HEWLETT® PACKARD

Effective Machinery Measurements using Dynamic Signal Analyzers

Application Note 243-1

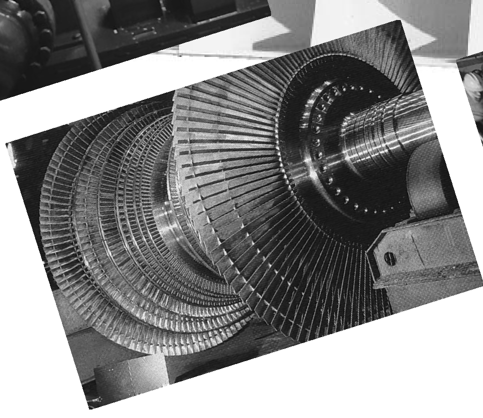


Table of Contents

Chapter 1
Introduction

The analysis of machinery vibration is characterized by a number of distinct application areas. In evaluating machinery vibration its paramount to ask “What is the purpose of the measurement?”. In general, the analysis will fall intoone of three distinct categories:

1) Product/machine research and development

2) Production and quality control (this includes rebuilding and overhaul)

3) In service maintenance and monitoring

In the cases of these differing application categories; the general principles and measurements are often the same, but the performance characteristics, measurement flexibility functionality and data presentation formats can vary.

The implementation of machinery vibration analysis has been made practical by the development of analysis instruments called Dynamic Signal Analyzers (DSAs). Machinery vibration is a complex combination of signals caused by a variety of internal sources of vibration. The power of DSAs lies in their ability to reduce these complex signals to their component parts. In the example of Figure 1-2, vibration is produced by residual imbalance of the rotor, a bearing defect, and meshing of the gears — each occurring at a unique frequency. By displaying vibration amplitude as a function of frequency (the vibration spectrum), the DSA makes it possible to identify the individual sources of vibration.

Figure 1-1 Dynamic Signal Analysers are (DSAs) are the ideal instrument for analyzing machinery vibration.

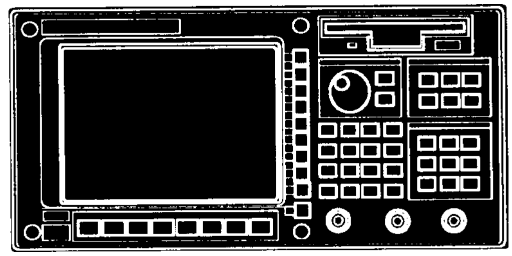
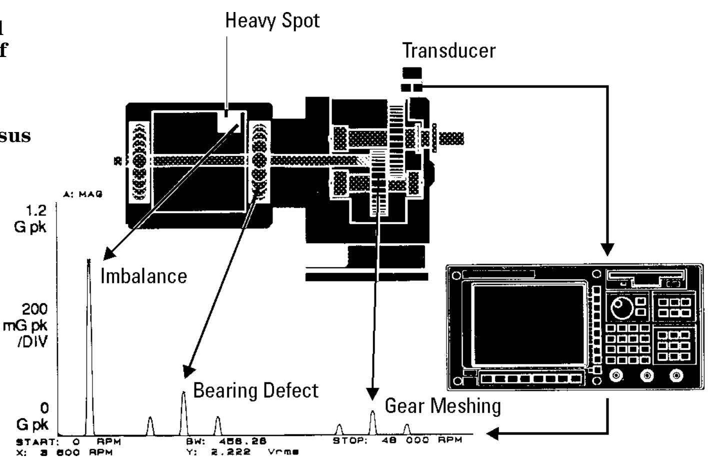


Figure 1-2 The individual components of vibration are shown in DSA displays of amplitude versus frequency.



Dynamic Signal Analyzers can also display the vibration amplitude as a function of time (Figure 1-3), a format that is especially useful for investigating impulsive vibration (e.g. from a chipped gear). The waterfall/spectral map format (Figure 1-4) adds a third dimension to vibration amplitude versus frequency displays. The third dimension can be time, rpm or a count triggered by an external event (e.g. load, delay from top-dead-center, etc.). DSAs come in many shapes, sizes and configurations. They range from stand- alone battery operated portables, to bench-top precision instruments, to rack-mounted computer controlled systems. They range from single-channel units up through multi-channel (~500) systems. Virtually all can be computer automated and controlled, and a wide variety of post-processing capabilities and programs are available.

This application note is a primer on analyzing machinery vibration with Dynamic Signal Analyzers. Each of the important steps in the analysis process from selecting the right vibration transducer to interpreting the information displayed is covered. The techniques described provide insight into the condition of the machinery that eliminates much of the guesswork from analysis, troubleshooting and maintenance.

1.1 Benefits of Vibration Analysis

The ability to analyze and record vibration data has existed for a considerable time. Its only recently with the advent of modern DSAs that the actual detailed analysis of vibration data has become widespread and effective.

Figure 1-3
DSA display of amplitude versus time are especially useful for analyzing impulsive vibration that is characteristic of gear and rolling element defects.

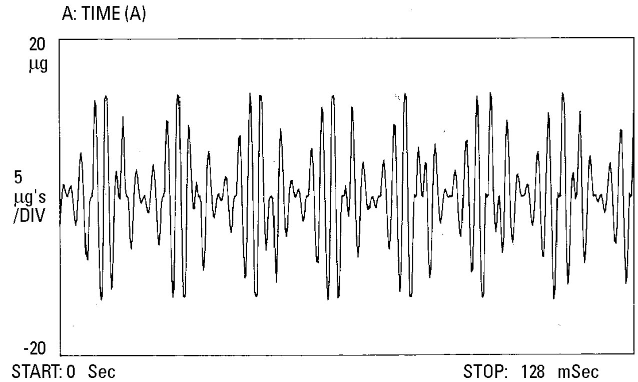
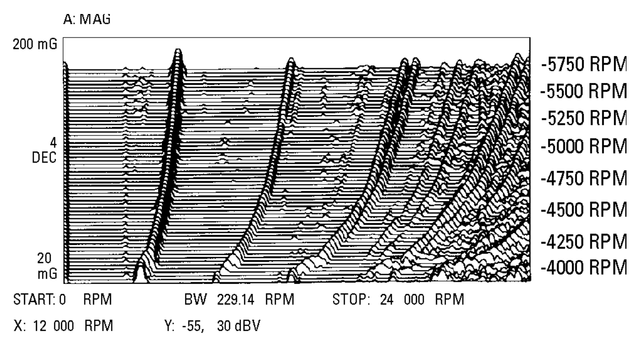


Figure 1-4
DSA map displays illustrate changes in vibration with rpm, load, or time. This map is a collection of vibration measurements made during a machine runup.



The principle objectives in analyzing the vibration data are:

1) Simplify and reduce the vibration data into a more compact easily interpreted form.

2) Associate characteristics of the vibration to specific features of the machine vibrating.

3) Provide a consistent, repeatable measurement by which to characterize the vibration of a machine.

4) Identify characteristics that change with time and operating conditions, or both.

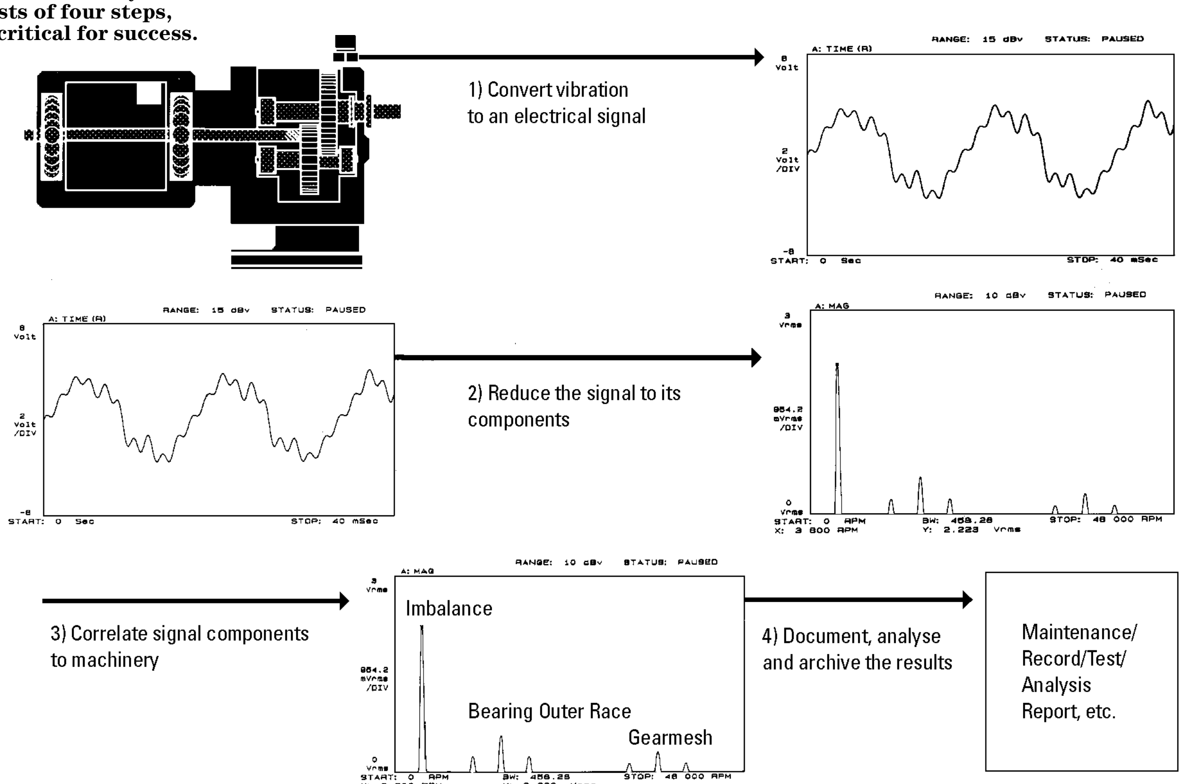
Figure 1-2 illustrates the principles presented; the vibration data is broken down into its individual frequency components by the DSA; the analysis can associate these components to particular elements in the machine and depending upon the objective, determine whether an individual component of the vibration is abnormal. The total energy in any single component is generally small and the ability of a DSA to individually segregate this component make it a very sensitive measure of the machine. Often a very large change in an individual component will cause an extremely small change in the overall vibration level.

1.2 Using this Application Note

This application note is organized around four key steps in the analysis process shown in Figure 1.2-1: (1) converting the vibration to an electrical signal, (2) reducing it to its components, (3) correlating those components with machine defects, and (4) documenting, archiving and analyzing the results. Each of these steps is vital to analysis, and viewing the process in this manner promotes a systematic approach that increases the probability of success. The contents of each chapter, and their relation to the steps in Figure 1.2-1 are discussed below.

Two subjects beyond the scope of this note are rotor dynamics and the vibration characteristics of specific types of machinery. Rotor dynamics is required for complete analysis of the rotors used in most turbomachinery (i.e. flexible rotors) although most of the information in this note still applies (we will note circumstances when it does not).

Figure 1.2-1
The process of machinery vibration analysis consists of four steps, each critical for success.



Understanding the vibration characteristics of specific types of machinery is important for effective analysis. This information can be obtained from machinery manufacturers, independent training centers, and from well documented experience with similar machines.

The analysis of machinery vibration is not an easy task, and you will not fully understand each and every measurement, nor will you easily predict the effects of changes or an impending failure. What vibration analysis does provide is a valuable tool to give you additional insight into the dynamics of a rotating machine, the ability to predict most failures and diagnose the cause of excessive vibration.

Chapter Overview

Chapter 2: *Converting Vibration to an Electrical Signal*
Vibration is converted to an electrical signal with transducers, and effective analysis requires a signal that accurately represents the vibration. This chapter gives you the information needed to select and mount transducers.

Chapter 3: *Reducing Vibration to its Components - The Frequency Domain*
The key to successful analysis is reduction of the complex signal to simple components. As shown in Figure 1-2, this is best done with a display of vibration amplitude vs. frequency – a perspective known as the frequency domain. The objective of this chapter is to provide a good working knowledge of the frequency domain.

Chapter 4: *Characteristic Vibration of Common Machinery Faults*
Each type of machine fault has distinctive characteristics that can be used for identification. This chapter describes the characteristics of some of the most common machinery faults.

Chapter 5: *Advanced Analysis and Documentation*
This chapter focuses on solving some of the practical problems encountered in machinery vibration analysis, such as identifying spectral relationships, order analysis, orbits, limit testing, automation and other advanced techniques.

Chapter 6: *Dynamic Signal Analyzers*
DSAs feature measurement capabilities that make them the ideal instrument for machinery vibration analysis. This chapter explains why these capabilities are important, describes key aspects of each and helps discriminate between the different analyzers ranging from single-channel up through large multi-channel systems.

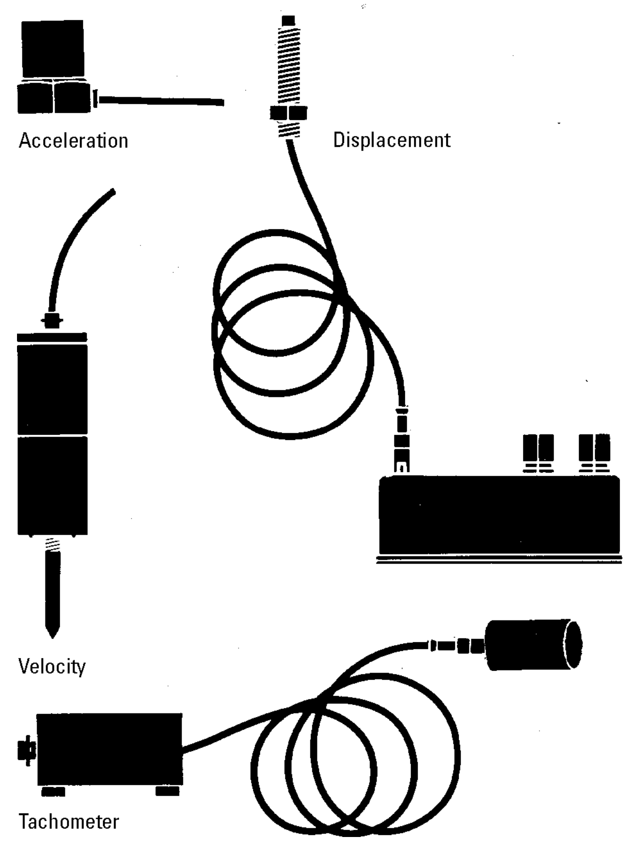
Chapter 2
Converting Vibration to an Electrical Signal

Before analysis can begin, vibration must be converted to an electrical signal — a task performed by vibration transducers. The key considerations in obtaining a signal that accurately represents the vibration are: (1) selecting the right type of transducer, and (2) locating and installing it correctly. The four types of transducers commonly used for machinery vibration are shown in Figure 2-1. They are differentiated by the parameter measured (i.e. displacement, velocity, or acceleration), and by the machine component measured (i.e. shaft or housing). Selection depends on the characteristics of the machine and its expected faults. Installation requires correct placement, secure mounting, and proper signal conditioning.

In addition to the motion transducer, for many measurements the operating speed of the shaft is of importance. The transducer used for this is called a tachometer; and provides a pulse type signal as opposed to the analog data normally found in motion transducers. The tachometer normally produces a fixed number of "pulses" per revolution which is in turn converted to a rotation speed by a frequency counter. Common types of tachometers include the use of the displacement probe and/or optical or magnetic sensors.

This chapter begins with a discussion of basic vibration concepts that are fundamental to understanding transducers and their installation. This is followed by a description of each of the three types of motion transducers and two common tachometer configurations. The final section of the chapter provides transducer installation guidelines.

Figure 2.1
Four types of transducers commonly used to convert machinery vibration to an electrical signal.



2.1 Vibration Basics

Before starting our discussion of the details of transducers and vibration analysis, it is important to establish some basic concepts. The three topics we will focus on are:

(a) Vibration Parameters.
Using commercially available transducers, we can measure the displacement, velocity, or acceleration of vibration. Selecting the right parameter is critical for effective analysis.

(b) Mechanical Impedance.
What we can measure with transducers is the response of the machine to vibration forces caused by machinery characteristics; not the forces themselves. The mechanical impedances of the machine shaft/rotor and housing determine how they respond to vibration forces and can alter significantly the characteristics of the signal we measure. These characteristics are often non-linear in nature.

(c) Natural Frequencies.
When a structure is excited by an impact, it will vibrate at one or more of its natural frequencies or resonance. These frequencies are important because they are often associated with critical speed of the machine, where residual imbalance excites the resonance. They can cause large changes in the vibration response with changes in rpm and are often associated with critical operation conditions.

Vibrations Parameters

We will start our discussion of vibration parameters by examining the vibration produced by simple imbalance. Referring to the machine rotor in Figure 2.1-1, note that the heavy spot produces arotating force that appears sinusoidal from any fixed reference position. At points A and C, the force in the direction of the reference is zero. At points B and D it is at positive and negative maximums, respectively.

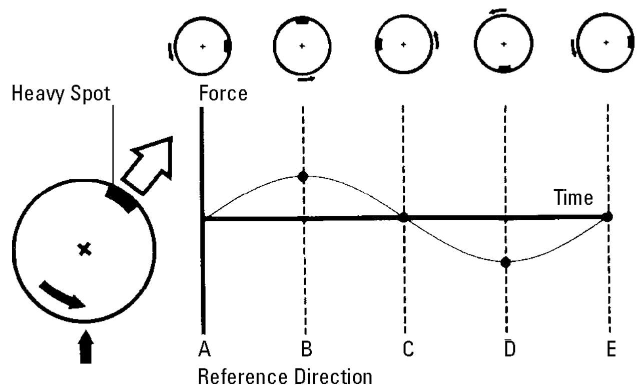


Figure 2.1-1
A heavy spot on a machine rotor results in a rotating force vector that appears sinusoidal from a fixed reference.

The response of the rotor to such a force is a displacement which moves the center of rotation away from the geometric center (Figure 2.1-2).[1] A displacement measurement performed on the rotor results in approximately the same waveform as the force, with a signal amplitude approximately proportional to the magnitude of the force. It is not exactly the same because the dynamics of the rotor affect the response. This is an important point in vibration analysis, and is discussed in more detail in the next section.

The velocity and acceleration parameters of the vibration are offset in phase relative to displacement — an important consideration when using phase for analysis. Phase relationships are shown in Figure 2.1-3. Velocity, for example, is offset from displacement by 90°. At point B, when the displacement is maximum, the velocity is zero. At point C, when displacement is zero, velocity is maximum. Following the same reasoning, acceleration can be shown to be offset 90° from velocity, and thus 180° from displacement.

The amplitude of the vibration parameters also vary with rotation speed (rpm) — an important consideration in transducer selection. Velocity increases in direct proportion to frequency (f), while acceleration increases with the square of frequency. This variation with frequency, and the phase relationships shown in Figure 2.1-3 , are illustrated in the equations below. In these equations, which apply only to sinusoidal vibration, A is the vibration displacement amplitude and f is the rotor frequency of rotation (cps or Hz).

$$\text{Displacement} = A \sin (2p\, f\, t)$$
$$\text{Velocity} = 2p\, f\, A \,\text{COS}\, (2p\, f\, t)$$
$$\text{Acceleration} = - (2p\, f)^2 A \sin (2p\, f\, t)$$

The three vibration parameters are thus closely related and, in fact, can be derived from each other by a Dynamic Signal Analyzer (see Section 6.6). However, the variation in vibration amplitude with machine speed, and transducer limitations, often mean that only one of the parameters will supply the information necessary for analysis.

The impact of variations in amplitude with rotation speed is illustrated in Figure 2.1-4. In this example, potentially dangerous vibration levels are present in a low-speed fan and a high-speed gearbox. The two items to note are: (1) displacement and acceleration levels differ widely, and (2) velocity is relatively constant.

From the first, we can conclude that frequency considerations are important in selecting a vibration parameter. Acceleration is not as good a choice for very low frequency analysis, while displacement does not work well for high frequencies. Note that these are limitations of the vibration parameter, not the transducer.

[1] Note: This applies to shafts that do not bend in operation (i.e. rigid shafts). Flexible shafts respond somewhat differently to imbalance forces.

Frequency range limitations of transducers are also an important consideration in parameter selection, and are discussed in Section 2.2.

The fact that velocity is a good indicator of damage, independent of machine speed, implies that it is a good parameter for general machine monitoring. That is, a vibration limit can be set independent of frequency. (Velocity remains constant with damage level because it is proportional to the energy content of vibration.) Velocity is also a good parameter for analysis, but the upper frequency limitation of velocity transducers can be a problem for gear and high-speed bladeanalysis.

Mechanical Impedance
A key point illustrated by Figure 2.1-5 is that we are measuring the response of the machine to vibration forces, not the forces themselves. Thus the response characteristics of the machine — its mechanical impedance — have a direct impact on the measured vibration. The two key results of this are: (1) if the response is small, the vibration will be difficult to analyze, and (2) if response changes drastically with frequency, changes in running speed can produce misleading changes in measured vibration level. These are important considerations in selecting and installing transducers.

Figure 2.1-2 The imbalance force produces a vibration whose displacement has approximately the same waveform as the force itself.

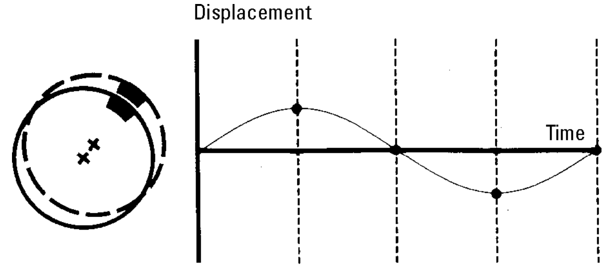


Figure 2.1-3 Velocity and acceleration of the vibration are offset 90° and 180° in phase from displacement.

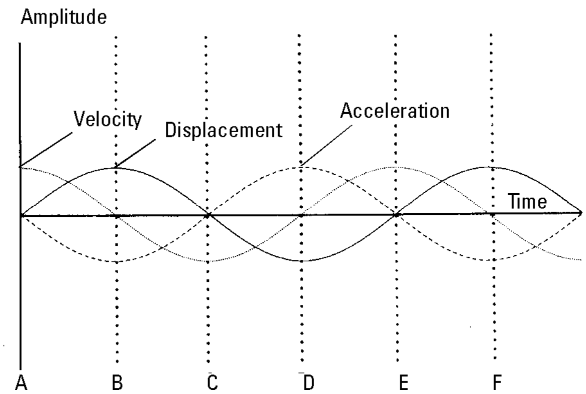
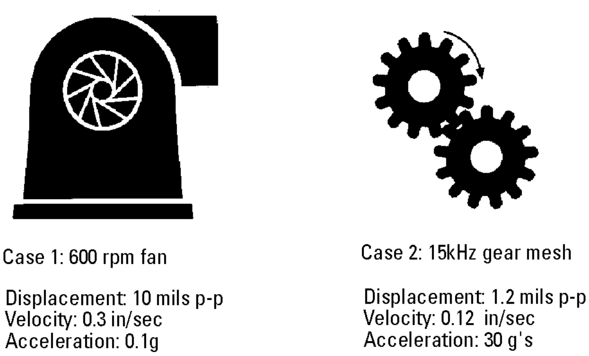


Figure 2.1-4 Two cases which illustrate the variation of vibration parameters with machine speed.



The most common example of low-level response involves machines with relatively light rotors and fluid-film bearings, mounted in heavy casings. Very little shaft vibration is transmitted to the casing, and shaft vibration must be measured directly (see Figure 2.1-6). Rolling element bearings are much stiffer than most fluid-film bearings, and transmit shaft (and their own) vibration to the machine case well.

An example of mechanical impedance that changes noticeably with speed is shown in Figure 2.1-7. This measurement shows how the ratio of acceleration response to input force might vary with frequency on a machine. Note that measurements made at speeds A and B would differ markedly in amplitude, even if the source of vibration remained the same. This illustrates why simple level measurements made on a machine whose speed varies can be misleading.

Natural Frequencies

In the plot of Figure 2.1-7, the response peaks occur at natural frequencies. These are the frequencies at which a structure will vibrate "naturally" when hit with an impact. A good illustration of natural frequency vibration is a tuning fork, which is designed to vibrate at a specific frequency when impacted (see Figure 2.1-8). When a vibration force occurs at a natural frequency, the structure will resonate (i.e. respond with a large amplitude vibration).[1]

Figure 2.1-5
Vibration measured on a machine is the response to defect force, not the force itself.

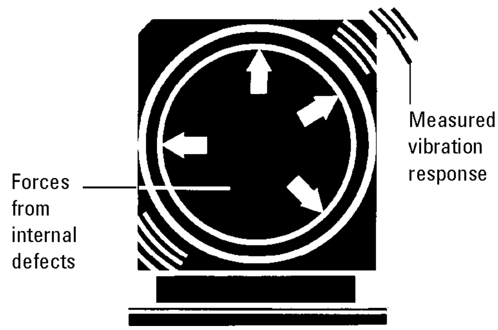


Figure 2.1-6
A relatively light shaft turning in fluid-film bearings transmits little vibration to the machine housing. Its vibration must be measured directly with a displacement transducer.

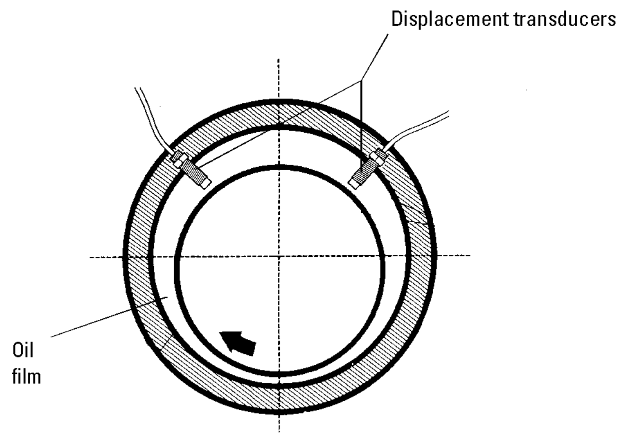
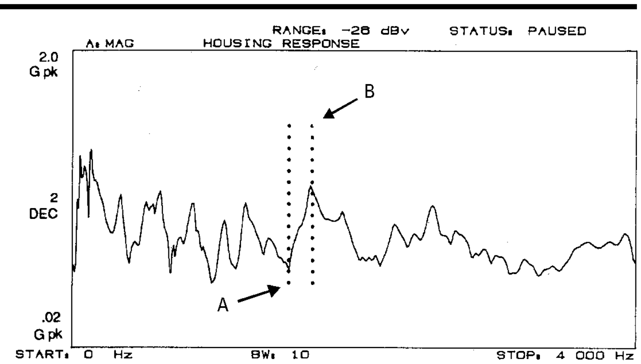


Figure 2.1-7
A plot of vibration response versus frequency for a machine housing shows how measured vibration level can change with rpm. A defect force at frequency B produces a much larger vibration response than the same force level at frequency A.



Natural frequencies relate to machinery vibration analysis in three important areas: (1) resonances of the structure can cause changes in vibration level with rpm, (2) the dynamics of rotating shafts change significantly near natural frequencies (or critical speeds), and (3) resonances of transducers limit the operating frequency range of velocity transducers and accelerometers. Changes in vibration response with frequency are shown in Figure 2.1-7. Shafts which operate above or near a natural frequency of the shaft are classified as flexible, and are discussed briefly in Section 3.4. Natural frequency limits on the useful frequency range of transducers are described in the next section (2.2).

A relationship worth noting at this point is the variation in natural frequency with mass and stiffness. The equation for the natural frequency of the simple mechanical system in Figure 2.1-9 is given below, where k is stiffness and m is mass. Note that natural frequency goes up with increasing stiffness and decreasing mass.

Natural frequency $(\omega_n) = (k/m)^{1/2}$

If you think of piano wires or guitar strings, the tight, lightweight ones are higher in frequency than the loose, heavy ones. This relationship is important when determining a solution to resonance problems.

Figure 2.1-8
When excited by an impact, a tuning fork vibrates at its natural frequency.

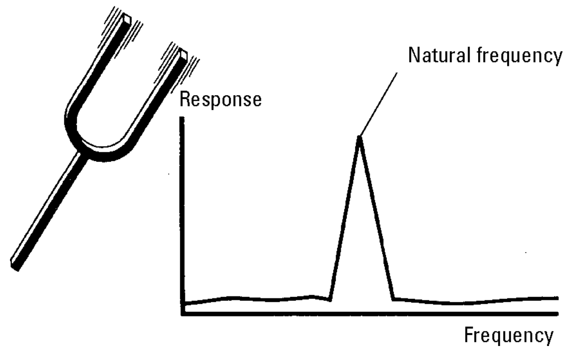
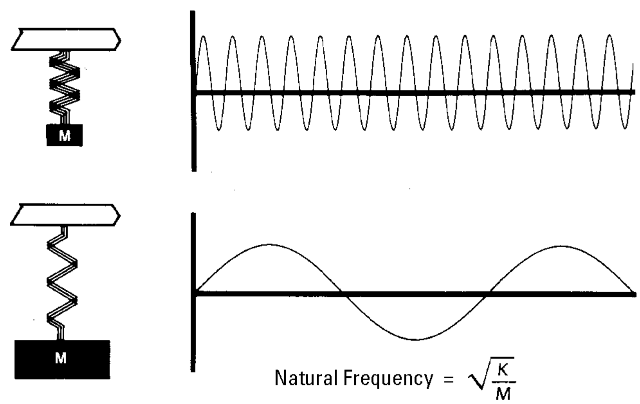


Figure 2.1-9
The natural frequency of a simple mechanical system varies with mass and stiffness.



[1] Note: The subject of resonances and structural vibration is dealt with in more detail in Hewlett-Packard Application Note AN243-3. Though extremely important to analyzing and understanding machine vibration; the focus of this note is more on analyzing operating machines than on structural analysis.

2.2 Transducers

In this section, each of the transducers shown in Figure 2.1 will be described. We will discuss how each one works, its important characteristics, and the most common applications. We will also discuss some common tachometer type transducers used to obtain rotation speed information on the machine.

Displacement Transducers
Noncontacting displacement transducers (also known as proximity probes[1]), like the one in Figure 2.2-1, are used to measure relative shaft motion directly. A high frequency oscillation is used to set up eddy currents in the shaft without actually touching it. As the shaft moves relative to the sensor, the eddy current energy changes, modulating the oscillator voltage. This signal is demodulated, providing an output voltage proportional to displacement. This is illustrated in Figure 2.2-2.

In practice, noncontacting displacement probes are used on virtually all turbomachinery because their flexible bearings (fluid film) and heavy housings result in small external responses. Some gas turbines, especially those used on aircraft, use relatively stiff rolling-element bearings, and can thus use housing-mounted transducers (velocity and acceleration) effectively.

[1] Note: We will limit our discussion to eddy current probes as they are by far the most commonly used type.

Key characteristics of displacement transducers

(a) Displacement transducers measure relative motion between the shaft and the mount, which is usually the machine housing. Thus, vibration of a stiff shaft/bearing combination that moves the entire machine is difficult to measure with displacement transducers alone.

(b) Signal conditioning is included in the electronics. Typical outputs are 200 mV/mil or 8mV/micron (1 mil is 0.001 inches; 1 micron is 0.001 millimeters). Technically, the frequency response of displacement probes is up to 10,000 Hz (or 600,000 rpm), but as a practical matter the displacement levels at these frequencies is so low that the actual useful frequency range of proximity probes is about 500 Hz (30,000 rpm).

(c) Shaft surface scratches, out-of-roundness, and variation in electrical properties due to hardness variations, all produce a signal error. Surface treatment and run-out subtraction can be used to solve these problems [11,12].

(d) Installation is sometimes difficult, often requiring that a hole be drilled in the machine housing.

(e) The output voltage contains a dc offset of 6 – 12 volts, requiring the use of ac coupling for sensitive measurements. AC coupling is a feature of all DSAs, and simply means that an input capacitor is used to block the dc. The practical disadvantage of ac coupling is reduced instrument response below 1Hz (60 rpm).

Figure 2.2-1 Noncontacting displacement transducers include a probe and an oscillator module.

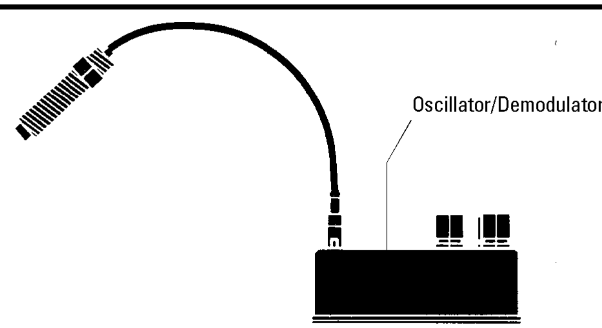
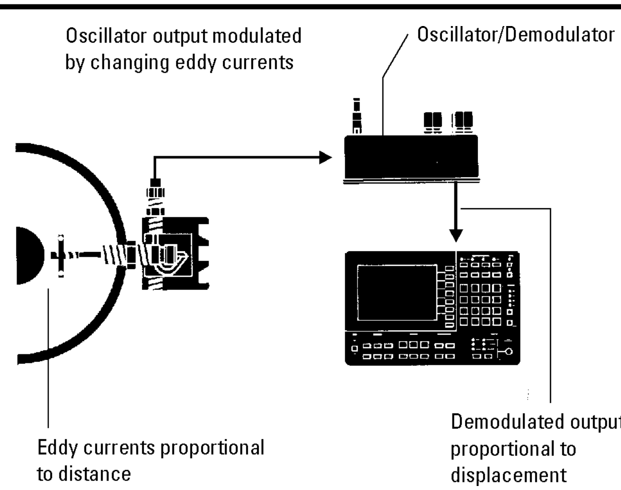


Figure2.2-2 Schematic diagram of a typical noncontacting displacment transducer installation.



Velocity Transducers

Velocity transducers were the first vibration transducer, and virtually all early work in vibration severity was done using velocity criteria. Velocity transducer construction is shown in Figure 2.2-4. The vibrating coil moving through the field of the magnet produces a relatively large output voltage that does not require signal conditioning. The amplitude of the voltage is directly proportional to the velocity of the vibration. As shown in Figure 2.2-5, the spring-mass-damper system is designed for a natural frequency of 8 to 10 Hz, which allows the magnet to stay essentially fixed in space. This establishes a lower frequency limit of approximately 10 Hz (600 rpm). The upper frequency limit of 1000 to 2000 Hz is determined by the inertia of the spring-mass-damper system.

Historically, the velocity transducer was widely used in machinery vibration measurements; but in recent years most transducer manufacturers have replaced this technology with accelerometers that have electrically integrated outputs which provide the same functionality as velocity probes but with wider frequency range and better stability. DSAs also provide for internal integration of acceleration signals; making accelerometers the transducer of choice — due to its wider frequency response, greater accuracy and more rugged construction.

Figure 2.2-3
A typical velocity transducer with extension probe installed.

