HEWLETT® PACKARD

Effective Machinery Measurements using Dynamic Signal Analyzers

Application Note 243-1

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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.

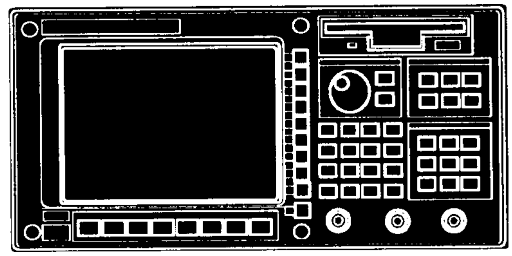


Figure 1-1
Dynamic Signal Analysers are (DSAs) are the ideal instrument for analyzing machinery vibration.

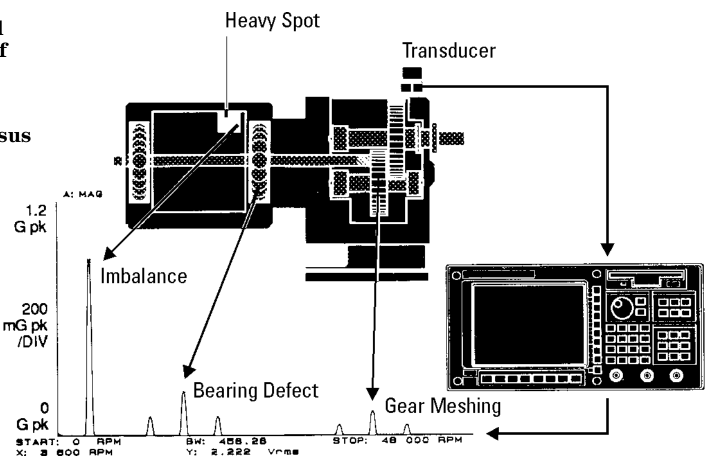


Figure 1-2
The individual components of vibration are shown in DSA displays of amplitude versus frequency.

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.

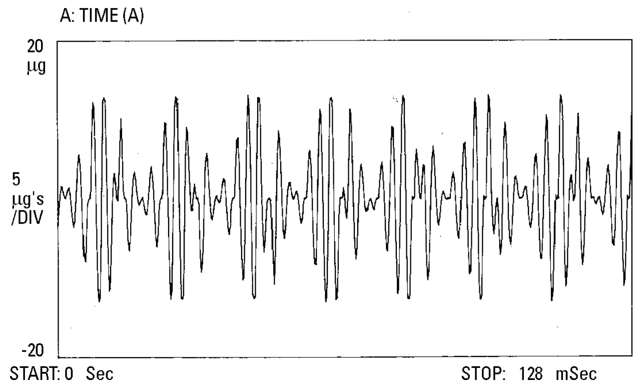


Figure 1-3
DSA display of amplitude versus time are especially useful for analyzing impulsive vibration that is characteristic of gear and rolling element defects.

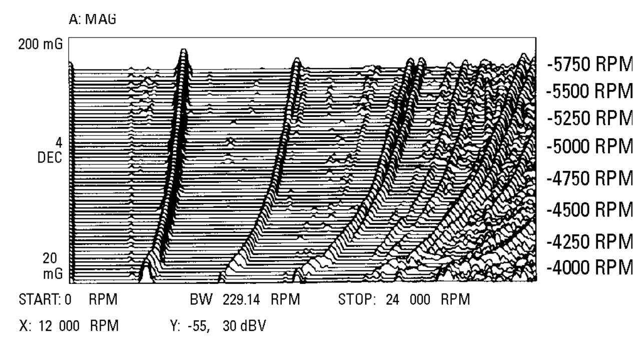


Figure 1-4
DSA map displays illustrate changes in vibration with rpm, load, or time. This map is a collection of vibration measurements made during a machine runup.

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.

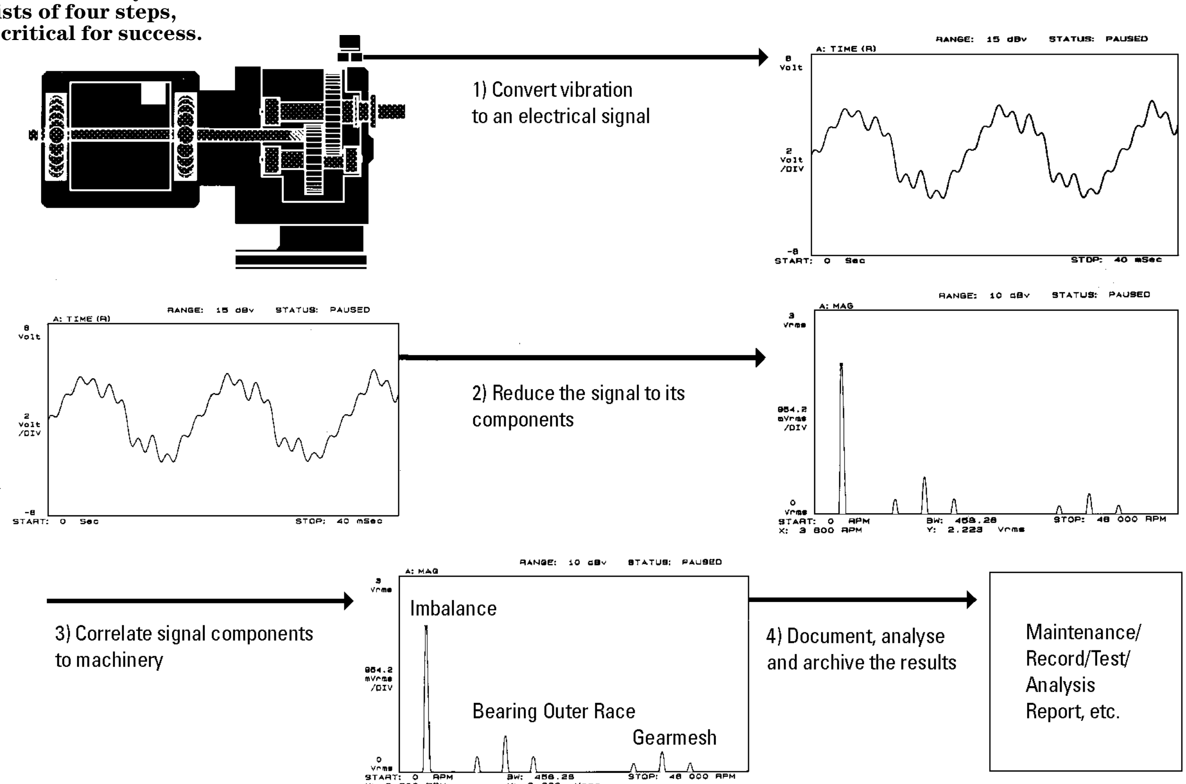
Figure 1-2 illustrates the principles presented; the vibration data is broken down into its individual frequency components by the DSA; the analysis can associate these components to particular elements in the machine and depending upon the objective, determine whether an individual component of the 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.

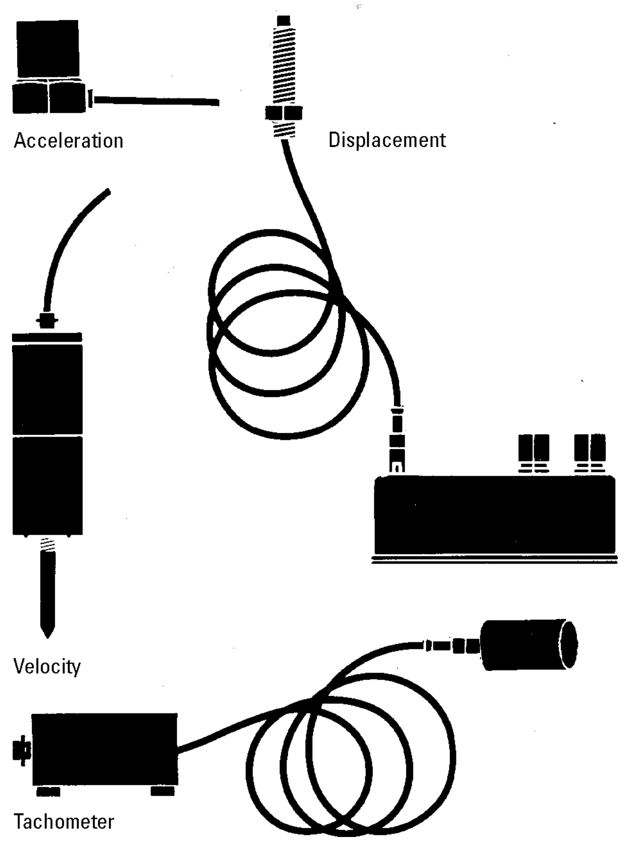
Chapter 2
Converting Vibration to an Electrical Signal

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.

