Vibration analysis for reciprocating compressors
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Abstract
Vibration analysis of reciprocating machines creates some unique challenges. This article explains the reasons and gives clarity on recommended monitoring and analysis practices and tools. Years of field experience have demonstrated that techniques which may be well understood for measuring and analyzing the vibration of purely rotating machinery can produce confusing results when applied to reciprocating machinery.

Vibration associated with rotational speed is the dominant motion for most industrial rotating machines. This "synchronous" (1X) behavior allows the direct application of traditional vibration analysis concepts towards addressing common machinery malfunctions – such as rotor unbalance. The typical frequencies observed with those common rotor-related malfunctions generally occur between a quarter of running speed and twice running speed and correlate excellently with machine mechanical conditions. Consequently, principles and diagnostic methodologies for these machines are broadly accepted and harmonized within the machinery diagnostic community.

This is not quite true for reciprocating compressors. Vibration analysis of these machines creates some unique challenges; many forcing functions produce a complex vibration signature that makes any attempt of using standard analysis techniques used for rotating equipment ineffective.

Compressor frame vibration
Vibration measured at the frame results principally from the response of the mechanical system to the forces and movements that are occurring in the machine at the normal running conditions. These include the following factors:

Gas load forces: These forces act on the piston and stationary components at 1X and at integer multiples of running speed. They are generally significant up to about 10X and in the direction of the piston rod travel. For large slow speed compressors (up to roughly 500 rpm), gas forces are typically the largest contributor to piston rod and compressor frame load.

Inertial load forces: These forces are caused by the acceleration of the reciprocating components (piston, piston rod, and crosshead). These components represent a large amount of mass to be accelerated back and forth with each stroke. Inertial loads of 400,000 Newton (~90,000 pounds) of force or more are not uncommon with very large compressors.

Reciprocating & rotating masses unbalance forces: These forces are predominant at 1X and 2X compressor speed, and are caused by asymmetrical crankshaft design and imperfect manufacturing tolerances. They are usually much smaller than inertial and gas load forces.

Gas unbalance forces: These are caused by pressure in the pulsation bottles and pulsation at the cylinder nozzle area and on piping. Allowable pulsation levels are defined in API-618. Although these pulsating forces are usually much smaller than the forces listed above, they can be destructive to piping and piping support systems if they happen to correspond to resonant frequencies for the structures.

As a consequence of these factors, the extent of vibration is inherent with the reciprocating compressor design and its response to all the applied forces and movements. This causes these machines, even when in good condition, to vibrate much more than a comparable rotating machine. The examples in Figures 2 and 3 shows that many harmonics are produced by the complex shape of the frame velocity waveform.

Frame vibration frequencies typically include components below 10 Hz. For this reason, a velocity transducer (with extended low frequency response) is usually better suited...
than an accelerometer for detecting an increase of rotation-related forces (due to gas load or inertial loads, imbalance, foundation looseness, excessive rod load, etc.). The preferred location for the frame vibration transducer is on the side of the frame oriented in the direction of piston rod travel, on the centerline of the crankshaft and at a main bearing where dynamic load is transmitted (Figure 1). Magnitude for a filtered frame velocity signal is usually low (less than 7 mm/s); however, at low frequencies, even small amplitudes of measured velocity may correspond to large amounts of displacement.

On the other hand, measuring only frame vibration can be insufficient for effective condition monitoring, as the increase in frame velocity from incipient failures developing at the running gear or cylinder assembly will be small and typically covered by the larger signal that is produced by normal machine movement. Experience has shown that by the time the malfunction has been detected by the frame velocity transducer and the compressor shutdown, major secondary damage may have already occurred because of the malfunctions. These malfunctions include liquid or debris carryover, loose piston or piston nut, loose crosshead nut, or loose cylinder liner, and typically manifest themselves as impacts transmitted at the crosshead.

**Monitoring vibration & impact**

Vibration transducers monitoring rotating machinery generate “stationary” signals; this means they have constant frequency content over each revolution of the rotor (Figure 4).

