APPLICATION AND OPERATING HISTORY OF MODERATE-SPEED API 618 RECIPROCATING COMPRESSORS

by

George M. Kopscick

Director of Marketing and Process Sales

Ariel Corporation

Mount Vernon, Ohio

George M. Kopscick is Director of Marketing and Process Sales for Ariel Corporation, in Mount Vernon, Ohio, and is involved with their entry into the downstream market. He joined Ariel in 1997 as Director of International Sales. Mr. Kopscick has worked in the compression industry since 1974, beginning as an application engineer for Ingersoll-Rand working on centrifugal compressors for refinery applications. Other positions with Ingersoll-Rand included involvement with gas turbines, hot-gas power recovery trains, and reciprocating compressors for both upstream and downstream applications; creating and operating the project management department in their England facility with department responsibility for final contract negotiations through shipment and ultimately startup of the equipment; Marketing Manager for the Centrifugal Air Compressor Division; and sales of third party own and operate contracts for large gas compression and gas processing facilities. Mr. Kopscick is also a member of the API 618 Fifth Edition Task Team.

ABSTRACT

API Standard 618 (1995), "Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services," is written to cover both moderate-speed and low-speed reciprocating compressors for critical service applications but does not define the piston and rotating speed characteristics of each design. This paper looks at historic piston and rotating speed limits, what these limits are today, and how these two design alternatives can be applied to achieve the reliability objectives set forth in API 618. The basic differences between the two alternative designs are reviewed in light of both physical characteristics of the designs and changes in materials and manufacturing technology. Finally, the design, applications, and operating history of a number of moderate-speed units operating in refinery service are described and discussed.

INTRODUCTION

API Standard 618 (1995), "Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services," has evolved from its first edition, issued in April 1964, which covered only low-speed compressors for refinery service, to a standard for "moderate-to-low speed" compressors for critical petrochemical, chemical, and industrial gas services. The specification's lack of a definition for moderate- and low- speed compressors, together with no quantitative limits on either piston or rotating speed, has led to confusion and misunderstanding as to what constitutes a moderate-speed compressor and its acceptability for critical service applications.

The objective of this paper is to provide a definition of low- and moderate-speed compressors and the differences between the two designs. Additionally, piston and rotating speed limits for the refining industry are quantified. Design and application limits needed to ensure reliable operation is discussed for both design alternatives. Operating history of various moderate-speed units is provided.

API 618

The API 618 specification is an accumulation of design and application guidelines based on operating experience with reciprocating compressors. Its primary purpose is to provide minimum compressor design and application requirements to achieve safe, reliable compressor installations with long maintenance intervals. The first paragraph of Section 2, Basic Design, of API 618 (1995)states two general design objectives:

- Provide a minimum of 20 year service life
- Provide three years of uninterrupted service

It qualifies the second objective with the statement, "It is recognized that this is a system design criteria." This was added in recognition that the uninterrupted time between required maintenance is influenced by things outside the control of the compressor supplier such as proper process gas composition and correctly operating the unit within the application range for which it was designed. It also recognizes that the technology needed to ensure operating a nonlubricated unit for three uninterrupted years may not be available.

API 618 is continually updated to incorporate new knowledge and experience. The benefits gained from the specification have led to an expansion of its application to new or alternative compressor designs and to additional industries. API 618 originally applied only to low-speed reciprocating compressors for refinery service. As new editions of the specification were written, the scope was expanded to include moderate-speed compressors as well as chemical and gas industry applications.

API 618 (1995) states that it is applicable to both low- and moderate-speed compressor designs, but does not define these terms or designs. The primary reason for this is because a commonly accepted industry definition does not exist. To add to the confusion, the term "moderate-speed" does not really exist in Europe. Most European companies refer to compressors as either low- or high-speed designs.

Rotating and piston speed limits are also not defined in API 618 (1995). The specification includes the following guidance under paragraph 2.2, "Allowable Speeds":

> - "Compressors shall be conservatively rated at a speed not in excess of that known by the manufacturer to result in low maintenance and trouble-free operation under the specified service conditions."
> - "The maximum acceptable average piston speed (in meters per second or feet per minute) and the maximum acceptable speed (in revolutions per minute) may be specified by the purchaser where experience indicates that specified limits should not be exceeded."

Because no piston or rotating speed limits were established, many purchasers incorporated them into their company specifications. Most customer limits were established initially for the low-speed designs covered in the first edition of API 618. Unfortunately, as technology changed and the moderate-speed design was added to API 618, many of these limits were never amended.

API 618's compressor design requirements are stated in Section 2, "Basic Design." The first edition of the specification based these on experience and knowledge gained from low-speed reciprocating compressor installations. As technology and the specification evolved, there was no segmentation of design requirements that did not apply to either low- or moderate-speed designs. Rather, the paragraphs are made applicable to both designs with the proviso that the user has the option to accept alternative designs.

Moderate-speed designs were created years after the basic low-speed designs were developed and incorporated technology that was not available previously. This enabled them to accomplish the reliability and durability objectives of API 618 using different designs than those used in low-speed units. The differences result in some deviations to specific design paragraphs of API 618, but not the overall objectives. The main deviations will be defined and discussed later in this paper.

DEFINITION OF LOW- AND MODERATE-SPEED

No formal or generally accepted definitions of low- and moderate-speed units exist in the compressor industry. While some relate it to the piston speed, most people think of compressor speed as the rotating speed of the crankshaft.

Low- and moderate-speed compressors are categorized by their rotating speed. The rotating speed range that identifies each type of equipment will always be a subject of debate, but a reasonably accepted definition based on the common designs available at the time the first edition of API 618 was issued, as low-speed, is provided in Table 1.

Table 1. Low and Moderate Compressor Rotating Speeds.

	Low-Speed	Moderate-Speed
RPM	200 – 600	600 - 1200

The rotating speed and the compressor's categorization as either a low- or moderate-speed design does not determine or define the piston speed of the unit. Piston speeds are dependent on the rotating speed and the length of the piston stroke length. Because of this, application limits on rotating and piston speed must be independently reviewed.

