Diagnosis and Solution of the Exciter Rotor Vibration Problem

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ABSTRACT

This paper addresses the experimental and theoretical investigation of unacceptable high vibration during startup and coastdowns on the #7 exciter bearing of a 384 MW unit. The Encina Unit #5 Turbine Generator has had the vibration problem since the unit became operational in 1978. The high vibration occurs during startup and coastdowns at 2500 RPM, which is the first critical speed of the exciter and the second critical speed of the generator. Shaft vibration occurs during cold starts had been in excess of 25 mils (peak to peak). Numerous balances on the exciter were attempted over the years, as well as several on the generator, to correct the problem. Vibration amplitudes were optimized, but the problem was not solved. Vibration data showed that the #7 bearing is very lightly loaded. A detailed theoretical analysis of the rotor-bearing system indicated that static bending stresses at the generator-exciter coupling are "zero" under current alignment and the exciter vibration problem is the result of a compromised alignment, possibly done to resolve the generator instability problem in the past. The recommended solution consisting of minor change in the design of bearing #7 and in the alignment of bearing #6 and #7 was presented and obtained the expected effect. Now the unit operates at allowable vibration level, and responds better to balancing.

Keywords: Vibration, exciter rotor, bearing

BACKGROUND

An Diego Gas & Electric Company is an investorowned utility that serves over one million customers in San Diego and Orange counties. Currently, over fifty percent of the power distributed by SDG&E is imported from other power producers. The remaining power is generated at either the 950 MW Encina Power Plant in Carlsbad, California or the 700 MW South Bay Power Plant in Chula Vista, California. The turbine generator that is addressed in this paper is the Encina #5 generating unit. Encina #5 is not based loaded unit, it is a cycling unit. The unit load-cycles daily, and can have numerous on/off cycles during the year.

1. INTRODUCTION

During the Fall 1994 outage, a new Siemens LP rotor had been installed replacing the other OEM on 385 MW Turbine-Generator. The new LP turbine is heavier and has different rotor dynamic properties. A recommended alignment was implemented. Deviating from the full catenary was very visible at the joint between generator and exciter. The purpose of the original alignment change was done to avoid probably the instability problem of the generator and to lower the vibration at the critical for the #7 bearing by offsetting the exciter-generator coupling. The alignment with new LP rotor was maintained as close to original rotor chain slope as possible, to avoid extensive pedestals elevation changes. The generator was balanced in the High-speed Balance and Overspeed

Facility. However when the Unit was re-started and synchronized, an ongoing high vibration problem continued to exist at the Exciter #7 bearing. (Table I). Numerous efforts had been made by plant engineers to balance the exciter with no significant improvement. Balance attempts had shown that by reducing vibration at running speed, the vibration increased at critical speed, and reducing critical speed vibration had increased running speed vibration. Based on analysis of the vibrational data, it was shown that the exciter journal centerline within own bearing rises to a large level (Fig. 1) and instability within the exciter #7 bearing may exist[1]. In order to determine the reasons for this problem, the dynamic model of the turbine-generator system was built. The calculation focused on two points: load distribution and stability analysis of #7 bearing.

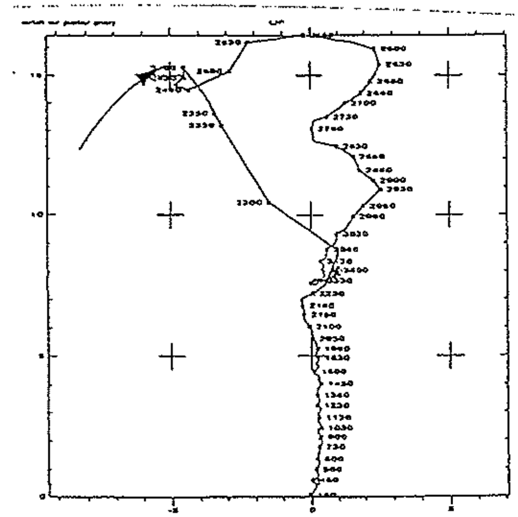


Fig. 1 The Exciter Journal Centerline at #7 Bearing

Table I Vibration of Generator-exciter

Vibration	Exciter (#7 bearing)		Generator (#6 bearing)	
	2550 RPM	3600 RPM	2550 RPM	3600 RPM
Relative Shaft (vert.)	22.8 mil	2.1 mil	4.7 mil	2.4 mil

2. MODEL AND ANALYSIS OF ROTOR'S DYNAMIC AND STATIC DEFLECTION[1][2]

The turbine-generator system was simplified to lumped (reduced) mass and beam-type model using Sprint® finite element program. The finite element model generated was used for calculation of the static deflection or bearing loading and also for the dynamic analysis of the turbine-generator. The gravitation sag and bearings reaction were calculated for three cases with various initial bearings vertical offsets (Table II).

