45TH TURBOMACHINERY & 32ND PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 12 – 15, 2016
GEORGE R. BROWN CONVENTION CENTER

Your Gas Compression Application – Reciprocating, Centrifugal, or Screw?

Greg Phillippi
Director Process Marketing and Sales
Ariel Corporation
35 Blackjack Road
Mount Vernon, OH 43050
740-397-0311
gphillippi@arielcorp.com

Tim Manthey
Manager – Process Gas Division
Aerzen USA
108 Independence Way
Coatesville, PA 19320
484-718-3702
tmanthey@aerzenusa.com

Bruce McCain
Engineering Consultant
Oxy Permian
5 Greenway Plaza, Suite 110
Houston, TX 77046
713-366-5492
bruce_mccain@oxy.com

Jonathan Sutter
Application Engineer
Elliott Group
901 North Fourth Street
Jeannette, PA 15644
724-600-8362
jsutter@elliott-turbo.com

Ben Williams
Application Engineer
Ariel Corporation
35 Blackjack Road
Mount Vernon, OH 43050
740-397-0311
bwilliams@arielcorp.com

Greg Phillippi is the director of process compressor marketing and sales for Ariel Corporation in Mount Vernon, Ohio. Greg has a BSME (1978) from Ohio Northern University and an MBA (2000) from Ashland University. He began his career as a design engineer with Cooper Energy Services in 1978. In 1985 he accepted a position with Ariel Corporation where he worked as a design engineer and manager through 1999. In 2000 Greg accepted a position with ACI Services, Inc. in Cambridge, Ohio where he had responsibility for marketing, sales and engineering. In January 2004 he returned to Ariel in his present role. Greg has significant experience in the design, application, marketing and sales of reciprocating compressors for oil and gas industry applications.

Jonathan Sutter is an application engineer for Elliott Group in Jeannette, Pennsylvania. Jonathan received a BSME (2008) from The Ohio State University and is a licensed Professional Engineer in Ohio. In 2010 Jonathan started as a project and packaging engineer within Elliott Group's Engineered Products division. He transitioned to application engineering in 2013 and is responsible for equipment selection and proposal preparation.

Tim Manthey is Manager of the Process Gas Division for Aerzen USA in Coatesville, Pennsylvania. Tim received a BSME (2003) from Grove City College and began his career as a design engineer with PacTec Corporation. In 2007 he accepted a position with Aerzen USA as a Project Manager, and shortly thereafter transitioned into technical project management work within the Process Gas Division of the company. As Project Manager, Tim was responsible for complete project execution with special emphasis on technical and quality compliance. In early 2016 he accepted the position of Manager of the Process Gas Division at Aerzen USA. Tim is on the current API 688 task force and has extensive experience with screw compressor system design, application, controls, and components, with additional experience in rotordynamics and machine monitoring practices.

Ben Williams is a process application and account manager for Ariel Corporation located in Mount Vernon, Ohio. His responsibilities include the sizing and selection of Ariel's API618 Moderate Speed Process compressors. Ben has over 30 years of experience with reciprocating compressors' primarily within the refinery and petrochemical industries. He is a graduate of the U.S. Naval Nuclear Power School (1979) and a veteran of the United States Naval Submarine Force.

Bruce McCain is an Engineering Consultant at Oxy Permian. His experience spans over 25 years as an end user, primarily in the upstream sector, providing technical support for rotating, reciprocating, and stationary equipment both as a plant engineer and in a staff functional role. Bruce is a licensed Professional Engineer in Texas, a Certified API 510 Pressure Vessel Inspector, member of the TAMU Turbomachinery Advisory Committee, and is active on the API Std 688 (Pulsation and Vibration Control in Positive Displacement Machinery Systems) Taskforce. He has a BSME from Texas Tech University.

ABSTRACT

This tutorial addresses the question of which compressor type is better suited to a given application - reciprocating, centrifugal or rotary screw- with the screw being subdivided into oil-free and oil-flooded types. The application guidelines will be addressed from the standpoint of reliability, cost, efficiency, size, and other more general application parameters such as molecular weight, compression ratio, and turndown capability.

INTRODUCTION

Choosing the proper compressor type for an application is a critical decision. An uninformed choice will invariably lead to increased operating and maintenance costs. Selecting which compressor type, reciprocating, screw, or centrifugal, to use for a specific application requires consideration of numerous parameters such as suction and discharge pressure level, gas mole weight, and required flow. Furthermore, an understanding of how each type works, along with their specific strengths and weaknesses is essential to selecting one appropriate for a given application.

The tutorial provides guidelines and comparative information to be used by contractors and users to determine which of these three compressor types may be the best fit for their particular application. It is important to note that in addition to reciprocating, rotary screw, and centrifugal compressors, other compressor types may be used. Among other types used are rotary vane, diaphragm, liquid ring, and axial compressors. This document will focus only on reciprocating, rotary screw, and centrifugal compressors as they are the most widely used.

The tutorial is organized into six sections. The first three are "How-To" sections explaining how each compressor type works. The goal of these three sections is to serve as a primer for the rest of the tutorial. The fourth section includes example applications and describes what makes one compressor type better than the others for specific applications. The fifth section provides the end user perspective on factors that must be considered when selecting what type of compressor to use. The sixth section contains a comparison chart for reference when making a compressor selection.

HOW A RECIPRICATING COMPRESSOR WORKS

How a reciprocating compressor works will be explained by reviewing a pressure versus volume diagram (P-V diagram, Figure 1). A reciprocating compressor is a positive displacement machine in that a volume of gas is drawn into the compression chamber where it is trapped, compressed and released. Figure 2 is a simplified cross-section drawing of a double-acting compressor cylinder assembly with some major parts identified. The P-V diagram is a plot of the pressure of the gas versus the volume of the gas trapped in the compressor cylinder's compression chamber. In Figure 1, P_S represents the pressure of the gas at the inlet to the compressor cylinder. P_D represents the pressure of the gas at the outlet from the compressor cylinder. V_{MAX} represents the maximum volume of gas trapped in the compression chamber and V_{MIN} the minimum. The difference between V_{MAX} and V_{MIN} is known as the piston displacement, or how much volume is displaced in one stroke length of the piston.

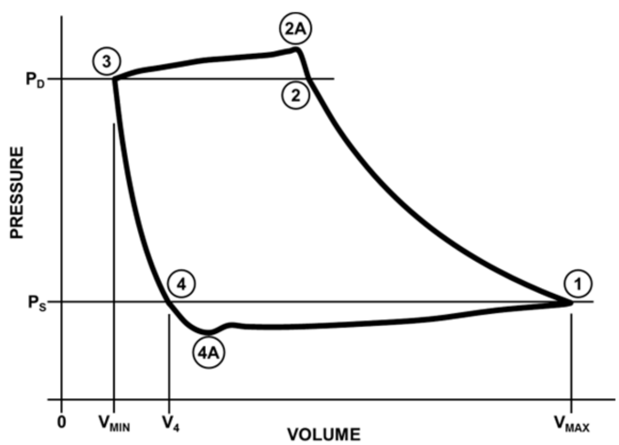

Figure 1: A typical P-V Diagram

Referring to Figure 1, the P-V diagram is made up of four basic segments or events, 1-2, 2-3, 3-4, and 4-1, and each will be explained. Position 1, has the maximum volume (V_{MAX}) of gas trapped in the compression chamber and that gas is at suction pressure and temperature. At position 1 all of the compressor valves are closed and the piston is at rest. All of the compressor valves are simple spring-loaded check valves and are not actuated by any outside means. As the crankshaft rotates and causes the piston to move, the volume inside the compression chamber decreases. As the volume decreases, the gas pressure increases (the gas is trapped in the compression chamber). When the pressure inside the compression chamber becomes slightly higher than discharge pressure (P_D, at position 2), the discharge compressor valve will open. This completes the first segment and is called the compression event.

At position 2, the discharge valve opens and, as the piston continues to move, gas at discharge pressure and temperature is pushed out of the compression chamber, through the discharge compressor valve and into the discharge gas passage.

At position 3, the piston comes to rest, the discharge compressor valve closes and this segment ends. Segment 2-3 is called the discharge event. The piston has moved through one stroke length from position 1 to 3. Position 3 represents the minimum volume of gas trapped inside the compression chamber (V_{MIN}). Note this is not zero volume.

At position 3 the piston will reverse and travel the opposite direction. As it moves the volume inside the compression chamber will increase and the trapped gas will increase in volume and decrease in pressure. Segment 3-4 is called the expansion event.
At position 4, the pressure inside the compression chamber will be slightly less than suction pressure (P_S) causing the suction compressor valve to open. With the suction valve open, the compression chamber is open to the suction gas passage and as the piston continues to move, the volume continues to increase and the compression chamber fills with gas at suction pressure and temperature.

The piston will return to position 1, come to rest, and the process repeats. Segment 4-1 is called the suction event.