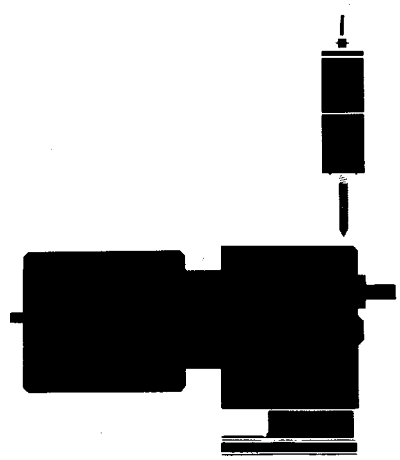


Figure 2.2-4
Velocity transducer output is a current generated in the coil as it moves through the field of the stationary magnet.

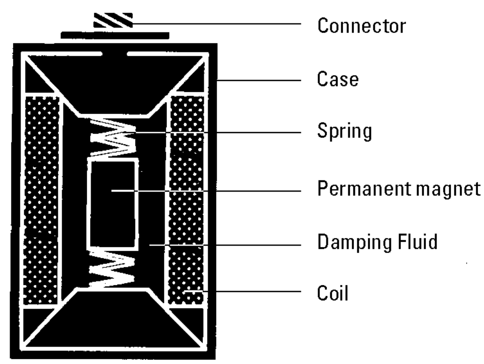
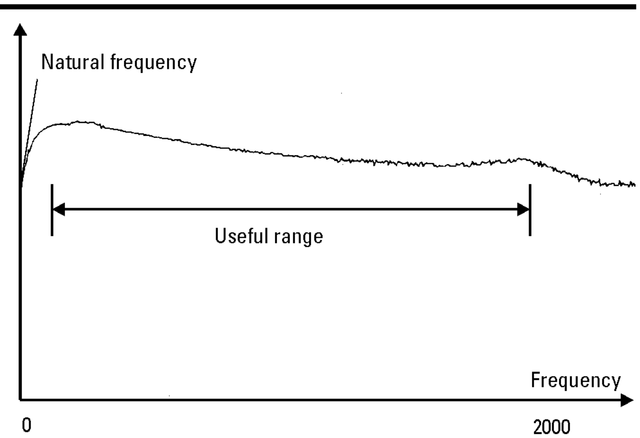


Figure 2.2-5
Frequency response of a typical velocity transducer. Note that the natural frenquency of the magnet-spring-damper system is below the operating range.



Accelerometers

Accelerometers are the most popular general purpose vibration transducer. They are constructed using a number of different technologies, but for general purpose measurements and machinery vibration, the most common design is the piezoelectric quartz accelerometer. Our discussion will be limited to this type and its derivatives. Construction of a simple accelerometer is shown in Figure 2.2-7. The vibrating mass applies a force on the piezoelectric crystal that produces a charge proportional to the force (and thus to acceleration).

The frequency response of a typical accelerometer is shown in Figure 2.2-8. Note that the natural frequency is above the operating range of the transducer (unlike the velocity transducer). Operation should be limited to about 20% of the natural frequency.

Accelerometer sensitivity is largely dependent on the size of the mass, with a larger mass producing more output. High output is especially important for increasing the usability of accelerometers at low frequencies. However, in our previous discussion of natural frequency, we noted that natural frequency decreases as mass increases. Thus increased sensitivity tends to lead to lower operating frequency range and larger physical size.

Key characteristics of accelerometers

(a) Accelerometers offer the broadest frequency coverage of the three transducer types. Their weakness is at low frequency, where low levels of acceleration result in small output voltages. Their large output at high frequencies also tends to obscure lower frequency content when the transducer is used for measuring overall level. This can be overcome by models with built-in integrators giving velocity output, or by added signal processing.

(b) The low frequency response of piezoelectric accelerometers is limited to approximately 5 Hz. This can be improved with special low frequency versions of the accelerometer. An inherent problem still exists in measuring acceleration at low frequency since its level tends to decrease dramatically at low frequencies.

(c) Accelerometers are very sensitive to mounting. Handheld models are available but repeatability is very dependent upon the individual. This is increasingly true for high frequencies. When possible, accelerometers should be securely mounted using a threaded stud, high strength magnet, or industrial adhesive. The mounting surface should be flat and smooth — preferably — machined. Frequently, special mounting studs are bonded or welded in place where repeated measurements are to be made.

Figure 2.2-6 Accelerometers feature wide frequency range and ruggedness. They should be securely mounted on a flat surface for best results.

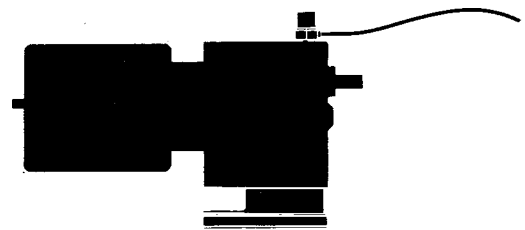
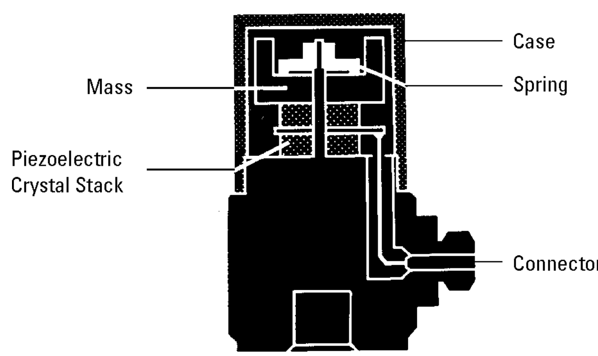


Figure 2.2-7 The output voltage of an accelerometer is produced by the accelerating mass squeezing the piezo-electric crystal stack. The force — and thusthe output voltage —is proportional to acceleration.



Accelerometer output is a low-level, high-impedance signal that requires special signal conditioning. The traditional method is to use a separate charge amplifier, as shown in Figure 2.2-9(a). How-ever, accelerometers are available with built-in signal conditioning electronics that require only a simple current-source supply. The accelerometer can be directly connected to most DSAs (Figure 2.2-9(b)). Another advantage of this type of accelerometer is that expensive low-noise cable required of normal piezoelectric accelerometers is not required. This can be especially important when long or multiple cables are required.

Tachometers

Tachometers are devices used to measure the rotation speed of a machine shaft. They are useful in determining accurate operating speed and identifying speed related components of the velocity. The transducer itself normally provides a pulse of some fixed amplitude at a rate related to rotation speed (typically, once per revolution). We will discuss two common types, the proximity probe and the optical tachometer.

Figure 2.2-8 Accelerometer high frequency response is limited by the natural frequency of the spring-mass system.

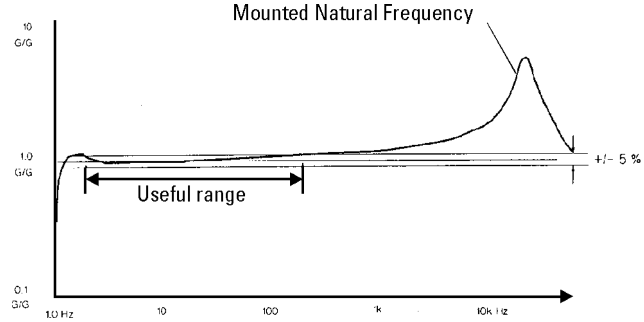


Figure 2.2-9a Traditional accelerometers require an external charge amplifier for signal conditioning.

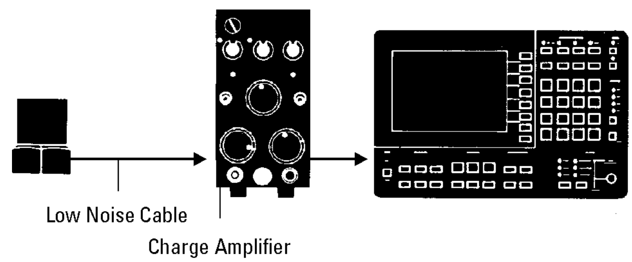
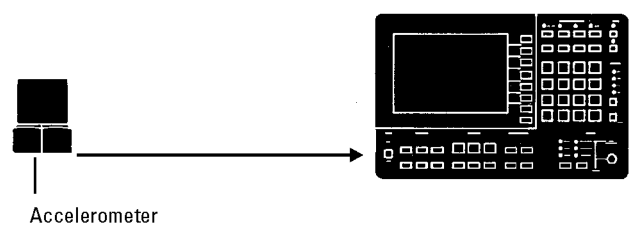
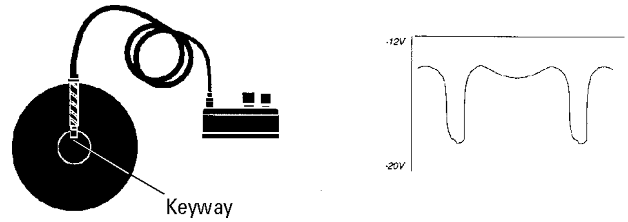


Figure 2.2-9b Integrated Circuit Piezoelectric accelerometers, with built-in signal conditioning, can be connected directly to a compatible DSA.



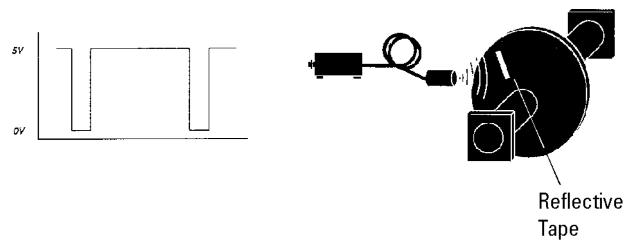
The proximity probe is the same as previously discussed, however, it is not used to get accurate displacement information in this mode. It is commonly used to detect the presence of something such as a keyway slot (often referred to as a keyphaser) or gear tooth. Figure 2.2-10(a) illustrates a proximity probe detecting a keyway to provide a once per revolution signal. This transducer has many of the limitations previously described.

Figure 2.2-10a Proximity probe used as tachometer to provide signal with repetion rate proportional to shaft velocity (rpm).



The other common tachometer transducer is the optical tachometer. It generally consists of either an optical or infrared light source and a detector (Figure 2.2-10(b)). Optionally, a lens for focusing the beam can be provided. The beam is trained on the rotating shaft and detects the presence of a reflective indicator (usually, a piece of tape or reflective paint).

Figure 2.2-10b Optical tachometer which measures reflection of light from a rotating object to provide signal proportional to rotation speed.



The output of the tachometer is handled in one of two ways. On multi-channel DSAs the tachometer is fed into a channel of the DSA where the once-per-rev pulse train will produce a large frequency component at the rotation speed of the machine. This is useful in obtaining valuable phase information about the response channels. An alternative is to measure the rotation speed directly with specialized hardware interfaced directly to the DSA's external sample control. It is also common to connect the tachometer signal directly to the trigger input of the DSA to obtain an accurate phase reference.

Tachometers differ from motion transducers in the fundamental variable measured. They measure the timing of an event, i.e. like the passing of a reference, such as a keyphasor.

2.3 Selecting the Right Transducer

Selecting the right transducer for an application is a straight forward process that is described below. Table 2.4 in the next section is a guide for the application of transducers to several general types of machinery.

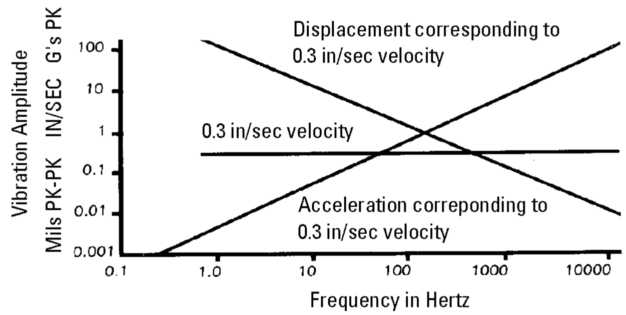


Figure 2.3
A vibration nomograph shows how the levels of displacement and acceleration change with frequency, relative to the level of velocity. Note that the acceleration response is very low at 1 Hz (less than 100 µV with a 10 mV/g accelerometer).

Step 1: Determine the Parameter of Interest.
If you are interested in monitoring a critical clearance or relative displacement, the only choice is a displacement transducer. Although acceleration and velocity can be converted to displacement, it will be an absolute measurement, rather than the relative measurement given by a displacement probe. If the parameter is a quantity other than a clearance or relative displacement, go on to next step.

Step 2: Mechanical Impedance Considerations.
If the vibration is not well transmitted to the machine case, you must use a displacement transducer to measure the shaft runout directly. This will be the case with a flexible rotor-bearing system working in a heavy casing. If the shaft is not accessible (as an internal shaft in a gearbox), or if the rotor-bearing system is stiff, you should use a casing mounted velocity or acceleration transducer. In borderline cases, it may be appropriate to use both absolute and relative motion transducers.

If Steps 1 and 2 indicate a displacement transducer, it is the one that will provide the best results. If a housing-mounted acceleration or velocity transducer is indicated, go on to Step 3.

Step 3: Frequency Considerations.
If the frequency of the expected vibration is greater than 1000 Hz, you must use an accelerometer. (You will have a much better idea of frequencies to expect after reading Chapter 4). If the vibration will be in the 10 to 1000 Hz range, either velocity or acceleration transducers can be used. Generally, an accelerometer will be the choice in these cases. The important thing to consider is the individual specifications of the accelerometer. Choose one designed for the frequency range and vibration level anticipated. The vibration nomograph of Figure 2.3 can be used to help determine the required performance. In many cases for low frequency (<20 Hz) applications or applications where the overall level is important for accessing machinery health a velocity output is required. This will dictate using either a velocity transducer or more commonly an accelerometer with integrated output proportional to velocity.

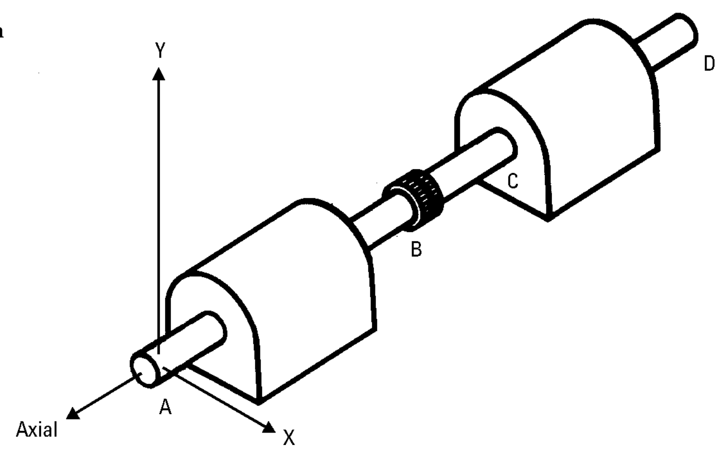
Table 2.4 Transducer Application Summary.

Machine Description	Transducer Variable	Location
Steam turbine/large pump or compressor A,B,C,D. with fluid-film bearings.	Displacement	Radial horizontal at Redundant axial at A and D.
Gas turbine or medium size pump	Displacement	Radial horizontal and vertical at A and B.
	Velocity	Radial horizontal or vertical at A and B.
Motor/fan both with fluid-film bearings	Displacement or Velocity	One radial at each bearing. One axial displacement to detect thrust wear.
Motor/pump or compressor with rolling element bearings	Velocity or Acceleration	One radial at each bearing. One axial, usually on motor, to detect thrust wear.
Gear box with rolling element bearings	Acceleration	Transducers mounted as close to each bearing as possible.
Gearbox shafts with fluid-film bearings	Displacement	Radial horizontal and vertical at each bearing. Axial to detect thrust wear.

2.4 Installation Guidelines

After the transducer has been selected, it must be properly installed for the best results. Figure 2.4 is an example of a machine combination that is used for the application summary in Table 2.4. The machine combination could be a small motor and pump, or a steam turbine and generator. In general, the number of transducers used on a machine combination is determined by the purpose of the measurement. Table 2.4 is intended to show typical applications and considerations that can be used as a guide in selecting measurement points and transducers.

Figure 2.4 Transducer locations referenced in Table 2.4



When troubleshooting a vibration problem it is critical to get information on vibration of key components in the principle directions. The inclusion of phase information is critical to diagnosing many machine dynamics problems. On the other hand, characterizing a non-critical machine for machinery health monitoring purpose; the goal is often to find a “representative” measurement which can characterize the general condition of the machine with the minimum number of measurements. When selecting measurement points and transducers the ultimate goal should be kept in mind. Careful transducer selection; bearing in mind manufacturers specification; proper mounting of the transducer can be critical. One particular caution: the transducer should never be mounted to a sheet metal cover, since resonances may easily be in the operating speed range and can easily mask the real objective of the measurement.

Chapter 3

Reducing Vibration to its Components: The Frequency Domain

The signal obtained from a machinery vibration transducer is a complex combination of responses to multiple internal and external forces. The key to effective analysis is to reduce this complex signal to individual components, each of which can then be correlated with its source. Techniques for reducing vibration to its components are the subject of this chapter, while the process of correlating these components with machinery vibration is discussed in Chapters 4 and 5.

Two analysis perspectives are available for determining the components of vibration: (1) the time domain view of vibration amplitude versus time and (2) the frequency domain view of vibration amplitude versus frequency. While the time domain provides insight into the physical nature of the vibration, we will see that the frequency domain is ideally suited to identifying its components. The advantage of Dynamic Signal Analyzers for machinery analysis is their ability to work in both domains.

This chapter begins with a discussion of the relationship between the time and frequency domains. Waterfall/spectral maps, which add the dimension of machine speed or time to the frequency domain, are presented next. The frequency phase spectrum, an important complement to the more familiar amplitude spectrum, is discussed in the following section. This chapter closes with a description of the type of instruments available for frequency domain analysis. Information on the time and frequency domains in this application note is focused on machinery vibration. For a more general discussion of the subject refer to Hewlett-Packard application note AN 243.

3.1 The Time Domain

One way to examine vibration more closely is to observe how its amplitude varies with time. The time domain display in Figure 3.1-1 clearly shows how vibration due to an imbalanced rotor varies with time (we are using a displacement transducer to simplify the phase relationship). The amplitude of the signal is proportional to the amount of imbalance, and the speed of rotation. This signal is easy to analyze because we are using an idealized example with a single source of vibration – real world vibration signals are much more complex.

When more than one vibration component is present, analysis in the time domain becomes more difficult. This situation is illustrated in Figure 3.1-2, where two sine wave frequencies are present. The result of this combination is a time domain display in which the individual components are difficult to derive. The time domain is a perspective that feels natural, and provides physical insight into the vibration. It is especially useful in analyzing impulsive signals from bearing and gear defects, and truncated signals from looseness. The time domain is also useful for analyzing vibration phase relationships. However, the individual components of complex signals are difficult to determine. A perceptive that is much better suited to analyzing these components is the frequency domain.

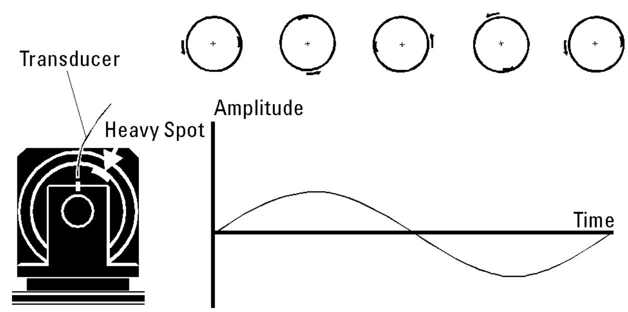


Figure 3.1-1 A time domain representation of vibration due to rotor imbalance.

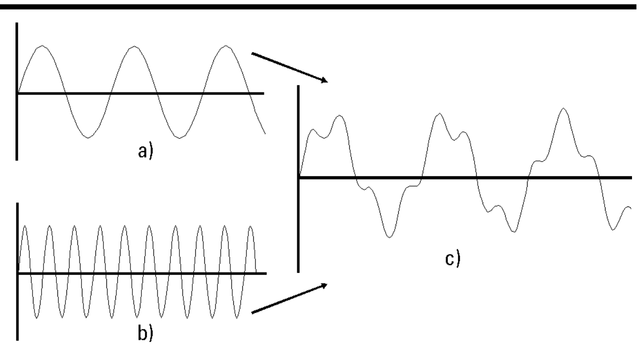


Figure 3.1-2 Waveform (c) is the combination of signals (a) and (b). The nature of these components is hidden in the time domain view of their sum.

3.2 The Frequency Domain

Figure 3.2-1(a) is a three-dimensional graph of the signal used in the last example. Two of the axes are time and amplitude that we saw in the time domain. The third axis is frequency, which allows us to visually separate the components of the waveform. When the graph is viewed along the frequency axis, we see the same time domain picture we saw in 3.1-2. It is the summation of the two sine waves which are no longer easily recognizable.

However, if we view the graph along the time axis as in Figure 3.2-1(c), the frequency components are readily apparent. In this view of amplitude versus frequency, each frequency component appears as a vertical line. Its height represents its amplitude and its position represents its frequency. This frequency domain representation of the signal is called the spectrum of the signal.

The power of the frequency domain lies in the fact that any real world signal can be generated by adding up sine waves. (This was shown by Fourier over one hundred years ago.) Thus, while the example we used to illustrate the frequency domain began as a summation of sine waves, we could perform a similar reduction to sine wave components for any machinery vibration signal. It is important to understand that the frequency spectrum of a vibration signal completely defines the vibration – no information is lost by converting to the frequency domain (provided phase information is included).

A Machinery Example

Figure 3.2-2 should give you better insight into frequency domain analysis applied to machinery. The internal sources of vibration in this example are rotor imbalance, a ball bearing defect, and reduction gear meshing. For purposes of illustration in this example, the sources of vibration and their resulting frequency components have been somewhat simplified. (Details of the frequency components that each of these defects produce are given in Chapter 4.)

Imbalance produces a sinusoidal vibration at a frequency of once per revolution. If we assume a single defect in the outer race of the ball bearing, it will produce an impulsive vibration each time a ball passes over the defect – usually around four times per revolution. To simplify the example, we will assume that this is a sine wave. The two smaller sine waves around this frequency are caused by interaction (modulation) of the

Figure 3.2-1 The relationship between the time and frequency domains.

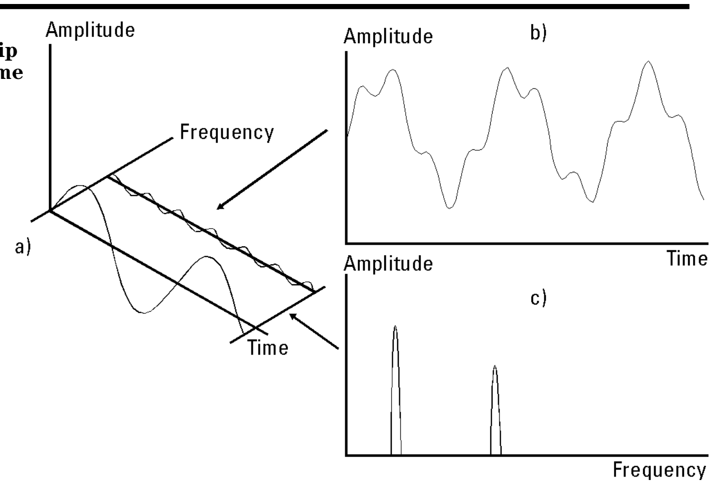
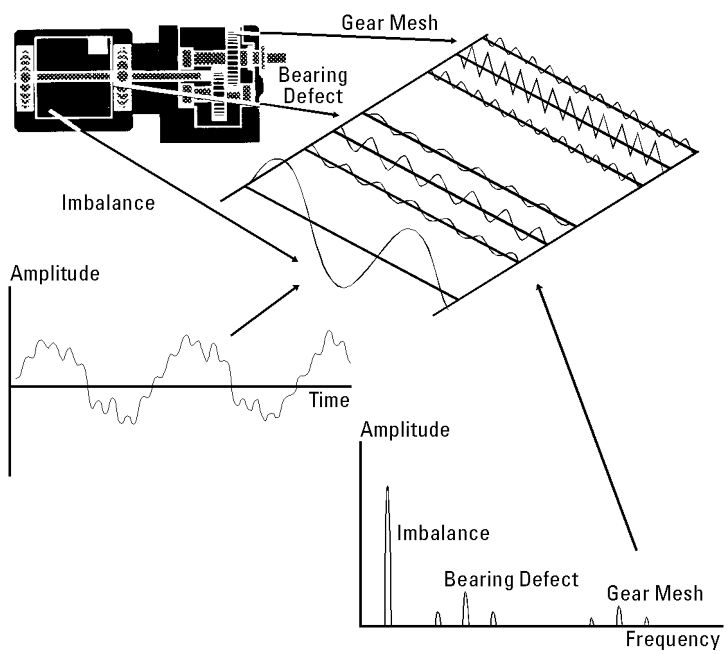


Figure 3.2-2 Machinery vibration viewed in the time and frequency domains.

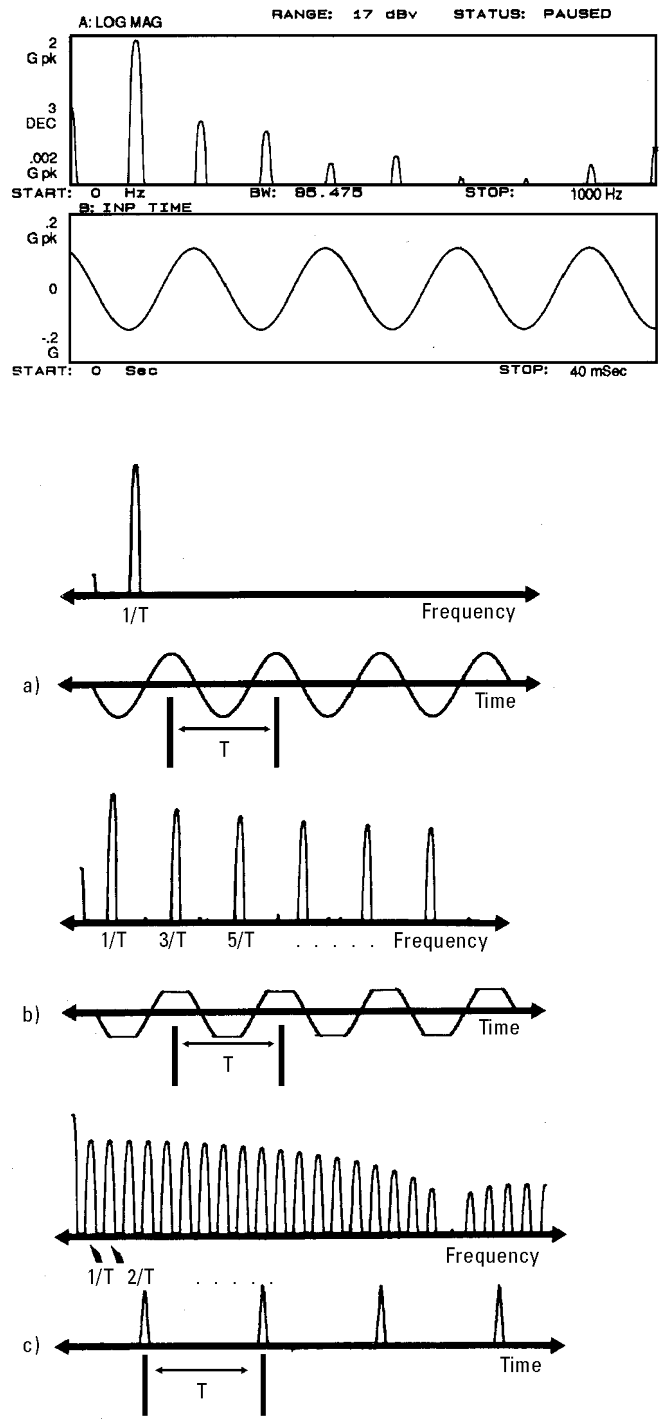


bearing defect force with the imbalance force. These signals are called sidebands, and occur often in machinery vibration. They are spaced at increments of plus and minus the running speed from the defect frequency. These components are often referred to as sum and difference frequencies, and are discussed in Section 5.3. The gear mesh frequency appears at running speed multiplied by the number of teeth on the main shaft gear, which here we assumed to be ten. The running speed sidebands around the gear meshing frequency usually indicate eccentricity in the gear. While this is a greatly simplified view of machinery vibration, it demonstrates the clarity with which vibration components can be seen in the frequency domain.

Early Warning of Defects
As we pointed out in the introduction, DSAs are used to make machinery vibration measurements in the frequency domain. This is because the low level vibration produced by early stages of some defects cannot be detected by an overall vibration meter. (In effect, it is "buried" by the relatively large residual imbalance component.) This is especially true of rolling element bearings, and is one of the reasons this particular problem is one of the most difficult to detect.

A major advantage of the frequency domain is that low level signals are easy to see – even in the presence of signals 1000 time larger. This is illustrated in the time and frequency domain displays of Figure 3.2-3, where the low-level signals that are readily apparent in the frequency domain cannot be seen in the time domain. A key to this capability is logarithmic display of amplitude.

Figure 3.2-3
Small signals that are hidden in the time domain are readily apparent in the frequency domain. By using a logarithmic amplitude scale, signals which vary in level by a factor of over 1000 can be displayed.



While most people prefer the more natural feel of a linear display, logarithmic displays are an aid to displaying the wide dynamic range of data present in a DSA. (Dynamic range is discussed in section 6.4). We will present examples using both linear and logrithmic scales in this application note.

Spectrum Examples
Figure 3.2-6 shows the time and frequency domain of four signals that are common in machinery vibration.

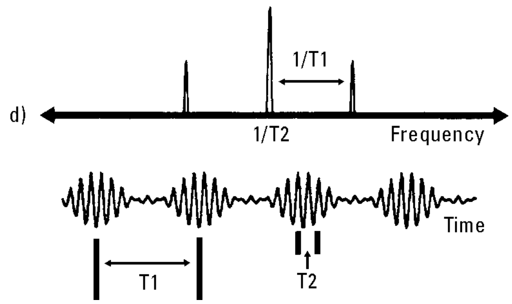
(a) The frequency spectrum of a pure sine wave is a single spectral line. For a sine wave of period T seconds, this line occurs at 1/THz.

(b) A distorted sine wave, produced by "clipping" the signal at some prescribed amount on both the positive and negative directions. This is much like the truncated signal produced by mounting or bearing cap looseness and is made up of a large number of odd harmonics. Harmonics are components which occur at frequency multiples of a fundamental frequency. In machinery analysis, we often refer to harmonics as "orders" of the fundamental running speed.

(c) Bearings and gears often produce impulsive signals that are typified by harmonics in the frequency domain. These harmonics are spaced at the repetition rate of the impulse.

(d) Modulation can result when some higher characteristic frequency interacts with a lower frequency, often the residual imbalance. The frequency spectrum of a modulated signal consists of the signal being modulated (the carrier), surrounded by sidebands spaced at the modulating frequency.

Figure 3.2-6 Examples of frequency spectra common in machinery vibration.



3.3 Spectral Maps/Waterfalls

The vibration characteristics of a machine depend on its dynamics and the nature of the forces acting upon it. The change of these characteristics with machine speed has two important implications for analysis: (1) the vibration resulting from a defect may not appear in all speed ranges, and (2) insight into the nature of the machine may be obtained from observing the change in vibration with speed. Spectral maps[1], such as the one in Figure 3.3-1 are three dimensional displays that effectively show variation in the vibration spectrum with time. These are also called cascade plots.

Rpm spectral maps usually consist of a series of vibration spectra measured at different speeds. A variety of other parameters, including time, load, and temperature are also used as the third dimension for maps and waterfalls. A common method for mapping the variations in the vibration with rpm is to measure successive spectra while the machine is coasting down or running up in speed. If the machine is instrumented with a tachometer, the speed can be monitored and used to trigger the measurement thus obtaining vibration spectra at uniformly spaced rpm. Figure 3.3-1 illustrates such a map and the setup used to produce it with a Hewlett-Packard DSA with this built-in capability (Figure 3.3-2).

In addition to showing how vibration changes with speed, spectral maps/waterfalls quickly indicate which components are related to rotational speed. The components will move across the map as the speed changes, while fixed frequency components move straight up the map. This feature is especially useful in recognizing machine resonances (critical speeds), which occur at fixed frequencies.

3.4 The Phase Spectrum

The complete frequency domain representation of a signal consists of an amplitude spectrum and a phase spectrum. While the amplitude spectrum indicates signal level as a function of frequency, the phase spectrum shows the phase relation between spectral components. In machinery vibration analysis, phase is required for most balancing techniques. It is also useful in differentiating between faults which produce

[1] Note: The distinction between maps and waterfalls is often disputed. For the purposes of this note a waterfall is a "live" continuously updating display that constantly updates itself with the latest spectrum while discarding the oldest. A map is a display of multiple spectrums taken at different times/conditions; often requiring complete regeneration to add additional spectra.

similar amplitude spectra. DSAs are unique among commonly used frequency domain analyzers in providing both amplitude and phase spectra.

The concept of phase relationships is most easily seen in the time domain. In Figure 3.4-1, phase notation has been added to the waveform we used in our first time-domain example. One 360° cycle of the rotor corresponds to one cycle of the vibration signal. This relationship holds regardless of where we start on the circle, but absolute phase numbers mean nothing without a reference. In Figure 3.4-1, we have defined the reference point as A. This means that in effect when the keyphasor passes point A the time of the first data point of the block is defined as t=0. The actual phase is also dependent upon the orientation (and type) of the transducer. By convention for a single-channel measurement, a cosine wave (i.e. positive maxima at t=0) is defined as the zero phase reference.

Just as absolute phase can be defined relative to a reference point, we can define the relative phase of two signals of the same frequency. The signals shown in Figure 3.4-2 are separated by 1 quarter of a cycle, or 90°. We say that the phase of the trace A leads that of trace B because its peak occurs first.

In the frequency domain, each amplitude component has a corresponding phase. Figure 3.4-3 is a DSA display of our imbalance example, indicating a 90° phase relationship between the frequency component and the trigger signal (amplitude is shown as a dashed line). The phase is -90° because the peak of the signal occurs after the trigger.

Figure 3.3-1
Spectral maps show variation in the vibration spectrum with time or rpm.

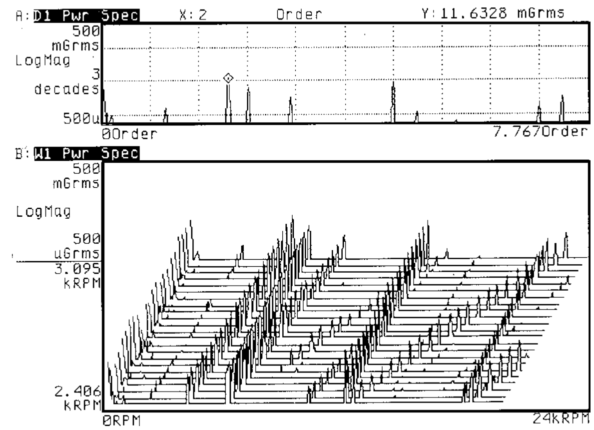


Figure 3.3-2
Set-up of DSA to produce spectral map.

