A Simple Procedure for Assessing Rotor Stability

BY

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ABSTRACT

This paper describes a procedure for assessing rotor stability using perturbation tests in an at-speed balancing machine. The equipment required to perform the tests is readily available and could be used at most at-speed balancing facilities. Additional time required over and above normal balancing time is minimal. Results of using this procedure for evaluating a five stage, 2800 hp hydrogen recycle compressor are presented. Comparisons of calculated and measured stability characteristics are made using the concept of "mechanical logarithmic decrement" (δ_m). Three different bearing configurations were tested. The test data clearly showed the sensitivity of rotor stability (system damping) to rotor speed as well as bearing design parameters. The measured results were used to optimize bearing parameters (clearance and preload) to improve rotor stability.

INTRODUCTION

The evaluation of rotor stability has become an important aspect of turbomachinery rotor design. Self excited rotor instabilities have been responsible for millions of dollars of downtime, lost production, and maintenance costs as rotor performance (speed, horsepower, pressures, etc.) continues to increase [1, 2]. High speed centrifugal compressors are particularly susceptible to destabilizing effects from oil ring seals, impeller-diffuser interaction, fluid-film bearings, balance piston labyrinths, etc. Although the state-of-the-art in rotor stability calculation has progressed to a high degree of sophistication in recent years, there are still cases when rotors exhibit stability characteristics that differ from predicted behavior.

Rotor stability is usually expressed in terms of the system "logarithmic decrement" or log dec. The log dec is essentially a measure of system damping. Lund [3] described a method for calculating the log dec for a general flexible rotor in fluid-film journal bearings, considering internal shaft damping and destabilizing aerodynamic forces. Others [4–7] have adapted this method for use with industrial machinery and most turbomachinery manufacturers typically perform rotor stability calculations in the design stage of high speed rotors.

The theoretical details concerning rotor stability can be found in the literature. The basic concept involves solving the equations of motion for the rotor-bearing system and computing the damped natural frequencies. The damped natural frequency or *eigenvalue* is a complex quantity of the form

$$s = \lambda + i\omega \tag{1}$$

where: λ = damping exponent
ω = undamped natural frequency

The logarithmic decrement (δ) is then defined as

$$\delta = \frac{-2\pi\lambda}{\omega} \tag{2}$$

Further, the log dec is related to the system "Q" or amplification factor (AF) according to the following equation

$$\delta = \frac{\pi}{AF} \tag{3}$$

where: $AF = \frac{1}{2\zeta}$
$\zeta = \frac{c}{c_c}$ = damping ratio

For values of $\lambda < 0$ ($\delta > 0$), the free vibrations will be damped out with time, Figure 1, and the system is considered stable. For values of $\lambda > 0$ ($\delta < 0$), the free vibrations will increase with time, Figure 2, and the system is considered unstable.

