Advanced Online Condition Monitoring and Diagnostics support Operational and Maintenance Decisions in an Offshore Gas Compression and Export System Unit

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1 Summary
Chevron North Sea Limited operates two 3-stage reciprocating compressors on the offshore platform which are in service of the Gas Compression and Export System unit.

Essential information to evaluate the condition of the reciprocating compressor are a base for operation decisions, therefore Chevron North Sea Limited installed an online condition monitoring system for reciprocating compressors to increase the reliability and safety protection for their compressors.

This paper presents several case studies about malfunctions like a broken rod, loose crosshead shoes and misalignment, as well as lessons learned which helped to reduce outage time and costs.

2 Introduction
Chevron – Captain Field and Facilities
Chevron is one of the world’s leading integrated energy companies being engaged in every aspect of the oil, natural gas, and geothermal energy industries, including hydrocarbon exploration and production; refining, marketing and transport; chemicals manufacturing and sales; and power generation. The upstream business combines innovation and the effective use of technology to maximize mature fields, discover new resources and meet the world’s growing demand for energy.

Chevron has key operations in the world’s most important oil and gas regions, including the North Sea within which is the Captain Field.

The Captain Field lies approximately 90 miles (145 km) north-east of Aberdeen, Scotland, in the Outer Moray Firth, in water depths of roughly 346 feet (105.5 m). Discovered in 1977 in Block 13/22a, the Captain Field achieved first production in March 1997, thanks to key technology developments in horizontal drilling and down-hole pumps in well bores.

The field currently includes a wellhead protector platform (WPP) and bridge linked platform (BLP) connected to a floating production, storage and offloading (FPSO) vessel and two subsea manifolds tied back and connected to the platforms by a suite of pipelines.

Captain crude oil is offloaded from the floating production, storage and offloading vessel to a dynamically positioned shuttle tanker and taken either directly to customers or to the Nigg Bay terminal. Captain gas is exported and imported via subsea pipeline to the Frigg U.K. Gas Transportation System and then on to St Fergus gas terminal.

In 2016, daily production averaged 24,000 barrels of liquids and 3 million cubic feet of natural gas.

3 Gas System Overview
The Captain Field gas liberated through partial stabilization of produced fluids is utilized to meet the fuel gas requirements of the platform power generation turbines. The remaining gas is compressed and dehydrated for export to the Frigg pipeline system via a 70 km 8” export flow line.

Gas in excess of fuel requirements is routed to the Gas Compression Package A-4760. The maximum A-4760 throughput is 18 mmscf (million standard cubic feet per day).

Following removal of entrained liquids, the gas stream is split and routed to two identical Reciprocating Gas Compression Trains, A-4760A and A-4760B. Each train consists of three stages of compression with inter stage cooling and liquid knockout. Liquids removed from the gas stream in each train are routed under level control to a Gas Scrubber Vessel.

High pressure gas from Compressor After coolers is connected and routed to a Common Discharge KO Drum. From there the gas enters a Glycol Dehydration System which dehydrates the gas by contact with Tri-Ethylene Glycol (TEG) to the required export water dew point specification. The gas is then metered and delivered to the Frigg pipeline for export.

The Gas Compression and Export System is supported by lubricating oil and cooling water systems, and also interfaces with other utilities systems, Inert Gas (Nitrogen), Drains, Compressed Air and Flare Systems.

The Gas Export Compressor K-4760A is an electrically-driven, 3-stage, 4-cylinder reciprocating compressor.

Gas enters Compression Train A-4760A and the first compression stage K-4761A via a 1st Stage Suction Volume Bottle. There is a volume bottle at the suction of each stage. The purpose of these bottles is to minimise gas pulsation and starvation in the compressor suction resulting from the intermittent gas flow in and out of the re-ciprocating compressor cylinder.
The first stage of compression in cylinders 2 and 4 raises the pressure of the gas to 21.6 barg. Gas then enters the 2nd compression stage via a 2nd Stage Suction Volume Bottle. The 2nd stage of compression raises the pressure of the gas from approximately 21 barg to 67.6 barg maximum. Compression is carried out in cylinder 3. The gas is then directed to the 3rd compression stage from the last Suction Volume Bottle. The 3rd stage of compression raises the pressure of the gas to approximately 187.3 barg. Compression is carried out in cylinder 1. Gas from V-4764 is then routed to the Glycol Dehydration System and Glycol Contactor for dehydration to Frigg export pipeline water dew point specification.

