48TH TURBOMACHINERY & 35TH PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 9-12, 2019
GEORGE R. BROWN CONVENTION CENTER

DRY GAS SEAL FAILURE SIMULATION METHODOLOGY

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ABSTRACT
The failure of dry gas seals is a critical event in oil & gas industry which if not correctly managed can lead to uncontrolled gas release to atmosphere. It is important to properly estimate the potential risks due to the failure and find appropriate solutions to mitigate them. In the past few years an increasing interest in the topic has been demonstrated by both manufacturers and end-users. The recent introduction of the API 692, covering the minimum requirements for dry gas sealing system for axial, centrifugal, rotary screw compressors and expanders, confirmed that this event can be taken as reference for a safer design of the seal gas system.
The new normative clarifies which are the crucial elements that shall be verified in a dry gas seal failure event and it gives hints for the analysis to be performed in case a vent study is needed.
The methodology developed to evaluate the system behavior in case of dry gas seal failure is described in the present paper. The analysis is performed with a process simulation software and it is capable of estimating pressure profile within the dry gas seal cartridge, the centrifugal compressor casing and the surrounding system. Moreover, outcome of the calculation is also the flow distribution of process gas within venting system or systems adjacent to the sealing one.
With such analysis it is possible to highlight critical points and evaluate mitigation measures, with the final scope of making sure that the sealing system design is robust and aligned to the most stringent API 692 requirements.
Main assumptions and failure modes that may be expected according to manufacturer's experience are covered, giving also details on the simulation model and its components, specifically created for the purpose of this analysis. Results that can be obtained and the acceptance criteria to be fulfilled are then described together with the most effective mitigation measures, selected thanks to the expertise gained in these years.
Finally, a case study is presented showing how the methodology was applied to revamp an old system design to comply with new End-User requirements based on API 692, implementing corrective actions to further mitigate risks in the event of a dry gas seal failure.

1. INTRODUCTION
The first edition of API 692 "Dry Gas Sealing Systems for Axial, Centrifugal, Rotary Screw Compressors and Expanders" was released in June 2018. This new standard covers the minimum dry gas sealing system requirements in association with compressors and expanders for use in the petroleum, chemical, and gas industry services. This regulation consolidates the DGS design approach among End-Users, OEMs and DGS suppliers experience and introduces new requirements to further improve the level of reliability and safety.
One of the components impacted by the new standard is the separation seal: over its task to separate the DGS from lube oil in normal operation, the regulation requests that it contributes to contain gas in a scenario of simultaneous failure of inboard and outboard rings. In particular, the regulation requires that in the event of a dry gas seal failure, the separation seal shall be designed to maintain sealing integrity and limit the amount of process gas entering the bearing housing. In addition, it provides guidelines to conduct a vent study to establish the vent capability in case of a seal failure.
This is a new approach to the design, as previously the common practice, in designing the DGS system, was to consider the failure of only one seal at the time (primary or secondary), having the other one as back-up for the shut-down transient.
During the last years, to satisfy end-users increasing requirements on dry gas seals safety and at the same time anticipating new API 692 standards introduction several steps have been taken: the present work gives an overview of a methodology that was developed to estimate the system behavior in case of dry gas seal failure, at the same time providing basis for improvement in the separation seal mechanical design and detailed analysis of seal failure mode. In particular, this paper refers to the case of tandem dry gas seals, but the methodology is applicable to other configurations.
The paper is organized as follows: failure mode and assumptions are presented in section 2; section 3 and section 4 give more detailed explanations of simulation boundaries, methodology and main components' modelling; the analysis outcomes and their acceptance criteria are described in section 5; section 6 lists some of the mitigation measures that can be used in case there criteria are not fulfilled; an exemplifying case study is reported in section 7; section 8 and section 9 elaborate on future developments and conclusions.

2. FAILURE MODE AND ASSUMPTIONS
The failure mode of each item is assumed to achieve the worst scenario in terms of leakage of process gas from the compression loop to the dry gas seal area. Figure 1 shows a simplified DGS cross section and the assumed failure modes.
The regulation requires to simulate seal failure in such a way that *seal faces provide no restriction*, i.e. considering the complete absence of the rings. This assumption is the most conservative approach as it leads to the maximum flow leakage. The failure modes observed by OEM in real cases not always show this behaviour, as depicted in Figure 2, case (a) where rings were reduced to debris, causing a certain restriction to the flow. On the other hand, Figure 2, case (b) gives an example of a failure in line with the regulation, where rings were completely transformed to dust. This assumption can be discussed case by case with DGS manufacturer or End-User; if no indications are given, the API 692 assumption of 'no restriction' is used.

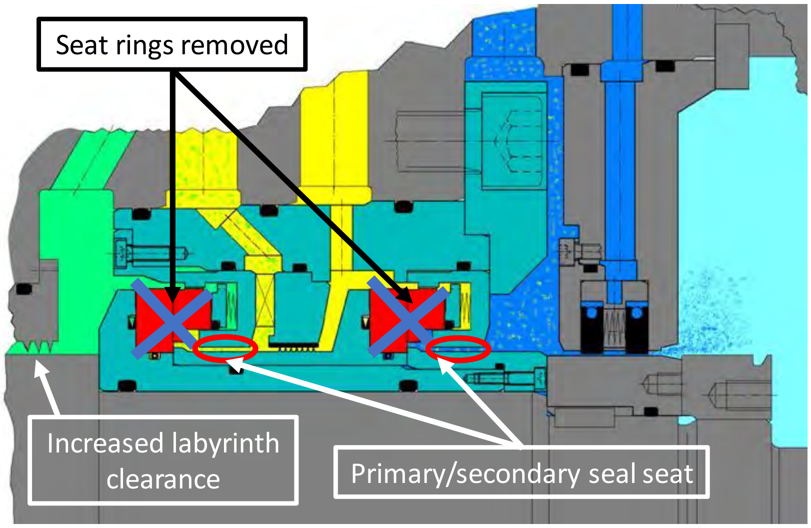

Figure 1. Dry Gas Seal section drawing with failure modes.