This completes the basic reciprocating compression process. One cycle around the P-V diagram represents one revolution of the crankshaft and two stroke lengths of the piston: one stroke from positions 1 to 3 and another from 3 to 1.

The amount of gas that is compressed by this P-V diagram is the volume difference between positions 1 and 4:

$$\boldsymbol{Capacity = V_{max} - V_4}$$

This volume is influenced by the compression ratio and V_{MIN}. The higher the compression ratio, the less gas is compressed. The larger the V_{MIN} is, the less gas is compressed. V_{MIN} is also referred to as the fixed clearance volume.

The power required to compress this quantity of gas is represented by the area enclosed by the P-V diagram, as:

$$\boldsymbol{Work = \int PdV}$$

The compression and expansion events are modeled thermodynamically as adiabatic processes, meaning it is assumed that no heat is transferred to or from the gas during these events.

Inefficiency in this process is essentially the pressure drop incurred in moving the gas from the inlet flange into the compression chamber and in moving the gas from the compression chamber to the outlet flange. Overcoming this pressure drop requires energy. This energy is represented by the areas 4-4A-1-4 and 2-2A-3-2 in Figure 1.

The capacity can be varied in a reciprocating compressor utilizing many different means:

- Varying speed. This is a simple and energy efficient capacity control method. Capacity will vary directly with speed. Varying the rotating speed varies the number of P-V cycles per minute but does not significantly alter the shape of the P-V diagram. As the speed is reduced, the power per capacity improves. A disadvantage of speed variation is that it is highly likely that a torsional natural frequency (or two) will be encountered somewhere in the speed range and operation there will be blocked. This means “infinite” speed range is typically not possible.

- Recycle. This is also very common and simple but is not energy efficient as the compressor will always be operating at a higher power than is necessary.

- End deactivation. This is a mechanical method to reduce the piston displacement. Several devices are available to accomplish this such as finger-type and plug-type suction valve unloaders. Both devices open the head or crank end compression chamber to the suction gas passage during the complete revolution of the crankshaft. Also very common on compressors in process applications, it is essentially an internal bypass and does consume some power making it somewhat less efficient than other methods.

- Add fixed clearance. Mechanical devices are available to add fixed clearance to the compression chamber. Adding fixed clearance changes the slope of line 3-4 in Figure 1 (moves 4 to the right), thus reducing the volume of gas that is brought into the compression chamber during the suction event - volumetric efficiency is reduced. Adding fixed clearance also changes the slope of line 1-2 moving position 2 to the left. Adding fixed clearance does not change power per capacity, thus it is an efficient method. Adding fixed clearance is much more effective (more percentage capacity change for a given percent fixed clearance change) at higher compression ratios.

- Timed suction valve closing. Mechanical devices are available that resemble finger-type suction valve unloaders but only hold the suction valve open for part of the compression event, then allow the suction valve to close and compression to take place. This is an infinitely variable method and slightly less energy efficient than adding fixed clearance.

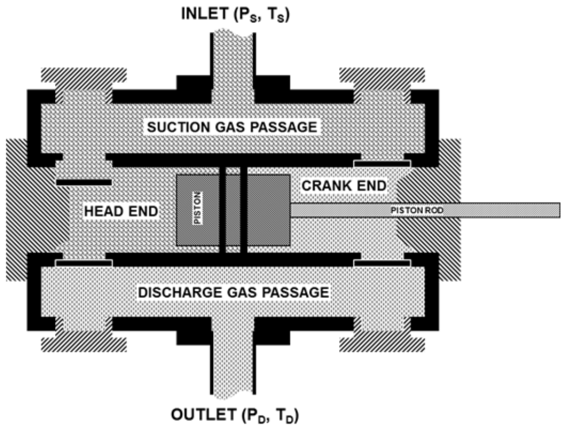

Figure 2: A simplified cross section of a reciprocating compressor

HOW A CENTRIFUGAL COMPRESSOR WORKS

Centrifugal compressors are the dynamic type, meaning that compression is accomplished through the conversion of kinetic energy to static energy. The defining characteristic of centrifugal compressors is that head is determined by the volume flow through the unit. A brief overview of the components will aid in understanding how this type of compressor works (reference Figure 3).

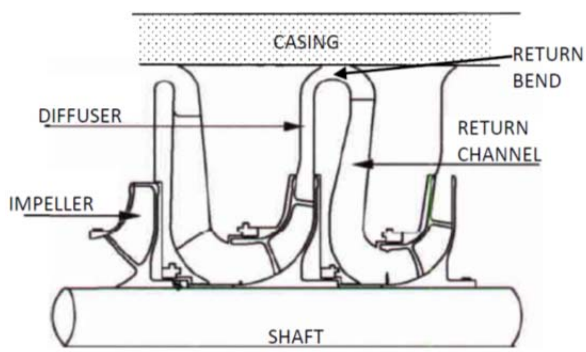

Figure 3: A simplified cross section of a centrifugal compressor

It is worth noting that there are multiple ways flow is directed into the eye of a centrifugal impeller. In the case of an axial-inlet centrifugal compressor (typical of integrally geared units), an appropriate length of pipe serves to smooth the flow of gas into the eye of the first impeller. For the radial-inlet compressor, an appropriate length of pipe and inlet guide vanes direct the gas towards the center of the unit before a 90° bend redirects the flow (axially) into the eye of the first impeller. In the multistage centrifugal compressor, a vaned radial return channel is used to straighten and smooth gas flow preceding a 90° bend into the eye of the next impeller.

The impeller spins on a shaft and is the means by which energy (work) is imparted on the gas. Gas enters the eye of the impeller and once again makes a 90° turn resulting in a radial flow from the center of the impeller to the outer diameter. Additionally the gas encounters rotating blades in the impeller. These blades push the gas in a circular motion resulting in a static pressure rise (compression) due to the centrifugal force of rotation. For the impeller with the typical backward-leaning blade, approximately two thirds of total static pressure rise of the stage is obtained within the impeller. Upon leaving the impeller, the flow has two velocity components: the first is a component in the radial direction, V_R, and the second is a component in the tangential direction, V_T (impeller velocity components, Figure 4). The high velocity gas then enters the diffuser.

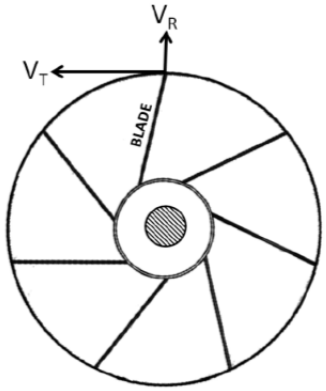

Figure 4: Velocity components at the impeller tip

The diffuser is a stationary component which primarily converts velocity (kinetic energy) to pressure (static energy). The diffuser is a radial passage that is roughly the same width as the impeller blade, but the radial area expands and provides the desired diffusing effect. As the gas exits the diffuser and enters the return bend, the gas has obtained the majority of the static pressure rise for the centrifugal stage (a centrifugal stage being the combination of the impeller and diffuser).

The return bend is found in multistage centrifugal compressors and is a stationary component which redirects the gas from a radially outward flow in the diffuser to a radially inward flow into the return channel which contains the next stage's guide vanes (as previously mentioned, these vanes help to straighten and smooth the gas flow preceding the downstream impeller). The return bend is a 180° passageway that is often partially integral with the upstream stage's diffuser and partially integral with the downstream stage's guide vanes.

Performance of a centrifugal compressor is often evaluated through the compressor's head versus flow curve and power versus flow curve. The shape of the head versus flow curve is very much a characteristic of the impeller geometry (Head vs. Flow, Figure 5).

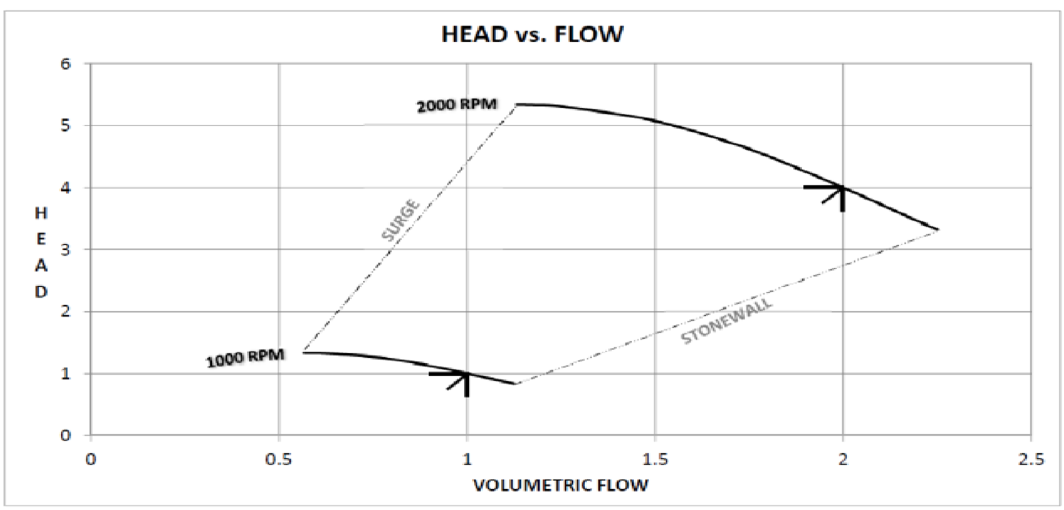

Figure 5: A typical Head vs. Flow curve for a centrifugal compressor

For the typical compressor stage, decreasing the required head increases the volumetric flow. Likewise, increasing the required head reduces the volumetric flow. This can be readily seen by following the head vs. flow curve in Figure 5. The ends of the curve are defined by surge at the low-flow end and stonewall at the high-flow end.

