

DESIGN AND MITIGATION TECHNIQUES FOR APPLICATIONS WITH POSSIBLE LIQUID CONTAMINATION OF THE SEALING GAS DRY GAS SEAL SYSTEM

by

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ABSTRACT

A compressor in a natural gas gathering service experienced multiple seal failures on the discharge end. Synthetic oil mist contained in the sealing gas was identified as a significant source of contamination that caused the seal failures. Since similar dry gas seals in other compressors at the same location were operating satisfactorily, a study of a particular compressor was undertaken. The seals were instrumented with thermocouples to monitor the temperature distribution across the seals.

This paper discusses the findings of the study, resulting modifications of the dry gas seals, seal controls, and the compressor itself. This paper also outlines techniques that have been developed to mitigate the situation without having to shut the unit down. Also presented is a design approach to the sealing and seal monitoring in processes where liquid ingress into the sealing area may occur, including seal material selection, control system philosophy, seal head, and bearing cavity design.

INTRODUCTION

An oil and gas production company purchased a centrifugal compressor to increase the gas sales volume. Their chosen configuration consisted of an electrical motor driven centrifugal booster, and special attention was given to the design of the compressor's dry gas seal system (Figure 1). The system control philosophy was based on remote diagnostics of all critical and noncritical seal system operating parameters. The system had also been designed to prevent process gas release into the atmosphere in case of a catastrophic seal failure.

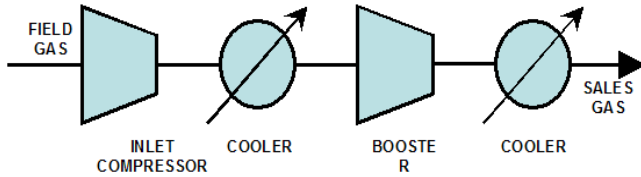


Figure 1. Facility Flow Diagram.

SYSTEM DESIGN

The control system design (refer to APPENDIX A) incorporated two stages of filtration: coalescing and particulate. The particulate filters (AJ117A/B) were placed downstream of coalescing (AJ117A/B) to prevent seal contamination by fibrous debris in case of a coalescing element rupture. The sealing gas supply (PDV-120) was based on the differential pressure control principle (PDY-120), with balance piston cavity pressure used as a reference. Since the pressure control principle has been utilized for primary seal supply and not flow control, each compressor end seal supply flow was monitored by flow transmitters (FIT-120/125) to guarantee adequate supply volume.

Traditionally dry gas seal performance is evaluated based on its leakage. Since in this application primary vent flow consisted of roughly the sum of primary seal leakage and secondary seal supply flows, in order to enable efficient leakage monitoring and prevent primary seal leakage dilution, the secondary seal supply employed flow control (FY-160/166). The primary and secondary leakage flows (FIT-150/151/155/156) and corresponding leakage cavity pressures (PIT-150/151/155/156) were monitored to provide indication of seal health. All critical and noncritical operating parameters were monitored and controlled by a discrete control system.

The selected dry gas seal design incorporated a hard silicon carbide (SiC) rotating face running against a soft carbon stationary face. The interfaces between the seal and compressor head and the shaft were sealed with fluorocarbon O-rings, while spring energized polymers served as the seal's internal secondary sealing elements. A circumferential carbon ring pressure bushing was used to separate the dry gas seal from the bearing cavity. The seals were installed into a separate seal head, which in turn, was installed into the compressor body. The supply and leakage porting was cross-drilled and piping connections were located on a circle terminating at flanges.

