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ABSTRACT

In today’s business environment, centrifugal compressor reliability is more critical than ever. Unscheduled equipment downtime has a negative impact on plant operations, oftentimes resulting in a temporary halt of production. In most centrifugal compressor applications, loss of production results in a substantial loss of revenue, sometimes measured in the hundreds of thousands or even millions of dollars per day. This lost revenue has a direct impact on the profitability of the plant.

There are tens of thousands of centrifugal compressors currently operating with outdated technology, resulting in increased maintenance costs and reduced reliability. However, product upgrades can improve compressor operation and increase reliability and availability by incorporating the latest technology into existing rotating equipment. The initial investment for these mechanical upgrades is very low in comparison to current operating and maintenance costs and the lost revenue resulting from unscheduled downtime. In most cases, the payback period is measured in months or even weeks.

This paper identifies the following mechanical (nonaerodynamic) compressor upgrades and discusses the technical and commercial advantages of each, calling on actual field examples to illustrate achieved benefits:

- Floating ring oil seals
- Dry gas seals
- Seal gas conditioning systems
- Damper seals
- Polymer labyrinth seals
- Noise attenuation

FLOATING RING OIL SEALS

Most old floating ring oil seal designs require substantial maintenance, are difficult to assemble, unreliable, and consume a substantial amount of seal oil. These oil seals can be upgraded to today’s state-of-the-art oil sealing technology, which can dramatically reduce oil consumption.

Features and Benefits

There are two sealing surfaces in a typical floating ring oil seal—the inside diameter of the seal rings and the mating surfaces between the seal rings and the seal housings. For a detailed description of floating ring oil seals and their support systems, refer to Wilcox (2000). Today’s floating ring oil seals include several features that make them superior to older seal designs in terms of performance, maintainability, and reliability (Figure 1). First and foremost is the windback feature of the inner seal ring. The windback feature improves the performance of the most critical sealing surface in the floating ring seal—the seal between the inside diameter of the inner seal ring and the shaft. The windback consists of a groove machined into the inside diameter of the inner seal ring (Figure 2), which creates a small counter-pressure when rotating, pumping the seal oil outboard toward the bearing, thus reducing sour seal oil leakage and oil migration into the compressor. Typical sour oil leakage rates are around 10 gallons per day per seal.
Older floating ring seals use O-rings on the sealing surface between the seal rings and the seal housing, and also to load the inner and outer seal rings in the axial position. Today's floating ring oil seals employ a high velocity oxygen fuel (HVOF) thermal spray hard surface coating on the mating surfaces of the seal rings instead of an O-ring. Further, the O-ring used to load the inner and outer seal rings in the axial position has been replaced with a spring. Replacement of these O-rings eliminates the risk of seal ring "hang up" caused by sticking O-rings, eliminates the risk of O-ring degradation, and reduces the wear rate of the ring and housing mating surfaces. The result is an improvement in the ability of the seal rings to remain concentric with the shaft and a longer life of the seal components, further reducing sour oil leakage.

In recent years, reduction of gas emissions from compressor seals has become a major effort for many compressor operators striving to become compliant with environmental regulations. Because the seal oil comes in contact with the process gas on the inboard side, gas becomes entrained in the oil. This sour oil is usually drained from the seal to a trap system, where the process gas is vented, usually to the plant flare system. Since the sour oil leakage is greatly reduced with a windback seal, the "load" on the trap package is reduced accordingly. This allows the trap vents to be isolated from the flare system, eliminating gas emissions. To further assure zero gas emissions from the compressor during normal operation, a nitrogen buffer injection can be added to the labyrinth seal inboard of the inner seal ring (Figure 3).

In the vast majority of cases, the new floating ring seals can operate with the existing seal support system, but this must be reviewed and confirmed when considering a seal upgrade project.

Feasibility Considerations

There are two primary technical issues that must be examined when considering the feasibility of a floating ring seal upgrade:

• **Existing seal cavity dimensions**—The new, upgraded floating ring seals must be able to fit within the existing seal cavity in the compressor. This is typically not a problem, but the original seal cavity dimensions must be known and the new seals designed to fit. In most cases upgraded seals can be installed without the need for reworks to the compressor shaft or heads. The seal change-out can be done during a normal compressor overhaul.

• **Existing oil seal support system**—In the vast majority of cases, the new floating ring seals can operate with the existing seal support system, but this must be reviewed and confirmed when considering a seal upgrade project.

Economic Justification

An oil seal upgrade is relatively easy to justify. Most payback periods are measured in just a few months or even weeks. Cost savings can be expected in the following areas after an oil seal upgrade:

• **Seal oil consumption**—The new windback seals will consume substantially less seal oil (either lost to the process by leaking into the compressor or lost to the sour oil sewer). This sour oil must be replaced by adding new oil to the seal oil reservoir to maintain the correct operating level. The cost of this replacement oil is easily quantified.

