UNEXPECTED ROTORDYNAMIC INSTABILITY IN A "PROVEN" FCC WET GAS COMPRESSOR

by
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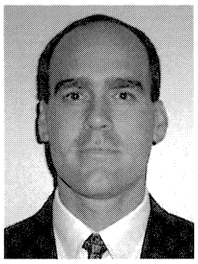

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ABSTRACT

The cause and effects of a large subsynchronous vibration (3.5 mils at 3490 cpm) in an FCC wet gas compressor are examined. The compressor is driven at 7850 rpm by a 6000 hp electric motor, through a speed-increasing gearbox, and compresses hydrocarbon gases from 20 psig to 235 psig. A rotordynamic analysis of the compressor determined that even though the compressor had a long history (approximately 25 years) of smooth operation (overall amplitudes less than 0.5 mils), it had a very low margin of stability. This lack of stability was caused by a high bearing to shaft stiffness ratio as well as destabilizing forces in the balance piston and shaft seals. The pressure dam bearing design is not destabilizing to the rotor system because the journal is very heavily loaded; however, the high direct stiffness that is produced reduces the effective damping of the system. Labyrinth and oil seals are usually not considered to have large effects on a compressor with such low pressures. However, the atmospheric side of the oil seal was determined to contribute the largest destabilizing effect. The effective length to diameter (L/D) ratio of the oil seal was reduced by 50 percent to lower the amount of cross-coupled stiffness the seal produces. This modification reduced the subsynchronous vibration from 3.5 mils to 0.1 mils. The rotordynamic analysis also identified other modifications to the bearing and balance piston design that should be made to increase the rotor's margin of stability. Both steady-state and transient spectral vibration data are provided before and after the oil seal modification to validate the rotordynamic model of the compressor.

INTRODUCTION

The compressor was first commissioned in 1971 and had a history of very low vibration levels (< 1.0 mil). It is a straight through, horizontal split machine that operates at 7850 rpm. It has two sections, with three impellers in the first section and four in the second (Figure 1). It was rerated three times between 1973 and 1992, but none of the rerates involved drastic design changes (i.e., smaller impellers were installed, speed decreased 200 rpm, three impellers were redesigned). The compressor operates at relatively low pressures with suction and discharge equal to 20 psia and 235 psia, respectively. The molecular weight of the gas varies from approximately 35 to 45.

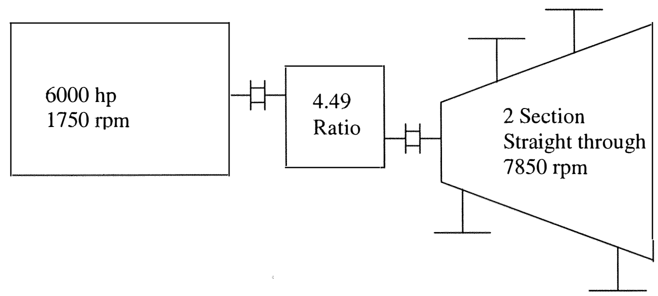


Figure 1. FCC Wet Gas Compressor.

The compressor was originally installed with pressure dam bearings. The bearing effective L/D was decreased during the second rerate because the rotor weight decreased slightly and it was believed that the OEM wanted to maintain the same bearing loading/eccentricity to prevent instability. The OEM recommended changing to spherically seated tilt-pad bearings during the last rerate; however, the user decided not to make the change due to historically low vibration levels. The unbalance response analysis conducted before the last rerate did not include any oil seal effects (only aerodynamic). It did show a very high amplification factor, approximately 18.

On September 26, 1997, over five years after the last rerate, the compressor shut down due to an electrical power outage. A few days after the compressor was restarted, the compressor outboard bearing vibration increased to 2.5 mils. The largest amplitude was at approximately 0.4× to 0.45×. The vibration levels were very erratic, but seemed to be affected by horsepower, molecular weight, oil pressure, and oil temperature. Several attempts were made to reduce the vibration online by varying the oil temperature and pressure, but they only achieved limited success. After each change, the overall magnitude of the subsynchronous vibration would increase. It was decided that a detailed rotordynamic analysis of the compressor needed to be performed to determine the cause of the subsynchronous instability.

High Subsynchronous Vibration

The compressor is equipped with two radial proximity probes mounted at 45 degrees from the vertical at each bearing. The vibration monitoring system was upgraded in 1995 to include an online real time/spectrum analyzer that allowed monitoring of all the spectra online. The high overall vibration levels were caused by a subsynchronous component at approximately 0.45× (3400 cpm), with the highest levels on the compressor outboard bearing (Figure 2). As can be seen, the subsynchronous component was approximately six times the synchronous vibration levels. In the cascade plot (Figure 3), the subsynchronous component appears at 6850 rpm, then disappears. Also, the vibration levels were very erratic, but continued to grow with time after the September 26, 1998 shutdown (Figure 4). Because the vibration represented a significant increase in past vibration levels, it was considered to be a serious problem. The vibration levels were affected by the following:

- Horsepower
- Process gas molecular weight
- Oil pressure
- Oil temperature

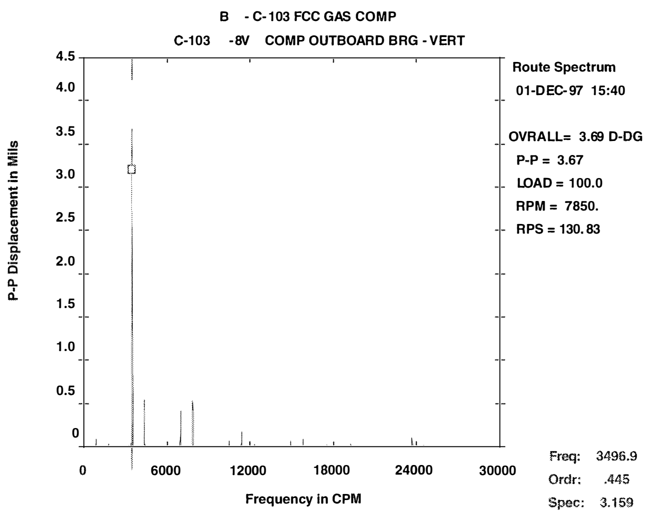


Figure 2. Compressor Outboard Vibration.

