CENTRIFUGAL COMPRESSOR APPLICATIONS—UPSTREAM AND MIDSTREAM

by

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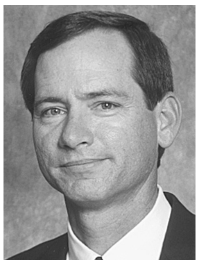

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ABSTRACT

Centrifugal compressors are used in a large number of different compression applications. This tutorial addresses applications for centrifugal compressors in the upstream and midstream sector, such as LNG compression, refrigeration, pipeline compression, compression for gas injection, gas lift, gas gathering, export compression, air compression, and others.

The focus is on the specific compressor requirements for these upstream and midstream applications, not only regarding the design of the compressor, but also its performance characteristics and control requirements. Specific challenges from different process gases are addressed.

To understand the interaction of the compressor with the system, different control mechanisms (speed, variable vanes, recycle, throttling) are explained, as well as the requirements for antisurge control.

GENERAL CHARACTERISTICS OF CENTRIFUGAL COMPRESSORS

The performance of centrifugal gas compressors is best displayed in a map showing isentropic efficiency and isentropic head as a function of the actual inlet flow. (The authors prefer isentropic head and efficiency to polytropic head and efficiency because the isentropic head can be directly derived from station design parameters [i.e., gas composition, suction temperature and pressure, discharge pressure], whereas the definition of the polytropic head requires the additional knowledge of the compressor efficiency or the discharge temperature. In particular, two compressors for the same process conditions will show the same isentropic head, but different polytropic head if their efficiency is different.) The means of control are added parameters. Figure 1 shows the map of a speed controlled compressor. Such a map also needs to define the operating limits of the compressor.

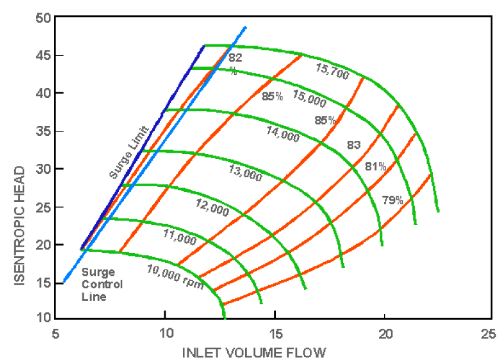

Figure 1. Typical Compressor Map (Variable Speed).

The limitation for lower flow is the surge limit. Some manufacturers also limit the operation of their machines in the choke region, while others allow the operation of their machines anywhere in choke, as long as the head remains positive. Other limits include the maximum and minimum speed, limits of vane settings, temperature limits and others. The head-flow map does not automatically define the temperature limits of the compressor because the discharge temperature depends also on the gas composition and the suction temperature. With the information in the head-flow-efficiency map and known suction conditions, it can be calculated. The speed limits are either rotor dynamic limitations or stress limits. It must be noted that a performance map as described will not change even if the inlet conditions are changed within limits.

THERMODYNAMICS OF GAS COMPRESSION

For a compressor receiving gas at a certain suction pressure and temperature, and delivering it at a certain output pressure, the isentropic head represents the energy input required by a reversible, adiabatic (thus isentropic) compression. The actual compressor will require a higher amount of energy input than needed for the ideal (isentropic) compression. This is best illustrated in a Mollier diagram (Figure 2). Head is the amount of work one has to apply to affect the change in enthalpy in the gas. (Physically, there is no difference between work, head, and enthalpy difference. In systems with consistent units [such as the SI system], work, head, and enthalpy difference have the same units [e.g., kJ/kg in SI units]. Only in inconsistent systems [such as US customary units], one needs to consider that the enthalpy difference [e.g., in BTU/lb_m] is related to head and work [e.g., in ft lb_f/lb_m] by the mechanical equivalent of heat [e.g., in ft lb_f/BTU].)

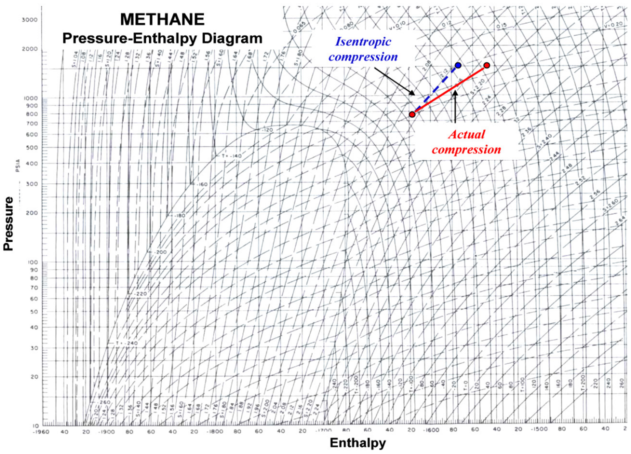

Figure 2. Compression Process in a Mollier Diagram for Methane.

The compressor head (H) can be determined from the suction and discharge pressure and temperature, assuming the gas composition is known. The compressor is usually assumed to be an adiabatic system (otherwise, neither the isentropic nor the polytropic work and efficiency definitions are very useful). Intercooled compressors can only be considered piecewise adiabatic. The relationship between the pressure, temperature, and enthalpy (h) are defined by the equations of state described below. By using the equations of state (Kumar, et al., 1999), the relevant enthalpies for the suction, the discharge, and the isentropic discharge state can be computed.

The isentropic head (H*) is:

$$H^* = h(p_d, s(p_s, T_s)) - h(p_s, T_s) \tag{1}$$

with the isentropic efficiency,

$$\eta_s = \frac{H^*}{H} \tag{2}$$

the actual head (H), which defines the power requirement as well as the discharge temperature, is:

$$H = \frac{H^*}{\eta} = h(p_d, T_d) - h(p_s, T_s) \tag{3}$$

It should be noted that the polytropic efficiency is defined similarly, using the polytropic process instead of the isentropic process for comparison. The actual head, which determines the absorbed power, is not affected by the choice of the polytropic or isentropic process. In order to fully define the isentropic compression process for a given gas, suction pressure, suction temperature, and discharge pressure have to be known. To define the polytropic process, in addition either the polytropic compression efficiency, or the discharge temperature have to be known.

In particular, both the isentropic and the polytropic process are reversible processes. Both apply to adiabatic systems, but only the isentropic process is adiabatic. The polytropic process is the succession of an infinite number of isentropic compression steps, each followed by an isobaric heat addition. This (reversible) heat addition generates just the same temperature increase as the (irreversible) losses in the real process would generate.

The polytropic efficiency η_p is defined such that it is constant for any infinitesimally small compression step, which then allows to write:

$$H = \frac{1}{\eta_p}\int_{p_1}^{p_2} v\,dp = \frac{H_p}{\eta_p} \tag{4}$$

and:

$$H_p = \int_{p_1}^{p_2} v\,dp \tag{5}$$

or, to define the polytropic efficiency:

$$\eta_p = \frac{H_p}{H} \tag{6}$$

For designers of compressors, the polytropic efficiency has an important advantage: If a compressor has five stages, and each stage has the same isentropic efficiency η_s, then the overall compressor efficiency will be lower than η_s. If, for the same example, one assumes that each stage has the same polytropic efficiency η_p, then the polytropic efficiency of the entire machine is also η_p.

The actual flow (Q) can be calculated from standard flow or mass flow (standard conditions are p = 14.7 psia, T = 60°F. Similarly, normal conditions are specified at p = 101.325 kPa, T = 0°C), once the density is known (from the equation of state), i.e.,:

$$Q(p,T) = \frac{\rho_{std}}{\rho(p,T)} Q_{std} = \frac{W}{\rho(p,T))} \tag{7}$$

Finally, the aerodynamic or gas power of the compressor, then, is determined to be:

$$P_g = \rho_1 Q_1 H = \frac{p_1}{Z_1 R T_1} Q_1 H \tag{8}$$

Mechanical losses occur in the gas compressor and the gearbox (if one is used). The adiabatic efficiency of a compressor does not include the mechanical losses, which typically amount to about 1 to 2 percent of the absorbed power. The predicted absorbed power of a compressor should include all mechanical losses. By introducing a mechanical efficiency (η_m), typically 98 to 99 percent, to account for bearing losses, the absorbed compressor power (P) becomes:

$$P = \frac{P_g}{\eta_m} \tag{9}$$

Real Gas Behavior and Equations of State

Understanding gas compression requires an understanding of the relationship between pressure, temperature, and density of a gas. An ideal gas exhibits the following behavior:

$$\frac{p}{\rho} = RT \tag{10}$$

where R is the gas constant, and as such is constant as long as the gas composition is not changed. Any gas at very low pressures ($p \to 0$) can be described by this equation.

For the elevated pressures seen in natural gas compression, this equation becomes inaccurate, and an additional variable, the compressibility factor Z, has to be added:

$$\frac{p}{\rho} = ZRT \tag{11}$$

Unfortunately, the compressibility factor itself is a function of pressure, temperature, and gas composition.

A similar situation arises when the enthalpy has to be calculated. For an ideal gas, one finds:

$$\Delta h = c_p \cdot \Delta T = \int_{T_1}^{T_2} c_p\,dT \tag{12}$$

where c_p is only a function of temperature.

In a real gas, one gets additional terms for the deviation between real gas behavior and ideal gas behavior (Poling, et al., 2001):

$$\Delta h = (h^0 - h(p_1))_{T_1} + \int_{T_1}^{T_2} c_p dT - (h^0 - h(p_2))_{T_2} \tag{13}$$

The terms $(h^0 - h(p_1))_{T1}$ and $(h^0 - h(p_2))_{T2}$ are called departure functions, because they describe the deviation of the real gas behavior from the ideal gas behavior. They relate the enthalpy at some pressure and temperature to a reference state at low pressure, but at the same temperature. The departure functions can be calculated solely from an equation of state, while the term $\int c_p dT$ is evaluated in the ideal gas state.

Equations of state are semiempirical relationships that allow to calculate the compressibility factor as well as the departure functions. For gas compression applications, the most frequently used equations of state (EOS) are Redlich-Kwong, Soave-Redlich-Kwong, Benedict-Webb-Rubin, Benedict-Webb-Rubin-Starling and Lee-Kessler-Ploecker (Poling, et al., 2001).

In general, all of these equations provide accurate results for typical applications in pipelines, i.e., for gases with a high methane content, and at pressures below about 3500 psia. Kumar, et al. (1999), Beinecke and Luedtke, et al. (1983), and Sandberg (2005) have compared these equations of state regarding their accuracy for compression applications. For higher pressures, and gases with high CO_2 and H_2S content, the Benedict-Webb-Rubin type EOS, and in particular its derivative, the Lee-Kesler-Ploecker EOS provide the most accurate results.

GAS COMPRESSOR COMPONENTS

The typical centrifugal compressor is either a single stage machine with overhung rotor or a machine with multiple stages with a beam style rotor. A stage consists of the inlet system (for the first stage) or a return channel (for subsequent stages), the impeller, the diffuser (either vaneless or with vanes), and after the last stage a discharge collector or (in more modern machines) a discharge volute.

The gas enters the inlet nozzle of the compressor (Figure 3) and is guided (often with the help of guide vanes) to the inlet of the first impeller. An impeller consists of a number of rotating vanes that impart mechanical energy to the gas. The gas will leave the impeller with an increased velocity and increased static pressure. In the diffuser, part of the velocity is converted into static pressure. Diffusers can be vaneless or contain a number of vanes. If the compressor has more than one impeller, the gas will be again brought in front of the next impeller through the return channel and the return vanes. After the diffuser of the last impeller in a compressor, the gas enters the discharge system. The discharge system can either make use of a volute, which can further convert velocity into static pressure, or a simple cavity that collects the gas before it exits the compressor through the discharge nozzle.

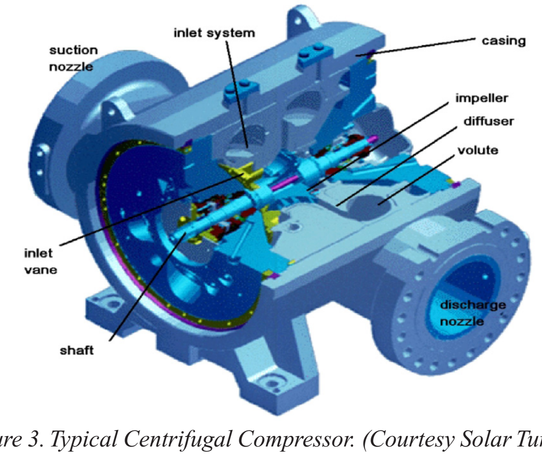

Figure 3. Typical Centrifugal Compressor. (Courtesy Solar Turbines Incorporated)

The rotating part of the compressor consists of all the impellers and the shaft. This rotor runs on two radial bearings (on all modern compressors, these are hydrodynamic tilt pad bearings), while the axial thrust generated by the impellers is balanced by a balance piston, and the resulting force is balanced by a hydrodynamic tilt pad thrust bearing. The compressor shaft can be either a solid shaft, with the impellers shrunk or keyed on, or in modular rotors, the impellers can form part of the shaft (Figure 4).

