DESIGN IMPROVEMENTS ENHANCE DRY GAS SEAL'S ABILITY TO HANDLE REVERSE PRESSURIZATION

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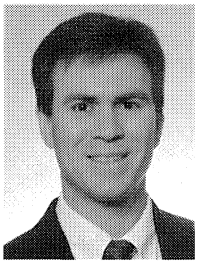

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ABSTRACT

Within the past three years, refrigeration compressors operating intermittently at subatmospheric pressures have experienced two dry gas seal failures which have been attributed, either all or in part, to reverse pressurization of the seal. Failures of this type occur while operating at subatmospheric suction pressures and/or high seal vent pressures (flare header). These failures have resulted in significant production losses and maintenance costs. The design of the seal and buffer system controls, failure analysis, corrective actions implemented by seal design changes, and buffer gas control improvements are discussed herein. Emphasis is given to the seal manufacturer's advanced modelling capabilities and operating/testing experience which has allowed refinements in the seal's design to tolerate reverse pressurization. Limitations of these design changes are also discussed.

INTRODUCTION

There are two refrigeration compressors in the ethylene plant at DuPont's Sabine River Works that are operating with tandem dry gas seals (Figure 1). The first machine, a 35,000 hp, four stage propylene compressor was retrofitted in May 1991. This machine has two compression cases; a low pressure and high pressure case, operating with suction pressures at about 1.0 psig and final discharge pressures of 255 psig. In November 1991, the plant was shut down for repairs, at which time the inboard dry gas seal on the low pressure case failed during startup. The primary seal vent on the dry gas seal ties into the plant flare header, which at the time of the failure, was operating at a higher pressure than normal due to high flare rates. Investigation into the failure concluded that the seal had experienced "reverse pressurization," which caused the seal hard faces to contact and fail. A system modification was made to prevent reoccurrence.

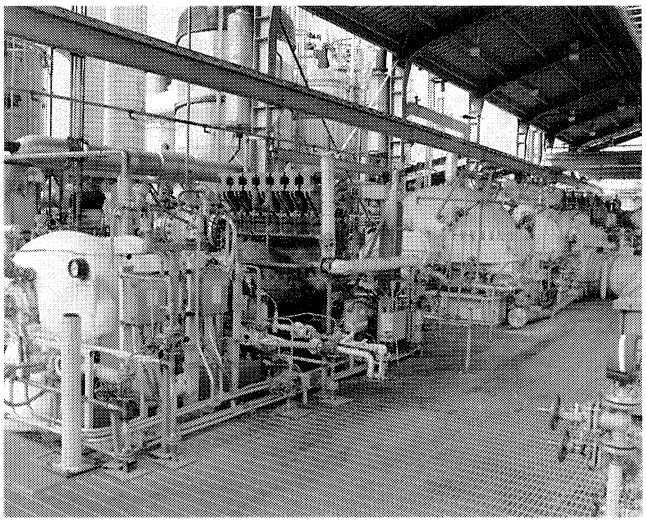

Figure 1. Ethylene and Propylene Compressors.

The second machine, a 6000 hp, three stage ethylene compressor, was retrofitted in 1992. This machine operates with suction pressures at about 0.5 psig and discharge pressures of 260 psig. During startup after a 1995 plant turnaround, the suction side dry gas seal failed after several hours of operation (Figure 2). Prior to this failure, there were liquid carryovers into the compressor

suction that lead to the compressor surging and being shut down until the problem could be resolved. These excursions were first detected when radial vibration alarms sounded. About eight hours after restart, the seal failed. During the investigation, it was noted that the suction pressure was operating at subatmospheric levels for several brief durations. Investigation also revealed that the compressor had experienced subatmospheric conditions a multitude of times since its initial startup. A review team studied the failure and confirmed that the subatmospheric suction pressure operation had caused the seal to experience "reverse pressurization," the magnitude of which was difficult to determine. It was the investigation team's opinion that the failure had occurred, in part, due to "reverse pressurization," which was aggravated by the liquid ingestion experienced earlier.

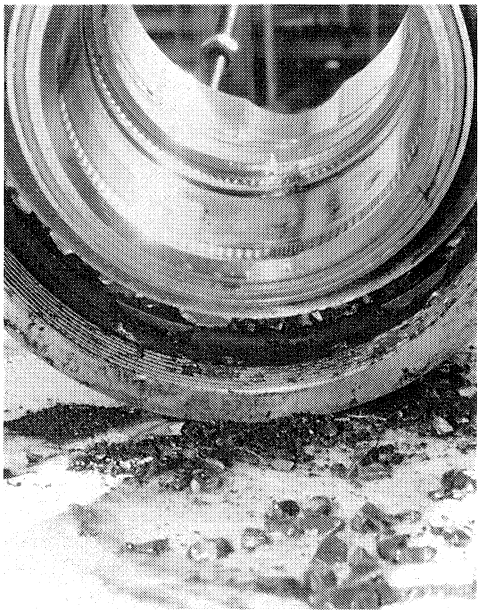

Figure 2. Failed Seal from Ethylene Compressor, May 1995.

