ONLINE, CONTINUOUS MONITORING OF MECHANICAL CONDITION AND PERFORMANCE FOR CRITICAL RECIPROCATING COMPRESSORS

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ABSTRACT

This paper outlines the typical problems seen in reciprocating compressors and shows that the existing type of monitoring systems, consisting of vibration level, rod drop, rod runout, and valve cap temperatures, often misses many of them. The advanced compressor monitoring system, which uses dynamic compressor cylinder pressure, directly targets these areas. This system not only provides the information available in the old type monitoring system, but provides much more information concerning the compressor's loading, performance, and efficiency. Operators and maintenance personnel will now have much more information for critical decisions concerning letting a compressor continue in service or shutting it down to help prevent further damage.

A point by point discussion of the type of data available from such a system and the diagnostic information provided by each parameter is provided. Finally, several examples are presented showing the use of this type of advanced continuous monitoring system on a hydrogen compressor in a refinery setting.

INTRODUCTION

Improvements in catalyst and other aspects of refinery operation have greatly extended the time interval that these plants may be in operation without a maintenance shutdown. This has led to increased reliability and performance demands being placed on the hydrogen compressor feeding these processes. To achieve these goals, operators and maintenance personnel need as much information concerning the health and operation of their equipment as possible. This allows a better decision to be made as to whether a compressor showing a problem needs to be taken offline as soon as possible or if it will be able to sufficiently perform until a scheduled shutdown. Additionally, this information can be used to help ensure that all necessary repairs are made during maintenance downtime and, when returning equipment back to operation, that no new problems have been introduced.

In the past, the information available to operations and maintenance has often been limited to what was available from a process control system's measured parameters, such as line pressures and temperatures, and either measured or estimated flow rates. On critical reciprocating machinery, some attempts have been made to add continuous monitoring. These systems mainly monitored the mechanical condition of the machinery, looking for bearing looseness and wear of components by using rod drop and rod runout measurements and vibration levels. Sometimes these systems were also fitted with valve cap temperature monitoring. This allowed an indirect estimate of the internal health of a compressor. Unfortunately, the valve cap temperature measurement method can miss problems and is sensitive to changes in process and environmental conditions, which can make interpreting the information difficult. It is also sensitive to valve design; if there is a large distance between the valve and cap, the temperature probe may have to be mounted internally, greatly increasing the cost. A final downside to valve cap monitoring is the case where the compressor has multiple valves per end. Since wiring and a sensor for each valve cap are necessary, the cost for the system can increase quickly.

Even if we choose to spend the capital investment for valve cap temperature monitoring, we only have a cloudy view of the internal health of a compressor and no measurement of its efficiency, performance, or loading. Fortunately, there are now more advanced continuous monitoring systems for reciprocating compressors that provide information, not only on the mechanical health of the unit, but also a clear picture of the internal health of the unit and its efficiency, loading, and performance. These systems not only monitor the same proximity and vibration sensors of the old monitoring systems, but also the dynamic internal pressure of the compressor. This has become practical within the last three to five years with pressure sensor manufacturers producing sensors with a life expectancy of several billion cycles at normal compressor operating temperatures (less than 350°F). Using the dynamic pressure and suction and discharge temperatures, the system monitor is able to accurately detect leaking valve, ring, and packing conditions. It is also able to measure the loading, performance, and efficiency of the compressor, providing horsepower consumption, flow calculations, flow balance, rod loading, rod reversal, calculated clearance, theoretical (isentropic) horsepower, discharge temperatures, and efficiency. Not only does the advance system provide much more information, it typically has fewer sensors and less wiring required than the old valve cap temperature monitor.

This paper outlines the typical problems seen in reciprocating compressors and shows that the information provided by the advanced compressor monitoring system directly targets these areas. It then discusses the type of data available from such a system and the diagnostic information provided by each parameter. Finally, several examples showing some of the typical diagnostic information available will be presented, showing the use of this type of advanced continuous monitoring system on a hydrogen compressor in a refinery setting.

BACKGROUND

Continuous online monitoring has long been an accepted practice for critical rotating machinery. However, complete online continuous monitoring of critical reciprocating compressors has been difficult to achieve. Unlike centrifugal machines that have one primary moving element, a reciprocating compressor can have hundreds of moving parts so there is not one primary measurement parameter such as bearing vibration that can be used to monitor the overall mechanical condition of a reciprocating compressor.

In order to successfully monitor the condition of a reciprocating compressor, a variety of sensors are required. This paper explores the necessary sensors and analysis techniques required for an online condition monitoring system for critical reciprocating compressors.

TYPICAL RECIPROCATING COMPRESSOR MAINTENANCE COST AND FAILURE MODES

In most installations, the maintenance costs for valves, packing, and rings amount to approximately 90 percent of the overall maintenance budget (Smith, et al., 1997). Table 1 highlights the percentage costs for the separate compressor components (Smith, et al., 1997).

Leaking valves, packing, and rings are not the major concern, even though they are the most significant element of the overall maintenance costs. In today's economic environment, where critical reciprocating compressors are run at full capacity and the cost for an unscheduled shutdown can be upward of $100,000.00 per day in lost production (Leonard, 1997), normally gas leakage alone is not sufficient reason to justify taking a machine offline. Without the additional information provided by an advanced compressor monitoring system over the valve cap temperature monitoring system, no economic evaluation of the cost of operating with leaking components can be made.

Thus, the traditional approach of measuring valve cap temperatures to detect leaking valves and rings is inadequate. Additionally, what is "critically important" is what effect leaking valves and rings have on the dynamic forces of the compressor. Table 2 denotes the causes of compressor failures (Smith, et al., 1997).

Table 1. Breakdown of Maintenance Cost on Reciprocating Compressor.

Valves	50 %
Packing	20 %
Piston Rings	20 %
Piston Rider Bands	7 %
Piston Rods	2 %
Cylinder Liners	0.5 %
Bearings	0.5 %

Table 2. Cause of Failure in Reciprocating Compressors.

