

IMPROVING THE RELIABILITY OF A HIGH SPEED REFRIGERATION COMPRESSOR

by

Ray D. Kelm

Owner

Kelm Engineering

Angleton, Texas

and

Malcolm E. Leader

Turbomachinery Consultant and Owner

Applied Machinery Dynamics

Dickinson, Texas



Ray Kelm is the Owner of Kelm Engineering, in Angleton, Texas. His primary duties involve numerical modeling and field testing of dynamic systems including rotating, reciprocating, and static machinery. He has 24 years of experience in the petrochemical and engineering consulting businesses.

Mr. Kelm received a BSME (1983) from Texas A&M University, and an M.S. degree (Mechanical and Aerospace Engineering) from the University of Virginia. He is member of ASME and the Vibration Institute, and is a Category VI ISO certified vibration specialist.



Malcolm E. Leader is a Turbomachinery Consultant and Owner of Applied Machinery Dynamics, in Dickinson, Texas. He is currently involved in the design, testing, modification, and installation of rotating equipment. He spends time doing theoretical design audits and working in the field implementing changes and overseeing installations.

Mr. Leader obtained his B.S. (1977) and M.S. (1978) degrees from the University of Virginia. While there, he worked extensively on experimental rotordynamics and hydrodynamic bearing design. He has written several papers on the subjects of experimental rotordynamics, bearing design, design audits for rotating equipment, and practical implementation of rotordynamic programs. Mr. Leader is a member of ASME, Sigma Xi, the Houston Chapter of the Vibration Institute, and is a registered Professional Engineer in the State of Texas.

ABSTRACT

A high speed (15,200 rpm) refrigeration centrifugal recycle compressor was experiencing severe reliability problems resulting in short run times between major overhauls. Typical run times were one to 12 weeks. The symptoms identified at the start of the project included oil foaming in the closed loop oil sump on the end of the compressor, sudden and severe damage to the balance piston and interstage seals shortly after startup, high thrust bearing oil drain temperature (due to high balance piston leakage), and high levels of process contaminant fouling on the impellers during operation. A solution was pursued using a systems approach including review

of repair practices, hardware installation, operating practices, lubrication system design, and rotordynamic analysis of the rotorbearing system.

Review of casing vibration during a startup cycle of the compressor was found to include evidence of surging, and generally high amplitude casing vibration even though the casing to rotor weight ratio was very large (~40:1). Inspection and review of the lubrication system indicated that the system had previously been modified on several occasions from the original equipment manufacturer (OEM) design in various attempts to improve the compressor reliability. In addition, several grades of oil including synthetic were used at various times. However, the oil sump continued to exhibit heavy foaming not just during startup but under normal continuous operation. Although the rotor design appeared to be consistent over the life of the machine, the journal bearings were reported to have been changed to several different styles including pressure dam bearings, fixed lobe, and with various bearing materials of construction.

Oil foaming problems under continuous operating conditions were eliminated by repair of plugged oil mist coalescing elements in the oil sump, and improving the oil/refrigerant flow paths into the oil sump. A complete lateral rotordynamics study found that the existing bearings were not a good choice and that the rotor was operating on the first critical speed. New bearings were optimized and a "dummy" impeller was added to the rotor to reduce the critical speed to well below operating speed. The results of the bearing and rotor modifications reduced the amplification factor from 19 to less than 2.5 at the first critical speed, and predicted a 90 percent reduction in center-span operating speed vibration amplitudes for a given level of rotor unbalance, greatly reducing the risk for internal rubs.

The tremendous success of this project emphasizes the clear requirement to address problems of this sort using a systematic, engineered approach as opposed to a trial and error solution strategy. The charge of the reliability improvement team was to identify positive changes that could be reasonably implemented to lengthen run time to repair. The final solution included some rather simple maintenance modifications (oil system) as well as some very sophisticated hardware modifications (bearing and rotor design changes). However, review of the whole system was necessary to identify the key components that would lead to rapid success in the plant.

INTRODUCTION

A multistage centrifugal compressor was used in a chemical processing plant in a combined refrigeration and process service. The service was initially intended to provide process gas recycling

and refrigeration for various cooling demands within the plant. The process gas stream was primarily a hydrocarbon refrigerant, with some powder/dust carryover. Although various filtration systems were used, some level of powder was always located downstream of the filters that would foul the compressor.

The compressor in question is a 15,200 rpm, 1400 hp centrifugal compressor with three impellers, shown schematically in Figure 1. The compressor casing was originally selected from a standard size that was designed to accommodate up to five impellers. The original process requirements for this application dictated that only three impellers were needed. As a result, the balance piston was located toward the center of the rotor bearing span and a spacer device was used to position the balance piston seal stationary components near the center of the compressor casing.

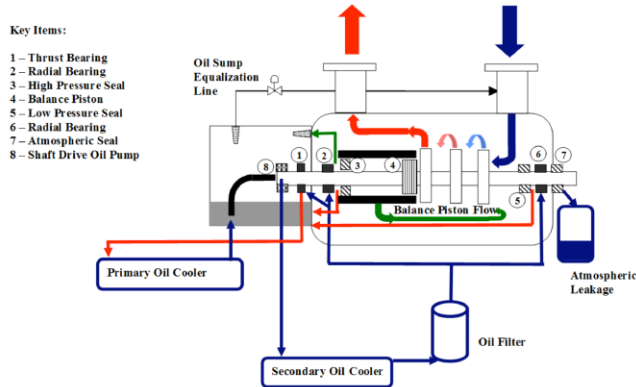


Figure 1. Oil and Refrigerant Circulation Path Diagram.

A general process flow diagram is drawn in Figure 2. The refrigerant is cooled with liquid that is recycled from the condenser, and blended in the receiver vessel with gas evaporated from a reboiler. The reboiler provides chilled water to the process and produces vapor to feed the compressor. A knock out drum is installed directly upstream of the compressor to prevent liquid from entering the compressor suction.

