

DYNAMIC ANALYSIS OF AN 8000 HP STEAM TURBINE OPERATING NEAR ITS SECOND CRITICAL SPEED

by

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He spent the first three years of his engineering career at Babcock and Wilcox in Lynchburg, Virginia, where he was responsible for seismic structural design,

stress analysis and rotordynamics analysis. He spent two years as a Project Engineer in the Compressor Division of Allis-Chalmers Corporation in Milwaukee, Wisconsin. He was responsible for rotordynamics analysis of multi-stage centrifugal compressors and 3-D finite element stress analysis for horizontally-split fabricated casings.

For more than five years, Mr. Salamone was Chief Engineer for Centritech Corporation in Houston, Texas. In this capacity, he was responsible for rotordynamics analysis and bearing design consultation on industrial turbomachinery for vibration problems solutions.

In 1984, Mr. Salamone founded Salamone Turbo Engineering, Incorporated. Turbo is a engineering consulting company that provides services in rotordynamics analysis and bearing

design consultation for the solution of turbomachinery vibration problems in the utility, petroleum and chemical industries.

Mr. Salamone is a member of ASLE, ASME, NSPE, Society of Sigma Xi, and the Vibration Institute. He is also a registered professional engineer in the State of Texas.

ABSTRACT

A rotordynamics analysis of an 8000 hp steam turbine which drives an ethylene compressor is presented. This machine characteristically exhibited increasing vibration levels with increasing speed, in the upper portion of the speed range. It was suspected that this ramping vibration was due to operation near a critical speed.

The purpose of this analysis was to identify the source of this high vibration with the present rotor and bearings and then to determine the best solution to the problem. The computer analysis determined that the original rotor/bearing system was operating near the second critical speed. The predicted peaks were within nine percent to thirteen percent of running speed (8400 cpm).

The optimum solution was to shorten the shaft at the governor end and replace the bearings. The shaft modification consisted of cutting off the governor worm gear assembly and replacing it with an electronic governor disk. This new bearing design was also a four-shoe tilting pad journal bearing, but it had longer pads and higher preload than the existing design. Because there was concern about coupling gear tooth lock-up in the original design, a dry, flexible-element coupling design was evaluated. The coupling change was also of interest because of its potential effect on moving the second critical speed. However, in this case, the analysis determined that the rotor vibration characteristics were very close with either the existing gear coupling or the dry coupling. Therefore, the justification for changing couplings would be that the dry coupling would not be susceptible to drive tooth lockup and unbalance eccentricity due to tooth wear.

INTRODUCTION

A rotordynamics analysis of an 8000 hp steam turbine which drives an ethylene compressor is presented. The turbine and compressor unit is shown in Figures 1 and 2. The minimum and maximum operating speeds are 6000 cpm and 9240 cpm, respectively. In this report, an operating speed of 8400 cpm will be assumed. The shaft is presently supported by two four-shoe

rocker pivot tilting pad bearings that are separated by a 67.4 in span. A cross-section of the machine is shown in Figure 3.

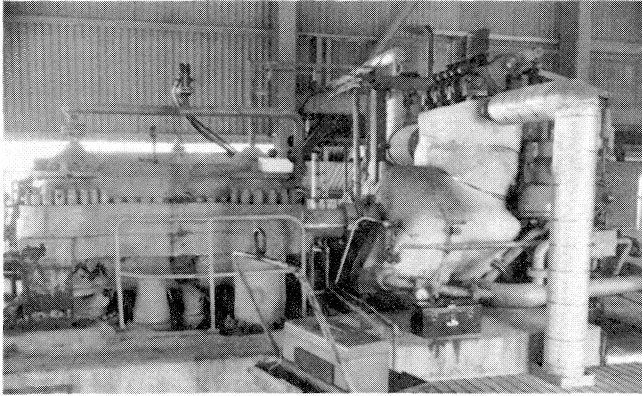


Figure 1. 8000 HP Steam Turbine and Compressor Train.

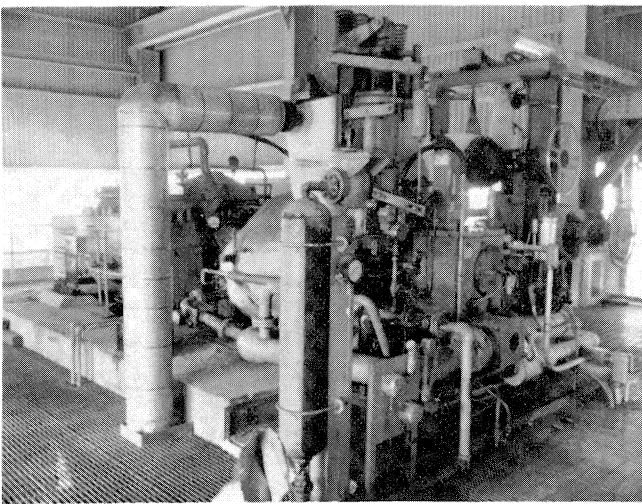


Figure 2. 8000 HP Steam Turbine and Compressor Train Viewed from Governor End.

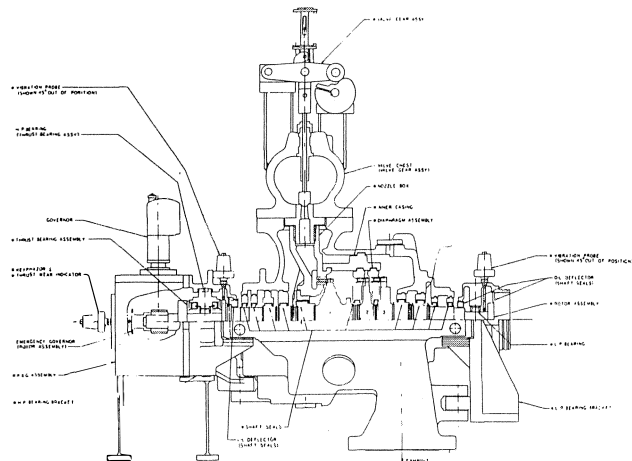


Figure 3. Cross Section of 8600 HP Steam Turbine.

This machine characteristically exhibited increasing vibration levels with increasing speed in the upper portion of the speed range. It was suspected that this ramping vibration was due to operation near a critical speed. A rotordynamics analysis was performed on the system to identify the source of this high

vibration in the machine with the present rotor and bearings and to determine the best solution to the problem.

DESCRIPTION OF ROTOR SYSTEM

The computer model, (Figure 4), consists of a series of lumped mass stations containing the weight and inertia properties of the turbine wheels, couplings, thrust collar, etc. The mass stations are connected by elastic beam elements. The rotor model also includes the bearing speed-dependent stiffness and damping properties.

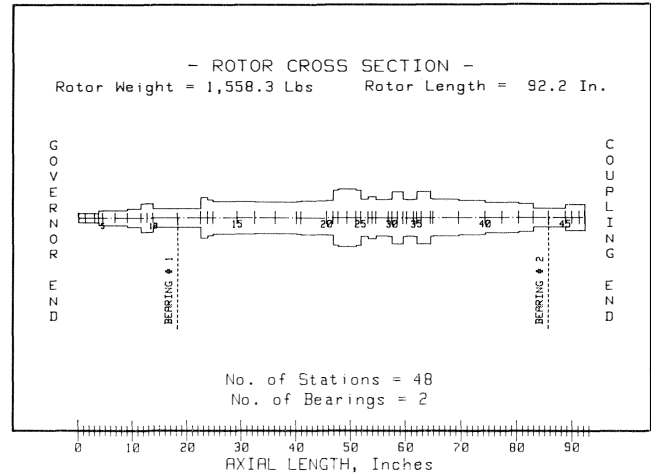


Figure 4. Computer Model of Turbine Rotor.

The turbine rotor consists of three bucketed wheels that are integral with the shaft. The weight of the wheel sections minus the base shaft are 276.7 lb, 131.7 lb, and 176.4 lb at the first, second, and third stages, respectively. The exhaust end of the turbine rotor is coupled to the compressor by a gear-type coupling, which has a half-weight of 26.4 lb. The overall shaft length is 87.25 in and the total weight, including the half-coupling weight, is 1558.4 lb.

The shaft was originally supported by two four-shoe, rocker pivot tilting pad journal bearings. The journal bearings had identical geometries, which included 4.0 in journal diameters, 1.5 in long pads, preloads of 0.22, and load-between-pad orientations. The radial bearing gravitational loads are 791.8 lb and 766.6 lb at the steam and exhaust ends, respectively. The bearing span is 67.4 in.

UNDAMPED CRITICAL SPEEDS

Critical Speed Map

The critical speed map for the turbine rotor is shown in Figure 5. This is a plot of the first three undamped critical speeds for a range of bearing stiffness values. The critical speeds increase with increasing bearing stiffness to a limit. When this limit is reached, the critical speeds are insensitive to increasing bearing stiffness because the bearings appear rigid relative to the stiffness of the shaft. Therefore, in the rigid bearing condition, the bearing damping is no longer effective for the suppression of vibration.

The rotor mode shapes indicate the relative rotor amplitudes at the undamped critical speeds. The first three undamped critical speed mode shapes for the existing rotor and coupling are shown in Figures 6, 7, and 8. The two bearings were assigned the actual stiffness values for the original bearings, as noted in the figure. Note that the frequency of mode 2, (Figure 7), is 8264 cpm. This indicates that the rotor is operating close to the

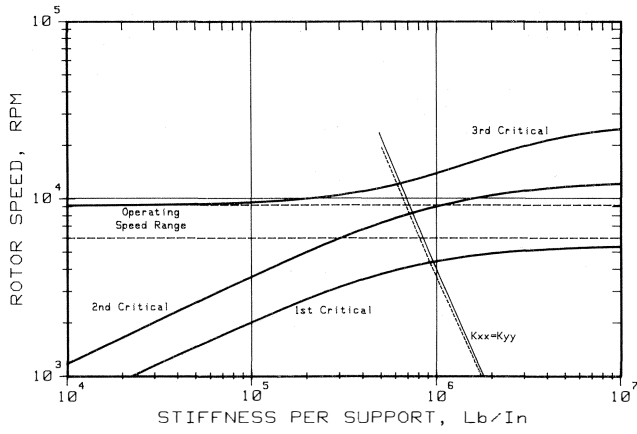


Figure 5. Undamped Critical Speed Map of Unmodified Steam Turbine Rotor.

