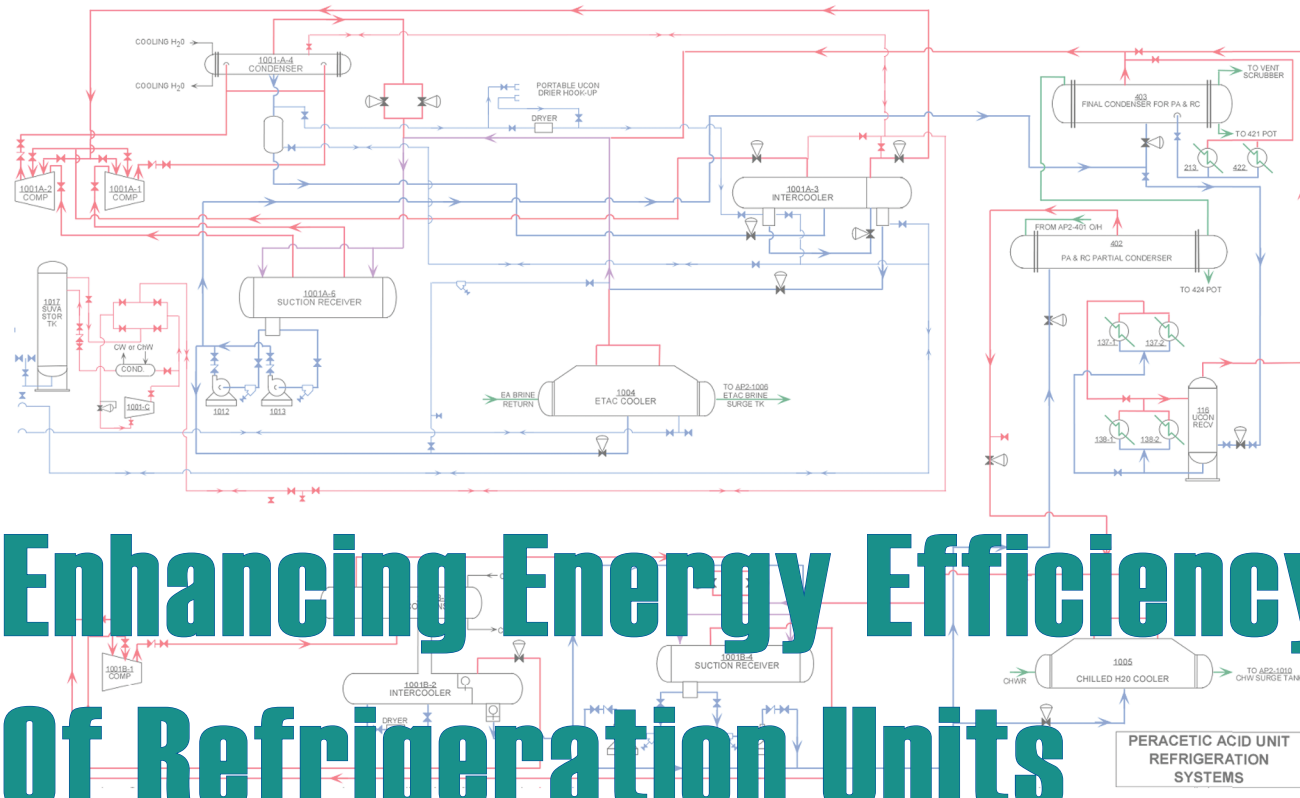

Enhancing Energy Efficiency Of Refrigeration Units

By Riyaz Papar, P.E., Member ASHRAE; **Kevin Zugibe, P.E.,** Member ASHRAE; and **Jeffrey Heitler, P.E.**

This article focuses on the systems approach and the results of energy-efficiency enhancement of two steam-turbine driven refrigeration units in the PXC plant at The Dow Chemical Company (St. Charles Operations, Hahnville, La.).

The plant manufactures specialty chemicals and is spread over an area of about 40 acres. Similar to other Dow chemical plant sites, this site operates a cogeneration facility that is managed by Dow Chemical Energy Services Group.

