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KNOWLEDGE-BASED TOOL DEVELOPMENT AND DESIGN FOR RECIPROCATING COMPRESSORS

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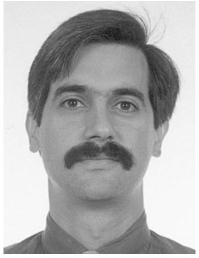

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ABSTRACT

Condition monitoring data collection capabilities have grown over the last decade in large part due to advances in integrated circuits (IC). These advances now permit real time thermodynamic and process calculations on desktop computing platforms. With these capabilities, the sophistication and amount of data available to the users of rotating and reciprocating equipment has grown exponentially. In particular, online reciprocating compressor monitoring systems generate significant amounts of real time data beyond the traditional values of pressure, temperature, vibration, and speed.

With the growth in the amount and manipulation of data, it has become imperative to assist the engineer or operator in quickly locating indications of a potential problem. Many users and condition-monitoring suppliers have designed, developed, and tested knowledge-based tools to achieve this goal. The intent of this paper is to describe the knowledge-based tool design and development process for reciprocating compressors, including the basic assumptions that must be considered and the design features that can make an effective knowledge-based system. A discussion of basic research into reciprocating compressor knowledge-based tool development will be presented in an effort to help those who undertake this development effort understand potential obstacles. The important aspects of an effective system will be described. Finally a series of case histories from operating process plants will show instances where the system worked, as well as instances where problems occurred and how they were addressed.

INTRODUCTION

Today's petrochemical and refining plants utilize a wide variety of equipment classes (fixed equipment, rotating machinery, and reciprocating machinery) to produce the chemicals and fuels required by society. Advances in design tools, information technology, and condition monitoring have enabled reliability improvement for all classes of equipment. Expectation of continuing operation excellence at the plants, environmental concerns, and economic drivers encourage plant stake holders to reduce emissions, unplanned outages, and flare events.

Yet, reliability has not increased equally across each of these three classes. Reciprocating equipment reliability improvement lags behind the other two. In order to close this gap plant owners have turned to condition-monitoring technologies to improve the reliability of their reciprocating compressors.

Condition-monitoring vendors responded to this need by introducing a variety of new technologies, such as the microelectromechanical systems (MEMS) embedded in the online cylinder pressure transducer. These new technologies, along with advances in information systems, have increased the amount of data available to reciprocating compressor operators. In addition, computers can now run real-time thermodynamic, process modeling, and calculation programs creating additional data.

For the rotating equipment engineer tasked with operating a unit or plant, extracting useful information from this large collection of data presents a formidable analytic challenge.

Condition-monitoring systems for reciprocating compressors, which have many more moving parts than an equivalent centrifugal compressor, generate a large amount of raw data. For a critical, large reciprocating compressor, the condition-monitoring measurements include those shown in Table 1. (Less critical or spared reciprocating compressors utilize a subset of these measurements.)

Table 1. Reciprocating Compressor Monitored Points.

Component	Monitoring Point	Transducer
Electric Motor	Stator Temperature	RTD or Thermocouple
	Bearing Temperature	RTD or Thermocouple
	Rolling Element Bearing Vibration	Velocity Transducer
Frame Assembly	Main Bearing Temperature	RTD or Thermocouple
	Crankshaft Position Reference	Proximity Probe/ multi-event wheel
	Frame Vibration	Velocity Transducer
Crosshead	Machine Vibration	Accelerometer
	Guide Temperature	RTD or Thermocouple
Pressure Packing	Case Temperature	RTD or Thermocouple
	Vent Line Temperature	RTD or Thermocouple
Piston and Rod	Rod Position	Horizontal and Vertical Proximity Probes
Cylinder	Internal Cylinder Pressure	DC Coupled Pressure Transducer
	Cylinder/Valve Acceleration	Accelerometer
	Valve Temperature	RTD or Thermocouple
	Discharge Temperature	RTD or Thermocouple
	Suction Temperature	RTD or Thermocouple

Although some of the measurements change slowly (i.e., static data), other measurements require high-speed data collection during the revolution of the crankshaft (i.e., dynamic data). Condition-monitoring systems typically collect frame velocity, crosshead acceleration, cylinder acceleration, valve-cover acceleration, cylinder pressure, and rod position data as dynamic data.

Assuming the measurements in Table 1 are applied to a typical four throw reciprocating compressor, the condition-monitoring system manages over 200 static points and 44 dynamic points (assumes 2 frame velocity [3 static, 2 dynamic values], 4 crosshead accelerometers [6 static, 2 dynamic], 8 rod position [5 static, 2 dynamic], and 8 cylinder pressure [19 static, 2 dynamic]). The analytical tools in most reciprocating compressor condition-monitoring systems provide numerous plotting options for each point, representing the potential for scores, or even hundreds, of plots to review.

Recognizing that in today's economic climate it is not practical to expect an engineer to audit the data presented by this system at regular intervals, many companies and organizations have turned to knowledge-based rules to flag suspect data and potential problems. This paper describes the knowledge-based tool development and design process for reciprocating compressors, including the basic assumptions that must be considered, and the design features that can make an effective knowledge-based system.

TOOL DEFINITION AND DESCRIPTION

The use of outside resources to store and document knowledge has been with the human race for thousands of years, from the ancient Sumerian cuneiform marks on clay to the paperback books commonplace today. Computers brought changes in not only the way we store information, but in the type of information that could be stored.

Writing and drawing enabled the storage of only descriptive information. The content could only say, "go do this" or "check that" but could not perform any of the operations. Computers, programming languages, and digital media enabled storage of both descriptive and operational information. The handheld calculator presents a great example of this. Unlike a math book, which describes how to perform addition, subtraction, and other basic math operations, the calculator actually takes data, performs the operations, and returns a result. By pressing the "PI" button, the calculator also provides descriptive information.

The term "knowledge-based tools" refers to a collection of computational procedures containing both descriptive and operational information. The knowledge base may include first principles or model-based structures, empirically determined structures, or any combination of the two. Obviously, a reciprocating compressor condition monitoring knowledge base includes a large amount of both descriptive and operational information.

Knowledge-based tools do not represent the only mechanism for analysis of the data generated by condition-monitoring systems. Other approaches, such as statistical analysis or, in some cases, neural net analysis can be used. These approaches do not require knowledge of how a particular machine should behave. Instead these approaches detect anomalous behavior in the data set based on past performance. These paradigms have the advantage of never missing an event, but the disadvantage of requiring the local knowledge to disposition each event and develop a baseline for each unique compressor configuration and operational state. As such, these anomaly detection tools fall outside the scope of this paper.

In general, knowledge-based tools can include a wide variety of analytical toolsets and databases. This paper focuses on a subset of those tools, fixed-threshold alarms, fault dictionaries, model-based diagnosis and decision trees, and the ways in which these tools can be combined to define *rules*.

Fixed-Threshold Alarms

Fixed-threshold alarms, by far, find the greatest application in reciprocating compressor monitoring. When a measured value exceeds a preset limit, the monitoring system triggers an alarm. Threshold alarms represent the most primitive knowledge-based tool as they contain only two items: a single descriptive piece of information (the set point) and a single piece of operational information (trigger an alarm). This tool requires so little computational effort that it has been implemented in mechanical, pneumatic, and electronic systems.

Configuring parallel preset limits on the same measurement enables multiple severity alarms. Although very simple to set, process upsets or transient events may trigger multiple alarms requiring the operator to quickly sift through the triggered alarm list to discover the cause of the root alarm event.

Fault Dictionary

When discrete events, such as threshold alarms or measurements, are grouped together in such a way that a particular pattern, or set of patterns, of alarms or events drives an output this system is referred to as a fault dictionary. For example, a simple fault dictionary might consist of two measurement points: a velocity transducer at the drive end of the compressor and a second velocity transducer at the nondrive end of the compressor. If the direct vibration amplitude of either transducer crosses a threshold alarm, a contact closes turning on a light behind a panel that reads "HI FRAME VIB." When both measurements cross, or have crossed, the threshold a contact closes turning on a light behind the "HI HI FRAME VIB" panel. Diverse and complex fault dictionaries can be compiled using computer software programmed with Boolean logic steps.

