

A Discussion of the Various Loads Used to Rate Reciprocating Compressors

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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. “Rod load” is one of the most widely used, but least understood reciprocating compressor descriptors in industry. Typical end users know that rod load is a factor used to “rate” a compressor, but they don’t generally have a good understanding of how this rating is developed and how to utilize it for machinery protection.

This paper discusses the various definitions of rod load, including historical and current API-618 definitions, manufacturer’s ratings, and various user interpretations. It also explains that there are really load limits based on the *running gear* (moving parts such as pistons, rods, crosshead, crankthrow, etc.) as well as load limits based on the *stationary components* (frame, crosshead guide, etc.).

The basic kinematics and forces acting on a slider-crank mechanism will be reviewed to provide a better understanding of the various definitions that are used. Analytical results and field rod load measurements will be compared to illustrate the various factors that influence rod load on typical compressor installations.

Basic Theory

Consider the typical double-acting compressor cylinder geometry 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.

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.

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:

$$F_{Tension} = (P_{Discharge} \times A_{CE}) - (P_{Suction} \times A_{HE}) \quad (1)$$

$$F_{Compression} = (P_{Discharge} \times A_{HE}) - (P_{Suction} \times A_{CE}) \quad (2)$$

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%)
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 through 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. In any case, the difference should be small provided the losses are within the typical values listed above.

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 $P_{Suction}$ and $P_{Discharge}$, 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.

Of course 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.

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

$$x = r \cos \theta + l \sqrt{1 - \frac{r^2 \sin^2 \theta}{l^2}} \quad (3)$$

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

$$x = r \cos \theta + l \left(1 - \frac{r^2}{2l^2} \sin^2 \theta \right) \quad (4)$$

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:

$$F_I = m_{recip} r \omega^2 \left[\cos(\omega t) + \frac{r}{l} \cos(2\omega t) \right] \quad (5)$$

where:

m_{recip} = mass of reciprocating components

r = crank radius

ω = angular velocity, $\dot{\theta}$

l = connecting rod length

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.

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.

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 less 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.

API-618 5th edition is scheduled to be published this year. The rod load definitions have only one minor change, they are to be calculated every 5 degrees instead of every 10 degrees.

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. At IR interpretation was gas only load not including valve losses. 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 Gas Load (MACGL). Term used in 4th and 5th edition of API-618. Includes internal gas pressures inside the cylinder (flange gas + 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**.

Crosshead Pin Load. Term commonly used with no API definition. Typically means the same thing as the current API-618 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.

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 *Rod Load* 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.

User's Perspective

Various OEMs used other terms such as “Maximum Allowable Frame Load”, “Maximum Allowable Gas Load”, etc. but only **MACCRL** and **MACGL** are defined in API-618 and only since the 4th Edition (1995). It is not clear that all OEMs, users, analysts, operators, analyzer vendors, etc. recognize these terms and agree that reciprocating machinery should be rated in this manner.

Oxy Permian owns, operates, and maintains in excess of 400,000 horsepower of compression in West Texas and Eastern New Mexico. This includes screw, centrifugal, and reciprocating machines. Most of these machines are reciprocating compressors and range in age from a few months to several decades with the majority being installed prior to 1986 (i.e. pre-3rd Edition of API-618). The OEM published “rod load” limits range from 18,000 to 225,000 lb_f. These machines vary by service, manufacturer, speed, loading, installation, and operating philosophies and yield an array of equipment configurations. Longevity of service requires many machines to be subjected to a variety of process conditions that result in full utilization of “rod load” capabilities applied to these machines at commissioning. Several factors affect the actual rod load on a machine including declining field pressures, process changes, improper operation of unloaders, valve degradation, ring failure, process upsets, and machinery modifications.

Oxy Permian performs compressor analyses on the majority of reciprocating compressors at approximately six-week intervals. This snapshot of compressor operation presents the facility with the machinery health at the time of data collection. The rod load is presented, generally in a “percentage of allowable rod load” format, and the facility makes maintenance and operation decisions based on various components of these health reports. The question is: What exactly are we looking at and how do we compare our measured “rod load” to the OEM recommended maximum? For practical purposes a facility is able to measure the gas load on a compressor by using either peak pressures generated inside the cylinder (periodic measurements by experience analyst with portable equipment), or flange pressures (typical pressure gages, transmitters, etc.), along with the cross sectional area of the piston. If the flange pressures are used, then the resulting loads must be compared to allowable “flange loads”, but that is not an API definition and all OEMs do not provide such allowable loads. If the appropriate reciprocating weights are known (piston, piston rod, nuts, rings, riders, crosshead, bushings, pins, etc.) then the inertial loads can also be calculated. These inertial loads can be added to the gas load to develop the combined rod load. However, the reciprocating weights are not always known as modifications may have been made over the years and the degree of recordkeeping may be in question. A thorough understanding of what the rod load rating implies is generally not apparent to the typical machinery analyst/engineer/operator. The analyzer software will typically display both gas loads and combined rod loads, but only one allowable value is available (see Figure 10). Many times the combined rod load is

lower than the gas rod load, but this is not always the case as will be shown later. The standard rod load reports (see Figure 11) do not compare the measured values to both **MACCRL** and **MACGL**.

Facilities must continually do more with less and are generally capital constrained such that we must obtain every pound of load capability, in addition to all available horsepower, from a given machine. As we pull liners from cylinders, install new cylinders, and push the machines to the max, we have found that in some cases we do not actually understand the rod load limits of the compressor. Economic viability of a new project, whether revamping an existing machine, or adding either parallel or series compression, is usually based on three items: power availability, additional throughput, and machinery limitations. It is imperative that the OEM be included when evaluating design options because as an end user, we are not always privy to all design limitations on a machine. For example, the original installation may have been designed with custom distance pieces on one or more throws, and that may limit the load carrying capability of the frame. The following examples serve to illustrate the issues at hand.

Performance Study to Evaluate Compressor Re-Rate

A group of compressors was being considered for a re-rate project to increase capacity. The economics of the project depended on (among other things) the capital cost to modify the existing compressors if the increased capacity resulted in overload conditions. The compressors in question were from two different OEMs and were pre-1995 (prior to API-618 4th Edition) vintage. The load ratings for one type of compressor were provided in terms of “Rated Rod Load” and “Maximum Allowable Rod Load”. The load rating for the other compressor model were provided in terms of “Rated Rod Load”, “Rated Flange-to-Flange Load”, “Maximum Allowable Rod Load”, and “Maximum Allowable Flange-to-Flange Gas Load.”

