A Discussion of the Various Loads Used to Rate Reciprocating Compressors

By

Martin Hinchliff
Principal Engineer
Dresser-Rand

Shelley Greenfield
Vice President Design Services
Beta Machinery Analysis

Wally Bratek
Principal Engineer
Beta Machinery Analysis

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Introduction

Reciprocating compressors are usually rated in terms of horsepower, speed, and rod load. Horsepower and speed are easily understood; however, the term “rod load” is interpreted differently by various users, analysts, OEMs, etc. This paper discusses the various definitions of rod load, including historical and current API-618 definitions. Examples are provided of what rod loads ratings applied prior to the current edition of API-618 meant when compared to the current definition. Also, in general most performance programs make assumptions of uniform pressure inside the cylinder, ideal valve motion, constant pressure at the cylinder flange and neglect gas inertia effects. This paper will discuss the effects of real world compression including pressure pulsation at the outside of the valve, non-uniform piston-face pressure distributions and torsional pulsation effects on rod load. Examples will be provided along with guidance on when these effects will be significant.

The user will find this helpful for understanding rod load ratings on existing machinery. It will help in interpreting real world PV card analysis compared to compressor performance data. It will provide guidance for when compressor performance assumptions regarding rod load may result in significant deviation from real rod load.
Basic Theory

A typical double-acting compressor cylinder is illustrated in Figure 1. The loads (forces) that are generally of concern include the piston rod loads, the connecting rod loads the crosshead pin loads, the crankpin loads, and the frame loads. As the crankshaft undergoes one revolution, all of these loads vary from minimum to maximum values. The loads are generated by both gas and inertia forces as discussed in the following paragraphs.

![Figure 1: Double Acting Compressor Cylinder](image)

Gas Loads

As the compressor piston moves to compress gas, the differential pressures acting on the piston and stationary components result in gas forces as illustrated in Figure 2. An ideal pressure versus time diagram for a typical double acting compressor cylinder is shown in Figure 3. The pressures acting on the piston faces (head end and crank end) result in forces on the piston rod. The force acting on the piston rod due to the cylinder pressures alone alternates from tension to compression during the course of each crankshaft revolution. It is straightforward to compute the net force on the piston rod due to pressure. A plot of this force versus crank angle for the ideal P-T diagram is shown in Figure 4. The forces due to pressure also act (equal and opposite) on the stationary components.
\[ F = P_{CE} A_{CE} - P_{HE} A_{HE} \]

\( P \) = Static & Dynamic Pressure
\( A \) = Area of Piston

**Figure 2: Gas Loads**

**Figure 3: Ideal P-T Diagram**
Figure 4: Rod Loads Based on Ideal P-T Diagram

The maximum compression force due to pressure occurs when the head end is at discharge pressure and the maximum tensile force due to pressure occurs when the crank end is at discharge pressure. Therefore the equation shown in Figure 2 is often evaluated at the extremes as follows:

\[
F_{\text{Tension}} = \left( P_{\text{Discharge}} \times A_{\text{CE}} \right) - \left( P_{\text{Suction}} \times A_{\text{HE}} \right) \tag{1}
\]

\[
F_{\text{Compression}} = \left( P_{\text{Discharge}} \times A_{\text{HE}} \right) - \left( P_{\text{Suction}} \times A_{\text{CE}} \right) \tag{2}
\]

Now consider a more realistic pressure versus time diagram as shown in Figure 5. “Line pressure” refers to the pressure at the line side of the pulsation bottle (suction or discharge). “Flange pressure” refers to the pressure at the cylinder flange. As shown, the in-cylinder discharge pressure exceeds the nominal discharge line pressure and the in-cylinder suction pressure is less than the nominal suction line pressure due to several effects:

1. Pressure drop due to valve and cylinder passage losses (typically 2-10%)
2. Pressure drop due to pulsation control devices (typically 1-2%)
3. Pulsation at cylinder valves (typically <7%)
4. Valve dynamics (inertia, sticktion, flutter, etc.)
API-618 specifies that the internal pressures must be computed, but does not define any procedure for the calculations. There are several methods for accounting for the non-ideal effects. One common method is to model the valve as an orifice and then the pressure drop though the valve (valve loss) is proportional to the square of the piston velocity (flow). This is illustrated in Figure 5. Theoretically, it would be more accurate to use the results of the valve dynamics analysis coupled with the digital pulsation simulation to model the instantaneous pressure at the valves. This is not practical to do until all of the piping and valve details are known. Even when all the information is available, the difference should be small provided the pulsations at the valve are below the API 618 criteria, therefore, there is typically minimal benefit to combining the valve and pulsation models.

