

RECYCLE GAS COMPRESSOR DESIGNED FOR HIGH UNBALANCE TOLERANCE AND STABILITY

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ABSTRACT

A process recycle gas compressor was needed that would be highly reliable and tolerant of suction contaminants of sticky material that is abrasive and corrosive. Squeeze film dampers, pinched diffusers, and other technologies created a robust design that proved highly tolerant of extreme process upsets as well as normal demands. Impeller tip speed limits help control erosion and an open-faced, radial vane impeller minimizes fouling. Custom five-axis milled Inconel® 625 impellers with splitter vanes and pinched diffusers allowed adaptation of a standard compressor frame to meet thermodynamic needs without excess flow. Squeeze film damper bearings with flexible pivots yielded a rotorbearing system highly tolerant of unbalance with excellent cross-couple stability. The compressor was test run four hours with a shaking force two and one half times rotor weight while running below vibration and bearing temperature interlocks. Other features were also provided to improve reliability. During two extreme process upsets, the impeller blade tips were almost completely eroded away in just 10 days, but operation was able to continue until a scheduled outage.

INTRODUCTION

Compressor suction contaminants can normally be controlled with a variety of separation technologies to obtain reliable compressor operation in most Dupont processes. Unfortunately, this process stream is corrosive and routinely contains sticky, abrasive solids that have proven exceedingly difficult to confine to either the solid or liquid state. Thus, a process recycle gas compressor was needed that would be highly reliable while being tolerant of these suction contaminants.

The process pressure requirements could be met with a single compression stage and offered several advantages. Frequent cleaning of the compressor would be required and an axially split casing could be cleaned more quickly than the horizontally split casings in similar lines and would also be easier to seal. Using a single stage would also reduce the installed cost and the cost of periodic rebuilds for this difficult service. API standards were used to guide decisions during the specification and design process, but there are many major departures from API as well.

CONCEPT OF COMPRESSOR

A single-stage overhung compressor is driven by an induction motor through double helical gears and mounted on a base frame via a bearing pedestal (Figure 1, motor not shown). The oil system is integrated on the base frame (Figure 2, serves also as oil

reservoir); local instrumentation is wired up to junction boxes at the edge of the skid. Both couplings (motor-to-gear and gear-to-compressor rotor) are full metal disc couplings. The compressor runs on constant speed and is flow controlled via a cooled bypass. The seal is a single chamber labyrinth that is purged by nitrogen. The gearbox and the oil system operate under a nitrogen blanket.

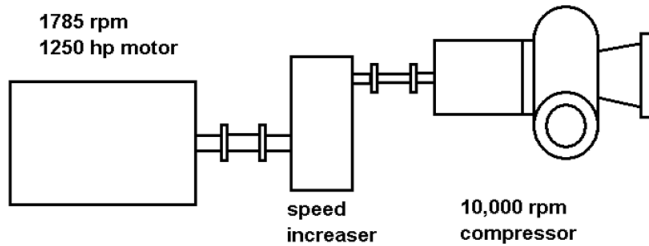


Figure 1. Compressor Configuration.

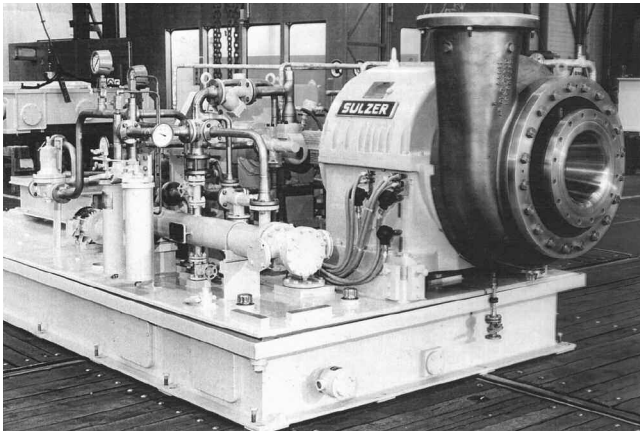


Figure 2. Compressor on Base Frame.

In order to meet the customer's requirements, a custom compressor design was conducted, adapting only a few existing components within the manufacturer's line of integral gear compressors. Figure 3 gives a cut view through the compressor with its bearing pedestal. The main challenges of this project were linked to the nature of the compressed medium as well as to the known extreme operating conditions linked to the process. A number of design features linked to the medium are noted below. The other rationale driving this design is linked to the rotordynamic behavior of the machine under all operating conditions.

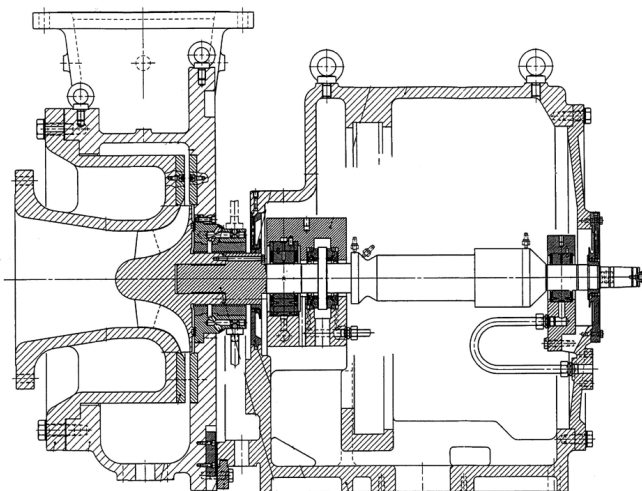


Figure 3. Longitudinal Cut Through Compressor.

