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SLOW SPEED VIBRATION SIGNAL ANALYSIS: IF YOU CAN'T DO IT SLOW, YOU CAN'T DO IT FAST

Mel G. Maalouf P.E.
GE Energy, Minden, Nevada,
United States of America

ABSTRACT

For many people, the interpretation of vibration signals for a machine at running speed is complicated and foreign, and is considered an art in many circles. Interpreting the rich characteristics of the raw signals during run-up and coast down requires even more skill and experience. For some, interpreting the signals at slow speeds (sometimes called slow roll speeds) is so difficult that the signals are often ignored and discredited as not useful data. This paper will communicate the author's experience in using this valuable, yet sometimes difficult, data to correlate and corroborate with high-speed data. This data and interpretation are used to understand the dynamic behavior of the machine while the forces on the rotor are driving the response characteristics at run up, full speed and coast down. In the sports arena, good coaches often say that if you cannot execute skills in slow motion, you likely won't be able to execute them at normal speeds and absolutely not in high-pressure game situations. This is also true for vibration diagnostics: if you don't do correct slow speed analysis, full speed and transient (startup and coast down) analysis may be misleading or just wrong. In this instance, the analyses and diagnostic calls that were made by using slow speed signal analysis include:

Selecting Slow Roll Values
Shaft Surface Quality
Direction of Rotation
Rotor Bow (Gravity)
Rotor Bow (Thermal)
Locked up Coupling
Non-Concentric Coupling
Reverse Rotation

This paper will describe the methodologies for collecting data and the analysis of the data to make the above calls on specific examples experienced by the author and his colleagues.

INTRODUCTION

The methods used in this paper are for interpreting data from proximity probes on rotating machinery at slow speeds. The data used for vibration analysis can be difficult to interpret and it can be made even more difficult or impossible to interpret without the right correction of the data for shaft surface defects, bows, and other anomalies that may present themselves under the view of the probes. This data can be used to correct the full speed data to better interpret the dynamics of, and forces on, the machine. It is additionally helpful in determining the mechanical condition using some unique techniques of interpreting the data collected while there are no forces or dynamics normally associated with high horsepower and high-speed machines.

NOMENCLATURE

Slow Roll: Used interchangeably with glitch and runout. This is the data collected at slow speeds of the machine with proximity probes while there are no significant dynamic forces on the machine.

Slow Roll Speeds: Speeds of machinery defined as rotational speeds at which there are no significant dynamic forces on the machine, such as unbalance forces or other forces generated by the rotor or processes around the rotor. Slow roll speeds are typically 10% or less of the full operating speed of the machine or the first balance resonance.

1X Compensation: Usually used for compensating the vibration signals for problems with the shaft surfaces, such as scratches, out of roundness of rotors, stray magnetism, and other sources of errors to which proximity probes are susceptible. This is normally done with the familiar 1X (Filtered at 1 times shaft rotational speed) vector data, which is comprised of an amplitude and phase.

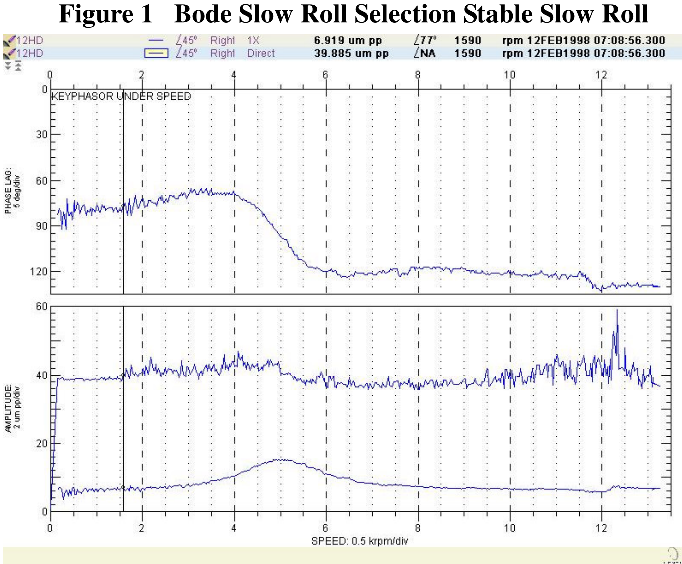
Waveform Compensation: This is a more sophisticated compensation than 1X. It entails taking a digital waveform of the proximity probe signal with reference to a repeatable once per turn signal, sometimes referred to as a Keyphasor® signal. This is effective in eliminating defects such as scratches, burrs, and other shaft surface abnormalities that may get filtered out from the 1X compensation.

Selecting Slow Roll Values

Many of the currently available data acquisition systems allow for the collection of data of high- speed machinery as they pass through startup and coast down procedures. It is very important to have a system that allows for collection of data at slow speeds. Many of the latest systems can take data as low as 1 rpm or lower, so it is not an issue to take slow roll data with modern systems.

If it is possible, a range of slow roll speeds should be identified, which are characterized by no significant amplitude or phase changes in the data over a range of speeds in the slow roll speed range. If this is the case, then any sample can be selected within this range. This is shown in **Figure 1**, which shows a stable speed range up to approximately 1500 rpm, which consequently is roughly 10% of the full speed of the machine: 13,000 rpm. We see that the phase values for the samples in this range are not particularly stable, but this is of little consequence since the 1X filtered value for the slow roll readings are below 10 µm pp. The cause of these instable phase readings is usually related to low amplitudes as compared to the full-scale ranges of the data acquisition setup. The phase accuracy of the systems is lower when the amplitudes are typically less than 10% of the full-scale range of the systems. It is also noteworthy that the direct or overall values shown in the data is quite elevated as compared to the 1X data.

Figure 1 Bode Slow Roll Selection Stable Slow Roll



This is due to shaft surface irregularities that will be discussed later in this paper.

