orbit

Volume 30 | Number 1 | 2010

A Technical Publication for Advancing the Practice of Operating Asset Condition Monitoring, Diagnostics, and Performance Optimization

From the Desk of **Don Marshall**

Chief Marketing Officer, Bently Nevada Asset Condition Monitoring

Orbit: Looking Forward with the Experience of the Past

2010 has proven to be a year of change. The economic crisis of 2009 was a difficult time for all of us and this year began with a lot of uncertainty. However, as we near the end of 2010, I think we can all agree that our environment feels a bit more stable and we all have much to look forward to.

Here at Bently Nevada, a cornerstone of our strategy is to "protect the core." This "core" represents all of the things you have come to expect from Bently Nevada. The deep domain knowledge you expect from our sales team, the superior expertise of our services team, the service you receive from our local offices, the fast turn-around you expect from our manufacturing team and the overall quality you expect from our products. These are all important aspects of our legacy and our future.

Another strategic aspect of our legacy that we will not forget is that of the Orbit magazine. In this issue, we will be highlighting a cornerstone of our legacy: the need for using the right transducer suite to make the right measurement to solve the right problems. This "back to basics" message is found in the use of XY proximity probes for measuring vibration AND position. We will reinforce this message via three types of machines highlighted in this edition of Orbit: horizontally mounted machinery, (turbines and compressors), vertically mounted machinery (hydro turbine-generators) and reciprocating machinery.

Bently Nevada has been a pioneer in the use of proximity probes for industry-accepted measurements on these machines, accommodating diagnostic plots that include the orbit, shaft centerline and rod position. We will continue this legacy into the future as we apply solid engineering concepts to facilitate condition monitoring for an ever-increasing range of machines that cause problems for our customers.

As we look to the future of the Orbit magazine, I am pleased to announce that we will be bringing a new look and feel to Orbit in 2011. Don't worry; our technical content will only be enhanced by our continuing desire to make this magazine the benchmark of the industry. To meet this goal, we will be bringing a new editor on-board for 2011. This will be combined with the goal of increased frequency of publication and an on-going commitment for enhancing our technical leadership. Bently Nevada is the world's leader in condition monitoring products and services and the Orbit will continue to be one of our primary educational and informational tools. 2011 will truly be an exciting year.

In this **Issue**

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Are XY Radial Proximity Probes Really Necessary?

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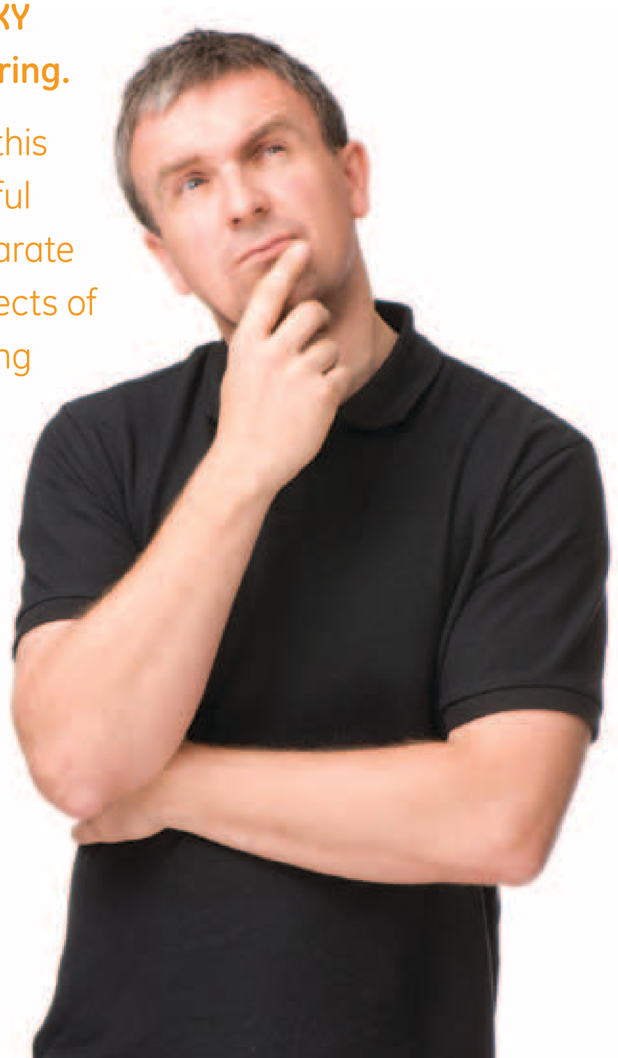
The title of this article is a very common inquiry from customers considering retrofits on older machines that previously had minimal (or no) vibration instrumentation. **For critical machines with fluid film bearings, the short answer is yes. You really do need XY probes at each bearing.**



To understand why this is the case, it is helpful to examine two separate but interrelated aspects of a vibration monitoring system's purpose: machinery protection and machinery management. It is also helpful to understand the nature of radial vibration inside a machine's bearing clearances.

Machinery Protection

An extremely important aspect of a machinery vibration monitoring system, and indeed often the primary purpose by which it is economically justified, is to prevent mechanical problems from progressing to catastrophic failures without prior warning – in other words, to save the machine, the process, and a significant amount of downtime. To this end, a vibration monitoring system is designed to provide operator annunciation alarms (i.e., ALERT alarms) when vibration levels exceed user-configured setpoints, and to shut the machine down automatically when second-level alarm limits (i.e., DANGER alarms) are reached.

As shown in Figure 1, a shaft within a horizonal rotor system will vibrate within its bearing clearances in a characteristic pattern that is typically elliptical in nature rather than perfectly circular. The ellipticity of this orbit is a function of numerous parameters in the machine, such as stiffnesses, that are different in one axis than another (a machine is typically stiffer in the vertical direction than in the horizontal) and preloads such as gravity which tend to keep the rotor in the bottom half of the bearing . However, stiffness and preloads are only two such parameters. Machinery alignment conditions, balance conditions, lubricant conditions, rotative speed, rotor critical speeds, and other rotordynamic aspects will all affect the orbit shape under both normal and malfunction conditions.

As shown in figure 2, when XY probes are present, the full 2-dimensional motion of the shaft can be ascertained as well as the maximum excursion of the shaft. This is because any 2-dimensional shape can be uniquely identified by the path its x- and y- coordinates trace out. A simple analogy, is that the orbit path is what you would see if a pencil lead at the shaft's centerline traced the shaft's path on a piece of paper (amplified of course).

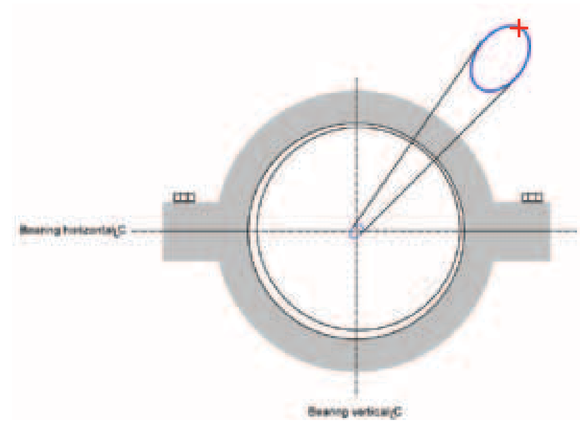


Figure 1: The path traced out by the shaft centerline as it vibrates about the bearing clearance centerline is known as the shaft orbit. At the instant depicted, the shaft centerline is in the position shown by the red cross-hairs.

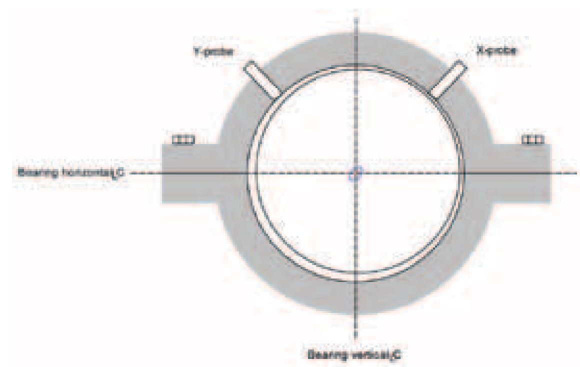


Figure 2: With an orthogonal probe pair, the shape of the orbit can be ascertained, rather than just the amplitude observed by one probe or the other. In this example, the major axis of the elliptical orbit aligns primarily with the X probe; consequently, it will observe approximately twice as much vibration amplitude as will the Y-probe.

DEFINITION:

Radial proximity probes in an orthogonal 'XY' pair are mutually perpendicular. That is, their angular orientation is 90 degrees apart, as shown in Figure 2.

In contrast, a single radial transducer is only able to detect vibratory motion in a single axis. For example, if only the X transducer were installed in Figure 3, it would be unable to differentiate between the orbits of Figures 3A, 3B and 3C. The orbit in Figure 3A would normally not indicate a machinery malfunction. However, the orbits in Figures 3B and 3C have very large amplitudes in the 'cross-axis' direction that would not be detected by the X transducer. It is clear that if we had only the X transducer installed, machinery protection and alarming would be seriously compromised, and malfunction analysis would be nearly impossible.

While one might be inclined to dismiss such scenarios as "infrequent" or "unlikely" this is not necessarily the case. By their nature, machinery malfunctions are unpredictable. For example, following the turnaround of a large steam turbine, the bearing might be assembled or shimmed incorrectly; or, the machine might be aligned improperly; or, thermal expansion conditions might cause the machine to assume incorrect alignment. All of these are examples of conditions that can lead to extreme preload on a shaft, constraining its vibration to primarily a single plane, and resulting in a highly planar orbit shape. Further, it is impossible to predict in advance the direction in which this preload might act. A case in point occurred where a customer installed a balance correction weight on their 650 MW steam turbine – based on vibration measurements that were made using only a single X probe on each bearing. We recommended against this action without the addition of Y probes to prove that the root problem was in fact balance related. We suspected that balance was not the root cause because we knew that in the winter, the vibration levels would change significantly if a loading bay door was opened (indicative of a thermal alignment sensitivity). Orbit plots, and shaft centerline data would have provided us with the confidence to proceed, or recommend an alignment change.

Rubs are another very common machinery problem that can result in not only highly planar orbit shapes, but also unusual orbits that exhibit either exterior loop (so-called "Figure 8") patterns or interior loop patterns. This occurs when the vibration frequency in one axis is markedly different from the frequency in another axis. Depending on the probe orientations, one probe may see a single amplitude peak for each shaft revolution, while the orthogonal probe may see two or more peaks per shaft revolution.

Figure 4 shows an example of a classic "figure 8" orbit shape incurred during a rub. Notice that one probe sees only one-fourth as much vibration amplitude as the other probe, even though something abnormal is clearly occurring in the machine. For a conventional vibration monitoring system that monitors only the amplitude from a probe mounted in a single plane, the orbit in Figure 4 would not generate any kind of machinery alarm, assuming that the amplitude alarm for the X probe was 3 mils (a typical alert level for large 3600 rpm steam turbines) and only the X probe was installed in the machine.

As has been shown, only when orthogonal probes are present at each radial bearing can the true amplitude of the vibration be ascertained, regardless of the orbit shape or orientation. And only when this amplitude can be ascertained can adequate machinery protection be ensured.

The most prominent industry standard pertaining to vibration monitoring systems for machinery protection applications is American Petroleum Institute standard API 670. The usefulness and applicability of this standard transcends the petrochemical industries and has been employed across many industry's as part of their purchasing specifications for vibration instrumentation. However, for some machines, such as hydro-generators (highlighted in this issue of *Orbit*), this standard may not be applicable. To understand the standard's wide appeal with end users, it is helpful to consider that nearly every paragraph in the standard is the result of an end users experience with vibration monitoring systems, leading to "best practice" recommendations that were often acquired through "doing it the wrong way" and then embedding in the standard a series of guidelines that would prevent other users from making similar mistakes.

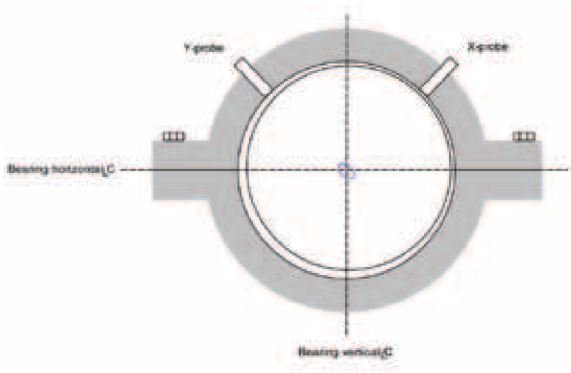


Figure 3A: This orbit is fairly normal and is slightly elliptical. The Y-probe observes approximately twice as much vibration as the X-probe.

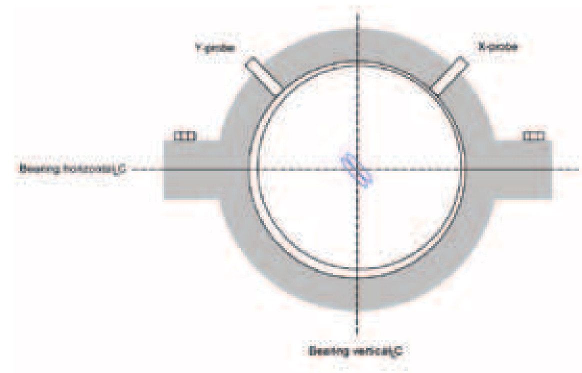


Figure 3C: This orbit again has the same amplitude and frequency (1X) as the orbits of Figures 3A and 3B. However, something is clearly different as shown by the "figure 8" pattern of the orbit. With only an X-probe installed, this change in condition would be missed. Indeed, the monitoring system would detect no change at all between Figures 3A, 3B, and 3C. It is only when a Y-probe is installed that such important changes can be detected.

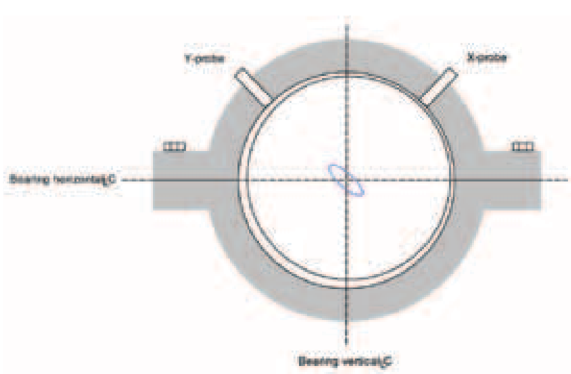


Figure 3B: This orbit has the same amplitude in the X-axis as the orbit of figure 3A, while the amplitude in the Y-axis has approximately doubled. If only an X-probe were installed, it would indicate no change in vibration from the orbit of Figure 3A, and would be oblivious to the fact that amplitude in the Y-direction was nearly 3 times as large as that in the X-direction.

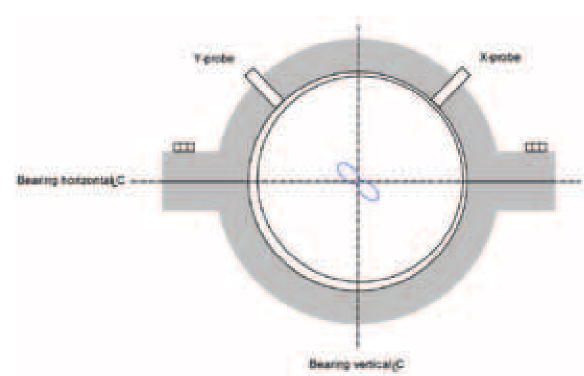


Figure 4: This orbit shows approximately 4 times as much vibration in the X-axis as in the Y-axis. If only amplitude from the X-probe was observed, as would be typical from a radial vibration monitor with only a single probe, such a machinery condition would be missed. Indeed, the X-probe would observe exactly the same vibration amplitude as in Figures 3A, 3B, and 3C.

Central to this is the mandate within API 670 for XY probes at each radial bearing. The petroleum industry are extremely machinery-intensive and users of this standard have learned the hard way that a single probe is "penny wise, but pound foolish" when a machinery failure goes unnoticed and an expensive outage and/or machinery repair results.

Another benefit of XY probe pairs is that if a transducer should fail, the bearing will still have *some* vibration monitoring capabilities, although they will be severely limited. For critical applications where probe access is difficult or impossible during machine operation, it is often beneficial to install redundant spare probes. It is a simple task to disconnect a failed transducer and connect the installed spare to regain full protection and monitoring capability without interrupting plant operation.

Machinery Management

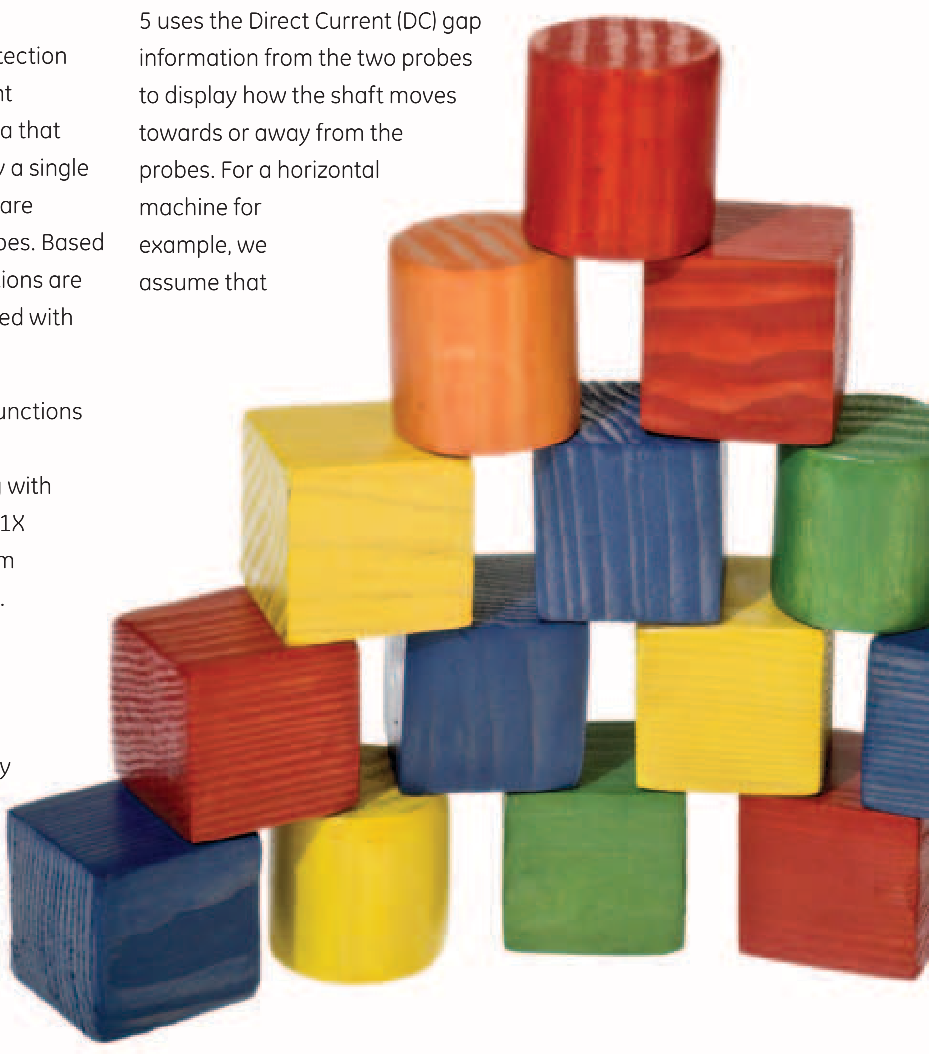
We discussed the implications of a single probe versus XY orthogonal probes for machinery protection considerations above. Another equally important consideration is a comparison of the type of data that can be obtained from XY probe pairs versus only a single probe. As Machinery Diagnostics Engineers, we are passionate about the need for both X and Y probes. Based on our experience, even very common malfunctions are difficult to diagnose when machines are equipped with only a single probe per bearing.

