APPLICATION OF ROTOR DYNAMIC ANALYSIS FOR EVALUATION OF SYNCHRONOUS SPEED INSTABILITY AND AMPLITUDE HYSTERESIS AT 2ND MODE FOR A GENERATOR ROTOR IN A HIGH-SPEED BALANCING FACILITY

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ABSTRACT

The paper presents a high speed balancing facility case history of a generator rotor with a long turbine end overhang. The rotor had experienced rapid increase in vibration at 3600 RPM and amplitude hysteresis at the 2nd mode between run-up and run-down during shop balancing. This behavior raised concerns about the possibility of excessive generator rotor vibration on site.

A rotor-bearing-support system model was created to study the observed generator behavior in the balance facility. Critical speed, unbalance response and damped eigenvalue analyses were performed. Examination of the rotor kinetic and potential energies for the 2nd mode showed that over 78% of the rotor 2nd mode kinetic energy was associated with the overhang. The analysis indicated that when the rotor operated at 3600 RPM, between the 2nd and 3rd modes, the overhang motion increased due to amplification of both modes and very little damping. As a result, vibration at the bearings increased and when the rotor decelerated through the 2nd mode the increased motion on the coupling generated excessive vibration.

The model was modified by adding coupling constraints to represent operating conditions of this rotor in the unit. The 2nd mode was shifted out of operating speed range, which was confirmed by the field operation.

Two rotors were balanced and both are now operating with acceptable vibration levels.

This paper illustrates how rotor dynamics analysis help to explain unusual rotor behavior and provided assurance that vibration performance of this rotor on site will not be affected.

Vibration plots from the balancing facility, DyRoBes models, critical speed analysis, unbalance response plots, and field vibration data are included as illustrations.

Keywords: **high speed balancing, rotor dynamics, mode shape, unbalance response, critical speed analysis.**

INTRODUCTION

A small, 50 MW generator as shown in Fig. 1 has displayed somewhat unexpected behavior during shop balancing. While operating at 3600 RPM, the rotor experienced rapid increase in vibration with the highest readings reaching unacceptable levels on the turbine end within several minutes. Vibration levels on run-down peaked at the 2nd mode over-ranging instrumentation. This behavior was repeated during each balance run with 2nd mode amplitudes on run-down depending mostly on vibration levels achieved at 3600 RPM, not on vibration readings taken on run-up (Fig. 2, Fig. 3).