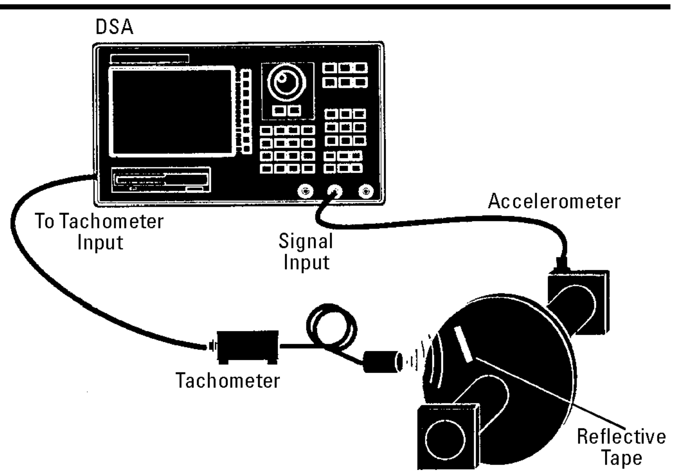


Figure 3.4-1
The phase of the imbalance signal corresponds to the direction of the displacement. One 360° rotation of the rotor corresponds to one 360° cycle of the signal.

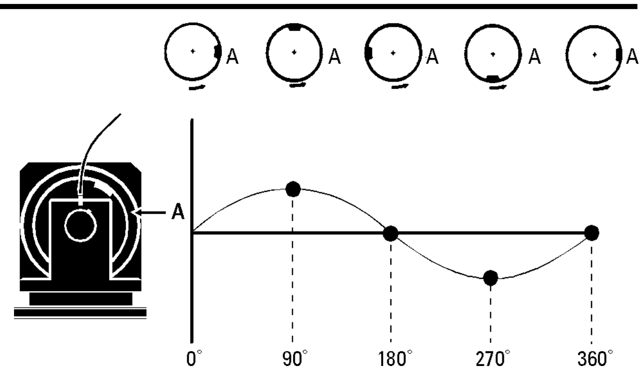
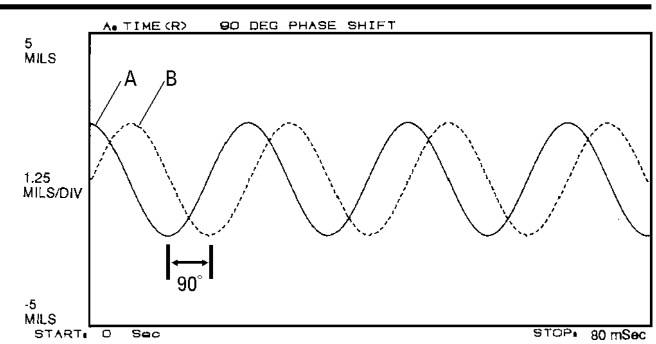


Figure 3.4-2
Two sine waves with a phase relationship of 90°.



Balancing
The most common application for phase spectrum is in trim balancing. Recall from Figure 3.4-1 that we need a reference for absolute phase to be meaningful. In machinery analysis, this reference is most often provided by a keyphasor – a displacement or optical transducer which detects the passage of a keyway, set screw, or reflecting surface. Figure 3.4-4 shows a keyphasor added to our example machine. With the transducer 90° behind the keyphasor (in the direction of rotation), and the keyphasor and heavy spot lined up, the resulting time domain waveforms are offset in phase by 90°. The corresponding phase spectrum of the vibration signal is as shown in Figure 3.4-3. In this case, the keyphasor is used to trigger the measurement.

Figures 3.4-3 and 3.4-4 indicate the location of the heavy spot relative to the keyway. This information can be used in balancing to locate a compensation weight opposite the heavy spot. This will readily give information about the location of the imbalance, but little information about the magnitude of the imbalance weight. Unless the system has been calibrated previously on the same or similar machines a two-measurement scheme which uses trial weights is required to get accurate data on the magnitude of the imbalance. For balancing, it's important to note, that the previous discussion assumed a displacement transducer; velocity transducers and accelerometers have additional 90° and 180° phase shifts that must be accounted for.

We have also assumed that the rotor is rigid. There are two areas of caution. First is that a magnetic phase detector (i.e. keyphasor) can cause phase shifting errors. This is due to the changing waveform shape with speed causing the trigger point to move. The other area is in balancing speed. It is advisable that balancing not be done close to resonance frequencies as the phase changes very rapidly with speed near resonances and this can lead to considerable measurement error.

Other Applications of Phase
The phase spectrum is also useful for differentiating between defects that produce similar amplitude spectra. In Section 4.4, we will describe how axial phase measurements can be used to differentiate between imbalance and misalignment. Section 5.2 explains how the relative stability of phase can be used to gain insight into the nature of defects.

Rigid and Flexible Rotors
We mentioned in the introduction that flexible rotors required an understanding of the shaft dynamics for complete analysis. As the name implies, a flexible rotor is one which bends during operation. This bending occurs at a natural frequency of the rotor, often referred to as a critical speed. A flexible rotor has several critical speeds, each with a specific bending shape (or direction). These shapes are called modes, and can be predicted through structural modeling and measured using orbit analysis. The distinction between rigid and flexible rotors is important because the dynamics of a rotor change significantly as it approaches and passes through a critical speed. The amplitude of the vibration response peaks, and the phase response shifts by 180°.

This phase shift is shown in the plot of Figure 3.4-5 (commonly referred to as a Bode plot). When phase is measured at a speed well above the critical, the high spot measured by the displacement transducer is at a point opposite the imbalance – a phase shift of 180°. When operating speed is near the critical speed, the phase response will be shifted between 0° and 180°, depending on the dynamics of the rotor.

Accurate interpretation of phase spectra measured on flexible rotors requires an understanding of rotor dynamics that is beyond the cope of this application note. Unless otherwise noted, all statements about the use of phase in analysis refer only to rigid rotors (those which operate well below the first critical speed).

3.5 Frequency Domain Analyzers

Instruments which display the frequency spectrum are generally referred to as spectrum analyzers, although DSAs are also commonly referred to as real-time or FFT analyzers. There are three basic types of spectrum analyzers: (1) parallel filter, (2) swept filter, and (3) DSA. This section will give a short description of each, along with advantages and disadvantages. For a more detailed discussion, refer to Hewlett-Packard application note AN 243.

A simple block diagram of a parallel-filter analyzer is shown in Figure 3.5-1. These analyzers have several built-in filters that are usually spaced at 1/3- or 1-octave intervals. This spacing results in resolution that is proportional to frequency. For a 1/3-octave analyzer, resolution varies from around 5 Hz at low frequencies to several thousand Hertz (kHz) at high frequency. A variation of the parallel-filter analyzer that is sometimes used in machinery work has a bank of filters that can be individually selected.

Parallel-filter analyzers offer a good compromise between resolution and frequency span when very large spans are required such as in acoustics. They tend to be expensive and do not have the resolution required for many machinery analysis applications.

Figure 3.4-3 DSA frequency domain display of a 90° phase relationship. Referring to Figure 3.4-2, this is the phase of trace B when trace A is used to trigger the measurement.

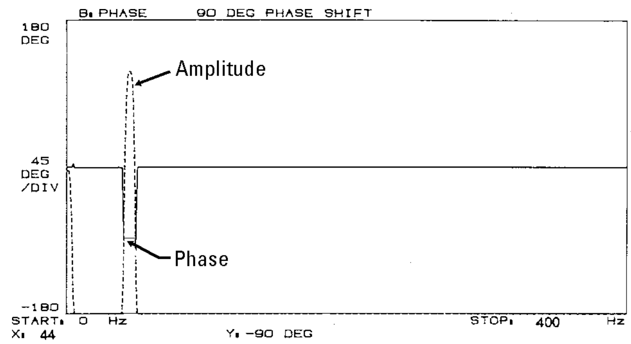


Figure 3.4-4 Since the heavy spot on the rotor passes the transducer 90° after the keyway passes the keyphasor, the imbalance signal lags the keyphasor pulse by 90°. The corresponding frequency domain phase spectrum is shown in Figure 3.4-3.

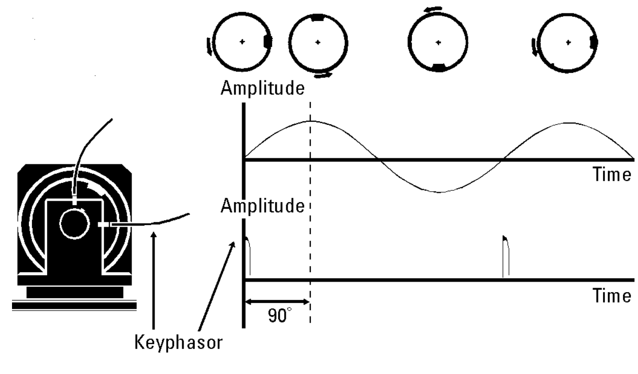


Figure 3.4-5 The vibration response of a flexible rotor shifts 180° in phase as rpm passes through a critical speed.

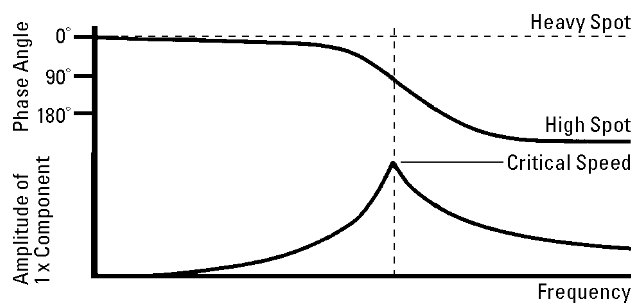
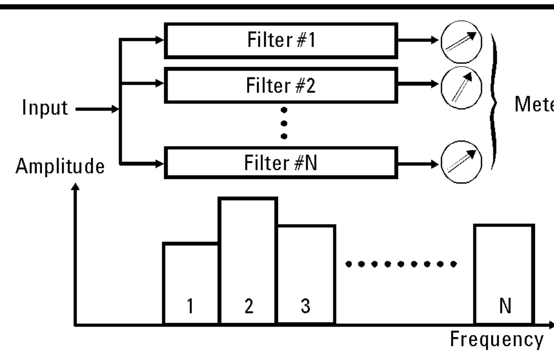


Figure 3.5-1 Parallel filter analyzers have insufficient frequency resolution for machinery analysis.



Swept-filter analyzers use a tuneable filter, much like a radio receiver. The block diagram for this type of analyzer is shown in Figure 3.5-2. The frequency resolution of these instruments is on the order of 1 to 5 Hz – better than parallel-filter analyzers but not good enough for many vibration analysis applications. They are much slower than the parallel-filter analyzers as they must analyze each individual frequency one at a time. The slowness of the operation not only increases the measurement time; it makes the technique unacceptable for situations where non-steady data is present.

DSAs use digital techniques to effectively synthesize a large number of parallel-filters. The large number of filters (typically 400 or more) provides excellent resolution, and the fact that they are parallel means that measurements can be made quickly. DSAs also provide time- and phase-spectrum displays, and can be connected directly to computers for automated measurement. The DSA essentially uses up FFT to create filters of constant-bandwidth resolution; unlike the parallel filters that tend to be proportional bandwidth. Being digital in implementation, some DSAs have the ability to analyze the data in much the same way as the parallel analyzers in addition to its normal FFT mode; thus allowing addition flexibility. This is referred to as digital real-time octave analysis.

Figure 3.5-2
Swept filter analyzers provide better frequency resolution than parallel filter analyzers, but are too slow for machinery analysis.

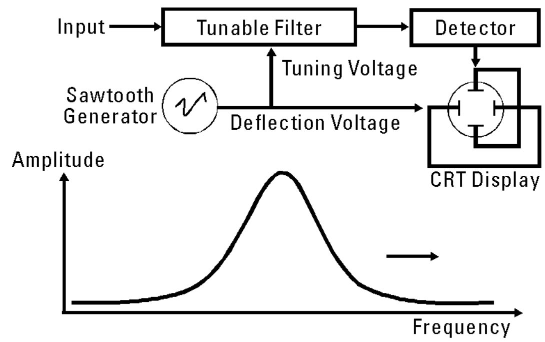
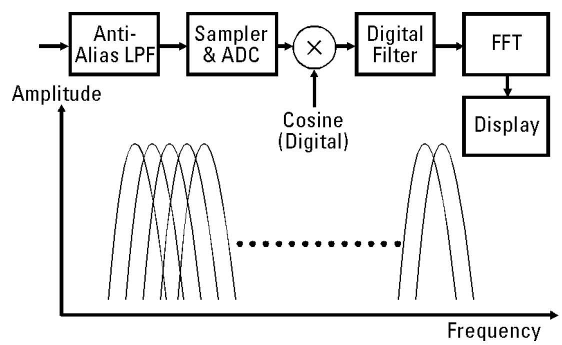


Figure 3.5-3
DSAs digitally simulate hundreds of parallel filters, providing both high speed and excellent frequency resolution. DSAs also provide time and phase displays not available on the other frequency domain analyzers.



Chapter 4

Vibration Characteristics of Common Machinery Faults

In the last chapter, we saw how a complicated time domain vibration signal can be reduced to simple spectral components using the frequency domain. In Chapters 4 and 5, we will take the next step in analysis – correlating these components with specific machine characteristics or faults. This chapter provides the basic theory, while Chapter 5 addresses some of the common analysis problems and techniques.

Each machine defect produces a unique set of vibration components that can be used for identification. This chapter describes these vibration patterns or “signatures” for the most common machinery defects. Where appropriate, frequency calculation formulas and details of spectrum generation are also included. The descriptions will give you the basic information needed to correlate vibration components with defects; the details provide insights that will improve your ability to analyze unusual situations.

The table in Section 4.10 summarizes the vibration pattern descriptions of Chapter 4. It is important to understand, however, that correlation is rarely as easy as matching vibration components on a DSA display with those in a table. Machinery dynamics, operating conditions (e.g. load and temperature), multiple faults, and speed variation all affect vibration, complicating the correlation process. Methods of dealing with these problems are the subject of Chapter 5.

Converting a vibration spectrum to a detailed report on machine vibration is another challenging aspect of vibration analysis. Chapter 4 and 5 are a starting point, providing a basis for building your skills through experience.

4.1 Imbalance

Rotor imbalance exists to some degree in all machines, and is characterized by sinusoidal vibration at a frequency of once per revolution. In the absence of high resolution analysis equipment, imbalance is usually first to get the blame for excessive once per revolution vibration – vibration that can be caused by several different faults. In this section, we will discuss spectral characteristics that can be used to differentiate these faults from imbalance, eliminating unnecessary balancing jobs.

Phase plays a key role in detecting and analyzing imbalance, and it is important to remember the phase shifts associated with flexible rotors (see Figure 3.4-5). A state of imbalance occurs when the center of mass of a rotating system does not coincide with the center of rotation. It can be caused by a number of things, including incorrect assembly, material build-up/loss, and rotor sag. As shown in Figure 4.1-1, the imbalance can be in a single plane (static imbalance) or multiple planes (coupled imbalance). The combination is referred to as dynamic imbalance. In either case, the result is a vector that rotates with the shaft, producing the classic once per revolution vibration characteristic.

Distinguishing Characteristics of Imbalance

The key characteristics of vibration caused by imbalance are:

(1) it is sinusoidal at a frequency of once per revolution (1x)

(2) it is a rotating vector, and

(3) amplitude increases with speed (i.e. $F=mw^2$). These characteristics are very useful in differentiating imbalance from faults that produce similar vibration.

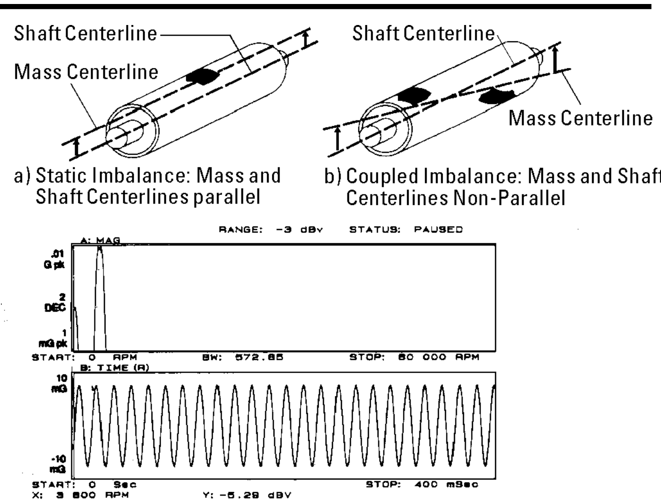


Figure 4.1-1 Imbalance, whether static or coupled, results in a spectral peak at a frequency of once per revolution (1 x).

The driving force in imbalance is the centrifugal forces caused by a mass rotating about a center point; as such, the vibration caused by pure imbalance is a once-per-revolution sine wave, sometimes accompanied by low-level harmonics. The faults commonly mistaken for imbalance usually produce high-level harmonics, or occur at a higher frequency. In general, if the signal has high harmonics above once per revolution, the fault is not a simple imbalance. However, high-level harmonics can occur with large imbalance forces, or when horizontal and vertical support stiffnesses differ by a large amount (see Section 4.4).

Because the imbalance force is a rotating vector, the phase of vibration relative to a keyphasor follows transducer location, while the amplitude changes are generally small. As shown in Figure 4.1-2, moving the transducer 90° results in a 90° change in phase reading with approximately the same amplitude[1]. It is also common for the stiffness to vary to some extent from vertical to horizontal; this can under some circumstances cause wide variations in phase readings with flexible rotors.

4.2 Rolling-Element Bearings

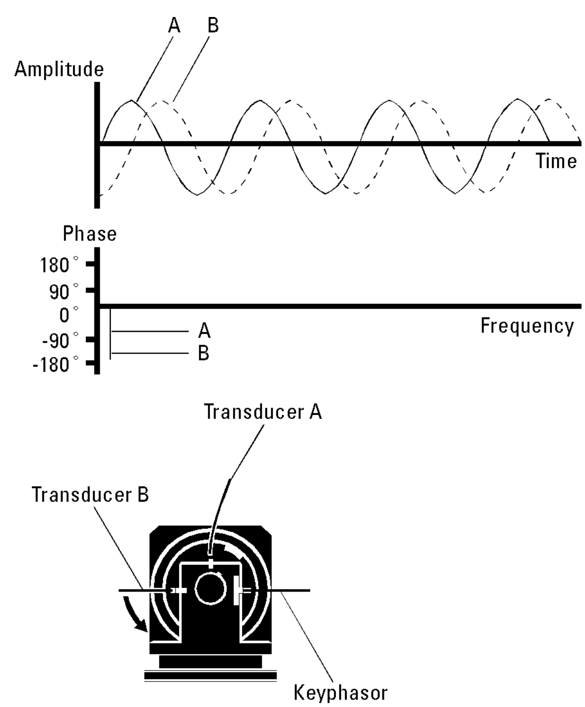
Rolling-element (anti-friction) bearings are the most common cause of small machinery failure, and overall vibration level changes are virtually undetectable in the early stages of deterioration. However, the unique vibration characteristics of rolling element bearing defects make vibration analysis an effective tool for both early detection and analysis of faults.

The specific frequencies that result from bearing defects depend on the defect, the bearing geometry, and the speed of rotation. The required bearing dimensions are shown in Figure 4.2-1, and are usually available from the bearing manufacturer. Included in this section is an HP Instrument Basic[2] program that computes the expected frequencies given bearing parameters and rotation speed. One caution: parameters of the same model-number bearing can change with manufacturer.

The major problem in detecting the early stages of failure in rolling-element bearings is that the resulting vibration is very low in level and often masked by higher level vibration. If monitoring is performed with a simple vibration meter (or in the time domain), these low levels will not be detected and unpredicted failures are inevitable (see Figure 3.2-3). The advantage of a DSA is that with the high resolution and dynamic range available, vibration components as small as 1/1000th the amplitude of higher level vibrations can be measured and detected.

Interestingly, some early indications of a bearing failure can later be obliterated in the later stages as the failure develops. For example, often in the early stages of failure a very succinct vibration

Figure 4.1-2
The rotating nature of the imbalance force results in a phase reading (relative to a key phasor) that follows transducer location. This is useful in differentiating imbalance from faults which produce directional vibration.



[1] The amplitudes obtained from these two readings can vary with support stiffness and running speed. With flexible rotors, small speed variations between the two measurements will result in a phase relation very different from the one pictured.

[2] Instrument Basic is an HP implementation of the Basic programming language that runs resident in many DSA analyzers.

component will be present. As the failure develops, the overall energy of the fault will increase, but often become more broad band in nature and difficult to detect in the presence of the other vibration components of the machine (Fig 4.2-3). This appearance of "healing" can be misleading. The example also illustrates a characteristic of frequency-spectrum analysis: it's usually easier to detect a distinct low-level narrow-band tone than a wide-band signal of high levels in the presence of other signals or noise.

Frequencies Generated by Rolling-Element Bearing Defects

Formulas for calculating the frequencies resulting from bearing defects are given in Table 4.2. The formulas assume a single defect, rolling contact, and a rotating inner race with fixed outer race. The results can be expressed in orders of rotation by leaving out the (RPM/60) term. The I-Basic program listing in Figure 4.2-2 will compute the bearing frequencies automatically.

If bearing dimensions are not available, inner- and outer-race defect frequencies can be approximated as 60% and 40% of the number of balls multiplied by the running speed, respectively. This approximation is possible because the ratio of ball diameter to pitch diameter is relatively constant for rolling-element bearings.

While it isn't necessary to understand the derivation of these formulas, two points of explanation may give you a better feel for them. (1) Since the balls contact both the shaft-speed inner race and the fixed speed outer race, the rate of rotation relative to the shaft center is the average, or 1/2

Figure 4.2-1
Using the parameters shown, the basic frequencies resulting from rolling element bearing defects can be completed.

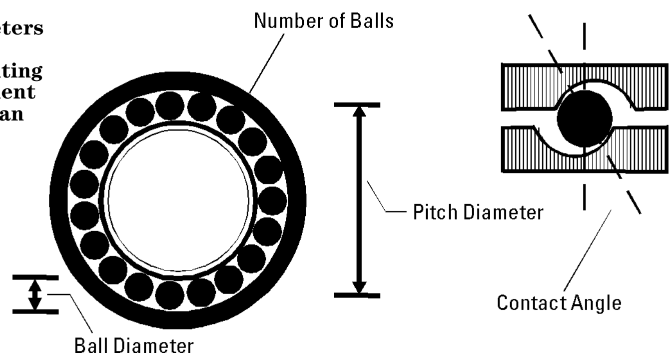


Table 4.2
Bearing Characteristic Frequencies.

Defect on outer race (Ball pass frequency outer) $= \frac{(n)}{2}\frac{(RPM)}{60}\left(1 - \frac{Bd}{Pd}\cos\phi\right)$ (1)

Defect on inner race (Ball pass frequency inner) $= \frac{(n)}{2}\frac{(RPM)}{60}\left(1 + \frac{Bd}{Pd}\cos\phi\right)$ (2)

Ball defect (ball spin frequency) $= \frac{(Pd)}{2Bd}\frac{(RPM)}{60}\left[1 - \left(\frac{Bd}{Pd}\right)^2\cos^2\phi\right]$ (3)

Fundamental train frequency $= \frac{1}{2}\frac{RPM}{60}\left(1 - \frac{Bd}{Pd}\cos\phi\right)$ (4)

Pd = Pitch diameter
Bd = Ball diameter
n = Number of balls
ϕ = Contact angle

Figure 4.2-2
The I-BASIC program to compute bearing characteristic frequencies. The specific unit used in lines 230 and 240 is not critical, as long as it is the same for both.

```
10 !Bearing frequency calculation
20 ! for HP Instrument Basic
30 !
40 ASSIGN @Hp35665a TO 800
50 DIM D$[32],L$[7]
60 DEG
70 CLEAR SCREEN
80 OUTPUT @Hp35665a;"DISP:PROG:MODE LOW"
90 INPUT "Enter bearing description:",D$
100 INPUT "Enter ball diameter,pitch diameter:",B,P
110 INPUT "Enter contact angle, # of balls:",A,N
120 INPUT "Enter RPM(0 for ORDERS)",F
130 IF F=0 THEN
140   L$=" Orders"
150 ELSE
160   L$=" Hertz"
170 END IF
180 IF F=0 THEN F=60
190 F=F/60  !Convert RPM to Hz
200 PRINT USING "/"
210 PRINT D$
220 PRINT USING "/"
230 PRINT USING "14A,2D.3D,7A";"Ball diameter:",B," inches"
240 PRINT USING "14A,2D.3D,7A";"Pitch diameter:",P," inches"
250 PRINT USING "14A,2D,8A";"Contact angel:",A," degrees"
260 PRINT USING "16A,DD";"Number of balls:",N
270 IF F>1 THEN PRINT USING "6A,5DZ.DD,4A";"Speed:",F*60," rpm"
280 PRINT "        -------------"
290 IMAGE 17A,4D.2D,7A
300 PRINT USING 290;"Ball pass--outer:",F/2*N*(1-B/P*COS(A)),L$
310 PRINT USING 290;"Ball pass--inner:",F/2*N*(1+B/P*COS(A)),L$
320 PRINT USING 290;"Ball spin:",F/2*(P/(2*B)*(1-B/P*COS(A))^2),
330 PRINT USING 290;"Fund. train:",F/2*(1-B/P*COS(A)),L$
340 END
```

the shaft speed. This is the reason for the factor of 1/2 in the formulas. (2) The term in parentheses is an adjustment for the diameter of the component in question. For example, a ball passes over defects on the inner race more often that those on the outer race, because the linear distance is shorter. Vibration components at the fundamental-train frequency, which occurs at a frequency lower that running speed, is usually caused by a severely worn cage.

Rolling-element bearing frequencies are transmitted well to the machine case (because the bearings are stiff), and are best measured with accelerometers. For bearings which provide axial support, axial measurements often provide the best sensitivity to defect vibration (because machines are usually more flexible in this direction).

Example Spectra
The example spectrum of Figure 4.2-4 is the result of a defect in the outer race. A printout of the bearing data and characteristic frequencies, computed with the program given in Figure 4.2-2, appears below the spectrum. Note the sidebands at running speed which are characteristic of most beginning spectra.

The spectrum in Figure 4.2-5 is also the result of a defect in the outer race. In this case, the characteristic ball-pass frequency has disappeared, but its harmonics remain. The component around 200 Hz is the gearmesh vibration.

Some Details of Spectrum Generation
To give you better insight into how bearing spectra are generated, we'll take a look at some simulated bearing signals and their resulting spectra. The characteristics we will focus on are:

(1) the impulsive nature of bearing vibration (which produces high frequency components),

(2) the effect of multiple defects, and

(3) modulation of the bearing characteristic frequencies by running speed.

In contrast to the sinusoidal vibration produced by imbalance, vibration produced by bearing defects is impulsive, with much sharper edges. The effect of these sharp edges is a large number of higher frequency harmonics.
In Figure 4.2-6, the lower trace is a time display of a simulated defect and the upper trace is the corresponding frequency spectrum. The defects are spaced at 10 ms intervals, resulting in a harmonic spacing of 100 Hz (1/10 ms) in the frequency spectrum.

Factors That Modify Frequency Characteristics

While the computation of characteristic bearing frequencies is straightforward, several factors can modify the vibration spectrum that results from bearing defects.

A. Bearing frequencies are usually modulated by residual imbalance, which will produce sidebands at running frequency (see Figure 4.2-9). Other vibration can also modulate (or be modulated by) bearing frequencies, and bearing spectra often contain components that are sums or differences of these frequencies (see Section 5.3).

B. As bearing wear continues and defects appear around the entire surface of the race, the vibration will become much more like random noise, and discrete spectral peaks will be reduced, or disappear. This will also be the case with roughness caused by abrasive wear or lack of lubrication. Another variation that occurs in advanced stages is concentration of the defect energy in higher harmonics of the bearing characteristic frequency (see Figure 4.2-6).

C. Some of these frequencies will appear in the vibration spectrum of a good bearing. This is usually due to production tolerances, and does not imply incipient failure.

D. To modify the formulas for a stationary shaft and rotating outer race, change the signs in equations (1) and (2) of table 4.2.

E. Contact angle can change with axial load, causing small deviations from calculated frequencies.

F. Small defects in stationary races which are out of the load zone will often only produce noticeable vibration when loaded by imbalance forces (ie. once per revolution).

Two important consequences of the high frequency content are:

A. High-frequency resonances in the bearing and machine structure may be excited, resulting in non-order related components not produced by other defects (except gears). One type of vibration meter designed for early detection of bearing defects de-pends on these high frequencies (20-50 kHz) to excite the natural frequency of a special accelerometer. (With no exciting frequency in this range, the output of the transducer is very low.) This type of instrument can pro-duce misleading results if the accelerometer is not carefully mounted or if the defect is such that little high frequency energy is produced.

B. High-frequency content tends to indicate the seriousness of the flaw, since shallow defects will tend to be more sinusoidal, producing fewer high-frequency defects.

Figure 4.2-3
As bearing defects progress, the vibration becomes more like random noise, and spectral peaks tend to disappear.

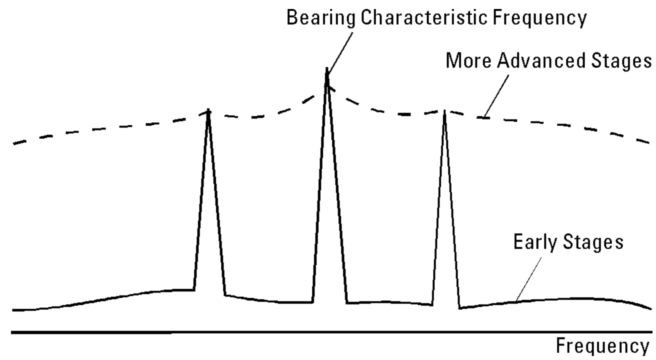


Figure 4.2-4
HP Instrument Basic print out of bearing data and characteristic frequencies with corresponding spectra. The result of a single defect on outer race is evident.

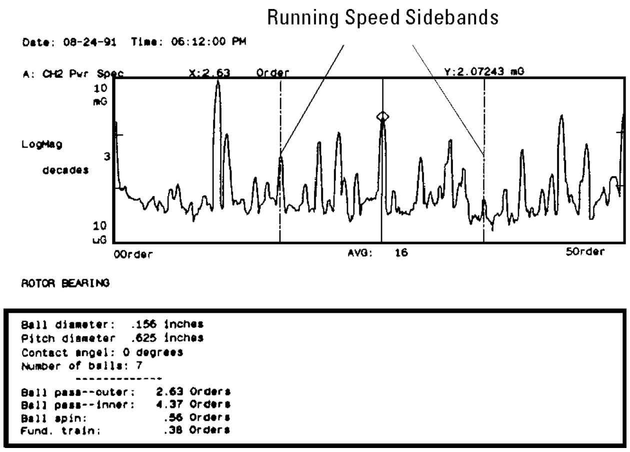


Figure 4.2-5
In this example of an outer race defect, the component at the ball pass (outer race) frequency has disappeared, but its harmonics remain. This is characteristic of advanced stages of a defect.

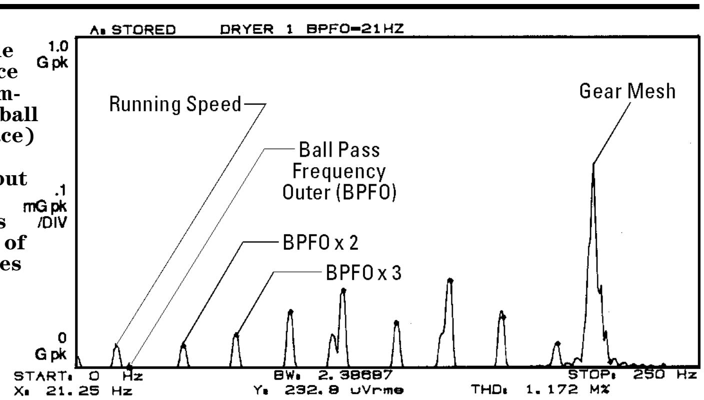
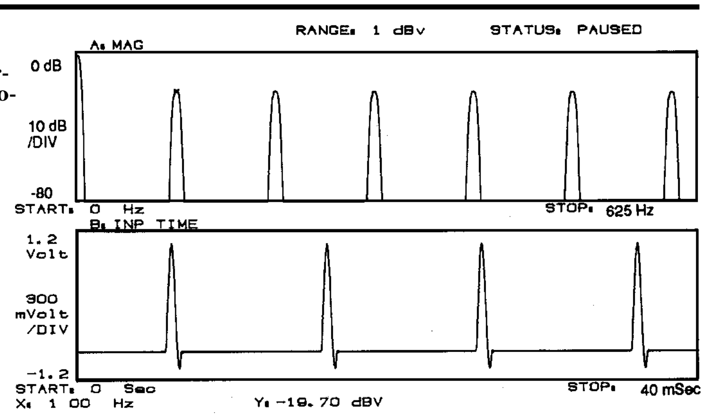


Figure 4.2-6
The impulsive nature of bearing defects produces a large number of harmonics spaced at the characteristic frequency.



Multiple Defects and Running Speed Sidebands

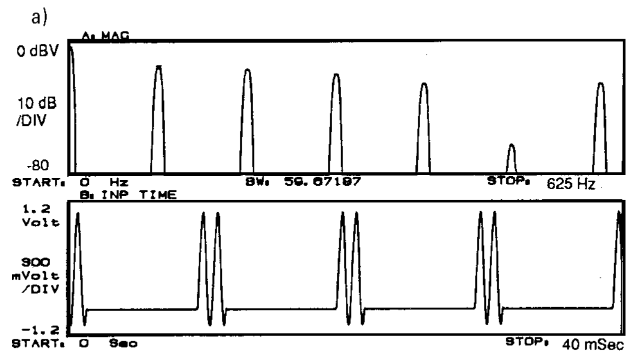
The characteristic spectrum of multiple bearing defects is difficult to predict, depending heavily on the nature of the defects, Figures 4.2-7(a) and (b) show two simulated multiple defects and their resulting spectra. Note that as long as the sequence repeats itself at the appropriate characteristic frequency, the spacing of the harmonics will be at that frequency. in this case, only the harmonic amplitudes will change.

Every machine has some residual imbalance which will amplitude modulate the bearing frequencies. In Figure 4.2-8, a bearing defect pulse is being modulated by imbalance. The imbalance component appears at the 21 Hz running speed, and as sidebands around the bearing frequency harmonics. This type of spectrum is common with bearing defects. Note that other defects, such as looseness or misalignment, will also modulate the bearing frequencies.

4.3 Oil Whirl in Fluid-Film Bearings

Rotors supported by fluid-film bearings are subject to instabilities not experienced with rolling element bearings. When the instability occurs in a flexible rotor at a critical speed, the resulting vibration can be catastrophic. Several mechanisms exist for producing instabilities, including hysteresis, trapped fluid, and shaft vibration interacting with bearings. In this section we will discuss only fluid-bearing instabilities, which are the most common.

Figure 4.2-7 Two simulated examples of multiple defects. Note that the harmonic spacing remains at the characteristic frequency.



