TURBINE INSTABILITY SOLUTION–HONEYCOMB SEALS

by

Jay Armstrong

Senior Engineering Technologist

and

Frank Perricone

President

VibroDynamics Inc.

Mississauga, Ontario, Canada

Jay Armstrong is a Vibration Specialist with seven years experience in vibration analysis and troubleshooting rotating machinery, piping, and structural problems. He operates a Plant Predictive Maintenance Program monitoring the condition of approximately 1000 rotating machines with vibration and oil analysis.

Mr. Armstrong also spent five years involved in destructive and nondestructive testing in the aerospace industry.

Mr. Armstrong graduated as a Mechanical Engineering Technologist from Confederation College in Thunder Bay, Ontario. He is a member of the Canadian Machinery Vibration Assoc. and Vibration Institute.

Frank Perricone is President and a Machine Dynamics Specialist for Vibro Dynamics Inc. in Mississauga, Ontario. He has over 22 years of experience in troubleshooting and resolving rotating equipment and structural dynamics along with piping structural dynamics and pulsation problems.

Prior to VibroDynamics, Mr. Perricone was a Unit Head of the Machine and Structural Dynamics group for Ontario Hydro Research. In this capacity, he supervised professional and technical staff who provided services primarily to the Nuclear and Thermal power stations in the province of Ontario. The group's services included preparation of equipment dynamics specifications, design audits, commissioning of plant equipment in three new nuclear plants as well as developing and installing predictive maintenance online and walk-around vibration monitoring systems. He was also responsible for training engineering and maintenance staff at the power plants.

Mr. Perricone graduated from the University of Toronto (1973) with a B.A.Sc. degree and from the University of Waterloo (1979) with a M.A.Sc., both in Mechanical Engineering. He is a member of ASME, CMVA, and the STLE.

ABSTRACT

A 40,000 hp steam turbine-generator in a utility service experienced a history of high vibration alarms and trips from an online vibration monitoring system. The excursions of high vibration restricted the operational load performance of the turbine.

The cause of the vibration was determined to be a 'rotor instability,' identified as a subsynchronous vibration in the form of steam whirl of the rotor. This whirling excites a rotor critical speed resulting in a very unstable rotor system. The instability could lead to destructive levels of vibration.

A rotordynamics model of the rotor-bearing-support system was developed. Then, using the computer model, a stability analysis was performed and the sensitivity of the instability was evaluated for various system parameters (such as seals, rotors, and bearings).

Numerous possible solutions were investigated, of which two exhibited significant increases in the stability of the system. First, a newly designed rotor and second, replacing the rotor interstage labyrinth seals with honeycomb seals. The latter had a high probability of success and a greatly reduced capital cost.

The turbine underwent a major overhaul in August 1992 when the honeycomb seals were installed. The machine is steadily running at loads well above the expected project justification levels.

Aspects of the project, including before and after vibration analysis, rotordynamics analysis, and seal technology will be discussed.

INTRODUCTION

The occurrence of self-excited rotordynamics instability is of significant importance to users and manufacturers of modern high performance turbomachinery, particularly with the present trend towards higher speeds and loading conditions. Typically, the subsynchronous vibration occurs suddenly at some combination of speed, flow, or power level. The vibration can jump to very large amplitudes which can severely damage or destroy the machine and/or cause annoying and costly system trips.

Traditionally, tilt pad hydrodynamic bearings have provided greater stability margins and have been used extensively to stabilize machinery. In fact, it was sometimes felt that machines with this type of bearing could not experience instability problems. However, due to continuous pushes in machine design and performance requirements, it is not uncommon to find modern turbomachinery with stability problems even when they have tilt pad bearings. For example, the steam turbine in the case history presented herein has tilt pad bearings and suffered a rotordynamics instability problem. This stresses the need to fully understand the mechanisms of instability and to recognize that machine components other than bearings can also contribute to destabilizing forces.

The field of rotordynamics instability is highly complex and research in this field, such as the work of Dara Childs of the

Turbomachinery Laboratory at Texas A&M University, is still progressing at a fast rate. In particular, his pioneering research in the theoretical analysis and accompanying experimental investigations has had a significant influence on advancing the use of honeycomb seals in stabilizing machinery [1, 2]. The NASA space shuttle program has been a major beneficiary of this research in stabilizing the engine's turbo pumps. There are a few occasions of use in stabilizing high performance turbomachinery such as the cases by Zeidan, et al [3], but for the most part, the use of honeycomb seals has remained in the aerospace industry.

The purpose herein is not to present an indepth discussion on rotordynamics instability, but to outline basic engineering concepts that plant engineers and maintenance personnel can use to quickly identify this phenomena in their plants and to better understand the destabilizing mechanisms and the possible solutions.

The authors present a rotordynamics instability case history involving the use of honeycomb seals to stabilize a 30 MW power plant steam turbine at a world scale petrochemical plant.

ROTORDYNAMICS INSTABILITY

In rotating equipment, there are a number of mechanisms that can cause nonsynchronous whirling in a rotor system, but the most serious one is the occurrence of self excited whirling as caused by hydrodynamic bearing instability, aerodynamic cross coupling, internal friction, etc. Instability implies a motion that can increase without limit. In fact, the vibration can be so severe as to cause destruction for the rotor system. Thus, it is critical for plant and maintenance engineers to be able to recognize the occurrence of this potentially dangerous situation so as to change the rotor operating conditions or speed. It is most desirable that turbomachinery be designed to minimize the possible occurrence of a rotordynamics instability.

In some cases, the nonsynchronous motion can occur with bounded nondestructive amplitudes. Since a small change in the system parameters can destabilize the system and produce a quick growth in the rotor motion, it is imperative that these cases be carefully monitored.

Typically, the subsynchronous vibration motion appears at some speed or power level. The vibration levels grow rapidly until the equipment fails or the rotor motion is stabilized at some amplitude due to nonlinearities in the system.

BASIC CONCEPTS OF ROTORDYNAMICS INSTABILITY

The mathematical definition of a rotordynamics instability is as follows:

a solution to linear differential equations of motion characterized by a complex eigenvalue positive real part assume the solution

$$x(t) = Ce^{st} \quad (1)$$

where:

$S = \lambda \pm i\,\omega$

is the eigenvalue

the displacement time dependent amplitude of vibration is given by:

$$x(t) = e^{\lambda t}\,(C_1 \cos \omega t + i\, C_2 \sin \omega t \quad (2)$$

λ = is the stability predictor

ω = is the whirl frequency

Computer rotordynamics programs for rotordynamics stability analysis calculate the eigenvalues. The real part of an eigenvalue gives the growth (or decay, if negative) factor of the solution; the imaginary part gives the frequency. The solution is a function which describes the time dependent amplitude of vibration. Unstable motion is shown in Figure 1.