Model-Based Diagnosis

In model-based diagnosis, a model of the process runs concurrently with the physical process. Input values originating in the field drive the model, and the output of the model is compared to the measured output of the physical process. Differences in the observed values and model's predicated values can be evaluated. For example, the compression process inside a reciprocating compressor cylinder closely matches an isentropic compression process. From this observation, a mathematical model can be constructed where discharge temperature is calculated as a function of gas composition, suction temperature, and compression ratio.

Using a simple threshold alarm, the discharge temperature of the model can be compared to the measured discharge temperature at the cylinder. In contrast to a fixed set point, which could be crossed due to process upset or other transient conditions, this threshold changes as conditions inside the cylinder change. Multiple combinations of threshold alarms and fault dictionaries can be used with model-based diagnosis.

Decision Trees

Decision trees provide a method for mapping diagnostic methodology. Figure 1 shows a simple decision tree describing the problem solving methodology for selecting a generic soda over a name-brand soda when the potential for a name brand coupon exists. Threshold alarms, fault dictionaries, and model-based diagnosis can all be combined within decision trees to provide flexible, robust knowledge-based tools.

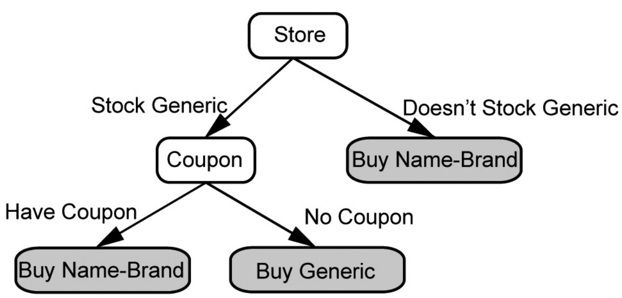

Figure 1. Sample Decision Tree.

Figure 2 shows a decision tree incorporating threshold alarms and fault dictionaries. The decision tree consumes temperature data from "LP Stg3 Suct W" and "LP Stg 3 Suct Temp," performs basic mathematical operations and compares the result to the constant "Suction Valve Coefficient" in the first threshold alarm, labeled "Temperature Threshold Alarm." In a separate branch of the tree, "Leak – Cylinder to…" value is compared to a severity constant. Based on the result of this combination of threshold alarms and fault dictionary, the rule delivers different severities for the "Suction Valve Leak" alert.

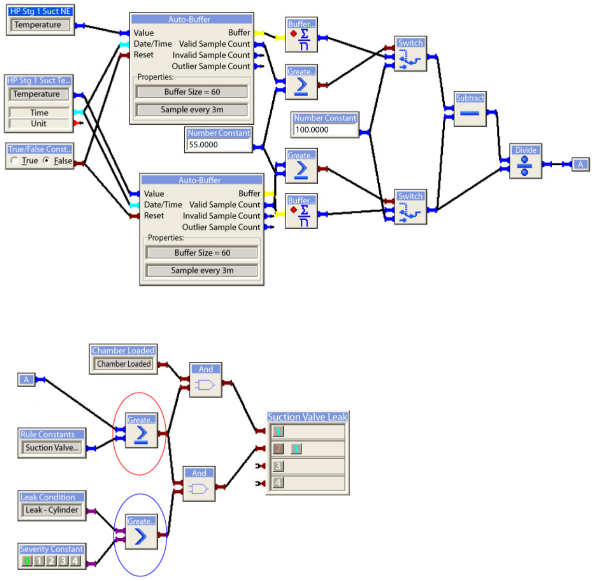

Figure 2. Knowledge-Based Tool Showing Decision Tree Incorporating Fault Dictionary and Threshold Alarms. Red Ellipse Indicates Temperature Threshold and Blue Ellipse Indicates Leak Threshold Alarms.

ROBUST TOOL DESIGN

Threshold alarms, fault dictionaries, model-based diagnosis, and decision trees all seem to be useful tools, but what is the best way to synthesize a useful set of tools for reciprocating compressor condition monitoring? The following describes the design process and lessons learned from knowledge-based tool development for heavy-duty, slow-speed, API-618 (2007) reciprocating compressors.

Characteristics of Successful Tools

A successful rule has only two functions: where no malfunction indicator or malfunction exists, the rule should be silent; where an indicator or malfunction exists, the tool must provide an alert.

Moving from this accurate, but simple, characterization to a robust knowledge-based tool requires that many implicit needs be addressed, some of which have a direct impact on the design process. Here are some of the concerns that arose during the rule development process for reciprocating compressors:

- The rule and rule infrastructure design need to address the human-interface needs. For example, what happens to an unacknowledged alert when the severity increases? How does the display prioritize and drive focus to high-priority alerts during multiple events?
- The rule must perform these functions at all operating and load conditions, including the presence of other malfunctions. Reciprocating compressors have a variety of capacity-control devices including valve unloaders, pocket unloaders, and hydraulically actuated "stepless" unloaders making this a real challenge.
- Condition-monitoring system infrastructure must be pristine. Transducer installation, wiring, system configuration, and information technology (IT) infrastructure must be in excellent condition prior to deployment of the rule to avoid false/missed alerts.
- The knowledge-based tool must be able to differentiate between harmless transient conditions (such as debris passing through a valve) and serious changes in machine condition.
- The tool must be sufficiently flexible to fit into the cultural DNA of the organization. For example, some plants change every compressor cylinder valve during an outage; some change only the

one in distress. To the first plant a knowledge-based tool that indicates exactly which valve is in distress has no value; to the second, it is essential.

- To every rule for reciprocating compressors, there exists an exception. The rule and development environment must be flexible enough to allow deployment over a large fleet and accommodate these exceptions.

Beyond the technical needs, the development process also highlighted infrastructure needs at the plant. Some of those include:

- Involve information technology at the earliest opportunity. Condition-monitoring systems often require data from the distributed control system (DCS) and, therefore, the plant IT locates them in the control room and only connects them to the control network. In this case, information does not get from the rule to the people that need it. Directing this information appropriately requires that the condition-monitoring system be connected to the business network as well as to the control network.
- Establish condition-monitoring system ownership. The way in which a plant designates ownership of the system has a direct impact on the value rule alerts return to the organization. For example, consider a broken wire. In a plant where the rotating equipment engineer owns the machine, but instrumentation and controls (I&C) own the transducer, the rotating equipment engineer detects the problem and must issue a work order describing the problem to the I&C shop. In contrast, if the rotating equipment engineer owns the transducer and wiring, the engineer can make the repair directly.
- Who should get the alerts? The unit operators are in the best position to use the data immediately to take corrective action. But if every alert goes to the operators, they quickly become resistant and ignore the rule alerts. The machine stakeholders need to decide who gets what information. The rule must be flexible enough to allow the alerts to flow to a variety of levels within the organization.
- Rule infrastructure has as much to do with the success of the rules as does the technical content. Data presentation, alarm management, data storage, and other factors all provide important value and features to the end users. Even something as relatively minor as a color scheme can create confusion if the user needs' and expectations are not understood.

Tool Design Process

Figure 3 shows an overview of the design process used today to develop rules for reciprocating compressors. The process looked quite different in the first development phase so the chart in Figure 3 includes a variety of lessons learned.

For example, the initial rule design process began with the effort to map the failure effect on measurements and then proceeded directly to the assembly of the decision tree. During the proof-of-concept deployment, questions from the plant personnel about operation and delivery of alert messages raised several concerns about how to best manage the results generated by the rule. This resulted in the second step shown in Figure 3.

Another gap in the initial rule process became apparent as the initial results of the proof of concept deployment were integrated into the rule design. In adding or modifying decision tree components within the rule, changes could inadvertently result in unexpected operation during regimes where the tool had previously been stable. To ensure that changes to the rule do not impact stability, a test program must be designed and implemented prior to rule development. This test program should incorporate several features to ensure proper rule behavior.

First, the test suite must cover a broad range of machine operation. In practice, this usually reduces to a collection of databases describing normal operation and malfunctions. Data from these sources flows through the rule in a controlled environment. Figure 4 shows such a development tool in which the various components of a decision tree can be monitored during testing. Next, the test suite must be able to execute quickly. Generally, the test suite should take less than 10 minutes to execute. In addition to speed, the test suite should be automated as this improves error trapping and feedback to the developers. Lack of speed and feedback causes developers and rule designers to avoid testing, resulting in poor rule operation. Finally, the test suite should be viewed as a living document, readily modified as lessons learned accumulate from deployment of the rule.

