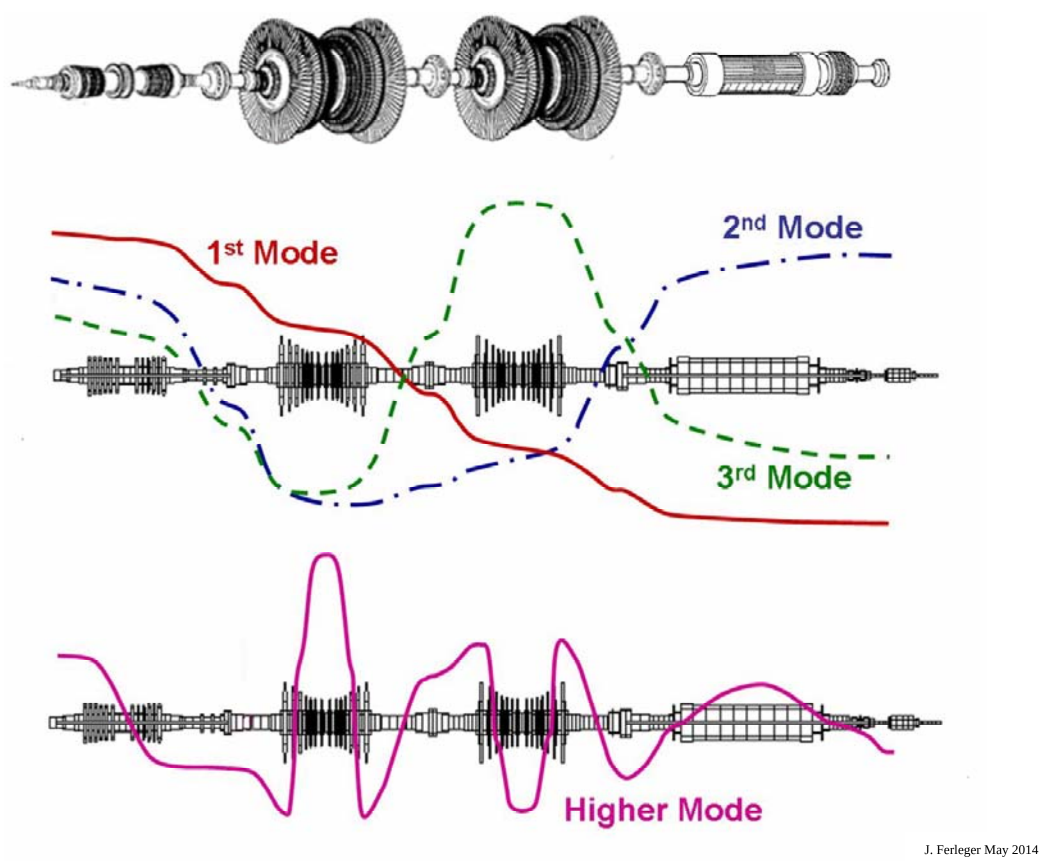


Figure 2: Conceptual Outline of Torsional modes of a power train consisting of an HPIP, DFLP, DFLP and a Generator

- **Blade vibrations**: Synchronous blade vibrations are typically illustrated on a Campbell diagram, on which the frequency of each blade mode is plotted as a function of turbine running speed (frequencies usually increase as a function of turbine speed because of the increasing stiffening effect of the centrifugal force). Each blade mode can actually be characterized as a family of modes which includes a zeroth nodal diameter mode (a.k.a. umbrella mode), 1st nodal diameter mode, second nodal diameter mode, etc as shown in Figure 3. All the subject modes, except the "infinite nodal diameter mode", couple with disc frequencies. The 1st, 2nd, etc, nodal diameter modes can be excited when turbine speed coincides with the 1st or 2nd multiple of turbine speed, respectively, usually referred to as 1st or 2nd harmonic. The umbrella modes, which cannot be shown on a Campbell diagram, can only be excited by electrical grid perturbances, which exert fluctuating torques on the generator. Each blade row has an umbrella mode corresponding to the 1st mode, 2nd mode, etc. Typically, only the two lowest umbrella modes are evaluated as a part of rotor torsional response. However, as mentioned in the previous bullet, for a large power train which includes 2 DFLPs, a number of similar umbrella modes, which correspond to the first or second blade mode, may exist. This is because, for each rotor, both an in-phase and out-of-phase umbrella modes on opposite discs in the same rotor, typically with very similar frequencies, can be excited.

