PUMP ROTOR CRITICAL SPEEDS DIAGNOSIS AND SOLUTIONS

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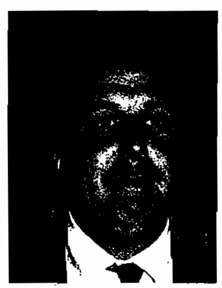
Mr. Marscher holds three patents, is author of structural design and analysis of section of Sawyer's Gas Turbine Handbook, *and was winner of the ASLE Dodson Award, 1983, for best technical paper by a member under 35 years of age.*



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Mr. Reid's current position is with the Production Operations Technical Staff of the Delmarva Power and Light Corporation. His major responsibility with the utility is to improve station availability by solving generic rotating machinery problems. This includes troubleshooting, analysis, and if necessary, procurement of upgraded equipment.

Mr. Reid is a member of ASME and the Vibration Institute. He holds two patents and has authored a number of technical papers in the areas of design and analysis of power generation equipment.

ABSTRACT

Delmarva Power Indian River Power Station's Unit 4 has had boiler feed pump reliability problems since startup in 1980. The pump has been rebuilt nine times with post overhaul operability ranging from 30 minutes to 16 months. Many of the failures have been attributed to high shaft vibration. The analysis techniques used to diagnose and develop solutions to the chronic rotor critical speed problems are described.

The analysis techniques described are general in nature and can be applied to all rotating equipment, but are particularly well suited to high speed multistage pumps where some rotor "bearings" that determine the rotor behavior are difficult to characterize. There have been many papers written concerning

pump rotordynamic computer modelling [1, 2, and 3] and modal analysis [4] of pump rotors, but there is no published case study where the computer modelling and field modal testing have been combined to diagnoses and develop solutions to a real problem. The combination of the two methods provides a powerful tool that accurately represents the behavior of the rotor and allow designers to engineer proper solutions without trial and error experiments on the pump.

INTRODUCTION

"The Indian River Power Station consists of four coal-fired units. Unit 4, the largest, produces approximately 403 MW at full load. Replacement energy costs for each week of downtime can exceed one-half million dollars" [5]. Delmarva Power estimates the main feed pump to be responsible for up to two percent of the plant's unavailability and is, therefore, considered a critical component. Specific recurring problems included; suction impeller cavitation damage (recirculation type), intermediate stage cover diaphragm wall cracking, bearing Babbitt fatigue (drive end bearing only) and rapid increase of clearances at the annular stage sealing surfaces (impeller wear rings, hydraulic thrust balance sleeve and bushing and shaft seals). During the life of the pump, the power company employed no less than three industry consultants to diagnose the pump problems and suggest solutions. In 1988, EPRI offered the electric power industry's fossil fuel fired power plants the opportunity to become host sites to demonstrate newly developed pump technology [6], Delmarva was one of the first in line to take advantage of the opportunity.

Subsequent to the site interview and pump component review, the EPRI Technology Demonstration Expert Panel recommended that vibration and modal analysis of the pump be performed. Delmarva contracted with a vibration measuring instrument manufacturer to provide, install and calibrate accelerometers to monitor the absolute movement of the bearing housings and proximity probes to monitor shaft movement relative to the bearing housings. W. D. Marscher, Dresser Pump Division, was contracted to perform the modal analysis of the pump case, rotor and foundation. The manufacturer's engineers performed computer modelling of the pump rotor (concurrent with the power company's engineers). Results of the field modal analysis were compared with results of computer models of the pump rotor then the assumptions used to develop the computer models were reviewed and modified until the computer model results behaved the same as the real rotor measured in the field.

PUMP DESCRIPTION AND OPERATION

A cross section is shown in Figure 1 of the pump which is a four stage (single suction first stage, two intermediate stages and a small kicker stage for desuperheat spray) machine designed in 1976 by the manufacturer. Steam turbine driven, the pump is rated at 7200 gpm at 7828 ft total discharge head, while operating at 5600 rpm (approximately 16000 BHP). Initially designed with floating ring type shaft seals, the pump was modified to use fixed bushing type seals in 1987. There were no 'special' design features peculiar to this particular machine. The problems encountered are apparently the product of small, seemingly unrelated evolutionary changes normally incorporated during the course of custom engineered product design. The performance tests in the manufacturer's plant were run at full speed and full load and do not indicate any significant vibration signatures (vibration data recorded during performance testing are maintained on magnetic tape just for such purposes). The differences between the factory tests which showed no signs of vibration problems and the poor field results could be attributed to changes in the internal rotor alignment (all pumps of this size are realigned after installation) or lube oil viscosity. However, there is no specific evidence of significant differences between the factory and site conditions which leaves a bit of mystery to be solved.

