

## RECIPROCATING COMPRESSOR CONDITION MONITORING

**Steven M. Schultheis, P.E (TX and DE USA)**  
Rotating Machinery Team Lead-Upstream East  
Shell Global Solutions (Malaysia) Sdn Bhd  
Kuala Lumpur, Malaysia

**Robert Parchewsky**  
Regional Manager, Connectivity Europe, Remote Monitoring  
Shell Global Solutions International  
Rijswijk, Netherlands

**Charles A. (Chuck) Lickteig**  
Senior Engineer  
Shell Global Solutions (US) Inc.  
Houston, Texas, USA

### ABSTRACT

Technology for reciprocating compressor condition monitoring has been around since the 1950's, however until the last 20 years or so it seemed that only the pipeline companies spent much effort on this activity. Technology has advanced, and there are very effective approaches to monitoring and protecting reciprocating compressors on the market today. While pipeline operations are pulling out their reciprocating compressors, this machine is still the workhorse of refineries, chemical plants, and oil production facilities. As a result a new generation of interest has developed in effective condition monitoring of reciprocating compressors. This paper will discuss risk based decision making in regard to measurements and protective functions, online versus periodic monitoring, proven and effective measurement techniques, along with a review of both mechanical and performance based measurements for assessing machine condition. Case histories will also be presented to demonstrate some of the concepts.

### INTRODUCTION

Each year at the Turbomachinery Symposium in the reciprocating compressor discussion group the focus of the discussion is primarily on condition monitoring. With all the other technical issues related to reciprocating compressors that could be discussed, this is usually the topic that generates the most interest. Past topics have included vibration monitoring, rod drop monitoring, pressure-volume analysis, and temperature measurements. In these discussions there are several thematic questions that have come out: How is condition monitoring of reciprocating compressors justified? What measurement techniques are really effective? Should these parameters be used as protective functions or not? Should the measurements be online or periodic? What can I do easily to improve the monitoring on my existing machine, especially without doing a major retrofit project? Is it really worth it? All legitimate questions, but unfortunately there is not one answer for all equipment. Based on risk, a compressor in a refinery hydrotreater service may require a very different condition monitoring approach than a gas lift machine in an oil field application, or a nitrogen compressor in a chemical plant.

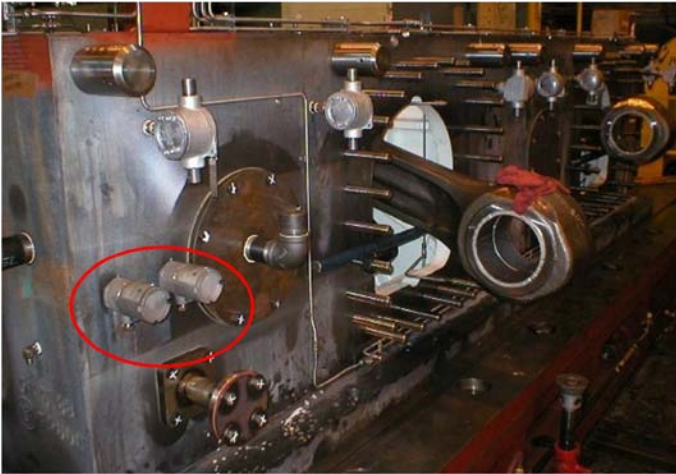
Sorting out the risk and applying the right monitoring approach is the primary objective of any condition-monitoring project.

### AN OVERVIEW OF PROVEN MEASUREMENT TECHNIQUES

For centrifugal compressor trains there is good agreement across industry on how to effectively monitor machine condition. These approaches are summarized in American Petroleum Institute (API) standard 670, and there is little question as to what suite of instrumentation will be used to monitor centrifugal machines. When it comes to reciprocating compressors in process plants, there is much less agreement on which monitoring techniques should be standard but at least API 618 contains some basic requirements. For an ISO 13631 high speed reciprocating compressors, particularly in oil field service there is even less agreement in the industry on applicable monitoring systems. API standard 618 monitoring and protection requirements include high discharge temperature, low frame lube-oil pressures and level, cylinder lubricator system failure, high oil filter differential pressure, high frame vibration, high level in the separator, and jacket water system failure. API standard 670 describes the requirements for installing proximity and casing transducers on reciprocating compressors, but the details of what measurements to make, and how to apply those measurements are left to the OEM and the purchaser to decide. The following are some techniques that have been proven effective across a wide range of applications.