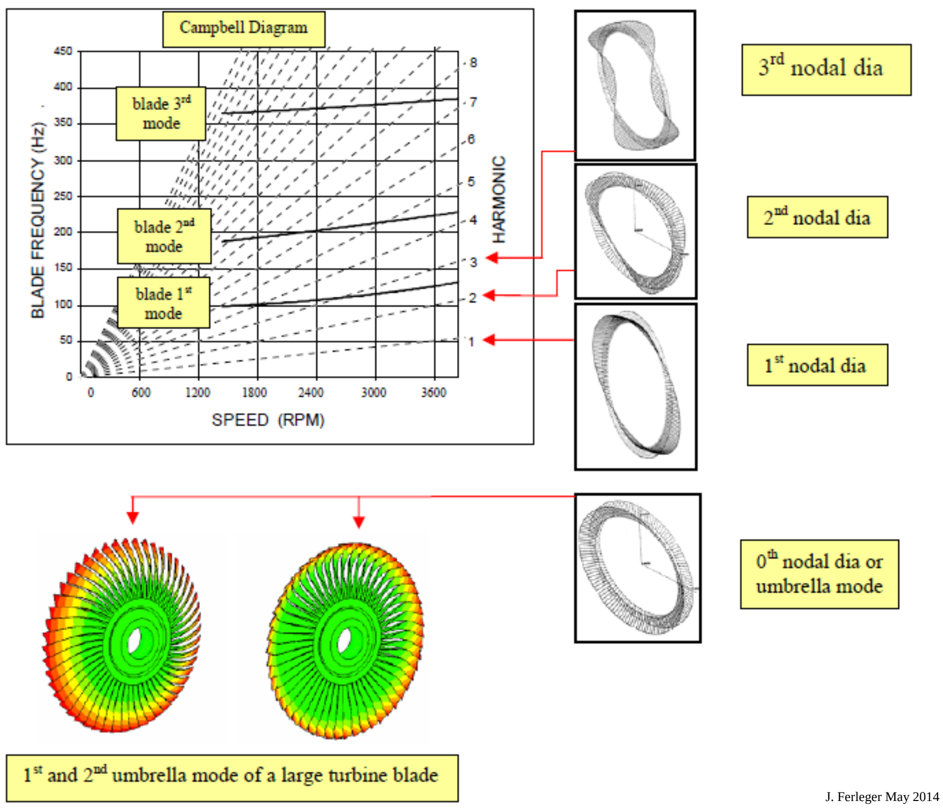


Figure 3: Outline of Turbine Blade Vibratory Characteristics

OUTLINE OF ELECTRICAL TORSIONAL STIMULI ON TG SHAFTLINE

Torsional vibrations of the turbine generator power train cannot be excited by steam flow or rotor imbalance. Torsional stimuli on the turbine generator shaftline arise due to electrical imbalances imposed on the generator rotor.

Normally, a three-phase balanced supply voltage, applied to a set of symmetrical three-phase windings, generates a constant-magnitude flux in the airgap of the generator. Such flux rotates at synchronous speed around the circumference of the machine. However, during any perturbance, such as line-to-ground fault, the supply voltage or currents become unbalanced. (During an electrical imbalance, the currents in the faulted and intact phases incorporate three 60 Hz components, referred to as zero, positive and negative sequence currents.) As a result of an electrical fault, an additional flux, which matches the fundamental frequency of the system, appears in the airgap of the generator. This flux rotates in the opposite direction from the generator rotor and thus induces a voltage and body current at twice the fundamental frequency of the rotor windings. Such current is referred to as negative sequence current, designated as I_2. The larger the imbalance, the higher the I_2 component. (Under a balanced electrical load, there is a steady positive sequence current and the negative and zero sequence currents have zero value.)

