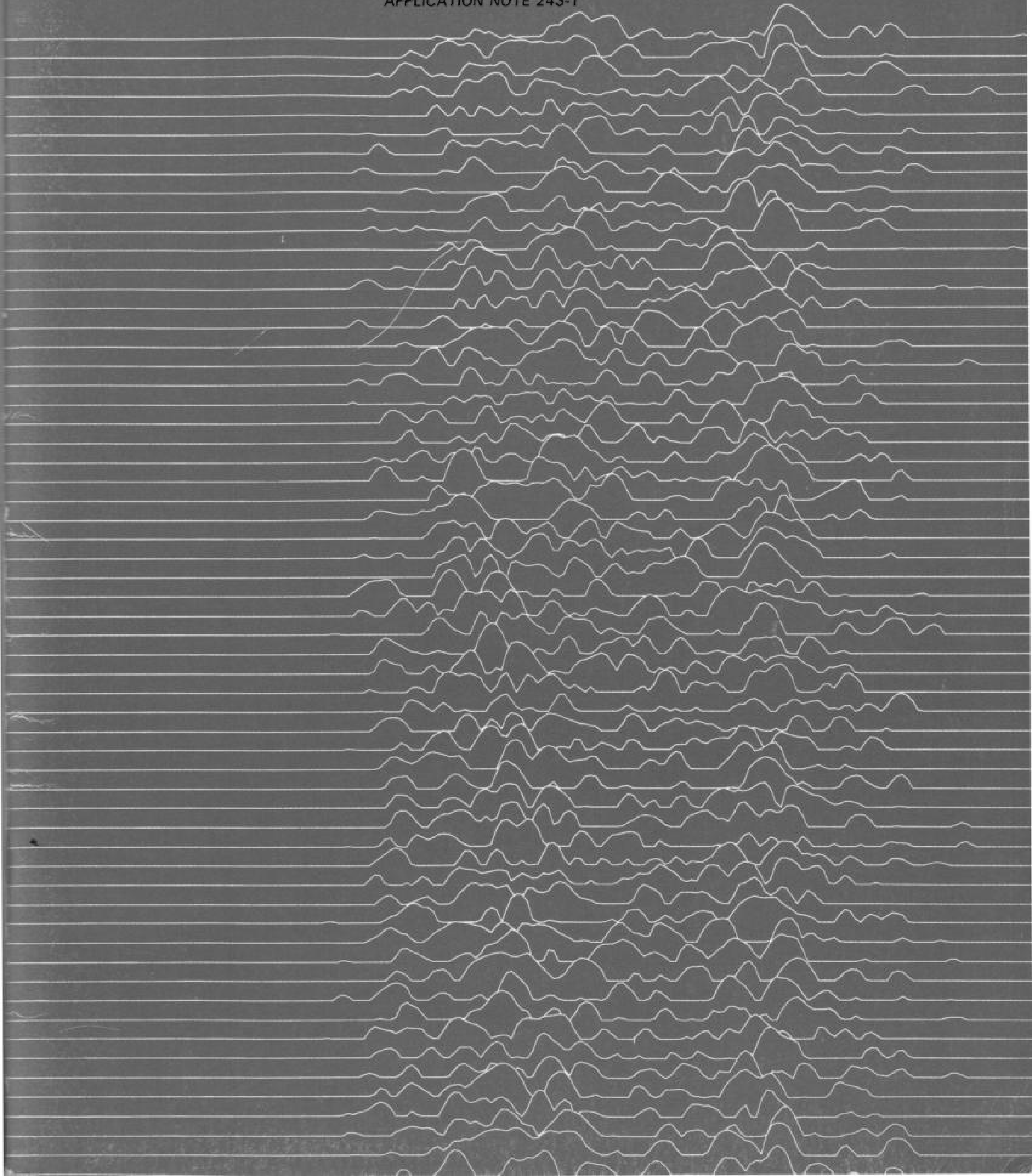
HEWLETT PACKARD

DYNAMIC SIGNAL ANALYZER APPLICATIONS
Effective Machinery Maintenance Using Vibration Analysis

APPLICATION NOTE 243-1



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

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CHAPTER 1:
INTRODUCTION

In the traditional approach to maintenance, scheduling repairs is difficult because the need for repair usually cannot be assessed without disassembling the machine. If a problem is serious enough to be readily apparent, damage has probably already occurred. Without a means to externally determine machine condition, scheduling is inaccurate: machines in perfect working order are taken out of service, while machines on the verge of failure are ignored. Modern technology provides a number of methods for externally determining the condition of machinery. The most effective of these is vibration analysis.

When a defect such as a bad bearing occurs in a machine, the result is an increase in vibration level. By regularly measuring this level, defects can be detected before they have a chance to cause extensive damage or failure. More importantly, the characteristics of the vibration are unique to the specific defect. By analyzing the vibration signal, the nature of the defect can often be determined. The key advantage of this approach is that need for repair — and the specific nature of any problems — can be assessed without disassembling the machine, or even taking it out of service.

The implementation of machinery vibration analysis has been made practical by the development of analysis instruments we call Dynamic Signal Analyzers (see Figure 1-1). Machinery vibration is a complex combination of signals caused by a variety of internal sources of vibration. The power of Dynamic Signal Analyzers (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.

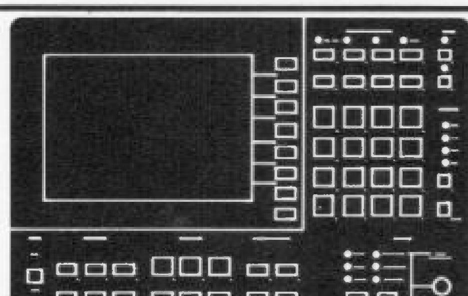


Figure 1-1
Dynamic Signal Analyzers (DSAs) are the ideal instrument for analyzing machinery vibration.

Dynamic Signal Analyzers can also display 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 spectral map format (Figure 1-4) adds a third dimension to vibration amplitude versus frequency displays. The third dimension is most often rpm, but can also be time or load — any variable that changes the vibration characteristics of the machine. DSAs can be connected to computers for automatic data storage and analysis, and are available in lightweight (less than 35 lbs) models that are ideal for machinery vibration analysis.

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 by the DSA, is covered. The techniques described provide insight into the condition of machinery that eliminates much of the guesswork from maintenance and troubleshooting. Primary benefits — and the focus of the note — are in the area of machinery maintenance; however, the techniques presented also have important benefits for machinery development and manufacturing.

In the next two sections, we will discuss the benefits of machinery maintenance programs based on vibration analysis, and the organizational philosophy of the note.

1.1 MACHINERY MAINTENANCE BASED ON VIBRATION ANALYSIS

The objective of maintenance is to keep machines running, especially those that are critical to plant production. Unexpected catastrophic failures cause both loss of production, and large repair bills. The classic maintenance strategy for avoiding such failures is to periodically disassemble critical machines for inspection and rebuilding. This process results in costly downtime, often to inspect machines that are in perfect working order. Because the process is expensive, it is only applied to a few critical machines, which usually account for a small percentage of maintenance expense. In addition, faulty reassembly or damage in transit from the repair shop sometimes results in a machine in worse condition than before the maintenance.

A more effective approach is to schedule repairs on the basis of machine condition, as determined by vibration analysis. This "predictive" maintenance strategy can be applied to all the major machines in a plant, and has proven its effectiveness in hundreds of maintenance organizations. In a typical program, overall vibration level is measured regularly with a vibration meter and compared to established severity limits (Section 5.1) or past readings. The vibration level of critical machinery is often monitored on a continuous basis, and compared against preset limits. If an excessive level is detected, a Dynamic Signal Analyzer is used to determine the severity and nature of the problem (see Figure 1.1).

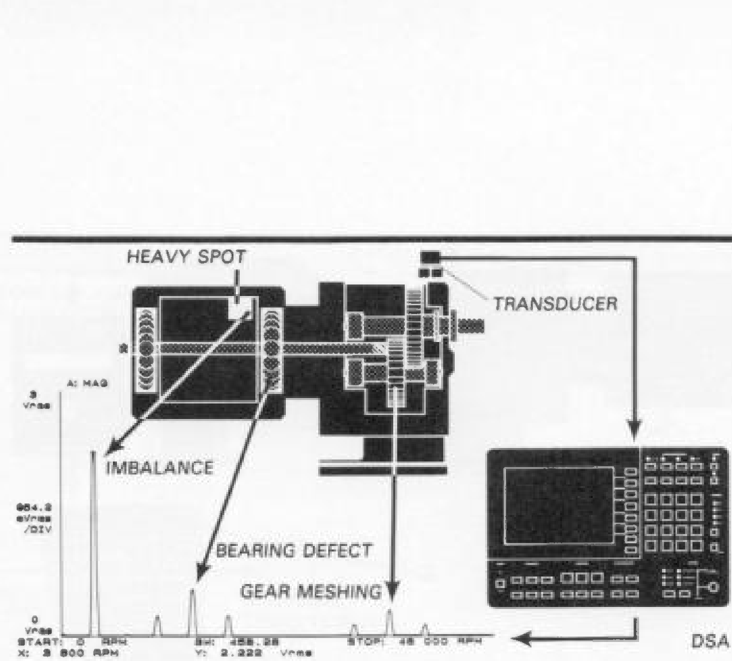


Figure 1-2
The individual components of vibration are shown in DSA displays of amplitude versus frequency.

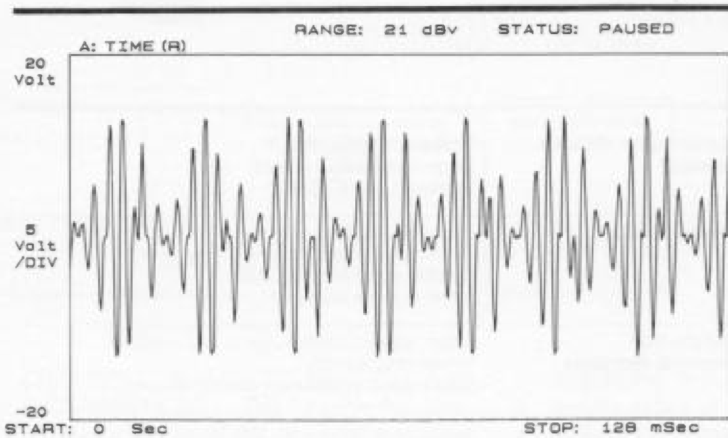


Figure 1-3
DSA displays of amplitude versus time are especially useful for analyzing impulsive vibration that is characteristic of gear and rolling element bearing defects.

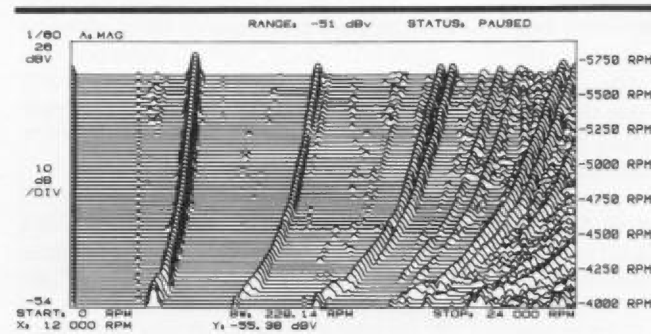


Figure 1-4
DSA map displays illustrate changes in vibration with rpm, load, or time. This map is a collection of vibration measurements made during a machine runup.

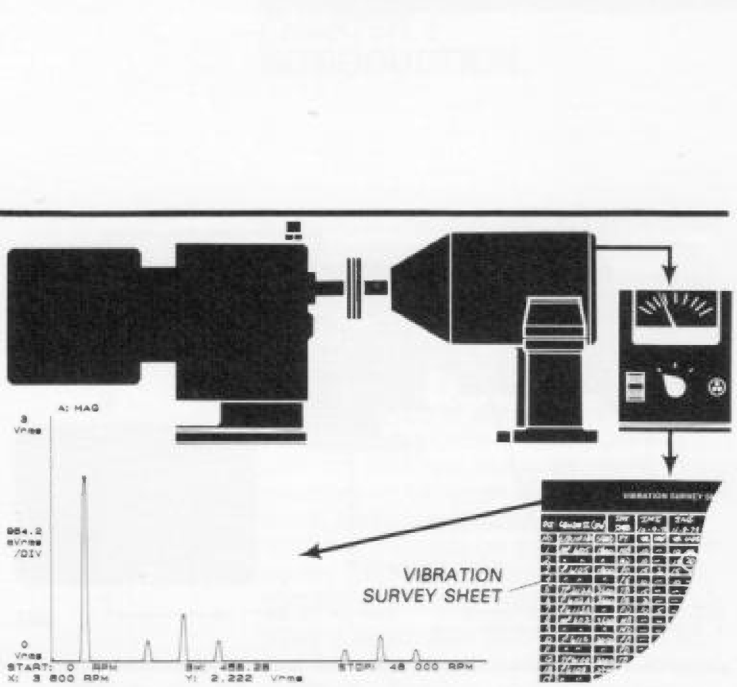


Figure 1.1
In a typical predictive maintenance program, vibration level is measured on a regular basis and compared with past readings. When a excessive level is detected, a DSA is used to determine the nature and severity of the problem.

Table 1.1
Economic Benefits of a Predictive Maintenance Program

Catastrophic Failures Avoided	• Increased production. • Large repair bills avoided. • Improved plant safety
Planned Repairs	• Improved repair quality. • Overtime expense reduced. • Smaller spare parts inventory (time to order the parts).
Insight Into Machine Problems	• Faster repairs (knowledge of the problem before disassembly). • Determine the cause of chronic failures.

Analysis is a critical part of the process for several reasons:

1. Overall vibration level can change with load and operating speed, thereby presenting a misleading picture of machine condition. Analysis of the vibration spectrum indicates whether or not a serious problem exists. This is an important step in avoiding unnecessary repairs.

2. Taking a machine out of service for repairs can rarely be done without some impact on production, so it is important to know just how severe the problem is. Analysis can help you decide, for example, whether the machine can be run until the next scheduled plant shut-down. Thus analysis is valuable in maximizing the effectiveness of a maintenance program.

3. Repair time is minimized because the nature of the problem is known. Technicians won't spend valuable time looking for the fault, and the necessary replacement parts can be ordered prior to disassembly.

The advantages of analysis make it appropriate in some cases to monitor the levels of individual components (rather than overall vibration level) to detect faults. Monitoring individual components also gives earlier warning of failure. This is an especially important consideration in highly loaded machines using rolling element bearings, whose condition can deteriorate rapidly.

The benefits of a predictive maintenance program based on vibration analysis are summarized in Table 1.1. Several authors, including Jackson [18] and Mitchell [3], provide details on establishing a program of predictive maintenance.

1.2 USING THIS APPLICATION NOTE

The application note is organized around the four key steps in the analysis process shown in Figure 1.2: (1) converting the vibration to an electrical signal, (2) reducing it to its components, (3) correlating those components with machine defects, and (4) implementing necessary repairs and documenting 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, are discussed below.

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 three 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.

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 transducers, and a procedure for selecting the type that best fits the application. The final section of the chapter provides transducer installation guidelines.

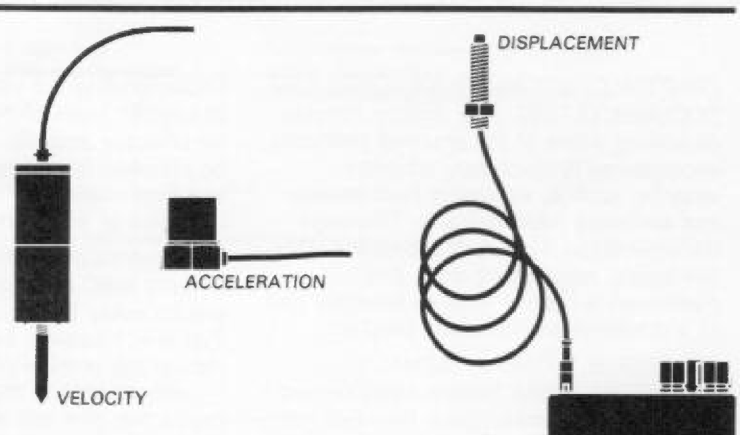


Figure 2-1
Three types of transducers are commonly used to convert machinery vibration to an electrical signal.

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 defects, not the forces themselves. The mechanical impedances of the machine shaft and housing determine how they will respond to vibration forces, and can alter significantly the characteristics of the signal we measure.

(c) NATURAL FREQUENCIES. When a structure is excited by an impact, it will vibrate at one or more natural frequencies. These frequencies are important because they limit the operating frequency range of velocity and acceleration transducers, and because they can cause large changes in vibration response with changes in rpm.

Vibration 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 a rotating 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.

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).* 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.

*Note: This applies to shafts that do not bend in operation (i.e. rigid shafts). Flexible shafts respond somewhat differently to imbalance forces (see Figure 3.4-5).

Velocity, for example, is offset from displacement by 90° (one complete cycle is 360°). 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 varies with rpm — an important consideration in transducer selection. Velocity increases in direct proportion to speed, while acceleration increases with the square of speed. This variation with speed, 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 amplitude and f is the rotor frequency of rotation.

$$\text{DISPLACEMENT} = A \cdot \sin(2\pi ft) \quad (1)$$

$$\text{VELOCITY} = 2\pi f A \cdot \cos(2\pi ft) \quad (2)$$

$$\text{ACCELERATION} = -(2\pi f)^2 A \cdot \sin(2\pi ft) \quad (3)$$

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 variation in amplitude with 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 a good 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. 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 monitoring work. 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 the 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 blade analysis.

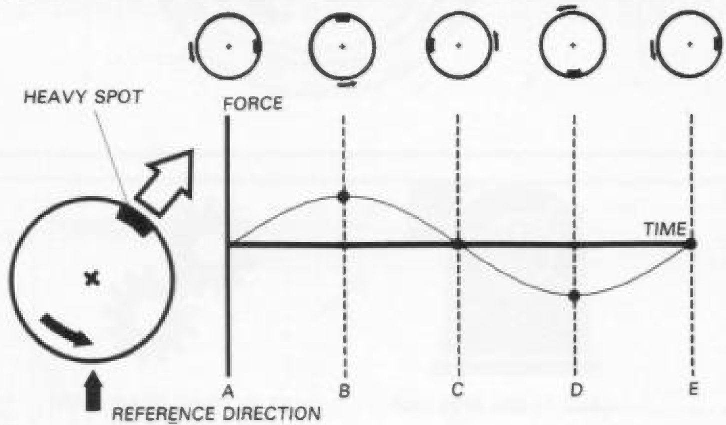


Figure 2.1-1
A heavy spot on a machine rotor results in a rotating force vector that appears sinusoidal from a fixed reference.

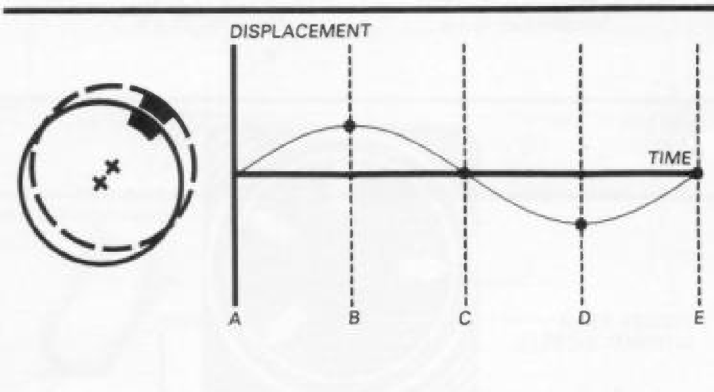


Figure 2.1-2
The imbalance force produces a vibration whose displacement has approximately the same waveform as the force itself.

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

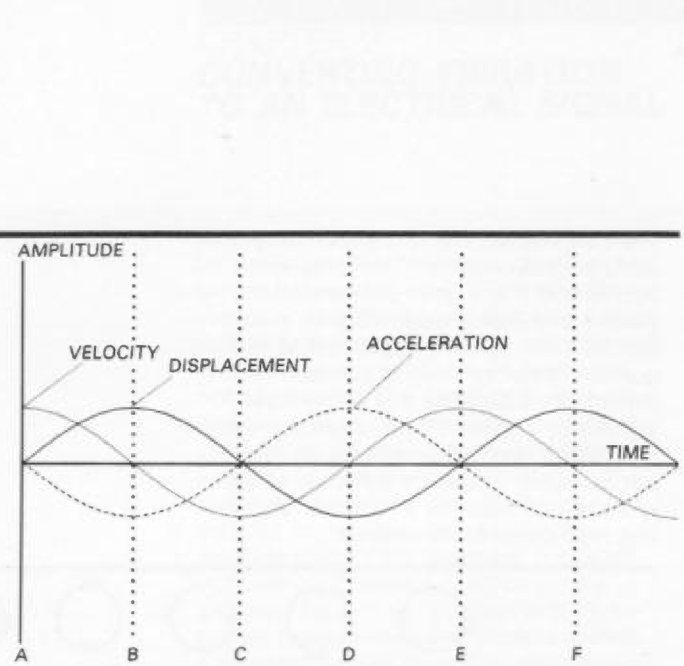


Figure 2.1-4
Two cases which illustrate the variation of vibration parameters with machine speed.

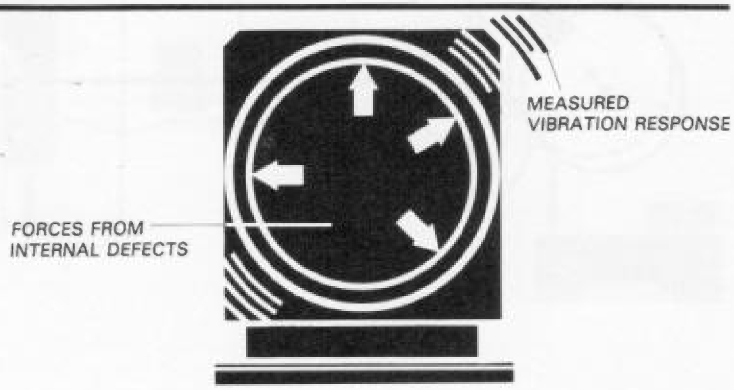
CASE 1: 600 RPM FAN

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

CASE 2: 15kHz GEAR MESH

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

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



Mechanical Impedance

A key point illustrated by Figure 2.1-2 is that we are measuring the *response* of the machine to a vibration force, not the force itself. 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 the selection and installation of transducers.

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

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

Natural Frequencies

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

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 (or "critical") speeds, and (3) resonances 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 are classified as flexible, and are discussed briefly in Section 3.4. Natural frequency limits on the useful frequency range 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. If you think of piano wires

$$\text{Natural frequency} = \sqrt{\frac{k}{m}}$$

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.

2.2 TRANSDUCERS

In this section, each of the three types of transducers shown in Figure 2.0 will be described. We will discuss how each one works, its important characteristics, and the most common applications.

Displacement Transducers

Noncontacting displacement transducers (also known as proximity probes), like the one in Figure 2.2-1, are used to measure relative shaft motion directly. A high frequency oscillator 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 signal proportional to displacement. This is illustrated in Figure 2.2-2.

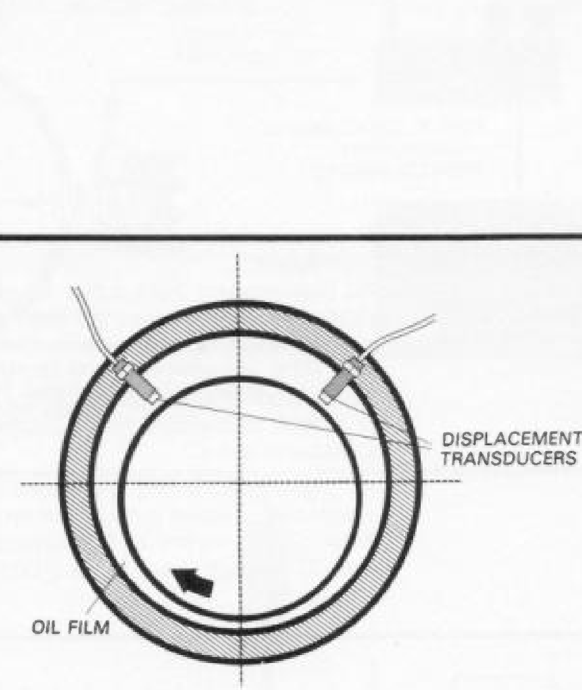


Figure 2.1-6
A relatively light shaft turning in fluid-film bearings transmits little vibration to the machine housing. Its vibration must be measured directly with a displacement transducer.

