42nd Turbomachinery
29th Pump SYMPOSIA

Proceedings of the Forty-Second Turbomachinery Symposium
October 1-3, 2012, Houston, Texas

CO_2 CORROSION ON NATURAL GAS COMPRESSORS: CRITERIA AND GUIDELINES

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ABSTRACT

Liquid water and CO_2 can easily lead to carbon steel corrosion, especially at high pressures. Corrosion can happen either during normal running or during pressurized standstill conditions. The present paper describes three real cases of gas lift compressors operating in offshore platforms that suffered internal damage due to the formation of carbonic acid and unfavorable process conditions. The damages found were slightly different on each machine due to specific design characteristics (such as the presence or not of inlet guiding vanes), different manufacturers and process conditions. Full description of the failure mode and relevant root cause analysis as well as the modifications implemented to solve the issue are provided. Additionally, a review of the monitoring tools used to diagnose internal corrosion will be presented, including typical responses that can be expected in compressors subjected to CO_2 corrosion. Finally, detailed criteria to define CO_2 corrosion potential are proposed as well as comprehensive guidelines on material selection to avoid similar problems.

INTRODUCTION

Natural gas centrifugal compressors are always subjected to suffer effects of gas inherent contaminants. Most common and dangerous contaminants, with regards to its mechanical integrity, are carbon dioxide (CO_2) and hydrogen sulphide (H_2S), especially for parts manufactured in carbon or low alloy steel. CO_2 becomes a dangerous contaminant when is associated with high pressure and liquid water.

Although compression systems are designed to avoid liquid at compressor suction by removing liquids (either hydrocarbon condensate or water) in scrubbers and conducting the satured gas in thermal insulated lines, this arrangement can never reach 100% of efficiency: liquid droplets will always be present at compressor suction.

Combination of CO_2, liquid water and high pressure leads to formation of carbonic acid (H_2CO_3), which can heavily corrode carbon steel of piping, valves and compressor components.

In the cases described in this paper, corrosion of carbon steel compressor components resulted in, depending on the particular case, geometry loss of aerodynamic parts, severe erosion process (caused by the continuous detaching of particles) at impeller blades and leakage paths and clogging of the gas path, especially in the first and second stages diffusers.

The consequences of these processes ranged from performance loss by plugging of diffuser channel, to compressor stalling even at high flow rate and crack nucleation at impeller outer diameter, with posterior impeller failure.

For each case, root cause analysis was carried out by operator and manufacturer in order to understand the failure mechanism and find possible modifications to avoid further repetition of the problem.

The root cause analysis for one of the cases (case A) will be deeply described, highlighting the differences from the other two cases (cases B and C). Additionally, monitoring tools used to identify and diagnose the internal damage, as well as criteria and guidelines on material selection are presented in order to avoid these unwanted effects.

SYSTEM DESCRIPTION

From the three compression systems where this corrosion issues occurred, two of them (cases B and C) are installed in oil and gas production vessels (FPSOs), while case A is at a semi-submersible (SS) oil and gas platform, all of them located in Campos basin, Brazil. Their main services are gas lift and exportation.

Each system contains three compressor trains, each divided in three compression sections, to raise natural gas pressure from 8 bar to 200 bar. Gas coolers and scrubbers are provided at each section inlet. On case A, one casing is dedicated to the first section, in a straight-through arrangement, while a second casing combines second and third sections in a back-to-back arrangement. Each casing is driven by a dedicated electric motor. On cases B and C, the first two sections are on a back-to-back casing and the third section on a straight-through casing, with both sharing the same electric motor. Figure 1 presents a schematic description of the systems.

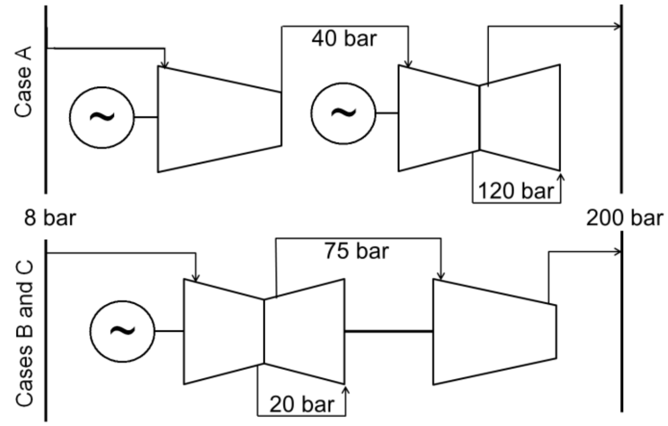

Figure 1. Natural gas compression system.