As noted previously, these failures resulted in significant production losses and maintenance cost. To ensure that this type of failure would not occur again, a thorough review of the seal design, buffer system controls, and piping was conducted. This review led to several recommendations involving the seal and control systems, and piping upgrades.

BACKGROUND

Seal Description

The seal is a tandem dry gas seal (Figure 3). The seal has primary and secondary units that operate with a buffer gas. The primary seal is closest to the process and uses filtered process gas from the compressor discharge as the buffer and is controlled to maintain a ΔP of 0.5 psi above the compressor suction pressure, which is measured at the balance line (Figure 5). Approximately 99 percent of this gas leaks past the primary labyrinth seal, back into the compressor suction. The remaining one percent leaks past the tungsten/carbon hard face seal into the primary vent. The primary vent has a three way valve which is normally lined up to the flare, until the vent pressure reaches 5.0 psig, at which time it automatically lines up to atmosphere.

The secondary seal employs an intermediate labyrinth to ensure that no primary seal leakage flows through the secondary seal. A dry nitrogen buffer gas is injected at the secondary seal and is controlled to maintain a ΔP of 0.5 psi above the primary vent pressure. Approximately 99 percent of the Nitrogen leaks past the intermediate labyrinth seal into the primary vent. The remaining one percent is pumped past the secondary hard face seal and into the secondary vent, which is routed to atmosphere at all times. A second N_2 buffer is applied between two labyrinth seals with the flow split at approximately 50/50. This keeps bearing oil from migrating into the dry gas seal. The supply for this N_2 is taken downstream of the N_2 control valve to the secondary seal.

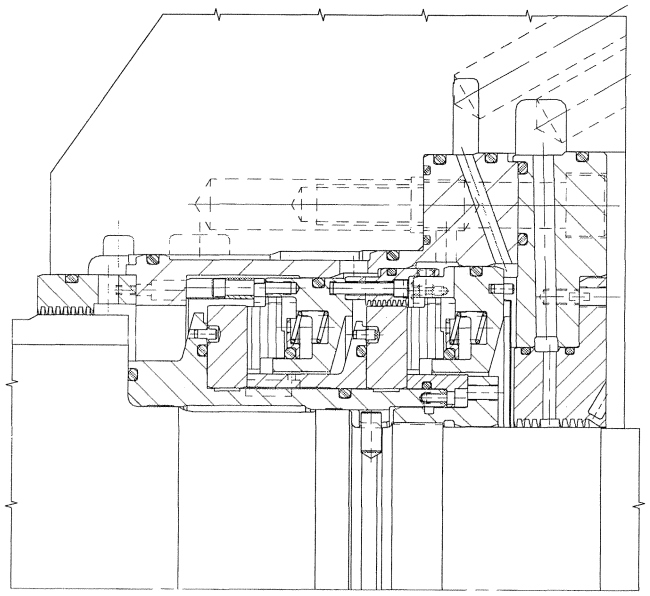

Figure 3. Cross Section of Tandem Dry Gas Seal.

The heart of the dry gas seal is the stationary carbon and rotating hard faces which are noncontacting during operation. This noncontacting feature is what makes this seal unique since it results in low heat generation, low power consumption, and ultimately longer life without the necessities of oil lubrication. This noncontacting operation is achieved by specially designed spiral grooves in the rotating tungsten carbide disc face that produce a pumping action which generates a pressure gradient across the seal faces, from the outside diameter (OD) to the inside diameter (ID) (Figure 4). This pressure gradient, coupled with a positive pressure buffer gas, causes the seal faces to separate by approximately 0.0001 in to 0.0002 in against the spring forces.

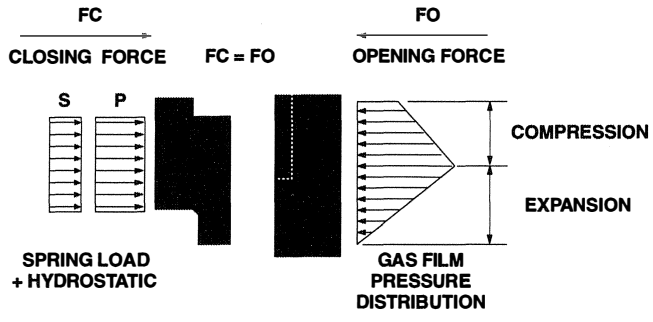

Figure 4. Pressure Gradient across Spiral Groove Rotating Face.

