PROVEN METHODS FOR EVALUATING THE HEALTH CONDITION OF PISTON COMPRESSORS

by
Alex Deitermann
Managing Director
KÖTTER PROGNOST Systems GmbH
Rheine, Germany
and
David Jetelina
Director of Operations
Kotter Technologies Inc.
Houston, Texas

ABSTRACT

Very often, especially in the oil, gas, and chemical industry, the efficiency of a plant depends on the reliability of reciprocating compressors. Telemonitoring systems can help to make the shutdowns and outages for maintenance purposes predictable. Additionally, consequential damages can be avoided by early detection. This paper describes different methods of evaluating the condition of a particular system. Practical examples from more than 10 years of experience with an online telemonitoring system for reciprocating compressors are presented as proof for the effectivity of permanent online operations monitoring.

INTRODUCTION

Longer machine operating times and the avoidance of hazard-related damage are of critical importance in the operation of reciprocating machines in industrial plants. This is especially true of reciprocating compressors working with process gas in refineries, in the field of natural gas transportation and storage, and compressors in chemical plants, such as for polyethylene production. Large portions of a plant are often dependent on the operational capacity of compressors, so the early planning of shutdowns for maintenance purposes and the avoidance of consequential damage are essential. Permanent online monitoring of operations is also an absolute necessity, for safety reasons, where machinery is operated in a potentially hazardous or explosive environment.

Consequently, recent years have seen increased use of automatic, computer-aided condition monitoring systems for industrial machinery, facilitating both conventional and innovative methods of maintenance and monitoring with the aid of modern metrology and computer technology. Moreover, state-of-the-art, specialized online monitoring systems provide the necessary basis for the introduction of condition-dependent maintenance.

The PROGNOST®-NT condition monitoring system has been in use for more than 10 years, designed specially to meet the requirements of reciprocating compressor monitoring, evaluation, and diagnosis. This system undergoes continuous development with the benefit of the user’s input. Table 1 surveys the history of development and installation of the system.

This paper describes some principles of machine condition evaluation, then continues, with reference to practical examples, to demonstrate experience gained during online monitoring accumulated in the course of routine maintenance work.
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Objectives of General Machine Condition Monitoring

Essentially, the objectives of any monitoring system are

treefold:

- Operating condition monitoring,
- Early fault detection and diagnosis, and
- Efficiency monitoring and improvement.

Operating Condition Monitoring

The job of a pure machinery-monitoring system is to detect

operational faults in the machine, so there are only two conditions,

namely “fault” and “fault-free operation.” Systems of this kind

may be installed as fully automatic facilities, or simply as a

warning system. An automatic system itself initiates disconnection

of the machine when a fault occurs. The warning system just

provides an alarm signal, and the operators themselves must decide

whether the machine should be disconnected, or if the problem

may be solved by some other means.

Early Fault Recognition and Diagnosis

The early recognition of faults is an important factor when it

comes to detecting defects and avoiding consequential damage. Where a fault is signaled at an early level, the machine in question may possibly continue in operation for a limited period of time. However, the development of such a situation must be closely observed, and a shutdown for maintenance should be scheduled to intervene as required.

Faults arising or produced can be diagnosed by referring to machine data collected for monitoring and early-detection purposes. Various analyses are carried out in the process, localizing the fault in terms of place and operational effect, so that an estimate can be arrived at regarding necessary replacement parts and anticipated period of the outage. Trend analysis is particularly important in this context. Trends are calculated, for example, by hourly averaging of recorded machine data, and provide a means of tracing the (long-term) development of machine state.

Efficiency Monitoring

Efficiency is another characteristic component of the condition of a machine. The primary purpose of efficiency monitoring is not to detect faults, but to record changes in process parameters or machine parameters that influence the efficiency of energy transfer to the compressed gas.

The efficiency of a reciprocating compressor may be determined, for example, by finding how indicated compression power relates to the drive power required—for example, the power consumption of an electric motor.

Systematic Condition Evaluation

A specific monitoring strategy has to be defined for every single machine, taking into account the requirements of safety monitoring and machine status evaluation, and involving mainly the following factors:

- Failure and maintenance history of the particular machine,
- Importance to the production process,
- Design or type of construction of the compressor,
- Plant operating mode, and
- Availability of the machine.

