

SLOW-SPEED

Vibration Signal Analysis

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Good coaches often require their athletes to execute skills in slow motion, knowing that only when the individuals have mastered the skills at slow speeds will they be able to execute at normal speeds—not to mention under high-pressure game situations. This also holds true for vibration diagnostics: an inability to correctly analyze slow-speed data often translates to misleading or wrong interpretation of full-speed and transient (startup and coastdown) data. Unfortunately, because data collected at slow-roll speeds is all-too-often erroneously viewed as nothing more than “noise,” this data ends up being ignored and/or discredited as useless.

This article provides a tutorial on so-called “slow-roll” data—the data collected from proximity probes observing a rotating shaft when the rotational speeds are too slow to contribute any meaningful rotordynamic effects in the signal. In addition to electrical and mechanical shaft surface imperfections (known as runout) such as scratches and residual magnetism, certain machinery problems, such as misalignment and bows, manifest themselves at slow-roll speeds, allowing them to be distinguished from other malfunctions that yield nearly identical data at full-speed conditions. Also, it is only by understanding the behavior of the machine when it is not subject to the dynamics of rotational acceleration, deceleration, process con-

ditions, and rated-speed operation that the effects of these forces can be properly understood. As a result, slow-roll data is vital to proper machinery diagnostics and proper machinery rotordynamic understanding.

Practical information will be provided in the pages that follow for using and interpreting this valuable data, along with guidance on the sometimes-difficult task of correlating and corroborating slow-roll data with high-speed data. Several case history examples are also provided, showing how the author and his colleagues have used slow-speed signal analysis techniques to isolate specific conditions.

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NOMENCLATURE

Slow-Roll

Data collected from proximity probes while the machine is rotating at speeds for which rotor-dynamic effects are negligible.

Slow-Roll Speeds

Rotational speeds at which no significant dynamic forces act on the machine, such as unbalance forces or other forces generated by the rotor or processes around the rotor. A rule of thumb is that slow-roll speeds are typically 5% or less of the full operating speed of the machine, or its first balance resonance if it operates above this resonance.

1X Compensation

The slow-roll waveform may be filtered to contain only its 1X (rotational speed) component, and then represented as a vector with a characteristic magnitude and phase lag. When this vector is subtracted from the high-speed vibration data from that same probe, it is known as 1X compensation.

Waveform Compensation

Compensation of the high-speed vibration waveform whereby the entire slow-roll waveform is subtracted rather than merely the 1X vector. When available, this is a preferred method of compensation because it utilizes the entire runout signature rather than just the (usually dominant) 1X aspect of the signature.

Selecting Slow-Roll Values

Collection of slow-roll data requires a system that can capture data at very low frequencies, typically 300 rpm or lower. Fortunately, this is rarely a problem with many data acquisition systems designed to handle proximity probes. When possible, a range of slow-roll speeds should be identified, which is characterized as exhibiting no significant amplitude or phase changes within the upper and lower speeds. Once this range has been identified, any convenient speed within this range can be used for collection of slow-roll data. The turning gear speed, for example, is often convenient for large steam turbines.

Figure 1 shows a Bode plot with the cursor positioned at 1590 rpm. It exhibits a nearly unchanging amplitude response below this speed. Although the phase values for the samples in this range are not particularly stable, this is not the result of actual changing phase, but rather the inability of the system to trigger on extremely low vibration amplitudes and provide accurate phase information. Generally, this effect will only be pronounced when the amplitudes are less than 10% of the full-scale range of the data acquisition system. It can be readily seen that the phase variations are centered at approximately 80 degrees, and a slow-roll 1X phase value can often be visually identified quite easily in this manner. It is also noteworthy that the direct (unfiltered) amplitude is considerably elevated compared to the filtered 1X amplitude (compare the top trace with the bottom trace in the amplitude portion of the plot). This is due to shaft surface irregularities that will be discussed later in this article.

In contrast to Figure 1, the dynamic response of some rotor systems will often become apparent almost immediately upon application of increasing speed as the forces interact with the rotor system (Figure 2). In this case, it is virtually impossible to select a slow-roll sample that is not being influenced by dynamic forces. In these situations, select the lowest speed where amplitude and phase readings are stable. In Figure 2, this corresponds to approximately 600 rpm where the amplitude is about 1.5 μm and the phase is about 80 degrees.

Figure 1 – Bode plot slow-roll selection under stable conditions.