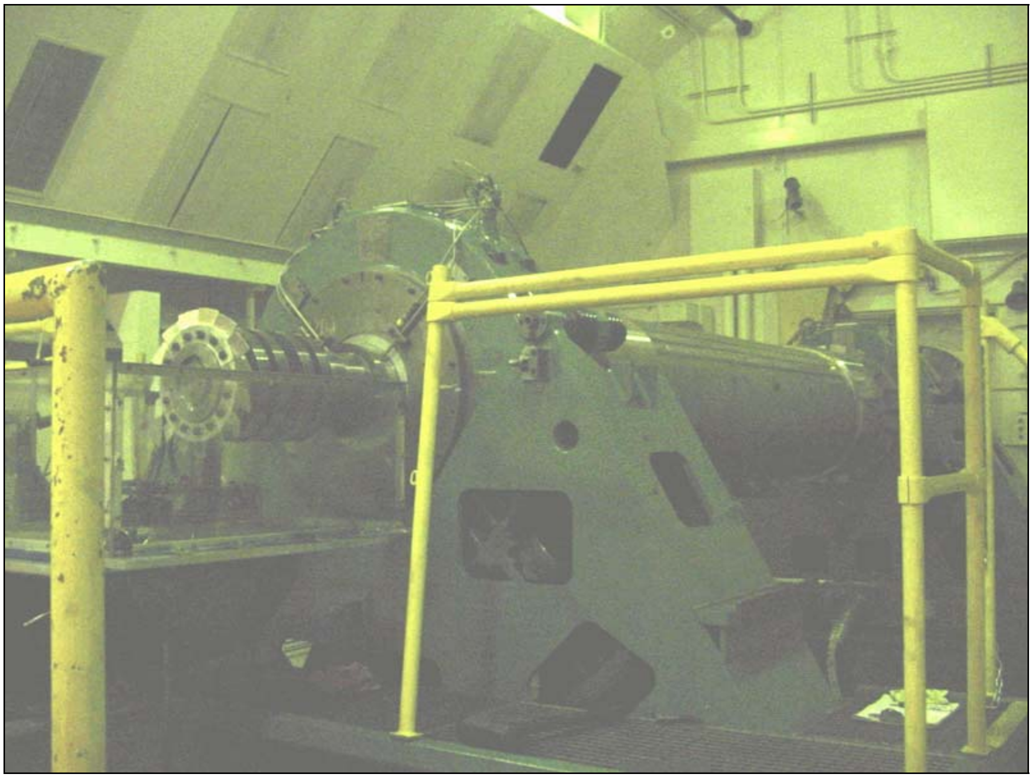

Fig. 1 Generator Rotor in High Speed Balancing Facility

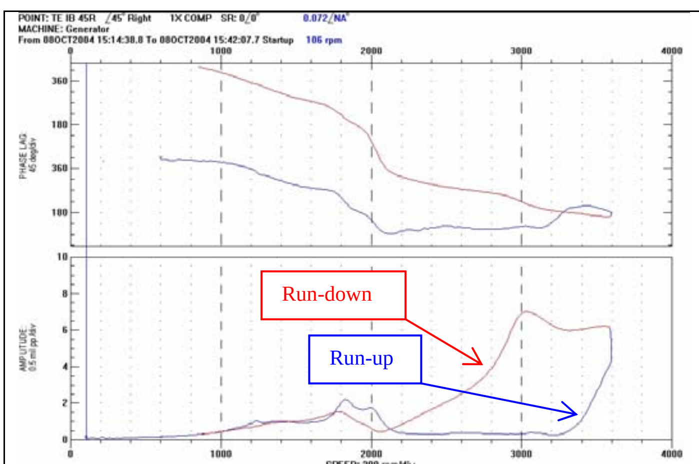

Fig. 2 Run-up and run-down TE Inboard Probe

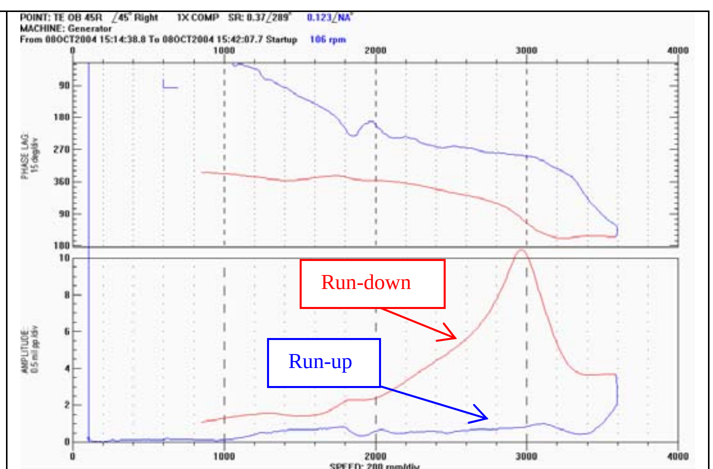

Fig. 3 Run-up and run-down TE Outboard Probe

Previous experience with several turbine rotors with long overhangs, which had displayed similar behavior (hysteresis), led to conclusion that the observed data was clearly due to rotor's overhang influence on operating mode shapes. Fig. 2 represents the TE inboard probe and Fig. 3 represents the outboard probe. These probes are set at 45^o and hence record components of motion in both the horizontal and vertical directions. Note that the TE outboard bearing response shows a distinct peak at 3,000 RPM on run-down. This effect is not observed on run-up. The response on run-down is a result of the nonlinear characteristics of the turbine end bearing under large whirl motion inside the bearing. Similar response effects have been observed on overhung rotors in rolling element bearings with dead band clearance. This condition is balance pit specific, and while it complicates balancing process, it does not exist in operation.
Inspection for oil seal rubs was conducted to make sure that rubbing is not a cause of rapid increase of vibration. No signs of rubs were discovered.

1. MODEL TO STUDY OBSERVED CONDITIONS

A rotor model was created using DyRoBes finite element rotor bearing dynamics software (Fig. 4). The goal was to compute critical speeds, mode shapes, unbalance response and energy distribution. The generator is supported on elliptical bearings as shown in Fig. 5. The 50% bearing preload causes higher stiffness and damping values for the vertical direction as compared to the horizontal. The bearings are supported in pedestals with measured X-Y stiffness values of approximately 3E6 Lb/In. For accurate rotor dynamics modeling of large turbines and generators, the pedestal effects most be included.

