



## **Transient Speed Vibration Analysis - Insights into Machinery Behavior**

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## **Abstract**

This paper discusses the need for and benefits of analyzing machinery vibration data taken during startup and shut down to help more fully understand machinery dynamics and to resolve vibration and operational problems that are not readily solved using only steady state / spectral data.

Many analysts focus on acquiring steady state vibration data, often as part of ‘Predictive Maintenance’ or PdM programs. Such programs have proven their worth and are often a plant’s first-step in identifying and resolving reliability problems.

PdM programs typically focus on using portable data collectors to acquire and analyze spectral data and to a lesser degree the time waveform data. This data is usually taken during constant speed operation, and is generally not phase-referenced. It achieves its intended goal of providing trended data to identify arising problems, while also providing data that can be analyzed for frequency content and severity. And it is the frequency content that allows us to begin our analysis process and identify possible fault mechanisms.

However, steady state spectral analysis remains just a single tool – the identification of frequency versus amplitude. We may or may not be able to accurately identify a root cause to a vibration problem from the spectral data. This is often the case with journal bearings, whose vibration signatures usually show just a predominant one-times rotational speed frequency component, and the analyst is left with several fault possibilities to choose from.

Our paper will review the equipment and techniques we use to acquire additional vibration data during startup and shut down. This ‘transient’ speed data provides exceptional insight into machinery dynamics, and allows us to accurately sort out most machinery problems that are not readily solvable using only steady state data.

We will discuss how to properly set up for and sample transient data, discussing vibration transducers, band width filters, sample times, and required data resolution. We will review the types of transient data plots typically used in analysis, including: polar; bode; waterfall; cascade; orbit / timebase; and shaft centerline. We will discuss how to identify the major classes of machine faults within the transient data: mass unbalance; shaft misalignment; rotor resonances; structural resonances; shaft centerline movement; rotor to seal rubbing; and oil whirl / whip. And we will conclude with case histories highlighting the identification and resolution of specific problems.

## Introduction

Have you ever analyzed vibration data, only to discover there was more than one possible root cause for the same frequency response? Have you ever analyzed journal bearing problems, only to discover that most faults will generate a 1X response? Have you ever felt that “vibration charts” still left you with too many possible causes?

If you answered yes to these questions, you are not alone. Every year we encounter many analysts facing the same problems. And this mostly results from the ‘industry’ focusing too intensely on the collection of steady state data with walk-around programs and “portable data collectors”.

This paper discusses how to start solving this dilemma. We look at the need for, and the benefits of, analyzing machinery vibration data taken during startup and shut down using more sophisticated analysis equipment. We also show how to acquire and use this data to more thoroughly understand machinery dynamics, and how to resolve vibration and operational problems that are not readily solved using only steady state / spectral data.

Looking at today’s market, we see that the standards of performance have significantly improved for Predictive Maintenance (PdM) program analysts over the past two decades. Advancements in technology have dramatically improved the quality of our data acquisition hardware and software, while also continuing to reduce costs. And industry in general has recognized the benefits and return-on-investment that can be achieved with a quality vibration analysis program.

One of the more important reasons behind the increased quality of our analysts, and the results of our programs, is the high-quality training & certification programs that have become available. By following the ISO and ASNT vibration analyst guidelines, we now have multiple sources for meeting our training needs, and can effectively advance an analyst from novice to expert following a well defined training path.

Although today’s analyst has advanced hardware, software, and training, we find many users are still trying to solve all of their problems using steady state<sup>1</sup> spectrum analysis. Spectrum analysis is an excellent tool, and is rightfully the backbone of PdM programs. It allows us to quickly identify many faults, assess their severity, and plan for corrective maintenance.

But in many cases spectrum analysis alone cannot resolve the problem. This is where advanced training can help. One area that merits specific attention is transient speed vibration analysis. It can often provide the missing data and get us to a solution. We will discuss the general concepts of transient vibration analysis, provide data sampling guidelines, explain the various types of data plots used in transient vibration analysis and the problems we can identify in them, and provide case histories with examples of various problems and how we identify and resolve them.

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<sup>1</sup> Steady state data: data typically taken while a machine is operating at normal, full-load operating conditions, usually at a constant speed (rpm).

## **The Root of a Problem**

Predictive Maintenance (PdM) is usually thought of as the use of condition monitoring technologies to detect machinery faults at an early stage, allowing planned corrective maintenance on an as-needed basis. These technologies include vibration, thermography, ultrasound, motor current, and oil analysis. Of these, we are concerned here specifically with vibration analysis, how it has evolved, and some potential implications on an analyst's skill-set.

Vibration-based PdM programs have proven their worth in managing rotating machinery, and the techniques and technology have developed to a very mature state. So successful is this technology that it is common for vibration-based PdM programs to provide a Return on Investment (ROI) of less than one year when the savings in unplanned downtime, reduced machinery damage, and lost production are contrasted against the hardware, software and manpower training costs.

PdM programs typically use portable data collectors to acquire our spectral and waveform data on a periodic basis. The data is generally taken during steady state operation, and is usually not phase-referenced. This process achieves the intended goal of providing data that can be trended to identify arising problems and providing data that can be analyzed for frequency content and amplitude severity. And it is the frequency content that allows us to begin our analysis process and identify possible fault mechanisms.

Because of their effectiveness, vibration-based PdM programs are often a plant's first step into PdM, the identification and resolution of machinery reliability problems, and moving from a reactive to proactive maintenance environment.

And because of that, considerable focus is often placed on the training and technology that is required up-front to produce an effective, efficient analyst that can carry out the required job functions.

What has transpired in the industry over the past two decades or so has been the development and homogenization of a very effective palette of training courses from a variety of vendors. And in parallel with this has been the development of training-related standards by both the International Standards Organization (ISO) and the American Society of Non-destructive Testing (ASNT) in an effort to provide common industry-wide guidelines for training and certification of advancement. Specific standards applicable to vibration analysis training include ISO 18436.2, and ASNT Recommended Practice SNT-TC-1A.

ISO 18436.2 specifies 4 levels of vibration analyst certification, along with the corresponding levels of practical experience; ASNT specifies 3 levels of analyst certification. Naturally, there is some overlap when comparing the two standards, and there are minor variations regarding course content, examination certification, and administration. However, whether an individual pursues an ISO or ASNT-based certification process, they can be assured that either will provide an effective basis for training.

In surveying the ISO and ASNT guidelines, and the various seminars available from a variety of vendors, it quickly becomes apparent that the main focus of analysis is placed upon spectral data analysis. This is a logical starting point for the novice or Level 1 analyst, and it is easy for even the lay-person to understand how different faults generate different characteristic frequencies, and that we can then show these in an FFT / spectrum plot.

In contrast, it is far more difficult to explain frequency content within any waveform that is much more complicated than that for simple harmonic motion. (Imagine explaining how to detect rolling element bearing faults in a time-waveform to a novice analyst!)

So data analysis training typically begins with learning to understand spectrum plots, and then learning to recognize the common machinery faults such as unbalance and misalignment that are easily identified. It then continues along to more advanced machinery and problems as the analyst progresses in his or her training. Let's look at an example of what our analyst might need to dissect on a motor-driven pump unit that has a speed-increasing gearbox.

On any induction motor we can have discrete frequencies generated by the motor's design characteristics:

- 1-times & 2-times Line Frequency
- Slip & Pole Passage Frequency
- Rotor Bar Passage Frequency
- Potential sidebanding around 1xLine, 2xLine, and Rotor Bar Passage

If we have a 60 Hz electrical system, and a 6-pole motor operating at 1,182 rpm that has 48 rotor bars, the following frequencies would be calculated:

- Slip Frequency ..... 18 cpm
- Slip Ratio ..... 0.015
- Pole Pass Frequency ..... 108 cpm
- Rotor Bar Current Passage ..... 54 cpm
- Rotor Bar Passage ..... 56,736 cpm

Now, if the motor is equipped with rolling element bearings, they too would introduce a set of potential fault frequencies for analysis.

Identifying the discrete frequencies generated by rolling element bearings is one of the most common uses of spectral analysis, because these faults usually are readily identified. Much has already been written about inner and outer race faults (BPFI, BPFO), ball spin (BS), fundamental train (cage) frequency (FTF or CF) for various types of bearings, and we will not belabor those calculations here. Suffice it to say that many of the current PdM software packages identify these frequencies automatically if you provide the bearing identification number. For example, consider two SKF rolling element bearings - a #6316 and a #22228. We would have the following fault frequencies (in terms of running speed)<sup>2</sup>:

	FTF	BS	BPFO	BPFI
6316	0.39x	2.07x	3.09x	4.92x
22228	0.43x	3.60x	8.21x	10.80x

Beyond these “look up” fault frequencies, bearing failure can be further refined as Stages 1 – 4, providing an indication of the progress relative to the observed frequency patterns. This provides some degree of insight (however subjective) into the useful remaining bearing life. But it still remains fundamentally a frequency response identification issue.

