EXPERIENCES IN ANALYSIS AND MONITORING COMPRESSOR PERFORMANCE

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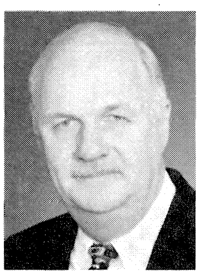

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ABSTRACT

Evaluations of life cycle costs in typical turbomachinery installations have shown that for reliable machines, energy costs normally far outweigh purchase, installation, and maintenance costs. Experience has shown that flexibility in compressor system capacity can often be extremely valuable because many manufacturing plants are limited by compressor capacity. These factors have led to increased awareness of the values of efficiency, capacity flexibility, and performance analysis and monitoring, in addition to the traditional values for reliability.

Typical compressor system life cycle costs estimates are discussed along with the value and benefits from the use of computerized performance analysis and monitoring techniques. Several typical examples of the application of performance analysis and monitoring technology are shown and the resulting benefits are presented.

INTRODUCTION

Life cycle cost analysis has shown that efficiency and reliability are by far the two most important factors to consider in selecting new machinery. Plant compressor requirements often change several times over the life of an installation, and careful analysis of performance is necessary to assure that high efficiencies will always be provided at the changing conditions. Once analysis shows that high efficiencies are obtainable, the next step is to be sure that the high efficiencies are actually met during normal operation and that the full capacity range is always available if needed.

In past years, operating personnel have not had the technology tools or time to identify compressor performance deficiencies even though the stakes were often substantial. Further, when processes operate at conditions significantly different from the original design conditions, the performance evaluation becomes more complicated. With the current wide use of personal computers with spread sheets and specialized software, performance analysis and monitoring are becoming easier and more common. As a result of successes in this area, efforts are expanding in performance analysis and monitoring techniques to many more compression systems.

LIFE CYCLE COSTS

In an effort to optimize use and minimize costs of turbomachinery, the authors began to look at life cycle costs. They found that purchase costs vary considerably because some designs are very complex and require special materials while other machines are more simple and use standard materials and designs. Some of the installations are very elaborate with special buildings with cranes while others are located outside with bare essentials depending on the needs of the business. Energy used to drive the machines may in some cases be free from waste process heat or from burning waste gases. In other situations, the energy may be very expensive. The analyses of life cycle costs were relatively simple but are valuable for deciding relative magnitudes of factors involved in life cycle costs.

After reviewing a wide variety of machines, installations, applications, and energy situations the authors picked rough cost estimates that they believed to be generally reasonable. They used installed costs of \$200/hp, maintenance costs of \$40/hp yr, and energy costs of \$333/hp year. Most large petrochemical companies have several million horsepower in installed turbomachinery. Looking at one million horsepower with a 30 year life cycle, it can be seen that energy costs far exceed any other predictable costs (Figure 1).

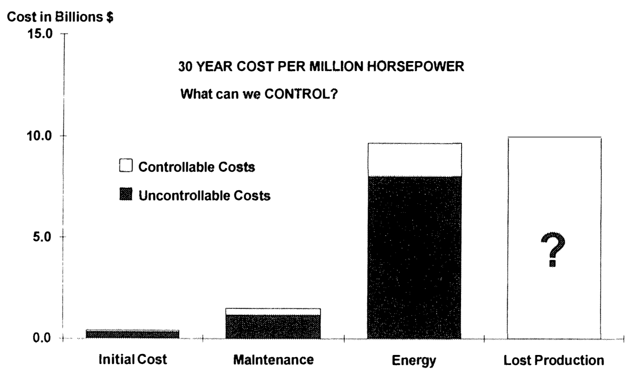

Figure 1. 30-Year Life Cycle Costs.

However, another bar for Figure 1 that is not easy to estimate is one that represents production losses due to equipment failures. It could be very low for reliable spared machinery or very high for single line trains with frequent failures. Because of this, we consider reliability to actually be the most important factor in selecting new machinery.

With basically reliable designs, comparing life cycle energy costs to initial purchase costs leads to the conclusion that additional money is justified to purchase machinery with the highest available efficiency. To verify this, life cycle costs were examined for two extremely different compression systems. The first (Figure 2) is a 20,000 hp process air compressor. This machine handles clean air with minimal corrosion effects. The energy slice of the pie greatly exceeds any other costs. The second example (Figure 3) is a smaller compressor in very dirty and corrosive service. Special materials of construction were required for corrosion and erosion prevention thereby raising initial costs. Frequent disassembly is necessary to clean out solids buildup, and repairs are often needed to repair eroded and corroded components. This makes the pieces of the pie have different proportions. However, the largest piece of the pie is still energy costs by far.