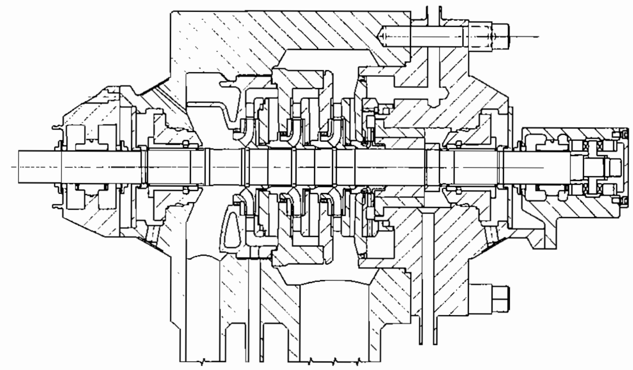

Figure 1. Boiler Feed Pump Cross Section.

The unit swings in load from 158 MW to 403 MW, using variable speed to control the flowrate. There is a 30 percent capacity electric motor driven startup pump normally used to bring the unit online. However, there are provisions to use steam from another unit to drive the main boiler feed pump turbine and pressurize unit 4's boiler during startup.

During normal operation, the main feed pump speed varies between 4600 and 5750 rpm. During low load operation, a minimum flow of 3000 gpm is maintained by a recirculation valve. The unit was designed as a base load unit, but in the last several years it has been cycled down at night, and last year, the minimum flow was increased from 1800 gpm to the 3000 gpm value to avoid the recirculation damage found on the suction stage.

COMPUTER MODEL OF PUMP ROTOR

The main feed pump rotor geometry is shown in the rotor sketch (Figure 2) and the associated physical properties are tabulated in Table 1. All major components have an interference fit onto the shaft (impellers have relatively tight fit near the split ring and light interference fit at the hub end to minimize friction and its destabilizing effect). All close clearance annular seals have multiple helical grooves on both the rotating and stationary surfaces. Bearing coefficients for the oil lubricated bearings and the close clearance annular seals are tabulated in Table 2.

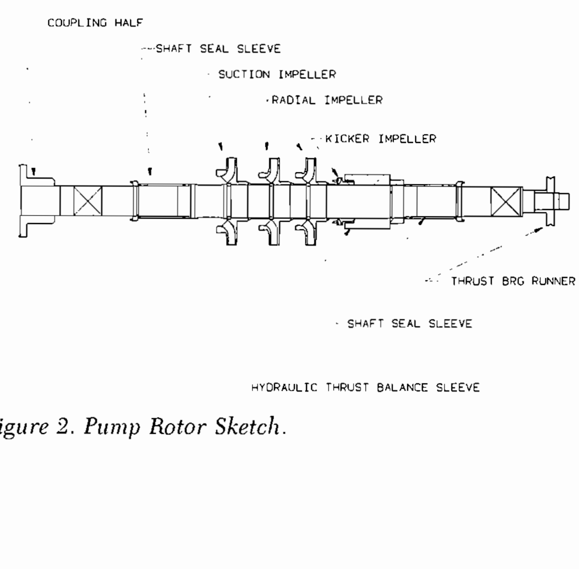

Figure 2. Pump Rotor Sketch.

Table 1. Rotor Component Data.

Component	Weight (LBS)	Moment of Inertia (LB-FT*FT)
Coupling Half	112.70	12.44
Shaft	730.00	21.25
Shaft Seal Sleeve	18.50	1.16
Suction Impeller	56.00	11.48
Radial Impeller	65.00	13.30
Kicker Impeller	14.60	1.28
Hydraulic Thrust Balance Drum	112.60	14.00
Thrust Bearing Runner	23.70	1.97

Table 2. Bearing Coefficients, Computer Code Calculation Results.

Component	Stiffness Coefficients lbf/in			
	Kxx	Kxy	Kyx	Kyy
Journal Bearing (Drive End)	1109000	2449000	-2652000	532200
Journal Bearing (Non-Drive End)	1028000	2426000	-2621000	498300
1st Stage Impeller Wear Ring	430800	90478	90478	430800
Normal Stage Impeller Wear Rings	385400	82426	82426	385400
All Impeller Hub Wear Rings	53185	1332	1332	53185
Hydraulic Thrust Balance Drum	1499000	5575000	5575000	1499000