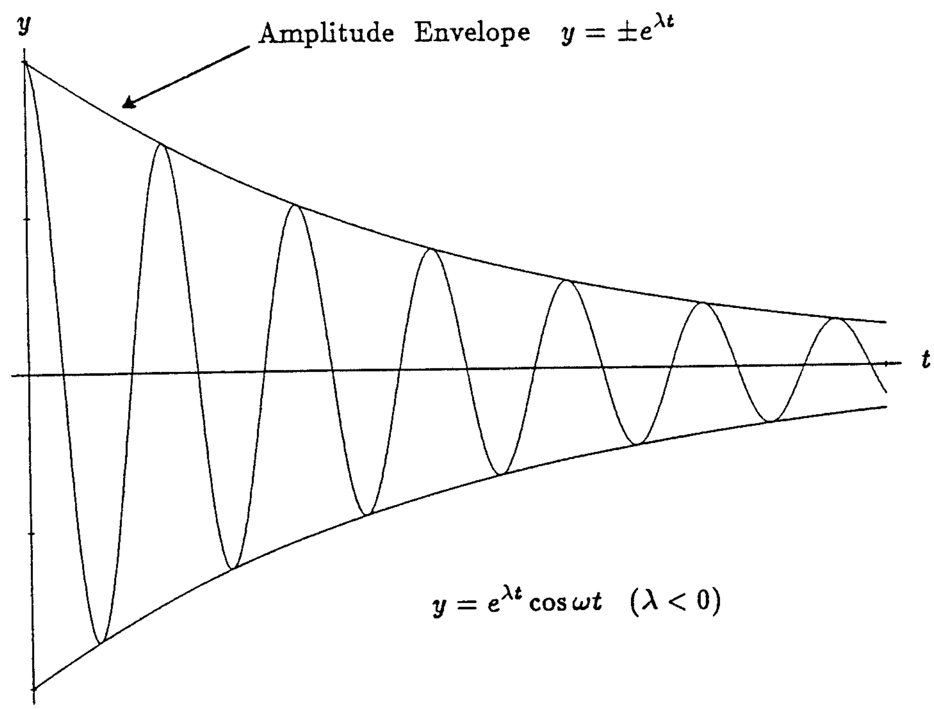

FIGURE 1. Stable System

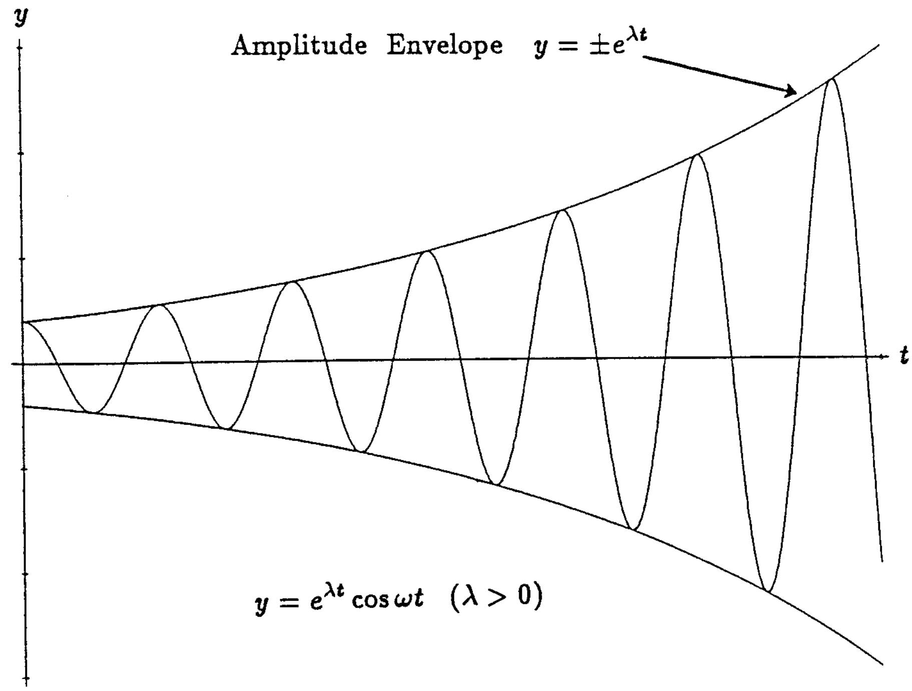

FIGURE 2. Unstable System

In practice, a rotor is modeled as a series of masses connected by springs and supported at various locations by springs and dampers, Figure 3. The stiffness, damping, and mass matrices used to formulate the eigenvalue problem include all effects of added masses (impellers, balance piston, couplings, etc.), bearing and seal coefficients (stiffness and damping as well as cross-coupling), fluid dynamic loading, internal friction, etc. Of these considerations, the rotor mass and stiffness properties are the best understood and most accurately predicted. Fluid film bearings are also modeled with good accuracy. Rotor-stator interaction at interstage seals, impeller-diffuser areas, oil ring seals, and balance pistons are the subject of continuing research. Methods do exist [8–11] to handle these components, but the degree of accuracy is somewhat less than desired. Further complications involve rotor-stator eccentricity effects and internal shaft friction.

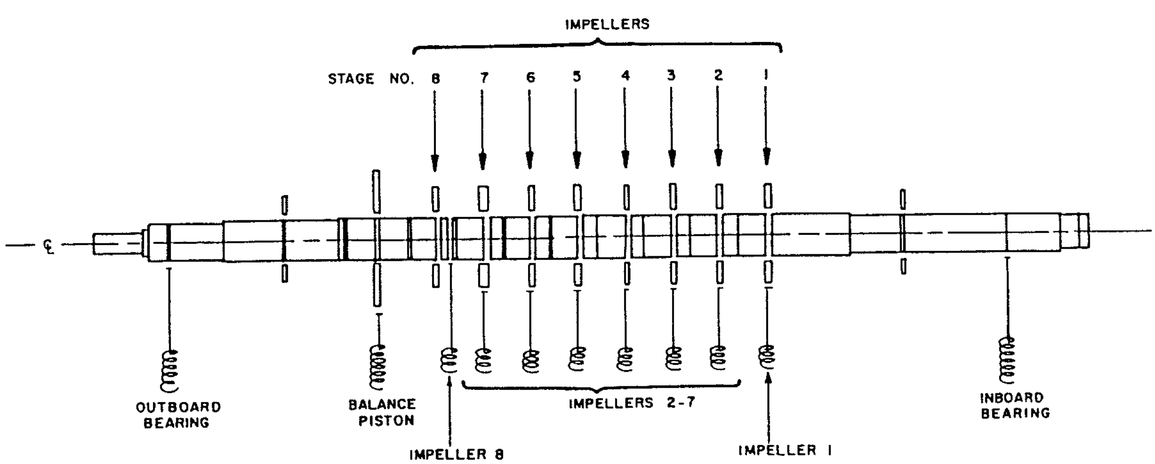

FIGURE 3. Schematic of Rotor Model for Multistage Industrial Machine

In cases where the amount of aerodynamic loading is unknown, the log dec is mapped versus aero loading to determine the sensitivity of the rotor design to this parameter. An example stability map for a light gas recycle compressor is shown in Figure 4. For this rotor design, the system stability was relatively insensitive to aerodynamic loading, since the log dec did not significantly reduce as the loading increased.