Often the dynamic response of the rotor system begins to react to the forces on the rotor immediately on changing of speeds in the range of concern. It is sufficient to select one of the speeds at the lowest speed where the readings are stable, but possibly changing with respect to speed as shown in **Figure 2**. In this case, it is virtually impossible to select a slow roll sample that is not being influenced by dynamic forces. There are two main causes for the changing indications of the data:

1. The amplitudes are very low, ~3 µm pp in this case, and are influenced by very small dynamic forces on the rotor system.
2. The forces on the system are significant at low speeds and are immediately causing the rotor system to respond noticeably at slow speeds.

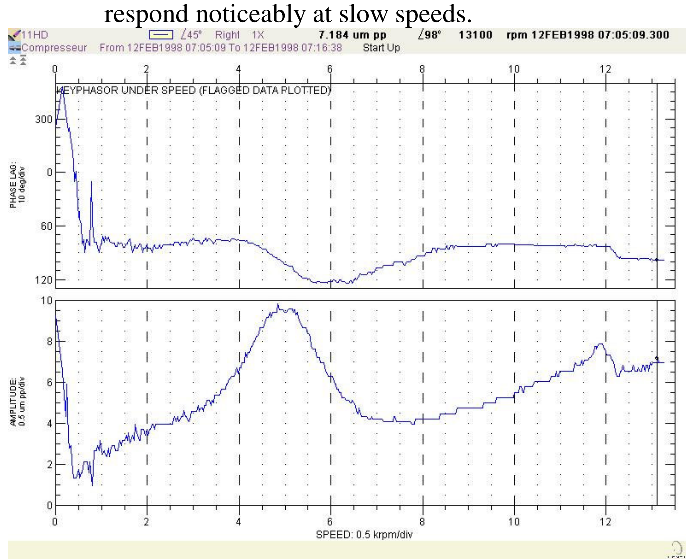


Figure 2 Bode Slow Roll Selection Changing Conditions

Shaft Surface Quality

API 670 has defined the acceptable shaft surface quality as indicated by proximity probes to be less than 6.35 µm pp. These levels have been reduced in recent years since the acceptable vibration levels of machinery has been reduced, so it is therefore necessary to have fewer defects on the shafts to be able to properly monitor the lower vibration levels of the machines.

The quality of the shaft surface can be evaluated by observing the orbit-timebase data at slow speeds. From this data presentation, it is somewhat easy to view the shaft irregularities, which is a mixture of shaft bow, non-circularity, scratches and burrs. The shaft surface depicted in **Figure 3** shows very little 1X data but is a good example of what scratches and burrs would look like. A scratch and a 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. A shaft with a burr will have a spike in the direction of the probe, since proximity probes respond to a positive going signal for a surface that would approach the tip of the probe. A brief approach of the surface to the probe tip, such as a burr will appear to be a positive going impulse, as is shown in **Figure 3,** with two peaks right after the Keyphasor mark on the 9VD probe. It is evident that the surface for this shaft is in poor condition. There are many scratches and burrs that would distort the readings at full speed.

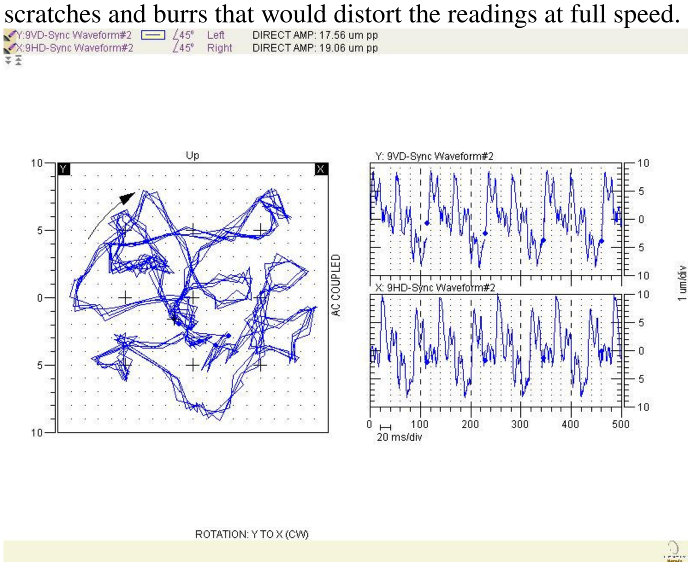
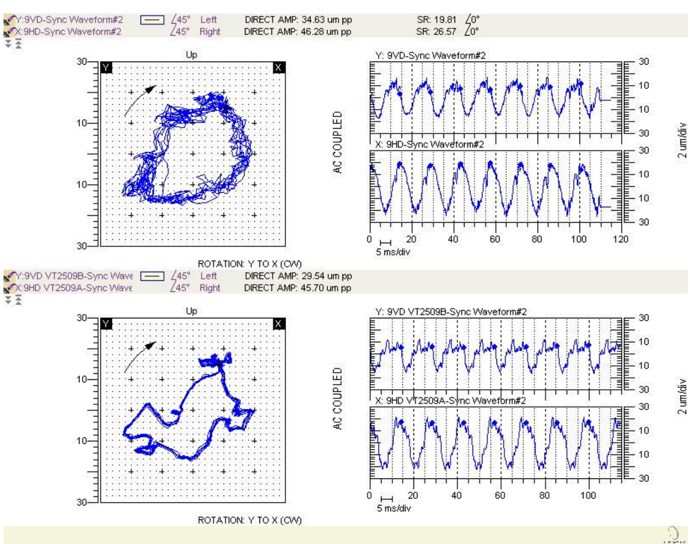


Figure 3 Orbit-Timebase Shaft Surface Evaluation

Once a reference waveform sample is selected, it can be used to compensate data at full speed. An example of compensating full speed data is shown in Figure 4. The dynamic sample compensation allows the engineer to look at the real dynamic data. In this case, the burrs and scratches make the uncompensated data look like there may be erratic movement of the shaft that could be misinterpreted as impacts or other mechanical influences on the shaft orbit.

