Vibration Diagnostics and Condition Assessment As Economic Tool

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ABSTRACT

Based on the interpretation of field vibration data, the four pole generator rotor from the generator rated at 1150 MW was declared by the OEM as defective and in need of a rewind. Since the rewinding process could take up to 6 months, which is too long for a Power Plant to be idle, the OEM recommended to the client to order a new rotor with a lead time of two years and at a price of $6,000,000.00. With a new rotor on hand, the exchange could be done in 4 weeks during a normal reactor re-fueling outage.

This paper deals with re-evaluation of the customer's field data upon which the recommendation was based, and with the additional vibration data collected by the author under controlled operating conditions needed. The additional data was needed to find the root cause of the currently observed generator rotor vibrations. Further verification was done by Finite Element Method modeling of the rotor, calculating its critical speeds, its elastic constant and simulating the effect of balancing.

The results of the data evaluation was contradictory to the OEM findings, and it allowed the rotor to continue operation until some assembly correction could be made within the time frame of a normal re-fueling outage. The total savings were at least $6,000,000.00 by avoiding the unnecessary expenditures for a new spare rotor.

KEYWORDS

Rotor Dynamics, Vibration, Measurements, Analysis, Generator

INTRODUCTION

In the world around us, and so it is in the technical world, each and every event becomes evident via symptoms.

In the technical world, symptoms of any event should be used only as a piece of the puzzle on the way to finding the root cause of that event. Root cause is defined only then, when there are no more questions to be answered about the event. Everything else is an assumption. Unfortunately it is a widely spread fact that often the assumptions, with large doses of "experience," are substituted for the "Root Cause".

All activities on this Earth sums up to events, symptoms, and root causes. Most often people in every field of activity confuse "symptoms" with the root cause. Symptoms from some activity are often misused to create an atmosphere, e.g., fear, for the purpose of leading others in the direction of their choosing for achieving their own objectives. But, that seems to be human nature. This paper deals with one such case.

The generator rotor, rated at 1150 MW at a nuclear station, had experienced some unexplained vibration changes. Before following the OEM recommendations and spending $6 to $10,000,000, the plant owner wanted a "second opinion". The first task was to isolate the symptoms and to define the root cause. For that it was necessary to acquire vibration data from various sensors and under various speed transients and operating load conditions. The data collected for generator and exciter rotors under available operating conditions was analyzed, and symptoms were separated into those of mechanical and electrical origin.

AC Generator– Hydrogen Cooled

RATING	@60 PSIG Hydrogen Pressure
kVA	1,333,200
Stator Voltage	25,000
Stator Amperes	30,789
Power Factor	0.90 1,800 rpm, 3 phase, 60 Hz
Excitation Ampers	7,461
Excitation Voltage	450
Temperature rise not to exceed	Stator <50ºC, Rotor <64ºC

MECHANICAL EVALUATION

a) Slow Roll Runout (<40 RPM)

Although considered very important to the Author, it was not possible to record the crucial data of DC gaps at standstill since the Unit was already on turning gear before testing time. The Unit was rolling at ~40 RPM.

DC gap at standstill was necessary to evaluate bearings' alignment. Slow roll runout at 40 RPM with multiple samplings per each revolution gave us the journals' TIR and a rotor coupled eccentricity.

The largest value was 0.001" (1.00mil) at bearing #10. (Fig. 1a). For a rotor this size (i.e. weighing 389,000 lbs) that is equivalent to approximately 6.0 lbs of unbalance at the rotor OD. The centrifugal force of the equivalent balancing weight could correct the elastic deformation amplitude, but it would have NO effect on the eccentricity displacement amplitude measured. The eccentricities can originate from exceeding OEM machining tolerances, forging deformation in operation, or from the assembly. Some effect of the eccentricities can be corrected by high speed balancing in the bunker (more access planes than in the field), some must be corrected by machining and for some, the operating condition must be corrected (proper cooling, or elimination of shorted turns).