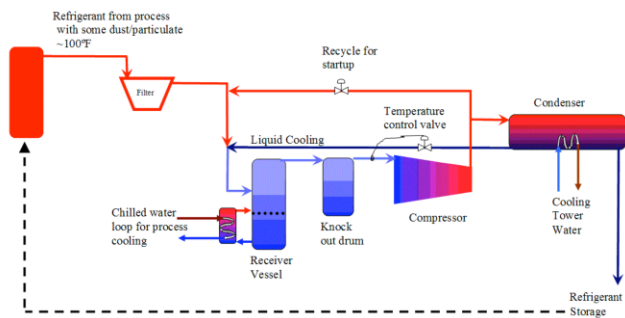


Figure 2. Refrigerant Process Flow Diagram.

The compressor discharge is sent to a condenser with the condensing pressure controlled by the cooling duty on the condenser. Liquid is continuously recycled from the condenser to cool the inlet process stream, which flows into the supply line upstream of the receiver. The original process design conditions for the compressor were:

- Suction pressure: 54 psia
- Suction temperature: 13°F
- Discharge pressure: 268 psia
- Discharge temperature: 180°F
- Suction flow rate: 2332 ACFM

Frequent overhauls of the compressor were required due to either process plugging of the compressor or inlet piping or because of excessive balance piston pressure. In situations where the compressor or inlet piping was partially plugged from powder fouling, the compressor had to be shut down since the loss of production from restricted flow was unacceptable. When this occurred, the compressor internals were observed to have a thick layer of buildup on the impellers, inlet guide vanes, and other internal passages. The high levels of fouling were also accompanied by severe damage to the balance piston and process seals due to excessive rotor response from imbalance. In some instances, the compressor flow rate was adequate for process demands but the mechanical condition required repair.

The excessive forced response sensitivity of the rotor was consistently found to severely damage all the seals in the compressor. In some cases, the severe damage was observed in less than several days of operation. Although the seals were found damaged, radial bearings were normally not damaged. Active thrust bearing shoes were often found to have some brown areas, suggesting that the thrust bearing temperature was high. The common indication for shutdown was high thrust bearing drain temperature.

In addition to severe damage of radial seals in the compressor, oil foaming was also present during normal operation. This problem had not been associated with bearing damage, but did result in some compressor trips due to low oil pressure, presumably from gas or foam entrained in the oil.

The initial scope of work was to include a rotordynamic analysis to reduce the rotor vibration to known fouling deposits and an analysis to improve the oil system to prevent foaming during normal operation. One obvious solution was to eliminate the powder from the process gas stream; that solution was addressed by a different team on a parallel path. In addition, it was accepted from the initial scope definition that some fouling due to powder would be expected even after improvement of filtration hardware, making improvement of forced response a logical solution to extend run times.

TYPICAL FAILURE

In events where the compressor was shut down due to mechanical damage as opposed to loss of flow due to plugging, the symptoms observed included the following:

- High thrust bearing drain temperature ($>165^{\circ}\text{F}$)
- High balance piston chamber pressure (>90 psig with suction pressure of 42 psig and discharge pressure of 210 psig)
- Low oil pressure shutdown switch trip
- Excessive foaming in the oil sump
- Significant damage to internal seals and particularly the balance piston seal

After review of components from failed compressors, it became obvious that most if not all of these symptoms were interrelated.

During operation (and often shortly after startup), the balance piston chamber pressure would begin to increase above suction pressure. A newly rebuilt compressor will normally have a balance piston line pressure of about 1 to 5 psi higher than the suction pressure on the compressor. However, after a short time of operation, the balance piston pressure would increase rather quickly to about 70 to 80 psig, then gradually continue to climb.

Higher thrust bearing drain temperature was caused by the increase in balance piston chamber pressure and the higher thrust loading that resulted from high balance piston differential pressure. This was also verified by the dark color of the thrust pads.

The cause of occasional low oil pressure was not immediately obvious, since radial bearings were seldom found to have any evidence of lack of lubrication. However, increases in the intensity of foaming with higher balance piston pressure may have resulted in foaming oil being pumped through the system.

OBSERVATIONS DURING STARTUP AND OPERATION

A typical startup was observed to help identify if there was the possibility of compressor surging or other startup events that could result in premature seal failure. The compressor was started as expected with the bypass valve open and a high level of recycle. The bypass was slowly closed and the compressor was operated on partial recycle until process gas was available.

During the startup process, there was continual evidence of compressor surging as suggested by a pulsing noise at about 2 Hz. The motion was visually observed on the casing with amplitudes at times reaching about 30 mils peak-to-peak.

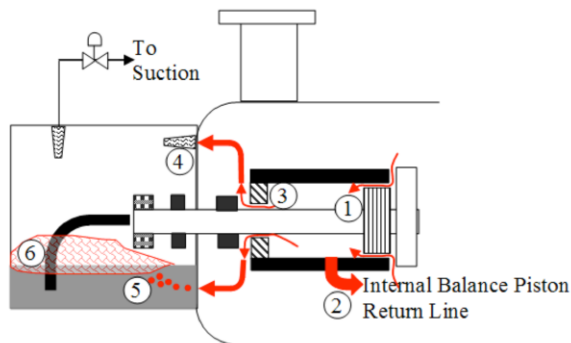
In addition, foaming of the oil sump was moderately excessive during the starting process, with the intensity of foam governed by how quickly that the sump pressure was reduced toward suction pressure. This type of compressor has an oil sump that starts with the oil pressurized to the stagnation pressure of the oil and refrigerant mixture. As the sump pressure drops toward suction pressure after startup, the refrigerant that is saturated in the oil at higher pressure will flash off causing the foaming.

The compressor oil system had been previously modified by others by adding two additional oil coolers in an attempt to reduce foaming. Product literature on the compressor operation indicates that foaming during startup can be reduced by heating the oil to a higher temperature, but foaming during normal operation can be reduced in general by operating at cooler temperatures. The addition of the second pair of oil coolers provided more oil in the system with saturated refrigerant that has to flash off during a start, making control of the rate of change of the oil sump pressure a key parameter to prevent the compressor from tripping on low oil pressure.

Several oils had been used in the system in an attempt to find some relief to foaming. The oils used included several viscosities of mineral-based oil designed for refrigerant service as well as synthetic oils. Replacement of the mineral oil with the synthetic did require cleaning and flushing. After the synthetic was installed, there was no decrease in the amount of foaming.