Overhung compressors (i.e., compressor impellers are located outside the bearings) are widely used in various industries—API 672-type (1996) machines for air compressors have been used for many years. Recently, applications in the process industry can cite API 617 (2002) in its newest edition where these machines are addressed specifically. The basic design principles are well understood. Operation above certain critical speeds is common and usually leads to a very smooth running behavior. During startup and coastdown, the existence of these critical speeds is usually noticeable through the vibration monitoring equipment. The observed peak vibratory displacement (Figure 4) is linked to the general balance condition of the rotor and the so-called amplification factor. For well-balanced rotors, even a large amplification factor can be tolerated. However, this feature can lead to rather undesirable situations if the balance condition changes during normal operation—and changes are usually deteriorations. Even if the vibratory level at operating speed has increased but is not critical, it will mean almost certain destruction of vital machine components when the machine is shut down and has to pass through the amplification factor region while coasting down.

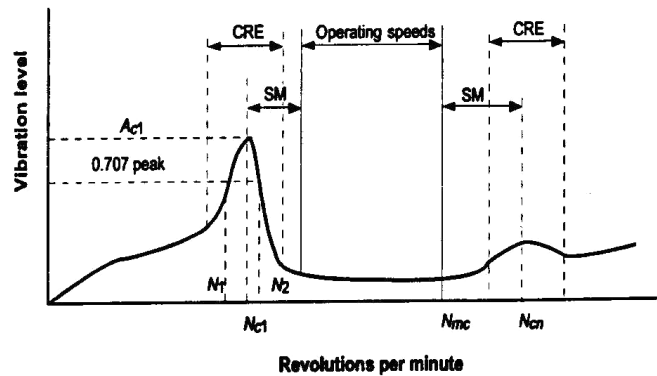


Figure 4. Vibration Amplitude Versus Speed. (Courtesy API 617, 2002)

A solution to this dilemma is obviously to design a machine that exhibits no noticeable amplification factor.

Thermodynamic Layout—Stage Design

Contrary to many other designs, this machine was not primarily optimized for aerodynamic performance but for reliability and robustness. The customer specifically advised radial discharge blades for the impeller, vaneless diffusers, and a volute designed for minimal dead spaces, which might lead to the accumulation of dust. Subsequently, an existing zero-backsweep impeller (Figure 5) was adapted by increasing the blade thickness to provide more material against the abrasive nature of the process. The effect on flow capacity due to the blade thickness increase was estimated and compensated for by an increase in blade height. This change was checked during the shop test—it turned out that the compensation was adequate, so no remachining of the impeller contour was found to be necessary.

The vaneless diffuser space was provided by two “consumable” plates (Figure 6), which could be easily adapted in case the capacity of the stage should have to be adjusted for (by applying more or less “pinch,” i.e., a reduced channel height downstream of the impeller exit) or replaced if found to be worn out due to erosion during operation.

The volute was developed to provide a simple flow collecting space without the usual emphasis on optimum aerodynamic performance. During outages, the unit is washed off-line by water injection regularly (Figure 7), therefore it was imperative to avoid any dead spaces that might lead to accumulation of dust and/or fluids with increased potential for chemical activity.

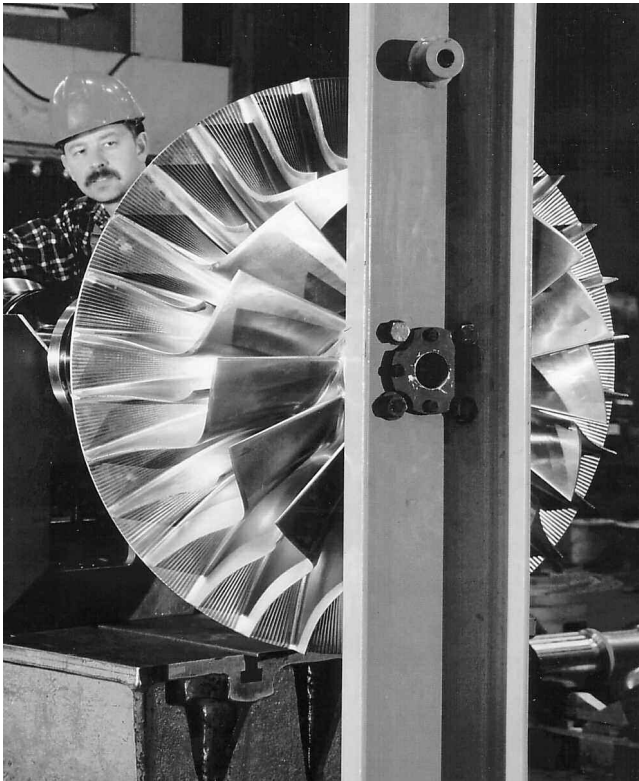


Figure 5. Radial Discharge Impeller.