Component	Damping Coefficients lbf-sec/in			
	Cxx	Cxy	Cyx	Cyy
Journal Bearing (Drive End)	9290	-1871	-1877	8748
Journal Bearing (Non-Drive End)	9107	-1745	-1751	8705
1st Stage Impeller Wear Ring	320	N/A	N/A	320
Normal Stage Impeller Wear Rings	291	N/A	N/A	291
All Impeller Hub Wear Rings	85	N/A	N/A	85
Hydraulic Thrust Balance Drum	19723	N/A	N/A	19723

The computer program used to model the pump rotor was the APDS finite element solver [7 and 8] with pre and post processors written by the manufacturer to simplify data entry and data output interpretation. Results which include Campbell diagrams, (Figure 3) and residual unbalance response (Figure 4) show the rotor to be stable. The bending mode shapes are shown in Figure 5 with an associated logarithmic decrement (log dec) value. The log dec is an indicator of damping, values equal to or less than zero are considered unstable. In each of the solution boxes of Figure 5 there are two curved lines representing the shape of the shaft. The solid and dashed lines represent the vertical and horizontal, respectively, projections of the shaft shape. Also, to aid interpretation a small rotor sketch with vertical lines indicating the position of the journal bearings has been added. When comparing the residual imbalance response amplitudes at various stations to the mode shapes given in the stability analysis, good correlation is found. This is considered a basic check because the results come from independent computer codes.

Campbell diagrams provide a visual tool to evaluate critical speeds by plotting the damped natural frequency *vs* running speed. As the pump speed changes, the bearing characteristics of the oil lubricated bearings and the close clearance annular seals change and result in a family of damped natural frequen-

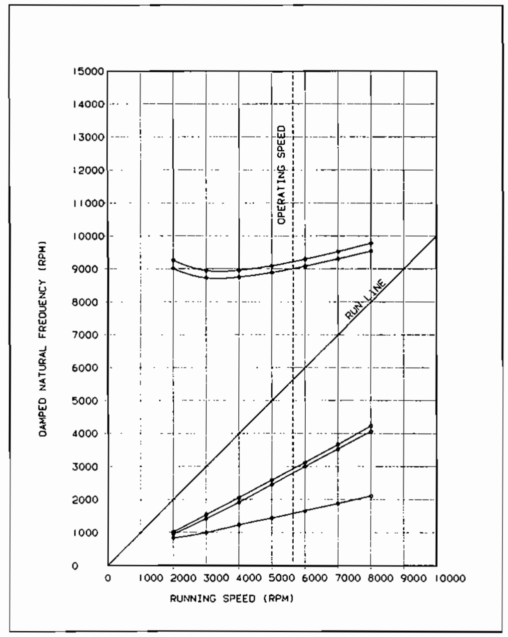

Figure 3. Campbell Diagram, "Stiff Drive End Bearing."

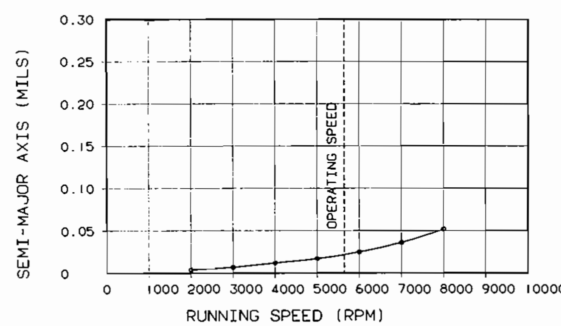

Figure 4. Rotor Mode Shapes at Incremental Speeds, "Stiff Drive End Bearing."

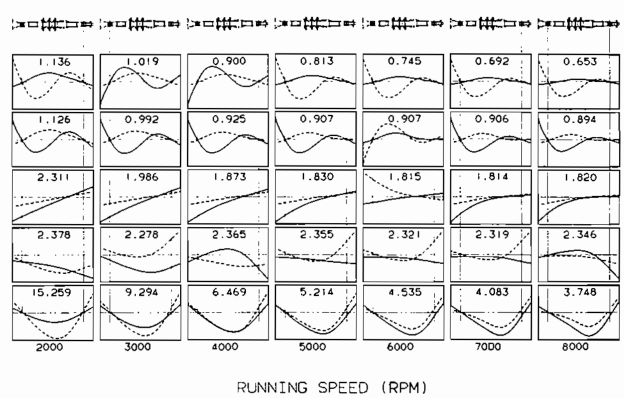

Figure 5. Residual Unbalance Response, "Stiff Drive End Bearing."

cies. The discrete damped natural frequencies have been connected at incremental running speeds of a common mode shape (first through fifth bending modes) with a 'mode' line to ease interpretation. If there is an intersection of a bending mode with the 'run line,' a review of the log dec will determine if the intersection will result in a significant critical speed or a damped critical speed. The authors' experience is that most computer model rotor analyses for pumps are not absolutely accurate but they are good trend indicators and show sensitivity to changes in bearing characteristics and/or geometry.