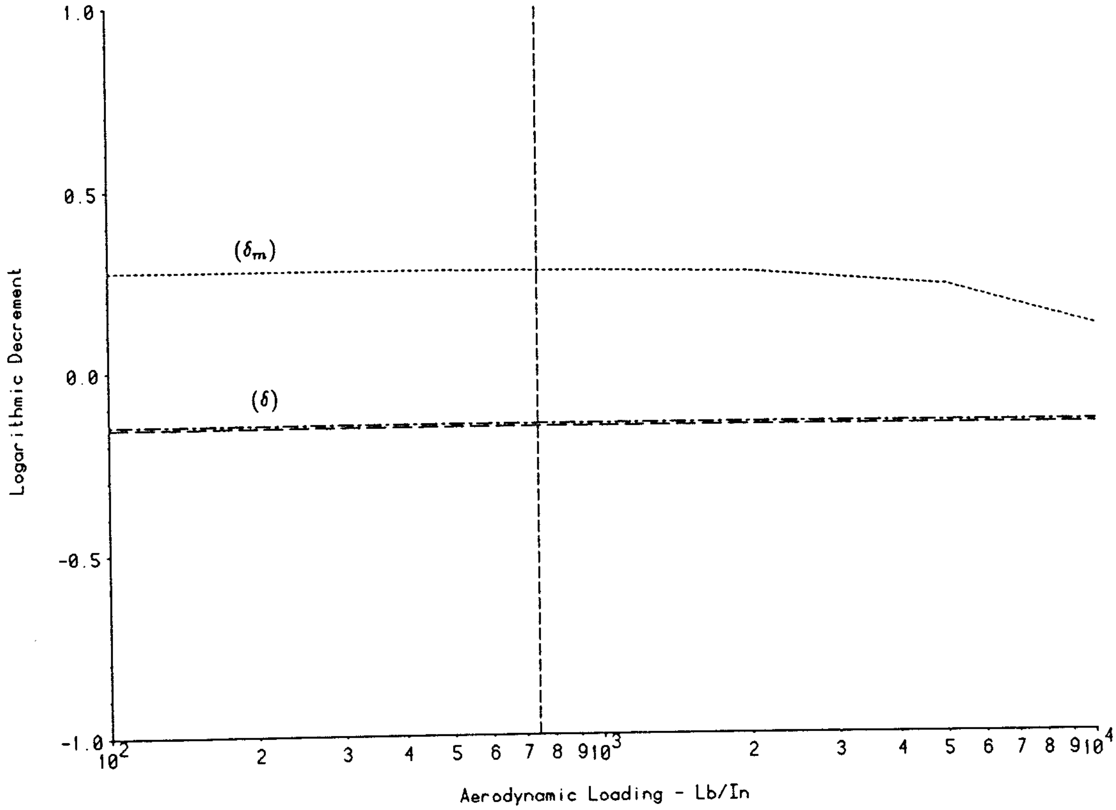

FIGURE 4. Hydrogen Recycle Compressor Plot of First Forward Whirl Mode vs Aerodynamic Loading

With all the expected system effects included, the log dec is usually plotted versus speed to determine the "stability threshold ", or the speed at which the log dec becomes negative. A typical plot of log dec versus speed is shown in Figure 5.

It is often desirable to consider the rotor stability (versus either speed or aerodynamic loading) from two standpoints. First, since the rotor and bearing models are best understood, the stability can be calculated ignoring all other effects. Second, the other destabilizing mechanisms can be applied. If the rotor-bearing system alone is only marginally stable, then major modifications such as bearing span or shaft diameter changes, addition of squeeze-film dampers, etc. are generally required. If seal effects, balance pistons, or aerodynamic loading dominate the stability characteristics, then the component causing the system to be unstable can often be redesigned. For the example plot of log dec versus speed shown in Figure 5, the oil seals dominate the stability characteristics, indicating that design modification efforts should be focused in the seal area.

To account for the uncertainties, the rotor designer typically aims for a positive log dec on the order of 0.3 for the basic rotor-bearing system and attempts to maintain a positive log dec of 0.1 or greater throughout the rated speed range when all destabilizing mechanisms are considered.

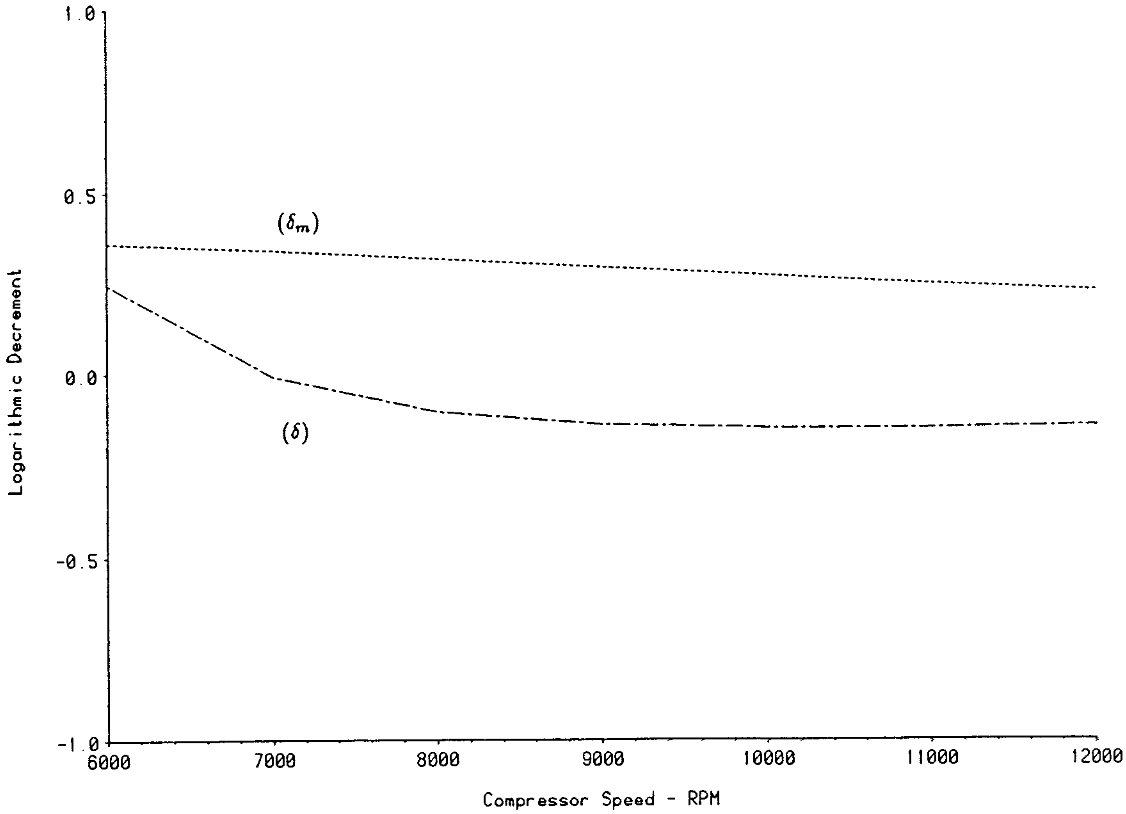

FIGURE 5. Hydrogen Recycle Compressor Plot of First Forward Whirl Mode vs Speed

For the purposes of this paper, the term δ_m (i.e. *mechanical* log dec) refers to the basic rotor-bearing stability parameter and δ refers to the rotor stability with all destabilizing effects included. The following sections describe a particular rotor stability problem. The rotor's history is presented, the stability analysis is discussed in terms of both δ_m and δ, and a test procedure for measuring the δ_m of the rotor is described. Comparisons of calculated and predicted δ_m are made and the benefits of measuring δ_m are also discussed.

