HOW INTERNAL GAS FORCES AFFECT THE RELIABILITY OF RECIPROCATING COMPRESSORS

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INTRODUCTION

To keep reciprocating compressors operating safely and reliably, users must always take great care to use them in the manner they were designed to operate. While there are other factors that can affect compressor reliability, such as operating conditions, gas composition, lubrication issues, and foundation issues, in this short article, we will concentrate on two key issues that potentially can shorten the useful life span of your reciprocating compressors:

1. Exceeding the compressor’s rod load rating and
2. Operating a compressor in a non-reversing gas load condition.

GAS LOADS

Let’s begin with a brief overview of gas loads within healthy reciprocating compressors.

As the compressor piston (Figure 1) 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 caused by the 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:

\[
F_{\text{tension}} = P_{\text{discharge}} A_{\text{CE}} - P_{\text{suction}} A_{\text{HE}}
\]

\[
F_{\text{compression}} = P_{\text{discharge}} A_{\text{HE}} - P_{\text{suction}} A_{\text{CE}}
\]

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 618 specifications.

“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.

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:

- Pressure drop caused by valve and cylinder passage losses (typically 2 to 10%)
- Pressure drop as a result of pulsation control devices (typically <1%)
- Pulsation at cylinder valves (typically <7%)
- 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 illust.
trated 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 until all 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 from 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 if 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.

**NON-REVERSING GAS LOADS**

During every rotation of a compressor crank, the load acting on every crosshead pin is designed to reverse completely. In other words, the load acting on every compressor throw must go from compression to tension through each rotation. Without a sufficient reversing load, the crosshead bushing will not be provided with sufficient lubrication and will eventually fail due to inadequate lubrication (Figure 6). Pin load reversal is characterized by two components, degrees and percent. These components represent the duration of the reversal and the magnitude of the reversal. Both values must meet or exceed minimum values. For example, Ariel compressors require a reversal for at least 30 degrees of the crank rotation, with at least 25% change in load magnitude throughout each rotation. The percent load magnitude is defined by the smaller of the tension or compression force divided by \( \frac{P_{\text{discharge}}}{P_{\text{suction}}} \) and the magnitude of the reversal. Both values must meet or exceed minimum values. For example, Ariel compressors require a reversal for at least 30 degrees of the crank rotation, with at least 25% change in load magnitude throughout each rotation. The percent load magnitude is defined by the smaller of the tension or compression force divided by \( \frac{P_{\text{discharge}}}{P_{\text{suction}}} \) and the magnitude of the reversal.

Normally, compressor designers dutifully design compressors to operate in a safe load reversing mode. The designers review all the expected loading conditions to ensure reliable service under all process load conditions. There may be rare occasions when a certain speed condition, unloader position, or set of pressures could lead to a non-reversal failure. Purchasers must be careful to review all rod load and reversal possibilities to ensure there are no design deficiencies that could lead to field problems.

However, in the real world, there are unexpected compressor conditions that can arise due to mechanical deficiencies. Valve failures are the most common component failures encountered in the field, usually caused by entrained solids or liquids. When valve failures occur, unexpected pressure balances can result across a piston, which can lead to unfavorable rod loading conditions.

**NON-REVERSING ROD CONDITIONS MATRIX**

Let’s look at a case where a single valve failure can lead to a severe non-reversing load condition. If we have a single-acting compressor, one that only compresses on the head end of the cylinder, a single discharge valve failure can lead to a severe non-reversing load condition and a catastrophic failure due to the resulting abnormal pressure conditions acting on the piston. To better understand the relationships between compressor design, load conditions, and the effect of valve failures, let’s take a closer look at Table 1.

Table 1 was developed by a compressor specialist with many years of experience with real field compressors. He has seen numerous non-rod reversal failures during his career. He decided to classify the possible scenarios that could lead to non-reversing failures. The failure scenarios assume that a major valve failure occurs while the compressor is running. For example, the first row is labeled “HE normal.” If we move horizontally to the column labeled “CE suction valve fails,” we find a yellow highlighted square, which means this condition may lead to light non-reversing condition.

Next, let’s go to the column labeled, “CE suction valve fails” and go across on the row labeled “HE discharge valve fails” until these two intersect, there you’ll find a rose-colored square. A rose-colored square means “severe non-rod reversal” will most likely occur. The formula in the landing square is used to determine the force generated by the component failure. In this failure mode, we see the following formula: \( A_{\text{HE}} \times P_{\text{discharge}} - A_{\text{CE}} \times P_{\text{suction}} \).