RELATIONSHIP BETWEEN ROTATING AND PISTON SPEED

Piston speed is an important reciprocating compressor design parameter because it is a significant factor in determining the life of compressor wear parts. The instantaneous piston speed varies from zero at the end of the stroke, to a maximum near the middle of the stroke, and back to zero. However, as a matter of practicality, "average" piston speed is used to establish limits, make comparisons between units, and determine wear rates of wear components. Any desired piston speed can be achieved at any rotating speed by selecting the appropriate piston stroke length. Piston speed, stroke length, and operating speed are related by the following formula:

$$Average\ Piston\ Speed\ (ft/\min) = \frac{Stroke\ (inch) \times rpm \times 2}{12\ (inch/ft)}$$

$$or \qquad (1)$$

$$Average\ Piston\ Speed\ (m/s) = \frac{Stroke\ (mm) \times rpm \times 2}{60(s/\min) \times 1000(mm/m)}$$

Depending on the stroke length of a compressor it can be applied at various synchronous speeds to achieve desired piston speed limits. Figure 1 illustrates how both low- and moderate-speed units can achieve the same piston speed at different rotating speeds.

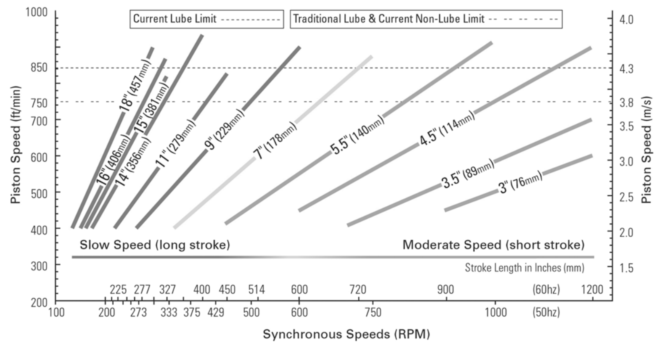

Figure 1. Piston Speed Versus Driver Speed.

The exact rotating speed that separates low- from moderate-speed compressors will always be a matter of discussion. The 600 rpm demarcation stated in this paper is based on the highest rotating speed that was commonly accepted as being "low-speed" when the first edition of API 618 was published in April 1964. Some users may select a lower speed than this based on their experience at the time they created their internal specifications.

HISTORICAL PERSPECTIVE OF COMPRESSOR DESIGNS

Low-Speed Designs

API 618, First Edition, published in April 1964 was based on compressor operating experience from the 1950s and early 1960s. These compressors had evolved from designs that operated at approximately 100 rpm in the early 1900s. The units were driven by steam or internal combustion cylinders mounted on the same crankshaft as the compressor cylinders or by an electric motor. Many of the electric drivers used support bearings that were common to the compressor system. Improvements in design, materials, and manufacturing technology resulted in operating at increased speeds and reliability throughout the first half of the 1900s. The increase in operating speeds was driven by the fact that higher speed units were smaller and therefore less costly.

Most reciprocating compressors installed in refineries during the 1950s and 1960s were either integral engine/compressors, electric motor, or steam turbine driven. The electric motor could be part of a common compressor/motor shaft or a separate motor driving a compressor frame. Steam turbines primarily drove compressors through a separate gearbox. Many of the same components were used for both the integral engines and compressor frames. This was done for both practical and economic reasons. The compressor manufacturers had a large variety of existing designs for cylinders, running gear, distance pieces, and other components that could be used as part of an integral engine unit or to create separate compressor frames that were driven by electric motors or steam turbines. Doing anything else would increase costs, add risk associated with new designs, and make supplying parts more difficult.

The speeds at which these designs could operate were limited by a number of factors. Piston speeds needed to be limited to about 750 ft/min (3.8 m/s) in order to achieve the desired three years of uninterrupted operation between required maintenance. This limit was arrived at empirically through operating experience. Nonlubricated compressors were typically applied with piston speed limits at or below 700 ft/min (3.6 m/s). Three years of operation between maintenance was not realistic for a nonlubricated compressor at that time.

Design and material technology available at that time determined the maximum rotating speeds at which engines and compressors could operate and achieve the desired operating time between required maintenance. Integral engines operating at up to 500 rpm were available, but those operating over approximately 300 rpm did not provide the uninterrupted service life desired. Because of this, operating at about 300 rpm became the standard at that time.

Compressor designs were based on piston stroke lengths that enabled the unit to operate within both piston and rotating speed limits. The most common designs in North America were based on 15 and 14 inch (381 and 356 mm) strokes that operated at 300 and 327 rpm, respectively. The drive speed and stroke length of these compressors resulted in piston speeds of about 750 ft/min (3.8 m/s). Other designs were built for various synchronous drive speeds. These included 16 inch (406 mm) units running at about 277 rpm, 11 inch (279 mm) stroke at 400 rpm, 9 inch (229 mm) stroke at 514 rpm, and 7 inch (178 mm) stroke at 600 rpm. Integral engine compressors that operated at 400 and 500 rpm utilized the 11 and 9 inch (279 and 229 mm) stroke components. The 7 inch (178 mm) and many of the 9 and 11 inch (279 and 229 mm) stroke units were driven primarily by electric motors. Older designs operating at lower speeds were available but were not as economic as the higher speed unit and had disappeared by this time.

The lack of any quantified piston and rotating speed limits in the First Edition of API 618 issued in April 1964 resulted in many companies incorporating the limits into their company specifications. Companies that sold refinery process designs also incorporated them into their equipment specifications that they required the customer's equipment meet in order to qualify for the process guarantees. Many of the speeds in these specifications have remained unchanged since they were originally issued. The maximum rotating speed of 400 rpm, or in a few cases 600 rpm, seen in many current specifications can be related directly to compressor designs that were common in the 1950s and 1960s.