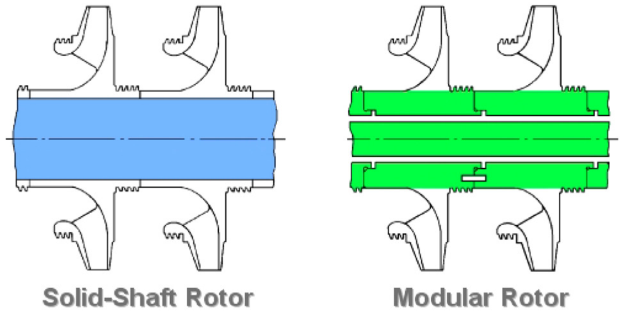

Figure 4. Rotor Designs for Centrifugal Compressors.

The stability of the rotordynamic behavior of a compressor, that is, to operate without excessive vibrations within the desired range of speeds and pressures, is not only the key requirement for successful operation, but it also often limits the application of a compressor for certain applications (Nicholas and Kocur, 2005; Kocur, et al., 2007; Baldassare and Fulton, 2007). Particular importance relates to the capability to sufficiently dampen the various possible excitations the rotor system may be subject to, be it from seals, impellers, unbalance, and others.

To keep the gas from escaping at the shaft ends, dry gas seals are used on both shaft ends, except for overhung impellers, which only require one seal. Other seal types have been used in the past, but virtually all modern centrifugal compressors use dry gas seals, except for applications in air or nitrogen compression, where often carbon ring seals or labyrinth seals are used. For dry gas seals, the sealing is accomplished by a stationary and a rotating disk, with a very small gap (about 5µm) between them. At standstill, springs press the movable seal disk onto the stationary disk. Once the compressor shaft starts to rotate, the groove pattern on one of the disks causes a separating force, making the seals run without mechanical contact of sealing surfaces.

The pressure containing casing is either horizontally or vertically split (Figure 5). The casing, as well as the compressor flanges have to be rated for the maximum discharge pressure the compressor will experience. Horizontally split casings are typically used for lower pressure applications (up to about 40 bar [600 psi] discharge pressure), while vertically split (barrel type) casings have successfully been used for discharge pressures up to 800 bar (12,000 psi).

The maximum amount of impellers in a casing is usually limited by rotordynamic considerations. Therefore, the maximum amount of head to be generated in one casing is limited. If more head is required, multiple casings, driven either by the same driver, or by separate drivers, have to be used. Another limitation for head may be the temperature limits of the compressor (typically, the discharge temperatures are limited to about 175°C/350°F). If more head is required, the gas has to be cooled during the compression process.

Various design alternatives are available (Figure 6):

- Multibody tandems, that is up to three compressor casings driven by the same driver, possibly with a gearbox either between the driver and the compressor train, or between two of the compressors.
- Compound compressors: This compressor contains multiple compartments, each with its own suction and discharge nozzle. All impellers are on the same shaft and face in the same direction.

- Back-to-back compressors: This compressor contains two compartments, each with its own suction and discharge nozzle. The impellers are on the same shaft, but the impellers in the first compartment face in the opposite direction from the impellers in the second compartment.
- Integral gear type compressors: Overhung impellers are located at each end of multiple pinions, driven from a central bull gear.

Either of these configurations does not only allow intercooling, but also sidestreams or gas off-takes.

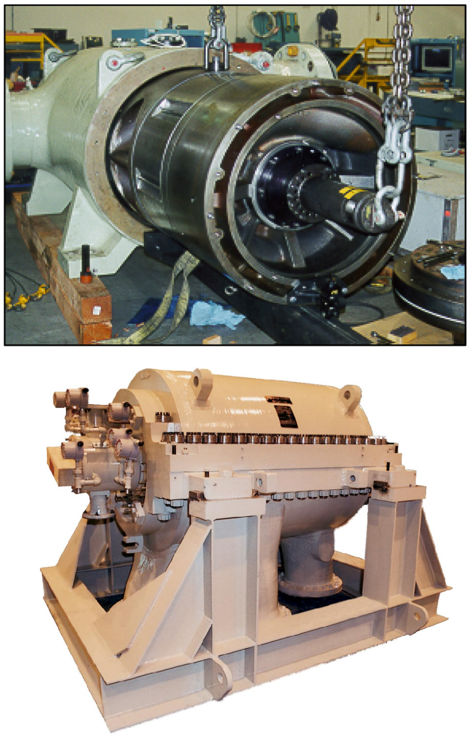

Figure 5. Casing Designs for Centrifugal Compressors. (Top: Barrel Type Compressor During Bundle Removal [Courtesy Solar Turbines Incorporated], Bottom: Horizontally Split Compressor [Courtesy Dresser-Rand Company]).

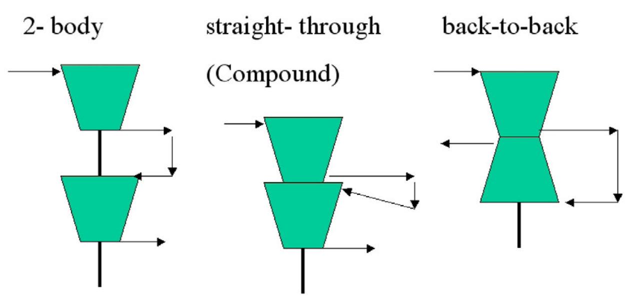

Figure 6. Multibody, Compound and Back-to-Back Compressors.

OFF-DESIGN BEHAVIOR (CONSTANT SPEED)

A compressor, operated at constant speed, is operated at its best efficiency point (Figure 7). If one reduces the flow through the compressor (for example, because the discharge pressure that the compressor has to overcome is increased), then the compressor efficiency will be gradually reduced, due to the increase in incidence losses. At a certain flow, stall, probably in the form of rotating stall, in one or more of the compressor components will occur. At further flow reduction, the compressor will eventually reach its stability limit, and go into surge.

If the flow through a compressor at constant speed is reduced, the losses in all aerodynamic components will increase. Eventually the flow in one of the aerodynamic components, usually in the diffuser, but sometimes in the impeller inlet, will separate. It should be noted that stall usually appears in one stage of a compressor first.

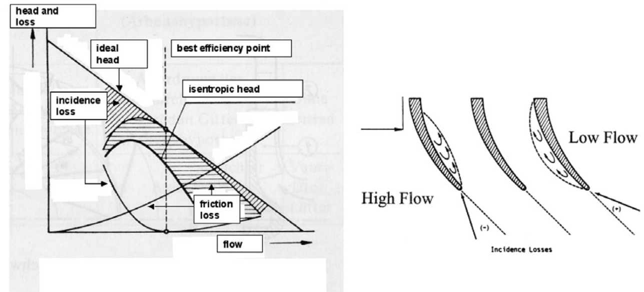

Figure 7. Compressor Operating at Off-Design Points.

Flow separation in a vaneless diffuser means that all or parts of the flow will not exit the diffuser on its discharge end, but will form areas where the flow stagnates or reverses its direction back to the inlet of the diffuser (i.e., the impeller exit).

Stall in the impeller inlet or a vaned diffuser is due to the fact that the direction of the incoming flow relative to the rotating impeller changes with the flow rate through the compressor. Usually, vanes in the diffuser reduce the operating range of a stage compared to a vaneless diffuser. Therefore, a reduction in flow will lead to an increased mismatch between the direction of the incoming flow the impeller was designed for and the actual direction of the incoming flow. At one point this mismatch becomes so significant that the flow through the impeller breaks down.

Flow separation can take on the characteristics of a rotating stall. When the flow through the compressor stage is reduced, parts of the diffuser experience flow separations. Rotating stall occurs if the regions of flow separation are not stationary, but move in the direction of the rotating impeller (typically at 15 to 30 percent of the impeller speed). Rotating stall can often be detected from increasing vibration signatures in the subsynchronous region. Onset of stall does not necessarily constitute an operating limit of the compressor. In fact, in many cases the flow can be reduced further before the actual stability limit is reached.

If, again starting from the best efficiency point, the flow is increased, one also sees a reduction in efficiency, accompanied by a reduction in head. Eventually the head and efficiency will drop steeply, until the compressor will not produce any head at all. This operating scenario is called choke. For practical applications, the compressor is usually considered to be in choke when the head falls below a certain percentage of the head at the best efficiency point. Some compressor manufacturers do not allow operation of their machines in deep choke. In these cases, the compressor map has a distinct high flow limit for each speed line.

The efficiency starts to drop off at higher flows, because a higher flow causes higher internal velocities, and thus higher friction losses. The head reduction is a result of both the increased losses and the basic kinematic relationships in a centrifugal compressor. Even without any losses, a compressor with backward bent blades (as they are used in virtually every industrial centrifugal compressor) will experience a reduction in head with increased flow (Figure 7). "Choke" and "stonewall" are different terms for the same phenomenon.

Surge Margin and Turndown

Any operating point A can be characterized by its distance from the onset of surge. Two definitions are widely used: The surge margin:

$$SM(\%) = \frac{Q_A - Q_B}{Q_A} \cdot 100 \qquad (14)$$

which is based on the flow margin between the operating point and the surge point at constant speed, and the turndown:

$$Turndown(\%) = \frac{Q_A - Q_C}{Q_A} \cdot 100 \tag{15}$$

which is based on the flow margin between the operating point and the surge point at constant head.

COMPRESSOR CONTROL

Previously, the authors have discussed the operating characteristic of a compressor without any other control adjustments. In this case, the head the compressor has to generate uniquely determines the flow.

There are several ways to enhance the capability of the compressor to cover a wider range of different operating conditions:

- Variable speed
- Inlet vanes
- Diffusor vanes
- Throttling (suction, discharge)
- Recycling

Variable Speed

Compressor drivers that can operate at variable speed (two shaft gas turbines, steam turbines, turboexpanders, electric motors with variable frequency drives or variable speed gearboxes) allow the compressor to operate over a range of different speeds. The faster the compressor runs, the more head and flow it generates, and the more power it consumes. The efficiency characteristics of the compressor are retained for different speeds, so this is a very efficient way of adjusting the compressor to a wide range of different operating conditions. Figures 1 and 8 show the resulting map.

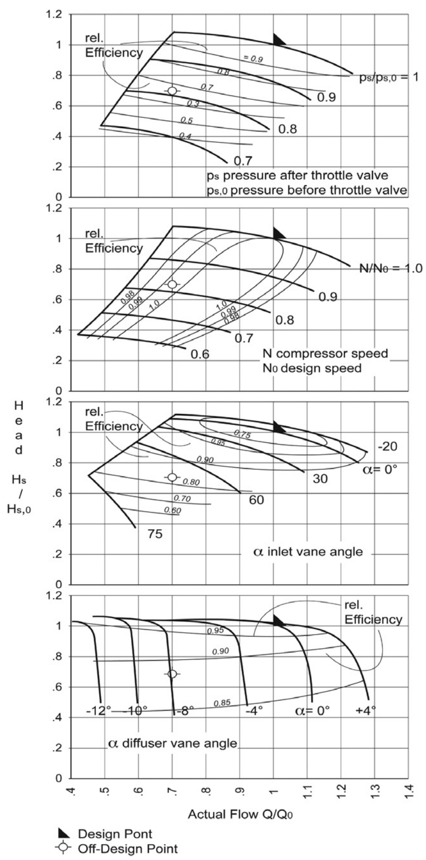

Figure 8. Single Stage Compressor Maps for (Top to Bottom) Suction Throttle, Variable Speed, Variable Inlet Vanes, and Variable Diffuser Vanes.

Adjustable Inlet Vanes

Modifying the swirl of the flow into the impeller allows to modify the operating characteristics of the stage (Figure 8). This can be accomplished by adjustable vanes upstream of the impeller. Increasing the swirl against the rotation of the impeller increases the head and flow through the stage. This is very effective to increase the range for a single stage. In multistage compressors, the range increase is limited if only the first stage has adjustable vanes. The technical difficulty for high pressure compressors lies in the fact that complicated mechanical linkage has to actuate from outside the pressure containing body.