The computer model of the original rotor shows no 'critical speeds' (modes) near running speed and the system to be stable. There is one pair of lightly damped modes at 9500 cpm and one pair of critically damped modes at 2700 cpm and a single mode at 1500 cpm at the running speed of 5300 cpm.

The results of the Texas A&M's 'APDS' solution were compared to the EPRI FEATURE and University of Virginia 'ROMAC' [9, 10] codes and found to be very consistent (both the FEATURE and APDS codes use finite element solvers while the ROMAC code is a transfer matrix solution). The APDS results are shown herein because of superior post processor graphics available.

INITIAL FIELD MODAL ANALYSIS

The location and type of vibration instrumentation shown in Figure 6 was used to gather the various vibration data. A specially designed 12 pound hammer applied constant artificial excitation force at all frequencies from zero to vane passing (40,000 cpm) and the pump was instrumented to detect these frequencies. This information was used to develop "transfer functions" (vibration at a given location divided by known impact excitation at another given location) [4].

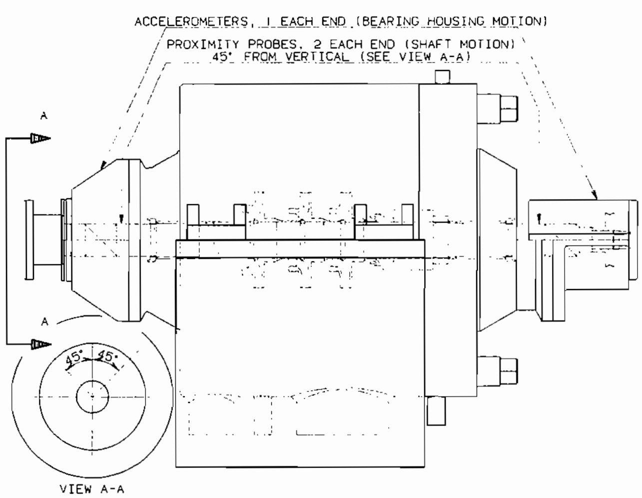

Figure 6. Vibration Instrumentation used for Modal Analysis.

There were two types of "transfer function" tests. The "driving point" (Modal DP) tests where the impact and vibration measurement were at the same location and in the same direction. These tests were used to determine those natural frequencies which were dominated by the mass and stiffness near the chosen test points and, hence, to locate and sort out "rotor modes," "housing modes," "casing modes," and "baseplate/foundation modes." The second type of tests, "modal," are where the impact and measurement points were different. Because only two measurement points were available on the shaft, these tests were done only to determine mode shapes and system damping of structural (non-rotating) natural frequencies.

All results were obtained while the pump was operating, in particular the rotor modal DP tests. This allowed both the bearing and close clearance seal (impeller wear rings, hydraulic thrust balance sleeve and bushing and shaft seals) stiffness, cross coupling and damping to be at their actual values for the pump rotor being tested. To perform such tests while the pump is running requires special proprietary signal processing techniques. These techniques ensure that the natural excitation response spectrum does not overwhelm the much weaker modal response spectrum.

Modal analysis data is shown in Figures 7 and 8. All spectrum plots are in dB:

$$dB = \log 10 \text{ (vibration units)}$$

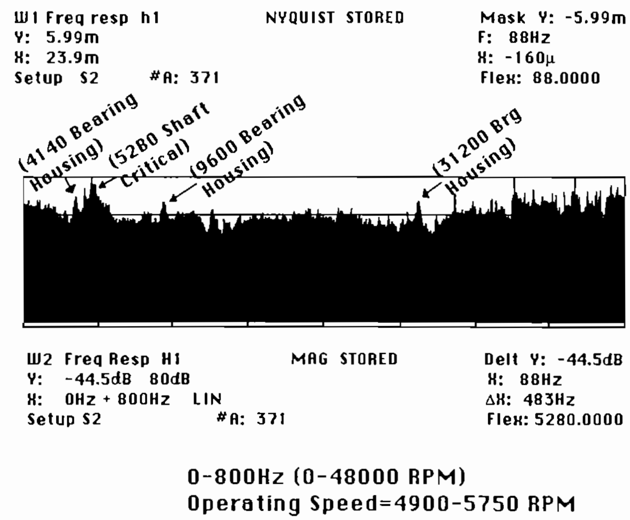

Figure 7. Drive End Bearing Housing Modal Analysis Frequency Response Plot.