Gearboxes are a prime target for spectral-based frequency analysis. Whether they are single or double-helical, bevel, worm or planetary gears, they will all generate their own distinct frequency responses.

Most analysts can calculate and identify the Gear Mesh frequency and its harmonics, and can likely identify pinion and gear frequency sidebands. These are the important first steps in gear analysis. Equally important, though generally less well understood, are the characteristic frequencies for Tooth Repeat and Assembly Phase Passage.

<sup>2</sup> Source: DLI Engineering/ExpertALERT software.

Continuing our example, if the bull gear has 33 teeth and is driving a 121 tooth pinion, the following gear-set related frequencies would be calculated:

- Tooth Repeat (Hunting) ..... 394 cpm
- Gear Speed ..... 1,182 cpm
- Pinion Speed ..... 4,334 cpm
- Assembly Phase Passage .... 13,002 cpm
- Gear Mesh ..... 143,022 cpm

On the pump our job becomes a little easier. If we have a single-stage impeller with 6 vanes, we might expect vane passing vibration at 6X, in addition to the usual 1-times running speed vibration from residual unbalance. And there might also be some broad-band noise related to recirculation or cavitation.

If this were a multi-stage unit we might expect vane passing vibration for each stage, as well as the potential for sum and difference frequencies. And on both the gearbox and the pump we would again need to consider the bearing type used in each location.

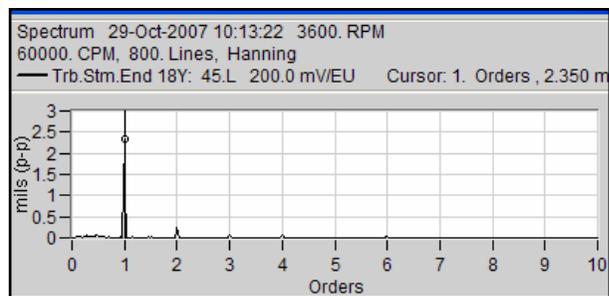
Identifying all these frequencies is a necessary part of the analysis process – when they are present! It is obvious that spectral analysis is really the only way to properly sort through the myriad of frequencies. This is why such an emphasis is placed on frequency identification in analyst training.

Field experience indicates that many times we will find problems at the prescribed fault frequencies. However, and very interestingly, that same experience also shows that in many situations, perhaps the majority, the largest responses seen occur at 1-times running speed, or 1X. Or, even in the presence of other faults, the 1X response may likely be dominant. As a quick look at any of the common spectral analysis ‘cheat-sheet’ charts shows, we have a variety of problems that can occur at 1X.

These 1X forcing functions include:

- Unbalance (mass & electrical)
- Misalignment (shaft and/or bearing)
- Bent or Bowed Shafts
- Resonance (rotor or structure)
- Rotor to Stator Rubbing
- Shaft Cracking
- Mechanical Looseness; Loose Bearings
- Mounting Problems (soft-foot)
- Journal Bearing Wear

So, while frequency identification is necessary, much of our work ultimately comes down to resolving a rather bland spectrum plot with a predominant 1X vibration that looks something like this:



Or, we may find asynchronous frequencies that do not occur at an expected fault frequency, or be wondering why a particular fault frequency may be particularly amplified. It is generally at this point that spectrum analysis, by itself, may not allow us to accurately identify a root cause.

It is in these cases where transient vibration analysis can often help us get to the root of the problem. Even when no particular problems are apparent in the steady state spectral data, transient vibration data presents a wealth of information for analysis and provides much deeper insight into the machinery condition. It is not at all unusual to detect problems within the transient data that are not apparent in the steady state testing.

## What is Transient Vibration Analysis?

Transient vibration analysis, or perhaps more correctly for our use here, transient speed vibration analysis, is the acquisition & analysis of data taken while a machine is being started or stopped. By sampling as a function of speed, we gain significant insight into the rotor and structural dynamics that cannot be had with only steady state analysis. This information includes:

- Unbalance “Heavy Spot” Locations
- Rotor Mode Shapes
- Shaft Centerline Movement / Alignment
- Bearing Wear
- Shaft Runout
- Critical Speeds / Resonances
- Rotor Stability
- Bearing Wear
- Foundation Deterioration, and others

As a first try, an analyst may try to capture several spectra during a transient run using a portable data collector. This may be helpful, but the sparse data sampling and only 1 or 2 transducers falls far short of the data that can be gathered using more dedicated instrumentation.

Good transient analysis generally involves acquiring data from multiple transducers simultaneously. For example, a small steam-turbine generator machine train would typically have four radial bearings, with two proximity probes installed at each bearing in an X-Y orientation, giving us 8 radial vibration measurements. If we also monitor thrust position, which is usually measured using a two-probe setup, we have 10 channels of data. Finally, we need a tachometer channel to monitor and measure speed. During startup or shut down this data is then sampled versus rpm, with samples often being taken in increments of 5 to 10 rpm.

On larger units, there may be multiple turbine casings. The largest units in nuclear service have 12 radial bearings, and some units have both proximity and seismic transducers at each bearing. Including thrust, that is a total of 50 channels of data! Similarly, units having gearboxes, or units with multiple compressor casing will all likely have in excess of 12 channels of data.

Two other requirements generally not considered under in PdM data sampling must be considered. First, all channels should be sampled in a truly simultaneous manner. This allows generation and analysis of the data plot types we will discuss shortly. And second, all data should be referenced to a once-per-revolution speed probe, which we will also discuss.

Some analysts may feel that transient analysis is mostly applicable to large, critical turbomachinery. While this machinery certainly merits the time and effort involved, we find transient analysis very applicable to balance of plant equipment as well. Some of this less critical equipment is often poorly designed and/or supported, and suffers from chronic poor reliability. Using transient analysis has allowed us to solve many problems that were otherwise not resolved through ordinary PdM analysis.

We have many case histories where no transient vibration data had ever been recorded. Because startups and shut downs generally are not performed regularly, we recommend acquiring transient data whenever possible. This aids future analysis, and lets us benchmark machinery against future changes. This is particularly true of the critical machinery in many plants – turbine/generators, boiler feed pumps, gas compressors, and the like.

## Instrumentation<sup>3</sup>

Because of the need for simultaneous sampling of multiple channels with a speed reference, the typical PdM data collector generally will not be sufficient.

So just what are some of the requirements for acquiring and analyzing transient data? Aside from the usual requirements of configuring frequency spans, lines of resolution, spectral windows, and transducers types, here's a "short" list of desired abilities for transient vibration analysis:

- Minimum channel count of 8, with 16 or more channels preferred
- Synchronous sampling of all channels
- 2 or more tach channels for the high-speed and low-speed shafts of units containing gearboxes or fluid drives
- Accurately sample data at low rotor speeds (< 100 rpm)
- Measure DC Gap Voltages up to -24 Vdc and produce DC-coupled data plots (for shaft centerline & thrust data)
- Provide IEPE / accelerometer power
- Electronically remove low speed shaft runout from at-speed data
- Display bearing clearances; plot shaft movement with available clearance
- Specify RPM ranges for sampling, and RPM sampling interval
- Produce bode, polar, shaft centerline, and cascade plots for data analysis
- Tracking filter provides 1X and several other programmable vector variables

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<sup>3</sup> While we normally like to remain 'vendor-neutral' in our papers and discussions, our clients and customers often want to know what works for us, to flatten their learning curve and become effective more quickly. And effective transient vibration analysis is far more demanding of instrumentation & software in terms of sampling and data plotting requirements. So we feel a discussion of relevant instrumentation is warranted.

We currently use an IOtech Zonic Book/618E data acquisition system in conjunction with their eZ-Tomas software<sup>4</sup> for rotating machinery steady state and transient vibration analysis. The system consists of an 8-channel ZonicBook base unit, and can be expanded in 8-channel increments by adding WBK18 modules. A total of 7 - 618E modules can be added, for a total channel count 56. The system is easy to use, light weight, portable, and the per-channel costs are among the most affordable in the industry. For a copy of the most recent product information, check this link:

[http://www.iotech.com/catalog/cat\\_pdf/ZonicBook618E.pdf](http://www.iotech.com/catalog/cat_pdf/ZonicBook618E.pdf)

The ZonicBook system is powered by a PowerPC processor, and all acquired data is transferred to the PC in real time at 2+ Mbytes per second. This means that every acquired data point resides on your PC's hard drive, making recreation and post acquisition analysis of acquired data as precise as possible. And all time-domain measurements are transferred, not just spectral data, which means there's no data loss when analyzing acquired waveforms. Data storage is only limited by the amount of hard disk memory on your PC, or available on a network. And all channels are measured synchronously, providing 1 degree phase matching between channels.