BACKGROUND

History

A 2800 hp, 10,200 rpm, five-stage hydrogen recycle compressor in a Gulf Coast refinery was operated with reasonable success for several years in the 7500–9500 rpm speed range. Higher unit charge rates required higher flow from the recycle compressor and continuous operation near 10,000 rpm. Although the compressor design speed was 11,000 rpm, the unit experienced excessive subsynchronous vibration levels at the higher speeds. The high vibration levels resulted in shaft end oil seal failures. This limited the speed range and thus the compressor flow rates. Detailed field vibration measurements were made and a comprehensive rotordynamics analysis was initiated to characterize the problem and develop a solution.

It was concluded from the analysis and field testing that a subsynchronous instability problem was responsible for the high vibration levels. The cause of the instability was believed to be high cross-coupled stiffness values resulting from oil seal lock-up. To eliminate these effects, gas seals were installed as described in [12]. The rotordynamics model indicated an adequate log dec could be realized with gas seals. Baseline vibration data gathered after startup with the gas seals showed no significant subsynchronous components. It was felt that the use of gas seals resulted in a significant improvement in the rotor stability.

Because of rotor fouling causing imbalance, it was decided to change compressor rotors during the next planned unit outage. The compressor rotor was removed and was found to be heavily fouled. No significant subsynchronous vibration components were seen during the compressor startup with the spare rotor. During the plant startup, several operational problems were encountered. The compressor surged violently several times for extended intervals. During these severe surges, subsynchronous vibration grew to unacceptable levels. Vibration levels exceeded 10 mils (at 4500 cpm) when the speed was increased to approximately 10,000 rpm. Subsynchronous vibration accounted for 90% of the total amplitude. Acceptable vibration amplitudes could only be re-established by reducing speed below the first lateral critical speed. This again forced a compressor shutdown.

After these vibration excursions, the 5SLOP tilt pad bearings were removed for inspection. Some bearing shoes were found to be locked and the bearing wear pattern indicated the shoes were misaligned relative to the rotor journals. The bearing clearance was found to be 3 mils diametral, which was the minimum clearance specified by the manufacturer.

Tight bearing clearances and locked pads were both thought to contribute to reduced stability margin. The most expedient course of action available to get the machine back on-line was to install the spare bearings. The spare bearings were found to have 4 mils diametral clearance. During the installation of the 4 mil bearings, great care was taken to ensure proper bearing alignment. Even with the 4 mil bearing clearance and proper bearing alignment, the compressor again went unstable (>5 mils). Attention was focused on the possibility that internal damage to interstage and balance piston labyrinths might be the primary contributor to the subsynchronous frequency components. Compressor efficiency was low, rotor axial position was significantly different than past history indicated it should be, and thrust bearing metal temperatures were high. The machine speed was limited and the stability model from the gas seal retrofit project was used to evaluate potential modifications for further improving rotor stability.

Stability Analysis

The stability model included the tilt pad bearings, the interstage labyrinth seals, the balance piston labyrinth as well as aerodynamic cross-coupling at the impellers. For nominal values of bearing and labyrinth seal clearance, the system log dec (δ) at rated speed was predicted to be 0.24 at 10,000 rpm. Because of the surge event, it was felt that excessive interstage and balance piston seal clearances were likely. The computer model was used to predict the effects of labyrinth seal clearance of double their nominal values. Although the system δ reduced slightly to 0.23, it was still sufficiently positive to expect stable rotor behavior.

Another case used an arbitrarily high value for aerodynamic cross-coupling of $-40,000$ lb/in which was required to drive the rotor unstable. For this light gas recycle machine, the only feasible means of generating this level of aero loading was thought to be unexpectedly high labyrinth seal clearance (greater than 3$\times$ nominal) or excessive impeller eccentricity.

Rotor performance data showed that the machine was, in fact, achieving much lower efficiency than normal, indicating high labyrinth seal leakage rates. Due to production requirements and a lack of additional spare parts, an extended outage for rotor repair could not be taken. Therefore, the stability analysis was used to select an optimum bearing to replace the damaged bearing and the plan was to run at reduced throughput until the extended outage could be scheduled.

The analysis showed that the maximum clearance, minimum preload bearings resulted in the best rotor stability. A third party bearing vendor was used to make the bearing because of delivery time considerations. The upper end of the compressor manufacturer's clearance tolerance was chosen and bearings with 0.2 preload and 7 mils diametral clearance were manufactured. Pertinent rotor-bearing system parameters are summarized in Table 1.

TABLE 1. Rotor-Bearing System Parameters

Total Rotor Weight = 500 lbs | Bearing Span = 54.72 inches
Rotor Length 63.88 inches | Shaft Diameter = 2.9995 inches
Bearing Length = 1.375 inches | Oil Viscosity = 2×10^{-6} reyns
Bearing Type = 5 Shoe Load-on-Pad

Assembled Diametral Clearance (inches)	Preload	Bearing Coefficients at 11,000 rpm			
		Kxx	Kyy	Cxx	Cyy
0.005	0.5	2.85×10^5	3.79×10^5	286.4	335.5
0.007	0.3	8.35×10^4	2.25×10^5	132.3	221.4
0.008	0.2	4.93×10^4	2.11×10^5	98.8	205.4

Rotor Operating with 7 mil Bearing

The compressor was retrofitted with a self-aligning, ball and socket, five pad tilting shoe bearing. Bearing clearances were opened to 7 mils diametral (three inch journal) and pad curvature was modified to achieve a 0.2 preload. After startup, the subsynchronous peak was still present at 9000 rpm. The stability margin appeared to be slightly better, but was still unacceptable.

Based on the stability calculations, it was felt that internal damage to the compressor was responsible for the apparently high destabilizing forces. Figure 6 shows a plot of the subsynchronous vibration component versus compressor ΔP. Although the vibration levels were relatively low (< 2 mils, p–p), this was because the rotor speed was limited to about 8700 rpm. This sensitivity to pressure differential indicated that the effective destabilizing forces were greater than anticipated.

Plan for Scheduled Outage

With the unit operating at reduced rates, efforts were directed toward ensuring that, when the scheduled outage could be taken, the optimum rotor-bearing system was installed. Many hours of brainstorming led to four key items to be evaluated.