The compressor has a sister unit, K-4760B, that is the same type and specification of machine. It is currently out of service for structural redesign, giving K-4760A a heightened criticality to production with high availability required for efficient operations. This provided the justification for deployment of an online condition monitoring system.

4 Reciprocating Compressor Health Monitoring

The two reciprocating compressors are monitored by an online condition monitoring system. With the goal to establish a condition-based maintenance strategy, a detailed knowledge of the machines’ behavior is essential. Vibration measurements at each crosshead slide and at frame, in combination with dynamic indicated pressures and rod position are constantly collected and analyzed with hardware and software thresholds. The online condition monitoring system synchronizes the vibration signals with crankshaft rotation to associate vibration peaks to the piston position along the stroke. This provides machinery protection and detailed information of machine condition.

5 Crosshead Acceleration

The mechanical safety of each throw is monitored by accelerometers, which are mounted on the top of each crosshead slide. Placing accelerometers over each crosshead slide provides the best method to detect machinery problems due to impact-type events. Acceleration spikes in the waveform refer to the response of impulse events, occurring during normal operation.

As parts wear over time, the mechanical clearances increase. The extra clearance gives the parts more distance to gain velocity and the impacts between the components become more energetic. As examples, such impacts can occur between the crosshead and its guide, as the babbitt of the crosshead shoes slowly erode. Loose rod nuts or bolts, worn pins as well as liquid ingestion can be detected in early stages of development.

The mechanical impacts produce a high frequency structural vibration, which is characterized by a high amplitude peak, followed by a “ring down.” Crosshead acceleration signals are normally in the range of about 0.7 to 2.0 g.

6 Frame Velocity

Two frame velocity transducers on the side of the frame monitor the response of the system to the main forces and moments that are occurring on the balanced-opposed compressors. If used for machinery protection, these velocity measurements provide additional safety monitoring capabilities.

Frame velocity measurements monitor malfunctions such as imbalance due to unusual pressure differential or inertial unbalance, looseness in the foundation attachment (such as deteriorating grout or shims) and high moments caused by excessive rod load. However, installing frame vibration transducers only may be insufficient—as the increase in frame vibration due to incipient failures developing at the running gear or cylinder assembly will be small and typically covered by the larger vibration contribution due to machine movement. By the time the malfunction has been detected by the frame velocity transducer and the compressor shut down, major secondary damage may have already occurred. [1]
Cylinder Pressure and Forces Monitoring

The single most effective way of determining the overall health of a reciprocating gas compressor is by examining the cylinder pressure profile. Online access to the internal pressure for each compressor cylinder enables continuous monitoring of cylinder pressures, compression ratios, peak rod loads, and rod reversal. This provides valuable information on the condition of suction valves, discharge valves, piston rings, packing glands, and crosshead pin. In addition, the functionality of capacity control devices, such as unloaders, clearance pockets and stepless control systems can be analyzed.

From the diagnostic standpoint, the dynamic cylinder pressure measurement provides great value. The ability to correlate events in the crosshead acceleration waveform with events in the pressure, rod load curve and crosshead vertical force plots is essential. This enables to make precise maintenance decisions, to exchange the failed parts, only.

As the crankshaft turns, the combined rod load (inertia and gas load) at the crosshead pin varies. As a result, the forces acting at the crosshead can be calculated directly by the monitoring system, using the cylinder pressures and geometrics of the reciprocating components.

The crosshead transmits the axial acting pressure loads to the piston through the piston rod and absorbs the dynamic forces produced in a direction perpendicular to the horizontal axis of the motion.

The figure below shows a force reaction diagram with summation of all of the forces that are acting on the crosshead pin. The angle \( \Phi \) can be found from the known dimensions of the crank-slider mechanism.