The ZonicBook has a 10/100BaseT Ethernet interface and can be used in a point-to-point application, or can be attached to a network for remote monitoring. The system also has four dedicated Tachometer inputs, and can also use any analog channel as a tachometer input.

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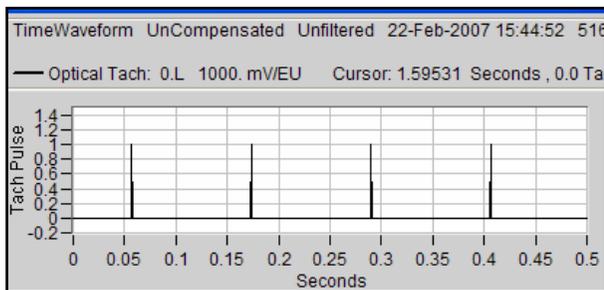
<sup>4</sup> Most of the vibration data graphics contained in this paper were produced using the eZ-Tomas software package.

## The Need for (Rotor) Speed

A key component to successful transient analysis is a once-per-revolution tachometer signal, often referred to as a Keyphasor<sup>® 5</sup>. This signal provides a triggering pulse for the data acquisition instrument tracking filter, and allows us to establish the rotor phase angle reference required for transient data analysis.

For machines without a permanent vibration monitoring system, a tachometer pulse is easily provided using a portable laser tachometer that observes piece of optically reflective tape attached to the shaft. We have had excellent results with Monarch Instrument's PLT-200. The PLT-200 can sense the optical tape from a distance of about 25 feet, and at angles of  $\pm 70^\circ$ ! The viewing distance and angle provides for excellent flexibility in the field.

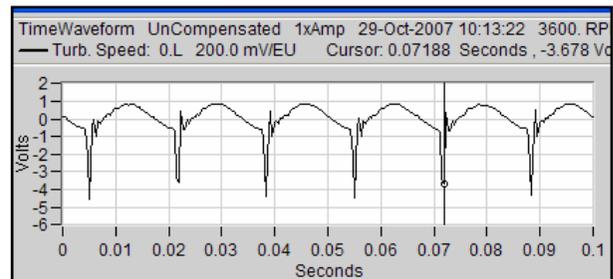
The PLT-200 provides a reliable once-per-revolution TTL output pulse that is fed directly into the data acquisition instrumentation. The figure below shows a typical pulse output from the PLT-200 observing optical tape. Note the clean, with well defined positive and negative slopes to the pulse, with no significant overshoot at the beginning or end of the pulses. This provides for a very reliable speed and phase reference trigger when used in conjunction with the ZonicBook dedicated Tachometer input channels.



<sup>5</sup> Keyphasor is a registered trademark of Bently Nevada Corporation.

On machines with permanent vibration monitoring systems, a proximity probe is often used to observe a notch or keyway in the shaft, providing a DC voltage pulse output. This normally works very well when used as an analog tach input on the ZonicBook. However, some signals create triggering problems due to signal quality issues. The prox-probe tach pulse below is typical of field installations. There are several problems present that might cause triggering issues:

- Overshoot / ripple, which may cause multiple triggers per revolution
- The overall signal also contains an AC vibration signal
- The bottom of each pulse is not at the same voltage level



If your instrumentation does not properly trigger using default settings, you may be able to adjust the trigger voltage level. In the figure above, a trigger setpoint of -2.0 to -3.0 Vdc would work nicely. Your goal is to set the voltage so the instrument sees that voltage level and corresponding slope (+ or -) only once per revolution.

If reliable triggering cannot be established, a signal conditioner such as Bently Nevada's TK-15 Keyphasor Conditioner can be used to modify the signal. It can simultaneously 'clip' the top and bottom portions by applying bias voltages, thus removing any ripple / overshoot from the pulse, and producing a more TTL-like pulse.

If a machine is already running and you are preparing for a shut down, you will probably have time to check your triggering adjustments before sampling begins. But be aware if you are preparing for a startup, you could miss data if you are adjusting the trigger setpoints while the machine is starting.

On steam turbines and VFD drives, you will likely have some leeway operationally, and can perhaps request the machine to be held at low speed while you adjust the signal and triggering. But on induction motors the starts will be very fast and you probably won't know if the tach is working properly until full speed is already achieved. Be ready to make some quick changes!

### **Vibration Transducer Selection vs. Machine Design**

Specific transducer recommendations depend on the design of the machine being analyzed, and the type of data desired. While a detailed discussion is beyond the scope of this paper, we feel some general comments are in order to help ensure that the correct transient data is available for analysis.

We can begin by loosely segregating machinery into two classes: those with journal (sleeve) or tilt-pad bearings, and those with rolling-element bearings. And some machines will contain both types!

Journal bearing equipped machines include the various designs of babbitted bearings – cylindrical, lemon bore, offset, pressure-dam, multi-lobe, axial groove, and tilt-pad designs. The common feature among them is that the shaft rotates within a (mostly) cylindrical, lubricated surface, and that the inside diameter of the bearing is slightly larger than that of the journal (shaft).

You may wonder how can we 'see' the shaft moving within the bearing. We cannot do this with the accelerometers used with typical portable data collector, which only measure the bearing casing movement. The answer is by using 'proximity' probes. 'Prox' probes are non-contacting, eddy current transducers that measure shaft movement without making physical contact with the shaft surface. They allow direct measurement of the shaft vibration and position. This direct measurement is important because transmissibility losses through the oil film can significantly attenuate the shaft motion that is finally transferred to the bearing cap.

Depending on the bearing design being monitored, proximity probes can be mounted directly through a bearing cap to observe the journal. Or, they can be placed in small brackets and mounted to the bearing or seal faces, or installed using long 'stingers' to penetrate large bearing enclosures.

Two probes are usually mounted in an X-Y configuration, 90° apart from each other, at each bearing. This allows measurement of the shaft orbital and average centerline movement, and allows us to produce the associated data plots for transient analysis. These X-Y based plots are powerful analysis tools, and are not available if only a single probe is used.

In general, proximity probes should be used on machines equipped with journal or tilt-pad bearings. If we compare the shaft vibration measurements from a prox probe to the bearing cap vibration from a seismic probe (accelerometer or velocity probe) at the same bearing, we will usually see significantly less vibration on the bearing cap.

While it may make your manager happy to have lower vibration reported, it will likely decrease the accuracy of your analysis! This is not to say that good analysis cannot be per-

formed on journal bearings with an accel, but you will be working with limited quality data.

The line between when to use proximity probes and when to use an accel becomes a bit more blurred if we consider a machine equipped with journal bearings that has a soft (compliant) bearing support, and/or the machine casing mass is relatively light compared to the rotor mass.

In these cases, the more compliant bearing pedestal and/or lighter casing will more readily follow the shaft vibration. We can expect lower transmissibility losses, and a higher percentage of the original shaft vibration being present on the bearing cap.

For example, let's consider a multi-stage centrifugal barrel-style compressor. Here we would have a relatively light, flexible rotor, and the bearings would be mounted in the compressor end bells, which are rigidly attached to the compressor barrel. If we compare the casing mass to the rotor mass, we would see a very high case to rotor mass ratio. This means that for whatever vibration originates on the rotor, the casing would not show very much movement because of its much larger mass; the rotor motion would be readily absorbed by the heavy case (and the oil film).

Next, consider an industrial gas turbine engine with journal bearings. One end of the machine will typically be mounted on supports that are horizontally and axially flexible, such as a series of vertically mounted rods, supporting the compressor end of the machine. Due to their lateral flexibility, the machine will generally show comparable seismic and shaft vibration levels so that monitoring the seismic vibration, in addition to the shaft vibration, is warranted. In fact, many gas turbine manufacturers use seismic vibration as the main input to their vibration monitoring / pro-

tection systems to provide automatic shut down.

Moving on to rolling element bearings equipped machines, transducer selection becomes easier. Here our primary concerns are using a transducer that captures the frequency range of interest, and mounting the transducer where it sees the best transmission of bearing related signals. This usually means as close to the shaft centerline as possible, avoiding the transmission losses experienced across each mechanical joint. Accelerometers are generally used by most analysts to measure seismic vibration. Accels can also be used to gather vibration data from journal bearing caps, and from machine supports, foundations, frames, piping, etc..