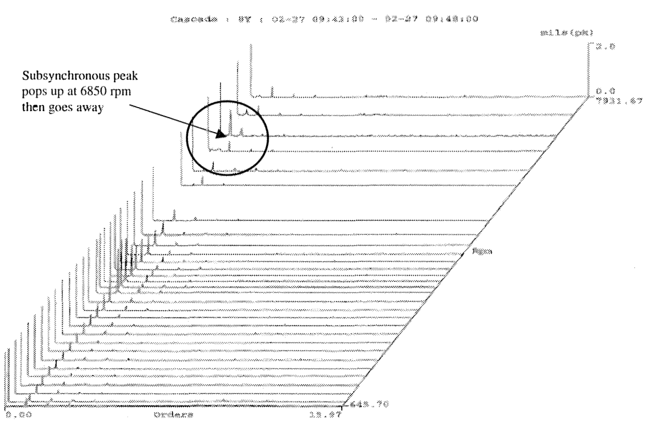


Figure 3. Compressor Outboard Radial Vibration on Startup.

Figure 5 shows the overall levels of vibration on the compressor outboard vertical probe immediately before the compressor was shutdown on January 6, 1998. As can be seen in the plot, the levels dropped from approximately 3.0 mils to 0.5 mils when the power was reduced from 4500 hp to 2500 hp. Likewise, the vibration dropped when the oil temperature was *lowered*, but would not stay low for an extended period of time (Figure 6). Raising the oil pressure also lowered the vibration (Figure 7), but the vibration levels increased later even though the oil pressure was not changed.

The vibration appeared to be very consistent with oil whirl in the journal bearings because:

- The largest component was at 0.45×.
- The vibration was affected by oil temperature and pressure.
- The problem started after an electrical power outage that might have caused the bearings to wipe.

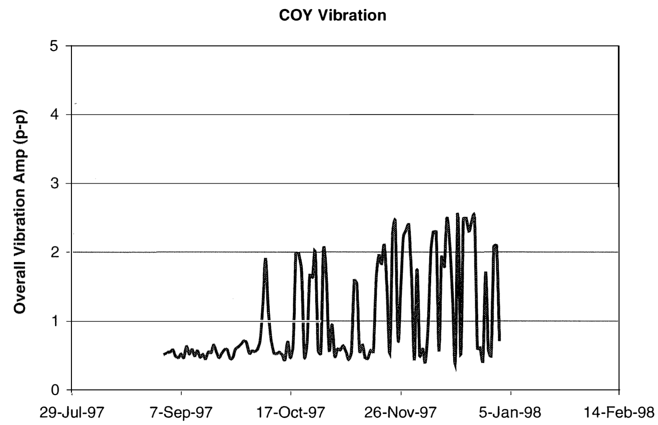


Figure 4. Increase of Vibration Amplitude Over Time.

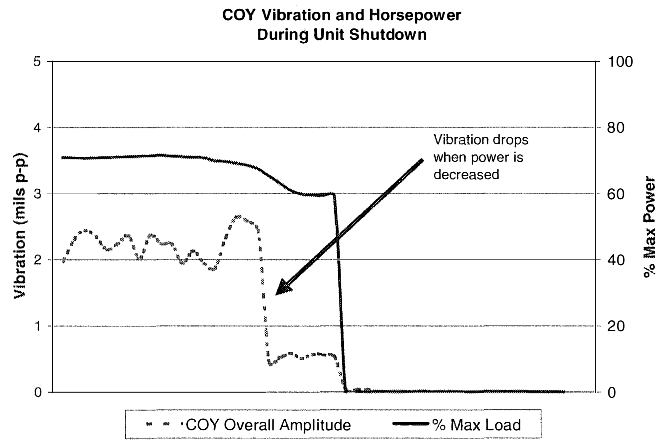


Figure 5. Effects of Load on Vibration.

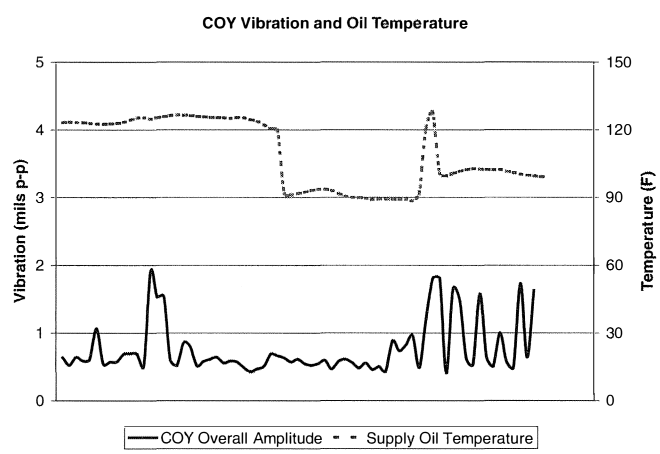


Figure 6. Effects of Lube Oil Temperature.

- The bearings were fixed geometry pressure dam instead of tilting-pad.

However, several facts were not consistent with oil whirl:

- The high vibration did not start until a few days after the compressor was restarted. A wiped bearing would have normally caused a problem immediately.
- The main lube oil pump is turbine driven, so lube oil pressure was available during the shutdown.
- No high bearing temperatures were noted during the shutdown or subsequent startup.

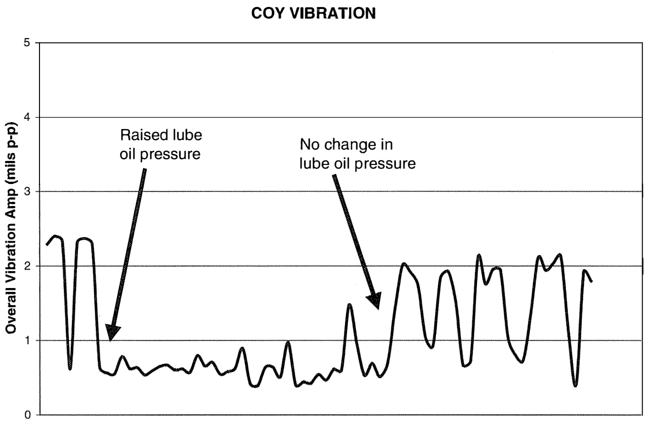


Figure 7. Effect of Lube Oil Pressure.