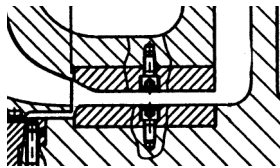


Figure 6. Vaneless Diffuser.

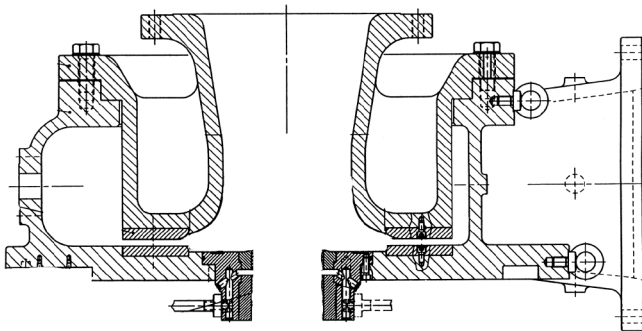


Figure 7. Volute and Seal Cartridge.

Mechanical Design Specifics

The impeller shaft attachment was specifically adapted to address the severe media requirements. Axial bolts and pins are not directly exposed to the process gas. Stress levels in the impeller are generally low due to the relatively low tip speed and the “bore”-less design. A balance groove on the backside of the impeller was provided to vary the balance condition during shop testing (to check the rotordynamic behavior under severe unbalance conditions). The volute is attached to the bearing pedestal via three radial keys to ensure concentricity between shaft centerline and the shroud geometry under all thermal conditions. Vibration monitoring as well as axial shaft position is arranged between the bearing

span. Limits for alarm and shutdown at these locations have to take the rather large distance to the impeller blades into account to prevent rubbing contact of the impeller with stationary parts (as there the most likely rubbing might occur).

Materials of Construction

As material of choice for the impeller, a nickel-based alloy (2.4621, NiCr22Mo9Nb) was recommended from customer experience. The volute was cast from AISI 316L (1.4404, G-X2CrNiMoN1810). Due to the main constituent of the process gas, the area of potential contact by rubbing between the open impeller and the shroud portion of the casing was covered by a cladding layer of the nickel-based alloy. This precaution was taken due to the risk of a rather violent reaction (“combustion”) above a relatively low “ignition” temperature for the process, which is well known, e.g., Bartos (1995). The seal cartridge was made from AISI 316L with a nickel-based alloy insert. The shaft was made from AISI 4140 with nickel plating in the seal area. The bearing pedestal is made of cast-iron; the base frame is fabricated from mild steel.

Fabrication

The impeller size permitted five-axis milling at reasonable cost for a one-of-a-kind situation—for higher production numbers the use of investment castings would seem feasible. The volute was cast and its integrity qualified by employing various nondestructive testing (NDT) techniques and subsequent weld repairs. The nickel-base alloy cladding region required additional care and a special welding technique to ensure an adequate bond between the two materials, free of hot cracks.

Rotordynamics and Bearings

As has been noted above, the suppression of the amplification factor is a necessary feature for this application. As with all resonance phenomena, damping is crucial to achieve this. While the hydrodynamic bearings provide a certain amount of damping, it is not sufficient for this application. Additional damping can be provided by squeeze film dampers (Zeidan, 1995). These dampers can be incorporated in the bearing body design. Together with springs that provide for centering of the bearing under the rotor weight (static displacement), the system’s rotordynamic characteristics can be tailored to achieve the desired shape (Figure 8). The remainder of the bearing system employed usual components: four-pad flexure pivot pad journal bearings for the radial bearings, eight-pad tilting pad axial bearings arranged around a thrust collar. The oil quality is specified for ISO VG 32. Figure 9 gives a cut through the bearing with integrated squeeze film damper; Figure 10 shows one half of the bearing. Note the spring elements connecting the outer and inner bearing body. The bearing is of the “closed” design (flood-lubricated with labyrinths on both sides limiting the oil release). Total containment of the squeeze film inside the labyrinths eliminates the need for piston rings or O-rings that often prove troublesome for reliable squeeze film applications. The impeller end bearing stiffness and damping characteristics are shown in Figures 11 and 12, and the resulting rotorbearing system critical speed map is shown in Figure 13.