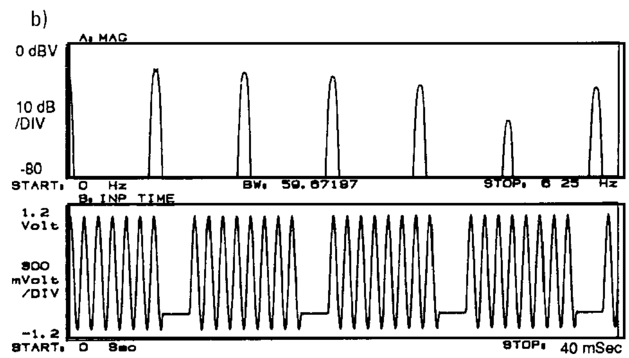
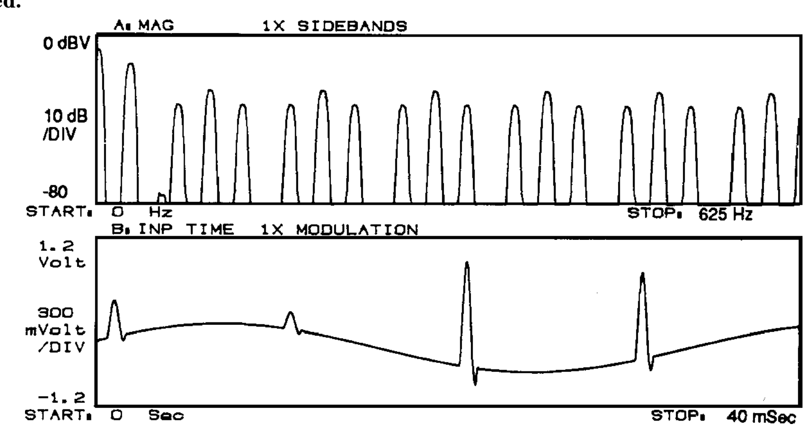


Figure 4.2-8 Bearing frequencies are almost always modulated by residual imbalance at running speed.



A basic difference exists between vibration due to instability, and vibration due to other faults such as imbalance. Consider the case of a shaft imbalance. Vibration of the shaft is a forced response to the imbalance force, occurs at the same frequency, and is proportional to the size of the force. Instability, on the other hand, is a self-excited vibration that draws energy into vibratory motion that is relatively independent of the rotational frequency. The difference is subtle, but has a profound effect on measures taken to address the problem.

Oil Whirl and Whip

Deviation from normal operating conditions (attitude angle and eccentricity ratio) are the most common cause of instability in fluid-film bearing supported rotors. As shown in Figure 4.3-1, the rotor is supported by a thin film of oil. The entrained fluid circulates at about 1/2 the speed of the rotor (the average of shaft and housing speeds). Because of viscous losses in the fluid, the pressure ahead of the point of minimum clearance is lower than behind it. This pressure differential causes a tangential destabilizing force in the direction of the rotation that results in a whirl – or precession – of the rotor at slightly less than 1/2 rotational speed (usually 0.43 - 0.48).

Whirl is inherently unstable, since it increases centrifugal forces which in turn increase whirl forces. Stability is normally maintained through damping in the rotor-bearing system. The system will become unstable when the fluid can no longer support the shaft, or when the whirl frequency coincides with a shaft-natural frequency.

Changes in oil viscosity or pressure, and external preloads are among the conditions that can lead to a reduction in the ability of the fluid to support the shaft. In some cases, the speed of the machine can be reduced to eliminate instability until a permanent remedy can be found. Stability sometimes involves a delicate balance of conditions, and changes in the operating environment may require a bearing redesign (e.g. with tilting pad or pressure dam designs). Whirl may also cause instability when the shaft reaches twice critical speed. At this speed, the whirl (which is approximately 1/2 running speed) will be at the critical speed, resulting in a large vibration response that the fluid film may no longer be able to support. The spectral map display of Figure 4.3-2 illustrates how oil whirl becomes unstable oil whip when shaft speed reaches twice critical and the oil whirl coincides with a rotor-natural frequency. Whirl must be suppressed if the machine is to be run at greater than twice the critical speed.

Figure 4.3-1 A pressure differential in fluid-film bearings produces a tangential force that results in whirl.

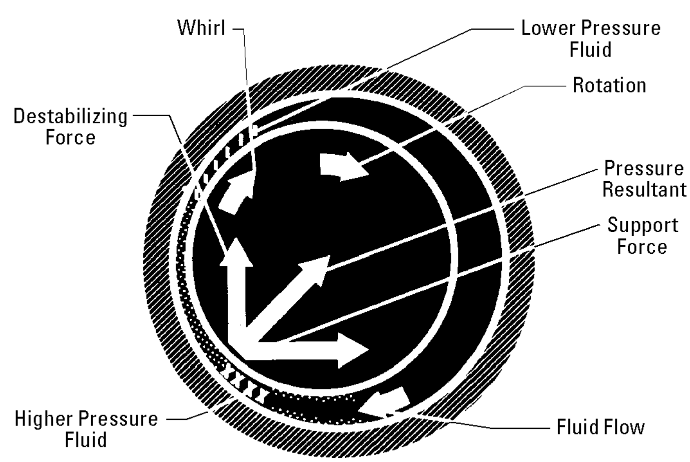
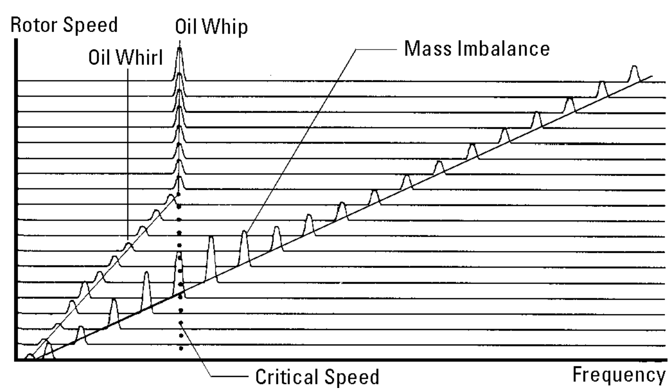


Figure 4.3-2 A spectral map showing oil whirl becoming oil whip instability as shaft speed reaches twice critical.



4.4 Misalignment

Vibration due to misalignment is usually characterized by a 2x running speed component and high axial vibration levels. When a misaligned shaft is supported by rolling-element bearing, these characteristic frequencies may also appear. Phase, both end to end on the machine and across the coupling, is a useful tool for differentiating misalignment from imbalance.

Misalignment takes two basic forms: (1) preload from a bent shaft or improperly seated bear-ing, (2) offset of the shaft center lines of machines in the same train and (3) angular misalign-ment. Flexible couplings increase the ability of the train to tolerate misalignment; however, they are not a cure for serious alignment problems. The axial component of the force due to misalignment is shown in Figure 4.4-2. Machines are often more flexible in the axial direction, with the result that high levels of axial vibration usually accompany misalignment. The high axial levels are a key indica-tor of misalignment.

High second harmonic vibration levels are also a common result of misalignment. The ratio of 1x to 2x component levels can be used as an indicator of severity. Second harmonics are caused by stiffness asymmetry in the ma-chine and its supports, or in the coupling. This asymmetry causes a sinusiodal variation in response level – a form of rotating imped-ance vector. The vibration that results from the rotating force and impedance vectors contains a component at twice the rotating frequency, as shown in Figure 4.4-1.

Vibration due to misalignment often also contains a large number of harmonics, much like the characteristic spectra of loose-ness and excessive clearance. The key distinguishing feature is a high 2x component, especially in the axial direction.

Using Phase to Detect Misalignment

As shown in Figure 4.4-2, the axial vibration at each end of the machine (or across the coupling) is 180º out of phase. This relation-ship can be used to differentiate misalignment from imbalance, which produces in-phase axial vibration. This test cannot be used in the radial direction, since imbalance phase varies with the type of imbalance. Relative phase can be measured with a single-channel DSA using a keyphaser reference, or directly with a dual-channel DSA (see Section 6.8).

Several notes of caution relative to phase measurements are appropriate at this point:

A. Machine dynamics will affect phase read-ings, so that the axial phase relationship may be 150° or 200° rather than precisely 180°.

B. Transducer orientation is important. stages of gear defects are often difficult to analyze. Transducers mounted axially to the outside of the machine will most often be oriented in opposite directions. If this is the case, a 180° phase relationship will be measured as 0°.

C. Great care must be exercised when meas-uring relative phase with a single-channel DSA. Two measurements are required, each referenced to the shaft with a keyphasor (or similar reference). These measurements should be made at the same speed. In gen-eral, you should make more than one meas-urement at each point to insure that phase readings are repeatable.

Figure 4.4-1 Alignment problems are usually char-acterized by a large 2 x running speed component, and a high level of axial vibration.

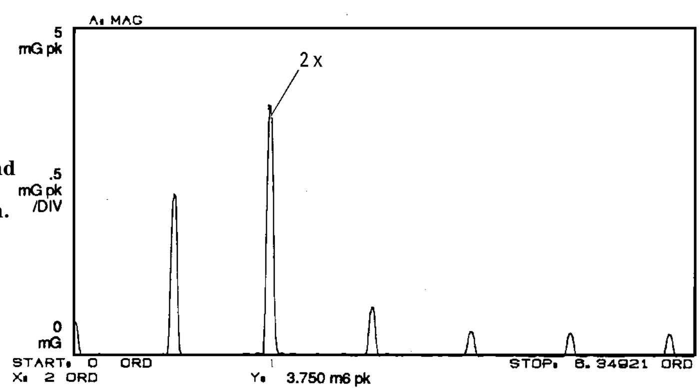
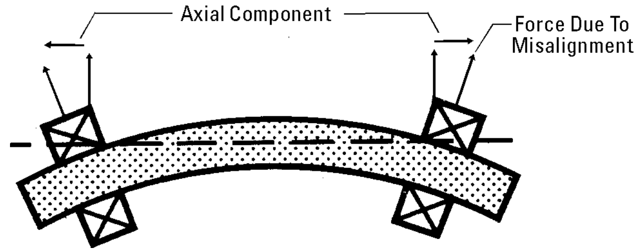


Figure 4.4-2 A bent or mis-aligned shaft results in a high level of axial vibration.



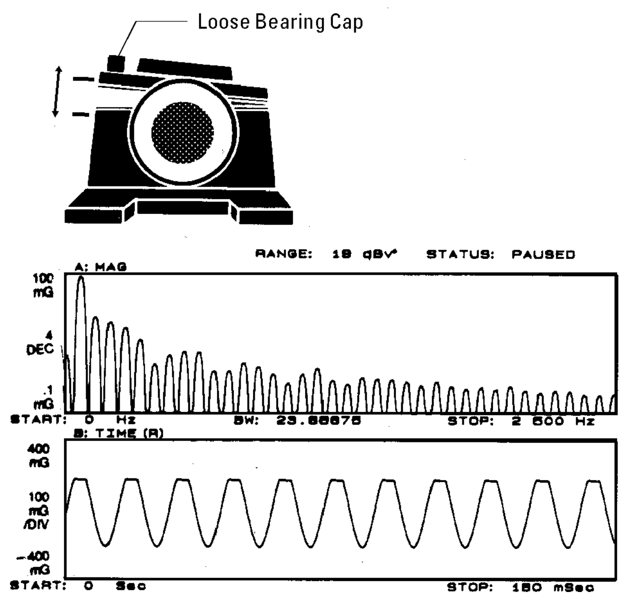
4.5 Mechancial Looseness

Mechanical looseness usually involves mounts or bearing caps, and almost always results in a large number of harmonics in the vibration spectrum. Components at integer fractions of running speed may also occur. Looseness tends to produce vibration that is directional, a characteristic that is useful in differentiating looseness from rotational defects such as imbalance. A technique that works well for detecting and analyzing looseness, is to make vibration measurements at several points on the machine. Measured vibration level will be highest in the direction and vicinity of the looseness. Also measuring vibration level on a bolt and comparing the level measured on the housing can pinpoint where to shim and torque.

The harmonics that characterize looseness are a result of impulses and distortion (limiting) in the machine response. Also, measuring vibration level on a bolt and comparing the level measured on the housing can pinpoint where to shim and troque. Consider the bearing shell in Figure 4.5. When it is tight, the response to imbalance at the transducer is sinusoidally varying. When the mounting bolt is loose, there will be truncations when the looseness is taken up. While these waveforms are idealized, the mechanism for producing harmonics should be clear. The general term for deviation from expected behavior, as when the sinusoidal vibration is interrupted by a mechanical limit, is non-linearity.

Belt drives present one situation where looseness does not result in a large number of harmonics. In this case, the impacts and sharp truncations are damped by the belt and the resulting vibration is largely once per revolution. The directionality that usually accompanies looseness results in vibration levels that vary significantly with transducer direction. In other words, while imbalance response is usually about the same in horizontal and vertical directions, looseness in a mount that produces a large vertical component may produce a much smaller horizontal component.

Figure 4.5 Looseness usually results in a truncated waveform that produces a spectrum with a large number of both odd and even harmonics.



4.6 Gears

Gear problems are characterized by vibration spectra that are typically easy to recognize, but difficult to interpret. The difficulty is due to two factors: (1) it is often difficult to mount the transducer close to the problem, and (2), the number of vibration sources in a muilti-gear drive result in a complex assortment of gear mesh, modulation, and running frequencies. Because of the complex array of components that must be identified, the high resolution provided by a DSA is a virtual necessity. It is helpful to detect problems early through regular monitoring, since the advanced stages of gear defects are often difficult to analyze. Baseline vibration spectra are helpful in analysis because high-level components are common even in new gear boxes. Baseline spectra taken when the gearbox is in good condition make it easier to identify new components, or components that change significantly in level.

Hints On Gear Analysis

A. *Select And Mount Transducers Carefully.* If gearmesh or natural frequencies above 2000 Hz are expected, use an accelerometer. Mounting should be in the radial direction for spur gears, axial for gears that take a thrust load, and as close to the bearings as possible.

B. *Determine Natural Frequencies.* Since recognition of natural frequencies is so important for analysis, take every opportunity to determine what they are. This can be done by impacting the shaft of the as-sembled gearbox, and measuring the vibration response of the housing. This measurement should be done with a two-channel DSA for best results (Section 6.8), but a single-channel measurement will give you an idea of the frequencies to expect.

C. *Identify Frequencies.* Take the time to diagram the gearbox, and identify gear-mesh and shaft speed frequencies. Even if you don't know the natural frequencies, shaft speed sidebands will often indicate the bad gear.

Characteristic Gear Frequencies

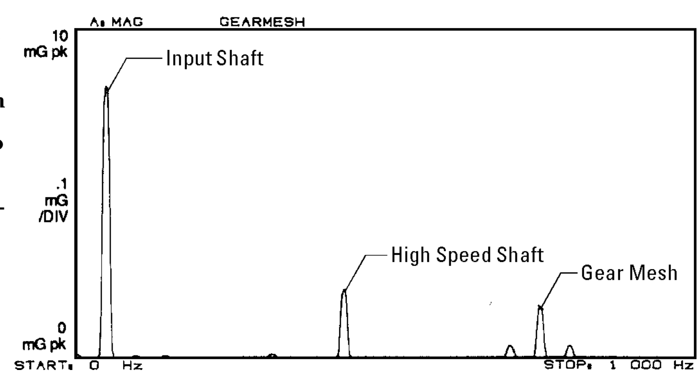
A. Gear Mesh: This is the frequency most commonly associated with gears, and is equal to the number of teeth multiplied by the rotational frequency. Figure 4.6-2 is a simulated vibration spectrum of a gearbox with a 15-tooth gear running at 3000 rpm (50 Hz). The gear-mesh frequency is 15 x 15 = 750 Hz. This component will appear in the vibration spectrum whether the gear is bad or not. Low-level-running-speed side-bands around the gearmesh frequency are also common. These are usually caused by small amounts of eccentricity or backlash.

The amplitude of the gearmesh component can change significantly with operating conditions, implying that gearmesh level is not a reliable indication of condition. On the other hand, high-level sidebands or large amounts of energy under the gearmesh or gear-natural-frequency components (Figure 4.6-2), are a good indication that a problem exits.

B. Natural Frequencies: The impulse that results from large gear defects usually excites the natural frequencies of one or more gears in a set. Often this is the key indication of a fault, since the amplitude of the gearmesh frequency does not always change. In the simulated vibration spectrum of Figure 4.6-2, the gearmesh frequency is 1272 Hz. The broad-band response around 600 Hz is centered on a gear-natural frequency, with sidebands at the running speed of the bad gear. The high-resolution-zoomed spectrum of 4.6-2(b) shows this detail.

C. Sidebands: Frequencies generated in a gearbox can be modulated by backlash, eccentricity, loading, bottoming, and pulses produced by defects. The side-bands produced are often valuable in determining which gear is bad. In the spectrum of Figure 4.6-2(b), for example, the sidebands around the natural frequency indicate that the bad gear has a running speed of 12.5 Hz. In the case of eccentricity, the gearmesh frequency will usually have sidebands at running speed.

Figure 4.6-1 The characteristic spectrum of a gearset in good condition contains components due to running speed of both shafts, and gear-meshing frequency.



4.7 Blades and Vanes

Problems with blades and vanes are usually characterized by high fundamental vibration or a large number of harmonics near the blade or vane passing frequency. Some components of passing frequency (number of blades or vanes x speed) are always present, and levels can vary markedly with load. This is especially true for high speed machinery, and makes the recording of operating parameters critical. It is very helpful in the analysis stage to have baseline spectra for several operating levels.

If a blade or vane is missing, the result will typically be imbalance, resulting in high 1x vibration. For more subtle problems such as cracked blades, changes in the vibration are both difficult to detect and difficult to quantify. Detection is a problem, especially in high- speed machinery, because blade vibration can't be measured directly. Strain gauges can be used, but the signal must be either tele-metered or transferred through slip rings. Indirect detection produces a spectrum that is the result of complex interactions that may be difficult to explain. This, combined with the large variation of levels with load, make spectra difficult to interpret quantitatively.

One characteristic that often appears in missing- or cracked-blade spectra is a large number of harmonics around the blade passing frequency. Figure 4.7 shows how a space in the vibration signal greatly increases the number of harmonics without changing the fundamental frequency.

Figure 4.6-2 Gear natural frequencies, excited by impulses from large defects, are often the only indication of problems. The zoom spectrum in (b) shows the natural frequency, with sidebands that correspond to running speed of the bad gear.

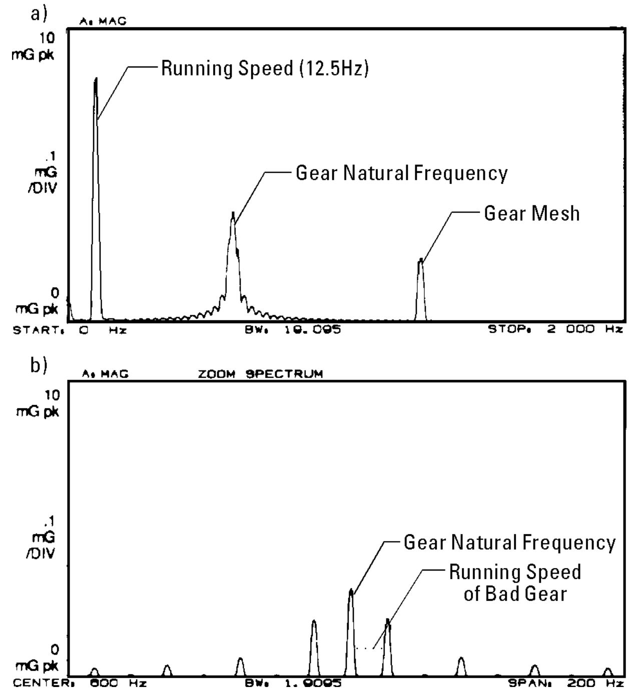
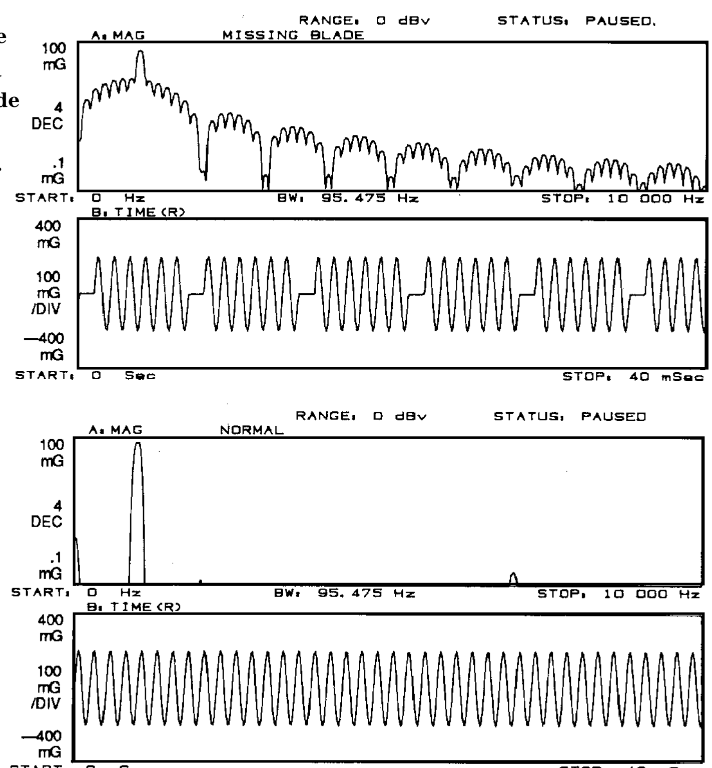


Figure 4.7 A space in the vibration signal caused by a missing blade results in a large number of harmonics. A missing blade usually also causes enough imbalance to significantly increase the 1 x level.



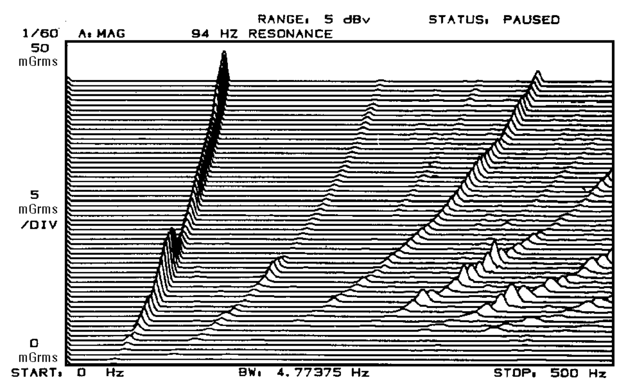
4.8 Resonance

Problems with resonance occur when natural frequencies of the shaft, machine housing, or attached structures are excited by running speed (or harmonics of running speed). These problems are usually easy to identify because levels drop appreciably when running speed is raised or lowered. Spectral maps are especially useful for detecting resonance vibration because the strong dependence on rotational speed is readily apparent (see Figure 4.8). Phase is also a useful tool for differentiating resonances from rotationally related components. Say, for example, that you encounter a high level of vibration at 16-times running speed. If the vibration is rotationally related (e.g. a blade passing frequency), the phase relative to a keyphasor signal or residual imbalance will be constant. If the vibration is a resonance, the phase will not be constant. This is a useful technique when it is not practical to vary the speed of the machine.

Piping is one of the most common sources of resonance problems. When running speed coincides with a natural frequency of the pipe, the resulting vibration will be excessive, and strain on both the pipe and the machine can lead to early failure. The most logical approach is to change the natural frequency of the pipe. It can be raised by making the pipe shorter or stiffer (e.g. by adding a support), or lowered by making the pipe longer (see Figure 2.1-9). The same rules apply to any attached structure. Structural analysis of the structure by measuing operating mode shapes is useful in determining optimal positioning of supports and braces.

Shaft resonance problems in high-speed machinery are sometimes caused by changes in the stiffness provided by fluid-film bearings, load changes, or by the effects of machines added to the train. Bearing wear, for example, can reduce the stiffness of the shaft/bearing system, and lower the resonant frequency to running speed multiples. Coupling changes can raise or lower torsional natural frequencies to running speed. The dynamics of these situations can be quite complex, and are beyond the scope of this note. Hewlett-Packard application note AN 243-3 deals with the topic of measuring the resonance and structural properties of machines in some detail. The key is to understand that maintenance and installation related factors can alter assumptions made in the rotor design.

Figure 4.8 Spectral Maps are especially useful for analyzing vibration due to resonances.



4.9 Electric Motors

Excessive vibration in electric motors can be caused by either mechanical, or electromagnetic defects. The latter can often be isolated by removing power: vibration caused by electrical or magnetic defects will disappear. The high frequency resolution of DSAs is key for analyzing electrical problems in induction motors, since running speed and power-line related components are often very closely spaced (see Section 6.3 on resolution).

Vibration caused by electrical problems in induction motors can be analyzed to determine the nature of the defect. In general, a stationary defect such as a shorted stator produces a 2 x powerline frequency component. A rotating defect, such as a broken rotor bar, produces 1 x running speed with 2 x slip frequency sidebands. (Slip frequency = line synchronous frequency – running frequency).

The vibration spectrum of induction motors always contains significant components at powerline frequency times the number of poles. A great deal of research has been done on the subject of relating the spectrum of the electric supply current to specific problems. A number of commercially available software products which can readily identify electric-motor faults from frequency spectra of the current taken with a DSA and a current probe.

4.10 Summary Tables

Tables 4.10-1 (below) and 4.10-2 (next page) summarize the vibration characteristics information in this chapter. This information should be used as a guide only, since the vibration resulting from specific defects can be modified by machinery dynamics.

Table 4.10-1
Phase Characteristics of Common Vibration Sources

Source	Characteristics*
Rolling element bearing defect	Unstable
Electrical	Unstable unless synchronous motor
Gear mesh	Unstable
Imbalance	Stable unless caused by uneven loading or cavitation. Phase follows transducer location (4.1)
Looseness	Unstable; may be highly directional
Misalignment	Stable; relation between axial phase at shaft ends should be approximately 180°
Oil whirl	Unstable
Resonance	Unstable; large phase change with change in speed in rpm.

Table 4.10-2
Vibration Frequencies Related To Machinery Faults

Frequency	Possible Cause	Comments
1 x rpm	Imbalance	Steady phase that follows transducer. Can be caused by load variation, material buildup, or pump cavitation.
	Misalignment or Bent Shaft	High axial levels, ~180° axial phase relation at the shaft ends. Usually characterized by high 2x level.
	Strain	Caused by casing or foundation distortion, or from attached structures (e.g. piping).
	Looseness	Directional – changes with transducer location. Usually high harmonic content and random phase.
	Resonance	Drops off sharply with changes in speed. From attached structures or changes in attitude angle or eccentricity ratio.
	Electrical	Broken rotor bar in induction motor. 2x slip frequencies sidebands often produced.
2 x rpm	Misalignment or Bent Shaft	High levels of axial vibration.
Harmonics	Looseness	Impulsive or truncated time waveform; large number of harmonics.
	Rubs	Shaft contacting machine housing.
Sub-rpm	Oil whirl	Typically 0.43 - 0.48 rpm; unstable phase
	Bearing cage	See formula in Table 4-2.2
N x rpm	Rolling element bearings	See formulas in Table 4-2.2 Usually modulated by running speed.
	Gears	Gearmesh (teeth x rpm); usually modulated by running speed of bad gear.
	Belts	Belt x running speed and x 2 running.
	Blades/Vanes	Number of blades/vanes x rpm; usually present in normal machine. Harmonics usually indicate that a problem exists.
N x powerline	Electrical	Shorted stator; broken or eccentric rotor.
Resonance		Several sources, including shaft, casing, foundation and attached structrues. Frequency is proportional to stiffness and inversly proportional to mass.

Chapter 5

Advanced Analysis and Documentation

5.1 Practical Aspects of Analysis

In the literature and discussions on the subject of machinery vibration analysis, several factors are regularly mentioned as keys to success. In this section, we will discuss five of these factors: (1) documentation, (2) machinery knowledge, (3) severity criteria, (4) instrumentation, and (5) analysis personnel. Its important to note that a clear objective as to the purpose and scope of the measurements must be established. As noted in Chapter 1, there are a number of reasons for undertaking a vibration analysis program (for example, new machine development, quality improvement, maximize service life, maintenance program, and field balancing). It is important at the onset of the program to have a clear understanding of the purpose of the measurement; why and how each measurement is to be used; and a sound basis for making the measurement.

Chapter Overview

5. *Advanced Analysis and Documentation*
Chapters 1 through 4 provide the basic information needed for the analysis of machinery vibration. Chapter 5 contains practical information that will help in determining specific defects, and in assessing their severity.

5.1 *Pracitical Aspects of Analysis*
A discussion of 5 practical aspects for successful analysis.

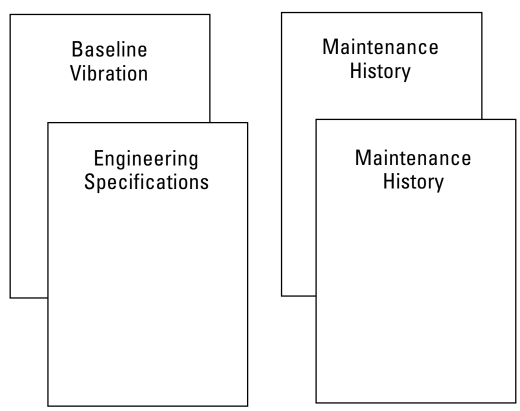
5.2 *Using Phase for Analysis*
We have discussed the importance of phase in analyzing imbalance and misalignment. This section is an extension of that discussion, and an introduction to the related concept of time averaging.

5.3 *Sum and Difference Frequencies*
Multiple defects often produce vibration components that are sums and differences of characteristic frequencies.

5.4 *Speed Normalization*
A common problem when making direct spectral comparisons is shift in frequency of vibration components caused by changes in running speed. This section discusses solutions to this problem.

5.5 *Baseline Data Collection*
Records of vibration spectra taken when a machine is in good condition or of a similar machine can provide significant insight into the interpretation of machinery vibration data. This section presents guidelines for collecting baseline data.

Figure 5.1-1
Complete documentation consists of baseline vibration spectra, maintenance history, and engineering data.



Documentation

Thorough documentation often provides the information required to successfully analyze a vibration problem. Complete documentation includes baseline vibration spectra, machine maintenance history, and engineering data.

The baseline vibration measurements made on a machine provide a reference for detecting changes or differences which indicate problems — and for identifying significant components when problems do occur. Without this information, you can easily waste time determining the source of a vibration component that is perfectly normal and expected.

History records include machine failures, and vibration spectra before and after significant modifications or repairs. These records are often in the form of computer digital data and organized in an information data base. You should, for example, be able to immediately identify the changes in the vibration spectrum of a machine that has had a major modification. This will immediately give some insight into the relationship between major machinery components and the vibration spectrum.

Engineering data includes bearing and gear parameters used to calculate characteristic frequencies, and machine dynamic models used to predict vibration response characteristics (or as an alternative, the results of a structural test conducted on the machine).

Also useful are manufacturer's data on vibration limits and characteristics. This data will not always be easy to obtain — the key is to collect the available data before hand so it is available for correlation with measured vibration data.

Computers have become an indispensable tool in organizing records and data. A number of software suppliers offer products that organize vibration data, analyze trends (Fig 5.1-1), and provide detailed correlation of spectra with known characteristic frequencies. In some cases such as ball bearings, they even provide a data base of characteristic frequencies by product number.

Machinery Knowledge
The design and operating characteristics of a machine determine both the type of defects that are possible, and the vibration response to those defects. Vibration analysis is difficult without a working knowledge of these characteristics. Another important consideration is the effect of changes in operating condition on measured vibration. By understanding how vibration changes with such variables as load and temperature, you will be better able to determine whether an increased level of vibration is due to a defect, or to a change in operating conditions.

The best sources of information on these characteristics are the manufacturer of the machine, and historical records on the same or similar machines. In applications such as machinery maintenance, courses from manufacturers can provide insight into both the possible defects, and the mechanisms of vibration response for specific machines. Several baseline spectra taken under different operating conditions are useful for documenting the effects of changing operating parameters.

Severity Criteria
Once a vibration spectra is measured and its individual components identified and correlated, the problem of interpreting the severity of the vibration level (amplitude) often arises. Machines will inevitably vibrate and will also undoubtedly produce spectral components at characteristic frequencies given the resolution and sensitivity of modern DSAs. The issue then becomes — what is an acceptable level? It is difficult to generalize here but a number of sources are useful.

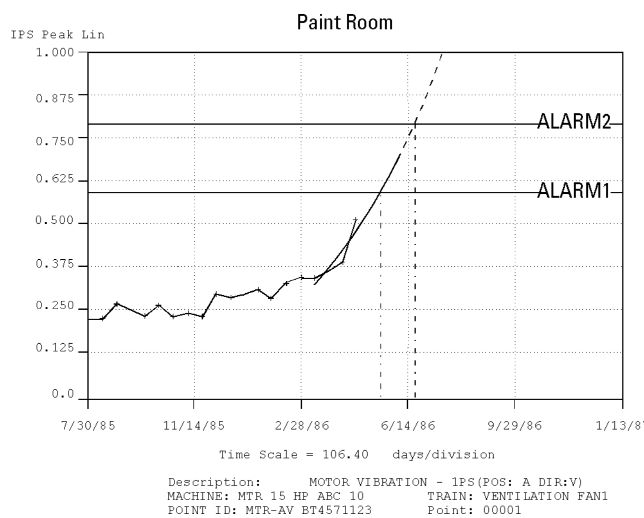


Figure 5.1-1
An example of a "trend analysis" simplified from vibration spectrum data over a period of weeks/months by post-test machinery analysis software.

Figure 5.1-2
Tables of vibration severity, like this one published by the ISO,* are most useful as guidelines rather than absolute limits.

Vibration Severity In./sec.	mm/sec	Support Classification Hard Supports	Soft Supports
.017	.045	good	good
.028	.071		
.044	1.12		
.071	1.8	satisfactory	
.11	2.8		satisfactory
.18	4.5	unsatisfactory	
.28	7.1		unsatisfactory
.44	11.2	impermissible	
.71	18.0		impermissible
1.10	28.0		
2.80	71.0		

* This material is reproduced with permission from International Organization for Standardization Standard 3945-1977, Mechanical Vibration of Large Rotating Machines with Speed Range from 10 to 200 rev/s - Measurement and Evalutation of Vibration Severity in Situ, copyrighted by the American National Standards Institute, 1430 Broadway, New York, NY 10018.

References for severity include published vibration standards and historic vibration measurements.

The table in Figure 5.1-2 is an example of a published vibration standard. This particular standard is from the International Standards Organization (ISO). To make the standard more applicable to a wide range of machines, a distinction is made in the severity criteria between soft and hard supports. The essential problem with such published standards is that they are too general to allow us to make high accuracy judgements with the power and accuracy available with modern DSAs. They were originally devised as a tool for interpreting severity on the basis of overall level measurements. Modern analyzers can identify components of the vibration which contribute negligibly to the overall spectral energy, but may be indicative of very important local phenomena.