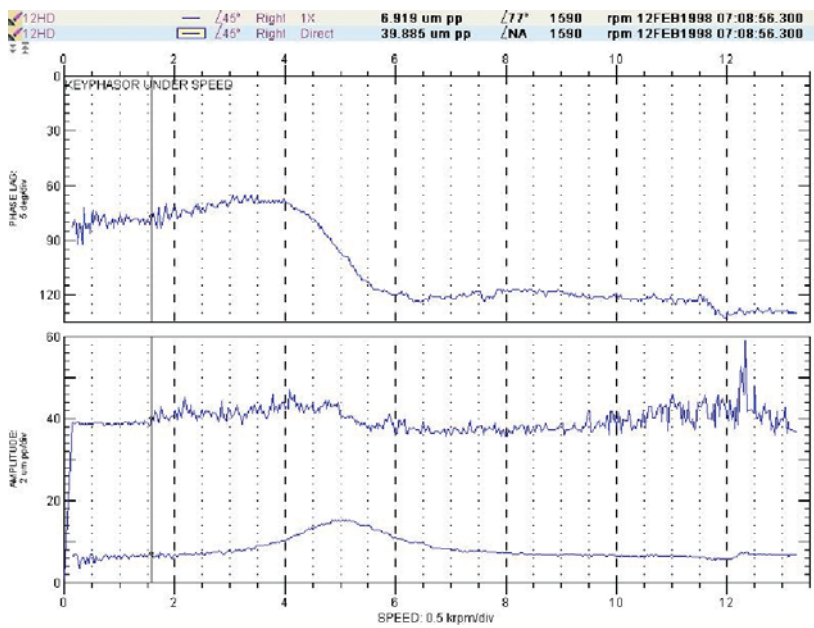
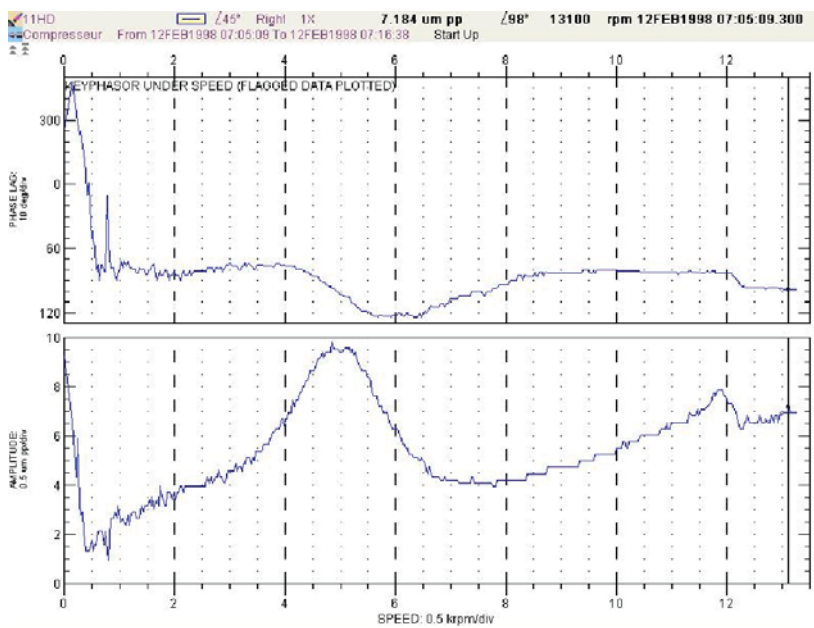


Figure 2 – Bode plot slow-roll selection under changing conditions.



Shaft Surface Quality

API 670 defines the acceptable shaft surface quality as indicated by proximity probes to be less than 6.35 $\mu\text{m pp}$ (0.25 mils). These levels represent a trend over the last several years to lower the acceptable vibration levels of machinery. As the allowable vibration levels have decreased, so have the allowable levels of runout so that the ratio between maximum allowable vibration and runout will be maintained. This corresponds to fewer defects on the shafts and can represent significant challenges for machinery manufacturers and repair shops.

[Editor's Note: A companion article dealing with measurement and reduction of runout is provided on page 36 in this issue of ORBIT.]

The quality of the shaft surface can be evaluated by observing the orbit/timebase data at slow speeds. From this data presentation it is relatively easy to view the shaft irregularities, which are a mixture of shaft bow, non-circularity, scratches, burrs, and electrical runout. The shaft surface depicted in Figure 3 shows very little 1X data but is a good example of what scratches and burrs look like.

A scratch or burr can be identified by a repetition of the pattern in the signal on each of the orthogonal probes. If the rotation is clockwise, or Y-to-X, then the pattern will start on the Y-probe and then be repeated 90 degrees later on the X-probe. If the rotation is counterclockwise, or X-to-Y, then the pattern will be the reverse of the Y-to-X situation. The primary difference between a scratch and a burr is that a burr results in a positive-going pulse (towards the probe) while a scratch results in a negative-going pulse (away from the probe) as shown in Figures 3 and 4 respectively. In Figure 4 it is particularly obvious that the negative-going pulse first appears on the X-probe and then 90 degrees later on the Y-probe, exactly as would be expected for X-to-Y rotation as noted at the bottom of the plot.

“A SCRATCH OR BURR CAN BE IDENTIFIED BY A REPETITION OF THE PATTERN IN THE SIGNAL ON EACH OF THE ORTHOGONAL PROBES...THE PRIMARY DIFFERENCE BETWEEN A SCRATCH AND A BURR IS THAT A BURR RESULTS IN A POSITIVE-GOING PULSE...WHILE A SCRATCH RESULTS IN A NEGATIVE-GOING PULSE.”

