The Vibration Analysis Handbook

A Practical Guide for Solving Rotating Machinery Problems

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THE VIBRATION ANALYSIS HANDBOOK

First Edition
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CHAPTER ONE:
INTRODUCTION TO MACHINERY VIBRATION

THEORY OF VIBRATION

The physical movement or motion of a rotating machine is normally referred to as vibration. Since the vibration frequency and amplitude cannot be measured by sight or touch, a means must be employed to convert the vibration into a usable product that can be measured and analyzed. Electronics, mechanics, and chemical physics are closely related. Therefore, it would logically follow that the conversion of the mechanical vibration into an electronic signal is the best solution. The means of converting the mechanical vibration into an electronic signal is called a transducer. The transducer output is proportionate to how fast the machine is moving (frequency) and how much the machine is moving (amplitude). The frequency describes what is wrong with the machine and the amplitude describes relative severity of the problem. The motion can be harmonic, periodic, and/or random. All harmonic motion is periodic. However, all periodic motion is not harmonic. Random motion means the machine is moving in an unpredictable manner.

Harmonic Motion

Harmonic motion is characteristically a sinusoid or some distorted version, depending upon the harmonic content, as in Fig. 1-1. All harmonic motion is periodic, meaning it repeats at some point in time. In a linear system, imbalance in rotating equipment could generate harmonic motion. However, with many variables such as gear problems, looseness, bearing defects, misalignment, etc., such sinusoids are not often found. It is important to understand that a sine wave is simply a plot of a circle against time. Notice how the circle in Fig. 1-1 can be plotted as a sine wave, proving that linear motion is harmonic. All harmonic motion is repeatable and is just one form of periodic motion.

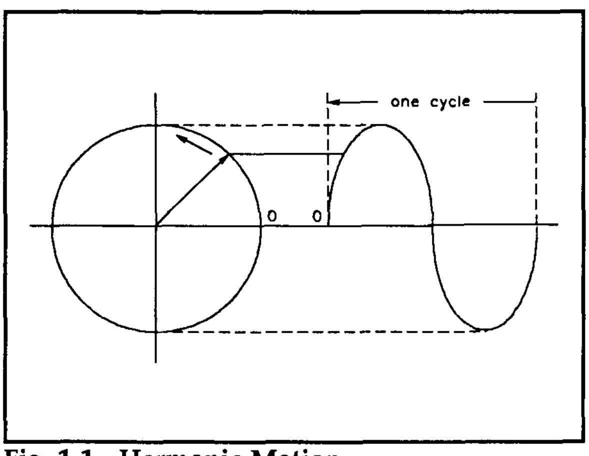


Fig. 1-1. Harmonic Motion.

Periodic Motion

Periodic motion is all motion that repeats periodically. This includes harmonic motion, pulses, etc. Periodic motion is any motion that repeats itself in equal time periods. For example, a misaligned motor coupling that is loose could have a bump once per revolution of the shaft. Although this motion is not harmonic, it is periodic. The time signal will have one pulse every x seconds as indicated in Fig. 1-2.

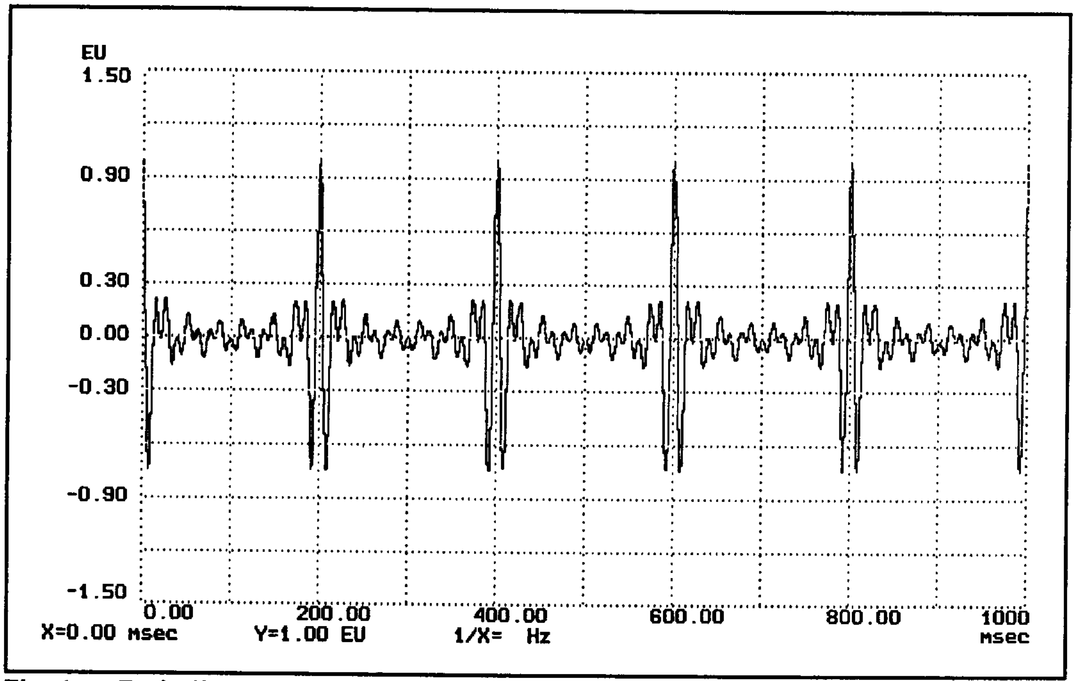


Fig. 1-2. Periodic Motion.

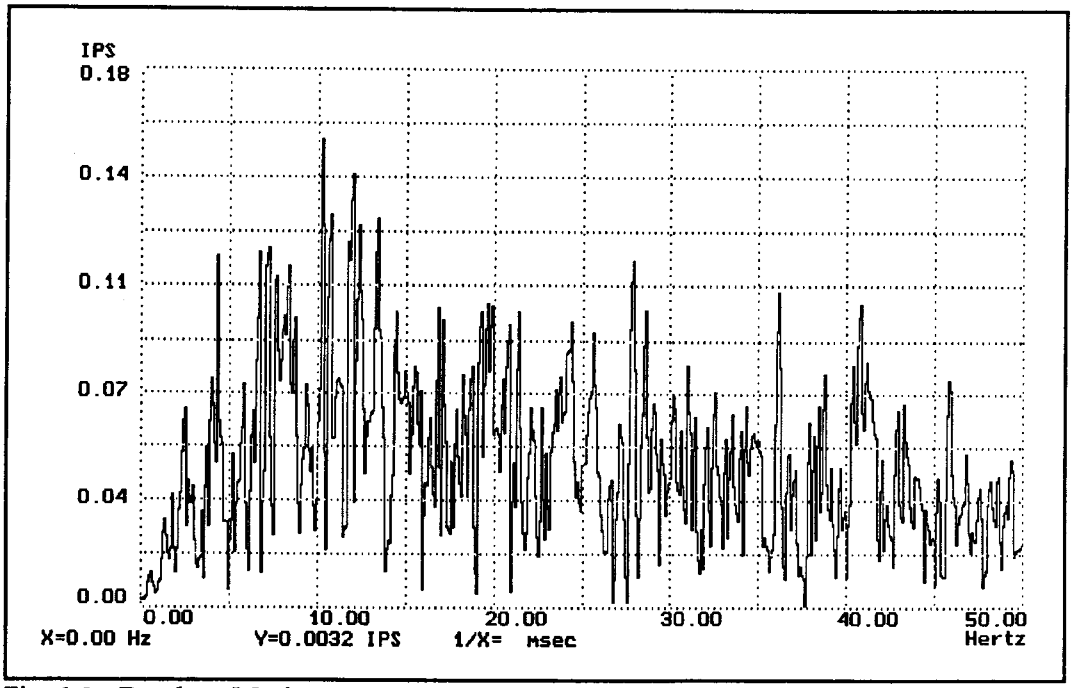


Fig. 1-3. Random Motion.

Random Motion

Random motion occurs in an erratic manner and contains all frequencies in a particular frequency band. Random motion is any motion that is not repeatable. Popcorn in a popper, rain hitting a roof, and bowling pins being knocked over are examples. Random motion is also called noise. When random noise is generated by a machine, a recording of the noise played back ten times faster than it was recorded can sound like a TV set after the station has signed off the air. A time signal of random noise will contain all frequencies in a given range. The frequency spectra from such time signals will be up off the baseline as indicated in Fig. 1-3. Often, random motion in a machine is caused by severe looseness.

THE RELATIONSHIP BETWEEN TIME AND FREQUENCY

Time

When we say that AC line frequency is 60 cycles per second, this means if a one second time period was observed, 60 cycles would be present as indicated in Fig. 1-4. However, it is not always practical to observe one second of time and count the number of cycles.

We can measure the time period for one cycle and calculate the frequency. We can also calculate the time period for one cycle if the frequency is known. Time and frequency are the reciprocal of each other. For example, if 60 cycles occur in one second, divide one by 60 to get the time period for one cycle. When determining the frequency from the time period for one cycle, divide the time period for one cycle into one (1):

$$F = \frac{1}{T} \quad F = \frac{1}{0.0167} \quad or\ 60$$

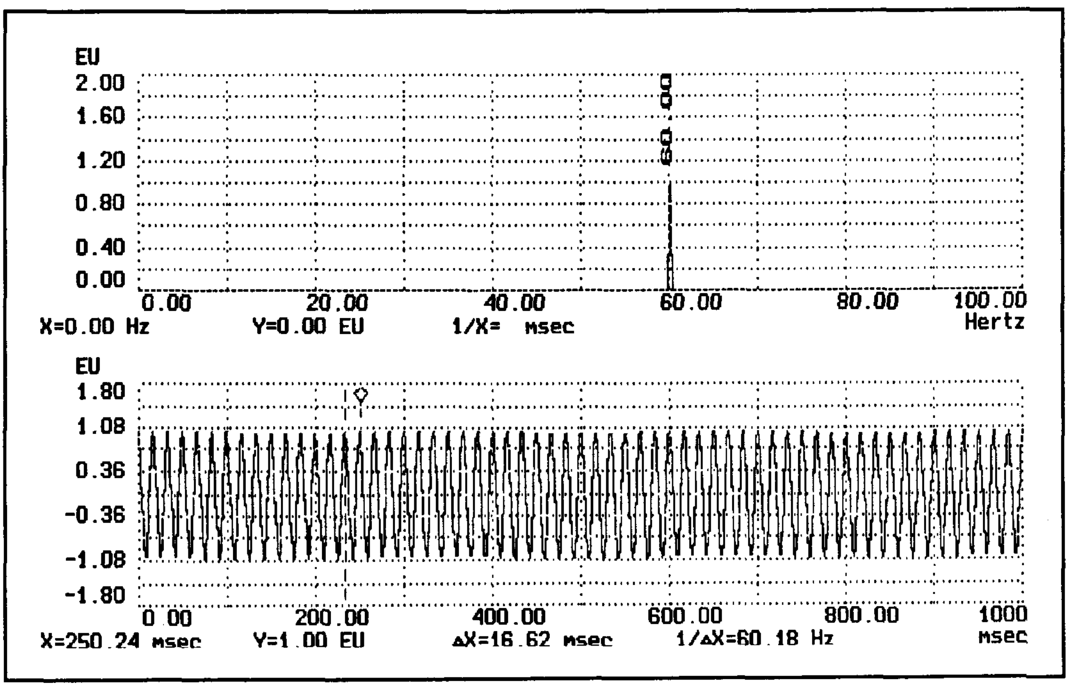


Fig. 1-4. Time and Frequency.

If 60 cycles occur in one second and the time period for one cycle is 0.0167 seconds, the calculation can be verified by: $F \times T = 1$ *or* $60 \times 0.0167 = 1$. Please note that the time period for one cycle of all frequencies above 1 Hz will be less than one second. Also note that if frequency is in cycles per second, time must be measured in seconds (generally fractions of a second).

Frequency

Frequency is the number of cycles that occur in one time period, usually one second. Until a few years ago, frequency was identified as cycles per second (CPS). CPS was changed to Hertz, honoring the man who developed the frequency theory. Today Hertz (cycles per second) is the standard measurement of frequency. Machine speed is measured in revolutions per minute (RPM), but the frequencies generated by those machines are measured in Hertz.

From the above discussion, the formulas for frequency and time can be derived:

$$F = \frac{1}{T} \qquad T = \frac{1}{F} \qquad FT = 1$$

For the beginner, it may be helpful to construct a triangle such as in Fig. 1-5. To solve for 1, F, or T, simply cover the unknown and the formula can be seen. For example, to solve for F, cover the F, and 1 over T is left.

Where F equals frequency or the number of cycles that can occur in one second, T equals the time period for one cycle, and (1) equals 1 second in this case.

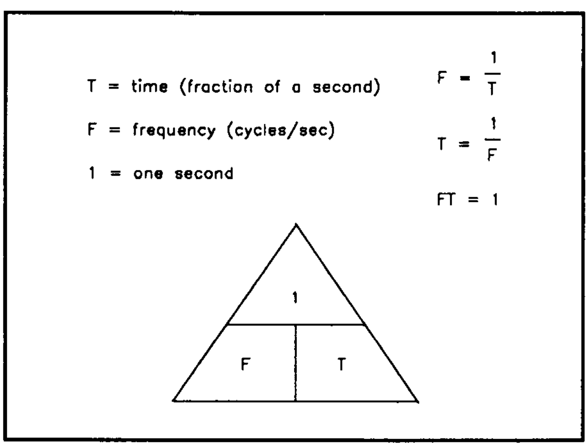


Fig. 1-5. Relationship of Time and Frequency.

Example 1-1: What is the time period for 1 cycle if the frequency is 29.6 Hz?
Answer:

$$T = \frac{1}{F}, \quad T = \frac{1}{29.6\ cyc/sec}$$

$$T = 0.0338\ \text{sec}\ \textit{or}\ 33.8\ ms$$

Most electronic instruments measure time in milliseconds or thousandths of one second. To convert milliseconds to seconds, move the decimal point to the left three places. For example, one millisecond (ms) is equal to 0.001 seconds.
Therefore:

$$T = 1\ ms \times (0.001\ \text{sec}/ms), \quad T = 0.001\ \text{sec}$$

Example 1-2: What is the frequency of a time period of 50 milliseconds?
Answer:

$$T = 50\ ms \times 0.001 = 0.05\ \text{sec}, \quad F = \frac{1}{T}, \quad F = \frac{1}{0.05\ \text{sec}}\ or\ 20\ Hz$$

In the above formula, when determining frequency in cycles per second, time must be in seconds.

Example 1-3: What is the frequency of a signal if the time period is 0.0338 seconds?
Answer:

$$F = \frac{1}{T}, \quad F = \frac{1}{0.0338\ sec}, \quad F = 29.6\ Hz$$

The conversion of cycles per second or Hz into cycles per minute or RPM to determine machine speed is often required. One minute contains 60 seconds and frequency is measured in cycles per second.
Therefore:

$$1\ cyc/\text{sec} \times 60\ sec/\text{min} = 60\ cyc/\text{min} = 60\ RPM$$

Therefore, multiply Hertz by 60 to obtain CPM or RPM and divide CPM or RPM by 60 to obtain Hertz. Please note the industry standard for measuring machine speed is revolutions per minute (RPM). The industry standard for measuring frequency is Hertz (cycles per second). In this text, the industry standards shall be used.

Example 1-4: What is the speed of a machine that generates a fundamental frequency of 29.6 Hz?
Answer:

$$29.6\ Hz \times 60\ \text{sec/min} = 1776\ RPM$$

Example 1-5: What is the fundamental frequency a machine will generate if the machine speed is 1180 RPM?
Answer:

$$\frac{1180\ cyc/\text{min}}{60\ \text{sec/min}} = 19.7\ Hz$$

AMPLITUDE MEASUREMENT

The four different ways to express the vibration amplitude level are: peak-to-peak, zero-to-peak, RMS, and average. Peak-to-peak is the distance from the top of the positive peak to the bottom of the negative peak. The peak-to-peak measurement of the vibration level is shown in Fig. 1-6. This type of measurement is most often used when referring to displacement amplitude.

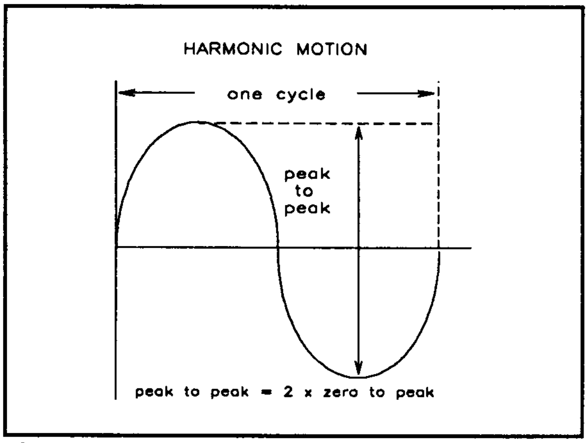


Fig. 1-6. Peak to Peak.

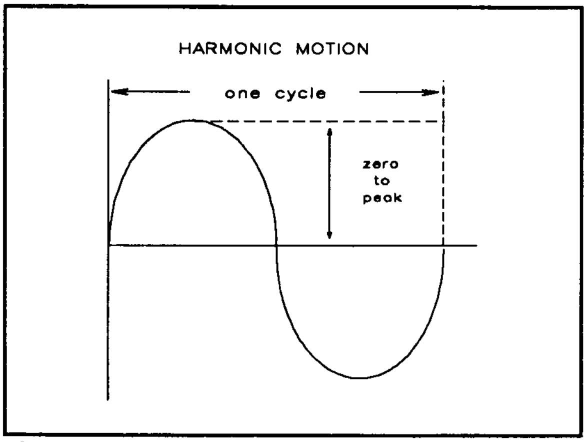


Fig. 1-7. Zero to Peak.

Zero-to-peak or peak is the measurement from the zero line to the top of the positive peak or the bottom of the negative peak. The zero-to-peak value of the vibration level is shown in Fig. 1-7. This type of measurement is used to describe the vibration level from a velocity transducer or accelerometer.

The Root Mean Square (RMS) is the true measurement of the power under the curve. In Fig. 1-8, the RMS value is the cosine of 45 degrees times peak (0.707 x peak only applies to pure sine waves). The true RMS value is calculated by the square root of the sum of the squares of a given number of points under the curve.
For example:

$$True\ RMS = \sqrt{\frac{(P_1^2 + P_2^2 + \ldots + P_N^2)}{N}}$$

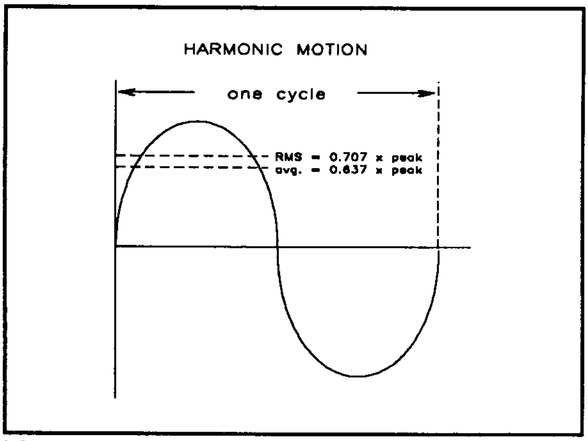


Fig. 1-8. RMS and Average.

When calculating true RMS, the crest factor and duty cycle must be considered for signals that contain pulses. The crest factor (CF) is the ratio of the peak value to the RMS value with the DC component removed.
For example:

$$CF = \frac{P - DC}{RMS}$$

A crest factor of 7 is normally required for accurate measurement of pulses. The duty cycle is the ratio of the pulse width (PW) to the pulse recurrence frequency (PRF).
For example:

$$Duty\ cycle = \frac{PW}{PRF}$$

Several forms of pseudo RMS are used in some equipment.
For example:

$$0.707 \times PEAK, \quad \sqrt{AC^2 + DC^2}$$

Analog meters measure average amplitude. Various constants are then used to calculate peak, peak-to-peak, or RMS. Most measurements that are not true RMS measurements are either overstated or understated.

When describing the vibration level of a machine, the RMS value should be used if possible. However, some cases require peak-to-peak measurements, for example, when measuring mils of displacement. Other cases require zero-to-peak displacement measurements such as high places on a roll.

The average value is 0.637 times the peak of a sine wave. See Fig. 1-8. Average values are measured by analog meters. Average is then converted to peak by multiplying a constant of 1.57. These calculated values are accurate only when measuring pure sinusoids. The following constants may be helpful. However, they apply to true sine waves only. The more the signal deviates from a true sine wave, the more error is introduced.

$$Average = 0.637 \times Peak$$

$$Average = 0.90 \times RMS$$

$$Peak\ to\ Peak = 2 \times Peak$$

$$Peak = 1.414 \times RMS$$

$$Peak = 1.57 \times Average$$

$$RMS = 0.707 \times Peak$$

$$RMS = 1.11 \times Average$$

SOURCES OF FREQUENCIES

The three sources of frequencies in machines are: generated frequencies, excited frequencies, and frequencies caused by electronic phenomena.

Generated Frequencies

Generated frequencies, sometimes called forcing frequencies, are those frequencies actually generated by the machine. Some examples are imbalance, vane pass frequency (number of vanes times speed), gearmesh frequency (number of teeth times speed), various frequencies generated by antifriction bearings, ball passing frequency of the outer race, ball passing frequency of the inner race, ball spin frequency, and fundamental train frequency. Generated frequencies are the easiest to identify because they can be calculated if the internal geometry and speed of the machine are known.

Some of the calculated frequencies may be present in most machines without indicating a vibration problem. These frequencies, at acceptable levels without sidebands, include but are not limited to: imbalance, vane pass frequencies, blade pass frequencies, and gearmesh frequencies.

Other calculated frequencies should not be present in any form at prescribed calibration levels. These frequencies include, but are not limited to: ball pass frequencies of the outer and inner races, ball spin frequency, fundamental train frequency, etc. Calculated frequencies should not be modulated with any degree of significance by other frequencies. If any of these frequencies are generated, a vibration problem exists.

When a rotating unit has a mass imbalance, it will generate a sine wave that has very little distortion. This signal can be observed in the time domain. The frequency domain spectrum will have a spectral line at one times speed of the unit. For example, a 1776 RPM fan that is out of balance will have one spectral line at 29.6 Hz.

Most pumps and fans can generate vane or blade pass frequency, which is the number of vanes or blades times the speed of the unit. A high vibration at this frequency could be the result of buildup on the vanes or blades, the vanes or blades hitting something, or looseness associated with the rotating unit.

Example 1-6: What is the blade pass frequency of a 1776 RPM fan with four blades?
Answer:

$$\frac{1776\ RPM}{60} = 29.6\ Hz$$

$$29.6\ Hz \times 4\ Blades = 118.4\ Hz$$

Gearmesh frequency is normally seen in data taken from a gearbox or gear train. The frequency is the number of teeth on a gear times the speed of that gear. For two gears in mesh, the gearmesh frequency will be the same for each gear; the ratio of number of teeth to gear speed is a constant. In a gear train, all gears will have the same gearmesh frequency. This vibration is caused by teeth rotating against each other. Multiples and submultiples of gearmesh frequency are sometimes observable in the frequency spectrum and will be discussed later.

Example 1-7: A bull gear with 67 teeth is in mesh with a 22-tooth pinion gear. The bull gear is rotating at 6.4 Hz. a) What is the gearmesh frequency (GF)? b) What is the speed of the pinion gear?
Answer:

$$GF = no.\ of\ teeth \times gear\ speed$$

$$a)\ GF = 67 \times 6.4\ Hz = 428.8\ Hz$$

b) Since GF is the same for both gears, the speed of the pinion is:

$$\frac{428.8\ Hz}{22\ teeth} = 19.5\ Hz$$

There are many other generated frequencies such as misalignment, bent shafts, bearing frequencies, looseness, etc. These sources will be discussed in detail later.

Excited Frequencies

Excited frequencies, (natural frequencies), are a property of the system. Amplified vibration, called resonance, occurs when a generated frequency is tuned to a natural frequency. Natural frequencies are often referred to as a single frequency. Vibration is amplified in a band of frequencies around the natural frequency, as in Fig. 1-9. The amplitude of the vibration in this band depends on the damping.

When we refer to the natural frequency, we often mean the center frequency. Natural frequencies can be excited by harmonic motion if the harmonic motion is within the half-power points of the center frequency and contains enough energy.

The half-power points are down 3 dB on either side of the center frequency. The

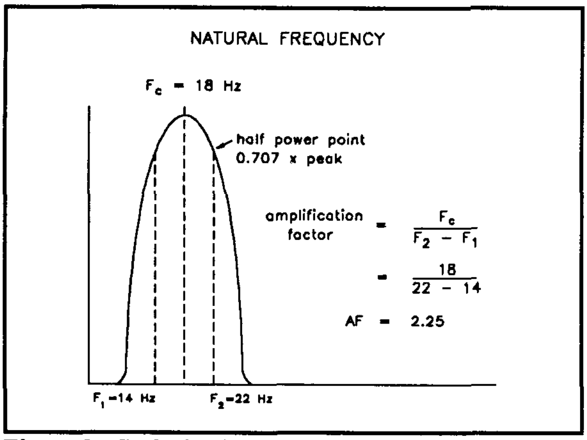


Fig. 1-9. Relatively Damped Frequency.

frequency range between these half-power points is called the bandwidth of the natural frequency. The half-power point, or 3 dB, is 0.707 times peak at the center frequency. It is a general rule to stay at least 10% away from each side of the center frequency. If some frequency is within the bandwidth of the natural frequency and this frequency contains enough energy to excite the natural frequency, the natural frequency will be present. The term "critical speed" means the rotating speed of the unit equals the natural frequency. When this occurs, the natural frequency is considered unacceptable by some experts.

Damping is the measure of a machine's ability to absorb energy. Therefore, a relatively undamped vibration signal can be high in amplitude and relatively narrow banded. A relatively damped signal can be low in amplitude and relatively wide banded. Fig. 1-9 displays a relatively damped frequency. F_c is the center frequency and is equal to 18 Hz. F_1 is the low frequency at the half power point (0.707 times peak) and is equal to 14 Hz. F_2 is the high frequency at the half power point and is equal to 22 Hz. From this information, an amplification factor (AF) can be calculated:

$$AF = \frac{F_c}{(F_2 - F_1)} = \frac{18}{(22 - 14)} = \frac{18}{8} \text{ or } AF = 2.25$$

Some experts agree that the amplification factor should be less than eight (8). Note that the result is the same if RPM is substituted for Hertz. Fig. 1-10 contains a relatively well-damped frequency where:

$$F_c = 18\ Hz, \quad F_1 = 17\ Hz, \quad F_2 = 19\ Hz,$$

$$\text{Then } AF = \frac{18}{(19 - 17)} = \frac{18}{2} \text{ or } AF = 9$$

Some standards for fluid film bearing machines require an amplification factor of less than eight (8). However, for rolling element bearings, it is hard to get less than twelve (12).

In Fig. 1-9, machine speeds could vary from 0 to 840 RPM and from 1320 RPM up to the second critical. Therefore, the operating range is restricted. However, if you operate on

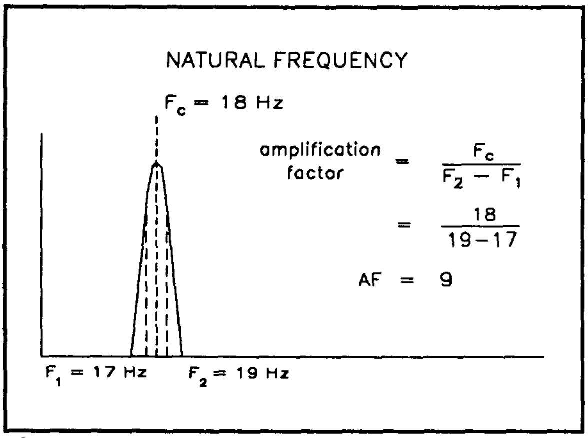


Fig. 1-10. Relatively Well-damped Frequency.

the critical, the problem is less severe. In Fig. 1-10, the operating range is less restricted. However, if you operate on critical, the problem could be severe.

Natural frequencies can be helpful when they act as a carrier, transporting the source of excitation to the measurement point. This normally occurs when natural frequencies are excited by periodic motion such as hits, bumps, or impacts. In such cases, the natural frequency can be present with spectral lines on each side. The number of these spectral lines, or sidebands, is determined by damping and distortion of the wave form. The difference frequency between the spectral lines identifies the frequency or source of excitation. More than one natural frequency may be present and harmonics of the natural frequencies can occur if distortion is present.

Natural frequencies can be identified because they are generally not a calculated frequency, but they are modulated by a calculated frequency. Fig. 1-11 is an example of an excited natural frequency.

In such cases, if the source of excitation is removed, the natural frequency will not be excited. When the oil film is destroyed in antifriction bearings, a natural frequency can be excited by the metal-to-metal contact between the balls or rollers and the raceways, as in Fig. 1-12. This problem can be solved by adding oil or changing the viscosity of the oil. The worst situation exists when the natural frequency equals the generated frequency. In such cases, the amplitude can be quite high.

The most simple solution is to change the generated frequency or raise the natural frequency above the generated frequency.

An excited natural frequency is a resonant condition. Resonance in rotating machinery is the same as an amplifier in electronics. Therefore, excessively high vibration amplitudes are often encountered. The Resonance and Deflection Calculator Program (RADC) is necessary in every predictive maintenance program to identify and calculate natural frequencies, critical speeds, mode shapes, and wavelengths. Critical speeds and mode shapes require some explanation. For explanation purposes we shall use a simple H-beam. The theory applies to more complex systems such as motors, turbines, gears, etc.

However, the complex explanations are beyond the scope of this book.

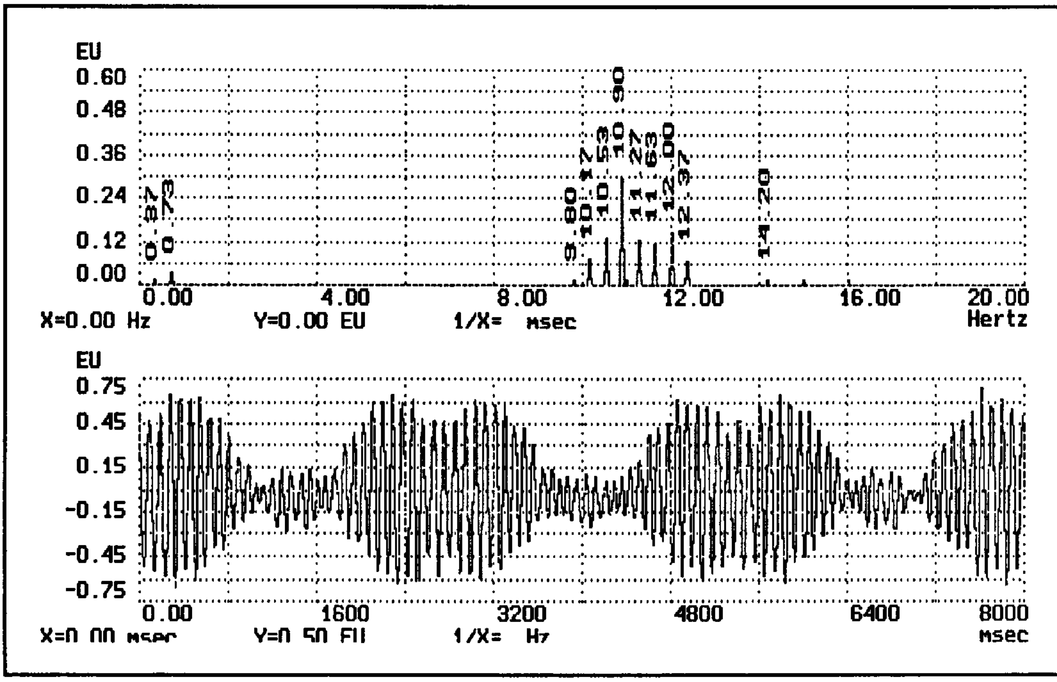


Fig. 1-11. Excited Natural Frequency.

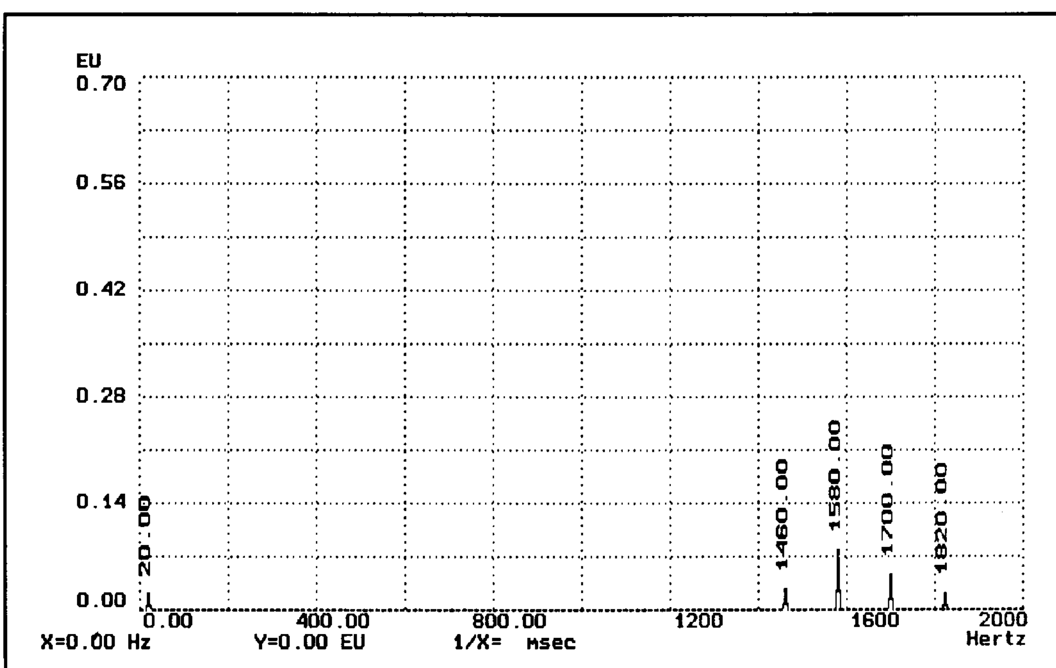


Fig. 1-12. Inadequate Lubrication.

A critical speed occurs when a machine speed equals a natural or resonant frequency of the machine or some machine part.

The mode shape explains the way the beam or machine is bending. The first mode equals the first critical, the second mode equals the second critical, etc. The second or higher criticals are seldom a harmonic of the first or higher critical.

The wavelength or lambda (λ) is the physical length of a beam required for one cycle to occur. The wavelength should not be confused with the natural or resonant frequency, because the natural frequency occurs at one quarter or three quarters of a wavelength. The reason for this seeming paradox is loops and nodes. A node or a nodal point is the place where the curve crosses the zero point. A loop is the area where the curve is not crossing zero. Zero motion or vibration occurs at a node and maximum vibration occurs at the peak of a loop.

In order to keep the mathematics to a minimum, the RADC software package is used. This discussion explains the natural frequencies and mode shapes of a 6 by 6 by 0.25 inch H-beam. The H-beam is six feet long and one end is fixed in concrete. Fig. 1-13 contains the mode shape for the first natural frequency. The first natural frequency is 26.86 Hz. The node is at near zero and below. The maximum loop is at 72 inches and the quarter wavelength is also 72 inches. The second, third, and fourth natural frequencies are 170.58, 482.47, and 946.42 Hz respectively.

Fig. 1-14 contains the mode shape for the second natural frequency of 170.58 Hz. Nodes occur at zero and 62 inches, which are zero and one half wavelength, respectively. Maximum loops occur at about 36 inches and 72 inches, which are one quarter and three quarters wavelength, respectively.

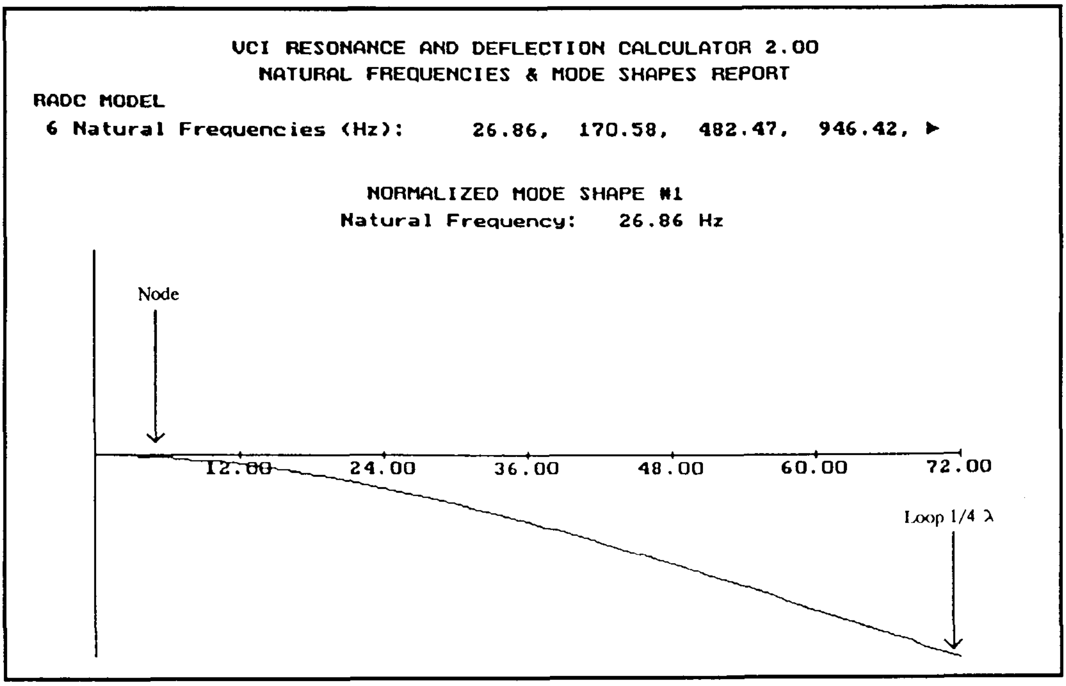


Fig. 1-13. Normalized Mode Shape #1.

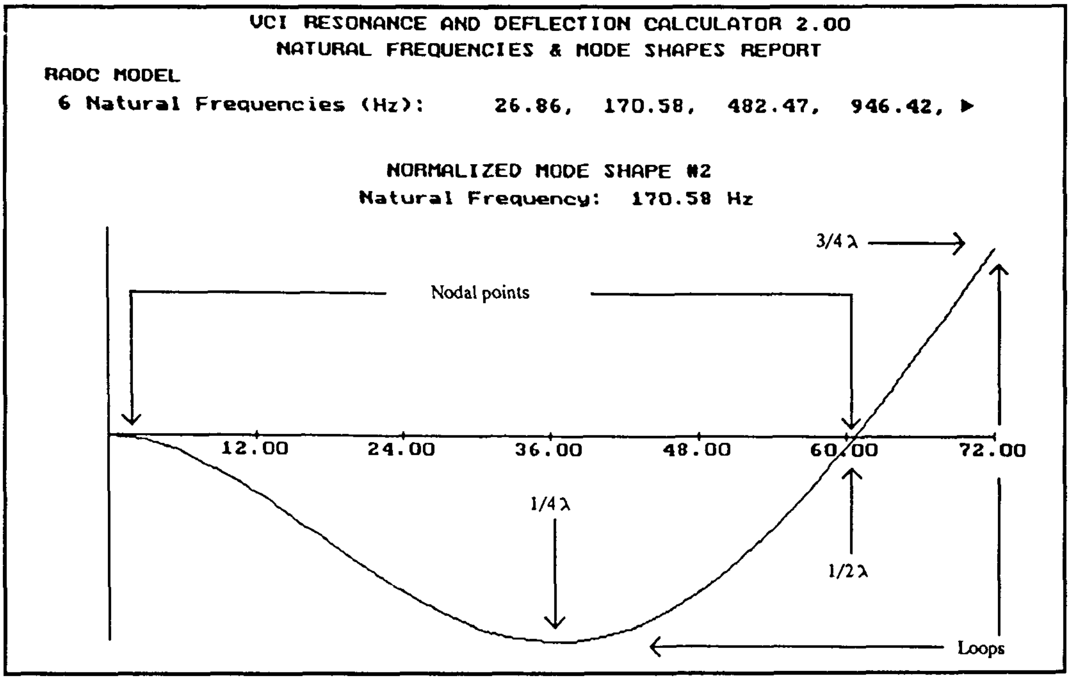


Fig. 1-14. Normalized Mode Shape #2.

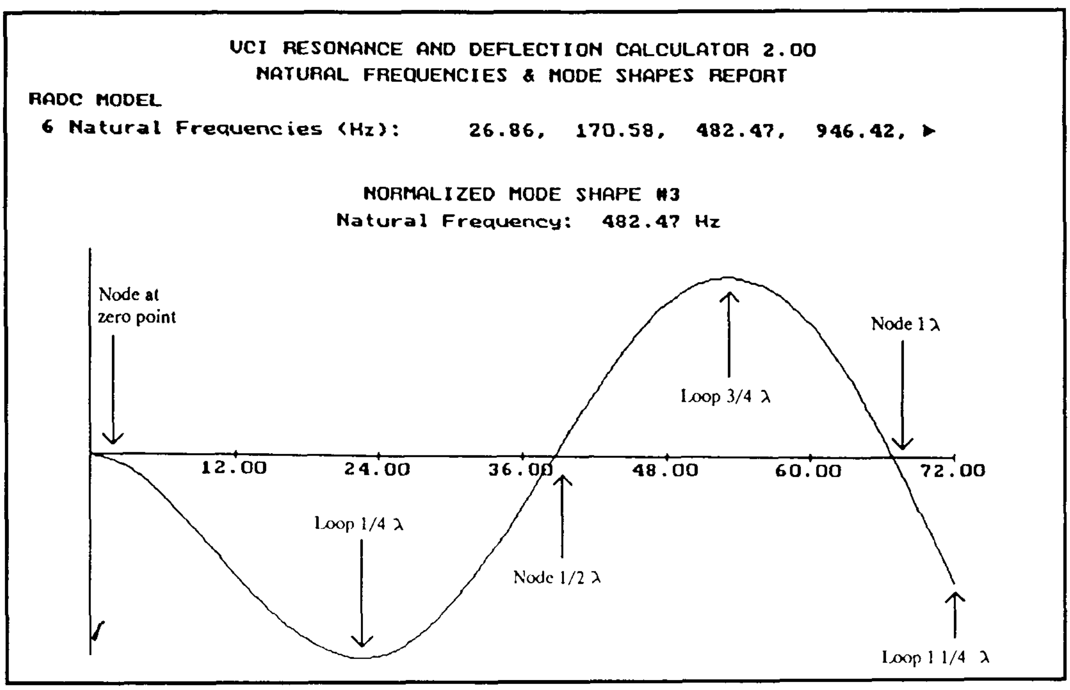


Fig. 1-15. Normalized Mode Shape #3.

Fig. 1-15 contains the mode shape for the third natural frequency of 482.47 Hz. This mode shape contains three loops and three nodes. One wavelength is about 68 inches long at a frequency of 482.47 Hz in this case. Please note the first mode or natural frequency contains one loop and one node. The second mode contains two loops and two nodes. The third mode contains three loops and three nodes. This pattern is repeatable at higher modes.

This software package removes the mystery and the complex mathematics from analyzing natural frequencies. This software also permits you to simulate a repair to determine if the proposed repair will solve the problem, and allows you to evaluate any new problems that may be generated from the repair. For example, what would the effect be if the beam was shorter or longer, or if it was heavier or lighter? What if the H-beam was made into a box or channel, or was braced at various points, etc?

Following are some simple rules to remember when working with resonant frequencies:

1. Mass - If mass is added, the frequency goes down. If mass is removed, the frequency goes up.

2. Stiffness - If stiffness is increased, the frequency goes up. If stiffness is decreased, the frequency goes down.

3. Increase in length lowers stiffness.

4. Decrease in length increases stiffness.

5. Braces increase stiffness.

Another major feature of the RADC software is the ability to calculate the resonant frequency and the deflection or sag in a beam or shaft. Fig. 1-16 is a 20-inch diameter 4140 steel pipe 300 inches long with a wall thickness of 0.5 inches. Fig. 1-17 contains the first, second, and third natural frequencies of 24.35, 96.74, and 205.39 Hz, respectively.

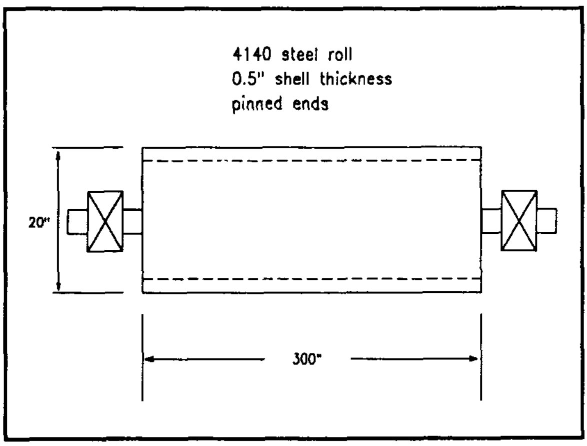


Fig. 1-16. Steel Pipe Model.

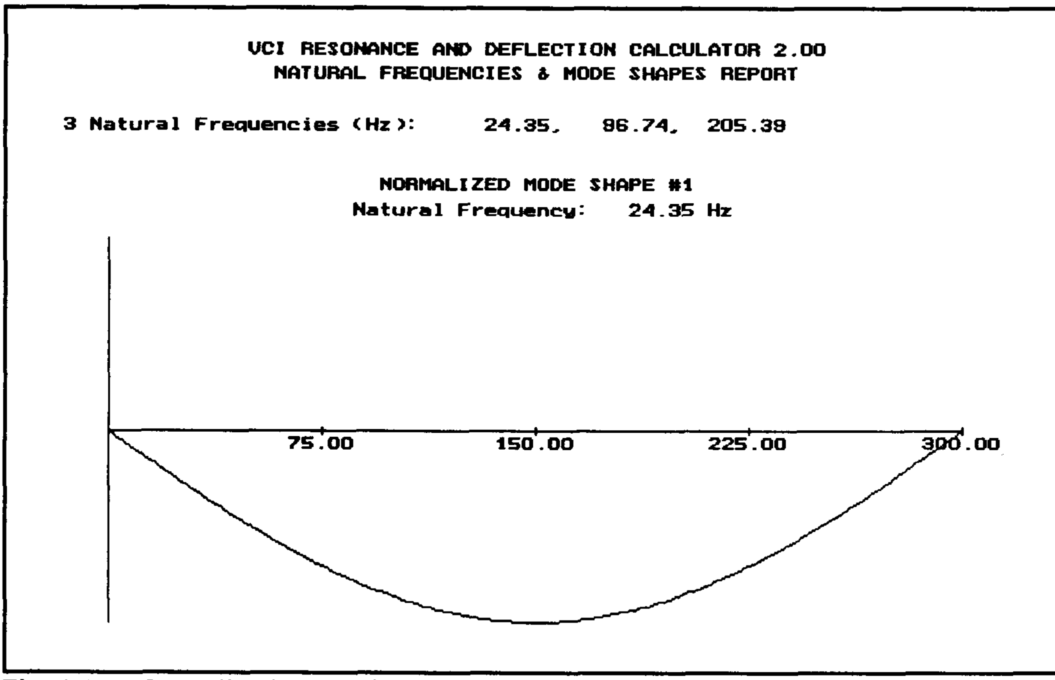


Fig. 1-17. Normalized Mode Shape #1.

Table 1-1. Static State Values.

VCI RESONANCE AND DEFLECTION CALCULATOR 2.00
STATIC STATE REPORT

Location (inches)	Shear (lbf)	Moment (inch-lbf)	Slope (deg)	Deflection (inches)
0.00	1300	0.00	-0.0128	0.00
30.00	1040	35107	-0.0121	-0.0066
60.00	780.16	62412	-0.0101	-0.0124
90.00	520.11	81916	-0.0073	-0.0170
120.00	260.05	93619	-0.0038	-0.0199
150.00	0.00	97519	0.00	-0.02
180.00	-260.05	93619	0.0038	-0.0199
210.00	-520.11	81916	0.0073	-0.0170
240.00	-780.16	62412	0.0101	-0.0124
270.00	-1040	35107	0.0121	-0.0066
300.00	-1300	0.00	0.0128	0.00

Maximum State Values

Shear: 1300 at L = 0.00 Moment: 97519 at L = 150.00
Slope: -0.0128 at L = 0.00 Deflection: -0.02 at L = 150.00

Table 1-1 lists the static state condition for this pipe. The maximum deflection occurs at 150 inches and the pipe deflects or sags 0.0209 inches. The Resonance and Deflection Calculator Program puts high technology in the hands of the predictive maintenance department. Until now, this technology was only available to the more sophisticated engineering departments.

Frequencies Caused by Electronic Phenomena

In certain situations, false or misleading signals can be present. For example, when a sinusoid is clipped (as occurs when the input to the Real-Time Analyzer (RTA) is overloaded slightly), it can cause a string of harmonics. The amplitude of the harmonic content is normally quite low as in Fig. 1-18.

If the clipping is serious enough to cause a square wave or near square wave, only the odd harmonics may be present, or the odd harmonics can be higher in amplitude than the even harmonics. See Fig. 1-19.

Also, if a sinusoid is distorted and the time period for each halfwave is not equal, harmonics can be generated as in Fig. 1-20. When true harmonic motion is present, the number of positive going peaks observed for one time period is equal to the true harmonic content. The amplitude of these harmonics is normally high in the frequency domain. When distortion occurs with true harmonic content, the number of harmonics in the frequency domain can exceed the number of positive peaks for one period in the time domain as in Fig. 1-21.

If the construction of a machine is sound and the various parts are not defective, the machine will behave in a linear manner. Pure imbalance will generate a single frequency

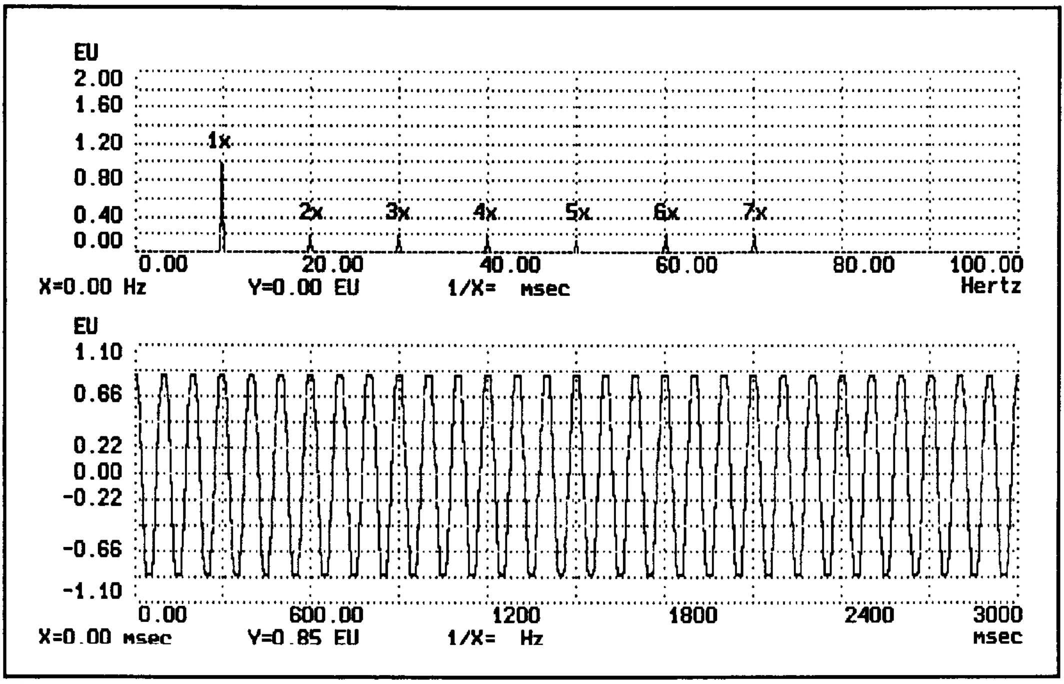


Fig. 1-18. Clipped Time Signal.

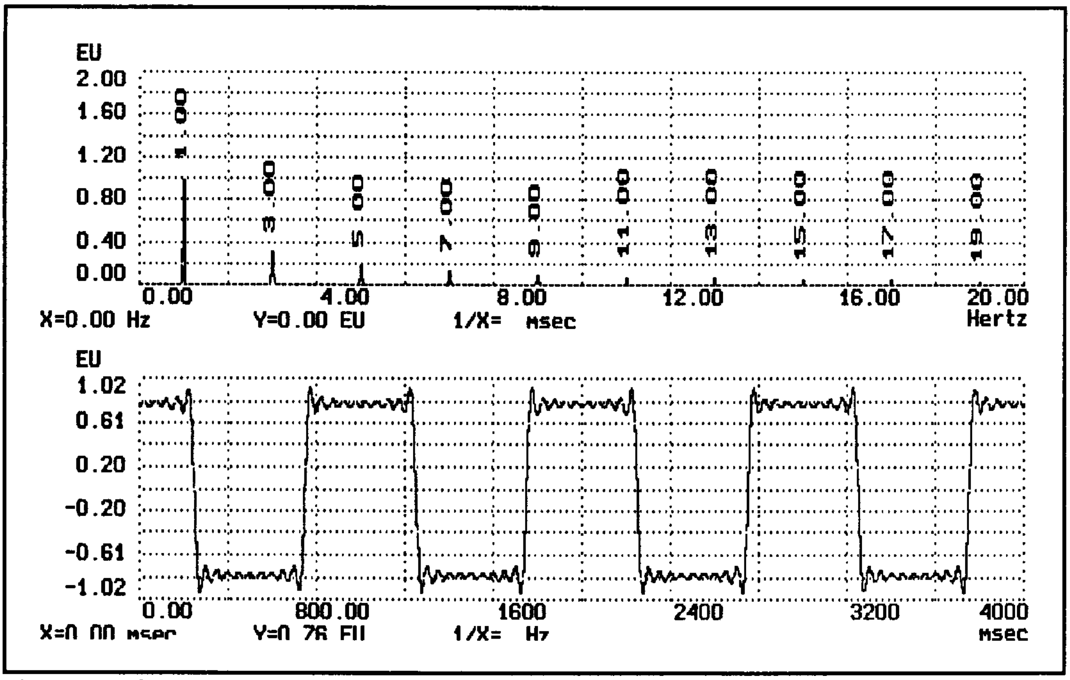


Fig. 1-19. Square Wave.

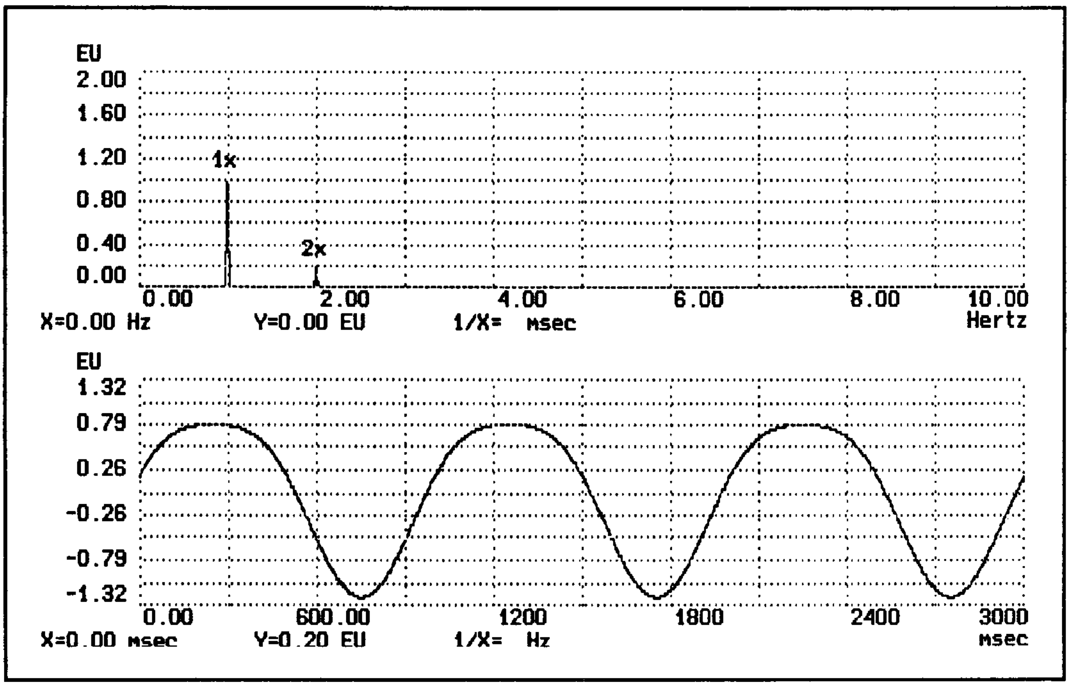


Fig. 1-20. Distorted Wave Form.

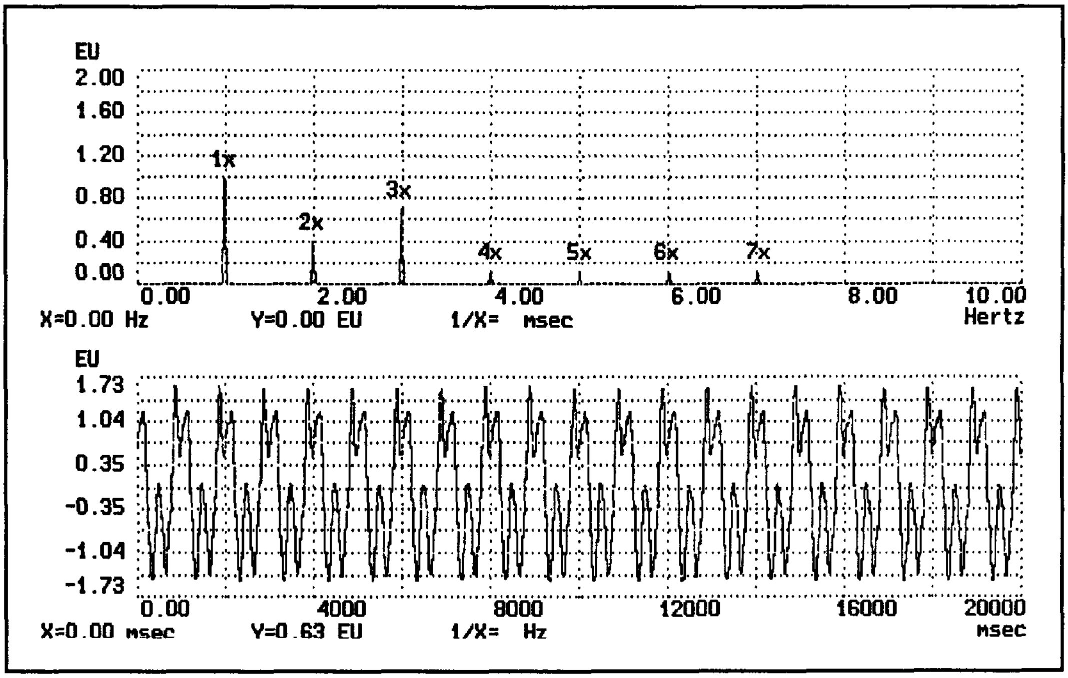


Fig. 1-21. True Harmonic Content.

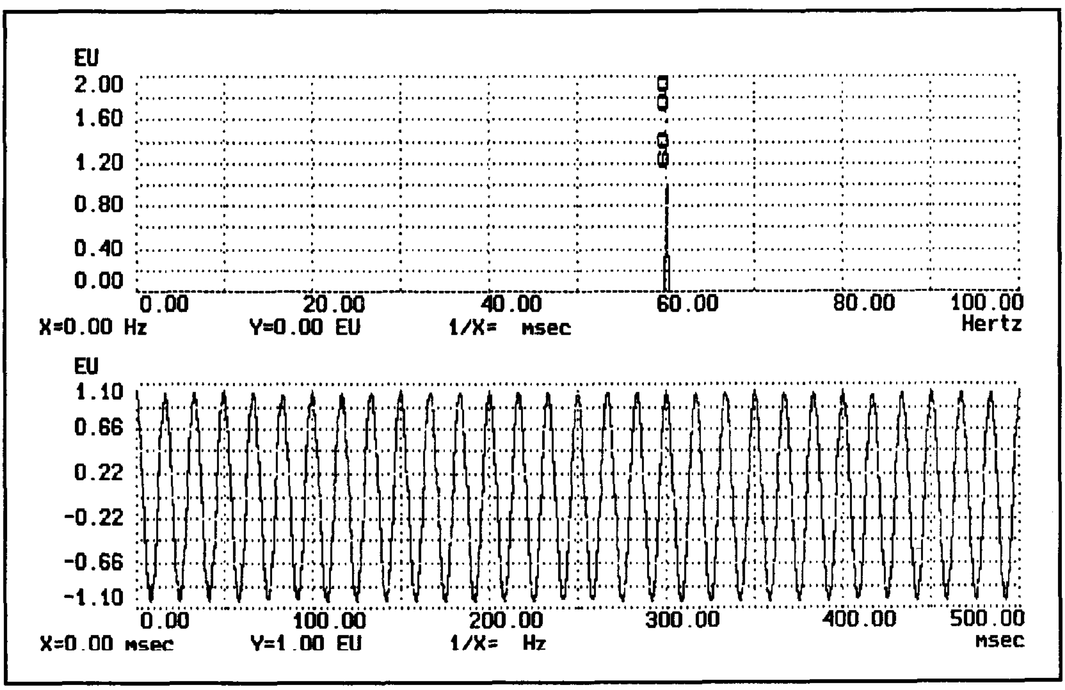


Fig. 1-22. Pure Imbalance.

without harmonics or modulation as in Fig. 1-22.

There are several forms of nonlinear problems in a machine. Misalignment, for example, is nonlinear and is not speed related above certain speeds depending upon shaft size, coupling type, amount of misalignment, etc. It is general knowledge that at higher speeds, less misalignment is allowed. After the speed increases, making misalignment a problem, the vibration level will increase to a given point and remain relatively constant at higher speeds.

Looseness is probably the most common form of a nonlinear machine problem. Looseness is also the most common problem in operating machines. When some form of looseness is present in a machine, the machine becomes "tuned for maximum sensitivity." When this occurs, a small amount of imbalance can cause high vibration levels. Also, the machine is easily shaken by a small excitation from other machines. Looseness causes distortion in the signal. The distortion causes harmonics, and if clipping or truncation occurs, sum and difference frequencies will be present in the spectra. A good time domain signal is necessary to identify all frequencies in the frequency spectra and accurately diagnose machinery problems.

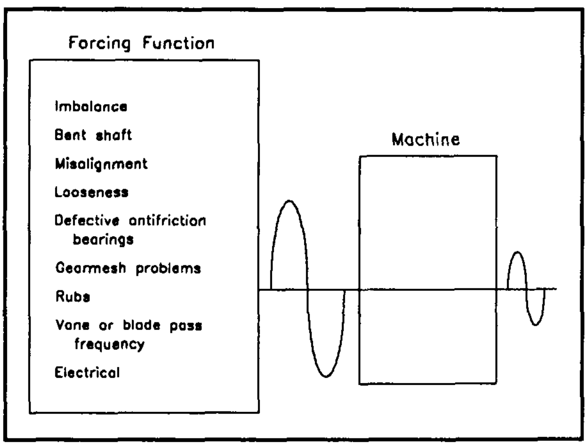


Fig. 1-23. Physical Nature of Vibration.

FORCING FUNCTION

When a machine vibrates, the cause must be determined. The cause of the vibration is often referred to as the forcing function. A number of possible problems, any combination of these problems, and an infinite degree of each problem can cause the machine to vibrate. The amplitude of each problem can be either overstated or understated depending on transfer function, resonance, damping, frequency addition, and frequency subtraction. Therefore, very low amplitudes can be very serious problems in some cases and very high amplitudes can be relatively minor problems in other cases. Frequency measurement and analysis makes vibration analysis work. Each forcing function has its own frequency, as indicated in Fig. 1-23. If the problem is present, the frequency is present. The frequency can manifest itself as a discrete frequency or as a sum and/or difference frequency. Resonance, damping, frequency addition, and frequency subtraction do not affect the frequencies caused by vibration. This fact explains why frequency analysis is accurate, while amplitude measuring and trending are not accurate when diagnosing problems in rotating machinery.

Defects in rotating machinery are caused by normal wear, improper lubrication, overloading, improper installation, improper manufacturing, etc. Vibration analysis identifies the problems or problem areas. Management, engineers, and technicians must determine the cause of the problems and direct the necessary repairs, or find the appropriate solutions to the problems.

Vibration analysis of rotating machinery involves calculating the frequencies the machine can generate and measuring the frequencies the machine is generating. Then, relating the measured frequencies to the calculated frequencies and machine installation can identify the problem. If the forcing function is a hit or an impact, the hit or impact may not be measurable. However, the machine response to the impact can be measured. Problems in machinery induce vibration and result in wear, malfunction, and/or structural damage.

COMBINATIONS OF MACHINE PROBLEMS

The amplitudes of various frequencies and the frequencies in rotating machines can add and subtract. These frequencies mix, detect, amplitude modulate, and frequency modulate in much the same way as signals in electronic mixers, detectors, and modulators. An electronic amplifier performs in much the same way as resonance. An attenuator performs in much the same way as damping.

The simplest form of amplitude addition and subtraction is imbalance. When a rotor becomes out of balance, one side is heavier than the other side. This imbalance condition or heavy place generates a signal at one times RPM, as shown by the solid line in Fig. 1-24.

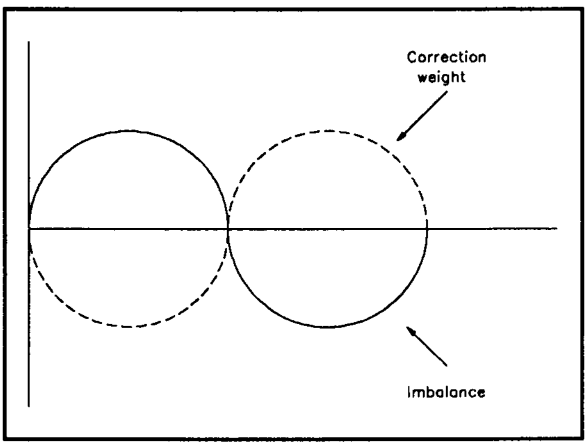


Fig. 1-24. Simplest Form Imbalance.

Balancing the fan is accomplished by adding an equal amount of weight to the light place. This weight generates a one times RPM equal in amplitude and 180 degrees out of phase with the imbalance, as indicated by the broken line in Fig. 1-24. Since we live in an imperfect world, the signals may not cancel completely. Therefore, a small amplitude of the resulting imbalance may be present as indicated in Fig. 1-25.

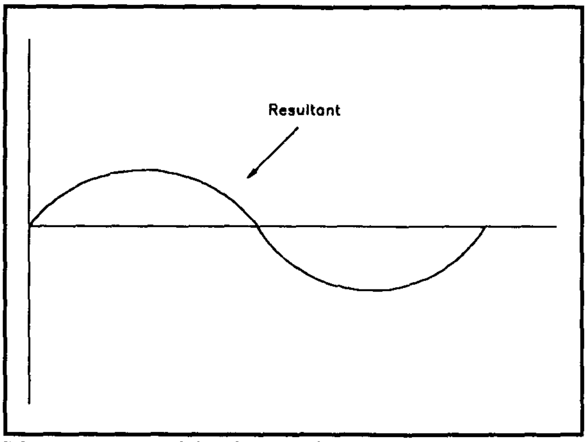


Fig. 1-25. Residual Imbalance.

MIXING FREQUENCIES

When two frequencies are present in a machine and a cause and effect relationship is not present, the high frequency will be riding the low frequency and the Fast Fourier Transform (FFT) will yield spectral lines at frequency one and frequency two. If there is a cause and effect relationship and the two frequencies can mix together, the result is amplitude modulation. Without getting mathematical, amplitude modulation is a time varying amplitude. Amplitude modulation is caused when the equipment has some form of nonlinearity. This nonlinearity permits the amplitude of the two signals to add together when the signals are in phase, or subtract when the signals are out of phase. With amplitude modulation, the carrier frequency will be the frequency with the highest amplitude. The envelope of the varying amplitude will be the difference between the two frequencies. An FFT of these signals can yield spectral lines at frequency one, and frequency one plus and/or minus frequency two.

For example, suppose gearmesh frequency is modulated by gear speed, gearmesh frequency is 1200 Hz, and gear speed is 20 Hz. An FFT of this signal would then yield spectral lines at 1200 Hz, 1200 + 20 = 1220 Hz, and/or 1200 - 20 = 1180 Hz.

Descriptions of these frequencies are:

1. 1200 Hz is gearmesh frequency.

2. 1220 Hz is gearmesh frequency plus gear speed. This is a sum frequency.

3. 1180 Hz is gearmesh frequency minus gear speed. This is a difference frequency.

4. The difference between 1200 and 1220 Hz, or 1200 and 1180 Hz is 20 Hz, and this is also a difference frequency.

5. The source of excitation, or the problem shaft or gear is usually expressed as a difference frequency.

If the amplitude modulated signal is clipped or truncated, the FFT can yield an array of sum and difference frequencies of the two frequencies, including frequency one plus frequency two. There are many forms of sum and difference frequencies. These frequencies will be discussed later.

ELECTRICAL AND MECHANICAL RELATIONSHIP

As mentioned previously, the basic physics of electronics and mechanics are the same. This similarity explains why we can convert a mechanical motion into an electrical signal, then diagnose the mechanical problem with the electrical signal. For example, Table 1-2 displays the equivalent electrical and mechanical terms.

Table 1-2. Comparison of Electrical and Mechanical Terms.

ELECTRICAL	MECHANICAL
E = Voltage	F = Force
I = Current	V = Velocity
R = Resistance or Impedance (Z)	R = Friction
P = Power	HP = Horsepower
F = Frequency (CPS)	S = Speed (RPM)

In electronics there is electromotive force or voltage; in mechanics there is force. In electronics there is current; the mechanical counterpart is velocity. In electronics there is resistance, which is purely resistive resistance, and impedance which is the total opposition to current flow. This includes resistive, capacitive, and inductive resistance. The mechanical counterpart to resistance is friction. In electronics there is power; in mechanics there is horsepower or work. In electronics frequency is used; in mechanics speed is used.

Table 1-3 contains some of the electrical formulas established by Ohm's Law and the mechanical equivalents. Some liberties were taken with the mechanical equivalents for ease of explanation.

Each mechanical action or occurrence is related to an electrical action or occurrence. Vibration analysis begins with mechanical energy and ends with electrical energy. In examining a rotating object mechanically, a force causes the object to rotate. Electrically, voltage causes the object to rotate. We can mechanically measure the velocity of the rotating object, or electrically measure the current of an electric motor. For example, to compare the mechanical and electrical worlds, we can slow down the rotating object by mechanically applying friction, or electrically applying resistance. Speed (mechanical) or frequency (electrical) measures how fast the object is moving. As explained in the introduction of this text, a transducer is used to convert mechanical energy into electrical energy. The electrical energy is then processed and analyzed for possible machine problems. Even though mechanical and electrical terms may appear to be different, they are very much alike.

Table 1-3. Electrical and Mechanical Equivalents.

ELECTRICAL	MECHANICAL
$E = IR$	$F = VR$
$I = \frac{E}{R}$	$V = \frac{F}{R}$
$R = \frac{E}{I}$	$R = \frac{F}{V}$
$P = IE$	$HP = VF$
$P = \frac{E^2}{R}$	$HP = \frac{F^2}{R}$

TIME AND FREQUENCY DOMAIN

The frequency domain is all around us. However, sometimes we call frequency by another name. For example, light is a frequency. The color red is a frequency. Sound is a frequency. We do not refer to these items as frequencies; we call them light, color, and sound. As light, color, and sound are varied, the frequency changes. The audio range of frequencies is from 20 to 20,000 Hz. It is called the audio range because it is the frequency response of the human ear. However, most people cannot hear 20 or 20,000 Hz, and some cannot hear several frequencies between these limits. Low-pitched sound contains low frequencies, as in a baritone singer. A high-pitched sound contains high frequencies; an opera singer is an example.

Some things are better measured in time and others are better measured in frequency. For example, consider the sentence, "We go to work once a day." We would not say that "once a day" is the frequency. Rather, we say we work eight hours each day or 40 hours each week, which are amounts of time.

Life and the world are full of misconceptions because we do not understand our surroundings. For example, we say it is dark. There is no such thing as darkness, only the absence of light. Darkness cannot be measured. We say it is cold. There is no such thing as cold, only the absence of heat. However, heat and light can be measured. These misconceptions carry over to other areas. Many times the author has heard a client say "I looked at the bearing (or equipment) and I did not see anything wrong." The client was telling the truth, but this does not mean the problem is not there just because he did not see it. Many times a mechanic, supervisor, or engineer has used a screwdriver to listen to a machine after the machine has been diagnosed as having a problem. This person would then say: "It sounds okay to me." However, the machine does have a problem; the frequency may be out of the audio range or it could be masked by other noises.

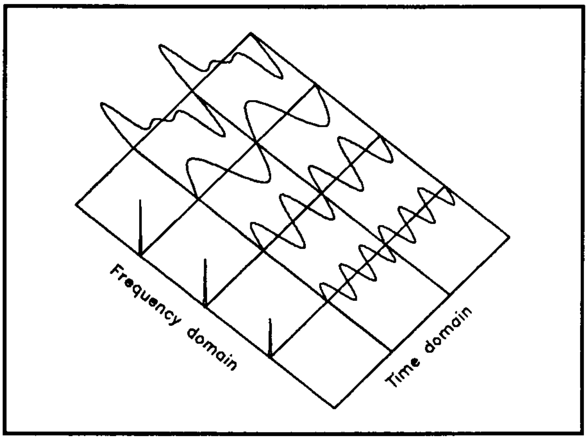


Fig. 1-26. Relationship between Time and Frequency.

Analysis of some problems such as imbalance can be diagnosed in the time domain. However, time domain signals from rotating machines are often very complex. Such signals must be analyzed in the frequency domain. If imbalance is diagnosed in only the frequency domain, errors will often occur. In vibration analysis, use of both the time domain signal and the frequency domain spectra are required for complete, accurate analysis.

To move from the time domain to the frequency domain, we must perform a Fast Fourier Transform on the time domain signal. Mr. Fourier was a French mathematician who proved all complex wave forms could be broken down into their individual frequency components mathematically. However, this brilliant technology was not used extensively until the advent of the computer. Utilizing the transformation of the time signal to a frequency spectrum, Mr. Fourier's technology, and the computer's rapid capabilities, can produce a Fast Fourier Transform or FFT.

Fig. 1-26 contains a time signal and the corresponding FFT. The first (left-most) time signal contains the fundamental, second, and third harmonic. This statement is true because each time period has three positive going peaks. The first spectral line is the fundamental frequency and its time signal. The second spectral line is the second harmonic and its time signal. The third spectral line is the third harmonic and its time signal. If the time signals of the three frequency components were added together, the result would be the first (left) time signal, as indicated in Fig. 1-26.

RELATIONSHIP BETWEEN VELOCITY, DISPLACEMENT, AND ACCELERATION

Velocity is the measurement of how fast an object is moving from zero-to-peak and is normally measured in tenths of one inch per second (IPS). The effective frequency range of most velocity transducers is from about 10 to 2,000 Hz. Velocity is the most accurate measurement because it is not frequency related. For example, 0.15 IPS is the same at 10 Hz as it is at 2,000 Hz.

Displacement is the measurement of how far an object is moving from peak-to-peak and is normally measured in thousandths of one inch (mils). Displacement is frequency related. Therefore, any measurement of displacement must be at a specified frequency. For example, three mils at one times RPM of a 1200 RPM motor is a meaningful statement. However, an overall measurement of three mils is a meaningless statement. Three mils at 1200 RPM is about 0.2 IPS of velocity. However, three mils at 3600 RPM is almost 0.6 IPS of velocity. The effective frequency range of noncontacting displacement transducers is from about 0 to 600 Hz. For contacting displacement transducers, the effective frequency range is about 0 to 200 Hz.

Acceleration measures the rate of change of velocity from zero-to-peak and is normally measured in units of gravitational force (g's). This means that high frequencies generate high g levels, and acceleration is frequency related. For example, 3 g's at 20 Hz equals about 9 IPS of velocity. Three g's at 2,000 Hz is about 0.09 IPS of velocity. The effective frequency range for low frequency accelerometers is from about .2 to 500 Hz. The effective range of high frequency accelerometers is from about 5 to 20,000 Hz. The Vibration Calculator Program is useful for comparing various Engineering Units (EU). Fig. 1-27 displays the frequency response curves for displacement, velocity, and acceleration. All curves are plotted against a constant value of 0.15 IPS of velocity.

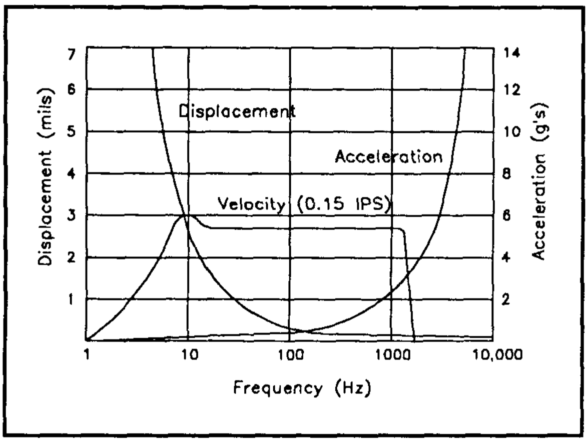


Fig. 1-27. Frequency Response Curves.

Please note the displacement curve is downward and outward sloping. This indicates low frequencies generate high levels of displacement and high frequencies generate low levels of displacement. Therefore, the displacement transducer most effectively measures lower frequencies. The frequency response of the velocity transducer is relatively flat from about 10 Hz to about 2,000 Hz. It is the most accurate transducer to use in this frequency range. The acceleration curve is an outward and upward sloping curve, which means that high frequencies generate high levels of acceleration. The accelerometer must be used for frequencies above 2,000 Hz and may not be as effective for frequencies below 100 Hz. A strong word of caution is due at this point. Two transducers must be used when making measurements on equipment that generate both high and low frequencies, one transducer for measuring low frequencies and an accelerometer for measuring high frequencies. If this procedure is not used, the high levels from the high frequencies require equipment setups that could mask the low frequencies. If so, low frequency problems may not be identified. Except for real-time monitors, velocity should be used for most vibration monitoring and analysis. Velocity levels remain relatively constant over a wide frequency range. Therefore, velocity should be used whenever possible; displacement and acceleration are both frequency related. The frequency of the vibration source must be known for the measurement in either displacement or acceleration to be meaningful.

UNITS OF MEASUREMENT

The following symbols are used in the vibration field:

D = Displacement in inches peak-to-peak

V = Velocity in inches per second zero-to-peak

g = Acceleration in g's zero-to-peak

F = Frequency in Hertz

RELATIONSHIPS

The relationships between these measurements and the conversion from one engineering unit to another can be accomplished with the use of the following equations:

1. $V = \pi F D$
2. $V = 61.44 \dfrac{g}{F}$
3. $g = 0.0511\ F^2 D$
4. $g = 0.0162\ VF$
5. $D = 0.3183 \dfrac{V}{F}$
6. $D = 19.57 \dfrac{g}{F^2}$

The following examples may be helpful:

Example 1-8: The vibration level on a variable speed motor operating at 400 RPM is 0.12 IPS (measured with a velocity transducer). The displacement in peak-to-peak can be calculated by using equation 5:

$$D = 0.3183 \frac{V}{F}$$

First, frequency in Hertz must be determined.

$$F = \frac{400\ RPM}{60} = 6.67\ Hz, \qquad D = 0.3183 \left(\frac{0.12}{6.67}\right)$$

Then, $D = 5.7\ mils$

Example 1-9: If the vibration level for the motor in Example 1-8 is determined with an accelerometer to be 0.013 g's, then the displacement can be calculated by using equation 6:

$$D = 19.57\left(\frac{g}{F^2}\right) = 19.57\left(\frac{0.013}{6.67^2}\right) = 5.7\ mils$$

Example 1-10: A pump driven by the motor in Example 1-8 is turning at 111.4 RPM. The vibration level is measured with an accelerometer which reads 0.01 g's. The vibration level in IPS can be calculated from equation 2:

$$V = 61.44\ \frac{g}{F}$$

Again, F must be calculated. For this problem,

$$F = \frac{111.4\ RPM}{60} = 1.86\ Hz,\ and$$

$$V = 61.44\ \frac{g}{F} = 61.44\left(\frac{0.01}{1.86}\right) = 0.33\ IPS$$

Warning: These calculations are true only for 0.01 g's at the given motor speed.

Example 1-11: The vibration of the pump in Example 1-10 is 40 mils when measured with a displacement probe. The vibration level in terms of velocity (IPS) can be calculated directly from equation 1:

$$V = \pi FD = \pi(1.86)(40 \times 10^{-3}) = 0.23\ IPS$$

Example 1-12: The vibration level on a motor running at 400 RPM is 0.10 IPS. The corresponding acceleration is calculated using equation 4:

$$g = 0.0162\ VF = 0.0162\ (0.10)\left(\frac{400}{60}\right) = 0.011\ g's$$

Example 1-13: The vibration level for the pump in Example 1-10 is 10.1 mils (measured with a displacement probe). An accelerometer would read, from equation 3:

$$g = 0.0511\ F^2D = 0.0511(1.86)^2\ (10.1 \times 10^{-3}) = 0.0018\ g's$$

Example 1-14: The vibration level on a motor running at 1800 RPM is 0.3 IPS (measured with a velocity transducer). The displacement in mils can be calculated by using equation 5:

$$D = 0.3183 \frac{V}{F}$$

First, frequency must be calculated in Hertz.

$$F = \frac{1800\ RPM}{60} = 30\ Hz$$

Substituting this value in equation 5:

$$D = 0.3183 \left(\frac{0.3}{30}\right) = 3.18\ mils$$

Example 1-15: Vibration level of the motor in Example 1-14 can be determined in g's by using equation 4:

$$g = 0.0162\ VF = 0.0162\ (30)\ (0.3) = 0.146\ g's$$

Example 1-16: If the vibration level of the motor in Example 1-14 is measured with an accelerometer to be 0.146 g's, then the displacement can be calculated using equation 6:

$$D = 19.57 \frac{g}{F^2} = 19.57 \left(\frac{0.146}{30^2}\right) = 3.18\ mils$$

Example 1-17: The vibration level of the motor in Example 1-14 can also be computed in IPS by using equation 2:

$$V = 61.44 \frac{g}{F} = 61.44 \left(\frac{0.146}{30}\right) = 0.3\ IPS$$

The above examples are useful. However, it may not be convenient to perform the mechanical calculations each time conversion is needed and the formulas and constants may not be available. A simpler, better, and faster method is the Vibration Calculator Program. The main screen contains three frequency response curves for a pure imbalance condition: displacement, velocity, and acceleration. A menu is also provided. Table 1-4 contains the menu contents.

The operator enters the frequency of interest and the vibration level in engineering units: either displacement, velocity, or acceleration. The program then calculates the other two engineering units. The program also calculates engineering units from dB and transducer sensitivity or calculates dB from transducer sensitivity and engineering units.

Fig. 1-28 contains the frequency response curve for displacement. The x axis is frequency in Hertz and the y axis is amplitude in engineering units. Please note the displacement is 1 mil at 20 Hz, about 2.3 mils at 30 Hz and 4 mils at 40 Hz.

Table 1-4. Vibration Calculator Program.

MAIN SCREEN MENU

Frequency (Hertz):	20.00
Displacement (Mil):	1.00
Velocity (IPS):	0.06
Acceleration (G):	0.02
Sensitivity (mV/EU):	375.00
Vibr. Level (EU):	1.00
dB:	51.48

Fig. 1-29 contains the frequency response curve for velocity. The x and y axes are the same as in Fig. 1-28. Please note the velocity is 0.07 IPS at 20 Hz, 0.21 IPS at 30 Hz, 0.45 IPS at 40 Hz, and 1 IPS at 50 Hz.

Fig. 1-30 contains the frequency response curve for acceleration. The x and y axes are the same as stated previously. Please note the acceleration is 0.02 g's at 20 Hz, 0.1 g's at 30 Hz, and 0.34 g's at 40 Hz.

Figs. 1-28 through 1-30 represent the effects of imbalance in displacement, velocity, and acceleration as speed is increased or decreased. These graphs can be compared with actual coast down data to identify imbalance, misalignment, bent shaft, and resonance problems.

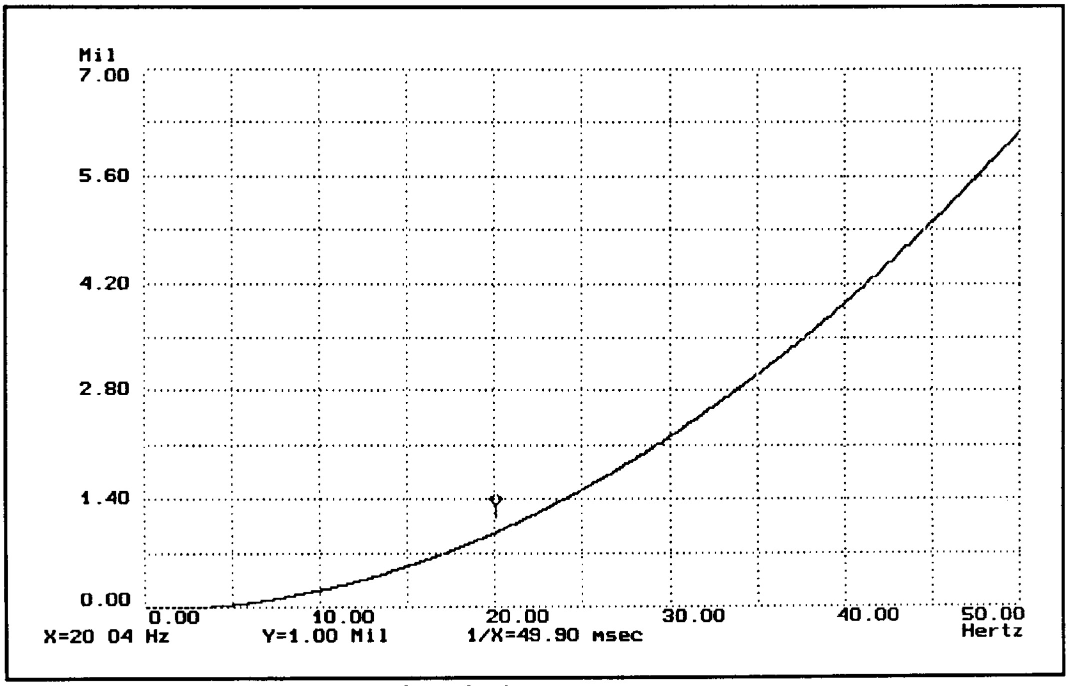


Fig. 1-28. Imbalance Response for Displacement.

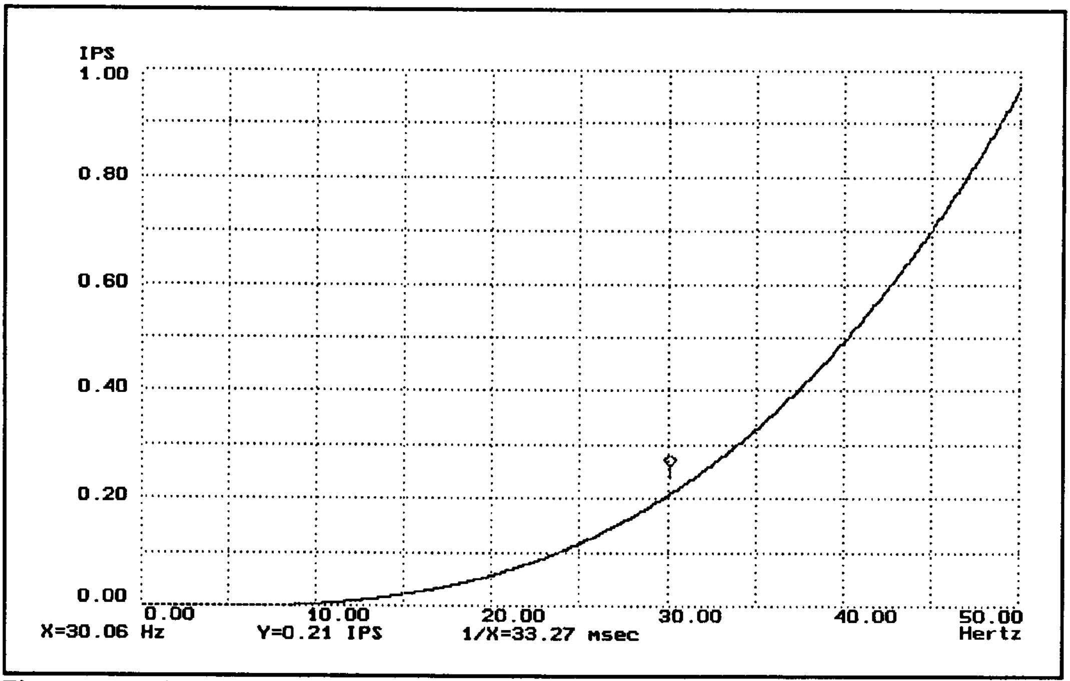


Fig. 1-29. Imbalance Response for Velocity.