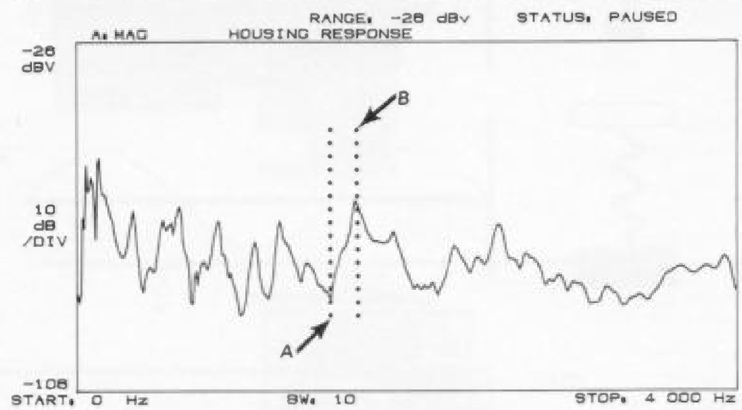


Figure 2.1-7
A plot of vibration response versus frequency for a machine housing shows how measured vibration level can change with rpm. A defect force at frequency B produces a much larger vibration response than the same force level at frequency A.

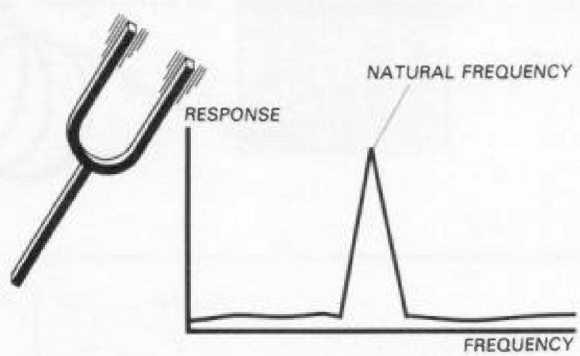


Figure 2.1-8
When excited by an impact, a tuning fork vibrates at its natural frequency.

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 a displacement transducer alone.

(b) Signal conditioning is included in the electronics. Typical outputs are on the order of 200 mV/mil or 8 mV/micron (1 mil is 0.001 inches; 1 micron is 0.001 millimeter).

(c) Shaft surface scratches, out-of-roundness, and variation in electrical properties all produce a signal error. Surface treatment and run-out subtraction can be used to solve these problems [10, 11].

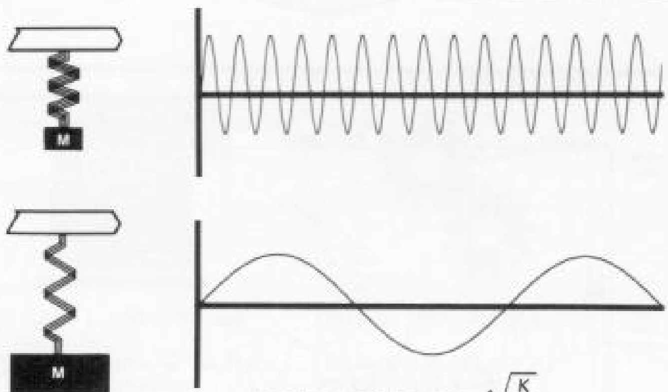
(d) Installation is sometimes difficult, often requiring that a hole be drilled in the machine.

(e) The output voltage contains a dc offset of approximately 10 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 dc. The practical disadvantage of ac coupling is reduced instrument response below 1 Hz (60 rpm).

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

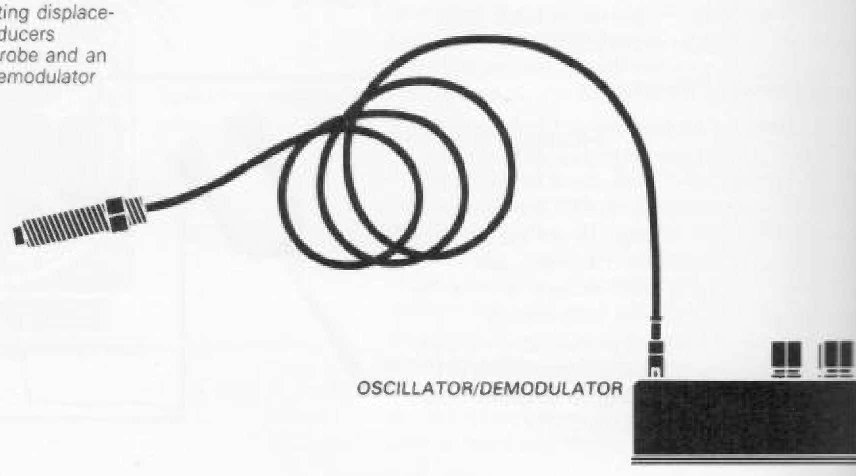
Displacement transducers are also commonly used as tachometer signals by detecting the passage of a keyway (see Section 3.4).

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



$$\text{NATURAL FREQUENCY} = \sqrt{\frac{K}{M}}$$

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



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 size 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.

Key characteristics of velocity transducers:

(a) The frequency range of approximately 10 Hz to 1000 Hz is ideal for most machinery work. Major applications outside this frequency range are gears, and blading on high-speed turbomachinery. The lower frequency limit can be extended slightly by compensating for the roll-off. (Note that phase measurements will be in error because of the 180 degree phase shift at the natural frequency.)

(b) Installation of velocity transducers is relatively noncritical, and extension probes and magnetic mounts work well. In addition, no signal conditioning is required.

(c) Because they are an electro-mechanical device with moving parts, velocity transducers can change calibration over time and wear out. High temperature operation can also cause changes in calibration.

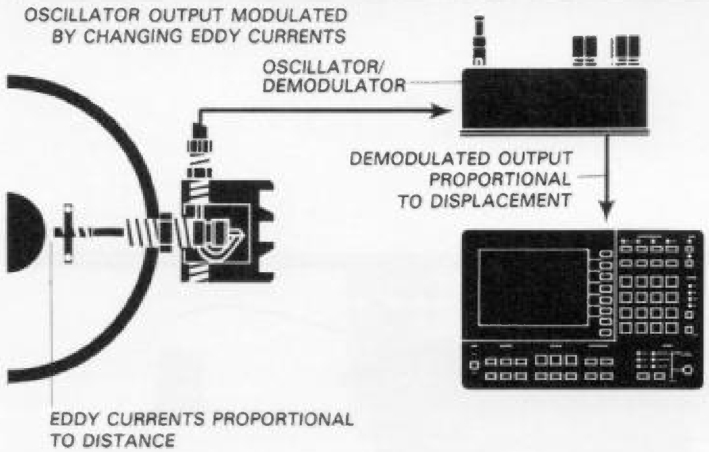


Figure 2.2-2
Schematic diagram of a typical noncontacting displacement transducer installation.

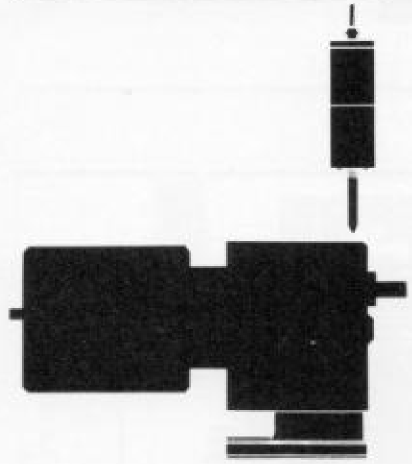


Figure 2.2-3
A typical velocity transducer with extension probe installed.

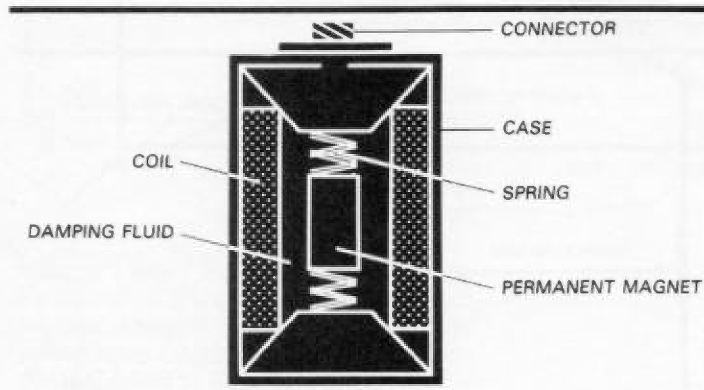


Figure 2.2-4
Velocity transducer output is a current generated in the coil as its moves through the field of the stationary magnet.

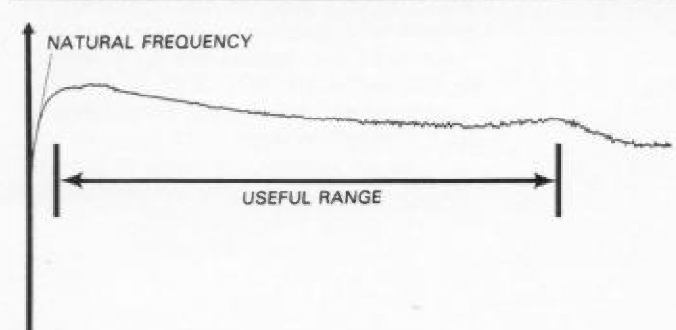


Figure 2.2-5
Frequency response of a typical velocity transducer. Note that the natural frequency of the magnet-spring-damper system is below the operating range.

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

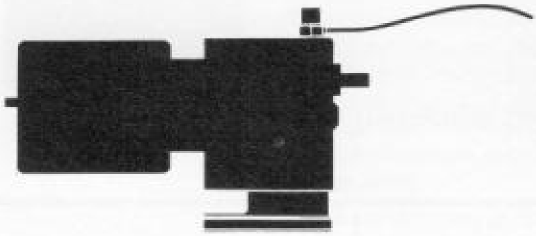


Figure 2.2-7
The output current of an accelerometer is produced by the force of the accelerating mass squeezing the piezoelectric crystal stack. The force — and thus the output current — is proportional to acceleration.

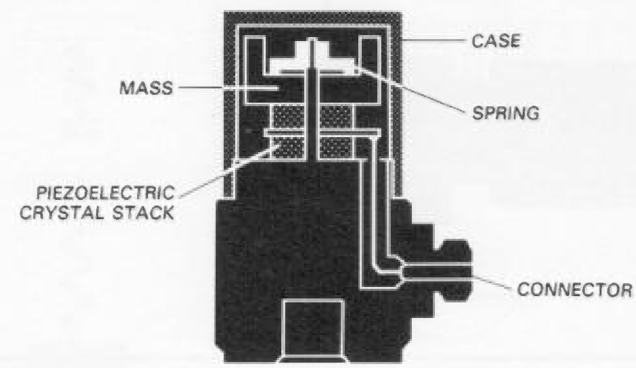
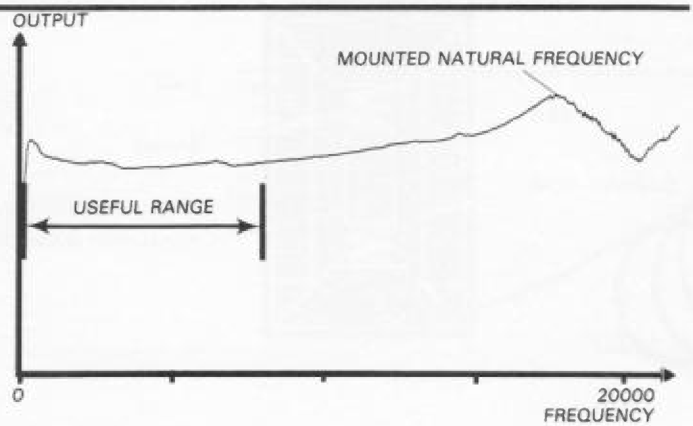


Figure 2.2-8
Accelerometer high frequency response is limited by the natural frequency of the spring-mass system.



Velocity transducers were once the standard for vibration monitoring work on machines other than turbomachinery, and they are especially good for hand-held measurements. Their popularity has declined somewhat because accelerometers are typically more rugged, and offer wider frequency response.

Accelerometers

Accelerometers are a popular transducer for general vibration analysis because they are accurate and rugged, and available for a wide range of applications. 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 current 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 (it is below the operating range in velocity transducers). Operation should be limited to about 30% 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 move the operating range down in frequency.

Accelerometer output is a low-level, high-impedance signal that requires signal conditioning. The traditional method has been to use a charge amplifier, as shown in Figure 2.2-9(a). However, accelerometers are available with built-in signal conditioning electronics that require only simple current-source supply. These accelerometers, sometimes referred to as ICP (for Integrated Circuit Piezoelectric), can be directly connected to a compatible DSA (Figure 2.2-9(b)). Another advantage of the the ICP accelerometer is that expensive low-noise cable required to connect traditional accelerometers to the charge amplifier is not required. This is especially important when long cables are involved.

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 defects when overall level is measured (this is not a problem with the wide dynamic range of DSAs).

(b) Accelerometers require signal conditioning, however as noted above, the development of the ICP accelerometer has eliminated this disadvantage.

(c) The low frequency response of piezoelectric accelerometers is limited to approximately 5 Hz. Piezoresistive and cantilever-beam accelerometers are available that respond down to dc, although it is worth remembering that acceleration level is low at very low frequencies.

(d) Accelerometers are very sensitive to mounting, and should never be hand-held. They should be securely attached with a threaded stud, high strength magnet, or industrial adhesive. The mounting surface should be flat and smooth — preferably machined.

2.3 SELECTING THE RIGHT TRANSDUCER

Selecting the right transducer for an application is a straightforward 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.

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 transducer. If the parameter is a quantity other than a clearance or relative displacement, go on to the 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 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.

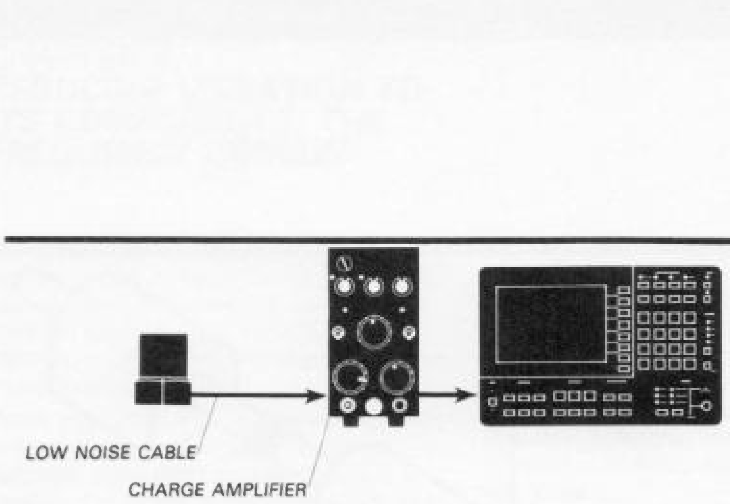


Figure 2.2-9a
Traditional accelerometers require an external charge amplifier for signal conditioning.

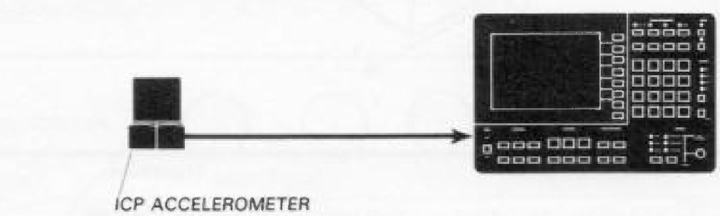


Figure 2.2-9b
Integrated Circuit Piezoelectric (ICP) accelerometers, with built-in signal conditioning, can be connected directly to a compatible DSA.

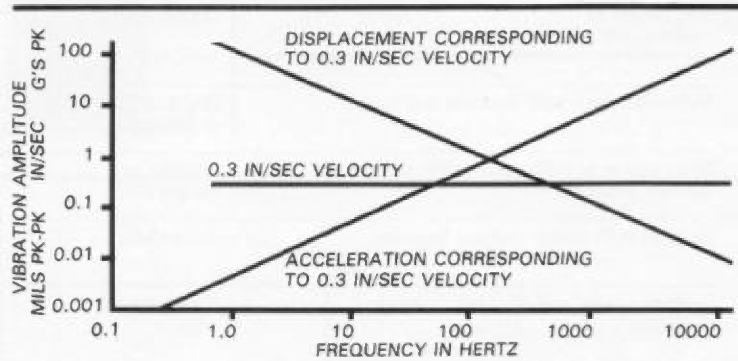


Figure 2.3
A vibration nomograph shows how the levels of displacement and acceleration change with frequency, relative to the level of velocity. Note that acceleration response is very low at 1 Hz (less than 100μV with a 10 mV/g accelerometer).

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 range of 10 Hz to 1000 Hz, either velocity or acceleration transducers can be used. The vibration nomograph of Figure 2.3 can be used to determine whether an accelerometer will produce sufficient output.

Figure 2.4
Transducer locations referenced in Table 2.4.

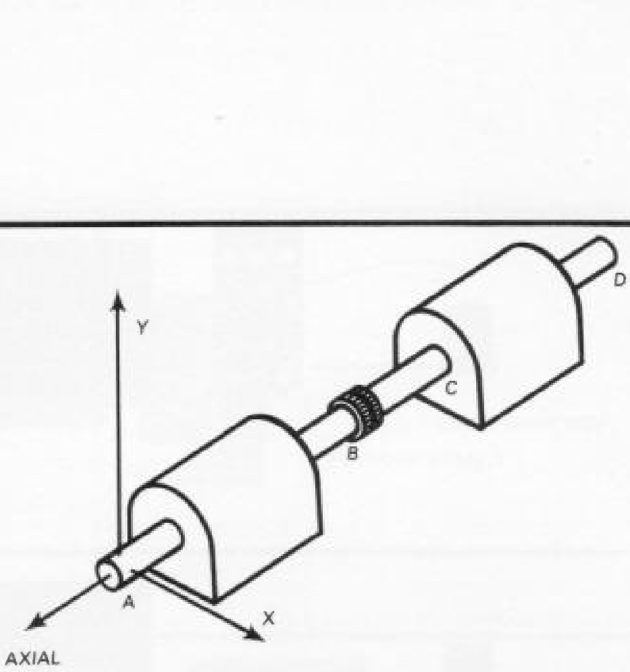


Table 2.4
Transducer Application Summary

Machine Description	Transducer	Location
Steam turbine / large pump or compressor with fluid film bearings	Displacement	Radial horizontal and vertical at A,B,C,D. 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.
Gearbox 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.

In general, use a velocity transducer if:

(a) Overall level is being measured to detect defects.

(b) The transducer will be hand-held.

(c) The machinery being analyzed is low-speed (i.e. less than 1200 rpm).

Use an accelerometer if:

(a) High frequency (above 1000 Hz) blading or gear defects are being analyzed.

(b) Structural response is being measured.

(c) Long transducer life (more than 2 years) is required.

(d) High temperatures are encountered (although high temperature velocity transducers are available). Note that the operating temperature range of ICP accelerometers is usually limited to 250° F (120° C).

After the type of transducer has been determined, consult transducer manufacturers for specific model recommendations.

2.4 INSTALLATION GUIDELINES

After the transducer has been selected, it must be properly installed for best results. Figure 2.4 is an example machine combination that is used for the application summary of Table 2.4. The machines 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 how critical the machine is to the process, and how expensive it is to repair or replace. Table 2.4 is intended to show typical applications, and should be used only as a guide.

Proper mounting of the transducer to the machine is also critical, especially with displacement and acceleration transducers, and manufacturer's recommendations should be followed closely. One particular caution: the transducer should never be mounted to a sheet metal cover, since resonances may easily be in the operating speed range.

CHAPTER 3:
REDUCING VIBRATION TO ITS COMPONENTS: THE FREQUENCY DOMAIN

he signal obtained from a machinery vibration transducer is a complex combination of responses to multiple internal (and sometimes 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 correlation 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. 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 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 of 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 cycle repeats once per revolution. 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 perspective that is much better suited to analyzing these components is the frequency domain.

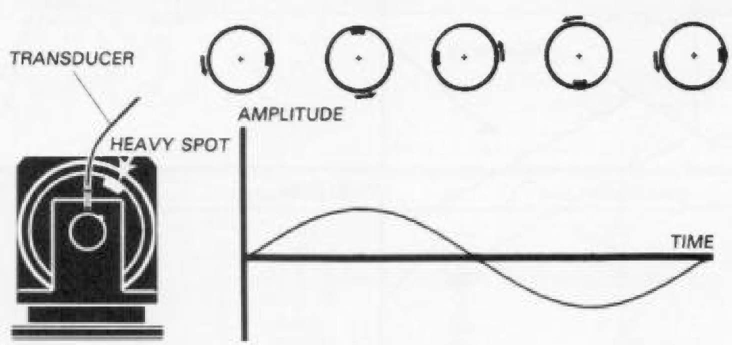


Figure 3.1-1. A time domain representation of vibration due to rotor imbalance.

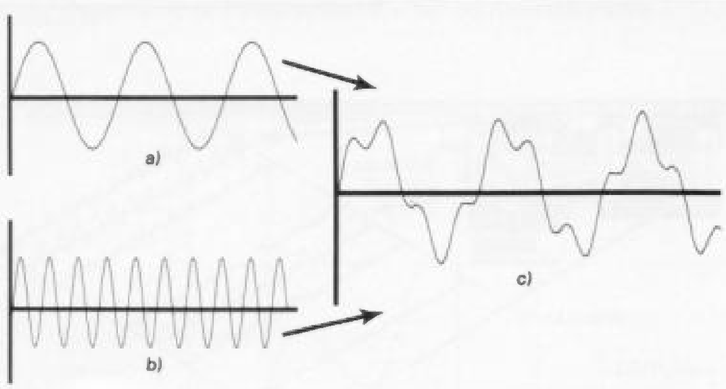


Figure 3.1-2
Waveform (c) is the combination of signals (a) and (b). The nature of these components is hidden in the time domain view of their sum.

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 recognizable.

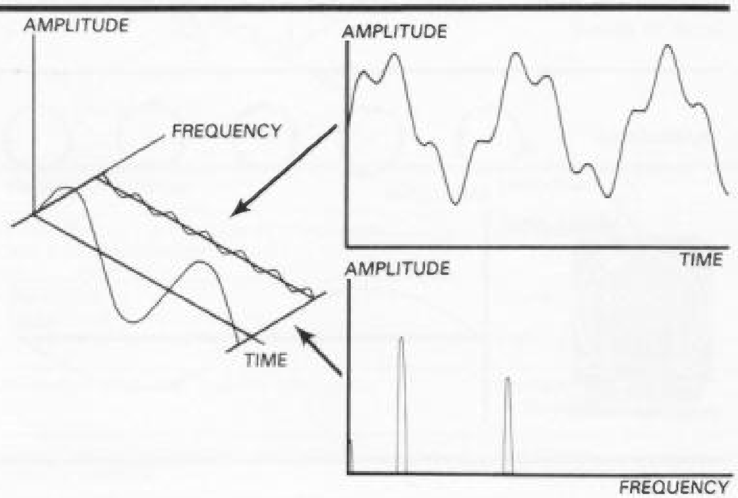


Figure 3.2-1
The relationship between the time and frequency domains.