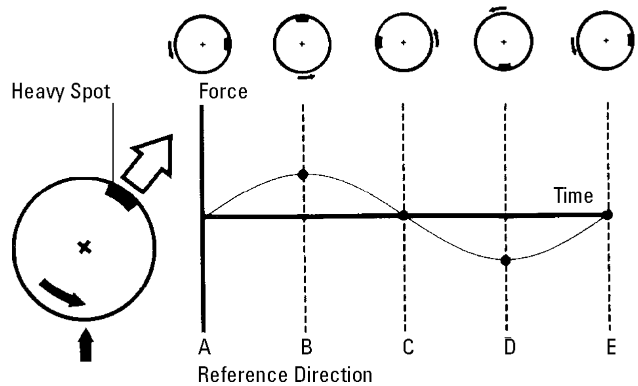


Figure 2.1-1
A heavy spot on a machine rotor results in a rotating force vector that appears sinusoidal from a fixed reference.

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).

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

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.

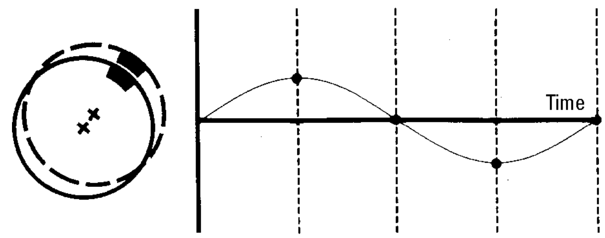


Figure 2.1-3 Velocity and acceleration of the vibration are offset 90° and 180° in phase from displacement.

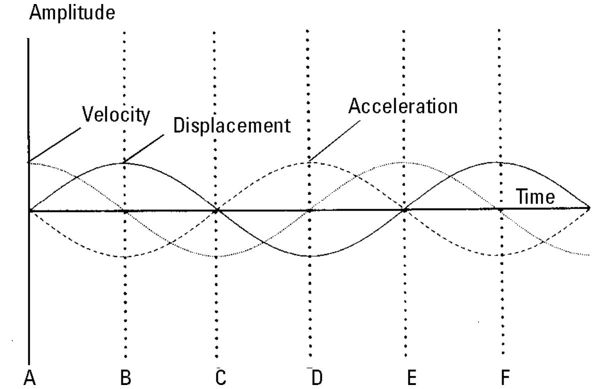


Figure 2.1-4 Two cases which illustrate the variation of vibration parameters with machine speed.

Case 1: 600 rpm fan

Displacement: 10 mils p-p
Velocity: 0.3 in/sec
Acceleration: 0.1g

Case 2: 15kHz gear mesh

Displacement: 1.2 mils p-p
Velocity: 0.12 in/sec
Acceleration: 30 g's

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.

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

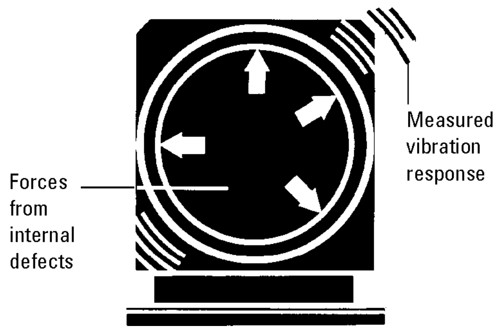
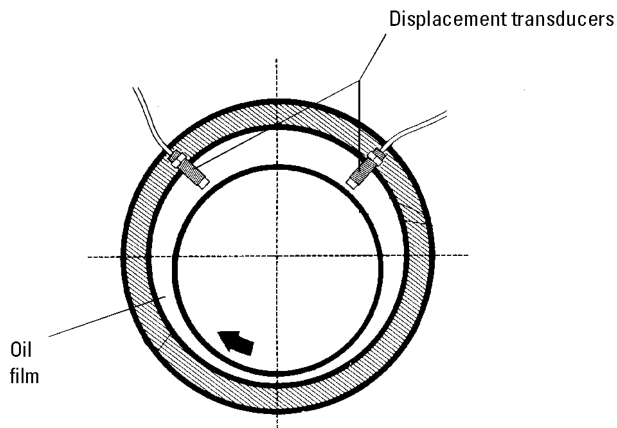


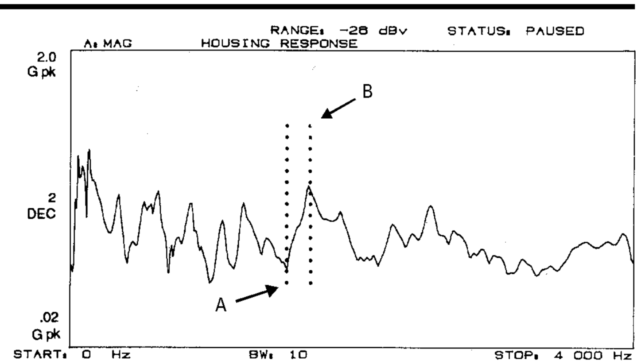
Figure 2.1-6 A relatively light shaft turning in fluid-film bearings transmits little vibration to the machine housing. Its vibration must be measured directly with a displacement transducer.



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 2.1-7 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 (ω_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.

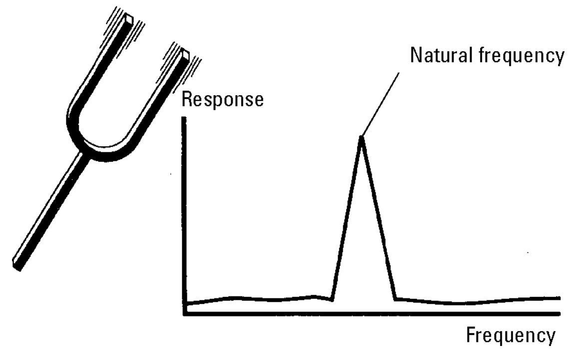
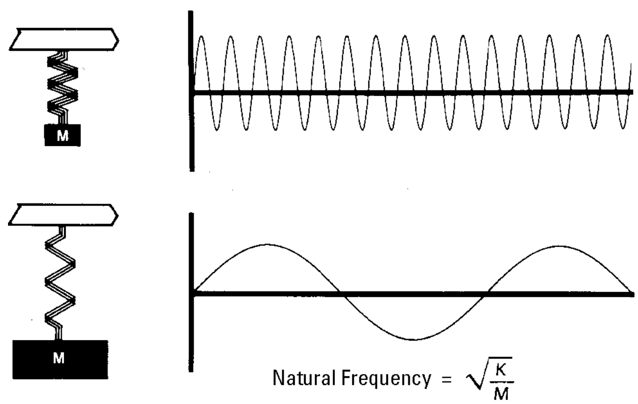


Figure 2.1-9 The natural frequency of a simple mechanical system varies with mass and stiffness.



[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.

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.