Overloading	28 %
Liquid or Foreign Object Ingestion	18 %
Lubrication Problems	12 %
Fatigue, Excessive Stress	9 %
Freeze Damage	6 %
Other Causes, Undetermined Cause	27 %

As Table 2 indicates, leaking valves, packing, and rings are not the primary cause of compressor failure. However, when compressor valves and rings leak, the following occurs:

- Gas discharge temperatures rise
- Percent of rod reversal may diminish
- Rod loads increase on other stages

The effects of these changes can have a significant impact on how long a compressor will operate safely and efficiently. It has been shown that when gas discharge temperatures approach 300°F, valves and rings deteriorate faster, the lubricating ability of oil is diminished, and Teflon products start to degrade. When the percent of rod reversal is minimized, the crosshead pin and bushing do not get adequate lubrication and clearances begin to open. When rod loads are exceeded, components break or have accelerated fatigue failure due to excessive stress. Once again, the advanced compressor monitoring system allows anticipation and measurement of these effects, while the old valve cap temperature monitoring system will be blind to much of this.

Additionally, in the healthy compressor with no leaking components, the valve cap temperature monitor gives no information at all, whereas the advanced monitoring system could warn of overloading due to changes in process conditions or the occurrence of lack of rod reversal.

REASONS RECIPROCATING COMPRESSORS ARE TAKEN OFFLINE

Table 3 shows the results of a survey sent to over 200 users of process reciprocating compressors (Leonard, 1997). It is obvious from Table 3 that operation and maintenance personnel must often decide between taking a leaking compressor offline or running to a scheduled shutdown period. With the valve cap temperature monitor, one cannot accurately tell the magnitude of a leak, only that it might be getting worse because temperature is increasing. The advanced compressor monitoring system, using dynamic, pressure measurements, can gauge the extent of the leak, help estimate the capacity still being pumped, and determine if the leakage is causing other problems, such as lack of rod reversal or lack of capacity. With these data, an informed, economically based decision can be made as to whether the compressor should be shut down. It is known, from the downtime cost provided by Leonard (1997), that a compressor running an extra week could mean a cost savings of a quarter to three quarters of a million dollars for a refinery.

Table 3. Reciprocating Compressor Systems and Components Reported to Cause Unscheduled Reciprocating Compressor Shutdown.

Compressor Valves	36 .0 %
Pressure Packings	17.8 %
Process Problems	8.8 %
Piston Rings	7.1 %
Rider Bands	6.8 %
Unloaders	6.8 %
Cylinder Lube Systems	5.1 %
Instrumentation	5.1 %
Other	6.5 %

Thus, it is seen that one of the most critical parameters to monitor with any online reciprocating compressor condition monitoring system is dynamic cylinder pressure. This is the heart of what we are calling the advanced continuous compressor monitoring system. In the past, this has been uneconomical because the pressure sensor's diaphragm would fail due to cyclic fatigue. On the other hand, if the diaphragm was thick enough to last, its response was slow, distorting the dynamic pressure wave form. Within the last three to five years, pressure sensor technology has improved to where continuous online monitoring of dynamic cylinder pressures is possible and practical.

By measuring head and crank end dynamic pressure versus crank angle and suction and discharge temperature, the critical parameters of the compressor can now be measured, monitored, and trended. The next section will outline the information available from a comprehensive condition monitoring system and the last section will provide some examples from an installed advanced continuous monitoring system on a reciprocating compressor.

INFORMATION AVAILABLE FROM AN ADVANCED CONTINUOUS RECIPROCATING COMPRESSOR MONITORING SYSTEM

Typical Process Points

These include the static temperatures and pressures from various process points.

- Suction temperatures for each compressor end are used in many of the performance and efficiency calculations. It may often be a single temperature point for both ends of a compressor cylinder.
- Discharge temperatures for each compressor end are used in many of the performance and efficiency calculations. If possible, measure each end's discharge temperature.
- Compressor oil temperature and pressure are optional but important points.
- Miscellaneous process temperature and pressure points can be any point the plant may wish to trend.

Vibration Points

- Accelerometer, typically mounted on the crosshead housing, is used to look for mechanical looseness in the compressor's moving reciprocating parts
- Accelerometer, mounted on the frame, is used to detect mechanical failure and foundation failures

Proximity Points

- Rod drop is used to trend the wear of the rider band on the compressor piston. This is typically measured at a fixed crank angle reference or some type of averaging is done to the rod runout information to obtain a static reading.
- Rod runout is also used to monitor rod, cylinder, and crosshead component wear. This is measured from the peak-to-peak proximity readings as the compressor rod moves through a full rotation of the crankshaft.

Trended Calculated Points

These points are calculated or reduced from the crank angle based compressor pressure curves measured by the advanced monitoring system. The cylinder suction and discharge temperatures are also typically used in these calculations. By trending these parameters, the health of a compressor can easily be monitored without having to resort to detailed analysis of dynamic pressure curves. Only in the event of these parameters exceeding their alarm limits will the typical plant operator need to consult with a reciprocating machinery analyst to study the detailed dynamic pressure curves. The monitoring system normally should already have the data stored that the analyst will require for detailed analysis.

- Indicated horsepower is the power required by the compressor to complete the measured gas cycle. This is used to determine the load on the compressor. Also, this is compared to the isentropic horsepower calculated for the gas cycle to obtain isentropic efficiency. These values should be fairly steady with time, if the load remains constant. Load changes and ambient conditions can cause some change in the measured efficiency. The typical percent efficiency can range from the high 90s to the low 70s. The absolute efficiency number is affected by the design and service of the compressor and is normally of little use. However, we are interested if the measured efficiency drops with time.
- Capacity is calculated based on suction and discharge conditions and averaged. This is the volume of gas pumped by the compressor adjusted to standard conditions (PCRC 84-10a, 1990). If independent process flow information is available, it can be very useful to compare this calculated value to a measured value.
- Flow balance is the ratio of the capacity calculated using the suction conditions of the compressor, divided by the capacity calculated using the discharge conditions of the compressor. By conservation of mass, this ratio should be one, unless the compressor is storing mass. Typically, mass is not stored in the compressor, but there can be leakage through valves, rings, packing, and gaskets. This will cause the ratio to be other than one. The ratio goes above one when a leaking suction valve is present and falls below one for a leaking discharge valve. The acceptable flow balance range can vary significantly from one machine, process, or compression ratio to another. Normally, leakage is considered to be acceptable while the flow balance ratio stays within 1.10 and 0.95. In other cases, the range might be ± 3 percent. In the event of suction and discharge valves both leaking or, in some cases where there is ring leakage, the flow balance ratio may still be near 1.0. Other calculated parameters need to be consulted to ensure that these faults are not present, such as comparing calculated discharge temperature to measured discharge temperature or comparing the calculated compressor clearance to the known clearance value. PV and Log P (pressure) and Log V

(volume) plots can also be useful in looking for these conditions. The next section further explains pattern interpretation.