1. Change Rotor — The rotor exhibiting the unstable behavior was the spare rotor, installed when the original rotor was found to be heavily fouled. The question was raised as to whether the two rotors could have different levels of internal hysteretic damping, which could affect rotor stability. Since there was no way to determine this, it seemed that installing the original rotor, which had operated successfully, was the best choice. This was also desirable from a scheduling standpoint, since the rotor would be stacked and ready to change anyway.

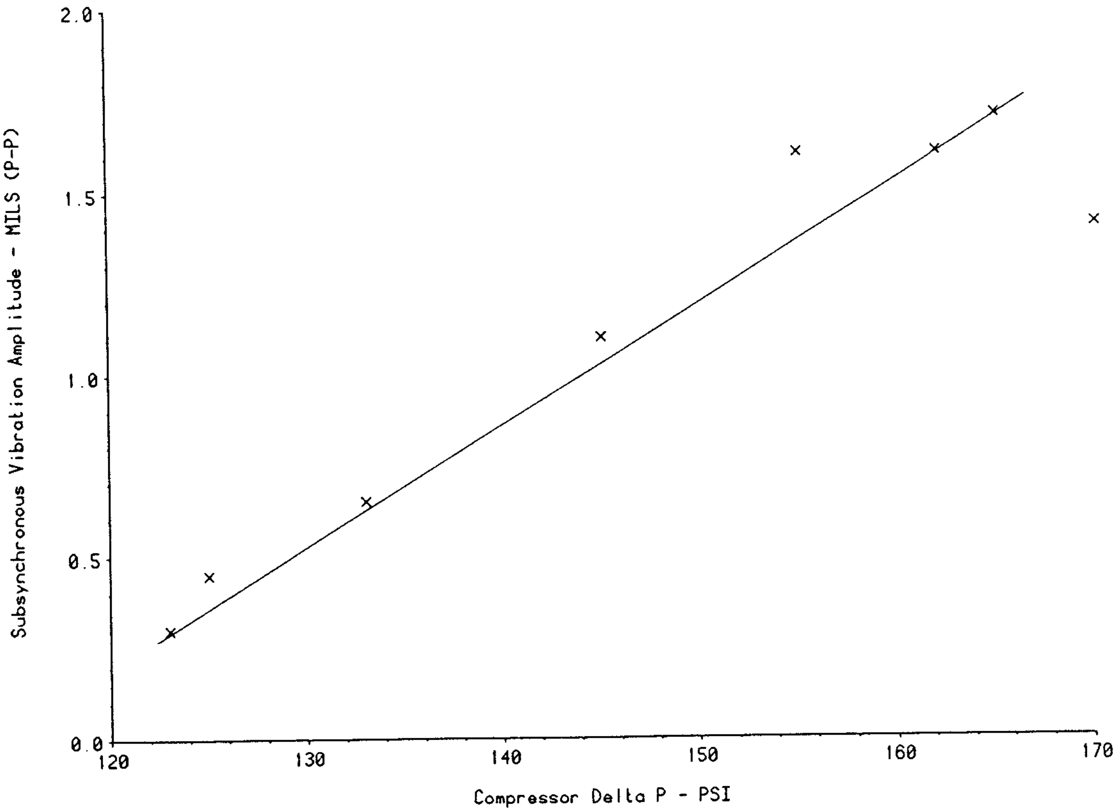

FIGURE 6. Hydrogen Recycle Compressor Plot of Subsynchronous Vibration Amplitude vs Pressure Rise

2. Further Optimize Bearings — Calculations were made to see if any other stability improvements could be realized by bearing design changes. Various designs, including 4 shoe load on pad, 5 shoe load between pad and four shoe load between pad were considered. The 5SLOP design was found to be the best from a stability standpoint. Discussions with the manufacturer led to an increased maximum clearance of 8 mils diametral with 0.2 preload. This was a slight improvement over the 7 mil bearings.

3. Oil Seal Effects — Oil ring seals have been thought to provide damping and, in some cases, improve stability [12]. For this compressor, the destabilizing cross-coupling components from the seals exceeded the damping components. Therefore, the oil ring seals did not improve stability. Consequently, the gas seals were preferable from a stability standpoint.

4. Additional Tests of Rotor During Rebuild — The original rotor was scheduled to be completely disassembled, re-stacked and high speed balanced in preparation for the outage. It was felt that perturbation testing to determine the mechanical log dec (δ_m) of the first damped eigenvalue of the rotor in its bearings would be useful in assessing rotor stability. Perturbation tests have been discussed in the literature [13–15] and have provided insight into rotor-bearing system dynamic behavior.

PERTURBATION TESTS

Theory

Based on the stability analysis, the mode of interest was the first forward whirl mode. This is often the case for large industrial rotors which experience unstable vibration problems. From a purely mathematical standpoint, it would be desirable to excite a shaft in its journals with a nonsynchronous rotating force excitation so that whirl direction could be determined and forward and backward whirl modes could be identified [13]. For this type of rotor, these modes exist relatively close in frequency. It is difficult and costly to develop a method of exciting the rotor directly with a rotating force excitation and such a method was not available within time and budget constraints for this analysis. Therefore, a small electrodynamic shaker was used to provide nonsynchronous, variable frequency excitation to the bearing pedestals. The simplified method described herein was shown to be an excellent method to obtain meaningful results of the rotor stability for varying parameters.

This unidirectional pedestal excitation proved to give valuable results identifying stability trends. For example, the observed behavior was that the nonsynchronous vibration levels increased as the speed increased. Therefore, the system log dec would be expected to decrease as the rotor speed increased. Additionally, the computer model predicted that the larger clearance, lower preload bearings were best from a stability standpoint. Consequently, perturbation tests at several speeds and with several bearing clearance values were performed.

Test Procedures

The rotor was prepared for at-speed balancing in a vacuum chamber, Figure 7. Appropriate bearing retainers were machined to fit the balance machine pedestals. Three sets of bearings were prepared, one with 5 mils diametral clearance and 0.5 preload, one with 7 mils diametral clearance and 0.3 preload and a third with 8 mils diametral clearance and 0.2 preload. Complete shaker data was obtained at rotor speeds of 7000–11000 rpm in 1000 rpm increments. At each speed, the shaker was swept in frequency near the expected value of the first damped eigenvalue.