• **Seal oil disposal**—In some cases, the user chooses to dispose of the sour oil rather than treat it for reclamation. There is a cost associated with the disposal of sour oil, which will be reduced in direct proportion to the reduction of sour oil leakage.

• **Gas emissions**—As the sour oil leakage rate is reduced, the rate of gas emissions from the sour oil traps is reduced accordingly. Therefore, a cost saving is realized in terms of the reduction of process gas lost to flare.

• **Seal repair costs**—An alternative to upgrading floating ring oil seals is to restore the existing seals to "as new" condition by repairing or replacing seal components. This cost is avoided when replacing the existing seals with an improved design.
- Unscheduled downtime—As the existing seals wear they can become unreliable, oftentimes leading to unscheduled compressor downtime in order to perform maintenance on the seals. This unscheduled downtime and maintenance results in increased labor and maintenance costs and also lost revenue due to reduced or stopped production. These costs can be greatly reduced or eliminated by upgrading to the state-of-the-art floating ring seal design.

**DRY GAS SEALS**

A popular alternative to upgrading existing oil compressor seals is to replace them with dry gas seals. In addition to replacing the actual compressor seals, a dry gas seal upgrade also requires decommissioning and removal of the existing oil seal system, and installation of a new dry gas seal support system.

**Features and Benefits**

Dry gas seals represent the state-of-the-art in compressor shaft sealing technology. Nearly all process gas compressors manufactured today are equipped with dry gas seals. Dry gas seals are mechanical face seals, consisting of a mating (rotating) ring and a primary (stationary) ring. During operation, grooves in the mating ring generate a fluid-dynamic force causing the primary ring to separate from the mating ring creating a very narrow running gap between the two rings. Inboard of the dry gas seal is a labyrinth seal, which separates the process gas from the gas seal. A sealing gas is injected between the inner labyrinth seal and the gas seal, providing the working fluid for the running gap and the seal between the atmosphere or flare system and the compressor internal process gas. Dry gas seals are available in a variety of configurations, but the tandem gas seal (Figure 4) is typically applied in process gas service.

![Figure 4. Tandem Gas Seal.](image)

Dry gas seals offer many advantages over oil seals. The elimination of seal oil and the oil seal system results in reduced oil usage, elimination of the need to dispose of sour oil, reduced maintenance of the oil system, and improved safety (due to the elimination of flash point issues that occur when gas is absorbed into seal oil). Other advantages include the reduction of process gas losses due to seal emissions, reduced operating costs (parasitic power losses are much lower with gas seals), and improved seal reliability.

**Field Experience**

A refinery in the UK operates a model 2M8-6 wet gas compressor as part of their fluid catalytic converter (FCC) process. The compressor was equipped with oil film seals, and a nitrogen buffer gas was injected into the inboard labyrinth seal to prevent hydrogen sulfide (H₂S) contamination of the seal oil from the process gas. The refinery embarked on a project to upgrade to dry gas seals in order to eliminate the seal oil system and reduce nitrogen usage. The normal sealing pressure was a relatively low 66 psig, so the compressor manufacturer recommended a model L90B double opposed gas seal (Figure 5). The back-to-back configuration of the double opposed seal, compared to the traditional tandem gas seal, was very appealing to the refiner for the following reasons:

- Due to the low sealing pressure, an inert gas such as nitrogen can be used as the seal gas medium (instead of process gas from the compressor discharge, as is typical with tandem gas seals).
- Since the seal gas is injected between two gas seal elements, the total seal gas consumption is equivalent to the sum of the leakage of the two seals, which is very low. Nitrogen consumption was reduced from 137 scfm to about 6 scfm.
- Seal gas leakage into the process is very low, equivalent to the leakage of the inboard seal.
- Leakage to the outboard vent is also very low, contains no process gas (zero process gas emissions), and can be vented locally. Further, when nitrogen is used, the risk of an explosive mixture in the vent system is greatly reduced.
- The plant nitrogen source is very clean and dry, which greatly reduces the potential for gas seal contamination (a major concern when sealing with treated process gas), increasing gas seal reliability.
- There is no primary vent system, which simplifies the gas seal support system.

![Figure 5. Double Opposed Gas Seal.](image)

After the gas seal conversion, the compressor now operates with zero process gas emissions, and the complications associated with the old oil seal system have been completely eliminated.