In contrast, vibration measurements on reciprocating compressors present both stationary and non-stationary content. In particular, the signal generated by an accelerometer placed vertically on a crosshead guide is characterized by different frequencies with different amplitudes that occur at specific points in the revolution.

Figure 5 shows a typical waveform from a crosshead accelerometer. The signal shows high amplitude, short duration impulse peaks followed by a “ring down” that occur at certain parts of each crankshaft revolution. This signal is not filtered so the transducer is picking up the widest range of frequencies (typically from 10 Hz to 30 kHz).

![Figure 4](image)
**Figure 4** Example of stationary vibration sample taken at an electric motor bearing. The higher frequency components are typical of the characteristic vibration produced by the interaction of the rolling elements with the bearing races.

These acceleration peaks can be referred to as responses to impulse events occurring during compressor operation (valve opening and closing, gas flow turbulence, crosshead pin shifting at load reversal, etc.). Such impulses excite the structural resonances of the machine components - resulting in high frequency free vibration and the characteristic impact/ring-down profile.

As mentioned, the main source of vibration on the compressor frame is related to periodic forces. While the overall frame vibration increase is certainly a concern, the primary interest of crosshead vibration monitoring is detecting peaks associated with structure response to impulsive events. Conditions that increase the excitation of such resonances are generated by developing faults such as fractured or loose components or excess clearance.

Loose rod nuts, loose bolts, excessive crosshead slipper clearance, worn pins as well as liquid in the process can be detected at early stages of development using crosshead impact monitoring, thus allowing appropriate countermeasures and avoiding potential catastrophic consequences.

Of all vibration measurements that can be applied to reciprocating compressors, crosshead acceleration is probably the most effective protection measurement available, if appropriately employed.

While crosshead acceleration has proven itself to be a sound measurement for detecting mechanical failures, industry has little experience in applying and analyzing it, resulting in increased risks of false or missed alarms, and poor diagnostic value from diagnostic systems. The following paragraphs describe some basic requirements for a reliable monitoring system and diagnostic software.

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Requirements for monitoring systems

General considerations on the effective employment of crosshead acceleration for monitoring and protection are described here:

**Transducer selection:** Amplitude measurement units should be generally selected based upon the frequencies of interest. For crosshead vibration monitoring an accelerometer should be selected as it emphasizes the higher frequency components. The unit of measurement used should be the natural units of the transducer used (signal integration is not a recommended tool for this purpose).

**Transducer mounting:** Frequency response is sensitive to mounting techniques and may be affected by any reduction of the mechanical coupling between accelerometer and mounting surface – such as the use of an adhesive, magnetic isolation base, or non-flat mounting surface. The transducer should be installed directly on the machine structural component to be measured, avoiding brackets or plates as a support, or mounting on flanges or covers. Accuracy of an accelerometer can also be affected by ground loops, base strains, and cable noise. These can be minimized by following the recommendations from transducers and monitoring systems manufacturers as well as applying appropriate cable tie-downs.

**Signal processing & alarming:** One of the concerns in applying crosshead vibration measurement for compressor shutdown is the risk of false alarms due to spurious peaks in the signal. The peak detection circuit in the protection system should be designed to manage impulsive vibration in order to avoid nuisance alarms; this can be accomplished by counting the number of readings that exceed an alarm threshold in a set time before triggering an alarm.

Additionally, an appropriate time delay needs to be configured for the alert and shutdown thresholds. Careful setting of these thresholds, counts and alarm delays will allow us to minimize the possibility of false alarms. The recip Impact/Impulse channels in the Bently Nevada® 3500/70M monitor include these features.

**Signal filtering:** Another essential aspect to carefully consider is signal filtering. As described previously, an accelerometer can detect vibration components up to very high frequencies. While acceleration analysis in a broad frequency range may have diagnostic value, the main object of crosshead impact monitoring is protecting the machine from the consequences of mechanical failures. A signal with too high corner frequency for the low-pass filter may introduce the risk of false alarms due to the presence of high frequency content signal, and for malfunctions such as loose foundation or load unbalance, this energy content relates well with machine condition, as well as operator perception of machine condition.