All the plants on the Hahnville site receive their electrical power from the central cogeneration facility. The thermal needs of the plants are satisfied by steam supplied at different pressures from the cogeneration facility. The utility cost structure to individual plants on the site is interlaced with costs and credits for the supply and return steam to the plant headers.

As a first step towards a systems approach for enhancing the operating energy efficiency of the refrigeration units, The Dow Chemical Company initiated a feasibility study (gap analysis). This study required development of detailed system and individual equipment analysis models to understand bottlenecks and inefficiencies. The evaluation was done on a load profile basis from data collected over a year. The results of this study are presented here, and the projects that were done are described. Most of the study's findings have been implemented, resulting in significant improvement in system operation. The systems approach and analysis incorporated the supply and demand-sides and targeted all cost savings achievable for the overall refrigeration system. The supply-side included steam flow rate, superheat (temperature, pressure), cost of steam, etc. The demand-side included production rates, process loads, cooling water temperature, etc. After the projects were completed, operational data was

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collected and analyzed to quantify the cost savings and level of optimization. The plant has operated reliably for more than three years and also has undergone a refrigerant decontamination procedure during planned shutdown. Results of the plant operation before and after decontamination are presented here.

System Description

The overall PA unit refrigeration system consists of two individual systems, A and B, that operate at different temperature levels and serve a myriad of process loads as well as provide chilled water for the plant. *Figure 1* provides a process flow diagram for the complete system. The total plant capacity is ~6,500 tons (22.9 MW) and has a total charge of 225,000 lbs (102 272 kg) of R-134a refrigerant. Each of the systems has a steam-driven turbine and is supplied by a common utility steam header at 600 psig (4238 kPa). Historically, both systems have been observed to be operating in the surge region of the compressors. Hence, the compressor speed, inlet guide vanes, and the hot gas bypass (HGBP) are set up with a microprocessor-based controller.

Mississippi river water is used for the refrigeration system heat rejection in the condensers. As the river water temperature increases during the summer, the refrigeration capacity and the chilling temperature constrain production rates of the PA unit processes. The rise in the river water temperature leads to high condenser pressures and near surge conditions when process loads are reduced. The plant's control mechanism tries to control the compressor speed to avoid surge. Both systems operate close to the surge point, and the HGBP opens to avoid surge, whenever required. This increases the suction pressure, avoiding surge but at a loss of efficiency and an increase in the evaporator temperature.

Each system has a suction receiver that collects the refrigerant liquid before pumping it to the different loads. Pneumatically actuated valves, based on operating levels in the individual coolers, control refrigerant flow to each of the coolers. Each system also is equipped with a refrigerant liquid spray desuperheater to remove any superheat from the individual coolers as well as the hot gas bypass. Additional details specific to each system are described in the following sections.

System A

System A provides refrigeration capacity at ~0°F (–18°C). Two identical three-stage compressors have impellers that were redesigned when the system was changed from R-12 to R-134a.

As per design, the overall cooling capacity is expected to be 3,340 ton (11.7 MW). The combined cooling load of the Ethyl Acetate (EA, ETAC) chiller insulator, EA vent scrub cooler and the oxygen/ozone coolers is small (less than 2.5%) compared to the total cooling load of system A and is expected to remain fixed during year-round operation. System A has two intercoolers (high pressure and low pressure). Both have pneumatically actuated control valves to maintain liquid levels. The steam turbine on System A receives 600 psig (4328 kPa), 670°F (354°C) steam and exhausts to a steam header of the PA unit at ~75 psig (618 kPa).

System B

System B provides cooling capacity at ~48°F (8.9°C). It has one two-stage compressor, and its impeller was redesigned when the system was changed from R-12 to R-134a. System B supplies cooling capacity to the chilled water cooler and the PA and RC partial condenser.

As per design, the overall cooling capacity of System B is expected to be 2960 ton (10.4 MW). System B has one intercooler that has an internal float valve to maintain a desired level of liquid in the vessel. The steam turbine on System B receives 600 psig (4328 kPa), 670°F (354°C) steam and exhausts to a steam header of the PA unit at ~200 psig (1480 kPa).

