Centrifugal Compressor Operations

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Understanding the interaction between fundamental principles of centrifugal compression, operating changes that influence compressor performance and basic control is important. Since the wet gas compressor plays a central role in FCC operations, it will be used to review compressor performance fundamentals, common operating conditions that influence operations and basic control philosophy.

Wet Gas Compressor

The FCC wet gas compressor's major function is reactor pressure control. The machine must compress gas from main column overhead receiver to gas plant operating pressure while maintaining stable regenerator-reactor differential pressure (Figure 1). Typically reactor-regenerator differential pressure must be controlled within a relatively narrow +2 to -2 psi (+0.14 to -0.14 bar) range to permit stable catalyst circulation. The wet gas compressor and its control system play a vital role in maintaining steady reactor operating pressure. Even though optimum FCC operations require balancing regenerator and reactor pressures against wet gas and air blower constraints, throughout this paper reactor pressure is presumed constant to simplify discussions.

Reactor operating pressure is regulated by main column overhead receiver pressure and system pressure drop from the reactor to the overhead receiver. The wet gas machine needs to have sufficient capacity to compress receiver wet gas to the gas plant operating pressure. Reactor effluent composition, overhead receiver pressure and temperature, and gasoline endpoint all influence the amount of wet gas and its molecular weight. Variability in main column overhead receiver pressure or unstable system pressure drop produce reactor pressure swings. These can cause catalyst circulation problems and other operability concerns.

Reactor Pressure

Reactor operating pressure is set by main column overhead receiver pressure and system pressure drop. System pressure drop depends on equipment design and operation, while compressor and control system performance set receiver pressure. Wet gas compressors operate at fixed or variable speed. Fixed speed compressors throttle compressor suction while variable speed machines use steam turbines or variable speed motors to control receiver pressure. If necessary, compressor surge control systems recycle gas to ensure inlet gas flow rate is maintained above the minimum flow (surge point or line). Even when receiver pressure is controlled, rapid system pressure drop changes from tray flooding and dumping will cause rapid changes in reactor pressure.

Most motor driven compressors operate at fixed speed using suction throttle valves to vary pressure drop from the main column overhead receiver to the compressor