ROTORDYNAMIC STABILITY ANALYSIS

A rotordynamic stability analysis was conducted to determine the source of the subsynchronous vibration and recommend a solution to eliminate or reduce the vibration to an acceptable amplitude. The rotor model developed for this machine included the shaft model (Figure 8) as well as the following shaft support component models:

- Bearings
- Oil seals
- Balance piston seals
- Impeller eye seals
- Interstage shaft seals
- Aerodynamic effects on impellers

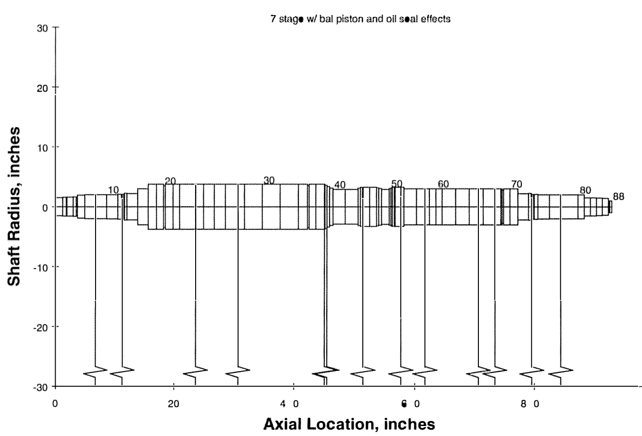


Figure 8. Shaft Model.

Bearings

The bearings are 120-degree arc pressure dam journal bearings. As can be seen in Figure 9, there are two relief tracks in the bottom half of the bearing, one on each side. These relief tracks reduce the effective L/D of this bearing to be 0.25! The diametrical clearance is 0.006 inch to 0.008 inch. This design results in a *very* heavily loaded (180 psi) bearing with a high eccentricity ($\in = 0.85$ to 0.9), confirmed by shaft centerline plot, in Figure 10, and very axisymmetric principal stiffness ($K_{yy} = 5 \times K_{xx}$, Figure 11). Of course this also makes the bearing have a stabilizing effect on the rotor as well, since $K_{xy} < 0$. A simplified stability map for this bearing with a point mass rotor is shown in Figure 12. As can be seen, the bearing operating point is in the stable region, but it does not have an infinite onset speed of instability (i.e., it can be driven unstable). This stability map does not indicate the stability of the compressor, but does show that the bearing has no destabilizing effects. The only possible way for this bearing to be destabilizing at its original design point was for it to be mechanically damaged or have the radial load on the journal decreased.

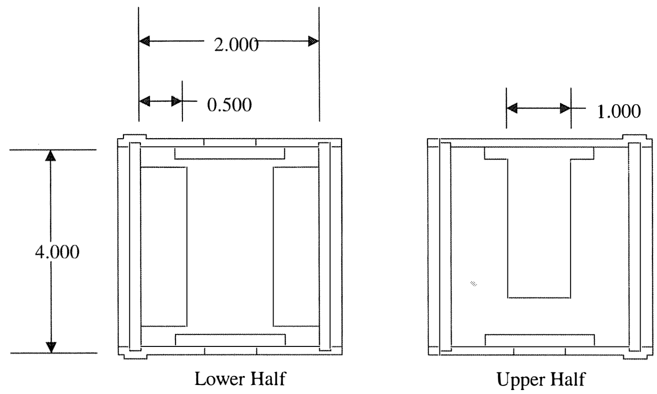


Figure 9. Pressure Dam Bearing.

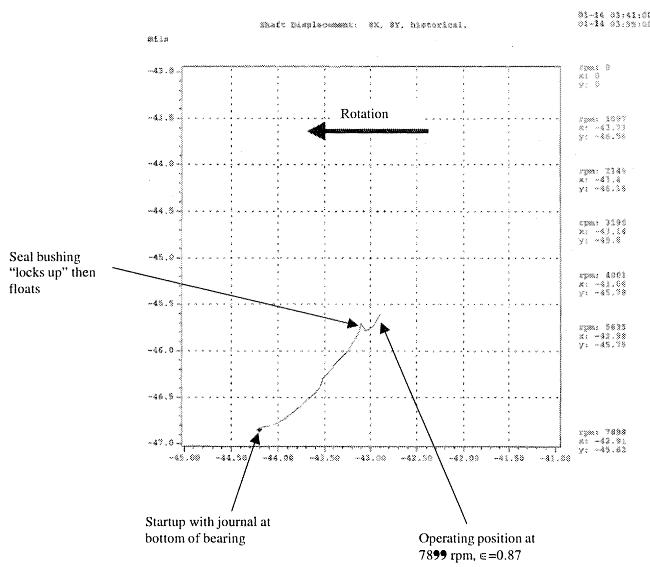


Figure 10. Compressor Nondrive End Shaft Centerline Plot.

However, this bearing design does not come without a cost to the total system stability. The high bearing to shaft stiffness ratio (K):

$$K = \frac{2K_B}{K_S} = 24 \tag{1}$$

reduces the effective damping in the system, because the rotor is like a pined connection at each end. A simple method to predict stability based on K was developed by Barrett, et al. (1978). Based on this method, the amplification factor (Q) is approximated by:

$$Q = 2(1+K) \cong 50! \tag{2}$$

Likewise, the cross-coupled stiffness required in the middle of the rotor to drive the system unstable is:

$$K_{XY} = \frac{M\omega_{cr}^2}{2(1+K)}\left[\frac{1+2K}{2(1+K)}\right]^{\frac{1}{2}} \cong 3725 \text{ lbf / in} \tag{3}$$

and the optimum direct damping in the bearing is:

$$C_{OPT} = \left(\frac{K_B c}{W} + \left(\frac{c\omega_{cr}^2}{2g}\right)\right)\frac{W}{\omega c} \cong 7800 \text{ lbf} - \text{s / in} \tag{4}$$