Systematic condition evaluation using the described methods is based on direct and indirect condition variables, the following of which have greatest influence according to particular requirement:

- Machine vibration,
- Indicated cylinder pressure,
- Wear parameters, e.g., piston rod position, and
- Quasi-static process data, such as temperatures, flow rates, etc.

Once these factors have been evaluated, the necessary monitoring modules are chosen individually to suit the particular situation. The described systems, installed at present, base their operations on one, several, or all of the modules listed (Figure 1).

![Figure 1. Sensor Positions.](image)

### Table 1. Telemonitoring—History of Development and Installations (Excerpt).

<table>
<thead>
<tr>
<th>Industry</th>
<th>Installed</th>
<th>Online System For</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical</td>
<td>2000</td>
<td>Triplex diaphragm pump (5000 psi/940 hp)</td>
</tr>
<tr>
<td>Chemical</td>
<td>2000</td>
<td>Various process gas compressors</td>
</tr>
<tr>
<td>Chemical</td>
<td>2000</td>
<td>LDPE hyper compressors (2750 bar/36,000 psi)</td>
</tr>
<tr>
<td>Refinery</td>
<td>1999</td>
<td>4 cylinder process gas recip (2.5 MW/3500 hp) for multiple services</td>
</tr>
<tr>
<td>Chemical</td>
<td>1999</td>
<td>4 cylinder vertical piston compressor in a polypropylene plant</td>
</tr>
<tr>
<td>Chemical</td>
<td>2000</td>
<td>Safety monitoring at reciprocating compressors and pumps</td>
</tr>
<tr>
<td>Chemical</td>
<td>1999</td>
<td>10 cylinder LDPE hyper compressors (33,000 psi/15,700 hp)</td>
</tr>
<tr>
<td>Refinery</td>
<td>1989/90, 93/97/98</td>
<td>10 large multi-crank compressors for damage avoidance and online condition monitoring (0.8 - 3.2 MW/500 - 4500 hp)</td>
</tr>
<tr>
<td>Natural gas</td>
<td>1999</td>
<td>4 cylinder compressor (2.5 MW/3500 hp)</td>
</tr>
<tr>
<td>Natural gas</td>
<td>1999</td>
<td>4 cylinder natural gas compressor (2.5 MW/3500 hp e-motor)</td>
</tr>
<tr>
<td>Natural gas</td>
<td>2000</td>
<td>Two storage compressors (1.7 MW/2400 hp)</td>
</tr>
<tr>
<td>Natural gas</td>
<td>1995</td>
<td>4 cylinder storage compressors (1.8 MW/2500 hp) for unmanned operation</td>
</tr>
<tr>
<td>Refinery</td>
<td>1996</td>
<td>Safety and condition monitoring at 6 large process gas recip (2/4-cylinder) and e-motor</td>
</tr>
<tr>
<td>Chemical</td>
<td>2001</td>
<td>LDPE primary/hyper compressors (46,350 psi/9000 hp)</td>
</tr>
<tr>
<td>Technical gas</td>
<td>1998</td>
<td>Oxygen compressors</td>
</tr>
<tr>
<td>Natural gas</td>
<td>1994</td>
<td>Condition assessment of a 2 cylinder acid gas compressor (2 weeks) with mobile version</td>
</tr>
<tr>
<td>Refinery</td>
<td>1997</td>
<td>Various multicylinder and midstage process gas recip</td>
</tr>
<tr>
<td>Chemical</td>
<td>2001</td>
<td>Compressors in an ethylene gas plant</td>
</tr>
<tr>
<td>Natural gas</td>
<td>1999</td>
<td>4 cylinder recip (2.5 MW/3500 hp)</td>
</tr>
<tr>
<td>Natural gas</td>
<td>1999</td>
<td>Recips and as a basis for service contracts with service companies</td>
</tr>
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</table>
Even more important than the measurements taken are the evaluation and analysis strategy to create not only data but information! Since it has been standard for years to measure, display, trend, and store measured data, the essential improvement is to fully integrate those data into an automated and "intelligent" analysis procedure to create "clear text messages" for users, who are not data analysis experts. More than 20 years of measurement and analysis experience with reciprocating compressors are combined with well known and new scientific methods such as neuronal networks. This intelligence inside makes the system easy to use.

One proven method for the automatic and "intelligent" evaluation of piston compressors is:

- Pattern recognition.