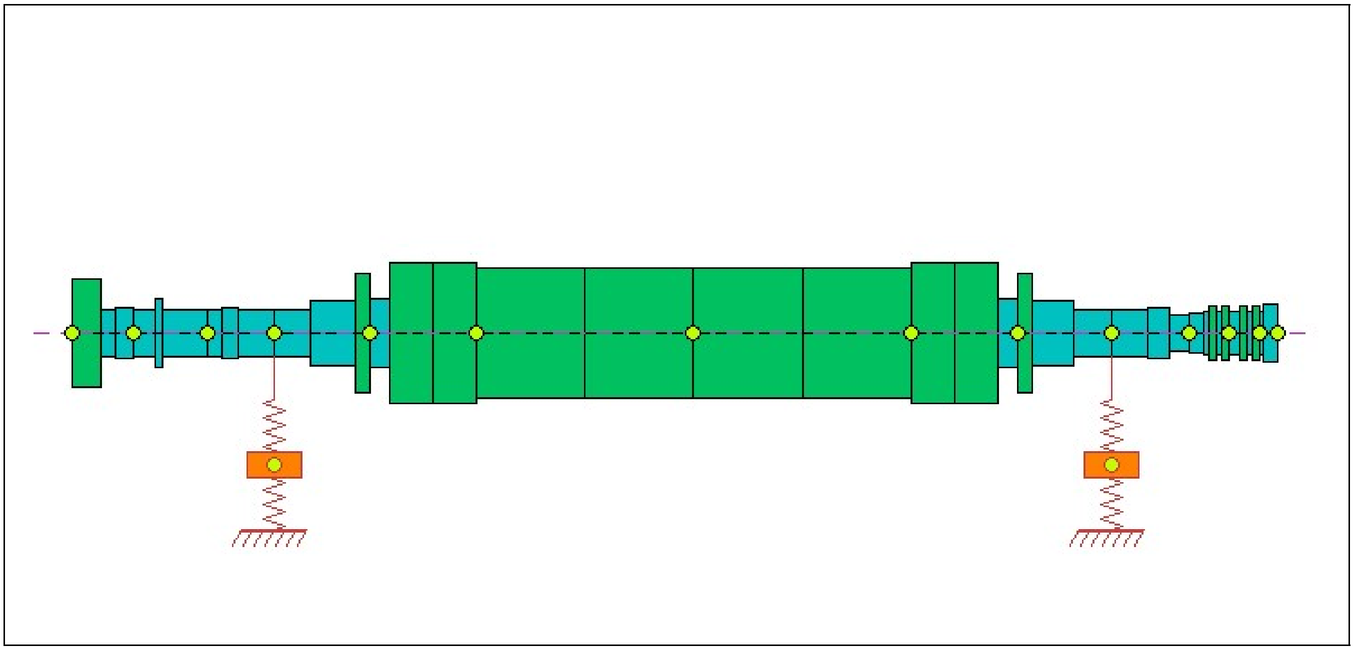

Fig. 4 Rotor Model

In general, 2-pole generator rotors have slots machined in the rotor core, which are filled with copper, insulation materials and wedges, creating two different stiffness axes in the rotor. If generator bending inertia asymmetry is high, then superharmonic vibrations of 2X may be observed. In normal rotor dynamics modeling, shaft asymmetry effects are ignored and the rotor is modeled as a cylinder. In general, this effect may be ignored in critical speed and unbalance calculations. The critical speed calculations provide useful information on rotor mode shapes and kinetic and potential energy distribution in the various modes. Table 1 represents the generator undamped critical speeds based upon nominal stiffness values for the first three computed modes.

	1st Mode	2nd Mode	3rd Mode
Measured, RPM	1810	3100	4070
Calculated, RPM	1928	2962	4208
Diffrence, %	6.12	4.66	3.28

Table 1 Calculated and Measured Critical Speed Comparison

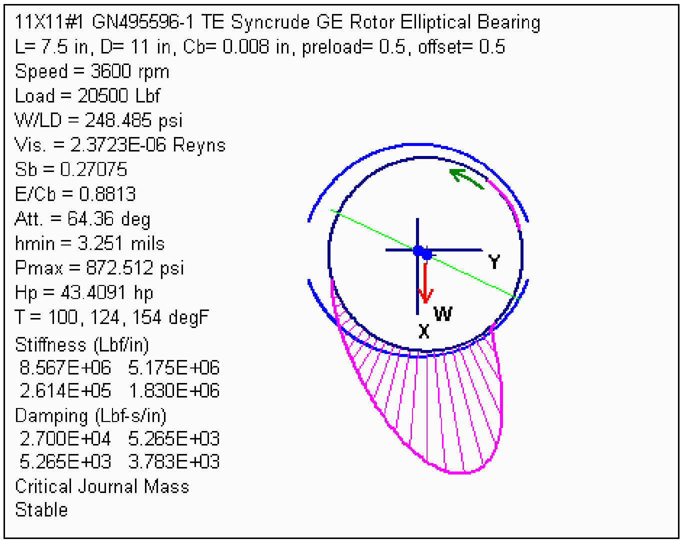

Fig. 5 Bearing Pressure Distribution at 3600 RPM

The first three computed mode shapes are presented in Figs. 6,7 and 8. The graphs clearly demonstrate that the majority of the motion on 2nd and 3rd modes is occurring at the coupling. Critical speed map (Fig. 9) illustrate that the rotor operating speed in the balancing facility fall between 2nd and 3rd modes. Fig. 10 and Fig. 11 show run-up to 20% overspeed that was performed as a part of standard balancing. The Bode plots show all three modes measured by the turbine end probes.

