LOW FREQUENCY INSTABILITY IN A PROCESS COMPRESSOR

by

John A. Kocur, Jr.

Machinery Engineer

ExxonMobil Research & Engineering

Fairfax, Virginia

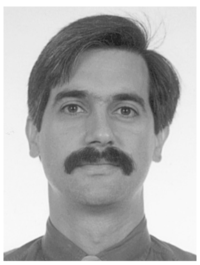
and

George C. Hayles, Jr.

Machinery Engineer

ExxonMobil Research & Engineering

Baytown, Texas



John A. Kocur, Jr., is a Machinery Engineer in the Plant Engineering Division at ExxonMobil Research & Engineering, in Fairfax, Virginia. He has worked in the turbomachinery field for 20 years. He provides support to the downstream business within ExxonMobil with expertise on vibrations, rotor/aerodynamics and health monitoring of rotating equipment. Prior to joining EMRE, he held the position of Manager of Product Engineering and Testing at Siemens Demag Delaval Turbomachinery. There Dr. Kocur directed the development, research, engineering, and testing of the compressor and steam turbine product lines. He has also held positions with Pratt & Whitney and Amoco Corporation.

Dr. Kocur received his BSME (1978), MSME (1982), and Ph.D. (1991) from the University of Virginia and an MBA (1981) from Tulane University. He has authored papers on rotor instability and bearing dynamics, lectured on hydrostatic bearings, has been a committee chairman for NASA Lewis, and is a member of ASME.

George C. Hayles, Jr., is a Machinery Engineer in the Plant Engineering Division at ExxonMobil Research & Engineering, in Fairfax, Virginia. He started at EMRE in 2000 with a primary role of assisting in design evaluation of centrifugal compressors being purchased or revamped. His analyses mainly involved performance simulation and rotordynamics. In 2003 he began an assignment at the ExxonMobil Baytown, Texas, refinery as a rotating equipment contact engineer. There he provides turnaround, repair, and day-to-day operations support for 11 units. His roles include engineering, troubleshooting, and root cause failure analysis of general and special-purpose rotating equipment.

Mr. Hayles received his B.Eng. degree (Mechanical Engineering, 2000) from Stevens Institute of Technology, in Hoboken, New Jersey, and is currently a licensed E.I.T.

ABSTRACT

A catalytic gas process compressor experienced low frequency vibrations in excess of trip levels following installation of a honeycomb balance piston. After 12 years of operation, low seal oil differential forced an unplanned unit shutdown. The source was identified as caustic corrosion of the aluminum balance piston. A stainless steel (SS) honeycomb seal was installed in place of the aluminum labyrinth seal. During the next two weeks, several trips of the compressor train were experienced caused by high levels of subsynchronous vibrations. Ties of the vibration problem to process conditions were studied and tested. Rotordynamic analysis identified the combination of damper bearing and honeycomb balance piston seal as the source of the low frequency vibrations.

This paper describes the operating history of the compressor, the vibration data recorded during a vibration event, teardown inspection results, and the subsequent rotordynamic analysis. The combination of low support stiffness with negative honeycomb stiffness lowered the frequency of the first natural frequency to a point that the honeycomb seal became destabilizing. To minimize unplanned capacity loss of the unit, an aluminum labyrinth seal was reinstalled, the caustic source corrected, and the unit restarted. The high levels of low frequency vibration have been eliminated and the unit has operated without vibration incidents for over 12 months.

INTRODUCTION

The C-30 catalytic gas compressor, installed in 1991, experienced vibration excursions in excess of the trip alarm for the first time in March 2003. A low frequency component, ≈6 percent of running speed, was identified as the source of the high vibration levels. The vibration occurred randomly over a two to three week period and would appear suddenly without apparent cause. Vibration levels would peak over 6 mils on the discharge end of the compressor.

Efforts were made to identify a correlation between the vibration and operating condition of the unit. Flow rates, pressure, and speed trends were examined during excursion events. An operating condition was selected based on tests where no subsynchronous vibrations were present. The compressor was operated at these conditions overnight. During that period, several vibrations were experienced but brought under control by lowering speed and flow.

Following the operational test, the compressor was shut down for inspection. One of the possible causes was thought to be rotor instability as an outcome of the installation of the honeycomb seal. The paper presents the results of the inspection and the vibration data taken during the excursions. The rotordynamic analysis used to identify the problem is described and the results of the stability analysis are presented. The solution path was influenced by the desire to limit further unit downtime.

UNIT DESCRIPTION

At the refinery, the light ends unit takes catalytic (cat) and refinery gas from numerous sources. The unit's principle purpose is to recover ethane and heavier hydrocarbons such as propane and propylene from the feed gases, which are sent to other units in the complex.

When the cat gases enter the light ends unit, condensate and liquids are removed from the stream via a knockout (KO) drum. Before the C-30 compressor compresses the cat gas, ammonia, carbon dioxide, and hydrogen sulfide are stripped. Ammonia is removed first by using a water wash tower. An amine absorber tower next strips away the majority of hydrogen sulfide and carbon dioxide. Any traces of amine that may have carried over from the previous tower are neutralized by use of a caustic/water wash tower. A water wash neutralizes the caustic. After passing through a final KO drum, the cat gas stream is ready to be compressed. After compression, the stream is condensed and sent to the fractionator.