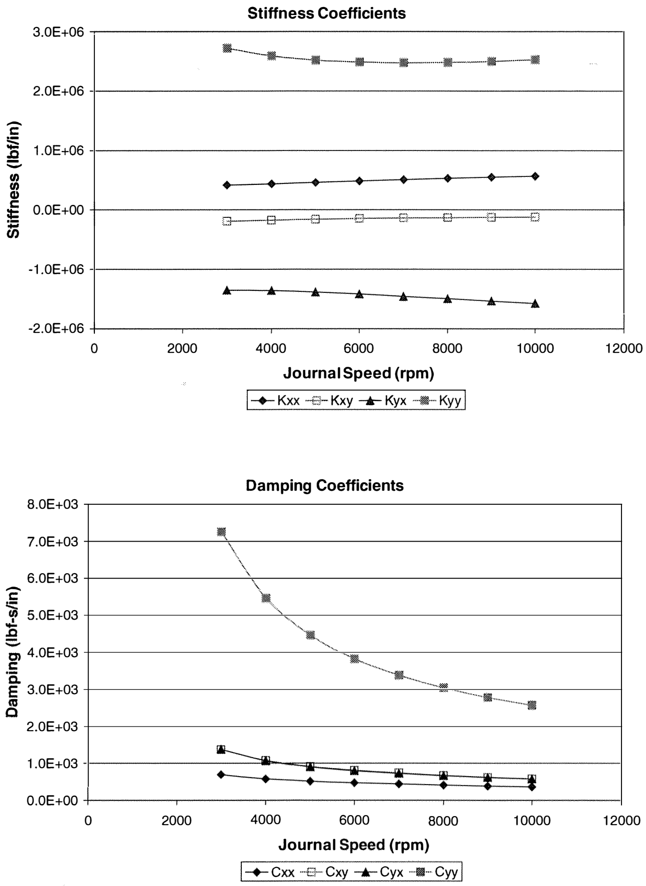


Figure 11. Stiffness and Damping Coefficients for Pressure Dam Bearing.

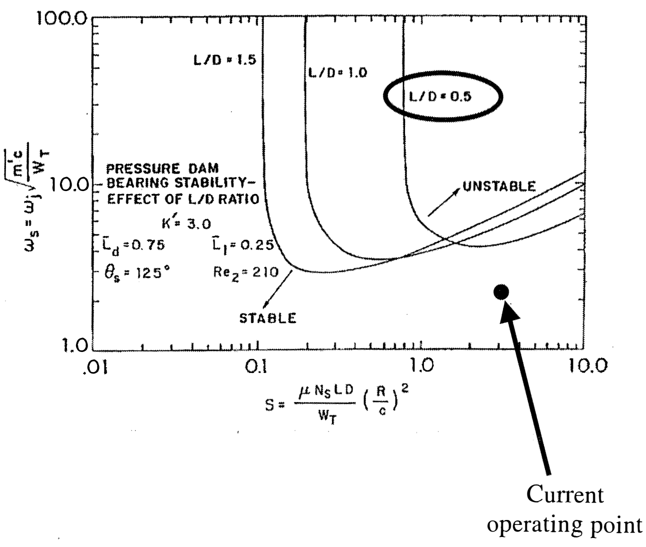


Figure 12. Stability Map for Pressure Dam Bearing.

which is twice the actual value. This is only an approximate method for a simple model (i.e., the rotor's mass has been lumped together into one modal mass, gyroscopics are not included, and the bearings are considered to be identical). Obviously, this method over-predicts the synchronous amplification factor because it does not include the direct damping provided by the oil seals. However, it provides an easy way to estimate the sensitivity of the rotor system and the optimum bearing damping required. This will be confirmed later by the system stability analysis using the transfer matrix method.

Oil Seals

The shaft end oil seals are the mechanical face type, with rotating and stationary faces, as well as a carbon face in between the two. The process seal itself has very little effect on the rotor, other than to add a little damping. It is the floating bushing seal that is the big concern (Figure 13). The bushing seal floats in the seal housing and allows the seal oil that does not leak past the process seal to return to the lube oil reservoir, through the lube oil return line. If the friction force generated by the differential pressure across the seal is greater than the hydrodynamic forces generated by the oil between the seal and the shaft, the seal ring will no longer float in the radial direction. Floating seals at these low pressures are usually not considered to be a problem, since the low differential pressures do not allow for a potential "lock up." The bushing seal can float 0.030 inch diametrically in the housing, but only has 0.003 inch axial clearance. It was originally designed with a relatively low diametrical clearance of 0.005 inch to 0.007 inch between it and the shaft. Note: this clearance is less than the bearing's, which is 0.006 inch to 0.008 inch.

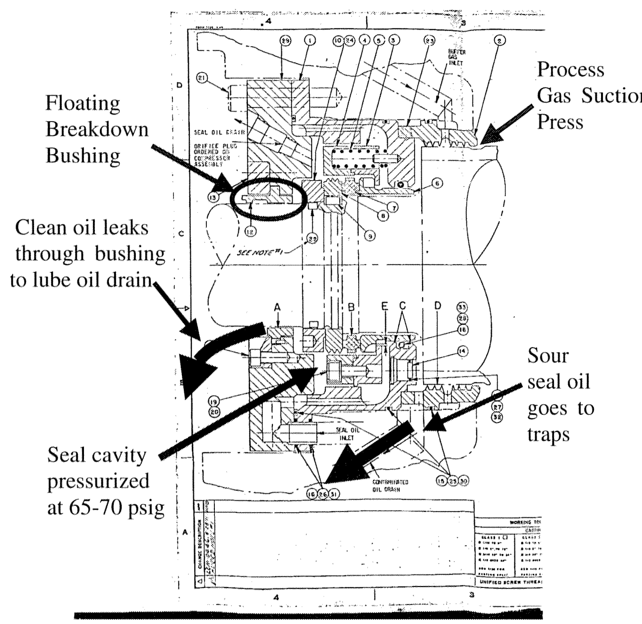
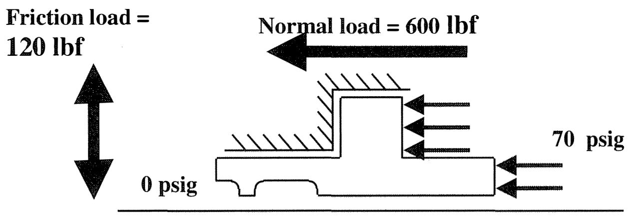


Figure 13. Shaft End Seal.