REVIEW OF THE OIL SYSTEM

The oil system on a pressurized oil sump refrigeration compressor of this type is shown in Figure 3. A shaft driven oil pump located inside the sump provides pressurized oil to the added secondary cooler and oil filters. The filtered oil then flows to the bearings on both ends of the compressor.



Key Points:

- 1 – Excessive Balance Piston Leakage
- 2 – Restrictive Balance Piston Return Line
- 3 – Excessive Process Seal Leakage Into Oil Sump
- 4 – Plugged Screen
- 5 – Hot Gas Blowing Through Oil Return Port
- 6 – Foam Developed In Oil Sump

Figure 3. Cause of Oil Foaming Problem.

Oil return from the drive end bearing is fed back to the sump through an external return line. Oil from the outboard end bearing flows directly back into the oil sump. The thrust bearing oil drain line (~ 80 percent of total oil flow) drains from the compressor casing into a thrust drain oil cooler. The cooler outlet is fed back to the sump and into the oil pump with some makeup oil drawn from the sump.

Oil that feeds to the radial bearings on both ends is returned to the oil sump by drain connections between the bearing and the internal process seal adjacent to each bearing. Since the outboard end bearing is close to the oil sump, there is a gravity feed port that flows from between the bearing and the internal process seal.

Flow of the gas back to the suction of the compressor is controlled using a local valve that is connected to the compressor suction.

Oil foaming on this type of compressor will occur in two different operating modes: during startup (or suction pressure variations) and during normal operation. Oil foaming would then be addressed for each case separately.

For the startup case, the problem is the level of saturation of refrigerant into the oil at stagnation conditions. Most refrigeration compressors will have a stagnation pressure about half way between the normal suction and discharge pressure. This stagnation pressure also pressurizes the oil sump, allowing refrigerant to dissolve into the oil.

When the compressor is started, the oil sump pressure will decrease from stagnation pressure to near suction pressure since the sump is vented to the compressor suction. Things that should be done to minimize foaming during the startup include:

- Heating the oil sump to flash off as much refrigerant as possible at stagnation pressure.
- Limiting the amount of oil volume in the system so the total mass of refrigerant in the oil is lower.
 - Increasing the volume of oil in the system also increases the total amount of refrigerant in the oil (not percent by weight, but total mass of refrigerant) that must be flashed out in the oil sump over the same surface area.
- Controlling the rate of decrease in the oil sump pressure using the manual valve that vents back to the suction.
 - This allows the sump pressure to decrease in a more controlled fashion and results in less aggressive foaming over a longer period of time following the start until the oil/refrigerant ratio becomes stable at normal operating conditions.
- In some cases, it may be necessary to replace the oil in the sump with new oil and start the machine as quickly as possible before the refrigerant has an opportunity to saturate into the oil again.

The characteristic of the refrigerant dissolving into the oil can be described using Henry's law. This law indicates that, for a constant temperature, the concentration of refrigerant saturated into the oil is dependant on the pressure of the solution. Henry's law is as follows:

$$e^p = e^{kc} \quad (1)$$

where:

- p = Partial pressure of the solute
- k = Henry's law constant
- c = Concentration of the solute

This formula can be reduced by taking the natural logarithm of each side and rearranging terms to produce:

$$c = p / k \quad (2)$$

This indicates that the concentration of refrigerant is approximately proportional to the pressure of the solution. What this means is that, as the pressure is dropped for a saturated solution, gas must be liberated to produce a stable refrigerant/oil mixture at the lower pressure.

Although these relationships appear easy to apply, the actual Henry's law constant (k) is very difficult to determine for a given solution. The interaction of the saturated gas and liquid should, in general, follow these relationships, but the actual saturation concentration cannot be established without extensive laboratory testing with the specific compounds. Therefore, the use of Henry's law concepts is only presented to aid in understanding the relative response of the solution to variations in pressure and temperature.

STARTUP RESPONSE OF OIL SYSTEM

During the startup condition, the oil will be saturated with refrigerant at the stagnation pressure. As the pressure of the oil sump decreases toward suction pressure after startup, the excess refrigerant in the oil must flash off. For this application, the difference in dissolved concentrations between stagnation and operating is about a ratio of almost 2:1. Therefore, a lot of refrigerant must flash off prior to the concentration stabilizing at the actual suction pressure.

To reduce the extent of flashing, the oil in the sump is normally heated to a higher temperature to help reduce flashing prior to startup. The characteristic of the solution of refrigerant in oil that supports this practice is the variation of the Henry's law constant with temperature. As the temperature is increased, the constant changes as follows:

$$k(T) = k(T_{ref}) \times e^{[-C \times (1/T - 1/T_{ref})]} \quad (3)$$

where:

- C = A constant for a specific gas
- T = Absolute temperature of the gas
- T_{ref} = Reference temperature for Henry's law constant (normally 298°K)

This characteristic will result in lower levels of solubility of the gas in the oil at higher temperature. To accomplish this, there is an oil sump heater installed to heat the oil in the sump prior to starting. To prevent the oil from cooling and allowing higher concentration building in the oil system, the auxiliary oil pump is normally not run except right before starting the compressor. This allows the primary oil volume in the oil sump to "degas" prior to starting.

Since there was controllable foaming during the startup process by limiting the rate of pressure drop in the oil sump, only moderate foaming was observed in this condition. However, if the pressure was lowered too quickly, aggressive foaming would occur and oil pressure would occasionally drop below the minimum set point and trip the compressor.

NORMAL OPERATION OF OIL SYSTEM

Once the compressor is running and stable, lower oil sump temperature will allow higher concentrations of refrigerant to remain in the oil than is in the sump. This property is referred to as the air release property concerning air entrainment in hydraulic oils. Lower solution temperature will reduce the rate of bubble formation and collapse where higher temperature promotes bubble formation and quick release of the solute. This characteristic would result in decreased foaming with lower sump temperature during normal operation, but improved gas liberation when not operating for elevated sump temperature.