**Vibration:** There are two primary vibration measurements that have been proven effective, measurement on the crankcase, and measurement on the crosshead/distance piece. This is due to the way forces are applied in reciprocating machines. The most common machine design is a balanced opposed configuration and in this configuration the reaction forces in the cylinders are balanced across the machine by the opposing cylinder. Since the cylinders are offset, moments are also set up in the crankcase, and these moments are balanced as much as possible by cylinder placement and cylinder timing. The balance of forces and moments is fine tuned by designing the weight of reciprocating parts and by applying counter weights

on the crankshaft, but it is rare for the balance to be perfect and if a malfunction occurs that upsets this balance of forces and moments, the result is high vibration at 1X or 2X running speed. The flip side is that if the machine support stiffness is reduced, for instance due to grout deterioration or the loosening of foundation bolts, then even in the presence of normal forces and moments the vibration will increase. Catastrophic events such as breaking a piston rod, or losing a counter weight results in a sudden increase in unbalanced forces and moments, and may result in very high crankcase vibration.



*Figure 1: Crankcase vibration transducers.*

Crankcase vibration (figure 1) has been used as a basic protection parameter for decades, usually with a mechanical “earthquake” switch. API 618 has specifically eliminated the mechanical switch as an option because of instrument reliability issues and this measurement is now mostly made with solid-state electronic devices. Most compressor OEM’s specify a crankcase velocity alarm/shutdown level and the most common configuration is to mount the transducers on each end of the crankcase about halfway up from the base plate in line with a main bearing. This configuration enables the transducers to see the effects of both unbalanced forces and moments while minimizing the number of measurements. It is then also possible to monitor changes in relative vibration amplitude and phase in the event that grout breaks down or forces and moments become unbalanced in the machine. An alternate approach is to install a transducer at each main bearing, but this is much less common. The advantage of this approach is that information is available online regarding the operating deflection shape of the crankcase.

Since the purpose of these transducers is to measure running speed related vibration, a low frequency transducer is required, and velocity is the normal measurement parameter. The crankcase vibration acceptance criteria used by most OEM’s is also in velocity. An option is to use a low frequency accelerometer and run the signal to a dual path monitor, allowing both the measurement of low frequency running speed related vibration as well as the high g excursions associated with a major impact event in the crankcase.



*Figure 2: Crosshead Accelerometer installation.  
Courtesy of Bently Nevada.*

The other primary vibration measurement that has proven to be effective is to measure acceleration on the crosshead or distance piece of each cylinder (Figure 2). In this case the idea is to measure the mechanical response of the assembly to impact events. Malfunctions such as liquid carryover, loose piston nuts, loose crosshead attachments valve problems, clearance problems and many others can be identified with this measurement. For more than 100 years mechanics have been listening to the knocks and rattles of reciprocating compressors to assess the health of their machines. An accelerometer on the distance piece of a cylinder is in essence a microphone that allows those sounds to be recorded, and allows for consistent alarms and shutdowns. Of all the vibration measurements that could be made, this is probably the most effective vibration protection measurement available. If a machine is undergoing catastrophic distress, it will typically be picked up on the crosshead accelerometers. As a result, even for very small, spared, or non-critical compressors in hydrocarbon service a simple (and cheap) accelerometer measurement on the crosshead or distance piece of the machine is easily justified.