What is Six Sigma

Six Sigma refers to a rigorous quality process that measures improvement opportunities based on substantially reducing the number of defects-per-million opportunities in existing systems or designing out defects in new systems. The Six Sigma term stems from a definition of reliably making "on spec" product 99.9996% of the time.

This is a fundamental concept of zero defects that stresses that all errors are preventable. The Six Sigma programs can be broken down into two major categories, MAIC and DFSS. MAIC (measure, analyze, improve, control) is a systematic approach to reducing defects in existing systems or operations, while DFSS (design for Six Sigma) is an approach to reducing defects and breakthrough technology in the design of new systems.

Data Collection and Models

The data collection was done systematically and in a tiered structure. In the preliminary stage, most of the effort was spent on understanding the system from the piping and instrumentation diagrams (P&IDs) and the process flow diagrams (PFDs). This was followed by detailed walkthroughs of the PA unit. At the intermediate stage, effort was focused on collecting detailed design information and studying previous reports and engineering efforts. In the final stage of data collection, six-hour interval averages for data points from Systems A and B, as well as process parameters that were closely related to the refrigeration system were collected for one year. The total hours of operational data that was collected and analyzed was ~6,200 hours, implying ~70% run hours annually. A statistical analysis was carried out on this data to ensure the authenticity of the information, identify trends and relationships and eliminate any erroneous information that may skew the feasibility analysis.

Thermodynamic system models capable of predicting and modeling the system operation were developed. Each system model consisted of the calculation of thermodynamic property values at each state-point, heat and mass balances

over all the heat exchangers, pressure vessels (intercoolers, receivers), prime movers (turbines, compressors, pumps) and auxiliary components (HGBP valves, flow control devices, etc.). Additional detailed models were also developed for the condensers, steam turbines and the orifice plates. A systematic top-down load profiling exercise was then performed to understand the operation of the PA unit refrigeration system.

Six Sigma Process Methodology

From a Six Sigma perspective, the process for improvements to existing facilities is termed MAIC (measure, analyze, improve and control). Some of the components of each phase are as follows:

Measure

- Define the opportunity. In this case it is the excess steam consumption (costs) due to system inefficiencies;
- Establish the baseline. This is the comparison of current operation versus optimized. The defect would be the difference between the two (opportunity for energy and cost savings). This difference is used as the as-is Sigma level;
- Define process steps and establish input variables that affect process performance; and
- Establish the metric for measuring performance. A detailed model was used to predict the opportunity, but it had to be simplified for continuous tracking.

Analyze

- Root cause analysis of system inefficiencies; and
- Determine key input variables.

Improve

- Fix problems areas;
- Monitor performance;
- Determine input variable limits and optimum conditions; and
- Determine "improved Sigma" level.

Control

- Implement control schemes (engineering or administrative) to maintain improvements; and
- Document changes.

Energy-Efficiency Enhancement

The refrigeration system energy-efficiency enhancement project goals were as follows:

- Enhance energy efficiency of the refrigeration system by reducing steam consumption per ton of refrigeration by 5% or more; and
- Obtain nameplate design refrigeration capacity or more.

The measure and analyze parts of the Six Sigma methodology helped to identify several projects for optimizing the performance of the refrigeration system. *Figures 2a* and *2b* show the difference between the current operation and the optimized operation for Systems A and B, respectively. The optimization analysis indicated an overall potential operating energy savings of ~30%. An attempt was made to isolate the effect of each of the identified opportunities and estimate the percentage energy savings that could be attributed to that individual opportunity. This is one of the key requirements of the Six Sigma methodology. In some cases, isolating the individual opportunity and its effect on system savings was difficult. A second important parameter is the metric that defines the improvement on an ongoing basis. These metrics were developed by the plant based on the instrumentation available on the system to assist in day-to-day operations and ensuring that the system efficiency was being maintained.