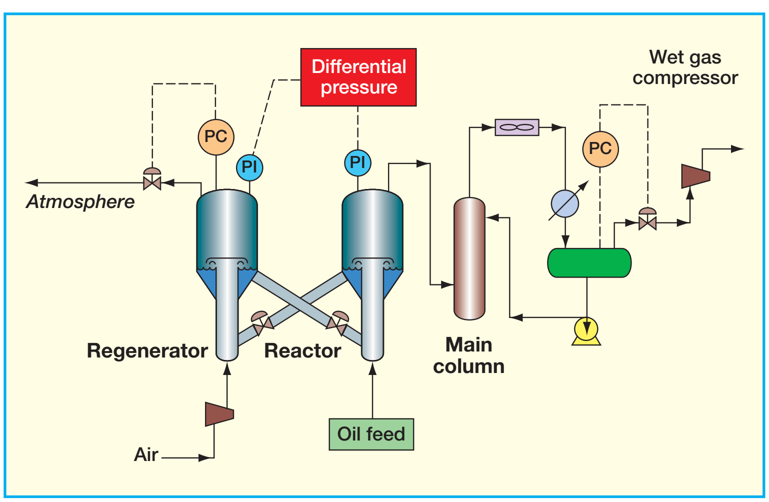

Figure 1. *Regenerator-reactor differential pressure*

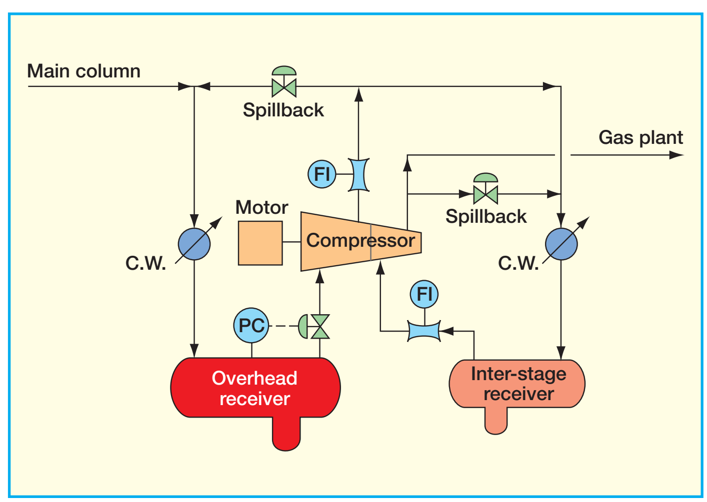

Figure 2. *Fixed speed compressor and inter-condenser system*

nlet (Figure 2). The pressure controller manipulates the throttle valve position and pressure drop to maintain constant receiver pressure. Normal system pressure drop variation is slow and predictable, thereore, receiver pressure can be adjusted to maintain constant reactor pressure. As ong as the throttle valve is not fully open, hen the compressor has excess capacity. Once the throttle valve is fully open and spillback valve closed, the machine can no onger compress wet gas flow to the gas plant operating pressure. Generally, reacor temperature or feed rate is reduced to permit the compressor throttle valve to egain pressure control so that flaring can be avoided.

Variable speed compressors use steam urbines or variable speed motors to conrol receiver pressure. Speed is adjusted to change the operating point on the compressor map to meet the system flow-head equirements for stable reactor pressure control. As system pressure drop increases

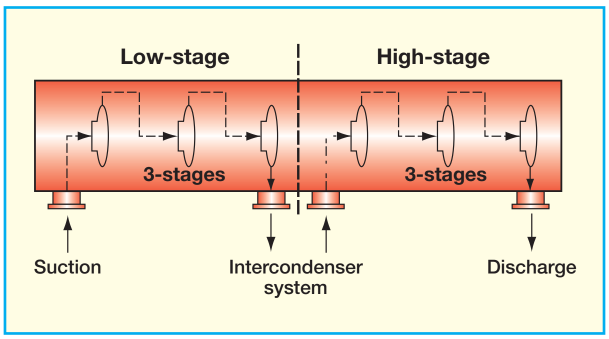

Figure 3. *Compressor schematic with inter-condenser*

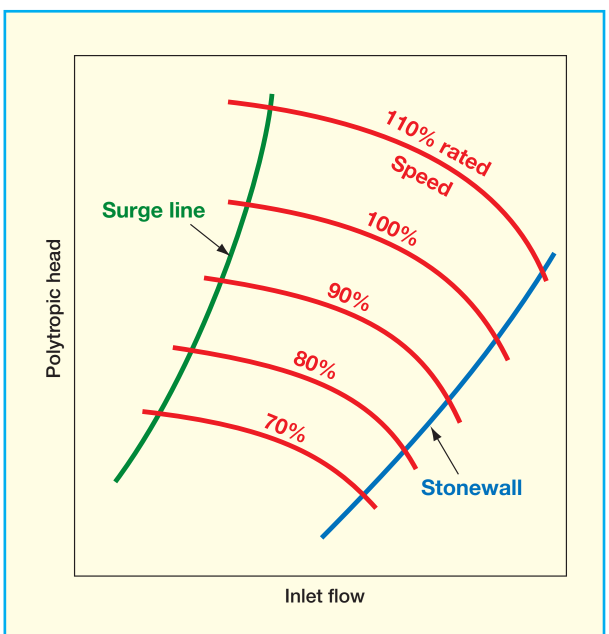

Figure 4. *Variable speed flow-head map*

receiver pressure is reduced, thus machine speed must be increased to compress the higher gas flow rate and to meet higher head requirements. Once the turbine governor is wide-open or variable speed motor is operating at maximum speed or amps, feed rate or reactor temperature must be reduced to lower wet gas rate to the compressor capacity.

Fixed or variable speed motors and turbines must have sufficient power to compress the mass flow rate of gas while meeting the differential head between the overhead receiver and the gas plant. Otherwise, reactor temperature or feed rate must be reduced to decrease the amount of receiver wet gas flow to the driver limit.