- Rod load is calculated from the crank angle pressure curves measured from the head and crank end of the compressor, and the inertial effects of the reciprocating mass. This is used to determine if the compressor is too heavily loaded either in tension or compression. The maximum tension and compression values are compared to limits provided by the compressor manufacturer to determine if a problem exists.

- Rod reversal is calculated from the same information as the rod load. The rod must reverse in load from compression to tension and back for a sufficient amount of degrees of rotation in order to provide proper lubrication to the crosshead pin and bushing. A rough guide is that reversal must occur for at least 15 degrees of rotation and have a magnitude of greater than three percent of the loading in the opposite direction. For more information about rod load and rod reversal, consult API 618 (1995). Alarms can be set to check for a minimum number of degrees of reversal for both directions of loading.

- Peak pressure seen in the compressor can be useful in detecting fluid build up in the compressor or anything else that is reducing the top dead center (TDC) clearance and causing higher compression. It can also be used to detect flow restrictions.

- Clearance estimation is the clearance volume calculated using the measured crank angle pressure trace and the compression ratio measured from the toe points on that trace. Refer to Figure 1 for the definition of toe points. This is useful in comparing it to the known design clearance. It may be possible to detect the accumulation of fluid or oil in a compressor clearance pocket using this calculation. Leakage will also affect this calculation. Refer to both GPSA (1998) and PCRC 84-10a (1990) for methods to calculate the clearance volume.

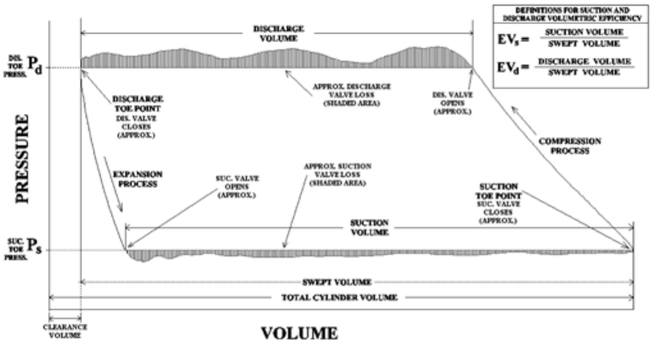

Figure 1. Pressure Versus Volume Plot of a Compressor Pressure Trace Showing Definitions of Toe Points and Volumetric Efficiencies.

- Suction toe point is measured from the crank angle pressure curve. It is the pressure measured in the cylinder at the end of the suction stroke or point of maximum cylinder volume, as shown in Figure 1. This is approximately the suction line pressure and the point the suction value is assumed to have closed.

- Discharge toe point is measured from the crank angle pressure curve. It is the pressure measured in the cylinder at the end of the discharge stroke or point of minimum cylinder volume, as shown in Figure 1. This is approximately the discharge line pressure and the point the discharge value is assumed to have closed.

- Theoretical horsepower is calculated using the discharge toe point pressure and suction toe point pressure to fix a cycle that has isentropic compression and expansion processes. The cycle is assumed to have zero valve loss. The horsepower of this ideal cycle is then compared to the actual, and an isentropic efficiency is calculated. This is the minimum amount of work that is required to pump the given gas mixture from the suction toe pressure to the discharge toe pressure. Refer to PCRC 84-10a (1990) for a method to calculate theoretical gas horsepower.

- Theoretical discharge temperature is calculated using the toe points to set the ratio of compression, the specified gas properties, and the measured suction temperature. The compression process is assumed to be isentropic. It is useful to compare this calculated value to the measured discharge temperature. Variations in the normal relationship between theoretical and measured temperatures can indicate problems with valves, piston rings, unloaders, etc. As a rough guideline, once the measured discharge temperature exceeds the theoretical discharge temperature by 15°F to 20°F would normally indicate that gas recirculation is occurring for some reason. However, the magnitude of the temperature difference can also be affected by compressor design, type of service, and loading. The best way to use this parameter is to watch for a change in the difference in measured and calculated discharge temperature with time. Refer to GPSA (1998) for a method to calculate theoretical discharge temperature.

Crank Angle Information—Pattern Interpretation

Crank angle based curves are taken using a once per turn trigger and, assuming constant speed, taking 360 equally time-spaced readings. A superior method is to provide a once per degree trigger and have the sampling driven by both these signals.

- Internal cylinder pressure taken versus crank angle is often plotted and, with experience, the user can obtain diagnostic information from the curve. This type of plot is commonly referred to as a PT plot. The pressure trace can also be plotted versus swept volume of the compressor. This is known as a PV plot. This also contains useful information for the experienced pattern interpreter. Often the PV is compared to a theoretically drawn PV curve. The theoretical PV curve can be based on either the specific heat ratio (this assumes isentropic expansion and compression processes), the measured polytropic constant for each process (expansion and compression), or using the averaged value of the two measured polytropic constants. Each theoretical model gives a different PV curve. All three types can be useful in spotting problems, but each requires its own specific method of pattern interpretation. The following guidelines for pattern interpretation are based on using polytropic theoretical curves. Bowing of the expansion or compression lines is a visual indication of valve leakage; an "S" shaped curve is an indication of ring leakage. A final way the pressure trace can be plotted is using a Log Pressure versus Log Volume plot. This is often called a log-log plot. This is the total volume of the compressor cylinder, not just the swept volume, thus you need to know the TDC clearance volume of the compressor to use this plot. This converts the PV curve to a parallelogram-looking curve. This can also be viewed versus a theoretically calculated curve. In a healthy compressor, the expansion and compression lines become straight lines with nearly identical slopes. As with the PV curve, bowed lines indicate valve leakage while ring leakage once again shows up as an "S" shaped curve.

