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 3600RPM 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 (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).

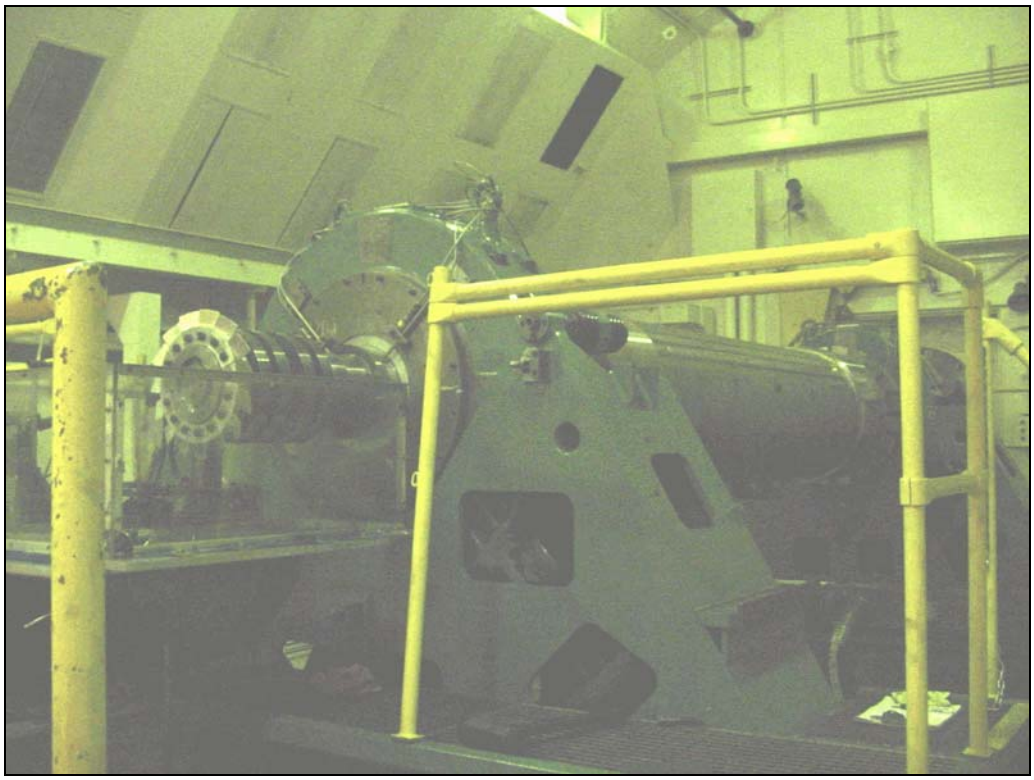

Fig. 1 Generator Rotor in High Speed Balancing Facility

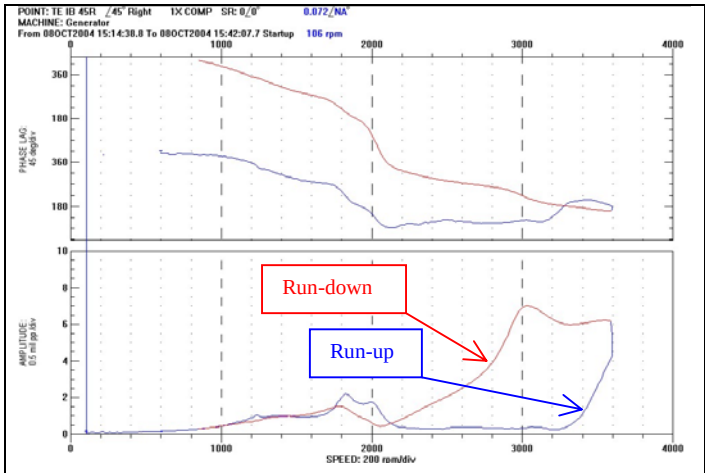

Fig. 2 Run-up and run-down TE Inboard Probe

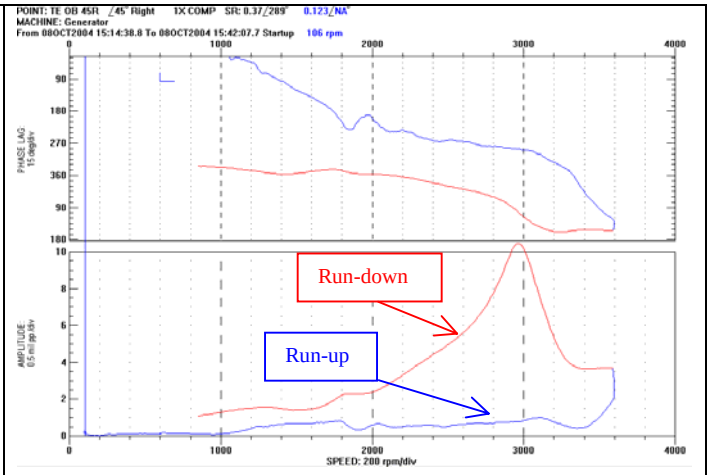

Fig. 3 Run-up and run-down TE Outboard Probe

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. 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. Bearing properties (Fig. 5) were previously