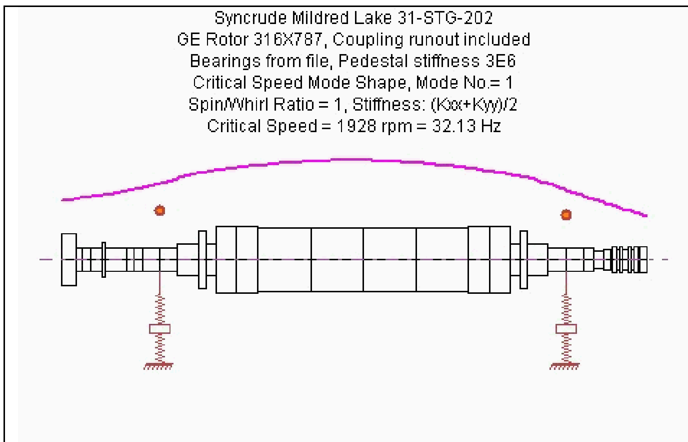

Fig. 6 Mode 1

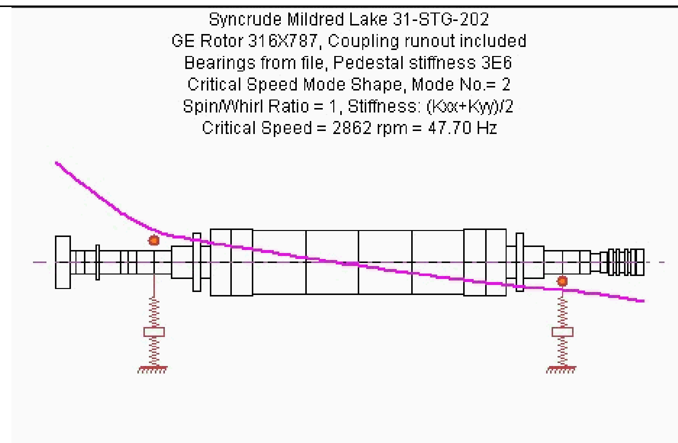

Fig. 7 Mode 2

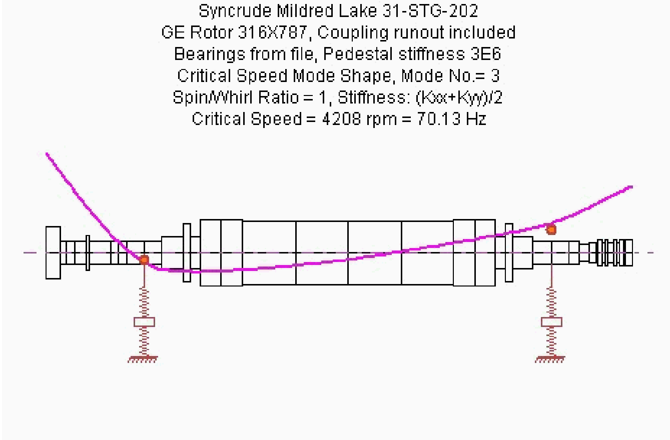

Fig. 8 Mode 3

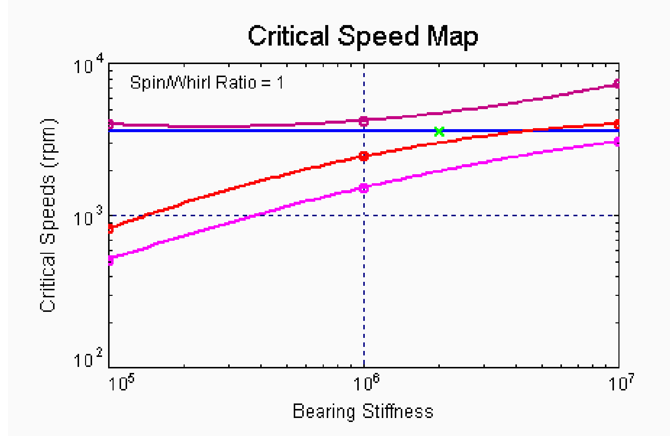

Fig. 9 Critical Speed Map

Fig.7.