**Feasibility Considerations**

There are several technical issues to consider when embarking on a gas seal upgrade:

- Existing seal cavity dimensions—The new dry gas seals must be able to fit within the existing seal cavity in the compressor. If the existing seal cavity is insufficient, the compressor heads may require machining to accommodate the gas seal. It may also be necessary to rework the shaft to add a drive mechanism for the rotating component of the gas seal. Finally, a shaft sleeve is usually required under the gas seal, which will need to be installed on the shaft.
- Existing seal porting—A tandem gas seal requires at least four gas ports: seal gas supply, primary vent, secondary vent, and barrier seal supply. If the existing oil seals have inadequate porting, compressor head machining may be required.
Compressor rotordynamics—A compressor rotordynamics study should be completed before commencing with a gas seal upgrade. Removal of the existing oil seals will result in reduced rotor damping, which can have an adverse effect on the rotordynamic characteristics of the compressor. These effects must be quantified with a complete lateral rotordynamics analysis. If the results of the rotordynamics analysis are unsatisfactory, additional rotordamping equipment such as damper bearings and/or damper seals (discussed later) may be required.

Compressor operating conditions and proposed seal gas—It is imperative to dry gas seal reliability that the compressor operating conditions and the proposed seal gas are thoroughly defined before designing the gas seal support system. Of primary concern is the potential for liquid condensation within the gas seals. This can be evaluated by constructing a phase diagram of the proposed seal gas (Stahley, 2001).

Economic Justification

Cost savings can be expected in the following areas after a dry gas seal upgrade:

- Seal oil consumption—With dry gas seals, seal oil consumption (either lost to the process by leaking into the compressor or lost to the sour oil sewer) is totally eliminated.
- Seal oil disposal—In some cases, the user chooses to dispose of the sour oil rather than treating it for reclamation. There is a cost associated with the disposal of sour oil, which is totally eliminated.
- Maintenance costs—Dry gas seal support systems are typically much simpler than oil seal systems. Reduced maintenance costs are usually realized after a gas seal upgrade.
- Gas emissions—Due to the very small running gap between the stationary and rotating rings in a dry gas seal, gas leakage to the primary and secondary vents is very low in comparison to oil seals. This results in cost savings in terms of reduced process gas emissions to flare.
- Operating costs—An energy savings is realized when operating with dry gas seals. There is a mechanical loss associated with oil shear forces inherent with oil film seals. The power loss due to gas shear forces in a dry gas seal is much less. In addition, the energy required to operate the seal oil pumps is totally eliminated.
- Seal repair costs—An alternative to installing dry gas seals is to restore the existing seals to “as new” condition by repairing or replacing seal components. This cost is avoided.
- Unscheduled downtime—Providing the gas seal support system has been designed correctly for the application, unscheduled downtime associated with unreliable oil seals can be reduced or eliminated. The costs associated with unscheduled downtime are reduced accordingly.

SEAL GAS CONDITIONING SYSTEMS

Contamination is a leading cause of dry gas seal degradation and reduced reliability. Ingress of foreign material (solid or liquid) into the narrow seal running gap between the seal’s rotating and stationary rings can cause degradation of seal performance (excessive gas leakage to the vent) and eventual failure of the seal. A primary source of gas seal contamination is the seal gas supply injected into the seal. Contamination from the seal gas supply occurs when the sealing gas is not properly treated upstream of the dry gas seal. Seal gas conditioning systems can be employed to reduce or eliminate contamination from the seal gas supply, thus increasing gas seal reliability/availability.

Features and Benefits

Seal gas conditioning systems typically include a prefiter/liquid separator, heater, and/or pressure booster, all mounted on a single baseplate installed upstream of the existing compressor gas seal system (typically mounted on a panel adjacent to the compressor). The system is intended to condition the seal gas before it flows into the compressor gas seal system in order to assure a certain gas quality at the gas seal. In regard to gas seal contamination from the sealing gas, there are three areas of concern: seal gas quality, composition, and pressure. Each of the problem areas can be addressed with a seal gas conditioning system:

- Seal gas quality—Gas seal manufacturers have stringent requirements for seal gas quality. Typically, the sealing gas must be dry and filtered of particles 3 micron (absolute) and larger. Coalescing-type filters are normally provided in the gas seal system to address this requirement, but such devices may be inadequate depending on the source of seal gas supply. At a minimum, a prefilter should be provided as part of the seal gas conditioning system in order to remove solid particles. If entrained liquids are present in the seal gas, a liquid separator should also be provided.
- Seal gas composition—“Heavy end” hydrocarbons and water vapor in the sealing gas will have a tendency to condense as the gas flows through the gas seal system. Components of the gas seal system such as filters, valves, orifices, and the seal faces themselves will cause seal gas pressure drops during operation. As the seal gas expands across these components, the Joule-Thomson effect will result in a corresponding decrease in the gas temperature. A gas heater can be provided as part of the seal gas conditioning system in order to superheat the seal gas and reduce the risk of liquid condensation.
- Seal gas pressure—The seal gas source must be available at sufficient pressure to cover the entire operating range of the compressor. The author has previously suggested a minimum seal gas source pressure of at least 50 psi above the sealing pressure (Stahley, 2001). This may not be achievable during some transient conditions. In these cases, a seal gas pressure booster can be provided with the seal gas conditioning system, or an alternate source of seal gas can be employed.

