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THE ROLE OF ONLINE AERODYNAMIC PERFORMANCE ANALYSIS

by

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ABSTRACT

Continuous online aerodynamic performance monitoring of turbomachinery is crucial to plant operation and maintenance and is a key part of a thorough machine reliability program. Integrating machine real-time performance along with mechanical parameters (vibration data, thrust, bearing temperatures, and oil condition) makes for an all-encompassing condition-based maintenance program resulting in better maintenance and reliability decisions. Case studies of machine problems and the need to extend the time between overhauls are discussed. The benefits that are realized from implementation of online performance monitoring are presented.

INTRODUCTION

Just like racing driver Jimmie Johnson strives to keep his car in top condition in order to keep out in front of the pack, every plant manager must work hard to keep his plant in top operational condition. It is that little bit of added performance that is the cream, the profits that make the difference between surviving and thriving in today's competitive economic climate.

While a good, effective maintenance program is not free, and the cost of maintaining that program cuts into the bottom line, the added revenue from high onstream factors more then offsets this cost. The goal is to have a low-cost program with real benefits. Instant feedback on information with minimal intervention is key. Basing a maintenance program on equipment condition rather than operating time will go a long way toward saving money.

While there is always the risk of too much information, knowing how the equipment is running is just as important as knowing plant profits on a real-time basis. Online performance monitoring of critical turbomachinery equipment adds value by knowing instantly when something goes wrong or is starting to go wrong, and the data provided helps scheduling maintenance and troubleshooting for root cause. Getting to problems quickly cuts losses and adds to the bottom line. And, if the equipment is running fine, then why spend the time and money to open it and inspect/replace all the wearing parts?

MAINTENANCE AND OPERATIONAL IMPROVEMENTS

In addition to confirming you are getting what the equipment manufacturer promised you would get, online performance monitoring will give you instant feedback for maintaining peak efficiency. While there are other indirect means to monitor compressor efficiency, directly calculating efficiency is the best method to assure the problem is in the compressor and not somewhere else. Monitoring the performance can help with maintenance scheduling such as cleaning (Figure 1) and scheduling a major turnaround. When it comes to considerations of a rerate for increased capacity, the current and historical data will be invaluable in making the decision on how to expand.

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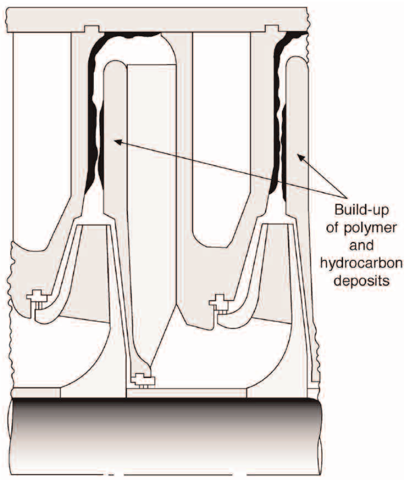

Figure 1. Polymer Buildup in the Diffuser Passage of a Cracked Gas Centrifugal Compressor. Online Monitoring Shows the Rate of Decay in Performance and Aids in Scheduling Maintenance Such as Spray Washing or Disassembly for Manual Cleaning. (Courtesy Gresh, 2001)

Plant operators are constantly making adjustments to improve the plant processes. Knowing the effect on the compressor via instant feedback from online performance monitoring can expedite their work immensely saving valuable man-hours.

Performance monitoring can help with mechanical as well as efficiency issues. Some very basic information gathered from performance monitoring like where the compressor is operating on the performance curve can help prevent impeller failures or at lease help in troubleshooting efforts. Documenting surge events and choke events are vital in finding the root cause of a failure and provide the data necessary to change procedures for preventing future failures.

Startups and other transients like trip situations can lead to severe upsets that may not show up as a significant event on the vibration monitors but may push the compressor into deep choke or surge, and unless online monitoring is implemented, will go unnoticed (Figure 2). Improper startup of a refrigeration compressor can result in liquid ingestion and damage to the compressor internals especially the impellers. Normal performance testing will not show these phenomena since a performance test is typically conducted during stable, steady-state conditions.