PROBLEM DETECTION

For case A, in the very beginning of the compressors operation, a deviation on the thermodynamic performance of the third section was detected by the maintenance team (Figure 2). The first hypothesis considered was wearing off labyrinths clearances due to a high number of starts.

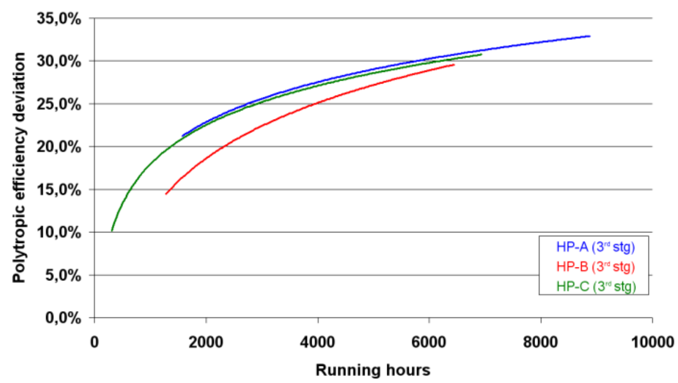

Figure 2. Historical trend of polytropic efficiency deviation for case A compressors.

Due to the premature lack of capacity, an overhaul was scheduled for each HP compressor at 7,000 running hours. During the first overhaul of compressor B, it was noted that clearances were OK. Heavy deposit was found at first impeller and diffuser of second phase and suction volute was found corroded, as can be seen on Figure 3. Compressor B was completely cleaned and return to operation with restored performance.

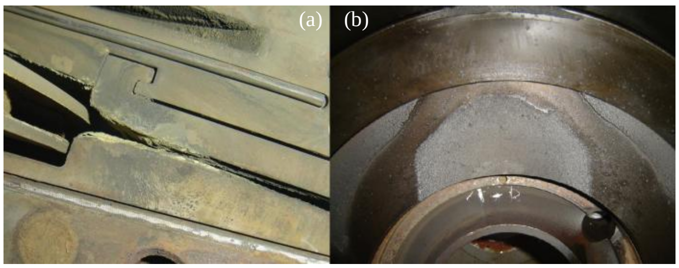

Figure 3. Plugged diffuser (a); corrosion at suction volute (b).

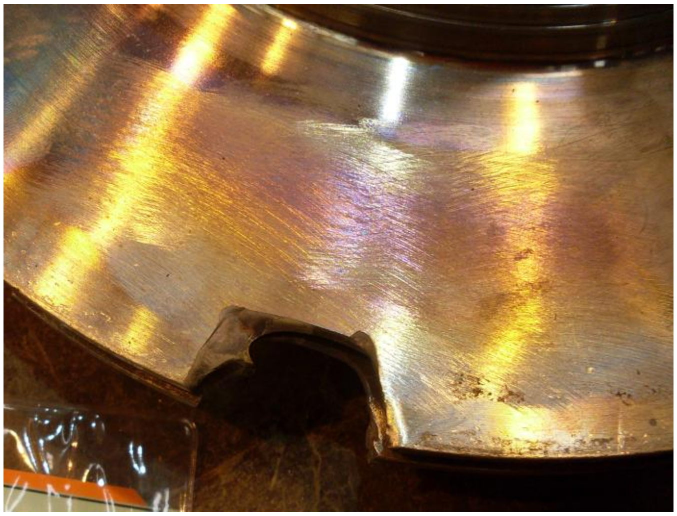

Figure 4. Failed impeller (cleaned).

During the second overhaul (compressor A), beside the same deposit and corrosion found on compressor B, the first impeller of second section was found damaged, with detached parts of the disc and cover at the outer diameter (Figure 4). Significant erosion on blades trailing edge was also detected, as presented on Figure 5. When later overhauled, compressor C was found in similar conditions.