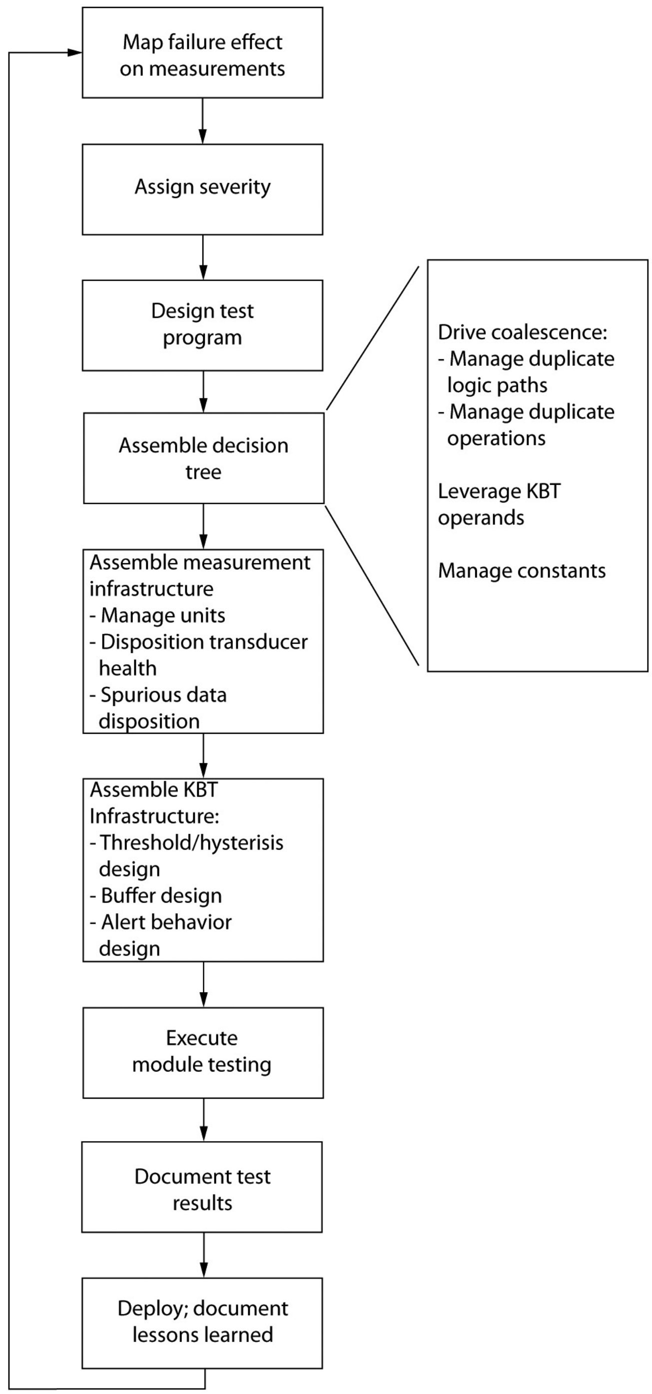

Figure 3. Reciprocating Compressor KBT Design Process.

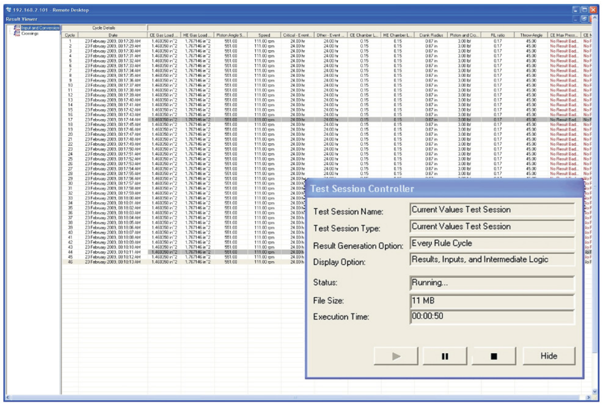

Figure 4. Knowledge-Based Tool Test Session.

Another important change to the process includes the cyclical nature of the development processes. Just as Antony van Leeuwenhok discovered new worlds of microbiology when he applied the tool of the microscope to everyday items, so too will users of rules on reciprocating compressors discover many new and unexpected details about the operation of their compressors. In many cases, this new learning will impact the failure effect mapping and, therefore, trigger the process over again. As the organization deploys rules, this cycle enables corporate learning to be leveraged by the entire enterprise.

The remainder of the paper describes some of the findings from the application of rules to reciprocating compressors.

CASE HISTORIES

Effect of Buffer Settings and Hysteresis on Rule Operations

Easily one of the most frustrating experiences with deployment of rules has been designing the necessary infrastructure for managing data, alarms, and alarm set/reset events.

Anyone tasked with auditing data coming from a reciprocating compressor has likely seen spurious transient events in the data. Problems such as debris passing through a valve, process changes, or gas composition changes can cause short-term deviations that typically do not warrant an alert. How does one manage the rules so that these problems do not trigger an alert, but real machine problems do?

Part of the answer has been buffering, in which the rule averages the data coming in over a fixed amount of time. To date, no real first-principles approach has emerged to determine appropriate values for these buffers. The values in Table 2 have been determined empirically, based on results of installation on API-618 (2007) style compressors.

Table 2. Buffer Settings.

Variable	Buffer Size	Sample Time (sec)	Buffer Length (hours)
Valve Temperature	60	180	3
Packing Temperature	60	360	6
Cylinder Pressure Performance Parameters	60	180	3

Using buffers provides improved rule operation, but the rules require additional infrastructure in order to operate robustly. In order to understand this infrastructure, recall that the opening of this paper included a definition of fixed-threshold alarms and mentioned that such systems had been implemented in many different mechanical, pneumatic, and electronic systems. A simple pressure switch, as shown in Figure 5, provides an example of a "simple" mechanical system that provides fixed threshold alarm capability.

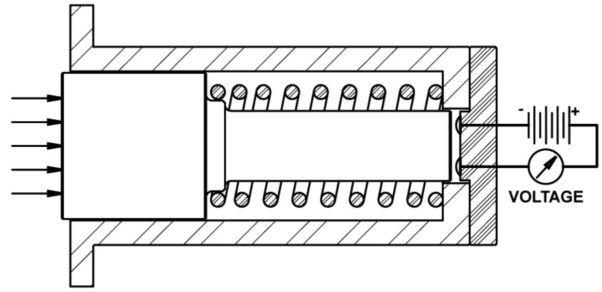

Figure 5. Mechanical Pressure Switch.

Consider what happens to the switch as pressure on the piston face increases: a voltage potential exists at one contact and a voltmeter connects the other contact to the potential source as shown in Figure 5. As the pressure force begins to overcome the spring force, the piston moves toward the contacts. At Point 1 in Figure 6 the piston has experienced sufficient displacement to enable the conducting stem to touch the contacts. This completes the circuit, allowing current to flow from one contact to the other. As a result the voltmeter indicates a value. Point 2 in Figure 6 represents this state.

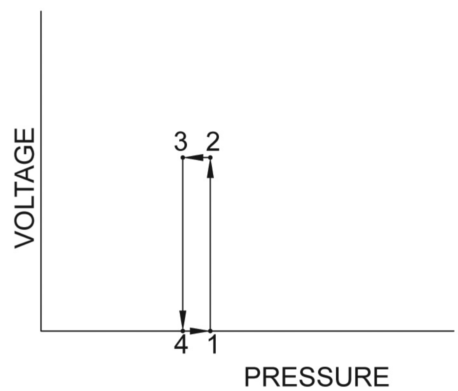

Figure 6. Voltage Potential and Pressure.

Now consider the switch reaction to a decrease in pressure. Reduction of pressure does not immediately change the value of the voltage at the contacts. Friction exists between the piston and the walls, and the piston, stem, and spring have inertia resulting in resistance to movement. The line from Point 2 to Point 3 represents the switch state for this condition. Eventually, the pressure drops low enough that the spring force overcomes friction and inertia allowing the stem to move away from the contacts, breaking the circuit. Point 4 represents this state.

The difference between the pressure that causes the switch contacts to close and the pressure that causes them to open again is referred to as hysteresis. All mechanical switches have some degree of hysteresis and designers for these systems strive to keep the values bounded as excessive hysteresis has a strongly negative influence on switch operation.