A number of techniques have been developed for refining the method of defining the severity critera for frequency bands in the vibration spectrum. Figure 5.1-3 is an example of one technique that breaks the spectrum into 6-frequency bands and specifies the allowable severity level for each band based on the predominant vibration mechanism present in each particular band.

Historic vibration measurements are an excellent reference for severity measurements, because they are specific to the machine or type of machine in question. In the case of machine development or modification the historical data on a machine or class of machines can provide a valuable "yard stick" by which to began the evaluation.

While absolute vibration limits for a machine may not be known, there is a high probability that large changes in vibration level indicate something significant with respect to the operating condition of the machine. The process of monitoring vibration level for changes is referred to as, trend analysis. Since the vibration level in a machine is variable, it isn't always obvious how much change is tolerable. The best approach is to analyze the statistics of variability for each machine, and base change limits on that. An increase in vibration that exceeds two standard deviations is usually a sign of a problem. In the absence of this type of analysis, you can use a factor of 2 increase as an approximate change limit threshold.

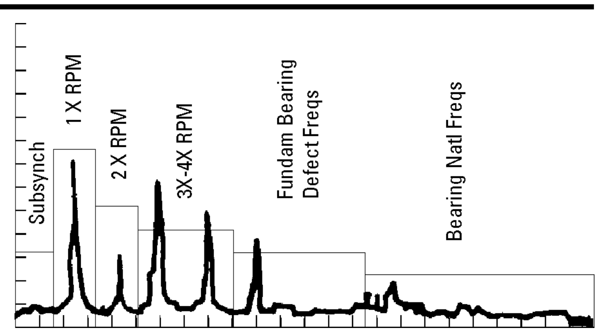
When significant changes are detected, vibration level and other key operating parameters should be monitored regularly. The rate of change of these parameters is a good indication of the severity of the problem. The existence of a consistent pattern of change is indicative of a developing problem and/or changing operating condition.

Instrumentation

A wide variety of features and capabilities are available in instrumentation ranging from transducers to DSAs to applications software. Chapter 6 addresses many of the issues in the selection of DSAs but it's important to put the instrumentation requirements in the context of the individual task being addressed. In many applications particular performance issues or features can be critical to the measurement, while others are convenience features or totally superfulous.

For example, in predictive maintenence or troubleshooting applications portablility and battery operation could outweigh considerations of dynamic range,

Figure 5.1-3
A specification of machine severity criteria by specifying individual frequency band criteria.*



* This material is reproduced with permission from reference [8]: Berry, James E., *Proven Method for Specifying Spectral Band Alarm Levels and Frequencies Using Today's Predictive Maintenance Software Systems*, Technical Associates of Charlotte, Inc., 1990).

real-time bandwidth or programmability. In the case of production quality testing of automobile engines; automation and measurement time will be more important than portability. It would be nice to have an instrument, that is portable, fast, programmable, etc.. But these attributes are not easily attainable, often not necessary and undoubtedly expensive.

Though expense is frequently the driving factor; it is important to put cost in its proper prospective, and access the return on investment. A computer and applicable software that automatically retrieves vibration data and analyzes trends can quickly pay for itself. Instrumentation with a general purpose feature set, high performance, a convenient-user interface, often finds itself being used in many applications not initially envisioned. Programmability is another cost factor that will direcly impact use ability. Instruments with built in HP Instrument Basic can greatly reduce measurement automation tasks and can circumvent the need for an external computer to control and automate processes. Easy programmability will allow less skilled personnel to collect data and do preliminary level severity checks.

People
The quality and effectiveness of a vibration analysis program is most often limited by the availability of capable and skilled personel. Successful programs are characterized by people who are properly trained and given a chance to develop analysis expertise. In some applications it is neither practical nor desirable to train individual operators as vibration analysts. An example would be a production assembly person. In this case the need for training can be aleviated to some extent by built-in automation capability which can make repetitive measurements and access the results.

Also consider and experienced consultant to help setup and establish your maintenance program. The consultant can help overcome the extensive expertise required early on in the establishment of a machine vibration monitoring program. With time, more and more of the task can be taken over by local personnel.

5.2 Using Phase for Analysis

The usefulness of the phase spectrum as a means for differentiating between defects with similar amplitude spectra has already been discussed. We will present a more general discussion of the subject in this section. Time averaging, a powerful processing technique related to phase, will also be described.

Table 5.2-1 Phase Characteristics of Common Vibration Sources

Source	Characteristics
Rolling element bearing effect	Unstable
Electrical	Unstable unless synchronous motor
Gear mesh	Unstable
Imbalance	Stable, unless caused by uneven loading or cavitation. Phase follows transducer location (4.1)
Looseness	Unstable , may be highly directional
Misalignment	Stable, relation between axial phase at shaft ends should be approximately 180°
Oil Whirl	Unstable
Resonance	Unstable, large phase change with change in speed in rpm.

In general, the phase of vibration caused by a defect will either be stable or unstable relative to a fixed reference (i.e. keyphasor). The nature of this relationship is shown in Table 5.2-1. Figure 5.2-1 is a sequence of vibration spectra that shows phase for imbalance (1x), and running speed harmonics, and unstable phase for powerline related components. Also, the relative phase relationship between vibration at different points on a machine can be used to differentiate between faults — as in the case of misalignment and imbalance (see Section 4.4).

Instrumentation required for phase measurements is shown in Figure 5.2-2 and 5.2-3. The keyphasor senses shaft rotation and serves as the phase reference. The phase of vibration that is synchronous (i.e. an integer multiple) with rotation is constant, while that of nonsynchronous vibration varies. Relative phase measurements can be made sequentially, as long as the same reference (i.e.

keyphasor) is used (see Section 6.8 on a dual-channel DSA). Running speed should remain constant between measurements to minimize the phase effects of mechanical impedance. Relative phase measurements on flexible rotors must include considerations of shaft dynamics.

The keyphasor, which is often a proximity sensor that detects a keyway or setscrew, provides a relatively good signal for triggering. Because the gap of the proximity probe can vary with speed, there can be some error in the phase measurement as the trigger point shifts with the gap causing the actual position on the shaft of the trigger point to vary to some degree. A better trigger is often obtainable with an optical sensor and reflective tape or paint. Sometimes an electrical signal such as the spark ignition on a gasoline engine is used, though here again, there is a propensity for this to shift under differing operating conditions (vacuum advance).

The actual vibration signal is usually not suitable for triggering even though there exists some instrumentation designed for this purpose. This method relys on the fact that the imbalance (1x) is often the largest component; however, noise in the spectrum adds uncertainty to level, and thus trigger timing. If an independent tachometer reference is not practical, then it may be possible to use a band-pass or low pass filter to reduce the level of noise and higher frequency components in the vibration signal to provide a more stable trigger.

When measuring relative phase between two ends of a machine, it is important to mount the transducer with the same orientation. When measuring axial vibration, for example, if both transducers face the machine, they are mounted 180° out of phase.

Thus vibration due to misalignment, which you would expect to be 180° out of phase, will be measured as in phase. It is likewise

Figure 5.2-1
A sequence of vibration spectra with phase shows constant phase for imbalance (1 x and harmonics), and unstable phase for power-line components (60 Hz harmonics).

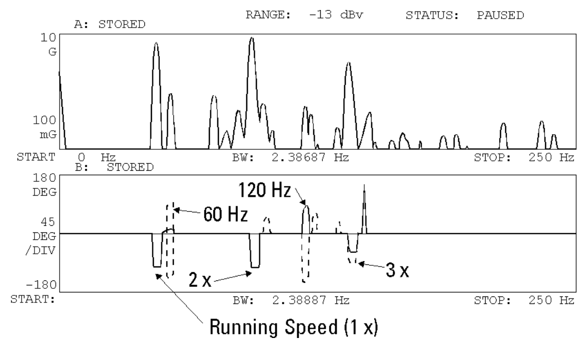


Figure 5.2-2
Instrumentation setup for phase measurements and time averaging.

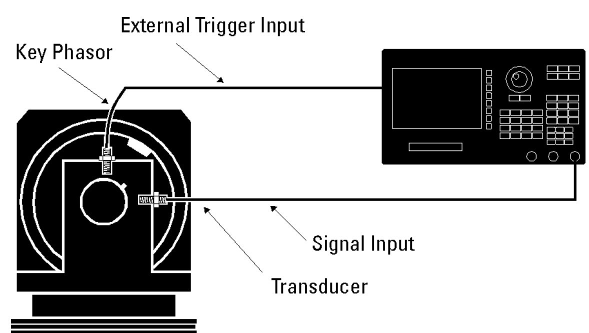
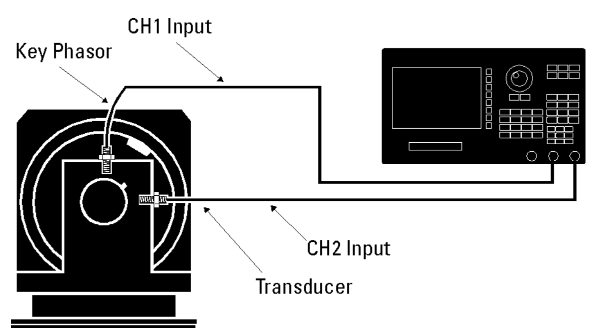


Figure 5.2-3
Alternate instrumentation setup for relative phase mesurements using 2 channel DSA.



important to remember that phase response of a system is related to the variable measured; that is, displacement and acceleration measurements are 180° out of phase and if the phase between these is being compared, it is necessary to take into account any phase difference between different types of motion variables.

Time averaging is explained in Section 6.4, and is a powerful technique for eliminating nonsynchronous components from a vibration spectrum. It is most useful for reducing the level of background noise, especially vibration from other machines. It must be used with care, however, since it will reduce the level of all vibration components that are nonsynchronous, including bearing and gear frequencies. In the plot of Figure 5.2-4, a time-averaged spectrum (dashed line) is overlaid on a non-averaged spectrum. The synchronous components have not changed in level, while the nonsynchronous background noise components are greatly reduced.

5.3 Sum and Difference Frequencies

Vibration spectra often contain components that are the result of interaction between multiple vibration mechanisms. These components appear as sum and difference frequencies of the mechanisms involved, and can be useful as indications of specific problems, especially in gears and bearings. When the major frequency components are closely spaced, the difference frequency is often audible. These "beat" frequencies are common in rotating machinery and are the result of a process called "modulation".

In Figure 5.3-1, the difference between running speed at 144 Hz and the 2nd harmonic of the line frequency at 120 Hz is 24 Hz. This component appears at 24 Hz and as sidebands around the harmonics of the rotational speed.

The exact mechanisms which generate sum and difference frequencies can be quite complex and a detailed mathematical analysis is beyond the scope of this note. However, you can get a feel

Figure 5.2-4
Time averaging is effective in reducing the level of background vibration.

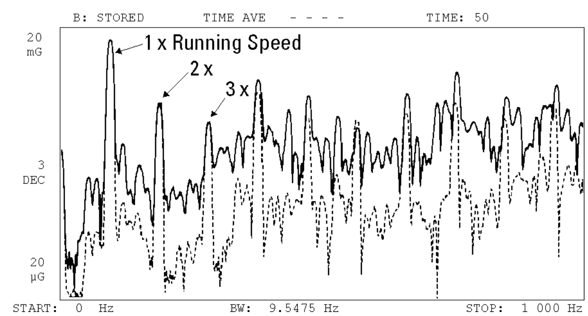


Figure 5.3-1
A vibration spectrum with sum and difference frequencies. The 24 Hz difference between rotational frequency and the 120 Hz powerline component appears both as a discrete signal, and as side-bands around rotational speed harmonics.

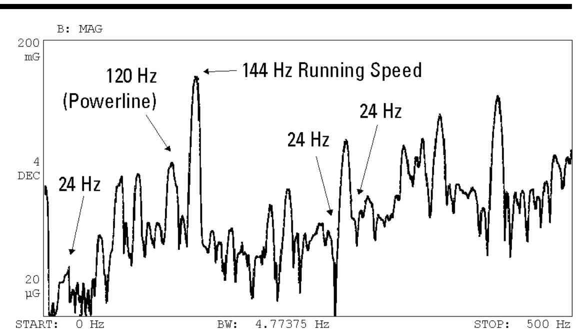
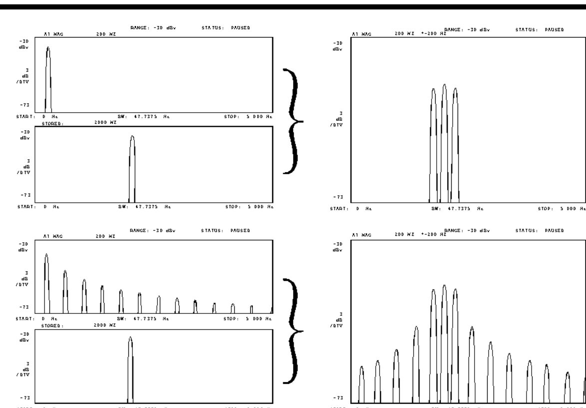


Figure 5.3-2
The number of sum and difference components depends on the number of harmonics in the signals involved.



[1] Phase and frequency modulation are also present and produce many of the same sum and difference frequencies but their phase relationships differ. A detailed discussion of this is beyond the scope of this note.

for the interactions involved by thinking of them as a form of amplitude modulation[1]. In the trigonometric identity given below,

$$\cos(f1) * \cos(f2) = 1/2[\cos(f1+f2) + \cos(f1-f2)]$$

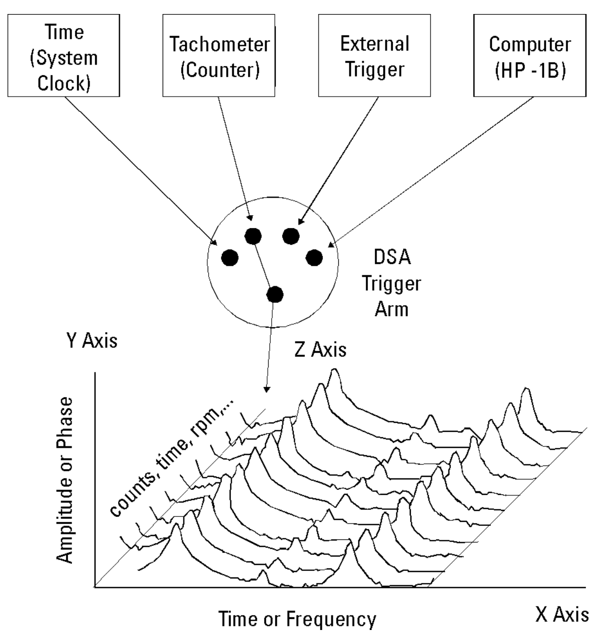
it is apparent that the interaction of one frequency with another results in sum and difference frequencies. If one of the signals contains a large number of harmonics, then multiple sum and difference frequencies will appear. This is illustrated in Figure 5.3-2. Phase can be an aid in identifying sum and difference frequencies, since it will be unstable unless the phase of both sources is stable.

The most common faults indicated by sum and difference frequencies are associated with rolling element bearings and gears.

A. Rolling element bearings. Defects in rolling element bearings are almost always modulated by residual imbalance. As the wear progresses, and characteristic frequencies are replaced by noise, these running frequency sidebands may be the only indication of trouble (see Section 4.2).

B. Gears. As pointed out in Section 4.6, gear defects often appear as gear natural frequencies with sidebands at the running speed of the defective gear. These running speed sidebands may also appear around the gearmesh frequency.

Figure 5.3-3
The Z axis in a map or waterfall display can be precisely controlled by the DSA trigger arming function.



Spectral Map and Waterfall Displays

Waterfalls and spectral maps are a useful technique for detecting changes with time and identifying speed related components in a variable-speed machine. Technically, a waterfall differs from a map only in the way in which the map is updated. Waterfalls generally are a continuously updating display with the newest spectrum appearing at the top of the display and the oldest scrolling off at the bottom (hence the name waterfall, from the appearance of the data migrating down the trace like a waterfall).

Spectral maps on the other hand are generally a fixed set of data starting at a defined time or condition (rpm) and ending as some predetermined time or number of spectra later. These are sometimes referred to as cascade plots and appear the same once the measurement has been paused. Many people use the terms interchangeably causing some confusion as to what is meant.

Figure 5.3-3 shows the topology of this type of three dimensional display. The third dimension (z-axis) is the number traces, rpm increments, or time increments in the display. This is controlled by the analyzer's trigger- arming capability which determines when a spectrum will be acquired and displayed. This allows for the very precise determination of when data will be taken and analyzed.

5.4 Speed Normalization

A common problem in machinery vibration analysis is running speed variation — both long-term and short-term. Short-term variations in speed make real-time analysis difficult. Long-term variations make point-by-point comparisons between current and baseline spectra virtually impossible. Synchronous sample control (also known as order tracking) can be used to compensate for both problems while the measurement is in progress.

Traditionally, these systems where limited to direct control of the analog-to-digital sampler by an external source synchronized to the machine running speed. Recently however, advances in digital technology have allowed for the sampling synchronization to be performed digitally, thus avoiding some of the problems of the older analog technology. Either scheme seeks to lock the sample rate to the speed of the machine so that speed related components appear at a stationary frequency. This is very useful in machinery analysis, as was discussed in Chapter 4, since most machinery defects are related to some shaft- rotation frequency. The details of controlling the sample rate and of digital order tracking are discussed in Section 6.7 and Appendix A. A good way to illustrate the effects of synchronous (or external) sample control is with spectral maps made in external and normal sample modes, as shown in Figure 5.4-1. These maps were made during a run-up. Note in the normal sample-mode map of (a), rotational speed-related components move to the right as speed increases, while fixed frequency components (e.g. structural resonances and powerline related) move straight up. In the external sample-control map of (b), rotational speed-related components move straight up the map, while fixed frequency components move to the left (they are relatively lower in frequency as speed increases).

The main advantage of synchronous sample control is that real-time displays of the order related spectral components remain fixed

Figure 5.4-1
Two spectral maps of a machine run-up illustrate the effect of external sample rate control.

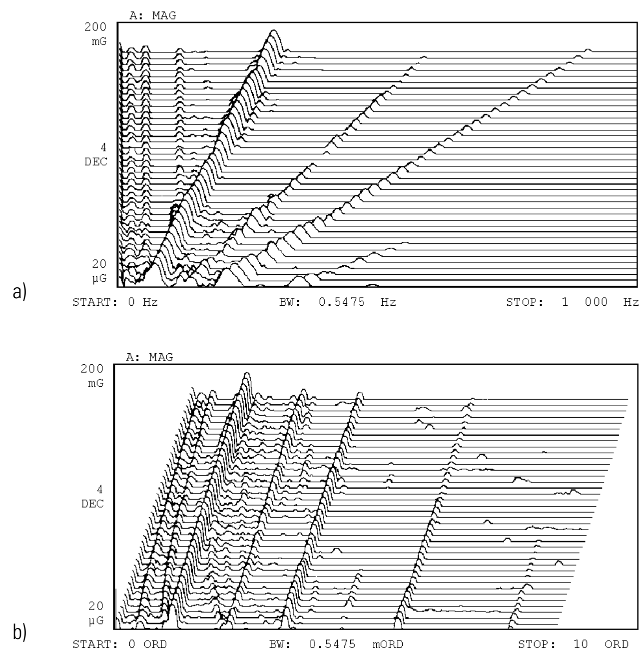
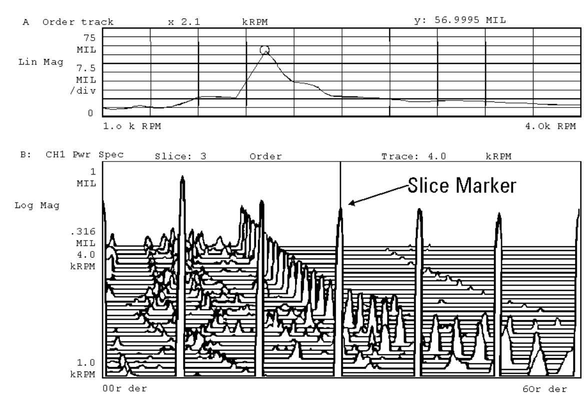


Figure 5.4-2
A "slice marker" function is used to extract the order related information from the synchronously sampled map display (lower) and and constructs an order track (upper) for the 3rd order.



within the horizontal position speed. During individual measurements (or especially with averaging) speed variations do not cause a “smearing” of the frequency over a range. Another advantage is the extraction of order tracks is greatly simplified and the accuracy improved. An order track is the plot of an individual order as the rotation speed changes. Since the frequency of these components has been normalized to a fixed value, a simple marker function can be used to extract the order track from the map display (see figure 5.4-2). Frequency can also be normalized to rotational speed after a measurement. In the display of Figure 5.4-3, note that the frequency axis and the readout are in terms of orders of rotation (multiples of running speed), rather than in frequency. This technique simply amounts to re-scaling the frequency axis when the running speed is known or can be deduced, it is not normally useful with map/waterfall displays. This normalization does not work in real time, and resolution is not a constant percentage of running speed (as with synchronous sample control). However, it is useful when no tachometer/keyphasor signal is readily available.

Short-term speed variation causes a broadening of spectral lines in the vibration spectrum, as shown in Figure 5.4-4. As speed changes during the sampling interval for one measurement, the DSA is effectively analyzing several different spectra. This results in the broadened spectral components of Figure 5.4-4(b).

Figure 5.4-3 DSA display in which the horizontal (frequency) axis is calibrated in multiples (orders) of running speed.

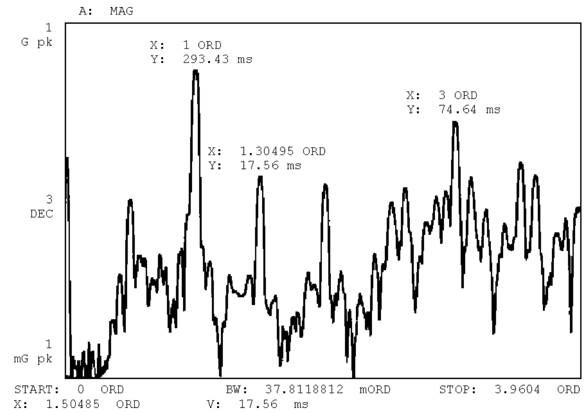
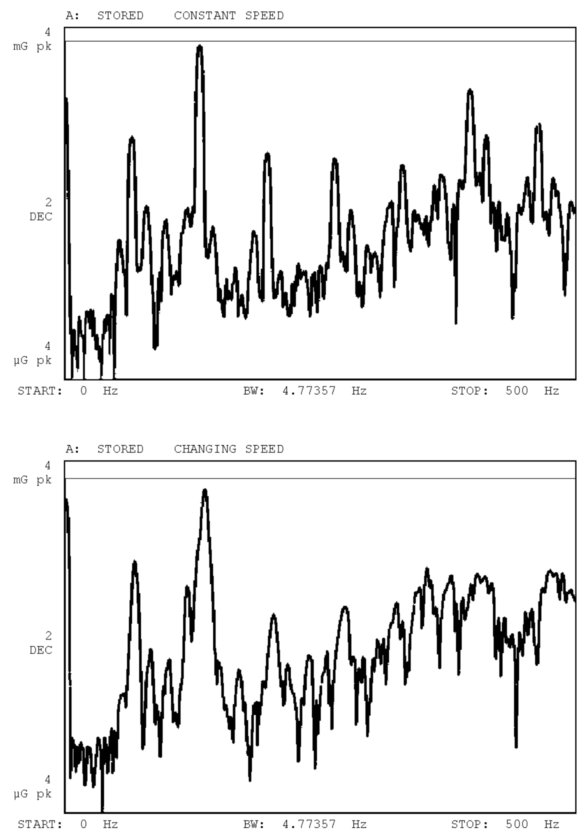


Figure 5.4-4 Short-term speed variation results in a broadening of spectral components (b).



a) Constant running speed

b) Changing running speed

5.5 Baseline Data Collection

Baseline vibration spectra are reference data that represent normal machine condition, and are essential to effective analysis. In the event of trouble, they quickly indicate the frequency components that have changed. Baseline data is also the basis for trend monitoring; it is a much more specific indicator of normal vibration than generalized vibration severity charts. To be most useful, the guidelines below should be followed in collecting baseline data. The key objective of the process is to understand the characteristics of the machine.

A. *Normalize for Speed.*
Normalizing the vibration spectrum for speed is required for direct spectrum comparison. Section 5.4 discussed the alternative methods for accomplishing this. Whichever method is chosen, some provision should be made when taking baseline data.

A spectral map/waterfall of a run-up or coast-down is also useful in dealing with changes in speed. A spectral map can quickly show how vibration level changes with speed, and the resonances and other fixed frequencies that are present in the vibration spectrum.

B. *Be Complete*
You can't take baseline data after the machine has a problem, so it is important to take all the data you can when it is operating normally. Follow the guidelines in Chapter 2 for transducer selection and placement. For machines with rolling-element bearings or gears, consider taking high- and low-frequency spectra. The low-frequency spectrum (0-500 Hz) provides good resolution for most analysis, while the high frequency spectrum (0-5 kHz) will provide a baseline for the high frequencies that can indicate problems with bearings and gears.

In addition to vibration data, operating parameters such as oil pressure. temperature, load, bearing and gear parameters should be collected. Also, any information available from the machine manufacturer regarding vibration characteristics and failure mechanisms should be included.

Thermal gradient in a machine can cause temporary misalignment; so making several temperature readings along the machine may be useful in diagnosing vibration problems.

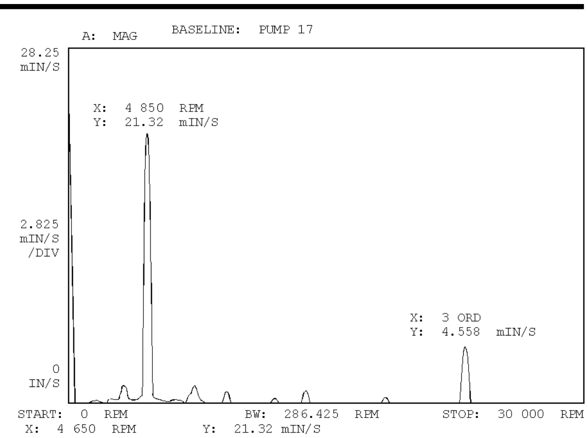
C. *Check Statistical Accuracy*
This just means that one measurement may not be representative of normal operation. For example, an adjacent machine may be vibrating excessively when baseline signatures are taken, or an older machine may already have excessive vibration levels.

The best approach is to take several spectra over time and perform a statistical analysis to yield mean and standard deviation. This results in a representative average level, and also provides a quantitative basis (e.g. 1 or 2 standard deviations) for determining whether a change in level is significant. The accuracy of these statistics can be improved by updating them with data from regular vibration monitoring data.

D. *Document the Effect of Load Vibration*
This is not strictly required, but can be invaluable when determining whether a change is due to a fault, or just a change in load.

E. *Update Regularly*
Baseline data should be updated after major repairs or changes in operating conditions.

Figure 5.5-1 Baseline data should include fully documented vibration spectra and engineering data, such as bearing and gear parameters, which can be invaluable for analysis.



Chapter 6

Dynamic Signal Analyzers

Chapter 6 describes the important measurement capabilities of DSAs as they relate to machinery vibration analysis. For a more detailed discussion of DSAs and how they work, refer to Hewlett-Packard applications note AN 243.

Chapter Overview

6.1 *Types of DSAs*
Dynamic Signal Analyzers are available in a number of different form factors and capabilities. Generally, the "classes" of DSAs can be broken into handheld/portable, benchtop instrument and computer controlled systems.

6.2 *Measurement Speed*
Machinery vibration is a dynamic phenomenon that can change quickly — so quickly that slower swept spectrum and some DSAs can completely miss key events. DSAs can capture a typical vibration signal and transform it to the frequency domain within seconds. Another method involves capturing data in a digital form and post- test processing it; thus allowing for much higher acquisition rates than is possible with on-line processing. Post-test processing also allows data to be processed and presented in different forms.

6.3 *Frequency Resolution*
Closely spaced machinery vibration signals often must be resolved for accurate analysis. Often industry standards and technical requirements dictate the use of 1/3-octave analysis, especially in the areas of noise control, acoustics and transients.

6.4 *Dynamic Range*
Vibration components are often very small relative to vibration from residual imbalance or other machines. The wide dynamic range of DSAs allow them to resolve signals less than 1/1000 the level of the background vibration or residual imbalance.

6.5 *Digital Averaging*
Machinery vibration signals often contain large amounts of background vibration that can reduce accuracy and obscure small signals. The digital averaging feature of DSAs can be used to reduce both of these effects.

6.6 *HP-IB and HP I-BASIC*[1]
The Hewlett-Packard Interface Bus is a standardized interface that makes it easy to connect a DSA to a computer, printer or digital plotter. It is important in automating repeated tests and transferring data to computer data bases and/or analysis programs. The processing power of modern DSAs has advanced to the point where HP has implemented a version of the BASIC programming language resident in the instrument. This allows for extreme flexibility in adapting the instrument to dedicated tasks.

6.7 *User Units and Unit Conversion*
DSA displays can be calibrated in vibration units such as inches/seconds and rpm. Units of vibration amplitude can also be converted to other parameters (e.g. acceleration to velocity) using the processing capabilities of DSAs.

6.8 *Synchronous Sample Rate Control*
By controlling the data-sampling rate with a tachometer pulse, the frequency axis can be normalized to rotational speed. Traditionally, this was done with external analog circuitry directly controlling the sampling rate of the analyzer. Increased digital processing power has allowed this task to be handled digitally, bringing with it increased capability and accuracy.

6.9 *Two-Channel Enhancement*
While single-channel DSAs address most of the needs of machinery analysis, dual- and multi-channel DSAs provide important enhancements such as real-time phase comparisons and transfer function measurements.

[1] HP-IB: Not just IEEE-488, but the hardware, documentation and support that delivers the shortest path to a measurement system.

Figure 6.1-1 Hewlett-Packard DSAs representative of the handheld, benchtop and systems types.

6.1 Types of DSAs

There is a wide variety of DSAs on the market; and generally, they can be broken down into three categories:

1) Handhelds
2) Benchtop Instruments
3) Computer Controlled Systems

Though there are variations on this theme this covers by far the majority of systems on the market. Obviously, there is a long list of features and trade-offs to consider; including speed, portability, number of channels, display resolution and price. Figure 6.1-1 is a photograph of three Hewlett-Packard products representative of these categories.

The product offerings change rapidly as technology advances, it is hard to make generalizations, but the following are some key considerations for each DSA type.

Handhelds are light-weight, portable and battery operated. They generally have a LCD display that limits their resolution and display update rate. The power consumption considerations lead to design compromises, that make this class of DSA the lowest performing as a group in terms of speed, dynamic range and accuracy. They tend to be lower in cost but have a relatively robust set of capabilities. In machinery analysis applications these are well suited for maintenance and troubleshooting where portability is very important.

The advantage of this type instrumentation in predictive maintenance is the operator receives immediate vibration spectral results and preliminary vibration severity analysis.

Benchtop instruments range from relatively low-cost, low-performance instruments to high-performance instruments. Generally, the benchtop instruments give exceptional performance in a small easily operated package. The tight coupling of the hardware and software within the instrument, leads to very high display updates, extremely powerful analysis capability and generally very high accuracy and dynamic range. In machinery analysis these are generally used in troubleshooting, research and development, and in certification testing where portability is less an issue and a small number of channels is necessary.

DSA systems consist of an instrumentation mainframe connected to a computer. The system is actually a DSA instrument that uses the computer as the user interface and data storage device. Most often, systems are multichannel in nature having from 2 to 500 channels of data acquired simultaneously. Systems are also capable of being customized by users or software developers to perform dedicated or high-performance tasks. They are generally used when multi-channels are required and the computer interface is desirable. In machinery analysis they are used in many of the same general areas as benchtops and in continuous machinery monitoring applications.

6.2 Measurement Speed

Speed is important in machinery analysis because vibration characteristics can change quickly. This is illustrated in the spectral map of Figure 6.2-1, where measurements of a machine run-up spaced at 0.5 s intervals show significant variation. Speed is also important for reducing the time required to characterize a machine. The time required to make a measurement with a DSA is determined by two factors: (1) measurement resolution, and (2) transform computation time. High resolution measurements require a long data sampling time (frequency resolution spaced at 1 Hz intervals requires a 1 s measurement time). This is a physical fact, independent of the design of the DSA. Computation time, however, varies widely among DSAs, and can make a noticeable difference in measurement time. Computation time is usually expressed in terms of real-time bandwidth — the frequency span at which data sampling and computation times are equal (higher real-time bandwidth implies faster computation). This is theoretically the maximum bandwidth that data can be collected without gaps while simultaneously computing spectra.

Real-time bandwidth example

Suppose you were making measurements with a 2000 Hz frequency span. The data sampling time for this span on DSAs with 400-lines resolution (see Section 6.2) is 0.200 s. If the computation time were also 0.200 s, then sampling would never have to stop to let the computation catch up. (This computation time would correspond to a real-time bandwidth of 2000 Hz). If the computation time were 1 s (a real-time bandwidth of 400 Hz), the analyzer would miss large amounts of data while waiting for the computation.

Actual real-time bandwidth and specified bandwidth can vary considerably. Its important to note that a number of factors influence the actual real-time bandwidth. Often the processor is required to perform a number of additional calculations to display the data; this can have a considerable effect on the real-time bandwidth. Although an increase in the data block size will increase data sampling time the calculation time increases at a faster rate for larger blocks; thereby, reducing the real-time bandwidth for larger block sizes and increasing it for smaller. Obviously, the number of simultaneous channels will also effect the rate. For these reasons, the real-time bandwidth is normally specified for a block size of 1024 time sample points and with the display update turned off (i.e. fast averaging). Therefore, if real-time performance (i.e. gap free) is necessary, caution should be exercised in interpreting the specification.

One way of circumventing the problem is to buffer up the raw-time- domain digital data in either the analyzer memory (RAM) or on a high-speed data-storage device (normally a hard disk drive). These capabilities are referred to as time capture and throughput, respectively. Figure 6.2-3 illustrates the concept of time capture. In this manner the analyzer can acquire gap free data, store it in memory, and analyze it later without relying on the processing speed of the analyzer being able to keep up with the data acquisition rate. An additional benefit, is the ability to go back and reanalyze the data in a number of different ways after the fact, without reacquiring the data.

Figure 6.2-1 Machinery vibration spectra can change very quickly, as this spectral map of a run-up test illustrates. Slower swept spectrum analyzers can miss these changes.

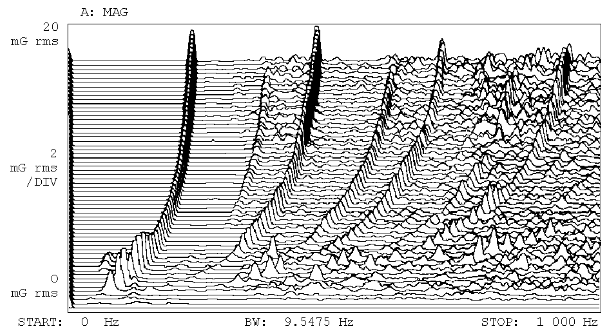


Figure 6.2-2 Total DSA measurement time is the sum of data sampling time and computation time. While sampling time is fixed for a given resolution, computation times vary widely among available DSAs.