Table II Gravity Load and Elevation

CASE	I. Tmax=2849 lb/in^2 (Zero Alignment)		II. Tmax=5911 lb/in^2 (Existing Alignment)		III. Tmax=2850 lb/in^2 (Recommended Align.)	
Bearing Location	Brg. Vert. Displ. (in)	Bearing Force (lb)	Brg. Vert. Displ. (in)	Bearing Force (lb)	Brg. Vert. Displ. (in)	Bearing Force (lb)
#1 HP O.B.	0.0	22,810	0.108	24,080	0.100	24,320
#2 HP 1.B.	0.0	27,410	0.12	25,080	0.10	23,870
#3 LP O.B.	0.0	38,880	0.0	39,640	0.0	41,020
#4 LP 1.B.	0.0	41,140	0.0	50,240	0.0	44,110
#5 Gen O.B.	0.0	59,090	0.20	47,040	0.20	54,610
#6 Gen 1.B.	0.0	55,090	0.315	58,540	0.215	56,610
#7n Exciter	0.0	2,832	0.322	2,634	0.265	2,831
TL Brg Load	-	247,172	-	247,254	-	247,251

In the first case, no initial vertical offsets of the bearing pedestals were specified; in the second case, the existing vertical alignment provided by the power plant was used; in the third case, a recommended alignment was presented to increase the load of #7 bearing.

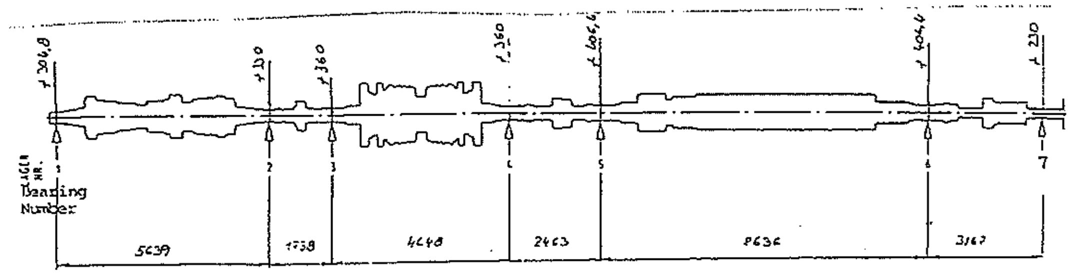


Fig. 2 Schematic outline of 385 MW Turbine-Generator

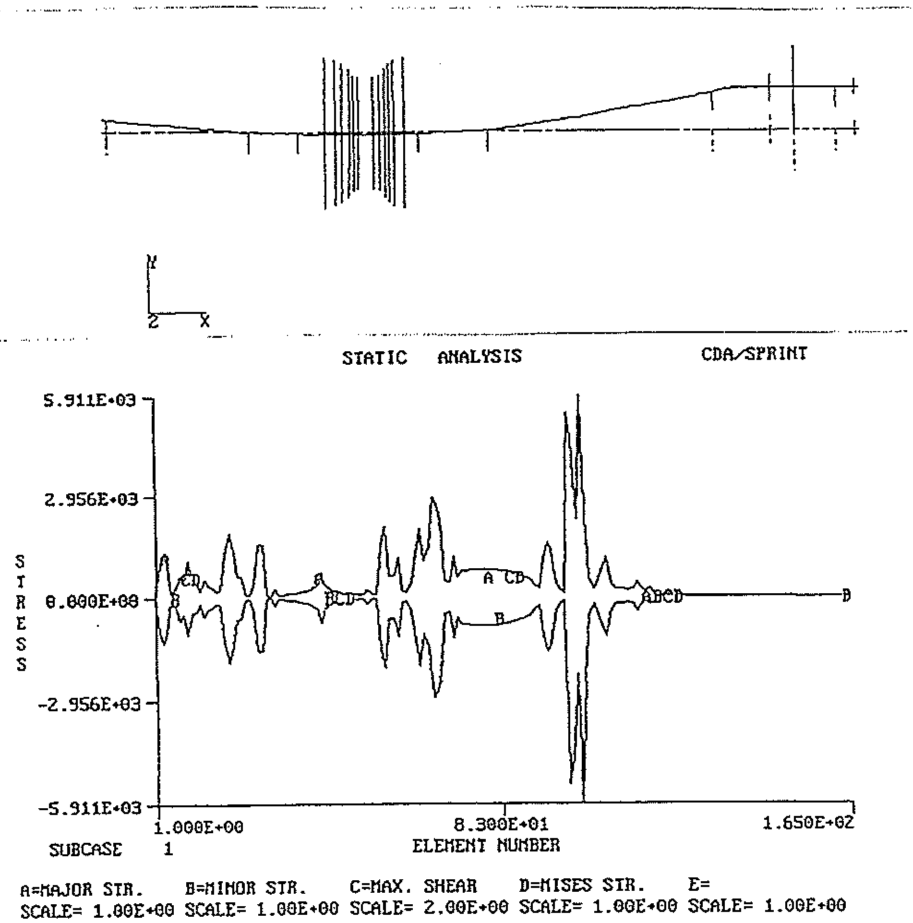
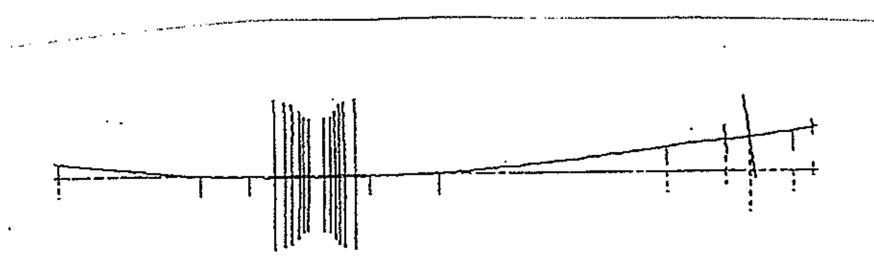


