SOLVING STABILITY PROBLEMS WHILE COMMISSIONING A 100 MW TURBINE GENERATOR SET

by
John K. Whalen
Engineering Manager and President
TCE/Turbo Components and Engineering, Inc.
Houston, Texas
and
Malcolm E. Leader
Turbomachinery Consultant and Owner
Applied Machinery Dynamics
Dickinson, Texas

John K. Whalen is the Engineering Manager and President of TCE/Turbo Components and Engineering, Inc., in Houston, Texas. He spent seven years at Turbodyne Steam Turbines (Dresser-Rand) as a Product Engineer in the Large Turbine Engineering Department and as an Analytical Engineer in the Rotordynamics Group of the Advanced Engineering and Development Department. In 1988, Mr. Whalen accepted a position with Centritech, as the Assistant Chief Engineer. In 1989, he was promoted to Manager of Engineering. In 1991, he left Centritech to help start TCE. At TCE, he is responsible for the engineering department and engineering for the product lines, which include babbitted journal and thrust bearings, labyrinth seals, and related engineering services.

Mr. Whalen received his B.S. degree (Mechanical Engineering, 1981) from the Rochester Institute of Technology. He is a member of ASME, STLE, and the Vibration Institute, and is a registered Professional Engineer in the State of Texas.

Malcolm E. Leader is a Turbomachinery Consultant and Owner of Applied Machinery Dynamics, in Dickinson, Texas. He is currently involved in the design, testing, modification, and installation of rotating equipment. He spends time doing theoretical design audits and working in the field implementing changes and overseeing installations.

Mr. Leader obtained his B.S. (1977) and M.S. (1978) degrees from the University of Virginia. While there, he worked extensively on experimental rotordynamics and hydrodynamic bearing design. He has written several papers on the subjects of experimental rotordynamics, bearing design, design audits for rotating equipment, and practical implementation of rotordynamic programs. Mr. Leader is a member of ASME, Sigma Xi, the Houston Chapter of the Vibration Institute, and is a registered Professional Engineer in the State of Texas.

ABSTRACT

A new cogeneration installation was in the process of being commissioned when severe subsynchronous vibration was detected on the generator during no load tests. Subsequently, two rotordynamics problems were discovered. These problems prohibited the steam turbine generator set from reaching full speed. Additional vibration problems were anticipated once the generator was loaded and the steam flows through the turbines increased.

The first subsynchronous vibration problem, observed during initial full speed, no-load run attempts, was found to be most severe at the inboard generator bearing and the adjacent turbine bearing. The second problem, uncovered during analysis of the generator vibration problem, was the presence of low-level subsynchronous vibration at a different frequency on both bearings of the high pressure-intermediate pressure (HP-IP) turbine rotor. The concern was that this vibration would be exacerbated once the machine was loaded and additional aerodynamic cross-coupling was introduced by steam flow at seal and wheel locations.

This paper details the discovery of the problems, initial attempts to address them, and the use of rotordynamics tools to engineer a solution to the problems by the design, manufacture, and installation of optimized bearings.

INTRODUCTION

The steam turbine generator (TG) set (Figure 1) is of a mixed pressure design. The steam turbine unit consists of a single cylinder low pressure (LP) section rigidly coupled to a second turbine containing a high pressure (HP) section and an intermediate pressure (IP) section. From left to right there is the #1 journal bearing, the LP blade section (with axial flow exhaust), the #2 bearing, and a rigid coupling between the LP and HP-IP rotors. The HP-IP turbine is made up of the #3 (HP) bearing followed by the HP blade section, a long labyrinth seal, the IP blade section, and finally, the #4 (IP) bearing. The turbines are connected to the generator by another rigid coupling. The generator bearings are designated as #5 bearing (inboard), and the #6 bearing next to the exciter. An overhung exciter is outboard of the #6 bearing. All six journal bearings were initially believed to be elliptical sleeve bearings.

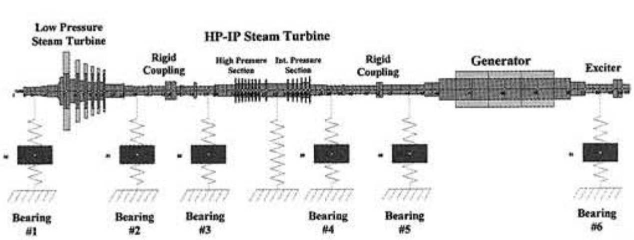

Figure 1. Cross Section Schematic of Steam Turbine-Generator Train.

Initial Problem Discovery

While commissioning this train the generator rotor at the #5 bearing location (next to the turbine) experienced large amplitude subsynchronous vibrations that prohibited operation at the 3600 rpm running speed. Figure 2 is a cascade spectrum plot at the #5 location. This is a series of vibration spectra taken during a startup at equal speed increments. Note the clear presence of high amplitude subsynchronous vibration at 1080 cpm. It was thought that possibly the static loading on the #5 bearing was low. This could introduce substantial cross-coupling, resulting in oil whip. In an attempt to increase the unit load on this bearing, an alignment change was made. The #5 bearing elevation was raised causing this bearing to accept more static journal load. It was hoped that the increase in the bearing loading would reduce the cross-coupling enough to avoid the instability. However, this action did not have the desired effect, as there was no change in the vibration behavior.

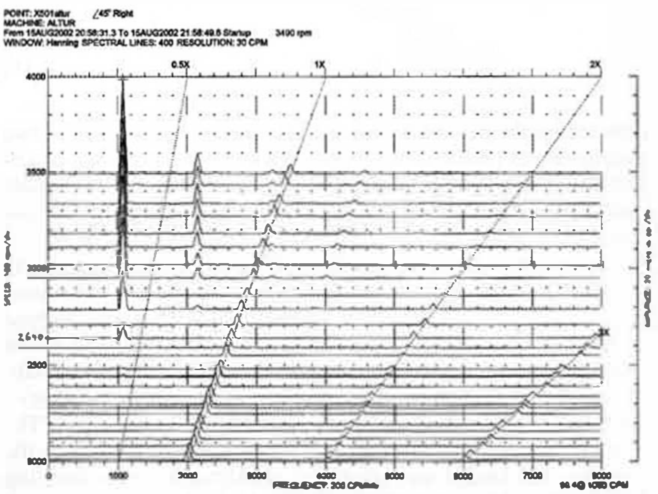

Figure 2. Cascade Spectrum Plot of Original Generator Vibration Problem.