Even with the water wash and KO drum upstream of the compressor, there is a possibility of caustic carrying over. Caustic is known to corrode some metals (e.g., aluminum), so the original design specifications for the compressor called for materials of construction to be resistant to caustic attack.

C-30 Compressor Description

The C-30 compressor was manufactured in the late 1980s and uses a steam turbine driver to accommodate variations in the cat gas stream. The interstage seals are labyrinth (laby) style with straight teeth. Due to the possibility of caustic entering the compressor, these labys are made of a thermoplastic material. The original balance piston in C-30 was also going to utilize thermoplastic material. However, during factory testing it was found that the balance piston was deforming under the design pressures. A honeycomb design made with stainless steel was considered. This option involved a long manufacturing lead-time but provided a seal resistant to caustic corrosion. A standard aluminum labyrinth would shorten the manufacturing cycle but not provide the caustic corrosion resistance.

Due to the timing constraints of installation and startup, the compressor was retrofitted and tested with the aluminum laby balance piston. A honeycomb was also manufactured and placed into storage at the refinery. The plan was to retrofit the aluminum laby with the stainless steel honeycomb at the next opportunity (e.g., the first overhaul). The design operating conditions for C-30 are presented in Table 1.

Table 1. Compressor Rotor Characteristics.

Unit	C-30	Service	Cat Gas	Molecular Weight	28
P_1, psia	150	P_2, psia	550	Temp In - Temp Out	100 °F - 270 °F
Rated Power, hp	9000	Rated Speed, rpm	10,736	MCS Speed, rpm	11,273
Driver Type	Steam Turbine	Rotor Weight, lbm	697	Shaft Length in	Renk (1)
# of Impellers	6	Bearing Span, in	57	Shaft Diameter @ mid Span, in	5.25

COMPRESSOR OPERATION

Initial Compressor Shutdown

The C-30 compressor train was commissioned in 1991. For the first 12 years of operation, vibration levels stayed within acceptable limits, and the compressor was able to meet process demands. On February 20, 2003, in the early morning C-30 tripped due to low seal oil differential pressure. The compressor was restarted a number of times, and each time the compressor would trip from the same interlock. Increasing the seal oil pressure did not alleviate the problem.

The C-30 seal oil system uses a reference pressure from the low-pressure side of the balance piston, which is tied to compressor suction by the balance line. By applying a set delta pressure to this reading, the seal oil system can provide the correct amount of oil pressure to the seals for proper sealing and cooling. Normal suction pressure of the compressor is ≈150 psi. The reference pressure at the time of the trip was ≈250 psi. The preliminary causes of this higher pressure were narrowed down to two possible situations that can cause the pressure in the seal reference cavity to increase.

- The balance line was plugged thus restricting flow to the compressor suction.
- Increased balance piston leakage flow due to wear or other damage opening the seal clearance. Increasingly large flows will create a large pressure drop across the balance line, which raises the cavity pressure.

Additional data relating to the thrust bearing taken prior to the trip also pointed to a problem with the thrust balance. Over a period of two days before the trip event, the thrust bearing temperature on the active side rose from 130° to 200°F as seen in Figure 1. The change in thrust bearing temperature independent of process conditions was an indication that the balance piston and/or balance line condition was affecting the net thrust of the compressor.

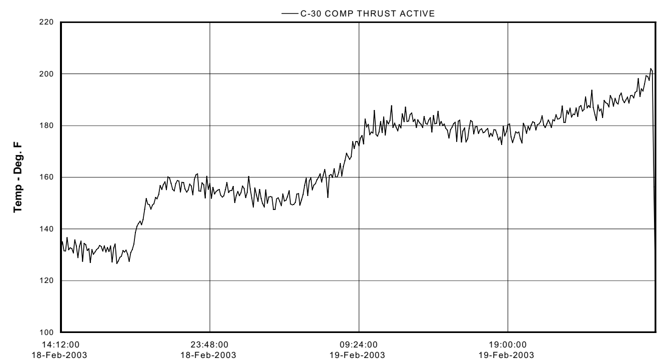


Figure 1. Active Thrust Bearing Temperature.

Due to concerns around the oil seals after the restart attempts, the compressor was shut down. The compressor was sent to the manufacturer's service shop for disassembly and overhaul due to the compressor run length since the initial startup (12 years). When the compressor was disassembled, an assessment of the compressor components identified these pertinent findings:

- Considerable buildup of material was found within the compressor. The balance piston stator was clogged with this material (Figure 2). This material is believed to be caustic material that had carried over from the caustic washing facilities upstream of the compressor.

Figure 2. Labyrinth Seal Balance Piston Following 12-Year Run.

- Upon cleaning up the balance piston for measurement, the diametrical clearance was measured at 70 mils versus a design specification of 10 to 14 mils.
- Disassembly of the thrust bearing revealed heavy varnishing on the active thrust pads. On the inactive side, uneven wear was seen on the pads. This was seen as support for the thrust bearing temperature readings that were recorded two days prior to the trip event.

- When the seals were taken apart, the Teflon® ring that serves as the shutdown seals was deformed on the discharge end and destroyed on the intake end. This was an indication of the shutdown seals having been activated multiple times due to the multiple trips from the restart attempts.

- Inspection of the balance line revealed that it was not clogged. Without an artificial restriction, the increase in pressure was therefore due to excessive flow in the line caused by caustic corrosion opening the labyrinth clearance.