Figures 2 and 3 show sets of internal cylinder pressure from compressors in good mechanical condition with minimal leakage. Figure 4 shows an example of a compressor cylinder with a leaking discharge valve on the head end. The compression line shows the classic out-bowing, when compared to the polytropic based theoretical curve. At any given volume, the pressure is higher than the ideal curve would indicate. This is due to leakage from the high pressure source of the discharge line through a discharge valve. The plots also indicate some ring leakage is occurring. This can be seen in the slight distortion at the beginning of the compression line and the outward bowing of the expansion process. Normally, the line should be bowed inward if the only leakage was from a discharge valve. The flow balance on this curve, 0.79, gives a strong indication of the discharge leakage occurring. Figure 5 shows a severely leaking discharge valve in the head end of the compressor. The internal pressure of the head end never reaches the suction line pressure, thus the suction valves never open. The head end is recycling leakage from the discharge line. The work being consumed by this end of the compressor is totally wasted since it has a zero throughput of gas. Figure 6 shows an example of a compressor cylinder with a leaking suction valve on the head end. The compression line shows the classic in-bowing, when compared to the polytropic based theoretical curve. At any given volume, the pressure is lower than the ideal curve would indicate. This is due to leakage from the higher pressure source of the compressing gas, through a suction valve back to the suction line. The expansion line also shows lower pressure than predicted by the polytropic process line. The flow balance, 1.45, on this curve gives a clear indication of the suction leakage. Figure 7 contains a PV showing the classic S shape pressure curve of a leaking set of compressor rings when compared to the polytropic compression process for both the head and crank ends of a compressor cylinder. This figure also shows the PV curves after the rings have been replaced.

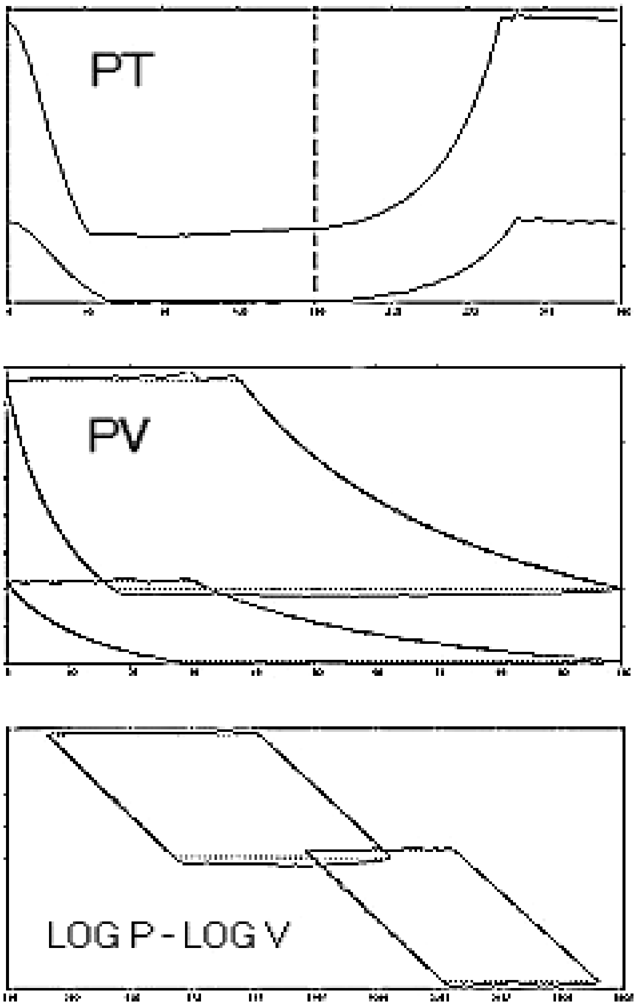

Figure 2. PT, PV, and Log P-Log V Plot for Head End of First Stage and Second Stage Cylinders in a Hydrogen Compressor with Little to No Leakage Occurring.

• Rod loading can be plotted versus crank angle if both head and crank end pressure curves are used. The rod load based on the pressure difference, or gas loading, is calculated for each degree of rotation. The inertial rod load is also plotted. This is from the force produced due to the reciprocating motion of the piston and rodmasses. Finally, a curve is plotted based on the combination of the gas load and inertial load. This is known as the total rod load. Refer to Figure 8 for a typical rod load plot.