Field experience shows us that 80% of the malfunctions that machine's experience are unbalance and misalignment. Both of these malfunctions along with common rubs, and shaft crack initiation exhibit 1X vibration characteristics. The differentiators from a diagnostics viewpoint is the shape of the orbit. Single probes are not capable of displaying orbital information. It takes two orthogonal X and Y probes.

For many users, the ability to perform machinery diagnostics while the machine is in service is becoming increasingly important. With many plants incurring downtime costs of $100,000 an hour or more, even a single hour saved by the ability to perform better machinery diagnostics will more than offset the costs of XY probes compared to X-only probes at each bearing.

Consider again the orbit of Figure 4 exhibiting the so-called "figure 8" shape. Depending on where a single transducer was oriented, it may see two vibration peaks per shaft revolution (2X vibration) or one vibration peak per shaft revolution (1X vibration). The malfunctions that exhibit high 1X vibration versus 2X vibration are quite different, and the ability to differentiate such conditions can easily be the difference between a relatively benign change in machinery condition such as imbalance and a serious problem requiring immediate intervention such as a shaft crack, severe misalignment, or a heavy rub.

Although not discussed earlier, another significant diagnostic tool that XY probes provide is the Shaft-Centerline plot. Figure 5 uses the Direct Current (DC) gap information from the two probes to display how the shaft moves towards or away from the probes. For a horizontal machine for example, we assume that



when the rotor is at rest, it is sitting on the bottom of the bearing. As the rotor starts to turn, an oil film is developed between the journal and the bearing. This oil film lifts the rotor off of the bottom of the bearing, and as speed increases, continues to lift it up further. The diagnostician will look to see where in the bearing clearance the rotor is riding. Typically, for a clockwise rotating machine, the rotor will rise and move towards the left. It should not rise above the centerline of the bearing clearance. For a rotor with tilting pad bearings, it is not uncommon to have the rotor rise almost to the center of the bearing, but for most rotors that are aligned well, the rotor will remain in the bottom half of the bearing (remember gravity?). While shaft centerlines are rarely used as an absolute arbiter of machinery condition, they are relied upon regularly to substantiate a diagnosticians conclusion. The shaft centerline exhibited here is not atypical, and represents a startup to 3600 rpm for a large steam turbine rotating in the clockwise direction.

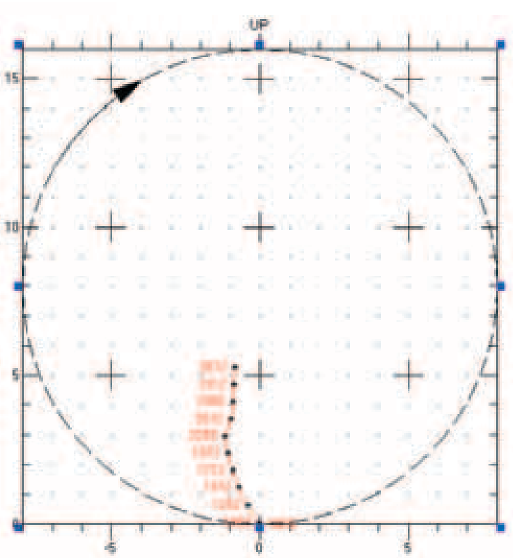


Figure 5: Typical Shaft Centerline Plot.

As was previously mentioned, machinery malfunctions are by their nature, unpredictable. Once they manifest themselves, the ability to conduct adequate diagnostics almost always proves to be invaluable and often will pay for the entire machinery monitoring system in just a single "event." When only a single radial probe is installed, diagnostic capabilities are greatly reduced. In most cases, the diagnostician will find it necessary to install temporary probes in the other axis in order to isolate the malfunction, wasting valuable time and incurring extra costs that will often exceed the costs of simply installing permanent XY probes in the first place.

Conclusion

The issue of installing XY probe pairs versus a single probe at each radial bearing in rotating machinery is primarily an economic decision. While it may be appealing to instrument the machine with only a single probe at each bearing when only initial costs are considered, the resulting system will essentially be unable to perform adequately from both a machinery protection and machinery diagnostics perspective. It should also be noted that in our experience, when all installation costs are accounted for, adding a second "Y" probe is usually only a small incremental cost, certainly not one that costs "twice as much." This cost differential will be eclipsed at the first instance in which a machinery problem is incurred or in which a comprehensive picture of machinery behavior is required. Consequently, most vibration monitoring system suppliers – including GE – recommend following API 670 guidelines to include XY probe pairs for shaft-relative measurements.

Synopsis of Savings

System 1* Software Used to Identify Problems in Reciprocating Pumps

A large refinery uses System 1 software to consolidate more than 1900 points of process and vibration data into a single condition monitoring 'dashboard' for their operators and engineers. In this instance, more than 90% of the monitored points in System 1 come from the plant's Distributed Control System (DCS) and monitoring hardware other than Bently Nevada* vibration systems, illustrating the diversity of the software for consolidating data from a variety of sources. Machinery monitored by the system includes five reciprocating pumps, used for wash water. During startup of the pumps as part of a new unit in the plant, a loud and unusual noise was heard coming from cylinder #3 on one of the pumps. The customer immediately wanted to shut down the pump and inspect cylinder #3. However, System 1 data indicated that the problem was actually in cylinder #2. The customer questioned the integrity of the monitoring system performed a number of loop checks to ensure the instrumentation was indicating problems in the correct cylinder. After satisfying themselves that the system was configured properly and no points were cross-wired, they elected to inspect both cylinders #2 and #3. The problem was found in cylinder #2, validating the integrity of the System 1 data, and they found that the noise was simply resonating to cylinder #3. Had the system not been in place, only cylinder #3 would have been inspected, and the problem would have persisted, potentially causing significant damage and necessitating additional shutdowns (possibly several) and attendant lost production.

System Identifies Rub, Saves $120,000

A large power plant monitors its critical equipment using Bently Nevada machinery protection systems and online condition monitoring software. High vibration readings on the plant's #1 steam turbine generator were originally thought to be a rotor bow. However, the system allowed plant engineers to identify that the problem was actually a rub. The problem was corrected and the plant estimates savings in excess of $120,000 for this single event alone.

Online Monitoring Provides Huge ROI for South American Mine

A South American mining operation replaced portions of their condition monitoring program relying upon portable data collection with a Bently Nevada Trendmaster* System. The system eliminates the need for manual data collection by using automated, online data collection. In the first six months, the savings provided were more than 3X the cost of the system, including installation.

Condition Monitoring Helps Improve Mean Time Between Failures (MTBF) on Critical Centrifuges by 25X

A chemical plant uses Bently Nevada continuous machinery protection systems and condition monitoring software to monitor numerous critical compressor trains and centrifuges. Previous to installation of these systems in one of the process units, MTBF for the centrifuges was less than six months. The plant's goal was to increase the MTBF while reducing their maintenance spend. The condition monitoring software allowed the customer to see valuable information as the centrifuges were being loaded, identifying that the centrifuges were being loaded too quickly, negatively impacting the MTBF. The system even enabled the plant to make several piping and process changes to improve the operation of the unit. The customer has driven MTBF for the units from less than six months to more than 150 months. The maintenance spend was reduced by a factor of 20, and shop labor requirements were reduced by more than 50%.

EfficiencyMap* Software Saves $40,000 Per Month

A combined cycle power plant requested that GE develop a custom graphical user interface (GUI) for their EfficiencyMap software installation, showing operators when they are optimized against grid constraints. The GUI enables the customer to easily optimize on a single power block, or on a whole-plant basis. These capabilities are allowing the customer to save $40,000 per month on an ongoing basis.

EfficiencyMap Software Saves Millions Per Year By Optimizing Partial Load Operation

A large electric utility operates numerous combined cycle plants and employs GE's EfficiencyMap* software to assist operators in optimizing partial load operation of the plants at night. One plant estimates savings exceeding $5 million per year while another plant estimates savings approaching $2 million per year.

EfficiencyMap Software Helps Optimize Cooling Tower Operation

A plant in the Southwestern USA uses EfficiencyMap software to optimize the cold end of their power generation process. By running their cooling fans using setpoints recommended by EfficiencyMap, they can save up to 250 kW of auxiliary power per fan under operating condition constraints that are in place approximately 3 months out of the year.

Upgraded Monitoring Systems Yield Rapid Payback

A refinery in the UK upgraded its older Bently Nevada 7200 Series Monitoring Systems to 3500 Series Systems and added online condition monitoring software as well. After extensive review, the plant elected to use our services organization for both project management and system installation. The upgrades were conducted during a planned plant outage, allowing startup to proceed on schedule with a fully operational condition monitoring and machinery protection solution.

The newly installed condition monitoring systems proved immediately useful both during startup and thereafter. They detected a number of machinery and process-related problems that previously would have gone unnoticed until more extensive damage occurred. Further, even if detected, the problems would have been time-consuming and difficult to diagnose without the aid of the new systems.

- During start up of a turbine-driven compressor, an improper interaction between the rotor and the balance piston resulted in an increase in vibration. The condition monitoring system allowed this problem to be immediately identified and corrected.
- Four months after start up, the rotating machinery engineer came in one morning and noted an increase in vibration as captured by the condition monitoring software on one of the turbine-driven compressors. Further investigation revealed that the lube oil had been contaminated with steam. The machine was shut down and the lube oil replaced. The customer advised that this one save alone had paid for the entire system.
- Carry-over from a steam boiler was causing a build-up of deposits on steam turbine blading, restricting steam flow and resulting in high axial thrust loading. By carefully observing process, vibration, and thrust position data, operators were able to keep the machine running without extensive damage until a planned outage, allowing them to procure spare parts in advance, minimize downtime, and conduct the repairs when there would be no incremental production losses – saving hundreds of thousands of dollars in avoided production losses.
- Software alarms were set in the condition monitoring system at levels below those set in the machinery protection system, providing earlier warning of developing problems. These software alarms alerted operators to water contamination in the lube oil system of a wet gas compressor and its associated mechanical-drive turbine, allowing early intervention by changing out the lube oil on both machines without interrupting production.

Had this condition gone unnoticed, a probable machine wreck would have ensued, resulting in costly repairs and lost production.

By using software alarms to flag changes in vibration vectors (amplitude and phase) on a compressor/expander train, the customer established to ability to detect the buildup of deposits on compressor blades. This allows them to accurately predict when a water wash of the compressor is needed, and schedule these washes at optimal times. The software also helps predict when thermal washes (i.e. steam injection) are necessary on the expander to clear excessive catalyst build-up on the blading.

XY Proximity Probes Identify Gearbox Problem Not Detected by Casing-Mounted Transducers

A Middle Eastern petroleum company utilizes a gas turbine generator package to provide electric power at one of its oilfield locations. A speed reducing gearbox is used to run the generator at 50 Hz, and the entire train is equipped with Bently Nevada proximity probes and companion machinery protection system. Since the first day of commissioning, high shaft vibration was observed by the proximity probes on the #1 journal bearing in the gearbox, even though seismic transducers on the gearbox casing indicated normal vibration amplitudes. The unit was subjected to close monitoring for the next 12 months, but no further increase in vibration was noted. However, during a planned outage and maintenance on the unit, the bearings were inspected and the gearbox bearing exhibiting high vibration was found to have wiped babbit on the lower half. The bearing was replaced, alignment was checked and improved, and the unit returned to service. Unfortunately, the vibration problem persisted. It was suggested that the rigid coupling between turbine and gearbox be replaced by a flexible coupling, but the vibration problem continued unabated. Finally, the plant elected to collect transient vibration data during startup and shutdown conditions using a Bently Nevada ADRE* system. Examination of this data revealed extremely large runout at slow-roll conditions and the decision was made to again open the gearbox, suspecting some type of mechanical runout problem. As expected, the customer found that the quill shafts where not concentric or parallel. The shafts were properly reassembled, trueness at the coupling faces was improved, and the gearbox returned to service. Upon startup, vibration data and runout data was greatly diminished, and the unit has since operated without incident.

The customer used the system to identify improper assembly of the quill shafts, modifying their maintenance procedures to ensure the quill shafts would be assembled properly in the future. Without this diagnostic data, the problem would have continued to plague the customer, greatly reducing the MTBF of the gearbox bearings, and possibly leading to catastrophic failure. So impressed was the customer with the ADRE system and the value of XY probes, they presented their experience at a gas turbine user's conference. Their presentation underscored the importance of using vibration data collected during speed transients for identifying problems, and the value of using XY proximity probes rather than relying solely upon seismic casing-mounted vibration transducers for machines with fluid-film bearings.

Oil Whirl Problem Identified and Rectified Using ADRE System

During refurbishment and upgrading of a 50-year-old fossil fired generating plant, the turbine generator was retrofit with XY proximity probes at each radial bearing. At start up, a Bently Nevada ADRE system was used to collect baseline data from the machine and a vibration instability was apparent. The diagnoses was complex and was carried out by the customer's machinery engineering team in conjunction with the contractor who had been awarded the refurbishment contract. Ultimately, oil whirl was identified, and the situation was remedied by slightly altering the bearing geometry to increase the loading and raise the margin of stability. Without the XY probes installed on these bearings, the problem would have been very difficult to isolate and rectify. The customer was extremely pleased with the contractor's ability to diagnose and resolve the problem, making full use of the capabilities of the Bently Nevada sensors and ADRE system.

Note: Oil whirl is a cyclic rotor instability excited by the circulation of the lubricating oil film between the shaft journal and the bearing. It usually occurs at a frequency slightly less than 1/2X (where 1X is the synchronous frequency corresponding to the rotating speed of the shaft). Oil whirl can be triggered by factors such as excessive bearing clearances, inappropriate bearing design, or improper radial loading of the bearing caused by machine misalignment.

Permanent Monitoring System Pays for Itself

A mechanical drive steam turbine is used to power a cooling water pump in a U.S. chemical complex. The turbine's original 8-stage design had been modified to a 6-stage design by removing two turbine wheels. During startup, the Bently Nevada monitoring system detected very high vibration levels as the unit passed through its first balance resonance. Analysis of the data provided by GE's condition monitoring software showed that the shaft was bowing, resulting in a rub. The rub was so severe that it actually slowed the unit, and resulting vibration caused piping insulation to fall off. A rotor dynamic analysis, enabled by both a model of the rotor system and the online data collected by the Bently Nevada system, showed that the rotor system was highly sensitive to unbalance and modifications would be necessary to correct the problems. First, the bearing stiffness needed to be reduced and the available damping needed to be more effective. Second, the loss of mass when the two turbine wheels had been removed had raised the balance resonance speed, and it needed to be lowered to a range closer to the original 8-stage design.

Two non-functional ("dummy") turbine wheels were added, and the bearings were modified. These changes reduced the rotor amplification factor from 14.8 to 4.2, allowing the machine to pass through its first balance resonance with far less vibration and completely eliminating the rotor rubs. The permanently-installed continuous monitoring system more than paid for itself compared to the costs that would have been required to install temporary instrumentation to first diagnose the problem and then confirm the results of the turbine modifications. In addition, the unit benefits from ongoing permanently installed machinery protection and condition monitoring which would not be the case had temporary instrumentation been used.

* denotes a trademark of Bently Nevada, Inc., a wholly owned subsidiary of General Electric Company.

XY Rod Position Monitoring Detected Piston Rod Connection Issue

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In the operation of a reciprocating compressor, crosshead looseness, piston rod bow, piston rider band wear and other problems contribute to irregular piston rod motion apart from the normal movement of a piston assembly. Rod position monitoring measures the rod position from the geometric center of the cylinder bore. The measurement comes from an orthogonally installed proximity probe pair on each rod at the pressure packing case. Continuous online monitoring of rod position along with the crank angle points to the sources of movement. Trending the amplitude and direction of piston rod vibration, the degree of crankshaft revolution, when the peak amplitude occurs, is an important tool for diagnosing compressor problems. Some compressor designs dictate that rod position measurements are much more suitable than traditional rod drop measurements. The following case history demonstrates that rod position monitoring provides unique insight of piston rod movement that would not be available in a rod drop measurement. Equipped by rod position information, the refinery was able to detect piston rod connection issues, reduce maintenance time, prevent catastrophic failure and avoid production loss.

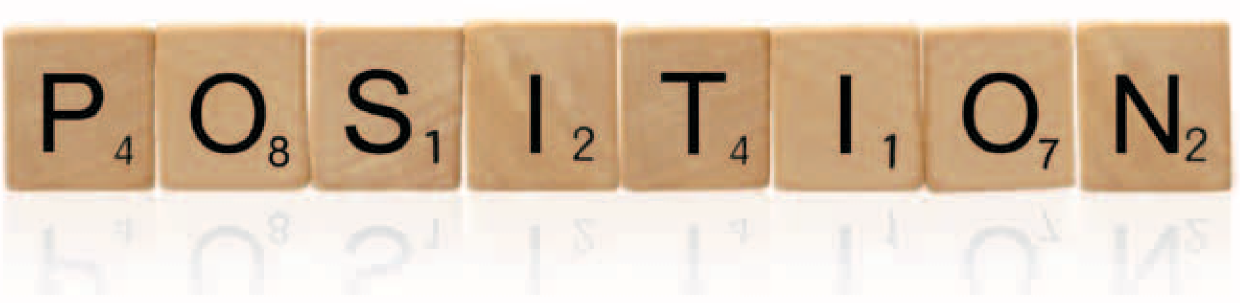
In a major North American refinery, two reciprocating compressors compress hydrogen for make-up service. In order to for the unit to run at full output, two compressors must run in parallel at all times except during start-up.

One of the two-stage reciprocating compressors has two horizontal throws, with double acting cylinders on each throw. The cylinders are lubricated with non-metallic piston rider rings. It is noteworthy that the piston material is aluminum, while piston rod is made of steel. The piston rod is connected to the piston and to the crosshead by Superbolt® fasteners. Because of the difference in thermal expansion characteristics of the aluminum piston and the steel piston rod, a compensating collar is installed in the piston hub. This collar is made of a special iron-nickel alloy called minovar, which has a very low coefficient of thermal expansion.

The operating condition of the reciprocating compressor has been continuously monitored by Bently Nevada's 3500 system and System 1* software. The parameters monitored include frame velocity, crosshead acceleration, cylinder pressure, rod position and various temperature points.

Two 11mm proximity probes for rod position measurement are installed on the packing case face. From the paired measurement, Bently Nevada's 3500 Rod Position monitor returns the maximum rod movement from the theoretical bore centerline as a vector quantity (Position Magnitude and Piston Angle) and the crank angle when the maximum displacement occurred. Each channel independently returns their peak-peak displacement.

Figure 1: System 1* View of the machine train: motor, gearbox, reciprocating compressor, and parameters.



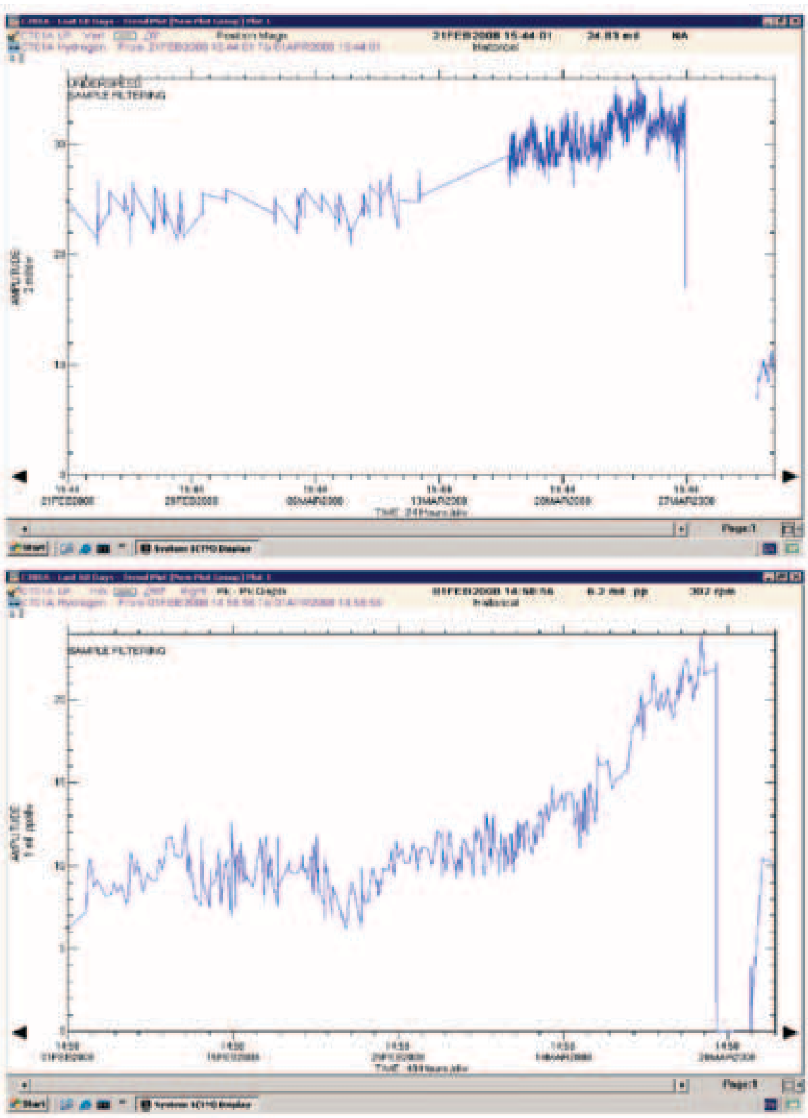


Figure 2: System 1 Trend plot for Rod Position magnitude (top) and horizontal peak-peak displacement (bottom)

DEFINITION: **Position Magnitude**

The maximum values of the position vector magnitudes are calculated every one degree of crank rotation over one cycle. The position vector magnitudes are the rod center position relative to a zero position representing the piston being concentric to the cylinder bore.