As can be observed in Figure 5, the failed impeller (as all others in these rotors) was cut back. During shop acceptance tests, it was necessary to machine the impellers vanes outer diameter to reduce compressor head and required power, in order to match API requirements and avoid future operational problems, as motor overloading.

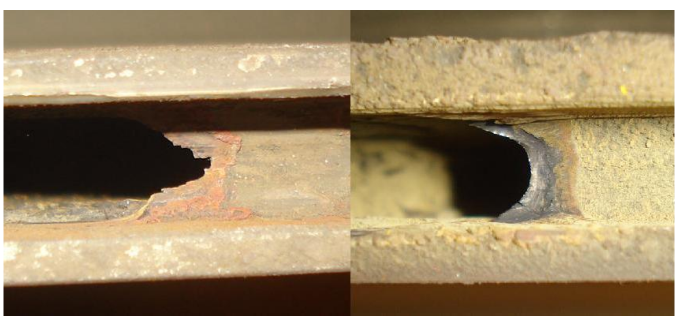

Figure 5. Erosion on the blades trailing edge of first impeller of third section.

It should be noted that only the impellers vanes were machined, the impeller disc and cover remained with its original diameters.

Although continuously monitored by both vibration protection and vibration monitoring system, the impeller failure was not detected prior to the compressor disassembly. After this fact, historical data was retrieved and analyzed, but no sign of sudden increase of 1X vibration, as one would expect as a result of the rotor losing localized mass, could be found. Even with significant unbalance, smooth continuous operation was possible due to compressor robustness.

In the other two cases, the issue was only identified after 2 to 4 years of operation. As there wasn't plugging in the gas passages, no performance loss greater than the expected due to compressor fouling was identified. The issue only appeared once the compressors started tripping due to sudden very high vibration at the high pressure, straight-through, casing drive end. This would be the side of the suction of the third compression section. Further investigation using online vibration monitor systems identified the vibration as being majorly subsynchronous, roughly at 0.2X. Figure 6 presents this vibration behavior as captured by the vibration monitoring software. It was also observed that this particular vibration signature was highly dependable on flow through this compression section: at high flow, no particular vibration frequency other than the unbalance (1X) response, but as flow was decreased a diffuse, low amplitude rash would appear at low frequency (0.1-0.3X), before suddenly concentrating in a single peak that would in most times trip de compressor train.

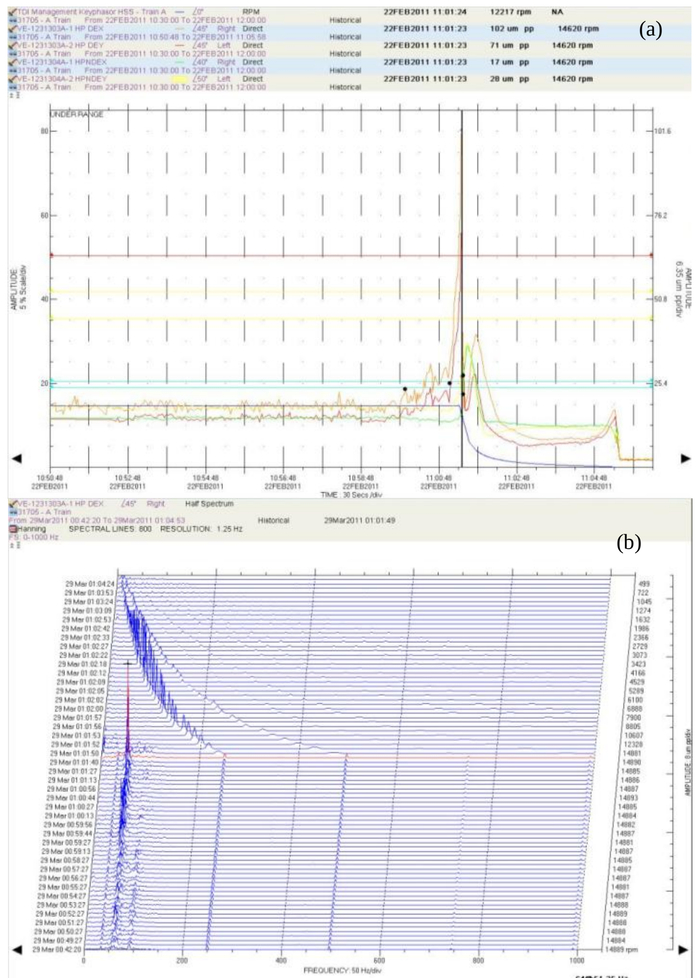

Figure 6. Vibration trend showing sudden peak (a) and waterfall showing the subsynchronous frequency (b).