When the signal is converted to the frequency domain via Fast Fourier Transform (FFT) processing, it can often end up rendering a scratch or burr indistinguishable from other malfunctions, resulting in misdiagnosis (Figure 5). Thus, a timebase representation will often be a better method of identifying a scratch or burr, rather than a single spectrum. Looking only at the spectrum can easily lead to a misdiagnosis where a more serious problem is suspected, but is in fact only a scratch or burr.

However, spectrum presentations do have a role in diagnosing a scratch or burr, particularly when ambiguity is present in the waveform. This is done by inspecting the cascade spectrum data as the machine goes through a startup or coastdown. Scratches and burrs will exhibit fairly constant amplitude harmonics of shaft rotational frequency throughout the speed range of the machine. This is illustrated in Figure 6.

“LOOKING ONLY AT THE SPECTRUM CAN EASILY LEAD TO A MISDIAGNOSIS WHERE A MORE SERIOUS PROBLEM IS SUSPECTED, BUT IS IN FACT ONLY A SCRATCH OR BURR.”

Figure 3 – Orbit/timebase plot at slow-roll speeds exhibiting a burr on the shaft surface, as noted by the positive-going pulse (towards the probe).

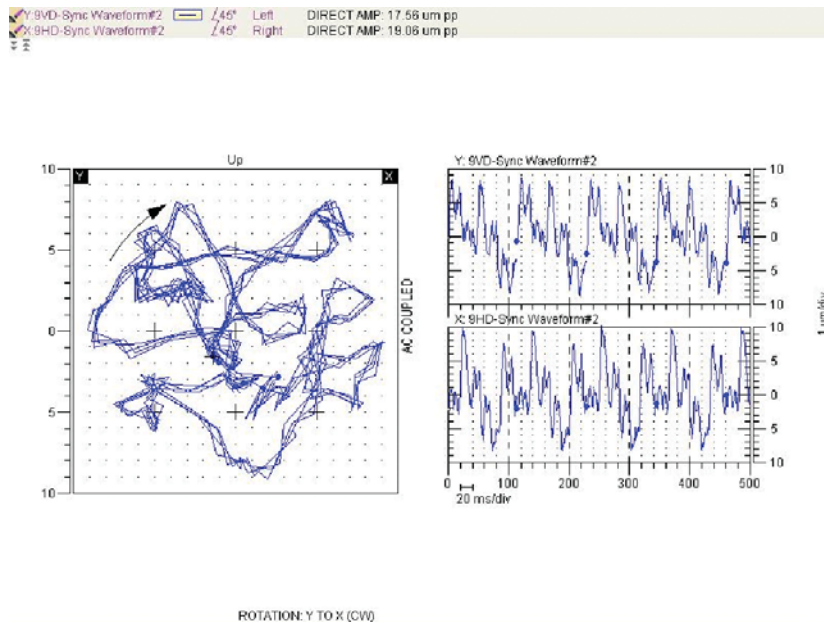


Figure 4 – Orbit/timebase plot at slow-roll speeds exhibiting a scratch on the shaft surface, as noted by the negative-going pulse (away from the probe).

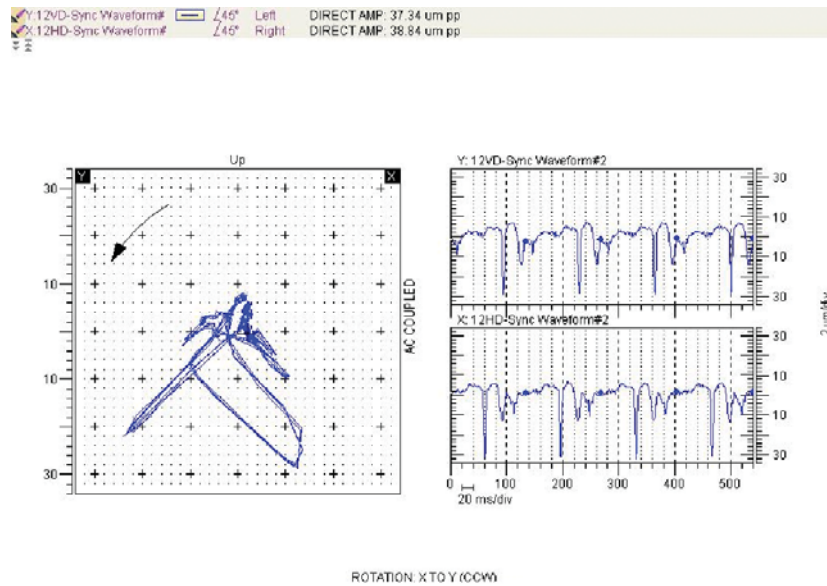


Figure 5 – A shaft scratch appears as multiple harmonics in a spectrum presentation, making it indistinguishable from other malfunctions such as periodic impacting events that yield many harmonics.

