RETROFIT OF GAS LUBRICATED FACE SEALS IN A CENTRIFUGAL COMPRESSOR

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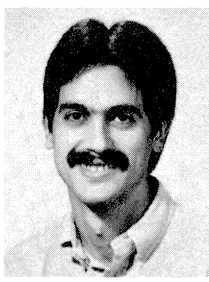

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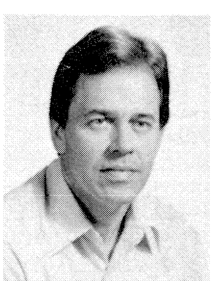

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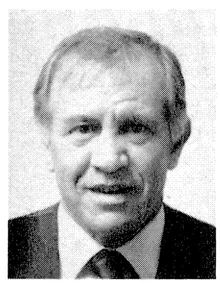

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ABSTRACT

There are significant advantages in using gas lubricated face seals (dry gas seals) in centrifugal compressor service. Foremost among these are the elimination of the seal oil system resulting in lower maintenance, increased safety, and higher operating availability. For these reasons, one of four identical compressors at an installation having severe problems with seal oil contamination was selected for trial conversion to dry gas seals. The rotordynamic engineering portion of this job was done by the compressor manufacturer in conjunction with the gas seal supplier. The seal assembly was designed so that it would be essentially a drop-in conversion from a mechanical standpoint. Rotordynamic studies indicated that the conversion would result in a "better" machine. Unfortunately, sustained operations were not possible, due to excessive vibration levels at startup with the new seals. The shaft vibration exceeded 0.007 in, peak-to-peak, at a subsynchronous frequency of 4900 rpm (the machine rated speed is in excess of 10,000 rpm). These levels were sufficiently high to cause extensive damage to all internal

labyrinths. The midspan labyrinths were wiped open in excess of 0.060 in, radial. Analysis of tape recorded data indicated that the vibration was due to a rotor/bearing system dynamic instability. Additional computer simulations of the compressor rotordynamics revealed that the oil seals had provided sufficient damping to the system to bound the instability. This extra damping was not being provided by the gas seals. Bearing redesign to increase stability and realignment of the rotor within the bundle to remove suspected excitation appear to have eliminated the problem.

INTRODUCTION

Background

In 1985, an installation of four "identical" multistage natural gas compressors was chosen for retrofit with gas seals. The intent of the retrofit was to eliminate the safety hazard of gas contaminated seal oil and reduce maintenance costs associated with the oil seals. Since the technology was seen as new, one compressor was selected as a testbed for the gas seal conversion. This compressor had a history of subsynchronous vibrations. It had originally been installed with only two stages. As the process conditions changed, additional stages were added until the current configuration of six stages was achieved. After each rerate, some fine tuning was required to reduce the vibration levels. However, once the fine tuning was completed, the vibrations had not affected the operation of the compressor. The amplitude of these vibrations had remained bounded at low values.

The compressor selected for the retrofit will be referred to as the B-1 compressor. These compressors discharge into a common header and are used to deliver gas to a commercial distribution facility. The B-1 compressor was somewhat unique, since it had a history of performance problems. To achieve the same the discharge pressure as the other three machines, its operating speed needed to be 150 rpm higher.

A lateral rotordynamic analysis to study the effects of the gas seals on the compressor was performed by the OEM. The gas seals, compared to bushing type oil seals, theoretically have greatly reduced levels of both damping and cross-coupled stiffness in the lateral rotor/bearing system. The study indicated that the net effect of the seal change would be a more stable machine. However, a higher amplification factor was predicted at the first critical speed. Consequently, special emphasis was placed on rotor assembly, balancing and minimizing seal eccentricity. The gas seals were installed in the compressor in mid-1985.

Subsynchronous Vibration Problem

In November 1985, as the demand for natural gas shipments rose, the speed of the B-2, B-3, and B-4 compressors was increased until the 150 rpm differential of the B-1 compressor was lost, resulting in a B-1 compressor surge which re-excited the first critical. The resulting high vibration levels caused an emergency shutdown (ESD). All efforts to bring it back on line were unsuccessful. Recordings were made of the compressor startup. Analysis of these recordings revealed that the frequency of the high vibration was at 48 percent to 49 percent of running speed, and the vibration increase did not occur until the operating speed of the machine reached approximately 9100 rpm. A waterfall plot of the compressor thrust end during runup is shown in Figure 1. Notice that there are no subsynchronous vibrations present in the compressor signature until the rotor speed reaches 9100 rpm. As the rotor speed exceeds 9100 rpm, the subsynchronous vibration appears. The amplitude of the subsynchronous component increases rapidly as the rotor speed approaches 10,000 rpm and has a frequency of 4920 rpm (49 percent)at that speed. At this point, the subsynchronous component amplitude is 0.004 to 0.006 in, peak-to-peak resulting in an ESD. Subsequent examination of the labyrinths revealed a midspan excursion greater than 0.060 in, radial. At this time, a complete rotordynamic analysis was undertaken by Amoco, in an attempt to identify the problem.

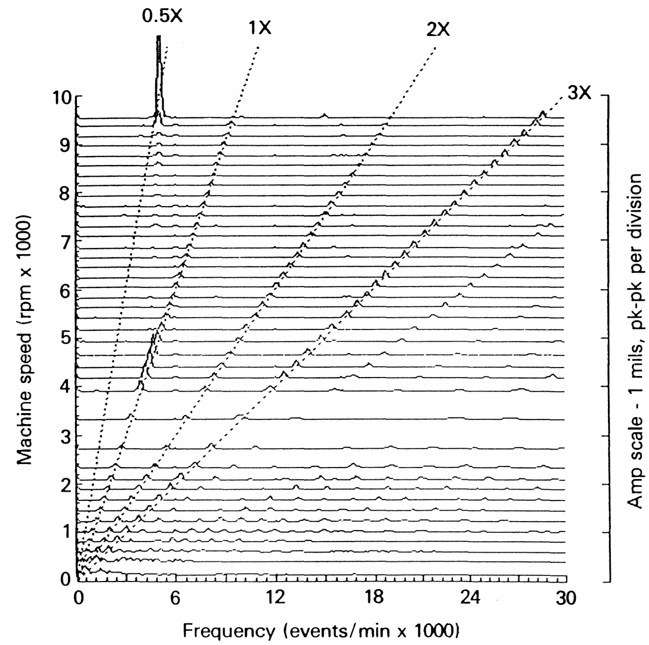

Figure 1. Waterfall Plot of B-1 Compressor Runup—Thrust End Vertical Probe.