At the time, the issue was diagnosed as aerodynamic stall events. The diagnosis was later confirmed once the compressor bundle was opened and inspected. Corrosion and erosion had severely damaged the inlet guide vanes and the low solidity diffuser vanes of the third section, causing a significant gas path geometry change that led this section to be prone to stalling, as shown in Figure 7.

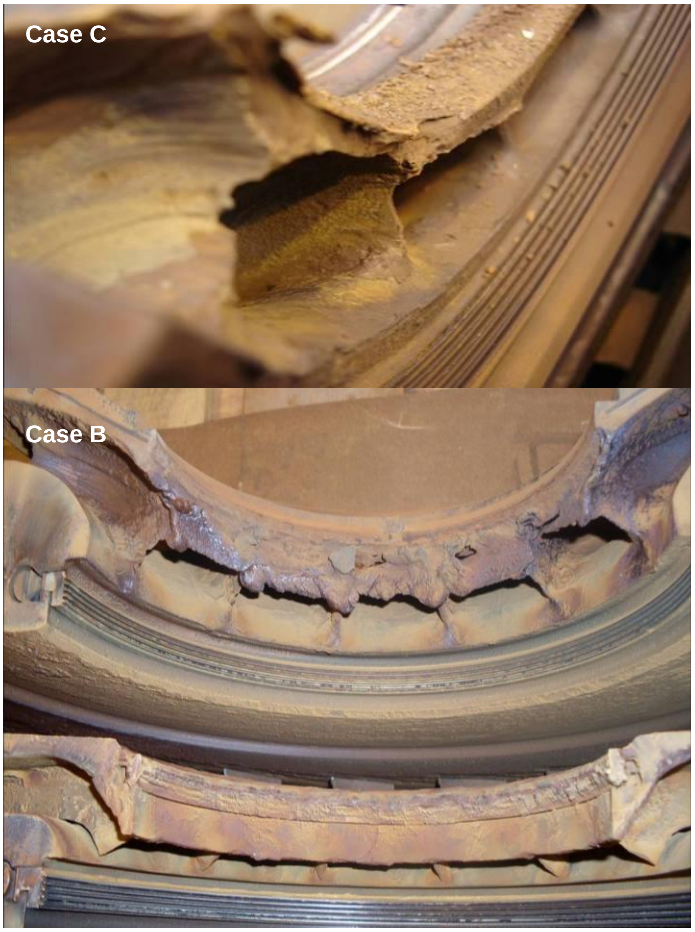

Figure 7. Damages found at inlet guiding vanes.

ROOT-CAUSE ANALYSIS

After case A compressor overhauls, it was clearly noted that two main problems should be investigated: heavy deposits at first diffuser and the broken impeller. The relation between problems should also be under analysis.

The starting point of the investigation was the analysis of the deposits found at the first impeller and diffuser. An X-ray diffraction analysis detected that the deposits were mainly Siderite ($FeCO_3$), which is the corrosion product of carbon steel in CO_2 environment.

This corrosion process can only occur with the presence of liquid water, but as the gas dehydration plant is downstream of compressors, a small fraction of liquid water is always found in the gas. The gas upstream of the compression plant is saturated in both water and hydrocarbons. After pressure increase in a compressor section, it is cooled down, resulting in the condensation of both hydrocarbons and water. This liquid fraction are, in theory, completely separated from the gas stream at the scrubber vessel before the next compression section suction, however some mist will inherently be carried over. The scrubber design was deeply investigated and found to be well within the acceptable limits. Additional liquid may condensate due to heat loss in the piping between the scrubber and the compressor (accentuated if no thermal insulation is installed).

The next step was checking the amount of CO_2 in the natural gas. Compressors were designed to process natural gas with CO_2 amount around 0.7%. Gas samples taken during operation presented CO_2 fraction up to 1.5%.