The application of a system integrated pattern recognition for reciprocating compressors requires some special considerations to allow the exchange experiences of similar machines. The basic idea is quite simple. Whenever critical conditions are detected by threshold violations, a condition matrix is generated containing all prevailing condition categories at a particular point of time. Condition patterns are saved automatically and users can add their own comments in plain text, e.g., "discharge valve plate broken." When a new critical state occurs, the monitoring system compares the current pattern with all other existing saved in the database and provides the user with a list by similarity. Pattern recognition is an accepted analysis procedure that supports passive accumulation of experience and also allows operators to compare most varied damage situations and use the percentage match with previous damage situations to make diagnosis as accurate as possible (Figure 2).

Figure 2. Condition Pattern of Vibration Signals. A Graphic Condition Evaluation Is Represented by a Traffic Light System with Red, Green, or Yellow Spots.

EXPERIENCE IN THE RECOGNITION OF RECIPROCATING COMPRESSOR FAULTS

Faults in Suction Valves and Discharge Valves

Faults in suction valves and discharge valves count among the most frequent reasons for an unplanned outage of reciprocating compressors. As a result, either of these valve types is regarded as one of the main wearing parts, so particular importance is attached to the early recognition and precise determination of a defective valve (Figure 3).

The status inside the compressor can be evaluated directly by permanent measurement of pressure in the cylinder. The measured time characteristic of this pressure is also presented in the form of a p-V diagram (indicator diagram), from which various characteristic values are calculated and duly monitored in relation to a limiting value of their own.

Figure 3. Damaged Valve Plate.

Additional condition information on these valves is provided by measurement of vibration in the cylinders, which is analyzed in 36 segments for every crank cycle with each 10 degrees of crank angle and monitored in relation to limiting values. This method is especially created for recips. In comparison with other vibration monitoring strategies that were originally designed for centrifugal machines, this method allows the combination of measured vibration with the function of the compressor, and therefore it renders causes and effects transparent.

The following example demonstrates the characteristic changes in signals associated with damage to a pressure valve. This example is of a refinery operating a two-stage, double-acting reciprocating compressor with an output of about 950 hp to compress hydrogen. Figure 4 shows the vibration measured at cylinder 1, and the cylinder pressure characteristic of the crank-end compression chamber, with valves working normally. The pressure characteristic indicates a uniform curve containing nothing really conspicuous. Cylinder vibration is minimal.

Figure 4. Crank-End (CE) Cylinder Pressure Curve and Vibration Signal in First Stage Cylinder with Valves Working Normally.
Figure 5 shows p-V diagrams of the crank-end compression chamber in the first and second stages. In this example, the monitoring system signaled exceeding of the vibration limits in the first stage cylinder and disturbance of the p-V characteristic values of the crank-end compression chamber.

Figure 6 reproduces the vibration characteristics and pressure characteristics recorded at the time of the alarm. There is increased vibration in the cylinder here during almost the entire revolution of the crank, less vibration being evident only within the discharge phase (70 to 180 degrees crank angle) of the crank-end compression chamber. Compression now rises more steeply and reexpansion is distinctly flatter than before, beginning about 10 degrees later (refer to Figure 2). The changes are even more obvious in comparison with the first and second stage p-V diagrams (Figure 7).

The p-V diagram of the first stage is visibly distorted and no longer reaches the intermediate pressure of 294.4 psi. The compression ratio in the second stage rises because of the fall in discharge pressure in the first stage. The displayed interstate pressure is determined by the sister machine. Analysis produces the following conclusions:

- The compression curve is steeper, but the reexpansion curve is flatter, so in these regions, gas is passing into the cylinder from the discharge side. This can only be attributable to the high level of pressure on the discharge side, and is normally prevented by the sealing effect of the discharge valves. Accordingly, the problem could be due to a damaged pressure valve.
- Any other cause, such as leakage along the piston rings, for instance, can be ruled out, because in that case, the direction of gas overflow alternates between crank-end and head-end cylinder space during one revolution of the crank. Compression and reexpansion are then steeper as a result.
- The detected increased vibration (which has also produced an alarm signal) is caused by flow-induced noise and chattering of the damaged valve.

The special benefit of a condition monitoring system for this kind of “standard” failure is the effective determination of the damaged valve that leads to shorter shutdown periods and reduces the number of replaced valves. As a consequence the maintenance strategy changes: the compressor is only shut down for the replacement of the faulty valves. All other valves remain on the compressor so the number of needed spare valves also reduces. The total cost reduction will therefore result in saving production, spare parts, and maintenance personnel.