Figure 4 Compensated and Uncompensated Orbit-Timebase



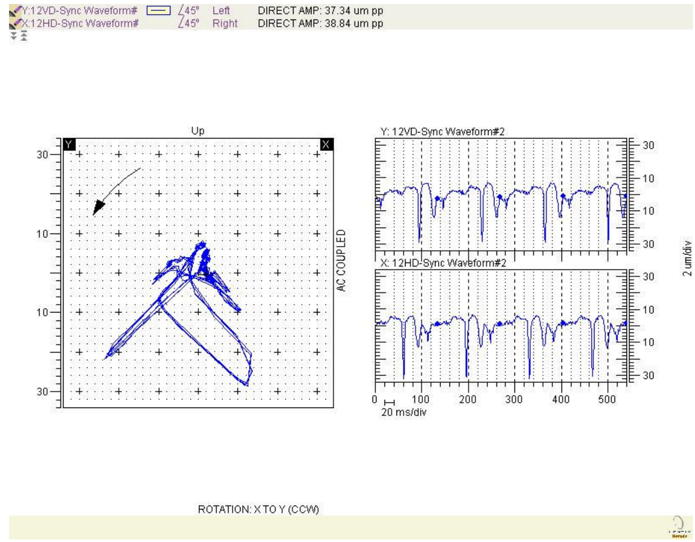


Figure 5 Orbit-Timebase Scratch

Similarly, a scratch, which is a brief distancing of the surface from the probe tip, is often a scratch which would appear as a downward movement of the shaft on the waveform. As shown in **Figure 5**, the scratch appears on the X probe 90 degrees before the Y probe observes the scratch since rotation is in the counterclockwise or X to Y direction. After the signal is passed through FFT processing, it will appear just like an impact would appear as shown in **Figure 6.** Impacts could be misinterpreted as a rub or other impact event. Another way to identify would be to inspect the cascade data as the machine goes through a startup or coastdown. Scratches and burrs would have a fairly

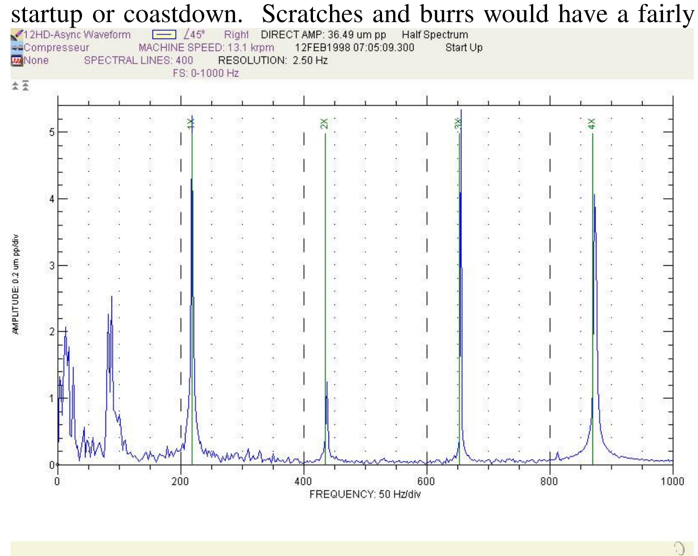


Figure 6 FFT at Full Speed of Scratched Shaft Surface

constant amplitude of harmonics of shaft speed throughout the speed range of the machine. This is observed in **Figure 7.** It can be observed that the peaks are somewhat constant, but are a little distorted because of the rapid deceleration of the machine. The double peaks are due to the speed slowing during the sampling of the FFT. This phenomenon is called smearing. It can be noted that there is some response on the 1X frequency line that indicates the machine is responding to its first balance resonance.

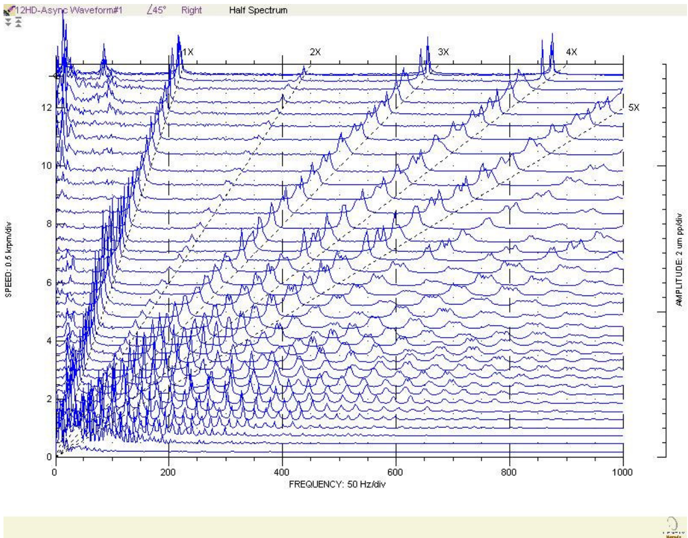


Figure 7 Cascade Plot with Scratch

It has been shown that both the orbit and spectrum analyses can be perturbed by shaft imperfections. It is therefore important to ensure that shaft imperfections do not hamper or give false indications of a machine malfunction.

A note of caution is merited: 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.

Direction of Rotation

Reverse precession of vibration is defined as the vibration direction being in the opposite direction of shaft rotation. The stress that can result from reverse precession vibration can fatigue the shaft very quickly since the cyclic stresses on the shaft because this type of vibration 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 vibration direction. The shaft rotation direction can be verified with 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 is on the design documentation. It is important to verify this physically on the machine. One method commonly used is to use a pencil eraser and place it on an exposed surface of the shaft. While effective, this method may expose the operator to some danger, so more safety-minded and sophisticated methods should be used. Another method is to use shaft imperfections to identify the direction of rotation. At full speed, a large scratch can be used to devise the direction of rotation, as the scratch will pass under each of the probes in the direction of rotation. This will happen despite the direction of precession of the vibration. The full speed orbit of the orbit in **Figure 5** is shown in **Figure 8**. As shown in the slow speed and full speed orbits, the scratches pass underneath the X probe 90 degrees before the Y probe. This confirms the configured direction of rotation as shown by the direction of rotation arrow on the orbit plots. In general, the direction of precession of most vibration signals is also in the same direction of rotation, which is termed as forward precession. Therefore, if a large imperfection is not present, the direction of precession of vibration at slow speeds can also be used to determine shaft rotation direction.