Adjustable Diffusor Vanes

Vaned diffusers tend to limit the operating range of the compressor because the vanes are subject to increased incidence at off-design conditions, thus eventually causing stall. Adjustable diffuser vanes allow to adjust for the changing flow conditions, thus effectively allowing for operation at much lower flows by delaying the onset of diffuser stall (Figure 8). They will not increase the head or flow capability of the stage. In multistage compressors, the range increase is limited if only one stage has adjustable vanes. The technical difficulty for high pressure compressors lies in the fact that complicated mechanical linkage has to actuated from outside the pressure containing body. Another issue is that for the vanes to operate, small gaps between the vanes and the diffuser walls have to exist. Ubiquitous leakage through these gaps causes efficiency and range penalties, in particular in machines with narrow diffusers.

Throttling (Suction, Discharge)

A throttle valve on the suction or discharge side of the compressor increases the pressure ratio the compressor sees, and therefore moves the operating point to lower flows on the constant speed map. It is a very effective, but inefficient way of controlling compressors (Figure 8).

Recycling

A controlled recycle loop allows a certain amount of the process flow to go from the compressor discharge back to compressor suction. The compressor therefore sees a flow that is higher than the process flow. This is a very effective, but inefficient way to allow the compressor system to operate at a low flow.

Process Control with Centrifugal Compressors Driven by Two Shaft Gas Turbines

The following is a description of a typical control scenario, in this particular case for compressors with a gas turbine driver that can operate at variable speeds. Centrifugal compressors, when driven by two shaft gas turbines, are usually adapted to varying process conditions by means of speed control. This is a very elegant way of controlling a system, because both the centrifugal compressor and the power turbine of a two shaft gas turbine can operate over a wide range of speeds without any adverse effects. A typical configuration can operate down to 50 percent of its maximum continuous speed, and in many cases even lower. Reaction times are very fast, thus allowing a continuous load following using modern, programmable logic controller (PLC) based controllers.

A simple case is flow control. The flow into the machine is sensed by a flow metering element (such as a flow orifice, a venturi nozzle, or an ultrasonic device). A flow set point is selected by the operator. If the discharge pressure increases due to process changes, the controller will increase the fuel flow into the gas turbine. As a result the power turbine will produce more power and cause the powerturbine, together with the driven compressor, to accelerate. Thus, the compressor flow is kept constant (refer to Figure 8). Both the power turbine speed and the power increase in that situation.

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If the discharge pressure is reduced, or the suction pressure is increased due to process changes, the controller will reduce the fuel flow into the gas turbine. As a result the power turbine will produce less power and cause the power turbine, together with the driven compressor, to decelerate. Thus, the compressor flow is kept constant.

Similar control mechanisms are available to keep the discharge pressure constant, or to keep the suction pressure constant. Another possible control mode is to run the unit at maximum available driver power (or any other constant driver output). In this case, the operating points are all on a line of constant power, but the speed will vary. The control scheme works for one or more compressors, and can be set up for machines operating in series as well as in parallel.

If speed control is not available, the compressor can be equipped with a suction throttle, or with variable guide vanes. The latter, if available in front of each impeller is rather effective, but the mechanical complexity proves usually to be prohibitive in higher pressure applications. The former is a mechanically simple means of control, but it has a detrimental effect on the overall efficiency.

SYSTEM INTERACTION

The compressor operating point is always the result of the interaction between the compressor and its controls (speed, power, vane settings, etc.) and the characteristics of the system within which it operates. The system usually can be characterized by the relationship between compressor head and flow. Typical cases are:

1. Systems requiring an increase in head with increase in flow.

2. Systems requiring (more or less) constant head with change in flow.

There are systems, with a significant amount of storage capacity, where:

3. The head requirement is a function of the previous flow.

Examples for 1 are all pipeline systems, including transmission pipelines, but also gas gathering systems. Type 2 systems often involve separators or other process equipment that need to operate at constant pressure. Another example is large reservoirs, and situations where the gas has to be fed into a pipeline that operates at more or less constant pressure. Air compressors, providing plant air (at constant pressure), are another example. Refrigeration processes usually also require constant pressure independent of flow. Type 3 systems are often found in storage/withdrawal applications.

It should be noted that the steady-state characteristic of most systems is different from the transient characteristic. In transient operations, the gas inertia as well as mass storage effects have to be considered.

For any compressor characteristic (represented by its performance map) the compressor operating point is determined by the intersection of the system characteristic and the compressor characteristic.

The concept is explained for a pipeline application follows. For a situation where a compressor operates in a system with pipe of the length L_u upstream and a pipe of the length L_d downstream, and further where the pressure at the beginning of the upstream pipe p_u and the end of the downstream pipe p_e are known and constant, one has a simple model of a compressor station operating in a pipeline system (Figure 9). The pressure gradient in the pipeline can be described by the Fanning equation:

$$\frac{dp}{dx} = -32 f \frac{\rho_{std} Q_{std}{}^2}{\pi^2 D^3} \tag{16}$$

which can be integrated, assuming the friction factor f to be constant. For a given, constant flow capacity Q_{std}, the pipeline will then impose a pressure p_s at the suction and p_d at the discharge side of the compressor.

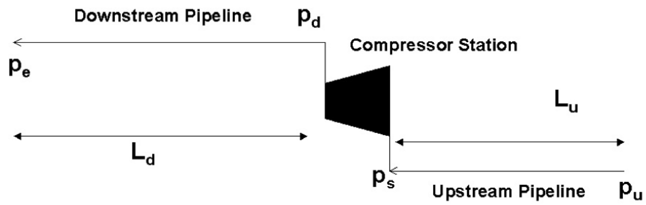

Figure 9. Pipeline Schematic.

Kurz and Lubomirsky (2006) show, that for a given pipeline, under steady-state conditions, the head-flow relationship at the compressor station can be approximated by:

$$H^* = C_p T_s \left[\left(\sqrt{\frac{1}{1 - \frac{C_3 + C_4' \cdot Q^2}{p_d^2}}} \right)^{\frac{k-1}{k}} - 1 \right] \tag{17}$$

where C_3 and C_4 are constants (for a given pipeline geometry) describing the pressure at the two ends of the pipeline, and the friction losses, respectively.

Equation (17) shows a direct relationship between the flow transported in the pipeline and the required head, assuming the pipeline losses are known, and the station discharge pressure is defined. This seems somewhat limiting, but one must consider that, in order to maximize flow in a pipeline system, the station discharge pressure will be at, or close to, the maximum operating pressure of the pipeline. This means that for a compressor station within a pipeline system, the head for a required flow is prescribed by the pipeline system (Figure 10). In particular, this characteristic requires the capability for the compressors to allow a reduction in head with reduced flow, and vice versa, in a prescribed fashion. The pipeline will therefore not require a change in flow at constant head (or pressure ratio).

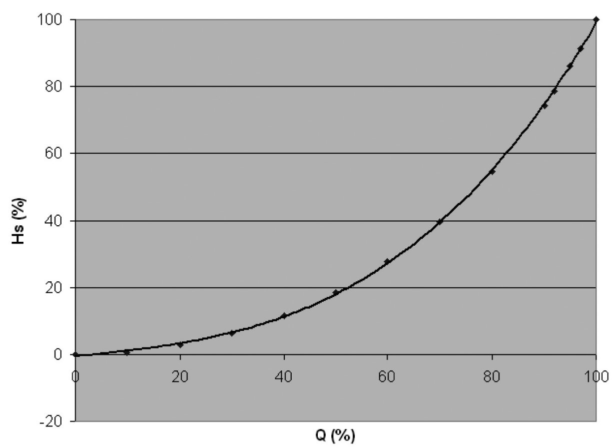

Figure 10. Station Head-Flow Relationship Based on Equation 17.

Compressor Control

Based on the requirements above, the compressor output must be controlled to match the system demand. This system demand is characterized by a relationship between system flow and system head or pressure ratio. Given the large variations in operating conditions experienced by compressors, an important question is how to adjust the compressor to the varying conditions, and, in particular, how does this influence the efficiency.

Centrifugal compressors tend to have rather flat head versus flow characteristic. This means that changes in pressure ratio have a significant effect on the actual flow through the machine. For a centrifugal compressor operating at a constant speed, the head or pressure ratio is reduced with increasing flow.

The resulting operating point of a compressor is determined by the head-flow characteristic of the system, the map of the compressor, and the available power to drive the compressor.

Surge Control

Surge Control systems are by nature surge avoidance systems. In general, the control system measures the gas flow through the compressor and the head it generates. The knowledge of head and flow allows to compare the present operating point of the compressor with the predicted surge line (Figure 1). If the process forces the compressor to approach the surge line, a recycle valve in a recycle line is opened (see above). This allows the actual operating point of the compressor to move away from surge (Kurz and White, 2004; White and Kurz, 2006).Well designed surge control systems can allow to reduce the station flow to zero while keeping the compressor online. They will also make the transition from fully closed recycle valve to an increasingly open recycle valve smooth and without upset to the process.

Control for Multiple Process Streams

Compressor or compressor trains often have to handle multiple process streams. To control N streams, the compressor train needs to have N control devices. For example, a two body tandem with one sidestream can be controlled by a combination of speed control (for the entire train) in combination with a throttle valve for the sidestream. Another possibility for this case may be to use a unit recycle loop for control.

Start-Up and Shutdown

The start-up and shutdown processes require particular attention. Issues to be considered involve the speed-torque capability of the driver, as well as the thermal balance in the recycle system. The former is in particular an issue with electric motor drives, where the available motor torque often depends on the capability of the electric grid, or the electric power generation. The latter involves the fact that in many instances the compressor will pump gas in the recycle loop until the pressure it generates is high enough to go online. In a recycle loop that is not cooled, the temperature will increase (due to the work the compressor puts into the gas), and if the temperature exceeds certain thresholds, the start has to be aborted. This issue is discussed in more detail by White and Kurz (2006).

SPARES AND MULTIPLE UNITS

For a given application, the overall process pressure ratio, or the total flow can be divided into multiple units. This is often advantageous because it provides for more process flexibility. If, for example, the flow demand is reduced, one unit can be shut down completely, therefore avoiding operation in recycle.

Consideration has also to be given to the availability of the station. This includes planned maintenance events, as well as shutdowns due to damage of the equipment. Possible configuration could include one complete spare unit. The spare unit would be sized to duplicate the capability of the main unit. If the process is divided into multiple, smaller units, the shutdown of one of these units causes only a proportional reduction in process flow, and a spare unit may not be necessary.

HOW ARE OIL AND GAS PRODUCED?

Once an oil or gas reservoir is discovered and assessed, the task is to maximize the amount of oil or gas that can ultimately be recovered. Oil and gas are contained in the pore spaces of the reservoir rock. Some types of reservoirs allow the oil and gas to move freely, making it easier to recover. Other reservoirs restrict the flow of oil and gas and require special techniques to move the oil or gas from the pores to a producing well. Even today, with advanced technologies, in some reservoirs more than two-thirds of the oil present may not be recoverable. Figure 11 shows the path of the gas from well to user. Many oil and gas wells are on the ocean floor, and production requires an offshore platform (Figure 12).

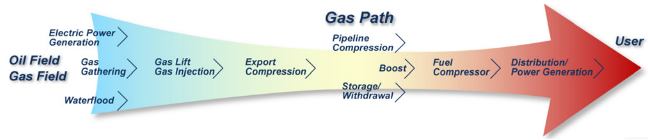

Figure 11. Gas Path for Natural Gas.

Figure 12 Offshore Platform.

To prepare the well for production, the bore hole is stabilized with a casing (lengths of pipe cemented in place). A small-diameter tubing string is centered in the wellbore and held in place with packers. This tubing will carry the hydrocarbons from the reservoir to the surface.

The reservoirs are typically at elevated pressure. A series of valves and equipment ("Christmas tree") is installed on top of the well, to regulate the flow of hydrocarbons out of the well.

Early in its production life, the underground pressure will often push the hydrocarbons all the way up the well bore to the surface. Depending on reservoir conditions, this "natural flow" may continue for many years. When the pressure differential is insufficient for the oil to flow naturally, mechanical pumps must be used to bring the oil to the surface. This process is referred to as artificial lift.

The production pattern for most wells follows a predictable pattern where production will increase for a short period, then peak and follow a long, slow decline (Figure 13). The shape of this decline curve, how high the production peaks, and the length of the decline are all driven by reservoir conditions. The decline curve can be influenced by cleaning out the wellbore to help oil or gas move more easily to the surface, by fracturing or treating the reservoir rock with acid around the bottom of the wellbore to create better pathways for the oil and gas to move through the subsurface to the producing well, or by drilling additional wells, or by employing enhanced oil recovery techniques.