An extension of vibration technology is ultrasound analysis. Ultrasound has proven to be the preferred approach to analysis of valve condition. Ultrasonic energy is most often associated with gas leaks, so a valve that leaks is a strong generator of ultrasonic energy. Ultrasound measurements are usually taken in conjunction with compressor pressure-volume analysis, which will be discussed later (figure 3).

Critical to successful implementation is properly set trip levels that are just high enough over the normal operating level to react to mechanical failures, but not high enough to miss the failure prior to catastrophic release. Unfortunately, even when systems are properly configured, catastrophic failures often still occur due to operations restarting the machine to confirm that the trip was real (case history 1). In these cases when restarted the failure is escalated, leading to a large consequence failure. Proper operating procedures need to be in place to ensure that reciprocating compressors are not restarted without proper inspection and engineering assessment.

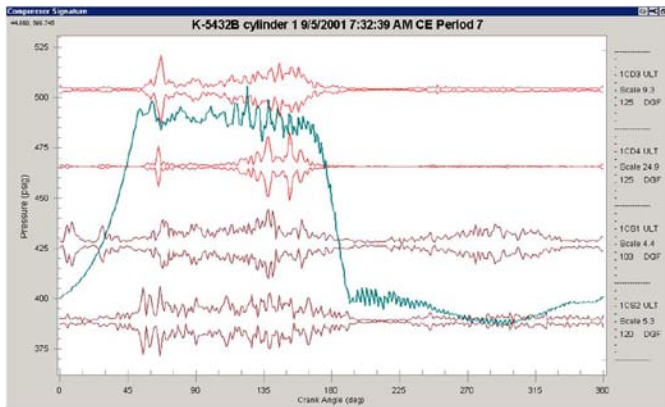


Figure 3: Pressure and ultrasound traces for valve analysis.

Temperature- Machine temperatures are a valuable indication of machine condition and are a primary tool for reciprocating compressor condition monitoring. The primary temperature measurements include cylinder discharge temperature, valve temperature, packing temperature, crosshead pin / big end bearing temperature, and main bearing temperature. Cylinder discharge temperature is one of the protection parameters recommended by API 618 since leaks in rings and valves result in recompression of gas that will raise the discharge temperature. Valve temperatures have proven to be valuable in identifying individual valve problems, but are most effective if the measurement is made in a thermowell in the valve cover (figure 4) so that the reading is taken close to the valve plates or poppets, rather than measuring contact temperature of the valve covers. Packing case temperature or packing leak off temperature can give an indication of packing leakage, while main bearing temperature measurement has been proven effective in preventing major damage due to main bearing failures. Many machines have been saved by shutdowns associated with eutectic or “turkey popper” temperature devices in wrist pin and big end bearings, but there are technology developments currently underway to make this a wireless measurement allowing for a temperature trend. Fossen, Gemdjan, and Turney describe one such technology in the proceedings of the 35th Turbomachinery Symposium in their paper, on radar-based sensors. This and other similar technologies are a major improvement in the protection and condition assessment of connecting rod bearings.



Figure 4: Valve temperature measurement, thermo wells in the valve covers.

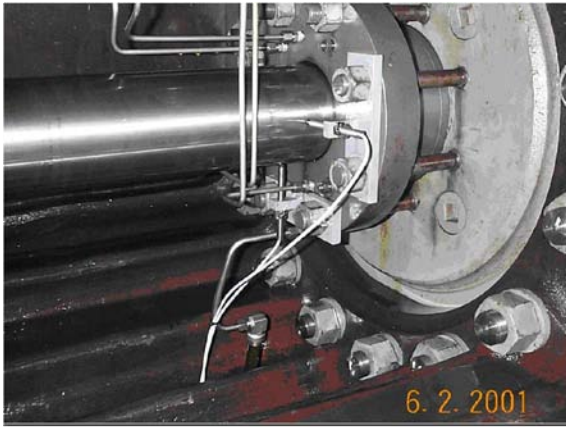
Rod Drop/Rod Runout: In their paper “Rod Drop Monitoring, Does it Really Work,” in the proceedings of the 29th Turbomachinery Symposium, Schultheis and Howard discuss the significant problems associated with rod drop monitoring. While advances have been made to this technology since then, it is still a difficult measurement with many sources of error. At the same time, rider band wear and failure is still a significant maintenance issue with many machines and as a result rod drop monitoring still has a place in the reciprocating compressor condition monitoring toolbox (figure 5).