Since subsynchronous vibration was also present on the IP turbine bearing a decision was made to modify the #4 bearing in an attempt to reduce the cross-coupling from that location. The turbine manufacturer had representatives onsite to approve this modification, and they were willing to share information. At this time it was difficult to get the generator manufacturer (an overseas supplier) to agree to make any changes to the generator bearings or even supply useful data.

One way to increase the bearing unit loading, and thus the stability, is to reduce the effective load carrying area. Thus, the babbitt was removed in two circumferential arcs, one from each end of the #4 bearing (Figure 3). The reduction in load capacity increases the journal eccentricity and decreases the attitude angle. This change should have resulted in a decrease in cross-coupling in the bearing, and it was hoped that this reduction in cross-coupling at the #4 location would help eliminate the instability at the #5 location. However, the installation of the modified #4 bearing had no appreciable impact on the #5 bearing vibration problem.

A decision was made to convert the #4 bearing to a tilting pad journal (TPJ) bearing design to virtually eliminate its destabilizing contribution. Since this was now a radical change in bearing design, it was agreed that a machinery consultant should be brought in to analyze the situation and perform an optimization study for the bearing upgrade at the #4 location. A bearing manufacturer was also called in to initiate the generation of manufacturing drawings for the TPJ, and to work with the consultant on feasible TPJ designs. This unit was in commissioning and the startup was being delayed due to this vibration problem. Substantial penalties were at stake so the bearing optimization, bearing design for manufacture, and even the manufacture of the bearing needed to be performed concurrently.

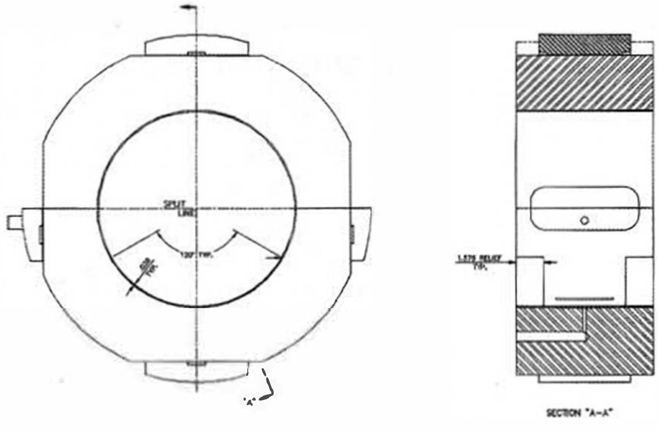

Figure 3. Drawing of Modified IP Elliptical Bearing.

ROTORDYNAMICS ANALYSIS

The computer model of the train was shown previously as Figure 1. The rotor cross-section is shown supported by springs that represent the stiffness and damping of the fluid film bearings. The "box" in the middle of each spring represents the mass of the support structure and the spring beneath each box models the stiffness and damping of the support to ground. For this analysis specific bearing pedestal information was not available. However, based on experience with similar machinery trains, each pedestal was assigned a modal mass of 10,000 lb with a horizontal stiffness of 7 million lb/in and a vertical stiffness of 8 million lb/in. Ten percent of critical damping was assigned to each support based on previous testing on similar machines. There is another spring attached to the center of the HP-IP turbine used as a means to simulate aerodynamic cross-coupling at that location.

Since there are no flexible couplings in this train to isolate the rotors laterally, a complete train model was required. The turbine manufacturer supplied sufficient drawings of the turbine rotors and the #2, #3, and #4 bearings. This information was used in modeling the turbines for the rotordynamics analysis (RDA). Reasonable assumptions were made about the #1 bearing. However, no drawings were available for the generator rotor. The only information available for #5 and #6 generator bearings was basic dimensions supplied by the turbine vendor. At this time the generator bearings were believed to be elliptical.

The model of the generator was constructed based on what information was known. This included some shaft dimensions, journal diameters, bearing span, and total rotor weight. In addition, the measured first critical speed frequency was known from testing as well as the whirl frequency during unstable operation. The analyst had previously modeled similar generator rotors and was able to construct a generator model that was sufficiently accurate.

Undamped Critical Speeds

As a first step toward understanding the nature of any rotor-bearing system, the undamped critical speed mode shapes are calculated. The actual frequencies are not important at this stage since the inclusion of damping can have a large influence on the speeds at which maximum amplitudes are observed. The first critical speed mode shape for this train is plotted in Figure 4. This is a cylindrical mode of the generator with maximum amplitude near the center of the generator rotor. While some of the effects of this resonance carry over to the IP end of the adjacent turbine, these are minor. The next resonance, Figure 5, is the first critical speed mode shape of the HP-IP turbine. This is also a cylindrical mode and is fairly well isolated to that body. The first critical speed of the LP turbine is not shown. All three of these first mode resonances are well below operating speed and will be encountered during every startup and shut down. The only other resonance that is significant to this analysis is the second critical speed of the generator, Figure 6. This mode is pivotal and due to the high relative amplitude at the bearings, it is expected to be a fairly well damped resonance.

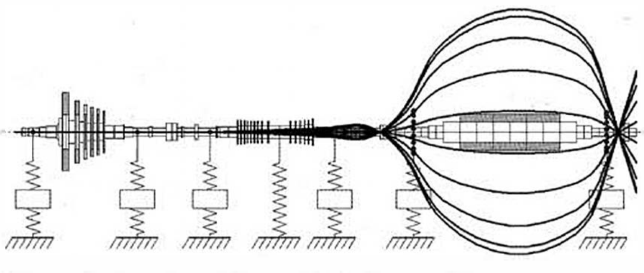

Figure 4. First Critical Speed Mode Shape of Generator.