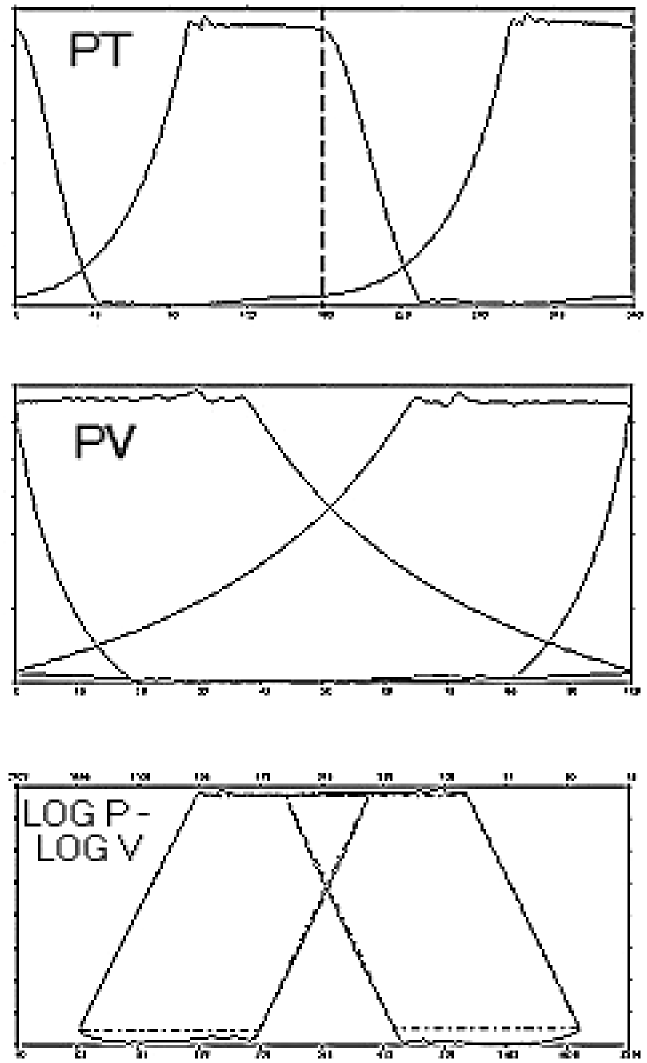

Figure 3. PT, PV, and Log P-Log V Plot for Head End and Crank End of Second Stage Cylinder in a Hydrogen Compressor with Little to No Leakage Occurring.

• Vibration level versus crank angle is also a very useful diagnostic. Typically, the user is looking for vibration events occurring at rod loading reversal. They may wish to plot the crank angle vibration trace and the rod load trace on the same plot to make identification of the reversal points easier. Figure 8 shows the effect that the reciprocating mass can have on shifting the reversal point versus what would be estimated using the gas load.

• Ultrasonic traces are also very useful when plotted versus crank angle. With knowledge of the cycle and when valve events should be occurring, the experienced user can see ultrasonic emissions from leaking valves easily. Leaking valves will show ultrasonic emissions when the valve is seated and closed. Healthy valves will have no ultrasonic emission during this period.

• Rod drop can also be plotted versus crank angle. This can show interesting information, especially if there are mechanical problems present. This is normally only used by a very experienced analyst.

Additional Calculated Information

Volumetric efficiency is measured from the PV pressure curve. The suction volumetric efficiency is the percent of the suction stroke that the suction valves are open. The discharge volumetric efficiency is the percent of the discharge stroke that the discharge valve is open. Refer to Figure 1 for a graphical explanation.

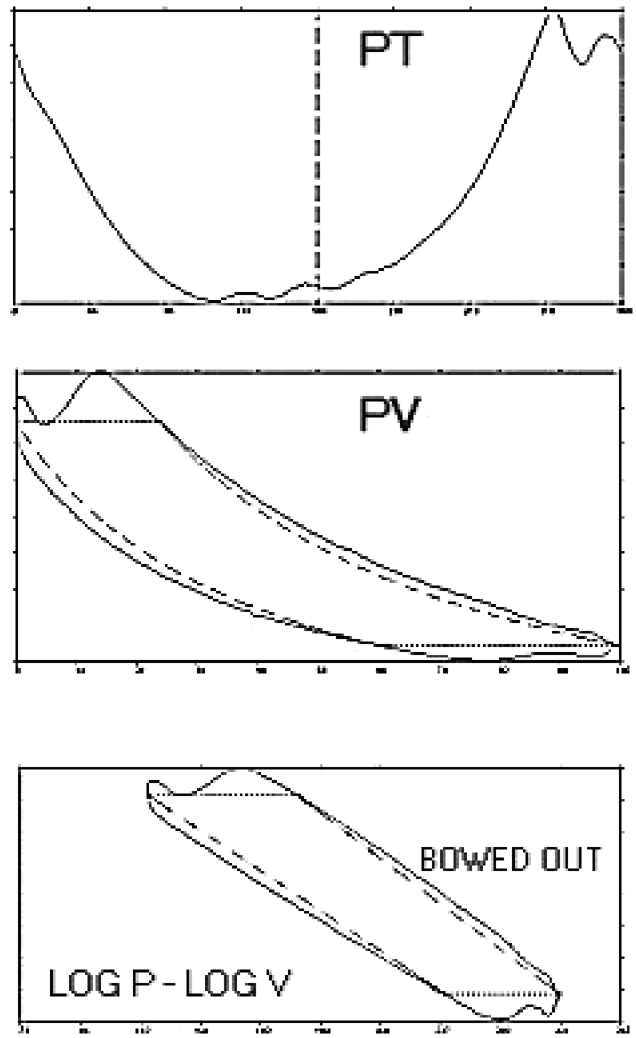

Figure 4. PT, PV, and Log P-Log V Plot for Head End of Compressor with Discharge Valve Leak and Possible Ring Leakage. Theoretical Lines Shown on PV Are Based on Measured Polytropic Constants.

• Valve loss is the amount of work required to force the gas through the suction and discharge valves once they are open. This parameter is useful in determining changes in the valve spring tension and lift. This can also be used to evaluate performance when new valves are installed. Care should be taken because pulsation effects often complicate getting an accurate valve loss number, especially if the toe point pressure line is being used as the zero valve loss reference (refer to Figure 1). Sometimes a special nozzle trace pressure curve is taken and this is used as the reference for the valve loss calculation. This helps to remove pulsation effects.

• Compression ratio is the ratio of the absolute discharge toe point pressure divided by the absolute suction toe point pressure.

• Theoretical efficiency or isentropic efficiency is the theoretical horsepower divided by the measured horsepower times 100.