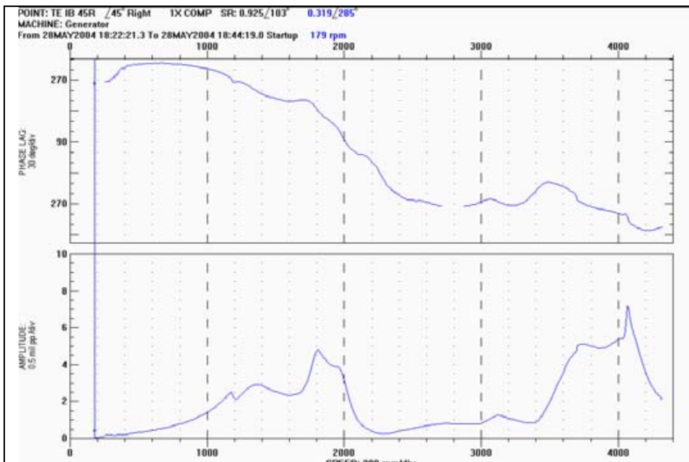

Fig. 10 Run-up to 4320 RPM TE Inboard Probe

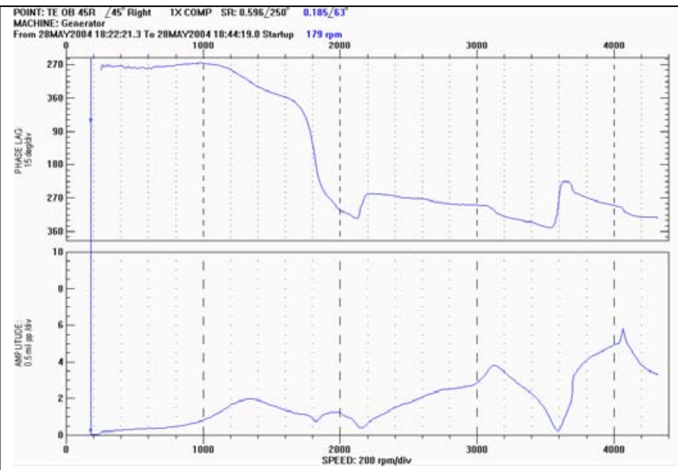

Fig. 11 Run-up to 4320 RPM TE Outboard Probe

2. ENERGY EVALUATION

Examination of the rotor kinetic and potential energies for the 2nd mode (Fig. 14, Fig. 15) revealed that over 78% of the rotor 2nd mode kinetic energy was associated with the overhang. Three-dimensional plots of the damped eigenvalues (Fig. 18, Fig. 19) show that the majority of rotor's motion is occurring at turbine end overhang. The analysis indicated that when the rotor operated at 3600 RPM, between the 2nd and 3rd modes, the overhang motion increased due to amplification of both modes. There is also very little damping associated with the second mode. As a result, vibration at the bearings increased and when the rotor decelerated through the 2nd mode the increased motion on the coupling generated excessive vibration.