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 bearing defect force with the imbalance force. These signals are called *sidebands*, and occur often in machinery vibration. They are spaced at in-

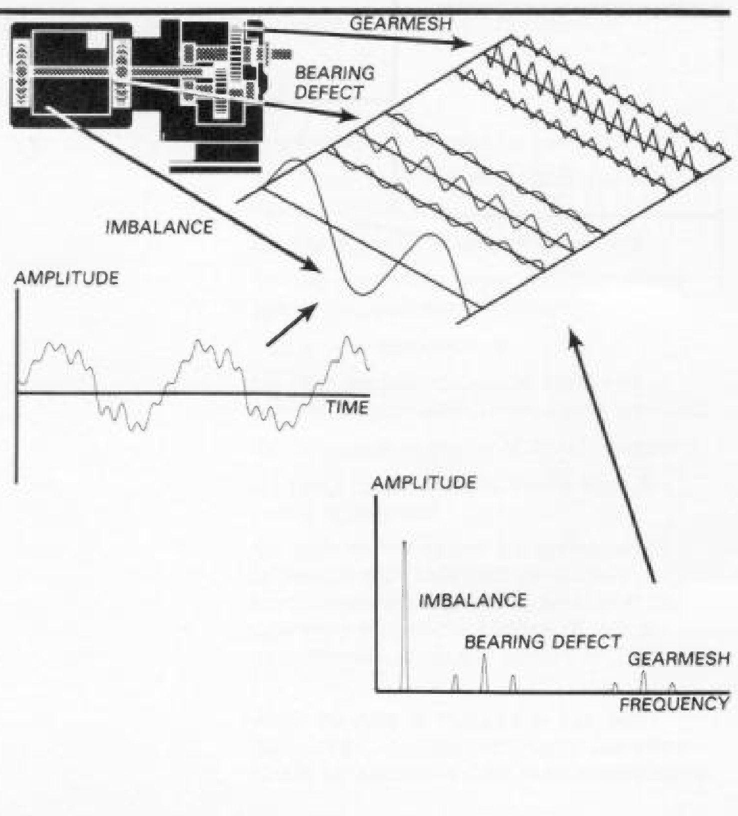


Figure 3.2-2
Machinery vibration viewed in the time and frequency domains.

crements of plus and minus 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 we have 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.

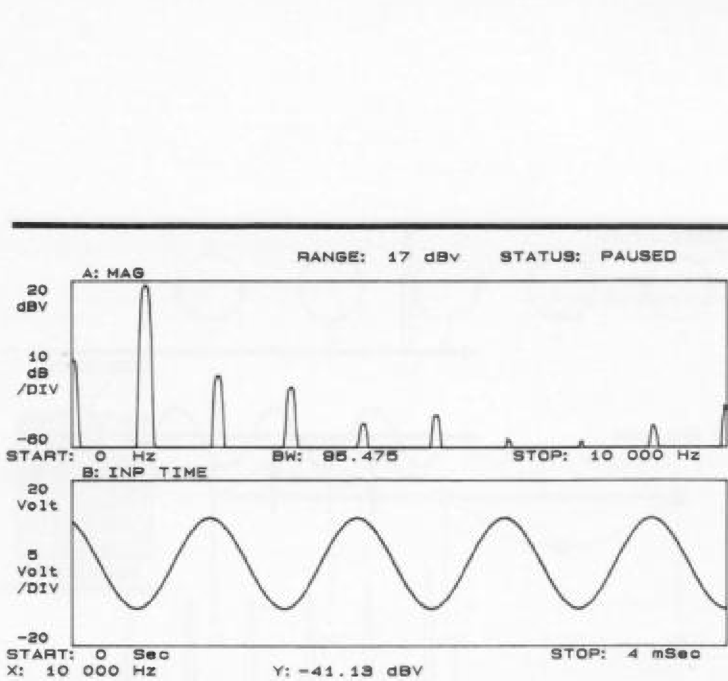


Figure 3.2-3
Small signals that are hidden in the time domain are readily apparent in the frequency domain. By using a logarithmic (dB) amplitude scale, signals which vary in level by a factor of over 1000 can be displayed simultaneously.

The Need for Decibels (dB)

A major advantage of the frequency domain is the ability to resolve small signals in the presence of large ones. To utilize this capability, we must use a logarithmic (dB) amplitude scale.

Suppose we wish to measure a defect component that is 0.1% (1/1000) of the residual imbalance component. If we set the fundamental to full scale on a 4 inch (10 centimeter) screen, the smaller component would be only 0.004 inch (0.1 millimeter) tall. Such a signal would be difficult to see, let alone accurately measure. Yet many analyzers are available with the ability to measure signals even smaller than this.

Since we want to be able to see all the components easily at the same time, the only answer is to change our amplitude scale. A logarithmic scale would compress our large signal amplitude and expand the small ones, allowing all components to be displayed at the same time.

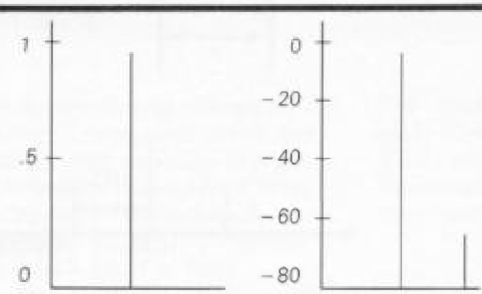
Alexander Graham Bell discovered that the human ear responded logarithmically to power difference and invented a unit, the Bel, to help him measure the ability of people to hear. One tenth of a Bel, the deciBel (dB) is the most common unit used in the frequency domain today.

A table of the relationship between volts, power and dB is given in Figure 3.2-4. From the table, we can see that out 0.1% component is 60 dB below the residual imbalance component. If we had an 80 dB display as in Figure 3.2-5, the 0.1% defect component would occupy ¼ of the screen, not 1/1000 as in a linear display.

dB	POWER RATIO	dB	VOLTAGE RATIO
+20	100	+40	100
+10	10	+20	10
+3	2	+6	2
0	1	0	1
-3	½	-6	1/2
-10	1/10	-20	1/10
-20	1/100	-40	1/100

dB = 10 LOG (POWER RATIO) = 20 LOG (VOLTAGE RATIO)

Figure 3.2-4
The relationship between decibels, power and voltage.

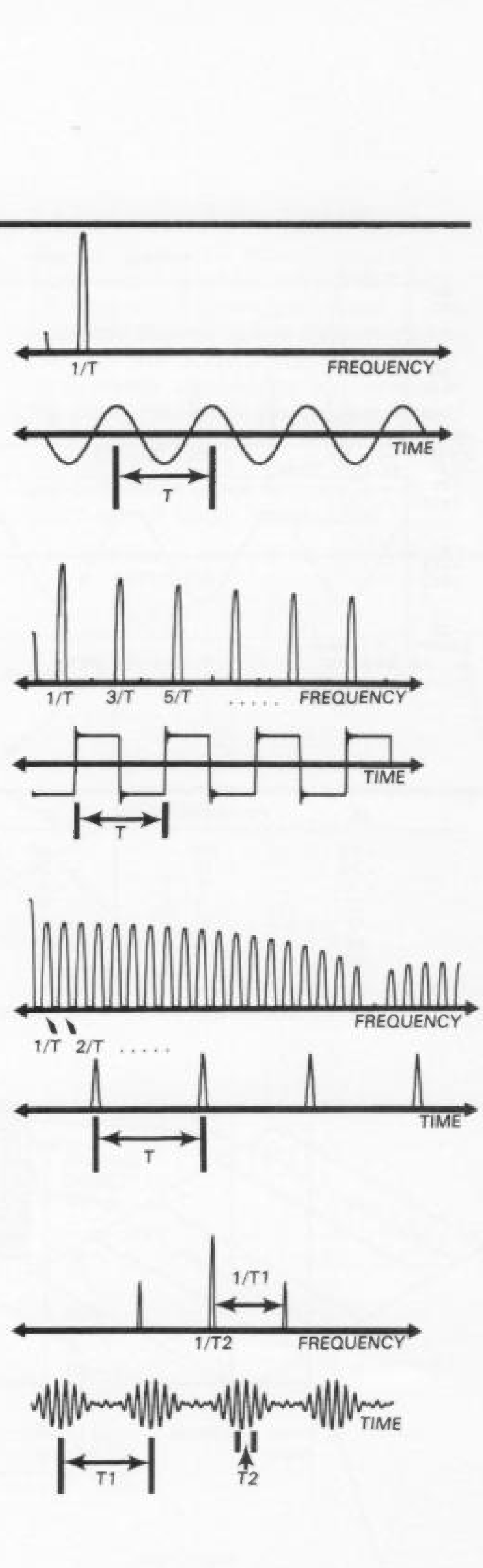


a) Linear Amplitude Scale

b) Logarithmic Amplitude Scale

Figure 3.2-5
Small signals can be measured with a logarithmic amplitude scale.

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



Early Warning of Defects

As we pointed out in the introduction, many maintenance programs use DSAs to monitor machinery vibration in the frequency domain. This is because the low level vibration produced during the 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 fault 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 times 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. In this display format, amplitude is expressed in decibels, or *dB*. (See the boxed explanation of dB and the need for logarithmic scales.)

While most people prefer the more natural feel of a linear (i.e. non-logarithmic) display, logarithmic displays are an aid to earliest detection of defects such as those characteristic of rolling element bearings. Example spectra in this note will use both logarithmic and linear amplitude scales.

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/T Hz.

(b) A square wave, which is much like the truncated signal produced by mounting or bearing cap looseness, is made up of an infinite number of *odd harmonics*. Harmonics are components which occur at frequency multiples of a fundamental frequency. In machinery analysis, we usually refer to harmonics as orders of the fundamental running speed. Because square wave type signals from machinery are not ideal, their spectra often contain both odd and even harmonics.

(c) Bearing and gear defects usually produce impulsive signals that are typified by harmonics in the frequency domain. These harmonics are spaced at the repetition rate of the impulse.

(d) Finally, as mentioned above, many defects are modulated by 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.

3.3 SPECTRAL MAPS

The vibration characteristics of a machine depend on its dynamics and the nature of the defect. 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 a defect may be obtained from observing the change in vibration with speed. Spectral maps, such as the one in Figure 3.3, are three dimensional displays that effectively show variation in the vibration spectrum with machine speed.

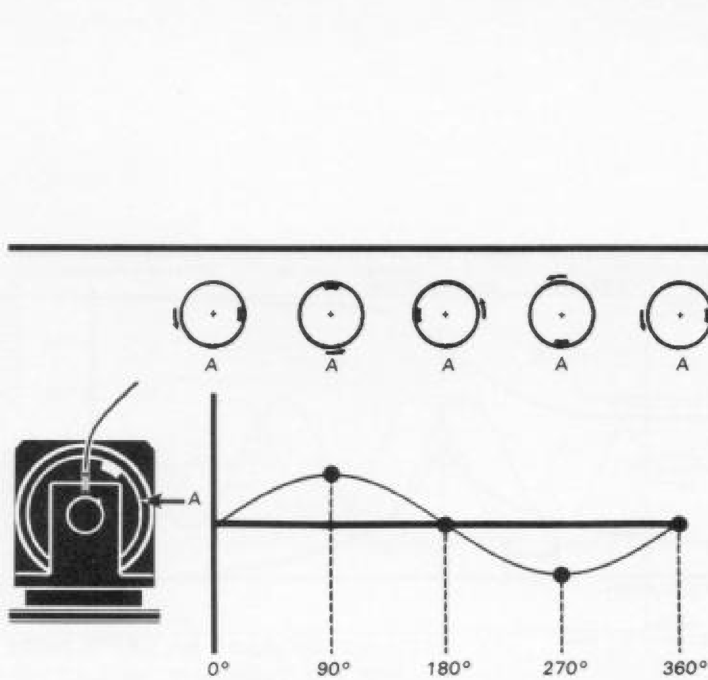


Figure 3.4-1
The phase of the imbalance signal corresponds to the direction of the displacement. One 360° rotation of the rotor corresponds to one 360° cycle of the signal.

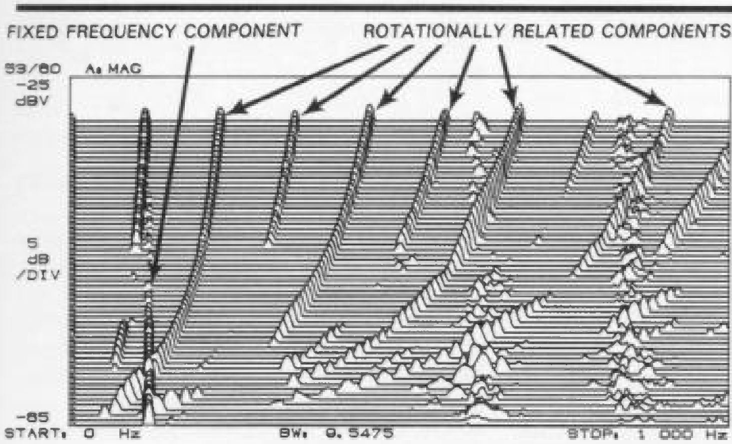


Figure 3.3
Spectral maps show variation in the vibration spectrum with time or rpm.

Spectral maps (also known as cascade plots) usually consist of a series of vibration spectra measured at different speeds. A variety of other parameters, including time, load, and temperature, are also good third dimensions for maps. The most common method for mapping the variation in vibration with rpm is to measure successive spectra while the machine is coasting down or running up in speed. The map in Figure 3.3 was made on a Hewlett-Packard DSA with this capability built in.

In addition to showing how vibration changes with speed, spectral maps quickly indicate which components are related to rotational speed. These components will move across the map as speed changes, while fixed frequency components move straight up the map. This feature is especially useful in recognizing machine resonances, 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 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

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

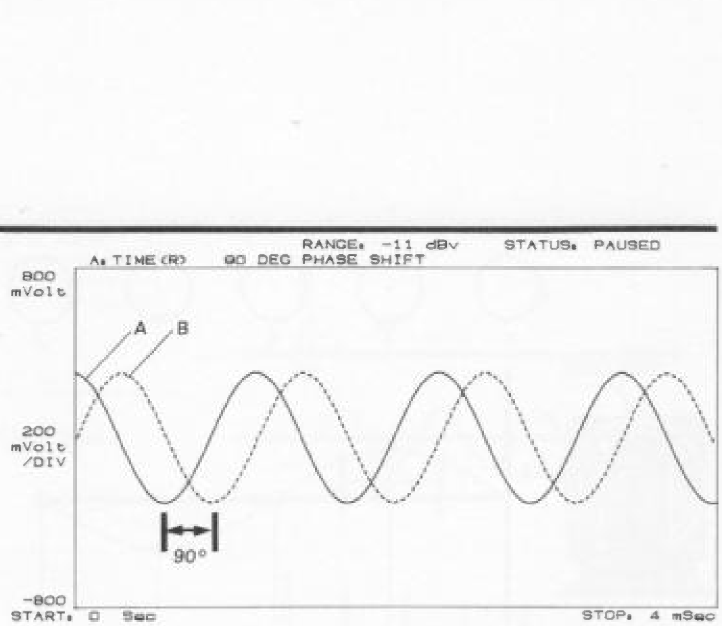


Figure 3.4-3
DSA frequency domain display of a 90° phase relationship. Referring to Figure 3.4-2, this is the phase of trace B when trace A is used to trigger the measurement.

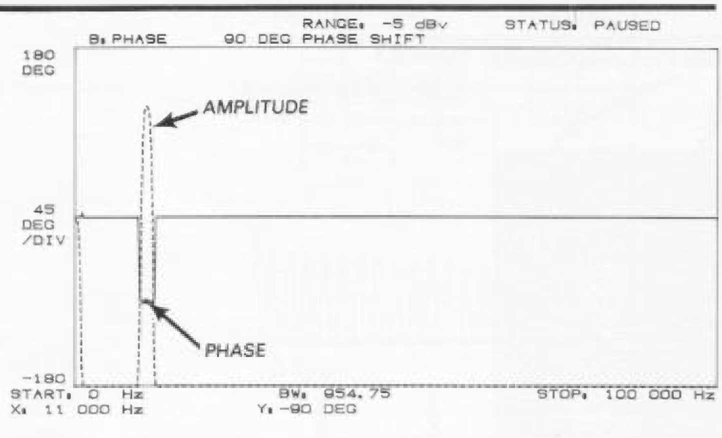
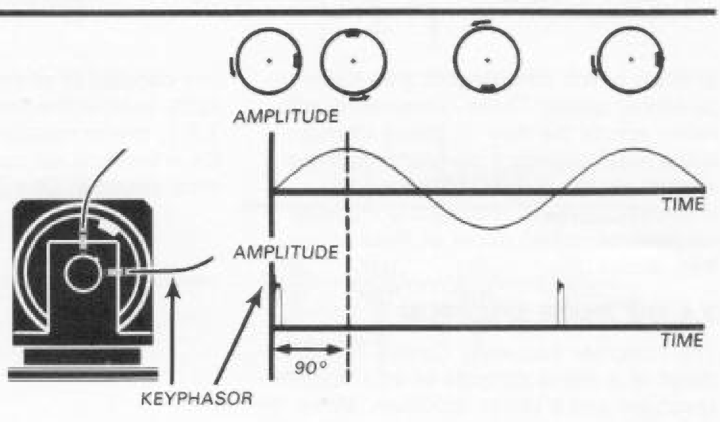


Figure 3.4-4
Since the heavy spot on the rotor passes the transducer 90° after the keyway passes the keyphasor, the imbalance signal lags the keyphasor pulse by 90°. The corresponding frequency domain phase spectrum is shown in Figure 3.4-3.



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.

Just as we can define the phase of a signal relative to a reference point on a rotor, we can define the relative phase of two signals. 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.

Balancing

The most common application for the phase spectrum is in 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 TM — 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*

* Single-channel DSAs require trigger delay to read phase relative to a trigger correctly — refer to operating instructions.

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. Once the relationship between vibration amplitude and imbalance weight has been determined, all the information needed for single-plane balancing can be obtained with one measurement. For balancing, you should remember that changing the vibration parameter, or moving the keyphasor relative to the transducer, will introduce a phase change that must be taken into account. (Vibration parameter phase relationships are described in Section 2.1.)

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 measurement 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 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, usually referred to as a critical speed. A flexible rotor has several critical speeds, each with a specific bending shape. These shapes are called modes, and can be predicted through modeling. 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°.

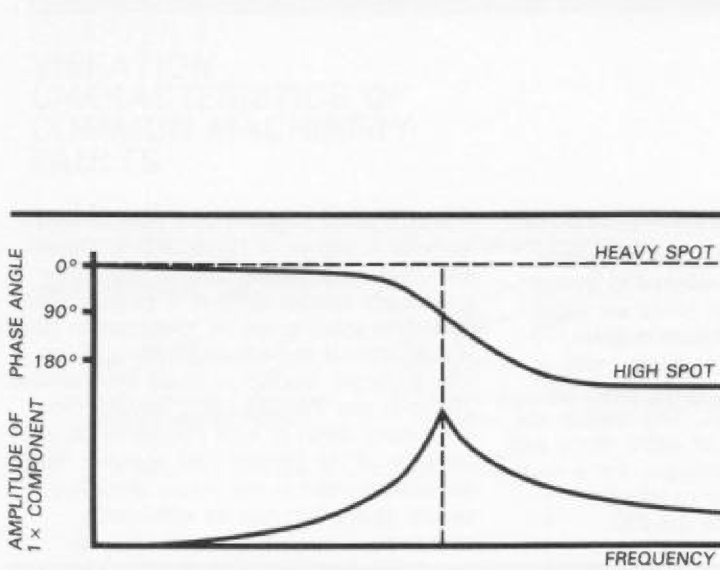


Figure 3.4-5
The vibration response of a flexible rotor shifts 180° in phase as rpm passes through a critical speed.

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, 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 scope 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).

* Keyphasor is a Bently Nevada trademark.

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 analyzer: (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 ⅓ or 1 octave intervals. This spacing results in resolution that is proportional to frequency. For a ⅓ octave analyzer, resolution varies from around 20 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 several filters that can be individually selected.

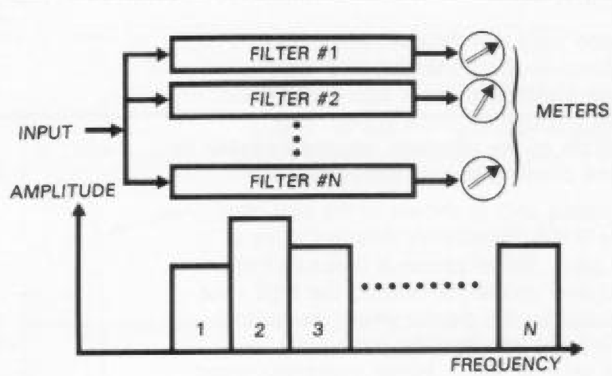


Figure 3.5-1
Parallel filter analyzers have insufficient frequency resolution for machinery analysis.

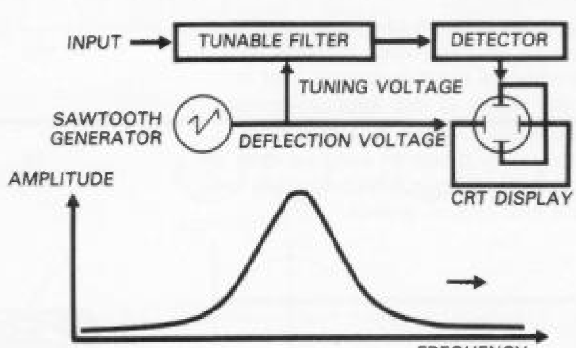


Figure 3.5-2
Swept filter analyzers provide better frequency resolution than parallel filter analyzers, but are too slow for machinery analysis.

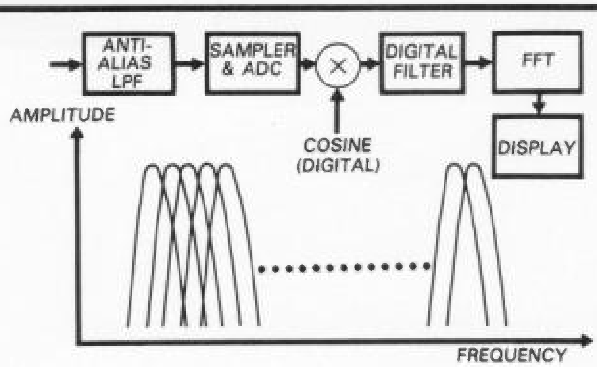


Figure 3.5-3
DSAs digitally simulate hundreds of parallel filters, providing both high speed and excellent frequency resolution. DSAs also provide time and phase displays not available on the other frequency domain analyzers.

Parallel filter analyzers are usually relatively low in cost and battery operated, but their resolution is not nearly good enough for analysis. This is especially true for gear and bearing analysis, where frequency resolution of a few Hertz is often required. Electrical problems in induction motors often require frequency resolution of a few Hertz. (Section 6.2 describes the importance of 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 5 Hz — better than parallel filter analyzers, but not good enough for complete vibration analysis. Their operation, however, is much slower than the parallel filter analyzers. This is because transient responses in the filter must be allowed to settle at each new frequency. This slow operation increases the time required to complete a vibration survey. Also, because of the time required to sweep through the frequency spectrum, swept-filter analyzers can miss short duration events.

DSAs use digital techniques to effectively synthesize a large number of parallel filters. The large number of filters (typically 400) provides excellent resolution, and the fact that they are parallel means that measurements can be made quickly. DSAs also provide both time and phase spectrum displays, and can be connected directly to computers for automated measurement. For these reasons, DSAs are the best frequency domain analyzer for vibration analysis.

CHAPTER 4: VIBRATION CHARACTERISTICS OF COMMON MACHINERY FAULTS

In the last chapter, we saw how the vibration signal can be reduced to simple components using the frequency domain. In Chapters 4 and 5, we'll take the next step in analysis — correlating these components with specific machine faults. This chapter provides the basic theory, while Chapter 5 addresses some of the common problems encountered in analysis.

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 tables in Section 4.10 summarize 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 condition (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 condition is the most challenging aspect of vibration analysis. Chapters 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 characteristics that can be used to differentiate these faults from imbalance, eliminating unnecessary balancing jobs.