Compressor Design

Wet gas machines use 6-8 impellers (stages) to compress gas from the main column overhead receiver to the gas plant operating pressure. Most have inter-stage condensing systems after the first 3 or 4 stages (low-stage) that cool the compressed gas, condense a small portion and separate the gas and liquid phases (Figure 3). Inter-stage receiver gas is then compressed in the last 3 or 4 stages (high-stage). Inter-stage condensers reduce gas temperature and raise compressor efficiency by 5-7%, but they also generate pressure drop. Separate flow-polytropic head and flow-polytropic efficiency curves are needed to evaluate overall compressor system performance.

These curves have inlet gas flow rate on the X-axis and polytropic head developed and polytropic efficiency on the Y-axis. Consequently overall compressor performance and power consumption depend on each compressor sections' performance curves and the effects of the inter-condenser system. Evaluating overall performance of these compressors is more complex than for a machine without inter-cooling, but fundamentally the same.

Some compressors do not have inter-condenser systems. A single flow-polytropic head and flow-polytropic efficiency curve represent overall performance. The machines have lower efficiency and the gas temperature leaving is generally near 300°F rather than 200°F with an intercooled design. These machines must compress all wet gas from inlet conditions to the gas plant operating pressure hence power consumption is higher.

Wet Gas Compressor: Stable Operating Range

Each compressor section must be operated within its stable flow range. At fixed speed the compressor curve begins at the surge point and ends at stonewall, or choke flow. Surge point is an unstable operating point where flow is at minimum. At surge, the compressor suffers from flow reversals that cause vibration and damage. At the other end of the curve is the choke (or stonewall) point. At the choke point, the inlet flow is very high and the head developed very low. Flow through the machine approaches sonic condition or a Mach number of 1. Polytropic efficiency also drops rapidly near stonewall.

For variable speed compressors, there is a region between the surge and stonewall lines where there is stable machine performance (Figure 4). The compressor flow-polytropic head can be varied anywhere within this region. Because there is no throttling, all power goes into compression which minimizes power consumption.

Stable compressor performance is defined between these two flow-head limitations. Some machine designs can vary flow by 25% or more between surge and stonewall points, while others have only 6-8% flow variation between these limits. Compressors with small stable flow regions need to have robust surge control systems. The head-flow curve basic slope is relatively flat near the surge point and becomes steeper as inlet flow is increased. The impeller blade angle determines the shape of the curve and the compressor efficiency.

Basic Compressor Control

Reactor yield and condenser operating temperature and pressure change throughout the day, hence the gas rate from the main column receiver is variable. Consequently the compressor control system must be capable of maintaining constant receiver pressure. Thus fixed speed compressors have suction throttle valves and variable speed machines change speed to compensate for gas rate changes. Because the compressor inlet gas flow rate is not constant and may be below the surge point or line, the compressor is typically designed with a surge control system.

Surge control ensures that inlet flow rate is maintained above minimum (surge point or surge line) at all times. A flow meter in the compressor suction or discharge and inlet temperature and pressure are used to calculate the actual flow rate (ICFM) into the low- and high-stages of compression. As suction flow (ICFM) decreases toward the surge line (or point), the spillback control valve opens to recycle gas from discharge to suction to raise inlet flow rate. Spillback flow is kept at minimum to reduce power consumption. Compressors with inter-condensers need two independent spillback systems from discharge to the suction of each section. Spillback streams should be routed in front of upstream exchangers so the heat of compression is removed (Figure 2).