Positive and negative sequence currents affect the generator rotor differently. When the negative sequence current is imposed on the stator, the rotor, which rotates at 60 Hz, is subject to the magnetic field that rotates 60 Hz in the opposite direction (hence the name "negative sequence current"). Therefore, the generator rotor is temporarily exposed to a 120 Hz rotating magnetic field. This introduces negative sequence currents, which tend to flow close to the surface of the rotor. Such fluctuating negative sequence currents produce fluctuating torques, which act on the generator rotor, at twice the grid frequency. While the permissible values for continous I_2 currents to be withstood by a generator without injury are contained in ANSI/IEEE C50.13, no equivalent I_2 thresholds have been published to minimize High Cycle Fatigue (HCF) damage in turbine rotating components.

The frequency of the positive sequence current, which rotates in the opposite direction from the I_2 current, is 60 Hz. A change in the positive sequence current arises whenever a there is a mismatch between the electrical load demand and supply. Electrical excitation at 60 Hz can excite a rotor twisting mode which has a frequency close to 60 Hz, if the mode shape includes generator participation. However, the 60 Hz torque decreases rapidly to negligible values. Therefore, the 60 Hz stimulus disappears quickly, before any significant excitation of the rotor train occurs, and is ignored by some turbine manufacturers. The 120 Hz transient torque, on the other hand, does not decay as rapidly as the positive sequence torque. In general, it is truncated by the generator protection system. However, it may be sustained for a few cycles until the breaker protection system intervenes. Therefore, this torque can potentially result in a large transient response in the rotor train before the fault is cleared.

The negative sequence current torques can become quite large during a grid transient, particularly if the grid fault is not cleared by protective relays in a timely manner. The large torques act as impulse loads on the unit and may excite many torsional modes regardless of their natural frequency. However, individual mode response depends on its excitability. For example, a torsional mode that includes in-phase generator rotor twist will be readily excited by an impulse load, whereas a mode with little or no generator participation will not be excited. In addition, a mode with a node in the middle of the generator rotor span cannot be excited by a uniform excitation force due to torque cancellation. This effect is conceptually illustrated in Figure 4 where it can be observed that a uniform excitation force is out of phase with a mode of the rotor, i.e. half of the excitation force tends to excite this particular mode, whereas the other half of the same force tends to suppress it. Thus, a mode with a node at the middle of the generator rotor, for which the twist of each half of the generator body is equal but 180° out of phase, cannot be excited by a uniform air-gap torque acting along the length of the generator body. This is true even if the damping in the particular mode is negligible [5].

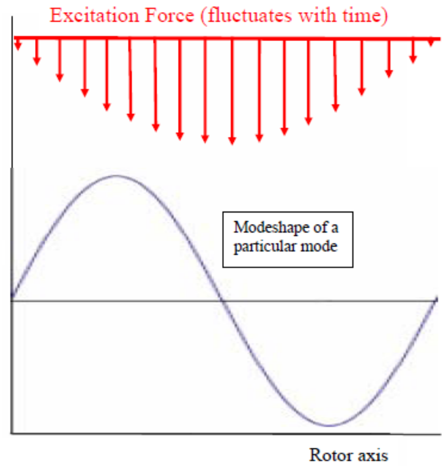


Figure 4: Outline of Torque Cancellation Concept

DESIGN CONSIDERATIONS

All Original Equipment Manufacturers (OEMs) impose frequency separation requirements between the electrical excitations and shaftline torsional frequencies. The specific design criteria of each OEM are proprietary and are not entirely consistent. Some manufacturers consider the double line excitation only whereas others avoid the single line excitation, as well. A number of manufacturers will allow a resonance close to the 120 Hz double line if it can be shown that the response level associated with such mode is acceptable. The recently issued ISO 22266 outlines recommended industry standards, summarized in Table 1 for the 60 Hz market. In the ISO standard, the separation requirements depend on the available frequency verification testing. In addition, some basic requirements have also been proposed by the Nuclear Electric Insurance Loss of Control Manual [4]