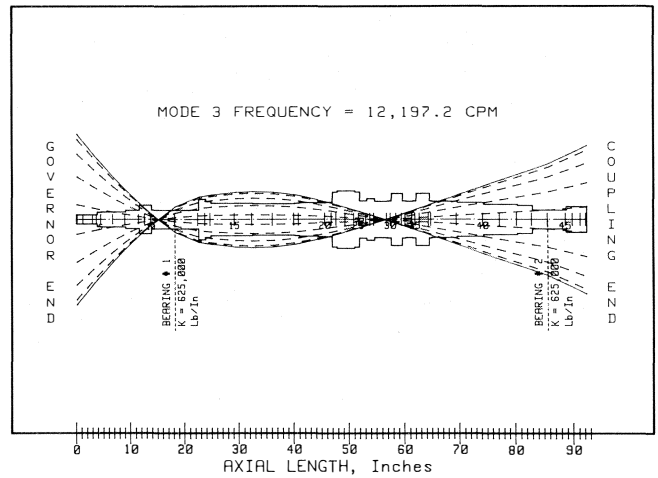


Figure 8. Mode Shape at Third Undamped Critical Speed (12197 CPM) for Unmodified Rotor Supported on Original Bearings ($K_b = 6.25 \times 10^5$ LB/IN).

ORIGINAL BEARINGS

Description

The original governor-end and coupling-end journal bearings are a four-shoe rocker pivot tilting pad design, as shown schematically in Figures 9 and 10, respectively. Both radial

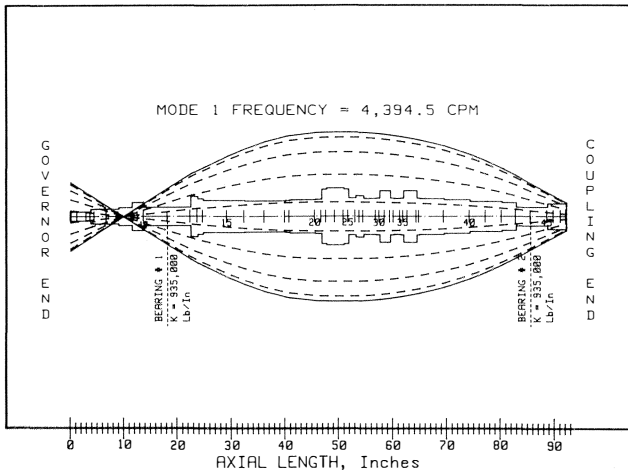


Figure 6. Mode Shape at First Undamped Critical Speed (4395 CPM) for Unmodified Rotor Supported on Original Bearings ($K_b = 9.35 \times 10^5$ LB/IN).

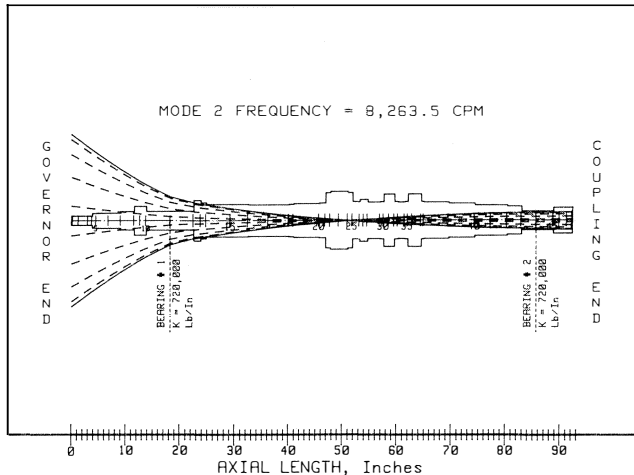


Figure 7. Mode Shape at Second Undamped Critical Speed (8264 CPM) for Unmodified Rotor Supported on Original Bearings ($K_b = 7.20 \times 10^5$ LB/IN).

second undamped critical speed. Also note the large amplitudes at the governor end of the rotor. These observations will be addressed in more detail in the unbalance response analysis.

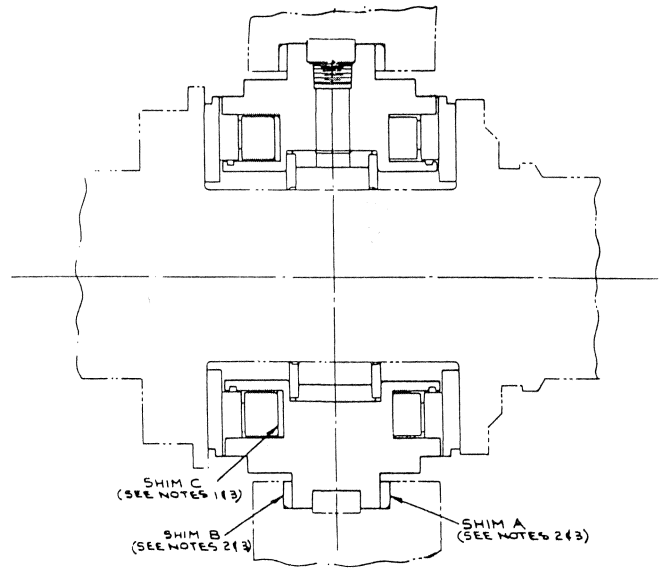


Figure 9. Original Governor-end Tilting Pad Thrust and Journal Combination Bearing.

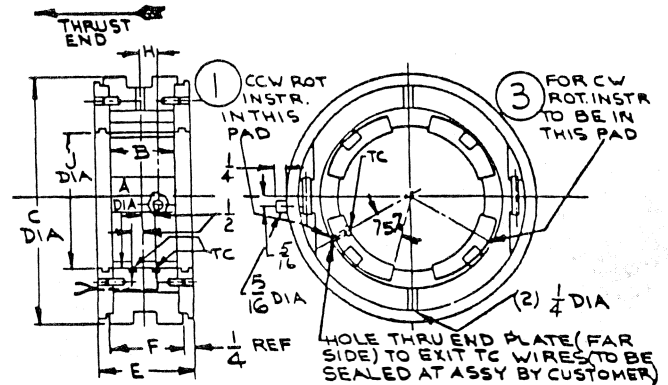


Figure 10. Original Coupling-end Tilting Pad Journal Bearing.

journal bearings have 4.00 in journal diameters, 1.5 in pad lengths, 0.007 in to 0.009 in diametral set clearances, preloads of 0.22, and load-between-pad orientations. The lubricating oil is an International Standards Organization (ISO) 32, which has an absolute viscosity of 1.57 microreyns at 145°F.

Bearing Coefficients

The stiffness and damping coefficients for the original governor-end bearing are shown in Figure 11. The coupling-end bearing coefficients are similar. Note that the four shoe bearing

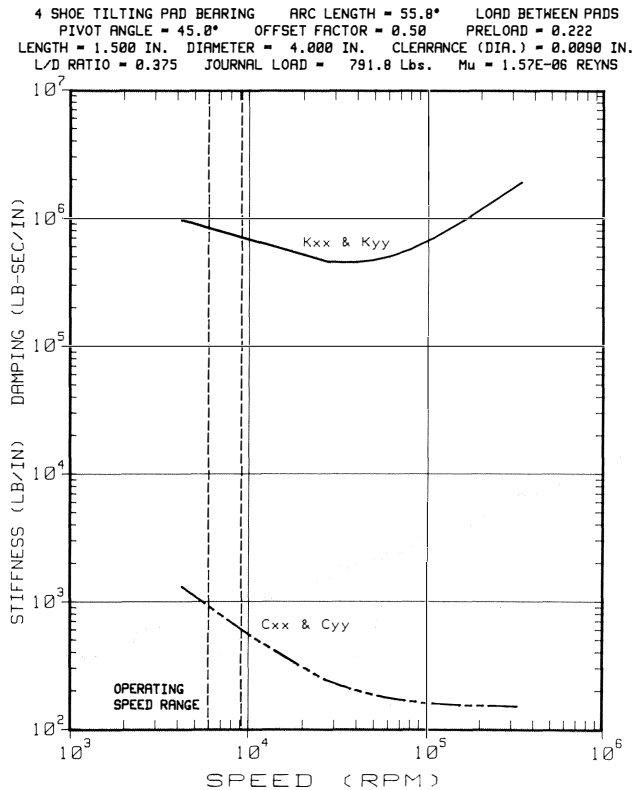


Figure 11. Original Governor-end Bearing Stiffness and Damping Coefficients vs Speed.

loaded between pads has symmetric coefficients in the horizontal and vertical planes. The critical speed map shown in Figure 5 also indicates that the bearing stiffness changes with the running speed. It is observed that the principal stiffness values for these bearings intersect the second mode curve very close to the operating speed of 8400 cpm. These intersections, which are the undamped critical speeds, are listed in Table 1.

Table 1. Undamped Critical Speeds of Unmodified Turbine Rotor Supported on Original Tilting Pad Journal Bearings.

First Critical	4,400 cpm
Second Critical	8,300 cpm
Third Critical	12,200 cpm

UNBALANCE RESPONSE FOR EXISTING ROTOR IN ORIGINAL BEARINGS

General

The unbalance response analysis simulates the rotor amplitudes of vibration during a startup or coastdown. These results indicate the rotor sensitivity at the critical speeds [1, 2].

In this analysis, the response was examined over a speed range from 1500 cpm to 13950 cpm.

All of the amplitudes herein are equivalent to half of the peak-to-peak values. These amplitudes should be used only for comparison purposes, since they are dependent on the assumed amount and location of the unbalance. It should also be noted that this analysis assumes that the bearings are supported by a rigid casing and base. This assumption must be used with discretion, but it was reasonable in this analysis.

Unbalance Criteria

The rotor unbalance criteria consists of the following:

- In-Phase: ten percent of the rotor weight at each of the turbine wheels (at 90 degrees).
- Out-of-Phase:
 - Ten percent of the governor-end bearing load at the worm gear assembly (at 0 degrees).
 - Ten percent of the coupling-end bearing load at the coupling (at 180 degrees).
 - Coupling: 2.0 mil eccentricity of half coupling (at 180 degrees).

Predicted Response

The predicted peak responses and the running speed amplitudes, respectively, for the existing rotor supported in the original bearings are summarized in Tables 2 and 3. Note that there are response peaks between 9150 cpm and 9450 cpm. These peaks are within nine percent to thirteen percent of running speed (8400 cpm). Also relatively high governor-end amplitudes at running speed are indicated in Table 3. Response plots for this case are depicted in Figures 12, 13, and 14. Note that part "a" is the Bodé amplitude plot and part "b" is the polar plot. Descriptions of the rotor station numbers referred to on the figures are presented in Table 4. The absolute amplitudes across the rotor are illustrated in Figure 15. The absolute amplitude is the magnitude of the amplitude vector. Therefore, it does not indicate the phase relationship.