Fig. 3 Static Deflection and Stress Distribution with Current (existing) Alignment Line



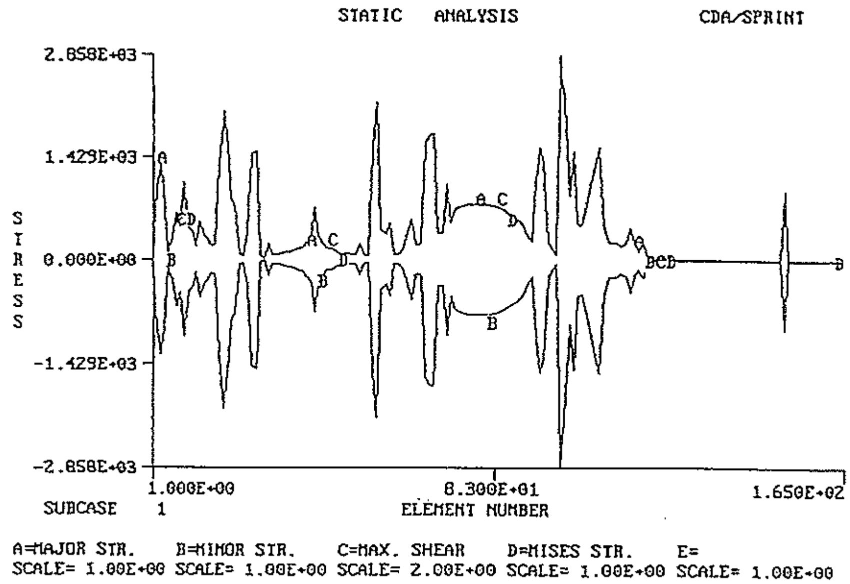


Fig. 4 Static Deflection and Stress Distribution with Recommended Alignment Line

From the calculated results, an interesting observation was made when looking at stress distribution plots (Fig. 3). While current alignment-line did what it is supposed to do with generator operation, static bending stresses at the generator/exciter coupling was "zero"; however, some stresses were introduced when the unit operates and the exciter journal was running in upper bearing quadrant (Fig. 1), moreover, the current alignment caused high stresses over 5,911 lb/in^2 in the generator as shown in Fig. 3; in case III (Fig. 4), the #7 bearing reaction are improved and the stresses are reduced below the case II. The maximum stress is reduced from 5,911 lb/in^2 to 2,850 lb/in^2. In order to check the accuracy of the calculated static deflection, the sum of the bearing reactions must equal the gravitational load of turbine-generator. In Table 2, it is seen that the total bearing loads are in very close agreement to each other.

3. BEARING ANALYSIS

The hydrodynamics characteristic of the exciter bearing was computed. The exciter #7 bearing was a 4 tilting pads bearing. Through an analysis of the bearing, with clearances as currently installed, pressure distribution (Fig. 5.) over the speed range indicates a drop, exactly in the region of the exciter rotor resonance's 1860-2484 RPM.