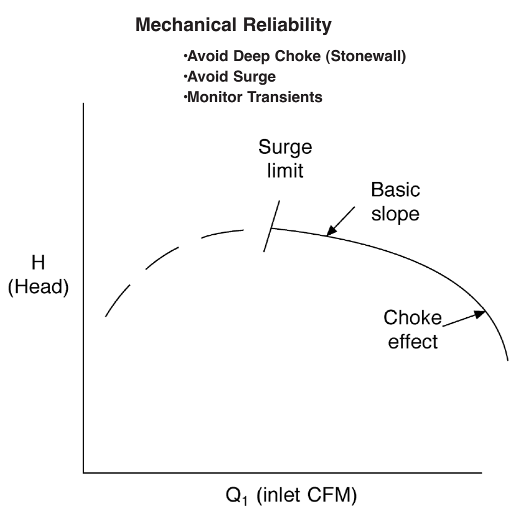

Figure 2. Monitoring and Documenting Where the Compressor Is Operating on its Performance Curve Can Enhance Mechanical Reliability. Damage to Equipment Is Known to Occur in the Choke Region as Well as in Surge. (Courtesy Gresh, 2001)

Online Performance Monitoring System

Online performance monitoring utilizes compressor operating data and compares the processed results to the manufacturer's expected performance to determine the compressor health. Pressures, temperatures, speed, and flow rates are processed to determine operating work input, head, and efficiency (Tables 1 and 2). These calculated results are then displayed on a chart (Figure 3) and compared to original equipment manufacturer (OEM) expected values. A historical data log (Figure 4) of raw data and calculated results is maintained for use in determining maintenance schedules and troubleshooting problems.

Table 1. Input Data for GB1701 Methanation Compressor Pulled out of the Plant Historian Software. Input Data Then Moves Through the Spreadsheet below and Then to the Calculation Engine. Table 1 below Shows Live Compressor Information Being Received and (Table 2) the Calculation Results.

GB1701 – Methanation Compressor – 02/24/2006

GB1701

Input	Tag #	Last Update Time 02/24/2006 8:56:08	
Inlet Pressure	PI17005	321.3	PSIA
Inlet Temperature	TI17056	532.2	F
Discharge Pressure	PC17004	379.6	PSIA
Discharge Temperature	TI17049	573.7	F
Speed	SI17601	4488.4	RPM
Inlet Volume	FC17005	26227.7	ACFM
		18.0	MMSCFH

Table 2. Output Data for GB1701 Methanation Compressor.

GB1701 **Output**

	Compressor Data
Date	02/24/2006
Time	8:54:00 AM
Inlet flow (ACFM)	26292.30
Mass flow	12751.49
Head	16217.00
Speed	4481.70
Gas power	7977.00
Mole weight	15.94
Delta head	154.25
Delta work	6.91
Delta efficiency	0.83
Efficiency	78.56

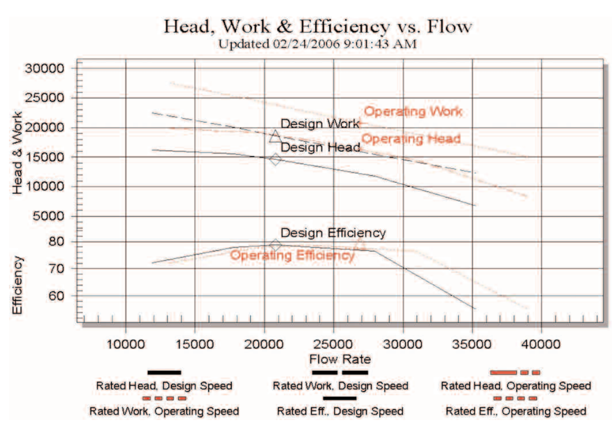

Figure 3. Head, Efficiency, and Work for GB1701 Methanation Compressor. Data Shows Compressor to Be Operating as Expected.