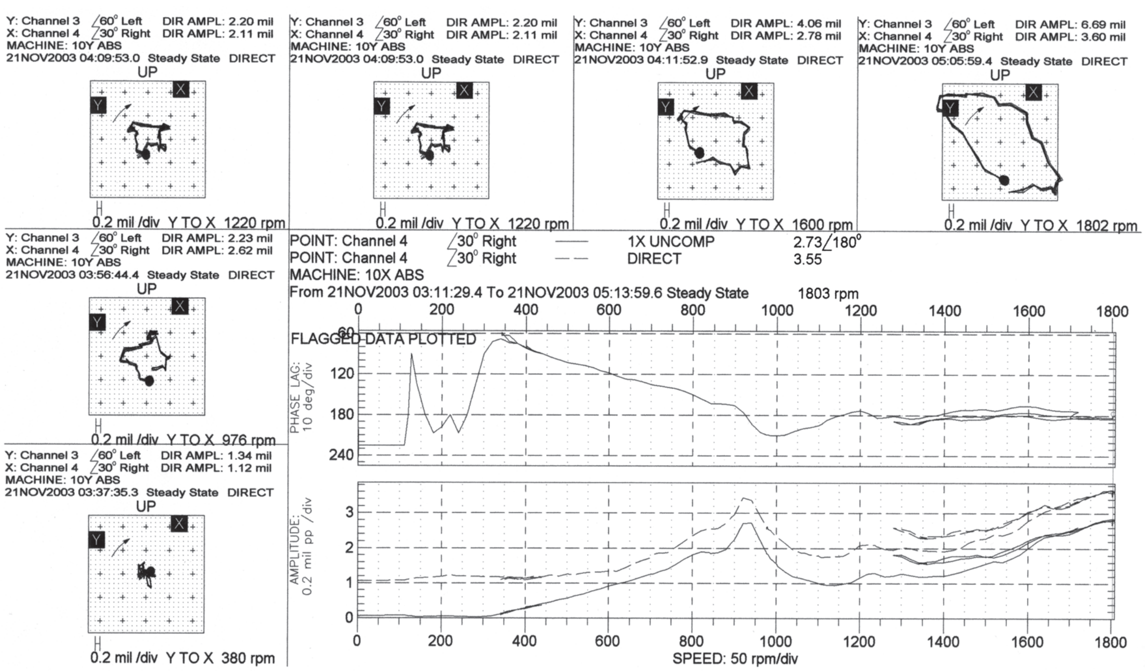


Figure 1a: Bode Plots Bearing 10X

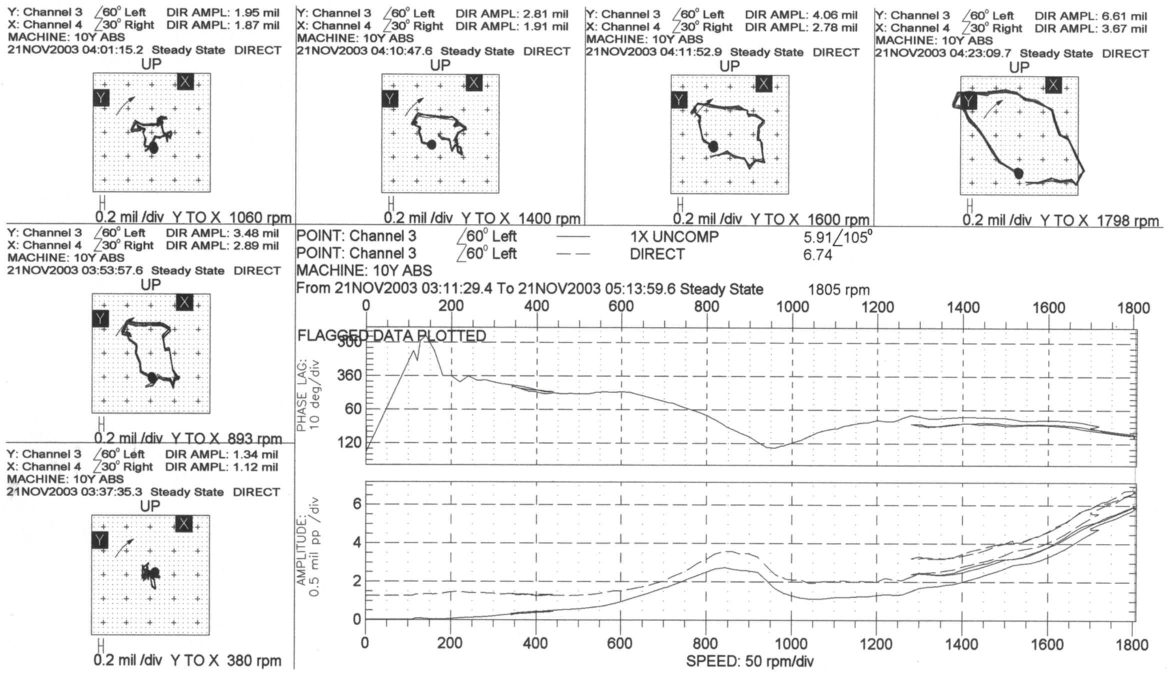


Figure 1b: Bode Plots Bearing 10Y

b) Slow Roll Runout (<200 RPM)

These values are visible on Bode Plots and indicate the eccentricity (pk-pk) at the measuring location of the journal. The eccentricities are the fundamental cause of all vibrations. The largest value was measured at journal #10Y at 0.001” (1.0 mil). It is significant that this eccentricity was visible also in the form of residual eccentricity in the transition speed range observed from 1st to 2nd critical, at journals #9 and #10. That confirms