Table 1: Summary of ISO 22266 Frequency Separation Requirements for 60 Hz Network

Verification requirements	Single line separation requirements	Double line separation requirements	Separation requirements as %
None	± 3.6 Hz	± 7.2 Hz	± 6%
Full-speed shop dynamic test on the generator rotor and a static test on the adjacent LP turbine rotor	± 3 Hz	± 6 Hz	± 5%
Full speed shop dynamic test on the generator and the associated LP	± 2.4 Hz	± 4.8 Hz	± 4%
Full-speed test carried out on the fully installed shaft system	± 2.1 Hz	± 4.2 Hz	± 3.5%

Experience has showed that torsional frequency analytical models and the resultant calculations can be assumed to be accurate within ± 2%. This is a fairly typical tolerance in determining the frequency of a complex system such as a turbine rotor shaftline. A similar margin of error is often assumed in calculation of the frequency of a large turbine blade, where both bending and torsion are likely to be present. It can therefore be surmised that it is not possible to calculate the higher natural torsional frequencies accurately. However, since the resonance peaks tend to be sharp due to low system damping, the probability of the excitation frequency coinciding with the response frequency is low. Unfortunately, as outlined in the following section, the consequences of a such a coincidence and the resultant torsional resonance, in particular at 120 Hz, can be severe.

SUMMARY OF INDUSTRY EXPERIENCE

The oldest reported failure due to torsional resonance occurred at Cleco Coughlin fossil plant in 1968. LP blade failures at the plant were attributed to high generator phase imbalance. Another series of events, which were attributed to torsional resonance, occurred in 1973-1974 at Prairie Island Nuclear Plant. Three separate L-2 and L-1 blade loss incidents, which resulted in forced outages, were reported. To address the torsional resonance problem, the L-2 blade was redesigned, the disc under the L-1 blade was re-contoured and a weight was added at the turning gear location [7].

The most significant torsional event occurred in 1986 in the Far East [8]. No information that clearly identifies the cause of this incident is available in the public domain. However, it is known that a number of last stage blades were lost. The resulting significant imbalance caused large rotor vibrations which, in turn, caused the bearing caps to fail. A lubricant fire ensued. The affected unit and a sister unit were retrofitted by a competitor.

Two additional torsional vibration incidents were reported in 1986. Details of these incidents have not been published. Two torsional incidents also occurred in 1993. (In one of these events, the penultimate blade failure caused a forced outage. The failure was subsequently attributed to 120 Hz torsional resonance. Addition of an inertia ring to de-tune the unit by approximately 3 Hz was recommended by the manufacturer).

A complex turbine generator system may have up to 25 torsional vibration modes below the excitation frequency of 120 Hz, including several frequencies near 120 Hz [9]. Welded rotors, which are stiffer, feature fewer torsional modes in the spectrum of interest. This simplifies the process of tuning away from torsional excitations.

OPERATING HISTORY OF THE AFFECTED UNIT

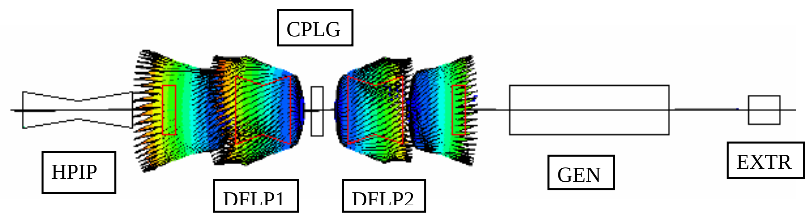
The unit that is the subject of this paper has over 20 years of operating history. A Non Destructive Examination (NDE) was performed in 2011. No high cycle fatigue cracks, which could be attributed to torsional vibrations, were found [11].

A decision was made to replace the LP rotors for unrelated reasons. During the preparation of the procurement specification and subsequent bid evaluation, it was realized the currently operating turbine generator power train may have a torsional mode in the vicinity of 120 Hz. Although the HPIP element had been replaced by a non-OEM in 2002, the retrofit is very unlikely to have affected the torsional frequency of the generator in a measurable way.