This formula defines the magnitude of the gas forces “pinning” the compressor rod to one side of the crosshead pin. The force is the head end piston area times the discharge pressure minus the crank end piston area times the suction pressure. Because of the HE discharge valve failure, the pressure on the HE will be a constant discharge pressure. If, in this example, we would experience a constant suction pressure on the CE, then the pressure acting on

(A) Some bushing designs (such as grooved bushings) have proven to be reliable with as little as 15 degrees of rod reversal with at 3% load magnitude. Simple bushings designs (un-grooved) may require a minimum of 45 degrees of rod reversal and a 20% load magnitude. The manufacturer should provide the actual reversing load requirements to the purchaser at the time of their proposal (See section 6.6.4 of API 618).
both sides of the piston would be static so the net pressure on the piston would be static and oriented toward the crank direction. Let’s look at another example. If you go to the column labeled “CE normal” and go across on the row labeled “HE discharge valve fails” until these two intersect, you’ll find an orange-colored square. An orange-colored square means “moderate non-rod reversal” will probably occur.

A severe non-rod reversal condition will probably lead to a catastrophic failure in a short period of time, where a moderate non-reversal condition will probably lead to a failure over time. The two worst-case scenarios are:

- **CE discharge valve fails + HE suction valve fails or is completely unloaded**
- **CE suction valve fails or is completely unloaded + HE discharge valve fails**

Note: This matrix is only meant to be used to educate compressor users on the consequences of various single- and double-cylinder valve failure modes.

Refer to Figure 7 to better understand what we mean by the terms “normal loading,” “net load is toward CE,” and “net load is toward HE” used in Table 1. As you see failed crosshead bushings in the field, you can begin matching the observed wear with the various loading possibilities shown in Figure 7. Hopefully, your bushing wear observations, Table 1, and the actual valve failure information will help you determine the mechanical root cause of your particular failure.

**NON-REVERSING GAS LOAD EXAMPLES**

Let’s go through a few examples so you can get a feel for the magnitudes of non-reversing gas forces that can be encountered due to abnormal conditions. For these examples, we selected representative values for the suction pressure, discharge pressure, rod area, crank-end effective piston area, and the head-end effective piston area. After entering our input values (Table 2), our spreadsheet returned load values (gas loads only) for “light non-rod reversal,” “moderate non-rod reversal,” and “severe non-rod reversal” conditions.

**Calculation Results:**

<table>
<thead>
<tr>
<th>Condition</th>
<th>Load Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light non-rod reversal</td>
<td>219.8 lbs. (99.7 kg)</td>
</tr>
<tr>
<td>Moderate non-rod reversal</td>
<td>565.2 lbs. (256.4 kg)</td>
</tr>
<tr>
<td>Severe non-rod reversal</td>
<td>20,852 lbs. (9458.3 kg)</td>
</tr>
</tbody>
</table>

It’s easy to see that the severity of the pinning load varies from 219.8 lbs. to a whopping 20,852 lbs. As we have stated, the static load created by the non-reversing conditions does not allow a healthy oil wedge inside the connecting rod bearing to form and therefore lead to rapid bearing failure. When the pinning load is light, gradual wear occurs; however, when the pinning load is severe, the crosshead bushing will fail rapidly and often fail catastrophically.

**How Inertial Loads Can Affect The Total Rod Loads**

Calculations in the above examples do not take into account the inertial loads acting on the piston. It is possible that in light and moderate non-reversal cases, the inertial loading may be sufficient to permit adequate reversing loads for reliable operation. However, most compressor designers would never intentionally allow a compressor to operate normally with non-reversing gas loads. Reversing gas loads are vital to ensuring healthy reversing loads at the crosshead bushing.

**“ONE FAILURE FROM DISASTER”**

Reliability engineers always strive to design systems that require two or more component failures before a catastrophic failure occurs. For example, you might have a system that requires a control valve failure and relief valve to fail before a process vessel can overpressure and possibly rupture. Here’s another example: Let’s say we have a critical centrifugal pump with a control valve on its discharge line to regulate flow. If, in this example, the control valve fails closed, its failure would force the pump to operate at zero flow. A no-flow condition could eventually cause...
Advice To Consider When Choosing Single-Acting Compressors

If a compressor’s operating requirement changes to a point where the compressor needs to be configured to run a single-acting state, it is preferable to use the head end of a cylinder. Since a single-acting compressor reduces the duration of reversal, safety alarm and shutdown parameters, like discharge temperatures and pressures, should be adjusted as tightly as possible to operating conditions to allow for the early detection of valve failures.

If a single-acting compression design is chosen, the user needs to keep in mind the possible risks associated with their decision. Let’s re-examine the “CE suction valve fails and HE discharge valve fails” case listed in Table 1. If the compressor was designed to be single-acting, i.e., suction valves are removed on the head end, then a single crank end discharge valve failure would result in a rapid and catastrophic connecting rod failure. The “CE suction valve fails and HE discharge valve fails” case represents a single failure scenario where a rapid failure would likely occur. However, there are ways to mitigate the potential consequences of a non-rod reversal event in single-acting configurations.

