DEBOTTLENECKING HYDROGEN RECIPROCATING COMPRESSORS AND SYNCHRONOUS MOTORS

by

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ABSTRACT

A Midwest refinery realized significant gains in hydrocracker throughput and to its bottom line by debottlenecking its hydrogen makeup compressors. Maintaining and even improving compressor reliability and operating flexibility was one of the guiding principles throughout this process.

Two high pressure reciprocating compressors supply makeup hydrogen to the hydrocracker reactors. The compressors were originally installed in 1968 when the unit was first built; no modifications had been made until the recent debottlenecking efforts.

Working extensively with the compressor OEM, nine options were developed to increase the compressor flowrate ranging from 101.7 percent to 230 percent of their original design. Both compressors have been debottlenecked successfully to 112.4 percent of their original design capacity for a minimum cost. An option to debottleneck to 144.1 percent of their design capacity has

been developed and recommended for the future. A plan to debottleneck the existing motors to 125 percent of their rated horsepower has also been developed to be implemented in conjunction with the 144.1 percent option.

INTRODUCTION

Sunoco's Toledo Refinery is a nominal 140,000 bpd crude oil refinery with a high degree of conversion capacity. A vital part of this conversion capacity is obtained in the hydrocracker complex (HCC). This complex consists of a hydrocracker, a hydrogen plant, a naphtha reformer, a motor gasoline reformer, and a PSA unit. The primary products from this complex are hydrogen, gasoline, benzene, toluene, xylene, and a low sulfur distillate blending component. The main economic benefit from this complex is the volume gain it achieves. Because of the high volume gain, this complex is a vital contributor to refinery profitability.

Central to the Toledo's HCC complex are two makeup compressors (C-9202 and C-9203) that compress hydrogen produced by the reformers, PSA, and the hydrogen plant to supply hydrogen to the HCC. These compressors were each originally designed to compress 27.5 mm scfd of hydrogen gas. Both machines are operated continuously to provide maximum throughput to the HCC. Typically, the main throughput limitation to the HCC complex had been the capacity of the makeup compressors. The reliability of the compressors has been historically good. Unplanned compressor downtime will usually result in throughput reductions in the HCC, thus incurring a severe economic penalty.

PROJECT DEVELOPMENT

Refinery units are designed to operate at a rated throughput capacity. An increment in unit throughput could directly add to the refinery's overall processing capacity. Since the refinery has a fixed base operating cost, incremental processing capacity will usually reduce the processing cost per barrel and, therefore, will increase the overall profitability. The objective is to identify opportunities that offer very high returns.

Toledo Refinery's HCC was designed to operate at 24,000 bpd capacity. However, throughput at the HCC had been increased over the years to 29,000 bpd. For some time prior to 1996, unit operators had identified that *"the hydrocracker was limited by makeup hydrogen."* In 1996, a question was asked: "What do you mean the HCC is limited by hydrogen?" There were three possibilities:

- A limitation of hydrogen gas availability, or
- A limitation in hydrogen compression, or
- A limitation of both hydrogen gas availability and compression.

In this case, it turned out that we had plenty of hydrogen gas available, however, we were limited by compression capacity.

Historically, the hydrogen makeup compressors at the Toledo Refinery have operated very reliably. They are overhauled once every two years. Compressor performance reviews have shown that the compressors are operating efficiently and as per design.

In the summer of 1996, the compressor OEM, Cooper Energy Services, was contracted to study the compressors and develop debottlenecking options based upon current and future system operation. Toledo Refinery carried out a performance test to establish compressor baseline performance. The compressor OEM was to work within two constraints:

- Develop low cost options (no major equipment replacement or addition), and
- Implementation must be achieved within the scheduled maintenance window for the compressor overhauls.

As a result of their initial study, the compressor OEM presented two options as shown in Table 1.

Table 1. First Set of Debottleneck Options.

Proposed Modification	Flow (MMSCFD)	Flow Increase Over Base (%)	Motor BHP Required (HP)
Base Compressor (No Modifications)	27.5	---	2,886
Remove 1st stage double deck valve	27.9	1.7	2,936
Change 1st stage cylinder from 19.5" to 20" dia. And remove 1st stage double deck valve	29.4	6.8	3,078

In the spring of 1997, a project team was established to evaluate compressor debottlenecking opportunities. The project team reviewed the options proposed by the compressor OEM. A process review was carried out to evaluate hydrogen line size and intercooler capacity. After an economic evaluation, it was decided to implement the 6.8 percent flow increase option on the C-9202 compressor during the upcoming overhaul scheduled for December 1997.

After ordering parts for the 6.8 percent option, the project team started looking at the possibility of other cost effective options that may provide larger increases from these compressors. A further review by the compressor OEM resulted in three additional options as shown in Table 2.

Table 2. Second Set of Debottleneck Options.

Proposed Modification	Flow (MMSCFD)	Flow Increase Over Base (%)	Motor BHP Required (HP)
Change 1st stage cylinder from 19.5" to 20.5" max. dia. And remove 1st stage double deck valve	30.9	12.4	3,241
Change 1st stage cylinder from 19.5" to 20.5" max. dia., remove 1st stage double deck valve and change 2nd stage cylinder from 17.0" to 17.25" dia.	31.4	14.3	3,327
Change 1st stage cylinder from 19.5" to 20.5" max. dia., remove 1st stage double deck valve, change 2nd stage cylinder from 17.0" to 17.75" max. dia. And install new 13" 3rd stage cylinder	32.0	16.2	3,365

The 12.4 percent and 14.3 percent options provided a significant increase in flow for small incremental cost. However, the 16.2 percent option was considerably more expensive since it involved a third stage cylinder change out, and motor horsepower also became a concern.

