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Examination of Methods of Dry Gas Seal Supply, Regulation, and Monitoring

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ABSTRACT

Compressor Dry Gas Seal Failure is a common issue that affects Turbomachinery train availability. Since most turbomachinery trains are un-spared critical assets, unscheduled shutdowns have a major impact to the operating facility. As operators look to extend equipment run time, the quality and delivery of the primary seal gas supplied to the dry gas seal is vital to the long term operation. In case of a failure, automated trips can be applied to mitigate catastrophic failures but in many cases the machinery engineer needs to make an operations decision due to changing seal performance and needs to answer the following questions:

- Is gas delivered to the seal at required volume and quality at all operating conditions?
- How is the seal monitored?
- What is the data telling me?
- What is causing the abnormal indication?
- What action should be taken?

This tutorial examines the two main philosophies for delivering seal gas, comparing them with two calculated examples before describing a recommend primary vent monitoring setup and the associated instrumentation. The tutorial concludes with several examples based off real world failures to further illustrate the monitoring and diagnostic abilities of the dry gas seal system.

INTRODUCTION

Compressor shaft sealing technology has evolved over the last 35 plus years from oil seal systems to the now standard dry gas seal. These systems have been developed to operate within the range of conditions and services seen in our industry today. However, as the technology developed and applications grew, failures inevitably followed. In 2008 a study was done across a collection of user companies to identify the major issues experienced with compressor dry gas seals. The overwhelming response identified the condition and delivery of the seal gas as the primary cause of seal failures. This tutorial does not focus on the gas condition but the delivery and monitoring of the gas streams. This tutorial will first discuss the two main control philosophies for controlling seal gas: pressure control and flow control. Then the tutorial will examine common methods of monitoring seal health and diagnosing failures.

Single, tandem and double seals are used for various applications but tandem dry gas seals are the most common dry gas seal arrangement on the market. The tandem seal with an intermediate labyrinth can be applied to most all pressure ranges, process gas compositions and potentially eliminates atmospheric emission of process gasses when applied with a secondary seal gas. For simplicity, this tutorial will cover tandem seals with an intermediate labyrinth but the overall concepts can be applied to the other seal configurations.

Tandem Seal with Intermediate Labyrinth

Tandem seals consist of primary and secondary seals with an intermediate labyrinth located between the two. A typical cross-section of a tandem dry gas seal with an intermediate labyrinth identifying the major seal components is shown in figure 1.

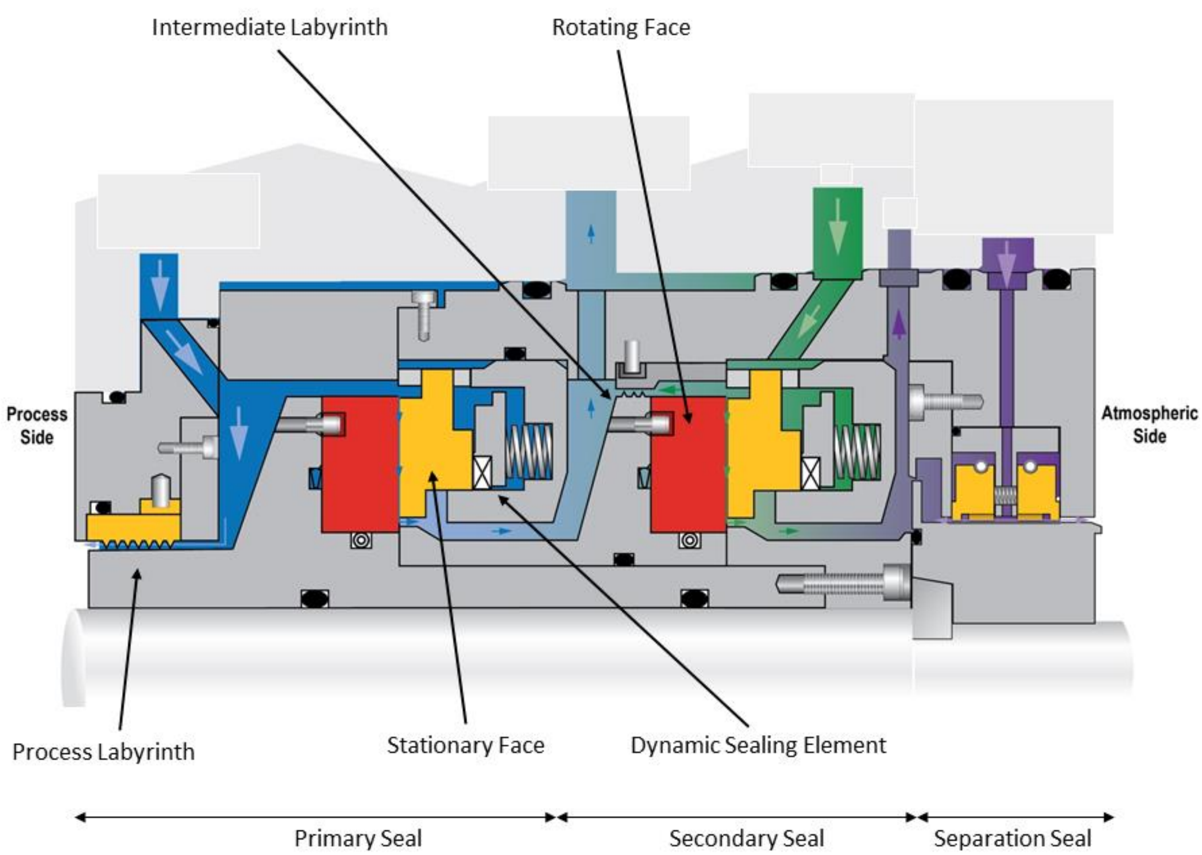

Figure 1 - Major Seal Components

The dry gas seal system can be broken down into five separate streams: seal gas supply, secondary seal gas supply, separation gas supply, primary vent, and secondary vent as shown in figure 2. Seal gas is introduced to the primary seal via the port labeled “seal gas supply”. Most of this seal gas flows across the process labyrinth and back into the compressor, while the rest flows across the primary seal and into the interstage cavity between the two seals. This gas is commonly referred to as primary seal leakage. Secondary seal gas is introduced through the port labeled “secondary seal gas supply” into the interstage cavity between the two seals.

Most of the secondary seal gas flows across the intermediate labyrinth and sweeps the primary seal leakage into the primary vent, typically connected to a flare system. The rest of the gas flows across the secondary seal and out the secondary vent. This gas is commonly referred to as secondary seal leakage. The pressure differential across the secondary seal is quite low, so the leakage is very small compared to the primary seal.

A separation seal is also part of the overall seal assembly. This seal provides a separation between the secondary seal and the oil bearing cavity, by injecting a gas, typically nitrogen, between the two sections through the port labeled separation seal gas supply. This gas flows partially into the bearing cavity, and partially out the secondary vent where it mixes with secondary seal leakage. These separation seals generally fall into three categories: Labyrinths, contacting bushings, and non-contacting bushings. Each with its own benefits and drawbacks.