One additional characteristic of this system that will tend to make it more susceptible to the generation of foam is the presence of small dust particles in the process gas that migrate into the oil system. The presence of particulate in the oil will increase the potential for the development of foam as the gas is released.

Oil circulation in the system is limited to primarily circulating the same oil with limited makeup taken from the sump due to the oil feed system design. Most of the oil that is pumped through the system is returned through the thrust bearing drain line and is fed directly back to the oil pump after going through an oil cooler. The amount of makeup oil from the sump may be about 10 to 20

percent of the total flow through the oil pump. This design characteristic allows the oil that returns to the sump from the bearings that will have higher concentrations of refrigerant to have a higher residence time in the sump, promoting better gas liberation.

Oil that returns from the area between the bearing and process seals will have the highest amount of refrigerant in solution since this oil is in direct contact with the refrigerant. To improve the gas/oil separation on the oil that returns from the drive end of the compressor, the oil is returned to a section of the sump that has a weir and an additional heating element. This slight heating of this return stream will dramatically help degas this stream prior to returning it to the main oil sump.

The oil that returns from the nondrive end bearing between the bearing and the process seal will be the primary source of oil foaming since this stream is returned in a saturated mixture directly back to the oil sump. In reality, the gas/oil mixture includes two paths as shown in Figure 3. The two paths include a saturated oil return directly back to the sump by gravity flow as well as a gas refrigerant path back to the top of the oil sump through a coalescing element.

Once the oil is pressurized by the oil pump, the solution will not flash unless the oil pressure is decreased below the partial pressure for the gas at saturation, or if the temperature rises resulting in lower saturation concentrations. For this application, the minimum rise in oil pressure of about 60 psid will overcome the rise in temperature from 120°F up to a minimum temperature of about 177°F and prevent flashing of the refrigerant in the oil supply and in the bearing passages. Since the oil drain temperature is normally near 150 to 165°F, flashing of oil in the bearing supply passages and return lines is not expected.

CAUSE OF FOAMING DURING OPERATION

Once the compressor was operating, there were varying degrees of foaming that existed at all times. However, as the balance piston chamber pressure would increase, the level of foaming would always increase as well. If the balance piston seal is damaged and excessive flow passes into the balance piston chamber, this area begins to pressurize as the flow increases. This machine was provided with several internal balance piston return tubes that were also found to foul and plug after short-term operation. When that would happen, the flow across the internal process seal would increase even more forcing balance piston seal leakage past the internal process seal and into the oil reservoir on the nondrive end of the compressor.

The design of the oil system flow causes the gas flow past the process seal on the outboard end to flow back into the oil sump, then through the vent return line that attaches to the top of the oil sump. This connection into the sump was provided with a coalescing screen. The purpose of the screen is believed to be to prevent oil foam during startup from migrating backward to the passage between the process seal and the outboard bearing.

The resulting gas flow across the process seal would then flow through the oil sump and back to compressor suction through the oil sump equalizing line. After detail investigation of the oil system, it was discovered that the coalescing screen on the seal return line was plugged. When this occurred, the gas pressure generated by the elevated flow would only have to overcome the static pressure in the oil sump before gas would vent back to the sump through the lower port where oil normally drains back into the sump.

The oil foaming during normal operation was reduced by replacing the screen on the gas return line back to the oil sump with a new screen. The original screen included an external screen with some packing material that resembled steel wool. After the new replacement screen plugged reasonably quickly due to the particulate in the gas stream, a screen with larger mesh and no packing was installed. This modification reduced the foaming to acceptable levels.

VIBRATION DATA

The compressor was originally purchased with proximity probes installed at each bearing. These were removed at some point. No permanent vibration monitoring equipment was installed. Temporary accelerometers were mounted on the casing to observe the casing velocity during a typical startup.

The casing to rotor weight ratio was large with a rotor weight of about 100 pounds and a casing weight over 4000 pounds. With such a heavy casing, large amplitude casing vibration was not expected. The vibration immediately after a startup following an overhaul with the compressor operating on full recycle is shown in Figure 4. The overall vibration was about 0.05 in/sec peak, with a 1× rpm amplitude of about 0.027 in/sec peak. Two× and 3× rpm harmonics were also present.

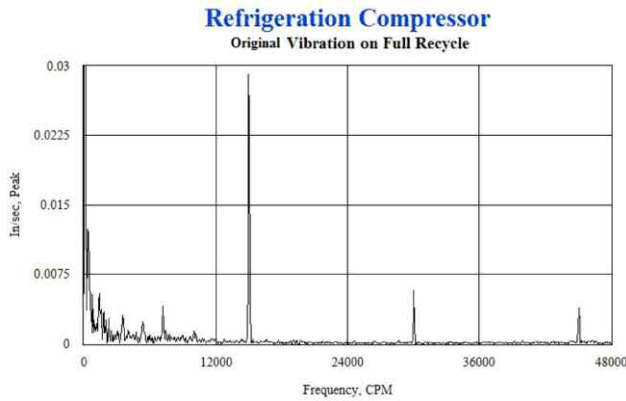


Figure 4. Measured Case Vibration on Compressor Before Modifications.

Casing amplitudes of 1× rpm vibration in the heavily fouled condition were observed to be as high as 0.2 in/sec peak. The vibration data also indicated there were high vibration amplitudes near 2 Hz caused by surging.

ROTOR DYNAMICS ANALYSIS OF COMPRESSOR

The history of the compressor suggested that it was very sensitive to imbalance. Not all of the changes made in the past were fully documented, but it appeared that at least one bearing design change had been previously implemented. No other analysis models were available so the spare rotor was carefully measured and all parts weighed. The compressor rotor, Figure 5, is driven at 15,200 rpm by an induction motor through a speed increaser. The flexible coupling between the pinion and the compressor is a hybrid design with one flexing element and a long quill shaft. Thus, to fully define the compressor dynamics a combined model of the pinion, coupling and compressor was constructed as seen in Figure 6. A closeup view of the flexible diaphragm portion of the coupling can be seen in Figure 7.