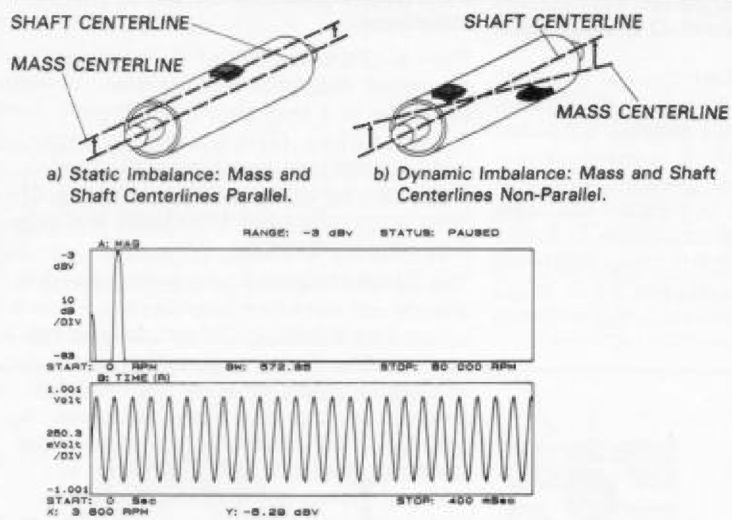


Figure 4.1-1
Imbalance, whether static or dynamic, results in a spectral peak at a frequency of once per revolution (1 ×).

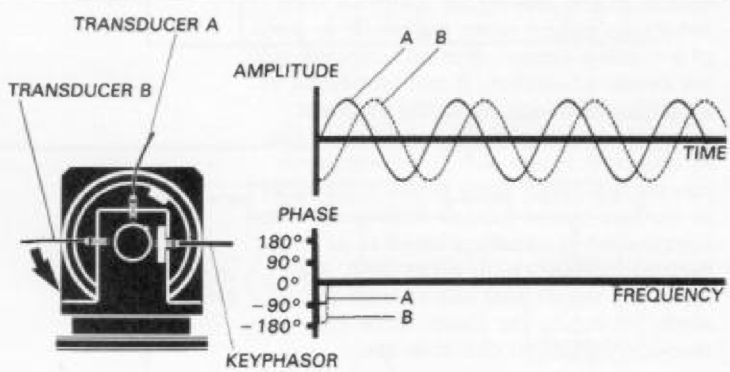
Phase plays a key role in detecting and analyzing imbalance, and it is important to remember the phase shifts associated with flexible rotors (see Figure 3.4-5). A state of imbalance occurs when the center of mass of a rotating system does not coincide with the center of rotation. It can be caused by a number of things, including incorrect assembly, material buildup, and rotor sag. As shown in Figure 4.1-1, the imbalance can be in a single plane (static imbalance) or multiple planes (couple imbalance). The combination is usually 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 the vibration caused by imbalance are: (1) it is sinusoidal at a frequency of once per revolution (1 ×), (2) it is a rotating vector, and (3) amplitude increases with speed. These characteristics are very useful in differentiating imbalance from faults that produce similar vibration.

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 harmonics above once per revolution, the fault is not 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).

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



Because the imbalance force is a rotating vector, the phase of vibration relative to a keyphasor follows transducer location, while amplitude changes little. As shown in Figure 4.1-2, moving the transducer 90° results in a 90° change in phase reading, with approximately the same amplitude.* Two such readings made on directional vibration (e.g. caused by looseness) will vary widely in amplitude.

Faults Commonly Mistaken For Imbalance

The following faults are often mistaken for imbalance because they result in increased levels of vibration at running speed. However, each has distinctive characteristics which can be used for identification.

A. MISALIGNMENT. The key characteristics that identify misalignment are a large second harmonic component, and a high level of axial vibration. Bent shafts and improperly seated bearings are special cases of misalignment, and produce similar vibration. The relative phase of axial vibration due to misalignment, measured at the ends of the shaft, is typically 180°. These signals are usually in phase when the shaft is out of balance (see Section 4.4).

B. LOAD VARIATION. Uneven loading in material handling machines and retained fluid in pumps result in imbalance. Higher machine loading can also cause an increase in running speed vibration level. The key to correct interpretation of these increases is a good understanding of the machine's operating characteristics. When measuring baseline vibration, it is important to check variation with key operating parameters such as load, pressure, and temperature. If the variation is significant, it can be roughly characterized by taking baseline spectra under a variety of operating conditions.

C. MECHANICAL LOOSENESS. The vibration spectrum that results from looseness almost always includes higher harmonics, although these can be damped out in a belt-driven machine. Since looseness is usually highly directional, it can be identified by relatively large level changes with transducer location (see Section 4.5).

*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.

D. RESONANCE. A machine resonance (natural frequency) at running speed will produce a high 1× vibration level. Resonance is usually identifiable because vibration level will be significantly reduced at frequencies higher or lower than resonance. Resonances are usually the result of installation faults (see Section 4.8).

E. EXCESSIVE CLEARANCE IN FLUID-FILM BEARINGS. The increase in running speed vibration level is usually accompanied by higher frequency harmonics.

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 detection and analysis.

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 a computer program that computes the expected frequencies given bearing parameters and rotational speed. One caution: parameters for 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 low 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).* A good solution is regular monitoring of critical machinery with a DSA, since the high resolution and dynamic range can show components as small as 1/1000 the amplitude of higher level vibration.

An added benefit of early detection is that indications of the cause of failure, which may be obliterated in later stages, are still visible. An example of this would be false brinelling caused by excessive vibration on a stationary machine. By understanding the cause of problems such as this, the source of chronic failures can be determined.

Frequencies Generated by Rolling Element Bearing Defects

Formulas to calculate the frequencies resulting from bearing defects are given in Table 4.2 (refer to Figure 4.2-1). The formulas assume a single defect, rolling contact, and a rotating shaft with fixed outer race. The results can be expressed in orders of rotation by leaving out the (RPM/60) term. The BASIC program in Figure 4.2-2 will compute the bearing frequencies automatically. Again, remember

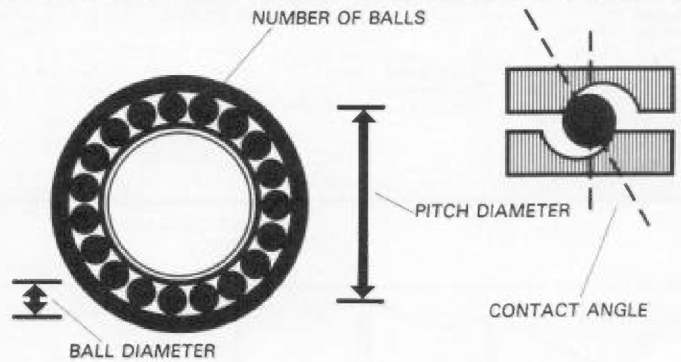


Figure 4.2-1
Using the parameters shown, the basic frequencies resulting from rolling element bearing defects can be computed.

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)

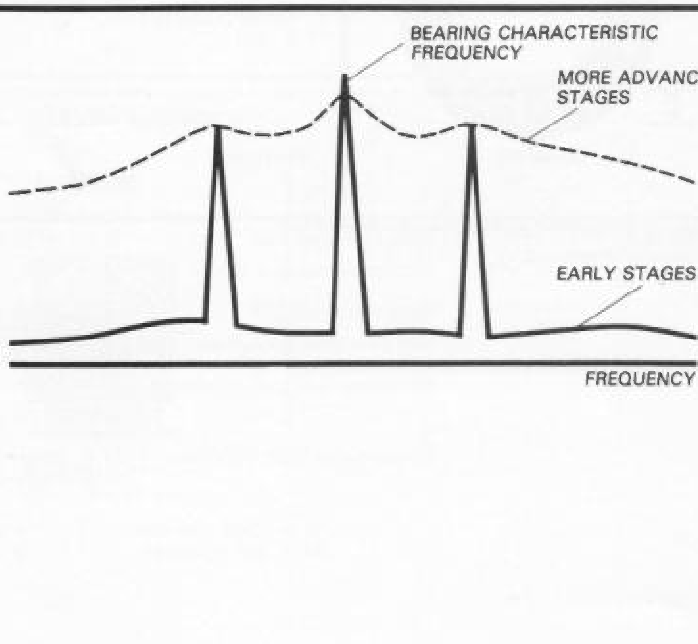
Pd = Pitch diameter
Bd = Ball diameter
n = Number of balls
ϕ = Contact angle

* Specialized vibration meters are available for early detection of bearing defects. These depend on filtering or special transducers to heavily weight readings toward bearing frequencies. These instruments usually detect defects earlier than normal meters, although readings can sometimes be misleading, and they cannot replace DSA's for analysis.

Figure 4.2-2
A BASIC program to compute bearing characteristic frequencies. The specific unit used in lines 170 and 180 is not critical, as long as it is the same for both.

```
10 ! Bearing frequency calculation program for HP 85
20 !
30 DIM D$[32]
40 DEG @ CLEAR
50 DISP "Enter bearing description:"
60 INPUT D$
70 DISP "Enter ball diameter, pitch dia-  meter:"
80 INPUT B,P
90 DISP "Enter contact angle, # of balls:"
100 INPUT A,N
110 DISP "Enter RPM (0 for ORDERS)"
120 INPUT F
130 IF F=0 THEN L$=" Orders" ELSE L$=" Hertz"
140 IF F=0 THEN F=60
150 F=F/60 ! Convert RPM to Hz
160 PRINT D$
170 PRINT USING "14A,2DZ.3D,7A" ; "Ball diameter:",B," inches"
180 PRINT USING "15A,2DZ.3D,7A" ; "Pitch diameter:",P," inches"
190 PRINT USING "14A,2DZ.D,8A" ; "Contact angle:",A," degrees"
200 PRINT USING "16A,DD" ; "Number of balls:",N
210 IF F>1 THEN PRINT USING "6A,5DZ.DD,4A" ; "Speed:",F*60," rpm"
220 PRINT "          -------------"
230 IMAGE 17A,4DZ.2D,7A
240 PRINT USING 230 ; "Ball pass--outer:",F/2*N*(1-B/P*COS(A)),L$
250 PRINT USING 230 ; "Ball pass--inner:",F/2*N*(1+B/P*COS(A)),L$
260 PRINT USING 230 ; "Ball spin:",F/2*(P/(2*B))*(1-(B/P*COS(A))^2),L$
270 PRINT USING 230 ; "Fund. train:",F/2*(1-B/P*COS(A)),L$
280 PRINT USING "4/"
290 END
```

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



that bearing parameters can change with manufacturer.

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 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 outer race, the rate of rotation relative to the shaft center is the average, or ½ the shaft speed. This is the reason for the factor of ½ in formulas (1) through (4). (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 than those on the outer race, because the linear distance (which is proportional to diameter) is shorter.

The fundamental train frequency, which occurs at a frequency lower than 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 or velocity probes. 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).

An interesting new development in bearing transducers [13] involves high sensitivity displacement probes which measure the actual deflection of the bearing outer race. This measurement is very sensitive to bearing defects, and eliminates the effects of case impedance. Installation, however, requires disassembly of the machine.

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 completely. 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. Comparison with a baseline spectrum will help to avoid misinterpretation.

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

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

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

Example Spectra

The example spectrum of Figure 4.2-5 is the result of a defect in the outer race. A printout of bearing data and characteristic frequencies, computed with the program given above, appears in Figure 4.2-4. Note the sidebands at running speed which are characteristic of most bearing spectra.

The spectrum in Figure 4.2-6 is also the result of a defect in the outer race. In this example, the characteristic ball pass frequency has disappeared, but its harmonics remain. The component around 200 Hz is gearmesh vibration.

```
Ball diameter:  0.156 inches        Ball pass--outer:   2.63 Orders
Pitch diameter:  0.625 inches       Ball pass--inner:   4.37 Orders
Contact angle:  0.0 degrees         Ball spin:          0.94 Orders
Number of balls: 7                  Fund. train:        0.38 Orders
```

Figure 4.2-4
Bearing data and characteristic frequencies for the spectrum of Figure 4.2-5.

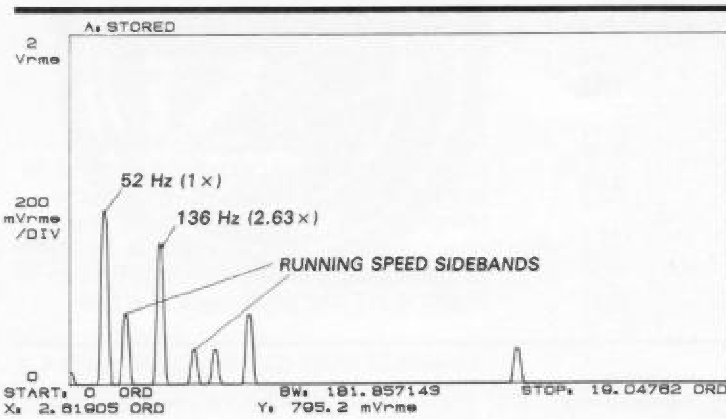


Figure 4.2-5
Spectrum resulting form a single defect in the outer race of a bearing. The sidebands at running speed are characteristic of bearing defects.

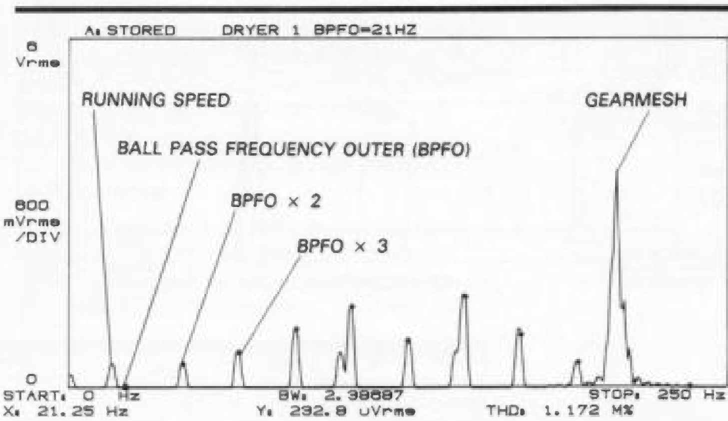


Figure 4.2-6
In this example of an outer race defect, the component at the ball pass (outer race) frequency has disappeared, but its harmonics remain. This is characteristic of advanced stages of a defect.

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 the frequency spectrum. In Figure 4.2-7, 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 1 msec intervals, resulting in a harmonic spacing of 1 kHz (1/1msec) in the frequency spectrum.

The important consequences of the high frequency content are:

A. The sensitivity of accelerometers to high frequency vibration implies that a large amount of energy outside the range of characteristic bearing frequencies will be included in measurements with these transducers. This is not a problem when analysis is performed with a DSA, but it does make overall vibration measurements misleading (i.e. the reading includes vibration responses not necessarily related to defects). For this reason, velocity is a better choice for monitoring vibration level. Velocity can be derived from acceleration by integration (see Section 6.6), or measured directly with a velocity transducer.

B. High frequency resonances in the bearing and machine structure may be generated, 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 depends on these high frequencies (20-50kHz) to excite the natural frequency of a special accelerometer. (With no exciting frequency in this range, the output of

Figure 4.2-7
The impulsive nature of bearing defects produces a large number of harmonics spaced at the characteristic frequency.

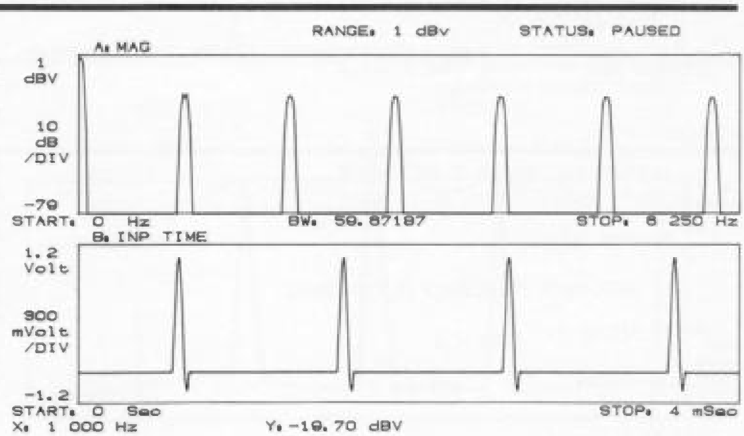
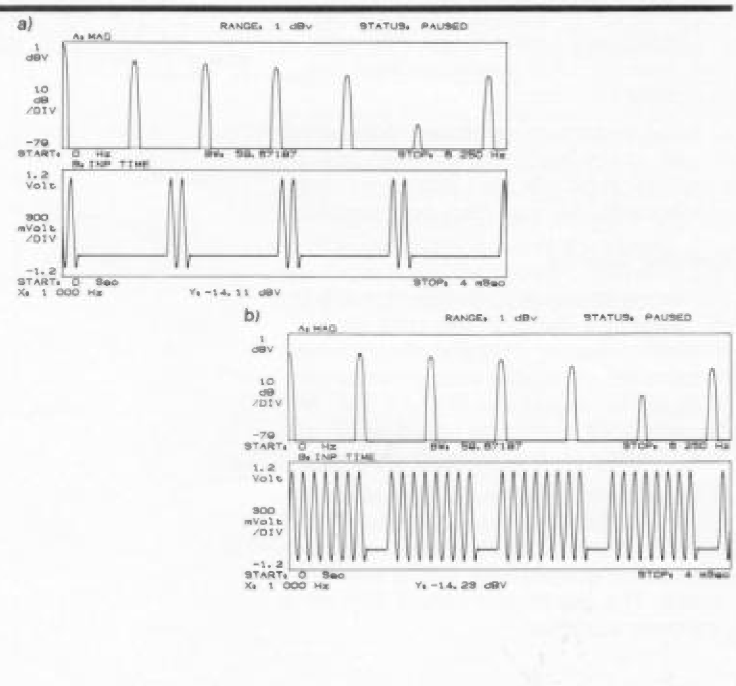


Figure 4.2-8
Two simulated examples of multiple defects. Note that the harmonic spacing remains at the characteristic frequency.



the transducer is very low.) This type of instrument can produce misleading results if the accelerometer is not carefully mounted, or if the defect is such that little high frequency energy is produced.

C. 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.

Multiple Defects and Running Speed Sidebands

The characteristic spectrum of multiple bearing defects is difficult to predict, depending heavily on the exact nature of the defects. Figures 4.2-8(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 4.2-9, a bearing defect pulse is being modulated by imbalance. The imbalance component appears at the 280 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.

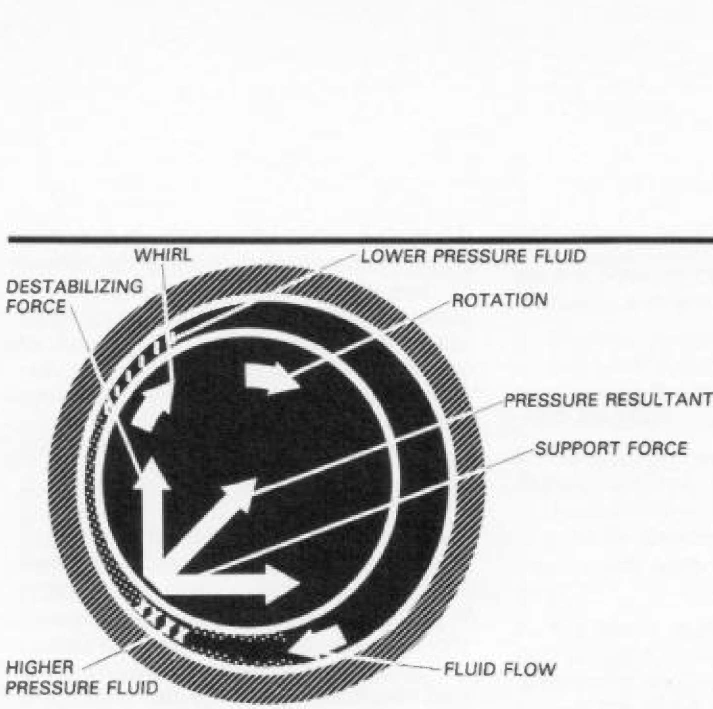


Figure 4.3-1
A pressure differential in fluid-film bearings produces a tangential force that results in whirl.

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 fluid-film bearings [25, 26]. In this section we will discuss only fluid-bearing instabilities, which are the most common.

A basic difference exists between vibration due to instability, and vibration due to other faults such as imbalance. Consider the case of shaft imbalance. Vibration of

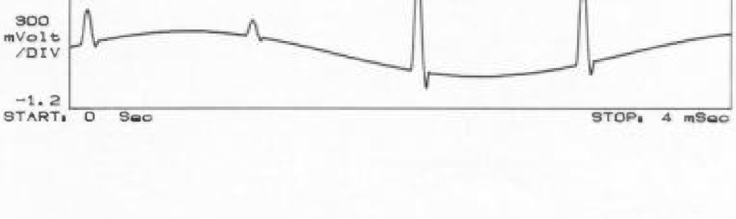


Figure 4.2-9
Bearing frequencies are almost always modulated by residual imbalance at running speed.

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 rotational frequency. The difference is subtle, but has a profound effect on measures taken to cure the problem.

Oil Whirl and Whip

Deviations 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 ½ 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 rotation that results in a whirl — or precession — of the rotor at slighlty less than ½ 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 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 ½ 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 of Figure 4.3-2 illustrates how oil whirl becomes unstable oil whip when shaft speed reaches twice critical. Whirl must be suppressed if the machine is to be run at greater than twice the critical speed.

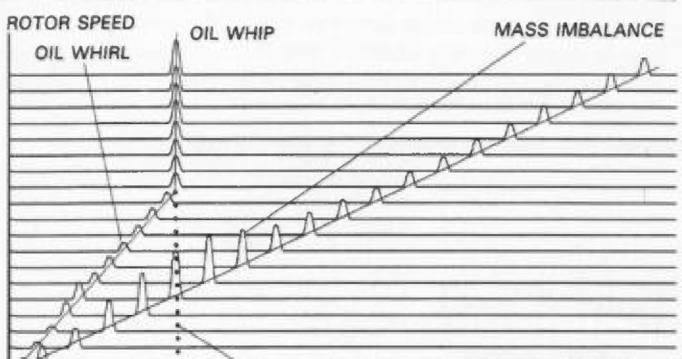


Figure 4.3-2
A spectral map showing oil whirl becoming oil whip instability as shaft speed reaches twice critical.

4.4 MISALIGNMENT

Vibration due to misalignment is usually characterized by a 2× running speed component and high axial levels. When a misaligned shaft is supported by rolling element bearings, 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 bearing, and (2) offset of the shaft center lines of machines in the same train. 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. These high axial levels are a key indication of misalignment.

High second harmonic vibration levels are also a common result of misalignment. The ratio of 1× to 2× component levels can be used as an indicator of severity [3]. Second harmonics are caused by stiffness asymmetry in the machine and its supports, or in the coupling. This asymmetry causes a sinusoidal variation in response level — a form of rotating impedance 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 looseness and excessive clearance. The key distinguishing feature is a high 2× 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 relationship 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 imbalance. Relative phase can be measured with a single-channel DSA using a keyphasor 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 readings, so that the axial phase relationship may be 150° or 200° rather than precisely 180°.

B. Transducer orientation is important. 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. The phase relationships described hold only for rigid rotors. Using phase for diagnostics on flexible rotors requires knowledge of the rotor's dynamics.

D. Great care must be exercised when measuring relative phase with a single channel DSA. Two measurements are required, each referenced to the shaft with a keyphasor. These measurements must be made at the same speed unless trigger delay or external sample control (Section 6.7) is used. In general, you should make more than one measurement at each point to insure that phase readings are repeatable.

