MODIFICATION of the EXCITER BEARING

on LARGE NUCLEAR TURBINE GENERATORS

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Abstract:

This paper deals with the experimental and theoretical investigation of unstable vibrations which were observed at the #11 bearing at the exciter. Vibrations of the #11 bearing change with the load change of the unit and are much larger in the horizontal than in the vertical direction. An experimental investigation has indicated that the #11 bearing is very lightly loaded. A detailed theoretical analysis shows that the increase of gravity load and/or reducing the clearances of the existing bearing would lead to another problem, i.e., it would bring the rotor "first critical" even closer to operating speeds of 1800 RPM. The solution found is to change to another type of bearing. The selected bearing is a cylindrical offset halves bearing. With this bearing the rotor system critical will be moved above the unit operating speed, thus becoming less "sensitive" to variations of excitations due to changes of the operating parameters.

1. Introduction

The Grand Gulf exciter's bearing #11 had high shaft vibrations and vibration seems to change over time, much more visible in the horizontal direction.

Table 1 Vibrational Trend of #11 Bearing

Date	4/2/92	7/9/92	12/2/93
Ampl.Vertical Dir. (mils)	3.0 p-p	6-7 p-p	6-7 p-p
Ampl. Horizontal Dir. (mils)	-	-	14 p-p

Exciter rotor had required balancing after the start-up following an outage, but operated well after balancing. Some time later, during the operation, there were some electrical disturbances in the distribution system, after which a change in #11 bearing vibrations were noted. Subsequent balancing (in single plane) showed small improvement, but a negative effect was observed on the adjacent bearings. Following the historical analysis, it was decided to perform an in depth study of the exciter rotor/bearing system.

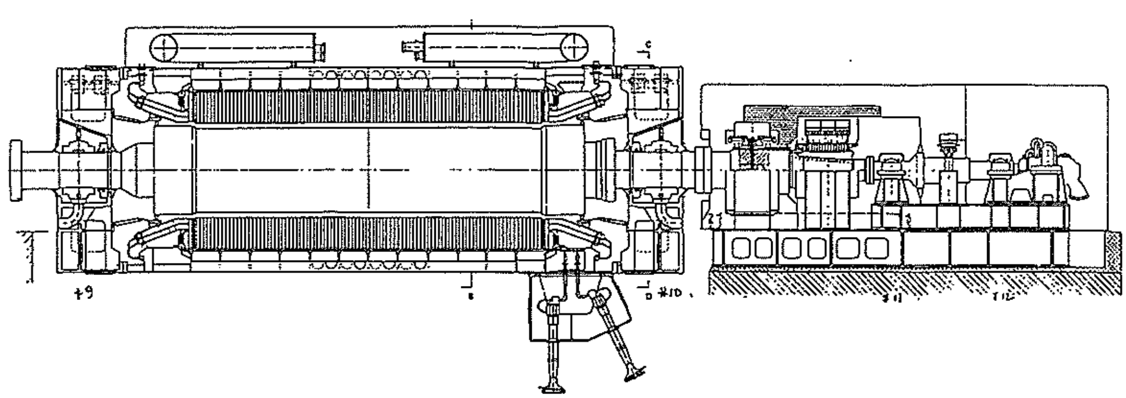


Fig. 1 Schematic Outline of Generator-Exciter

In order to solve the problem, the dynamic model of generator and exciter was built, and the calculations were focused on three points: Load distribution and alignment, the analysis of properties of the original #11 bearing with the changes in gravity load and clearances, and the analysis of recommended new type of bearing.

2. Model and Analysis of Generator-Exciter Dynamics and Static Characteristic

The generator-exciter was simplified to lumped mass and beam-type model using Sprint Finite Element program, the finite element model generated was used for calculation of the alignment curve and bearing loading, and also for the calculation of unbalance response. The gravitation sag and bearings' reaction were calculated for two cases, one is based on original alignment, the other is for changed alignment. (Table 2)

Table 2 The Bearing Static Load

Case	Original Alignment		Changed Alignment	
Bearings	Brg.offset (in)	Load (lbf)	Brg. Offset (in)	Load (lbf)
Gen #9	0.000	220600	0.000	221300
Gen #10	0.000	268900	0.000	265900
Exc. #11	0.008	2187	0.012	5901
Exc. #12	00.20	7838.	0.020	6481

Table indicates that the original alignment caused light loading of #11 bearing and it seems to be a good way to raise the elevation of the #11 bearing, but this would cause another problem. At the original alignment case, the first critical speed was 1625 RPM in field, however the operational speed is 1800 RPM. In this case, if the elevation of #11 bearing is raised, it will lead to increase of stiffness of #11 bearing, and the increase of the stiffness will result in the first critical speed of exciter to move closer to the operational speed. Therefore, before making any change of #11 bearing, a detailed calculation of bearing is necessary.