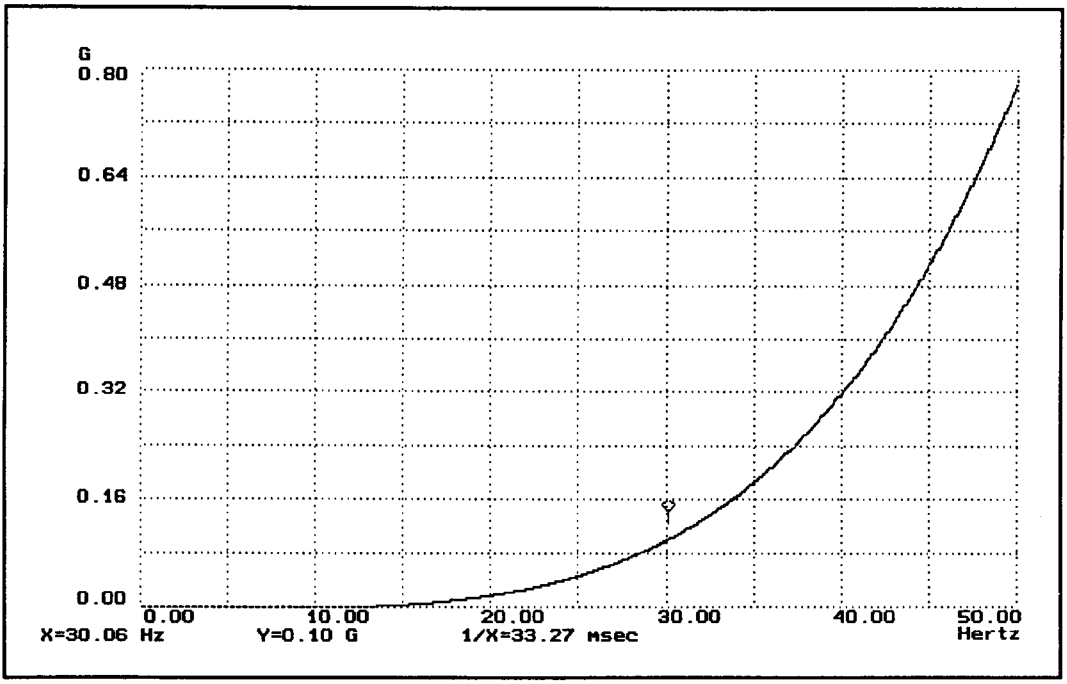


Fig. 1-30. Imbalance Response for Acceleration.

The program and graphs provide quick conversions between the three engineering units without having to remember formulas or perform calculations. The program also calculates vibration level from transducer sensitivity and dB, or dB from transducer sensitivity and vibration level.

WAYS OF MEASURING VIBRATION

Vibration can be measured by displacement, velocity, and acceleration. Transducers, meters, data collectors, and real-time analyzers are some of the tools capable of measuring vibration levels. Displacement is measured with a contacting displacement transducer. The contacting displacement transducer is a mechanical device that measures relative motion between two surfaces. The tip of the contacting displacement transducer is a plunger which moves back and forth with the relative motion between the two surfaces. The moving plunger changes the transformer's differential output. The resulting signal is the measured displacement.

The noncontacting displacement transducer measures relative motion with magnetic flux. (Magnetic flux is generated by the driver.) The noncontacting displacement transducer measures the dynamic displacement using magnetic fields. The tip of the noncontacting displacement transducer is a small coil which measures the change in magnetic flux when the probe tip is near a ferromagnetic material. The contacting displacement transducer and noncontacting displacement transducer must be used when the frequencies generated are below 10 Hz. The contacting displacement transducer should not be used above 200 Hz. The noncontacting displacement transducer may not be useful above 600 Hz.

Velocity is measured with a velocity transducer that has a relatively flat frequency response between 10 and 2,000 Hz. The velocity transducer measures the vibration using a permanent magnet that is fixed, and a coil of wire mounted on a spring. When a vibration source is applied to the transducer, the coil moves over the magnet, producing a signal. In most applications, the velocity transducer is the best tool with which to measure and record vibration.

Acceleration is measured with an accelerometer. The frequency response is greatest when measuring high frequencies. An accelerometer measures the vibration generated by using piezoelectric crystals. A piezoelectric crystal produces a voltage signal when the crystal is deflected. A change in speed will cause the crystal to deflect and display a change in vibration. An accelerometer must be used when the expected frequencies are greater than 2,000 Hz.

It is possible to measure displacement with an accelerometer, acceleration with a velocity transducer, etc. However, the signals must be either differentiated or integrated. These conversions can be calculated with the equations given above or with the Vibration Calculator Program.

RELATION BETWEEN DIAMETER, SPEED, AND RPM

The speed paper or other material travels through a machine can be obtained at various readout locations on the machine. Paper speed is useful in determining the RPM of a specific roll. The displacement of an element of paper is equal to the displacement of an element on the surface of the roll. Therefore, the speed of the paper over a roll is equal to the speed of the roll surface. The equation for the circumference of a circle, $C = \pi d$, indicates how far the roll element travels during one complete revolution, where d equals the roll diameter in feet. Given the paper speed in FPM, the RPM of a roll can now be determined from:

$$d = \frac{ps}{(RPM)\pi}, \qquad ps = \pi d(RPM), \qquad RPM = \frac{ps}{(\pi d)}$$

where ps is the paper speed.

Example 1-18: Paper speed through a dryer section is 1750 FPM. The diameter of a dryer roll is 5 feet. The speed of the dryer in RPM is:

$$RPM = \frac{ps}{\pi d} = \frac{1750}{\pi(5)} = 111.4$$

HOW TO DETERMINE MACHINE SPEED IN FPM FROM THE VIBRATION DATA

Vibration data from a felt roll or the fundamental frequency generated by a felt roll are typically from about 2 Hz to about 18 Hz. Look for these frequencies in the spectrum. If a spectral line is found that could be the roll speed, do a trial paper speed calculation. For example, if a spectral line is present at 6.5 Hz and the felt roll diameter is about 16 inches, convert 6.5 Hz to RPM by multiplying by 60: 6.5 x 60 = 390 RPM. Next, convert roll diameter in inches to feet by dividing by 12:

$$\frac{16}{12} = 1.333$$

Multiply roll diameter in feet by pi: $1.33 \times \pi = 4.178$.

This is the roll circumference in feet. Now multiply the roll circumference, 4.178, by the roll speed of 390: 4.178 x 390 = 1629 feet per minute (paper speed).

This speed can be verified by gearmesh frequency from the dryer roll. For example, if the gearmesh frequency is 138.4 Hz, divide the gearmesh frequency by the number of teeth on the gear (80 teeth):

$$\frac{138.4}{80} = 1.73\ Hz\ (dryer\ speed)$$

1.73 x 60 = 103.8 RPM is the dryer speed. Most dryer cans are 15.7 feet in circumference. Thus, 15.7 x 103.8 = 1629 FPM. This is the same speed obtained in the first test.

Paper machine speed can also be verified with bearing frequency. For example, an SKF 22318 bearing that has a defect on the outer race can generate a frequency of 40.17 Hz, if the roll is rotating at 6.5 Hz or 390 RPM. A fundamental frequency at 40.17 Hz and harmonics in a spectrum from a felt roll would indicate a defect on the outer race. Divide

40.17 by 6.18 (the ball pass frequency of the outer race at 1 Hz): 40.17/6.18 = 6.5 Hz. The roll must therefore turn at 6.5 Hz or 390 RPM in order to generate a frequency of 40.17 Hz if a defect is on the outer race. Finally, 390 x 4.178 (roll circumference) = 1629 FPM machine speed.

CONCLUSION AND EFFICIENCIES

Thus far, this textbook has explained the basics in machinery vibration analysis: Why and how a mechanical signal can be converted into an electrical signal and analyzed; various types of motion; different ways to measure motion; sources of frequencies; complex machine problems (mixers, detectors, resonance, and damping); relationships between velocity, displacement, and acceleration; units of measurement for velocity, displacement, and acceleration; and problem examples. These basics are the first step to understanding vibration analysis.

Vibration analysis is one of the best ways for companies with heavy industrial rotating machinery to save money. With the use of the above basics, companies can stop fixing equipment that is not broken, improve the quality of spares, predict impending failures, and reduce downtime.

CHAPTER TWO: TIME AND FREQUENCY ANALYSIS TECHNIQUES

INTRODUCTION

Accurate diagnosis of problems in rotating machines requires a thorough understanding of the time domain signal and the frequency domain spectra. Following are some of the reasons:

1. The time signal is a plot of amplitude versus time. This signal contains all frequencies, harmonics, and subharmonics. The phase relationships of these frequencies are also contained in the signal. Pulses, amplitude modulation, frequency modulation, truncation, and distortion are also present.

2. The frequency spectra are plots of amplitude versus frequency. These spectra contain frequency, harmonics, subharmonics, and sum and difference frequencies. The FFT produces the frequency spectrum from the time signal, based on electronic physics. However, during the process, some information is lost. For example, phase, true amplitude of pulses, nature of the pulses, bandwidth, and the various forms of modulation are not easily identified in the frequency spectrum.

3. A mechanical machine may not generate a fundamental plus harmonics in the same way as in the electronic world. However, a rotating machine does generate relatively linear signals when a linear problem exists, such as imbalance. A machine can generate a distorted signal as a result of a nonlinear problem. This distorted signal is a composite signal, as would be obtained after various frequencies and harmonics are combined.

4. For the above reasons, various time signals can produce the same frequency spectrum. This explains why the time signal must be considered. Costly errors in diagnostics and loss of credibility could occur if the time signal is not analyzed.

Before analyzing the time signal, an understanding of how frequencies add and subtract, and the effects of the phase relationships is required. It may be helpful to remember that multiplication is a series of additions, and division is a series of subtractions. This chapter explains the time signal in a nonmathematical way. However, extensive use of a computer and the **Signal Analysis Program (SAP)** are necessary to provide graphic representations of various signals and how they mix. The **SAP** software performs the required mathematics.

BASIC PHYSICS

All things in the universe obey the basic laws of physics. Vibration signals from rotating machinery must obey these same basic laws of physics. This is why we can take data from either side of a motor and receive the same results. (Some slight variations can occur in nonlinear systems because of transfer functions.)

In a pure linear system, data taken in different directions around a motor should be the same, except for phase.

CHAPTER ONE made the point that a sine wave is the plot of a circle against time. All complete circles contain 360 degrees and all complete sine waves contain 360 degrees. The phase of a signal can be anything from 0 to 360 degrees, depending on the reference point. For example, in comparing a signal taken from the horizontal direction with a signal taken from the vertical direction, one signal should lag the other signal by 90 degrees. This is because the positions from horizontal to vertical are 90 degrees apart on the machine. This phase relationship should also apply to other data taken at various points around the machine.

The time signal is continuous from start to stop of the machine. Concerning the phase of the fundamental frequency, the starting point happens purely by chance, depending on the exact instant of time when data collection is started. Once again, we are saved by the basic laws of physics. We can start taking data at any instant of time or location, and it does not make any difference. The reason for this is the phase relationship between the fundamental and other frequencies, or a once-per-revolution marker, will remain constant.

These frequencies will add and subtract, depending on the phase relationship. When the signals are in phase, the amplitudes will add. This is why positive sidebands occur on some frequencies. When the signals are out of phase, they will subtract. This is why negative sidebands occur. This also explains how and why truncation occurs.

Because of the above physics, a given frequency spectrum and time signal can take many forms, depending on the phase relationships of the various signals, including harmonics. In rotating machines, several different problems can generate the same frequency spectrum. For example, a machine that is loose can generate a fundamental and the second harmonic. A machine that has a bent shaft can also generate a fundamental and the second harmonic. The only way to determine which problem exists is to determine the phase relationship between the fundamental and the second harmonic. If these two signals are in phase, the shaft is bent. If the two signals are out of phase, the machine is loose. Currently, the only way to determine this phase relationship is with the time signal. The meanings of various phase relationships for specific problems are discussed in more detail in other chapters.

SINGLE FREQUENCY

A single frequency is often referred to as a discrete frequency and is the simplest form of frequency data. Fig. 2-1 contains the time signal of a single frequency and the resulting frequency spectrum from the **Signal Analysis Program**. The time period for each cycle is 0.01667 seconds and the signal is sinusoidal.

Sinusoidal simply means the signal follows the sine function. Mathematically, the time signal is:

$$y = A \cos (360\, f\, t + \theta),$$

where

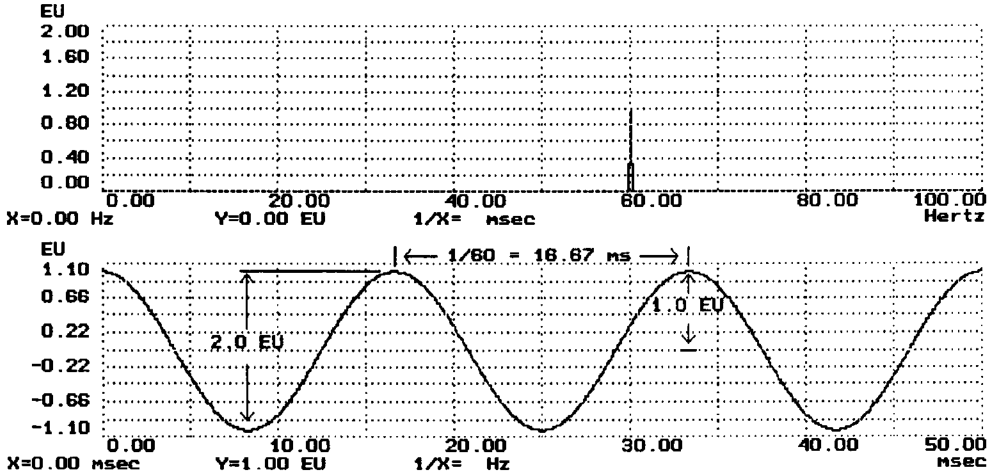
y = *instantaneous value of the signal*

A = *the zero-to-peak amplitude*

θ = *the phase angle (degrees)*

t = *the time variable (seconds)*

f = *the frequency variable (Hertz)*



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-1. Single Frequency.

The time signals for **SAP** are based on the cosine function. The reason for using the cosine instead of the sine is because the starting point at time zero of the fundamental is the highest point. The cosine is the same as the sine, except for a 90 degree phase shift. Fig. 2-1 is a cosine function with a peak amplitude of 1.0 engineering units (EU). In the real world, the signal can start at any point between 0 and 360 degrees. Fig 2-2 is a representation of a sine function with a peak amplitude of 1.0 EU. Comparing this signal to the signal in Fig. 2-1, a phase shift of -90 degrees is present. The signal in Fig. 2-2 represents the sine wave with the phase starting at zero degrees. The difference between the signals in Fig. 2-1 and Fig. 2-2 is the effective difference between taking data in the horizontal and vertical positions. The signals are 90 degrees out of phase.

Cosine is used for **SAP** because the real part of the FFT is based on the cosine function. The following mathematical description of the FFT is given for those readers interested in the mathematics used to break the complex wave forms down into the discrete frequencies. The FFT is based on:

$$\cos(x) - j\sin(x),$$

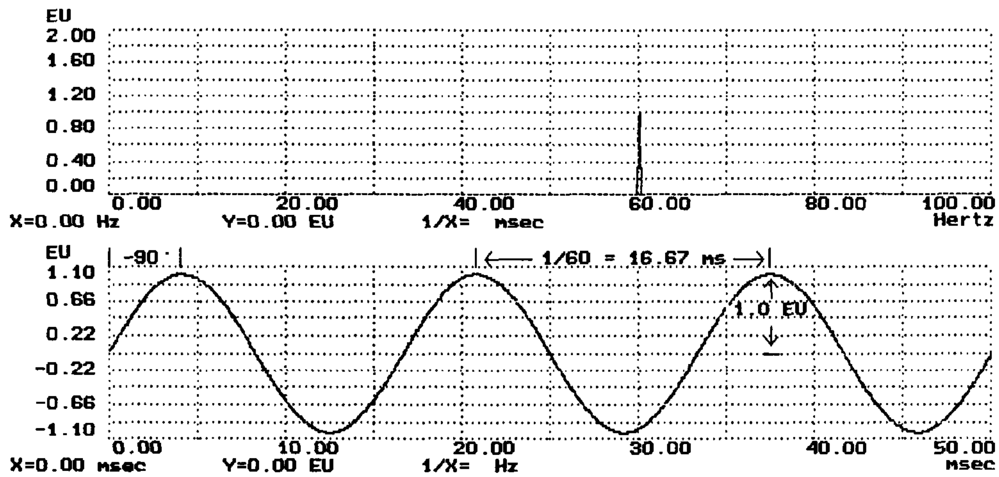
where

$\cos(x)$ - *real component of signal*

j - *imaginary variable* ($j^2 = -1$)

$\sin(x)$ - *imaginary component of signal*

When the time signal is processed by the FFT, it is divided into the amplitudes and phases of the individual cosine and sine functions. When a spectrum is displayed, the plot is an amplitude plot of the frequencies. If a complex FFT is performed, both the



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	-90.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

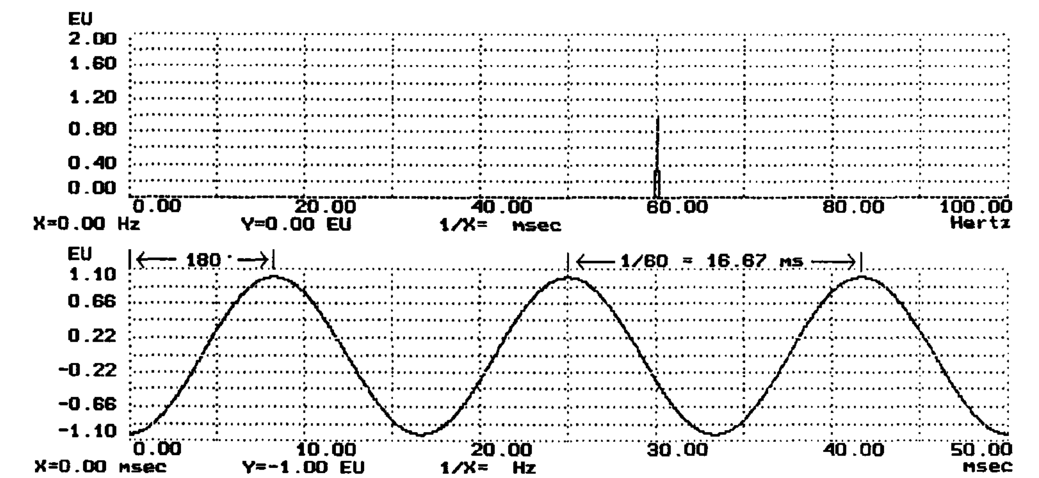
Fig. 2-2. Single Frequency -90 Degree Phase Shift.

amplitude and phase are available, but only the amplitude is displayed. In the introduction, the frequency spectrum was said to lose the phase. In actuality, it may not be lost. It is just not listed or is discarded. Unless the phase is retained and viewed, the time domain signal must be used to identify phase relationships.

To reconstruct the time signals from the frequency domain, a starting point must be selected. When time equals zero, the cosine starts at maximum amplitude and the sine starts at zero amplitude. It is more consistent to start at the maximum amplitude of all signals, so the signals add at a time equal to zero.

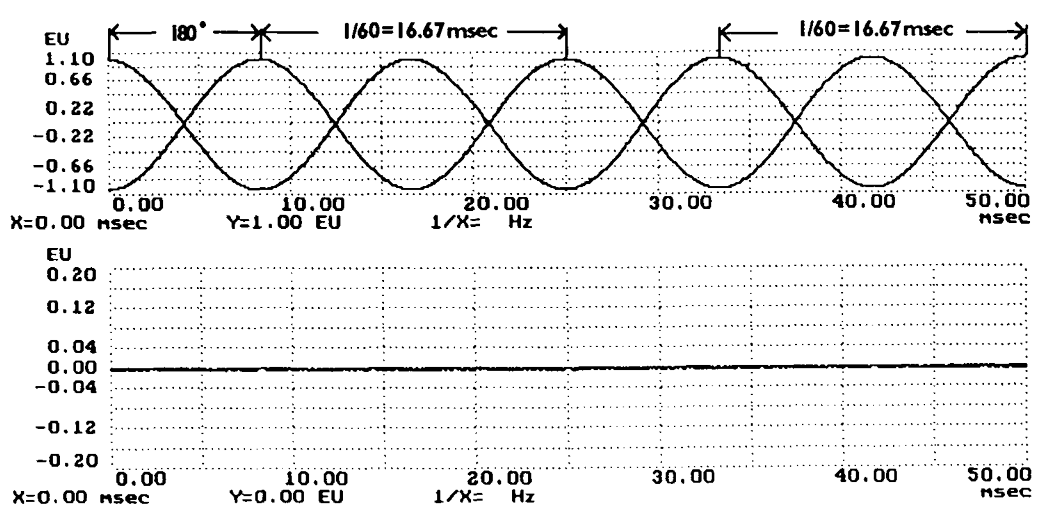
The signal in Fig. 2-3 starts at maximum negative amplitude. This is a cosine function with a 180 degree phase shift. In rotating machinery, data taken from opposite sides of a motor should be the same, except the phases of the fundamentals should be 180 degrees apart. This obeys all laws of physics.

Amplitudes of signals in rotating machinery will add or subtract, depending on their phase relationships. The top grid in Fig. 2-4 contains two frequencies at 60 Hz which are 180 degrees out of phase. The time signal in the bottom grid has zero amplitude because the two signals are equal in amplitude, 180 degrees out of phase, and one signal cancels the other signal. This is the analytical proof. The empirical proof can be observed in balancing. Equipment is out of balance because one side is heavier than the other side. This generates a signal, as in Fig. 2-1. The equipment is balanced by placing a weight on the light side equal to the heavy place. This generates another signal 180 degrees out of phase, as in Fig. 2-3. If the two signals were equal in amplitude and 180 degrees out of phase, they would cancel, as in Fig. 2-4. In the real world, such perfection does not exist. However, it does prove that the amplitudes of out-of-phase signals subtract.



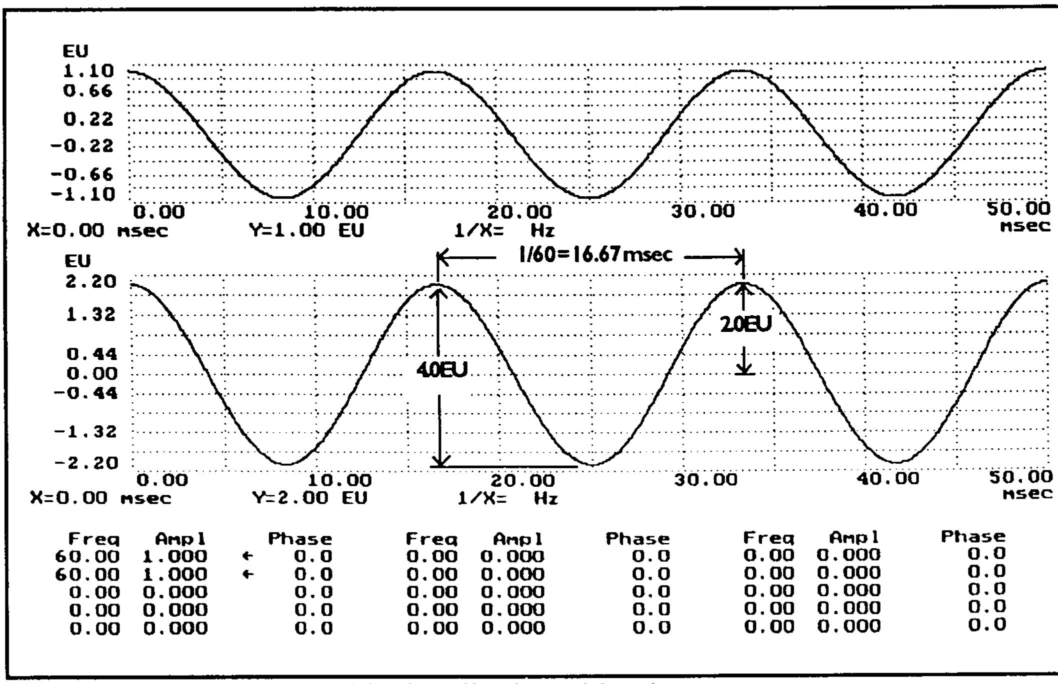
Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	180.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-3. Single Frequency 180 Degree Phase Shift.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	← 0.0	0.00	0.000	0.0	0.00	0.000	0.0
60.00	1.000	← 180.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-4. Two Signals 180 Degrees out of Phase.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000 ←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
60.00	1.000 ←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-5. Two Signals Equal in Amplitude and in Phase.

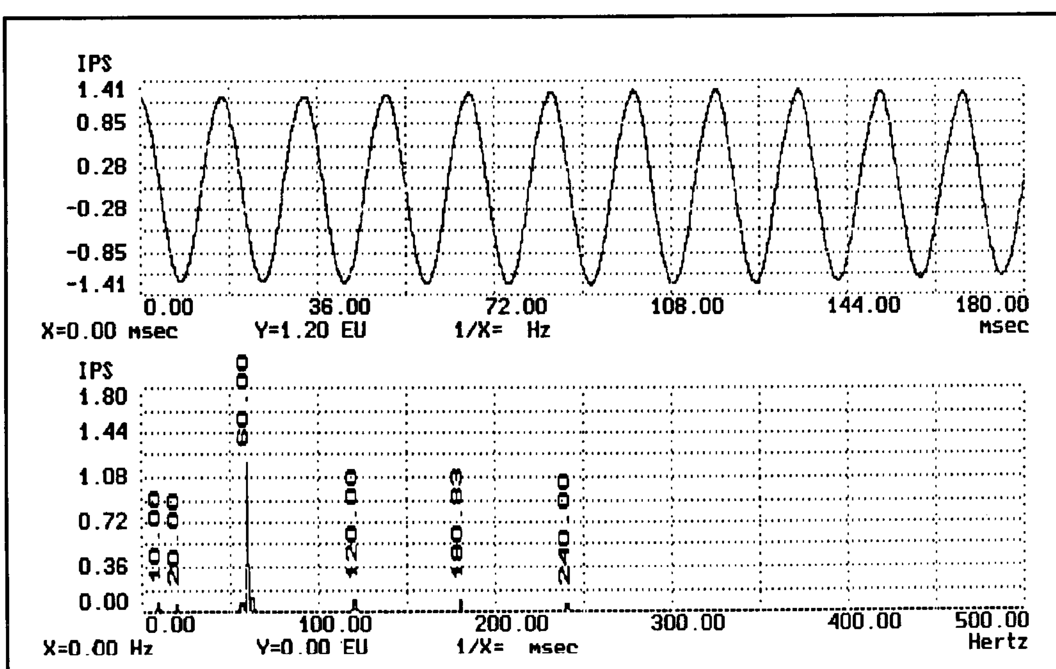


Fig. 2-6. Boiler Feedwater Pump with Imbalance.

The top grid in Fig. 2-5 contains two in-phase signals equal in amplitude. The top grid appears to contain only one signal. The signal in the bottom grid in Fig. 2-5 is the result of signal 1 plus signal 2, and the amplitude has doubled. The cases discussed in Figs. 2-4 and 2-5 prove that the amplitudes of frequencies can subtract when out of phase, and add when in phase. This data also proves the overall vibration measured in most machines can be either over or understated. Therefore, phase must be accounted for to accurately identify the severity of a problem. This explains why "amplitude trending" is misleading.

Fig. 2-6 is an example of a single frequency. The data was taken from a 3600 RPM Boiler Feedwater Pump that is out of balance. This example has some small differences from the theoretical single frequency. The real world case is not a perfect signal and shows some effects of distortion. The distortion in the time signal may not be visible to the unaided eye, but it is present. The spectral line at 60 Hz is the frequency of the time signal. The low frequency noise and the harmonics of 60 Hz are caused by distortion. In this case, the frequencies at these levels are considered insignificant because they could be caused by very minor nonlinearity in the machine or some other source.

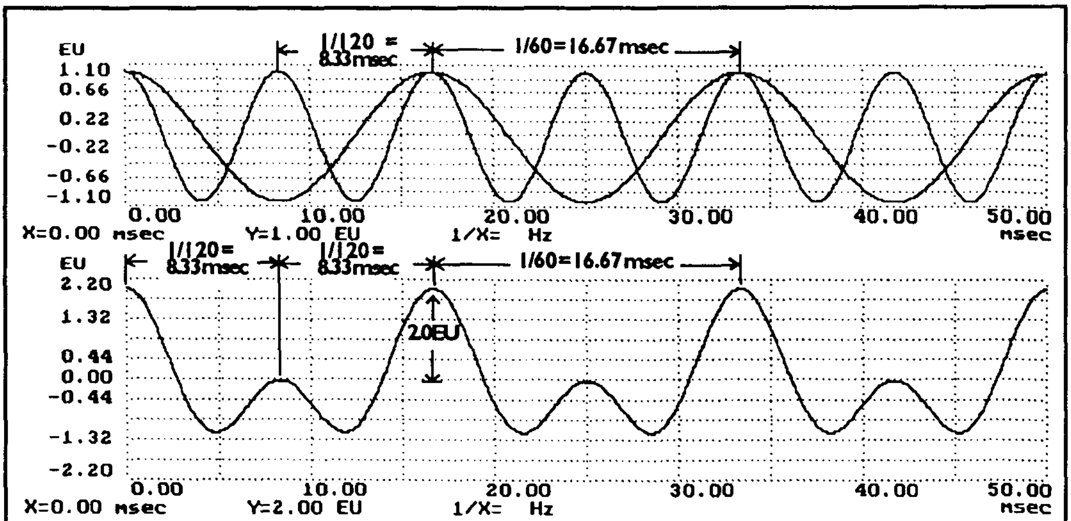
SINGLE FREQUENCY WITH HARMONICS

The same physics that permit two signals to add and subtract apply to a single frequency with harmonics. A harmonic is some exact multiple of a discrete frequency. The discrete frequency, called the fundamental, is the first harmonic. The second frequency, which is two times the fundamental frequency, is the second harmonic. The second, third, fourth, etc., harmonics can be either in phase or out of phase with the fundamental. The phase relationships between the fundamental and the harmonics are valuable in diagnosing problems in rotating machines. Failure to understand and use the time signal and harmonic phase can result in diagnostic errors.

A single frequency without harmonics will have one positive-going peak per time period, as indicated in Fig. 2-1. The number of positive-going peaks in one time period of the fundamental frequency identifies the highest number of true harmonics. This is true for a single frequency with harmonics only, and is true regardless of the phase relationships between the fundamental and the harmonics. The amplitudes of the fundamental and the harmonics determine the amplitudes of the positive-going peaks. However, the phase relationships of the harmonics to the fundamental determine the locations of the positive-going peaks in the signal. For example, the top grid in Fig. 2-7 contains a fundamental and second harmonic. The second harmonic is in phase with the fundamental and goes completely out of and back into phase with the fundamental once each time period of the fundamental. The bottom signal in Fig. 2-7 is the result of the top two signals adding and subtracting. It is important to note several points:

1. The amplitude of the positive-going composite signal is doubled because both signals are the same amplitude and the positive-going signals are added.

2. The amplitude of the negative-going composite signal is less because one cycle of the second harmonic is 180 degrees out of phase with the negative-going portion of the fundamental.



Freq	Ampl		Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	1.000	←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-7. Single Frequency with an In-Phase Harmonic.

3. The second positive-going peak is at the bottom of the fundamental, and the amplitude of the composite signal is zero. Once again, the reason for this is both signals are equal in amplitude and they subtract.

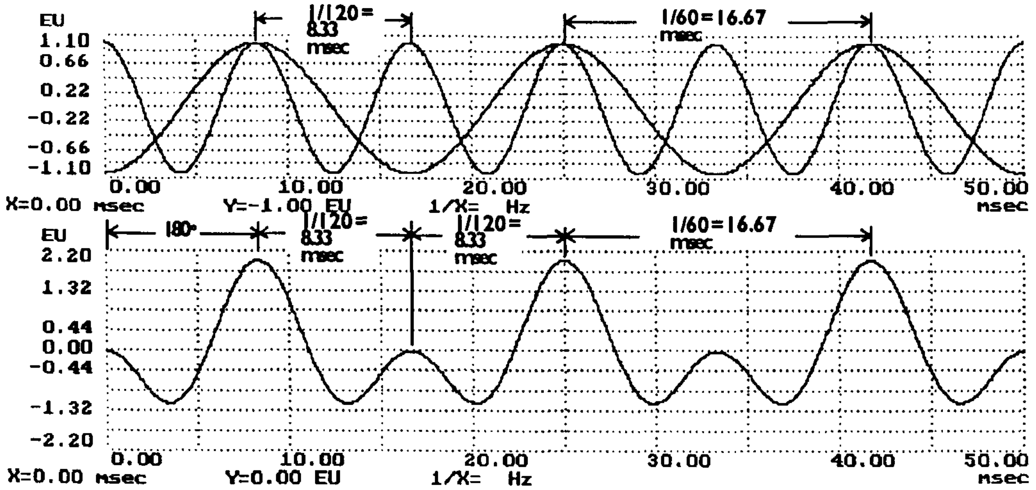
4. The composite signal is truncated on the bottom or negative side.

The time period for each frequency is marked on each grid.

The phase of the fundamental is changed 180 degrees in Fig. 2-8. The two signals are still in phase because the harmonics are referenced with respect to the fundamental, as can be seen in the top grid. Both signals in Fig. 2-7 start at the maximum value. In Fig. 2-8, the fundamental starts at the maximum negative value, and the second harmonic starts at maximum positive value.

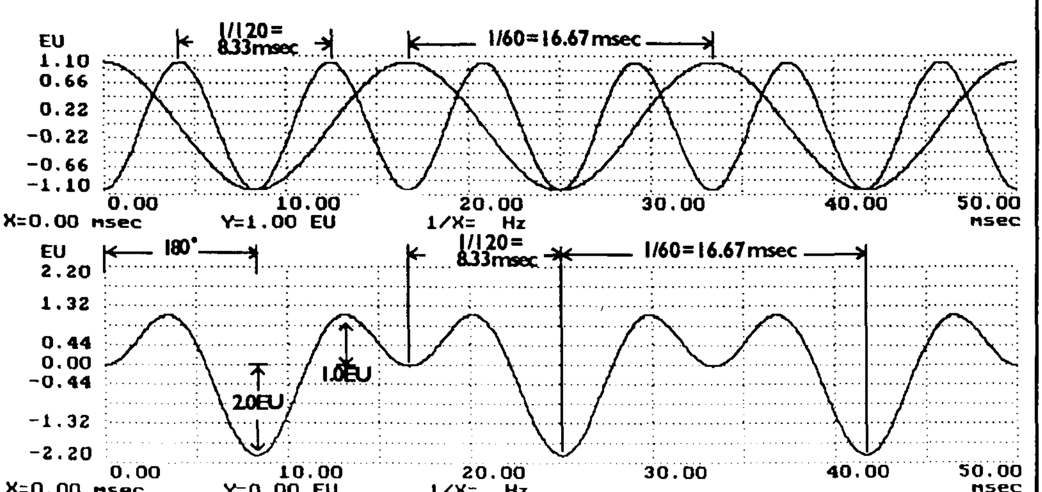
It is important to note that the phase of the harmonic in reference to the fundamental is at the start of the cycle. In Figs. 2-7 and 2-8, the harmonic is in phase with the fundamental. The highest point of both signals occurs at the same point. At any other point in the cycle, the harmonic goes out of phase and back into phase with the fundamental.

The net effect of the 180 degree phase shift is that the fundamental in the top grid and the composite signal in the bottom grid are shifted 180 degrees. The overall effect does not change the shape of the composite signal. The phase relationship between the fundamental and second harmonic is unchanged. In rotating equipment, the signals in Figs. 2-7 and 2-8 are the same result as could be obtained by taking data on opposite sides of a motor. That is, the fundamentals would be 180 degrees out of phase, but the second harmonics would still be in phase with the fundamentals. Again, this proves that data



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	← 180.0	0.00	0.000	0.0	120.00	1.000	← 0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-8. Single Freq. with a 180 Degree Phase Shift and Harmonic.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	← 0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	1.000	←-180.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-9. Single Freq. with a 180 Degree out-of-Phase Harmonic.

can be taken from either side of a machine without affecting the phase relationships of the harmonics.

The top grid in Fig. 2-9 contains a fundamental and a second harmonic. The second harmonic is 180 degrees out of phase with the fundamental. When this occurs, the second harmonic is in phase with the fundamental when both signals are maximum negative, and out of phase when the fundamental is maximum positive. This can be seen because the signals add when the fundamental is at maximum negative amplitude and subtract at maximum positive amplitude. The result is a higher negative amplitude in the composite signal in the bottom grid, and the two positive-going peaks are at the top. The top grid contains both signals and displays how they go into and out of phase as they did previously. However, the second harmonic subtracts from the fundamental at the start of the fundamental cycle. Please note the composite signal still has two positive-going peaks that identify the fundamental and second harmonic.

However, both positive-going peaks are at the top of the signal. This can only occur if the second harmonic is 180 degrees out of phase with the fundamental.

Several points concerning amplitude must be made for Fig. 2-9:

1. The zero-to-peak amplitude of the negative-going peak below zero is equal to the zero-to-peak amplitude of $F_1 + F_2$.

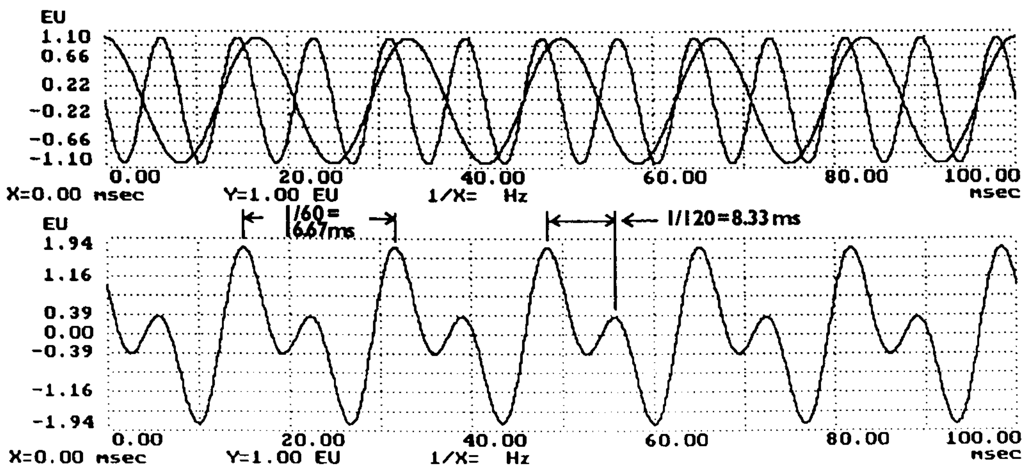
2. The zero-to-peak amplitudes of each positive-going peak are equal to the zero-to-peak amplitudes of F_1 and F_2.

3. These amplitude relationships are true only for two signals equal in amplitude and 180 degrees out of phase.

These phase and amplitude relationships hold true for linear systems. However, most real applications contain nonlinearities, called distortion. The distortion can appear in the signal as a phase shift in one or more of the harmonics. Distortion of the signal can also generate additional harmonics in the frequency domain which are not true harmonics of the signal. Therefore, the number of peaks in the time signal must be checked for true harmonic content.

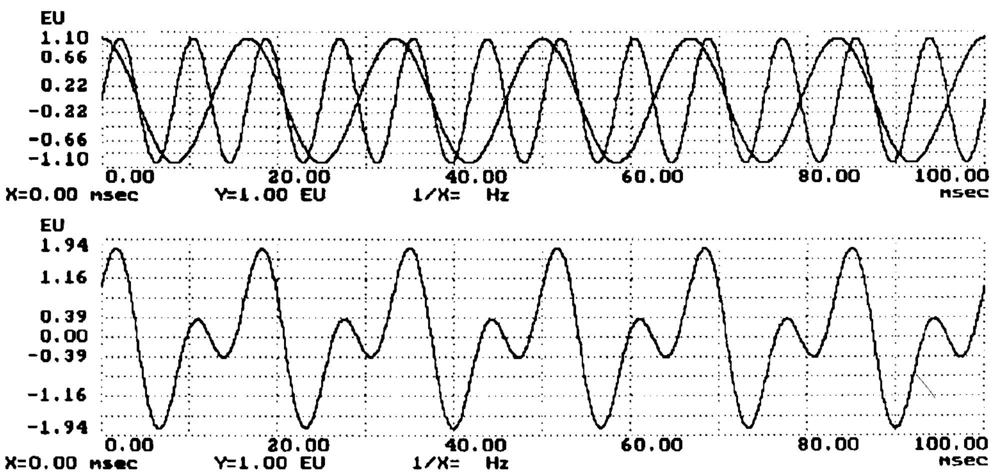
Continuing with phase relationships, the next step is to observe a phase shift of 90 degrees. Figs. 2-10 and 2-11 contain signals with a plus 90 and a minus 90 degree phase shift of the second harmonics, respectively. In Fig. 2-10, the composite signal still has two peaks per cycle, denoting the second harmonic. However, instead of a discrete high and low peak, the lower peak is shifted to the left, on the downward slope of the signal. Fig. 2-11 has the same uneven peaks, except the lower peak has shifted to the right and is on the upward slope of the signal. It can also be shown that any angle between 0 and 180 degrees can be approximated by looking at the various phase shifts. For example, a positive phase shift of the second harmonic will move the lower peak to the left. As the phase approaches 180 degrees, the lower peak will approach the same amplitude of the higher peak and the signal will look like the signal in Fig. 2-9.

In addition to phase shifts, the amplitudes can change. Fig. 2-12 shows the second harmonic changed to one half the amplitude of the fundamental. The starting amplitude of the composite signal is still the sum of the two amplitudes: 1.0 EU + 0.5 EU = 1.5 EU.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	← 0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	1.000	← 90.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-10. Single Freq. with 90 Degree Phase-Shifted Harmonic.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	← 0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	1.000	← -90.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-11. Single Freq. with a -90 Degree Phase-Shifted Harmonic.

However, the lower peak is smaller and does not have a zero amplitude. The two signals do not cancel each other because the second harmonic only subtracts one half of the fundamental amplitude when it is 180 degrees out of phase.

After seeing the effect of changing amplitude in Fig. 2-12, one can identify the effect of changing the amplitudes in other ways. Changing amplitude only affects the amplitude of the composite peak. It does not affect the number of peaks or the phase relationship of the composite.

The signals in Figs. 2-7 through 2-12 illustrate the effects of changing the phase and amplitude. With **SAP**, the amplitude, phase, and harmonic content can be identified, and these phase relationships between the fundamental and harmonics contain valuable diagnostic information. For example, if the spectra from a machine contain only a fundamental frequency generated by rotating speed and a second harmonic that is in phase with the fundamental, the shaft may be bent. If the second harmonic is out of phase with the fundamental, the machine may be loose. If the second harmonic of gearmesh frequency is out of phase with the fundamental, the gears have a backlash problem or are oscillating. If the second harmonic is in phase with the fundamental, the gears may be bottoming out. More examples and complete descriptions of problems will be addressed in later chapters.

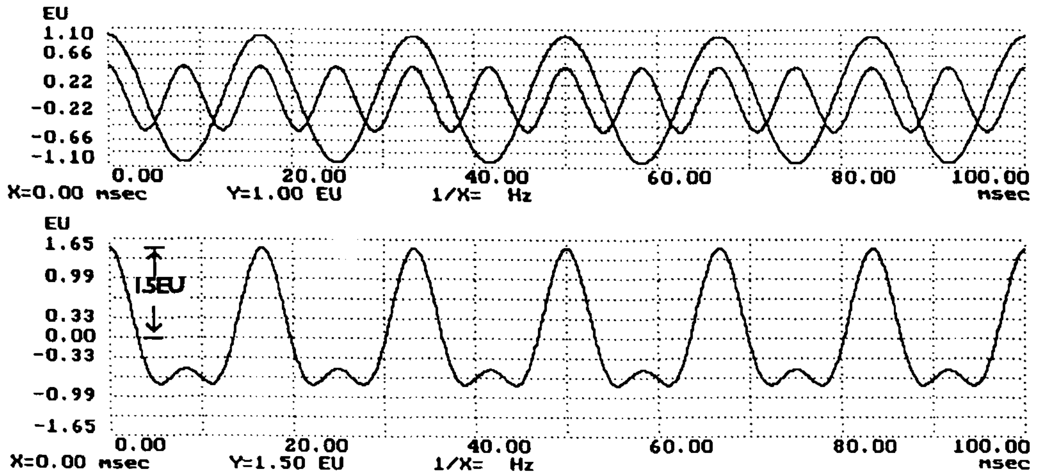
A good rule to remember is that if the source of the harmonic is tied to the source of the fundamental, such as a fixed, geared, or bolted coupling, the harmonic should be in phase. If the source of the harmonic is not tied to the source of the fundamental, the harmonic should be out of phase. This is also discussed in more detail later.

The addition of a third harmonic will now be examined, along with the effects of changing the phase and amplitude. Changing the amplitude changes the amplitude of the individual peaks, as with two harmonics. Fig. 2-13 contains a signal with three harmonics. Three positive peaks per cycle are present, indicating the three harmonics.

The amplitudes of the positive peaks above zero are equal to $F_1 + F_2 + F_3$: 1.0 EU + 1.0 EU + 1.0 EU = 3.0 EU. The high peaks occur at the fundamental frequency of 60 Hz. The lower peaks occur at the third harmonic frequency of 180 Hz.

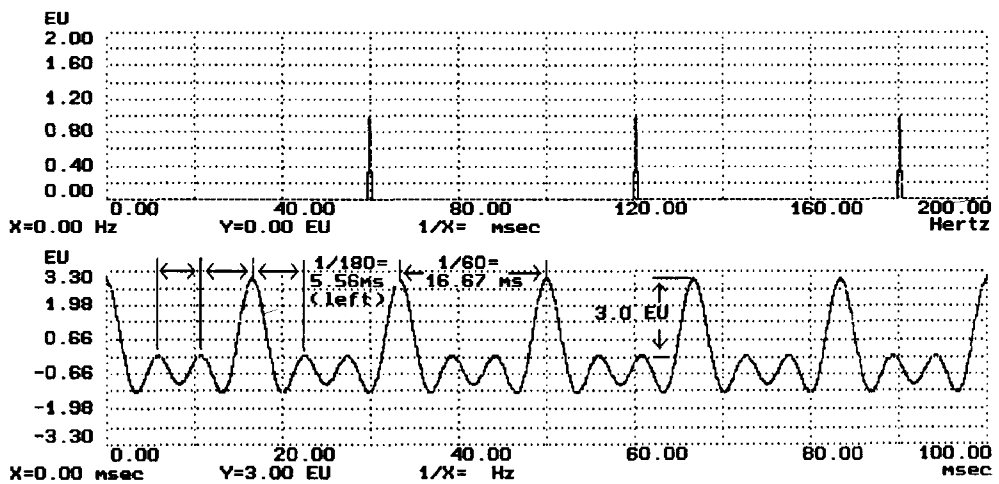
The signal in Fig. 2-14 contains only the first and third harmonics. The negative portion of the signal is more pronounced because the second harmonic is not present to subtract out the third harmonic. In fact, one cycle of the third harmonic is in phase with the negative portion, causing the two signals to add. The amplitudes of the negative and positive portions are equal to $F_1 + F_3$: 1.0 EU + 0.5 EU = 1.5 EU.

When the first three harmonics are distinctive, misalignment is indicated. This fact was documented several years ago by VCI. However, until recently, the differences between the many time signals that can generate the same spectrum were not evaluated. These differences can now be explained because of the phase relationships of the harmonics. The phase relationships of the harmonics can significantly alter the time signal without affecting the frequency spectrum. There are an infinite number of phase and amplitude combinations possible. Figs. 2-15 through 2-20 show representative signals that have been identified on live data. For example, Fig. 2-15 is a result of a misaligned, rigid coupling. Misalignment is indicated because of the distinct first three harmonics, which must be in phase because of the rigid coupling. If a rigid coupling is not used, the phase of the



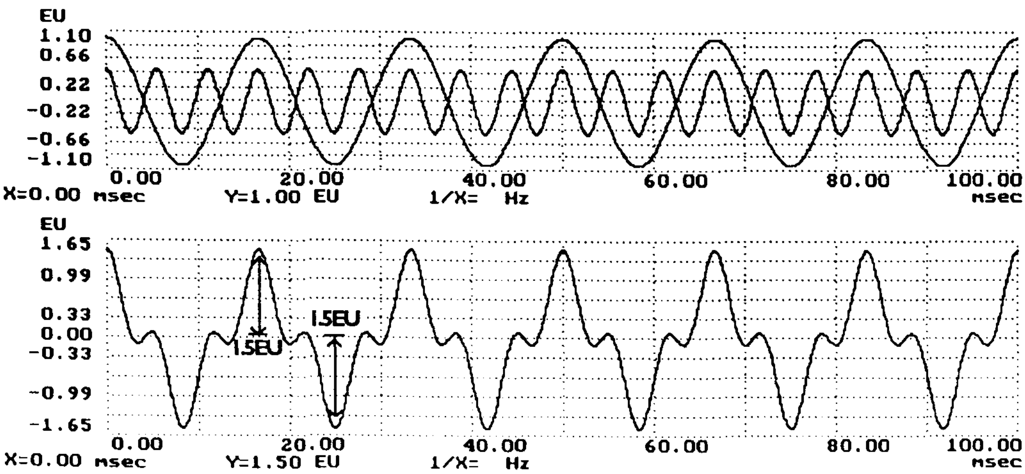
Freq	Ampl		Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	0.500	←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-12. Single Frequency with a Lower Amplitude Harmonic.



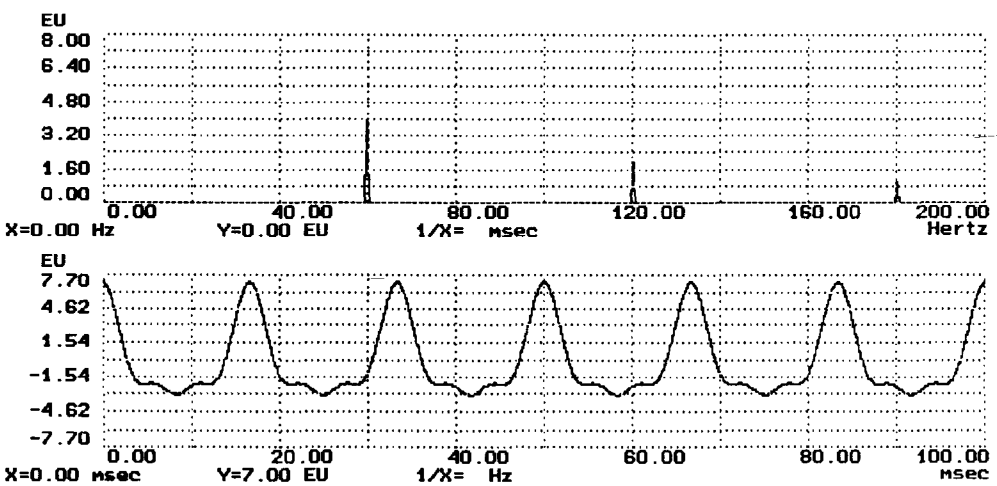
Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
180.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-13. Single Frequency with Two Harmonics.



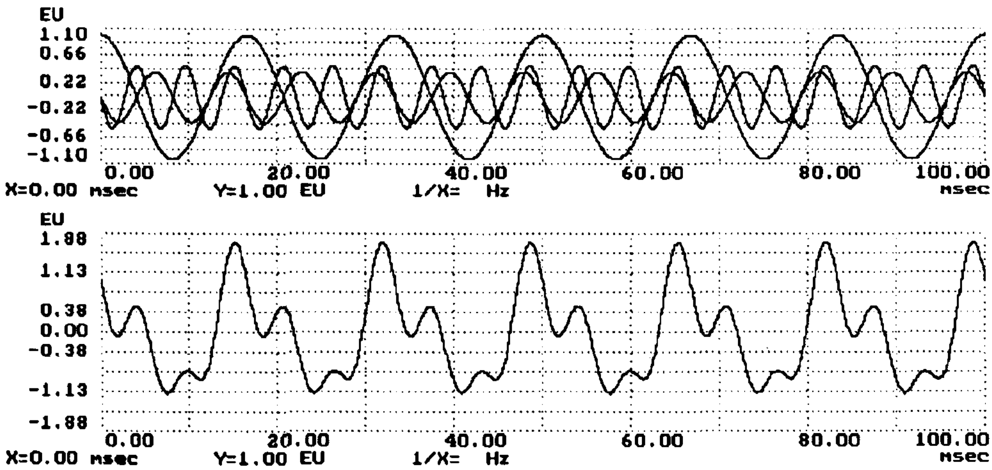
Freq	Ampl		Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0
180.00	0.500	←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-14. Single Frequency with Only Third Harmonic.



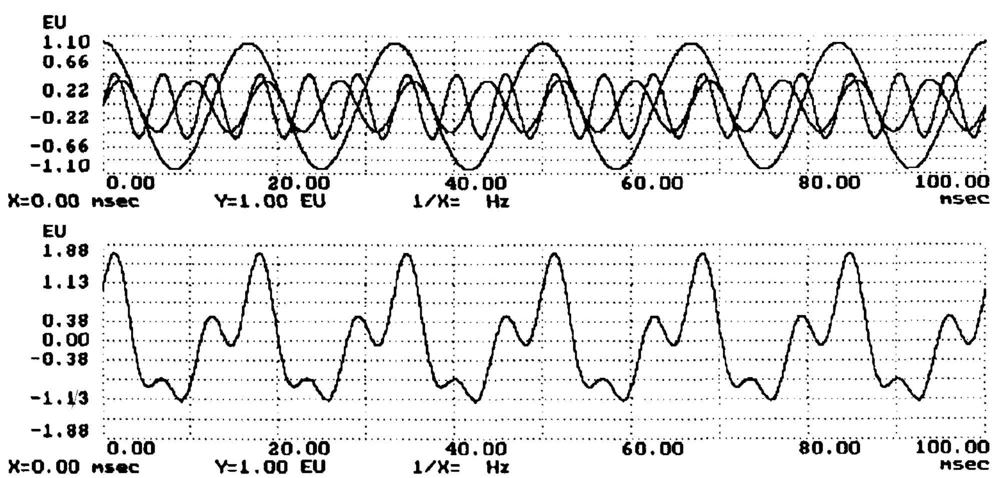
Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	4.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	2.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
180.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-15. Single Frequency with Two Harmonics.



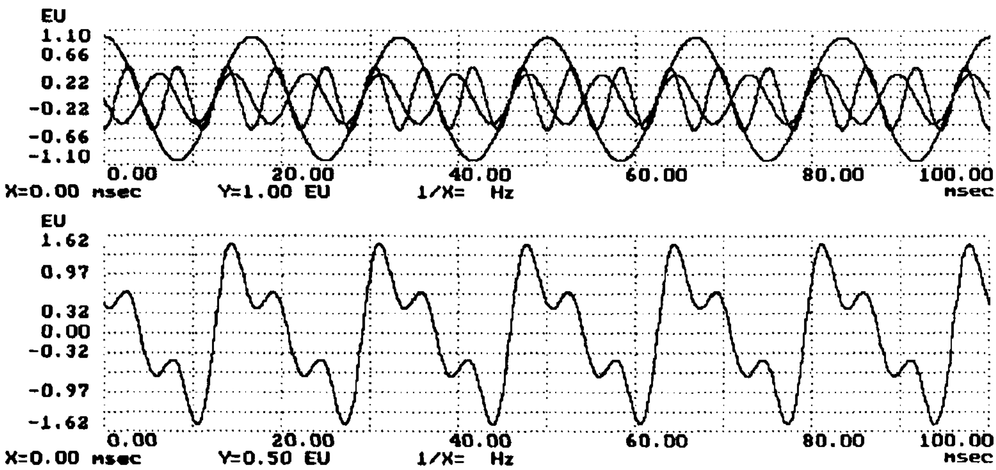
Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	← 0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	0.400	← 90.0	0.00	0.000	0.0	0.00	0.000	0.0
180.00	0.500	← 90.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-16. Single Frequency with Two Phase-Shifted Harmonics.



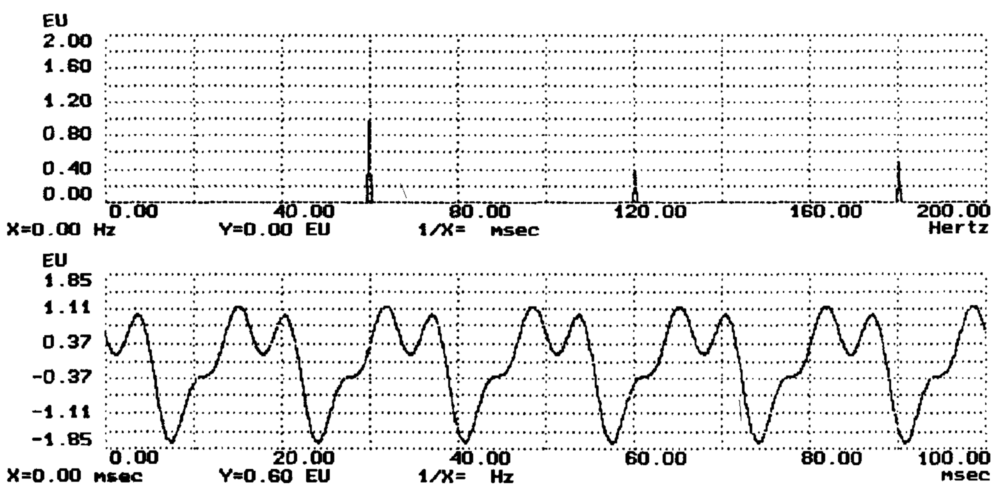
Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	← 0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	0.400	← -90.0	0.00	0.000	0.0	0.00	0.000	0.0
180.00	0.500	← -90.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-17. Single Frequency with Two Phase-Shifted Harmonics.



Freq	Ampl		Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	←	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	0.400	←	90.0	0.00	0.000	0.0	0.00	0.000	0.0
180.00	0.500	←	180.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000		0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-18. Single Frequency with Two Phase-Shifted Harmonics.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	0.400	180.0	0.00	0.000	0.0	0.00	0.000	0.0
180.00	0.500	90.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-19. Single Frequency with Two Phase-Shifted Harmonics.

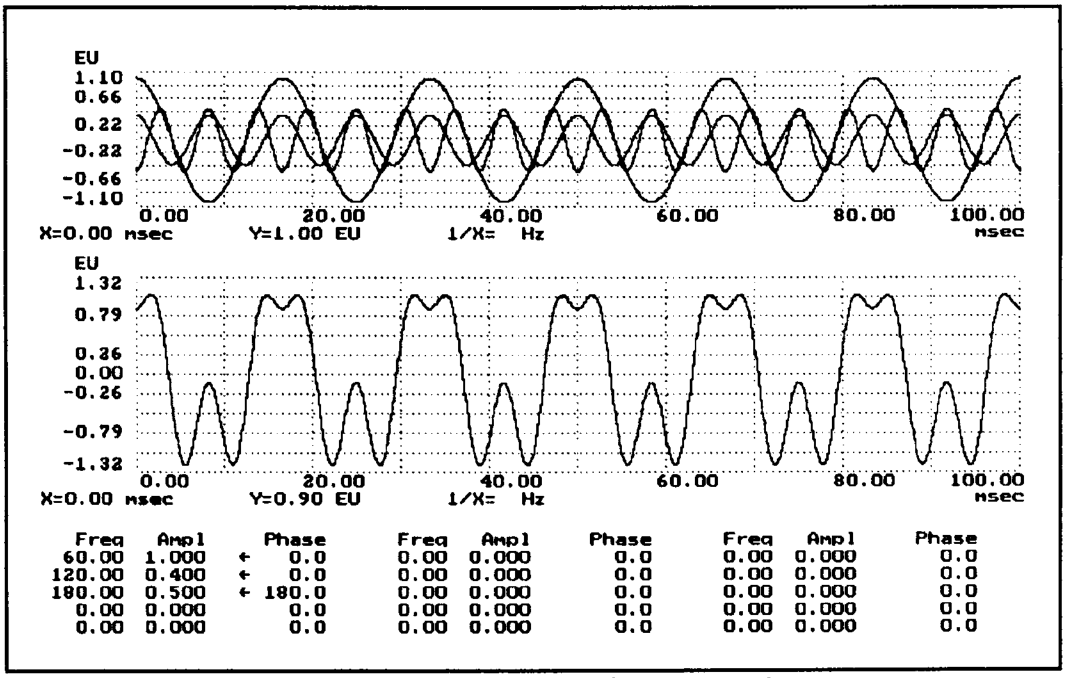


Fig. 2-20. Single Frequency with Two Phase-Shifted Harmonics.

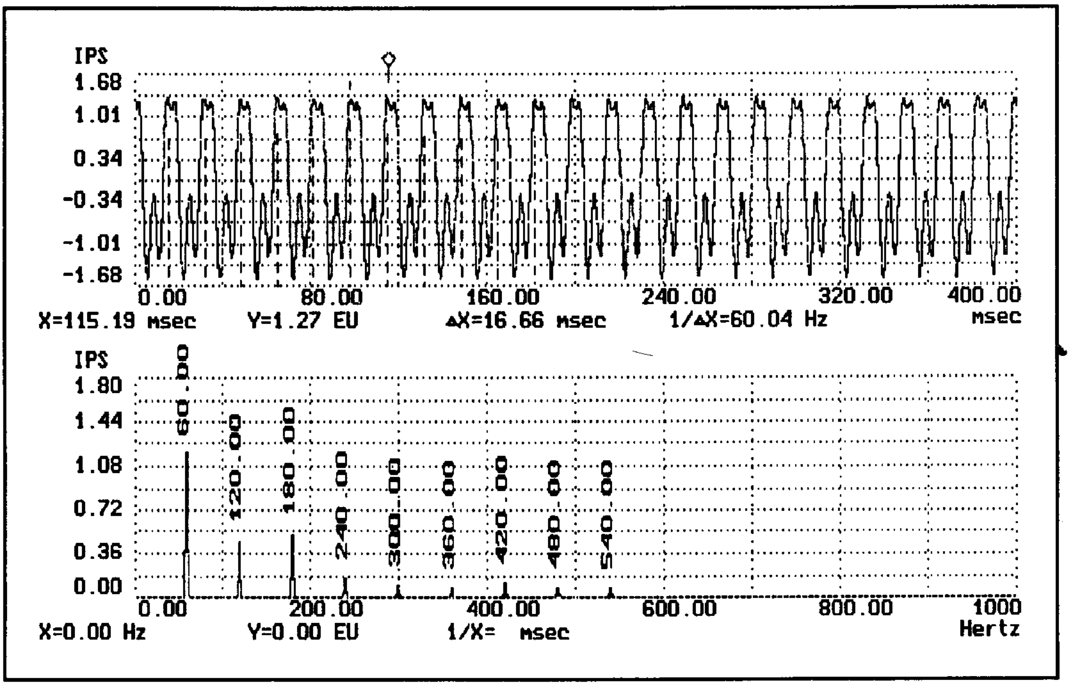


Fig. 2-21. Data from Main Turbine Generator.

harmonics will change, depending on the coupling. If the coupling is rigid, it acts in a linear manner. If the coupling is loose or nonrigid, it will appear nonlinear and out of phase. More examples with specific problems are discussed later.

Fig. 2-21 contains a spectrum from a main turbine generator. The time signal contains the basic features of the signal in Fig. 2-20. Three harmonics are present and the third harmonic is 180 degrees out of phase with the fundamental.

Note the signal amplitude is less on the positive half than the negative half. This is a form of distortion called truncation. Truncation occurs when the machine is more flexible in one direction than in the other direction. The first three harmonics in the frequency spectrum are also distinctive. These spectral lines are the result of the three positive-going peaks in the time signal. The low level harmonics in the frequency spectrum are a result of the distortion in the time signal, and are not true harmonics. The first three harmonics are true harmonics in this spectrum. The time domain signal is necessary to verify which harmonics are true and which harmonics are caused by distortion. In this case, the distortion is caused by various nonlinearities and is not required for analysis of this problem. However, this is not always the case.

CLIPPING

A signal is said to be clipped when a slight amount of the positive or negative signal is flattened, as in Fig. 2-22. The upper signal is an undistorted time signal. The lower signal is clipped at the bottom.

Fig. 2-23 contains a time signal and a spectrum from a motor. The time signal is clipped. Such a signal can be generated when a machine goes against a stop in one direction and cannot move further in that direction for a small period of time. As the cycle continues, the machine moves away from the stop in a relatively linear manner. The signal is distorted because the time period for the negative and positive portion is not the same. Clipping is also a form of distortion. The frequency spectrum contains very little harmonic content, because in order for harmonic content to be generated, the signal distortion must be repeatable. In this case, the signal distortion is not repeatable. The nonrepeatable distortion is noise. The "skirts" on either side of the 29.6 Hz spectral line, predominately on the low side, contain the noise. The spectrum is up off the baseline to the left of the spectral line, which indicates looseness or random motion.

Square Wave

A special case to note is a single frequency with only odd harmonics present, as in Fig. 2-24. The harmonics tend to cancel each other out, except for one positive and one negative peak per time period of the fundamental. The peaks have an amplitude equal to the sum of all the amplitudes added together. A special case of odd harmonics is shown in Fig. 2-25. Fig. 2-25 has only odd harmonics, and every other odd harmonic is 180 degrees out of phase. The resultant signal is a square wave. The amplitudes correspond to the amplitude of the fundamental divided by the harmonic number.

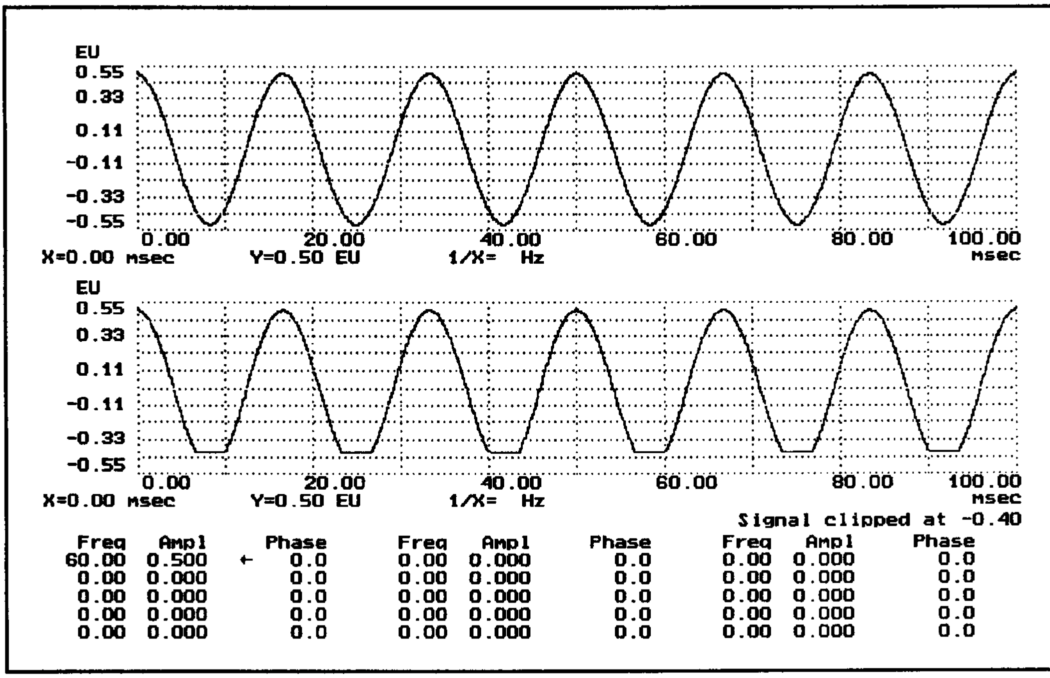


Fig. 2-22. Clipped Signal.

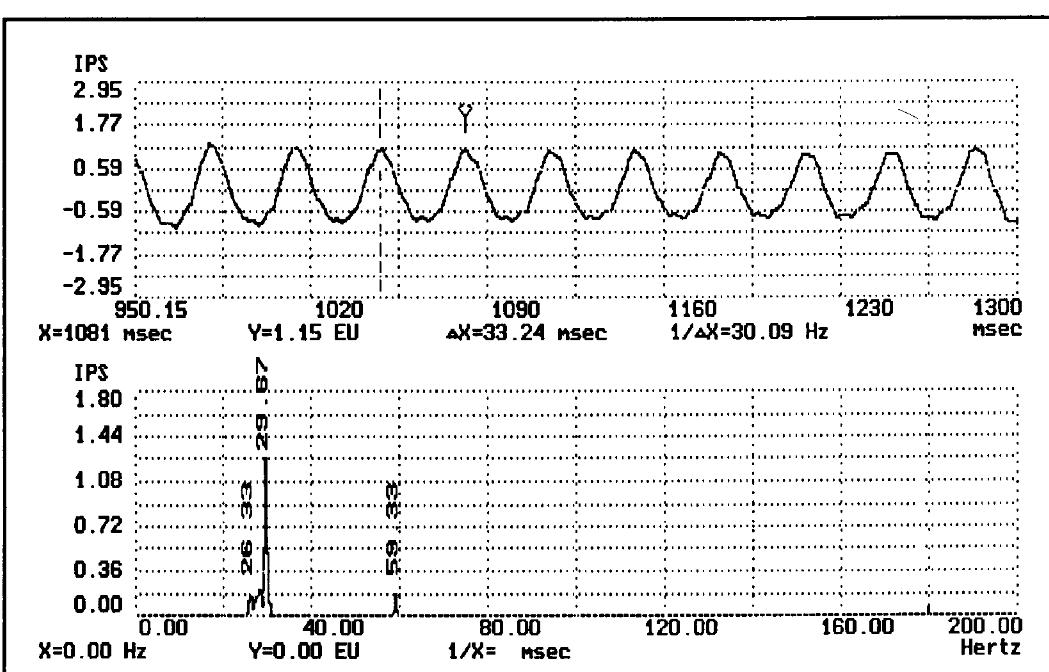
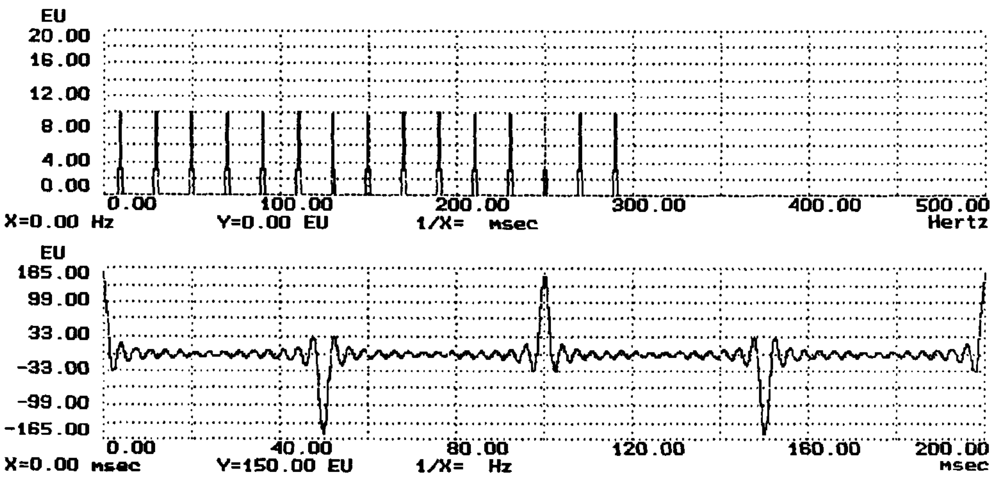
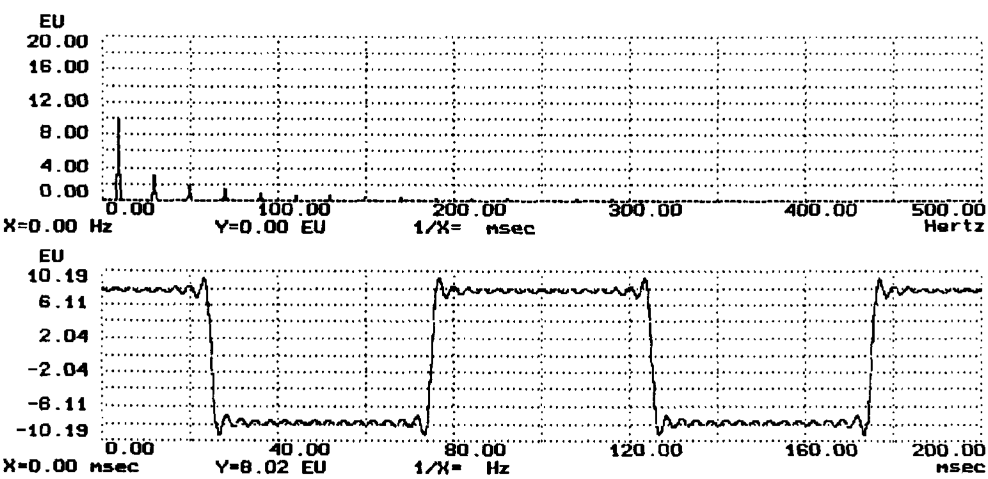


Fig. 2-23. 1800 RPM Motor Belt Driving a Fan.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
10.00	10.000	0.0	110.00	10.000	0.0	210.00	10.000	0.0
30.00	10.000	0.0	130.00	10.000	0.0	230.00	10.000	0.0
50.00	10.000	0.0	150.00	10.000	0.0	250.00	10.000	0.0
70.00	10.000	0.0	170.00	10.000	0.0	270.00	10.000	0.0
90.00	10.000	0.0	190.00	10.000	0.0	290.00	10.000	0.0

Fig. 2-24. Single Frequency with Only Odd Harmonics.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
10.00	10.000	0.0	110.00	0.909	180.0	210.00	0.476	0.0
30.00	3.333	180.0	130.00	0.769	0.0	230.00	0.435	180.0
50.00	2.000	0.0	150.00	0.667	180.0	250.00	0.400	0.0
70.00	1.428	180.0	170.00	0.588	0.0	270.00	0.370	180.0
90.00	1.111	0.0	190.00	0.526	180.0	290.00	0.345	0.0

Fig. 2-25. Single Frequency with Only Odd Harmonics.

For example:

Harmonic	Amplitude
1	10.0/1=10.000
3	10.0/3=3.333
5	10.0/5=2.000
⋮	⋮

Note that the signal is not exactly square, but has ripples. This is due to the fact that a limited number of odd harmonics are contained in the signal. A true square wave contains all odd harmonics, which cannot be truly simulated on a computer as a sum of the cosine functions.

It may appear impossible for a piece of equipment to generate a square wave, but it is possible to generate a signal with square wave features. This can occur in a motor if the motor has a loose mount. If the mounting bolts are loose, the motor will tend to move up and down. If the motor moves up and is stopped by the mounting bolt, and then moves down and is stopped by the motor support, a square wave can be generated. If clipping occurs on both the top and bottom of a signal and the clipping is significant, the result will resemble a square wave.

NATURAL FREQUENCIES

A special case occurs when natural frequencies are excited. A discrete frequency is not present in such cases because the natural or excited frequency is a band of frequencies. In this situation, a band of spectral lines occurs. The bandwidth is determined by damping, and the difference frequency between the spectral lines is the source of excitation. A high degree of measurement accuracy is required to determine if a given frequency is an excited frequency (natural) or a generated frequency (events times speed).

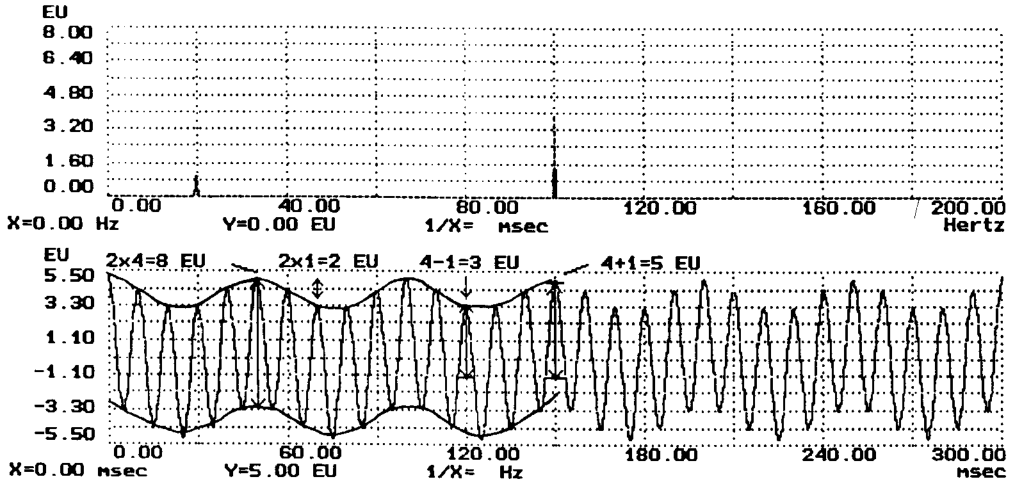
Amplitude modulation can occur between two generated frequencies, or an excited frequency can be modulated by the source of excitation, i.e., a generated frequency. When a generated frequency equals an excited frequency, other tests and measurements must be employed to identify the excited and generated frequencies. These techniques will be discussed in following chapters.

MULTIPLE FREQUENCIES - LINEAR SYSTEMS

High Frequency Riding a Low Frequency

When two independent frequencies are present in a linear system, they cannot add together in amplitude or frequency. When this occurs, the two frequencies mix, and the high frequency will ride the low frequency, as in Fig. 2-26. At first glance, amplitude modulation appears to be present. However, close inspection indicates that the two envelopes enclosing the peaks of the time signal are in phase. This means when the positive peaks of the signal move upward, the negative peaks of the signal move upward. The time period for one cycle of the envelopes is 50 milliseconds or 20 Hz:

$$F = \frac{1}{T} = \frac{1}{50\ msec} = 20\ Hz$$



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
20.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
100.00	4.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-26. High Frequency Riding a Low Frequency.

which is the low frequency. Five cycles of the signal are contained in each cycle of the envelopes, and the time period for each cycle of the signal is 10 milliseconds or 100 Hz:

$$F = \frac{1}{T} = \frac{1}{10\ msec} = 100\ Hz$$

which is the high frequency. The peak-to-peak amplitude of each cycle is two times the zero-to-peak amplitude of the 100 Hz frequency: 2 x 4.0 EU = 8.0 EU. The peak-to-peak amplitude of each envelope is two times the zero-to-peak amplitude of the 20 Hz frequency: 2 x 1.0 EU = 2.0 EU. The zero-to-peak value of the highest envelope peak is the amplitude of the 100 Hz frequency plus the amplitude of the 20 Hz frequency: 4 EU + 1 EU = 5 EU. The lowest point of the envelope is the amplitude of the 100 Hz frequency minus the amplitude of the 20 Hz frequency: 4.0 EU - 1.0 EU = 3.0 EU. Since harmonics are not present and the high frequency is riding the low frequency, there is not a cause-and-effect relationship. The two signals are generated independently. It is important to note that a high frequency which is an exact multiple of a low frequency will cause the amplitude of the high frequency peaks to be the same in each period of the low frequency. A high frequency that is not a multiple will cause the amplitude of the high frequency peaks to vary during each period of the low frequency.

MULTIPLE FREQUENCIES - NONLINEAR SYSTEMS

Two or more independent frequencies can be mixed together if a machine has some form of nonlinearity or other problem. There are many forms and degrees of frequency mixing. Examples of amplitude modulation, sum and difference frequencies, pulses, and frequency modulation are discussed in the following sections.

Amplitude Modulation

Amplitude modulation occurs when two frequencies are added together algebraically. Frequencies will not add in a machine that behaves in a linear manner. Therefore, a problem must exist before amplitude modulation can occur. There are several forms of amplitude modulation; one form is a beat. A beat occurs when the amplitudes of two frequencies are added together. Assume there are two frequencies of 29.6 Hz and 25.6 Hz, as in Fig. 2-27.

When the amplitudes of the two frequencies go into phase, they add together. Then, as the two frequencies go out of phase, the amplitudes subtract until they are 180 degrees out of phase. The two frequencies continue to go into and out of phase, forming a time-varying amplitude signal called a beat.

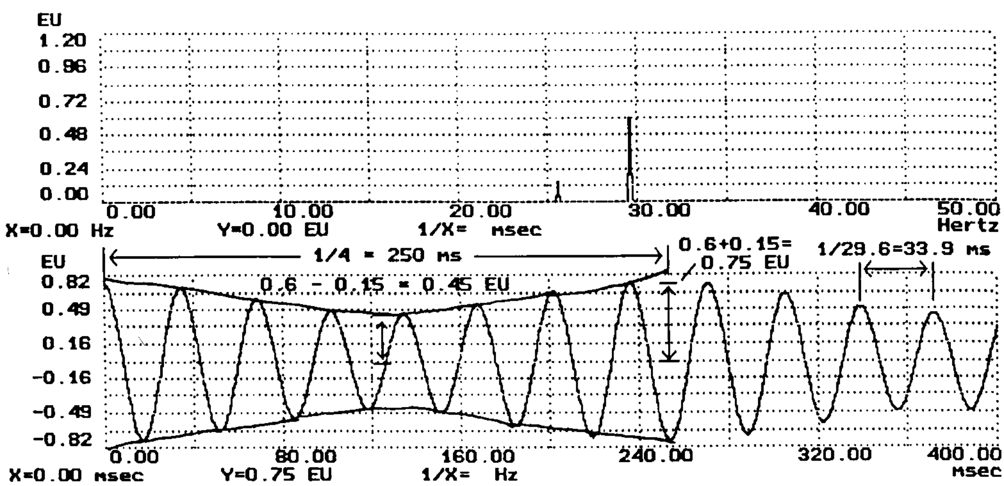
There are similarities between a high frequency riding a low frequency and amplitude modulation. The maximum zero-to-peak envelope amplitude is equal to the amplitude sum of $F_1 + F_2$:

$$0.6\ EU + 0.15\ EU = 0.75\ EU = \textit{zero-to-peak envelope maximum}$$

The amplitude difference of $F_1 - F_2$ is the minimum zero-to-peak envelope amplitude:

$$0.6\ EU - 0.15\ EU = 0.45\ EU = \textit{zero-to-peak envelope minimum}$$

The time period for each cycle in the time signal is 33.9 milliseconds, which is the 29.6 Hz frequency.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
25.60	0.150	0.0	0.00	0.000	0.0	0.00	0.000	0.0
29.60	0.600	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-27. Beat-Amplitude Modulation of Two Frequencies.

$$T = \frac{1}{29.6\ Hz} = 0.0339\ \text{sec}$$

The envelope frequency, however, is not the 25.6 Hz frequency, as it was for a high frequency riding a low frequency. For amplitude modulation, the envelope frequency is the difference between the two frequencies: $F_1 - F_2$ = 29.6 Hz - 25.6 Hz = 4 Hz. Fig. 2-27 contains a complete cycle of the difference frequency of 4.0 Hz. The envelope occurs at a time period of 250 milliseconds:

$$T = \frac{1}{4.0\ Hz} = 0.25\ sec = 250\ ms$$

Amplitude modulation is the result of a cause-and-effect situation and the amount of modulation can be expressed as a percentage. The frequency spectrum identifies the percent modulation as the amplitude of the lower frequency divided by the amplitude of the higher frequency:

$$\frac{0.15\ EU}{0.60\ EU} = 0.25\ or\ 25\%$$

The percent modulation in the time signal is expressed as the maximum envelope amplitude (M_x),

$$where\ M_x = F_1 + F_2 = 0.6 + 0.15 = 0.75$$

minus the minimum envelope amplitude (M_i),

$$where\ M_i = F_1 - F_2 = 0.6 - 0.15 = 0.45$$

divided by the maximum envelope amplitude plus the minimum envelope amplitude:

$$\frac{M_x - M_i}{M_x + M_i} = \frac{0.75 - 0.45}{0.75 + 0.45} = \frac{0.3}{1.2} = 0.25 - 25\%$$

Some comparisons between a high frequency riding a low frequency and amplitude modulation are:

1. The envelopes at the top and bottom of the signal are in phase for a high frequency riding a low frequency. The envelopes are out of phase for amplitude modulation.

2. For a high frequency riding a low frequency, the zero-to-peak amplitude of the envelope is equal to the amplitude of the low frequency. For amplitude modulation, the zero-to-peak amplitude of the envelope is also equal to the amplitude of the low frequency. The zero-to-peak amplitude of the highest peak is equal to $F_1 + F_2$, and the zero-to-peak amplitude of the lowest peak is equal to $F_1 - F_2$ for both a high frequency riding a low frequency and amplitude modulation.

3. In rotating machinery, a high frequency riding a low frequency there is

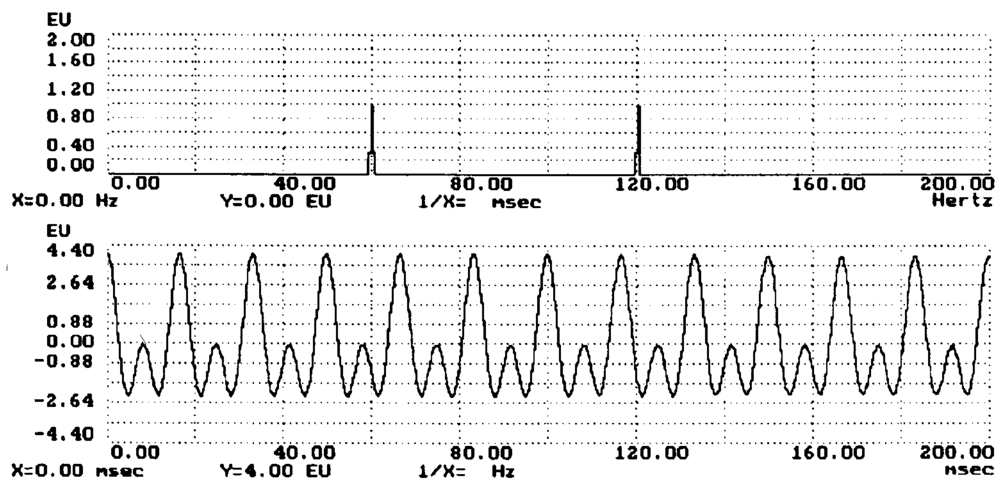
no cause-and-effect relationship. There are two independent frequencies and two independent problems. In amplitude modulation, there is a cause-and-effect relationship because there are two or more frequencies and only one problem.

4. When a high frequency rides a low frequency, the reciprocal of the envelope time period is equal to the low frequency. In amplitude modulation, the reciprocal of the envelope time period equals the difference between frequency 1 and frequency 2 (F_1 - F_2).

When two frequencies beat together, such as in Fig. 2-27, there is a cause-and-effect relationship. Therefore, two frequencies are present, but only one problem exists. The 29.6 Hz frequency is the carrier frequency and is the effect of the problem, i.e., the motor is loose. The 25.6 Hz is the modulation frequency or modulator and is the speed of the fan driven by the motor. The only way the fan can shake the motor, in this case, is when the motor is loose. Looseness in the motor allows the motor to shake, and the two frequencies to beat.

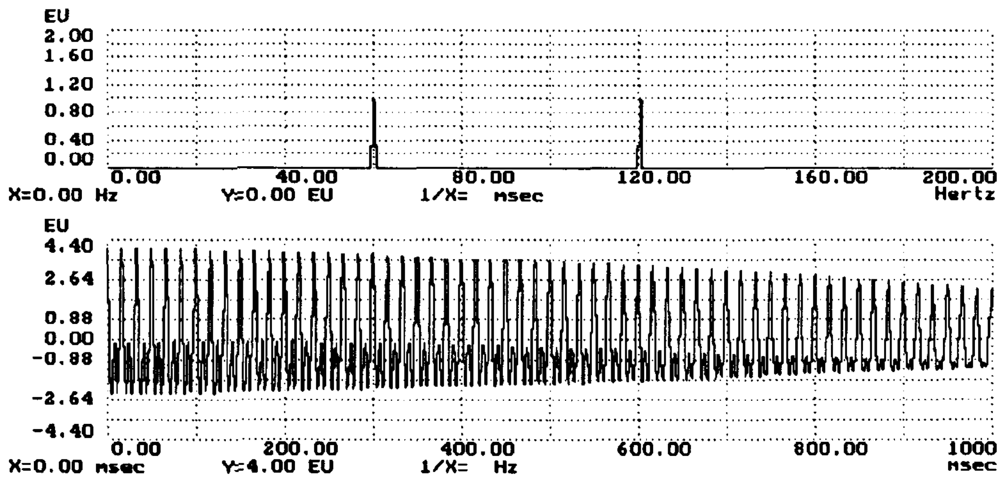
Amplitude modulation also occurs when two frequencies are not exact multiples. Fig. 2-28 contains two frequencies with second harmonics. The two frequencies are almost the same, but not exact. The signal seems to be constant.

