DETERMINING THE ROOT CAUSES OF SUBSYNCHRONOUS INSTABILITY PROBLEMS IN TWO CENTRIFUGAL COMPRESSORS

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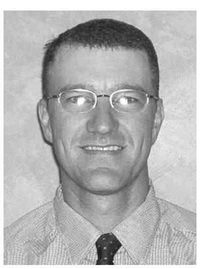

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ABSTRACT

This paper discusses the rotordynamic instability problems experienced with two separate centrifugal compressors. While the root causes of the instabilities are very different, the analysis methodology of reconciling the rotordynamic model with measured vibration data was the same.

The first problem occurred with a propylene compressor in a Gulf Coast chemical plant that had experienced high journal bearing temperatures for several years. A modified bearing was installed to alleviate the temperature problem; however, a large subsynchronous vibration appeared after the new bearings were installed. A lateral stability analysis showed that the compressor with the modified bearings was very stable with the aerodynamic destabilizing effects predicted by the Alford and/or Wachel equation. A comparison analysis was made of the stability predicted with the original bearings (which were stable) as well as the modified bearings (which were not). This allowed the user to determine the magnitude of the destabilizing forces present in the compressor and design a new bearing that was both stable and would operate at an acceptable temperature. The new bearing was installed and the compressor has operated without the subsynchronous vibration for the past year and a half.

The second problem occurred with a very similar ethylene compressor in a Midwest ethylene plant. This compressor had operated for over two years after an overhaul with low vibration. Then a subsynchronous vibration appeared that was very erratic, but was slowly increasing in amplitude over time. To solve the problem, a rotordynamic analysis was performed that suggested that replacing the bearing would solve the stability problem. However, comparison between the measured field vibration and the rotordynamic model did not agree on all points. A more indepth look at the compressor revealed that the increase in subsynchronous vibration was tracking very closely with the balance line differential pressure. This fact, along with the characteristics of the balance piston seal, suggested that a bearing change alone may not completely address the problem. A new balance piston seal was designed to reduce its destabilizing effects on the rotor. The compressor was inspected during the next scheduled downtime to determine the cause of the high vibration and install the new balance piston seal and bearings. Examination of the internals revealed a large seal rub in the compressor, but at the dry gas seals, not the balance piston. The rub was addressed and the subsynchronous vibration was eliminated. While the exact source of the destabilizing force was not correctly "guessed" before disassembly, the indepth rotordynamic analysis did reveal that there was a large destabilizing force in the compressor, and a bearing change alone would not eliminate the vibration. The compressor has been operating without the subsynchronous problem for the past year since the modification.

CASE STUDY 1—GULF COAST PROPYLENE EXPORT COMPRESSOR

Background

The first compressor is a horizontally split, eight-stage, intercooled compressor in a Gulf Coast chemical plant that pumps propylene gas from 30 to 280 psig. The compressor operates between 10,000 and 12,500 rpm and is driven by a 3000 hp turbine through a speed-increasing gearbox. The original bearing design was a five-pad load-on-pad (LOP) bearing with center pivots and relatively tight clearance (0.004 to 0.005 inch on a 3 inch journal). In October of 2000, the compressor was overhauled as part of a normal unit turnaround. After the overhaul, the radial bearing temperatures were excessively high (~220°F) and would rapidly spike to over 300°F at times. This compressor had experienced high bearing temperatures in the past as well. To solve the problem, the bearing design was changed to a load-between-pad (LBP) design with offset/spherical pivots and higher radial clearance (Figure 1). The analysis of the new bearing design predicted that the bearing temperature would be lowered by at least 25°F. Likewise, the rotordynamic analysis of the compressor with the new bearing predicted that the synchronous vibration would be low with the new bearings (Figures 2 and 3). A lateral stability analysis was not performed by the contractor, mainly because the sole objective of the bearing design was to lower the bearing temperature, and because the vibration of the compressor had always been so low. Likewise, a stability analysis had been performed during the conversion to dry gas seals in 1994, which showed that the compressor rotor was very stable (Figure 4). Since the calculated aerodynamic cross coupling for the entire rotor was approximately 4300 lbf/in, the compressor was considered stable. These new bearings were installed in May of 2001. The radial bearing temperatures were at acceptable levels (< 175°F) after the bearing modification; however, the radial vibration increased from 0.3 mils to 3.5+ mils. Additionally, most of the vibration was at a subsynchronous frequency that had not been present before. This frequency coincided with the rotorbearing system's first natural frequency.

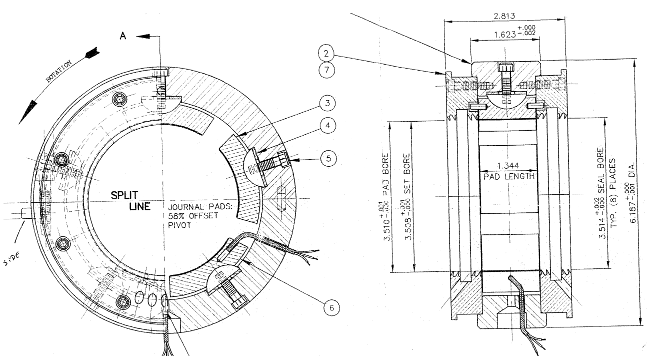

Figure 1. New LBP Bearing Installed in Compressor in 5/20/01.

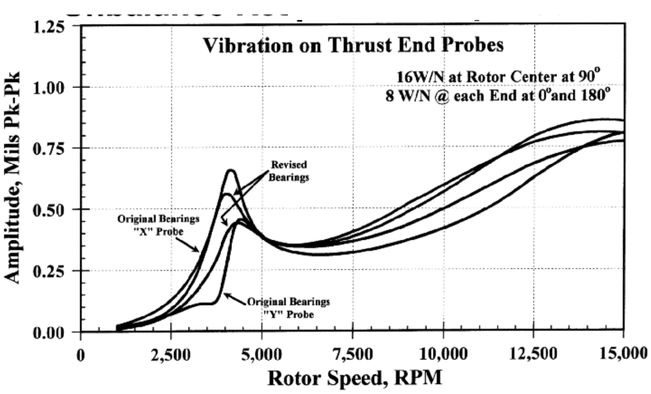

Figure 2. Predicted Thrust End Synchronous Response with New Bearings.

Description of the High Vibration Problem

Even though the temperature of the modified bearings was low, the shaft vibration on the compressor was much higher than

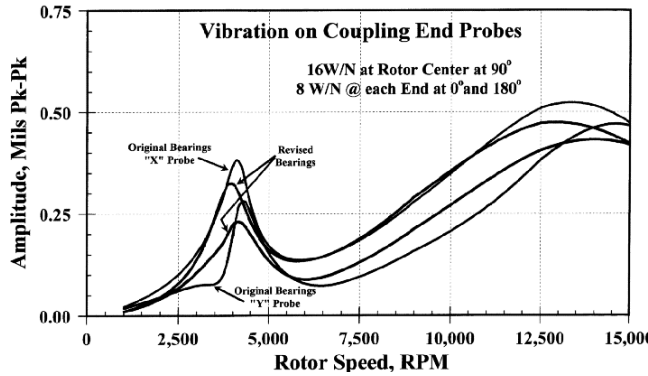

Figure 3. Predicted Coupling End Synchronous Response with New Bearings.

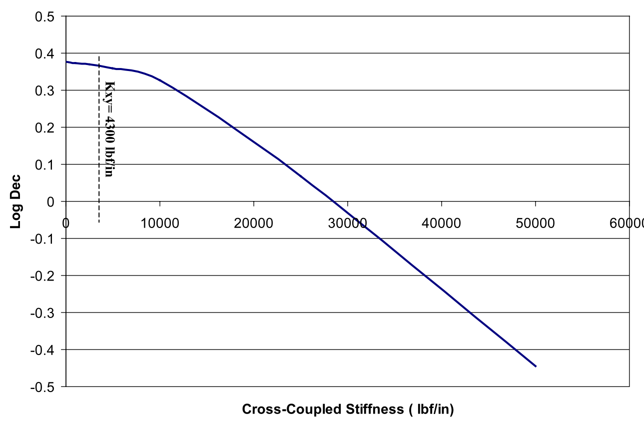

Figure 4. Effect of Aerodynamic Cross-Coupling on Stability from Analysis in 1994.

expected (Figure 5). A spectrum of the vibration revealed that the largest portion was a subsynchronous component, at approximately 4100 cpm (Figure 6). This subsynchronous component would begin increasing around noon, reaching a maximum around 7:00 p.m. (refer to overall and subsynchronous components in Figures 7 and 8). This frequency corresponded to the first lateral mode, which was confirmed from the transient data recorded on startup, as well as the rotordynamics study (Figures 2 and 9). As expected, the first critical speed had shifted down approximately 500 to 700 cpm after the bearing modification, in comparison to that seen during the coastdown with the original bearings (i.e., the new bearings with higher clearance and reduced stiffness lowered the critical speed). Also, note the rapid increase in synchronous amplitude at running speed (i.e., the far right-hand side of Figure 9). This appears to be a result of the compressor operating so close to its second critical speed (Figure 10).