The Events

Shortly after the overhaul in June 2007, an unusual rod position pattern was observed. The center of the rod position deviated from its normal position and the piston rod position moved with patterns that did not repeat. Eventually the rod position magnitude alarm came in and our Machinery Diagnostics Services (MDS) team was called to review data. Toward the end of February 2008 operations reported hearing a slight knock on the first stage. The plant engineers reviewed all the data and the rod position probe mounting, and concluded that the rod movement was real. Since the amplitude change occurred in both horizontal and vertical directions and the probe mounting was not impaired visibly, the malfunction of the probes themselves was ruled out as the cause for the rod position movement pattern change (Figure 2).

Since the crosshead movement also contributes to piston rod motion, the crosshead acceleration and frame velocity trends were evaluated. Initially the data didn't reveal significant change. Performance of the compressor calculated from cylinder pressure parameters did not change either.

Due to the fact that a production penalty would occur if the compressor was shut down for inspection, the plant engineers decided to keep a close watch on these parameters, particularly verifying that crosshead acceleration stayed below the alarm limit, while the unit continued to operate. At the beginning of March 2008, the crosshead acceleration started to increase while the rod horizontal peak-peak displacement increase persisted (Figure 3).

The plant engineers suspected a rod connection problem at the crosshead when the rod position horizontal peak-peak displacement started to increase. Information was sent to another outside vendor, who confirmed the suspicion. Based on available information, a shut down decision was made. On Wednesday March 26th, operations were advised to prepare the compressor for a planned, controlled shutdown. On Thursday March 27th the Hydrocracker had to be shut down due to a fin-fan exchanger leak. This presented a window of opportunity for compressor inspection. The Process operation immediately prepared the compressor for the required maintenance inspection.

First, the crosshead Superbolt® was inspected and was found to be in good condition. All crosshead and crosshead pin clearances were found to be within specification. Next, the cylinder head was removed, revealing the piston nut damage. The minovar collar was found to be severely deformed and the piston rod had suffered thread damage. The collar and piston assembly were replaced, and the compressor returned to service without production loss.

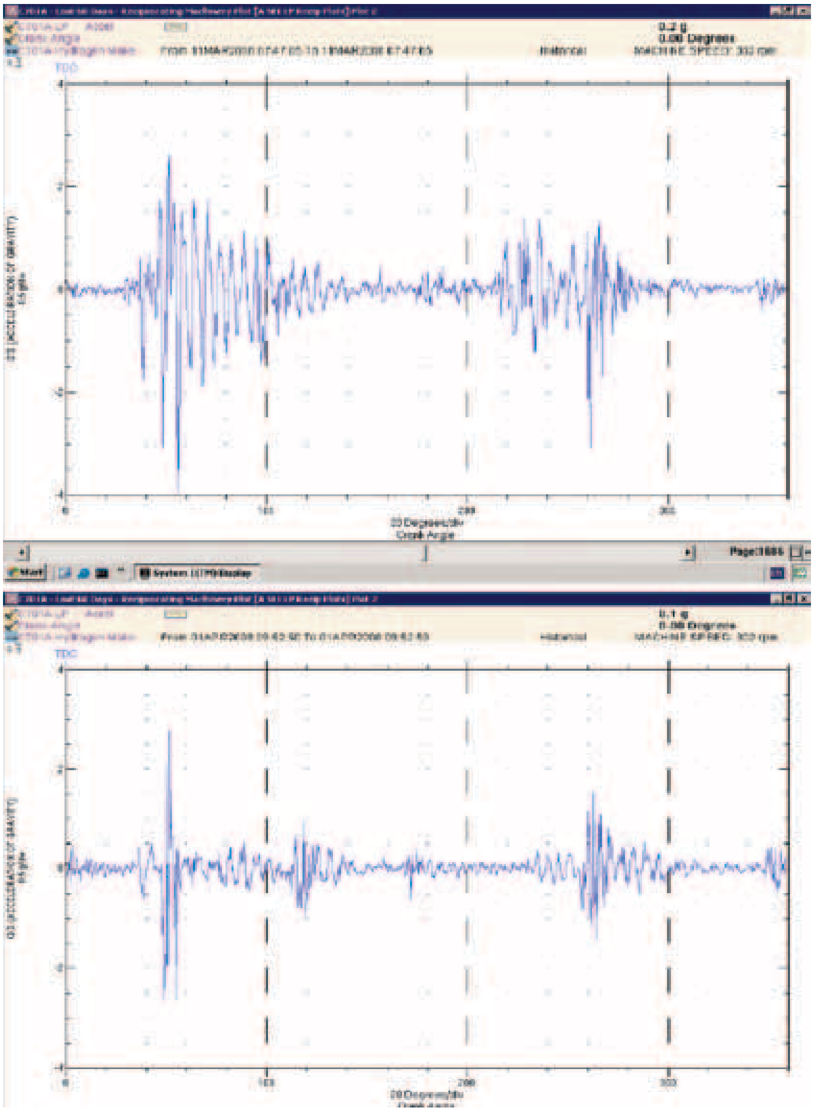


Figure 3: System 1 plots for crosshead acceleration versus Crank Angle, before (top)/after (bottom) repair. The screenshots showed only one stroke cycle, the historical data is stored continuously for every stroke of the compressor.

Figure 4: Minovar Piston Collar Damage

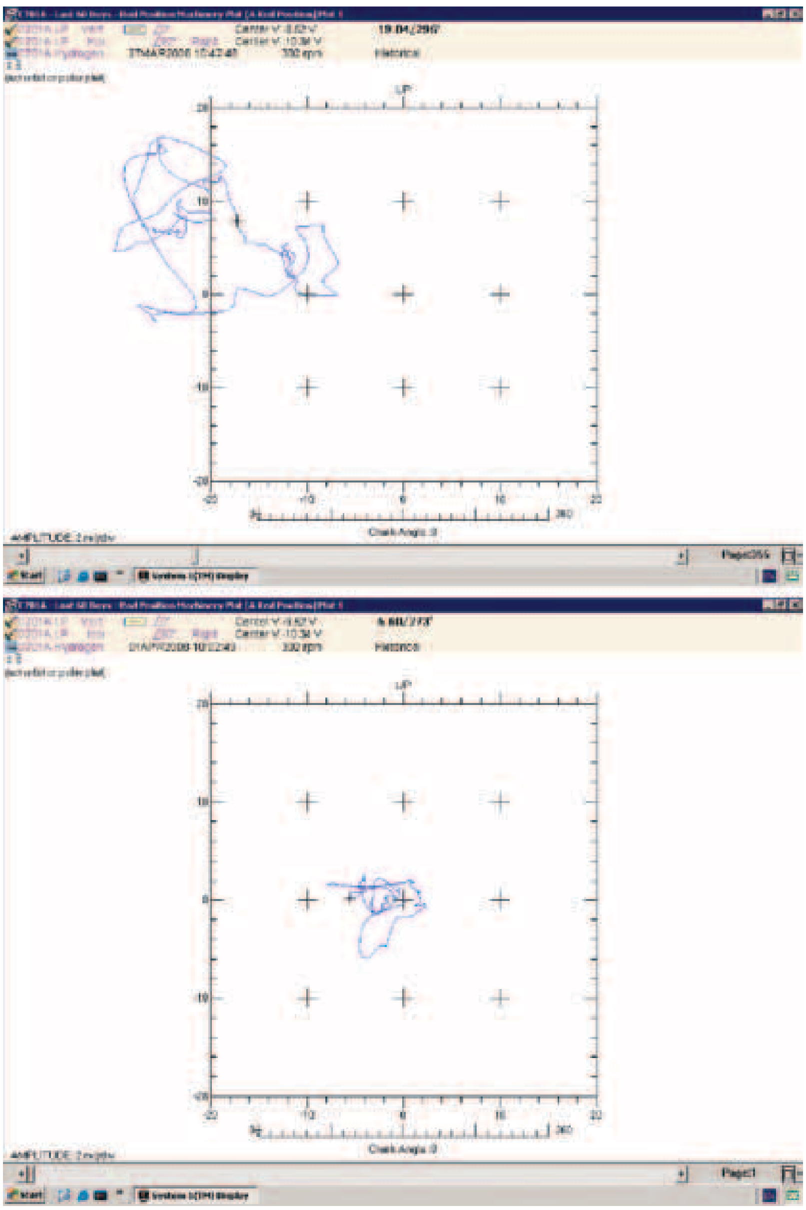


Figure 5: System 1 Rod Position plots before (top) and after (bottom) repair.

DEFINITION: **Crank Angle**

The rotational angle of the crank corresponding to the axial position of the rod (referenced from piston Top Dead Center) when the position vector is at its maximum magnitude. Represents where the rod is located in its stroke when the position vector is at its maximum magnitude. The crank angle is referenced from the piston angle (TDC being 0 degrees) with positive rotation in the direction of crank rotation.

Although the initial diagnosis of crosshead failure was not exactly correct, it was close enough to allow plant engineers to narrow the problematic areas, plan the shut down and reduce the time spent locating the broken part during the shut down. This prevented major failures with more severe consequences. If this had gone unnoticed, the broken piston collar pieces might have escaped into the cylinder and caused damage to the cylinder liner. Also the piston rod could have broken because of its loose connection. In any of these worst cases, the plant might have to extend the shut down from two days to two weeks, with production loss about one hundred thousand dollars per day.

After repair, the rod position plot showed rod displacement reduced and rod position moved to the center (Figure 5).

The rod position plot displays the dynamic horizontal and vertical movement of the piston rod during one complete stroke. The rod position plot displays each sample in the synchronous waveform according to crank angle. The total movement of the piston rod is displayed in both the horizontal and vertical direction. The center (zero in both vertical and horizontal axes) of the plot is the cylinder bore center. For example, in the upper plot of Figure 5, the cursor on the curve is at 19.04/295, which shows a Position Magnitude of 19.04 mil/Position Angle 295 degrees, where Crank Angle is at zero on the axis at the bottom of the plot.

Conclusion

The root cause for the broken piston collar was improper torque leading to distortion of the collar. A different material, hardened steel, is now more commonly used by the compressor vendor for collars on aluminum pistons.

The rising trend of rod position, especially horizontal displacement, proved to be critical in alerting the plant engineers. A traditional rod drop monitor would have missed the crucial information. The lesson learned from this case is that Rod position monitoring is an invaluable tool for accessing reciprocating compressor condition and planning outages. Excessive rod movement, particularly in the horizontal direction, can indicate a connection issue either at the piston or the crosshead. Proper design and installation of a rod position monitoring system ensures the effectiveness of such a system. The System 1 reciprocating compressor solution returned several machine saves to the refinery. According to plant engineers, the rod position monitoring made the difference "between running machines blind and having valuable information when something does happen. We were able to avoid any lost production by taking advantage of a plant outage and knowing what we were looking for going into it". ■

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DEFINITION: **Position Angle**

The angle made by the vector representation of the maximum position magnitude. It represents where the center of the rod is with respect to the zero position when the position vector is at its maximum magnitude. 0 degrees is defined as vertical and the angle increases in the clockwise direction as viewed from the crank end.

Cylinder Pressure and Rod Load Measurement in High-Pressure Upstream Reciprocating Compressors Using System 1*

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Three (3) two throw, two-stage, balanced opposed horizontal reciprocating compressors provide natural gas compression for the Boqueron field. Each compressor at full load provides approximately 49 mmscfd of compression capacity for injection into the wells. At this time, the field requires at least two units operating, HP-4100 at full load and HP-4200 with recycle on both stages (89 mmscfd between the two units.)

As part of on ongoing reciprocating compressor management program, various machine parameters are continuously monitored and trended with on-line instrumentation. Parameters monitored include cylinder pressure (pressure vs. volume or PV measurements), valve temperature, suction and discharge temperature, frame velocity, crosshead acceleration and piston rod position. A 3500 monitoring system provides alarms on parameters such as low rod load reversal, high vibration and changes in gap voltage. All parameters are collected and displayed in System 1.

Machine Description

These compressors, driven by a Caterpillar® 3616 reciprocating engine, provide high-pressure gas for re-injection into the Empresa Mixta Boquerón (PDVSA-BP) gas field. The table below summarizes the nominal operating conditions for these compressors:

	Pressure psig(kPa)	Temperature °F (°C)
1st Stage Suction	3069 (21160)	105 (40.6)
1st Stage Discharge	5855 (40370)	184 (84.4)
2nd Stage Suction	5691 (39240)	105 (40.6)
2nd Stage Discharge	10,101 (69640)	172 (77.8)

In late 2004, plant personnel and the local services team retrofitted these machines with a reciprocating compressor condition monitoring system, including on-line dynamic cylinder pressure. Figure 1 shows the transducer layout on one of these machines, C-4100. The others have been instrumented similarly.

Each machine is monitored for several parameters including flow balance, maximum rod load (compression), minimum rod load (tension) degrees of load reversal, peak cylinder pressure, discharge pressure, minimum cylinder pressure, suction pressure, compression ratio, overall crosshead acceleration levels and overall frame velocity levels.

On February 7th, 2007 at 2:26 in the morning, the 3500 rack issued a hardware alarm on the crosshead acceleration point B41V-41807. As can be seen in Figure 2, the compressor shut down 2:27 am as a result of this high vibration[1].

Data Analysis

Unlike purely rotating equipment, where rotating unbalance, misalignment, rubs, etc. place large stresses on rotating parts, the largest stresses on reciprocating compressors are caused by the combination of the pressures acting on the piston and inertial loads from starting and stopping the reciprocating masses at each end of their stroke. Experience with reciprocating compressors has shown that the crosshead pin bushing is the weakest link in the chain. The mounting location of B41V-41807 at the crosshead results in a signal containing both crosshead knocks and valve noise. In order to understand the impulse events within the signal, the diagnostic engineer plots both rod load and transducer waveform in the crank angle domain.

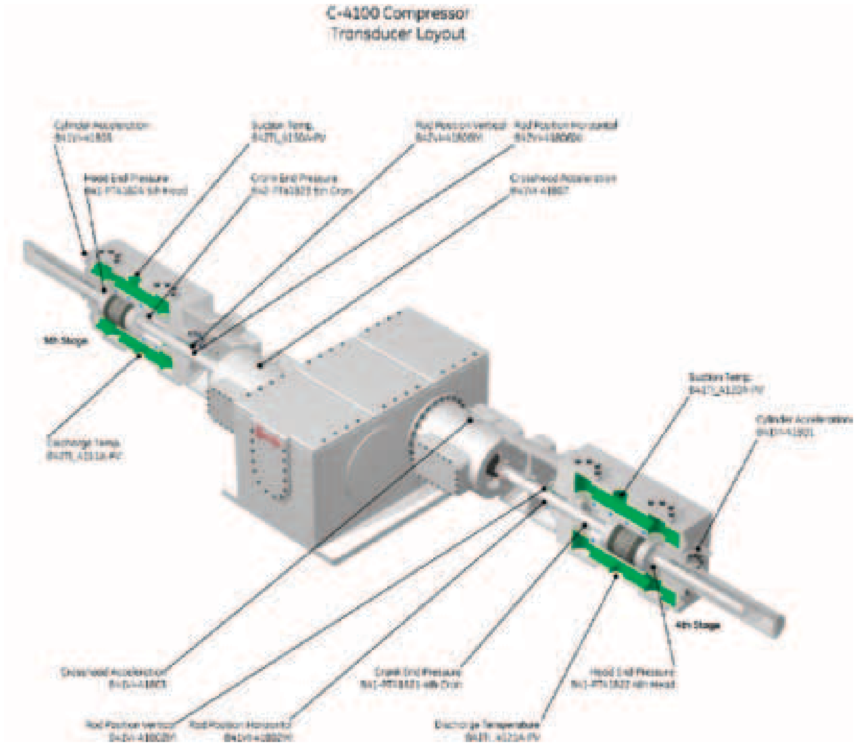

Figure 1: Transducer Layout

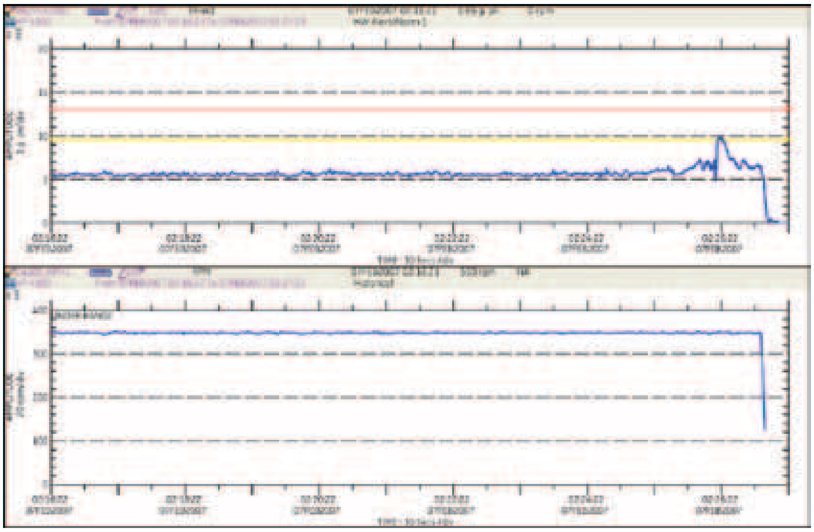

Figure 2: 5th Stage Unfiltered Crosshead Accelerometer Vibration Trend (upper plot) Machine rpm (lower plot)

[1] In this case the shutdown came from the unfiltered crosshead vibration signal because the customer wanted to shutdown on either valve problems or mechanical knocks. Frequently customers desire a shutdown for only high amplitudes originating from mechanical knocks. Applying a low pass filter results in waveform data that contains content related to mechanical knocks. For this reason many customers will configure a shutdown on the filtered, rather than unfiltered, waveform data.

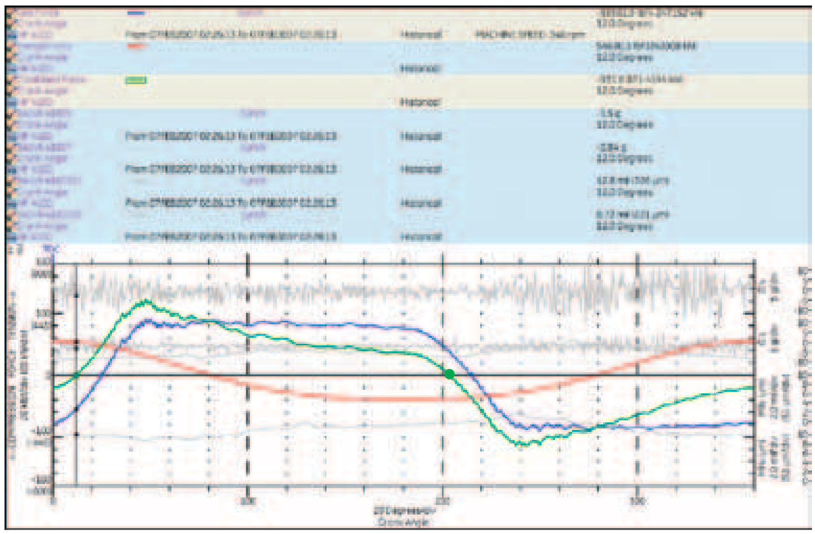

Figure 3: 5th Stage Rod Load, Cylinder Acceleration, Crosshead Acceleration and Rod Position Versus Crank Angle. 07 Feb 2007 02:26:13.