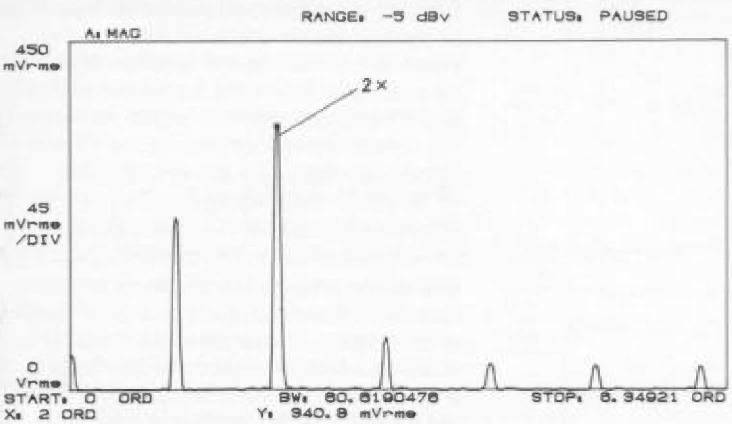


Figure 4.4-1
Alignment problems are usually characterized by a large 2× running speed component, and a high level of axial vibration.

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

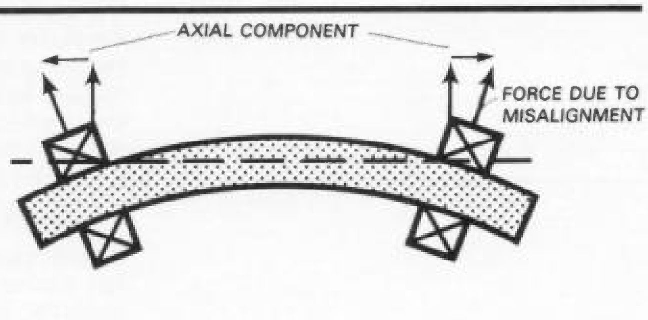
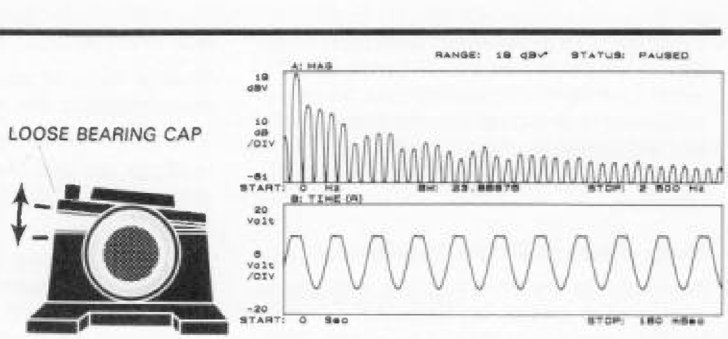


Figure 4.5
Looseness usually results in a truncated waveform that produces a spectrum with a large number of both odd and even harmonics.



4.5 MECHANICAL 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 (velocity transducers work well for this). Measured vibration level will be highest in the direction and vicinity of the looseness.

The harmonics that characterize looseness are a result of impulses and truncation (limiting) in the machine response. Consider the bearing shell in Figure 4.5-1. When it is tight, the response to imbalance at the transducer is sinusoidally varying. When the mounting bolt is loose, there will be truncation 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 truncation 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.

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 multi-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 gearboxes. Baseline spectra taken when the gearbox is in good condition make it easier to identify new components, or components that have changed significantly in level.

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 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 gearmesh frequency is 15 × 50 = 750 Hz. This component will appear in the vibration spectrum whether the gear is bad or not. Low level running speed sidebands 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 indicator 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 exists.*

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 indicator 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 gear mesh frequency will usually have sidebands at running speed.

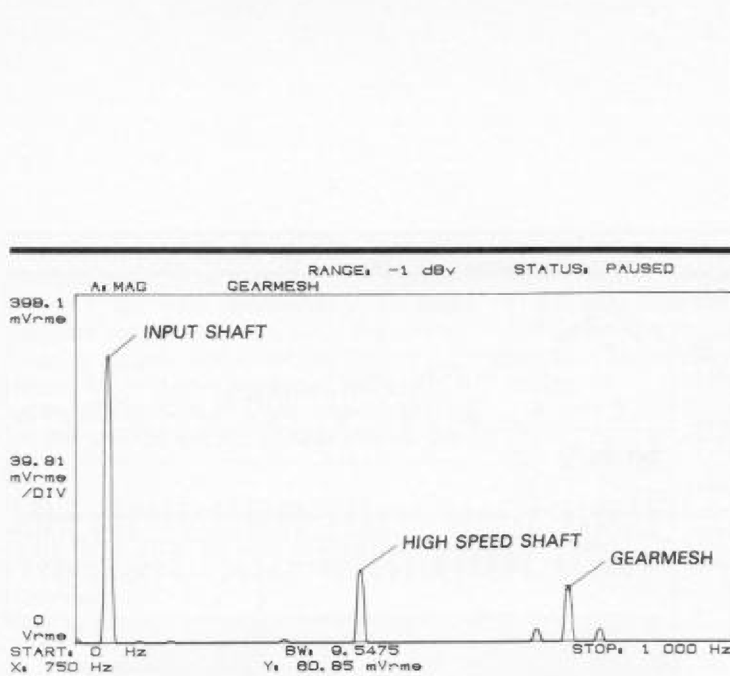


Figure 4.6-1
The characteristic spectrum of a gearset in good condition contains components due to running speed of both shafts, and gear meshing frequency.

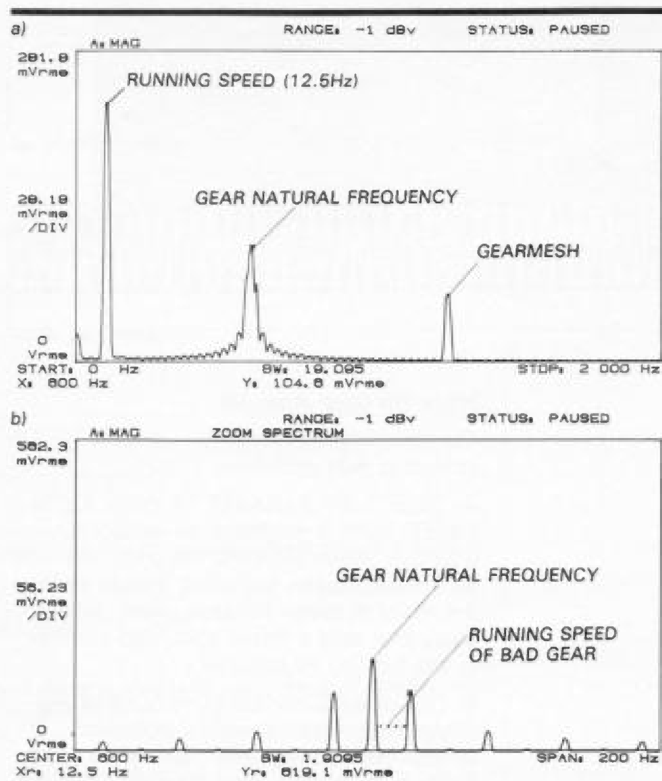
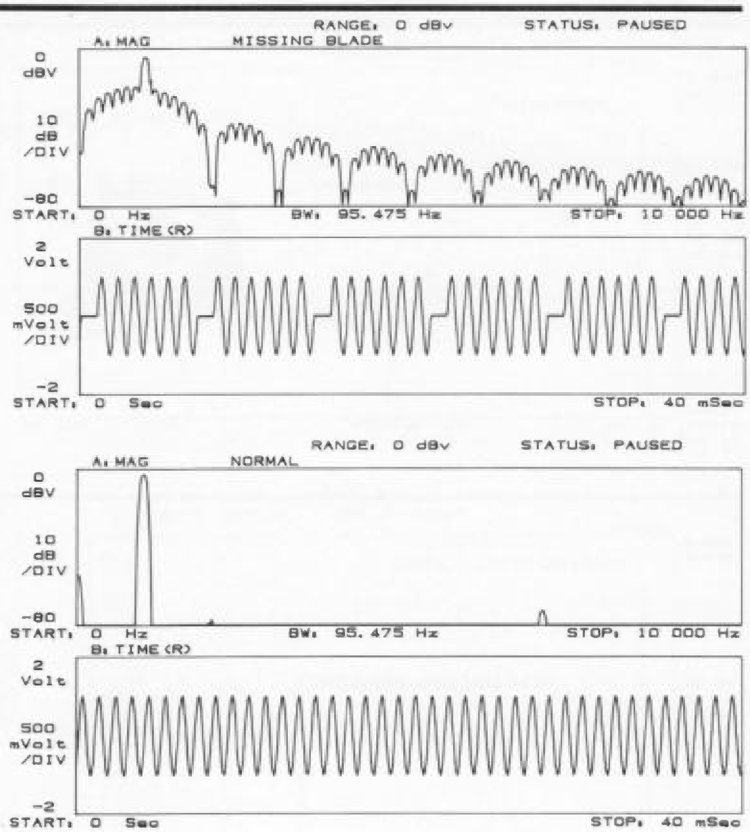


Figure 4.6-2
Gear natural frequencies, excited by impulses from large defects, are often the only indication of problems. The zoom spectrum in (b) shows the natural frequency, with sidebands that correspond to running speed of the bad gear.

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 × level.



Hints On Gear Analysis

The following hints may be useful in analyzing gear problems.

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 bearing 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 assembled 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 gearmesh and shaft speed frequencies. Even if you don't know the natural frequencies, shaft speed sidebands will often indicate the bad gear.

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 × speed) are always present, and levels can vary markedly with load. This is especially true for high speed machinery, and makes recording of operating parameters for historical data 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 1 × 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 telemetered or transferred through slip rings. Doppler detection techniques show promise, but have not been sufficiently developed for practical use. 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, makes spectra difficult to analyze quantitatively.

One characteristic that often appears in missing- or cracked-blade spectra is a large number of harmonics around the blade passing frequency. Figure 4.7-1 shows how a space in the vibration signal greatly increases the number of harmonics without changing the fundamental frequency.

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, 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.

Shaft resonance problems in high speed machinery are sometimes caused by changes in the stiffness provided by fluid-film bearings, 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. 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. The key is to understand that maintenance and installation related factors can alter assumptions made in the rotor design.

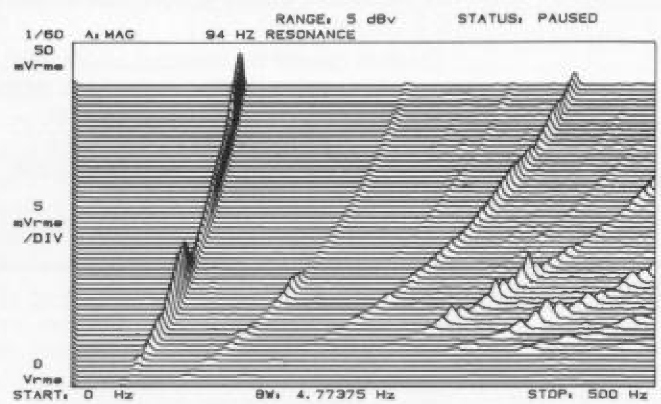


Figure 4.8
Spectral maps are especially useful for analyzing vibration due to resonances.

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.

*Relative to running speed

4.9 ELECTRIC MOTORS

Excessive vibration in electric motors can be caused by either mechanical, or electromagnetic defects. The latter can be quickly 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 powerline related components are often very closely spaced (see Section 6.2 on resolution).

Vibration caused by electrical problems in induction motors can be analyzed to determine the nature of the defect [20]. In general, a stationary defect such as a shorted stator produces 120 Hz vibration (100 Hz with a 50 Hz powerline). A rotating defect, such as a broken rotor bar, produces 1 × vibration with 2 × 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. Baseline vibration signatures taken on machines in good condition are the best criteria for judging the relative severity of these levels.

4.10 SUMMARY TABLES

Tables 4.10-1 and 4.10-2 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 4.10-2
Vibration Frequencies Related To Machinery Faults

Frequency	Possible Cause	Comments
1 × 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 frequency 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 × running speed and ×2 running.
	Blades/vanes	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 structures. Frequency is proportional to stiffness and inversely proportional to mass.	

CHAPTER 5:
ADVANCED ANALYSIS AND DOCUMENTATION

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

5.1 PRACTICAL ASPECTS OF ANALYSIS. A discussion of 5 practical aspects of successful analysis.

5.2 USING PHASE FOR ANALYSIS. We have discussed the importance of phase in analyzing defects such as imbalance and misalignment. This section is an extension of that discussion, and an introduction to the related concept of time averaging.

5.3 SUM AND DIFFERENCE FREQUENCIES. Multiple defects often produce vibration components that are sums and differences of the defect characteristic frequencies.

5.4 SPEED NORMALIZATION. A common problem when making direct spectral comparisons is shift in frequency of vibration components caused by changes in running speed. This section discusses solutions to the problem.

5.5 BASELINE DATA COLLECTION. Records of vibration spectra taken when a machine is in good condition save significant amounts of time in analysis. This section presents guidelines for collecting baseline data.

5.1 PRACTICAL ASPECTS OF ANALYSIS

In the literature and discussions on the subject of machinery vibration analysis, several factors are regularly mentioned as keys to success. In this section, we will discuss five of these factors: (1) documentation, (2) machinery knowledge, (3) severity critera, (4) instrumentation, and (5) analysis personnel.

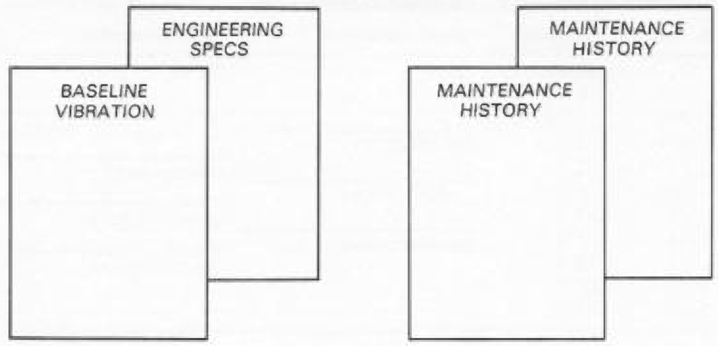


Figure 5.1-1 Complete documentation consists of baseline vibration spectra, maintenance history, and engineering data.

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

The baseline vibration measurements made on a machine in good condition provide a reference for detecting changes which indicate problems — and for identifying significant components when problems do occur. Without this information, you could easily waste time determining the source of a vibration component that is contained in the spectrum of a good machine. Baseline data collection is discussed in Section 5.5.

Maintenance history includes records of machine failures, and vibration spectra before and after repairs. These records form a library of known defects and vibration spectra that is invaluable in analysis. You should, for example, be able to immediately identify a defect that has occurred previously in a machine. These records also provide insights into design or specification changes that will improve reliability.

Engineering data includes bearing and gear parameters used to calculate characteristic frequencies, and machine dynamic models used to predict vibration response. Also useful are manufacturer's data on vibration limits and characteristics. This data will not always be easy to obtain — the key is to collect the available data before problems occur.

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

Vibration Severity		Support Classification	
in./sec	**mm/sec**	**Hard Supports**	**Soft Supports**
.017	0.45	good	good
.028	0.71		
.044	1.12		
.071	1.8	satisfactory	
.11	2.8		satisfactory
.18	4.5	unsatisfactory	
.28	7.1		unsatisfactory
.44	11.2	impermissible	
.71	18.0		impermissible
1.10	28.0		
2.80	71.0		

Computers are a virtual necessity for efficiently keeping the records described above. Several companies offer software for this purpose that will work on a variety of computers and DSAs.

2. MACHINERY KNOWLEDGE. The design and operating characteristics of a machine determine both the type of defects that are possible, and the vibration response to those defects. Vibration analysis is difficult without a working knowledge of these characteristics. Another important consideration is the effect of changes in operating condition on measured vibration. By understanding how vibration changes with such variables as load and temperature, you will be better able to determine whether an increased level of vibration is due to a defect, or to a change in operating conditions.
The best sources of information on these characteristics are the manufacturer of the machine, and plant records of machinery defects and their associated vibration. Courses on machine maintenance and design from manufacturers can provide insight into both possible defects, and the mechanisms of vibration response for specific machines. Several baseline spectra taken under different operating conditions are useful for documenting the effects of changing operating parameters.

3. SEVERITY CRITERIA. Once a defect has been detected and identified, its severity must be determined for the repair to be most effectively scheduled. We have already mentioned the variability of vibration level with changes in speed and operating parameter, which makes it difficult to assign severity solely on the basis of level. On the other hand, the appearance of vibration components at key frequencies — such as those which characterize bearing defects — is a good indication that problems exist.

Assessment of vibration severity should consider both the level of vibration, and its characteristics. References for severity include published vibration standards, and historic vibration measurements.

The table in Figure 5.1-2 is an example of a published vibration standard. This particular standard is from the International Standards Organization (ISO). To make the

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

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 5.2-1 Phase Characteristics of Common Vibration Sources

standard more applicable to a wide range of machines, a distinction is made in the severity criteria between soft and hard supports. The essential problem with such published standards is that they are too general to be used for high accuracy judgements of vibration severity. Where one machine may operate normally at a vibration level of 0.1 in/sec, that vibration level in another machine may indicate imfailure. Thus, unless the standard is for a specific machine, it is best used only as a guideline.

Historic vibration measurements are an excellent reference for severity measurements, because they are specific to the machine in question. While absolute vibration limits for a machine may not be known, there is a high probability that large changes in vibration level indicate a problem. The process of monitoring vibration level for changes is referred to as trend analysis. Since vibration level in a good machine is variable, it isn't always obvious how much change is tolerable. The best approach is to analyze the statistics of variability for each machine, and base change limits on that. An increase in vibration that exceeds two standard deviations is usually a sign of trouble. In the absence of this type of analysis, you can use a factor of 2 (6 dB) increase as an approximate change limit.

When a significant change is detected, vibration level and other key operating parameters should be monitored regularly. The rate of change of these parameters is often a good indication of the severity of the problem.

4. INSTRUMENTATION. The large reductions in machinery downtime and maintenance expense provided by vibration analysis make it unwise to sacrifice capability to save a small amount of capital investment. A DSA should have all the key machinery analysis features discussed in Chapter 6. A computer and applicable software that can automatically store vibration spectra and analyze trends can quickly pay for itself. It is important, however, to limit equipment to a manageable quantity. A program can fail to pay back its investment quickly if too much equipment is purchased without real insight into analysis and monitoring requirements. Also keep in mind that some skill is required for analysis, and that this skill can take time to develop.

5. PEOPLE. The quality and effectiveness of a vibration analysis program is most often limited by the availability of capable personnel. Successful programs are characterized by people who are properly trained and given a chance to develop analysis expertise.

5.2 USING PHASE FOR ANALYSIS

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

In general, the phase of vibration caused by a defect will either be stable or unstable relative to a keyphasor. The nature of this relationship is shown in Table 5.2-1 (a reproduction of Table 4.10-1). Figure 5.2-1 is a sequence of vibration spectra that

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

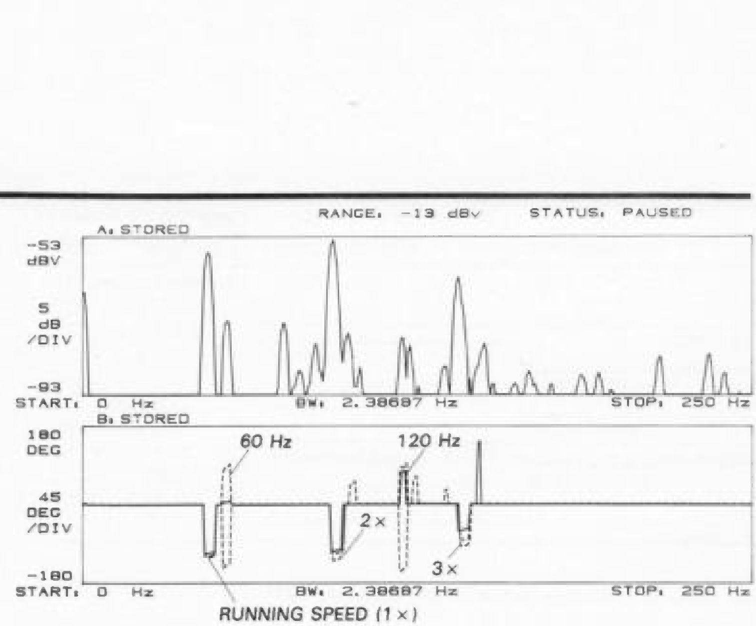


Figure 5.2-2
Instrumentation setup for phase measurements and time averaging.

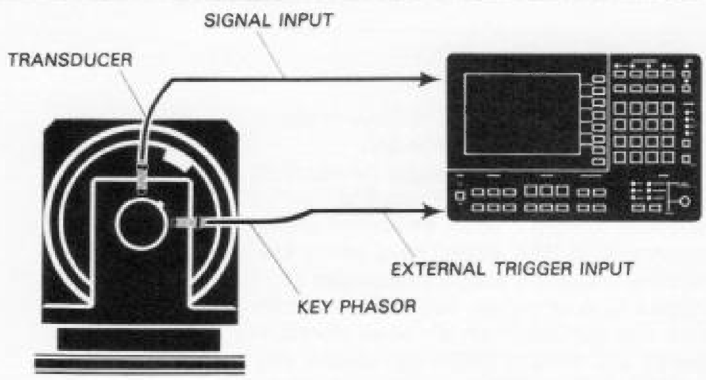
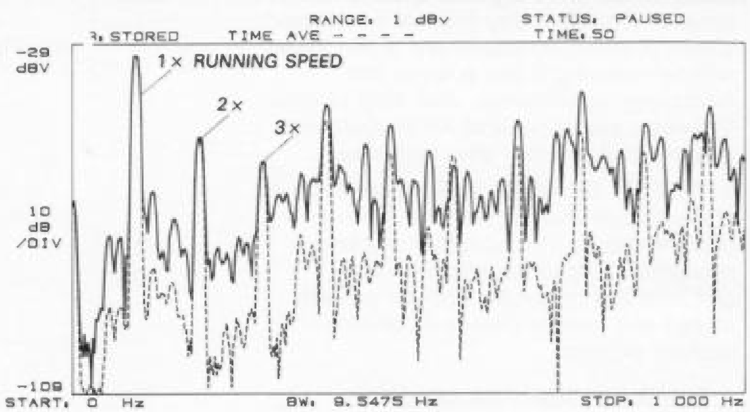


Figure 5.2-3
Time averaging is effective in reducing the level of background vibration.



shows phase for imbalance (1 ×), and running speed harmonics, and unstable phase for powerline-related components. Also, the relative phase relationship between vibration at different points on a machine can be used to differentiate between faults — as in the case of misalignment and imbalance (see Section 4.4).

Instrumentation required for phase measurements is shown in Figure 5.2-2. The keyphasor senses shaft rotation and serves as the phase reference. The phase of vibration that is synchronous (i.e. an integer multiple) with rotation is constant, while that of nonsynchronous vibration varies. Relative phase measurements can be made sequentially, as long as the same reference (i.e. keyphasor) is used (see Section 6.8 on easier relative phase measurements with a dual-channel DSA). Running speed should remain constant between measurements to minimize the phase effects of mechanical impedance. Relative phase measurements on flexible rotors *must* include considerations of shaft dynamics.

The keyphasor, which is usually a proximity sensor that detects a keyway or setscrew, provides a clean signal for triggering. It is required because the vibration signal is usually not suitable as a trigger source: although its largest component is usually 1 × rpm imbalance, noise in the spectrum adds uncertainty to level, and thus trigger timing. If a keyphasor is not available, it may be possible use a band-pass or low-pass filter to reduce the level of noise and higher frequency components in the vibration signal to make it usable as a trigger.

When measuring relative phase between two ends of a machine, it is important to mount the transducers with the same orientation. When measuring axial vibration, for example, if both transducers face the machine, they are mounted 180° out of phase. Thus vibration due to misalignment, which you would expect to be 180° out of phase, will be measured as in-phase.

Time averaging is explained in Section 6.4, and is a powerful technique for eliminating nonsynchronous components from a vibration spectrum. It is most useful for reducing the level of background noise, especially vibration from other machines. It must be used with care, however, since it will reduce the level of all vibration components that are nonsynchronous, including bearing and gear frequencies. In the plot of Figure 5.2-3, a time averaged spectrum (dashed line) is overlaid on a non-averaged spectrum. The synchronous components have not changed in level, while the non-synchronous background noise components are greatly reduced.