Field Experience

Seal gas conditioning systems have been field proven to improve gas seal reliability. Consider the following example.

Gas Gathering Compressors

A national oil company has three gas turbine-driven centrifugal compressors, operating in parallel, in gas gathering service offshore Southeast Asia. All these compressors were equipped with dry gas seals. Immediately upon commissioning, and in the first two years of operation, the user experienced multiple incidents of excessive gas leakage to the gas seal primary vents, resulting in numerous unscheduled shutdowns of all three compressors. The resulting gas seal repairs and production losses proved to be extremely costly. After an extensive engineering review and onsite investigation, it was determined that the gas seals were being contaminated by liquids and solids contained within the seal gas. The liquids appeared to be both entrained in the seal gas and the result of condensation as the gas experienced pressure drops throughout the gas seal support system. While the degree of contamination varied dramatically from incident to incident, liquids were observed in all the damaged gas seals. The seal gas filters provided with the gas seal support system were overwhelmed by the amount of contaminants within the gas, as can be seen in one of the more extreme incidents (Figure 6).

The solution to this gas seal reliability problem was to install a seal gas conditioning system. The system included the following features (Figure 7):

- The main source of seal gas was relocated from the discharge of each compressor to a common point in the platform discharge header piping, serving all three units. This was necessary to
The seal gas conditioning system as described above has been proven to be successful. There have been no incidents of high seal leakage/gas seal contamination since the system was installed.

Feasibility Considerations

A thorough engineering review of the seal gas supply source and the existing gas seal support system will determine the extent of seal gas conditioning required. It is important that the composition, quality, and source of the seal gas be well defined at the time of the system design. The pressure-temperature relationship of the seal gas must be considered to assess the potential for liquid condensation. This can be done by simulating the seal gas pressure and temperature drops expected across the various components within the gas seal system and plotting the results on a phase diagram of the seal gas (Stahley, 2001). Pending the results of the engineering study, the seal gas conditioning system may include one or more of the following components:

- Alternate seal gas source
- Prefilter
- Liquid separator
- Heater
- Pressure booster

Economic Justification

Unreliable gas seals can prove to be extremely costly, which simplifies the economic justification for installing a seal gas conditioning system. Costs savings can be realized in two areas:

- **Gas seal repair costs**—A damaged gas seal should be returned to a repair facility certified by the gas seal manufacturer for repair and testing. Since gas seal testing typically requires two seals be installed in the dynamic test rig, the mating seal (from the opposite end of the compressor) must also be returned to the repair facility even if it has not been damaged. Due to the tight machining tolerances required for most gas seal components, gas seal repair and testing can be quite expensive, sometimes approaching the cost of a new seal. These repair costs can be avoided by installing a seal gas conditioning system.

- **Unscheduled downtime**—Dry gas seal performance is typically assessed by monitoring the seal leakage to the primary vent. An increasing leakage trend is indicative of a deteriorating gas seal condition and eventual failure, leading to unscheduled downtime. There is a significant labor cost to remove both gas seals for return to the manufacturer for repair and also the lost revenue associated with reduced or stopped production. Upgrading to a seal gas conditioning system can eliminate these costs.

**DAMPER SEALS**

Rotordynamic instability in centrifugal compressors manifests itself in the form of subsynchronous rotor vibration, typically in high pressure applications. Such vibration can easily lead to damage of compressor internal parts or, at the very least, accelerated wear. Existing balance piston and/or division wall labyrinth or honeycomb seals can be replaced with a hole pattern damper seal (Figure 8) when increased rotor damping is required to improve rotor stability. The hole pattern damper seal offers rotordynamic benefits equivalent to or better than conventional honeycomb seals without the performance penalty.

**Features and Benefits**

Hole pattern damper seals are typically manufactured from the same aluminum material as a labyrinth seal, employ a single piece construction, and can usually be designed as a direct replacement for existing labyrinth or honeycomb seals without further modifications to the compressor.
The hole pattern damper seal offers many advantages over honeycomb and labyrinth seals. The hole pattern damper seal has been proven to effectively increase rotor damping. In fact, full-load, full-pressure factory testing of a back-to-back compressor has proven that the hole pattern damper seal can actually increase the rotor’s logarithmic decrement (log dec) as the discharge pressure increases, which is in direct contrast to previous industry experience (Moore, et al., 2002).

In addition to the rotordynamic benefits, the hole pattern damper seal offers a substantial operating benefit in improved compressor performance. Unlike honeycomb seals, the hole pattern damper seal is manufactured from the same aluminum material as a labyrinth seal. This allows the compressor designer to use the same shaft clearance for a hole pattern damper seal as is used for a labyrinth seal. When compared to the generous clearances required for honeycomb seals, the hole pattern damper seal can reduce internal seal leakage by 50 percent, which equates to a substantial power savings.