No correlations between process or lube oil conditions could be found that explained the erratic change in vibration. The compressor was not operating close to a surge condition in either section. Likewise, the compressor flows, temperatures, and pressures did not change with the vibration. The only correlation with the vibration that peaked at approximately 7:00 p.m. was sunshine on the compressor/gearbox/baseplate. Several observations determined that the sun would set behind several distillation towers at approximately this time of the day and shade the compressor. It was concluded that this might be slightly affecting the alignment and/or oil temperature. Obviously, the possible changes in alignment were not drastic because there were no indications in the shaft orbit or fast Fourier transform (FFT) that would indicate misalignment. Likewise, the supply oil temperature varied by less than 5 degrees. However, it did indicate that the compressor was very sensitive to small changes (i.e., marginally stable). Furthermore, to determine if the unstable

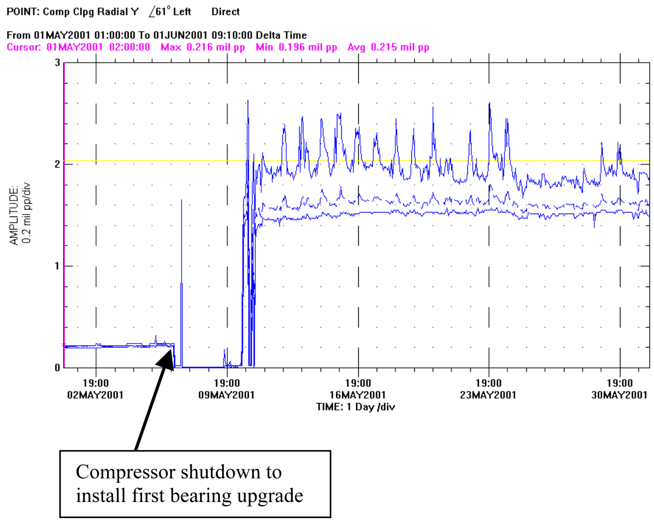

Figure 5. Compressor Inboard Radial Vibration before and after First Bearing Upgrade.

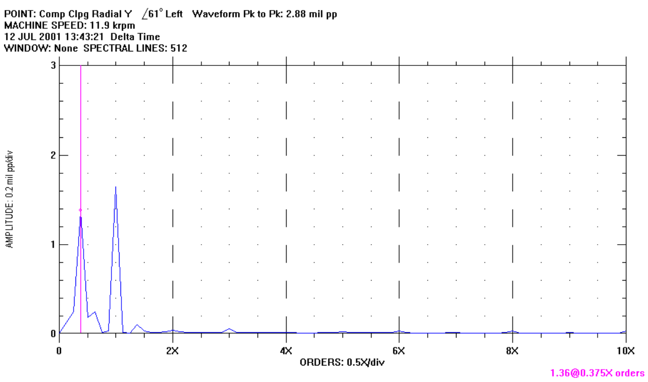

Figure 6. Spectrum of Compressor Inboard Radial Vibration Showing Large Subsynchronous Component.

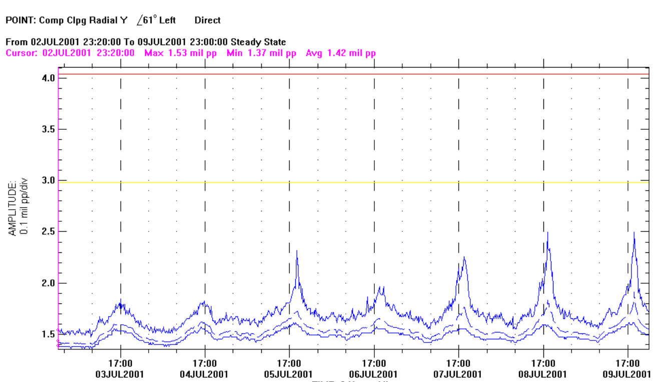

Figure 7. Erratic Nature of Compressor Inboard Radial Vibration, Overall.

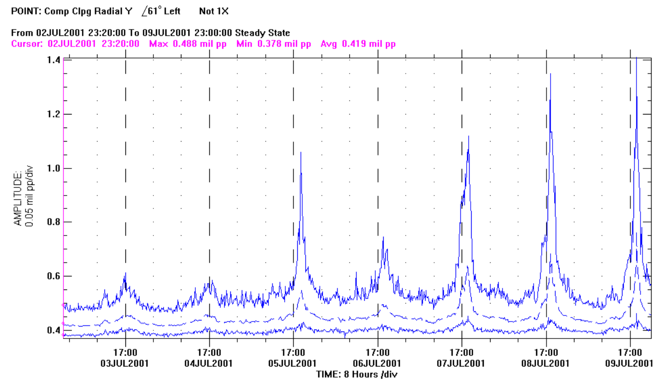

Figure 8. Subsynchronous Component of Compressor Inboard Radial Vibration.

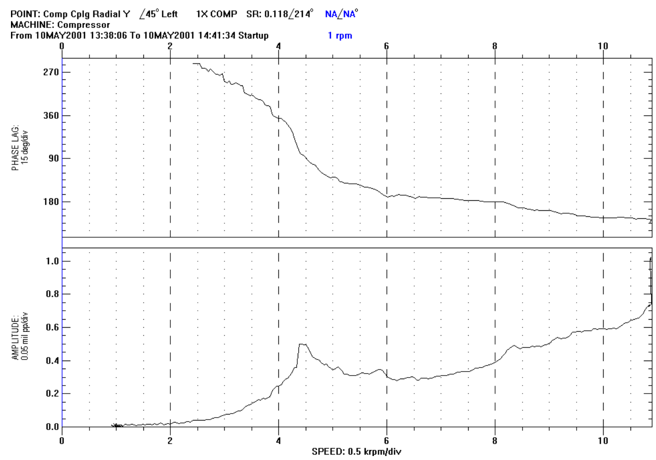

Figure 9. Bodé Plot of Compressor Inboard Radial Vibration During Startup after Bearing Modification.

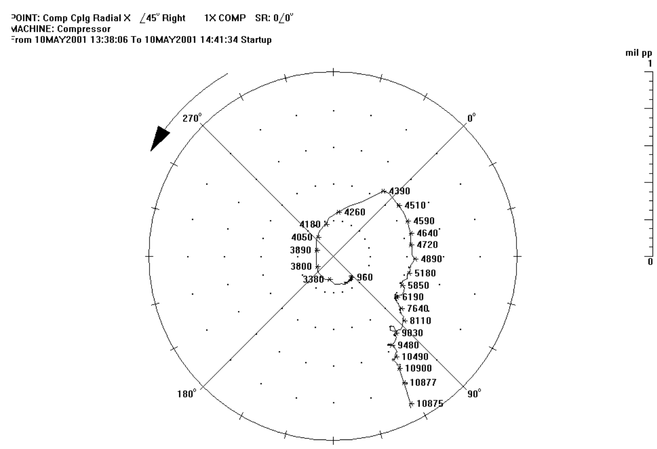

Figure 10. Nyquist Plot Showing Rotor Is Approaching its Second Critical (I.E., 360 Degree Phase Shift).

condition of the compressor could be attributed to load (i.e., horsepower), on 7/12/01, the speed of the compressor was increased from 11,000 to 12,200 (Figure 11). The radial vibration jumped from an overall value of 1.6 to 3.3 mils, most of which was the subsynchronous component. The drive end shaft orbits before and during the speed increase are shown in Figures 12 and 13. As can be seen, the internal loops are representative of a large subsynchronous vibration at 1/3 of shaft speed. The speed was lowered back to 11,000 rpm, and the vibration dropped back down to its previous level. This further confirmed that the problem was indeed stability related.