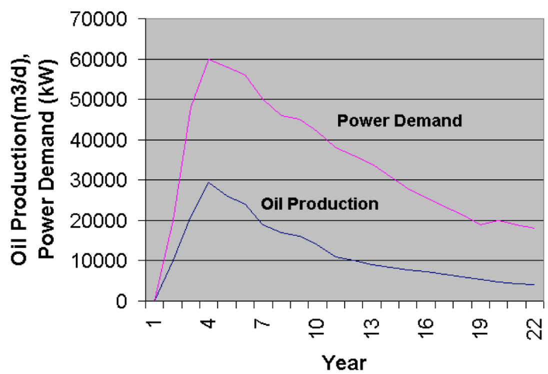

Figure 13. Typical Decline in Oil Production and Power Demand in an Offshore Application. (Courtesy Miranda and Brick, Turbomachinery Laboratory, 2004; Kurz, and Sheya, 2005)

These techniques are collectively referred to as enhanced oil recovery (EOR) and, depending on reservoir conditions, employ water injection ("waterflooding") or the injection of various other substances (hydrocarbons, steam, nitrogen, carbon dioxide) into the reservoir to remove more oil from the pore spaces to increase production.

When geologists began studying time-lapse seismic monitoring results ("4-D"), they were surprised to discover that one of the most basic notions about the movement of oil in a reservoir—that it naturally settles between lighter natural gas above and heavier groundwater below—oversimplifies the behavior of real oil fields. Actually, most wells produce complex, fractal drainage patterns that cause the oil to mix with gas and water. It also became clear that traditional techniques may leave 60 percent or more of the oil behind (Anderson, 1998). This led to the strategy to pump natural gas, steam, carbon monoxide, or nitrogen into the reservoirs. This injection spreads through the pores in the rock and pushes oil that otherwise would have been abandoned toward the existing wells. Application where gas is injected into the oil reservoir for pressure maintenance and to enhance oil recovery by miscible flooding with lean (methane rich) gas is usually referred to as gas reinjection.

Most oil wells produce oil, gas, and water. This mixture is separated at the surface. Initially, the oil well may produce mostly oil with a small amount of water. Over time, the percentage of water increases. This produced water varies in quality from very briny to relatively fresh. Where this water cannot be used for other purposes, it may be reinjected into the reservoir—either as part of a waterflooding project or for disposal (returning it to the subsurface).

The oil is then sent to an oil treatment plant. There it is processed in a gas-oil separation system where its pressure is reduced in several stages. In each decompression stage the associated gas (also called flash gas) is released in a separator until the pressure is ultimately reduced to slightly above atmospheric pressure. The crude oil is then sent to a stabilizer column where it is heated and cascaded through a series of bubble trays spaced throughout the column. Hydrogen sulfide (if present) and remaining light hydrocarbons boil off in this process and are collected at the top of the column, while the sweetened heavy crude is drawn off from the bottom. The stabilized oil is then cooled and stored. The streams collected from the top of the stabilizer unit are treated in accordance with environmental regulations.

Natural gas wells do not produce oil but usually some amount of liquid hydrocarbons, which are called condensate. In addition to condensate, natural gas liquids (ethane, propane, butane) are removed at a gas processing plant, along with other impurities, such as hydrogen sulfide and carbon dioxide. Natural gas liquids often have significant value as petrochemical feedstock. Natural gas wells also often produce water, but the volumes are much lower than is typical for oil wells.

Natural gas is almost always transported through pipelines, except in cases where a pipeline cannot economically be built. In that case, the gas can be liquefied (liquefied natural gas [LNG]) and transported on a ship. As part of the transportation process in pipelines, gas can be stored in storage facilities, which often use former gas fields or salt caverns. This allows to balance differences in supply and demand on a seasonal or daily basis.

Usually, all applications upstream and including a gas plant are considered "upstream" applications, while the applications related to bringing gas to the ultimate users are referred to as "midstream." Applications in refineries, chemical and processing plants are considered "downstream" applications.

APPLICATIONS

For this section, the authors have made an effort to use the definitions most widely used in the industry. However, some of the definitions are used interchangeably, and some applications might be combined in a single compressor or compressor train. The different applications are illustrated by worked out examples. Unlike older publications that demonstrate hand calculations, the authors will use a computer program to perform equation of state calculations (as the authors would recommend for anyone who has to do similar calculations), i.e., the authors will directly calculate head and actual flow from process data, as described in the section about thermodynamics above. This removes one of the major inaccuracies of using ideal gas calculations (or modifications thereof) for real gas calculations. The authors also deviate from the often used practice of polytropic calculations, and use typical isentropic efficiencies for the compressors employed (this is more realistic, because the assumption of a constant polytropic stage efficiency does not hold in reality).

Many installations include the facilities for a number of different duties (Figure 14 and 15).

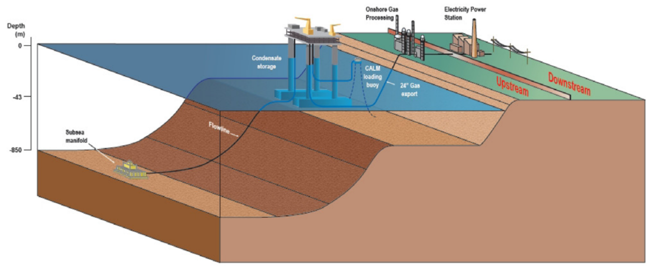

Figure 14. Offshore Platform with Depletion Compression (Gas-Reinjection), Condensate Stabilization (Gas Gathering), Export Compression to Onshore Gas Plant via Subsea Export Pipeline.

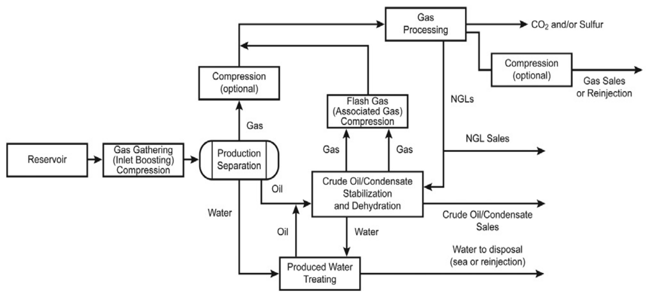

Figure 15. Oil and Gas Field Compression.

Upstream Applications

Gas Gathering (Inlet Boosting, Flash Gas Compression) for Associated Gas

Application where associated gas from the oil production is compressed. At the wellhead, a mixture of hydrocarbons is present, and the goal is to separate the crude oil from its more volatile constituents, mostly natural gas. Typically, gas is separated from crude oil by flashing it at several pressure levels, leading to gas streams at a number of pressure levels, and different gas compositions (typically, the lower the pressure of the gas stream, the heavier is the gas). This gas is recompressed to about 70 to 100 bar (1000 to 1500 psi), and either used for gas lift, gas injection, or as sales gas (Figures 16, 17).

In oilfield facilities, there are a number of conditions that often require the use of compressors. The most common use is the recompression of flashed gas for sale to a gas pipeline. The gas may have been at a low pressure for one of the following reasons:

- The wells may require a low flowing pressure in order to produce economic quantities, or
- Multiple stage separation may be necessary for proper fluid stabilization or other process requirements.

The gas may also need compression for reinjection into the formation, either as an interim measure awaiting a gas market or to maintain reservoir pressure. A typical example is provided in APPENDIX A.

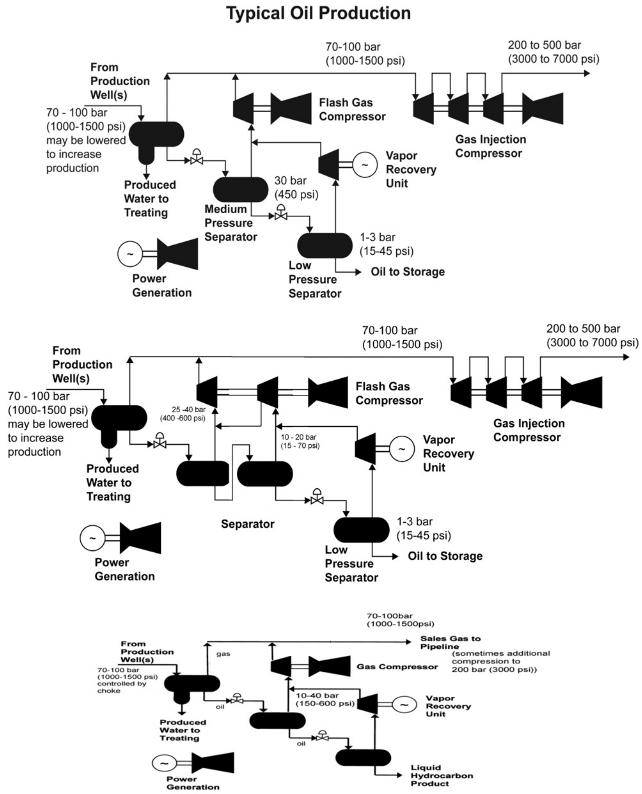

Figure 16. Offshore Oil Field Application, Combining the Gas Gathering, Crude Stabilization, Gas Injection, or Gas Export Duties.

Figure 17. Offshore Gas Compressor Train. (Courtesy Solar Turbines Incorporated)

Gas Gathering in Gas Fields

Application where gas from gas wells is compressed and either fed to a gas plant or a pipeline. Usually more than one well in a geographic area are piped via "flow lines" to a booster compressor or even multiple booster compressors in an inlet compression station. After compression, the gas is usually processed to meet sales quality and then sold. The compressor station is usually close to the wellhead, and upstream of the gas plant. The gas usually comes from a number of wells, which often produce at different pressure levels. The applications usually have low suction pressures (3 to 20 bar, 50 to 300 psi), and the gas is compressed to about 70 to 100 bar (1000 to 1500 psi). A typical scenario involves small compressors close to the wellhead feeding to centrally located larger compressor stations.

The inlet pressures vary with the dynamics of the reservoirs. A gas gathering system usually starts off with a relatively low ratio (1.25 to 1.5 range) and high flows. The reservoirs usually decline in ability to produce and require lower pressures to maintain volume rates. Eventually the losses of the gathering system well tubing and flow line piping dominate and no amount of compression will maintain the flowrate. Many reservoirs under compression will draw down to near vacuum at the wellhead prior to abandonment. This leads to booster compression requirements of high ration and low flow, usually utilizing all the possible horsepower installed in the initial operating case. Ideally this is following a constant horsepower, rising pressure ratio, and falling volume scenario.

The usual strategy for designing booster compression facilities is to consider the reservoir pressure versus flow decline curve and to plan an economically optimized life of field compression scheme. This may involve justifying to invest in compressor configurations that allow higher pressure ratio (thus usually a larger number of impellers), and lower flow future aerodynamic components.

In some gas fields, the incoming gas is cascaded to a lower pressure level to separate the liquids from the gas. This also leads to gas available at different pressure levels. In other instances the gas compression is straight through, without any sidestreams. A typical example is provided in APPENDIX B.

Gas Plant Compression

Gas plants are designed to produce dry export gas (i.e., gas with very little water, a low hydrocarbon dewpoint, limited amounts of CO_2 and other contaminants) and liquefied petroleum gas (LPG) products (ethane, propane, and butane). For the range of gas compositions at the inlet, the plants have specified recovery targets for the heavier hydrocarbons. The process steps inside the plant (Figure 18) include (Brown, et al., 2005):

- Primary separation
- Front end compression (boost compression, inlet compression)
- Carbon dioxide removal
- Mercury/chloride removal
- Gas dehydration
- Gas expansion (turboexpander)
- LPG/condensate fractionation
- Dry (sales) gas compression
- Storage
- Utilities

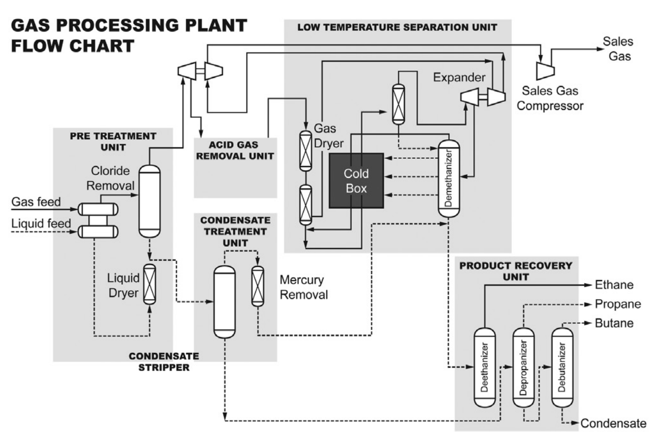

Figure 18. Gas Plant.