calculated during design phase and were directly imported into the model. Flexible supports with known stiffness were also used.

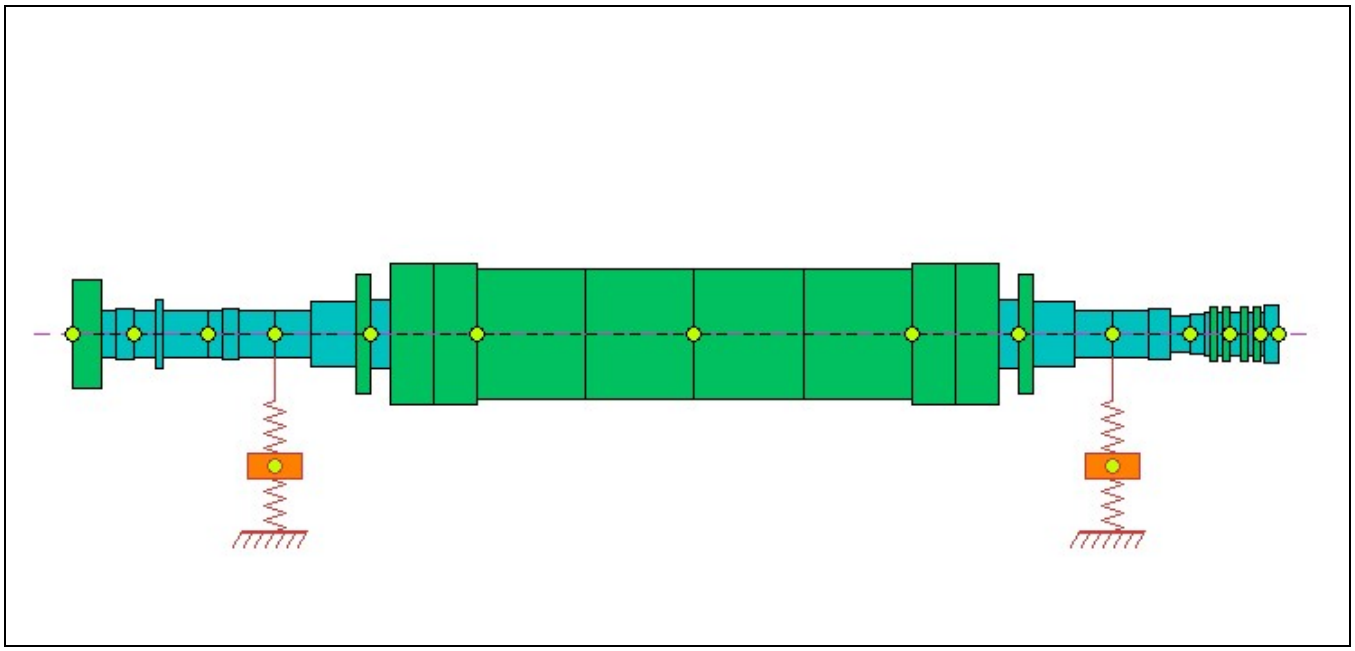

Fig. 4 Rotor Model

Practically all large 2-pole generator rotors have slots machined in rotor's body, which are filled with copper, insulation materials and wedges, creating 2 different stiffness axis's. To simplify and speed-up the modeling process the body was modeled as a cylinder. That probably led to some differences between calculated and measured critical speeds (Table 1), but the values were close enough for practical use.