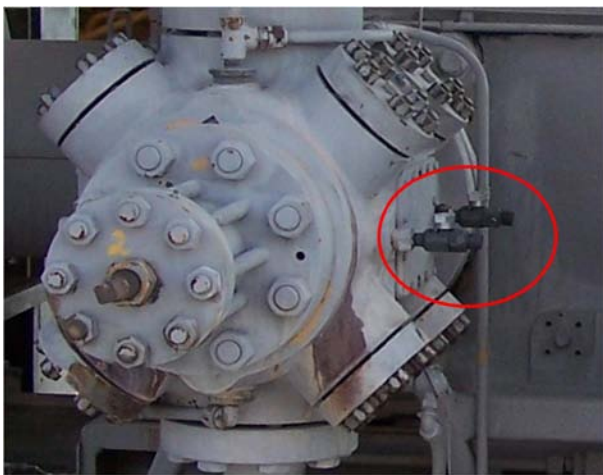
Figure 5: Rider band failure.

An under-utilized parameter however is rod run-out monitoring. Rod drop measurement is accomplished by using the DC component of the proximity probe signal, which is proportional to position. This locates the average distance between the probe tip and the target. In a reciprocating compressor proximity probes are typically located under the piston rods, and are used to measure the rod position, which can be converted to rider band wear (figure 6). There is also an AC component of the probe signal, which is representative of operating rod run-out. Cold run-outs are usually held to about 2 mils pp, and operating run-outs under normal conditions are typically somewhat greater than that, on the order of 2 to 6 mils pp. In the event of a malfunction such as a cracked piston rod attachment, a broken cross-head shoe, or even a liquid carry over to a cylinder, the operating rod run-out will increase significantly. This is data that has historically been ignored, since the rod drop monitors on the market were only measuring the DC component of the signal. Paralleling the signal from rod drop probes into a vibration channel, or configuring the rod drop monitors to display rod run-out in addition to rod drop adds tremendous value to the condition data set on a reciprocating machine.

PV Analysis: PV analysis is a technique that has proven to be very effective in assessing the condition of reciprocating machinery and has been used for more than fifty years. Personal computer technology has significantly reduced the cost of this kind of measurement, and improvements in



*Figure 6: Rod drop probe installation. Courtesy of Bently Nevada.* transducer technology have overcome the technical obstacles such that PV analysis is now available online. Dynamic pressure transducers are used to measure the pressure inside the cylinder over the course of the stroke (figure 3). This allows the analyst to evaluate the condition of the rings, valves, and packing, while at the same time calculating the dynamic rod load, which is the source of the forces and moments described in the discussion on vibration. This requires that a pressure transducer be installed in the cylinder, either on a temporary basis using a valved port (figure 7) or for an online measurement using a permanent transducer installation (figure 8). As described in the next section, it is often possible to predict the PV diagram from existing instrumentation and modeling.



*Figure 7: PV measurement valves on the side of a cylinder.*

## EFFECTIVELY USING THE DATA ALREADY AVAILABLE

In many reciprocating compressor applications, a tremendous amount of data is gathered as a matter of course, and while it is displayed to the operator through a DCS or other control system it is otherwise wasted. In many if not most applications, that data is stored, most often in a data historian, but this is functionally like the black box on an aircraft, data gathered and stored so that it is available in the event of a