Several system optimization opportunities to enhance energy efficiency were identified in the "Analyze" section. Plant per-

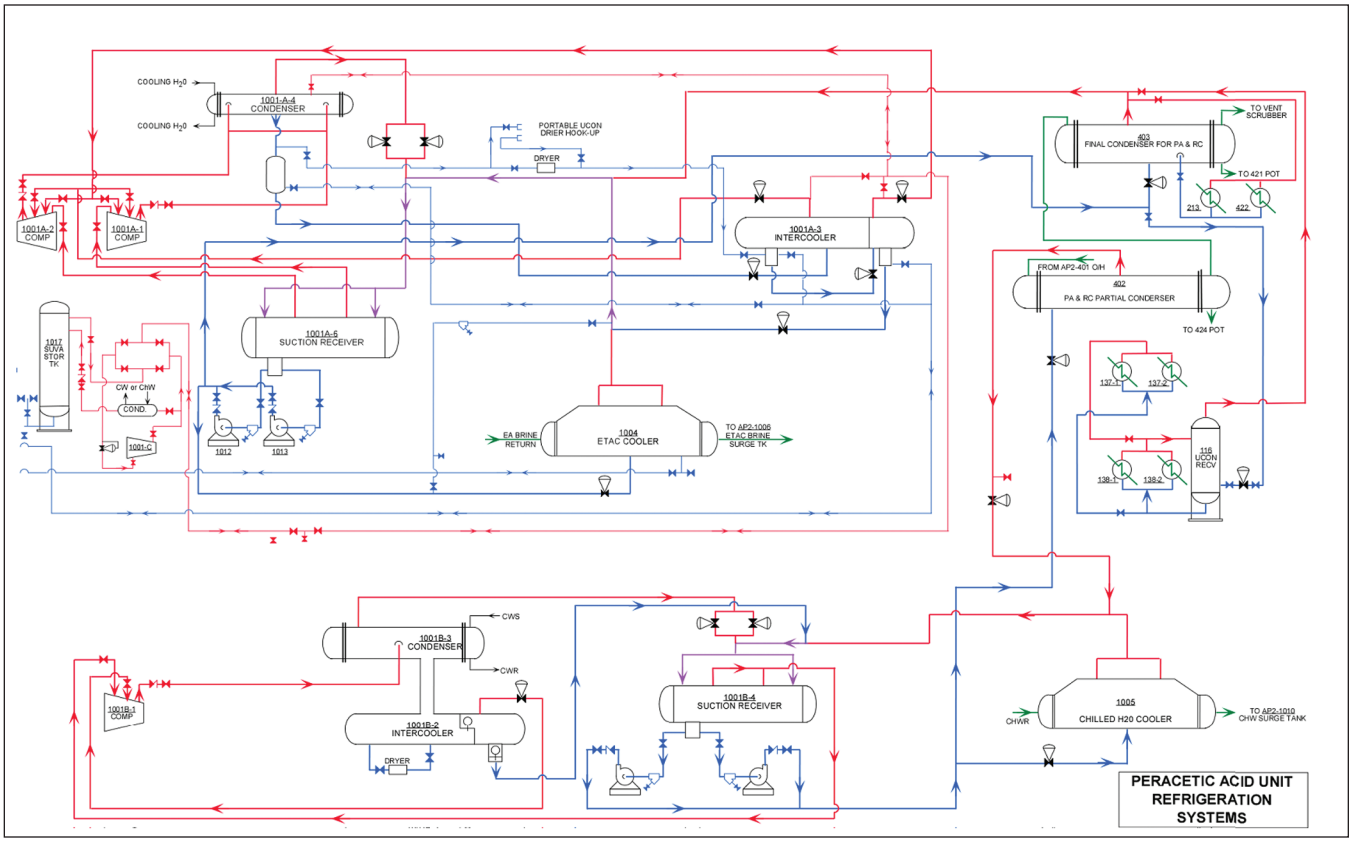

Figure 1: Process flow diagram of the PA unit refrigeration systems.