From Figures 1, 2, & 3 it can be seen that to minimize life cycle costs the most efficient and reliable machines must be bought, and they have to be maintained and operated in such a way as to benefit from the high efficiency design. The best way to be sure that turbomachinery is operating at its expected efficiency level is to monitor its performance. This would be simple if each machine always ran at constant flows, inlet conditions, coolant temperatures, speeds, inlet guide vane positions, etc. The problem with performance monitoring is that the machines are almost never run at the design point or at any other constant operating point. This means that performance comparisons to determine deterioration must involve more than simple measurements of flow, pressure, temperature, and power.

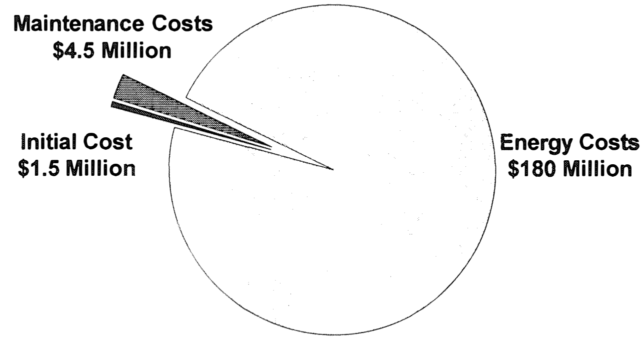

Figure 2. 30-Year Life Cycle Costs for Large Air Compressor.

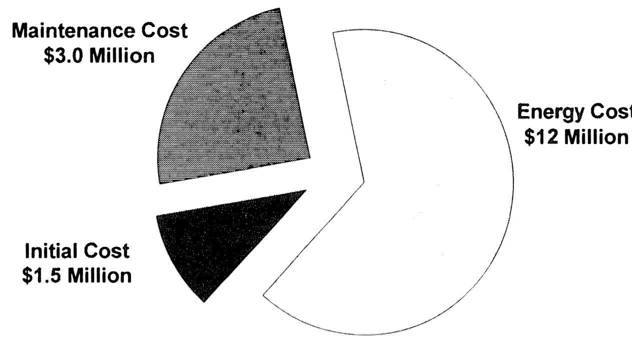

Figure 3. 30-Year Life Cycle Costs for Small Process Gas Compressor in Corrosive, Dirty Service.

Many compression systems have to meet requirements of capacity and operating conditions that are quite different from the original design. Many plants need their machines to operate efficiently at extreme ends of their operating ranges to meet rapid changes in product demands. It may be beneficial to selectively use refrigeration to boost compressor capacity in the hot summer months.

It may be advantageous to install an inlet booster blower or "supercharger" to boost capacity when needed and provide a much broader operating range. The ability to quickly evaluate feasibility and efficiency of all options through performance analysis models can be very valuable.

Also, existing machines can sometimes be used efficiently at new operating conditions that had previously been thought to require purchase and installation of an entire new compression system.

PERFORMANCE ANALYSIS

In order to assure efficient operation at varied flows and conditions, some performance analysis is required. It is necessary to determine operating points on each individual section curve to avoid surge, stonewall, and low efficiency areas on each section curve. Heat transfer calculations in intercoolers, condensers, and evaporators must also be made to evaluate operation in changed conditions. Hand calculations, spreadsheet evaluations, and more sophisticated performance analysis software can be used. For compression section calculations the authors use curves of polytropic efficiency and polytropic head coefficient versus flow coefficient as the basis for all predictions [1]. These curves can normally be obtained in some form from the OEM, can be partially generated in a field test, or can be calculated in sophisticated computer analysis programs.

One possible way to greatly increase the maximum capacity of a compression train is to use an inlet booster blower. The blower can be bypassed to give the normal operating range or turned on and valved-in to boost system capacity. Typical capacity increases might be 20 to 30 percent. Some additional power will be required from the existing driver, but a large part of the increased power will be in the booster blower's drive. Since the inlet density will increase, the possibility of surging must be reviewed. Also, there will be some changes in stage pressures that could require rerating of maximum allowable pressures in compressor cases or intercoolers.

If a higher discharge pressure is needed, a discharge booster compressor might be chosen; and analysis of the complete operating range might help select the type of performance curve needed in the booster. Other possibilities might be speed changes or stage changes in the existing compressor. Calculations of total power requirements in all options should help reduce long-term energy requirements.

Adding or removing compression system side streams with flows either entering or leaving can significantly change stage efficiencies and overall performance. This might be especially critical if adjacent stages are near surge or stonewall.

Another option for boosting summertime capacity is the use of refrigeration in a heat exchanger to cool inlet gas or in selected intercoolers to reduce interstage temperatures. Careful evaluation is necessary because arbitrary cooling might cause surging or no appreciable capacity increase. Usually inlet cooling is most effective and refrigerating some intercoolers has more or less effect than refrigerating others. Use of refrigeration to increase capacity is normally an inefficient method, but it can be done very quickly using rental refrigeration units to get immediate boosts in capacity.