At the shop, the honeycomb balance piston stator was installed as the spare part. When the compressor was returned to the refinery after the overhaul, action was taken to provide an additional way to assess the condition of the balance piston area as well as the balance line in the future. A bleeder valve was installed on the balance line to enable use of a pressure gauge. This gauge will help determine if a problem within the balance piston area and/or balance return line exists. C-30 was installed and subsequently started up on March 6. Upon investigation it was found that caustic did carry over from the upstream caustic/water wash tower. The caustic combined with entrained water vapor attacked the aluminum balance piston.

Second Shutdown

After eight days of operation at normal vibration levels (average reading was 0.5 mils peak-to-peak on the compressor inboard [IB] end, and 0.8 mils peak-to-peak on the compressor outboard [OB] end), C-30 experienced an excursion on the morning of March 14. Vibration readings on the IB end of the compressor rose to about 3.6 mils. OB vibration readings rose to about 1.8 mils. Vibration levels did not drop to below 1 mil for another 82 minutes, but also did not rise to the compressor trip alarm level. The compressor vibration monitoring system is set to trip the compressor if displacement exceeds 6 mils.

A second excursion exceeding the 6 mils trip level occurred that evening. Vibration data showed the trip vibration levels to be only on the IB end (the same side where the balance piston is located). The OB end reached about 4 mils in this excursion. The IB vibration magnitude reached 6 mils briefly.

In these and later excursions that occurred between March 14 and 18, the spectrum and orbit patterns on both ends were very similar regardless of the vibration level. Figure 3 shows the source of the increase in vibration levels to be due to a subsynchronous vibration component at 0.063× running speed. The subsynchronous vibration was larger in magnitude at the IB end of the compressor (discharge end).

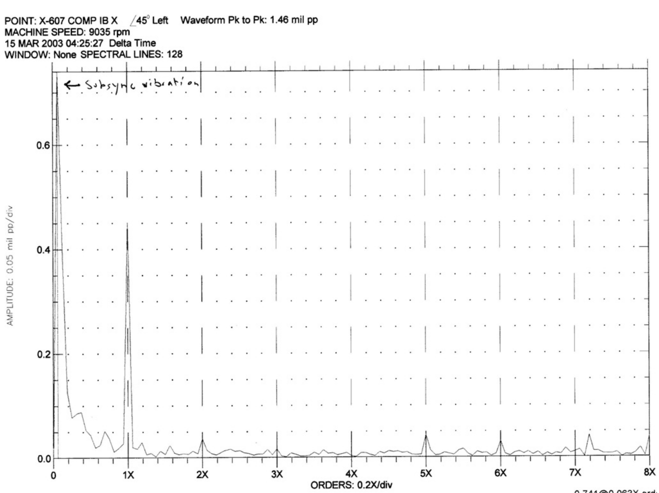


Figure 3. Compressor IB End Spectrum Plot During Vibration Excursion.

The problem area was initially believed to be the balance piston based on evaluation of the vibration data. Other potential factors were considered, e.g., the amount of water in the lube/seal oil system. But the water levels were comparable to levels during the 12-year run and this was discounted as a factor. One theory that warranted checking was that the excursions were related to the load on the compressor (i.e., can only happen at a certain speed and flow).

To identify the possible correlation of the vibration with load, a test was performed on March 17. In this test, C-30 was first set at a reduced load as a starting point; compressor speed was ~9.45 krpm, and flowrate was ~69 mmscfd. The load was slowly increased (by means of increasing speed and flow) while the vibration readings were monitored.

Throughout the test no vibration excursions were witnessed; the magnitude of IB and OB vibration did not exceed 1 mil. At the end of the test, the compressor was running at ~10.2 krpm, and the flowrate was ~90 mmscfd. This was at a much higher speed and flow when compared to the conditions when the first excursion events occurred. With the operating and mechanical conditions normal and stable, the compressor was left to run overnight.

Three vibration events occurred during the overnight run, but were brought under control before the vibration reached trip levels. The operator was able to adjust the speed and flow to slightly reduce the load on the compressor, such that the magnitude of vibration did not reach levels that would trip the compressor. Again, the orbit and spectrum readings were very similar to past excursions. Figures 4, 5, and 6 show similar patterns to the March 15th data. Notice that the subsync vibration at 0.063×. There were occasions in which an excursion reached trip levels rapidly, preventing the operator from making adjustments. The compressor IB end vibration is plotted against time on Figure 7. The rapid rise (though not instantly) of the vibration is apparent.

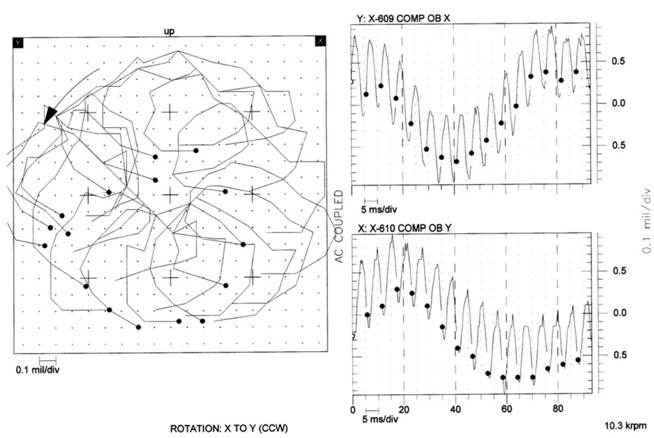


Figure 4. Compressor OB End Vibration During Excursion.