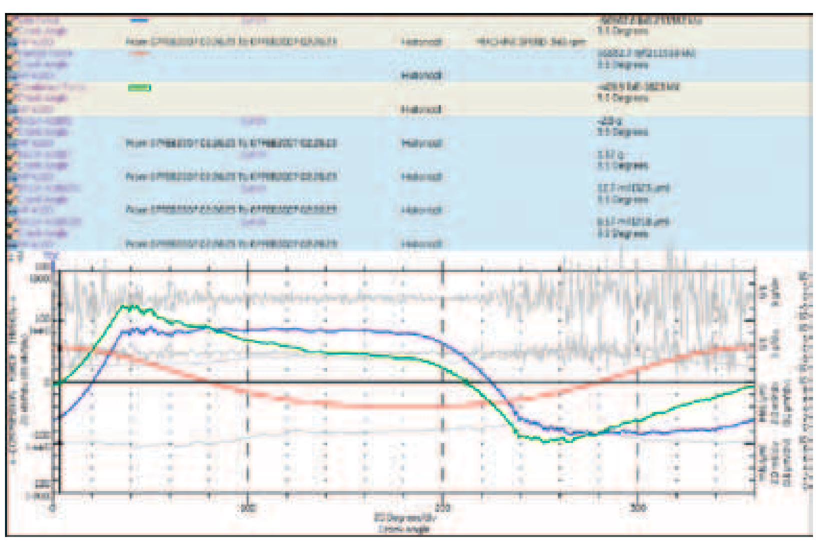

Figure 4: 5th Stage Rod Load, Cylinder Acceleration, Crosshead Acceleration and Rod Position Versus Crank Angle. 07 Feb 2007 02:26:23.

Figure 3 shows the combined rod load data for one full rotation of the crankshaft just a few seconds prior to the crosshead acceleration alarm. The red line is the inertial load curve, the blue line is the gas load curve, and the green line is the combination of the two. The vertical scale on the left side of the plot shows calculated load force in US units (lbf), while the horizontal scale at the bottom shows crank angle from 0 degrees, which represents Top Dead Center (TDC), to 360 degrees. In addition to the colored curves, four gray waveforms are shown. From the top down, these waveforms show cylinder acceleration, crosshead acceleration, piston rod vertical piston, and piston rod horizontal position, respectively. The vertical scale on the right side of the plot shows units of acceleration (g) and rod position (mil). Other than Figure 7 which is a trend plot, the rest of the plots in Figures 4 through 11 use this same arrangement.

The vertical load scales of Figure 3, minimum, -180,000lbf and maximum, 180,000lbf, reflect the maximum allowable crosshead pin, or combined rod load, ratings of the machine (Ref 1). The plot indicates 123 klbf of tension load and –114 klbf of compressive load. Both values fall well below the original equipment manufacturer (OEM) limits indicating operating conditions have not overloaded the machine.

In Figure 3 the green combined rod load line first crosses the neutral axis at 12° after top dead center (TDC). A green dot and the cursor position mark this point. As the forces at the crosshead pin reverse from compression to tension, the crosshead pin moves from the back, or frame side, of the crosshead bushing to the front, or cylinder side, of the crosshead bushing. The lubricating oil rushes in the fill this void on the frame side and cools and lubricates this void. At 204° ATDC the forces transition from tension to compression, this time marked with a green dot in Figure 3. This reversal of load forces causes the crosshead pin to move from the cylinder side of the crosshead bushing to the frame

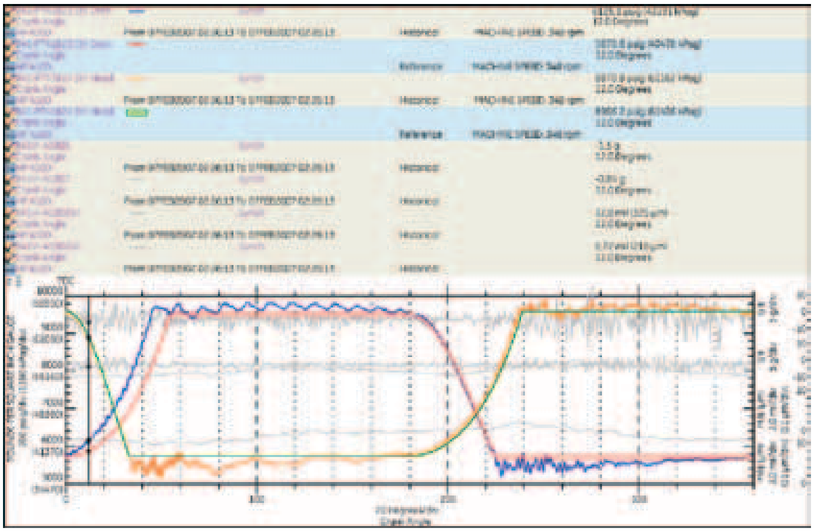

Figure 5: 5th Stage Cylinder Pressure, Cylinder Acceleration, Crosshead Acceleration and Rod Position Versus Crank Angle. 07 Feb 2007 02:26:13.

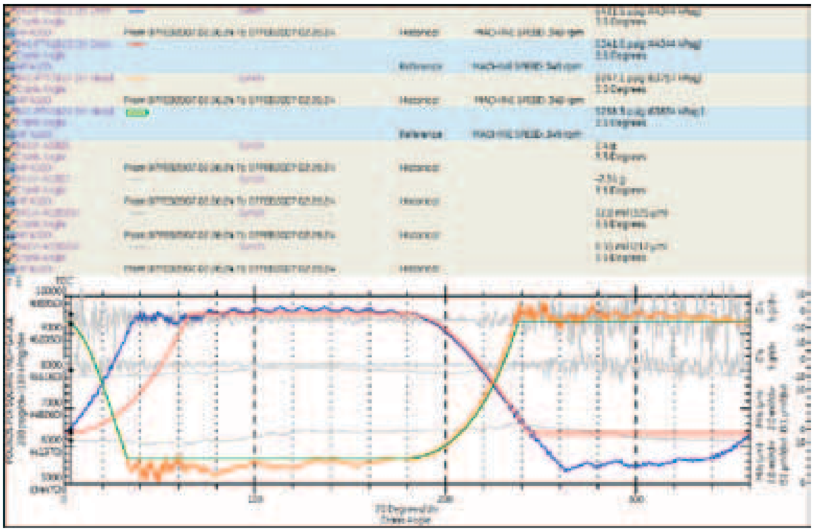

Figure 6: 5th Stage Cylinder Pressure, Cylinder Acceleration, Crosshead Acceleration and Rod Position Versus Crank Angle. 07 Feb 2007 02:26:13.

side of the crosshead bushing. Now lubricating oil fills the void on the cylinder side and provides cooling in this area.

Ideally, the compressor would have 180 degrees of reversal, but practically this is difficult to achieve. For this machine, the OEM recommends at least 60 degrees of reversal. In this case, the rod load curve shows 204°-12°=192° in tension and 360°-192°=168° in compression. So, in this case, the machine has 168° of reversal, well above the OEM limit of 60°.

If excessive clearance existed between the crosshead pin and bushing, the resulting buildup of kinetic energy would have been dissipated as a ringing of the crosshead and crosshead guide (Ref 2). The crosshead guide accelerometer waveform does not show any impulse events near gas or combined rod load crossing suggesting no looseness or mechanical problems.

Neither loading nor the accelerometer waveforms indicates a malfunction. Figure 4 shows another data set 4 seconds after the alarm occurred.

The rod load curve in Figure 4 displays similar values as before the alarm. Although combined rod load in Figure 4 shows slightly less reversal, 151°, the value still falls well above OEM limits.

As mentioned previously, discrete impulse events near either the combined or gas load points could indicate a problem with the machine; however neither accelerometer waveform shows these types of impulse events.

Based on this analysis, it appears that the alarm does not indicate a problem with crosshead-pin loading or mechanical looseness.

Referring to Figure 4, it can be seen that the crosshead vibration, from approximately 230° to 360°, has increased. In addition, the crosshead vibration, from 0° TDC to approximately 100° after TDC, has also increased. A leaking crank end discharge valve or a leaking head end suction valve can cause this pattern of vibration.

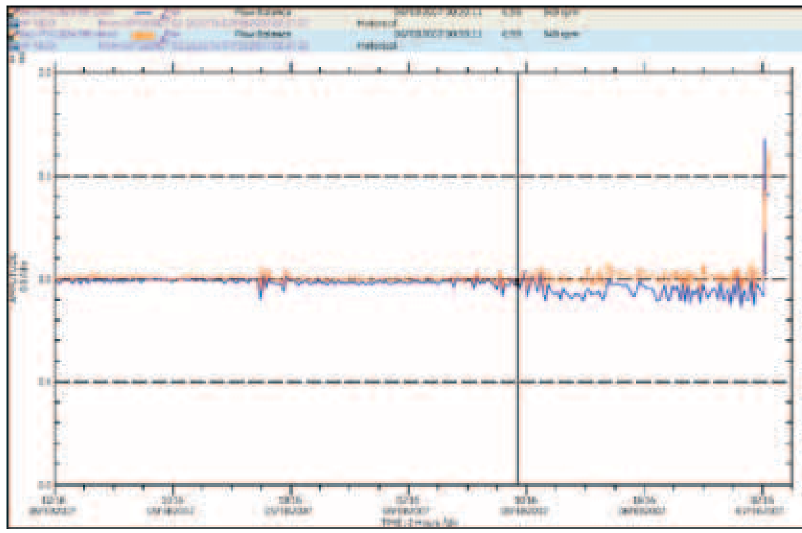

Figure 7: 5th Stage Flow Balance Trend.

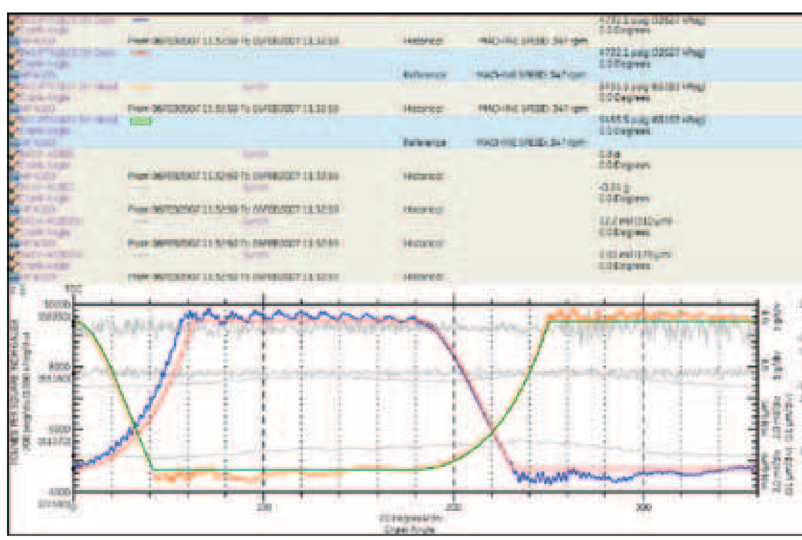

Figure 8: 5th Stage Cylinder Pressure, Cylinder Acceleration, Crosshead Acceleration and Rod Position Versus Crank Angle. 06 Feb 2007 11:32:59.

To isolate the end responsible for the amplitude change, the diagnostic engineer changed the plot format to show both indicated and theoretical cylinder pressure curves in the foreground, rather than rod load curves.

Within Figures 5 and 6 the blue line shows the indicated cylinder pressure curve on the crank end and the orange line shows the indicated cylinder curve on the head end. The red and green lines show the pressure curve for the isentropic compression process for the crank and head end respectively. As can be seen, in both plots the head indicated and isentropic curves agree well; however, in the case of the crank end, the difference between the indicated and isentropic curves increases between the two figures.

For a properly installed and configured system, a difference between the theoretical and indicated pressure curves suggests a leak path within the sealing systems of the cylinder. In this case, the indicated pressure inside the cylinder rises faster than the theoretical curve during the compression stroke of the crank end. Pressure can only rise faster if gas enters the cylinder from another source. The cylinder discharge manifold is the only potential source for this leak and the crank end discharge valve seems the most likely candidate.

The waveform shape of the cylinder, and to a lesser extent, crosshead accelerometers also supports this diagnosis. When the pressure inside the cylinder falls below the pressure of the discharge manifold, gas flows across the broken valve into the cylinder. Note how the accelerometer signal grows just after the pressure begins to fall and decreases as the pressure inside the cylinder approaches the discharge pressure.

When the internal cylinder pressure equals or exceeds the discharge pressure, the accelerometer signal amplitude greatly decreases. This interaction between cylinder pressure and manifold gives rise a characteristic shape associated with valve failure.

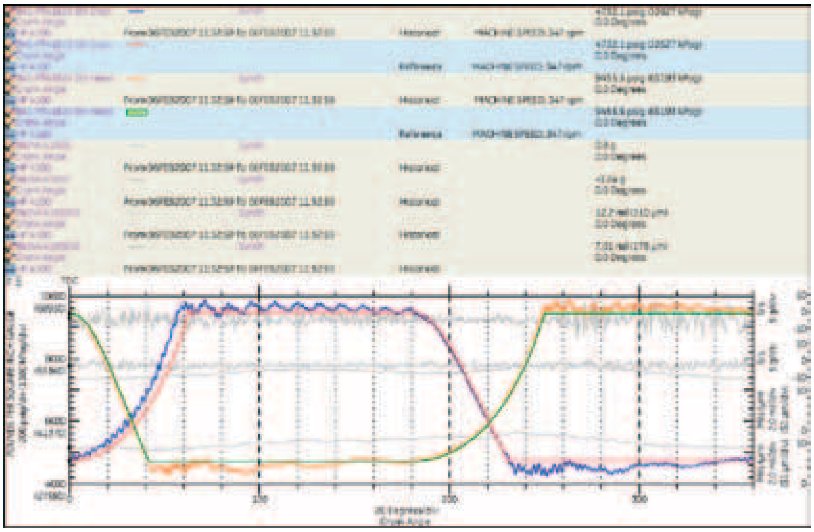

Figure 9: 5th Stage Cylinder Pressure, Cylinder Acceleration, Crosshead Acceleration and Rod Position Versus Crank Angle. 07 Feb 2007 19:06:31

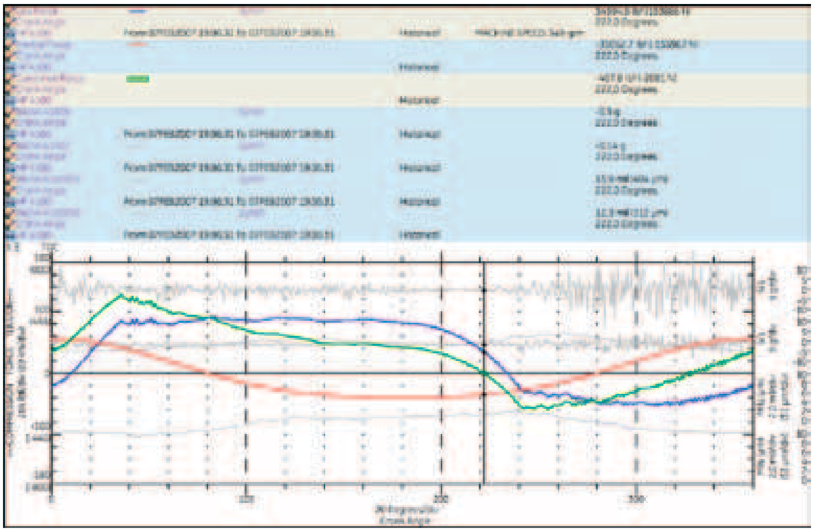

Figure 10: 5th Stage Rod Load, Cylinder Acceleration, Crosshead Acceleration and Rod Position Versus Crank Angle. 07 Feb 2007 19:06:31.

The two plots show that the leak existed prior to the alarm and that the leak seems to be worsening. In order to gauge probable valve life, the diagnostic engineer needed to understand when the leak began. In other words, the diagnostic engineer needed to understand when the indicated pressure began to deviate from the isentropic compression process.

The difference between the indicated and theoretical pressure can be quantified in a value referred to as flow balance. The flow balance takes advantage of the fact that a reciprocating compressor operates as a positive displacement machine. For such a machine, the number of molecules that come into the cylinder must equal the number of molecules that exit the cylinder. Practically, the software calculates flow balance as the ratio of indicated cylinder capacity at suction conditions divided by the indicated cylinder capacity at discharge conditions. In cases like this where the discharge valve leaks, the indicated discharge capacity increases and indicated suction capacity decreases resulting in a flow balance that falls below unity. Leaks that allow high-pressure gas inside the cylinder to escape to a low-pressure reservoir result in a flow balance greater than unity.

Figure 7 shows the flow balance value for both the crank end and head chambers of this cylinder from approximately 2 days prior to the alarm and up through and including the shutdown. Initially the values where quite close to unity. As time progresses, the crank end flow balance values (blue line) begin to fall below unity, indicating a discharge valve leak. Interestingly, this plot shows fluctuating flow balance values beginning at 9:30 am and then the valve failure at approximately 11:30 am. Figure 8, showing the cylinder pressure, acceleration and rod position data, confirms that the discharge valve leak had begun late in the morning of 06 February.


SYSTEM
RECIPS

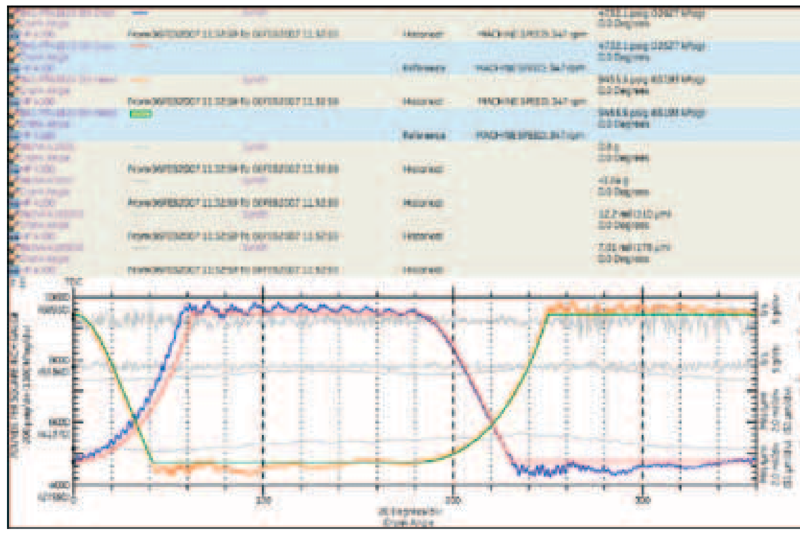

Figure 11: 5th Stage Cylinder Pressure, Cylinder Acceleration, Crosshead Acceleration and Rod Position Versus Crank Angle. 08 Feb 2007 15:36:55.

As the malfunction appeared to be related to the valve and an on-line system provided continuous monitoring, the plant staff decided to allow the compressor to run.

Valve condition deteriorated more rapidly than expected. Figure 9 shows the cylinder pressure, acceleration and rod vibration data seventeen hours after the alarm. The leak in the valve has become so severe that internal cylinder pressure on the crank end never reaches suction pressure. More critical than valve condition, as can be seen in Figure 10, the first reversal occurs at 222° ATDC and the second at 328° ATDC. Degrees of reversal have dropped from the 161° shown in Figure 3 to 106°. Although not yet near OEM limits, the rapid change of rod load and reversal over a seventeen-hour period clearly shows lubrication conditions at the crosshead pin to be deteriorating rapidly.