For brand “X”, the Rated Rod Load was 150,000 lbs, and the Maximum Allowable Rod Load was 180,000 lb. The performance study concluded that overload conditions would occur, based on comparing the calculated pin loads to the 150,000 lb limit. It was later clarified that the **MACCRL** was 180,000 lbs and the **MACGL** was 180,000 lbs for this application.

The brand “Y” machines had a Rated Rod Load of 175,000 lbs, a Rated Flange-to-Flange Gas Load of 187,500 lbs, a Maximum Allowable Rod Load of 210,000 lbs and a Maximum Allowable Flange-to-Flange Gas Load of 225,000 lbs. The performance study also concluded that overload conditions would occur based on comparing the pin loads to the 175,000 lb limit. It was later clarified that the **MACCRL** was 210,000 lbs and the **MACGL** was 225,000 lbs for this application.

The predicted overload situation initially led to proposed modifications to the compressors. This would have added a significant capital cost to the project. After the rod load ratings and operating conditions were reviewed with the OEM (and the current API definitions were applied), it was verified that there was not an overload condition.

The project was delayed while the overload issue was resolved and later deferred, due to market conditions.

Combined Load Exceeds Gas Load

Many users mistakenly assume that the combined rod loads (gas plus inertia) will always be lower than the gas loads. This is not true for low ratio (high volumetric efficiency) applications. As an approximate rule, if the discharge volumetric efficiency (VE) exceeds 50%, the gas load will reach a maximum prior to 90 degrees, while both inertia load and gas load are same sign thus are additive. If the discharge VE is less than 50% then the gas load does not reach a maximum until after 90 degrees and so it is opposite in sign to the inertia load and the combined rod load will be less than the gas load. This is illustrated in Figure 12, which shows measured rod load data for a 250 RPM single-stage compressor in natural gas transmission service.

Distorted Pressure Measurements = Distorted Rod Loads

Another common mistake is to report distorted rod loads based on distorted pressure measurements. This is most often due to the “channel resonance” effect present in nearly all in-cylinder pressure measurements. This effect is illustrated in Figures 13 and 14. The rod load plot without channel resonance correction is shown in Figure 13, while the corrected plot is shown in Figure 14. The reported rod load is higher when the channel resonance is present.

Conclusions

Compressor rod load ratings are often misunderstood and misapplied. It is important to understand that the API definitions of **MACCRL** and **MACGL** are not actually rod loads, but refer to crosshead pin loads and gas loads, respectively. The API definitions help to avoid confusion, but these ratings are not always available for pre-1995 vintage machines.

“Measured” rod loads are actually computed rod loads based on measured pressures. The forces based on the measured pressures are combined with inertia forces based on the weights of reciprocating components that are input into the analysis software. If the pressure measurements are distorted and/or the reciprocating weights are not accurately known, then the combined rod loads reported will be erroneous.

There is some logic in using the simplified gas rod load calculations presented in equations 1 and 2. The trends will be correct (i.e. higher differential pressure results in higher rod load). However, if nominal flange pressures are used to rate a compressor, care must be taken to include enough margin to account for the maximum possible in-cylinder pressures due to pressure drop, valve losses, pulsation and valve dynamics. These effects vary for each application.

The user must make a decision when a compressor is revamped on whether to use current API definitions and ratings, or the ratings in effect when the machine was first installed. Again, it is imperative that the OEM be included when evaluating design options because as an end user, we are not privy to all design limitations on a machine.