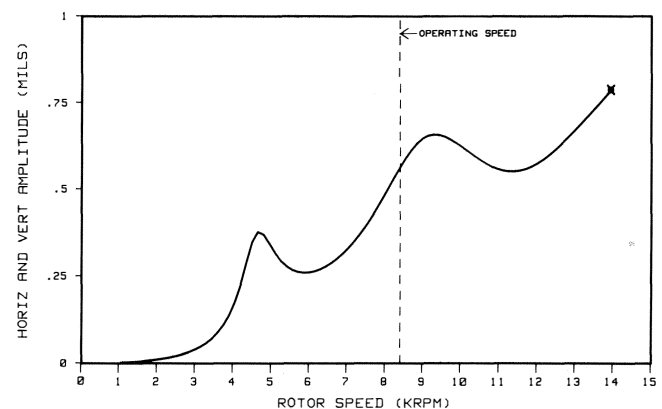


Figure 12a. Synchronous Response Amplitudes vs Rotor Speed at Governor-end Bearing Location. Unmodified turbine rotor supported on original journal bearings.

ROTOR STABILITY FOR EXISTING ROTOR IN ORIGINAL BEARINGS

Rotor stability is assessed by calculating the damped eigenvalues of the rotor/bearing system [3, 4]. The damped eigen-

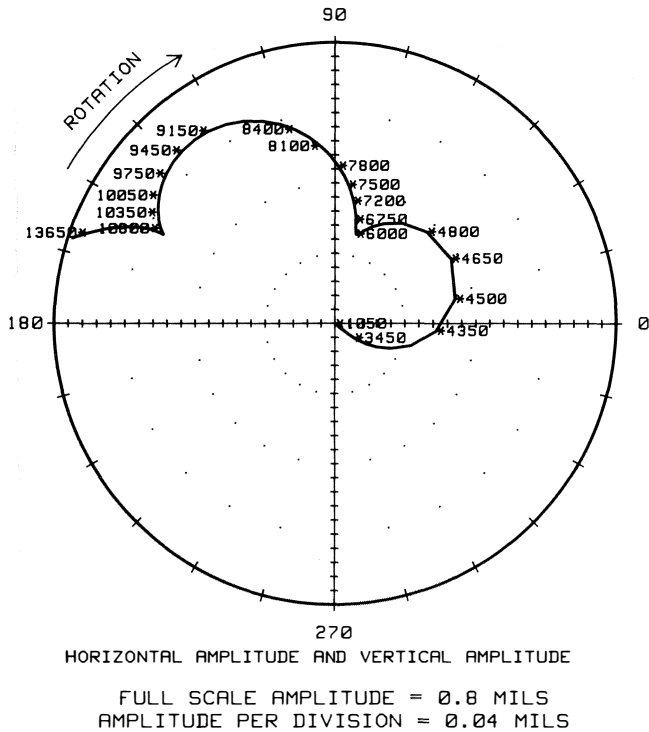


Figure 12b. Nyquist Plot of Response at Governor-end Bearing Location. Unmodified turbine rotor supported on original journal bearings.

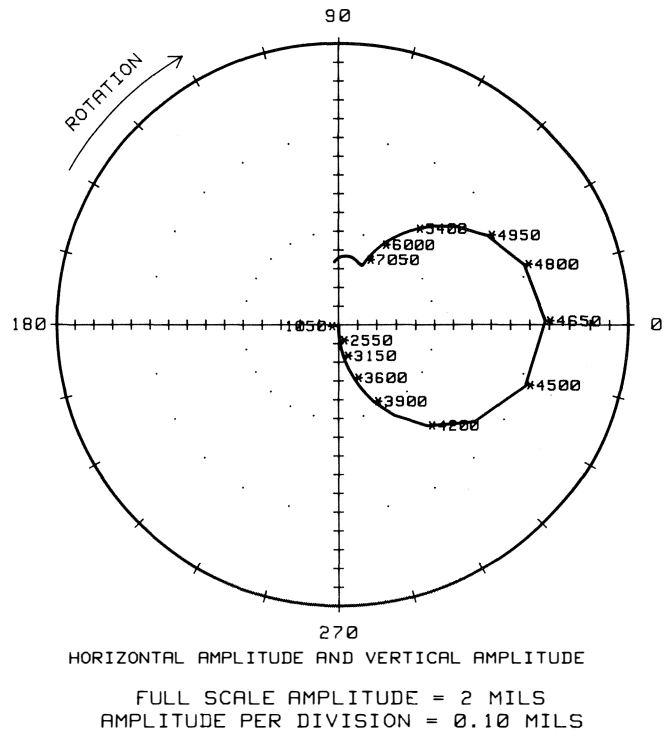


Figure 13b. Nyquist Plot of Response at First Stage Wheel Location. Unmodified turbine rotor supported on original journal bearings.

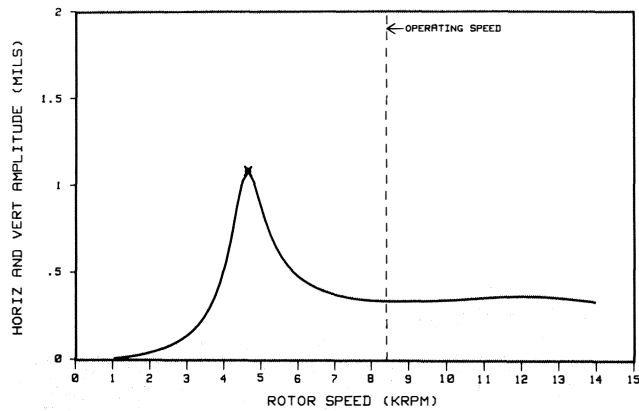


Figure 13a. Synchronous Response Amplitudes vs Rotor Speed at First Stage Wheel Location. Unmodified turbine rotor supported on original journal bearings.

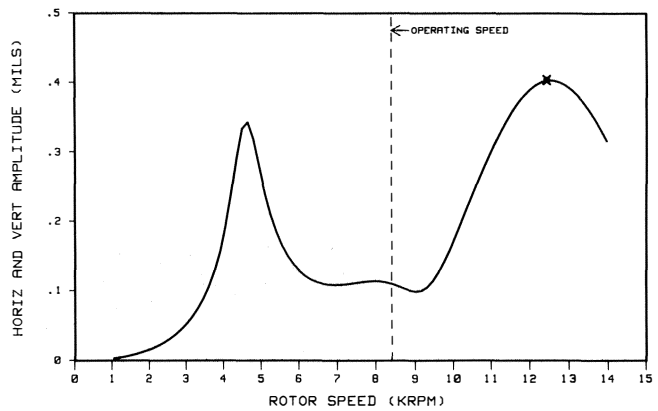


Figure 14a. Synchronous Response Amplitudes vs Rotor Speed at Coupling-end Bearing Location. Unmodified turbine rotor supported on original journal bearings.

value ($s = \lambda + i\omega_d$) consists of a growth factor (λ) and a damped natural frequency (ω_d). The growth factor is the rate of exponential decay or growth of the vibration amplitudes.

The logarithmic decrement (δ) is used to determine the stability of a system. It is defined as the natural logarithm of the ratio of two successive vibration amplitudes. The logarithmic decrement is computed from the complex eigenvalue by the formula: $\delta = -2\pi\lambda/\omega_d$. If δ is positive, the system is stable. If δ is negative, the system is unstable. From experience, it is desirable to have a logarithmic decrement greater than or equal to +0.25.

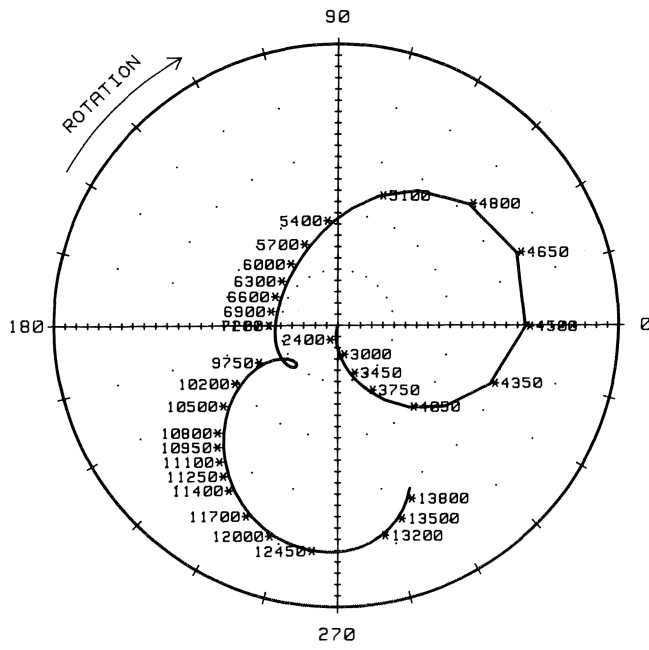
The stability results for the original rotor/bearing system are listed in Table 5. The rotor is stable in the original bearings

without external steam cross-coupling effects. The logarithmic decrement is +0.58 at the first forward whirl mode.

SUMMARY OF ORIGINAL SYSTEM DYNAMICS

This analysis determined that the original rotor/bearing system was operating near the second critical speed. The predicted second peak response speed range was between 9150 cpm and 9450 cpm in the horizontal and vertical planes. These peak speeds were within nine percent to thirteen percent of the running speed (8400 cpm).

From the predicted response of the original system, it is important to observe the high vibration amplitudes at the



HORIZONTAL AMPLITUDE AND VERTICAL AMPLITUDE

FULL SCALE AMPLITUDE = 0.5 MILS
 AMPLITUDE PER DIVISION = 0.02 MILS

Figure 14b. Nyquist Plot of Response at Coupling-end Bearing Location. Unmodified turbine rotor supported on original journal bearings.

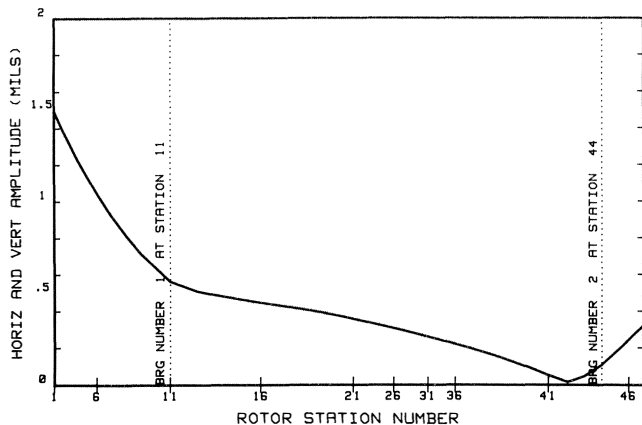


Figure 15. Magnitude of Synchronous Response Amplitude Vectors (Magnitude Only) Along Rotor Shaft for Unmodified Turbine Rotor Supported on Original Journal Bearings. Rotor speed is 8400 cpm.