Software systems suffer from no such limitations. In fact, software alarms have no intrinsic hysteresis at all. This leads to some difficulties. Figure 7 shows a typical repetitive advisory scenario. The top pane shows the process variables, in this case rod load tension, rod load compression, and degrees of reversal. The middle plot shows the intermediate buffered values, averaged over three hours. As expected, the buffering reduces the appearance of the transient events at 16:21 and 16:35 in the output of the rod load and reversal functions. Yet, as shown in the lower plot in Figure 7, the rule still changed state. The cause of this change in rule state requires close examination of the arithmetic sum of both the rod load and the reversal function (used as input to the rule). As Figure 8 shows, the buffered values move above and below the severity threshold several times a day. Note the small changes with respect to the scale on the left vertical axis.

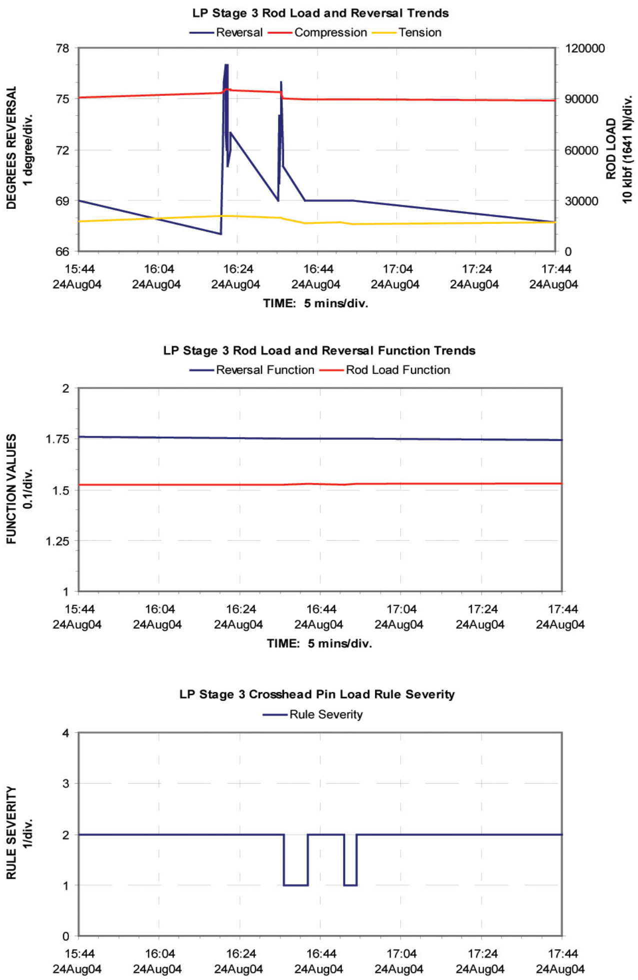

Figure 7. Trend Plots Showing Repetitive Severity. Top Pane Shows Inputs to Rule, Middle Pane Shows Buffered Intermediate Values Within Rule, and Bottom Pane Shows Rule Alert Status.

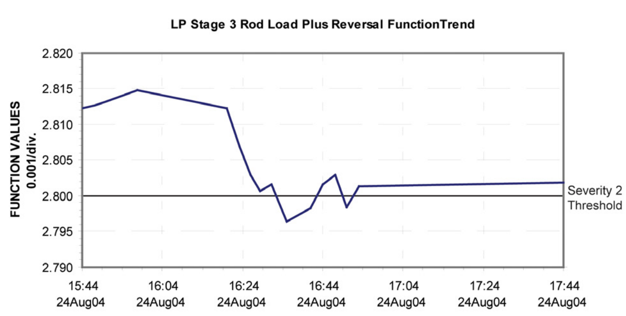

Figure 8. Arithmetic Sum of Rod Load and Rod Reversal Functions.

Clearly the small changes in values, and subsequent rule advisories, do not represent actual changes in machine condition. In order to introduce a hysteretic effect in a software tool, a dedicated operand, shown in Figure 9, needs to be included in the toolset to control the actuation of the alerts. When the "Set" flag is true the hysteresis operand passes the value through after the "Set Duration" time has elapsed. The value remains set until the "Reset Duration" time has elapsed and the flag at the "Reset" input equals true. The operand then remains in the reset condition until the "Set" is true and the set duration time has elapsed.

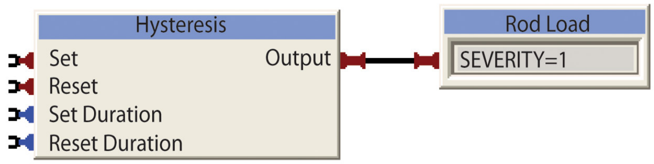

Figure 9. Threshold Operand.

Hysteresis settings for reciprocating compressors have been determined empirically, as shown in Table 3. With these buffer settings and hysteresis values, repetitive alerts have been eliminated.

Table 3. Threshold Settings.

Reset	90% of Set Value
Set Duration	0 seconds
Reset Duration	0 seconds

Suction Valve Failure

In this case history, a collection of rules is designed to assess the condition of the cylinder trim components: valves, piston rings, and packing rings. This collection includes rules to detect:

1. Pressure packing leak

2. Frame loading

3. Crosshead pin loading

4. Leak—cylinder to low pressure

5. Leak—high pressure to cylinder

6. Suction valve leak

7. Discharge valve leak

Likely, with the exception of the fourth and fifth items, most operators and owners of reciprocating compressors understand the malfunctions. The fourth and fifth items refer to distortion of the pressure versus volume (PV) curve caused by the leaks. In the case of the fourth item, a leak from the cylinder chamber to a low-pressure reservoir (such as the suction valve manifold, distance piece, or atmosphere) occurs and the cylinder end in distress can no longer build up pressure fast enough during the compression process and pressure falls as piston velocity slows near top dead center (TDC)/bottom dead center (BDC). The top two graphs in Figure 10 show this case. The fifth item addresses the opposite case in which gas from a high-pressure source leaks back into the cylinder. In this case, the pressure builds up too quickly during the compression process and cannot fall quickly enough as piston velocity slows near TDC/BDC. The bottom two plots in Figure 10 show this case.

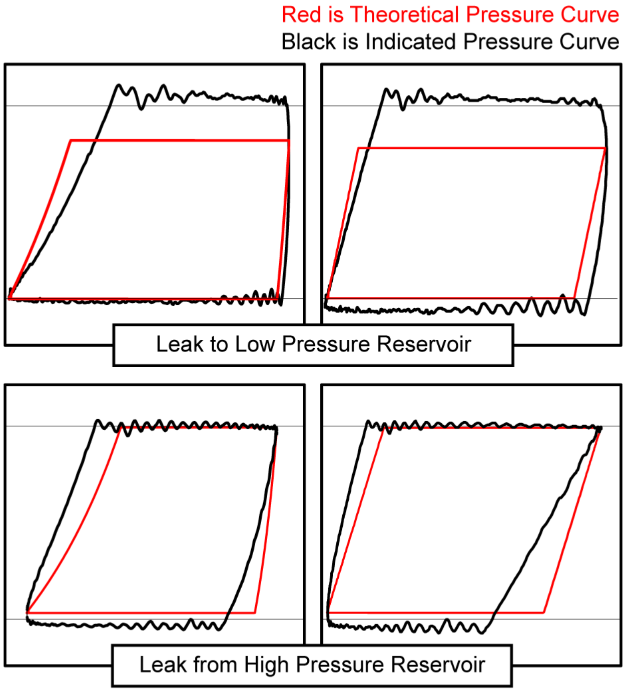

Figure 10. Chamber Leak Effects on Pressure-Volume Curves.

In the case of a leaking suction valve, the indicated discharge pressure drops with respect to line pressure, while, in the case of a leaking discharge valve, the indicated suction pressure rises with respect to line pressure. The leak detection algorithms use these pressure data points from the PV curve to determine if a leak exists on the cylinder and whether the leak originates in a high-pressure or low-pressure reservoir.

In order to pinpoint the component in distress, the rules combine the above cylinder pressure PV information with suction temperature, discharge temperature, and valve cover temperature information.

In this example, a four-throw, balanced-opposed, horizontal reciprocating compressor provides compressed hydrogen service at a large refinery. A head end (HE), suction valve plug-type unloader on each cylinder and a variable pocket clearance unloader on the head end of the first stage provide capacity control. Each compressor provides three stages of hydrogen make-up and one stage of recycle service. Each compression stage consists of one double-acting compressor cylinder. Shell and tube heat exchangers provide interstage cooling between each compression stage. Compressors have been instrumented as per Table 1 plus valve cover acceleration.