*Figure 8: Permanent pressure transducer installation through a valve cover. Courtesy of Bently Nevada.*

problem. That data is in essence under-utilized and since it is available it should be used as a part of the condition monitoring for a compressor, rather than leaving it to collect dust until a problem arises. One approach to utilizing this data that is proving successful is to use a modeling program that compares the theoretical performance of the compressor to the actual performance in order to determine if the compressor performance is deviating from predictions. Such a model can easily reside on the same computer with the data historian, gathering its inputs from that database, and writing results back to the historian. The modeling program is first populated with design data and physical properties of the compressor, e.g., piston diameter, stroke, rod diameter, etc. Then the model is validated against predicted performance information. Next the model is integrated into the network where the data resides and data on the actual process conditions are fed into the model to determine the actual performance of the machine. For a given set of process conditions the theoretical and actual performance of several compressor performance variables can be calculated. Finally performance and deviation from theoretical are written back to the data historian for trending. Software alarms on the calculated values can also be generated to alert operations or maintenance of developing performance issues.

The theoretical model uses first principal type equations to calculate, volumetric flow rate, volumetric efficiencies, power, rod loads, inter-stage pressure, outlet temperatures, and rod reversal. Each of these values for each cylinder or stage is then evaluated against the design or actual values to determine if the performance of the compressor is deviating from prediction. Deviations generally indicate degradation in cylinder wear parts or off design operating conditions. There may be accuracy issues associated with the field instrumentation, however trending performance deviations from a baseline can be as valuable as absolutely accurate performance data.

While it is true that a direct measurement of the compressor pressure volume trace utilizing a dynamic pressure transducer gives the most accurate picture of cylinder condition, a performance calculation based on the pressures, temperatures, and flows already available in the process control system provides a valuable overview of the condition of the

machine. Establishing a baseline of performance when it is known that there are no problems with the machine makes it possible to identify deviations from normal, alerting operations and maintenance to a developing problem, and perhaps giving operations a chance to take action to mitigate the problem before it causes a machine failure.

### A RISK BASED APPROACH TO CONDITION MONITORING

The most common objections to condition monitoring on reciprocating compressors include: “We never needed it before, why now? We don’t have problems with these machines, why monitor them? Monitoring is too expensive.” All of this may be true, it may not be necessary, the machine may not have problems, and a complete system may be expensive. However if the compressor is moving hydrocarbon gas, it becomes easy to justify at least a simple accelerometer based vibration system costing on the order of \$1000 USD, to shutdown the machine in the event of a catastrophic event and to install a performance calculation tool using existing instrumentation and databases. Unfortunately purchasing a condition monitoring system is much like buying an insurance policy. If there is never a failure then there is really no need to have it. One thought to consider is that at least 10% to 20% of all reciprocating compressor (based on author’s experience) suffer a catastrophic failure or a failure that could have been catastrophic if protection systems did not stop the event from progressing. In buying insurance, or in buying condition monitoring the first step is to assess the risk, and then purchase what is appropriate to mitigate the risk specific to the machine and the service. The outcome of the risk assessment should be a list of parameters that will be used to protect the machine, as well as parameters for determining machinery condition. The approach to risk assessment is usually a risk matrix that includes aspects of safety, business impact, environmental impact, and reputation impact. Figure 9 is an example of such a risk matrix. The outcome from this kind of analysis can be either a level of criticality, a safety integrity level, or a standard monitoring approach. One of the ways this type of analysis can be used is to determine if monitoring should be online or periodic. Machines with high criticality ranking will typically require online monitoring and protection.

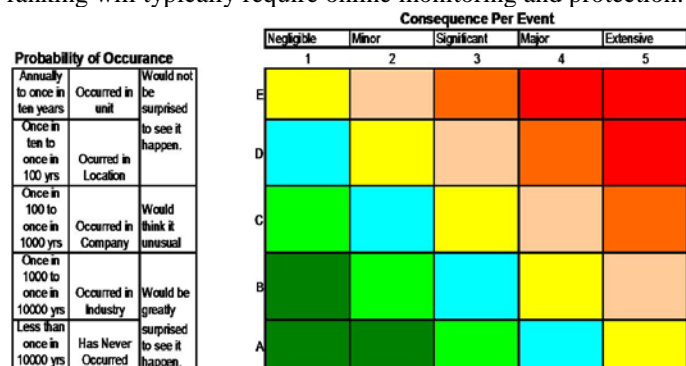


Figure 9: Risk assessment matrix.