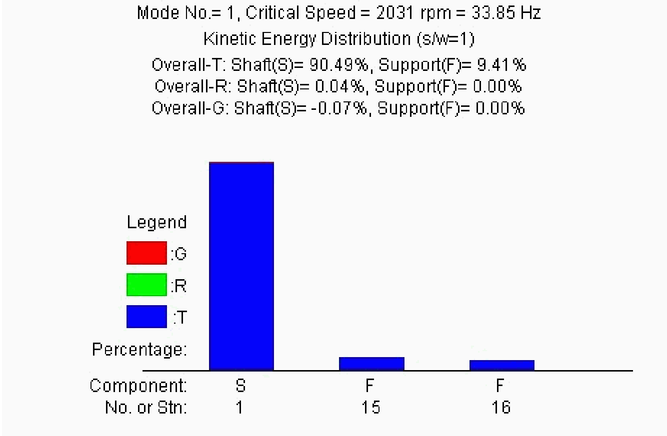

Fig. 12 Kinetic Energy Distribution, Mode 1

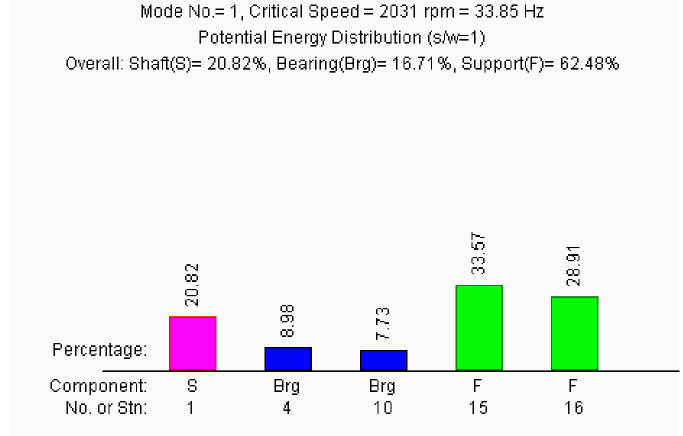

Fig. 13 Potential Energy Distribution, Mode 1

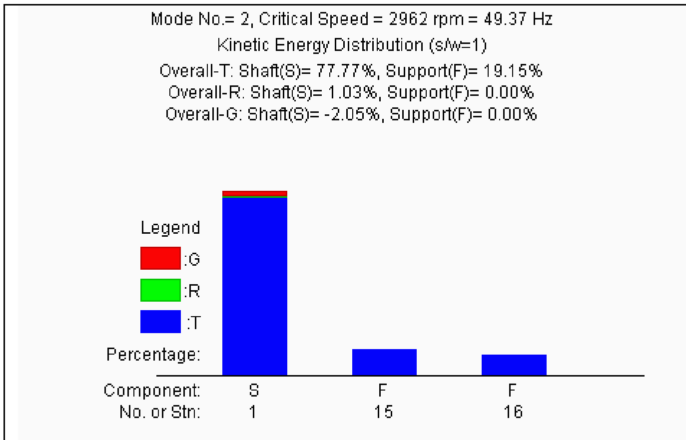

Fig. 14 Kinetic Energy Distribution, Mode 2

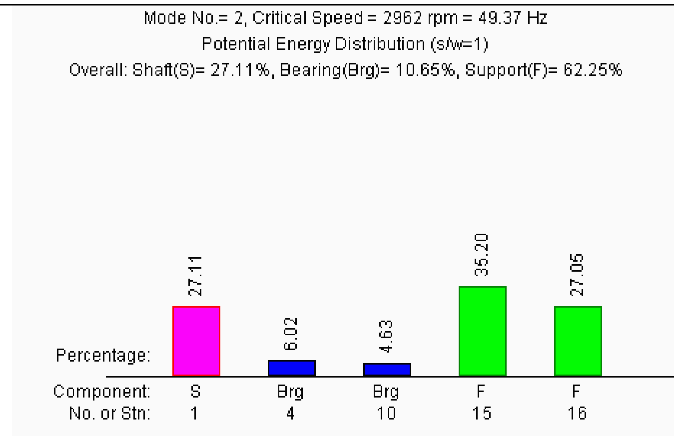

Fig. 15 Potential Energy Distribution, Mode 2

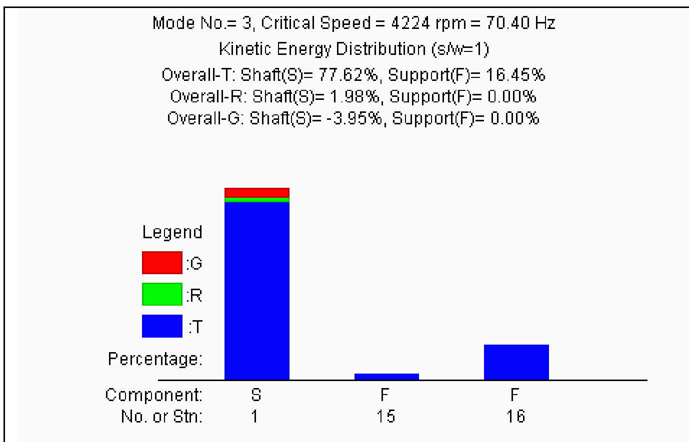

Fig. 16 Kinetic Energy Distribution, Mode 3

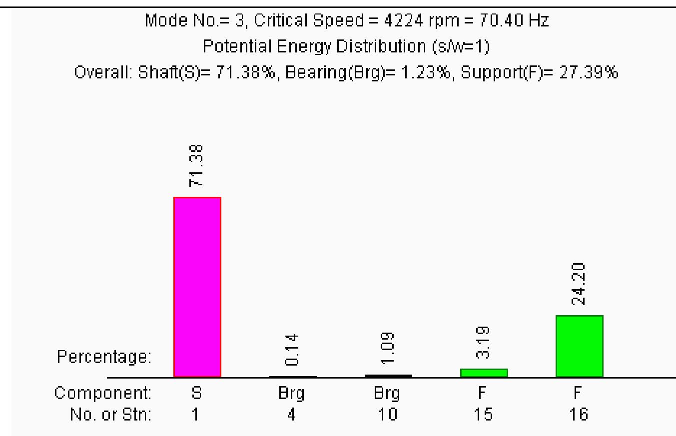

Fig. 17 Potential Energy Distribution, Mode 3

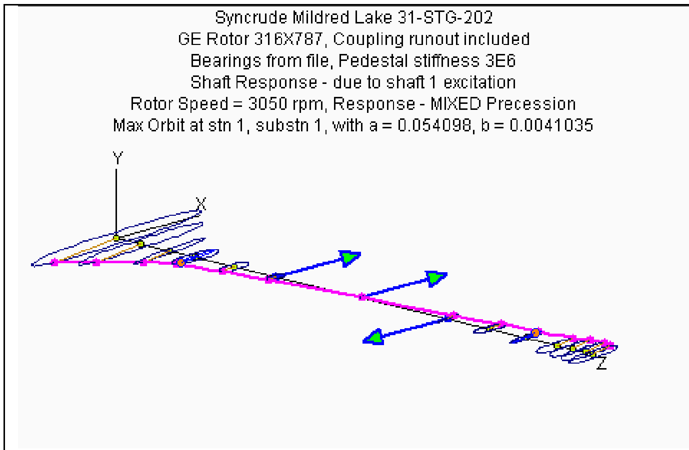

Fig. 18 Unbalance Response (3D) at 2nd Mode

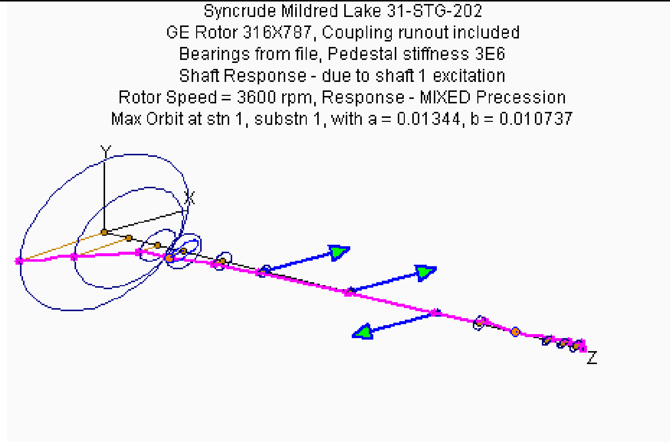

Fig. 19 Unbalance Response (3D) at 3600 RPM

3. MODEL MODIFICATION TO SIMULATE OPERATING CONDITIONS

The model was modified by adding constraints to the turbine end coupling to simulate connection to steam turbine. This can be done in two ways: adding a plain constant stiffness bearing or using constraints feature in DyRoBes. The latter option was selected, using pinned constraints. The position of the 1st critical speed was changed slightly, as expected, to 2020 RPM (Fig. 20), but the 2nd mode moved over 1000 RPM to 4113 RPM (Fig. 21).

It was confirmed with the customer that in operation 1st critical speed is observed about 2000 RPM and the is no 2nd critical within operating speed range.

Comparison of critical speed mode shapes revealed that the 3rd mode in balance pit (Fig. 8) corresponds to the 2nd mode in operation (Fig. 21). Thus the 2nd mode with high vibration levels and hysteresis between run-up and run-down seen in the facility was mostly due to unsupported overhang.