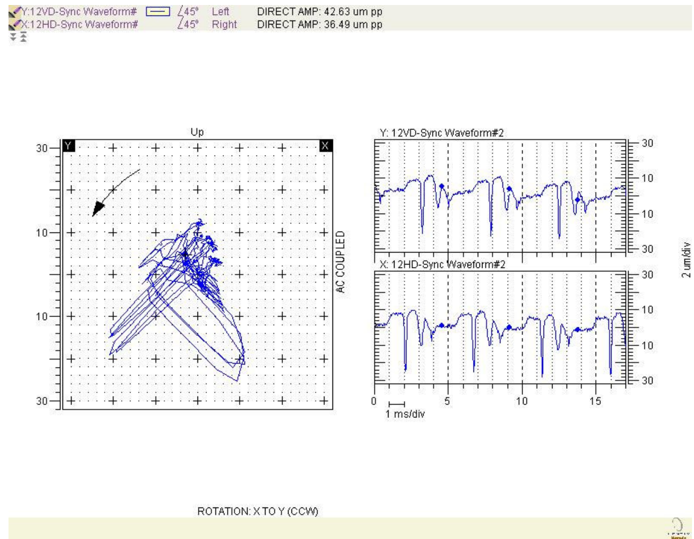


Figure 8 Orbit with Scratch at Full Speed.

As previously stated, at slow speeds, it is uncommon for a machine to have reverse precession. It has been this author's experience that comparing slow roll data from run to run is very important. In the data that is presented, the direction of precession is forward at full speed as shown in the bottom left orbit of **Figure 9**. As the machine decelerates, the orbit remains in forward precession until we see the rotor re-accelerate to approximately 2100 rpm and then coastdown. During this phase of re-acceleration and coastdown, the orbits are in reverse precession.

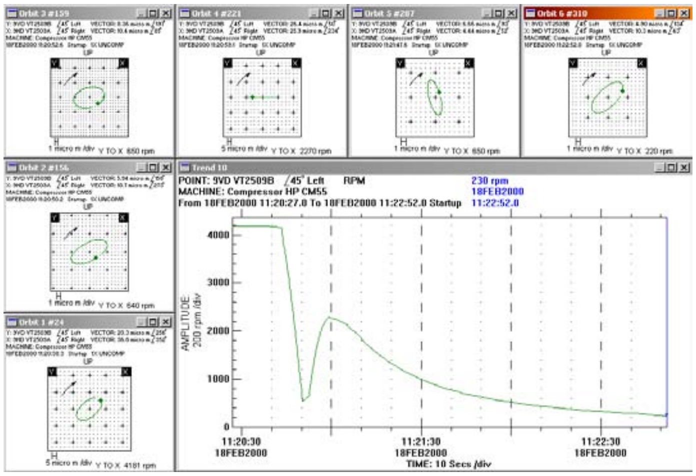


Figure 9 Coastdown-Reverse Rotation

It was discovered that this machine, which consists of two turbines casings and three compressor casings, had a problem with the discharge check valve. A check valve is typically installed in compressors to allow the flow in the piping to flow in one direction, preventing the back flow of the process gas. In this case, the machine was shut down, and the check valve was not operable. As is evident in the speed trend, the machine slowed down very abruptly in less than 10 seconds and subsequently accelerated to approximately 2100 rpm rapidly and then the speed decayed in the natural logarithmic fashion. It is likely that the discharge pressure dissipating through the machine until it no longer drove the machine in the reverse direction drove this logarithmic decay. In this case and many other cases, there are a few severe safety risks with this type of situation:

1. Many times, in the process industry, the compressors' fluids are flammable or hazardous. If the high-pressure gas goes to places where it is not designed, mainly up stream of the compressor stages, the sealing and other mechanical devices may fail under the high pressure or reverse flow dynamics of the process gas.

2. The braking action may cause reverse stresses on the couplings, seals, and other hardware, which may cause high fatigue or even yield the components.

3. The machine may overspeed in the reverse direction. Steam turbines and gas turbines have overspeed shutdown systems, which shut off the energy source (steam or fuel gas) in the event of an overspeed. In the event of a reverse rotation, where a check valve fails, the energy source cannot be shutdown and it is possible for the machine to overspeed in the reverse direction. If there is enough energy in the system to drive the system above the overspeed threshold, the flywheel forces may cause the wheels on the machines to liberate part of their structure.

To help identify and prevent this type of issue, there are reverse rotation monitors that can alert users and control systems of a reverse rotation event.

Rotor Bow (Gravity)

Many large machines have very heavy shafts and the bearing spans can cover long distances. This can cause a shaft to bend under its own weight due to gravity. It is for this reason that many shafts have devices installed called turning gears. These devices engage the shaft once it comes to a stop or comes close to a stop and turns it very slowly so the shaft does not remain at rest and become susceptible to a bow due to gravity. Often there are machinery protection systems: zero speed monitors that can alert personnel and also automatically engage the turning gear.

If a rotor becomes bowed while it is at rest it may become lodged on its internal seals and therefore be prevented from starting up. More commonly, the bow is induced and the machine starts up until it starts to respond to its first balance resonance. Once this happens, the bow behaves like a very large imbalance and it can vibrate excessively, especially in the midspan, and can rub out the seals that are normally required to have small clearances to ensure efficient operation. **Figure 10** shows the startup of a large industrial gas turbine. The turbine starts with approximately 50 µm pp of slow roll levels. Once the machine started to reach its balance resonant speed range, the unbalance due to the gravity bow caused 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 as the machine starts up and the machine subsequently shut down as it exceeded its vibration set points. The coastdown data is on the same plot.