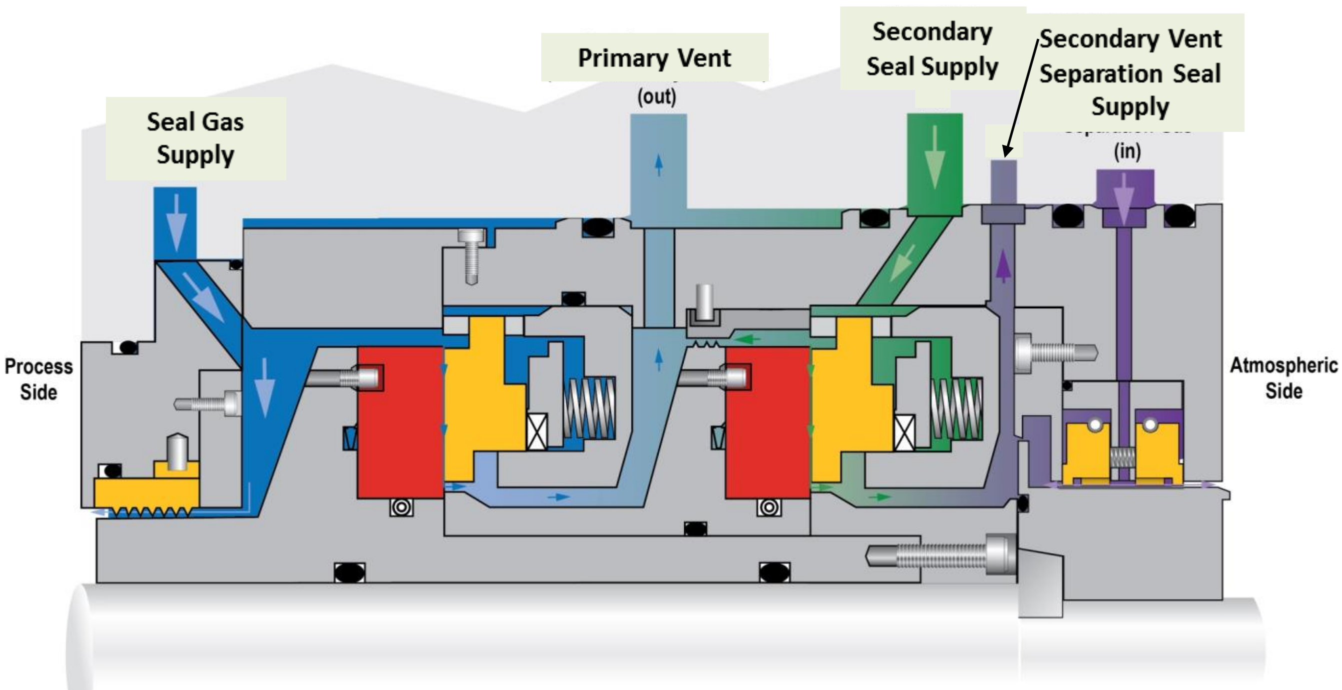

Figure 2: Tandem Seal Gas Flows

Secondary Seal Gas

The secondary seal gas is typically an inert gas, such as nitrogen. Most all of the supplied gas passes through the intermediate labyrinth where it mixes with the primary seal leakage and exits through the primary vent. The primary vent flow is monitored to indicate primary seal health, therefore the flowrate of the secondary seal gas is controlled to ensure it is relatively constant at various operating conditions.

Secondary Vent

The secondary vent is typically connected directly to atmosphere, and will contain the secondary seal leakage and the separation seal leakage. The secondary seal operates at such a low DP, that the secondary vent should remain unobstructed to avoid reverse pressurization and pushing any gas back into the bearing housing. Consequently, the secondary vent pressure and flowrate are typically unmonitored.

Separation Seal Gas

The separation seal gas is typically nitrogen or air and will be injected across the separation seal. Roughly half will flow into the bearing cavity, and the rest will mix with secondary seal leakage and flow out the secondary vent. This gas prevents bearing oil from entering and damaging the secondary seal.

The three streams above are important for maintaining the health of the dry gas seal, but for the sake of brevity this tutorial focuses on the two streams that most significantly impact the primary seal: Seal Gas Supply and Primary Vent. The principles that follow will generally apply to the other streams, but secondary seal monitoring and separation seal monitoring are both equally important for the health and functionality of a dry gas seal. Neither stream should be neglected in proper system design, especially as all seals have the potential for secondary seal failure without primary seal failure. For further information on secondary seal monitoring consult item one in the bibliography titled “Monitoring a Tandem Dry Gas Seal’s Secondary Seal”.

SEAL GAS SUPPLY

. In a tandem seal, seal gas is defined as the gas supplied to the high-pressure side of a primary seal, which flows through the primary seal faces and across the process labyrinth. It is typically a process gas that has been conditioned appropriately. The seal gas is the most important gas in the system, and as such is a primary concern for both control and monitoring. It acts as the first line of defense for the seal and can provide early indications of the potential failures.

Seal gas is typically process gas, but it is important that all seal gases, including nitrogen, be conditioned. Any gas going across the seal faces should be clean and dry. Clean means the gas should ideally contain no particulates. Obviously in practical terms this goal is unachievable, so gas streams should be restricted to having particles no larger than one micron. Dry means that the gas should be completely free of liquids and no liquid should condense from the gas during the pressure drop from seal gas inlet pressure to atmospheric pressure. These requirements are achieved by the use of heaters, separators, and filters, as required. Alternate gas sources may also be used in transient conditions or when available. For the purposes of this paper we will assume that all seal gas has already been conditioned before it arrives to the control system.

It's important to note that, while the seal gas does travel across the primary seal face, the vast majority of it (approximately 99%) flows across the process labyrinth and back into the compressor. This flow ratio means the seal gas volumetric flow requirements are not primarily driven by the dry gas seal, but by the geometry of the process labyrinth.

Required Gas Velocities

As mentioned above, the process labyrinth determines the required volume of the seal gas supply. The main concern is that unconditioned process gas could ingress across the process labyrinth and enter the seal cavity hindering seal performance. To prevent this, conditioned and regulated seal gas is provided by the seal gas control system. This gas is supplied at a higher pressure than the sealing pressure, and will consequently have positive flow across the process labyrinth.

However a larger volume of gas flowing across the process labyrinth necessitates a larger system. Heaters and pipework will have to be larger to accommodate increased flowrates and the more gas that flows through the filters the more often they will require changing. A control system is in place to optimize the gas consumption to the required amounts, but this begs the question: what is the required amount?

In order to prevent unconditioned process gas ingress, a minimum velocity should be maintained across the labyrinth at all operating conditions. This velocity is typically 5 m/s (~15 ft/s). These values have been shown to adequately protect the seal from contamination by reverse flow in different compressor types. The control system is designed to maintain this minimum velocity, but velocity is difficult to measure and control directly. To overcome this difficulty, two different control philosophies are used to control the velocity across the labyrinth indirectly.

Flow Control

The first seal gas control philosophy is flow control, but before discussing the details of a flow control system, it's important to define what flow control means. The term "flow control" refers to the *philosophy* behind the control scheme. It is independent of the variables being manipulated. When a flow control system is described, it means ~~that~~ the system is designed to ensure a proper gas velocity by controlling the flowrate across the process labyrinth. A sample flow control system is show below in figure 3.

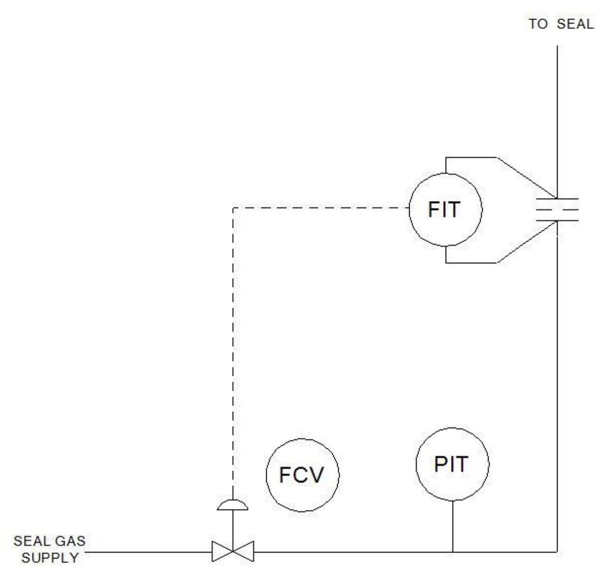

Figure 3 Flow Control System

Flow control systems are based on the following principal: A pressure differential across the orifice in the flowmeter is set to satisfy flow velocity requirement across the process labyrinth at a particular compressor operating point. The set point entered as a control parameter corresponds to the flowrate that provides the required velocity across the process labyrinth. Usually it corresponds to the flowrate that yields a velocity of 5m/s (~15 ft/s) at a double radial clearance of the labyrinth at normal operating conditions. Change in the sealing pressure propagates a pressure wave with the speed of sound towards the flow transmitter. This wave changes the pressure differential detected by the differential pressure transmitter indicating a change in the flowrate. Change of the pressure differential across the orifice from the preset value signals the flow control valve to open or close adjusting the flowrate until the differential pressure across the orifice returns to the preset value. In the case where a single control valve is controlling the flow to two seals, a low select is employed. The low select ensures that the flow to each seal will be greater than or equal to the predefined requirement.

The flow control system works by having the control valve act as the greatest restriction, choking and fully determining the flowrate. Flow control systems greatly simplify the calculations required in designing systems. As such there is less uncertainty introduced by engineering calculations, and a more accurate flowrate can be supplied to maintain the minimum velocities.

The major issue in flow control systems is the accuracy of the flow measurement for the transmitters. Every flow transmitter should be calibrated for a specific set of operating conditions. The density of the gas is used when determining the flow, and as such any change in the density of the seal gas will affect the reading of the transmitter. Users should be aware of the inaccuracies of the transmitter readings when setting and responding to the alarm conditions in the compressor, especially during startup and shutdown conditions. At these conditions the pressure or gas composition are often different from normal operating pressure, which affects gas density and consequently the instruments won't be able to read accurately.