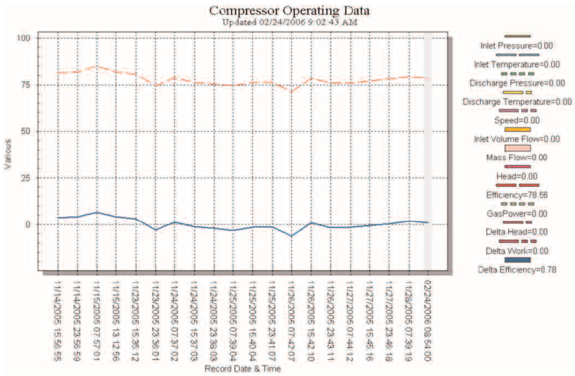

Figure 4. Historical Data Chart for GB 1701 Methanation Compressor Showing Efficiency (Upper Curve) and Delta Efficiency (Lower Curve).

In a typical online performance monitoring system, compressor operating data are transferred from a plant data collection system that, for most new plants, includes all the necessary data required for determining the compressor performance such as head, power, and efficiency. Some older plants do not collect all of this necessary data and some pressures and temperatures will need to be added to the existing data collection system in order to implement the online monitoring system. These data are transferred via dynamic data exchange (DDE) to the performance software calculation engine where the data are processed.

While some may consider it possible to monitor data by comparing discharge pressure or pressure ratio over time, this is not really a good procedure and can create more confusion than help. A change in inlet temperature or pressure will affect the discharge pressure and even the pressure ratio. So, it is wise to monitor head and efficiency. When calculating head, the best method is by using Benedict-Webb-Rubin (BWR) equations of state. This will provide the best means of comparing calculated results with the OEM performance curves for the compressor.

Along with head and efficiency, the work input is also calculated. By monitoring the work input, it is possible to monitor the quality of the compressor data being collected. Work input is representative of the energy transferred from the compressor impeller blades to the gas. Any degradation in the interstage seals, corrosion, or fouling will show up as a loss in efficiency and head, but the work input is fixed and remains the same regardless of these losses. Thus, monitoring work input is a good means to confirm that the input data are accurate.

Case History A

While online performance monitoring does not in itself enhance the mechanical integrity of the equipment, it is another tool for obtaining additional information otherwise not available or difficult to obtain or decipher. Such information can offer additional guidance to operators for avoiding conditions that could potentially cause equipment failure. Specifically, online performance monitoring provides an easy method of monitoring where the compressor is operating on its performance curve making it easy to assure the compressor is operating within the OEM limits. If a failure does occur, the information available from online performance monitoring will be valuable in the troubleshooting efforts.

In 1997, a newly installed propylene refrigeration compressor in southern Louisiana tripped on vibration after just 20 hours of operation (Kushner, et al., 2000). The compressor was a two-section centrifugal with all new internals and driven by a constant speed motor. Inspection of the compressor found that the cause of the high vibration was mechanical failure of the last (fifth) stage impeller. A triangular section of the impeller back plate (Figure 5) was missing and several cracks were found near each vane tip.

Figure 5. Damage at Tip of Fifth Stage Impeller of Propylene Compressor. (Courtesy Kushner, et al., 2000)

The second section of the compressor and specifically the last stage impeller were found to be operating well beyond the end of the curve, in deep choke conditions (Figure 6).

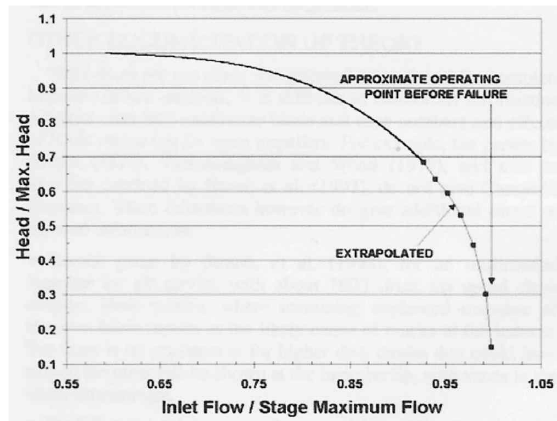

Figure 6. Head Ratio Versus Inlet Flow Ratio for Last (Fifth) Stage Impeller of Propylene Compressor. (Courtesy Kushner, et al., 2000)