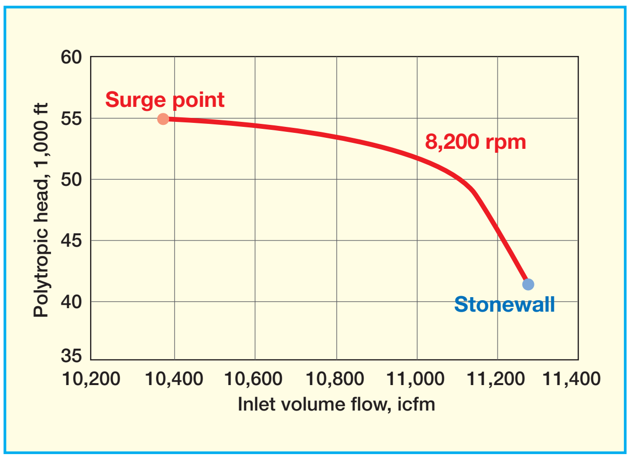

Figure 5. Flow-head curve for a 6-stage compressor

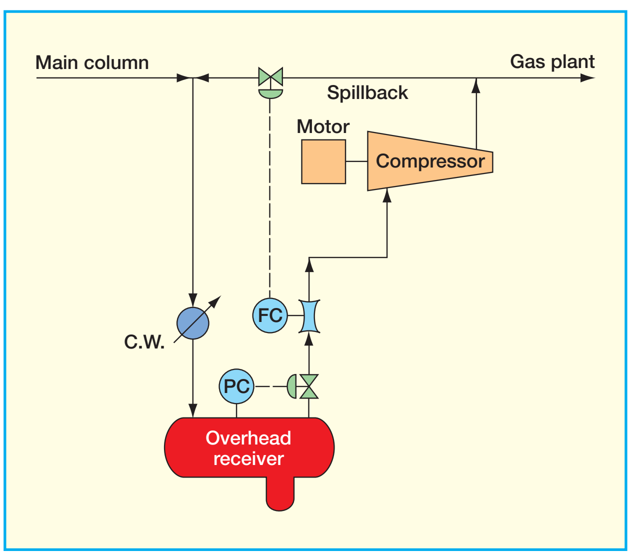

Figure 6. Receiver pressure control- compressor suction throttling

Wet Gas Compressor Operation: No Inter-condenser

The simplest compressor to evaluate is a fixed speed machine with no inter-condensers. It has a single flow-head curve (and single flow-efficiency curve) with a surge point rather than a line.

Although molecular weight does affect developed head, typical molecular weight variations in a gas oil cracker do not materially change the compressor flow-head performance curve. Whereas, resid crackers processing varying amounts and quality of residues may have as much as 8 number variations in molecular weight, thus the flow-head performance curve is affected. The manufacturer should provide curves at maximum and minimum molecular weight.

Figure 5 is the performance curve for a 6-stage FCC wet gas compressor that will be discussed in the case study presented later in this paper. The compressor develops a fixed polytropic head for a given inlet flow rate and the curve can be used to predict compressor performance at different process conditions.

The suction throttle valve plays an important role for a fixed speed motor driven compressor. Throttle valve pressure drop controls overhead receiver pressure (Figure 6) so that reactor pressure is stable. Throttle valve position and pressure drop compensate for changes in receiver gas flow rate or receiver pressure set point changes. Because compressor discharge pressure is held constant by the gas plant pressure controller, suction pressure will vary and follow the flow-head curve. When gas rate leaving the overhead receiver is higher than flow at the surge point, the spillback is closed. Hence, compressor suction pressure will ride up and down the flow-head curve as long as the throttle valve is generating pressure drop and not fully open. As compressor inlet flow rate approaches the surge point, the spillback valve opens recycling gas to ensure sufficient inlet flow into the machine. When the spillback is open, spillback flow rate determines the operating point on the curve. Flow rate must always be maintained above the surge point with suction pressure determined by the polytropic head generated at the minimum flow control point.

Since the amount of gas leaving the overhead receiver depends on reactor effluent composition and overhead receiver conditions, the compressor suction pressure will vary. As long as the suction throttle valve is not fully open, the compressor has unused capacity.

When establishing operating conditions to stay within an existing machine capacity or if considering a revamp, determining the compressor suction pressure needed to meet the proposed operation is critical. Compressor suction pressure is calculated from the compressor performance curve. Because the flow and head terms are affected by suction pressure, estimating this pressure is an iterative process. Centrifugal compressors generate a fixed polytropic head at a given inlet flow rate and not a fixed discharge pressure, with suction pressure, gas molecular weight and gas temperature all influencing both inlet flow rate and polytropic head. The polytropic head equation is shown in Equation 1.