Figure 5. Photograph of Unmodified Compressor Rotor. (Courtesy of Vibration Institute)

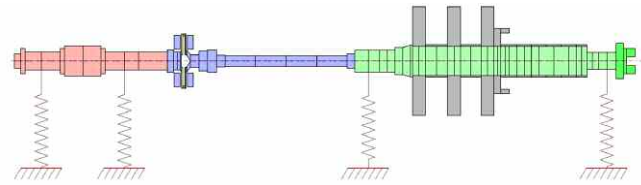


Figure 6. Finite Element Model of Pinion, Coupling, and Compressor Rotor.

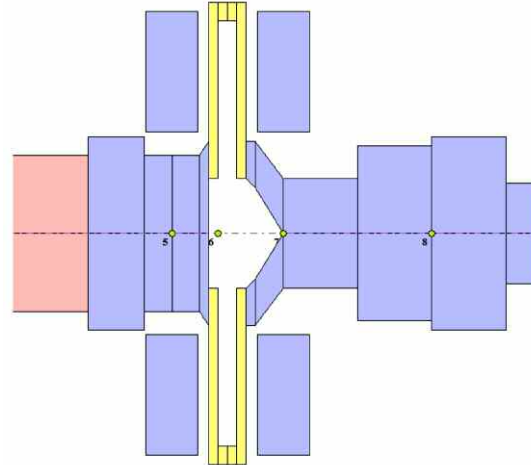


Figure 7. Enlarged View of Flexible Coupling.

The coupling ends are rigidly connected to both the pinion and compressor shafts. The flexible diaphragm section of the coupling accommodates misalignment along with the inherent flexibility of the long thin quill shaft. In the finite element model, the diaphragm sections have their elastic and shear moduli adjusted to reflect the actual lateral and torsional stiffnesses.

The existing compressor journal bearings were pressure dam configurations. Table 1 lists the geometry of each bearing. The design of the pinion bearings was unknown. However, the dynamics of the pinion had virtually no effect on the compressor so the pinion bearings were assumed to have a constant stiffness of 3 million lb/in.

Table 1. Existing Pressure Dam Bearing Dimensions.

BEARING	DIAMETER	LENGTH	DIAMETRICAL CLEARANCE	DAM LOCATION	DAM WIDTH	DAM DEPTH
Coupling End	2.5 Inches	2.375 Inches	4 to 5 Mils	135 Degrees	1.75 Inches	7 to 8 Mils
Outboard End	2.0 Inches	1.00 Inches	3 to 4 Mils	135 Degrees	0.72 Inches	6 to 7 Mils

At operating speed, Table 2 lists the dynamic characteristics of the original bearings. Table 3 summarizes the effective dynamic stiffness of these bearings at 15,200 rpm. The dynamic stiffness is $[K^2 + (C\omega)^2]^{-.5}$ and is useful for comparison later.

Table 2. Existing Pressure Dam Bearing Characteristics.

BEARING	Kxx LB/IN	Kyy LB/IN	Cxx LB-SEC/IN	Cyy LB-SEC/IN
Coupling End	1.46 X 10 ⁶	1.58 X 10 ⁶	1,879	4,437
Outboard End	6.3 X 10 ⁵	6.6 X 10 ⁵	802	1,304

Table 3. Existing Pressure Dam Bearing Dynamic Stiffnesses.

BEARING	Kdx LB/IN	Kdy LB/IN
Coupling End	3.3 X 10 ⁶	7.1 X 10 ⁶
Outboard End	1.4 X 10 ⁶	2.2 X 10 ⁶

An undamped critical speed analysis indicated that there could be a direct interference of operating speed and the first critical speed. The undamped critical speed map, Figure 8, shows that the bearings are very stiff compared to the rotor in this application since they intersect the first critical speed curve in its asymptotic section.

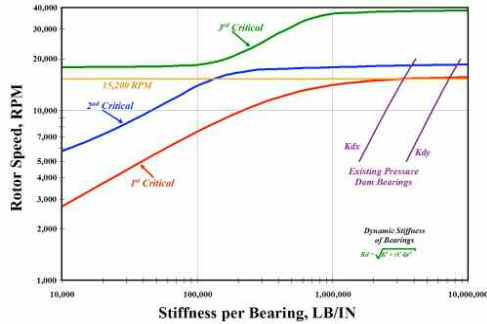


Figure 8. Undamped Critical Speed Map of Unmodified Rotor and Original Bearings.

The undamped first critical speed mode shape, shown in Figure 9, indicates that the quill shaft portion of the coupling is a strong influence. The pinion is not significantly involved in the dynamics as it is isolated by the flexible membrane. It is important to note that the maximum deflection at this resonance is very close to the balance piston location in the compressor.

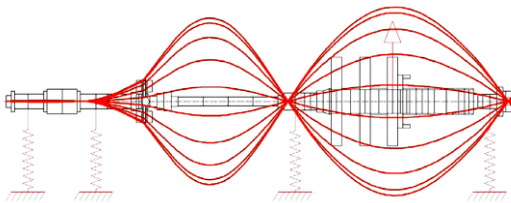


Figure 9. Undamped Critical Speed Mode Shape of Unmodified Compressor Train.

Figure 10 is the damped unbalance response synchronous amplitude and phase predicted for the center of the compressor rotor for a 4W/N imbalance placed at that same location. This confirmed that the compressor was operating directly on resonance in its current condition.

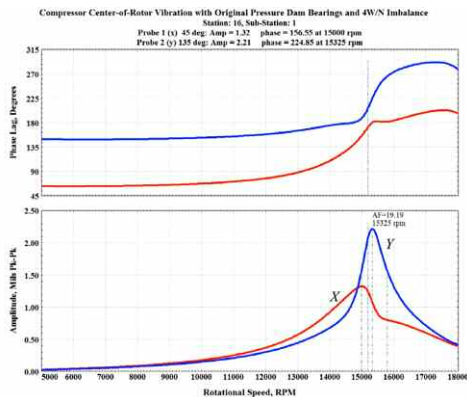


Figure 10. Predicted Unbalance Response at Compressor Rotor Center with Original Bearings.

With the direct interference and the high amplification factor, it became evident that the bearings needed to be softened. Many different bearing types were considered including a squeeze film damper arrangement. Only thin liner types of bearings were applicable due to space limitations. After many iterations, the best bearing was found to be a three-lobe design as shown in Figure 11.