Speed changes will also modify performance. Of course, only speeds below the maximum continuous operating speed should be considered.

Effects of removing impellers from multistage compression sections can be calculated. This might sometimes meet capacity requirements at reduced energy consumption.

One of the major current problems is converting refrigeration units for use with new safer refrigerants without losing the original cooling capacity. Performance models can be used to calculate required speed increases or other changes needed to get equivalent performance.

PERFORMANCE MONITORING

For performance monitoring, the same basic analytical model of the compression system is used, but drivers are also included. Calculations are made to evaluate the condition of each individual system component. Measurements of pressure and temperature are made at the inlet and outlet of each compression section, heat exchanger, and turbine or expander. Speed is also measured along with inlet guide vane positions and appropriate flowrates. Using measured inlet conditions, "good condition" outlet conditions are predicted for each system component. If the measured and predicted outlet conditions are significantly different, the model alerts the user that a problem exists in this component, calculates more information regarding the degree of deterioration, and provides the most likely causes of the problem.

Using a model in this way to monitor performance allows detecting and trending deterioration of various system components. It can be determined exactly where the problems are and where there is nothing wrong. The degree of deterioration and seriousness of the problem can usually be determined. This information allows consideration of things that might be done without shutting down, such as online solvent washing, changing cooling tower water additives, finding leaks around inlet filters, or finding antisurge valves that are not completely closed.

Knowing the condition of the compression system allows better scheduling of downtime for cleaning and repairs. Also, lost production for downtime may be minimized because disassembly and inspection of components in good condition may be avoided or postponed. This also reduces maintenance costs and eliminates possible new problems that result from errors in disassembly and reassembly of the machine.

CASE HISTORIES

Case 1: 20,000 Horsepower Process Air Compressor (Figure 4)

This machine consists of four compression sections separated by intercoolers and driven through a speed increaser by a 1200 rpm synchronous motor. Process requirements for air have abruptly cycled many times over recent years. Minimum demands were so low that air was sometimes vented through the antisurge valve, and at other times 50 percent more than the maximum capacity was needed.

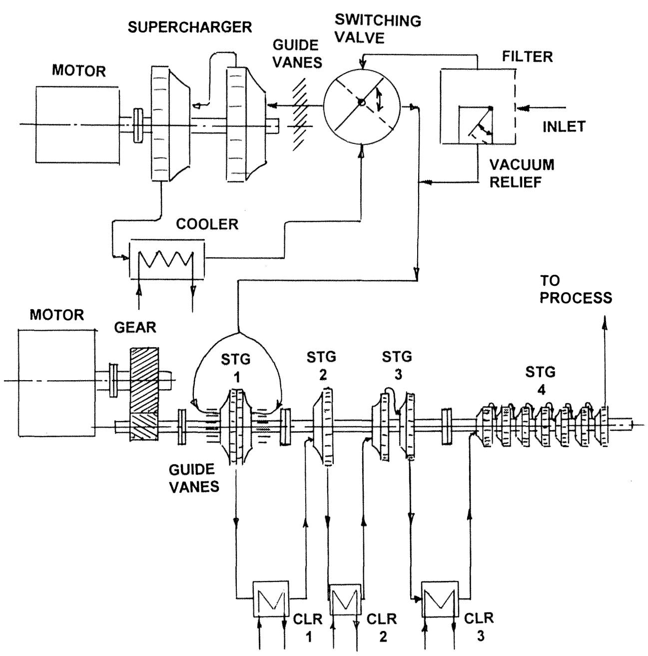

Figure 4. Case 1: 20,000 Hp Process Air Compressor.

This wide range of capacity requirements was evaluated by use of a computer model of the compression system performance. Various options were analyzed including speed changes, replacement of one compression section, installation of an inlet booster blower, inlet air refrigeration, and selective refrigeration of water to the intercoolers. Based on the results of the analysis, an efficient inlet booster blower or "supercharger" as it is called in the plant was chosen. This could be turned on or off as needed and would produce up to 4.0 psig inlet pressure to the original first stage. A supercharger aftercooler was also installed to reduce inlet temperatures back to near ambient. Use of the supercharger boosted the average system capacity by roughly 30 percent, and a large portion of the power associated with the capacity increase was furnished by the supercharger driver. The model showed that the third compression section maximum allowable pressure limit could be exceeded. This required installation of a pressure relief valve in this area and an inlet-guide-vane-position override control to close guide vanes as maximum pressure is approached. Model results showed that in the cold winter months, this would meet capacity demands; but in the hot summer months, temporary use of rental refrigeration equipment was a good choice if maximum capacity was needed.

