DEVELOPMENT AND FIELD APPLICATION OF A SINGLE ROTOR DESIGN DRY GAS SEAL

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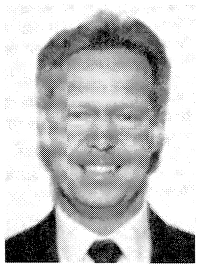

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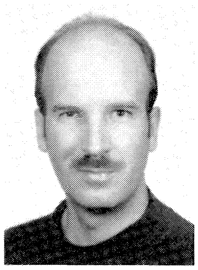

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ABSTRACT

Improving centrifugal compressor seal performance and reliability can be accomplished through development of new technology and/or by improvement of existing technology. An improved dry gas lubricating seal of single rotor design has been designed for high pressure gas compressors. A more durable seal of simpler design has been created through implementation of ductile materials and reduction of components. Currently, the tandem dry gas seal configuration is the accepted standard for high pressure natural gas pipeline compressor applications. These applications still suffer problems with process and lube oil contamination, brittle fracture of carbide materials, and explosive decompression of O-rings. Real costs associated with these problems are considerable. Development and potential benefits of the single rotor design seal, along with field application, in a high pressure natural gas centrifugal compressor is discussed.

INTRODUCTION

A variety of mechanical seal oil systems (wet seals) and dry gas lubricating seal systems (dry seals) have been devised to seal centrifugal high pressure gas compressors. Early compressor applications utilized wet seal technology. Problems and life cycle costs of wet seal systems were discussed by Hesje and Peterson [1] in 1984 and compared to newly developed dry gas seal installations. Pennick [2] subsequently described the advantages of typical dry gas seal design and the benefits of these seals applied in the process industry in 1985. Continuing work has led to the acceptance of the dry gas sealing system as an industry standard.

Unfortunately, current applications of the dry gas sealing technology still suffer problems from process and lube oil contamination, brittle fracture of carbide materials, and explosive decompression of O-rings. Real costs associated with these problems can be considerable. Elimination or reduction of these problems can translate into significant cost savings to the user.

The authors believe the best measure of compressor sealing performance is the ability to effectively seal the compressor with minimal leakage and to do so in a cost effective manner that addresses several variables including: reliability, durability, confidence, reduction of costs, and simplicity of operation. The use of tungsten carbide (WC) for seal mating rings continues to have a negative effect on several of these variables. Factors such as rotordynamics effects, ease of installation and removal, along with maintenance costs are also of concern. It is shown herein that use of ductile mating ring materials provides significant benefits.

Currently, the tandem dry gas seal configuration is the accepted standard for high pressure natural gas compressor applications. A single stage dry gas lubricating seal that utilizes a ductile mating rotor, of complex geometry, in conjunction with a barrier seal and control system is discussed. Laboratory and field tests in natural gas service have proven the durability of the materials in the single rotor design seal. To date, the single rotor design seal has been subjected to a plethora of laboratory tests and has successfully accumulated over 2,200 hours of service in a high pressure natural gas pipeline compressor. The seal was inspected after 2,100 hours of service, was reinstalled, and continues to operate successfully.

BACKGROUND

Traditional dry gas seals were first successfully implemented by a pipeline company in centrifugal gas compressors in the early 1980s. The currently accepted configuration for high pressure applications is a single cut, tandem (two stage) seal capsule. As shown in Figure 1, the traditional tandem dry gas seal is comprised of several components. Conventional carbide materials are used for the mating ring(s). From a design (risk) perspective, the primary reason for implementing a tandem seal for high pressure service is to provide 100 percent redundancy in the event of a catastrophic failure of the first stage seal elements. Buffered labyrinth seals are typically installed at both ends of the seal to prevent process and lube oil contamination from entering the seal.

The majority of problems originally encountered with traditional dry gas seals were the brittle fracture of WC mating rings and "explosive decompression" of the O-rings [1]. Although most problems with explosive decompression have been resolved, the use of WC mating rings remains problematic in rigorous applications.

The inherent problem with the use of WC is the propensity for the material to easily heat check when subjected to thermal shock. As experienced by the pipeline company, heat checking of WC mating rings is caused by seal face contact resulting from excessive shaft excursion or seal face contamination. Cracks of critical length can quickly propagate, resulting in catastrophic failure of the mating ring.

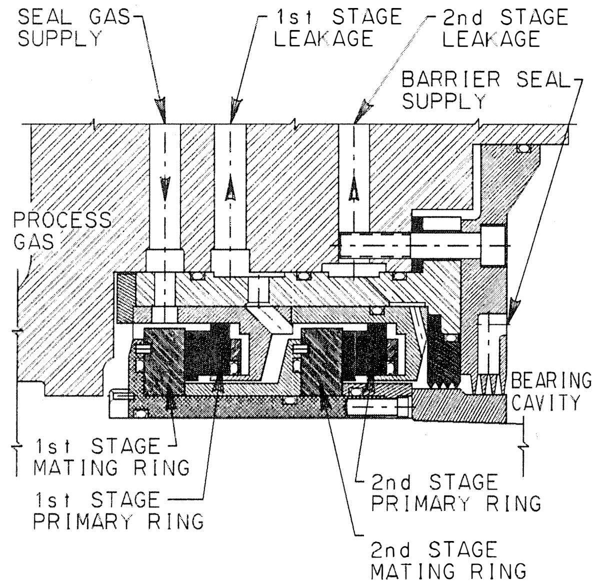


Figure 1. Traditional Two Stage (Tandem) Dry Gas Seal.

In 1994, the pipeline company had eight incidents where heat-checking of the WC mating ring was observed. The WC mating rings were replaced at considerable expense. Four of these failures where catastrophic, i.e., complete fracture and disintegration of the WC mating ring. Collateral damage to peripheral components increased costs and downtime of these failures. In three of the four catastrophic failures, no indication of the impending failure was observed. The inability to confidently predict WC mating ring failure results in a high degree of uncertainty when attempting to establish seal life expectancy. This led the pipeline company to consider the implementation of scheduled seal inspections, solely for the purpose of checking the condition of the WC mating rings. The fourth catastrophic failure resulted from continued operation of a dry seal with excessive leakage. Impending failure was anticipated by steadily increasing seal leakage.