However, Fig. 2-29 contains the same signal with a longer time period, and it is clear the signal is changing. Fig. 2-30 contains a full beat and amplitude modulation is evident. The difference between frequency 1 and frequency 2 is 0.2 Hz. Therefore, one beat takes 5 seconds to complete. As the frequencies get closer together, the time to complete one beat becomes longer. Therefore, it is extremely important to determine whether the frequency is a harmonic or is just close to the harmonic frequency. The beat in Fig. 2-30



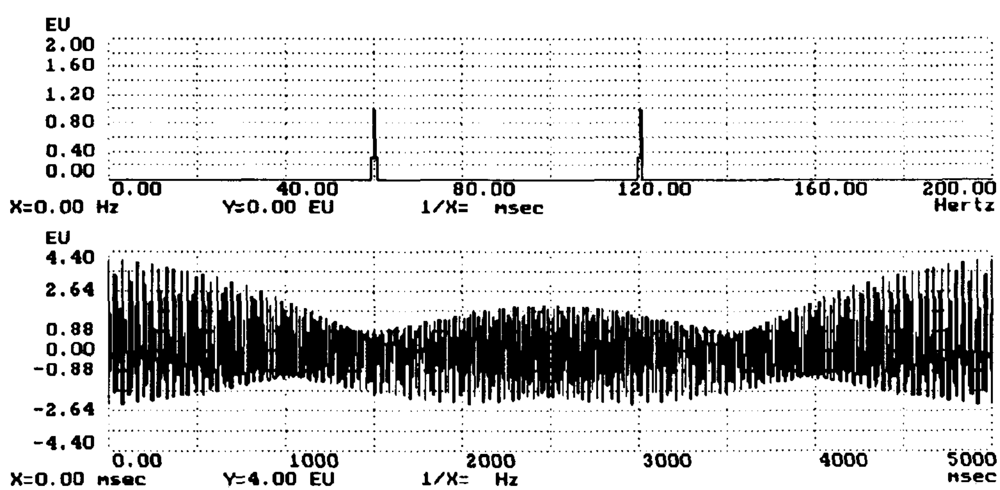
Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
60.20	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.40	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-28. Two Similar Frequencies with Second Harmonic.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
60.20	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.40	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-29. Two Similar Frequencies with Second Harmonic.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
60.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.00	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
60.20	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
120.40	1.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-30. Beat of Two Similar Frequencies.

is a common signal from a 3600 RPM motor that has an electrical problem. The motor speed is 60 Hz minus the slip frequency, when under a load. Line frequency is 60 Hz. In this case, the motor speed and line frequency must be examined with enough resolution to identify the true problem.

SUM AND DIFFERENCE FREQUENCIES

Another type of amplitude modulation occurs when one component is eccentric. A frequency of 100 Hz with a sum and difference frequency of 4 Hz is presented for analysis. The 100 Hz carrier frequency is contained in Fig. 2-31. One example of sum and difference frequencies is gear eccentricity. When one gear is eccentric or out-of-round, the amplitude of gearmesh frequency increases when the high place or places go into mesh. If the gear has only one high place, the signal amplitude will be higher once each revolution. In either case, amplitude modulation is caused by the eccentric gear. The associated spectra contain a spectral line at gearmesh frequency with sidebands of gear speed. If the gear has more than one high place, then the difference frequency between the gearmesh frequency and the sidebands is equal to the number of high places times the speed of the problem gear. If two high places are present, the difference frequency is two times gear speed. Three high places would generate a difference frequency of three times gear speed, four high places would generate four times gear speed, etc.

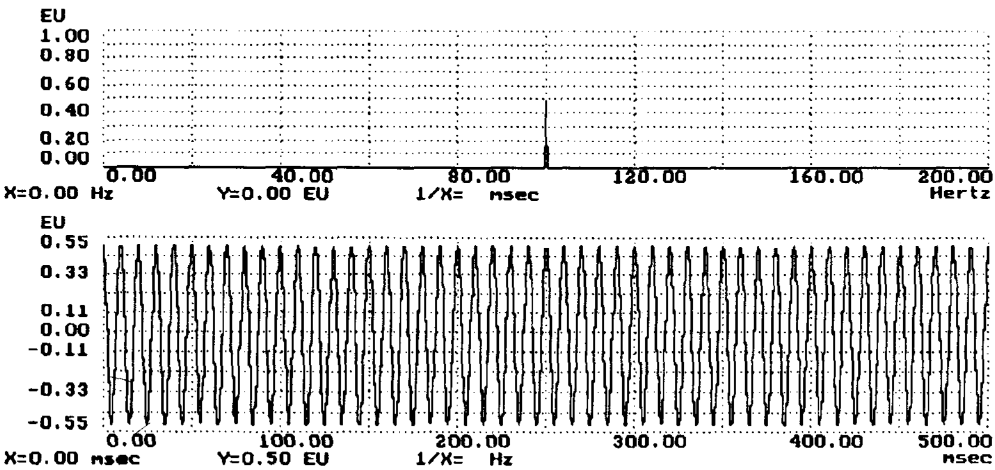
Assume the 100 Hz frequency is gearmesh frequency and gear speed is 1 Hz, if the gear has 100 teeth. If the gear has four high places or eccentricities, four times gear speed is 4 Hz. Fig. 2-32 shows the 100 Hz frequency with the sum frequency of 104 Hz: 100 Hz + 4 Hz = 104 Hz.

This signal has amplitude modulation. The carrier frequency is the gearmesh frequency of 100 Hz. The modulator or cause of the problem is the 4 Hz frequency of four times gear speed. The envelope frequency is 4 Hz and is noted in Fig. 2-32 as 250 milliseconds. One hundred cycles of gearmesh frequency occur each revolution of the gear. The modulation occurs at 25 cycles of gearmesh frequency or 4 times per revolution of the gear. The modulation is 10 percent. Fig. 2-33 contains the time period for one revolution of the gear. Four modulations can be observed:

$$T = \frac{1}{1\ Hz} = 1\ \text{sec} = 1{,}000\ msec$$

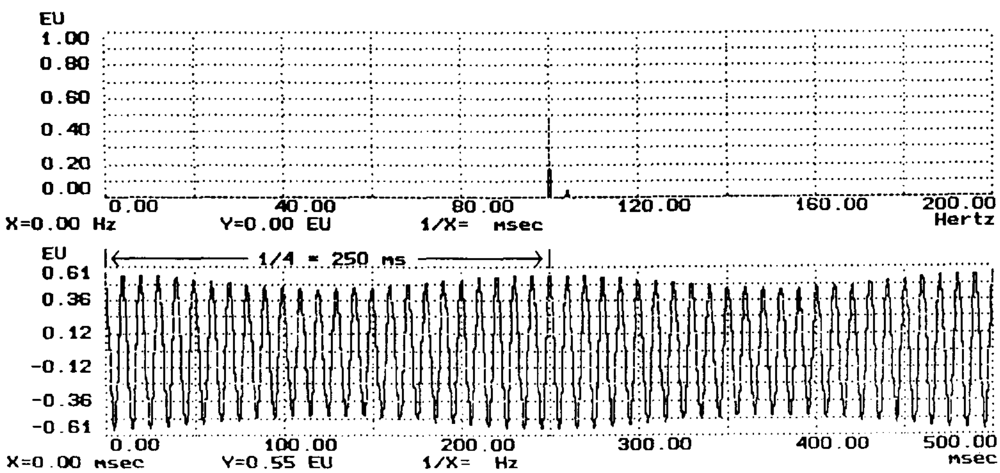
If the eccentric gear has not caused looseness, sidebands will occur at gearmesh frequency plus gear speed or multiples of gear speed. In other words, the sidebands will be on the high side of gearmesh frequency. The frequencies add, in this case, because the phase relationship between the carrier and the modulator is constant. As stated earlier, the machine is behaving in a linear manner.

Fig. 2-34 contains the 4 Hz as a difference frequency of 96.0 Hz: 100 Hz - 4 Hz = 96.0 Hz. The time signal appears similar to Fig. 2-32. When a gear or geared shaft system is loose, the looseness causes the modulator to subtract from the carrier because the two frequencies are out of phase. When the two frequencies are in phase, they add. Looseness causes an out-of-phase condition. Eccentricity is an in-phase condition.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
100.00	0.500	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-31. 100 Hertz Carrier Frequency.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
100.00	0.500	0.0	0.00	0.000	0.0	0.00	0.000	0.0
104.00	0.050	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0
0.00	0.000	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-32. 100 Hertz with 104 Hertz Sum Frequency.

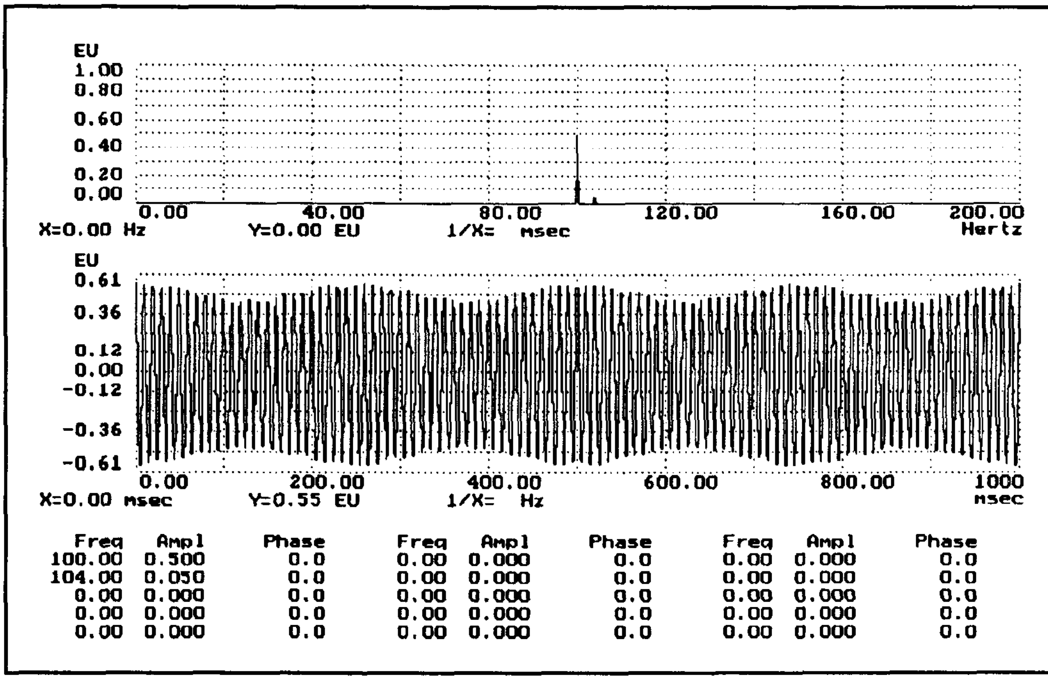


Fig. 2-33. One Revolution of Gear with Four Eccentricities.

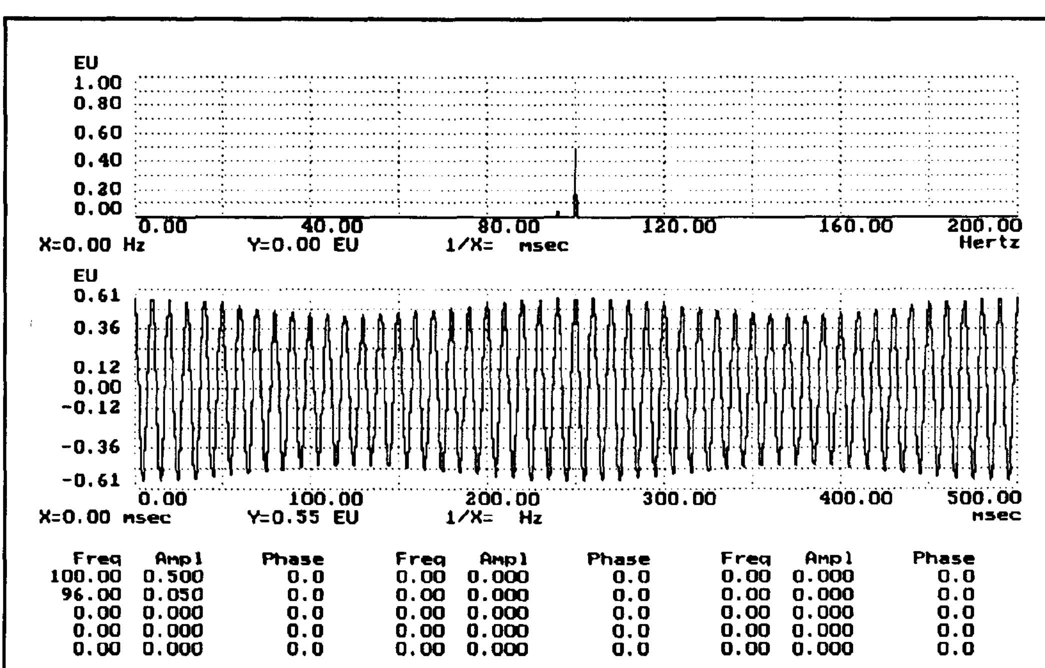


Fig. 2-34. 100 Hertz with 96 Hertz Difference Frequency.

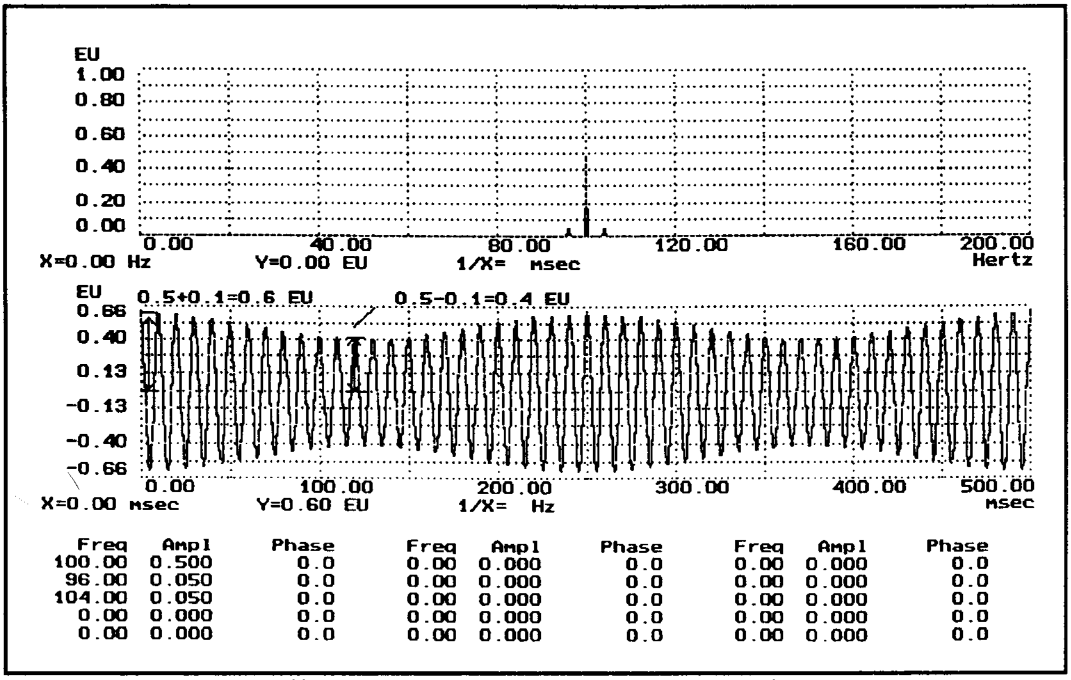


Fig. 2-35. Sum and Difference Frequency with No Phase Shift.

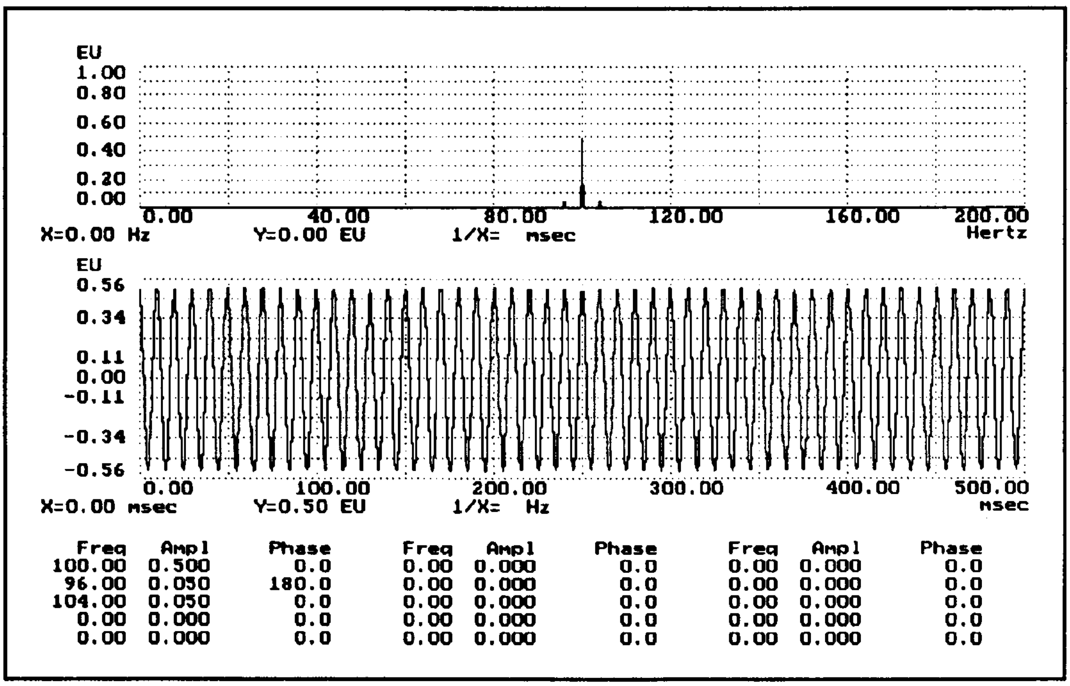


Fig. 2-36. Sum and Difference Frequency with Phase Shift.

Fig. 2-35 contains both the sum and difference frequencies. The beat is now more pronounced and the modulation is 20 percent:

$$\frac{0.6 - 0.4}{0.6 + 0.4} = \frac{0.2}{1.0} = 20\%.$$

This occurs when both the 96.0 Hz and 104.0 Hz sidebands are in phase. Fig. 2-36 contains both the sum and difference frequencies, but the 96.0 Hz difference frequency is 180 degrees out of phase. In this case, the modulation is zero percent. Modulation on the high side, or a sum frequency, indicates a linear problem. Low side modulation, or a difference frequency, indicates a nonlinear problem. Therefore, because of the nonlinearity, the phase of the difference frequency will be changing. However, the sum frequency will maintain its phase relationship.

If gear eccentricity has caused looseness (nonlinearity) associated with the problem gear, sidebands can occur on both sides of gearmesh frequency. If looseness is the more severe problem, the amplitudes of the sidebands will be higher on the low side of gearmesh frequency. If eccentricity is the more severe problem, the amplitudes of the sidebands will be higher on the high side of gearmesh frequency. If looseness is the only problem, then the sidebands occur only on the low side of gearmesh frequency.

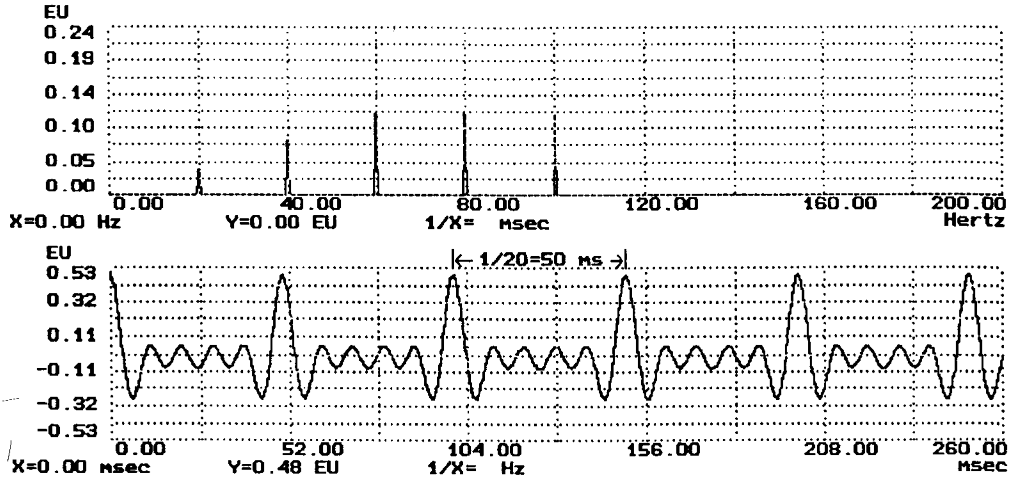
For example, suppose a gear was mounted on a shaft rotating at 29.6 Hz, gearmesh frequency was 1280 Hz, and the bearings were loose in the housing. Gearmesh frequency would be present and one sideband could occur at 1160 Hz: 1280 Hz - (4 x 29.6 Hz) = 1160 Hz. Four times running speed is an indication of slippage, so 4 x 29.6 Hz is generated. The frequencies subtract when looseness is present because the phase relationship between the signals is not constant, which means the machine is acting in a nonlinear manner.

This technology is not new. It has been used for years to identify moving targets with radar, and for space object identification to determine the shape as spherical, cylindrical, triangular, bell-shaped, etc. It has also been used to identify the way an object is moving, i.e., amount of spin, tumble, pitch and yaw. However, these principles had not been used for rotating machinery until the author applied the technology. The principles described for gearmesh frequency apply to other generated frequencies such as blade or vane pass frequencies, bearing frequencies, frequencies from multiple defects, and frequencies from bars or corrugations on press rolls.

PULSES

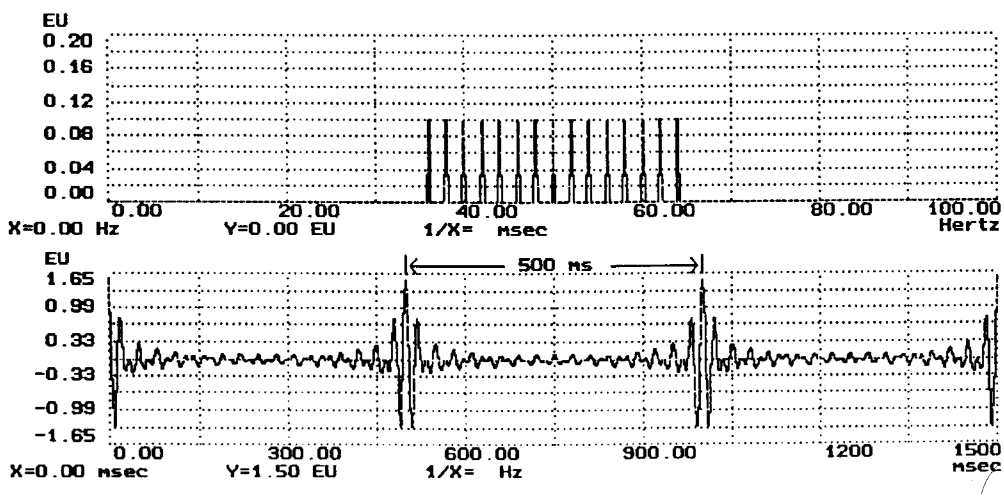
A pulse is caused by a hit or an impact. Pulses fall into one of three categories: empty pulses, frequency modulation, and amplitude modulation. Fig. 2-37 contains an empty pulse. This occurs when excited or generated frequencies are not present. It is called empty because it contains no generated or excited frequencies. A pulse is identified by a series of spectral lines. The repetition rate of the pulse is equal to the difference frequency between the spectral lines. The empty pulse has a low level spectral line at shaft speed, and the amplitude increases with each succeeding harmonic.

Frequency modulation is a time-varying frequency. This frequency modulation can appear as a series of bursts or beats, as contained in Figs. 2-38, 2-39, and 2-40. Generated



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
20.00	0.040	0.0	0.00	0.000	0.0	0.00	0.000	0.0
40.00	0.080	0.0	0.00	0.000	0.0	0.00	0.000	0.0
60.00	0.120	0.0	0.00	0.000	0.0	0.00	0.000	0.0
80.00	0.120	0.0	0.00	0.000	0.0	0.00	0.000	0.0
100.00	0.120	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-37. Empty Pulse.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
36.00	0.100	0.0	38.00	0.100	0.0	40.00	0.100	0.0
42.00	0.100	0.0	44.00	0.100	0.0	46.00	0.100	0.0
48.00	0.100	0.0	50.00	0.100	0.0	52.00	0.100	0.0
54.00	0.100	0.0	56.00	0.100	0.0	58.00	0.100	0.0
60.00	0.100	0.0	62.00	0.100	0.0	64.00	0.100	0.0

Fig. 2-38. Generated or Excited Pulse.

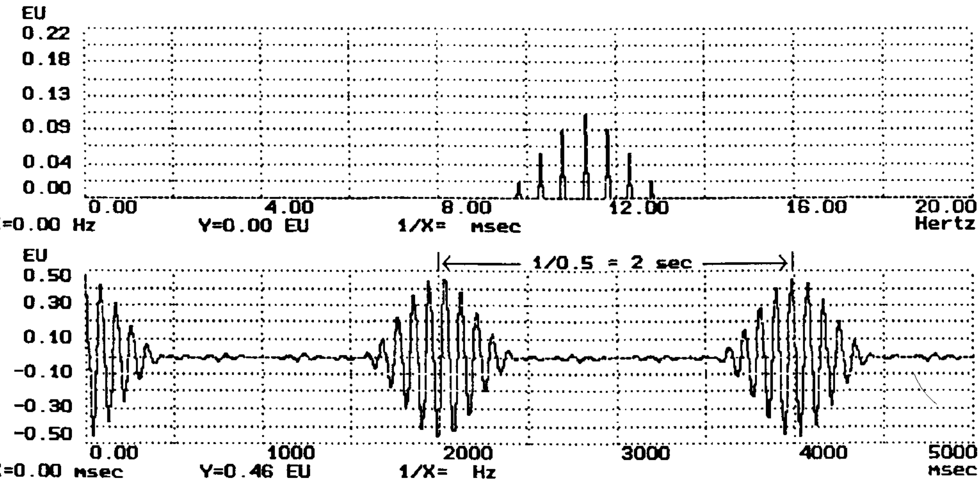
or excited pulses are usually caused by a once-per-revolution impact or excitation. Fig. 2-38 contains a series of spectral lines. The pulse repetition rate is equal to the difference frequency between the spectral lines, or 500 milliseconds. This would be similar to an impact once per revolution. Fig. 2-39 is a representation of a felt problem. The pulse or bad place in the felt is exciting a natural frequency of the loading diaphragm. The pulse repetition rate is 0.5 Hz, or 2 seconds, which is once per revolution of the felt. The signals in these figures contain characteristics of both amplitude and frequency modulation. Again, it is impossible to determine the character of the time signal from the frequency spectra.

FREQUENCY MODULATION

Frequency modulation is a time-varying frequency, as opposed to amplitude modulation, which is a time-varying amplitude. The lower frequency is the carrier, and the higher frequency is the modulator. The modulator is normally an excited frequency, and the source of excitation is normally the speed of the rotating unit.

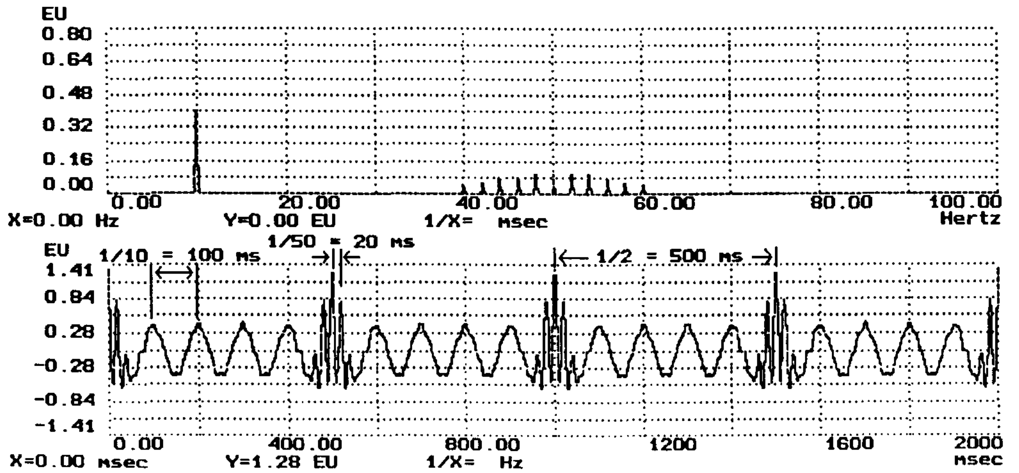
Fig. 2-40 is an example of frequency modulation. The low frequency carrier is 10 Hz, the excited high frequency is 50 Hz, and the source of excitation is the 2 Hz difference frequency. The high frequency bursts occur at 2 Hz intervals. This could occur in a pump when pump speed is 2 Hz, and the impeller has 5 vanes. The 10 Hz carrier frequency is vane pass frequency. The problem or source of excitation is one impact per revolution of the impeller. If one vane hit the housing, the 50 Hz frequency could be the natural frequency excited by the impact.

Frequency modulation can be a series of high frequency bursts similar to a pulse, or the high frequency can occur periodically with a low frequency. Since the frequency response



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
9.80	0.024	0.0	12.80	0.024	0.0	0.00	0.000	0.0
10.80	0.090	0.0	11.80	0.090	0.0	0.00	0.000	0.0
12.30	0.060	0.0	0.00	0.000	0.0	0.00	0.000	0.0
10.30	0.060	0.0	0.00	0.000	0.0	0.00	0.000	0.0
11.30	0.110	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-39. Pulse Exciting Natural Frequency.



Freq	Ampl	Phase	Freq	Ampl	Phase	Freq	Ampl	Phase
50.00	0.100	0.0	52.00	0.100	0.0	54.00	0.100	0.0
48.00	0.100	0.0	46.00	0.090	0.0	44.00	0.090	0.0
56.00	0.080	0.0	58.00	0.060	0.0	60.00	0.050	0.0
42.00	0.060	0.0	40.00	0.050	0.0	0.00	0.000	0.0
10.00	0.400	0.0	0.00	0.000	0.0	0.00	0.000	0.0

Fig. 2-40. Frequency Modulation.

of an accelerometer is best at high frequencies, such problems may be best measured in acceleration. Frequency modulation occurs most often in impacts, such as defects on the inner race of cylindrical roller bearings, or when two shafts are rotating very close to each other. Frequency modulation can occur in screw compressors, vacuum pumps, and blowers when one shaft is bent enough to permit an impact once each revolution.

One last comparison should be noted to clarify the differences between a high frequency riding a low frequency, amplitude modulation, and frequency modulation.

1. High frequency riding a low frequency - No looseness is present. High and low frequencies may be exact multiples of each other. No mixing of signals occurs. Changes in the phase have little or no effect.

2. Amplitude modulation - High frequency is the carrier; low frequency is the modulator. Signals go into and out of phase.

3. Frequency modulation - Low frequency is the carrier; high frequency is the modulator.

CONCLUSION

All machines obey the basic laws of physics. Therefore, vibration problems are repeatable and can be identified. Both the time signal and frequency spectra must be analyzed to accurately identify problems. This was proven both empirically and theoretically. The basics of time domain analysis were discussed and used to explain the frequency spectrum. Single frequencies and single frequencies with harmonics were explained to identify the effects of amplitude changes and phase relationships.

Using these basic principles, multiple frequencies were introduced to identify linear and nonlinear problems for more complex analysis. Multiple frequencies include high frequencies riding low frequencies, amplitude modulation, beats, pulses, and frequency modulation. It was also noted that if there is a cause-and-effect relationship in the signal, a problem exists in the equipment.

The fundamental concepts in this chapter are the foundation for understanding and explaining the proceeding chapters. Using these fundamental concepts, all time and frequency relationships can be understood. However, with the many combinations of phase shifts and amplitude changes, each case needs to be analyzed to identify all the information contained in the vibration signal. Analysis can be complex and time consuming, but the use of **SAP** simplifies the analysis. The result is a complete and more accurate diagnosis of machinery vibration problems.

CHAPTER THREE: HARDWARE AND SOFTWARE REQUIRED FOR ACCURATE DIAGNOSTICS

HARDWARE

INTRODUCTION

The improvements in hardware, software, and technology for accurate diagnosis of machinery problems over the past twenty years is unparalleled in history. In 1973, when this author first became involved in diagnosing rotating machinery problems, some people were still using a piece of sharpened soapstone for balancing rotating equipment. Hand-held vibration meters were widely used to measure the vibration amplitude. These amplitude readings were recorded and plotted on graph paper to indicate trends. A variety of balancing instruments were also available. These instruments employed hand-tunable filters. Some frequencies could be identified, however, the accuracy was not very good because the filter "Q" was of low quality. Software was nonexistent, and hand-held calculators were being introduced at a cost of several hundred dollars. The technology for diagnostics was in initial stages of development. Some of the more sophisticated engineers were using an oscilloscope to view the time domain signal. Not very many problems were identified and equipment often failed.

In the next few years several companies introduced real-time analyzers (RTA) that could produce a frequency domain spectra from the time domain signal using a Fast Fourier Transform (FFT). This was a giant leap forward, however, some problems still existed. For example, these units had only 256 lines of resolution. A frequency translator or zoom was not available, and the time signal displayed was analog and could not be synchronized. Technical papers on how to identify defects in antifriction bearings and other problems were being published. On-line condition monitoring systems using proximity probes were being introduced.

Around 1980, several manufacturers introduced new real-time analyzers. These new instruments had 400 lines of resolution, a frequency translator/zoom, and a digitized time signal. This improved equipment provided the capability for accurate diagnostics. However, the technology for accurate diagnostics was just emerging. These new analyzers with the improved frequency spectra caused some people to stop using the time domain signal. In retrospect this was a mistake, because now we know the time signal is required for diagnostics.

The personal computer (PC) was making its debut and the RTA could be controlled with a serial port. In 1980 the author introduced the first expert system. This system used a SD-345 real-time analyzer, an HP-85 desktop calculator, a Nagra IV scientific tape recorder, and software that could diagnose defects in antifriction bearings. The system was very accurate, however it was very slow. The HP-85 had 16k of memory and for $795 you could add another 16k memory module. These were limitations that required too many operator entries. This system required recording one data point for 1.5 minutes, and it took about 15 minutes to produce a fully annotated spectra. Today, we record four data points for 30 seconds and can produce almost three fully annotated spectra in one minute.

The 1980's were very dynamic in the predictive maintenance field. New and more powerful analyzers were introduced in rapid succession. Also, a lot of less powerful instruments such as shock pulse meters and data collectors were being introduced. The data collectors were the big success story of the 1980's. These instruments were used with IBM compatible computers and had sophisticated databases, with capabilities of up and down loading data, and trending with associated worker programs. These computerized predictive maintenance systems are widely used today even though the diagnostic ability does not match that of the real-time analyzers. The weak technology for diagnostics gave rise to several new concepts such as the Cepstrum, which is a spectrum of a spectrum. Spike energy, envelope detection, demodulators, and many new versions of the shock pulse meter. Each of these concepts/instruments provided varying degrees of success, however they did not provide the total solution. More powerful real-time analyzers were designed and built to perform specific tasks. If a new task or function was required, the RTA required modification. This caused some RTAs to be obsolete in a few short years.

The current state-of-the-art RTA is a multi-channel signal processor. This signal processor is controlled by a personal computer. The major advantage of this arrangement is that it uses the power of the PC to perform complex calculations, to manipulate data, and to store, recall, and display the data in a user friendly format. If additional tasks or functions are required, they can be provided with software changes or additions. This is more cost effective, and delays obsolescence. Today, we are seeing A to D converters for the new Personal Computer Memory Interface Adaptor (PCMCIA) slots on notebook computers. In the not too distant future, we may see an FFT signal processor on the PCMCIA cards.

Arguments and discussions of the advantages and disadvantages of the data collectors and real-time analyzers may continue for some time. It is also true the data collectors of today are very much improved from the ones originally introduced. With the trend toward smaller, faster, and more powerful, the data collectors may someday achieve the power of a real-time analyzer. However, it will still be difficult to get the engineer or technician to collect enough data at the equipment location for accurate diagnosis because of the hostile environments.

Fig. 3-1. Personal Computer.

Personal Computer

Fig. 3-1 contains a photo of the current state-of-the-art personal computer. The PC's of today are a marvel over those produced two or three years ago. Today, RAM of up to 16 megabytes is common. Hard disk drives with 500 megabytes are the standard. The capability to add stand-alone disk drives provides virtually unlimited disk storage. The use of cache systems and DSP chips with a clock speed of 66 MHz, processes data at a very high speed. The introduction of the new Pentium chip will make parallel processing and multi-tasking in the personal computer a reality.

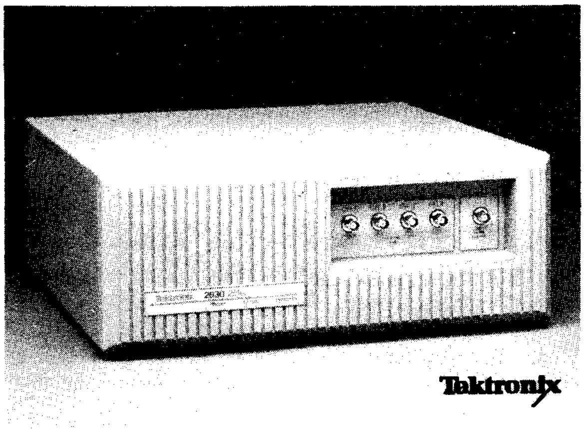


Fig. 3-2. Tektronix 2630.

Real-time Analyzer

The real-time analyzer has enjoyed significant improvements as well. The current state-of-the-art system employs an instrument with a multi-channel signal processor, and a signal generator output. Fig. 3-2 contains a photo of the Tektronix TEK 2630, 4-channel analyzer. These units are controlled by a PC, and the PC screen is used to display the data. The operating software is stored in the computer. If a new requirement is identified, new software can be developed to satisfy the requirement. This can often be accomplished without a hardware change. The following features are required for a real-time analyzer:

1. Anti-aliasing protection. This feature prevents false signals from being presented in the data.

2. A flat frequency response from DC to 20 KHz.

3. Operator selectable frequency range from 5 Hz to 20 KHz in a 1, 2, 5 sequence.

4. A zoom/frequency translator with a user-selectable center frequency and bandwidth.

5. Variable lines of resolution of up to 1600. The lines of resolution determine the measurement accuracy. Frequency range divided by lines of resolution equals measurement accuracy. $M_a = \frac{F_r}{L_r}$

6. A real-time rate of at least 10 KHz. This means the analyzer can perform an FFT on the signal in about the same time it takes to fill the memory period on the 10 KHz range.

7. Sampling rate. Physics requires the data to be sampled at 2.56 times the highest frequency. This sample rate prevents the loss of data, and a higher sampling rate does not improve the data.

8. A good time domain signal. The digitized time domain signal is required because of stability. As the machine zero motion point is zero on the time signal, phase relationship between the various signals can be observed. Truncation, peak amplitude of each cycle, amplitude modulation, frequency modulation, pulses, and beats are also observable in the time domain signal.

The use of the time domain signal requires some explanation. The amount of time displayed is determined by the frequency range used. The length of time displayed is the reciprocal of the measurement accuracy. For example, on the 200 Hz range using 400 lines of resolution, the measurement accuracy is $\frac{200}{400} = 0.5\ Hz$. The frequency measurement accuracy is 0.5 Hz. The reciprocal of 0.5 is $\frac{1}{0.5} = 2\ \mathrm{sec}$. This means on the 200 Hz frequency range, two seconds of time data would be displayed. This time period is also called the memory period. If you are using a 200 Hz frequency range, frequencies above 200 Hz will not be observed in time data displayed. The measurement accuracy in frequency is improved by using lower frequency range and/or using more lines of resolution. Measurement accuracy is improved in the time domain by using less time. Less time is obtained by using a higher frequency. Let's look at a couple of examples using 400 lines of resolution, which is a 1024 data point transform.

Using the above example, 2 sec = 2000 ms, and $\frac{2000}{1024} = 1.95\ ms$. This means that the data contains 1024 data points and that each of the data points are 1.95 ms apart. Time could then only be measured accurately within 1.95 ms. This may not carry much meaning for some readers. The following explanation may be helpful. Suppose we wanted to measure a frequency in the time signal of 29.6 Hz. The time period for one cycle is $\frac{1}{29.6} = 0.0337838\ \mathrm{sec}$ Then $0.0337838\ \mathrm{sec} = 33.78378\ ms$

and $\frac{33.78378\ ms}{1.95\ ms} = 17.32$ This means that 33.78378 ms cannot be measured because measurement accuracy is based on data points, and 33.78378 ms does not fit evenly between any two data points. We cannot measure fractions of data points, so we must measure either 17 or 18 data points. Therefore 17 x 1.95 = 33.15 ms, and 18 x 1.95 = 35.1 ms are the theoretical measurements.

Then $\frac{1}{0.03315\ \mathrm{sec}} = 30.16\ Hz$ and $\frac{1}{0.0351\ \mathrm{sec}} = 28.49\ Hz$
Therefore 29.6 Hz would be measured as 30.16 Hz in the time signal.

To measure 29.6 Hz in the frequency spectrum the process would be similar. We will use the same 200 Hz frequency range and 400 lines of resolution, $\frac{200\ Hz}{400\ lines} = 0.5\ Hz$ This is the distance between each data point in the frequency spectrum. $\frac{29.6\ Hz}{0.5\ Hz} = 59.2$ This means that 29.6 Hz cannot be measured exactly. We would have to measure using 59 or 60 data points. Each data point is 0.5 Hz so that 59 data points would give you 59 x 0.5 = 29.5 Hz, and 60 data points would give you 60 x 0.5 Hz = 30 Hz. In this case the 29.6 Hz signal would be measured as 29.5 Hz. In the above example, the 29.6 Hz frequency is measured more accurately in the frequency domain.

Now suppose we have the same frequency, 29.6 Hz, on a 2000 Hz frequency range. Then the measurement accuracy of the frequency spectrum would be $\frac{2000\ Hz}{400\ lines} = 5\ Hz$. Since $\frac{29.6\ Hz}{5\ Hz} = 5.92\ Hz$, the closest number of data points is 6. This means that 29.6 Hz could be theoretically measured as 6 x 5 Hz = 30 Hz on a 2000 Hz range in the frequency spectrum. In the time signal, the measurement accuracy would be

$$\frac{1}{5\ Hz} = 0.2\ \text{sec} = 200\ ms \qquad \frac{200\ ms}{1024} = 0.195\ ms$$

Now each data point in the time signal is only 0.195 ms apart. To measure 29.6 Hz:

$$\frac{1}{29.6\ Hz} = 0.03378378\ \text{sec} = 33.78378\ ms \qquad \frac{33.78378\ ms}{0.195\ ms} = 172.97$$

The closest number of data points is 173. This means that 29.6 Hz or 33.78378 ms could be theoretically measured as $173 \times 0.195\ ms = 33.735\ ms$ *or* $\frac{1}{33.735\ ms} = 29.6\ Hz$ on a 2000 Hz range in the time signal.

In this case the 29.6 Hz could be measured more accurately in the time signal. The frequency range was increased from 200 Hz to 2000 Hz, which improved the measurement accuracy in the time domain by using less time. By contrast, the measurement accuracy in the frequency spectrum decreased from 0.5 Hz to 5 Hz with the increased frequency range. This supports the earlier statement that measurement accuracy is improved by using a lower frequency range in the frequency domain and a higher frequency range in the time domain.

The screen on the personal computer, data collector, or real-time analyzer also affects measurement accuracy. The data points on the screen or display are called pixels. The industry standard for the horizontal axis is 600 pixels. Regardless of the lines of resolution in frequency, or the data points in time, the display will plot 600 points across the screen and connect each point with a line. Increasing the lines of resolution above 600 has little effect on measurement accuracy unless an expansion or zoom is used.

9. The three types of averaging required are: arithmetic mean averaging, peak hold, and synchronous time averaging.

The arithmetic mean average is equal to the summation of the x's over N. $M_a = \frac{\Sigma x}{N}$. This is the type of averaging used most, however, it is not often understood. This type of averaging takes the average power contained in a frequency during one memory period for each average. If eight (8) averages are taken, then the average power in each frequency in each memory period is added together eight times and the sum is divided by eight. Since averaging enhances the signal, sixteen (16) averages may be required to ensure all frequencies are obtained. However the enhancement obtained by the last eight averages is small, and most people agree that eight averages is enough for most applications.

If the signal contains a frequency during the complete memory period, and each memory period contains the same frequency, then the result of the averaging is satisfactory. However, if the memory period contains the frequency for a short time period, pulses for example, then the energy can be understated by a factor of five or more. Some examples are pulses generated by a broken tooth on a gear, some inner race defects, and defects on rollers in antifriction bearings. In these cases, the amplitude must be obtained from the time signal and/or peak hold averaging.

Peak hold averaging takes the peak amplitude for each frequency in the memory period and writes it to the screen. Then, each time the amplitude of a frequency is higher, the screen is updated with the higher amplitude. This type of averaging is used as indicated above, for coast down data, when determining speed variations of machines, and bump tests.

Synchronous time averaging employs a trigger from some source to start filling the memory period at exactly the same point each time. Often some event or pulse in the time signal can be used as a trigger. This type of averaging employs a comparator method and all frequencies that are in phase with the previous memory period are retained. All frequencies that are out of phase with the previous memory period are canceled according to the phase difference. Since different frequencies go into and out of phase with each other, the frequencies could be in phase part of each memory period. However, the frequencies are out of phase most of the time and will eventually average out to a point of little or no concern. This is why the synchronous to nonsynchronous data is improved by the square root of N. Therefore $SNR \propto \sqrt{N}$, when N equals the number of averages.

DATA COLLECTION

Accurate diagnosis of rotating machinery problems requires three or more frequency spectra and one or more time domain signals. The route/database of some data collectors allows only one frequency range per point. The slow speed (500 Hz real-time rate) of some data collectors and unpleasant work environment of some machines make it difficult to get a human to collect enough data for accurate diagnosis.

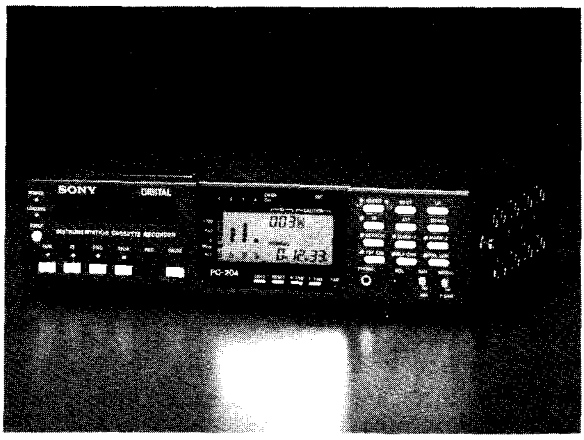


Fig. 3-3. Sony Digital Audio Tape Recorder.

The best way to collect the data is to collect an uninterrupted time signal at the equipment location and then process and analyze the data in a better environment. The amount of data required depends upon machine speed and the type of processing involved. For most machines about 25 or 30 seconds of data is enough. On slower machines, and when synchronous time averaging is required, several minutes of the time signal and triggers may be required.

Currently, the best way to collect this data is with a tape recorder. The best recorder is the Sony PC-204. Fig. 3-3 contains a photo of this scientific quality, digital audio tape recorder (DAT). It can record four (4) data points in 25-30 seconds or longer if needed. If a 120 minute tape is used, and the recording time is 30 seconds, then 120 min x 60 s/m = 7200 sec ÷ 30 = 240 data points. These data points are recorded per channel, or 240 x 4 = 960 data points can be recorded on one 120 minute tape, if data is recorded on all four channels. If a 180-minute tape is used, the numbers would be 360 data points per channel, and 4 x 360 = 1440 points per tape.

Administrative tasks for route information, equipment identification, and data identification numbers are simplified by controlling the DAT with a pen-based, hand-held computer. These portable, battery-powered instruments can collect more data faster than other equipment on the market today. Since the hand-held computer is a full-functioning personal computer (PC), the operator can go off route, change routes, or create a new route at any time. The 10 or 20 megabyte plug-in disk installed in the PCMCIA slot provides ample storage for all routes and data points in most plants or mills. Fig. 3-4 contains a photo of a hand-held computer. These instruments with control software make a fast and efficient data collection system. The hand-held computer has a minimum of two megabytes of RAM, which is adequate for most programs. This revolutionary data collection system allows the operator to record horizontal and axial data on the drive and driven unit in about 30 seconds. This is enough data on most motors and pumps, etc.

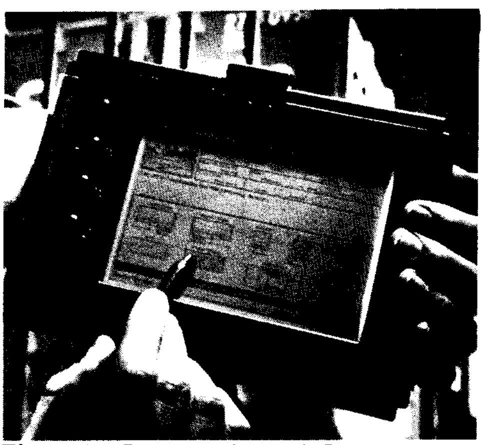


Fig. 3-4. GRID PalmPad Computer.

The processing and analyzing of the volume of data required for accurate diagnostics is a big workload if it were done manually. However, the host PC can now control the DAT, real-time analyzer, and the printer. This automated process is still time consuming, however the human interface is reduced to making hardware connections, starting the software, and changing the tape when required. The computer time for processing is offset by a factor of more than 10 to 1 by the results of accurate diagnosis of machinery problems.

The technology contained in this book has made possible the development of Expert software modules to diagnose problems from the data. This truly Expert software makes specific diagnosis and assigns a priority of 1, 2, or 3. Such as "Defect on the outer race priority 3" or "Defect on the inner race priority 2", etc. Currently other expert systems do not make specific diagnosis. Instead they provide a list of possible problems with a percentage listed by each problem. The workload of the analyst using good expert software is reduced to reviewing output products for accuracy.

Printers

The various types of laser jet and ink jet printers have the best quality and are widely used for many operations. When these printers are used for graphics, the quality is still good, however, it takes a long time to down load the graphics from the computer to the printer.

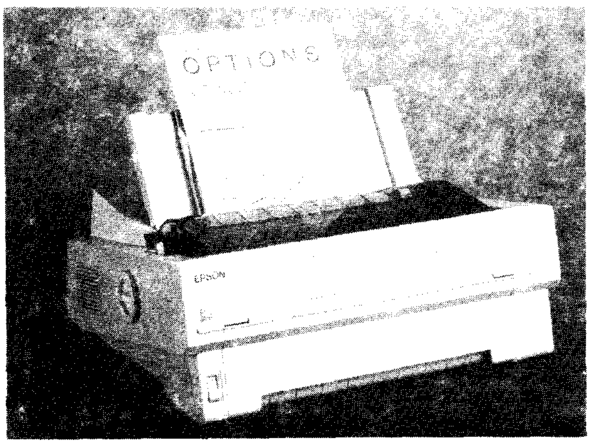


Fig. 3-5. Dot-matrix Printer.

The dot-matrix printer is the best choice for large volumes of printed graphics such as frequency spectra and time signals. Fig. 3-5 contains a photo of a dot-matrix printer. These printers are fairly fast and can print a fully annotated spectra in about 20 seconds. The quality of these prints is quite good.

Transducers

The electro-mechanical device used to convert mechanical motion into an electronic signal is called a transducer. These transducers can be divided into five categories. Each category contains one or more transducers for the various required measurements. The five categories are:

1. Displacement transducers measure how far something is moving, normally in thousandths of one inch or mils. The displacement transducer is unique because it measures relative motion, i.e. how much one component is moving relative to another.

2. Velocity transducers measure how fast a component is moving in inches per second (IPS) of velocity. The measurements are made in tenths of an inch per second. Most measurements are less than one (1) inch per second.

3. The accelerometer measures the rate of change per time period.

The units of measurement are in g's of acceleration. Low frequencies normally have low g levels, or a fraction of one g. Higher frequencies can have levels of several g's.

4. Pressure transducers measure pressure fluctuations. The selection of pressure transducers is based on static pressure, amplitude of the fluctuations, and the frequency of the fluctuations.

5. Microphones measure audible signals in the audio range from 20 Hz to 20 KHz. The major problem with microphones is they are not very discriminating. They pick up all audible signals in the audio range. However, the fact remains that the only way to pick up an audible signal is with a microphone. There are many applications for the microphone and it should be used more often. For example, the only way to measure sound in a room is to use the microphone.

NOTE: Please refer to the frequency response curves in Fig. 1-27 when selecting the correct transducer for the job.

Displacement Transducers

The two basic types are noncontacting and contacting. Both types require firm mounting. A bolted installation is required for permanent installation. However, a heavy duty magnetic base and a Starrett flex-o-post are satisfactory for most portable measurements. The frequency response varies by type and manufacturer, therefore, you must review the manufacturers' specifications to obtain frequency response, transducer sensitivity, etc. Generally speaking, the displacement transducers should be used to measure most low frequencies, below 10 Hz, and all relative motion measurements.

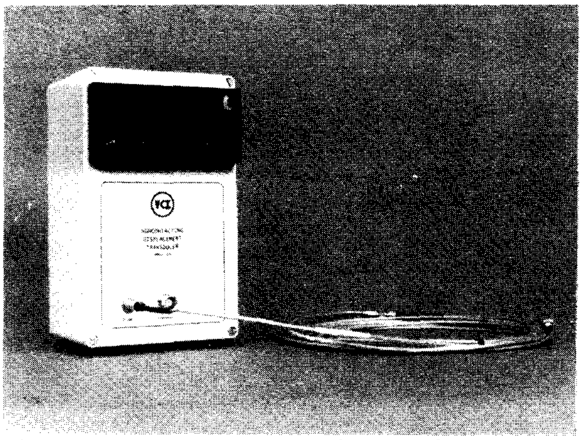


Fig. 3-6. Noncontacting Displacement Transducer.

Noncontacting displacement transducers require a power supply, a modulator/demodulator, normally called a driver, and a volt meter. Fig. 3-6 contains a photo of a portable noncontacting displacement transducer. The transducer has a small wound wire coil at the end. When the tip of the transducer is placed near the target, about .040", the relative motion between the component supporting the transducer and the target can be measured. The driver generates a high frequency. When the frequency passes through the coil, a magnetic field is generated. The motion between the mount and target intersects the magnetic field. This movement is measured when the driver demodulates the signal. If the target is moving slowly, the depth of the finish or other metal disfigurations can be measured. However, at turbine speeds those items are not measured because the frequency response at such high frequencies is not very good. This means the relative motion between a turbine housing and shaft across the coupling can be measured, even when the shaft has teeth marks left from turning the shaft with a pipe wrench. The total runout or the amount of bend in the shaft can also be measured. If a keyway is under the transducer, a pulse will be generated. The data on the shaft can still be used if enough attenuation is used to prevent saturation by the pulse during data acquisition. After the data is collected, enough attenuation must be removed in order to measure the shaft runout. Saturation of the pulse does not affect the data after the data is collected. This measurement may not be as accurate, because of the amplitude scale, as when a keyway is not present. However, the data is still useful. Please note these measurements must be made in the time domain, and shaft speeds should be about 6000 RPM or below.

The noncontacting displacement transducer has been used for years in high-speed turbo machinery. These permanently installed transducers with the associated vibration instruments are used for on-line condition monitoring. These systems may alert by turning on a light or ringing a bell. If the vibration becomes serious enough, the instrumentation may shut the machine down. When the proximity transducer is used in high speed, all specifications concerning shaft runout, etc. are much more critical. For example, 0.5 mils on a machine that operates at 5,000 RPM is about 0.13 IPS; however, 0.5 mils at 10,000 RPM is about 0.23 IPS, and 0.5 mils at 20,000 RPM is about 0.5 IPS. The specifications for transducer installation in turbo machinery are found in the API specification 670. In slow speed applications, 0.5 mils is of little consequence.

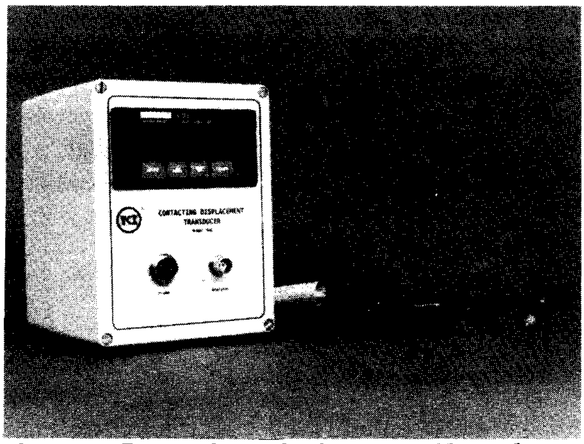


Fig. 3-7. Contacting Displacement Transducer.

Figure 3-7 contains a photo of the contacting displacement transducer. This transducer must make contact with the target. The contacting portion is a machined rounded surface. The tip can be removed and has the same thread size as the tips on a dial indicator. This permits replacement of the tip with a variety of "shoes" or "rollers." This transducer uses the Linear variable differential transformer (LVDT) principle. This transducer came out of the gauging industry, however, it is very effective in measuring low frequencies from DC to about 200 Hertz. The measurement range and sensitivity varies. However, if the range is ± 200 mils, the normal sensitivity is 25 mV/Mil. NOTE: THE USER SHOULD ALWAYS REFER TO THE MANUFACTURERS' SPECIFICATIONS FOR THE EQUIPMENT BEING USED.

Dial indicators have been used for many years and will continue to be used for a long time. The usefulness of this tool is widely recognized. The development of the contacting and noncontacting displacement transducers establishes the dawning of a new era. The new era is the electronic dial indicator. If engineers, mechanics, millwrights, and technicians would start using the electronic dial indicator while the machine is in operation, to the same extent dial indicators are used on non-operating machines, untold millions of dollars could be saved in lost production and wasted manhours and materials.

Fig. 3-8. Velocity Transducer.

Velocity Transducers

These transducers are voltage generators and do not require an external power source. Fig. 3-8 contains a photo of a velocity transducer. Most velocity transducers employ a permanent magnet mounted on a stud. A coil of wire surrounds the magnet and the coil is supported by leaf springs. When mounted on a machine, the motion causes the wire coil to move through or intersect the magnetic field. You may recall from basic physics that any time a conductor is passed through a magnetic field, a voltage is produced. The coil is wound to produce an output proportional to the movement -- normally four or five hundred millivolts per inch per second. Since calibration is affected by wire size and the number of turns in the coil, calibration is determined during manufacturing, and recalibration is not required. The rugged construction and very low failure rate make this transducer the best choice for data collection with a portable instrument or a permanent installation for on-line systems. Before a final decision is made for any transducer, consideration must be given to temperature tolerance and frequency response. Once again, the manufacturers' specifications must be consulted. Generally speaking, the frequency response rolls off sharply below 10 Hz. This does not mean frequencies below 10 Hz cannot be measured. Quite the contrary, frequencies down to about 2 Hz can be measured, however, the amplitude is understated for frequencies below 10 Hz. The upper frequency range is to about 2 KHz. However, this author has measured frequencies well above 2,000 Hz with a velocity transducer. Accurate measurements below 10 Hz should be made with a displacement transducer. Accurate measurements above 2,000 Hz should be made with an accelerometer.

Accelerometers

Three different accelerometers are required in most plants. Fig. 3-9 contains a photo of an accelerometer. A low frequency accelerometer is required to measure the low frequencies that cannot be measured with a displacement transducer, for example, low frequency vibrations in buildings, the earth, and some radar antennas. The frequency response of these accelerometers is from DC to about 500 Hz. Since low frequencies generate low g amplitudes, the sensitivity is often 500 mV/g or more. A medium range accelerometer is also required for various measurements. The frequency response is often from about 2 Hz to about 10 KHz. However, the sensitivity may vary 15 or 20% on the high end. The sensitivity of these accelerometers is often 100 mV/g. High frequency accelerometers are also required in most plants. These accelerometers can measure frequencies up to 100 KHz, however the amplitude accuracy may be ± 20%. The transducer sensitivity is often 25 mV/g or lower because high frequencies generate high g levels. The mounting of the high frequency accelerometer is quite critical. The transducer should be screwed down, the mounting surface should be machined, and a couplant should be used between the transducer and machined surface.

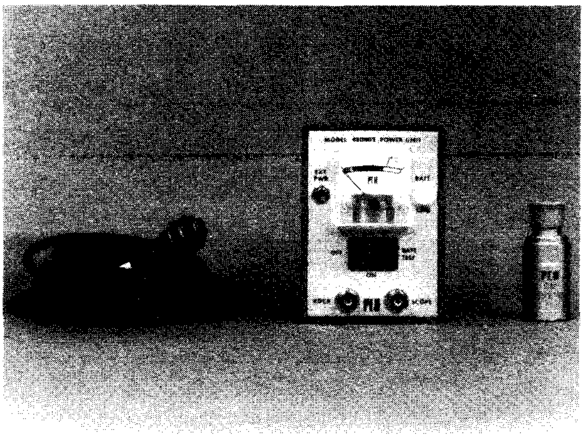


Fig. 3-9. Accelerometer Kit.

Most accelerometers are based on the piezoelectric crystal principle. A piezoelectric crystal responds to a mechanical motion by generating an electric voltage. The voltage amplitude is quite low, in the micro (μ) volt area (1×10^{-6}). The piezoelectric crystal responds to an electrical voltage with a mechanical vibration. These crystals are used in this manner also for delay lines in some radar applications. In most applications, the frequency response of a crystal is determined by size. The larger the crystal, the lower the frequency. The smaller the crystal, the higher the frequency. Since the output of a crystal is so low, the signal must be amplified in, or very close to, the accelerometer. If this amplification is not accomplished, the signal may get lost in the electronic noise.

When the accelerometer is used, it should be hard mounted, i.e. either glued or screwed down. A large segment of industry is collecting data with hand-held accelerometers, or accelerometers with a magnetic base. Such practice should be discouraged because some frequencies can be over or understated by as much as 12 dB, (about 4 times the actual). Also, if you use the zoom feature, the resulting frequency window may contain noise rather than discrete frequencies. The reason data collected with a velocity transducer using a magnetic base are acceptable, is because the velocity transducer is more forgiving. For example, a slight amount of movement caused by the mounting of the velocity transducer may not be observable with the unaided eye. This is true because first, the sensitivity is high, about 400 mV/IPS; second, because of the roll off of the frequency response at the low end. Also, looseness is a low frequency phenomena. By contrast, all motion of the accelerometer is amplified.

Pressure Transducers

Pressure transducers measure pressure fluctuations. Fig. 3-10 contains a photo of a pressure transducer. They are similar to accelerometers, except the output is calibrated in pounds per square inch (psi). This is why the amplitude and frequency of the fluctuations must be considered before selecting a pressure transducer. These transducers measure pressure fluctuations of gases and liquids. Such measurements are often required for accurate diagnosis of problems in pumps and compressors. These transducers must be mounted in a pipe or fitting, near the discharge of the unit, where the pressure fluctuations can be felt by the transducer. In other words, the gases or liquid must contact the end of the transducer.

Fig. 3-10. Pressure Transducers.

Microphones

A microphone is required to measure audible sound, or the audio range from 20 Hz to 20 KHz. Fig. 3-11 contains a photo of a microphone. The microphone is the only instrument that can measure sound. The units of measurement are dB above 1 mV. A wide range of microphones is available. The prices vary anywhere from less than $100 to several thousand dollars. The discussion of sound measurement here applies only to determining the frequency of sound for noise abatement and equipment diagnostics. The measurement of sound to satisfy OSHA requirements is beyond the scope of this book.

The measurement of sound can be accomplished with a microphone. The microphone will pick up all sound in the audio range. When this data is processed on the RTA a frequency spectra is produced.

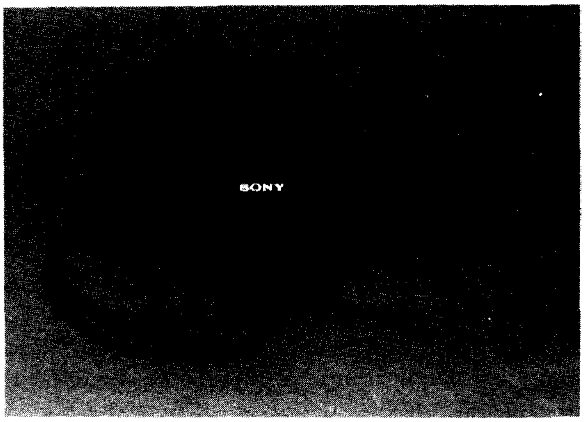


Fig. 3-11. Microphone.

Once-Per-Revolution Markers

When balancing rotating machinery and performing synchronous time averaging and other tasks, a trigger or a once-per-revolution marker is required. There is no single best instrument for this task. Often, more than one instrument is required, depending on the variety of circumstances encountered. The two basic types of instruments that generate a trigger are magnetic and optical. There is a wide variety in each type. The discussion of each type is beyond the scope of this book.

If the once-per-revolution marker is engineered for a pickup and installed on a rotor, then a wide variety of instruments can be used. If you have to trigger from a shaft or component, your best option may be the noncontacting displacement transducer installed over the keyway. **Warning: Before attempts to install are made, the shaft should be viewed with a strobe light to ascertain that a key is not present.**

If the trigger must be obtained from color contrasts on slow speed equipment, the fiber optic pickup may be the best choice. See Fig. 3-12. A good example is the trade line on a felt. If the green light fiber optic is used, a brown mark on a tan background can be identified.

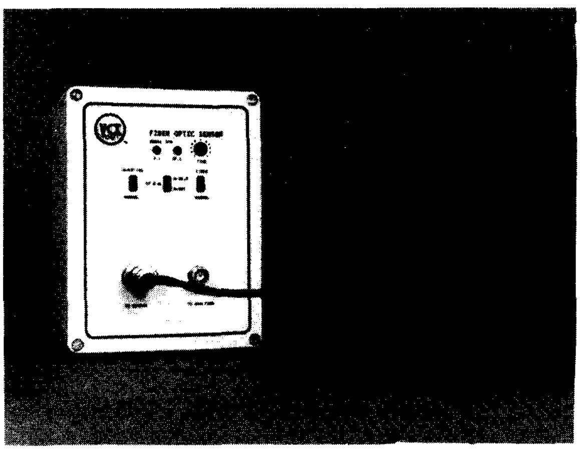


Fig. 3-12. Once-per-revolution Marker.

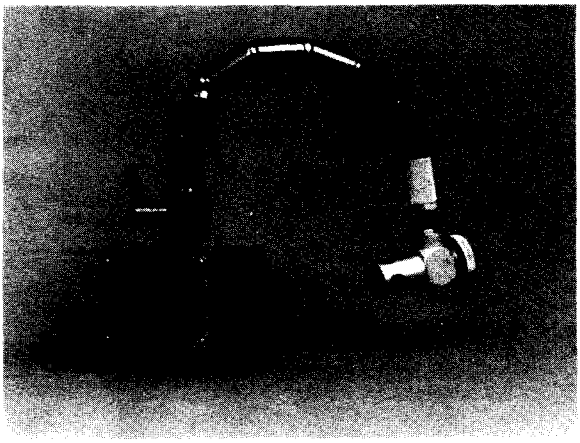


Fig. 3-13. Flex-o-post and Heavy Duty Magnetic Base.

A laser tachometer is also quite useful and has a broad application. However, it may not be useful in every situation. Also as with other once-per-revolution markers, mounting sometimes poses a challenge.

The Starrett heavy duty magnetic base and flex-o-post is one of the best mounts for portable units. See Fig. 3-13. When using this type of mounting, the unit should be tied off so the unit will not fall into the machine if it falls or is knocked off. The best mounting method is to drill and tap a hole, then screw the flex-o-post onto the machine. When the flex-o-post is removed, a pipe plug could be used to fill the hole.

Multiplexer

Multiplexers, sometimes called switching units, are available in two basic types. The first type is a master or control unit that controls other multiplexers. The second type is the slave unit that takes instruction from the master unit. Either unit can have manual control options and/or computer control. Some examples of computer control include 16-bit parallel, IEEE 488, RS-232, etc. Depending on the configuration, these units can control over 8000 data points.

A typical installation would have installed transducers similar to Figure 3-14. The cable is a shielded, twisted pair and depending upon the environment the cable should withstand high temperature. Coaxial cables are normally required for high frequency applications. The other end of the cable is connected to a slave unit. The slave unit has many inputs as in Figure 3-15 and normally 3 or 4 outputs. A status feedback should be employed to identify broken wires, failed transducers, and inoperative circuits.

The computer tells the master unit to access a particular slave unit, to obtain data from the desired point. After data is collected, the next point is selected. This routine continues until data from all points have been taken, processed, and analyzed. The entire process starts over again. Figure 3-16 contains a photo of a typical master unit.

On-line systems using installed transducers and computer-controlled multiplexers with expert software provide identification of defects soon after they occur. The safety aspect

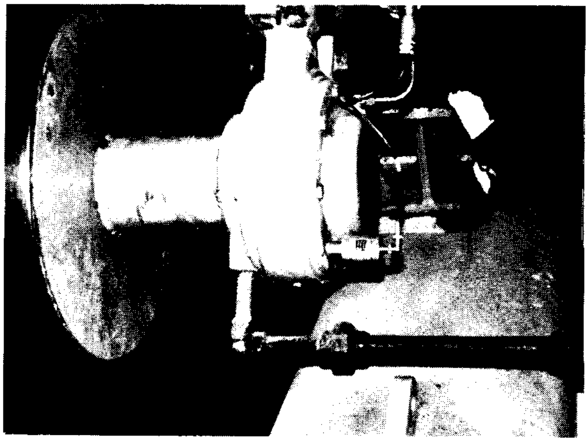


Fig. 3-14. Permanently Installed Transducers.

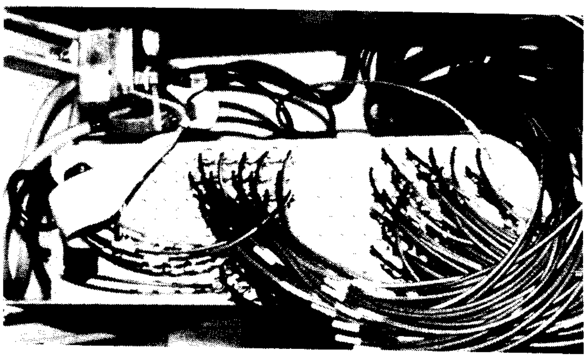


Fig. 3-15. Rear View of a Slave Unit.

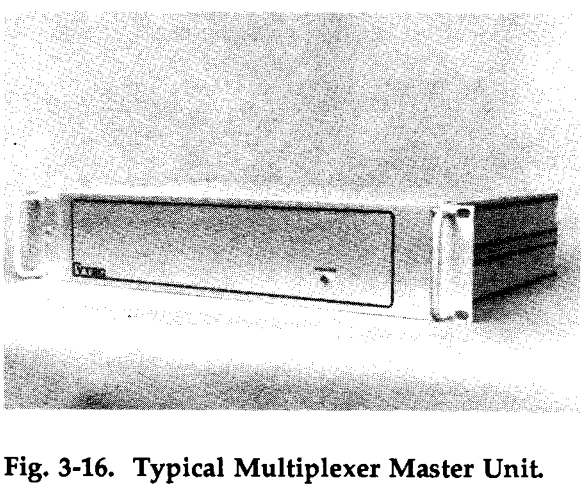


Fig. 3-16. Typical Multiplexer Master Unit.

is very important because personnel spend less time in hostile environments. Production is increased and the return on investment is often 600 percent in the first year. This author is convinced that on-line systems will be the preferred method in the near future.

Gauss Meter

A small gauss meter should be used to measure the magnetism in bearings, shafts, or other items that may contain small amounts of magnetism. See Fig. 3-17. This gauss meter measures ± 20 gauss. The plus and minus indicate the north and south poles of the magnet. The strength of magnetism measured is a function of the strength of magnetism in the piece and the distance between the meter and the piece. Magnetism can be measured through paper, wood, etc. Therefore, the magnetism in a bearing can be measured before the bearing is unpackaged. If a bearing contains magnetism on receipt, it should be rejected. Magnetism all by itself may not be harmful. However, if the magnetism is strong enough to attract tiny metal particles, the particles could cause a premature failure. If a bearing has too much magnetism from the manufacturer, it should be rejected. Magnetism caused by induction heaters may not cause a problem unless the bearing does not have seals or shields. Bearings used in circulating oil systems should not contain magnetism or magnetized spots.

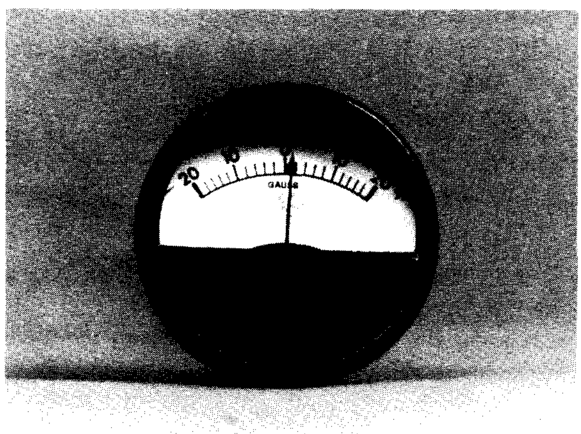


Fig. 3-17. Gauss Meter.

SOFTWARE

INTRODUCTION

The development of the personal computer created the demand for new software. The new electronic technology made possible the development of digital equipment for vibration measurements. Most plants were measuring vibration amplitude with hand-held meters, and plotting the amplitude trends manually. During this period, the predictive maintenance industry was divided into two groups. One group designed and developed data collectors and software programs for displaying, storing, retrieving, and trending data. The data collectors and software programs used with the personal computer satisfied the current demand. This new computerized predictive maintenance programs were and still are a huge success. These hardware and software systems permit the collection, display, and trending of huge volumes of data. The original systems had limited diagnostic capabilities. However, some improvements have been made and these improvements continue.

The efforts of the other group were toward developing new technology for diagnosing problems. This book is one result of that effort. This group also believed time domain signals should be recorded for processing in a controlled environment. The supporting facts are that three or more frequency spectra/zooms and one or more time signals are required for accurate diagnosis. Since all signals may not appear in every memory period, several memory periods may be viewed before one captures the desired signal. Also, a person may not be able to stay at the equipment location long enough to process and analyze the data because of temperature and safety. The collection of four measurement points in 30 seconds greatly simplifies and speeds up the data collection process. It is the best and fastest way to collect data. This task is made possible with a DAT recorder and computer control during data collection and processing. The analysis of this volume of data is simplified with Expert software that performs specific diagnosis and assigns priorities. A considerable amount of software is required to support this effort.

All software manufacturers claim their software is user friendly, easy-to-use, etc. Software should meet two tests in addition to being easy to use, fast, etc. The first test is one of utility. The utility test defines whether the job done by the software really needs to be done. For example, some software is developed to perform a function, regardless if the function should be performed. Such software is often in search of a problem to solve.

The second test is one of efficiency, proficiency, and accuracy. Efficiency has to do with conserving memory, task speed, and human interface. The working memory and storage memory should be allocated to improve efficiency and fast access. Task speed is a function of the hardware used and software design. This is accomplished by using state-of-the-art hardware, and reducing user entries by storing required data for program access. Also when a user entry requires changes of other parameters, the program should automatically change these other parameters. The human interface is concerned with the operator. The operator should not have to go off screen, except when entering or exiting the program. This is accomplished by pull-down menus. When the operator wants to change a setup, the menus are pulled down on the screen, the changes are made, and the menu is then put away until needed again. The function key, "F1," should provide the required on-screen help messages for each program function. The operator should not be required to make duplicate entries. For example, the operator should not be required to enter the I.D. number of the data to be stored, if the data contains the I.D. number. The best software is developed by a highly skilled engineer who knows the details of what should be done, and how the task should be done.

To achieve maximum results the required hardware and technology must be supported by software. This chapter presents some of the software available to perform advanced vibration analysis. This software system, called Machine Doctor (MACHDOC), contains various software packages. The first group of programs can be used as stand-alone programs to support the technology in this book. These programs are also integrated with the MACHDOC system by using the function keys, and on-screen display of results. This Diagnostic Toolbox software consists of Signal Analysis Program (SAP), Vibration

Calculator, Resonance and Deflection Calculator (RADC), Bearings Program, Gears Program, and Roll Ratio/Rusch Chart Program. These programs can be used on any personal computer that uses MS-DOS 5.0 or higher. These programs calculate and display frequencies and time signals to show the operator how the signals add, subtract, and mix. They also convert between engineering units, and calculate generated and excited frequencies and improper roll ratios. These programs aid the analyst in diagnosing the problem and determining the cause of failure.

Group 2 of the software is hardware dependent and can be used only with a personal computer that uses MS-DOS 5.0 or higher and the Tektronix 2630 two-channel or four-channel analyzer. This software is the MACHDOC Base system with the following options: Polar Plot, Time Plot, Balancing, Diagnostics data base, diagnostic modules, and the Roll Quality Assurance Program.

Group 3 of this software is hardware dependent and requires the hardware and software listed in Group 2. It contains the Digital Option using the hand-held computer and DAT recorder for data collection and processing.

The data collection routes are built into the host PC. These routes are then downloaded onto the disk of the hand-held computer. Except for major changes and additions, this is a one-time operation because the plug-in disk in the hand-held computer can hold all routes in most plants.

The hand-held computer displays the routes, controls the DAT, and keeps track of the equipment I.D. numbers and related tape I.D. numbers. After the data is collected each time, the equipment I.D. and associated tape I.D. are downloaded into the host PC. The DAT is connected to the host PC and the RTA. The control software then processes the data on user-selected frequency ranges and time periods. All processed data is stored and can be printed during processing or at a later time. If the diagnostic modules are installed, this expert software can diagnose problems, assign priorities, and print listings of the diagnosed problems.

Group 4 of the software is also hardware dependent and requires the hardware and software listed in Group 2, plus the real-time option using installed transducers connected to a multiplexer. The control software takes data from each installed transducer. After the data is processed, analyzed, stored, and the problem diagnosed, the next transducer is accessed and the procedure is repeated. The collected data is stored. The overall RMS value and the RMS value of each diagnosed problem can be plotted as a trend over a user-defined period in the form of a history plot. This system alerts and alarms from diagnosed problems, not preestablished amplitude levels.

These various hardware and software groups and options have been engineered and designed to satisfy the needs for accurate diagnosis of rotating machinery problems in small as well as large plants. They can be used to troubleshoot and diagnose on-machine problems, to collect and analyze data from routes, and to make on-line diagnostics with expert software. This system uses any transducer, and can be integrated with other on-line monitoring systems using currently installed transducers. The major economic advantage of this system is that the base system can be connected to the on-line system at night, on weekends, and on holidays for on-line diagnostics. Then during the day, the base system can be taken off line and used for test and diagnostics or used with the digital system for processing, analysis, and diagnostics of route data.

Group 1, Toolbox Software

The Signal Analysis Program (SAP)

SAP is a versatile program designed to investigate/teach/demonstrate simple signals, or sinewaves, harmonic content, the effects of a phase difference between the signals, the results of truncation, amplitude modulation, pulses, etc. Fig. 3-18 contains the display of a screen from the SAP program. The top screen contains the frequency spectrum, or can be changed to display the time signals before the signals are added together, as in the top screen of Fig. 3-19. The bottom screen display in Fig. 3-18 contains the composite time signal and the frequency amplitude and phase of two or more frequencies.

The Signal Analysis Program can be used as a tutorial for the user to learn how signals add, subtract, truncate, modulate, beat, or otherwise mix. The program can also be used as a diagnostic tool. For example, the frequencies and amplitude of the spectrum from a machine can be entered. Then the phase of the signals can be adjusted until the time signals from SAP is similar to the time signal from the machine. The phase of the signals can be obtained from the SAP screen. Remember, the frequency spectra are a result of averaging, and the time signal is one memory period. This explains why the result in the SAP time signal may be a little different from the equipment time signal.

Vibration Calculator Program

The vibration calculator program displays displacement, velocity, and acceleration as a function of speed for pure imbalance. The program also calculates and presents the various conversions between displacement, velocity, and acceleration. It also calculates transducer sensitivity, engineering units, and dB when any two items are entered. See

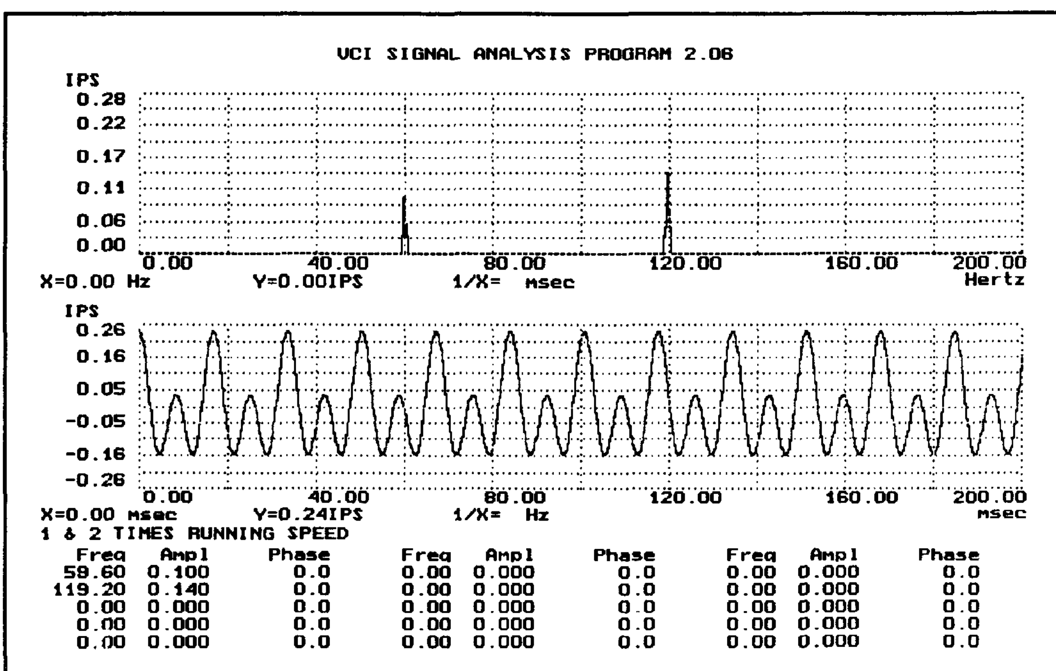


Fig. 3-18. Signal Analysis Program Display.

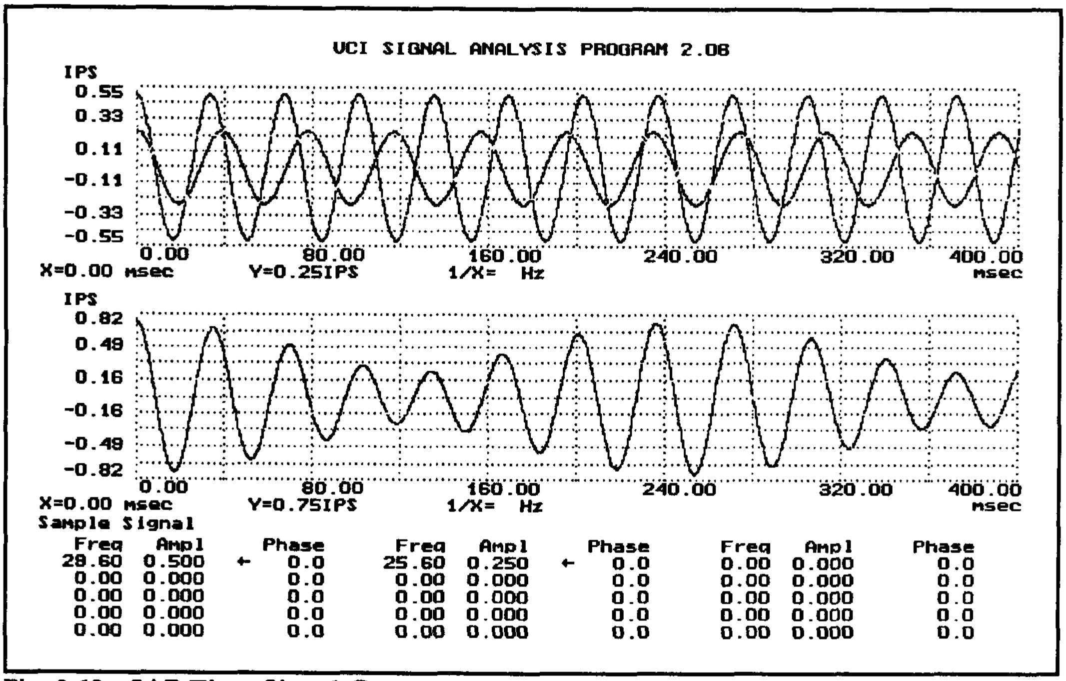


Fig. 3-19. SAP Time Signal Component.

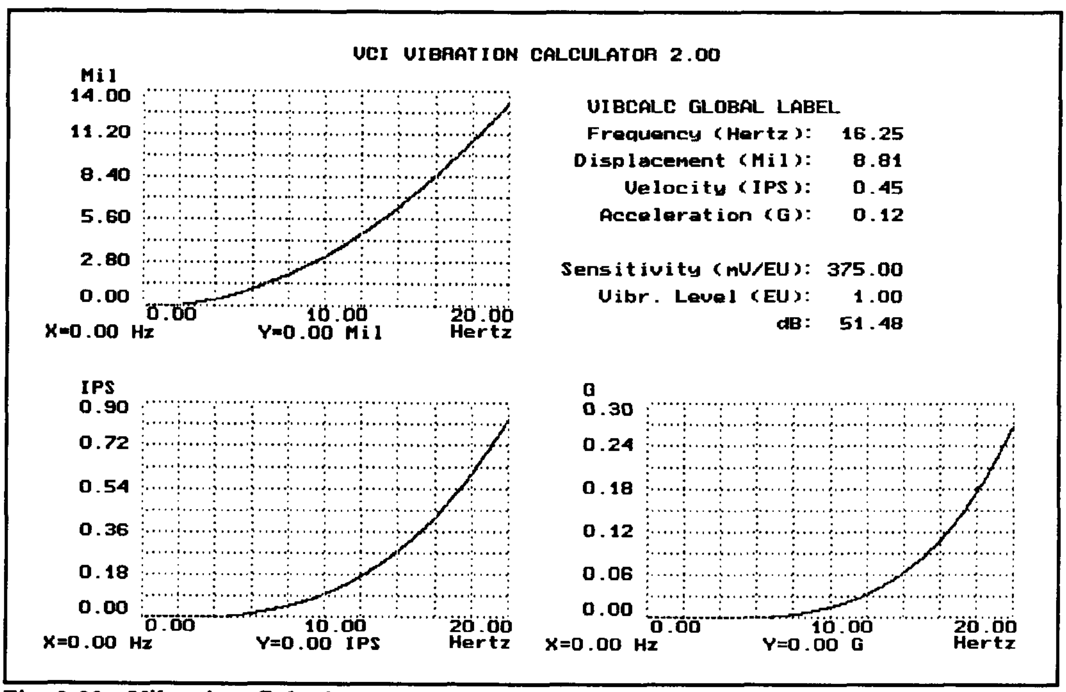


Fig. 3-20. Vibration Calculator Program Main Display.

Fig. 3-20. When actual startup or coast down data are compared to the theoretical curves, it is easy to identify imbalance.

Resonance and Deflection Calculator (RADC)

This program calculates resonant frequencies, mode shapes, shear, moment, slope, and deflection for beams, shafts, and rotors. Fig. 3-21 contains the main screen display. The program stores various cross sections, mass type, and weights. This keeps user entries to a minimum. Fig. 3-22 contains typical loading conventions and cross section types. From the operator entries, the static state data in Fig. 3-23 are generated. This display contains the shear, moment, slope, and deflection of the modeled piece. A deflection list is also generated. This list contains the amount of deflection at various points along the piece. The program calculates natural frequencies and displays the mode shapes, as in Fig. 3-24. The program permits the user to model proposed fixes to determine if the fix will solve the problem, and if new problems will arise as a result of the fix. This program improves diagnostic accuracy because the resonant frequencies can be calculated.