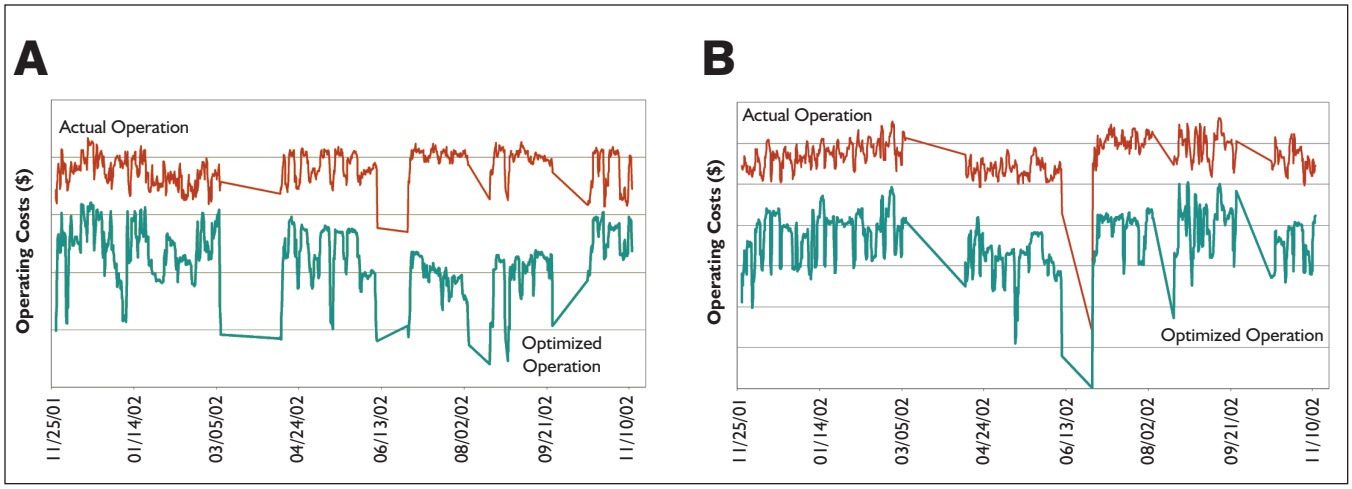

Figure 2a: System A delta analysis. Figure 2b: System B delta analysis.

sonnel implemented most of these opportunities and the results are briefly described as follows (*Figures 2a* and *2b*).

Increase Condenser Cooling Water Flow

Because the condensers on Systems A and B used Mississippi river water, they were prone to heavy fouling. Condenser A was operating at water flow rates ~600,000 lb/h (75.8 kg/s) less than normal operating conditions. Condenser B, was operating at water flow rates ~100,000 lb/h (12.6 kg/s) less than normal operating conditions. This strongly indicated flow restrictions in the condenser and the associated piping. When the condensers were opened for inspection, a large amount of silt and debris was found in the condensers. Additional investigations revealed that restrictive orifice plates were located downstream of the condensers on the return headers. These had been placed in service several years ago when the cooling water flow was limited and other users were starved of cooling water due to the flow requirements of the refrigeration system condensers.

Plant personnel ensure that the condensers are cleaned annually, prior to peak demand. Back flushing is done at every possible opportunity. Finally, the cooling water restrictive orifice plates have been removed from the system. All these measures have increased the cooling water flow rate by 24% and 18% to condensers A and B, respectively. *Figure 3* shows this improvement.

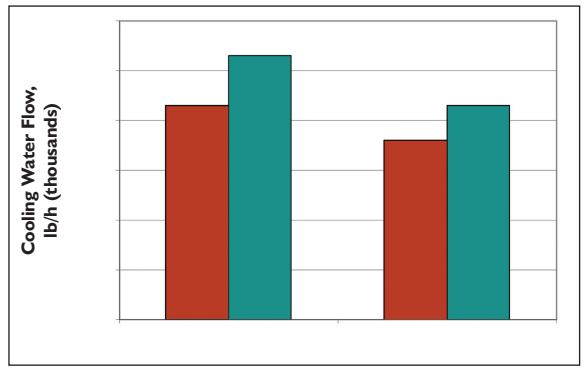

Figure 3: Condenser water flow improvement.