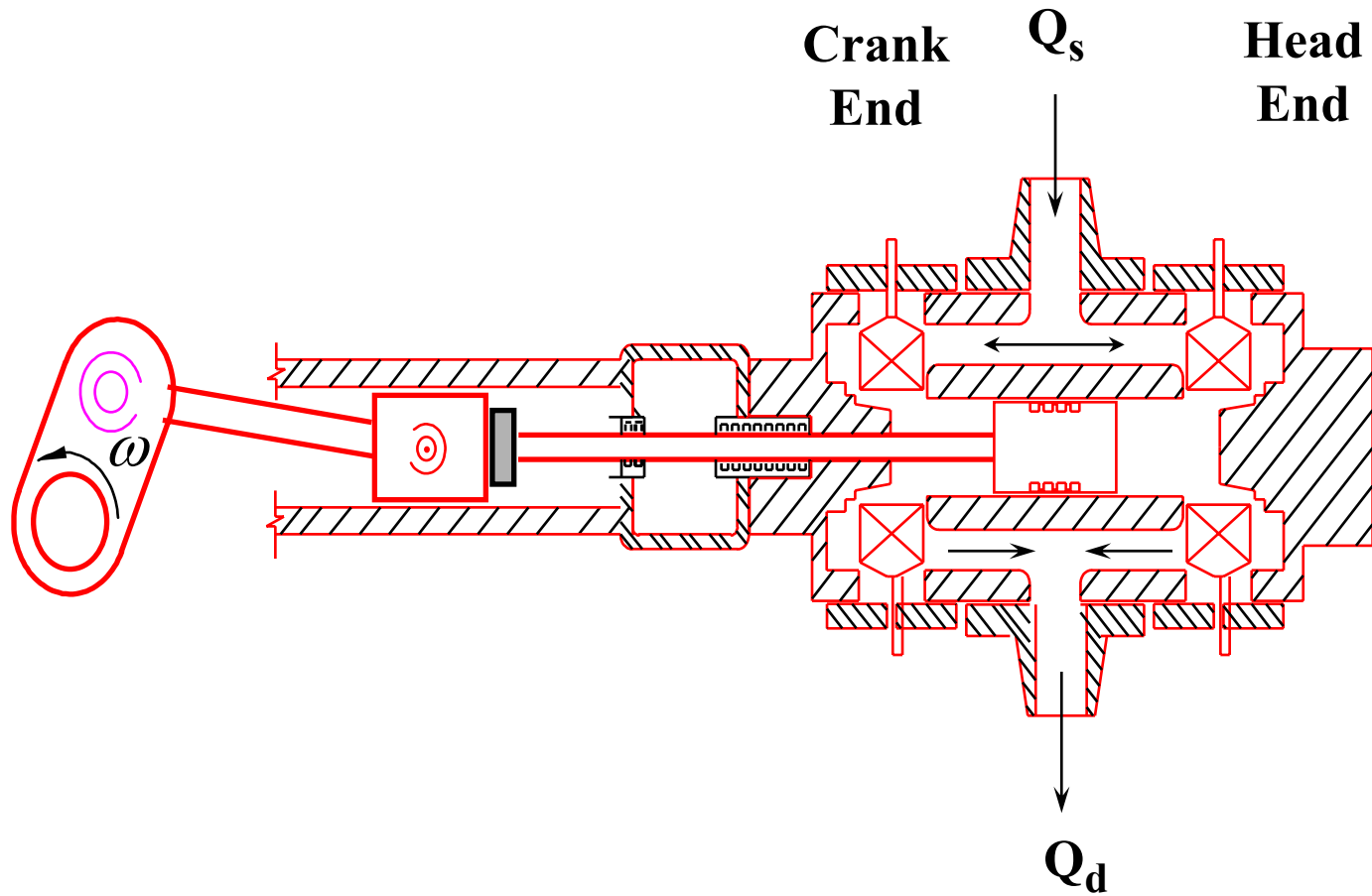
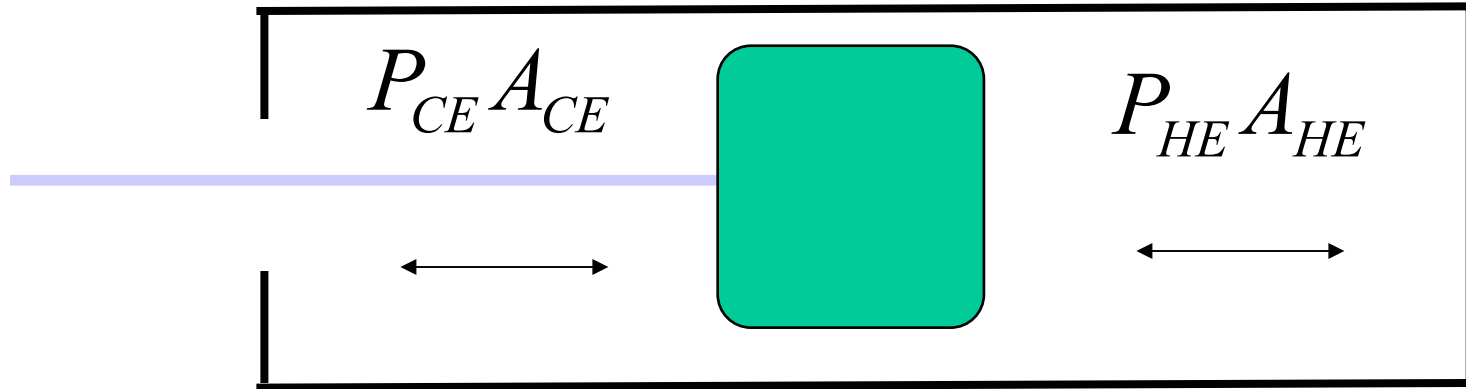


