ROTOR INSTABILITY PROBLEMS IN AN INTEGRALLY GEARED COMPRESSOR SUPPORTED BY TILTING PAD BEARINGS

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ABSTRACT

The appearance of a strong subsynchronous instability in an overhung compressor highlights the fact that rotordynamic instabilities are not restricted to high-pressure multistage between-bearing compressors but can also occur in lower pressure compressors with overhung rotors. Another important point is that many users mistakenly believe that tilting pad bearings eliminate all cross-coupling forces that cause subsynchronous instability. While this is true of the cross-coupling coefficients of the bearings themselves, a substantial cross-coupling force from labyrinth seals can overcome the beneficial characteristics of tilting pad bearings.

This paper describes the installation of an integrally geared compressor that initially had problems with high bearing pad temperatures. The manufacturer attempted to modify the bearings to reduce the pad temperatures but this was unsuccessful due to unacceptably high vibration amplitudes caused by rotordynamic instability. The rotordynamic instability was caused by a strong

subsynchronous excitation of a cantilevered rotor natural frequency located at 47 percent of running speed. The subsynchronous excitation was due to labyrinth seal cross coupling effects. Based on the recommendation of the authors, the manufacturer installed swirl brakes to decrease the subsynchronous vibration excitation, which eliminated the rotordynamic instability and allowed the bearings to be modified in order to reduce bearing pad temperatures.

INTRODUCTION

A combined service six-stage (three plus three) integrally geared compressor was installed in an industrial gas facility. The compressor is electric motor driven with three stages for high-pressure nitrogen compression and three stages for high-pressure dry air compression. The compressor was commissioned with bearing pad temperatures that were operating continuously over 200°F. Subsequent bearing inspections showed excessive oil coking and wear. Attempts to modify the bearings were unsuccessful due to rotor instability causing high vibration trips. Approximately six months after startup, the compressor had a series of sudden high vibration trips on the first stage of the dry air service. After each trip, the compressor was successfully restarted and would operate under normal conditions until the next unexpected vibration trip event.

The authors visited the site to secure more detailed data in order to analyze the problem and determine a solution with the manufacturer. The data collection involved a series of process and bearing oil tests to recreate the trip phenomena. Subsequent rotordynamic analysis further supported the conclusions reached in the field from testing.

In the course of the investigation, the manufacturer attempted several bearing modifications to reduce bearing pad temperatures. These were all unsuccessful as each modification reduced the rotor stability further, resulting in unacceptably high vibration amplitudes.

While the manufacturer's initial design calculations indicated acceptable rotor stability, it became clear from the field data and analytical analysis that the rotor was in fact less stable than predicted. Redesigning the bearings could increase the stability somewhat but would not eliminate the excitation source. The manufacturer ultimately agreed to install swirl brakes to reduce the excitation force and modify the bearings to reduce pad temperatures. This paper describes the reasoning behind the fix, and the results of its implementation.

MACHINE DESCRIPTION

The train consists of an induction motor coupled to a six-stage integrally geared, multiservice compressor. A central bull gear drives three high-speed pinions with a total of six impellers. Each service uses three impellers, i.e., three stages. The 6800 kW induction motor is coupled to the bull gear via a dry flexible disc coupling. The services include nitrogen compression, designated as CP53, and dry air compression, designated as CP14. There is an intercooler between each stage. Tables 1 and 2 summarize the compressor/rotor information and Figure 1 illustrates the configuration. When viewed from the motor, rotor one is at the nine o'clock position (upward driven), rotor two is at the three o'clock position (downward driven), and rotor three is at the 12 o'clock position (horizontally driven). The bull gear rotates clockwise and all pinions rotate counterclockwise.

Continuously adjustable inlet guide vanes (IGVs) are provided at the CP53 first and third stages (positions A and C) and CP14 first stage (position D). The CP53 IGVs are mechanically linked to modulate flow simultaneously.

Each pinion journal location is equipped with X-Y proximity probes. There is one keyphasor probe on each rotor. The probe transmitters provide a 4 to 20 mA input to the plant programmable logic controller (PLC) system so overall vibration levels can be monitored and trended. Tripping, due to high vibration, is via the plant PLC.

Table 1. General Compressor Information.

Service	Stage	Bearing	Rotor	Speed RPM
CP53 Nitrogen	1	A	1	29,150
	2	B		
	3	C	2	28,210
CP14 Dry Air	1	D		
	2	E	3	23,635
	3	F		

Table 2. Rotor 2 Operating Conditions.

Service	Stage (Brg Pos)	Inlet Press Psia	Inlet Temp °F	Power KW
CP53 Nitrogen	3 Stage (C)	330	95	660
CP14 Dry Air	1 Stage (D)	280	95	1200

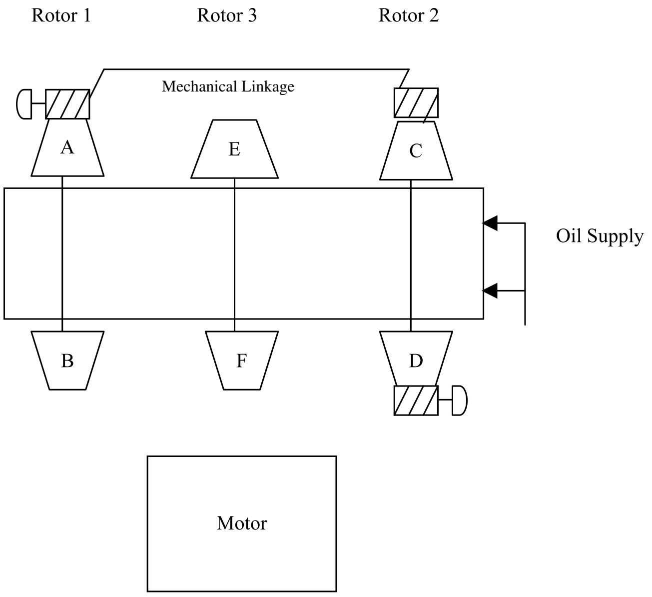

Figure 1. Compressor Layout.