(a)

(b)

Figure 2. Two Examples of Failed Dry Gas Seals.

For the labyrinths (process, intermediate and separation seal if applicable) the worst assumption, also indicated by the regulation, considers a clearance equal to two times the maximum clearance at assembly. This assumes that the root cause that determines the catastrophic scenario leads also to damage of the seal labyrinths. Also for the labyrinths, failure mode can be discussed case by case with DGS manufacturer or End-User.

The metal parts of the dry gas seal cartridge, such as carriers and holes, are considered intact in the analysis.

During the failure event the process gas will flow from the compression loop towards the dry gas seal through the process labyrinth and the gap between retainer and shaft sleeve of the primary seal. Then, part of the flow goes to the primary vent and the remaining gas flows towards the gap downstream of the secondary seal. Since the gas flow increases compared to normal operating condition, the pressure in the secondary vent cavity may increase depending on the capacity of the vents. The increased pressure acts on the separation seal which, according to API 692, has to maintain its mechanical integrity to limit the amount of process gas entering the bearing housing. In case bushing separation seals are adopted a standard failure mode cannot be considered since it depends on the seal mechanical arrangement. First of all, it is important to assess if the carbon ring is able to withstand the increment of pressure. Thus, a preliminary simulation is performed considering the separation seal is intact to evaluate the maximum pressure that can build up upstream of the separation seal. If the pressure is higher than the maximum allowable for the carbons, it will be assumed that the carbon rings suffer shear failure and the process gas flows through the gap between the remaining metallic clearance. The following step is to evaluate if the assembly device that connects the separation seal to the dry gas seal cartridge is able to withstand the increased pressure.

If the separation seal and the main dry gas seal are packed together with screws, the stress on the screws is calculated and compared with the maximum allowable stress. The failure of the screws will lead to an axial displacement of the separation seal and to a consequent bypass between the secondary seal cavity and the separation seal injection cavity. An alternative configuration is that the main dry gas seal and the separation seal are packed together with a shear ring on stationary part designed for full pressure.
Regarding the clearance of the carbon ring, it may increase in case of non-contacting bushing, while it may decrease closing the gap between the ring and the shaft. In any case the higher pressure upstream of the separation seal may lead to an axial movement of the internal carbon ring.
However, the details of the separation seal failure mode are usually discussed for each specific case with the seal vendor, since it depends on the design of the seal.

3. SIMULATION BOUNDARY

In order to properly simulate dry gas seal failure event, the first step is the definition of the boundary limits of the simulation model. As depicted in Figure 3, the typical gas inlet and outlet of a dry gas seal system are the following:

- <u>*Inner Compressor*</u>: the main gas release source upon DGS failure is the internal part of the compressor as the only containment element is the process labyrinth. Gas inside the compressor is usually considered to be a constant pressure source of gas at Settle Out Pressure, being the most conservative condition, or at nominal suction pressure in case of normal operating scenario analysis (not covered by the present document).
- <u>*Seal gas*</u>: coming from an external source at a given pressure and composition, or from compressor discharge in case of auto-buffer. Seal gas source is also considered to be at constant pressure which again depends on the scenario analyzed as for the inner compressor.
- <u>*Primary vent (to flare)*</u>: it is the first venting point considered to be at constant pressure equal to the maximum foreseen back-pressure. Primary vent piping up to vent header is included in the simulation model to take into account pressure drops up to the point where the pipe diameter increases and the pressure can be considered constant up to the flare. If vent header dimensions are available, it can be included in the simulation model as well for a better predictability of the back-pressure.
- <u>*Secondary vent (to atmosphere)*</u>: it is simulated up to the point where no back-pressure can be assumed.
- <u>*Secondary seal gas and separation gas*</u>: boundary point is placed on the gas header (usually nitrogen) considered to be at constant pressure and from where gas is taken under pressure control and sent to the DGS.
- <u>*Final outlet*</u>: after the separation seals, the gas released during a DGS failure event flows to the bearing house and then to the lube oil console. Usually, the last boundary point of the DGS failure simulation model is placed just after the separation seals. The model calculates the flow of gas migrating to the oil console as output. Depending on the scope of the analysis, this result can be then used to extend further the scope of the analysis including the lube oil system where the main outcome of interest is the level of pressurization the oil console reaches due to the gas flow released from the DGS system. The present work focuses on the simulation of the DGS only so hereafter it will be considered to have the last boundary on the separation seals where the gas flow migrating to the oil console is the output of the simulation.

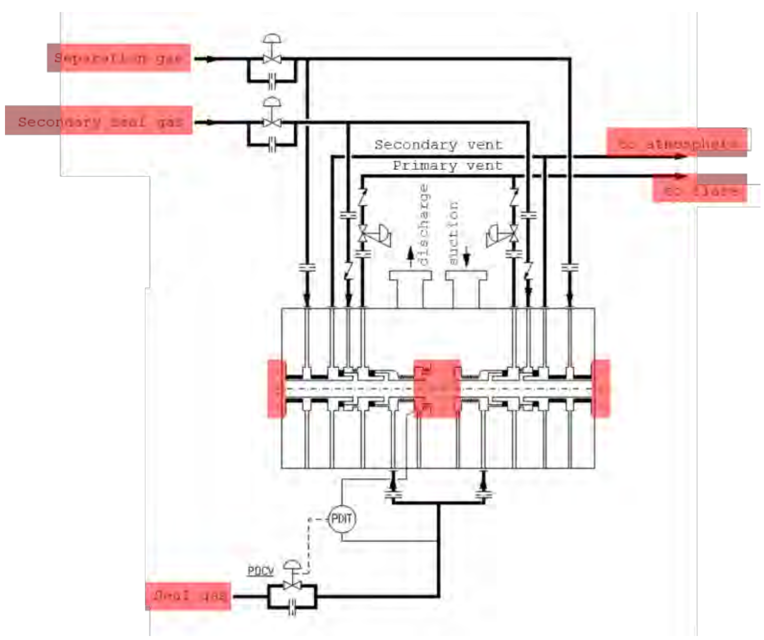

Figure 3. Boundary limits of Dry Gas Seal system simulation model. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