The results from deposits and gas analysis indicated that carbon steel was being corroded by CO_2 and its product was getting stuck in the first diffuser. The result was a reduced gas passage area (roughly to 30% of the original area), limiting the compressor performance.

This location for the deposits was also relevant. Usually thin solid particles pass through the compressor or accumulate in a reasonably homogeneous fashion through-out the gas path. The greater concentration at the first diaphragm indicated the presence of liquids at the section suction. The liquid served as a binder to the corrosion product and once evaporated by the temperature increase at the diaphragm, it left the corrosion product, once a thin dust, as a hard compact incrustation at the diaphragm walls. Impeller erosion rate was significantly influenced by the increased size and mass this solid/liquid agglomerate could have.

The source of particles could be any part made by carbon steel. Throughout the compression plant suction lines, recycle lines (downstream of recycle valve), valves, coolers and scrubbers are either made in or lined with stainless steel, while discharge and recycle lines (upstream of recycle valve) and compressor casing are made in carbon steel.

In principle, this discharge lines have the gas at a temperature well above its dew point during operation, some corrosion could have occurred during pressurized stops. The authors deem very unlikely that the short pressurized stops played a significant role as source of the corrosion product particles. Posterior piping inspection showed no significant corrosion. As these particles were heavily concentrated on third and non-existing in the first compression section, an upstream source was discarded by the user.

On cases B and C, some deposits were also found in the compressor internals (although not as many deposits as in case A) and sent to analysis. As expected, the results indicated the deposits to be Siderite. As in case A, since all the piping and equipments subjected to saturated gas is manufactured with stainless steel, the Siderite particles could only have come from compressor corrosion.

Regarding impeller failure, which only occurred in case A, the first analysis made by manufacturer was checking if failures could be happening due to excitation of impeller resonance frequencies. Possible sources of excitation were investigated, especially those from coupling of number of blades and vanes, and no crossing frequencies could be found.

Inspections detected that fractures that led to detaching of parts were started at blades trailing edge. As can be seen in Figure 8, these edges were found heavily eroded: only a very sharp edge was left at the blade. This led to a stress concentration even higher than that resulting from the blades machining after the shop acceptance tests. Finite element analysis showed that the impeller cutback increased stress at edges but still far from yielding strength. The high localized stress at the blade edge led to crack nucleation and, over time, impeller failure. Dye penetrant test was performed at failed impellers and clearly detected cracks initiated at blades trailing edges (Figure 8).

Figure 8. Dye penetrant test at impeller outer diameter.

Two possible causes for the blades trailing edges erosion were considered: liquid ingestion and corrosion product particles. Although the presence of particles was confirmed, it was not clear if the amount of particles could have resulted in such fast and severe erosion. On the other hand, there was no reason to suspect that a significant amount of liquid was being ingested by the compressor: very few condensate (hydrocarbon or water) is formed at the third section suction scrubber, most of it having been knocked-out in previous compression sections, and there were no records of high liquid levels at these vessels.

Most likely, a combined effected resulted in the observed erosion. The liquid present at the compressor suction might have aggregated the thin corrosion product particles into big heavy drops that could more easily erode the impeller.

On the other cases, although no erosion was identified at the compressor impellers, it occurred at the low solidity diffuser vanes, to extend of causing some to detach from their mount on the diaphragms.

MODIFICATIONS IMPLEMENTED

In case A, a performance loss could be detected with less than 1,000 hours of operation, while 7,000 hours was enough to cause impeller failure. These short periods indicate that the whole failure mechanism was able to quickly affect the compressor. In the other two cases, although the effects were only perceived after a few years of operation, there are no stand-by units in these plants. Therefore, the actions to be taken should be robust: modifications to both the compressor and the process it is inserted into were carried out.