Based on the problems originally encountered with traditional dry gas seal applications, a program was initiated by the pipeline company to develop a more durable dry gas seal of simpler design, through the use of ductile materials and a reduced number of components. A five year program was implemented to demonstrate the viability of ductile mating rings and, based on success, develop the single rotor design seal concept. As a result of continued operating problems associated with traditional dry gas seals, the progression of this work remains valid.

PREVIOUS MATERIALS WORK — CONTINUED INVESTIGATION

Development of ductile mating ring materials that provided improved high pressure gas sealing performance over traditional brittle carbide materials was the basis for the materials investigation. Currently, carbide materials such as WC and silicon carbide (SiC) are accepted as mating ring materials in dry gas seals. Work on alternate ductile materials had been previously performed by Gardner [3, 4], Sedy [5, 6] and Gabriel [7], although this work was not pursued. Evenson, Peterson, and Hanson [8] investigated the application of ductile mating rings in 1993. They showed that National Association of Corrosion

Engineers (NACE) specification condition ferrous alloys could be used as mating rings in dry gas seal applications. Subsequent work has shown ductile mating rings to be potentially more durable than WC in natural gas service. Nitrided AISI 4140 low alloy steel and 17-4PH stainless steel mating rings were experimentally developed and tested. The AISI 4140 mating ring was subsequently installed in the inboard dry seal capsule of an operating natural gas compressor.

High and low pressure static and dynamic testing of the ductile mating rings indicated performance similar to that of traditional WC mating rings. The ductile mating rings were subjected to numerous starts and stops at both high speed/high pressure and high speed/low pressure conditions. Leakage's were within acceptable limits and the ductile mating rings showed no indication of significant face deterioration other than light scuffing.

One nitrided NACE condition AISI 4140 mating ring was installed in an operational gas compressor in February 1992. The nitrided mating ring replaced the high pressure (first stage) WC mating ring of a traditional gas seal. As shown in Table 1, the dry seal operated for 5,200 hr before removal due to excessive leakage of the low pressure (second stage) seal. The seal capsule was found to be heavily contaminated with process condensate and bearing oil. The AISI 4140 mating ring showed no significant wear or deterioration. The mating ring was cleaned and reinstalled in the seal capsule.

Table 1. Field Performance of Nitrided AISI 4140 Mating Ring.

	CONDITIONS	SUCCESSFUL STARTS	HOURS
Field Installation #1	7,130 kPa. 14,900 rpm	30	5,200
Field Installation #2	8,550 kPa. 14,000 rpm	22	2,050
TOTAL		55	7,250

The seal capsule was subsequently installed in another operational compressor in March 1994. Of note was that the first stage WC mating ring of the seal removed from the compressor had suffered heat checking and was discarded. The newly installed seal with the AISI 4140 mating ring operated for an additional 2,050 hr. Subsequently, the seal was removed from operation due to loss of seal gas supply and a steadily dropping leakage rate. Severe process contamination of the first stage sealing area was observed on disassembly. The nitrided mating ring grooves were filled with contamination. This is believed to have caused the low seal leakage. A problem was identified with the seal gas supply which, in turn, allowed migration of dirty process gas into the seal. It is the pipeline company's experience that a WC mating ring subjected to the same operating conditions would heat check. The ductile mating ring was removed from the seal and showed no visible signs of excessive wear. The ring was subsequently cleaned, successfully retested, and is intended for service at the next available opportunity. The successful field application of ductile mating rings, as described above, provided confidence to continue development of the single rotor design seal.

SINGLE ROTOR DESIGN SEAL CONCEPT

The "single rotor design seal" concept is based on the development of a single stage seal comprised of a one piece mating rotor fabricated from ductile ferrous material. This concept reduces at least six rotating components of relatively simple geometry found in a traditional single stage seal assembly to one component of complex geometry called a mating rotor. A typical single rotor design seal assembly is compared to a typical traditional single stage seal assembly in Figure 2. Consolidated components include the mating ring, drive sleeve, spacer or retaining sleeve, multiple drive pins, flexible centering mechanism at the inner diameter of the mating ring, and an O-ring at the rear face of the mating ring. It was determined that introduction of ductile seal face materials and elimination of the O-ring behind the mating ring would result in a less complex, and potentially more forgiving and reliable seal.

Traditional high pressure dry gas seals utilize stationary (primary) rings and rotating (mating) rings, of simple symmetric