Eliminate Noncondensables

Noncondensables enter the systems at several different locations and times. Most of them are introduced during routine system maintenance. No clear procedure exists for proper evacuation of noncondensables from the system prior to refrigerant charging. Detailed system analysis agreed with the vapor samples from the individual condensers. System A had ~10% noncondensables, whereas System B had ~1%. The presence of noncondensables in the system artificially increases the head pressure and results in a capacity as well as an efficiency loss. On a percent basis, 10% noncondensables in System A resulted in 7.5% efficiency loss. Correspondingly, 1% noncondensables in System B resulted in 1% efficiency loss.

The plant has now developed and documented a procedure for proper system evacuation before charging it with refrigerant. Second, noncondensables are periodically monitored and removed as required.

Monitor Contaminants and Decontaminate Refrigerant

The system has had a history of refrigerant contamination by the process fluids. Refrigerant contamination by the process fluids can affect the system in one or more of the following ways:

- Reduce system efficiency;
- Reduce evaporator capacity;
- Process fluid breakdown leading to acid formation (corrosion) and noncondensables;

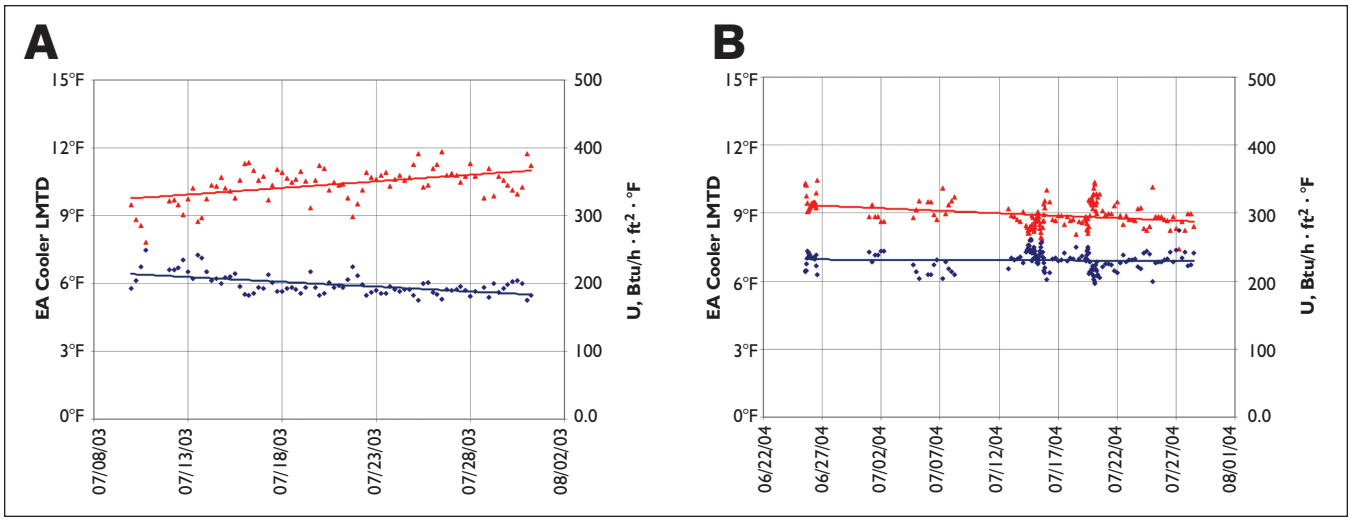

Figure 4a: ETAC cooler performance (07/2003). Figure 4b: ETAC cooler performance (07/2004).

- Fouling of heat exchangers; and
- Reaction with oil.