Pressure Control

As with flow control, pressure control doesn't necessarily mean that pressure is the variable being measured and controlled. The phrase "pressure control" refers to the philosophy employed. In this case a differential pressure is maintained across the process labyrinth between a reference (balance piston cavity) and the pressure downstream of the control valve. This differential pressure should ensure that the minimum velocity is maintained across the process seal.

In order to ensure the proper pressure differential is maintained, it's important to account for pressure losses throughout the system in the preset differential pressure. There are pressure losses due to the following factors: Gas friction (proportional to the pipe length and the square of the gas velocity), pressure loss in pipe bends (proportional to pipe length and square of gas velocity), pressure loss/drop in compressor porting (cross-drilling of a diameter smaller than pipe cross-section), gas expansion in the primary seal supply cavity (port annulus area is smaller than the cavity cross section), and pressure drop across the process seal.

Two example pressure control systems are shown below.

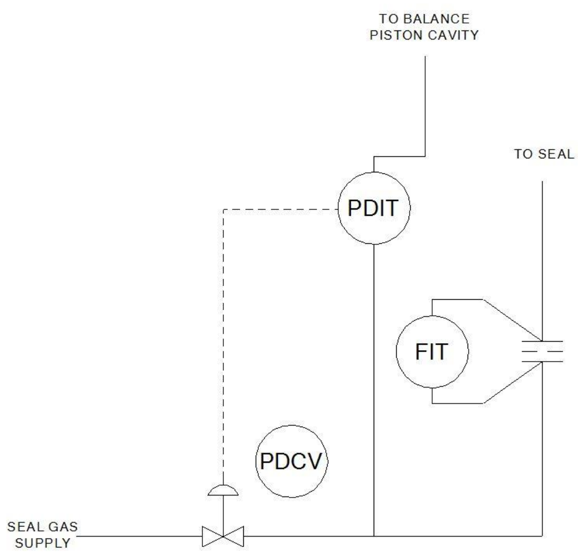

Figure 4 Typical Pressure Control Setup

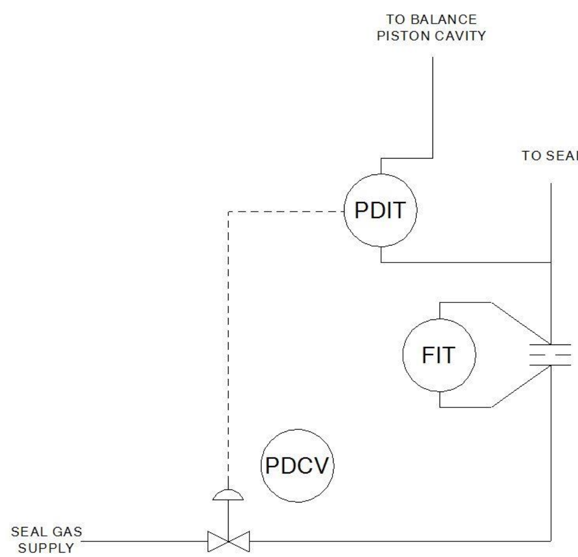

Figure 5 Recommended Pressure Control Setup

The recommend pressure control module, figure 5, functions by having a differential pressure transmitter communicate with a control valve. The differential pressure transmitter records the pressure differential between the seal gas supply pressure and the sealing pressure (the pressure directly downstream of the process labyrinth). The control valve then acts to maintain a specified differential pressure between these two points. By maintaining this differential pressure, the system is designed to maintain the minimum velocity across the labyrinth. The sealing pressure reference has historically been taken from the balance piston cavity. It is important that the tap for the sealing pressure reference line is located at the top of the cavity. This prevents possibility of liquids entering the sensing line.

The tap referencing the seal gas pressure should be placed as far downstream from the control valve in the line as possible, and ideally, immediately upstream of the compressor casing. Additionally, the tap should be as physically close to the compressor casing as possible. The recommended system shows this configuration, whereas figure 4 shows a more common configuration frequently found in the field. Placing the seal gas pressure reference tap as far downstream as possible helps to maintain an accurate measurement by minimizing the effects of any pressure losses caused by piping losses, gas expansion in the compressor annulus, or any other sources of pressure loss. As mentioned above, when sizing pipework and the control valve for a pressure control system, piping losses should be factored into any calculations, particularly when determining the set points of the control valve.

In addition to the differential pressure transmitter, a flow transmitter is included to aid in seal monitoring and failure analysis.

Pressure control systems suffer from a few drawbacks. Mainly the differential pressure required to maintain the minimum velocity across the labyrinth is generally incredibly small. So small as to be difficult to both measure and control with typical equipment. To compensate for this, additional differential pressure has to be provided in order to yield sufficient operating margin, and cover the design assumptions.

EXAMPLE CALCULATION

To better highlight the differences between the two approaches an example labyrinth velocity calculation has been performed for both philosophies. In this example and all those that follow in this section, a shaft of 110mm (4.33in) was considered, with suction and discharge pressures at 20bar (290psi) and 70bar (1015psi) respectively. A labyrinth process seal with a design radial clearance of 0.25mm (0.01in) was assumed.

For flow control the calculation is relatively simple. Maintaining a velocity of 5 m/s (~15 ft/s) at double the radial clearance (a standard industry safety factor), the required flowrates across a single process labyrinth are shown in table 1 below.

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Result	SI units	US units
Mass Flow	30 kg/hr	66 lbm/hr
Actual Flow	62 l/min	2.2 CFM
Volume Flow	740 Std l/min	26 SCFM
Velocity	5.0 m/s	16.4 ft/s

Table 1 Flow Control at 0.5mm Clearance (2x design)

For differential pressure control the process is slightly more complicated. Calculating the required DP across the process labyrinth to maintain 5m/s (~15 ft/s) velocity at double the radial clearance gives a differential pressure requirement of 0.018 bar (0.26psi). This differential pressure is so small as to be impractical to control. So when designing the system the control valve will be given a preset DP value of 0.5bar (7.25 psi) to maintain. The DP is always set at a rounded up value. Approximating the losses due to piping, expansion in the cavity, etc., as 0.4 bar (5.8 psi) the portion of the differential actually maintained across the labyrinth would be 0.1bar (1.45psi) . Calculating a pressure drop of 0.1bar (1.45psi) across the labyrinth yields the seal supply rates shown in Table 2.

Result	SI units	US units
Mass Flow	210 kg/hr	470 lbm/hr
Actual Flow	440 l/min	15 CFM
Volume Flow	5200 Std l/min	190 SCFM
Velocity	36 m/s	120 ft/s

Table 2 Pressure Control at 0.5mm Clearance (2x design)

These numbers are significantly larger than those of the flow control example above, but both examples rely on the same safety factor of doubling the radial clearance. In part this safety factor is in place to account for labyrinth wear, which is a virtual certainty in all machines. Each philosophy responds to this wear in a different way.

Labyrinth Wear

As the labyrinth wears, the radial clearance begins to increase. This increase affects the performance of each control philosophy. Consider the scenario where the radial clearance starts at 0.25 mm (0.001 in) and then wears to 0.5mm (0.002 in). Table 3 shows the effects this wear would have on a flow control system, assuming the system was designed for 0.5mm (0.002 in) clearance, and that the system maintains a constant flowrate of 740 Std l/min (26 SCFM).

Result	0.25mm (0.010 in)		0.5mm (0.020 in)	
	SI units	US units	SI units	US units
Mass Flow	30 kg/hr	66 lbm/hr	30 kg/hr	66 lbm/hr
Actual Flow	62 l/min	2.2 CFM	62 l/min	2.2 CFM
Volume Flow	740 Std l/min	26 SCFM	740 Std l/min	26 SCFM
Velocity	10 m/s	33 ft/s	5.0 m/s	16.4 ft/s

Table 3 Flow Control at Varying Clearances

Because the system is actively attempting to maintain the same flowrate, the consumption of gas remains the same. The only difference is in the velocity across the labyrinth. This velocity varies inversely with increasing gap, so when designing a flow control system it's important to always account for any change in radial clearance that may occur, whether due to shaft growth, wear, or some other reason.