Because of these effects, the forces due to differential pressures are higher on both the running gear and the stationary components than those calculated based on nominal line pressures. However, equations 1 and 2 are still applicable as long as the appropriate pressures (discharge pressure higher than nominal discharge pressure, suction pressure lower than nominal line pressure) are used. If the nominal pressures at the suction and discharge cylinder flanges are used for $P_{\text{suction}}$ and $P_{\text{discharge}}$, then these tension and compression forces represent the term “Flange Loads” as interpreted by some users. Equations 1 and 2 are easy to evaluate and for many years were the basis for rating “rod loads” of reciprocating compressors.

For the general non-ideal compressor cylinder, the maximum discharge pressure on the head-end will not necessarily occur at the same instant that the minimum suction pressure occurs on the crank-end and vice versa. Therefore, it is common to evaluate the gas forces versus crank angle at discrete steps (e.g. every 5 or 10 degrees). The history of these types of calculations is discussed below, but computing the instantaneous force due to differential gas pressures is easily accomplished with computer based software. If the actual in-cylinder pressures are used and the extremes are evaluated, these forces are then the “Gas Loads” referred to in the API specifications.

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**Figure 5: Non-Ideal P-T Diagram**
**Piston Rod Loads**

The basic slider crank mechanism is illustrated in Figure 6. The exact equation for the position of the crosshead with respect to the x-direction shown is

\[ x = r \cos \theta + l \sqrt{1 - \frac{r^2 \sin^2 \theta}{l^2}} \]  

\( (3) \)

![Figure 6: Slider Crank Geometry](image)

The piston (crosshead) motion is usually approximated using the first two harmonics of the Taylor series as follows:

\[ x = r \cos \theta + l \left(1 - \frac{r^2}{2l^2} \sin^2 \theta\right) \]  

\( (4) \)

The piston rod loads can be evaluated by considering the free body diagram in Figure 7. The forces acting on the piston rod are the gas forces due to differential pressures acting on head end and crank end piston areas plus the inertia forces due to the reciprocating mass. If the reference point is chosen as the crosshead end of the piston rod, then the reciprocating weight will include the piston rod and the piston assembly (piston, rings, rider bands, etc.). The reciprocating inertial force \((F=ma)\) can be computed using the following equation:

\[ F_I = m_{recip} r \omega^2 \left[ \cos(\omega t) + \frac{r}{l} \cos(2\omega t) \right] \]  

\( (5) \)

where:

- \( m_{recip} \) = mass of reciprocating components
- \( r \) = crank radius
ω = angular velocity, \( \dot{\theta} \)

\( l \) = connecting rod length

\[ F_{\text{Rod}} \quad \rightarrow \quad F_1 + F_G \]

**Figure 7: Piston Rod Load**

The combined piston rod load is the sum of the gas force and the inertial force. In accordance with API-618, this value is routinely calculated in the design stage and used along with the rod area at the minimum cross-section to compute tensile and compressive stresses in the piston rods. The stress in the piston rod is one factor to consider in the design, and in some cases it may be the limiting factor or the “weakest link in the chain.” However, this load is not the *rod load* to which API-618 refers.

**Crosshead Pin Loads**

The free body diagram for the system including the crosshead pin is shown in Figure 8. Here the mass of the crosshead assembly (crosshead, balance weights, crosshead shoes, etc.) must be considered, but the same equations apply. The combination of the gas loads and inertia loads evaluated at the crosshead pin *in the direction of piston motion* are the “combined rod loads” to which API-618 refers. This load does not consider side forces on the crosshead or the 1/3 of the connecting rod weight that is usually considered to be reciprocating. Thus, “rod load” by API definition is not really a rod load, but actually a pin load.

**Figure 8: Forces Acting on Crosshead Pin**
Crankpin Loads

If the loads and torques throughout the system are evaluated, then the rotating and reciprocating inertias as well as the side forces are included. Equations are applied for computing x and y components of crankpin and wrist pin loads, crank throw torques, main bearing loads, etc. The typical output of the computer program used to evaluate these loads is shown in Figure 9. All of these loads are typically considered in the design stage. Different OEMs evaluate the loads per their own experience. API guidelines are discussed in the following section.