Figure 5 outlines the summary of torsional frequencies provided by the manufacturer and illustrates some torsional modeshapes of interest. As it can be observed, the calculation predicts the 2nd generator mode to be in the vicinity of the 120 Hz double line excitation.

Mode	Description	Frequency (Hz)
1	System	18.56
2	System	32.95
3	System	39.52
4	L-0 Blade Out-of-Phase	112.46
5	L-0 Blade Out-of-Phase	114.17
6	Generator Twist	119.18
7	L-0 Blade In-Phase	128.05
8	L-0 Blade In-Phase	128.32
9	L-1 Blade Out-of-Phase	159.74

Some modes, including the generator 1st twisting mode, are in the range of 39.52 -112.46 Hz. These modes were not provided by the OEM



Conceptual Depiction of One of the Out-of-Phase Last Stage Blade Umbrella Modes

Figure 5: Summary of the torsional modes provided by the OEM

SUMMARY OF TORSIONAL TEST FINDINGS

To determine a “baseline” vibratory characteristic of the unit before the LP retrofit, a Torsional Vibration Test was performed on the turbine generator shaftline. Three telemetry collars were secured, one each at the bearing between the two LP rotors and at the two bearings adjacent to the generator, as shown in Figure 6. (Two sets of strain gauges were attached to the shaft 180º apart at each collar [6].) The strain gauges were mounted at locations where, based on analytical predictions, the highest torsional deflection was expected in the mode of interest. The unit was tested at sub-synchronous and at synchronous speed, at various load points. The following observations were made after a detailed review of test results:

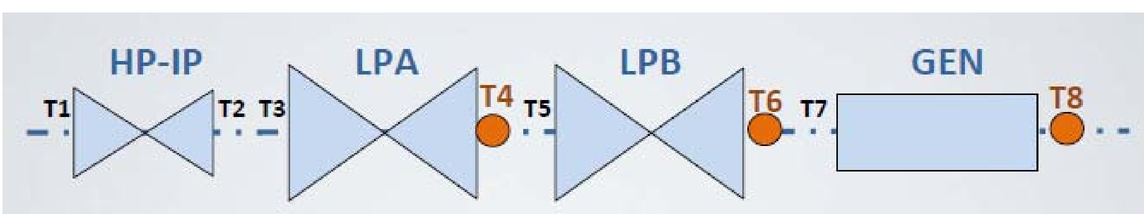


Figure 6: Location of torque collars

- A torsional mode was detected in the vicinity of 120 Hz at the initial three test points: (1) 3000 rpm, (2) Full Speed No Load (FSNL) conditions, and (3) 10 MW load.

- The subject mode was primarily detected at the bearings adjacent to the generator. No measurable response was observed at the bearing located between LPA and LPB. This corroborates the fact that the subject mode is a generator twist mode, outlined conceptually in Figure 7. (The manufacturer predicted a 2nd generator twist mode at 119.2 Hz.)