Special attention was given to prevention of uncontrolled gas release into the atmosphere in case of a seal failure. It was determined that, in case of a catastrophic primary seal failure, in order to depressurize primary seal leakage cavity and redirect gas flow from the primary seal leakage chamber into the flare system, an additional primary seal venting area was required. To accommodate the requirement, two vent ports were incorporated into the primary seal leakage annulus. Also, to increase the venting area, control system secondary seal supply piping was transformed into vent piping by two sets of pneumatically actuated ball valves. In case of a trip on high high flow, one set of valves (SDV-160/166) would close, isolating the supply system upstream, and the other set (BDV-161/167) would open, connecting the secondary seal gas supply and, thus, seal the primary leakage cavity to the vent system. The secondary seal cavity depressurization scheme was

developed based on similar calculations. By design, the system in case of a catastrophic seal failure of a primary and/or secondary seal would prevent uncontrolled gas release into atmosphere.

OPERATING BACKGROUND

The compressor was successfully commissioned and started. After a month of operation, a seal failure occurred. The compressor's drive end seal failed catastrophically. Debris originating from the seal's SiC rotating face damaged the seal head cavity during the failure and scored the lands during seal removal (Figure 2). Prior to the failure, operating personnel had noticed a decrease in barrier seal supply flow. At factory disassembly, the nondrive-end seal exhibited signs of contamination by both liquid and particulate. Upon disassembly of the barrier seal, it was noticed that seal O-rings were thermally hardened. The degree to which the O-rings hardened was depending on O-ring location: the farther the location was from the bearing cavity, the lesser was the degree of O-ring hardening. Although this fact had been noticed, no related action was taken at that time.



Figure 2. Head Damage at the O-Ring Location Resulting from Rotating SiC Ring Catastrophic Failure.

The compressor head was repaired, a new seal installed, and the unit was restarted. In time, the flow to the barrier seal on the drive end progressively declined and eventually was reduced to a few liters per minute. At a later date, the unit was shut down to evaluate the barrier seal condition. The pattern of O-ring hardening persisted. During the shutdown, a small volume of liquid was found in the seal gas particulate filters located downstream of the coalescing ones (Figure 3).



Figure 3. Liquid Accumulation in Particulate Filter Housing.

An analysis identified the liquid as a mixture of water and triethylene glycol (TEG). The gas used for the sealing supply was provided to the coalescor at 100°F. To filter the mist contained in the gas, it would have been necessary to increase the gas temperature by incorporation of a heater into the stream. But the panel valve temperature rating limited the supply gas temperature increase and prevented additional gas heating. Various modifications to the existing coalescing filter elements were undertaken. The modifications eliminated liquid accumulations in the filter bowl itself, but liquid continued to exit the piping drain valve downstream of the filter. The existing coalescor was replaced by a high efficiency coalescing filter element. The liquid coalesced in the new filter chamber was identified as a mixture of polyalkylene glycol (PAG), TEG, and water. The PAG was used as lubrication/sealing oil in upstream screw compressors. The second seal failure occurred in a manner similar to the first one a year later (Figure 4).



Figure 4. Catastrophic Seal Failure—SiC Fragments that Damage the Seal Head.

It is widely accepted that in order to maintain reliable operation, a dry gas seal requires clean and liquid-free gas. In reality, a seal is tolerant, to a certain degree, of liquid contamination by vaporizable liquids, while reacting poorly to nonvaporizable liquids. Incompressible fluid ingress onto the seal faces produces two detrimental effects: sealing film instability and heat generation. In steady-state compressor operation, if pressure differential across the compressor does not suddenly change, the rotor usually does not experience rapid axial movements (Figure 5). Thus, if a non vaporizable liquid ingress into the seal has occurred at a steady-state compressor operation, heat generation in liquid shear becomes more of a concern than sealing film instability. Generated heat, if not dissipated, leads to a thermal expansion of the rotating parts of the seal and generation of high stresses. The typical mode of such a failure would be a fracture of a rotating sealing face followed by its disintegration due to consequential impacts in rotation. Interestingly enough, the same site had other compressors with dry gas seals that have been successfully operating for years on the same sealing gas, supplied from a common header. Nevertheless, the dry gas seals in these units appeared to be tolerant of the sealing gas composition, operating up to and in excess of five years.



Figure 5. Mixture of PAG and TEG in the Seal Supply Annulus.