However, root mean square (rms) calculation applied to an impulsive frequency-rich operator signal such as crosshead vibration (Figure 5) does a poor job in correlating with other critical conditions such as mechanical knocks, which have relatively little energy content, but proves analysis provides little value due to the discontinuous frequencies involved.

The most appropriate analytic methodology is therefore based on signal timing: Bently Nevada 3500 synchronizes the vibration signal with crankshaft rotation to associate peaks to a piston position along the stroke. Individual monitoring and alarming on crank angle "bands" allows association of peaks to the problem area.

**Amplitude measurement**

Our last important note is about vibration measurements taken in either rms, zero-to-peak (peak or pk), or peak-to-peak (pp) amplitude measurement systems. A few international standards recommend rms measurement for assessing machinery health based on overall casing vibration and this is traditionally adopted by many practitioners. Rms values provide an indication of the energy content of a vital in assessing machine condition. For these types of malfunctions, peak-amplitude measurement is recommended as it correlates well with both high-energy and low-energy malfunctions typical of reciprocating compressors. Applying rms processing to crosshead vibration signals would provide under-predicting values.

**Crank angle domain analysis**

When viewed in the time domain, the non-stationary crosshead vibration signal looks like multiple disconnected events (Figure 5), so diagnostic methodologies such as spectral or example, a peak occurring when the piston is travelling toward the end of its stroke near Top Dead Center (TDC) can be correlated to liquid or debris ingestion in the compression chamber. When the piston moves towards its TDC position, the impact with the non-compressible material will generate an impulse event. The monitoring system will then raise an alarm for the corresponding crank angle band (for example, starting 10 degrees before TDC and ending 10 degrees after). Figure 6 shows case of liquid ingestion as detected by the crosshead guide accelerometer.

**Understanding frequency content**

Additional advanced analysis tools are available in System 1® diagnostic software. As noted before, not all impulse response events within the crosshead accelerometer signal contain the same frequencies. Mechanical knocks excite resonances of the reciprocating compressor components such as crosshead guides, distance pieces, etc., that generally lie below 2 kHz. In contrast, events originating in gas flow noise, valve opening or valve closing events express a much higher frequency.

Searching for a mechanical event in an acceleration signal that contains the whole transducer frequency response range is practically impossible due to the high amplitude and frequency peaks that cover smaller, yet more critical, peaks related to mechanical events. Such overlap prevents early indication of an incipient malfunction. It is for this reason the signal must be filtered. Figure 7 shows crosshead acceleration in the crank angle domain using 3 to 30 kHz (left plot) and 3 to 2 kHz (right plot) band pass filtering. The peaks present in the narrower pass-band correspond to mechanical impacts, which are difficult to distinguish in the signal with broader filter corners.

System 1 software is integrated with the 3500/70M monitor to allow dual signal processing and both storing and displaying the accelerometer signal with two different filter settings.
Figure 6
Crosshead acceleration in crank angle domain, presenting a high peak at Top Dead Center (TDC). The horizontal axis represents 360 degrees of crankshaft rotation (one full revolution), where 0° indicates TDC. The System 1 plot also displays a Throw Animation in the upper right corner of this plot showing the piston movement synchronized with the plot cursor. In this example the cursor is set at 2.5 degrees, and the animation shows that the piston is very close to the TDC position.

Figure 7
The 3500/70M module returns two waveform samples to System 1 software from a single crosshead acceleration signal with two different filtering characteristics.

Diagnostic approach
To wrap up, let us consider how we can effectively associate a malfunction to a specific vibration pattern and to obtain an early failure diagnostic. Experience has shown that associating vibration with additional measured dynamic parameters such as rod load have proven to be of great value in pinpointing a specific component failure.

Due to the complexity of the signal content and the vibration signatures that differ from case to case based on operating conditions and failure modes, several different automated diagnostic approaches have been developed. This includes rule-based and model-based approaches that are driven by data or by “first principles” of Physics relationships.

References
1. GE Energy Brochure, Condition Monitoring Solutions for Reciprocating Compressors, GEA-14927

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