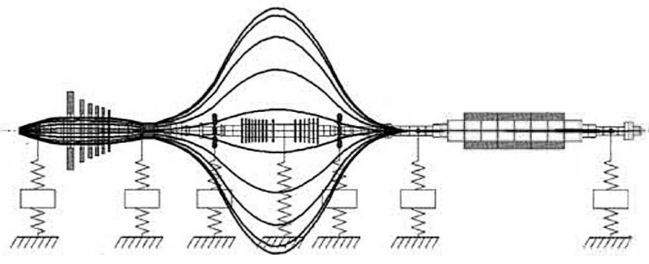

Figure 5. First Critical Speed Mode Shape of HP-IP Turbine.

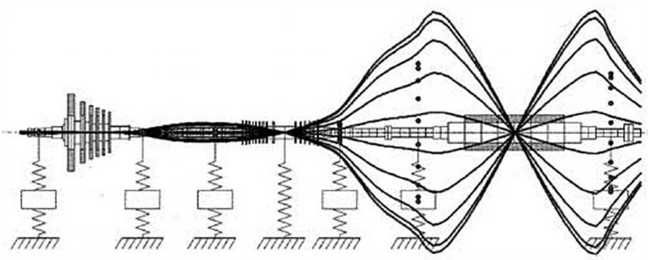

Figure 6. Second Critical Speed Mode Shape of Generator.

Bearing Coefficient Determination

In a rigidly coupled, multibearing rotor train, the bearing loads are statically indeterminate, and the bearing bores are never all collinear. If these rotors were to be set with all the bearings at the same elevation, Figure 7, the rotor sags would result in large bending forces in the shafts. Instead, the elevation of each bearing pedestal is adjusted so that the mating flanges between rotors have no moment forces in operation. In this case, with three rotors and two bolted joints, the pedestals on the LP turbine and on the generator are elevated. Since the generator rotor is most flexible, the exciter end is elevated nearly 3/16 of an inch so that the catenary curve causes the bolted flanges to meet squarely. Figure 8 illustrates the static rotor elevations and the catenary curve for this turbine-generator set. Once the pedestal elevations are set, the actual bearing loads can be calculated.

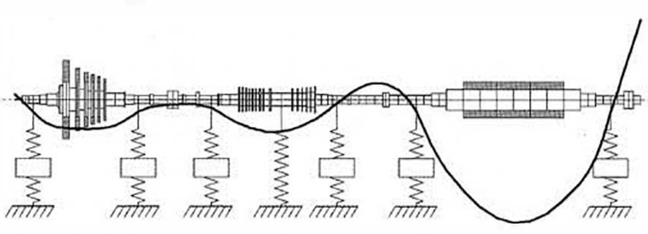

Figure 7. Rotor Deflection Shape with Collinear Bearings.

It was determined that the generator bearing nearest the turbine carried a static load of 32,770 lb. This bearing is 12.4 inches in diameter and 10.67 inches long. The unit loading (W/LD) is 248 psi. The exciter end bearing carried 23,730 lb on a bore of 11.02 inches and a length of 9.41 inches. The unit loading is 229 psi.

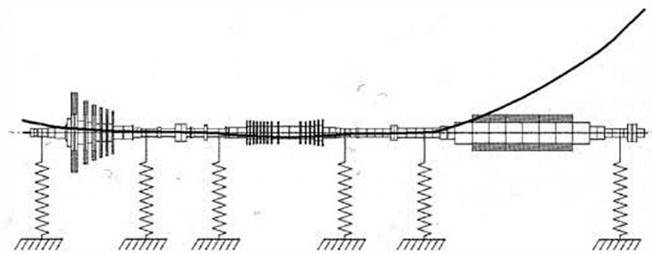

Figure 8. Rotor Catenary Shape with Properly Elevated Bearing Pedestals.

Using a finite element-based program, the stiffness and damping coefficients of all six bearings in the train were calculated over the range of 500 to 4000 rpm at 250 rpm intervals. Initially, the generator bearings were thought to be elliptical and the calculated stiffness and damping coefficients for 3600 rpm are shown in Table 1 using the average clearance given by the manufacturer.

Table 1. Generator Elliptical Bearing Coefficients.

Elliptical Bearing Coefficient	Bearing #5	Bearing #6
Principal horizontal stiffness Kxx lb/in	2.48×10^6	2.05×10^6
Cross stiffness Kxy lb/in	6.38×10^5	9.41×10^5
Cross stiffness Kyx lb/in	-6.53×10^6	-5.25×10^6
Principal vertical stiffness Kyy lb/in	9.42×10^6	7.66×10^6
Principal horizontal damping Cxx lb-sec/in	7.9×10^3	7.6×10^3
Cross damping Cxy lb-sec/in	-6.2×10^3	-4.2×10^3
Cross damping Cyx lb-sec/in	-6.2×10^3	-4.2×10^3
Principal vertical damping Cyy lb-sec/in	3.6×10^4	3.0×10^4

Unbalance Response Analysis

Once the rotor modeling has been accomplished, it is always interesting to compare the predicted synchronous imbalance response of the rotor to actual measured field data. This was done for all three machines with good agreement. Figure 9 is a Bodé plot from a startup showing the synchronous generator vibration amplitude and phase as a function of speed. The amplitudes are low due to a very low level of rotor imbalance. Figure 10 is the associated predicted unbalance response for the same shaft location. While not a perfect match, there is excellent agreement on the first critical speed frequency at 1250 rpm. The generator second critical speed is visible but it is very well damped.

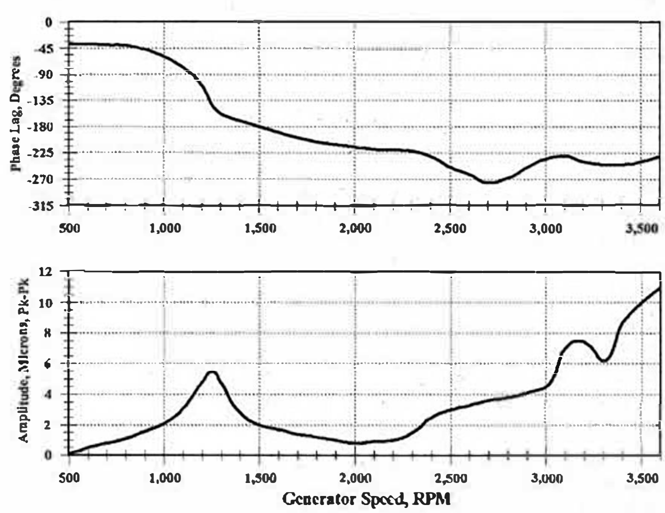

Figure 9. Measured Synchronous Startup Vibration from Generator Inboard Bearing Location.