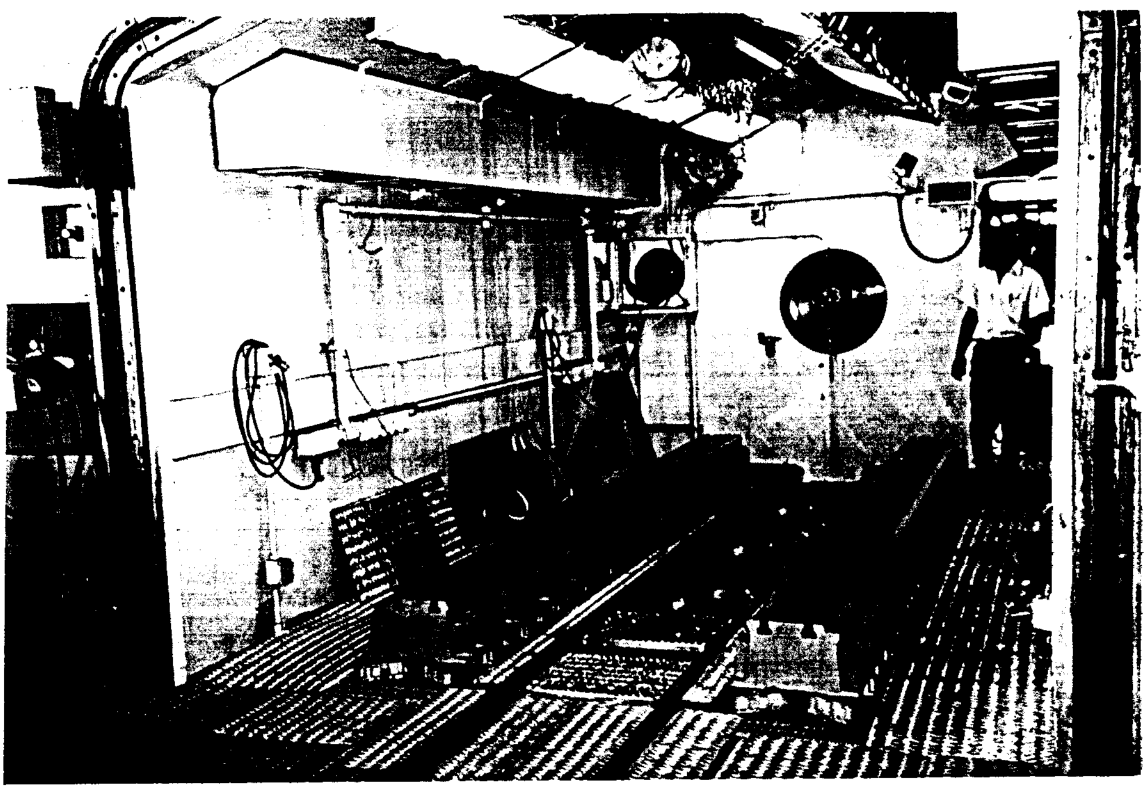

FIGURE 7. Vacuum Test Bunker

A photograph of the rotor installed in the balance machine is shown in Figure 8. A close-up view of the electrodynamic shaker is shown in Figure 9. The shaker had a force rating of 50 lb (p–p). The specified frequency range was 5–20,000 Hz which was more than adequate. The balance stand manufacturer specified that the pedestal stiffness was approximately 2×10^6 lb/in. For a 50 lb (p–p) force, the response would be only 0.008 mils (p–p) off resonance. On resonance, with an amplification factor (AF) of 12, the expected response amplitude would be approximately 0.10 mils (p–p).

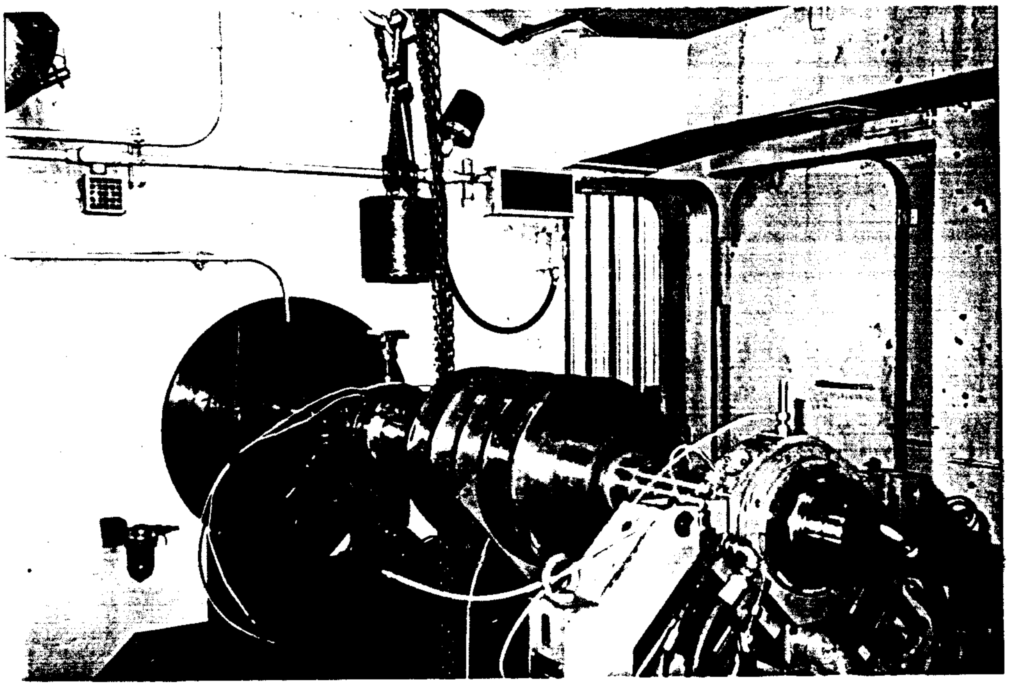

FIGURE 8. Rotor Mounted in At-Speed Balance Stand

In order to measure these low levels of vibration, proximity probes were installed at the normal probe locations. Since the maximum modal amplitude would be in the center of the rotor for the expected mode shape, a pair of probes was also mounted near the rotor midspan. With the midspan probes, acceptable amplitudes were obtained; however, a larger force shaker could have improved the signal to noise ratio.