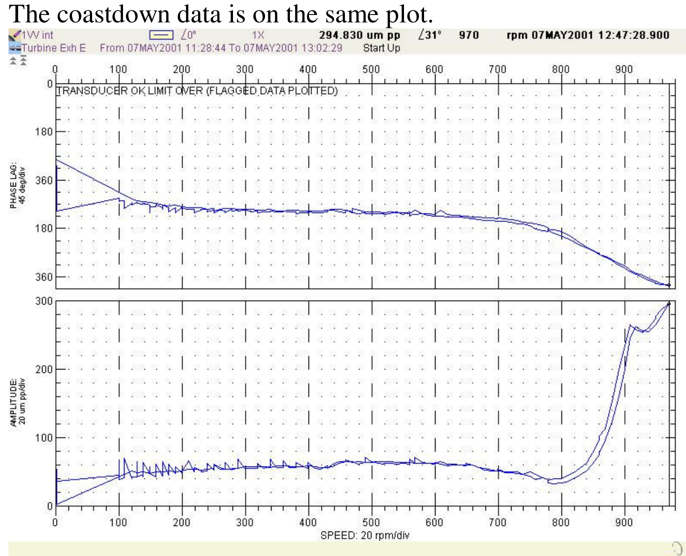


Figure 10 Turbine Startup with Gravity Bow

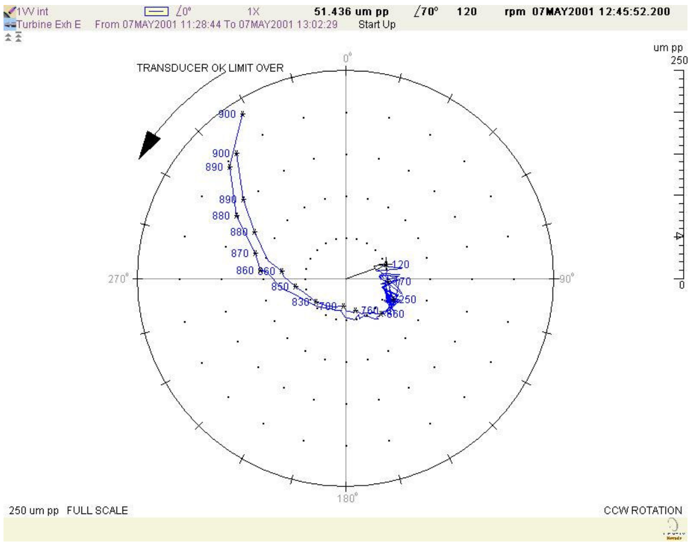


Figure 11 Slow Roll Vector and Startup

The data acquisition system was used to record the bow of the turbine after it shutdown on vibration. Since the turning gear speed was 7 rpm, which was slower than the capabilities of the hardware to record data, the system was used to record the DC gap readings of the proximity probe as the shaft was on turning gear. The data in **Figure 12** shows almost a 50 µm pp reading. This was also done initially with a simple voltmeter as the shaft rotated on turning gear.

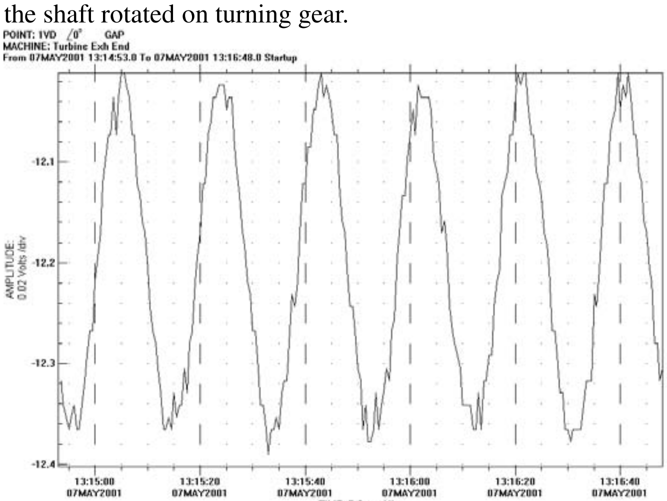


Figure 12 Slow Roll After Trip on Vibration

The machine was left on turning gear overnight and the slow roll readings reduced to a level that was acceptable for a subsequent startup. See **Figure 13.** On that startup, the vibration did not exceed 80 µm pp at the first balance resonance.

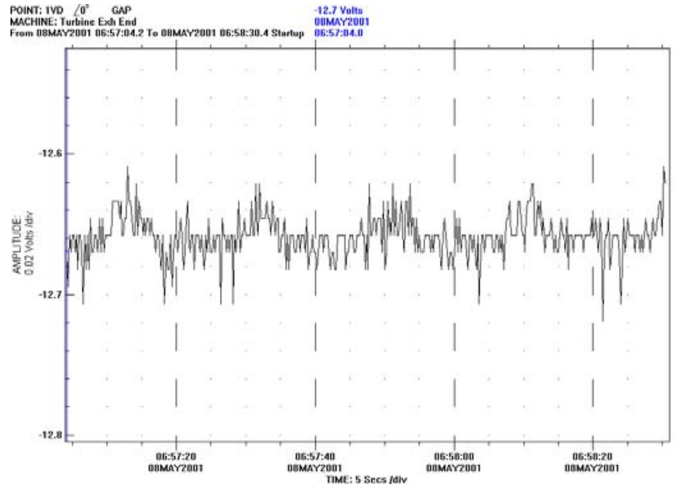


Figure 13 Slow Roll After Overnight on Turning Gear

For large machines that are susceptible to gravity bows, there are eccentricity monitors which measure the slow roll values of a machine at very low speeds using special filters and compare them to a maximum allowable setpoint. This monitor which will send a permissive to the control system to allow the machine to start. Machines that expose the 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.