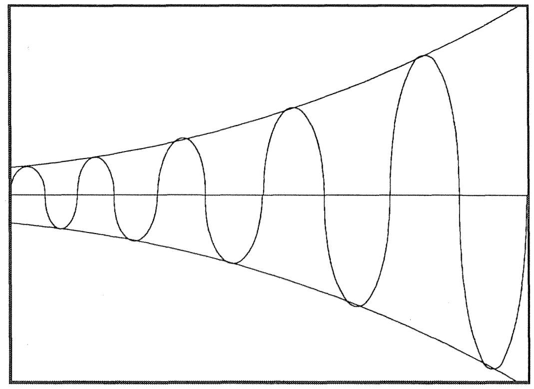

Figure 1. Unstable Motion.

In linear theory, unstable motion is unbounded with time (i.e., it keeps on getting larger and larger with no limit). In reality, the rotor will start rubbing against seals and blades will start rubbing against stators that will cause nonlinear vibrations and bounded vibrations. This is often referred to as limit cycle whirl (Figure 2).

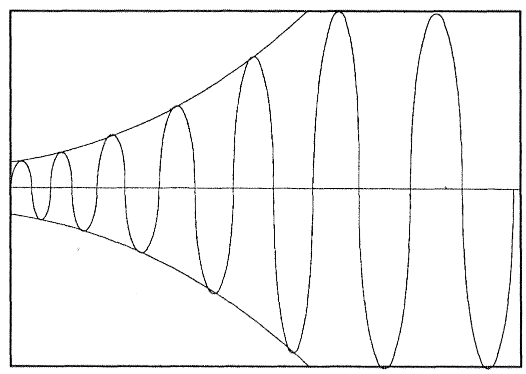

Figure 2. Limit Cycle Whirl.

Important Facts on Rotordynamics Instability

- Unstable motion is not caused by unbalance, and is normally subsynchronous
- Not all subsynchronous motion represents unstable motion, such as:
  - Oil whirl (predecessor to oil whip)
  - Subharmonic resonance
  - Excitation sources fed through from other machines
- Unstable motion is often caused by variation of a fluid dynamic pressure around the circumference of a rotor component
- Unstable motion is normally the free or self-excited response of a vibration mode that has negative damping

DESTABILIZING FORCES– "FORWARD WHIRL DRIVING FORCES"

Destabilizing Forces in a Fixed Geometry Bearing as an Example

The simplest machine components to help understand why rotating equipment experience instability is the fixed geometry journal bearing [3].

In a journal bearing, rotor weight is supported by developing a hydrodynamic pressure in a converging wedge (Figure 3). The unsymmetric oil film pressure profile produces an attitude angle between the line of center and the load vector which results in a cross coupling force in the bearing. This cross coupling stiffness force is the source that produces the self excited vibrations and instabilities in rotating equipment. A graphic representation of this force is described by Vance [4] and is shown in Figure 4.

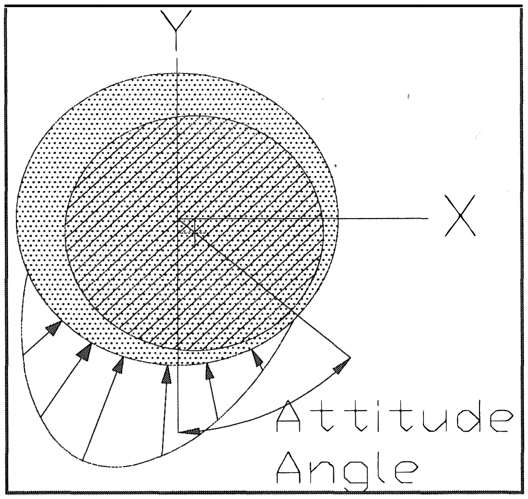

Figure 3. Pressure Profile in a Journal Bearing.

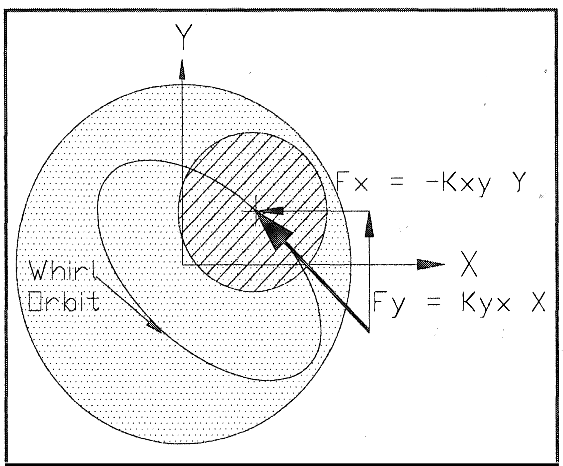

Figure 4. Schematic Representation of a Cross Coupled Force.

The driving force on the rotor is represented as the resultant force components due to (Kxy) and (Kyx) stiffness terms. A positive (Kxy) and a negative (Kyx) will result in a net force that is tangential to the whirl orbit and in the direction of rotation. This force acts against the damping forces in bearing and rotor systems. A schematic representation is shown in Figure 5 of the destabilizing cross coupled force opposing the stabilizing damping force. It is the balance of these two forces that determines whether a rotating machine is stable.

When a rotating machine has a stability problem, the task of the rotordynamics engineer is to find the methods of increasing the damping in the system, reducing the cross coupling forces or accomplishing both.

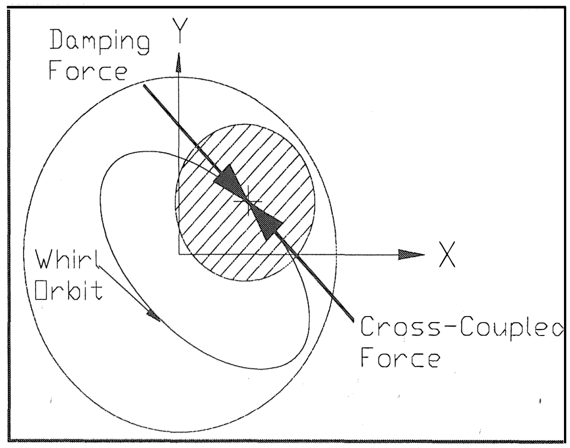

Figure 5. Cross Coupled Stiffness and Damping Forces.

The Main Frequency of Vibration of an Unstable System

At some rotor speed above the limiting value, when the destabilizing force exceeds the stabilizing external damping, any small system disturbance will cause the shaft to whirl at its critical speed, independent of the rotational speed. Since the cross coupled stiffness produces a tangential force proportional to shaft deflection, the force grows larger as the amplitude of whirl grows. This is the reason why instability is called "self excited" motion [5].

Unstable rotor systems have characteristically large subsynchronous vibrations. Typically, the whirl frequency ratio f/fs is 0.3 to 0.8. In general, rotor systems are most sensitive to instabilities when the rotor speed is greater than two times the first critical speed.