The power vs. flow curve is also stage dependent. For a given stage, the shape of the power curve is a function of flow, head, and efficiency (Power versus Flow, Figure 6).

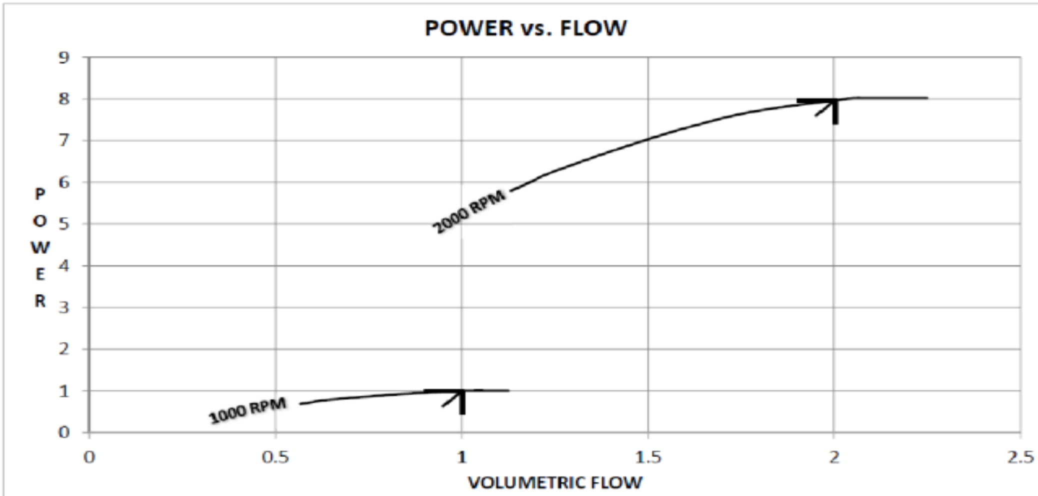

Figure 6: A typical Power vs. Flow curve for a centrifugal compressor

Fan laws are used to simplify performance estimation for centrifugal compressors. The three main relationships of fan law are:

1) Volumetric flow is proportional to speed - $Q \propto N$
2) Head is proportional to the square of speed - $H_{poly} \propto N^2$
3) Power is proportional to the cube of speed - Power $\propto N^3$

Figures 5 and 6 demonstrate these relationships between two hypothetical speed lines. Since these laws assume ideal gas characteristics with constant K and Z values, they apply with reasonable accuracy to single-stage compressors or multistage compressors with low pressure ratios. Large variation in speed and/or high pressure ratios will cause these relationships to deviate from fan laws.

Multiple operating conditions with varied inlet conditions (pressure and temperature) and/or varied gas properties (MW, K, and Z) add complexity to the compressor application. The relationship of these variables is defined by the equation for polytropic head:

$$\mathrm{H}_{poly} = \mathrm{Z}\frac{\mathbf{1545}}{\mathrm{MW}}\mathrm{T}_s\frac{\boldsymbol{n}}{\boldsymbol{n}-\mathbf{1}}\left[\left(\frac{\mathrm{P}_d}{\mathrm{P}_s}\right)^{\frac{n-1}{n}} - \mathbf{1}\right]$$

where temperature is °R, pressure is PSIA, n is the polytropic exponent, and head is ft-lbf/lbm. Recognizing that the head-output of a centrifugal impeller is defined by the volumetric flow rate, one can observe that altering inlet conditions and/or gas properties has a direct effect on discharge pressure. In consideration of multistage compressors, the effect becomes exaggerated through each additional stage of compression.

With respect to controlling operational range, the order of effectiveness is typically as follows:

• Recycling a portion of the flow from the discharge back to the inlet of the compressor is an effective method of control, but reduces compressor efficiency. An additional downside is the potential increase in after-cooler size if the recycle flow is large.

• Speed variation results in a large operating range. The operational map for the compressor is usually limited by mechanical constraints, such as rotor dynamics or impeller stress.

• Adjustable inlet guide vanes add a component of pre-whirl to the gas before entering the upstream impeller. This is a very efficient means of compressor control, however the effect can be limited when there are many stages of compression (usually the guide vanes are only applied preceding the first impeller).

• Throttling via a control valve preceding the compressor is also an effective means of controlling the volumetric inlet flow to a centrifugal compressor but introduces inefficiency into the overall process. Throttling can also be applied at the discharge of the compressor.

HOW A SCREW COMPRESSOR WORKS

A screw compressor is a twin-shaft rotary piston machine functioning on the principle of positive displacement combined with internal compression.

The medium handled is conveyed from the suction port to the discharge port, entrapped in steadily diminishing spaces between the convolutions of the two helical rotors, being compressed up to the final internal pressure before it is released into the discharge nozzle.

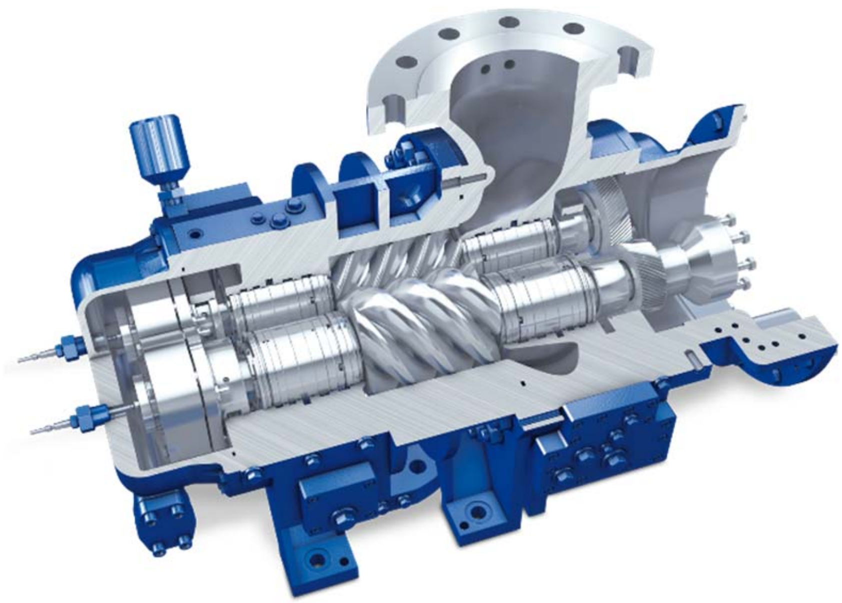

Figure 7: Cutaway of an oil-free screw compressor with dry gas mechanical seals

The spaces referred to are those formed between the cylinder walls and the interlocking convolutions of the two helical rotors. The position of the edge of the outlet port determines the so-called "built-in volumetric ratio", V_i. The "built-in compression ratio", π_i, results from the equation:

$$\boldsymbol{\Pi_i = V_i^K}$$

where K is the ratio of specific heats:

$$\mathbf{K = \frac{C_p}{C_v}}$$

P-V Diagram:

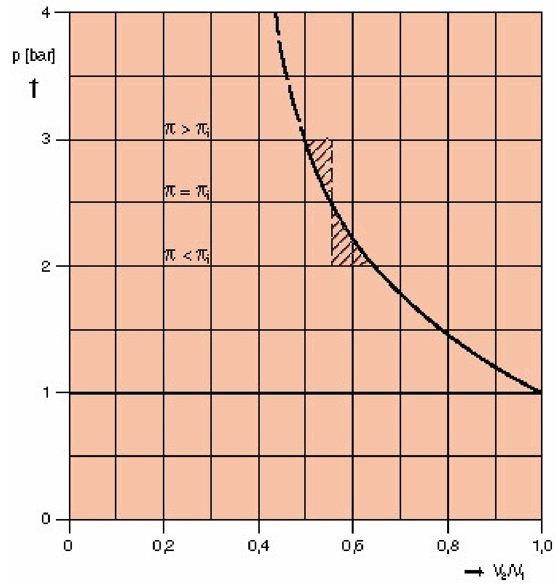

Figure 8: Typical P-V diagram for a screw compressor

The compression that occurs internally in the compressor is isentropic: the volume is being reduced.