Bearing Calculation Program

The Bearing Calculation Program contains over 4500 bearings and provides for user updates. The program locates bearings and calculates bearing defect frequencies. The main program display is shown in Fig. 3-25.

The program features include: speed entry, or speed calculation based on machine speed and roll diameter; manufacturer file selection; bearing location; bearing editing and adding with option to calculate fundamental frequencies based on ball diameter, pitch

```
                 VCI RESONANCE AND DEFLECTION CALCULATOR 2.00

Section Number:  2

Section Length:    12.00                 Material Type: carbon steel
Cross Section Type: I-beam               Concentrated Mass:  496.00
  WD:     6.00  HT:     6.00  TH:   0.35 Inertial Disk Diameter:     0.00
Spring Stiffness:    0.00                Mass Eccentricity:    0.00

Static Force:    0.00                    Harmonic Force:    0.00
Static Torque:    0.00                   Harmonic Torque:    0.00
Distributed Load (Left):   41.00         Distributed Load (Right):   41.00


UNITS: English (inches,pounds)           VIBRATION TYPE: in-plane bending
LEFT END CONDITION: fixed                RIGHT END CONDITION: free
LABEL: RADC MODEL                        NUMBER OF DATA POINTS: 200
```

Fig. 3-21. RADC Main Screen Display.

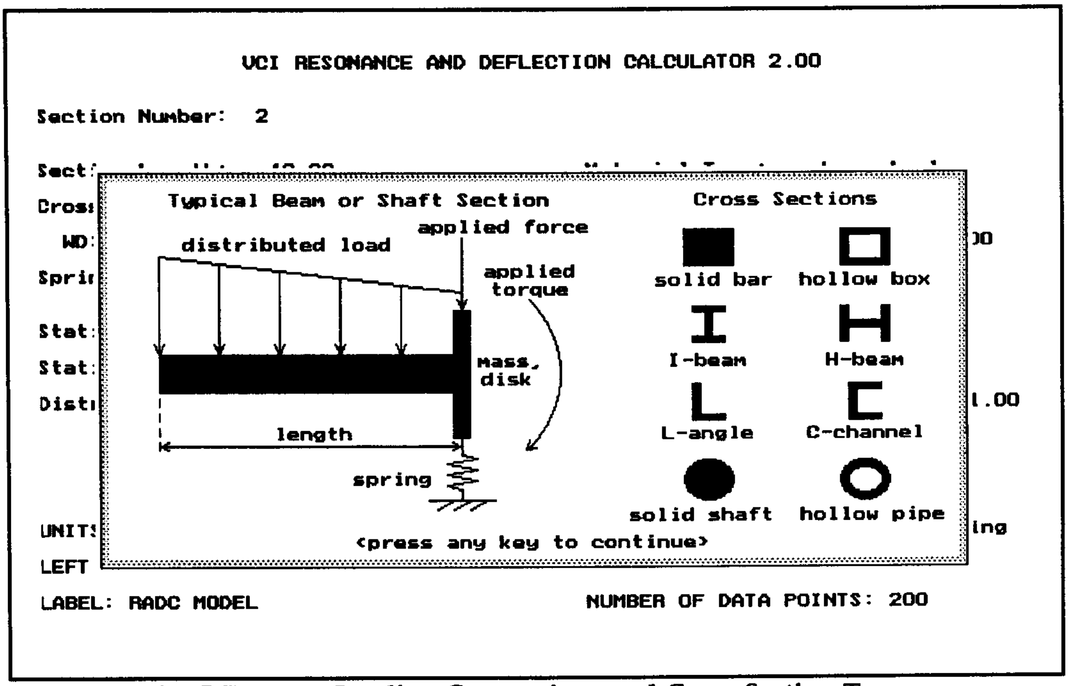


Fig. 3-22. RADC Program Loading Conventions and Cross Section Types.

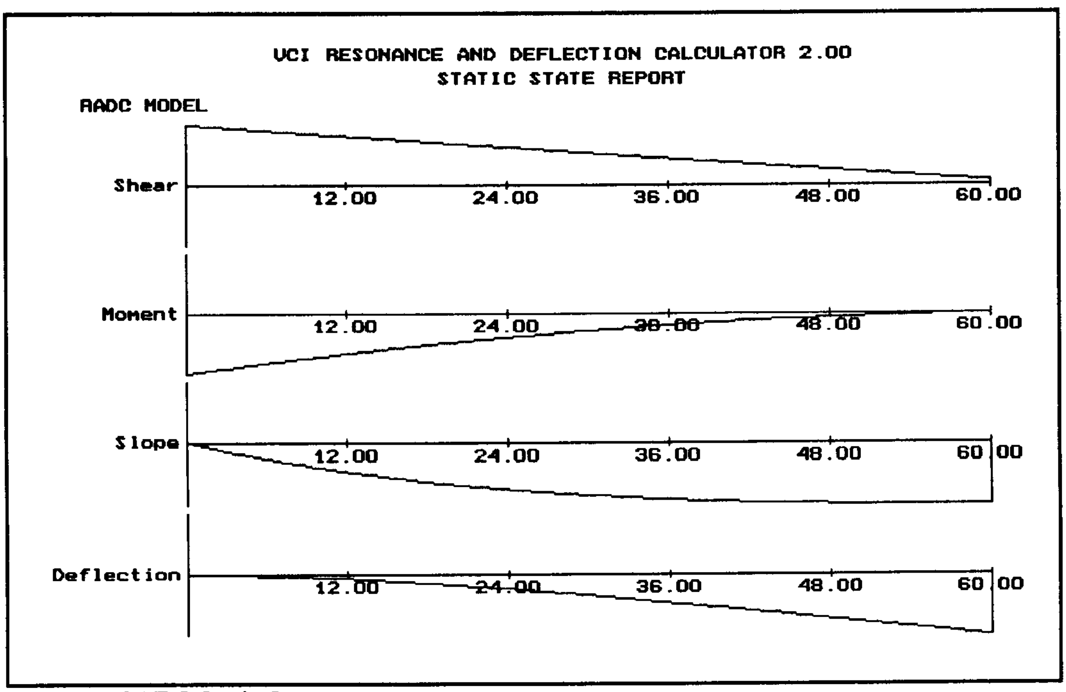


Fig. 3-23. RADC Static State.

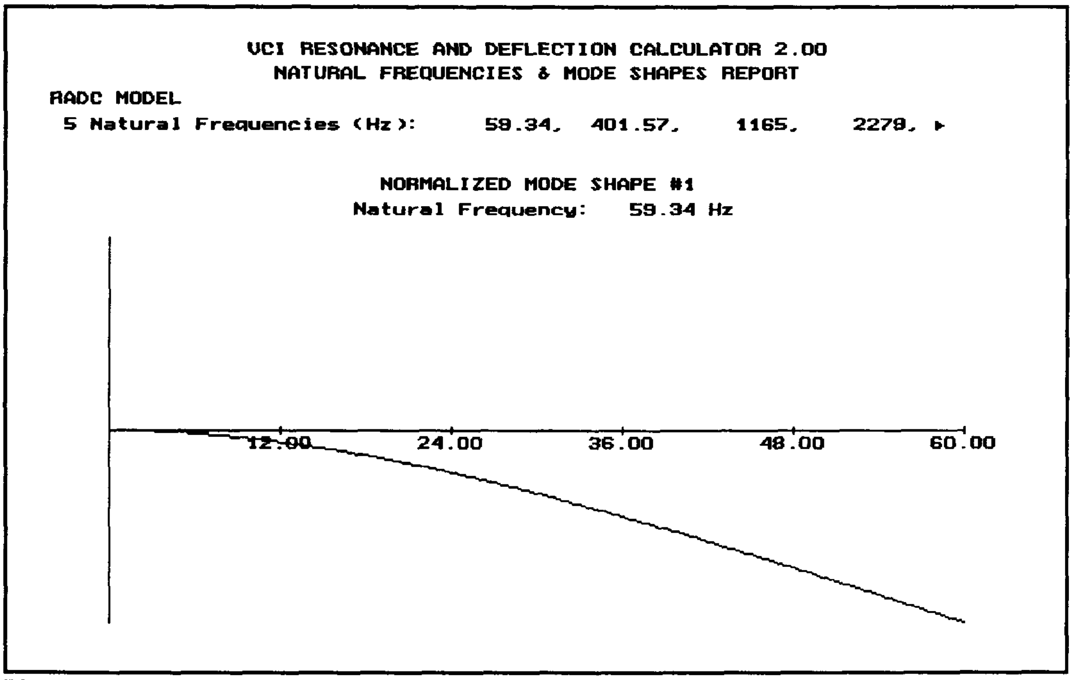


Fig. 3-24. RADC Natural Frequencies and Mode Shape.

Add Edit File selection Import DOS file Speed

File : SKF - Bearings

ID	Number of Balls	Contact Angle	BPFO	BPFI	FTF	2XBSF
22252	19	10	8.160	10.800	0.430	6.980
22252CA	18	10	7.630	10.370	0.420	6.340
22256CA	19	10	8.150	10.850	0.430	6.820
22260CA	18	10	7.640	10.360	0.420	6.400
22264CA	18	10	7.640	10.360	0.420	6.360
22272CA	19	10	8.180	10.820	0.430	6.980
22308C	13	10	5.240	7.760	0.400	4.840
22309C	14	10	5.750	8.250	0.410	5.260
22310C	14	10	5.700	8.300	0.410	5.040
22311C	14	10	5.730	8.270	0.410	5.180
22312C	14	10	5.720	8.280	0.410	5.140
22313C	14	10	5.760	8.240	0.410	5.280
22314C	14	10	5.720	8.280	0.410	5.260
22315C	15	10	6.170	8.830	0.410	5.340
22316C	14	10	5.730	8.270	0.410	5.200

Label : GLOBAL BEARING LABEL
Bearing : 22308C BPFO = 157.200 BPFI = 232.800
Speed = 30.000 Contact Angle = 10 FTF = 12.000 2XBSF= 145.200
File : TORRINGTON Bearings
Bearing : 22308 40SD23 BPFO = 157.500 BPFI = 232.470
Contact Angle = 0 FTF = 12.120 2XBSF= 146.040

F1=Help E=Exit program

Fig. 3-25. Bearing Calculation Program Main Screen Display.

diameter, contact angle, and number of balls; contact angle adjustment; and file importing to allow new bearing files to be created. The bearings are separated into files by manufacturer. The manufacturers include: Fafnir, FAG/Stamford, Link Belt, MRC, SKF, Timken, Torrington, and others.

Eight harmonics are calculated and listed for the outer and inner race frequencies. Contact angle from ball pass frequency of the outer race (BPFO) or the ball pass frequency of the inner race (BPFI) is calculated to identify bearings in a thrust load and the angle of the thrust. Cross reference is made for different manufacturers. The program calculates two times ball spin frequency (BSF). The fundamental train frequency (FTF) with inner or outer race rotating is also calculated.

Gears Program

The Gears Program simulates two gears in mesh. Fig. 3-26 displays the gears with simulation completed. The number of teeth on each gear, the gear identification, and the speed of one gear or the gearmesh frequency is entered. From this data the other speeds and frequencies are calculated. This includes gearmesh frequency and harmonics, fractional gearmesh frequencies, hunting tooth frequency, common and uncommon factors, and ratio. When the diameter of a gear is entered, the pitch-line velocity is calculated and the AGMA gear quality number is listed. The program also features a pull-down window for calculating gearmesh frequencies for planetary and sun gears. A visual display is provided on how an eccentric gear with an improper ratio can cause accelerated wear on certain teeth. The reduction in gear life expectancy is calculated based on the common factor.

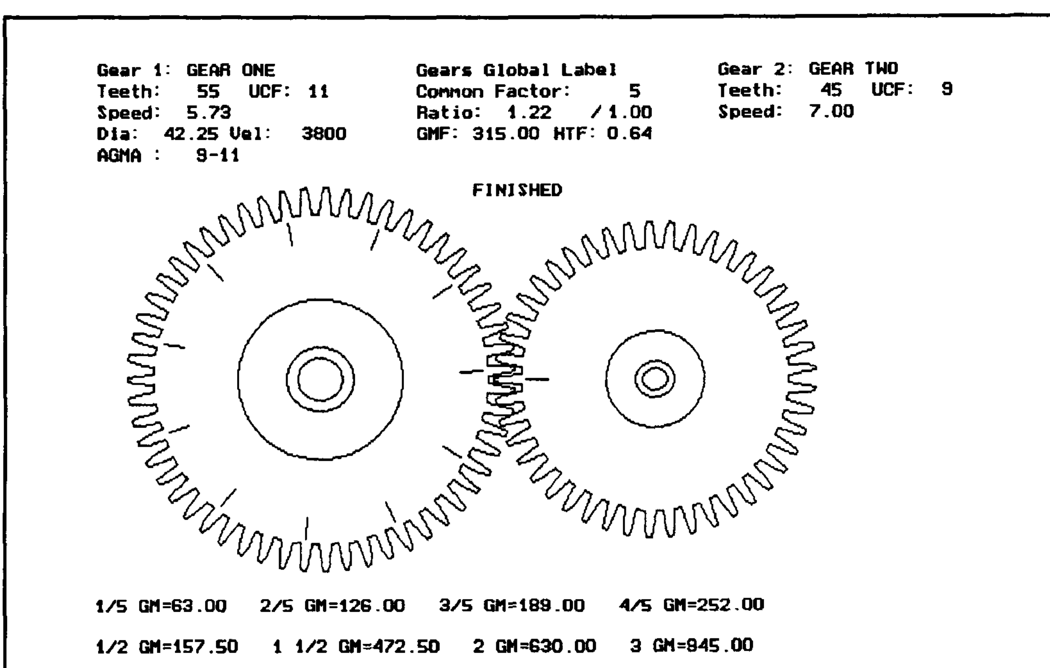


Fig. 3-26. Gears Program Display of Finished Simulation.

Paper Makers Inc, #3 Paper Machine

Roll 1: 412.00 to 412.70 inches (Yankee Dryer)
Roll 2: 87.00 to 87.50 inches (Pressure Roll)

#BARS	BAR INTERVAL	ROLL 1	ROLL 2
33 / 7	12.48	412.00	87.40
33 / 7	12.49	412.17	87.43
33 / 7	12.49	412.33	87.47
33 / 7	12.50	412.50	87.50
52 / 11	7.92	412.10	87.18
52 / 11	7.93	412.36	87.23
52 / 11	7.93	412.62	87.29
71 / 15	5.80	412.15	87.08
71 / 15	5.81	412.51	87.15
85 / 18	4.85	412.25	87.30
85 / 18	4.85	412.67	87.39
90 / 19	4.58	412.20	87.02
90 / 19	4.58	412.65	87.12
109 / 23	3.78	412.47	87.03
109 / 23	3.78	412.56	87.05
109 / 23	3.79	412.66	87.08
118 / 25	3.49	412.24	87.34
118 / 25	3.49	412.41	87.37
118 / 25	3.50	412.57	87.41
123 / 26	3.35	412.05	87.10

Fig. 3-27. Improper Roll Ratio Table.

Roll Ratio Program and Rusch Chart

Special problems are presented when two or more rolls are operated in nip. These problems also affect other items going through the nip such as a felt and product. These problems are similar to gears in that an improper ratio exists if the circumference of the rolls and/or felt have a common factor. When a common factor is present and either roll or felt has an eccentricity, the rolls and felts can get bars/corrugations/streaks. These irregularities can cause problems in the product and associated equipment failure. The distance between the bars is normally greater than the width of the nip before these problems occur.

In an effort to solve these problems the roll ratio program was developed. The maximum and minimum circumference of two components is entered. Then all size combinations of the two components that will cause barring are entered. Figure 3-27 contains a sample of the improper roll ratio table. The table lists the number of bars that will occur on each roll, the distance between the bars, and the size of each roll that can cause the bars. This table is quite large and difficult to use. To simplify the volume of data, the Rusch Chart was developed. This chart carries the name of the design engineer, Terry L. Rusch. Figure 3-28 contains the Rusch Chart. The circumference of one roll is plotted on the y-axis and the circumference of the other roll or component is plotted on the x-axis. The lines indicate where bars or improper roll ratios can occur. These lines are actually sunburst lines. The lines appear parallel because only a small segment is displayed. The blank spaces where lines do not occur indicate roll combinations where bars should not occur. The box inset can be moved around, made bigger or smaller, and then expanded to fill the screen. This provides better resolution.

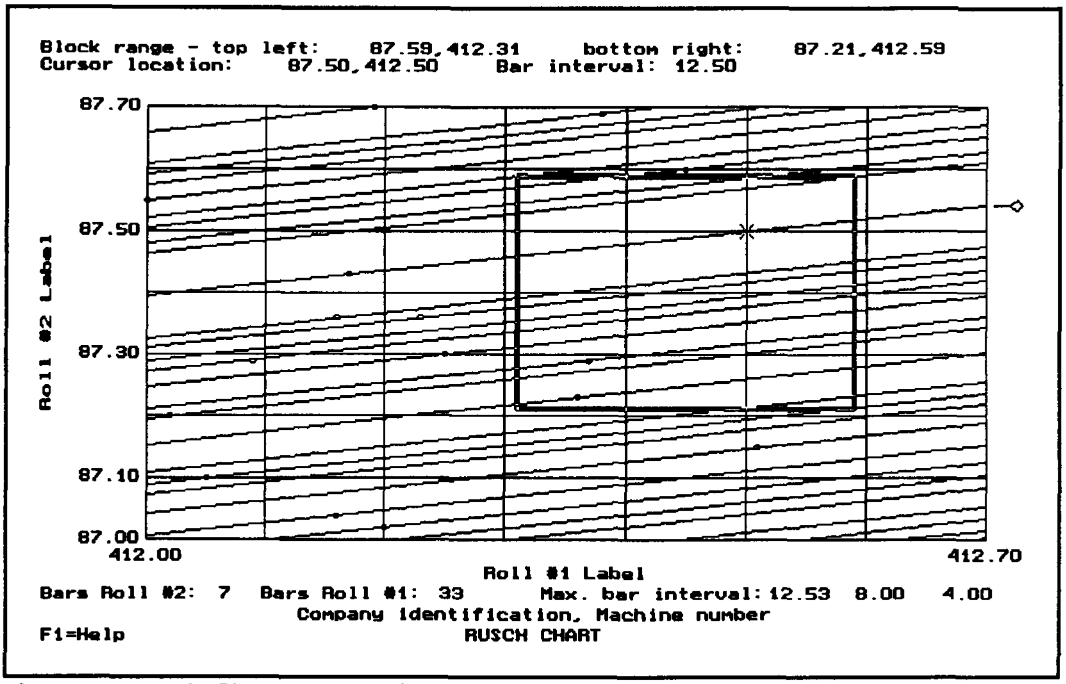


Fig. 3-28. Rusch Chart Display from Roll Ratio Program.

Group 2, Machine Doctor (MACHDOC)

This software is the vibration analysis platform for the processing of all data. It operates the base system. The digital and real-time systems are also designed to operate with the base system, and integration consists of installing additional software. The hardware required is a personal computer, the Tektronix 2630, two or four-channel analyzer, and a printer. Other software options include Polar Plot, Time Plot, Balancing, Diagnostic Database, and Expert Diagnostic Modules.

MACHDOC

This software provides the graphics for displaying the frequency spectrum and time signal. The operator can use a single display as shown in Figure 3-29, or a double display as in Figure 3-30. Up to four displays can be selected. Hardware setup is accomplished with pull-down menus. These menus are accessed with the ALT key and the first letter of each word across the top of the display. In Figure 3-29, the triggering menu is displayed. Some of the major features of the software are:

1. Cursor, with harmonics, and difference cursors in frequency and time

2. Single key labeling of up to 50 peaks in frequency and time

3. Diagnostic label storage and recall

4. Horizontal and vertical on-screen labeling

5. On-screen conversion and display of time and frequency

6. Zoom and expansion of frequencies. Expansion only of the time signal. Center frequency and bandwidth are user selectable

7. Toolbox, Polar Plot, Time Plot, and DOS accessible from function keys. When bearings or gears are accessed, the calculated frequencies are displayed on the screen upon return to the MACHDOC screen

8. Primary and secondary equipment identification labels of up to 25 characters each

9. Scaling, printing, get data or go, etc. are accessed by the first letter of the function. i.e. 'S' for scale, 'P' for printing, 'G' for go, etc.

10. Function key, "F1" help available at each function level

11. Single key storage of data, and data recall

Some of the above features are displayed in Figure 3-30.

In addition to storage and recall of data, the data can be recalled and plotted/printed as stacked spectra. Figure 3-31 contains a sample display. Some people call this a waterfall or cascade plot, however, it is not. A true waterfall or cascade plot is a larger series of

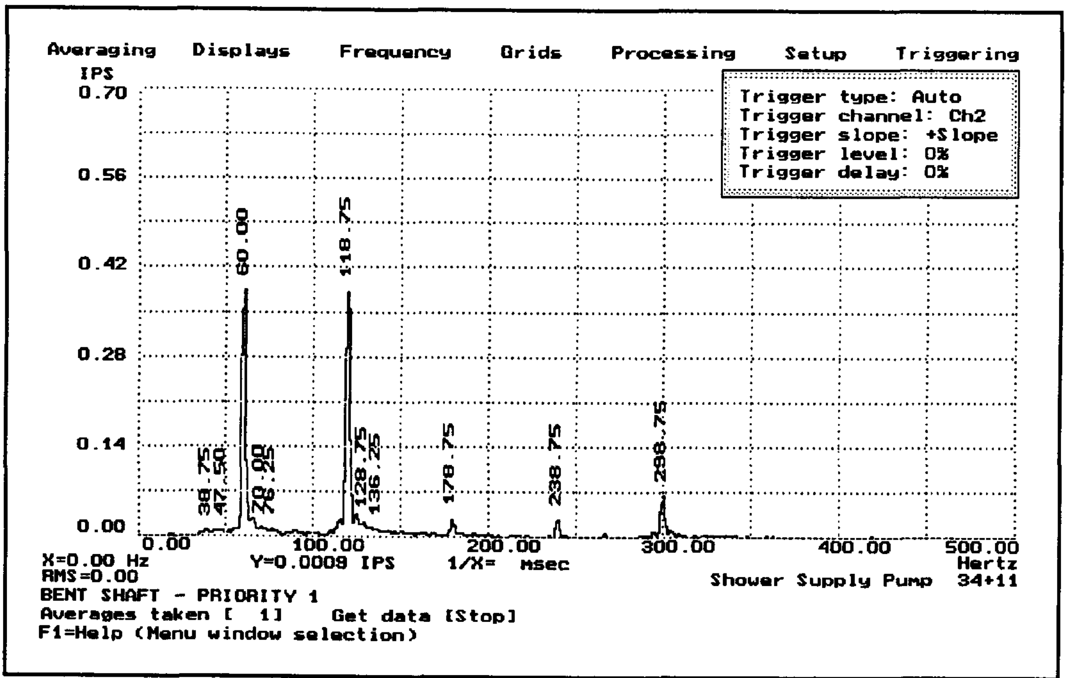


Fig. 3-29. MACHDOC Display.

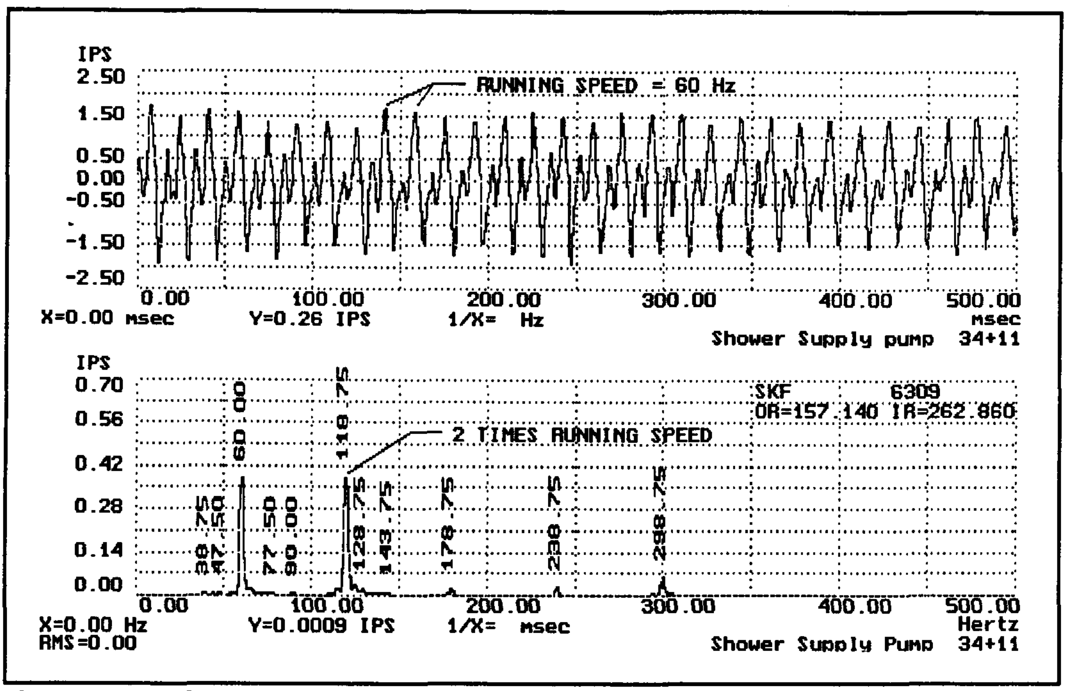


Fig. 3-30. MACHDOC Display with Labeling.

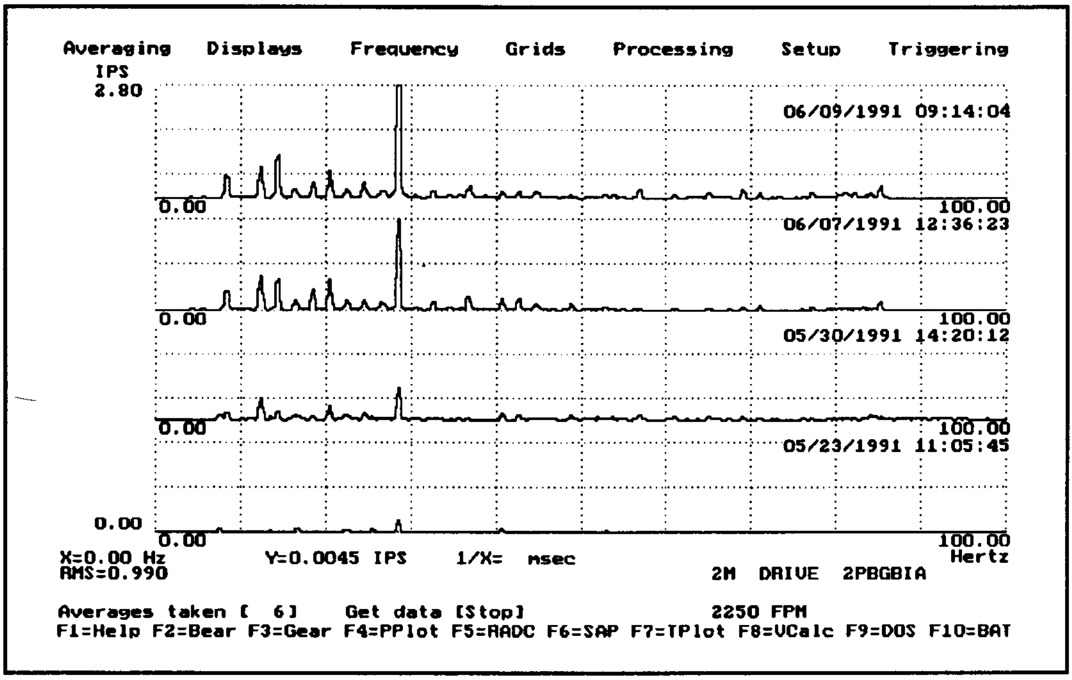


Fig. 3-31. MACHDOC Stacked Spectra Listing.

simple spectra taken a few seconds apart, while the machine is starting up or slowing. These plots are available in the Tektronix software that comes with the TEK 2600 series analyzers. These plots are also available with other real-time analyzers.

Polar Plot

Polar Plot is used to plot the time signal of one revolution around a circle. It is used for defining roll shape when synchronous time averaging is used for analyzing rolls in nip. It can also be used as an aid in diagnosing imbalance, misalignment, etc.

During synchronous time averaging and other operations where a trigger is used, this advanced design displays the trigger, averaged time signal, and averaged spectra on one screen, as in Figure 3-32. This permits the operator to observe all three functions at once. If the trigger becomes unstable or is lost, the speed changes, etc., and the operator can identify these problems when they occur.

When a trigger is used, the "F4" function key will obtain the Polar Plot as in Figure 3-33. Polar Plot contains some of the MACHDOC features such as equipment I.D. labeling, scaling, etc. The phase adjustment feature is used to adjust the phase angle so the zero point on the Polar Plot is equal to the once-per-revolution marker on the piece. This is accomplished by entering the angle between the transducer and the once-per-revolution pickup in the phase adjustment. After the angle is entered, the computer makes the adjustment. Then all problems can be referenced to the once-per-revolution marker.

When a trigger is not available, the time screen is expanded to one revolution full-scale and then you can use "F4" to obtain the Polar Plot.

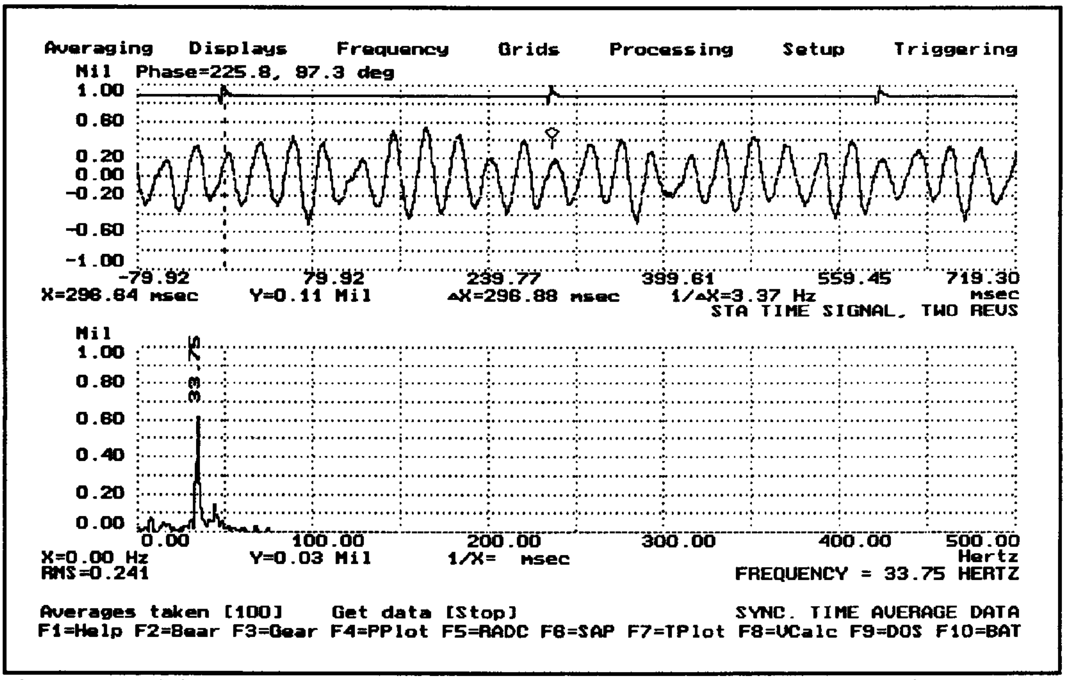


Fig. 3-32. MACHDOC Display with Synchronous Time Averaged Signal for the Polar Plot Program.

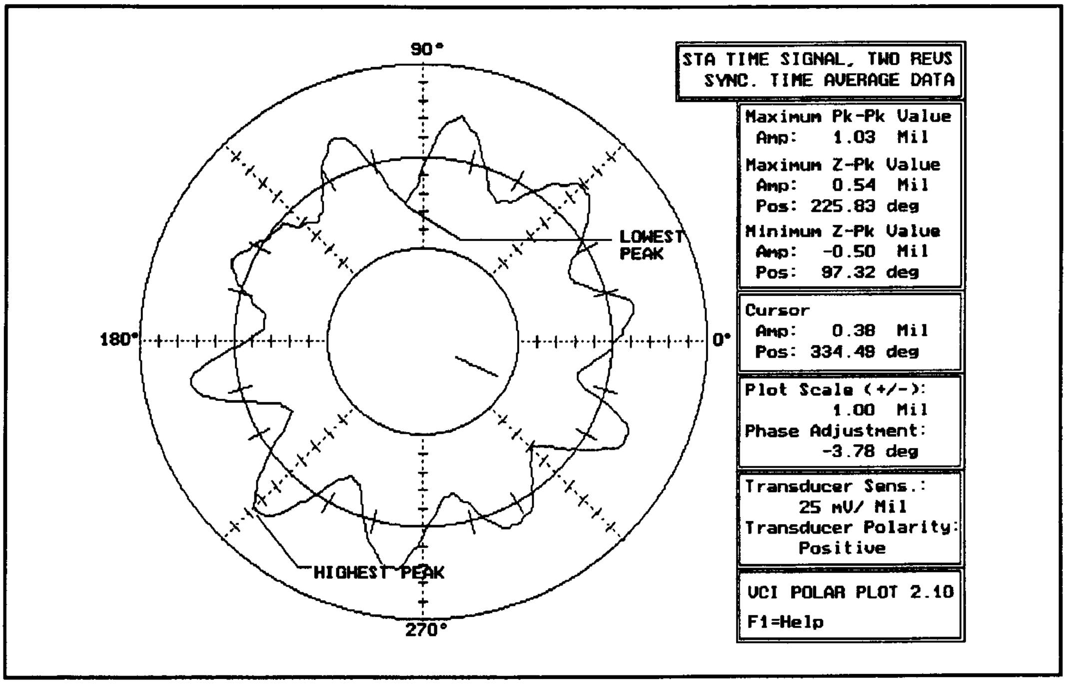


Fig. 3-33. Polar Plot Display.

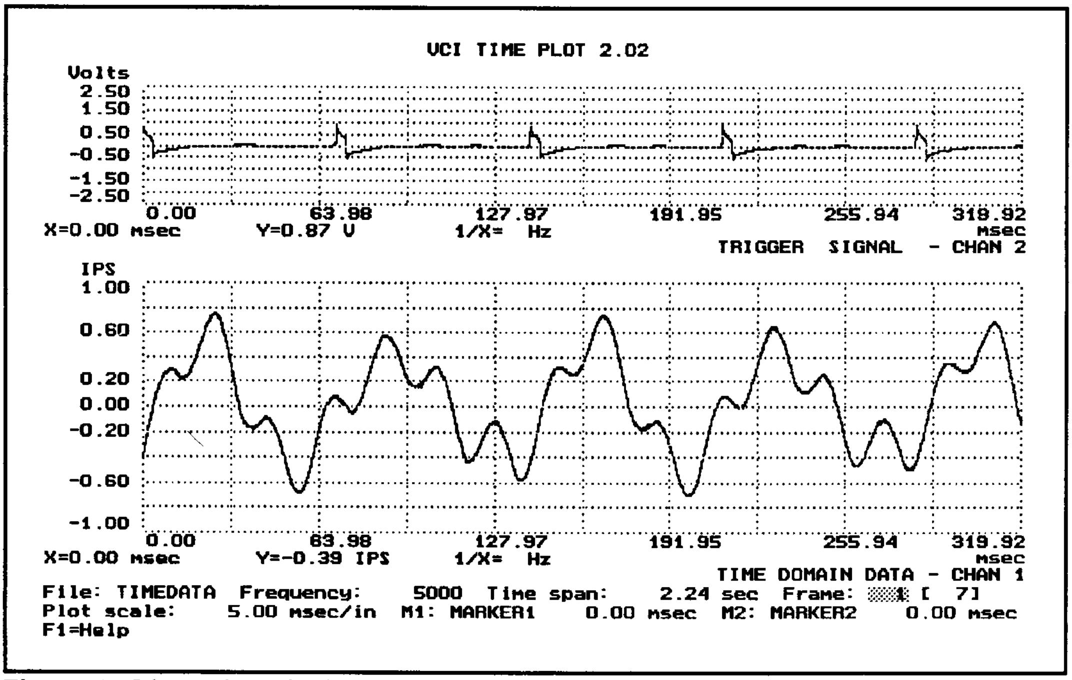


Fig. 3-34. Time Plot Display with Trigger.

Time Plot

The Time Plot Program is used with the high-speed parallel interface option in the Tektronix 2630 analyzer. This option permits the writing of an uninterrupted time signal and trigger to the hard disk in the computer. The only limitation is the space available on the hard disk. Figure 3-34 contains the Time Plot screen display. The recorded data can then be printed with user selectable resolution with two computer generated timing marks. This time signal is required to analyze the eccentricity of each gear, hunting tooth frequency, and other long time events that cannot be observed on one screen of the time domain signal.

Figure 3-35 contains a small sample of the Time Plot output with trigger and timing marks. When long time periods are used with a lot of resolution, it may take several hours to print the time signal.

Balancing

Imbalance may not be the most prolific diagnosis made concerning rotating machinery. Untold thousands of dollars are wasted annually by trying to solve vibration problems by balancing when imbalance is not the problem. However, if a rotor has imbalance, it must be balanced. All rotors should be balanced in two planes before installation. The balance tolerance for rigid rotor in Figure 4-1 should be used as a reference. Installed rotors sometimes require single plane balancing, sometimes called trim balancing. Single plane, or trim balancing, is often selected for installed rotors because it is quicker, easier, and requires less downtime. Sometimes a two plane balancing is required on installed rotors.

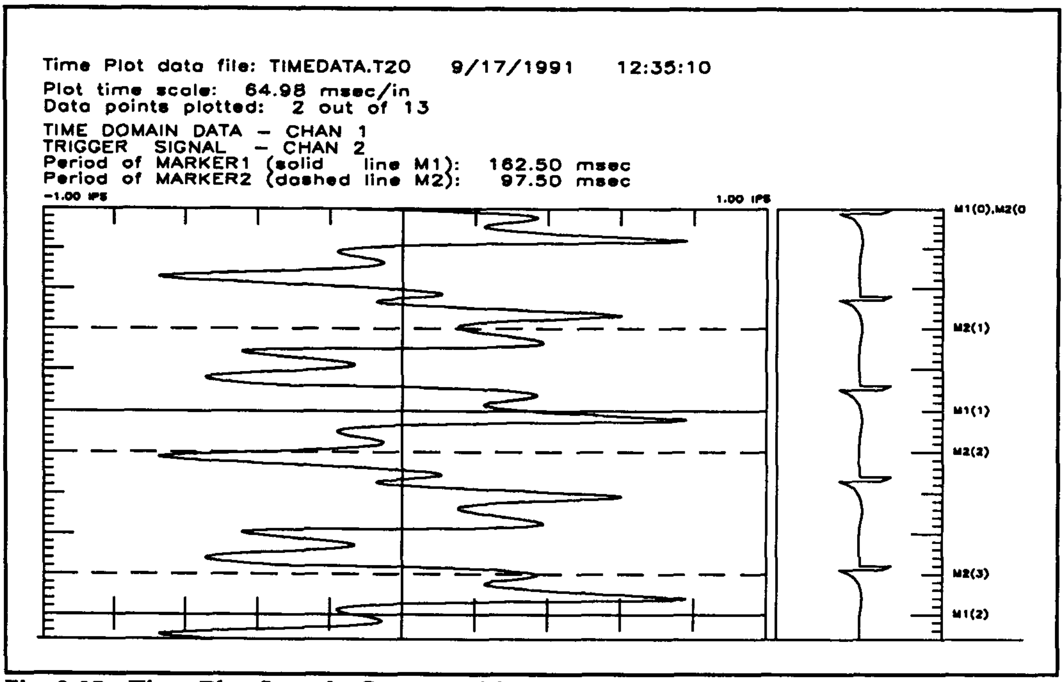


Fig. 3-35. Time Plot Sample Output with Trigger and Timing Marks.

When balancing is accomplished using a computer, all information such as whether rotor operates above or below first critical, phase lag and balance sensitivity can be stored and recalled for future balancing of the same piece. Figure 3-36 contains the single plane balancing window and Figure 3-37 contains the two plane balancing window. The computer calculates the angles, weight, and weight location. The operator installs/ connects the equipment and moves the weights.

Diagnostic Database

The diagnostic database contains the internal geometry and speed for each data point. From this information, the computer calculates the generated frequencies. It is used with the diagnostic modules for accurate diagnosis of problems with the expert software. A tree structure is used so all data can be displayed on one screen. Figure 3-38 contains the main screen display. The database is arranged by machine or group, section, and measurement point. The diagnostic information is listed for each point. This information contains required items such as transducer sensitivity and type, speed, diameter. Several bearings and/or gears can be entered for each point. Also the number of vanes on impellers, blades on fans, etc. can also be entered.

The Add, Delete, Edit, List, Mode, Path, Replicate, and Tach can be accessed with the ALT key and the first letter of each word. When more than one measurement point contains the same information, the multiple entries can be made as a single entry. For example, TR 1-6 and WRR 1-8 in Figure 3-38 represent Table Roll 1 through 6, and Wire Return Roll 1 through 8. When different sections have the same points, the replicate feature can be used to duplicate the sections. These advanced design features reduce the time required to build the database. For example, if 3 hours are used to build a database with

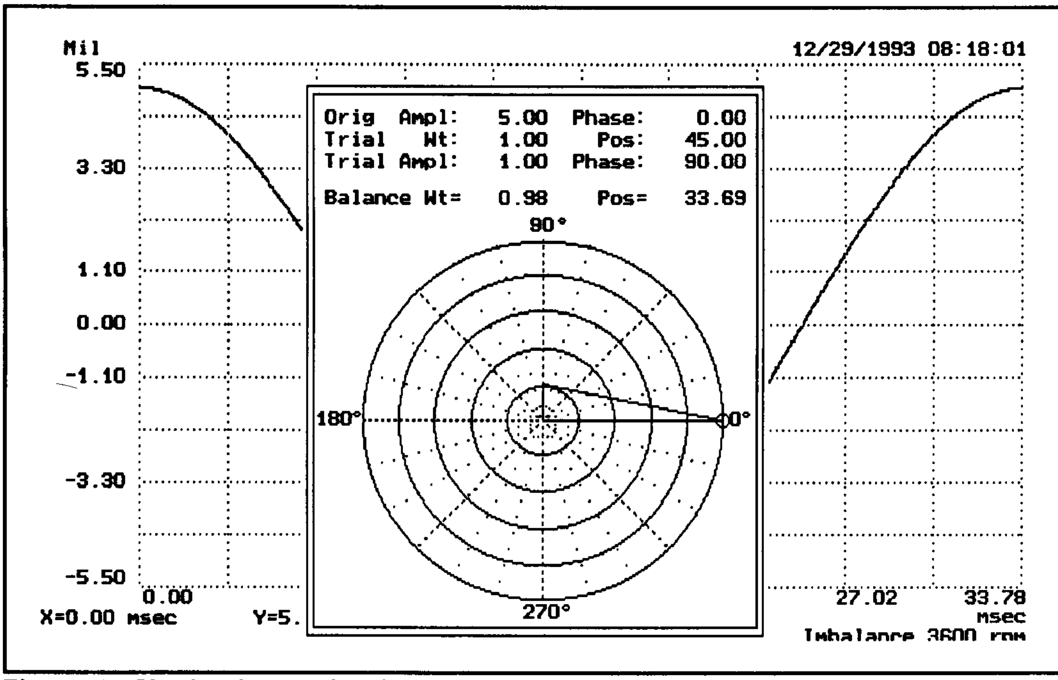


Fig. 3-36. Single Plane Balancing.

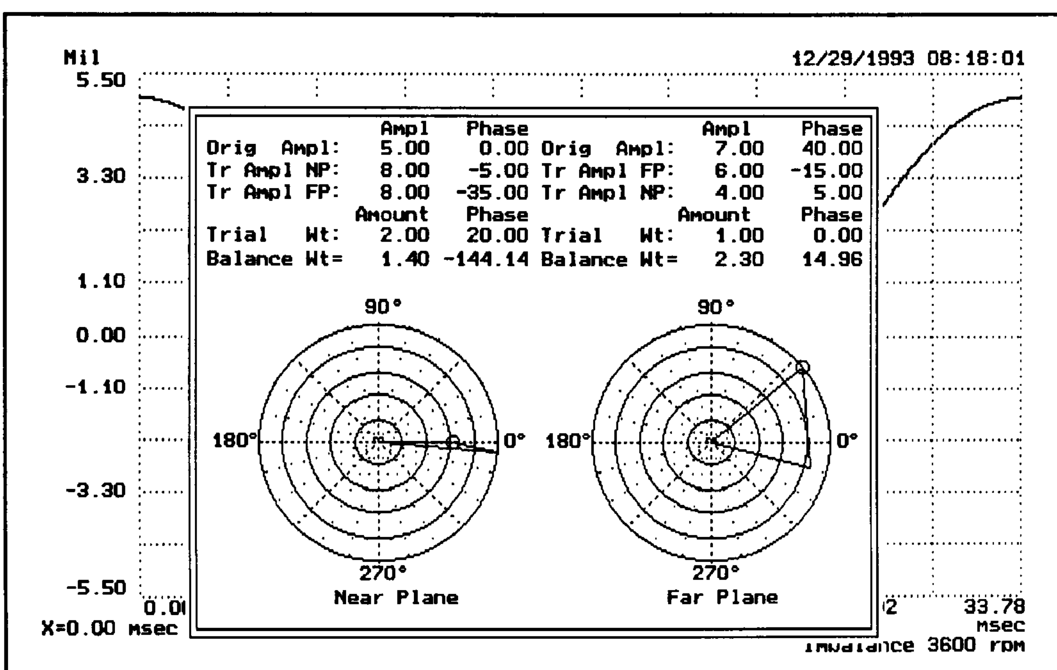


Fig. 3-37. Two Plane Balancing.

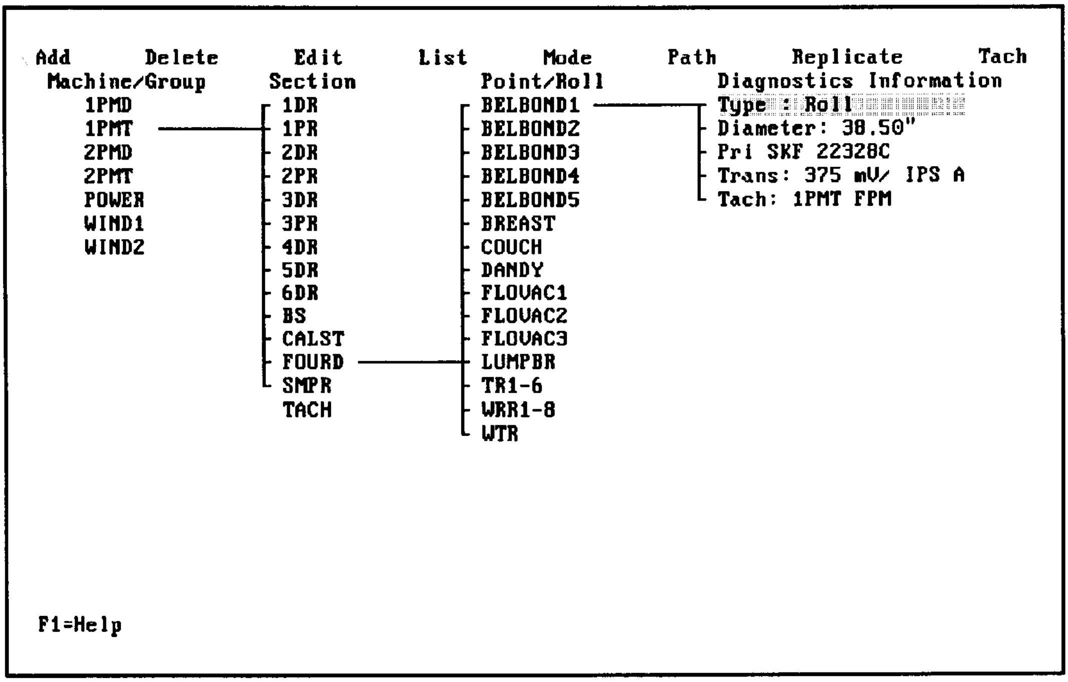


Fig. 3-38. The Diagnostics Database Main Display.

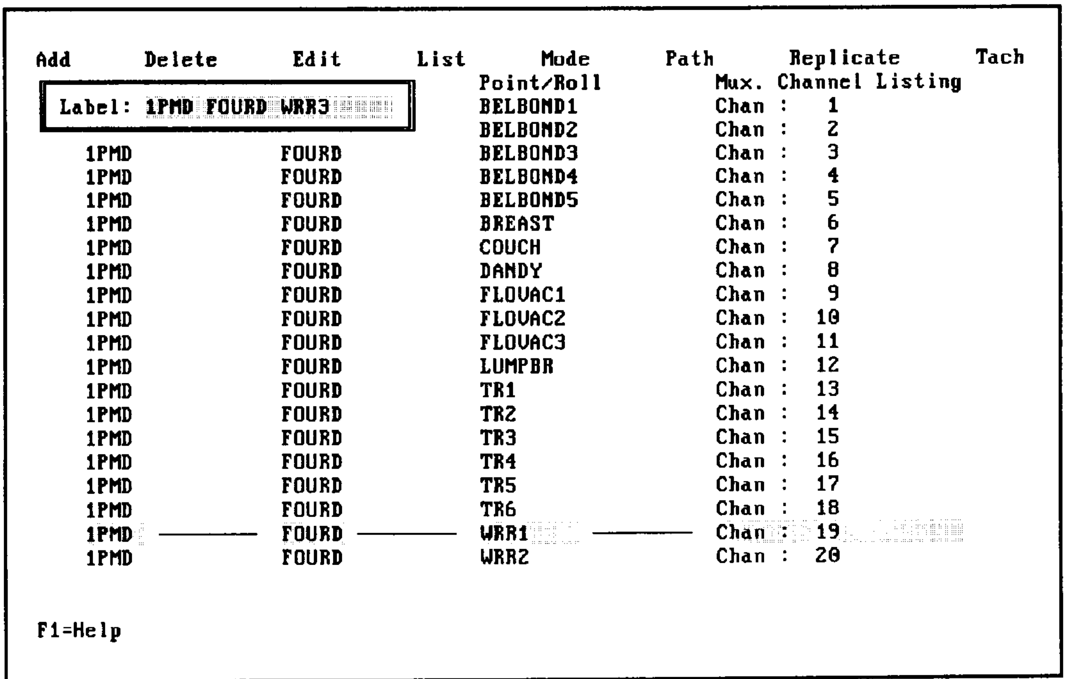


Fig. 3-39. The Diagnostics Database Real-time Option with Multiplexer Listing Display.

MACHDOC, 30 hours may be required on a less sophisticated system.

The mode selection permits access to the multiplexer listing in Figure 3-39. This listing is used with the real-time option and lists the multiplexer channel for each point.

The route listing used with the digital option is also accessed through ALT-Mode. The listing in Figure 3-40 displays measurement point, tape I.D. number, recorder channel, machine speed, length of recorded data in seconds, and the date the recording was made. This information is required for the computer to process the recorded data.

For large plants with many databases, the path function is used to access the various databases.

The list selection permits access to the various reports. To obtain these reports, the diagnostic modules must be used on stored data, recorded data, or real-time data. The priority listing contains Priority 1, 2, and 3 problems by machine group, section, and point. Figure 3-41 contains a sample report.

A comparison report is also available. This report compares the priorities between any two user-selected periods.

A sample of the History Report is contained in Figure 3-42. This report plots the trends for the overall RMS and the RMS value of each diagnosed problem, along with the machine speed. The time period is user selected. This report identifies when the defect occurred and indicates the deterioration rate.

Add Delete Edit List Mode Path Replicate Tach

Mach/Grp	Section	Pt/Roll	ID#	Chan	Speed	Time	Date
2MD	2DS	DR11	101	1	1250	35	03/01/93
2MD	2DS	DR12	101	2	1250	35	03/01/93
[redacted]							
2MD	2DS	DR14	101	4	1250	35	03/01/93
2MD	2DS	DR15	102	1	1250	35	03/01/93
2MD	2DS	DR16	102	2	1250	35	03/01/93
2MD	2DS	DR17	102	3	1250	35	03/01/93
2MD	2DS	DR18	102	4	1250	35	03/01/93
2MD	2DS	DR19	103	1	1250	35	03/01/93
2MD	2DS	DR20	103	2	1250	35	03/01/93
2MD	3DS	DR21	103	3	1250	35	03/01/93
2MD	3DS	DR22	103	4	1250	35	03/01/93
2MD	3DS	DR23	104	1	1250	35	03/01/93
2MD	3DS	DR24	104	2	1250	35	03/01/93
2MD	3DS	DR25	104	3	1250	35	03/01/93
2MD	3DS	DR26	104	4	1250	35	03/01/93

F1=Help File: TAPE1

Fig. 3-40. The Diagnostics Database Digital Option with Route Listing Display.

```
                        VCI PRIORITY LISTING
                           GROUP :  4MT
                         SECTION :  1DR
The following is considered Priority 1 for repair :

     Point         Problem

1.   1TFR6         LOOSE-PRI1, OUTER-PRI3(0.08), INNER-PRI3; ID-(9/9);
                    OVERALL-PRI1
2.   1TFSR4        LOOSE-PRI1, CAGE-PRI3; ID-(3/4); OVERALL-PRI1
3.   1TFR1         LOOSE-PRI1, OUTER-PRI3, INNER-PRI2, ROLLER-PRI3;
                    ID-(29/43); OVERALL-PRI1



                 <press any key to advance to next section>
```

Fig. 3-41. The Diagnostics Database Sample Screen Listing of Priority Report.

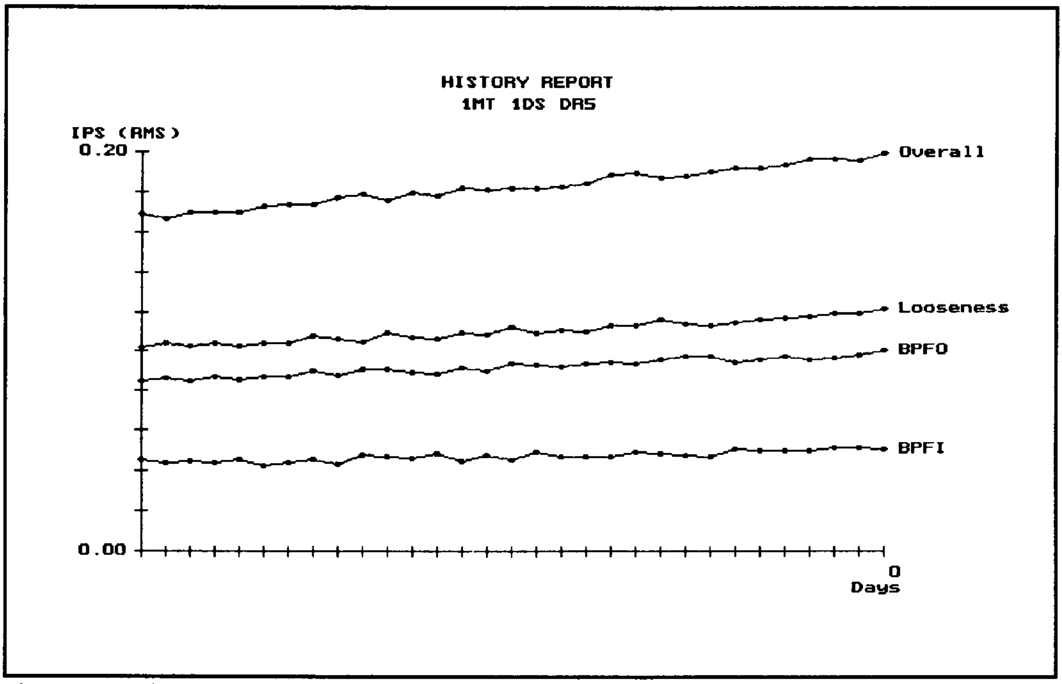


Fig. 3-42. History Report

Diagnostic Modules

The diagnostic modules perform the actual diagnostics. Since all industrial plants do not have the same equipment, the diagnostic software has been divided into four (4) broad categories. These categories or modules diagnose problems in antifriction bearings, gears, rolls in nip, and general machinery condition such as imbalance, misalignment, etc. The diagnostic database is required for the diagnostic modules. The modules are the Expert software for diagnostics.

The computer diagnostics can be performed on stored data, taped data, and real-time data from the multiplexer. The major difference between these diagnostic modules and other expert systems is: These modules diagnose specific and multiple problems, and assign a priority for each problem and an overall priority. These modules alert and alarm on actual diagnosed problems.

Figure 3-43 contains an analyzed spectra from MACHDOC. The data in the upper right corner identify the various problems, frequency, and RMS value. The letter 'L' is looseness, 'O' is outer race defect, 'I' is inner race defect, 'C' is the cage frequency, and 'R' is roller speed. These defect codes are also printed on each spectral line. The diagnosis from the expert software indicates a looseness, inner race, and roller problem at a priority 3. The outer race and overall is a priority 1. The ID-(27/37) indicates the software identified 27 of a total of 37 spectral lines. A visual observation of the data in Figure 3-12 indicates all spectral lines were identified. The human eye may not see all frequencies because of the vertical scaling. However, the computer can "see" those frequencies. Where the diagnostics were tested, a qualified engineer analyzed the data. Then the computer analyzed the same data. An analysis of the results indicated the computer found all the problems the engineer found plus several more low level problems. These are the results when a qualified diagnostic engineer programs the

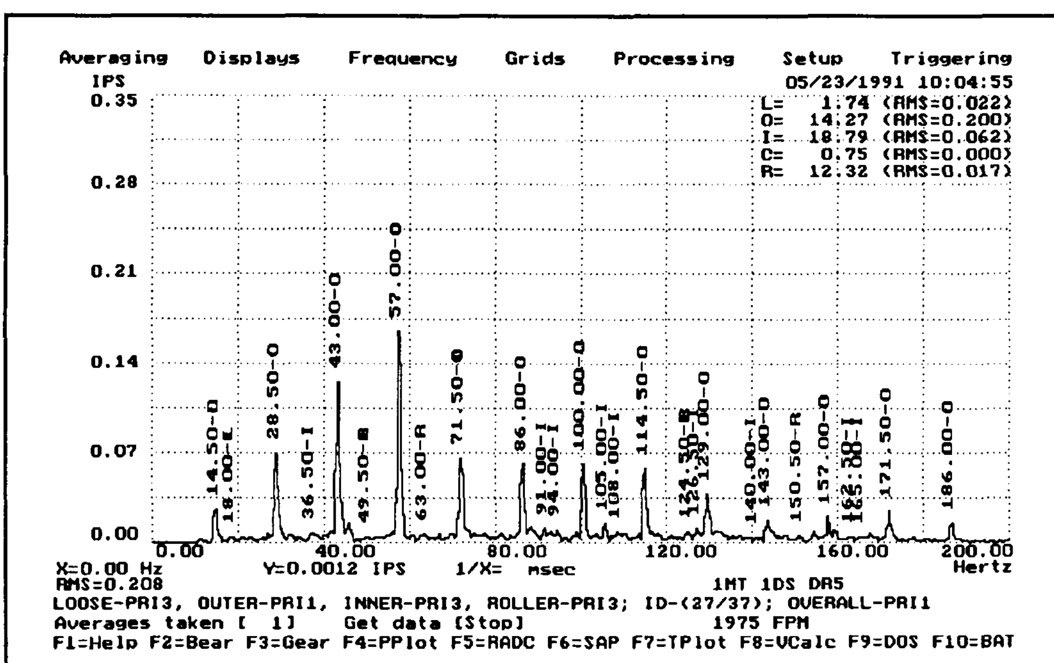


Fig. 3-43. MACHDOC Display after Diagnostics Have Been Performed.

computer to diagnose problems using the same engineering logic. These results are quite impressive. The reason the computer found more problems than the engineer, is that the engineer could not see all spectral lines because of scaling levels. The computer could identify those spectral lines.

Roll Quality Assurance Program

When the plant is operating rolls that are in nip, considerable savings can be realized by the improvement of roll management and roll quality. When rolls are operated in nip, i.e. one roll is rotating against another roll, all parameters must be held to a closer tolerance. For example, imbalance and runout should be less than 1 mil (0.001"). Most materials take a set. Some rolls may be measured round while in the grinder and be a problem when installed. When rolls with a set are installed and operated under load, the hard and/or soft places could appear as bars or corrugations from before the grind. In such cases a repeat failure could occur.

Roll roundness should be measured under operating conditions before the roll is removed. This can be accomplished by relative motion measurements between the rolls and/or dynamic measurement of steel rolls. Synchronous time averaging should be accomplished in both cases.

Roll roundness and both targeted and actual crown should be measured before and after the grind. The roll roundness should be measured again when the roll is installed, and before the roll is removed.

If the above measurements were made, and the Roll Ratio/Rusch Chart in Table 7-1 and 7-2 were used, much would be learned about the cause of roll failures, transfer function, and what should be done to prevent roll failures. Some of the new roll grinding machines are equipped with the instrumentation to measure roll roundness while in the roll grinder. For those that must continue to use what they have, the Roll Management program may be helpful. This program operates with the above hardware. While the roll is in the grinder, measurements are made with two displacement transducers mounted 180 degrees apart, in order to subtract out way and bed sag.

The roll is divided into a user-determined number of segments and measurements are made at each segment. The following reports are generated:

> The Roll Diameter Report lists average, minimum, and maximum diameter at each roll segment and location.
>
> The Polar Report displays the roll shape and runout for each segment and location.
>
> The Crown Report displays the targeted crown and the actual crown. The program measures circular, sine, and straight crowns.
>
> The Surface Report is also available. The computer presents the round surface as a flat matrix. The display can be rotated and viewed from different angles.

A database is provided for storing the reports for later retrieval and comparison.

Group 3

The digital option is a method to collect a large volume of data in a short time period and the resulting computerized processing of the data.

The measurement/data collection routes can be built in the host PC and then downloaded to a hand-held computer. Figure 3-44 contains a hand-held computer screen of a route or portion thereof on a paper machine. This software is operated by touching the computer pen to the screen and/or writing on the screen. The hand-held computer controls the DAT recorder and keeps track of the tape recorder channel, and tape I.D. number when each point is recorded. When the route is completed, the measurement point, DAT channel and I.D. number are downloaded from the hand-held to the personal computer. The DAT is then connected to the personal computer. The operator installs the tape and starts the program, and the computer processes the data. The computer can also analyze the data and diagnose the problems, if the diagnostic database and diagnostic modules are installed. There is also a subroutine that permits the analyst to screen large volumes of data for serious problems. When the computer is processing data unattended and an error occurs, the error is written to a specific file. When the analyst returns, the error file can be listed and the analyst can make necessary corrections.

Group 4

This software uses the multiplexer listing from the database and controls the multiplexer for processing and analyzing data from permanently installed transducers.

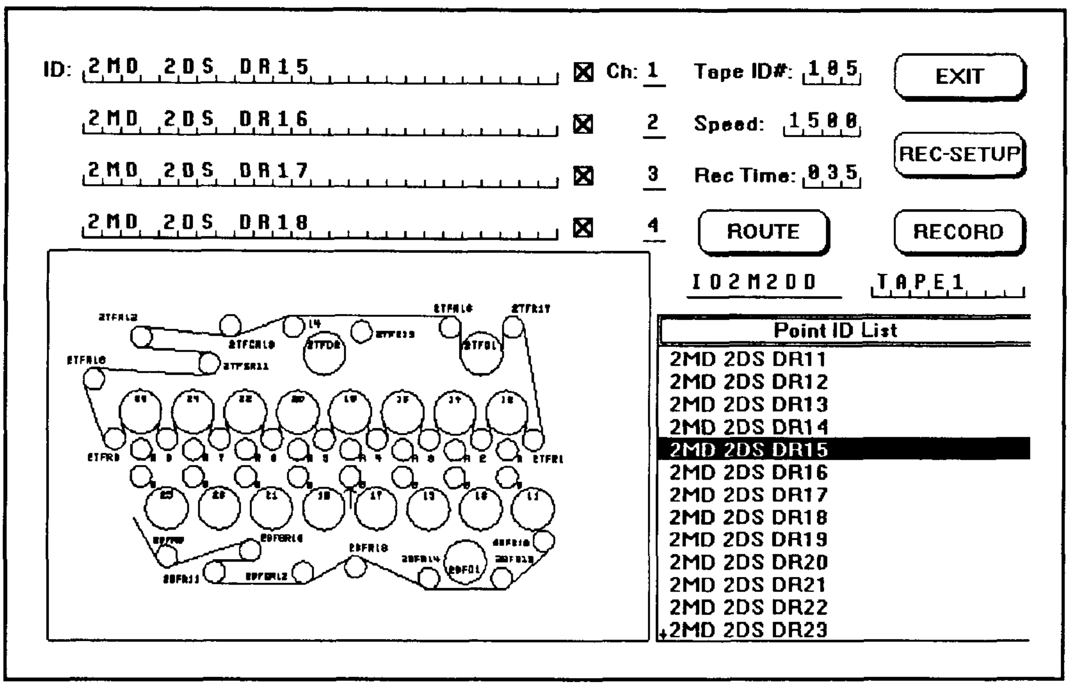


Fig. 3-44. PalmPAD Screen of Displayed Route.

CHAPTER FOUR: ACCURATE EVALUATION OF MACHINERY CONDITION

INTRODUCTION

This chapter describes the procedures for accurate evaluation of machinery condition. Diagnosis of bearings, gears, and rolls are covered in other chapters in this book. Accurate evaluation of machinery condition is more difficult because:

1. The same frequency spectra can be produced from various time domain signals. This is because the time signal contains phase and the various forms of signal distortion.

2. Some frequencies are acceptable below certain levels and other frequencies are not acceptable at all within prescribed calibration levels.

3. Some frequencies can be caused by more than one problem. For example, the fundamental or one times RPM can be caused by looseness, imbalance, bent shaft, misalignment, resonance, loading, pump starvation, open iron/broken rotor bars, and soft foot/casing distortion. The fundamental can also be measured in a machine coupled to another machine that has one of the above problems.

4. Accurate analysis of one frequency often depends upon the presence of one or more other frequencies.

The importance of the frequency domain spectra in various frequency ranges and the ability to zoom in on various frequency windows are stressed. The requirements for the time domain signal in various time ranges and the understanding of the phase between various signals are also emphasized.

As pointed out in Chapter Two, some valuable data is lost when converting from the time domain to the frequency domain. This is why the same or similar frequency spectra are presented for very different problems, and why the time domain signal is required for accurate diagnostics. In fact, several time periods in various time ranges are required. This is true because all events may not be occurring in a given time period. High frequencies are not visible if the time period is too long, and low frequencies are not visible if the time period is too short.

Problems that can occur in all machines, such as imbalance, bent shaft, misalignment, looseness, etc., are discussed separately. Problems unique to certain machine groups, such as motors, pumps, fans, turbines, compressors, etc., are discussed with that group. The importance of calibration and adjusting sensitivity for speed is also discussed.

THEORY

The measured frequency tells what is wrong with a machine, and amplitude identifies the relative severity. Frequencies do not lie, although they can be misunderstood, measured improperly, or interpreted incorrectly. Amplitude, on the other hand, can be over or understated. Amplitude is not understood by most people and is often misinterpreted. For these reasons, amplitude, all by itself, is of little value and should not be relied upon. For example, a small fan could vibrate at levels up to 1 IPS for years and not fail. On the other hand, a bearing with a cracked inner race could fail within days with an amplitude of 0.02 IPS.

Most of the vibration severity charts were designed for imbalance only. However, over the years these charts have been used and applied to all machinery problems. The net result is that machines fail and the vibration level does not go up, in fact it often goes down. Consequently, people think it is a catastrophic failure. If diagnostics had been performed along with or instead of condition monitoring, some warning would have been obtained. Fig. 4-1 is the balance tolerance nomograph which contains a good balancing specification for rigid rotors during shop balancing. When using this graph, place one end of a straightedge on the rotor weight scale, and place the other end on the rotor operational speed scale. Then read the allowable gram inches of residual imbalance from the intersection of the straightedge and the center scale. This graph is based on ISO 1940, and ANSI S2.19 - 1975 with some refinements.

Fig. 4-2 contains the operating machinery imbalance guide. This chart is a good guide for general severity of imbalance only. We do not know the origin of this chart. However, it may be a derivative of the Rathbone Chart. Various versions have been published in several places. In some circles it is referred to as the IRD Chart. Regardless of who deserves the credit, it is a good chart for imbalance severity and should not be used for other problems.

Some frequencies are permitted at acceptable levels in some machines. Some examples are imbalance (1 X RPM), gearmesh frequency, vane pass frequency, etc. These frequencies should not contain harmonics (except for 1 X RPM in fluid film bearings), modulation, or truncation. In other words, a "good" machine could have the above discrete frequencies at amplitudes of less than 0.2 IPS. Other frequencies are not permitted at any level within the prescribed calibration levels. Some examples of these frequencies are oil whirl, all bearing frequencies, pulses (except in cutters, etc.), noise, and fractional discrete frequencies. Most real-time analyzers have enough sensitivity to measure the balls or rollers passing under the transducer. If the calibration standards are not observed, or if auto-ranging is used, bearing frequencies could be observed without the presence of a significant defect.

In Chapter Two, we discussed how frequencies add, subtract, modulate, and truncate based on physics in an electronic environment. Sometimes rotating machines can behave in a very similar manner. A good example is a beat: If a 1776 RPM motor was belt-driving a 1500 RPM fan, and the motor and fan were isolated from each other, the motor would be vibrating at a frequency of 29.6 Hz, and the fan would be vibrating at a frequency of 25 Hz. This is true when there is no mechanism to allow the two frequencies to beat. If the motor was mounted on the fan pedestal, then the two frequencies could beat. The flexibility of the fan pedestal is the mechanism that permits the two frequencies to beat. If the fan and motor mounting were isolated and the motor was loose, the beat would be felt on the motor and not on the fan. If the fan was loose, the beat would be felt on the fan and not on the motor. In both cases, the looseness is the mechanism that permits the beat. In these examples, the nonlinearity (looseness and flexibility in this case) permitted the frequencies to beat. In other cases, two frequencies can be present, generated by the same machine, and they will not beat. An example would be imbalance and vane pass frequency. In such cases, a spectral line is present at rotating speed, and another spectral line is present at vane pass frequency. In this case, we have two frequencies and two problems (imbalance and a vane pass frequency problem). The time signal is that of a high frequency riding a low frequency. On the other hand, if the impeller was loose on the shaft, the spectral line at vane pass frequency could be modulated by unit speed. Modulations of speed could appear below and above

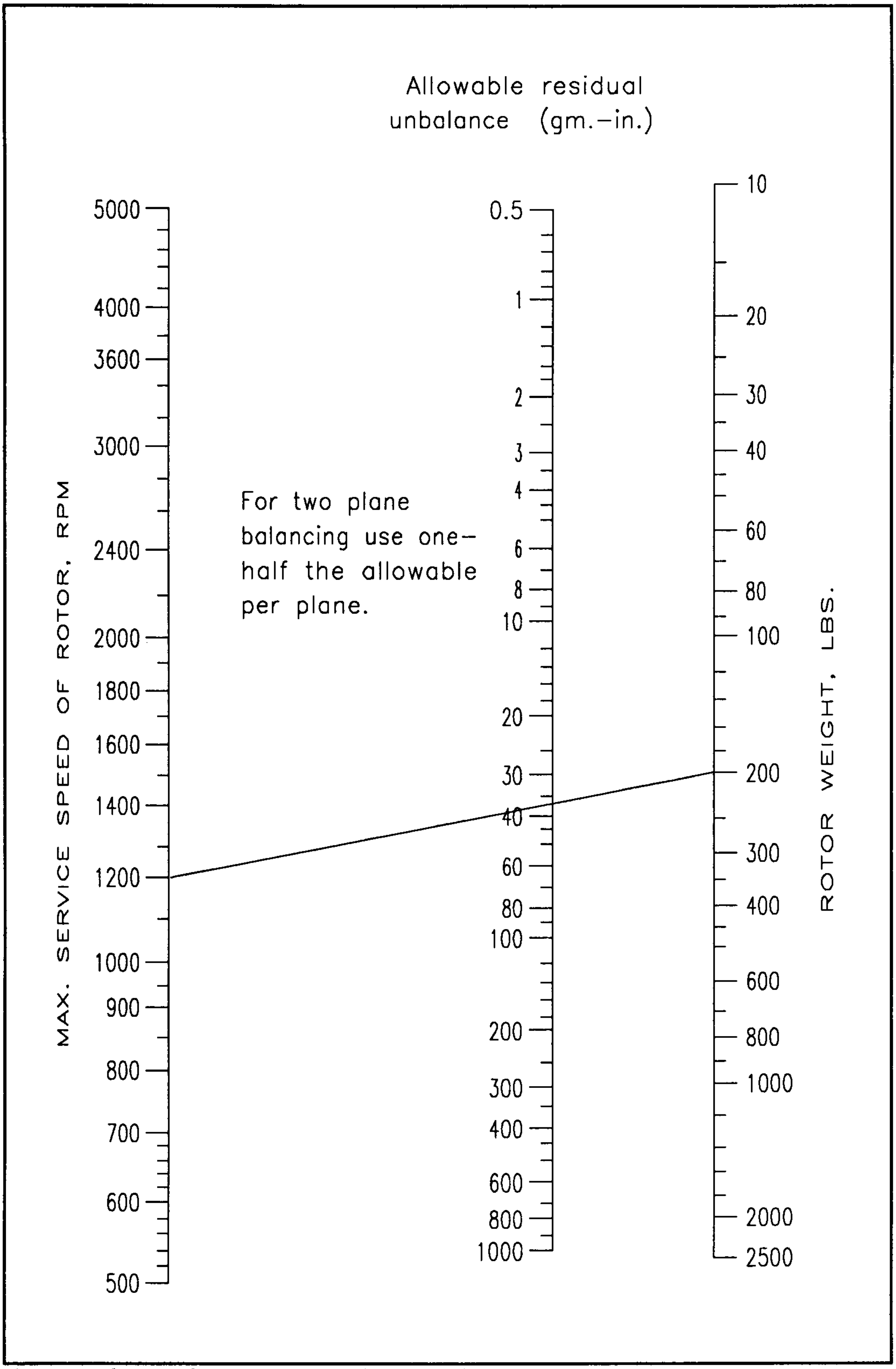


Fig. 4-1. Balance Tolerance Nomograph for Rigid Rotors.

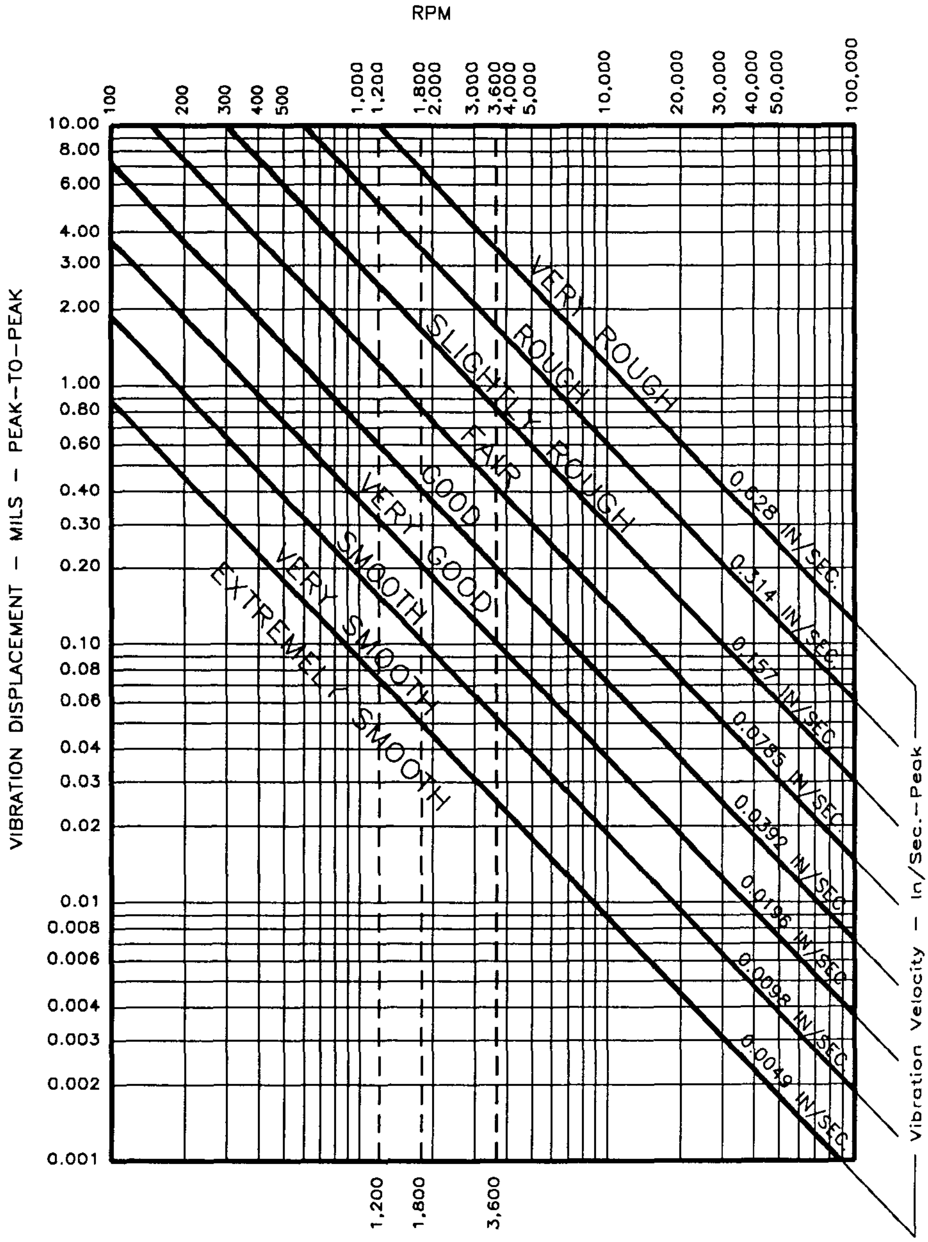


Fig. 4-2. Imbalance Severity Chart.

vane pass frequency. However, more energy may be contained in the sidebands below vane pass frequency. Sidebands below the discrete frequency are an out-of-phase condition or looseness, and sidebands above are an in-phase condition or eccentricity. In some looseness problems, sidebands can occur on the high side when eccentricity is not present. This occurs because looseness goes in and out of phase. Remember, if frequencies are out of phase, they subtract, if they are in phase, they add. In either case, amplitude modulation occurs.

In other cases, the various frequencies are not generated and then added together. In these cases, only the resultant time signal is produced. Remember, a perfect sinusoid contains only a discrete frequency. The only way to distort the sinusoid is to add some harmonic content. The distorted signal can take many forms, depending upon the harmonic content and the phase relationship of the harmonics. This signal is normally the discrete or fundamental frequency along with various harmonics and their phase relationships. Since phase is not presented or is discarded after the FFT is performed, the time signal must be used to obtain phase for accurate diagnosis. If your analysis does not include the time signal, your diagnosis could contain up to a 20% error rate.

CALIBRATION

Some frequencies at acceptable amplitudes are found in "good" machines. Some examples are imbalance, vane pass frequency, gearmesh frequency, etc. Other frequencies are not acceptable at any level within prescribed calibration standards. Some examples are the various frequencies generated by antifriction bearings, pulses, bars, or corrugations on rolls, etc. In addition, spectra taken from "good" machines should not be modulated by shaft speed or any multiple of shaft speed. However, the data from most machines contain some sidebands. The reason for this paradox is that it may not be economical to fix or repair all identified problems. Most of the problems in determining what is good or bad can be eliminated by strict adherence to calibration standards and adjusting sensitivity to speed. This writer has used the speed-adjusted calibration standards for twenty years with outstanding success. See Table 4-1.

Table 4-1. Speed-adjusted Calibration Standards.

ROTATING MACHINE SPEED IN RPM	PAPER MACHINE SPEED IN FPM	ZERO-TO-PEAK IPS
1,000 and above	2,400 and above	0 to 0.70 IPS
500 - 1,000	1,700 - 2,400	0 to 0.35 IPS
200 - 500	1,000 - 1,700	0 to 0.18 IPS
Below 200	Below 1,000	0 to 0.10 IPS

Another valuable utility of Table 4-1 is that problems of the same severity appear at about the same percentage of full scale for all speeds. For example, an imbalance problem on a 1200 RPM fan at 0.35 inches per second (IPS), or half scale of zero to 0.7 IPS, is just as bad as 0.05 IPS, or half scale on a 150 RPM fan. Of course, the latter fan would not require balancing. However, if a similar situation occurred with looseness or bearing defects, a fix would be required. Therefore, automatic ranging should be used only in rare cases.

FREQUENCIES GENERATED

The fundamental frequency or one times RPM can be caused by imbalance. When a rotor has one heavy place, the weight of the heavy place, distance from the shaft center line to the center of the heavy place, and angular rate of velocity or speed all combine to create a forcing function or vector at some magnitude and direction. If the rotor has more than one heavy place, the weight will be the vector sum of all the heavy places. As the heavy place rotates with the rotor, a frequency at the speed of the rotor is generated. Several conclusions should be made from this simple example:

1. The industry standard for measuring machine speed is revolution per minute (RPM) or feet per minute (FPM).

2. The industry standard for measuring frequency is cycles per second (CPS), now called Hertz, to honor the man who developed the frequency theory.

3. Frequency identifies the problem. If the discrete frequency is not present, it will always manifest itself as a sum or a difference frequency.

4. Amplitude identifies relative problem severity. Amplitude can be overstated by looseness and resonance. Amplitude can be understated by mass, rigidity, and damping. Amplitude can also be overstated or understated by the way most instruments measure amplitude. Harmonic content and signal distortion are other good ways to determine problem severity.

All generated frequencies are equal to events times speed. For example, half-shaft speed is generated when an event is occurring every other revolution, as in some looseness problems. If a rotor has imbalance, the heavy spot rotates at the same speed as the rotor. When two events are occurring each revolution, as with looseness, bent shaft, or two high places on a roll, the fundamental and second harmonic are generated. If three events are occurring each revolution, as with misalignment, the fundamental, second, and third harmonic are generated. It also follows that:

$$VP = S \times N_V$$

where VP = *vane pass frequency*

S = *speed*

N_V = *Number of vanes on the impeller*

Blade pass frequency:

$$BP = S \times N_B$$

Gearmesh frequency:

$$GM = S \times N_T$$

Some frequencies vary. For example, frequency is equal to the number of events times speed. If a rotor becomes loose or improperly adjusted, it could hit or strike the housing or some other object. The frequency generated can vary, depending on how many times it hits each revolution. The frequency will be the number of hits times speed.

Note that vane pass, blade pass, and gearmesh frequency are all a specific harmonic of rotor speed. In some cases, it may be difficult to distinguish between a harmonic and another generated frequency, such as vane pass frequency, ball pass frequency, etc. In other cases, frequencies such as ball pass frequency inner race (BPFI) or ball pass frequency outer race (BPFO) may be close to a harmonic of rotating speed, or vane pass or ball pass frequency. In such cases, a frequency translator or zoom feature and the time domain signal are required for differentiation between the various frequencies. Accurate analysis also requires determining the phase relationships between the fundamental and the harmonics.

DATA COLLECTION

Data should be collected in the load zone with due respect to flexibility. For example, imbalance is normally a radial-loaded problem. However, imbalance in some cantilevered fans can cause an axial load. Looseness can be either radial or axial-loaded, depending on where the machine is loose and how the machine is installed. Horizontal and vertical offset misalignment cause a radial load. Angular offset misalignment causes an axial load.

Monitoring of small simple machines can often be accomplished by collecting data in the horizontal and axial directions on the drive and driven units across the coupling. Depending on time, workload, etc., one monitoring point at 45° between horizontal and axial may suffice for monitoring. However, data must be taken in the horizontal, vertical, and axial directions when trouble-shooting and establishing baseline data for new installations, and when new equipment is added to the program.

Most failures occur on the drive and driven end of machines. Therefore, data must be taken on both sides of the coupling. Data should also be taken on the "off" end of the drive and driven unit in some cases. For example, data should be taken on the off end when the rotor is between two bearings. The location of where the data is taken also depends on what you want to measure. If a motor has the same bearing on both ends and one bearing is defective, often the location of the defective bearing can be identified by taking data from both ends.

If the problem is a radial-loaded problem, then data must be taken in the radial direction. Imbalance is a good example. If the problem is an axial-loaded problem, then data must be taken in the axial direction. Antifriction bearings and angular misalignment are good examples of axial-loaded problems. When trouble-shooting, data should be taken in the horizontal, vertical, and axial directions on the drive and off end of each machine in the complete machine train.

For routine monitoring, it may be adequate to take data in the horizontal direction on the drive and driven end of each machine. For large motors, fans, and other equipment, it may be desirable to monitor both ends. **Note:** These data collection procedures apply to equipment that doesn't have permanently installed monitors. Different rules apply to high-speed turbo machinery. The appropriate specifications should be reviewed in such cases.

TRANSDUCER SELECTION

Transducer selection depends upon the job. For example, defects in antifriction bearings for some speeds are best measured with a velocity transducer or an accelerometer. Lower speeds require a displacement transducer. If you want to measure how much one object is moving relative to another (commonly called relative motion measurement), a displacement transducer must be used. Some examples are: relative shaft motion in fluid film bearings and press/nip rolls; clearance in antifriction bearings; amount of misalignment; and bend in a shaft.

Much has been written on the selection and use of transducers, and most of it is inaccurate. The following rules may be helpful:

1. If you want to measure the amplitude of frequencies below 10 Hertz (Hz) or measure relative motion, a displacement transducer must be used. Some displacement transducers can be used to measure frequencies up to 1,000 Hz.

2. For measuring frequencies between 10 Hz and 2,000 Hz, a velocity transducer is the best selection in most cases.

3. For measuring all frequencies above 2,000 Hz, an accelerometer must be used. When an accelerometer is used, it must be either screwed down or glued down. If this rule is not followed, the amplitude of some frequencies can be over or understated as much as four times. Also, some higher frequencies can appear as wide-banded noise and not as discrete frequencies with sidebands. Both situations cause serious errors in analysis. Accelerometers can be used to measure frequencies below 2,000 Hz. However, the requirement for hard mounting is time consuming, and would not be the choice of an informed analyst.

4. When diagnosing problems in compressors and pumps, data from a pressure transducer is often required. These transducers are calibrated in mV/PSI and measure the pressure fluctuations in gases and liquids.

Refer to the frequency response curves in Fig. 1-27 for transducer selection. Data is often required from more than one type of transducer for accurate diagnosis.

CONTINUOUS MONITORING

Without question, high-speed turbo machinery and other high-speed machines should be monitored on a continuous basis. The displacement transducer is the best choice for measuring relative motion of the shaft. One of the biggest problems is the failure to monitor transducer gap voltage. In some cases, a machine can fail, and the vibration level may not increase significantly. However, the gap voltage will increase substantially. The other problem is a misunderstanding of the frequency response curves discussed in Chapter One. The result is that accelerometers are not installed to monitor high frequencies. It is essential to monitor high frequencies with an accelerometer. If this is not done, the machine can wreck, and a warning will never be observed by the displacement transducer. The major cause of such failures is normally high frequency rubs.

Diagnostic technology has developed to the point that the computer is programmed to diagnose problems. This on-line or off-line expert system is now available and is economical for most plants.

COMMON PROBLEMS

A group of problems that is common in many machines includes imbalance, bent shaft, misalignment, soft foot, looseness, resonance, rubs, and problems that cause pulses. Each of these problems/conditions is discussed in the following sections.

IMBALANCE

Imbalance is a linear problem. If a rotor is out of balance, it should be out of balance by the same amount, through 360° of rotation. Each cycle in the time domain signal will have the same amplitude and the time signal will be sinusoidal. The first four or five harmonics of rotating speed at low levels (about 0.05 IPS) are normally present in fluid film bearings. The following conditions cannot be present with pure imbalance:

1. Noise skirts at the base of the fundamental
2. Half shaft or fractional shaft speed
3. Harmonics of speed in antifriction bearings or high-amplitude harmonics in fluid film bearings
4. Wide-banded noise
5. Any form of a beat, or amplitude modulation

Fig. 4-3 contains a frequency spectrum and time signal of imbalance. Fig. 4-4 contains a frequency spectrum containing some of the above undesirable frequencies. **Warning:** If a rotor is out of balance and other problems are present, all other problems must be repaired before the unit is balanced. If you have ever tried to balance a rotor and did not succeed after four trial runs, you either made an error or you were trying to solve a vibration problem by balancing when imbalance was not the problem. If you have ever observed a rotor that has balance weights welded all around the rotor, you have observed many attempts to solve a vibration problem by balancing when imbalance was not the problem. Rotors with a vibration problem should not automatically be assumed to be out of balance. True imbalance can be diagnosed using this technology and some perseverance.