Conclusions

Leveraging the data from the on-line condition monitoring system provided critical information, not only to diagnose the valve, but to extend operational times safely without risking collateral damage to other components. Additional benefits from the system have included:

- Reducing cycle time of failures detection, avoiding catastrophic failures of components such as cylinders, pistons, and rods.
- Reducing downtime by 60%, focusing repair disassemblies only to failed components.
- Reducing maintenance costs, replacing only the failed parts and using efficiently the human resources.
- Increasing the availability of compressor, only operating with optimal performance conditions.

References

1. Atkins, K.E., Martin Hinchliff and Bruce McCain. "A Discussion of the Various Loads Used to Rate Reciprocating Compressors." Proceedings of GMC 2005.

2. Howard, Brian. "Loose Piston Detection for Reciprocating Compressors." Compressor Tech Two, April 2009.

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XY Measurements for Radial Position and Dynamic Motion in Hydro Turbine Generators

As an introduction to our "Hydro Corner" series we are presenting a back to basics perspective regarding the importance of XY measurements on hydro-electric turbine generators. Hydro power generation remains a reliable, low cost, renewable source of energy. Many hydro generators have been in operation for over four decades. The changes in energy demand have contributed to changes in operating modes for many of these units which urges operators to apply and explore condition monitoring techniques for improved asset management.

Therefore, we begin our series with focus on mechanical vibration as one of the most important measurements providing information about machinery physical condition. For example, vibration measurements are used for hydro-generators to measure: (a) rotor relative vibration, (b) rotor absolute vibration, (c) seismic vibration of chosen hydro-generator components. For some units it is also important to measure (d) rotor torsional vibrations. In this article, the focus is rotor oriented transverse vibration measurements.

Rotor vibration was used as a turbine condition measurement in the first half of the 20th century. For that purpose, non-contact transducers are currently used. These are mostly eddy current transducers, which were introduced successfully into the industrial environment by Donald W. Bently in the 1950s. These non-contact transducers provide not one, but two simultaneous measurements:

(a) Variable voltage (Vac) informs about dynamic movement of the observed surface (usually the rotor)

Figure 1: Examples of metallic (up) and rubber (down) segments of various guide bearings.

Figure 2: An example of XY transducer configuration where probes are connected to guide bearing cover.

(b) Constant voltage (Vdc) that provides information about average distance between transducer tip and the surface that allows for an estimate of rotor average position in a bearing clearance. This measurement is very popular for high speed rotating machinery with sleeve bearings. For many types of machines this measurement is used not only as a basic protection parameter, but as a source of additional fundamental data for asset diagnostics where a predictive maintenance strategy is applied. In the case of turbo machinery, these non-contact vibration transducers are fixed close to machine bearings. This same approach is usually taken for hydro-generators.

Careful consideration of transducer location is important due to significant construction differences of High speed machinery and low speed hydro-generators. For Example:

- High-speed machines usually operate above the first rotor resonant frequency (or even higher) whereas high power hydro-generators usually operate below the first balance resonance.
- There are usually disks with blades between turbo-machinery bearings whereas the high power hydro-generators usually have a "bare" (inner) shaft and easy access space between the turbine and the generator guide bearings;
- Horizontally mounted machinery usually have relatively stiff bearings (both radial and thrust), whereas vertically mounted machinery, with very heavy rotors, usually have a relatively stiff thrust bearing support system, and relatively less-stiff guide bearings;
- Horizontally mounted machinery use mostly metallic bearings (and for chosen applications gas or magnetic ones) whereas vertical hydro-generators, turbine runners use either metallic or rubber guide bearings ; the rubber bearings have less impact on the environment because they are lubricated by water and therefore the risk of oil leaking to the running water is minimized.

  Figure 1 presents examples of various bearing segment surface coating (metallic and rubber) used for turbine guide bearings of slow speed hydro-generators;
- Horizontally mounted machinery frequently has various radial clearances (vertical, and left/right side) whereas vertically mounted machines use circular guide bearings with uniform clearance. The inner diameter of hydro-generator bearings is set just larger than the diameter of shaft bearing section, and the difference, referred to as the bearing clearance, is expressed as the size of the gap between the bearing and the surface of the shaft if it were centered in the bearing.

Relative Vibration

There are two approaches for connection of shaft rotor vibration transducers on vertical hydro-generators. The first one uses non-contact transducers that are attached orthogonally (XY-configuration) to a chosen component of the guide bearing.

The connection can be done to the bearing or directly to the guide bearing pads. Such a connection allows checking of shaft position in a bearing clearance and additionally allows measuring rotor vibration relative to the guide bearing as presented in Figure 2.

The orientation of XY transducers at different bearings preferably should be in line. For horizontal machines, XY-transducer orientation (+45° and -45°) from vertical is very common, but other orientations are acceptable given consistent conventions. For vertical machines, the suggested XY-measurement directions are upstream and 90° to that.

Absolute Vibration

For vertical hydro-generators, relative motion between the shaft and the transducers is frequently small compared with the absolute shaft motion (and even smaller when the transducers are connected to the bearing pads). Therefore, measured absolute vibration is the second approach and is a necessary additional measurement. It can be done using the same XY-transducer configuration, but now the transducers are also connected to orthogonally oriented rigid frameworks, fixed to the generator or turbine pit wall, that allows for measurement of rotor absolute vibration with respect to the structure. Figure 3 presents examples of such applications for [a] generator guide bearing and [b] inter-shaft dynamic measurements. The same approach can be used for guide bearing absolute vibration measurements if the transducers measure movement of the guide bearing housing.

In accordance with ISO standards [1] "signals from these transducers can only be regarded as representative of the absolute shaft vibration when the absolute vibration of the supporting structure itself at the point of attachment of the transducer is less than 10% of the measured peak-to-peak value, with 25um as an upper limit".

Both types of shaft vibration measurement provide useful information about rotor system dynamics. However, from a diagnostic viewpoint, the proper transducer arrangement allows measurement of relative rotor vibrations and enables additional monitoring of bearing clearance and shaft alignment in the bearing.

In the case of basic rotor vibration monitoring, the non-contact transducers are usually located close to the guide bearings. However, when considering XY transducer installation as part of a monitoring system retrofit, the following recommendations should be considered:

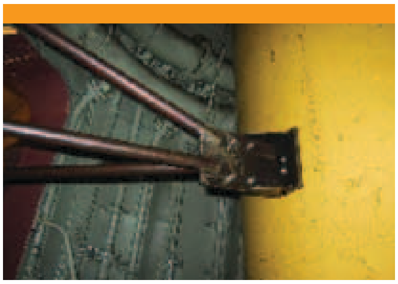
1 What is the rotor system operating resonance frequency? There are two possible answers:

A. Operating over the first resonant frequency. At this speed, the highest deflection of the shaft

Figure 3: Examples of XY transducer fixings allowing measurements of rotor absolute vibrations:

a) Two rigid bars holding non-contact transducers are connected to a static structure

b) An example of transducer holder inside of a hydro-generator chamber: holder fixing to the chamber wall (top), and holder head – allowing non-contact transducer connection (bottom).



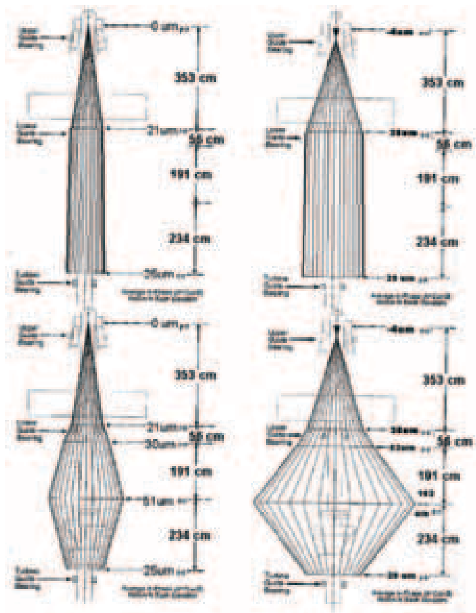


Figure 4: Estimations of shaft mode shape based on XY transducers located close to guide bearings (3 bearings – up) and using 2 additional XY-planes (down) at 60 CPM (left) and 660 CPM (right – after load rejection) [1].

occurs somewhere between bearings. Therefore, locating the XY transducers between guide bearings provides much better indication of rotor dynamic behaviors.

B. Operating at first rotor resonant frequency (rotor frequency corresponds to rotor operational speed). We can expect that during some malfunctions, the modal stiffness of the rotor system will decrease, and consequently the rotor system resonant frequency decreases. Therefore, the rotor system resonance is affected by the rotor operational speed which will cause the rotor system to change its operating mode from under resonance to over resonance.

In this scenario, it is important to add additional XY-transducers (or at least one set of them) between guide bearings.

This additional measurement plane can significantly help with estimation of the real rotor shape.

Figure 4 [2] presents an illustration of two various estimations of a hydro-generator rotor shape depending on the number of planes of XY-measurements. Accuracy of dynamic shaft bending mode is more accurate when a higher number of XY- measurement plains are used. This conclusion is true as well for very low (60RPM) and very high (660RPM) speed hydro-generator operations.

Various modes of shaft bending for hydro-generator operating conditions are described in [3].

The referenced example describes a 3-bearing unit operating in a pump storage station, presented shaft motion for generator, pump, and synchronous condenser modes respectively, before and after overhaul. The shaft bending modes are very different in the various operation modes, due to the significant eccentricity between the dynamic electromagnetic, thrust bearing and hydraulic centers. Identifying real shaft bending during various operation modes is easier when having more planes of XY measurements.

2 What is the bearing clearance during a serious bearing malfunction?

The answer to this question is especially important for hydro-generators that use a rubber bearing as the turbine guide bearing. Usually clearance of radial bearings is on the order of several hundred micrometres (microns). Therefore, for monitoring of shaft movement with XY transducers, fixed close to guide bearings, it is recommended to use transducers with a minimum of 2mm range of operation. However, there have been cases of units with rubber bearings in a hydro-station, that in the case of catastrophic failure, bearing motion can be even in the range of 3–4 mm. Therefore, for rubber bearings, it is recommended to apply transducers that have an even larger linear operation range than the expected maximum bearing clearance.

3 Is it possible to get significant rotor unbalance that would destroy the guide bearing?

It is be possible for propeller type runners to lose a blade. This failure mode immediately results in very significant rotor system unbalance. If guide-bearing construction is not robust enough, the unbalance destroys the bearing and consequently can destroy the XY transducers fixed to the bearing cover.

Depending on available features of the monitoring system used, various outcomes of such events are possible. The following features of a modern monitoring system are important for such an event:

- Wide hardware auto-diagnostics including a proper indication of "NOT OK" status for channel or monitor operation.
- Bypass capability of some channels for valid reasons.
- Availability of various types of voting in a protection relay system (for a scenario such as a significant increase of vibration. In this case, one or both channels of the monitoring system could be "Bypassed" if in the "NOT OK" status.

Therefore, when considering a hydro-generator condition monitoring system, one should not only consider proper choice of transducers and optimal places of transducer location, but additionally available features of monitoring and protection systems according to the role of the hydro-generator in a production system. Sudden failure can destroy critical components of the condition monitoring and protection system.

Overall reliability and effective operation of a monitoring and protection system is related to a variety of factors including: required range of transducers, location of XY transducers, transducer cable routing and available functionality of the monitoring system.

Figures 2 and 3 show a transducer arrangement where both the probe and shaft surface are very accessible. In this scenario, it is easy to replace a transducer if there is an operational problem. However, for some hydro-generators, transducers have to operate in an enclosed space, where quick probe replacement can be problematic. Therefore, for hydro-generators, it is important to consider installing redundant XY-transducers to increase the reliability of the monitoring and protection system. The redundant transducers can be fixed:

1) Opposite of the current shaft observing XY-transducers, and/or
2) Without significant angular shift when compared to the existing XY-transducers, and/or
3) Without significant axial shift when compared to the existing XY-transducers.

The term "shift" means that the distance between the two sets of probe tips has to be greater than the probe separation recommendations in the transducer's technical documentation. If this condition is not met, then an interaction between both transducers can occur (often called cross-talk) decreasing signal to noise ratio.

Figure 5 presents an example of redundant dual radial proximity probes mounted through the casing. Depending on the criticality of the hydro-generator, these 'redundant' probes may simply be installed spares with cables connected to a Proximitor* sensor. They would not actually be wired to the monitoring system unless the normally connected probe fails.

Figure 5: An example of redundant XY transducer observing shaft in the closed space (only one of XY transducer and its redundancy is visible at the photo)

When a hydro-generator rotor system is balanced and aligned properly, the shaft should spin within the confines of the guide bearings without much force being exerted against these bearings. Based on XY-measurements:

- Clearance of guide bearings can be estimated based on data that is acquired during unit startup. This is because the shaft moves in a random "orbit" throughout the clearance set by the guide bearings for the first few revolutions during unit startup. Therefore, when measuring shaft movement for the first few revolutions, (when the radial forces are not significant because of low speed – e.g. 8 orbits after start-up) guide-bearing clearance can be estimated quite accurately using orbit analysis. This data can be collected for various temperature conditions of the guide bearings:

  — cold condition
  — hot conditions

It is also very useful to establish a coefficient of temperature influence on the bearing clearance

- During normal operation (normal speed and bearing temperature) you should know:

  — Rotor position in the bearing (Vdc measurements)

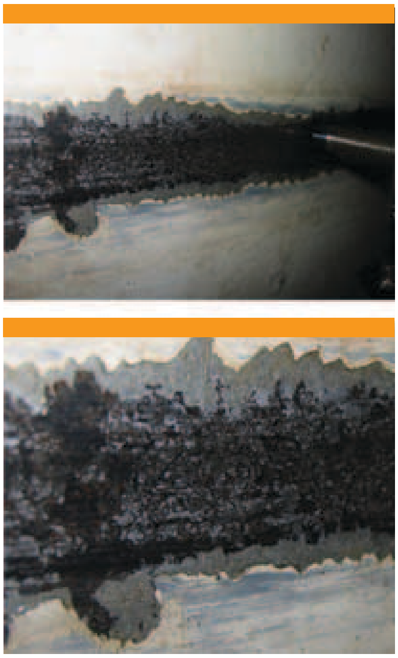


Figure 6: An example of eddy current transducer operating with a 250MW hydro-generator shaft in area close to the guide turbine bearing (upper) and closer view of the shaft surface (lower).

  — Rotor dynamic motion relative to the bearing clearance (Vac measurements).

From a diagnostic point of view, it is important to compare vibration (Vac) with bearing clearance. The observed shaft vibration at operating speed has often appeared to be greater than the nominal clearance of the bearing. However when disassembling the bearing later it was found there was no visible proof of the expected rubbing in the bearing.

This occurred because the nominal bearing clearance during the assembly process is measured during a cold bearing condition. During operating conditions we expect bearings to operate in a hot condition (which results in changes of linear static dimensions):

- There is a dynamic component of the radial clearance as a consequence of deflection of the bearing segments (pads), and
- Deflection of the bearing shell (e.g. under the bearing pad, possible deflection of the shell in points where XY probes are fixed).

Because of these effects, the apparent bearing clearance indications seen during operation are often less trustworthy than the data that is collected at slow speed during a startup. If it is available, oil analysis data can provide useful confirmation of bearing wear by indicating the presence of wear particles or elements that are present in babbitt (copper, lead, tin, etc.).

For non-contact rotor measurements, eddy current transducers provide a local electric current induced in a conductive material by a magnetic field produced by the active coil in the probe tip which in turn changes the inductance in the coil. When the distance between the target and the probe changes, the impedance of the coil changes accordingly. Because the observed surface takes an active role in transducer operation, it is required to prepare the surface to be observed for proper transducer operation.

The following is one of XY-eddy current transducers operating with an unprepared shaft area close to the turbine guide bearing. One can see the very bad surface of the shaft (Figure 6). This is actually typical for this rotor section after long periods of operation in corrosive environments. However, even with this corrosion the signal quality can be sufficient to collect reliable data about rotor system dynamics and shaft position in bearings.

As shown in Figure 7, dynamic data was collected from the transducer arrangement. The data is of sufficient quality (adequate signal-to-noise ratio) to allow reliable inference about rotor-system technical condition. Changeable amounts of humidity or the presence of oil do not influence operation of the eddy-current transducers substantially. Even though a trained expert can interpret this data, these plots point out the noise and error sources that can be induced by the observed shaft surface.

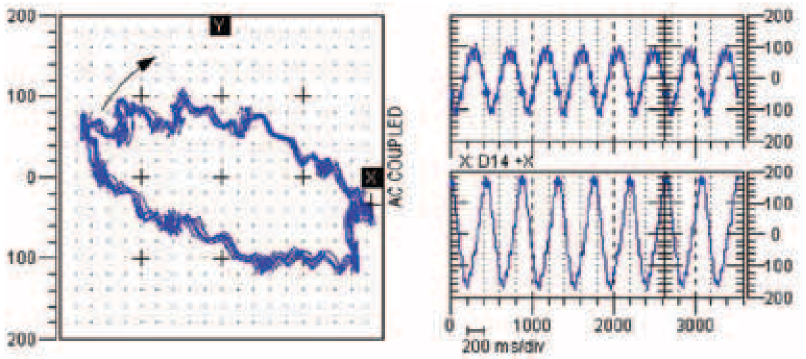


Figure 7: Dynamic data (waveforms and corresponding ORBIT) collected from the XY eddy current transducers introduced in Figure 6.

Recommended non-contact transducers for XY measurements on hydro-generators (=HG)		
Turbine description	**Place of transducer fixing**	**Size of 3300 XL series transducers**
Horizontal HG:	Close to a bearing housing	8 mm standard
Vertical HG with Francis turbine:	Close to a guide bearing	8 mm standard
Vertical HG with a propeller type turbine:	Close to a guide bearing	11 mm standard
Turbines with a rubber bearing:	Close to a guide bearing	11 mm standard
Inter-shaft:	Around of mid section of the inter-shaft	11 mm standard
Measurement range of the advised transducers in a hydro station application: • 8 mm transducers: 0.25 to 2.5 mm • 11 mm transducers: 0.5 to 4.7 mm		

Monitoring systems that can be used for XY measurements provided for hydro-generators:

1. 3500 monitors /40, /42, and /46: 3500 Systems are dedicated to larger hydro-generators that need monitoring and protection.
2. 1900/65A: 1900 Systems are dedicated to smaller hydro-generators that need monitoring and protection
3. Trendmaster* DSM Systems do not provide automatic machine trips. They can be used alone for hydro-generators that only require condition monitoring, or in conjunction with 3500 or 1900/65 Systems for machines that require protection.

These monitoring systems perform signal processing to provide the following measurements: Gap, 1X vector, 2X vector, nX vector, not 1X, S_{max}, and more.

Literature

[1] INTERNATIONAL STANDARD ISO 7919-5; Mechanical vibrations of non-reciprocating machines – Measurements on rotating shafts and evaluation criteria – Part 5: Machine sets in hydraulic power generation and pumping plants.

[2] Robert C. Eisenmann, Sr., Robert C. Eisenmann, Jr., Machinery Malfunction Diagnosis and Correction, Hewlett-Packard® Professional Books, ISBN 0-13-240946-1, Ch. 10

[3] Marcus Craham, III; Duane A. Gettman, Technalysis, Irvine California; Dramatically improved hydroelectric generator performance through total unit centerline erection – a case history; presented at the Annual Meeting of the American Power Conference 86-04-14

[4] R V Jones, J C S Richards; The design and some applications of sensitive capacitance micrometers. Journal of Physics E: Scientific Instruments, Volume 6, 1973, p. 589-600.

[1] In old hydro stations it is still possible to see bearings with wooden strips on some smaller units. They are lubricated by water similarly to the rubber bearings.