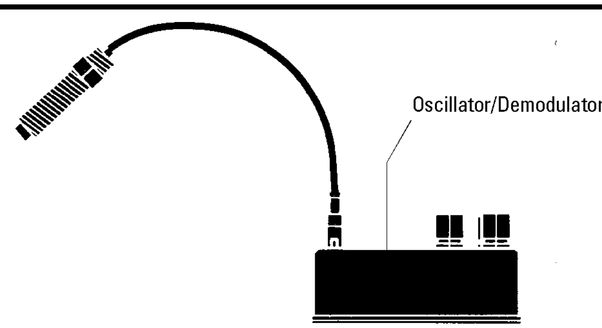
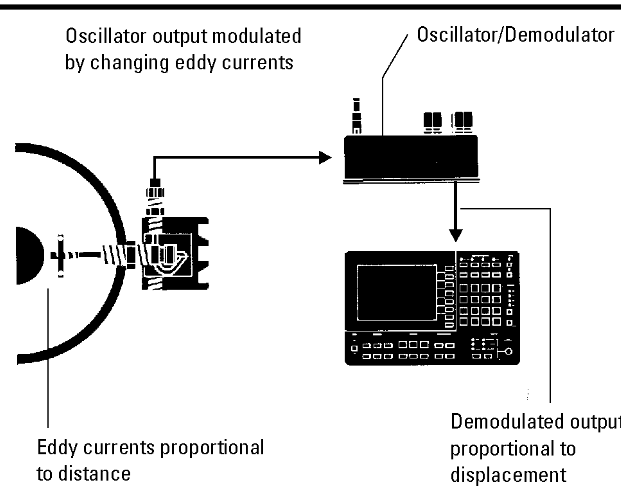


Figure2.2-2 Schematic diagram of a typical noncontacting displacment transducer installation.



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

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.

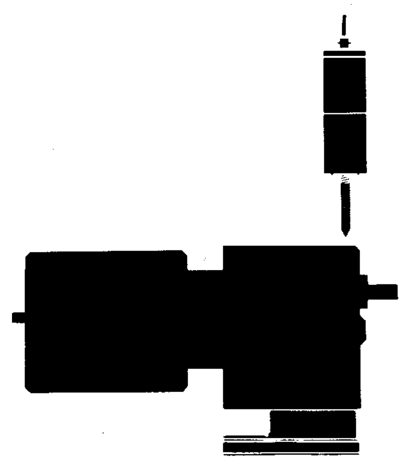


Figure 2.2-4
Velocity transducer output is a current generated in the coil as it moves through the field of the stationary magnet.

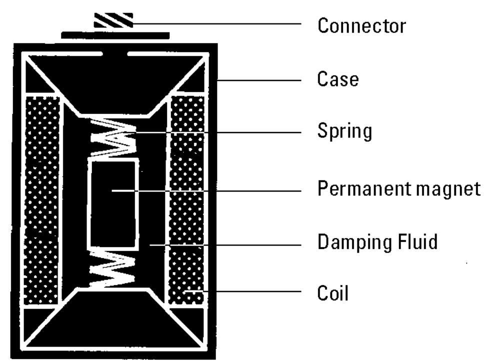
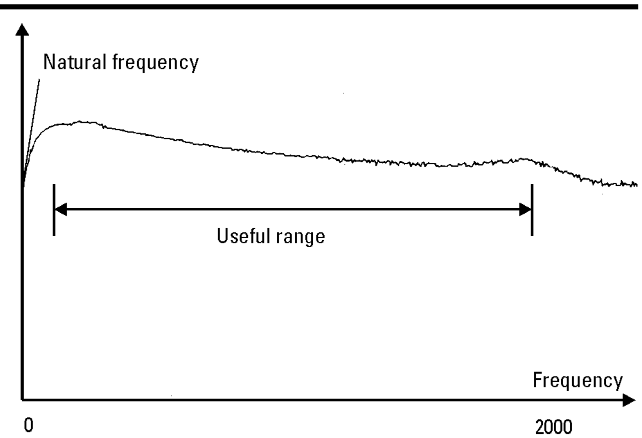


Figure 2.2-5
Frequency response of a typical velocity transducer. Note that the natural frenquency of the magnet-spring-damper system is below the operating range.



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.

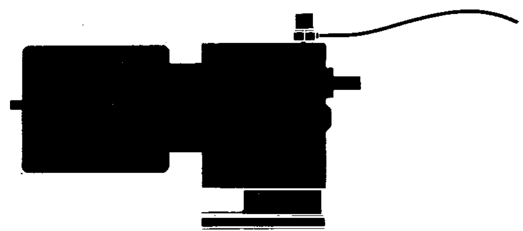
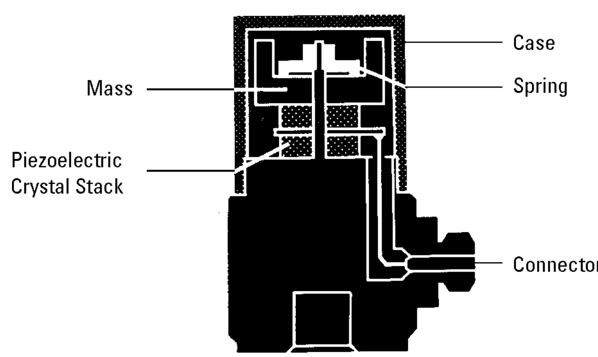


Figure 2.2-7 The output voltage of an accelerometer is produced by the accelerating mass squeezing the piezo-electric crystal stack. The force — and thusthe output voltage —is proportional to acceleration.



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.

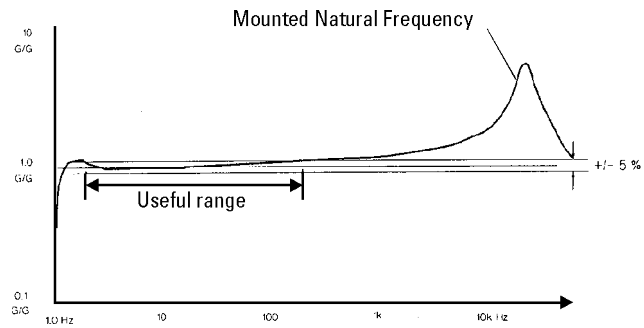


Figure 2.2-9a Traditional accelerometers require an external charge amplifier for signal conditioning.

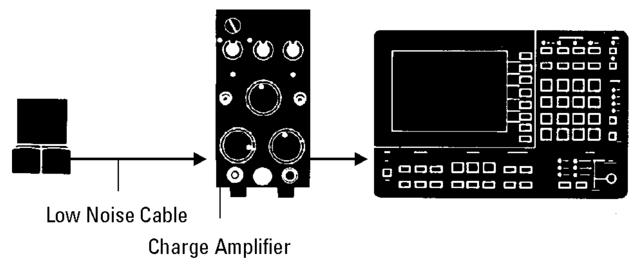
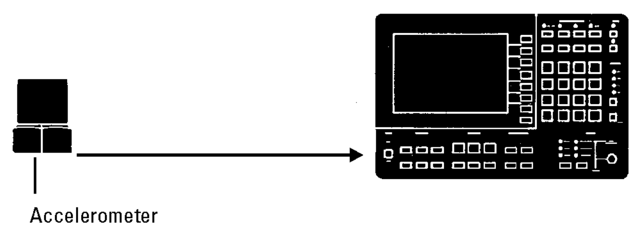
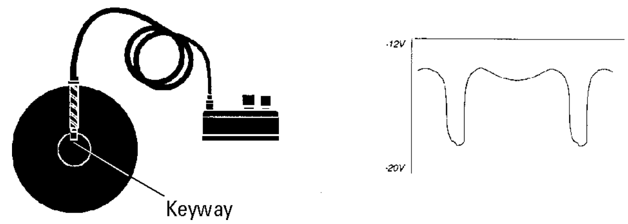


Figure 2.2-9b Integrated Circuit Piezoelectric accelerometers, with built-in signal conditioning, can be connected directly to a compatible DSA.