that the measured runout at low speed was not just an irregularity at the measuring location, but the true rotor eccentricity. (Fig. 1a)

c) Run-up, Critical Speeds and Overspeed

First critical speed is visible at ~920 RPM peaking at maximum of 0.004” (~4.0 mils) (pk-pk direct). This is an indication of a dynamically very well balanced rotor with good alignment. (Fig. 1a, 1b).

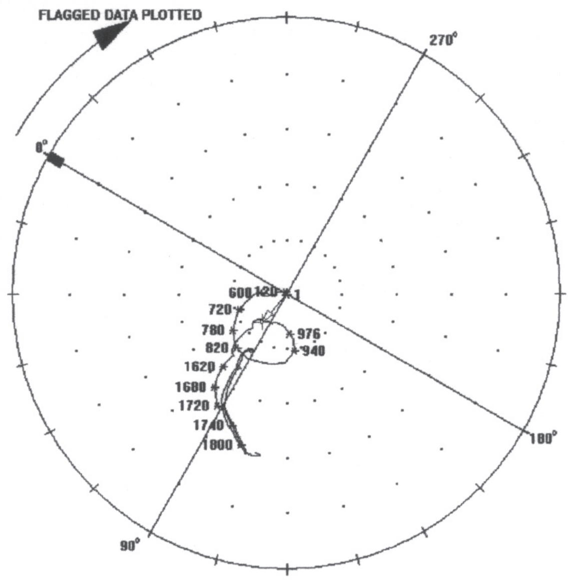
Coming out of the 1st critical, “vibrations” are not resolved, but remain at 0.001” (1.0 mil) because of the existing eccentricity, which causes an additional excitation of the 2nd critical (rocking mode).