The condition-monitoring system, which includes rules, first provided an alert to a suction valve problem on 13:59:47 on 22 February 2008, as shown in Figure 11. Review of waveform data, shown in Figure 12, shows the cylinder pressure curves associated with this event. The plot shows a significant difference between the indicated pressure curve (orange) and the adiabatic pressure curve (green) on the head end. (The adiabatic pressure curve assumes an isentropic thermodynamic process and uses the indicated pressures at TDC and BDC to generate the curve.) The difference appears most pronounced during the compression process, 100 percent to 75 percent of cylinder displacement, when the indicated pressure curve rises slower than the theoretical pressure curve. Looking at the same lines just after TDC, it can be seen that the indicated pressure falls faster than the adiabatic curve. This pattern of difference between the indicated and adiabatic pressure curves signifies a leak from the cylinder to a low-pressure area, such as is found with a suction valve leak.

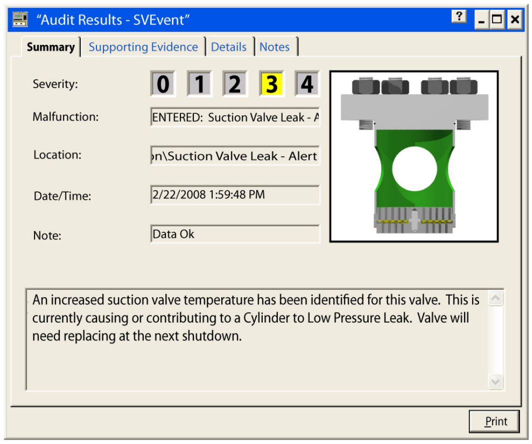

Figure 11. Rule Suction Valve Alert Advisory.

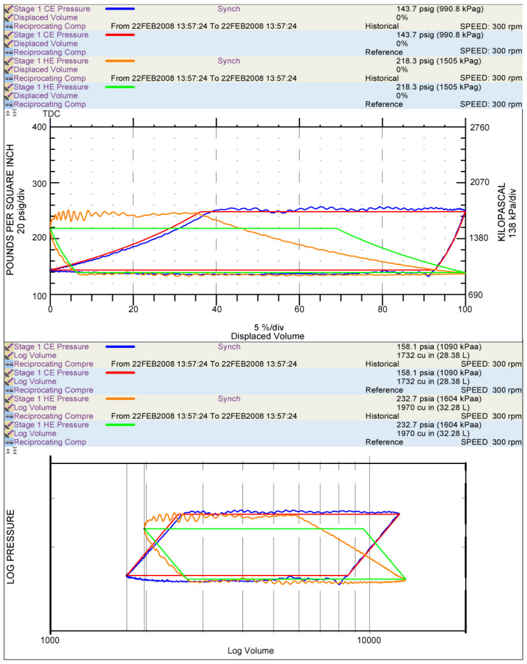

Figure 12. First Stage Cylinder Pressure Versus Volume.

Typically, a valve temperature measurement provides confirming evidence to determine which valve has failed with multivalve cylinders. For the cylinder in question, valve temperature data taken 30 minutes prior to the PV diagram (Figure 13) does indicate that the 1HS1 valve temperature had risen far above the suction temperature: 171°F (77.2°C) valve cover temperature versus 70°F (21.1°C) suction gas temperature. This high temperature difference confirms a leaking valve, as reported by the alert.

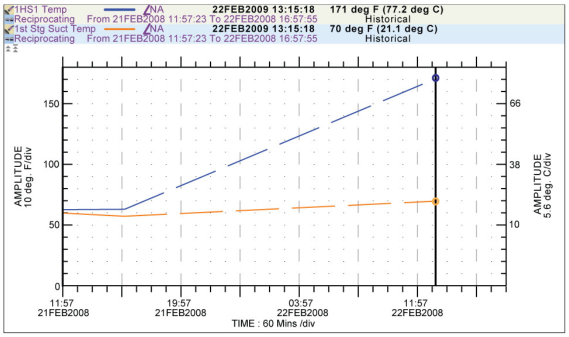

Figure 13. 1HS1 Valve Temperature and Suction Temperature Trend.

Interestingly, the customer's inclusion of a measurement not typically found in online reciprocating compressor condition-monitoring systems (valve cover vibration) provides an opportunity for additional analysis. The vibration from this transducer can be plotted with the cylinder pressure data, as shown in Figure 14.

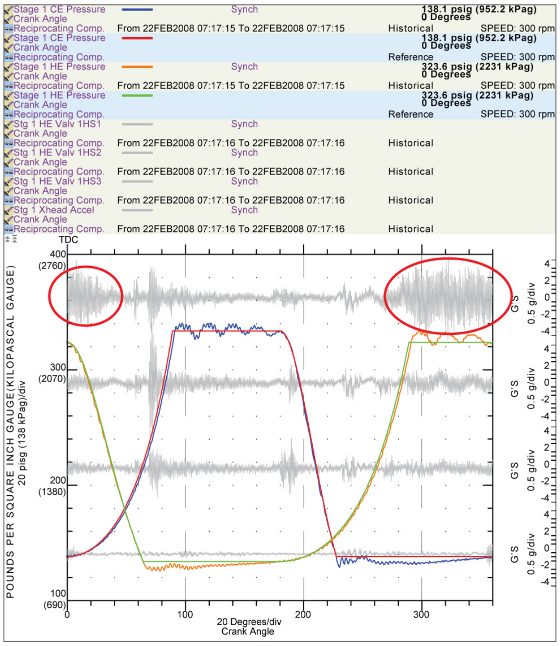

Figure 14. Cylinder Pressure, Head End Suction Valve Acceleration and Crosshead Acceleration, 22 Feb 2008, 7:17:16.

In the areas highlighted by the red ellipses in Figure 14, gas begins to leak past the valve into the cylinder suction manifold. This flow of gas creates an acoustic noise detected by the accelerometer. The red ellipses in Figure 14 highlight this noise

The good agreement between the theoretical and indicated pressure curves in Figure 14 illustrates that this incipient leak has not yet significantly affected the cylinder performance and would not be detected using a PV diagnostic tool.

The presence of these signatures at 7:16 in the morning of 22 February 2008—nearly seven hours prior to the suction valve alert—suggests valve cover vibration may provide the earliest indication of valve distress.

The valve condition continued to deteriorate until the plant took the unit out of service on 27 February.

Machine outage and overhaul commenced on 27 February 2008. Referring to Figure 15, the 1HS1 suction valve experienced both concentric ring and valve seat failure. The plant replaced the valve assembly and returned the unit to service.

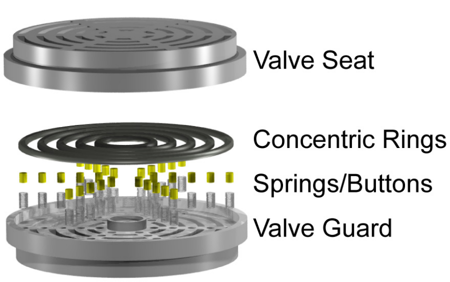

Figure 15. Typical Suction Valve Assembly.

As Figure 16 shows, the good agreement between the theoretical and indicated pressure curves and low amplitude valve cover accelerometer signals confirm the new valve to be in good health and operating correctly.

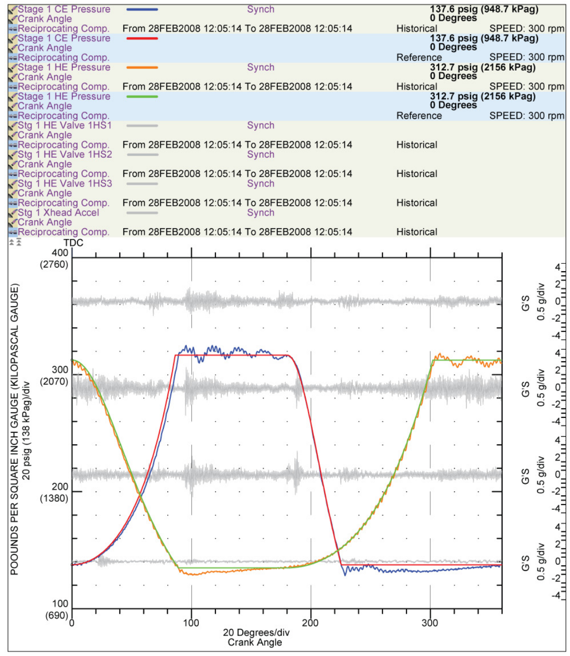

Figure 16. Stage 1 Cylinder Pressure, Suction Valve Cover, and Unfiltered Crosshead Accelerometer Signal, 28 Feb 2008, 12:05:14.