Moderate-Speed Designs

Whereas the low-speed unit came from an integral engine heritage, moderate-speed units descend from the high-speed compressors designed to be driven by gas engines operating at 1000 rpm and higher. These gas engines operated at significantly higher speeds than integral units. This enabled size and therefore cost of gas engine compressor packages for a specified power rating to be reduced. These "high-speed" engines were not as reliable and did not have the ability to provide long periods of uninterrupted service compared to their integral alternatives. The market for which they were originally designed was oil and gas production, where the unit's reliability and length of uninterrupted service was not as critical as it was in a refinery application, but the reduced cost was a significant advantage to the user. The large number of compressor packages purchased by the oil and gas production companies provided the economic incentive to build and continuously improve both the reliability and length of uninterrupted service offered by high-speed compressor designs. This development started in the 1950s and continues today.

High-speed compressors differed from low-speed designs in both rotating and piston speeds. A conscious decision was made to reduce wear part life in order to achieve the greater oil and gas production output desired by the users. Piston speeds were increased to about 1000 ft/min to maximize the flow from the compressor during the one year period that the engine could be operated before major maintenance. The economic benefit of the increased flow exceeded the cost of the wear part replacement at the same time the engine needed to be serviced.

Early high-speed compressor designs were not known for their reliability. During the ensuing years much was learned about how to design frame and bearings systems for higher rotating speeds, how to correctly balance a unit to reduce vibration, cylinder lubrication, and most importantly valve designs for operation at high cyclic rates. The number of compressors built provided the economic incentive to create the technology required for reliable operation. Improved valve design technology and the development of nonmetallic materials for valve plates and compressor wear parts were key technical innovations that enabled reliable high-speed compressors to be built. By the 1990s a high-speed compressor's design could be expected to provide 98 percent availability without changing any wear parts during the 18 month period gas engines are now operated between major maintenance requirements. A common 1500 horsepower high-speed compressor design operates at 1400 rpm with a stroke length of 4.5 inches resulting in a piston speed of 1050 ft/min. While the majority of compressors used in the upstream industry in the early 1950s were low-speed designs, by the 1990s they were almost exclusively high-speed units.

Moderate-speed compressors suitable for critical service applications were developed by using design and materials technology advances that enabled the creation of reliable high-speed units. Piston speed limits equivalent to the proven low-speed designs were selected. Valve and compressor design methods developed for high-speed units were then used to create a unit that met the selected piston speed limits while operating reliably at higher rotating speeds. Many frame and running gear components created for high-speed units met the design requirements. These components would operate at one-half to two-thirds of what they were originally designed for. Many other items such as distance pieces had been designed in accordance with API 618 and could be used without modification. Other components such as pistons and cylinders for low molecular weight gas were specifically created for typical downstream applications. The result of these design efforts was a moderate-speed compressor with components that met API 618's requirements for uninterrupted operation periods at rotating speeds between 600 and 1200 rpm. As an example, the 4.5 inch (114 mm) stroke frame and running gear from a high speed compressor designed for 1500 rpm can be used in moderate-speed applications. When applied in critical service refinery application, it would be operated at rotating speeds of either 900 or 720 rpm. The resulting piston speeds will be either 675 or 540 ft/min (3.4 or 2.7 m/s), respectively.

Moderate-speed compressors are now starting to be used in critical downstream applications. Components specifically designed for the applications, in conjunction with current engineering and material technology, enable them to provide the reliability and uninterrupted service required for these applications. The moderate-speed design's inherently lower first and installed cost was one element that made companies look at using it in place of the proven low-speed compressors. The other reason was that insurance spares, such as crankshafts, connecting rods, and other running gear components, were typically available from the manufacturer's factory stock or had very short lead times. This was because the parts were also used in high-speed compressors that are built in relatively high volumes. Their availability eliminated the user's need to purchase insurance spares to eliminate lengthy outages if an infrequently needed but long delivery part was required.

ALLOWABLE OPERATING SPEEDS

Any discussion of allowable compressor speeds must be divided into two sections: piston and rotating speed. Piston speed affects the operating life of the piston rings, wear band, and packings. Rotating speed affects the valve cyclic (fatigue) life and the forces and moments created by the compressor.

Piston Speed

Piston speed limits are primarily defined by operating experience. When the first edition of API 618 was published in April of 1964, most refinery compressors were limited to a maximum piston speed of 750 ft/min (3.8 m/s) for lubricated and 700 ft/min (3.6 m/s) for nonlubricated services. Nonmetallic materials and new compressor valve designs have enabled piston speeds to be increased. Piston speed limits for the majority of lubricated applications are now commonly offered as high as 825 to 875 ft/min (4.2 to 4.4 m/s). Nonlubricated compressors are typically applied up to a piston speed limit of 750 ft/min (3.8 m/s). Though three years of uninterrupted operation is realistic for a lubricated compressor, it should not be assumed for nonlubricated applications. API 618 recognizes this, and it is one of the reasons that the statement "It is recognized that this is a system design criteria," is appended to the requirement for three years of uninterrupted operation.

Many factors such as process variables, operating temperature and pressure, and material selections can greatly affect component operating life. No definitive equation exists to calculate the expected life of wear components. A first order approximation of the main factors that affect wear rate can be expressed as:

$$Wear\ depth = (k)(P)(V)(t) \tag{2}$$

where:
k = Empirical wear factor
P = Pressure loading per unit area of the wear component on the counterface (the surface against which the wear component is moving)
V = Sliding velocity (piston speed)
t = Running time

This relationship assumes that the wear material is applied within its temperature and other application limits, and that the chemical and mechanical characteristic of the counterface surface is correct.

The relationship indicates that the wear rate of the material is a function of piston speed and is independent of rotating speed or stroke length other than how they contribute to defining the piston speed. Within a reasonable variation around the point at which the empirical wear factor was determined, wear rate is linearly related to piston speed and the pressure loading of the wear component. Low- and moderate-speed compressors operating at the same piston speeds and wear band loads can be expected to exhibit similar wear rates. Decreasing the piston speed and pressure loading on the components can extend wear component life.