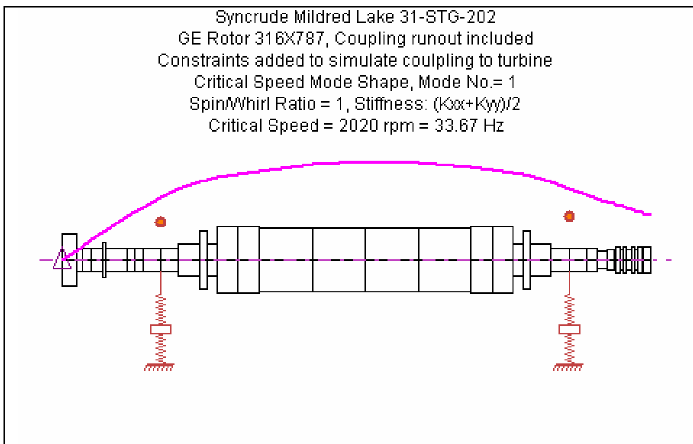

Fig. 20 Mode 1 in Operation

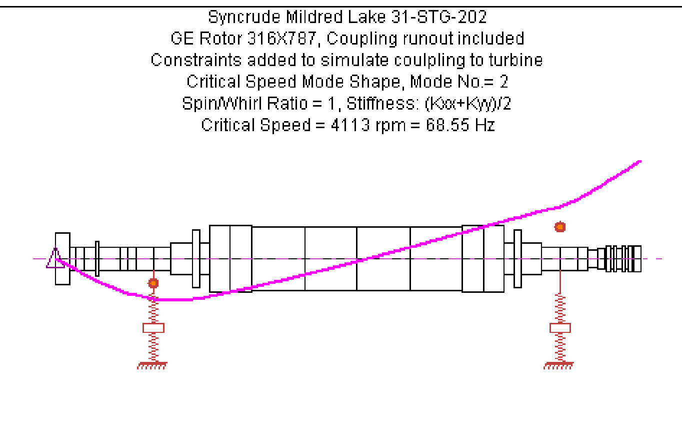

Fig. 21 Mode 2 in Operation

The presented data demonstrates once again, that a high speed balancing facility is a close, but only, approximation of operating conditions, [1]. Many factors, including connection to other rotors, stiffness and damping properties of the support systems and thermal and electrical forces are influencing vibration behavior in the unit.

4. DAMPED NATURAL FREQUENCIES

In order to add further insight into the dynamical behavior of the generator, the damped natural frequencies (complex eigenvalues) and 3 dimensional mode shapes (complex eigenvectors) were computed (Fig. 22-25). The complex modes were computed using the 8 bearing stiffness and damping coefficients as shown in Fig. 5 for the preloaded elliptical bearing. In addition, the bearings are acting in series with pedestals or foundations with a mass of 5,000 Lb and support stiffness values of 3E6 Lb/in.

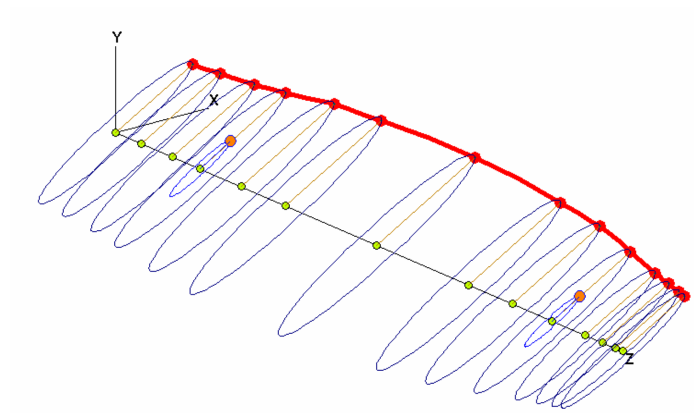

Fig. 22 Mode 1 - 1501 RPM (1st Horizontal)

Log Dec =.221, Ac =14

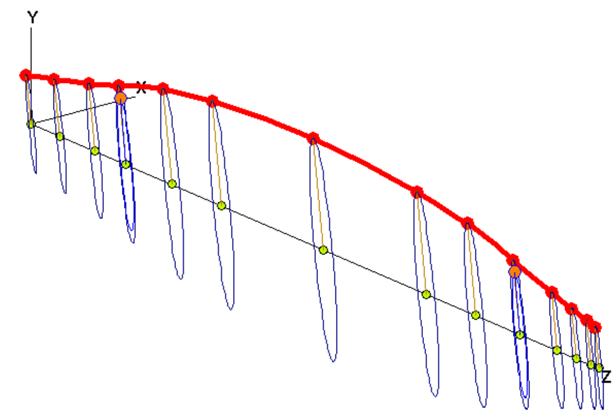

Fig. 23 Mode 2 - 2064 RPM (1st Vertical)