Interesting here is that this “unbalance” at bearing #10 (relative) does not change phase through the speed range from 1000 RPM to 1800 RPM, while the “vibration vector” is proportional to square of speed. The lack of phase change differentiates an eccentricity-caused displacement from a “dynamic” or elastic response and corresponding phase lag. (Fig. 1a, 1b.)

d) The effect of “Loose” Bearing

It is important to know that some OEMs design their bearing assemblies with liberal side to side and top clearances. This feature in itself is not significant, but it allows for bearing “rocking” when sufficiently excited by shaft displacement. Shaft measured displacement may indicate elastic deformation of the rotor, eccentricity “whirling” at higher speeds, or “wobbling” at lower speeds due to bearing looseness. The effect of loose bearing is indicated by very different vector pattern seen on Polar plots for “shaft absolute” and “shaft relative” vibration signals. (Fig. 1c, 1d).

Based on all information so far, it was determined that some eccentricity exists on the rotor, large enough to cause the observed vibration symptoms.



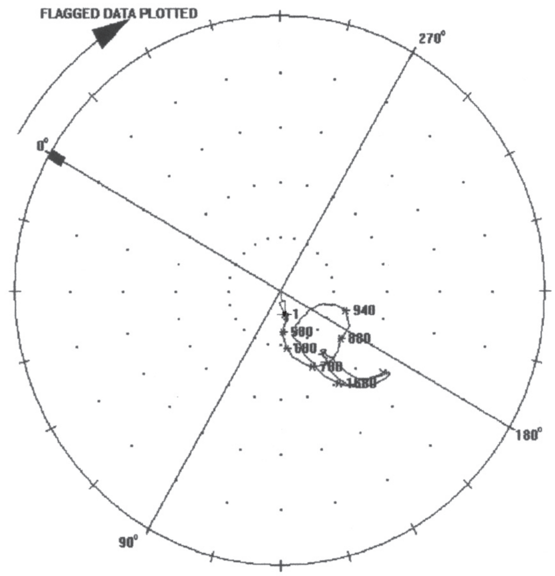


Figure 1c: Polar Plot Bearing 10Y– Absolute **Figure 1d: Polar Plot Bearing 10Y– Relative**

FE MODELING

In the past, the operators at the plant attempted to "balance" the elevated shaft amplitude at bearing #10. In general, the purpose of "Balancing" is a process to reduce the forces generated from an "unbalanced" rotor, transmitted to a bearing and its structure, best expressed in units of inch/sec. A rotor displacement defines rotor motion and should be analyzed separately for better understanding of the total dynamic picture. The guiding criteria for determination of vibration severity should be pedestal vibration.

The generator rotor was modeled using DYROBES®, a FE software. The purpose of using a FE model was to save the cost for physical testing. Verification of the model validity was done by calculating rotor sag and critical speeds. (Fig. 2-5)

After verification, a simulation of the unbalance response was done. A simulated shot of 100 oz. was placed in the model at the GEN-EXC coupling i.e., at location where operators were placing the "balance