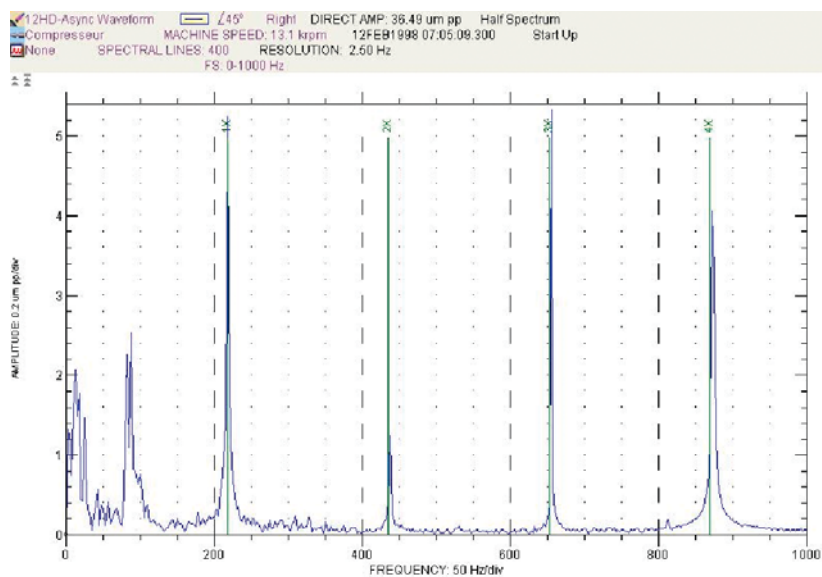
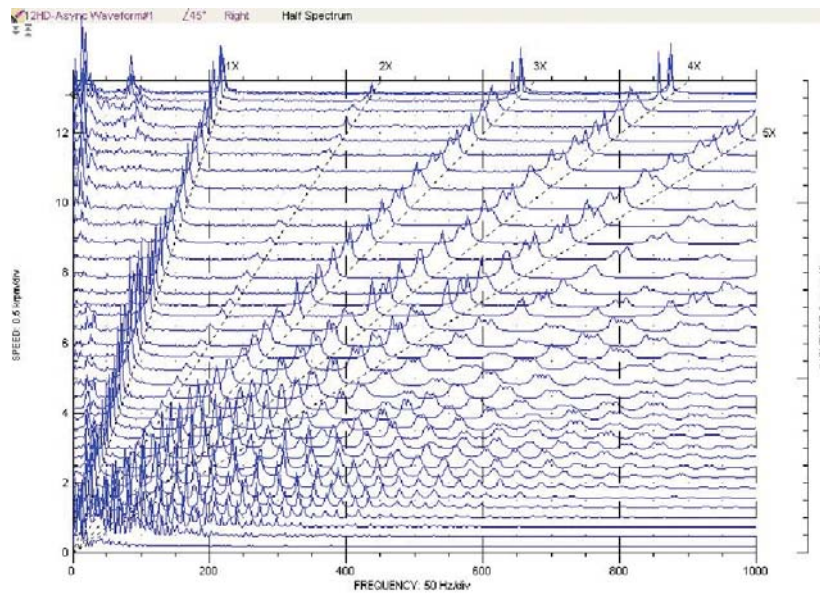


Figure 6 – A spectrum cascade showing evidence of a shaft scratch. A scratch yields harmonics of nearly constant amplitudes at all shaft speeds, since its amplitude is independent of shaft dynamic motion.



Compensation

Because vibration data can be affected by shaft imperfections, it is important to ensure that these imperfections are suitably small. This will prevent them from resulting in false indications of a machine malfunction in either the monitoring or diagnostic system.

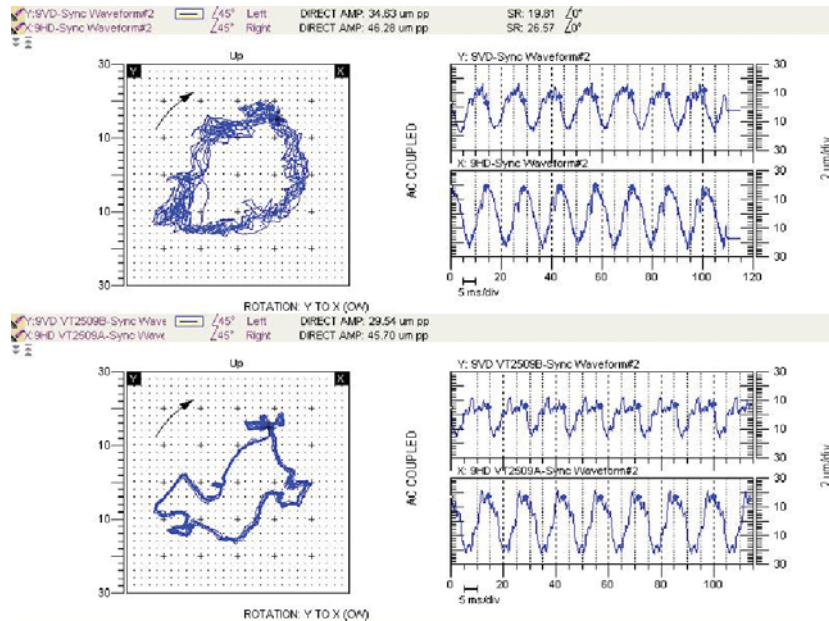
One way to reduce the effect of shaft imperfections is to use compensation, whereby either the 1X slow-roll vector or the entire slow-roll waveform is subtracted from the non-slow-roll data, such as a waveform obtained at full running speed. An example of such compensation is shown in Figure 7. Here, a reference slow-roll waveform has been used to compensate data at full speed by subtracting the slow-roll waveform from the full-speed waveform. The dynamic sample compensation allows the engineer to look at the actual vibration, rather than the “apparent” vibration corrupted by slow-roll noise.