LATERAL ROTORDYNAMICS ANALYSIS

In order to analyze and predict the behavior of the B-1 compressor, an accurate model was needed. The OEM cooperated to the fullest extent by supplying the appropriate shop drawings and their analytical model. This information was used for auditing the OEM design of the compressor rotating assembly. The shaft sleeves were treated as additional mass without increasing the shaft stiffness at the sleeve location. Impeller wheels were added as equivalent disks (the polar and transverse moments of inertia and the mass of the disk equals that of the corresponding impeller). The coupling was included and is located on the high pressure end of the compressor. The high pressure stages are located on the opposite end of the rotor from the thrust disk with the midspan as the reference point. This end will be referred to as the high pressure end. The model is presented in Figure 2.

The undamped natural frequencies of the compressor rotor were calculated using the bearing stiffness only. The deflection shapes of the first three undamped modes are presented in Figure 3. The bearing coefficients used were those calculated for the rotor operating speed. The calculated frequency of the first mode was 3720 rpm. Using bearing coefficients valid for the rotor speed at the first critical, the frequency of the first mode dropped to 3655 rpm. While this agreed closely with the OEM calculation, it was considerably lower than the observed first critical at 4350 rpm. Although it is not unusual for the observed first critical speed of a turbomachine to occur at a higher frequency than the calculated undamped first mode, the 16 percent discrepancy in this instance was not considered acceptable.

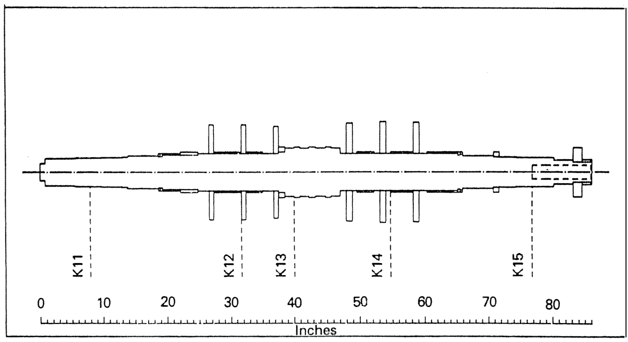

Figure 2. B-1 Compressor Rotor Model.

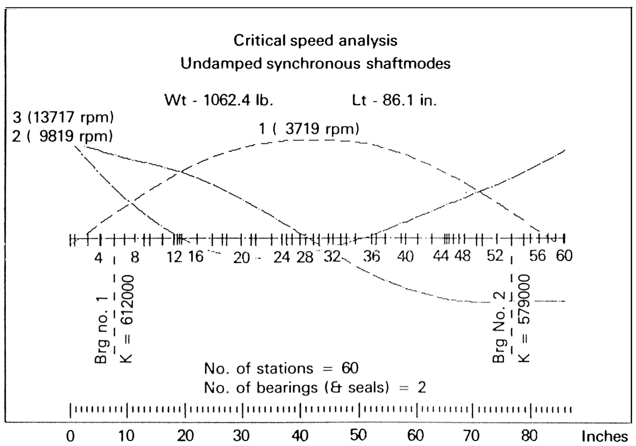

Figure 3. Undamped Mode Shapes with 9800 RPM Bearing Coefficients.

Balance Piston Labyrinth Effects

To improve the analytical model, the effect of the balance piston labyrinth was considered. This compressor consists of two sections of three wheels each in one case. The wheels are mounted such that their inlets face the opposite ends of the shaft. This places the balance piston at the midspan between the bearings of the rotor. Since the balance piston sees a 600 psi differential (from a 1200 psi discharge), it was not thought unreasonable to expect the development of dynamic bearing coefficients at this location. Inclusion of an additional, arbitrary, stiffness at the rotor midspan was selected in an attempt to eliminate some of the discrepancies between the model and the machine. This approach worked. Values of direct stiffness in the range of 50,000 to 100,000 lb/in placed the model response in close agreement with the real machine. It is interesting to note that, within this range, the value of the stiffness was not significant, only that some stiffness was present. However, it is still not completely understood why this was necessary in this particular study but not others in the authors' experience.

Encouraged by these results, it was decided to attempt to calculate the interstage gas labyrinth coefficients. This was done using a code supplied to Amoco by the ROMAC Industrial Research Program at the Department of Mechanical and Aerospace Engineering of the University of Virginia. The analysis is based on a method developed by Iwatsubo [1, 2, 3, 4, 5]. The impeller rear shroud, impeller eye and balance piston labyrinths were modelled. However, the calculated principle stiffness terms generated by the code for these seals was negative in sign. Application of these numbers to the rotor model lowered the calculated frequency of the first undamped mode. Negative principle stiffness in labyrinths have been reported by other researchers [6, 7] for both analytical and experimental derivations. Since the goal was to improve the model and, since some question as to the calculation accuracy of the principle stiffness terms existed, only the damping and the crosscoupled stiffness terms were used in the calculation (changing the theory to fit the data is somewhat better than changing the data to fit the theory). Unfortunately, as will be shown later, this left the rotor model with a significant discrepancy to the real machine. The coefficients for the impeller labyrinths were lumped together and modelled as apparent bearings in two places, one in the low pressure or thrust end and one in the high pressure or coupling end. These are locations K12 and K14 in Figure 2. The coefficients for the center labyrinth were also modelled as an apparent bearing. They are entered at location K13 in Figure 2.