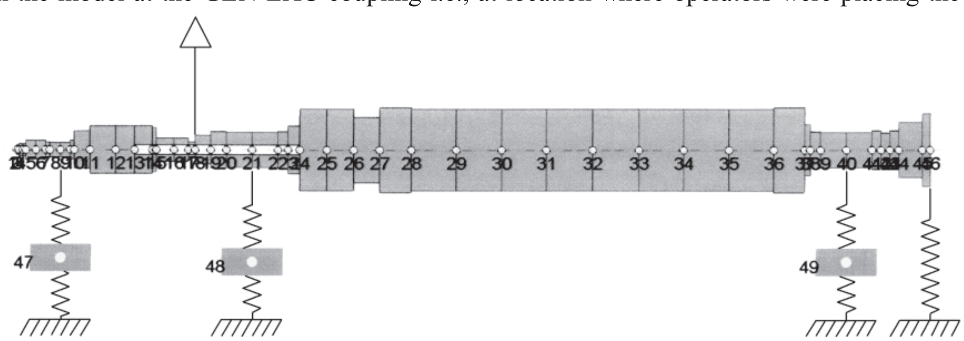


Figure 2: Generator Rotor Model with Simulated Weight

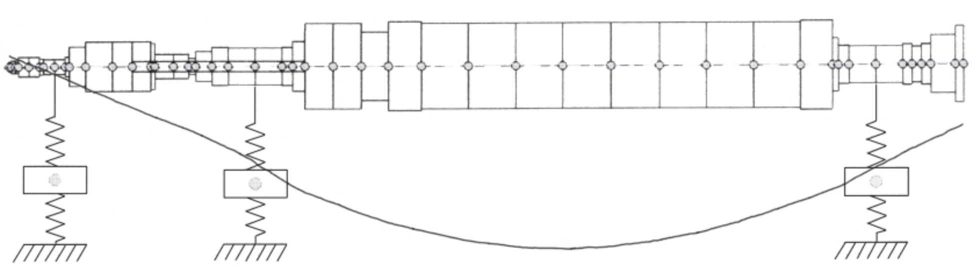


Figure 3: Generator Rotor Model with Gravity Sag Line

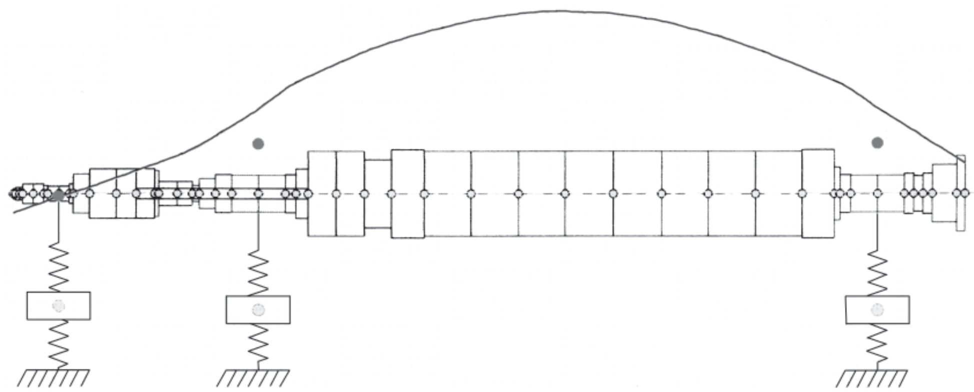


Figure 4: Generator Rotor Model– First Mode at 924 rpm

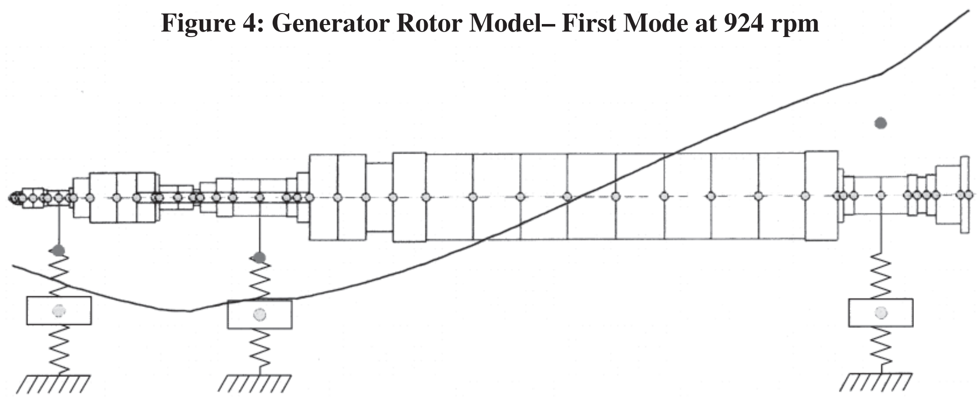


Figure 5: Generator Rotor Model– Second Mode at 1933 rpm

shot". The model confirmed "improvement" of ~ 0.001" (1.0 mil) at bearing #10, but worsened substantially the response at the exciter bearing #11, exactly as it was occurring during the actual balancing. This seems to be in agreement with the past history of "balancing". It also shows that field balancing is not a recommended procedure to solve current rotor vibration problems. (Fig. 6-7).