4. METHODOLOGY

Between the boundaries just described, the simulation model includes all the restriction elements that mainly affect the gas distribution toward the various release points. In the following paragraphs, each element type is described, together with the criteria for its modeling, underlining the key elements that more affect the overall simulation results.

Piping

Seal gas, primary vent, secondary vent, secondary seal gas and separation gas line pipes shall be implemented in the simulation model, paying particular attention to secondary vent where the gas flow may be quite high and pressure drops due to pipe frictions are the only ones present: length, diameter and roughness of the pipe may significantly affect the calculated gas flow and so the pressurization level reached in the secondary vent chamber.

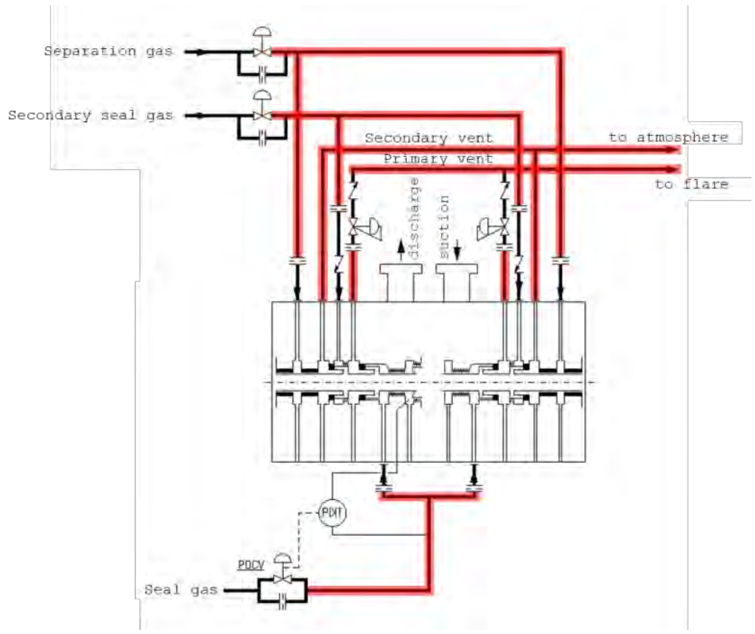

Figure 4. Pipes simulated in the DGS failure model. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

When the compressor pressure is high, the gas flow in the secondary vent may reach sonic condition, requiring a dedicated friction model to take into account this flow regime.

In case of standard flow (i.e. not sonic) the Moody diagram [3] can be used to estimate the pressure drops across the pipe, using the following equation:

$$\Delta P = \left(f \cdot \frac{L}{D} + K\right) \cdot \rho \cdot \frac{v^2}{2} \quad \textbf{Eq. 1}$$

where K is the resistance coefficient of the fittings installed in the pipe. Typical values of K coefficient for some fittings are reported in Table 1.

Table 1. Typical values of resistance coefficient ([3], [4])

type of fitting	K value
45° elbow (standard)	0.35
45° elbow (long radius)	0.20
90° elbow (standard)	0.75
90° elbow (long radius)	0.45
90° elbow (square or miter)	1.30
Sudden contraction	0.50
Sudden enlargement	1.00

Several equations are present in literature that fit the Moody diagram.
Eq. 1 is valid only for incompressible fluids, hence when the sonic condition is approached the equation relevant to adiabatic compressible flow with friction in a duct of constant cross section shall be used [3].

Orifices

Orifices installed on seal gas, primary vent, secondary seal gas and separation gas lines usually represent the main source of pressure drop on these parts of the system. If that is the case, orifices may be the only element to be simulated to properly estimate the gas flow in those lines.

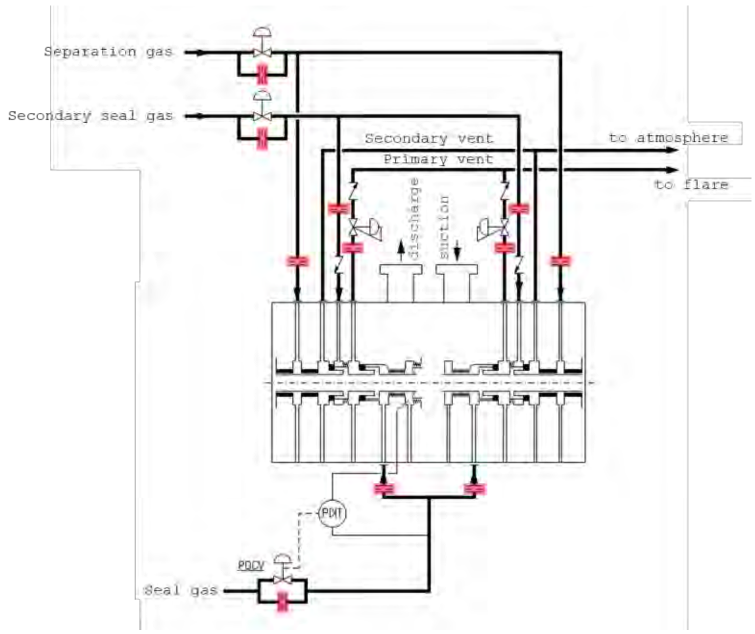

Figure 5. Orifices simulated in the DGS failure model. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

Moreover, a sonic condition is usually established across orifices, hence the equation in [5] can be used to simulate these elements taking into account choked flow.