Closed impellers, with labyrinth eye seals, are used at positions C, D, E, and F. The CP53 position's A, first stage, and B, second stage, are semi-open impellers (no impeller eye seal). The shaft seals located at the back of the impellers for positions A, B, C, and D are labyrinth type seals, typical for this type of compressor. Dry face mechanical seals are used for the shaft seals of CP14 stages two and three, positions E and F, in order to minimize leakage losses.

The original pinion bearings (unmodified) are a radial type utilizing five tilting pads (Table 3). The pad material is copper-chrome to provide increased heat dissipation. Bearing load is between pads at full gear load. The pads use a center-supported spherical seat (0.50 offset) with pad arc lengths of 58 degrees. The C, D, E, and F bearings are identical in design and size. The final modified bearing utilized on the C, D, E, and F positions are chamfered on the trailing edge in order to help reduce bearing temperatures. This chamfer effectively reduces the pad arc length to 53 degrees, creating a 0.547 offset.

Oil supply to the bearings is from a shaft-driven main oil pump with an oil pressure regulator maintaining 30 psi. Oil temperature

Table 3. Bearing Data.

Pinion	A/B	C/D	E/F
Load press, Bar	19	18	19
Circ. Speed, m/s	91.6	96.0	80.4
Bearing Dia, mm	60	65	65
Relative Width (L/D)	1	1	1
Rel Clearance, 1/1000	2.68	2.63	2.63

is controlled via a three-way oil temperature control valve. Depending upon the ambient and cooling water temperature, the oil supply temperature ranges from 110°F to 115°F.

OPERATIONAL HISTORY

The compressor is part of an industrial gases plant providing both high-pressure nitrogen and oxygen to a chemical plant complex. The plant was originally commissioned in August-September 1999 but was then taken offline until the customer was prepared for production. The plant and compressor were restarted in April 2000.

During the shop test, the compressor manufacturer was concerned with higher than expected bearing pad temperatures. Bearing temperatures were running well above 200°F. In order to decrease the bearing pad temperatures, the manufacturer provided a specially designed bearing to provide more direct injection of cold oil into the pad. These bearings were field-installed during the original commissioning in 1999. The compressor could not operate with these bearings due to high vibration. The bearings were removed and the original bearings were reinstalled, and the commissioning process was completed with higher than desired oil temperatures.

After the plant was put into full operation in April 2000, the compressor continued to run for approximately three months until there was a series of sudden unexplained trips due to high vibration. The trips occurred on each of four successive days. All trips were caused by high vibration on the CP14 first stage (position D). It was noted that the trips occurred during the hottest weather days since the compressor had been started.

Typical trip scenarios involved the plant personnel operating the plant in steady-state conditions and, without warning, the compressor tripping on high vibration. One of the trips is shown in Figure 2 and is indicative of what the plant personnel would see in the control room. The compressor would be running steady-state with all parameters "flatlining"; the process flows/pressures/temperatures, oil supply pressure/temperature, and vibrations would be unchanging. The CP14 first stage vibration would start to increase until a sharp spike initiated a shutdown. From the time of the first increase in vibration till shutdown was two and half minutes. The sudden increase in vibration amplitude occurred in a matter of seconds.

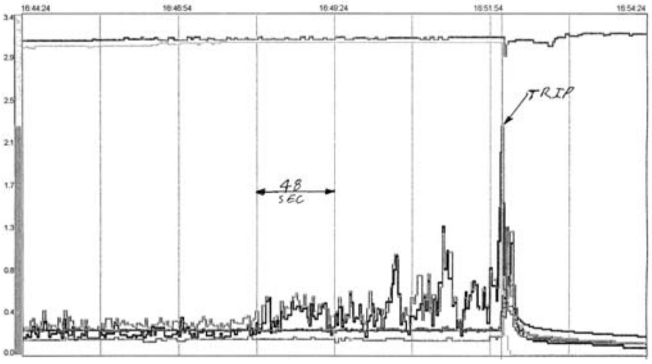

Figure 2. Sudden Vibration Trip as Seen in Plant Control Room.

After the fourth trip, the authors traveled to site to investigate the cause of the trips. It was the intent of the authors to try to reproduce the operating conditions that would cause the trip and to observe and record the trip with a spectrum analyzer. Up to now, the main information available included overall compressor operation (pressures, temperatures, IGV positions, overall vibration levels, oil pressure/temperature, etc.). No vibration data in the form of fast Fourier transforms (FFTs) were available.

INITIAL FIELD TESTING

Upon arrival on site, the available field data were analyzed to determine if there were any discernable trends prior to the high vibration trips. This review showed the plant and compressor running steady-state with most trends "flatlining," including oil temperature and pressure. This analysis was inconclusive, although small changes in IGV position and compressor load were observed just prior to the trips.

The authors initiated a variety of tests in order to isolate the cause of the high vibration trips. The testing was intended to simulate the operating conditions immediately before a trip and was ultimately successful in causing a trip during oil temperature variation tests.

Instrumentation

The main measurement techniques involved using real-time high frequency FFT spectra plotted as a function of time (with a high degree of sample-to-sample overlap) to examine various transducer outputs:

- *Accelerometers*—Such phenomena as vane pass, IGV, and diffuser vane frequency could be examined based on the casing axial sidewall and bearing housing radial accelerometer measurements.
- *Pressure transducers*—Pressure transducers were used to examine the dynamic nature of the pressure at both the inlet and discharge of the first stage of CP14. Pressure measurements were also examined on the labyrinth rotor seal discharge location to look for evidence of aerodynamically induced cross-coupled forces.
- *Proximity probes*—Proximity probes already mounted on the machine were used to obtain direct measurements of the rotor vibration relative to the bearing housing.

Process Variation Tests

A number of process variable tests were conducted to investigate whether process conditions were contributing to the sudden vibration trips. These tests included variations of the IGV angles at various flow rates, depressurization of the nitrogen stages of the compressor, and unloading of the air stages of the compressor. No sudden changes in vibration levels were observed through this testing, although the subsynchronous component later found to be causing the vibration trips was seen to lessen in magnitude as the air compressor stage was unloaded.