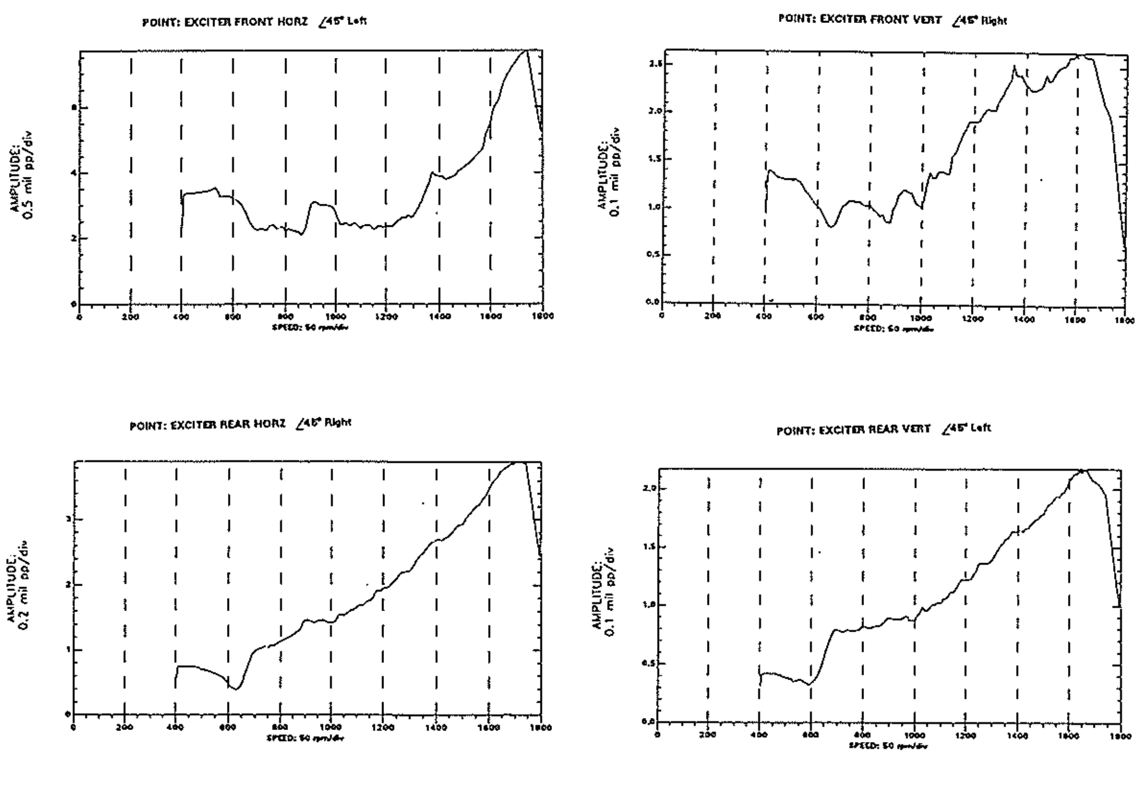
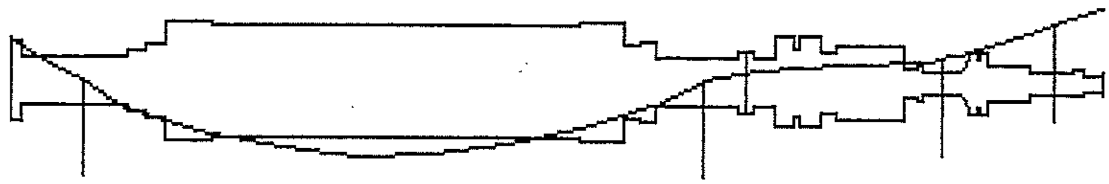


Fig. 1 Bode Plots



a. Original Alignment

b. Changed Alignment

Fig. 2 Deflection Curve of Generator-Exciter

3. Bearing Analysis

The hydrodynamics bearing characteristic of the #11 bearing of the exciter were computed; the exciter #11 and #12 bearings were partial arc bearing, the contacting arc is 59 degrees. Based on suggested solution of the vibrational problem of the #11 bearing, four cases of calculations were made, (see Table 3 and Fig. 3-6).