	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

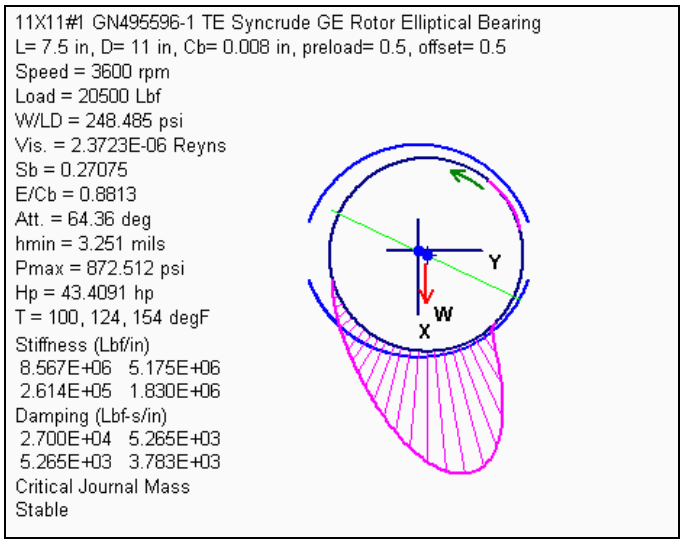

Fig. 5 Bearing Pressure Distribution at 3600 RPM

Calculated mode shapes are presented on Fig. 6, Fig. 7 and Fig. 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 3 modes measured by the turbine end probes.