The operating life of compressor wear components can be reduced by factors other than piston speed and the pressure loading of the components. Items that must be avoided if the desired service life is to be obtained include:

- Lack of particulate filtration or liquid separation from the process gas stream.
- Lack of or incorrect cylinder lubrication.
- Incorrect nonmetallic material selection for nonlubricated applications. (This is primarily caused by improper process gas or dew point information.)
- Operating the compressor outside of the range for which it was designed. (The normal problem is operating at a point or gas composition that results in discharge temperatures that exceed the application limits of the nonmetallic materials.)

These items affect low- and moderate-speed designs equally and reduce wear component life in a significant number of installations.

Rotating Speed

The rotating speeds at which a compressor can operate and meet the reliability objectives of API 618 have increased appreciably since the first edition of the specification came out in 1964. Compressors can now operate at speeds of up to 1200 rpm for three years before maintenance is required. The technological improvements that enable this include:

- Compressor design techniques created during the development of reliable high-speed reciprocating compressor and engine designs.
- Nonmetallic valve plate materials.
- Improved valve designs.
- Improved valve dynamic calculation capability.
- The availability of software to predict stress levels, thermal growth, vibration modes, and cyclic stress.

Valve design technology together with the advent of nonmetallic materials is the primary reason compressor rotating speeds can be increased while still achieving the uninterrupted operating goal set forth in API 618. Nonmetallic has replaced metallic valve plate materials in most low- and moderate-speed compressor valves. This occurred because the nonmetallic materials enabled valves to be operated reliably at higher lifts and therefore efficiency. The cyclic fatigue life of valve plate material is a major factor in determining the operating life of a valve. The stress created by the impact velocity of the valve plate on the seat normally is the most important limiting factor. Metallic valve plates could theoretically operate for an infinite time at impact velocities of 3 m/s (591 ft/min) or lower. Above that, they were limited by the cyclic fatigue life of the material. Nonmetallic materials had finite cyclic fatigue life for all stress levels but could operate at higher stress levels for a period. The evolution of technology now enables moderate-speed compressor valves with nonmetallic plates to operate for three years at impact velocities equivalent to those used in low-speed designs, typically 7 m/s (1378 ft/min). Nonmetallic valve plates also provide better sealing and more tolerance to dirt and liquid in the process gas flow.

The life expectancy of a moderate-speed compressor valve can be imputed from the operating history of high-speed compressors. The moderate-speed compressor operating history that is provided later in this paper also verifies it. Properly applied valves in high-speed compressors operate reliably (98 percent+) at impact velocities of 10 m/s (1969 ft/min) for a minimum of 18 months and longer. If the compressor operated at two-thirds its rated speed, the valve would take 2¼ years to open and close the same number of times. Because of this, the life of the valve would be expected to be a minimum of 2¼ years. All valve designs are a tradeoff of flow efficiency versus life. In oil and gas production operations customers want maximum flow and high valve reliability during the 18 month time between maintenance of the gas engine driver. Valves are rebuilt when engine maintenance is done. Because of this, valves are typically applied in high-speed compressors at higher valve lifts and impact velocities than in refineries where the requirement is for high reliability and three years' life. Moderate-speed units achieve the three year uninterrupted run time by decreasing the valve impact velocity to 7 m/s (1378 ft/min). This reduces the impact stress and therefore increases the cyclic life of the nonmetallic plate material. Since valve plate impact energy is related to the impact velocity squared, impact energy is cut in half. This in conjunction with the reduced cyclic rate of operation enables valves to operate for the three year uninterrupted period specified in API 618.

If both low- and moderate-speed valves are correctly designed, there are a number of other design requirements that have a larger impact on valve life than the cyclic life of the valve plate material. These include:

- Selecting the most reliable type of valve design for the application.
- Eliminating particulate matter and liquid carryover in the process gas through inlet filters or separation.
- Defining all potential operating points and associated gas compositions.

• Conducting a dynamic analysis to define the correct valve components and ensure the valve plate impact velocities are within conservative limits that enable the three year operating life criterion to be met.

• Selecting the correct valve sealing element (i.e., plate, ring, or poppet) material for the application.

One or more of the above factors are typically the reasons for a valve not achieving its expected operating life. Inlet filtration and separation increases the initial cost of an installation but can have dramatic effects on the reliability of the compressor installation. New compressor performance calculation software that includes basic valve dynamic analysis is now available. Using it to verify the suitability of the valve design over a wide range of operating conditions and plot maps showing any areas that should be avoided helps eliminate problems.

Forces and Moments

The forces and moments created by reciprocating compressors affect both foundation design and mechanical vibration. Moderate-speed designs exploit both the geometric layout of the cylinders and balance methods to enable them to operate at higher rotating speeds. The inertial force of the reciprocating components generates most compressor forces. The gas loading on the piston normally acts in the opposite direction of the inertial force and acts to reduce the forces. The inertial force generated by one throw of a compressor can be calculated with the following formula:

$$F_{Inertial} = (m)\left(\omega^2\right)(R)\left(Cos\ \theta + R / L\ Cos\ 2\theta\right) \quad (3)$$

where:
m = Mass of reciprocating components
ω = (Rpm) 2π/60
R = Crank throw (stroke/2)
L = Connecting rod center distance
θ = Crank angle in degrees

Low-speed units are built in horizontal, vertical, and "Y-style" configurations. Horizontal configurations included provisions for balancing, but not to the level of balance provided in moderate-speed compressors. Additionally, opposing throws of low-speed units typically were not component balanced during assembly to determine and correct for variations in the actual weights of component castings versus that assumed in the engineering design. The forces created by the compressor were accounted for by mounting the unit on a concrete foundation with sufficient mass to absorb them. A 1 MW low-speed horizontally opposed compressor is typically balanced to about 50 lb (23 kg). Units can easily be balanced to about half of this amount and less if required.

Moderate-speed compressor designs always utilize horizontally opposed cylinders. The inertial forces of the opposing cylinders act in opposite directions and, to the extent that the reciprocating masses of the opposing throws are equal, cancel each other out (Figure 2).