BENT SHAFT

A bent shaft is a form of imbalance and balancing can reduce the vibration level.

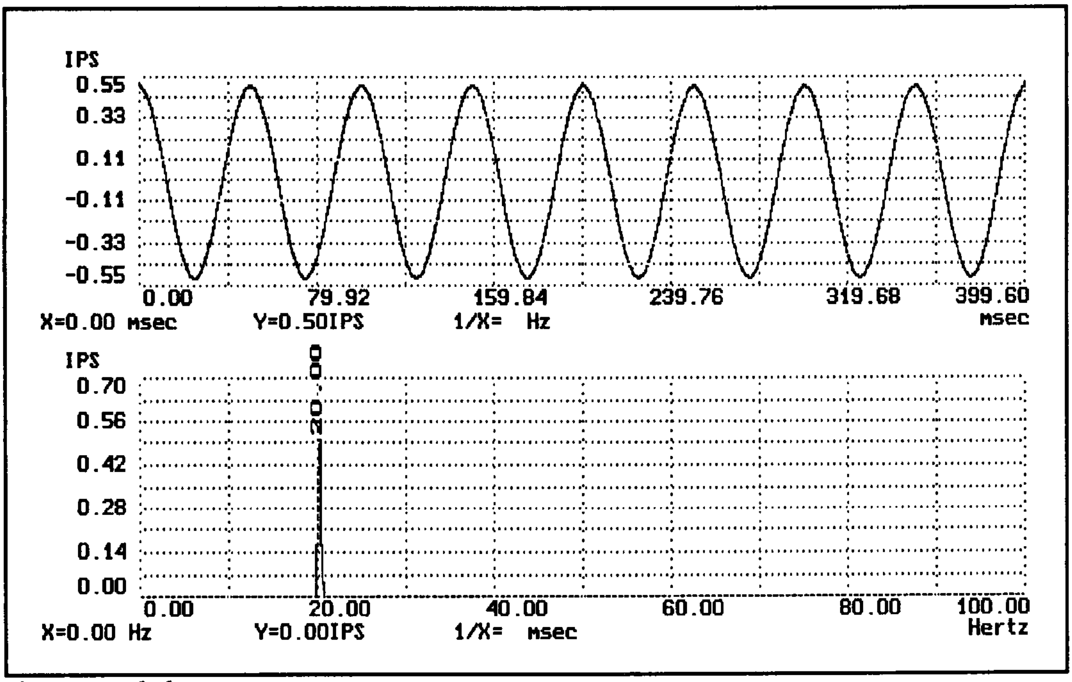


Fig. 4-3. Imbalance.

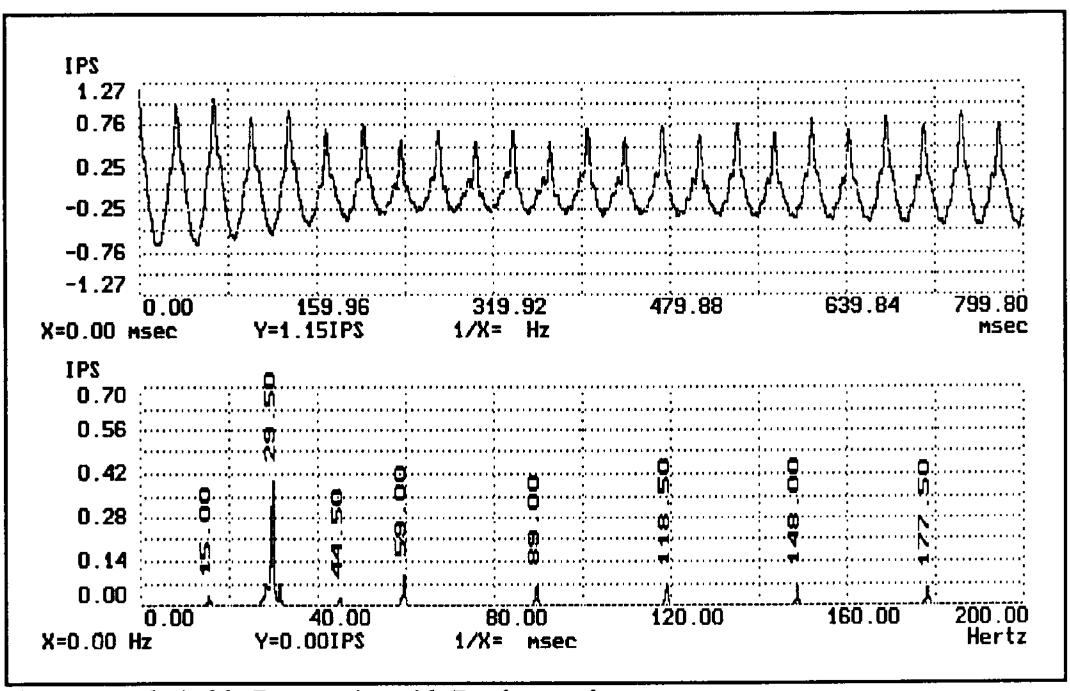


Fig. 4-4. Undesirable Frequencies with Fundamental.

However, balancing cannot straighten the shaft. The bent shaft prevents adequate alignment in some cases and causes clearance problems in others, depending on where the bend occurs.

The characteristic spectrum of a bent shaft is a distinctive second harmonic, as in Fig. 4-5. If the shaft is bent enough to cause a misalignment problem, one of the indications of misalignment could also be present. If a shaft is bent, the end of the shaft will rotate in an orbit. When this occurs, a force vector will be felt on the coupled shaft through 360 degrees of rotation. This force produces a high amplitude signal at the fundamental speed, as in Fig. 4-3. For example, if a motor has a bent shaft and is coupled to a pump, the time signal and frequency spectra could be similar to Fig. 4-5. The second harmonic is distinctive and is in phase with the fundamental. The frequency spectra and time signal from the pump indicate imbalance. However, the problem is caused by the bent shaft on the motor. The time signal is a good sinusoid. This is true because the shaft is bent by the same amount through 360° of rotation. The only distortion would be that caused by the coupling, looseness, etc. If the second harmonic is not in phase or is changing phase with the fundamental, the unit is loose and the shaft may not be bent. Where the unit is operating above first critical, there is a 180° phase shift in the residual imbalance. Since the imbalance is caused by the bend in the shaft, the truncation could move to the top of the time signal in some cases, and the second harmonic would be 180 degrees out of phase with the fundamental. The section on coast down data provides more insight. When trouble-shooting, data should be taken in the horizontal, vertical, and axial directions. A bent shaft on a motor that belt drives another unit could look like imbalance, as in Fig. 4-3, or the second harmonic could be present, as in Fig. 4-5. As a simple test, insert a wooden pencil in the center hole at the end of the shaft. If the pencil vibrates, the shaft is bent. The VCI Model 340 proximity probe can also be used to measure the total indicated runout (TIR) of the shaft while the machine is running.

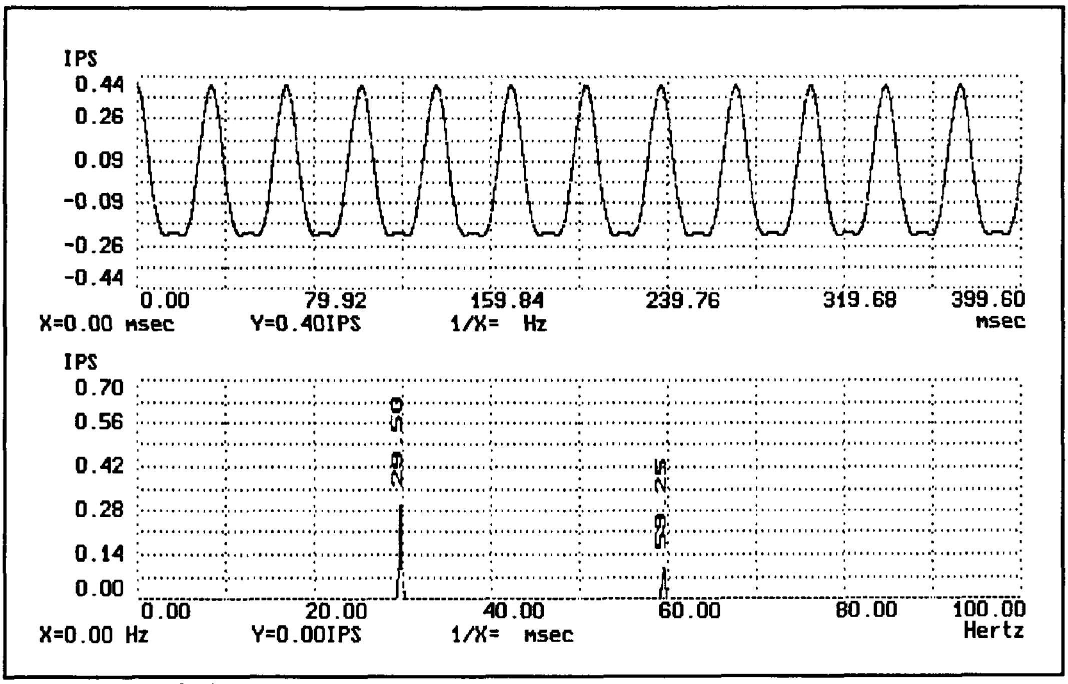


Fig. 4-5. Bent Shaft.

SOFT FOOT

Soft foot can generate a high level fundamental or second harmonic. Soft foot can and often does indicate a misalignment problem. The best way to prove soft foot is to observe the vibration level while tightening or loosening the bolts on each foot. The vibration amplitude will increase or decrease significantly when pressure is applied to the bolts on the soft foot. The frequency spectrum and time signal can indicate imbalance, bent shaft, looseness, and/or misalignment, depending on how the rotor and casing are distorted by the soft foot. When a soft foot is present, the machine may have to be removed, reset, and realigned.

MISALIGNMENT

Misalignment can occur in several places in rotating machinery. For example, misalignment can occur along the shaft centerline between bearings. Misalignment can also occur in meshing gears (refer to Chapter Six). However, misalignment more frequently occurs when two machines are coupled together. When misalignment occurs, the first three harmonics are generated. If coupling halves are misaligned, the first three harmonics of rotating speed are generated. If gears are misaligned, the first three harmonics of gearmesh frequency can be generated. For these reasons, solo data should be taken on all motors and turbines before they are coupled to the driven unit.

If you want to know the type of misalignment, take data in the horizontal, vertical, and axial directions on the drive and off ends of the drive and driven units. If the first three harmonics are distinctive in:

1. the horizontal direction, there is vertical offset misalignment
2. the vertical direction, there is horizontal offset misalignment
3. the axial direction, there is angular misalignment
4. all directions, there are horizontal offset, vertical offset, and angular misalignment

If the first three harmonics are distinctive, then misalignment is present, regardless of the amplitude within prescribed calibration levels. The problem is that most people think low amplitude levels indicate a good machine. This could be true in some situations, however, there are severe problems that can occur with low amplitudes. Misalignment is one of them. In the "real world," most misalignment problems go undetected and unrepaired until the coupling, shaft fits, or the bearing(s) are worn out.

There are many types, combinations, and degrees of misalignment. The signals obtained can also vary with coupling type, shaft size, and speed. Because of these factors, a single example of misalignment is not adequate. However, this writer has observed several distinctive patterns associated with misalignment. Please remember various combinations of amplitude and phase can cause some signals to be very different. Fig. 4-6 contains a spectrum and time signal of misalignment. Please note:

1. The first three harmonics of speed are distinctive. The amplitude

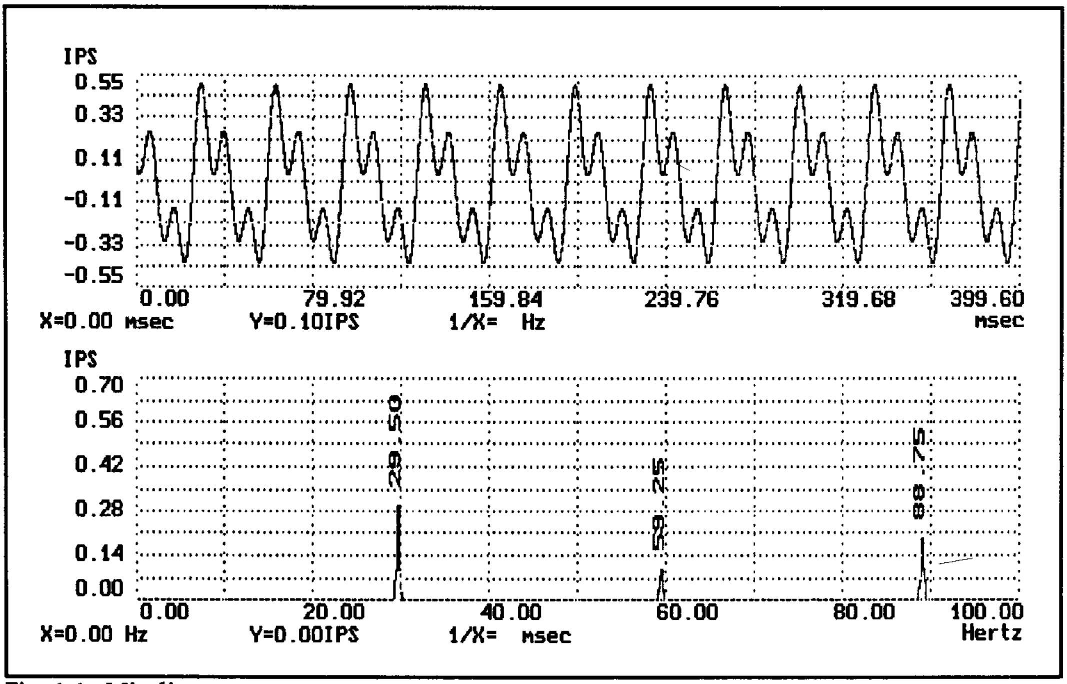


Fig. 4-6. Misalignment.

relationships could vary significantly. If the amplitude of the second harmonic is higher than the first and third harmonics, the shaft may be bent.

2. The time signal contains a fast rise time on one side and two positive-going peaks on the other side of each cycle. As the shafts rotate, the coupling bends and then frees up once each revolution. When the coupling is binding, the three peaks are generated. When the coupling frees up, the fast rise time is generated.

3. The time signal is normally truncated on the negative side because the signals are in phase. However, in some cases, the signals could be out of phase. Operating speeds above first critical is a good example.

4. The time period for each half revolution may not be equal. The unit speed decreases when the coupling is in a bind and increases when the coupling is free.

The amplitudes of the first three harmonics can approach a 6 dB per octave slope, as in Fig. 4-7, or a 12 dB per octave slope, as in Fig. 4-8. The 6 dB per octave slope indicates a level of increased binding of the coupling. When the signals approach a 12 dB per octave slope, the amplitude may increase, the coupling may be locked and rigid, and severe wear and/or failure may occur relatively quickly.

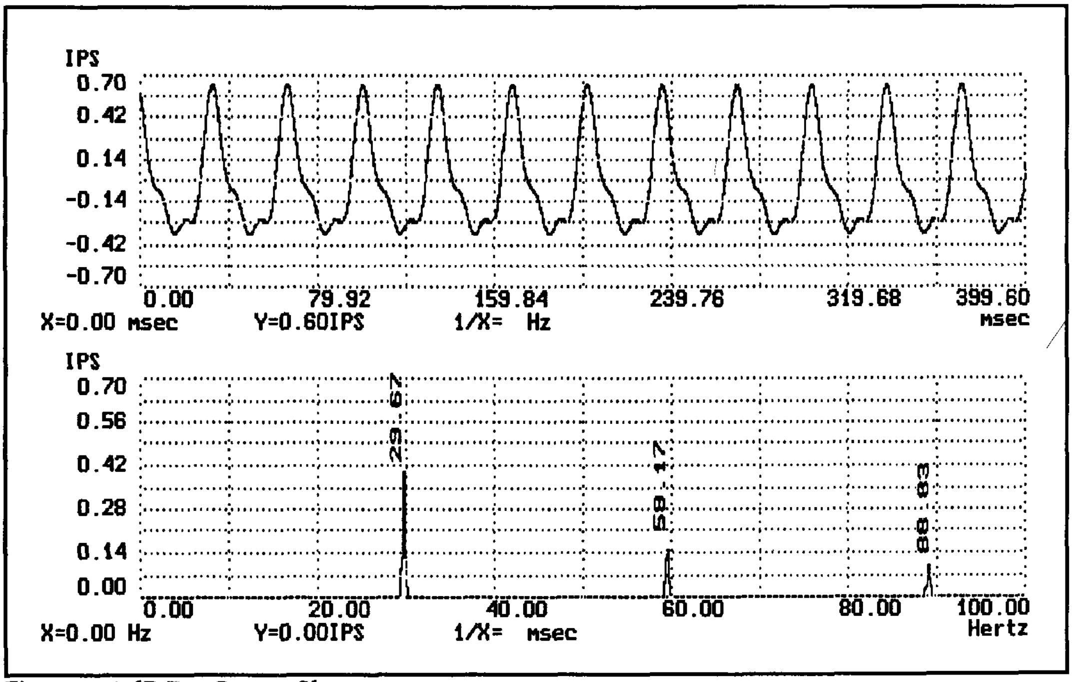


Fig. 4-7. 6 dB Per Octave Slope.

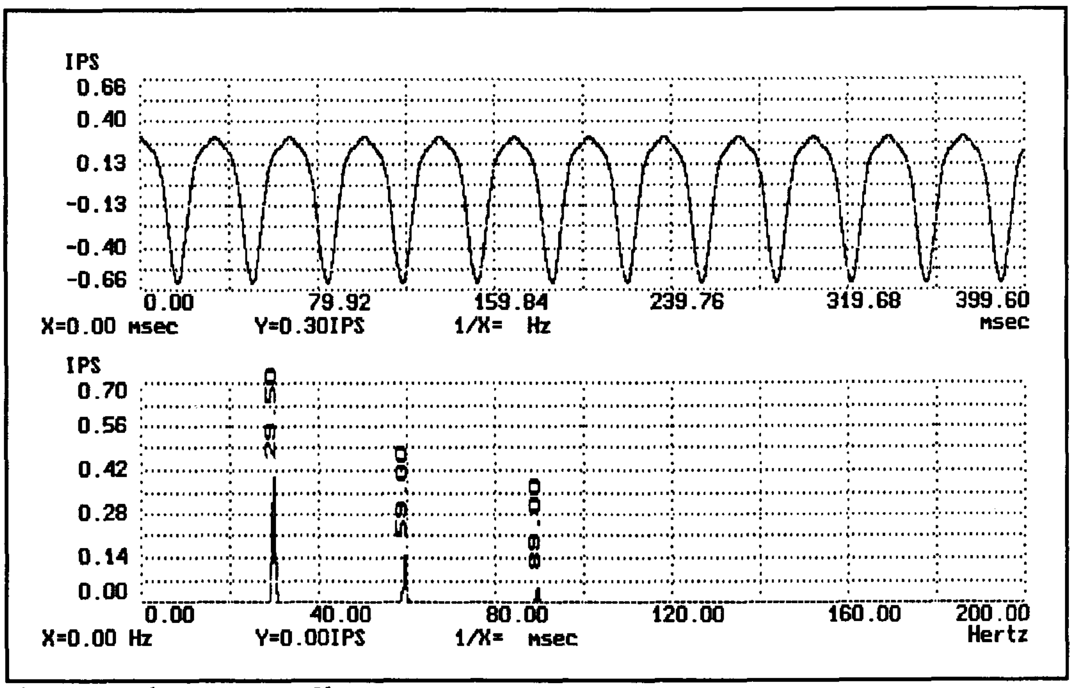


Fig. 4-8. 12 dB Per Octave Slope.

LOOSENESS

Looseness can and does take many forms. The various forms are created by the type of looseness (i.e. is the machine or rotating unit loose?), the amount of looseness, and other associated problems such as imbalance, misalignment, defective bearings, etc.

Bearings Loose on the Shaft

When a bearing is loose on the shaft, the type of signal generated depends upon how the unit is installed. If the inner race is turning on the shaft in an electric motor that is belt driving another unit, the frequency spectra may appear as imbalance with a high amplitude spectral line at motor speed, as in Fig. 4-3. However, the time signal may be distorted in some manner. The time signal may vary in amplitude, be truncated, contain harmonics, contain a beat, or the time period may be different at each half cycle. Some of these characteristics may produce harmonics and other frequencies, such as the speed of the driven unit.

On direct coupled units, if the bearing is turning on the shaft, the frequency spectra may contain a spectral line at unit speed and another spectral line a little lower than unit speed. The lower spectral line is the speed at which the inner race is turning. The difference frequency or delta F is equal to the unit speed or how fast the bearing is turning relative to the shaft, as in Fig. 4-9. The time signal will contain a beat caused by the two frequencies going in and out of phase with each other.

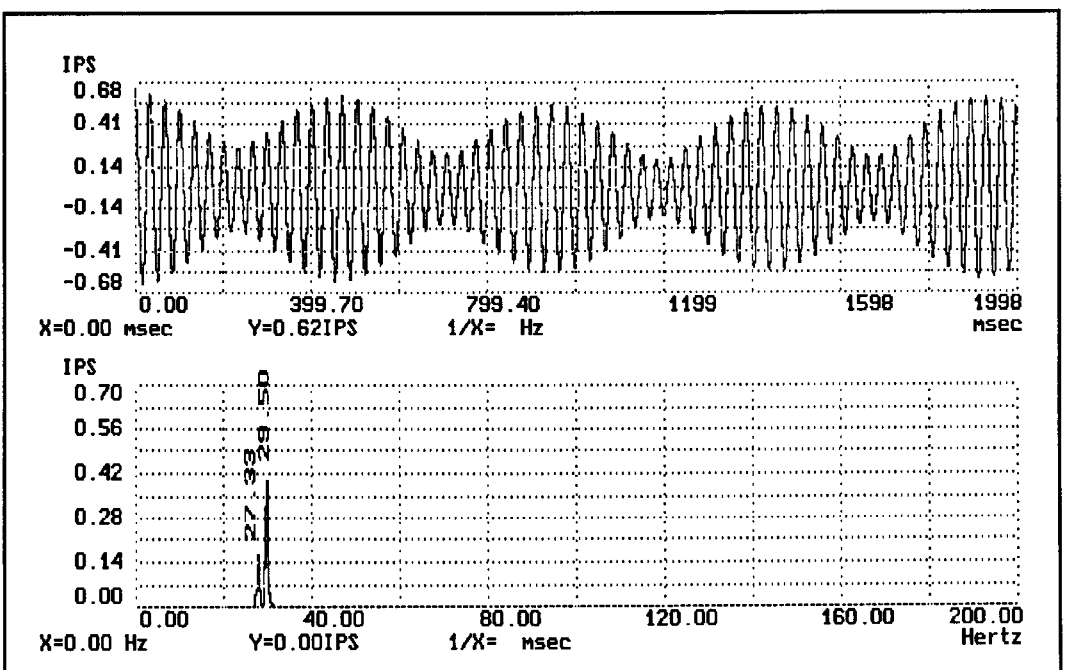


Fig. 4-9. Delta F Equals the Speed of the Bearing Turning Relative to the Shaft.

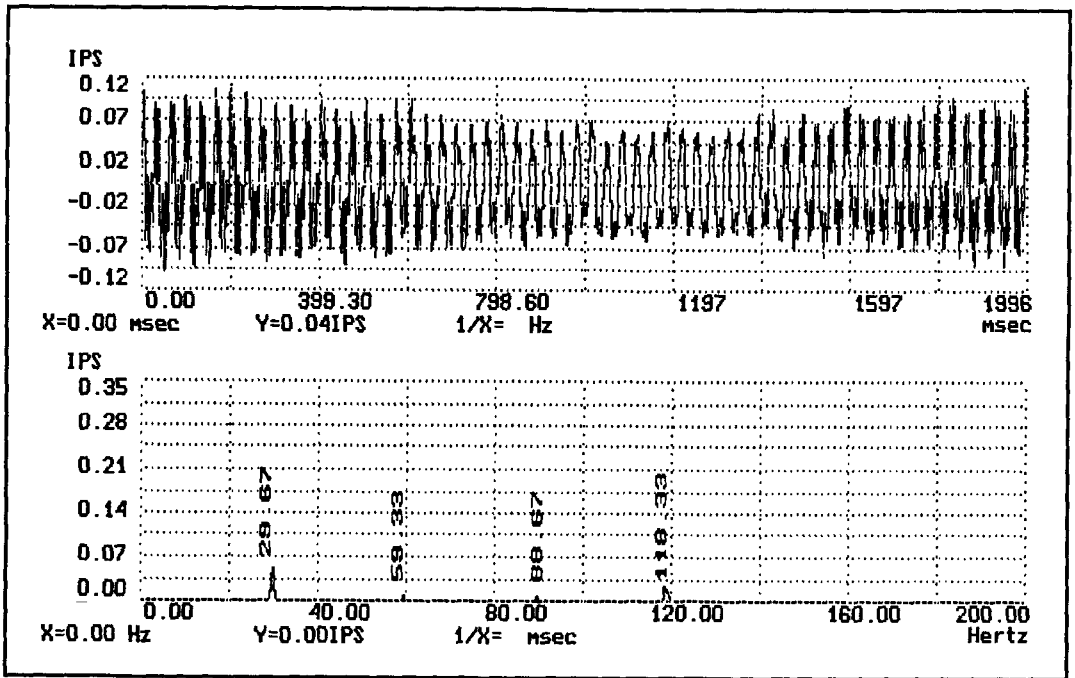


Fig. 4-10. Data from a Motor Running Solo.

Bearings Loose in the Housing

When the fourth harmonic of rotor speed is distinctive, the bearing may be loose in the housing. The data in Fig. 4-10 was taken from a motor running solo. Note the overall amplitude is only 0.1 IPS, and a serious problem exists. The fourth harmonic is distinctive, and the second and third harmonics are present. These harmonics should not be visible with prescribed calibration levels. Caution should be used in diagnosing bearings that are loose in the housing when a pump has four vanes on the impeller because the fourth harmonic may be caused by vane pass frequency. Some press rolls may also contain the fourth harmonic of speed. This can occur when the roll has four

high places. There are many more examples, i.e. screws that have four flights, etc. The only way to determine which problem exists is to look at the phase relationship between the fourth harmonic and the fundamental. If the fourth harmonic is out of phase or is changing phase, the bearing may be loose in the housing. If the fourth harmonic is in phase and maintains a constant phase relationship, the bearing may not be loose in the housing.

Another indication that the bearing is loose on the shaft or loose in the housing is careful analysis of bearing frequencies, particularly ball pass frequencies of the outer and inner races. When calculating bearing frequencies, the rotating unit speed is used. The normal assumption is that the bearing is rotating at the same speed as the shaft, and the fixed race is not rotating. These assumptions are wrong in many cases. The bearing frequencies are actually determined by the relative speed between the inner and outer races. If either race is turning on the shaft or is loose in the housing, the bearing frequencies will be less than those calculated. Once again, care must be used because if the contact angle is increased, the ball pass frequency of the inner race will decrease.

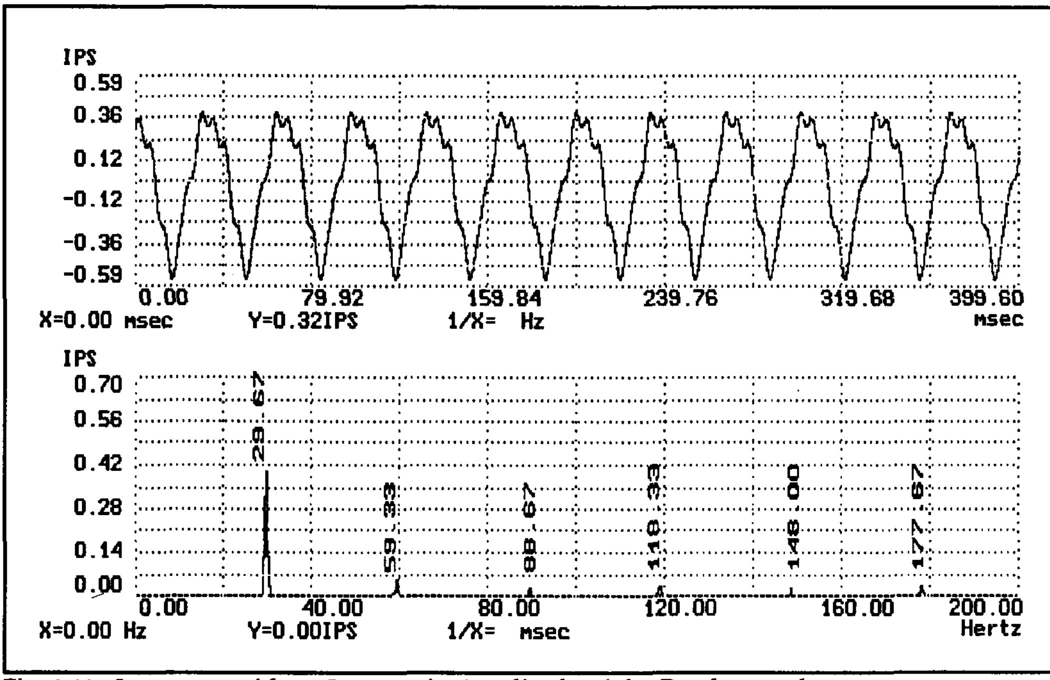


Fig. 4-11. Looseness with an Increase in Amplitude of the Fundamental.

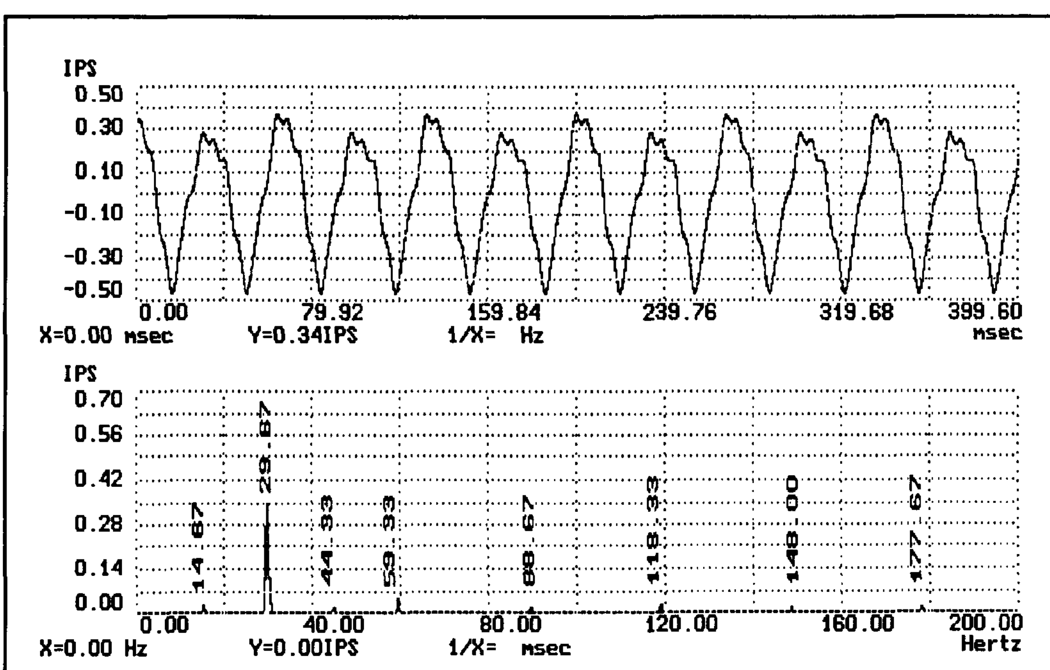


Fig. 4-12. Amplitude of Every Other Revolution is Higher.

Careful analysis of bearing frequencies can identify bearings that are loose on the shaft or in the housing.

Common Forms of Looseness

The more common forms of looseness could start with an increase in the amplitude of the fundamental and the harmonic content, as in Fig. 4-11. Many unskilled analysts try to solve these problems by balancing. Such efforts may not be successful. As the looseness increases, the harmonic content also increases, and the amplitude of the fundamental and the overall RMS value can actually decrease. The next stage of looseness could be the appearance of spectral lines at fractional shaft speed. These spectral lines could occur at 1/2, 1/3, 1/4, shaft speed, etc., depending upon the type and amount of looseness. The author has observed 1/7 shaft speed when the spacer and outer race were switched on a cylindrical roller bearing. Fractional shaft speed is often generated when the rotor does something different every X revolution. One example is when the amplitude of every other revolution is higher, as in Fig. 4-12. If an event such as a pulse is occurring every third revolution, then 1/3 shaft speed will be present. These events may not occur each and every revolution. However, they must occur several times during the averaging process to generate spectral lines at fractional shaft speed. The amplitudes of these spectral lines can be quite low because they may not be present in every time period.

Noise

The last stage of looseness is often noise. The noise can be wide-banded "white noise" or narrow-banded "pink noise." Noise contains all frequencies in a defined bandwidth. This machine is moving in an unpredictable manner and is generating all frequencies in the

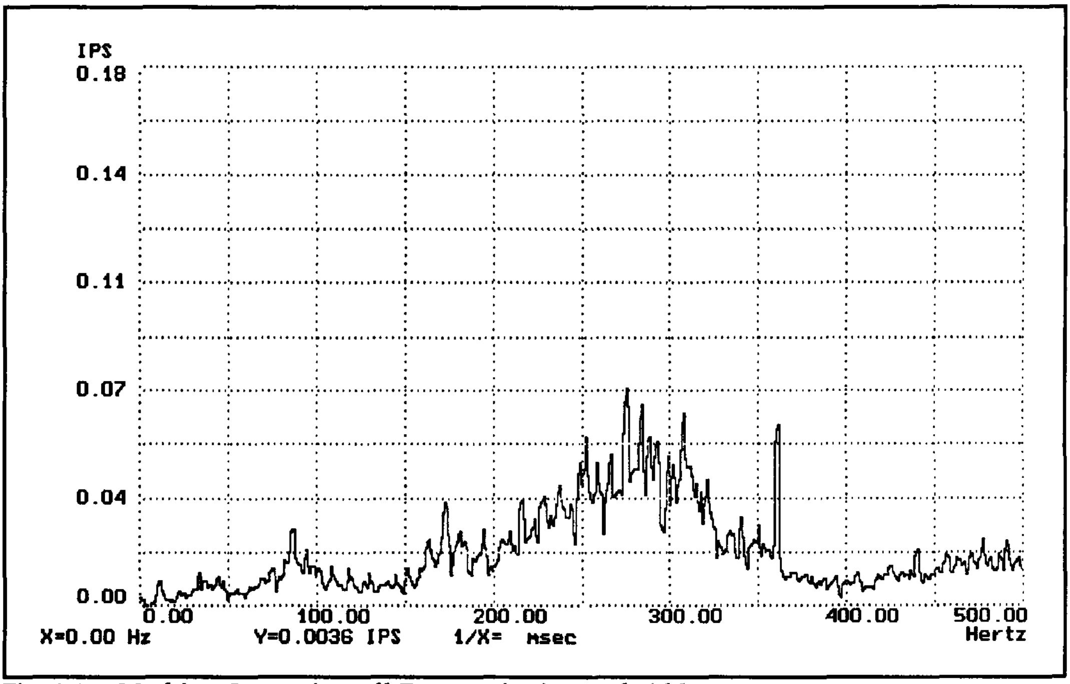


Fig. 4-13. Machine Generating All Frequencies in Bandwidth.

bandwidth, as in Fig. 4-13. Severe looseness is the only problem that can generate such frequencies except for cavitation in pumps. When cavitation occurs, vane pass frequency is generated, and the spectral line at vane pass frequency is normally on top of wide-banded noise, as in Fig. 4-41. Various operational parameters will verify the cavitation problem. Some machines fail before noise occurs. When noise occurs, the machine can and normally does fail in a short period of time. The time period could be hours, days, weeks, or months in some less severe cases, but not years.

Diagnosis of Looseness

The earlier a machine is diagnosed, the more precise and accurate is the diagnosis. The later the diagnosis is made, the more general and less accurate is the diagnosis. For example, early stages of looseness, misalignment, bearing defects, and gear problems can be accurately diagnosed. In the later stages, only wide-banded noise may be present, and the diagnosis may be looseness when, in fact, there are bent shaft, misalignment, and defective bearings present. This is why diagnostics of rotating machinery may not always identify all problems. Accurate diagnosis will always identify the most severe problem, which is often looseness.

RESONANCE

Resonance is a very complex subject. It is beyond the scope of this book to provide all the mathematical descriptions. However, a practical explanation is provided.

All things have one or more resonant frequencies. These frequencies are often called modes, i.e. first mode, second mode, etc., as explained in Chapter One. Resonant frequencies are often called natural frequencies and critical speeds. Natural frequency may be a better term because it refers to the natural frequency at which the machine is inclined to vibrate. The term "critical speed" is used when a resonant/natural frequency equals the unit speed. When this occurs, the machine may have exceptionally high vibration levels.

A resonant frequency is often referred to as though it were a single frequency. However, in actuality, a resonant frequency is often a band of frequencies. The bandwidth is determined by damping. A resonant frequency can be well-damped, which means it will be low in amplitude and wide-banded. A resonant frequency can also be undamped. This means the amplitude will be high and it will be narrow-banded. The resonant frequency can be anywhere between well-damped and undamped. Resonant frequencies that are well-damped contain low vibration levels. Undamped resonant frequencies contain high vibration levels. A measure of how violent the resonance may be is called the amplification factor, which was discussed in Chapter One.

Natural frequencies can be determined in one of three ways:

1. The resonant frequency can be calculated or the **Resonance and Deflection Calculator Program** can be used. Since we live in an imperfect world, calculated frequencies may not be exact. However, they can be very close.

2. The resonant frequency can be determined with a "bump test." Under this test, the machine or piece is bumped or hit, and the resonance is measured. For example, if you strike a crystal glass with a fork, it will ring or vibrate at its resonant frequency. The energy put into the glass by the fork excites the glass into vibrating at the resonant frequency.

3. If a machine is operating above the first critical speed, coast down and startup data can be used to identify the resonant frequency, bandwidth, and amplification factors. Coast down data can be taken only from machines that will coast. Startup data can only be used to identify resonance if the machine can be started slowly.

As can be seen from the above list, all problems cannot be identified from bearing cap vibration while the machine is running. Special tests must be performed in some cases. These special tests are explained in the last section of this chapter.

RUBS

Some people think of rubs as surface-to-surface contact over long time periods, causing much friction and heat. Some rubs can occur in this manner. The signal generated is wide-banded noise, and the result is a sudden and catastrophic failure.

A more common form of rub is when the rotating unit hits something once or many times each revolution. When one hit occurs per revolution, a pulse is often generated. The reciprocal of the time period between the pulses is unit speed.

A rub can also occur at a specific multiple of rotating speed. For example, if each blade on a motor cooling fan is hitting something, the frequency generated can be the number of blades on the fan times the speed of the fan. The spectral line at blade pass frequency can have plus and minus sidebands at unit speed, depending on the characteristics of the time signal. If the time signal contains pulses, amplitude, and/or frequency modulation, these sidebands will be present.

A rub can also occur when the surface of the rotating unit hits another object a number of times each revolution. This type of rub can generate a series of spectral lines, and the difference frequency between the spectral lines will be equal to unit speed. The center frequency could be equal to the number of hits times speed, or an excited natural/resonant frequency. See Fig. 4-14. If two series of spectral lines occur, then two resonant frequencies are being excited.

PROBLEMS THAT CAUSE PULSES

A pulse is caused by a hit or an impact. Four measurable characteristics of a pulse are:

1. Repetition rate

2. Amplitude

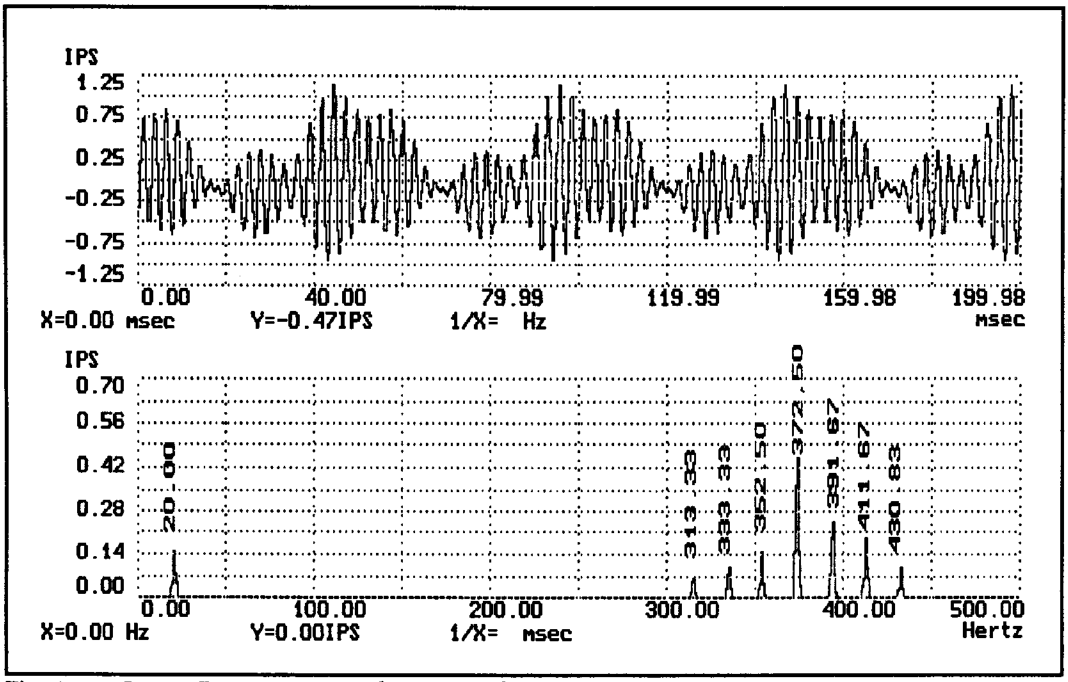


Fig. 4-14. Center Frequency Equal to Natural Frequency.

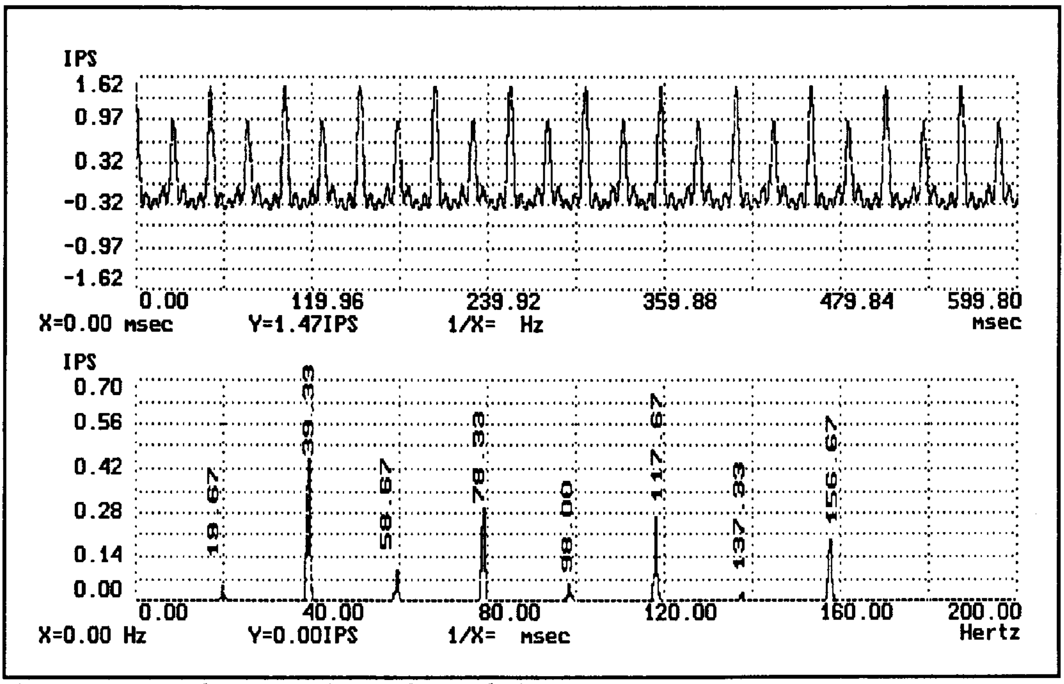


Fig. 4-15. Two Pulses Occurring Each Revolution.

3. Frequency

4. Width

The repetition rate identifies how often the pulse is occurring. The repetition rate is normally unit speed. When more than one pulse is occurring each revolution, the pulses can appear random in the time domain. However, the frequency spectrum will have a series of spectral lines, and the difference frequency will be unit speed. If two pulses are occurring each revolution, every other spectral line will be higher in amplitude, as in Fig. 4-15. If four pulses are occurring each revolution, every fourth spectral line can be higher in amplitude, etc. Pulses can also occur at other generated frequencies, such as ball pass frequencies of the inner and outer races, or gearmesh, vane, and blade pass frequencies. Pulses can occur randomly. When random pulses occur, the frequency spectrum contains noise, the rotating unit is flopping and banging around in an unpredictable manner, and failure is imminent.

Amplitude is affected by problem severity, resonance, damping, and loading. Pulse amplitude in the time domain can be very low or very high, depending on the aforementioned points. In the frequency domain, the spectral lines range from very low amplitude to nonvisible because of the low energy or lack of energy between the pulses. For this reason, pulses are often overlooked if the time domain signal is not analyzed. If a pulse is suspected, data taken with an accelerometer often yields a more pronounced pulse. This is true because an accelerometer measures rate of change per time period, and pulses often have a fast rise time.

A pulse can be empty, which can mean it does not contain a frequency. An empty pulse occurs when a natural frequency is not excited in amplitude, or the impact is measured. In this case, the fundamental in the frequency spectrum will be low or nonvisible. The second, third, fourth, fifth, etc., harmonic can be higher in amplitude, as in Fig. 4-16. When an impact excites a natural frequency, the pulse will contain the excited frequency, and a series of spectral lines will be around the excited frequency. Once again, the amplitude will be quite low if the signal is damped because there will be little or no energy between the pulses.

When a pulse is empty, the pulse width will be narrow and only half a cycle wide. If a natural frequency is excited, the pulse width will be determined by the damping. Fig. 4-17 contains a well-damped time signal and frequency spectrum. Fig 4-18 contains a relatively undamped time signal and frequency spectrum.

Once the above pulse parameters have been identified, it may be difficult, in some cases, to identify the specific problem. This is true because more than one problem can cause a pulse at unit speed. Following are some examples:

1. When coupling halves are set too close together, (they can hit once each revolution)

2. A broken, cracked, or chipped tooth on a gear

3. A circumferential crack in a roll

4. Steam joints

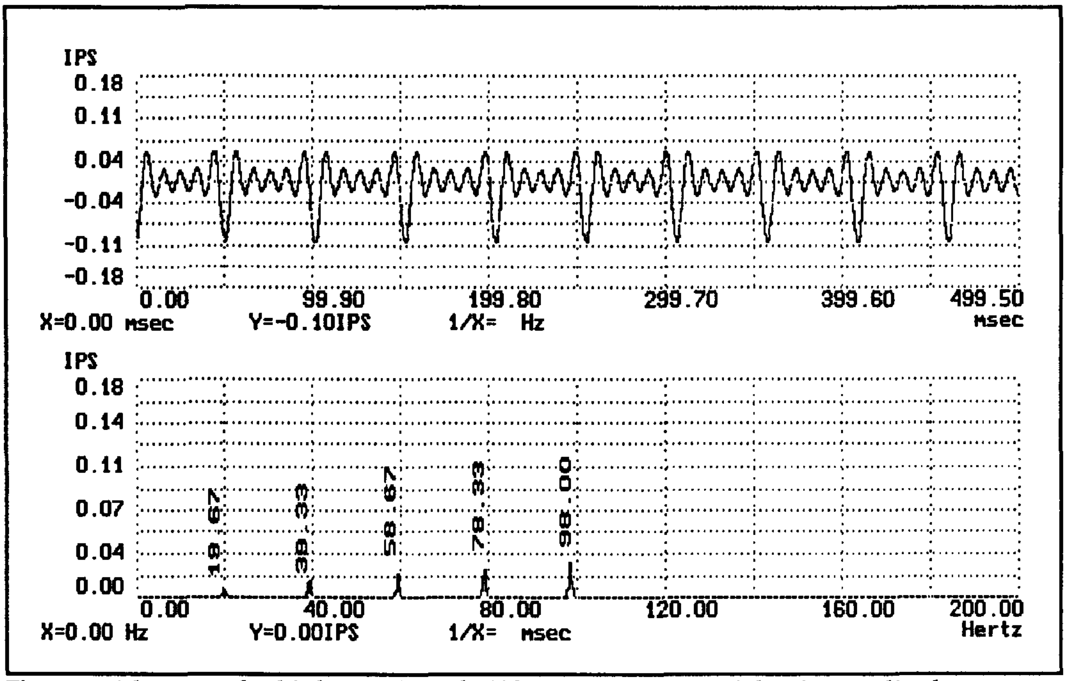


Fig. 4-16. The Second, Third, Fourth, and Fifth Harmonics Are Higher in Amplitude.

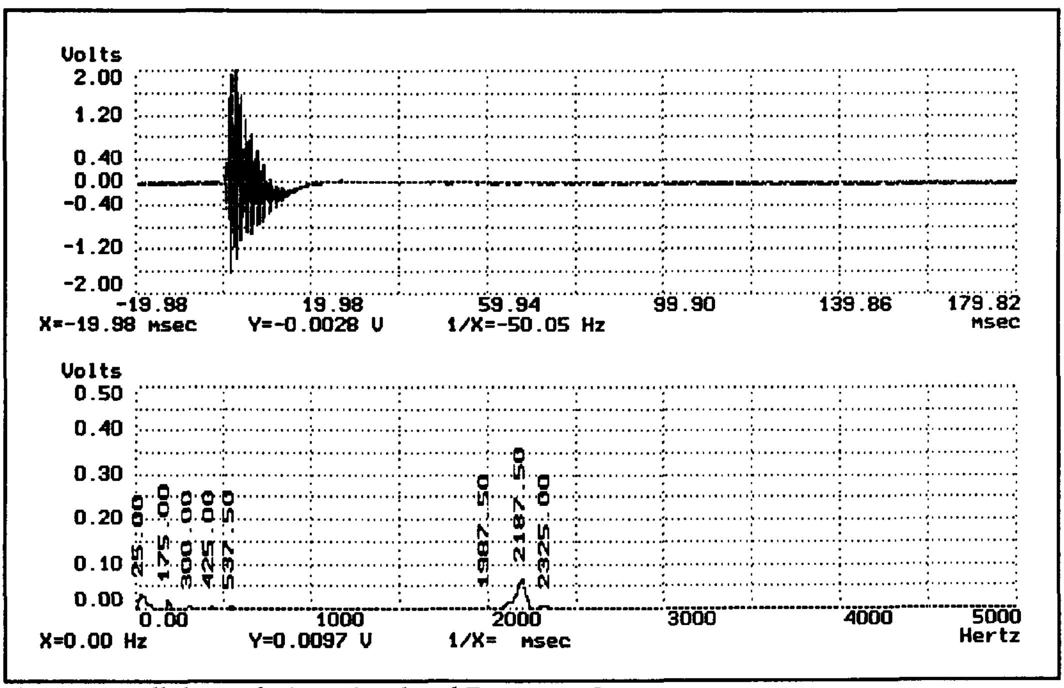


Fig. 4-17. Well-damped Time Signal and Frequency Spectrum.

5. Looseness in bearings

6. A bearing with a cracked inner race

7. Loose headbolts, manhole covers, and syphons on rolls

OIL WHIRL

In some lightly loaded, plain journal bearings, a phenomenon called oil whirl can occur. This is caused when the rotating unit pushes up an oil wedge in front of the shaft. As the shaft tries to roll over the oil wedge, the shaft whirls around the bearing while rotating at a given speed. The general view is that oil whirl frequency varies between 40 to 50 percent of shaft speed. Oil whirl may not occur in tilted pad, pressure dam, and crush fit bearings. This writer agrees with these generally held views. However, additional information can be learned from the time domain signal and frequency domain spectrum. Fig. 4-19 contains the time signal and frequency spectrum from an oil whirl problem. Turbine speed is 93 Hz, and oil whirl frequency is 43 Hz. In most machines that have oil whirl problems, a knocking noise can be heard. This noise is generated because the oil whirl does not occur continually. Rather, the shaft makes a few whirls, stops for a couple of revolutions, then makes a few more whirls. This intermittent whirling can sound like a knock. The time period between points 1 and 2 in Fig. 4-19 is 0.02325 seconds or 43 Hz, which is oil whirl. The time period between points 3 and 4 is 0.01075 seconds or 93 Hz, which is turbine speed. The time period between points 5 and 6 is 0.1428 seconds or 7 Hz. This is the frequency of maximum oil whirl amplitude. There is a spectral line at 50 Hz, which is 43 Hz + 7 Hz. There is also a spectral line at 36 Hz, which is 43 Hz - 7 Hz.

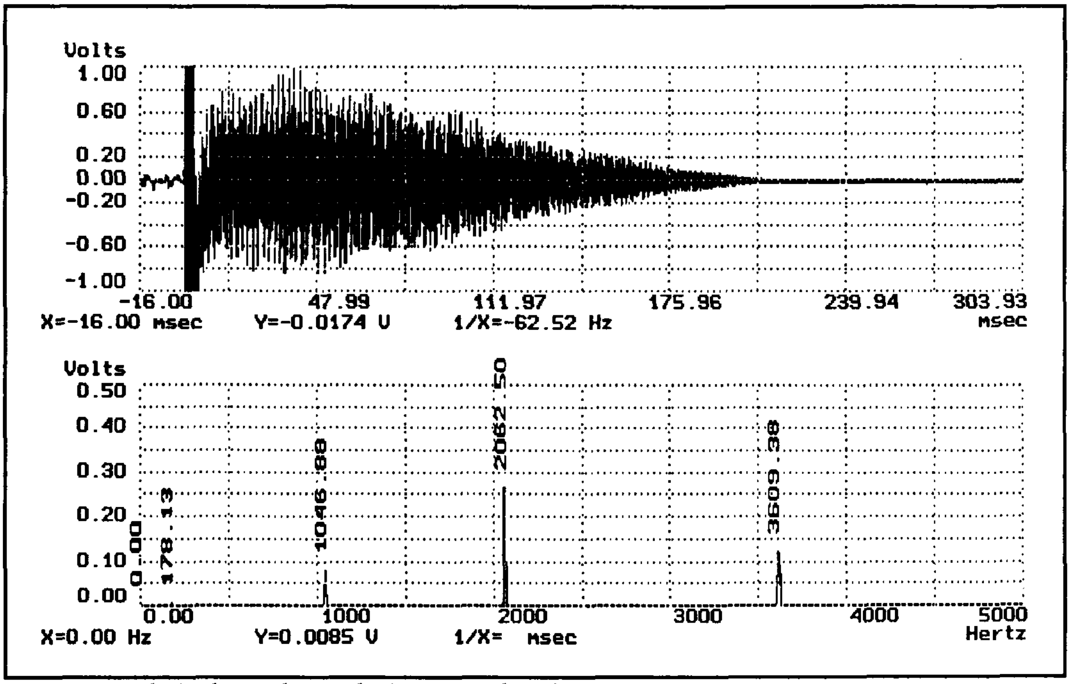


Fig. 4-18. Relatively Undamped Time Signal and Frequency Spectrum.

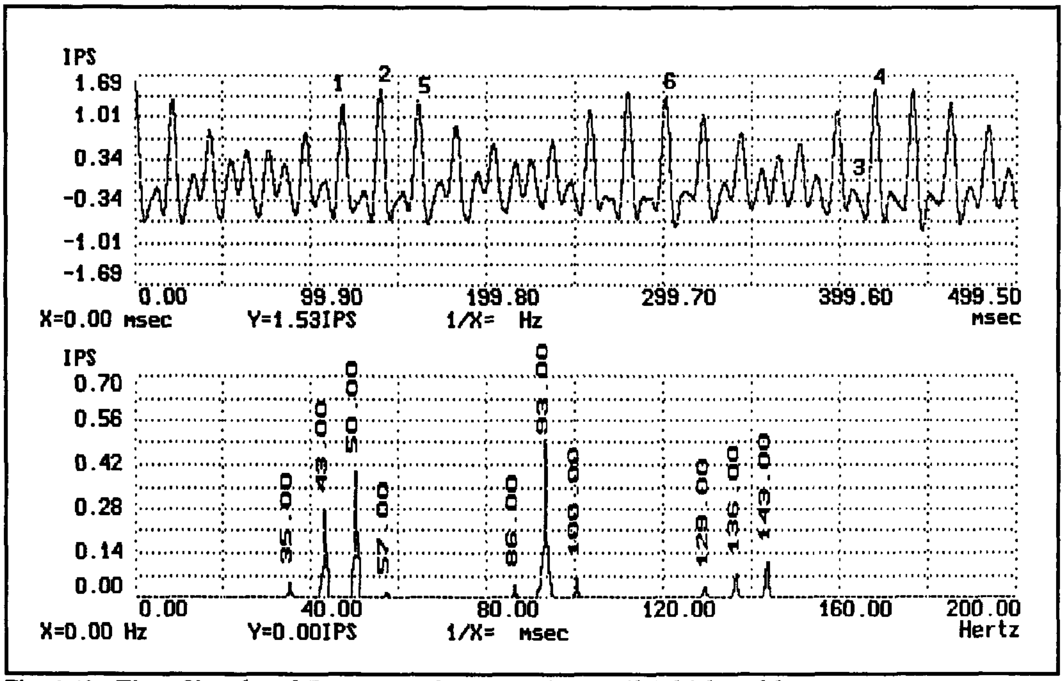


Fig. 4-19. Time Signal and Frequency Spectrum from Oil Whirl Problem.

The spectral line at turbine speed has plus and minus sidebands of 7 Hz, which are 100 Hz and 86 Hz. A spectral line is also located at 136 Hz (43 Hz + 93 Hz), which is the sum of oil whirl frequency and turbine speed. This frequency also has plus and minus sidebands of 7 Hz (129 Hz and 143 Hz).

Note: The spectral line at 50 Hz is a sideband or sum frequency and is higher in amplitude than the discrete frequency at 43 Hz. A sideband cannot be higher than the discrete frequency, except when the sideband equals a resonant frequency or contains more energy than the discrete frequency.

From the above empirical data, it would appear that oil whirl frequency is equal to turbine speed minus 7, divided by 2, or:

$$F_{ow} = \frac{S_T - 7}{2} \quad or \quad \frac{93 - 7}{2} = 43$$

This formula meets the current description that oil whirl is somewhere between 40 and 50 percent of turbine speed. We recognize that some refinements to the formula may be required in the future.

ANALYSIS OF ELECTRIC MOTORS

Electric motors can have a limited number of problems, just as any other equipment group. However, some problems occur only in electric motors.

Electro-mechanical problems in electric motors must be further divided into induction and synchronous motors. The following electro-mechanical problems occur primarily in induction motors:

1. Motor out-of-magnetic center
2. Broken bars in the rotor
3. Turn-to-turn shorts in the stator windings
4. Vibration at line frequency, 50 or 60 cycles
5. Siren effect
6. Overloading

Induction motors are designed to operate at a fixed speed. For example, 900, 1200, 1800, and 3600 RPM are the most common speeds. However, these motors seldom operate at the synchronous speed. Generally, they operate below the synchronous speed. The difference between the synchronous speed and operating speed is called the slip frequency. The exact operating speed is affected by motor efficiency and load. A high efficiency motor operating with little or no load may run at near synchronous speed.

The stator windings are called poles. The number of poles determines the speed of the motor. To calculate the number of poles in a motor, divide 7200 by the synchronous speed of the motor. For example:

$$\textit{A 3600 RPM motor has } \frac{7200}{3600} = 2 \textit{ poles}$$

$$\textit{An 1800 RPM motor has } \frac{7200}{1800} = 4 \textit{ poles}$$

$$\textit{A 1200 RPM motor has } \frac{7200}{1200} = 6 \textit{ poles}$$

$$\textit{A 900 RPM motor has } \frac{7200}{900} = 8 \textit{ poles}$$

Frequency is equal to events times speed. When some of the above problems occur, the motor speed and harmonics are modulated by slip frequency times the number of poles. These frequencies normally do not appear as discrete frequencies. Rather, they appear as sidebands or difference frequencies around motor speed and harmonics. The data from motors out-of-magnetic center, broken or open rotor bars, or turn-to-turn shorts in the windings are similar, and detailed analysis is required for accurate diagnosis.

<u>Motors Out-of-Magnetic Center</u>

This condition normally occurs when the rotor is not positioned in the magnetic center of the stator in the axial or lateral direction. An out-of-magnetic center condition could occur in the radial direction when there is an uneven air gap between the rotor and stator. This problem may not generate excessive vibration levels and is often overlooked because the data is not processed with enough resolution to identify the problem. For example,

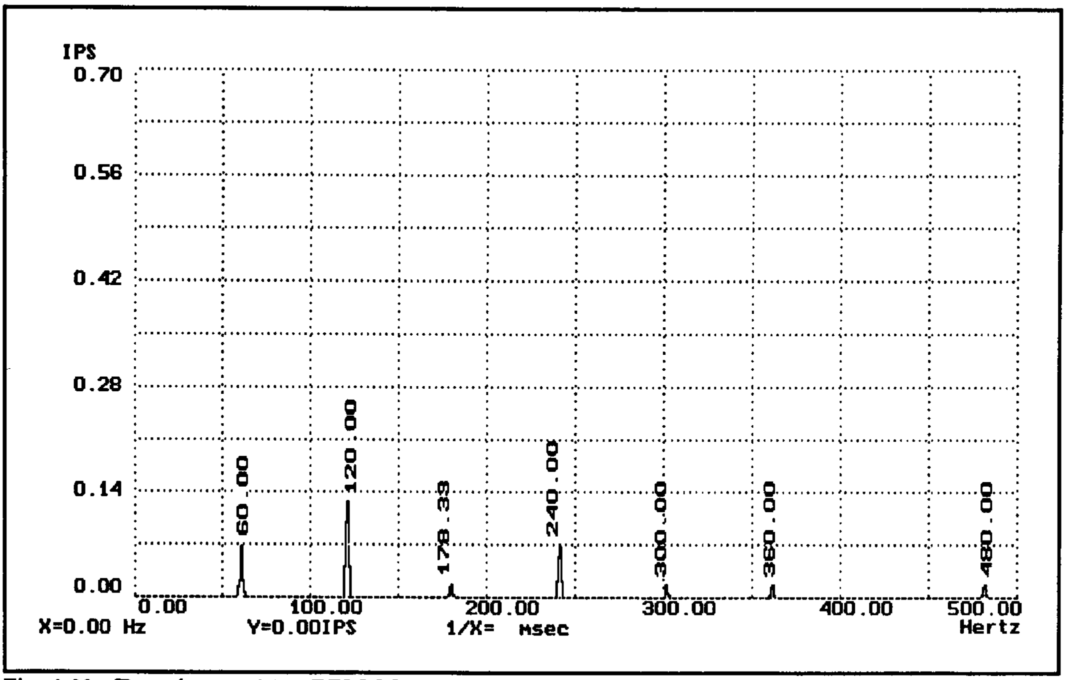


Fig. 4-20. Data from a 3600 RPM Motor.

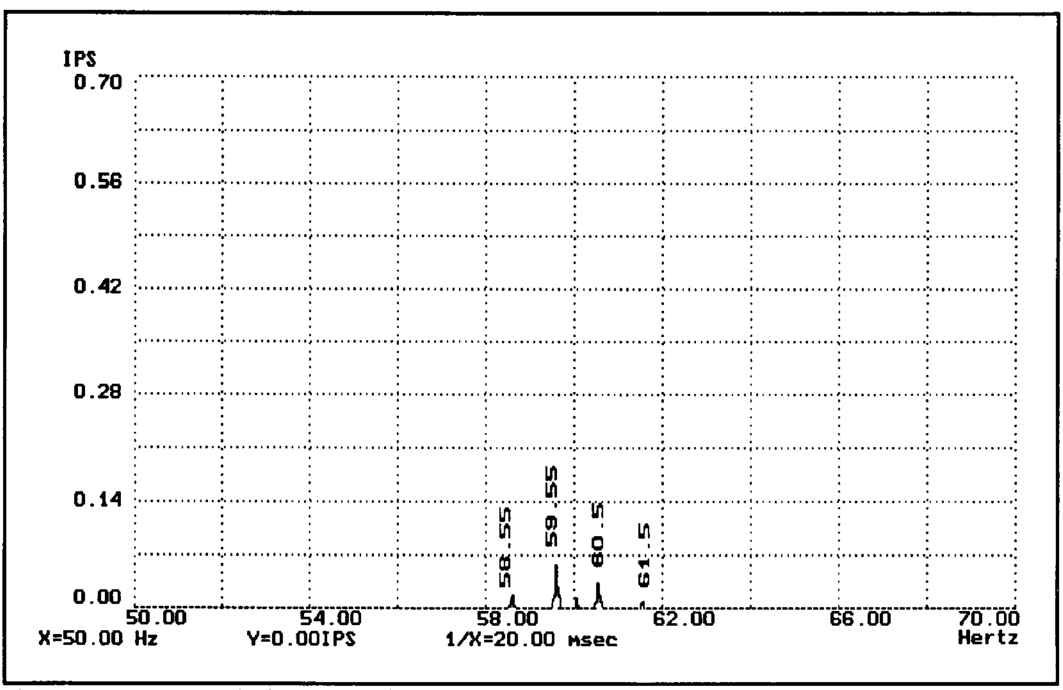


Fig. 4-21. A 20 Hz Window around 60 Hz with an Out-of-Magnetic Center Motor.

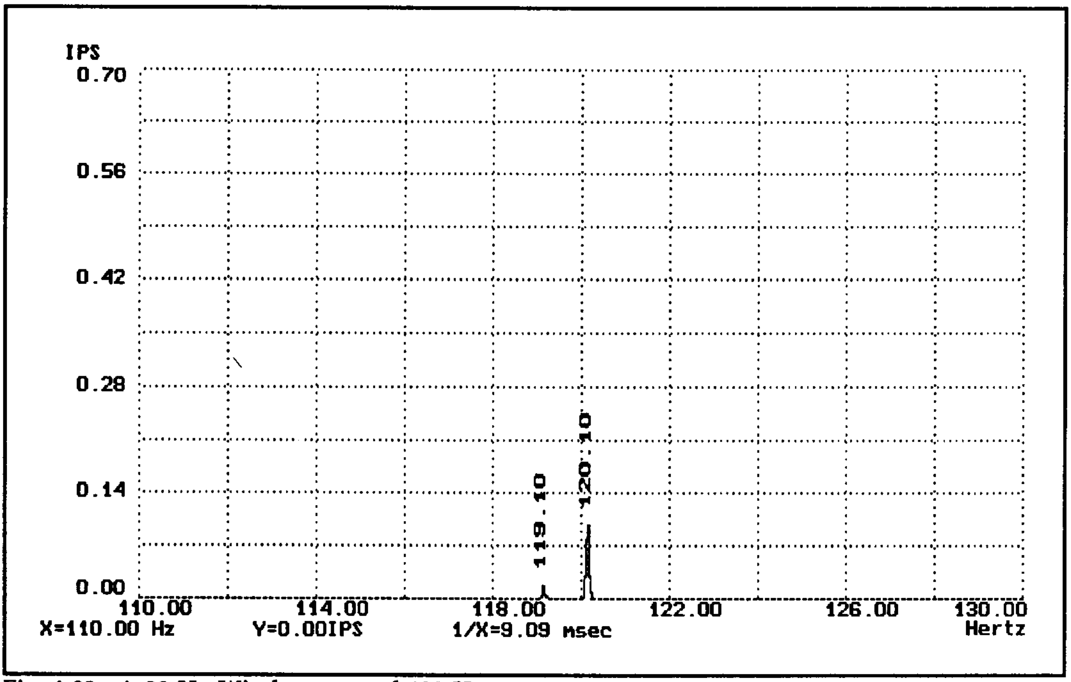


Fig. 4-22. A 20 Hz Window around 120 Hz.

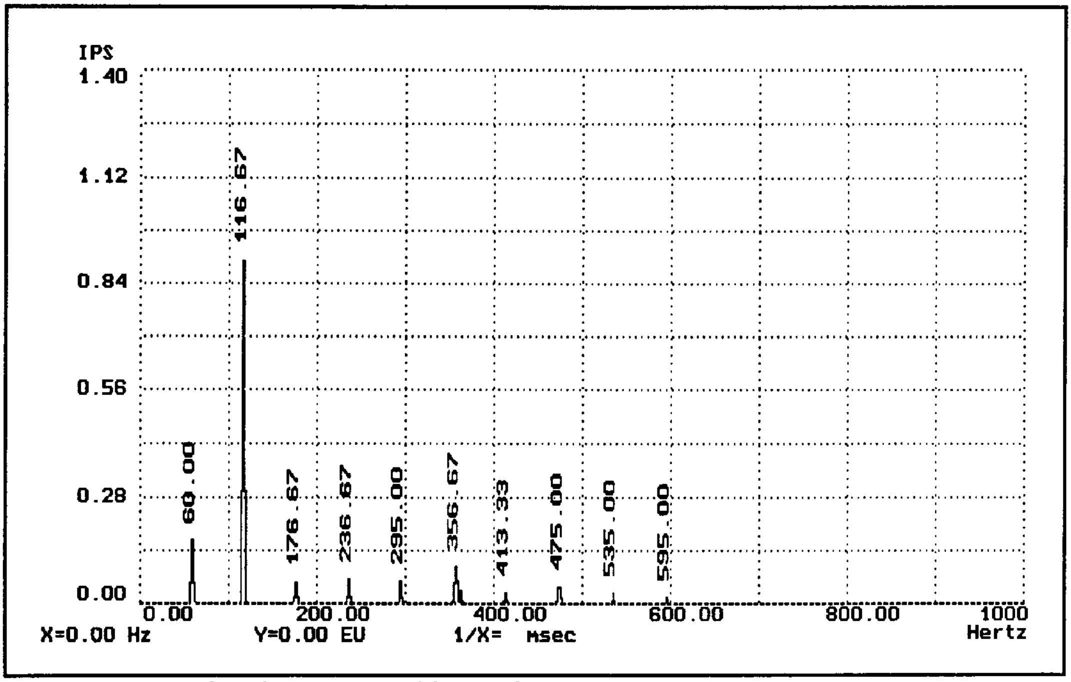


Fig. 4-23. Data Taken from Motor with a Broken Rotor Bar.

the data in Fig. 4-20 was taken from a 3600 RPM motor. This data could be erroneously diagnosed as looseness, misalignment, or even bent shaft because more resolution is required for accurate diagnosis. Fig. 4-21 contains a 20 Hz window around 60 Hz. This data indicates that the motor is out of magnetic center. This data contains spectral lines with a line frequency of 60 Hz and a motor speed of 59.5 Hz. Motor speed has sidebands at a slip frequency of 0.5 Hz times the number of poles (2), or 0.5 Hz x 2 = 1 Hz. The sidebands are 58.5, 60.5, and 61.5 Hz. **Please note:** Line frequency does not have sidebands. Fig. 4-22 contains a 20 Hz window around 120 Hz. This data has enough resolution to verify the second harmonic of motor speed of 119 Hz, which is not a problem. The predominate frequency is 120 Hz, which is two times line frequency.

The beat is available in 1800 and 3600 RPM motors with an out-of-magnetic center condition. Slower motors (1200 RPM and lower) may not be affected as much by out-of-magnetic center problems because higher harmonics of motor speed are near 60 Hz. The 60 Hz frequency may appear as a discrete frequency when these motors have an out-of-magnetic center condition.

Broken Rotor Bars

This problem is similar to the out-of-magnetic center problem. However, there are distinctive differences. The similarity is that rotor speed is modulated by slip frequency times the number of poles. A broken rotor bar, sometimes called open iron, creates a dead spot in the rotor. The resulting electrical imbalance can generate significant vibration levels at one and two times RPM. Since this problem is not interactive with line frequency, a spectral line at 60 Hz may not be present. Once again, the data must be processed with enough resolution for accurate diagnosis. Fig. 4-23 contains data taken from a motor that has one or more broken rotor bars. On the 1,000 Hz range, the

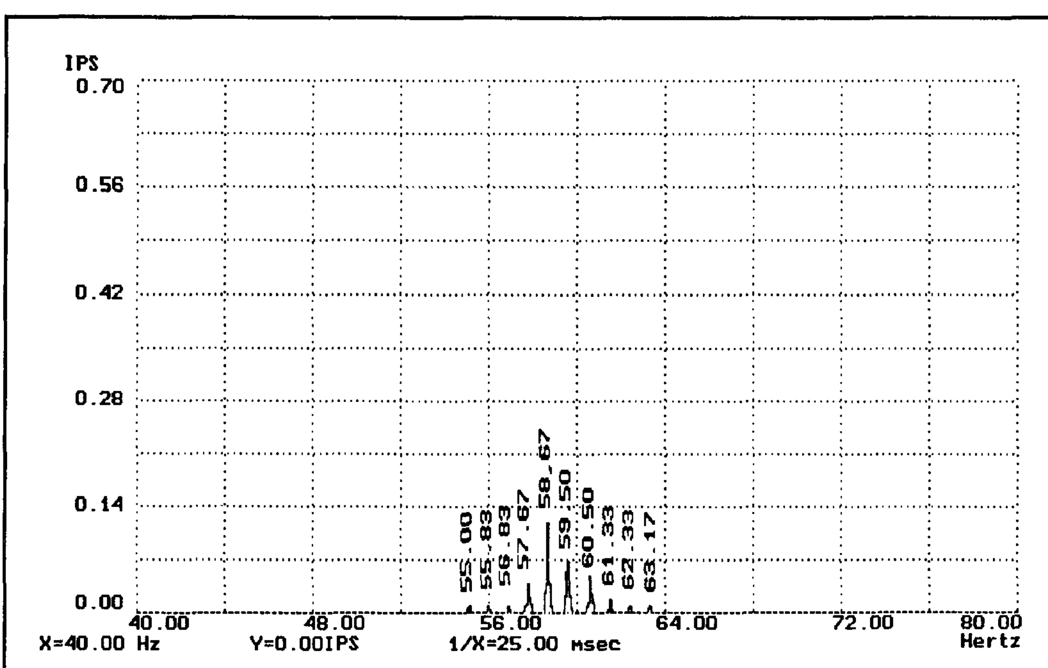


Fig. 4-24. 40 Hz Window around 60 Hz.

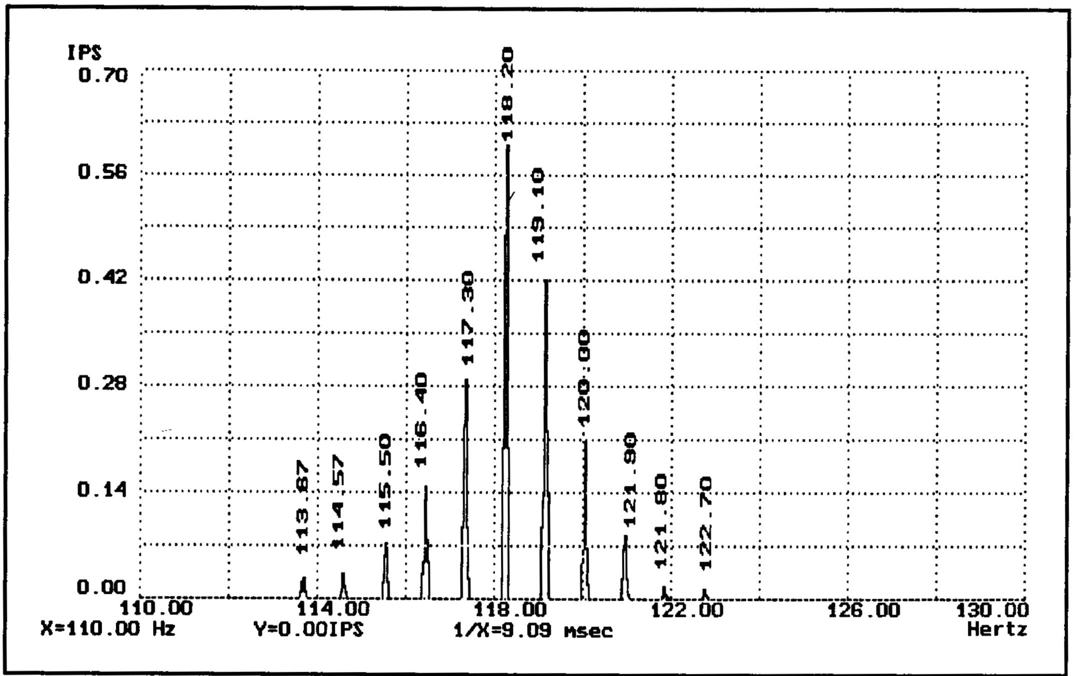


Fig. 4-25. 20 Hz Window around 120 Hz.

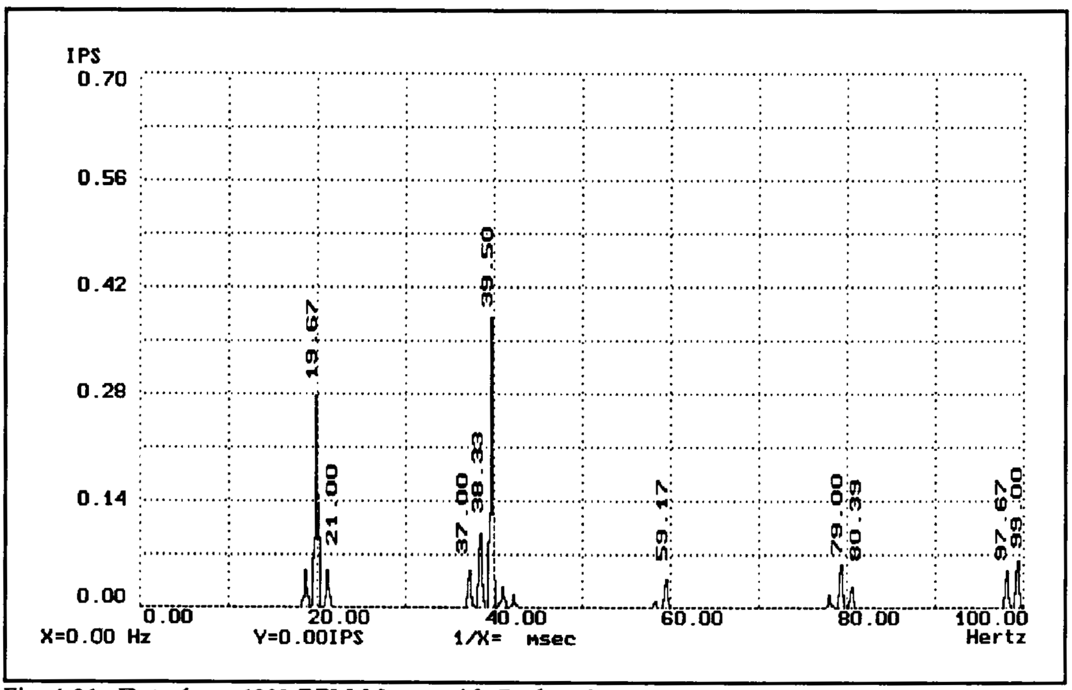


Fig. 4-26. Data from 1200 RPM Motor with Broken Rotor Bars.

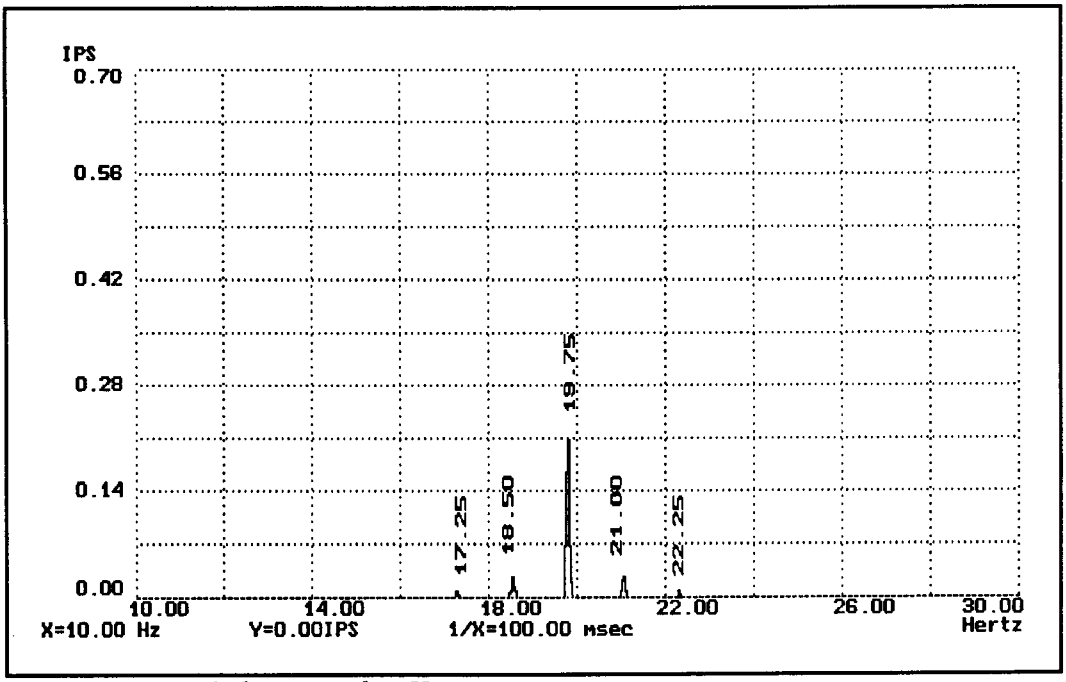


Fig. 4-27. 20 Hz Window around 20 Hz.

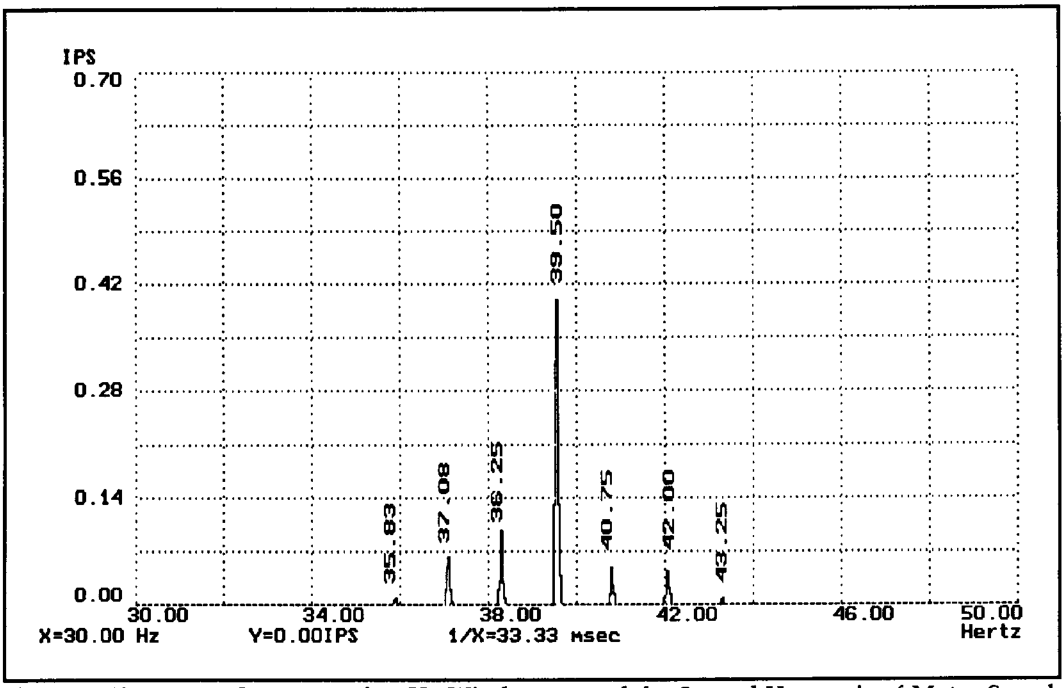


Fig. 4-28. Frequency Spectrum of 20 Hz Window around the Second Harmonic of Motor Speed.

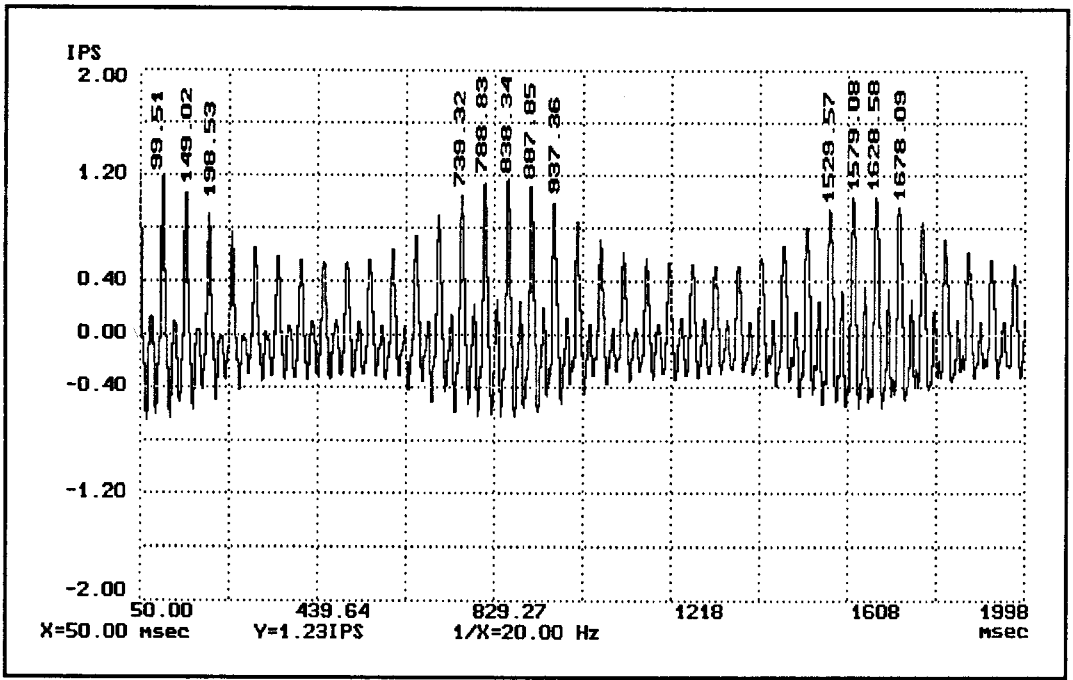


Fig. 4-29. Time Domain Signal from 1200 RPM Motor with Broken Rotor Bars.

diagnosis could be looseness and bent shaft. Fig. 4-24 contains a 40 Hz window around 60 Hz. Motor speed is 59.5 Hz, and plus and minus sidebands of 0.9 Hz are present. Slip frequency is 0.45 Hz x 2 poles = 0.9 Hz (within measurement accuracy). The sideband at 58.6 Hz is higher in amplitude than motor speed (59.5 Hz). This occurs when the amplitude of the modulator is greater than the carrier. In such cases, phase reversal occurs and can be observed in the time signal. This could indicate more than one broken rotor bar. Fig. 4-25 is a 20 Hz window around 120 Hz. Two times motor speed is 119.1 Hz, and sidebands 0.9 Hz apart are present. Also note that the sideband at 118.2 Hz has an amplitude greater than the second harmonic of motor speed. The above explanation also applies.

An example of broken rotor bars from a motor at a different speed may provide some clarity. Fig. 4-26 contains data from a 1200 RPM motor that has broken rotor bars. The vibration amplitude is quite high and the sidebands are visible on the 100 Hz range. Fig. 4-27 contains a 20 Hz window around 20 Hz. The sidebands are 1.25 Hz apart. The delta F is equal to slip frequency times the number of poles, or 0.21 Hz x 6 = 1.26 Hz. Fig. 4-28 contains a 20 Hz window around the second harmonic of motor speed. The sidebands are slip frequency times number of poles. In this case, the amplitude of the spectral lines at motor speed and the second harmonic are higher than the sidebands. Fig. 4-29 contains the time domain signal. The time period between points 1 and 2 is motor speed. The time period between points 3 and 4 is two times motor speed, and the time period between points 5 and 6 is slip frequency times number of poles.

Fig. 4-30 is a photo of a rotor with a broken rotor bar.

Fig. 4-30. Rotor with Broken Rotor Bar.