While the compressor was an older two-section machine, the installation was new. The compressor had been rerated for the new site with new diaphragms and new rotor. The motor was oversized significantly for the application so that it could be a spare for another compressor. During startup, liquid propylene was sprayed into the inlet piping upstream of the knockout drums to provide gas to operate the compressor. Unfortunately, the flowmeters, which were new, were installed correctly, but the wrong factor was input into the computer resulting in a calculated flow rate that was significantly less than the actual flow rate. So, when the operators noted the flow was looking low, they were concerned about surging the compressor, and increased the liquid injection to increase the gas to the compressor and bring it further away from surge. Unknowing to the operators, they were pushing the compressor deep into choke. Further adding to the problem was the knockout drum design. The drum was horizontal and did not have demister pads.

Since several cracks were found near each vane tip on the toe of the fillet welds, resonance of an impeller natural frequency was at the top of the list of possible causes of the failure. The design of the impeller was a cutback design. The blade outer diameter was less than the impeller hub and cover outer diameter. So, the resolution of the problem was to cut the hub and cover back to the blade diameter, stiffening the impeller and raising the impeller natural frequencies.

There were no metallurgical problems found with the failed impeller. The metallurgical examination by the OEM showed that cracking was not a result of preexisting (manufacturing) defects. Stress levels at the crack initiation site were calculated to be 15.4 ksi.

The only impeller resonance found was for plate modes at the impeller outside diameter (OD). While these frequencies could be excited by return vanes at the impeller inlet, they should dissipate greatly, according to the vendor, by the time flow reached the impeller OD where the failure occurred. In the vendor's opinion, it would be difficult to have this high excitation unless there were liquids involved. The diffuser did not have vanes that might excite the impeller tip.

Flow through the compressor's second section during this 20 hour period was calculated to be 7950 icfm. This is 5 percent beyond the end of the OEM performance curve and 15 percent higher than the original rated flow.

Borer, et al. (1997), did research that clearly shows that flow within a volute and around its tongue or cutwater causes a nonuniform static pressure field that will act as an excitation force at the impeller OD. While this pressure gradient is present throughout the compressor operating range, computational fluid dynamics (CFD) analysis shows that it becomes stronger in the choke region (Figures 7 and 8). Also, full load testing with straingauges demonstrated that overload dynamic stresses are significantly higher than dynamic stresses near surge.

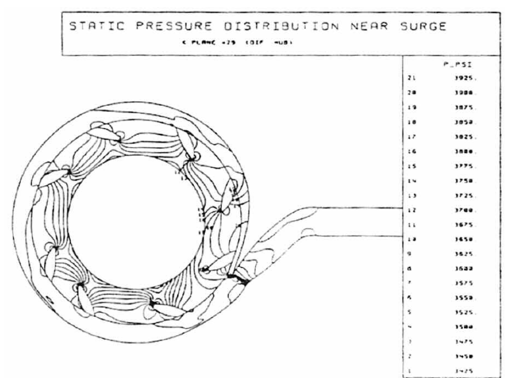

Figure 7. CFD Profile of Pressure Gradient Around the Periphery of a Discharge Volute for a Compressor Operating near Surge. Note How the Pressure Is Relatively Uniform. (Courtesy Borer, et al., 1997)

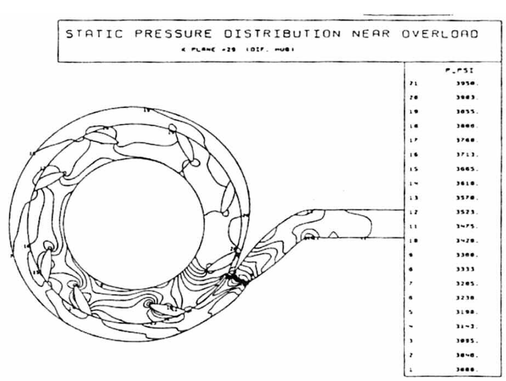

Figure 8. CFD Profile of Pressure Gradient Around the Periphery of a Discharge Volute for a Compressor Operating in Overload Conditions. Note the Large Pressure Change at the Volute Cutoff. (Courtesy Borer, et al., 1997)

Based on this, prudence dictates operation within the boundaries of the OEM curve avoiding choke as well as surge.