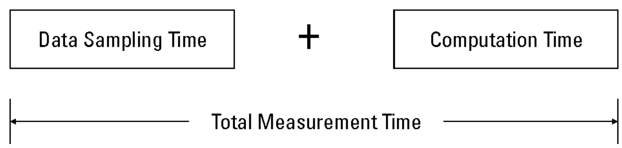
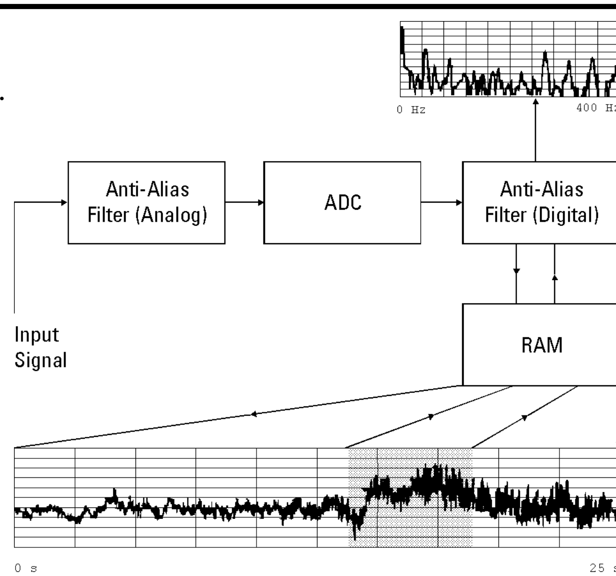


Figure 6.2-3 Block diagram of data flow in a DSA using time capture. The data is first acquired and stored in RAM; then analyzed to product spectra, etc.



This mode is similar to tape recording data and then post-processing it. The advantage of throughput or time capture is the integration of the process into the DSA eliminating the extra calibration and setup steps of a separate recorder. Post-processing is greatly simplified, saving time and allowing previewing of the analyzed data immediately.

One of the shortcomings of this is that the real-time spectral display update is not available, since the data isn't analyzed until after the test. DSA systems generally have the capability to monitor input data and acquire the throughput/ time capture simultaneously. Also, since, all of the data is held in memory it requires a considerable portion of the analyzer memory resources; especially, if long records are required. Many times, what is required is not real-time data acquisition — the real issue is display update rate. It is possible to keep track of fast changing spectrums and not have real-time performance; the gaps do not materially effect steady or pseudo-steady state conditions. Generally, real-time performance is required when transients are present and non-real-time analysis would risk missing an important event.

6.3 Frequency Resolution

High resolution is required for analysis when vibration signals are closely spaced, or when the frequency of a component must be read with high precision. A common example of closely spaced signals are the 1 x and powerline components of induction motor vibration, which can be separated by a few Hz. The sidebands around rolling- element bearing and gear frequencies are often closely spaced. High precision is required when the characteristic vibration frequencies of two possible sources are close together, as in the case of a bearing frequency and a running speed harmonic.

Frequency resolution in a DSA is determined primarily by the number of filters (or lines of resolution), and the ability to zoom. The filters of a DSA are shown in Figure 6.3-1. Signals must lie in different filters to be resolved, so resolution depends on the spacing of the filters. If the number of filters is fixed, filter spacing is determined by the number of filters and the analysis span. More filters imply better resolution for a given span.

Figure 6.3-1 Frequency resolution in a DSA is determined primarily by the number of filters, and the ability to zoom. In a zoom measurement, the component of interest is made the center frequency of the analysis, allowing the use of an arbitrarily narrow frequency span.

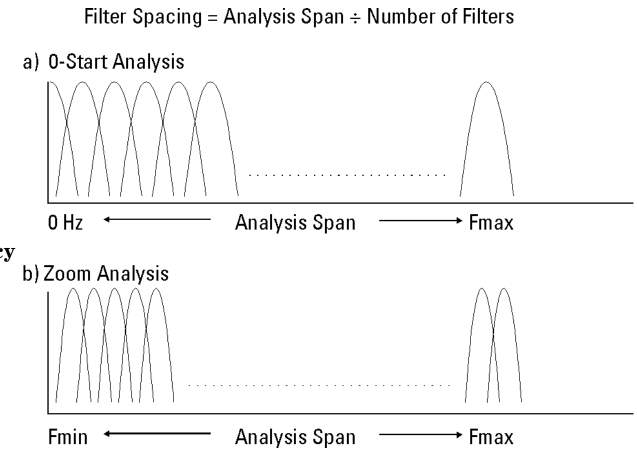
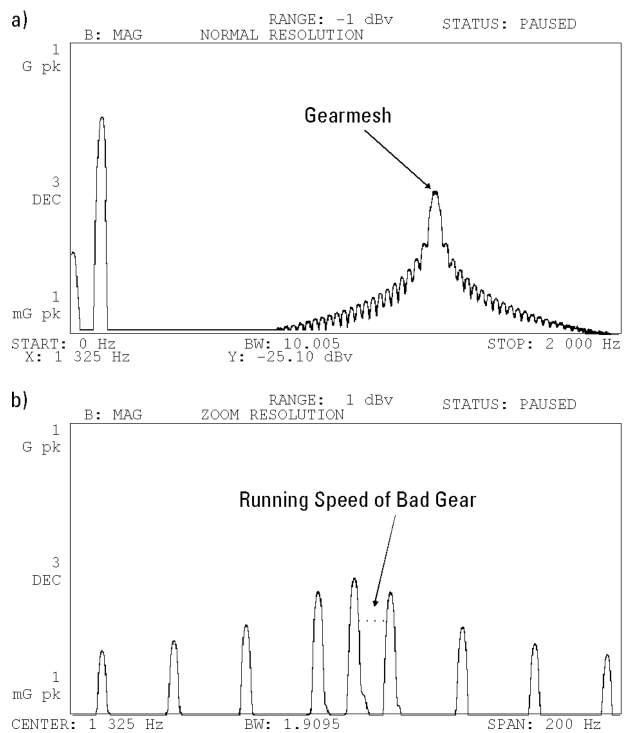


Figure 6.3-2 An example gear vibration spectrum that illustrates the need for zoom. The sidebands around gearmesh often indicate the bad gear, but are too closely spaced to to resolve in (a). The zoom measurement in (b) centers a narrower frequency span on the gearmesh, increasing resolution.



If the span required for the desired resolution is too narrow to include all the frequencies of interest, then the analysis must start at a frequency above zero. This process is referred to as zooming (because it involves zooming in on an arbitrary center frequency), and is a feature of most DSAs. Ideally, the zoom feature should allow frequency spans down to 1 Hz to be centered on any frequency in the analysis range. Typically, implementation of zoom should have no effect on the real-time bandwidth of the analyzer, since the process is normally handled by dedicated hardware that operates independently of the other computational hardware.

The gear spectrum in Figure 6.3-2 illustrates why the ability to zoom is so important. In the low-resolution spectrum of (a), the sidebands around the gearmesh frequency indicate a problem, but the exact spacing (which will indicate which gear has the defect) is difficult to determine. Since the gearmesh is at a relatively high frequency, a span narrow enough to resolve the sidebands cannot cover the entire frequency range starting at 0 Hz. Thus we must zoom on the gearmesh frequency to complete the analysis. (See Section 4.6 for more information on gear analysis.)

Figure 6.3-3
A comparison of 3 common window types: a) flat top, b) Hanning and c) uniform.

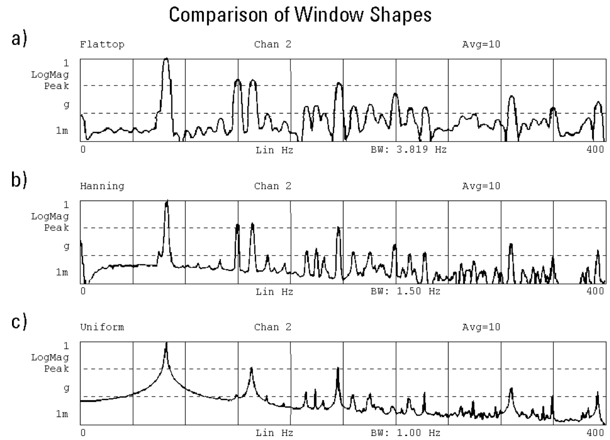
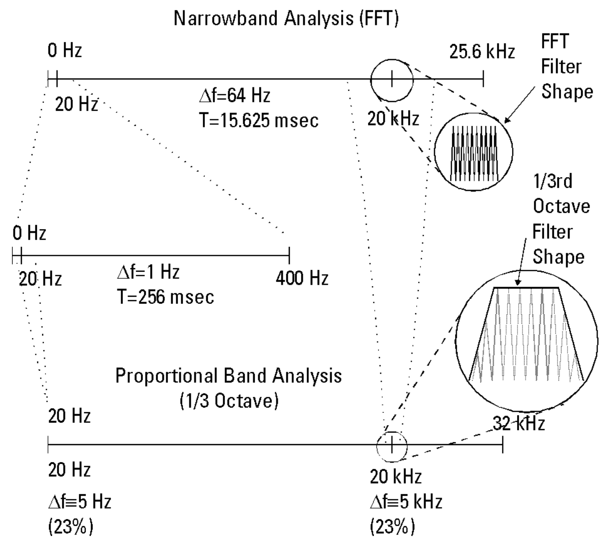


Figure 6.3-4
Synthesising 1/3 octave data from FFT requires multilple FFT bands of analyses. The lower fre-quencies of the 1/3 octave have resolution requirements of Δf 5Hz while the high frequencies require Δf 5kHz.



Window Functions

Frequency resolution is also affected by the shape of the filters — determined in a DSA by the window function selected. The window function shapes the input data to compensate for discontinuities in the sampling process (see application note AN 243). Figure 6.3-3 shows the same vibration spectrum measured with the three windows commonly available on DSAs.

A. The Flat Top window is optimized for level accuracy, with a response variation with frequency of 0.1%. This is the window to use unless maximum frequency resolution is required, or you are capturing a transient.

B. The Hanning window provides improved frequency resolution (note the Bandwidth notation at the bottom of the display), but sacrifices amplitude accuracy. Variation with frequency is up to 15%.

C. The Uniform window provides no weighting, and should be used only for totally observed transients, or specialized signals. The wide skirts, known as leakage, severely restrict frequency resolution. (Leakage is what weighting in the other two window functions minimizes.) Amplitude variation is up to 36%.

Octave Band Analysis

During most of this note we have concentrated on FFT analysis because it is most useful in machinery vibration. This is because, many of the vibration components are very narrow in bandwidth and repeat themselves in a uniformly spaced relationship with respect to frequency (i.e. harmonics and sidebands). Sometimes it is desirable to have the frequency resolution spaced in a logarithmic or octave fashion, with the frequency resolution depicted proportionally rather than uniformly as in FFT analysis. This type of analysis is useful when high resolution is not required, and for reasons beyond the scope of this note, is useful in detecting and analyzing transient behavior. This type analysis is referred to as octave-band analysis.

One method of obtaining this type of analysis is to resynthesize the octave analysis data from high resolution FFT data. Figure 6.3-4 illustrates the concept of synthesis of 1/3-octave data from multiple passes of FFT data (1/3-octave refers to a doubling of the frequency for every third data point). Figure 6.3-5 is a comparison of comparable 1/3- and 1/12-octave analysis and narrow-band FFT analysis of the same data. Since the analysis requires multiple "passes" and differing data acquisition times, the process of synthesis is NOT real-time. Therefore, it is only useful for steady state analysis.

Real-time octave analysis is a process of obtaining octave data by programming the analyzer to perform digital filtering on the data rather than FFTs. The digital filtering process is inherently proportional and logarithmic in nature and readily yields the octave spectra. Though this is beyond the scope of this note, this capability is often required for noise and acoustics problems associated with machines. It is specified in a number of international standards and the capability is available in a number of DSAs.

Figure 6.3-5
A comparision of measurements of the same spectrum made with narrowband (FFT), 1/3 octave and 1/12 octave.

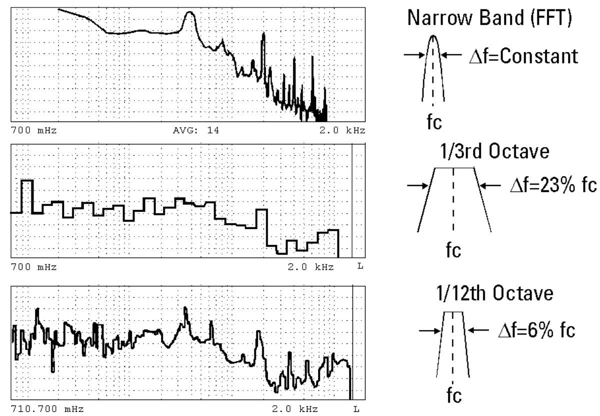
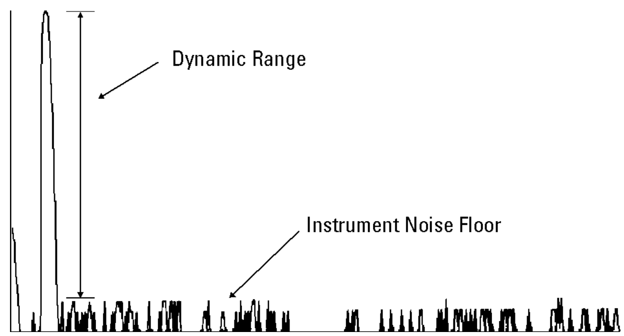


Figure 6.4-1
Dynamic range is defined as the ratio between the largest and smallest signals that can be analyzed at the same time.



6.4 Dynamic Range

Dynamic range is another aspect of resolution. It is a measure of the ability to analyze small signals in the presence of large ones, as shown in Figure 6.4-1. DSAs feature wide dynamic range, with most able to display signals that differ in amplitude by factors of 1000 or more. Logarithmic display scales are used to take advantage of this measurement capability.

Wide dynamic range is important for analyzing low-level vibration signals in the presence of large residual imbalance components.

Dynamic range is also important when the component to be analyzed is small compared to the total power level. That is, a large number of relatively low-level signals result in a high total power level that limits input sensitivity in the same way a single large signal would. This is often the case, for example, when analyzing low frequency vibration with an accelerometer.

Dynamic range in an analyzer is a cumulative specification of a DSAs ability to distinguish small signals in the presence of larger signals. It is effected by a number of components in the data acquisition portion of the analyzer: Analog-to-digital converter resolution (number of bits, linearity etc.), input amplifier noise floor, anti-aliasing filter performance, spurious signals within the analyzer, digital signal processor performance, etc.. Generally, the specification is for the worst case situations (i.e. max requency span and lowest input range) and typical performance in the frequency range for most machinery measurements and reasonable input ranges is significantly higher (20 dB is common). It is important to understand the difference between specified dynamic range (i.e. guaranteed) and typical (i.e. expected under common conditions).

6.5 Digital Averaging

Machinery vibration spectra often contain large levels of background noise, vibration from adjacent machines, or components that vary in amplitude. Three types of digital averaging are available to reduce the problems that these conditions imply for analysis.

A. RMS. The result of a RMS average of successive spectra is an improved estimate of the mean level of vibration components. RMS averaging should be used when component levels vary significantly.

B. TIME. While RMS averaging reduces the variance of signal levels, it does nothing to reduce unwanted background noise. This background noise may mask low-level components, or add unrelated components to the spectrum. Time (or synchronous) averaging effectively reduces components that are not related to a once per revolution trigger, which is usually a key-phasor. Time averaging can be used when background noise or vibration from adjacent machines interferes with analysis. Time averaging requires very good speed regulation to be effective. Time averaging in the computed order tracking mode will eliminate the speed regulation requirement.

C. PEAK. It is often desirable to hold peak vibration levels during a run-up or coast-down, or over a period of time. The result of peak averaging is a display of the maximum level at each frequency point.

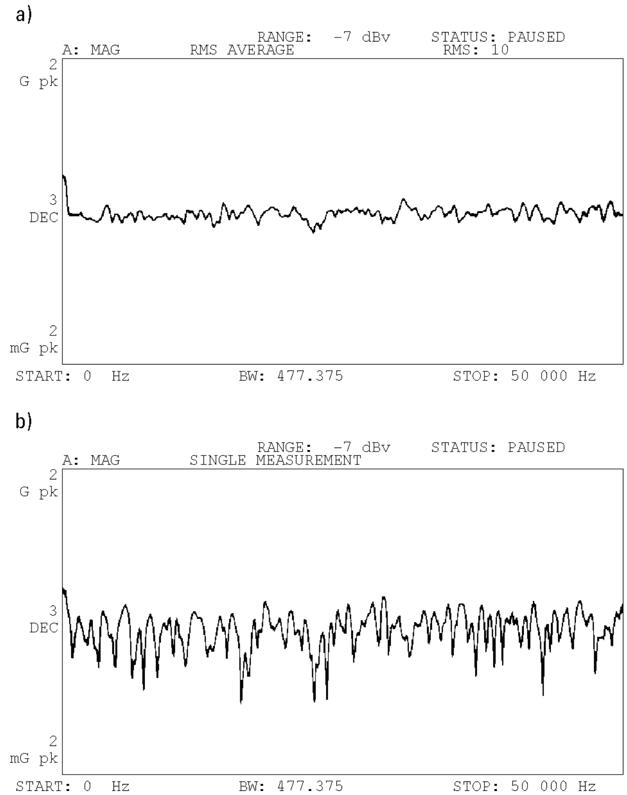


Figure 6.5-1 When RMS averaging is performed, components which vary in amplitude converge to their mean value, providing a better statistical estimate of amplitude. a) random noise average over 10 records. b) un-averaged random noise.

RMS Averaging

Because noise can cause spectral components to vary widely in amplitude, a single measurement is not statistically accurate. While watching the components vary in amplitude, you could visually average them and determine the mean level. This is essentially what RMS averaging does, and the more averages you take the better the accuracy will be. RMS averaging can be thought of as amplitude averaging, since phase is ignored. (RMS, or root mean square, is the square root of the mean of the squared spectra.) The effect of RMS averaging is shown in Figure 6.5-1.

RMS averaging improves the statistical accuracy of a noisy spectrum, and does not require a trigger, but it does not actually reduce the noise level.

Time Averaging

Time averaging is a technique that can be used to reduce the level of noise, and thus uncover low-level signals that may have been obscured by the noise. Sometimes referred to as linear averaging, this type of averaging requires a synchronizing trigger — usually a keyphasor.

Time averaging can be implemented in either the time or frequency domains, but the time domain is traditional (thus the name). In this form, the blocks of time data that are transformed by the analyzer to the frequency domain are averaged before the transformation. Signals that are fixed in the time record (i.e. synchronous with the trigger) will remain, while nonsynchronous signals eventually average to zero. This is shown in Figures 6.5-2(a) and (b), where time averaging a noisy square wave has reduced the noise level, while keeping the square wave intact. An example with machinery spectrum can be found in Section 5.2.

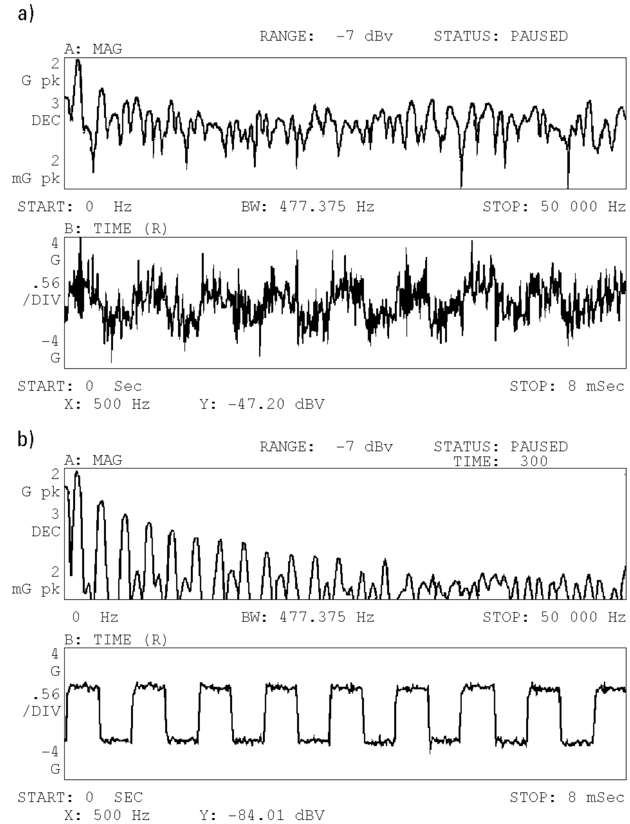


Figure 6.5-2
The time averaged displays in (b) show a reduction in the level of components that are nonsynchronous with the trigger.

Peak Hold

Peak hold is a function usually grouped with averaging in DSAs. By displaying the maximum level at each frequency over a number of samples, this feature provides a history of peak levels. Two applications are shown in Figure 6.5-3. In (a), peak hold has been used during a machine coast down, providing a simple track of the maximum level (which is usually 1 x rpm)*. The display in (b) is a peak hold over a relatively long period that shows the range of speed variation of a nominally constant speed motor. This could be used, for example, as an indication of load variation. Peak hold is also useful for recording momentary vibration peaks (e.g. from start-ups or load changes).

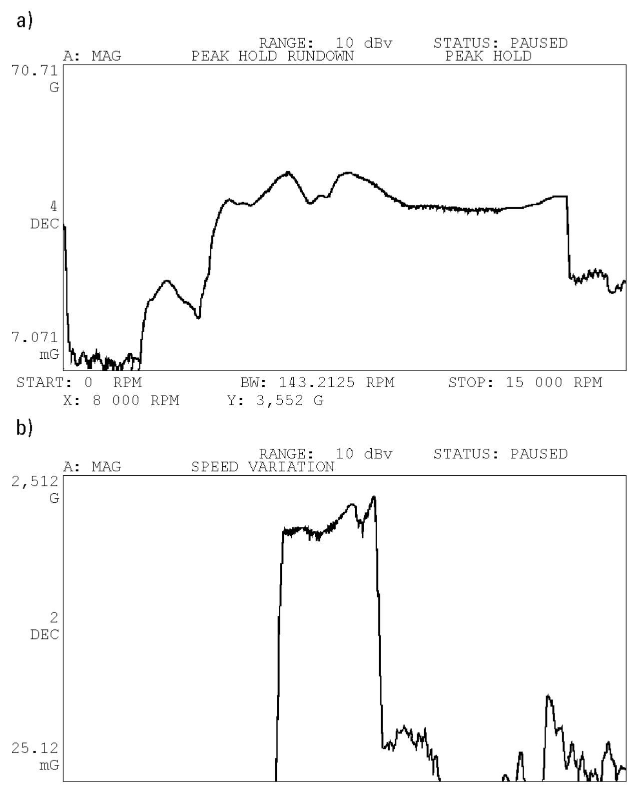


Figure 6.5-3
Peak hold used (a) to track peak level during a coast-down, and (b) to indicate variation in speed over time.

* In applications where a tachometer signal is available, an order track measurement is preferable to this method (see sec. 6.8).

6.6 HP-IB and HP Instrument BASIC

The Hewlett-Packard Interface Bus (HP-IB) is a standardized interface that can be used to connect a number of instruments, plotters, printers and computers together. Most DSAs come standard with a HP-IB interface and a set of commands that allows virtually unlimited possibilities for automatic data storage, presentation, and analysis. Most commonly the interface is used to provide for hardcopy output of DSA results using a digital printer or plotter that can be connected and controlled directly.

Computer Data Storage and Analysis

A common problem encountered in machinery vibration monitoring is the need to organize, store, plot and archive large amounts of data. Often extensive post-test data processing is required depending on the specific application. It is often convenient to put the data in a data base type applications program so that trends can be analyzed and specific data easily retrieved.

Though most DSAs have extensive data storage capabilities, it is often desirable to transfer the data to a computer, most commonly, using HP-IB. It is also desirable to place data collected on different DSAs on a common platform for comparison and analysis. Hewlett-Packard has standardized on a common data storage structure for its DSA analyzers allowing for easy transfer of data between instrument types and application programs using a set of utility programs and a Standard Data Format (SDF). Figure 6.6-1 illustrates the concept of the standardized format and the use of the utilities. The utilities consist of a set of MS-DOS® programs for viewing, converting, and transferring data.

Figure 6.6-1
Use of a Standard Data format (SDF) allows data to be easily interfaced and simplifies the conversion of non-SDF DSAs data to a standard format.

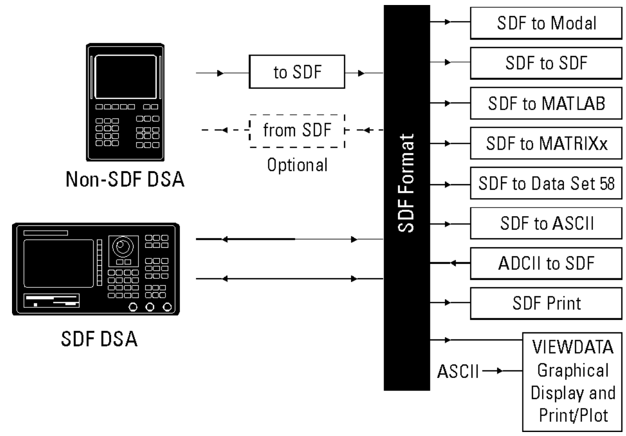
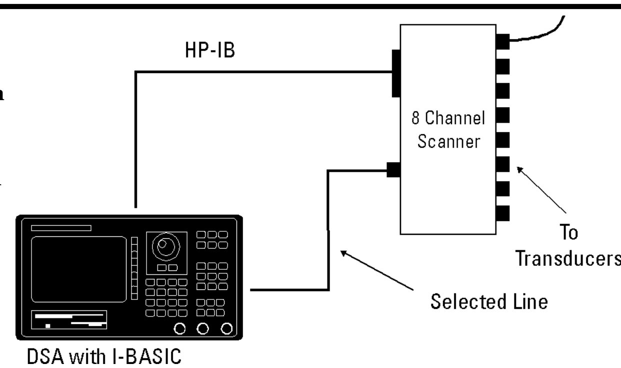


Figure 6.6-2
HP-IB system for scanning a number of transducers. Such a system can be configured to take appropriate action (e.g. sound an alarm or remove power) when current vibration level exceeds pre-defined limits.



Instrument Systems and I-Basic

Hewlett-Packard has implemented a proprietary version of the BASIC programming language that allows for many of the capabilities of the computer/DSA system without needing an external computer. HP Instrument Basic (I-Basic) is a version of the HP Basic language which is designed to run inside many HP instruments. I-Basic is also available to run in DOS and MS Windows®.

I-Basic is optimized for instrument control applications, letting the user customize measurements. It is most commonly used for automatic and repetitive tests. Not only can I-Basic address the host DSA, it can communicate over the HP-IB to other instruments or peripherals that are attached. Figure 6.6-2 illustrates the advantage of this in a particular application.

Whether through the use of I-Basic or some other programming language the HP-IB capability on test equipment allows for tying together a number of instruments over the HP-IB and controlling them in concert with each other to make automated and complex measurements using an Instrument System concept. The HP DSA Systems are a specific example of this approach, but the same concept can be applied to traditional DSAs.

* MS-DOS and MS Windows are U.S. registered trademarks of Microsoft Corporation.

6.7 User Units and Waveform Math

Vibration displays are easier to interpret if they are presented in units that are relevant to machinery. DSAs provide the capability for user calibration of amplitude units, and a selection of units for the frequency axis. DSAs can also convert spectra from one vibration parameter to another through integration and differentiation.

User units calibration is accomplished by entering a calibration factor (such as, 10 mV/g) or using the marker function to specify a known value (such as, marker value equals 94 dB SPL). The DSA performs the conversion and displays the vibration spectrum in the desired units, usually referred to as "EU" (Engineering Units). DSAs also provide for custom labeling of user defined units (e.g. g's, in/s, mils, etc.).

The frequency units used for machinery vibration analysis include Hertz, rpm, and orders. Orders refer to "orders of rotation", and are harmonics of the rotation speed. Orders are handy for analysis because many vibration problems are order-related. By using external-sample control, orders can be fixed on the display while speed changes (see Section 6.8).

Often it is desirable to convert from one vibration motion variable to another. Referring to the formulas for displacement, velocity, and acceleration in Section 2.1, it should be apparent that they are related by frequency and a phase shift. For example acceleration can be converted to velocity through division by $j\omega = j(2\pi f)$. This operation is commonly referred to as artificial integration (the "j" term is an operator that implies a 90° phase shift), and is a feature of most DSAs. Figure 6.7-1 shows an integrated acceleration spectrum overlaid on an actual velocity spectrum measured at the same point.

Figure 6.7-1
A comparison between an integrated acceleration spectrum and an actual velocity spectrum (dashed line).

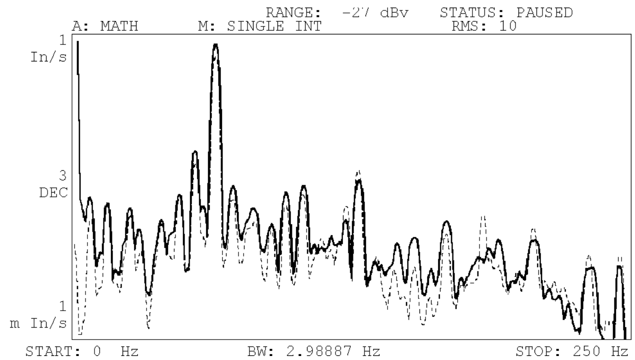


Table 6.7
Vibration Parameter Conversions

Conversion	Operator	Description
Acceleration ⇑ velocity	/$j\omega$	Single integration
Acceleration ⇑ displacement	$1/\omega^2$	Double integration
Velocity ⇑ displacement	$1/j\omega$	Single integration
Velocity ⇑ acceleration	$j\omega$	Differentiation
Displacement ⇑ velocity	$j\omega$	Differentiation
Displacement ⇑ acceleration	$-\omega^2$	Double differentiation

Table 6.7 summarizes vibration parameter conversion. Two things to note about these conversions: (1) integrating absolute velocity will not result in relative displacement (i.e. integrated measurements from a case-mounted velocity transducer will not give the same result as a displacement transducer that measures the shaft directly),and (2) differentiation is usually not recommended, since noise in the spectrum to be differentiated tends to give misleading results.

A feature of many DSAs which actually implement this capability is waveform math. It fundamentally allows you to define mathematical relationships between data traces within the DSA and calculate supplemental data from the results of measurements; much the same way as a calculator can be used to calculate the results of static measurements. Some DSAs implement this conversion explicitly. Order analysis conversion is more complex but possible.

Historically, these conversions were made with analog hardware circuitry built into the DSA or signal conditioning. This is being largely replaced by the math calculation which gives good results and does not require the costly additional circuitry. Though these operations are time proven and straightforward to implement; whenever performing this type operation it is recommended that the operation and units be checked carefully and tested to insure that there has not been some unexpected error introduced.

6.8 Synchronous Sample Control and Order Tracking

One of the complications encountered in analyzing rotating machinery is variation in speed. For machines that will operate over a wide range of speeds, it is desirable to measure vibration over the entire range. With a fixed-frequency axis, spectral components are constantly moving with the changes in speed. For machines that run at a nominally constant speed, even small changes can make point-for-point comparisons difficult.

The problem of wide-speed variation is sometimes addressed by post-test manipulation of the data. The frequency display can be normalized (i.e. calibrated to fixed location, orders of rotation) through software manipulation. But often what is required is the order track (i.e. the locus of points characterized by the amplitude as a function of rotation speed for a particular order, see figure 6.8-1). There are a number of problems with this approach: (1) it does not provide real-time display update of the data; (2) the number of orders measured changes with measurement speed and the resolution appears different at different running speeds; and (3) the scheme assumes that the software can calculate or somehow determined the running speed.

To circumvent these problems, external sample control was introduced. By controlling the data-sampling-rate with a signal tied to rotating speed, the display will have a fixed calibration in orders of rotation. (See Section 5.4). This is a result of the analyzer, in effect, sampling at a constant delta angle of rotation.

Figure 6.8-1
The order track is the locus of points of a particular order as a function of machine speed. Illustrated is the plot of this data super-imposed on a waterfall plot.

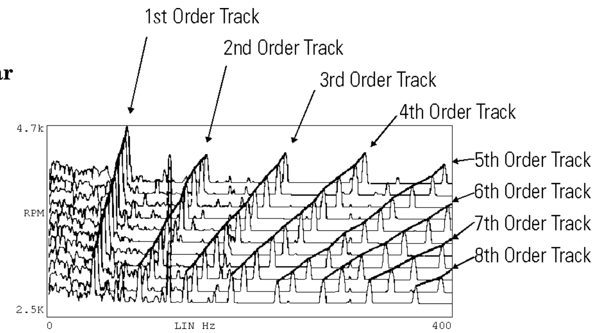


Figure 6.8-2
Instrumentation setup for controlling sample rate externally.

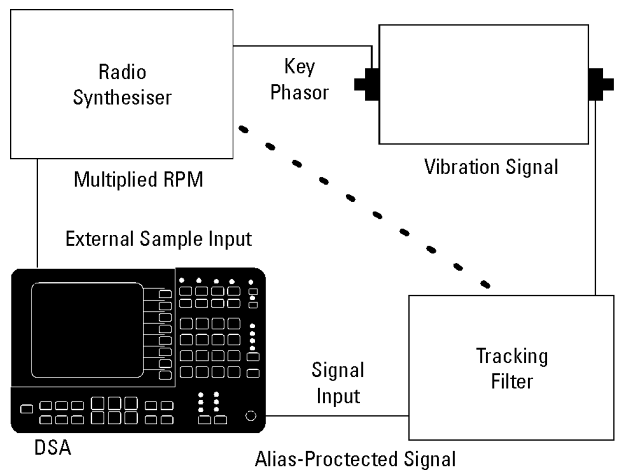


Figure 6.8-2 shows the instrumentation required for controlling the sample rate externally. Typically, a once per revolution pulse multiplied by a ratio synthesizer is used for sample control. The ratio synthesizer is required because DSAs typically sample at a rate of 2.56 times the frequency span. Since it is usually desirable to look at several orders, the once per revolution tachometer pulse must be multiplied by 2.56 times the number of orders to be analyzed. (If you needed to analyze a machine out to a frequency of 100 orders with a once per revolution tachometer signal the ratio synthesizer would be required to produce 2.56 * 100 = 256 sample pulses per revolution. If the block size was 1024 points, there would be exactly 4 revolutions in one data record). An important requirement for the ratio synthesizer is anti-aliasing protection. Aliasing occurs when the data-sample rate is too slow, allowing high-frequency signals to be misrepresented as low-frequency signals. Aliasing is avoided if a filter is used to limit input signals to frequencies less than 1/2 the sample rate (See Hewlett-Packard Application Note AN 243 for more information on aliasing.) Since the sampling rate is varying its necessary to have a variable (or tracking) filter.