Capacity and Process Effects on Combined Rod Loading and Reversal

In this example a six-throw, balanced-opposed horizontal reciprocating compressor provides make-up hydrogen service at a large refinery. An HE suction valve plug-type unloader on each cylinder and a variable pocket clearance unloader on the head end of the first stage provide capacity control. Each compressor has two distinct services, based on process stream requirements; each of these services has three compression stages; and each compression stage consists of one double-acting compressor cylinder. Shell and tube heat exchangers provide interstage cooling between each compression stage. The compressor is instrumented as per Table 1.

Shortly after the commissioning of the collection of rules, the plant engineer observed that one of the throws indicated an alert. Opening the event viewer, the plant engineer discovered that an alert had been issued (Figure 17).

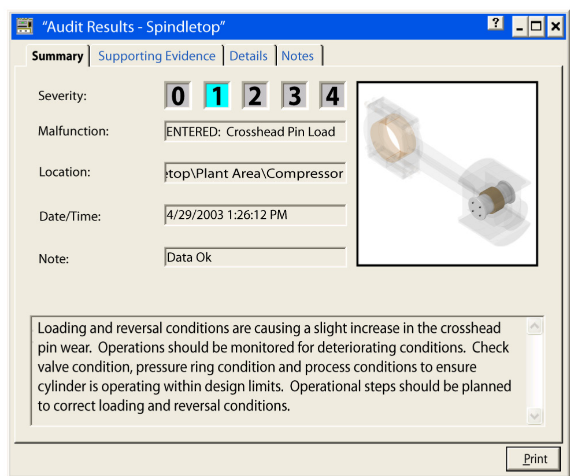

Figure 17. Crosshead Pin Loading Condition Alert.

As the rules had been recently installed and not completely commissioned, the plant engineer initially believed the alert to be false. In addition, plant operations had begun the process of shutting down the compressor, which lowers the pressure across the compressor. It seemed unlikely that a reduction in pressure would have caused loading problems on a crosshead pin.

Evaluation of the rule configuration confirmed correct configuration. The supporting evidence indicated that a change in rod load and reversal had occurred and driven the alarm. To confirm that the data consumed by the rule accurately reflected machine condition, the rod load curves were reviewed. (Refer to APPENDIX A for definition of terms and how the condition-monitoring system generates these curves.) Figure 18 shows these curves. At first glance, the two curves appear quite similar; however, closer inspection revealed that a critical change in forces at the crosshead pin had occurred.

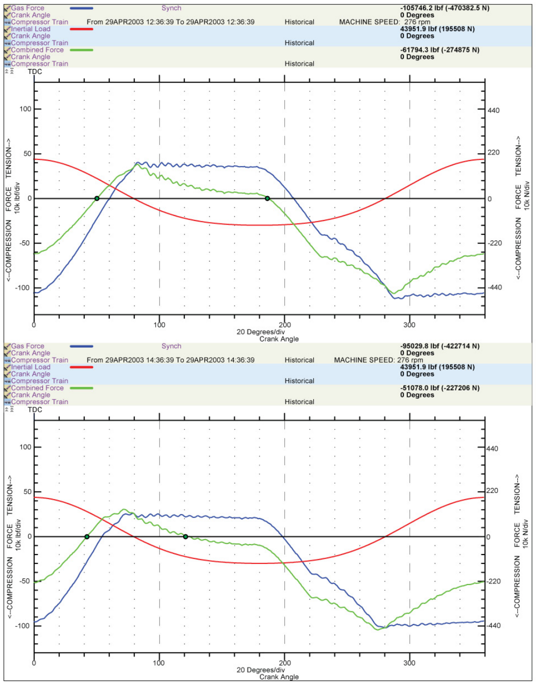

Figure 18. Rod Load Curves Before (Top) and After (Bottom) Alert.

Figure 19 shows a crosshead assembly and nomenclature. The crosshead pin connects the connecting rod to the crosshead. The crosshead pin does not turn by any large amount. In order to achieve complete lubrication, the crosshead pin forces must alternate so that the pin moves from one side of the bushing to the other. If the forces at the crosshead pin do not alternate sufficiently, metal-to-metal contact occurs between the pin and the bushing. Unchecked, the wear leads to excessive clearance and high-amplitude shock events to the connecting rod, which can cause the connecting rod to fail.

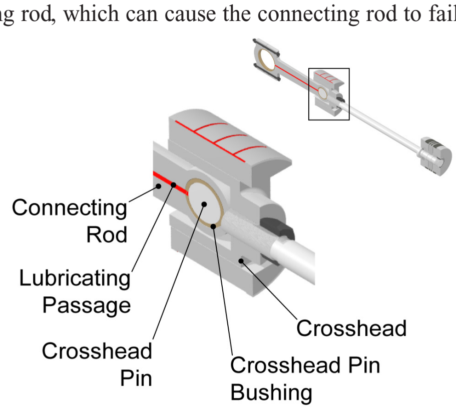

Figure 19. Typical Direct Rod Connection Crosshead Assembly.

The plots in Figure 18 revealed that the alert had correctly identified a shift in the rod load forces at the crosshead pin. The blue lines represent the force exerted by the gas acting on the piston. The red line represents the inertia load imposed upon the pin by the reciprocating motion of the crosshead assembly, crosshead nut, piston rod, and piston assembly. Summing the inertia and gas forces results in the net force experienced by the crosshead pin. The green line in Figure 18 shows this combined rod load force.

In each plot, the combined rod load crosses the neutral axis at two points, shown by black dots. The minimum distance between crossings, referred to as degrees of reversal, serves as a good indicator of lubrication between the pin and crosshead bushing. As the degrees of reversal decrease, the pin has less time to flush out old lubricant and allow new lubricant to flow in.

In addition to degrees of reversal, the ratio of maximum tensile load to maximum compressive load also impacts the lubrication condition at the crosshead pin. Ideally, the magnitude of the compressive and tensile forces would be equal. The more unequal the forces, the more difficulty the pin has in achieving good lubrication.

As can be seen in Figure 18, both degrees of reversal and symmetry in tensile/compressive forces decreased. The rule alert had correctly identified a problem that could have led to substantial machine damage.

As a result of this analysis, the plant technician consulted with operations to review the process for machine shutdown. It was discovered that prior to lowering discharge pressure, the shutdown procedure decreased the head end clearance volume of the first stage cylinder.

Figure 20 shows the cylinder pressure curves before and after the volume pocket change on the first stage cylinder. Prior to the volume change, the third stage cylinder had the highest compression ratio. The change in clearance volume on the first stage cylinder caused this cylinder to operate with a higher compression ratio, but at the expense of the compression ratio on the third stage.

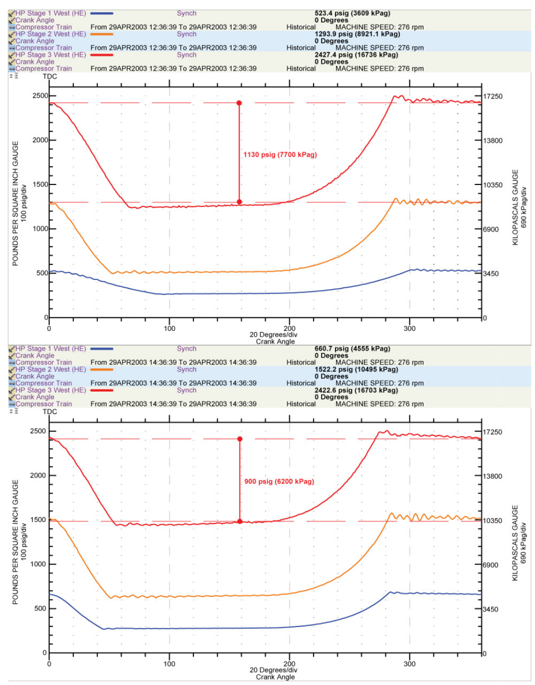

Figure 20. First Stage, Second Stage, and Third Stage Cylinder Pressure Curves Before (Top) and After (Bottom) Change in First Stage Head End Volume Clearance.