Observing the spectrum on a real-time basis, a subsynchronous (i.e., below running speed) component could be observed "bobbing" up and down while the rest of the spectra including one times running speed and multiples thereof were of constant magnitude. The overall magnitude of this subsynchronous frequency and of the overall vibration was quite low however, and could not be conclusively pinpointed as causing the problem. The authors began to suspect the possibility of an aerodynamically induced whirl phenomena in the seals, and therefore attempted to observe the pressure at the problem seal location in hopes of seeing some confirmation of subsynchronous whirl in the spectrum of the pressure probe. However, investigation of the CP14 first stage labyrinth shaft seal pressure failed to detect any unusual subsynchronous pressure phenomena.

Even though the seal gas tests were inconclusive, the authors still suspected rotordynamic instability based on the unsteadiness of the randomly excited critical speed at 47 percent of running speed. Turbomachinery rotors tend to whirl at rotations somewhat less than 50 percent running speed, since the flow in tight clearances (such as the labyrinth seal, or in plain or fixed-pad journal bearings) will be at zero speed on the stationary wall, and at rotor speed on the rotor surface. This leads to an average speed roughly halfway between the two biased toward the stationary wall because the stationary wall surface area is greater than the shaft surface area. This is a simplification, since swirl can be significantly affected by other factors, such as leakage along a radial side passage prior to the close clearance radial gap. However, the leakage flows in this case suggested fluid swirl, and therefore rotor whirl in response to it, would be in the range of 43 to 49 percent of rotor speed, which is historically typical of rotordynamic instability in compressors and other types of turbomachinery.

A BRIEF EXPLANATION OF ROTORDYNAMIC INSTABILITY

Rotordynamic instability has been thoroughly discussed in the literature by various authors, such as Lund (1974), Ehrich (1987), Kirk (1985), and Ehrich and Childs (1984). It is generally explained as a condition in which the rotor cross-coupled stiffness, which acts perpendicular to the rotor radial movement and points in the direction of the shaft whirl, cancels out and overcomes the rotor damping, which also acts perpendicular to the rotor movement but points in the direction opposite to the rotor whirl. Since damping tends to decrease vibration (like in an automobile where the shock absorber offers damping to provide a smoother ride), and because cross-coupling acts in a direction opposite to damping, many have considered it as a form of "negative damping," increasing vibration rather than decreasing it. The threshold of where the cross-coupled stiffness force exactly cancels the damping force (as added up along the entire rotor) for whirl of a given mode would be the "threshold" of rotor instability. Once this threshold is crossed, the rotor modal vibration tends to grow without bounds, unless the increasing orbit causes the cross-coupling to decrease and/or the damping to increase so that the damping force once again dominates.

The threshold of rotor instability occurs at a given speed, known as the "threshold speed." At speeds above this threshold speed, the rotor vibrations grow without bound, usually at a vibration frequency equal to the first insufficiently damped natural frequency. Since damping force typically increases with speed at least as quickly as cross-coupling force does, one would think that as speed increases a rotor should be able to "run through" to a speed above the point at which damping and cross-coupling forces are equal, to a point where damping once again is dominant. The physical reason offered by the authors for why this does not occur is that when the whirl of the rotor, driven by the net fluid swirl forces surrounding it, is at a frequency that matches the rotor natural frequency, the rotor no longer responds to the force of swirling pressure fields as soon as those forces are applied. Instead, its response is delayed by a quarter turn of the rotor's whirl orbit, because of the well-known 90 degree phase lag between force and response that occurs when a rotor is excited at its critical speed. This means that the cross-coupled force, if it dominates over the damping, makes its existence felt when the rotor whirl close-clearance pinch-point rotates to the clock position where the cross-coupled force points exactly in the direction of the pinch, instead of being parallel to it. This will of course tend to close the pinch gap further. Since cross-coupled force is typically caused by the presence of a pinch gap in the first place and increases as the pinch gap tightens, the increased pinch situation leads to even higher cross-coupling. This feedback continues in a vicious spiral, in principle until the entire gap is used up, and the rotor is severely rubbing in its clearances. If cross-coupling dominates over damping forces, the unstable growth in vibration happens even if balance and alignment are nearly perfect, because movement causes force, causing more movement, causing more force, no matter how small the initiating force might be. Once the threshold speed is passed, this unhappy chain of events dominates any future vibration, and the rotor continues to self-excite itself at its critical speed, "locking on" to this critical speed even if the swirl forces would have otherwise continued to increase in frequency (typically at about ½ rotor speed).

Note that rotordynamic instability is fundamentally different from 1× resonance at a rotor critical speed, even though sometimes people will refer incorrectly to the high vibration, which occurs at a critical speed, as "unstable vibration." In a critical speed, the vibration may become high, but eventually reaches a limit depending on how much or little damping is present. Also, the rotor can be "driven through" a critical speed, so that, for example, 1× running speed moves above the shaft natural frequency, relieving the high resonant vibration since the oscillation of the force is no longer in tune with the natural vibration movement. In the case of a rotordynamic instability, once the "threshold" speed is reached, it cannot be "driven through," and the unstable growing vibration can only be lowered by quickly dropping speed to below the point where damping once again dominates over cross-coupling, typically when the roughly half rotor speed whirl drops below the self-excited critical speed.

SECOND TEST SERIES

Given how damaging rotordynamic instability can be, it was important to determine if instability was the key issue. If it was, certain types of specific fixes should be implemented to decrease the ratio of rotor system cross-coupled stiffness to damping, and/or to shift the rotor critical speed that was near 47 percent of running speed to a higher value. To decrease cross-coupled stiffness, swirl brakes could be installed in the labyrinth seals, since this would be expected to be the major source of cross-coupling in a tilting pad bearing supported rotor (tilting pad bearings have very little cross-coupling, unlike most other types of fluid film bearings). To increase damping, modified lubricant temperature or bearing clearances (to provide a more highly loaded pad) could be considered, although this would likely result in the already unacceptably high bearing temperatures becoming even higher. To significantly increase the critical speed of the impeller cantilever mode would require a redesigned rotor assembly, which appeared impractical from a field-retrofit point of view. On the other hand, if the problem were some other factor rather than rotordynamic instability, such as internal misalignment because of improper tolerancing, or a structural mounting critical speed, then an entirely different set of fixes would be required.