Both General Electric (GE) synchronous motors are rated for 3500 hp at 300 rpm. API Standard 618 (1995), Section 3.1.2.1, recommends designing the motor with a 10 percent margin over the required design horsepower. This started to create some concerns since the options listed in Table 2 were cutting into this 10 percent safety margin. There was much discussion as to whether or not cutting into the 10 percent safety margin would be going against the API 618 Standard (1995) and good design practices. Details of this discussion are covered later in the "COMPRESSOR DESIGN AND RELIABILITY CONSIDERATIONS" section.

There was also a concern about the post debottleneck operation and reliability of the compressors. The 6.8 percent option for the C-9202 compressor was more than a month away from being implemented. To address this concern, a four step strategy was developed:

1. Identify, review, and address the possible operational and reliability concerns as a result of the debottlenecking,
2. Talk to other end-users who have debottlenecked their compressors,
3. Debottleneck in smaller flow increments rather than one large change, and
4. Evaluate mechanical performance of the C-9202 compressor with 6.8 percent flow increase prior to debottlenecking the second compressor.

Economic evaluation and the four step review of the options resulted in a recommendation to proceed with the 12.4 percent option for the C-9203 compressor to be implemented during a planned HCC outage in February 1998. Due to long delivery requirements, parts were ordered in the first week of December 1997 while we were implementing the 6.8 percent option on the C-9202 compressor.

The C-9202 compressor debottlenecking work was completed in December 1997 with very little difficulty and within cost and schedule. A couple of flow performance reviews three to four weeks apart validated the expected flow increase of 6.8 percent. Also, motor performance was confirmed as well. No reliability concerns arose as a result of the testing.

Although the C-9203 compressor was not due for an overhaul in February 1998, an HCC outage for scheduled maintenance work provided an opportunity to implement the 12.4 percent flow increase option. This conversion also went very well. A performance review of the C-9203 compressor confirmed a 12.4 percent increase in the compressor flow. The C-9202 compressor was also converted with 12.4 percent flow increase compared to original at the last maintenance overhaul opportunity in January 2000.

A better understanding of compressor debottlenecking opportunities made us curious to explore yet larger flow increase options with no restrictions on motor horsepower; the only restriction for these options was compressor frame horsepower. They included maximizing piston sizes for each cylinder class and installing larger cylinders. Working with the compressor OEM, various options were discussed and five additional options were developed, as shown in Table 3.

Table 3. Third and Final Set of Debottleneck Options.

Proposed Modification	Flow (MMSCFD)	Flow Increase Over Base (%)	Motor BHP Required (HP)
New 23" dia., 1st stage cylinder with a new pulsation damper, change 2nd stage Cylinder from 17.0" to max. 17.5" dia., and a new 13.5" dia., 3rd stage cylinder and a new pulsation damper	39.6	44.1	4,132
All new cylinders, new pulsation damper and refurbished 5500 HP motors	48.3	75.5	5,002
All new cylinders, pulsation bottles and motors at 300 RPM	58.0	111	6,046
All new cylinders, pulsation bottles and motors at 327 RPM	63.3	130	6,607

The options described in Table 3 require significant compressor part replacements as well as power demand in excess of 3500 hp. The synchronous motors were rated for 3500 hp at 300 rpm and 0.8 power factor (pf). The electrical engineer identified the possibility of potentially increasing the motor horsepower by changing the pf to 1.0. Based on a quick review of the motor V curve (shown later in Figure 10), it appeared that a 25 percent increment in motor horsepower available could be realized as a result of changing the pf to 1.0. To further evaluate this possibility, the motor OEM was contracted to carry out a study and provide recommendations. Their study concluded the motors were capable of providing 4375 hp at 1.0 pf. And this change in pf would require no modifications to the existing motors. However, some additional capacitors may be required to maintain the refinery power system balance.

As a result of the motor OEM study, the 44.1 percent option appeared to be a real and viable option with the existing motors. In addition to compressor modifications, the first stage suction intercooler and compressor water jacket cooler would also require additional capacity. Intercoolers, knockout drums, piping, and other process equipment were found acceptable for the 44.1 percent flow option. The motor OEM also calculated the torque values for the motors at 4375 hp. These torque values were reviewed by the compressor OEM for the compressor torque requirements for 44.1 percent option. The OEM evaluation concluded the available torque, although reduced, would still meet the compressor requirements.

The 44.1 percent flow option provided an opportunity to once again review the hydrogen availability and demand at the HCC complex. It was learned that other HCC throughput limits and hydrogen gas production capability limits were reached before encountering makeup compressor limitations. The refinery has set up a team to review the debottlenecking opportunities that may exist in both the HCC and the hydrogen plant for the next major turnaround.

COMPRESSOR DESIGN AND RELIABILITY CONSIDERATIONS

Compressor Description

Two Cooper Bessemer, model LM-3, compressors boost hydrogen gas from 250 to 1450 psig at a rate of 27.5 mm scfd. The compressors are three-stage units operating in parallel. The pressure increase is accomplished in approximately equal steps. Piston sizes on the machines were 19.5 inches for the first stage, 16.25 inches for the second stage, and 11.0 inches for the third stage. The pistons are double acting, pumping gas in the forward and reverse direction. The compressors are each driven by a 3500 hp GE synchronous motor.

Reliability

The compressors were also evaluated to determine what improvements may be needed to ensure reliability with the additional demand on the equipment after the rerate. Although updates have been made to the equipment since installation in 1968, the compressors lacked some of the features available on new compressors. The improvements considered include: a valve design change, a new lubricator system for rider band and packing case lubrication, rod drop detection, and frame vibration monitoring. These improvements were evaluated and no changes were made for the 12.4 percent option. However, 44.1 percent or larger options would include many of these improvements.