On a typical motor-pump unit we would mount accelerometers horizontally and vertically (X-Y) at each of the four radial bearings. We would also mount transducers axially on the drive-end bearing of each component to monitor axial vibration. And we would likely place some transducers on the baseplates or mounting frames to measure vibration there, and to detect any differences between the frame and machinery.

## Configuration & Sampling Guidelines

There are sampling considerations in transient data collection that must be considered, which affect the quality & reliability of the acquired data. And with the limited opportunities to acquire transient data that we generally have, it is important to understand each one, and its effect on the quality of the data. Some of the major considerations would include:

### Machine Speed Range

The total speed range over which data must be sampled, and whether speed will oscillate during the transient, is our first consideration. We must know the total rpm range to properly determine how many samples will be gathered during a particular run.

On motor-driven machinery, this is easily identified by looking at the nameplate data. Typical AC motor full speeds on a 60Hz electrical systems would include 900, 1200, 1800 and 3600 rpm. Since the motors are ON or OFF, the startup and shut down process and the RPM range is well defined.

On variable speed machinery such as a steam turbine, the upper and lower speed limits are easy to identify, but the actual startup process can be quite drawn out. For example, consider a steam turbine driving a 60 Hz AC generator. The total nominal speed range would be 3,600 rpm. But, during a cold startup operators must typically hold turbine speed at several points during startup to allow the turbine casing and rotor temperatures to equalize and avoid high differential expansion conditions. During the hold points the rotor speed will often vary over a 50 – 200 rpm window, and the startup may take several hours to accomplish. And, it is common after a major outage to experience rotor rubs as new seals ‘rub in’. This, and other problems, often result in multiple false starts before the unit can finally be successfully brought up to synchronous speed.

### $\Delta$ RPM & $\Delta$ Time Sampling Intervals

In conjunction with the Machine Speed Range, we must establish a  $\Delta$ RPM interval for our transient sampling. These two items will determine the final size of our database.

If we use an 3,600 rpm AC motor as an example, and we sample with a  $\Delta$ RPM of 5, then a total of  $(3600 / 5) = 720$  samples will be acquired.

However, if we consider a steam turbine under cold startup conditions, we might be looking at something like this:

- Ramp up from 0 to 500 rpm; heat soak at 500 rpm for 1 hour
- Ramp up 500 to 1,000 rpm; heat soak at 1,000 rpm for 30 minutes
- Ramp up to 1,000 to 2,500 rpm; heat soak at 2,500 for 1.5 hours
- Ramp up from 2,500 to 3,600 rpm

In this case we would want to capture the pure  $\Delta$ RPM samples, but we should also capture  $\Delta$ Time samples during the heat soak periods as vibration conditions may change significantly while the turbine is warming up. In this case, we would recommend a  $\Delta$ RPM sample rate of 5 - 10 rpm, and also using a  $\Delta$ Time sample rate of perhaps 20 seconds. So our total sampling for the startup would be the sum of the  $\Delta$ RPM and  $\Delta$ Time as follows:

$(500 \text{ rpm} / 5) =$	100
$(60 \text{ min.} * 3 \text{ s./min}) =$	180
$((1,000 - 500) / 5) =$	100
$(30 * 3) =$	90
$((2500 - 1,000) / 5) =$	300
$(90 * 3) =$	270
$((3,600 - 2,500) / 5) =$	<u>220</u>
<b>Total samples =</b>	<b>1,260</b>

Keep in mind this total does not take into account any speed variations that usually occur during the heat soak periods. These would get sampled at the  $\Delta\text{RPM}=5$  rate, and it would not be unusual to capture 200 or more additional samples, bringing the total to at least 1,460 samples.

The above calculations might not seem important as many systems can utilize your entire free hard drive space for the database. But there is often a database size specified during software configuration that actually limits the number of samples acquired before a new database is started. You want to avoid starting a new database in the middle of a transient run as a significant portion of the run will be missed, and your data plots may not be able to concatenate the two databases. We will review these settings in our examples that follow.

Another consideration: if rubs or other problems during startup required the unit to be shut down before being restarted – and we wanted to show all of this data in one database, which is advisable – then the database size could easily double.

We generally like  $\Delta\text{RPM}$  sampling rates of 5 to 10 rpm for most machinery. This produces high quality data plots, while keeping database sizes reasonable. For  $\Delta\text{Time}$  sampling during startup we have considerable latitude to choose a rate appropriate with our desired data density. From a practical standpoint, unless process conditions are changing rapidly, there is little to be gained beyond 3 or 4 samples per minute.

As a final consideration, the  $\Delta\text{RPM}$  rate also needs to take into account the Ramp Rate and Frequency Span settings relative to the time required to gather each individual sample. These factors will determine the final quality of our transient data.

### **Ramp Rate vs. Frequency Resolution**

In conjunction with the speed range,  $\Delta\text{RPM}$  and  $\Delta\text{Time}$  sample rates, we must consider the rotor acceleration or “ramp rate” during startup and shut down. If ramp rates are too fast relative to our data acquisition settings, we may have poor quality data and/or miss data samples entirely. From a data acquisition standpoint, the worst situation generally occurs on AC induction motor startups.

On typical AC motors the startup will be very fast, with the rotor accelerating quickly and smoothly from zero to full speed. The startup will only last perhaps 10 – 40 seconds after the breaker is closed. While there are calculations that can be done to determine the total startup time, they are neither practical nor necessary for the vibration analyst.

As an example, consider a 3,600 rpm motor that accelerates to full speed in 40 seconds. That produces an average ramp rate of  $(3600 / 40) = 90$  rpm per second.

If we attempt to sample at a  $\Delta\text{RPM}$  of 5, we would be expecting our system to capture  $(90 / 5) = 16$  samples per second.

So, what can we expect from our data? To answer this we must look at our data acquisition settings for frequency span ( $F_{\text{max}}$ ), the lines of resolution (LOR), and the number of averages used per sample. These three factors are interrelated and determine the time required per sample.

As an example, when using the ZonicBook system for our 3,600 rpm motor above, if we were to set an  $F_{\text{max}}$  of 2,000 Hz, with 1,600 lines of resolution, the time required to capture 1 sample would be 0.8 seconds. Considering our average ramp rate from above, this means that in the 0.8 seconds required to capture a sample, that motor speed would have changed by  $(90 \times 0.8) = 72$  rpm. So, when we started acquiring any given sample, by the

time the sample was finished the machine speed would have increased 72 rpm.

If think about our spectral data in those terms, our 1X peak would have ‘moved’ 72 rpm during the sample. This would in effect give us an inaccurate picture of the 1X response during that sample.

If we change the Fmax to 1,000, guess what happens? The sampling time doubles to 1.6 seconds.

Now, if we keep Fmax at 1,000 Hz and then decrease LOR from 1,600 to 400, we have a sample time of 0.4 seconds per sample. For these settings the machine speed would change by  $(90 \times 0.4) = 36$  rpm per sample. Decreasing LOR further to 200 produces a sample time of 0.2 seconds, during which speed will change only 18 rpm during the sample. Keep in mind that at Fmax=1,000 Hz with 200 LOR, our frequency resolution will only be 5 Hz.

We can see from this that data acquisition time is proportional to LOR, and inversely proportional to Fmax. This is not unique to the ZonicBook system, it is a function of the digital data waveform sampling where the sampling time =  $LOR / Fmax$ .

So what should be done on induction motor startups? We normally will set a  $\Delta RPM$  of 20 – 30, with an Fmax = 1,000, and LOR = 200. This will yield a reasonable number of samples during startup, and allow us to track the 1X responses for resonance evaluation.

Naturally, motors controlled through Variable Frequency Drives and Wound-Rotor AC motors can be brought up to speed in a more controlled manner, and our typical  $\Delta RPM$  of 5 – 10 rpm, in conjunction with Fmax=1,000 and LOR = 400 or 800 would likely be sufficient.

During a motor shut down we might expect a relatively long period for the rotor to coast down due to the mass of rotating elements. This will hold true for fans and compressors, where the air does not present much resistance. In those cases we keep our Fmax and LOR reasonably high for good data resolution.

If we are analyzing pumps we will find that they will decelerate quickly due to the fluid within the pump casing. Experience will be your best guide, but it is advisable to not use a LOR that is higher than needed for the shut down. Typically 400 or 800 LOR will prove adequate.