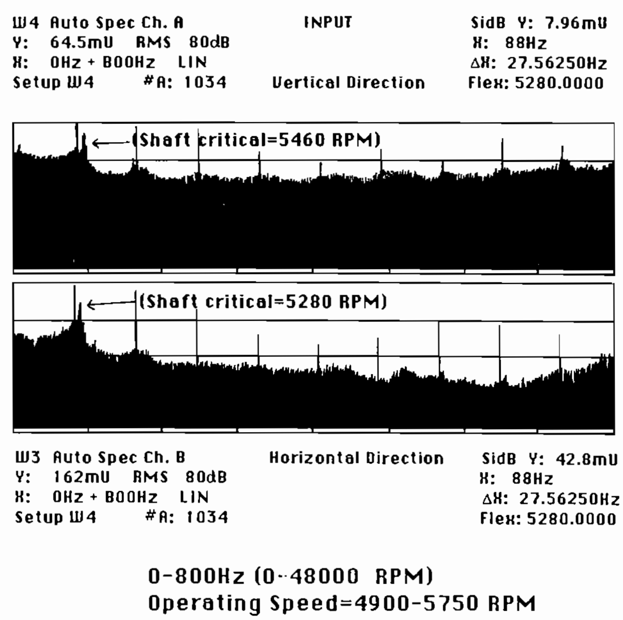

Figure 8. Shaft Modal Analysis (at Drive End) Frequency Response Plot, Prior to Bearing Modification.

Although dB is more difficult to interpret than linear vibration, it provides a focus on broad band low level vibrations along with the narrow band multiples of running speed (the harmonics). Relative peaks in the smooth broad-band or "floor" of the dB vibration *vs* frequency spectrum are indications of natural frequency or hydraulic forcing frequency values, particularly when observed over a range of load. It should be noted that the modal analysis was performed shortly after overhaul and the wear clearances were essentially "new."

The results are shown in Figure 8 of the Modal DP test of the drive end of the shaft in the horizontal direction, indicating that the first underdamped shaft natural frequency occurs at 5280 cpm. The natural excitation load cycle tests verify that this is a natural frequency and not a hydraulic force (hydraulic forces track at a constant percentage of speed). The tests also verify that the natural frequency is associated with the shaft (primarily the drive end), and not the stationary structural components. Modal analysis curve-fit of the 5280 cpm shaft mode indicated its damping value was only 1.8 percent, which would result in an amplification factor of about 30, which is five to 10 times higher than typical shaft bending modes in horizontal centrifugal pumps.

There is a second underdamped shaft natural frequency shown in Figure 8 at 13020 cpm and indications of two critically damped shaft mode pairs centered near 1500 cpm and 2950 cpm, and a possible heavily damped shaft mode at 31200 cpm.

Although the structural responses were interesting and are being addressed with structural changes, they are beyond the scope of this study.

COMPUTER MODEL MODIFICATION

In addition to the computer rotordynamic analyses described previously, approximate manual "beam on an elastic foundation" analysis was performed to provide ball park verification of the numerical results, and to better understand the parametric influences of the shaft *vs* bearing *vs* annular seal "Lomakin" stiffness.

Referring to Figure 3, there is no reasonable situation that can be found which would increase the first (lowest) or the second mode shape line to intersect the run line at the running speed of 5300 cpm as these modes are controlled by the close clearance annular seals and are insensitive to radical changes in the bearing stiffness and damping coefficients (confirmed by parametric studies). Nor can the fourth or fifth mode shape line be reduced to intersect the run line at 5300 cpm, because these mode shapes are dominated by the rotor geometry (again, confirmed by parametric studies). However, the third mode shape line, which corresponds to the cantilever type mode shape shown in Figure 5, can be made to move about by changing the drive end bearing stiffness and damping coefficients. It is significant that the third mode is a cantilever shape. Measurement of shaft vibration is made near the journal bearings and all mode shapes except the cantilever have very limited deflections at the drive end journal bearing. The location of the proximity probes near the bearing, where the deflection is minimal for most bending modes, provides sufficient information to evaluate most rotordynamic problems associated with pumps. Only with specially designed pumps and proximity probes can the shaft vibration within with pressure casing be measured.