OEM Considerations

Most existing reciprocating compressors have the potential to be rerated such that additional process gas can be compressed by basically the same machine. However, when considering the rerate of an existing compressor, there are several items that should be carefully studied early in the project development stage. The service history of the compressor needs to be considered as it relates to previous failures and their subsequent repairs, which may have had an undesirable impact on the compressor and its ability to be uprated to the theoretical full potential. Other user related items include the maintenance practices employed throughout the years and the resulting current condition of the compressor as well as the support systems and their ability to handle additional duty.

If the initial investigation does not reveal anything that would obviously make the candidate compressor a bad choice for rerate, the desired process conditions should be looked at to determine the general requirements in terms of cylinder sizing, rod loading, and compression power. The compressor OEM, who generally knows the design evolution of the equipment, can determine if uprate potential is likely such that further pursuit is warranted.

Frame and Rod Load Evaluation

The historical reliability of the particular LM-3 compressors at the Toledo Refinery seemed to make them good candidates for a possible uprate. Existing process conditions were provided to the compressor OEM. Baseline compressor performance was calculated with the current conditions. A series of possible modifications to increase compressor capacity was then developed. Two important compressor frame design parameters were checked before deciding to proceed further with the rerate investigation. First was the frame nominal horsepower rating. Although not always a hard and fast parameter, the new power requirement was compared against the nominal frame rated horsepower to gauge how close to the application limits the new conditions placed the compressor. Typically, the highest frame and crankshaft stresses are induced by rod load and not by input horsepower. Therefore, it is often perfectly acceptable to exceed the nominal frame rating as long as the rod loading is not exceeded. For the LM-3 compressors at the Toledo Refinery, the nominal frame horsepower rating is 7330 bhp. Frame horsepower limitations were not a concern, as the drive motors are rated at 3500 bhp and the initial rerate configurations were to utilize the existing motors.

The most significant consideration in regard to frame suitability for the rerate is that of comparing the new rod loads at relief valve set pressures against the compressor rating. Since the highest frame and crankshaft stresses are usually induced by rod load and not horsepower input, paying particular attention to them is imperative so as not to compromise compressor reliability after the rerate. The OEM engineering team, during the original design of any compressor, determines gas and combined rod load limits. Generally, a target machine-rating is established at the outset of the design phase, and then all compressor components are designed to operate safely at these load levels plus some predetermined factor of safety. Exceeding these limits for any significant amount of time may result in catastrophic failure of the compressor running gear or static components.

If a particular compressor frame is rated below the modern day limits, it may be possible to increase those ratings. This is because there have often been several relatively minor design changes that resulted in the increase of the compressor rod load rating. Two relatively simple improvements have been the material and thread manufacturing method (cut threads versus rolled), e.g., the clamp bolt, used in the eye of the master compressor rod as well as the rod cap. With some compressor designs, the bolting of the crosshead guide to the frame has also been modified during rod load uprating. In these instances, the existing frame casting may or may not have the necessary material in the proper areas to support the new bolting. This needs to be evaluated when considering a rod load rerate.

When calculating the rod load, it is important to consider the suction and discharge pressure internal to the cylinder that produce the actual stresses on the equipment. The techniques for calculating rod load based on internal pressures have been improved since many of the machines in operation today were installed. The first check is often of the traditional or nominal rating that most machines were furnished under, i.e., the calculated rod load based on pressures at the cylinder flange. Currently, this rating is not generally recognized by the requirements of API 618, Fourth Edition (1995), but it can be a good first check.

The accepted criteria of API 618 (1995), and the industry in general, are twofold: the gas loading on the compressor static parts (cylinders, heads, distance pieces, crosshead guides, and bolting) and the combined rod loading on the running gear. Both criteria must meet OEM ratings. The combined rod loading considers gas plus inertia effects. Both criteria consider internal cylinder pressures, including valve losses, to calculate the rod loads.

The simpler method of using cylinder flange pressure recognized the limitations of this calculation; therefore, it was used along with generally a higher safety factor to rate the compressor. The more sophisticated calculation methods available today generally result in a higher compressive rod load rating, although the tensile load typically is not changed.

The nominal frame loading limits for the refinery's LM-3 compressors are 150,000 lbs (150 kips) in both compression and tension. Using the more sophisticated techniques, the OEM allowed a gas rod load of up to 160 kips in compression and 150 kips in tension, and a combined rod load of up to 175 kips in compression and 150 kips in tension. The gas/combined rod loads for the baseline case and the rerate cases of 6.7 percent, 12.4 percent, and 44.1 percent additional flow are summarized in Tables 4, 5, 6, and 7.

Table 4. Gas Rod Load/Combined Rod Load for the Baseline Case.

Stage	Compressive (kips)	% of Allowable Stress	Tensile (kips)	% of Allowable Stress
1st	49.0/57.3	30.6/32.7	40.7/46.0	27.1/30.7
2nd	107.7/106.3	67.3/60.7	91.4/84.6	60.9/56.4
3rd	70.9/77.5	44.3/44.3	41.1/45.4	27.4/30.3

Table 5. Gas Rod Load/Combined Rod Load for the 6.7 Percent Option.

Stage	Compressive (kips)	% of Allowable Stress	Tensile (kips)	% of Allowable Stress
1st	60.2/68.8	37.6/39.3	51.5/55.2	34.3/36.8
2nd	113.3/111.8	70.8/63.9	95.9/90.5	63.9/60.3
3rd	66.7/75.6	41.7/43.2	36.2/40.8	24.1/27.2

Table 6. Gas Rod Load/Combined Rod Load for the 12.4 Percent Option.