Pressure reversal (or reserve pressurization) occurs when the vent pressure (point C on Figure 5), or pressure on the ID of the seal, is higher that the supply pressure on the OD of the seal (point B on Figure 5). When the force from the reverse pressure is great enough to overcome the pressure generated by the pumping action of the spiral grooves, gas film between the seal faces is lost, which initiates contact.

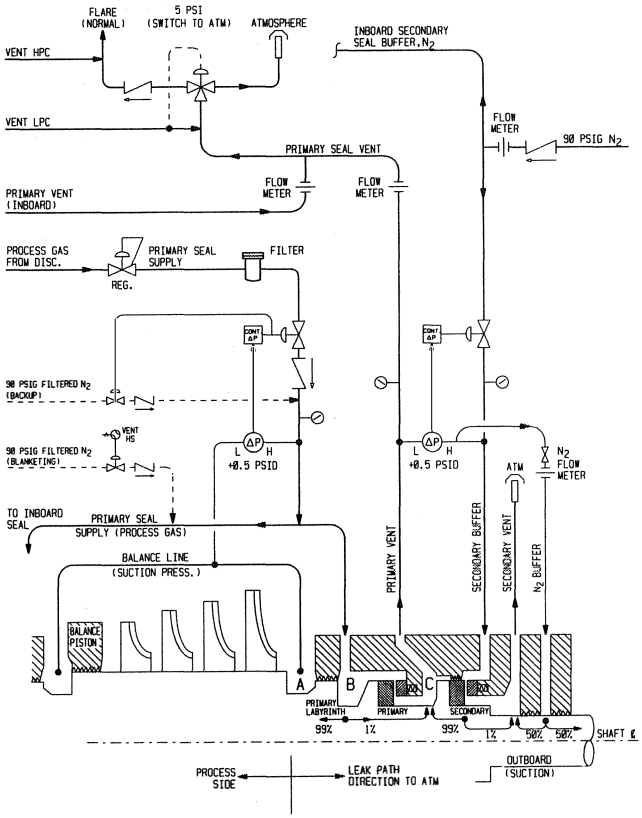

Figure 5. Original Control System.

The damage created during these periods of contact is cumulative and dependent on many factors including: speed, pressure differential across the seal, and the duration of contact.

Propylene Compressor Seal Failure

Originally the three-way valve on the primary vent was set to switch from the flare header to atmosphere when the flare header pressure reached 12 psig. The dry gas seal failed during startup when the flare header pressure was at 8.0 psig, at which point, there was approximately a 6.0 psig reverse pressure on the seal. To prevent reoccurrence, the setting on the three-way valve was changed from 12 psig to 5.0 psig, which at that time was considered to be a safe limit. This arrangement is shown in Figure 5.

Ethylene Compressor Seal Failure

The seal that failed is located on the suction side of this machine. Review of the process variables revealed that a "pressure reversal" on the seal hard faces has occurred many times since the seals were originally installed in 1992. Referring to Figure 5, this phenomena occurs when the suction pressure, point A, drops below atmospheric. The primary seal supply gas (fresh ethylene) is regulated 0.5 psi above the suction pressure via a ΔP controller. The primary vent is either routed to the flare or atmosphere, depending on flare header pressure. If the suction pressure (point A) drops below −0.5 psig, the primary seal supply control valve closes since the 0.5 ΔP is satisfied. If the vent pressure (point C) is 0 psig (when routed to atmosphere) and the suction pressure (point A) is −5.0 psig, the supply pressure (point B) is −4.5 psig, then a 4.5 psi pressure reversal occurs. Refer to Table 1 for examples of different suction pressure and vent pressure conditions. In most cases, the primary vent is routed to the flare which results in a higher pressure reversal, depending on the flare header pressure. Note that a pressure reversal of −19.5 psig could occur if the flare (vent) pressure is 5.0 psig and the suction pressure drops to −10.0 psig (Table 1). When the flare header pressure reaches 5.0 psig, a three-way valve automatically routes the vent to atmosphere, thereby reducing the backpressure.

Table 1. Original Control System Operating Characteristics.