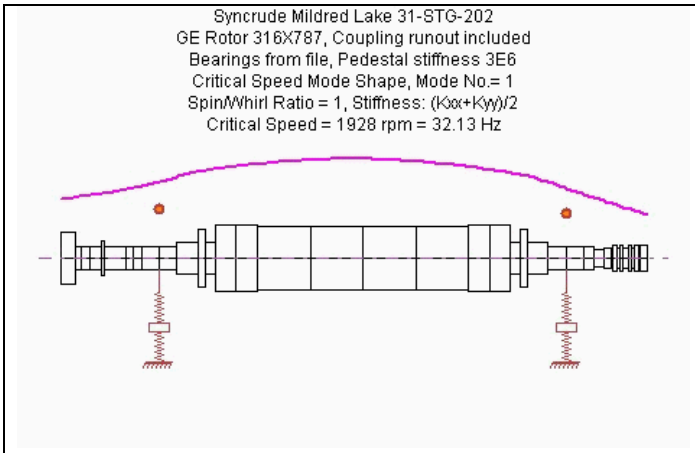

Fig. 6 Mode 1

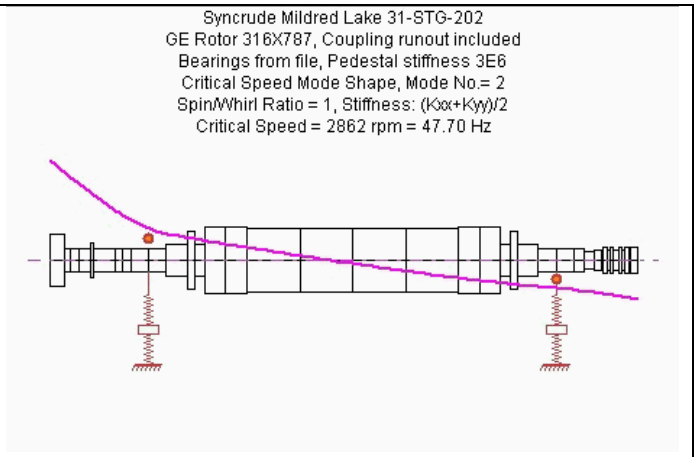

Fig. 7 Mode 2

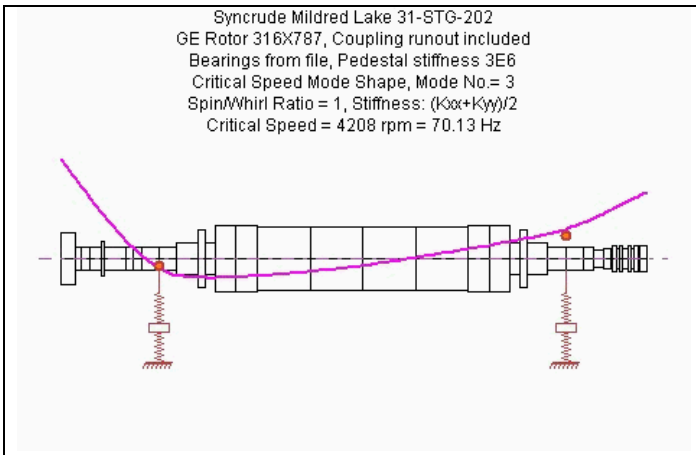

Fig. 8 Mode 3

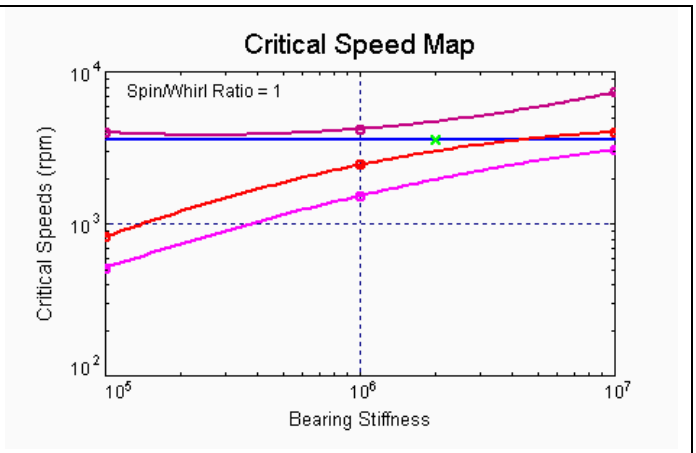

Fig. 9 Critical Speed Map

Fig.7.