Stage	Compressive (kips)	% of Allowable Stress	Tensile (kips)	% of Allowable Stress
1st	70.2/76.5	43.9/43.7	61.2/62.9	40.8/41.9
2nd	117.6/99.5	73.5/56.9	99.5/94.8	66.3/63.2
3rd	63.4/72.7	39.6/41.5	32.5/39.4	21.7/26.3

Table 7. Gas Rod Load/Combined Rod Load for the 44.1 Percent Option.

Stage	Compressive (kips)	% of Allowable Stress	Tensile (kips)	% of Allowable Stress
1st	91.8/102.3	57.4/58.5	82.5/85.0	55.0/56.7
2nd	104.7/111.6	65.4/63.8	88.0/90.1	58.7/60.1
3rd	102.6/112.9	64.1/64.5	73.4/75.4	48.9/50.3

As can be seen by the results listed in each table, the rod loads were not a concern for any of the rerate cases. Therefore, LM-3 compressors at the Toledo Refinery did, and will continue to, operate well below the rod load ratings. While these results are at the operating pressures, relief valve set pressure rod loads were also checked and, as expected based on the low percent of rated rod load values, did not pose any concerns.

The final consideration, related to rod load, is the compressor piston rod, thread root stress. The thread root stress is calculated from the maximum rod load; however, the OEM may use separate criteria. Considering the standard compressor rod diameter for any given unit, the root stress limit is related to the rod load limit such that if the rod load limit is not exceeded, the thread root stress rating is not exceeded either. However, many users have their own root stress limit. In those cases, it may be necessary to use a larger-than-standard diameter rod to lower the thread root stress. However, this refinery did not specify any root stress limits, so the compressor OEM standard ratings were considered.

Relief Valve Horsepower

The relief valve horsepower was an important consideration when reviewing compressor and motor limitations. Relief valve horsepower is the amount of power required by the compressor during a pressure relieving situation. This is dictated by the relief valve set pressure in the compressor discharge piping. The compressor and motor must be capable of handling the new hydraulic loads without equipment failure and without compromising reliability. Both the compressor and the motor may be mechanically capable of handling the relief valve horsepower for the rerated conditions; however, the motor may be electrically overloaded during a pressure relieving situation.

The original relief valve setting in the compressor discharge piping was 1750 psig. The original design pressure was 1570 psig. This was sufficiently high for all debottlenecking options considered based on the design discharge pressure. With the operating pressure at 1450 psig, however, the relief valve set point could have been lowered to 1600 psig and still provide an adequate margin over the design discharge pressure. The result would be a lowering of the relief valve horsepower.

Moreover, it was determined that a slight increase in horsepower above the motor rating during a relieving situation was temporary and controllable such that motor reliability would not be compromised. Motor amperage and winding temperature could be monitored for emergency compressor unloading that could furthermore be automated through the use of pneumatic unloaders or a process recirculation line.

Pulsation Damper Pressure Drop

Assumptions about the pressure drop across suction and discharge pulsation dampers directly influence the resulting horsepower calculations. The greater the assumed pressure drop across the dampers, the greater the horsepower requirements are for a set of given process conditions. This may influence motor size selection, or in this case, limit the optimum compressor rerate based on available motor size.

The pressure drop across a pulsation damper is based on a percentage of the average pressure at the inlet to each damper. According to API 618 (1995), Section 3.9.2.2.4, the maximum pressure dropped is calculated by the following equation:

$$\Delta P(\%) = \frac{1.67(R-1)}{R} \tag{1}$$

where $\Delta P(\%)$ is the maximum pressure drop based on steady flow through a pulsation suppression device, as a percentage of the average absolute line pressure at the inlet of the device. R is the pressure ratio across the cylinder.

By this equation, the maximum pressure drop across a pulsation damper for the compressors was 0.92 percent and occurred in the second stage of the compressor. In the original compressor design calculations, the compressor OEM had used a 1.0 percent pressure drop for all the dampers, which is consistent with the maximum allowable pressure drop calculated from Equation (1).

However, for the rerate calculations, 2.5 percent of design pressure was used based on uncertainties in baffle integrity, vessel cleanliness, and a desired conservatism in the required horsepower calculation. The compressor OEM completed an analog study at the time of the original compressor design. Once the potential rerate flows were calculated using the 2.5 percent assumed pressure drops, their analog engineering department made a cursory review of the original study data as compared with the new expected flows and determined a complete analog study was not warranted. While there was some risk to this, experience has shown the risk was justified. However, when increasing the flow by 44.1 percent, a new pulsation study would be required since the pulsation dampers would have to be changed to accommodate the new cylinders.

The level of conservatism used for pressure drop assumptions is subjective, depending upon the level of knowledge about the condition of existing equipment. The pressure drop assumption could have been modified to more accurately reflect existing conditions through further testing, inspection, and cleaning. A visual inspection and cleaning had been completed during two previous outages. Figures 1 and 2 illustrate the impact of pressure drop on the flow and horsepower predictions.

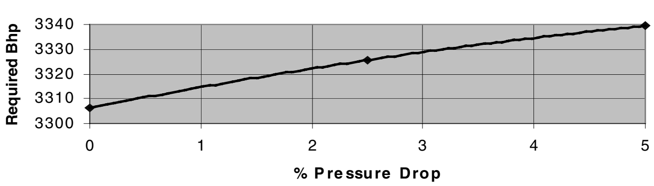

Figure 1. Pulsation Damper Pressure Drop Versus Brake Horsepower.