Rotor Bow (Thermal)

Some rotors develop a thermal bow as they go from full speed, no load to a full load condition. One method of confirming a thermal bow is that often the slow roll data before, and after, a start will be different. Since the rotor will assume a bowed condition due the sensitivity to heat, it will retain those characteristics after a hot shutdown. Once the heat is removed, the bow will return to a simple mechanical bow if any exist. Often, running the machine on the turning gear will relieve any thermally induced bows

The machines that are sensitive to thermal bows are often electric in nature such as generators and motors. Steam and gas turbines are also sensitive to thermal heating. Machines that exhibit thermally sensitive 1X vibration vectors are due to the fact that normally the thermal bow will cause the balance state of a rotor to change. These changes are sometimes called a thermal vector or a load vector. Often these are repeatable and predictable. These changes are often due to shorts on the rotor electrical assemblies, which cause local heating and cause a rotor bow or simply an eccentric mass due to the thermal expansion of the shorted area and surrounding the shorted area. The vibration data from a thermally sensitive motor is displayed in **Figure 14** and shows the vibration from full speed no load to full load conditions.[1] This data was repeatable from run to run so it was decided to attempt to balance the motor so that the vibration amplitudes before and after the heat soak would be similar. This was achieved by a brilliant balancing job to focus the thermal through the origin of the polar plot. The thermal bow in this particular case was due to the thermal treatment process of the rotor forging which left the thermal growth characteristics from one side of the rotor to another.

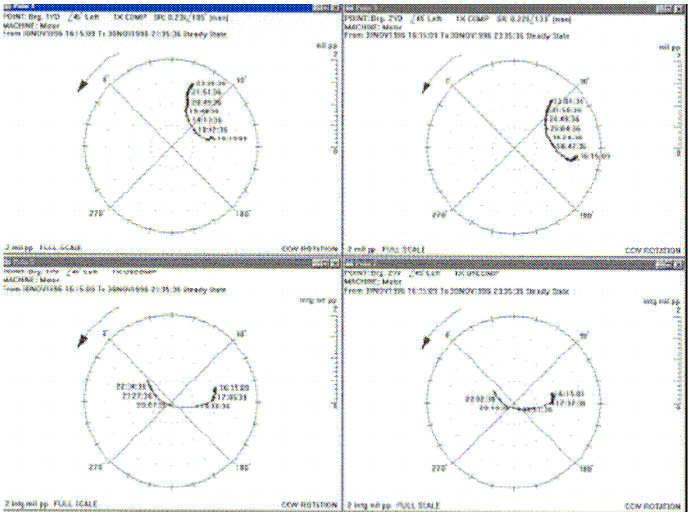


Figure 14 Thermally Sensitive Motor

Locked up Coupling

With the advent of flexible couplings, many machine operators have benefited from the advantages of allowing machinery to run with minor amounts of misalignment. However, with more advanced couplings, some complications can arise from these pieces. Sometimes one or more of the components may have defects or assembly issues on the machines that cause machines to vibrate. Gear type couplings are robust and allow for difficult service, but these couplings are the most frequently plagued couplings. Often gear couplings require continuous lubrication to ensure the gear teeth are able to allow very small movement that may be due to very small misalignment. If there is a problem with the wear of the teeth, as can be the case with old couplings, or some mechanical interface and assembly issues, the gear coupling can lock up and lose its flexible nature. The most common symptom is a high 1X vibration that looks much like a rotor imbalance. Most modern couplings come with the facility to allow in-situ balancing, with either balance holes or the ability to add washers to the coupling bolts. One clear indication that there may be coupling problems shows up in the 1X nature of the vibration that remains down to slow speeds of the machine.

The following case shows the clear indications that can be associated with a flexible coupling not being able to flex freely. The machine operator called out an engineer to balance their machine because of high 1X vibration. The data as found upon arrival revealed the slow roll data as shown in **Figure 15**. The orbits shown are on either side of a flexible disc pack coupling. We can see that the phase in the slow roll orbit of bearing 10 is in phase with the high-speed orbit of bearing 11. **Figure 16**. The full speed orbits also show that the phase across the coupling is generally in phase.