[2] A transducer connected to the bearing segment observing shaft motion from position of the segment, and transducer connected to the bearing shell observing resulting movement: the shaft movement in the bearing, bearing segment deformation, and the shell deformation (as consequence of the bearing segment pressure) under radial action of the shaft. Therefore, (a) usually the measurements of shaft relative movement observed from the bearing pads are lower than shaft relative movements observed from the bearing shell, and (b) in the case of shaft relative movement measurements provided from the shell, if the movements are not significantly bigger than bearing clearance, no evidence of rubbing is visible after the bearing disassembling.

[i] Alternatively for XY measurements noncontact capacitance transducers can be used.

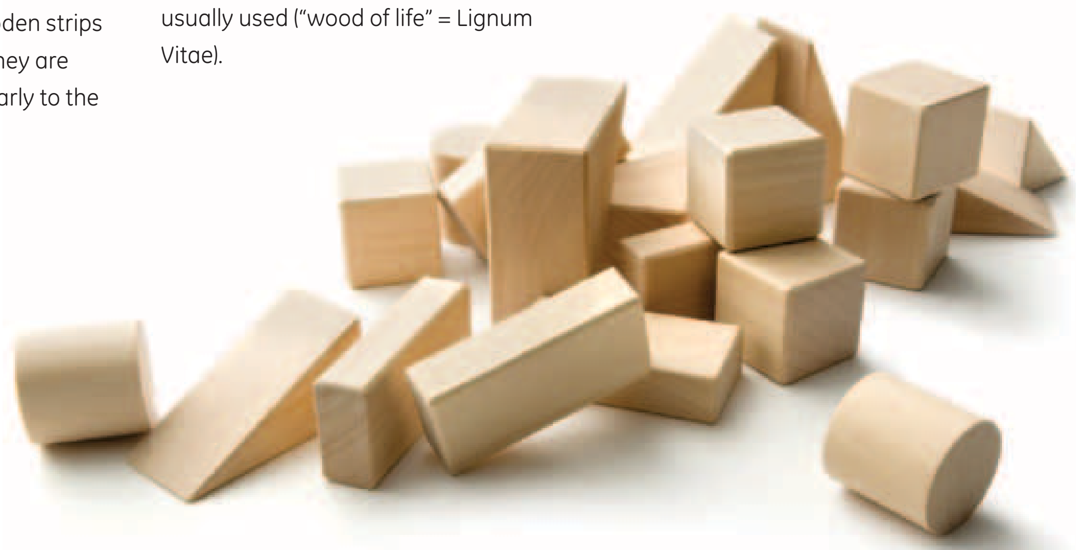
[ii] In some old hydro stations it is possible to find hydro generators that have wooden guide bearings. For such applications, wood has been usually used ("wood of life" = Lignum Vitae).

[iii] It is not possible to use capacitance transducers if they have to be connected to the bearing pads. This space contains air with a changeable amount of fluid used for the bearing lubrication (oil or water – depends on bearing construction). This strongly impacts capacitance transducer sensitivity.

[iv] There is usually no damage to XY probes when they are fixed to bearing pads. However, routing of non-contact transducer cables should be provided in a way that the cables are not broken during discussed type of runner malfunction.

[v] BENTLY NEVADA 3500 SYSTEM can provide Normal "AND" Voting and "True AND" Voting.

[vi] Some users look for very simple condition management systems that have very limited functionality, and are used only as provider of basic measurements to a DCS. However, many of the necessary functionality that is recommended can only be found in a more advanced monitoring and protection system.



Cracked Bearing Race Detection in Wind Turbine Gearboxes

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[Note: This article is based on a paper presented at the China Wind Power 2010 conference in Beijing, China in October 2010.]

Wind turbines are becoming established as an economically viable alternative to fossil-fueled power generation. Wind farms consisting of hundreds of units are now adding a significant amount to world generating capacity. As the size of wind farms continues to increase, business economics dictate careful asset management to minimize downtime and maximize availability and profits. A wind turbine condition monitoring system is essential to achieving that goal.

Wind turbine condition monitoring tends to focus on two primary groups of vibration frequencies: gear mesh frequencies and bearing defect frequencies. On typical 1.5 MW machines, the gearbox increases the input rotor speed of around 16 to 20 rpm to the generator speed of from 1400 to 1800 rpm. Most of these gearboxes have one planetary stage followed by two parallel stages. These stages typically produce three fundamental gear mesh frequencies (and their harmonics) that result from the meshing of a sun gear, planets, and a ring gear, plus two other pinions and gears.

Bearings in wind turbines are typically of rolling element design, and the gearbox contains five shafts supported by twelve or more bearings, each of which produces a set of five defect frequencies (outer race element pass, inner race element pass, cage or fundamental train, element spin, and twice element spin). It is easy to see that the combination of mesh frequencies, harmonics, and bearing defect frequencies can make frequency analysis a formidable task.

In addition, wind turbines are variable speed and power machines. Because of this, specific bearing and gearbox fault frequencies change with speed and must be carefully tracked. Variable power means variable torque and associated changes in gear meshing forces that can affect vibration amplitudes and mesh harmonic frequency content.

GE Energy has incorporated this knowledge into a new monitoring platform specifically designed for wind turbine condition-based monitoring, the Bently Nevada ADAPT.*wind monitoring system. The system includes a Bently Nevada 3701/60 monitor installed uptower in each wind turbine on the farm and a central farm server. The monitor uses an Ethernet connection to connect to the farm Supervisory Control And Data Acquisition (SCADA) network, which communicates to the farm server. The farm server is typically located at the farm central office, where it collects data from each 3701, and stores it in a local historian. ADAPT.wind software on the farm server is then used to view alarms and data collected from the monitors.

The ADAPT.wind platform automatically adjusts for wind turbine speed changes and extracts specific fault frequencies. To help compensate for the variable power and torque effects, the operating power range of the machine is divided into five bands, or modes, each with separate control over alarm levels. This allows more consistent comparison when trending data over long periods of time.

Recently, a number of new Bently Nevada 3701 ADAPT.wind monitoring systems were installed on a fleet of 1.5 MW wind turbines. These machines typically have three-stage increasing gearboxes: a planetary stage followed by two parallel stages. The 3701 system uses one accelerometer on the main bearing, three on the gearbox, and two more on the generator inboard and outboard (upwind and downwind) bearings. The three gearbox accels are mounted on the planetary ring gear, the high-speed intermediate shaft downwind bearing, and immediately adjacent to the high-speed output shaft downwind radial bearing. In addition, a two-axis accel is built into the 3701 for monitoring tower sway. The system can also utilize a Keyphasor* probe, mounted on the high-speed shaft to provide a speed reference. For bearing condition monitoring purposes, the 3701 monitors many vibration-based variables, including direct (overall) vibration amplitude, kurtosis, and crest factor; and it extracts timebase waveforms. It also tracks and extracts specific bearing fault frequency amplitudes from envelope spectra. The ADAPT.wind software has the capability to display trend plots, timebase waveforms, and both conventional and demodulated (enveloped) spectra.

In this case history, we will be discussing enveloped data from the gearbox high-speed shaft (HSS) accel (Figure 1). The diagnostic process utilized the high sensitivity provided by acceleration enveloping, so we will begin with a brief review of the enveloping process.

Bearing Faults and Acceleration Enveloping

Rolling element bearings usually fail by fatigue of the raceways or elements or, in generators, by electrostatic pitting of the raceways and elements. Another failure mode can involve cracking and fracture of the raceway. When a rolling element passes over a defect, the temporary loss of support causes the element to deflect slightly. When the element encounters the far side of the defect, the contact can usually produces a sharp impact to the bearing and support structure, similar to a car driving over a pothole in the road.

This impact causes a ringing of the bearing and support structure at the structural natural frequency. Each time an element encounters the defect, an impulse/response free vibration waveform is produced. The sequence of successive element passages produces a series of impulse/response events that repeat at the bearing defect frequency. If the defect is on an outer race, the events repeat at the outer race defect frequency; if the defect is on an inner race, the events repeat at the inner race defect frequency. Element faults produce defects related to the spin period of the element. All types of defects produce repeating series of impulse/response events that are visible in a timebase waveform.

Inner race faults have an additional characteristic. As the inner race rotates with the shaft, the defect moves from within the load zone of the bearing to outside the load zone, where impacting is less likely to occur. This causes the amplitude of the impulse/response events to increase and decrease with each shaft rotation. It causes amplitude modulation of the impulse/response vibration, and the modulation pattern repeats at the shaft rotational frequency. This modulation is in addition to the defect modulation. Element faults can also produce additional modulation at the fundamental train, or cage, frequency as the damaged element moves into and out of the load zone. Because the outer race of a bearing is usually fixed, outer race faults do not produce this additional modulation.

Bearing defects produce a series of amplitude modulated impulses that are visible in a timebase waveform. This effect is analogous to an AM radio signal, where the content appears as changes in the amplitude (modulation) of a carrier signal. In AM radio, this carrier is around 1 MHz. In wind turbines, the carrier frequency is the structural natural frequency, which for typical machines is in the range of 4 to 10 kHz. Just as an AM radio signal carrier frequency is demodulated to extract the voice or music information, acceleration enveloping demodulates the high-frequency structural carrier frequency to extract the bearing fault repetition frequency.

When properly configured, acceleration enveloping will usually have a much higher sensitivity to faults that produce impacting. The structural natural frequencies "amplify" the vibration resulting from the impact and make them easier to detect.

Acceleration enveloping can be thought of as a four-step process: filter, rectify, envelope, and FFT. The filter band-pass region is set to include the range of structural natural frequencies (3701/60 monitor uses a 4-10 kHz bandpass filter for 1.5 MW turbines). The filtering removes rotor and gear mesh related vibration that could obscure the relatively small bearing fault signatures. After filtering, the signal is rectified, and an envelope is constructed that follows the extremes of the rectified signal fairly closely. The envelope typically has an approximately sawtooth wave shape, where a fast rise at the impulse is followed by a relatively slow decay. The envelope is then processed using a Fast Fourier Transform (FFT), producing a spectrum that is inspected for bearing fault frequencies. Note that the sawtooth envelope waveform shape will often produce a harmonic series based on the fundamental fault frequency. These harmonics are, typically, artifacts of the process and should be interpreted with caution. The fundamental fault frequency (and any sidebands) is usually the primary focus for bearing diagnostics. If an additional source of amplitude modulation is present (for example, an inner race defect where the impulse amplitude is modulated by rotor rotation frequency, or an element defect modulated by cage), an additional spectral line at this second modulation frequency will often be visible. We will see examples of this shortly.

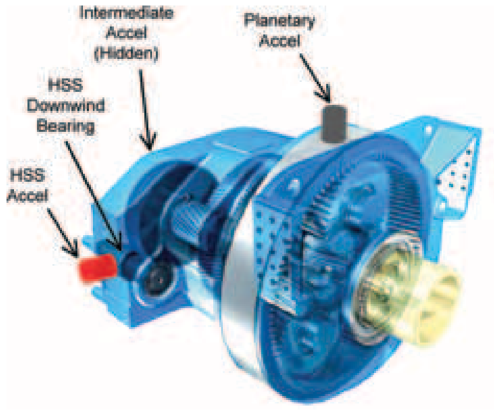


Figure 1: Wind turbine gearbox, viewed from the main rotor side. The high-speed shaft downwind bearing (HSS DW) and accel locations are shown. The high-speed accel location used for this case history is shown in red.

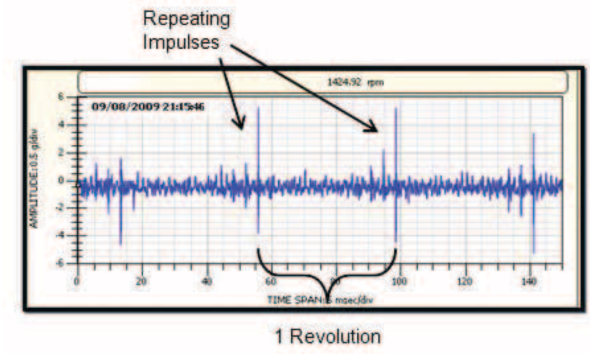


Figure 2: High-speed acceleration waveform from ADAPT.wind software showing periodic impulse patterns. The patterns repeat once every revolution.

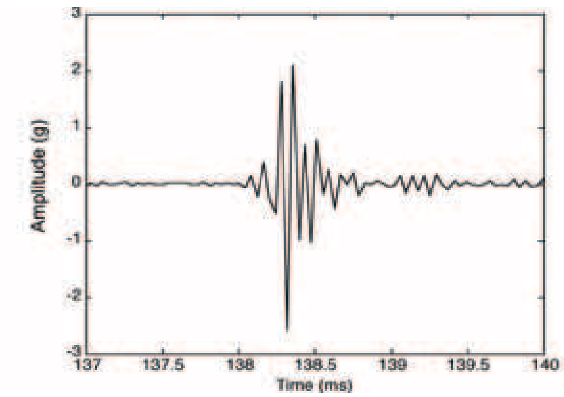


Figure 3: Zoom on a typical spike showing mechanical impulse/ response behavior.

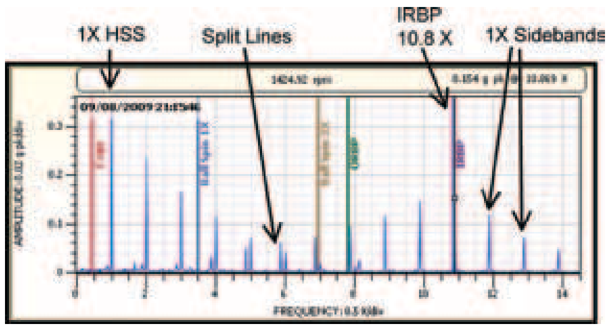


Figure 4: Acceleration envelope spectrum of waveform in Figure 2 (September 2009). The key bearing fault frequency lines are overlaid on the plot. The IRPB fault line is a perfect fit to the data. The split lines show that the two harmonic series are separated. Data processed in ADAPT.wind software.

Initial Detection

After deployment of the ADAPT.wind monitoring systems, some interesting vibration signatures were observed on a few gearboxes. On one, the timebase waveforms from the high-speed gearbox accelerometer showed a repeating pattern of symmetric impulses of very short duration (Figure 2). This waveform was captured in September 2009. The waveform shows a pattern of three impulses that increase and decrease in amplitude. As the inner race rotates through one revolution, the defect moves in an out of the radial load zone of the bearing, producing the long-wave modulation pattern. As it enters the load zone, the defect encounters successive rolling elements (cylindrical elements in this case). Each defect/element encounter produces an impulse in the waveform (the short period modulation pattern). The impulses increase in amplitude until a maximum is reached when the defect is approximately aligned with the radial load direction in the bearing. Then, as the defect begins to leave the load zone, the impulses rapidly die away. This basic pattern is repeated once per revolution of the high-speed gearbox shaft and is highly suggestive of a single, inner-race defect in a bearing on that shaft.

Initially, the extremely narrow, sharp spike appearance in Figure 2 suggested an electrical noise problem on this accelerometer channel. To verify the data quality, several of the "spikes" in the signal were zoomed. Figure 3 shows a zoom from a similar waveform that has been exported and high-pass filtered at 5 kHz using MATLAB® software. The spike is clearly an impulse/response signature indicating mechanical vibration, not electrical noise. The vibration occurs at the "carrier" frequency that will be used for envelope demodulation. The enveloping bandpass filter was set from 4 kHz to 10 kHz, so this 5 kHz high-pass filtered plot shows approximately what the enveloping algorithm would see.

Enveloped spectra from this accelerometer display a rich harmonic series (Figure 4). This figure shows enveloped frequencies in orders of high-speed shaft (HSS) speed. The lowest fundamental corresponds to the HSS speed (1X HSS), and there are several harmonics. The next major feature is the inner-race ball pass (IRBP) frequency at 10.8X. This frequency agrees with the theoretical IRBP defect frequency for the high-speed downwind radial bearing. A series of 1X sidebands are present on both sides of the central fault frequency line. (Most often, sidebands occur in spectra because of amplitude modulation of vibration happening at the center frequency.) The IRBP frequency at 10.8X is close to a harmonic of HSS speed at 11.0X, so at first glance, all the lines in the spectrum appear to be equally spaced harmonics of 1X. Close inspection shows that there really are two distinct harmonic series, one series based on 1X, and the other based on 1X sidebands from 10.8X. You can clearly see that the two families of spectral lines nearly overlap, and that separation is visible in the split lines near 5X and 6X.

The very rich spectrum occurs because of the extremely sharp impulses in the original waveform in Figure 2. The theoretical spectrum of a single, sharp, periodic impulse is an infinite harmonic series of narrow lines starting at the fundamental repeat frequency. In this case, we have a more complicated repetition pattern, but the sharp nature of the repeating impulses produce a similar effect in the envelope spectrum.

There are two sources of amplitude modulation in the waveform. The 10.8X line is obviously related to the inner race defect. This defect causes amplitude modulation of the waveform at the element pass period (the relatively small impulse spacing in Figure 2). The 1X line is caused by relatively large scale amplitude modulation as the inner race rotates in and out of the load zone. Both sources of modulation are present, so both modulation frequency lines appear in the envelope spectrum, along with their harmonic series.

The monitor also calculates the kurtosis of the waveform. This is the fourth statistical moment of the waveform which emphasizes extreme values, so spikes in amplitude will tend to increase the kurtosis. The kurtosis of a pure sinusoidal waveform is 3. Because of the impulse/response behavior, the kurtosis of the waveform in Figure 2 is around 150.

Given the data so far, we concluded that a single defect existed on the HSS downwind radial bearing inner race. The extreme narrowness of the impulse/response signatures led us to suspect that the defect was a crack in the inner race; we reasoned that a broader spall would likely have produced a less well-defined impact signature. But at this time, because the overall amplitude of vibration was only 5 g, we recommended that the operator wait and watch. This would allow us to see how quickly the damage evolved. If it increased significantly, we would recommend shutting down the turbine.

More Data Arrives

For a few months after initial detection, the vibration signature remained relatively unchanged. However, when we looked again in April 2010, the vibration waveform had become more complex (Figure 5).

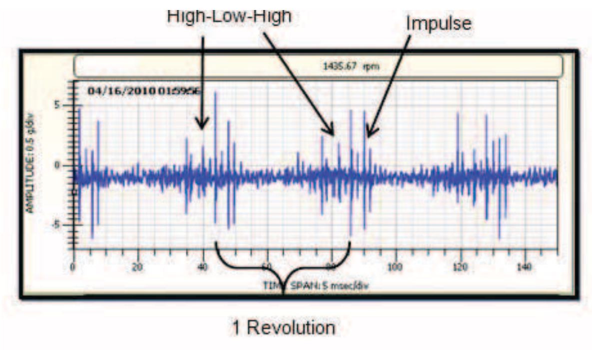


Figure 5: Data from the same bearing as Figure 2, but several months later. The vibration signature has become more complex.

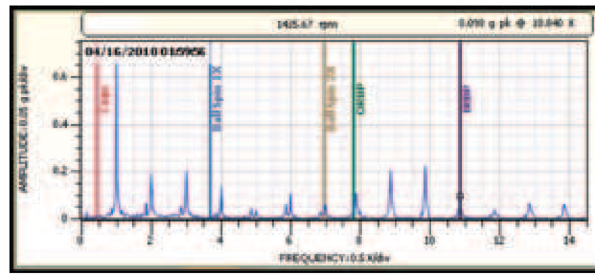


Figure 6: Data from April 2010. Compare to Figure 4. The amplitude of the IRBP line is lower, while the 1X line amplitude is higher.