Seal

The seal design had to address the media requirements, which require positive process separation from the environment as well as from the low alloy, steel components of the shaft. Due to the bearing concept, the chosen rotor system with relatively large static deflections leads to large clearances. To top this off, the seal must be operational during washing in standstill without contamination of the environment. Figure 14 indicates the chosen configuration: nitrogen is introduced as a purge into a single chamber. The stationary labyrinths are cut out of a nickel-base alloy insert. A jet connected to an external water line allows in situ washing of the backface of the impeller (during standstill). Drain lines both in the

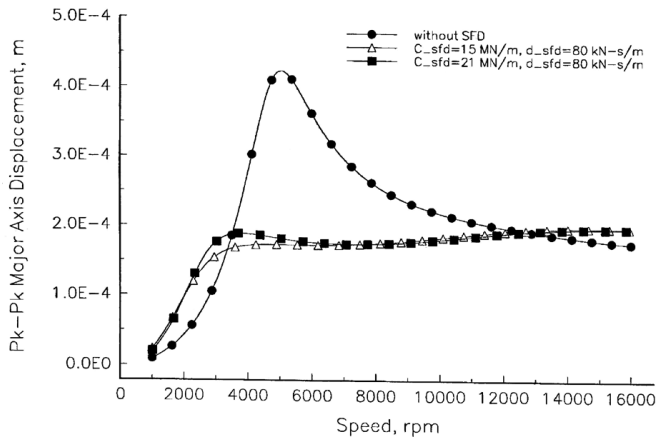


Figure 8. Predicted Vibration Characteristics with and without Squeeze Film Damper.

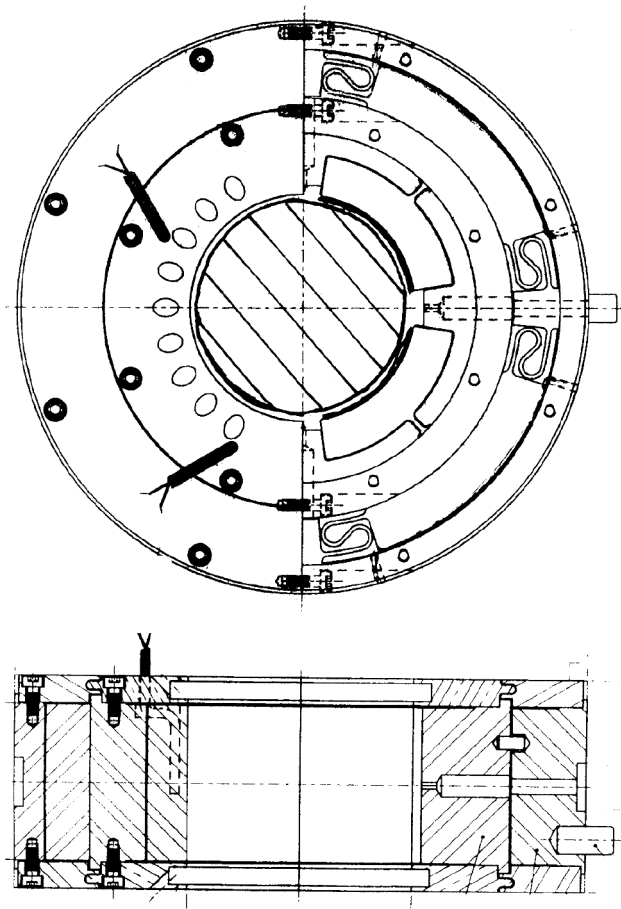


Figure 9. Bearing with Integrated Squeeze Film Damper.

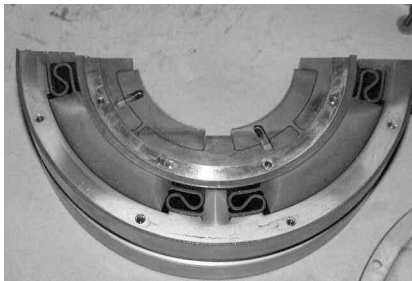


Figure 10. Bearing Half.

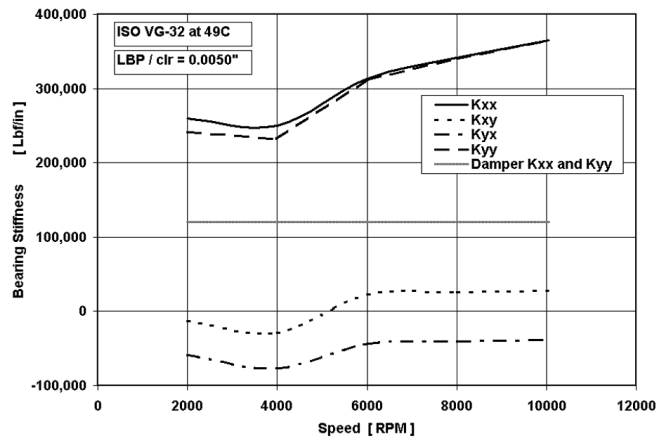


Figure 11. Impeller End Bearing Stiffness.