The balance piston labyrinth had an anti-swirl feature incorporated in its design. Gas is bled from the discharge of the compressor and injected radially into the center of the labyrinth. The intended purpose of this feature depends on the beliefs of the compressor designer (there are at least two schools of thought in this matter). Some designers use this device to reduce the magnitude of the crosscoupled stiffness at the impeller tip. They maintain that this is accomplished by preventing the gas exiting from the impeller (which has a very high tangential velocity component) from entering the close clearance space between the impeller back shroud and the diaphragm. Other designers use this exact same device to reduce the magnitude of the crosscoupled stiffness produced in the labyrinth [6]. They maintain that this is done by establishing a negative pressure drop across part of the labyrinth, thereby not allowing the gas from the impeller (with its circumferential velocity component) to enter the labyrinth. In this manner, the total velocity of the gas in the labyrinth annulus is minimized.

In its original concept, the design of the B-1 compressor incorporated the anti-swirl feature to prevent the development of the crosscoupled stiffness at the impeller tip. However, the OEM's thinking tended to treat the device as a means to minimize the crosscoupled stiffness in the labyrinth. In fact, as an interim attempt to control the rotor, the OEM supplied the device with the inlet ports drilled at an angle against rotation. The hypothesis was that by inducing negative crosscoupled stiffness, the positive crosscoupled stiffness causing the vibration problem would be canceled out. One major problem with this approach is that there is some indication in current research that too much antirotation swirl can lead to instability [8]. This device will be referred to hereafter as a "negaswirl."

The damped natural frequencies of the system were calculated with an arbitrary amount of crosscoupled stiffness placed at the impeller wheels. The inclusion of this crosscoupling, which is assumed to be aerodynamic in origin, is based on past experience. Rotor stability of each mode was measured using the logarithmic decrement. The first damped natural frequency was calculated to be 3710 rpm with the stiffness and damping coefficients for a rotor speed of 9800 rpm. This was still 15 percent below both the observed first critical and 25 percent below the observed subsynchronous vibration frequency. An unstable mode was indicated by the logarithmic decrement, − 0.44 for this natural frequency. Thus, while the frequency was still being under predicted, the log decrement was indicating an unstable mode as was observed in the field.

Original Bearings

The original bearing configuration was a tilt pad design with five single degree-of-freedom pads with load between pads.

The bearings are at locations K11 and K15 in Figure 2. The speed-dependent dynamic coefficients of the bearings were calculated for shaft speeds up to 10,000 rpm. These coefficients were used in the analysis. The dynamic coefficients of the tilt pad bearings and the labyrinths for a rotor speed of 9800 rpm are presented in Table 1.

Table 1. Dynamic Coefficients for the Tilt Pad Bearings and Labyrinth Seals—at 9800 RPM Compressor Speed.

	Plain End Bearing	*HP Imp Labys*	*LP Imp Labys*	*Bal Drum Labys*	*Thrust End Bearing*
Kxx (lb/in)	482,000	0.0	0.0	0.0	468,000
Kxy = −Kyx (lb/in)	0.0	− 65,250	− 33,500	4,700	0.0
Kyy (lb/in)	612,000	0.0	0.0	0.0	579,000
Cxx (lb − s/in)	469.0	32.0	16.0	56.0	463.0
Cxy = Cyx (lb − s/in)	0.0	− 64.2	− 29.2	− 3.5	0.0
Cyy (lb − s/in)	534.0	32.0	16.0	56.0	519.0

First Bearing Change

Due to operational commitments, minimizing down time for the compressor was the highest priority short-term objective. Thus, only minor modifications to the OEM bearings were permitted. Initially, extensive modifications to the bearing housing or to the compressor were not examined. With these limitations set, the effect on the system stability of modifying the existing bearing clearance was examined. With the aid of the OEM, a more optimal bearing design was achieved. The bearing clearance was changed to 0.004 in radially from the 0.003 and 0.0035 in that originally existed in the compressor. This corresponds to a decrease in the preload to 0.33 from 0.42 and 0.5. A typical tilt pad bearing geometry, with the preload, m_b, defined is displayed in Figure 4. The net effect of the

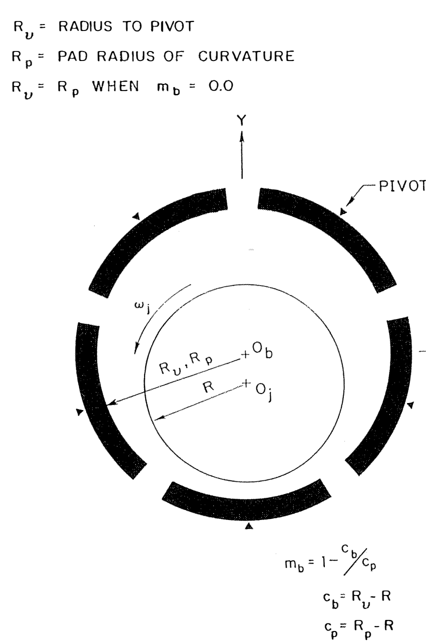

Figure 4. Typical Tilt Pad Bearing Geometry.

change was to soften the rotor supports and increase the effective damping at the center of the rotor. This led to an increase in the logarithmic decrement of the first mode to − 0.291. The new bearing coefficients for 9800 rpm are given in Table 2.

Table 2. Dynamic Coefficients for the OEM/Amoco Redesign of the B-1 Compressor Tilt Pad Bearings—at 9800 RPM Compressor Speed.