Some guidelines proposed by Mowery to avoid non-reversal conditions:

a. Slow Speed Operation
   Low speed operation by itself is not necessarily a problem. But, with other conditions present, slow speed could be a significant contributor to a non-reversing rod load.

b. Single-Acting Operation
   Non-reversing rod loads occur in single-acting operations more than in any other situation. Single-acting head end operation (SAHE) is always more susceptible to non-reversals than single-acting crank end operation (SACE).

c. Small bore sizes in double-acting cylinders approach a single-act-
   "condition and are prone to be non-reversal.

d. Low volumetric efficiencies (VE) often produce non-reversals. Low volumetric efficiencies result from high clearances, particularly in unloading sequences where clearance is deliberately added. When performing unloading, one should always remember that SAHE is more susceptible to non-reversals than SACE. The head end pockets should be opened first to avoid the non-reversal.

e. High compression ratios are apt to produce non-reversals.

f. High cylinder pressures are a natural for non-reversing rod loads. They usually mean high gas loads, small cylinder bores, and sometimes single-acting operation – all of which are susceptible to non-reversals.

WAYS TO PROTECT YOUR COMPRESSOR

Four ways to detect valve failures and cross-head bushing failures before a catastrophic failure occurs:

1. Temperature sensing on discharge valve caps with trending capabilities can help you detect upward temperature trends due to valve failures. Perhaps you could alarm whenever any valve cap temperature reads 10°F (–12°C) higher than adjacent valve caps.

2. Discharge temperature monitoring in the cylinder discharge piping or discharge bottle can help identify valve failures.

3. Monitoring cylinder pressure differentials can help you sense abnormal inter-stage pressure conditions that could result in rod loads at unsafe levels. Note: You should use data from field performance monitoring equipment (Windrock, Reciptrap, etc.) to determine what is normal and then look for deviations.

4. There are several aftermarket methods available for monitoring the condition of cross-head bushings under full load conditions. These monitoring methodologies involve continuously sensing the actual temperature of the Babbitt in the cross-head bearing. These systems can be either mechanical in nature, where a eutectic device melts and activates a switch or a wireless system, where the temperature of the bearing is transmitted to a stationary antenna. These systems can be pricey and must be maintained to work properly.

CONCLUSIONS

Hopefully this short article has instructed you about internal gas forces and how they can affect the reliability of reciprocating compressors. Remember that the compressor designer begins the process by designing a compressor and control system that limits rod loads to safe levels under all expected operating conditions; but, it is the user/operator who must faithfully operate the compressor as it was intended and be vigilant to subtle changes in its mechanical condition. As we have seen, a well-designed compressor can experience a non-reversing condition due to a single valve failure. If everyone does their job properly and owners carefully mind their equipment, reliable and safe service will be your reward.

ACKNOWLEDGMENTS

The authors thank Bruce McCain, engineering consultant for Oxy, and Ken Atkins, director of field engineering services at Engineering Dynamics Incorporated, for their help in reviewing this paper and helping improve its content.
<table>
<thead>
<tr>
<th></th>
<th>CE normal</th>
<th>CE suction valve fails or is completely unloaded</th>
<th>CE discharge valve fails</th>
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</thead>
<tbody>
<tr>
<td>HE normal</td>
<td>Normal</td>
<td>$A_{rod} \times P_{suction}$</td>
<td>$A_{rod} \times P_{discharge}$</td>
</tr>
<tr>
<td>HE suction valve fails</td>
<td>$A_{rod} \times P_{suction}$</td>
<td>$A_{rod} \times P_{suction}$</td>
<td>$A_{CE} \times P_{discharge} - A_{HE} \times P_{suction}$</td>
</tr>
<tr>
<td>HE discharge valve fails</td>
<td>$A_{HE} \times P_{discharge}$ - $A_{CE} \times P_{suction}$</td>
<td>$A_{rod} \times P_{discharge}$</td>
<td></td>
</tr>
</tbody>
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<table>
<thead>
<tr>
<th>Rod Reversal</th>
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</tr>
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<tbody>
<tr>
<td>Complete rod reversal</td>
<td></td>
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Table 2: Examples Of Large Gas Pinning Forces (No Inertial Loads Considered)
Note: The formulas in the table approximate the minimum non-reversing load acting on the crosshead bushing during every rotation for various cylinder conditions. (The only gas piston forces are considered, i.e., inertial loads are ignored.)

REFERENCES