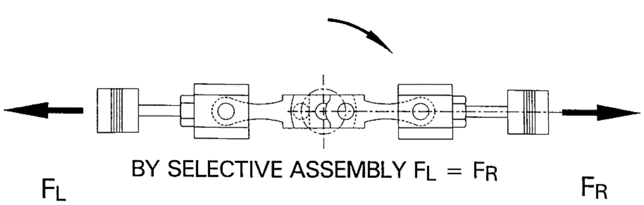

Figure 2. Equal Masses Result in Equal but Opposite Inertial Forces.

Moderate-speed compressors component balance the reciprocating mass of opposing throws to within about 2 lb. Balance is achieved by selecting crossheads and piston rod nuts of the correct weight to accomplish this (Figure 3). Balance weights that bolt on are not used because of the risk of loosening and detachment.

Figure 3. Balance Nuts and Crossheads.

The inertial force equation can be used to compare the forces created by low- and moderate-speed compressors in similar applications. Table 2 compares the forces created by a 1 MW compressor in a typical refinery service. The stroke length of all three alternatives is calculated to provide a piston speed of 750 ft/min (3.8 m/s) so that force created at each speed is directly comparable. The actual stroke of the unit would depend on the compressor manufacturer and would be selected to provide a piston speed equal to or less than the limit desired for the project.

Table 2. Comparison of Forces Generated by Low and Moderate-Speed Compressors in Like Services.

Compressor Type	Low-Speed	Moderate-Speed	Moderate-Speed	Moderate-Speed
RPM	300	720	900	1200
Piston Speed Ft/Min (m/s)	750 (3.8)	750 (3.8)	750 (3.8)	750 (3.8)
Stroke Inch (mm)	15 (381)	6.25 (159)	5.0 (127)	3.75 (95)
Unbalance Lbs. (Kg)	22 (10)	2.5 (1.15)	2.5 (1.15)	2.5 (1.15)
%(M ω^2 R)	100%	27%	34%	45%

Table 2 shows that the forces created by moderate-speed units are actually less than for a comparable low-speed unit. Though the force increases with the square of the rotating speed, it decreases linearly with the stroke length reduction required for maintaining the specified piston speed. It also reduces linearly with unbalance mass.

The forces created by a reciprocating compressor are handled by transmitting them to a sufficiently large foundation through anchor bolts to prevent any relative motion between the compressor and the foundation. The larger the force, the larger the foundation needs to be and the more energy is transmitted through the anchor bolts. Reducing the forces and moments created by a compressor helps eliminate long-term foundation problems that occur as the concrete degrades over time due to age and the effects of a plant environment.

MODERATE SPEED COMPRESSOR DESIGNS

Moderate-speed compressors used in critical refinery service are designed and built in accordance with API 618. Component designs are equally robust in both low- and moderate-speed units and equally compliant to API 618. The only exceptions to this are a few design requirements of low-speed designs such as cylinder water jackets and liners that are not required by moderate-speed compressors. The reasons for these deviations will be explained later in the paper.

Moderate-speed compressors are not just high-speed compressors slowed down to provide piston speeds acceptable to a refinery application. Not all component designs used in high-speed designs are appropriate for critical service applications. The following component designs will be part of a moderate-speed design but are not always found in a high-speed unit:

- Frame and running gear
  - Forged steel crankshaft and connecting rods
  - Precision fit, replaceable bearings on the crankshaft and big end of the connecting rod
  - Conservative bearing loads and rod loads
- Distance pieces and packing
  - A variety of distance pieces designed in accordance with API 618 to select from
  - Options for purged and water-cooled pressure packing with a variety of material selections for both lubricated and nonlubricated applications
  - Intermediate and wiper packing with designs and options such as purge capability and nonmetallic materials suitable for the intended application
- Cylinder, piston, and valves
  - Piston designs that include nonpressure loaded wear bands designed in accordance with API 618 with material options for both lubricated and nonlubricated applications
  - Piston designs with the additional piston rings required for low molecular weight gas
  - Compressor valve design options appropriate to the intended application

Each manufacturer will have a unique list of comments and exceptions to API 618. Most of the items are for deviations when the supplier has a different design with proven operational reliability. Moderate-speed compressor designs will not comply with the requirement for cylinder water jackets and liners. This is because these designs are required for the majority of low-speed design applications but are not needed by the moderate-speed alternative.

COMPRESSOR CYLINDERS

Low- and moderate-speed compressor designs are equally compliant with the majority of the API 618's (1995) cylinder design requirements. The two areas where the low- and moderate-speed cylinder designs deviate are the inclusion of water jackets and cylinder liners.

Water Jackets (Cylinder Cooling)

API 618 (1995) specifies that cylinders operating above stated discharge temperatures must have water jackets that are either static filled, have an atmospheric or pressurized thermosyphon, or have forced liquid coolant depending on the discharge temperature. Additionally, any cylinder operating fully unloaded for extended periods shall have a water jacket with a forced liquid coolant.

There are two reasons for this requirement. The first is that water jackets were needed when API 618 was first written for the dimensional stability of the cylinder bore of long-stroke, low-speed compressors. The length of the cylinder and the casting quality available at that time resulted in the cylinder bore deforming as cylinder temperature increased. Water jackets were required to ensure the heat effect was evenly spread around the cylinder bore to increase dimensional stability. This was critical in view of the use of metallic rings, and the fact that the bore of the cylinder was soft (approximately 20 to 25 Rc). If the cylinder bore distorted dimensionally, the hard piston rings in use at the time the first edition of API 618 was issued would scrape the cylinder bore and damage it.

The second reason that water jackets were required was to carry away the heat generated by cylinders operating fully unloaded (both head end [HE] and crank end [CE]) for extended lengths of time.