In a gas plant, several compression duties have to be covered (Lynch, et al., 2005):

- Boost compression (inlet compression) to bring the gas from delivery pressure (from the gas gathering system) to plant pressure.
- Recompression (sales gas compressor) to bring the natural gas from plant pressure to pipeline pressure. This duty may also be referred to as pipeline head station (essentially depending on whether the compressor is operated by the gas plant or the pipeline operator).
- Turboexpander/compressor for the low temperature cryogenic cycle.

Export Compression (Sales Gas Compressor)

Application where gas is compressed (usually on an offshore platform) to feed a subsea pipeline that transports the gas to shore. Discharge pressures are often high (typically 70 to 140 bar (1000 to 2000 psi), but sometimes up to about 200 to 240 bar (3000 to 3500 psi) to reduce pipe diameter, and also because usually the gas cannot be recompressed between the platform and the shore. Depending on whether this compressor gets gas at well pressure, or whether there is a gas gathering train upstream, configurations can vary from machines with only a few stages to triple body trains (Figures 16,19). Tradeoffs are often the required compression power on the platform versus cost of the pipeline especially if pressure is not dictated by already existing systems.

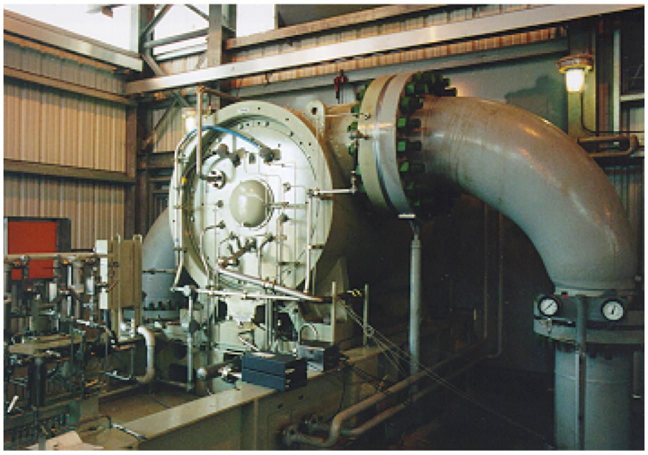

Figure 19. Export Compressor. (Courtesy Solar Turbines Incorporated)

Gas Lift

Application where gas is injected into the oil well to aerate the crude, thus enhancing the flow of crude to the surface (Figures 20, 21). The application may be combined with a gas gathering operation. Some operators use the same compressor train to both feed a gas lift service and export compression to feed gas into a pipeline.

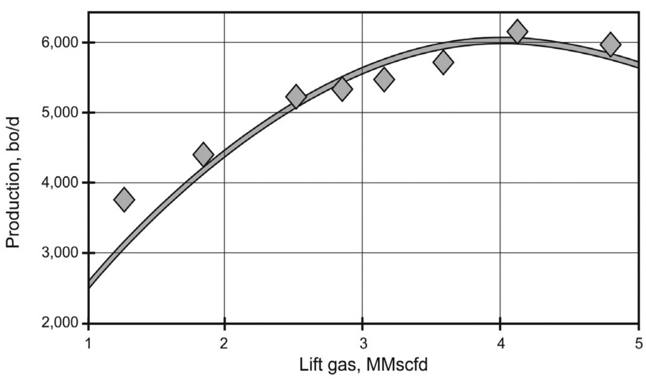

Figure 20. Gas Lift Affects Production. (Courtesy Mayo, et al., 1999)

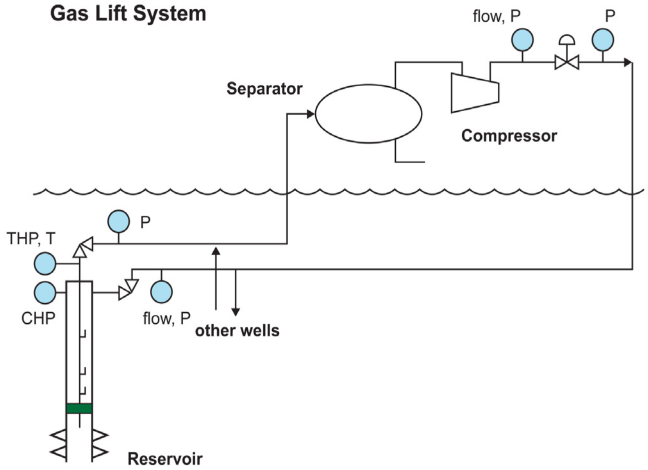

Figure 21. Gas Lift.

Wells may require artificial lifting and "gas lift" may be used. This is a process in which low pressure (3 to 7 bar, 50 to 100 psi) produced gas is compressed to a higher pressure and recycled down the well casing and through a gas lift valve into the tubing at a predetermined depth to lighten the column of liquid in the tubing. Compressor discharge pressures are typically 100 to 120 bar (1400 to 1700 psi), but sometimes up to 200 bar (2900 psi) may be required for such applications, necessitating compressors with relatively high throughput and a high compression ratio.

Gas Reinjection

Reinjection is used as a method of enhanced oil recovery to compensate for the natural decline of an oil field production by increasing the pressure in the reservoir, thus restoring the desired level of production and stimulating the recovery of additional crude oil. Using this technique the field exploitation can be increased by up to 20 percent. The gas that is reinjected is usually the associated gas separated from the crude oil in the flash and stabilization phases. Other gases, such as nitrogen, or carbon dioxide, may also be used. The gas is reinjected into the reservoir in dedicated wells and forces the oil to migrate toward the well bores of the producing wells. Gas Injection projects may also involve the injection of CO_2 or nitrogen into the reservoir. Especially for deep reservoirs, very high compressor discharge pressures (140 to 820 bar [2000 to 12,000 psi]) are required (Figure 16). Recent material technology advances allow associated sour gases containing high percentages of H_2S and/or CO_2 to be reinjected without the need for sweetening.

Depending on the depth and physical characteristics of the field, very high injection pressures may be required. High pressure barrel compressors are normally used in this application. As part of the compression process, the gas has to be dehydrated, usually at pressure levels between about 40 and 70 bar (600 and 1000 psi). A typical application is outlined in APPENDIX A.

Subsea Compression

For offshore applications, the services described above are installed on platforms (fixed leg for shallow waters, floating for deeper waters). Potentially, these services can be located on the seafloor instead. The development of subsea compressor packages is a current challenge, and there are no subsea compressors in commercial production at this time. Challenges involve the issue of separation of gas solids and liquids close to the well (or, conversely, the capability of the compressor to compress gas with significant amounts of entrained liquids), as well the design of compressors motor and drive systems to survive for extended periods of time on the seafloor.

Midstream Applications

Pipeline Compression

Another common use for compression is found in cross-country transmission service. Many pipelines require periodic booster compression, increasing pressure to overcome pressure drop caused by friction in the pipeline (Figure 22). Natural gas can be transported over large distances in pipelines. Optimal pipeline pressures, depending on the length of the pipe, as well as the cost of steel, are in the range of 40 to 160 bar (600 psi to 2500 psi) balancing the amount of power required to pump the gas with the investment in pipe. Most interstate or intercontinental pipeline systems operate at pressures between 60 to 100 bar (1000 and 1500 psi), although the pressures for older systems might be lower. The gas usually has to be compressed to pipeline pressure in a head station (usually coming from a gas plant). This head station often sees pressure ratios of 3. The pipeline compressors are arranged in regular distances along the pipeline, usually spaced for pressure ratios between 1.2 and 1.8. The distinction is sometimes made between mainline stations (that basically operate continuously) and booster stations that are only in operation sporadically to assist mainline compression.

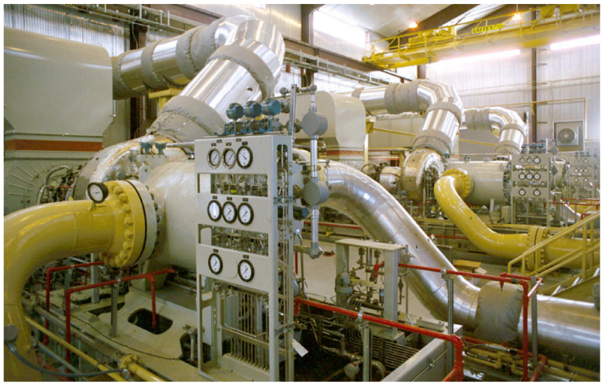

Figure 22. Pipeline Compressor Station with Three Compressors in Parallel Operation. (Courtesy Solar Turbines Incorporated)

Subsea pipelines often only have a headstation, but no stations along the line. They are either used to transport gas to shore from an offshore platform (refer to the *Export Compression* section above), or to transport gas through large bodies of water. In either case, relatively high pressures (100 to 250 bar, 1500 to 3700 psi) are common (Tobin, et al., 2005).

A few onshore pipelines worldwide make use of the added super compressibility of the gas at pressures above 140 bar (2000 psi, depending on gas composition) and operate as "dense phase" pipelines at pressures between 125 and 180 bar (1800 and 2500 psi). Not only natural gas is transported in pipelines, but also CO_2. CO_2 is noncorrosive, as long as it is dehydrated. Most applications transport CO_2 in its dense phase, at pressures above 140 bar (2000 psi), in particular to avoid two phase flows when ambient temperatures drop.

If the throughput of a pipeline has to be increased, two possible concepts can be used: Building a parallel pipe (looping), or adding power to the compressor station (i.e., adding one or more compressors to the station), or a combination of both. These means can also be combined. If power is added to the station, the discharge pressure can be increased (assuming this is not already limited by the pipeline maximum operating pressure). The station will therefore operate at a higher pressure ratio. The added compressors can either be installed in parallel, or in series with the existing machines. If the pipeline is looped, the pressure ratio for the station typically is reduced, and the amount of gas that can be pumped with a given amount of power is increased. In either scenario, the existing machines may have to be restaged (for more pressure ratio and less flow per unit in the case of added power, for more flow and less pressure ratio in the other case.

In general, pipelines that have many takeoffs and interconnects (like in the United States) tend to operate more frequently in transient, nonsteady-state conditions. Long, transcontinental pipelines that basically transport gas from point A to point B tend to operate closer to steady-state conditions. In most cases, compressors experience a significant range of operating conditions (Figure 23). A typical example is outlined in APPENDIX C.

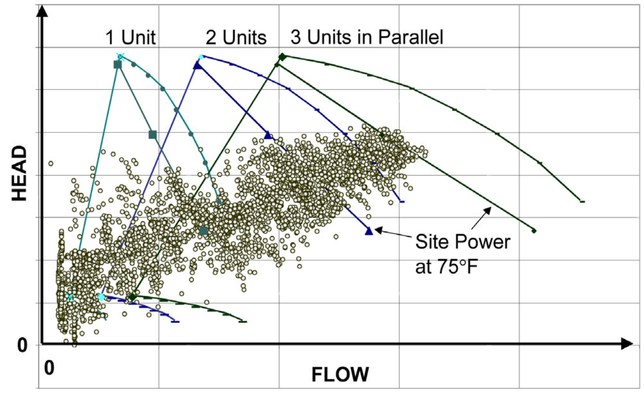

Figure 23. Operating Points Collected over a Six-Month Period in a Gas Compression Station with Three Units in Parallel. (Courtesy Kurz, et al., 2003)

Standby Units

Because the failure or unavailability of compression units can cause significant loss in revenue, the installation of standby units must be considered. These standby units can be arranged such that each compression station has one standby unit, or that some stations have a standby unit, or that the standby function is covered by oversizing the drivers for all stations. It must be noted that the failure of a compression unit does not mean that the entire pipeline ceases to operate, but rather that the flow capacity of the pipeline is reduced. Since pipelines have a significant inherent storage capability ("line pack"), a failure of one or more units does not have an immediate impact on the total throughput. Additionally, planned shutdowns due to maintenance can be planned during times when lower capacities are required.

Standby units are not always mandatory because modern centrifugal compressor sets can achieve an availability of 97 percent and higher. A station with two operating units and one standby unit thus has a station availability of $100\ (1 - 0.03^2) = 99.91$ percent (because two units have to fail at the same time in order to reduce the station throughput to 50 percent). A station with one standby unit and one operating unit also yields a 99.91 percent station availability. However, while failure of two units in the first case still leaves the station with 50 percent capacity, the entire station is lost if both units fail in the second case. Arguably, installing two smaller 50 percent units rather than one larger 100 percent unit could avoid the need for installing a standby unit (Figures 24, 25).