Figure 1. Double-Acting Compressor Cylinder



$$F = P_{CE} A_{CE} - P_{HE} A_{HE}$$

P = Static & Dynamic Pressure

A = Area of Piston

Figure 2. Gas Loads

Figure 3. Ideal P-T Diagram

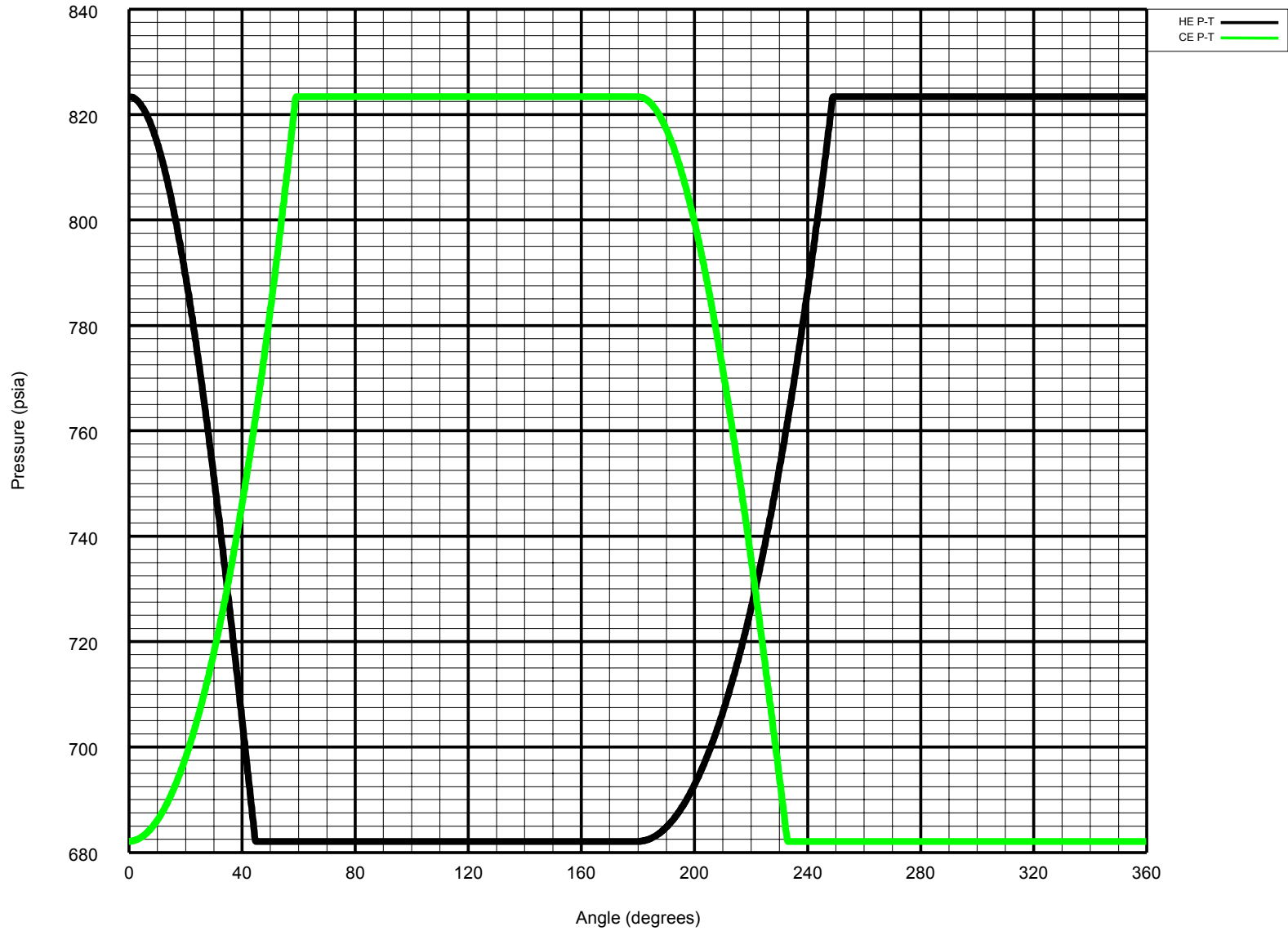


Figure 4. Rod Loads Based on Ideal P-T Diagram

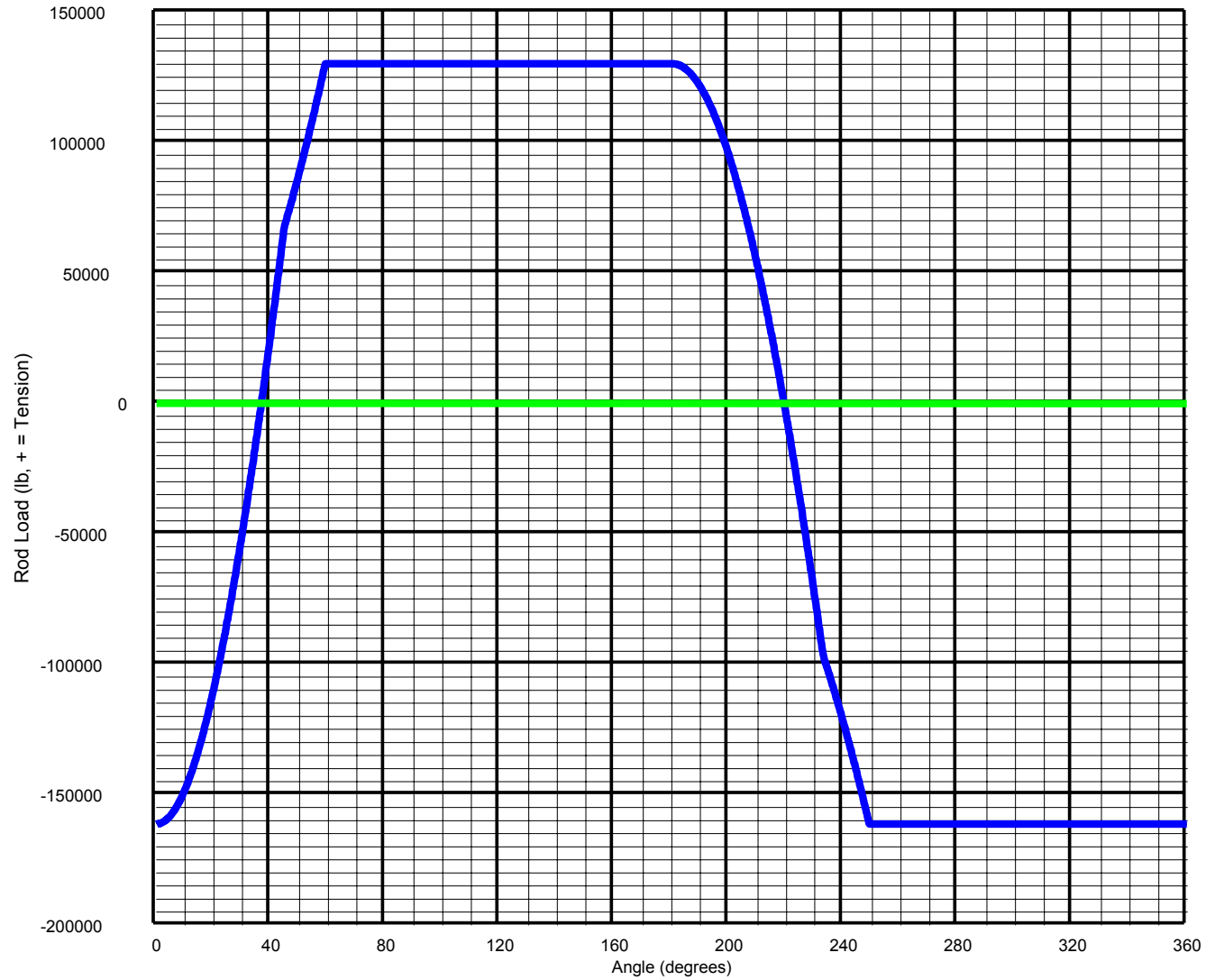
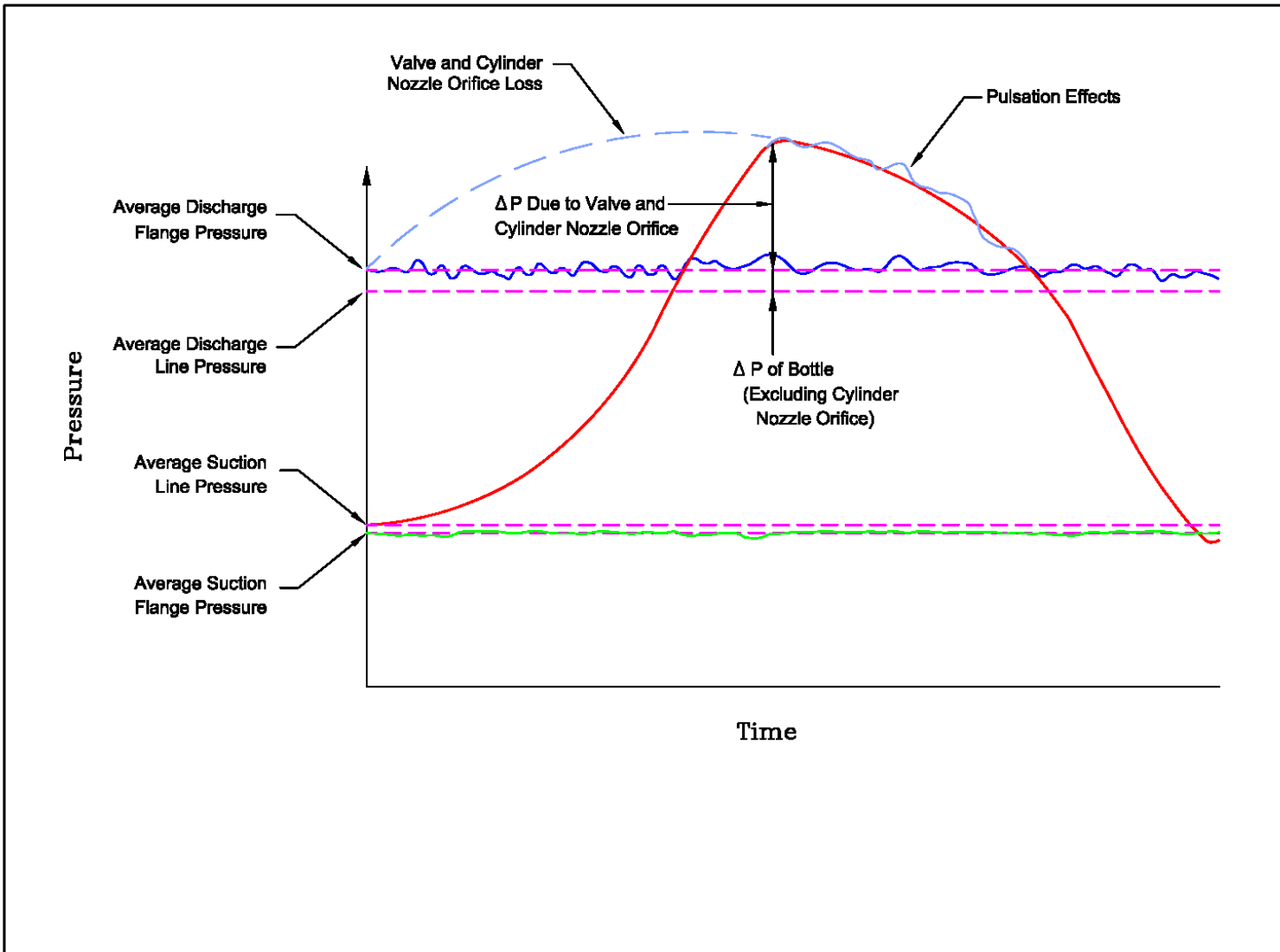