Valve Modifications

The removal of the unloader valves was one of the first considerations in the compressor rerate study. The unloading valves provided additional operating flexibility by loading and unloading the compressor in varying steps depending upon unit requirements. However, unloader type valves have more fixed clearance than standard valves. Therefore, throughput was reduced because of the added fixed clearance. There was a total of seven suction side unloader valves and one discharge side, double deck valve. The high clearance, double deck valve was in place on the discharge of the first stage to facilitate the installation of a clearance volume valve cap unloader that was under the discharge valve.

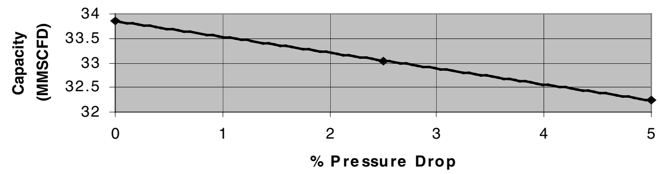

Figure 2. Pulsation Damper Pressure Drop Versus Capacity.

The unloader under the double deck valve provided the least change in the compressor flowrate, only five percent. By removing this valve and replacing it with a standard nonunloading plate valve, the first stage fixed clearance was reduced, thus increasing gas flow through the compressor. This modification resulted in a flowrate increase of 1.7 percent, requiring an additional 50 hp. Minimal operating flexibility was lost by this modification.

Another compressor rerate option involved the removal of all seven suction unloading valves and replacing them with standard, nonunloading, plate valves. These valves unloaded the compressor in several increments from 100 percent to zero percent flow depending upon the valve sequence. Removing these valves would greatly restrict the operating flexibility of the compressor, requiring the addition of a recirculation line from the third stage discharge to the first stage suction. The recirculation line would allow startup of the compressor without the horsepower required for the fully loaded condition. The only other alternative in operational flexibility would be the two head-end clearance pockets on the first and second stage cylinders. They provide approximately 10 to 15 percent unloading, depending upon the sequence.

Only a 1.5 percent flowrate increase could be realized with the removal of all seven suction unloading valves from the compressor. The loss in operating flexibility and reliability was of greater value than the small increase in flowrate. The addition of the recirculation line required by this option would also necessitate the installation of other process equipment such as heat exchangers, knockout drums, etc. This was a very costly alternative with little benefit. Therefore, this option was not pursued further.

The suction unloader valves would provide the operating flexibility to allow for machine startup without the risk of interstage pressure buildup or excessive horsepower requirements. Automated valve unloaders were considered for rerate options where the motor would operate very near 100 percent full load. This would allow the compressor to be unloaded quickly in the event of process excursions that caused the horsepower to increase.

Cylinder Modifications

Several possible modifications were developed as a result of running compressor performance calculations with the revised process conditions. Enlarging cylinder bore sizes provided numerous, relatively inexpensive, possibilities. This is because most of the compressor cylinders employed on reciprocating process compressors are equipped with replaceable liners. The advantage to this is that the cylinder bore can be increased or decreased, within the design range of the particular cylinder class, in an attempt to meet the revised duty requirements.

These liners are usually an interference fit in the cylinder body, so the entire cylinder should be removed from the frame and taken into a capable machine shop to make the modifications, as done by the refinery for the completed rerate cases (refer to photos in Figures 3 and 4). If the modification is to enlarge the bore, the existing liner can usually be bored to the larger diameter. If a cylinder must be made smaller, then the existing liner must be cut out and a new one installed with the necessary interference fit. This requires heating the compressor body and/or chilling the liner.

Figure 3. Toledo Refinery's Hydrogen Compressor.

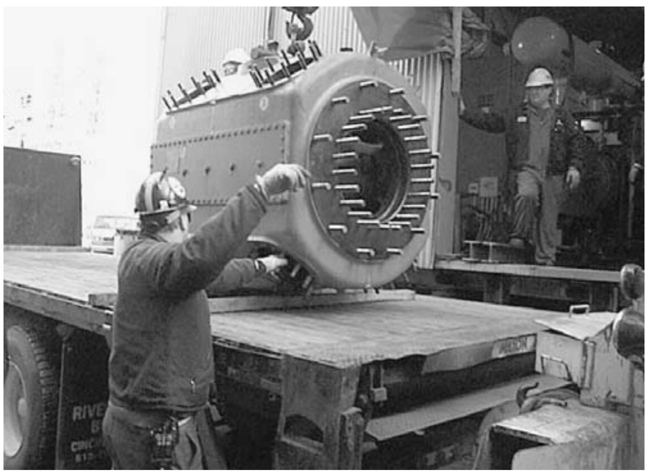

Figure 4. Removed First Stage Cylinder.

By design, the three particular cylinders on the Toledo Refinery LM-3 machines could have bore sizes as large as 20.5 inches, 17.75 inches, and 11 inches, respectively. Only the existing third stage was at its maximum size. Initially, the first stage cylinder liner bore in the C-9202 compressor was opened from the original 19.5 inches to 20.0 inches, a 6.8 percent increase in flowrate, because we wanted to make incremental changes in the compressor flowrate. The cylinder liner on the C-9203 compressor was then bored to 20.5 inches, a 12.4 percent flowrate increase.

During the review process, it was discovered that the minimum liner thickness acceptable to the compressor OEM differed slightly from that suggested in API 618 (1995), Section 2.6.2.3. Their standards were more liberal than those outlined by the API 618 Standard. After discussions with the compressor OEM, justification for the discrepancy was deemed acceptable. It is important to consider this during the engineering review. The manufacturer may have practices in place that do not agree with those outlined in the API standards; this should be identified where possible. While in theory, completely removing the liner and running on the virgin cylinder bore would allow the largest possible piston diameter and thus maximize the flow through the compressor, this was not considered due to the extreme risk involved. The purpose of the liner is to serve as a sacrificial wearing surface. Without it, a damaged cylinder bore would likely result in the need to procure a new cylinder body. The high cost and long lead time of doing so for cylinder bodies of this size make this option an unacceptable risk.