Movement of the third mode (cantilever mode) perfectly matches the modal test data. It is reasonable (in hindsight) that this situation could occur because, as shown by the manual analysis, the shaft is not flexible enough to prevent a triple bearing situation. In such a situation, unless radial loads are fairly symmetrical and the shaft/casing alignment is very good, two of the bearings will dominate, and will unload the third. The sum of the direct stiffness of the close clearance seals is approximately twice that of a single oil lubricated bearing (Table 2) and is quite capable of supporting the entire rotor weight with a deflection of less than 0.001 in. It is not difficult to imagine the rotor pivoting about the balance sleeve and unloading the inboard bearing reducing the load and, consequently, reducing the direct stiffness and damping allowing the natural frequency to intersect the run line at 5300 cpm.

There is an infinite number of combinations of bearing stiffness and damping values that would allow the cantilever mode shape to intersect the run line at 5300 cpm (Figure 9). This set of curves was developed from a matrix of computer model solutions where the drive end journal bearing stiffness and damping values were varied. The 100 percent values were calculated using the assumption that bearing load was the simple static load of ½ the rotor weight. Since bearing coefficients are a function of many factors, including bearing load, the researchers were keen on finding the effect of the more general bearing stiffness and damping terms on the cantilever mode shape's damped natural frequency. This would show the sensitivity of the rotor system to fundamental bearing characteristics regardless of the source of inaccuracy; bearing load assumptions, manufacturing error, oil properties assumptions, or even bearing characteristics calculation error.

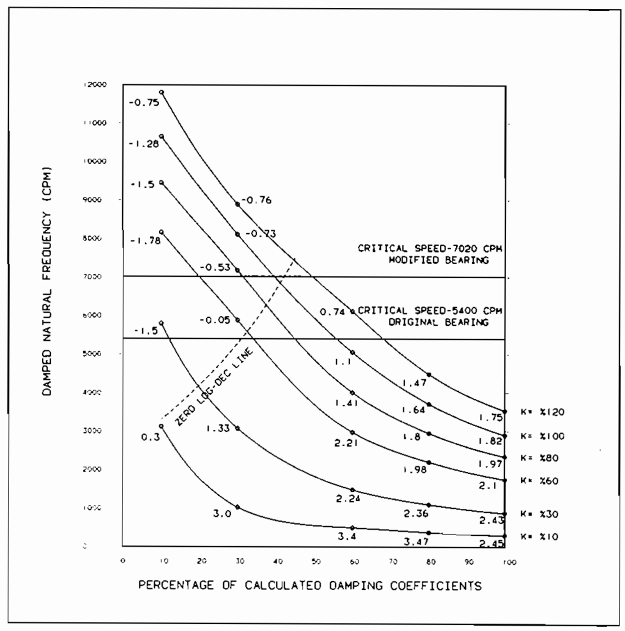

Figure 9. Damped Natural Frequency of the Cantilever Bending Mode vs Journal Bearing Damping, with Isostiffness Lines.

The reasons for the change in damped natural frequency with a small change in bearing characteristics is not intuitive but it is consistent with various computer aided analysis codes and an approximate manual "beam and an elastic foundation" analysis.

Again, referring to Figure 9, the log dec value is noted on the isostiffness lines. The log dec value allows better insight of the rotor response to the various stiffness/damping combinations causing the cantilever mode intersection with the run line in Figure 10. While there are many combinations that can cause the intersection only the combinations with very low or negative values of log dec will result in high amplitude vibration. The dashed line in Figure 9 represents the boundary of stability (values of 'zero' log dec).

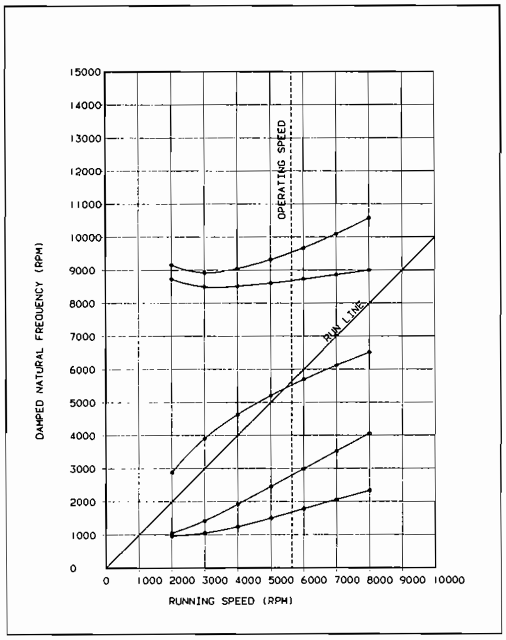

Figure 10. Campbell Diagram, "Soft Drive End Bearing."