5.3 SUM AND DIFFERENCE FREQUENCIES

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

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

The exact mechanisms which generate sum and difference frequencies are not well understood, and a complex mathematical analysis is beyond the scope of this note. However, you can get a feel for the interactions involved by thinking of them as a form of amplitude modulation. In

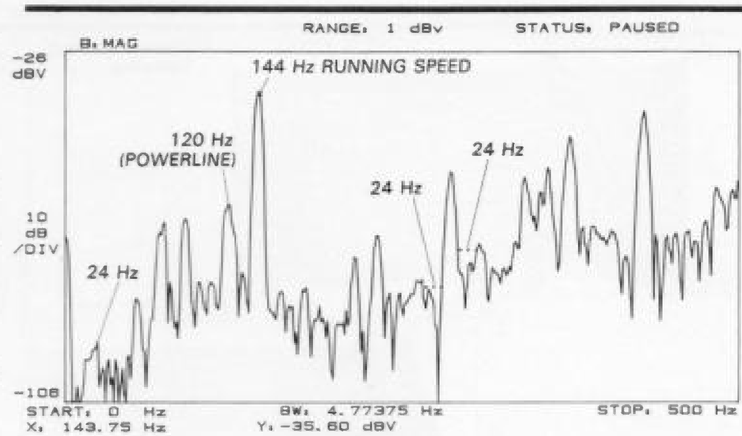


Figure 5.3-1
A vibration spectrum with sum and difference frequencies. The 24 Hz difference between rotational frequency and the 120 Hz powerline component appears both as a discrete signal, and as sidebands around rotational speed harmonics.

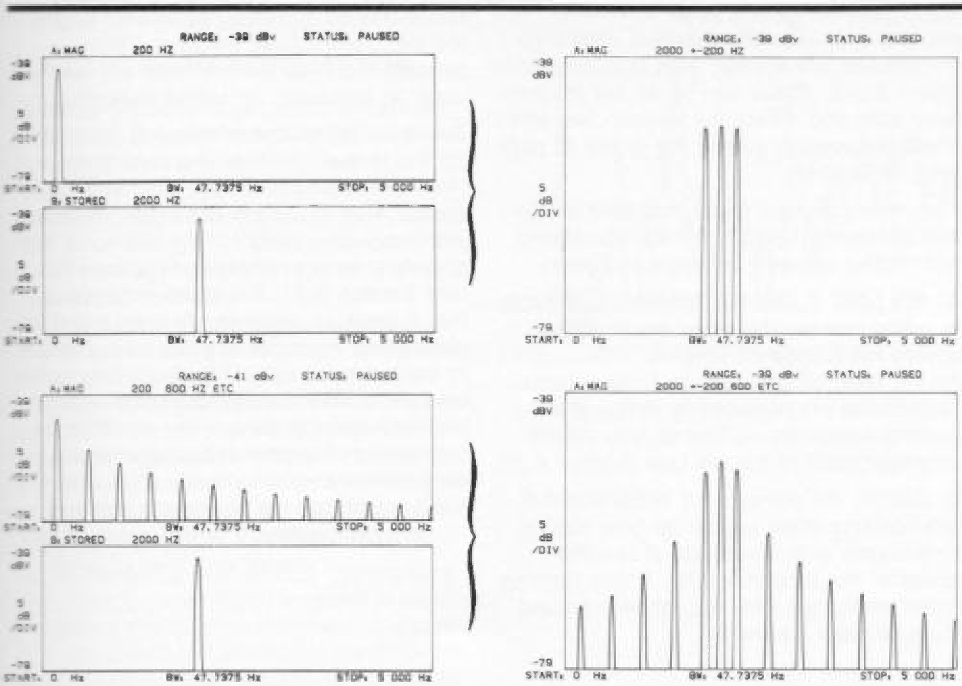
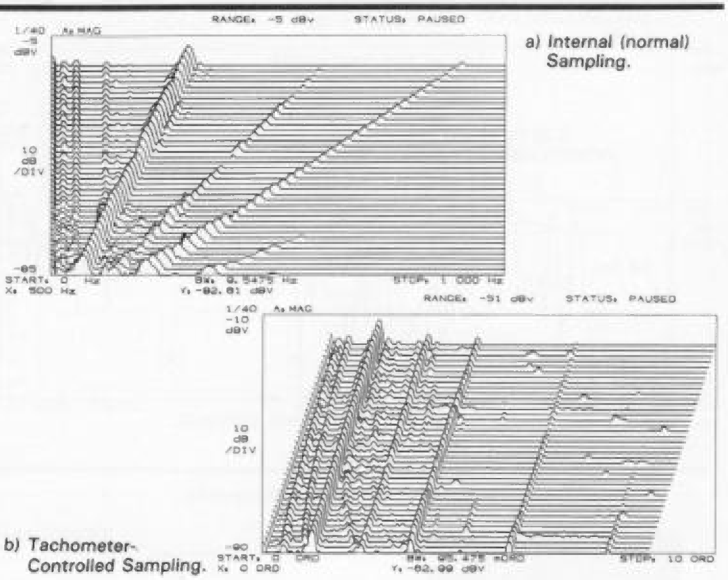


Figure 5.3-2
The number of sum and difference components depends on the number of harmonics in the signals involved.

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



a) Internal (normal) Sampling.

b) Tachometer-Controlled Sampling.

the trigonometric identity given below, it is apparent that the interaction

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

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

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

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

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

5.4 SPEED NORMALIZATION

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

External sample rate control locks the analysis process to a tachometer signal, and is especially applicable when large speed variations are encountered. The details of controlling the sample rate of a DSA externally are given in Section 6.7. A good way to illustrate the effects of external sample control is with spectral maps made in external and normal sample modes, as shown in Figure 5.4-1. These maps were made during a run-up. Note in the normal sample mode map of (a), rotational speed-related components move to the right as speed increases, while fixed frequency components (e.g. powerline related) move straight up. In the external sample control map of (b), rotational speed-related components move straight up the map, while fixed frequency components move to the left (they are relatively lower in frequency as speed increases).

The main advantage of external sample control is that real time displays of the vibration spectrum remain fixed with speed. Also, the 15% amplitude variation with frequency which DSAs without a high accuracy window experience is avoided (see Section 6.2). The disadvantages are that frequency calibration is lost, and a key phasor and multiplier are required. Frequency calibration is critical for analyzing fixed frequency vibration due to resonances and electrical defects. Frequency can also be normalized to rotational speed after a measurement. In the display of Figure 5.4-2, note that the frequency axis and

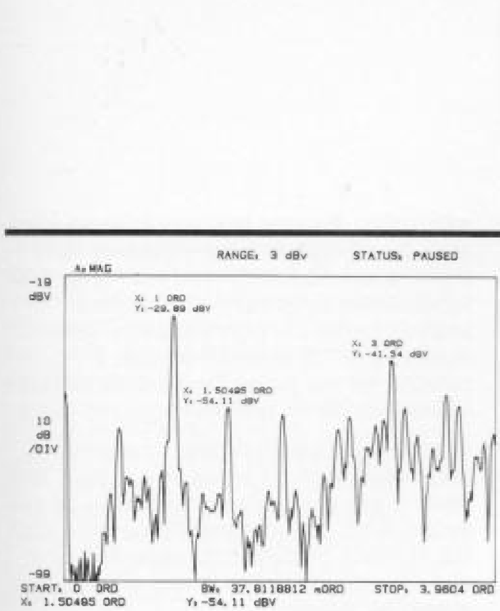
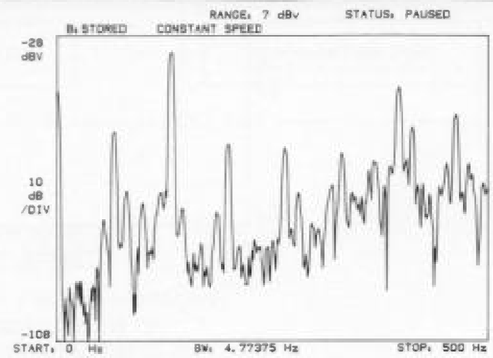


Figure 5.4-2
DSA display in which the horizontal (frequency) axis is calibrated in multiples (orders) of running speed.

marker readout are in terms of orders of rotation (multiples of running speed) rather than frequency. This normalization does not work in real time, and resolution is not a constant percentage of running speed (as with external sample control). However, frequency information is retained, and a keyphasor signal is not required.

Short-term speed variation causes a broadening of components in the vibration spectrum, as shown in Figure 5.4-3. As speed changes during the sampling interval for one measurement, the DSA is effectively analyzing several different spectra. This results in the broadened spectral components of 5.4-3b.

a) Constant Running Speed



b) Changing Running Speed

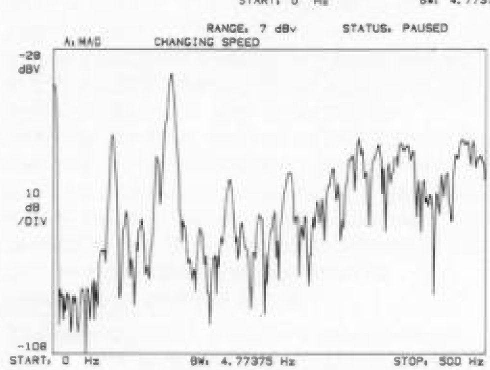


Figure 5.4-3
Short-term speed variation results in a broadening of spectral components (b).

5.5 BASELINE DATA COLLECTION

Baseline vibration spectra are reference data that represent normal machine condition, and are essential for effective analysis. In the event of trouble, they quickly indicate the frequency components that have changed. Without the baseline data, you can easily waste time analyzing spectral components that are present in normal operation. Baseline data are also the basis for trend monitoring, being a much more specific indicator of normal vibration than generalized vibration severity charts. To be most useful, the guidelines below should be followed for collecting baseline data. The key objective of the process is to understand the characteristics of the machine before a problem occurs.

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

A spectral map of a run-up or coast-down can also be useful in dealing with changes in speed. Such a map can quickly show how vibration level changes with speed, and the resonances and other fixed frequencies that should be expected in the vibration spectrum.

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

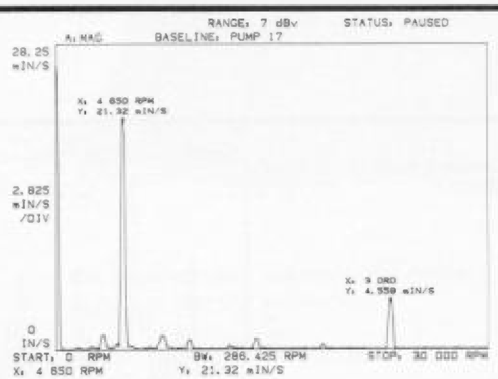


Figure 5.5-2
Baseline data should include fully documented vibration spectra and engineering data, such as bearing and gear parameters, which can be invaluable for analysis.

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

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

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

D. DOCUMENT THE EFFECT OF LOAD VARIATION. This is not strictly required, but can be invaluable when determining whether a change is due to a fault, or just a change in load.

E. UPDATE REGULARLY. Baseline data should be updated after major repairs or changes in operating conditions.

CHAPTER 6: **DYNAMIC SIGNAL ANALYZERS**

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

6.1 MEASUREMENT SPEED. Machinery vibration is a dynamic phenomenon that can change quickly — so quickly that slower swept spectrum analyzers may completely miss key events. DSAs can capture a typical vibration signal and transform it to the frequency domain in less than 1 second.

6.2 FREQUENCY RESOLUTION. Closely spaced machinery vibration signals often must be resolved for accurate analysis.

6.3 DYNAMIC RANGE. Vibration components that are the first indications of trouble are often very small relative to vibration from residual imbalance or other machines. The wide dynamic range of DSAs allows them to resolve signals less than 1/1000 the level of background vibration or residual imbalance.

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

6.5 HP-IB*. The Hewlett-Packard Interface Bus is a standardized interface that makes it easy to connect a DSA to a digital plotter for hard-copy results, or to a computer for automated data storage and analysis.

6.6 USER UNITS AND UNITS CONVERSION. DSA displays can be calibrated in vibration units such as inches/second and rpm. Units of vibration amplitude can also be converted to other parameters (e.g. acceleration to velocity) using the post-processing capabilities of DSAs.

6.7 EXTERNAL SAMPLE RATE CONTROL. By controlling data sampling rate with a tachometer pulse, the frequency axis can be normalized to rotational speed. This is convenient for real time analysis of machines whose speed varies widely.

6.8 TWO-CHANNEL ENHANCEMENTS. While single-channel DSAs are most commonly used for machinery analysis, dual-channel DSAs provide important enhancements such as real time phase comparison and transfer function measurements.

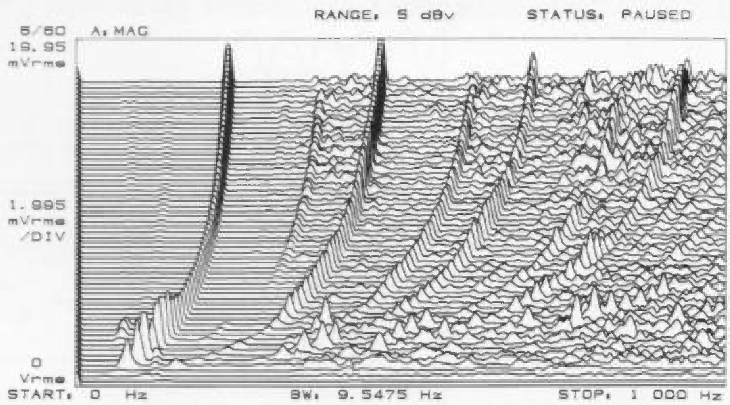


Figure 6.1-1 Machinery vibration spectra can change very quickly, as this spectral map of a run-up test illustrates. Slower swept spectrum analyzers can miss these changes.

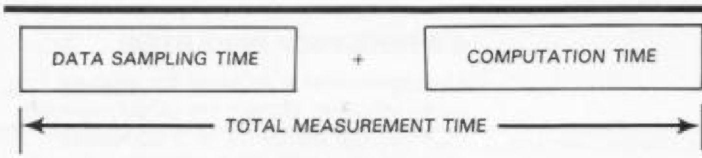


Figure 6.1-2 Total DSA measurement time is the sum of data sampling time and computation time. While sampling time is fixed for a given resolution, computation times vary widely among available DSAs.

6.1 MEASUREMENT SPEED

Speed is important in machinery analysis because vibration characteristics can change quickly. This is illustrated in the spectral map of Figure 6.1-1, where measurements of a machine run-up spaced at 0.5 second intervals show significant variation. Speed is also important for reducing the time required to characterize a machine. A high resolution measurement (500 Hz bandwidth) takes about 2 minutes with a swept analyzer; it takes less than 1 second with a DSA. The time differential can be significant when vibration is measured at multiple points.

The time required to make a measurement with a DSA is determined by two factors: (1) measurement resolution, and (2) transform computation time. High resolution measurements require a long data sampling time (resolution of events spaced at 1 second intervals requires a 1 second measurement time). This is a physical fact, independent of the design of the DSA. Computation time, however, varies widely among DSAs, and can make a noticable difference in measurement time. Computation time is usually expressed in terms of real time bandwidth — the frequency span at which data sampling and computation times are equal (higher real time bandwidth implies faster computation).

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

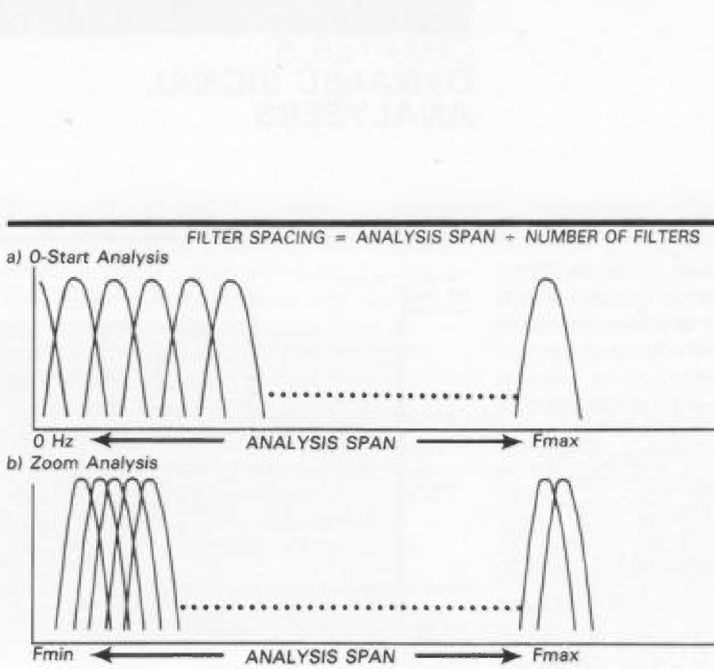


Figure 6.2-1
Frequency resolution in a DSA is determined primarily by the number of filters, and the ability to zoom. In a zoom measurement, the component of interest is made the center frequency of the analysis, allowing the use of an arbitrarily narrow frequency span.

6.2 FREQUENCY RESOLUTION

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

Frequency resolution in a DSA is determined primarily by the number of filters (or lines of resolution), and the ability to zoom. The filters of a DSA are shown in Figure 6.2-1. Signals must lie in different filters to be resolved, so resolution depends on the spacing of the filters. Since the number of filters is fixed, filter spacing is determined by the number of filters and the analysis span. More filters imply better resolution for a given span.

If the span required for the desired resolution is too narrow to include all the frequencies of interest, then the analysis must start at a frequency above zero. This process is referred to as zooming (because it involves zooming in on an arbitrary center frequency), and is a feature of most newer DSAs. Ideally, the zoom feature should allow frequency spans down to 1 Hz to be centered on any frequency in the analysis range.

The gear spectrum in Figure 6.2-2 illustrates why the ability to zoom is so important. In the low resolution spectrum of (a), the sidebands around the gearmesh frequency indicate a problem, but the exact spacing (which will indicate which gear has the defect) is difficult to determine. Since the gearmesh is at a relatively high frequency, a span narrow enough to resolve the sidebands cannot cover the entire frequency range starting at 0 Hz. Thus we must zoom on the gearmesh frequency to complete the analysis. (See Section 4.6 for more information on gear analysis.)

REAL TIME BANDWIDTH EXAMPLE

Suppose you were making measurements with a 2,000 Hz frequency span. The data sampling time for this span on DSAs with 400 line resolution (see Section 6.2) is 0.200 seconds. If the computation time were also 0.200 seconds, then sampling would never have to stop to let the computation catch up. (This computation time would correspond to a real time bandwidth of 2000 Hz.) If the computation time were 1 second (a real time bandwidth of 400 Hz), the analyzer would miss large amounts of data while waiting for the computation.

Window Functions

Frequency resolution is also affected by the shape of the filters — determined in a DSA by the window function selected. The window function shapes the input data to compensate for discontinuities in the sampling process (see Application Note AN 243). Figure 6.2-3 shows the same vibration spectrum measured with the three windows commonly available on DSAs.

A. The FLAT TOP window is optimized for level accuracy, with a response variation with frequency of 1% (0.1 dB). This is the window to use unless maximum frequency resolution is required, or you are capturing a transient.

B. The HANNING window provides improved frequency resolution (note the Bandwidth notation at the bottom of the display), but sacrifices amplitude accuracy. Variation with frequency is approximately 15% (1.5 dB).

C. The UNIFORM window provides no weighting, and should be used only for transients or specialized signals. The wide skirts, known as leakage, severely restrict frequency resolution. (Leakage is what weighting in the other two window functions eliminates.)

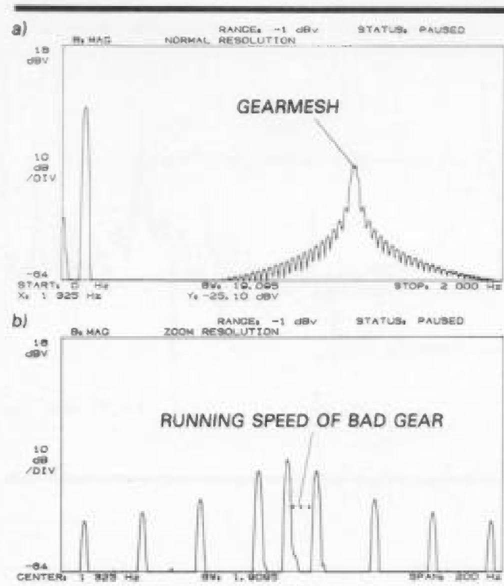


Figure 6.2-2
An example gear vibration spectrum that illustrates the need for zoom. The sidebands around gearmesh often indicate the bad gear, but are too closely spaced to resolve in (a). The zoom measurement in (b) centers a narrower frequency span on the gear mesh, increasing resolution.

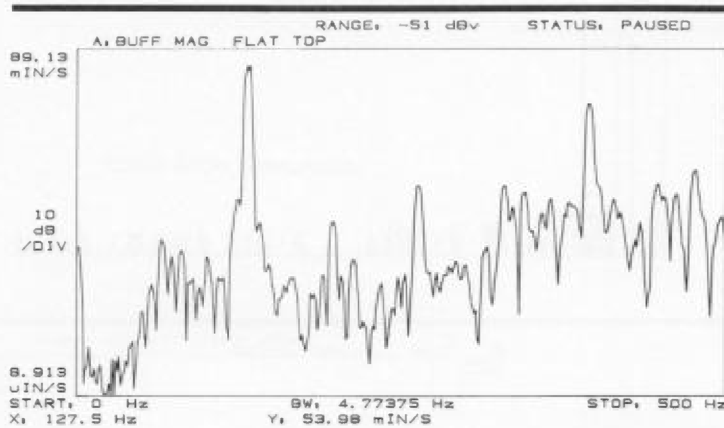


Figure 6.2-3a
The Flat Top window is optimized for amplitude accuracy.

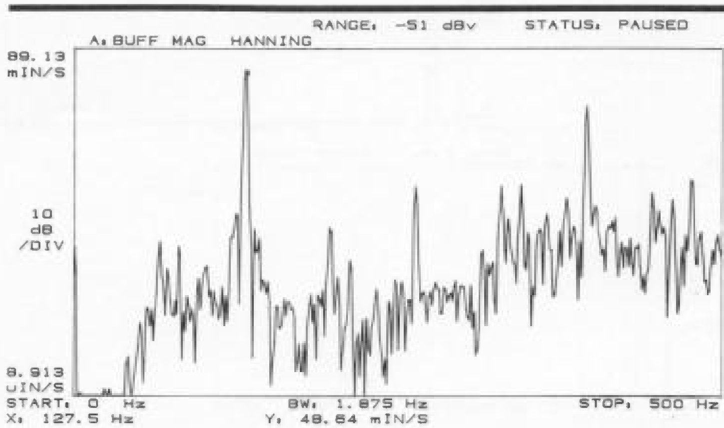


Figure 6.2-3b
The Hanning window provides better frequency resolution than the Flat Top window, but amplitude accuracy is reduced.

Figure 6.2-3c
Leakage (wide filter skirts) is a problem with the Uniform window, and it should only be used for transients or specialized signals.

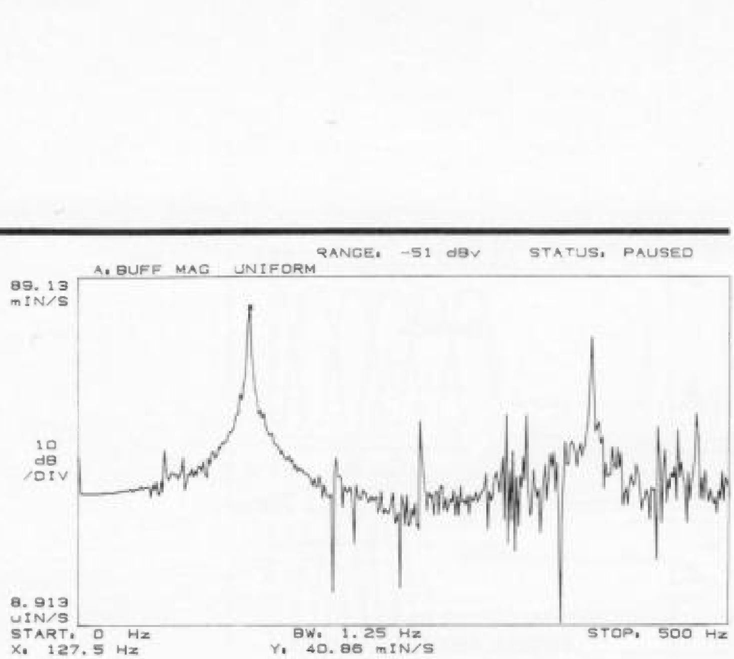


Figure 6.3-1
Dynamic range is defined as the ratio between the largest and smallest signals that can be analyzed at the same time.