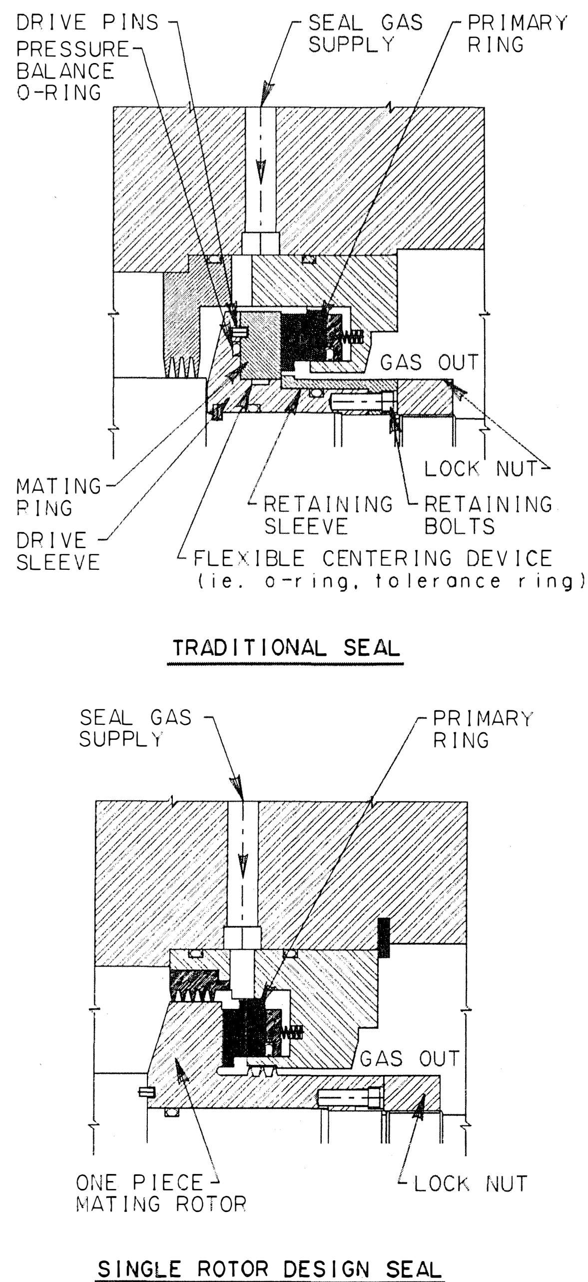


Figure 2. Single Stage Comparison: Traditional Vs *Single Rotor Design.*

geometry, which maintain a lubricating film (gap) between their respective sealing faces by means of hydrostatic and hydrodynamic (pumping action of shallow grooves) fluid forces. These seal designs attempt to minimize deflections or tilt of the mating ring sealing face. Mating seal ring tilt can occur due to pressure, centrifugal loading, and thermal deformation. Geometric symmetry and application of pressure balancing techniques about the mating ring ensure the seal face remains essentially perpendicular to the axis of the shaft and fundamentally parallel to the stationary sealing face (with slight convergence for stability) throughout all operating conditions. A nonsymmetrical mating ring would cause the sealing face to tilt in a diverging or converging manner due to centrifugal loading, as speed is varied, which could result in unstable operation and/or seal face touchdown. The O-ring at the rear face of the mating ring provides a seal face alignment feature that balances the pressure distribution about the face throughout the operating range. This prevents seal face distortion or tilting from an unbalanced pressure distribution. Gross quasistatic mating ring tilt could occur without implementation of these mating ring seal face alignment features. The single rotor design developed in this study compensates for gross mating ring tilt in converging and diverging directions resulting from elimination of these alignment features.

An advancement in dry seal technology has been made through realization that geometry and material selection of the seal components (rotating and stationary) can be engineered to account for the missing pressure balance and seal face rotational alignment features. The lack of these alignment features cause the mating rotor in a single rotor design seal to tilt significantly throughout its operating range. The mating rotor tilt is accommodated by precisely engineering the flexibility of the primary ring. The primary ring is bound by a precise combination of moment of inertia (I), centroid location, and Young's modulus (E) to provide the flexibility for the primary ring sealing face to follow the tilting sealing face of the mating rotor. The flexibility's of both the primary ring and mating rotor are designed to provide the optimum coning angle and optimum hydrostatic and hydrodynamic pressure distributions between the rotating and stationary sealing faces. This ensures stable seal performance throughout the seal's operating range. The combined flexibility of the primary ring and mating rotor can be tuned to accommodate seal designs that range from pure hydrostatic operation through full hydrodynamic operation. An appropriate groove can be designed and introduced into one of the sealing faces to provide the desired hydrodynamic component. Ductile ferrous or nonferrous materials are currently the most practical choice for the mating rotor as they can be machined in complicated shapes and are not prone to catastrophic failure like brittle materials.

POTENTIAL BENEFITS OF THE SINGLE ROTOR DESIGN SEAL CONCEPT

Advantages of a "single rotor design seal" design include:

• Elimination of the requirement for the redundant seal (second stage) through use of ductile materials.

• Reduced capital and maintenance costs due to fewer seal components per sealing stage.

• Improved seal reliability by reducing the design complexity and number of elastomers within the seal capsule.

• Increased safety of operation by eliminating the risk of catastrophic seal failure.

• Increased available axial space so that more effective seal gas supply and bearing cavity barrier seals may be implemented.

• Enhanced field retrofit feasibility resulting from reduced axial envelope design requirements.

• Improved compressor shaft rotordynamics by reducing the rotating seal mass.

Dry gas seal performance is measured by several variables including reliability, durability, cost effective operation, simplicity of operation, and user confidence. The single rotor design seal potentially delivers tangible and intangible benefits in all areas.

Tangible benefits lie in the potential for reduced seal purchase costs along with reduced operation and maintenance costs. By nature of the single rotor design, a reduction in manufacturing costs can be realized through the utilization of conventional materials, reduction of components, and reduction of associated quality assurance costs. For example, costs associated with over speed testing, an industry standard for rotating WC mating rings, are eliminated.

Potential reduction of operating and maintenance (repair) costs associated with the single rotor design seal concept is significant. Based on ductile mating ring performance to date, a cost benefit is anticipated from increased durability and life expectancy and by eliminating the requirement to replace heat checked WC seal rings. The number of critical O-rings (located on the opposite side of the mating ring and primary ring sealing faces) has been reduced by 75 percent. Inventory carrying costs could be reduced by up to one half as the number of components in a single rotor design seal is less than one half the number of components found in a traditional tandem seal.

Overhaul and repair of traditional gas seals is labor intensive due to a high quantity of components with tight assembly tolerances and fits. Quality assurance procedures and documentation required to ensure proper workmanship is rigorous. Labor requirements for repair, overhaul, and assembly of the single rotor design seal are estimated to be 50 percent less than for traditional tandem dry seals.

Intangible benefits, namely, increased reliability and safety, are expected through utilization of the single rotor design seal. Fewer components and less build complexity translate to fewer potential problems during operation and repair. Elimination of the risk of catastrophic brittle mating ring failure and peripheral damage will provide safer, more reliable operation with less unscheduled down time.

Additional axial space, available through elimination of the second sealing stage, allows implementation of a more effective seal gas supply and bearing cavity barrier seals. Improved barrier sealing will reduce seal contamination, a significant cause of compressor downtime. Conversely, the reduced axial space requirement of the single rotor design seal could make compressor dry seal retrofit economically feasible by eliminating major compressor modifications, often required to accommodate traditional tandem dry gas seals. More reliable products of simpler design will contribute to increased user confidence and acceptance of the technology.

PROTOTYPE DEVELOPMENT

The single rotor design seal development program has progressed through several phases including: a mating seal face material investigation; development of pseudo seal prototypes to emulate a single rotor design; development of a laboratory single rotor seal prototype; and field application of a prototype single rotor design seal.