In order to get more conclusive proof of whether or not rotordynamic instability was responsible for the vibration trips, the authors decided to attempt to decrease the damping of the system by variations in oil temperature, to see at what point, if any, the dominance of cross-coupling forces would result in unstable subsynchronous vibration.

Oil Temperature Test 1

Throttling the water to the oil cooler was used to increase the oil supply temperature to the bearings. The oil temperature was increased from normal operating 115°F to a maximum of 128°F and then returned to the normal operating 115°F.

During this first oil temperature test, there was a significant vibration response to increasing oil temperature but the compressor did not trip. Each time the oil temperature increased, a response could be clearly seen in the trends. Please refer to Figure 3 for the vibration response as seen in the control room. In the middle of the test, the vibration peaked at 0.49 mils and settled down to 0.35 mils at 125°F oil supply temperature. As the oil temperature decreased to the normal operating value, 115°F, the vibration also returned to normal. Figure 4 shows the oil and bearing temperature variation during this test.

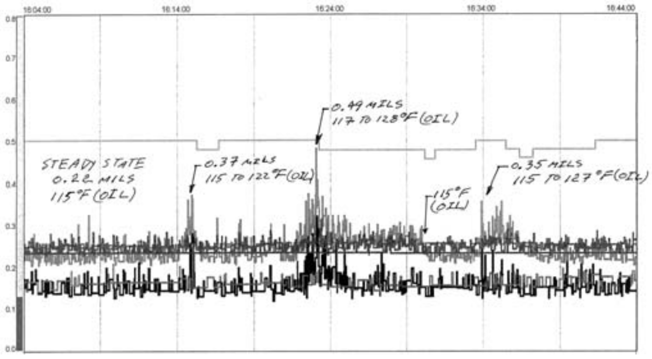

Figure 3. Vibration Response as Seen in Plant Control Room During Oil Temperature Test 1.

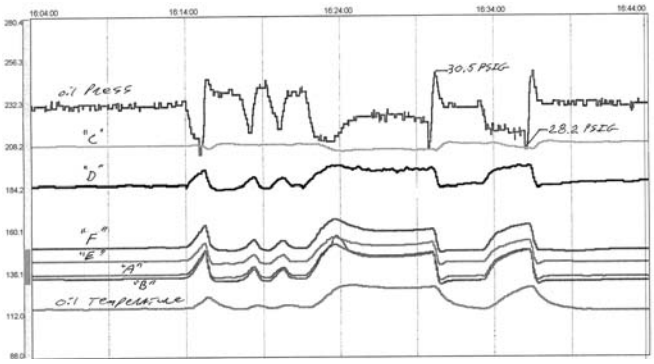

Figure 4. Oil Temperature as Seen in Plant Control Room During Oil Temperature Test 1.

The waterfall plot shown in Figure 5 shows this event as observed by the proximity probes. As can be clearly seen, variations in the oil temperature create a marked increase in the subsynchronous component of the vibration. From these spectra it is clear that a subsynchronous component at 221 Hz (13,260 cpm) is very strong, while the running speed is evident at 470 Hz (28,210 rpm). Figure 6 shows the overall vibration level versus time of the spectrum during this flare-up as seen at the proximity probes. This test helped to further convince the authors that the most likely cause of the excessive vibration at trip was a subsynchronous instability. In order to further validate this theory, additional oil variation tests were conducted.

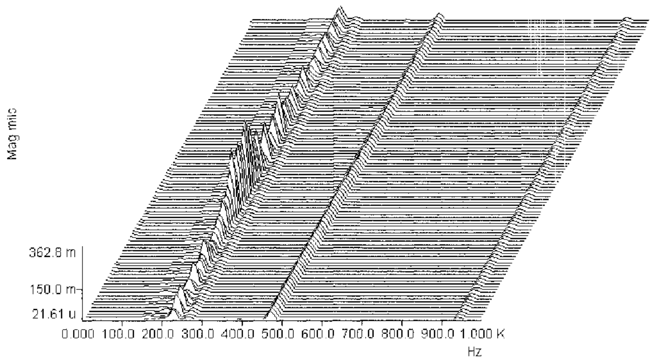

Figure 5. Waterfall Plot of Proximity Probe Response During Oil Temperature Test 1.

Oil Temperature Test 2

This was a cold oil temperature test. The three-way oil temperature control valve has a manual override that bypasses cold oil directly to the oil supply header. During this test, the oil supply

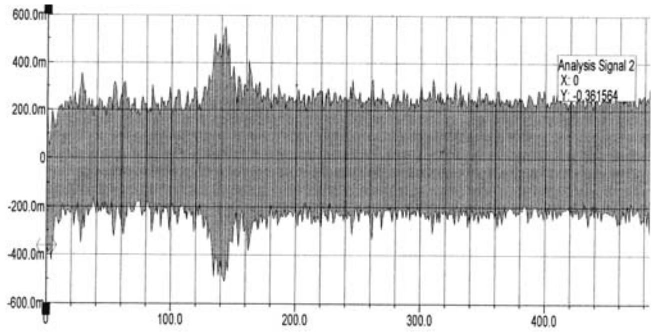

Figure 6. Overall Time Domain Response of Proximity Probe During Oil Temperature Test 1.

temperature became the oil temperature directly from the oil cooler. The oil temperature decreased from the normal operating value of 115°F to 94°F and was then returned to the normal operating value of 115°F.

There was an interesting vibration and bearing temperature response during this test. The bearing temperatures for rotors one and three (A, B, E, and F bearings) decreased 12°F to 18°F, showing a similar response to the oil supply temperature decrease of 19°F, as can be seen in Figure 7. The response for rotor two (C and D bearings) was decidedly different. The C bearing temperature actually increased, while the D bearing temperature decrease was barely 6°F.