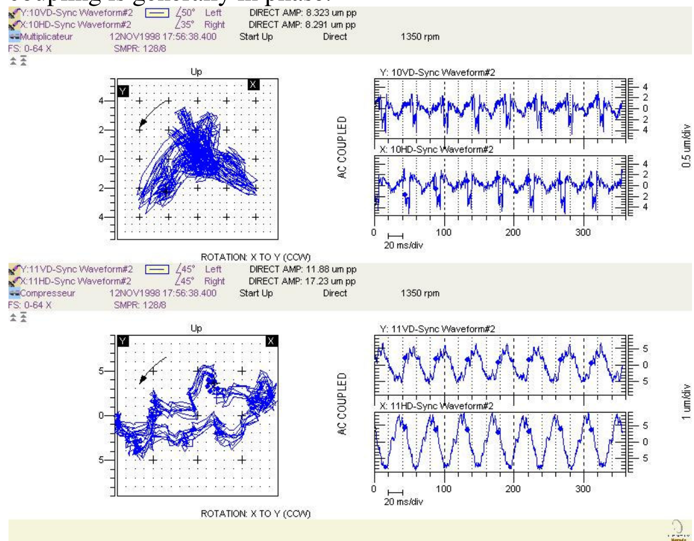


Figure 15 Initial Slow Roll Orbits

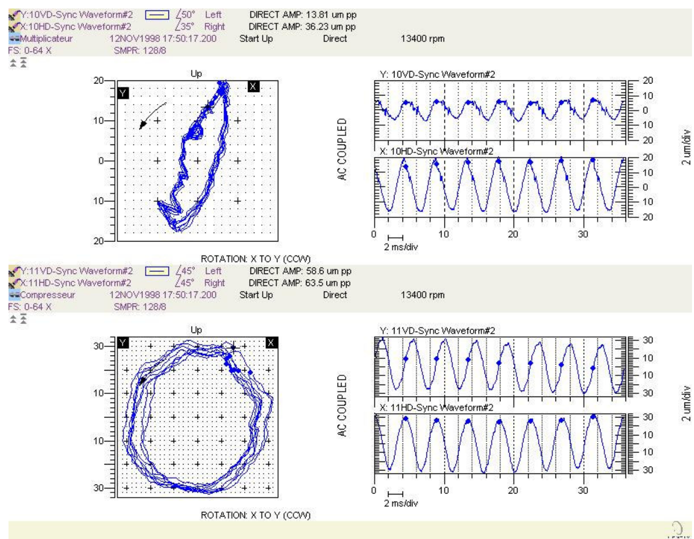


Figure 16 Initial Full Speed Orbits

After two attempts to place weights in the coupling to balance the 1X vibration, it was evident that the root cause of the problem was not unbalance. The coupling had a tendency to vibrate at the same amplitude and direction despite the attempts. See **Figure 19.** It was suspected that there were coupling problems. After the coupling was disassembled for a bearing replacement, the slow roll and full speed orbits in **Figure 17** and **Figure 18** were observed. We can see that the phase of the slow roll orbits of bearing 11 are 180 degrees out of phase. It also follows that the full speed orbits show that the vibration response of the probes on either side of the coupling is 180 degrees out of phase from the previous initial set of data. The phase across the coupling remains in phase as well.

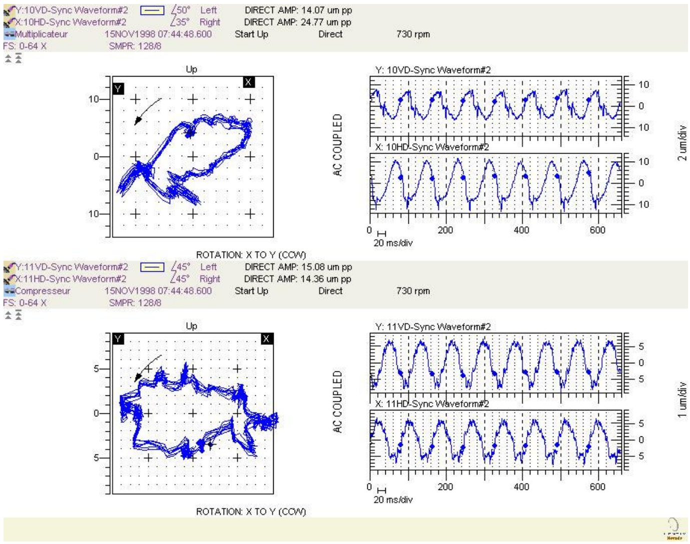


Figure 17 180 deg Out Slow Roll Orbits

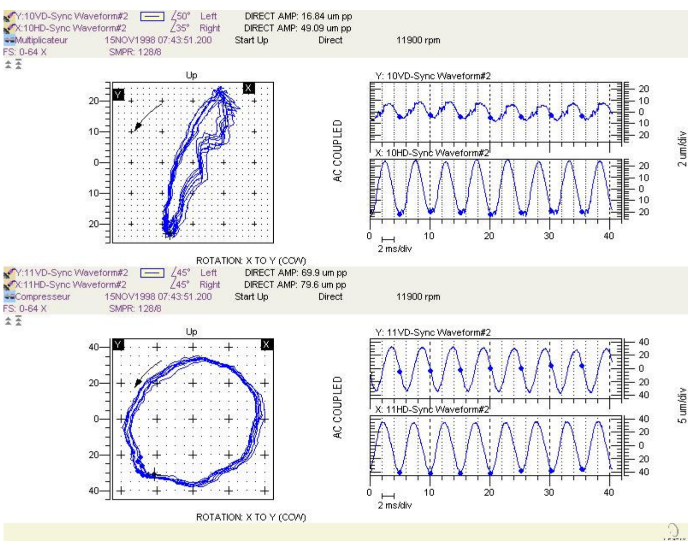


Figure 18 180 deg Out Full Speed Orbits

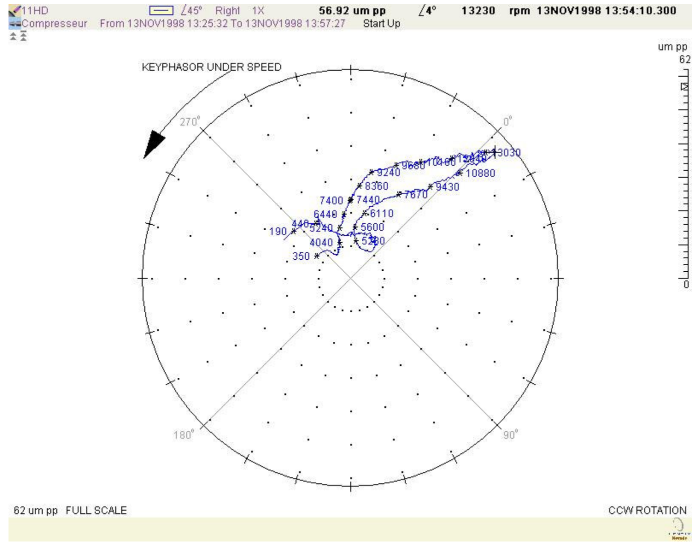


Figure 19 Polar Plot with Mechanical Constriction

After discussion with the mechanics working on the machine, it was discovered that a coupling spacer piece (approximately 5 mm thick) was bent. This spacer piece, which was fabricated initially during construction to account for an error in the axial distance between machines foundations, was causing the coupling to take on a bent attitude. When the bearing was replaced, the spacer piece was installed 180 degrees out of phase from its installation, causing the slow roll and full speed responses to be opposite one another. This bent angle was behaving like a 1X unbalance, but restricting the normal balance resonance response to the direction of the bend. The spacer was not match marked as the rest of the components of the coupling were, so there was no reference as to how to re-install the piece. The spacer piece was re-installed by carefully tightening the bolts to flatten it, so that the coupling was no longer bent. A suggestion was made to correct the foundation errors to alleviate the need for the spacer, but the costs associated with this fix were prohibitive. A new spacer piece was to be fabricated.