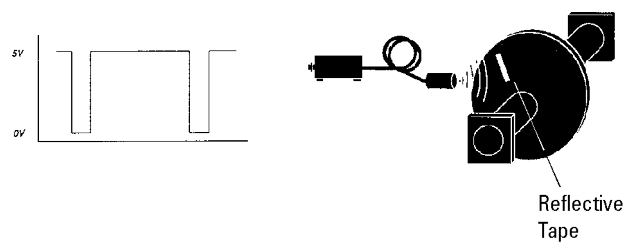
The proximity probe is the same as previously discussed, however, it is not used to get accurate displacement information in this mode. It is commonly used to detect the presence of something such as a keyway slot (often referred to as a keyphaser) or gear tooth. Figure 2.2-10(a) illustrates a proximity 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.

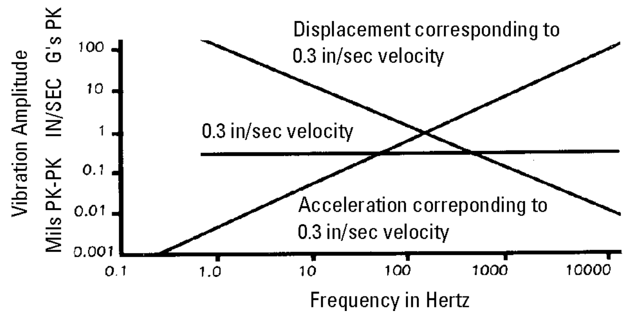


Figure 2.3
A vibration nomograph shows how the levels of displacement and acceleration change with frequency, relative to the level of velocity. Note that the acceleration response is very low at 1 Hz (less than 100 µV with a 10 mV/g accelerometer).

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.

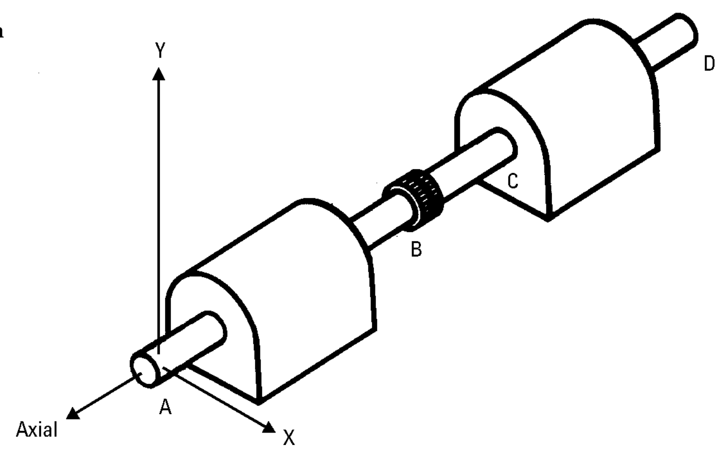
Table 2.4 Transducer Application Summary.

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.

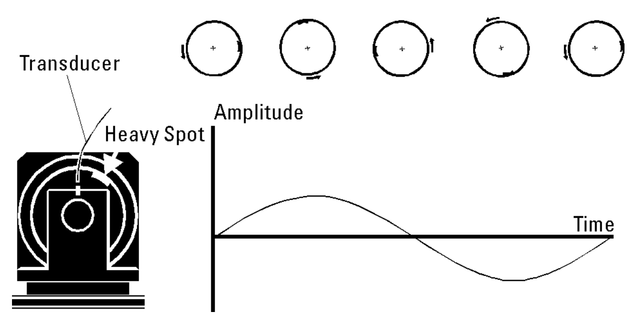
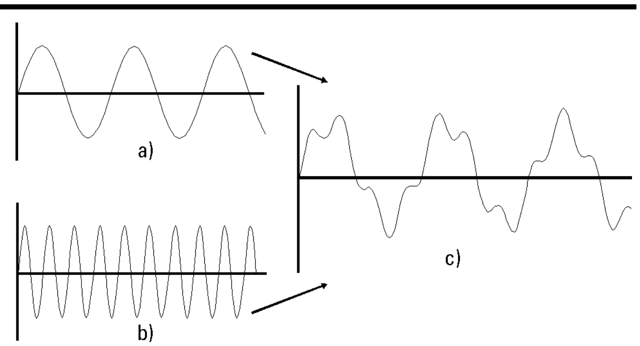


Figure 3.1-2 Waveform (c) is the combination of signals (a) and (b). The nature of these components is hidden in the time domain view of their sum.



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.

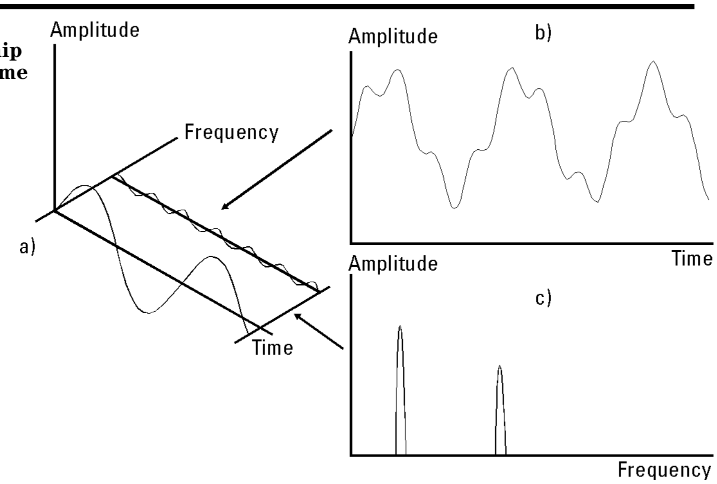
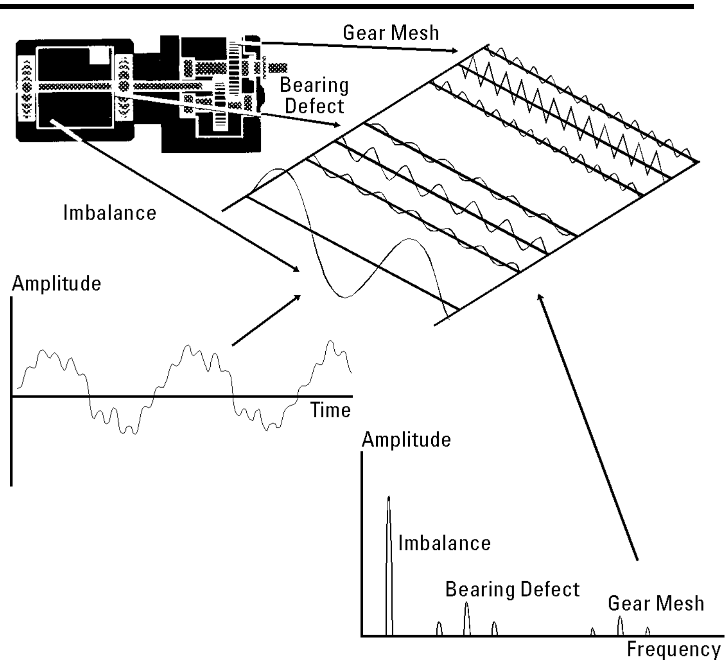


Figure 3.2-2 Machinery vibration viewed in the time and frequency domains.



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.