Rotordynamic Stability Analysis

The erratic behavior of the shaft vibration indicated that the rotor was only slightly unstable, since the vibration did not continue to grow unbounded. A rotor model was built for the compressor to determine the cause of the high vibration. The validity of the model was checked against the free-free modal vibrations previously measured on the spare rotor. This confirmed that the model accurately predicted the stiffness of the compressor rotor without the bearing effects.

The stability of the first lateral mode, calculated without any seal effects, showed a very positive logarithmic decrement (ld) (i.e.,

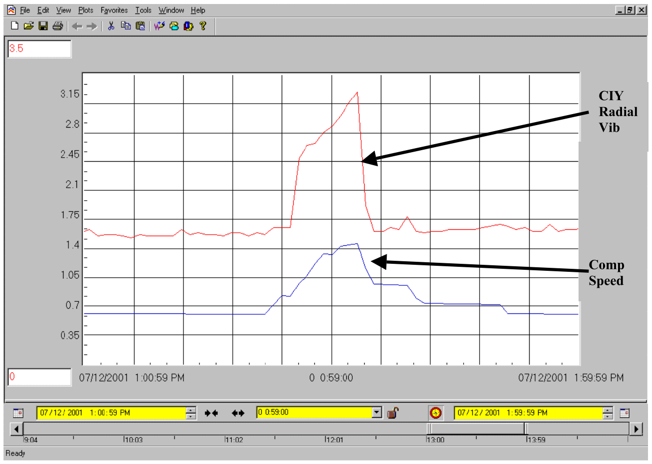

Figure 11. Compressor Inboard Radial Vibration During Speed Increase on 7/12/01.

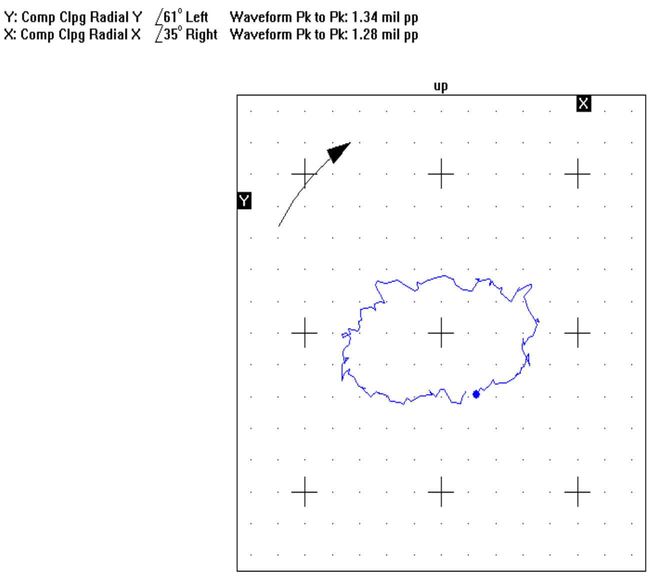

Figure 12. Shaft Orbit at 10,900 RPM (Before Speed Test).

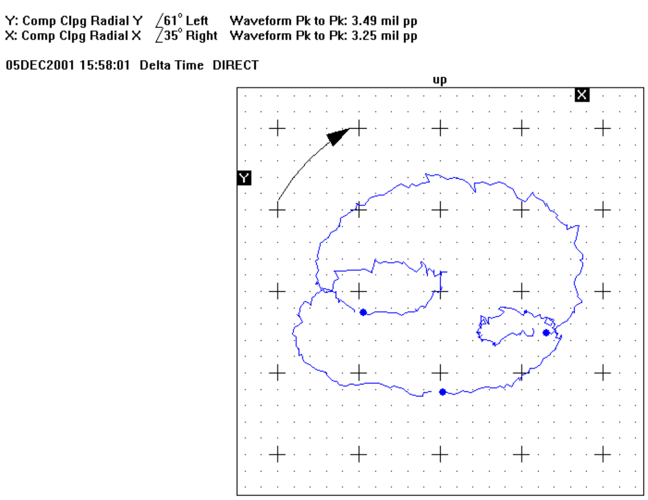

Figure 13. Shaft Orbit at 12,000 RPM (During Speed Test).

large degree of stability, Figure 14). Notice that the calculated ld of 0.32 corresponds closely with the value shown in Figure 4 on the far left of the curve. However, this did not correspond to the erratic subsynchronous vibration seen in the field. Since tilting pad bearings do not produce significant destabilizing forces, the rest of the compressor was examined. The most significant rotor support in the compressor, besides the bearings (because the compressor has dry gas seals), is the center labyrinth seal that separates the discharge of the two compression sections (Figure 15). This seal is a rotating labyrinth design with an abradable stationary. Even though this was a low pressure application, this seal was suspected as the source of instability, since rotating labyrinths typically produce more cross-coupled stiffness than stationary labyrinths. Likewise, this seal was in the center of the rotor, where it would have the most effect on the first mode (i.e., much more displacement at the center of the rotor). The dynamic coefficients of the center labyrinth seal were calculated using a two control volume bulk flow model (Figure 16). Adding the seal coefficients to the rotor model with the new bearings causes the calculated log decrement of the first mode to drop considerably (Figure 17). However, these calculated coefficients seem to be unusually large because, if the seal coefficients are added to the model with the old style bearing, it also predicts an unstable system (Figure 18). Since this compressor has a long history of operating with low vibration, this indicated that either the model for the center labyrinth seal was incorrect or the destabilizing effects in the compressor had changed since the new bearings were installed.

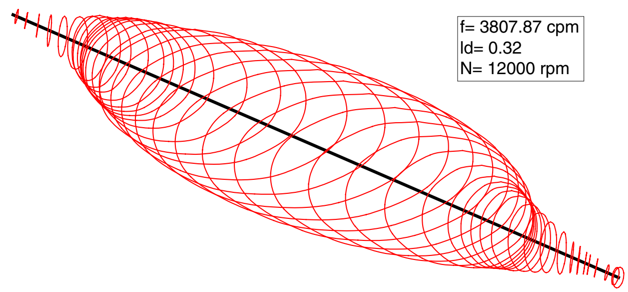

Figure 14. Calculated First Mode Shape Without Seal Effects.

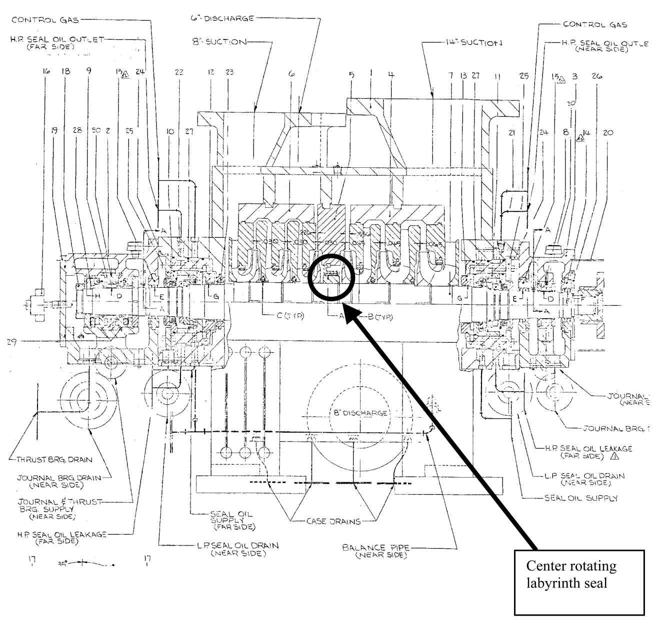

Figure 15. Cross-Section of Compressor, Showing Center Seal.