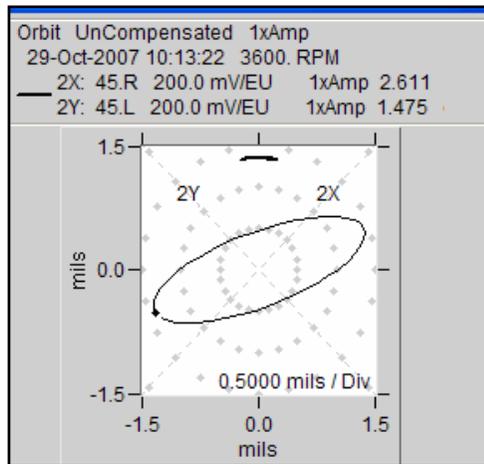
Looking at steam turbine driven units during startup, the ramp rates are typically held to 100 – 300 rpm per minute. It is rare to encounter a situation where an Fmax of 1,000 or 2,000 Hz coupled with a LOR of 800 or even 1,600, respectively, would not provide good data.

Similarly, the shut downs are generally very slow, with large units often taking 15 – 30 minutes to coast down completely. For 60 Hz turbine-generator shut downs we generally like to use an Fmax = 500 Hz with LOR=800. This produces excellent quality waterfall and cascade plots for transient analysis, which we discuss shortly.

### Channel 'Pairs'

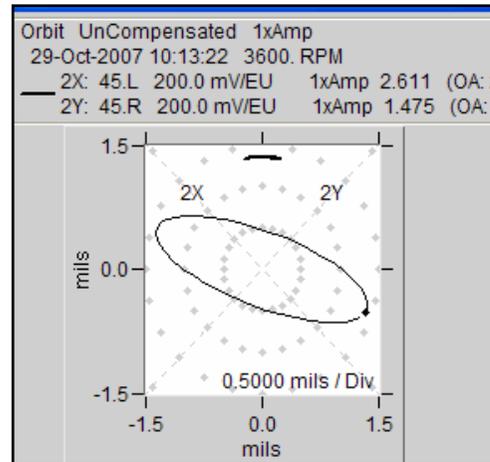
To produce certain types of data plots – orbits, shaft centerline, and full spectrum – it is necessary to have a pair of transducers mounted in an ‘X-Y’ configuration at a given location. This generally applies to proximity probes, where we are interested in the orthogonal shaft motion. Probes are typically mounted 90° apart from each other, and are usually located ±45° from top-dead-center at each bearing, although the 90° probe pair can be rotated to any position that is mechanically expedient for installation.

It is important that the actual probe installation angles are correctly specified during software configuration. These angles will then determine the orientation of the associated data plots. For example, consider the 1X-filtered orbit plot below, which was correctly configured with the probes at 45° to the right & left from top-dead-center:

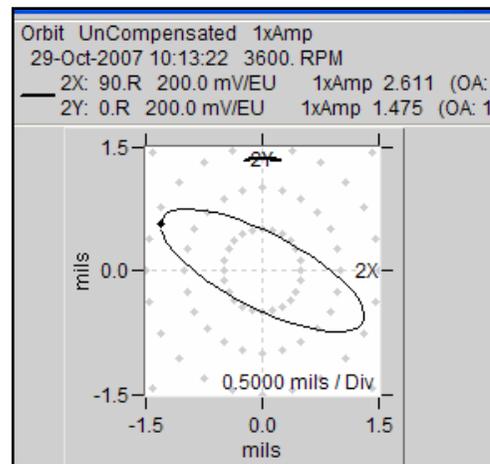


We see the orbit is elliptically shaped, with the major axis being oriented up and to the right. Also note the probe names (2X and 2Y) are shown on the plot at the probe angles, and the plot header contains the angle data, as well as the probes’ scale factors.

If the same channel pair was incorrectly configured by reversing the probes angle, the 1X-filtered orbit plot would be as follows:



Note that the orbit’s is now a mirror image of the original plot. As a final variation, see what happens if we place the X probe at 90°Right, and the Y probe at top-dead-center:



Notice the difference in the two plots above. They are almost identical except for the small dot & blank spot along the outside of the orbit. This is the reference mark for phase angle measurement, and occurs when the tachometer signal is triggered. Note that the two plots above show the blank/dot sequence to be in nearly opposite positions. In terms of analysis, this would produce different phase angle readings of 180°. And more to the point – an analysis of these three orbits would lead to drastically different machinery condition conclusions. Always verify your installed probe angles, and insure your wiring and software configuration is correct.

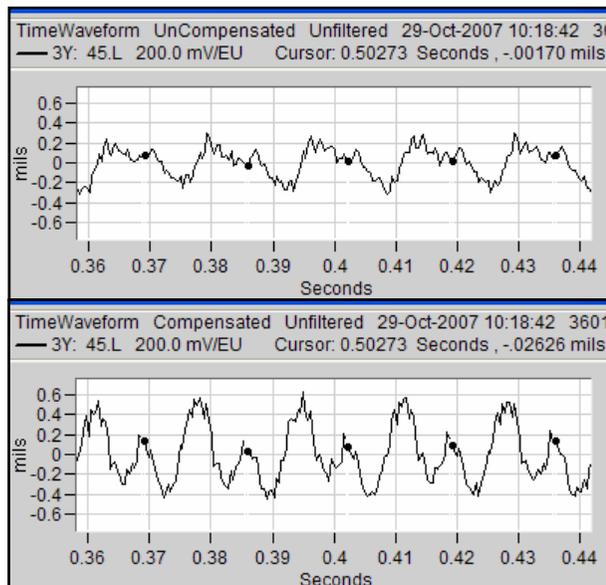
### Additional Notes for Journal Bearings

Before discussing transient data plots there are several important items regarding journal bearing analysis with proximity probes that should be reviewed. Note these items do not generally apply when using accelerometers or velocity probes.

#### Slow Roll / Runout Compensation

Slow-roll refers to the mechanical and electrical runout in the target area of a proximity probe. These defects create a non-dynamic 'false' vibration signal that adds to the true dynamic vibration at any speed. Unfortunately, a proximity probe cannot distinguish between the runout and true dynamic motion.

For most turbo-machinery, if we sample vibration at low speeds, typically below 300 rpm, we can be reasonably sure that there will be little dynamic shaft motion. The measured signal will contain the runout of the probe target area. Most data acquisition systems then allow the user to store this runout signal and have it digitally subtracted from any at-speed vibration. The differences can be dramatic, as shown below in the uncompensated waveform (on top) and the compensated waveform:



Looking at the uncompensated waveform, we see a predominant 1X frequency, with small amounts of 2X and higher order harmonics. However, once compensated the 2X vibration becomes particularly noticeable, with 2 strong peaks per revolution. Notice also that the peak-to-peak amplitude actually increased after compensation. This is because the subtraction is done in terms of vector relationships.

To do a vector subtraction, we first add 180° to the phase angle of the component being subtracted, and then add the two vectors. If the phase angle for a runout component is initially out of phase with the at-speed signal, when we add the 180° it will become additive with the at-speed signal. This is an important relationship that many analysts do not fully understand, or utilize in their analysis.

We strongly recommend that slow roll data be sampled as the machine is coasting down, after being operated on-line for some time. This helps ensure the rotor has thermally expanded to its normal operating location (axially), while the shut down places the rotor in a nearly torque-free state. If slow roll data can only be acquired during startup, it should be viewed with some caution. The effects of startup torque and a cold rotor will generally not yield the same runout pattern that the machine has once it is running on line and has thermally stabilized.

It should also be noted that runout compensation can be performed on the overall vibration signal, as shown here, and any individual vectors, such as 1X. When balancing a rotor, the slow roll compensated 1X vector provides the correct information regarding the vibration amplitudes and the balancing process. As a practical note, slow roll amplitudes much below 0.2 mil-pp can be largely neglected except in high speed applications where tolerances are much tighter.

## Transient Data Plot Types

In addition to the usual spectrum and time-waveform data plots used during steady state analysis, there are several other plot types available for transient vibration analysis. These plots provide views into the machinery dynamics that are not apparent in steady state analysis, and greatly enhance our ability to diagnose machinery faults.