Primary Labyrinth Seal supply control valve set to maintain +0.5 psid.					
Comments	Suction Pressure (Point A)	Primary Seal Supply Pressure (Point B)	Vent Pressure (Point C)	Primary Labyrinth ΔP (Point B - A)	Primary Seal ΔP ** (Point B -C)
Slightly Positive Suction Pressure.	+0.5	+1.0	0	**+0.5**	**+1.0**
Slightly Negative Suction Pressure; equal to control ΔP setting	-0.5	0	0	**+0.5**	**0**
Negative Suction Pressure.	-5.0	-4.5	0	**+0.5**	**-4.5**
Negative SuctionPressure with increasing Flare (Vent) pressure. Danger Zone!	-5.0	-4.5	+5.0	**+0.5**	**-9.5**
Suction pressure continues to decrease.	-10.0	-9.2	+5.0	**+0.5**	**-19.5**

Note: 1. ** Negative values indicate "Reverse Pressurization".
2. Pressures are in psig.
3. These values are based on the Primary Labyrinth Seal supply control valve set to maintain +0.5 psid between point A and B (Figure 5).

The damage created during these periods of contact is cumulative and dependent on the magnitude of the contact. The magnitude is dependent on the speed, pressure differential across the seal, and the duration of contact. In the case of this failure, the speed was high (12,000 rpm), the reverse ΔP is estimated to be low (1.0 to 2.0 psi), and the durations are short (a few minutes). An increase in any of these variables would increase the damage.

The failure of this seal is also coincidental with a liquid intrusion that occurred earlier that day. The seal that failed is on the suction side, which is where the liquid entered. Although there were no data to support the possibility of the liquid causing the failure, it is believed to have aggravated the seal wear and to have contributed to the failure.

SYSTEM DESIGN IMPROVEMENTS

Interim Operating Changes

It was recognized, with the current control arrangement, that the 0.5 psi ΔP setpoint across the primary labyrinth seal could be increased to +5.0 psi, which would reduce the possibility of reverse pressurization. With the higher ΔP, the compressor suction pressure would have to decrease an additional 4.5 psig, under the same vent conditions, to cause reverse pressurization. The only negative aspect would be an increase in buffer gas flow, which slightly reduces compressor efficiency. This change was made immediately on the ethylene machine. Refer to Table 2 for the operating characteristics with the increased ΔP.

This proved to be a problem on the propylene compressor, which had high leakage rate across the primary labyrinth since the original installation and had problems maintaining the recommended 5.0 psi ΔP. As a result of the high leakage rate, only 0.5 psi could be maintained without the backup valve opening and flooding the compressor. The ΔP set point was left at 0.5 psi since it was undesirable to operate with a continuous N_2 flow into the compressor.

Future Upgrades

Additional discussions with the seal manufacturer were conducted to determine more permanent fixes. From these discussions came the following recommendations, some of which are included in Figure 6:

• Incorporate a pressure override system for the primary seal which will monitor and control the ΔP across the primary seal. This

Table 2. Original Control System Operating Characteristics with Increased ΔP.

Primary Labyrinth Seal supply control valve set to maintain +5.0 psid.					
Comments	Suction Pressure (Point A)	Primary Seal Supply Pressure (Point B)	Vent Pressure (Point C)	Primary Labyrinth ΔP (Point B - A)	Primary Seal ΔP ** (Point B -C)
Slightly Positive Suction Pressure.	+0.5	+5.5	0	**+5.0**	**+5.5**
Slightly Negative Suction Pressure.	-0.5	+4.5	0	**+5.0**	**+5.0**
Negative Suction Pressure equal to control ΔP setting.	-5.0	0	0	**+5.0**	**0**
Negative Suction Pressure with increasing Flare (Vent) pressure.	-5.0	0	+5.0	**+5.0**	**-5.0**
Suction pressure continues to decrease. Danger zone!	-10.0	-5.0	+5.0	**+5.0**	**-10.0**

Note: 1. ** Negative values indicate "Reverse Pressurization".
2. Pressures are in psig.
3. These values are based on the Primary Labyrinth Seal supply control valve set to maintain +5.0 psid between point A and B (Figure 5).