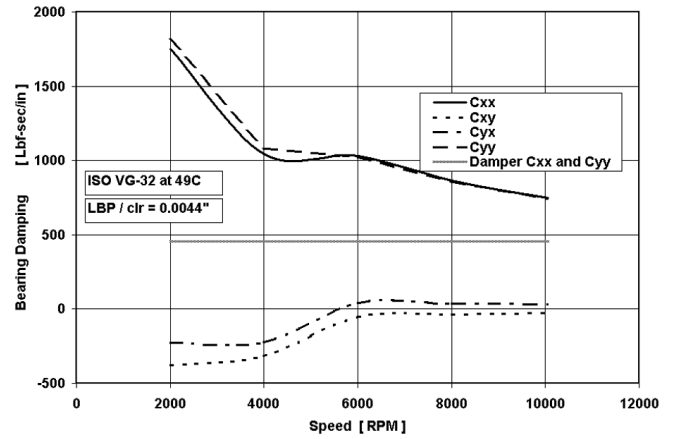


Figure 12. Impeller End Bearing Damping.

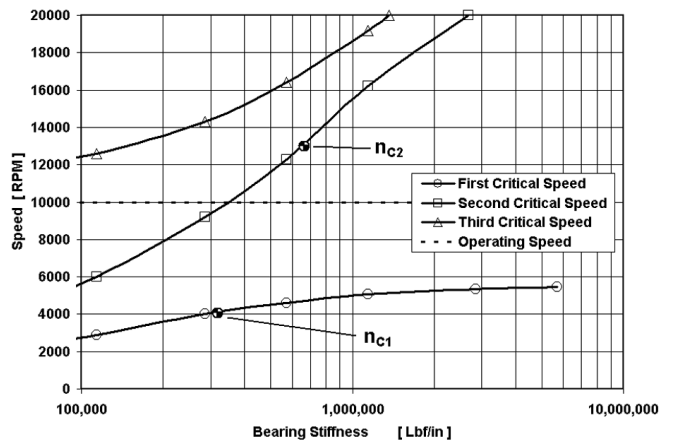


Figure 13. Rotorbearing Critical Speed Map.

low point of the purge chamber as well as in the low points of the labyrinth area closest to the impeller backface allow defined removal of excess water and their liquid reaction products. The seal system is augmented by a nitrogen feed line, a water line for washing, and a drain line together with the necessary valving.

SHOP TEST

Obviously, the most important result of the shop test (Figure 15) outside the usual topics was the behavior of the squeeze film damper bearings. After the normal balance condition was tested, additional unbalance weights of 4000 gmm (gram-millimeters) were attached to the balancing groove. This unbalance corresponds

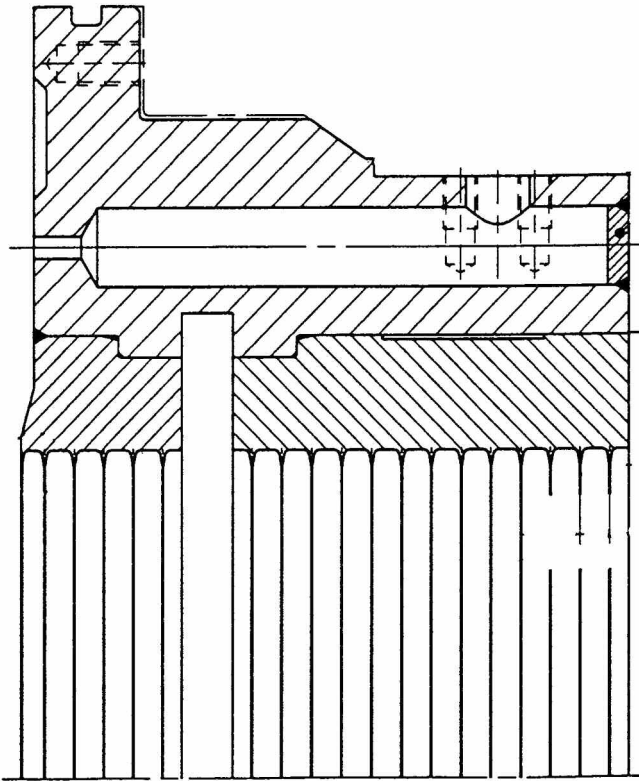


Figure 14. Seal Details.

to deterioration by a factor of more than 10 with respect to the normal balance quality. As can be seen in Figure 16, the desired characteristic was not quite reached: the rotor does exhibit a range of speeds below the operating speed with a significantly increased vibration level. A subsequent check of base frame vibratory interaction did not reveal any obvious contribution to these observations. A numerical study conducted afterward in order to simulate the observed behavior led to the postulation of an effective damping coefficient of the squeeze film damper of about four times the design value. The location of the critical speed (was predicted at 40 percent of operating speed but was actually just below 60 percent as observed by phase shift during test) also leaves some open questions with the current ability to accurately predict characteristics of such combined configurations. For the actual application, however, the results were satisfying as the actual damping capability of the rotor was almost as predicted (the maximum vibrations were about 10 percent above the predictions). Due to the larger than expected differences in vibration level at operating speed (significantly lower than predicted), the alarm and shutdown levels for the operating speed regime had to be adjusted accordingly.

OPERATING EXPERIENCE

In most of the lead author’s company’s processes it is technically possible to do a good clean up on the compressor suction flow contaminants. Several company chemical engineering consultants have developed leading edge designs that are capable of a very high degree of separation when the gas contains either clearly defined dry solids or liquids. Unfortunately, this particular process stream is corrosive and routinely contains sticky, abrasive solids that have proven exceedingly difficult to confine to either the solid or liquid state (Figures 17 and 18). Thus, we have had to learn to be content with doing the best we can under the process conditions. To obtain high reliability, it was necessary to design the compressor to be more tolerant of the normal process operating conditions.