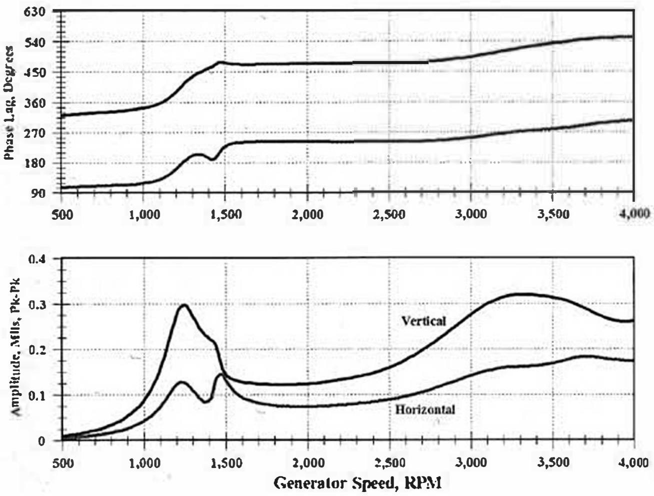

Figure 10. Predicted Synchronous Startup Vibration from Generator Inboard Bearing Location.

Stability Analysis

As the analysis progressed it was discovered that the stability problem observed at the #5 bearing location was not being predicted. This left two options: refine the modeling efforts and analysis to determine why the model was not agreeing with the observed behavior, or accept the analysis as a valid base case. In the interest of time it was decided to keep the current model and quickly optimize a TPJ bearing for the #4 location that would maximize the stability at the #5 location. In order to compare various bearing design cases, enough cross-coupling was analytically applied at the generator midspan until the logarithmic decrement (log dec) went to zero. This is the theoretical stability threshold and would be the base case.

When the #4 bearing design was fully optimized, the analysis was still predicting instability in the generator. This made it clear that a change to the #4 bearing alone would not solve the problem. All these analyses were run with the analytically applied cross coupling in place at the generator midspan to force the model to be calibrated to the observed behavior. Thus, it was decided that the #5 bearing needed to be upgraded to a TPJ. Meanwhile, work commenced on the drawings and manufacture of the #4 TPJ, based upon the optimization performed.

A window presented itself where a modified #5 sleeve bearing could be tried as an interim solution. Further commissioning testing could proceed if the unit could be run up all the way to trip speed. The goal was to determine what geometry could be machined in the bore of an existing #5 bearing to maximize stability. Initially a reduced axial length design with a circumferentially shortened lower half was considered, and drawings for the modification were made.

Meanwhile the #5 bearing arrived at the bearing manufacturer's site and was carefully measured. It was quickly discovered that the bearing was cylindrical, not elliptical as originally reported. When new bearing coefficients were calculated and this information was incorporated into the stability analysis, excellent agreement was obtained between the model and the field data. No cross-coupling had to be applied to the generator rotor to calibrate the model. Confidence in the rotor and bearing models increased substantially, resulting in a high level of agreement that the optimized #4 and #5 bearings would solve the problem.

Figure 11 is a summary of the stability calculations for the generator with plain bearings. The stability threshold is predicted to occur at about 2600 rpm with no external cross-coupling applied. Looking at Figure 2, the first signs of subsynchronous vibration are seen in the spectrum taken at 2640 rpm. The predicted operating speed logarithmic decrement is −0.149, which had already been confirmed by the severe vibrations.

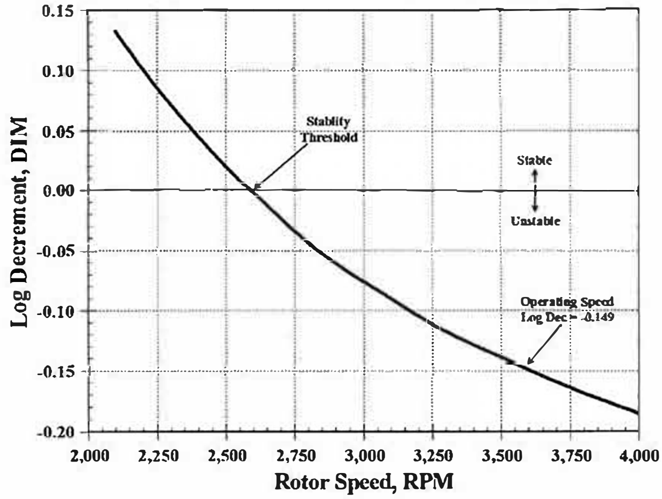

Figure 11. Logarithmic Decrement Prediction for Generator with Actual Plain Bearings.

It is interesting to compare the bearing coefficients of the actual plain bearings with those shown for the elliptical bearing above in Table 1. Table 2 lists the coefficients for the actual generator bearings at 3600 rpm. The principal horizontal stiffness and damping have both increased. The principal vertical stiffness is about the same, while the vertical damping has increased. However, the Kyx cross-coupled stiffness is the dominant controller of stability, and this has doubled compared to the elliptical bearings and is what is driving this rotor into the subsynchronous whip. Figure 12 is a three-dimensional plot of the forward unstable motion of the rotor at 3600 rpm.

Table 2. Generator Cylindrical Bearing Coefficients.