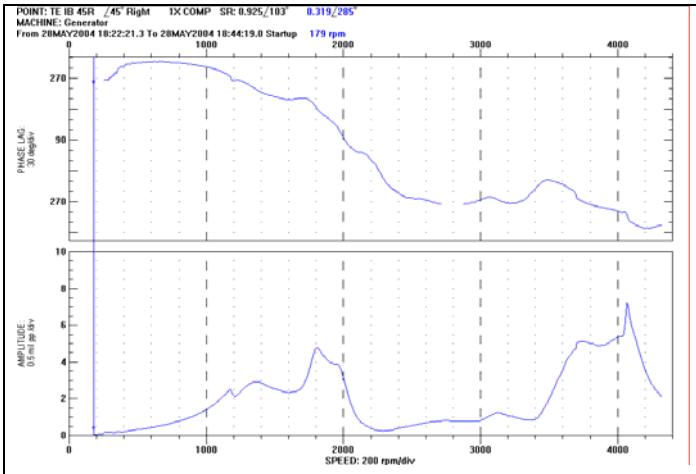

Fig. 10 Run-up to 4320 RPM TE Inboard Probe

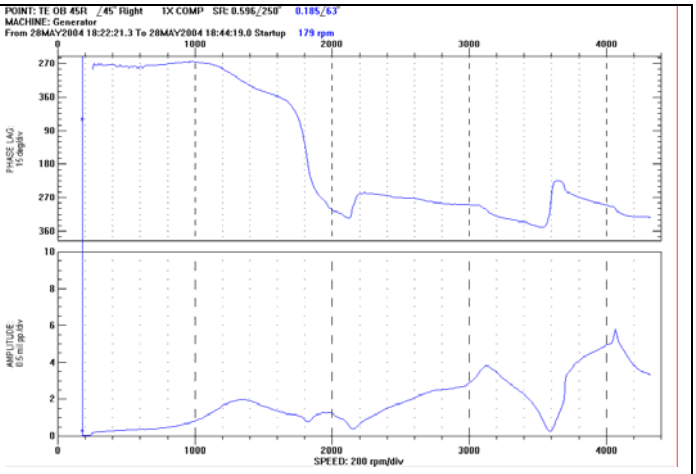

Fig. 11 Run-up to 4320 RPM TE Outboard Probe

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.