Two problems arise out of this scheme because of the additional hardware used. First, the ratio synthesizer is a phase lock loop and it has some inherent time lag and phase error problems. In order tracking very small errors in frequency can cause significant amplitude and phase errors. Second, the tracking filter can be one of a number of types, the most popular is switched capacitance due to its low cost and ease of implementation. Though it is low cost, it typically has a limited dynamic range because of the switching "spurs" (noise spikes that appear as signals) making it unsuitable. (See appendix A).

To avoid these problems HP developed DIGITAL SYNCHRONOUS SAMPLE CONTROL schemes, they use the digital signal processing power available inside the DSA to perform the ratio synthesizer functions and to digitally resample the data to produce conceptually the same effect as the external sample control without the addition of analog hardware. Additionally, the flexibility and power of digital processing allowed for the elimination of many problems inherent in the older technology. Figure 6.8-3 shows a block diagram of this digital implementation of synchronous- sample control. Appendix A of this note contains a more detailed discussion of the implementation of this scheme. When the data is synchronously sampled, producing the order track data is very easy, because it's simply the locus of points at a fixed frequency (in the order domain) as a function of rotation speed.

Figure 6.8-3 Instrumentation for synchronous sample control for DSA with digital synchronous resampling.

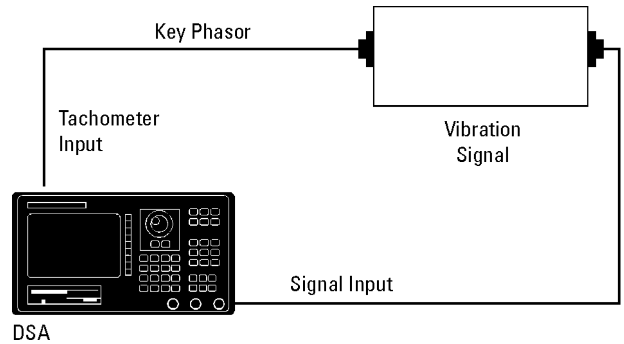


Figure 6.8-4(a) Bode plot of a machine runup for a simple flexible rotor system showing the critical speed. The plot represents magnitude as linear magnitude and the "X" axis is linear to conform to normal convention, though the DSA's can also scale the data in a Logrithmic format.

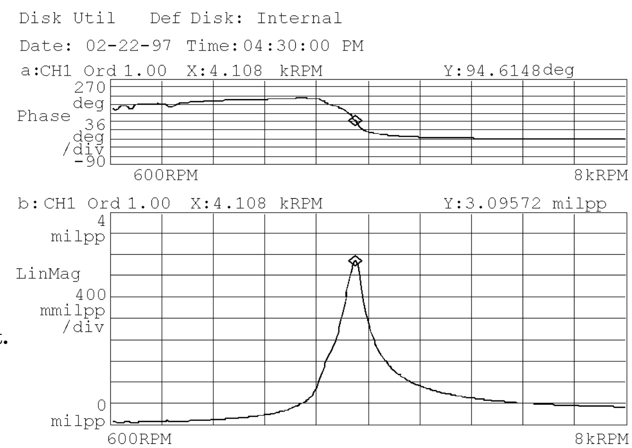
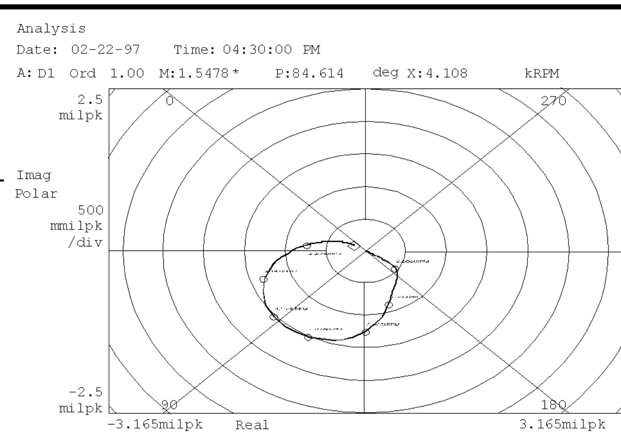


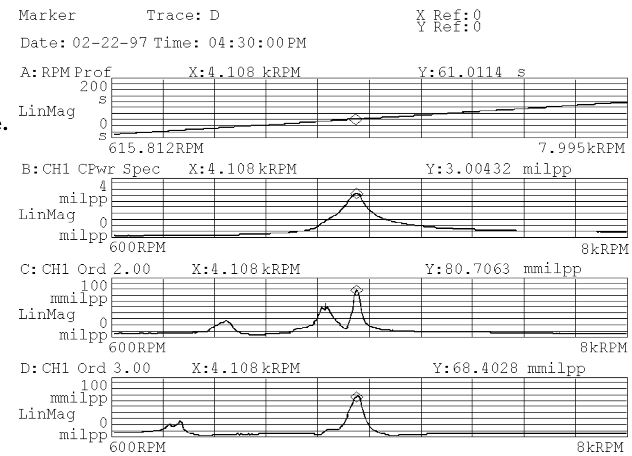
Figure 6.8-4(b) Polar plot of runup depicting the magnitude and phase plotted in a polar fashion. Note the rotation intended to compensate for positioning of the phase reference and the displacement transducer.



Machine Runup Measurements
An important measurement made using the order tacking capability of DSA's is the machine runup/down. In many machines the only time they operate at certain important speeds (ie. critical speeds, at structural resonances, etc.) is during a runup or rundown. This measurement is an important indication of machinery health and is commonly used to qualify new and overhauled high speed machinery. The measurement uses the residual imbalance in the machine to excite it at different frequencies as it runs up to operating speed and measures the response (magnitude and phase) as a function of speed. This utilizes the basic order tracking capability of the analyzer coupled with special display features required of this measurement.

Two common display formats are used with this measurement; one is the Bode diagram* and the other is the polar display. The bode plot depicts the magnitude and the phase response of the system to the runup as a function of speed (RPM). A benefit of the DSA in this measurement is its ability to simultaneously track multiple orders and display them in addition to the fundamental rotation speed (1st order); as well as the overall level and the RPM profile Figure 6.8-5.

Figure 6.8-5 Runup depiction of the 2nd and 3rd orders as well as the overall level and the RPM profile.

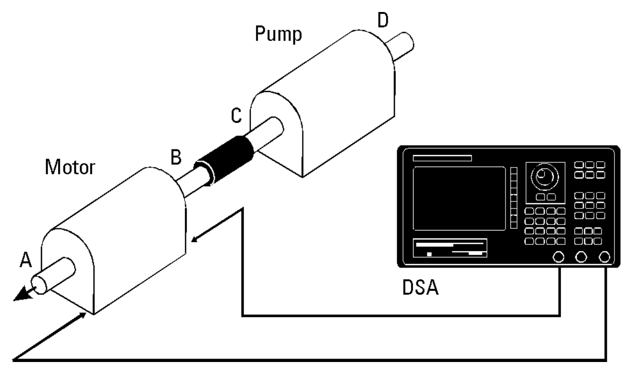


6.9 Dual/Multi-Channel Enhancements

Most of our discussion has center around spectrum measurements which can be made with a single-channel DSA, and in some cases a reference trigger to obtain phase information. This is not to imply that a single-channel DSA is the best solution to vibration analysis problems, it merely points out that many measurements CAN be made with a single-channel analyzer. In fact, advances in technology have significantly reduced the price differential between 1-, 2-, and multi-channel DSAs; making their usage common in machinery vibration analysis.

A multi-channel DSA is much more than multiple separate analysis channels, because it can measure the amplitude and phase relationships between two signals or sets of signals. Another relationship that is commonly measured is called the frequency response function. It is especially useful for performing real-time phase comparisons, and identifying the source of vibration or noise in a machine. The frequency-response function can also be used to determine natural frequencies of shafts, gears, and machine housings that can be critical for analysis. Multi-channel DSAs can display shaft orbits. These displays give insight into the path of the shaft as it rotates, and are especially useful in high-speed machinery. For a more general discussion of dual-channel DSA capabilities, refer to Hewlett-Packard application note AN 243.

Figure 6.9-1 Misalignment is indicated by a 180° phase relation between A and B. For transducer oriented as shown (180° relation), the relative phase will be 0°.



Real-Time Comparisons
Comparative phase measurements are a powerful tool for analysis, especially for differentiating between similar forms of vibration (see Section 4.4 and 5.2). This measurement is made both easier and more accurate with a dual-channel DSA. Referring to the motor-pump combination in Figure 6.9-1, suppose that you are not sure whether the high-vibration level is due to imbalance or misalignment. As pointed out in Section 4.4, the relative phase of axial vibration at A and B will be 180° if misalignment is the problem (assuming a rigid-rotor). With a single-channel analyzer, you would use a keyphasor as a reference, and measure the two ends one at a time. With a dual-channel analyzer, all you have to do is connect an end to each channel and measure the transfer function phase (this connection is diagrammed in Figure 6.9-1.) Thus, relative phase measurements can be made with a single-channel DSA, but are much easier (and less error-prone) with a dual- or multi-channel DSA.

* Bode diagram conventions for rotating machine applications differ from electrical and servo conventions; here the convention common to machinery vibrations are used.

Another important place where a dual- or multi-channel DSA is useful is in two- or multi-plane balancing. The vibration level at multiple planes can be monitored simultaneously, while at the same time utilizing the external trigger to provide for phase information for all channels. This can effect large reductions in the number of measurement runs required to balance a machine.

Cause and Effect Relationships: The Coherence Function
A common problem in machinery vibration analysis is that vibration from one machine in a train is coupled to the other machines. The coherence function can help with these problems by indicating the cause and effect relationship between vibration at two locations.

The coherence display covers a range of 0 to 1, and indicates the percentage of power in channel 2 that is coherent (i.e. linearly related) with channel 1. Let's suppose that vibration levels at points A and D on the motor pump combination of Figure 6.9-1 are similar, and rather high. You would like to know whether they are independent or related. A low value of coherence between vibration components from A and D indicates that they are not related. A high coherence value for a component implies that there may be a causal relationship. (The high coherence component could, for example, be from a third source of vibration.) Coherence measured between end points on a motor and pump is shown in Figure 6.9-2. Note that coherence is high for all the major vibration components except 240 Hz indicating that this vibration is not from the motor. The technique will not work 100% of the time, and you will have to get a feel for what constitutes a high level of coherence, but it may save in disconnecting machines to isolate the source of vibration.

Figure 6.9-2
Coherence measured between a pump and motor clearly indicates which components are unrelated.

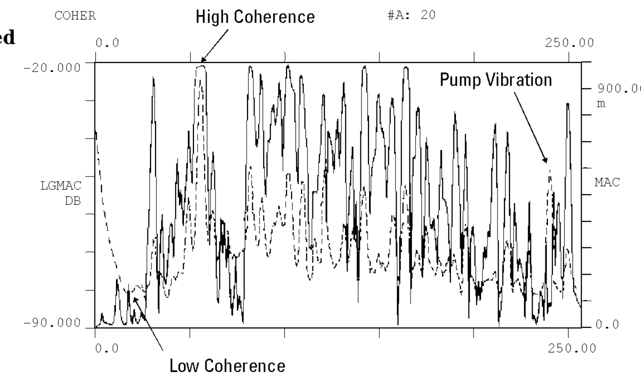
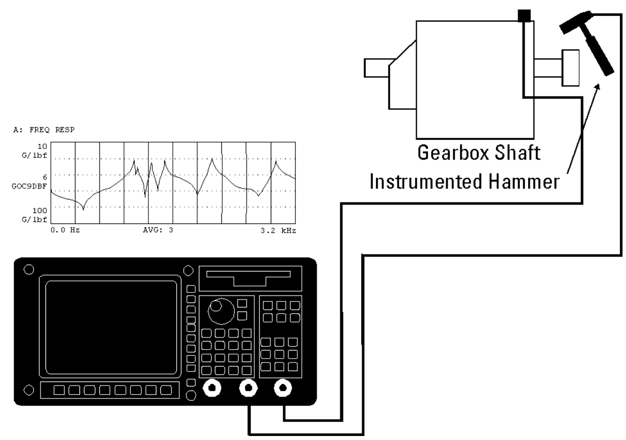


Figure 6.9-3
The transfer function of a gearbox can be measured with an instrumented hammer and a two-channel DSA.



Natural Frequency Measurements

The natural frequencies of a machine housing or foundation can be easily determined through what is sometimes referred to as a “bump” test. A single impulsive signal produces a broad spectrum of energy. If the housing is impacted with sufficient force (typically with a block of wood), all the natural frequencies will be excited. The response can be measured with a single-channel DSA, but an imperfect impact may result in a misleading spectrum.

A better way to make this measurement is with a multi-channel analyzer and an instrumented hammer. This is shown diagrammatically in Figure 6.9-3, where the natural frequencies of a gearbox are being determined. As we saw in Section 4.6, gear defects often show up at the natural frequencies, so this information is valuable. It can also help identify critical rotor frequencies in high speed machinery. Hewlett-Packard application note AN 243-3 contains more detailed information on measuring the response of mechanical structures.

Orbits

HP dual-channel DSAs have the ability to display orbit diagrams, Figure 6.9-4 shows how an orbit diagram is generated utilizing two proximity probes mounted at 90° to each other. Figure 6.9-5 shows diagramatically a typical orbit diagram. They are useful for gaining insight into rotor motion in

Figure 6.9-4 Measurement setup for orbit measurements using orthogonal proximity probe and a 2 channel DSA.

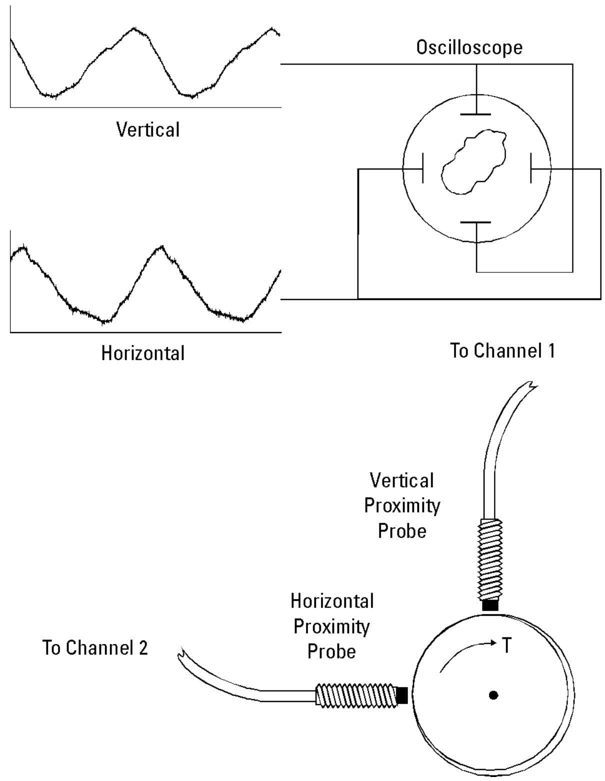
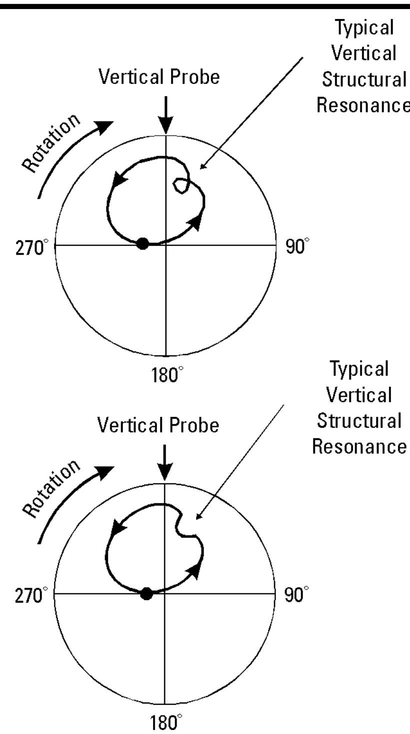


Figure 6.9-5 The orbit capability of some two-channel DSAs provides insight into rotor motion in machines with fluid-film bearings.



turbo-machinery. The subject of orbit interpretation is covered well in Reference 30. If a multi-channel system (≥4 channels) is used it is possible and often desirable to make multiple simultaneous orbit measurements at different shaft locations.

The orbit diagram is fundamentally a time domain measurement and it should be emphasized that a DSA will typically low pass filter the data before collection. In some cases unfiltered time data is desired, and many DSAs allow for bypassing the filters just for this purpose.

On the orbit diagram as you move about the orbit pattern the independent variable is time. If the measurement is triggered by a shaft reference (i.e. keyphasor) then the actual position can be marked on the diagram and if the rpm is known the location of any point can be calculated. An alternative method is to use synchronous sample control in orbit measurement (see Section 6.8) with an external reference for trigger (this can be the same signal as the tachometer signal). Now the independent variable is not time but shaft position and can be read directly from the DSA's display as shown in Figure 6.9-6.

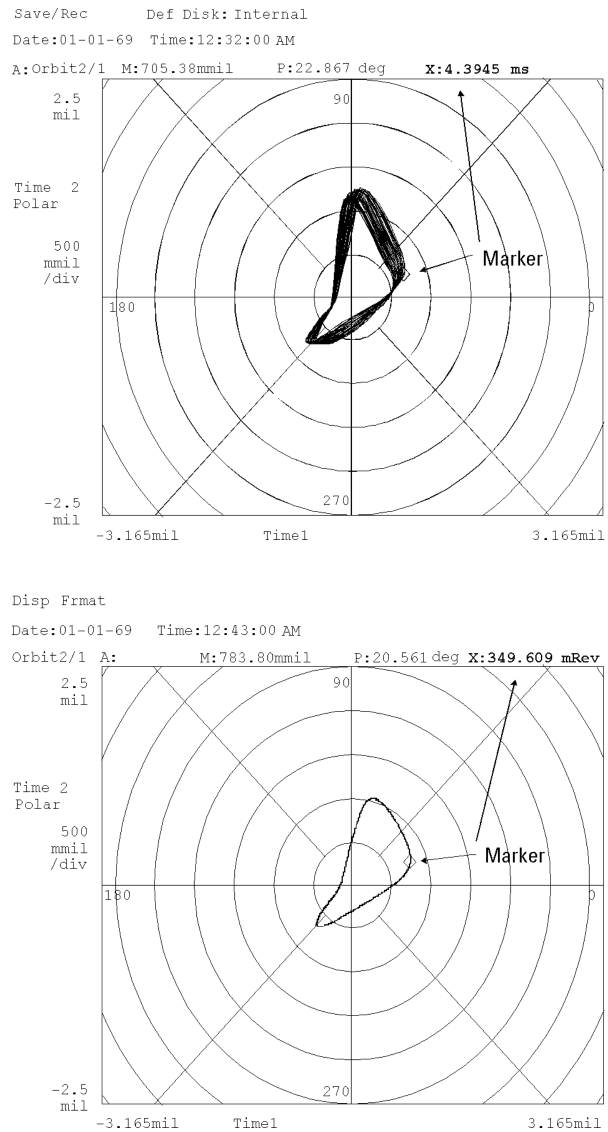


Figure 6.9-6 Orbit diagram using synchronous sampling will accurately represent the angular position (note the marker read out).

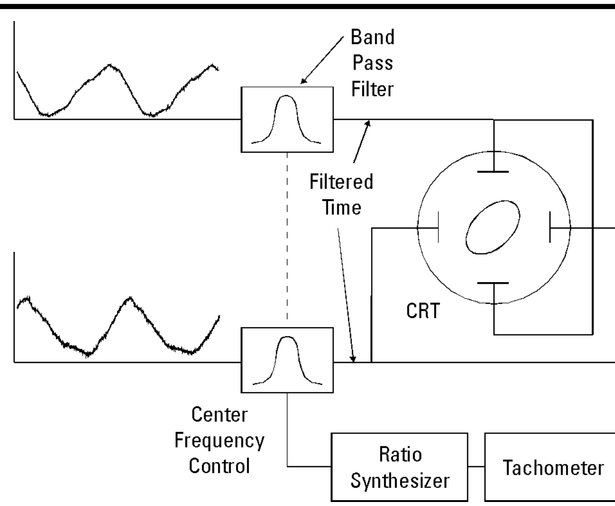


Figure 6.9-7 Traditional method for measuring filtered orbits using tracking filters and an oscilloscope.

Filtered Orbits

Normal orbit diagrams represent the contribution of all frequencies within the bandwidth of analysis. These include contributions from surface defects and anomolies, higher (or lower) order components, powerline harmonics, machine noise and the like. Filtered orbits is a technique for focusing the orbit analysis on a select frequency range (or ranges) of interest. This was traditionally done by placing a tracking narrow bandpass filter in the analysis stream (Fig 6.9-7). This allows the analysis to focus on contributions to the orbit associated with that particular frequency or order. The traditional technique requires special purpose analog hardware and has many of the limitations associated with the use of similar techniques discussed elsewhere in this note.

In an FFT analyzer the filtered orbit information is contained within the linear spectrum or order ratio spectrum measurements (Note: phase information is required, ie. power spectrum measurements are not sufficient). The individual frequency components of the spectrum represent precisely the same sine wave that would be extracted by narrow band filtering. The advantages are the presence of all frequencies simultaneously, the precision and accuracy of the digital impementation and in the convenient user interface.

Figure 6.9-8 is an example of a traditional orbit diagram and a filtered orbit (of the fundemental rotation frequency) for the same

Figure 6.9-8(a) Complex orbit diagram of a rotating shaft at low speed with significant noise and distortion present. All frequency components are represented.

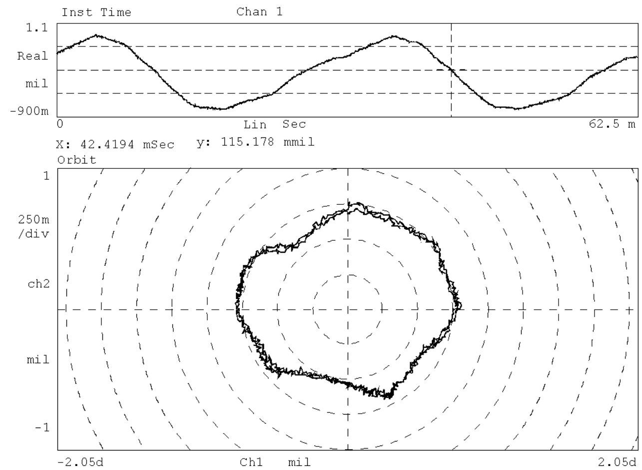


Figure 6.9-8(b) Filtered orbit of frequency component corresponding to first order. Only the single frequency component is displayed for same measurement as Figure 6.9-8(a). This data was extracted from linear spectra frequency domain data measured at a single rotation speed.

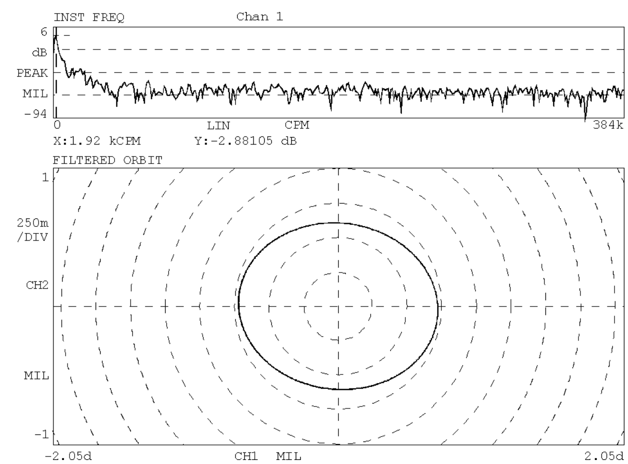
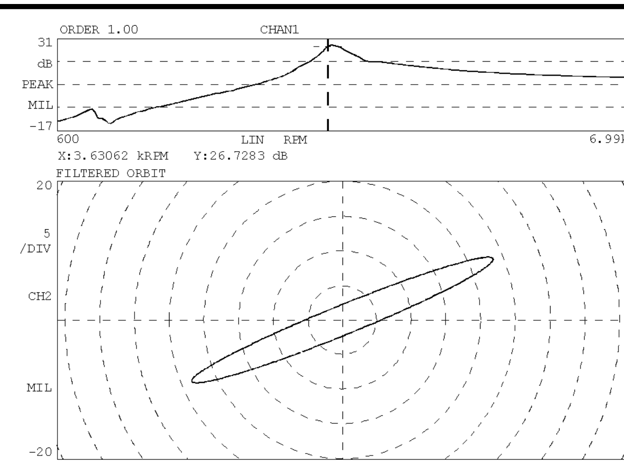


Figure 6.9-9 Filtered orbit of the first order frequency component taken at the peak response of the order track for the first order (note position of the marker in the upper trace). This data was extracted from measurement of a run-up using order tracking mode.



signal. The higher order information has been effectively “filtered out”. Both measurements were actually made simultaneously using the frequency domain measurement mode. The measurement could also be made in the order domain using either order tracks or order ratio spectra. Figure 6.9-9 is a filtered orbit taken from a run up measurement using order tracking; in this measurement an entire range of filtered orbits as a function of RPM become part of the measurement set allowing any orbit shape to be review for any time/rpm in the run-up. A capability of the digital implementation is the ability to add together related components to obtain a composite filtered orbit containing only those frequency (or order) components desire. (Figure 6.9-10).

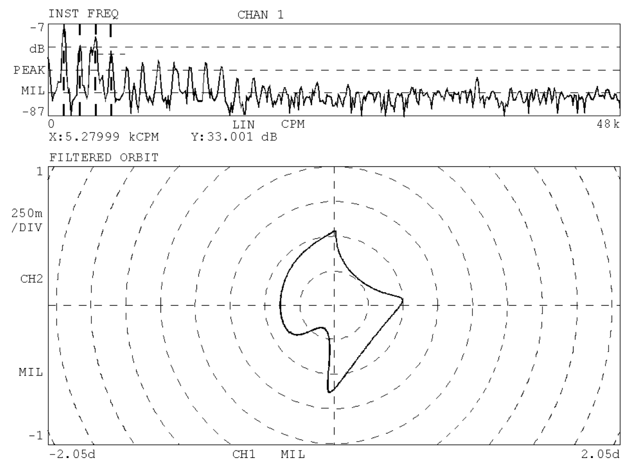


Figure 6.9-10(a) Filtered orbit consisting of the contributions of 4 individual orders to the orbit shape. The marker positions of the upper trace specify the components to include in the representation.

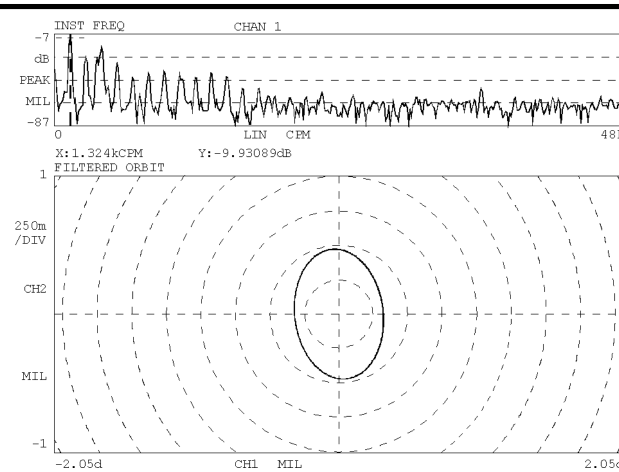


Figure 6.9-10(b) The filtered orbit diagram for same data containing only the contribution of the first order.

Appendix A

Computed Synchronous Resampling and Order Tracking

Digital Resampling

Digital signal processing (DSP) hardware and software have continually improved, allowing the substitution of digital processing in many areas which have traditionally used analog processes. This appendix discusses the implementation of DSP techniques in the processing of time sampled data to produce rotating machinery order domain information.

In DSAs the traditional method for order analysis involved varying the actual sample rate (Δt) of the data to correspond to some multiple of machine rotation speed, so that sampling is locked to a constant angle of shaft rotation. This generally led to a requirement for a significant amount of ancillary equipment, making the measurement less practical and less commonly used (Figure A-1). The use of ratio synthesizers (phase locked loops) led to a number of problems, particularly in machines with fast run-up rates or where high order numbers were being analyzed.

Figure A-1
Traditional external sampling method of order domain analysis.

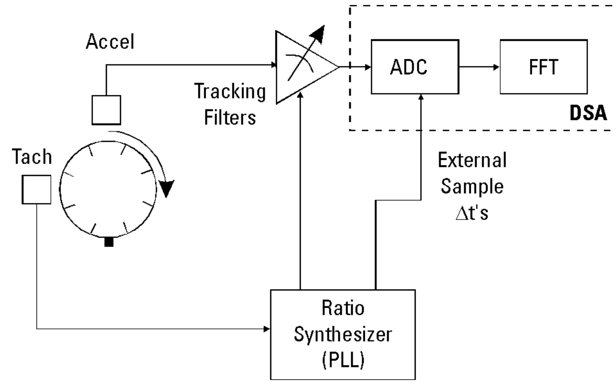
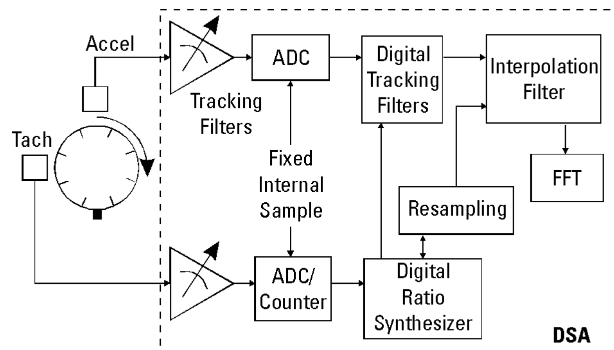


Figure A-2
Block diagram of digital resampling method of order domain analysis.



A significant improvement is possible by applying the power of the DSP and microprocessors in high-performance dynamic signal analyzers (DSAs) to replace external sampling and low-pass tracking filter hardware with a digital implementation. One immediate advantage is the use of existing general-purpose DSA hardware, since the entire process is carried out in software (Figure A-2). Many of the shortcomings of older techniques can be overcome by using digital technology.

There are three basic contributions through digital implementation:

1) Calculation of the resample times avoids many of the pitfalls of ratio synthesizers and approaches the “theoretically” perfect resampling times that would be present from a physical device, such as a shaft encoder.

2) Eliminates the need and the limitations of tracking, low-pass anti-aliasing filters by replacing their functionality with equivalent digital filters.

3) Allows the digital data to be captured in mass memory and post-test analyzed using techniques without extraneous and unwieldy recording, and AD/DA conversions required of the analog approach.

External Sampling Techniques

The ideal technique for measuring an order spectrum has long been considered the use of an encoder physically attached to a shaft to generate sampling pulses at uniform angular intervals around some reference shaft. This directly determines the sampling times as a function of shaft position. Then, if various transducers are sampled at these times, the resulting frequency spectrum will show components that depend upon multiples of the shaft-rotation rate as stationary lines, independent of shaft rpm. This sort of spectral display is plotted versus order (multiples of the shaft rotation rate), instead of frequency. Thus, if the shaft rotation rate is changed, any frequency components that are locked to this rotation rate will appear stationary in the order spectrum, while the spectra of any fixed frequency components will appear to move (Figure A-3).

Unfortunately, the appropriate shaft encoders are not always practical to install and do not, in themselves, address the problem of aliasing, so other approaches must often be considered. The classical method of bypassing the requirement for a shaft encoder is using a phase-locked loop (PLL) to generate a sampling frequency that is a suitable multiple of some shaft rotation rate by synchronizing the loop to a small number of pulses per revolution. For example, an optically reflective stripe might be attached to the shaft, giving one synchronizing pulse per revolution. Then the phase-locked loop might be set to generate exactly 256 sampling

Figure A-3 Sample plots of analysis done in the frequency (a) and the order (b) domains.

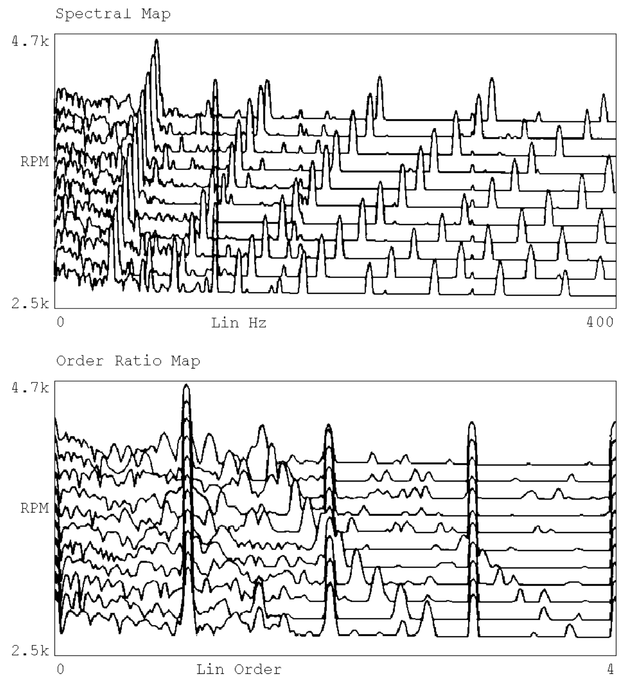


Figure A-4 Deviation of an "ideal" PLL's estimate of shaft angular position from actual shaft with constantly increasing RPM.

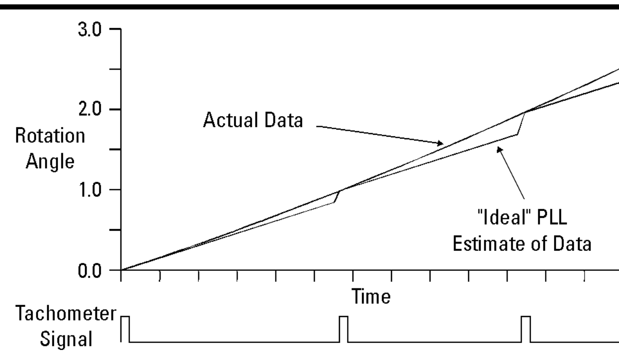
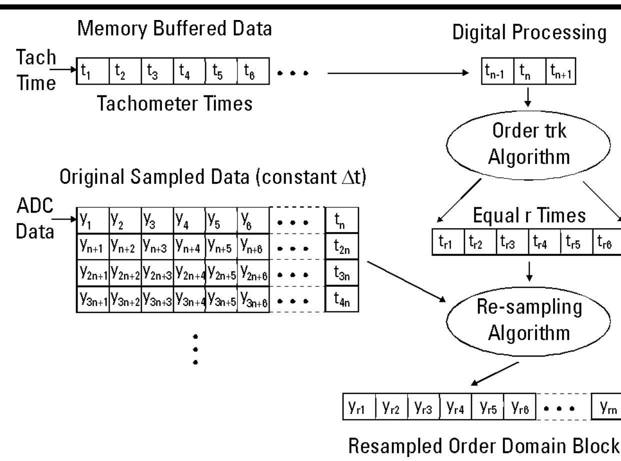


Figure A-5 Digital data processing of tachometer and data for order domain processing prior to FFT.



pulses per shaft revolution, no matter what the shaft rotation rate might be (Figure A-1). This would yield an analysis typically up to 100 orders on an FFT analyzer utilizing external sampling.