Vertical Crosshead Force

Throughout the piston's stroke, the horizontal component of the connecting rod force (FCR) balances out the Rod Load Force (FRL) generated at the piston rod. With respect to crankshaft rotation, the vertical force (FCV) applied to the crosshead is affected. With the exception of the dead center positions, the acting forces at the connecting rod include a horizontal force (to counter the gas and inertia forces acting on the wrist pin) and a vertical force, which is in downward or upward direction. When the crank angle is 0 or 180 degrees, the connecting rod is directly in line with the cylinder bore, and it is incapable of imparting vertical forces to the crosshead. At these crank angles, the vertical force on the crosshead will be zero regardless of the combined rod load. At all other crank angles, the loaded connecting rod passes a vertical force component to the crosshead pin.

If this vertical force is in upward direction and exceeds the weight of the crosshead, the crosshead will physically lift off the lower guide and be pressed against the upper guide. The downward restraining force that is being applied to the crosshead pin is caused by the crosshead weight and the reaction between the upper crosshead shoe and the upper surface of the crosshead guide. This is the secondary motion of the crosshead motion which takes place across the clearance between crosshead and crosshead guide.

Thus, the crosshead moves periodically against the inner surface of the crosshead's lower and upper guide which introduces...
consequently impact-induced vibration, which is one of the sources of vibration in reciprocating compressors.

At balanced opposed vertical compressors, in the case of the left crosshead (which is being pulled downward by its connecting rod), the weight of the crosshead adds to the downward vertical load component. For the right crosshead (which is being pulled upward by its connecting rod), the weight of the crosshead is subtracted from the vertical load component. However, for a small time period in either case, the force does act opposite of the expected direction.

9 Rod Position Monitoring

Horizontal and vertical proximity probes, mounted in an orthogonal arrangement, observe the piston rod movement. Continuous monitoring over time measures the rider band wear. But in addition, the peak to peak displacement along the piston stroke allows conclusions to the condition of the rod.

In figure 7, the three rod position plots (orange, blue, red) illustrate three different stages of a machine’s condition. The yellow & red dashed arcs represent estimated alarm thresholds (alert and danger, respectively) from the bore clearance extrapolated back to the measurement plane. If the piston rod shifts more than 60 mils/1520 µm from the geometric center in any direction, then an alert condition would be raised to prevent actual piston to cylinder liner contact.
The orange trace shows the piston rod motion in reference/baseline conditions with new rider bands. The low amplitude and proximity of the piston rod centerline to the geometric centerline indicates that the throw is running smoothly with low peak-peak displacement and Position Magnitude values.

The blue trace shows the rod position at a later time as the condition progresses. The peak-peak displacement has increased slightly; however, the Position Magnitude is much larger and more significant.

And finally, when the machine is in an alarm condition (red sample), the peak-peak is much larger along with the rod position clearly indicating an alert condition. In this state there is greater potential for contact with the cylinder wall; however, it should be noted, that the equivalent vertical displacement reading is still indicating much less than the alarm value. This example shows the benefit of the horizontal transducer for a fully two dimensional measurement. [2]

10 Cases – Condition Monitoring helps to define precise maintenance decisions

10.1 Case: Broken Rod at Throw 2, 1st stage

On July 25th the Machinery Protection System tripped the reciprocating compressor due to high frame velocity vibration. The trend showed a sudden increase from 3 mm/s RMS to above 40 mm/s. The accelerometer, mounted above the crosshead slide at throw 2, 1st stage displayed the same symptoms of rapid vibration increase. Also observed in the preceding minutes to trip was an indicated fall in throw 2 rod position.

Following the machine stop, the throw 2 covers were removed for internal inspection by the mechanical team and the compressor was barrowed over. It was reported that nothing unusual was observed in the course of these activities and that next a restart of the compressor should commence.

However, based on the initial data review, it was considered that prior to attempting a machine start a more detailed analysis of the diagnostics should be performed to fully understand the shutdown characteristics.