The first phase of the development program concentrated on investigating the application and performance of ductile ferrous mating ring materials subjected to high pressure natural gas service. Continuation of this work has been discussed herein.

Development of single rotor design seal laboratory prototypes was discussed by Evenson, Mason, and St. Onge in 1994 [9]. Continuation of this development work has focused on design and field application of a prototype single rotor design seal and its control system.

PROTOTYPE SINGLE ROTOR DESIGN SEAL

Previous materials and secondary seal (labyrinth) work led to a prototype single rotor seal design intended for field application. A schematic of the field prototype is shown in Figure 3. The single stage seal comprises a one piece ductile mating rotor opposite a carbon graphite primary ring mounted in a housing. The assembled housing is shown in Figure 4. Labyrinth barrier seals are located at the process side and bearing cavity side of the housing. The mating rotor, shown in Figures 5 and 6, was fabricated from UNS S17400 (17-4PH) material, heat treated and surface hardened (nitrided) to H 1150-M condition per NACE MR0175-88 for sulfide stress-cracking resistant metallic materials subjected to sour gas environment. Aluminum seal gas supply labyrinths are located toward the process gas side of the seal face and run opposite nitrided surfaces of the one piece rotor. The labyrinth seal located on the mating rotor runs against an aluminum sleeve that is press fit in the housing. Seal gas leakage is vented through porting with an effective area of approximately 0.45 in^2. The rotating 17-4PH barrier labyrinth (Figure 7) that is mounted on the mating rotor, runs opposite a babbitted aluminum sleeve with an initial design clearance of 0.002 in. This seal was subjected to numerous laboratory tests to define the requirements for a control system to manage the seal in a field compressor.

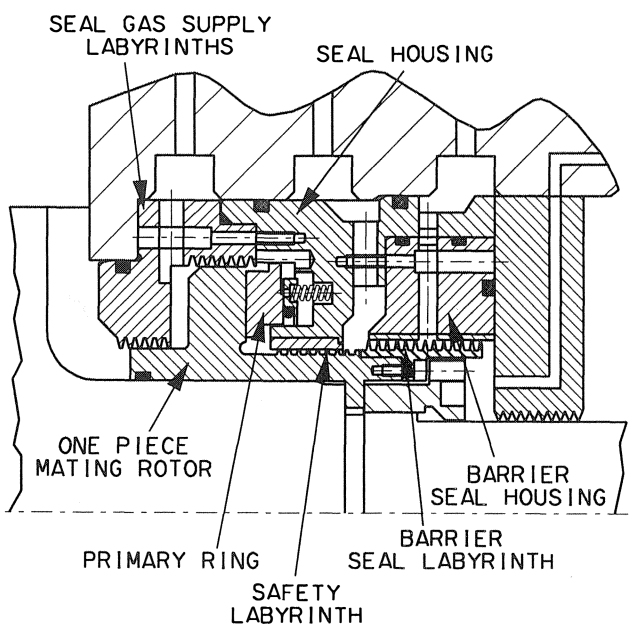


Figure 3. Prototype Single Rotor Design Schematic.

CONTROL SYSTEM DESIGN

One focus in this study was to develop a barrier seal system that prevents high pressure process gas from entering the compressor bearing cavity during both normal operation and a worst case seal failure. Development of a control system for the single rotor design seal is dependent upon performance of the seal gas supply labyrinths, the safety labyrinth, and the barrier seal system. It was determined through laboratory testing that a

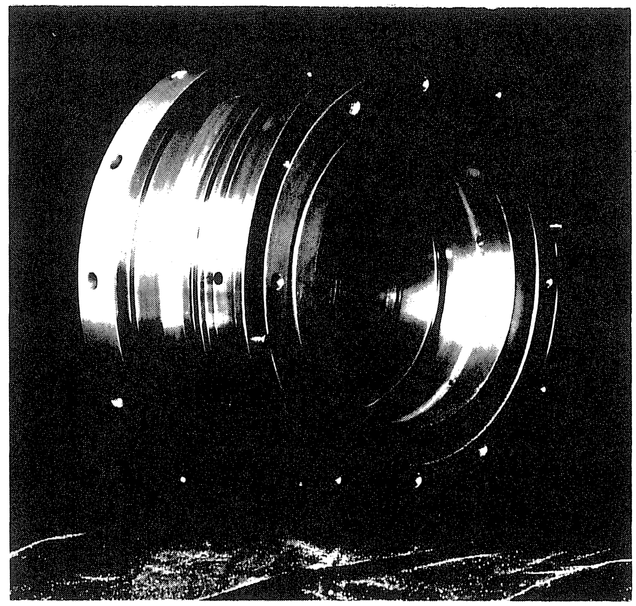


Figure 4. Prototype Seal Housing Assembly.

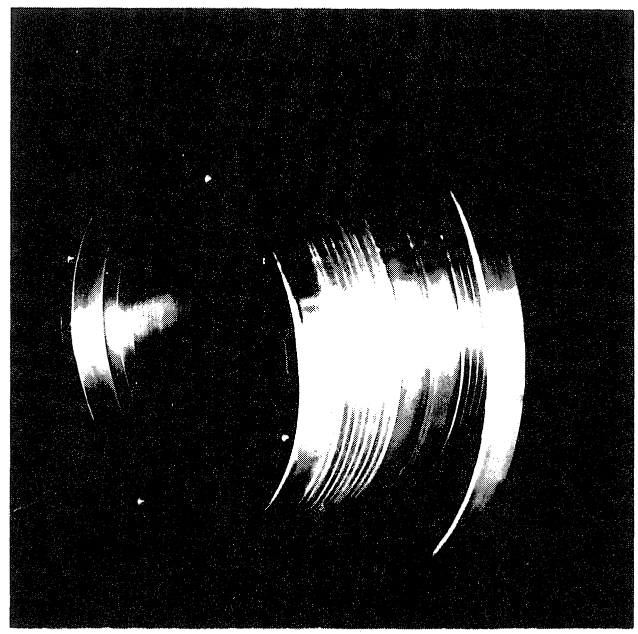


Figure 5. Prototype Mating Rotor—View of Specular Sealing Face.

labyrinth seal system offered the optimum combination of design features and performance for the barrier seal application.