The Coal Gasification Plant

The commercial-scale coal gasification plant in Beulah, North Dakota, began operating in 1984 and today produces more than 54 billion standard cubic feet of natural gas annually. An abundant lignite resource underlying the rolling North Dakota plains supplies the plant with the fuel source. The plant consumes more than six million tons of coal each year (Figure 9).

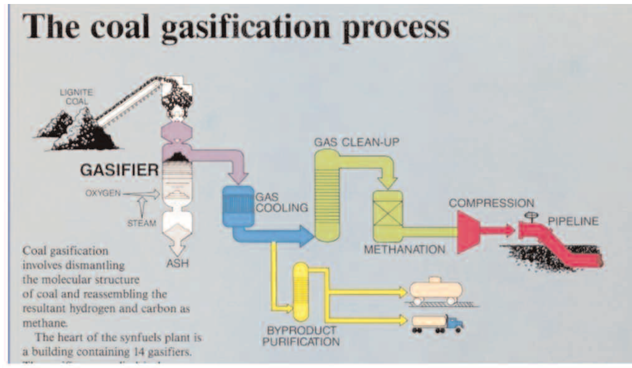

Figure 9. The Coal Gasification Process. (Courtesy Stelter, 2001)

Coal gasification involves dismantling the molecular structure of coal and reassembling the resultant hydrogen and carbon as methane. The heart of this plant is a building containing 14 gasifiers. Lignite is fed into the top of the gasifiers. Steam and oxygen are fed into the bottom of the coal beds causing intense combustion (2200ºF). The resulting hot gases break down the molecular bonds of the coal and steam, releasing compounds of carbon, hydrogen, sulfur, nitrogen, and other substances to form a raw gas that exits the gasifiers. Ash discharges from the bottom of the gasifiers.

The raw gas is cooled and the tar, oils, phenols, ammonia, and water are condensed from the gas stream. Other byproducts are used as boiler fuel for steam generation.

Methanation takes place by passing the cleaned gas over a nickel catalyst causing the carbon dioxide to react with free hydrogen to form methane. Final cleanup removes traces of carbon monoxide.

Synthetic natural gas leaves the plant through a 2-foot-diameter pipeline, traveling 34 miles south. There it joins a 1249 mile interstate natural gas pipeline system, which transports the gas to pipeline companies. These companies supply thousands of homes and businesses in the United States.

In addition to natural gas, the coal gasification plant produces numerous products from the coal gasification process that have added great diversity to the plant's output. These products include ammonium sulfate and anhydrous ammonia, which are fertilizers that supply valuable nitrogen and sulfur nutrients for agricultural crops. Other products include phenol for the production of resins in the plywood industry, cresylic acid for the chemical industry, liquid nitrogen for refrigeration and oil field services, methanol for solvents, naphtha for gasoline blend stocks, carbon dioxide for enhanced oil recovery, and krypton and xenon gases for the nation's lighting industry.

In 2004, this plant began implementation of a continuous online monitoring system for critical machinery because of the following events causing the company +$10,000,000 in lost profits:

- 30,000 hp condensing mechanical drive steam turbine was opened for inspection after 15 years of operation and was found to have severe erosion in the nozzle block area resulting in extending the scheduled outage
- 30,000 hp axial air compressor blade failure resulting in a 50 percent reduction in production (Case History B)
- 14,000 hp centrifugal gas compressor was partially fouled causing a 10 percent reduction in plant rates (Case History C)

Setting Up and Monitoring Compressors

While the Beulah coal gasification plant has realized for some time the value of adding online performance monitoring to their condition-based maintenance program, limited instrumentation has hampered progress. There are 22 critical unspared compressor trains that directly affect the plant revenue stream plus other spared compressor and expander trains. All of these machines are not now monitored continuously because of instrumentation deficiencies. Most of these machine trains have been in service for more then 20 years. As the machine control panels and instrumentation are updated, all of the compressor trains will eventually have continuous performance monitoring. In this interim period, manual data are taken from local instrumentation to calculate machine performance.