Figure 5. Non-Ideal P-T Diagram



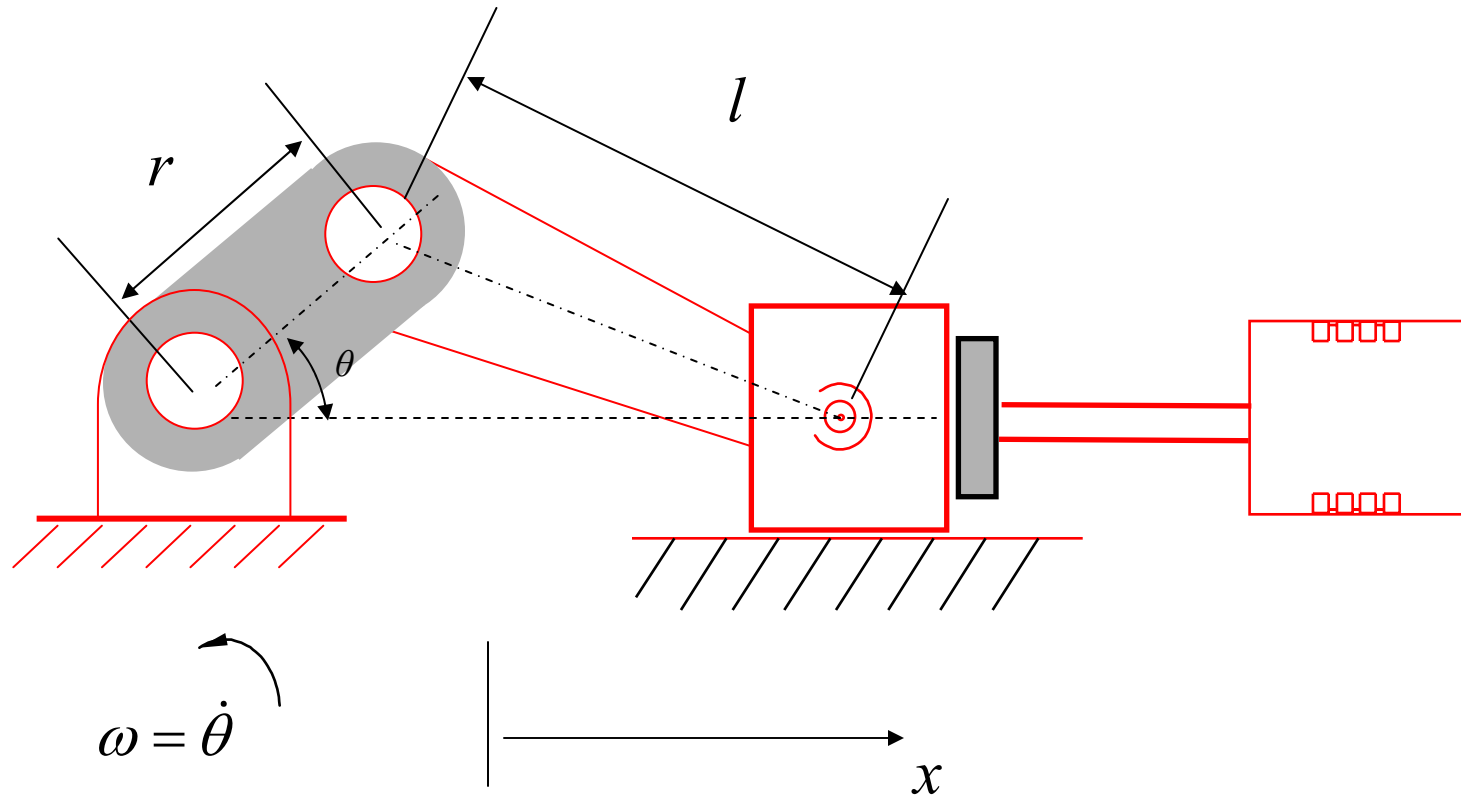


Figure 6. Slider Crank Geometry

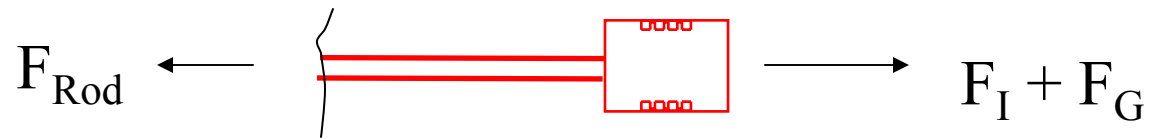


Figure 7. Piston Rod Load

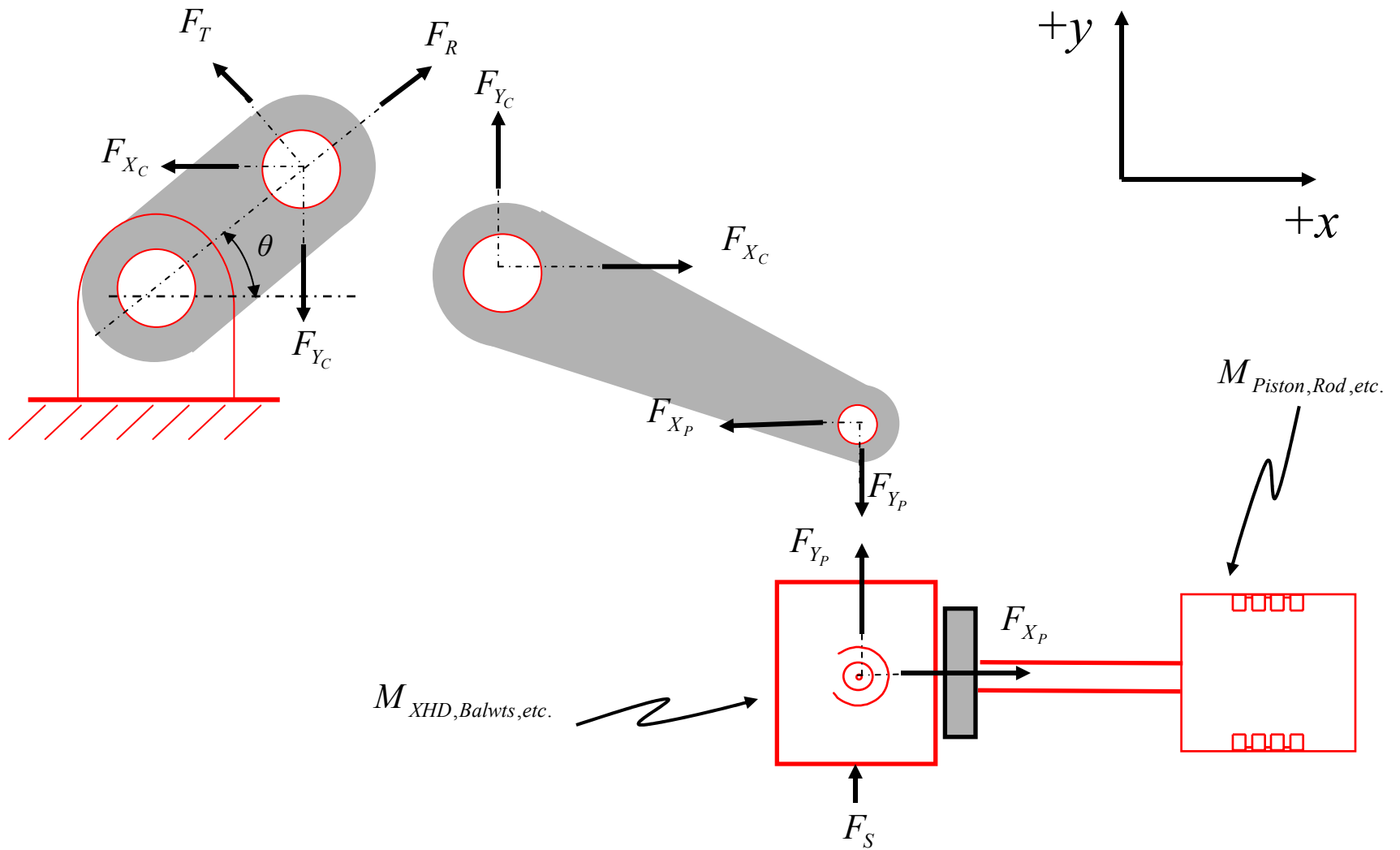


Figure 8. Forces Acting on Crosshead Pin

Figure 9. Design Calculation Results

BDC									
Crank Ang. (Theta)	Local Crank Ang. (Alpha)	X-M.Pin* Force (Lb)	Y-M.Pin* Force (Lb)	Comp. Piston Rod Force (Lb)	Master Side Force (Lb)	Torque* (In-Lb)	Press. Torque* (In-Lb)	Press. H.E. (Psi)	Press. C.E (Psi)
0	0	-47279.7	0	-47597.9	0	0	0	1698.2	822.5
1	1	-47156.7	139.3	-47474.9	139.3	7698.6	18334.4	1697	822.6
2	2	-46788.7	276.5	-47106.9	276.5	15273.7	36526.8	1693.4	822.8
3	3	-46178.8	409.2	-46497	409.2	22603.7	54437.1	1687.4	823.2
4	4	-45332	535.4	-45650.2	535.4	29570.7	71928.8	1679.2	823.8