Pressure control is more complicated. If the system was sized to maintain a DP of 0.1bar (1.45 psi) across the labyrinth at 0.5 mm (0.002 in), that DP will not necessarily be the pressure differential across the labyrinth at a different clearance. Maintaining the 0.1 bar (1.45 psi) differential at a smaller gap requires lower flowrates. Because the pressure losses mentioned above are dependent on flowrate, the lower flowrates mean that the pressure losses will be smaller. For the sake of this example, it is assumed that differential pressure across the labyrinth will remain 0.1 bar (1.45 psi), but in reality the flowrates and DP across the labyrinth would be slightly higher. Table 4 compares the flowrates across the labyrinth at two different radial clearances assuming that the differential pressure across the labyrinth remains a constant 0.1 bar (1.45 psi).

Result	0.25mm (0.01 in)		0.5mm (0.02 in)	
	SI units	US units	SI units	US units
Mass Flow	71 kg/hr	66 lbm/hr	210 kg/hr	470 lbm/hr
Actual Flow	150 l/min	52 CFM	440 l/min	15 CFM
Volume Flow	1800 Std l/min	62 SCFM	5200 Std l/min	190 SCFM
Velocity	24 m/s	79 ft/s	36 m/s	120 ft/s

Table 4 Pressure Control at Varying Clearances

A pressure control system will consume a variable amount of seal gas at varying clearances, and is able to adjust ensuring that the velocity always remains above the required 5 m/s (~15 ft/s).

Operating Conditions

Each control system should be capable of handling all the transient operating conditions in the compressor. All operating conditions have differing pressures and special considerations that affect both pressure control and flow control systems, but they often affect them in different ways.

At startup there is often a lack of available head pressure for the system. The available pressure affects both systems equally in the typical case where seal gas is taken from discharge or an interstage of the compressor. If no alternate seal gas supply is available, and there is no booster installed in the system, then process gas will be free to ingress across the labyrinth at startup, until discharge pressure, and consequently seal gas pressure, is generated. Shutdown is the same process as startup but in reverse. The system steadily begins to lose head pressure, and will struggle until the compressor casing is depressurized.

After startup the compressor will be in normal operation. Here the pressure control system and the flow control system are both operating as previously described. This should be considered the default state for both systems.

Settleout shares similar concerns to startup when it comes to the available source of high pressure gas. Due to environmental or production concerns oftentimes compressors will not be blown down to atmospheric pressure while in stand still. If the compressor is not blown down, then suction and discharge pressures within the casing will equalize. When the primary seal gas supply is taken from discharge of the compressor, there will be no head pressure available to the support system to provide a flow across the process labyrinth. The seals however will continue to leak, and this leakage will induce a flow that draws unconditioned gas back across the process labyrinth and into the sealing cavity. This flow can bring contamination and condensate which may cause damage to the seal. Head pressure can be supplied by including a booster in the gas seal system, or by utilizing an alternate seal gas supply during settle out conditions.

Here flow control and pressure control both fare about the same. Without head pressure both systems are completely ineffective, and until such a time as flowrate can be generated both will remain inadequate.

PRESSURE VS. FLOW DIRECT COMPARISON

When selecting the type of control desired, the largest concern for most end users is the amount of gas consumed by the system. As the seal gas is taken from discharge and recirculated to suction and flare, it has a minor effect on compressor efficiency.

The required gas volume affects the size of the piping and accessories of the system, and, affects the size of the compressor port drillings. An increased flowrate also necessitates more frequent changing of filters or other such devices. Table 5 below shows the consumption rates, during normal operation, for both setups calculated based on the parameters and caveats discussed above.

Result	Pressure Control		Flow Control	
	SI units	US units	SI units	US units
Mass Flow	210 kg/hr	470 lbm/hr	30 kg/hr	66 lbm/hr
Actual Flow	4400/min	15 CFM	62 l/min	2.8 CFM
Volume Flow	5200 Std l/min	190 SCFM	740 Std l/min	26 SCFM
Velocity	36 m/s	120 ft/s	5.0 m/s	16.4 ft/s

Table 5 Pressure vs. Flow Direct Comparison

Flow control emerges as the clearly superior choice for most applications as it consumes less than $1/4^{th}$ of the mass flow of gas as pressure control. But only 99% of the gas will end up flowing across the process labyrinth. The other 1% will flow across the primary seal and through the primary vent, and by monitoring this flow, a plant engineer can discern the health of the primary gas seal.

PRIMARY VENT

In addition to controlling the gas streams delivered to the seal, a good seal gas panel helps monitor the health of the gas seal. The additional information helps predict or prevent failures, and aids in diagnosing the root cause of a failure if one occurs.

The primary seal is generally the area of greatest concern, as it is the main seal of the dry gas seal. The best indicator of seal health is seal leakage, and the only place to monitor seal leakage outside of the compressor casing is the primary vent. As such, any troubleshooting or failure examination generally starts with the primary vent.

Primary vent systems can potentially experience full sealing pressure in the event of a failure. Consequently it is important that failure conditions are considered when designing a primary vent system. The system and associated componentry, including pipework, should be rated for sealing pressure, or some sort of pressure relief device should be fitted.

The primary vent can be monitored on the basis of pressure, flow, or both. Figure 6 shows a potential vent monitoring setup.

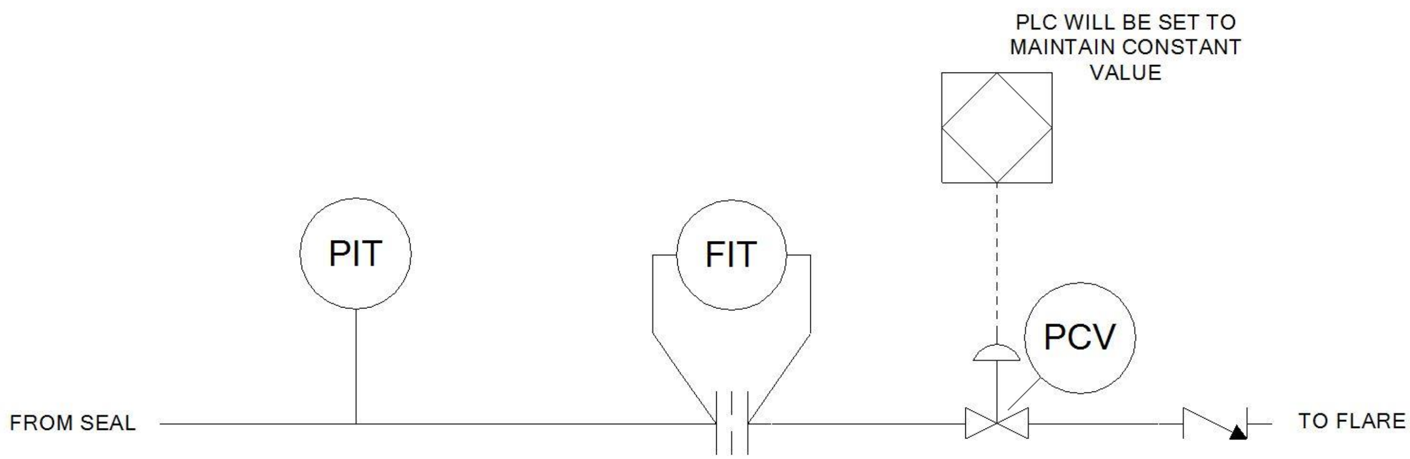

Figure 6 Typical Primary Vent Setup

This setup features a pressure transmitter to monitor pressure, a flow transmitter to monitor the flow and a check valve to prevent any backflows from the flare system. The system also features a pressure control valve. This valve in place to generate a backpressure in the primary vent line.

Backpressure

Backpressure can improve secondary seal performance, but one of the biggest benefits of creating the backpressure is improved ability to monitor secondary seal failure. Without this backpressure in place, it is very difficult to set low alarms that will reliably detect secondary seal failure without causing numerous false alarms. However without secondary seal gas, the end user can have difficulty building up backpressure in the primary vent, because the only flow is primary seal leakage.

While generating a backpressure is not required for the operation of the dry gas seal, backpressure creation can greatly improve the performance of the seal system, helping to isolate transmitters in the primary vent from any flare pressure fluctuation. The flare is already operating at low pressures, typically 0.2 bar – 1.0 bar (3 psi – 14.5psi) where minor fluctuations create a significant amount of instability in the flow transmitter signals. Keeping a constant backpressure in the primary vent, usually of maximum flare pressure (1bar, 14.5psi), helps to mitigate this issue and provide a stable signal to monitor. Utilization of pressure transmitters for alarms in vent lines routed directly to flare can generate false alarms due to variable flare pressure, but the presence of the stable backpressure mitigates this problem.