A heat exchanger model was developed to analyze the performance of the ETAC cooler. Historical data was retrieved for the same time period (July 2003) and compared to data after refrigerant decontamination was performed (July 2004). Analysis of logarithmic mean temperature differences (LMTD) (approach temperatures at inlet and outlet) and overall heat transfer coefficient (U-value) shows that the performance of the EA cooler has improved. The average U-value in July 2003 was ~197.6 Btu/h·ft^2·°F (1122 W/m^2·K), whereas the average U-value for July 2004 is ~228.8 Btu/h·ft^2·°F (1300 W/m^2·K). That is an increase of ~16%. Correspondingly, much closer approaches and tighter LMTDs have been observed in the July 2004 data compared to the July 2003 data. This improvement in the heat exchanger performance can be credited directly to the decontamination of the refrigerant charge. *Figures 4a* and *4b* compare the LMTDs and the U-values for the July 2003 and July 2004 data.

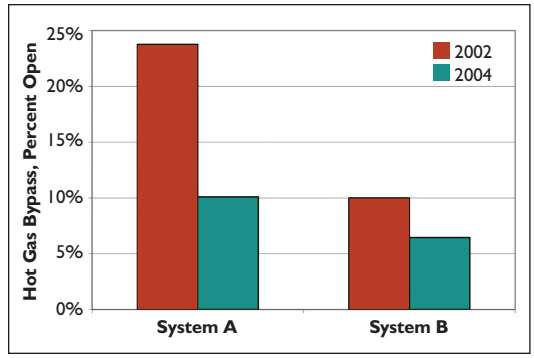

Figure 5: Reduction in HGBP valve operation.

Reduce Surge and Excessive HGBP

There are several factors in the PA unit refrigeration system that resulted in the compressors operating in the surge area. These factors include:

- Low load conditions;
- High head due to noncondensables;
- Inadequate cooling water flow;
- High cooling water temperatures;
- Low turbine speed (horsepower); and
- Heat exchanger fouling.

The system improvements that mitigate this problem fall into several categories. To best understand this problem and remedy it, a root cause analysis was done that started from the basics and identified the cause, effects and penalties associated with the surging problem. Eliminating compressor surge results in tremendous compressor energy savings and operational benefits such as:

- Increased compressor efficiency;
- Lower process temperatures;
- Increased capacity;
- Less wear and tear on the compressor; and
- Reduced refrigerant pumping power.

The annual average HGBP valve position in System A was 23.8% open. At these conditions, there is a 25% increase in energy costs, and almost a 12% increase in the condenser load. A similar scenario existed on System B also. The annual average HGBP valve position in System B was 9.9% open. Because only one compressor is in System B, this was equivalent to the HGBP valve being 19.8% open when compared to System A.

All the improvements related to high discharge pressures were undertaken. The surge protection safety factors in the surge controller program were optimized to prevent compressor hunting, incipient surge and instability. Increasing the safety factor (surge protect ratio) actually generates more hot gas bypass. At the start of the project, both Systems A and B had surge protect ratios set higher than established design setpoints of 1.05 on System A and 1.09 on System B. Apparently, these surge protect ratios had been increased in the past to help stabilize the compressor operation during periods of high discharge pressures and near surge conditions. The reduction in the head pressures, higher

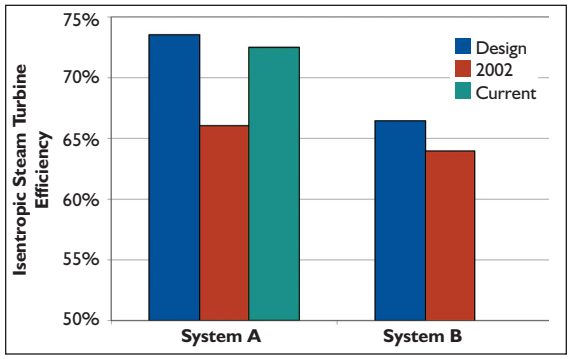

Figure 6: Steam turbine efficiency comparison.

production rates, and improvements on the steam turbines helped both the systems to operate at more stable conditions. Therefore, Systems A and B surge protect ratios were set back to the original design ratios. A combination of these changes resulted in the overall reduction in hot gas bypass. Production was planned to operate at design conditions. A procedure was set in place to monitor any changes in the surge curve and initiate overhauls as necessary. Finally, better controls were incorporated for the inlet guide vanes to limit HGBP operation. *Figure 5* shows the HGBP operation before and after the optimization.