MODIFICATIONS

The issue of the compressor seal reliability can be approached from two directions: elimination of the liquid in the seal gas supply or creation of an operating environment for the seals that ensures that the liquid ingress impact is minimized. The first approach is always preferred because it eliminates the cause of the seal failures. Unfortunately, in many situations, such a source of clean gas does not exist and its creation is economically prohibitive. Since the other compressors at the site were operating satisfactorily and no other source of sealing gas existed at the site, a decision was made to study and modify the dry gas seal system. The goal was to reduce to a minimum concentration of the nonvaporizable liquid in the seal supply gas and to eliminate all additional sources of heat generation in and around the seal.

Dry Gas Seal Supply Gas Conditioning

A number of proprietary filter elements were tested to determine an optimum choice. While testing one, a two-phase liquid sample

was rejected by a coalescer that consisted of PAG and TEG/water. PAG, viscous synthetic oil, is used in flooded screw compressors in the facility for sealing and bearing lubrication. The increase in fluid viscosity adversely affects the coalescing ability of the filters. A filtration element vendor designed a special high-efficiency element (99.9998 percent at 0.1 micron aerosol) to enable reduction in the synthetic oil carryover. The element utilized the liquid's velocity and temperature (viscosity) to maximize its coalescing ability. The second filter in a duplex arrangement was repiped to operate in series with the first one (Figure 6). As a result, a maximum concentration of PAG and TEG equated to 40 ppmw and water concentration to 160 ppmw.

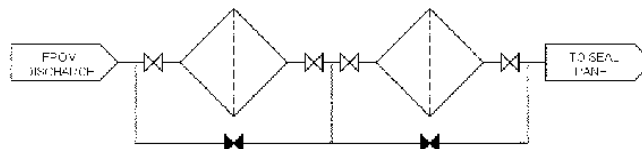


Figure 6. Prefilter Arrangement.

Compressor Seal Cavity

Investigation showed that the design of a cavity between the balance piston and dry gas seal provided a potential for liquid accumulation. The same applied to the primary seal supply and leakage annulus, as the correspondent ports were located in the upper quadrants. In case liquids were introduced into the primary seal, the accumulations would occur in the primary seal supply and leakage cavities annulus (Figure 7). To prevent the liquid accumulation, the cavities were outfitted with the drains (Figure 8).

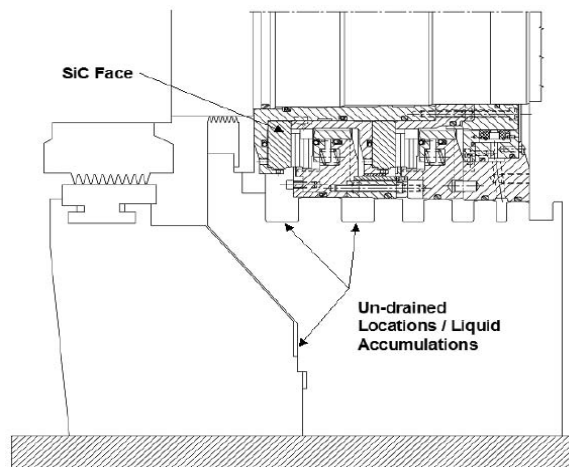


Figure 7. Seal and Seal Head Prior to Modifications.

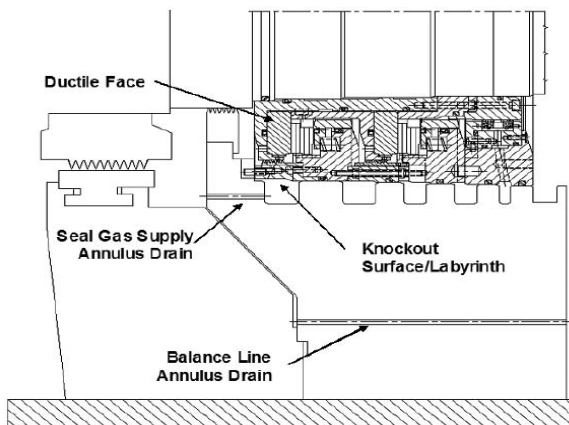


Figure 8. Seal and Seal Head after Modifications.