The industry utilizes many different methods to generate a backpressure in the primary vent. The simplest of which is to include a high cracking pressure check valve. Check valves suffer from a number of issues preventing the method from being the preferred solution. At low flowrates check valves can chatter, and the backpressures they generate are not as consistent as other options. Contamination frequently builds up on the seat preventing the valve from closing fully. For a more robust solution a backpressure regulator can be fitted. This regulator maintains the backpressure more reliably than a check valve, and allows for alarms to be set on the pressure and flow transmitters. However, because the regulator will endeavor to maintain a constant backpressure, the only reliable indicator of an increase in leakage will be the flowmeter. Control valves offer benefits in certain applications where the end user does not wish to set a trip based solely on high primary leakage. The control valve provides as reliable a backpressure as the regulator while also allowing for position monitoring. Position monitoring gives the end user greater visibility into the actual conditions of the primary vent, and in turn more insight into the health of the primary seal.

Primary Seal Health

Leakage is one of the best indicators of primary seal health, and the best place to monitor seal leakage is the primary vent. In the case where secondary seal supply is present, all the seal leakage will flow out the primary vent. As such, the primary vent flow will be equal to Seal leakage + Secondary Seal Gas Supply – Secondary Seal leakage. Secondary seal leakage is generally so small that it can be ignored for the purposes of this tutorial. This combination of gasses means that the flow out the primary vent will be some mixture of the seal gas and the secondary seal gas. If no secondary seal gas supply is utilized, then the flow through the primary vent will be Seal Leakage –Secondary Seal Leakage. Again, the secondary seal leakage is still quite small and can generally be ignored.

It is common practice within industry to set alarms (high and low) and shutdowns (high-high alarms) as multiples of the guaranteed seal leakage. Table 6 below illustrates common multipliers used for this purpose. While high and high-high alarms may be set on both flow and pressure transmitters, low alarms will typically only be set on pressure transmitters. The flowrates in the primary vent are typically very low, making it difficult to specify a flow transmitter that can accurately handle the whole range of expected flowrates.

Seal Vent flow	Seal guaranteed leakage flow	Intermediate Labyrinth flow	Indication
A= X+Y	X = (1 x Guaranteed Seal Leakage)	Y = Secondary Seal Supply flow rate	Normal
B= X+Y	X= 2.5 times Guaranteed Seal Leakage	Y = Secondary Seal Supply flow rate	High
C= X+Y	X= 5.0 times Guaranteed Seal Leakage	Y = Secondary Seal Supply flow rate	High-High

Table 6 Typical Alarm Setpoints for Flow Instruments

The flow transmitter (orifice style) in the primary vent should be sized for the flow rates defined in Table 1. If no secondary seal gas injection is present, the above table still applies except Y=0 for all cases. If secondary seal supply is present then the gas composition used in sizing will be a mixture of secondary seal gas (typically nitrogen) and process gas. The gas composition of the primary vent affects the alarm setpoints used, so a clear understanding of the gas composition is paramount. If the gas composition changes and the instruments used to monitor the seal are not updated, the system can trigger false alarms and even false shutdowns.

Low Alarms, if utilized, should always be reviewed in detail, as they may be infeasible for a given instrument or set of operating conditions. Improper low alarm settings can cause false alarms, especially during certain compressor operating conditions such as startup. Pressure alarms will generally be used for low alarms, which will indicate a secondary seal failure. Note this is usually only possible in the case where a sufficient backpressure is created, and secondary seal gas injection is included.

Primary Seal Vent – Gas Composition

The gas composition has a significant effect on the accuracy of transmitters, and directly impacts orifice sizing for an orifice style flowmeter. The following example illustrates the difference in measured differential pressure across an office with varying gas compositions and constant flowrates. Consider a hydrogen recycle compressor compressing hydrogen at a suction pressure of 20 bar (290 psi), and with a 110mm (4.3 in) shaft running at 13000 rpm. Assuming a normal leakage of 1.7 Nm3/hr (1.1 SCFM) and a secondary seal gas flowrate of 5.0 Nm3/hr (3.1 SCFM) the proper alarm values for the flowmeter can be calculated. Because the flowmeter being used is a differential pressure orifice style flowmeter, the alarm set points are expressed as differential pressures. The alarm differential pressures for the normal case, hydrogen seal gas, are shown in table 7.

Alarm Value	Primary Seal Leakage		Measured Differential Pressure	
	SI Units	US Units	SI Units	US Units
Normal Leakage	1.7 Nm3/hr	1.1 SCFM	.16 bar	2.3 psi
High Alarm	4.3 Nm3/hr	2.7 SCFM	.26 bar	3.8 psi
High High alarm	8.5 Nm3/hr	5.3 SCFM	.41 bar	5.9 psi

Table 7: Calculated Alarm Pressures Seal Gas: Hydrogen

These calculated values provide guidance for an operator on alarming the system, but these values are dependent on the gas composition. If the compressor operates on an alternate seal gas, the same flowrates will produce different differential pressures. In the case below, natural gas substitutes for the hydrogen as an alternate seal gas and will leak across the seal mixing with the secondary seal gas in the primary vent. The new differential pressures, at the same assumed flowrates, are shown in table 8.

Alarm Value	Primary Seal Leakage		Measured Differential Pressure	
	SI Units	US Units	SI Units	US Units
Normal Leakage	1.7 Nm3/hr	1.1 SCFM	.21 bar	3.0 psi
High Alarm	4.3 Nm3/hr	2.7 SCFM	.41 bar	5.9 psi
High High alarm	8.5 Nm3/hr	5.3 SCFM	.76 bar	11 psi

Table 8: Calculated Alarm Pressure Seal Gas: Natural Gas

The pressures generated by the natural gas are much higher than the ones by the hydrogen. If the orifice flowmeter is not adjusted when the secondary seal gas is in use, the system could indicate a seal failure when none exists. In the above case, the trip of the compressor, configured for the hydrogen seal gas, would be activated by the high alarm flowrate of the natural gas. If the user does not understand the instrumentation used in the system, production could be halted while the seal is perfectly functional.

INSTRUMENTATION

In order to properly control and monitor the gas streams, correct instrumentation is required. There are three key variables of concern when discussing the gas seal environment: pressure, flowrates and temperature.

Pressure monitoring

Pressure transmitters are simple instruments well understood by industry. They will typically be connected to an impulse line running to a tap at whichever point the pressure measurement is desired. This tap can be placed without causing a significant pressure drop or flow restriction, giving great freedom of placement. Pressure transmitters will generally have a turndown ratio of 10:1.

Temperature monitoring

In order to prevent condensation in the sealing cavity or on the seal faces, gas temperature should always be maintained above its dew point and monitored. Seal gas heaters are supplied specifically for this purpose. The temperature reference should be located as close to the compressor as physically possible to eliminate temperate loss in the lines or due to expansion.

Flow monitoring

Flow monitoring is more complicated than pressure monitoring. It is important to distinguish between the different types of flow measurement when discussing flowrates in systems. There is mass flow (kg/hr, lbm/hr), actual volumetric flowrate (l/min , CFM) and standard volumetric flowrate (SL/min, SCFM). Flowmeters in most applications will express their readings in some form of standard volumetric flowrate. While standard volumetric flowrate is independent of density, any flowmeter reporting a standard flowrate will first utilize some calculation to determine actual volumetric flowrate, or mass flow. These calculations always involve an assumption of the actual density meaning that while the transmitter may be outputting standard flowrates, the measured flowrate will still vary as the actual density differs from the assumed density. There are several different types of flowmeters each with their own operating principles, and their own advantages and drawbacks. A few of the more common types are detailed below. In most cases, the measured flow is accurate only for the exact gas composition, pressure, and temperature used for its calibration. Most end users find these limitations acceptable because the actual value of the flowrate is less important than the trend of the measured flowrate.

Orifice DP Flow Meter

Orifice style flow meters are one of the most common flow transmitters in the field. By measuring the differential pressure across a known geometry, the flowrate through said geometry can be calculated using the modified Bernoulli equations. In this case the differential pressure is measured using a differential pressure transmitter. An orifice provides the known geometry and for a measured pressure drop a flowrate can be calculated. Orifice style flowmeters are simple in construction and have no moving parts, making them reliable. They are also economical and have a well-defined calibration curve with respect to flow vs. differential pressure. Orifice flowmeters are not covered by API 14.3.1 or recommended for use in applications where the Reynolds number drops below 4,000. As the Reynolds number decreases, the device's measurement uncertainties grow exponentially. (API 14.3.1)

If the gas conditions change from the calibrated values, (pressure, temperature, gas comp, etc.) then the flow meter reading becomes inaccurate. However, the error is insignificant if the variance between the actual case conditions and calibrated conditions are minimal. The orifice should also be sized properly. The orifice by necessity will cause a pressure drop, but an improperly sized orifice can drastically affect the pressure, flow, and the flow measurement.