System Symmetry Effects on Rotordynamics Instability

A rotor system that has symmetric bearings (i.e., circular) is more sensitive to instability than one that has a significant amount of asymmetry. For example, fixed geometry journal bearings achieve a higher stability threshold by incorporating modifications to reduce fluid rotation and alter the circular geometry, such as the addition of grooves, altering the circular bearing geometry by using preloaded lobes, lemon shaped, or elliptical geometries. The asymmetry in the bearing generally tends to enhance stability.

Asymmetry combines with the cross coupled stiffness coefficients to influence the total energy added to the dynamic system [5]. Energy added is destabilizing, while the energy dissipated due to direct damping is stabilizing. The net effect on stability can be physically explained by the schematic shown in Figure 6 and the following equation.

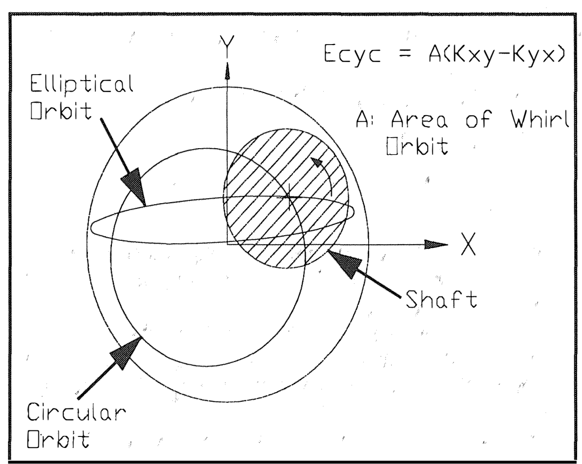

Figure 6. Schematic of the Influence of Asymmetry on Stability.

$$Ecyc = (Kxy - Kyx) \quad (3)$$

The energy added to the system is calculated by integrating the force due to the bearing cross coupled coefficients over the displacement around the closed curve of the whirl orbit. Therefore, the energy added (destabilizing) to the dynamics system is equal to the product of the whirl orbit area (A) times the net value of (Kxy – Kyx). In conclusion, the more asymmetric the whirl orbit, the smaller the orbit area and less destabilizing energy is added to the dynamic system.

In summary:

- Rotors are destabilized by cross coupled stiffness coefficients which produce the tangential reaction forces
- Stability is enhanced by increasing external damping (it can be shown that too much damping can be destabilizing; however, for most practical cases damping improves stability)
- Rotors are more sensitive to instability as the ratio of running speed to critical increases
- Rotor-bearing system asymmetry helps to reduce instability

KNOWN INSTABILITY DRIVERS IN TURBOMACHINERY

Some of the mechanisms and machine components that can induce instabilities in turbomachinery are as follows [3, 5]:

- Hydrodynamic bearings
- Locked floating oil seals
- Annular seals
- Labyrinth seals
- Trapped fluids
- Internal friction
- Alford's force
- Impeller shroud forces
- Torque whirl
- Rubbing (backward modes)

GENERAL APPROACHES TO CORRECT AN INSTABILITY PROBLEM

- Introduce stiffness asymmetry
- Increase damping in the supports
- Increase stiffness to increase the natural frequency of the unstable mode
- Increase support stiffness to improve hydrodynamic bearing damping
- Minimize or defeat the destabilizing mechanism

A ROTORDYNAMICS INSTABILITY PROBLEM CASE HISTORY

Background and History

The 3600 rpm 30 MW turbine generator was installed in 1982. During startup of the machine there were many high vibration alarm occurrences and several trips.

Displacement spectra taken on several occasions showed that the major component of vibration was at 1630 cpm (0.458 times the operating speed). This component was much larger than the 1.0 × rpm component.

After startup, the load was restricted to less than 20 MW, since operation above this load setting would definitely cause vibration alarms to occur.

In 1989, the machine had its first major overhaul. When the turbine was opened up, the labyrinth seals were found to be in bad shape. The seals had undergone severe rubbing, were bent, completely broken, or missing.

New labyrinth seals were installed. After the overhaul, many high vibrations occurred. The log books show that up to 12 to 15 alarms occurred per shift. On several occasions system trips occurred directly related to high vibration. Trip settings were at 150 μm p-p rotor displacement.

After a month of rough operation, the machine vibration levels settled out. The operations personnel found a comfort zone around 19 MW and operated the machine around this power level. In addition, a significant effort was made not to fluctuate from this load condition and to control any disturbances from the boiler.

Experimental Stability Investigation

The vendor of the turbine generator performed experimental tests on the machine in March 1990. The main tests results showed that the machine was unstable at 15.5 MW and that the instability vibration component was at 28 Hz (1680 cpm–at 0.47 times the operating speed of 3600 rpm). In addition, the tests results showed that instability rotor vibration component measured at the turbine inboard and outboard bearings were significantly higher than those measured at the generator bearings.

The vendor confirmed that the cause of the high vibration levels was a rotor instability.

Theoretical and Experimental Stability Investigation

In 1992, a rotordynamics stability analysis was requested. The main objectives were to:

- Develop an accurate rotordynamics model of the rotor-bearing-support system
- Perform a stability analysis (i.e., damped eigenvalue analysis) using the computer model
- Evaluate sensitivity of instability to various system parameters
- Provide recommended actions to significantly increase the stability of the system

Recommendations were made to change the labyrinth seals to honeycomb seals. The design changes were implemented and the installation was completed in August and September 1992. When the turbine was opened during this overhaul period, the condition of the labyrinth seals (Figure 7) indicated that the rotor had undergone excessive displacement and contacted the seals.

The startup of the turbine generator system was in September 1992. The machine could only be tested up to 23.5 MW during the startup because of limitations on the maximum boiler output capacity; however, the system has operated up to 29 MW after the startup. Since startup, the system has been normally operating near 25 MW. Since startup, during the initial testing and under normal operating conditions, there has been no signs of instability.

The following provides an overview of the theoretical rotordynamics analysis and the main test results after the design modifications were implemented.

THE THEORETICAL ROTORDYNAMICS ANALYSIS MODEL

Rotor

The rotor computer model is shown in Figure 8. The rotor model consists of 161 mass stations and 22 bearings. Four bearings were used to model the turbine and the generator bearings and 18 bearings were used to model the destabilizing forces at the turbine wheel tips and at the turbine wheel and the labyrinth seals. Information for the model was obtained from detailed drawings provided by the steam turbine manufacturer.