Labyrinths

Upon dry gas seal failure, labyrinths remain the main elements that segregate the various sections of the dry gas seal cartridge. Labyrinths present in the system are:

- **Process labyrinth**: it is the main element that affects the gas flow coming from the inner compressor
- **Intermediate labyrinth** (not always present): it is installed between primary vent and secondary seal gas injection avoiding process gas migration into the secondary vent during normal operation. Upon DGS failure it provides additional resistance to the flow release from inner compressor
- **Separation labyrinths** (if present): they can be the main responsible for the amount of flow which leaks towards the oil console.

Labyrinths' model calculates the mass flow across the labyrinth as a function of:

- Radial Clearance
- Teeth Pitch
- Labyrinth Diameter
- Number of Teeth
- Labyrinth type (stepped or straight)
- Teeth type (straight or oblique)

The model is valid and provides a good estimation of the flow passing through the labyrinth only when the pressure drop across the labyrinth is small, or anyhow whenever the flow condition is sub-sonic. During DGS failure, labyrinths (especially process labyrinth) could experience a very high pressure drop and the flow could fall to sonic condition. In such cases, the gas flow tends to be overestimated (in some conditions up to the double, according to experience).When pressure drop across the labyrinth is high, it is usually better to move to a more accurate labyrinth model based on experimental data.

Valves

The universal gas sizing model is used to simulate the valves usually present in the seal gas, primary vent, secondary seal gas and separation gas lines. The control logics that regulate the valves opening is also implemented in the DGS failure simulation model in order to properly take into account the behavior of the controllers during a DGS failure.

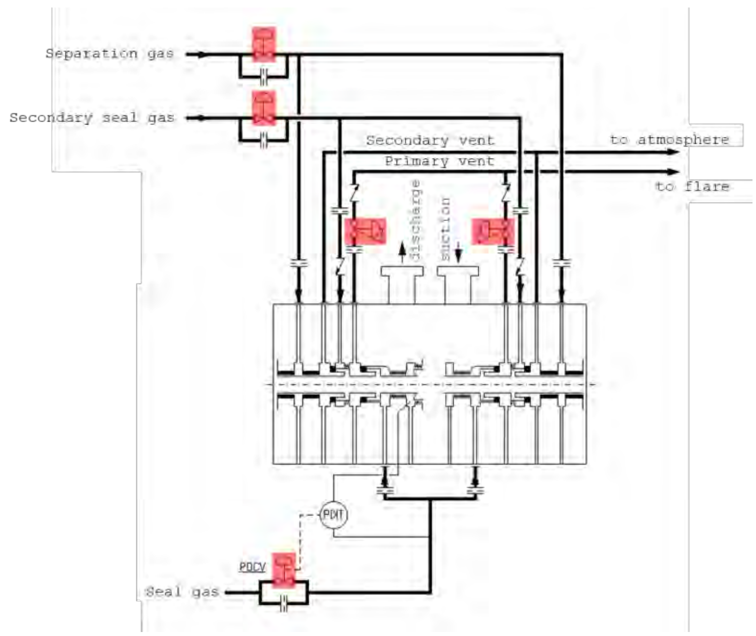

Figure 6. Valves simulated in the DGS failure model. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

Casing holes

Holes in the compressor casing also provide resistance to the flow, especially in the primary and secondary vent and so they shall be properly taken into account in the simulation model.

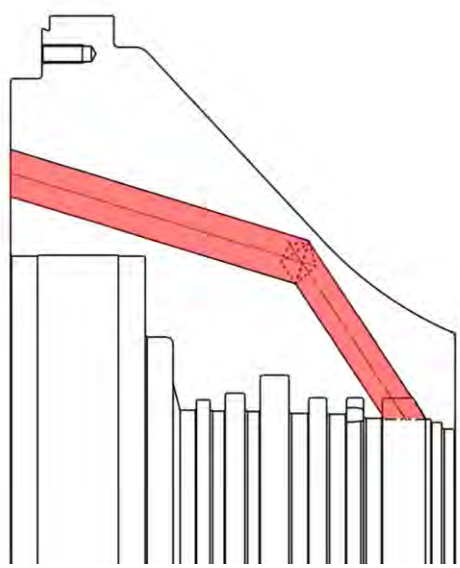

Figure 7. Example of casing hole simulated in the DGS failure model. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

Since compressor casing holes are simple ducts with circular cross section, they are modeled as pipes, taking into account elbows, contractions and enlargements.

General restrictions

All the other geometries present in the system that do not fall into the previous categories, are modeled with a general equation based on the piping equation (Eq. 1) where the pipe diameter is replaced by the Equivalent Diameter defined as:

$$D_{eq} = \frac{4A}{p} \qquad \textbf{Eq. 2}$$

where:

- p is the wet perimeter;
- A is the cross-sectional area.

This equation is typically used for all the restrictions inside the dry gas seal cartridge.