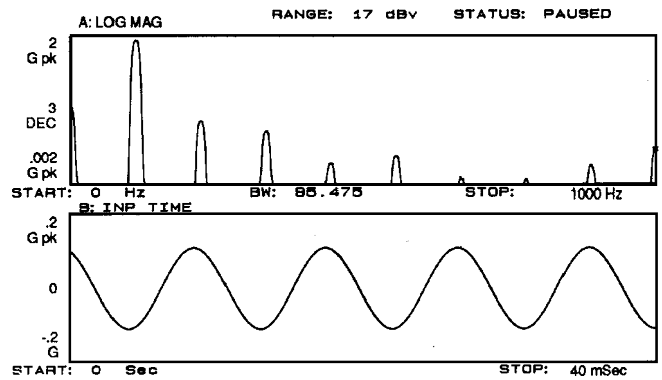
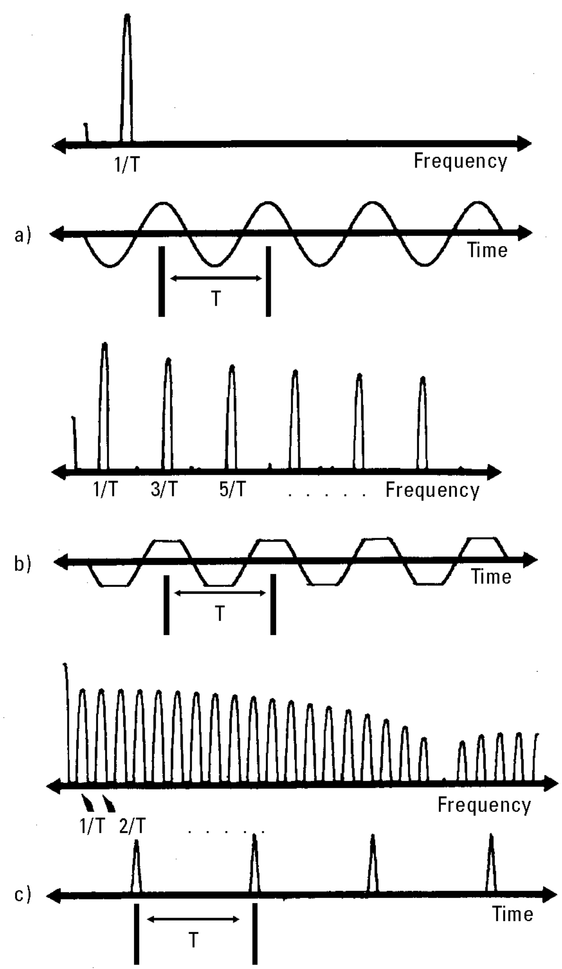


Figure 3.2-3
Small signals that are hidden in the time domain are readily apparent in the frequency domain. By using a logarithmic amplitude scale, signals which vary in level by a factor of over 1000 can be displayed.



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.

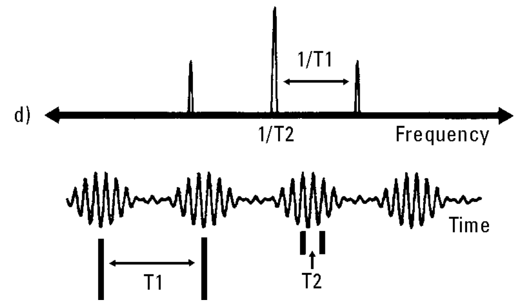
(a) The frequency spectrum of a pure sine wave is a single spectral line. For a sine wave of period T seconds, this line occurs at 1/THz.

(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.

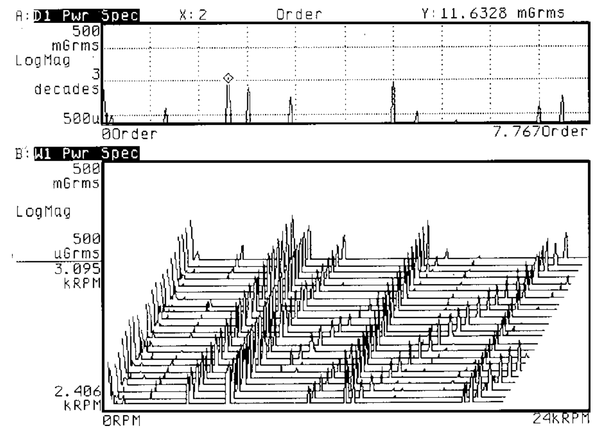


Figure 3.3-2 Set-up of DSA to produce spectral map.

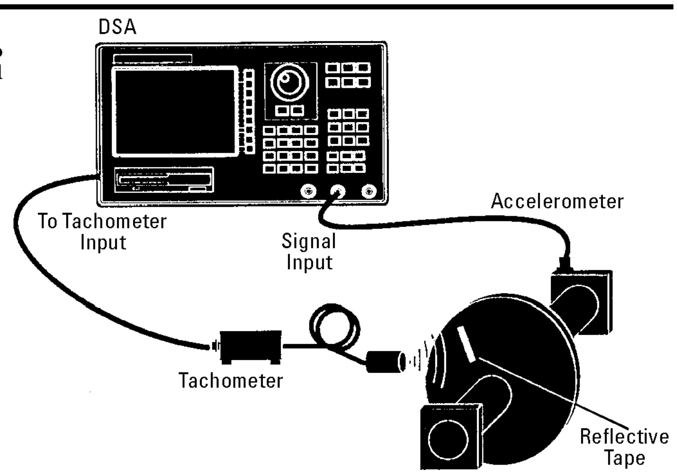


Figure 3.4-1 The phase of the imbalance signal corresponds to the direction of the displacement. One 360° rotation of the rotor corresponds to one 360° cycle of the signal.

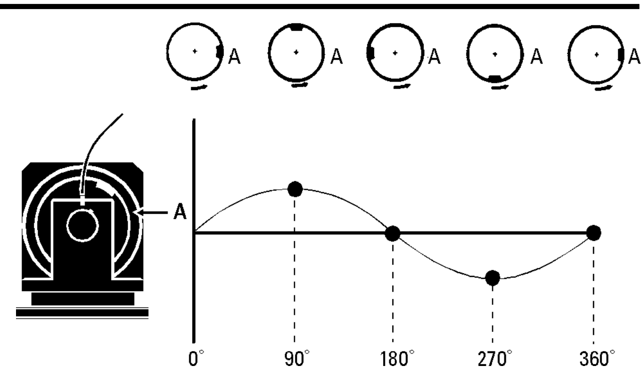
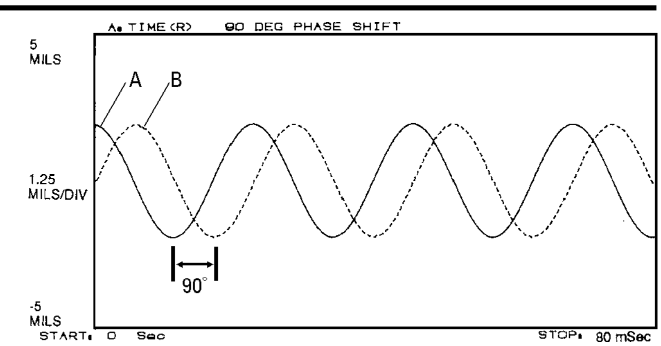


Figure 3.4-2 Two sine waves with a phase relationship of 90°.



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.

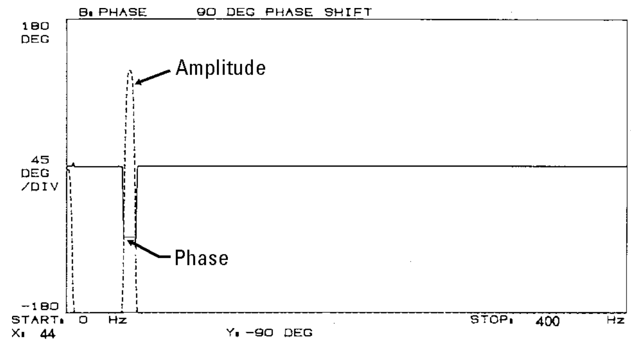


Figure 3.4-4
Since the heavy spot on the rotor passes the transducer 90° after the keyway passes the keyphasor, the imbalance signal lags the keyphasor pulse by 90°. The corresponding frequency domain phase spectrum is shown in Figure 3.4-3.

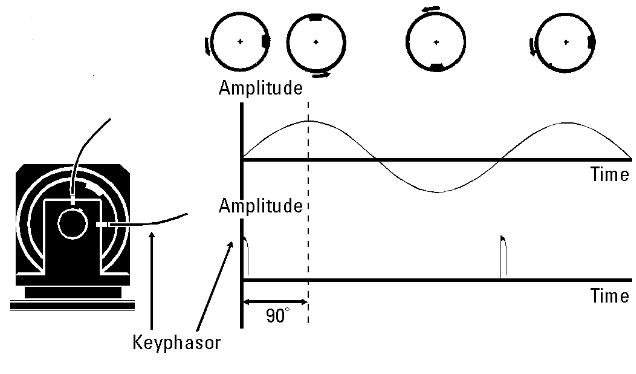


Figure 3.4-5
The vibration response of a flexible rotor shifts 180° in phase as rpm passes through a critical speed.