The center labyrinth seal might have been damaged since the bearing modification, which would have caused it to produce more destabilizing forces on the rotor. The most likely cause of this would be a rub during startup of the compressor.

```
* Case # 1: test run 11000 rpm
*
* Tooth on Rotor Labyrinth Analysis
* Operating Analysis Point
*
*         Radial Clearance           =     .0030 in.
*         Inlet Rotor Diameter       =    6.4920 in.
*         Tooth Pitch                =     .1450 in.
*         Tooth Height               =     .0610 in.
*         Tooth Tip Width            =     .0200 in.
*         Cavity (Depth-Width)       =     .0050 -     .0200 in.
*         Number of Teeth            =        13
*         Shaft Speed                =    11000. rpm
*         Inlet Swirl                =       .60
*         Supply Pressure            =    270.00 psia
*         Exit Pressure              =    120.00 psia
*         Supply Temperature         =    200.00 F
*         Supply Viscosity           = .739E-05 lbm/ft/s
*         Ratio of Specific Heats    =     1.180
*         Gas Constant               =      .047 BTU/lbm/R
*         Compressibility Factor     =       .97
*
*
* The FLOW IS UNCHOKED
* THE MASS FLOW RATE IS      .111 lbm/s
*
* THE WHIRL FREQUENCY RATIO IS    1.00
*
* KXX        KXY        KYX        KYY        CXX        CXY        CYX        CYY
* (lbf/in)                                    (lbf-s/in)
*
 -.115E+05  .327E+05 -.327E+05 -.115E+05   28.5       3.77      -3.77       28.5
```

Figure 16. Center Seal Rotordynamic Coefficients.

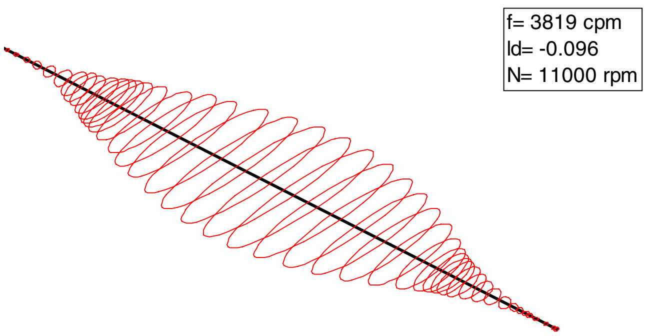

Figure 17. Calculated First Mode Shape with Center Seal Effects (New Bearings).

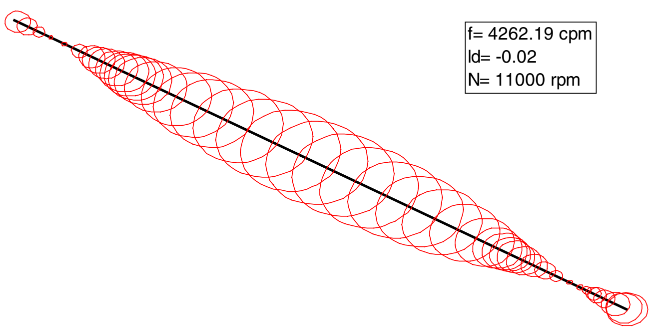

Figure 18. Calculated First Mode Shape with Center Seal Effects (Old Bearings).

Damaged Center Labyrinth Seal?

The most likely damage to the center labyrinth seal would occur during startup of the compressor on 5/10/01. Since it is a rotating labyrinth seal with an abradable stationary, the rub would not be as detrimental as a stationary aluminum labyrinth against a steel sleeve. To determine if the center seal did rub, the measured synchronous response, first mode shape, and predicted synchronous response must be used. As can be seen in Figure 9, the synchronous amplitude measured on startup at 11,000 rpm is approximately 0.6 mils peak-to-peak. This is very comparable to the amplitude predicted in the calculated synchronous response (Figure 3). This verifies that the assumed unbalance is close to the actual. Likewise, the predicted synchronous response at the center seal is shown in Figure 19. As can be seen, the maximum synchronous amplitude is approximately 1.1 mils peak-to-peak. Since the diametrical clearance of the center seal is 0.006 inch, it seems unlikely that it rubbed on startup of the new bearings. Another interesting note is that the synchronous response at the center seal with the original bearing is actually higher (Figure 20). While at first this seems incorrect, it is important to remember that the amplification factor of a rotor system is a strong function of the bearing to shaft stiffness ratio. As the bearing stiffness goes up for the same rotor, the amplification factor goes up as well. Since the original bearings had a much higher stiffness, this increased the amplification factor (even though the clearance was lower). However, it is unlikely that the center seal rubbed with the old bearings as well because the maximum amplitude is 1.6 mils peak-to-peak, plus there were no vibration problems prior to the bearing modification.

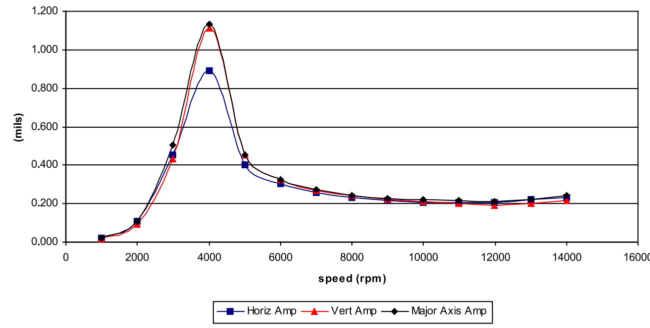

Figure 19. Forced Synchronous Response at Center Seal with Modified Bearing.

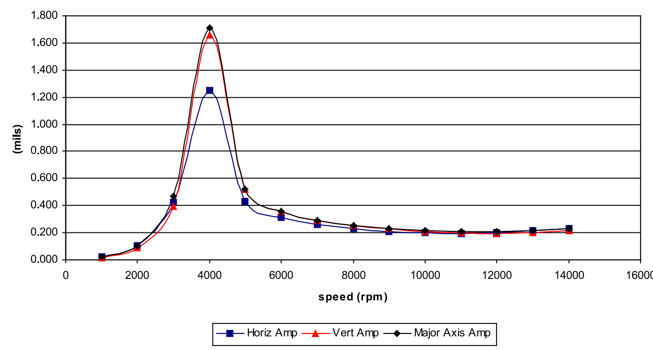

Figure 20. Forced Synchronous Response at Center Seal with Original Bearings.

A stability map of the compressor with the old LOP bearings and the new LBP bearing is shown in Figure 21. Based on these calculations, it seems reasonable that the stiffness coefficients used for the center labyrinth seals are too large, since there were no stability problems prior to the bearing change. Therefore it seems reasonable to assume that the cross-coupling produced by the center seal was approximately 15,000 to 18,000 lbf/in (i.e., the point on the horizontal axis where the existing LBP bearing curve goes negative, but the original LOP bearing curve is still positive).

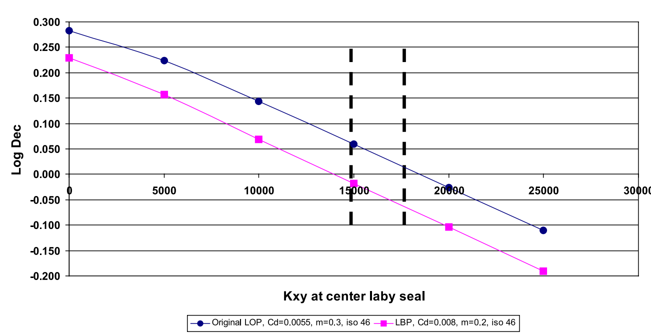

Figure 21. Compressor Stability Map for Prior LOP and Current LBP Bearings.

Further review of the results from the labyrinth seal model showed that the calculated whirl frequency ratio ($K_{xy}/\omega C$) is equal to 1.0 (Figure 16). Experimental results have shown that the whirl frequency ratio should not be higher than the inlet velocity ratio (Childs, 1993). Since, this was a teeth-on-rotor seal, the inlet velocity ratio was assumed to be 0.6. Setting the whirl frequency ratio equal to 0.6 and solving for K_{xy} results in the corrected value of 19,000 lbf/in. This is very close to the range shown in Figure 21 above.

Second Bearing Modification

LOP bearings can be more stabilizing than LBP designs because of the asymmetric stiffness they produce. LOP bearings tend to produce more elliptic orbits, which reduces the amount of energy and/or torque produced by cross-coupled forces (Vance, 1988; Zeidan, 1991). LBP bearings produce more circular orbits because of the symmetric stiffness coefficients, which are more susceptible to cross-coupled forces. While LOP bearings are not always more stable than LBP, they appear to be for this particular machine. The drawbacks of the LOP design are:

- The difference in stiffness between the two axes can be large enough to cause a split critical.
- The LOP does not produce as much direct damping as the LBP, which may cause the synchronous response to be higher.
- For LOP bearings, most of the load is supported by one pad instead of two, which can cause higher bearing temperatures.