Laboratory Testing: Barrier Labyrinth Seal—Normal Operating Conditions

Dynamic tests were performed to determine the requirements of the barrier supply gas that will ensure containment of seal leakage ranging from normal to a defined shutdown level. Based on normal high pressure seal leakage of approximately 1.0 standard cubic foot per minute (scfm), standard design practices determined that an additional 2.0 scfm and 5.0 scfm seal leakage above normal would constitute alarm and shutdown conditions respectively. Using the aforementioned criteria, alarm and shutdown conditions were simulated in testing by adding "auxiliary"

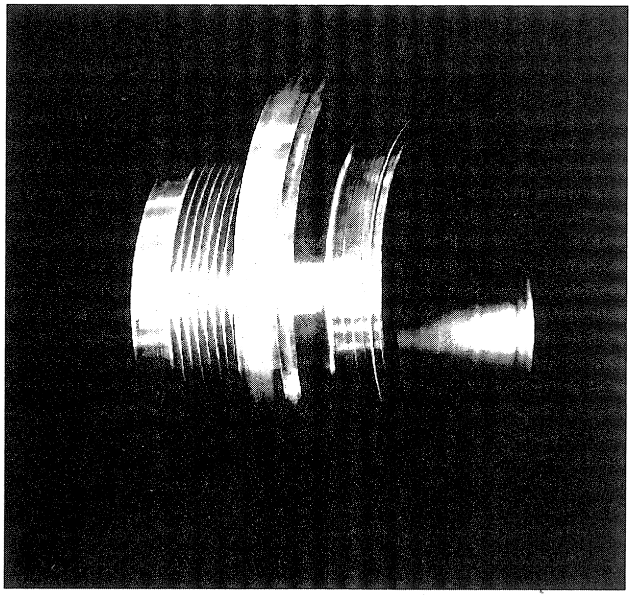
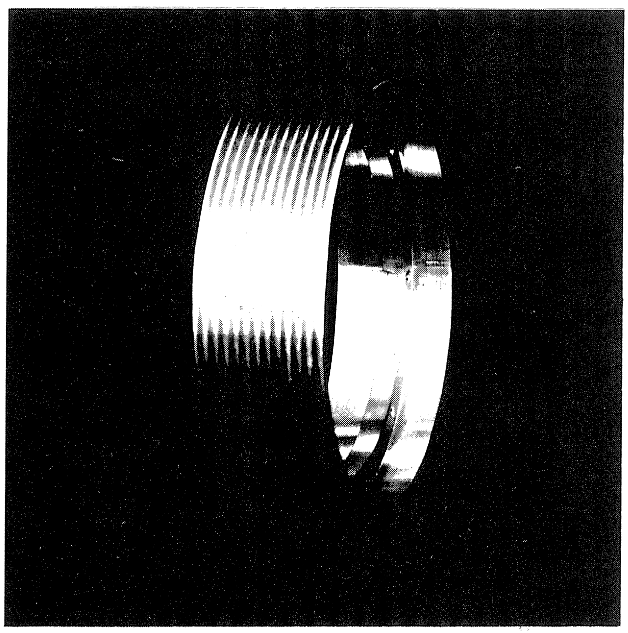


Figure 6. Prototype Mating Rotor—View of Rear Shoulder.



air to the seal leakage at various barrier air supply conditions. The test rig used to establish alarm and shutdown conditions for the field application is shown in Figure 8. Although a range of leakage orifice sizes were tested, larger diameters created lower back pressures in the seal leakage vent, reducing pressure and flow requirements of the barrier air supply. Careful attention was given to the exit flow areas for seal leakage and the bearing cavity vent to emulate the flow areas available for an actual field application. The test rig setup used to evaluate leakage port area is shown in Figure 9. Pressure and flow results from this testing are illustrated in Figures 10 and 11, respectively.

Laboratory Testing: Combined Labyrinths—Blowout Condition

One of the key system design requirements was to determine the resultant pressure that would be created in the seal leakage

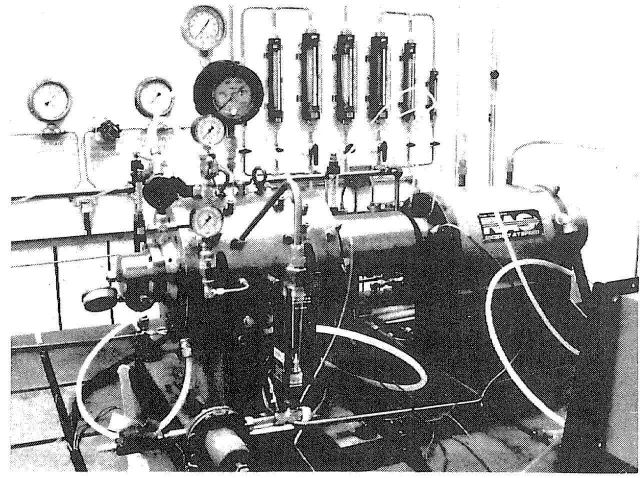


Figure 8. Seal Test Rig; Showing Additional Flowmeter Used to Establish Alarm and Shutdown Set Points.

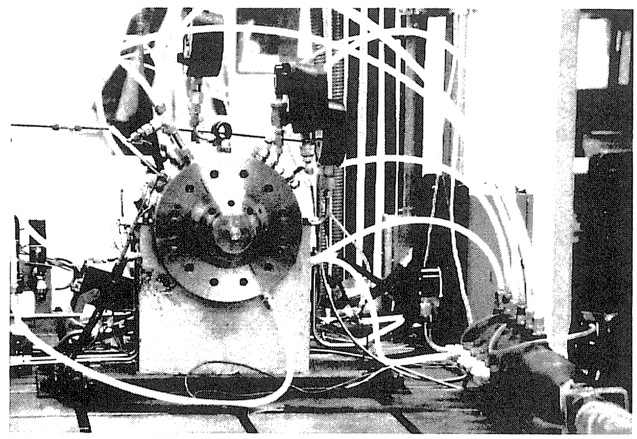


Figure 9. Seal Test Rig; Set up to Establish Seal Leakage Port Sizing.

cavity during a worst case seal failure. This worst case scenario was defined as complete destruction of the carbon graphite primary ring (total blowout) subjecting the seal supply cavity to full process pressure. Given the ductile nature of the material, the nitrided steel mating ring would remain substantially intact except with some degradation of its polished face. The total blowout condition was simulated in the test rig by complete removal of the primary ring in conjunction with application of 1,200 lb/in^2 gage (psig) to the seal supply port. As the test rig was originally designed for high pressure/low flow applications, a maximum resultant air pressure of 115 psig at the test rig seal supply cavity was attained. Subsequent curve fitting and extrapolation (Figure 12) predicted a maximum leakage pressure of 108 psig at the seal leakage port for a process gas supply pressure of 1,300 psig.