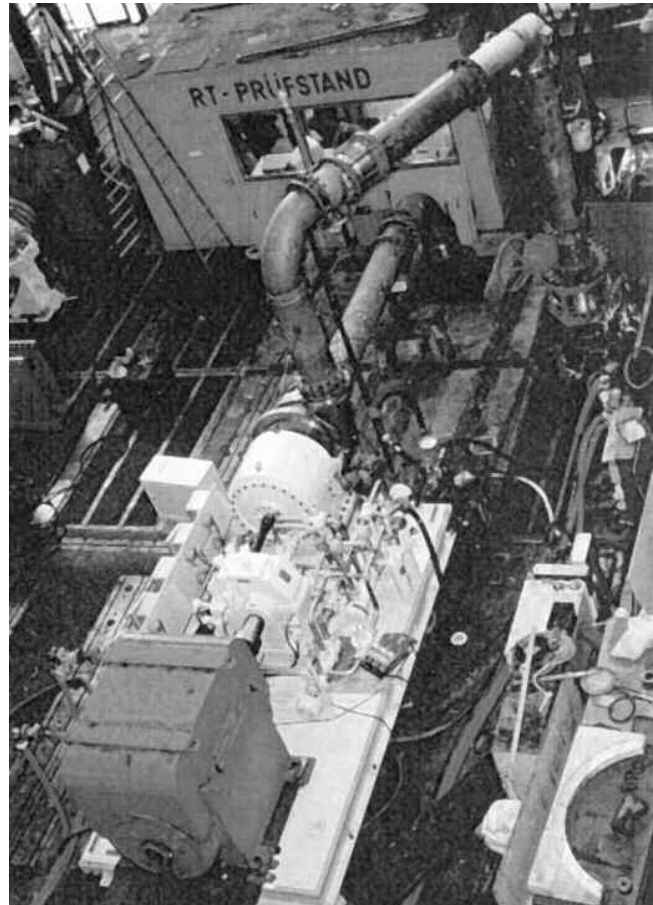


Figure 15. Shop Test Setup.

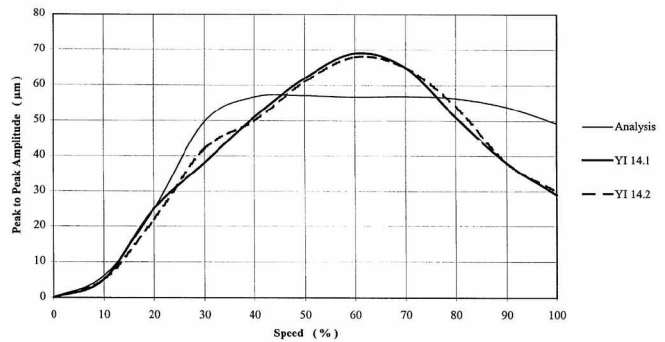


Figure 16. Measured Vibration Characteristics with 4000 GMM Unbalance.

Unbalance Tolerance in Normal Operation

Buildups on the impeller routinely occur in this line, and the rotor and squeeze film damper bearing design has been proven in tolerating the routine buildups much better than a similar single-stage machine on another process line. The normal buildups do not cause any significant wear on impeller and stationary parts. Although the buildups sometimes cause vibration to go over one mil on this 10,000 rpm rotor, the squeeze film damper absorbs the energy sufficiently to avoid excessive bearing temperatures and allows operation to continue until the rotor and casing can be cleaned during scheduled outages.

Startup Oil Contamination Due to Valve Setup Errors

Most compressor systems have some startup problems, and this installation was no exception. Soon after the compressor was put in

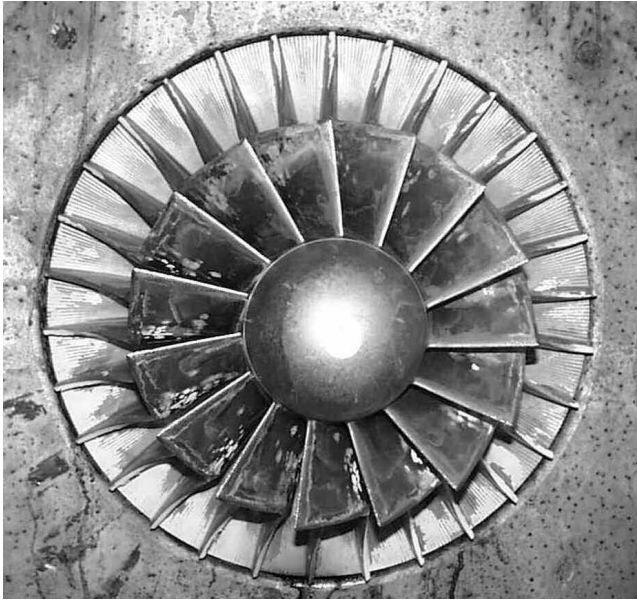


Figure 17. Impeller after Washing—Impeller Still in Good Condition.