Table 2. Synchronous Peak Response Amplitudes (Mils) for Unmodified Turbine Rotor Supported on Original Tilting Pad Journal Bearings.

	Shaft Location				
	1	11	23	44	47
	Axial Position Disk	Governor End Bearing	First Stage Wheel ¹	Coupling End Bearing	Coupling
First Peak Amplitude	0.58	0.38	1.08	0.34	No Peak ²
Peak Speed	4,500	4,650	4,650	4,650	—
Second Peak Amplitude	1.74	0.66	0.37	0.40	0.69
Peak Speed(s)	9,150	9,300-9,450	11,400-12,600	12,150-12,900	12,000-12,150

NOTES: (1) The First Stage Wheel Is Close to the Center of the Bearing Span
 (2) No Peak Found Up to 13,950 CPM

Table 3. Synchronous Response Amplitudes (Mils) at Running Speed(8,400 CPM) for Unmodified Turbine Rotor Supported on Original Tilting Pad Journal Bearings.

	Shaft Location				
	1	11	23	44	47
	Axial Position Disk	Governor End Bearing	First Stage Wheel	Coupling End Bearing	Coupling
Amplitude (Horizontal & Vertical)	1.50	0.56	0.34	0.11	0.29

Table 4. Cross Reference for Unmodified Rotor Computer Model Station Locations.

Station Number	Location
1	Axial Position Disk
11	Governor End Bearing
23	First Stage Wheel (Close to Center of Bearing Span)
44	Coupling-End Bearing
47	Coupling

Table 5. Stability for Unmodified Turbine Rotor Supported on Original Tilting Pad Journal Bearings. Rotor Speed is 8,400 cpm.

Damped Frequency (CPM)	Log Decrement	Condition	Whirl Direction
4221	+0.58	Stable	FWD
4204	+0.57	Stable	BWD

governor-end of the shaft, relative to those at other rotor locations. These amplitudes could cause distress at the governor gears. It is also important that the predicted steam-end bearing amplitudes are higher than those at the exhaust end. This characteristic is opposite to the original field measurements, and it indicates that the governor gear mesh forces are probably significant. However, the characteristic of higher relative amplitudes at the steam end was observed in the balance machine. This is important because in the balance machine the governor worm was not engaged.

Because it is difficult to analytically simulate the true influence of the worm gear, its potential effect was noted but was not included in the rotor model. Instead, the analysis was directed toward identifying the optimum modifications that would move the second critical speed out of the operating speed range. The modifications that were considered included three feasible alteration possibilities:

- **Bearing Redesign**
 - This option would consider hydrodynamic stiffness and damping changes to shift the second critical speed or dampen it out. However, care must be taken to avoid increasing the sensitivity at the first critical speed.
- **Shaft Modification**
 - A change to the shaft overhangs can have a very significant effect on the second and third critical speeds. However, shaft changes must be made judiciously because of their irreversibility. Also, there are other mechanical considerations, such as encroachment upon worm gear drives and mechanical over-speed trip pins, which must be considered. This option could also include changes in the bearing span.

• *Coupling Redesign*

This option would consider changes in weight and over-hung moment. Therefore, the coupling change can significantly affect the second and third critical speeds. In considering this option, the analyst must ensure that the coupling change does not adversely affect the torsional critical speeds.

NEW TILT PAD JOURNAL BEARING DESIGN

The first modification that was considered in this analysis was a bearing redesign that would fit into the existing bearing brackets without requiring any shaft changes.

Description

The new governor and coupling-end bearings are illustrated in Figure 16. The new radial bearings are four-shoe tilt pad bearings with a longer pad length, higher preload and larger pad arc. Specifically, the new design has a length-to-diameter ratio of 1.0, preload of 0.33, pad arc of 72 degrees, and orientation of load-between-pad.

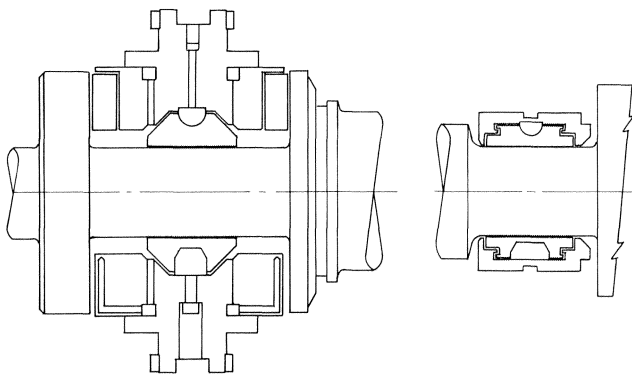


Figure 16. New Bearing Design Layout.

Bearing Coefficients

The stiffness and damping coefficients for the new governor-end journal bearing are presented in Figure 17. The coupling-end coefficients are similar. The critical speed map for the original rotor, showing a cross-plot of the new bearing stiffness values, is shown in Figure 18. Note that the new stiffness still intersects the second critical speed curve in the operating speed range. However, the redesign has improved stiffness and damping characteristics compared to the original design. A comparison of the original and new bearing stiffnesses is also illustrated in Figure 18.

Comparison of Original and New Bearings

The stiffness and damping ratios computed from the formulas presented by Barrett, et al. [5], and discussed by Salamone [6, 7], are listed in Table 6. Note that for both bearings the stiffness ratios are well below the recommended value of 6.0 [5]. The important difference is that the new bearings provide more flexibility and more damping than the original bearings. Most important, the new bearings provide 109 percent of the optimum damping compared to the original 85 percent. Next, the unbalance response and stability calculations will be presented to provide further insight into the merits of one design over the other.

STABILITY AND UNBALANCE RESPONSE WITH NEW BEARINGS

Stability

The original rotor with new bearings has a very stable logarithmic decrement of +1.25 (Table 7).

4 SHOE TILTING PAD BEARING ARC LENGTH = 72.0° LOAD BETWEEN PADS
 PIVOT ANGLE = 45.0° OFFSET FACTOR = 0.50 PRELOAD = 0.333
 LENGTH = 4.000 IN. DIAMETER = 4.000 IN. CLEARANCE (DIA.) = 0.0105 IN.
 L/D RATIO = 1.000 JOURNAL LOAD = 791.8 Lbs. $\mu = 1.57E-06$ REYNS

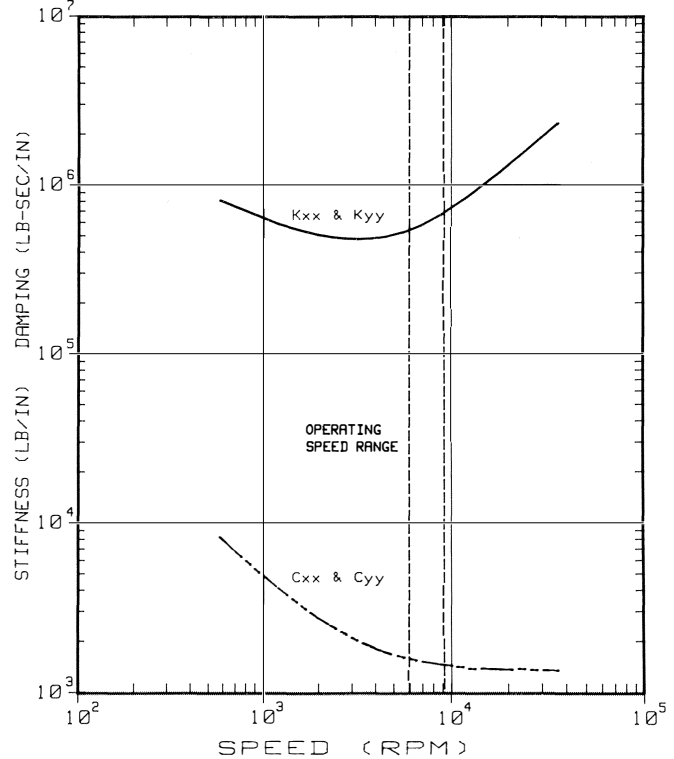


Figure 17. New Governor-end Bearing Stiffness and Damping Coefficients vs Speed.

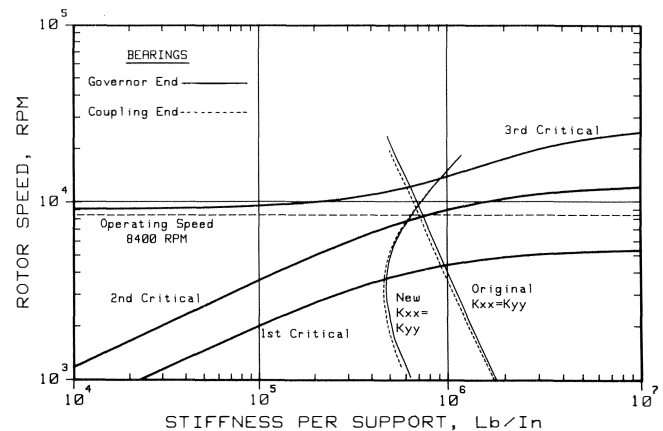


Figure 18. Undamped Critical Speed Map for Unmodified Rotor Showing Comparison Between Original and New Tilting Pad Journal Bearings.

Unbalance Response

The peak responses and amplitudes at the running speed are listed in Tables 8 and 9. For comparison, these tables also list the base case, which considers the original rotor, bearings and coupling. The percentage changes from the base case give a good indication of improvement with the new bearing design.

The most significant improvement with the new bearings is the effect on the second critical speed. The second peak responses are damped out with the new bearings (Table 8). In addition, the peak amplitudes at the first critical are reduced by 9 to 43 percent. Significant amplitude reductions of 3 to 57 percent at the running speed are reflected in Table 9. The rotor center amplitude is increased by six percent, but this increase is

Table 6. Comparison of Stiffness and Damping Ratios for Original and New Journal Bearings. Considers Unmodified Rotor.

	Bearing Type	
	Original 4 Pad TPJ 1. L/D=0.375 2. Preload=0.22	New 4 Pad TPJ 1. L/D=1.0 2. Preload=0.33
Stiffness Ratio:		1.54
\bar{K} (DIM)	2.57	
Optimum Damping Ratio:		1.27
ξ_{opt} (DIM)	1.79	
Actual Damping Ratio:		1.39
ξ_{act} (DIM)	0.85 48% ξ_{opt}	109% ξ_{opt}

Table 7. Stability for Unmodified Turbine Rotor Supported on New Tilting Pad Bearings. Rotor Speed is 8,400 cpm.