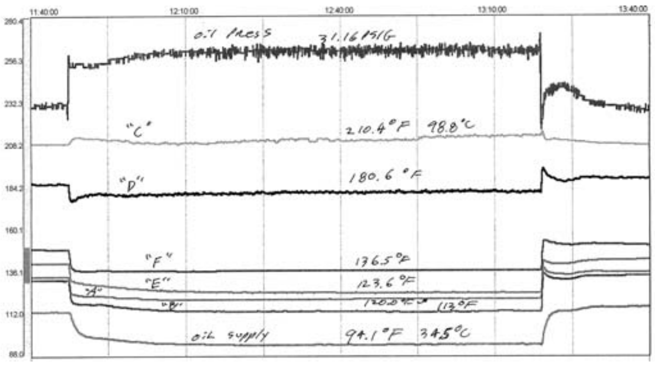

Figure 7. Oil Temperature as Seen in Plant Control Room During Oil Temperature Test 2.

In Figure 8, the CP14 first stage, D bearing, overall vibration amplitudes became progressively lower over time with spikes to 0.4 to 0.5 mils. The large vibration spike to 0.75 mils was due to a rapid oil temperature increase when the oil temperature control valve resumed normal operation. Figure 8 also shows the C bearing, same rotor as D bearing, vibration response. While the C bearing is much more stable, overall levels increased when the oil supply temperature decreased. This test, while interesting, did not provide conclusive evidence of whether or not instability was occurring, and so a third test was performed.

Oil Temperature Test 3

Because excitation of the subsynchronous vibration was observed during previous oil temperature increase tests, the high oil temperature test was repeated. As the oil temperature was increased from 111°F, the compressor tripped on CP14 first stage high vibration. The oil supply temperature was 122°F at the time of this trip. Figures 9 and 10 show the D bearing rotor vibration response during this test as seen in the control room. As the oil temperature increased above 120°F, there was an exceptionally large and rapid increase in vibration leading to the compressor trip. The total time for the system to be tripped was approximately 30 to 45 seconds from the beginning of the temperature change. Once

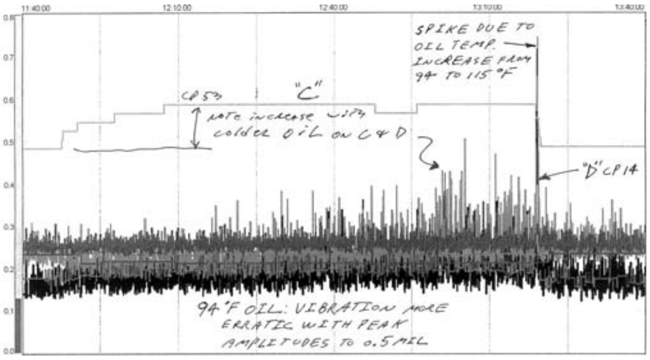

Figure 8. Vibration Response as Seen in Plant Control Room During Oil Temperature Test 2.

the rapid growth in vibration took place, it had the clear earmarks of rotordynamic instability, such as nearly all the vibration frequency occurring at 47 percent of running speed, and the orbit having a round "double loop" characteristic shape.

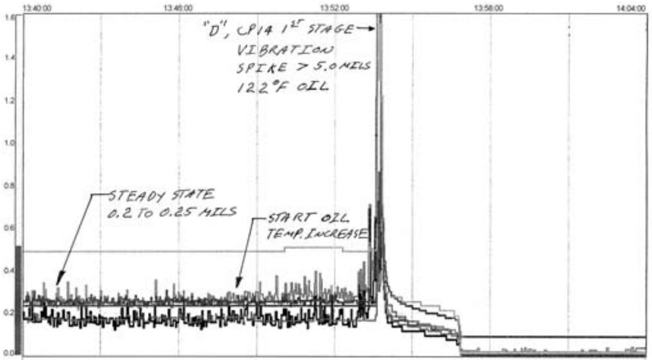

Figure 9. Vibration Response as Seen in Plant Control Room During Oil Temperature Test 3.

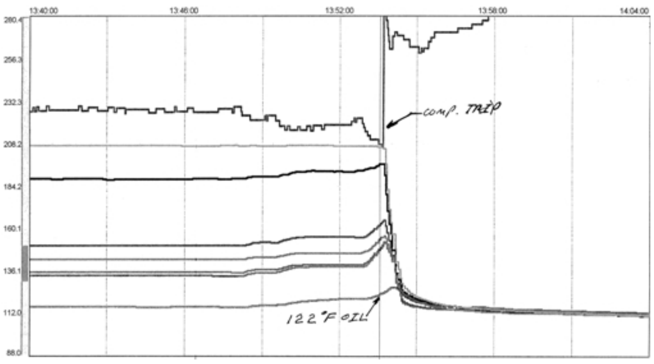

Figure 10. Oil Temperature as Seen in Plant Control Room During Oil Temperature Test 3.

The waterfall plot provided in Figure 11 clearly shows the sudden increase in the subsynchronous rotor vibration component, until the machine trips. Figures 12 and 13 show the orbits and vibration spectra just prior to trip. Figure 14 shows the overall response through time. As can clearly be seen, once the apparent instability begins, it increases the vibration amplitude exponentially. The vibration levels actually reached 6 mils radial motion (12 mils peak-to-peak) before the unit finally tripped. In this particular case, the high vibration trip event was preceded by a smaller increase in vibration very similar to the type seen during Oil Temperature Test 1. In this test however, the flare-up did not die down, but was followed by a more severe unbounded vibration resulting in trip.

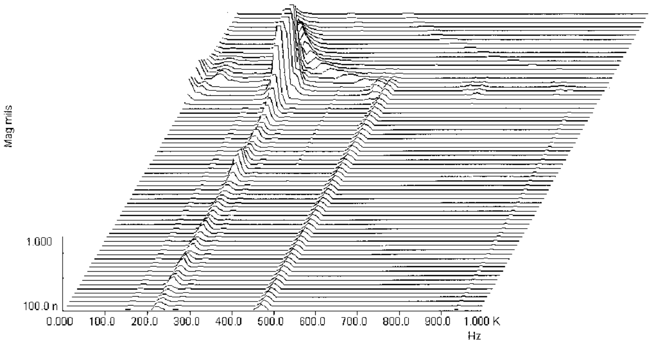

Figure 11. Waterfall Plot of Proximity Probe Response During Oil Temperature Test 3.