	Kxx	*Kyy*	*Cxx*	*Cyy*
Plain End	253,000	393,000	357.0	476.0
Thrust End	236,000	358,000	323.0	415.0

The redesigned bearings were installed in the machine in late November. Also at this time, the anti-swirl device (not the "negaswirl") was left in the balance piston labyrinth. Operation in the recycle mode was achieved to the maximum operating speed of the compressor without any large excursions of the subsynchronous amplitude. A subsequent attempt to place the compressor online was successful. However, one week after the installation of the bearings, the compressor tripped due to high vibration levels. From information obtained from field personnel, it was determined that the high vibrations were surge related. The first critical speed was now observed at 4250 rpm. A waterfall plot of the compressor plain end vibration (Figure 5), shows the re-excitation of the first mode natural frequency. Faults in the surge protection system were found and corrected. This enabled the compressor to operate for the next 1200 hours without significant downtime.

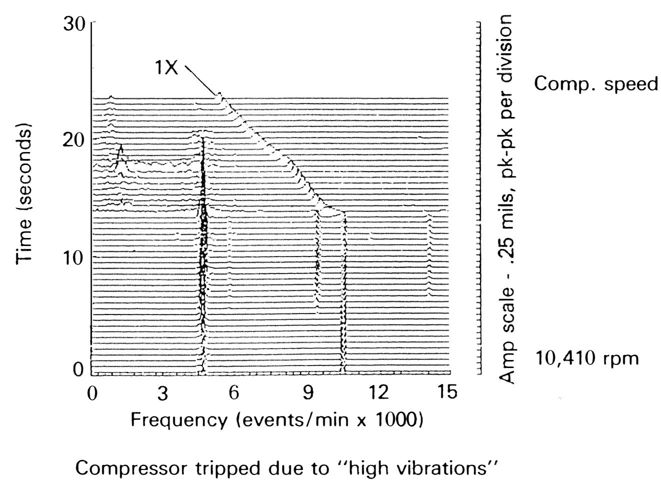

Figure 5. Re-Excitation of First Critical Due to Surge—Plain End Vertical Probe.

Second Bearing Redesign

In March 1986, after 1200 operating hours, efforts to bring the compressor back online after a vibration trip were unsuccessful. Subsequently, it was reported that during the three months when the 1200 hours were logged, the compressor had experienced several vibration trips. On each of these occasions, the speed differential between the B-1 compressor and the other compressors was lost, resulting in a B-1 compressor surge. However, after each of the previous trips, the B-1 compressor was put back online without difficulties.

Recordings were made of the current vibrations. Analysis revealed that the excitation of the subsynchronous component, as the recycle valve was closed to bring the compressor online, was causing the high vibration levels. Due to a reduction in operational commitments, the B-1 compressor was not needed for service. This allowed time for more extensive modifications.

The second bearing modification was directed at maximizing the stability of the compressor by further increasing the effective damping in the rotor system. The problem was diagnosed as a continuing instability, since the high vibration was due entirely to the subsynchronous component. To increase the damping, lengthening of the bearings was examined. It was determined that the existing housings could accommodate a bearing up to 2.75 in in length. This was 0.75 in longer than the bearings then in the compressor. At the new length, the L/D ratio of the bearings was increased to 0.595.

Stability Analysis

A stability analysis was undertaken to optimize the bearing design at the 2.75 in length. The pad arc length, groove arc length, oil type, number of pads, and the load placements were held constant. The bearing clearance and preload were set to 0.004 radially and 0.35, respectively. At this value, a ratio of 0.0017 in clearance per inch of shaft diameter was achieved. The bearing clearance and pad clearance were then varied around this point. The model used in this investigation was identical to that previously used, with the exception of the bearing characteristics. The stability of the compressor was calculated for each new bearing configuration. A plot of the results is presented in Figure 6. The curves illustrate that, in this instance, varying the pad clearance had the largest affect on the rotor stability. A bearing design with a 0.004 in bearing clearance and a 0.005 in pad clearance was selected. The zero preload point was not selected in order to avoid operating in the negative preload region. (This might arise due to manufacturing, assembly and/or maintenance tolerances.) A final optimizing step was performed to ensure that the best bearing design was selected. The clearances were varied 0.001 in in each direction and, the stability of the compressor was calculated for each case. The results, presented in Table 3, assured us that the best practical bearing design was achieved. Additionally, the bearing was specified with two degree-of-freedom pads. While the benefits of this type of pad can not be shown analytically, it is felt this represents a superior design over single degree-of-freedom pads. The two degree-of-freedom pads reduce axial misalignment and thus, edge loading of the bearing.

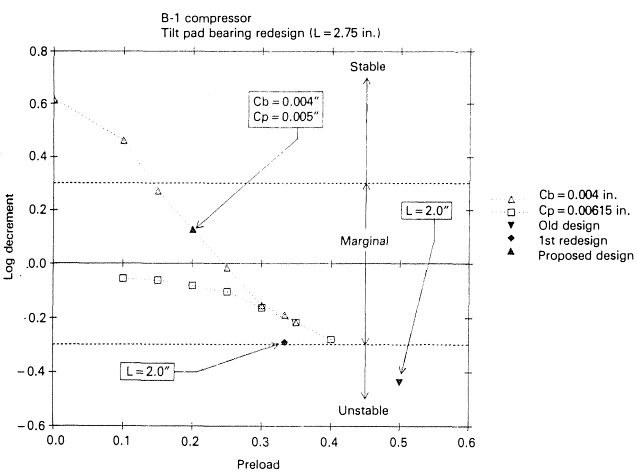

Figure 6. Stability Calculations for B-1 Compressor with Various Bearing Designs.

Forced Response Analysis

A forced response analysis was undertaken to study the sensitivity of each bearing design to excitation of subsynchronous whirl. The model from the previous studies was used for this analysis. The rotor speed was held constant at 9800 rpm. A sinusoidal force was applied at the rotor center with a magnitude approximately one quarter of the rotor weight. The frequency of the force was varied throughout the rotor speed range. The subsynchronous response of the rotor was calculated and compared at the first critical. The responses of the original, the OEM/Amoco redesign and the Amoco redesign (L = 2.75 in) are plotted in Figures 7, 8, and 9. The response of the longer bearing with the clearance reduced by 0.001 in to a level of 0.003 in radially is shown in Figure 10. The findings of the forced response study are summarized in Table 4. Further testing of the compressor was planned for late July 1986. At that time, quantitative data on the rotor response would be gathered.