API 618 (1995) uses the term cylinder cooling for the water jacket. Inlet water temperature is specified to be a minimum of 10°F (6°C) above the gas inlet temperature to eliminate gas condensation, primarily during shutdowns. The static and thermosyphon designs specified in API 618 will maintain bore stability but have no real mechanism to transport heat away from the cylinder. The inclusion of these designs reinforces the fact that water jackets are used to maintain bore stability rather than to transport heat away from the cylinder. The forced liquid cooled cylinder design specified in API 618 does transport heat away from the cylinder in the water. In operation, the majority of the heat is picked up in the discharge passage of the cylinder where the maximum gas to liquid temperature differential exists. The amount of heat transferred from the surface of the bore is small. If the cylinder is viewed as a single pass shell-and-tube heat exchanger with a 15 inch (381 mm) long "tube" of 1 to 1.4 inch (25 to 35 mm) cast-iron having a mean gas to water differential temperature of 108°F (60°C), it is apparent that little heat can be transferred from the cylinder bore. A forced liquid cooled cylinder does provide enough heat transfer to reduce heating problems in extended periods of unloaded operation.

Moderate-speed compressor designs utilize cylinders with shorter stroke lengths. Current casting technology, the uses of nonmetallic wear materials, and the short bore length enable dimensional bore stability to be maintained without the need for water jackets. The reliability of these designs has been proven over the last 30 years in oil and gas production and gas transportation applications. Whereas previously water-jacketed cylinders were standard in these applications few if any are utilized now.

Cylinders that do not require water jackets have the benefit of eliminating the need for cylinder cooling water systems, the maintenance of those systems, and the probability of bore distortion if cooling water flow is lost or the water jacket becomes plugged. Nonwater jacketed cylinders do have the problem of not being suitable for operating totally unloaded for extended periods of time.

Cylinder Liners

API 618 (1995) specifies that cylinder liners be provided. Low-speed compressor designs are typically supplied with liners, while moderate-speed designs may or may not.

Originally, long-stroke cylinders had problems with bore dimensional stability if the casting did not grow uniformly as the temperature increased. This in conjunction with the relatively hard rings used at the time API 618 was first published and the low bore hardness could result in damage to the bore's surface. Cylinder liners were included to help get a cylinder with bore damage back in service quickly and as cost effectively as possible. A cylinder could be taken to a repair facility for liner replacement, and the unit would be back in operation within five to seven days.

API 618 (1995) requires that piston rods be surface hardened in the packing area to increase their service life and help protect against damage. The option of hardening the surface of the cylinder bore to obtain these same benefits was a technology that did not exist at the time low-speed units were designed. Piston rods could be heat treated to increase their surface hardness while retaining suitable base material physical characteristics. A cylinder could not be heat treated to harden the bore surface because the base material at that time would not provide suitable pressure vessel characteristics.

The development of technologies such as ion-nitriding enables manufacturers to harden the surface of cylinder bores to 58+ Rc without changing the base material characteristics. The hard surface provides the increase in life and resistance to damage that a hardened piston rod provides.

Moderate-speed cylinders can be obtained with liners, without liners, and with hardened unlined bores depending on the manufacturer. The short stroke of moderate-speed cylinder designs enables bore stability to be better maintained than on a long-stroke design. The use of nonmetallic rings and the improvement in casting technology eliminates problems with thermal growth causing the piston to scrape the cylinder wall.

The inclusion of a liner in a moderate-speed cylinder degrades the performance of the compressor because of the increased fixed clearance added in the valve area. For this reason, most manufacturers do not include a liner unless the customer requests it.

A cylinder, either low or moderate speed, is still susceptible to bore damage due to particulates in the gas, lubrication problems, or operating the unit with worn wear bands and/or piston rings. A hardened cylinder bore helps protect against bore damage and increases the life and reliability of the cylinder.

If damage does occur to the cylinder bore, a liner can be replaced or the entire cylinder body can be replaced. Moderate-speed compressor manufacturers normally make the cylinder from castings with relatively high production volumes. Thus, these castings are typically in stock. The cost of replacing a new moderate-speed cylinder body is about equal to the cost of taking a slow-speed cylinder to a machine shop, removing an interference-fit liner, and replacing it with a new liner. This is possible because a moderate-speed cylinder is about half the size of the equivalent slow-speed cylinder and is produced with stock castings that are ordered in quantity. It takes about one week to replace a liner, the same time required to obtain a new moderate-speed cylinder from a manufacturer that has stock castings.

Low- and moderate-speed cylinders can also be damaged due to liquid slugs carried into the compressor or by ingesting a foreign object such as a valve part. A cylinder liner does not help place the unit back in service in either of these cases because other areas of the cylinder are often damaged. A new or rebuilt cylinder is typically required to correct these problems. Being able to source a new moderate-speed cylinder in a week eliminates much of the downtime normally incurred in these situations.

OPERATING HISTORY

The majority of reciprocating compressors operating in refineries and industrial gas applications are low-speed designs. Operating experience gained from these units has provided an understanding of the design and application limits needed to achieve three years of uninterrupted service. Much of this knowledge is directly applicable to moderate-speed compressors in similar applications. Similarly, the operating histories of high-speed compressor designs are useful in verifying the reliability and application limits applied in critical services. The longevity and bore stability of short-stroke, nonwater jacketed cylinder designs has been verified by over 30 years of experience.

A number of high-speed units found their way into refinery service in the past. Some of these were operated at lower rotating and therefore piston speeds. Many were applied at the same speeds as in oil and gas production applications. The majority of these utilized pistons were originally designed for natural gas applications, even in low molecular weight gas service. If properly applied, they have operated reliably on refrigeration, acid gas, vapor recover, and other services. When misapplied, especially on light molecular gas services, they do not operate reliably. Because of their low initial cost, there was a large incentive to attempt to use these for purposes for which they were never designed. Their poor reliability has hindered the introduction of moderate-speed units into the downstream industry.

A number of API 618 moderate-speed units have been installed in refinery applications and have been operating for a number of years. These units are configured with piston, distance pieces, and other components correctly designed for the application. They operate at piston speeds needed for the desired long uninterrupted service but at higher rotating speeds typical of moderate-speed designs. The operating history of correctly configured moderate-speed units applied in refinery service verifies their ability to meet the reliability and operating time between maintenance required by API 618 (1995). The following is the operating history of four units, built by two different manufacturers.