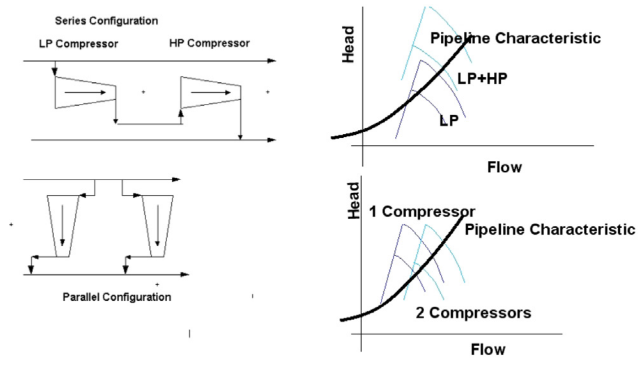

Figure 24. Series and Parallel Compressors Can Cover the Same Duty.

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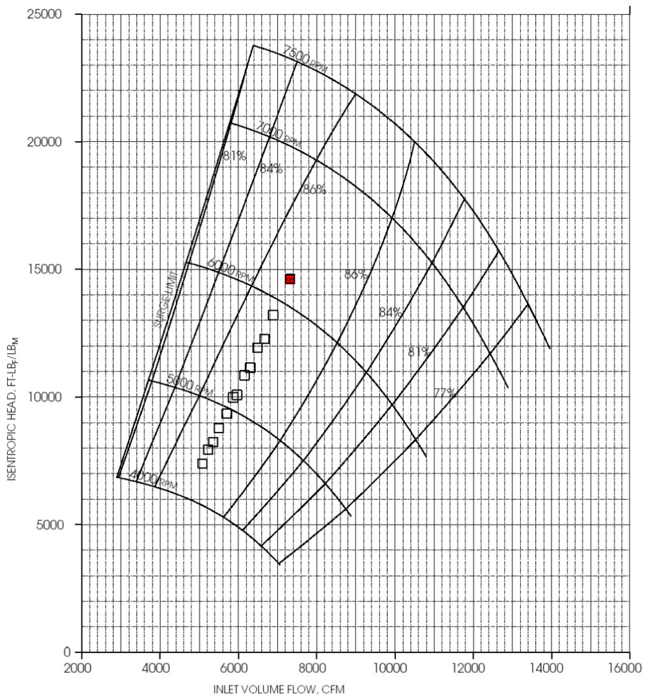

Figure 25. Typical Pipeline Steady-State Operating Points Plotted into a Typical Compressor Performance Map. (Courtesy Kurz and Brun, 2007)

A concept that has been discussed from time to time is the approach to have "power" back up rather than "unit" standby on a station. The idea is to have an oversized driver, operating at part load during normal operation rather than an additional standby unit. In this case, the driver has to be sized such that it, together with other (equally oversized) units located further downstream along the pipeline, can pick up the duty of an unavailable station upstream. The advantage of this concept lies in the reduction in the number of units. The disadvantage is that the driver operates at part load and, thus, possibly at a lower efficiency for most of the time. Oversized units also limit the turndown capability of a station significantly.

It has often been assumed that for two-unit stations without a standby unit, a parallel installation of the two units would yield the best behavior if one unit fails. However, Ohanian and Kurz (2002) have shown that usually a series arrangement of identical compressor sets yields a lower deficiency in flow than a parallel installation (Figure 26). This is due to the fact that pipeline hydraulics dictate a relationship between the flow through the pipeline and the necessary pressure ratio at the compressor station. This follows from the fundamental flow equation for a pipeline: For parallel units, the failure of one unit forces the remaining unit to operate at or near choke, with a very low efficiency. Identical units in series, upon the failure of one unit, would initially require the surge valve to open, but the remaining unit would soon be able to operate at a good efficiency, thus maintaining a higher flow than in the parallel scenario. Figure 26 shows the operating point of the respective compressor after the other unit is shut down at T = 1.0. While the remaining unit in the series arrangement requires to recycle initially, it moves, starting at about T = 2.0, slowly to efficient operating points. The remaining parallel unit, on the other hand, moves further and further into choke. In the example described by Ohanian and Kurz (2002), either scenario would eventually let the pressure at the pipeline outlet drop below the minimum level, but later for the series arrangement. Therefore, the compressor in series arrangement gives the operator more time for corrective action. Given the fact that the gas stored in the pipeline will help to maintain the flow to the users, a series installation would often allow for sufficient time to resolve the problem. Therefore, a standby unit could be avoided.

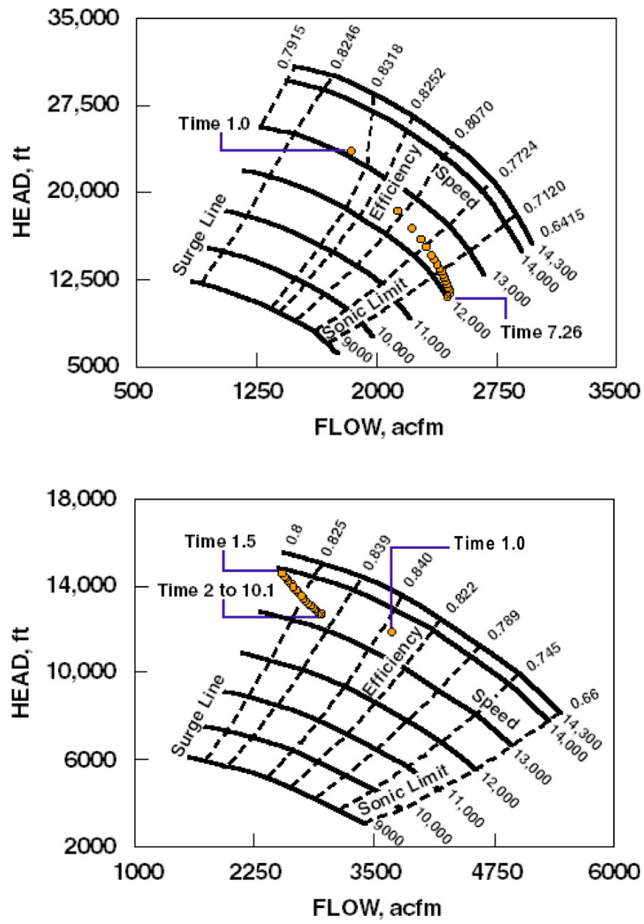

Figure 26. Operating Points of a Parallel (Top) and a Series (Bottom) Compressor Arrangement after One of the Two Units Fails at Time = 1.0. (Courtesy Ohanian and Kurz, 2002)

Gas Storage/Withdrawal

The first natural gas storage facilities date back to the early 20th century in the US and Canada. They were used to provide local natural gas supply during the winter heating season. This became necessary because the high demand in winter frequently exceeded the capacity of the local pipeline and production infrastructure. The introduction of a gas spot market in the mid 80s has led to an increased demand for gas storage facilities. Currently, there are over 400 facilities in North America, and over 130 in Europe in operation. The vast majority of these gas storage facilities use depleted hydrocarbon reservoirs, aquifers, or salt caverns for storage (INGAA, 2007). The former two options involve storage in porous rock layers, while the latter is created by washing a cavity out of a salt dome. These types of storage facilities are very safe, preventing reliably leaks or other safety hazards. In either case, the pipeline company injects natural gas into the storage field when demand is low and withdraws it from the storage field during times of high demand (Figure 27).

Historically, storage was used to respond to the peak needs of cold winter days. Natural gas demand used to be at its highest in winter, primarily due to home heating requirements. In recent years, however, mostly due to increased demand from natural gas fired power plants, demand has become less seasonal. Because of this shift, well-placed natural gas storage has become even more important to natural gas operations.

Today, North American natural gas storage plays a key role in balancing supply and demand, particularly consumption during peak-demand periods.

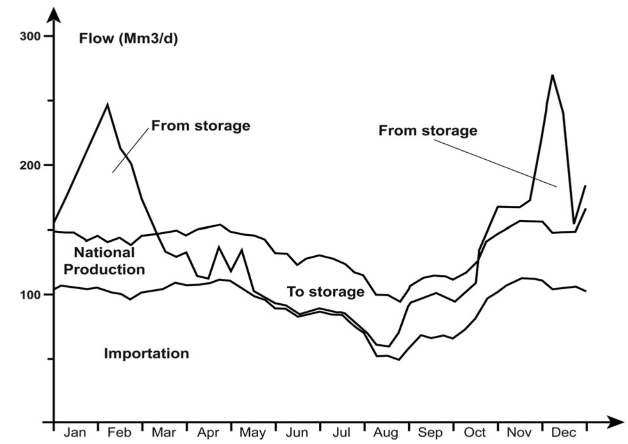

Figure 27. Gas Storage in Italy. (Courtesy Zampieri and Damiani, 1993)

- Storage can reduce the need for both swing natural gas production deliverability and pipeline capacity by allowing production and pipeline throughput to remain relatively constant.
- Customers may use storage to reduce pipeline demand charges, to hedge against natural gas price increase, or to arbitrage gas price differences.
- Pipelines and local distributors use storage for operation flexibility and reliability, providing an outlet for unconsumed gas supplies or a source of gas to meet unexpected gas demand.
- Storage at market trading hubs often provides balancing, parking, and loan services.
- In the future, additional conventional storage will be needed to meet growing seasonal demands and high deliverability storage will be required to serve fluctuating daily and hourly power plant loads.

Gas supply and demand in many pipeline systems shows significant seasonal changes, which is further aggravated by the periodic influx of liquefied natural gas. Gas storage facilities, where gas is stored during times of low demand or high supply, and removed during times of high demand or reduced supply are an important means in managing the gas supply (Kurz and Brun, 2009).

Gas compressors are required to inject gas from a pipeline into the underground for storage, and to extract gas from the storage and feed it into the pipeline. Typical pipeline pressures range from 40 to 100 bar (600 to 1500 psi), and from this pressure, the gas has to be compressed to final storage pressure, typically between 100 and 200 bar (1500 and 3000 psi). The compressor duty is cyclical in nature (Figure 27). Traditionally, the cycles were seasonal, with fluctuating pipeline and storage pressures gradually changing during the course of the season. However, under spot market activity conditions, daily demand cycles, market conditions, or short-term weather patterns can require a facility to change their operating patterns several times a week.

Gas compression has to be used to fill the storage facility, as well as for recompressing gas when the facility is emptied. The compression task is therefore described as filling a large, constant volume with gas, with the limiting factor being the available driver power. This yields very uncharacteristic operating conditions for the compressor. Initially, the low pressure ratio allows for high flow conditions. The pressure ratio has to increase with an increasing amount of gas in the facility, therefore reducing the possible flow for a power limited compression system (Figure 28). A typical example is outlined in APPENDIX D.

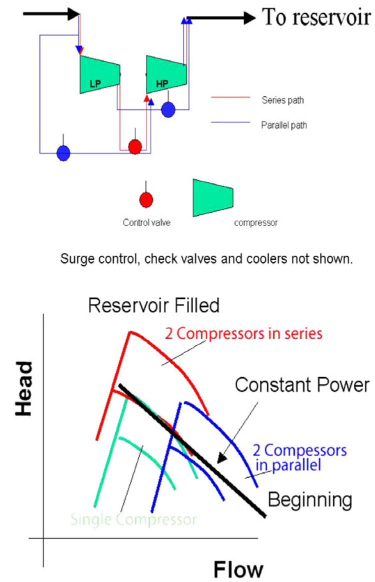

Figure 28. Series and Parallel Operation for Gas Storage Applications.

Air Compression

Many chemical processes require compressed air. Examples are ammonia plants and air separation plants. Compressed air is also used to drive mechanical equipment, and serves many other purposes in chemical and manufacturing plants. It may be used for mine ventilation. Certain integrated gasification combined cycle (IGCC) plants also require large air compressors for air separation. Where the required quantities make centrifugal compressors attractive, one usually finds multistage machines, in general, driven by electric motors. The number of impellers is determined by the required pressure ratio. Efficiency is often important, since it impacts the operating cost of the plant. The capability to efficiently provide air over a range of flows is also often a requirement. Since air is neither combustible nor poisonous, the seal requirement is less onerous than for other compressors. Various types of centrifugal compressors are used, but this application has long been one of the areas where many integral gear type machines are used (Fingerhut, et al., 1991). Several compressor stages are mechanically coupled via a single or multistage spur gear ("bull gear"). The compressor stages are overhung and sit on individual pinions.