Momentary existence of 108 psig gas pressure at the seal leakage port during compressor emergency shutdown mode establishes a requirement for the barrier seal to resist this pressure and ensure process gas does not enter the compressor bearing cavity. Furthermore, the effective vent area downstream was evaluated to ensure that sufficient flow area is available in the actual compressor cavity porting and vent piping connections. This is to ensure that during similar seal failure conditions a maximum of 108 psig would exist in the actual field installation.

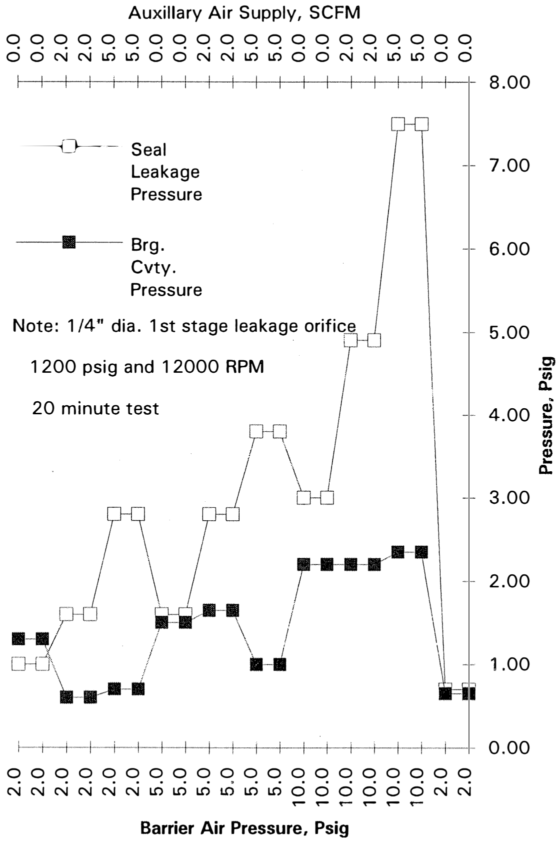


Figure 10. Measured Pressure Data Used to Establish Seal Leakage Alarm and Shutdown Set Points.

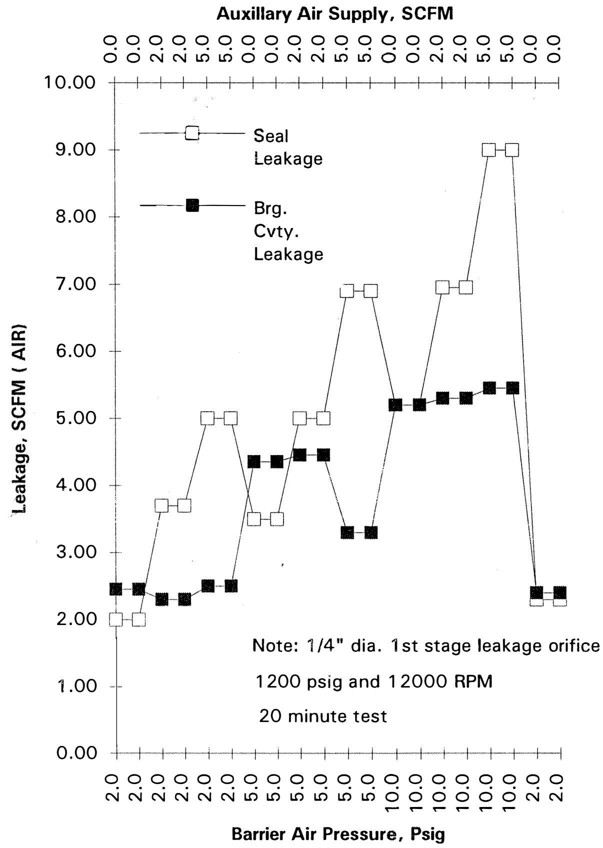


Figure 11. Measured Leakage Data Used to Establish Seal Leakage Alarm and Shutdown Set Points.

Field Application System Design

All possible locations for field installation of the single rotor design seal use a seal leakage monitoring system which measures the back pressure created by a known size of fixed orifice in the seal leakage vent line. For reasons of interface simplicity and proven past performance of these systems, a similar flow monitoring system was adopted in the actual single rotor design seal application. The design of the control system was divided into two portions; one which focused on primary leakage containment during a unit emergency shutdown (ESD) as a result of seal failure, the other for seal leakage containment during normal operation up to ESD conditions.

Design of the ESD system to control worst case seal failure was based upon test results shown in Figure 12. A design pressure of 108 psig is required to be resisted at the seal leakage port. The actual field conditions would require the single rotor design seal to resist only a transient peak pressure of 1,300 psig during a severe process upset condition. Given that the compressor would be sequencing through the ESD procedure prior to the vent valve opening, the casing would be basically at suction pressure that is substantially lower than 1,300 psig. This allows standard plant air pressure of 115 psig to be utilized as a barrier fluid to prevent process gas from entering the compressor bearing cavity. A schematic of the single rotor design seal and its control system is shown in Figure 13. Application of the ESD air to the barrier seal is controlled by a solenoid valve. The solenoid valve operation is controlled by a pressure switch (PSHH) that monitors seal leakage and initiates a compressor ESD in the event of excessive leakage pressure and associated excessive flow. Piping loss calculations were performed to ensure the system's ability to meet the flow and pressure design requirements of the barrier labyrinth system.