Damped Frequency (CPM)	Log Decrement	Condition	Whirl Direction
4552	+ 1.25	Stable	FWD
4532	+ 1.21	Stable	BWD

Table 8. Synchronous Peak Response Amplitudes (Mils) for Unmodified Turbine Rotor Supported on New Tilting Pad Journal Bearings.

	Shaft Location				
	1	11	23	44	47
	Axial Position Disk	Governor End Bearing	First Stage Wheel (1)	Coupling End Bearing	Coupling
Present Case					
First Peak					
Amplitude	0.53	0.27	0.62	0.20	No Peak ⁽²⁾
% Change from Base	-9	-29	-43	-41	—
Peak Speed(s)	4,800	4,950-5,550	5,100-5,250	4,800-4,950	—
Second Peak					
Amplitude	2.52	0.57	No Peak	No Peak	No Peak
% Change from Base	—	—	—	—	—
Peak Speed(s)	12,450-12,900	12,600-13,050	—	—	—
Base Case⁽³⁾					
First Peak					
Amplitude	0.58	0.38	1.08	0.34	No Peak
Peak Speed(s)	4,500	4,650	4,650	4,650	—
Second Peak					
Amplitude	1.74	0.66	0.37	0.40	0.69
Peak Speed(s)	9,150	9,300-9,450	11,400-12,600	12,150-12,900	12,000-12,150

Notes: (1) The First Stage Wheel Is Close to the Center of the Bearing Span
 (2) No Peak Found Up to 13,950 cpm
 (3) Original Rotor, Bearings and Coupling

Table 9. Synchronous Response Amplitudes (Mils) at Running Speed (8,400 cpm) for Unmodified Turbine Rotor Supported on New Tilting Showing New Tilting Pad Bearing Stiffnesses.

	Shaft Location				
	1	11	23	44	47
	Axial Position Disk	Governor End Bearing	First Stage Wheel	Coupling End Bearing	Coupling
Present Case					
Amplitude (Horizontal & Vertical)	0.65	0.25	0.36	0.09	0.28
% Change from Base	-57	-55	+6	-18	-3
Base Case⁽¹⁾					
Amplitude (Horizontal & Vertical)	1.50	0.56	0.34	0.11	0.29

Note: (1) Original Rotor, Bearings and Coupling

from 0.34 to 0.36 and is insignificant. The response amplitude plots and polar plots are presented in Figures 19, 20, and 21. The absolute amplitudes along the rotor at the running speed are illustrated in Figure 22.

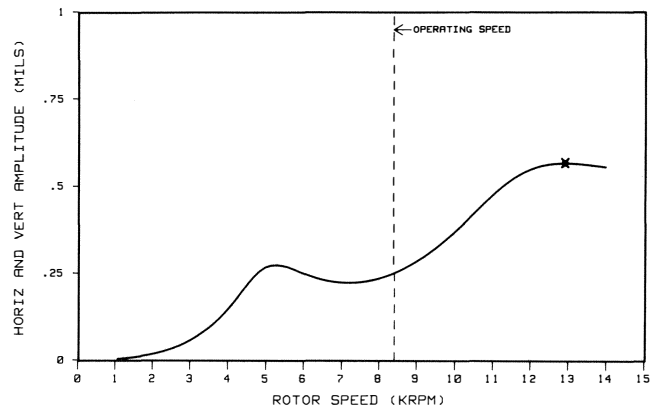


Figure 19a. Synchronous Response Amplitudes vs Rotor Speed at Governor-end Bearing Location. Unmodified turbine rotor supported on new journal bearings.

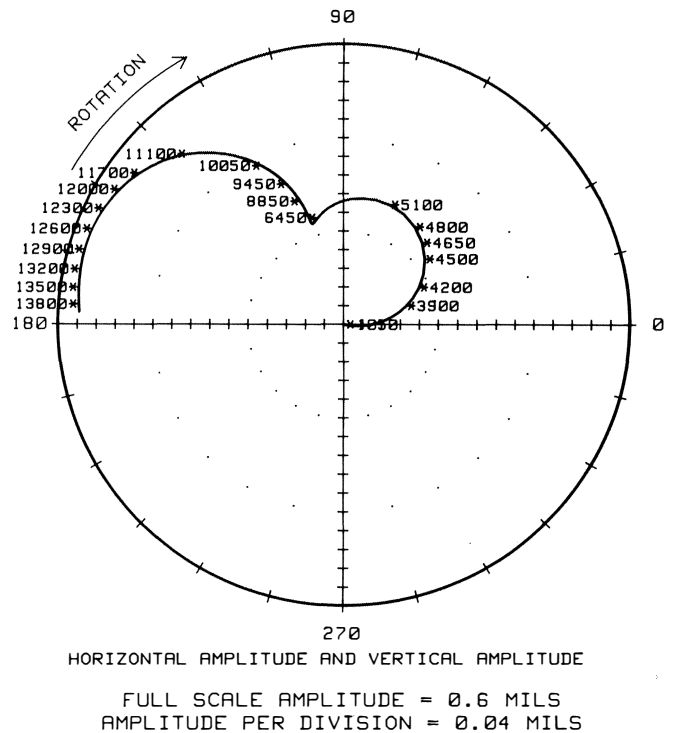


Figure 19b. Nyquist Plot of Response at Governor-end Bearing Location. Unmodified turbine rotor supported on new journal bearings.

ROTOR MODIFICATION

The second level of modification considered in this analysis is a shaft modification.

Description

The rotor modification (Figure 23) considered cutting off the end of the shaft, just outboard of the thrust collar, and replacing the worm gear governor drive with an electronic governor disk. In this modified-rotor computer model, the governor end of the shaft extends 3.5 in outboard of the thrust disk into the 3.25 in diameter section, where the worm gear was mounted on the

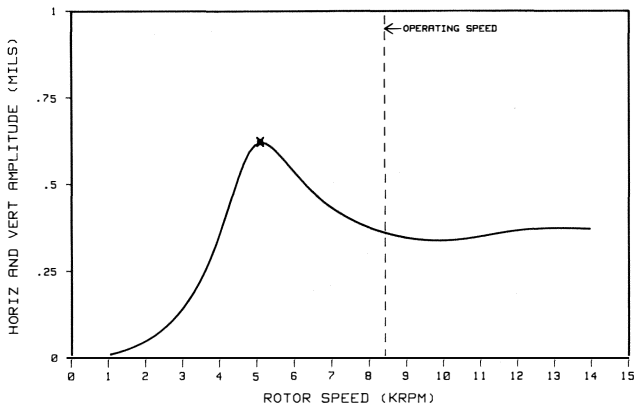


Figure 20a. Synchronous Response Amplitudes vs Rotor Speed at First Stage Wheel Location. Unmodified turbine rotor supported on new journal bearings.

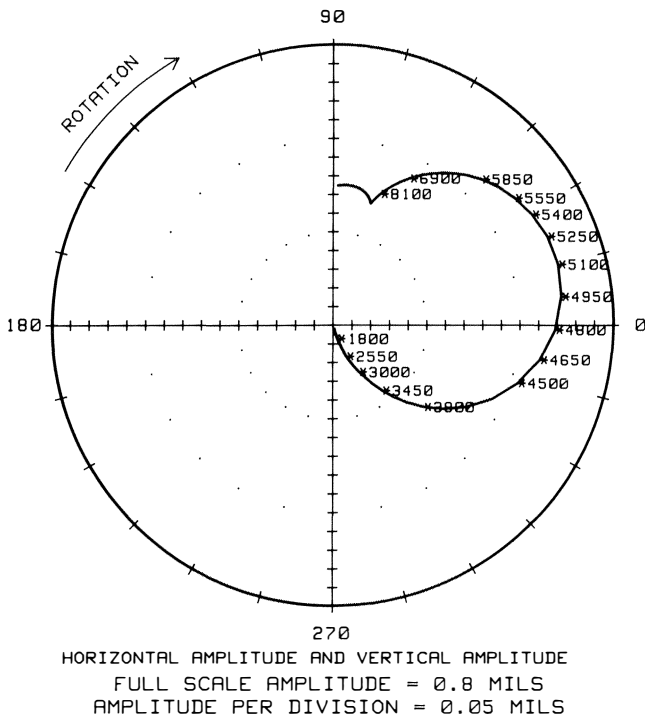


Figure 20b. Nyquist Plot of Response at First Stage Wheel Location. Unmodified turbine rotor supported on new journal bearings.

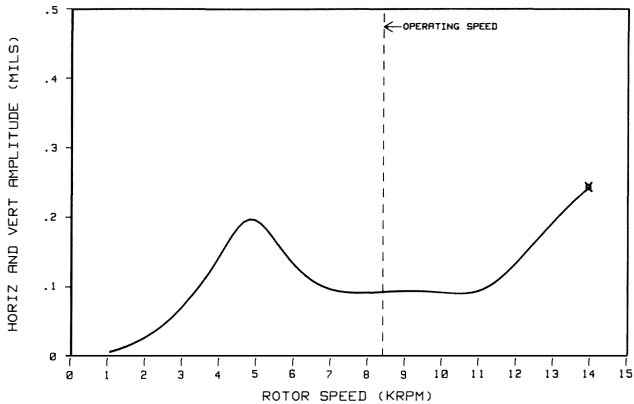


Figure 21a. Synchronous Response Amplitudes vs Rotor Speed at Coupling-end Bearing Location. Unmodified turbine rotor supported on new journal bearings.

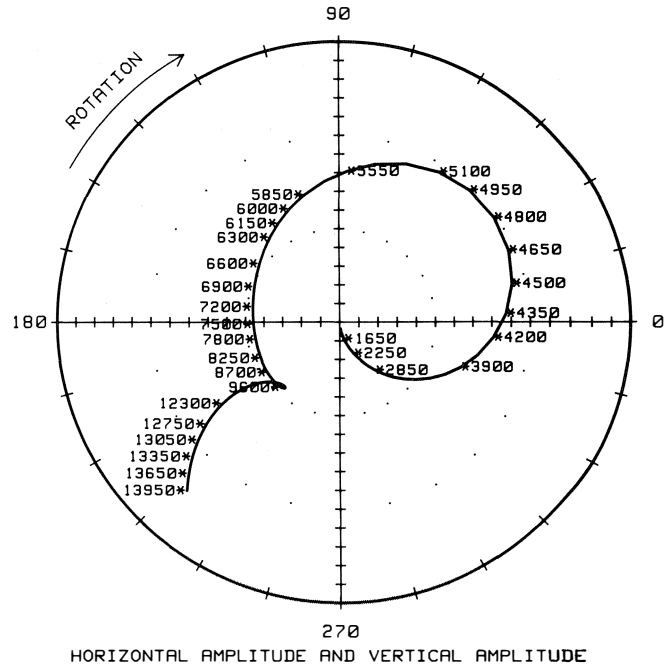


Figure 21b. Nyquist Plot of Response at Coupling-end Bearing Location. Unmodified turbine rotor supported on new journal bearings.