Considering the expected life of the compressors, which should be, at least, 25 years of operation, and the fact that it was one of the possible sources of the corrosion product particles (see Figure 2), an action was needed to stop corrosion at suction volute of second section of high pressure compressor. A corrosion inhibitor system would be easier to implement, however, the operational cost would be much higher than protecting carbon steel surfaces of the compressors. Many alternatives were taken into consideration and, in the end, electroless nickel plating was chosen as a coating. Advantages of this process are its hardness (effective to avoid erosion and diffuser plugging), small thickness (almost negligible to final dimensions of parts), strong adhesion and good capability to reproduce the base surface. Moreover, it could be applied on parts locally, without the need to ship the parts back to the compressor manufacturer, and at a far shorter time range than the manufacture of new parts in corrosion resistant materials. Casing, counter-casing, 1st and 2nd diaphragm sectors were coated. As all impellers where manufactured in 17-4 PH stainless steel and treated to achieve high hardness, there was no significant benefit to coat them as well.

As described above, the most probable cause for the adhesion of the particles at the first diffuser was the presence of condensate, which acted as a glue to gather the particles. Gas from scrubbers to suction flange is theoretically satured, but few mist carry over is expected. Additionally, even if lines are completely insulated, a very small quantity of liquid is inherent to this kind of application. To completely avoid hydrocarbon and water condensate at compressors inlet, gas shall be superheated. The dew point of the gas coming from the 2nd compression section at this section discharge pressure was calculated to be around 48°C. As per design, compressor should run with 35°C at all the three sections suctions. For the third section suction, this meant that there would be liquid formation at suction scrubber and that the scrubber gas outlet would be saturated, if not with some liquid carry-over.

Raising suction temperature above gas dew point, the discharge temperature limit would be easily reached. Manufacturer reviewed temperature limit of compressor and discharge lines, reducing project margins. As a result, discharge temperature limit could be raised by around 10°C, which allowed raising suction temperature to 50°C

To guarantee that the gas at compressor inlet is superheated, a margin to dew point should be added to account for expected small fluctuations in gas composition and the different heat loss through the piping thermal insulation depending on the weather conditions. This margin was added by reducing the suction temperature of previous sections. In this way, more water and heavier hydrocarbons were knocked-out of the gas at the first and second section scrubbers, resulting in a lighter gas at third section, reducing its dew point and adding a safe margin.

At the moment that this paper is written, all three operating units described in case A have been fully modified and are running. The longest operating modified unit has been running for over 20 months, without any evidence of rapid performance loss observed previously.

Because of these good, field-proven, results, the electroless nickel coating, jointly with stainless steel sleeves for exposed shaft parts, was adopted as solution for other similar cases and are under implementation. Although the superheating of the compressor suction gas could not be implemented in this other units due to operational limitations in the cooling water system, the authors believe that the electroless nickel coating will be enough to avoid large scale corrosion.

MONITORING STRATEGIES

In all the presented cases, early detection of acid corrosion issue in the compressor internals was only possible due to the in place monitoring routine for critical turbomachinery. This monitoring is centered in two aspects of turbomachinery engineering: vibration and performance/thermodynamic analyses.

In case A, the problem was first detected by performance monitoring of each compressor section. In short term routine, the existing field process instrumentation is used to evaluate weekly deviations in polytropic efficiency from the compressor shop test curves. Deviations in polytropic head may also be evaluated, but usually the degradation trend is less visible in the parameter. Such calculations are done as described by Schultz.

For the calculation of each trend point a sufficient stable operational period shall be identified. The ASME PTC-10 has a good basic criterion to establish when the evaluated compressor section is in steady state. However, due to process oscillations, usually it is not possible to keep the flow variation inside the ASME PTC-10 variation limits. When needed, a broader limit may be used without significant loss in trend quality. Gas samples for chromatography should be regularly taken (in three to six months intervals) to update gas composition in this calculation.

In a long term routine, a more detailed performance analysis may be done, evaluating the actual performance curves and the maximum compression capacity for current process conditions in each machine, supporting the decision in whether to overhaul or not the machine.

In the other two cases, no accelerated performance loss was observed. The corrosion issue manifested itself as frequent sudden high vibration trips. Periodical vibration data collection most likely would not be able to detect the stall phenomenon caused by the internal corrosion and erosion, unless specifically looking for it, due to its high dependence on operation point. On the other hand, the continuous monitoring system adopted by the user for critical turbomachinery allowed for a fast diagnosis, leading orientations to field operators that permitted an extended compressor operation while the resources for intervention were being gathered.