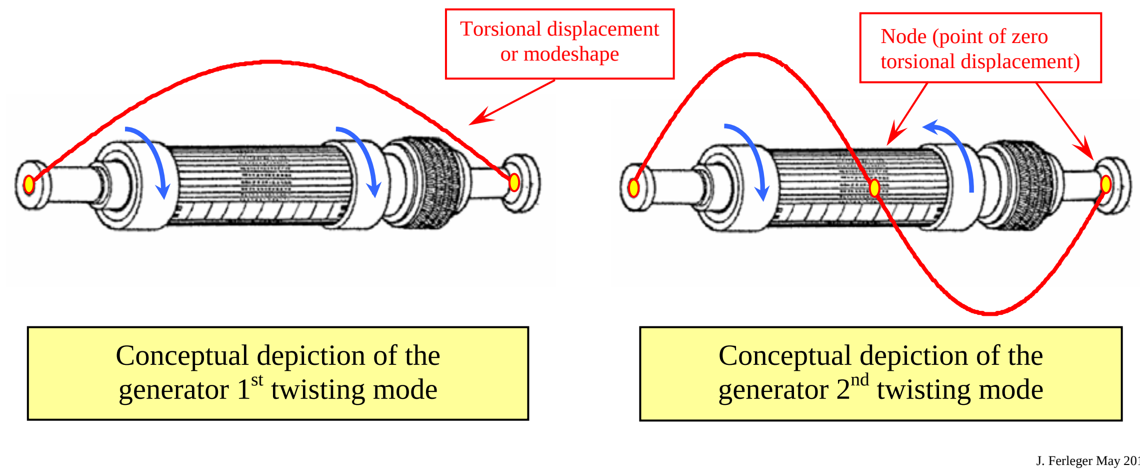


Figure 7: Outline of generator 1st and 2nd twist modes

- Compared to the first three fundamental system modes identified in the table in Figure 5, the stress level associated with the generator second twist mode appears low. This can be clearly detected in Figure 8. (This observation is particularly significant given the fact that the strain gauges at T6 and T8 bearings were strategically placed to detect the 2nd twist mode of the generator.)
- The L-0 out-of-phase and in-phase bucket modes were detected at 112.5 Hz and 128.6 Hz, respectively, at 10 MW load test point, summarized in Figure 8 below. The predicted frequencies of these modes at full load are given in the table in Figure 5 (as described in the following section, the torsional frequencies at 10 MW and at full load conditions are slightly different.)

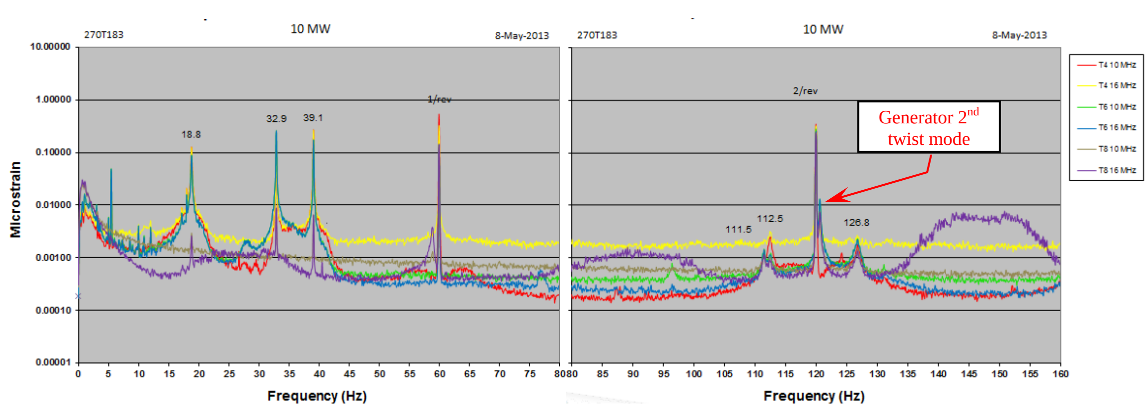


Figure 8: Torsional frequency spectrum as measured during the test at 10 MW load [6]

- The 2nd generator twist mode merges with the with the "2/rev" electrical spike, identified in Figure 9, at all test points at loads higher than 10 MW as shown. (The frequency of 1st generator twist mode, depicted in Figure 7, was not provided by the OEM.)