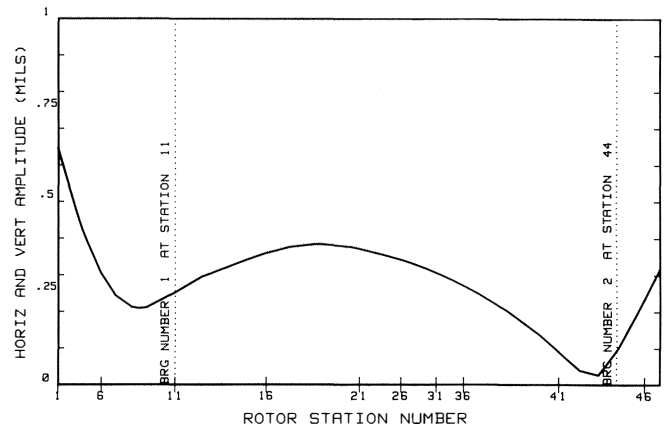


Figure 22. Magnitude of Synchronous Response Amplitude Vectors (Magnitude Only) Along Rotor Shaft for Unmodified Turbine Rotor Supported on New Journal Bearings. Rotor speed is 8400 cpm.

original shaft. Therefore, on the modified rotor, the amplitudes at the axial position disk refer to the end of this electronic governor disk. The new electronic governor disk was assumed to be a 5.875 in diameter solid disk with a four-in diameter counter-bore that is 0.188 in deep, as shown in the figure. For reference purposes, this disk would weigh approximately 8.4 lb. The disk mounting arrangement is shown only for illustration purposes.

Effect on Undamped Critical Speed Map

A comparison of the original and modified rotor critical speeds is shown in Figure 24. Note the significant increase in the second and third critical speed curves. The map indicates that the shaft modification is very effective in moving the second critical speed.

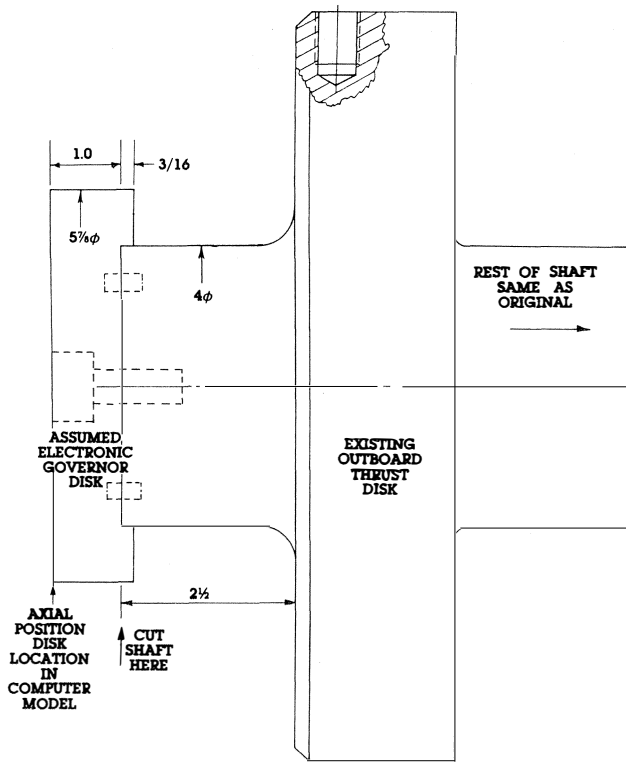


Figure 23. Shaft Modification to Shorten Rotor at Governor End. The Governor-end is reduced in length by 9.0 in.

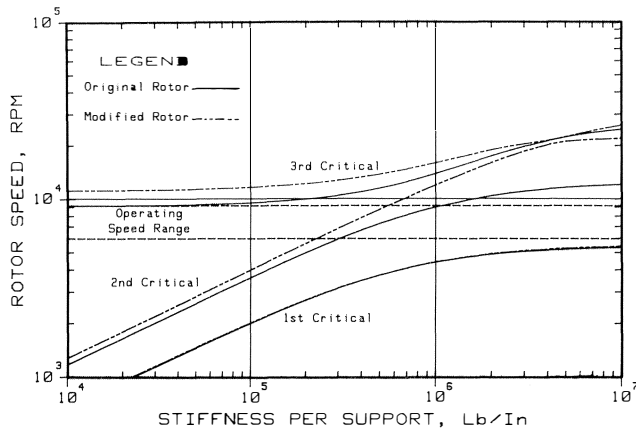


Figure 24. Undamped Critical Speed Map Showing Comparison of Unmodified and Modified Rotors.

Critical Speeds of Modified Rotor with Original Bearings

The modified rotor is lighter and has a different center of gravity than the original rotor. This reduces the governor-end bearing load by six percent. The coupling-end bearing load change was insignificant. Therefore, only the governor-end bearing coefficients were recalculated. The modified rotor critical speed map with the original journal bearing stiffnesses is shown in Figure 25. The undamped critical speeds for this case are listed in Table 10. Note that the first undamped critical speed is not changed, but the second and third critical speeds are significantly increased. Most importantly, the second undamped is increased from 8300 cpm to 10000 cpm.

Response of Modified Rotor with the Original Bearings

The peak responses and the amplitudes at the running speed, respectively, for the modified rotor in the existing bearings are

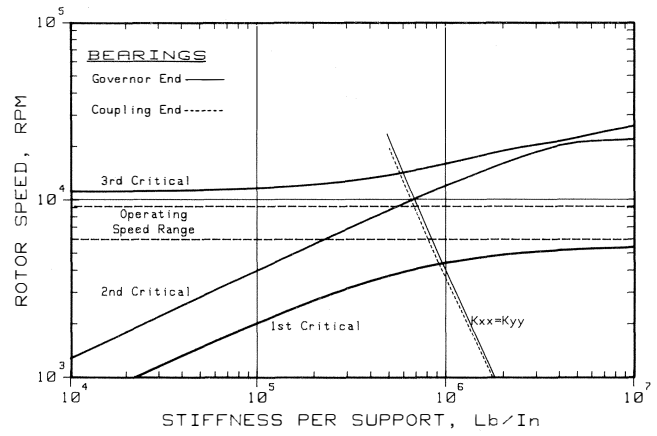


Figure 25. Undamped Critical Speed Map for Modified Turbine Rotor Showing Original Tilting Pad Bearing Stiffnesses.

Table 10. Undamped Critical Speeds for Modified Turbine Rotor Supported on Original Tilting Pad Journal Bearings.

First Critical	4,400 cpm
Second Critical	10,000 cpm
Third Critical	14,100 cpm

summarized in Tables 11 and 12. Note the 52 percent reduction in peak amplitude at the axial position disk (Table 11). Also, there are no peak responses between 9150 cpm and 9450 cpm. The amplitudes at the running speed (Table 12) are reduced from 1.50 mils to 0.36 mils (a 76 percent reduction). The amplitude comparisons must be reviewed relative to the amount in the base case. For example, the exhaust end bearing shows an 82 percent increase with the modification, but the amplitude change is only from 0.11 to 0.20 mils. Therefore, this increase is insignificant.

Table 11. Synchronous Peak Response Amplitudes (Mils) for Modified Turbine Rotor Supported on Original Tilting Pad Journal Bearings.

	Shaft Location				
	1	6	18	39	42
	Axial Position Disk	Governor End Bearing	First Stage Wheel ⁽¹⁾	Coupling End Bearing	Coupling
Present Case					
<i>First Peak</i>					
Amplitude	0.28	0.38	1.01	0.31	No Peak ⁽²⁾
% Change from Base	-52	0	-6	-9	—
Peak Speed(s)	4,500	4,650	4,650	4,650	—
<i>Second Peak</i>					
Amplitude	No Peak	0.68	No Peak	No Peak	No Peak
% Change from Base	—	—	—	—	—
Peak Speed(s)	—	12,150-12,600	—	—	—
Base Case⁽³⁾					
<i>First Peak</i>					
Amplitude	0.58	0.38	1.08	0.34	No Peak
Peak Speed(s)	4,500	4,650	4,650	4,650	—
<i>Second Peak</i>					
Amplitude	1.74	0.66	0.38	0.40	No Peak
Peak Speed(s)	9,150	9,300-9,450	11,400-12,600	12,150-12,900	—

Notes: (1) The First Stage Wheel Is Close to the Center of the Bearing Span
 (2) No Peak Found Up to 13,950 cpm
 (3) Original Rotor, Bearings and Coupling

Critical Speeds of Modified Rotor with New Bearings

The modified rotor critical speed map with the new bearing stiffnesses is shown in Figure 26. The undamped critical speeds from the map are summarized in Table 13. In this case, the second undamped critical speed is increased to 10600 cpm. The

Table 12. Synchronous Response Amplitudes (Mils) at Running Speed (8,400 cpm) for Modified Turbine Rotor Supported on Original Tilting Pad Journal Bearings.

	Shaft Location				
	1	6	18	39	42
	Axial Position Disk	Governor End Bearing	First Stage Wheel	Coupling End Bearing	Coupling
Present Case					
Amplitude (Horizontal & Vertical)	0.36	0.37	0.34	0.20	0.38
% Change from Base	-76	-34	0	+82	+31
Base Case⁽¹⁾					
Amplitude (Horizontal & Vertical)	1.50	0.56	0.34	0.11	0.29

Note: (1) Original Rotor, Bearings and Coupling

mode shapes of the modified rotor supported in the new bearings are illustrated in Figures 27 and 28. Notice that this second mode shape shows more coupling-end bearing amplitude than the original second mode shape. This additional motion provides better utilization of the fluid film damping in the bearings. This point is further confirmed by comparing the stiffness and damping ratios, previously discussed. A comparison of the original and new bearings with the modified rotor is presented in Table 14. Note that the actual damping ratio increases from 47 percent to 110 percent of the optimum.