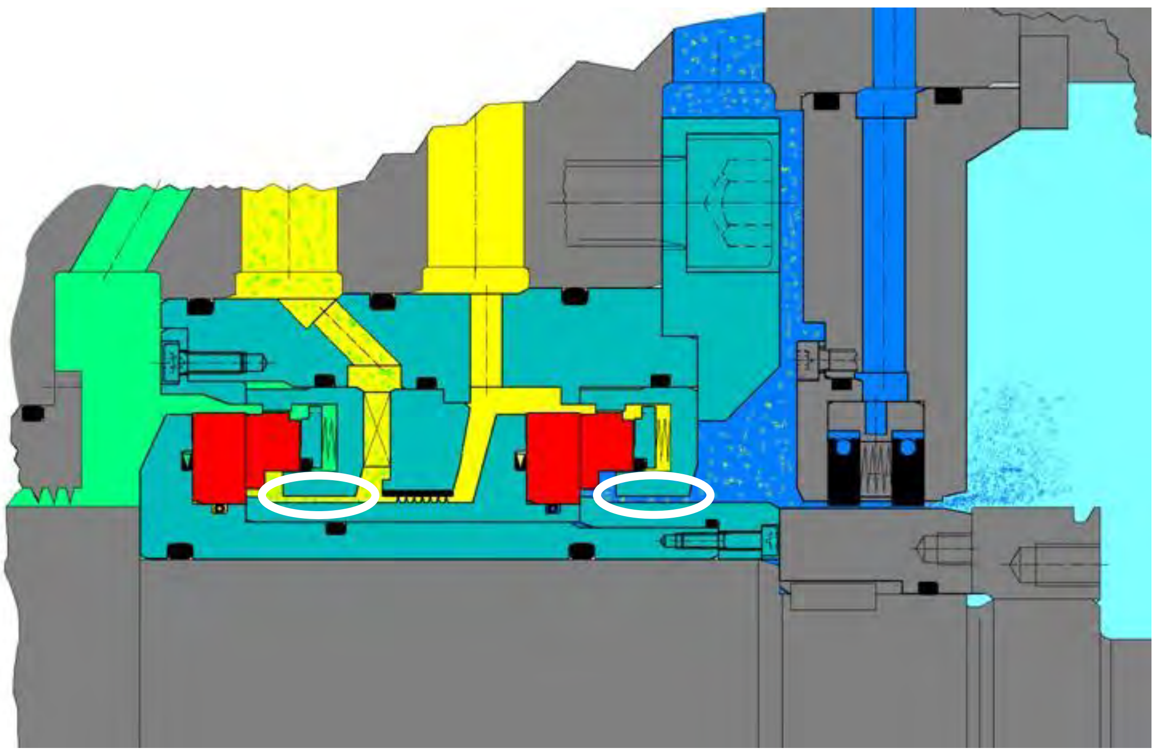

Figure 8. Restrictions in the DGS cartridge simulated in the DGS failure model. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

All the various openings in the DGS cartridge have been analyzed building a model that contains all the restrictions present. From this complete model it has been found that the main sources of pressure drops are only the gaps between retainer and shaft sleeve just after the primary and secondary seat rings (as highlighted in Figure 8) and the holes in the DGS cartridge itself (as highlighted in Figure 9). All the other openings produce negligible pressure drops, or such to not affect the overall flow distribution in the system, hence they can be disregarded in the DGS failure analysis.

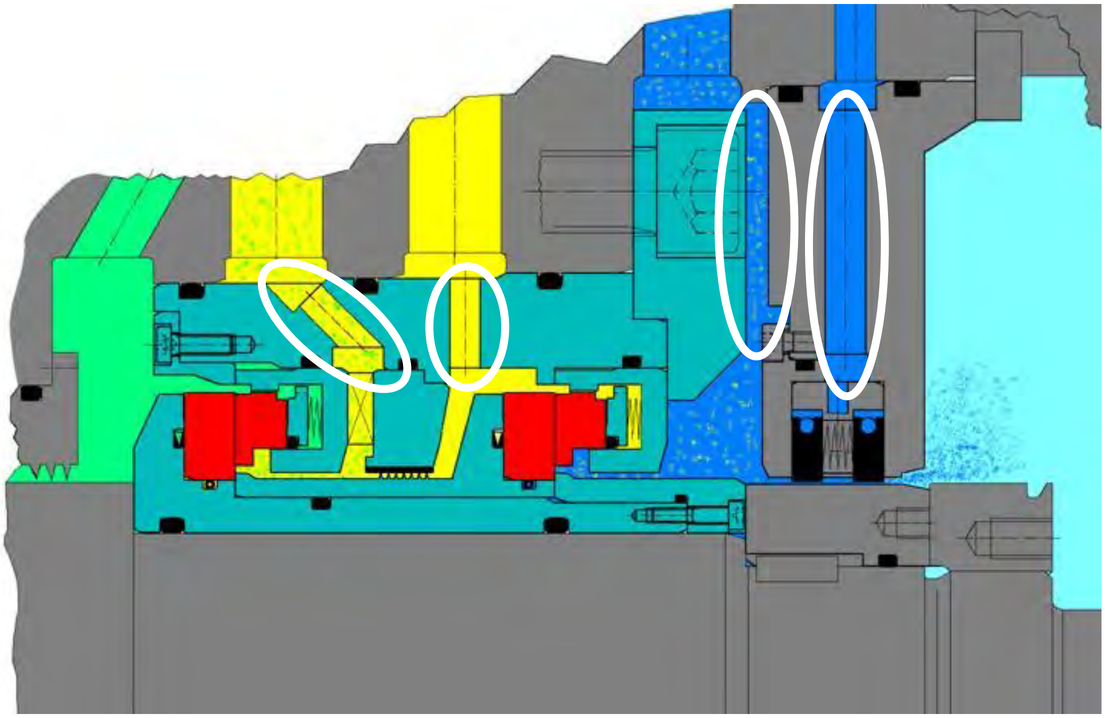

Figure 9. Restrictions in the DGS cartridge simulated in the DGS failure model. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

5. ANALYSIS OUTCOMES AND ACCEPTANCE CRITERIA

As said, API 692 was only recently introduced in the industry. The dual dry gas seal failure analysis therefore can be performed not only during design phase of new units, but also used as verification for already installed machines.