Table 3 Calclulating Condition of #11 Bearing

Case	1	2	3	4
Type of Bearing	Partial Arc Bearing	Partial Arc Bearing	Partial Arc Bearing.	Offset Cylindrical Bearing
Load of Bearing (lbf)	2187	2187	5901	2187
Bearing Clearance (in.)	0.0077	0.0060	0.0077	0.0060
Pad Clearance (in.)	0.0077	0.0060	0.0077	0.0120

The calculation shows that current #11 bearing (case 1) had low damping and stiffness, and negative cross stiffness. The attitude angle indicated that minimum film area was out of contacting area of partial arc bearing.

Partial Arc Bearing, ARC = 59 degrees

Offset
0.0000

Preload
0.0000

Shaft Speed
1800 RPM

Bearing Load
2187.00 Lbf

Max. Film Pressure
68.79 psi

Friction Power Loss
5.24 hp

Stiffness (Lbf/in)
9.73E+05 1.46E+06
-2.74E+04 8.08E+04

Damping (Lbf-s/in)
1.56E+04 4.41E+02
4.41E+02 3.87E+02

Critical Journal Weight = 2541.40 Lbf

Sommerfeld no
2.6015
E/Cb=0.3017
Attitude Angle
41.31

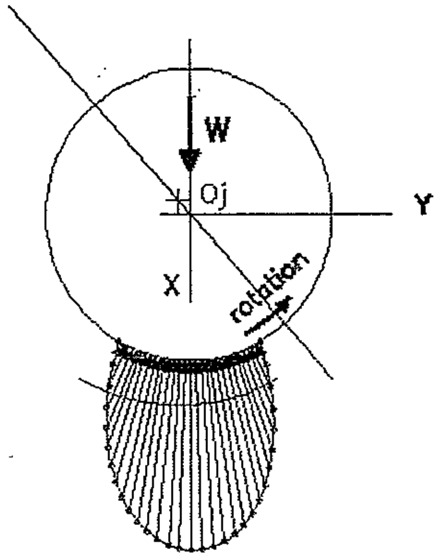


Fig. 3 Profile Pressure of #11 Bearing (Case 1)

Partial Arc Bearing, ARC = 59 degrees

Offset
0.0000

Preload
0.0000

Shaft Speed
1800 RPM

Bearing Load
2187.00 Lbf

Max. Film Pressure
68.37 psi

Friction Power Loss
6.03 hp

Stiffness (Lbf/in)
1.13E+06 2.28E+06
-4.81E+04 9.37E+04

Damping (Lbf-s/in)
2.43E+04 5.05E+02
5.05E+02 5.94E+02

Critical Journal Weight = 3062.62 Lbf

Sommerfeld no
4.2845
E/Cb=0.2127
Attitude Angle
49.65

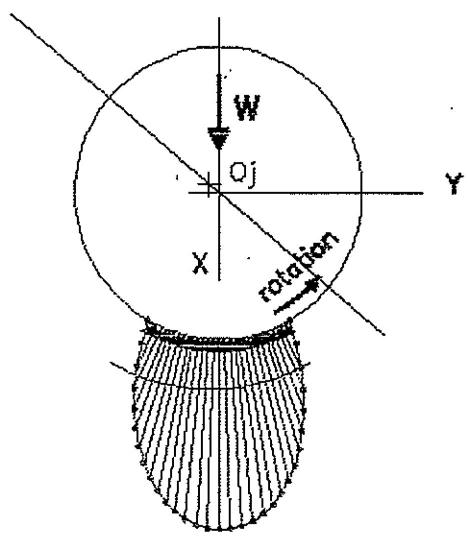
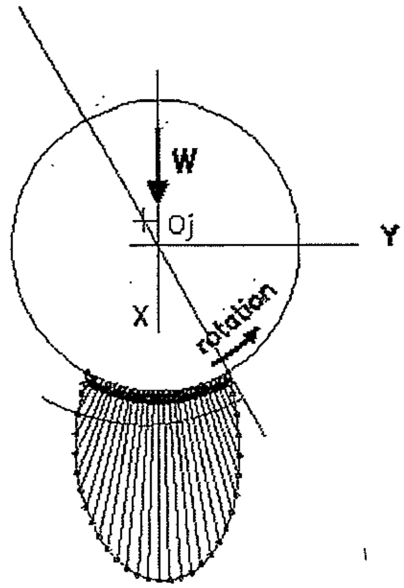