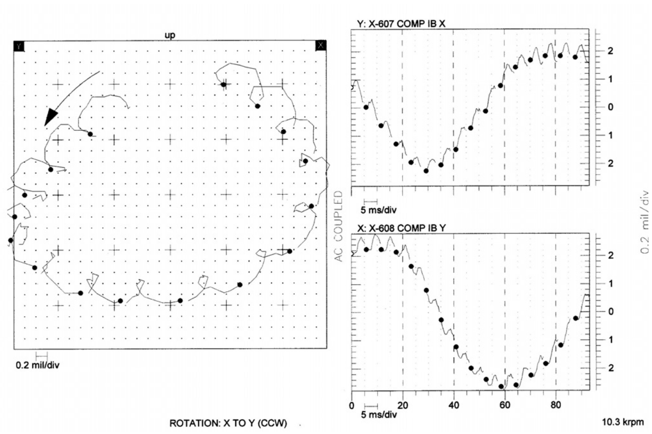


Figure 5. Compressor IB End Vibration During Excursion.

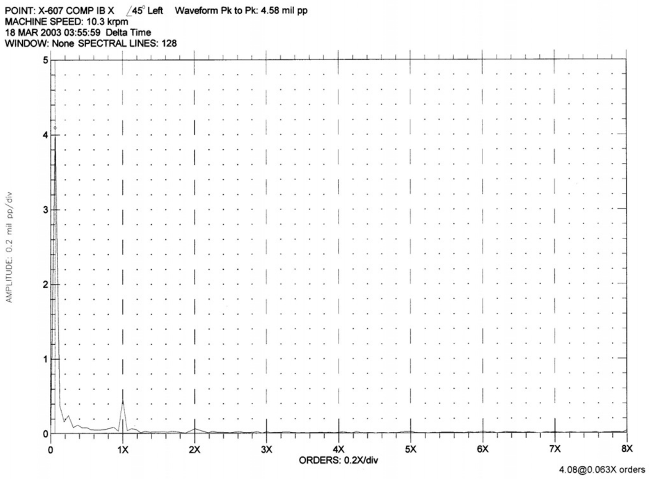


Figure 6. Compressor IB End Vibration Spectrum.

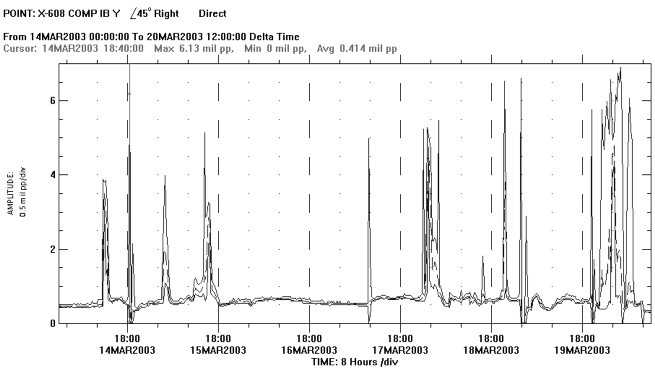


Figure 7. Compressor IB End Vibration Versus Time.

Figure 8 is a graphical representation of the speed and flow at the various excursion events between March 14 and 20. The compressor speed and flow over this period, combined with the vibration trends, indicated that the excursion phenomenon could happen at different compressor loads and speeds. Changing the load slightly was usually enough to bring the compressor out of the whirl and return vibration levels to normal.

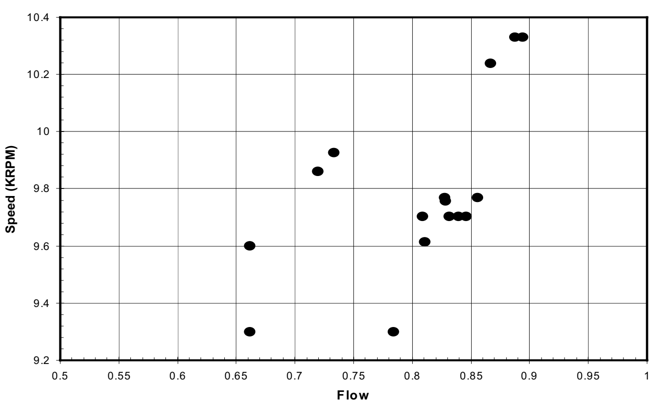


Figure 8. Speed Versus Flow During Vibration Events.

On the morning of the 20th, the compressor was shut down for a second time and sent back to the manufacturer's service shop for an evaluation of the compressor's condition. When the compressor was opened, the honeycomb balance piston was found mostly plugged with what was later identified as caustic (Figure 9). The plugged cells were on the inlet or high-pressure side of the balance piston.

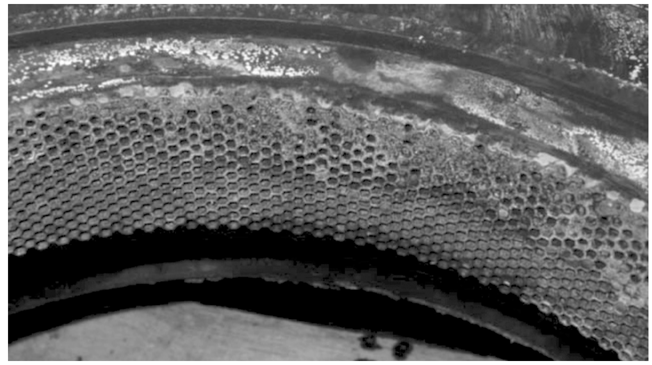


Figure 9. Honeycomb Balance Piston Plugging Due to Caustic Carryover.