Dry Gas Seal Supply Path

Part of the study was to determine what differed with regard to the compressors operating successfully at the facility from the one experiencing seal failures. When a seal gas supply path was examined, it was noted that the sealing gas was delivered immediately to the sealing faces in the problem unit, while in the other units, it was first directed onto a solid surface acting as a knock-out plate and then injected between the seal and a process side labyrinth, thus allowing for oil knock-out with further drainage away from the faces. To modify the existing design, an additional process side labyrinth was introduced. Now, the sealing gas entering the cavity is injected on the outer diameter (OD) surface of the new process labyrinth, which acts as a knock-out (Figure 8). Separated liquid is drained back to suction through an introduced drain.

Dry Gas Seals

In both seal failures, the compressor seal cavity was damaged by shattered SiC rotating faces. To avoid the lengthy cavity repair process resulting from damage by SiC bits, the seals were retrofitted with ductile (stainless steel with tungsten carbide [WC] coating) rotating faces. Since heat generation was the reason for both seal failures, the drive-end (DE) dry gas seal and barrier seal were retrofitted with temperature monitoring devices.

Compressor Bearing Cavity

The issue with oil being found in the secondary vent line started shortly after the initial startup. The barrier seal's O-rings were found to be hardened. To eliminate all possible sources of heat generation, the barrier seal was redesigned to reduce the windage and eliminate the whipping action of protruding screw heads. A swirl brake was incorporated into the bearing cavity to facilitate oil drainage.

OBSERVATIONS

After the modifications were completed, thermocouple readings were taken twice a day and compared to the drain's content in various locations. The temperature measurements established that the primary seal temperature was not subject to a high rise. The temperature increase in the secondary seal was the highest and started occurring shortly after the startup. The barrier seal temperature closely followed the temperature of the secondary seal, though the rate of increase was not as high.

Since the liquid carryover was minute, the correlation between temperature rise and liquid presence in the filter drains and other drains proved to be complicated. Either there was no direct correlation or it was too difficult to discern the rise in the secondary seal temperature and the misting of the drains. Half a year after the startup, the secondary seal temperature was approaching 300°F with the barrier seal temperature trending high (Figure 9).

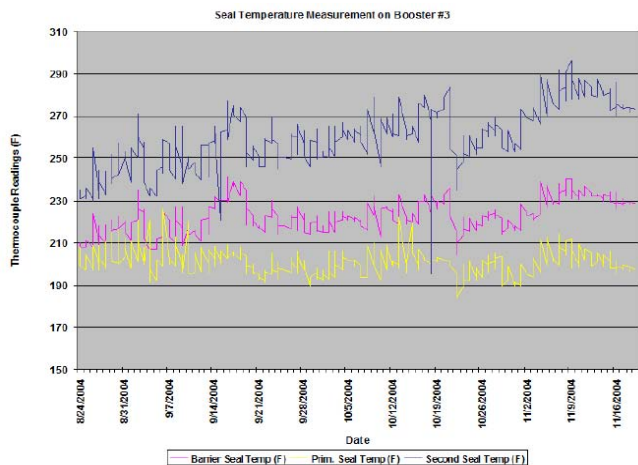


Figure 9. Seal Temperature Trends Prior to Mitigation.

ANALYSIS

The thermocouple readings validated that the secondary seal appears to be the dominant source of heat generation. The liquid ingress into the seal cavity was minimized, but some liquid continued to be accumulating on the secondary seal faces. By design, in order to conserve nitrogen and allow for an accurate reading of actual seal leakage, the intermediate labyrinth supply was flow controlled to 1 acfm (velocity of 5 to 7 fps).