The 70 psid oil pressure exerts approximately 600 lbf of axial force, which presses the bushing into the outer ring of the seal housing. High pressure seal faces are normally lapped together to produce a very smooth finish and/or spray coated with a hard material such as tungsten carbide to reduce the amount of friction between the two. However, because it is such a low pressure application, the bushing seal in this seal is just machined. This increases the coefficient of friction between the floating bushing and seal housing, resulting in a radial force of approximately 120 to 150 lbf that can "lock up" the bushing seal in an eccentric position (Figure 14). The emergence of the subsynchronous vibration at 6850 rpm, in Figure 3, is evidence that the floating bushing "locked up" momentarily on startup, but then started floating again by the time the compressor reached full speed. The radial load carrying capability of these seals is not enough to lift the rotor off the bearings (the rotor weighs 1450 lbf). However, these bushings can affect the rotor stability in two ways:

- *Decrease the radial load on the bearings*—Decreasing the radial load on the bearings would reduce the eccentricity of the journal in the bearings and potentially cause the journal to "whirl" in the bearing. However, the downward force of the pressure dam is approximately 400 lbf. This force along with the rotor half weight of 725 lbf is more than enough to keep the bearing in a stabilizing mode at this speed.



Cd = 0.005 inch
Axial float = 0.003 inch
Radial float = 0.030 inch

Figure 14. Floating Seal Oil Breakdown Bushing.

• *Act as an additional support with a high cross-coupled stiffness*—In addition to affecting the radial load on the bearing, the bushing seal acts as a bearing as well. The bushing seal is similar to a straight journal bearing with no pressure dam, an L/D = 0.1875, and an axial pressure drop of 70 psid. The floating bushing develops stiffness and damping in a similar manner to a journal bearing, except that the flow is obviously more turbulent and the oil does not normally cavitate until it reaches the low pressure side of the seal.

Because of the balance line, the floating bushings on each end of the compressor see the same seal oil differential pressure, but not the same oil temperature. Because of limited buffer gas on the seals, the discharge temperature of the compressor causes the seal oil temperature on the drive end to be approximately 175°F (52 SSU); whereas, on the nondrive end, the oil enters the sleeve at approximately 110°F (100 SSU). Because the flow through the seal is nearly laminar, the viscous effects are much more important than inertial forces. This lower viscosity on the drive end makes a large difference in the stiffness coefficients between the two seals of the compressor. As can be seen in Figures 15 and 16, the nondrive end seal has a much higher cross-coupled stiffness value.

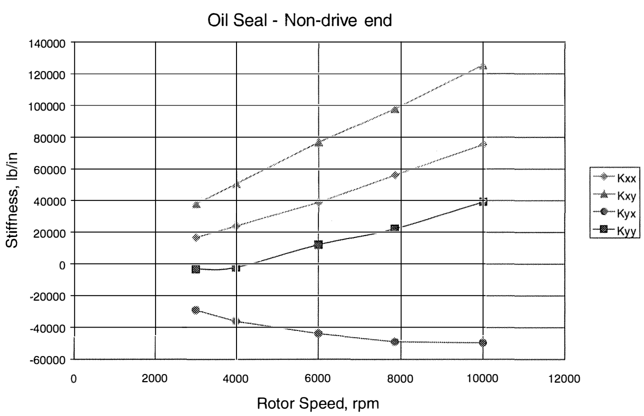


Figure 15. Stiffness Coefficients for Nondrive End Oil Seal.

Note: the computer model for the bushing seal requires the seal eccentricity as input. Since the eccentricity is unknown, it is entered iteratively until the calculated radial load capacity matches the frictional load in Figure 14.

Balance Piston Seal

The balance piston seal has a teeth-on-rotor (TOR) labyrinth (Figure 17). This type of labyrinth is very common on wet gas services because of its excellent resistance to fouling (i.e., it centrifuges out any buildup in the labyrinth). However, TOR labyrinths have been shown to be more unstable than teeth-on-stator (TOS) seals (Childs, 1993). The location of the balance

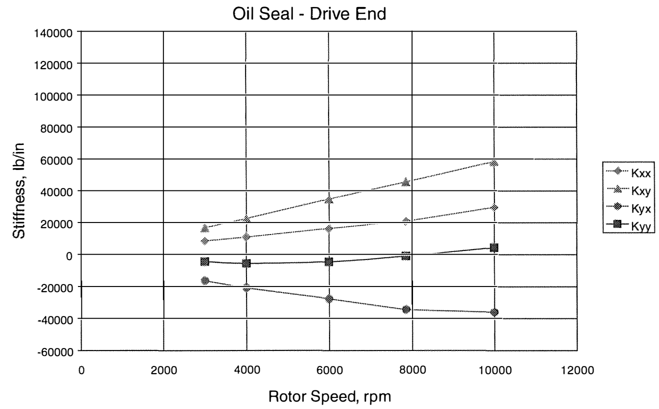
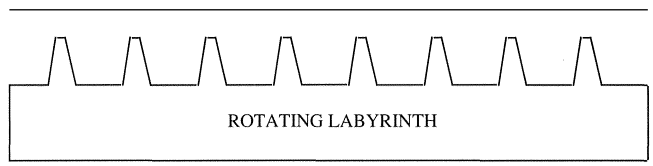


Figure 16. Stiffness Coefficients for Drive End Oil Seal.

piston seal relative to the antinode of the first bending mode (i.e., larger displacement means larger force produced) is the reason this balance piston has a large affect on the stability of the compressor, not the magnitude of the stiffness coefficients (Figure 18). Obviously, the balance piston seal is much closer to the center of the rotor than the bearings or oil seals. Because of the shape of the first bending mode, this causes the effect of the balance piston to be magnified.



Rotating labyrinth
21 teeth, 0.19 in equal spacing, 0.18 in tall
Diametrical clearance - 0.010 - 0.020 inch
External diameter - 12.993
P_1=235 psia P_2= 35 psia
Mw=35-45

Figure 17. Balance Piston Seal.

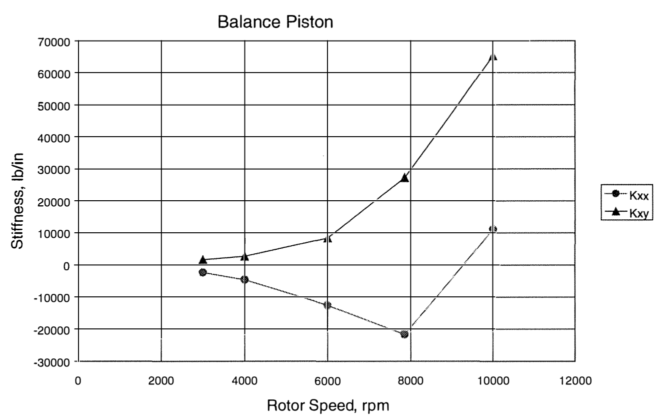


Figure 18. Balance Piston Seal Stiffness Coefficients.