Equation 1

$$\text{Head} = \frac{1{,}545}{MW}\;\frac{Z_{avg}T_1}{\left(\frac{n-1}{n}\right)}\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n}{n-1}\right)} -1\right]$$

Where: MW = Molecular weight
Z_{avg} = Average compressibility
T_1 = Suction temperature, °R
n = Compression coefficient
P_1 = Suction pressure, psia
P_2 = Discharge pressure, psia

Understanding each variable's impact on inlet flow rate and polytropic head is important. Molecular weight and suction pressure have a significant influence on performance, while compressor discharge pressure (P_2) is fixed and temperature effects are small. Gas molecular weight (MW) is primarily controlled by reactor effluent composition. As molecular weight decreases the inlet flow rate increases. Because the compressor discharge pressure is fixed, compressor suction pressure must be high enough to generate the head corresponding to the inlet flow rate into the compressor. For a fixed speed compressor, as long as the compressor throttle valve is not fully open, it has unused capacity. Thus, changes in molecular weight simply cause throttle valve position and pressure drop to adjust to maintain receiver pressure. But once the throttle valve is fully open, the machine is operating at maximum capacity.

Inlet gas temperature has little influence on compressor capacity because it is based on absolute temperature. Hence, a 20°F ise in temperature changes the head term by only 3% and the flow term a similar amount.

Compressor suction pressure has a large nfluence on inlet gas flow rate. For a fixed mass flow rate, raising suction pressure decreases inlet volume. At constant receiver pressure, compressor inlet pressure is determined from the flow-head curve for a ixed speed compressor. Using the compressor curve shown in Figure 5, the sucion pressure will be that needed to satisfy he inlet flow and head term simultaneousy. Throttle valve pressure drop will vary to maintain inlet flow rate between 10,400 and 11,100 ICFM while meeting the gas plant discharge pressure. As long the eceiver gas flow rate is above the surge point, then the spillback valve will be closed. However, as gas flow approaches surge, the spillback valve opens to mainain flow in a stable region of the curve. Suction pressure is a dependent variable as long as the throttle valve has pressure drop. Once the control valve is wide open he gas rate must be reduced or the suction pressure increased to reduce the inlet volume into the compressor.

Variable speed compressors have an operating region between the surge and stonewall lines (Figure 4). Because there is no throttle valve, main column overhead eceiver pressure controller varies speed to maintain receiver pressure. As long as the eceiver gas flow rate is above the surge ine flow rate, there will be no spillback. However, if receiver gas flow is below the surge line, then the spillback will open to meet the minimum flow. In Figure 4, maxmum compressor capacity occurs when he compressor is operated at maximum speed. As long as the compressor driver has sufficient energy then minimum overhead receiver operating pressure will be based on the flow-head developed at 110% of the rated speed for the curve shown in Figure 4.

Compressor Capacity: Driver Power

Compressor power consumption is a function of the mass flow, polytropic head, polytropic efficiency, and gear losses. Compressor shaft horsepower (SHP) is shown in Equation 2:

Equation 2

$$\text{Compressor SHP} = \left[\left(m\, H_p\right) / \left(n_p\, 33{,}000\right) \right] 1.02$$

Where:
H_p = Polytropic head
n_p = Polytropic efficiency
SHP = Shaft horsepower
m = Mass flow rate of gas
1.02 = 2% gear losses

Fixed speed compressors have suction throttle valves to control overhead receiver pressure. Satisfying the flow-head curve requires pressure drop across the control valve and throttling always wastes energy. Because variable speed compressors change speed to match the process flow-head requirements, they consume less energy. However, the capital cost of steam turbines or variable speed motors is higher than a fixed speed motor.