Rotameter

Rotameters, are variable area, in-line type flowmeters. Inside the rotameter is a small float which rests at the narrow end of a thin tapered tube. As gas flows through the tube, drag force acts on the float which pulls it up against the force of gravity. The tube tapers outward causing the velocity of the gas (and consequently the drag force) to decrease. Eventually the float comes to a rest in an equilibrium position. The position of the float at equilibrium can be equated to a particular flowrate given a set of gas properties (gas composition, temperature, pressure etc.) and float material.

Rotameters suffer from some major drawbacks. Like the orifice style flowmeter, they should be calibrated for a particular set of operating conditions (gas density and anything that affects it). Any deviation from these operating conditions will deviate the measurement from the true value. Additionally, as they rely on gravity to achieve equilibrium, they should always be installed vertically. Any liquids or contamination within the system have the potential to contaminate the float, changing the weight of the float or permanently damaging it, altering the readings of the flowmeter. Because they are analog devices, rotameters also have the lowest accuracy of all flowmeters, the accuracy is half the smallest division of the scale employed.

Thermal Mass Flow meter

Mass flow meters are available in a variety of styles each employing a different operating principle. For gas seal systems, thermal dispersion mass flow meters are one of the most commonly used types. Thermal-dispersion flowmeters are designed for low flow and low pressure applications rendering them well suited to the conditions found in the vent lines. An example of a thermal mass flowmeter is show below in figure 7.

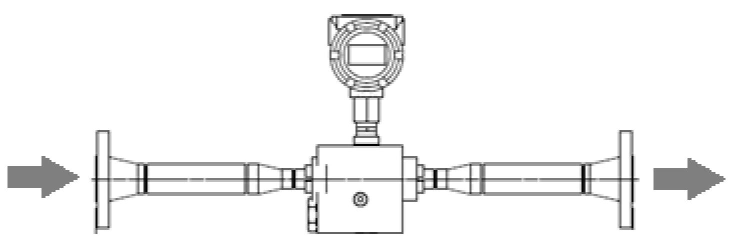

Figure 7: Thermal Mass Flow Meter

Thermal mass flow meters also should be calibrated to a particular set of operating conditions and as such a change in those operating conditions will affect the accuracy of the measurement. Unlike orifice style flowmeters and rotameters, thermal mass flow meters give a measurement of the mass flowrate instead of the volumetric flowrate. They also have a superior turndown ratio generally around 1:100.

Thermal mass flowmeters employ a 4-wire system, which means that additional cable will be required to power their heating circuit.

DRY GAS SEAL TROUBLESHOOTING

The dry gas seal support system not only regulates gas streams to the dry gas seal, but also transmits the necessary information to monitor the seal's condition. Support system instrumentation, with appropriate alarms, helps users detect and analyze issues. Not all failures are catastrophic and quite often values will change slowly over time. In such scenarios the system is used to identify the problems as they occur.

Alarms

It is important to distinguish the difference between alarms and trips. Alarms trigger an alert that a condition outside the normal range of operation has been reached. Trips initiate an automatic shutdown of the compressor. Alarms should be actionable and allow operations to respond and correct the condition preceding any trips. The high (H) alarm is the "alarm", and the high high (HH) alarm is the "trip". Traditionally, alarm values are simple set points so increasing flow rates, valve movements, or increasing pressures may go un-noticed as conditions degrade. This significantly reduces the ability of the site personnel to catch issues early and implement corrective actions or plan for maintenance. Trend alarms, alarms based on change in measurement value over a set time period, can be applied within site condition monitoring systems to alert the technical team prior to a traditional alarm triggering.

Trends

When troubleshooting, absolute values are not necessarily the main concern. As with most parameters in equipment monitoring values from seal gas system instruments are never steady or constant. Values will vary or drift over short periods of time and can cause some concern to unexperienced personnel. This is particularly true of flow instruments for the reasons discussed in the instrumentation section. While large step changes are significant and need to be addressed, small variations are normal. In order to accurately diagnose problems, data needs to be trended over time to show minimum, maximum and average values. A change in trends over time is a key point to help identify potential issues.

Correlation of data

One of the fundamental steps in any diagnostic activity is to validate the data. Instrument calibrations and redundant transducers are useful for accuracy but may not validate a potential failure mode. Here we rely on correlation of data between different types of transducers for validation. All sections of the system contribute to failure analysis, because each particular failure mode can affect each section differently. A leaking seal can increase the primary vent flow, but so can a malfunction of the secondary seal gas supply regulator. To properly diagnose the problem, the end user should examine not only the flow transmitter in the primary vent, but all of the transmitters in the system to seek out any abnormalities. A simple cause and effect can be completed on the dry gas seal support system around common seal failure modes. This allows the facility to improve monitoring, change conditions, plan for an outage, and address the root cause of problems, instead of responding to a failure and equipment trip.

EXAMPLES

In order to illustrate the points discussed above, several examples of dry gas seal failures are provided below.

Example 1 - Particulate contamination.

Background

A set of dry gas seals were installed and commissioned in a new application which continued to operate for over a year without incident. The compressor was then shut-down for an issue unrelated to the dry gas seal. However, shortly after the restart, the drive end primary vent transmitter recorded a step change in leakage. Although the leakage was high, it remained below the trip value and the compressor continued to operate. Approximately 40 days later the leakage value spiked again this time initiating a shutdown. The leakage trends are illustrated in Figure 8. The dry gas seals were subsequently removed and sent to the factory for examination.

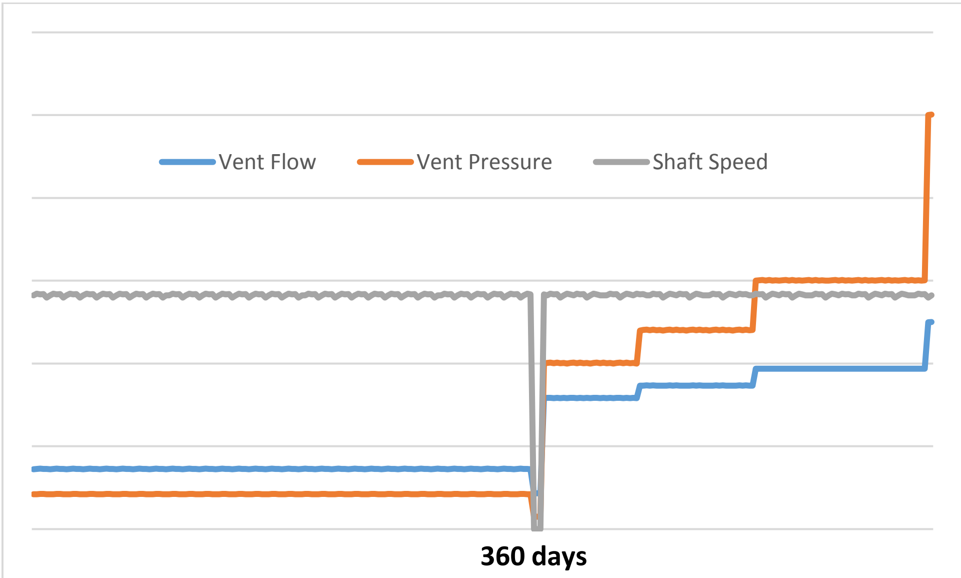

Figure 8 Primary Vent Leakage Trend

Seal Observation

Inspection of the dry gas seal revealed particulate contamination throughout the primary seal components (figure 9). Further examination of the stationary components showed a build-up of particulate where the dynamic sealing element interfaces with the balance diameter sliding surface (figure 10).