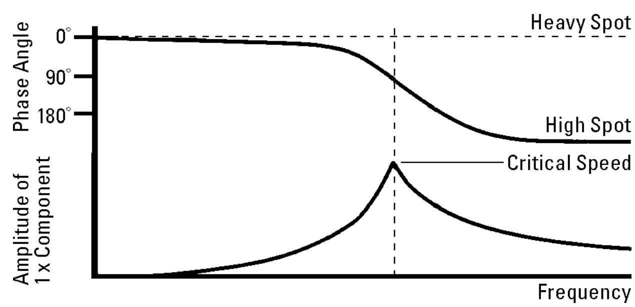
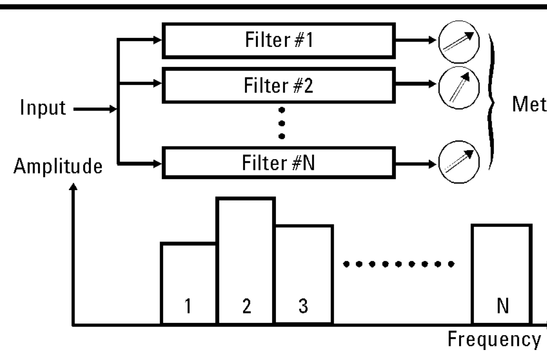


Figure 3.5-1
Parallel filter analyzers have insufficient frequency resolution for machinery analysis.



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.

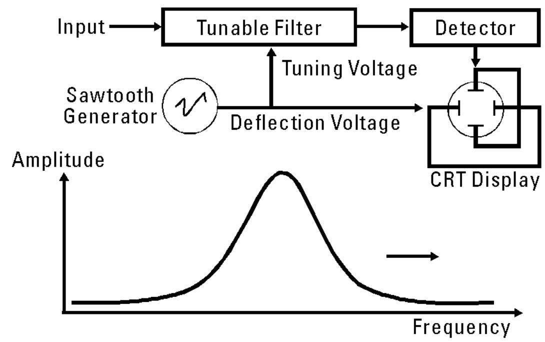
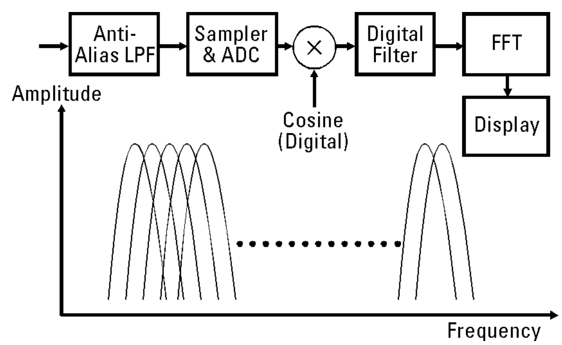


Figure 3.5-3
DSAs digitally simulate hundreds of parallel filters, providing both high speed and excellent frequency resolution. DSAs also provide time and phase displays not available on the other frequency domain analyzers.



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.

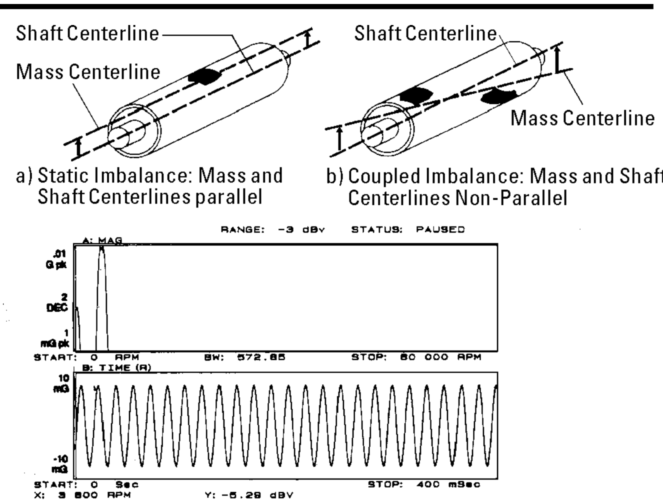


Figure 4.1-1 Imbalance, whether static or coupled, results in a spectral peak at a frequency of once per revolution (1 x).

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

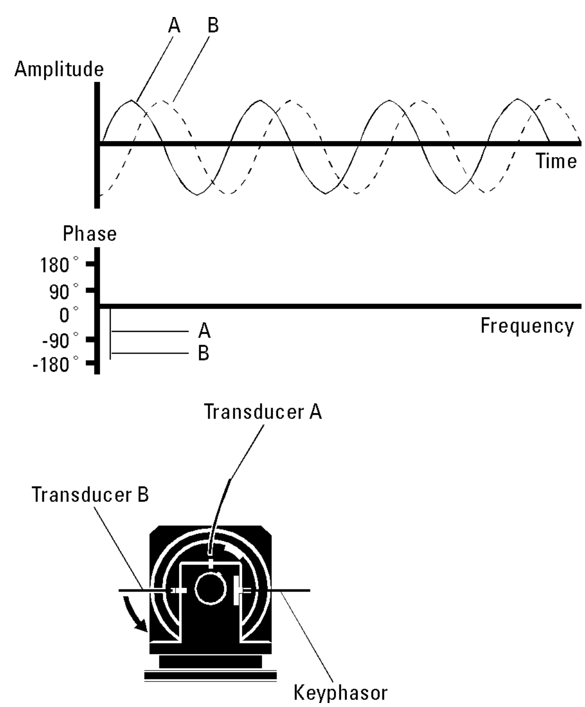
Rolling-element (anti-friction) bearings are the most common cause of small machinery failure, and overall vibration level changes are virtually undetectable in the early stages of deterioration. However, the unique vibration characteristics of rolling element bearing defects make vibration analysis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

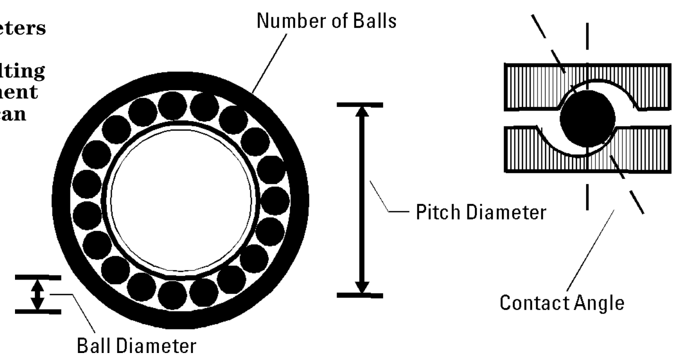


Table 4.2
Bearing Characteristic Frequencies.

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 vib-ration becomes more like random noise, and spec-tral peaks tend to disappear.