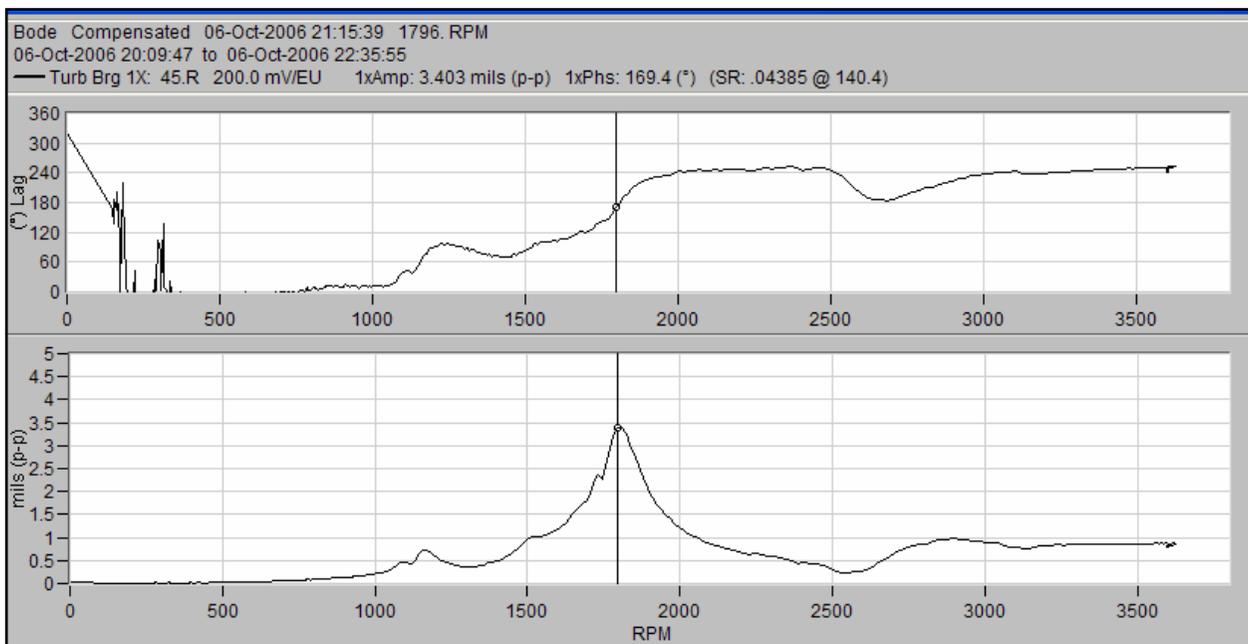
### Bode Plots

The Bode plot is an excellent point to begin discussing transient data. It is a single-channel plot commonly used to plot the 1X filtered vibration amplitude and phase lag angle versus speed. The figure below shows the response from 'X' probe of the outboard bearing of a steam turbine bearing.

Note that this plot has been compensated for slow roll, as indicated in the plot header information. To select the proper speed for our slow roll data, we first viewed the uncompensated bode plot and selected a low speed region where there were no significant amplitude or phase changes occurring. This data was then stored and available to compensate any further data in the database

Bode plots are typically made to show the 1X vector response. They help provide the following information:

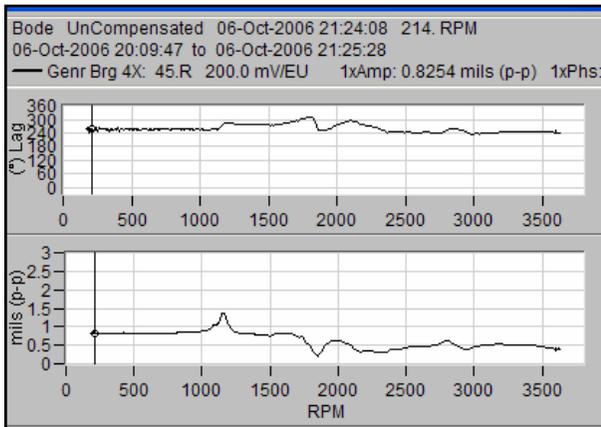
- The proper speed range for selecting slow roll data for slow roll compensation
- The “High Spot”, i.e., the rotor’s vibration response
- The “Heavy Spot”, i.e., the physical location of a residual unbalance directly on the rotor
- The location (speed) of rotor and structural resonances
- The presence of ‘split’ resonances
- The amplification factor (‘Q’) and damping ratio for any single mode / resonance
- The separation margin between resonances and operating speed
- The difference in vibration amplitudes between the unfiltered and filtered signals



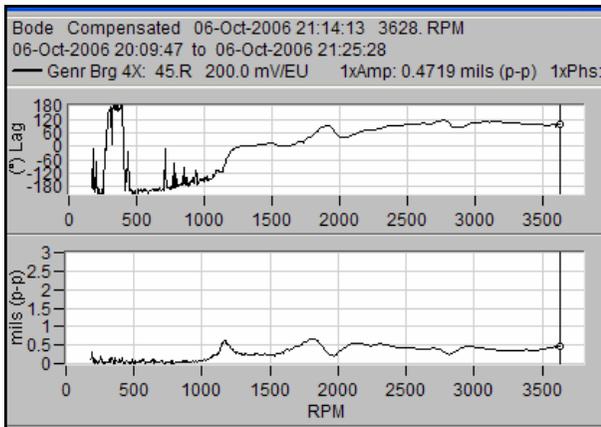
Slow Roll Data Selection

As noted above, it is important to select the proper speed range for slow roll data storage to ensure the at-speed data is properly compensated. Regardless of the operating speed of the machinery, the key characteristics that we seek are little to no amplitude and phase changes in the low rpm region of the bode plot.

The bode plot below shows the uncompensated 1X data from a proximity probe. Note that from 750 rpm to the minimum speed of 214 rpm there is practically no change in the vector’s amplitude or phase angle. We can select a sample anywhere in this region, and receive good compensation results.



From a practical standpoint, it is usually preferable to choose a point from the lower speed area of this region, say between 215 and 500 rpm. Using the data at 214 rpm for compensation produced the following plot:



So how do we know we selected a good sample for performing compensation? By examining the amplitude curve, we see the data has been reduced to almost zero amplitude. Note that some noise is now present in the amplitude trace, and the phase appears unstable until about 500 rpm is reached. This occurs because the amplitude is too low for the instrument’s tracking filter to achieve a good lock on the trace. As soon as speed begins to increase above 500 rpm and the vibration begins increasing very slightly, the tracking filter is better able to resolve the amplitude and phase.

“High Spot” & “Heavy Spot”

The term “High Spot” and “Heavy Spot” are primarily related to rotor balancing.

Heavy Spot describes the physical location of a mass unbalance on the rotating element. If we consider a rotor with a disk that is operating well above its first lateral balance resonance (or “critical”), the Heavy Spot will lag behind the High Spot by 180°. When we balance a rotor we are correcting for the Heavy Spot.

High Spot is the phase angle measured by our data acquisition instruments. It simply refers to the rotor response phase angle at any give speed, i.e., the direction of the rotor vibration, relative to our once-per-turn tachometer signal.

Resonances and Amplification Factors

All rotors have natural frequencies of vibration, or resonances. When the rotor operates at this frequency the resonance will be excited by any residual unbalance in the rotor, or any 1X driving forces or moments such as results from misalignment. It has been the industry standard for many years to call these resonance conditions “critical speeds” due to the sometimes excessive vibration that is experienced at resonance.

As we saw in the bode plot on page 15, at low speed we had little to no vibration. As speed increased above 1,000 rpm the vibration began increasing in earnest. At 1,810 rpm it reached the ‘critical’ – which is more correctly termed the rotor’s 1<sup>st</sup> lateral balance resonance. At resonance we saw the characteristic amplitude peak and 90° phase lag increase in the High Spot, compared to low speed operation. This 90° shift indicates that the High Spot is now lagging behind the Heavy Spot by 90° as the shaft rotates.

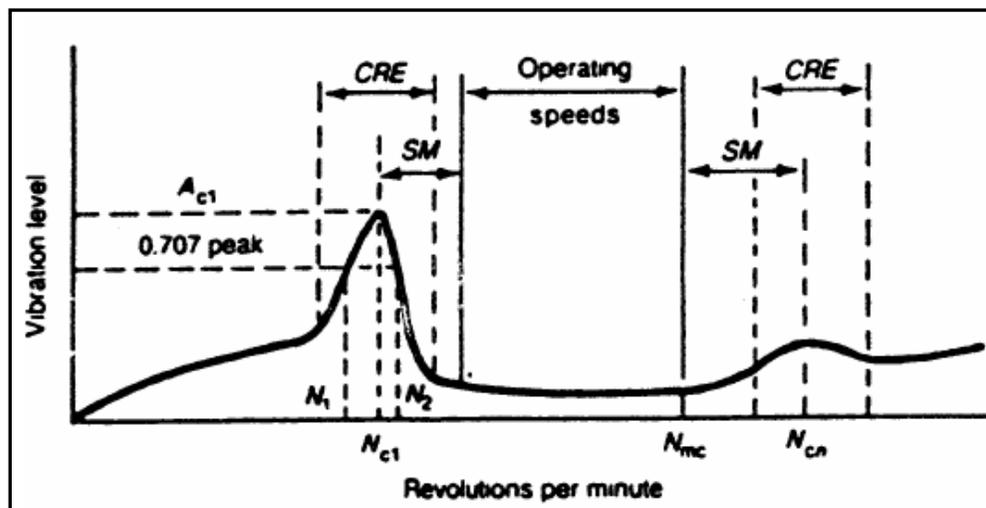
As speed continued to increase above the 1<sup>st</sup> resonance the amplitude subsided and the phase continued to lag. If we proceed far above the 1<sup>st</sup> resonance, the phase angle change approaches a full 180° shift from low speed, provided other resonances and modal effects do not begin to dominate the response.