As the third stage is a high-pressure cylinder, the rod diameter is not so different from the diameter of the piston. The resulting inequality in crank-end and head-end piston area makes the cylinder sensitive to changes in load. Contrary to "common sense" thinking, an increase in the compression ratio of the third stage cylinder in fact results in improved degrees of reversal and symmetry of force magnitudes. Lower compression ratio, as in this instance, actually reduces degrees of reversal and symmetry of force magnitudes.

As a follow-up action based on the advisory, the plant altered shutdown and operational procedures to avoid loading the first stage cylinder.

In this case, the alert provided a timely notification that allowed the plant to both update the operating instructions and to continue monitoring to ensure that the machine remained below design conditions.

CONCLUSION

The previous case history illustrates the importance of getting timely, meaningful alerts to the right people. Designing and deploying knowledge-based tools has challenges. Yet, organizations that successfully deploy and manage knowledge-based tools to leverage information provided by condition-monitoring systems recognize advantages in their plant operations.

APPENDIX A— ROD LOAD TERMINOLOGY AND CALCULATION

Introduction

The term “rod load” has been used for decades to describe the maximum forces a reciprocating compressor assembly can withstand. The term has some ambiguity, but recent papers have clarified some of the key definitions and terms.

With the improved reciprocating compressor condition monitoring understanding beginning to permeate the industry, customers have begun to ask questions about how condition-monitoring systems calculate these values. The purpose of this Appendix is to answer those questions.

The terminology and context for this application note are American Petroleum Institute (API) Standard 618, Fourth Edition and Fifth Edition (2007). Previous editions of API 618 used other descriptions and definitions for rating values of reciprocating compressors. For ease of reference, the terms and definitions used herein appear at the end of this application note.

Calculation Methodology

Figure A-1 shows the rod load curves and data generated by a reciprocating compressor condition-monitoring system.

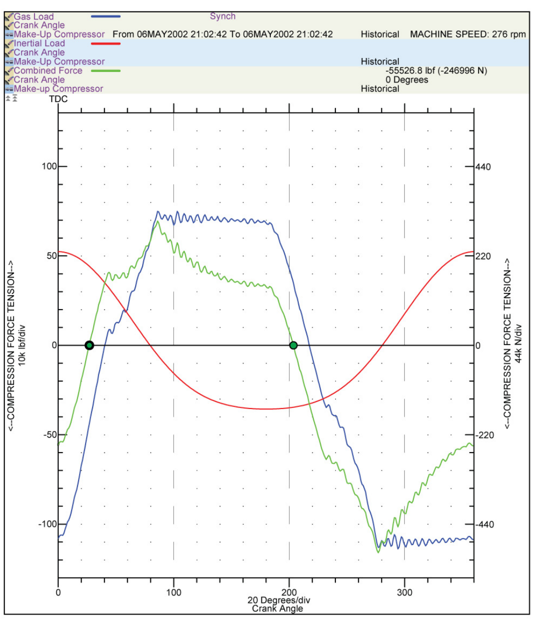

Figure A-1. Rod Load Curves Generated by Condition-Monitoring System.

Inertial Force

The red line in Figure 21 represents the forces due to inertia. The inertia mass for nearly all reciprocating compressor condition-monitoring installations includes the crosshead assembly, crosshead nut, piston rod, and piston assembly. This collection of mass for inertia is consistent with the definition provided in API-618 Fifth Edition (Data Sheets, Page 7, line 31). The condition-monitoring system does allow the user to exclude the crosshead mass from the inertia force calculations (Figure A-4); however, the configuration is rarely encountered in the refining/petrochemical segments and is not consistent with API-618 Fourth or Fifth edition terminology. For these reasons, this definition falls outside the scope of this application note.

Gas Force

The blue line in Figure A-1 represents the gas forces acting on the compressor’s static components and running gear. This force is the gas load referenced in API-618 (paragraph 6.6.2). To obtain this force, the indicated cylinder pressure on the head end is multiplied by the head-end area of the piston. This is shown in green in Figure 22 for a typical double-acting cylinder. The resultant force is then subtracted from the indicated crank-end cylinder pressure times the crank-end piston area (shown in brown in Figure A-2).

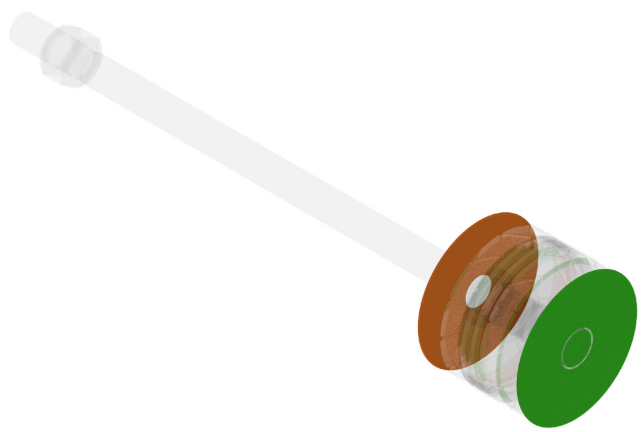

Figure A-2. Piston Areas Used for Gas Rod Load Calculation.

This summation represents the net force (sign convention for forces in rod load show compressive forces as negatives and tensile forces as positive) acting on the piston rod and can be written as:

$$F_{GasLoad} = \left(A_{CrankEnd} \times P_{CrankEnd}\right) - \left(A_{HeadEnd} \times P_{HeadEnd}\right) \quad \text{(A-1)}$$

The cylinder pressure varies continuously throughout the revolution of the crankshaft so the calculations must be performed multiple times throughout the revolution to obtain a curve. For each 360 degrees of crankshaft rotation, a condition-monitoring system collects 720 indicated cylinder pressure data points simultaneously for both the head and crank end. The gas load calculation is thus performed 720 times for each revolution.

The gas pressures in the chamber act not only on the piston, but also on the heads of each cylinder. The combined gas force on the crank- and head-end heads has the same absolute value as the gas load calculated above, but acts in the opposite direction (i.e., has the opposite sign) (Figure A-3).

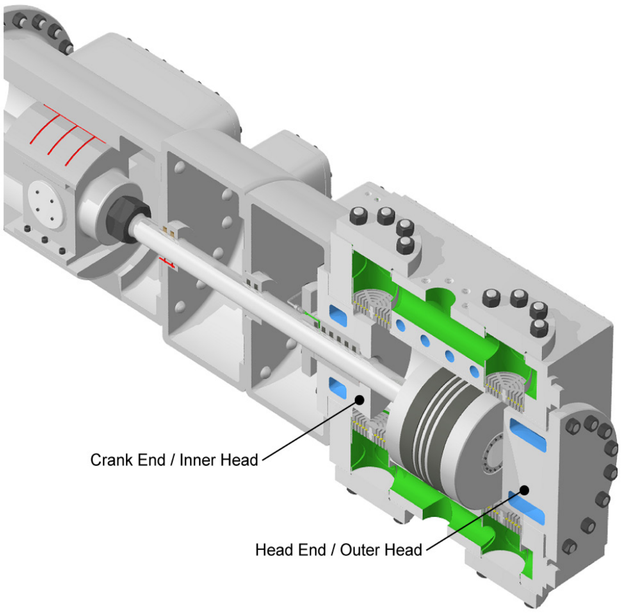

Figure A-3. Reciprocating Compressor.

Combined Rod Load

Finally, the green line in Figure A-1 represents the combined rod load, or crosshead pin load. This force is the combined rod load referenced in API-618 (paragraph 6.6.1). The gas load is added to the inertia force at each point of measurement to obtain this force. Since, as noted in the previous section, the gas load is computed 720 times for each crankshaft revolution, the condition-monitoring system also performs the combined rod load calculation 720 times for each crankshaft revolution. For each of these 720 points of measurement during the crankshaft revolution, the calculation can be written as:

$$F_{CombinedRodLoad} = F_{GasLoad} + F_{Inertia} \tag{A-2}$$

When the mass used in the inertia force calculation includes the crosshead, the smallest distance between the points of zero force (shown by black dots at approximately 35 degrees and 200 degrees of crank angle in Figure A-1) represents the degrees of rod reversal referenced in API-618 (paragraph 6.6.4).