Turn-To-Turn Shorts In Windings

Motor stator windings are often called poles. When some wires in the winding/pole are shorted together, the motor often slows down. When the motor speed is reduced, slip frequency increases. Fig. 4-31 contains the data from an 1800 RPM motor that has turn-to-turn shorts in the windings. Motor speed is 29 Hz x 60 = 1740 RPM, and the sidebands are 4 Hz apart (delta F = 4 Hz). These sidebands are still equal to slip frequency x number of poles, i.e. 1 Hz X 4 = 4 Hz. Fig. 4-32 contains the time signal. Motor speed, slip frequency, and number of poles are labelled.

The above descriptions of induction motor problems can be summarized as follows:

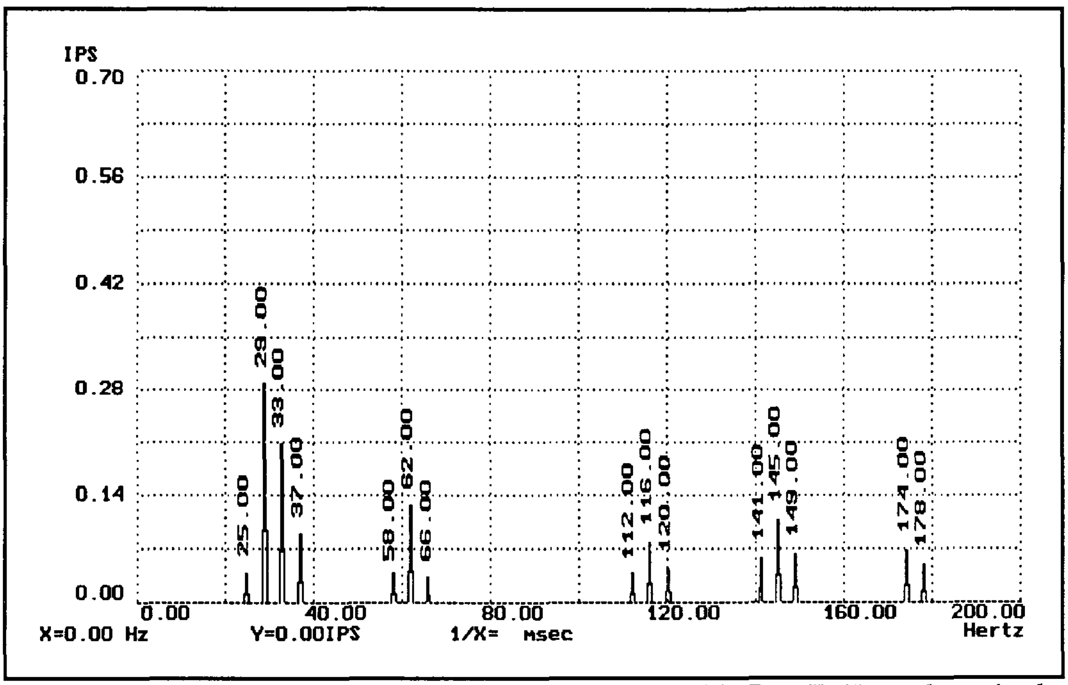


Fig. 4-31. Frequency Spectrum from an 1800 RPM Motor with Turn-To-Turn Shorts in the Windings.

1. Motors out-of-magnetic center with broken rotor bars or turn-to-turn shorts all contain sidebands of slip frequency times number of poles around motor speed and harmonics.

2. Motors out-of-magnetic center normally contain a spectral line at 60 Hz.

3. Motors that have broken rotor bars normally have high amplitudes at motor speed and/or the second harmonic.

4. Motors that have turn-to-turn shorts in the windings have slower speeds and higher slip frequencies.

Vibration Problems in Synchronous Motors

Half-wave rectification of a 60-Hz sine wave yields a ripple frequency of 60 Hz. Full-wave rectification yields a ripple frequency of 120 Hz. Half-wave rectification of one phase and full-wave rectification of another phase yields a ripple frequency of 180 Hz. Full-wave rectification of three phases yields a ripple frequency of 360 Hz. Some synchronous motors have power supplies that use diodes to rectify AC voltage and obtain the desired power for a desired speed. These motors can vibrate at significant levels when a problem occurs in the power supply or the controlling circuit board. The frequency of these electrical vibrations in synchronous motors can occur at 60 Hz and harmonics, 120 Hz and harmonics, 180 Hz and harmonics, and/or 360 Hz and harmonics. Fig. 4-33 contains data from a synchronous motor with an electrical problem. This problem was solved by replacing a circuit board in the power supply.

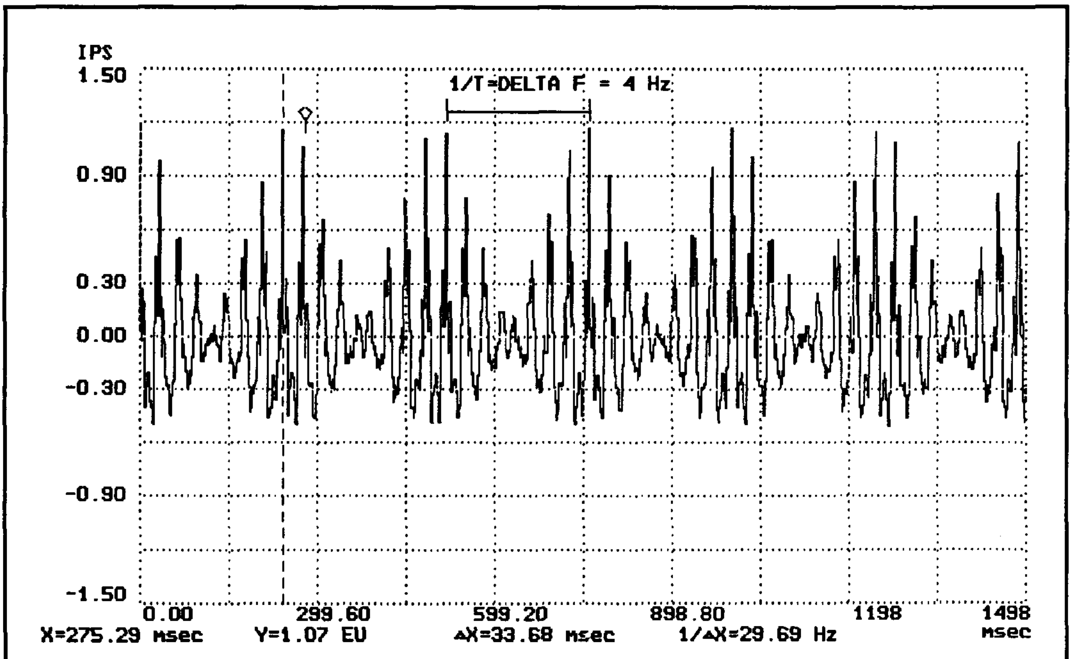


Fig. 4-32. Time Domain Signal from an 1800 RPM Motor with Turn-to-Turn Shorts in the Windings.

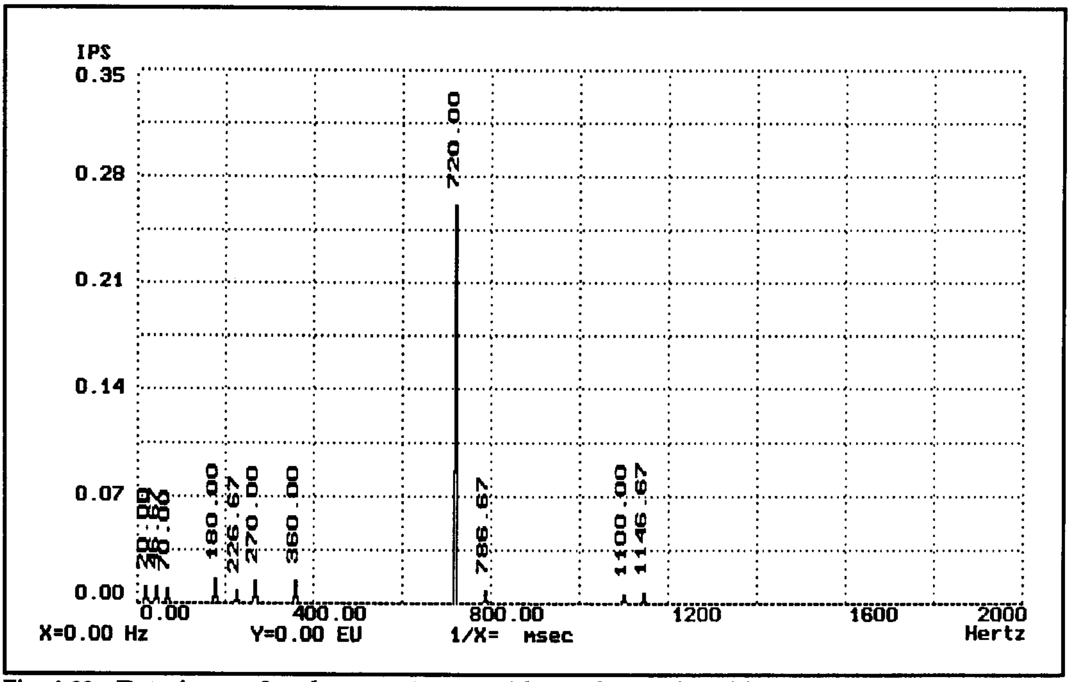


Fig. 4-33. Data from a Synchronous Motor with an Electrical Problem.

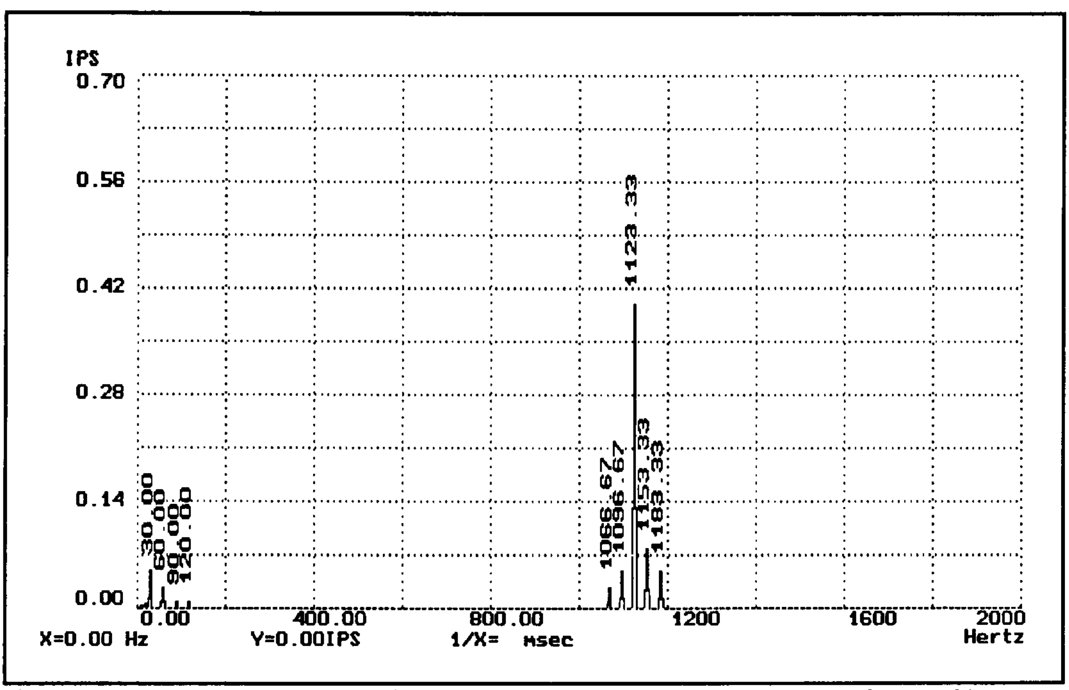


Fig. 4-34. An 1800 RPM Motor with 38 Rotor Bars Generating a Frequency of 1124.8 Hz.

Siren Effect

The siren effect in motors occurs as in any other siren. When a number of slots or grooves pass over other slots or grooves at the correct speed and air gap, a noise or sound pressure level is generated. Since this is a generated frequency, it is equal to events times speed. If a rotor has 38 bars/slots and the speed is 1776 RPM (29.6 Hz), the frequency generated would be 38 x 29.6 Hz = 1124.8 Hz. As these slots pass over one point, the frequency is generated. If the slots pass over several points, as in a motor, the same frequency is generated. However, the sound volume is higher and the frequency is modulated by motor speed. The vibration level on the motor may not be very high, 0.2 or 0.3 IPS, but the sound pressure level can be quite excessive. Siren effect sound pressure levels above 120 dB have been observed. (Pain is felt in the human ear at 120 dB.) An 1800 RPM motor with 38 rotor bars could generate a frequency of 1124.8 Hz, as in Fig. 4-34. Please note that plus and minus sidebands are present. The difference frequency between the sidebands is 29.6 Hz, or motor speed.

Solo Data on Motors

After the coupling has been installed and a motor has been set in place, the motor must be started and ran. After the motor finds its magnetic center (if it does), the shaft must be scribed. The scribe marker is essential for setting the proper distance between the coupling halves. While the motor is running solo, data should be taken. This data provides the last chance to identify imbalance, bent shaft, misalignment between the bearings' center lines, bearings loose in the housing, defects in the bearings, and inadequate lubrication. Every plant in the world could save hundreds of thousands of dollars per year if solo data were taken on all motors, and if repairs were made before the installation is completed.

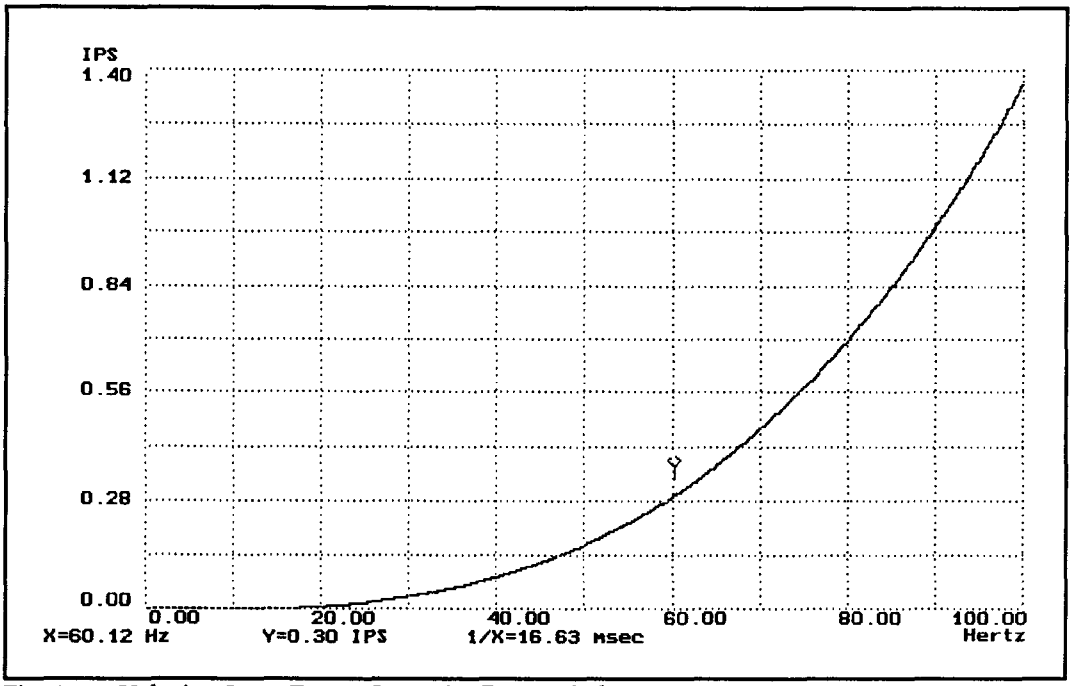


Fig. 4-35. Velocity Coast Down Curve for Pure Imbalance.

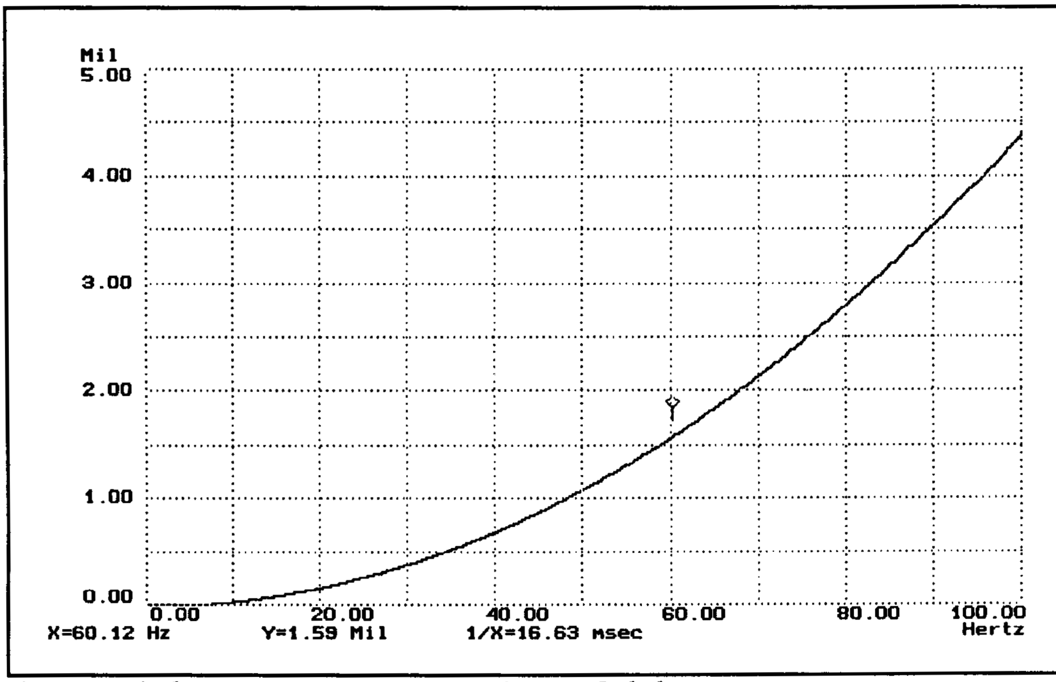


Fig. 4-36. Displacement Coast Down Curve for Pure Imbalance.

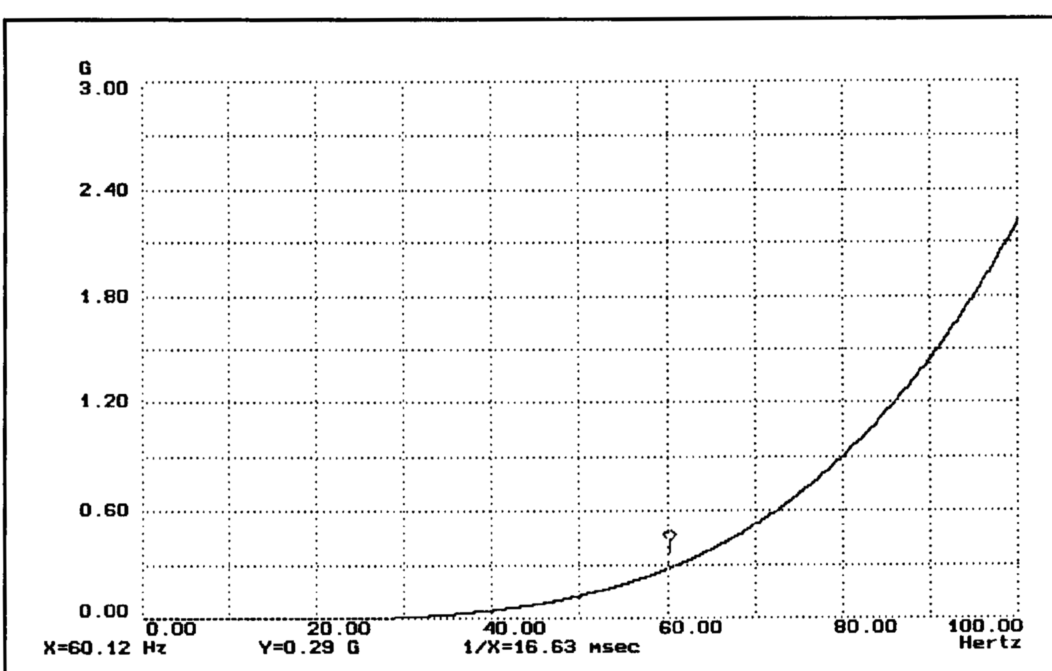


Fig. 4-37. Acceleration Coast Down Curve for Pure Imbalance.

The vibration should be recorded while the power is removed. Recording should continue until the motor speed is below 500 RPM. The coast down curves for velocity, displacement, and acceleration should be similar to Figs. 4-35, 4-36, and 4-37, respectively, for pure imbalance. The curves were produced with the **Vibration Calculator Program** and can be used to determine how much the vibration will increase for a given increase in speed.

Additional data should also be taken at startup after the installation is complete. This data identifies all problems with the drive unit that are installation related.

Solo, coast down, and startup data are essential for future reference, integrity of the new equipment, installation, and identification of potential premature failures.

STEAM TURBINES

Steam turbines require special attention because of the high speeds involved, extreme heat, and the small lightweight rotors. These factors contribute to rotor sag or bow. Therefore, most all turbines should be slow rolled for some period of time before alignment and operation. This slow rolling should remove the sag or bow from the rotor.

Solo data and coast down data must also be taken on turbines to determine the vibration response at various speeds and to identify imbalance, misalignment between the bearings, and natural frequency. Diagnostics can be severely complicated when the problem could have been identified from taking solo data first.

Startup and/or coast down data must also be taken for records, problem identification, resonant frequencies, bandwidth, and amplification factors.

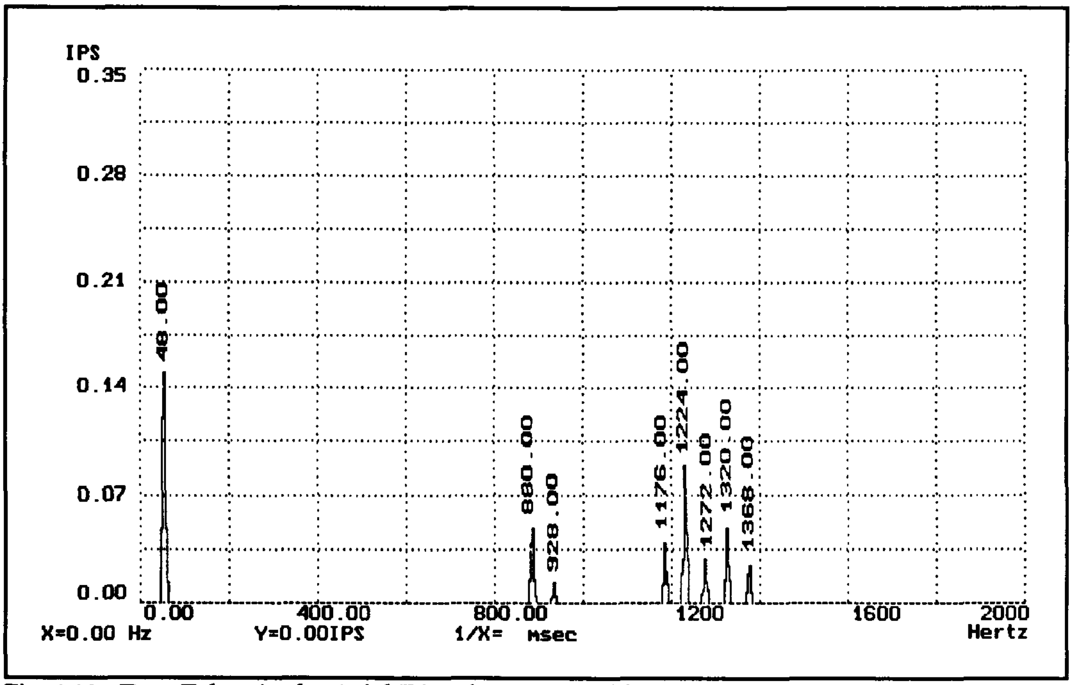


Fig. 4-38. Data Taken in the Axial Direction on a Turbine Housing.

If a turbine is not equipped with permanently mounted proximity probes, then data must be taken with portable proximity probes. Relative motion should be measured between the turbine bearing housing, the shaft, and the coupling.

Some manufacturers of proximity probes have over-emphasized the seriousness of mechanical runout, electrical runout and "glitches" to the point that many analysts will not use a proximity probe except in tightly controlled permanent installations. Portable probes have been used for years with a high degree of success. The portable proximity probe is quite versatile and is essential for relative motion measurements.

A limited number of problems can occur in turbines. For example, looseness, imbalance, misalignment, soft foot, resonance, and rubs. The net result is that most diagnoses rely solely on proximity probe data. Data should also be taken with a permanently mounted accelerometer. This is true because the frequency response curve for displacement is a downward and outward sloping curve. This means high frequencies generate very low displacement levels. The frequency response curve for acceleration is an upward and outward sloping curve. This means high frequencies generate large acceleration levels. See Fig. 1-27 in Chapter One. A rub in a turbine can generate high frequencies somewhere around the number of buckets times speed. The data in Fig. 4-38 was taken in the axial direction on a turbine housing. The 1224 Hz frequency is generated by the buckets hitting or rubbing 25.5 times each revolution. The 48 Hz difference frequency of the sidebands indicates that the frequency is modulated by turbine speed. The 880 Hz frequency indicates that the rotating unit is not hitting or rubbing the same number of times each revolution. Please note that 0.1 IPS at 1224 Hz equals 0.03 mils of displacement and 1.99 g's of acceleration. This problem would not have been identified with displacement. This points out the need to take data with an accelerometer on high-speed machines.

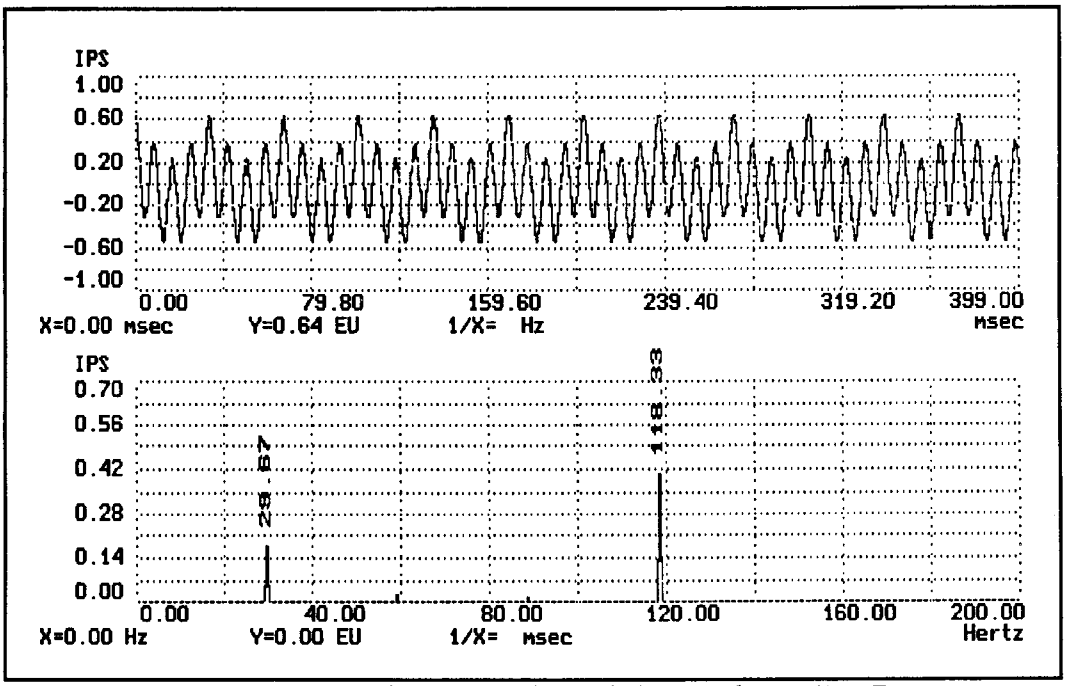


Fig. 4-39. Frequency Spectrum and Time Signal of Imbalance and Vane Pass Frequency.

PUMPS

In addition to the more general problems, a pump can have vane pass frequency and cavitation problems. Pump starvation, impeller loose on the shaft, and the impeller hitting something can all generate vane pass frequency.

Impeller Contact

When the vanes on an impeller are hitting something, vane pass frequency is equal to the number of vanes times speed. A pump rotating at 1776 RPM that has four vanes on the impeller would have a vane pass frequency of:

$$\frac{1776}{60} \times 4 = 29.6 \times 4 = 118.4\ Hz$$

When an impeller is hitting something, two types of problems normally exist: residual imbalance and vane pass frequency. In this case, there are two frequencies and two problems. Fig. 4-39 contains a frequency spectrum and time signal. The frequency spectrum has a spectral line at unit speed and at vane pass frequency. The time signal contains a high frequency riding a low frequency. The low frequency is unit speed, and the high frequency is vane pass frequency. Vane pass frequency is in phase with the fundamental.

If the impeller was loose on the shaft, the vane pass frequency would be modulated by unit speed. The data in Fig. 4-40 was taken from a pump that has five vanes. Vane pass frequency is changing its phase relationship to the fundamental because the impeller is loose on the shaft. The sidebands of unit speed on the low side of vane pass frequency

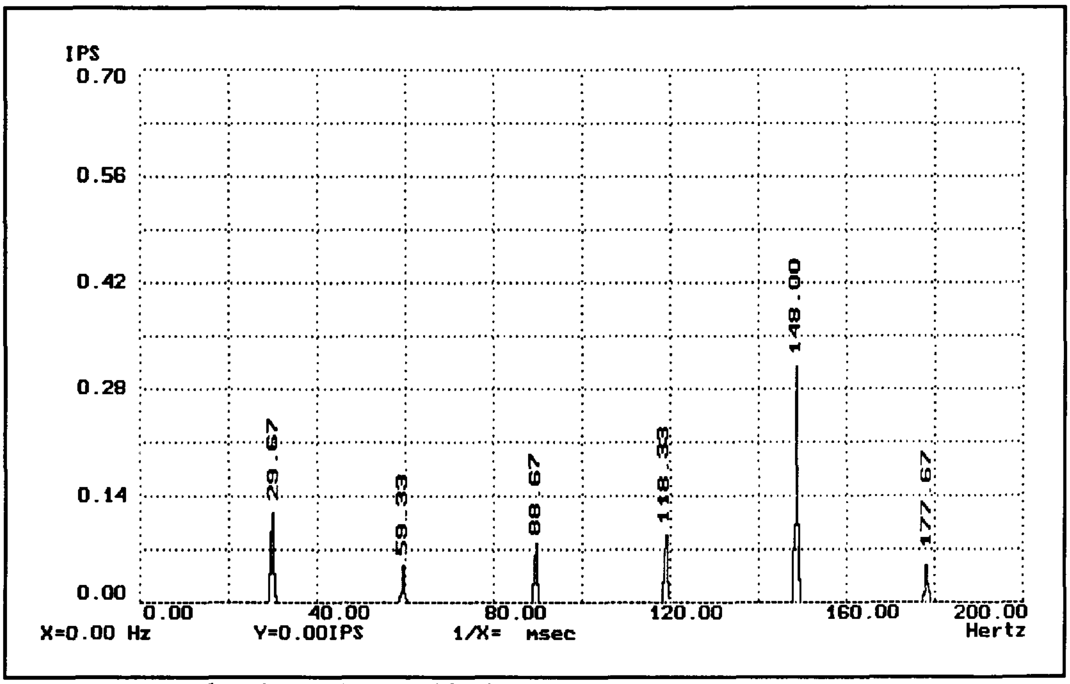


Fig. 4-40. Data Taken from a Pump with Five Vanes.

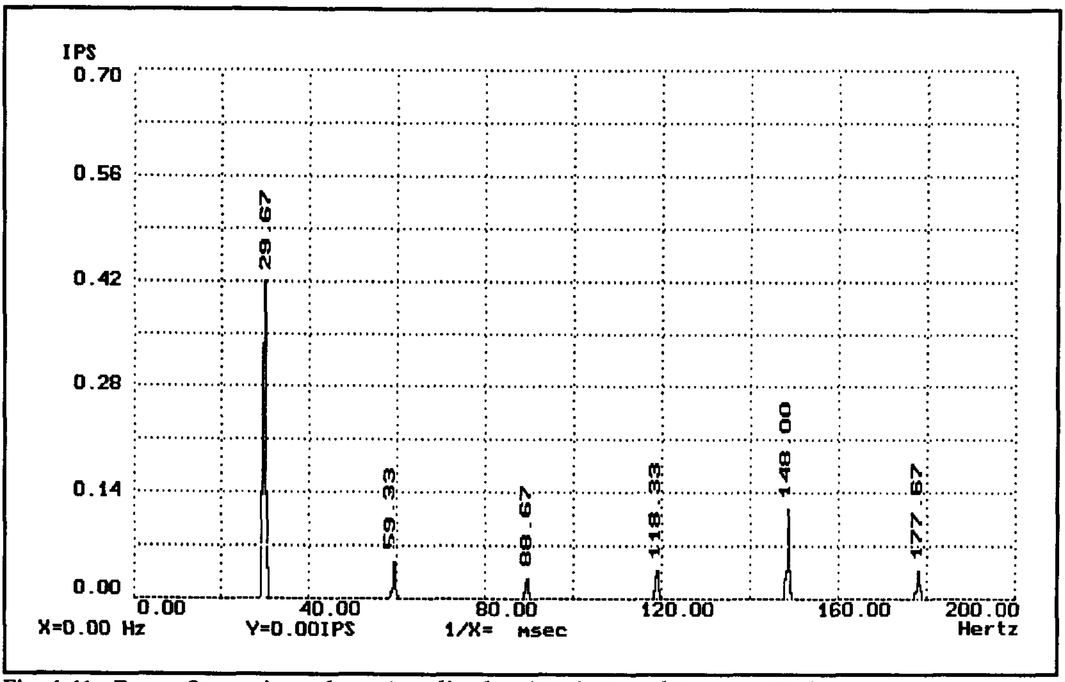


Fig. 4-41. Pump Starvation where Amplitude of Unit Speed Is Very High.

are higher in amplitude because the frequencies are out of phase most of the time. The lower level sidebands of unit speed on the high side of vane pass frequency are present because the frequencies are in phase some of the time. If one or two sidebands were only on the high side of vane pass frequency, this would indicate that the signals are in phase. The impeller is hitting something each revolution.

Starvation

Pump starvation occurs when enough liquid is not present to fill each vane on the impeller every revolution. When this occurs, the amplitude of unit speed can be quite high, as in Fig. 4-41. Pump starvation is often confused with imbalance. However, starvation has the following characteristics:

1. Since each vane on the impeller is filled to varying degrees each revolution, the amplitude can vary each revolution.

2. The time signal could have some distortion.

3. If the pump is driven by an electric motor, the amp load on the motor may be less than normal.

Starvation can also occur when foam is mixed with some slurries. Low amplitudes of vane pass frequency can also be present with starvation.

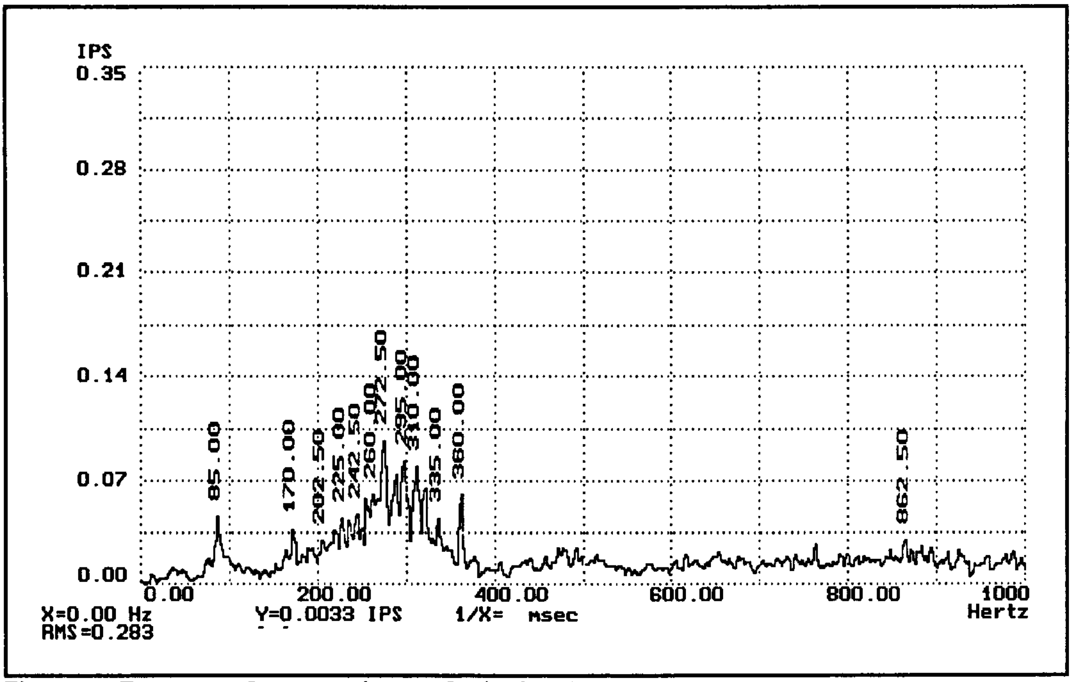


Fig. 4-42. Frequency Spectrum from a Cavitating Pump.

Cavitation

When a pump is not operating at the correct point for the pump curve, cavitation can occur. This phenomenon normally occurs when the suction intake is restricted, and the liquid tends to vaporize while coming off the impeller. This process generates noise and, to the human ear, may sound like the pump has failed. The frequency spectrum from a pump that is cavitating normally contains a spectral line at vane pass frequency on top of noise, as in Fig. 4-42. When cavitation occurs, operations or personnel should be contacted to correct the problem.

Vane pass frequency and harmonics at relatively high amplitudes are often measured on vacuum pumps. This can be caused by bent shaft and build-up in the pump, or improper clearance. However, this problem is most often caused by improper operation. These pumps need a water seal for proper operation; however, too much water can generate high amplitudes of vane pass frequency, as in Fig. 4-43.

Diagnostics of pump problems can be improved and expedited by installing a pressure transducer in the discharge line of the pump. Pressure transducers measure pressure fluctuations. They are calibrated in mV/PSI and the output can be processed on a real-time analyzer. The frequencies measured are the frequencies of the fluctuations, and the amplitude is the amount of fluctuations measured in PSI zero-to-peak.

COMPRESSORS

Compressors compress some form of gas, come in various configurations, and operate in

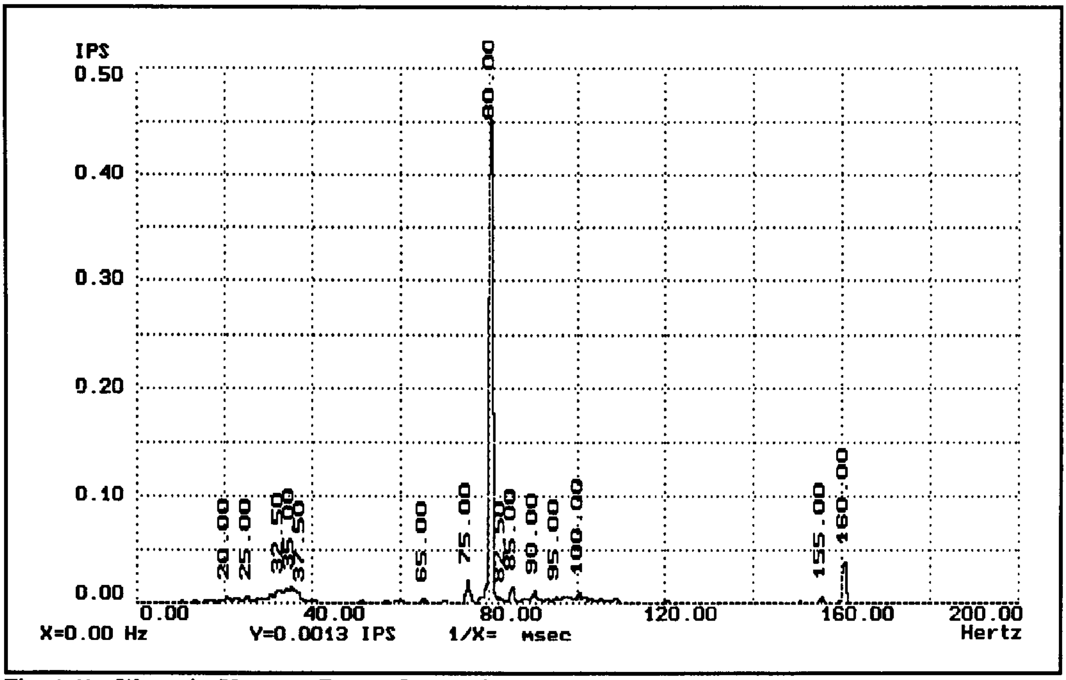


Fig. 4-43. Water in Vacuum Pump Generating Vane Pass Frequency.

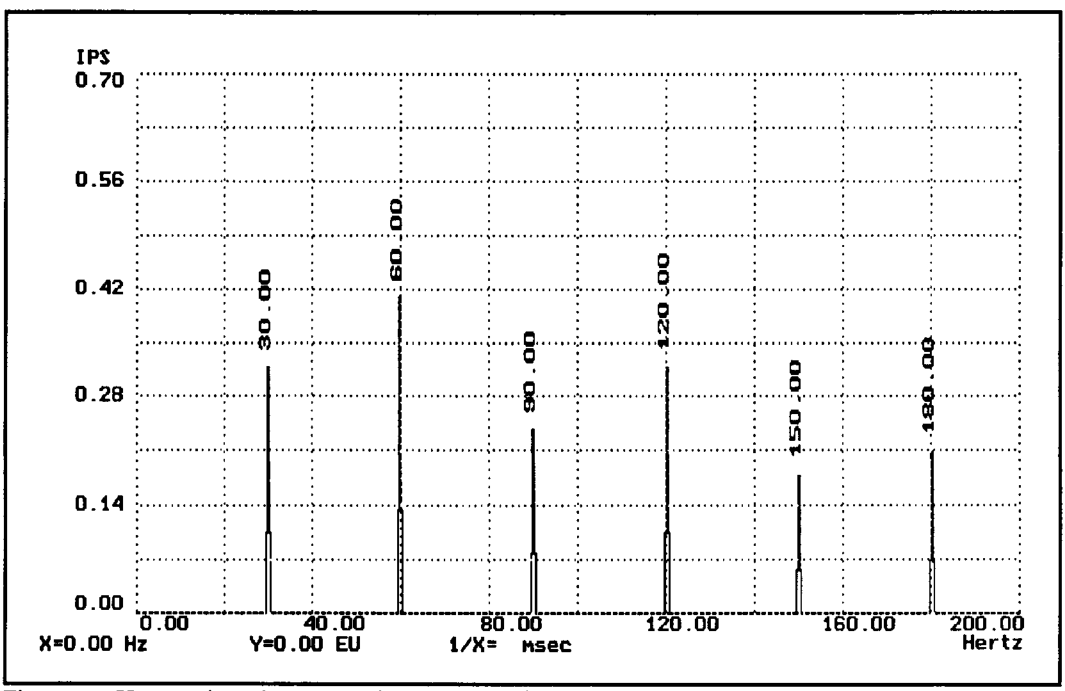


Fig. 4-44. Harmonics of Frequencies Generated in a Compressor.

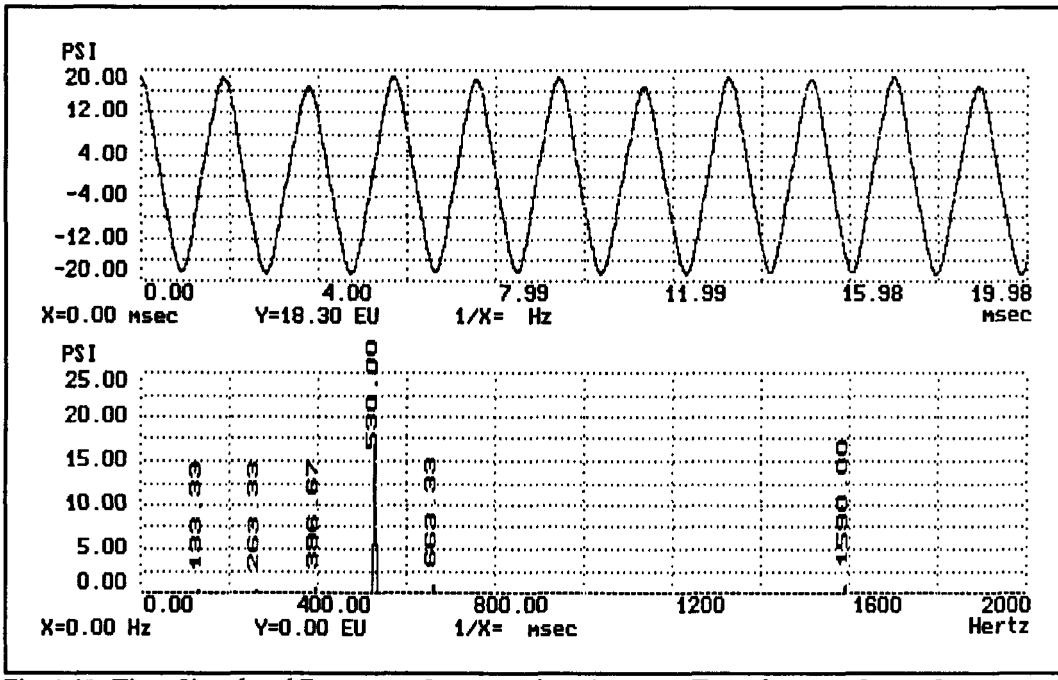


Fig. 4-45. Time Signal and Frequency Spectrum from Pressure Transducer on Screw Compressor.

much the same way as pumps. For example, some compressors have a given number of lobes or vanes on a rotor. They generate vane pass frequency as do pumps. However, compressors may not cavitate. On screw compressors, vane pass frequency is calculated by multiplying the number of flights on the screw times the speed of the screw.

Some compressors have two rotors, and each rotor has two or more lobes or vanes. The rotors can turn at the same speed and have timing gears. These compressors can generate frequencies at rotor speed, two times rotor speed because there are two lobes or vanes on each rotor, and four times rotor speed because there are four lobes or vanes on the two rotors. Harmonics of each frequency can also be generated, as in Fig. 4-44. With this design and the close tolerances involved, overall vibration levels often exceed 0.3 IPS.

Pressure transducers should be installed in the discharge line of compressors because the measurement of pressure fluctuation is often required for accurate diagnosis. Fig. 4-45 contains the time signal and frequency spectrum from a pressure transducer installed in the discharge line of a screw compressor with four flights. Vane pass frequency is 530 Hz. The static pressure is 92 PSI, and the measured pressure fluctuations are 17%. (Fluctuations are: **16 ÷ 92 = 17%**). This is excessive by most standards. In some applications, pressure fluctuations are held to two or three percent.

FANS

Fans can generate blade pass frequency, which is the number of blades times speed. This frequency normally occurs when the blades are hitting something, the housing is quite flexible, or an acoustical resonance is present which equals blade pass frequency. The more typical problems, such as imbalance, looseness, misalignment, bent shaft, and defective bearings are discussed in other sections.

SPECIAL TESTS

Accurate diagnosis of rotating machinery problems often requires one or more of the following tests:

1. Startup and coast down tests to measure machinery vibration at all speeds during startup and coast down
2. Bump tests to measure the machine response when bumped or hit with enough energy to cause the machine to vibrate
3. Sound recordings with a microphone to measure discrete frequencies heard with the ear
4. Synchronous time averaging to remove nonsynchronous vibration data
5. Relative motion measurements to determine how much one object is moving relative to another

Startup/Coast Down Data

This data is used to identify and verify imbalance, misalignment, resonance, and bent shafts. Startup and coast down data cannot be used in all cases because some machines will not coast when the power is removed. On other machines, the startup cannot be controlled. However, when a machine will coast or the startup can be controlled, these tests are valuable diagnostic tools.

For best results, all startup and coast down data should be recorded because once the data is recorded, it can be processed as many times as necessary to obtain the desired results. If the data is processed live, you often have only one chance to take the data. If the equipment is not set up properly or other operator errors are made, this valuable data may be lost. An exact setup procedure cannot be established because there are several variables involved. However, the following information may be helpful.

The real-time analyzer must have a peak-hold averaging capability. With this averaging mode, every time the peak amplitude changes at any frequency, the amplitude is written to the screen. This procedure provides a maximum amplitude plot of speed as the machine starts up or coasts down. The frequency range should be just high enough to contain the maximum speed. **Warning:** The lower the frequency range used, the longer the real-time analyzer takes to collect data for one average. When the machine is increasing or decreasing in speed much faster than the averaging is performed, a smooth amplitude plot or curve cannot be obtained. In this case, the curve will have sharp increases and decreases, and the frequency range should be increased to reduce the time required for averaging. The number of lines of resolution can also be decreased to speed up the averaging. Sometimes it is necessary to both increase the frequency range and decrease the lines of resolution.

The number of averages required depends on the real-time rate of the analyzer, frequency range, lines of resolution, and amount of time the machine will coast. For example:

Realtime rate = 10,000 *Hz*

Lines of resolution = 400

Frequency range = 200 *Hz*

Time required for startup/coast down = 3 *minutes*

Then

$$\text{Time to perform 1 average} = \frac{1}{10{,}000 / 400} = 0.04 \text{ sec}$$

$$\text{Time to collect data for 1 average} = \frac{1}{200 / 400} = 2 \text{ sec}$$

Total time for one average = 2 sec

Then

Since 3 *minutes X* 60 sec/min = 180 sec

$$\text{The number of averages} = \frac{180}{2} = 90 \text{ averages}$$

A total of 90 averages are required for the above given parameters. Startup and coast down data may require over 300 averages, depending on how long it takes to start up or coast down.

Figs. 4-35 through 4-37 contain the imbalance frequency response curves for velocity, displacement, and acceleration. The amplitude at 60 Hz is 0.3 IPS. The equivalent displacement is 1.59 mils, and the equivalent acceleration is 0.29 g's. These curves were generated with the **Vibration Calculator Program**. These curves provide the frequency response for a pure imbalance condition, and are used to compare with actual coast down data to identify imbalance. These curves are also used to determine the increase in vibration associated with an increase in speed. The coast down data from a machine that has pure imbalance and is operating below first critical should resemble these curves.

Coast down and startup data from a machine operating above first critical should resemble the curve in Fig. 4-46. When power is removed, the amplitude decreases sharply by a small amount. This decrease in amplitude represents the loading factor. When the machine starts to coast, the amplitude follows the slope of an imbalance response curve. When the speed approaches the critical speed, the amplitude increases until the center frequency of the critical speed is reached. After the speed passes through the center frequency, the amplitude again decreases. The resulting bell-shaped curve is the "envelope" of the resonant frequency. By measuring the center frequency and the frequency at the half-power points (down 3 dB or 0.707 times peak), the amplification factor can be calculated as discussed in Chapter One.

Coast down and startup data from a machine with misalignment should resemble the

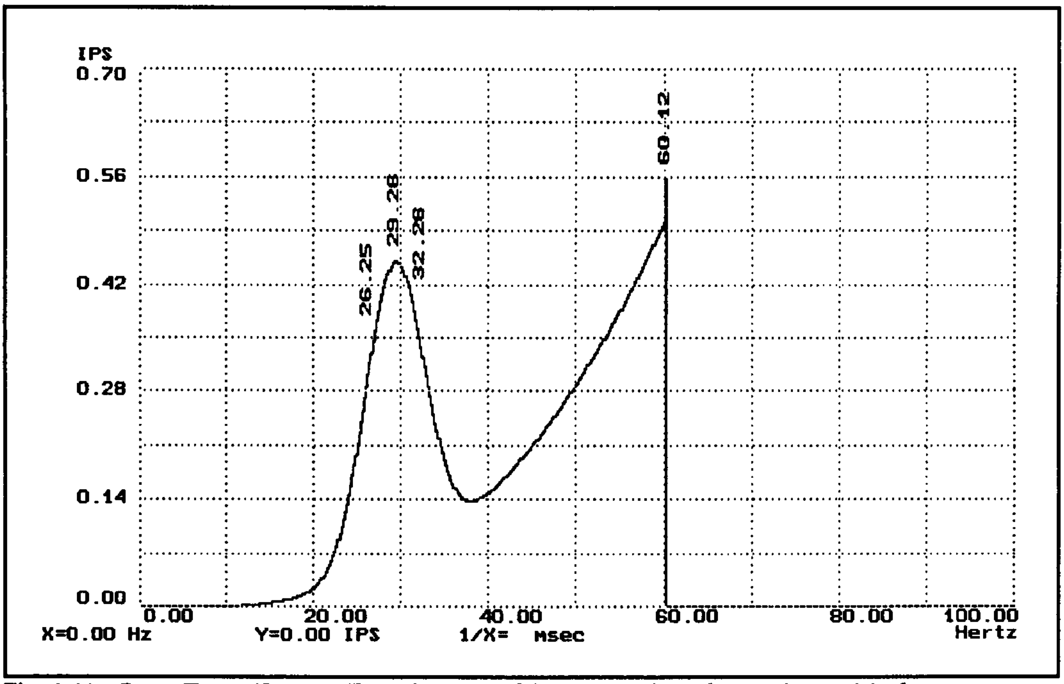


Fig. 4-46. Coast Down/Startup Data from Machine Operating above First Critical.

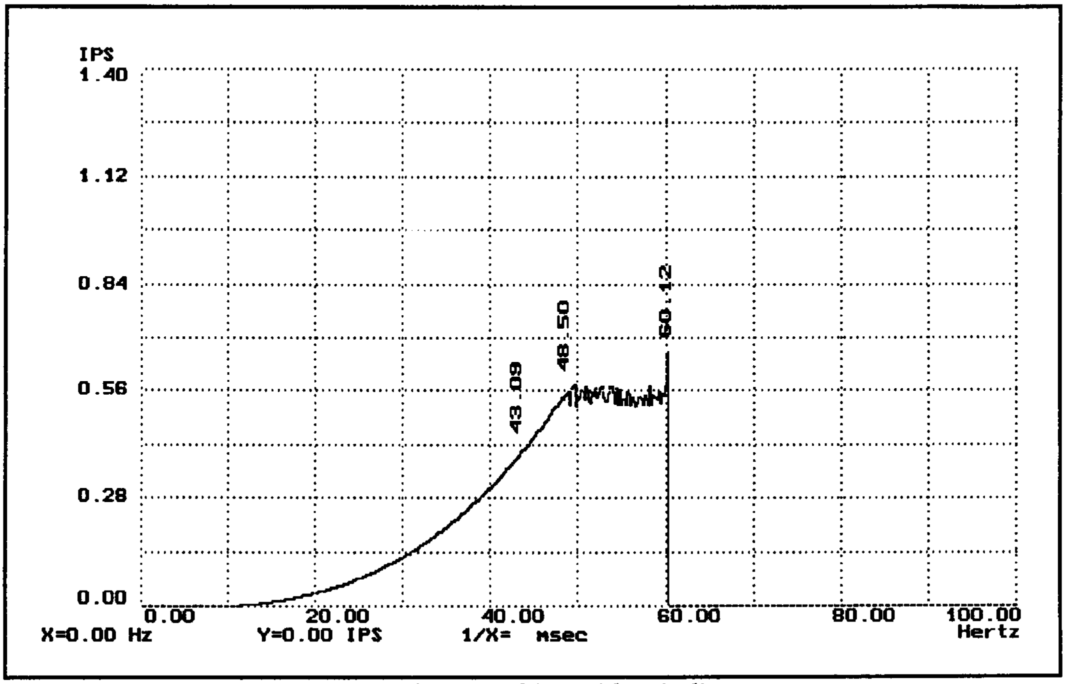


Fig. 4-47. Coast Down/Startup Data from Machine with Misalignment.

curve in Fig. 4-47. This data can identify misalignment between the bearing center lines in solo data, and misalignment between the coupling halves on operating machines.

When the power is removed, the amplitude drops sharply by a slight amount. This drop also represents the load. The amplitude then remains relatively constant or flat for a specific decrease in speed. The length of the flat line is determined by the amount of misalignment.

The flat line indicating a given amount of misalignment can cause a problem in a given machine. Once the problem occurs, reasonable increases in speed may not cause an increase in amplitude. In other words, misalignment may be speed related until the problem occurs and not speed related after the problem occurs. In this coast down data, calculate the amplitude at the half-power point (0.707 X the average peak of the shelf or flat line). For example, 0.707 X 0.55 = 0.39. Next, identify the frequency at the half-power point, 43 Hz in this case. Then measure the amount of misalignment between the coupling halves (0.015" in this case). This data indicates that this machine could be operated at 43 Hz or 2580 RPM with 0.015" misalignment. Several measurements such as this could reveal realistic alignment standards.

When a machine is operating above first critical and the shaft is bent, the coast down data could be similar to the data in Fig. 4-48. This data indicates that the bent shaft is not causing a misalignment problem. When the machine has a bent shaft and is operating above first critical, a distinctive dip should occur in the bell curve representing the critical speed. The reason the dip occurs is because the phase of the imbalance shifts 180 degrees each time the machine goes through a critical. The bend in the shaft stays at the same place, and when the imbalance shifts 180 degrees, the imbalance must be out of phase with the bend in the shaft for a very short period of time. When this occurs, the machine tries to balance itself, and the amplitude drops. Since the phase shift occurs very fast, the imbalance returns as the coast down continues.

Analysis is complicated when both misalignment and a bent shaft are present. The data in Fig. 4-49 was taken from the center bearing on a three-bearing turbine generator set. The shelf or flat line indicating misalignment is present. The distinctive dip in the critical speed envelope indicates a bent shaft. When this occurs, the questions that often arise are: "Is the misalignment causing a bow in the rotor?" or "Is the bow in the rotor causing the misalignment?" In either case, the machine must be removed from service, physical measurements must be made, and the necessary repairs must be accomplished.

Bump Tests

Bump tests are often required to determine natural frequencies when startup and coast down tests cannot be performed. A bump test is performed by placing a transducer on the machine or part and then hitting or bumping the unit with enough energy to excite the natural frequencies. The real-time analyzer should be set up for a dual display with both time and frequency data, as in Figs. 4-17 and 4-18. The analyzer should trigger from the time data with a delay to display the beginning of the pulse. The data in Figs. 4-17 and 4-18 was recorded with a microphone while bumping a crystal glass. The glass was filled with water to obtain some damping. The dip in the time signal in Fig. 4-18 reflects the impact when the glass is struck. The glass has three natural frequencies, and the higher frequencies are not harmonics of the lower frequencies. Also, note the frequency

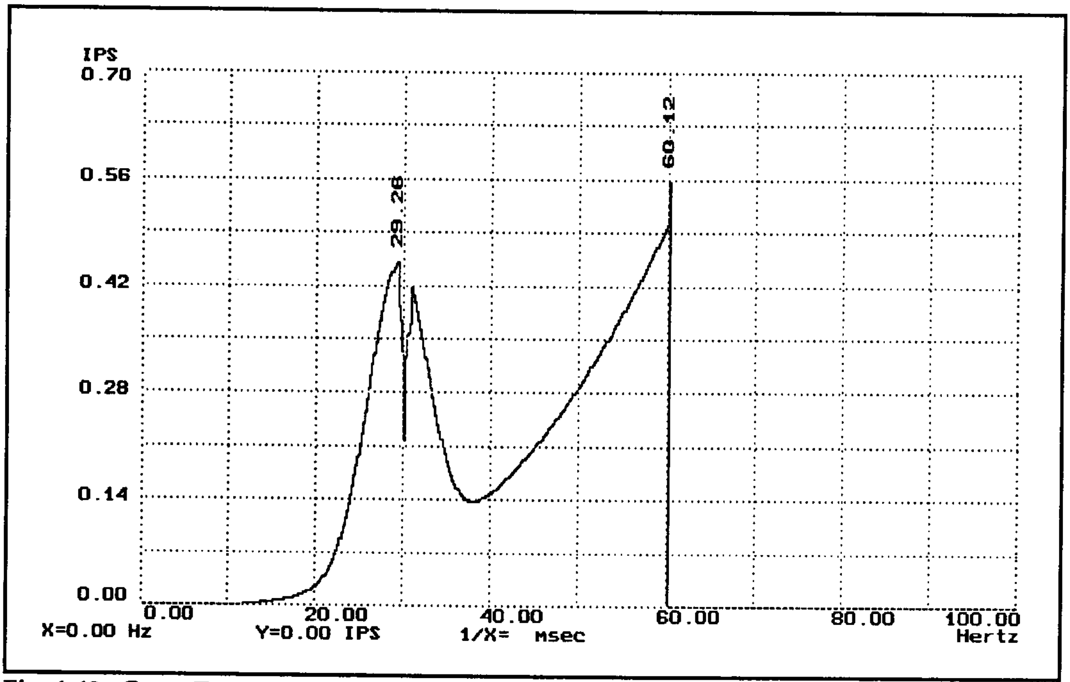


Fig. 4-48. Coast Down Data on Machine Operating above First Critical with Bent Shaft.

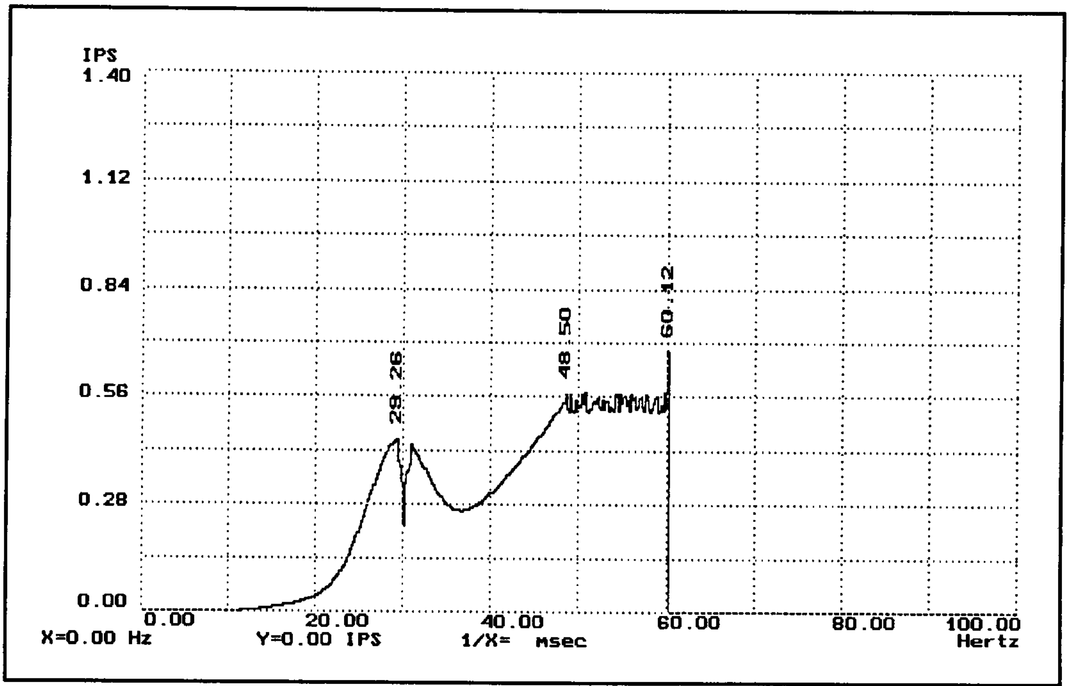


Fig. 4-49. Misalignment with a Bent Shaft.

spectral line at 487 Hz in Fig. 4-17 is wide banded, and the time signal is relatively well-damped. The frequency spectral lines in Fig. 4-18 are narrow banded, and the time signal is relatively undamped.

Bump tests are not effective on all machines or parts because of size, energy required, etc. When bump tests are performed, care must be used when placing the transducer. If the transducer is placed in the direction of the hit or bump, only the natural frequency of the transducer may be excited. Precise rules cannot be given on transducer placement for every case. As a general rule, the transducer should be placed on the part where the response is expected, and the part should be bumped in such a way as to cause the response. Trial and error of transducer placement, locations, and magnitude of bumps often yields the best results.

Noise Recording

A microphone and recorder cannot be used in all cases to diagnose problems in rotating machines. However, many cases arise when a microphone is the best transducer to use. Noise should be recorded to:

1. Prove or disprove that machine problems are causing the noise.
2. Identify the source of the noise.
3. Determine the frequency and amplitude for noise abatement projects.
4. Determine the correlation between sound pressure levels (SPL)

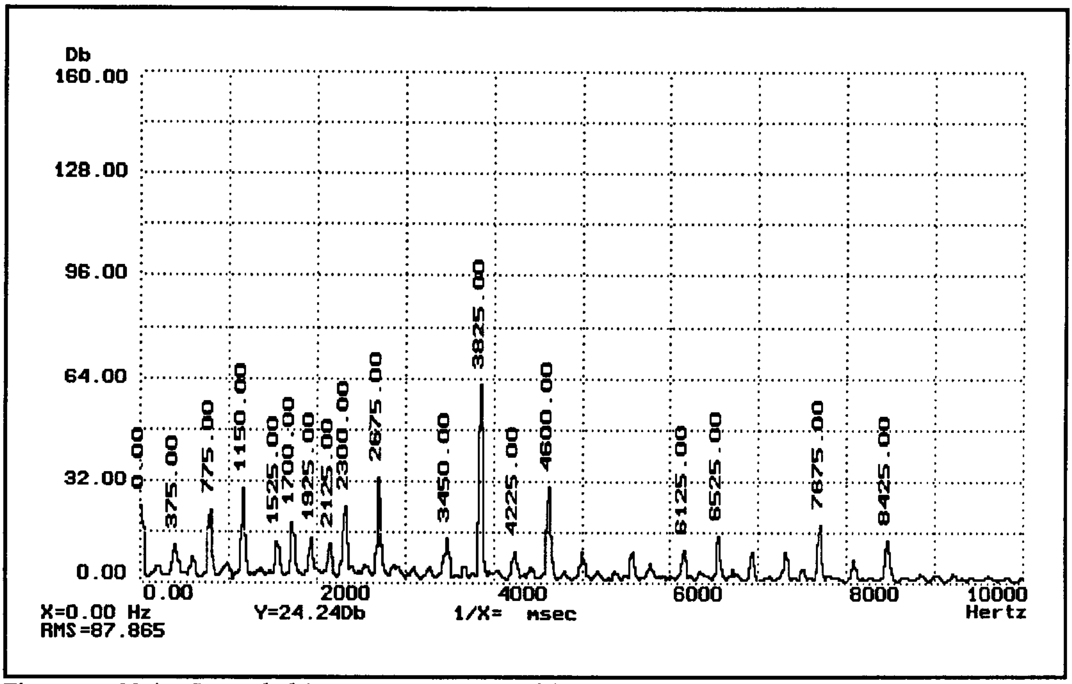


Fig. 4-50. Noise Recorded in a Compressor Building.

and vibration levels.

Fig. 4-50 contains the frequency spectrum of the noise recorded in a compressor building. The frequencies measured are vane pass frequency of the low pressure compressor (375 Hz) and the high pressure compressor (775 Hz) plus harmonics of these frequencies. The zero-to-peak value of the highest peak is about 64 dB. The RMS value is about 88 dB or an overall zero-to-peak value of 1.414 X 88 = 124.4 dB. At this level, pain is felt in the unprotected ear. The cause of the high noise in the room and the high vibration level on the compressor is vane pass frequency. When the problem in the compressor is fixed, the noise and vibration may be reduced to acceptable levels.

Synchronous Time Averaging (STA)

Since frequencies add, subtract, mix, and beat, and various combinations of some frequencies equal other frequencies and/or harmonics of other frequencies, accurate analysis is difficult, errors are made, and some problems are overlooked. In these situations, synchronous time averaging must be used to identify and isolate the contribution of each vibratory component. Synchronous time averaging is required for accurate diagnostics of:

1. Press and calender rolls in paper machines

2. Embossing/pattern and backup rolls in converting and plastics plants

3. Work and backup rolls in rolling mills

4. Certain gear problems

5. Defect size in antifriction bearings

The equipment required includes various transducers, a high phase or a once-per-revolution trigger, and a real-time analyzer capable of STA by triggering from an external trigger or the time domain signal.

Arithmetic averaging, as accomplished by most analyzers, randomly fills the memory with data, performs an FFT, and stores the data. As soon as these functions are completed, the analyzer starts filling the memory again, performs the other functions, and presents an arithmetic average of all samples taken. This is required in all applications to provide signal enhancement because all frequencies may not be present in every average of the data. The amplitude of each frequency can vary from average to average.

Synchronous time averaging synchronizes the start time for filling the memory, with an external trigger or a repeatable distinctive event in the time signal. The memory is filled and the data is stored. The next average is not started until the next trigger arrives. The memory is filled again and the second average is combined with the first average. This process continues until the total number of required averages is completed.

Since each average is started at trigger time, and the trigger or event occurs at shaft speed or during another event such as ball pass frequency of the outer race, etc., then all vibration related to the trigger/event should be in phase for every average. When the

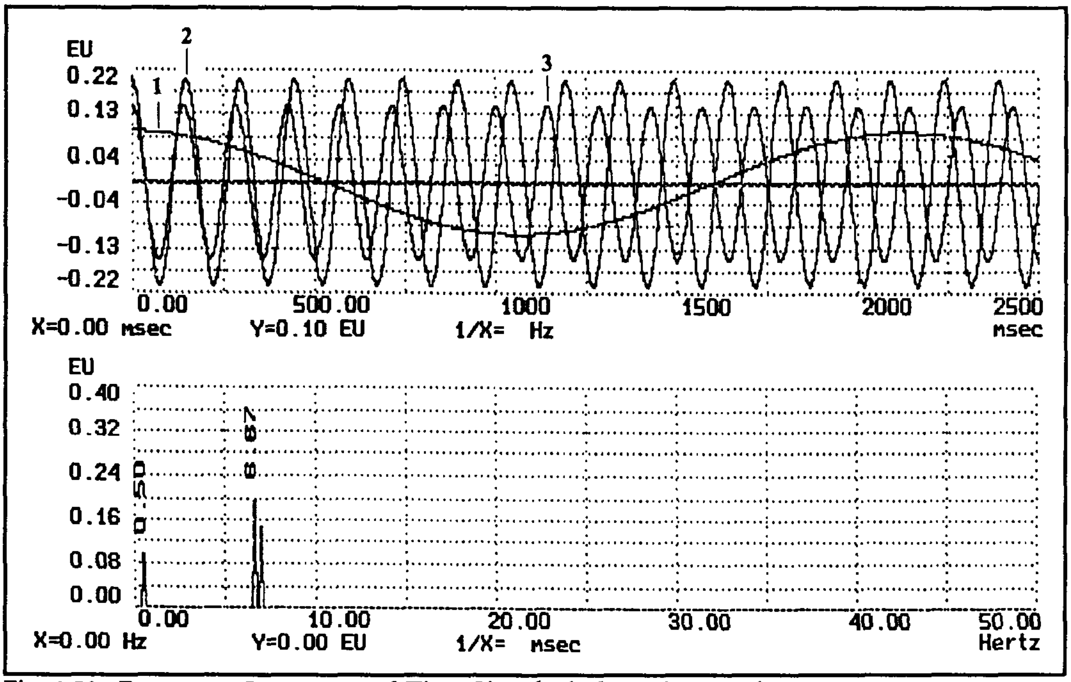


Fig. 4-51. Frequency Spectrum and Time Signal of Three Frequencies.

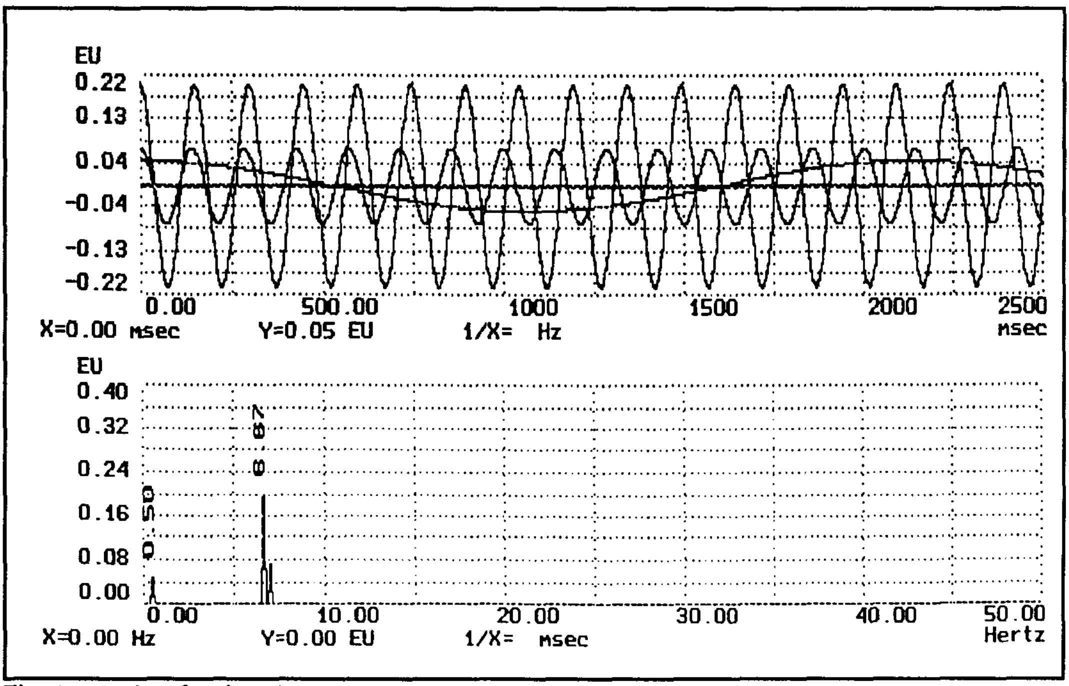


Fig. 4-52. Signals after Four Averages.

new data is compared with the stored data, all frequencies in phase with the trigger are retained and all frequencies out of phase with the trigger are subtracted out of the data. This means all frequencies occurring at event (trigger) speed or some exact multiple of event speed are retained. It also means that all frequencies that are not an exact multiple of event speed are averaged out of the data. Fig. 4-51 contains the frequency spectrum and time signal of three frequencies: felt speed = 0.47 Hz, roll 1 speed = 6.7 Hz, and roll 2 speed = 7 Hz. The time signal displays the frequencies before they are mixed. Point 1 is the 0.47 Hz signal, point 2 is the 6.7 Hz signal, and point 3 is the 7 Hz signal. At the beginning of the time period, the 6.7 and 7 Hz signals are in phase. However, at the end of the time period these two signals are out of phase. Since the time period between the 0.47 Hz signal and the other two signals is large, the lower frequency is always out of phase on either half of the cycle. Fig. 4-52 contains the same signals after four averages. This averaging was performed by triggering off the roll rotating at 6.7 Hz. Note that the amplitudes of the 0.47 and 7.0 Hz signals are reduced by 50%. Fig. 4-53 contains the same signals after 100 averages. The 0.47 and 7.0 Hz signals are still present, however, the amplitudes are now reduced by 90%. Since most frequencies will be in phase for some period of time during the averaging, it is hard to remove 100% of the nonsynchronous signals. If we define the synchronous data as the "signal" and the nonsynchronous data as "noise," then the signal to noise ratio is improved by the square root of the number of averages:

$$\sqrt{N}$$

where N is the number of averages. Some examples of synchronous vibration are:

1. Speed of the roll used as a trigger

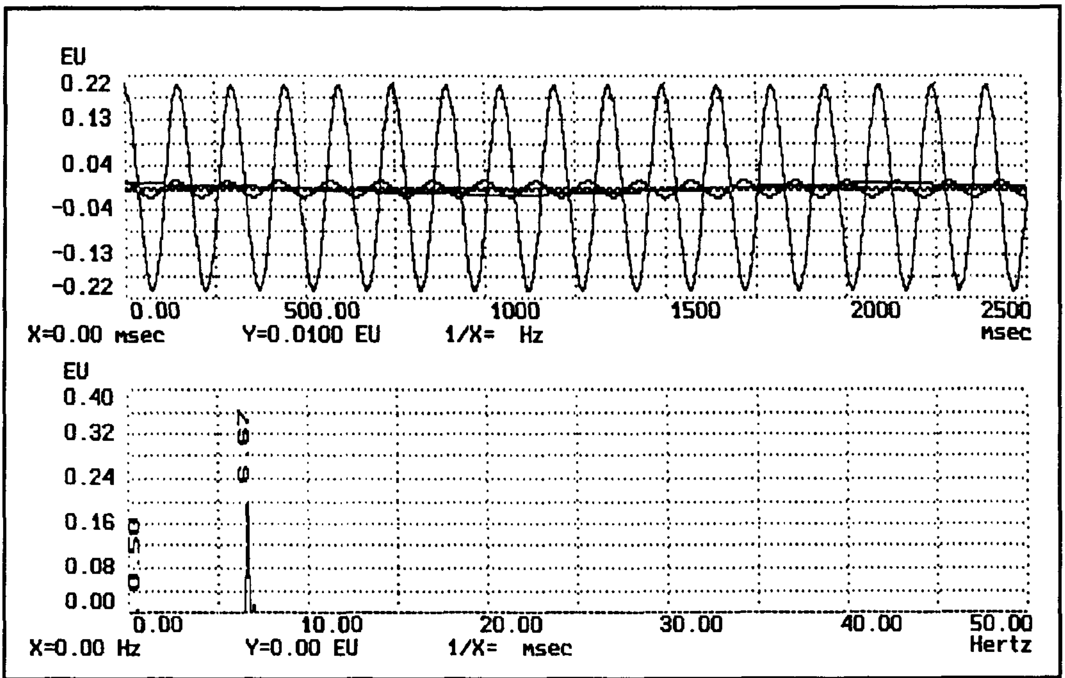


Fig. 4-53. Signals after 100 Averages.

2. Gearmesh frequency
3. Blade or vane pass frequency
4. Sideband frequencies caused by eccentricity
5. Frequencies generated by bars on rolls

Some examples of nonsynchronous vibration are:

1. Speeds of all rolls and shafts not used as a trigger
2. Noise
3. Bearing frequencies
4. Some sideband frequencies caused by looseness
5. Some harmonic frequencies caused by looseness

If a real-time analyzer will trigger from the time signal and the time signal contains a repeatable distinctive event, such as a defect on a bearing inner race or outer race, synchronous time averaging can be performed. For example, if a bearing has a defect on the outer race, STA sources using the time signal for a trigger could remove most of the vibration not related to the bearing defect. The sources of the data removed could be shaft speed and harmonics, gearmesh frequency, blade/vane pass frequency, and other bearing defect frequencies. The resulting improved signal facilitates a more accurate diagnosis and description of the problem. For more examples of synchronous time averaged data, please refer to Chapter Seven.

<u>Relative Motion Measurements (RMM)</u>

It is beyond the scope of this book to give an example of every case where relative motion should be measured. However, it is important to emphasize the need and general application of relative motion measurements. A relative motion measurement measures how much one part is moving relative to another part. The measurement is normally made in thousandths of an inch or mils. This is not a new measurement technique. It has been used for years with an instrument called the dial indicator. We now have electronic dial indicators called displacement transducers. These transducers can be of the contacting type or the noncontacting type. The contacting type, or linear variable differential transformer (LVDT), has been used for years in the gauging industry. The noncontacting type, or proximity probe, has been used for years in the turbo machinery industry to measure relative shaft motion in fluid film bearings. The use of these instruments in both applications has been quite successful.

Portable variations of these transducers have been used for years to accurately diagnose problems in rotating machines. The major advantages of these transducers over the dial indicator are:

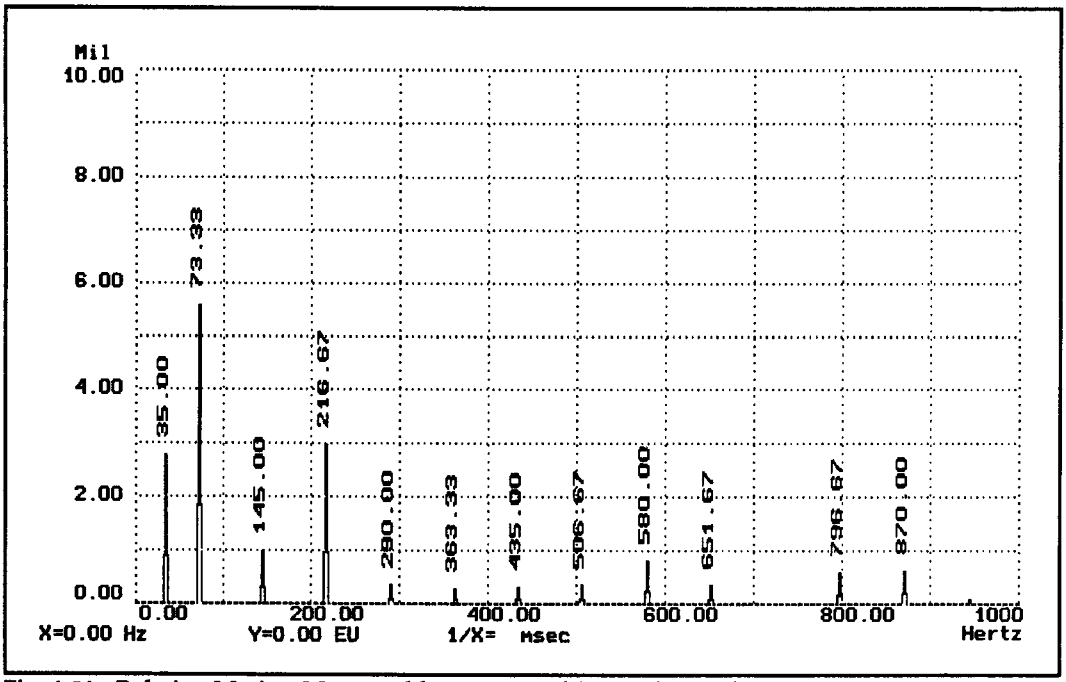


Fig. 4-54. Relative Motion Measured between Turbine and Coupling.

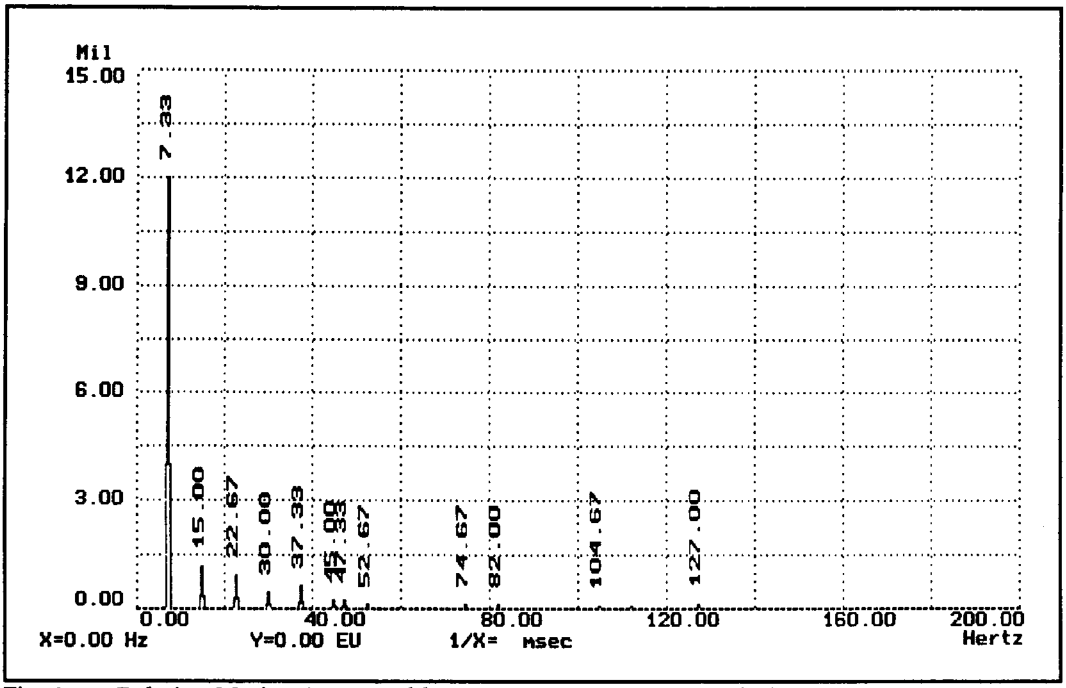


Fig. 4-55. Relative Motion Measured between Motor Housing and Shaft.

1. The output can be processed on an RTA to obtain a time signal and frequency spectrum.

2. They can be used while the machine is in operation, often without adversely affecting the operation.

3. They can be used to diagnose defects in antifriction bearings on shafts rotating less than 1 RPM. (Refer to Chapter Five for examples.)

4. They are required to diagnose gear problems in slow speed machinery when the gear and shaft weigh less than about 10% of the housing weight. (Refer to Chapter Six for examples.)

5. They are required to measure the roundness of operating steel rolls. (Refer to Chapter Seven for examples.)

6. They are required to measure the amount of looseness in a bearing, the bend in a shaft, and the amount of misalignment in a coupled machine.

7. They are required to measure the relative motion measured between two beams or structures before bracing is attempted.

The data in Fig. 4-54 shows the relative motion measured between a turbine and the coupling. The spectral line at 35 Hz is two times fan speed and is caused by a problem in the fan. Three mils at 35 Hz equals about 0.36 IPS of velocity. The spectral line at 72.5 Hz is turbine speed. Six mils at 72.5 Hz is about 1.4 IPS of velocity. These vibration levels are excessive, and the data indicates a problem in the fan and misalignment between the turbine and gearbox.

The data in Fig. 4-55 shows the relative motion measured between a motor housing and the shaft. The spectral line at 7.5 Hz is motor speed and indicates imbalance. Twelve mils at 7.5 Hz is equal to about 0.24 IPS of velocity.

Since more than one problem can generate the same frequency, relative motion measurements are often required to accurately identify the problem and distinguish between imbalance, looseness, bent shaft, etc.

In conclusion, the technology presented in this chapter should be helpful in accurate diagnosis of machinery problems. There are many quick fix principles and schemes available for machinery diagnostics and most, if not all, have some merit. However, people in the vibration analysis field have a responsibility to achieve as much accuracy as possible.

CHAPTER FIVE:
ACCURATE DIAGNOSIS OF ANTIFRICTION BEARINGS

INTRODUCTION

This chapter describes procedures for identifying defects in antifriction bearings by analyzing frequencies generated by the moving parts. Defects on bearing raceways, rolling elements, and the cage generate different frequencies. The spectrum shape, amplitude, frequency, sum and difference frequencies, and the time domain signal are useful in identifying the nature, location, combination, and size of defects.

Methods are presented to calculate the bearing frequencies; identify if the bearing is in a thrust or radial load; calculate the length of inner race defects with the time domain signal; and measure the length of a defect on the outer race in the frequency domain spectra. The nature of the defect, such as shallow flaking, deep fatigue spalls, corrosion, acid etching, fluting, and inadequate lubrication can be determined by analyzing the frequency and time domain data. These techniques can also identify bearings with excessive clearance and bearings that are not properly installed, such as those turning on the shaft or loose in the housing. Techniques used to predict the life span of a defective bearing are discussed.

DATA COLLECTION

Data collection is the most important step in evaluation of bearing condition. Data should be collected by placing the transducer in the bearing load zone with due respect to flexibility. If this is not done, the best signal definition may not be obtained. For example, with a radial bearing in a radial load, the best signal is obtained in the radial position. For an angular contact bearing or a radial bearing in a thrust load, the best signal definition is obtained in the axial direction. The machine internal geometry, as well as which problems generate radial or thrust loads, must be determined in order to properly place the transducer. Data should be taken where the transfer function is best; for example, put the transducer on a bolt head, not the cover. For pillow-block, tending side dryer, and similar bearings, the transducer should be placed near the top of the bearing. For bearings in gear housings, the transducer should be placed on a bolt head in the load zone.

TRANSDUCER SELECTION

Success or failure in diagnosing bearing defects often depends on the selection of the proper transducer. The discussions of transducer selection in other chapters apply. However, a few words should be added here. Spherical roller bearings rotating at 1200 RPM can generate harmonics of BPFO in the 3,000 Hz range when fluting occurs. An accelerometer must be used in such cases.

Low speed machines rotating as low as 2 or 3 RPM, other machines when bearing frequencies are below 10 Hz, and lightweight shafts installed in heavy housings all require either contacting or noncontacting displacement transducers for accurate diagnostics.

The velocity transducer is still the best choice for frequencies between 10 and 2,000 Hz.

GENERATED FREQUENCIES

In order to understand the relationships between the different rotating elements of a bearing, the equations describing the relative speeds must first be developed. These equations define the frequencies generated by antifriction bearings.