Table 3. Log Decrement Calculations of the Proposed Tilt Pad Bearing Redesign (L = 2.75 in) for Varying Clearances.

Pad Clearance (inches)	Bearing Clearance (inches) 0.003	0.004	0.005
4.0	– 0.107	0.616	-----
5.0	– 0.221	0.127*	0.217
6.0	– 0.410	– 0.188	– 0.039

----- negative preload
* proposed design point

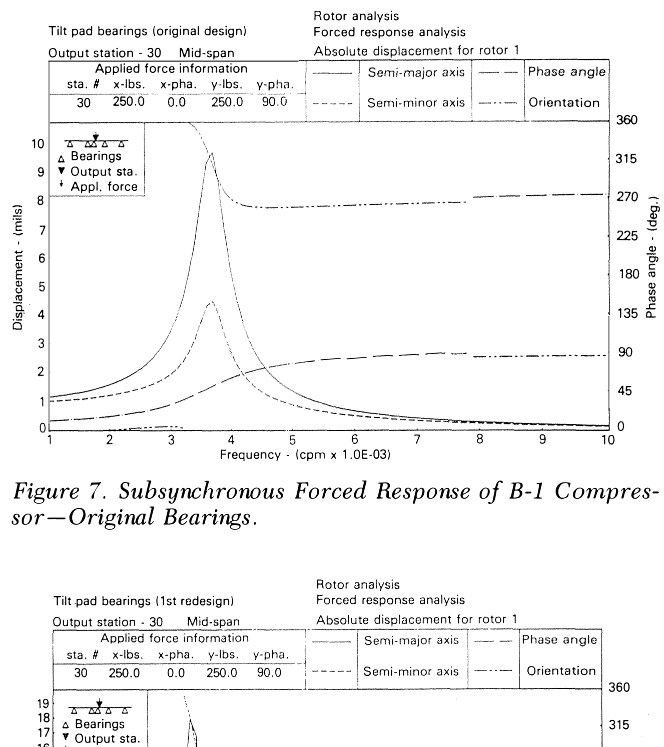

Figure 7. Subsynchronous Forced Response of B-1 Compressor—Original Bearings.

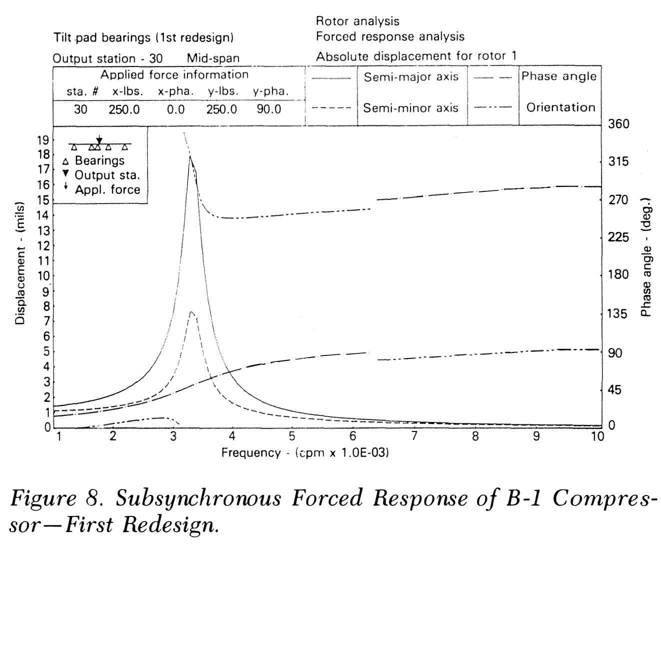

Figure 8. Subsynchronous Forced Response of B-1 Compressor—First Redesign.

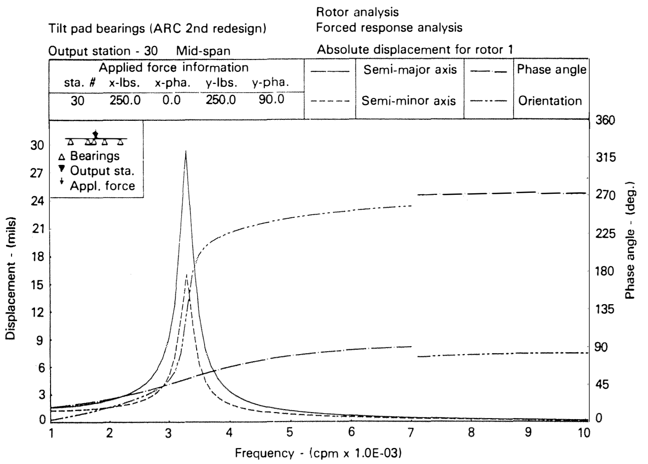

Figure 9. Subsynchronous Forced Response of B-1 Compressor—Second Redesign (L = 2.75 in).

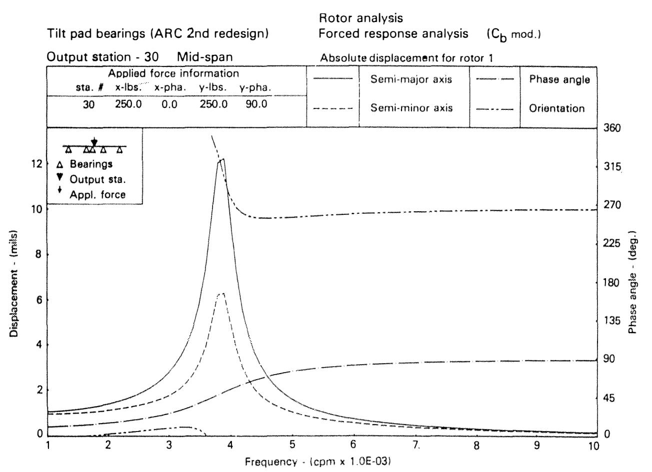

Figure 10. Subsynchronous Forced Response of B-1 Compressor—Second Redesign with 0.003 In Bearing Clearance.