Further benefits of the hole pattern damper seal over a honeycomb seal include shorter manufacturing cycle times, improved rub tolerance, and increased structural strength.

Field Experience

A hole pattern damper seal can improve compressor rotordynamics characteristics and operating efficiency. This is illustrated by the field examples below.

Rotor Stability Improvement

A South American national oil company has three gas turbine-driven centrifugal compressor trains operating in parallel in gas gathering service. The compressors are a back-to-back design with a honeycomb seal on the division wall between the first and second section (Figure 9). When operating at the design conditions of about 1800 psia discharge pressure at just over 10,000 rpm, the unit exhibited a subsynchronous vibration (Figure 10). A hole pattern damper seal was installed at the division wall, replacing the honeycomb seal, and the subsynchronous vibration was eliminated (Figure 11).

Operating Efficiency Improvement

A gas turbine-driven centrifugal compressor was operating offshore South America in gas injection service. The compressor is a back-to-back design with a honeycomb seal on the division wall between the first and second section. Field testing conducted after the compressor was commissioned indicated the unit was in low efficiency, and the compressor was returned to the manufacturer for factory testing to diagnose and resolve the problem.

This particular application had a very low mass flow through the compressor. Factory testing confirmed that the internal leakage across the division wall seal (from the second section of compression to the first section) resulted in a larger than expected first section discharge mass flow. To resolve the problem, the honeycomb division wall seal was replaced with a hole pattern damper seal. Subsequent factory testing confirmed that the hole pattern damper seal was successful in reducing the internal leakage to the first section.

As shown in Table 1, comparing factory test results with a honeycomb seal to factory test results with a hole pattern damper seal, the division wall leakage with the honeycomb seal was nearly 6 percent of the total mass flow through the compressor. Installation of the hole pattern damper seal reduced this leakage to less than 3 percent of the total mass flow through the compressor. This 50 percent reduction in division wall seal leakage results in compressor power savings of just over 3 percent, or about 200 hp, yielding an enormous operating cost reduction over the life of the compressor.
Feasibility Considerations

Hole pattern damper seals can be applied to most compressors to eliminate or reduce subsynchronous rotor vibration levels and/or improve compressor performance by reducing interstage seal leakage. The following points should be evaluated when considering a hole pattern damper seals upgrade:

- **Existing honeycomb or labyrinth seal design**—The attachment method of the existing seal must be evaluated for installation of a hole pattern damper seal. This can usually be accomplished without the need for machining of the existing compressor internal components.

- **Process gas composition**—The gas composition must be evaluated for potential corrosion or erosion of typical aluminum seal material. It may be necessary to use an alternate material, such as a polymer, which has been successfully applied in the field.

- **Seal differential pressure**—The differential pressure across the hole pattern damper seal may affect the seal material selection. A stronger grade material may be required for cases of high differential pressures.

Economic Justification

A hole pattern damper seal upgrade is a low cost solution to very complicated operating problems, and is therefore relatively easy to justify. The payback period is typically very short. An economic impact can be expected in the following areas after a hole pattern damper seal upgrade:

- **Improved operating efficiency**—A hole pattern damper seal will reduce the interstage leakage when replacing an existing division wall or balance piston labyrinth seal. The resulting power savings can be substantial (refer to the previous field example).

- **Increased production**—If the compressor operating range is restricted by an excessive subsynchronous rotor vibration level, a hole pattern seal can be applied to reduce or eliminate the problem. The compressor can then be operated across its full performance map, increasing plant output and sales revenue.

- **Extended operating life of internal wear parts**—If left unchecked, an excessive subsynchronous rotor vibration can reduce the operating life of compressor wear parts such as internal labyrinth seals, bearings, and process seals. Reducing or eliminating the vibration can avoid this accelerated wear and subsequent cost of replacement parts.

- **Alternate remedies**—Alternative methods of reducing subsynchronous rotor vibration levels, such as damper bearings or tilt-pad seals, are much more expensive than a hole pattern damper seal retrofit.

POLYMER LABYRINTH SEALS

Labyrinth seals are an integral part of any process compressor. A labyrinth is a noncontacting seal that uses a tortuous path to restrict gas leakage. A pressure drop occurs at each labyrinth tooth as the gas is squeezed between the labyrinth tooth and the rotor. A labyrinth’s sealing efficiency is directly proportional to its clearance over the rotor. Labyrinths are used as shaft end seals, impeller eye seals, interstage shaft seals, balance piston seals, and division wall seals (in back-to-back compressors). Engineered polymer (thermoplastic) labyrinth seals can replace existing metallic (usually aluminum) labyrinth seals to provide increased wear and corrosion resistance.