Often, people ask why such care must be taken to reduce shaft runout at the manufacturing stage instead of simply using waveform or vector compensation in

the measuring instrument. The answer is quite simple: runout can and does change. When compensation is used in permanent monitoring situations, rather than when examining a specific data set for which the slow-roll data has been computed, it can lead to erroneous results. Compensation of any kind is considered poor engineering practice in permanent monitoring systems, and is indeed prohibited in API 670. There are two primary reasons why compensation is strongly discouraged for permanent monitoring:

1. If a shaft has a tendency to grow or move axially with respect to the probes, it is possible that the shaft surface observed by the radial probes may move from underneath the probes at slow speed to a new position at full speed, resulting in an entirely different runout signal. The compensated signal will be erroneous and may actually be much higher or lower than the actual vibration, resulting in a false alarm or missed alarm.

Figure 7 – Compensated orbit/timebase data (top) compared with uncompensated data (bottom). The uncompensated data on the bottom could lead to erroneous diagnoses as it appears to show many locations where impacting is occurring, such as caused by a heavy rub.



- Even if the relative position between probe and shaft does not move axially, electrical runout will often change over time, resulting in an entirely different slow-roll signal. Compensation with an erroneous signal will cause the same problems as previously noted in #1.

Compensation is therefore recommended only in situations where machinery diagnostics are being conducted (not machinery protection), and a specific set of slow-roll data can be validated before use to compensate corresponding high-speed data.

Diagnostic Uses – Direction of Rotation

Reverse precession is defined as vibration direction opposite to shaft rotation direction. The stress that results from reverse precession can fatigue the shaft very quickly since this causes reverse cyclic stresses for each shaft revolution. It is therefore important to verify the shaft rotation direction of the machine and compare

it to the shaft precession direction. The shaft rotation direction can be verified using a variety of methods. The most common is to look at the design of the machine. The direction of rotation is often stamped on the machine or contained within the design documentation. However, regardless of what the documentation says, it is important to physically verify this on the machine.

One method of physical verification that has been employed historically (but is discouraged today for obvious safety reasons) is to touch a pencil eraser to the exposed shaft surface and feel the direction in which the frictional force acts. A considerably safer method—and one that does not require direct physical contact with the shaft—is the use of an orbit/timebase plot where shaft imperfections can identify the direction of rotation.

A scratch on a rotating shaft will always pass under each of the probes in the same sequence as the rotation, regardless of vibration precession direction.

Figure 8 – Orbit/timebase plot from same bearing as Figure 4, but at full-speed conditions. The shaft scratch passes under the X-probe (bottom plot) 90 degrees before passing under the Y-probe, confirming X-to-Y or counterclockwise rotation.