Table 4. Comparison of the Stability and Subsynchronous Response for the Bearing Redesigns of the B-1 Compressor.

Bearing Design	*Response at 1st Critical Speed (inches)*	*Stability (Log Decrement)*
Original	0.009652	– 0.438
OEM/Amoco	0.0179	– 0.291
Amoco	0.02939	0.127
2nd Amoco	0.012222	– 0.211
(Cb = 0.003 in)		

DISCUSSION

Field Testing of New Bearing Design

The longer bearings were installed in the B-1 compressor in late June 1986. Also at this time, the "negaswirl" device was installed in the balance piston labyrinth. No subsynchronous vibrations were recorded on the thrust end bearing throughout the operating range with the system in the recycle mode. The plain end bearing had subsynchronous levels below ten percent of the synchronous component. Operation throughout the speed range was achieved without inducing large excursions in the vibration levels. However, field reports seemed to indicate an increased sensitivity of the compressor to fluid forces. Specifically, attempts to take the compressor off recycle were unsuccessful. As the recycle valve was closed, putting the compressor online, an increase of the subsynchronous vibration component was experienced. This was a new phenomena which appeared to be the result of a fluid dynamics excitation. Unfortunately, the effective softening of the compressor bearings apparently developed a forced response problem in the machine.

Fluid Dynamics Excitation

An investigation as to possible sources of the fluid dynamics excitation was initiated. All suction and discharge lines and valves were checked for obstructions and for proper operation. The source of the problem was not discovered there. It was decided to reopen the machine and look for obstructions in the gas flow path and to remove the "negaswirl" device. Since the machine had operated successfully on recycle, but not at load, the two most probable sources of the fluid dynamic excitation were determined to be the "negaswirl" device and flow path restrictions. No obstructions were discovered in the machine. However, it was found that, despite the fact that drawing dimensions had been followed in the buildup, the rotating assembly was severely misaligned in the bundle. The eccentricity resulting from this misalignment was strongly suspected as the source of the fluid dynamics excitation. A new alignment procedure was developed which eliminated this problem.

Field Testing After Realignment

Operational tests of the B-1 compressor were conducted on July 21-24, 1986. The goal of the testing was to evaluate the response of the compressor to the newly redesigned tilt pad bearings and alignment procedure. Both were developed to reduce vibration levels in the compressor that had been hampering operation since the retrofitting of the compressor with gas seals. During the testing program, the compressor was subjected to several starts and loadings. Operation of the compressor in parallel with the B-3 machine was also accomplished. Surge protection at one operating point was checked. The results of the tests proved satisfactory with no experienced high vibration levels.

The reduction of the data obtained during the testing period and the presentation of the results is divided into two sections, transient and steady state operation. The transient section deals with compressor startups and shutdowns along with any significant change in compressor speed during testing. The steady state section covers any testing procedures performed at a constant compressor speed, i.e., loading, parallel operation, and surge protection tests. The results are presented in chronological order. Probe locations are at the 135 and 45 degree locations measured from the positive x-axis with clockwise shaft rotation. The 135 degree probe is designated as the vertical probe.

Transient Analysis

A rundown of the compressor was captured on July 21. Compressor speed was reduced from 7400 rpm to 2500 rpm. This range was used to show the location of the first critical speed which was expected to occur at 3720 rpm. The bode plot of the inboard vertical probe indicates a possible split in the natural frequency. The horizontal natural frequency occurs first at 4200 rpm, this is due to the fact that the rotor support in the horizontal direction at the probe location is softer than in the vertical direction. The vertical natural frequency occurs 200 rpm higher at 4400 rpm. The phase angle plot shows the expected shift in the readings at these frequencies. The outboard probe indicates only one critical speed at 4200 rpm. The response of the two probes is compared in Figure 11. Cascade plots reveal no other significant vibration components occurring within this speed range with the exception of the 2X component at the outboard end, Figures 12 and 13.

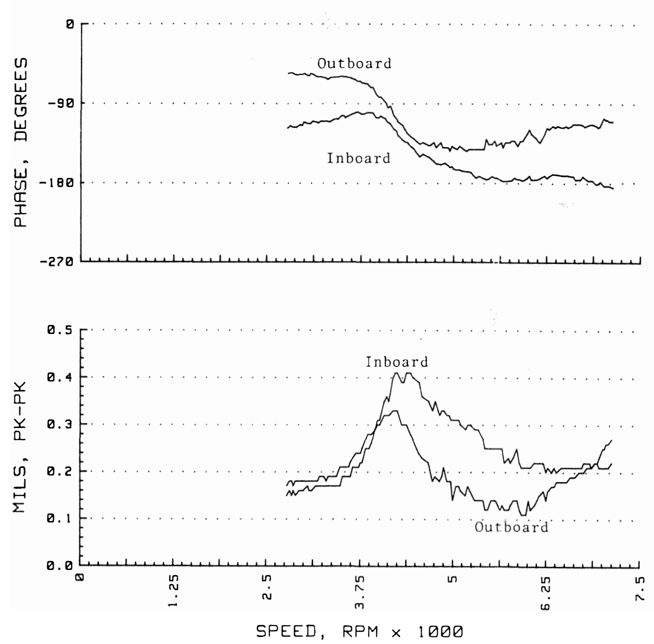

Figure 11. Bodé Plot of Coastdown from 7400 RPM to 2500 RPM—Vertical Probes.