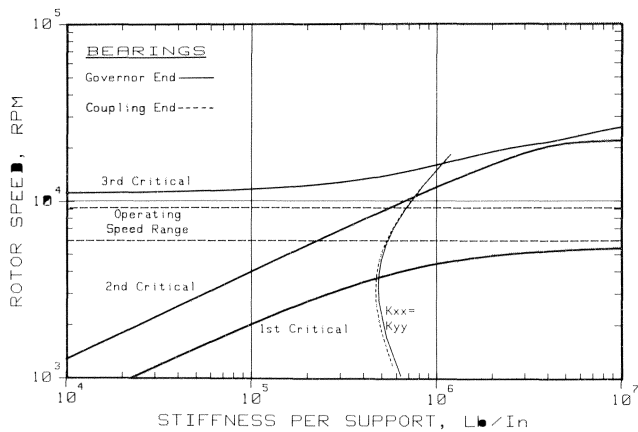


Figure 26. Undamped Critical Speed Map for Modified Turbine Rotor Showing New Tilting Pad Bearing Stiffnesses.

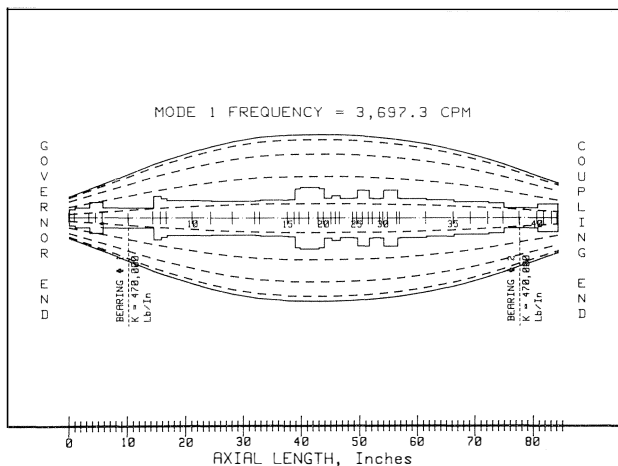


Figure 27. Mode Shape at First Undamped Critical Speed (3697 CPM) for Modified Rotor Supported on New Bearings ($K_b = 4.7 \times 10^5$ LB/IN).

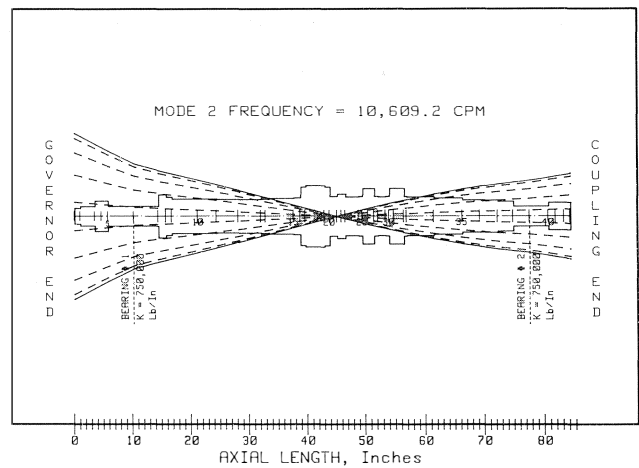


Figure 28. Mode Shape at Second Undamped Critical Speed (10609 CPM) for Modified Rotor Supported on New Bearings ($K_b = 7.50 \times 10^5$ LB/IN).

Table 13. Undamped Critical Speeds for Modified Turbine Rotor Supported on New Tilting Pad Journal Bearings.

First Critical	3,700 cpm
Second Critical	10,600 cpm
Third Critical	16,300 cpm

Table 14. Comparison of Stiffness and Damping Ratios for Original and New Journal Bearings. Considers Modified Rotor.

	Bearing Type	
	Original 4 Pad TPJ 1. L/D=0.375 2. Preload=0.22	New 4 Pad TPJ 1. L/D=1.0 2. Preload=0.33
Stiffness Ratio: \bar{K} (DIM)	2.48	1.51
Optimum Damping Ratio: ξ_{opt} (DIM)	1.74	1.26
Actual Damping Ratio: ξ_{act} (DIM)	0.83	1.38
	47% ξ_{opt}	110% ξ_{opt}

Stability and Response of Modified Rotor with Original Bearings

The modified rotor with new bearings has a very stable logarithmic decrement of +1.30 (Table 15).

Table 15. Stability for Modified Turbine Rotor Supported on New Tilting Pad Bearings. Rotor Speed is 8,400 cpm.

Damped Frequency (CPM)	Log Decrement	Condition	Whirl Direction
4512	+1.30	Stable	FWD
4492	+1.26	Stable	BWD

The peak responses and amplitudes at the running speed, including comparisons to the base case, as previously discussed, are listed in Tables 16 and 17. The rotor modification shows a very significant improvement in amplitudes at the first critical speed (Table 16). There are amplitude reductions between 18 percent and 33 percent at all locations. Finally, significant running speed amplitude reductions of 64 percent at the

governor-end bearing and 94 percent at the new governor/position disk are shown in Table 17. The amplitude increases are not out of proportion relative to the rest of the rotor. The response amplitudes at the running speed and the response plots are presented in Figures 29, 30 and 31. The absolute amplitude distribution across the rotor at the running speed is illustrated in Figure 32. The descriptions of each modified rotor station number are presented in Table 18.

Table 16. Synchronous Peak Response Amplitudes (Mils) for Modified Turbine Rotor Supported on New Tilting Pad Journal Bearings.

	Shaft Location				
	1	6	18	39	42
	Axial Position Disk	Governor End Bearing	First Stage Wheel (1)	Coupling End Bearing	Coupling
<i>Present Case</i>					
<i>First Peak</i>					
Amplitude	0.32	0.27	0.60	0.18	No Peak ⁽²⁾
% Change from Base	-45	-29	-44	-47	—
Peak Speed(s)	4,800-5,100	5,100-5,400	5,100-5,250	4,650-4,950	—
<i>Second Peak</i>					
Amplitude	No Peak	No Peak	No Peak	No Peak	No Peak
% Change from Base	—	—	—	—	—
Peak Speed(s)	—	—	—	—	—
<i>Base Case⁽³⁾</i>					
<i>First Peak</i>					
Amplitude	0.58	0.38	1.08	0.34	No Peak
Peak Speed(s)	4,500	4,650	4,650	4,650	—
<i>Second Peak</i>					
Amplitude	1.74	0.66	0.37	0.40	0.69
Peak Speed(s)	9,150	9,300-9,450	11,400-12,600	12,150-12,900	12,000-12,150

Notes: (1) The First Stage Wheel Is Close to the Center of the Bearing Span
 (2) No Peak Found Up to 13,950 cpm
 (3) Original Rotor, Bearings and Coupling

Table 17. Synchronous Response Amplitudes (Mils) at Running Speed (8,400 cpm) for Modified Turbine Rotor Supported on New Tilting Pad Journal Bearings.

	Shaft Location				
	1	6	18	39	42
	Axial Position Disk	Governor End Bearing	First Stage Wheel	Coupling End Bearing	Coupling
<i>Present Case</i>					
Amplitude (Horizontal & Vertical)	0.07	0.19	0.38	0.13	0.32
% Change from Base	-95	-66	+12	+18	+10
<i>Base Case⁽¹⁾</i>					
Amplitude	1.5	0.56	0.34	0.11	0.29

Note: (1) Original Rotor, Bearings and Coupling

Table 18. Cross Reference for Modified Rotor Computer Model Station Locations.

Station Number	Location
1	Axial Position/Electronic Governor Disk
6	Governor-End Bearing
18	First Stage Wheel (Close to Center of Bearing Span)
39	Coupling-End Bearing
42	Coupling

ALTERNATE COUPLING DESIGN

An alternate coupling design was evaluated, because there was concern about coupling gear tooth lockup in the original design. The alternate design is a dry-type coupling. This design

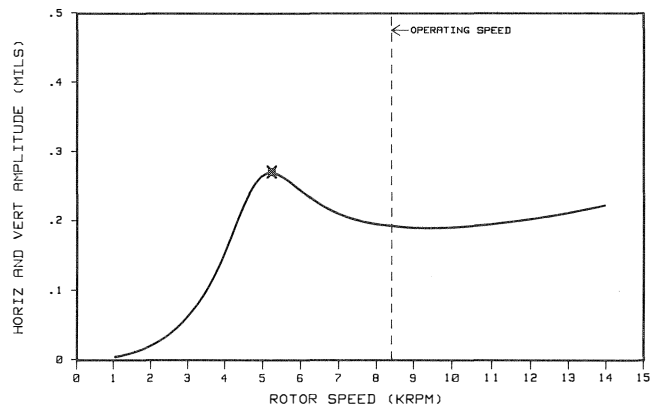


Figure 29a. Synchronous Response Amplitudes vs Rotor Speed at Governor-end Bearing Location. Modified turbine rotor supported on new journal bearings.

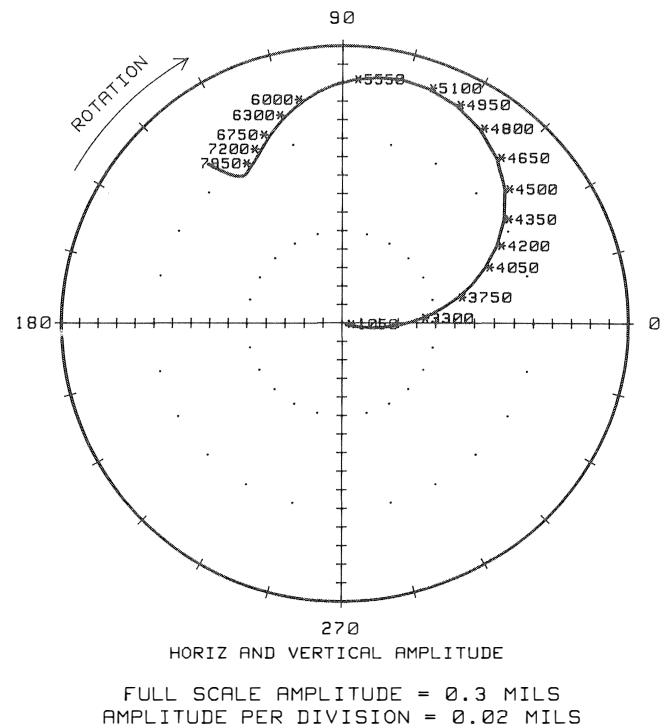


Figure 29b. Nyquist Plot of Response at Governor-end Bearing Location. Modified turbine rotor supported on new journal bearings.

drives through a multiple element, laminated disk-pack instead of gear teeth. The half-coupling weight of the alternate is 36.5 lb (not including the integral shaft flange).

In this analysis, the alternate coupling design yielded rotor vibration characteristics that were very similar to those obtained with the original gear coupling. Therefore, the justification for changing couplings would be that the dry coupling would not be susceptible to drive lockup and unbalance eccentricity due to tooth wear.