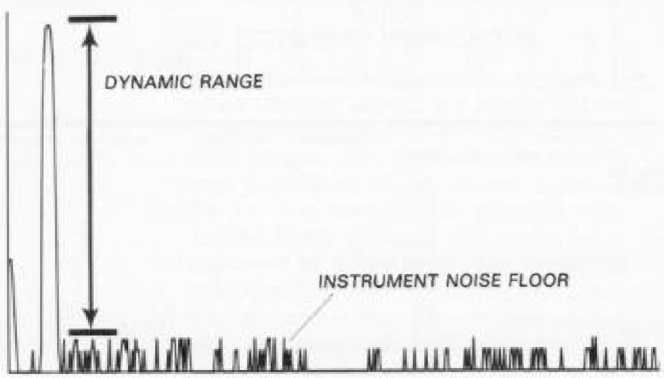
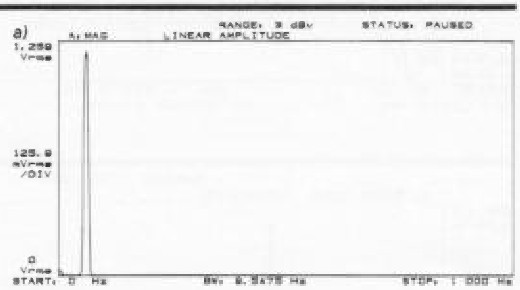
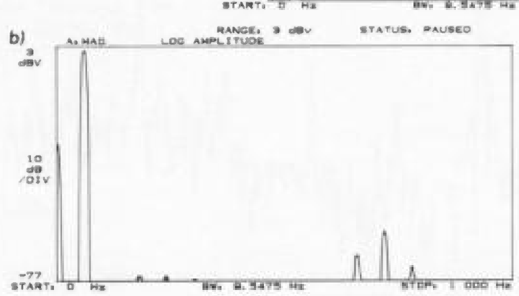


Figure 6.3-2
Bearing characteristic frequencies that are easily seen in the logarithmic (dB) amplitude display of (b) are not visible in the linear amplitude display of (a).

a)



b)



6.3 DYNAMIC RANGE

Dynamic range is another aspect of resolution. It is a measure of the ability to analyze small signals in the presence of large ones, as shown in Figure 6.3-1. DSAs feature wide dynamic range, with most able to display signals that differ in amplitude by factors of 1000 or more. Logarithmic (dB) display scales are used to take advantage of this measurement capability (see the explanation of dB on page 17).

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

This situation is shown in the displays of Figure 6.3-2. Here, an accelerometer is being used to check for bearing wear. As pointed out in Section 2.1, acceleration response increases with frequency, resulting in an accelerometer output that often has a large high-frequency energy content. This energy limits input sensitivity, and can easily result in bearing frequencies 40 dB (a factor of 100) below full scale. These low level signals are clear on the logarithmic display of (b), but not visible on the linear display of (a).

6.4 DIGITAL AVERAGING

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

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

B. TIME. While RMS averaging reduces the variance of signal levels, it does nothing to reduce unwanted background noise. This background noise may mask low level components, or add unrelated components to the spectrum. Time (or synchronous) averaging effectively reduces components that are not related to the trigger, which is usually a keyphasor. Time averaging should be used when background noise or vibration from adjacent machines interferes with analysis.

C. PEAK. It is often desirable to hold peak vibration levels during a run-up or coast-down, or over a period of time. The result of peak averaging is a display of the maximum level at each frequency point (recall that most DSAs display 400 frequency points for any selected span).

RMS Averaging

Because noise can cause spectral components to vary widely in amplitude, a single measurement is not statistically accurate. While watching the components vary in amplitude, you could visually average them and determine the mean level. This is essentially what RMS averaging does, and the more averages you take, the better the accuracy will be. RMS averaging can be thought of as amplitude averaging, since phase is ignored. (RMS, or root mean square, is the square root of the mean of the squared spectra.) The effect of RMS averaging is shown in Figure 6.4-1.

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

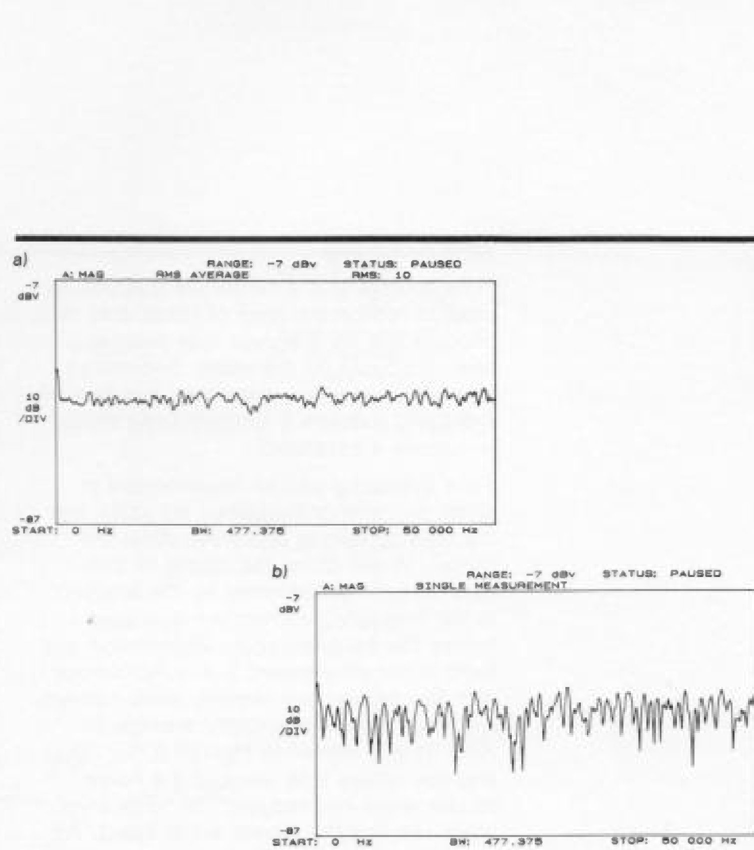


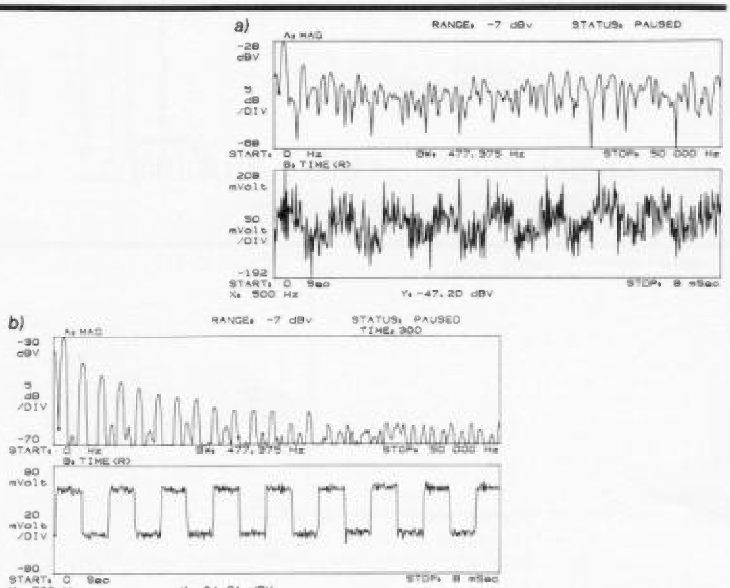
Figure 6.4-1
When RMS averaging is performed, components which vary in amplitude converge to their mean value, providing a better statistical estimate of amplitude.

Time Averaging

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

Time averaging can be implemented in either the time or frequency domains, but the time domain is traditional (thus the name). In this form, the blocks of time data that are transformed by the analyzer to the frequency domain are averaged before the transformation. Signals that are fixed in the time record (i.e. synchronous with the trigger) will remain, while nonsynchronous signals eventually average to zero. This is shown in Figures 6.4-2 (a) and (b), where time averaging a noisy square wave has reduced the noise level, while keeping the square wave intact. An example with a machinery spectrum can be found in Section 5.2.

Figure 6.4-2
The time averaged displays in (b) show a reduction in the level of components that are nonsynchronous with the trigger.



Peak Hold

Peak hold is a function usually grouped with averaging in DSAs. By displaying the maximum level at each frequency over a number of samples, this feature provides a history of peak levels. Two applications are shown in Figure 6.4-3. In (a), peak hold has been used during a machine coast-down, providing a simple track of the maximum level (which is usually 1 × rpm). The display in (b) is a peak hold over a relatively long period that shows the range of speed variation of a nominally constant speed motor. This could be used, for example, as an indication of load variation. Peak hold is also useful for recording momentary vibration peaks (e.g. from start-ups or load changes).

6.5 HP-IB

The Hewlett-Packard Interface Bus (HP-IB) is a standardized interface that can be used to connect digital plotters and computers to DSAs. (A cable and connector are shown in Figure 6.5-1.) Because of the large number of computers, plotters and instruments that are compatible with this interface, the possibilities for automatic data storage, presentation, and analysis are virtually unlimited. Some of these are discussed below.

Digital Plotters

Digital plotters produce high quality copies of DSA displays, complete with annotation. They are no more expensive than their analog counterparts, and many DSAs can interface to them directly, with no controller required. (All of the example DSA plots in this note were made on a digital plotter.) An advantage of these plotters over video display printers is that poor resolution, which results in the same stair-step effect as seen in raster scan displays, is eliminated.

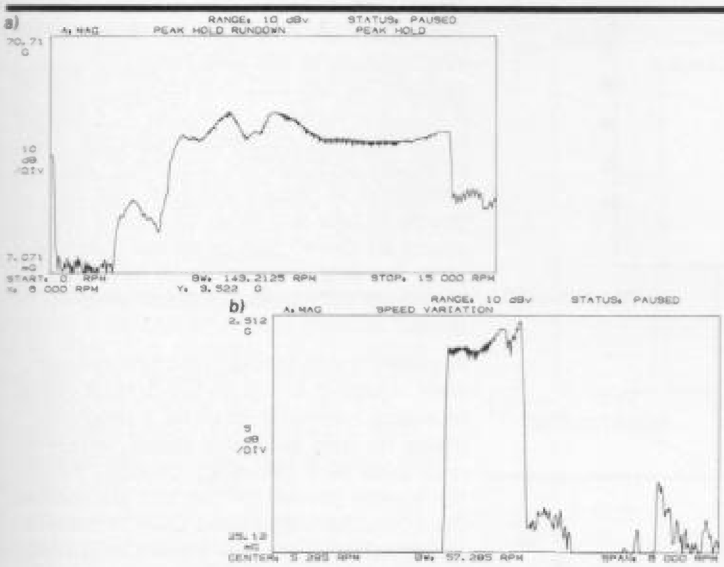


Figure 6.4-3
Peak hold used (a) to track peak level during a coastdown, and (b) to indicate variation in speed over time.

Computer Data Storage and Analysis

A common problem encountered when starting a vibration monitoring program is that large amounts of data must be stored. Plots of vibration spectra can be stored in a file, but several months of data can result in a rather cumbersome file. A much better solution is to use a computer to store the data on tape or disc. Data stored in this fashion can be automatically recalled for comparison purposes or further analysis. Once vibration data is in the computer, it can be analyzed and displayed in virtually any way you desire. Several companies offer software that is specifically designed for storage and analysis of machinery vibration data.

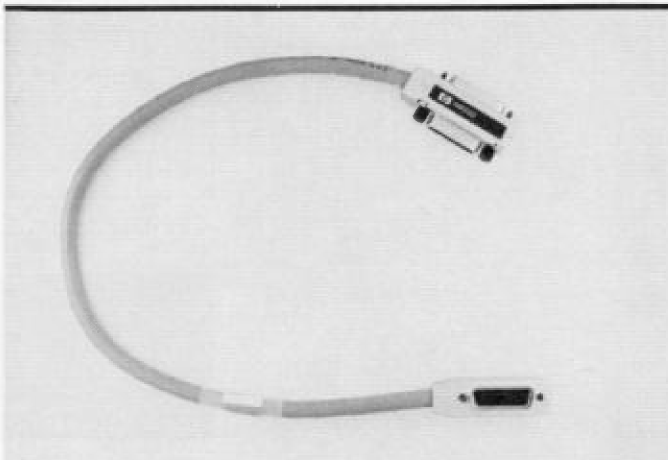


Figure 6.5-1
A standardized cable and connector simplify HP-IB interface connections.

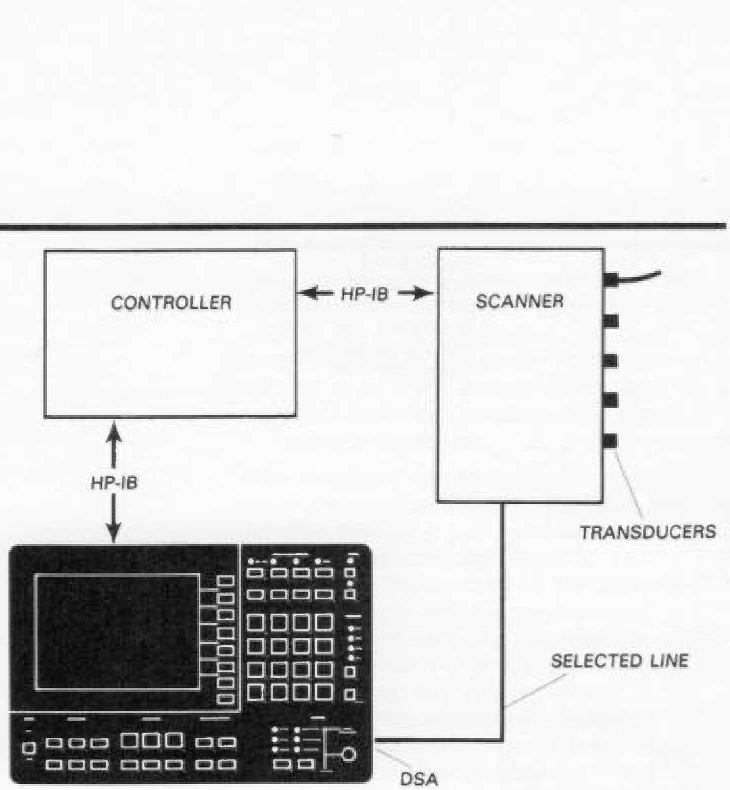


Figure 6.5-2
HP-IB system for scanning a number of transducers. Such a system can be configured to take appropriate action (e.g. sound an alarm or remove power) when current vibration level exceeds pre-defined limits.

Instrument Systems

The variety of compatible HP-IB instruments make possible automatic systems for test and monitoring. In Figure 6.5-2, a scanner is connected to a number of permanently installed transducers. The computer switches the scanner to each transducer at regular intervals, and can decide to take a number of actions (e.g. sound an alarm, shut down the machine, store the data) depending on how the data compares with stored vibration severity limits.

A system for automatic machine run-up tests is shown in Figure 6.5-3(a). A digital to analog converter provides a programmable dc level to control speed, which is monitored by a frequency counter. When the speeds desired for the test are reached, the computer triggers the DSA to make a measurement. The plot from such a setup is shown in Figure 6.5-3(b).

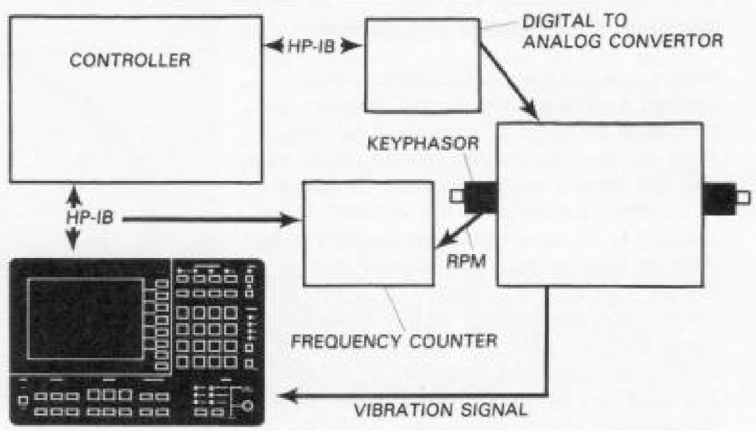


Figure 6.5-3a
An instrument system for performing run-up tests automatically.

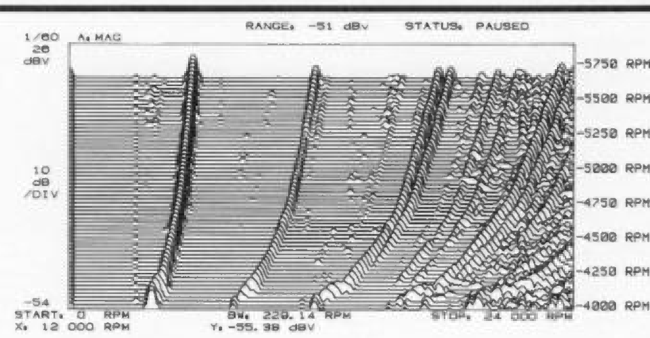


Figure 6.5-3b
A spectral map made with the system of (a).

6.6 USER UNITS AND UNITS CONVERSION

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

User units calibration is accomplished by entering a calibration factor, such as 10 mV/g. The DSA performs the conversion and displays the vibration spectrum in the desired units, usually labeled as "EU" (Engineering Units). Some newer DSAs have provision for custom labeling of user-defined units (e.g. in/sec, g).

The frequency units used for machinery vibration analysis include Hertz, rpm, and orders. Orders refer to "orders of rotation", and are harmonics of the rotating speed. Figure 6.6-1 illustrates calibration of the frequency axis in orders. Orders are handy for analysis because most vibration problems are order-related. By using external sample control, orders can be fixed on the display while speed changes (see Section 6.7).

Referring to the formulas for displacement, velocity, and acceleration in Section 2.1, it should be apparent that they are related by frequency and a phase shift. For example, acceleration can be converted to velocity through division by $j(2\pi f)$. This operation is commonly referred to as artificial integration (the "j" term is an operator that implies a 90° phase shift), and is a feature of most DSAs. Figure 6.6-2 shows an integrated acceleration spectrum overlaid on an actual velocity spectrum measured at the same point.

Table 6.6 summarizes vibration parameter conversion. Two things to note about these

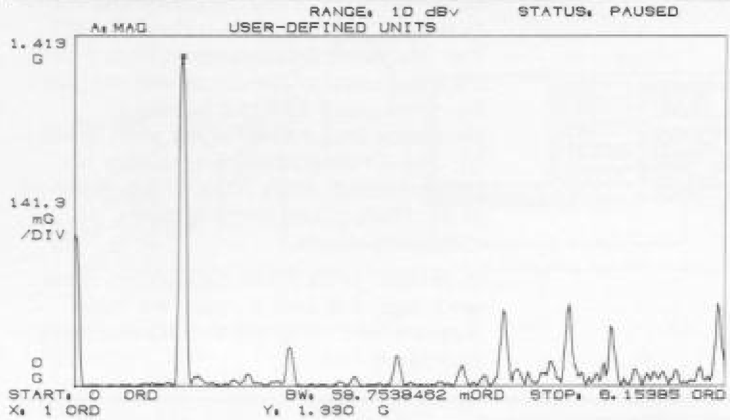


Figure 6.6-1
A machinery vibration spectrum calibrated in user-defined amplitude units (EU), and orders (harmonics) of rotational speed.

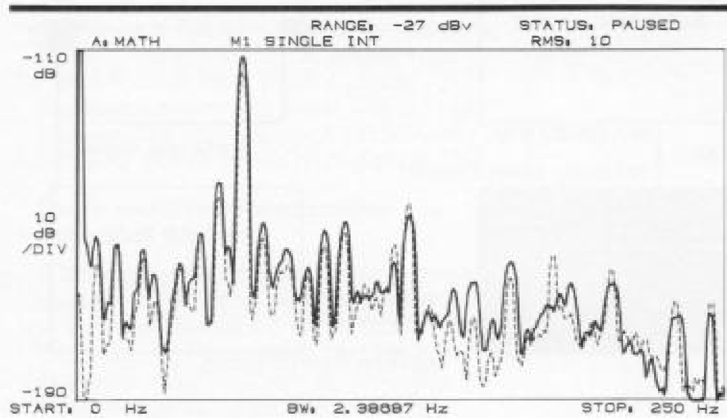


Figure 6.6-2
A comparison between an integrated acceleration spectrum and an actual velocity spectrum (dashed line).

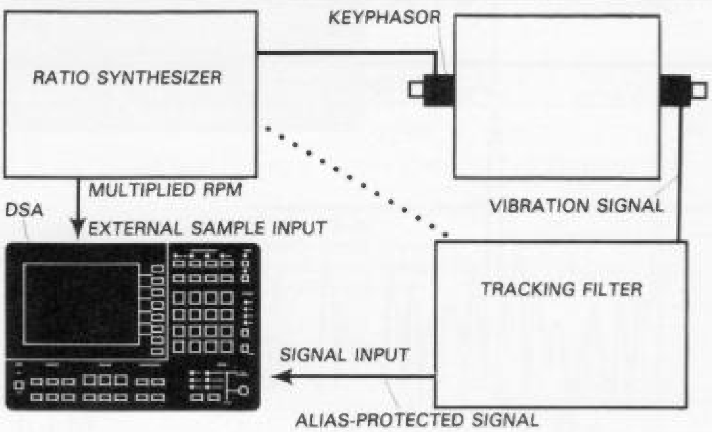
CONVERSION	OPERATOR	DESCRIPTION
Acceleration → velocity	$1/j\omega$	Single integration
Acceleration → displacement	$-1/\omega^2$	Double integration
Velocity → displacement	$1/j\omega$	Single integration
Velocity → acceleration	$j\omega$	Differentiation
Displacement → velocity	$j\omega$	Differentiation
Displacement → acceleration	$-\omega^2$	Dbl. differentiation

Table 6.6
Vibration Parameter Conversions

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

In general, while these conversions often work well, it is best to make the initial measurement using the desired end-result parameter.

Figure 6.7
Instrumentation setup for controlling sample rate externally.



6.7 EXTERNAL SAMPLE CONTROL

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

The problem of wide speed variation can be solved after the data is collected with a spectral map plot. The frequency display can be normalized (i.e. calibrated in fixed location orders of rotation) through software manipulation. However, the only way to normalize the frequency display in real time is with external sample control. By controlling data sampling rate with a signal tied to rotating speed, the display will have a fixed calibration in orders of rotation. (See Section 5.4 for a discussion of the relative merits of each method.)

Figure 6.7 shows the instrumentation required for controlling sample rate externally. Typically, a once/revolution pulse multiplied by a ratio synthesizer is used for sample control. The ratio synthesizer is required because DSAs typically sample at a rate of 2.56 times the frequency span. Since it is usually desirable to look at several orders, the once per revolution tach pulse must be multiplied by 2.56 times the number of orders to be analyzed. An important requirement for the ratio synthesizer is anti-aliasing protection. Aliasing occurs when the data sample rate is too slow, allowing high frequency signals to be misinterpreted as low frequency signals. Thus the spectrum may appear to have components that are not really there. Aliasing is avoided if a filter is used to limit input signals to frequencies less than ½ the sample rate. (See Hewlett-Packard Application Note AN 243 for more information on aliasing.)