Air Separation

Air consists of approximately 78 percent nitrogen, 21 percent oxygen, and 1 percent argon, along with smaller amounts of carbon dioxide, neon, helium, krypton, hydrogen, xenon, and water vapor. Air separation plants produce oxygen, nitrogen, and argon from atmospheric air, normally by one of two processes: cryogenic or noncryogenic. Noncryogenic plants produce oxygen or nitrogen gas product from compressed air at near ambient temperature by physical adsorption. Noncryogenic plants can be a cost-effective alternative where a single gas only product (either oxygen or nitrogen) is required, the production is relatively low, and high purity is not required. Cryogenic plants liquefy and distill ambient air to separate it into its components. Cryogenic plants can produce high production rates of gas and/or liquid products (oxygen, nitrogen, and argon) at high purity levels.

All air separation processes start with the compression of air. Additional compression may be needed to increase the pressure of the produced nitrogen or oxygen leaving the separation process. Further compression may also be needed for the refrigeration process if the products are to be supplied in liquid form.

The process begins with the intake of huge volumes of air from the atmosphere. The air is compressed and purified before entering the cryogenic equipment package. The air is cooled to about −300°F (−185°C) and then, relying on different boiling points, separated into its elemental components in the form of liquid oxygen, argon, and nitrogen. The gas products oxygen and nitrogen can then be compressed and fed into a pipeline network for further use.

The compressors are usually driven by electric motors and the cost of electricity is the largest single operating cost incurred in air separation plants, usually between one- and two-thirds of the operating cost. Compressor efficiency is thus of high importance.

CO_2 Compression

Some key issues have to be considered in the compression of CO_2:

- CO_2 being a heavy gas will usually lead to operation at conditions where the flow into the impeller is at or near the speed of sound at operating speeds that are below the mechanical limit for the operating speed of the impeller. This issue is shared with other compression applications utilizing heavier gases such as propane or other refrigerants. In multistage compressors, this also leads to relatively steep head-flow characteristics with limited operating range. The volume reduction per stage is significant, and, in higher pressure ratio, multicasing compression requirements, it is often useful to use a gearbox between the casings.

- The fluid and thermodynamic properties are well known. The critical pressure and temperature of pure CO_2 (73.5 bara/29°C, 1066 psia/85°F) are well within typical compression applications. At these conditions, there is an extremely high sensitivity of compressibility (i.e., density) and enthalpy to temperature. Lowering the temperature would lead to condensation, but any increase in temperature has significant effects on the required work.

  It also should be noted that any contamination of CO_2 with other substances leads to significant changes in its thermodynamic properties in that area.

- Carbon dioxide alone is inert and not corrosive. It has, however, a high affinity for water, and when combined, forms carbonic acid, which corrodes carbon steel.

Sour Gas Compression

Natural gas containing significant amounts of H_2S and CO_2 is usually referred to as sour gas (as opposed to sweet gas). A number of gas fields produce sour gas, and in many instances the removal of H_2S and CO_2 is part of the gas plant operation (see above). In some instances, sour gas is compressed untreated, in particular when it is used in gas reinjection applications (Hopper, et al., 2008). In particular higher levels of H_2S can lead to sulfide stress cracking of materials in an aqueous environment. In the presence of liquid water, H_2S and CO_2 form corrosive acids, and one of the issues when compressing sour gas into the dense phase region is that water can drop out if the temperature is lowered (Hopper, et al., 2008). ANSI/NACE MR0175 (2003) provides information on requirements and recommendations for the selection and qualification of carbon and low-alloy steels, corrosion-resistant alloys, and other alloys for service in equipment used in oil and natural gas production and natural gas treatment plants in H_2S -containing environments, whose failure could pose a risk to the health and safety of the public and personnel or to the environment. It also defines limits for H_2S partial pressure in the gas, depending on the pH value of the environment, beyond which special material considerations apply. The local pH value depends on the partial pressures of both H_2S and CO_2, the alkalinity of the water, represented by the sum of the bicarbonate (HCO_3) and disulphide (HS^-) contents, the ionic strength of water and, to some extent, the temperature. The high toxicity of H_2S also requires specific attention to avoid and detect leakages.

Refrigeration Applications

The basic refrigeration cycle (Figure 29) involves the heat transfer from the process fluid into the refrigerant (in its liquid state) by evaporation of the liquid. This is a latent heat increase (i.e., the temperature of the refrigerant does not change). The net refrigeration effect (NRE) of the process is defined as the heat per mass the refrigerant can absorb, i.e.:

$$NRE = H_3 - H_2 \tag{18}$$

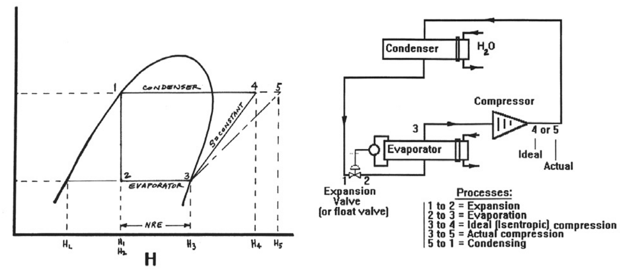

Figure 29. Basic Refrigeration Cycle.

The evaporated refrigerant is then compressed, and the heat from the process gas and the compression process is then transferred to another medium (for example cooling water or atmospheric air). In this process, the refrigerant is liquefied again. This liquid is then expanded to a lower pressure, during which some of the liquid is flashed into vapor, thus reducing the temperature of the liquid. The discharge pressure of the compressor is thus determined by the saturation pressure of the refrigerant at the temperature in the condenser. The condenser temperature in turn is determined by the temperature of the air or water used to cool it.

This is refrigeration process in its simplest form. More complicated cycles include cascades (using a low temperature and a high temperature cycle with different gases), and compound processes using two stage compression with de-superheaters and economizers.

Liquefied Natural Gas

Transportation of natural gas to markets without economic pipeline access is sometimes accomplished by liquefying it to −160°C (−256°F). The liquid natural gas can then be transported by special cryogenic ships to receiving terminals for storage and regasification into sales pipelines. LNG is usually feasible when the gas reserves are large enough to support the large development costs (Figure 30).

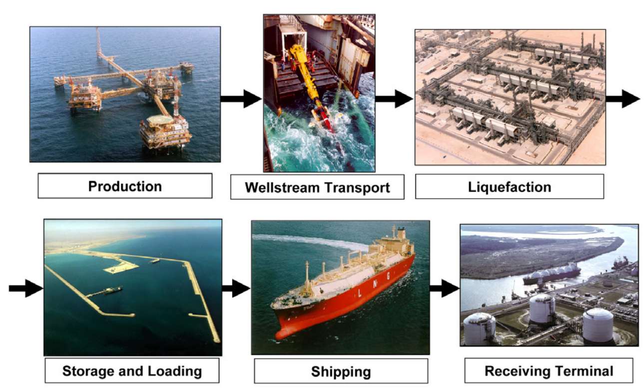

Figure 30. LNG Stations from Gas Production to the Receiving Terminal. (Courtesy ExxonMobil Corporation)

There are seven common LNG refrigeration processes, all of which require significant compression (Figures 31, 32):

1. Cascade: propane, ethylene, and methane refrigerant cycles

2. Single mixed refrigerant: more commonly used in peak shaving plants

3. Propane precooled: mixed refrigerant (C3MR). Used in over 80 percent of existing world liquefaction capacity.

4. Propane precooled: mixed refrigerant hybrid (AP-X). Higher capacity than C3MR.

5. Dual mixed refrigerant (DMR). Replace propane precooling with a mixed refrigerant.

6. Mixed fluid cascade

7. Three mixed

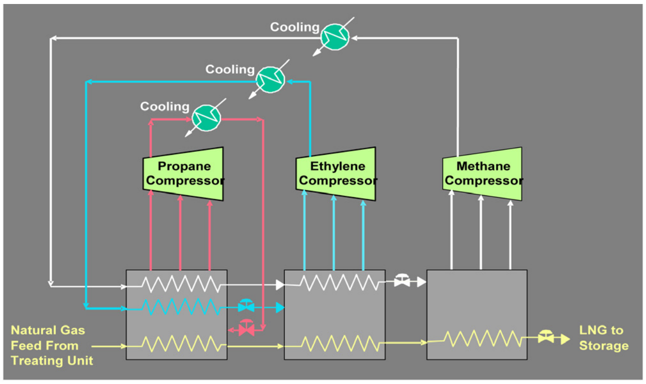

Figure 31. Cascade LNG Process. (Courtesy ExxonMobil Corporation)

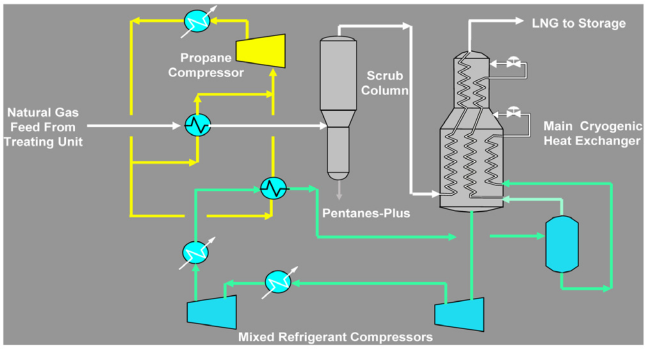

Figure 32. Propane Mixed Refrigerant LNG Process. (Courtesy ExxonMobil Corporation)

These refrigerant compressors operate with mostly fixed compositions and in closed loops. The capacity and power required is usually very large with powers ranging from 20 to 80 MW per compressor casing, and most designs requiring multistage compressors (Figure 33). The compressors are typically driven by gas turbines; however, steam turbines and electric motors are also used. Among the refrigeration cycles mentioned, Figures 31 and 32 outline two frequently used processes, and Figure 33 shows compressors typical for these applications, not the least to give an impression of the size of these machines. Part of the LNG process is also a compressor to recompress gas that has boiled off (Figure 34).

Figure 33. Large LNG Refrigerant Impeller and Mixed Refrigerant Barrel Compressor. (Courtesy General Electric Company)

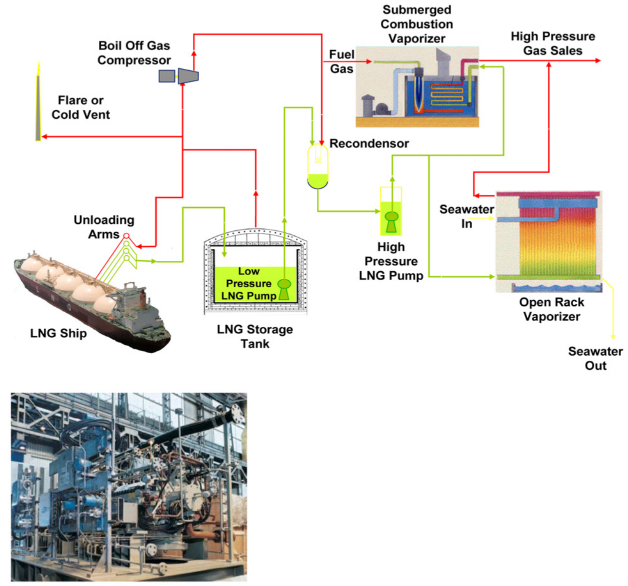

Figure 34. Boil-Off Gas Compressor in the LNG Receiving Terminal. (Photo Courtesy Siemens AG)

Some Lessons Learned

In this section, the authors discuss briefly two application examples that highlight the need for understanding the application to ask the right questions.

On an offshore gas depletion compression application a lesson was learned about asking the right questions. This was a well boosting project where the gas reservoir pressure was declining, limiting production rate. This application called for applying the maximum compression that could fit within the limited platform deck space. Three gas turbine driven centrifugal compressor packages would fit and achieve an initial production rate of 900,000 Nm^3/d (350 mmscfd) with a suction pressure of 55 bar (800 psi) and a discharge pressure of 80 bar (1150 psi). When operations were asked what is the maximum pressure the sales gas pipeline has ever operated, the response was "91 bar (1320 psi)." Indeed the line could be safely operated at this pressure; however, as a pipeline system it was restricted at this facility to 80 bar (1150 psi). The compressors were staged to operate with a discharge of 91 bar (1320 psi)when in fact they were never allowed to operate above 80 bar (1150 psi). This resulted in less production because the compressors were optimized for higher head than necessary as shown below. Using all available power, they therefore operated at higher flows than necessary for best efficiency. This issue also shortened the interval for restaging as the reservoir pressures declined causing acceleration of costs while producing less reserves, a negative impact to the economics.

The right questions to ask are: What is the sales pipeline pressure range? Are there exceptions to this? Is this the appropriate design basis?