Cylindrical Bearing Coefficient	Bearing #5	Bearing #6
Principal horizontal stiffness Kxx lb/in	5.93×10^6	4.81×10^6
Cross stiffness Kxy lb/in	2.56×10^6	2.13×10^6
Cross stiffness Kyx lb/in	-1.25×10^7	-1.02×10^7
Principal vertical stiffness Kyy lb/in	9.49×10^6	7.58×10^6
Principal horizontal damping Cxx lb-sec/in	2.1×10^4	1.7×10^4
Cross damping Cxy lb-sec/in	-1.9×10^4	-1.5×10^4
Cross damping Cyx lb-sec/in	-1.9×10^4	-1.5×10^4
Principal vertical damping Cyy lb-sec/in	6.5×10^4	5.4×10^4

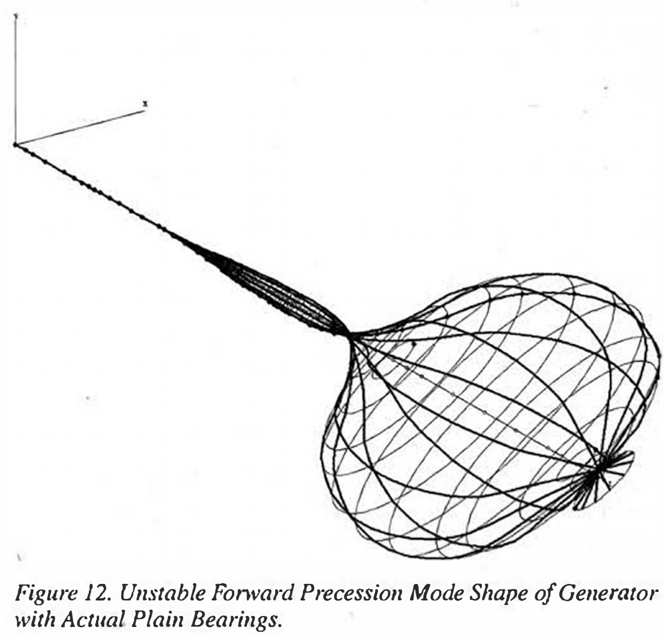

Figure 12. Unstable Forward Precession Mode Shape of Generator with Actual Plain Bearings.

The RDA and bearing optimization continued with a sidetrack to look at options available to modify the existing #5 bearing for increased stability. Now that it was known that the bearing bore was cylindrical, the option to modify it to actually be elliptical was available. Numerous design options were considered including pressure dam, shortened effective length, and elliptical designs. The full-length elliptical proved to be the best design for stability, and the bearing was rebabbitted and remachined with an elliptical bore. This bearing was installed and run successfully up to the 3960 rpm trip speed with no subsynchronous vibration observed at the #5 location. The owner decided that the new TPJ for the #5 location was still needed since the calculated log dec was low, and slight changes such as alignment variations, increased bearing clearances, or oil temperature swings could push the unit into unstable operation.

POTENTIAL HP-IP INSTABILITY FOUND

As the RDA was being run and compared to field test data it was discovered that a subsynchronous vibration, not related to the generator, was occurring in the HP-IP rotor. Figure 13 is a cascade plot of the vibration spectrum at the HP-IP turbine. A slight subsynchronous vibration near 2430 cpm is visible. This frequency

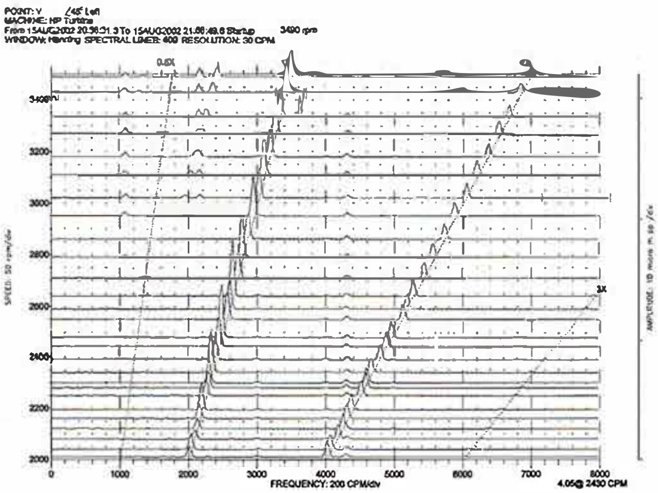

Figure 13. Cascade Spectrum Plot of HP-IP Turbine—Incipient Instability at No Load.

coincides with the first critical speed of the HP-IP rotor. An analytical study was conducted to determine the stability of the HP-IP rotor. Figure 14 is a plot of the calculated log decrements as a function of speed for the case of no aerodynamic cross-coupling

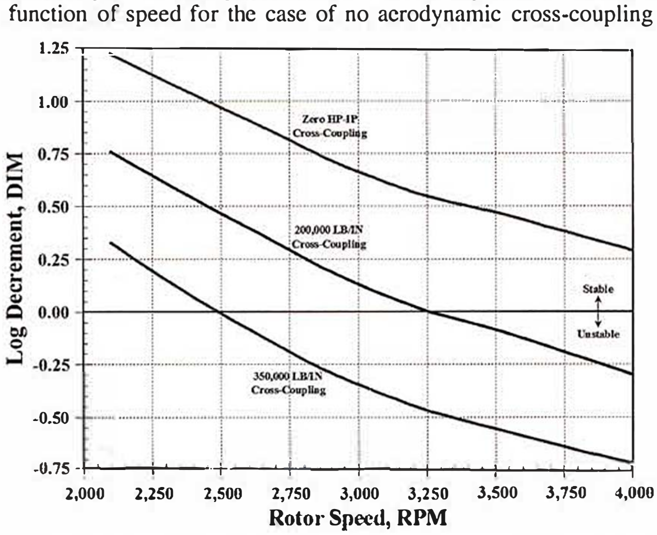

Figure 14. Predicted HP-IP Turbine Stability with Variable Cross-Coupling.

and two other cases. Since exact details of the long midspan steam seal were not known, it was estimated that this seal could produce between 200,000 lb/in and 350,000 lb/in cross-coupling. These results were sobering. There was now a real concern that, as load was applied with higher steam flow rates, the cross-coupling at blade and seal locations would be substantially increased. This would cause the HP-IP machine to suffer violent subsynchronous vibrations. Thus the owner decided that the #3 bearing needed to be upgraded to a TPJ as well. The bearing optimization study was then redirected to consider upgrades to tilting pad bearings at both HP-IP bearings and the #5 generator bearing.