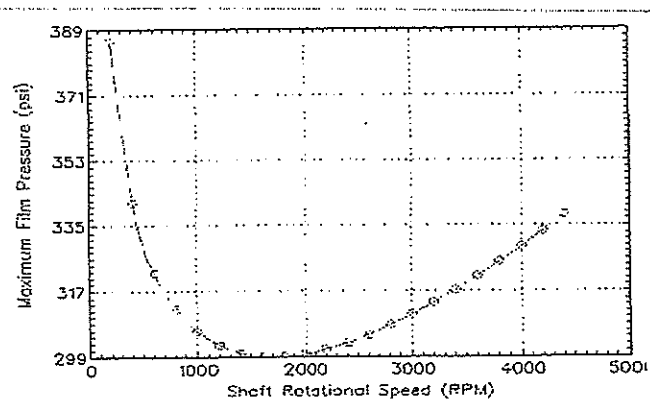


Fig. 5 Maximum Film Pressure vs. Speed

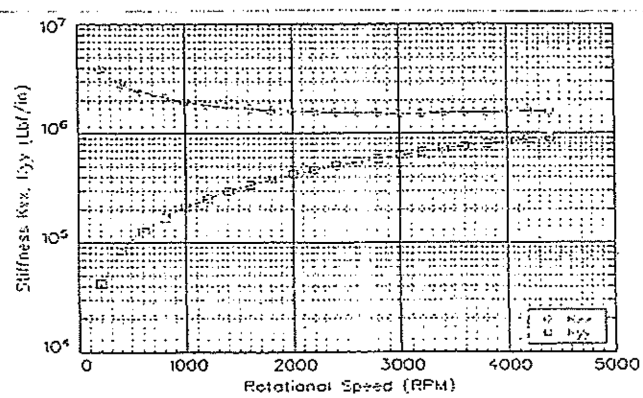


Fig. 6 Four Pad Stiffness Curves vs. Speed

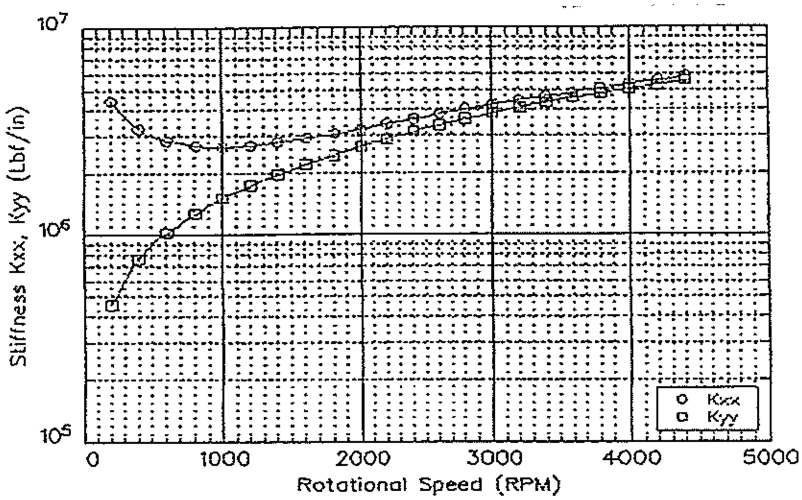


Fig. 7 Five Pad Stiffness Curves vs. Speed

4. RECOMMENDATIONS

Based on the experimental and theoretical investigation, two approaches were presented, and the choice depends on the economics and the practicality in application:

(1) Alignment change

A-1 Lower the generator bearing #6 by 0.1 in. from design value (assumed to be 0.315 in)

-2 Raise exciter bearing by 0.050 in. above generator bearing (assumed to be 0.265 in)

B-1 Leave generator as is.

-2 Raise exciter bearing #7 by 0.050 in. Above generator bearing #6.

Version A will reduce overall bending stresses on the entire train (LP, Generator) and distribute bearing loading more evenly. On the negative side it may renew a suspected instability problem with the generator bearing #6, if it existed in the past. Version B will leave Generator stresses at current level and bearing loading less evenly distributed. The alignment changes alone will not solve the vibration problems completely. For a complete solution, #7 bearing must be changed. Even here there are two options:

(2) Bearing Re-design

A Four pad tilting bearing #7 of the exciter will be better replaced with the five pad tilting bearing with load on the bottom pad, installation clearance of approximately 0.8 mils/inch of diameter, and preload of 40-50%.

B Reduce clearances of the existing bearing and assure there is 40% preload on the pads. The choice between A and B is purely of economic nature. Bearing characteristic are almost identical. The only difference is with consideration to alignment, since A bearing has 5 pads with load on the pad and bearing B has 4 pads with load between the pads, and rotor sinking between the pads, with four pad bearing must be taken into account.

C In either case verify, and machine as necessary the bearing fit, and check good anchoring of the pedestal.

5. CONCLUSION

After installing the redesigned bearing as described in step (2) A above, the high rotor vibration of the exciter #7 bearing were dramatically reduced. On February 3, 1996 the unit was given its first start after the bearing change out. The rotor vibrations through the critical were reduced significantly even without balancing. Through the investigation of this problem, it was shown that the clearances of a lightly loaded bearing , journal vibration, and total system stiffness and damping were important factors in influencing the working performance of the bearing.

REFERENCES

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2. E. J. Gunter, Rotor Modeling for System Dynamic Analysis (1994).