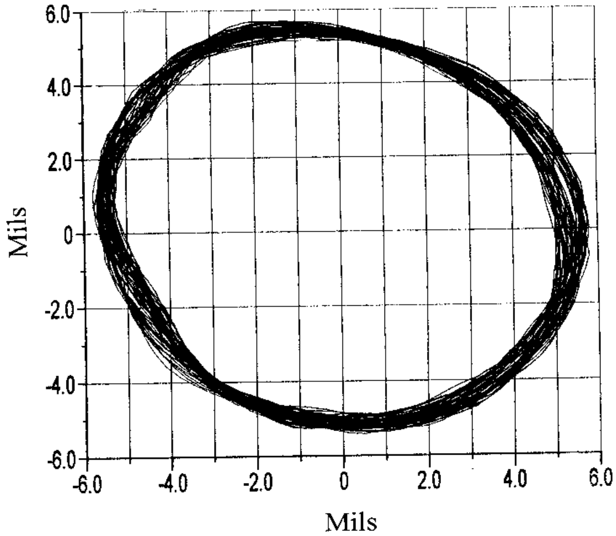

Figure 12. Orbit as Seen Immediately Before Trip During Oil Temperature Test 3.

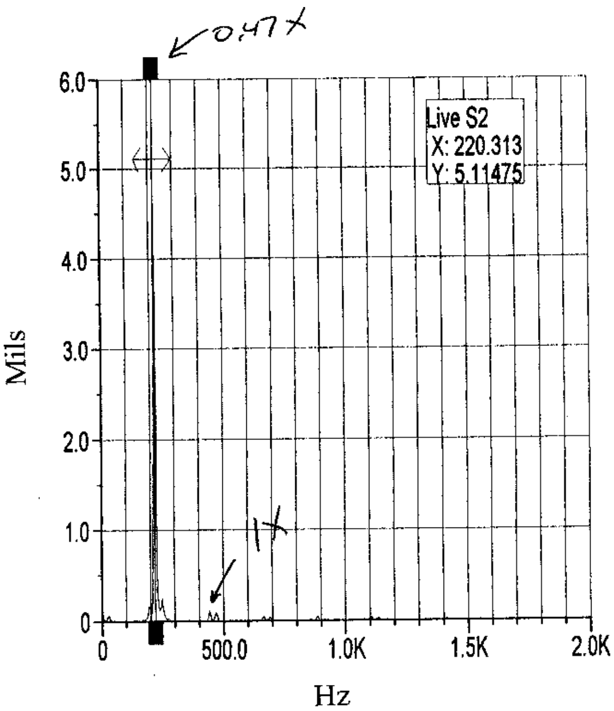

Figure 13. Vibration Spectra as Seen at Proximity Probe Immediately Before Trip During Oil Temperature Test 3.

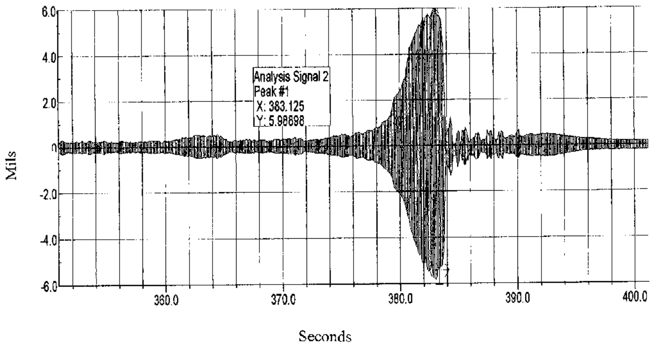

Figure 14. Overall Time Domain Response of Proximity Probe During Oil Temperature Test 3.

Vibration data during the compressor restart were also obtained. The restart was normal and vibration levels returned to steady-state conditions. It should be noted that the oil temperature was returned to normal operating levels and the delay on the vibration trip was shortened to two seconds from the previous four seconds in order to trip the machine well before any bearing damage might occur during future events.

EVALUATION OF TEST RESULTS AND DISCUSSION

As is clear by the summary presented thus far, a broad range of practical field testing was conducted to narrow the list of potential causes of the sudden vibration trips. Methodical testing eliminated most process variable issues such as IGV angle, sudden loss of flow, and antisurge control. The variable that had a strong and repeatable effect on the system was modification of the oil temperature of the bearings. The oil temperature variations categorically recreated and pinpointed the cause of the sudden trips. Based on this information, it was conjectured that the unusually hot days had most likely led to an increase in the bearing oil temperatures at the inlet to the bearing, and reduced the damping of the bearing enough to cause seal cross-coupling forces to exceed the damping forces, allowing the subsynchronous frequency to trip the machine out.

Therefore, results of the testing made it very clear that some type of classic fluid whirl phenomenon was at work in an unstable fashion. Because of the use of tilting pad oil journal bearings, which are whirl-resistant by nature, it was concluded that the whirl was not from the bearings themselves but rather from some other close clearance location. In this case the most reasonable source was the labyrinth seals of the machine. Evaluation of the shaft/bearing system with inclusion of a "nominal" estimate of the effect of cross-coupling of the labyrinth seals indicated that rotordynamic instability was indeed possible, even likely, confirming the results of testing as seen in the field. Without detailed information of the labyrinth and impeller geometry, a nominal amount of 20,000 lb/in of cross coupling was chosen. Use of the well-known Wachel equation with estimates of impeller geometry resulted in a calculated cross coupling of 20,700 lb/in. Figure 15 shows the predicted stability results of the OEM versus that of independent analysis. Table 4 summarizes the predicted stability results for variations in temperature. The analytical results support the oil temperature variations seen in the field. Specifically that increases in bearing oil temperature drive the system more unstable.