Considerations should be given to liner porosity long before any cutting begins in the field. Most liners are made of cast material; subsurface inclusions formed during the casting of the liner may appear during the machining process. Most nondestructive testing methods to identify such an inclusion are not practical in this instance. Because of concerns for process down time and replacement part lead time, a replacement liner was ordered well in advance of the compressor shutdown as insurance against finding subsurface inclusions, machining errors, and other unforeseen circumstances.

Capacity increases beyond boring the first stage cylinder to 20.5 inches required a change in one or more cylinder classes. These options are summarized in Tables 2 and 3. Options beyond 12.4 percent were not executed in the field during the time of the debottlenecking study. However, recommendations were made to fully utilize the available motor horsepower and modify the compressors to increase the capacity by 44.1 percent of the original design.

Piston Sizing and Crankshaft Balance

Compressor pistons are designed with relatively small diametrical clearance. There is little latitude available for reusing the existing piston when the cylinder bore is enlarged. New pistons were purchased for both the 20.0 and 20.5 inch cylinder modifications.

Increasing the unbalanced forces and couples acting on the machine foundation was a concern with the increased piston size. Depending on the internal design of the piston, design/manufacturing technique (cast versus fabricated), and the piston material, a substantial change in reciprocating weight on any given compressor throw can result in creating undesirable unbalanced forces and/or couples with potentially harmful effects. For this particular set of modifications, multipiece compressor pistons were chosen to replace the original first stage single piece cast iron piston. Besides ease of maintenance, one advantage of the multipiece piston is the fact the weight can be more easily controlled, even to the point of using different materials for the various sections.

During the piston change evaluation process, it is important to review and decide how to treat the compressor frame in the force balance analysis. The frame can be treated either as a rigid body or as a flexible body. A rigid body analysis is a more simplistic approach that sums the individual load components to arrive at an overall result. Opposing forces and couples are assumed to completely cancel each other with this approach. Typically the rigid body assumption is valid if no speed changes are being made, the unit does not operate near its speed limit, small changes in piston weight are made, and/or rod loading is not excessive. Whereas, a flexible body analysis is far more rigorous and complex and takes into account all the opposing forces and couples that may not necessarily completely cancel one another and therefore could influence the design. During the original design of the Toledo compressors, frame flexibility was included in the design and analysis.

For the 6.7 percent and 12.4 percent cases, the OEM considered the calculated design weight of the new larger multipiece pistons as compared to the original single piece piston, and concluded there would be no substantial change in the unbalanced forces and couples and that the more detailed approach to this calculation was not warranted. Since the existing foundations were adequately designed and in good enough condition such that the result had been satisfactory reliability since original installation, it was concluded satisfactory performance should continue to be achieved. Therefore, no further changes in reciprocating balance weight or crankshaft counterweight were deemed necessary. A more detailed analysis should be carried out for the future 44.1 percent increase option as it involves larger changes to the machine.

Assumptions and Safety Factors

It is very important to understand the assumptions and safety factors that are used in various calculations. A review of these assumptions and safety factors with the OEM may offer some opportunities that may be significant. The impact of pulsation damper pressure drop from assumed five percent to 2.5 percent on flow and horsepower has been discussed previously. As can be seen in Figure 2, the impact of the damper pressure drop on capacity is notable.

At the beginning of the project study, the refinery had provided the compressor OEM with expected operating data for the rerate conditions. In their analysis, the compressor OEM added some factors to account for unknowns and uncertainties. Below is a summary of the parameters that the compressor OEM had used in their compressor calculation model.

- 95 percent of the operating suction pressure
- 105 percent of the discharge pressure
- Five percent pressure drop across the suction and discharge pulsation dampers. After discussion, this was later changed to 2.5 percent.
- Three percent safety factor

Figures 5 and 6 illustrate the impact of suction pressure both on capacity and horsepower. Figure 7 shows the relationship between capacity and horsepower. As can be seen in Figure 5, the impact of suction pressure on capacity can be very significant. Therefore, it is strongly recommended to review and understand the assumptions and safety factors that are incorporated in the compressor rerate.

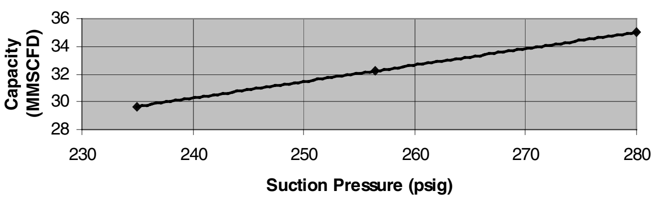

Figure 5. Suction Pressure Versus Capacity.

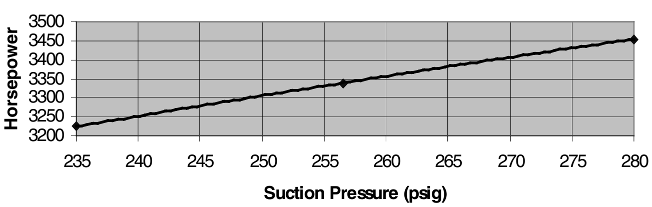

Figure 6. Suction Pressure Versus Brake Horsepower.