The compression that can occur in the discharge header is isochoric: more gas is being pushed into a fixed volume. Since a screw compressor operates on a positive displacement principle, it will continue to move the same volume of gas per rotation regardless of gas density and downstream pressure.

The screw compressor rotor is stiff and has a low moment of inertia. Precision balancing is not as critical and the operating range will normally be below the first torsional critical speed.

Screw compressor pressure limits are normally dictated by temperature, due to the heat of compression; however, rotor deflection and bearing life are also key factors in pressure limits.

In an oil-free screw compressor, sometimes referred to as a "dry" screw, the rotors do not contact one another, but are separated with extremely small clearances via timing gears. Seals at the conveying chamber ensure that the conveyed gas does not come into contact with the lubricating oil or outside air. Thus, the gas is never contaminated by the oil, and vice versa.

In an oil-injected screw compressor, sometimes referred to as a "flooded" or "wet" screw, there are no timing gears. Rather, one rotor drives the other directly. There are also no internal seals. The conveyed gas is mixed with the lubricating oil inside the machine, and gas and oil are discharged together. The oil is then separated from the gas stream before the gas is discharged into the downstream system. The oil is then cooled, filtered, and again injected into the compressor conveying chamber, bearings, and driveshaft seal. Because the gas is mixing with the oil, care must be taken in oil selection to prevent oil contamination by the gas or gas contamination by the oil. Capacity control is possible via an internal slide valve.

EXAMPLE APPLICATIONS

The following section contains three common compressor applications and describes why a specific compressor type (listed first) is better suited for the given conditions. It also provides information on why the other two compressor types may not be optimal for that specific application. The applications were selected to clearly show why one compressor would be the preferred selection.

Hydrogen Product

Gas Compressed – Pure Hydrogen
MW – 2.02
K – 1.41
Suction pressure (psia) – 450
Suction temperature (°F) – 100
Discharge pressure (psia) – 1450
Capacity (MMSCFD) – 63

Reciprocating Compressor – 4420 BHP These conditions are typical of a hydrogen plant application. Due to the low molecular weight of hydrogen and the required discharge pressure, this is an ideal application for a reciprocating compressor. A reciprocating compressor is a positive displacement compressor and therefore the molecular weight of the gas has little impact on the ability of the unit to compress the gas. For optimal performance and reliability, the compressor valves are configured based on the gas being compressed. The required power, capacity and discharge pressure of this application are well within typical reciprocating compressor designs.

Figure 9: Motor-driven reciprocating compressor units in hydrogen service

Due to hydrogen's relatively high ratio of specific heats (K), hydrogen applications typically have low compression ratios between stages. In this case, two stages of compression are required to keep discharge temperatures within acceptable limits.

In many cases, these applications will require load steps. There are a number of capacity control devices available, including suction valve unloaders and clearance addition devices. These are used separately or in tandem to achieve the desired flow rates.

Both lubricated and non-lubricated reciprocating compressors can be used for this application. The decision depends on the process design and whether oil will cause an issue with downstream components. In most cases, lubricated compressor cylinders are used.

Another consideration in hydrogen applications is that there is almost always a nitrogen run required for plant start-up in order to purge and dry the system. The nitrogen operating conditions are quite different than those required for hydrogen and as such the reciprocating compressor can rarely be operated at the same pressures and temperatures as the hydrogen run. That is because the valves are designed for low molecular weight gas and nitrogen is fairly heavy, with a molecular weight of 28. If the reciprocating compressor is operated with the existing hydrogen valves, the time spent operating on nitrogen should be kept as short as possible since this can have a negative impact on valve life.

Centrifugal Compressor - Impractical Despite the inlet flow of 1570 ACFM being within the limitations of many centrifugal compressor manufacturers, the low mole weight of pure hydrogen makes this an unrealistic application for a centrifugal compressor utilizing covered (shrouded) impellers. With a typical covered impeller producing 12,000 ft-lbf/lbm of head (reference 5), over 50 impellers would be required (assuming 85% polytropic efficiency). While multiple centrifugal units are applied quite regularly in other applications, this is clearly not a good fit when contrasted to a reciprocating compressor.

Screw Compressor - Impractical While there is at least one oil-flooded screw compressor on the market that can reach the hydrogen discharge pressure, it is unlikely that it can achieve it in one stage due to slippage and heat of compression concerns. As with other applications, a multi-stage screw compressor is generally at a disadvantage compared to a multi-stage reciprocating compressor because it usually means a separate housing, driveshaft, etc. Therefore it is unlikely that this would be considered a good application for a screw compressor.

Propylene Refrigeration

Gas Compressed – Pure Propylene
MW – 42.08
K – 1.24
Suction pressure (psia) – 50
Suction temperature (°F) – 5
Suction capacity (MMSCFD) – 41.55
Sideload pressure (psia) – 100
Sideload temperature (°F) – 44.5
Sideload capacity (MMSCFD) – 10.39
Discharge Pressure (psia) – 260

Centrifugal Compressor – 3800 BHP Propylene refrigeration is commonly found in ethylene plants and within the NGL (natural gas liquids) portion of natural gas processing. Refrigeration usually involves the use of one or more economizers, thus the necessity of multiple stages of compression. A centrifugal compressor is ideal for this type of service due to the potential to add an intermediate process connection between two impellers. For this particular example, a single body centrifugal compressor having one impeller preceding the sideload and two impellers downstream of the sideload can meet the process requirements. The nature of the refrigeration process makes pressures at the inlet, sideload(s), and discharge critical. The centrifugal compressor manufacturer optimizes performance by scaling aerodynamic components, resulting in a unit capable of operating at the customer's specified conditions for all process streams within tight tolerances.

Figure 10: A centrifugal compressor with an intermediate connection in refrigeration service

Reciprocating Compressor - Impractical Propylene and other heavy gases are frequently compressed with reciprocating compressors. However, in this case, due to the capacity required, the reciprocating compressor would require considerably more power than the centrifugal to meet the flow requirements. The centrifugal is significantly more efficient in this application. This is primarily due to the pressure drop that occurs as the gas moves across the compressor valves (or valve loss horsepower, VLHP) within the reciprocating compressor. To lower the VLHP you would need to reduce the piston speed of the compressor and increase the area of the valves.

Physical size of the compressor cylinders and the flow area of the compressor valves are the practical limits for a reciprocating compressor. If the cylinder is large enough to meet the flow requirements and the valve area is large enough to reduce the VLHP losses, generally the efficiency will be more comparable to the centrifugal.

Screw Compressor – 4370 BHP Oil-flooded screw compressors are often used in refrigeration applications. This application could be done with an oil-flooded screw, but would require a machine that is towards the larger end of size (capacity) available on the market today. The power required would be approximately 15% higher than that of the centrifugal in this case.

A screw compressor would be worth investigating if there are changes in the operating parameters. Screw compressors handle changing inlet conditions very well and maintain a high efficiency across a broad range of conditions. It would be worthwhile getting a comparative sizing of a screw compressor for this application if:

- The flow varies widely
- The load changes regularly (due to changing condensing temperature or other factors)
- The gas has some contamination that could cause imbalance issues for the centrifugal

Flare Gas Recovery

Gas Compressed – Flare Gas
MW – 20.0
K– 1.3
Suction pressure (psia) – 14.7
Suction temperature (°F) – 175
Capacity (SCFM) – 450
Discharge Pressure (psia) – 130

Screw Compressor – 130 BHP Flare gas recovery is a growing application due to the increasing desire to recover all energy sources, as well as increasing restrictions on emissions and flaring. The challenge for a flare gas compressor is to be able to handle varying molecular weights and inlet pressures as the makeup and supply of flare gas changes over time. This is one of the strong areas for a screw compressor – the positive displacement principle means that it is generally insensitive to changing inlet conditions, and the robust nature of the stiff rotor design combined with few moving parts makes it well able to handle occasional upset conditions.