Figure 18. Normal Dirt Buildups on Impeller—Vibration Can Move Up and Down Somewhat Depending on How Dirt Sheds. Impeller Still in Good Shape Here.

operation, it became apparent that we were getting backflow of process material into the lube oil system during machine shutdowns via the nitrogen purge gas line. Although the process labyrinth seal and the bearing labyrinth seals were fed from the nitrogen header through different regulators, operator valving error and inability to completely deinventory the process piping conspired to create the backflow from the process labyrinth seal into the bearing labyrinths (Figures 19, 20, and 21). This caused rapid, severe corrosion damage to the bearings, shafts, and gears, and several changes were required to prevent contamination:

- A valve configuration changed to adequately deinventory the process
- Added a check valve to the purge gas feed to the process shaft laby seal
- Removed the handle from a valve to prevent use in normal operation
- Converted from one common header for the bearing labyrinth purges to individual regulators

Many of our compressors successfully use one common header to supply purge gas to the bearing labyrinth seals. However, on this machine, it became very difficult to provide adequate flow to buffer the bull gear seal without overfeeding the other seals and vents, and the individual regulators solved the problem.

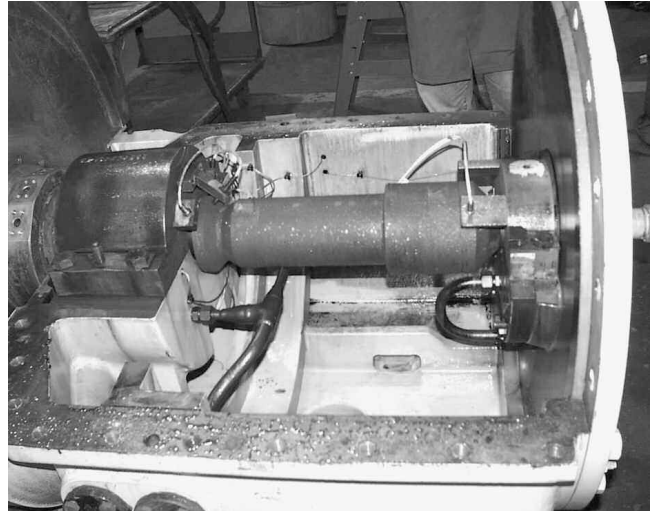


Figure 19. Corrosion Due to Oil Wetted Side Contamination—Oil Was Contaminated by Process Backflow from Process Shaft Labyrinth Seal into Bearing Labyrinth Nitrogen Purge System.

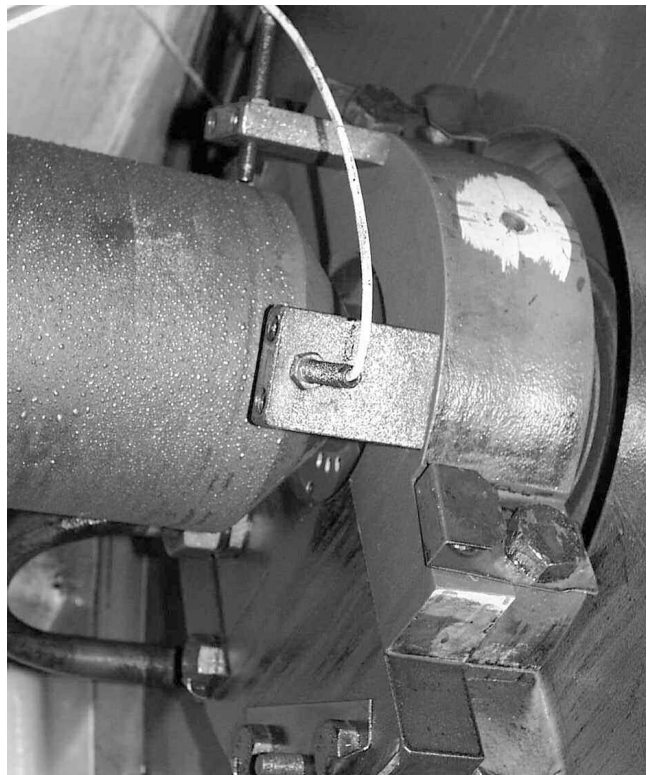


Figure 20. Corrosion Due to Oil Wetted Side Contamination—Note Heavy Scale on Steel Parts.

Demonstrated Advantages of Pinched Diffuser

The impeller tip speed was intentionally limited to speeds that were already proven successful on other process lines. This required use of a larger impeller and casing at lower rpm than would have normally been quoted. Excess flow would have had to be recycled with the standard diffuser design, but use of a pinched diffuser allowed the current capacity requirements to be met without recycle and wasted energy. This decision kept open the possibility for straightforward capacity increases in the future by removing metal from the stationary diffuser parts. Use of pinched diffusers with removable rings also provided a much easier diffuser repair



Figure 21. Thrust Bearing Heavy Babbitt Corrosion Due to Oil Contamination.

method than an integral diffuser passage. Pinched diffusers provided one additional bonus that has proven valuable. Since the impeller has more flow capability than the diffuser, the compressor capacity is also insensitive to process buildups as well as to unbalance.