Data required for online monitoring includes: inlet and discharge pressure and temperature at each compressor section nozzle, flow rate, and speed. Additionally, it is necessary to have an accurate gas analysis and have a copy of the compressor manufacturer's predicted performance curves. Gas data are entered into the computer along with the curve data for reference.

Accurate data are essential to obtaining good results with any condition-monitoring program. Calibrating all instruments involved will assure accurate data and correct results. One way to confirm the data are accurate is to monitor the compressor work input. If the work input falls on the curve, then the data are most likely accurate. If the measured work input does not fall on the work input curve, then the first thing that needs to be done is to check the accuracy of all instruments. This includes the flowmeters, pressure and temperature readings, and the gas analysis.

At the Beulah plant, data are pulled from the plant historian system to a Microsoft® Excel spreadsheet. From there the online monitoring system takes the data via DDE connection and processes the data to obtain the appropriate compressor parameters such as head and efficiency. A single point is displayed on the performance curve (Figure 3) and the curves are compensated for speed. A historical trend tracks various data (Figure 4) versus time to show degradation of the compressor efficiency and thus aid maintenance scheduling. Tracking the delta efficiency (the difference between the predicted and the actual efficiency) and delta head is a very good indication of compressor health. Tracking the delta work input (this value should be zero or near zero) is a good way of tracking the overall quality of the online performance monitoring system.

Case History B

In June 2002, an axial air compressor blade failed catastrophically in the first row of the low pressure section (Figure 10). The compressor is driven by a 30,000 hp fixed speed synchronous motor through a parallel shaft gear increaser at an operating speed of 4530 rpm. This compressor is one of two machines operated in the air separation facility for oxygen production used in the plant gasifiers and thus crucial to plant operation. The other machine is direct driven by a steam turbine. Both machines were placed in continuous operation in 1984. These machines operate at about 140,000 scfm and a discharge pressure of 90 psia. A similar failure had also occurred in the motor driven train in1995. Subsequent to this failure a protective coating was installed on the low pressure blading to protect it from erosion and corrosion. The impact to plant revenue of this second failure prompted an extensive investigation into the root cause.

Figure 10. Axial Air Compressor Blade Failure. The Second in the Compressor's 22 Year Operating History.

A thrust bearing failure on the steam turbine train in 1993 resulted in an upgrade to a directed lube tilt pad design in both compressors. In 1994 the speed was increased from 4425 rpm to 4530 rpm, which increased capacity by 12 percent. There were no incidents of high vibration, or high thrust bearing or journal bearing temperatures prior to the blade failure. Analysis of the failed blade showed it to be of the correct material and mechanical properties and was not the result of foreign object damage, corrosion, or erosion. Since only one minute averaged data were available from the plant historian, the operating point on a dynamic basis was not available making it difficult to determine that the machine had been operating in a choke condition. A finite element analysis (FEA, Figure 11) was performed to determine the structural response for the defined load conditions. Metallurgical examination suggested that the failure was due to fatigue associated with a relatively high mean stress. In addition, blade vibrations and plastic strain above the tolerance limit most likely assisted in the fatigue failure. It was the contention of the investigator that the crack leading to the blade failure was initiated from the concave (high pressure) side of the thin blade edge and progressed in the transverse direction (opposite direction to airflow) along about 60 percent of the fracture surface until the blade succumbed to ductile overload (Figure 12).