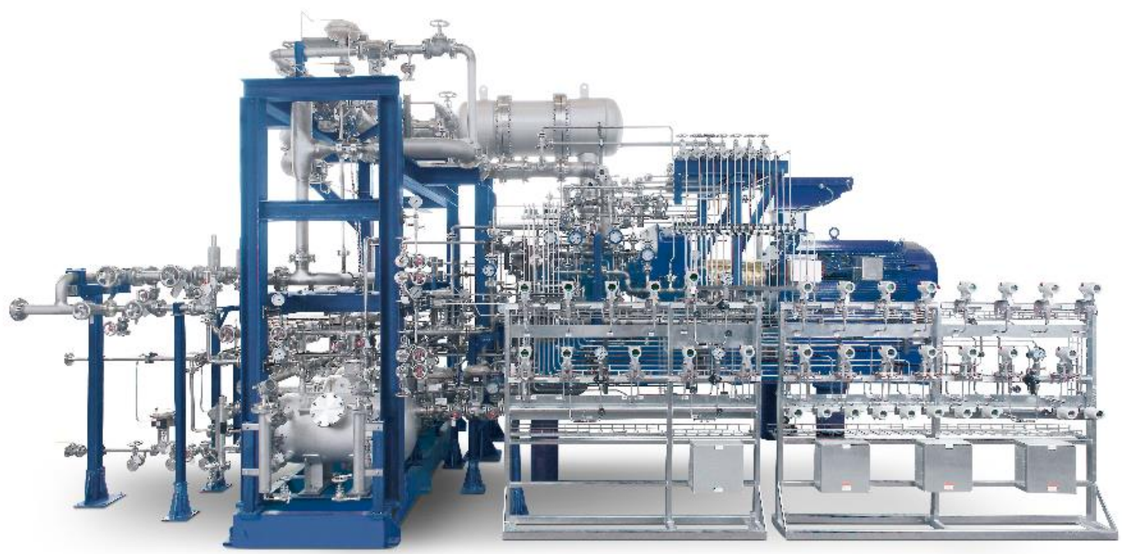

Figure 11: Two-stage oil free screw compressor system for flare gas recovery

With the screw compressor it is interesting to note that if the mole weight of the flare gas varies between 10 and 30, as long as the volumetric flow rate remains constant, the brake horsepower also remains nearly constant.

Flare gas applications often need to maintain either a constant inlet pressure to support the upstream system or a constant discharge pressure to the downstream users. A screw compressor is well suited to either case. For an oil-flooded screw compressor, an internal slide valve can be used for capacity control without varying the operating speed. Oil-free screw compressors are also used in this service, and in these cases a VFD or recycle line are used for capacity control.

If an oil-flooded screw compressor is used in this application, the oil will be in contact with the gas, and thus must be selected based on compatibility with the gas components. Oil-free screw compressors are less susceptible to issues related to gas composition, and as long as the gas is above the dew point (as is true most of the time with oil-free screw compressors), there is little risk to the materials of the compressor.

Reciprocating Compressor - Impractical The challenge for a reciprocating compressor in this application is the varied molecular weight of the gas. Although reciprocating compressors can compress a wide variety of gas molecular weights, in order to achieve the reliability required in the process industry, the compressor valves are configured based on the operating conditions and the gas molecular weight. Since the gas composition varies in flare gas service, it can be difficult to configure the valves to meet the entire range. The valves are never "optimized". In cases where reciprocating compressors are used in flare gas service, the valves are typically configured with low lifts to reduce the valve impact velocity, which can help improve valve life.

Centrifugal Compressor – Impractical In considering the use of a centrifugal compressor for this application, the first problematic attribute is the low flow rate of 550 ACFM. Due to the size of the flow passages in the centrifugal compressor the efficiency will suffer. Secondly, the ratio of specific heats, K, coupled with the lower efficiency will require at least one intercooler. Although there are centrifugal compressors that accommodate connections for up to two process-intercoolers on a single casing, the number of impellers required to reach the final discharge pressure at the lower mole weight of 10 will require multiple compressor bodies (due to rotor dynamics). Finally, there is the large mole weight variation between the operating cases. In theory, a recycle loop could be added after the process-intercooler (every four to six impellers) which would allow recycle flow to be added to each section of compression, thereby counter acting the effect of volume ratio mismatch, however the overall system efficiency and amount of equipment (multiple compressors and an intercooler) would quickly lead one to investigate another type of compressor.

AN END USER'S PERSPECTIVE ON COMPRESSOR SELECTION

As shown throughout this paper, there can be multiple compression solutions for a single application. The end user has to balance technical and non-technical requirements in order to finalize on a particular selection. Non-technical requirements such as familiarity with the machine, delivery, costs, maintenance, parts, quality, and operability can make or break the project. The ability of the compressor selection to handle off design operation (including startup and shutdown) has to be considered. How many times has the reader crossed their fingers and said "If we can just get it started, it will run great?" Oversight of these nuances can have consequences, some not becoming apparent until the equipment is installed.

It should be recognized that while technical requirements are critical for long term reliable service, the engineer may find themselves having to select a machine that is best for the project and not so much the application. The project always has technical constraints, but also commercial and schedule considerations to take into account. Also important are items such as availability of replacement parts, craft skills available for maintenance, repair facility options, staffing, and automation. Ultimately, the end user has to make a technical compromise to select the best machine the project can bear that will be delivered and installed in the promised time, and without exceeding the budget. Everything about machinery selection can be lumped into one of these three categories: Quality, Delivery, and Price. Whether a technical or "soft" issue, quality, delivery, and price should always be brought into the discussion.

As simple as it may sound, this entire process begins with having a thorough understanding of the project requirements. Input from operations personnel provides insight to operational deviations. For example, 95% of the time a gas gathering service in the upstream sector may carry an acceptable amount of water (or other liquids) entrained in the gas, but the other 5% of the time (18 days of a year) the amount of liquids in the gas is beyond the capacity of the scrubbing system, as initially designed. This additional liquid loading could be due to low ambient temperature, high winds, blowing snow, well treating chemicals, pigging of the pipeline, etc. If a reciprocating compressor is applied here, the user may not be able to replace compressor valves as fast as they fail, or scrubber high level alarms won't clear. Considering a centrifugal compressor with a water wash system, the end user may exhaust their reverse osmosis water supply, or the system freezes and there are extended periods without online washing. Alternatively a screw compressor is installed in a gathering service where a new well is tied in with a composition very different than the balance of the system. This can (and has) caused significant lube carryover problems and other lube issues that prevent the screw compressor from operating. In all cases, for one reason or another, the compressors won't run reliably. There may not be an easy solution for each of these problems every time, but the project staff should understand "off design" conditions and practical future expectations. The operations team must be involved when deciding how to handle off design issues that have occurred in the past. More times than not, in the upstream sector, off design becomes the new normal.

It is better to profit from the mistakes of others, or at the very least try to polish up the crystal ball and understand what the machine will be asked to do once project personnel have moved on to other endeavors. This shouldn't be construed as a complete review of considerations but rather a few points to keep in mind when deciding which type of compressor to select.

Reciprocating Compressors

The upstream sector uses a large number of reciprocating compressors. One of the reasons is the high degree of flexibility, or operating range. Reciprocating compressors can be very forgiving for a wide range of operating conditions, so long as those conditions are taken into account during the early phases of the project. It is recommended to seek the advice of operating personnel as they have a good understanding of what happens in reality.

Many installations are unmanned, either initially or are converted to unmanned. The degree of potential automation must be taken into account. If it's reasonable to believe it will eventually be unmanned, then one should design it as such.

One of the biggest problems end users of reciprocating compressors deal with is the change of seasons. The first freeze of the year, or a late freeze in the spring, causes problems if the interstage coolers (finfans) have manual louvers configured for warmer ambient conditions and/or constant speed fans. One successful option is to have automated louvers with variable speed fans on temperature control. This latter option is more costly to implement, but successful in avoiding condensation (water or hydrocarbon) that can carry over into the compressor causing downtime if it gets past the scrubber. Conversely, blowing sand can cause problems with automated louvers. Again, consider everything. A great package design for one location may be a poor design for another.

Suction temperature set points should have a large, practical margin to prevent problems due to pressure drop across bottles, orifice plates, nozzle, passages, and valves. There may not be a liquid level in a suction scrubber but this does not guarantee the avoidance of condensation very close to (or inside) the compressor cylinder due to the Joule-Thomson (JT) effect across the listed components. Properly sized recycle lines allow for startup in cold conditions. Many installations run just fine, if the operation can ever get started. Having a properly sized and placed recycle line with a temperature controller actuating the valve will allow the user to slowly heat the system with relatively hot discharge gas.

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Considering winter operation is not enough- one should consider winter startup from a fully depressurized, cold system. Compressor and engine frame heaters, oil circulation systems and the time it takes to heat those systems (including off skid piping) up should be reviewed.

Reciprocating compressors can handle a large range of conditions with the proper design. Reciprocating compressors generate gas pulsations at harmonics of running speed. These pulsations can couple mechanically with piping, fittings, vessels, nozzles, and if coincident with mechanical natural frequencies can amplify significantly causing component failures, generally fatigue related. There are two schools of thought on this topic: vibration control, and pulsation control.