Test Results

At each speed, from 7000 to 11000 rpm in 1000 rpm increments, the shaker frequency was varied near the expected location of the first damped eigenvalue, 60–75 Hz (3600–4500 cpm). A tracking filter was used to measure the shaft vibration amplitude versus frequency over the shaker frequency range. To enhance the signal to noise ratio, the transfer function was used by referencing the response amplitude to the shaker force input signal from a load cell mounted between the shaker and the bearing pedestal. To minimize the time spent on the test stand, the data were recorded using an FM tape recorder. This procedure was repeated for each set of bearings.

The transfer function (response/input force) was plotted in amplitude/phase versus frequency format (Bode plot) at each speed for each probe. Representative plots from the midspan X–probe with the 5 mil bearings are shown in Figure 10. From the Bode plots, the amplification factor (AF) and δ_m could then be approximated and compared to the calculated values.

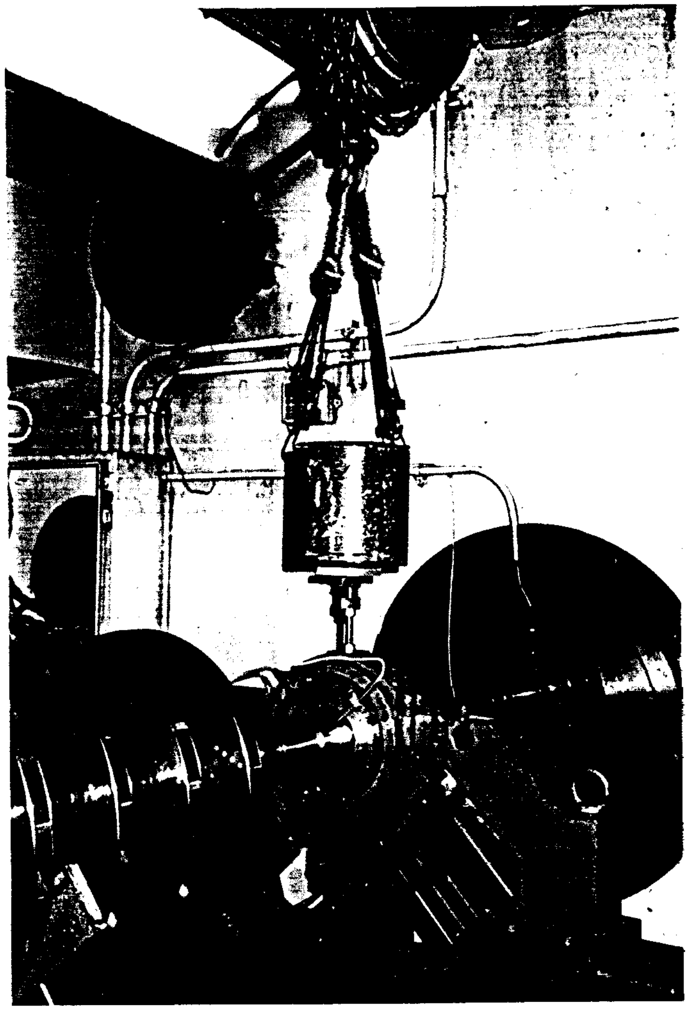

FIGURE 9. Electrodynamic Shaker Mounted on Bearing Housing

The original stability calculations for this machine were based on average values of bearing preload and clearance computed from the manufacturer's specified range of tolerances for journal diameter and pad bore. For the purposes of the perturbation tests, close attention to bearing dimensions was required. It is difficult to measure preload for a tilting pad bearing. The pad bore is usually known within 1 mil diametral before the bearing is split. After the pads are separated, the radius of curvature is difficult to determine accurately. The shaft diameter and assembled clearance are easier to measure. The preload value is computed based on the pad bore and assembled clearance.

For this compressor, the pad bore was specified as 3.009″ (+0.001″, -0.000″). The journal diameter was nominally 2.9995″, resulting in a machined diametral bearing clearance of 9.5–10.5 mils. Using the average value of 10 mils, the resultant nominal preloads for the three bearings tested were calculated.

The journal diameter actually ranged from 2.9992″ to 2.9995″ according to the inspection report. The diametral machined clearance could have been as high as 10.8 mils. Further, the bearing assembled clearance has a tolerance of ±1 mil diametral. The possible preload values are shown in Table 2.