Example 1

- Service: Recycle gas compressor for a hydrogen desulfurization unit located in a German refinery
- Compressor configuration:
  - Two-throw, single-stage
  - Stroke: 4.25 inch (108 mm)
  - API 618 Type C distance pieces
  - Purge provisions were provided for the distance piece compartments, pressure packing, and intermediate packing
  - Water-cooled pressure packing
  - Piston designs include nonpressure loaded wear bands sized for 5 psi loading and sufficient piston rings for low molecular gas applications.
  - Nonwater jacketed nodular iron cylinders with bores hardened to approximately Rc 61
  - Distribution block type cylinder lubrication system
  - Capacity control system
  - Online performance monitoring system
- Driver: Variable speed induction motor (Note: capacity control is normally accomplished using the capacity controllers on the suction valves of the compressor
- Rpm: 742
- Piston speed: 526 ft/min (2.67 m/s)
- Lubricated/Nonlubricated: Lubricated
- Power: 267 bhp (199 kW)
- Gas composition: Table 3 (Gas composition varies over time)

Table 3. Example 1 Gas Composition.

C1	13.3
C2	4.7
C3	3.5
I-C4	3.4
N-C4	0.0
I-C5	1.4
N-C5	0.0
C6+	0.0
H2S	0.2
H20	Saturated
H2	73.5

- Flow (min/normal/max): 2500/7000/9200 Nm^3/hr
- Suction pressure: 304 psia (21 bara)
- Discharge pressure: 507 psia (36 bara)

- Discharge temperature: 193°F (101°C)
- Startup date: December 10, 1999
- Maintenance:
  - 8000 hours: Replaced discharge valves with damaged plates. The root cause of the problem was that the discharge valves were incorrectly specified. The original valves were supplied with metal rather than nonmetallic valve plates. The wide flow variance provided by the capacity control system resulted in the discharge valve impact velocity during low flow operation exceeding the operating capability of the metal plates in the valve. The valve design was changed to incorporate nonmetallic valve plates. Suitable valve springs were selected for the range over which the compressor was operated. The new valve design resulted in some initial problems because the stack clearance of the valve and valve cap were incorrect. This was identified and solved quickly by refinery personnel.
  - 28,800 hours: All wear components were changed as part of a preventive maintenance program during a plant turnaround. The pressure packings and piston rod were worn to the point that they required replacement or rebuilding. All other components were in serviceable condition but were replaced as part of the preventive maintenance program. No components other than the discharge valve mentioned above were changed before this point.
  - 32,484 hours: The unit has operated for 32,484 hours as of January 15, 2004.

Example 2

- Service: Makeup gas compressor for a hydrocracker in a refinery located in Oklahoma
- Compressor configuration:
  - Four-throw, three-stage with two cylinders on first stage service
  - Stroke: 4.5 inch (114 mm)
  - API 618 Type C distance pieces
  - Purge provisions were provided for the distance piece compartments, pressure packing, and intermediate packing.
  - Water-cooled pressure packing
  - Piston designs include nonpressure loaded wear bands sized for 5 psi loading and sufficient piston rings for low molecular gas applications.
  - Nonwater jacketed nodular iron cylinders with bores hardened to approximately Rc 61
  - Distribution block type cylinder lubrication system
  - Variable volume clearance pockets on first stage cylinders
  - Ring-type valves
- Driver: Single speed induction motor
- Rpm: 885
- Piston speed: 664 ft/min (3.37 m/s)
- Lubricated/Nonlubricated: Lubricated
- Power: 1050 bhp (782 kW)
- Gas composition: Table 4 (Gas composition varies over time, H_2 varies from about 80 to 90 percent)
- Flow (design): 5.5 mmscfd
- Suction pressure: 95 psia (6.55 bara)
- Discharge pressure: 1735 psia (120 bara)
- Discharge temperature: 225°F (107°C)
- Startup date: May 2002
- Maintenance:
  - 200 hours (approximately): Crankshaft detuner added and cylinder lubricator pump replaced. The root cause of the problem was an error in the torsional study. Lack of sufficient separation at higher order frequencies resulted in a fatigue failure of the cylinder lubricator drive shaft. Shaft end torsional readings were taken before and after the detuner was added to confirm both the problem and that the correct separation levels were obtained. The problem was resolved and the unit was operating within three days of the initial occurrence of the problem.
  - 8000 hours (approximately): A scheduled preventive maintenance check was performed. This check was part of an established refinery practice based on operating experience obtained from other units in similar service. Cylinder lubrication was checked to ensure it was sufficient. The valves and forth stage cylinder bore were visually inspected. All components were in good condition and were put back in service.
  - 13,000 hours (approximately): As of February 2004, the unit has operated for approximately 13,000 hours without maintenance on the compressor. Maintenance has been required on the main motor starter and scrubber level switches. One significant difference has been noticed between this unit and other low-speed compressors in similar service. All other units in similar service were applied as three-stage units with discharge temperatures of slightly over 275°F (135°C). The shell and tube intercoolers on these units need to be rodded out every four months to eliminate scale buildup caused by the cooling water quality. The moderate-speed compressor was applied as a four-stage unit to reduce the operating discharge temperature to approximately 225°F (107°C). This was feasible because the installed cost of the four-stage moderate-speed unit was less than a comparable three-stage low-speed unit. The original intention was to prolong the life of the Teflon® based wear materials by running them cooler. The reduction in operating temperatures has eliminated the scale problems with the shell and tube heat exchangers, which have not required any maintenance since startup.

Table 4. Example 2 Gas Composition.