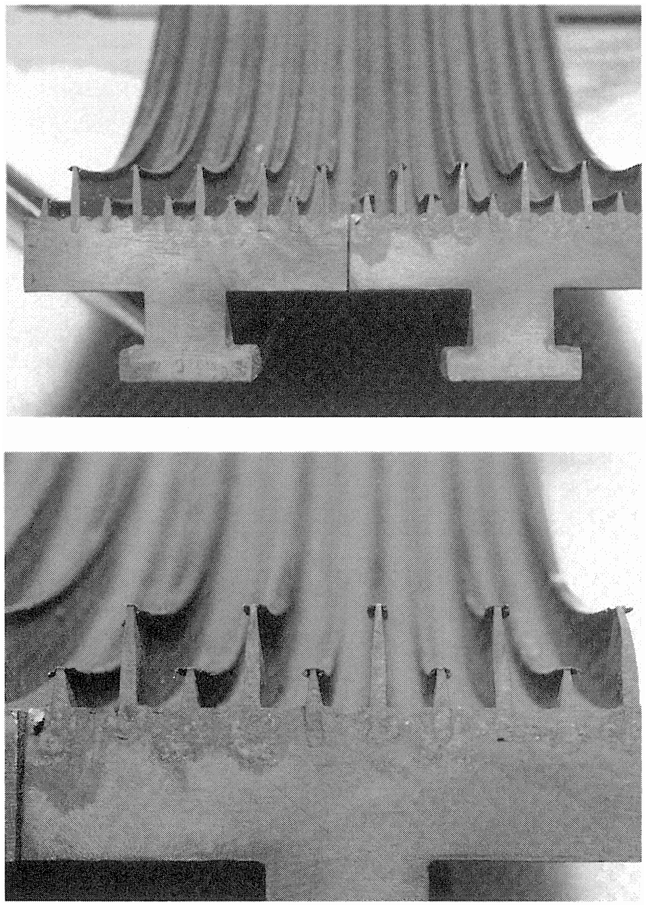

Figure 7. Damaged Turbine Labyrinth Seals Found During the August-September 1992 Overhaul.

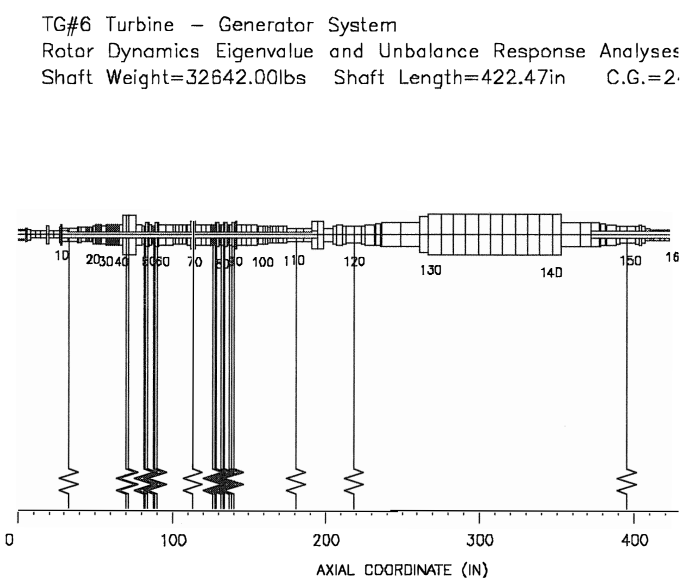

Figure 8. Turbine Generator Computer Rotordynamics Model.

Equivalent Bearing Stiffness and Damping Dynamic Coefficients

A significant effort was made in calculating the equivalent fluid film bearing stiffness and damping coefficients. The analysis included the effects of the bearing housing and pivot pad stiffnesses on the bearing oil fluid film dynamic coefficients. Experimental dynamics testing was performed on the four bearing housings in both the vertical and horizontal directions to determine their dynamic compliances.

Destabilizing Alford Forces, Tip and Interstage Labyrinth Seal Dynamic Coefficients

The destabilizing forces caused by the variation of blade-tip clearance around an unshrouded axial flow machine due to any deflection of the shaft away from the center of the housing was estimated using Alford's formula as follows:

$$Kxy = -Kyx = \beta T / DH \tag{4}$$

where:

β = the efficiency factor
T = stage torque
D = vane pitch diameter
H = vane height

The destabilizing seal dynamic coefficients were determined using the steam pressures and temperatures at each of the interstage labyrinth seals which were provided by the vendor of the turbine generator.

Both the Alford and seal destabilizing dynamic coefficients were calculated for a 20 MW operating condition.

MAIN RESULTS OF THE THEORETICAL STABILITY ROTORDYNAMICS ANALYSIS

The damped eigenvalue analysis predicted a turbine mode at 1646.5 cpm (27.44 Hz) with a logarithmic decrement value of 0.0672 (Figure 9). A negative logarithmic value implies a fully destabilized mode of vibration. Thus, the analysis predicted a marginally stable mode close to the measured instability frequency of 1680 cpm (28 Hz).

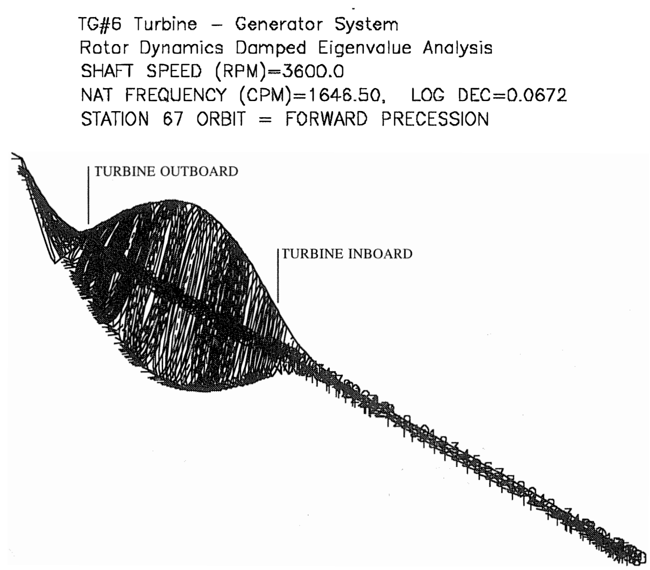

Figure 9. Existing System Rotordynamics Results.

The 1646.5 cpm mode had a forward precession and it was the third mode predicted by the analysis. A backward precession mode was predicted at 1298.9 cpm and a forward precession mode was predicted at 1551.4 cpm.

The unstable mode was characterized by (Figure 9):

- Large rotor amplitudes at the turbine and very low rotor amplitudes at the generator
- Rotor vibrations which were larger in the vertical direction than in the horizontal direction
- Significantly larger rotor vibration amplitudes at the turbine center than at the turbine bearings (i.e., a flexible mode of vibration)

The first two vibration characteristics of the theoretical analysis were reported by the vendor from his field testing of the machine.