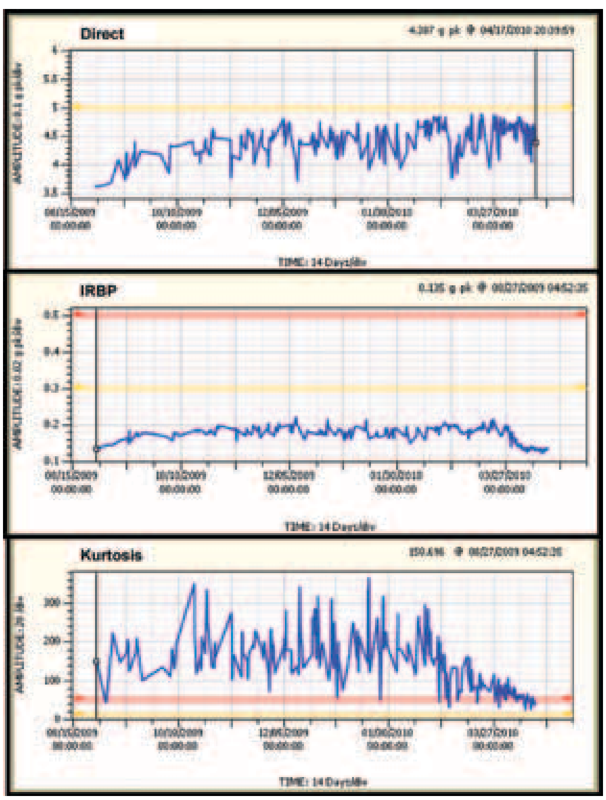


Figure 7: Trend plots from August 2009 to April 2010. Direct vibration (top) increases, but IRBP frequency amplitude (middle) and kurtosis (bottom) decrease near the end of the period.

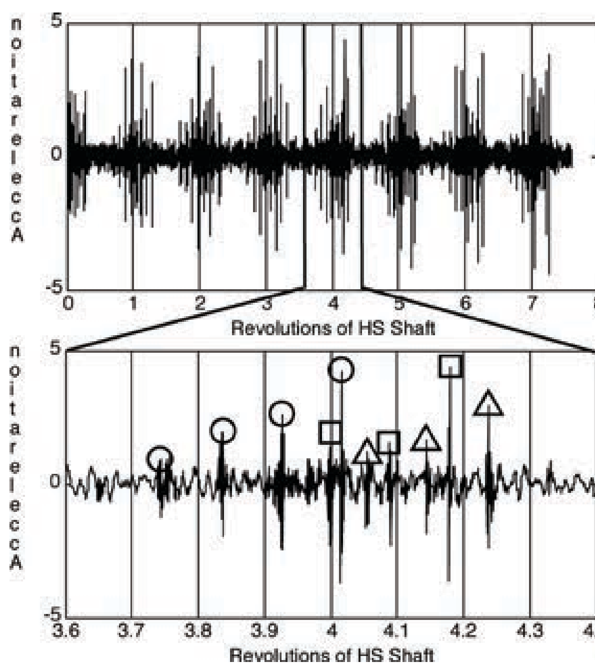


Figure 8: HSS accelerometer waveform from the second turbine gearbox. The circular, square, and triangular marks highlight three distinct patterns of fault induced impulses. This pattern suggests at least three separate inner race faults.

Although the amplitude of vibration remained about 5 g, there were two primary differences between the data in Figure 5 and the data in Figure 2. First, the impulse pattern in Figure 5 shows a repeating high-low-high impulse pattern with each revolution. This differs from the steadily rising amplitude pattern in Figure 2. Secondly, there is another set of lower-amplitude impulses that are offset by less than the element passage period from the larger set. This "double impulse" set appears at the same place every revolution. These changes suggest that more damage has developed on the inner race; the second change suggests that some of the additional damage is smaller than the original damage we observed before.

The envelope spectrum also shows some interesting changes. The 1X line has *increased* in amplitude, while the IRBP line has *decreased* (Figure 6, compare to Figure 4). The increase in 1X modulation has also increased the IRBP 1X sideband amplitude. This is interesting because it shows that more damage can actually reduce the IRBP amplitude.

Trends of direct vibration amplitude, IRBP amplitude, and kurtosis also changed over this period (Figure 7). Direct vibration slowly increases; this is a measure of the impulse amplitude we see in the timebase plots. The middle plot is the amplitude of the IRBP line in the envelope spectrum; it is approximately constant until it drops off near the end of the period. The kurtosis also declines near the end.

While the decline in IRBP is somewhat puzzling, the decline in kurtosis is caused by the larger number of impulse events in the waveform; in other words, the average signal excursion has increased. This reduces kurtosis, which is very sensitive to outliers. When everything is an outlier, then the outlier becomes the norm.

The impulses appear to form three distinct repeating groups (these are marked in the lower figure with circles, squares, and triangles); in each group, impulses are separated by the IRBP fault period (1/10.8X = 0.093 revolutions) for the HSS downwind radial bearing. All of the impulses disappear and reappear once per revolution of the high-speed shaft. We concluded that at least three different faults on the inner race of the bearing were repeatedly passing through the bearing load zone. Note that the impulse amplitudes are also 4 to 5 g. Because there were so many indications of damage, we felt that there was a very good chance that this bearing would have visible damage that could confirm the diagnosis. This bearing is located in an accessible part of the gearbox, so the operator decided to replace this bearing, work that could be done up tower without a crane.

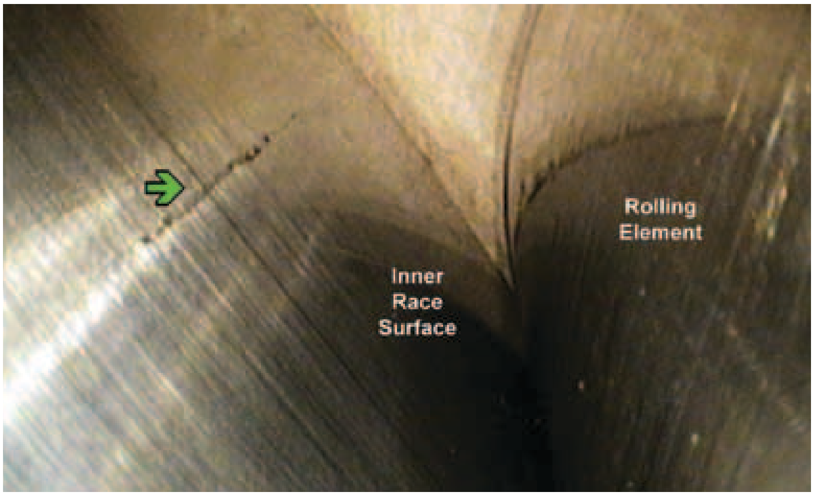


Figure 9: Crack in inner race of the high-speed downwind radial gearbox bearing (arrow). This was the largest of four race cracks found.

The impulse signatures still remained very narrow, and we suspected that a second crack had developed in the inner race. Still, at this time, overall vibration remained only about 5 g, so a decision was made to keep watching.

Another Turbine Exhibits Similar Behavior

In April 2010, another turbine was commissioned. This turbine was identical in every way to the first one. The HSS accelerometer data showed similar behavior to the first turbine (Figure 8, upper). This waveform shows a similar, very complex series of very narrow impulses.

In May 2010 the bearing was replaced, and the original was inspected. We discovered four radial cracks in the inner race that extended some distance axially across a significant part of the race (Figure 9 shows the largest of them). None of these cracks had propagated all the way through the race, so there was some remaining useful life in the bearing.

Summary and Conclusions

The diagnosis of a cracked inner race was based on the following evidence:

- A clear pattern of modulated impulse/response signatures in the timebase waveform. This was consistent with an inner race bearing defect.
- IRBP bearing defect and 1X modulation frequency in the envelope spectrum. This confirmed the inner race fault, and the IRBP frequency identified the offending bearing.
- Very narrow impulse/response signatures in the timebase waveform suggested a narrow defect.
- A second, identical turbine exhibited similar symptoms on the same bearing. The bearing was removed and inspected, and four inner race cracks were found. Because of this confirmation, we are now confident that the original turbine has the same problem.

Trend plots of the first turbine data showed interesting behavior. While direct vibration increased at first and remained approximately constant, IRBP fault frequency and kurtosis amplitudes declined as damage progressed. These two measures appear to be most sensitive in the very early stages of bearing failure. Kurtosis is most sensitive when only one defect is present, as it is more of a statistical outlier. The reduction in IRBP amplitude may have been caused by an increased number of lower amplitude impulse events. This effect also caused an increase in the 1X modulation line amplitude in the spectrum. This shows the importance of using different kinds of data to monitor machine behavior. You cannot simply assume that all measures will increase with increasing damage!

In the first turbine, it took several months to evolve from a single defect to multiple defects, and given the bearing inspection results in the second turbine, there are probably at least several months more remaining life for this bearing. The bearing can remain in service, be monitored, and a convenient time can be scheduled for replacement.

The very close proximity of the sensor to the bearing probably made this diagnosis easier than it might have been if the transducer were mounted farther away. When farther, vibration propagation and reflection can muddy the impulse signatures in the timebase plot. Fortunately, the sensor was mounted close to the bearing, making this diagnosis more clear-cut. The diagnosis of these defects as cracks depended on both timebase and envelope spectrum information. Spectrum is, in one sense, an averaging process; the values displayed are average values over the entire waveform. Short-term time information is lost, and the timebase plot supplies that information, making the combination of the two plots more powerful than either one alone. The timebase plot also allows a cross check to check on the quality of the data.

This case has shown that it is possible to detect bearing race cracks at a very early stage, providing plenty of time to monitor and schedule maintenance. Timely warning is a key part of optimizing economic performance of a wind farm. ■

3500 ENCORE* Series

Bently Nevada Asset Condition Monitoring

New Upgrade Path for Condition Monitoring

As this issue of *ORBIT* goes to press, we're pleased to announce the 3500 ENCORE Series Machinery Protection System. The 3500 ENCORE is your most cost effective, non-intrusive means of upgrading your existing 3300 monitoring system to the state-of-art condition monitoring technology, thus, extending the life of your investment while maintaining best-in-class protection of your production assets.

The 3500 ENCORE Series upgrade consists of removing the original 3300 modules and replacing them with modified 3500 modules. The resulting system provides the improved performance of a 3500 Monitoring System while retaining the transducer, Keyphasor* and relay field wiring from the original 3300 system.

* denotes a trademark of Bently Nevada, Inc., a wholly owned subsidiary of General Electric Company.

For more information please visit our 3500 dedicated microsite at

www.3500ENCORE.com

Delayed Coker Drum Monitoring

An innovative operator feedback solution that reduces coke cutting risks and costs

Many petroleum refineries employ the delayed coking process to crack heavy residual oils into lighter hydrocarbons, leaving behind solidified petroleum coke as a by-product. This solidified "pet coke" must then be removed from the coke drum using a high-pressure water jet.

The Problem

Operation of the water jet is a manual process, carried out by a Cutting Operator. Cutting Operators typically do not have instrumentation to guide them, making it difficult to know when a layer of coke has been cut from the walls of the drum and the water jet should be repositioned. Instead, they rely on simple sensory feedback (such as the sound of the water jet as it nears the drum wall) and their subjective experience interpreting this sensory feedback.

This leads to two significant drawbacks:

- **Elevated Personnel Risks**. When forced to rely on sight and sound alone, Cutting Operators can be more inclined to exit their protective cab to better see or hear the progress of the cutting jet. This removes them from the cab's protective environment.
- **Elevated Costs**. Even the best Cutting Operator will be less than optimal when forced to rely on subjective evaluation of his own senses. As a result, cutting operations will take longer than necessary, reducing coker unit throughput and wasting energy. In addition, longer cycle times expose the coker assets to a lengthier duration of the elevated temperatures and pressures inherent to the process. Thus, asset wear and tear per batch is increased, driving maintenance costs and downtime-related costs upward.

Further exacerbating the problem, these two drawbacks pit themselves against one another. A Cutting Operator can increase cutting effectiveness (and thereby reduce costs) by periodically exiting the protective cab to get better feedback on the cutting process. What is needed is a better method of feedback—one that keeps personnel inside their protective environments while maximizing cutting effectiveness.

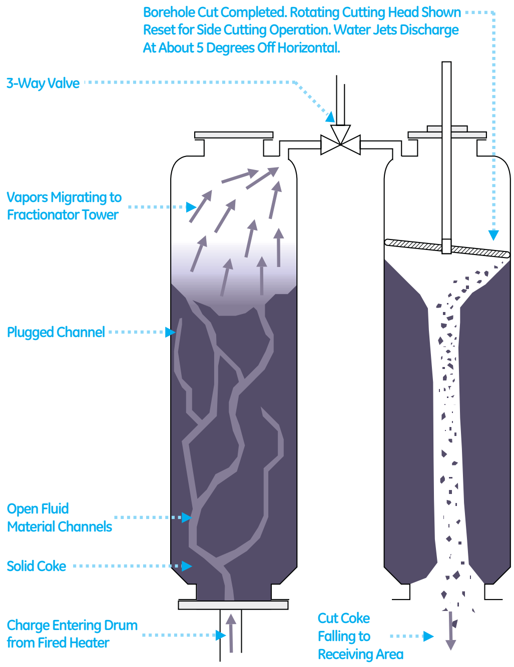
The GE Solution

Pioneering efforts by GE to provide more accurate operator feedback in the coke cutting process centered on the use of structural vibration measurements. Extensive testing showed that overall vibration amplitude measurements from accelerometers mounted at appropriate locations on the drum were a reliable and repeatable indicator for controlling the coke cutting process. Thus, by watching vibration levels during the cutting process, the operator can accurately determine when the coke has been completely removed, allowing him to reposition the water jet to the next area of the drum neither too soon nor too late.

The Delayed Coking Process and Coke Cutting

Following the fractional distillation of crude oil in a refinery, a certain amount of so-called "bottoms fractions" remain including tar, paraffin wax, asphalt, heavy residual oils, and others. Further processing must be performed to extract higher-value hydrocarbons from these lower-value "bottoms."

Delayed coking is the most widely used process for cracking one of these bottoms—heavy residual oil—into lighter hydrocarbon chains. The delayed coking process first superheats its heavy residual oil feedstock and then moves it into an insulated and vertically mounted pressure vessel known as a coke drum. After the vapors from the super-heated residual are extracted and further refined, a high-density hydrocarbon material known as petroleum coke is left behind in the drum. As the coke cools, it hardens and must be removed from the drum using a water jet cutting process. The jet is first used to bore a hole in the center of the solidified coke "slug". The cutting jet is then repositioned for side-cutting operation where it breaks the coke into smaller chunks that fall through the bore-hole. The removed coke is then used in additional carbon-based byproducts.

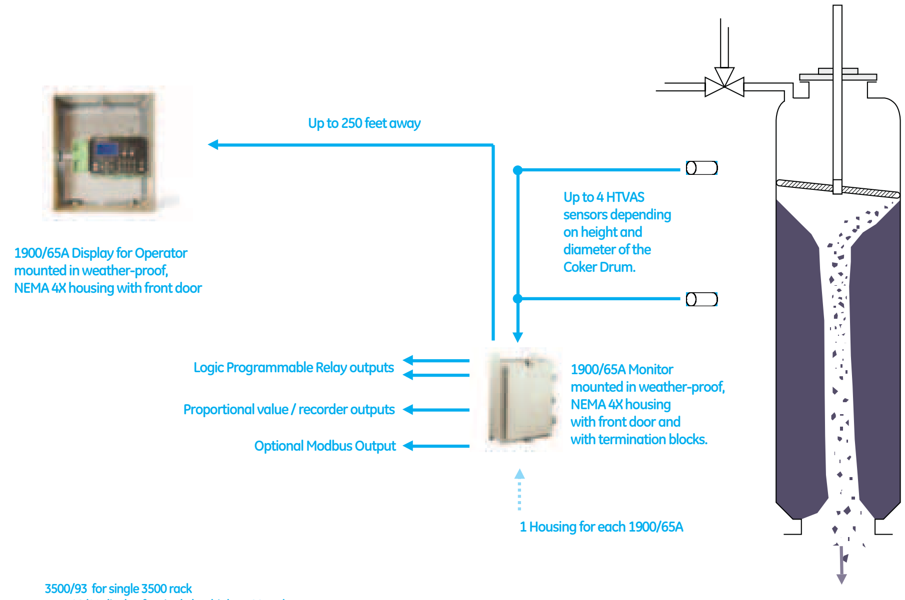


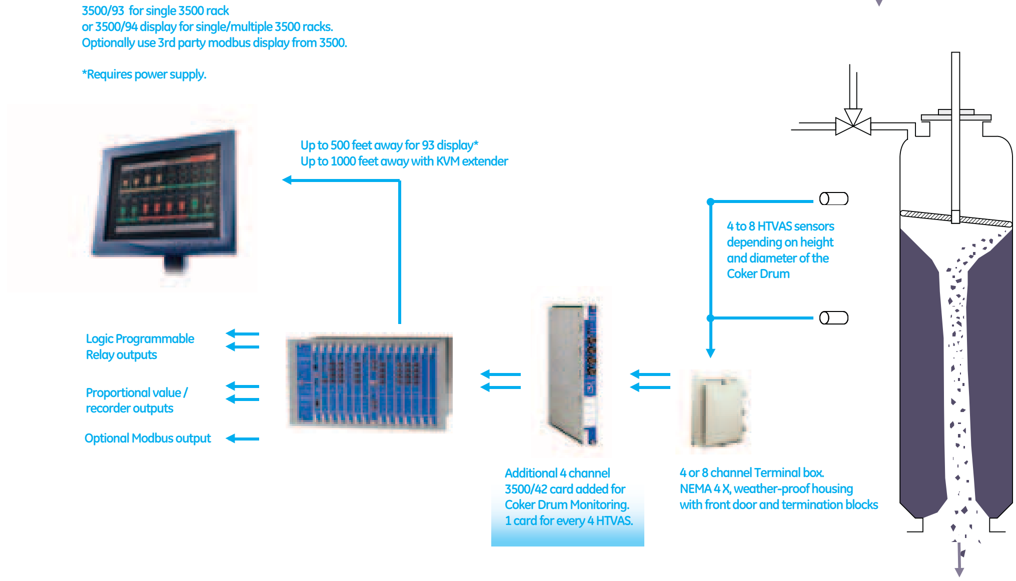
GE's Bently Nevada Delayed Coker Monitoring package provides customers with the necessary hardware and installation service options to instrument one delayed coke drum, providing operators with real-time status of the drum cleaning process. Depending on height and diameter, a coker drum requires anywhere from 4 to 8 high-temperature accelerometers, mounted on either the drum's outside wall or its top and bottom flange necks. The accelerometers are connected to an associated monitoring system that measures overall vibration amplitude and displays it in the Cutting Operator's protective cab via a variety of remote indication options.

Customers have the choice of a 3500-based system or a 1900/65-based system (refer to diagrams on following page). The 3500-based option is intended for customers with existing 3500 installations and adequate spare rack capacity for the requisite additional cards (one 3500/42 card is required for every 4 accelerometers). The 1900/65 option is intended for customers without installed 3500 systems, or where use of existing 3500 systems would be impractical due to wiring length limitations and/or costs. Both options utilize the same underlying monitoring methodology and provide the same basic vibration information to the Cutting Operator.

Benefits

- **Reduced exposure to hazardous environments**. Cutting Operators remain where they belong—*inside* their protective cabs—while receiving high-quality information necessary to control the cutting process.





- **Improved coker throughput and energy usage**. Consistent, repeatable measurements allow operators to minimize cutting time, enhancing coker throughput and conserving energy.
- **Increased productivity**. Vibration data removes much of the subjectivity inherent in sensory-based control of the coke cutting operation, resulting in less variation from one shift to the next. Every operator can now consistently minimize cutting time through use of a common, repeatable reference.
- **Enhanced profitability**. More efficient cutting operations reduce the time that coker assets are subjected to temperature/pressure extremes for each batch processed. As a result, asset wear and tear per batch is reduced, improving asset availability, reducing downtime, and ultimately increasing refinery profitability.

Learn more by contacting your nearest GE sales professional specializing in Bently Nevada Asset Condition Monitoring solutions or by visiting www.ge-energy.com/bentlyapplications. You can also request hardcopies of relevant literature via the Reader Service Card in this issue of ORBIT.

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System 1* at Agrium: Nutrient for Growth

Fertilizer sustains world food and energy demands. At Agrium, a leading global producer and marketer of agricultural nutrients and industrial products, System 1* is making a positive impact on operation and maintenance, helping feed the production demand.