Analysis of refrigeration use showed that the water to the first main intercooler must be refrigerated before inlet air to the supercharger or water to the supercharger aftercooler to keep the first compression section out of surge. After refrigerating water to the first intercooler, the inlet air and water to the supercharger aftercooler could be refrigerated. This amount of refrigeration added the most capacity to the system for the amount of cooling used. Refrigerating water to the second intercooler also had a significant effect on capacity, but refrigerating water to the third intercooler had very little effect.

By analyzing the compressor's performance with all options, the authors were able to select the choices best suited to meet their needs and to evaluate and avoid potential problems. The available summertime flow range was increased from the original 145,000 to 175, 000 lb/hr to the final 145,000 to 260,000 lb/hr. Installation of the supercharger and its aftercooler was found to be a relatively inexpensive choice to get extra operating range at high efficiency. Use of rental refrigeration equipment for occasional hot weather capacity increases met the capacity needs very quickly, but was not efficient and would not be a good long term choice.

Case 2: Parallel Process Air Compressors (Figure 5)

Two centrifugal air compressors operate in parallel to feed process air to a plant process. One compressor is steam turbine driven, and the other has a dual electric motor and steam turbine drive. Process expansions called for 25 percent flow increase with a 60 percent increase in discharge pressure to push the extra flow through the existing process. Available options were to replace the two compressors with a single dual drive unit rated for the increased flow and pressure, or to install a "supercharger" to boost performance of the existing compressors.

To investigate the performance with a supercharger, a spreadsheet model of each compressor was made. Both compressors are multistage but have single compression sections with no intercoolers. Original factory curves were converted to polytropic head coefficient and polytropic efficiency *vs* flow coefficient to make all new performance predictions. It was found that the new required output could be met with the existing compressors if the suction pressure was increased from ambient pressure to 4.0 psig. The new driver power requirement could be met with existing drivers because most of the additional power would be supplied by the supercharger drive. One concern was that with increased supply pressure, the maximum possible compressor discharge pressure was slightly above one of the compressors' pressure limits. A hydrostatic test of the case at 1.5 times this maximum pressure proved that it was safe.

The supercharger-with-aftercooler option proved to be the least expensive option by far and had the added advantage of efficient operation at much lower capacity, if future needs decreased.

Case 3: Parallel Process Air Compressors (Figure 6)

Two centrifugal air compressors similar to Case 2 operate in parallel to feed process air to a plant process. One compressor is

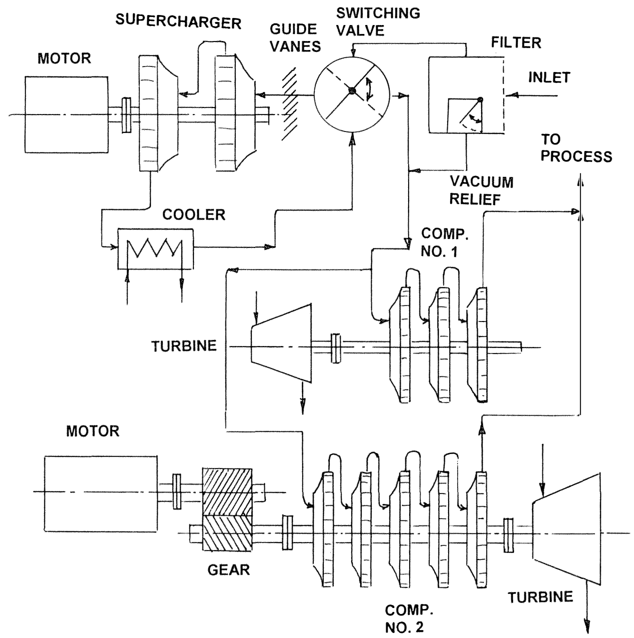

Figure 5. Case 2: Parallel Process Air Compressors.

driven by an electric motor with a speed increaser, and the other has a dual electric motor and steam turbine drive. Both compressors are rated for 5000 hp drivers. The capacity of the plant is limited by these compressors, and an increase in capacity of about 10 percent was needed.

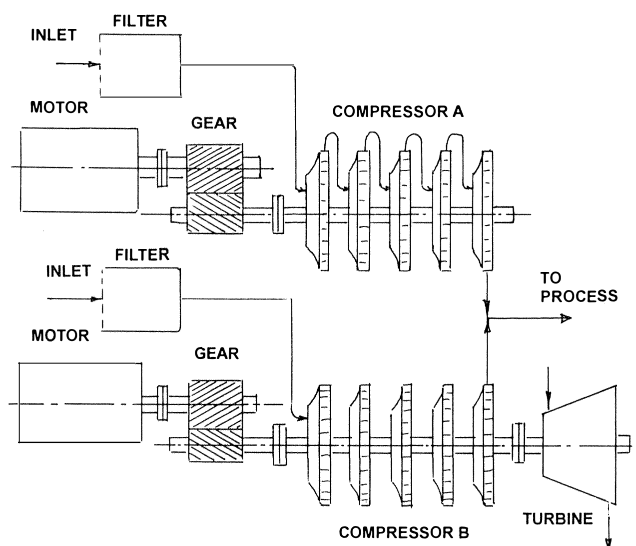

Figure 6. Case 3: Parallel Process Air Compressors.