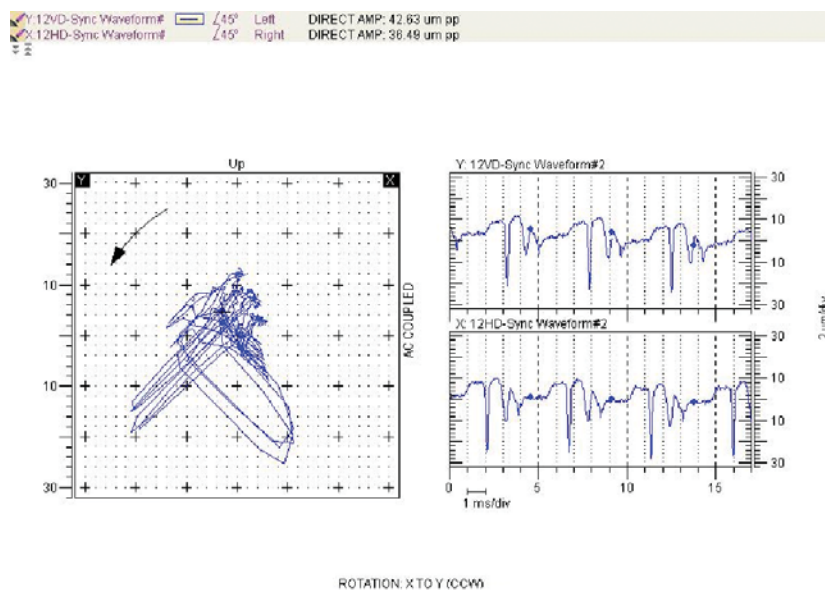


Figure 8 shows the same bearing and probe location as Figure 4 except the orbit of Figure 4 was collected under slow-roll speeds and the orbit of Figure 8 was collected at full speed. Both orbit/timebase plots show that the scratches pass underneath the X probe (bottom timebase) 90 degrees before the Y probe (top timebase). This confirms that the rotation is X-to-Y (counterclockwise).

One note of caution is in order, however: it is very important to confirm that the field wiring and software configuration is correct when relying upon this method. It is very common in the field to find systems where the X and Y probes have been inadvertently cross-wired. A loop check should be performed if in doubt by physically connecting/disconnecting the wiring from the X and Y probes (while bypassing the machinery protection system, of course) and watching the respective channels on the monitoring system and software screens.

In general, the direction of precession of most vibration signals is in the same direction as rotation, unless a serious malfunction is present. Consequently, if a machine is operating normally, the direction of vibration

precession is an indication of shaft rotational direction and vice-versa. Additionally, although reverse vibration precession is rare, its occurrence at slow-roll speeds is even more rare. As such, the direction of precession at slow-roll speeds can almost always be used to establish the direction of rotation.

Diagnostic Uses – Reverse Rotation

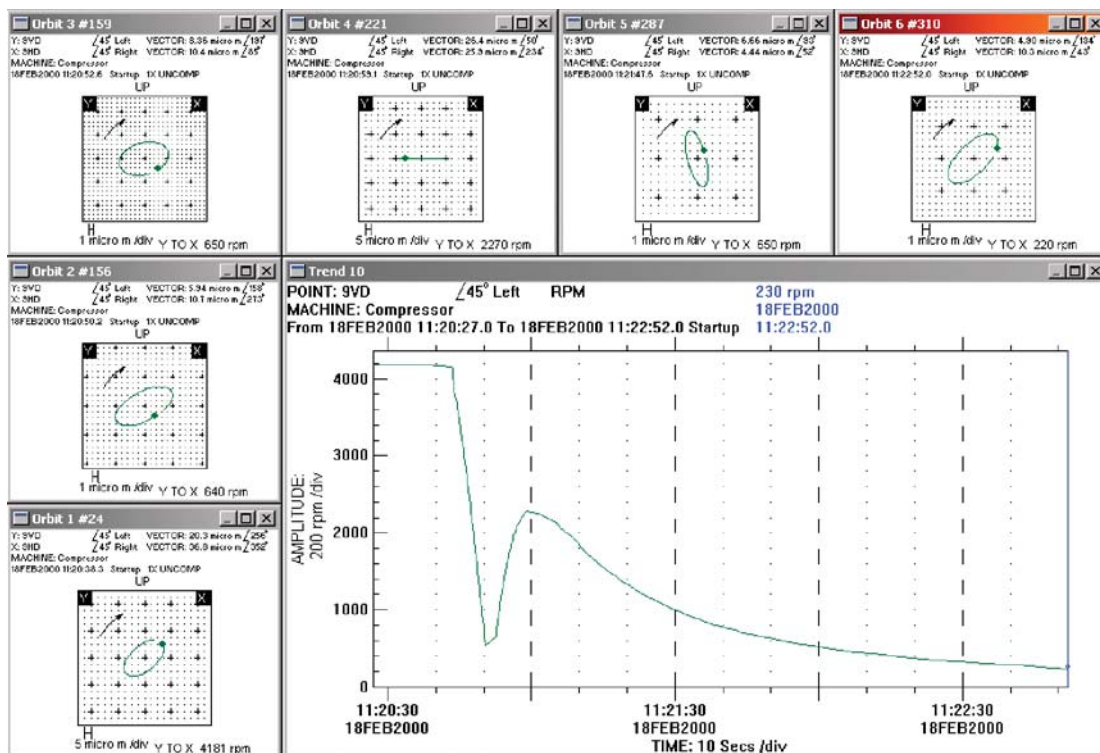
In the last section, we stated that the direction of shaft precession is almost always a reliable indicator of shaft rotational direction—particularly at slow-roll speeds—as long as the machine is not in a malfunction condition and as long as it has been confirmed that the X- and Y-probe wiring has not been reversed. By definition, at slow-roll speeds the rotodynamic forces that occur at higher speeds and cause a shaft to vibrate in any direction (whether opposite to the direction of rotation or in the same direction) are not at work. Thus, what appears to be reverse precession at slow-roll speeds will almost always be either: A) reverse rotation or B) cross-wired probes.