The decision was made to replace the LBP bearings with a LOP design that had a larger clearance than the original LOP. The proposed design was a five-pad LOP design, with 0.007 inch set clearance, 0.3 preload, and ampcolloy (copper with 1 percent chrome for strength) pads with a center pivot. The calculated coefficients and stability map for this design are shown in Figures 22 and 23. Likewise, forced synchronous response at each bearing and the rotor midspan are shown in Figure 24, 25, and 26.

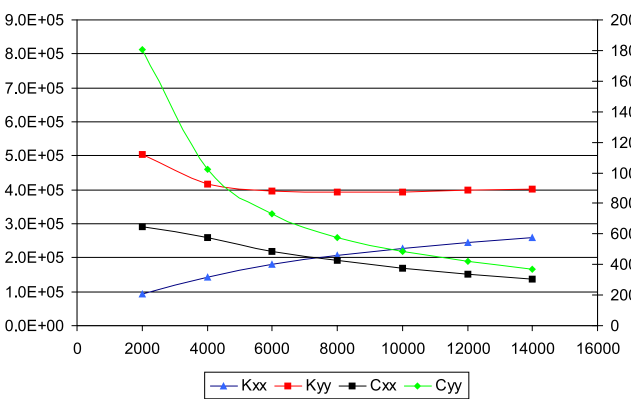

Figure 22. Bearings Coefficients of Five-Pad, LOP Design with Cd = 0.006 and M = 0.3.

A 58 percent offset pivot was considered for the LOP design to lower the operating temperature; however, the offset pivot causes the principal stiffness to increase dramatically, with very little increase in damping (Figure 27). The net result is a decrease in stability in comparison to the center pivot design (Figure 28). The ampcolloy pad material helps to compensate for this change because its thermal conductivity is approximately 4× that of bronze and 4.5× that of steel. The calculated maximum babbitt temperature with copper and bronze pads is shown in Figure 29.

In addition, this latest lateral analysis revealed why the original compressor original equipment manufacturer (OEM) bearing clearances had been so low. There is no doubt that the OEM knew that this compressor operated very close to its second natural mode. This mode is not considered a critical speed (by APIs definition because

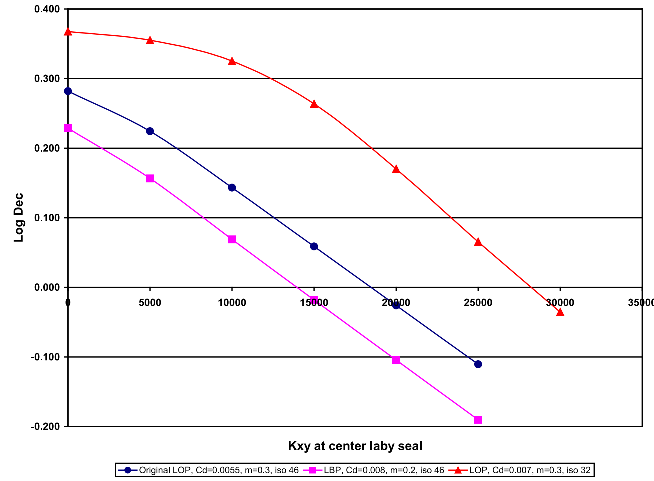

Figure 23. Stability Map Showing Prior LOP, Current LBP, and Proposed LOP Bearing Designs.

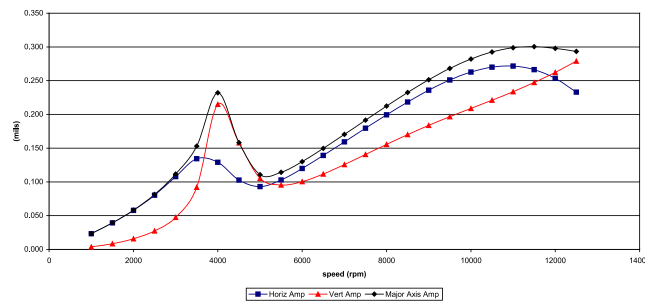

Figure 24. Thrust End Radial Bearing Response with Proposed LOP Bearing.

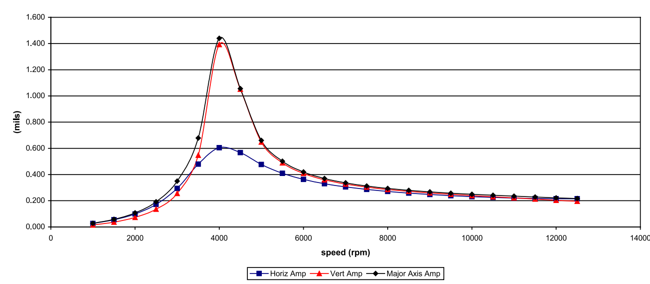

Figure 25. Response at Center Labyrinth Seal with Proposed LOP Bearing.

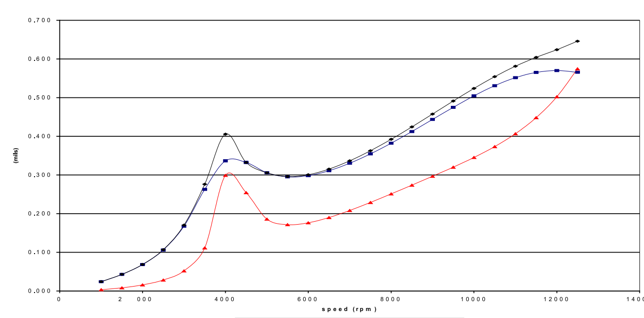

Figure 26. Response at Coupling End Bearing with Proposed LOP Bearing.

it is well damped, which results in an amplification factor less than 2.5. To compensate for this, tight clearance bearings were specified

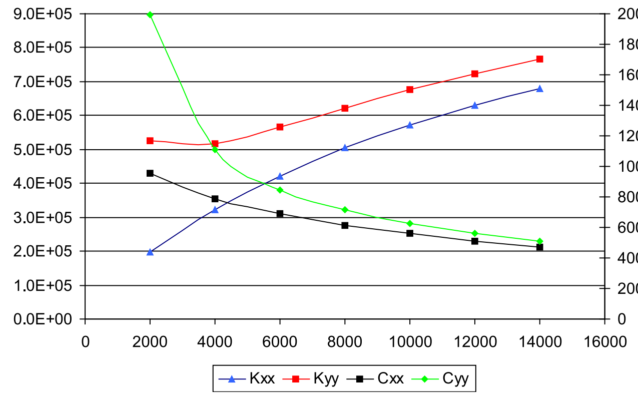

Figure 27. LOP Bearing Design with 58 Percent Offset Pivot.

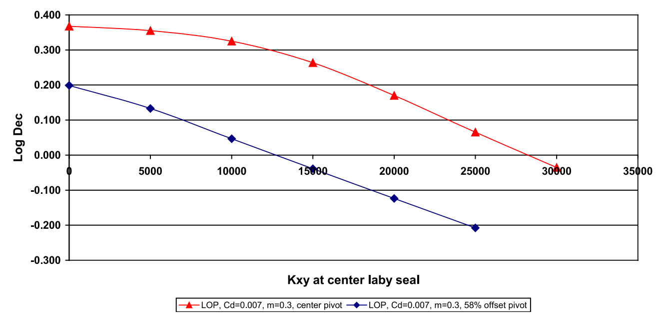

Figure 28. Stability Map for Proposed LOP with Center and 58 Percent Offset Pivots.

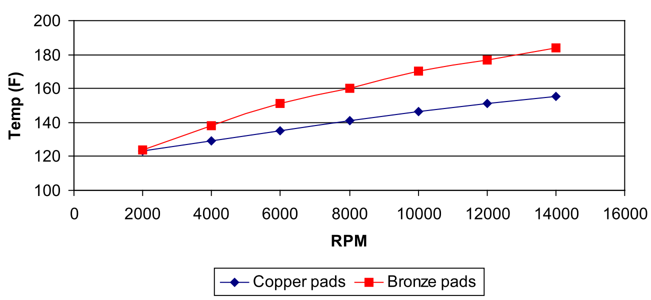

Figure 29. Predicted Maximum Pad Temperature for Copper and Bronze Pads.

to lower the synchronous response. Additionally, during the investigation into the root cause of the stability problem, it was discovered that one of the journals was actually 0.001 inch oversized. Since the bearing clearances were already tight, this certainly contributed to the original high bearing temperature problems. Furthermore, it seemed very odd for this compressor manufacturer to supply a thrust bearing of this type (i.e., journal bearing between two thrust collars). If a conventional journal/thrust arrangement (where there is only one thrust collar outside the journal) were used, the bearing span would be reduced by 1.73 inches. This would greatly improve the stability of the compressor (Figure 30). This is a significant contributor to the low margin of stability for this compressor.