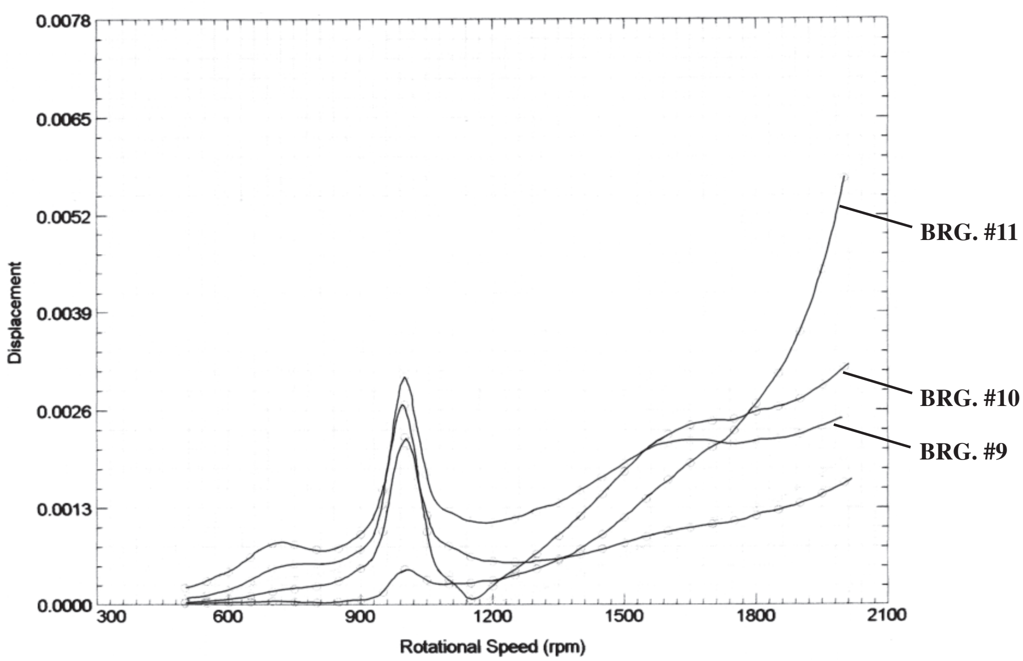


Figure 6: "Normal" Responses, as Found

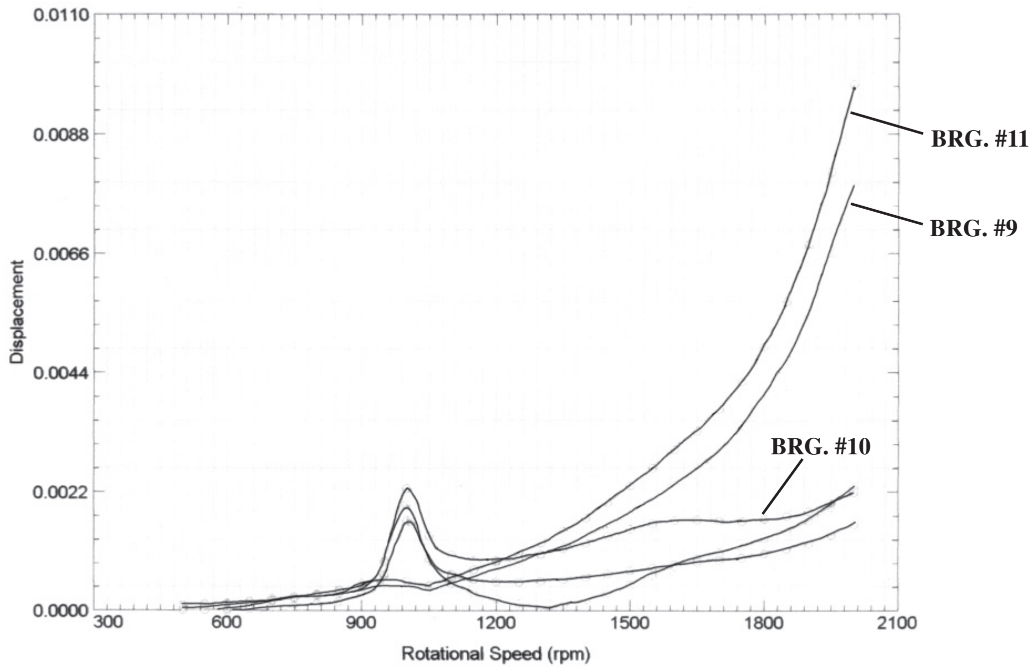


Figure 7: Responses with a Simulated Shot at GEN-EXC Coupling, Approximately 100 oz.

Since simulated balancing gave very similar response as was observed from the actual balancing in the past, it was safe to accept that simulating the expected eccentricity will yield the readings very similar to the currently measured data.

EVALUATION OF THE ELECTRICAL EFFECTS

Most of the unusual events, observed from the collected data, took place in the period from field excitation until some time after generator synchronization. That was also the moment when the alarm light would come on at bearing #10.

At the moment of field excitation, the rotor was spinning at 1800 RPM. When the exciter breakers closed, a magnetic field was created in the air gap which "shocked" and excited the rotor's running mode. The shock was evident by instantaneous amplitude rise and the phase reversal. (See points 1 and 2 respectively on Fig. 8)

The generator was synchronized shortly thereafter (approximately after nine minutes). As the rotor continued to be heated from the applied current, a small "thermal" vector was developing. This vector alone cannot be observed, but only as a vectorial sum change of the initial unbalance vector and thermal vector itself. This is marked as point 3 on Fig. 2.

The upward vibration spike visible at journals #10 and #11 is caused by the unequal circumferential distribution of the strength (hence the radial force) of the magnetic field at the moment of closing the field breakers. This force is longitudinally equally distributed along the length of the rotor coils. This force, since it acts along the rotor, is exciting the rotor's first mode. The angular location of this magnetic field differential force is exactly opposite the existing shorted turn.

Since the 1st mode (static unbalance) is superimposed over the operating deflection mode (second mode or couple unbalance), it is logical that one side will show an increase and the other side a decrease in shaft vibration amplitude. This additional vibration rise happened to be in addition to the mechanical eccentricity detected at generator exciter end (EE) side. The new, sum vector (eccentricity plus differential force from the magnetic field) rotates at synchronous speed and acts as increased unbalance.

Although the force from the field is created within the rotor, the force "pushes" off the stator core, deflecting the rotor. This effect also makes rotors' orbit more cylindrical, since the rotor is more elastic than the core. Additionally, the stator core may exhibit a small deformation in a 4 leaf clover "ovalisation" pattern, following