This encompassed a variety of indepth dynamic assessments detailed, as thus:

The following crank angle based vibration plots illustrate the sequence of events of the 1st stage, Throw 2 at K-4760 A on July 25th.
Forces

Dynamic Pressures: normal
Crosshead acceleration


Two minutes before the failure, the crosshead acceleration waveform (upper right plot) shows no indication of a developing failure. The piston rod peak-to-peak movement of 500 µm (lower right plot) indicates a typical movement. The cylinder pressures (lower left plot) and the calculated gas-, inertial-, and combined forces (upper left plot) are normal.

46 Seconds before the rod breaks, no indications are visible at the crosshead vibration, but peak to peak movement at the piston rod increases up to 950 µm. It seems that the rod is already cracked, because the highest peak to peak movement occurs in the area of the rod load reversals.

Forces - Rod load reversal area
Crosshead acceleration

Figure 10: 25 July 2015 17:39:59 h – 46 seconds before the rod breaks.

As the failure develops, the piston rod peak-to-peak movement increases significantly. The crosshead acceleration waveforms show a slightly higher acceleration in the load reversal areas—which would not provide an alarm, yet.
Forces

Dynamic Pressures: normal

Figure 11: 25 July 2015 17:40:41 h – 4 seconds before the rod breaks. Crosshead vibration signature changes slightly, but peak to peak movement at the piston rod increases up to 1300 µm.

The figure below represents a revolution, where the rod is already broken. When the reciprocating components travel from the Bottom Dead Center (180° crank angle) in direction to the Top Dead Center (0°/360° crank angle), they hit the disconnected piston at point A and pushed it towards Top Dead Center. Due to the acceleration of the piston, the head end cylinder pressure increases (orange line in lower left plot). The collision of the disconnected rod and disconnected piston is also visible in the high amplitude of the crosshead acceleration at point A (plot upper right). The amplitude remains in the Top Dead Center, as the piston pumped the cylinder head.

During machine stop, the following revolutions displayed the sharp impulses, when the reciprocating components hit the disconnected piston.
Forces

Piston is disconnected, no pressures are built up

Crosshead acceleration

Piston Rod movement

Figure 12: 25 July 2015 17:40:45 h – Rod is disconnected.

The detailed analysis of the reciprocating diagnostics provided the foundation for a decision not to attempt a restart of the compressor and to instead mobilize materials and Original Equipment Manufacturer (OEM) field engineers offshore for confirmation and replacement of a sheared piston rod at cylinder 2.
Lessons learned from this failure: during the course of the event investigation and data analysis a number of parameters were identified with the potential for providing early warnings as to the presence of a machine operating environment that could instigate a propagation of mechanical defects. In addition, a further layer of shutdown protection limits—to prevent significant secondary damage during failure occurrences—were defined. This enhanced machinery protection consisted of peak-to-peak displacement of the rod movement with Alert level at 750 µm peak-to-peak and 900 µm peak-to-peak for Danger and crosshead acceleration boundaries which were included into the trip logic.

From the Root Cause Analysis investigation it was highlighted that five days prior to failure, the 1st stage suction pressure upstream (red trend line in Figure 14) of the compressor started to become more erratic and that this was initiated by a gas scrubber pressure spike. Subsequently the gas scrubber behavior became erratic and the export compressor suction pressure started to increase in range. The cause of this was narrowed down to the gas recycle valve controlling irregularly.
Reviewing the liquid carryover potential at the time, the levels in the vessels hadn’t been any higher than normal. Deducing from all the associated data and information—fluctuating pressures and therefore changing in the compression ratio could have led to component stress and ultimately to the rod failure.

At time of compressor pressure instabilities, cylinder 2 and 4 compression ratio becomes very erratic and goes above the OEM rated for 1st stage is 4.4.