The cross-coupled stiffness of a labyrinth seal is a function of many parameters including surface speed, differential pressure, and gas molecular weight. Neither the speed nor differential pressure changed in Figure 5 (i.e., low load case); however, the molecular weight of the gas drops as the FCC unit is being shut down. This indicates that the balance piston seal (likewise the other internal labyrinth seals) may be causing the sensitivity to load. The stiffness coefficients for the balance piston seal with half the normal molecular weight is shown in Figure 19. They are approximately 75 percent lower than those in Figure 18. The effect of the balance piston seal, with the two different molecular weights, on the stability of the compressor will be discussed below.

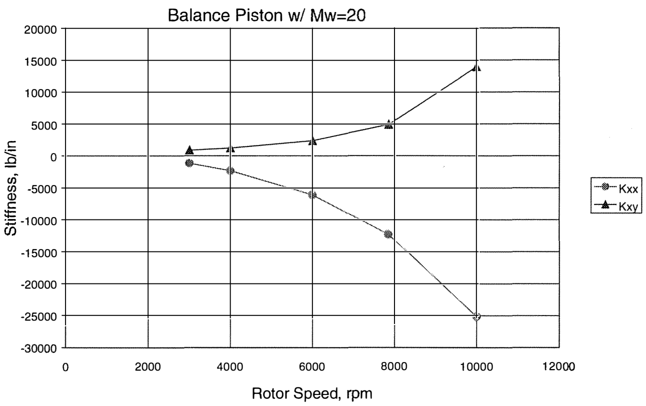


Figure 19. Balance Piston Seal Stiffness Coefficients with Mole Weight = 20.

Impeller Eye Seals/Interstage Shaft Seals

Obviously, the stiffness and damping coefficients for the small impeller eye and interstage shaft seals are quite low in comparison with the balance piston seal listed above. This is due to:

- Much lower differential pressure.
- TOS construction.
- Reduced L/D.
- Reduced number of teeth.

In fact, the impeller eye seals have such a small effect for a compressor of this type, with such low pressures, that their effect in the rotordynamic model is negligible.

Aerodynamic Effects on Impellers

Because of the relatively low pressures and horsepower, the cross-coupled stiffness values obtained from the Wachel equation were all less than 500 lbf/in per impeller, at full load with the high molecular weight. These values certainly add to the instability of the system just because they increase the amount of cross-coupled stiffness at the center of the rotor (where it has the most effect on the first bending mode), but overall they have a very minor effect on the stability of the compressor.

SUMMARY OF STABILITY ANALYSIS

The first step in any stability analysis is to verify the rotordynamic model of the machine. The calculated unbalanced response at the outboard bearing, Figure 20, shows the first damped mode to be in close agreement with the one shown in the measured Bodé plot (Figure 21).

Table 1 shows the calculated logarithmic decrement (log dec) and frequency of the first forward mode of the compressor for various models. The rotor with the bearings alone is not very stable, with a log dec of only 0.035 (model 1). Normally, a minimum log dec of 0.1 to 0.2 is required to ensure stability of the system. The stability improves slightly with the addition of the centered oil seals to the model because of the damping they add (model 2). Adding the balance piston seal lowers the calculated log dec to 0.022 (model 3), though it is still considered marginally stable. If the seals are assumed to be locked with the low molecular weight gas, the calculated log dec increases slightly to 0.035 (model 4), which explains the low vibration when the compressor was pumping lighter gas. As can be seen, it is only the combination of locked seals and high molecular weight that drive the compressor unstable (model 5). The frequency of the high subsynchronous vibration is within 200 cpm of the predicted first forward mode at 3240 cpm for this model (Figure 22). The calculated log dec of the first mode equals −0.055, which means that the vibration will not die out if it is perturbed (i.e., the system is unstable).

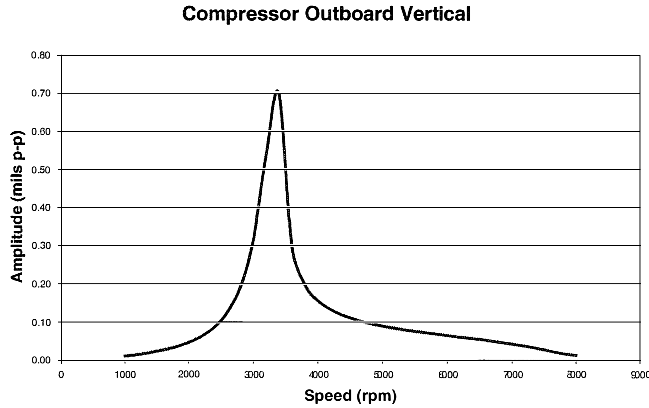


Figure 20. Predicted Unbalanced Response of Compressor Outboard Bearing.

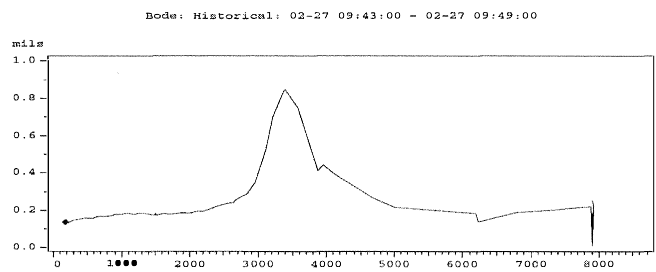


Figure 21. C-103 Measured Response, Compressor Outboard Bearing Vertical.

Table 1. Variations in Stability of Calculated First Forward Mode.