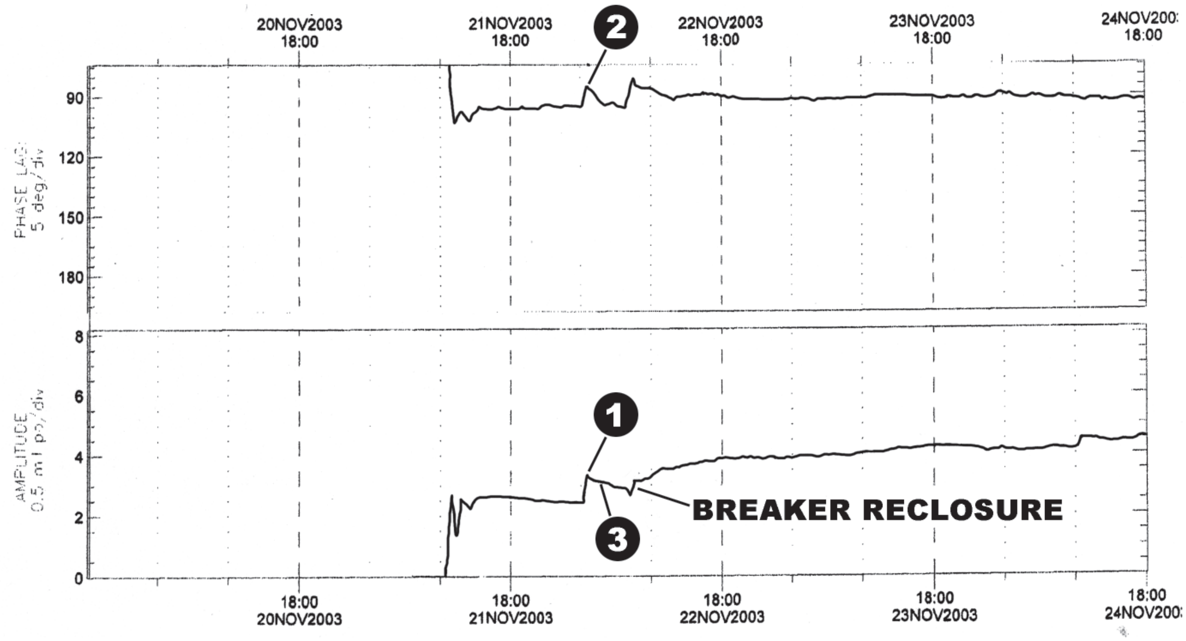


Figure 8: Time Response Plot

the rotation) of the uneven magnetic field producing vibration at 30 Hz (synchronous rotation) at the generator feet.

The magnitude of the stator core motion influence on the stator frame vibrations will depend on the core suspension springs condition and their stiffness. From the moment of the "spike" onward, the side with a stronger magnetic field heats more with applied rotor current than the opposite side with a shorted turn. That produces a "thermal vector" along the full length of the coils, again creating a static unbalance. (The event is time dependent). This "thermal vector" initially cancels the effect of the differential magnetic force on the rotor and rotor vibrations are lowered. As more current is applied to the field, the "thermal vector" rises proportionally to the square of the applied current. The magnitude of the magnetic differential force, and the severity of the short could be assessed from the known rotor's deflection and the rotor's stiffness constant.

The verification of the existence of a short and its severity could have been simplified were the flux probes installed in the air gap. The estimate is that this force is about 26,000 lbs, based on the FE model stiffness and the measured "spike" amplitude at the specified rotor element. The severity of the shorted turns if they are all located in one pole, could be estimated by the OEM, by calculating a change in magnetic flux strength due to shorted turns. The differential force could be compared to calculated spring force of the rotor derived above.

CONCLUSIONS

After a review and analysis of all available past and collected new data, it was concluded that the generator rotor vibrations are of mechanical nature. There is a small electrical effect, but it is not severe enough to justify the OEM's recommended expenditure. This paper shows that the proper vibration diagnostics can be used as a tool in economic decision making.

In Summary:

- The highest shaft vibration amplitude is recorded at bearing #10, i.e., the journal on EE side of the generator.
- Vibration was there for previous 8 years, but just below the plant preset alarm level.
- Vibration became a problem when the alarm was activated (a change of a mere 0.0005" pk-pk).
- The cause of vibration was induced by a "cocked" fan hub shrink fit, during the replacement of the Rotor

Retaining Rings. (This was confirmed by review of plant historical vibration data).
- There is a small rotor thermal sensitivity, which when superimposed to mechanical displacement, causes the "vibrations" to reach alarm set point.
- The root cause of vibrations is predominantly mechanical, and it can be corrected by re-assembling shrink on components on the rotor, verify and evaluate rotor eccentricities prior to balancing and rebalance the rotor in balancing facility prior to re-assembly.
- Install Flux Probes in generator, to avoid future confusion between mechanical and electrical effects on vibrations.
- There is no need for generator rotor replacement.

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