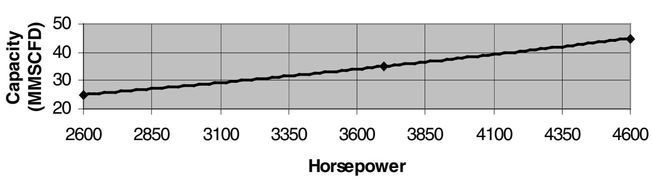

Figure 7. Brake Horsepower Versus Capacity.

MOTOR DESIGN CONSIDERATIONS

Each compressor is driven by a GE synchronous motor rated at 3500 hp, 300 rpm, 4000 V, and 0.8 pf. These are original motors installed in 1968.

API 618 (1995), Section 3.1.2.1, recommends the motor be sized to 110 percent of the compressor design horsepower. The 10 percent horsepower margin is adopted as safety margin to account for design uncertainties in both mechanical equipment and process design. Originally, the compressors were designed for 2735 hp and therefore required 3009 hp or larger motors. Obviously, the next frame size motors, 3500 hp, were selected to drive these compressors.

Compressor debottlenecking options requiring less than 3150 hp did not raise any concerns about the motor size. As options were developed that required more than 3150 hp, questions related to the API standard, safety, and reliability were raised. To understand and address these concerns, compressor and motor performance evaluations were carried out. Since the equipment has been in service for the last 30 years, their performance was established and most of the design uncertainties have been understood and eliminated.

Along with a process review, three independent methods were used to evaluate performance and long-term reliability of the motors. These were:

- Winding temperature monitoring,
- An online partial discharge test, and
- A polarization index test conducted during compressor maintenance outages.

Understanding the process unit took careful review of available data and good communication with personnel closest to the operation of the plant. As a first pass review, it was easy to determine where the motor was operating with respect to the original specified conditions. Motor winding temperature, amperage, horsepower, and various process variables were gathered from the historical database. The data were then plotted and analyzed for motor evaluation purposes.

Motor amperes and winding temperature rise over a two year period were obtained from the plant process database to ascertain their historical relationship and relative values. Winding temperature rise increased with amperage as expected and were found to be within acceptable levels when compared with maximum allowable winding temperature rise and maximum amperage (see Figures 8 and 9). Absolute winding temperature was also reviewed and found to be acceptable.

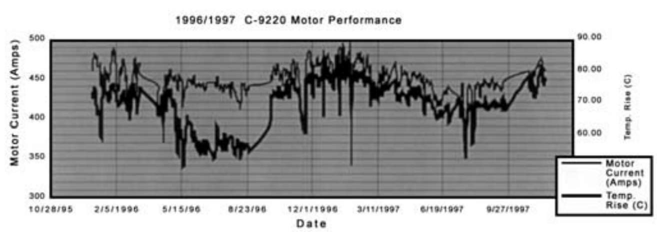

Figure 8. Time Versus Motor Current and Temperature Rise.

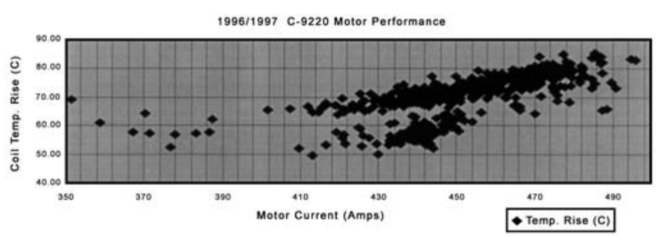

Figure 9. Motor Temperature Rise Versus Motor Current.

An online partial discharge test was carried out by Westinghouse to evaluate the motor integrity. They market this Russian technology. This test evaluates the condition of the motor insulation by measuring partial discharge. The partial discharge test found the motor insulation to be in good condition.

Offline testing of the motor windings was another step taken to ensure the electrical integrity of the motor. Polarization and winding resistance tests resulting from past and recent inspections confirmed the motors were in good condition. Electronic equipment was installed in both motors to allow online monitoring of motor condition.

After carefully reviewing API 618 (1995) and confirming with other compressor users, it was concluded there was no need to maintain the additional 10 percent horsepower margin anymore. Therefore, it was possible to consider options that reduced the 10 percent horsepower margin. Furthermore, it was decided to take a conservative approach and make small incremental changes to compressor capacity and closely monitor the results.

Power Factor

The original motor pf was 0.8. Changing the pf to 1.0 would allow the motors to operate at a higher horsepower for a given amperage. Refer to Figure 10 for a graphical explanation of the pf adjustment. Point "A" denotes where the motor was originally designed to operate, 100 percent full load amperage at 0.8 pf. By adjusting the pf, the operation of the motor at full load moves down the "full load" curve to the "unity pf" line, at point "B." The motor now draws only 80 percent of full load amperage. Loading the motor further by moving on the "unity pf" line, back to 100 percent full load amperage at point "C," it is now able to run at 25 percent higher than rated horsepower of 3500 hp at 1.0 pf. This increase in horsepower may also be expressed mathematically by the following equation:

$$HP = \frac{\sqrt{3} \cdot (\text{Volts}) \cdot (\text{Amps}) \cdot (\text{eff}) \cdot (\text{pf})}{746} \qquad (2)$$

where *HP* is the available horsepower of the motor, *pf* is the power factor of the motor, *Volts* is the line voltage at the motor leads, *Amps* is the rated amperage, and *eff* is the design efficiency.