ROTORDYNAMIC ANALYSIS

With a possible connection between the honeycomb condition and rotor vibration, a rotordynamic analysis of the C-30 compressor was undertaken. To help diagnose and eliminate the radial vibrations hampering operation, a mechanical/dynamic source was sought. This was assigned a high probability given that the balance piston seal configuration was changed following the first shutdown from a labyrinth to a honeycomb configuration. This represented the only change that could affect the compressor vibrations.

Compressor aerodynamic operation had not changed significantly since the installation of the machine with the exception of the small effects due to caustic fouling. As later determined, this condition had existed prior to the installation of the honeycomb balance piston and, in fact, was the cause of the aluminum balance piston corrosion. However, until the failure of the seal oil pumps to maintain a differential, the caustic carryover did not significantly affect performance or, more importantly, the vibration level. To explain the newly created vibration sensitivity, the analysis would focus on the change in dynamic behavior of the rotor/bearing system resulting from the change in balance piston design.

The nature of the vibration shared traits with classical instability problems in rotating machinery, i.e., rapid rise and fall in subsynchronous vibration. However, the extreme low frequency did not correlate with other documented incidents (Wachel, 1982). These were normally witnessed at ≈35 to 50 percent of running speed. In the C-30 compressor, the subsynchronous frequency was occurring at 6 percent of running speed. In addition, a honeycomb seal has classically been used to solve or prevent rotor instabilities (Zeidan, 1993), not create them. The same can be said for the squeeze film damper (SFD) radial bearings employed by the manufacturer. A scenario was required that would explain why an instability was created by components normally associated with solving that problem.

Two possible scenarios were examined that could produce the phenomena witnessed at the refinery. Both deal with the location of the first critical speed of the compressor. The first scenario assumes the critical speed is located at ≈600 rpm due to abnormally low support stiffness. The second hypothesizes that the first natural frequency is lowered by a relatively large negative stiffness during normal operation. To achieve the first, "very soft" support stiffness is required to lower the critical speed to the required speed. It can be approximated by the simple equation,

$$\omega = \sqrt{k / m}\,, \tag{1}$$

since the rotor will basically act as a rigid body at 600 rpm. For this case, the support stiffness would need to be 3000 lbf/in. This is unreasonably low even for O-ring supported dampers.

The second hypothesis would place the critical speed, as defined by the American Petroleum Institute (API), at an expected rotor speed for damper supported rotors. At normal operating speeds, the mode is then lowered in frequency by a pressure driven component, e.g., a honeycomb seal.

Results of the rotordynamic analysis are presented for 9875 rpm operating speed. This corresponds to the compressor speed at which the subsynchronous vibration typically appeared. For clarity, the paper will present the results of the damped eigenvalue analysis. While a complete analysis was performed, undamped and forced response calculations did not contribute to the explanation of and solution to the subsynchronous vibration problem.

Rotor Model

The C-30 compressor is a six-stage straight-through compressor. Train details and operating characteristics are shown in Table 1. Following standard techniques for lumped mass models, the rotor model takes the form as presented in Figure 10. The radial bearings are five pad tilt pad bearings. The bearing configuration is presented in Table 2.

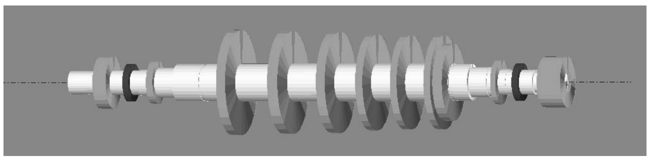


Figure 10. C-30 Compressor Rotor Model.

Table 2. Tilt Pad Bearing Geometry and Operating Parameters.

Type:	Tilting Pad	No. of Pads:	5	LOP	Pivot:	55% Offset	Pad Angle:	52°

	otal Load lbf	Unit Loading lbf/in^2	Length in	Dia. in	L/D	Preload	Diametrical Clearance, mils		Clearance Ratio C_b / Diameter (0/00)
							C_p	C_b	
Drive-end	332	63	1.5	3.5	0.4275	0.31	9.5	6.5	1.86
Thrust-end	365	69.5	1.5	3.5	0.4275	0.31	9.5	6.5	1.86

Bearing Operating Conditions

Preload Range			Oil Type	Normal Oil Temperature	Oil Temperature Range
0.21	0.31	0.41	ISO VG 32	120 °F	110 - 130 °F

The radial bearings are supported by O-ring centered squeeze film dampers. The stiffness of the O-rings plays an important role in determining the dynamic behavior of the rotor/bearing system (Kuzdzal and Hustak, 1996). In an attempt to better quantify that value, load versus displacement measurements were taken during the disassembly of the C-30 compressor. For the range of loading applied, a stiffness of 55,000 lbf/in was projected for loads equivalent to the half weight of the shaft. This was close to the 50,000 lbf/in value used by the manufacturer in the original rotordynamic analysis. While O-ring material has been shown to be frequency dependent (Lakes, 1998), at the low frequency under investigation, ≈10 Hz, it was assumed to behave similarly to static measurements. The damping contribution of the O-rings was neglected.