The 1646.5 cpm mode of vibration was highly susceptible to producing an instability condition for the following reasons:

- The mode has a whirl frequency below one-half the system operating speed of 3600 rpm (i.e., a prerequisite for instability)
- The mode has very low damping as indicated by the low logarithmic decrement
- The mode has large rotor displacements near the locations where large steam destabilizing forces are produced

Many turbine generator parameters were investigated to evaluate their sensitivity to significantly increasing the system stability (i.e., increasing the logarithmic decrement of the mode). The following outlines the main cases of the rotordynamics investigation:

CASE 1–The Effect of Having No Destabilizing Forces in the Turbine

For this case, all the destabilizing stiffness coefficients that were entered in the rotordynamics computer model were set to zero. If this parameter change showed a significant beneficial increase in the logarithmic decrement, then the implementation of swirl brakes in front of the important seals could have been a possible solution to consider. Since the cost of implementing the swirl brakes would be very high, and they cannot remove all the destabilizing forces, the necessary increase in the logarithmic decrement would have to be large, such as 0.5 to 0.75, for this solution to be even considered.

The results show that the logarithmic decrement increased marginally from 0.0672 to 0.1761 (Figure 10). The mode of vibration increased slightly from 1646.5 cpm (27.44 Hz) to 1666.46 cpm (27.77 Hz). Thus, stabilizing the turbine with swirl brakes was not pursued.

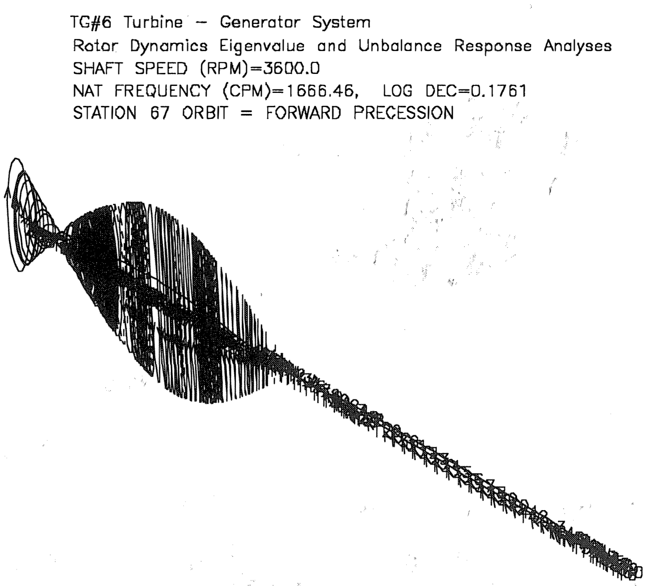

Figure 10. Case 1–The Effect of Having No Destabilizing Forces in the Turbine.

CASE 2–The Effect of Removing the Large Cross Coupled Stiffnesses in the Generator Bearings

In this case, the large cross coupled stiffness coefficients in the generator were set to zero in the computer rotordynamics model. This would simulate the beneficial effect of replacing the existing generator journal bearings with tilt pad bearings which have very small destabilizing cross coupled stiffness coefficients.

Typically when a system has a rotordynamics instability, replacement of fixed geometry bearings with tilt pad bearings is often considered. However, for the turbine generator under analysis, no significant increase in the logarithmic decrement of the unstable mode was expected with this parameter change since the generator rotor vibration orbits were very small in relationship to the turbine rotor vibration orbits.

The results showed that the logarithmic decrement of the unstable mode increased marginally from 0.0672 to 0.0894 when the generator fluid film cross coupled bearing stiffness coefficients were set to zero (Figure 11). Thus, replacement of the generator journal bearings with tilt pad bearings was not considered.

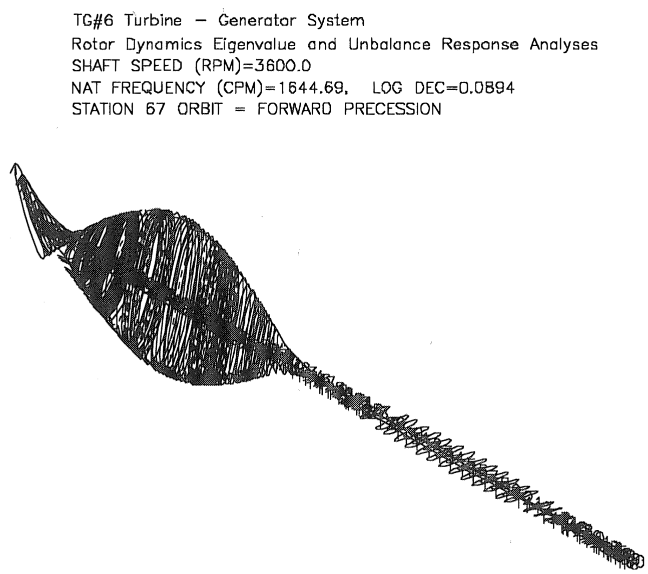

Figure 11. Case 2–The Effect of Removing the Large Cross Coupled Stiffnesses in the Generator Bearings.

CASE 3–The Effect of Rotating the Turbine Tilt Pad Bearings to a Load Between-Pad Configuration

The existing turbine tilt pad bearings have a load-on-pad configuration. In this case, the bearings are assumed to be rotated 36 degrees to have a load-between-pad configuration. The results showed that the logarithmic decrement of the unstable mode decreased significantly to -0.1413, indicating a fully destabilized mode (Figure 12).

This case clearly shows the destabilizing effect of bearing fluid film stiffness symmetry that is typically produced by a load-between-pad configuration. For the existing turbine generator, the turbine orbit is highly elliptical (Figure 9) because of the asymmetric bearing fluid film stiffness characteristics produced by the load-on-pad configuration (existing case logarithmic decrement was calculated to be 0.067). The parametric change studied in Case 3 resulted in a highly circular orbit by the load-between-pad configuration due to the symmetric bearing fluid film stiffness characteristics. The net result was a significantly smaller log decrement.

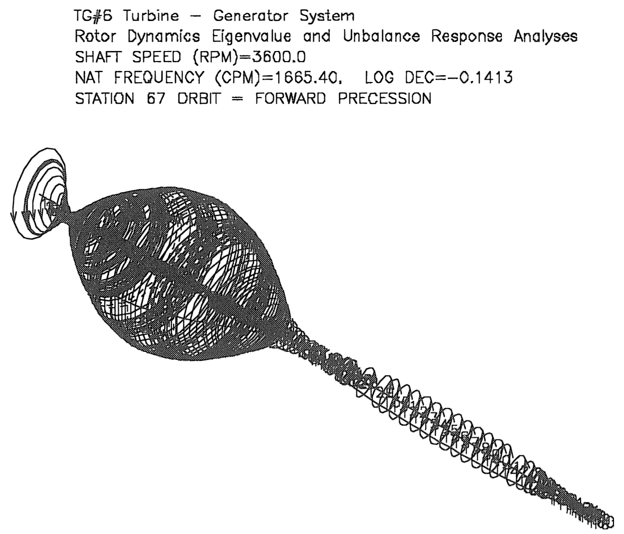

Figure 12. Case 3–The Effect of Rotating the Turbine Tilt Pad Bearing to a Load-Between-Pad Configuration.