The normal operating system's function is to provide adequate barrier air supply for all operating conditions except that of a compressor ESD. Control of this air supply source is provided by a solenoid which is activated by bearing lubrication oil pressure. The supply pressure to the barrier seal is determined by a preset pressure regulator. A portion of this supply will augment the seal leakage and contribute to the pressure sensed by the alarm and shutdown instruments. The air supply flow is monitored by a local flow meter. Pressure at seal leakage and air supply cavities is measured with pressure gauges to ensure correct control system operation. The oil trap connected to port 'C' (Figure 13) was used in the previous tandem seal

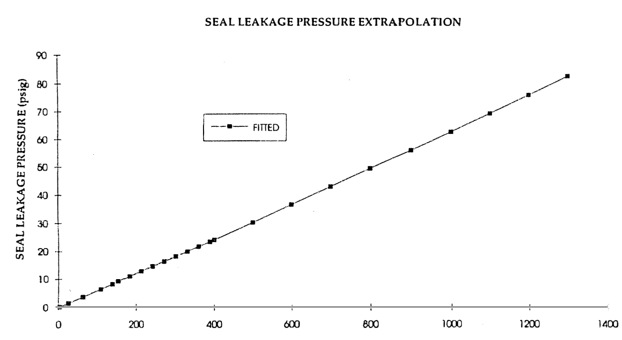


Figure 12. Fitted Seal Leakage at Predicted Seal Supply Pressure for 0.4452 In^2 Port Area.

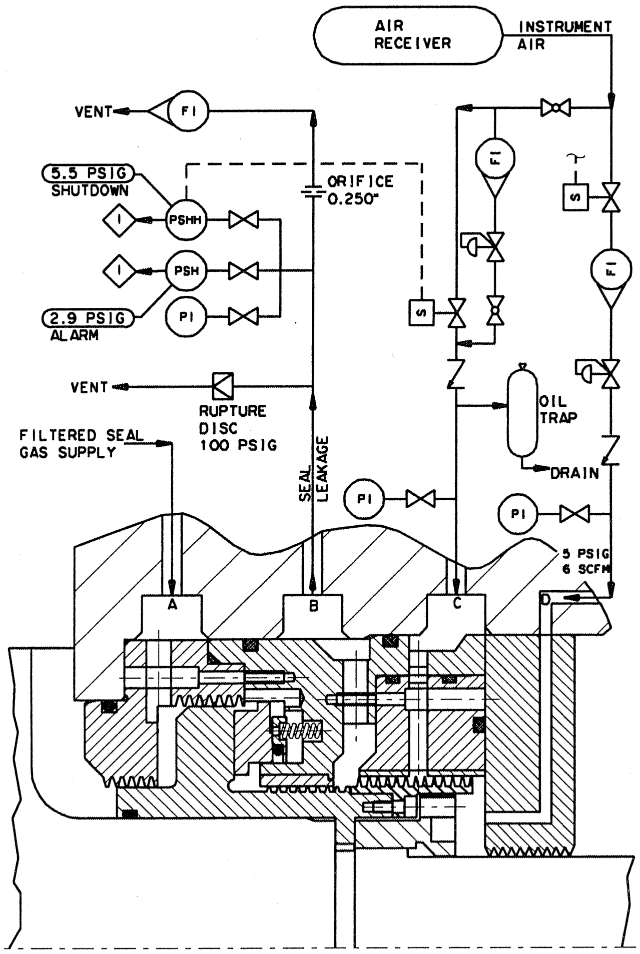


Figure 13. Prototype Single Rotor Design Field Application Control System.

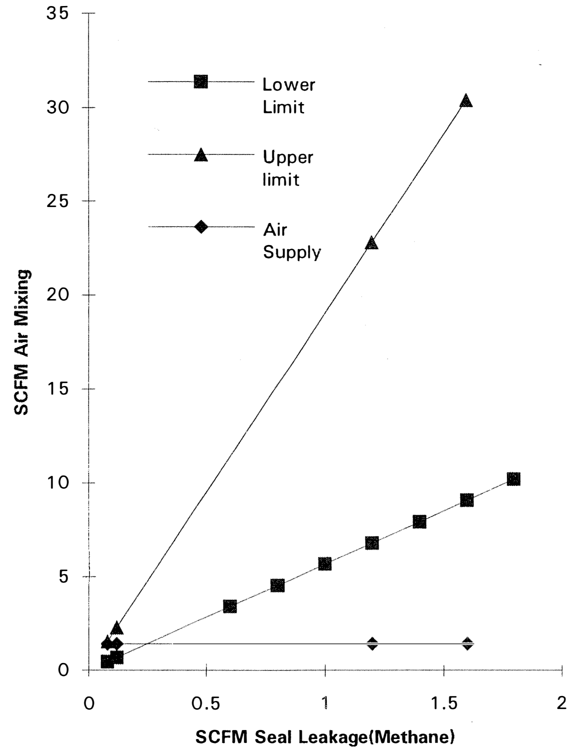


Figure 14. Field Application Flammability Characteristics.

installation and remains installed to facilitate drainage of any bearing oil that enters that area. From the laboratory test results (Figure 10), a barrier air supply of 5.0 psig at port 'D' is required to contain the process gas seal leakage in the vent up to (ESD) shutdown conditions. For the 5.0 psig supply pressure, the corresponding alarm and shutdown set points are established at 2.9 psig and 5.0 psig respectively with a predicted normal seal leakage pressure of 1.7 psig.

As an alternative to maintain flexibility in the field application, an auxiliary system to port 'C' for normal operation was set up in parallel to the ESD system.

Since air is the chosen media to supply the barrier seal which augments and mixes with seal leakage in this application, a discussion regarding combustible gas mixtures in the seal leakage cavity and conduit is appropriate. Calculations [10] that determined the flammability range were performed; the results are illustrated in Figure 14. The anticipated mixture during normal conditions is well below the lower limit; at 1.0 scfm seal leakage of natural gas it is calculated that 1.5 scfm of air will place the mixture in the nonflammable region. If the seal leakage increased, it would further increase the already methane rich mixture. In addition, the increase in seal leakage pressure will decrease the flowrate of air from the barrier seal due to a reduction of pressure differential across the barrier seal. If the seal experiences a complete failure as simulated by the "full blow out" testing condition, the ESD air supply (115 psig) is only slightly higher than the anticipated leakage pressure of 108 psig. The resulting barrier air flow to seal leakage will again be low resulting in a nonflammable mixture. This mixture will essentially be maintained until the compressor case depressurizes at some point during which the flammable mixture zone will be crossed. Since the unit is nonrotating state during this crossing, there is no potential source of ignition from churning of possible seal debris. In applications that involve sealing gas of larger flammability ranges for example hydrogen, it is suggested to use an inert barrier supply gas.