Results

The new LOP bearings were installed in May of 2002. After the bearing change, the overall vibration levels were less than 1.5 mils at the maximum operating speed of 12,500 rpm (prior spectra were only at 11,000 rpm) and the subsynchronous component had vanished (Figure 31). Likewise, the bearing temperatures are all at acceptable levels (< 175°F). The compressor has continued to operate at this same vibration level for the past year and a half.

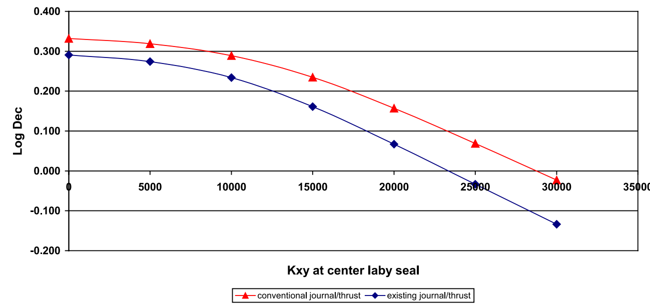

Figure 30. Stability Map Comparing Existing and Conventional Journal Thrust Arrangements.

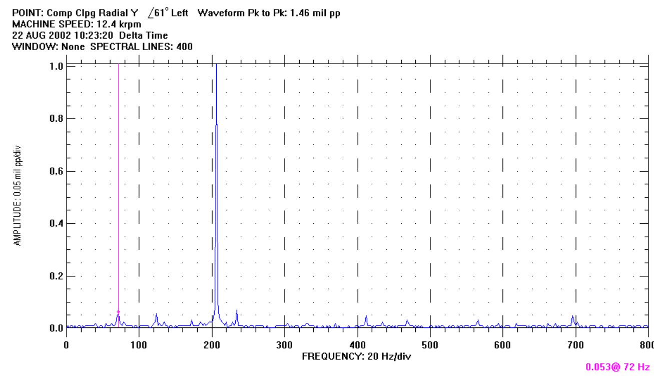

Figure 31. Compressor Drive End Radial Vibration after Bearing Modification.

CASE STUDY 2—MIDWEST ETHYLENE COMPRESSOR

Background

The second compressor is a horizontally split, multistage, side-load ethylene refrigeration/export compressor with dry gas seals in a Midwest olefins plant (Figure 32). The compressor typically operates between 10,500 and 11,000 rpm and is driven by a 6500 hp steam turbine. The original bearing design was a five-pad LOP bearing with center pivots. In 1997, the compressor was overhauled as part of a normal unit turnaround. A few years later, in 1999, a radial subsynchronous vibration that coincided with the first natural frequency of the compressor began to appear (Figure 33). At first the frequency was small in magnitude, but it continued to grow with time. The subsynchronous component was worse in the summer months and could only be kept at low levels by maintaining the oil temperature in a very tight band of a few degrees, toward the cool end of the normal range.

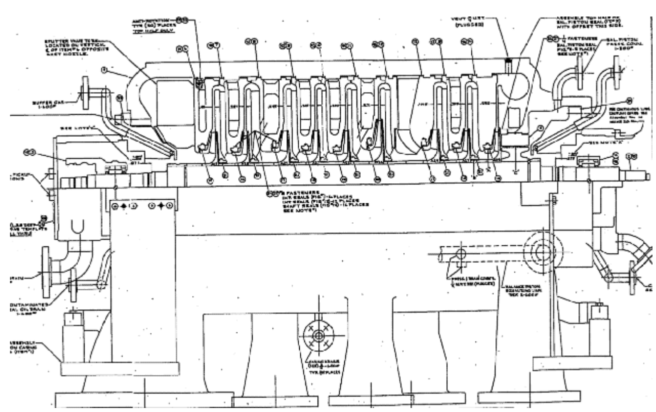

Figure 32. Cross-Section of Ethylene Compressor.

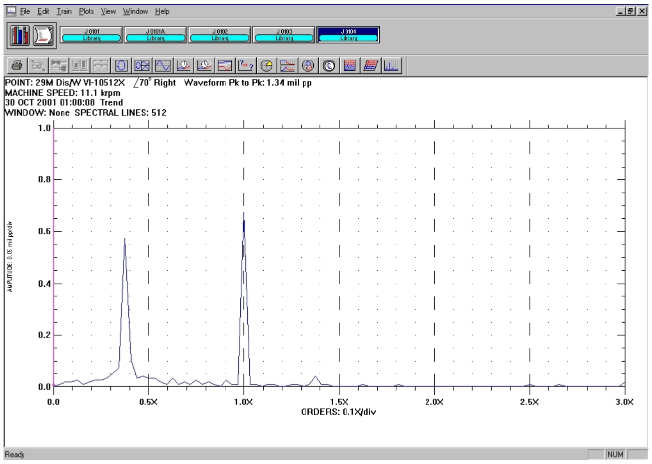

Figure 33. Spectrum from Compressor Drive End Radial Probe Showing Subsynchronous Peak at 4000 CPM.

A stability analysis was performed by an outside contractor that suggested that the stability problem could be eliminated by modifying the bearings. The proposed increase in stability was accomplished by increasing the width of the bearings from 1.875 to 2.125 and changing from a five-pad LOP to a four-pad LBP configuration. This causes the principal stiffnesses to be symmetric and increases the direct damping provided by the bearing (Figures 34, 35, and 36). The proposed stability of the first mode versus speed with the existing and the new improved bearings is shown in Figure 37. However, this stability map did not reflect the stability of the compressor in the field, because it shows that the compressor was unstable at all speeds above 8000 rpm (with the original LOP style bearings), but the compressor was only unstable intermittently and only after several years of operation.

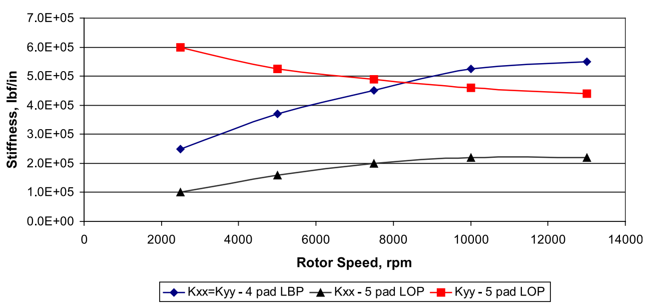

Figure 34. Comparison of Principal Stiffness Values for Four-Pad LBP and Five-Pad LOP Bearings.

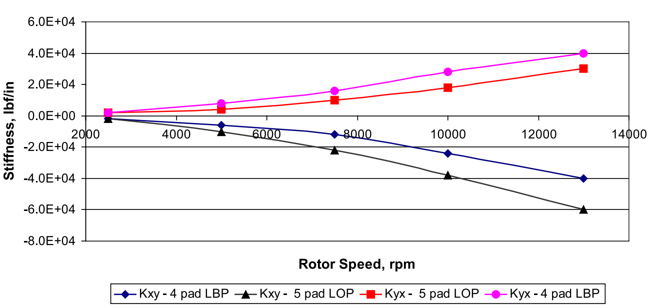

Figure 35. Comparison of Cross-Coupled Stiffness Values for Four Pad LBP and Five-Pad LOP Bearings.

At this point, a more indepth investigation into the root cause of the subsynchronous vibration was performed by plant personnel because a short outage opportunity was in the near future. The original plan was to replace only the bearings with the new modified

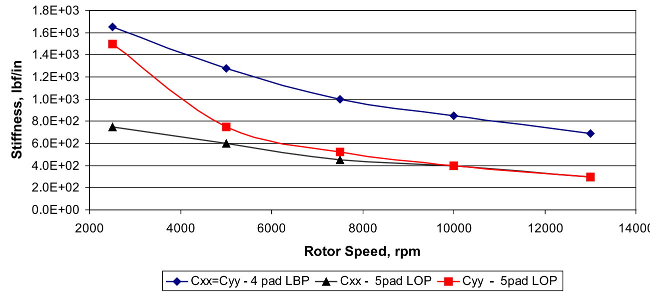

Figure 36. Comparison of Direct Damping Values for Four-Pad LBP and Five-Pad LOP Bearings.