The stability boundary is helpful in reducing the range of combinations of bearing stiffness and damping that can produce calculated results that match measured modal analysis results. If the assumption is made that a log dec equal to or less than zero is required to produce a 'critical' speed at 5300 cpm, the maximum bearing stiffness terms of the real drive end journal bearing are no greater than 60 percent of the calculated values listed in Table 2, and the maximum damping terms are no greater than 33 percent.

Rotor analysis results using these reduced stiffness and damping values for the drive end oil lubricated journal bearing show significant reduction in log dec for the cantilever mode shape (Figure 11), and 10 times the shaft deflection calculated at the drive end bearing (Figure 12), relative to the original rotor analysis.

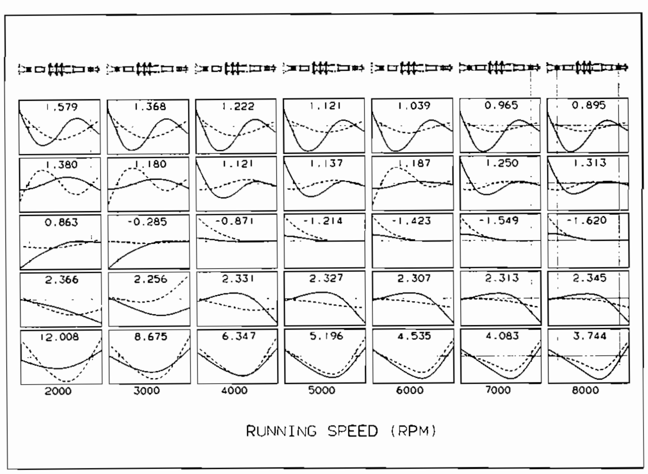

Figure 11. Rotor Mode Shapes at Incremental Speeds, "Soft Drive End Bearing."

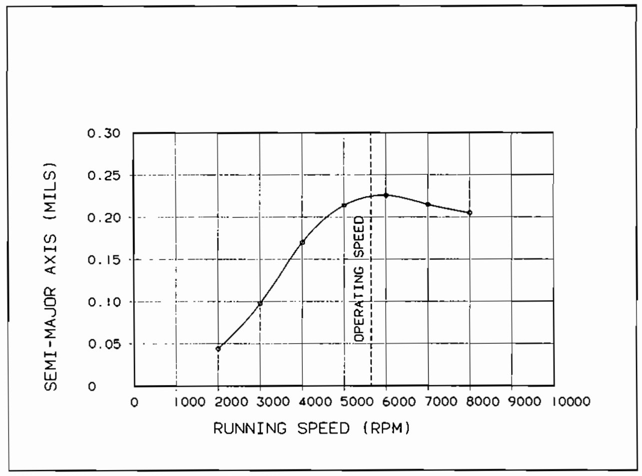

Figure 12. Residual Unbalance Response, "Soft Drive End Bearing."

JOURNAL BEARING REDESIGN

The original oil lubricated journal bearing was a simple pressure dam type design that has been in use for 30 or 40 years. The purpose of the pressure dam is to provide a self induced load on the bearings to maintain stability as plain journal bearings are subject to oil whirl when unloaded (pumps of this type are usually designed with lightly loaded bearings). Apparently, in this instance, the hydraulic forces within the pump were strong enough to support the rotor weight and counter the force generated by the pressure dam.

The original bearing design is depicted in Figure 13. Review of the geometry revealed that the pocket was deeper than modern designs by a factor of about 3. The bearing was redesigned with the following goals in mind:

- Provide increased stiffness terms
- Provide increase damping terms
- Provide strong self induced directional load
- Fit into existing bearing housing

The redesigned journal bearing is shown Figure 14. The pressure dam pocket depth was reduced to 0.010 to 0.012 in and a strip of material was removed from the lower surface to reduce the L/D ratio to 0.62. The reduced pocket depth should increase

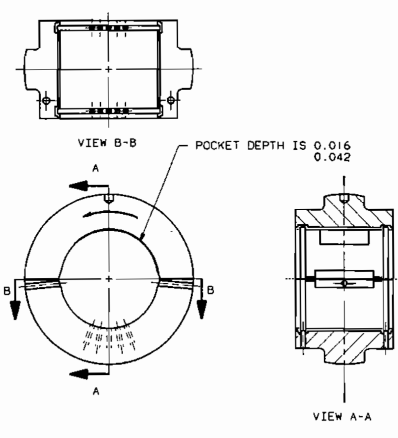

Figure 13. Original Pressure Dam Journal Bearing Design.

the self induced directional load and the reduced L/D should increase the direct stiffness and damping values. The researchers chose not to use a tilting pad bearing in this instance because of concern about the loss of crosscoupled terms (both stiffness and damping) which seems to drive the log dec results to zero.