EXAMPLES FROM AN INSTALLED ADVANCED CONTINUOUS MONITORING SYSTEM ON A RECIPROCATING COMPRESSOR

Rod Drop on Restart

On August 19, 2000, the machine was restarted after a rebuild. A connecting rod on the second stage recycle had broken, severely damaging the compressor, leading to this rebuild. One component

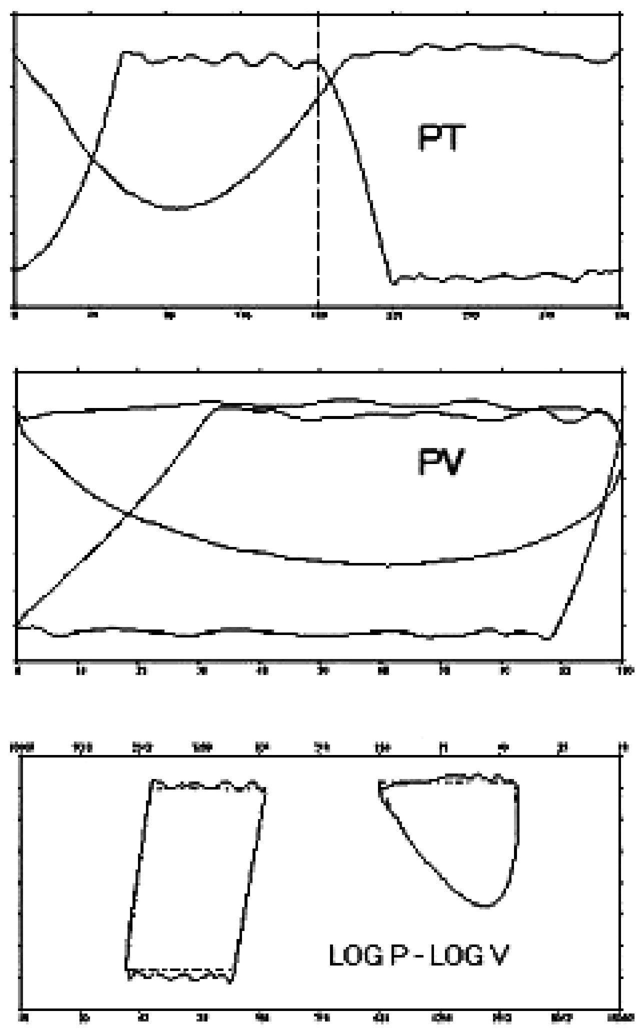

Figure 5. PT, PV, and Log P-Log V Plot for Head End and Crank End of a Compressor with Severe Discharge Valve Leak.

of the monitoring system looks at the rod drop and rod runout of each piston rod. During the first several hours after the restart, the monitoring system picked up excessive rod drop and rod runout of the second stage recycle cylinder. Figure 9 shows the rod drop during the first hours after startup.

From the plot in Figure 9, the rod position, which is measured as the average distance between the rod and a proximity probe located in the doghouse, dropped from 74 mils to about 10 mils during the 15 hours of operation. Because of this, operators, fearing catastrophic failure, shut the machine down and found that the crosshead shoe was severely worn, and the cause was a clogged lubrication line to the crosshead slide.

During the same time period, rod runout was observed to increase. Rod runout is the rod's vertical movement during a stroke. Figure 10 shows the rod runout during the same time period. The rod runout data show the vertical alignment between the piston and the cross-head. As seen in Figure 10, at startup, the alignment was acceptable, but during the first nine hours of operation the runout went from 2 mils to 30 mils. The dotted lines are the alarm points.

Fortunately, these data showed that a severe problem was developing and the compressor was shut down and repaired in less than 12 hours and put back into operation. Had the problem not been detected, it is felt that a likely scenario would have been a seizure of the cross-head and possibly another broken connecting rod. It is possible that this was related to the cause of the original crash and that the clogged lube line was never detected.

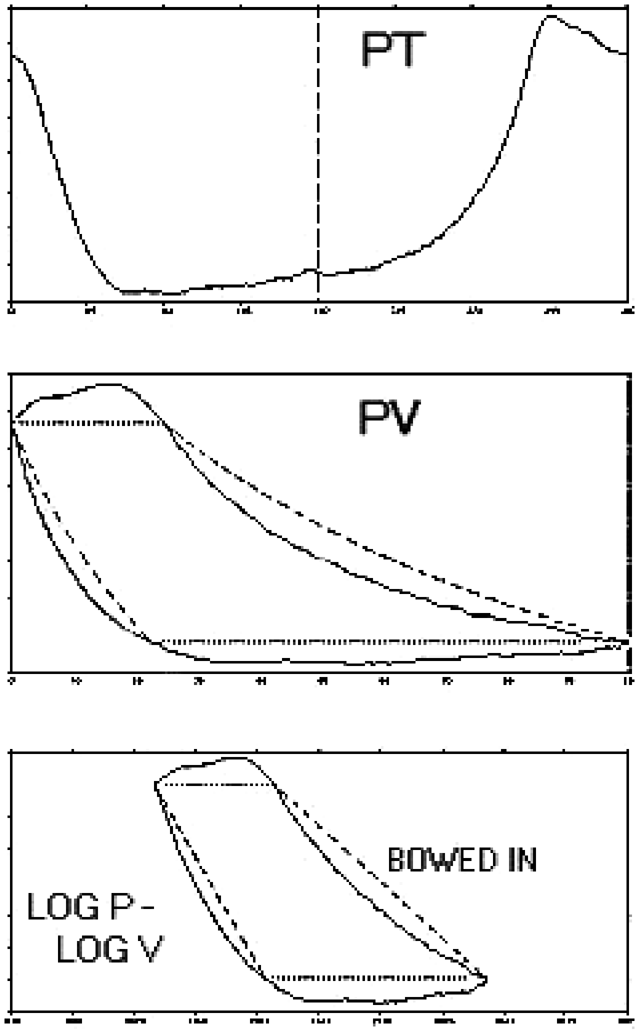

Figure 6. PT, PV, and Log P-Log V Plot for Head End of a Compressor with a Suction Valve Leaking. Theoretical Lines Shown on PV Are Based on Measured Polytropic Constants.

Rod Loading a Design Problem

This machine is heavily loaded due to a lack of hydrogen pumping capacity and, as a result, the rod loads are pushed to the limit. Prior to the installation of the online system, the compressor manufacturer believed the compressor was not overloaded. However, after being presented with data from the online system, the manufacturer discovered an error in their modeling and confirmed that the compressor had been overloaded.