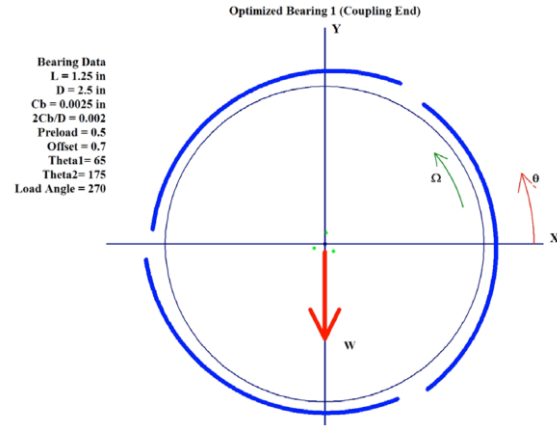


Figure 11. Optimized Three-Lobe Bearing for Refrigeration Compressor.

Based on an optimization routine, the coupling end bearing was axially shortened to produce the lowest stiffness and most effective damping. In addition, clearance, preload, offset, and lobe orientation were all varied to optimize the bearing design. Tables 4 and 5 list the final three-lobe bearing characteristics and dynamic stiffnesses. The calculated dynamic stiffnesses are much softer than the original bearing dynamic stiffnesses, particularly on the coupling end where the difference exceeds 5-to-1. While the drop in calculated damping may seem to be detrimental, the dramatic stiffness decrease actually allows the available bearing damping to be much more effective in attenuating critical speed vibration amplitude.

Table 4. Three-Lobe Bearing Characteristics.

BEARING	Kxx LB/IN	Kyy LB/IN	Cxx LB-SEC/IN	Cyy LB-SEC/IN
Coupling End	2.30 X 10 ⁵	2.34 X 10 ⁵	327	336
Outboard End	1.9 X 10 ⁵	1.9 X 10 ⁵	263	268

Table 5. Three-Lobe Bearing Dynamic Stiffnesses.

BEARING	Kdx LB/IN	Kdy LB/IN
Coupling End	5.63 X 10 ⁵	5.77 X 10 ⁵
Outboard End	4.51 X 10 ⁵	4.62 X 10 ⁵

With the drop in bearing stiffnesses it was hoped that this would solve the compressor's problem with critical speed interference. The undamped critical speed map, Figure 12, gave hope that this would happen. From this, the operating speed would appear to lie safely between the first and second critical speeds.

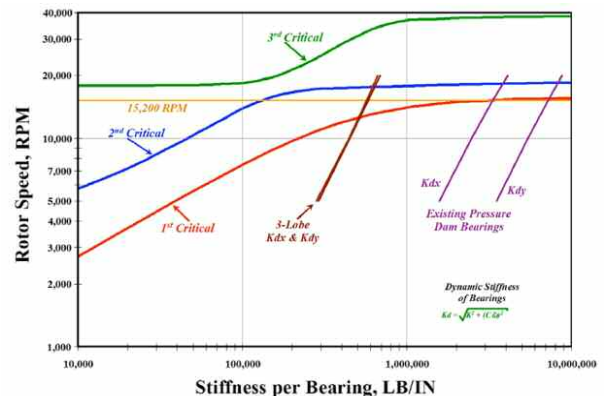


Figure 12. Undamped Critical Speed Map of Unmodified Rotor and Three-Lobe Bearings.

When the damped unbalance response was calculated, it was surprising to find that the resonance was still very near operating speed. The maximum center span amplitude had dropped from 2.2 mils to 0.9 mils and the amplification factor had dropped from 19.9 to 4.6 but the critical speed interference remained, as seen in Figure 13. The reason for this is that the more active bearing damping may be increasing the dynamic stiffness more effectively than before.

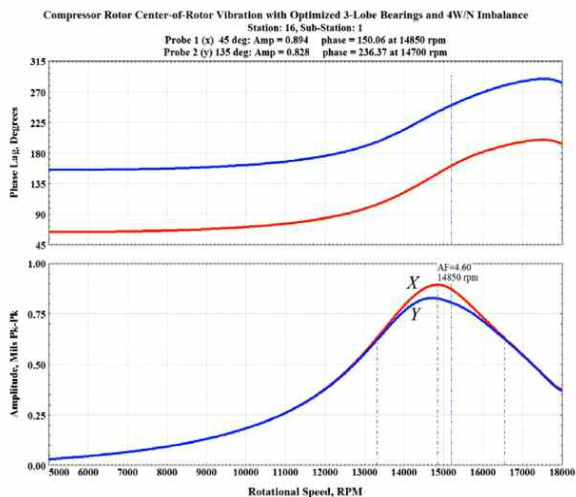


Figure 13. Predicted Unbalance Response at Compressor Rotor Center with Three-Lobe Bearings Only.

At this point rotor modifications were considered. Both shaft diameter changes and bearing span changes were not possible due to space limitations. The idea of adding mass to the rotor became the focus of the study. One idea was to make new wheels out of steel instead of aluminum but this was an expensive and long delivery route. Since there was axial room on the rotor, a dummy wheel was added next to the balance piston location as indicated in the new rotordynamics model, Figure 14. A photograph of the modified rotor, Figure 15, also features coated impellers. In the picture, a 12-inch scale rests on the impellers to help visualize the size of the rotor. This new disk added 34 pounds to the rotor's original 102 pounds or a 33 percent increase near the middle of the rotor. This was determined to be very effective in lowering the first critical speed frequency.

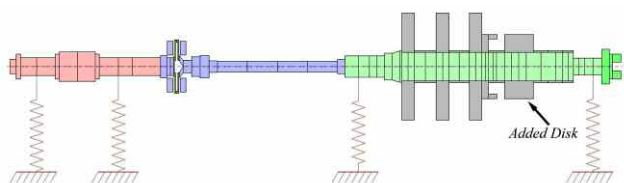


Figure 14. Finite Element Model of Compressor Train with Added Disk.



Figure 15. Photograph of Modified Compressor Rotor.