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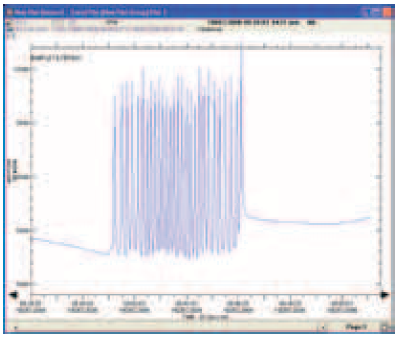
Agrium Inc. is a major retailer of agricultural products and services in North and South America, a leading global wholesale producer and marketer of all three major agricultural nutrients, i.e. nitrogen, phosphate and potash, and the premier supplier of specialty fertilizers in North America. Agrium's headquarters is located in Calgary, Alberta, Canada, with four major production facilities in Alberta. At all of these facilities, critical process equipment is equipped with Bently Nevada condition monitoring systems. Since 2005, GE's System 1 online condition monitoring and diagnostics software has been implemented. Reliability engineers at these facilities actively use System 1 tools and data to monitor and troubleshoot machinery malfunctions. System 1 enables operations and reliability personnel to make decisions to continue production or increase process rates without potential damage to the machines, as well as determine outage scope when necessary. Based on System 1 information, a predictive maintenance program is implemented to maintain equipment in good condition and prevent costly replacements. As explained by Gerry Kydd, senior specialist of rotating equipment at Agrium Technical Services, "System 1 saves us money. It allows us to know when a machine is in real stress and to monitor closely while keeping it running until we can shut it down in a controlled way. System 1 also allows us avoid secondary damage to the process that can occur in an emergency trip." The following case histories demonstrate how Agrium achieved these savings.

CASE HISTORY 1:
Steam Turbine Overspeed

Agrium Fort Saskatchewan Nitrogen Operation produces 465,000 gross tons of Ammonia and 430,000 tons of Urea per year.

In March 2005, the overspeed protection system on a steam turbine driven process air compressor was inspected; the trip bolt cleaned and overspeed condition tested. In December 2006, during a process upset, a maintenance engineer noticed in System 1 that the speed momentarily spiked to 10335 rpm, well above the rated trip speed of 9523 rpm. Due to slow data capture intervals, there was no indication of the overspeed condition in the DCS. The overspeed protection trip bolt didn't act. The maintenance engineer alerted operations of this anomaly. At the next available outage in June 2007, the trip bolt parts were examined, and were found to be contaminated with thick, sticky sludge that prevented the mechanism from functioning properly. Thorough cleaning and testing ensued and the steam turbine was overspeed trip checked successfully. In June 2008, the turbine was converted to an electronic overspeed trip. If the overspeed trip device malfunction had not been detected, the business impact for potential catastrophic machinery damage and process interruption was estimated at $30 million in this single incident. System 1 paid for itself by providing crucial data in an otherwise unnoticed overspeed condition. Even if the probability for such incident were 1 in 100, the cost of the risk would be $300,000.



The turbine speed spiked to 10335 rpm in December 2006, above the rated trip speed.

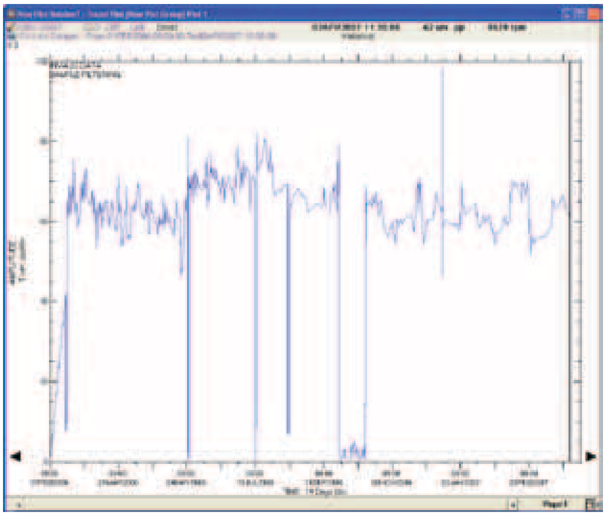
Sludge in trip bolt mechanism.

Process air compressor, compressor train.

1

CASE HISTORY 2:
Step Changes in Exhaust Bearing Vibration

The maintenance engineer uncovered another malfunction of the steam turbine in case history #1 in February 2006. After an outage, the radial vibration on the exhaust bearing step-changed from about 40 micron to 60 micron, and then went as high as 80 micron occasionally. During the outage of June 2007, the turbine was run solo as part of the overspeed test. The shaft centerline plot from the solo test rundown was compared to that of the coupled rundown before the outage, and distinctive differences between the shaft centerline movements and vibration levels were observed. It was concluded a significant component of the vibration was caused by internal and external misalignment. Turbine alignment keys and supports were lubricated with Molykote® 321 in June 2007, and vibration decreased from 60 microns to 50 microns. In the outage of July 2008, the plant replaced the coupling and re-aligned the turbine and compressor. After the outage, vibration dropped and settled out at 40 microns. As explained by the maintenance engineer, "System 1 allowed us to see dynamic response of the turbine in real time. Although the 40-micron level is higher than desired, it is acceptable, but the 80-micron level is just a step away from a shutdown and overhaul. The potential consequence of such a failure would be an unplanned outage at $5 million lost production. Even if the probability of this failure were 1 in 10, from this single incident we saved $500,000."



Vibrations were around 40 microns prior to February 2006 outage; the vibration jumped to 60 microns on the start up and went as high as 80 microns.

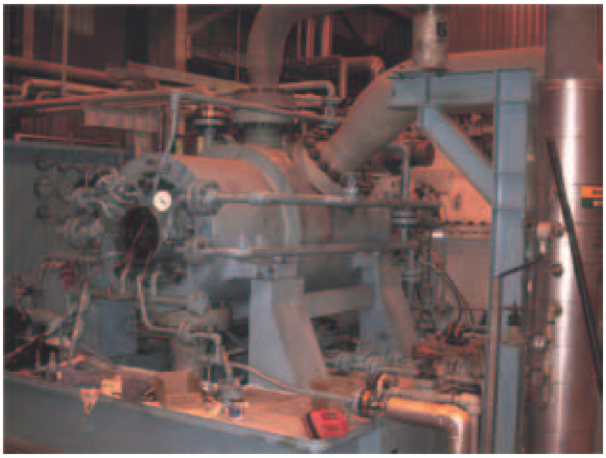
CASE HISTORY 3:
Thrust Bearing Damage by Electrostatic Discharge

It has been well documented that electrostatic discharge can gradually destroy bearings when arcing erodes the babbitt metal surface. This case provides new insights into the failure mode of thrust bearing damage caused by electrostatic discharge. Explained by Gerry Kydd, "The tilt pad thrust bearing that was experiencing electrostatic damage showed hardly any axial displacement even though the amount of babbitt erosion from the pads was severe. Temperature measured by embedded probes was actually giving an earlier indication. It turns out that the bearing pad can tilt progressively more and thus maintain axial positioning until the damage to the pad trailing edge is very severe. You may only get a couple of mils movement and think you are more or less OK, but it is misleading. At the point the axial position starts to move, the bearing pads may actually be badly damaged and starting to wear into the thrust disc. If the disc is integral with the shaft then expensive rotor replacement will be needed. For this failure mode, both temperature and thrust position are required for the diagnosis and this is something that I had not previously seen in the literature."

Joffre Nitrogen Operation produces 480,000 tons of ammonia per year. The heart of the process is synthesis gas compressor. The thrust bearings in the compressor high-pressure case are furnished with proximity probes and embedded temperature probes. At the beginning of November 2008, an apparent upward trend of bearing temperature was noticed at the inboard thrust bearing, but the outboard bearing temperature remained stable. In the previous three months, the inboard bearing temperature increased from 90C to 120C. During the same time period, the axial position measured at the inboard bearing only moved 3 mils. The orbit-timebase plot for inboard radial bearing vibration was examined and no distinct changes noticed.

The plant maintenance technician was concerned about possible thrust bearing damage, so an outage was planned. On November 17, 2008, the compressor was shut down. The inboard thrust-bearing pads had over 0.020 inch Babbitt removed from the trailing edge and were down to the steel backing. The root cause was electrostatic discharge. Consequently, grounding brushes were repaired and a back up brush installed. Since then, it has kept the problem from re-occurring. In this case, the plant maintenance team shut down the compressor on the strength of both System 1 data and DCS data. The damage was limited to only the thrust bearing and no other rotating or stationary parts were affected. The maintenance cost of the planned outage was minimized compared to emergency shut down at an inopportune time. System 1 saved the plant a potential loss of $250,000.

It is intriguing to note that the thrust movement, less than 5 mils, was far less than amount of the thrust bearing pads erosion, over 20 mils. Electrostatic damage showed up predominantly as a bearing temperature excursion. The change of bearing temperature was more indicative of damage than that of bearing axial position.



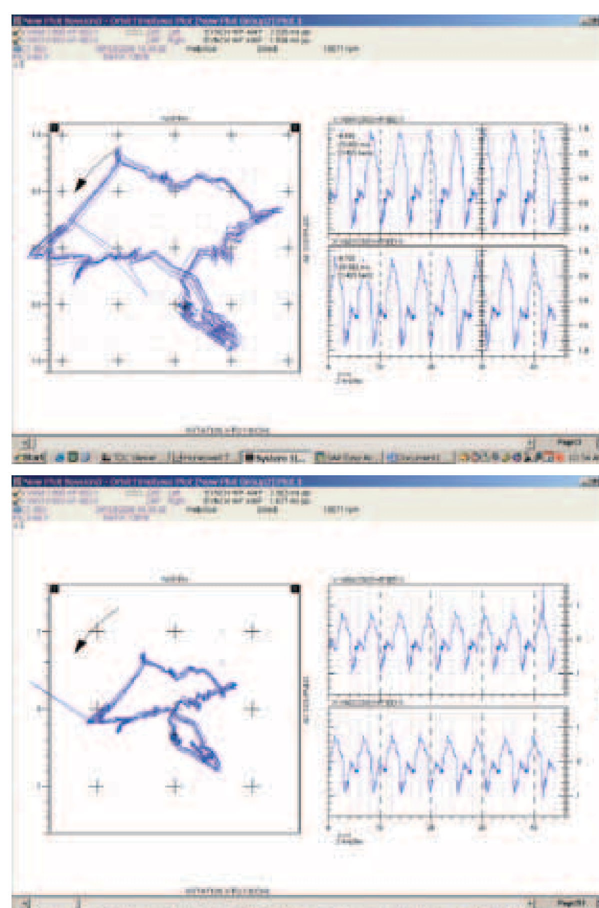
Synthesis Gas Compressor, High Pressure Case.

CASE HISTORY 4: Radial Bearing and Thrust Bearing Damage by Electrostatic Discharge

Redwater Nitrogen Operation is Agrium's largest nitrogen producer. It produces 960,000 gross tons of Ammonia per year, as well as Ammonium Nitrate, Ammonium Sulphate, Urea, and Nitrogen Solutions.

The synthesis gas and ammonia refrigeration compressors at the Redwater operation have experienced grounding problems from time to time, as static charges build up from impinging droplets of water in the wet stages of the steam turbines. In February 2008, the rotating equipment specialist at the plant observed erratic orbit patterns on both compressors and a slight increase in temperature of the syngas turbine thrust bearing. The random abnormal shaft vibration could not possibly represent any actual shaft movement. The enlarged orbit on the refrigeration compressor also indicated inboard bearing clearance had opened up. Electrostatic discharge was considered to cause the spikes. After reviewing System 1 data, the plant had a planned outage for other reasons in April and used the opportunity to inspect the syngas turbine thrust bearing and refrigeration compressor radial bearing. The grounding brushes were found badly worn on both machines. Severe surface erosion was found on the pads of the syngas turbine thrust bearing and some wear on the refrigeration compressor radial bearing leading to replacement of both. When the compressors returned to service, no more abnormal orbits were observed. The abnormality in the orbits was attributed to the radio frequency interference from the static discharges occurring in close proximity to the radial vibration probes. Accordingly, the plant revised the preventive maintenance program to inspect and replace the grounding brushes on a monthly interval or as necessary based on System 1 data. The static grounding brush maintenance has kept the bearing conditions stabilized. The plant engineer concluded, "System 1 allowed us to stay on top the situation until we could shut down and replace the thrust bearing. In some of our past experience, we have found that static discharge can destroy a bearing in only a few months. Even though System 1 was not designed to monitor static discharge within a piece of equipment, it is very beneficial to pay attention to any anomaly in the data. If we had not noticed this increase level of static discharge in our two machines, we would not have been able to run them to our 2009 turn around." A premature shut down could cost the plant many millions of dollars in production losses.

Similar to Case history #3, the syngas turbine thrust bearing experienced substantial damage, while the thrust transducers didn't show any significant axial movement. Again, the thrust bearing temperature also increased indicating the damage.



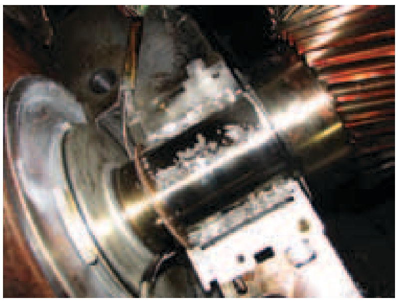
These random anomalies of the turbine vibration could not possibly represent any actual shaft movements. These were at the turbine HP casing radial bearing closest to the thrust bearing. Some anomalies still appeared at the bearings on the opposite end of the machine, quite a distance from the discharges themselves.

CASE HISTORY 5:
Air Compressor Oil Pump Failure

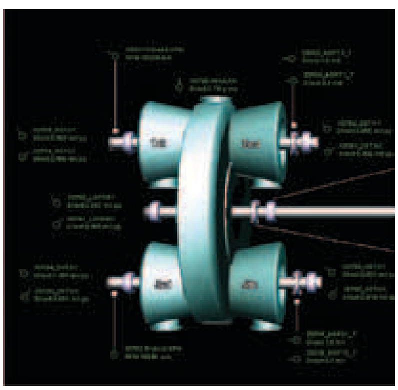
Carseland Nitrogen Operation produces 535,000 gross tons of Ammonia and 680,000 of Urea annually. System 1 was commissioned on critical equipment in 2008.

5

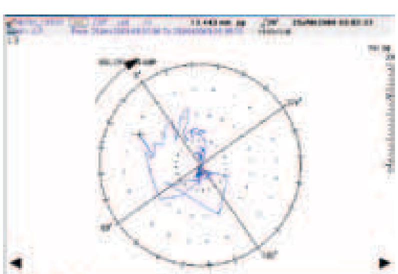
System 1 proved its value at 5AM Sunday, January 25, 2009. The main lube oil pump failed, and a few minutes later, the integral-gear air compressor tripped on high thrust vibration. The plant reliability engineer was alerted to the incident and quickly reviewed System 1 data. High axial movement was revealed at the thrust bearing on the high-speed shaft, starting about 20 seconds before tripping from losing pressurized lube oil. The high axial movement caused serious damage to the thrust bearing. First stage shaft radial bearings also experienced high vibration before and during the trip. The dominant radial vibration component was at running speed, 1x. Radial bearing damage was also found. To support the diagnostics, plant engineers reviewed DCS data, including data trends of thrust bearing temperature, lube oil pressure and running speed. Without this wealth of data, the likely scenario would have been restarting the compressor. But, based on data, a decision was made to remove the compressor cover for inspection and repair. Severe bearing damage and impeller damage was observed and a complete overhaul of the compressor was undertaken. The replacement parts cost more than $200,000 in material and labor, and it took more than a month to put the compressor back in service due to unavailability of some required parts; while the plant ran at reduced ammonia production. After this incident, several corrective actions were taken. The lubrication oil pump operation procedure was revised and an accelerometer installed on the main oil pump to catch pump faults early. The plant reliability engineer was convinced of the System 1 value in the unexpected trip situation. The trip data collected by System 1 proved to be extremely helpful in diagnosing the trip. Also, machine related parameters were in System 1, enabling diagnostics to be carried out in a timely fashion. The reliability engineer commented "as the compressor overhaul was a pretty big and costly job, it was crucial to make the right decision whether go forward and overhaul the compressor, or run it up based on uncertain thrust movement threshold setting. If the compressor had been restarted, it is inevitable that even higher production losses and repair costs would have occurred. Having data available in System 1 helped save money in terms of maintenance time, as we knew what was wrong and what parts to order."



Air Compressor High speed shaft thrust and radial bearing damage required a complete overhaul.



Motor driven 4 stage Air Compressor



Air Compressor 1X Vibration Polar plot just after failure from 5:00.

Summary

The cases presented here illustrate that System 1 provides critical data and diagnostic capabilities to Agrium facilities to detect and monitor equipment problems, make informed decisions for scheduling outages at the right time and carry on preventive maintenance programs based on equipment condition. ■

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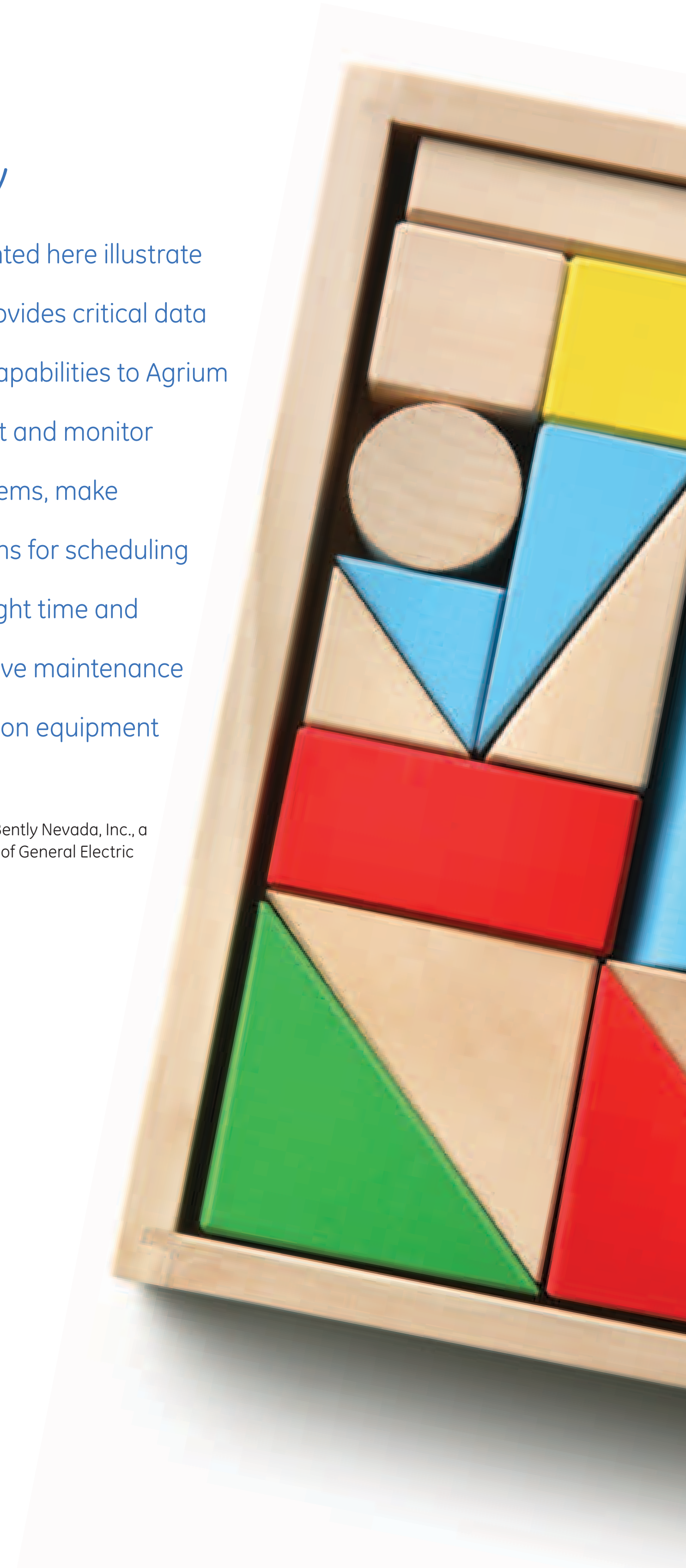

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GEA 18357