A compressor report from the system in Figure 11 shows that three of the cylinders are typically run with excessive rod loads. These are four inch rods and the limit is set to 100,000 lb. As a result, three piston rods have broken since August 2000. The loading on the compressor has now been adjusted to reduce the overloading on the unit and a redesign of the rods and piston is being considered.

Valve Leak

On January 26, 2001, the rod on the first stage makeup broke and the compressor was down for about a day to repair. Upon startup, operators noticed that the flow balance was high, above 1.10, and the difference between theoretical and actual discharge temperature was high on the second stage recycle. Figure 12 shows the flow balance before and after the shutdown. You can see that immediately following startup, the flow balance was higher for the

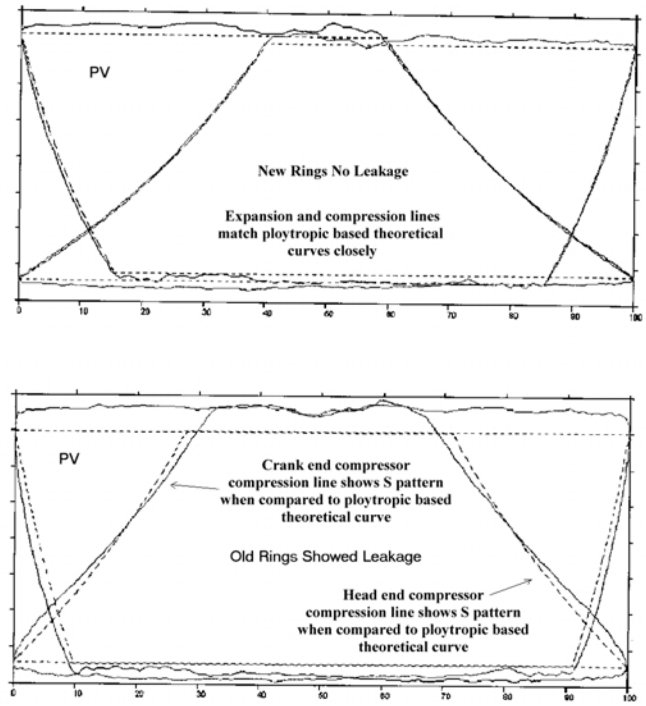

Figure 7. PV Plot for a Compressor with Ring Leakage Occurring, and PV Plot for the Same Compressor after Rings Have Been Replaced. Theoretical Lines Shown on PV Are Based on Average of Measured Polytropic Constants.

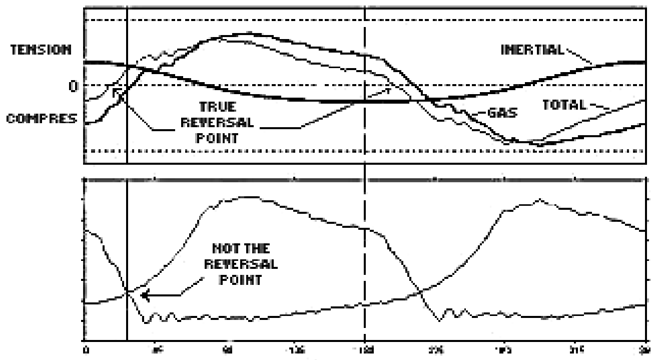

Figure 8. Rod Load Plot and the PT Plot of Pressure Curves Used to Calculate Rod Load Plot.

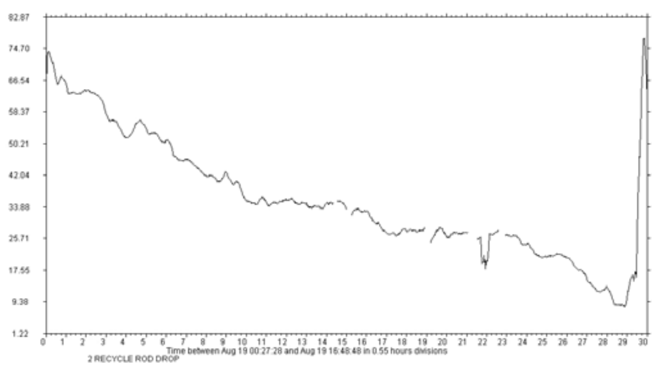

Figure 9. Rod Drop Versus Time.

head end, and soon the head end shot up even more. About a day later, the crank end went up. Since the online system confirmed that rod reversal was sufficient and, due to plant operations, the

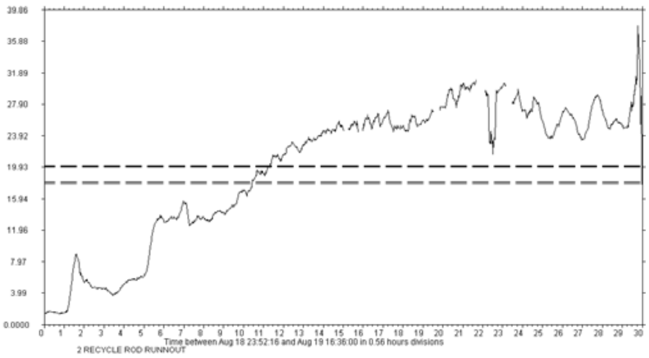

Figure 10. Rod Runout Versus Time.