Non-Concentric Coupling

Rigid couplings are not immune to mechanical problems that can be associated with the non-concentricity issues discussed above that can arise due to the flexible elements or other components not being installed correctly. One case included a steam turbine – speed reducing gearbox - generator set that included a shared bearing (bearing 4) between the generate and gearbox. This machine had a rigid coupling between the gearbox and generator. The generator had been re-worked and sent out to a shop for balancing prior to installation. The machine exhibited high 1X vibration as it passed through its resonance. **Figure 20.** It also included some elevated 1X runout. A few attempts were made to balance the machine in situ with little success. The machine was sent back to the shop for another balancing exercise and re-installed with little improvement. After close inspection of the slow roll data, especially as it appears on the polar plots, it was suggested that

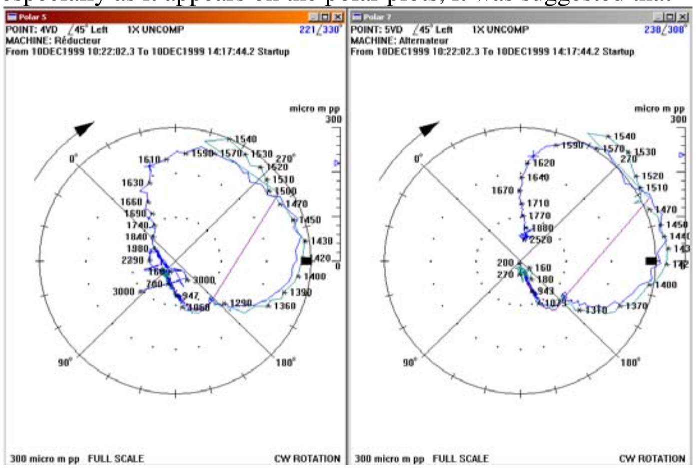


Figure 20 [1]**Generator Startup with High Vibration**

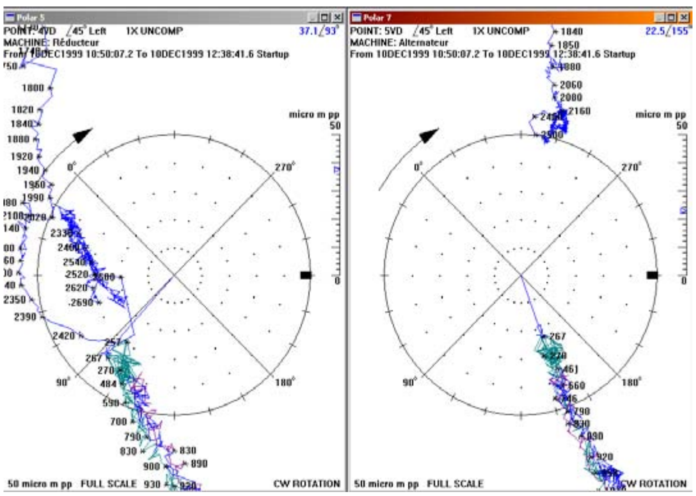
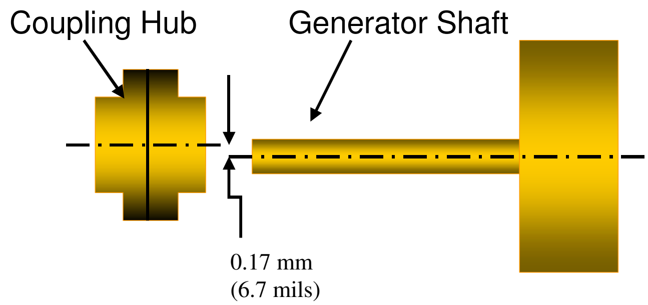


Figure 21 Slow Roll Direction and Unbalance

the coupling interface may be suspect. Although the data shown for bearing 4 does not show the slow roll vector directly in phase with the slow speed unbalance response, it does show it generally in phase with the unbalance direction. **Figure 21.**

Figure 22 Non-Concentric Hub



This slight discrepancy is likely due to the position of the probes and the mode shape of the machine. Bearing 5 shows a much clearer representation of a slow roll vector being in phase with the unbalance direction. A recommendation was made to measure the runout with a dial indicator referenced to the foundation of the coupling hubs, faces, and shafts. It was discovered that although the coupling hub had an interference fit with the shaft, it was almost 0.17 mm off the centerline of the shaft. This non-concentric hub was causing a severe unbalance and would have never been corrected with balancing.

Probe	Oct 99	Dec 99	Jul 2000
4VD	$41\angle 104°$	$37\angle 98°$	$30\angle 66°$
4HD	$17\angle 183°$	$19\angle 207°$	$27\angle 157°$
5VD	$24\angle 139°$	$27\angle 149°$	$10\angle 131°$
5HD	$28\angle 259°$	$36\angle 262°$	$15\angle 228°$

Table 1 Generator Slow Roll Trend

From **Table 1**, the slow roll data did not change from run to run. Only after the coupling hub was re-worked, did the slow roll data change to more reasonable levels. Although the bearing 4 data is from probes close to the coupling, it was distanced from the coupling and therefore does not show the 170 µm pp runout as tested with a dial indicator. The July 2000 data is not low by any means, but it does show that the slow roll data has changed and the subsequent startup data showed maximum amplitudes of 70 µm pp through the balance resonances.

REFERENCES

[1] Peyton Swan *Orbit* magazine, Bently Nevada, LLC. A Tale of Two Motors, December 1998.

Keyphasor is a trademark of General Electric Company. [2]