Tilt pad bearing coefficients were calculated using a solver based on the method developed by Nicholas, et al. (1979). The tilt pad bearing and SFD coefficients are combined in a manner described by Nicholas and Barrett (1986). At a compressor speed of 9878 rpm, the combined coefficients are presented in Table 3. (An assumed whirl frequency of 10 Hz is used to calculate the coefficients.) The support stiffness is significantly lower than the 300,000 lbf/in tilt pad bearing stiffness due to the softness of the O-ring support. This is desirable in SFDs. Motion in the damper, and not the bearing, maximizes the effectiveness of the damping produced in the squeeze film.

Table 3. Equivalent Support Stiffness for Combined Tilt Pad SFD Supported Bearings.

K_{xx} (lbf/in)	K_{yy} (lbf/in)	C_{xx} (lbf-s/in)	C_{yy} (lbf-s/in)
40819	48180	291	421

Cross-coupled coefficients were two orders of magnitude smaller than the principle terms and ignored for this analysis. An average gravity load is used for each radial bearing and identical coefficients used for each location.

Balance Piston Models

As noted earlier, the original configuration of the compressor as shipped contained a tooth-on-stator labyrinth seal. The seal consists of 15 teeth spaced 0.125 inch apart at a diameter of ≈10.4 inches. Since the compressor layout is straight through, the balance piston is exposed to the full pressure drop across the machine. The dynamic coefficients are based on a method developed by Kirk (1987). The seal analysis is tied to a proprietary performance program to get accurate flow properties at the entrance to the seal.

The honeycomb seal installed following the first shutdown was sized as a direct replacement for the labyrinth seal. Diameter and length remained unchanged. The honeycomb cell size was approximately the same as the tooth spacing for the labyrinth seal. The honeycomb seal dynamic characteristics were calculated using a method developed by Kleynhans and Childs (1997).

Both seals were exposed to the entire differential pressure across the compressor. The swirl ratio was determined to be 0.52 for the gas entering the labyrinth seal. This ratio was used for both seals. While the impact of inlet swirl on the dynamic behavior of gas seals is well known (Childs, 1993), time constraints to get the compressor back into service prevented any possibility of including anti-swirl devices or shunt injection.

The frequency dependence of the dynamic coefficients of honeycomb seals is well known to the authors' company. Camatti, et al. (2003), present a case history on a back-to-back injection compressor driven unstable in large part due to the honeycomb seal. During Type 1 testing, subsynchronous vibration was experienced on the first body of the injection train. The frequency of vibration was lower than expected. (However, not to the extent witnessed in the C-30 compressor.) Further investigation into the honeycomb seal behavior revealed that not only does the effective stiffness and damping decrease with lower whirl frequencies, the effective stiffness is strongly influenced by the clearance profile. For divergent profiles, effective stiffness could become largely negative of the frequency range. Coupled with the frequency dependence of the coefficients, the authors found very low natural frequencies and very negative logarithmic decrements.

With this experience and information in hand, the behavior of the labyrinth seal was compared against the honeycomb seal at 9878 rpm. The effective stiffness and damping coefficients are plotted against frequency on Figures 11 and 12 for both seals. Based on the linearized model developed by Childs (1993) for forces acting on a rotor undergoing centered circular orbits, the effective stiffness and damping are defined as presented in Equations (2) and (3).

$$K_{eff} = K - c \cdot \Omega \tag{2}$$

$$C_{eff} = C - \frac{k}{\Omega} \tag{3}$$

where:

K = Principle stiffness, lbf/in
k = Cross-coupled stiffness, lbf/in
C = Principle damping, lbf-s/in
c = Cross-coupled damping, lbf-s/in
Ω = Whirl frequency, rad/s
K_{eff} = Effective principle stiffness, lbf/in
C_{eff} = Effective principle damping, lbf-s/in

As can be seen in the Figures 11 and 12, both seals are frequency dependent in their behavior. Any nonsynchronous rotordynamic calculations must therefore take this behavior into account. For damped natural frequency calculations, the component frequency dependence must be entered directly or the eigenvalue iterated upon using assumed whirl frequencies to obtain the coefficients.

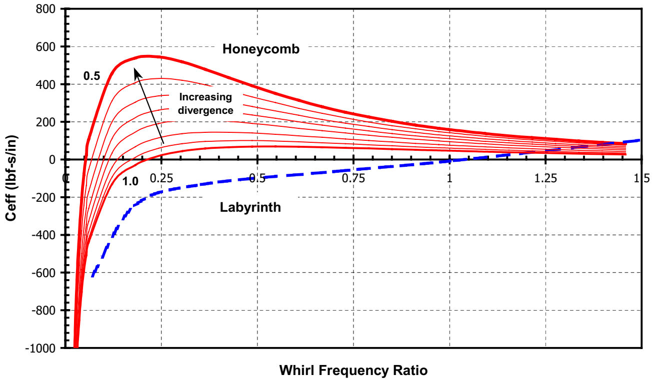


Figure 11. Effective Damping—Honeycomb and Labyrinth Balance Piston.