Fig. 4 Profile Pressure of #11 Bearing (Case 2)

Partial Arc Bearing, ARC = 59 degrees

Offset
0.0000
Preload
0.0000
Shaft Speed
1800 RPM
Bearing Load
5901.00 Lbf
Max. Film Pressure
189.44 psi
Friction Power Loss
7.09 hp
Stiffness (Lbf/in)
3.43E+06 3.32E+06
-3.84E+04 2.86E+05
Damping (Lbf-s/in)
3.61E+04 1.63E+03
1.63E+03 9.33E+02



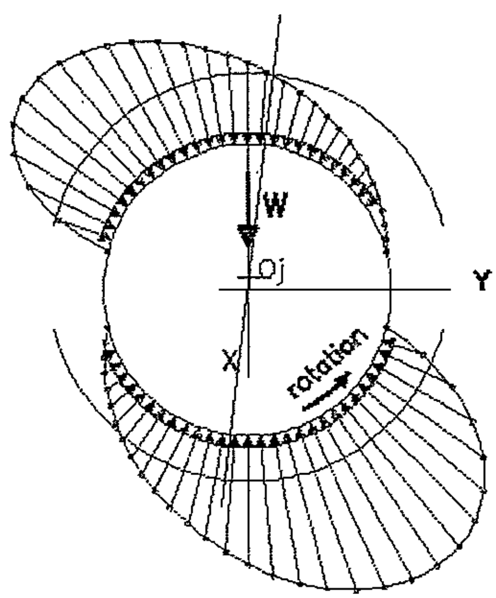
Sommerfeld no
0.9641
E/Cb=0.4916
Attitude Angle
29.80

Critical Journal Weight = 7880.81 Lbf

Fig. 5 Profile Pressure of #11 Bearing (Case 3)

Offset Half Bearing

Offset
1.1000
Preload
0.5000
Shaft Speed
1800 RPM
Bearing Load
2187.00 Lbf
Max. Film Pressure
150.11 psi
Friction Power Loss
16.07 hp
Stiffness (Lbf/in)
2.15E+06 3.47E+06
2.72E+05 2.44E+06
Damping (Lbf-s/in)
2.44E+04 9.74E+03
9.74E+03 1.14E+04



Sommerfeld no
1.0711
E/Cb=0.2196
Attitude Angle
-7.01

The Equilibrium Position is Stable

Fig. 6 Profile Pressure of #11 Bearing (case 4)

In general, two ways may be used to adjust the #11 bearing performing characteristics: (1) to increase the load of the bearing; (2) to reduce the clearance of the bearing. Fig. 4 shows that reducing clearance on the existing bearing is not a good solution for this type bearing.

Increasing the stiffness by closing bearing vertical clearance forces the attitude angle at a lower angle. This would be a positive side of reducing clearances. On the negative side, the increased stiffness would bring the exciter rotor first critical even closer to operating speed, not sufficiently higher above critical speed, creating larger sensitivity to unbalance, and larger vibration amplitudes.

Therefore, it is necessary to find the type of bearing which would increase system stiffness sufficiently to move the rotor resonance comfortably above operating speed. The best way to solve this problem is that the original design will not be changed too much, easy to accomplish and economical. One bearing which satisfies all criteria is an offset-halves bearing.

Fig. 6 shows the theoretical calculation result, and former experience which had demonstrated good performance in the field. The dynamic calculations show that the exciter with offset cylindrical bearing will have much better dynamic characteristic than that of the current bearing.

4. Comparison of the Dynamic Characteristic

4.1 Critical Speed Analysis

Assuming the #11 bearing stiffness of $(k_{xx} + k_{yy})/2$, the critical speeds of exciter were calculated based on the computation result of bearing analysis.

Table 4 Critical Speeds of Generator-Exciter

Type of Bearing	Partial Arc Cylindrical Bearing				Offset Cylindrical Bearing			
Number	1	2	3	4	1	2	3	4
Critical Speed	566	1049	1652	3186	568	1130	2162	3259
	1st Gen.	2nd Gen.	1st Exc.	2nd Exc.	1st Gen.	2nd Gen.	1st Exc.	2nd Exc.