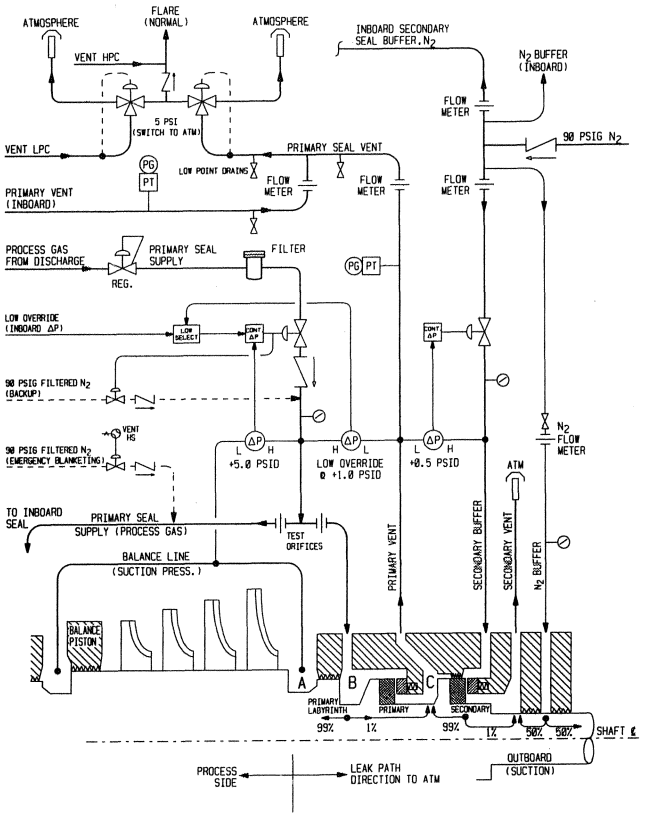

Figure 6. Future Upgraded Control System.

will override the current controller across the primary labyrinth ΔP. Refer to Table 3 and Figure 7 for the control system operating characteristics.

• Install pressure transmitters to determine actual seal pressure in addition to the ΔP across the primary seal.

• Install separate primary seal vent lines to the flare/atmosphere from each compressor. This will allow independent operation of each compressor and reduce backpressure from the other compressor seal vents.

• Add flowmeters to each secondary seal buffer supply to detect seal leakage variations.

Table 3. Future Control System Operating Characteristics.

Primary Labyrinth Seal supply control valve set to maintain +5.0 psid and Primary Seal Overide set to maintain +1.0 psid.					
Comments	Suction Pressure (Point A)	Primary Seal Supply Pressure (Point B)	Vent Pressure (Point C)	Primary Labyrinth ΔP (Point B - A)	Primary Seal ΔP ** (Point B -C)
Slightly Positive Suction Pressure.	+0.5	+5.5	0	**+5.0**	**+5.5**
Slightly Negative Suction Pressure.	-0.5	+4.5	0	**+5.0**	**+4.5**
Negative Suction Pressure. Low Primary Seal ΔP Override takes over.	-5.0	+1.0 (0 without override)	0	**+6.0**	**+1.0**
Negative Suction Pressure with increasing Flare (Vent) presure.	-5.0	+6.0 (0 without override)	+5.0	**+11.0**	**+1.0**
Suction pressure continues to decrease.	-10.0	+6.0 (-5.0 without override)	+5.0	**+16.0**	**+1.0**

Note: 1. ** Negative values indicate "Reverse Pressurization".
2. Pressures are in psig.
3. These values are based on the Primary Labyrinth Seal supply control valve set to maintain +5.0 psid between point A and B (Figure 6) and the Primary Seal Override set to maintain +1.0 psid minimum.

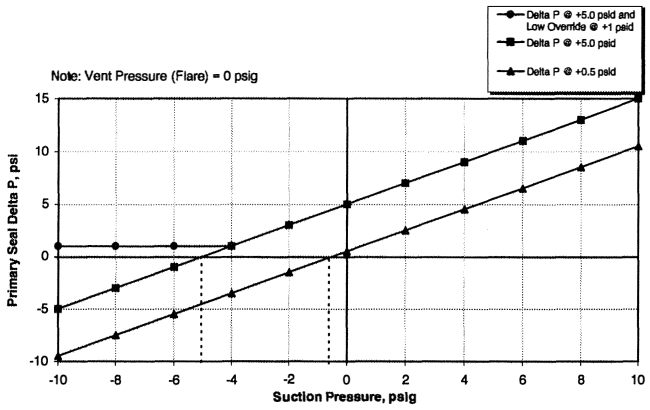

Figure 7. Control System Operating Characteristics.

• Add low point drains to the seal vent lines. This was recommended because of oil build up in the vent line, from its original installation as part of an oil type seal, which can cause restrictions in the vent line and increase backpressure.

• Relocate the N_2 source that supplies the outboard labyrinth of the ethylene refrigeration compressor seals. This takeoff was located downstream of the flow control valve to the secondary seal N_2 buffer, which caused the flow to the outboard labyrinth to vary as the flow control valve modulates. It was recommended to locate it upstream of the flow control valve and secondary seal flowmeter (Figure 6). Upstream of the secondary seal flowmeter will give a more accurate flow to the secondary seals.

• Modify the propylene machine's primary labyrinth to reduce the leakage rate. Without this modification, the required ΔP across the primary labyrinth cannot be maintained without using the backup N_2 to supplement the primary buffer gas. In this case, it is undesirable to inject N_2 into the propylene system since it will affect the compressor and refrigeration performance.