Machines that are spared, and thus represent significantly less risk, may only require minimal protection, with periodic monitoring. Machines with high criticality ratings, may not

only justify a complete set of monitoring and protection instruments, it might also make sense to establish the IT infrastructure for remote monitoring and diagnostics.

### CASE HISTORIES

Case History 1: A 1500kW, compressor was installed in 2002 that was deemed critical enough to justify an advanced monitoring and data acquisition system with remote monitoring and diagnostic capability. After 3000 running hours a failure occurred at the crosshead for cylinder 1. The machine had experienced a four-hour shutdown after a power failure and when this was resolved a restart was initiated. Shortly after startup a trip occurred due to vibration/impact at the accelerometer sensor for cylinder 1. The weather was cold and stormy (0 °F), and after examination of the crosshead the failure was obviously a matter of poor lubrication. This compressor had one common oil supply line to the top and bottom crosshead guides and based on the design there was certainly a preference in the oil supply to the top. That issue, in combination with the low temperature and associated viscosity changes in the oil along with startup conditions resulted in a lack of oil supply to the crosshead, and the resultant failure (figure 10). The compressor tripped on impact level set in the online monitoring system, which prevented the machine from more extensive damage and longer outage time. However the system showed a change earlier in both vibration and temperature, unfortunately no response was taken to this change in vibration level and temperature increase until the failure occurred (figure 11 and 12).



Lower crosshead guide

Figure 10: Damage to the lower crosshead guide.

Case History 2: At a Canadian gas plant, a series of reciprocating compressor failures occurred due to yielding of the aluminum pistons from high internal pressure loads resulting in rod failures (figure 13). The compressors were in acid gas service with high molecular weight gas, which resulted in high valve losses. By instituting, on-line performance monitoring of the compressor through the use of a compressor performance model interfaced to the data historian, and by highlighting to operations the appropriate operating envelope based on the results of the performance calculations, repeat failures have been eliminated. The same viewer was used for

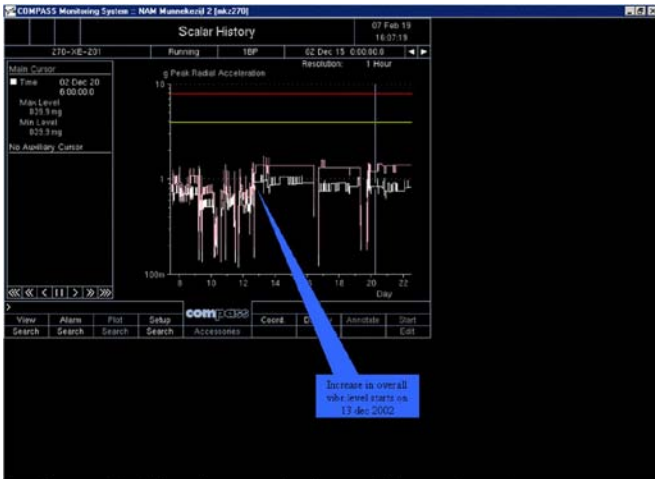


Figure 11: Vibration trend prior to failure (about 1g).



Figure 12: Vibration trip on 9 January, spiked up to 10g or more.

the data historian that the operators were already using for other tasks, so there was little training required to make this tool an effective operator interface (figure 14). On-line monitoring utilizing existing instrumentation and requiring no additional capital expenditures or machine modifications resulted in multimillion-dollar savings as the failures were eliminated.



Figure 13: Piston rod failure.