Improve Steam Turbine Performance

Steam turbine performance has been affected and no historical trends exist for the isentropic steam turbine efficiency. Compared to manufacturer's data, the operating steam turbine efficiency value is significantly lower. The design efficiencies of the steam turbine on System A are supposed to be at 73% but the average of the operating efficiencies was found to be 66%. A decision was made to rebuild the steam turbine because along with the higher energy costs the system also was becoming bottlenecked for capacity. The rebuild operations were successfully completed, and *Figure 6* compares the turbine efficiencies to the manufacturer's data. A routine procedure is now set up to periodically monitor and trend the turbine efficiencies and schedule a rebuild at regular intervals or at the first signs of turbine efficiency drop off. System B steam turbine is currently slated for a rebuild in early 2006.

In the PA unit refrigeration system, both steam turbines are designed for 600 psig (4238 kPa), 750°F (399°C) inlet conditions. Based on the annual operating data, the average inlet conditions were found to be 595 psig (4204 kPa), 670°F (354°C). This was due to a unit operation upstream of the PA unit that was exporting steam to the 600 psig (4238 kPa) header at conditions below the site criteria temperature. The pressure difference does not contribute much to the steam rate (lb/h/hp), but the temperature (superheat) has a profound effect on the steam rate. The higher the steam rate the greater the amount of steam required to do the same amount of work. *Figure 7* represents the steam rates with different steam inlet temperatures for the steam turbines in Systems A and B.

System analysis using the annual operating data reveals that the lower supply temperature results in a steam rate penalty of 8% on the System A turbine and 9.5% on the System B turbine. These penalties lead to excess steam requirements for normal day-to-day operations and consequently higher operating costs. The horsepower delivered by the steam turbines also becomes limited due to speed limitations of the turbines and the amount of steam that can enter the turbine. Therefore, the system cooling capacity becomes limited. This also intensifies surge problems at the compressor.

Results

On a cumulative basis, the overall system has shown a tremendous improvement in the overall energy efficiency of the refrigeration system to date. System A has shown improve-

ments on the order of ~11%, whereas System B has shown improvements of ~9% in the metrics that were used in the Six Sigma methodology. To put these in financial terms, these improvements have resulted in annual energy savings of ~$400,000 to date. Note that, as mentioned earlier, the overall refrigeration system optimization analysis indicated a potential energy cost savings of ~30%, however, due to physical and operational limitations, the system was not able to achieve the full potential energy savings. This shows the difference between the desire to optimize and the realities of what can be achieved in a real system. Nevertheless, the energy-efficiency improvement and the energy cost savings exceeded the original goals of the project.

Additionally, other projects that would contribute to enhancing the operational energy efficiency were identified during the feasibility analysis. These projects are being evaluated based on a life-cycle cost analysis as these require a significant amount of capital. These projects include:

- Automatic condenser cleaning and back flush;
- Automatic noncondensables purge units;
- Self-cleaning cooling water strainers;
- Real-time performance monitoring system;
- Conversion to an all electric drive system; and
- Use of steam-driven absorption chiller to offset load.

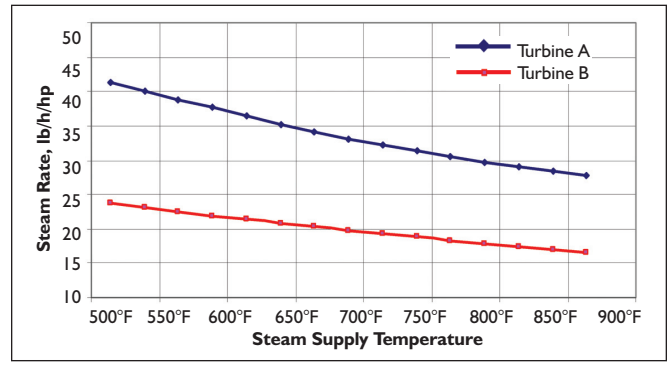

Figure 7: Steam rates for Turbines A and B.

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