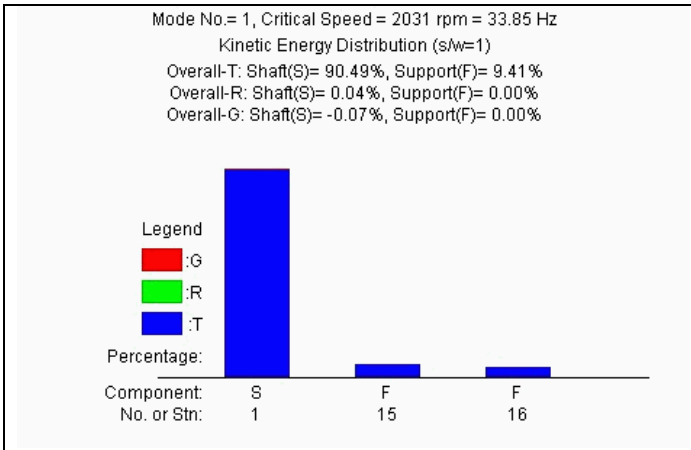

Fig. 12 Kinetic Energy Distribution, Mode 1

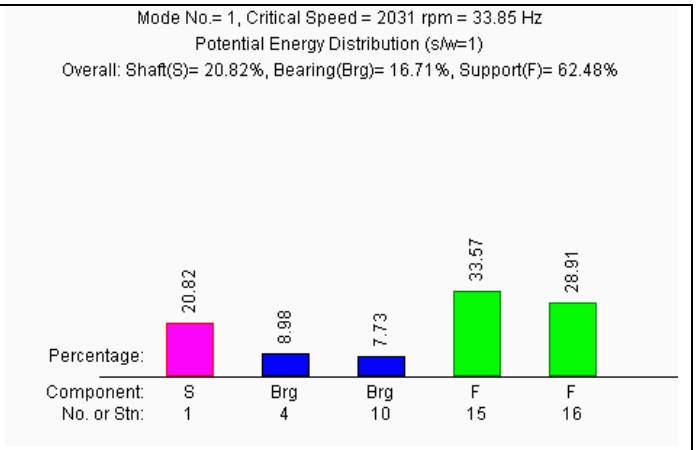

Fig. 13 Potential Energy Distribution, Mode 1

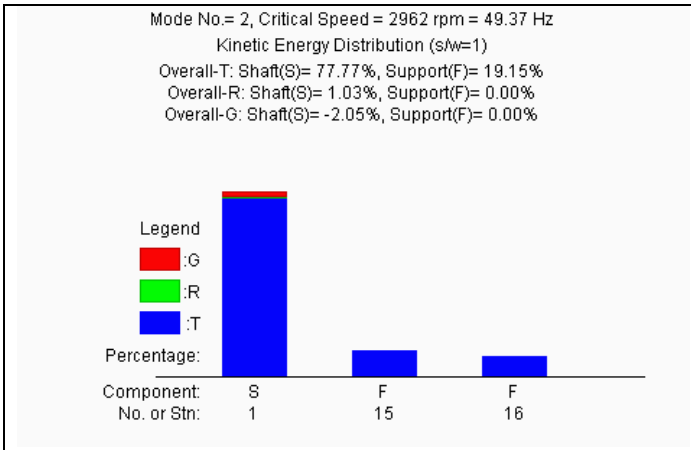

Fig. 14 Kinetic Energy Distribution, Mode 2

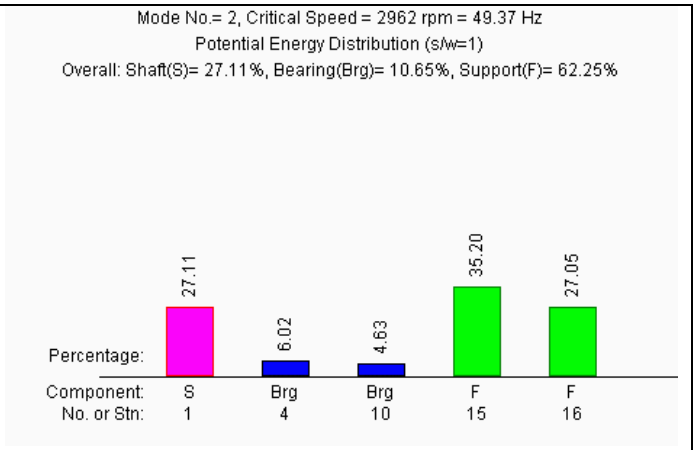

Fig. 15 Potential Energy Distribution, Mode 2

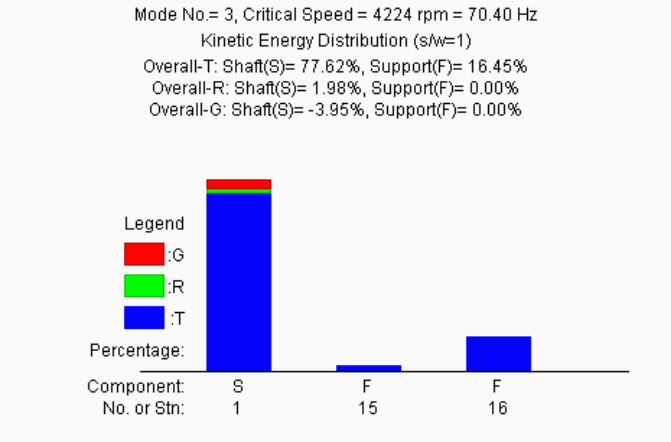

Fig. 16 Kinetic Energy Distribution, Mode 3

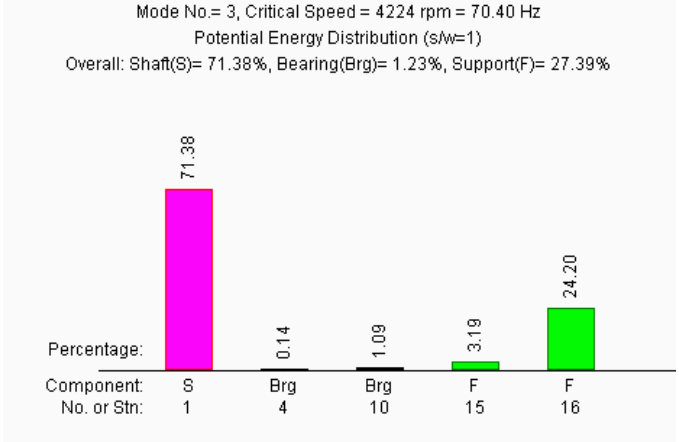

Fig. 17 Potential Energy Distribution, Mode 3