#	Model	Log Dec	Frequency
1	Rotor/bearings	0.035	3336
2	Rotor/bearings/floating oil seals	0.044	3305
3	Floating oil seals/balance piston/high MW-load	0.022	3247
4	Locked oil seals/balance piston/low MW-load	0.035	3280
5	Locked oil seals/balance piston/high MW-load	−0.055	3240

The following conclusions can be drawn from the stability analysis:

- The high bearing to shaft stiffness ratio causes the rotor to be very sensitive to even small amounts of cross-coupled stiffness.
- The instability of the compressor is caused mainly by the destabilizing forces in the shaft oil seals and balance piston labyrinth seals.
- The increase in vibration after the September 26, 1997 shutdown was probably caused by lockup of the oil seals, during the shutdown under load or subsequent startup.
- The subsynchronous vibration is not caused by destabilizing forces (oil whirl) in the journal bearings. The bearings are loaded heavily enough at this speed.

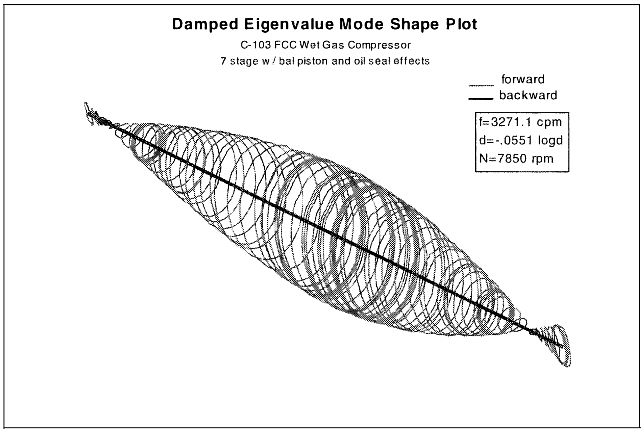


Figure 22. First Forward Mode Shape, Before Modifications.

- The oil seal floating bushing is very similar to a lightly loaded journal bearing, which has large amounts of cross-coupled stiffness.
- The lack of vibration during the low load condition probably comes from reduction of the cross-coupling in the balance piston labyrinth seal and the aerodynamic cross-coupling in the impellers, when the molecular weight of the gas and the horsepower of the compressor are both lower.
- Nondrive end seal effects are greatest because of higher seal oil viscosity.

PROPOSED MODIFICATIONS

The root cause of the subsynchronous vibration is the high bearing to shaft stiffness ratio, which decreases the effect of the system damping. The obvious answer to this problem is to replace the existing bearing design with a new design that is stabilizing but does not produce such high direct stiffness values. This can be accomplished with an optimized tilting-pad bearing. However, changing to a tilt-pad bearing design required modifications to the bearing housing and a considerable amount of downtime. Because of the compressor's long history of low vibration and the fact that the problem occurred after the electrical shutdown, the plant maintenance department still believed that the compressor bearings had been damaged. Also, only a short time period (approximately 24 hours' mechanical time) was available in January 1998 to work on the compressor without impacting plant throughput.

Since the oil seals could be modified easily, they were examined. It was decided to increase the diametral clearance of the bushing seal by 0.001 inch, and decrease the effective L/D of the seal by cutting a 1/16 inch square groove in the middle of the land of the seal (Figure 23). Increasing the clearance and cutting the groove in the bushing seal breaks up the hydrodynamic effect in the seal, which produces the high cross-coupled stiffness (Semanate and San Andres, 1993). It also reduces the radial load capacity of the seal, which causes the modified seal to operate with a higher eccentricity. Reduction of the hydrodynamic effect in the seal could have been accomplished by just increasing the seal clearance. However, this would have resulted in too much seal leakage, since the leakage rate is proportional to the cube of the clearance.

The predicted stiffness coefficients of the modified bushing seal are shown in Figure 24. As can be seen the cross-coupled stiffness, K_{xy}, is approximately 30 percent to 40 percent of the previous value in Figure 15. Its effect on the first mode is to raise the log dec to 0.022 (identical to the value predicted for floating seals, Figure 25). This value indicates a very slight margin of stability.

RESULTS OF MODIFICATIONS

On January 6, 1998, the compressor was shut down to inspect the bearings and make the necessary modifications to the floating

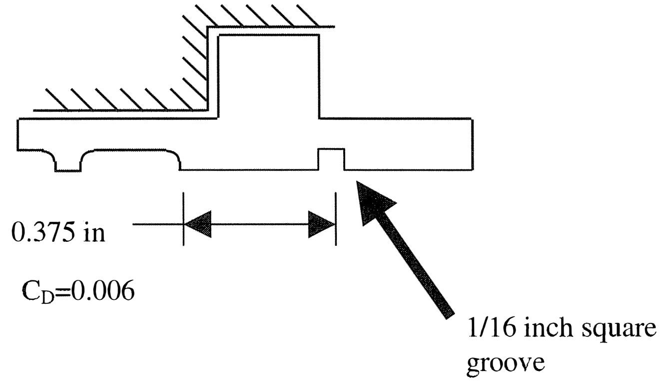


Figure 23. Modifications to Floating Seal Bushing.

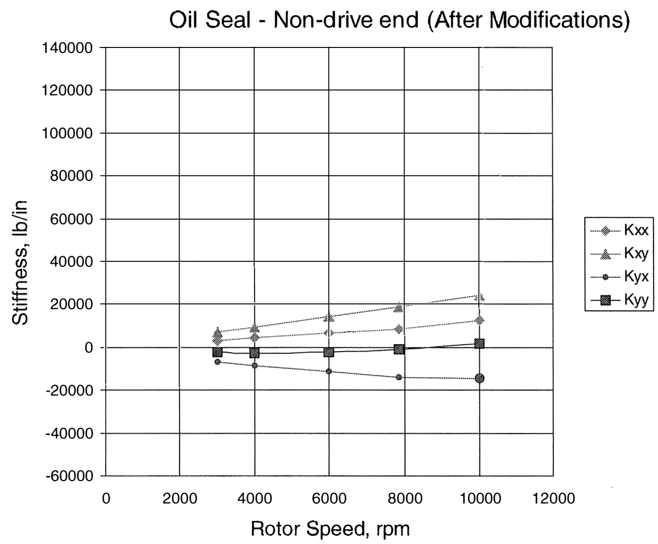


Figure 24. Seal Bushing Stiffness Coefficients After Modifications (One-Half of Seal).