C1	4.0
C2	4.0
C3	3.0
I-C4	0.0
N-C4	3.0
I-C5	0.0
N-C5	1.0
C6+	0
H2S	0.0
H20	Saturated
H2	85.0

Example 3

- Service: Hydrogen compressor for hydrogen production service located in a Gulf Coast refinery near Houston, Texas
- Compressor configuration:
  - Six-throw, two-stage with three cylinders on each stage
  - Stroke: 5.9 inch (150 mm)
  - API 618 Type B distance pieces
  - Purge provisions on distance piece and pressure packing
  - Water-cooled pressure packing
  - Piston designs include nonpressure loaded wear bands sized for 5 psi loading and sufficient piston rings for low molecular gas applications.
  - Water-jacketed cylinders with liners. Bores are not hardened.
  - Plate valves

- Driver: Single speed induction motor
- Rpm: 715
- Piston speed: 704 ft/min (3.56 m/s)
- Lubricated/Nonlubricated: Nonlubricated
- Power: 3375 bhp (2523 kW)
- Gas composition: Table 5

Table 5. Example 3 Gas Composition.

C1	0.0
C2	0.0
C3	0.0
I-C4	0.0
N-C4	0.0
I-C5	0.0
N-C5	0.0
C6+	0.0
H2S	0.0
H20	0.0
H2	100.0

- Flow (design): 51.6 mmscfd
- Suction pressure: 314 psia (21.7 bara)
- Discharge pressure: 875 psia (60.3 bara)
- Discharge temperature: 200°F (93°C)
- Startup date: November 1, 2001 (full time operation)
- Maintenance:
  - 10,500 hours: Preventive maintenance was conducted on the compressor during a planned two-week shutdown. The following parts were replaced:
    - Piston rings and wear bands
    - Pressure packings
    - Wiper
    - Valves

None of the parts showed any signs of undue wear.

  - 19,600 hours: The unit has been in continuous operation since the plant shutdown. As of March 15, 2004, the unit has operated continuously for approximately 9100 hours.

Example 4

- Service: Nitrogen compressor for nitrogen production service located in a Washington refinery
- Compressor configuration:
  - One-throw, single-stage
  - Stroke: 3.0 inch (76 mm)
  - API 618 Type C distance pieces
  - Water-cooled pressure packing (water-cooling provision not utilized in service)
  - Piston designs include nonpressure loaded wear bands sized for 5 psi loading and sufficient piston rings for low molecular gas applications
  - Nonwater jacketed nodular iron cylinders with bores hardened to approximately Rc 61
  - Fixed volume clearance pockets and finger type unloaders
  - Plate valves
- Driver: Single speed induction motor
- Rpm: 890
- Piston speed: 445 ft/min (2.25 m/s)
- Lubricated/Nonlubricated: Nonlubricated
- Power: 25 bhp (19 kW)
- Gas composition: Table 6

Table 6. Example 4 Gas Composition.

C1	0.0
C2	0.0
C3	0.0
I-C4	0.0
N-C4	0.0
I-C5	0.0
N-C5	0.0
C6+	0.0
H2S	0.0
H20	0.0
N2	100.0

- Flow (design): 0.30 mmscfd
- Suction pressure: 95 psia (7.2 bara)
- Discharge pressure (maximum): 315 psia (21.7 bara)
- Discharge temperature (maximum): 330°F (165°C)
- Startup date: December 1997
- Maintenance:
  - 8000 hours: The pressure packing was replaced as part of a preventive maintenance program. The original pressure packing was in serviceable condition when removed.
  - 23,150 hours: Nitrogen plant contract ends and unit is shut down. Other than the pressure packing no other parts or maintenance were performed.

The industrial company that owns and operates this unit has six other compressors similar to this. The initial units were supplied with single compartment distance pieces and required the pressure packings to be changed at approximately nine months due to oil ingression from the frame. Later units were supplied with API 618 Type C distance pieces. These units are operated on an 18 month preventive maintenance schedule. The operating history of these units indicate a 99 percent+ online availability during this 18 month period.

CONCLUSION

Reciprocating compressors for critical downstream applications must provide reliable, long operating periods between required maintenance intervals. API 618 (1995) and other compressor specifications were written to capture and communicate good design practices to the industry.

The first edition of API 618 was published in 1964. The materials, technology, and experience available at that time dictated that reciprocating compressors for refinery service be designed for drivers operating between approximately 200 to 500 rpm. The most common drive speeds were near 300 rpm. These rotating speeds and piston speeds of about 750 ft/min (3.8 m/s) for lubricated service resulted in the long-stroke compressor designs that have remained unchanged for years. The fact that API 618 was originally written around this type of machinery has helped to perpetuate it.

Gas engines operating at 1000 rpm began to be produced in the 1950s. These high-speed engines were directly coupled to high-speed compressors to provide a lower cost alternative to the low-speed integral compressors that were being used in oil and gas production applications. The large size of the oil and gas production market provided the economics needed to drive the development of technology and materials that enabled reliable high-speed compressor designs. Moderate-speed compressor designs evolved from this technology.

API 618 and other reciprocating compressor specifications have been written and updated over time. As they independently evolved, inconsistencies and inappropriate application of some sections of the specification occurred. Though the majority of design principles apply equally to both low- and moderate-speed compressor designs, some design requirements apply only to one or the other. The primary examples of this are the requirement for water-jacketed cylinders with liners. Both of these items were required for long-stroke cylinders built with the technology available when API 618 was first issued in 1964 but are not required for short stroke cylinder designs manufactured today.

API 618's lack of definition of moderate- and low-speed compressors, together with no quantitative limits on either piston or rotating speed, has led to confusion and misunderstanding about what constitutes a moderate-speed compressor and its acceptability for critical service applications. A definition of low- and moderate-speed compressors and the piston speed limits currently used for refinery applications have been provided in this paper.

The operating history of moderate-speed units in refinery service presented in this paper illustrates that they operate as reliably as the low-speed designs. The primary cause of problems with both low- and moderate-speed units is incorrect application followed by design errors in components that are uniquely designed for the application. Liquid and dirt in the process gas stream in excess of what was expected and designed for are examples of this. Other examples are inappropriate valve designs for the actual operating points, torsional and pulsation design errors, and errors in custom component designs.

REFERENCES

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