### Features and Benefits

The primary benefit of polymer labyrinth seals is their inherent wear resistance. Polymer labyrinths have the ability to deflect and return to their original shape after “touching-off” with the compressor rotor (i.e., elastic deformation). This allows the labyrinth clearance to be maintained after contact with the rotor. In contrast, when an aluminum labyrinth is subjected to a rotor rub, the tips of the labyrinth teeth can be worn as the softer aluminum material makes contact with the harder steel rotor (i.e., plastic deformation). The result is a permanent increase in the labyrinth clearance, which increases gas leakage across the labyrinth and decreases operating efficiency. Additionally, aluminum labyrinths can sometimes gall the compressor rotor, damaging shaft sleeves and further increasing the clearance. Rotor galling is not an issue with polymer labyrinths.

It has been estimated that efficiency gains of 0.5 percent to 1 percent per impeller can be realized with most polymer labyrinth upgrades (Whalen, 1994). This rule of thumb is often misinterpreted as a dramatic improvement in efficiency. For example, a polymer labyrinth upgrade for a compressor with six impellers is often mistakenly expected to result in an overall efficiency improvement of 3 percent (six impellers times 0.5 percent efficiency gain per impeller). However, a 0.5 percent efficiency gain for each of the six impellers results in an overall compressor efficiency improvement of 0.5 percent. Given the accuracy and availability of instrumentation in most compressor installations, it is very unlikely that field testing could detect this level of overall efficiency improvement.

These immediate efficiency gains are based on the assumption that installed polymer labyrinth clearances can be designed to be tighter than the existing aluminum labyrinth clearances, due to polymer’s ability to deflect during rotor contact. However, in the author’s experience, this is not always a valid assumption. In many cases, it is not possible to reduce the installed labyrinth clearance when using polymer materials due to thermal growth issues (discussed later).

While an immediate improvement in operating efficiency may or may not be observed, polymer labyrinths will provide a long-term efficiency gain. As explained previously, as aluminum labyrinths make contact with the rotor, they will wear over time until they reach a point of maximum running clearance and minimum efficiency. The compressor will be operated under these conditions until the next scheduled shutdown, when the aluminum labyrinths can be replaced. In this regard, polymer labyrinths are clearly superior. Polymer labyrinths will wear at a rate much slower than aluminum, and thus will maintain a tighter running clearance and higher efficiency over a longer period of time (this is displayed graphically in Figure 12). From this standpoint, the use of polymer labyrinths could increase the time between scheduled compressor turnarounds.

![Figure 12. Comparison of Aluminum Labyrinth Efficiency Versus Polymer Labyrinth Efficiency over Time.](image-url)
A second major benefit of polymer labyrinths is their corrosive resistance in certain operating environments. Certain elements, such as mercury and hydrogen sulfide, are corrosive to aluminum, and can attack and destroy the labyrinth (Figure 13). Polymer labyrinths can offer improved corrosive resistance in these cases, thereby increasing compressor reliability. PEEK (poly-ether-ether-ketone) material, for example, has excellent chemical compatibility characteristics.

Figure 13. Corrosive Attack of an Aluminum Labyrinth.

Field Experience

The case study below illustrates the corrosion resistance benefits of polymer materials.

Gas Injection Compressors

A private oil company has three gas turbine-driven centrifugal compressor trains, operating in parallel, in gas injection service offshore Southeast Asia. There are two compressors in each train, and each compressor is a back-to-back design. Final discharge pressure is about 1800 psig at about 9000 rpm. All these compressors were originally supplied with aluminum labyrinth seals.

After a short period of operation, it was determined that traces of mercury were present in the process gas (mercury was not identified in the original process gas composition). Mercury is known to attack aluminum, and this was believed to be the reason for accelerated labyrinth wear. The aluminum labyrinths became very brittle after exposure to mercury, and were easily damaged after making contact with the rotor. Polymer labyrinths were suggested as a means of increasing the useful life of the labyrinth seals. Since PEEK material is known to be resistant to mercury, all the aluminum labyrinths were replaced with PEEK labyrinths in early 2002.

As stated previously, it is not always possible to reduce installed clearances when replacing aluminum labyrinths with polymer labyrinths due to the thermal expansion of the polymer material. In this case, the installed polymer labyrinth clearances were increased compared to the original aluminum labyrinth clearances (Table 2). These compressors continue to operate today with no indication of corrosive attack of the polymer labyrinths.

Table 2. Increased Polymer Labyrinth Clearances Compared to Original Aluminum Labyrinths.