BEARING DESIGN AND MANUFACTURE

The #3 elliptical bore bearing, as supplied by the original equipment manufacturer (OEM), is virtually identical to the #4 so the drawings and manufacture of the #3 soon caught up to the #4. At this time the #4 design was nearly complete, and all material was on order and some was onsite. The bearing optimization was completed with the #3 TPJ bearing design being identical to the #4. This helped to shorten the time required for both the design and manufacture.

In the mean time, the modified #5 bearing allowed the turbine to be run to trip speed and some commissioning steps were completed while the tilting pad bearing designs were finalized. However the next step in commissioning was the loading of the turbine and there was a real concern that, without the TPJs at the #3 and #4 locations, the HP-IP machine would go into oil whip. Time was critical as it took three to four days to cool down the turbines enough to take the train off turning gear and replace the bearings.

Bearing Number 5 Design Challenges

The #5 bearing as supplied by the OEM had a rather thin wall making a tilting pad journal bearing upgrade difficult. Designing a thin outer shell and pads thick enough to take the load without excessive deformation restricted the bearing design to incorporate a rocker back pivot mechanism instead of the preferred ball and socket pivot design used on the #3 and #4 bearing upgrades.

Also, since the #5 bearing has outside diameter (OD) pads on the sides and at bottom dead center and top dead center, this had to be considered in the upgrade design process. OD pads are used on this bearing design to make rotor alignment changes by shimming under the OD pads to move the bearing bore. Figure 15 is a displacement plot from a finite element analysis (FEA) of the bearing outer shell with a load-between-pad (LBP) tilting pad journal bearing configuration. Note the excessive displacement at the two bottom pad pivot locations, which are 45 degrees from bottom dead center. This displacement under a normal static pad load of 17,700 lb is almost .003 inches. Dividing this deflection into the load

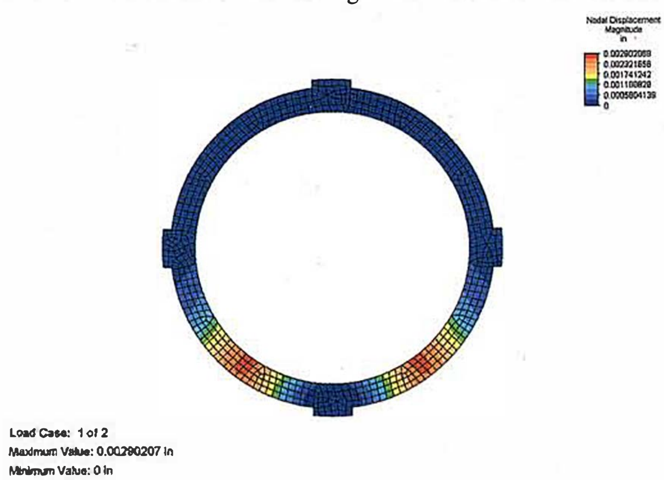

Figure 15. Displacement Result of FEA of #5 Outer Shell with LBP Configuration.

yields a stiffness of about 6 million lb/in, which is very low relative to the oil film stiffness. In order to take full advantage of available damping it became obvious that a load-on-pad configuration, with the bottom tilt pad located directly over the bottom OD pad, was required. Figure 16 is a plot of the results of the FEA analysis for

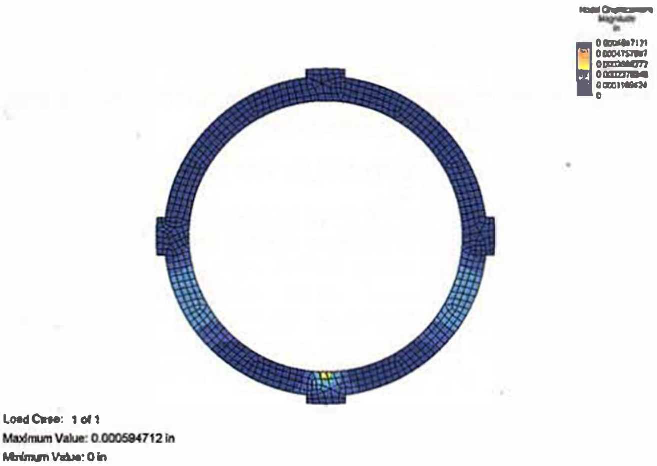

Figure 16. Displacement Result of FEA of #5 Outer Shell with LOP Configuration.

the load-on-pad configuration. For this case the bottom pad load is 29,600 lb and the adjacent pads are loaded to about 1750 lb each. The resulting stiffness is 50 million lb/in. Figure 17 is a photograph of the finished #5 TPJ.

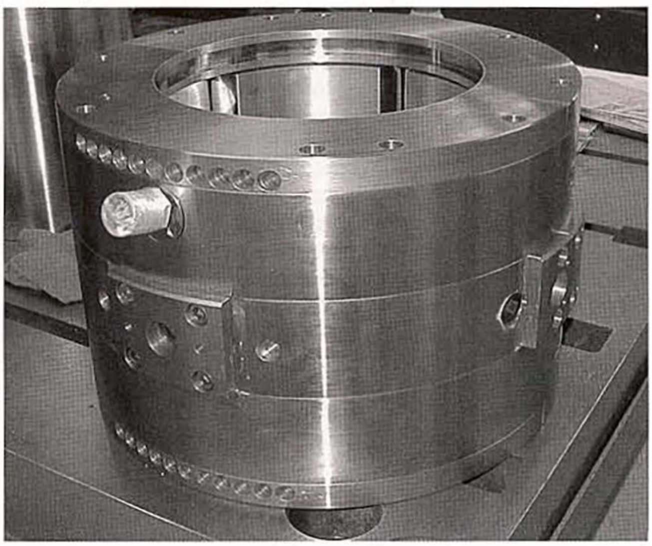

Figure 17. Photograph of #5 Tilting Pad Journal Bearing.

Concurrent Design and Manufacture of the TPJs

The need to get the unit through commissioning made the delivery of the new bearings the critical path. In order to deliver the completed bearings as quickly as possible they were being manufactured as the detail drawings were being completed. Manufacturing information was being sent to the shop as it was required. First, the raw material requirements were determined immediately after the decision to manufacture each bearing was made. This was followed by:

- Roughing information for the pads and outer shells.
- Split line bolt and dowel details for the outer shells.
- Bond line for the journal pads.