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Figure 9: Design Calculation Results

History of “Rod Loads”

The 1st Edition of API-618 was published in 1964 (34 pages). It included no definition of what was meant by the term “Rod Load.” However, the data sheets did call for the compressor manufacturer to specify the “Max Allowable Rod Loading” and “Rated Rod Loading”. So the definition of what that meant was left up to the compressor OEM.

In the 1963 edition of the Ingersoll-Rand (IR) frame ratings guide, the “piston loads” were defined. This document stated that “piston load” is frequently referred to as “rod load” which is a misnomer as it implies that the piston rod is the only limit in the establishment of a compressor load rating. It defined the piston load as the nominal pressure at the cylinder flange times the area of the piston. These loads were easily calculated from simple equations presented later in this paper (equations 1 and 2). It went on to say that the actual rod load would include the effect of inertia and valve losses, but these effects were considered in the piston rod load ratings, i.e. the piston rod load ratings were necessarily conservative. This approach served the industry well, but perhaps resulted in “over-designed” machinery.

This was before the advent of electronic calculators and digital computers, so combined rod load was tedious to calculate. The practice at the time was to look at and report simply the nominal gas load only with no valve losses or inertia loads considered. On rare occasions if it was judged necessary due to a combination of
high gas loads, high inertia forces and high volumetric efficiency (which can cause the gas load and inertia load to be additive), a manual calculation of combined rod load (gas + inertia + valve losses) would be done. This would consist of drawing a PV card including valve losses, using a planometer and slide rule to determine area (horsepower) and gas pressures at discrete degrees of rotation increments. Then inertia forces were calculated at each point and added to determine the combined rod load at the crosshead pin, forces in the connecting rod and crankshaft, and torque on the crankshaft. For a 6 throw compressor it would typically require 6 engineers (one cylinder each) and one week to perform this task.

The 2nd Edition of API-618 was published in 1974 (39 pages). The committee pushed the compressor manufacturers to advise how rod loads were calculated and to ensure that everyone would calculate rod loads the same way. This established the term “allowable rod load” and “actual rod loading.” The actual rod load was defined as the force due to the differential pressure across the piston plus the inertia of the reciprocating parts transmitted through the piston rod. It also stated that the actual rod load calculated on the basis of cylinder relieving pressure (RV setting) shall not exceed the vendor’s maximum allowable rod load.

By this time mainframe computers and programmable calculators were in widespread use. This allowed for more precise engineering calculations and the elimination of some of the conservatism in the design process. Practice was to calculate compressor performance and “gas load” using a programmable calculator, since the computations were relatively simple. Basic compressor sizing and feasibility studies used these methods.

The final performance including actual rod load (combined rod load) was obtained using mainframe computers (punch cards, overnight batch processing, etc.). Gas loads were still calculated and reported based on nominal cylinder flange gas pressures, but actual rod load included the effect of valve pressure drop and inertia loads. There was variability between various users and OEMs on the reference points used for the calculation of combined rod load. At IR and Worthington, the reference point was the crosshead pin, so all inertia outboard of the pin bearing was included in the combined rod load calculation. There was also a lack of consistency over the relief valve pressure. Some users and OEMs (including IR) used the final relief valve pressure rather than each stage RV setting.

In the 3rd Edition (1986), API-618 grew to 111 pages. The term Maximum Allowable Combined Rod Load (MACRL) was defined. The combined rod load was defined as the algebraic sum of the differential gas pressure on the differential piston area plus the inertia force. The reference point for inertia loads was defined as being at the crosshead pin. Additionally API established a minimum rod load reversal criteria (to ensure proper lubrication, 3% and 15 degrees), but that issue is outside the scope of this paper. Gas load still was reported based on nominal cylinder flange pressures and the relief valve pressure calculation was still inconsistent.

In the 4th Edition (1995) API-618 was at 166 pages. The calculation of rod load was defined much more precisely. The terms Max Allowable Continuous Combined Rod Load (MACCRL) and Max Allowable Continuous Gas Load (MACGL) were established. This was the first time that load limits based on running gear and load limits based on the stationary components were explicitly separated in the specification. Combined rod load was defined the same way as the 3rd Edition but with the clarification that it was to be at
the crosshead pin and only the component in the direction of piston motion was included. Note that the load in the connecting rod is higher due to geometry. Gas load was defined as being the gas pressure inside the cylinder (cylinder flange pressure plus valve and passageway losses). Combined rod load and gas load had to be calculated every 10 degrees of rotation. These loads had to be calculated and must be less than the manufacturer’s MACCRL/MACGL limit at the RV setting of each stage and the minimum pressure for each stage.