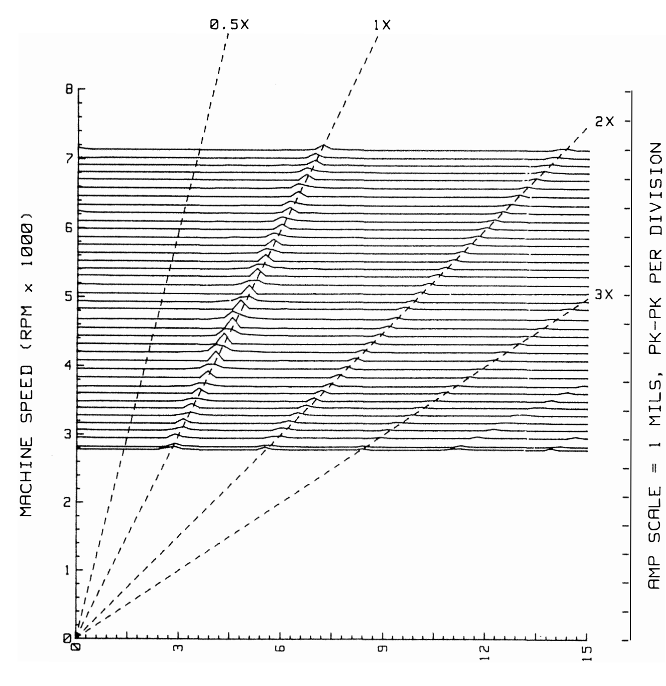

Figure 12. Cascade Plot of Coastdown from 7400 RPM to 2500 RPM—Inboard Vertical Probe.

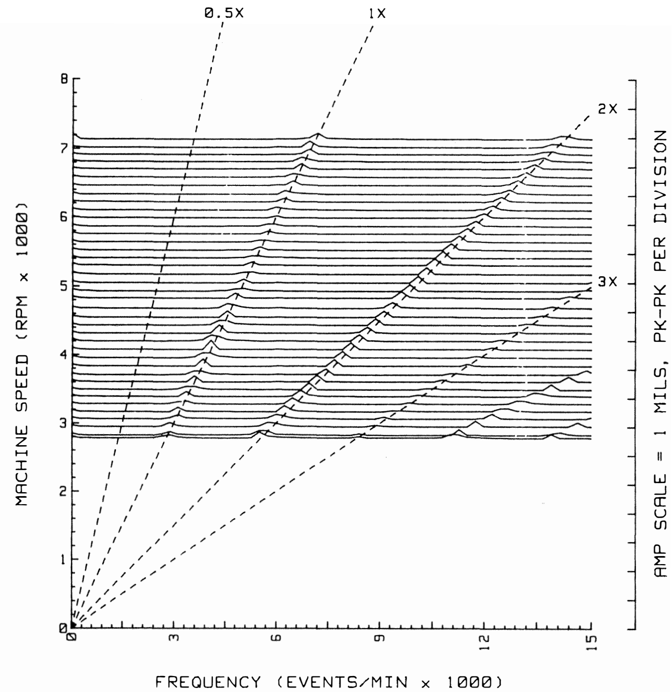

Figure 13. Cascade Plot of Coastdown from 7400 RPM to 2500 RPM—Outboard Vertical Probe.

The amplification factor for the first critical speed at the inboard end can be calculated using information contained in the previous plots. Using the API definition for the amplification factor, the calculated value is approximately 5.0 for this end. The amplification factor of the first critical at the outboard end is calculated to be approximately 4.1.

Steady State Analysis

The steady state analysis is used to identify problems during loading or surge protection testing. Additionally, comparisons of the steady state vibration spectra at different compressor speeds are examined.

The spectra for the inboard vertical, inboard horizontal, and outboard vertical probes at a compressor speed of 9270 rpm are displayed in Figure 14. Overall vibration levels are quite low. A slight subsynchronous component can be seen on the inboard vertical probe at 4600 rpm. Since the frequency of the subsynchronous vibration is about half the running speed, this marks the onset of oil whirl. The instantaneous data taken while the compressor was fully loaded at a speed of 10366 rpm are shown in Figure 15. The magnitude of the synchronous components has risen to 0.0004 in, peak-to-peak, on the average. This is an expected result of loading the compressor. Levels are still greatly less than the alarm points. The subsynchronous component has locked on to the first natural frequency at 4600 rpm. The magnitude of the subsynchronous vibration is dictated by the amount of crosscoupled forces generated in the compressor. These forces increase with increasing speed and load.

Data for the B-1 compressor in parallel operation with the B-3 compressor are contained in Figures 16 and 17. The speed of the B-1 compressor was 10350 rpm and 10780 rpm, as shown in Figures 16 and 17, respectively. Notice that the parallel operation had no affect on the vibration levels. These levels remained constant during surge protection testing of the B-1 compressor as well. During this test the B-3 compressor was used to drive the B-1 compressor into recycle. The opening of the recycle valve did not alter the vibration patterns of the B-1 compressor.

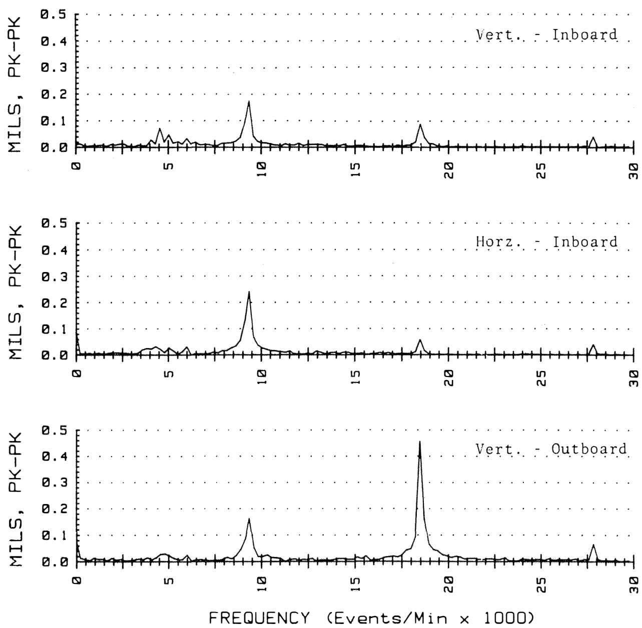

Figure 14. Vibration Spectra— 9270 RPM— Compressor in Recycle Mode.