In both cases, the main outcomes of the analysis are pressure profile within the dry gas seal itself and in the surrounding system, and the flow distribution of released gas among venting systems and towards the lube oil tank.

The sub-systems mostly impacted by the dual seal failure are the secondary vent, separation seals and lube oil system, which are pressurized by the process gas, while they normally work at low pressure and with limited contamination of process gas. On the other hand, this analysis does not influence the design of primary vent and intermediate injection lines (including all their components, such as pressure safety valves) since for them the most critical scenario is the primary ring failure which was taken into account even before the introduction of API 692.

For these reasons, the most critical parameters to be verified with the DGS failure analysis are the pressure in the secondary vent chamber and in the secondary vent line, and the flow rates established in the secondary vent line, separation gas line and towards the lube oil system.

In case of contacting and non-contacting carbons, the pressure build-up on secondary vent chamber may jeopardize their integrity. The maximum pressure value estimated by the simulation considering separation seal intact shall be within the vendor acceptance limit. If this condition is not fulfilled, even if mitigation measures are adopted on the system, the correct failure mode of the separation seal described in section 2 shall be considered in order to evaluate the consequences: increased process flow migrating in the lube oil system and into separation gas injection line. Some consequences are present if labyrinths are used as separation seal.

This event is critical especially because it may lead to high pressure in the lube oil drain system up to the main tank. These parts are the weakest point of the system as they generally work at pressures close or equal to atmospheric. Overpressure must be below the tank resistance in order to avoid loss of integrity that would lead to oil spillage.

In addition, separation gas injection line is normally equipped with check valves, therefore gas migration in the nitrogen system is not likely, but the high pressure possibly generated by failure shall be in any case verified against line rating.

Over-pressurization shall be checked also on secondary vent line which in normal operation operates with pressure close to atmospheric value: this event can therefore represent the design case based on which the secondary piping rating and size are selected.

Finally, the dual dry gas seal failure analysis allows also to calculate the amount of process gas that could flow through the secondary vent line after all the mitigation measures have been adopted to minimize it and to minimize over-pressurization reached in all sub-systems. The secondary vent is generally routed to atmosphere, in a secure area, and the quantity of process gas that reliefs from it can generate an explosive cloud. A dispersion study is necessary based on this flow, especially if the process gas is toxic. As outcome of this study, which shall be part of entire plant emission verification, the area surrounding the vent connection shall be classified as potentially explosive.

6. MITIGATION MEASURES

The most immediate measures to reduce the gas migration rate are the following:

- Increase primary vent relief capacity by adding relief devices (safety valves or rupture disks) and increasing size/number of connections;
- Increase secondary vent relief capacity by increasing size/number of connections.

It must be considered that all seals are assembled in a single cartridge, primary vent and secondary vents are realized with holes/slots in the cartridge itself, the passage area of these vents is limited, and it cannot be increased above a certain value. These restrictions often represent the bottleneck of relieving flow. For this reason the separation seal resistance required by API shall be verified mainly by the seal manufacturer.
Another bottleneck is the vent channel on machine casing, especially for small compressors where the size cannot be increased over a certain diameter.
Therefore, above a certain value, increasing the primary or secondary vent size cannot further improve the gas relieving capacity. Eventually the designer has to deal with a flow of process gas migrating to the lube oil system that cannot be further reduced, the lube oil console vent size shall be suitable to allow the gas relief without resulting in a tank overpressure. This is generally possible in normal lube oil consoles, in other cases, as alternative, the lube oil tank must be reinforced to increase the pressure resistance. The negative effect of this solution is the potential pressurization of oil drain pipes that can prevent the correct oil drainage from other machines.
Summarizing, the system can be designed or upgraded to withstand the event of dual seal failure without major damages, but in some cases the quantity of gas release to atmosphere can be consistent, the need to depressurize the compressor loop in case of such event appears evident.

7. CASE STUDY

A reinjection high pressure barrel compressor installed in the late 90's was selected as a case study. The objective of the analysis was to evaluate if a machine with seal gas system designed according to old design rules could be aligned to the new API 692 requisites and with which mitigation measures.
The compressor is equipped with tandem dry gas seals plus contacting carbon separation seals. The main dry gas seal cartridge and the separation seals are packed together with screws. Based on screws size, number and material, the maximum allowable pressure for mechanical integrity was calculated. The mechanical integrity of the carbons was however guaranteed by the supplier only up to a lower value, equal to 8% of the Settle Out Pressure (SOP), which became the acceptance criterion.
The analysis was performed considering the two DGS rings completely removed with the compressor in Settle Out Pressure condition and modelling the existing system as-is. The internal separation seal was considered functioning and with zero gap with the shaft: this is the most conservative assumption but also realistic since it is a contacting carbon. The pressure reached upstream of the separation seal resulted in 14% of the SOP, thus not acceptable. Results of pressures and flows measured in crucial points are shown in Figure 10. For sake of a better understanding, the presented values are normalized to the SOP and to the nominal inlet seal gas flow, respectively for pressure and mass flow. It can be easily seen in Figure 10 how the most significant gas flow was coming from inside the compressor, while the inlet seal gas was lower than the normal operating value. A high flow was also measured at the secondary vent.
Possible mitigation measures were analyzed and discussed with End-User. As a first thing, it was considered to substitute the pressure safety valve installed on primary vent with a rupture disk with a lower set point, since lower pressures were reached in the as-is calculation not causing PSV opening. The rupture disk was dimensioned based on the existing primary vent line. This first measure helped to lower the pressure upstream of the separation seal, but the target was still not reached.
Further modifications in the system were investigated, with the objective of increasing the process gas flowing out of the DGS cartridge exploiting the existing casing holes and vents. According to the calculations, sonic conditions were reached in the primary vent pipe section upstream of the rupture disk. The original line size was ¾". This line size was increased up to 1 ½", maintaining a similar layout to the existing configuration as first approximation. The results showed that sonic conditions were avoided and the mass flow towards the rupture disk line (and then flare) was doubled, as presented in Figure 11. The pressure upstream of the separation seals decreased within the acceptance limit.
These results were considered acceptable and demonstrated that with minor modifications to the existing system (PSV substitution with rupture disk, part of primary vent piping size increased) the compressor's safety could be improved, fulfilling the new API 692 requirement.