CASE 4–The Effect of Replacing the Existing Five Pad Load-On-Pad Turbine Bearings with Four Pad Load-On-Pad Bearings

Four pad tilt pad bearings typically have better damping than five pad tilt pad bearings and, therefore, this bearing type was investigated in Case 4. The results showed a marginal increase in the logarithmic decrement in the unstable mode (Figure 13). The logarithmic decrement increased marginally from 0.0672 to 0.1063 with this parametric change.

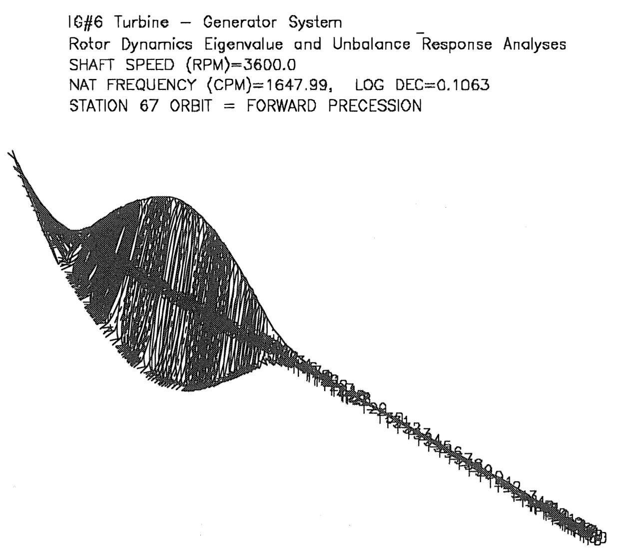

Figure 13. Case 4–The Effect of Replacing the Existing Five Pad Load-On-Pad Turbine Bearings with Four Pad Load-On-Pad Bearings.

CASE 5–The Effect of Stiffening the Turbine Bearing Housing in the Vertical Direction by Detuning the Structural Natural Frequency Near 28 Hz

During the structural dynamic testing performed on the turbine bearing housings, a structural natural frequency on the turbine outboard bearing in the vertical direction was measured near 28 Hz. In this case, the effect of doubling the bearing housing stiffness was investigated, This was considered to be a conservative estimate of the bearing housing structural stiffness increase, which could be produced by detuning the resonance.

The results showed that stiffening the turbine outboard bearing would only produce a marginal increase in the logarithmic decrement of the unstable mode (Figure 14). The analysis predicted that the logarithmic decrement would only increase from 0.0672 to 0.0724.

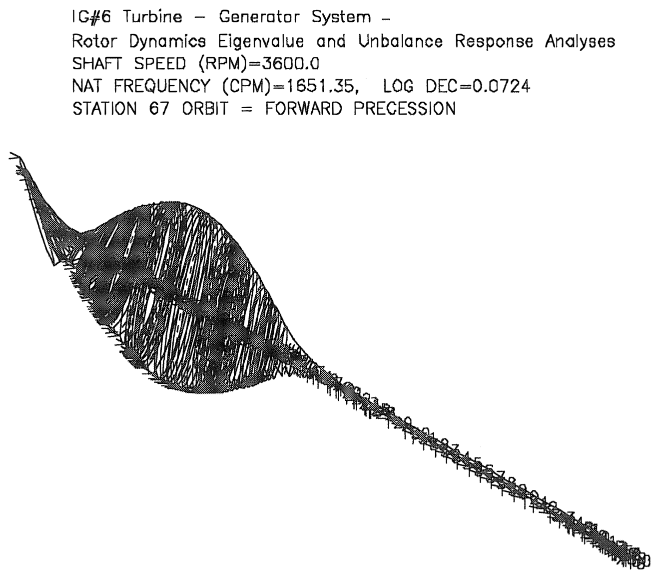

Figure 14. Case 5–The Effect of Stiffening the Turbine Bearing Housing in the Vertical Direction by Detuning the Structural Natural Frequency.

CASE 6–The Effect of Stiffening the Turbine Outboard Bearing in the Vertical Direction and Stiffening the Turbine Tilt Pad Bearing Pivots

In this case, the turbine outboard bearing housing and tilt pad pivot stiffnesses were doubled. A design change in the support system of the tilt pad bearing to a ball and socket or a flexible pivot bearing could increase the effective fluid film stiffnesses and damping.

The results showed that these design modifications would only produce a marginal increase in the logarithmic decrement of the unstable mode (Figure 15). The predicted increase would be from 0.0672 to 0.0972 with this design modification.

CASE 7–The Effect of Replacing the Turbine Labyrinth Interstage Seals with Honeycomb Seals

In this case, a small amount of damping (less than four percent of the maximum damping in the turbine tilt pad bearings) was added to the interstage seals in the rotordynamics computer model to simulate the possible damping which could be provided by honeycomb seals.

The results of the analysis showed that this small added amount of damping had a significant effect in increasing the logarithmic decrement of the unstable mode. The logarithmic decrement increased from 0.0672 for the existing case, to 0.5133 for the honeycomb seal investigation (Figure 16).

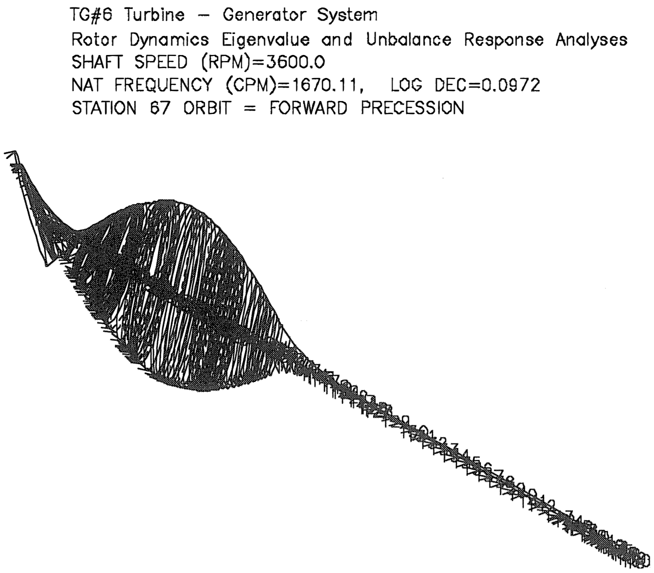

Figure 15. Case 6–The Effect of Stiffening the Turbine Outboard Bearing in the Vertical Direction and Stiffening the Turbine Tilt Pad Bearing Pivots.