5	5	-44255.2	653.1	-44573.5	653.1	36061.9	88871.1	1668.6	824.5
6	6	-42957.2	760.4	-43275.4	760.4	41971.6	105139.8	1655.9	825.4
7	7	-41448	855.4	-41766.2	855.4	47202.2	120619.5	1641.2	826.5
8	8	-39739.3	936.7	-40057.6	936.7	51665.6	135204.3	1624.4	827.7
9	9	-37844.1	1002.7	-38162.3	1002.7	55283.8	148798.9	1605.8	829.1
10	10	-35776.1	1052.3	-36094.3	1052.3	57990.2	161319.3	1585.5	830.6
11	11	-33550	1084.4	-33868.2	1084.4	59729.2	172693.2	1563.7	832.3
12	12	-31180.8	1098.3	-31499	1098.3	60457.3	182860.5	1540.3	834.2
13	13	-28684.1	1093.3	-29002.3	1093.3	60142.2	191773	1515.7	836.2
14	14	-26075.4	1069	-26393.6	1069	58763.3	199394.4	1490	838.5
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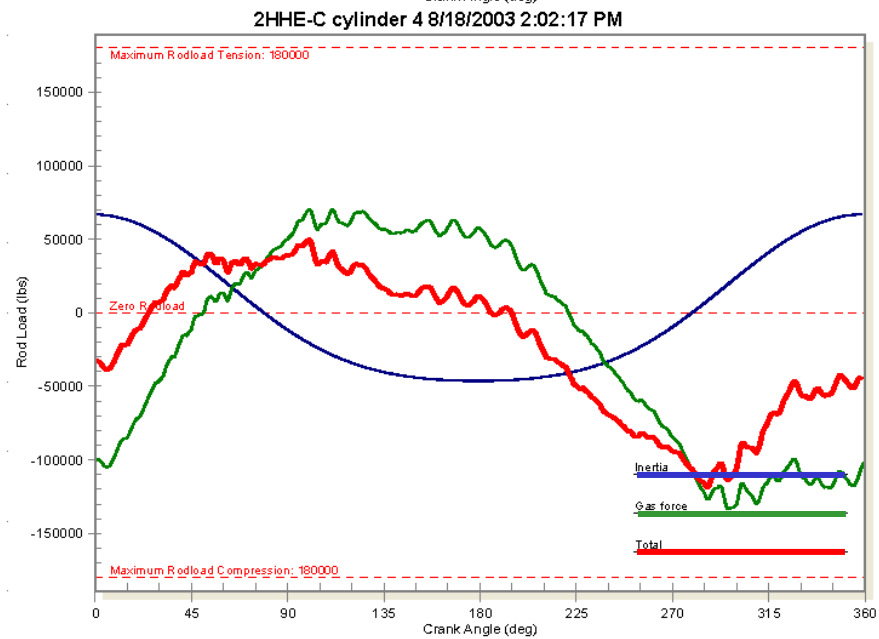
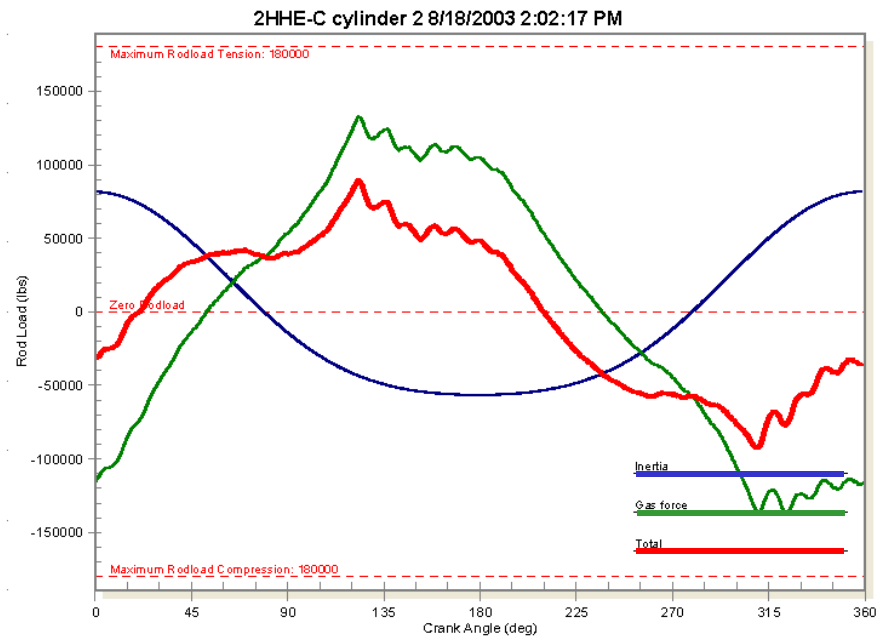


Figure 10. Measured “Rod Loads”