Nevertheless, the presence of a labyrinth-induced instability on a machine of this type seemed somewhat unusual considering that the pressure across the stage in question is not extraordinary. The suction is at 280 psig and discharge is at 453 psig. Perhaps the more telling parameter is the flexibility ratio of the rotor (flexibility ratio = max operating speed/1st critical speed). In this case the flexibility ratio is 2.14. The work of Kirk and Donald (1983) and Fulton (1984) is cited as a good overview of the importance of

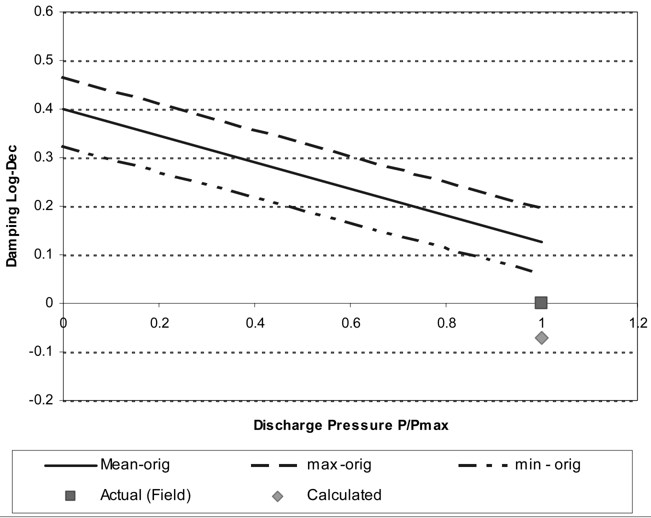

Figure 15. Predicted Stability of Unmodified System by OEM and Independent Analysis.

Table 4. Analytical Stability Results for Varying Bearing Oil Temperatures.

Bearing Inlet Temperature °F	Damped Natural Frequency (cpm)	Log Dec Calculated	Tmax °F
110	12586	-0.069	199
120	12599	-0.076	211
130	12612	-0.081	228

flexibility ratio and average discharge density across a machine. In the case of the machine at hand, direct comparisons to the empirical results presented by Fulton (1984), and Kirk and Donald (1983) are qualitative at best because this is a double overhung machine, whereas most of the previous investigations involved multistage between-bearing designs. In either case, flexibility ratio and average gas density across the stage still would be expected to indicate overall stability. Specifically the higher the flexibility ratio and the higher the gas density, the greater the likelihood of high cross-coupling causing stability problems.

While firm conclusions cannot be drawn from a single case, the key findings of Kirk and Donald (1983) and Fulton (1984) appear to hold true not only for between-bearing machines but also for a compressor with overhung or cantilevered rotors. In the case of overhung rotors, the location of the labyrinth eye seals at the end of the rotor is at the optimal location to excite cantilevered modes of the rotor. In the case of the machine investigated here, examination of the rotordynamic analysis predicts cantilevered modes at 12,500 cpm (208 Hz) and 13,500 cpm (225 Hz) (refer to the forced response analysis of Figures 16 and 17). It appears that in the case of the machine at hand, the labyrinth seals served to excite the cantilevered mode at 221 Hz (analytically predicted at 208 Hz). The stability results summarized in Table 4 further point to the presence of a lightly damped unstable mode at 12,599 cpm, which is coincident with the first cantilevered mode of the rotor. Figure 18 depicts the forced response mode shape of the critical speed at 12,500 rpm corresponding to the damped mode predicted at 12,599 cpm by the stability analysis. The possible excitation of the critical speed was not obvious when looking at the manufacturer's Campbell Diagram, as the diagram only considered the separation margin for the critical speed closest to running speed. In the present case, this was a mode occurring at 19,000 cpm. However, when examining the analytical forced response analysis, it could be clearly seen that the likely subsynchronous whirl excitation frequency (historically 43 to 49 percent of running speed) and the first critical were nearly coincident. This brings forth an important evaluation criterion when examining cantilevered machine configurations. Specifically, the location of the cantilevered modes, in relation to possible 43 to 49 percent excitation frequencies must be closely examined. Simply meeting API separation margins for synchronous excitation and integer multiples of running speed is not adequate.

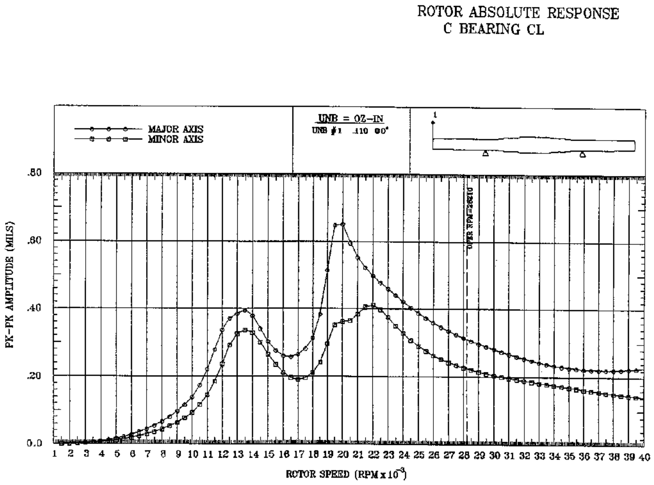

Figure 16. Analytical Forced Response of Unstable Rotor (A).

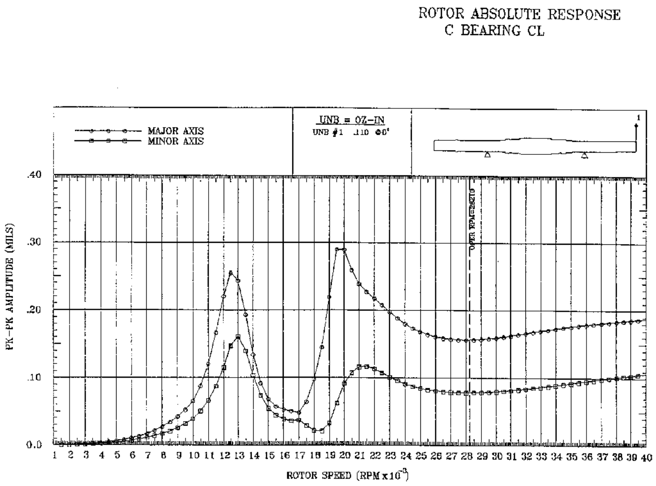

Figure 17. Analytical Forced Response of Unstable Rotor (B).