4.2 Unbalance Response

4.2.1 Unbalance response of exciter with partial arc #11 bearing

The unbalance response of the generator-exciter with partial arc #11 bearing was simulated in Fig. 7, the unbalance mass of 1000 g was placed in the main body of the exciter. It can be seen that the exciter is operating on top of the critical speed (and also in the critical speed regions). So its vibration is more sensitive to the stiffness change of #11 bearing, the major factor which influence the stiffness during operation was the change of elevation due to the change of load of generator.

4.2.2 Unbalance response of exciter with offset cylindrical #11 bearing

A simulation was performed in which the #11 bearing of the exciter was changed to offset cylindrical bearing in order to increase the damping and the critical speed. The offset cylindrical bearing's parameters are as follows; nominal diameter is 13 inches, length is 10.236 inches, preload m is 0.5, the offset is 1.1, the leading edge 105 degrees, the trail edge is 255 degrees, the vertical radial clearance is 0.006 inches, the pad radial clearance is 0.012 inches, the load of bearing is same as current #11 bearing. That means that no alignment changes are required to achieve the desired goal.

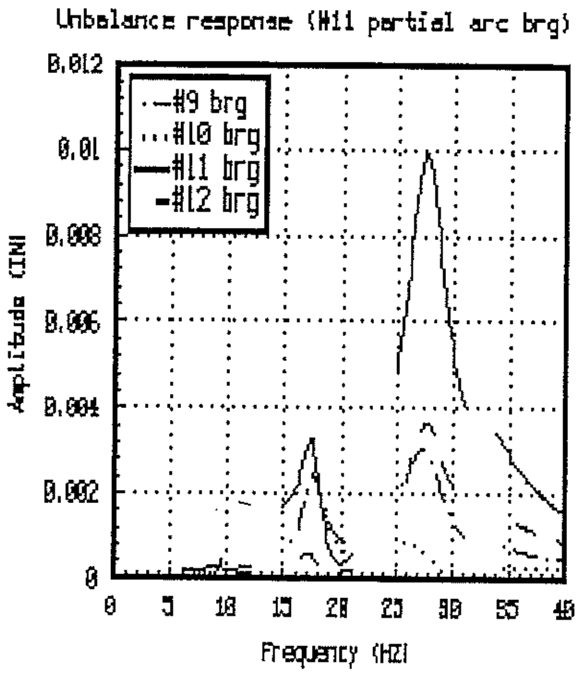


Fig. 7

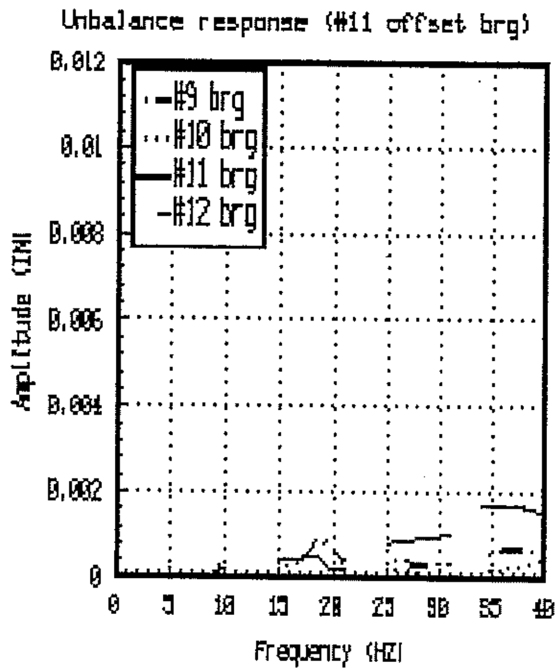


Fig. 8

The response of the exciter with offset cylindrical #11 bearing was calculated using same unbalance mass (see Fig. 8). The first critical speed of the exciter is changed from 1652 RPM to 2165 RPM, thus the exciter with this design concept is very insensitive not only to unbalance, but also to the change of the elevation during operation, compared to that of using the current #11 bearing.

5. Conclusion and Recommendation

This analysis and the case history shows that the behavior of the rotor bearing system may change over period of time. In special case when the system resonance is in the proximity of the machine operating speed, a subtle change in assembly clearances, aging of the concrete support or the alignment variations, the vibrational behavior of the system may change.

In order to make the system less sensitive, the solution may be to remove the resonance from the operating range, if possible, by modification of the bearing. Of course, full understanding of the behavior of various bearing types and the economic considerations are pre-requisites for the proper bearing selection in each particular case.