Wet Gas Compressor Operation: With Inter-condenser

Figure 3 is a schematic of a 6-stage compressor with 3-stages (Figure 7, low-stage) in front and 3-stages (Figure 8, high-stage) behind the inter-condenser. Gas must still be compressed from the overhead receiver pressure to the gas plant operating pressure, however, the low-stage discharges intermediate pressure gas (65-90 psig) to an inter-condenser where it is cooled from approximately 200°F to 100-130°F depending on whether air or cooling water

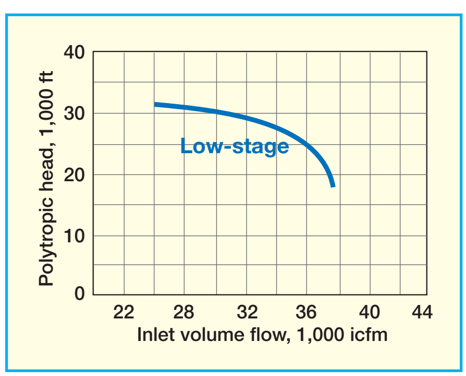

Figure 7. Low-stage curve

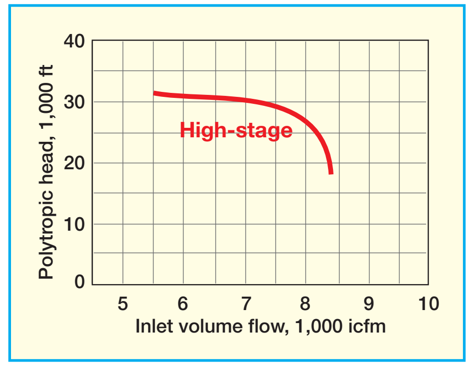

Figure 8. High-stage curve

is used. The condenser outlet stream is sent to a separator drum where the gas and liquids (oil and water) are separated. Interstage drum gas is fed to the high-stage section of the compressor. Each section has its own flow-polytropic head and flow-polytropic efficiency curves.

Because the inter-condenser system condenses a portion of the low-stage gas, mass rate and gas molecular weight are lower into the high-stage(typically 4-5 numbers lower than main column overhead receiver gas). This reduces compressor power consumption and allows for more efficient compressor design. Because the low-stage and high-stage sections have separate operating curves, each requires an independent surge control system. One part of the compressor may be operating with spillback to avoid surge, while the other may have the spillback closed.

Fixed speed compressors with an intercooler have a suction throttle valve into the low-stage to control main column overhead receiver pressure while high-stage discharge pressure is controlled by the gas plant sponge absorber or amine contactor pressure controller. Typically there is no throttle valve in the suction of the high-stage, consequently, discharge pressure from the low-stage and suction pressure to the high-stage are dependent variables.

Inlet flow rate into the low- and high-stages must always be above the surge point. Separate spillback systems from the low-stage discharge to inlet of the main column condensers and from the high-stage discharge to the inlet of the inter-condenser maintain flow above surge for both compressor sections. Thus, suction and discharge pressure from the first 3-stages and suction and discharge pressure of the last 3-stages are dependent on each other, inter-condenser system pressure drop, and the amount of material condensed.

While principles of compression are the

same, overall performance is more difficult to evaluate. Fortunately, process flow models, such as Simsci ProII® or Provision®, allow the low- and high-stage flow-polytropic head curves and flow-polytropic efficiency curves and the inter-condenser system to be rigorously modeled. Thus interstage operating pressures can be determined through an iterative process without excessive calculations.

Case Study: Minimizing Wet Gas Compressor Modifications

An FCC unit was revamped to increase unit capacity by 40%. In addition, gas plant limitations required undercutting gasoline to produce heavy naphtha from the main fractionator to reduce liquid loading through the gas plant. The existing compressor was a 7-stage machine with no inter-condenser and a 4400 horsepower motor. A suction throttle valve controlled main fractionator overhead receiver pressure. Prior to the revamp reactor pressure was 20 psig with a compressor inlet pressure of approximately 4-5 psig. The 40% higher feed rate and higher gas from undercutting together would have increased inlet flow rate by more than 50% if the 4 psig suction pressure was maintained. Thus, a new parallel compressor or one new larger compressor would have been required. Because this solution was high cost, alternates were considered.