At resonance the main controlling force on a rotor is the damping presented by the lubricating fluid within the bearings. If a rotor is said to be ‘well damped’, it will generally exhibit low amplification at resonance, and the amplitude peak will be broad. Conversely, a ‘poorly damped’ resonance would show a sharp, high amplitude peak. In either case a 90° phase change from low speed would still be present.

This amplification at resonance is a concern both for machine designers and analysts. From an analyst’s standpoint we can measure the existing amplification and use it both as a trending and a diagnostic tool. The graph below is taken from the American Petroleum Institute (API) Specification 617 for rotating equipment in refinery service, but it is applicable to any rotor. The nomenclature is as follows:

- Nc<sub>1</sub> = Rotor first critical speed, cpm
- Ac<sub>1</sub> = Peak Amplitude at Nc<sub>1</sub>
- N<sub>1</sub> & N<sub>2</sub> = Speeds at half-power points
- AFc<sub>1</sub> = Amplification factor  
= Nc<sub>1</sub> / (N<sub>2</sub> - N<sub>1</sub>)
- CRE = Critical response envelope
- SM = Separation margin

Note the half-power points are calculated as the peak amplitude, Ac<sub>1</sub> x 0.707.



The bode plot from page 15 is repeated below with some pertinent annotations. We found the resonance at 1,810 rpm, with 3.39 mil-pp of vibration, and showing the necessary 90° phase shift from low speed operation.

To calculate the Amplification Factor, the Half-Power amplitudes were calculated as:

$$(3.39 \times 0.707) = 2.4 \text{ mils}$$

The speeds  $N_2$  and  $N_1$  were then found by dropping down to the speed axis at each half-power point, indicating 1,870 and 1,750 rpm, respectively. The Amplification Factor and Separation Margins were then found:

$$AF = 1,180 / (1,870 - 1,750) = 15.1$$

$$SM = (3,600 - 1,810) / 3600 = 49.7\%$$

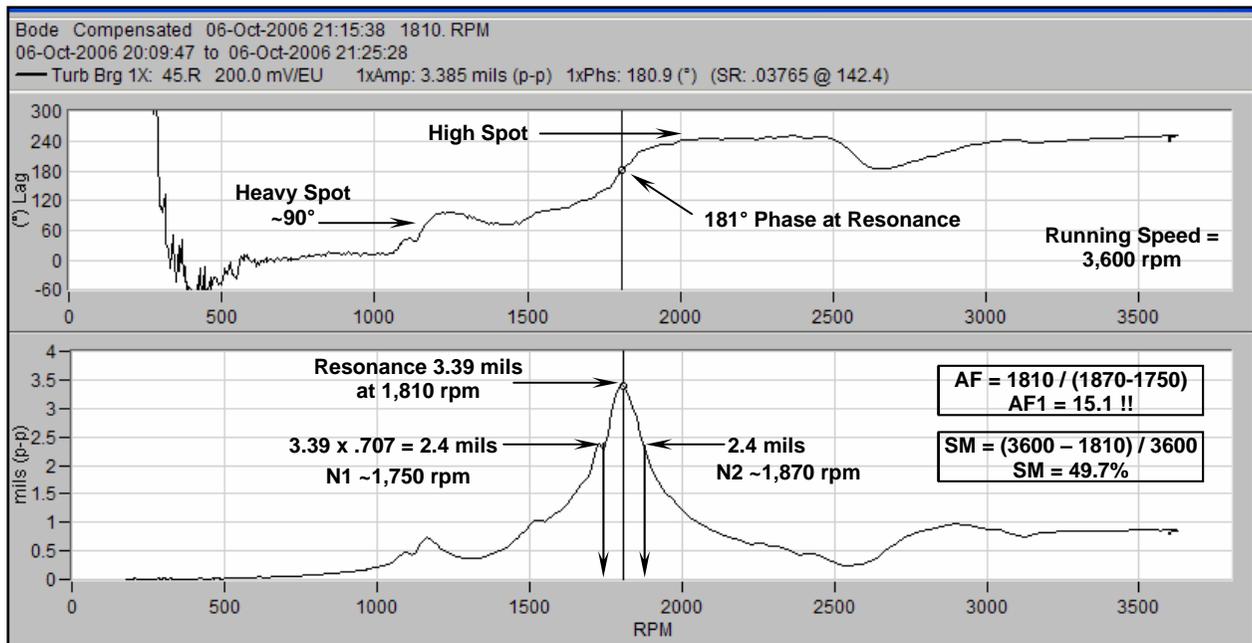
The Critical Response Envelope is not shown for clarity.

The AF of 15.1 represents a poorly damped rotor system that will be subject to high vibration if the rotor becomes unbalanced. Minor changes in balance could easily result in vibration of 10 mils or more.

API considers a system to be “critically” damped when the amplification factor is less than 2.5. Following are some of the API acceptance criteria for a damped unbalanced rotor analysis:

- $AF < 2.5$ : the response is considered to be critically damped and no Separation Margin is required.
- $AF = 2.5 - 3.55$ : Separation Margin of 15% above maximum continuous speed, and 5% below the minimum operating speed, is required.
- $AF > 3.55$  & critical response peak is below the minimum operating speed: the required Separation Margin (a percentage of minimum speed) is equal to the following:  $SM = 100 - \{84 + [6/(AF-3)]\}$
- $AF > 3.55$  & critical response peak is above the trip speed: The required Separation Margin (a percentage of maximum continuous speed) is equal to the following:  $SM = \{126 - [6/(AF - 3)]\} - 100$

So for our rotor, with  $AF = 15.1$  and critical response peak below running speed, the required  $SM = 100 - \{84 + [6/(15.1-3)]\} = 15.5\%$ , which we easily achieve – thankfully – due to the high amplification factor involved.



On critical machinery the Amplification Factors should be periodically calculated and trended during normal shut downs. For systems with poor damping (high AF) and/or that operate close to resonance at running speed, data on damping is important. Significant changes would be directly related to the condition of the bearing and lubrication system and warrant investigation.

It is important to note this approach does have accuracy limitations if the damping is low or the shape of the resonance curve is significantly influenced by adjacent modes or other factors, such as structural resonances. One major problem in turbomachinery that affects the usefulness of these calculation is the presence of a rotor to seal rub. This will effectively create a ‘flat-top’ amplitude peak, with a broader than normal resonance envelope, while also distorting the phase response. The flattened peak and broadened response will produce artificially low Amplification Factors. It is important that the data be free from such influences.

Another method for evaluating damping for a single mode of vibration using the half-power point data is to calculate the damping ratio as follows (Ehrich):

$$Q = AF = \frac{N_c}{N_2 - N_1} = \frac{1}{2(c/c_c)}$$

or,

$$\frac{c}{c_c} = \frac{1}{2AF}$$

where:

- Q = Quality Factor
- AF = Amplification Factor
- N<sub>c</sub> = Rotor Critical Speed
- N<sub>1</sub> & N<sub>2</sub> = Speeds at half-power points
- c/c<sub>c</sub> = Damping Ratio

For our previous data, we would calculate a damping ratio of:

$$c/c_c = 1 / (2 \times 15.1) = 0.033$$

Damping ratios vary from 0 (no damping) to 1 (no vibration). Typical damping ratios are as follows:

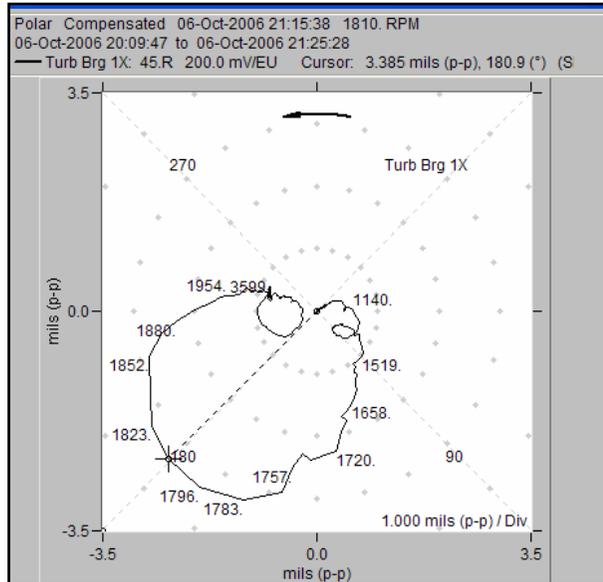
Steel.....	0.001
Rubber.....	0.05
Rolling-Element Bearing Machines.....	0.025
Fluid-Film Bearing Machines.....	0.03 – 1.0

We can see that our damping ratio of 0.033 falls on the low range for fluid-film bearing machines.