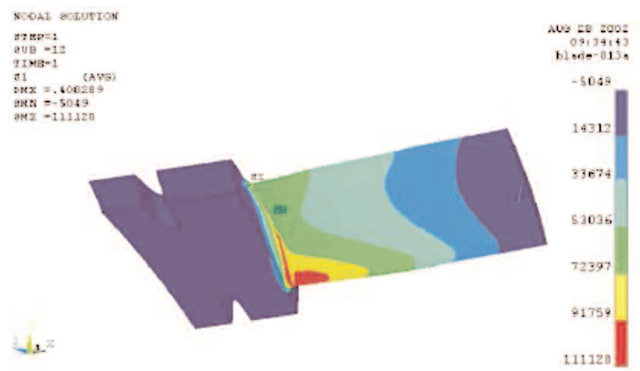

Figure 11. Finite Element Analysis of Failed Axial Blade. Note the High Stress on the Blade Trailing Edge.

Figure 12. Failed Axial Blade. Crack Initiated on High Pressure, Trailing Edge of Blade.

After returning the machine back to service with the spare rotor, high vibration was measured on the inlet air duct with an accelerometer at the blade pass frequency of the rotor's first row of blades. Vibration in g's ranged from 1 g to more than 75 g's. Dynamic loading data were recorded from pressure sensors mounted in the inlet ducting and in the compressor balance line. High energy levels were detected at various times suggesting that

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the blade design was marginal (Figure 13). This prompted a detailed look to see where the machine was operating on its curve. Observation of fluctuating downstream and upset conditions were absorbed by the motor driven train while the steam driven train was base loaded. This was causing the motor train to intermittently operate in a choke condition (Figure 14). Corrective actions included optimization of blade geometry, higher strength material and tighter blade manufacturing tolerances, and revised operating procedures. The authors believe that a continuous online performance monitoring system could have mitigated this situation or at least assisted in the analysis for root cause.

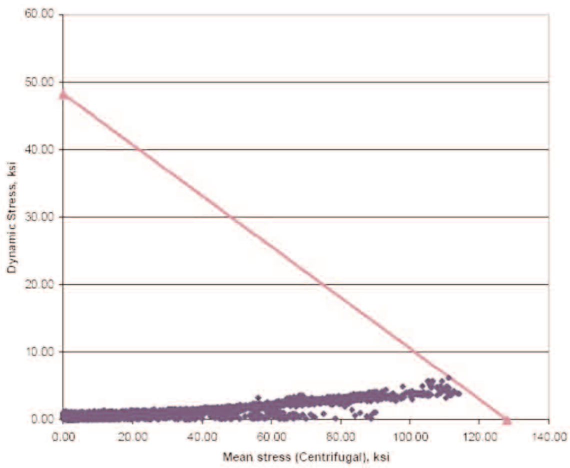

Figure 13. Goodman Diagram for Failed Axial Blade.

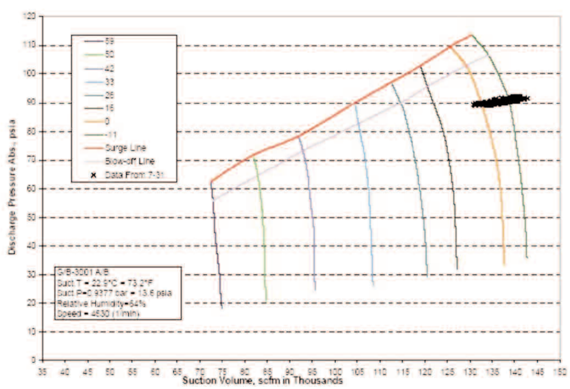

Figure 14. Axial Compressor Performance Map Showing Normal Operation. System Demands Occasionally Drop Discharge Pressure Resulting in Very Brief Periods of Choke Condition and Eventual Blade Failure.

Case History C

Synthetic natural gas is compressed by two parallel case steam driven compression trains after methanation for delivery into a natural gas pipeline. The trains operate at a maximum speed of 13,700 rpm. Suction conditions are 250 psig at a suction temperature of 110ºF. Discharge conditions are 1440 psig max and 110ºF after final cooling before entering the pipeline. Each train has the capability of delivering 85 mmscfd of gas. During periods of high pipeline pressures, the capacity was noticed to be less then expected causing a reduction in plant rates. Two problems were identified. The first was the suction temperature had increased to 150ºF. This problem was traced back to a fouled final heat exchanger in the methanation area. The second problem showed a decrease in efficiency of the first compressor case having five stages. An increase in 1× vibration of the first compressor case was also noted.