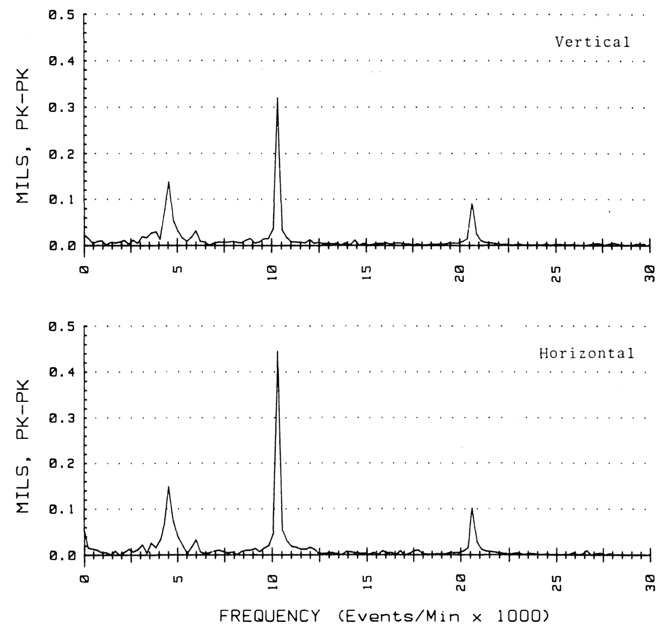

Figure 16. Vibration Spectra—10350 RPM—Compressor at Full Load in Parallel with B-3 Compressor—Inboard Probes.

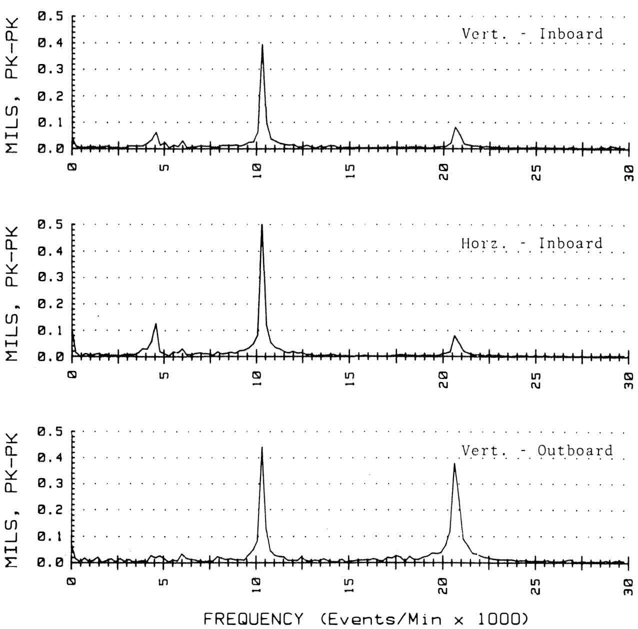

Figure 15. Vibration Spectra—10366 RPM—Compressor at Full Load.

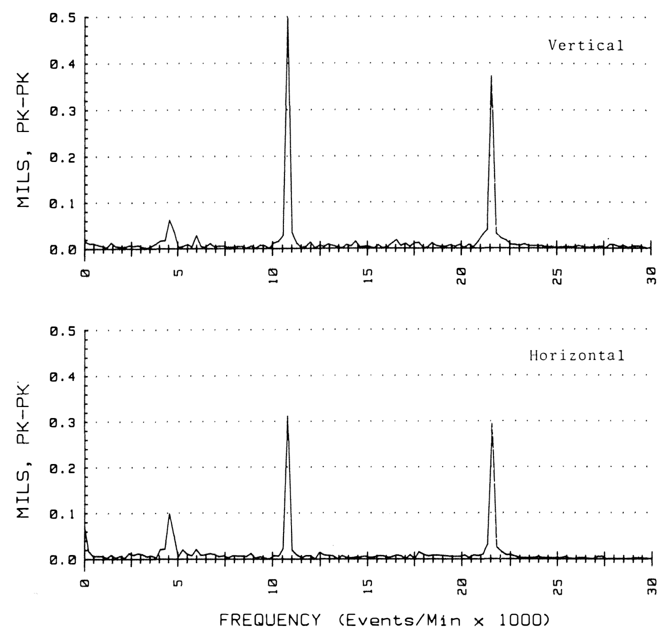

Figure 17. Vibration Spectra—10780 RPM—Compressor at Full Load in Parallel with B-3 Compressor—Outboard Probes.

CONCLUSIONS

The classical approach to oil bushing seals has been to accommodate the crosscoupled stiffness in a "worst case" scenario when designing a rotor/bearing system. The damping has been viewed as a positive influence which was assumed to make a design more conservative. In the case history presented here, however, the extra bit of damping provided by the oil seals was all that prevented a marginally stable rotor from going unstable. When this damping was removed by the replacement of the oil seals with dry gas seals, the rotor was no longer stable. It became necessary to provide additional damping by redesigning the bearings. In the course of doing this, the system became vulnerable to a forced response phenomena excited by a fluid dynamic force resulting from misalignment of the rotor in the bundle. Unfortunately, since the "negaswirl" device was removed at the same time as the rotor was realigned, no evaluation of it's performance can be made at this time. The dry gas face seals are extremely durable, as evidenced by the severe vibration excursions they have survived in this machine with no damage. However, a very close examination of the rotordynamics considerations is necessary, whenever they are to be utilized.

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