Vibration control seeks to hold components in place with clamps and supports, and design components such that Mechanical Natural Frequencies (MNF's) have adequate separation margin away from excitations (mechanical and acoustic). Pulsation control minimizes amplitude of pulsation and filters out frequencies above a specific value (Helmholtz Frequency). In reality, both are important. Predominate mechanical excitations on reciprocating compression generally occur at 1x and 2x running speed due the reciprocating masses (pistons, crossheads, etc.), while pulsations occur at all harmonics (1x, 2x, 3x, 4x, etc.). Predicting MNF's is not an exact science and has significant error, depending on stiffness of components/shapes, manufacturing, assembly, etc. When you take separation margins between excitations (mechanical and pulsation) into account, there is "nowhere to hide", as can be seen below in Figure 12. This chart shows an overlay of pulsation frequency range for a 900 rpm (15 Hz) reciprocating compressor (reference the blue bar at harmonics of running speed within ±10% range). Consider a mechanical natural frequency tuned halfway between the 2nd and 3rd order (37.5 Hz) pulsation frequencies. Applying a 10% margin around this mechanical frequency would yield a range of 33.75 to 41.25 Hz. As can be seen on the chart, there is overlap between the mechanical natural frequency range (red) and the pulsation frequency range (blue). Closer inspection of Figure 12 reveals that one cannot effectively design a system with mechanical natural frequencies between pulsation harmonics above 2x running speed- there will always be overlap, in other words, no separation margin. Using a variable speed driver further complicates the operation by having an even broader range of potential excitation frequencies. Understanding this concept is critical to the reciprocating compressor package engineer and user. All practical operating scenarios must be taken into account when designing the pulsation control system. Pulsation is important and must be controlled to maintain long term reliability.

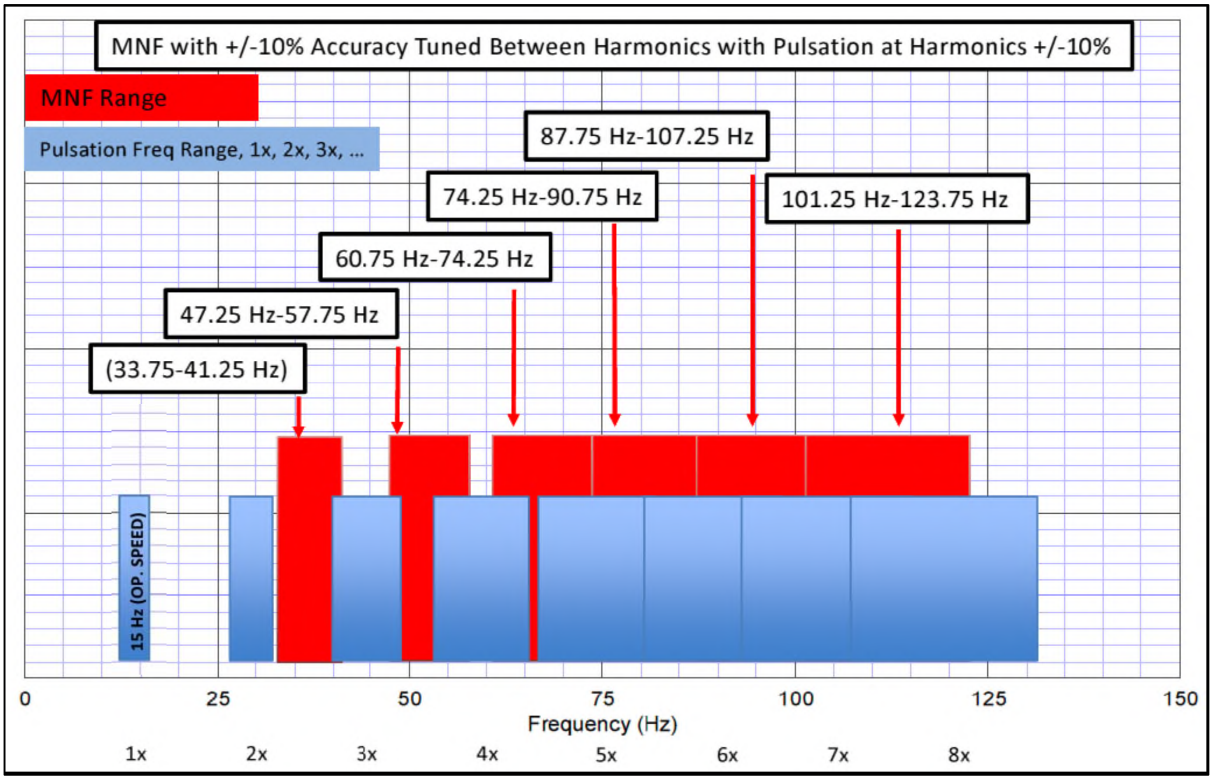

Figure 12: Overlap of pulsation frequencies and mechanical natural frequencies

Centrifugal Compressors

A typical upstream gathering application can be a rather difficult service as the gas is generally wet and can have a significant range of composition and pressure as the field matures over time. H2S and CO2 may increase which increases both corrosiveness and molecular weight. An accelerated drilling program may increase flow and pressure. Well chemicals can increase liquid handling requirements and add components to the gas stream that reacts with heat of compression causing fouling on impellers, balance lines, and cooler tubes. Neglecting these issues with a centrifugal compressor (or any other for that matter) can cause significant operational difficulties. What should one consider when looking at a centrifugal compressor for a particular application?

Understanding gas composition, specifically mole weight, is critical when considering centrifugal compressors. If the mole weight has a large range, centrifugal compressors may not be the best choice. As mole weight drops for a given operating point, speed has to increase (along with power requirements) to maintain mass flow and discharge pressure.

It is imperative to understand what it is the compressor will be (or might be) moving. Look at other equipment that may already be installed in the same gas stream. For example, if the same gas stream has a reciprocating compressor installed, is there any contaminant build-up on the reciprocating compressor's valves? Regardless of what a gas analysis says, these contaminates should be understood and taken into account. If installing centrifugal, screw, and/or reciprocating compression in parallel or in series, consideration should be given to acoustic interaction between the machines.

Understand what degree of conditioning the gas needs: scrubbers, coalescing filters, etc., and how they will be maintained. If little to no gas conditioning is to be installed in a typical gathering service, there could be significant operational impact to a centrifugal machine. Gas conditioning is critical to reliable operation in dirty services.

Understand requirements for compressor washing, both volumes and frequency. If water will be used, there will be cleanliness requirements. Does the facility have the capability of making that water, will facilities be constructed, or will the wash water be trucked in? The volume and frequency of the wash should also be discussed with the compressor manufacturer to ensure erosion throughout the gas path does not occur. Along with the wash system, injection nozzles must be installed through the compressor casing or inner casing/barrel assembly. Installation of wash nozzles is recommended, even if not needed during the early stages of operation due to unknowns with field maturity, well chemical program, drilling program. As this will be part of the compressor fabrication, wash nozzles should be specified early in the project.

Make sure the inside of the centrifugal machine (wetted parts) can handle the range of composition, including off design operations like pigging, well work, and the addition of new wells. Be very cautious of applying coatings to impeller wheels as there is difficulty in getting a sprayed-on coating deep inside an impeller. Also, be aware of potential damage to coatings due to abrasives from the product stream. Ultimately, a better solution is to ensure the wetted parts metallurgy can handle everything the gas could potentially contain. These discussions should be had early with the compressor manufacturer.

If purchasing a spare bundle assembly (vertically split casing, aka “barrel” machine) or spare rotor (horizontally split casing), where will it be stored? Assure there is adequate lifting capacity, overhead space, and anchor points to allow the bundle or rotor to be placed horizontally or vertically as needed.

Screw Compressors

Flooded screw compressors are often considered for applications with low suction pressure and a relatively clean gas, especially when a small footprint is desired. Flooded screw compressors are often selected over dry screw machines when a large compression ratio and single casing is needed. High compression ratio, and resulting high heat, requires a synthetic lubricant be used to seal the clearance in the screws and provide cooling. Dry screw compressors operating at higher temperatures may need special casing metallurgy. However, dry screw compressors are more suitable for dirty gas services, high water content, or has other properties that would contaminate the oil of a flooded screw.

For oil-flooded screws, accurate gas composition is critical so the correct lubricant can be selected. If the gas composition changes before or after startup, this must be addressed immediately to minimize risk of lubricant carryover and/or impact to machinery health. Oil-flooded screw compressors in upstream oil and gas service typically utilize a synthetic oil. In some services, the large expenditure for synthetic oil may be offset by lower frequency of oil replacement. Oil usage and costs should be taken into account when performing life cycle cost analysis: both initial fill and subsequent oil replacement. Some end users have been guilty of only looking at unit cost (per gallon) comparisons of synthetic and mineral oils. Synthetic oil unit costs may be more expensive, but the entire operation must be looked at. Mineral oil may breakdown faster, resulting in a significantly higher volume of mineral oil being used. The end user should also be aware that this premature breakdown could result in more maintenance requirements for the compressor. For these reasons, detailed discussions should be had with the screw compressor manufacturer and operations personnel during the selection process.