For an early detection of this issue through vibration, users should regularly monitor vibration for any unusual non-synchronous vibration. Small amplitude rash in a broad low frequency range of the vibration spectrum, visible especially at low flow operational points, could be a hint of pre-stall flow conditions. In such cases, compressor outage for overhaul could be postponed to allow for proper intervention planning, by operating at higher flows (for example, by partially opening the recycle valve), therefore avoiding the surge conditions. Such postponement may lead to more internal components being damaged and even consequences external to the compressor: in one case the anti-surge valves begun to stuck due to the presence of corrosion product particles in the gas.

RECOMMENDATIONS FOR FUTURE PROJECTS

With the lessons learned from the failures described above it was clear to the operator turbomachinery engineering that modifications should be done to its centrifugal compressor for upstream service technical specifications, especially when the majority of the user's future projects have indication of medium to high CO_2 content levels.

No change was needed in piping, heat exchangers and pressure vessels specifications, since they already required stainless steel materials to be used for low temperature (close to dew point) process gas.

Two different criteria were studied to avoid CO_2 corrosion on compressor internals: a CO_2 content criterion and a liquid water criterion. The first aiming to verify if enough CO_2 is expected on the compressed gas for the corrosion process to occur, while the second defined whether or not there might be liquid water in the gas (a requirement for the CO_2 corrosion process to occur).

For the CO_2 content criterion a search for references in other type of equipment in the oil & gas industry was made. An API standard for wellhead christmas trees (API SPEC 6A) with a criterion for CO_2 corrosion was found. According to it, CO_2 corrosion can be expected if the CO_2 partial pressure in the gas is above 7 psi-abs (48 kPa-abs), independently of gas flow speed. An internal standard for pipelines presented a criterion (reproduced at Table 1) to estimate CO_2 corrosion severity depending on CO_2 partial pressure and gas flow speed.

Table 1. CO_2 corrosion severities according to operator's internal standard for pipelines.

Parameters		**CO_2 partial pressure ($T<60°C$)**	
Corrosion potential	**Low**	$p_{CO2} < 4$ psi-abs $V < 5$ m/s	
	Moderate	4 psi-abs $< p_{CO2} <$ 15 psi-abs $V < 5$ m/s	$p_{CO2} < 4$ psi-abs 5 m/s $< V <$ 10 m/s
	Severe	4 psi-abs $< p_{CO2} <$ 15 psi-abs $V > 5$ m/s	$p_{CO2} > 15$ psi-abs Any velocity

The gas flow speed is an important parameter since the CO_2 corrosion process forms a protective carbonate layer that then greatly slows down the corrosion process. High gas velocities clear off this layer, allowing for a higher corrosion rate.

Taking a conservative approach it was chosen to adopt the 4 psi-abs (27 kPa-abs) CO_2 partial pressure criterion to define the boundary condition above which carbon steel and low alloy steels are prone to CO_2 corrosion.

Only high CO_2 content is not enough to initiate the corrosion process. For it to occur, liquid water needs to be present. Therefore a criterion defining when the gas should be considered wet was essential to complete the CO_2 partial pressure one. Stating that gas at any temperature above its water dew point is dry is not enough to avoid condensation, since variations in gas composition can easily change the water dew point of the gas. Therefore a safety margin was adopted: the gas should be considered wet if its temperature is lower than 10°C above the water dew point. This is the same margin defined by the NORSOK M-001 standard.

The same criteria of 10°C margin vs. water dew point should be recommended also in view of heavy hydrocarbons in order to eliminate any potential cause of liquid erosion and particles plugging.

Once the criteria for when CO_2 corrosion is expected are clearly defined, it was evident to the operator that a requirement needed to be added to its technical guidelines for centrifugal compressors in upstream services. In order to accommodate the uncertainties in gas composition available in early reservoir studies (for example, in case C negligible CO_2 content was expected in design phase, which proved to be untrue once the wells started operating), it was decided that all components in contact with wet gas (temperature below the 10°C above water dew point) should not be manufactured in uncoated carbon steel or low alloy steels. Additionally, this requirement will avoid issues if wells from new oil fields are connected to existing platforms. For piping and vessels, current requirements were considered adequate by operator.