In the author's experience, comparing slow-roll data from run to run is very important as it can yield important clues regarding malfunctions. In the next example, we show how slow-roll data collected during an unusual coastdown was markedly different than previously collected coastdown data, and how the shaft behavior at slow-roll speeds (in this case, less than 300 rpm) provided important clues as to the nature of the problem.

Figure 9 shows a machine exhibiting forward precession at full speed in orbit plots 1 and 2. Notice, however, that the blank-bright sequence in orbits 1 and 2 becomes bright-blank (reversed) in orbits 3–6. This indicates one of two things: Either the vibration direction changed

from forward to reverse, or the shaft rotation direction changed from forward to reverse. Examination of the trend plot in Figure 9 reveals what happened. As the machine coasted down, upon reaching approximately 600 rpm the unit abruptly reaccelerated to 2270 rpm and then coasted down once again, but more gradually. All of the orbits during this "re-accelerated" coastdown (orbits 3–6) exhibit what appears to be reverse precession, even at very slow speeds (below 250 rpm in orbit 6). In reality, this was not reverse precession; it was reverse rotation. Thus, the vibration precession was always forward (i.e., remained in the same direction as rotation), but swapped because the machine started rotating in the opposite direction.

Figure 9 – What appears to be reverse precession in orbit plots 3–6 is actually forward precession because the shaft was rotating in the reverse direction.



It was discovered that this machine, which consists of two turbine casings and three compressor casings, had a problem with the discharge check valve. Typically on compressors, a check valve is installed to constrain flow to a single direction, preventing backflow of the process gas. In this case, however, the machine was shut down and the check valve was not operable. As the unit coasted down, the backflow pressure caused the compressors to accelerate in the reverse direction until this excess pressure was dissipated, at which time the machine began to decelerate again while exhibiting a familiar logarithmic speed decay.

Reverse rotation can represent a severe safety risk in some situations, as noted below:

1. Many times in the process industries, the fluids being compressed are flammable or hazardous. If the high-pressure gas goes to places where it is not designed (primarily upstream of the compressor stages), the sealing and other mechanical devices may fail under the high pressures or reverse-flow dynamics of the process gas.
2. The braking action may cause reverse stresses on the couplings, seals, and other mechanical components which may introduce premature fatigue or even outright yield failures.
3. The machine may overspeed in the reverse direction, resulting in catastrophic failure such as blade liberation and case breaches. Steam turbines and gas turbines have overspeed shutdown systems that shut off the energy source (either steam or fuel gas). In the event of a reverse rotation where a check valve fails, the entrained energy source cannot be stopped, and the machine will continue to accelerate in the reverse direction until the energy is dissipated or the machine yields—whichever comes first.

To help identify and prevent reverse rotation on machines that are susceptible to such malfunctions, specialized reverse rotation monitors are available. In situations where they cannot prevent the reverse rotation from occurring, they at least flag the event so that operators will not inadvertently restart a machine that

may have incurred damaged seals or other components during the reverse rotation excursion.

Diagnostic Uses – Rotor Bow (Gravity)

Many large machines have extremely heavy shafts and the spans between bearings can be quite long, leaving the unsupported shaft to sag or bow under its own weight. To counteract this tendency for the shaft to bow when at rest, turning gears are employed that slowly rotate the shaft rather than allowing it to remain at rest when the machine is out of service.

If a rotor becomes bowed while it is at rest, it may become stuck in its internal seals and consequently prevented from starting up. More commonly, however, induced bow will allow the machine to start, but the bow will behave like a very large imbalance. As the shaft approaches its first balance resonance, this residual bow can lead to extremely large vibration deflections, particularly at the shaft's mid-span, and seals that are normally required to have small clearances to ensure efficient operation can be wiped.

Figure 10 shows the Bode plot for the startup of a large industrial gas turbine with a gravity bow. The turbine starts with approximately 50 μm pp of slow-roll levels. Once the machine starts to reach its balance resonance speed range, the unbalance due to the gravity bow causes the machine to vibrate excessively. It is very likely the machine started to rub at approximately 910 rpm as indicated by the abrupt change in amplitude progression. In this instance, the vibration became so severe that the unit tripped on high vibration and the coastdown data is noted by the nearly identical traces in Figure 10 (one trace shows the startup, the other shows the coastdown).

The data acquisition system was used to record the bow of the turbine after it shut down. This is shown in Figure 11a as a gap voltage trend plot while the machine was rotating on turning gear at approximately 3 rpm. The pp amplitude of this plot is approximately 340 mV, corresponding to 1.7 mils at 200 mV/mil (or approximately 43 μm). Almost all of this was due to residual shaft bow and not electrical/mechanical shaft surface imperfections (i.e., runout).

To relieve the bow, the machine was left on turning gear overnight, and the slow roll readings (Figure 11b) reduced to approximately one-third of their previous values. The subsequent startup was successful as vibration did not exceed 80 μm pp when the machine ran through its first balance resonance. Compare this to Figure 10 where the vibration amplitude reached 300 μm pp before tripping the unit.