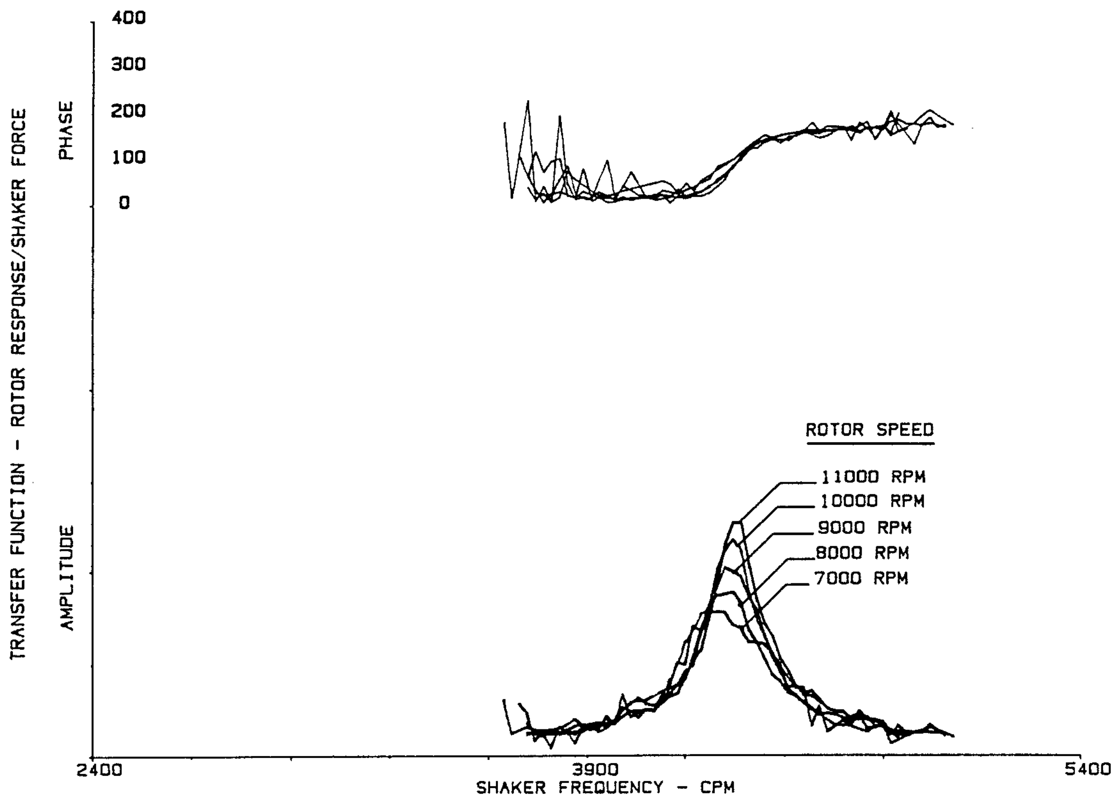

FIGURE 10. Midspan X-Probe — 5 Mil, 0.5 Preload Bearings

TABLE 2. Range of Bearing Preload Values

Assembled Diametral Clearance (mils)		Bearing Preload	
Minimum	Nominal	Nominal	Maximum
4	5	0.50	0.63
6	7	0.30	0.44
7	8	0.20	0.22

For the purposes of discussion, the nominal values of assembled clearance and preload were used. That is, the 5 mil bearings were designated as having 0.5 preload, the 7 mil bearings had 0.3 preload and the 8 mil bearings were designated as having 0.2 preload. The measured data correlated better with calculated data based on preload values higher than nominal.

Referring to Figure 10, it can be seen that the amplification factor increased and the log dec decreased, as the rotor speed was increased. The measured data along with calculated results are summarized in Table 3.

From this comparison, it can be seen that the measured results agreed favorably with the calculated values of system log dec. The predicted frequency was generally lower than the observed frequency, particularly for the 5 mil bearings. The authors have noticed a similar underestimation of the instability frequency on several other rotors of this type. It is thought that local deformation of the bearing pads due to thermal distortion may result in effectively increased preload and higher than expected bearing stiffness values. Non-linearities in the actual rotor-bearing system versus the linearized model may also account for some of this effect.

TABLE 3. Summary of Calculated and Measured Log Dec Values Versus Speed for the Three Bearing Configurations Tested

Rotor Speed (RPM)	M=0.5 5 Mil Assembled Clearance Bearings: Calculated Frequency (CPM)	Calculated Log Dec (δ_m)	Measured Frequency (CPM)	Measured Log Dec (δ_m)
7000	3598	0.175	4278	0.176
8000	3594	0.161	4323	0.153
9000	3597	0.147	4338	0.120
10000	3467	0.134	4331	0.087
11000	3608	0.122	4338	0.087
	M=0.3 7 Mil Assembled Clearance Bearings			
7000	3528	0.203	4038	0.204
8000	3501	0.196	4098	0.207
9000	3483	0.192	4068	0.185
10000	3464	0.185	4128	0.184
11000	3449	0.181	4038	0.146
	M=0.2 8 Mil Assembled Clearance Bearings			
7000	3519	0.204	3888	0.218
8000	3493	0.199	4008	0.212
9000	3467	0.194	3948	0.191
10000	3448	0.190	3948	0.167
11000	3427	0.185	3888	0.158

The measured trends in system log dec versus speed and bearing configuration generally agreed with the predicted results. Significant observations are noted below:

- System log dec decreased with increased rotor speed.
- System log dec decreased with increased bearing preload and decreased bearing clearance.
- Frequency of first mode was relatively insensitive to rotor speed. Measured value was generally higher than calculated value.
- Correlation between calculated and measured data was best for 7 and 8 mil bearings. Local pad distortion may explain some of the differences with 5 mil bearings.
- δ_m was lower than the design goal of 0.3. This was due to higher than expected bearing preload. Sensitivity of δ_m to possible values of bearing clearance and preload within the manufacturer's range of tolerances was significant. This emphasized that close attention to detail was required concerning the bearings for this machine to ensure stability.

CONCLUSIONS

Based on the tests and analysis results presented herein, the following conclusions are made concerning the simplified perturbation test method:

1. The procedure was simple in that the equipment used is readily available and would be applicable to most at-speed balancing facilities.

2. The procedure involved relatively little additional cost over the normal at-speed balancing procedure. Additional time in the vacuum test stand over the normal balancing time would be 8 hours or less.

3. The tests verified that the rotor-bearing calculations agreed reasonably well with the measured results. At high preload and minimum clearance values, bearing pad thermal distortion can apparently become significant.

4. The procedure provided a means for optimizing bearing parameters based on measured data. This increased the level of confidence that the best bearing was selected for installation in the machine during the outage.

5. The tests showed that the basic rotor-bearing design was sensitive to preload and had a δ_m value less than the design goal of 0.3 for bearings within manufacturer's range of tolerances. This emphasized that extreme care was required when specifying or replacing bearings in this machine.

Comments on Future Use and Development

It is felt that this simplified perturbation method could be useful in shop acceptance testing of critical machinery. It may also prove to be helpful in identifying uncalculable differences between primary and spare rotors. It is currently planned to repeat these tests for the spare rotor of the machine described in this paper.

Alternate methods of nonsynchronous excitation should also be pursued. Exciting the rotor directly may provide useful information concerning whirl direction, etc. This could be particularly relevant for overhung rotor designs in which gyroscopic influences are more pronounced.

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