With this information the shop had plenty of work to do while the detail drawings were completed and checked. The machinists could rough in the OD, inside diameter (ID) and length of the three outer shells, split the shells and machine the outer shell split lines, then bolt and dowel the outer shell halves. They could also machine the pads for babbitting and proceed to babbitt the three pad rings. By the time this work was done the detailed drawings were ready for release so the bearings could be finished quickly.

Fourteen days after the decision was made to manufacture the #4 bearing (Figure 18) it was shipped. This was followed the next day

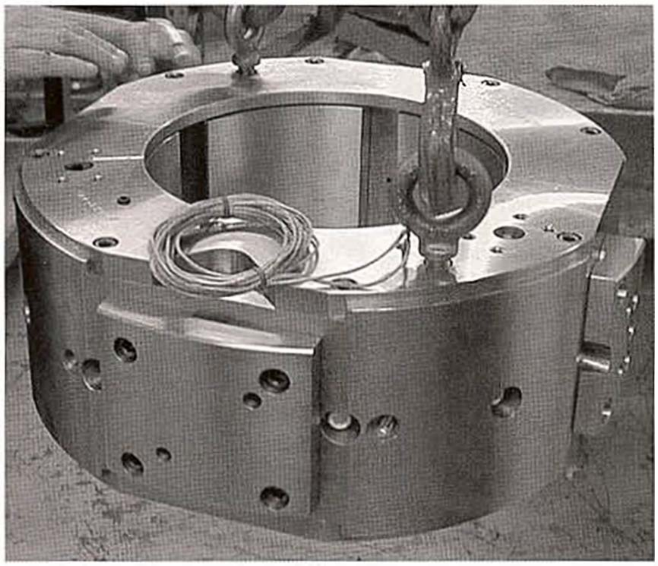

Figure 18. Photograph of #4 Ball and Socket Tilting Pad Journal Bearing.

by the #3 and the next day (day 16) by the #5 bearing. The bearings were installed and the unit was brought up to running speed with no subsynchronous vibrations observed. The unit was then fully loaded and no subsynchronous vibration problems were found on the generator or HP-IP rotor, or any other component of the train.

Figures 19 and 20 are a comparison of the shaft centerline plots for a no-load startup of the generator #5 bearing. The original plain

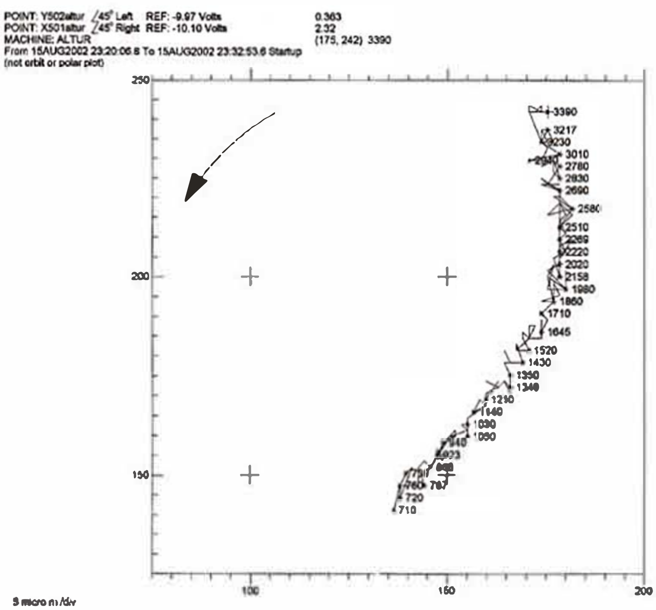

Figure 19. Shaft Centerline Position During Startup for Original Plain Generator Inboard Bearing.

bearing, Figure 19, shows a classic hook shape as the speed is increased. This is caused by the influence of large cross-coupling from the fluid film. In essence, with counterclockwise rotation, a vertical displacement induces a positive horizontal displacement. Thus the cross-coupling produces a force in the forward direction of rotation, which leads to the oil film instability. After the tilting pad bearings were installed, Figure 20, the shaft centerline is seen to move straight up. This confirms the absence of cross-coupling

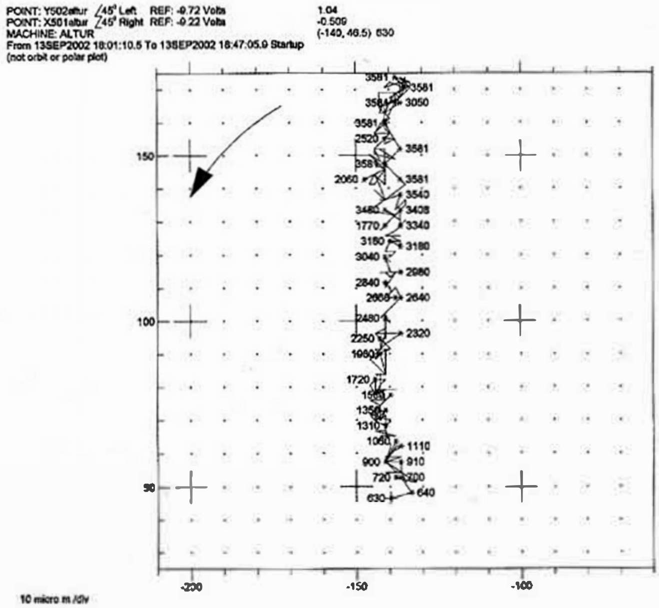

Figure 20. Shaft Centerline Position During Startup for Optimized Generator Inboard Tilting-Pad Bearing.

with this type of design and explains why oil whirl and whip are not possible. Figure 21 is the cascade plot for the final bearing configuration. There are no indications of subsynchronous vibration and none appeared when full load was applied.