Both compressors are identical with five stages in a single compression section with no intercoolers. Computer spreadsheet models were made of the machines using polytropic head coefficients and efficiency versus flow coefficient curves converted from the OEM supplied curves. Original plans were to evaluate the option of supplying both compressors with air from a single supercharger at 2.0 or 3.0 psig to get the capacity increase or possibly increasing operating speed. A brief evaluation with original curves indicated that the machines should already be capable of supplying more than the new capacity requirements. Field performance data were then collected and compared to performance predicted by the model. Performance evaluations showed both compressors to be in very poor condition. Each compressor was wasting about 1000 hp and was producing much less head and flow than design.

When disassembled, compressor internals were found to be very dirty, seals were deteriorated, a flexible suction piping connection was leaking unfiltered air into the system, suction piping obstructions were found, and an antisurge valve seat was missing. These problems had been wasting energy and reducing performance for many years and might not have been totally discovered without a thorough performance analysis and field data evaluation. Correction of these problems provided more than enough extra capacity to meet needs, and the increased efficiency reduced energy consumption substantially.

Case 4: 12000 Hp Process Air Compressor (Figure 7)

The compression system consists of four inline compression sections separated by intercoolers and driven by a condensing steam turbine. The reason for evaluating its performance was a general feeling that its capacity was lower than design and that the turbine was using more steam than it should for the air compressed.

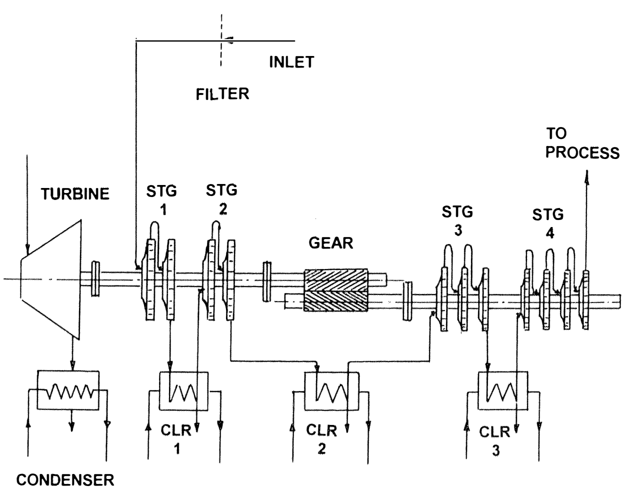

Figure 7. Case 4: 12,000 Hp Process Air Compressor.

A computer model was constructed to represent the compression sections and the intercoolers based on factory test data. Field data were taken to use in evaluating each individual section of the system. Calculations and field data led to the conclusions that close to 1000 hp was being lost in the compressor due to eroded seals and excessive impeller face clearances and that the turbine was not producing its rated power at the steam flow, pressure, and temperature supplied. Turbine power deficiencies were traced to inadequate exhaust vacuum. The exhaust pressure was higher than design because the turbine-to-condenser duct had been initially sized too small.

Use of computer modelling of the compressor system allowed us to evaluate each individual system component by comparing "good condition" performance to actual measured performance. With this, the authors were able to determine where problems existed and where problems did not exist.

Case 5: Integral Gear Air Compressor with Single Stage Booster (Figure 8)

This system consists of a steam-turbine-driven, four-poster-integral-gear air compressor and a discharge booster compressor driven through a speed increaser by a motor. This system

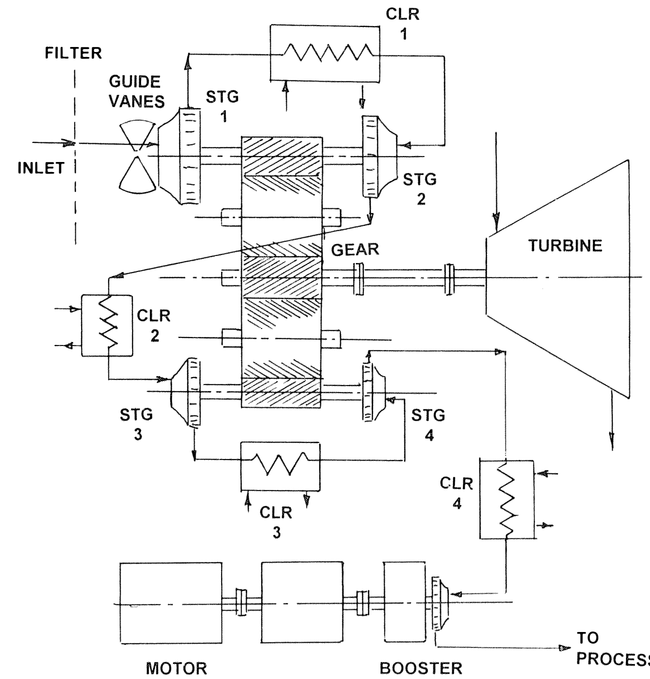

Figure 8. Case 5: Integral Gear Air Compressor with Single Stage Booster.

performance was analyzed to find whether or not the existing system could be uprated to meet increased flow requirements.