Although the details on the pressure and temperature increase along the compressor gas flow path is only known to the compressor manufacturer, operators may perform a preliminary evaluation on whether or not the above requirements are met having only the information available at the standard API 617 compressor data sheet. The aim of this evaluation is to identify which components are expected to be exposed to gas conditions that lead to CO_2 corrosion.

To do so, it is necessary to evaluate the pressure and temperature increase after each impeller and diffuser. A simple way to do so is to consider that the total polytropic head of a section is the sum of the polytropic head of each internal stage, weighted by their impellers external diameters squared (Equation 1) and that their polytropic efficiencies and specific work coefficients are the same as the compressor's whole section. To divide pressure and temperature increases between the impeller and the diffuser, a good starting value for the impeller reaction factor is 30%, which means that 30% of stage polytrophic head of a single stage is obtained at the impeller and 70% at the diffuser.

$$Hp_n = \frac{Hp_{total} \times D_n^2}{\sum D_n^2} \qquad (1)$$

Where,

Hp_n = polytropic head of the n-th impeller
Hp_{total} = total polytropic head of the section
D_n = external diameter of the n-th impeller

Temperature rise at the discharge of each impeller can be estimated through Equation 2.

$$\Delta T_n = \frac{g \cdot Hp_n}{C_p} \qquad (2)$$

Where g is the reaction factor.

With these results, a map with the estimated gas conditions at each compressor component can be made, allowing the identification of the components that may be prone to CO_2 corrosion (Figure 9).

Clearly, the above is a simplified evaluation of the gas conditions inside the compressor, but it allows the identification of critical components, whose material selection should be further discussed with the compressor manufacturer.

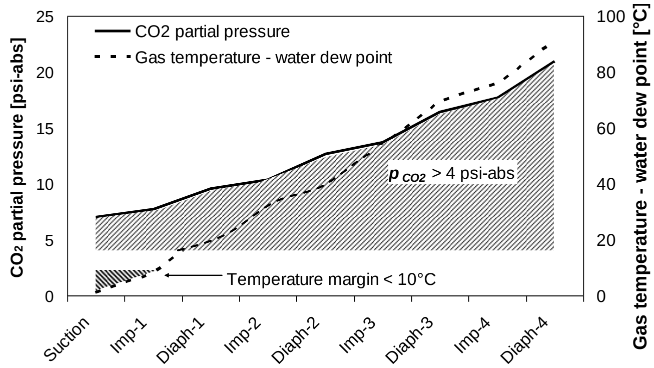

Figure 9. Example of a compressor CO_2 partial pressure and temperature to water dew point margin map. In this case, the gas path up to the first diaphragm is expected to be in conditions under which the CO_2 corrosion can occur.

For critical turbomachinery, in order to detect the issues described in this paper and others, the authors recommend routine performance evaluation and machinery vibration condition analysis. Infrastructure required for this condition monitoring is continuous acquisition of process parameters as well as a dedicated vibration monitoring system.

CONCLUSIONS

Medium to high CO_2 gas content is each time more frequently found on new projects. Up to recently, no special requirements were made by the operator regarding compressor components materials selection other than to follow NACE MR0175 standard when H_2S was predicted.

This paper presented example cases where CO_2 corrosion led to significant compressor performance loss, frequent tripping due to high vibration and internal components failure, resulting in prolonged equipment unavailability to carry out the necessary repairs and modifications.

In order to avoid these problems, this paper presented modifications made both to existing operating units and to operator's guidelines to new projects. In both cases, the modifications aimed at avoiding the contact of unprotected non-corrosion resistant steels with wet gas (since the CO_2 content is inherent to the production wells).

The data available so far indicates that the modifications done to the operating compressors cited in this paper successfully stopped the corrosion-erosion process.

These issues were early detected due to the availability of monitoring systems and specialized teams routinely evaluating critical turbomachinery, resulting in fast diagnoses, prolonged operation and reduced outages.

Regarding the updated guidelines for future projects, the authors are confident that the presented criteria will avoid future repetitions of such events.

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ACKNOWLEDGEMENTS

The authors would like to acknowledge all the work done by members of Petrobras’ turbomachinery maintenance, engineering, workshop and research center, as well as the different compressor manufacturer’s teams, which helped to diagnosis and solve the problem described on this paper.