A machine with a defective bearing can generate at least five frequencies. These frequencies are:

1. Rotating unit frequency or speed (S)
2. Fundamental train frequency (FTF)
3. Ball pass frequency of the outer race (BPFO)
4. Ball pass frequency of the inner race (BPFI)
5. Two times ball spin frequency (2 X BSF)

Figs. 5-1 and 5-2 show the axial and cross-sectional views of the geometry for a ball bearing, where v_o, v_c, and v_i are the linear velocities of the outer race, ball center, and inner race, respectively. B_d is the ball diameter, P_d is the pitch diameter of the bearing and is measured from ball center to ball center, and ϕ is the contact angle. If a vertical line is drawn through the bearing and another line is drawn where the ball contacts the inner and outer races, the angle between the two lines is the contact angle. See Fig. 5-2.

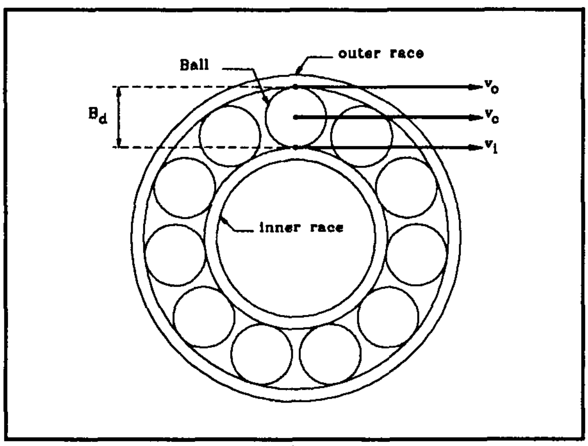


Fig. 5-1. Axial View of Bearing.

Fundamental Train Frequency

The train or cage frequency is equivalent to the angular velocity of the individual ball centers. From Fig. 5-1, the linear velocity of each ball center can be described as,

$$v_c = \frac{v_i + v_o}{2} \quad (1)$$

Angular velocity (ω) is defined as the linear velocity (v) about a radius (r) or,

$$\omega = \frac{v}{r} \quad (2)$$

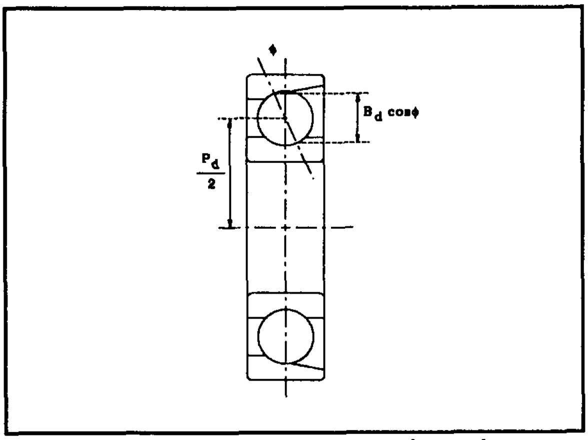


Fig. 5-2. Cross-sectional View of Bearing.

Therefore,

$$\omega_c = \frac{\frac{(v_i + v_o)}{2}}{\frac{P_d}{2}} \tag{3}$$

where ω_c is the angular velocity of the ball center or cage. Since $v = \omega\, r$ from Equation 2, ω_c can be expressed as Equation 4, where ω_i and ω_o are the angular velocities of the inner and outer bearing races. The quantity ω_c is also known as the fundamental train frequency.

$$FTF = \frac{\omega_i \left(\frac{P_d}{2} - \frac{B_d\cos\phi}{2}\right) + \omega_o \left(\frac{P_d}{2} + \frac{B_d\cos\phi}{2}\right)}{P_d}$$

$$= \frac{\omega_i}{2} - \frac{\omega_i B_d\cos\phi}{2P_d} + \frac{\omega_o}{2} + \frac{\omega_o B_d\cos\phi}{2P_d} \tag{4}$$

$$= \frac{1}{2}\left[\omega_i \left(1 - \frac{B_d\cos\phi}{P_d}\right) + \omega_o \left(1 + \frac{B_d\cos\phi}{P_d}\right)\right]$$

Ball Pass Frequency of Outer Race

The ball pass frequency of the outer race is defined as the frequency of the balls passing over a single point on the outer race. The BPFO can be described as the number of balls multiplied by the difference frequency between the cage and the outer race or,

$$BPFO = N_b \left| \omega_c - \omega_o \right| \tag{5}$$

which can be rewritten, using Equation 4, as Equation 6 where N_b is the number of balls.

$$\begin{aligned} BPFO &= \left| N_b \left(\frac{1}{2} \left[\omega_i \left(1 - \frac{B_d \cos\phi}{P_d}\right) + \omega_o \left(1 + \frac{B_d \cos\phi}{P_d}\right) \right] - \omega_o \right) \right| \\ &= \left| N_b \left(\frac{\omega_i}{2} - \frac{\omega_i B_d \cos\phi}{2P_d} + \frac{\omega_o}{2} + \frac{\omega_o B_d \cos\phi}{2P_d} - \omega_o \right) \right| \\ &= \left| N_b \left[\left(\frac{\omega_i - \omega_o}{2}\right) - \left(\frac{\omega_i - \omega_o}{2}\right) \frac{B_d \cos\phi}{P_d} \right] \right| \\ &= \left| \frac{N_b}{2} (\omega_i - \omega_o) \left(1 - \frac{B_d \cos\phi}{P_d}\right) \right| \end{aligned} \tag{6}$$

Ball Pass Frequency of Inner Race

The ball pass frequency of the inner race is defined as the frequency of the balls passing over a single point on the inner race. The BPFI can be described as the number of balls multiplied by the difference frequency between the inner race and the cage or,

$$BPFI = N_b \, | \omega_i - \omega_c | \tag{7}$$

which can be rewritten, again using Equation 4, as Equation 8.

$$\begin{aligned} BPFI &= \left| N_b \left(\omega_i - \frac{1}{2} \left[\omega_i \left(1 - \frac{B_d \cos\phi}{P_d}\right) + \omega_o \left(1 + \frac{B_d \cos\phi}{P_d}\right) \right] \right) \right| \\ &= \left| N_b \left(\omega_i - \frac{\omega_i}{2} + \frac{\omega_i B_d \cos\phi}{2P_d} - \frac{\omega_o}{2} - \frac{\omega_o B_d \cos\phi}{2P_d} \right) \right| \\ &= \left| N_b \left[\left(\frac{\omega_i - \omega_o}{2}\right) + \left(\frac{\omega_i - \omega_o}{2}\right) \frac{B_d \cos\phi}{P_d} \right] \right| \\ &= \left| \frac{N_b}{2} (\omega_i - \omega_o) \left(1 + \frac{B_d \cos\phi}{P_d}\right) \right| \end{aligned} \tag{8}$$

Ball Spin Frequency

The angular velocity of a ball about its center can be expressed in two different ways. First, considering the linear velocity of a point on the inner race in contact with the ball surface, the linear velocity (v_b) of a point on the ball surface is given as

$$v_b = (\omega_i - \omega_c) \, r_i \tag{9}$$

where r_i is the radius of the inner race. The ball angular velocity or ball spin frequency is then

$$BSF = \left| (\omega_i - \omega_c) \frac{r_i}{r_b} \right| \tag{10}$$

or, from Fig. 5-2,

$$BSF = \left| (\omega_i - \omega_c) \left[\frac{\dfrac{P_d - B_d \cos\phi}{2}}{\dfrac{B_d}{2}} \right] \right| \tag{11}$$

where r_b is the radius of the ball. Equation 11 can also be rewritten, using Equation 4, as Equation 12.

$$\begin{aligned} BSF &= \left| \left[\omega_i - \frac{1}{2} \left(\omega_i \left(1 - \frac{B_d \cos\phi}{P_d}\right) + \omega_o \left(1 + \frac{B_d \cos\phi}{P_d}\right) \right) \right] \left(\frac{P_d - B_d \cos\phi}{B_d}\right) \right| \\ &= \left| \left(\omega_i - \frac{\omega_i}{2} + \frac{\omega_i B_d \cos\phi}{2P_d} - \frac{\omega_o}{2} - \frac{\omega_o B_d \cos\phi}{2P_d}\right) \left(\frac{P_d - B_d \cos\phi}{B_d}\right) \right| \\ &= \left| \left(\frac{\omega_i - \omega_o}{2}\right) \left(1 + \frac{B_d \cos\phi}{P_d}\right) \left(\frac{P_d - B_d \cos\phi}{B_d}\right) \right| \\ &= \left| \left(\frac{\omega_i - \omega_o}{2}\right) \left(\frac{P_d}{B_d} - \cos\phi + \cos\phi - \frac{B_d \cos^2\phi}{P_d}\right) \right| \\ &= \left| \left(\frac{\omega_i - \omega_o}{2}\right) \left(\frac{P_d}{B_d}\right) \left(1 - \frac{B_d^2 \cos^2\phi}{P_d^2}\right) \right| \\ &= \left| \frac{P_d}{2B_d} (\omega_i - \omega_o) \left(1 - \frac{B_d^2 \cos^2\phi}{P_d^2}\right) \right| \end{aligned} \tag{12}$$

Second, considering the linear velocity of a point on the outer race in contact with the ball surface, the linear velocity of a point on the ball surface is given as

$$v_b = (\omega_c - \omega_o)\, r_o \tag{13}$$

Therefore,

$$BSF = \left| (\omega_c - \omega_o) \frac{r_o}{r_b} \right| \tag{14}$$

where r_o is the radius of the outer race. The ball spin frequency can then be expressed as Equation 15, which is exactly the same as Equation 12.

$$
\begin{aligned}
BSF &= \left| \left[\frac{1}{2} \left(\omega_i \left(1 - \frac{B_d \cos\phi}{P_d}\right) + \omega_o \left(1 + \frac{B_d \cos\phi}{P_d}\right) \right) - \omega_o \right] \left(\frac{P_d + B_d \cos\phi}{B_d} \right) \right| \\
&= \left| \left(\frac{\omega_i}{2} - \frac{\omega_i B_d \cos\phi}{2P_d} + \frac{\omega_o}{2} + \frac{\omega_o B_d \cos\phi}{2P_d} - \omega_o \right) \left(\frac{P_d + B_d \cos\phi}{B_d} \right) \right| \qquad (15) \\
&= \left| \left(\frac{\omega_i - \omega_o}{2} \right) \left(1 - \frac{B_d \cos\phi}{P_d}\right) \left(\frac{P_d + B_d \cos\phi}{B_d} \right) \right|
\end{aligned}
$$

$$
\begin{aligned}
BSF &= \left| \left(\frac{\omega_i - \omega_o}{2} \right) \left(\frac{P_d}{B_d} + \cos\phi - \cos\phi - \frac{B_d \cos^2\phi}{P_d} \right) \right| \\
&= \left| \left(\frac{\omega_i - \omega_o}{2} \right) \left(\frac{P_d}{B_d} \right) \left(1 - \frac{B_d^2 \cos^2\phi}{P_d^2}\right) \right| \qquad (15) \\
&= \left| \frac{P_d}{2B_d} (\omega_i - \omega_o) \left(1 - \frac{B_d^2 \cos^2\phi}{P_d^2}\right) \right|
\end{aligned}
$$

Application of the Bearing Formulas

Equations 4, 6, 8, and 12 are general formulas where either bearing race, or both, could be rotating. Also, the bearing can have either balls or rollers and the contact angle may be equal to zero as in deep groove ball bearings.

For the case where the outer race is stationary and the inner race is rotating,

$$\omega_o = 0, \qquad (16)$$

$$\omega_i = S, \qquad (17)$$

where S is the angular velocity or speed of the rotating system. Equations 4, 6, 8, and 12 can then be reduced to:

$$FTF = S \left(\frac{1}{2}\right) \left(1 - \frac{B_d \cos\phi}{P_d}\right), \qquad (18)$$

$$BPFO = S \left(\frac{N_b}{2}\right) \left(1 - \frac{B_d \cos\phi}{P_d}\right) \qquad (19)$$

$$BPFI = S \left(\frac{N_b}{2}\right) \left(1 + \frac{B_d \cos\phi}{P_d}\right), \qquad (20)$$

$$BSF = S \left(\frac{P_d}{2B_d}\right) \left(1 - \frac{B_d^2 \cos^2\phi}{P_d^2}\right) \qquad (21)$$

For the case where the inner race is stationary and the outer race is rotating,

$$\omega_i = 0, \quad (22) \qquad \omega_o = S \quad (23)$$

Equations 4, 6, 8 and 12 can then be reduced, in this case, to

$$FTF = S \left(\frac{1}{2}\right) \left(1 + \frac{B_d \cos\phi}{P_d}\right), \quad (24)$$

$$BPFO = S \left(\frac{N_b}{2}\right) \left(1 - \frac{B_d \cos\phi}{P_d}\right) \quad (25)$$

$$BPFI = S \left(\frac{N_b}{2}\right) \left(1 + \frac{B_d \cos\phi}{P_d}\right), \quad (26)$$

$$BSF = S \left(\frac{P_d}{2B_d}\right) \left(1 - \frac{B_d^2 \cos^2\phi}{P_d^2}\right) \quad (27)$$

The formula for BSF calculates the speed of the ball or roller. However, the frequency generated by roller speed is seldom measured. This is true because the balls or rollers are incased in the cage and between the inner and outer races. If one of the rolling elements has a defect of any kind, the defect can strike the inner and outer races and/or the front and back side of the cage alternately. This generates two times BSF because the timing for each event is exact and occurs when the roller rotates half a revolution.

All of these ideal bearing frequency formulas are based on the assumption of pure rolling contact between the rollers and races. The small error resulting from any slipping of these surfaces would produce somewhat lower values in the above equations. Also, if the bearing is turning on the shaft or in the housing, the bearing frequencies can be lower. When looseness is involved, the spectral lines at the bearing frequencies can be wide-banded. The outer race frequency may not be generated if the outer race is loose in the housing and the defect is not in the load zone.

Changes in the contact angle cause changes in the bearing frequencies. For example, with the outer race stationary and inner race rotating, if the contact angle increases, FTF, BPFO, and BSF increase and BPFI decreases. If the contact angle decreases, FTF, BPFO, and BSF decrease and BPFI increases. Careful analysis of the measured and calculated bearing frequencies using exact geometry, speed, and measurement accuracy often reveals useful information on the integrity of the bearing journal, housing, and thrust loads. This data is then used to determine when the bearing should be replaced, how long it will last, and the additional work required to prevent the new bearing from failing prematurely.

Further review of the above formulas indicates:

1. About 40% of the balls pass over a defect on the outer race each revolution, regardless of which race is rotating.

2. About 60% of the balls pass over a defect on the inner race each revolution, regardless of which race is rotating.

3. The FTF or cage frequency is about 40% of unit speed if the inner race is rotating and about 60% of unit speed if the outer race is rotating.

4. The BSF does not change regardless of which race is rotating. At first, it seems the BSF should increase if the outer race is rotating, because the cage speed increases. However, closer inspection indicates BSF must remain constant if BPFO and BPFI do not change.

In an effort to visually support the previous formulas, consider a simple eight ball bearing. Fig. 5-3 shows a bearing marked on the inner and outer races, and the cage at one ball. Assuming the inner race is rotating and the outer race is stationary, the inner race is rotated one revolution clockwise, as in Fig. 5-4.

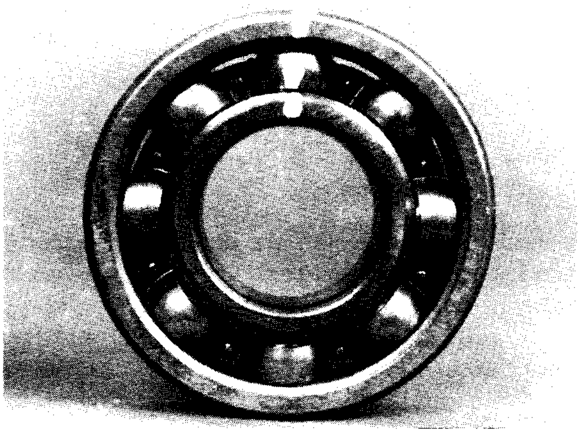


Fig. 5-3. A Ball Bearing with Eight Balls.

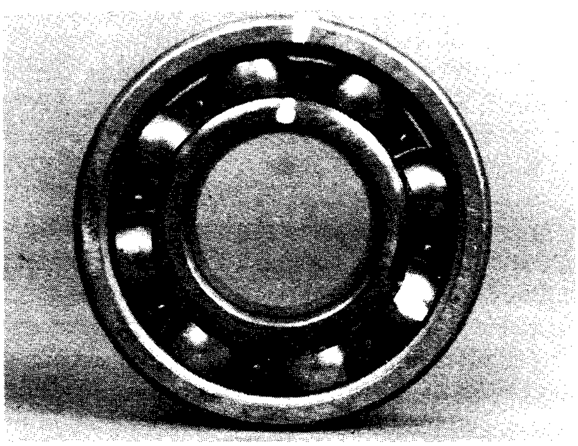


Fig. 5-4. Inner Race Rotated One Revolution.

Note that approximately five balls passed over the mark on the inner race. Since five divided by eight equals 0.62, about 60% of the balls passed over the mark on the inner race. Also note that approximately three balls, or about 40%, passed over the mark on the outer race. Finally, notice that the cage turned about 40% of one revolution.

Assuming the inner race is stationary, the outer race in Fig. 5-3 is now rotated one revolution, as shown in Fig. 5-5.

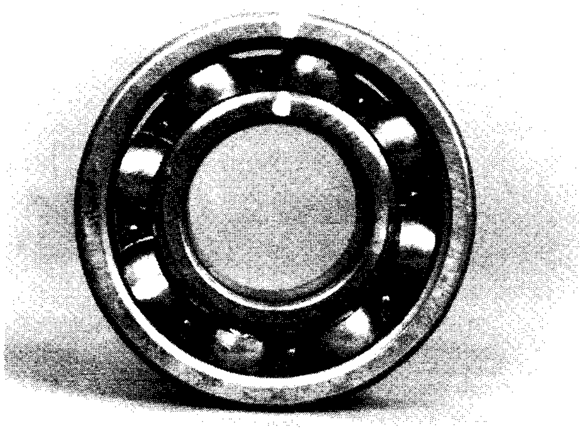


Fig. 5-5. Outer Race Rotated One Revolution.

Note that approximately three balls passed over the mark on the outer race, and approximately five balls passed over the mark on the inner race. This indicates that it makes no difference whether the inner or outer race is rotating; the ball pass frequencies are the same. Also note that the cage turned about 60% of one revolution while the outer race was rotated one revolution. These results are supported by the bearing frequency equations.

<u>Outer Race Analysis</u>

If a 6313 bearing containing eight balls is mounted with the inner race rotating at 1776 RPM or 29.6 Hz, the calculated ball pass frequency of the outer race is 90.9 Hz. The time it takes for the inner race to rotate one revolution is:

$$\frac{1}{29.6} = 0.0338 \; sec \; or \; 33.8 \; ms$$

The time for one outer race ball pass frequency is:

$$\frac{1}{90.9} = 0.011 \; sec \; or \; 11 \; ms$$

Therefore, every 11 milliseconds, a ball passes by a point on the outer race. Divide the time for one revolution by the time for one outer race ball pass frequency:

$$\frac{33.8}{11} = 3.07$$

This is the number of balls that pass the point each revolution. This value, divided by the number of balls,

$$\frac{3.07}{8} = 0.3837$$

indicates that a fraction over 38% of the balls are passing over a point on the outer race each revolution.

Inner Race Analysis

Using the above speed, the inner race frequency could be about 146.8 Hz.

$$\frac{1}{146.8} = 0.0068 \ \mathit{sec\ or}\ 6.8\ \mathit{ms}$$

This means a ball passes over a defect on the inner race every 6.8 ms. The time for one revolution is still 33.8 ms and

$$\frac{33.8}{6.8} = 4.97$$

This is the number of balls that pass over the defect each revolution. This value, divided by the number of balls,

$$\frac{4.97}{8} = 0.62$$

indicates about 62% of the balls pass over a defect on the inner race each revolution.

Ball Spin Frequency Analysis

Still using the above speed, the ball spin frequency is about 58 Hz:

$$\frac{1}{58} = 0.0172 \ \mathit{sec\ or}\ 17.2\ \mathit{ms}$$

This means it takes 17.2 ms for a ball or roller to make one revolution. Next, consider the ball is contacting the inner and outer race on two sides or curvatures, and the cage is on the other two sides or curvatures. If a ball or roller has one defect, the defect must strike the inner and outer races and/or both sides of the cage each revolution. This activity generates two times BSF. It is difficult to visualize how one times BSF can be generated.

We realize that a frequency equal to ball spin frequency can be measured in some bearings with serious defects. In such cases, the question arises as to whether these frequencies are BSF or a sum or difference frequency. When several balls/rollers have several defects, discrete frequencies may not be generated. Rather, wide or narrow banded noise is generated.

Fundamental Train Frequency Analysis

Using a speed of 29.6 Hz, the FTF is about 11.25 Hz and:

$$\frac{1}{11.25} = 0.0889 \; sec \; or \; 88.9 \; ms$$

This means it takes 88.9 ms for the cage to make one revolution. Several observations should be made:

1. The FTF is about 40% of unit speed if the inner race is rotating and about 60% of unit speed if the outer race is rotating.

2. The FTF cannot be generated in sufficient amplitude (within prescribed calibration standards) to observe unless there is an internal problem with the bearing, i.e. looseness, defects on balls/rollers, and/or missing rollers. Therefore, when the cage frequency is present, the problem may be serious.

3. A discrete frequency at FTF can be generated when looseness occurs, primarily in spherical roller bearings, and when balls/rollers are missing.

4. The FTF acts as a modulating frequency and produces sidebands to other frequencies when one or more balls/rollers have defects and severe looseness.

5. Because of the above, any measurable amplitude of FTF could mean imminent failure. This is with due respect to calibration standards.

VCI Bearing Calculation Program

The above bearing frequencies can be calculated by hand using internal geometry or the percentage method. However, most everyone is using a computerized calculation program. Most of these programs calculate bearing frequencies, harmonics, and roll speed. In addition to the above, the **Bearing Calculation Program** has the following distinguishing features:

1. A cross reference from one bearing manufacturer to another to compare frequencies

2. Calculates new bearing frequencies based on change in the contact angle

3. Calculates contact angle based on measured BPFO and BPFI, the contact angle identifies the amount of thrust load

4. Calculates two times BSF

5. Calculates FTF based on outer or inner race rotation

These features improve diagnostics, identify related problems such as looseness, provide the amount of thrust load, and help prevent future premature failures if all problems are repaired or solved. Far too many predictive maintenance programs are satisfied to identify a defect in a bearing at the last minute and replace the bearing. Managers and supervisors at all levels should insist that predictive maintenance identify all related problems so complete repairs can be made. Still, other programs operate with a bearing "mind-set." They are only concerned with identifying bearing defects and pay little or no attention to the plurality of other problems identified in this book.

BEARING DEFECTS

Defects in antifriction bearings can occur on the raceways, the rolling elements, the cage, or any combination. Such defects generate unique vibration signals. It is helpful to know the type of bearing installed because different types of bearings can generate different signals depending upon loading, internal clearance, and construction. For ball, cylindrical roller, and other bearings with a zero degree contact angle, defects on raceways can be identified by a narrow banded spectral line at the ball pass frequency of the race on which the defect exists. For spherical and tapered roller and ball bearings that have a contact angle, a defect on the outer race generates the fundamental BPFO and harmonics. The harmonics are generated because a large area of the outer race is in the load zone. In some bearings, 360 degrees of the outer race is in the load zone. The more harmonics generated by a fatigue spall, the larger the spall. Therefore, defect length can be determined by harmonic content for shallow flaking fatigue spalls.

Harmonics cannot be used to determine defect length for deep fatigue spalls that generate pulses, acid etching, corrosion, and fluting.

Defects on the inner race of ball and cylindrical roller bearings behave similar to outer race defects in that the fundamental BPFI and harmonics are generated and the harmonic content can be used to approximate defect size. Inner race defects in spherical and tapered roller bearings can generate a unique signal because these bearings have clearance opposite the load zone. Depending on the size of the bearing, the clearance varies from about 0.003" to over 0.014". If the inner race is rotating, the rollers may stop rolling when the defect is out of the load zone. The BPFI is only generated while the defect is in the load zone during each revolution.

Raceways

Outer Race

The spectrum in Fig. 5-6 was taken from an electric motor drive end, radial direction. The motor has a NU319 cylindrical roller bearing installed on the drive end. This bearing has 14 rollers. Bearings from different manufacturers can have a different number of rollers. This creates an occupational hazard, and everyone should be so advised. The **VCI Bearing Calculation Program** helps defend against this problem. Motor speed is 29.6 Hz. The calculated BPFO is:

$$BPFO = S\left(\frac{N_b}{2}\right)\left(1 - \frac{B_d}{P_d}\cos\phi\right)$$

$$BPFO = 29.6\left(\frac{14}{2}\right)\left(1 - \frac{1.024}{5.807}\times 1\right)$$

$$BPFO = 207.2 \times 0.8237 = 170.67$$

The frequency from the Bearing Program is 170.7. If the bearing had 13 rollers, the frequency would be 158.48. This type of bearing cannot have a thrust load because the inner race and rollers slip through the outer race. The frequency from the percentage method is 165.76 Hz (14 X 0.4 X 29.6 = 165.76). Note the percentage method is not very accurate at higher speeds. The percentage method is useful to test for bearing frequencies when the type of bearing and other geometry is not known.

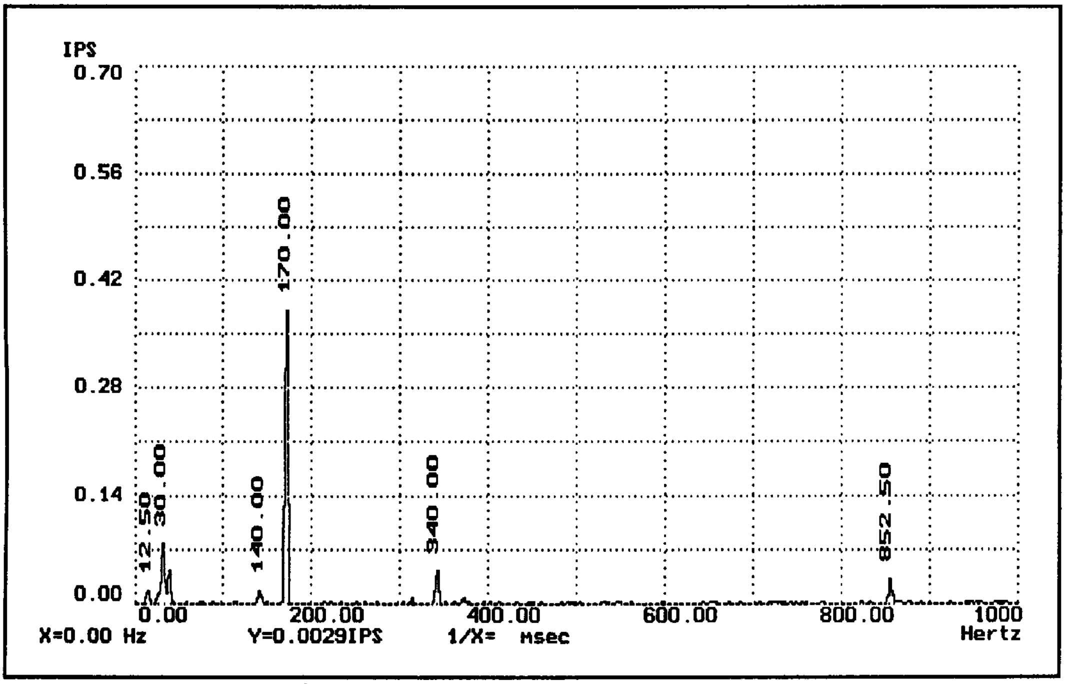


Fig. 5-6. Outer Race Defect, 200 HP Electrical Motor.

Several points should be made concerning the spectrum in Fig. 5-6:

1. The fundamental train frequency (12.5 Hz) is present with the second and third harmonics. This indicates looseness.

2. The unit speed is present and identifies residual imbalance, loading, looseness, etc.

3. The spectral line at 170 Hz is the measured BPFO. The sideband to the left at 140 Hz is the difference frequency (170 - 30 = 140 Hz) between BPFO and motor speed. This sideband indicates the defect is large enough to permit movement

of the shaft and approximates defect size.

4. The spectral line at 340 Hz is the second harmonic of BPFO. In this case, the second harmonic is probably caused by fragment denting and/or frosting and is not related to defect size.

5. The spectral line at 852.5 Hz is 5 X BPFO and is also caused by fragment denting and/or frosting.

6. A single spectral line at BPFO with an amplitude of 0.4 inches per second (IPS) normally occurs in outer race defects in bearings that have a zero degree contact angle. This is true because:

    A. The defect is always in the load zone.

    B. The signal travels through the least amount of interfaces in the outer race and the housing.

    C. The cause of failure in the outer race defect is caused by something external to the rotating unit. In this case, the "V" belts are too tight.

The photo in Fig. 5-7 contains the outer race defect that generated the spectrum in Fig. 5-6. The following points are relevant:

1. The defect is about 0.7 inches long.

2. The right side of the spall has surface cracks and shallow flaking. The left side of the spall does not have surface cracks, the spall is deep, and the edge is steep. This indicates the rollers were moving from right to left.

3. The frosting and fragment denting to the left of the spall is heavier than to the right. This also indicates the rollers are moving from right to left.

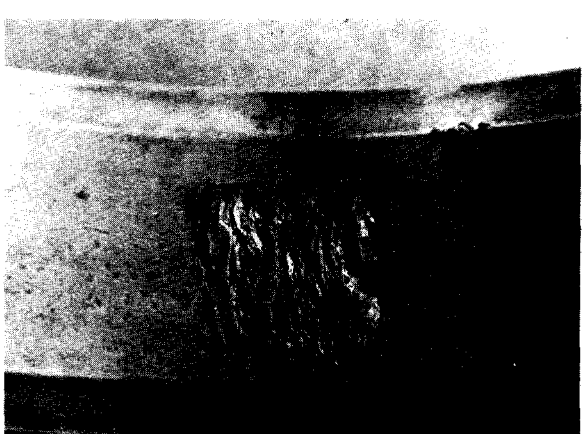


Fig. 5-7. Photo of Outer Race Defect. 200 HP Electric Motor.

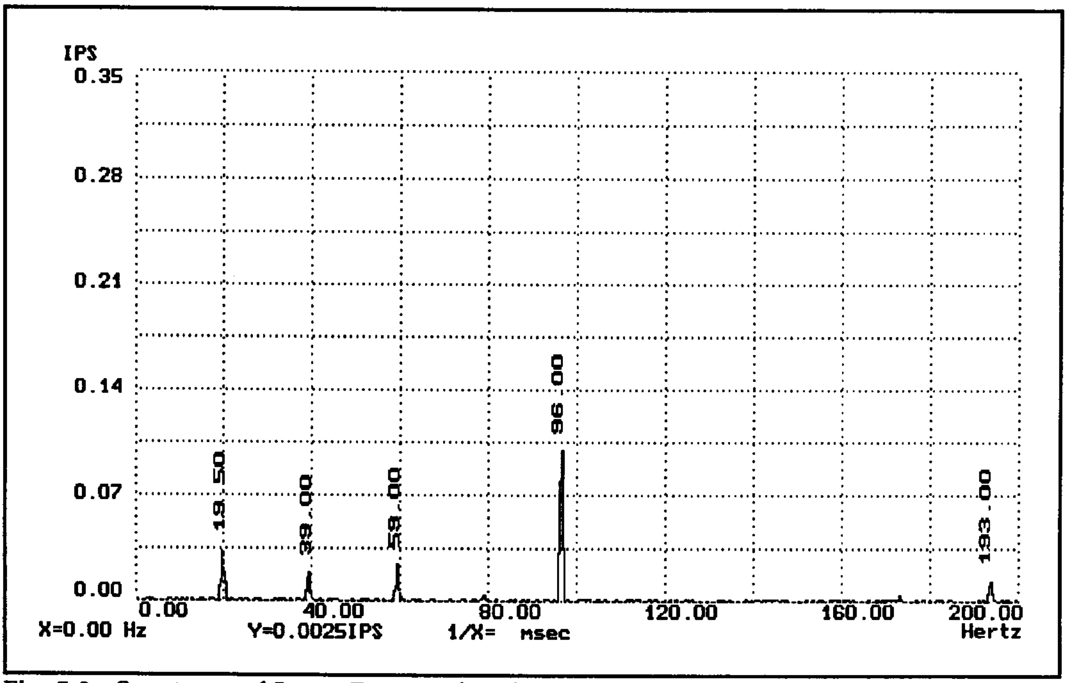


Fig. 5-8. Spectrum of Inner Race Defect from 300 HP Motor.

Inner Race

The spectrum in Fig. 5-8 was generated by an inner race defect in a 6313 bearing. The bearing was installed in the off end of a 300 HP motor. The motor speed generated a frequency at 19.6 Hz. The calculation for BPFI is:

$$BPFI = S\left(\frac{N_b}{2}\right)\left(1 + \frac{B_d}{P_d}\cos\phi\right)$$

$$= 19.6\left(\frac{8}{2}\right)\left(1 + \frac{0.937}{4.035} \times 1\right)$$

$$= 78.4 \times (1 + .2322) = 96.6$$

The following comments pertain to the spectrum in Fig. 5-8:

1. The low level spectral lines at 19.5 Hz and harmonics indicate looseness.

2. The spectral line at 96 Hz is BPFI, and it is not modulated by speed. This indicates the spall is small.

3. The spectral line at 193 Hz is the second harmonic of BPFI and is probably caused by frosting and/or fragment denting.

Fig. 5-9 contains a photo of the inner race defect that generated the spectrum in Fig. 5-8. This bearing was installed for over ten years and is a good example of a failure at the end of bearing life. Please note this is a shallow flaking spall. The entry point at the right is

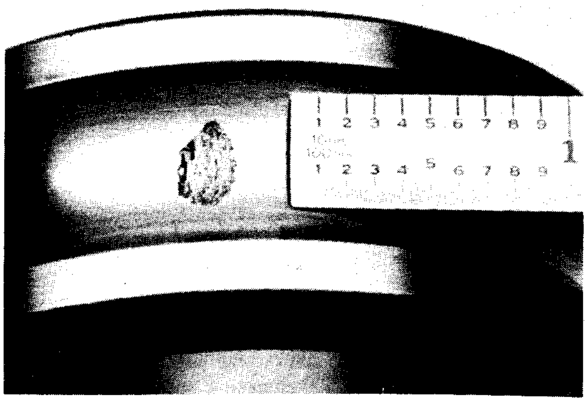


Fig. 5-9. Inner Race Defect from 300 HP Motor.

shallow and tapered. The exit point on the left is deeper and steep. Also, fragment denting and frosting are more pronounced on the left. This indicates the balls are moving from right to left. The ability to determine which way the balls or rollers are moving is useful in determining which way the bearing was installed on the shaft and the direction of thrust loads.

When a defect on the inner race is quite small, such as a cracked race or a small spall, the ball pass frequency may not be generated because at least two balls or rollers must hit the defect to generate ball pass frequency. Instead of generating the ball pass frequency, a pulse could be generated each revolution of the shaft. The basic frequency in the pulse is an excited natural frequency. The Fourier analysis of this pulse yields a series of spectral lines at harmonics of unit speed. The difference frequency between these spectral lines is the speed of the rotating unit. When this occurs, the ball pass frequency may not be present.

The time signal and frequency domain spectrum in Fig. 5-10 were taken from a 22338 bearing rotating at 120 RPM. The calculated ball pass frequency of the inner race is 17.6 Hz. The inner race of the bearing is cracked. Notice that a pulse is generated once each revolution and the ball pass frequency is not present in the spectrum. This indicates that more than one ball must hit the defect each revolution before the ball pass frequency is generated.

Depending on bearing configuration, length of load zone, and the width of the crack, a cracked inner race can generate a pulse as explained above or it can generate the BPFI and/or harmonics. Fig. 5-11 contains the frequency spectrum and time signal from a cracked inner race. The bearing is a 23260CA spherical roller bearing with 18 rollers. The speed is 0.62 Hz, BPFO is 4.795 Hz, and BPFI is 6.365 Hz.

First let's discuss the time signal. About 8,000 ms or 8 seconds are displayed. This time period contains 5 groups of pulses. The time period between pulse groups is 1.613 seconds or 1613 ms. This is roll speed. The time period between each pulse in each group is 0.157 seconds or 157 ms. This represents the time interval between rollers hitting the crack. Reading from left to right the first, fourth, and fifth groups of pulses indicate

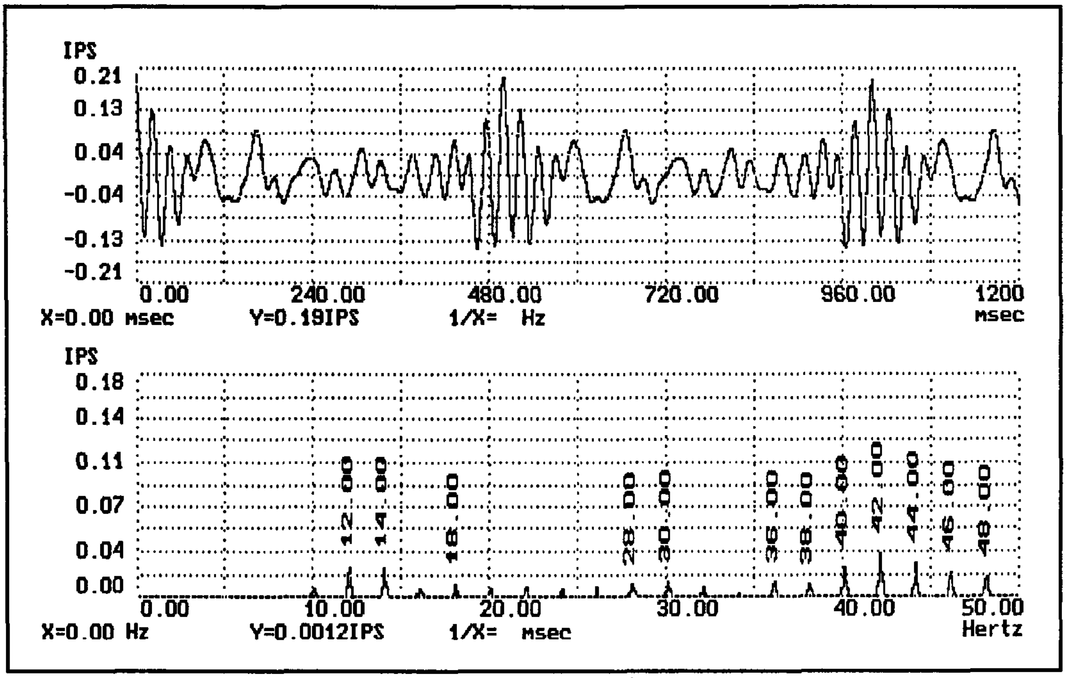


Fig. 5-10. Time Signal and Frequency Spectrum of a Cracked Inner Race.

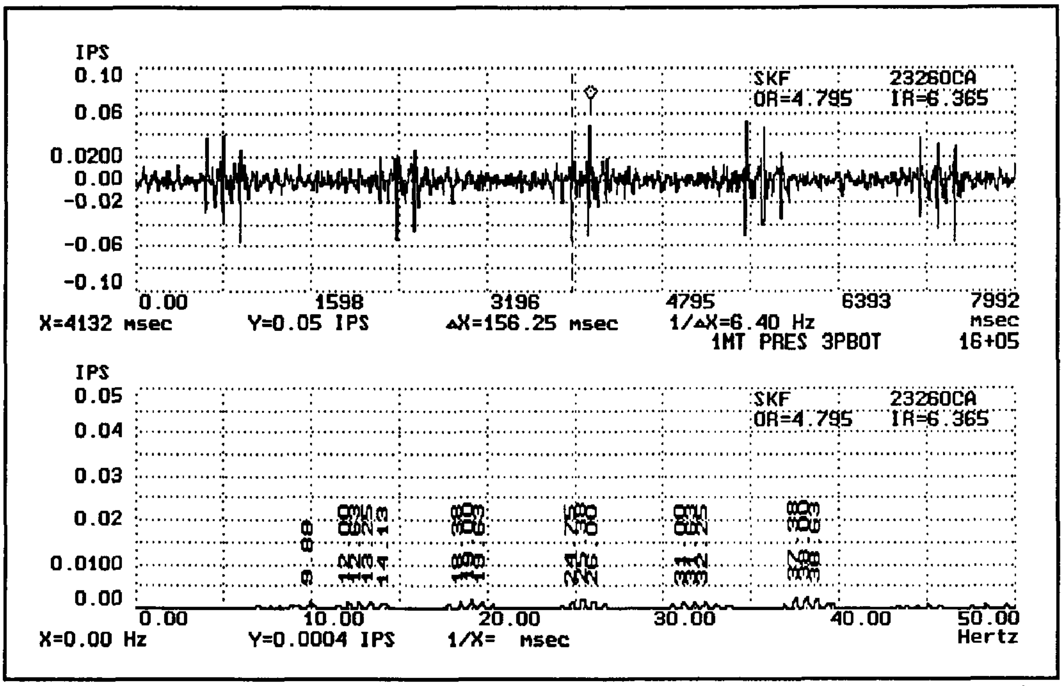


Fig. 5-11. Time Signal and Frequency Spectrum of a Cracked Inner Race, Showing BPFI.

three rollers hit the crack as the crack passed through the load zone. This is true because each of these groups contain three pulses. The second and third groups of pulses contain two pulses and indicate that two balls hit the crack when the crack passed through the load zone. The number of rollers that hit the crack when the crack goes through the load zone depends on the following.

1. Whether the crack hits a roller as it enters the load zone, or the crack enters the load zone between rollers

2. The length of the load zone

3. What happens after the crack occurs; does the crack remain open, and by how much, or does the crack close?

The Fourier analysis of the two or three pulses at BPFI, repeating once each shaft revolution, yields the frequency spectrum in Fig. 5-11. Several points should be made.

1. The fundamental BPFI is not present. This is characteristic when pulses are present.

2. Six harmonics of BPFI are present. These harmonics are modulated by shaft speed.

3. The amplitude of each harmonic is quite low, about 0.002 IPS.

4. The wide-band spectral lines and low amplitude discussed in points 2 and 3 are characteristic of a well damped pulse and have nothing to do with defect size.

5. Some of the pulses in the time domain signal are about 0.05 IPS. The overall RMS value in the frequency spectrum is only 0.008 IPS. The amplitude of the time domain signal is over six times higher than the overall RMS value (0.05 + 0.008 = 6.25). This is true because there is very little energy in the pulse, and the energy level between pulses is quite low.

6. This proves that in some areas, serious problems cannot be identified with amplitude measuring and trending overall amplitude.

7. If the amplitudes in preset frequency windows are set low enough to alert or alarm at these levels, slight amounts of noise/looseness would keep the point in alert or alarm at all times. On the other hand, if alerts and/or alarms are not set, the low amplitude serious problems can go undetected. THIS IS THE REASON WHY ACCURATE DIAGNOSIS MUST BE MADE AT EVERY POINT, AND ALERTS/ALARMS BASED ON AMPLITUDE ARE NOT ADEQUATE.

8. Alerts and alarms must be made on diagnosed problems.

9. If the time domain signal is not used in every case/point, serious problems can go undetected with normal calibration standards.

Outer and Inner Race Amplitude

Please note the amplitude of the BPFI in Fig. 5-8 is about 0.1 IPS. The amplitude of the BPFO in Fig. 5-6 is about 0.4 IPS. The outer race defect is larger than the inner race defect, however, the amplitudes of outer and inner race defects that are the same size can vary greatly. The author has seen inner race defects on 600 HP electric motors cause failure while the amplitude remained below 0.1 IPS until the rotor hit the endbell on the way out of the housing. The reasons for the low amplitude of inner race defects include:

1. Transfer function. When the balls hit a defect on the outer race, the vibration must travel through the outer race, the mechanical interface between the outer race and the housing, the housing, the mechanical interface between the housing and the transducer, and finally be converted into an electrical signal. When the defect is on the inner race, the vibration must go through the oil film between the inner race and the ball/roller, the ball/roller, the oil film between the ball/roller and the outer race, the outer race, the interface between the outer race and housing, the housing, the interface between the housing and transducer, and then converted into a signal. The net result is that the amplitude of the inner race defect is attenuated more than the amplitude of the outer race defect.

2. Loading. When a defect occurs on the outer race, it is in the load zone all the time and the BPFO is generated all the time. The result is the amplitude of an outer race defect can be quite high, over 0.5 IPS. When the defect is on the inner race, the defect may be in the load zone only once each revolution. When the defect is out of the load zone, the amplitude of the BPFI is smaller. In spherical roller and similar bearings, BPFI may not be generated at all when the defect is out of the load zone. This is true because some bearings can have up to 0.014" clearance and the rollers may stop rolling when out of the load zone. When the BPFI is generated only while the defect is in the load zone once each revolution, the Fourier analyzer yields BPFI plus harmonics, modulated by rotating unit speed.

Modulation of Ball Pass Frequency

A single spectral line can be generated at the ball pass frequency when the defect is not large enough to permit movement of the rotating unit. When the defect becomes large enough to allow movement of the rotating unit, the ball pass frequency becomes modulated with the speed of the rotating unit. This modulation generates a sideband to the ball pass frequency. The difference between the ball pass frequency and the sideband frequency is equal to the speed of the rotating unit. As the defect increases in size, more sidebands are generated and at some point, the ball pass frequency may no longer be generated. Instead, a series of spectral lines, the difference frequency of which is equal to the speed, is generated by various movements of the rotating unit during each revolution. This phenomenon occurs when the length of the defect is greater than the length required to generate one or two ball pass frequencies. This applies to both radial and axial loads.

Rolling Elements, Balls, and Rollers

When a bearing has a defective rolling element, two times ball spin frequency can be generated if the ball or roller is rotating in a manner that permits the defect to strike the outer and inner races. This frequency usually must pass through more mechanical interfaces and may not be measurable in some cases. Defects on balls strike or catch on the cage regardless of their rotation, resulting in generation of the fundamental train frequency. The fundamental train frequency generated by defective balls or rollers is seldom presented as a discrete frequency. Frequencies generated in this manner modulate other frequencies present, such as ball pass frequency, rotating unit frequency, etc. Intense modulation by the fundamental train frequency and wide-banded noise usually indicates that several balls or rollers are defective. Fig. 5-12 is a spectrum taken from a bearing that had three defective balls. The fundamental train frequency is about 11.4 Hz. If the BPFO or BPFI is present, the FTF can modulate these frequencies. In serious conditions such as Fig. 5-12, the FTF can excite one or more natural frequencies. When this occurs, the natural frequencies are modulated by the source of excitation, the FTF in this case. The band of frequencies around 243 Hz is an excited natural frequency. The difference frequency between the spectral lines equals the FTF. Balls in a bearing may never generate BSF or 2 x BSF because balls roll in one direction and spin in the other direction. This action virtually prohibits a defect on a ball from hitting anything with the repeatability required to generate one or two times BSF. The result is modulation of FTF, and wide-banded noise is generated. Fig. 5-13 is a photo of the defective balls.

Ball spin frequency is generated when a defect on the ball or roller strikes the raceway. The frequency generated, in most all cases, is two times the BSF because the defect strikes both races during each revolution of the ball or roller. The RMS value of the frequency generated is not very high because the ball is not always in the load zone when the defect strikes the race. For example, the ball or roller goes through the load zone once each

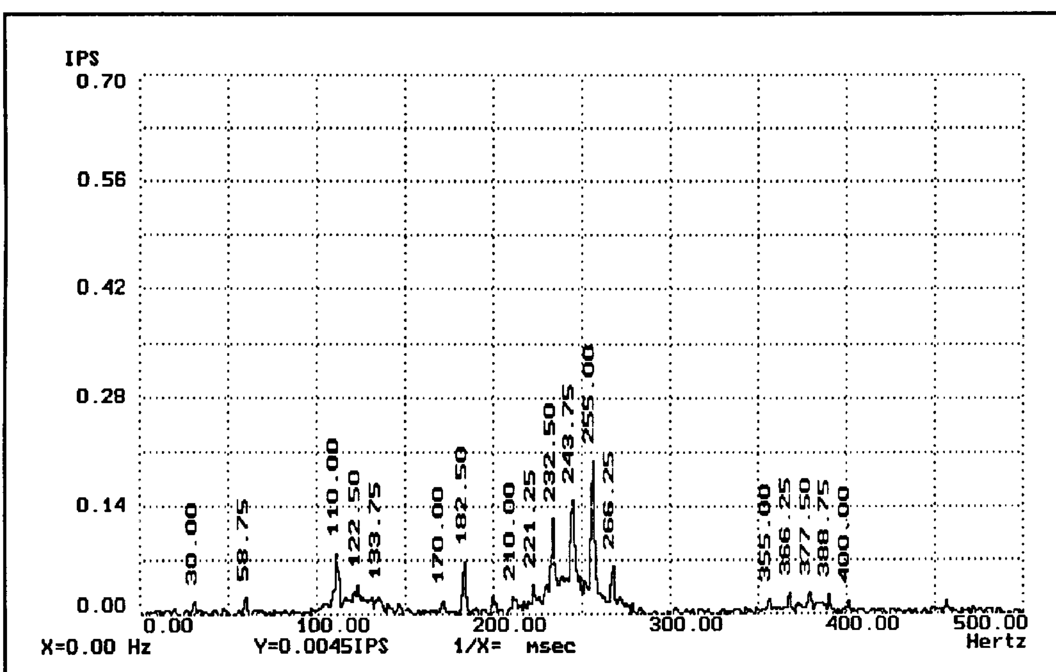


Fig. 5-12. Spectrum Taken from Electric Motor, Bearing Has Defective Balls.

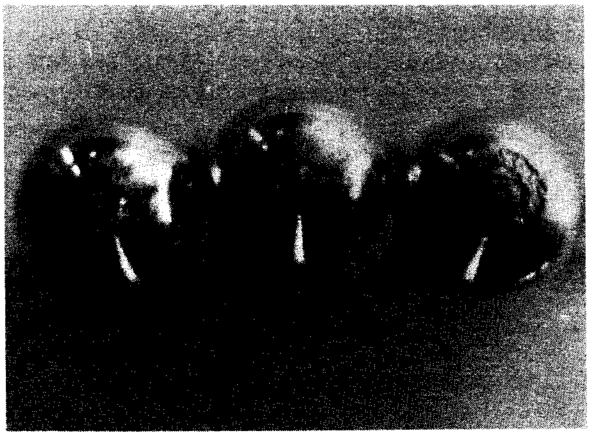


Fig. 5-13. Defective Balls That Generated the Spectrum in Fig. 5-12.

revolution of the cage. The BSF may not be generated when the balls/rollers are out of the load zone. The Fourier analysis in such cases yields modulation by the cage or FTF. Also, energy is lost because the signal passes through additional structural interfaces when the defect strikes the inner race. Cage frequency can also be generated when the balls are thrusting against the cage or the cage is broken.

The appearance of ball spin frequency seldom occurs. Two times ball spin frequency occurs when one or more rollers have one spall. When balls or rollers have several spalls, wide-banded noise is generated. Normally, defects on the balls or rollers are accompanied by a defective inner race and/or outer race. In such cases, the outer or inner race frequency can be modulated by cage frequency.

There are three major reasons why ball spin frequency is not measured very often. First, a defect on a ball may not generate ball spin frequency because the ball rotation may be such that the defect does not strike the races as it rotates. Second, a roller often gets defects all the way around the surface or at least around a majority of the surface.

When this occurs, ball spin frequency is not generated. Instead, noise is generated due to the large defective area on the roller. Third, if a single ball or roller has a defect, the defect can only strike the raceways while the roller is in the load zone. A specific ball or roller may be in the load zone only once each revolution of the cage. The energy level generated in such cases is therefore quite low because of the relatively long time period between defect strikes.

Fig. 5-14 contains the frequency spectrum and the time signal generated by a defective spherical roller. In the frequency spectrum there are several groups of spectral lines. The difference frequency between these spectral lines is 2.5 Hz. This is the fundamental train frequency. The difference frequency between the groups of spectral lines is 32.5 Hz. This is two times ball spin frequency. The time domain signal is more helpful in explaining how this signal was generated. There are two groups of pulses in the time signal. The time period between each group of pulses is 404.8 ms or 2.47 Hz. This is the time it takes for the cage to make one revolution, or the cage frequency. The time period between each pulse is 30.4 ms or 32.5 Hz. This is two times the ball spin frequency. The time periods between each cycle in the pulse are about five or six ms and are the excited

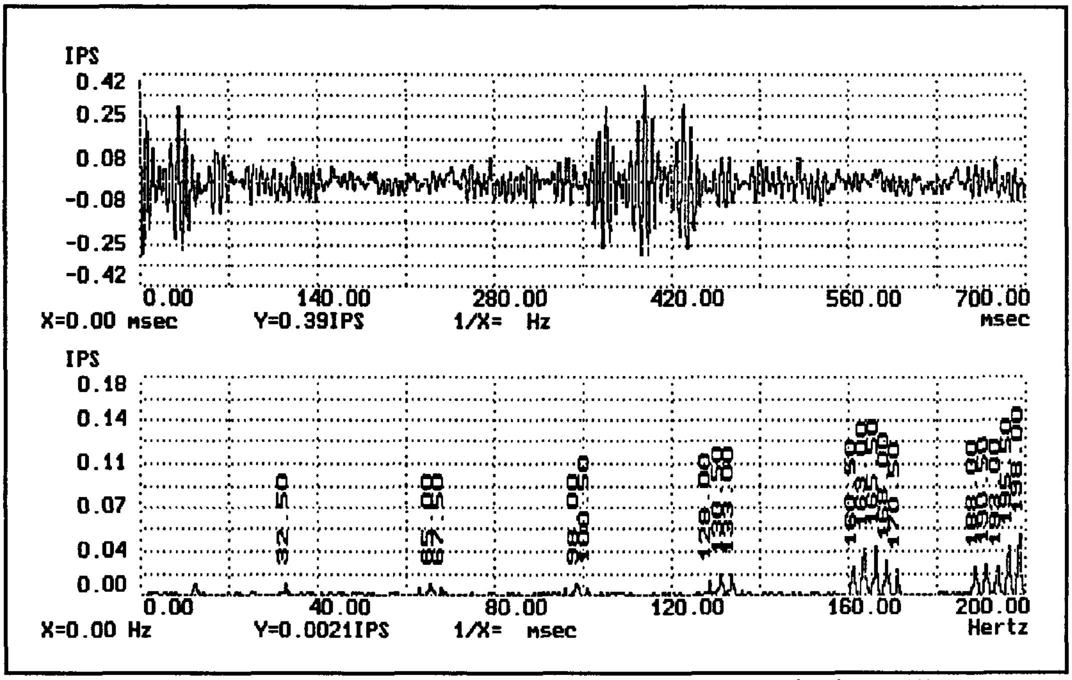


Fig. 5-14. Frequency and Time Signal Data Generated by a Defective Roller.

Fig. 5-15. Defective Roller That Generated the Data in Fig. 5-14.

natural frequencies of 195 and 165 Hz.

Several conclusions can be drawn from this data. First, there may be one roller that has a defect. When this roller is not in the load zone, the defect does not strike either race. However, when the roller goes into the load zone, the defect will strike each race alternately. When the defective roller clears the load zone, the defect does not strike a raceway until the roller starts to enter the load zone again. The period of this event is one revolution of the cage.

If the bearing is in a thrust load, the roller could be in the load zone for up to 360 degrees of each revolution. In such cases, the defect could strike the raceways several times. This would increase the energy level, and the vibration amplitude of the frequency domain spectra would increase.

Fig. 5-15 is the defective roller that generated the data in Fig. 5-14. The bearing had been installed for two weeks when the defect developed. The best explanation for a premature failure of this nature would be an oversized roller or a metallurgical flaw in the roller. This theory is supported by field reports of oversized rollers in bearings and observed defects on rollers in new bearings.

Cage

The fundamental train frequency is the rotating speed of the ball or roller cage assembly. This frequency is not encountered very often, but it can occur when some defect affects the rotation of the train. Following are some examples of problems that can cause the generation of the fundamental train frequency. These are in addition to other conditions already discussed.

1. In rare cases when one or more rollers are missing from a bearing, the FTF can be generated. The problem occurs as a pulse at the FTF. The frequency spectra contain a series of harmonics of the FTF. The amplitude of the first harmonic is quite low, the second, third, and fourth harmonics are higher in amplitude as determined by the pulse.

2. Sometimes, attempts to lubricate sealed or shielded bearings can cause the seal or shield to deflect inward. If the cage touches the seal or shield, the FTF and/or two times FTF plus harmonics can be generated.

3. Excessive clearance in an antifriction bearing can cause the generation of a discrete frequency at the FTF and/or modulations of the FTF at rotating speed and harmonics.

Except for defects that occur in bearing components during manufacturing, the cage is usually the last component to fail. The typical failure sequence is as follows: defects form on the races, the balls, and then finally the cage. A severely damaged cage can cause constant frequency shifts that are observable with the use of a real-time analyzer. When the cage is broken in enough places to allow the balls or rollers to bunch up, wide shifts in frequencies accompanied by loud noises can occur. When these signs are present, bearing seizure is imminent. Also see the discussion on looseness concerning Figs. 5-49 and 5-50.

Multiple Defects

Frequencies generated in defective antifriction bearings can add and subtract in much the same way as frequencies add and subtract in electronic mixers and detectors. This phenomenon frequently occurs, and the vibration signatures of some defective bearings do not contain some of the basic frequencies - BPFO, BPFI, BSF, or FTF. See Fig. 5-16. Instead, the frequencies of the spectral lines equal the sum or difference of the basic

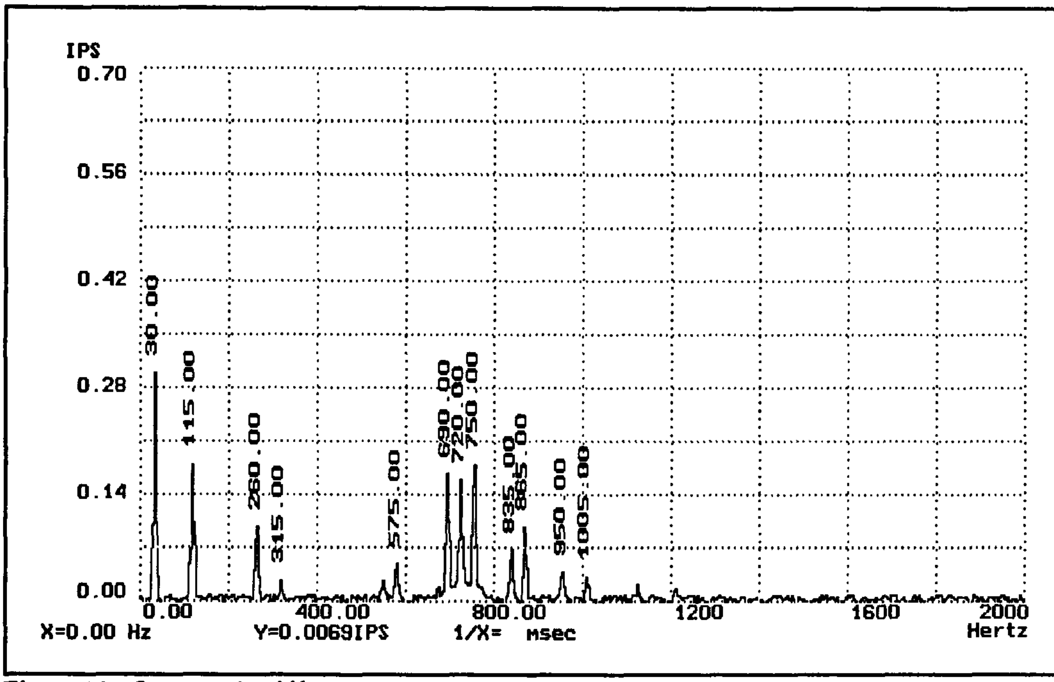


Fig. 5-16. Sum and Difference Frequencies, 300 Hp Electric Motor.

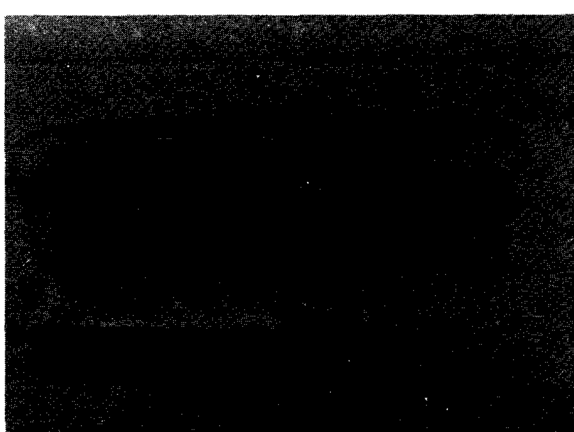


Fig. 5-17. Bearing with Defect around Inner Race.

frequencies generated by the bearing defects. Identification of the frequencies comprising each peak in the spectrum will characterize each defect in the bearing.

A single defect in a bearing can be identified by the frequency it generates. When several defects are present, some or all of them may be identified from the basic frequency, but sum and difference frequencies are almost always present in the spectra.

Analysis of complex spectra can be difficult. One approach is to first identify any basic frequencies. Multiples of the basic frequencies - 1x, 2x, etc. - must then be identified. Finally, any remaining frequency peaks are identified as combinations of the basic frequencies already identified.

The following explanation of the spectrum in Fig. 5-16 may be helpful. The defects on the inner race of the bearing in Fig. 5-17 generated these frequencies.

The bearing is a 6313 deep grove ball bearing. It was removed from a 300 HP electric motor. The bearing was in a thrust load with a contact angle of at least 15 degrees. The motor speed is 29.6 Hz and under normal conditions the following frequencies would be generated: BPFO = 91.43, BPFI = 145.37, FTF = 11.43, and 2 x BSF = 123.25. However, since the bearing has at least a 15 degree contact angle, the following frequencies were calculated using a 15 degree contact angle: BPFO = 92.35, BPFI = 144.45, FTF = 11.54, and 2 x BSF = 123.67. These frequencies are much closer to the generated frequencies.

The spectral line at 865 Hz modulated by unit speed of 29.6 Hz, is 6 x 144.2 Hz. When a size 13 ball bearing is failing from a fatigue spall, is in a thrust load, and has defects all the way around the inner race, it can generate 6 ball pass frequencies of the inner race. Therefore the spectral line at 865 Hz indicates the bearing is in a thrust load and has defects all the way around the inner race. Since we know the generated BPFI is 144.2 Hz, the **Bearings Program** indicates the average thrust load is 17 degrees. See Fig. 5-18.

The spectral line at 720 Hz is 5 x BPFI. A great amount of kinetic energy is required to generate the high amplitude modulations of 29.6 Hz and pull the spectrum up off the baseline. This indicates that the defect is very bad.

The spectral line at 260 Hz is two times BPFI minus unit speed (2 x 144.2 Hz - 29.6 Hz = 258.8 Hz). The spectral line at 115 Hz is BPFI minus unit speed (144.2 Hz - 29.6 Hz = 114.6 Hz). Finally, the spectral line at 30 Hz is unit speed.

File : SKF - Bearings

ID	Number of Balls	Contact Angle	BPFO	BPFI	FTF	2xBSF
6308	8	0	3.023	4.977	0.378	3.850
6309	7	0	2.619	4.381	0.374	3.720
6309-8	8	0	3.037	4.963	0.380	3.914
6310	8	0	3.047	4.952	0.381	3.962
6311	8	0	3.020	4.980	0.378	3.838
6312	8	0	3.064	4.936	0.383	4.040
6313	8	0	[illegible]	[illegible]	[illegible]	[illegible]
6314	8	0	3.076	4.924	0.385	4.100
6315	8	0	3.081	4.919	0.385	4.124
6316	8	0	3.086	4.914	0.386	4.146
6317	8	0	3.089	4.911	0.386	4.166
6318	8	0	3.093	4.907	0.387	4.182
6319	8	0	3.096	4.904	0.387	4.198
6320	8	0	3.073	4.927	0.384	4.082
6321	8	0	3.076	4.924	0.385	4.100

Label :

Bearing : 6313 BPFO = 91.434 BPFI = 145.366
Speed = 29.600 Contact Angle = 0 FTF = 11.426 2xBSF= 123.254

BPFI = 144.200
Contact Angle = 17

Fig. 5-18. Thrust Load Calculation Based on BPFI.

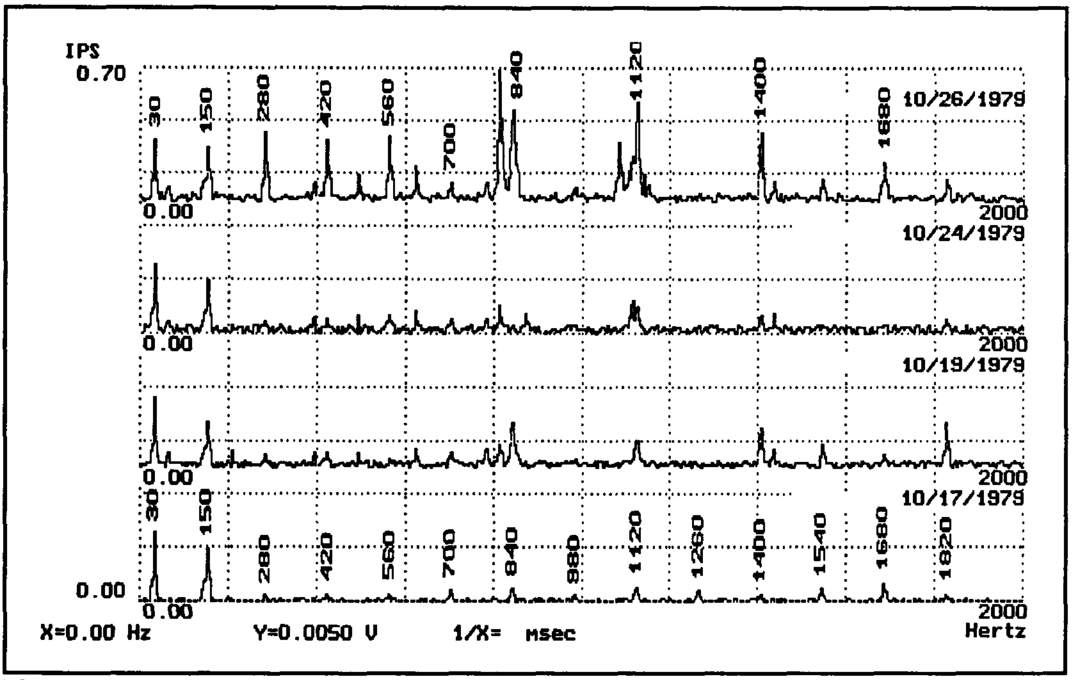


Fig. 5-19. Bearing Failure Spectrum Analysis - 10-17-79 to 10-26-79.

The best signal definition was obtained in the axial direction because of the severe thrust load. This was a premature failure and the bearings in these motors were lasting only four or five weeks.

Why were the bearings failing? When the bearings were disassembled and the inner races were placed on the workbench in the same relationship they were installed, the thrust loads on these bearings were in opposite directions. Properly installed motor bearings normally require an interference fit on the shaft and a slip fit in the housing. The lateral clearance should be sufficient to permit the motor to find its magnetic center, and allow for thermal expansion and contraction. These motors had no lateral clearance and the thermal expansion caused a thrust load on the inner race and the resulting premature failures. To solve the problem, the distance between the shoulders on the bearing journals was reduced.

This is a good example of how to improve equipment maintainability and reliability. Predictive maintenance accurately diagnosed the problem and the cause of failure. Then by working with engineering and the maintenance shops, the cause of failure was removed and the normal life expectancy was achieved. The following hints may be helpful when determining the cause of failure:

1. If a bearing is failing prematurely and the defect is on the inner race, the cause of failure is associated with the rotating unit, i.e. eccentric shaft, imbalance, or thrust load caused by the rotor.

2. If a bearing is failing prematurely and the defect is on the outer race, the cause of failure is external to the rotating unit, i.e. chip in the housing, poor surface contact in housing, tight belts, or a cocked outer race.

3. If a bearing is failing prematurely and the defects are on both races and balls/rollers, the cause of failure is common to all components, i.e. lubrication.

Progressive Bearing Failure

The following describes an actual bearing failure. A New Departure 5313 bearing was grease lubricated and installed in a vertical eight inch pump direct coupled to a 125 HP electric motor. The function of this pump was to pump 98% sulfuric acid through cooling coils. Imbalance, misalignment, cavitation, and other vibration problems normally associated with pumps were not factors in this failure.

This bearing is a double row ball bearing and each row has 12 balls with a ball diameter of 0.906 inches. The pitch diameter is 3.642 inches. The calculated BPFO is 139.3 Hz, BPFI is 215.9 Hz, FTF is 11.6 Hz, and 2x BSF is 113.5 Hz. A contact angle of 30 degrees was used in these calculations.

The bottom spectrum in Fig. 5-19 shows a frequency peak at 30 Hz, indicating a normal amount of residual imbalance. The 150 Hz peak is vane pass frequency and is not considered excessive. The low amplitude peaks at 280 through 1,820 Hz are generated by small defects on the outer race. This is because the frequency difference between the peaks is 140 Hz, which is ball pass frequency of the outer race. The low amplitude indicates that the defects are tiny and the number of harmonics indicates that the defects occur all the way around the outer race. At this point, there is no concern for immediate bearing failure.

The second spectrum in Fig. 5-19 was taken two days later. The amplitudes of the bearing frequencies are a little higher, and the vane pass frequency is also a little higher.

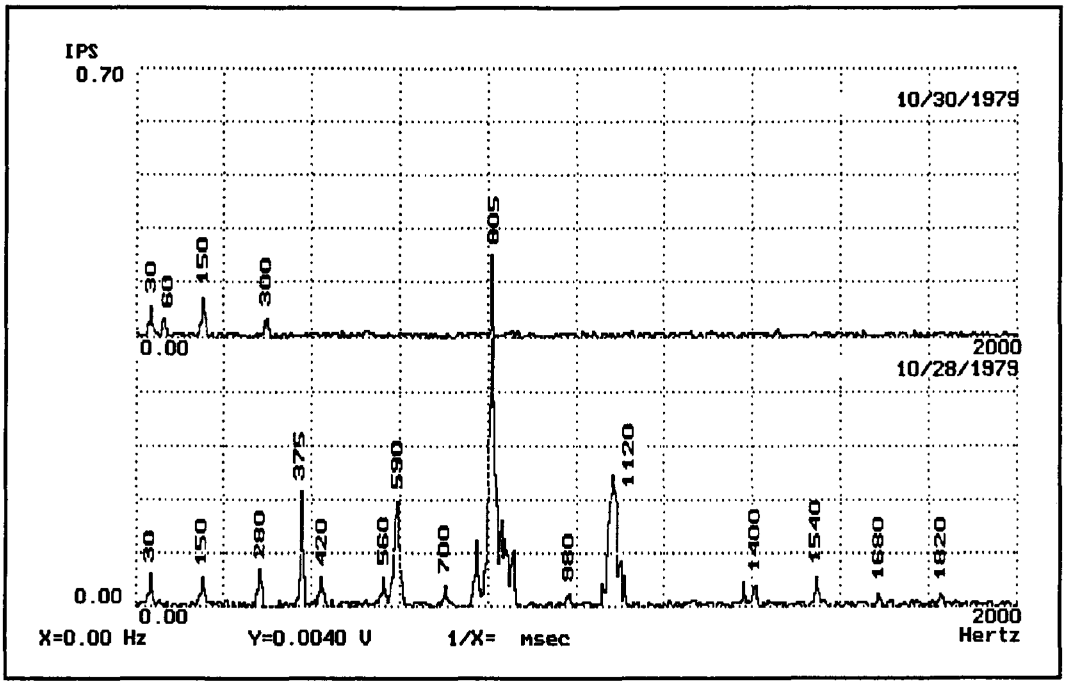


Fig. 5-20. Bearing Failure Spectrum Analysis - 10-28-79 to 10-30-79.

At this point, the defect could be slightly worse or the amplitude change could be caused by loading.

The third spectrum in Fig. 5-19 was taken five days later. The amplitudes of the bearing frequencies are higher still, but the spectral lines are no longer as distinct. The bearing frequencies are now modulated with the fundamental train frequency. The wide-banded peaks indicate that the defects on the outer race could be healing over due to the balls cold rolling the defects back into a smooth surface. Modulation by the fundamental train frequency indicates the balls may have spalls. The problem is still not very serious.

The top spectrum in Fig 5-19 was taken two days later. This spectrum indicates the overall bearing condition has deteriorated considerably. Notice the residual imbalance and the vane pass frequency have remained at about the same amplitude. However, the outer race bearing frequencies are much higher in amplitude, and modulation with the fundamental train frequency is more intense. It is interesting to note the difference frequency between 280, 420, 560, and 700 Hz is 140 Hz, or BPFO. The difference frequency between 840, 1120, and 1400 is 280 Hz, or two times BPFO. It is now clear that the bearing must be changed at the first opportunity.

The bottom spectrum in Fig. 5-20 was taken two days later. There are still defects on the outer race and balls, but new frequencies are present. The difference frequency between 375, 590, and 805 Hz is 215 Hz, or BPFI. The bearing now has defects all the way around the inner race and outer race, and severe spalls on the balls. This condition is serious. A down day was planned for two days later; the consensus was that the bearing would last for two more days.

The high amplitude, 805 Hz peak, is four times BPFI minus two times the speed of the rotating unit:

$$(4 \times 215.9\ Hz) - (2 \times 29.6\ Hz) - 804.4\ Hz$$

The most logical explanation for this peak is that since the defects are all the way around the inner race, the frequencies generated are in phase and the amplitude of the 805 Hz peak is the algebraic sum of the amplitudes of 395 Hz, 590 Hz, and the actual level of the 805 Hz frequency. This phenomenon indicates caution should be used when taking a signal amplitude at face value. In this case, the amplitude is overstated. The amplitude can be understated when the signal contains pulses with a low repetition rate, and when bearings with a radial load have a defect on the inner race. In the latter case, a high signal amplitude is generated only once each revolution, resulting in a significant difference between the peak signal and the averaged signal.

The top spectrum in Fig. 5-20 was taken with a new bearing, after the defective bearing was replaced. The overall vibration level is about 0.05 inches per second. This spectrum is characteristic of a good pump.

Fig. 5-21 is a photo of the inner race of the defective bearing. Notice the relative uniformity of the defects.

Fig. 5-22 is a photo of the outer race of the defective bearing. The defects are relatively uniform. Notice how the balls have smoothed out the defects on both races.

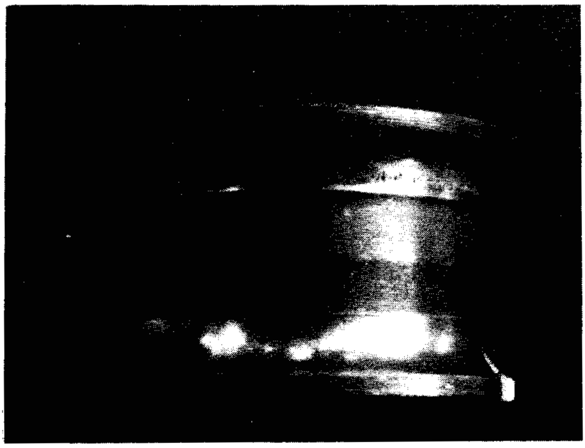


Fig. 5-21. Inner Race of Defective Bearing.

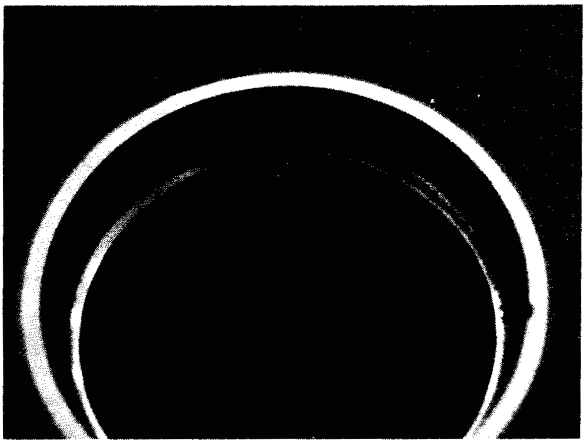


Fig. 5-22. Outer Race of Defective Bearing.

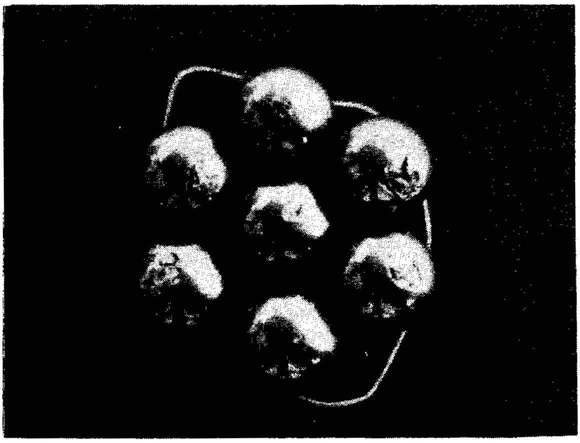


Fig. 5-23. Balls of Defective Bearing.

Fig. 5-23 is a photo of the defective balls of the bearing. Ten of the balls were spalled. These defective balls caused the cage to rattle with enough force to amplitude modulate most of the frequencies.

DEFECT SEVERITY

Bearing Behavior

A properly selected and installed angular contact bearing should have an evenly distributed thrust load on the entire surface of both races. Under such conditions the bearing typically develops small defects all the way around the race. The combination of a 360 degree load zone and multiple small defects produces a series of harmonics of ball pass frequency. The number of harmonics generated is related to the circumference of both races and/or the length of the load zone.

A properly selected and installed deep groove ball bearing should carry a radial load in about one third of the circumference. Under these conditions, the bearing develops a defect in the load zone that generates a single ball pass frequency. Two defects can generate the second harmonic of ball pass frequency. If the load -- as "felt" by the bearing -- is on the rotor, as occurs in imbalance, a spall should occur on the inner race (if the inner race is rotating and the outer race is stationary). The load zone is always at the same spot on the inner race. If the load is external to the rotor, as is the case with over-tight V belts, the defect should occur on the outer race at the load zone, because the load zone is always at the same place on the outer race.

If a thrust bearing is not evenly loaded over its circumference and has a radial load, it will behave like a radial bearing; that is, it will produce a single ball pass frequency. If a radial bearing has an abnormal thrust load, it will behave like a thrust bearing; that is, it will generate several harmonics of the ball pass frequency. Bearings that are subjected to the type of loads for which they were not designed fail quickly. Excessive loads can be identified with spectral analysis. This is accomplished by accurately measuring the generated BPFO or BPFI and then calculating the contact angle. The contact angle identifies the amount of thrust load.

A beginning defect on either race generates ball pass frequency. Early spall information can be identified when hairline cracks develop. Manufacturing defects, such as holes in the race the size of a sharp pencil point, can be identified. Ball pass frequency can be detected in unloaded machines, for example an electric motor not connected to a driven unit or a new motor on the shop floor.

After a defect has begun, it will get larger, and the spectral bandwidth will get wider until the spectrum is modulated with the speed of the rotating unit. The ball pass frequency and the ball pass frequency plus or minus the unit speed may be generated. Modulation can continue until the ball pass frequency is no longer apparent. (In some cases the amplitudes of the sum and difference frequencies are equal to or exceed those of the ball pass frequency.) The spectrum then becomes a series of frequency peaks whose difference frequency is equal to the unit speed. These phenomena occur when a growing fatigue spall is present on the race.

The limits of a significant change in vibration level depend on the type of equipment involved and the cause of the vibration. For a coupled pump and motor that normally vibrate at 0.07 IPS, a change of 0.05 IPS could be significant and should be inspected. On the other hand, a bucket elevator that normally vibrates within a range on either side of 0.5 IPS could be checked on a periodic basis or when a change of 0.2 IPS occurs. At the other extreme, a cracked inner race may not cause a noticeable change in the vibration amplitude.

Defects in antifriction bearings can be identified at 0.008 IPS; an increase of 0.1 IPS in a bearing could thus be important, whereas an increase of 0.1 IPS in an imbalance condition might cause little concern.

Excessive vibration can be defined as that level of vibration that experience has shown to be harmful to a particular piece of equipment. For a directly coupled pump and motor, excessive vibration could be 0.05 IPS at a bearing frequency. For a fan that weighs 2,000 pounds, 0.25 IPS at running speed could be excessive. At the other extreme, a 20 pound fan could vibrate at 2.0 IPS at running speed for long periods without incident.

Bearings in rotating machinery should be periodically checked with a frequency spectrum and time signal to detect and study developing defects on the outer and inner races. An accurate method for the calculation of bearing defect length is needed to allow a quantitative determination of the defect severity. With the defect size and progression of development determined, the remaining bearing life can be estimated.

Empirical measurements indicate that a 6313 bearing can generate six harmonics of BPFI if the bearing a) is in a thrust load; b) is failing from a shallow flaking spall; and c) has a defect all the way around the race. The same bearing can generate seven harmonics of BPFO if the above three conditions are met.

When deep fatigue spalls are present, pulses are generated. The FFT contains several harmonics when pulses are present. Etching, corrosion, and fluting can also cause many harmonics.

In spherical and tapered roller bearings, the length of the defect on the outer race can be approximated by harmonic content. If the bearing is failing from a shallow flaking spall, each true harmonic of BPFO equals about 0.5 inches of defect length. For example, a spherical roller bearing that is generating six harmonics of outer race ball pass frequency indicates an outer race defect about three inches long.

If the above knowledge and the following information on the inner race are used to approximate defect size, and applied to your diagnostics, it is helpful in assigning priorities for repair. For example, we may want to replace bearings with large defects before we replace bearings with small defects.

Inner Race Defect Length

Spherical roller bearings can be considered to have a predominant radial load, although a small thrust load may result from the contact angle of the rollers. This radial load is due to the weight of a rotating system and the forces applied to that system. Because of the clearance in this type of bearing and the existence of a radial load, only a small portion of the bearing actually experiences the load at any particular moment. This area along the bearing races is referred to as the load zone.

The clearance in a radially loaded spherical roller bearing allows an inner race defect to pass into and out of the load zone once each revolution of the system. (This condition applies for the case where the outer race is fixed and the inner race is rotating.) Because the bearing is loaded radially, the rollers do not come into contact with the inner race while they are not in the load zone. Therefore, pulses in the time signal generated by rollers hitting a defect only occur during the period that the defect is in the load zone. With the use of the time domain signal, the length of the defective area can be calculated. Fig. 5-24 shows a bearing with a large defect on the inner race.

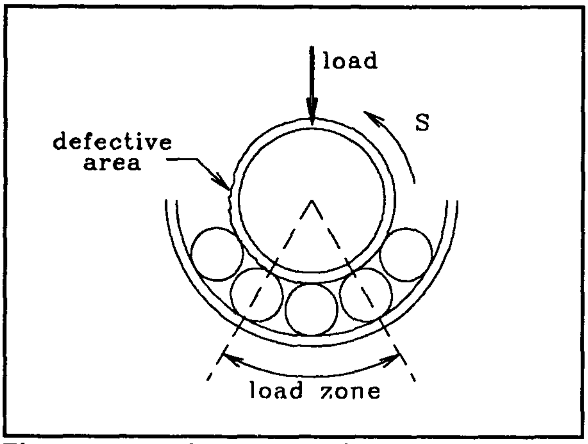


Fig. 5-24. Bearing with Defective Inner Race.

The actual length of time (t) that a defect remains in the load zone is a function of defect length (L_d), load zone length (L_{lz}) and inner race circumference (C_{ir}). This time period can be expressed as a fraction of the time required for each revolution of the inner race or,

$$t = \left(\frac{1}{S}\right)\left(\frac{L_d + L_{lz}}{C_{ir}}\right) \tag{28}$$

A significantly large defective area on the inner race will generate a time domain signal similar to that shown in Fig. 5-25 where several rollers strike the defect during each revolution.

With each revolution of the inner race, a distinct number of pulses (N_p) is generated in the time signal. The number of pulses equals the number of rollers that hit the defect while in the load zone. Because the inner race passes by the rollers at the same frequency that the rollers pass by the inner race, the number of pulses can be written as the inner race ball pass frequency multiplied by the time period that the defect is in the load zone or,

$$N_p = BPFI\,(t) = \left(\frac{N_b}{2}\right)\left(1 + \frac{B_d \cos\phi}{P_d}\right)\left(\frac{L_d + L_{lz}}{C_{ir}}\right) \tag{29}$$

Equation 29 may be rearranged to solve for the desired inner race defect length with,

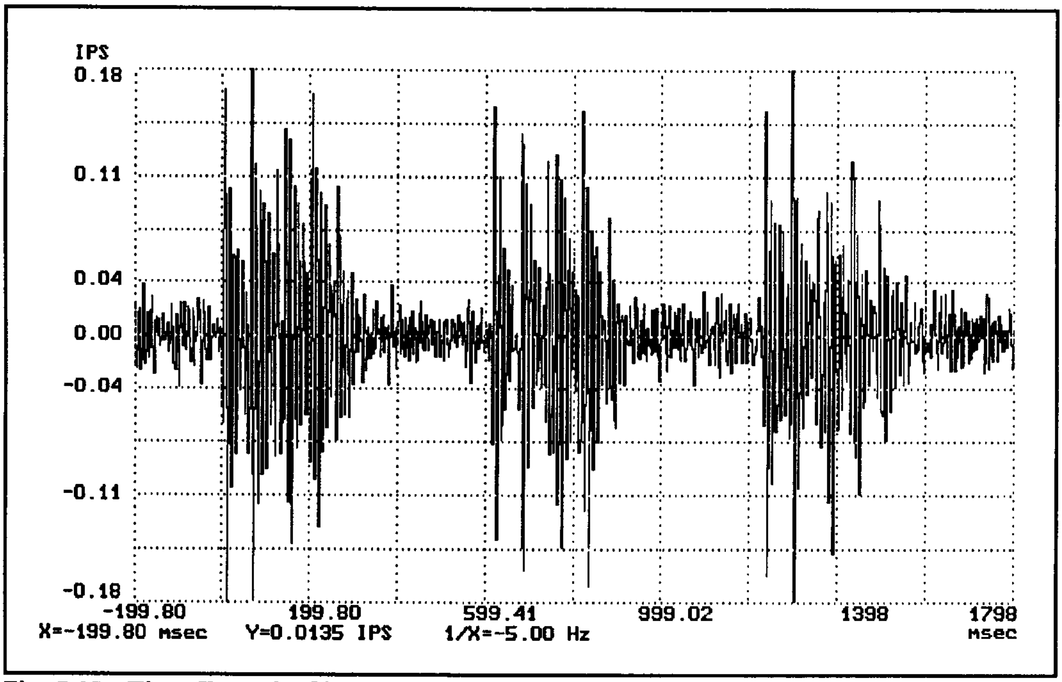


Fig. 5-25. Time Domain Signal Generated by Defect.

$$L_d = \left[\frac{N_p\,(C_{ir})}{\left(\frac{N_b}{2}\right)\left(1 + \frac{B_d \cos\phi}{P_d}\right)} \right] - L_{lz} \tag{30}$$

The defect length described by Equation 30 is the maximum value for a given number of pulses. This value can be exaggerated by as much as the length between pulses. For example, the actual defect length may be just short of that required to generate an additional pulse in the time signal. From Equation 30, the defect length that passes through the load zone between pulses (L_p) is equal to

$$L_p = \frac{1\,(C_{ir})}{\left(\frac{N_b}{2}\right)\left(1 + \frac{B_d \cos\phi}{P_d}\right)} \tag{31}$$

Therefore, the approximate length of an inner race defect is given by the inequality

$$\left[(N_p - 1)\,(L_p) - L_{lz}\right] \le L_d < \left[N_p\,(L_p) - L_{lz}\right] \tag{32}$$

In order to calculate the defect length from Equation 32, the length of the load zone must first be known or estimated. If the majority of the bearing load is assumed to be supported by three rollers, the load zone length can be expressed as twice the distance along the inner race circumference between two stationary rollers or,

$$L_{lz} = 2\left(\frac{C_{tr}}{N_b}\right) \tag{33}$$

Another estimate for load zone length can be expressed as a function of the overall clearance (c) in a bearing. Fig. 5-26 shows the inner race of a bearing resting on the inside of several rollers held together by a cage.

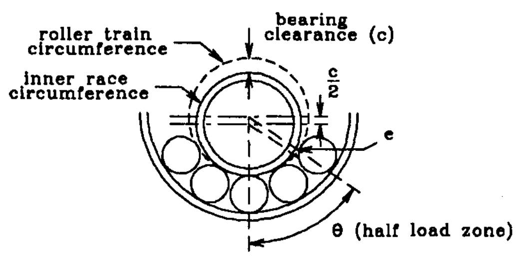


Fig. 5-26. Clearance between Inner Race and Rollers.

The load zone can be assumed to be the area along which the clearance between the inner race and the rollers is less than the oil film thickness (e).