Computer capabilities had increased to the point that the combined rod load calculations were readily available using PC based software and most machinery analyzers had the capability of computing the combined rod loads in real time as long as the measured in-cylinder pressures and the weights of the various components were properly utilized and interpreted.

In the 5th Edition (2007), API-618 was at 190 pages. The rod load definitions had only one minor change, they were to be calculated every 5 degrees instead of every 10 degrees.

API-618 6th Edition. This edition is in final revision and will be released soon. According to the current draft version, there are 2 changes. The term “Crosshead Pin Load” has been introduced. This has exactly the same meaning as the term combined rod load in the 4th and 5th edition and is defined as the algebraic sum of the gas load and inertia load on the crosshead pin parallel to the piston rod. The maximum crosshead pin load is to be calculated basis Relief Valve set pressure and every 5 degrees of crank rotation.

The other change is that the requirement has been eliminated for the manufacturers rod load rating to include a short duration 10% overload, called for in the 4th and 5th Ed. It is not clear what this overload was meant to accomplish, RV accumulation perhaps. However the 6th draft wording is the same as the 4th and 5th in that the max allowable rod load is basis RV set pressure which implies there could be a short term RV accumulation as gas actually blows through the RV, so there is no substantive change.
Glossary of Terms

Rated Rod Load (RRL). Term used in 1st edition of API-618 but without definition. At IR interpretation was gas only load not including valve losses. Was not to be exceeded on any normal operating load step (specified operating pressures, not relief valve setting).

Maximum Allowable Rod Load (MARL). Term used in 1st edition of API-618 but without definition. Was not to be exceeded at any upset including final discharge relief valve pressure. Operation at rod loads exceeding the MARL voided the warranty.

Actual Rod Load. Term used in 2nd edition of API-618 with definition. Included gas + valve losses + inertia loads in the calculation, but did not define the reference point for the calculation.

Maximum Allowable Combined Rod Load (MACRL). Term used in the 3rd edition of API-618 with definition. Same calculation as actual rod load (included gas + valve losses + inertia loads). Load was defined at the crosshead pin. This is the max load that can be applied at any load step including final RV pressure.

Maximum Allowable Continuous Combined Rod Load (MACCRL). Term used in 4th and 5th edition of API-618. Similar definition as MACRL except load to be calculated every 10 degrees (5 degrees in 5th edition) and load at the pin is the component in the direction of the piston motion. This is the current definition of the rated load that applies to the running gear.

Maximum Allowable Continuous Actual Rod Load (MACARL). Term used by some in industry. Same as MACCRL except the load reference point is on the rod (as opposed to the pin) at the rod’s weakest spot, and hence does not include any crosshead assembly weights for inertia calculations.

Maximum Allowable Continuous Gas Load (MACGL). Term used in 4th and 5th edition of API-618. Includes internal gas pressures inside the cylinder (flange gas pressure + valve + passageway losses). This is the current definition of the rated load that applies to the stationary components.

Internal Gas Load. Term commonly used with no API definition. Typically means the gas load based on pressure inside the cylinder, i.e. includes the valve and passageway losses. It is the same as the current API-618 definition of MACGL.

Maximum Allowable Crosshead Pin Load. Term used in the draft of API 618 6th edition. Means the same thing as the current API-618 (5th) definition of MACCRL. It is referred to as the crosshead pin load to avoid the frequent misconception that the combined rod load is at the crosshead end of the piston rod rather than the API defined crosshead pin reference point for inertia calculations.

Maximum Allowed Continuous Crosshead Load (MACCPL). Term used by some in industry. Same as MACCRL except the load reference point is on the crankpin (as opposed to the pin), and hence also includes a portion of the connecting rod weight for inertia calculations.
Terms Used by OEMs

**Total Rod Load.** Term used by Ariel with no API definition. It is used to define the maximum allowable load in tension plus compression that can be applied. Note the **Total Rod Load** limit is typically less than the sum of the internal gas load limit in compression and in tension. Ariel also lists a **Tension** and **Compression** limit which is defined as an internal gas load. A **Combined Rod Load** limit is not published; however, **Combined Rod Load** is calculated on the performance sheet and is used in determining acceptable rod load reversals at the crosshead pin.

**Maximum Allowed Internal Gas Rod Load.** Same as MARL but includes adjustment for pressure drops across valves and gas passageways to predict maximum and minimum internal pressures.