<table>
<thead>
<tr>
<th>Labyrinth Location</th>
<th>Increase in Polymer Labyrinth Clearance (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner Seal</td>
<td>50</td>
</tr>
<tr>
<td>Shaft</td>
<td>50</td>
</tr>
<tr>
<td>Impeller Eyesh (typical)</td>
<td>43</td>
</tr>
<tr>
<td>Division Wall</td>
<td>33</td>
</tr>
</tbody>
</table>

Feasibility Considerations

A thorough engineering review is key to a successful polymer labyrinth upgrade. There are several feasibility considerations:

- **Structural integrity of the labyrinth seal**—Polymer materials are not as strong as aluminum, particularly at elevated temperatures, so the structural integrity of the labyrinth geometry must be considered before upgrading to a polymer material. This is of particular concern for applications that are subject to high differential pressure loading, such as balance piston and division wall labyrinth seals. The geometry of the labyrinth, or the attachment method of the labyrinth may need to be modified to accommodate a polymer material.

- **Compressor operating conditions**—The compressor operating conditions must be considered when selecting an appropriate polymer labyrinth. The differential pressures across the labyrinth teeth must be evaluated for structural integrity as discussed above. The operating temperature may affect the polymer material to be used. PEEK, for example, is normally limited to about 250°F. The process gas composition can also affect the selection of a polymer material. The polymer material must be chemically compatible with the process gas. If other chemicals, such as cleaning solvents, are injected into the process, they too must be reviewed for compatibility.

- **Installed labyrinth clearances**—It should not be assumed that polymer labyrinths can always operate at tighter installed clearances than aluminum labyrinths. There are several factors to be considered when establishing the polymer labyrinth clearances. Since polymers typically have a much higher coefficient of thermal expansion than aluminum, the thermal growth of the labyrinth, its mating parts, and the rotor needs to be considered. Rotor sag should also be considered when determining optimal labyrinth clearances.

Economic Justification

Polymer labyrinth seals are usually more expensive than equivalent aluminum labyrinth seals, and the economic benefits, while real, are often difficult to quantify. However, economic benefits will be realized in the following areas:

- **Increased long-term operating efficiency**—As mentioned previously, an immediate efficiency improvement is not a given with a polymer labyrinth upgrade. Any immediate gains in efficiency would be small and difficult to measure in most field applications. However, there is a definite long-term efficiency gain over aluminum labyrinths due to the slower wear rate and sustained operation at design labyrinth clearances.

- **Improved corrosion resistance**—If the existing aluminum labyrinths are subject to premature wear due to corrosive attack, polymer labyrinths can offer increased corrosive resistance and therefore increase the operating time between turnarounds. This results in increased production and reduced maintenance costs.

NOISE ATTENUATION

In a centrifugal compressor, the dominant noise component is typically generated by high velocity gas in the impeller exit/diffuser entrance region, and occurs at the impeller blade passing frequency. This noise level increases when vanes are installed in the diffuser due to the aerodynamic interaction between the impeller and the diffuser vanes. Recent advances in acoustical technology have made it possible to reduce noise and vibration emanating from centrifugal compressors (Liu and Hill, 2001). A duct resonator array has been shop tested and field proven to reduce sound pressure levels.

Features and Benefits

Noise control can be addressed in three ways:

- By reducing the strength of the noise at its source
• By minimizing the transmission paths (e.g., covering the noisy structure with an acoustic insulation or an enclosure)
• By interrupting or minimizing the noise before it reaches the receiver (e.g., wearing earplugs)

For turbomachinery, the historical approach to noise control has been to build a barrier between the noise source and receiver with items like enclosures, lagging, or acoustic insulation. This approach is typically very expensive, increases weight, reduces accessibility, and does nothing to address the source of the noise. Reducing noise at the source is always the most desirable solution, but often more difficult to implement.

A duct resonator array attacks the noise at its source. To accommodate a duct resonator array in a centrifugal compressor, a recess is cut on each side of the diffuser wall and the array is rail-fitted or bolted to the diffuser wall (Figure 14). The volume, cross-sectional area, length, and other geometric parameters are specifically designed for each application. This allows for tuning of the duct resonator array to deliver maximum sound reduction at the compressor blade passing frequency, where the noise level is generally of the highest magnitude.

Excessive compressor noise can oftentimes result in vibration of the process piping, which can lead to structural damage and failure of pipe-mounted instrumentation. By reducing the noise level at its source, the duct resonator array also reduces the resulting pipe vibrations, reducing the potential for structural damage and instrumentation failures.

Field Experience

Duct resonator arrays have been field proven to reduce compressor noise and piping vibration in both single and multistage compressors. Consider the following example.

Gas Pipeline Compressor

A Canadian gas transmission company was operating a single-stage compressor with an overhung impeller design. The compressor was originally provided with diffuser vanes in order to achieve the desired aerodynamic performance. However, excessive compressor noise and piping vibrations resulted from the impeller-to-diffuser vane interaction. The excessive noise presented an environmental issue and the resulting piping vibrations had damaged locally mounted instrumentation. In order to reduce the noise and piping vibration levels, the user decided to remove the diffuser vanes. While this achieved the desired result, it also reduced the compressor efficiency and operating range and was therefore not an effective long-term solution.