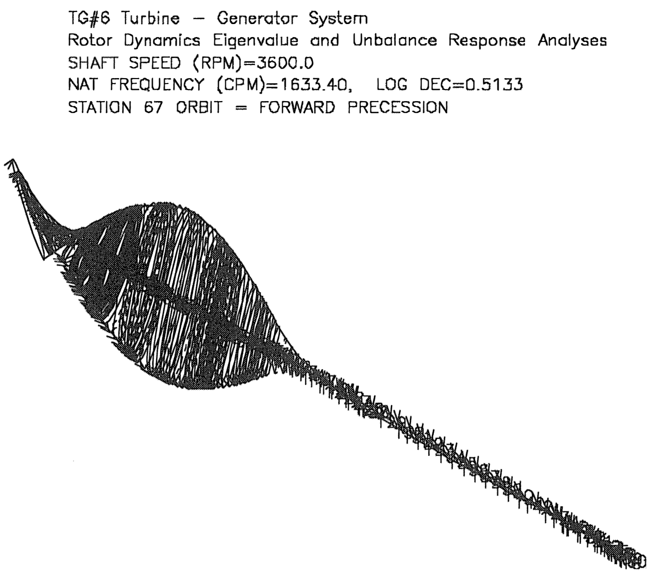

Figure 16. Case 7–The Effect of Replacing the Turbine Labyrinth Interstage Seals with Honeycomb Seals.

Based on the extensive analytical rotordynamics analysis and assessment of the various options for corrections, it was recommended that the turbine labyrinth seals be replaced with honeycomb seals. This recommendation was accepted as the most practical solution.

PREDICTED ROTORDYNAMICS RESULTS WITH THE RECOMMENDED HONEYCOMB SEALS

Detailed analyses were performed on the recommended honeycomb seals to determine their seal dynamic coefficients. The rotordynamics stability analysis with the recommended honeycomb seals showed that the third mode (1688 cpm–the unstable mode) for 20 and 30 MW generator loadings would be stable. The predicted logarithmic decrements for the 20 and 30 MW loadings were 0.2186 and 0.2280, respectively (Figures 17 and 18). The logarithmic decrements could have been significantly higher, but shaft undercuts near the seals greatly reduced the maximum damping capacity that the honeycomb seals could achieve. However, it was felt that the analyses were very conservative.

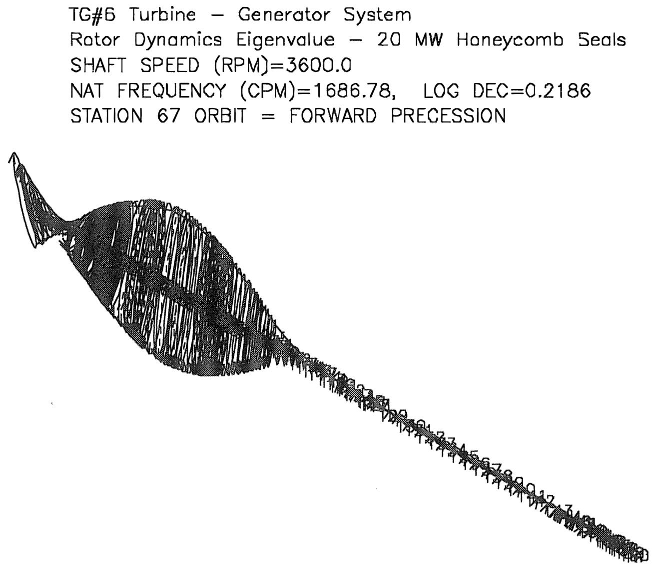

Figure 17. Predicted Rotordynamics Results for the Steam Turbine with Honeycomb Seals for 20 MW Loading.

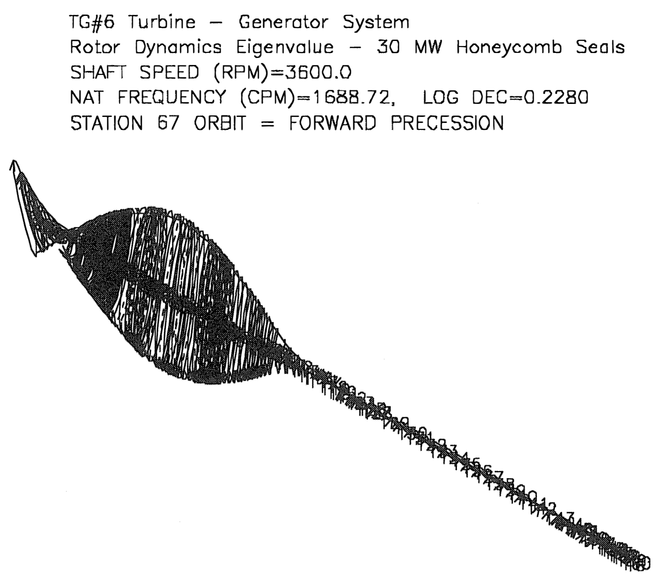

Figure 18. Predicted Rotordynamics Results for the Steam Turbine with Honeycomb Seals for 30 M W Loading.

GENERAL SUMMARY OF THE MAIN TEST RESULTS

The honeycomb seals were designed, manufactured, and then installed during a planned turbine generator overhaul. During startup of the turbine generator in September 1992, vibration measurements and analysis conclusively demonstrated the effectiveness of the honeycomb seals in eliminating the rotor system instability along with damping the rotor response, as predicted by the theoretical analysis. Since September 1992, there has been no evidence of any rotor instability (Figure 19).

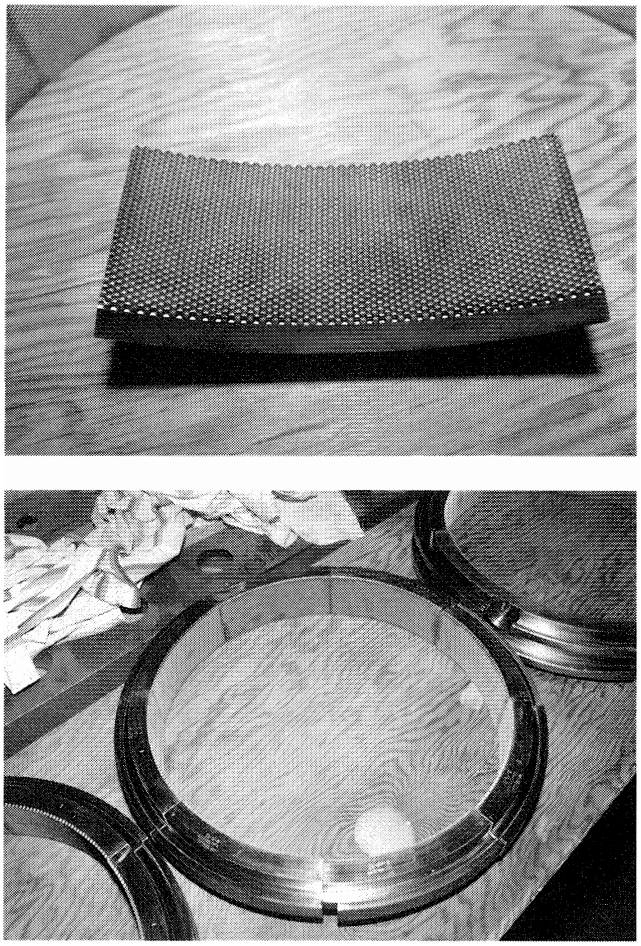

Figure 19. The New Honeycomb Seals.