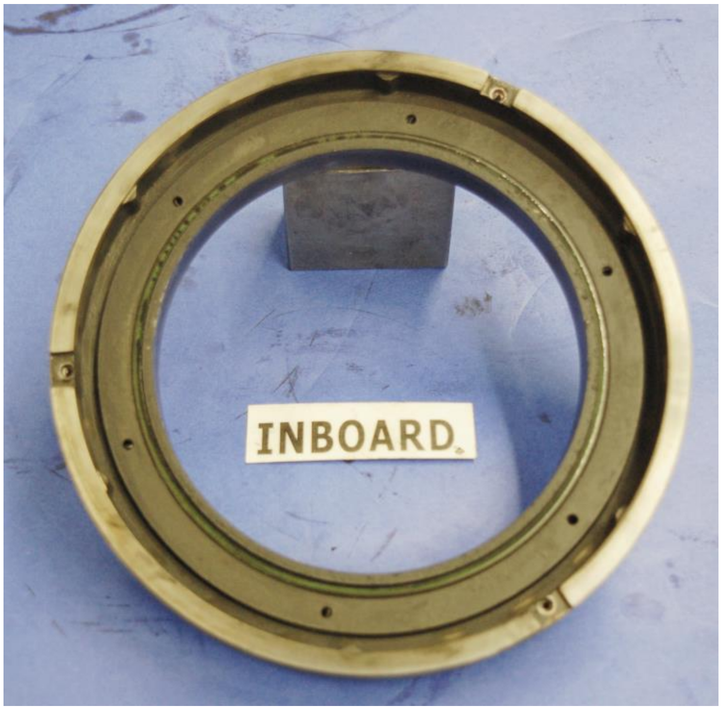

Figure 9: Inboard retainer

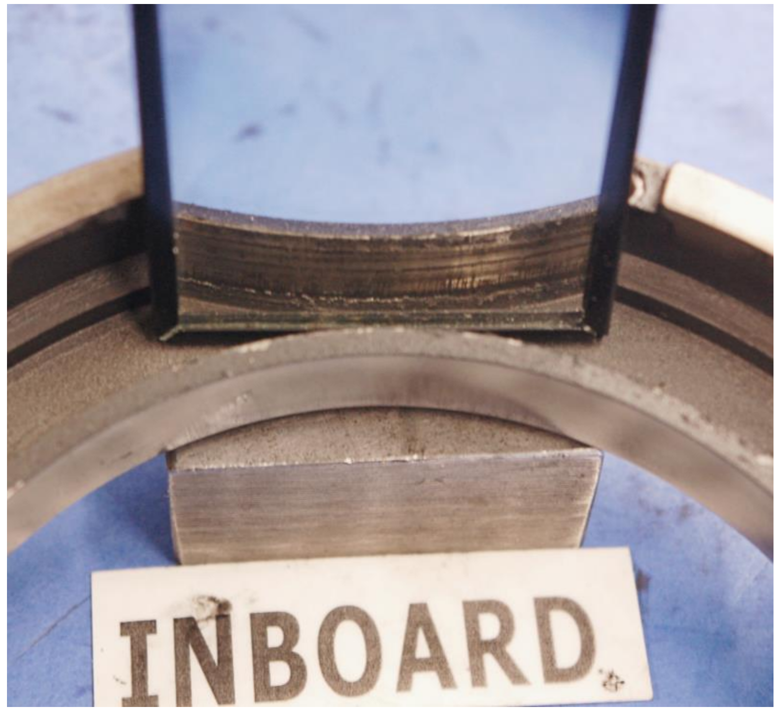

Figure 10: Close up of balance diameter

Discussion

Step changes in the primary seal leakage are typically associated with particulate contamination. Particulate contamination affects the seal in a myriad of ways, but the two most common are as follows. Firstly the particulates enter the seal faces increasing the gap, and secondly the contamination accumulates on the balance diameter reducing the seal's ability to track axial movement. The primary seal leakage data and physical appearance of the seal components support the second hypothesis. However, the high primary seal leakage and contaminated components are the symptoms, not the cause. Determining the root cause requires further analysis.

Analysis

There are essentially two ways particulate can enter the seal cavity. Particulate can enter the seal cavity via the seal gas supply if the filter is bypassed or the wrong filter element is used. The seal gas supply filters were checked to ensure they were installed correctly and the elements were found to be the correct rating. Contaminates can also enter the seal cavity via the process labyrinth if insufficient velocity is supplied across it. Conducting a review of the P&ID indicated that the gas velocity across the process labyrinth was inadequate to prevent reverse flow into the seal cavity during normal operation. Additionally as part of the RCFA, the start-up procedure was reviewed. The control system was properly equipped with a booster to supply clean gas during the startup and standstill, but the booster was inoperative due to water accumulating in the air driven portion of the booster.

Conclusion

The RCFA concluded that cause for the high primary seal leakage was insufficient velocity across the process labyrinth to prevent reverse flow of unfiltered process gas into the seal cavity hindering the tracking ability of the stationary seal face.

Example 2 - Axial runout (swash) and leaking dynamic sealing element

Background

During a scheduled turnaround, a set of dry gas seals were replaced and upon restart the seal performance was within acceptable parameters. As the unit continued to operate the Drive End primary seal exhibited increasing leakage. During the ensuing months the leakage values never stabilized and continued to increase until ultimately the high alarm threshold was reached and a shutdown was properly initiated. The leakage trends are illustrated in figure 11. The dry gas seals were subsequently removed and sent to the factory for examination.

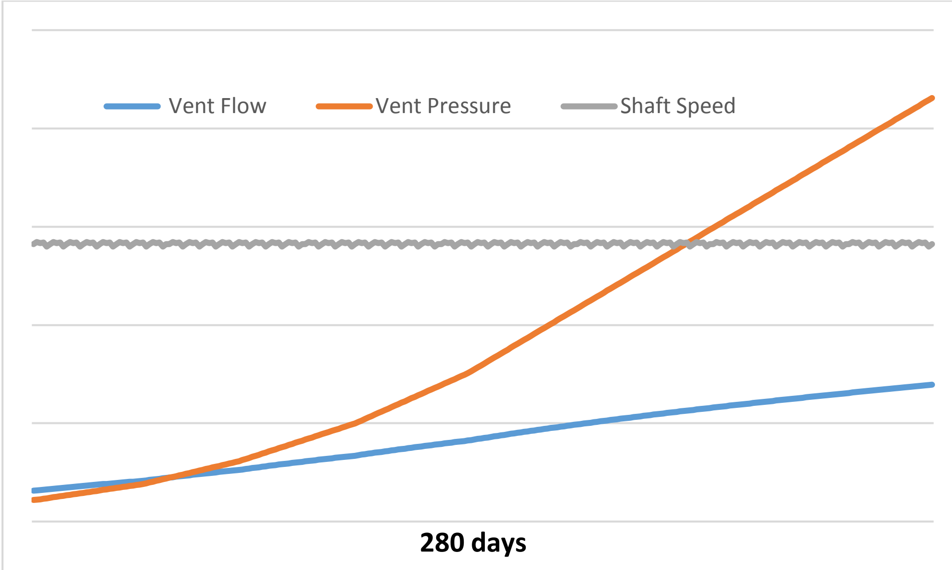

Figure 11: Leakage Trend

Seal Observation

The initial observations revealed seal faces in good condition with no significant signs of any issues. The ensuing examination of the core metal components did not reveal signs of distress or abnormalities that could explain the seal performance. The focus then shifted to the O-rings. The O-rings sealing in the compressor shaft and bore showed no signs of chemical attack or degradation, but the dynamic O-ring, on the balance diameter, showed signs of abrasion particularly where the sealing element interfaced with the balance diameter, shown in figure 12.

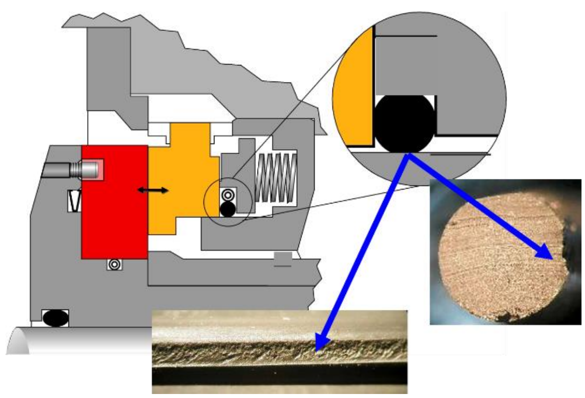

Figure 12: Balance Diameter Wear

Discussion

Slow increasing leakage can be caused by a variety of factors including liquid contamination, particulate contamination, chemical attack, or initial phases of extrusion of the balance diameter O-ring. However in this case, O-ring abrasion caused the leakage. Similar to the pervious example the increasing leakage and O-ring abrasion are symptoms, not the cause. The source of the abrasion can only be identified with further analysis.

Analysis

Typically abrasion is found in dynamic seals subjected to reciprocating, oscillating, or rotary motion. In a dry gas seal the balance diameter seal is designed to accommodate axial & oscillating motion linked with the stationary seal face tracking the rotating seal face. However if the surface finish is too rough, abrasive particles contaminate the sliding surface, or if the motion is excessive, abrasion can occur. The surface finish of the balance diameter was checked and within specification, and as mentioned in the observation section no abrasive particles were found within the gas seal. Axial vibrations data were reviewed and did not reveal any abnormal characteristics. An inspection of the rotating components followed revealing two deep scratches and displaced metal in a 90 degree segment of the locknut surface supporting the seal rotor. The raised surface from the scratches created an out of perpendicularity condition causing misalignment of the locknut as shown in figure 13.