Case History 3: A broken crosshead slipper was identified in the periodic data collection on the machine. During regular PV data gathering, the analyst also routinely gathered vibration data on the crosshead. Figure 15 shows the vibration trace of the cracked crosshead slipper on the top, along with two



Figure 14: Data display-rod load curves.

historical traces. The vibration signal on this cross head has changed significantly, and the predominant feature is the impact ring down seen in the waveform between 240 and 300 degrees. The crack in the slipper is shown in Figure 16.

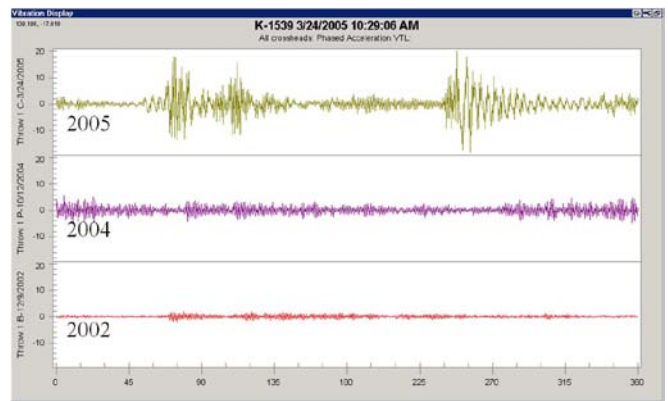


Figure 15: Routine crosshead vibration trace shows development of the vibration as the crosshead crack progressed.

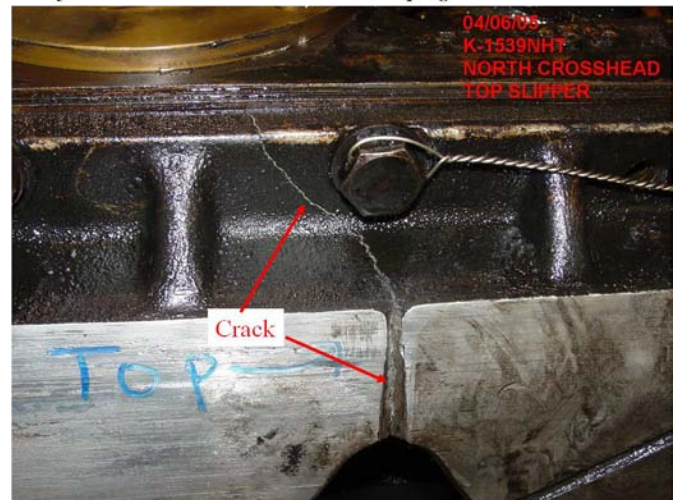


Figure 16: Broken crosshead slipper.

Case History 4: Analysis identified a leaking crank-end discharge valve as the reason for decreased performance. Inspection of the discharge valve during replacement showed what looked like pieces of piston ring or rider band material in the valve. The piston was removed from the cylinder and it was

confirmed that the piston rings were broken into segments and the edges of the rings were broken (Figure 17). The PT trace (Fig. 18) shows the crank-end discharge valve, 1CD3, ultrasonic data indicated this valve leaking during the suction portion of the stroke. Although the piston rings were modified by the failure to have more than one end gap, they stayed in the piston ring groove, between the piston and the cylinder and performed their intended sealing purpose. There was probably more by-pass than intended, but it wasn't noticeable in the flow rate.

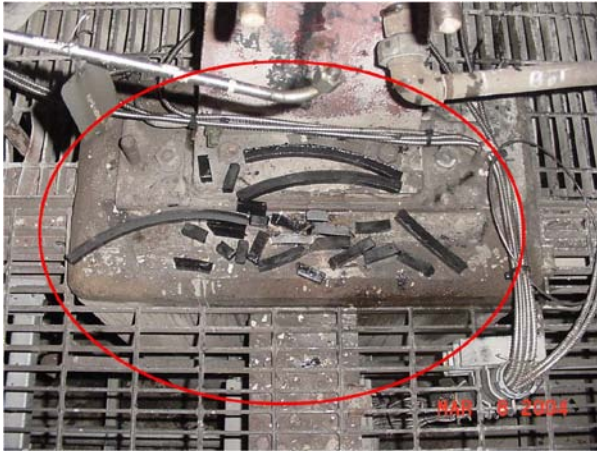


Figure 17: Broken piston rings laid out on the cylinder support pedestal during disassembly.