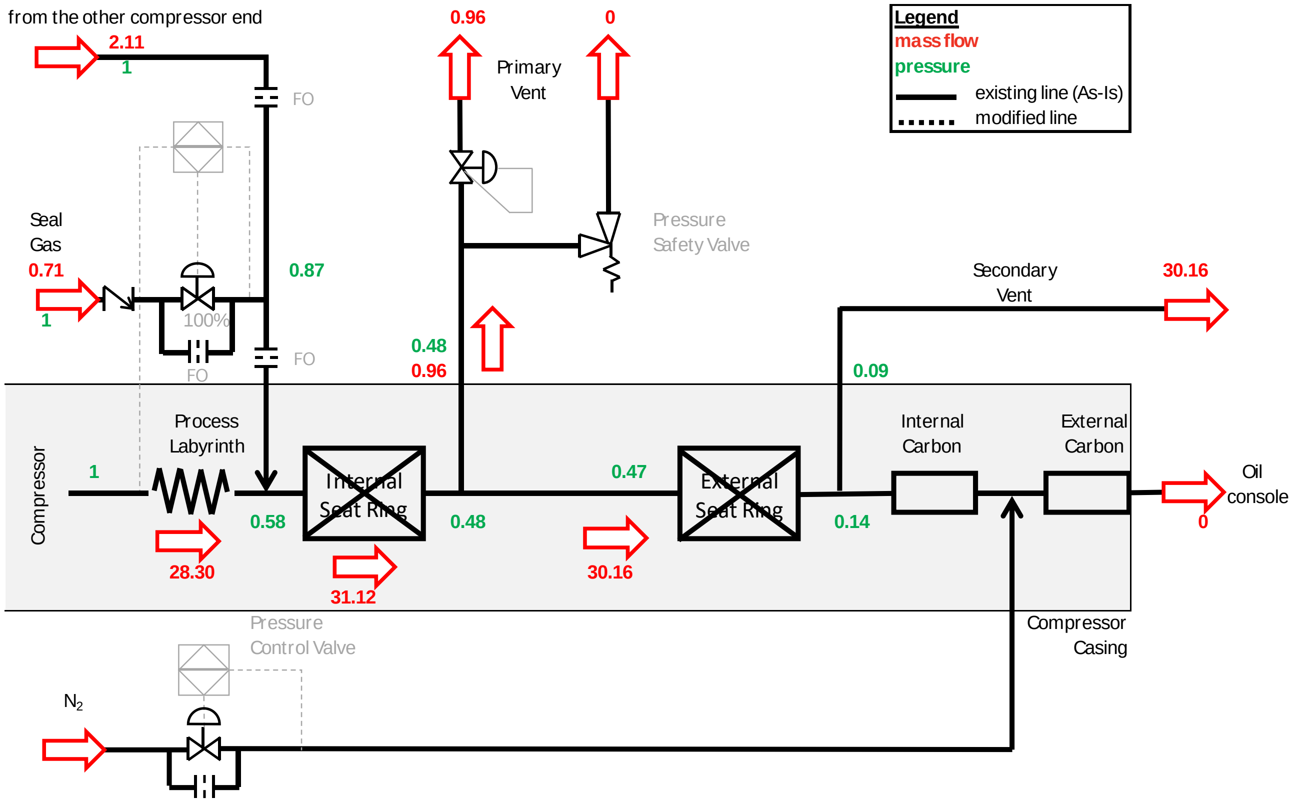

Figure 10. Dual DGS Failure Results: As-Is. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