The original design philosophy was based on an assumption that no liquids pass through the primary seal and, therefore, gas velocity of 5 to 7 fps would be sufficient to sweep clean leakage gas into the primary vent. But, this velocity was insufficient to prevent ingress of liquid trapped in the leakage gas into the secondary seal. Due to the fact that the primary leakage cavity pressure is small (around 1 to 2 psig), the secondary seal leakage is extremely low—a couple of NI/min. It is not sufficient to provide cooling and its flow is not sufficient to remove oil film off the secondary seal faces. In comparison, the primary seal was being cooled by more than 100 scfm of cool sealing gas and was operating at over 200 psig pressure. As long as the secondary seal operated at a low pressure with a small leakage, the above condition existed and liquids were not removed from the secondary seal faces, the amount of heat generated in this condition could not be completely dissipated. Essentially, the compressor's drive end seal head acted as a heat trap.

MITIGATION

There are very few ways of bringing heat generation under control in such a situation. The most easily attainable solutions include completely stopping the oil mist from depositing on the seal faces or to achieve an N2 flow to the intermediate labyrinth (secondary seal supply) that would be sufficient to remove the heat generated by the secondary seal faces. Such flow should remove the heat and prevent the sealing face fracture and/or create enough leakage flow through the faces to generate a velocity sufficient to remove a majority of the oil film of the faces or separate the faces.

To achieve a solution to the failure problem in this application, the flow to the barrier seal was increased in steps. Temperature measurements indicated no impact on the barrier seal cooling on the secondary seal temperature, confirming that the secondary seal was the source of the heat generation. Next, the flow to the intermediate labyrinth (secondary seal supply) was increased in steps, allowing time for temperatures to settle. After a flow of 6 scfm was achieved as a result of these stepped changes to the secondary seal supply, the secondary seal temperature dropped and remained stable at the 210 to 230°F level (Figure 10).

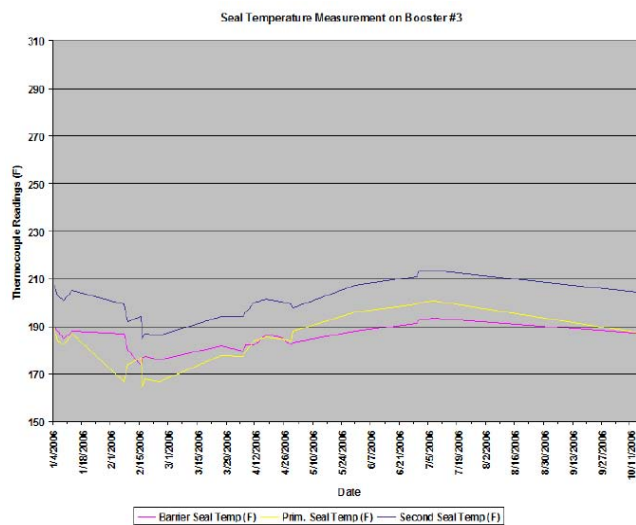


Figure 10. Seal Temperature Trends after Mitigation.

CONCLUSIONS

Although providing clean and liquid free seal supply gas remains a preferred option, many lessons could be learned from the study above. In order to enable a reliable operation of a centrifugal compressor in processes with a possibility of liquid ingress into the seal, a dry gas seal system and compressor design should incorporate the following:

- Dry gas seals should be outfitted with temperature monitoring instruments.
- Dry gas seal design, operating conditions permissive, should utilize ductile coated mating rings to prevent cavity damage in case of the seal failure.
- Sealing gas should be diverted from direct entry into the seal. A knock-out type entry should be provided in the seal gas supply cavity.
- All seal cavities and stagnant flow cavities on the process side of the seals should be equipped with drains located at the six o'clock position.
- Control systems should be designed to accommodate possible future increases in supply flow to both primary and secondary seals.
- Bearing cavity design should provide for a proper drainage of lubricating oil preventing pressure build up inside the cavity and untrapped oil accumulation.