When evaluating compressor performance, the complete system from the reactor thorough the compressor outlet needs to be evaluated as a single system. Undercutting gasoline increases wet gas flow rate if no other changes are made because it decreases the amount of liquid product from the main column overhead receiver. Gasoline "sponges" C_3+ hydrocarbons leaving the condenser, thus undercutting reduces sponging at fixed receiver temperature and pressure. For instance, undercutting 10% of the gasoline raises wet gas flow rate by approximately 5-7%. As the percent undercutting increases so does the amount of wet gas from the overhead receiver.

Increasing mass flow rate through the compressor without large changes in the inlet flow rate requires higher suction pressure. System pressure drop from the reactor to the compressor inlet must be reduced. In this case, the reactor pressure operated at approximately 20 psig and the system pressure drop was 16 psi, resulting in a 4 psig compressor suction pressure.

Table 1

Reactor system pressure drop

Components	ΔP, psi
Reactor cyclones	2
Reactor vapor line	1
Reactor line coke	4
Main column	3
Condenser	3
Piping	2
Flow control/metering	1
Total	**16**

High pressure drop components included reactor line coke formation, main column overhead system and column internals pressure drop. Table 1 shows each major component and its measured pressure drop. Substantial reduction in system pressure drop was required to lower compressor inlet flow rate to within 10,400 to 11,200 inlet cubic feet per minute stable operating range.

Increasing compressor suction pressure raises condensation in the overhead receiver, decreases inlet flow rate and increases compressor capacity. In this instance, main column overhead receiver pressure and temperature needed to be 12 Psig and 100°F to maintain inlet flow rate within stable operating range. Coking in the reactor line, main column pressure drop, piping, flow metering, and condenser pressure drop all had to be reduced. Otherwise, a new parallel compressor would have been needed.

Overhead condenser exchanger surface area, cooling water (CW) flow rate, and CW temperature all influence wet gas flow rate. Each 1°F reduction in receiver temperature lowers wet gas rate by approximately 1%; thus, decreasing temperature by 10°F lowers the wet gas rate by about 10%. Although temperature has little effect on gas volume, main column overhead receiver temperature has a large impact on condensation. Cost-effective changes that reduce receiver temperature should always be considered to maximize existing compressor capacity or to minimize the need for modifications.

In this instance, raising receiver pressure from 5 to 12 psig and lowering temperature to 100°F decreased the amount of wet gas produced so that the gas inlet flow rate was within the existing compressor capacity (Figure 9). While raising compressor inlet pressure (and lowering temperature) decreased inlet flow rate, it also raised compressor discharge pressure. Since centrifugal compressors develop fixed polytropic head, discharge pressure (P_2 in equation 1) from the existing 7-stages of compression would have been 350 psig,