**Rated Rod Load Definitions by OEM**

Below is a partial list of how published Rated Rod Loads are calculated by various OEMs:

- **Ariel**
  - Total Rod Load – MAARL
  - Tension, Compression – Maximum internal gas rod load

- **Dresser Rand**
  - low speed – MACGL and MACCRL
  - high speed – Maximum allowable operating load –MACCRL

- **GE Gemini**
  - Rod load (tension or compression) – MACRL
  - Combined rod load – MACRL tension + MACRL compression

- **Ajax**
  - Rod Load – MARL

- **Superior**
  - Total gas rod load – MACGL tension + MACGL compression

- **Neuman & Esser**
  - Maximum rod load - MACCRL – meant to provide the nominal load of the frame rather than real design load limit
Effects of Real World Compression

Effect of Pulsations External to the Valves

Reciprocating compressor performance, and therefore gas rod load, is commonly calculated based on steady line pressure on the outside of the compressor valves. The valves are considered ideal orifice plates, so the pressure drop through the valve is proportional to instantaneous piston velocity. Cylinder pressure measurements and acoustic analysis simulations indicate that there are cases where the pulsation on the outside of the compressor valve can significantly impact the compressor performance predictions.

Gas inertia effects in reciprocating compressors tend to create pulsation that is in-phase with the fundamental compressor cycle. When the compressor discharge valves are open, there is an increase in pressure as gas is forced out of the compressor cylinder through the discharge nozzle. Similarly, there is a reduction in suction pressure as gas is being pulled into the suction valves, when the suction valves are open. As the following examples will demonstrate, this effect is most significant when velocity head is high in the cylinder nozzles. Fundamental pulsation due to inertia effects increases the delta-P across the piston, and therefore also increases the compressor power and peak gas rod load. Note that cylinder nozzle orifice plates are not effective to reduce the pulsation that occurs in-phase with the compressor cycle. In fact, tighter cylinder nozzle orifice plates tend to increase this pulsation.

Pulsation at the compressor valves due to an acoustic resonance, typically at a higher-order of run speed (3x or 4x), is less of a concern when considering the gas rod load. The compressor valves have significant impedance, which tends to isolate the clearance volume from the gas passages. Also, these types of acoustic resonances can typically be controlled with the use of cylinder nozzle orifice plates.
Figure 10 shows the distortion of a PV (pressure-volume) curve due to pulsations, as predicted by acoustic simulation. This specific plot is from a 360 RPM Hydrogen compressor, and the PV distortion due to pulsation is minimal. Figure 11 shows the gas rod load that result from the distorted PV curve. As expected, the effect of pulsation on the gas rod load is insignificant in this case.

![Pressure Volume Curve of 360 RPM Hydrogen Compressor](image)

**Figure 10: Pressure Volume Curve of 360 RPM Hydrogen Compressor**

![Gas Rod Load Predictions of 360 RPM Hydrogen Compressor](image)

**Figure 11: Gas Rod Load Predictions of 360 RPM Hydrogen Compressor**

0.37% Increase in gas rod load
Alternatively, Figure 12 shows the distortion of a PV curve due to pulsations of a 360 RPM Heavy Gas (s.g. = 1.6) compressor. In this case, the pulsation at the compressor valves is substantial due to inertia effects. The result is significant distortion of the PV curve when compared with the ideal (steady-line pressure) PV prediction. Figure 13 shows how the distortion of the PV curve due to pulsation results in significant (17.5%) increase in gas rod load.

**Figure 12: Pressure Volume Curve of 360 RPM 1.6 SG heavy gas Compressor**

**Figure 13: Gas Rod Load Predictions of 360 RPM 1.6SG heavy gas Compressor**
Similar calculations were completed for a range of compressor applications, including high speed/low speed compressors, and various gas compositions. Results from these simulations are summarized in Table 1.