Volume ratio, V_i, should always be considered when evaluating screw compressor applications. V_i along with the ratio of specific heats (K), determines the internal pressure ratio, as previously described in the How A Screw Compressor Works section. Departure from the design V_i for gases should be understood before selecting a specific machine. If the volume ratio is too low for the system operating pressure, then under-compression will occur. Under-compression is an event where the gas being compressed in the screw thread nearest the discharge port does not reach the actual discharge piping pressure. As soon as that volume of gas is exposed to discharge piping pressure, a pulsation occurs from the discharge piping into the discharge port. If the volume ratio is too high, then over compression will occur. In this scenario the pressure at the discharge port is higher than the discharge piping pressure. As soon as the gas is exposed to the discharge port, a pressure pulse moves from the screw thread towards the discharge piping, thus equalizing pressure. Both under-compression and over-compression will require more power to move the gas. Since most screw compressors have a fixed V_i, this is often unavoidable, so the key is to select V_i for the normal running condition.

Additionally, both events will cause gas pulsations and will typically have the greatest amplitude at the Pocket Pass Frequency (PPF). This attribute is potentially damaging: the pulsations can mechanically couple with piping and generate vibration. If coincident with piping mechanical natural frequencies, then amplification can occur resulting in very large vibration amplitudes and resulting stresses on nozzles and attachments.

The design of the piping should take into account the Pocket Pass Frequency. Piping supports and clamps should impart the appropriate stiffness to allow required separation margin between excitation and mechanical natural frequencies. The key takeaway on this topic is that V_i is as important as any other consideration. For more detailed information on V_i related concerns, see reference 6.

Driver Considerations

Many projects do not consider drivers until after compressor selection takes place, which can cause delays and unnecessary rework if there is a driver requirement that limits compressor selection. Some decisions are easy- no electric power is (or will become) available, the user has a preference towards a certain driver type (e.g."we only install gas engines"), or one wishes to avoid the major issue of regulatory permitting that accompanies the use gas turbines or gas engine drivers. Whatever the case, here are a few considerations for each driver (note that this is not meant to be an all- inclusive list).

Electric Motors Electric motors, whether induction or synchronous, are a good driver selection for all the discussed compressor types and have lower maintenance costs, reduced downtime, and improved efficiency when compared to gas turbines and gas engines. Electric motors are generally more accepted in unmanned facilities due to their simplicity and reliability. The choice between induction and synchronous motors should take power factor and the power grid into account. Power factor correction is possible with synchronous motors and a credit can be taken in life cycle calculations. Consultation should be made with the appropriate expertise when deciding on a specific type of electric motor.

Recognizing that an electric motor will need an electric power supply is clear, but often this is where the forethought ends. The end user should ask themselves the following:

- Is the power supply already at site? If not, when will it be available? Can the wires handle the startup of the machinery? This could be critical path for startup and can cause significant delays up to the point of sacrificing machinery warranties.
- Will there be sufficient power available to support startup, not just normal operation? Startup philosophy needs to be discussed very early in the project. Different types of machines have different power requirements (in-rush current) during startup.
- Is the power clean or "dirty"? Power with harmonic issues can cause torsional problems that can be detrimental to any machine.
- Is the proposed location of the installation at the end of the grid? Other customers, even residential locations, can be negatively impacted by the project which can cause a public relations nightmare.

Gas Turbines If the site has primarily gas turbines in service, and the plan is to install a gas turbine driver, reciprocating compressors may not be the best compression choice. Some end users have done it, and many have proposed it, but there are significant challenges to overcome such as gear box design and ensuring torsional integrity of the train. Screw compressors would also typically require a speed decreasing gearbox, which can be integral to the oil-free screw compressor and supplied as a single unit from some manufacturers. A centrifugal compressor is commonly paired with a gas turbine driver. Care should be taken to understand impact of conditions such as ambient heat, air contaminants from salt spray or fog, blowing dust, and fuel quality. Whether in the fuel, air, or wash water, chlorides and H_2S can have a significant impact on time between hot-section-overhaul of the gas turbine.

Gas Engines Screw compressors and reciprocating compressors are generally paired with gas engine drivers. Utilization of gas engines requires a good understanding of fuel gas composition. Many engines have fuel gas heating value limits, ethane limits, dew point requirements, and H_2S limits. Be wary of providing a "typical" gas analysis to engine manufacturers. When utilizing "inventory" engines, take special care to assure the engine will be provided with appropriate equipment to handle the expected fuel gas in both normal and off design operation. When having the torsional design performed, assure that the system can handle failure of a power cylinder. Also, assure the torsional damper is sized correctly for the full range of expected operation. Again, when using an "inventory" engine assure it is compliant with recommendations from the torsional study. For example, if a double damper is required, then the end user should verify a double damper has been included with the engine. Compressor turndown should take into account that engine drivers can experience poor operation when in an unloaded state. Both gas turbines and gas engines may have significant de-rates that apply due to site conditions including elevation, gas composition, ambient temperature.

Similar to the electric motor, consideration must extend past the existence of the energy source. There may be a fuel gas source, but not all gas is the same. If planning to install a gas turbine or engine, consider the following.

- What type of gas (i.e. composition) will be used?
- Is the gas source a pipeline or a well head? If the latter, has sufficient conditioning been taken into account for the project?
- Is sufficient gas available for "blowing the line" from the sales gas line to the machinery without negatively impacting other users? Sufficient velocity is needed to clean the piping.
- Have sufficient plans been made for pigging the gas supply line (typical for gathering system gas)?
- If there is H_2S present in the gas, have the correct modifications been made to the engine to allow for it? If installing a gas turbine, have the economics taken into account the performance and life degradation for operation with H_2S?

Steam Turbines If selecting a steam turbine driver, a centrifugal compressor is the predominant selection since centrifugal compressors typically operate in the same speed range as steam turbines (2,500 to 20,000 RPM), thus a gearbox would not be required. Screw compressors do operate in this speed range, but not as typically as centrifugal units. Many screw compressors and virtually all reciprocating compressors would need a speed reducing gearbox if used with a steam turbine. The increased complexity of the installation by installing gearboxes and controlling torsional related issues may make this option both technically and commercially unattractive.

COMPRESSOR SELECTION CRITERIA

The Natural Gas Processor Suppliers Association (NGPSA) publishes an Engineering Data Book (reference 1) and included in the book is a chart showing the range of application for several different types of compressors. The chart (Figure 13) is reproduced here for reference. Applications which fit inside the regions defined by the various lines signify the application fits that machine. As depicted in the chart there is a fair amount of overlap involving reciprocating, screw, and centrifugal compressors. Determining which type of compressor is a best fit for a specific application requires an understanding of how these compressors operate and the tradeoffs associated with a particular selection.

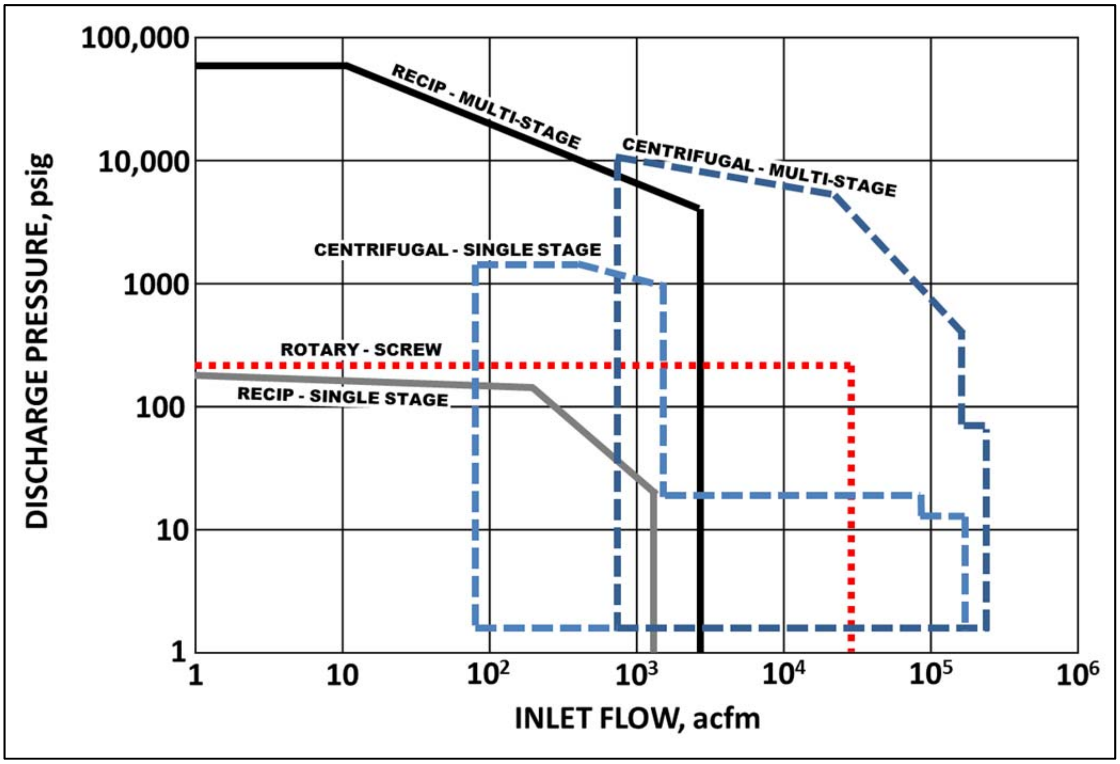

Figure 13: Compressor type based on discharge pressure and flow (NGPSA)

Note: The following table is strictly a guideline and is based on the opinions and experiences of the authors. When determining which compressor is best suited for an application, it will be necessary to consider multiple criteria. A ranking value of 1 is considered superior within this table.