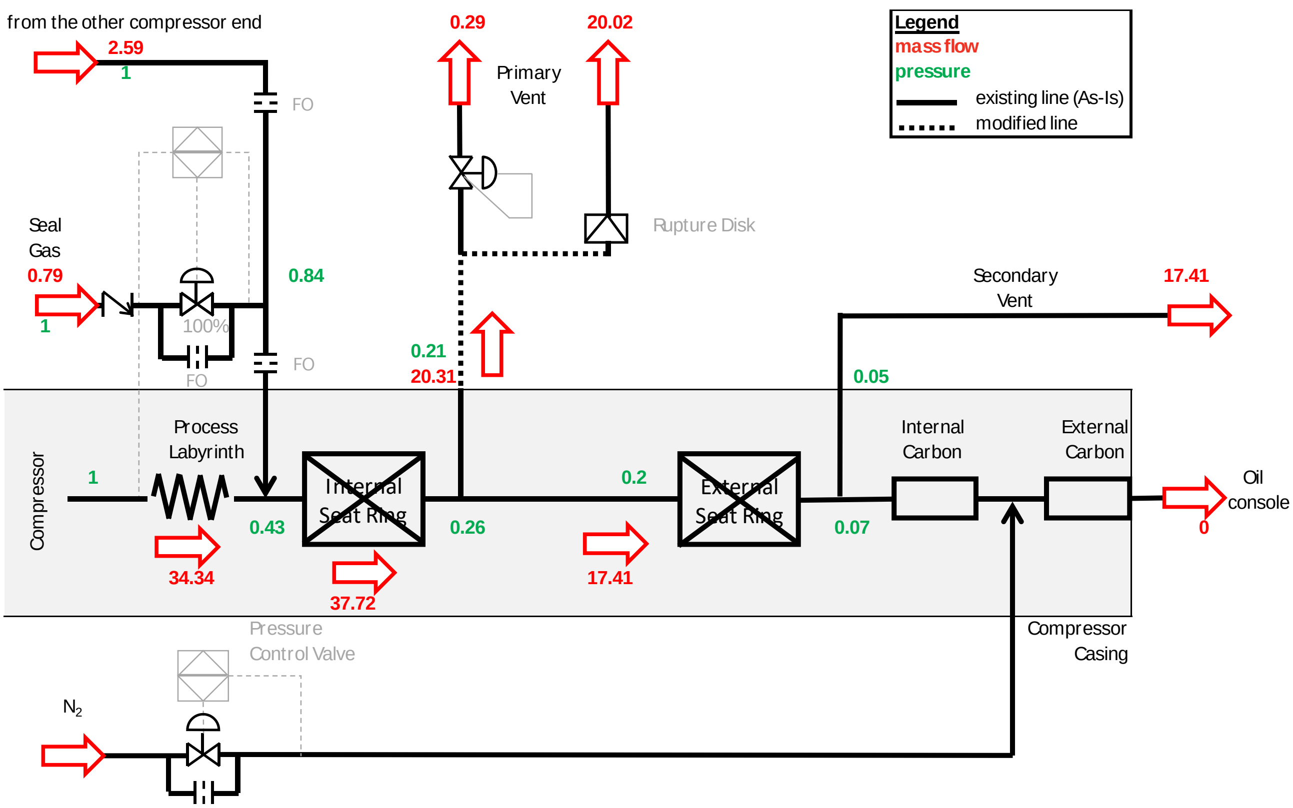

Figure 11. Dual DGS Failure Results: Mitigation measures. © 2019 Baker Hughes, a GE company, LLC - All rights reserved

8. FUTURE DEVELOPMENT

The plan for future development is to fine tune the analytical model in order to consider less conservative assumptions. A possible way is to develop a CFD model of the gas path during a dual ring failure scenario and compare the pressure drop at every restriction with the predicted one. A CFD model can evaluate the pressure drop compared with analytical model more precisely in particular for complicated gas paths as inside the DGS.
A further step can be to arrange a test bench using a real DGS and simulate the rings failure and measure the pressures at each vent to compare with the analytical model. This test can validate the model and improve the prediction capability.

9. CONCLUSIONS

The present paper provides an overview of the methodology developed to analyze the dry gas seal failure scenario. The main scope of the analysis is to estimate the pressure profiles and the gas distribution in the DGS cartridge and the surrounding system, identify the potential risks and find mitigation solutions to avoid uncontrolled gas release to atmosphere. This analysis aims at increasing the level of DGS system safety. In addition, the methodology can be applied to perform the analysis as required by API 692 regulation, which introduces a new step to further improve design: maintain the separation sealing integrity in the event of a dry gas seal failure, in order to limit the amount of process gas towards the bearing housing.
The peculiarity of the presented methodology is the level of refinement that was reached, including all the restriction elements that mainly affect the results of the analysis in the simulation model, giving the design engineer the possibility to identify possible risks in all parts of the system and to validate mitigations even in early design phases.
An example was given in the described case study that shows how the methodology can be applied to existing units and how systems design can be upgraded to new rules and requirements.
Currently the methodology is based on well-known formulas present in literature together with practices validated with experience and in some cases it may provide over-conservative results. As future development, the plan is to use data coming from CFD analysis and experimental tests to further refine the calculations and enhance the accuracy of the analysis in order to avoid over-conservativity and achieve the optimized design for the overall DGS system.

NOMENCLATURE

<u>Variables</u>

ΔP	= Pressure drop	($M \cdot L^{-1} \cdot t^{-2}$)
f	= Darcy friction factor	(-)
D	= Pipe diameter	(L)
L	= Pipe length	(L)
ε	= Pipe roughness	(L)
K	= Resistance coefficient	(-)
k	= Gas heat capacity ratio	(-)
ρ	= Gas mass density	($M \cdot L^{-3}$)
v	= Gas velocity	($L \cdot t^{-1}$)
M	= Mach number	(-)
G	= Mass velocity	($M \cdot L^{-2} \cdot t^{-1}$)
R	= Ideal gas universal constant	($M \cdot L^{2} \cdot t^{-2} \cdot mol^{-1} \cdot T^{-1}$)
T	= Gas temperature	(T)
P	= Gas pressure	($M \cdot L^{-1} \cdot t^{-2}$)
M_W	= Molecular weight	($M \cdot mol^{-1}$)
$\dot{m}$	= Mass flow	($M \cdot t^{-1}$)
Y	= Expansion Factor	(-)
C	= Discharge Coefficient	(-)
r	= Pressure Ratio	(-)
P_{vc}	= Pressure in the Vena Contracta	($M \cdot L^{-1} \cdot t^{-2}$)
β	= Diameter ratio	(-)
D_0	= Orifice diameter	(L)
r_c	= Critical Pressure Ratio	(-)
C_2	= Vena Contracta Enlargement correction factor	(-)
A_0	= Cross-sectional area of orifice	(L^2)
D_{eq}	= Equivalent diameter	(L)
p	= Wet perimeter	(L)
A	= Cross-sectional area	(L^2)

Acronyms

API = American Petroleum Institute
CFD = Computational Fluid Dynamics
DGS = Dry Gas Seal
OEM = Original Equipment Manufacturer
PSV = Pressure Safety Valve
SOP = Settled Out Pressure

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