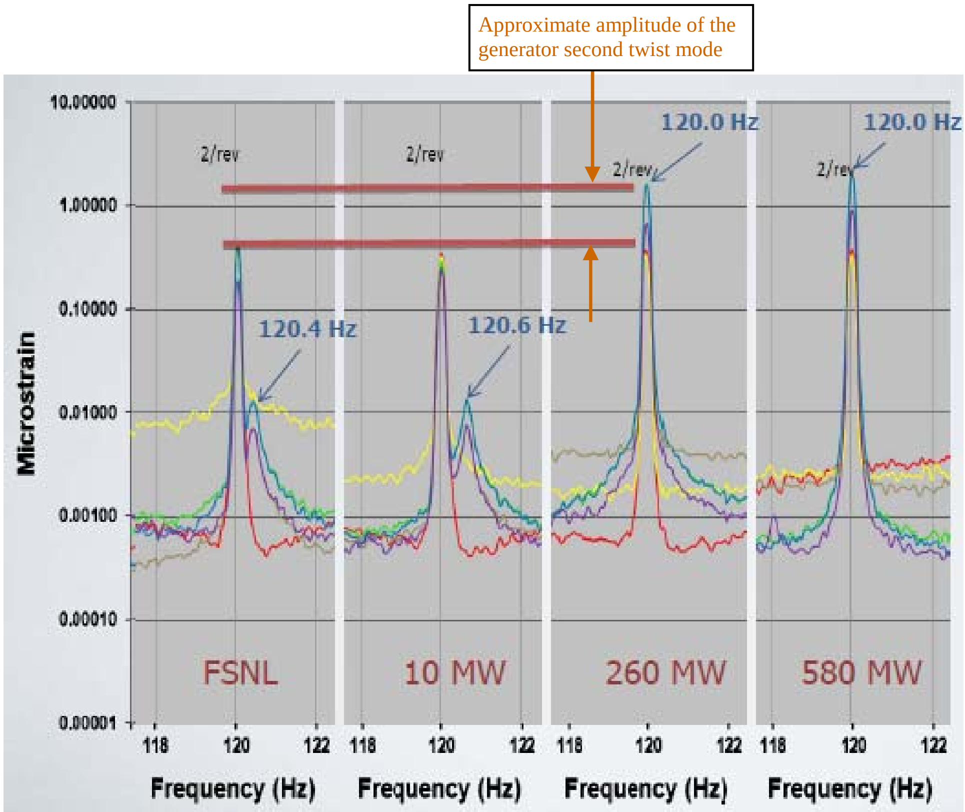


Figure 9: Partial summary of the torsional test [6]

DISCUSSION OF THE TORSIONAL TEST RESULTS

The OEM conducted a "monitoring test", which is performed while the unit is on line. Therefore, it cannot be completely ruled out that some non-torsional modes were detected by the test instruments. However, the test results match the predicted frequencies closely. Thus, it can be concluded with a high degree of probability that the unit features a 2nd generator twist mode in the close vicinity of 120 Hz. The frequency of this mode increases slightly between the full-speed-no-load test point and the 10 MW test point, as demonstrated in Figure 9, probably due to increase in stiffness of the generator rotor, which increases with increase in electrical torque. The same frequency drops somewhat at the higher load points because, as the load increases, so does the average temperature of the rotor. This results in a small drop in the Young's and Shear moduli and hence in a small frequency drop (Based on material properties given in ASME B31.1, Table C-1 [11] and the fact that the torsional frequency is a function of square root of Shear Modulus, a moderate temperature increase of approximately 50 ºF would cause a 0.5% frequency increase.) With the temperature effect included, the generator 2nd twist mode is practically indistinguishable from the 2 rev (120 Hz) peak. This would not be considered acceptable on a new unit, unless it can be unequivocally shown that the resultant response will not cause fatigue cracks within the life of the unit.

An analysis of whether the stress at the critical location in the subject mode is sufficient to cause fatigue failure can be performed [11]. Given the measured strain, it is possible to determine the actual vibratory stress at the highest stress point for this mode. Imposing the result on a Goodman diagram at an assumed or measured value for the negative sequence current should yield the accumulated fatigue damage, as depicted schematically in Figure 10.

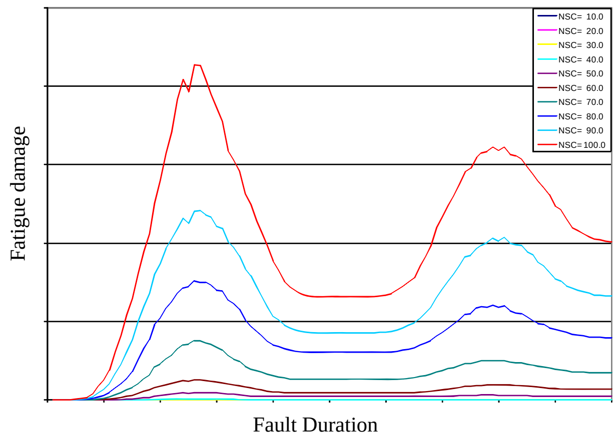


Figure 10: Fatigue life usage as a function of time and negative sequence current (courtesy of MPR)

In addition, a product line evaluation, if performed by the OEM, can provide insight into the performance of other similar units.