Topic	**Reciprocating**	**Centrifugal**	**Rotary Screw**	
			Oil-Free	**Oil-Flooded**
High Mole Weight (> 25)	Gas mole weight has no direct bearing on the reciprocating compressor's ability to compress gas. However, high mole weight gases can impact the ability of the compressor valves to function properly. Ensure proper lubricating oil selected.	Compression of a high mole weight gas is an ideal application for centrifugal compressors.	A dry screw will convey and compress gas regardless of mole weight. The mole weight of course impacts machine limits, but there is virtually no mole weight that is inherently unsuitable for a dry screw.	Gases with a mole weight >50 (heavier hydrocarbons) do not readily separate from the oil and can dilute it.
Ranking	**3**	**1**	**2**	**4**
Low Mole Weight (< 6)	Gas molecular weight has no direct bearing on the reciprocating compressor's ability to compress gas. With proper valves selection, low mole weight gases are easy to compress with a recip. This is an ideal application.	Similar to the compression of high mole weight gases, low mole weight gases are suited for centrifugal compressors when a high flow is desired.	Good at low differential pressures. At higher differential pressures, the slippage leads to inefficiency and high discharge temperatures.	Ideal since injected oil closes internal clearances.
Ranking	**1**	**2**	**4**	**3**
Wet Gas	Every effort should be made to prevent liquids from entering the compressor cylinder and compression chamber. Liquids can cause compressor valve failures and can dilute the lubrication in the compression chamber.	A centrifugal compressor can be used in a wet gas application if the size and quantity of any liquid droplets are very small, and process gas temperature limits can be met.	Generally not a problem. If droplets are expected to enter a dry screw, stainless steel should be considered to prevent erosion.	Can be done, but the discharge temperature should be kept >20°F over dew point to prevent dilution of oil by condensation.
Ranking	**2**	**2**	**1**	**4**

Topic	Reciprocating	Centrifugal	Rotary Screw	
			Oil-Free	**Oil-Flooded**
Dirty Gas	Solids in the gas stream can cause premature wear of sealing components thus leading to reliability issues. Gas with significant entrained solids should be filtered prior to entering a recip.	Depending on size and quantity, particulates and liquid droplets can have a significant erosion effect due to the high gas velocity within the machine. Performance will suffer due to the particulates' effect on aero components.	Small particles will pass through a dry screw without issue. Larger particles can cause scoring on rotors and casing.	Particles are not much of an issue unless they dilute or otherwise change the oil properties. Dirt will be caught in filters before the oil returns to the compressor.
Ranking	**3**	**4**	**1**	**2**
Low Suction Pressure (< 5 psig) **(high flow)**	While a recip can certainly compress gas from a low suction pressure it can make it relatively expensive. Low pressure typically means multiple large compressor cylinders which will add cost.	Suction pressures slightly above atmospheric are typical for some centrifugal applications. Sub-atmospheric conditions also can be accommodated.	Ideal for a screw, which can also be used for vacuum applications.	
Ranking	**3**	**1**	**2**	
Low Discharge Pressure (< 300 psig)	All compressor types work well with low discharge pressures.			
Ranking	**1**	**1**	**1**	**1**
High Suction Pressure (> 500 psig)	No issues at all for a recip.	A high suction pressure is within the mechanical design limitations of a centrifugal compressor. The frame size and quantity of hydrogen present will dictate the casing type (horizontally versus vertically split).	Few screws are suitable for this operating pressure. Due to internal compression, suction pressures this high must be treated with great care.	
Ranking	**1**	**1**	**3**	

Topic	Reciprocating	Centrifugal	Rotary Screw	
			Oil-Free	**Oil-Flooded**
High Discharge Pressure (> 3000 psig)	No issues at all for a recip.	Discharge pressures up to 15,000 psig can be accommodated through the mechanical design of the compressor. These high pressures would necessitate a vertically split casing.	Screw compressors cannot reach this pressure.	
Ranking	**1**	**2**	**3**	
Capacity Control (Turndown)	Many capacity control devices are available including suction valve unloaders and devices that add fixed clearance. Speed variation, suction throttling and recycle can also be used for capacity control.	Since head and flow are directly related in centrifugal compressor performance, methods of control (such as suction throttling, speed variation, or adjustable inlet guide vanes) are employed to achieve flexibility. Ultimately, aerodynamic events (surge and stonewall) will establish the limits of operation.	Turndown via speed control is possible but limited, especially if maintaining a constant discharge pressure.	With a capacity slide and sometimes a vi slide, an oil-flooded screw is specifically designed for varied operating cases without need for recycle or speed variation.
Ranking	**1**	**3**	**4**	**2**
Changes in Gas Properties *Positive displacement compressors (reciprocating and rotary screw) can handle changes in gas properties equally well.*		The amount of head that a centrifugal compressor stage can impart on a gas is highly dependent on mole weight, geometry, flow, and speed. With the geometry of a stage being fixed, variations in mole weight can be accommodated to a limited degree.		Ensure lubricating oil is suitable for changes in gas properties.
Ranking	**1**	**3**	**1**	**1**

Topic	Reciprocating	Centrifugal	Rotary Screw	
			Oil-Free	Oil-Flooded
Vibration	Vibration is certainly more significant for a recip than the other machines. Adherence to the recommendations of the pulsation study can alleviate possible vibration issues.	At-speed-balancing of the rotor, tight tolerances, and a continuous load on the rotating rotor result in very low vibration levels for a centrifugal compressor.	Generally low, but design of pulsation dampener and piping/supports is important to avoid high-frequency resonance.	
Ranking	**3**	**1**	**2**	
Capital Cost	Capital cost of a reciprocating compressor system is dependent on the gas composition, customer specifications and the application. Operating temperatures and pressures will also impact the system cost.	The capital investment for a low flow centrifugal compressor can be quite high relative to other compressor types. In contrast, savings can be realized when the application requires large flows that can be accommodated in a single centrifugal compressor body.	An oil-free screw compressor for process gas will almost always be higher in capital cost than an oil-flooded for the same duty, mainly due to internal seals and the seal support systems.	Can be quite reasonable with base materials and design, mainly because there is no need for internal seals, and thus no external seal gas system.
Ranking	**2**	**4**	**3**	**1**
Life Cycle Cost	A reciprocating compressor has more wearable parts than centrifugals or rotary screw compressors. Wear part replacement is typically required every one to three years.	Due to the low number of moving parts, constant loading, and low vibration, it is common for centrifugal compressors to be in continuous service for seven years or more.	Very few wearing parts. Five years between overhauls is normal; often longer. Wear is prevented indefinitely if the oil quality and sealing medium quality are properly maintained. Overhaul consists of bearings and seals if needed. Overhaul cost is <10% of new machine cost, generally <3% of complete package cost.	
Ranking	**4**	**1**	**3**	**2**

CONCLUSIONS

The Compressor Selection Criteria table will provide perspective on the ideal compressor type for a single characteristic of an application but by no means will the table provide the solution to an application. Additionally, the comparisons within the table are general and may not be representative of a particular manufacturer's special feature or design. Careful consideration of all of the application characteristics and working with the equipment manufacturers is the best way to ensure the right equipment is being utilized.

NOMENCLATURE

c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
H_{poly}	Polytropic head
K	Ratio of specific heats
MW	Mole weight
N	Rotational speed, RPM
n	Polytropic exponent
P	Pressure
P_D	Discharge Pressure
P_S	Suction Pressure
Q	Volumetric flow
T_D	Discharge temperature
T_S	Suction temperature
dV	Change in volume
V_i	Built-in volume ratio of a screw compressor
V_{MAX}	Maximum volume of gas within a reciprocating compressor chamber
V_{MIN}	Minimum volume of gas within a reciprocating compressor chamber
V_R	Radial velocity of the gas within a centrifugal compressor impeller
V_T	Tangential velocity of the gas within a centrifugal compressor impeller
Z	Compressibility factor
π_i	Built-in compression ratio of a screw compressor

FIGURES

REFERENCES

1. Figure 4: ***NGPSA Engineering Data Book***, Volume 1 Revised 10th Edition 1994. Compiled and edited in co-operation with the Gas Processors Association. Copyright 1987 Gas Processors Association.

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