Faulty Stuffing-Box

The stuffing-box seals the compression chamber from the outside along the piston rod. Just the crank-end compression room have packing in most double-acting compressors, although the outside cylinder sides have one as well in a limited number of cases.

Figure 8 illustrates two crank-end p-V diagrams of a two-stage double-acting hydrogen compressor in good condition. The compressor has lubricated cylinder liners.

After a period of time, in this example, the monitoring system signals that p-V characteristic value limits for the second-stage cylinder are being exceeded. The trend curve of the hourly mean values recorded over the six weeks prior to the first alarm messages is shown in Figure 9.
Figure 8. p-V Diagrams (Volume Display) of Crank-End Cylinder Spaces in First and Second Stage with Stuffing-Box Packings in Good Working Order.

The intersection point and the rise in the expansion curve have been changing continuously over recent weeks. The pressure curve is now reaching suction pressure at an earlier stage after passing through top dead center (TDC), so the expansion curve passes through the suction pressure at lesser crank angles (refer to Figure 8). The steepness of the expansion curve before passing through suction pressure is reducing all the time. The p-V diagrams shown in Figure 10 also clearly illustrate the changes described above.

Other conspicuous features appear in comparing the cylinder pressure curves shown in Figure 11 for one revolution of the crank, before and after occurrence of the alarm.

Reexpansion now begins even a little before reaching dead center, and then rises at a distinctly steeper rate than before, so now the pressure characteristic is already reaching suction pressure at a crank angle of about 200 degrees. The remaining cylinder pressure curve looks almost unchanged.

The cause is likely to be a leak, allowing gas to overflow from the cylinder to the outside. In theory, the problem could be attributable to a faulty suction valve, damage to the stuffing-box packing, or, possibly, to a worn piston sealing ring.

Figure 9. Trend Characteristic (Hourly Mean Values) for p-V Characteristic Values in Second Stage Within the Period of Approximately Eight Weeks Before Replacement of the Stuffing-Box.

In the present case, however, the defect is unlikely to be in the suction valve, since such a fault would not just express itself in changes in reexpansion. Overflowing of gas in the direction of the head-end compression chamber can also be ruled out, since no changes were to be seen at that point.

Accordingly, the reason for the change must lie in the stuffing-box of the second-stage cylinder. When the maximum pressure difference is reached between cylinder space and adapter, and where piston speed is low when close to dead center, quite a substantial amount of gas overflows from the cylinder space to the outside, duly producing a visibly flatter reexpansion curve. During the compression cycle, the leak is less severe because piston speed is distinctly faster, so only a small change will be seen at this point.

Wear inside the stuffing-box is usually increasing over a long-term period like months. Monitoring the p-V diagram is the way to observe the ongoing process of the wear and to determine the point of time when the replacement of the stuffing-box is also commercially effective.
Piston Ring Wear

One recognized method of determining piston rider ring wear is piston rod position or rod-drop analysis, in which a proximity sensor continuously measures dropping of the piston rod during operation. Over time, wear and tear in the piston rider rings produces a measurable drop in the piston rod.

In the described installation the proximity signal is recorded continuously and averaged 36 times per stroke—for every 10 degree segment. This seems to be necessary since many influences like misalignment, oil film, floating of pistons, various speed, and load affect the dynamic measurement permanently. Additionally it is absolutely necessary to provide various thresholds related to various operation states.

In the following example, a twin-crank, double-acting compressor operating with hydrogen mixed gas was producing alarm messages because it was exceeding the rod-drop limit. The trend curve reproduced in Figure 12 emerged after observing rod-drop measured values as they developed over the space of a few days.

The four chosen segments reproduce the mean clearance between piston rod and sensor within the crank-angle ranges zero to 10 degrees, 80 to 90 degrees, 170 to 180 degrees, and 350 to 360 degrees.

An exponential change in the measured clearance is evident over a period of about six or seven weeks. After initial warnings were given on reaching the first alarm threshold, the machine first remained in operation so that further developments could be observed. The piston drop was approximately 0.0062 inch at that time. About 10 days later, the second alarm threshold was passed as well, and the machine was shut down for a maintenance outage.

The piston rings were found to be severely worn on examining the machine. More detailed investigation of the cylinder’s lubrication revealed a partially obstructed supply of lubricating oil. Accordingly, the piston had dropped so markedly because the cylinder space was being insufficiently lubricated.