• Perform analytical reviews of the existing seal design to determine the maximum allowable reverse pressurization the seals can withstand. If unacceptable, redesign the seal face geometry using advanced seal design principles to increase the reverse pressurization capabilities. The results of this review is discussed in detail in the following sections.

SEAL DESIGN IMPROVEMENTS

The scope of this section is limited to noncontacting, single balanced, dry running spiral groove gas seals intended for compressor applications. The following information does not apply to dual balanced seals or noncontacting seals intended for pump applications.

As described earlier, even a relatively small pressure reversal (less than 5.0 psid) can have an adverse effect upon the dynamic operation of a gas seal. A pressure reversal or reverse pressure differential is defined as greater pressure at the seal face ID relative to the OD, while a positive pressure differential is defined as greater pressure at the seal face OD relative to the ID. It is often difficult to determine the exact magnitude or duration of the pressure reversal, and the surface speeds where failure may occur, but the effects appear to be cumulative. This is due largely to the collapse of the gas film between the seal faces at reverse pressure conditions.

One of the most arduous duties for a standard dry gas seal is dynamic operation during a prolonged, small pressure reversal. In this situation, lift off may not occur and the seal faces may begin to operate in a contacting mode, generating heat until the rotating hard face becomes heat checked and ultimately fails. Conversely, if the magnitude of the reverse pressure differential is large, in either a static or dynamic operating mode, the seal is likely to "blow open," reducing the probability of catastrophic failure.

Under these abnormal operating conditions, gas film stiffness diminishes as the magnitude of the applied pressure differential is increased. The relation between gas film stiffness and pressure differential is shown in Figure 8. In addition, gas film load capacity diminishes along with decreasing film stiffness.

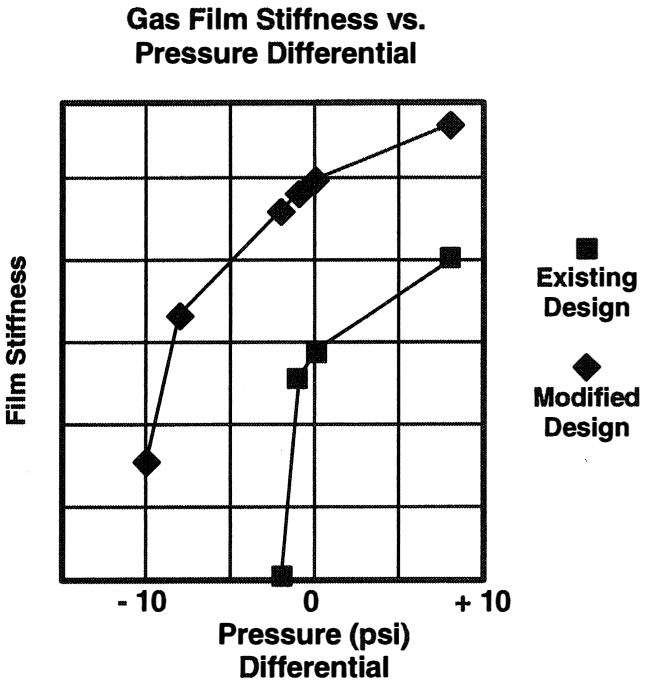

Figure 8. Seal Face Film Stiffness.

Film stiffness is a way to quantify the quality of the gas film and is typically given in terms of a force per unit length (lb/in). The stiffer the gas film, the greater its ability to support the closing forces (hydraulic load and spring force) on the seal faces. Furthermore, high film stiffness enables the gas seal to better cope with any transient conditions, such as compressor surge, shaft speed changes, temperature changes and/or pressure excursions. In the event gas film stiffness is substantially reduced or lost completely, seal face contact can occur.

Gas film stiffness is analogous to the stiffness constant of an ordinary coil spring which obeys the following linear equation commonly known as Hooke's Law.

$$F = k\Delta x \tag{1}$$

where:
f = force exerted by the spring (or gas film)
k = spring rate (or gas film stiffness)
Δx = change in spring length (or change in seal face gap)

(Unlike a linear spring, a gas film may exhibit non linear characteristics, typically at very small operating gaps.)

Stated quite simply, the more a gas film is "squeezed" between a pair of faces, the greater the force the gas film must exert back onto the seal faces. This has to be true since the summation of all forces (hydrostatic, hydrodynamic, and spring force) must be zero at equilibrium.