For large machines that are susceptible to gravity bows, eccentricity monitors are available that measure the slow-roll amplitude and interpret it as residual bow in the machine. When the amplitude exceeds a maximum allowable value, the monitor provides an alarm signal that is used by the control system. This signal is treated as a startup permissive (i.e., when the alarm is present, the machine is not allowed to start.)

Machines that expose their rotors to significant heating are even more susceptible to gravity bow since the yield strength of the rotor material will decrease with elevated temperatures. These machines include industrial gas turbines and many steam turbines.

Figure 10 – Bode plot of gas turbine starting up with gravity bow. Slow-roll amplitude is approximately 50 μm pp.

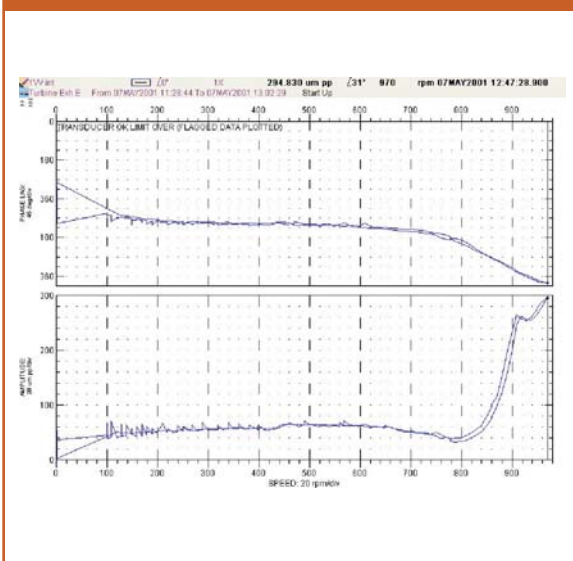
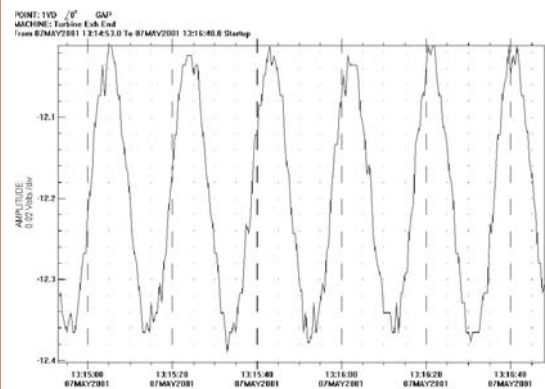
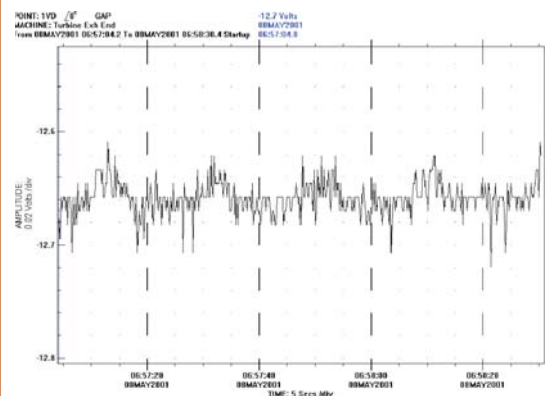


Figure 11 – Gap voltage trends with machine on turning gear at approximately 3 rpm. Top plot (a) shows levels were initially more than 40 μm . Bottom plot (b) shows levels had reduced to approximately 13 μm after running the unit on turning gear overnight to relieve residual gravity bow.

(a)



(b)



[Editor's Note: The original ASME paper contained three additional case histories dealing with locked couplings and thermal rotor bow. Space constraints did not permit them to be included in this reprint article. If the additional case histories are desired, a copy of the original paper is available upon request from the author.]

Conclusions

Slow-roll data is the signal provided from a proximity probe when the shaft is rotating so slowly that negligible rotordynamic forces are acting on the shaft. Although the signal consists of electrical and mechanical runout, it should never be dismissed as noise because certain malfunctions can be confirmed or eliminated using slow-roll data as they will be evident at slow speeds as well as high speeds. Examples including direction of rotation, reverse rotation, and shaft bows were discussed at length, showing how slow-roll data can be used when diagnosing these situations.

Compensation by subtracting slow-roll vectors or waveforms from high-speed data was also discussed as a useful technique for providing diagnostic clarity. However, it was shown that

compensation should be used only when conducting diagnostics, never in machinery protection schemes, and the reasons for this were outlined. Instead, mechanical and electrical runout levels should be reduced at the factory or repair facility to the lowest levels practical rather than relying upon electronic compensation in the measurement system.

Slow-roll data collection and use were shown to be straightforward and based on science, not art. Further, only when slow-roll data is correctly acquired and interpreted can corresponding high-speed data be interpreted with confidence and malfunctions correctly diagnosed. **O**

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