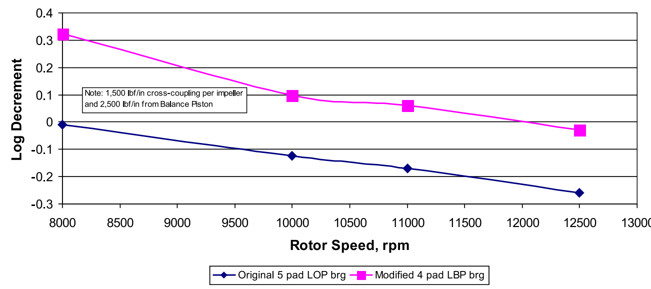

Figure 37. Stability Map from First Stability Analysis.

bearings; however, there was some concern that this change alone might not completely solve the problem. A review/recreation of the original stability analysis revealed the following:

- The LBP bearings did not greatly increase the stability of the compressor after all. A thorough review of the stability analysis was conducted with the same rotor dimensions, impeller, seal, and bearing coefficients. Even though the LBP bearings produced more direct damping than the original LOP design, the benefits of the nonsymmetric stiffness produced by the LOP design caused the change in stability to be negligible. As can be seen in Figures 38 and 39, the calculated log decrement with both designs is almost identical. Most important, the log decrement is very positive (i.e., stable) with the existing bearing design.
- The original stability analysis used 1500 and 2500 lbf/in as the cross-coupled stiffness produced by the impellers and balance piston seal, respectively. While the impeller coefficients were inline with industry practice (i.e., the Wachel equation), the values used for the balance piston seal appeared unusually low, considering the design of the balance piston seal. The seal is a teeth-on-rotor (TOR) design, with relatively high differential pressure and surface velocity, and moderate molecular weight (Table 1 and Figure 40). All these factors tend to indicate a labyrinth seal that can be very destabilizing.

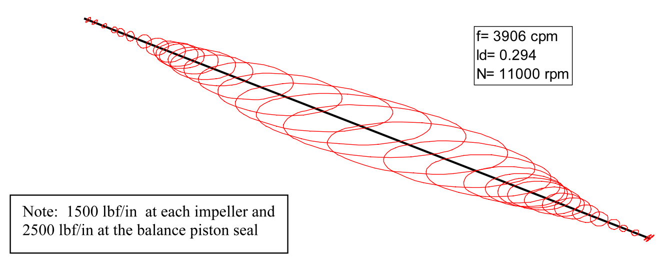

Figure 38. Calculated First Mode with Existing Bearings, Including Original Impeller and Seal Coefficients.

Because the balance piston seal had such a high potential to be destabilizing, a review of its performance was conducted. Due to problems with high leakage in the past, a differential pressure transmitter had been installed on the balance piston line prior to

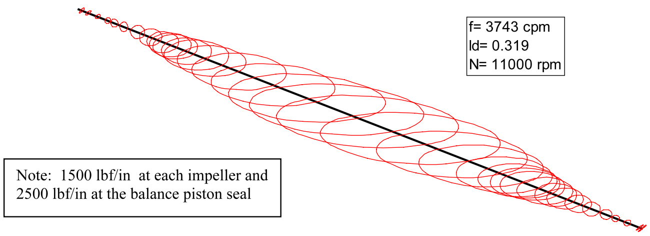

Figure 39. Calculated First Mode with Proposed Bearings, Including Original Impeller and Seal Coefficients.

Table 1. Balance Piston Seal Design Information.

DP	415 psid
Number of teeth	24
Length	4 in
Diameter	10.5 in nominal
Rotational speed (design)	11,000 rpm
Molecular weight of gas	28

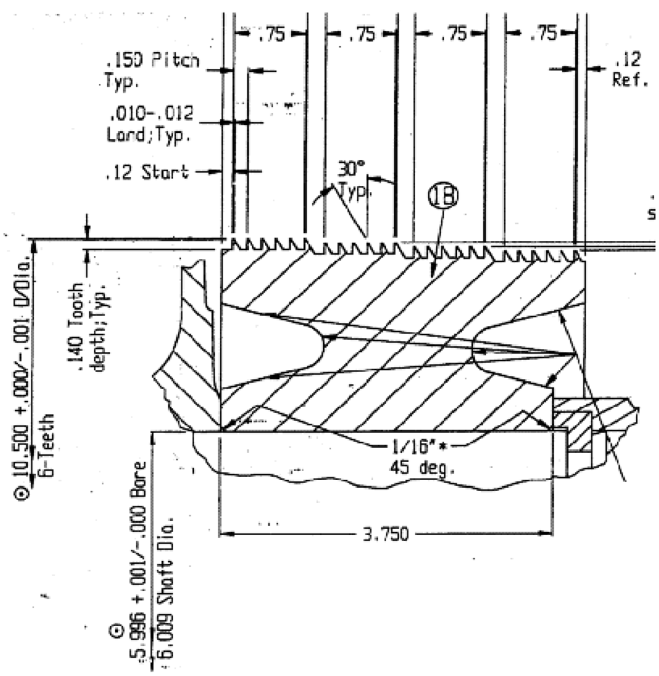

Figure 40. Cross-Section of Balance Piston Seal.

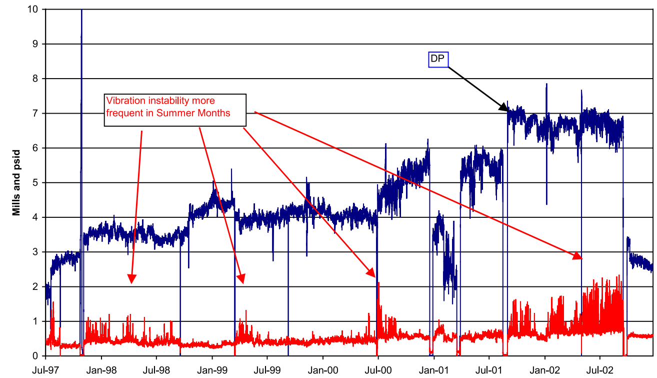

Figure 41. Balance Line Differential Pressure and Radial Vibration since 1997.

Table 2. Calculated Rotordynamic Coefficients for Balance Piston Seal for Different Clearances and Inlet Velocity Ratios.

Cr	Vel ratio	K_{xx}	K_{xy}	C_{xx}	C_{xy}
mils	non-dim	lbf/in	lbf/in	lbf-s/in	lbf-s/in
5	0.6	-49633	16380	23.7	39.2
5	0.7	-49205	19350	24	38.8
5	0.8	-49844	23380	25.4	39.4
10	0.6	-45610	19144	27.7	36.6
10	0.7	-46580	24029	29.8	37.7
10	0.8	-47502	29673	32.18	38.64
20	0.6	-30240	23580	34.11	24.18
20	0.7	-30645	30930	38.36	24.92
20	0.8	-30648	40000	43.426	25.34

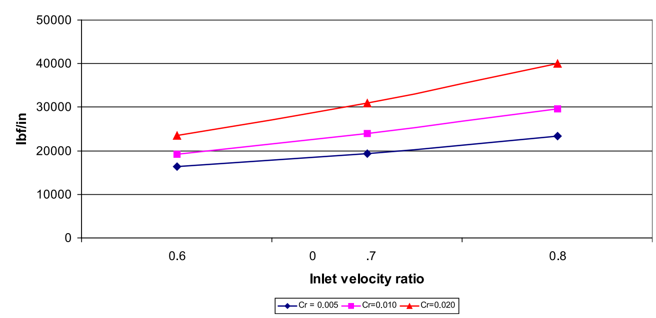

Figure 42. Cross-Coupled Stiffness Produced by Balance Piston Seal for Different Clearances and Inlet Velocity Ratio.

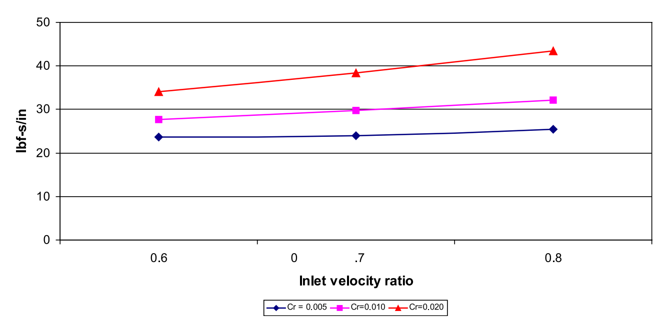

Figure 43. Direct Damping Produced by Balance Piston Seal for Different Clearances and Inlet Velocity Ratio.