Compressor Cylinder ID	IHP @ RPM	IHP/ MMSCFD	Capacity MMSCFD	Date	Time
1> 3 MAKEUP HE PRESS	669.0 @ 327.2	38.9	17.1887	1-26-01	10:49:25
2> 3 MAKEUP CE PRESS	528.8 @ 327.2	39.0	13.5712	1-26-01	10:49:27
3> 1 MAKEUP HE PRESS	664.3 @ 327.2	46.5	14.2887	1-26-01	10:49:11
4> 1 MAKEUP CE PRESS	698.2 @ 327.2	46.6	14.9698	1-26-01	10:49:12
5> 2 MAKEUP HE PRESS	668.1 @ 327.2	44.2	15.1155	1-26-01	10:49:18
6> 2 MAKEUP CE PRESS	617.5 @ 327.2	46.7	13.2305	1-26-01	10:49:20
7> 1 RECYCLE HE PRESS	351.8 @ 327.2	15.5	22.6335	1-26-01	10:49:34
8> 1 RECYCLE CE PRESS	270.2 @ 327.1	14.7	18.3810	1-26-01	10:49:35
9> 2 RECYCLE HE PRESS	943.9 @ 327.2	20.3	46.4804	1-26-01	10:49:41
10> 2 RECYCLE CE PRESS	826.5 @ 327.2	19.9	41.5115	1-26-01	10:49:43

	%VOL Dis	EFF Suc	%VALVE LOSS Dis	Suc	%Flow Bal Suc/Dis	Toe Press Pd	Ps	Comp Ratio	Temp F Dis	Suc	% Rod Load Ten	Comp
1>	51.5	87.8	5.0	5.8	1.13	1758.7	888.4	1.96	211.3	86.1	56.2	96.7
2>	51.3	84.4	4.0	4.4	1.06	1690.5	842.9	1.99	211.3	86.1	56.2	96.7
3>	44.0	88.7	3.1	2.4	1.08	404.7	164.4	2.34	229.3	89.1	100.6	87.9
4>	44.6	86.5	1.8	1.6	1.02	447.6	179.3	2.38	229.3	89.1	100.6	87.9
5>	44.7	86.0	3.6	2.2	1.08	871.7	380.7	2.24	227.2	89.7	75.8	100.3
6>	45.9	83.2	1.1	1.3	0.94	860.6	346.3	2.43	227.2	89.7	75.8	100.3
7>	79.1	95.3	10.3	12.8	1.07	1873.9	1500.5	1.25	198.2	144.9	11.1	51.5
8>	79.2	93.8	11.2	11.0	1.06	1880.4	1518.4	1.24	198.2	144.9	11.1	51.5
9>	82.8	96.2	13.7	14.7	0.77	1680.7	1315.4	1.27	190.0	319.6	62.4	113.7
10>	81.4	96.3	13.0	13.4	0.78	1633.6	1276.4	1.28	190.0	319.6	62.4	113.7

Figure 11. Various Calculated Parameters Including Percent Rod Load Limits.

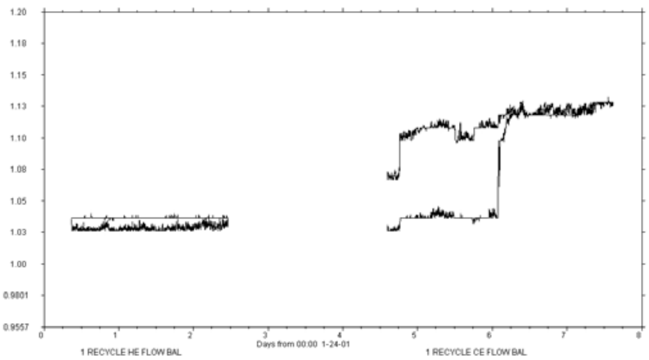

Figure 12. Head End and Crank End Flow Balance Versus Time.

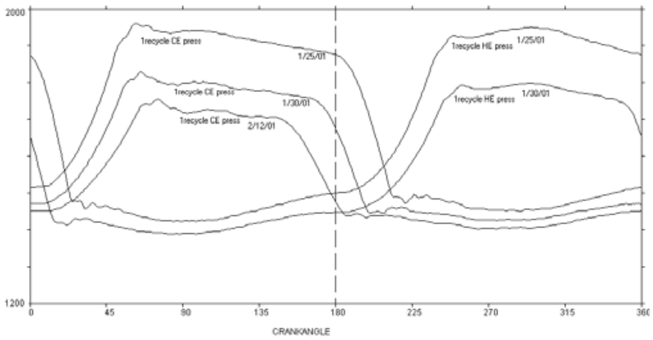

Figure 13. Head End and Crank End Dynamic Cylinder Pressure Traces Shown for Various Times.

compressor was not shutdown for two weeks. Observation of the PT curves confirmed the problem spotted by the trended parameters (refer to Figure 13).

In Figure 13, the PT curves from before and after the shutdown are shown. The PT curves from the system clearly show leaking suction valves. This is evident by the pressure drop on the crank end curve several degrees before 180 degrees and, in the head end curve, the same pressure drop several degrees before 360 degrees. The crank end curves show the severity of the leak continues to worsen.

As a result of the leak, the capacity of this cylinder is significantly reduced. If the leak is allowed to continue, the discharge temperature will rise. Rod loading or rod reversal could be pushed into the danger zone with a possible catastrophic failure as the end result. On February 14, 2001, the suction valves were replaced. Figure 14 shows the obvious damage on the crank end suction valve (top) and a crack in the head end valve to the lower right.

Figure 14. Damaged Suction Valves.

In the few months since the installation of this system, the monitoring system has proven its value in spotting the problems outlined above. In addition to the values shown in these examples, the system continuously monitors horsepower of each cylinder end, vibration and impacting of each cylinder, bearing temperatures, dynamic rod loading and rod reversal, gas pumping capacity, and various static pressures and temperatures.

CONCLUSION

With the development of dynamic pressure transducers capable of withstanding the cyclic and temperature conditions seen in most process compressors, the level of continuous monitoring of reciprocating compressors has taken a generational leap forward. These systems not only provide the information available in the old monitoring systems, they also provide much more information concerning the compressors loading, performance, and efficiency. Operators and maintenance personnel will now have much more information for the critical decision of letting a compressor continue in service or shutting it down to help prevent further damage.

An additional advantage with these advanced systems is the reduced number of sensors required versus many of the older monitoring systems. This leads to reduced installation costs and a system that is easier to maintain. One of the most common maintenance procedures on a compressor is a valve replacement.

With the old valve cap temperature monitoring system, there exists the problem of the temperature sensor and its wiring being in the way, thus being susceptible to damage. With the advanced monitor, there is no sensor involved with the valve cap. Its installed components should only need to be considered during the most major type of rebuild job on the compressor.

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