Performance models were made of the turbine driven compressor and the motor driven boost compressor. The analysis indicated the only change needed was an uprate of the turbine. After changing out the turbine internals, a system test failed to give the needed system capacity. The new bottleneck was the booster compressor. The booster compressor design was limited to 750 hp, but calculations showed that the uprated flow condition should be met at 680 hp.

The current measurement on the motor indicated that we were using over 800 hp, but the compressor flow, pressure, and temperature measurements led to a gas horsepower calculation of 650 ghp. This motor had been rewound; and its rerated performance values were open to question, so the spare motor was installed. It was new, had never been used, and was rated at 750 hp. The test was repeated, and the authors still failed to meet the desired flowrate because of booster motor overload indications. The motor current limit on the nameplate for its 750 hp rating was reached in the test, but the gas horsepower calculation was only about 650. Without the performance analysis they would have replaced the booster compressor with a larger unit, but instead they chose to do a full load shop test of both motors to measure their actual current values at 750 brake horsepower output. Neither motor came close to the rated current for 750 hp output in the test. The rewound motor was so inefficient that it was scrapped, and the new motor was reinstalled with the maximum current limit raised to match the measured value for 750 hp. The motor manufacturer's explanation was that it had never been tested and that it was not unusual to see a large variation from the nameplate rating values. The compression system was retested, and the new flow requirements were exceeded by a safe margin.

This experience taught the authors that an accurate performance analysis with accurate data is absolutely necessary when making rerate decisions on compression systems. Careful analysis and good data in this case probably saved the company the cost of purchase and installation of a new machine.

Case 6: Integral Gear Air Compressor

This machine is an intercooled four stage integral gear compressor that provides plant air in parallel with several other compressors. It is driven by a four pole 3500 hp induction motor and is controlled with an inlet butterfly valve. The rated capacity and turndown were adequate, but both had deteriorated since the previous overhaul. Several sources of fouling were evident, and an overhaul was needed to correct them. Since this compressor was 20 years old, the plant decided to upgrade to new OEM impellers that offered attractive improvements in efficiency and turndown.

After the overhaul, plant personnel noticed that the machine did not appear to meet expectations. Few of the company installations have sufficient data to do a complete PTC-10 analysis of performance, but there was sufficient information on the process computers to draw some strong inferences. Data available were uncompensated flow, discharge pressure, inlet air temperature and motor power. By plotting a year of hourly data from before and after the overhaul, the authors were able to average out the effects of changes in atmospheric pressure and cooling water temperature and show some fairly clear trends. They found that the compressor met expectations when the inlet and cooling water were cold, but missed performance significantly under warm humid conditions, as shown in Figure 9. Analysis of individual stage efficiencies revealed that the third stage discharge temperature was running substantially lower than expected, making a very strong case for water induction into that stage. The water could have been either from a cooling water leak or condensate which was not adequately separated from the air. A test with the cooling water pressure reduced below air pressure demonstrated that cooling water was not the source of water. Further tests showed that the condensate traps were not air bound and were draining all of the condensate which reached them. Also, very little of the condensate formed in the third stage was being removed by the separation chamber.

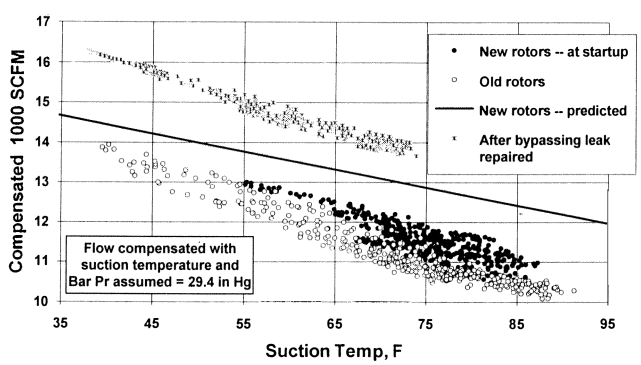

Figure 9. Case 6: Compressor Capacity in Thousands of SCFM Vs *Suction Temperature, F. Capacity on new impellers was below expectations, but improved after correcting internal bypassing.*

The OEM reopened the machine to find why the condensate was not being removed. They found that this particular machine had two carbon steel intercooler condensate lines routed through lower pressure parts of the casing, and one had corroded through in a hidden location. This leak allowed recycle of all of the intercooler condensate and about 2000 cfm of air. The lines were replaced with stainless steel, and the compressor substantially exceeded the OEM's efficiency and capacity predictions. After these corrections, the machine capacity, turndown, and efficiency exceeded the predictions (Figure 10).