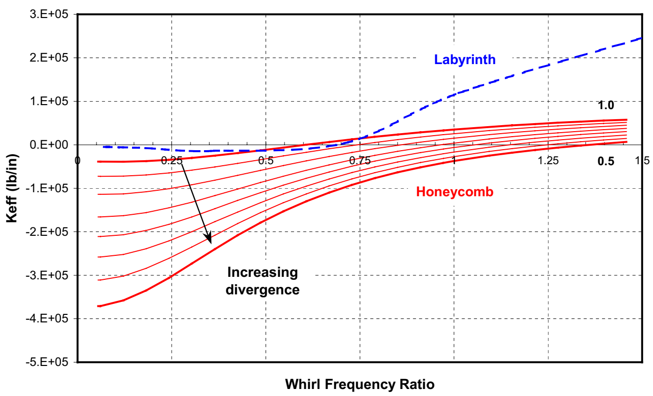


Figure 12. Effective Stiffness—Honeycomb and Labyrinth Balance Piston.

Both seal configurations share the same behavior of the effective damping versus whirl frequency, namely at low whirl frequencies both produce destabilizing forces of approximately the same magnitude. However, the defining difference in the two components is the relation of effective stiffness versus whirl frequency. The labyrinth seal is predicted to have slightly negative principle stiffness at low whirl frequencies. At these levels, the labyrinth seal would not alter the frequency of the damped eigenvalues of the system. With the honeycomb seal, the potential to lower the frequency does exist with a divergent clearance profile. This is similar to the behavior witnessed and predicted by Camatti (2003).

Divergence in the honeycomb seal was modeled by reducing the inlet clearance while holding the exit clearance constant. A divergence factor is shown on the plots and is simply the ratio of the inlet to exit clearance. For example, a factor of 1.0 refers to a cylindrical clearance profile. A factor of 0.5 indicates that the inlet clearance is 50 percent of the exit. The figures present the effect of this variation over a range of 1.0 to 0.5. At tapers approaching 0.5, the negative stiffness nearly equals the positive stiffness provided by the radial bearings.

Stability Analysis

A damped eigenvalue analysis of the C-30 compressor was performed for the two configurations run in the plant. A calculation including only the rotor and bearings was added as a basis for comparison. The calculations were performed at 9878 rpm and included the frequency dependence of the equivalent support stiffness models and balance piston. Table 4 presents the results of the analysis for the first forward damped natural frequency for the various configurations. In all cases, a destabilizing force equal to that specified in API 617, Seventh Edition (2002), was used.

Table 4. Damped Eigenvalue Results for the First Forward Mode.

Configuration	First Forward Mode	
	Frequency (cpm)	Log decrement
Rotor / bearings only	2245	2.12
w/ labyrinth balance piston	2432	0.36
w/ honeycomb balance piston (cylindrical clearance)	1478	1.22
w/ honeycomb balance piston (0.86 divergent clearance)	751	-6.18

The plot of the mode shape reveals the expected rigid shaft behavior. Figure 13 illustrates the 3-D mode shape associated with the compressor configured with the labyrinth balance piston. The mode shape for the compressor with a slightly divergent clearance is shown in Figure 14. The discharge end of the compressor can be seen as the driving force represented by the larger deflections at the respective end of the shaft.

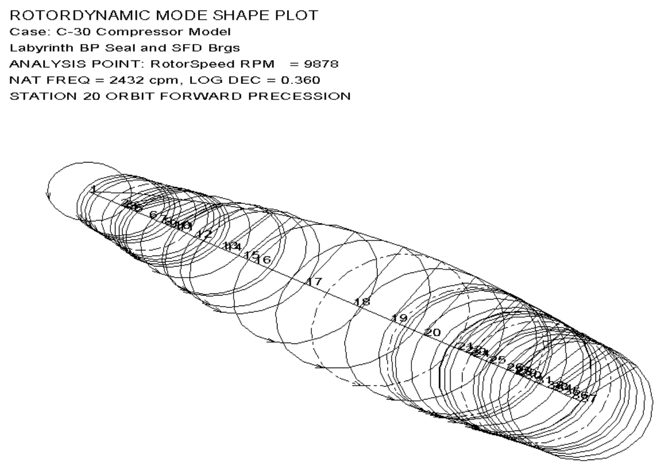


Figure 13. First Forward Mode Shape—Labyrinth BP.

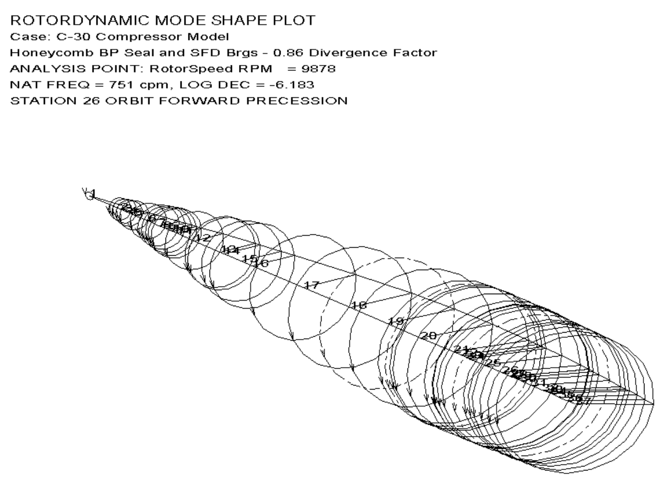


Figure 14. First Forward Mode Shape—Honeycomb BP (0.86 Divergence Factor).