SINGLE ROTOR DESIGN SEAL PROTOTYPE FIELD INSTALLATION

The single rotor design seal and control system, previously described, was retrofit (installed) in a Solar C-304 high pressure natural gas pipeline compressor and placed in service in early December 1994.

Upon installation, seal leakage pressure alarm and shutdown set points were set to design specifications of 2.9 psig and 5.0 psig, respectively. The set points have remained unchanged from the design specification to date. Seal operating data for the first 1,400 hours of operation is shown in Figure 15. Seal leakage pressure has remained at approximately 1.0 psig with approximately 2.8 scfm of combined seal and barrier air leakage. Seal leakage pressure and flow have remained relatively stable throughout this time period and have shown no degradation of performance to date. Suction and discharge pressures averaged 635 psig and 960 psig, respectively. Average compressor speed was 14,400 rpm.

Following approximately 2,100 seal operating hours, the compressor unit was shut down for scheduled maintenance. The seal was removed for disassembly and inspection. Prior to inspection, the seal was subjected to eleven nonseal related shutdowns and starts with no evidence of degradation of seal performance. Surviving compressor shutdowns and subsequent

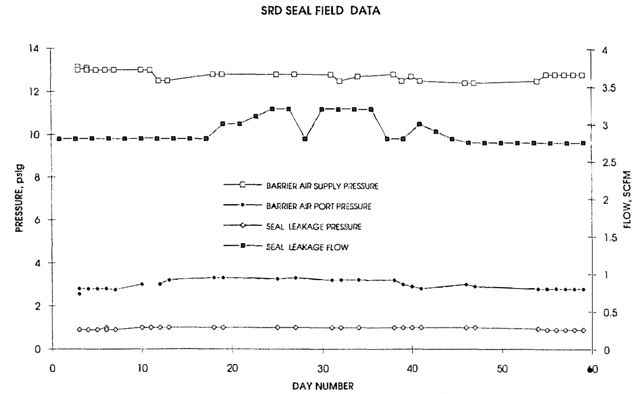


Figure 15. Field Application—Operating Pressure and Flow Data.

startups over time is an accepted measure of successful seal performance.

Inspection of the seal showed no evidence of oil contamination or debris at either the process or barrier air (oil bearing) side of the seal. The mating rotor and primary seal faces showed no sign of touchdown. The carbon graphite primary ring exhibited no color when subjected to a white glove test. This suggests excellent hydrostatic and hydrodynamic liftoff characteristics throughout the compressor's operating range. The barrier seal sleeve (Figure 16) shows the grooves cut by the rotating labyrinth teeth during compressor operation. The clarity and uniformity of the labyrinth grooves illustrate the durability and reliability of the barrier labyrinth system and its ability to successfully accommodate normal shaft excursions with little effect to labyrinth seal performance. The single rotor design seal was reassembled with existing components, including O-rings, and reinstalled in the compressor. The compressor and seal continue to operate successfully (with ongoing exposure to nonseal related shutdowns on a random basis).

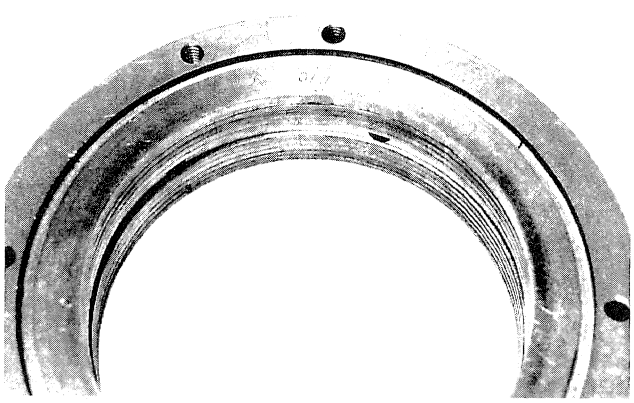


Figure 16. Barrier Seal Sleeve Showing Grooves Cut in Babbitted Surface.

Successful field performance of the single rotor design seal over time will result in removal of the redundant barrier air control system (loop), used in this application to account for existing compressor labyrinth performance uncertainties. Evolution to a control system based on combined gas/air mixture analysis will also simplify the control system. Nitrogen can be utilized as a barrier fluid in field applications that are not able to use standard plant air.

CONCLUSION

Development of a high pressure single rotor design seal has been described. The mating ring, drive sleeve, retaining sleeve, multiple drive pins, centering device at the inner diameter of the mating ring, and the O-ring at the rear face of the mating ring found in a traditional seal assembly were consolidated into one component of complex geometry called a mating rotor. Previously proven ductile ferrous materials were applied to the one piece mating rotor and successfully tested with a carbon graphite primary seal face designed to follow the tilting mating rotor seal face through its operating range.

The single rotor design seal can be applied in single or multiple stages as required. A single stage seal used in conjunction with an appropriate secondary or barrier seal can be designed to contain worst case leakage situations with increased confidence knowing that the ductile mating ring will not catastrophically rupture. A barrier seal may be composed of segmented carbons, labyrinths or other sealing devices.

Potential benefits of single rotor design seal technology applied to high pressure natural gas compressors include: increased safety by eliminating the potential for catastrophic mating ring failure through the use of ductile materials; fewer components including a reduced number of O-rings to decrease the possibility of explosive decompression and reduced risk of O-ring damage during assembly; improved rotordynamics on flexible, high speed applications due to lighter weight materials; less fabrication and quality assurance, and economic benefits from potentially lower purchase, operation, and inventory costs.

ACKNOWLEDGMENTS

The author's would like to acknowledge Nova Gas Transmission Ltd., Revolve Technologies Inc., and the Canada-Alberta Partnership Agreement Program for support of this development project.

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