With the new added disk the undamped critical speed map, Figure 16, showed that the first critical speed frequency had been significantly reduced so that even the original bearings would probably not have had a direct interference even if they were still too stiff.

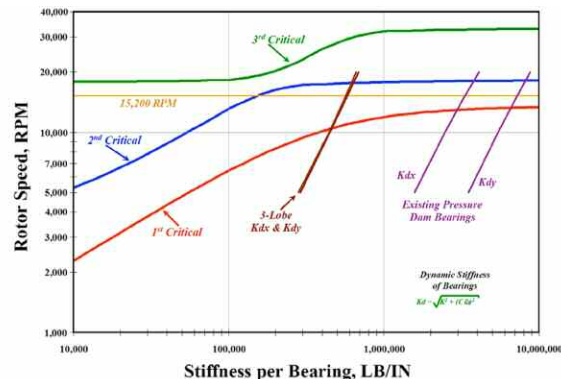


Figure 16. Undamped Critical Speed Map of Modified Rotor and Both Bearing Types.

The calculated first critical speed mode shape with the additional disk and the optimized bearings is plotted in Figure 17. While the amplitude is still a maximum in the rotor center, the contribution of the quill shaft to this mode has been reduced. In addition the relative amplitude at the bearings has been increased. This allows the bearing damping to be more effective than when there are nodal points near the bearing centers.

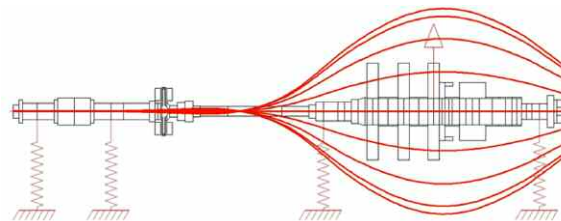


Figure 17. Undamped Critical Speed Mode Shape of Modified Compressor Train.

The redesigned rotor and bearing combination unbalance response, Figure 18, is predicting a very well controlled rotor. The maximum predicted amplitude is less than 12 percent of the original and the amplification factor is less than 1.0 with an operating speed amplitude of less than 0.2 mils peak-to-peak for a 4W/N imbalance. It should be noted that with the increased rotor weight, the applied 4W/N imbalance also increased and the bearing coefficients were recalculated for the new journal loads.

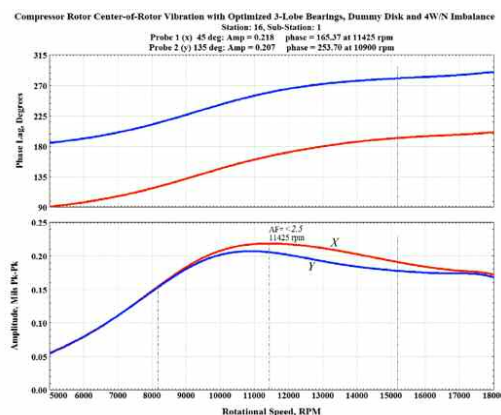


Figure 18. Predicted Unbalance Response at Compressor Rotor Center with Three-Lobe Bearings and Added Disk.

FIELD TEST RESULTS

New three-lobe bearings were manufactured and the dummy wheel was built and mounted during a compressor rebuild. When this modified machine was operated, there was a 56 percent reduction in 1x casing vibration velocity. The casing velocity spectrum at the same point is compared in Figure 4, the original configuration, and Figure 19, which is after the rotor and bearing modifications.

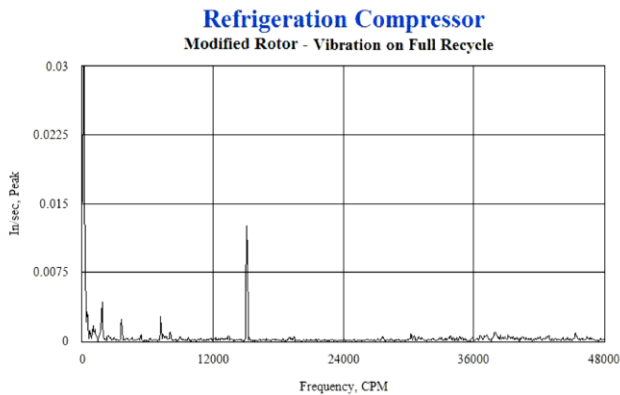


Figure 19. Measured Case Vibration on Compressor with Modified Rotor and Three-Lobe Bearings.

Subsequently, continuous surging of the machine along with material buildup on the impellers eventually caused the balance piston pressure to increase and caused a shutdown. The new three-lobe bearings were found to be damaged, just like the pressure dam bearings. To address this problem new three-lobe bearing liners were manufactured of unbabbitted bronze. This material is about 10 times stronger than Babbitt. This configuration was found to be able to run, without damage to the bearings or seals, for several months at a time until the impellers become choked with buildup. A process team is currently addressing this problem. The surging will continue until the rotor is modified or a recirculation system is implemented to increase the flow through the compressor. Figure 20 indicates the current compressor performance curves.

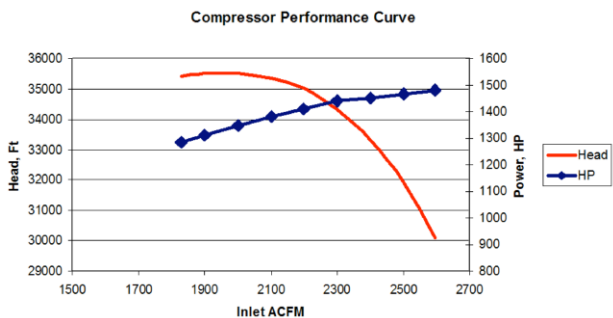


Figure 20. Compressor Performance Curve.

OBSERVED SURGING

Surging of the compressor was common during startup, which was the result of poor control timing between the recycle valve and the inlet guide vanes (IGVs). Based on the original design of the compressor controls, the IGVs and the recycle valve were operated from the same control signal. However, the timing of the recycle valve closure and the opening of IGVs was not properly managed so that in some cases the compressor would surge violently.

Once the compressor was started and operating with some flow from the process, the recycle valve would close while the IGVs would begin to control suction pressure. There was no dependable flow measurement in the system, and no dedicated surge control system.