As previously mentioned, if the applied pressure reversal is great enough, the seal faces will separate sufficiently, eliminating any danger of hard rubbing and subsequent failure. In this situation, the stationary seal face balance ratio is significantly reduced. (Balance ratio is defined as the ratio of hydrostatic closing force to opening force, and is therefore a function of the surface areas of the seal face and the applied pressures.) The static force balance on a positively pressurized seal face is illustrated in Figure 9 (a), while a quite different force balance on a reverse pressurized seal face is illustrated in Figure 9 (b). The balance ratio of a reverse pressurized seal can be approximated by the following equation:

$$BR_- = 1 - BR_+ \tag{2}$$

where:
BR_- = balance ratio of a reverse pressurized seal face
BR_+ = balance ratio of a positively pressurized seal face

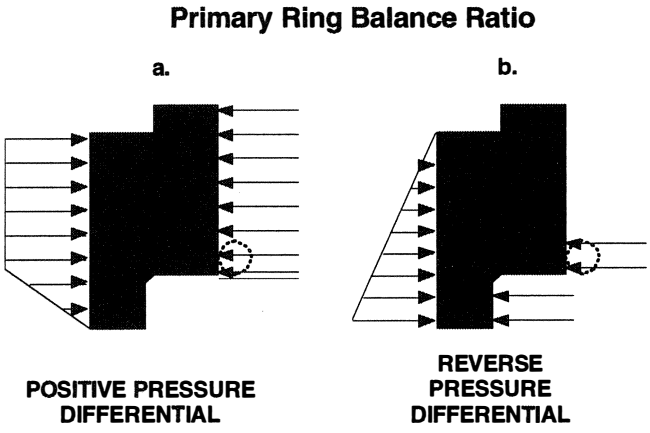

Figure 9. Primary Ring Balance Ratio.

Once the hydraulic force balance on the seal face exceeds the spring load, the seal faces will separate when the following equation is satisfied:

$$\Sigma F_o > \Sigma F_c \tag{3}$$

where:
ΣF_o = summation of opening forces
ΣF_c = summation of closing forces

Substituting pressure times surface area for force ($F = P \times A$) gives:

$$P(A_o) > P(A_c) + L_s \quad (4)$$

where:

P = magnitude of applied pressure reversal differential
A_o = surface area of stationary seal face
A_c = surface area bounded by the balance diameter O-ring and seal face ID
L_s = total spring load

With the aid of advanced computer modelling, analysis of the existing seal design revealed increased face gap, reduced film stiffness, and a change in the combined face angle (angle formed by the seal faces) from the desired convergent face pattern to a divergent face pattern as the magnitude of the dynamic pressure reversal is increased. (A convergent face profile is defined as a taper formed by a pair of seal faces where the gap is greatest at the face OD.)

The primary seal of the existing tandem arrangement was modified to accommodate a limited pressure reversal (approximately 8.0 psid), dynamically. Since the secondary seal was vented to atmosphere, and hence could not be reverse pressurized, only the primary seal was modified as described later.

The objective was to increase the pressure rise generated by the spiral grooves (at the minimum operating speed of the compressor), such that a barrier would be created against reverse flow. Reverse flow is defined as leakage between the seal faces that flows radially outward, i.e., from seal face ID to OD. In addition, the upgraded design must maintain a stable gas film and operate with a converging face profile throughout the full range of operating parameters including shaft speed, seal pressure, and temperature. All of these requirements were satisfied with a redesign that employed a more efficient spiral groove geometry and an increased spring load on the seal faces.

The modified spiral groove pattern alone tends to increase the dynamic operating gap which reduces film stiffness. To compensate, the total spring load was increased from the original design value to reduce the operating gap and hence restore some of the lost film stiffness. The increased spring load also improves the seal's ability to tolerate static pressure reversal before the faces separate completely.

A comparison between the dynamic pressure profiles generated by the standard spiral groove pattern and the modified spiral groove pattern is given in Figure 10. In each case, the seal OD is assumed to be at atmospheric pressure while the seal ID is pressurized to some positive value, i.e., the seal faces are reverse pressurized. The length of the vectors indicate the relative difference in the magnitude of the forces acting on the seal faces.

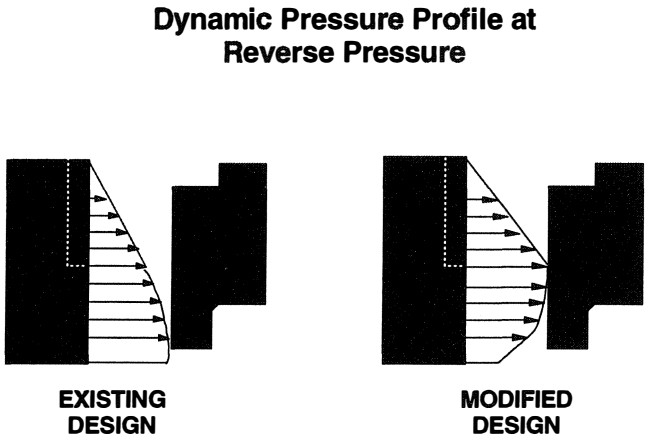

Figure 10. Dynamic Pressure Profile at Reverse Pressure.