Note that rolling-element bearing machines have very low damping by design. This foretells two things. First, they are generally designed to operate below rotor resonance. Because of the minimal damping inherent in rolling element bearings, they cannot effectively limit vibration during resonance, readily transferring it to the structure. And second, because they generally operate below resonance, we often will find the Heavy Spot and High Spot in close proximity to each other, and can estimate corrective balance weight locations with relative ease.

### Polar Plots

Polar plots present the same information as a bode plot, graphing amplitude versus phase. However, the data is plotted in polar coordinates rather than XY form, as shown below:



While the polar plot shows the same information as the bode plot, it is generally preferable for analysis and field work for several reasons:

- The plot is oriented to the angle of the vibration probe, and referenced to the machine casing. You can, quite literally, hold a sheet of paper that has the polar plot printed on it up to the end of the shaft, and directly see the probe orientation, the direction of rotation, and the direction of the rotor's responses.
- The High Spot and Heavy Spot have immediate physical meaning, being directly transferable from the plot to the machine.
- We can easily compensate any rotor resonance mode for runout and previous vibration.
- We can easily differentiate between rotor and structural resonances.

- We can establish the rotor mode shape to determine the ideal location for balance weights, and to assist with diagnostics.

More easily interpreted

The polar plot is used frequently for balancing. It directly shows the location of High Spot & corrective balance weight locations are easily determined.

More easily interpreted

Oriented to probe angle

Rotor resonance

Structural resonance

Amplification Factor / 'Q'

Phase lag plotted opposite the direction of rotation

Mode shape

Slow roll

Precession

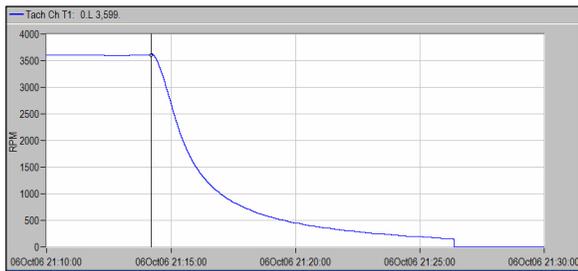
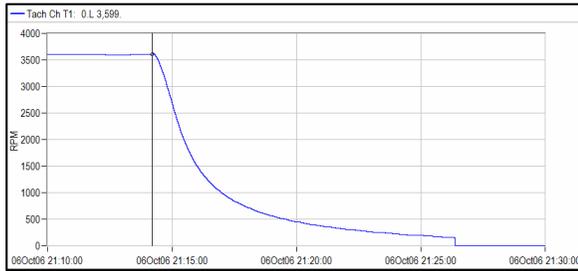
Split Resonance

Most resonance problems will manifest themselves as high or increased 1X vibration. If an analyst only uses steady state data and sees a large 1X component, they may believe they just have an unbalanced rotor. This may be true in many cases, but we have also many cases of high 1X resulting from excitation of a rotor or structural resonance that coincides with running speed or some other primary forcing frequency. When that occurs, any residual rotor unbalance is amplified via the resonance.

In solving a high 1X vibration problem, we might balance the rotor. But, if a rotor resonance were present near running speed, our balancing would need to be approached carefully due to amplification and rapid phase changes near resonance. And if the high 1X were instead caused by excitation of a structural resonance, we might want to consider structural modifications to move the resonance away from running speed. We therefore like to discern between any potential resonance issue and a pure non-resonance unbalance before deciding on a particular solution.

### Speed vs. Time

### Vibration vs. Speed





## **Cascade Plots**

Spectra versus speed

Horizontal and vertical relationships

Order lines / harmonic activity

Rotor vs. structural resonance

Instability

Cracked shaft

Looseness

Notes: spectra every sample versus every 10<sup>th</sup> sample.

**Waterfall Plots**

Waterfall plots are three-dimensional graphs of spectra at various machine speeds and times.

lplots Spectra vs. time

Different format from cascade, same data

Different view on the data

**DC Gap vs. RPM Plots**

Proximity probes only

Thrust probes

Single axis proximity probe

Gap reference voltage

### Shaft Average Centerline Plots

Proximity probes only

The shaft average centerline plot shows the movement of the shaft within the bearing, and is typically plotted in reference to the available diametral bearing clearance.

Used to confirm 'Hot' Misalignment and indicate if Optical Alignment is required

Need accurate bearing data

Hot shut down data should be sampled whenever possible. Cold startup data is OK but can be subject to significant error.

Gap reference voltages

Where to begin the plot

On large turbo-machinery, especially power generation units with very heavy rotors, the lube oil system will generally be equipped with a high-pressure jacking oil pump. This pump provides high-pressure oil directly beneath the rotor at low speeds, because at low speed a sufficient oil wedge will not form beneath the shaft to prevent bearing wear. The jacking oil will serve to lift the shaft a few mils off the bottom of the bearing. As speed increases and the oil wedge forms, the jacking oil will be turned off, generally around 600 rpm on most large turbine generator sets.

Oil is pumped into the bearing during operation via a lube oil system to provide lubrication between the two surfaces, and to minimize wear of the relatively soft babbitted bearing surfaces. Due to the oil's viscosity and its adhesion to the journal and babbitt, the rotating shaft drags the oil around within the available clearance, and the familiar 'oil wedge' is formed beneath the shaft. As the wedge develops with increasing speed, it lifts the shaft

within the bearing, and the shaft will move laterally to the left or right.

The direction of shaft rotation dictates where the oil wedge is formed and how the shaft will move in response to it. For clockwise rotation, we normally expect the wedge to develop along the bottom-right side of the shaft, pushing the shaft up and to the left. Conversely, for counter clockwise rotation, we expect to see the shaft move up and to the left. The diagram below shows these relationships:

*insert sketch/diagram showing wedge*

The observed shaft movement within a bearing is a function of rotor speed (rpm), bearing design, and lube oil pressure.

At low speeds, typically below 300 rpm on most machinery, the shaft will be very nearly in the bottom of the bearing, and riding on a thin oil film of perhaps 1 – 3 mils (0.001" – 0.003"), with little to no oil wedge present. This position will serve as our reference position during analysis.

As speed is increased and an oil wedge forms, the resulting wedge and the pressure it places on the shaft will increase as a function of speed. This will lift the shaft, as noted above, with most bearings showing the shaft to rise about 1/4 to 1/3 of the available clearance from the resting position.

For example, if we have a shaft journal measuring 8.000", we might expect the bearing inside diameter to be approximately 8.012", thereby providing a diametral clearance of 12 mils<sup>6</sup>. And we would expect the shaft to rise perhaps 3 – 4 mils from the reference position as full speed is reached (and to move laterally a few mils).

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<sup>6</sup> Typical turbo-machinery bearing design uses diametral clearances of 0.015" (1.5 mils) per inch of shaft diameter.

With the exception of tilt-pad bearings, most journal bearings will show the movement pattern noted above. Tilt-pad bearings, especially those with a between-pad loading, often show the shaft rising straight-up from the bottom of the bearing.

The expected positions noted above will then be influenced, often drastically, by the presence of misalignment, and **whether** or

DC gaps when stopped

Hot vs. Cold DC Gap

***Knowing design or last available bearing diametral clearances.***

## **Orbit Plots**

## **Identifying Machinery Problems**

***Shaft Runout***

***Bowed Rotors***

***Resonance***

## **Case Histories**

x

## **Bibliography / References**

Ehrich, Fredric F.; Handbook of Rotordynamics; ISBN 0-07-019330-4; McGraw-Hill Inc., 1992.

### **Time Investment:**

15 – 22 September: 2.5 hrs

Sunday, September 23, 2007: 2 hrs

Saturday 12/1: 5 hrous

Sunday 12/2: 6 hours

Monday 12/3: 4 hours