After the compressor was well loaded, field testing indicated it was continuously surging. This was identified by reviewing the casing vibration data as well as visual observations during operation. The typical casing vibration response with and without surge is shown in Figures 21 and 22.

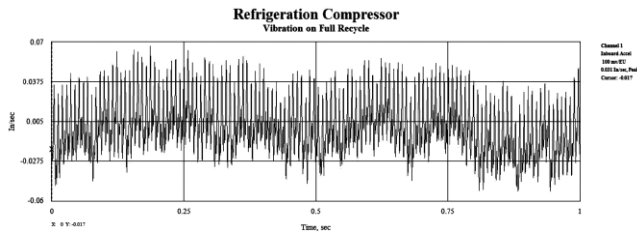


Figure 21. Compressor Vibration on Full Recycle.

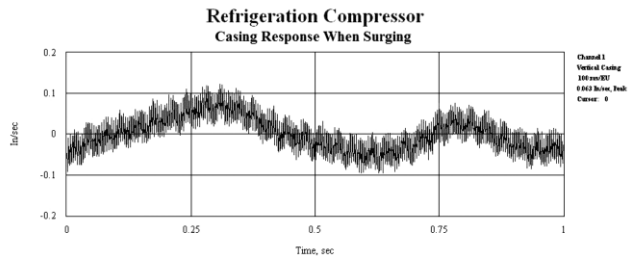


Figure 22. Compressor Vibration During Surging.

Figure 21 shows a typical casing vibration response time waveform when the compressor was on full recycle. A total peak amplitude of less than 0.1 in/sec was measured. Once the recycle valve was closed, Figure 22, which is normal operation, the casing response would shift with significant vibration near 2 Hz. At times, the casing response near 2 Hz was observed to reach as high as 0.4 in/sec peak.

SURGE REVIEW

After the surging response was identified by field measurements, review of the process was done to investigate the cause of the apparent compressor surging. Available instrumentation in the plant was limited so that it was very difficult to determine the actual flow rate into the compressor. Following extensive review of the plant operation, the normal flow was identified to be well below the flow required to prevent surge. In normal operation the IGVs would be operated between 50 percent and 100 percent open.

The typical performance curve with the IGVs fully open is shown in Figure 20. The minimum flow to prevent surge is slightly over 1800 ACFM with the guide vanes fully open. Once a dependable flow rate could be determined from the process, the actual flow compared to the published performance curve is plotted in Figure 23. Thus, the actual flow is less than about 50 percent of the minimum flow to prevent surge.

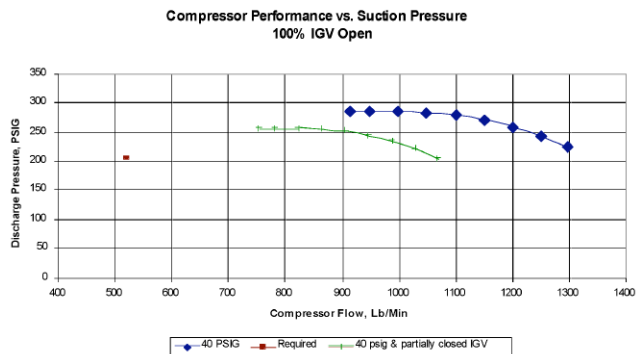


Figure 23. Compressor Performance Versus Suction Pressure and Operating Point.

This compressor does have inlet guide vanes upstream of the first impeller, but data were not readily available to characterize the performance at various guide vane settings. Other problems with predicting performance was the lack of dependable indication of IGV position. However, common operation was between 75 percent and 100 percent open.

Further investigation identified that the process had been dramatically changed from the original plant design. The primary variation was that the original design included a chilled water system that resulted in a high load on this compressor as shown in Figure 2. Modifications to the plant have resulted in almost eliminating the chilled water demand, and essentially removing 50 percent of the design gas flow through the compressor. This change resulted in low flow rates and the compressor operating in a continuous heavy surge.

ADDITIONAL REVIEW OF MACHINE DAMAGE

Once the surging was identified, additional review of the failure data led to the conclusion that most damage to the machine was most likely due to the continuous surging. The observations included:

- Very low vibration and no evidence of surging when in full recycle on a new machine (normal balance piston pressure, etc.).
- Evidence of quick damage to internal seals indicated by balance piston pressure increasing sometimes within a day of overhaul and certainly within a few days once on normal process duty.
- Extreme damage to the balance piston seal, which is near the center of the bearing span (typically almost 1/8 inch wiped off seal teeth!!).
- Significant damage to impeller eye seals on every overhaul.

These symptoms could be a result of high 1× rpm vibration due to the identified critical speed near compressor speed, or the result of surging. Since the damage described did not significantly change after the rotordynamics were changed to shift the rotor critical, surging is the most likely cause of the high damage.

SURGE TOLERANT DESIGN

In an effort to improve the machine's tolerance to surging until a proper surge control or surge prevention strategy could be employed, some changes were made to the compressor including:

- Bronze liner bearings instead of Babbitt (to improve fatigue resistance).
- Detuning the rotor critical speed away from operating speed.
- Nonmetallic seals for the balance piston and impeller eye seals.
- Adding offset pivot type thrust shoes to increase the thrust capability.

These changes have produced a compressor that has demonstrated the ability to operate for months without failure.

CONCLUSIONS

The original scope of this project was to produce a compressor that was much more tolerant to imbalance based on the initial presumption that the contaminant fouling was causing excessive vibration and failing the seals. The secondary problem of oil fouling was expected to be a separate issue.

After the rotor and bearing modifications were implemented to reduce the sensitivity to imbalance, similar failures were observed. Coating of the impellers and inlet guide vanes reduced the rate of fouling of the impellers. Additional investigation identified compressor surging as a key contributor to machine damage. Modification of the aluminum seals to thermoplastic seals allowed longer term operation with continuous surging.

The oil system was found to be adequate for the application. The primary cause of foaming was identified to be an inexpensive screen inside the oil sump that was plugged and forced gas refrigerant to vent into the sump below the liquid level. After this screen was replaced, fouling was eliminated.

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