Impeller Damage from Severe Process Upset—Twice—and It Still Ran!

Buildups shown above (Figure 18) are normal for this process and the rotorbearing system tolerated the unbalance very well for three years with little erosion. The high quality filtration upstream of the compressor is normally very reliable in this process. For example, one production line has had over 10 years of operation without major impeller damage.

While infrequent, when the upstream filtration fails, impeller damage is rapid and severe. Based on washing inspections, the extraordinary erosion damage shown here took place in just 10 days. Contrast the end view of the normal impeller appearance (Figure 22) with the severely eroded impeller (Figures 23, 24, and 25). Repairs were extensive enough to require complete replacement of most of the blade material (Figures 26 and 27). This happened twice to this compressor in 2002, yet despite the huge amount of wear on the impeller, the compressor was able to run below interlock vibration and temperature levels until scheduled outages for this product line. Fortunately a complete spare casing was ready to install for each of these major events.



Figure 22. Impeller Inducer Just after Cleaning, but before Heavy Erosion.

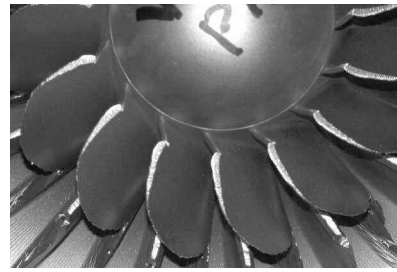


Figure 23. Impeller Inducer after 10 Days of Heavy Erosion.



Figure 24. Impeller Side View—after Heavy Erosion Damage.

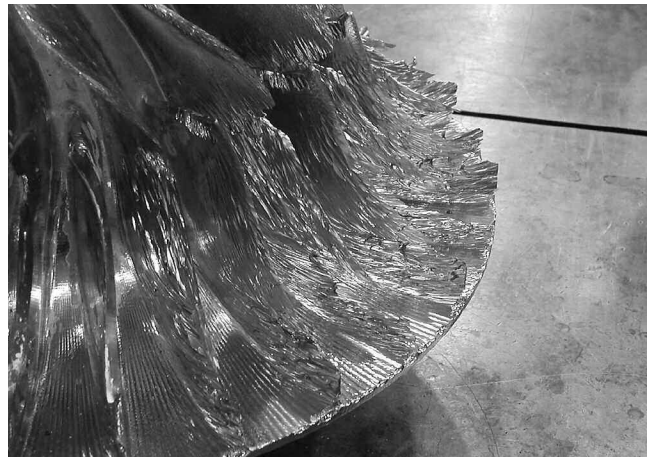


Figure 25. Impeller Closeup View—after Heavy Erosion Damage.



Figure 26. Impeller in Repair via Coupon Method.



Figure 27. Nearly Finished Repaired Impeller.

The diffusers also suffered heavy erosion (Figure 28). Although the impeller and casing repair were quite expensive, one round of bag filter replacements actually costs more than a new impeller. The impeller repair costs were essentially equal to the cost of a new rotor, but much faster than the time to fabricate a new rotor. While repair costs were very high, the production lines were sold out on both occasions and running until scheduled outages maximized the production.

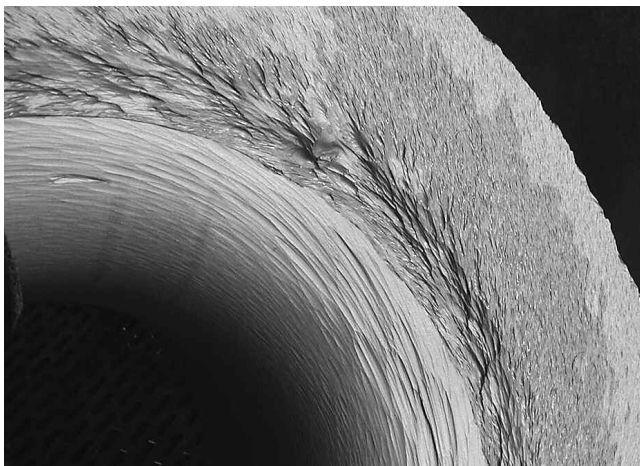


Figure 28. Diffuser Erosion Damage.

CONCLUSION

The most important feature for this compressor application was reliability, with the suction contaminants a given condition. By

sacrificing some efficiency in stage design and applying currently available squeeze film damper technology, a compressor was made and installed that has proven highly tolerant of the unbalance caused by process buildups. Use of pinched diffusers not only matched an existing compressor stage design to the process needs, the pinched diffuser also helped make the compressor flow output less sensitive to the process buildups. Other special provisions made offline cleaning of the impeller and casing easier and faster to accomplish, helping to reduce maintenance effort during scheduled outages and adding to the success of the application.

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