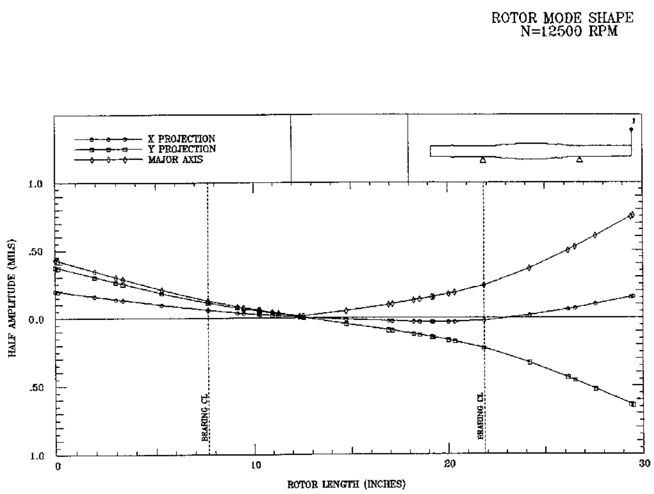

Figure 18. Analytical Forced Response Mode Shape.

DESIGN CHANGES

At the request of the authors, the OEM incorporated so-called "swirl brakes" to help disrupt the circumferential gas flow that causes the cross-coupling problems discussed above. The OEM was very cooperative and responsive to incorporating swirl brakes once it was agreed upon that the cause of the trips was a subsynchronous excitation. The swirl breaks were installed at impeller labyrinth eye seal locations on stages C, D, E, and F. The predicted stability margins of the OEM with the inclusion of swirl brakes are shown in Figure 19.

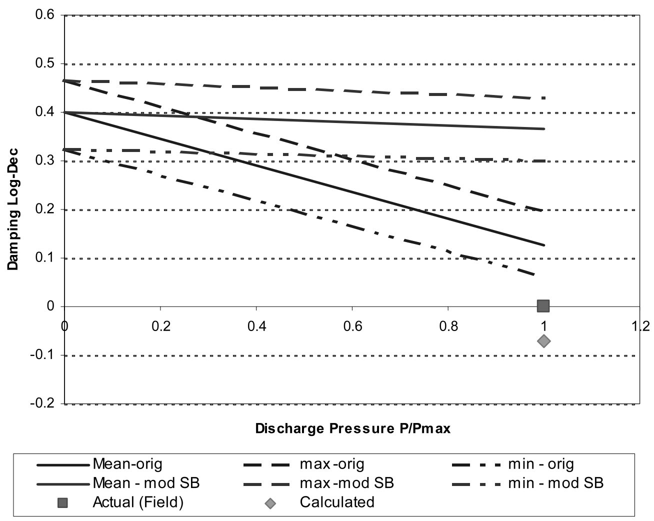

Figure 19. Predicted Stability of System with Inclusion of Swirl Breaks.

Once the swirl brakes were installed, design changes to the bearings could be implemented to reduce the bearing pad temperatures. These design changes involved chamfering the trailing edge of the bearings and thereby effectively reducing the pad arc lengths from 58 to 53 degrees. This change successfully reduced the pad temperatures as shown in Table 5.

Table 5. Bearing Instrumented Pad Temperatures Before and After Modification.

	Before Bearing Modification °F	After Bearing Modification °F
CP53 Stage 1, A	208	193
CP53 Stage 2, B	215	182
CP53 Stage 3, C	219	173
CP14 Stage 1, D	256	167
CP14 Stage 2, E	212	172
CP14 Stage 3, F	209	152

CONCLUSIONS

Normally manufacturers evaluate separation margin and may only look at the critical speed closest to running speed. It is equally important to evaluate other critical speeds to determine if they are in the whirl frequency range of 43 to 49 percent of running speed.

Another important point is that the use of tilting pad bearings does not guarantee stability. The addition of swirl brakes on stages C, D, E, and F was required to reduce the cross-coupling excitation to a point at which a bearing modification could be made to locations C and D without compromising the stability of the system. The final bearing modification, consisting of chamfering the trailing edge of each of the bearing pads, has resulted in a substantial reduction in bearing pad temperatures.

Most users are well aware of the potential for oil whirl instabilities present in nontilting journal bearings, and the fact that tilting pad style bearings tend to virtually eliminate the cross-coupling forces that cause this subsynchronous instability. This beneficial attribute of tilting pad bearings has been misinterpreted by many users to mean that the use of tilting pad bearings will guarantee stability. However the presence of substantial cross-coupling forces from labyrinth seals can still overcome the damping of tilting pad bearings and, therefore, this possibility needs to be closely examined. In the case of the machine investigated here, variations in oil temperature were enough to reduce the damping in the bearings to the point that the labyrinth cross-coupling forces could cause the system to run in unstable operation.

Perhaps the most important thing to note is that stability problems are not just limited to high-pressure machines as typical in petroleum reinjection service for example, but can also occur in flexible rotor systems with lower pressure stages. The combination of increasing flexibility ratios, increased speeds, and increased pressure rises has made the threat of instabilities in cantilevered machines a very real possibility. It is particularly important to closely examine the location of the first two cantilevered modes in comparison to the running speed range. As running speeds have increased and rotors have become more flexible, the moderately damped cantilevered modes of the high-speed pinions of integrally geared compressors can tend to drop into the range of 43 to 49 percent of the running speed of the machine, at which point these modes become prone to rotordynamic instability. The cantilevered impellers are the optimum location for cross-coupling forces from labyrinth swirl to reexcite the natural frequency and cause a substantial instability problem.

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