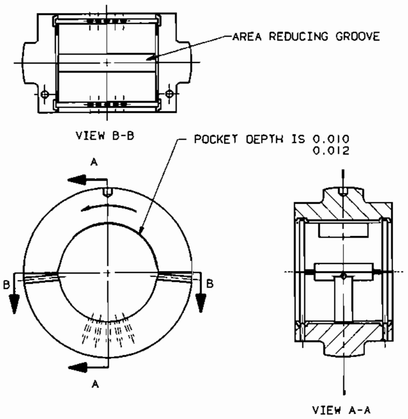

Figure 14. Redesigned Pressure Dam Journal Bearing.

SECOND FIELD MODAL ANALYSIS

Subsequent to the installation of the redesigned journal bearing a second modal analysis was performed to confirm that the "critical speed" was in fact removed from the operating speed range. This second modal analysis was completed four months after the initial modal analysis and there was concern about a change in wear clearances within the pump. However, the pump was disassembled six months later (10 months after the initial modal analysis) and the clearances were found to have very little wear and confirmed that the only difference between the initial and second modal analyses was the bearing design. The second modal analysis spectrum plots shown in Figure 15

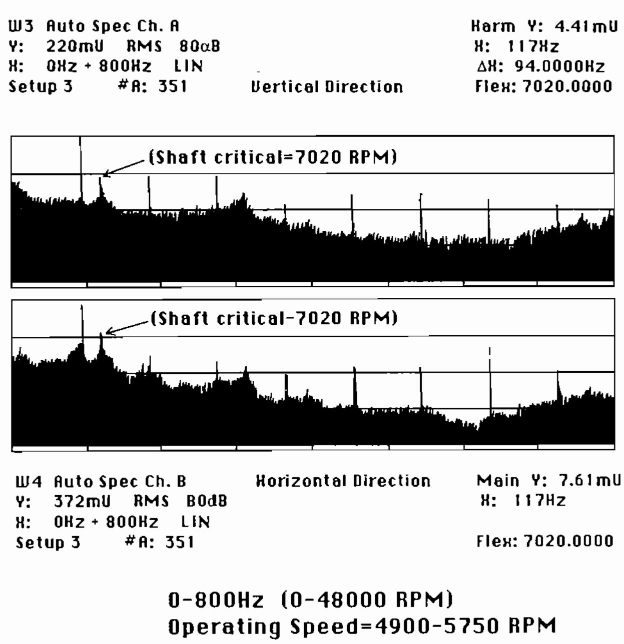

Figure 15. Shaft Modal Analysis (at Drive End) Frequency Response Plot, Subsequent to Bearing Modification.

clearly demonstrate that the 5300 cpm mode has moved to about 7020 cpm, well out of the operating range, while the two lower modes remained unchanged as predicted by the rotordynamics analysis. The redesigned bearing moved the cantilever mode shape damped natural frequency near the intersection of the zero log dec and 100 percent stiffness lines shown in Figure 9. Again, there are many combinations of stiffness and damping that would cause the cantilever bending mode damped natural frequency to intersect the run line at 7020 cpm. Modal test data analysis [4] results in a log dec of approximately 0.15 and allows an estimate of the drive end journal bearing stiffness and damping values to be about 105 and 40 percent, respectively, of the values listed in Table 2.

CONCLUSIONS

By working as a team with a common goal the utility and the manufacturer were able to identify and solve a long term problem. The tools for vibration modal analysis are now developed sufficiently to identify pump rotor critical speeds during operation and allow computer modelling codes to be calibrated and used efficiently to redesign stable rotors.

Redesign is a key word. This study is representative of how a computer model can be calibrated using modal analysis; however, during the initial design of a pump there is no rotor available for modal analysis. It is necessary for pump designers to perform parametric sensitivity analysis of the computer models to determine the significant variables and ensure that the design provides a margin for error. Of course, experience with a family of designs will make this task easier, but until automated and coordinated rotor analysis codes become available, this will be a very time consuming and expensive effort.

Although the elimination of the rotor critical speed problem was an important task there are several others just as important to completely resolve the poor availability at the Indian River Power Station. Hydraulic and structural changes have been made to reduce high frequency vibrations measured on the drive end bearing housing and to eliminate high cycle fatigue cracking in the interstage cover diaphragms. The authors intend to publish these extremely interesting and pragmatic solutions.

There is substantial evidence that there are many pumps that can benefit and provide increased plant availability through application of the emerging pump rotor analysis technology. However, it takes open cooperation between the user and the manufacturer to become objective and develop effective solutions to real problems.

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