During startup (September 11-13, 1992), the turbine generator was tested from 1.0 to 23.5 MW. Further testing up to the maximum generator's load of 30 MW was not possible during the startup, since the boiler capacity was limited to 23.5 MW. However, the machine did operate up to 29 MW approximately one month later.

The main result of the testing is shown in Figure 20. The plot clearly shows that the overall vibration level was predominantly due to the 1.0 × rpm component during the tests and not to any instability component. Typical rotor displacement spectra measured at the turbine thrust end (i.e., outboard end) and at the coupling end in the vertical direction are shown in Figures 21 and 22.

GENERAL CONCLUSIONS

- It is important that plant personnel be able to recognize rotordynamics instability and its potentially dangerous consequences and take actions to control it or eliminate it with machine design changes.

- There are numerous machine components, other than bearings, which can produce the forward whirl driving destabilizing forces responsible for rotordynamics instability.

- It is the balance of the destabilizing and damping forces which determines whether a rotating machine will be stable.

- A rotordynamics stability analysis was successful in predicting the instability characteristics of an unstable 30 MW steam turbine.

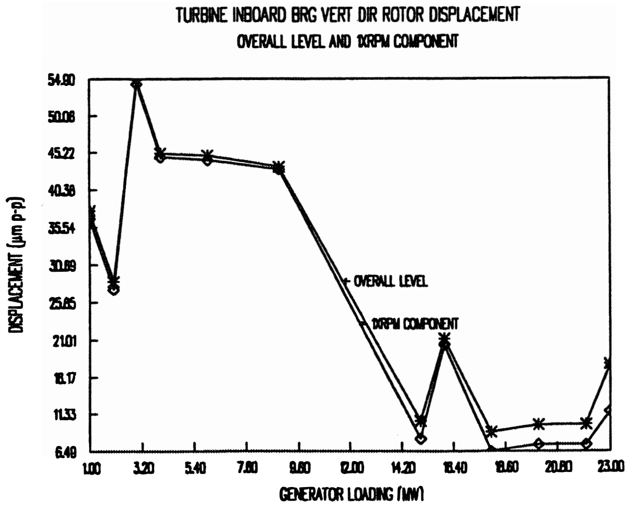

Figure 20. Turbine Inboard Bearing Vertical Direction Rotor Displacement Overall Level and 1.0 × Component Vs *Generator Loading.*

	PWR SPECTRUM	ChA	
1	21.50Hz	.133E+1	EU
2	26.50	.221E+1	
3	28.00	.114E+1	
4	60.00	.120E+2	
5	120.00	.340E+1	
6	179.50	.320E+0	
7	overall	.137E+2	
8			
9			
10			

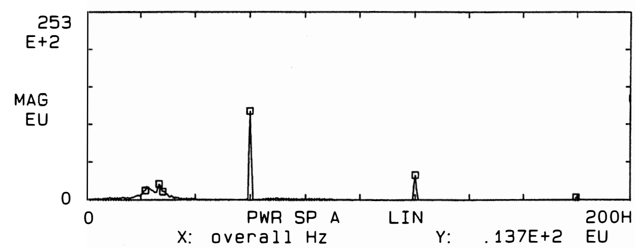

Figure 21. Turbine Outboard End Bearing Vertical Direction Rotor Displacement Spectrum 23 MW Generator Loading.

	PWR SPECTRUM	ChA	
1	26.50Hz	.472E+1	EU
2	28.00	.251E+1	
3	60.00	.117E+2	
4	120.00	.219E+1	
5	180.00	.894E+0	
6	overall	.179E+2	
7			
8			
9			
10			

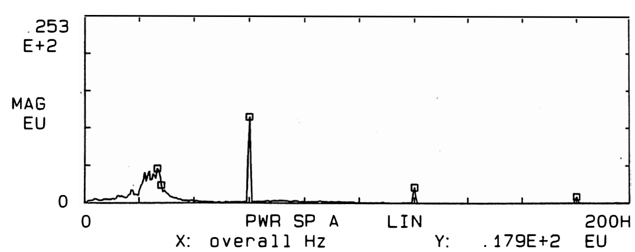

Figure 22. Turbine Inboard End Bearing Vertical Direction Rotor Displacement Spectrum 23 MW Generator Loading.

• Replacement of the steam turbine labyrinth seals with honeycomb seals was the only parameter change investigated in the rotordynamics analysis which resulted in a significant and sufficient increase in the logarithmic decrement of the instability mode.

• The honeycomb seals, as replacements for the labyrinth seals, were effective in stabilizing the turbine rotor over the load range tested up to 23.5 MW during the startup of the machine on September 11-13, 1992. Subsequently, the turbine generator operated with a load up to 29 MW without any evidence of instability.

REFERENCES

1. Childs, D., Elrod, D., and Ramsey, C., "Annular Honeycomb Seals; Test Results for Leakage and Rotordynamics Co-Efficients," Journal of Tribology, pp. 111, 293-301 (1989).
2. Childs, D., Elrod, D., and Ramsey, C., "Annular Honeycomb Seals; Additional Test Results for Leakage and Rotordynamics Co-Efficients," in IFToMM, Proceedings of the Third International Conference on Rotordynamics, Lyon, France, pp. 303-312 (1990).
3. Zeidan, F. Y., Perez, R. X., and Stephenson, E. M., "The Use of Honeycomb Seals In Stabilizing Two Centrifugal Compressors," *Proceedings of the Twenty-Second Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station , Texas (1993).
4. Vance, J. M., *Rotordynamics of Turbomachinery,* New York, New York: John Wiley & Sons (1988).
5. Tripp, H. A. and Murphy, B. T., "Eccentricity Measurements on a Tilting Pad Bearing," ASLE Trans., *28* (2), pp. 217-224 (1985).
6. Enrich, F. and Childs, D. W., "Self-Excited Vibrations In High-Performance Turbomachinery," ASME Mechanical Engineering, pp. 66-79 (May 1984).

ACKNOWLEDGMENTS

The commitment to the project and the willingness to be the first to install honeycomb seals in a steam turbine is commendable. From the results of the literature search and discussions with Dara W. Childs, honeycomb seals, to the best of our knowledge, have never been applied to a steam turbine.

The advice and assistance provided by Brian Murphy of the Center for Electromechanics of the University of Texas in Austin, Texas and Fouad Zeidan, Director of Engineering, KMC Inc., Houston, Texas are very much appreciated.