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<td>14.04%</td>
<td>9.95%</td>
<td>Moderate</td>
<td>37</td>
</tr>
<tr>
<td>90000</td>
<td>1.60</td>
<td>360</td>
<td>15</td>
<td>900</td>
<td>2725</td>
<td>5.67</td>
<td>776</td>
<td>17.65%</td>
<td>17.50%</td>
<td>High-significant</td>
<td>95</td>
</tr>
<tr>
<td>28500</td>
<td>0.97</td>
<td>890</td>
<td>5.5</td>
<td>815.8</td>
<td>3492</td>
<td>4.5</td>
<td>245</td>
<td>16.81%</td>
<td>25.60%</td>
<td>High-significant</td>
<td>165</td>
</tr>
<tr>
<td>60000</td>
<td>1.47</td>
<td>813</td>
<td>6</td>
<td>891.0</td>
<td>9448</td>
<td>3.84</td>
<td>151</td>
<td>15.27%</td>
<td>10.60%</td>
<td>High-significant</td>
<td>95</td>
</tr>
<tr>
<td>80000</td>
<td>0.59</td>
<td>1000</td>
<td>6.75</td>
<td>1125.0</td>
<td>1498</td>
<td>3.3</td>
<td>772</td>
<td>3.89%</td>
<td>2.02%</td>
<td>Low-insignificant</td>
<td>38</td>
</tr>
<tr>
<td>60000</td>
<td>0.78</td>
<td>996</td>
<td>6.5</td>
<td>1079.0</td>
<td>3761</td>
<td>2.9</td>
<td>280</td>
<td>8.57%</td>
<td>3.50%</td>
<td>Low-insignificant</td>
<td>41</td>
</tr>
</tbody>
</table>

Table 1: Change in Gas Rod Load Due to Pulsation

To qualify when pulsations may significantly increase the gas rod load, based on initial performance calculations, a Severity Index is proposed:

$$ Severity\ Index = \frac{Specific\ Power \times Gas\ Velocity \times s. g. \times RPM \times Column\ Length}{60000} $$

Where:

$$ Specific\ Power = \frac{Indicated\ Power}{Nominal\ Indicated\ Rated\ Power} $$

Nominal Indicated Rated Power (hp) = Average Piston Speed \( \frac{ft}{min} \) \times Rated xhd pin Load (kips) \times 0.015

Note that the nominal indicated power is derived from the area of a PV card bounded by gas pressures to give a gas load equal to the rated crosshead pin load, and 100% volumetric efficiency, ie a theoretical rectangular card. As a practical matter, a PV card greater than 50% is difficult to achieve, so the .015 factor for nominal indicated rated power is based on the power of the PV card that is 50% of the full theoretical area.

$$ Gas\ Velocity\ (ft/\min) = Average\ Gas\ Velocity\ in\ Cylinder\ Nozzle,\ based\ on\ Average\ Piston\ Speed $$

$$ Column\ Length\ (ft) = Average\ Length\ from\ compressor\ valves\ to\ pulsation\ bottle $$

The column length includes the (average) gas passage length from the compressor valves to the cylinder flange, and the length form the compressor flange to the pulsation bottle. If this is not known, the column length can be approximated as follows:

$$ Column\ Length\ Estimate\ (ft) = Cylinder\ Bore\ (ft) + Stroke\ (ft) + Nozzle\ Inner\ Diameter\ (ft) $$
Using the Severity Index as defined above, together with the guidelines proposed in Table 2, it is possible to estimate when pulsation due to the gas inertia effect will significantly increase the gas rod load.

Note that the severity index guidelines in Table 2 are based on simulated results, and assume that higher-order pulsations are within guideline levels. Examination of measured PV cards indicates that higher-order pulsations, those at frequencies greater than the fundamental compressor cycle (typically from ¼ wave acoustic resonance) could change the gas inertia effect on rod load.

<table>
<thead>
<tr>
<th>Severity Index</th>
<th>Expected Change in Gas Rod Load</th>
<th>Qualitative Pulsation Effect on Gas Rod Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt;32</td>
<td>&lt;5 %</td>
<td>Low – Insignificant</td>
</tr>
<tr>
<td>32-61</td>
<td>5-10%</td>
<td>Moderate</td>
</tr>
<tr>
<td>&gt;61</td>
<td>&gt;10%</td>
<td>High</td>
</tr>
</tbody>
</table>

*Table 2: Guidelines for Severity Index, Gas Inertia Effect*
Effect of pressure pulsations inside the cylinder

ie transverse pressure waves (ref 1,2).

A basic assumption in compressor performance is that the pressure inside the cylinder is everywhere equal. In fact this is often not the case. The worst point is when the discharge valve opens, then a particle of gas remote from the discharge valve can only begin to move towards the discharge valve when a pressure wave which travels at the speed the sound reaches that particle. Until that point in time the gas will continue to be compressed by the movement of the piston. This results in an overpressure in the cylinder. A pressure wave is set up across the face of the piston with the anti-nodes at the inlet and discharge valve (top and bottom for a horizontal cylinder). The node is at the center between the inlet and discharge valve. As the usual location of the PV analyzer pressure transducer is located at the node, this phenomenon is difficult to detect. Factors which cause an increase in internal pressure wave are large cylinder diameter, high rotative speed, low sonic velocity (ie high molecular weight gas), high instantaneous piston velocity and low internal volume inside the cylinder at the point in time that the discharge valve opens.