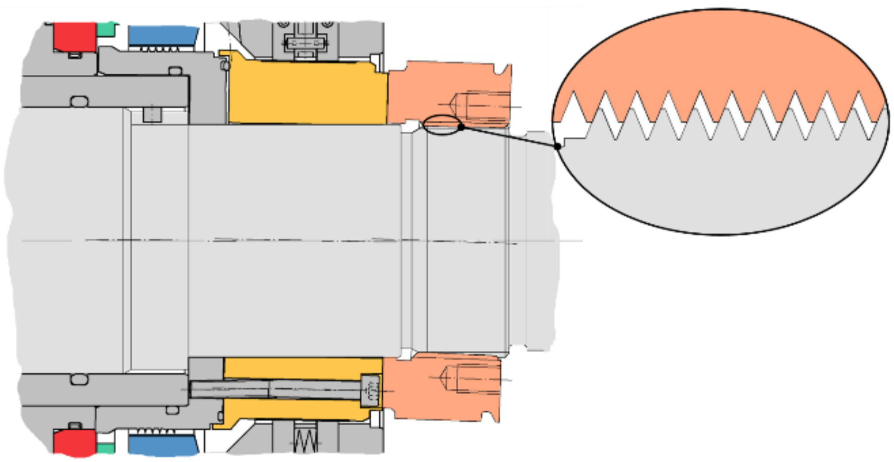

Figure 13: Misalignments of Axial Restraining Nut

One of the primary functions of the locknut is to support the thrust load of the seal rotor. As pressure is applied to the dry gas seal a reactionary force from the support surface it transmitted to the seal rotor. Therefore, any angular misalignment of this support surface will transmit the misalignment to the seal rotor and ultimately the rotating seal face resulting in a swash motion as shown in figure 14.

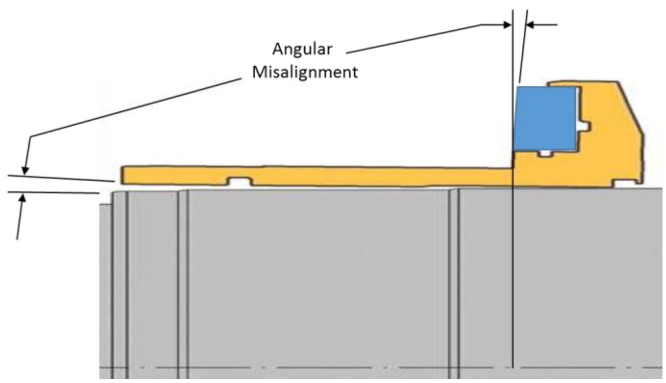

Figure 14: Angular Misalignment - Swash

The swash forces the stationary face and balance diameter O-ring to adjust their axial position with every rotation creating an oscillating motion that rapidly accelerates wear and abrasion. Depending on the severity of the deviation from perpendicularity and the operating speed/pressure, the time to failure will vary. This phenomena is not limited to O-rings and can also impact polymer based sealing elements.

Conclusion

The RCFA concluded that cause for the increasing primary seal leakage was abrasion of the balance diameter secondary seal attributed to excessive swash induced by the displaced metal on the supporting surface of the locknut.

Example 3 - Seal contamination by non-evaporable liquid.

Background

Following an extended shutdown, an offshore HP compressor tripped as the driver accelerated to normal operating speed. The primary vent transmitter recorded increasing primary seal leakage on both ends of the compressor immediately following shaft rotation. While the primary seal leakage on the non-drive end increased slowly, the drive end primary seal leakage increased rapidly as illustrated in figure 15. Provisions were made to remove the dry gas seals, but after removing the bearing the technicians realized the dry gas seal had suffered a catastrophic failure and was welded to the shaft. The seal could not be extracted offshore and the compressor was removed to an OEM facility for removal and examination.

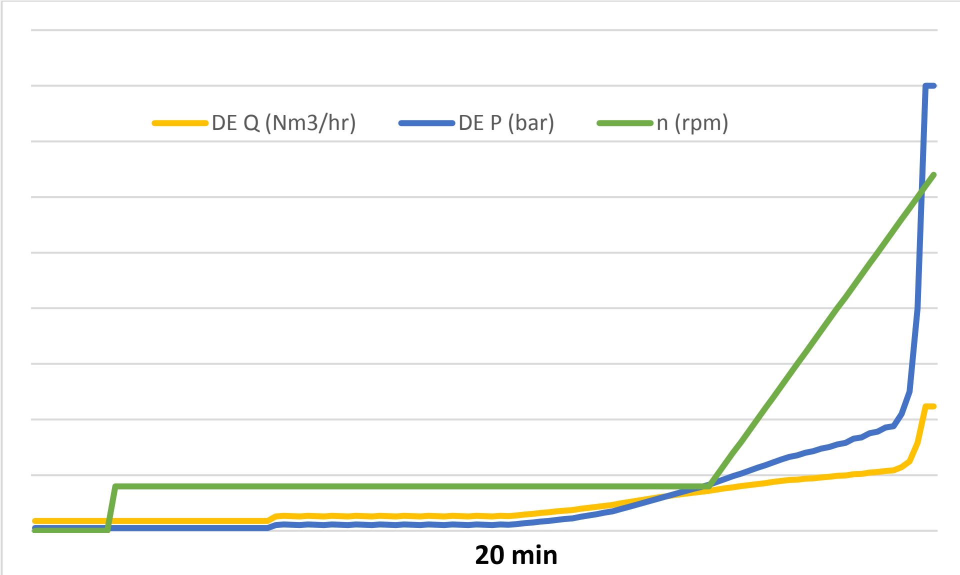

Figure 15: Primary Vent Leakage Trend

Seal Observation

Immediately upon extraction of the dry gas seal heavy liquid contamination was seen on the exterior of the dry gas seal as shown in figure 16. Further examination of the internal components revealed more liquid contamination as shown in figure 17.

Figure 16: External Liquid Contamination

Figure 17: Internal Liquid Contamination

Discussion

Dry gas seals can tolerate periodic ingress of a small amount of evaporable liquids, but, light hydrocarbon condensate, heavy hydrocarbons, crude, or any other type of non-evaporable fluid will inevitably lead to a seal failure if not properly addressed. Although, the reason for the seal failure was clearly the liquid contamination a RCFA was initiated to identify the source.

Analysis

There are essentially two ways liquids can enter the seal cavity in order to impact the performance of the dry gas seal: through the process labyrinth or through the seal gas supply. A review of the P&ID indicated that there was sufficient velocity across the process labyrinth to prevent reverse flow, but the seal system lacked the appropriate diagnostic modules to validate the calculated seal gas flow and associated velocity. If the seal gas is not properly conditioned, it can also carry liquids into the seal. A current gas analysis was obtained and the calculations performed concluded that due to the Joule-Thompson effect liquids condensed at start-up due to the low seal gas temperature. The analysis also concluded that during normal operating conditions the seal gas was below the dew point prior to reaching the compressor casing subjecting the dry gas seals to a constant flow of liquid contamination.

Conclusion

The RCFA concluded that liquid contamination from the seal gas supply caused the catastrophic failure. As the result of the RCFA a seal gas conditioning unit was installed that included a heater to ensure that the seal gas supply would no longer fall below its dew point at any point during operation.

SUMMATION

Dry Gas Seal Systems play an important role in both the control and monitoring of every dry gas seal. There are multiple philosophies and options available to design and operate the system. The concepts discussed in this paper are the optimum solutions for the vast majority of applications, but in reality represent a small number of the options available. The entire system is a complex piece of equipment with many intricate components that must work together as one. A properly designed, monitored and maintained system can be a powerful tool for safeguarding the environment of a dry gas seal and extending the equipment run time.

NOMENCLATURE

DP = Differential Pressure
FIT = Flow Indicating Transmitter
H = High
HH = High High
PCV = Pressure Control Valve
PDCV = Pressure Differential Control Valve
PDIT = Pressure Differential Indicating Transmitter
PIT = Pressure Indicating Transmitter
RCFA = Root Cause Failure Analysis

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2) API 14.3.1: Orifice Metering of Natural Gas and Other Related Hydrocarbon Fluids – Concentric, Square-edged Orifice Meters, Fourth Edition September 2012, Errata July 2013.