A duct resonator array was recommended as a means to resolve the noise and piping vibration problems, and still achieve the desired aerodynamic performance. A duct resonator array was designed specifically for this application and installed in September 2002. Diffuser vanes were also installed back into the compressor at the same time. An extensive field testing program was conducted immediately after installation of the duct resonator array (baseline testing was conducted before installation of the duct resonator array). Field testing included the measurement of sound pressure levels by four microphones mounted 3 feet from the compressor piping, and the measurement of piping vibration by nine accelerometers mounted in various locations on the piping. Field data were collected at nine different operating conditions at various flow rates and speeds, matching the baseline conditions as closely as possible.

Field testing confirmed that the duct resonator array was successful in reducing both noise and piping vibration. The sound pressure level was reduced by more than 10 dB at all nine operating points, with an average reduction of 12.9 dB (refer to Table 3). Discharge pipe vibration levels were reduced from 66 percent to 92 percent, with an average reduction of 83 percent (refer to Table 4). Further, there was no measurable effect on the aerodynamic performance of the compressor. The compressor now operates over its full design range and efficiency at a lower noise level than a parallel compressor with a vaneless diffuser operating in close proximity (Figure 15).

### Table 3. Reduction of Sound Pressure Level as Measured During Baseline and Final Field Testing.

<table>
<thead>
<tr>
<th>Test Point</th>
<th>Sound Pressure Level Reduction (dBA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>11.2</td>
</tr>
<tr>
<td>2</td>
<td>11.6</td>
</tr>
<tr>
<td>3</td>
<td>12.8</td>
</tr>
<tr>
<td>4</td>
<td>10.1</td>
</tr>
<tr>
<td>5</td>
<td>15.7</td>
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<tr>
<td>6</td>
<td>14.5</td>
</tr>
<tr>
<td>7</td>
<td>14.6</td>
</tr>
<tr>
<td>8</td>
<td>12.1</td>
</tr>
<tr>
<td>9</td>
<td>13.4</td>
</tr>
</tbody>
</table>

### Table 4. Comparison of Piping Vibration Levels Measured During Baseline and Final Field Testing.

<table>
<thead>
<tr>
<th>Test Point</th>
<th>Baseline (g/s)</th>
<th>Final (g/s)</th>
<th>Reduction (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>55.3</td>
<td>6.8</td>
<td>88</td>
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<td>2</td>
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<td>6.0</td>
<td>91</td>
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<td>6.8</td>
<td>84</td>
</tr>
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<td>6</td>
<td>41.9</td>
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<td>66</td>
</tr>
<tr>
<td>9</td>
<td>34.4</td>
<td>2.8</td>
<td>92</td>
</tr>
</tbody>
</table>

Feasibility Considerations

A duct resonator array can be applied to almost any compressor to reduce noise and/or process piping vibration levels. However, the following issues must be considered when assessing the feasibility of a duct resonator retrofit:

- **Existing diffuser wall size**—There must be sufficient radial space in the existing diffuser wall in order to accommodate the duct resonator array. Furthermore, the existing diffuser wall must be thick enough to allow machining of the recess required for the installation of the duct resonator array, while still maintaining structural integrity.
• Existing diffuser width—The existing diffuser width must be considered before applying a duct resonator array. Factory testing has shown that special measures must be employed to properly apply a duct resonator array when the existing compressor diffuser width is very narrow.

Economic Justification

To develop an economic justification for a duct resonator array retrofit, one must compare the costs of the retrofit project to the costs and effectiveness of other noise attenuation methods:

• Cost of doing nothing—There is a cost associated with taking no action to reduce noise or piping vibration. This cost could be in the form of financial penalties for noncompliance with federal environmental regulations or local ordinances. If piping vibration is an issue, there will be a cost associated with the damage or failure of locally mounted instrumentation.

• Cost of removing the compressor diffuser vanes—While this may be effective at reducing noise and vibrations, there is an associated cost in the form of reduced compressor efficiency and/or reduced compressor operating range. This can result in increased operating expenses and reduced process output.

• Cost of traditional noise attenuation methods—As mentioned previously, the traditional approach to turbomachinery noise control has been to install enclosures, lagging, or acoustic insulation on the compressor, creating a barrier between the noise source and receiver. This approach is typically more expensive and less effective than a duct resonator array because it does not address the source of the noise. There is also an increased maintenance cost associated with physical barriers, as they reduce accessibility to the equipment.

CONCLUSION

Continuous technological advancement has resulted in the availability of numerous compressor upgrades. These state-of-the-art designs can be applied to most existing centrifugal compressors, and can usually be installed during a regularly scheduled turnaround at relatively low cost. As has been demonstrated in this paper, when properly applied, these product upgrades can result in the resolution of chronic field problems, reduced operating costs, and improved equipment reliability and availability.

REFERENCES


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