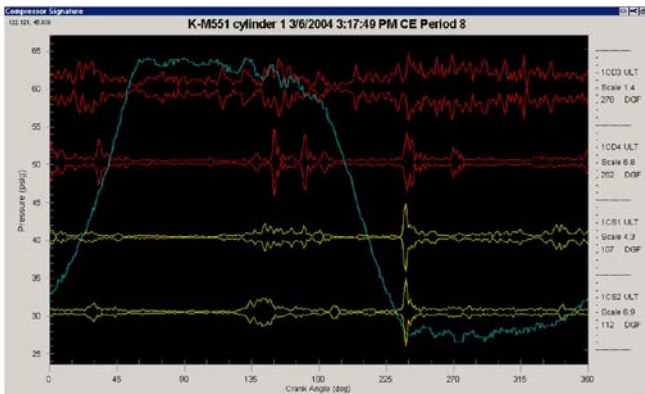


Figure 18: Pressure time and ultrasonic trace showing crank end discharge valve leak.

Case History 5: Again crosshead acceleration measurement saves the day as it tracks the development of a loose piston nut. The acceleration trace (figure 19) changed significantly between the May and June measurements, and there is a distinct impact/ring down pattern present after TDC and BDC indicating an impact as the load shifts from tension to compression and back. impeller.

## CONCLUSIONS

Reciprocating compressor condition monitoring has developed well beyond theory to the point that there are a number of proven and practical approaches to monitoring and protection of these machines. The first step to effective

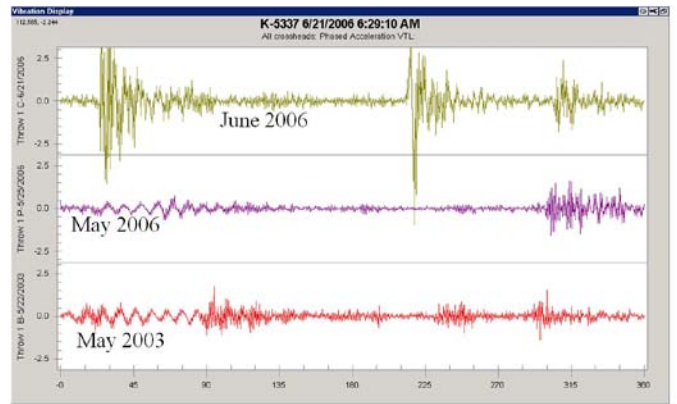


Figure 19: Crosshead acceleration traces showing the development of a loose rod nut. Crosshead knocks after TDC and BDC (ring down). condition monitoring is to take advantage of the information already available on the machine. From there a well thought out risk analysis should be used to guide the application of protective functions and condition analysis parameters, as well as determining whether monitoring should be online, periodic, or a combination of both. For normal wear parts such as valves, rings, and packing, some form of performance analysis is key to the early identification of cylinder problems, allowing for maintenance cost reduction through proactive maintenance planning.

When properly configured and utilized an on-line condition monitoring system that is tailored to the criticality of the equipment can significantly reduce the likelihood of catastrophic failures, and has the capability to create proactive action from operations, while being cost effective and repeatable. On-line monitoring can be quite effective even if only utilizing existing instrumentation and vibration transducers.

## NOMENCLATURE

- microns pp (mils pp) = Vibration displacement
- nX n=1, 2, 3= Frequency at integer multiples of running speed.
- g= acceleration equivalent to gravity.
- PT = pressure vs. time plot
- TDC = Top Dead Center
- BDC = Bottom Dead Center
- PV= Pressure vs Volume plot
- KW= Power measured in kilowatts.

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