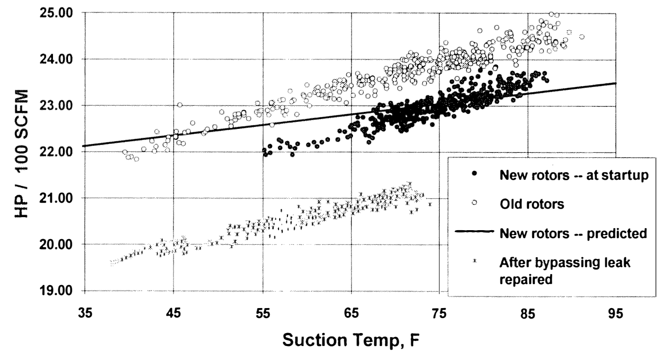

Figure 10. Case 6: Specific Power Was Higher than Expected after Installation of New Impellers, but Exceeded Expectations after Correction of Internal Bypassing.

Case 7: Process Gas Compressor Fouling (Figure 11)

This compressor handles a dirty process gas which occasionally deposits a tar-like material on the rotor and stationary flow passages. It is an uncooled single compression section with three impellers driven by a 1500 hp motor and gearset. Due to the erosive nature of the process material, metallurgists have been reluctant to attempt the use of coatings, and several process changes have succeeded in removing the majority of the contaminants from the compressor suction. Buildups of the tars degrade the performance and under extreme conditions have caused several major rotor wrecks.

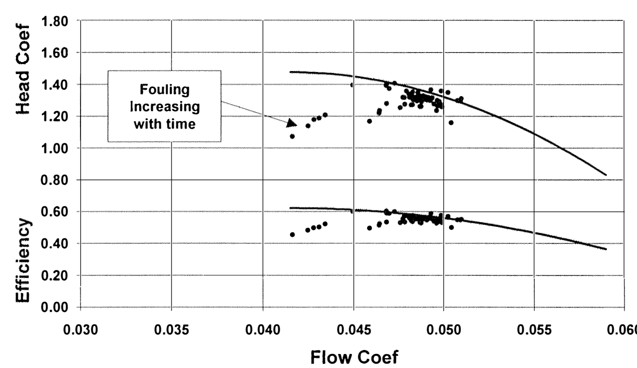

Figure 11. Case 7: Daily Performance Points Show Progressive Deterioration of a Process Gas Compressor for Several Days Prior to Cleaning.

To boost reliability and maximize capacity, the plant had adopted the practice of dismantling and washing the machine about every six weeks. Suction conditions vary considerably during operation, so discharge pressure and temperature versus flow are not reliable indicators of machine condition. The plant wanted to improve uptime and, above all, to avoid wrecking the machine.

An online computer performance model is used to calculate and monitor non dimensional head and flow coefficients, which enables the plant to continuously track machine condition. Converting performance data to nondimensional form removes the effects of changing suction conditions and allows comparison of operating data at all conditions. Daily snapshots are shown in Figure 11 of performance several days just prior to shutdown for washing. While this machine has rather poor efficiency by today's standards, the basic configuration has proven reliable in very demanding service. A more efficient machine of similar configuration has now replaced it. The plant can now accurately monitor fouling and wash the machine when there has been a significant degradation of performance. The six week wash cycle has now been increased to less than twice per year, and there have been no wrecks on this machine in over 10 years. This has paid off in both improved production uptime and reduced maintenance costs.

Case 8: Ethylene Compressor

A 35,000 hp ethylene compressor consists of three casings including four compression sections with intercooling and is driven by a condensing steam turbine through a speed increaser. As with Case 7, an online computer performance model was installed to compute and monitor non dimensional head and flow coefficients which enables the plant to continuously track machine condition. To accurately model this particular compressor, it was necessary to compensate for changes in compressibility factor and the ratio of specific heats through the flow path of the machine. It was also necessary to compensate for changes in molecular weight over time. Since this machine is subject to some fouling, the first use of the data was to assess performance improvements from washing.

After a rerate, the performance of the fourth compression section was found to be significantly lower than design and seemed to be more sensitive to fouling than the other three sections as shown on Figure 12. The data showed that the fourth stage was somewhat below design within 48 hours of startup, and decayed further over about two months. In addition to sensitivity to fouling, polytropic head coefficient was found to vary with suction temperature as shown in Figure 13. This indicated that sonic velocity had been approached somewhere in the machine. Subsequent measurements of individual impeller performance at the casing drains demonstrated that the last stage was causing most of the lost performance (Figure 14). This insight has helped to guide further retrofitting efforts with the OEM.