SUMMARY AND CONCLUSIONS

A case history of an 8000 hp steam turbine vibration problem and solution was presented. The field vibration data obtained from this unit indicated that it was operating near a critical speed. This characteristic caused ramping vibration levels at the bearings as the unit was brought up to its operating speed of 8400 cpm. Synchronous bearing vibration levels of 3.0 mils to

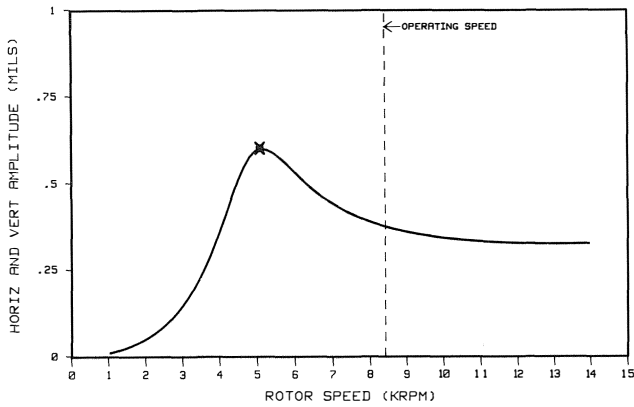


Figure 30a. Synchronous Response Amplitudes vs Rotor Speed at First Stage Wheel Location. Modified turbine rotor supported on new journal bearings.

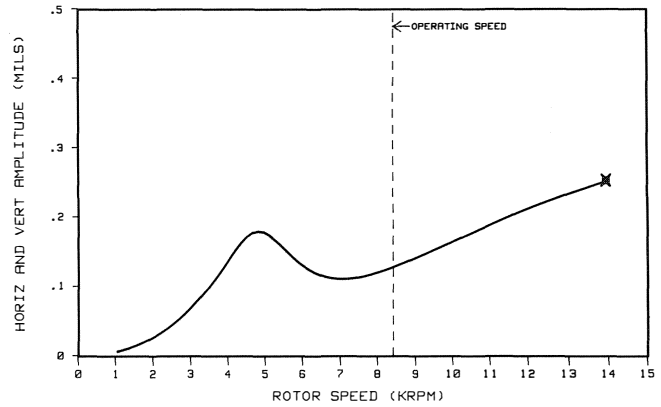


Figure 31a. Synchronous Response Amplitudes vs Rotor Speed at Coupling-end Bearing Location. Modified turbine rotor supported on new journal bearings.

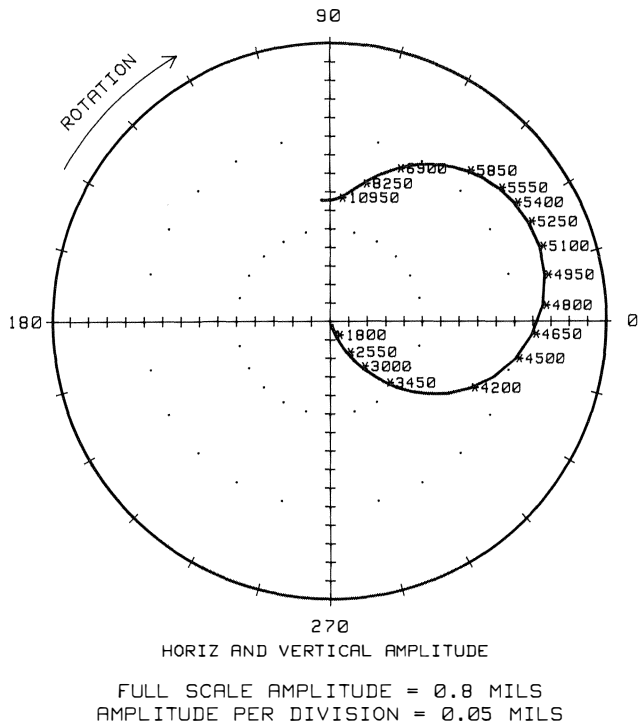


Figure 30b. Nyquist Plot of Response at First Stage Wheel Location. Modified turbine rotor supported on new journal bearings.

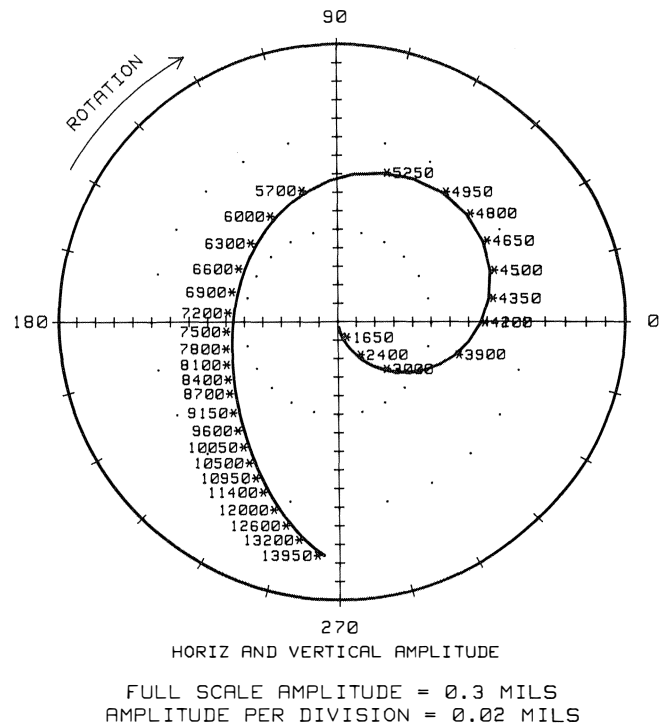


Figure 31b. Nyquist Plot of Response at Coupling-end Bearing Location. Modified turbine supported on new journal bearings.

4.0 mils were recorded. These symptoms caused concern for long-term reliability of this critical piece of turbomachinery, because continuous operation at these high vibration levels could significantly shorten bearing life and, potentially, rotor integrity. A rotordynamics computer simulation of the rotor/bearing system predicted that the second peak response speed range was between 9150 cpm and 9450 cpm in the horizontal and vertical planes. These peaks were within nine percent to thirteen percent of running speed.

Once the simulation identified the problem, it was necessary to determine where the most effective changes could be made in order to eliminate, or at least minimize, this critical speed sensitivity. It should be clarified that the feasible region for solutions is tightly bound, due to cost considerations. Therefore, the number of possible changes is limited. Three alteration possibilities were considered in this analysis:

- Bearing Redesign

- Shaft Modification
- Coupling Redesign

For this machine, the optimum feasible solution was to shorten the shaft at the governor end and replace the bearings. The shaft modification consists of cutting off the governor worm gear assembly and replacing it with an electronic governor disk. This new bearing design is also a four-shoe tilting pad journal bearing, but it has longer pads and higher preload than the existing design. This bearing has a length-to-diameter ratio of 1.0, preload of 0.33, and orientation of load-between-pads. It should be noted that the bearing change improves the vibration characteristics even if the rotor is not modified. However, this choice would be a compromise, because the shaft modification makes an even more significant improvement.

Because there was concern about coupling gear tooth lockup in the original design, an alternate coupling design was evaluated. A coupling change was also of interest, because of its

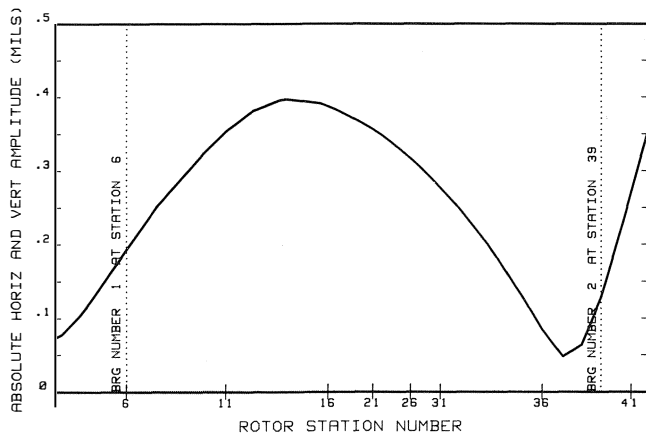


Figure 32. Magnitude of Synchronous Response Amplitude Vectors (Magnitude Only) Along Rotor Shaft for Modified Turbine Rotor Supported on New Journal Bearings. Rotor speed is 8400 cpm.



Figure 33. Lower Half of Governor-end (Outboard) Radial Journal Bearing After One Year of Service.

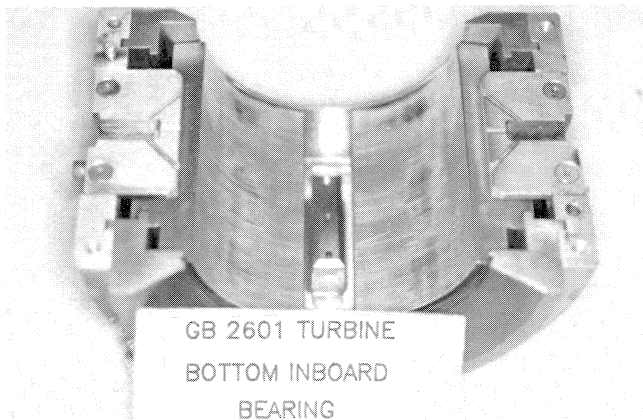


Figure 34. Lower Half of Coupling-end (Inboard) Radial Journal Bearing After One Year of Service.

potential effect on moving the second critical speed. The alternate design is a dry coupling which drives through a multiple element, laminated disk-pack instead of gear teeth. In this analysis, it was determined that the rotor response characteristics were very close with either the existing gear coupling or the dry coupling. Therefore, the justification for changing couplings was that the dry coupling would not be susceptible to

drive tooth lock-up and unbalance eccentricity due to tooth wear.

It should be clarified that the physical solution was approached in two phases. In this case, a significant improvement could be achieved with the bearing change. Therefore, the new bearings were installed on the first shutdown. This change alone reduced field vibration readings by approximately 50 percent. The rotor modification is more involved because of considerations previously discussed. However, an implementation plan is in progress because the rotor simulation indicates even further improvement.

One of the most important points to be emphasized herein was that the analytical study identified degrees of improvement without requiring experimentation on the actual machine. Therefore, the rotor modification was separated into a second phase that could be investigated while the rotor was up and running at full production rates.

As of this writing, the machine has been running for over one year with the new bearing design and the dry coupling. The vibration levels have been well controlled and there have been no production interruptions. Bearing photographs taken during a recent routine unit turnaround are presented in Figures 33 and 34. The scoring marks in the babbitt were made by dirt particles in the oil. However, the bearings were found to be in good general condition after this length of service. In fact, these bearings were reinstalled without changing the pads and are back in operation.

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