**11 When machines stop – problems start**

Subsequent to the verification of the broken piston rod on line 2 and further assessment by the OEM field service engineers—to enable a rapid turnaround restoration to get the machine back to an operational ready state—a repair was progressed utilizing materials from the B machine. The plan was to undertake a more extensive refurbishment during the planned 12,000 hour service in November. A successful post repair start of the compressor was initiated on the 17th of August.
An assessment of the compressor in operation highlighted a higher than expected Throw 2 crosshead acceleration value, exactly the throw with the disconnected piston rod. The vibration level was three times higher when evaluated against the usual level and in comparison to the other throws. This prompted further analysis and investigation.

The cause for the high crosshead acceleration became visible in the crank angle domain. A large impulse with an amplitude of 6g’s (Band pass-filtered: 3–2000 Hz) occurred at around 310° crank angle at the cursor position followed by a second impulse at 110° crank angle (Figure 18, upper right plot).

In addition, the piston rod moved slightly during these events (lower right plot).

To determine the nature and severity of an impulse/impact event from a crosshead acceleration signal waveform, the most appropriate analytic methodology is based on signal timing. Even when peaks’ position in crank angle domain is tracked, the large number of mechanical events occurring in a reciprocating compressor cylinder and running gear components present a real challenge for those attempting to determine the source of a knock. [1]

The synchronization of the vibration signal with crankshaft rotation did not associate vibration peaks to gas load or combined rod load reversal positions along the stroke (upper left plot). Thus, a loose piston to rod or cross-head to rod connection could be excluded.

As introduced previously in the "Vertical Crosshead Force" section, the secondary motion of the crosshead occurs across the clearance between crosshead and crosshead guides.

If this vertical force is in upward direction and exceeds the weight of the crosshead, the crosshead will physically lift off the lower guide and be pressed against the upper guide. After reaching the maximum vertical force, the crosshead will drop down and continue travelling on the bottom guide.

The overlay of crosshead vibration, piston rod position and vertical force showed that the impact-induced vibrations were related to these areas where the crosshead moves periodically against the inner surface of the crosshead’s upper and lower guide. In addition, the dynamic rod position displays unusual movement in these areas. Based on this it was recommended to keep monitoring for any deterioration and to check the crosshead shoe to guide clearance at the next planned service.
Due to the production critical function of the gas compressor and the extensive real time diagnostics and monitoring capability at the disposal of diagnostics engineers, it was decided to modify the surveillance attributes and introduce additional crosshead vibration into the trip logic of the machinery protection system. This allowed continued operation of the compressor with added confidence that if the fault deteriorated, the unit could be shut down safely without incurring additional damage.
One week later, instability of the crosshead vibration started and resulted in the monitoring channel blipping into the new alert level of 7.5g. The decision was made to stop the machine for the planned overhaul.

As the connections between the crosshead/rod and rod/piston were confirmed to be tight, the technicians inspected the crosshead assembly also and found the following issues:

- When the crosshead was removed, wear was found on the body due to contact with the con rod small end bush. The small end bushing was verified and the dimensions found to be ok, but the bushing was not centered on the connecting rod and was in fact flat on the pump side and out on the motor side.

- The bottom crosshead shoe was found to be slightly loose from the crosshead body. The lock plate was open.

Findings highlighted that the crosshead was working with reduced clearances as a result of the bottom shoe being slightly loose. This was as a consequence of the small end bush not being fully centered, initiating a state of no axial clearance and friction within the crosshead body surfaces.

The abnormalities were rectified and the machine was put back in service. The resultant crosshead vibration level of 2g pk was normal and three times lower than before. Condition monitoring had enabled swift location of the defect and ensured reduced downtime.

### 12 Conclusion

Machinery protection and condition monitoring system data can provide supplementary functionality and vital information when running on the edge of an equipment’s working envelope. Having the authority of extensive real time reciprocating diagnostics and the facility for continuous trending and modeling, this opens up additional opportunities for machine operation under extenuating circumstances. The augmented capability derived ensures prompt and expedient diagnosis of abnormalities and confidence in running with these defects while minimizing and mitigating the associated threats.

### 13 References

[1] Rossi, Gaia Localization of reciprocating compressor components faults using on-line dynamic correlation of impulsive vibration signatures with calculated forces and displacements at individual running gear components; 8th Conference of the EFRC, 2012