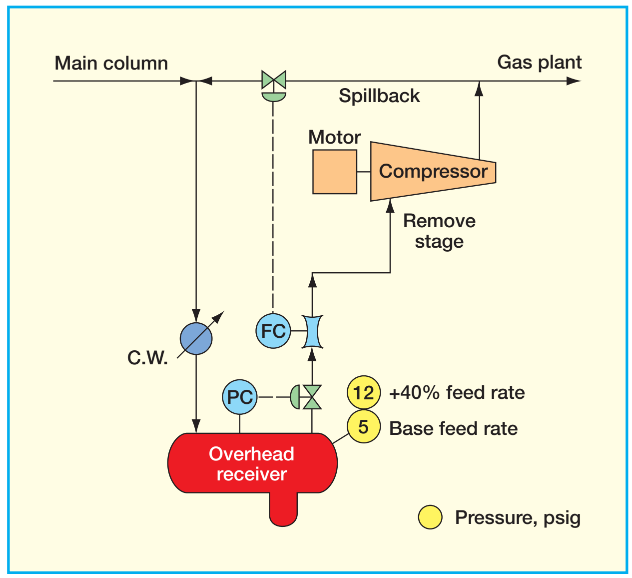

Figure 9. Receiver operating pressure- before and after revamp

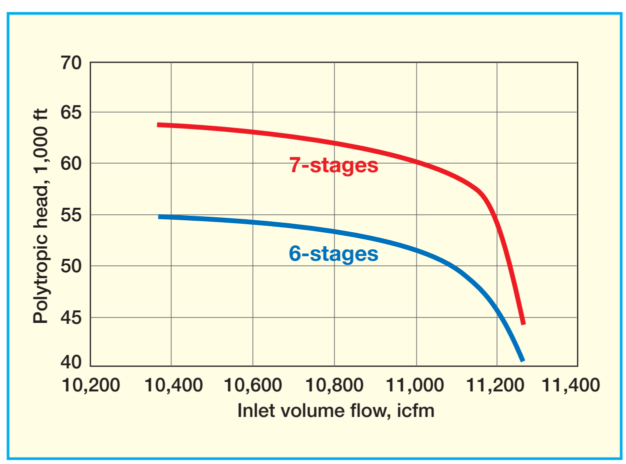

Figure 10. Flow-head curve for 7- and 6- stage compressor

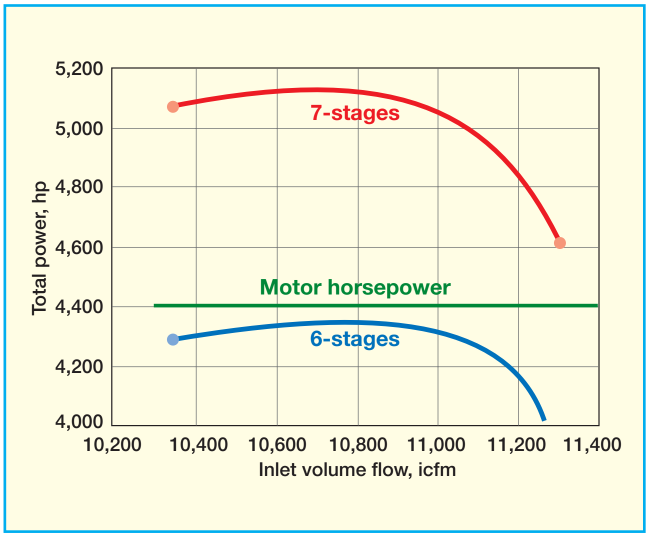

Figure 11. Flow-power requirements for 7- and 6- stage compressor

which exceeded the maximum allowable working pressure (MAWP) of the major equipment in the gas plant. Hence, compressor head had to be reduced.

Compressor power requirements also have to be considered. Both the flow-head and flow-polytropic efficiency curve is needed to evaluate power consumption. Power requirement would have been more han 5,100 horsepower due to the higher mass flow rate; hence, the existing 4,400 horsepower motor would have to be eplaced. But replacing a motor or steam urbine can be very costly because it often equires major utilities system modifica-ions as well.

Although it is compressor OEM or compressor expert who should do comprehensive compressor analyses, the revamp engineer can perform a preliminary review by making simplifying assumptions such as equal head rise per stage of compression to determine whether removing a stage of compression or trimming the mpellers is needed.

In this instance, the 7-stage compressor developed about 9,500 feet of head rise per stage of compression. Thus eliminating a stage would reduce developed head by about 9,500 feet. Because compressor discharge pressure needed to be approximately 210 psig to meet gas plant operating pressure and to stay below the 250 psig pressure relief valve (PSV) settings, the required polytropic head at 12 psig suction and 210 psig discharge pressure was approximately 46,000 feet of head. Therefore, destaging was a practical solution. Figure 10 shows the comparison between the flow-head curves for the original 7-stage machine and a destaged 6-stage compressor generated by the OEM.

Compressor drivers must also be able to supply the power requirement throughout the stable operating range of the compressor. Power consumption depends on mass flow rate through the machine, polytropic head developed and compressor polytropic efficiency. Flow-polytropic efficiency curves are supplied by the manufacturer and allow power consumption throughout the stable flow range to be easily calculated. Figure 11 shows a comparison between the power requirements of a 7- and 6-stage compressor. By eliminating one stage of compression, machine horsepower requirements were reduced below the existing 4,400 horsepower motor.

Though centrifugal machines are used in many refinery units to compress gases, the principles of compression are the same irrespective of the process units. Only the gases compressed and the process variables are different. Understanding centrifugal compressor flow-head and flow-efficiency curves is essential when evaluating process changes or when revamping.

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