Another case: In an onshore pipeline application, two gas turbine driven compressors were sized for a suction pressure of 55 bar (800 psia) and a discharge pressure of 83 bar (1200 psia). This could be accomplished with two single stage pipeline compressors in parallel. Some time later, the pipeline capacity was to be increased by adding a compressor to the station. The increased flow caused a reduction in suction pressure and an increase in discharge pressure. This caused two issues: The increased pressure ratio required two impellers in each of the compressors, and the actual flow through each of the compressors was reduced. While the reduced flow could have been accomplished by a restage, the higher head requirement forced the use of two impellers instead of one. The other option, to use the new compressor as a booster, i.e., the two existing compressors would feed onto the new compressor, was considered, but was unattractive due to the aerodynamic match.

Fortunately, the existing compressors were beam style machines, and had sufficient bearing span to accommodate two impellers, so the final configuration was for three identical units in parallel.

The important lesson is that planning should not just focus on the initial operating conditions, but also consider future modifications and expansions.

CONCLUSION

The authors have explained upstream and midstream applications for centrifugal compressors. Examples included LNG compression, refrigeration, pipeline compression, compression for gas injection, gas lift, gas gathering, export compression, air compression, and others.

As a basis for this discussion, the theory of compression was reviewed, the typical components of a centrifugal compressor, as well as typical configurations were outlined.

The topics of compressor performance characteristics, as well as control options were discussed in the context of the interaction of the compressor with the overall compression system, and thus the actual application. Lastly, challenges from different process gases were addressed and illustrated with specific examples.

NOMENCLATURE

a,b	= Constants
C	= Constant
c_p	= Heat capacity
f	= Friction factor
H	= Head
h	= Enthalpy
MW	= Mole weight
N	= Speed
P	= Power
p	= Pressure
R	= Gas constant
Q	= Flow
T	= Temperature
T	= Time
W	= Mass flow
x	= Distance
Z	= Compressibility factor
ρ	= Density
η	= Efficiency
ζ	= Loss factor

Superscripts

*	= Isentropic

Subscripts

e	= Pipeline delivery (exit)
d	= Discharge
d	= Downstream
m	= Mechanical
PT	= Power turbine
p	= Polytropic
s	= Suction
u	= Upstream
std	= Standard
1	= Suction, inlet
2	= Discharge, exit

APPENDIX A— TYPICAL CONDITIONS FOR A COMBINED GAS GATHERING AND GAS REINJECTION OPERATION

2 Low Pressure/Intermediate Pressure (LP/IP) Trains
LP: p_1=14.5 bara to p_2=37.2 bara, T_1=21°C, MW=22.8, SQ=482,000 Nm^3/d (210-540 psia, 70°F, MW=22.8, 18 MMSCFD)

IP: p_1=36.2 bara to p_2=81.4 bara, T_1=46°C, MW=21.5, SQ=2,000,000 Nm^3/d (525-1180 psia, 115°F, MW=21.4, 75 MMSCFD)

1 High Pressure (HP) Train
p_1=78.6 bara to p_2=372.4 bara, T_1=46°C, MW=20.6, SQ=2,950,000 Nm^3/d (1140-5400 psia, 115°F, MW=20.6, 110 MMSCFD)
After IP compressor: Dehydration and IP separator feed and feed to other platform of 0 to 2,150,000 Nm^3/d (0-80 MMSCFD)

Calculation

LP:
Isentropic head 104.4 kJ/kg (34,920 ft lb/lb) (single body machine)
Actual inlet flow: 1438 m^3/h (846 acfm)
Assumed isentropic efficiency: 80 percent
Assumed mechanical loss: 2 percent
Discharge temperature: 97°C (206°F)
Power consumption: 771 kW (1035 hp)

IP:
Isentropic head 98.7 kJ/kg (33,037 ft lbf/lbm) (single body machine)
Actual inlet flow: 2500 m^3/h (1471 acfm)
Assumed isentropic efficiency: 80 percent
Assumed mechanical loss: 2 percent
Discharge temperature: 117°C (242°F)
Power consumption: 2850 kW (3827 hp)

HP:
Isentropic head 216.3 kJ/kg (72,412 ft lbf/lbm)
Actual inlet flow: 1544 m^3/h (908 acfm)
Assumed isentropic efficiency: 80 percent
Assumed mechanical loss: 5 percent (gearbox necessary)
Power consumption: 9092 kW (12,204 hp)
Discharge temperature: 189°C (373°F)

Theoretically, this duty could be done in a single body, but the discharge temperature would be rather high. A better option would be a two body train, or a single body two section machine, either option allowing intercooling. The overall pressure ratio is 4.74, so each section would be sized for about $\sqrt{4.74}$=2.18 pressure ratio. Any configuration with a gas turbine driver would require a gearbox, because compressors for such relatively low flows would spin significantly faster than the power turbine of a gas turbine in that power class. Assuming 0.7 bar (10 psi) pressure loss for the intercooler, and a cooled temperature of 46°C (115°F), one gets:

Isentropic head 93.4/96.1 kJ/kg (31,261/32,169 ft lbf/lbm)
Actual inlet flow: 1544/644 m^3/h (908/379 acfm)
Assumed isentropic efficiency: 80 percent
Discharge temperature: 115°C/105°C (239°F/220°F)
Assumed mechanical loss: 2 percent + 3 percent for gearbox
Power consumption: 3807 kW+3924 kW+232 kW (gearbox)=7963 kW; (5110 hp + 5267 hp + 311 hp (gearbox)=10,688 hp)

By intercooling the gas, the discharge temperature was not only brought to an acceptable level, but about 800 kW (1200 hp) was also saved in power consumption.

APPENDIX B— TYPICAL CONDITIONS FOR A GAS GATHERING APPLICATION IN A GAS FIELD

Initial conditions:
P1=31.4 bara (455 psia) P2=93.1 bara (1350 psia), T_1= 46°C (115°F), Flow=167,500 Nm^3/h (150 MMSCFD), MW=19.2

Later:
P1= 14.8 bara (215 psia), p2=93.1 bara (1350 psia), T_1= 46°C (115°F), Flow=100,500 Nm^3/h (90 MMSCFD), MW=19.2

Final:
P1= 7.9 bara (115 psia), p2=93.1 bara (1350 psia), T_1= 46°C (115°F), Flow= 72,580 Nm^3/h (65 MMSCFD), MW=19.2

The initial conditions yield a head requirement of 157 kJ/kg (52,700 ft lbf/lbm), and an actual inlet flow of 99 m^3/min (3500 acfm). If a straight through compressor of 80 percent isentropic efficiency is used, this requires about 8050 kW (10,800 hp), with a discharge temperature of 146°C (295°F).

The later conditions require 297 kJ/kg (99,200 ft lbf/lbm) of head, with an inlet flow of 130 m^3/min (4590 acfm). For straight through compression (80 percent isentropic efficiency), this would yield a usually unacceptable discharge temperature of 220°C (428°F), and require 9100 kW (12,200 hp).

The final condition requires 428 kJ/kg (143,200 ft lbf/lbm) of head, with an inlet flow of 178 m^3/min (6290 acfm). For straight through compression (80 percent isentropic efficiency), this would yield an unacceptable discharge temperature of 283°C (542°F), and require 9500 kW (12,750 hp).

With time, the head requirement and the actual flow requirement for the compressors increase (despite the decline in standard flow). Furthermore, intercooling will be required to manage the discharge temperature. Depending on the timelines, it is possible to install initially one compressor on a skid, and add compressors (to the same skid) when necessary. The driver obviously has to be sized for the highest required power. One would start out with a two body tandem or a two compartment single compressor. Since the initial head is relatively low, that tandem can be operated at low speed to match the low inlet flow, and run it faster for the later conditions and higher inlet flow. This works well here, since the power requirements for the later and the final conditions are similar.

The train that matches the initial and later conditions would require the following conditions (IP/HP). Intercooler outlet temperature is assumed to be 54°C (130°F), and the pressure drop is 0.7 bar (10 psi).

Initial:
Head: 76.0/78.3 kJ/kg (25,420/25,200 ft lbf/lbm). Note that it does not make sense to add up the isentropic (or polytropic) head due to the intercooling employed.
Flow: 99/58 m^3/min (3500/2058 acfm)
Discharge temperature: 95/110°C (204/230°F)
Power: 3863/3893 kW (5220/5180 hp); Total (including 2 percent mechanical losses) 7911 kW (10,600 hp)

Later:
Head: 141.0/132.8 kJ/kg (47,100/44,400 ft lbf/lbm)
Flow: 130/51 m^3/min (4590/1810 acfm)
Discharge temperature: 132/144°C (270/291°F)
Power: 4325/4079 kW (5800/5470 hp); Total (including 2 percent mechanical losses) 8570 kW (11,500 hp)

Final:
Since the actual flow is increased significantly in the final conditions, it makes sense to add an LP compressor to the train. With this machine, the authors get the following for the LP/IP/HP combination (depending on the design of the compressors, restaging of some impellers may be necessary):

Head: 120.5/130.5/122.7 kJ/kg (40,300/43,700/41,050 ft lbf/lbm)
Flow: 178/84/34.5 m^3/min (6290/2960/1225 acfm)
Discharge temperature: 118/139/138°C (245/282/280°F)
Power: 2670/2900/2720 kW (3585/3890/3650 hp); Total (including 2 percent mechanical losses) 8455 kW (11,350 hp)

The interstage pressures were selected to basically load the compressor sections approximately equal, and the authors assumed that all compressors would achieve 80 percent isentropic efficiency. In actual designs, the designer may take advantage of loading the more efficient compressor more, or to adjust the interstage pressures to other process requirements (like dehydration, or fuel gas takeoff).

APPENDIX C— TYPICAL EXAMPLE FOR PIPELINE COMPRESSION

Typical requirements are:
P1=55.2 bara (800 psia)
P2=82.8 bara (1200 psia)
Flow=446,000 Nm^3/h (400 MMSCFD)
T1=21°C (70°F)
Gas moleweight: 17.3

Yielding:
Isentropic head 53 kJ/kg (17,790 ft lbf/lbm)
Flow: 130 m^3/min (4600 acfm)
Power (at 85 percent efficiency) 6270 kW (8410 hp)
T2=55°C (131°F)

A number of things should be noted: Pipeline applications usually require a very wide operating range of the compressor. It therefore often helps to load the individual stages lightly. In this example, the head can be generated by a single impeller, but using two stages may improve the efficiency. Also, in many installations, the limiting factor for compressor stations is the allowable temperature for the pipeline coating, often in the range of 55 to 60°C (130 to 140°F). Based on this, decisions have to be made whether an aftercooler is required. The aftercooler may also be appropriate to reduce the pressure loss in the pipeline.

APPENDIX D— TYPICAL STORAGE AND WITHDRAWAL APPLICATION

A typical application requires certain withdrawal and injection rates. The storage cavity can accommodate a maximum pressure of 125 bar (1800 psi) and can be emptied to 20 bara (300 psia), with a pipeline operating between 34 to 70 bara (500 and 1000 psia), and a suction temperature of 15°C (60°F). Gas is of pipeline quality, with a Moleweight of 16.5 (conditions are referred to at the compressor nozzles).

The driver available power is 7450 kW (10,000 hp)
This generates a number of operating conditions:
Storage (S1):
Highest pressure ratio: p1=34.5 bara (500 psia), p2=124.1 bara (1800 psia)
The required head is 204 kJ/kg (68,100 ft lbf/lbm), and allows up to 140,000 Nm^3/h (125.5 MMSCFD), or 68 m^3/min (2390 acfm).

Withdrawal (W1):
Highest pressure ratio: p1=20.7 bara (300 psia), p2=69 bara (1000 psia). This is not the highest head condition. The required head is 194 kJ/kg (64,700 ft lbf/lbm), and allows to flow up to 147,300 Nm^3/h (132 MMSCFD) or 122 m^3/min (4310 acfm) at 80 percent isentropic efficiency.

The highest head and lowest flow occur at point S1. That means, that this point determines the surge flow (minimum flow) requirement for the train. However, point W1, at a much higher flow and almost the same head, will require higher speed than point S1.

In many applications, the train will be a two-body tandem, with the option of running the two bodies in series or in parallel. This is due to the fact that, if the maximum available power is to be used, the compressors initially will operate at a low pressure ratio, both when filling the cavity, and when withdrawing form the cavity.

Therefore, the amount of flow that can be accommodated by the available power is rather large. In this example, for the situation S2, p1=34.4 bara (500 psia), p2=55.2 bara (800 psia), one could flow 424,000 Nm^3/h (380 MMSCFD) or 204 m^3/min (7235 acfm) at 80 percent efficiency. However, this is three times the actual flow at point S1, but the head is only 67 kJ/kg (22,460 ft lbf/lbm). The two compressors can now be run in the train in parallel where they can add their individual flow capacity, while the capability of the train to make head is effectively cut in half—which is still sufficient for the task.

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