Figure 14 shows a CFD output model of a 32.5” cylinder operating on CO2. This has a moderately high internal pulsation of 22 psi p-p or approximately 9% of the discharge pressure.

![Figure 14: CFD Model Output](image)
Figure 15 shows a slightly different operating case. This time the pressure difference p-p is 18 psi. This shows that when the discharge valve opens at approximately 300 degrees there is a large overpressure and a pressure wave is set up. This continues though 360 degrees (Top Dead Center) and decays during the expansion stroke (360-420 degrees ie 0-60 degrees) there is a small pulsation wave when the inlet inlet valve opens just after 420 degrees (60 degrees) then the pressure difference decays to nothing during the suction stroke and continues at almost zero during the compression (180-300 degrees).

![Figure 15: Pressure vs Crank Angle – Discharge Valve](image)

It has been determined that the best predictor of internal pressure pulsations is the crankshaft rotation angle that it takes for a sound wave to traverse across the piston face in the cylinder at discharge conditions, together with the volume in the cylinder and the instantaneous piston velocity when the discharge valve opens. (greater crank angle, higher piston velocity and lower internal volume cause increased pulsation). The volume fraction in the cylinder at valve opening can be approximated as the ratio $1/R^{1/n}$ where R is the compression ratio and n is volume exponent. The piston velocity is a maximum at midstroke, but it starts to drop significantly as the piston approaches 75% stroke (25% discharge volumetric efficiency). This implies that the severity of the pressure pulsations will increase with compression ratio (actually $R^{1/n}$), when $R^{1/n}$ reaches 2 then the volume in the cylinder (swept volume plus fixed clearance) has dropped to half and the discharge VE is less than 50% and so the piston velocity is dropping.

Figure 16 shows a severity chart for estimating pressure pulsations.
The 4 qualitative levels of internal pulsation approximately correspond to; negligible is below the “low” line, low is between the “low” and “moderate” line 5% p-p pulsation, moderate is between the “moderate” and “high” lines 10% p-p, and high is greater than the “high” line 15% p-p pulsation. Note that this chart is based on a cylinder clearance of 25%. If the clearance is less then the effect is a little more severe, if greater than 25% clearance then slightly less severe.

Angle of crank rotation for sound wave to traverse the piston face in radians=ω.D/u radians. Where ω is the angular velocity in rad/second, D is piston diameter inches, u sonic velocity in/sec

In customary units this is equal to A=N.D/(2.U) where A=Degrees of crank rotation, N=rotative speed rpm, D=piston diameter inches, U=sonic velocity fps at discharge gas pressure and temperature.
Table 3 Examples. In these examples you can see that due to the high sonic velocity hydrogen cylinders have very low internal pulsations. However heavier gases have lower sonic velocity so when used with large diameter cylinders and running at high rotation speeds can have significant internal pulsations. Note the example in the right most column is from Table 4 in the Steinruck paper referenced. Steinruck measured a pulsation of 23% and the value predicted in the severity chart shows good agreement at greater than 15%.

<table>
<thead>
<tr>
<th>Gas</th>
<th>Hydrogen</th>
<th>Nat Gas</th>
<th>CO2</th>
<th>CO2</th>
<th>51.2 MW</th>
<th>Nitrogen</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston Dia Inch</td>
<td>20</td>
<td>20</td>
<td>26</td>
<td>26</td>
<td>39.5</td>
<td>26.5</td>
</tr>
<tr>
<td>Compr Ratio</td>
<td>2.5</td>
<td>2.5</td>
<td>2.5</td>
<td>2.5</td>
<td>3.94</td>
<td>2.9</td>
</tr>
<tr>
<td>R&lt;sup&gt;1/ln&lt;/sup&gt;</td>
<td>1.91</td>
<td>2.07</td>
<td>2.07</td>
<td>2.07</td>
<td>3.19</td>
<td></td>
</tr>
<tr>
<td>Discharge P psia</td>
<td>250</td>
<td>250</td>
<td>250</td>
<td>250</td>
<td>75.8</td>
<td>92.8</td>
</tr>
<tr>
<td>Disch T deg F</td>
<td>270</td>
<td>218</td>
<td>223</td>
<td>223</td>
<td>229</td>
<td></td>
</tr>
<tr>
<td>D Sonic Vel fpm</td>
<td>5069</td>
<td>1551</td>
<td>966</td>
<td>966</td>
<td>881</td>
<td>1372</td>
</tr>
<tr>
<td>Rotative speed rpm</td>
<td>1000</td>
<td>1000</td>
<td>1000</td>
<td>720</td>
<td>360</td>
<td>1182</td>
</tr>
<tr>
<td>Deg crank rotation for sound to traverse piston</td>
<td>2.0</td>
<td>6.4</td>
<td>10.3</td>
<td>7.4</td>
<td>8.1</td>
<td>11.4</td>
</tr>
<tr>
<td>Qualitative level</td>
<td>Negligible</td>
<td>Moderate</td>
<td>High</td>
<td>Moderate</td>
<td>High</td>
<td>High</td>
</tr>
</tbody>
</table>