Log Dec=.221, Ac=14

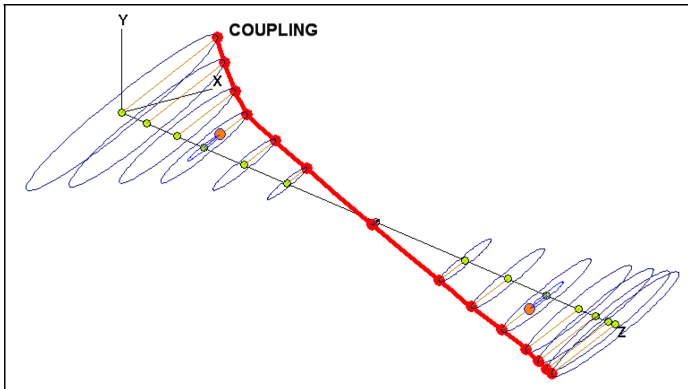

Fig. 24 Mode 3 - 2341 RPM (2nd Horizontal)

Log Dec=0.418

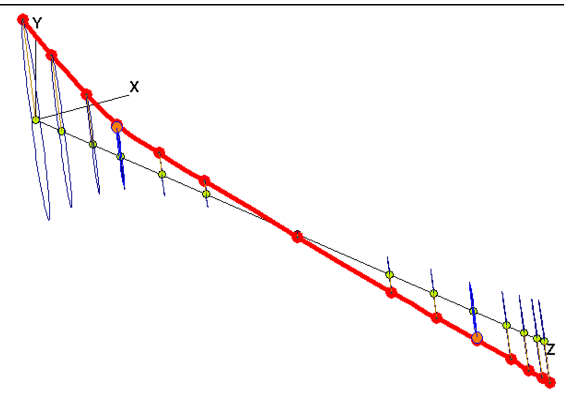

Fig. 25 Mode 4 2985 RPM (2nd Vertical)

Log Dec=0.18

Because of the bearing asymmetry, we observe different critical speeds in the vertical and horizontal planes. By having the balancing probes mounted at 45 deg, all modes are seen. The rotor amplification factor passing through a critical speed is related to the log decrement for that mode. The amplification factor Ac=PI/Log Dec. Of concern is the mode 5 as shown on Fig. 26. In this mode we have very high coupling motion with little amplitude of motion at the drive end bearing. This condition is known as bearing lockup and adds to the difficulty of balancing above 3,600 RPM due to high coupling motion. This condition will not occur when the drive turbine is connected to the generator coupling.

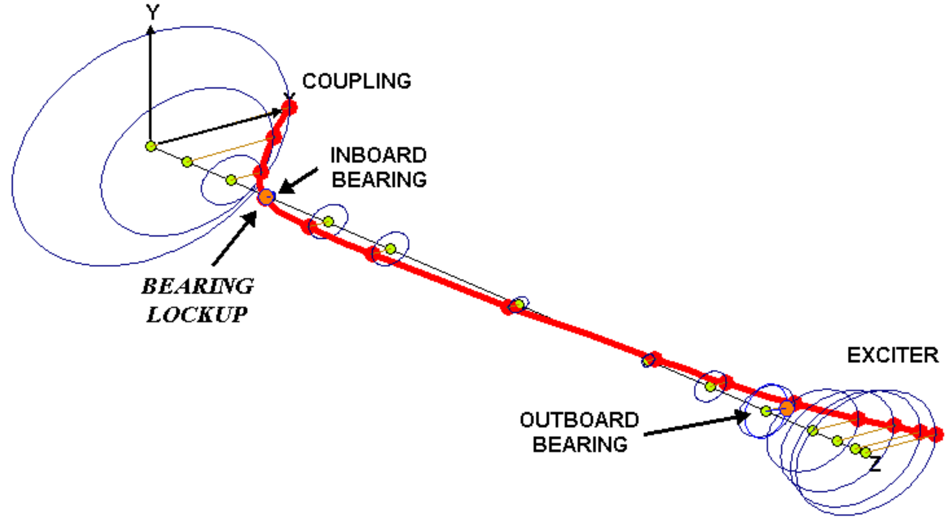

Fig. 26 Mode 5 - 4107 RPM (3rd Horizontal)

Log Dec =0.0139, Ac=220

5. BALANCING PROCEDURE

Based on the analysis and previous experience with other rotors it was decided to use transient readings at 3600 RPM for balancing and acceptance points. Amplitude hysteresis at second mode was disregarded as facility specific and not present in normal operation of the rotor. Run-up data was considered for acceptance through the full speed range. The customer has agreed with this assessment and the rotor was balanced to acceptable criterias.

After the rotor was put in service the measured vibration levels were acceptable at full speed and load ranges. The second rotor (sister unit) was balance following the same procedure several month later. Operational vibration data is presented below (Table 2).

	TE 45L, mils pp	TE 45R, mils pp	EE 45L, mils pp	EE 45R, mils pp
10 MW	1.51@321	0.61@62	1.72@165	1.23@20
50 MW	2.13@328	0.77@75	1.85@169	1.45@20

Table 2. 31-STG-201 Operational Vibration Data (not compensated)

CONCLUSIONS

This paper illustrates how rotor dynamics analyses help to explain unusual rotor behavior and provided assurance that vibration performance of this rotor on site will not be affected. A specific balancing process was developed which yielded good vibration performance of two units.

ACKNOWLEDGMENTS

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REFERENCES

1. L'vov, M., Flexible Rotors: Shop Balancing at "Operating Speed", *Proceedings 23rd Annual Meeting of the Vibration Institute*,(1999)