A scheduled overhaul of the compressor revealed fouling. Random flaking off of the fouling explained the higher vibration. Analysis of the deposits showed it to be predominantly iron oxide. An inspection of the carbon steel suction piping revealed significant corrosion on the bottom of the pipe. The first attempt to mitigate the fouling problem was to install a nonstick coating on the rotors. This did not solve the problem because the coating was eventually abraded and the fouling reoccurred.

An analysis of the process gas showed it to be saturated. Furthermore, the piping from methanation to the suction of the compressor is outside and not traced or insulated. Ambient conditions can range from –40ºF to 110ºF. It was realized that particularly in the winter, the cold piping was causing condensate to form in the bottom of the piping, which reacted with the gas to form carbonic acid. The acid formed then corroded the steel pipe. The corrosion products were transferred to the compressor by the high gas velocities. A suction knockout pot on the line to the compressor has not been effective in preventing the carryover from reaching the compressor. A glycol water removal unit is located between the first and second compressor cases to knockout the water before exporting the gas.

Injection of chemicals to neutralize the acid formation on the compressor suction piping was investigated and found to be very expensive. The current implementation plan is to steam trace and insulate the suction piping to keep the condensate from forming and to improve the efficiency of the suction knockout pot. Figure 15 shows a drop in efficiency prior to cleaning. Similarly, the compressor efficiency returned to normal after cleaning. Figure 16 shows buildup on the rotor. Continuous performance monitoring is assisting with tracking this situation.

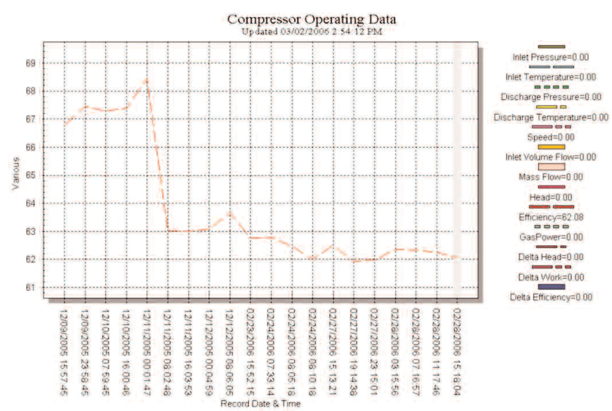

Figure 15. Synthetic Natural Gas Compressor Showing Drop in Efficiency.

Figure 16. Synthetic Natural Gas Compressor Rotor. Note Fouling in Impellers. Monitoring Compressor Performance Helps Track this Condition and Aids Scheduling of Maintenance.

CONCLUSION

In this day and age of high energy prices, revenue losses caused by machine deficiencies, and downtime at the Beulah gasification plant can exceed $600,000 per day. Over the years we have realized the importance of continuous machine vibration, thrust, bearing temperature, and oil condition monitoring to operating reliable and safe turbomachinery. Unfortunately, this does not tell us everything that we need to know to maximize revenue and minimize maintenance expenses.

By integrating machine real-time performance monitoring along with mechanical parameters (vibration data, bearing temperatures, and oil condition), we have realized that better-informed decisions can be made. Online performance monitoring is now a key part of the plant's machine reliability program. Key benefits that have been seen include:

- Knowing the machine performance immediately significantly aids the process of troubleshooting a machine problem and minimizing downtime/loss production.
- Trending of machine performance allows review of operating points that may have subsequently affected machine condition.
- The effects of process changes can be evaluated immediately.
- Knowing machine performance along with vibration, thrust, bearing temperature, and oil condition, scheduled maintenance and overhauls can be extended for well designed, maintained, and operated equipment.
- Performance monitoring provides valuable information when justifying an extended time between overhauls to an insurance carrier as well as minimizing insurance premiums.

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