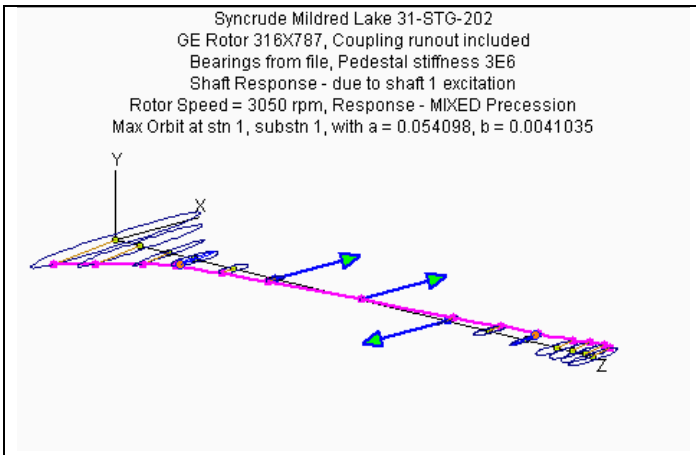

Fig. 18 Unbalance Response (3D) at 2nd Mode

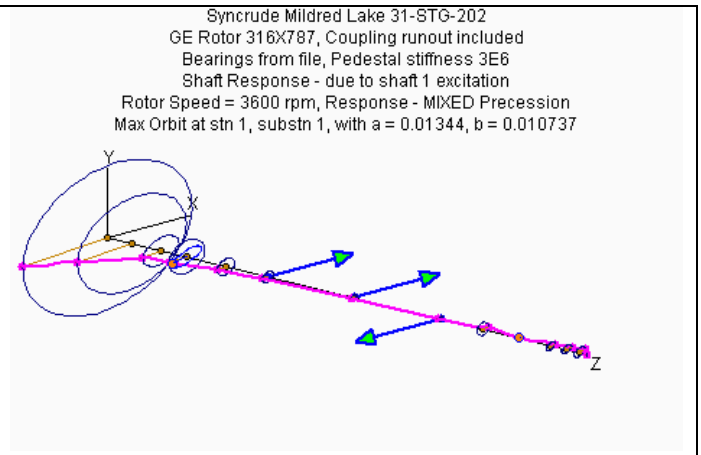

Fig. 19 Unbalance Response (3D) at 3600 RPM

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.

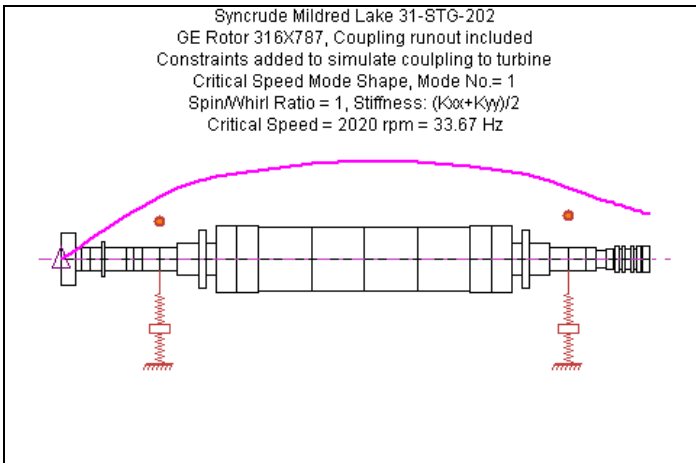

Fig. 20 Mode 1 in Operation

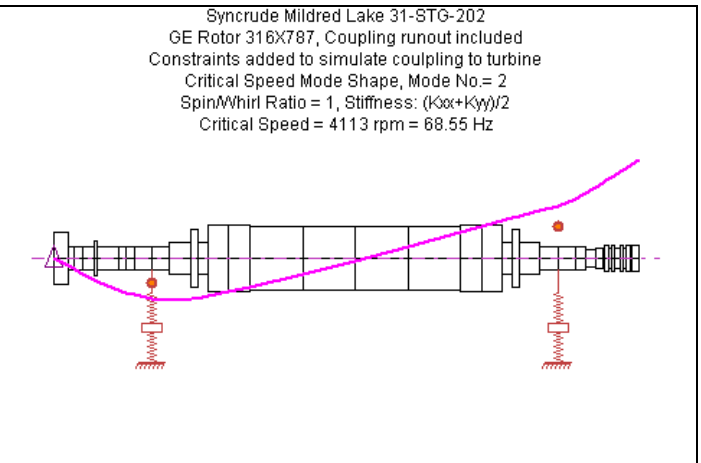

Fig. 21 Mode 2 in Operation

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. 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 analysis 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)