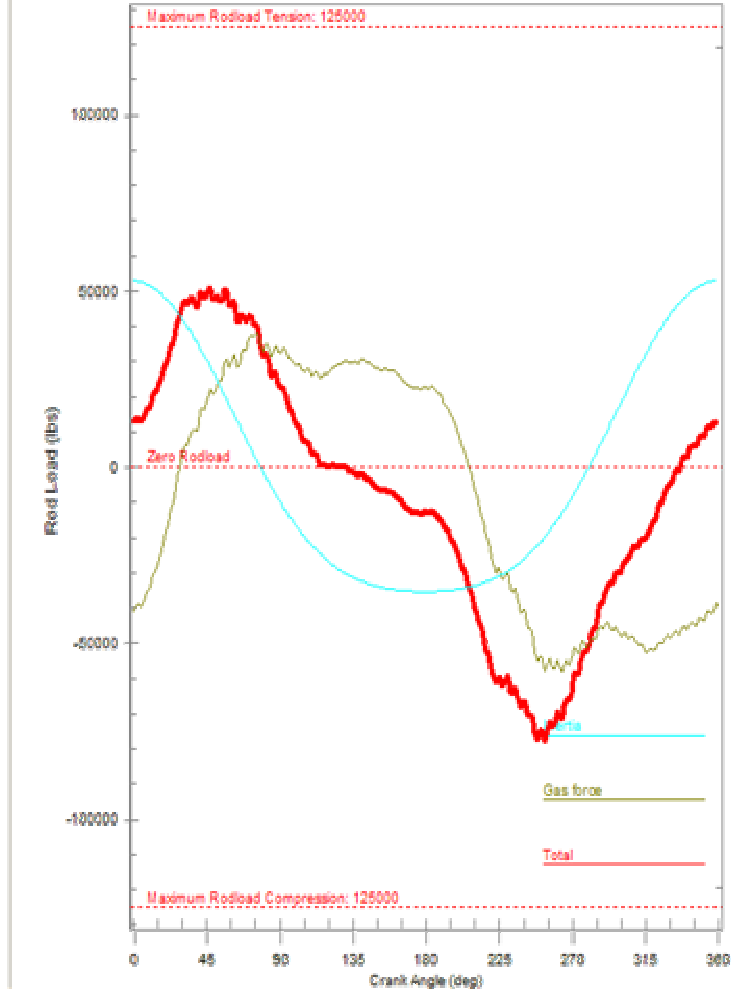
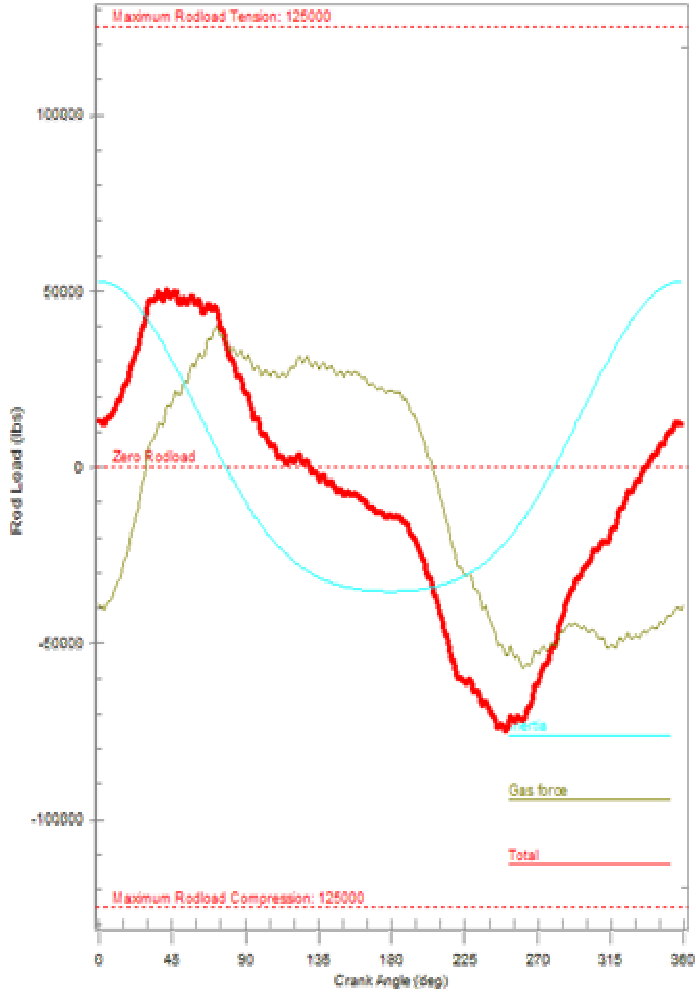
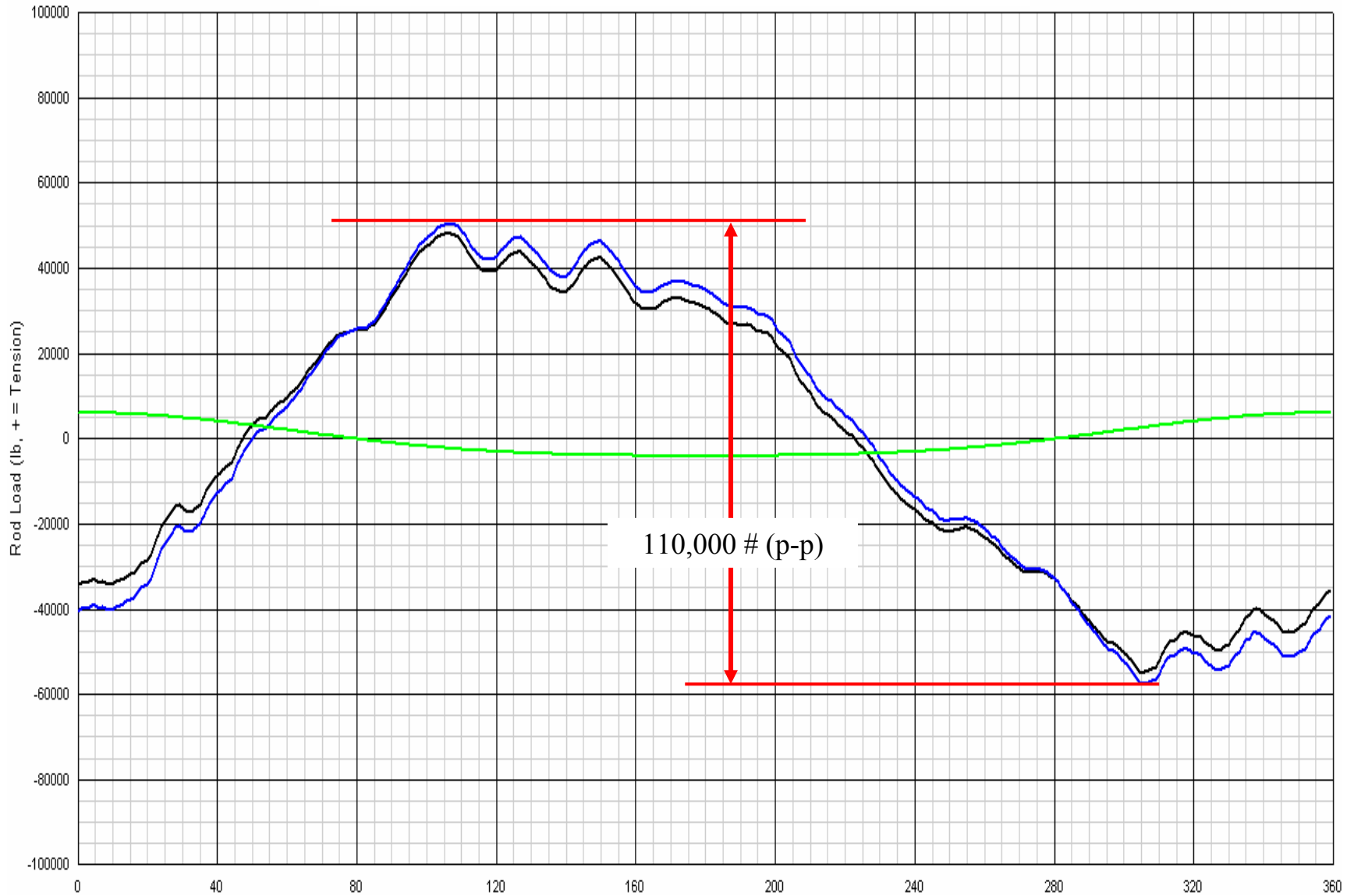


Figure 12. Combined Pin Load Exceeds Gas Load

Figures 13. Distorted Pressure Measurements



Figures 14. Corrected Pressure Measurements