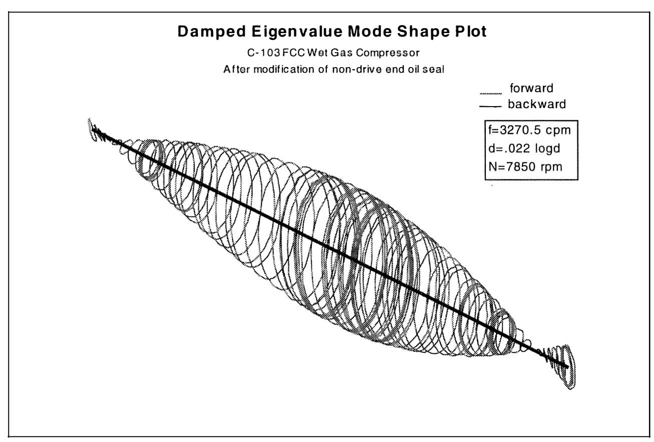


Figure 25. Mode Shape of Compressor After Nondrive Seal Bushing Modification.

bushing. The bearings were found to be well within the OEM clearance tolerance and showed no signs of wear. The modifications were made to the nondrive end seal bushing. After the compressor was started again, the overall levels of vibration were below 1.0 mil on all compressor probes. The spectrum from the compressor outboard vertical probe at full load shows that the subsynchronous component has been reduced from 3.5 mils to 0.1 mil (Figure 26).

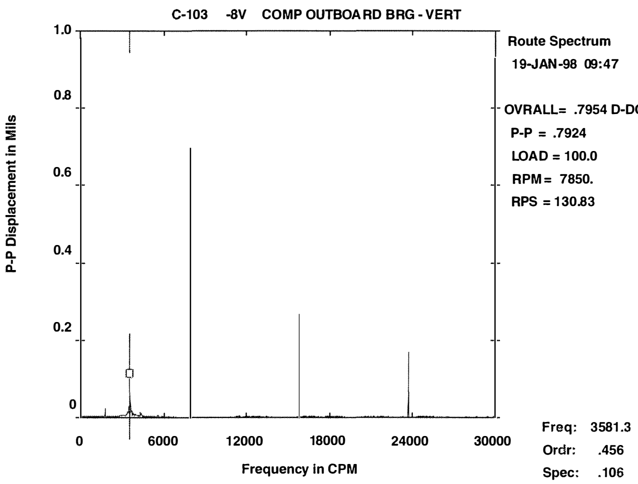


Figure 26. C-103 Compressor Outboard Vertical Spectrum After Seal Bushing Modification.

ADDITIONAL RECOMMENDATIONS

Even though the compressor vibration was acceptable after the modifications to the oil seal, the margin of stability needs to be increased. This can be accomplished by modifying the bearing and/or the balance piston seal design.

The bearing design should be changed to lower the bearing to shaft stiffness ratio (*K*). As mentioned above, this is the root cause of the stability problem. The bearing selected is a five pad, load-on-pad (LOP) bearing, with a preload of 0.2 and L/D = 0.4. Stiffness and damping coefficients for the proposed bearing are shown in Figure 27. This will improve the stability of the first forward mode (the calculated log dec is 0.5), as well as reduce the amplification factor of the rotor to 5.6 (Table 2).

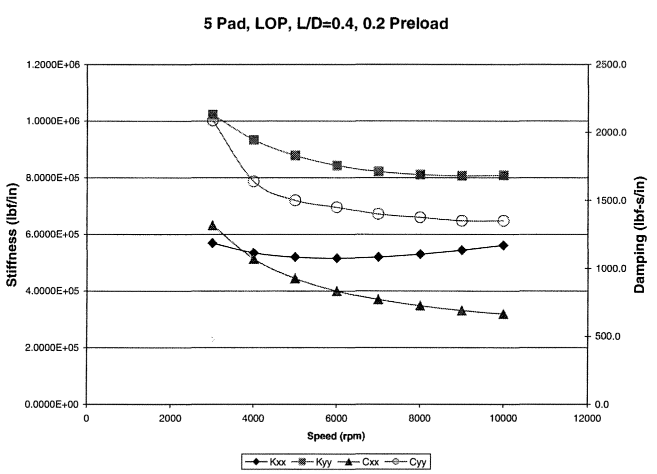


Figure 27. Optimized Tilting-Pad Bearing Stiffness and Damping Characteristics.

Table 2. Comparison of Pressure Dam and Tilt-Pad Bearings.

Modification	Log Dec	Amplification Factor
Pressure dam Bearing w/ seal mods	0.022	18.1
4 pad, LBP bearing w/ seal mods	0.51	5.6

The stability of the compressor can be increased further by replacing the existing TOR design balance piston seal with various designs that are less destabilizing, such as TOS labyrinth, honeycomb, or TAMSEAL™. All three of these options would undoubtedly improve the stability of the compressor, but they would all be very costly in comparison to the proposed bearing design modification and would require a much longer timeframe to install. Besides, the calculated stability with the tilt-pad bearing design does not warrant any further modifications, and the rotordynamic model has already been proven to be accurate.

CONCLUSIONS

Many machines operating today may have only a small margin of stability. Regardless of the number of years of smooth operation, even the most reliable machine may have hidden stability problems. The fixed geometry bearings in many older compressors may have been "optimized" so that the fixed geometry bearings themselves are stable; however, this may limit the overall stability of the compressor itself. The bearing to shaft stiffness ratio (*K*) is an important parameter in accessing the long term stability of a machine. Normally, the value of *K* should be maintained below 10 to 12 to guarantee a stable machine. Obviously, this limitation is dependent upon the amount of direct damping available in the bearings, seals, and/or bearing supports.

Just because a compressor operates at low pressures, does not mean that it cannot be susceptible to destabilizing forces from oil seals and labyrinth seals. The oil seals can easily lock up if the seal is not well balanced and/or the seal faces do not have a very smooth finish. Likewise, even a low differential pressure labyrinth can be a problem if the compressor is only marginally stable and the seal design is somewhat destabilizing.

NOMENCLATURE

- c = Bearing radial clearance
- C_{opt} = Optimum bearing damping
- K = Shaft to bearing stiffness ratio
- K_{ij} = Bearing stiffness tensor
- M = Rotor modal mass
- Q = Synchronous amplification factor
- W = Rotor modal weight
- ω_{cr} = Rotor rigid bearing critical speed

Subscripts

- B = Bearing
- S = Shaft

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