Figure A-4 shows a typical configuration screen that allows users to select rod load calculations at either the crosshead pin or piston rod. Had the user configured the condition-monitoring system to calculate rod load at the piston rod, the inertia forces would no longer include the crosshead mass. The combination of this inertia force and gas force results in the forces that act on the piston rod, next to the crosshead. Note that this force no longer reflects those acting on the crosshead pin, and therefore cannot be used to calculate rod reversal.

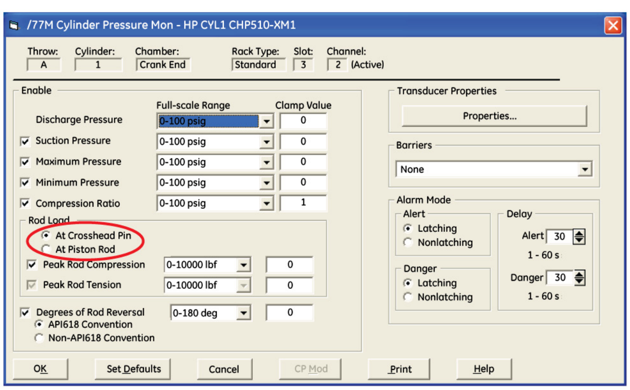

Figure A-4. Configuration Screen.

Terms and Definitions

The following definitions draw extensively from Atkins, et al. (2005). The reader is encouraged to consult this reference for a complete discussion of these terms and others, as well as a valuable historical perspective.

- **Combined rod load**—*The sum of actual gas load (including valve and passage losses) plus inertia loads at the crosshead pin, in the direction of the piston rod.*

This is the force curve labeled "Combined Force" in the condition-monitoring software's rod load plots when the inertia force includes the mass of the crosshead assembly. This force varies continuously throughout the revolution. The term appeared in both the Fourth and Fifth Edition of API-618 with the same definition; however the Fourth Edition required that it be calculated every 10 degrees and the Fifth Edition required that it be calculated every 5 degrees. Reference 3.7 of API-618 Fifth Edition for a formal definition.

- **Crosshead pin load**—*Same definition as "combined rod load."*

Note this term is not defined within API-618, but is a commonly encountered industry term used to clarify the component at which the combined rod load calculations are being done.

- **Gas load**—*The force resulting from the internal pressure in each chamber acting on the associated piston- and cylinder-head surfaces.*

This is the curve labeled "Gas Force" in condition-monitoring software plots. As it depends only on the pressure and cylinder geometry, it remains the same whether the force calculation is done at the crosshead pin or piston rod. The term appeared in both the Fourth and Fifth Editions of API-618 with the same definition; however, the Fourth Edition required that it be calculated every 10 degrees and the Fifth Edition required that it be calculated every 5 degrees.

- **Maximum allowable continuous combined rod load (MACCRL)**—*A value determined by the original equipment manufacturer (OEM) based on design limits of the various components in the compressor frame and the running gear (bearings, crankshaft, connecting rod, crosshead assembly, piston rod, piston assembly).*

With very minor exceptions (refer to 6.6.5 of API-618, Fifth Edition), no single value of combined rod load can exceed the manufacturer's ratings for maximum allowable continuous combined rod load (refer to paragraph 3.19 of API-618 Fifth Edition for a formal definition). OEMs may have individual limits for compressive rod load, tension rod load, and compressive plus tension.

- **Maximum allowable continuous gas load (MACGL)**—*A value determined by the OEM based on the design limits of the static components (frame, distance piece, cylinder and bolting).*

With very minor exceptions (refer to 6.6.5 of API-618, Fifth Edition), no single value in the gas load curve can exceed the manufacturer's ratings for maximum allowable continuous gas load. Reference 3.20 of API-618 Fifth Edition for a formal definition.

- **Rod reversal**—*The shortest distance, measured in degrees of crank revolution, between each change in sign of force in the combined rod-loading curve.*

Reference paragraph 3.49 of API-618 Fifth Edition for a formal definition; however, note that the API 618 definition is not entirely accurate as it references "piston rod loading" instead of "combined rod loading."

APPENDIX B— SELECT RECIPROCATING COMPRESSOR NOMENCLATURE

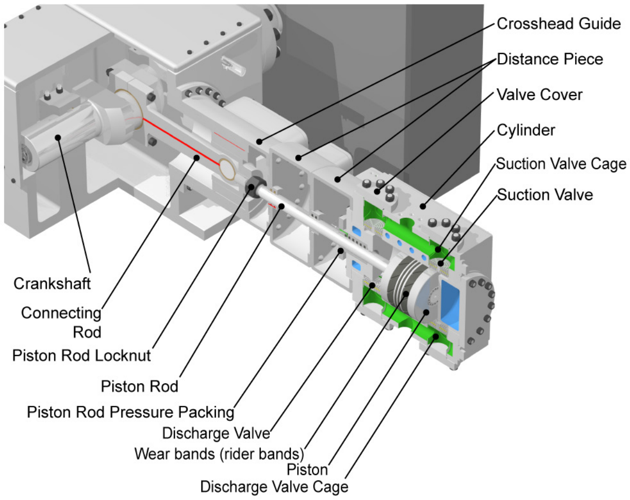

Figure B-1. Select Reciprocating Compressor Nomenclature.

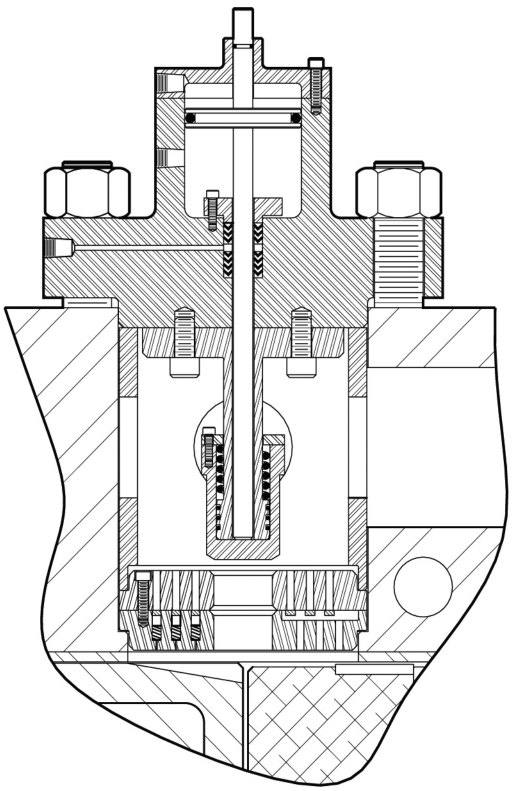

Figure B-2. Plug Type Loader Assembly.

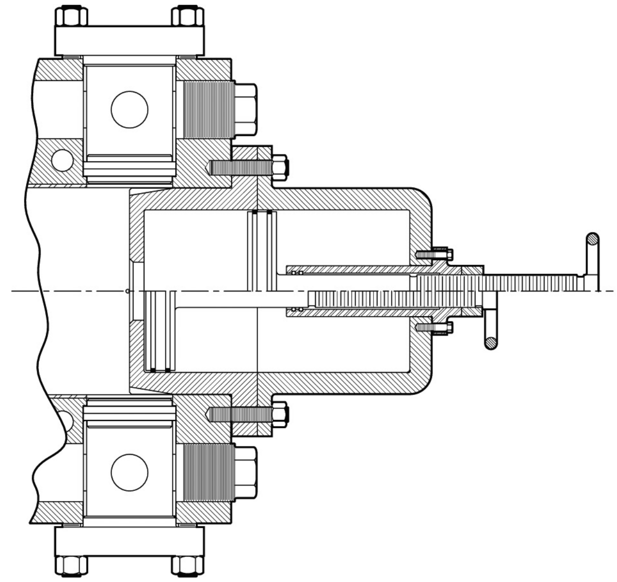

Figure B-3. Variable Clearance Pocket Assembly.

REFERENCES

Atkins, K. E., Hinchliff, M., and McCain, B., 2005, "A Discussion of the Various Loads Used to Rate Reciprocating Compressors," Proceedings of the Gas Machinery Conference.

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