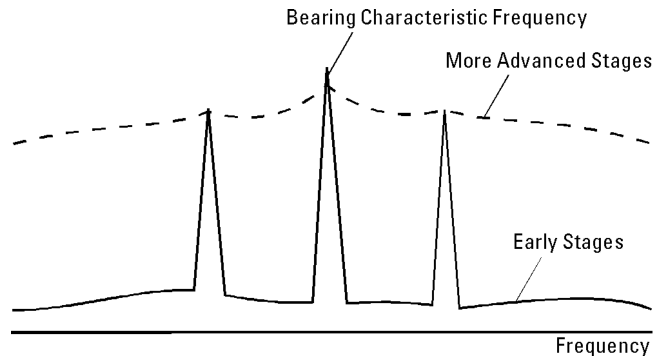
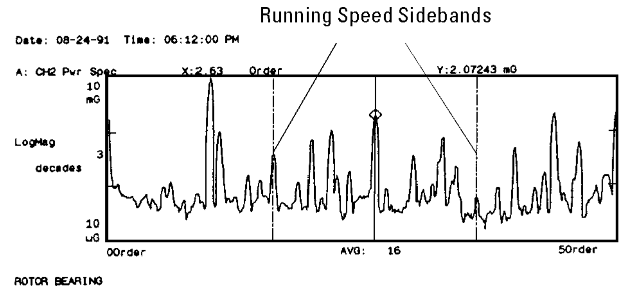


Figure 4.2-4
HP Instrument Basic print out of bearing data and characteristic frequencies with corresponding spectra. The result of a single defect on outer race is evident.




```
Ball diameter:   .156 inches
Pitch diameter   .625 inches
Contact angel: 0 degrees
Number of balls: 7
-------------
Ball pass--outer:   2.63 Orders
Ball pass--inner:   4.37 Orders
Ball spin:           .56 Orders
Fund. train:         .38 Orders
```

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

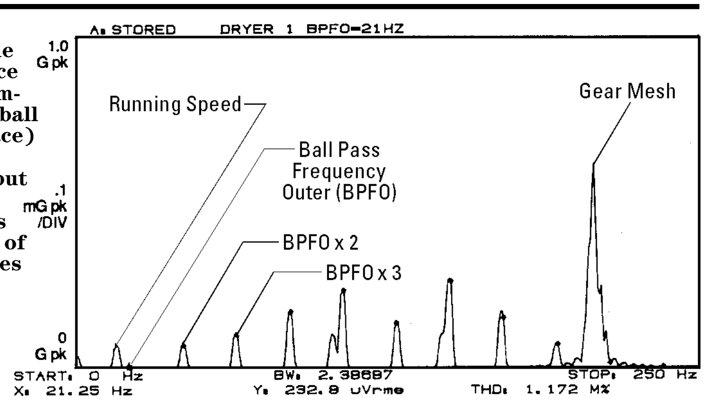
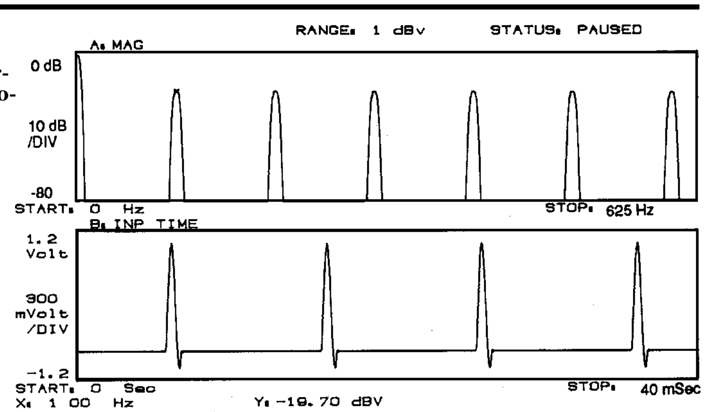


Figure 4.2-6
The impulsive nature of bear-ing defects pro-duces a large number of harmonics spaced at the characteristic frequency.



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.

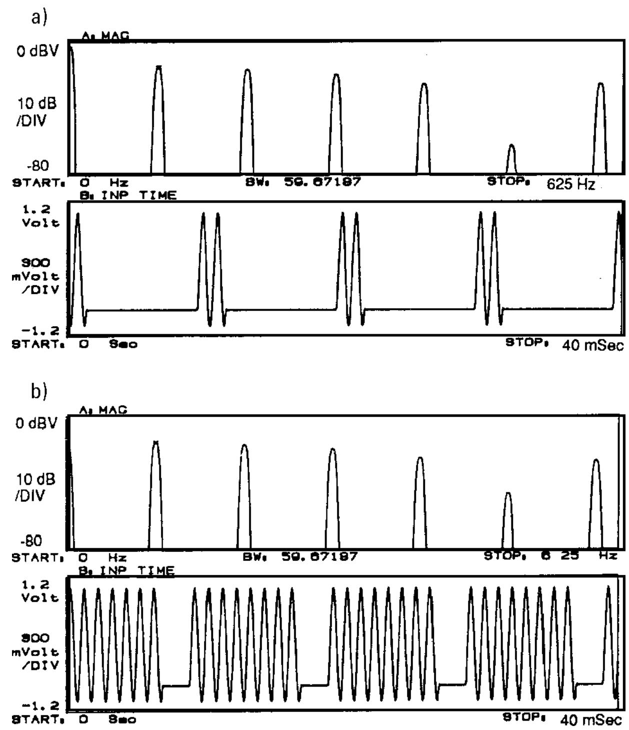
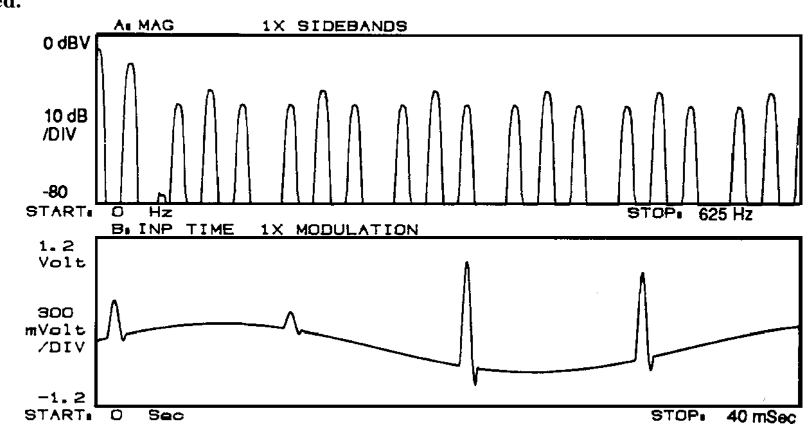


Figure 4.2-8 Bearing frequencies are almost always modulated by residual imbalance at running speed.



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.

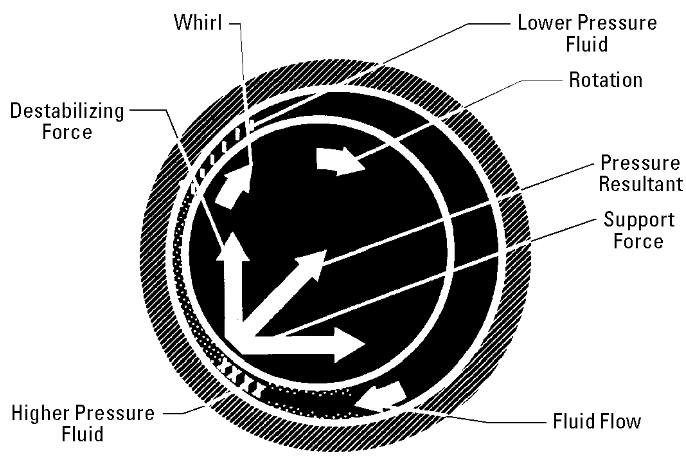
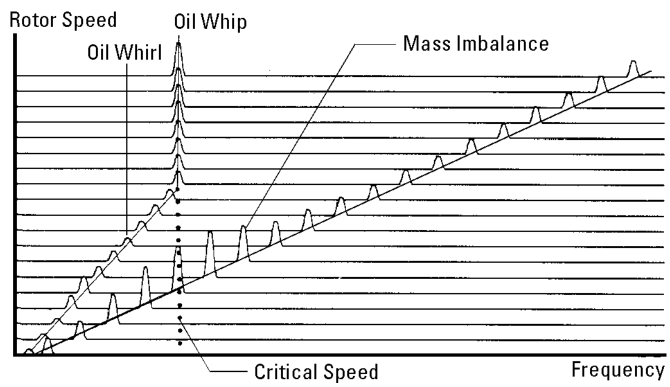


Figure 4.3-2 A spectral map showing oil whirl becoming oil whip instability as shaft speed reaches twice critical.



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 180° 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.