Given the pressure profile generated by the existing design, it is clear that the maximum pressure generated by the spiral grooves (which occurs at the spiral groove/sealing dam interface) is less than the pressure applied at the seal face ID. Under these circumstances, reverse flow through the seal faces will occur, contaminating the process with gas from the flare line. In this case, the gas film stiffness is relatively low.

Conversely, the pressure profile generated by the modified design illustrates that the spiral grooves generate a maximum pressure at the spiral groove/sealing dam interface greater than the pressure applied to the seal face ID. The increased pressure generated by the spiral grooves will ensure that flow through the seal faces is in the desired direction, radially inward, from OD to ID. This also eliminates the threat of process contamination from the primary seal vent line.

Another comparison based on film stiffness between the existing seal design and the upgraded seal design is given in Figure 8. Gas film stiffness is plotted as a function of the applied pressure differential for each design. Note the increased film stiffness generated by the upgraded design throughout the range of pressure differentials shown.

During the rework of the existing design, the high pressure capability of the modified design was compromised. This phenomenon is the result of a shift from the desired convergent face profile (present at low positive pressure differentials), to a parallel, and finally divergent face profile that occurs at high positive pressure differentials. It should be understood that a new seal may be designed to operate dynamically under reverse pressure conditions without compromise of the positive pressure limit.

The reduced high pressure limit of the modified design is typically not a concern, since most seals operating in low suction pressure applications only require modest maximum pressure limits. Similarly, high pressure seal applications are unaffected by abnormally high primary seal vent pressures.

TESTING

To verify the integrity of the modified design, extensive testing was performed at the seal manufacturer's facility. A schematic representation is shown in Figure 11 of the test set up that defines seal leakages, buffer flow path, and pressure.

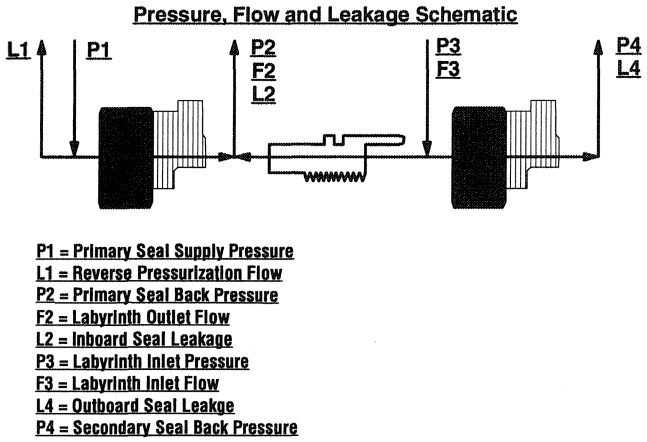

Figure 11. Pressure, Flow, and Leakage Schematic for Testing.

Buffer pressure was applied to the intermediate labyrinth (P3) and its corresponding flowrate (F3) was recorded. The primary seal leakage port (P2) was closed to insure that F2 and L2 were equal to zero. Primary seal pressure (P1) was maintained at atmospheric pressure (zero psig). A flowmeter was connected to L1 (the cavity between the pair of seal cartridges; one clockwise and one counter

clockwise) to detect any unwanted reverse flow through either of the two primary seals.

Following a standard static and dynamic test at normal operating conditions, reverse pressure was applied incrementally to the primary seal (P3), up to a maximum limit of 8.0 psig. Shaft speed was varied between 10,500 rpm and 12,000 rpm. At no point during the test was any reverse flow through the primary seal detected which proved the modified spiral groove's ability to overcome the 8.0 psi pressure reversal. Both cartridges were disassembled after the test to observe the condition of seal faces. No signs of distress were found which is indicative of a stable gas film (adequate film stiffness).

SUMMARY

Through a joint effort between the end user and seal manufacturer, combinations of abnormal operating conditions were identified as factors which contributed to unusual seal failures. Advanced modelling techniques confirmed the cause of failure and enabled the seal manufacturer to enhance the design to operate satisfactorily even under these abnormal conditions. In addition, a thorough review of the seal control system and subsequent improvements will significantly reduce the potential of reverse pressurization.

ACKNOWLEDGMENTS

The authors would like to acknowledge the contributions of Johnny Dugas, Sam Culpepper, E. I. DuPont de Nemours & Co., Inc., Orange Texas, Glenn Pecht, John Crane Inc., Morton Grove, Illinois, and Amrat Parmar, John Crane EAA, Slough Berkshire, England, in evaluating and implementing design improvements to the dry gas seals, and for help in preparing this paper.