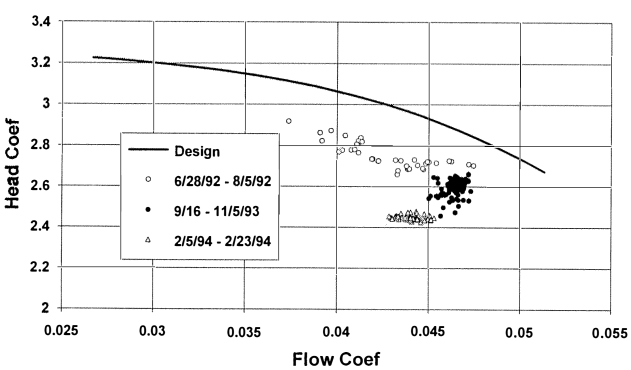

Figure 12. Case 8: Ethylene Compressor Performance Deteriorating over about Two Year Time Span.

Case 9: Fouling of One Section of a Process Air Compressor

A single line compressor driven by a 11,000 hp turbine feeds air to a process and consists of two casings and three compression sections. The plant had lost capacity over about two years, and it was thought that a complete overhaul would be required to restore the capacity.

A model was developed for the compressor, and the analysis of the three compression sections showed that only one section was fouled and was responsible for the lost capacity. This new

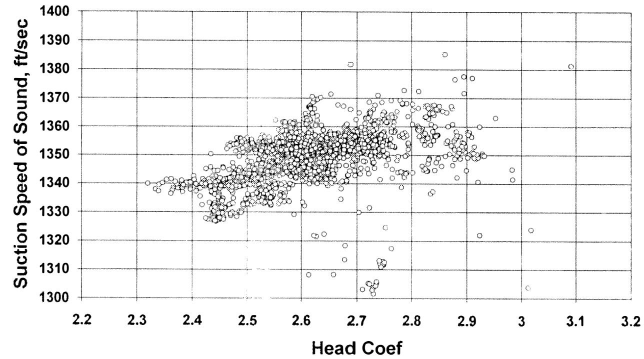

Figure 13. Case 8: Variation of Head Coefficient with Suction Sonic Velocity.

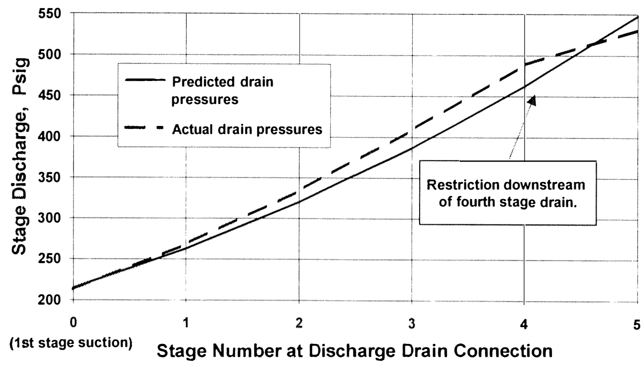

Figure 14. Case 8: Individual Impeller Pressures Measured at Casing Drains on Ethylene Compressor Fourth Section Show Last Stage Fails to Produce Expected Pressure.

information allowed the plant to reduce both the time and cost of restoring the lost capacity by cleaning only the one fouled casing. Intercooler carryover was found to be the root cause of fouling and will require modifications to the demister to prevent fouling.

CONCLUSIONS

Compressor system performance monitoring can help you to significantly reduce life cycle costs by:

- Detecting performance problems that occur in installation of equipment such as leaking antisurge valves, improperly sized lines, and flow obstructions.
- Detecting operating problems early enough to in some cases permit correction online and in others provide an early, planned shutdown to correct the problem before it gets worse.
- Isolating the problem or problems so that minimum down time is required for inspection and repairs.
- Permitting the detected problems to be watched to determine their rate of deterioration and their seriousness.
- Preventing surprises from unexpected deterioration that prevents desired capacities from being met when needed.
- Allowing proper planning for maintenance in shutdowns by giving advance information on problems to be corrected.
- Detecting machine assembly errors that affect performance immediately after startup so that corrections can be made quickly if necessary.

Compressor system performance analysis can also reduce life cycle costs and help meet rapidly changing production needs by:

• Analysis of all options available to meet new requirements.

• Evaluation of efficiency in off-design operation.

• Evaluation of possible problems that could result in off-design operation.

• Evaluation of refrigeration unit performance with new refrigerants.

• Reapplication of existing equipment to meet different application needs.

• Evaluate possible ways to make existing machines more efficient at their normal operating conditions.

• Evaluate possible ways to increase the operating range of existing machines.

REFERENCE

1. Duggan, J. B., "Compressor Performance Modelling and Monitoring," *Proceedings of the Sixteenth Turbomachinery Symposium*, The Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1987).