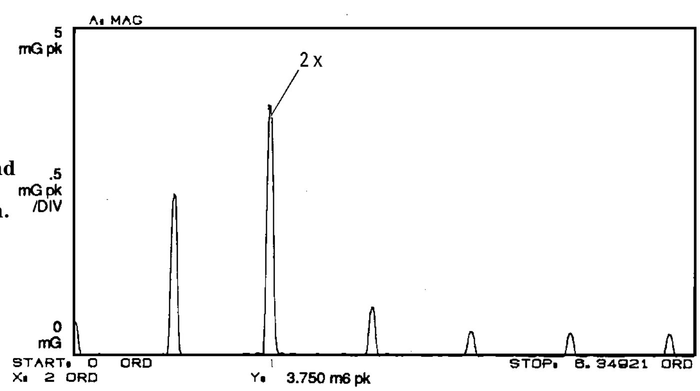
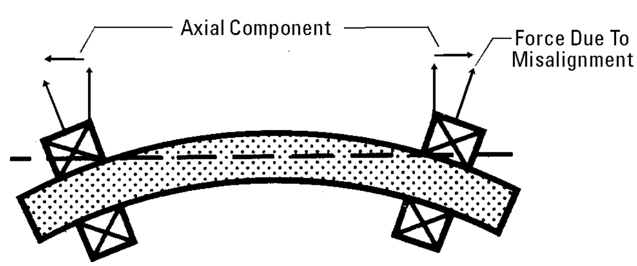


Figure 4.4-2 A bent or mis-aligned shaft results in a high level of axial vibration.



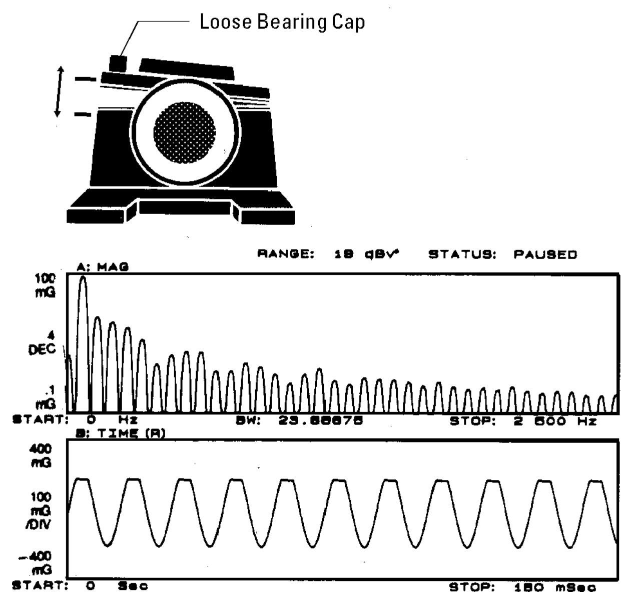
4.5 Mechancial Looseness

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

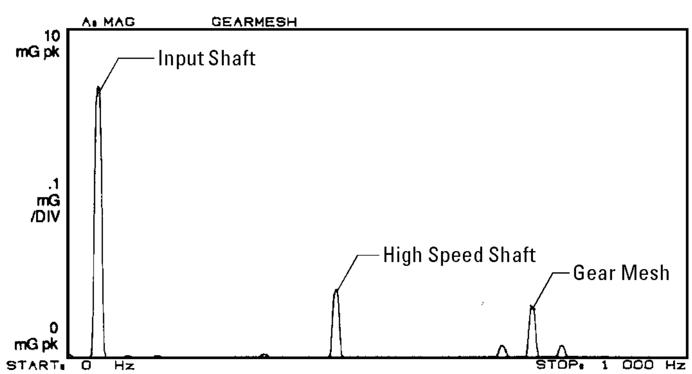
A. Gear Mesh: This is the frequency most commonly associated with gears, and is equal to the number of teeth multiplied by the rotational frequency. Figure 4.6-2 is a simulated vibration spectrum of a gearbox with a 15-tooth gear running at 3000 rpm (50 Hz). The gear-mesh frequency is 15 x 15 = 750 Hz. 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 broadband 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 sidebands 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.

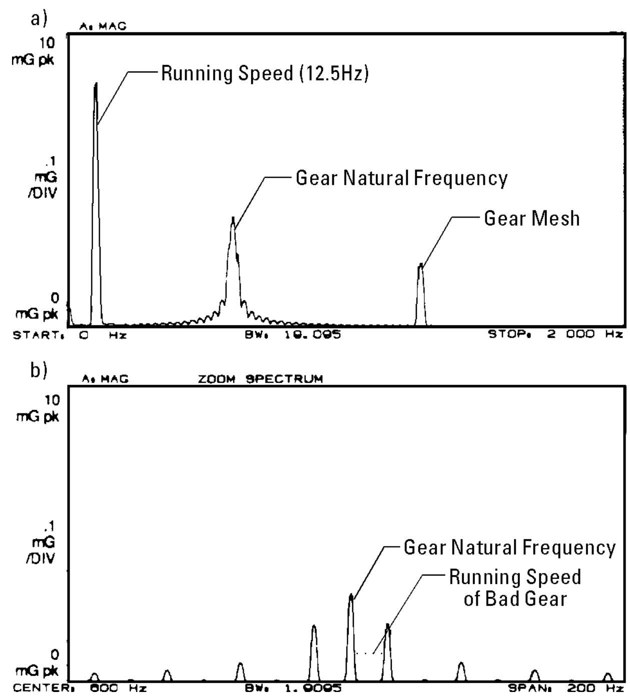
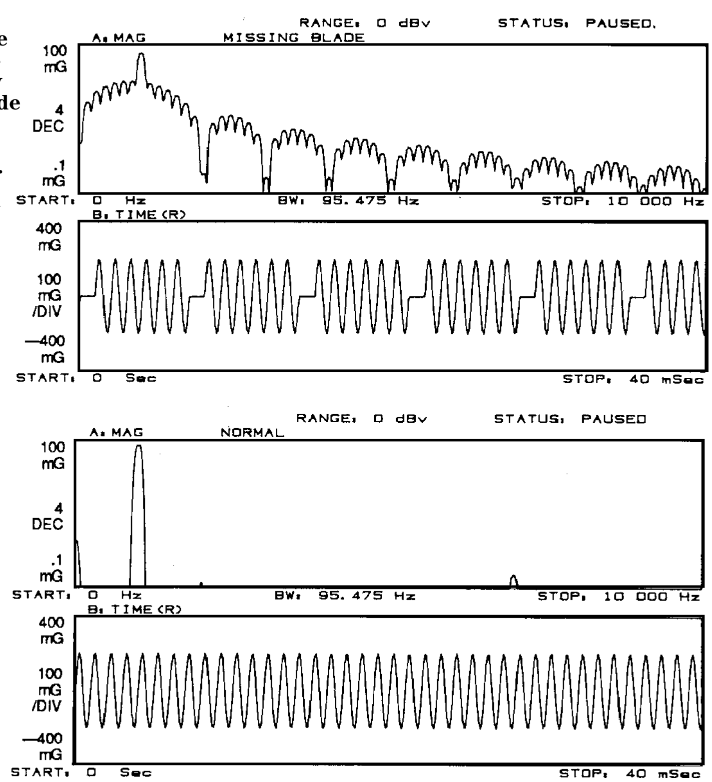


Figure 4.7 A space in the vibration signal caused by a missing blade results in a large number of harmonics. A missing blade usually also causes enough imbalance to significantly increase the 1 x level.



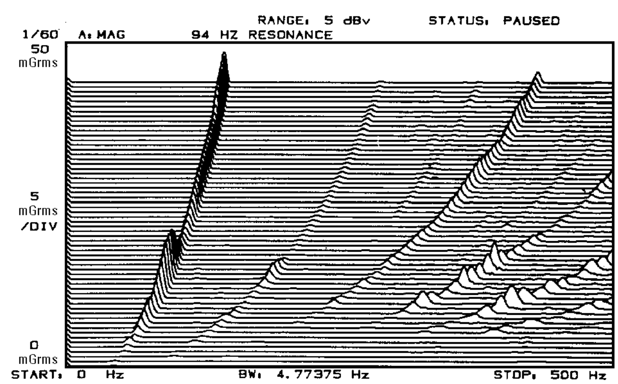
4.8 Resonance

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.