The shaft centerline plot for the original #4 elliptical bearing in the IP section, Figure 22, also shows a classic hook shape as the speed is increased. This is caused by the cross-coupling from the fluid film. The shaft centerline moves straight up after the ball and socket tilting pad bearings were installed, Figure 23. This confirms the absence of bearing cross-coupling. Figure 24 is the cascade plot for this location with the final bearing configuration. There are

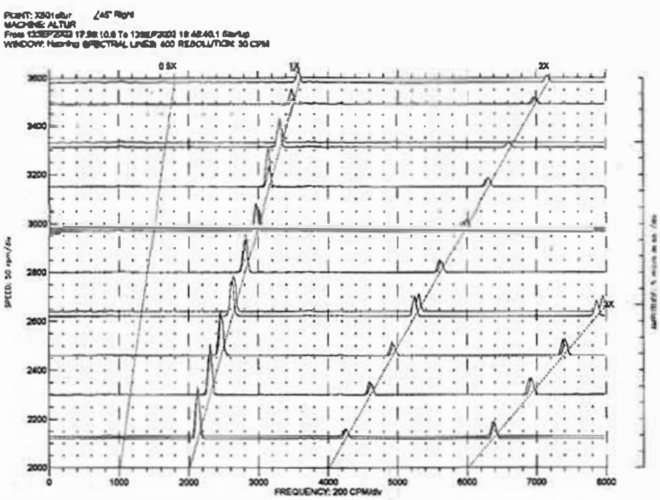

Figure 21. Cascade Spectrum Plot Showing Generator Vibration Problem Eliminated.

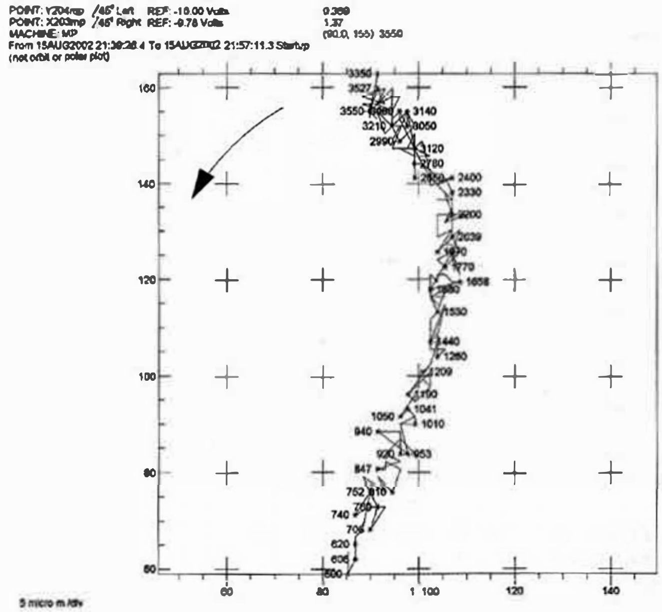

Figure 22. Shaft Centerline Position During Startup for Original Elliptical IP Bearing.

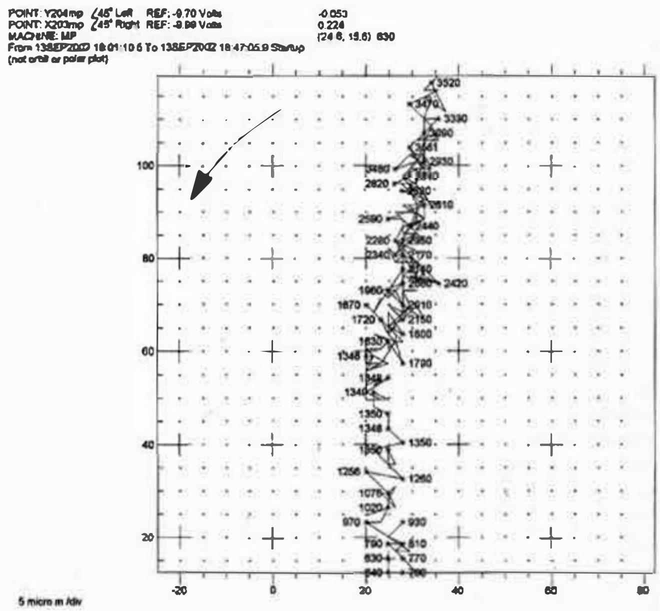

Figure 23. Shaft Centerline Position During Startup for Ball and Socket Tilting Pad IP Bearing.

no indications of subsynchronous vibration and none appeared when full load was applied.

CONCLUSION

This project went very smoothly when considering the complexity of the problems and the number of parties involved. The end user of the equipment was not even involved to any appreciable extent in this because they had not yet taken ownership of the equipment. The engineering and construction firm assumed full responsibility and saw to it that the turbine vendor, the generator manufacturer, the engineering consultant, and the bearing vendor all worked closely and quickly to engineer and implement the solution. Other than the problems with the generator information, and their bearing design, everything went smoothly.

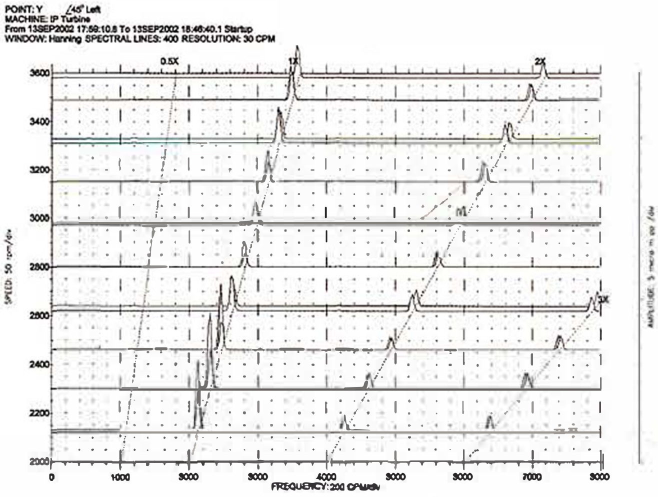

Figure 24. Cascade Spectrum Plot Showing HP-IP Vibration Problem Eliminated.

The unit has been running for several months now under various load conditions with no vibration problems.

ACKNOWLEDGEMENTS

The authors would like to thank those people at TCE who helped make the short deliveries of these three bearings possible. They include engineering, shop, and inspection personnel. Although we cannot name them, we would also like to acknowledge the professional work put in by the engineering firm, the end user of the equipment, and the turbine manufacturer.