MITIGATING FACTORS

As stated before, in spite of the fact that the 2nd generator twist mode coincides with the double line excitation of 120 Hz, the affected unit has not demonstrated any distress associated with torsional response. The explanation for this can be divided into two categories:

- **Absence of significant torsional stimulus/negative sequence current at the affected unit**: As can be deduced from Figure 10, fatigue damage will not occur unless a significant level of negative sequence current, acting on the generator rotor for a measurable length of time, is present. Absence of serious grid disturbances, i.e. absence of any significant torsional excitations, was given by the manufacturer as the main explanation for the absence of high cycle fatigue cracks, which can be attributed to the torsional excitations, during the life of the unit.
- **Torque cancellation effect**: Detailed mode shape of the generator 2nd twist mode was not provided by the manufacturer. Thus the location of the node, present in the middle of the generator, and the degree of torque cancellation in the particular mode, is not known. However, it is reasonable to assume that a fair amount of torque cancellation, illustrated in Figures 11 and 12, is present in the generator 2nd twisting mode. Therefore, energy input into the remainder of the power train is significantly reduced. The torque cancellation effect is likely to contribute to the fact that the amplitude of this mode, measured during the torsional test, was small compared to the amplitudes of the primary system modes.

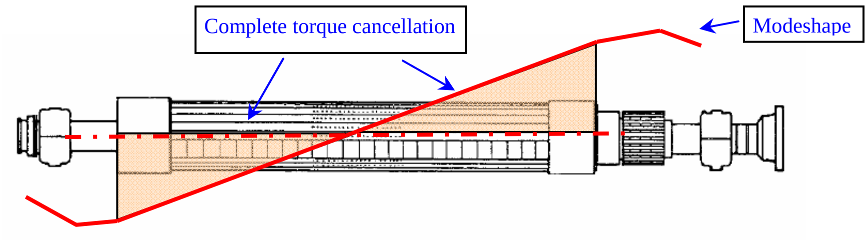


Figure 11: Mode with complete energy cancellation (zero net resultant torque)

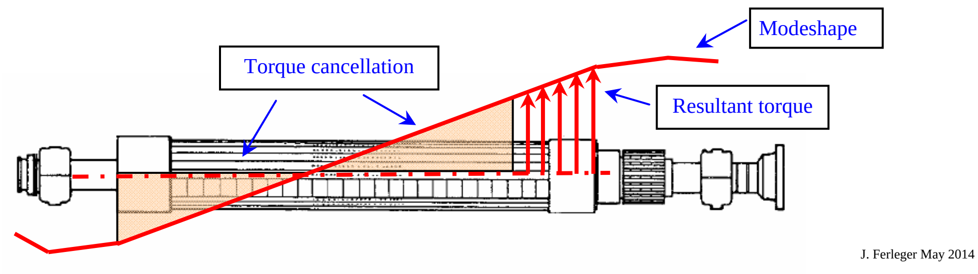


Figure 12: Mode with partial energy cancellation (non-zero net resultant torque)

SUMMARY

Presence of a torsional mode coincident with the double line excitation of 120 Hz has been predicted by calculations and essentially confirmed with a torsional test on the operating unit. However, consistent absence of any distress that can be attributed to such resonance in over twenty years of operation suggests that the response in this mode may be insufficient to initiate high cycle fatigue cracking of the rotating components. The absence of high cycle fatigue cracks can be attributed to energy cancellation effects which tend to suppress such mode and/or low level grid electrical excitations at this unit. Nevertheless, as recommended by the manufacturer for this unit, a coincidence between the 120 Hz electrical excitation and any generator twist mode should be avoided at any operating plant. Adhering to guidelines outlined in ISO 22266 is highly recommended.
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