This technique works reasonably well as long as the shaft speed does not change too quickly, and the phase noise generated by the phase-locked loop is negligible. However, when the shaft is accelerating rapidly, the phase-locked loop lags behind, since the loop cannot begin to adjust to a new rpm until after some change in speed has occurred (Figure A-4). In this situation, the samples are not spaced uniformly relative to the shaft angle, and the estimate of rpm can be in error. The response time of the phased-locked loop can be reduced by increasing the bandwidth of the loop, but this also increases the noise level. At some point, the resulting phase noise will begin to "smear" the higher order spectral lines. Small errors in the sampling rate with respect to rotation rate become much more critical at higher orders, where the error is effectively magnified by the order number.

Figure A-6 Processing of tachometer signals to obtain estimates of resample times of uniform shaft angle.

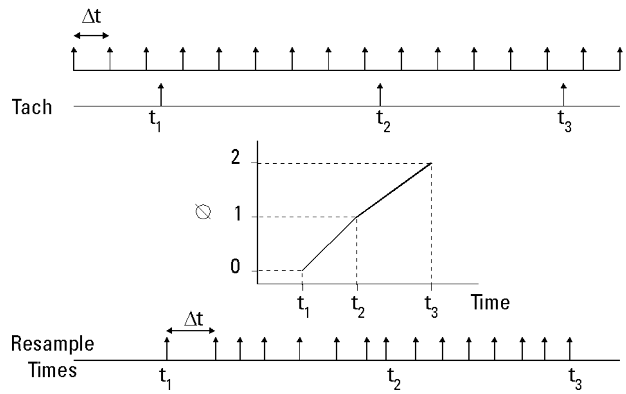
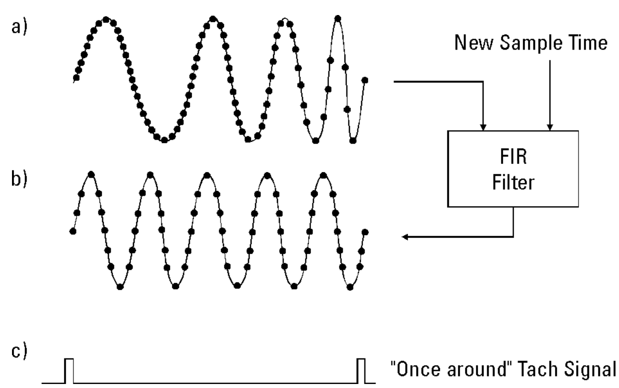


Figure A-7 Use of resampling on a fast sine-sweep at a frequency of five orders sampled at uniform time (a) and uniform angle (b).



Digital Resampling Times

Due to improvements in microprocessor performance (faster computations at lower cost), and to lower cost memory chips, it is feasible to design a tracking scheme that is independent of shaft acceleration and that has negligible internal phase noise. The idea is to collect measured data at some fixed rate, and to store this data in a large buffer memory. Simultaneously, the arrival times of each synchronizing tachometer pulse are measured and stored (Figure A-5). Then, the microprocessor can be used to determine the shaft angle and velocity at intermediate points between the tachometer pulses, based upon a model of constant shaft acceleration. The sampling times corresponding to the desired shaft angular increments can be calculated, and the stored measurement data can be interpolated in some optimum manner to obtain new samples at the desired time points (Figure A-6).

Figure A-7a shows a sinusoid chirp having a linear frequency versus time characteristic, sampled at uniform time intervals. Figure A-7b shows this same signal after resampling at uniform shaft angle increments. The frequency spectrum of the swept sine in Figure A-7a is "smeared" over a band of frequencies, while that for Figure A-7b occurs at only the 5th harmonic of the shaft rotation rate in the order domain (5th order), assuming that the plot is scaled to show exactly one revolution of the shaft (Figure A-7c).

Since the data is buffered in memory, it is possible to "look ahead," and to use data points that occur before the desired tachometer pulse times occur. This allows the design of a tracking algorithm that has no inherent time delay, and thus never gets behind, as long as the shaft is constantly accelerating (or running at a constant velocity). In addition, the shaft velocity can be correctly calculated at each instant in time. There is no significant internal phase noise introduced by this procedure, although it is important to measure the arrival time of each tachometer pulse very accurately to reduce the effects of time jitter.

Though in actual measurements the requirement that the shaft acceleration be constant is not met, the model can be updated at each tachometer pulse so errors introduced generally are quite small. It is possible to use a more complex model of the shaft acceleration, but this would introduce an additional performance penalty due to the increased computation time, and the increases in accuracy have not warranted this step. Generally, the traditional phase lock loop/ratio synthesizer implementation can be thought of as modeling the shaft position as constant velocity between tachometer pulses. Figure A-4 illustrates the problems with this assumption.

Figure A-8
Example of errors inherent in a simple two point linear interpolation scheme.

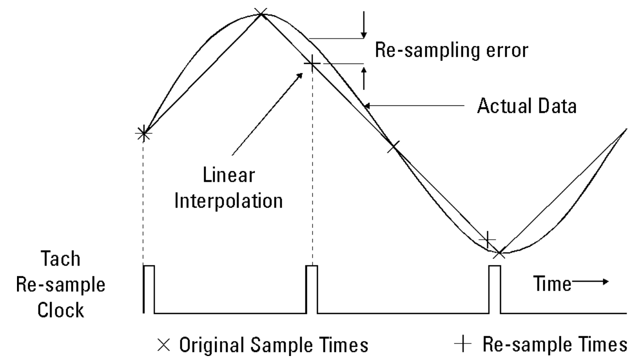
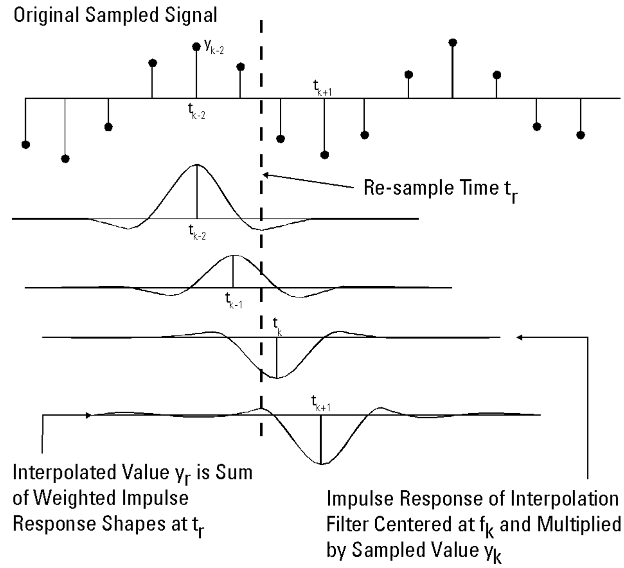


Figure A-9
Illustration of the implementation of a multi-point FIR interpolation scheme.



The computed order tracking approach requires considerably less in the way of special hardware, compared to the classical technique. For example, in addition to the need for a tracking ratio synthesizer, the classical order tracking method requires tracking anti-aliasing filters on each data channel and often a frequency counter to determine shaft velocity. With the new approach, a fixed analog anti-aliasing filter is used, and the remaining filtering operations are done digitally (this is the same hardware used in the normal FFT analysis mode). There is no need for an analog tracking ratio synthesizer, since the shaft position is continually calculated from the tachometer pulse arrival times.

Resampling Amplitude

With traditional ratio synthesizer/shaft encoder techniques, once the resampling signals have been created, the remainder of the order analysis can use standard FFT technology with variable A/D converter sampling rates to accomplish the remainder of the processing. This assumes that adequate alias protection is provided by some variable low pass filtering technique.

In the case of digital order tracking, the situation is not as straightforward since the data has already been anti-alias filtered and digitized at some fixed rate. The digital resampling process actually accomplishes two functions; first it provides the variable sample rates. Second, it provides the variable frequency low pass filtering (in conjunction with the fixed filtering of the DSA's digital and analog anti-alias filters) required for adequate alias protection. External sampling implementations must handle the anti-alias filtering as a separate step; normally adding a separate analog tracking filter to the input.

As described in the previous section, the desired sampling times were calculated based on the tachometer pulses and a linear acceleration model. In general, these times will lay between two fixed rate samples and an estimate must be made of the amplitude at the desired time based on existing data. The fixed sample data is buffered in computer memory so, again, we can look both ahead and back in time at the data to estimate the new resampled value. The simplest scheme would be to use linear interpolation between the two neighboring points. Though this would work to some extent, it's apparent from Figure A-8 that significant errors can be introduced.

This approach would lead to amplitude errors and a severe limitation on the system's dynamic range. In the case of simple linear interpolation, an amplitude error of 10% and an effective dynamic range of only 26 dB would be realizable. To reduce this error and increase the dynamic range, it's necessary to use more data in evaluating our resampled amplitude. The current implementation utilizes ten neighboring data points (five before and five after the resample time) to calculate the resample data point, formulated as a finite impulse response (FIR) filter (Figure A-9).

Theoretically, this formulation would lead to amplitude accuracy of .08% and a dynamic range of approximately 104 dB. To improve speed, the actual filter is implemented as a look-up table in memory, which leads to some round-off errors yielding the desired 80 dB dynamic range.

Example Measurements

To demonstrate the digital implementation under realistic conditions, run-up tests were performed on an automobile utilizing moderate run-up rates. The data was actually digitally recorded and analyzed using traditional external sampling/ratio synthesis techniques as well as the digital resampling technique. The following plots illustrate the differences between the two techniques.

In Figure A-12A the ramp rates were not particularly fast nor the order analysis very high, which are the conditions where the computed method would normally be expected to perform better. In spite of the low ramp rate, the automobile engine RPM was not particularly steady and would “jitter” about its average value. The traditional method’s phase lock loop tended to average these variations and consequently resulted in some loss of resolution. Figure A-10 is the computed order tracking measurement and the half and odd number orders are quite clear as are the dominant even order, which would be expected from a four-cylinder, four-cycle engine. Also visible is the 60 Hz noise component, which is lost in Figure A-11.

Figure A-11 is a similar measurement using external sampling and a tracking ratio synthesizer with a tracking low-pass filter. The conditions have been set to give equivalent resolution and low-amplitude orders are much less distinct and the 60 Hz is not discernible.

The differences can be attributed to phase lock loop delay. Reference [36] further illustrates this by examining the ability of a ratio synthesizer to track a square wave swept at known rates. Figure A-12a and b illustrate the order tracking capability of the digital technique which includes phase information. The FIR filter used in the digital implementation has essentially no phase shift or time delay allowing accurate phase measurements.

Figure A-10 Order ratio map of an automobile run-up with a four-cylinder, four-cycle engine utilizing computed resampling.

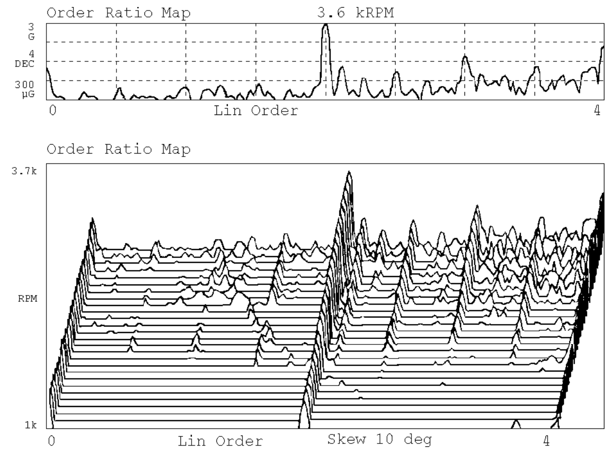


Figure A-11 Order ratio map utilizing external sampling and a PLL ratio synthesizer.

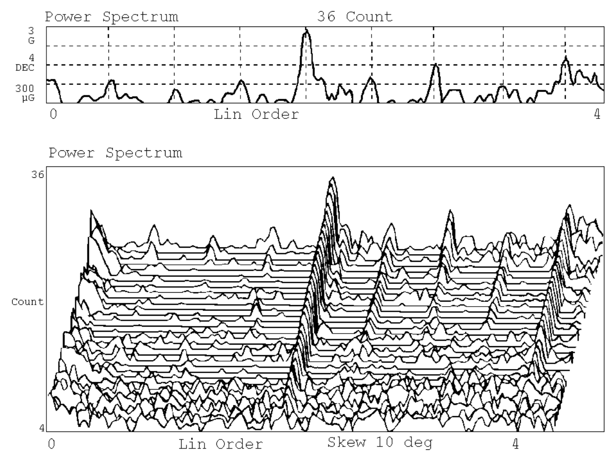
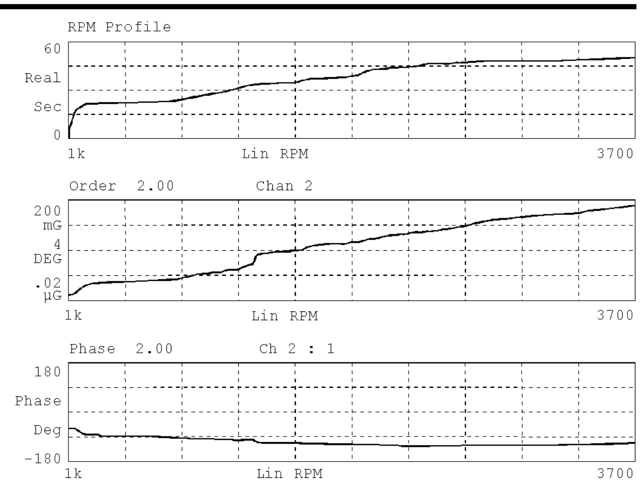


Figure A-12 Order track measurement made using digital resampling techniques. (a) Time vs RPM (b) Amplitude of two order vs RPM and (c) Phase of two order vs RPM.



Acceleration. The time rate of change of velocity. Typical units are ft/s/s, meters/s/s, and G's (1G = 32.17 ft/s/s = 9.81 m/s/s). Acceleration measurements are usually made with accelerometers.

Accelerometer. Transducer whose output is directly proportional to acceleration. Most commonly use piezoelectric crystals to produce output.

Aliasing. A phenomenon which can occur whenever a signal is not sampled at greater than twice the maximum frequency component. Causes high frequency signals to appear at low frequencies. Aliasing is avoided by filtering out signals greater than 1/2 the sample rate.

Alignment. A condition whereby the axes of machine components are either coincident, parallel or perpendicular, according to design requirements.

Amplification Factor (Synchronous). A measure of the susceptibility of a rotor to vibration amplitude when rotational speed is equal to the rotor natural frequency (implies a flexible rotor). For imbalance type excitation, synchronous amplification factor is calculated by dividing the amplitude value at the resonant peak by the amplitude value at a speed well above resonance (as determined from a plot of synchronous response vs. rpm).

Amplitude. The magnitude of dynamic motion or vibration. Amplitude is expressed in terms of peak-to- peak, zero-to-peak, or rms. For pure sine waves only, these are related as follows: rms = 0.707 times zero-to-peak; peak-to-peak = 2 times zero-to-peak. DSAs generally display rms for spectral components, and peak for time domain components.

Anti-Aliasing Filter. A low-pass filter designed to filter out frequenices higher than 40% the sample rate in order to prevent aliasing.

Anti-Friction Bearing. See Rolling Element Bearing.

Asymetrical Support. Rotor support system that does not provide uniform restraint in all radial directions. This is typical for most heavy industrial machinery where stiffness in one plane may be substantially different than stiffness in the perpendicular plane. Occurs in bearings by design, or from preloads such as gravity or misalignment

Asynchronous. Vibration components that are not related to rotating speed (also referred to as nonsynchronous).

Attitude Angle (Steady-State). The angle between the direction of steady-state preload through the bearing centerline, and a line drawn between the shaft centerline and the bearing centerline. (Applies to fluid-film bearings.)

Auto Spectrum (Power Spectrum). DSA spectrum display whose magnitude represents the power at each frequency, and which has no phase. Rms averaging produces an auto spectrum.

Averaging. In a DSA, digitally averaging several measurements to improve accuracy or to reduce the level of asynchronous components. Refer to definitions of rms, time, and peak-hold averaging.

Axial. In the same direction as the shaft centerline.

Axial Position. The average position, or change in position, of a rotor in the axial direction with respect to some fixed reference position. Ideally the reference is a known position within the thrust bearing axial clearance or float zone, and the measurement is made with a displacement transducer observing the thrust collar.

Balancing Resonance Speed(s). A rotative speed that corresponds to a natural resonance frequency.

Balanced Condition. For rotating machinery, a condition where the shaft geometric centerline coincides with the mass centerline.

Balancing. A procedure for adjusting the radial mass distribution of a rotor so that the mass centerline approaches the rotor geometric centerline.

Band-Pass Filter. A filter with a single transmission band extending from lower to upper cutoff frequencies. The width of the band is determined by the separation of frequencies at which amplitude is attenuated by 3 dB (0.707).

Bandwidth. The spacing between frequencies at which a band-pass filter attenuates the signal by 3 dB. In a DSA, measurement bandwidth is equal to [(frequency span)/(number of filters) x (window factor)]. Window factors are: 1 for uniform, 1.5 for Hanning, and 3.63 for flat top.

Baseline Spectrum. A vibration spectrum taken when a machine is in good operating condition; used as a reference for monitoring and analysis.

Blade Passing Frequency. A potential vibration frequency on any bladed machine (turbine, axial compressor, fan, etc.). It is represented by the number of blades times shaft-rotating frequency.

Block Size. The number of samples used in a DSA to compute the Fast Fourier Transform. Also the number of samples in a DSA time display. Most DSAs use a block size of from 250 to 8192. Smaller block size reduces resolution.

Bode Plot. Rectangular coordinate plot of 1x component amplitude and phase (relative to a keyphasor) vs. running speed.

BPFO, BPFI. Common abbreviations for ball pass frequency of defects on outer and inner bearing races, respectively.

Bow. A shaft condition such that the geometric centerline of the shaft is not straight. Also called shaft sag.

Brinneling (False). Impressions made by bearing rolling elements on the bearing race; typically caused by external vibration when the shaft is stationary.

Calibration. A test during which known values of the measured variable are applied to the transducer or readout instrument, and output readings varied or adjusted.

Campbell Diagram. A mathematically constructed diagram used to check for coincidence of vibration sources (i.e. 1 x imbalance, 2 x misalignment) with rotor natural resonances. The form of the diagram is a rectangular plot of resonant frequency (y-axis) vs excitation frequency (x-axis). Also known as an interference diagram.

Cascade Plot. See Spectral Map.

Cavitation. A condition which can occur in liquid-handling machinery (e.g. centrifugal pumps) where a system pressure decrease in the suction line and pump inlet lowers fluid pressure and vaporization occurs. The result is mixed flow which may produce vibration.

Center Frequency. For a bandpass filter, the center of the transmission band.

Charge Amplifier. Amplifier used to convert accelerometer output impedance from high to low, making calibration much less dependent on cable capacitance.

Coherence. The ratio of coherent output power between channels in a dual-channel DSA. An effective means of determining the similarity of vibration at two locations, giving insight into the possibility of cause and effect relationships.

Constant Bandwidth Filter. A band-pass filter whose bandwidth is independent of center frequency. The filters simulated digitally in a DSA are constant bandwidth.

Constant Percentage Bandwidth. A band-pass filter whose bandwidth is a constant percentage of center frequency. 1/3 octave filters, including those synthesized in DSAs, are constant percentage bandwidth.

Critical Machinery. Machines which are critical to a major part of the plant process. These machines are usually unspared.

Critical Speeds. In general, any rotating speed which is associated with high vibration amplitude. Often, the rotor speeds which correspond to natural frequencies of the shaft or the system.

Critical Speed Map. A rectangular plot of system natural frequency (y-axis) versus bearing or support stiffness (x-axis).

Cross Axis Sensitivity. A measure of off-axis response of velocity and acceleration transducers.

Cycle. One complete sequence of values of a periodic quantity.

Damping. The quality of a mechanical system that restrains the amplitude of motion with each successive cycle. Damping of shaft motion is provided by oil in bearings, seals, etc. The damping process converts mechanical energy to other forms, usually heat.

Damping, Critical. The smallest amount of damping required to return the system to its equilibrium position without oscillation.

Decibels (dB). A logarithmic representation of amplitude ratio, defined as 20 times the base ten logarithm of the ratio of the measured amplitude to a reference. dB readings, for example, are referenced to 1 volt rms. dB or Log amplitude scales are required to display the full dynamic range of a DSA.

Degrees Of Freedom. A phrase used in mechanical vibration to describe the complexity of the system. The number of degrees of freedom is the number of independent variables describing the state of a vibrating system.

Digital Filter. A filter which acts on data after it has been sampled and digitized. Often used in DSAs to provide anti-aliasing protection after internal re-sampling.

Differentiation. Representation in terms of time rate of change. For example, differentiating velocity yields acceleration. In a DSA, differentiation is performed by multiplication by *j*w, where w is frequency multiplied by 2p. (Differentiation can also be used to convert displacement to velocity.)

Discrete Fourier Transform. A procedure for calculating discrete frequency components (filters or lines) from sampled time data. Since the frequency domain result is complex (i.e., real and imaginary components), the number of points is equal to half the number of samples.

Displacement. The change in distance or position of an object relative to a reference.

Displacement Transducer. A transducer whose output is proportional to the distance between it and the measured object (usually the shaft).

DSA. See Dynamic Signal Analyzer.

Dual Probe. A transducer set consisting of displacement and velocity transducers. Combines measurement of shaft motion relative to the displacement transducer with velocity of the displacement transducer to produce absolute motion of the shaft.

Dual Voting. Concept where two independent inputs are required before action (usually machine shutdown) is taken. Most often used with axial position measurements, where failure of a single transducer might lead to an unnecessary shutdown.

Dynamic Motion. Vibratory motion of a rotor system caused by mechanisms that are active only when the rotor is turning at speeds above slow roll speed.

Dynamic Signal Analyzer (DSA). Vibration analyzer that uses digital signal processing and the Fast Fourier Transform to display vibration frequency components. DSAs also display the time domain and phase spectrum, and can usually be interfaced to a computer.

Eccentricity, Mechanical. The variation of the outer diameter of a shaft surface when referenced to the true geometric centerline of the shaft. Out-of-roundness.

Eccentricity Ratio. The vector difference between the bearing centerline and the average steady-state journal centerline.

Eddy Current. Electrical current which is generated (and dissipated) in a conductive material in the presence of an electromagnetic field.

Electrical Runout. An error signal that occurs in eddy current displacement measurements when shaft surface conductivity varies.

Engineering Units. In a DSA, refers to units that are calibrated by the user (e.g., in/s, g's).

External Sampling. In a DSA, refers to control of data sampling by a multiplied tachometer signal. Provides a stationary display of vibration with changing speed.

Fast Fourier Transform (FFT). A computer (or microprocessor) procedure for calculating discrete frequency components from sampled time data. A special case of the discrete Fourier transform where the number of samples is constrained to a power of 2.

Filter. Electronic circuitry designed to pass or reject a specific frequency band.

Filtered Orbit. An orbit diagram in which the vertical and horizontal displacement signals have been filtered. This is normally a bandpass filter centered at a running speed, however, digital systems are capable of multiple bandpass regions.

Finite Element Modeling. A computer aided design technique for predicting the dynamic behavior of a mechanical system prior to construction. Modeling can be used, for example, to predict the natural frequencies of a flexible rotor.

Flat Top Filter. DSA window function which provides the best amplitude accuracy for measuring discrete frequency components.

Fluid-Film Bearing. A bearing which supports the shaft on a thin film of oil. The fluid-film layer may be generated by journal rotation (hydrodynamic bearing), or by externally applied pressure (hydrostatic bearing).

Forced Vibration. The oscillation of a system under the action of a forcing function. Typically forced vibration occurs at the frequency of the exciting force.

Free Vibration. Vibration of a mechanical system following an initial force — typically at one or more natural frequencies.

Frequency. The repetition rate of a periodic event, usually expressed in cycles per second (Hz), revolutions per minute (rpm), or multiples of a rotational speed (orders). Orders are commonly referred to as 1x for rotational speed, 2x for twice rotational speed, etc.

Frequency Response. The amplitude and phase response characteristics of a system.

G. The value of acceleration produced by the force of gravity.

Gear Mesh Frequency. A potential vibration frequency on any machine that contains gears; equal to the number of teeth multiplied by the rotational frequency of the gear.

Hanning Window. DSA window function that provides better frequency resolution than the flat top window, but with reduced amplitude accuracy.

Harmonic. Frequency component at a frequency that is an integer multiple of the fundamental frequency.

Heavy Spot. The angular location of the imbalance vector at a specific lateral location on a shaft. The heavy spot typically does not change with rotational speed.

Hertz (Hz). The unit of frequency represented by cycles per second.

High Spot. The angular location on the shaft directly under the vibration transducer at the point of closest proximity. The high spot can move with changes in shaft dynamics (e.g., from changes in speed).

High-Pass Filter. A filter with a transmission band starting at a lower cutoff frequency and extending to (theoretically) infinite frequency.

Hysteresis. Non-uniqueness in the relationship between two variables as a parameter increases or decreases. Also called deadband, or that portion of a system's response where a change in input does not produce a change in output.

Imbalance. Unequal radial weight distribution on a rotor system; a shaft condition such that the mass and shaft geometric centerlines do not coincide.

Impact Test. Response test where the broad frequency range produced by an impact is used as the stimulus. Sometimes referred to as a bump test.

Impedance, Mechanical. The mechanical properties of a machine system (mass, stiffness, damping) that determine the response to periodic forcing functions.

Influence Coefficients. Mathematical coefficients that describe the influence of system loading on system deflection.

Integration. A process producing a result that, when differentiated, yields the original quantity. Integration of acceleration, for example, yields velocity. Integration is performed in a DSA by dividing by jw, where w is frequency multiplied by 2p. (Integration is also used to convert velocity to displacement).

Journal. Specific portions of the shaft surface from which rotor applied loads are transmitted to bearing supports.

Keyphasor. A signal used in rotating machinery measurements, generated by a transducer observing a once-per-revolution event.

The keyphasor signal is used in phase measurements for analysis and balancing.

Lateral Location. The definition of various points along the shaft axis of rotation.

Lateral Vibration. See Radial Vibration.

Leakage. In DSAs, a result of finite time record length that results in smearing of frequency components. Its effects are greatly reduced by the use of weighted window functions such as flat top and Hanning.

Linearity. The response characteristics of a linear system remain constant with input level. That is, if the response to input a is A, and the response to input b is B, then the response of a linear system to input (a + b) will be (A + B). An example of a non-linear system is one whose response is limited by mechanical stop, such as occurs when a bearing mount is loose.

Lines. Common term used to describe the filters of a DSA (e.g., 400 line analyzer).

Linear Averaging. See Time Averaging.

Low-Pass Filter. A filter whose transmission band extends from dc to an upper cutoff frequency.

Mechanical Runout. An error in measuring the position of the shaft centerline with a displacement probe that is caused by out-of-roundness and surface imperfections.

Micrometer (MICRON). One millionth (.000001) of a meter. (1 micron = 1 x E-6 meters @ 0.04 mils.)

MIL. One thousandth (0.001) of an inch. (1 mil = 25.4 microns.)

Modal Analysis. The process of breaking complex vibration into its component modes of vibration, very much like frequency domain analysis breaks vibration down to component frequencies.

Mode Shape. The resultant deflected shape of a rotor at a specific rotational speed to an applied forcing function. A three-dimensional presentation of rotor lateral deflection along the shaft axis.

Modulation, Amplitude (AM). The process where the amplitude of a signal is varied as a function of the instantaneous value of another signal. The first signal is called the carrier, and the second signal is called the modulating signal. Amplitude modulation produces a component at the carrier frequency, with adjacent components (sidebands) at the frequency of the modulating signal.

Modulation, Frequency (FM). The process where the frequency of the carrier is determined by the amplitude of the modulating signal. Frequency modulation produces a component at the carrier frequency, with adjacent components (sidebands) at the frequency of the modulating signal.

Natural Frequency. The frequency of free vibration of a system. The frequency at which an undamped system with a single degree of freedom will oscillate upon momentary displacement from its rest position.

Nodal Point. A point of minimum shaft deflection in a specific mode shape. May readily change location along the shaft axis due to changes in residual imbalance or other forcing function, or change in restraint such as increased bearing clearance.

Noise. Any component of a transducer output signal that does not represent the variable intended to be measured.

Nyquist Criterion. Requirement that a sampled system sample at a frequency greater than twice thehighest frequency to be measured.

Nyquist Plot. A plot of real versus imaginary spectral components that is often used in servo analysis. Should not be confused with a polar plot of amplitude and phase of 1x vibration.

Octave. The interval between two frequencies with a ratio of 2 to 1.

Oil Whirl/Whip. An unstable free vibration whereby a fluid-film bearing has insufficient unit loading. Under this condition, the shaft centerline dynamic motion is usually circular in the direction of rotation. Oil whirl occurs at the oil flow velocity within the bearing, usually 40 to 49% of shaft speed. Oil whip occurs when the whirl frequency coincide with (and becomes locked to) a shaft resonant frequency. (Oil whirl and whip can occur in any case where fluid is between two cylindrical surfaces.)

Orbit. The path of the shaft centerline motion during rotation. The orbit is observed with an oscilloscope connected to x and y-axis displacement transducers. Some dual-channel DSAs also have the ability to display orbits.

Oscillator-Demodulator. A signal conditioning device that sends a radio frequency signal to an eddy-current displacement probe, demodulates the probe output, and provides output signals proportional to both the average and dynamic gap distances. (Also referred to as Proximitor, a Bently Nevada trade name.)

Peak Hold. In a DSA, a type of averaging that holds the peak signal level for each frequency component.

Period. The time required for a complete oscillation or for a single cycle of events. The reciprocal of frequency.

Phase. A measurement of the timing relationship between two signals, or between a specific vibration event and a keyphasor pulse.

Piezoelectric. Any material which provides a conversion between mechanical and electrical energy. For a piezoelectric crystal, if mechanical stresses are applied on two opposite faces, electrical charges appear on some other pair of faces.

Polar Plot. Polar coordinate representation of the locus of the 1x vector at a specific lateral shaft location with the shaft rotational speed as a parameter.

Power Spectrum. See Auto Spectrum.

Preload, Bearing. The dimensionless quantity that is typically expressed as a number from zero to one where a preload of zero indicates no bearing load upon the shaft, and one indicates the maximum preload (i.e., line contact between shaft and bearing).

Preload, External. Any of several mechanisms that can externally load a bearing. This includes "soft" preloads such as process fluids or gravitational forces as well as "hard" preloads from gear contact forces, misalignment, rubs, etc.

Proximitor. See Oscillator/ Demodulator.

Radial. Direction perpendicular to the shaft centerline.

Radial Position. The average location, relative to the radial bearing centerline, of the shaft dynamic motion.

Radial Vibration. Shaft dynamic motion or casing vibration which is in a direction perpendicular to the shaft centerline.

Real-Time Analyzer. See Dynamic Signal Analyzer.

Real-Time Rate. For a DSA, the broadest frequency span at which data is sampled continuously. Real-time rate is mostly dependent on FFT processing speed.

Rectangular Window. See Uniform Window.

Relative Motion. Vibration measured relative to a chosen reference. Displacement transducers generally measure shaft motion relative to the transducer mounting.

Repeatability. The ability of a transducer or readout instrument to reproduce readings when the same input is applied repeatedly.

Resolution. The smallest change in stimulus that will produce a detectable change in the instrument output.

Resonance. The condition of vibration amplitude and phase change response caused by a corresponding system sensitivity to a particular forcing frequency. A resonance is typically identified by a substantial amplitude increase, and related phase shift.

Rolling Element Bearing. Bearing whose low friction qualities derive from rolling elements (balls or rollers), with little lubrication.

Root Mean Square (rms). Square root of the arithmetical average of a set of squared instantaneous values. DSAs perform rms averaging digitally on successive vibration spectra.

Rotor, Flexible. A rotor which operates close enough to, or beyond its first bending critical speed for dynamic effects to influence rotor deformations. Rotors which cannot be classified as rigid rotors are considered to be flexible rotors.

Rotor, Rigid. A rotor which operates substantially below its first bending critical speed. A rigid rotor can be brought into, and will remain in, a state of satisfactory balance at all operating speeds when balanced on any two arbitrarily selected correction planes.

RPM Spectral Map. A spectral map of vibration spectra versus rpm.

Runout Compensation. Electronic correction of a transducer output signal for the error resulting from slow roll runout.

Seismic. Refers to an inertially referenced measurement or a measurement relative to free space.

Seismic Transducer. A transducer that is mounted on the case or housing of a machine and measures casing vibration relative to free space. Accelerometers and velocity transducers are seismic.

Signal Conditioner. A device placed between a signal source and a readout instrument to change the signal. Examples: attenuators, preamplifiers, charge amplifiers.

Signature. Term usually applied to the vibration frequency spectrum which is distinctive and special to a machine or component, system or subsystem at a specific point in time, under specific machine operating conditions, etc. Used for historical comparison of mechanical condition over the operating life of the machine.

Slow Roll Speed. Low rotative speed at which dynamic motion effects from forces such as imbalance are negligible.

Spectral Map. A three-dimensional plot of the vibration amplitude spectrum versus another variable, usually time or rpm.

Spectrum Analyzer. An instrument which displays the frequency spectrum of an input signal.

Stiffness. The spring-like quality of mechanical and hydraulic elements to elasticity deform under load.

Strain. The physical deformation, deflection, or change in length resulting from stress (force per unit area).

Subharmonic. Sinusoidal quantity of a frequency that is an integral submultiple of a fundamental frequency.

Subsynchronous. Component(s) of a vibration signal which has a frequency less than shaft rotative frequency.

Synchronous Sampling. In a DSA, it refers to the control of the effective sampling rate of data; which includes the processes of external sampling and computed resampling used in order tracking.

Time Averaging. In a DSA, averaging of time records that results in reduction of asynchronous components.

Time Record. In a DSA, the sampled time data converted to the frequency domain by the FFT. Most DSAs use a time record of 1024 samples.

Torsional Vibration. Amplitude modulation of torque measured in degrees peak-to-peak referenced to the axis of shaft rotation.

Tracking Filter. A low-pass or band-pass filter which automatically tracks the input signal. A tracking filter is usually required for aliasing protection when data sampling is controlled externally.

Transducer. A device for translating the magnitude of one quantity into another quantity.

Transient Vibration. Temporarily sustained vibration of a mechanical system. It may consist of forced or free vibration or both. Typically this is associated with changes in machine operating condition such as speed, load, etc.

Transverse Sensitivity. See Cross-Axis Sensitivity.

Trigger. Any event which can be used as a timing reference. In a DSA, a trigger can be used to initiate a measurement.

Unbalance. See Imbalance.

Uniform Window. In a DSA, a window function with uniform weighting across the time record. This window does not protect against leakage, and should be used only with transient signals contained completely within the time record.

Vector. A quantity which has both magnitude and direction (phase).

Waterfall Plot. See Spectral Map.

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