If the speed range of the analysis is limited to approximately 20%, aliasing can be avoided by using high ratio synthesizer factors. Hewlett-Packard DSAs use a fixed frequency low pass filter on the input to provide aliasing protection for the broadest frequency span (highest sample rate). If the ratio synthesizer factor is adjusted so that the sample rate at the highest machine speed is equal to the analyzer's highest sample rate, the input low pass filter will provide alias protection. This protection ranges from complete at the maximum speed to very little at around 80% of the maximum speed.

For example, if the maximum machine speed is 3600 rpm (60 Hz), and the maximum analyzer sample rate is 256 kHz, the required ratio synthesizer factor for a once per revolution tach pulse is 256 kHz/60 Hz = 4266.7.

As machine speed is reduced from the maximum (and sample rate thus decreases), alias protection is reduced. At a speed of 80% of maximum, this results in alias components typically 20 dB below full scale (10% of full scale).

6.8 DUAL-CHANNEL ENHANCEMENTS

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

Real Time Comparisons

Comparative phase measurements are a powerful tool for analysis, especially for differentiating between similar forms of vibration (see Sections 4.4 and 5.2). This measurement is made both easier and more accurate with a dual-channel DSA. Referring to the motor-pump combination in Figure 6.8-1, suppose that you are not sure whether the high vibration level is due to imbalance or misalignment. As pointed out in Section 4.4, the relative phase of axial vibration at A and B will be 180° if misalignment is the problem (assuming a rigid-rotor). With a single-channel analyzer, you would use a key phasor as a reference, and measure the two ends one at a time. With a dual-channel analyzer, all you have to do is connect an end to each channel and measure the transfer function phase. (This connection is diagrammed in Figure 6.8-1.) Thus, relative phase measurements can be made with a single-channel DSA, but are much easier (and less error-prone) with a dual-channel DSA.

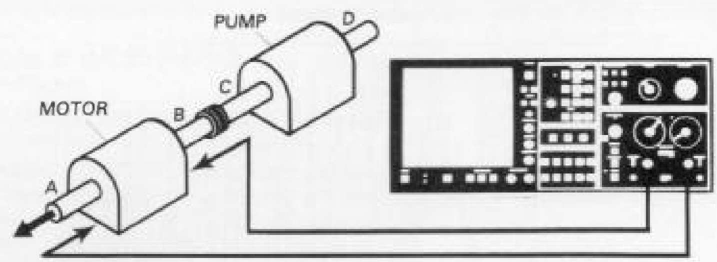


Figure 6.8-1
Misalignment is indicated by a 180° phase relation between A and B. For tranducers oriented as shown (180° relation), the relative phase will be 0°.

Cause and Effect Relationships: The Coherence Function

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

The coherence display covers a range of 0 to 1, and indicates the percentage of power in channel B that is coherent with channel A. Let's suppose that vibration

Figure 6.8-2
Coherence measured between a pump and motor clearly indicates which components are unrelated.

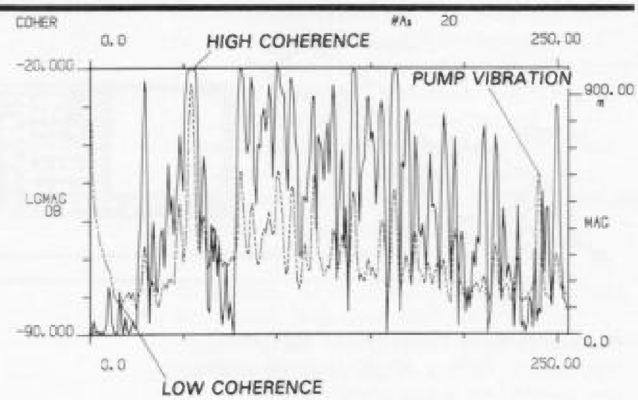


Figure 6.8-3
The transfer function of a gearbox can be measured with an instrumented hammer and a two-channel DSA.

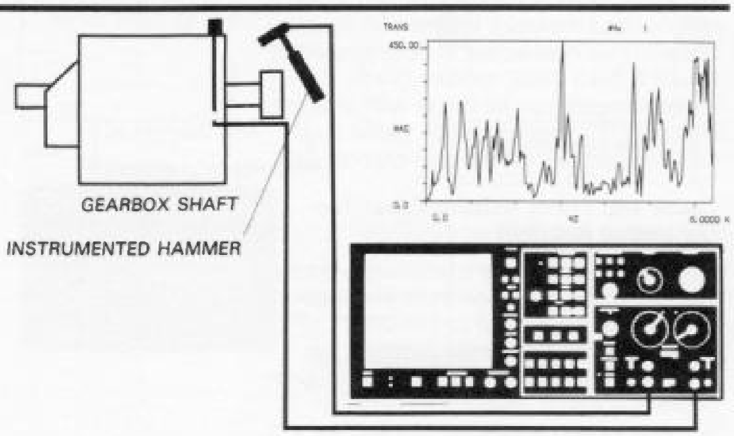
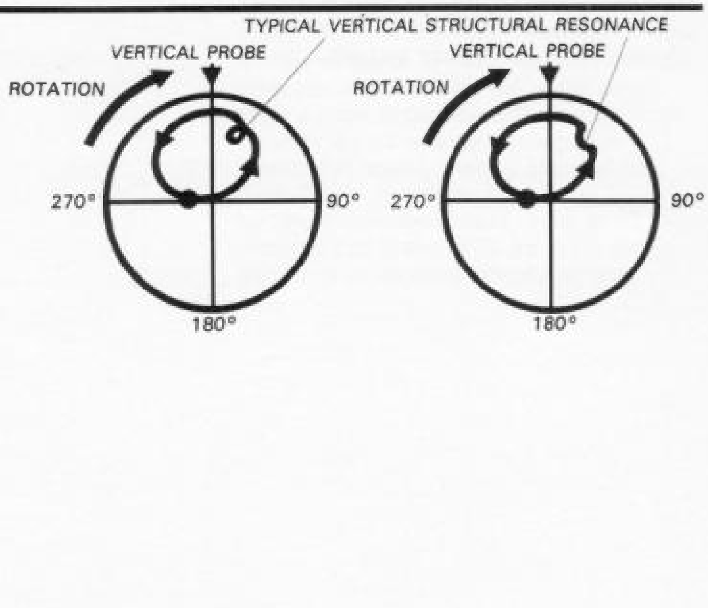


Figure 6.8-4
The orbit capability of some two-channel DSAs provides insight into rotor motion in machines with fluid-film bearings.



levels at points A and D on the motor-pump combination of Figure 6.8-1 are similar, and rather high. You would like to know whether they are independent or related. A low value of coherence between vibration components from A and D indicates that they are not related. A high coherence value for a component implies that there may be a causal relationship. (The high coherence component could, for example, be from a third source of vibration.) Coherence measured between end points on a motor and pump is shown in Figure 6.8-2. Note that coherence is high for all the major vibration components except 240 Hz, indicating that this vibration is not from the motor. The technique will not work 100% of the time, and you will have to get a feel for what constitutes a high level of coherence, but it may save time in disconnecting machines to isolate the source of vibration.

Natural Frequency Measurements

The natural frequencies of a machine housing or foundation can be easily determined through what is sometimes referred to as a "bump" test. Recall from Figure 3.2-4 that impulsive signals produce a broad spectrum of harmonics. If the housing is impacted with sufficient energy (typically with a block of wood), all the natural frequencies will be excited. The response can be measured with a single-channel DSA, but an imperfect impact may result in a misleading spectrum.

A better way to make this measurement is with a dual-channel analyzer and an instrumented hammer. This is shown diagramatically in Figure 6.8-3, where the natural frequencies of a gearbox are being determined. As we saw in Section 4.6, gear defects often show up at their natural frequencies, so this information is valuable. Hewlett-Packard Application Note AN 243 contains more detailed information on measuring the response of mechanical structures.

Orbits

Some dual-channel DSAs have the ability to display orbit diagrams, as shown in Figure 6.8-4. These are useful for gaining insight into rotor motion in turbomachinery. The subject of orbit interpretation is covered well in reference 29.

GLOSSARY

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

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

ALIASING. A phenomenon which can occur whenever a signal is not sampled at greater than twice the maximum frequency component. Causes high frequency signals to appear at low frequencies. Aliasing is avoided by filtering out signals greater than ½ the sample rate.

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

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

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

ANTI-ALIASING FILTER. A low-pass filter designed to filter out frequencies higher than ½ the sample rate in order to prevent aliasing.

ANTI-FRICTION BEARING. See ROLLING ELEMENT BEARING.

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

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

ATTITUDE ANGLE (STEADY-STATE). The angle between the direction of steady-state preload through the bearing centerline, and a line drawn between the shaft centerline and the bearing centerline. (Applies to fluid-film bearings.)

AUTO SPECTRUM (POWER SPECTRUM). DSA spectrum display whose magnitude represents the power at each frequency, and which has no phase. RMS averaging produces an auto spectrum.

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

AXIAL. In the same direction as the shaft centerline.

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

BALANCE RESONANCE SPEED(S). A rotative speed that corresponds to a natural resonance frequency.

BALANCED CONDITION. For rotating machinery, a condition where the shaft geometric centerline coincides with the mass centerline.

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

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

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

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

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

BLOCK SIZE. The number of samples used in a DSA to compute the Fast Fourier Transform. Also the number of samples in a DSA time display. Most DSAs use a block size of 1024. Smaller block size reduces resolution.

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

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

BOW. A shaft condition such that the geometric centerline of the shaft is not straight.

BRINNELING (FALSE). Impressions made by bearing rolling elements on the bearing race; tyically caused by external vibration when the shaft is stationary.

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

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

CASCADE PLOT. See SPECTRAL MAP.

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

CENTER FREQUENCY. For a bandpass filter, the center of the transmission band.

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

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

CONSTANT BANDWIDTH FILTER. A band-pass filter whose bandwidth is independent of center frequency. The filters simulated digitally in a DSA are constant bandwidth.

CONSTANT PERCENTAGE BANDWIDTH. A band-pass filter whose bandwidth is a constant percentage of center frequency. ⅓ octave filters, including those synthesized in DSAs, are constant percentage bandwidth.

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

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

CRITICAL SPEED MAP. A rectangular plot of system natural frequency (y-axis) vs bearing or support stiffness (x-axis).

CROSS AXIS SENSITIVITY. A measure of off-axis response of velocity and acceleration transducers.

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

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

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

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

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

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

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

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

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

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

DSA. See DYNAMIC SIGNAL ANALYZER.

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

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

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

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

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

ECCENTRICITY RATIO. The vector difference between the bearing centerline and the average steady-state journal centerline.

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

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

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

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

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

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

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

FLAT TOP FILTER. DSA window function which provides the best amplitude accuracy for measuring discrete frequency components.

FLUID-FILM BEARING. A bearing which supports the shaft on a thin film of oil. The fluid-film layer may be generated by joural rotation (hydrodynamic bearing), or by externally applied pressure (hydrostatic bearing).

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

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

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

FREQUENCY RESPONSE. The amplitude and phase response characteristics of a system.

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

GEAR MESH FREQUENCY. A potential vibration frequency on any machine that contains gears; equal to the number of teeth mutiplied by the rotational frequency of the gear.

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

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

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

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

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

HIGH-PASS FILTER. A filter with a transmission band starting at a lower cutoff frequency and extending to (theoretically) infinite frequency.

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

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

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

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

INFLUENCE COEFFICIENTS. *Mathematical coefficients that describe the influence of system loading on system deflection.*

INTEGRATION. *A process producing a result that, when differentiated, yields the original quantity. Integration of acceleration, for example, yields velocity. Integration is performed in a DSA by dividing by $j\omega$, where ω is frequency multiplied by 2π. (Integration is also used to convert velocity to displacement.)*

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

KEYPHASOR. *A signal used in rotating machinery measurements, generated by a transducer observing a once-per-revolution event. The keyphasor signal is used in phase measurements for analysis and balancing. (Keyphasor is a Bently Nevada trade name.)*

LATERAL LOCATION. *The definition of various points along the shaft axis of rotation.*

LATERAL VIBRATION. *See* RADIAL VIBRATION.

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

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

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

LINEAR AVERAGING. *See* TIME AVERAGING.

LOW-PASS FILTER. *A filter whose transmission band extends from dc to an upper cutoff frequency.*

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

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

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

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

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

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

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

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

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

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

NYQUIST CRITERION. *Requirement that a sampled system sample at a frequency greater than twice the highest frequency to be measured.*

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

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

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

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

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

PEAK HOLD. *In a DSA, a type of averaging that holds the peak signal level for each frequency component.*

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

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

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

POLAR PLOT. *Polar coordinate representation of the locus of the 1× vector at a specific lateral shaft location with the shaft rotational speed as a parameter.*

POWER SPECTRUM. *See* AUTO SPECTRUM.

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

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

PROXIMITOR. *See* OSCILLATOR/DEMODULATOR.

RADIAL. *Direction perpendicular to the shaft centerline.*

RADIAL POSITION. *The average location, relative to the radial bearing centerline, of the shaft dynamic motion.*

RADIAL VIBRATION. *Shaft dynamic motion or casing vibration which is in a direction perpendicular to the shaft centerline.*

REAL TIME ANALYZER. *See* DYNAMIC SIGNAL ANALYZER.

REAL TIME RATE. *For a DSA, the broadest frequency span at which data is sampled continuously. Real time rate is mostly dependent on FFT processing speed.*

RECTANGULAR WINDOW. *See* UNIFORM WINDOW.

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

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

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

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

ROLLING ELEMENT BEARING. Bearing whose low friction qualities derive from rolling elements (balls or rollers), with little lubrication.

ROOT MEAN SQUARE (RMS). Square root of the arithmetical average of a set of squared instantaneous values. DSAs perform rms averaging digitally on successive vibration spectra.

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

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

RPM SPECTRAL MAP. A spectral map of vibration spectra versus rpm.

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

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

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

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

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

SLOW ROLL SPEED. Low rotative speed at which dynamic motion effects from forces such as imbalance are negligible.

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

SPECTRUM ANALYZER. An instrument which displays the frequency spectrum of an input signal.

STIFFNESS. The spring-like quality of mechanical and hydraulic elements to elastically deform under load.

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

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

SUBSYNCHRONOUS. Component(s) of a vibration signal which has a frequency less that shaft rotative frequency.

TIME AVERAGING. In a DSA, averaging of time records that results in reduction of asynchronous components.

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

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

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

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

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

TRANSVERSE SENSITIVITY. See CROSS-AXIS SENSITIVITY.

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

UNBALANCE. See IMBALANCE.

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

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

WATERFALL PLOT. See SPECTRAL MAP.

REFERENCES

HEWLETT-PACKARD APPLICATION NOTES:

243 THE FUNDAMENTALS OF SIGNAL ANALYSIS.
The time, frequency, and modal domains are explained without rigorous mathematics. Provides a block-diagram level understanding of DSAs.

245-1 SIGNAL AVERAGING WITH THE HP 3582A SPECTRUM ANALYZER.
Provides an understanding of the signal averaging techniques commonly used in DSAs, and how they can be used to improve accuracy and signal-to-noise ratio.

245-2 MEASURING THE COHERENCE FUNCTION WITH THE HP 3582A SPECTRUM ANALYZER.
A theoretical and practical introduction to the coherence function, and its uses for indicating causality. A discussion of theoretical details is included.

MACHINERY MONITORING:

1. Dodd, V.R. and East, J.R., "The Third Generation of Vibration Surveillance," Minicourse notes, Machinery Monitoring and Analysis Meeting, Vibration Institute, Clarendon Hills, IL 1983.

2. Dodd, V. R., "Machinery Monitoring Update," Sixth Turbomachinery Symposium Proceedings, Texas A&M University, 1977.

3. Mitchell, John S., MACHINERY ANALYSIS AND MONITORING, Penn Well Books, Tulsa, OK, 1981.

4. Myrick, S. T., "Survey Results on Condition Monitoring of Turbomachinery in the Petrochemical Industry; I. Protection and Diagnostic Monitoring of 'Critical' Machinery", Vibration Institute, 1982.

5. Tiedt, Brian, "Economic Justification for Machinery Monitoring", Bently Nevada Publication L0377-00.

6. Wett, Ted, "Compressor Monitoring Protects Olefins Plant's Reliability," Bently Nevada Publication L0339-00.

7. Stewart, R.M., "The Specification and Development of a Standard for Gearbox Monitoring," Vibrations in Rotating Machinery, Mechanical Engineering Publications Limited, Inc., London, 1980.

TRANSDUCERS:

8. Bently, Donald, "Shaft Vibration Measurement and Analysis Techniques," Noise and Vibration Control International, April, 1983.

9. Dranetz, Abraham I. and Orlacchio, Anthony W., "Piezoelectric and Piezoresistive Pickups," in Shock and Vibration Handbook, C.M. Harris and C.E. Crede, eds., McGraw-Hill, 1976.

10. "Glitch: Definition of and Methods for Correction, including Shaft Burnishing to Remove Electrical Runout," Bently Nevada Application Note L0195-00, August 1978.

11. "How to Minimize Electrical Runout During Rotor Manufacturing," Bently Nevada Application Note L0197-00, July 1969.

12. Judd, John, "Noise in Vibration Monitoring," Measurements and Control, June 1983.

13. "REBAM (TM) — A Technical Review," Bently Nevada Publication, 5/83.

14. Stuart, John W., "Retrofitting Gas Turbines and Centrifugal Compressors with Proximity Vibration Probes," Bently Nevada Publication L0357-00, June 1981.

15. Wilson, Jon, "Noise Suppression and Prevention in Piezoelectric Transducer Systems," Sound and Vibration, April 1979.

VIBRATION ANALYSIS:

16. Ehrich, F.F., "Sum and Difference Frequencies in Vibration of High Speed Rotating Machinery," Journal of Engineering for Industry, February 1972.

17. Eshleman, Ronald L., "The Role of Sum and Difference Frequencies in Rotating Machinery Fault Analysis," Vibrations in Rotating Machinery, Mechanical Engineering Publications Limited, Inc., London, 1980.

18. Jackson, Charles, THE PRACTICAL VIBRATION PRIMER, Gulf Publishing Company, Houston, Texas, 1979.

19. Stewart, R.M., "Vibration Analysis As an Aid to the Detection and Diagnosis of Faults in Rotating Machinery," I Mech E, C192/76, 1976.

20. Maxwell, J. Howard, "Induction Motor Magnetic Vibration," Proceedings of the Vibration Institute Machinery Vibration Monitoring and Analysis Meeting, Houston, Texas, April 1983, Vibration Institute, Clarendon Hills, IL.

21. Taylor, James I., "An Update of Determination of Antifriction Bearing Condition by Spectral Analysis," Vibration Institute, 1981.

22. Taylor, James I., "Identification of Gear Defects by Vibration Analysis," Vibration Institute, 1979.

FLUID FILM BEARINGS AND ROTOR DYNAMICS:

23. Rieger, N.F. and Crofoot, J.F., VIBRATIONS OF ROTATING MACHINERY, Vibration Institute, 1977.

24. Bently, Donald E., "Oil Whirl Resonance," Bently Nevada Publication L0324-01, July, 1981.

25. Ehrich, E.F., "Identification and Avoidance of Instabilities and Self-Excited Vibrations in Rotating Machinery," ASME Paper 72-DE-21.

26. Gunter, E.J., "Rotor Bearing Stability," Vibration Institute, 1983.

27. Loewy, R.G. and Piarulli, V.J., DYNAMICS OF ROTATING SHAFTS, The Shock and Vibration Information Center, 1969.

28. McHugh, J.D., "Principles of Turbomachinery Bearings," Proceedings of the 8th Turbomachiney Symposium, Texas A&M University.

29. "Orbits," Bently Nevada Applications Notes.

VIBRATION CONTROL:

30. Beranek, L.L., NOISE AND VIBRATION CONTROL, McGraw-Hill Book Co., New York, NY, 1971.

31. "Fundamentals of Balancing," Schenck Trebel Corporation, Deer Park, NY, 1983.

32. Dodd, V.R., TOTAL ALIGNMENT, Penn Well Books, Tulsa, OK, 1975.

33. Gunter, E.J., Ed., FIELD BALANCING OF ROTATING EQUIPMENT, Vibration Institute, 1983.

34. Hagler, R., Schwerdin, H., and Eshleman, R., "Effects of Shaft Misalignment on Machinery Vibration," Design News, January, 1979.

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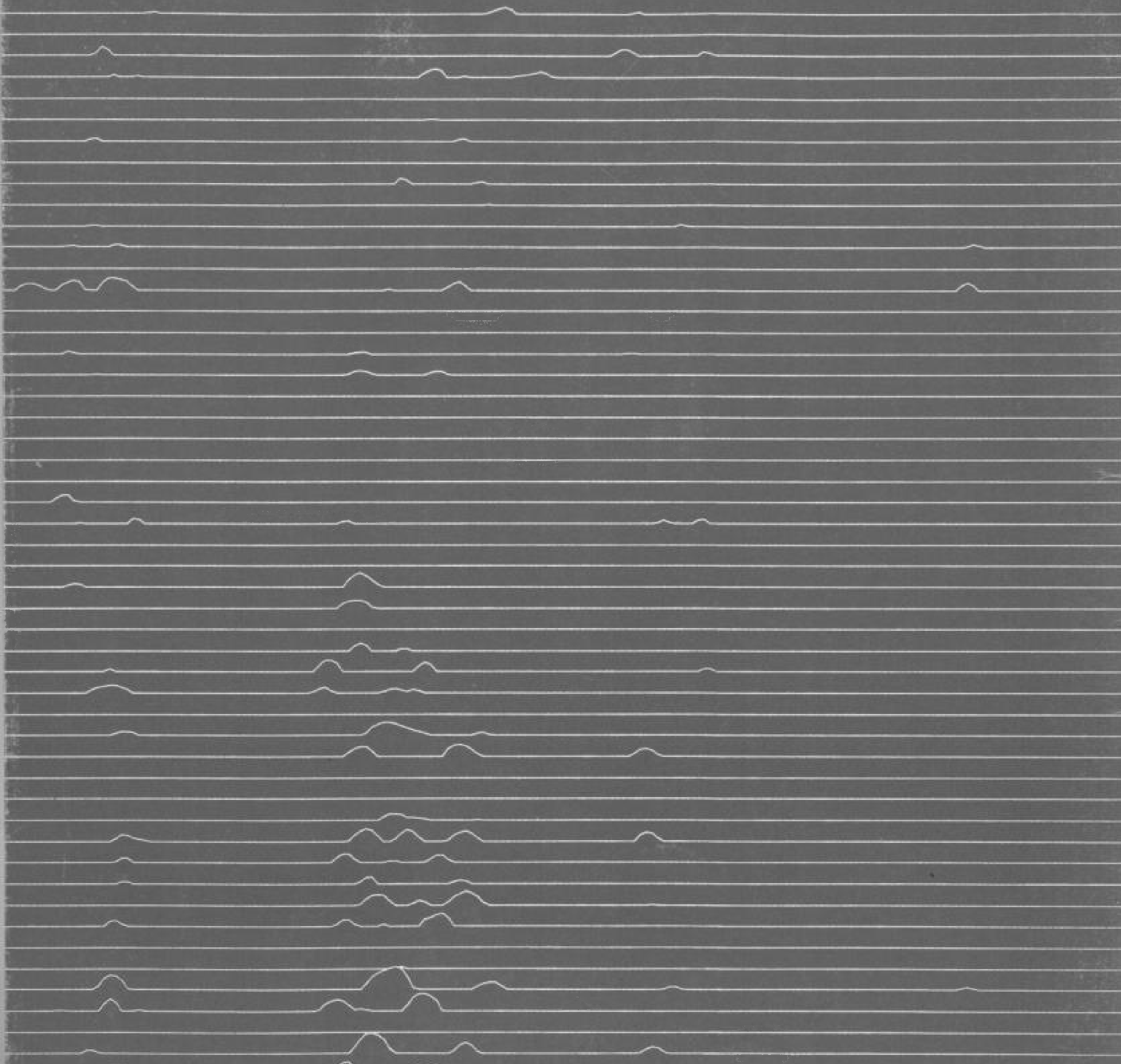

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