Several important findings can be drawn from the analysis:

- The soft support of the SFD radial bearings significantly lowers the location of the first natural frequency. As designed, the SFDs soften the support relative to the shaft stiffness. This increases the support damping at the rotor center and, therefore, the stability and response of the shaft. However, the soft supports lower the normal force exerted on the shaft increasing the sensitivity of the rotor to negative stiffness.

• The labyrinth balance piston, due to its almost zero effective principle stiffness, does not alter the frequency of the first damped eigenvalue. The destabilizing effects of the labyrinth seal are evident with the decrease in logarithmic decrement (log dec) of this mode versus the rotor and bearings only configuration. However, the compressor is still predicted to be stable and is confirmed by the 12-year operating history without subsynchronous vibration.

• The negative effective stiffness of the honeycomb balance piston, even with a cylindrical clearance profile, is sufficient to lower the first mode on soft supports by nearly 1000 cpm. The whirl frequency ratio (WFR) of this configuration is still high enough for the honeycomb to have a positive effective damping, thus increasing the log dec of the mode.

• Reducing the inlet clearance by 14 percent (roughly 1 mil radially) is sufficient to further drop the frequency of the first mode to 751 cpm. At this WFR, 0.076, the honeycomb is now largely destabilizing. This is reflected in the log dec indicating an unstable mode.

The authors feel that the dramatic change in the rotor stability based on small changes in the clearance profile helps explain the apparently random appearance of the subsynchronous vibration. A photograph of the balance piston following the two-week run (Figure 9) showed significant fouling of the honeycombs on the inlet side. Fouling due to caustic carryover would occur in distinct events rather than a gradual build up over time. These events would deposit material at the inlet of the honeycomb. While an analysis of a partially plugged honeycomb seal was unavailable, the authors believe that this phenomenon could be approximately modeled as a closing of the inlet clearance. The balance piston would trigger the instability as material deposits, occurring at distinct events, affected the inlet clearance.

RECOMMENDATIONS

Factors beyond analysis results influenced the decision on how to proceed with the solution to the C-30 compressor vibration. Always present is the desire to minimize unit downtime. The primary consideration was to find a solution that could return the unit to service in the shortest time possible but still ensure reliable operation until the next planned shutdown. Other recommendations for more drastic and long-term changes would then accompany this plan.

• *The aluminum labyrinth was recommended for installation in the compressor*—This represented the shortest turnaround time for the unit. The labyrinth balance piston also had a 12-year operating history in the compressor without vibration problems. Reliable operation was reasonably assured given that the caustic problem could be resolved.

• *Examination of the upstream scrubbers and knockout drums were required to ensure efficient removal of caustics/deposits from the gas stream*—As noted earlier, aluminum material will easily corrode in the presence of this contaminant combined with water. Without remedial action, the life expectancy of the balance piston seal would fall short of the desired run time of the compressor.

• *A long-term solution to the balance piston and rotor support was recommended*—With forces approaching those found in oil film bearings, proper design of the rotor/bearing system is critical when honeycomb seals are applied. Proposed alternate designs for the balance piston and bearings were:

1. *Honeycomb balance piston/tilt pad bearings (no dampers)*—While not presented in this analysis, an increase in the support stiffness to levels equivalent to tilt pad bearings without SFDs was sufficient to prevent the honeycomb seal from lowering the mode's frequency and driving it unstable.

2. *Rotating steel laby teeth and fluorosint stator with damper bearings*—More rub tolerant than a honeycomb seal, this configuration would need to overcome the design problems of the original attempt at the manufacturer's facility.

3. *Thermoplastic stationary laby teeth with damper bearings*—A frequently used substitute for option #2 that also supplies rub tolerance with resistance to caustic corrosion. The suitability of either option would depend on the thermoplastic's or fluorosint's ability to withstand the operating pressure and temperature of the application.

SUMMARY

Following 12 years of operation, the cat gas C-30 compressor damaged the original aluminum labyrinth balance piston through caustic corrosion. The manufacturer supplied spare part, a steel webbed honeycomb seal, was installed. After only two weeks in operation, the C-30 compressor experienced several trips due to vibration levels in excess of 0.006 inch. The source of the high levels was identified as a subsynchronous component. Failing to tie the vibration to a specific set of operating conditions, a rotordynamic study of the compressor was undertaken.

The analysis revealed that the combination of SFD supported tilt pad bearings and the honeycomb balance piston led to the creation of an unstable configuration of the C-30 compressor. While frequently used to solve instability problems, these components created a situation where the first lateral natural frequency was lowered to ≈10 Hz or 6 percent of operating speed. At these whirl frequencies, the honeycomb balance piston became destabilizing. The labyrinth balance piston, while predicted to have a similar destabilizing potential, did not share the same negative principle stiffness levels and, thus, did not lower the first mode. Due to pressures to get the unit back into operation, reinstalling the labyrinth seal was identified as an acceptable solution provided the source of caustic in the process could be identified and corrected.

FOLLOW-UP REPORT

Prior to the restart of the C-30 compressor train in late March 2003, the upstream components were examined. Problems with the upstream scrubbers were identified as the source of caustic in the C-30 compressor. These were corrected. While the source of water in the oil system was identified as the steam turbine, correction of this problem involved longer shutdown times than desired. The oil system is monitored closely for water content and replaced as necessary.

The compressor has operated 12 months without incident following installation of the labyrinth seal and correction of the caustic removal processes.

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