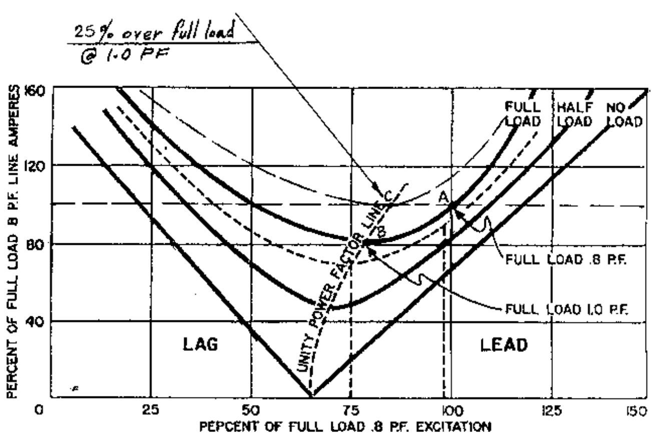

Figure 10. Power Factor Curve.

Of course, the available extra horsepower is not without penalties. For the subject motors, the pullout torque would be reduced by 20 percent. Furthermore, the change in the pf would have an impact on the refinery's overall power balance that would require the installation of additional capacitors to maintain the power balance.

The motor OEM confirmed the potential for an additional 25 percent of original nameplate horsepower output by changing the motor pf to unity. Therefore, by operating the motors at 1.0 pf, the existing motors can be operated at 125 percent of the original nameplate, or 4375 hp, with no capital investment in the motor and minimal capital investment in the electrical utilities.

Other options to increase the motor rated horsepower were also investigated. They included adding additional copper to the stator windings, increasing rotor size, and increasing the insulation grade. No latitude was available with these options.

PROCESS EQUIPMENT CONSIDERATIONS

The associated process equipment effected by the increase in compressor capacity was also evaluated for design limitations. The equipment was reviewed for acceptable pressure drop and flow velocity; included were heat exchangers, knockout drums, cooling water headers, and process headers. The equipment was found to be adequate and acceptable for the 6.7 percent and 12.4 percent flowrate increase options that were to be executed. However, for the 44.1 percent flowrate increase option, limitations were identified in the water side of the first stage intercooler. The required increase in the water flowrate resulted in an unacceptable pressure drop across the exchanger and would require a larger replacement or an additional exchanger in parallel.

Cooling water requirements for the compressor cylinder water jackets were also reviewed. The existing exchanger could handle the additional heat of compression for the short-term compressor rerate options. However, the larger compressor rerates, greater than 12.4 percent increase in flowrate, would require additional cooling capacity and a larger exchanger.

ECONOMICS

The strong economic incentives derived from the volume gain of the HCC complex demand that the facility be fully utilized at all times. During the life of this complex, the throughput of the HCC has been increased by various means. In 1997, the main limitation of obtaining more throughput or conversion to the HCC was the capacity of the hydrogen makeup compressors. The HCC complex had more hydrogen generating capacity than the HCC could utilize through its makeup compressors. Debottlenecking the makeup compressors would allow either more feed to be processed or more conversion to be achieved in the HCC.

The capacity of the C-9202 makeup compressor was first expanded to 106.7 percent of original design in December 1997. The C-9203 makeup compressor was expanded to 112.4 percent of original design capacity in February 1998, primarily based on the success of the first revamp. In January 2000, the C- 9202 makeup compressor was also expanded to 112.4 percent of original design to make both compressors the same. The total cost to perform these two revamps was approximately $300,000.

A post audit of these revamps was conducted in 1998 and the increases in hydrogen compressor capacity were documented. The refinery has since taken advantage of the expanded compressor capacity. The capacity increase has allowed the refinery to process additional barrels of feed through the HCC and has resulted in an increased profitability to the refinery of approximately $3.1 million per year.

With the final expansion of the C-9202 makeup compressor to 112.4 percent of its original design flowrate, the refinery is essentially in-balance with hydrogen production capacity and hydrogen compression capacity.

CONCLUSIONS

Reliability is designed into compressor systems with the guidance of industry standards and best practices. By carefully reviewing all design aspects of the compressor rerate and considering the intention of industry standards and best practices, we were able to harness some of the "design fat" in our compressor system to provide additional capacity to the unit from existing equipment.

Working directly with the compressor manufacturer provided valuable insight into the design assumptions. The original design assumptions and guidelines were scrutinized and changed to better match the current operation and performance. Measuring and evaluating both the equipment and unit performance allowed design margins to be reduced or eliminated. However, consideration was given in many areas of compressor design to ensure reliability was not compromised when the equipment was pushed to its design limits and beyond.

The refinery took an unconventional approach in developing and implementing the compressor rerate options. Reliability engineering, with the help of process engineering and technical services, took the lead in identifying, defining, pursuing, justifying, and executing the options. Additional options were being defined as the first and second rerate options were being executed in the field.

The smaller and incremental step rerate options were chosen and executed in the field for the following two reasons:

- *Minimize risk*—These compressors were the key to the HCC complex operation and had been very reliable. Therefore, it was decided to minimize the risk and proceed with smaller incremental steps. *To maintain or improve the reliability of the compressors was a must.*

- *Minimize cost*—The project was sold to management on the basis of low cost and high return. The low cost objective was met by making modifications during the maintenance overhaul windows.

By debottlecking both the C-9202 and C-9203 compressors by 112.4 percent of their design flowrate, the Toledo Refinery has added $3.1 million per year directly to their bottom line. Both of these modifications have proven very successful.

From a detailed review of the compressor motors, it was concluded that these motors can be operated at 4375 hp at 1.0 pf instead of the original design of 3500 hp at 0.8 pf.

Finally, the team made a recommendation to implement the 44.1 percent flow increase option in the future. This will require a larger effort as it will involve a review of the entire HCC complex to make other debottlenecks to take advantage of the compressor capacity. The refinery has put together a ream to study the HCC complex for the next turnaround.

REFERENCES

API Standard 618, 1995, “Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services,” Fourth Edition, American Petroleum Institute, Washington, D.C.

ACKNOWLEDGEMENT

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