Without rod-drop monitoring in this particular case there would have been no other indication of the increasing rider ring wear before the piston would hit the cylinder liner and cause serious damage. As an added value the continuous monitoring gives indication of how long the compressor could still be operated before it has to be stopped.

Damage to Crank Gear and Piston

Condition monitoring is very important in connection with damage in and around the crank gear. Loose connections between components in the flow of the load from crankshaft to piston can cause severe consequential damage and may impair the safety of the system as a whole.

Figure 13 shows the vibration signals, pressure characteristics, and rod-drop signals measured at the cylinder of a two cylinder, two staged double-acting compressor for hydrogen mixed gas in a refinery in northern Germany.

Alarm messages were triggered due to passing of the vibration limit for cylinder 1. Operators shut down the machine by manual emergency stop. Thanks to the transient ringbuffer automatically generated by the monitoring system and containing the time characteristics of all directly connected sensors, the cause of the fault could still be determined even with the machine shut down.

To provide for comparison with the good condition signals shown in Figure 13, Figure 14 reproduces the characteristics the endaround memory had recorded for cylinder pressure, piston rod position, and cylinder vibration.

Figure 13. Cylinder Pressure Characteristics, Rod-Drop, and Vibration in Cylinder 1 in Good Condition.

Figure 14. Time Characteristics of Measured Signals for One Revolution of the Crank at Cylinder 1 with Screw Connection Between Piston Rod and Crosshead Detached.
Severe vibrations are noticeable within the range from zero to about 40 degrees crank angle, caused by an impact at TDC. This impact was evidenced in situ by a clearly audible noise.

Given the high amplitude of the vibrations, the problem was unlikely to be caused by a sticking valve, or any other fault in and around the valves. However, the overcoming of mechanical clearance in a connection within the power train can produce a similar type of signal. The timing of the pulse, coinciding as it did with the change in direction of the piston, very much indicated mechanical clearance in the region of the piston, piston rod, or crosshead, all of them oscillating components.

The clear indication for cylinder 1 given by the monitoring system helped to reduce the downtime significantly. Instead of guessing where the strange noise came from, it took only 11 hours for checking and fixing the problem and to bring the compressor back into operation again. The examination of cylinder 1 revealed that the fault was caused by a detached screw connection between the piston rod and the crosshead. Because of the early detection of this mechanical problem, subsequent damages to the piston rod and the crosshead were prevented. In similar cases damages with more than $30,000 spare part costs for failed parts have been subsequently developed.

CONCLUSION

The use of specialized systems for the online condition monitoring of reciprocating machines allows rapid, reliable detection, and accurate allocation, of faults. Possible damage can therefore be recognized at an early stage, allowing prompt action to be taken before any more serious harm can be done. These systems create a foundation for the introduction of condition-based maintenance, making it easier to schedule appropriate and sensible intervals between duly shortened machinery shutdowns for maintenance purposes.

The analysis and diagnosis of faults may be based on a variety of dynamic measured variables like vibration, cylinder pressure, piston rod position, etc. Quasi-static measured values are added to these data, for instance temperatures, flow rates, etc. The monitoring and diagnostic system’s selective combination and integration of all measured data recorded within the system provides a well-founded evaluation of the machine condition, giving the maintenance operator reliable information of the location and operational significance of the cause of a fault, and allowing him to schedule the deployment of manpower and parts in advance—an important point in terms of economy.

As production capacity expands, with its associated reduction in manpower resources, the importance of systems for online condition monitoring of reciprocating compressors will grow. Plant operators have come to recognize the economic significance of machine condition monitoring, especially in regard to machines whose critical role could stop the production partly or of a whole plant during failure. Often, investment in such a system will pay for itself simply through avoiding a production stoppage lasting just one day. The industry acceptance of online systems has been accelerated, too, by the possibility of unmanned operation of compressor systems through remote monitoring, for example over the Internet.

In light of the results outlined above and based on industry demand, it is clear that online telemonitoring systems provide clear advantages to both—the plant operator and maintenance responsible can significantly affect plant profitability in terms of machine reliability, efficiency, scheduling of maintenance, and of course the important area of safety and environmental issues. Already it has been proven and it is envisioned that “intelligent” online monitoring technology will continue to be the key to modern maintenance systems of the future.