**Table 3: Severity Level Examples**

It is interesting that the work by D-R and the work by Steinruck were both done around the same time and independently. The approach is different however both agree that severity level (D-R) and classification number (Steinruck) is related to time for a sound wave to traverse the cylinder and the discharge event. As noted above the results are similar.

Effect of high lateral pulsations; High pulsation levels will have an adverse effect on discharge valve operation and impose a large bending moment on the piston and cylinder. This can result is piston rocking, piston reliability especially on sensitive pistons (large diameter compared to the piston length and rod diameter), cylinder vibration (rocking mode about horizontal plane at 90 deg to piston rod axis).
**Effect of Torsional Loads**

The response of the system to torsional loads results in a harmonic speed variation of the compressor crankshaft around the mean crankshaft rotational speed. The speed variation in turn impacts the rod loads.

Take the example of a large four throw compressor:

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Operating Speed</td>
<td>890 rpm</td>
</tr>
<tr>
<td>Piston weight</td>
<td>290 lbs</td>
</tr>
<tr>
<td>Connect rod length</td>
<td>18.5 in</td>
</tr>
<tr>
<td>Stroke</td>
<td>6.75 in</td>
</tr>
<tr>
<td>Allowable gas rod load</td>
<td>75,000 lb tension or compression</td>
</tr>
</tbody>
</table>

The speed variation is +/- 40 rpm and occurs at 8X the shaft speed. The overall piston rod load is 9% greater in tension, and 15% greater in compression with that superimposed speed variation. Figure 17 compares the piston rod load with and without the speed variation.

![Figure 17: Rod Load Due to Inertia and Crankshaft Speed](image)

It should be noted that although the speed variation occurs at 8X shaft speed this results in harmonics at 7X and 9X the shaft speed in the rod load. Table 4 is a comparison of the inertia component of rod load harmonic content with and without torsional oscillation.
<table>
<thead>
<tr>
<th>Order</th>
<th>Inertial Rod Loads without Torsional Oscillation (lbs)</th>
<th>Inertial Rod Loads with Torsional Oscillation (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>1</td>
<td>22020</td>
<td>22019</td>
</tr>
<tr>
<td>2</td>
<td>4051</td>
<td>4051</td>
</tr>
<tr>
<td>3</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>4</td>
<td>34</td>
<td>34</td>
</tr>
<tr>
<td>5</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>6</td>
<td>0</td>
<td>204</td>
</tr>
<tr>
<td>7</td>
<td>0</td>
<td>3030</td>
</tr>
<tr>
<td>8</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>9</td>
<td>0</td>
<td>5009</td>
</tr>
<tr>
<td>10</td>
<td>0</td>
<td>569</td>
</tr>
</tbody>
</table>

**Table 4: Load Rod Harmonic Content**

For this example case, the inertial rod load at 9X run speed approaches 7% of the allowable gas rod load for the compressor. In this case the torsional oscillation is 4.5% 0-p of 890 rpm at 8x or 119Hz. The guideline value (ref 3) is 1.8% (good) or 2.7% (marginal). This demonstrates that provided the torsional analysis gives torsional pulsations that are within guideline values, then this is not a significant factor. However, if torsional pulsations are high, then they can have a significant effect on the inertia component of rod load.

**References**


3. GMRC Guideline for high speed reciprocating compressor packages for natural gas transmission and storage applications. Figure 7.3a Published 2013 [www.gmrc.org](http://www.gmrc.org)