1997. A correlation between the balance line differential pressure and the radial vibration excursions on the compressor could be seen when plotted back to 1997 (Figure 41). Since the subsynchronous vibration seemed to be tied to increased flow through the balance line, it was suspected that the seal had suffered some type of failure that would allow increased flow. The director of a major university's research laboratory was contracted to provide the seal coefficients shown in Table 2, as well as Figures 42 and 43 for different clearance and inlet velocity ratios. As mentioned in the prior case study, the K_{xy} values were reduced so that the whirl frequency coefficients were equal to the inlet velocity ratios. The estimated leakage based on balance line differential pressure was close to the calculated leakage for the seal with 0.020 inch clearance.

A stability map for different values of cross-coupling at the balance piston (with the original five-pad LOP bearings) is shown in Figure 44. Since the calculated value of the cross-coupled stiffness produced by the balance piston seal was only 30,930, this indicates that the rotor is quite stable with the original LOP bearings. According to the calculations, the balance piston seal does not produce enough cross-coupled stiffness to drive the compressor unstable. It was deduced that the balance piston seal had suffered a severe failure that was causing it to produce more cross-coupled stiffness than the code predicted. The decision was made to split the case during the upcoming outage and inspect the balance piston seal, instead of just changing the bearings, because the analysis indicated a significant failure of an internal component to drive the machine unstable. Additionally, this plant had experience with a previous design stationary abradable seal failing catastrophically when process pressure closed the seal onto the shaft. To address the instability, an alternate stationary balance piston seal was designed using an aluminum honeycomb to run against the rotating teeth to provide a more robust seal and additional resistance to swirl.

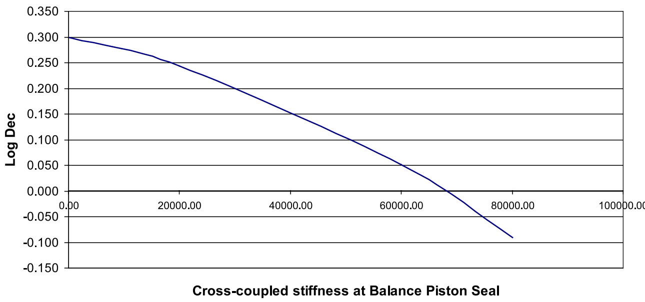

Figure 44. Effect of Cross-Coupled Stiffness on Compressor Stability with Original LOP Bearings.

Results

The compressor was shutdown in September of 2002 to inspect the balance piston seal and determine the source of the subsynchronous vibration. The balance piston seal and rotating teeth were found to be in excellent condition. However, the buffer gas labyrinth seal in front of the dry gas seals (Figure 45) had rubbed very hard (Figures 46 and 47). The rotating teeth ran against a stationary aluminum seal. The seal had rotated in the section (inboard side closing into the shaft, outboard side opening away from the shaft), apparently due to thermal distortion initiated by a rub. The distortion exacerbated the rub, causing further damage. The clearance on these seals had been quite tight by design (0.004 to 0.005 total) to reduce the amount of buffer gas consumption. The failure of these seals caused the high differential pressure in the balance line due to excessive buffer gas flow from the discharge end seal. Likewise, the rubbing undoubtedly excited the first natural frequency of the rotor. The compressor was reassembled with the new Fluorosint® stationary buffer gas seals with a larger clearance and the new LBP bearings. There was no subsynchronous vibration present after startup (Figure 48) or in the past year.

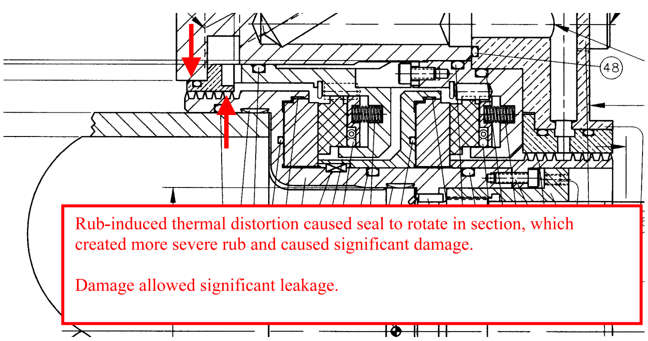

Figure 45. Dry Gas Seal with Buffer Gas Labyrinth.

CONCLUSIONS

Both these case studies are examples where a great deal of effort was expended to look into every possible detail of a stability problem. Without this additional effort, it is very likely that the stability problem(s) would have persisted after the first repair attempts. Especially in the second study, while the rotordynamic analysis did not exactly pinpoint the problem area, it did trigger the need for further inspection within the compressor, which led to the solution of the problem. Likewise, these two cases point out the potential inaccuracies in existing bulk-flow labyrinth seal codes and the need to apply experimental results to their output.

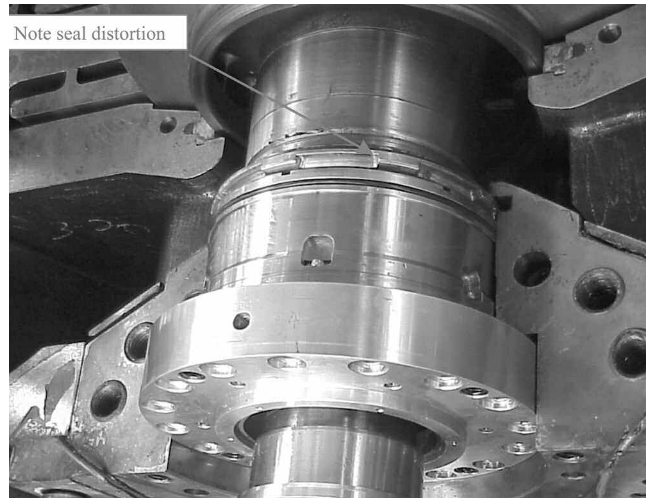

Figure 46. Damaged Buffer Gas Labyrinth.

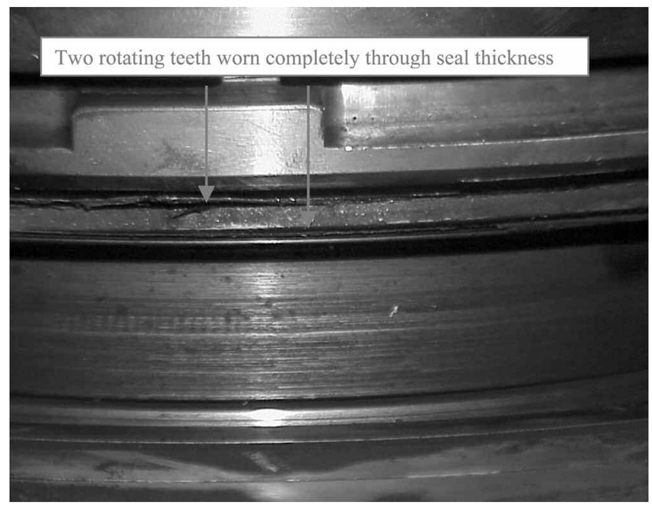

Figure 47. Damaged Buffer Gas Labyrinth Seal.

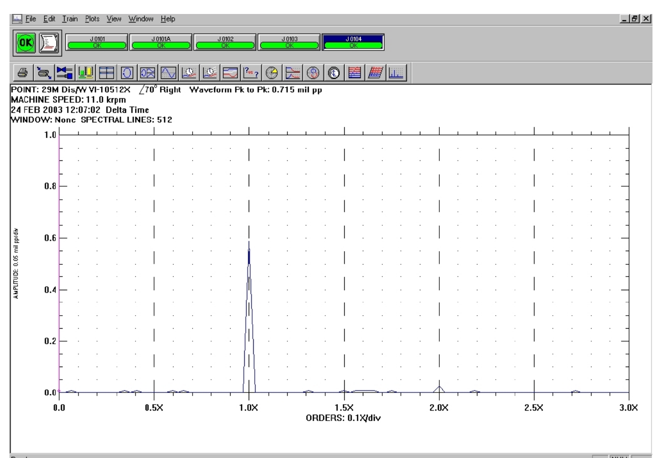

Figure 48. Radial Vibration Spectrum after Overhaul.

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