From Fig. 5-26, the oil film thickness can be expressed as the difference between the inner radius of the roller train (r_t) and the radius of the inner race at an angle θ to give

$$\begin{aligned} e &= r_t - \left[r_i + \left(\frac{c}{2}\right)\cos\theta\right] = (r_t - r_i) - \left(\frac{c}{2}\right)\cos\theta \\ &= \frac{c}{2} - \left(\frac{c}{2}\right)\cos\theta = \frac{c}{2}\,(1 - \cos\theta) \end{aligned} \tag{34}$$

where θ is the angle defining the length of the load zone over half of the bearing circumference. Equation 34 can be rearranged to solve for the angle θ with

$$\cos\theta = \left(1 - \frac{2e}{c}\right), \quad \theta = \cos^{-1}\left(1 - \frac{2e}{c}\right) \tag{35}$$

The length (l) along a curve is defined by the angle and radius of that curve as

$$l = r\,\theta \tag{36}$$

Therefore, the length of the load zone can be expressed as a function of bearing clearance to give

$$L_{lz} = 2\ (r_i)\ \cos^{-1}\ (1 - \frac{2e}{c}) \tag{37}$$

where the factor 2 is included because the angle θ only describes half of the load zone. The oil film thickness must be specified to calculate the load zone length as a function of bearing clearance. A value on the order of 10^{-5} should give good results. The actual or true load zone length is a complex function of the loading applied and the resulting deformation of the bearing elements.

Note that the approximation for load zone length has a significant effect on the value calculated for the length of a defect. This effect is least significant when the actual load zone length is relatively small and the defect length is large in comparison. Equation 33 seems to give good results for a load zone that is spanned by a specific number of rollers. An accurate approximation for load zone length can be made using Equation 37, if the values for bearing clearance and oil film thickness are known.

If the defect is shorter than the length of the load zone, as with a small spall, Equation 32 could produce a negative value for defect length when the load zone length is subtracted. Obviously, a defect cannot have negative length and this should be accounted for in certain cases by not subtracting the load zone length in Equation 32. One case where this problem could arise would be when there is only one pulse in the time domain signal. Another case would be if there are two pulses, but the length between pulses is less than the load zone length. Again, Equation 32 is most accurate for large defects and relatively small load zones.

If the defect is large enough to be partly within the load zone at all times, this condition should also be accounted for by adjusting Equation 32. When pulses are generated throughout the entire time signal for one revolution or if the number of pulses is equal to the BPFI, then the defect covers the whole inner race and the load zone length should not be subtracted. In this case, the defect length will equal the inner race circumference.

Defect Length Calculation

This example is from a press roll bearing. This SKF 23276 bearing was rotating at a speed of 97 RPM or 1.62 Hz. Fig. 5-27 is a frequency spectrum from the bearing which indicates an inner race defect.

From Equation 20, the BPFI is shown in Equation 38.

$$BPFI = S\ (\frac{N_b}{2})\ (1 + \frac{B_d \cos\phi}{P_d}) = 1.62\ (\frac{18}{2})\ (1 + \frac{2.76\ \cos 10}{18.9}) = 16.6\ Hz \tag{38}$$

Fig. 5-28 is a time signal showing the detail of the pulses generated during one revolution. The frequency of the pulses in this figure is 16.6 Hz which is equal to the BPFI. Although there appears to be five pulses per revolution, a close examination shows that there are actually two groups of three pulses each. The second group begins just after the third pulse of the first group. This seems to indicate that two smaller defects instead of a single large one are passing through the load zone in succession. The defect length between pulses, from Equation 38, is

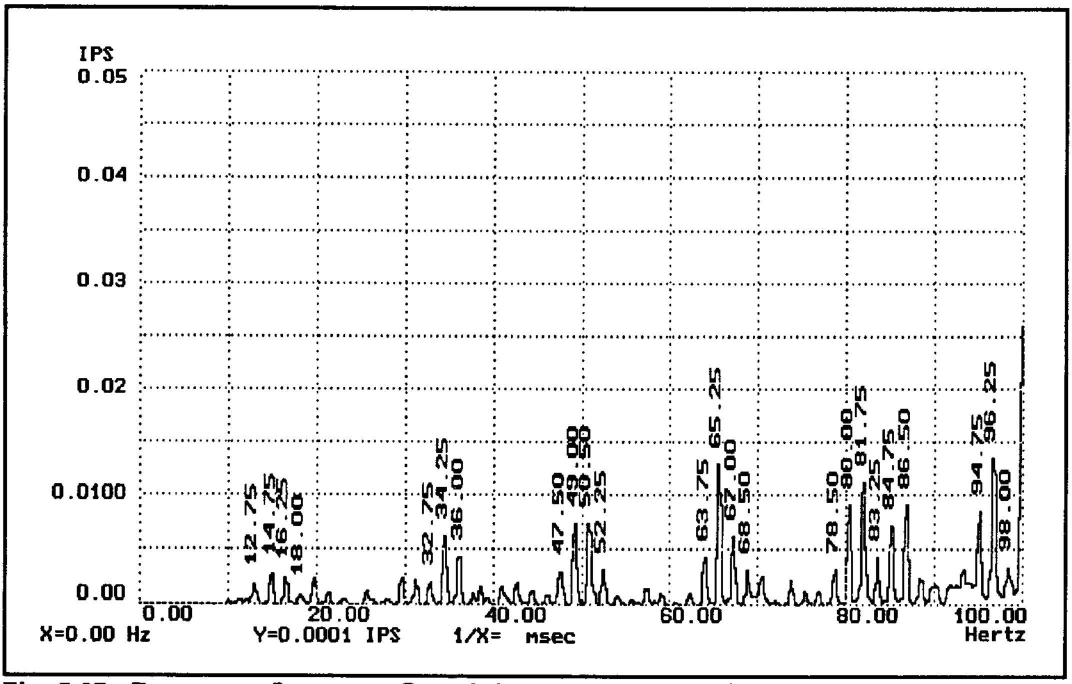


Fig. 5-27. Frequency Spectrum Containing Inner Race Defect.

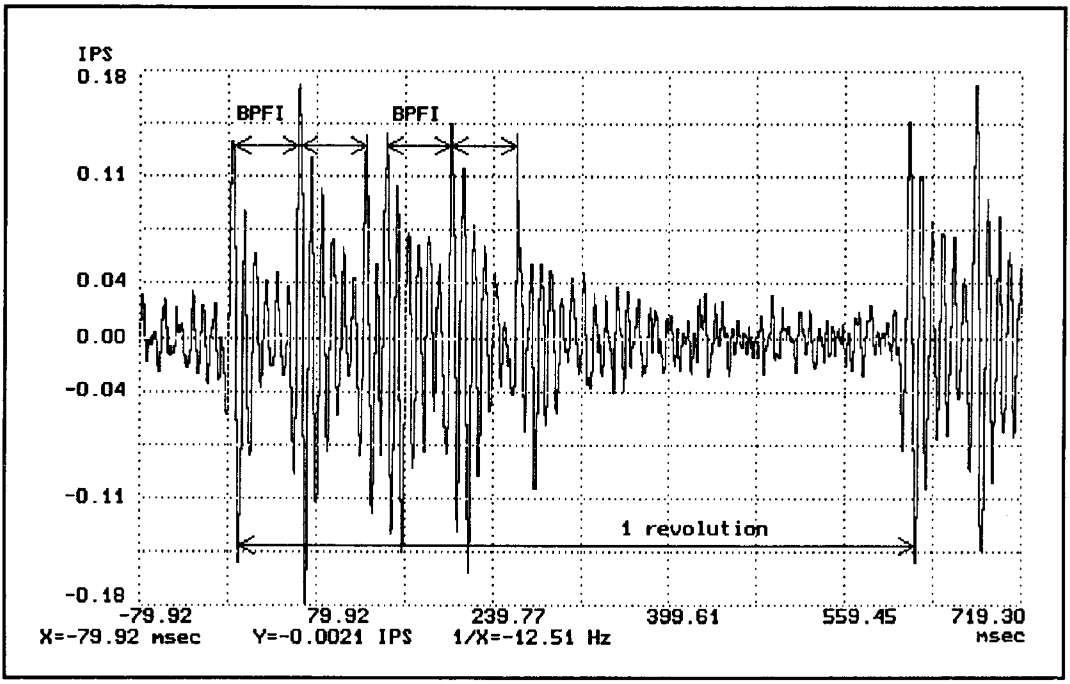


Fig. 5-28. Time Domain Signal Containing One Revolution.

$$L_p = \frac{1\ (C_{ir})}{(\frac{N_b}{2})\ (1 + \frac{B_d \cos\phi}{P_d})} \tag{39}$$

or, after substitution,

$$L_p = \frac{\pi(18.9 - 2.76)}{(\frac{18}{2})\ (1 + \frac{2.76\ cos10}{18.9})} = 4.93\ in \tag{40}$$

The load zone length is estimated using Equation 33 with

$$L_{lz} = 2\ (\frac{C_{ir}}{N_b}) = 5.63in \tag{41}$$

Finally, combining Equations 40 and 41 into Equation 32 will give the approximate defect length which generates three pulses as in Equation 42.

$$[(N_p - 1)\ (L_p) - L_{lz}] \leq L_d < [N_p\ (L_p) - L_{lz}]$$

$$[(3 - 1)\ (4.93) - 5.63] \leq L_d < [3(4.93) - 5.63] \tag{42}$$

$$or,\ \ 4.23\ in \leq L_d < 9.16\ in$$

The overall defective area consists of two defects having a size within the range approximated by Equation 41. In addition, referring to Fig. 5-28, this defective area will be similar in size to a single defect that generates five pulses in the time domain vibration signal.

<u>Deep Fatigue Spalls vs. Shallow Flaking</u>

Ball pass frequency is generated by balls or rollers passing over a defect. If the defect is a deep fatigue spall, cracked race, or a hole in the race, a pulse is generated each time a ball hits the defect. For radial loaded bearings with the defect on the inner race, only one ball may hit the defect each revolution. However, for defects on the outer race and the inner race when the bearing has a thrust load, pulses are generated at the ball pass frequency. A Fourier analysis of these pulses yields a series of spectral lines. The difference frequency between these spectral lines is the ball pass frequency, as in Fig. 5-29. When the defect is not very deep, as with shallow flaking, the balls passing over the defect generate a discrete ball pass frequency, as in Fig. 5-30.

When the nature of the defect is shallow flaking and two defects are present or one defect is of a certain length, two ball pass frequencies are generated. These frequencies are at some phase relationship depending on the distance between the two defects. At the same time, a frequency of two times the ball pass frequency is generated because twice as many balls are passing over the defective area, as in Fig. 5-31.

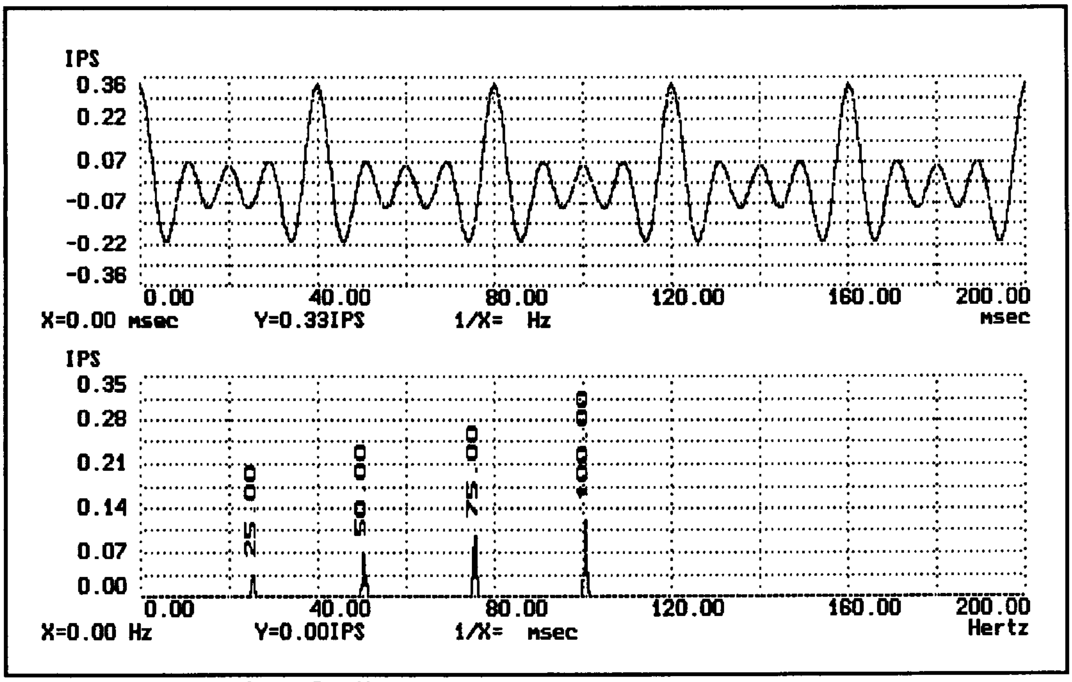


Fig. 5-29. Deep Fatigue Spall.

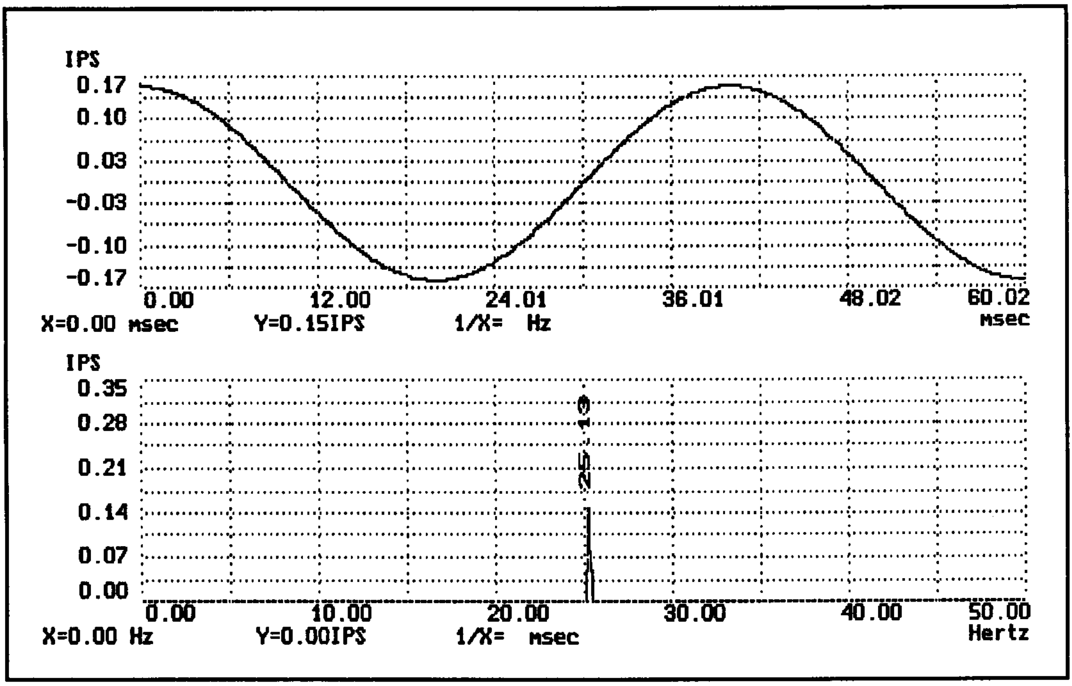


Fig. 5-30. Shallow Flaking.

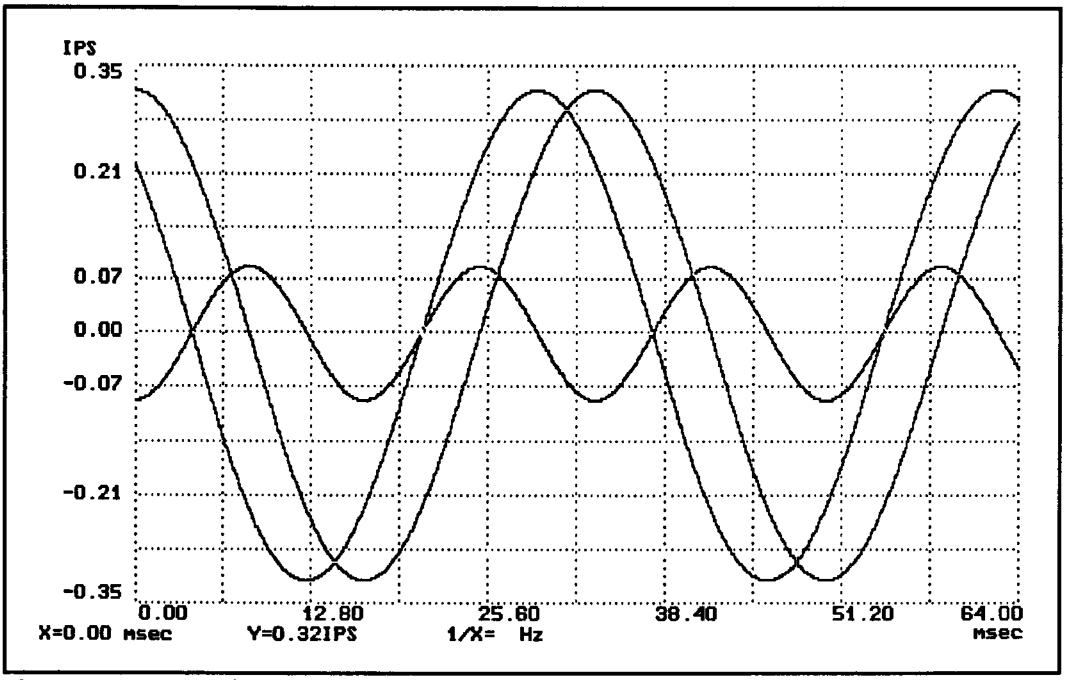


Fig. 5-31. Two Defects with Frequencies Generated.

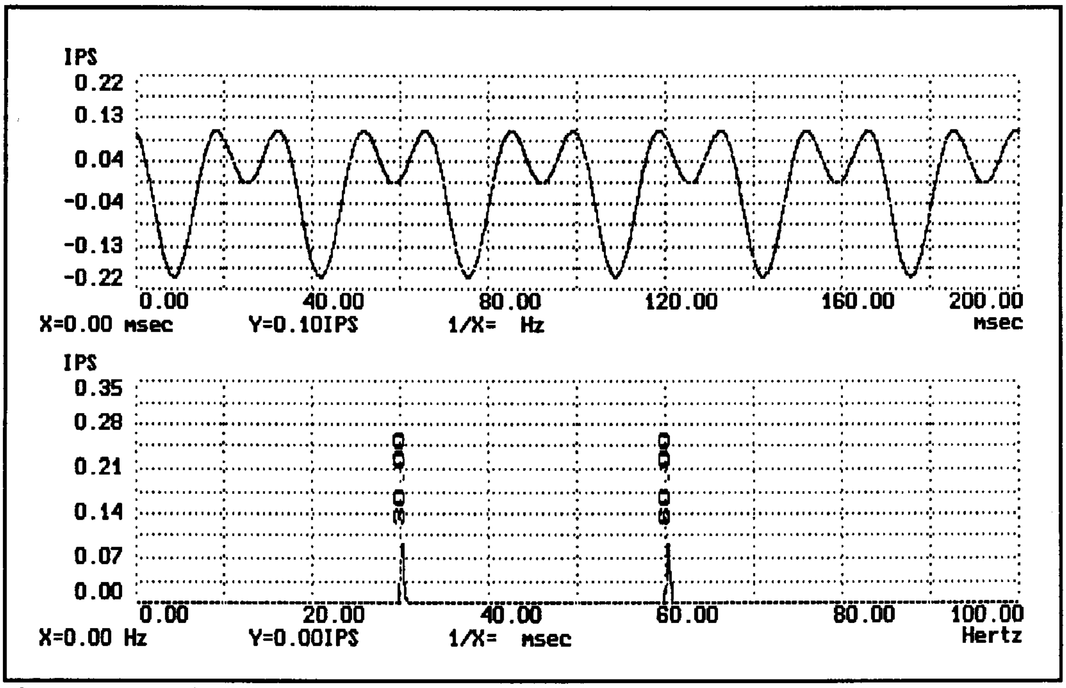


Fig. 5-32. Resultant Time Signal and Fourier Analysis.

The amplitudes of the two frequencies will increase or decrease depending on the phase relationship. The second harmonic amplitude modulates the lower frequency. The resulting time signal and Fourier analysis are indicated in Fig. 5-32.

This process can continue until the defect(s) extend all the way around the race. The end result is a series of peaks with difference frequencies equal to ball pass frequency.

This explains why only the second harmonic of bearing frequency is present in some newly installed bearings. For example, if the bearing was improperly hit with a hammer during installation, similar dents could occur opposite each other on the race. The two frequencies could be out of phase and equal in amplitude and would therefore cancel. However, the second harmonic would still be generated and would yield a spectral line at two times ball pass frequency.

As stated above, a bearing with one deep fatigue spall on the race can generate a pulse. The repetition rate of the pulse is the reciprocal of the ball pass frequency. A Fourier analysis of this signal yields a series of peaks with difference frequencies equal to ball pass frequency. Shallow flaking generates a similar spectrum. The best way to distinguish between the two defect types is to use the time domain signal. The time signal generated by a deep fatigue spall or other similar defect contains distinctive pulses. The best clue to the difference in the two frequency spectra is a low or nonexistent fundamental in the spectra generated by the pulses.

Fig. 5-33 contains the time signal and frequency spectrum taken from a 5313 bearing that has a deep fatigue spall. The bearing has 12 balls and the calculated ball pass frequency of the outer race is 140 Hz. Notice the distinct pulses occurring every 7.1 ms. Also note the low amplitude of the 140 Hz fundamental frequency. These are characteristic signals of a deep fatigue spall or similar defect.

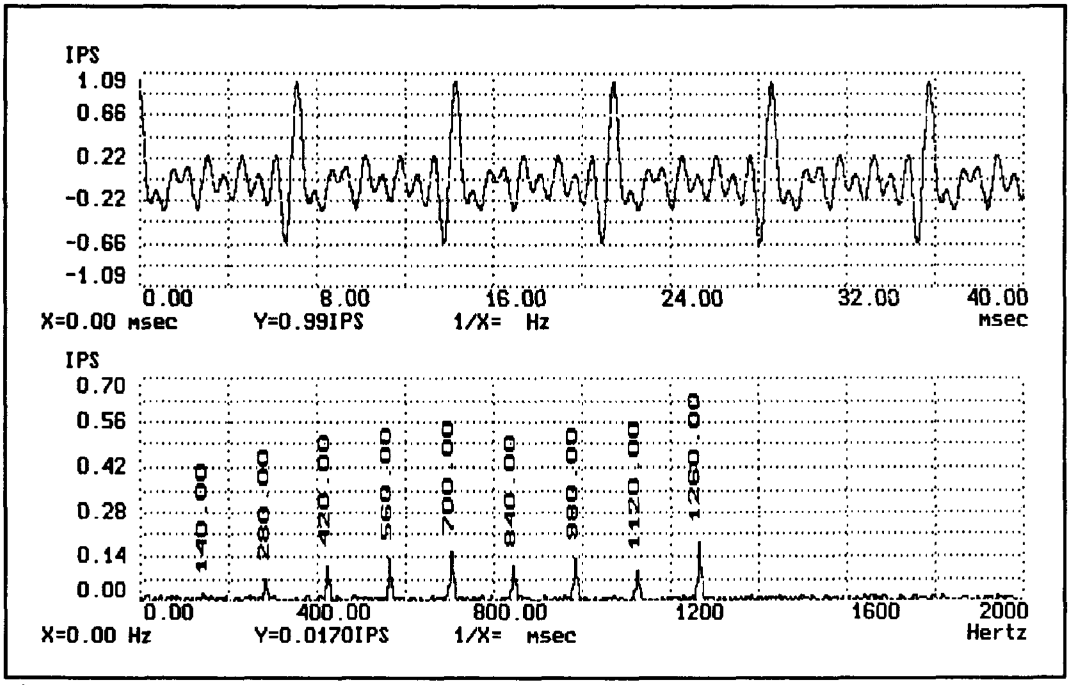


Fig. 5-33. Time Signal and Frequency Spectrum of Deep Fatigue Spall.

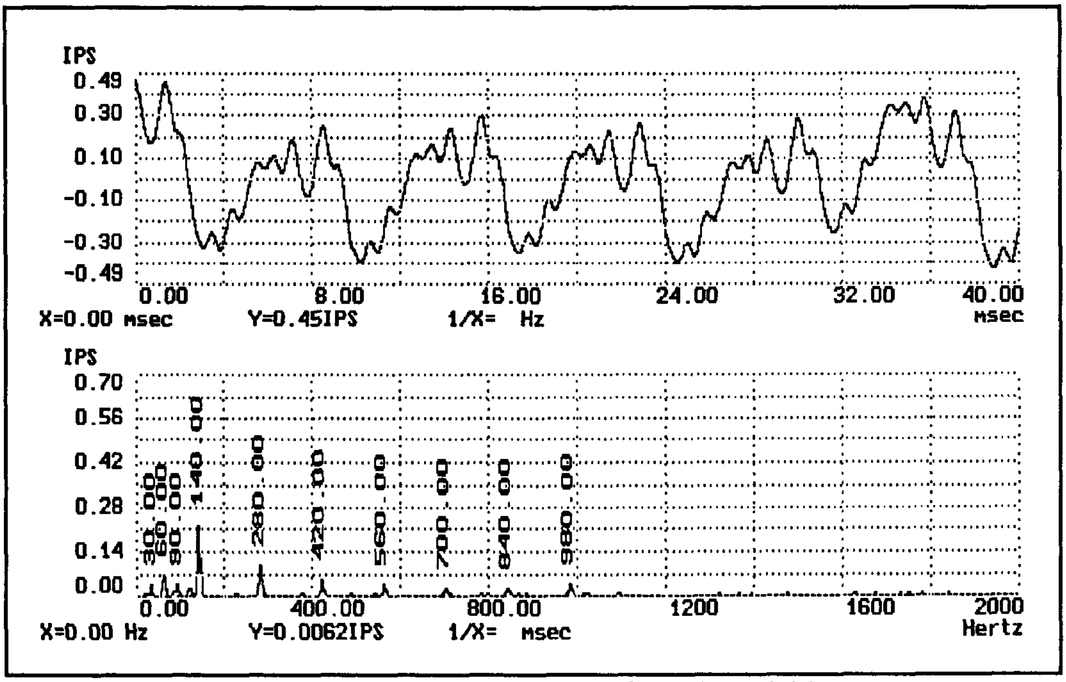


Fig. 5-34. Time Signal and Frequency Spectrum of Shallow Flaking.

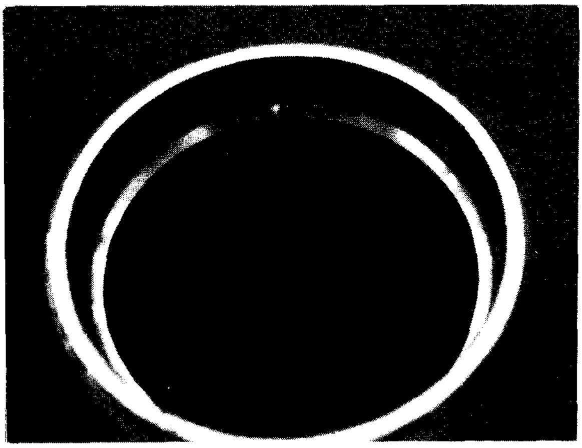


Fig. 5-35. Deep Fatigue Spall in Outer Race of Bearing.

Fig. 5-34 contains the time signal and frequency spectrum taken from a 5313 bearing that has shallow flaking. The number of balls and calculated ball pass frequency is the same. Notice the discrete frequency and high harmonic content in the time signal. Also notice the high amplitude of the fundamental, and some low amplitude harmonics. This is characteristic because some signals are out of phase.

Fig. 5-35 is a photo of the bearing defect that generated the pulses in Fig. 5-33. Fig. 5-36 is a photo of the bearing shallow flaking that generated the signals in Fig. 5-34.

Metal flakes from a single defect can become cold rolled on the race. When this occurs,

Fig. 5-36. Shallow Flaking on Outer Race of Bearing.

each flake can generate the ball pass frequency. A skilled analyst can distinguish between the two conditions because the frequencies from the metal flakes are very low in amplitude.

PROBLEM SOURCES

Acid Etching

When bearings are operated in an environment of high humidity and salt air, or areas where acids or acid vapors are present, corrosion or acid etching is often the nature of the bearing failure. These problems can be identified with spectral analysis by the distinctive shape and frequency content. The spectrum shape can have a low amplitude (0.15 IPS or below) with many spectral lines. An amplitude above 0.15 IPS may indicate spalling has occurred.

The fundamental BPFO, BPFI, or two times BSF may or may not be present. However, harmonics, sum, and difference frequencies may be present out to and above 2000 Hz. A ball bearing that has a shallow flaking fatigue spall all the way around the inner race can generate about six true harmonics of BPFI. If the defect is on the outer race, about seven harmonics of BPFO can be generated. For spherical and tapered roller bearings each true harmonic of BPFO is equal to about 0.5 inches of defect length. A general rule is not available for inner race defects because inner race defects for spherical and tapered roller bearings generate pulses at BPFI when the defect goes through the load zone. When bearings generate many harmonics (more than is reasonable for the above conditions) and the amplitude of the harmonics is low, the proper diagnosis is acid etching or corrosion.

The spectra in Figs. 5-37 and 5-38 were taken from a sulfuric acid product pump. The pump speed is 29.6 Hz, and the bearing is a 6308 deep grove ball bearing. The bearing frequencies are : BPFO = 89.5, BPFI = 147.3, FTF = 11.2, and 2 x BSF = 113.9 Hz.

In Fig. 5-37 note the high amplitude spectral lines at 1435 and 1580 Hz. This indicates inadequate lubrication. The higher amplitude indicates a resonant frequency. The

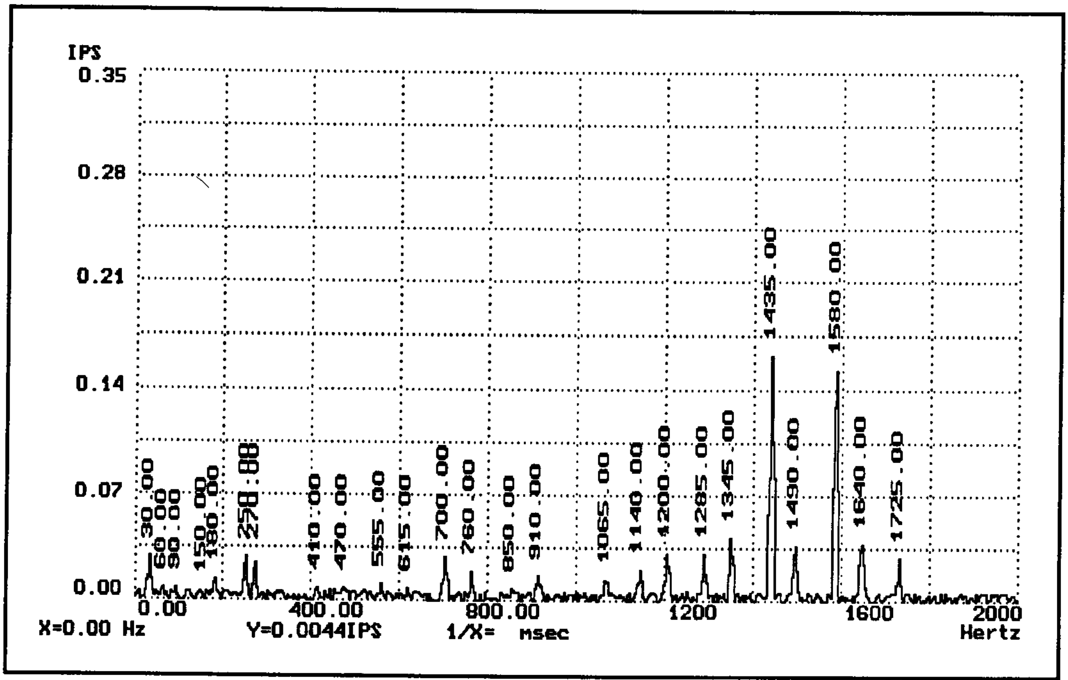


Fig. 5-37. 2000 Hz Spectrum from a Bearing with Corrosion on Inner and Outer Races and One Ball.

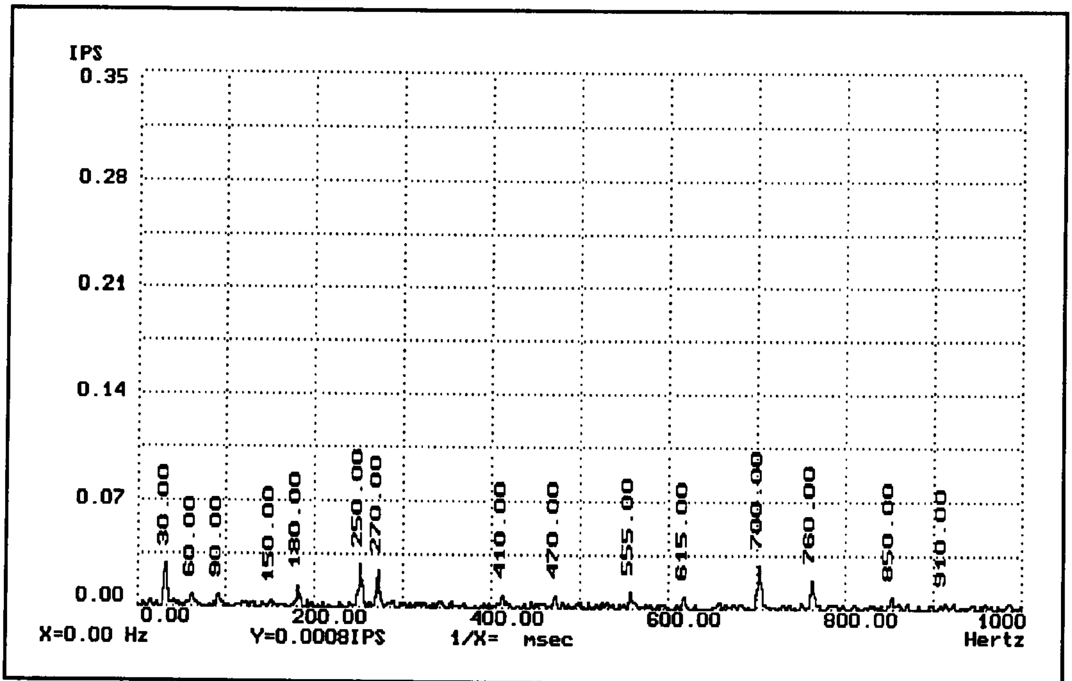


Fig. 5-38. 1000 Hz Spectrum from a Bearing with Corrosion on Inner and Outer Races and One Ball.

spectral lines at 1435 and 1580 Hz are not harmonics of BPFI because 1435 ÷ 142.3 = 10.08. However, the difference frequency is 1580 - 1435 = 145 Hz which is BPFI within measurement accuracy on the 2000 Hz range. Also note the plus and minus sidebands equal to BPFI. This is hard evidence that the inadequate lubrication spectra indicate an excited natural frequency and the source of the excitation can be BPFO and/or BPFI. Also note there is another set of spectral lines starting at 470, 615, 760, 910 Hz, etc. These are harmonics of BPFI. The amplitude of these harmonics is quite low because a resonance is not involved; these harmonics are caused by the defect, and the defect is quite small.

In Fig. 5-38 the first three harmonics of running speed indicate some slight looseness/misalignment. The pump has five vanes on the impeller and the spectral line at 150 Hz is vane pass frequency. The spectral lines at 90, 180, and 270 Hz are the first three harmonics of BPFO. The spectral lines at 470, 615, 760, and 910 Hz are harmonics of BPFI and they are caused by the defect. The difference frequency between 410, 555, 700, and 850 Hz is also BPFI. They are minus sidebands to the natural frequency excited by inadequate lubrication and the small defect on the inner race. However, they are not harmonics of BPFI.

Even though one ball had three acid etching marks, BSF or harmonics of BSF are not present in the spectra. Figs. 5-39, 5-40, and 5-41 contain the photo of the outer race, inner race, and one ball of the bearing that generated the above spectra. The day before the data was taken, an acid line broke and some acid was splashed on the pump bearing housing. Some of the acid entered the bearing, destroyed the lubrication, and left acid etching marks. This bearing failure was caused by an associated failure (the broken acid line) and further action was not required, other than the bearing replacement.

Fluting

When an electrical current arcs through a bearing, the resulting disfiguration is called fluting. The classical tale involves a welder that tack welds the ground on a fan housing and then arc welds on the fan. The path to ground is from the fan through the inner race, through the balls/rollers, through the outer race, and through the bearing housing. The resulting current flow causes arc lines on the bearing components. In today's better educated work force this problem, as caused by the welder, is seldom seen. However, fluting can and does still occur as a result of eddy current buildup and discharge. Fluting in the bearings supporting eddy current couplings has been observed. These couplings are used for speed control. Arcing can occur when the dielectric becomes contaminated and allows the eddy current to pass through the bearing. Fluting can also occur in some refiner drive motors. There has been some speculation that fluting in these motors may be caused by electrolysis from dissimilar metals in the pump and motor, although this has not been confirmed.

The most common case of fluting occurs in the newer model D.C. drive motors. Bearing failure caused by fluting can be quite prolific in these motors. Fluting can occur on the outer race, inner race, or balls on the drive or off end of the motor. Fluting appears to occur most often on the outer race of the off end bearing. Claims have been made that the installation of grounding brushes on the motor shaft drive end has stopped the fluting. However, there are documented cases where fluting has occurred on the outer race of the off end bearing after grounding brushes were installed on the drive end shaft.

Fig. 5-39. Outer Race of Bearing with Acid Etching.

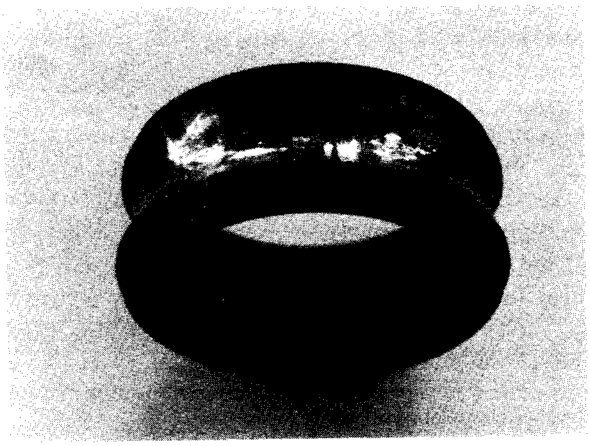


Fig. 5-40. Inner Race of Bearing with Acid Etching.

Fig. 5-41. Ball with Acid Etching.

When fluting occurs, the BPFI and/or the BPFO is generated. However, the fundamental bearing frequency normally does not occur until spalling starts. The characteristic spectra of fluting contain high frequencies (from 900 to 5000 Hz) modulated by ball passing frequencies. The exact position of where these frequencies occur is a function of speed, number of balls or rollers, and the density of the arc marks in the bearing.

Since the siren effect (discussed in Chapter 4), inadequate lubrication, and fluting can all generate frequencies in this range, some confusion has resulted in attempts to analyze these problems. The following hints may be helpful.

1. The siren effect is a generated frequency and should equal speed times number of bars on the rotor. The signal should also be modulated by motor speed.

2. The frequencies generated by fluting should be modulated by the ball pass frequencies.

3. Other generated frequencies that could fall into this range should be eliminated.

4. After the above are eliminated, the remaining frequencies should be analyzed for inadequate lubrication.

The question often arises as to why the newer motors are failing more often. We do not know the exact answer. However, a review of the events over recent years may be helpful. The industry demand for high efficiency motors has motivated manufacturers. One way to accomplish higher efficiency is to reduce the distance between the rotor and stator. This smaller air gap increases the transformer action, achieving more efficiency. This reduced air gap also has the following negative effects:

1. Increases the temperature

2. Increases the magnitude of the eddy currents

3. Increases the probability of the siren effect

The net result may be increased failure from broken rotor bars, increased failure from fluting in the bearings, and more motors producing the siren effect. The data from Fig. 5-42 was taken from an electrical motor, off end axial direction. The installed bearing is a 6236, the speed is 8.7 Hz, the calculated BPFO is 40.55 Hz, and the BPFI is 55.14. The spectral lines at 360 and 720 Hz are electrical frequencies, and may be normal in newer motors. The frequencies between 1000 and 2000 Hz contain a difference frequency of about 40 Hz.

The frequency window from 1600 to 2000 Hz in Fig. 5-43 contains a difference frequency of between 40 and 41 Hz. The variation is caused by measurement accuracy. These spectra indicate the off end bearing has fluting on the outer race.

Since the fundamental BPFO is not present, this indicates spalling has not started.

Fig. 5-44 contains a photo of the bearing that generated the above spectra. Please note the fine arc marks across the bearing, and the absence of spalling.

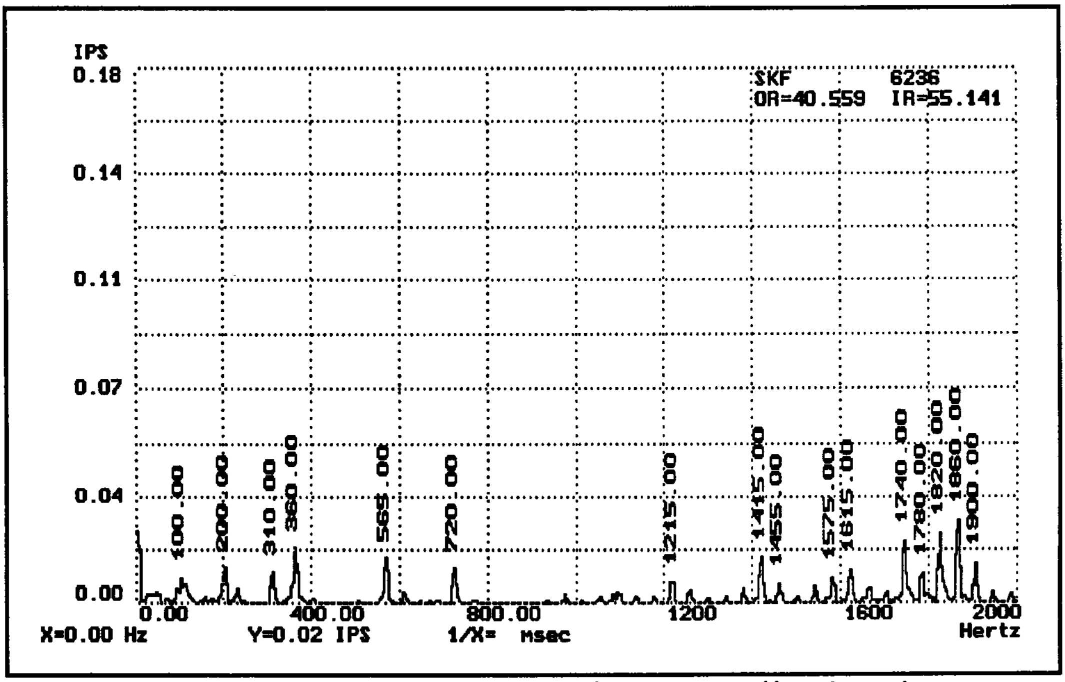


Fig. 5-42. Electrical Fluting, 2000 Hz Spectrum from Motor off End Bearing.

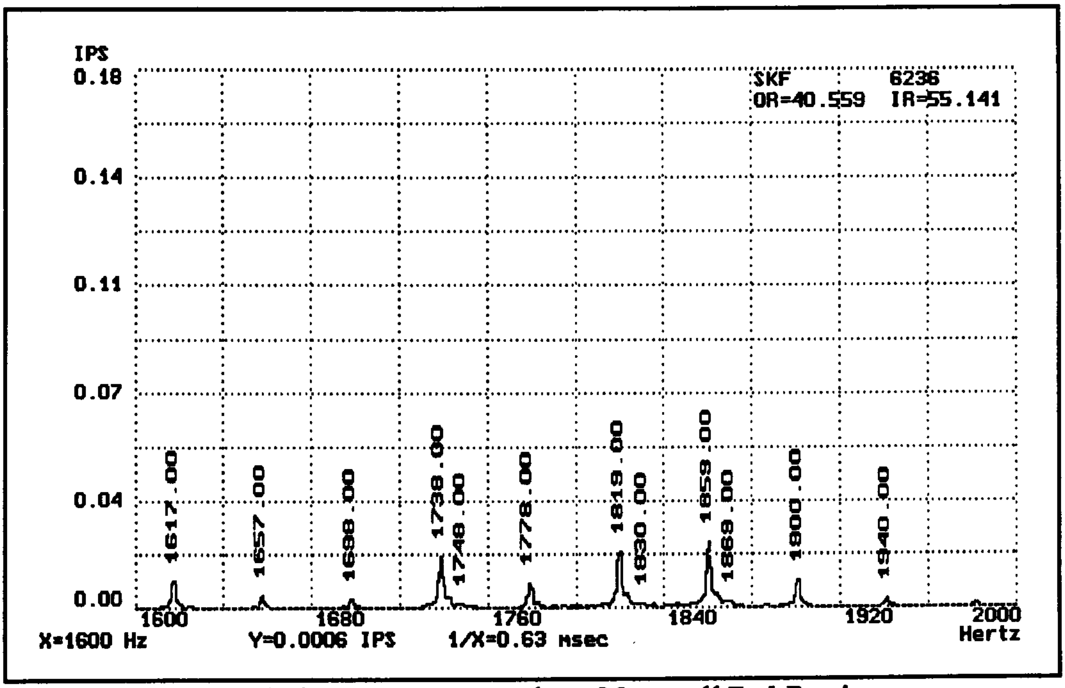


Fig. 5-43. Electrical Fluting, 400 Hz Zoom from Motor off End Bearing.

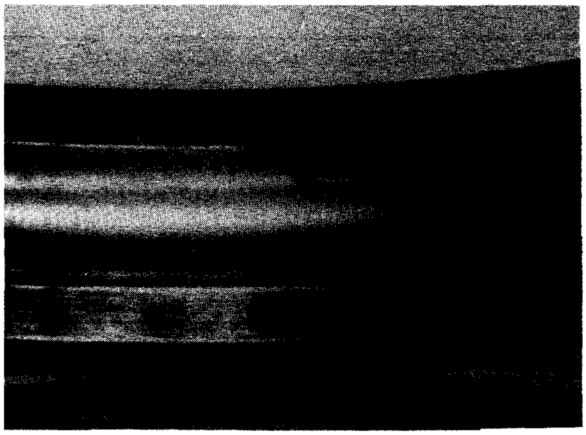


Fig. 5-44. Photo of Bearing with Fluting.

Inadequate Lubrication

One of the most important diagnoses that can be made in antifriction bearings is inadequate lubrication. When properly diagnosed, simply adding proper lubrication often prevents a catastrophic failure and increases equipment life.

Considerable time and effort have been spent in developing a technique to accurately diagnosis inadequate lubrication. Empirical measurements of lubrication problems indicate the following:

1. When the lubrication in a bearing becomes marginal, the lubrication may break down once each revolution. An example is when the heavy spot (caused by the residual imbalance, loading, etc.) passes through the load zone each revolution. When this occurs, the pulse or impact of the balls excites the natural frequency of the installed inner race assembly. The source of excitation is the BPFI. The oil film may be thinner on the inner race because of rotation, and the curvature of the inner race is more acute.

2. The frequencies generated by these events produce a series of spectral lines around the natural frequency. This natural frequency has sidebands or modulation equal to the BPFI.

3. The result is about four spectral lines somewhere between about 500 to 2500 Hz or more. The spectral line with the highest amplitude may be the natural frequency, and the difference may be equal to BPFI. Normally the natural frequency is not evenly divisible by BPFI.

4. When lubrication is added, the spectral lines either disappear or the amplitude decreases significantly.

Fig. 5-45 contains a spectrum taken from a dust blower. The blower speed is 19.6 Hz and the bearing is a 6313. The calculated BPFI is about 96 Hz. Note the spectral lines at 920, 1020, 1120, and 1220 Hz. The difference frequency is 100 Hz, and is within measurement accuracy of 96 Hz. The spectral line at 1020 Hz is the natural frequency and 1020 + 96 = 10.63. This proves the spectral line is not a harmonic of BPFI.

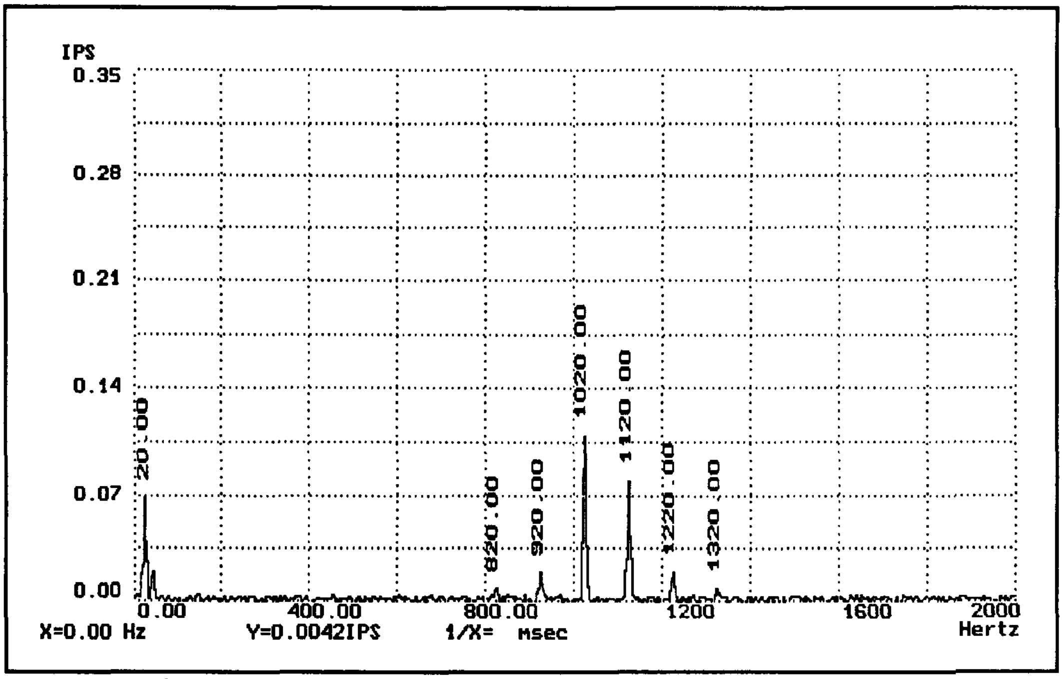


Fig. 5-45. Before Lubrication.

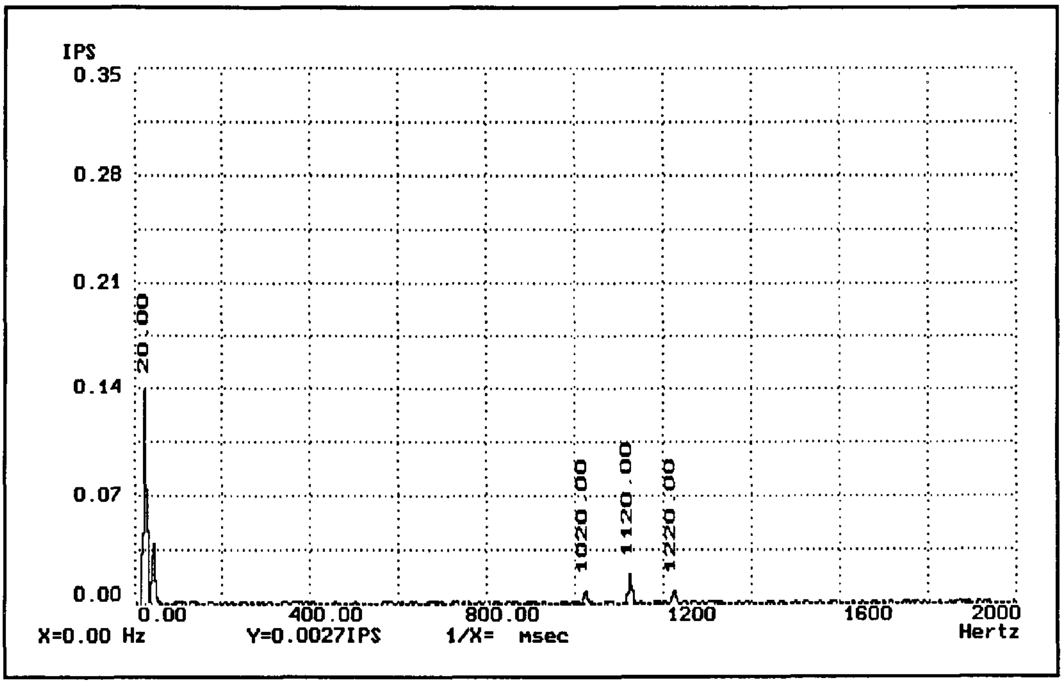


Fig. 5-46. After Lubrication.

This bearing was installed in a barrel assembly that supported the shaft and was oil lubricated. An oil leak had drained the assembly of oil and the spectrum in Fig. 5-45 was taken before oil was added. The spectrum in Fig. 5-46 was taken after oil was added. Note the amplitude has reduced significantly and three spectral lines are present. The amplitude of the spectral line at fan speed has increased. This is because the dust builds up on the blower and then falls off. This causes the amplitude at blower speed to vary with time.

The BPFO could excite a natural frequency in some special cases. For example, when the outer race is stationary, a portion of the outer race is in the load zone at all times. Balls or rollers passing over the load zone generate a discrete frequency. A discrete frequency excites a natural frequency only when the natural frequency is equal to the discrete frequency, a harmonic, or a subharmonic of the discrete frequency.

Accurate diagnosis of inadequate lubrication has become more difficult in recent years because of the high incidence of fluting and the siren effect in some high efficiency motors. Frequencies generated by these problems can and do occur in the same frequency range. However, each problem has its own identifying characteristics. For example:

1. Inadequate lubrication generates about four spectral lines with a difference frequency equal to BPFI. However, the frequencies are not harmonics of BPFI.

2. Fluting generates many spectral lines with a difference frequency equal to BPFO or BPFI. All the frequencies are harmonics of BPFO or BPFI.

3. The siren effect generates three or four spectral lines with a difference frequency equal to some multiple of rotor speed (i.e. two or three times motor speed). These spectral lines are harmonics of rotor speed (i.e. number of slots on the rotor times speed) and the spectral lines can be modulated by rotor speed.

Looseness

Excessive internal looseness is usually a problem in spherical roller bearings mounted with tapered bushings. Internal looseness can occur for a number of reasons: the bushing has become better seated on the shaft, the lock nut has come loose, or abrasives in the lubricant are causing excessive wear. Whatever the cause, the spectra can contain frequencies at one times speed and several harmonics of speed, as in Fig. 5-47. Other characteristics of looseness include low amplitude, a broadband spectrum of random noise, and a drastic change in balance sensitivity. Care should be exercised when diagnosing this type of looseness because several multiples of speed can also be caused by a bearing turning on the shaft or in the housing. Fig. 5-48 is a photo of a bearing with excessive internal looseness. The excessive wear pattern was caused by an abrasive contaminating the oil and behaving as a lapping compound to wear the outer race.

Bearings that have excessive internal clearance generate spectra very similar to bearings turning on the shaft. Both can have a high amplitude fundamental, a fundamental plus harmonics, and low frequency modulations. The difference between the two conditions is that bearings with excessive internal clearance normally generate the fundamental train frequency.

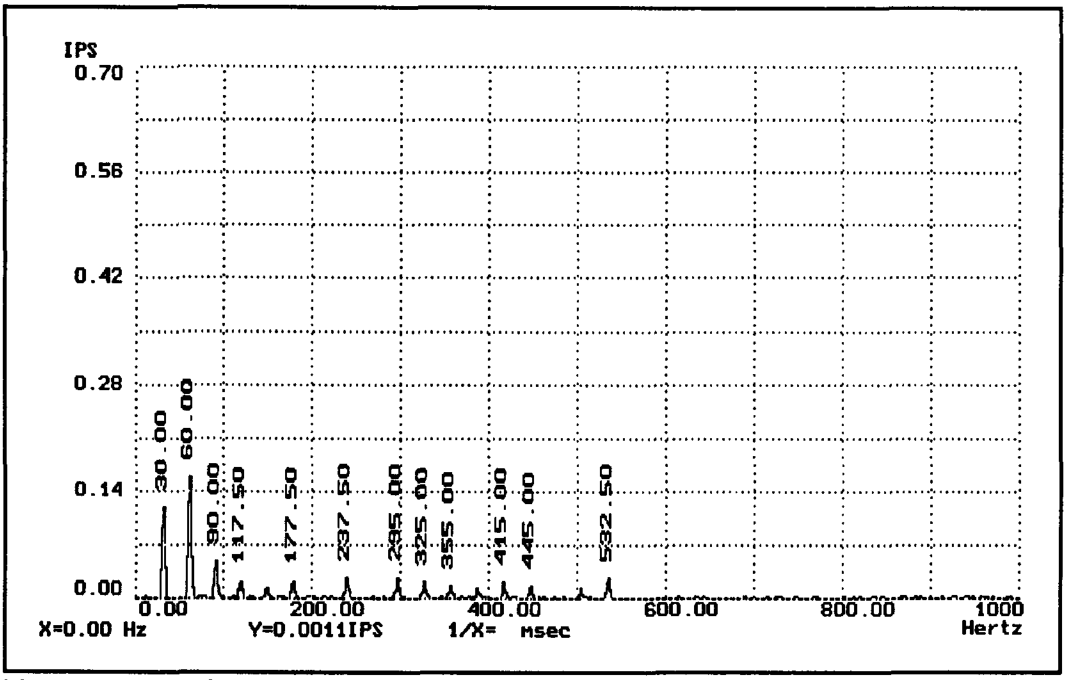


Fig. 5-47. Excessive Internal Looseness.

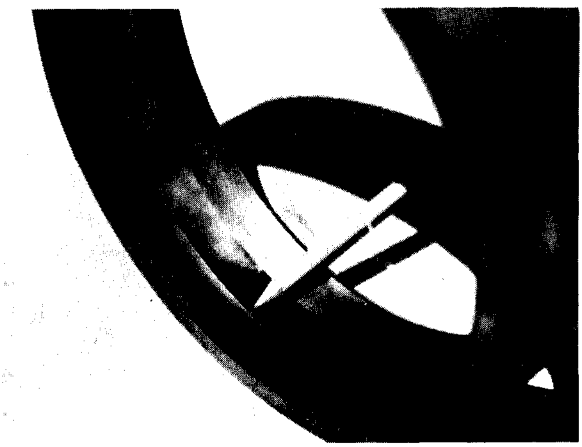


Fig. 5-48. Photo of Bearing with Excessive Internal Looseness.

The FTF can appear as a discrete frequency or as modulations of the fundamental and harmonics. Bearings that fail with a worn cage normally have small defects on the outer race. However, the severity of these defects is often overstated. In such cases, ball pass frequency of the outer race and harmonics are generated. The peaks at these frequencies are modulated with the FTF and often the rotor speed.

Figs. 5-49 and 5-50 contain the spectra taken from a 22318 bearing with a worn cage and small defect on the outer race. The roll is turning at 240 RPM, the BPFO is 24.2 Hz, and the FTF is 1.6 Hz. Note the modulations at rotating speed and the FTF. Fig. 5-51 is a photo of the worn cage taken from the bearing. Notice the excessive clearance between

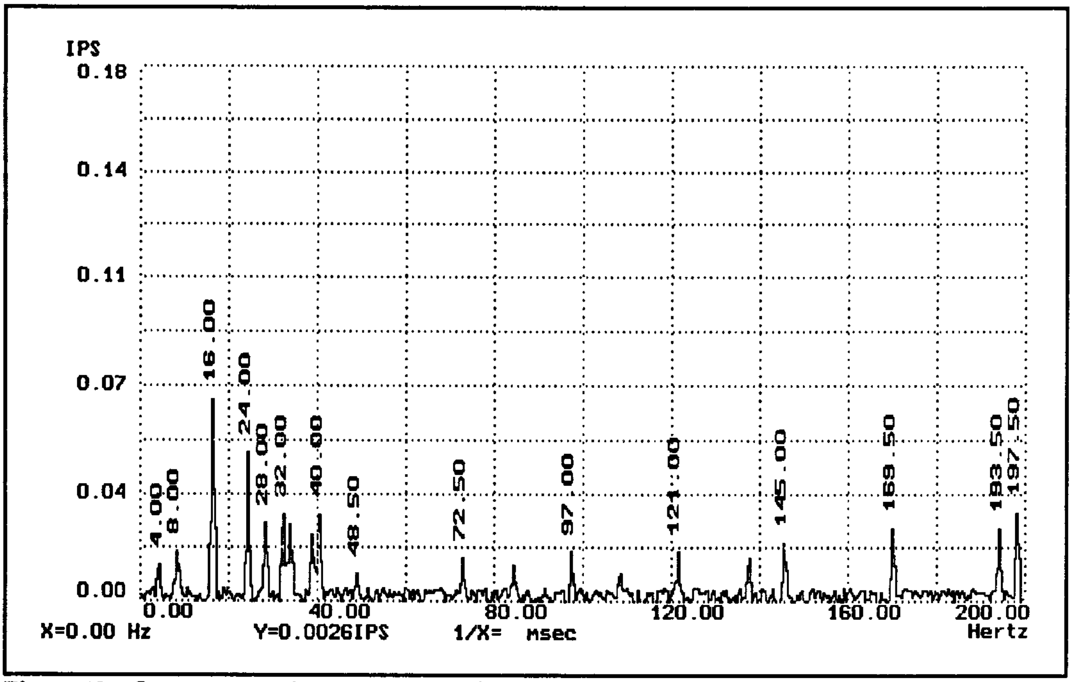


Fig. 5-49. Spectrum Taken at 200 Hz from a Bearing with a Worn Cage.

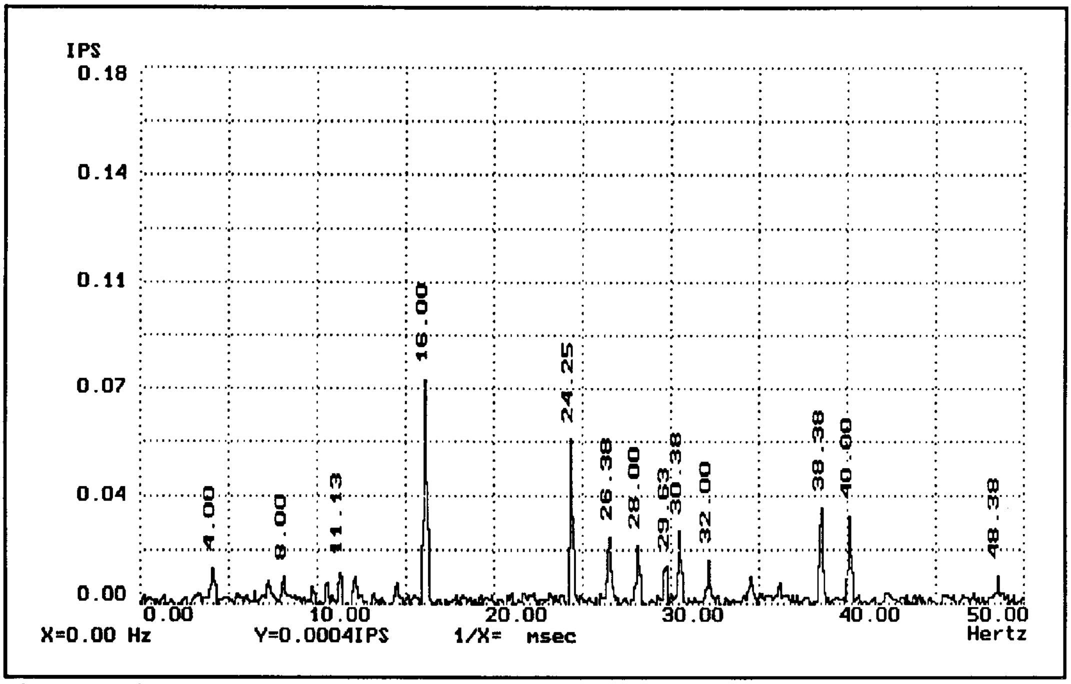


Fig. 5-50. Spectrum Taken at 50 Hz from a Bearing with a Worn Cage.

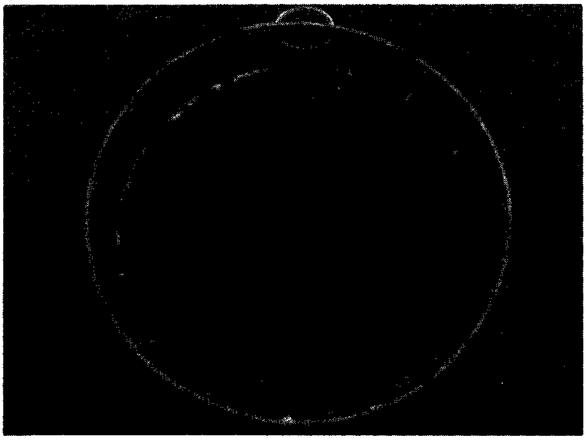


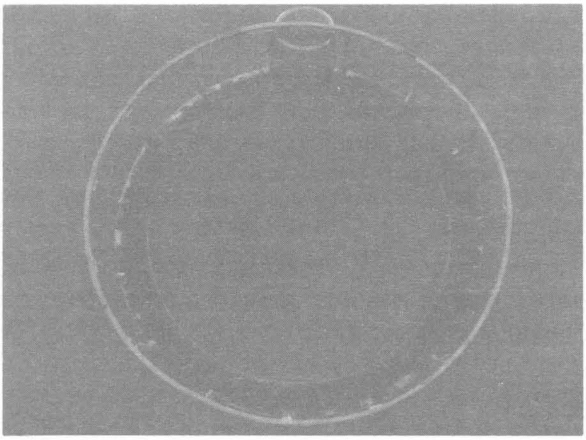
Fig. 5-51. Worn Cage That Generated the Spectra in Figs. 5-49 & 5-50.

the roller and the cage ribs. In some cases when the cage is severely worn, discrete frequencies may no longer be generated. In such cases, the spectrum contains wide-banded noise over several hundred Hz, and the amplitude is slightly above the baseline.

Looseness in bearings often generates the fundamental and several harmonics because of the hit-and-bounce motion of the rotor as it rotates. The third harmonic can be higher in amplitude, indicating truncation caused by the nonlinearity of the looseness. The hit-and-bounce motion of units with this problem often generates a pulse at some fraction of unit speed. A higher amplitude pulse every other revolution generates half shaft speed. A higher amplitude pulse every third revolution generates peaks at one-third shaft speed. Low frequencies up to one-seventh of shaft speed have been observed. These fractional shaft speeds can also modulate the fundamental and harmonics. A Fourier analysis of such signals yields spectral lines at the fundamental and harmonics with plus and minus sidebands. The difference frequency between the sidebands is equal to some fraction of shaft speed. These difference frequencies can equal belt speed when the outboard bearing on a belt-driven unit has looseness. The pulses that cause these low frequency modulations can be observed in the time domain. The amplitude may not be the same for each time period and sometimes the pulse is not visible at all. This explains the very low amplitude of the sidebands in the frequency domain. Also refer to the discussion on looseness in Chapter 4. Since looseness has so many forms and degrees, it is often difficult to determine exactly what is loose.

The following paragraphs discuss distinctive characteristics of bearings loose on the shaft or in the housing.

Bearings that are turning on the shaft can generate the fundamental frequency of unit speed and several harmonics, or a high amplitude fundamental, depending on the installation. An electric motor with the drive end bearing turning on the shaft can generate a high amplitude fundamental if the motor is belt driving another unit. This problem creates an imbalance condition because the rotor no longer rotates around its center of gravity. Attempts to balance rotors in this condition are often unsuccessful. When a bearing is turning on the shaft, a low amplitude spectral line is generated on the low side of rotor speed. This spectral line is generated by the inner race turning on the shaft. This distinguishable characteristic identifies the speed of the inner race. The



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spectral line at rotor speed obviously identifies the speed of the rotor. The difference frequency between the two spectral lines is how fast the bearing is turning on the shaft. See Fig. 4-9 in Chapter 4.

Bearings loose in the housing can generate a distinctive fourth harmonic of rotor speed. The second and third harmonics can also be present. However, the fourth harmonic is slightly higher in amplitude. See Fig. 4-10 in Chapter 4. Note the overall amplitude is less than 0.1 IPS. When a gear is mounted on a shaft and the shaft support bearing is loose in the housing, the gearmesh frequency can be modulated by four times rotor speed. Since looseness is an out-of-phase condition, the amplitude of the fourth harmonic will be higher on the low side of gearmesh frequency. Other harmonic and/or sidebands of rotor speed can also be present.

The data in Figs. 5-52 and 5-53 was taken from a 1776 RPM (29.6 Hz) motor that has a 43 tooth gear mounted on the shaft. Gearmesh frequency equals 29.6 x 43 = 1272.8 Hz. The spectral line at 1275 Hz is gearmesh frequency within measurement accuracy on the 2000 Hz range. The spectral line at 1155 Hz is 120 Hz less than gearmesh frequency and indicates the bearing is loose in the housing. The fourth harmonic is also distinctive in Fig. 5-53 and indicates the bearing is loose in the housing. The other harmonics of motor speed, half-shaft speed, and modulations of shaft speed also indicate looseness.

Testing for Bearing Frequencies

Other procedures in this chapter deal with calculated and measured frequencies. These procedures are good, however, the analyst is often required to diagnose machines, and internal geometry is not always available. The following procedure may be helpful in

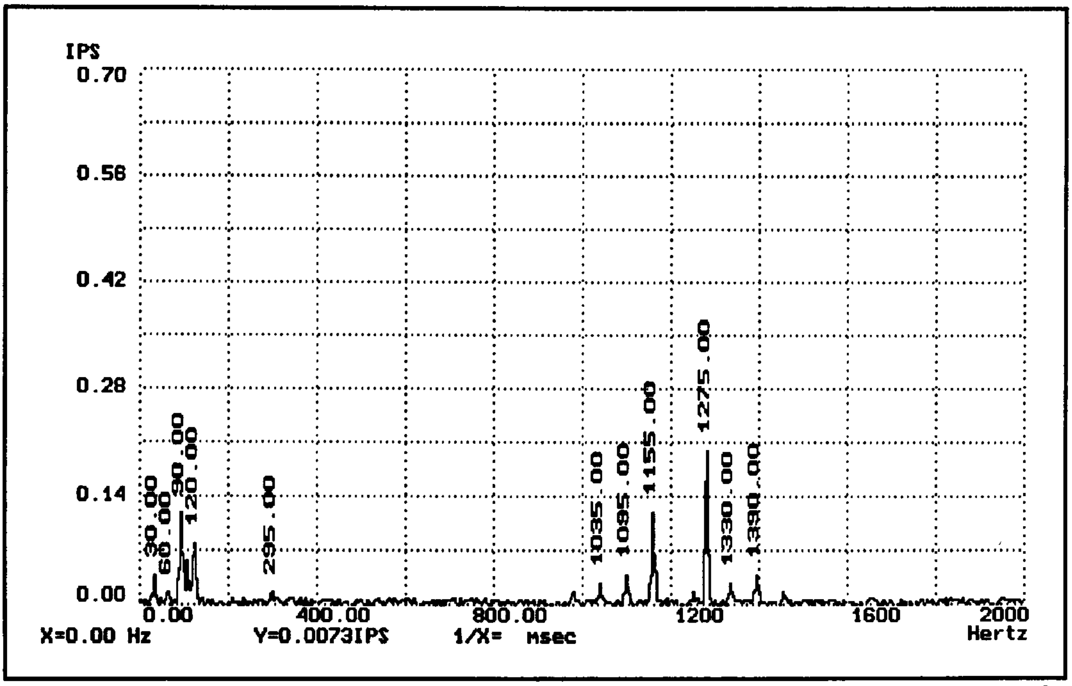

Fig. 5-52. Spectrum Taken at 2000 Hz from Electric Motor with Bearing Loose in Housing.

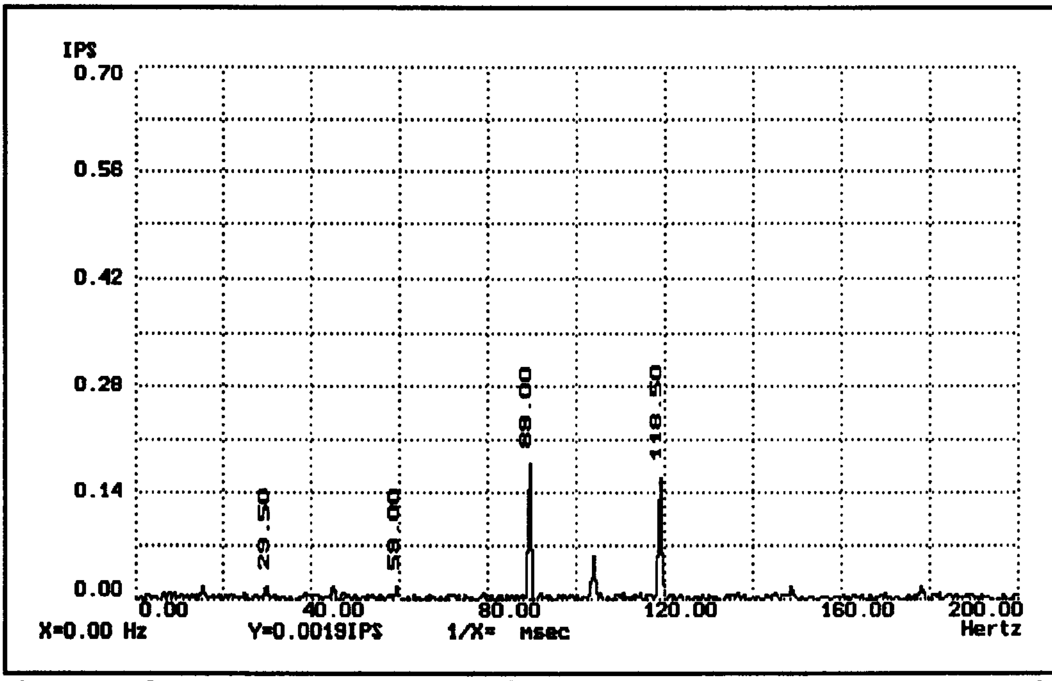

Fig. 5-53. Spectrum Taken at 200 Hz from Electric Motor with Bearing Loose in Housing.

such cases when the spectra contain one or more frequencies that could be a bearing frequency.

First, divide the suspected frequency by unit speed. If the frequency is some exact multiple of speed, with due respect to measurement accuracy, the frequency may be vane pass or gearmesh frequency, or some harmonic of these frequencies. If the frequency is not an exact multiple of speed, it could be a bearing frequency. Next, use the percentage method and calculate backwards to identify the number of balls the bearing may have. If the number of balls/rollers obtained is not realistic for that machine, the bearing may not be defective. Finally, the frequency could be a resonant frequency. Resonant frequencies normally do not have harmonics unless looseness is also present.

The formulae for calculating the number of balls are:

$$N_b = \frac{F/S}{0.4}$$

for a defect on the outer race, and

$$N_b = \frac{F/S}{0.6}$$

for a defect on the inner race

where F = the suspected frequency
0.4 or 0.6 = the percentage of balls or rollers that pass over the defect for BPFO and BPFI, respectively.

The above formulae are only approximate because the percentage method for BPFO actually varies between 38 and 41 percent, and the percentage for BPFI varies between 59 and 62 percent. Additionally, the contact angle is not known.

For example, consider data taken from an electric motor that contains a spectral line at 91.2 Hz. The motor speed is 29.6 Hz. Then, $\frac{91.2}{29.6} = 3.08$ This proves that 91.2 is not a harmonic of speed.

Next, $\frac{3.08}{0.6} = 5.1$ balls. This proves the frequency is not BPFI because there are no bearings with five balls.

Next, $\frac{3.08}{0.4} = 7.7$ balls. This value is very close to eight.

Next, $\frac{3.08}{0.38} = 8.1$ balls. An eight ball bearing is common in electric motors. Therefore, the spectral line at 91.2 Hz may be BPFO.

CONCLUSION

At first, most managers want all defective bearings identified so that the appropriate repair or replacement action can be scheduled. Once a defective bearing is identified, the next question is, "How long will it last?" This question is somewhat more difficult and has no simple answers, but the following criteria have been developed.

1. How long the bearing will last is proportional to the length of time the bearing has been in operation when the defect is first identified. If the bearing has been in operation for a number of years, then it will probably last for several months. At the other extreme, if the bearing has been in operation for only a few weeks when the defect occurs, the expected life may only be hours or days.

2. How long the bearing will last depends on the detailed condition of the bearing when the question is asked. If the bearing has one defect on one of the raceways, life expectancy could be many weeks. If the bearing has defective balls or rollers, or a damaged cage, life expectancy may only be a few hours.

3. How long the bearing will last depends on why the bearing is failing. If the bearing is failing because all the internal clearances have been lost, rapid destruction can occur. If failure is caused by a parasitic load, the failure rate is proportional to the load. If the parasitic load is removed, life can be extended. Complete loss of lubrication can also cause rapid failures.

4. How long the bearing will last depends on the speed of the rotating unit. Defective bearings in units rotating 3,600 RPM and above can fail very quickly. At the other extreme, for defective bearings in slow speed machines rotating 200 RPM or less, the time between defect identification and failure can be as long as several months.

WARNING: One of the major obstacles in determining how long a bearing will last, is the inability to determine the extent of metallurgical damage present in the bearing parts. The technology provided in this book gives instructions on how to identify defects and approximate defect size. However, there is no way to identify when a bearing part such as a ball, roller, race, or shaft will break. In this regard, care should be used when predicting bearing life. For this reason bearings should be replaced as soon as possible after a defect has been identified. It is much more economical to allocate resources to increase bearing life than it is to try to predict how long it will last after a defect occurs.

History or baseline data are not needed to identify a defective bearing. Once the defective bearing is identified, data should be taken periodically to determine the failure rate. When deterioration in the bearing condition can be detected over a 24 hour period, life expectancy is a matter of a few days. Of course, the four specifics above must be consider ed at all times.

Problems with bearings in rotating machinery can be accurately identified up to six months before the repair must be made. This lead time is helpful in scheduling the necessary repairs and can save thousands of dollars in terms of lost production, wasted manpower, and wasted materials. Accurate diagnosis puts the manager in charge of the machine instead of the machine being in charge of the manager. When the manager knows of a problem in a machine, he can direct the correction of that problem at his convenience. When he does not know of the problem, he must react when the machine breaks, regardless of the day or the hour.

CHAPTER SIX: ACCURATE EVALUATION OF GEARS

INTRODUCTION

The right tools are required for diagnosing gear problems. These tools include hardware, software, technology, and skilled personnel. This chapter describes the technology required for accurate diagnosis.

Vibration data is necessary for accurate identification and analysis of gear problems. The time domain signal is required. The ability to process and view this signal with various long time periods is also essential. The frequency domain spectra are also required. The ability to view these spectra on various ranges and zoom in on certain frequencies is equally essential.

Using the above data and the technology contained in this chapter, the following gear problems can be identified:

1. Improper tooth ratio of meshing gears
2. Out-of-round or eccentric gears -- This includes gears with one or more high places, pitch line runout, uneven tooth width, etc.
3. Gears mounted on a bent shaft
4. Looseness, including gears or bearings loose on the shaft, and bearings loose in the housing
5. Misalignment
6. Excessive backlash or backlash-type problems such as oscillating gears
7. Gears that have broken, cracked, or chipped teeth

We cannot tell how long gears will last. However, we can project the expected life based on the tooth ratio of meshing gears. We can identify worn gears and arrange the problems by degree of severity.

DATA ACQUISITION

How To Take Data

In order to diagnose gear problems accurately, it is imperative to obtain useful information. Achievement of this goal depends upon various parameters. They include knowing the expected frequencies, proper transducer selection, and knowing how to use the selected transducers.

Placement of a transducer is critical. If the transducer is not placed at the correct location, useful data may not be obtained. Correct transducer placement can vary according to the type of machine, type of gears used, nature of the problem, and the construction of the machine.

The correct place to take data on a specific gearbox can vary depending on which gear is defective and which side of the gear is defective. Best signal definition can be obtained by placing the transducer in the load zone as close to the gear support bearing as possible.

A few simple rules may be helpful in identifying the best place to take data:

1. For spur gears, best signal definition is obtained from the radial position.

2. For helical, herringbone, and other gears with a contact angle, best signal definition is obtained from the axial position.

3. The transducer should be placed as close to the gear as possible in one of the above positions.

4. If a tooth on the right side of a gearbox is broken, the best signal is obtained from the right side of the gearbox. The converse is also true.

5. If a gearbox is constructed with internal webs, the best signal is obtained by placing the transducer where the web joins the housing. This applies only for the gear that is supported by the web.

6. The nature or type of problem also affects the location. For example, a gear loose on the shaft or a shaft with a worn key or keyway may generate a strong signal in the radial direction.

7. Do not take data from gearbox covers or the thin plates over the oil seal. Use natural benchmarks such as bolt heads. The transmissibility of frequencies through gaskets, Permatex, etc., is not very good, and the signal can be attenuated. The transmissibility of frequencies through threads on a bolt is not attenuated. Therefore, the data from a bolt head yields the best results.

Transducer selection is important for the accurate diagnosis of gears. If the third harmonic of gearmesh frequency is less than 2,000 Hz, most velocity transducers should be adequate. If the third harmonic of gearmesh frequency is above 2,000 Hz, an accelerometer must be used.

If both low frequencies and high frequencies are present, data must be taken with two transducers. This is necessary when both low and high frequencies are present because the velocity transducer cannot pick up the high frequencies. The accelerometer can pick up both frequencies. However, the low frequencies may not be visible. This is true because the frequency response curve for the accelerometer is an outward and upward sloping curve. Enough attenuation to prevent saturation of the high frequencies could obscure the low frequencies. When an accelerometer is used, it should be hard-mounted to obtain accurate data.

When small gears and shafts are installed in gear cases with thick walls, a contacting or noncontacting displacement transducer must be used to measure relative motion between the shaft and casing. This is true because low speed, lightweight gears and shafts cannot generate enough energy to shake the heavy gearboxes. The bearing cap vibration signal may not be useful when the casing weight to rotor weight ratio is more than 10-to-1 [1].

In order to obtain accurate information about rotating speed, units turning below 500 RPM may require analysis with a displacement transducer. For accurate analysis of all problems in units rotating with speeds below 120 RPM, a displacement transducer must be used.

Fig. 1-27 in CHAPTER ONE contains a typical frequency response curve for displacement, velocity, and acceleration, and is helpful in selecting the proper transducer for various frequencies.

GEAR VIBRATION THEORY

When two or more gears are in mesh, the frequencies generated depend upon gear speed, number of teeth, eccentricity, and common factors. The following section discusses some of these topics.

Evaluation of gear ratios can be accomplished by factoring the number of teeth on each gear. An ideal set of gears does not have a common factor other than 1. Any number, even a prime number, has itself and 1 as factors. The number 12 for example, has the following factors: **1 x 2 x 2 x 3 - 12.** The number 10 has factors of: **1 x 2 x 5 - 10.** The numbers 12 and 10 have a common factor of 2 and uncommon factors of 2 x 3 and 5.

The number of teeth on each gear must be factored to determine if the gears have a common factor. Gears with a common factor other than 1 are considered an improper ratio. If one gear is eccentric, the common factor identifies the number of teeth between the defective teeth. The uncommon factor of the meshing gear also identifies the number of revolutions the other gear must make before the same teeth mesh again. The frequency of this event is called the hunting tooth frequency.

Factoring is necessary to identify the common factors and uncommon factors. The numbers 1, 2, 3, 5, 7, 11, 13, 17, 19, 23, etc., are prime numbers. When a number is factored, it is broken down to the prime factors. Two numbers can have one or more prime numbers as a common factor. The number 8 has 1 x 2 x 2 x 2 as prime factors. The number 12 has 1 x 2 x 2 x 3 as prime factors. The numbers 8 and 12 have a common factor of 4, or 2 x 2.

If a number ends in zero or an even number, the number is divisible by 2. The numbers 20 and 36 are both divisible by 2.

If the sum of the digits in a number is divisible by 3, the number is divisible by 3. For example, the number 123 is divisible by 3 because **1 + 2 + 3 - 6,** and 6 is divisible by 3. Therefore, $\frac{123}{3}$ **- 41.** The prime factors of 123 are 1, 3, and 41, or 1 x 3 x 41 = 123.

If a number ends in 0 or 5, the number is divisible by 5. For example, 20 = 1 x 2 x 2 x 5. Also, 125 = 1 x 5 x 5 x 5.

The **Gears Program** or a prime numbers table is the best way to determine if a number is prime. If neither is available, then these hints may be helpful.

Gearmesh frequency can be computed by multiplying the number of teeth on a gear by the speed of that gear.

$$GF = T \times S, \textit{ where}$$

$$GF = \textit{gearmesh frequency}$$

$$T = \textit{number of teeth}$$

$$S = \textit{speed}$$

The number of teeth on the drive gear times the speed of the drive gear must equal the number of teeth on the driven gear times the speed of the driven gear. For example, if a 43-tooth pinion was driving a 99-tooth gear, as in Fig. 6-1, and the pinion rotates at 1776 RPM, the gearmesh frequency would be the number of teeth on the drive gear times speed.

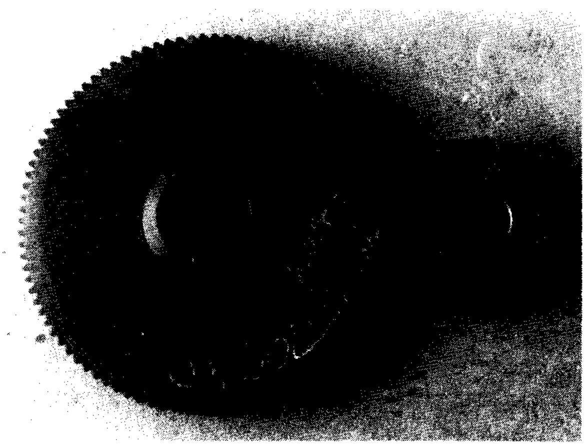

Fig. 6-1. 43-Tooth Pinion in Mesh with 99-Tooth Gear.

$$Speed = \frac{1776}{60} = 29.6\ Hz$$

$$GF = 29.6 \times 43 = 1{,}272.8\ Hz$$

The number of teeth on the driven gear times speed must equal the same number. For example, the 99-tooth gear must be rotating at:

$$Speed = \frac{1{,}272.8}{99} = 12.856\ Hz$$

$$\textit{because } 99 \times 12.856 = 1{,}272.74$$

This is also true for worm gears. The number of flights on the worm times the speed must equal the number of teeth on the driven gear times speed. As the pinion rotates against the driven gear, the individual cycles of the frequency generated are a profile of the individual teeth meshing. Harmonics of gearmesh frequency may also be present.

The second and third harmonics are very important. For example, if the second harmonic of gearmesh frequency has a high amplitude with respect to the fundamental and third harmonic, it would indicate a backlash-type problem. In other words, the gears may have too much backlash or one of the gears may be oscillating. In either case, the gears are meshing on both sides of the teeth. This double mesh generates the second harmonic. The phase of the second harmonic is 180° out of phase with the fundamental and appears at the top of the positive cycle of gearmesh frequency.

When the amplitudes of the first three harmonics of gearmesh frequency are distinctive, the gears are misaligned or have lead runout. The harmonic amplitude could be distinctive in several ways. For example, the amplitude of each harmonic could be about the same, or any one or two could be higher or lower than the other harmonic. The slope of the amplitudes could approach a 6 or 12 dB per octave slope.

Fractional gearmesh frequency can be generated by two gears in mesh. If the number of teeth on each meshing gear has a common factor other than 1, and one gear is eccentric, every Nth tooth (where N = common factor) on the good gear can be imprinted or worn by the eccentric gear, where N equals the common factor. This imprinting or wearing causes the Nth cycle of the gearmesh frequency to be higher in amplitude than the other cycles. The Fourier analysis of such a signal yields a fraction of gearmesh frequency equal to $\frac{1}{N}$. For example, if the gears have a common factor of 2, 3, or 5, then the frequency spectrum could have:

$$\frac{1}{2}, \frac{1}{3}, \frac{2}{3}, \frac{1}{5}, \frac{2}{5}, \frac{3}{5}, \text{ or } \frac{4}{5}$$

gearmesh frequency. The fractional gearmesh frequencies generated are caused by the common factor and the eccentric gear. If the gearmesh frequency has harmonics, then spectral lines can occur at the following fractions of gearmesh frequency:

$$1\frac{1}{2}, 1\frac{1}{3}, 1\frac{2}{3}, 1\frac{1}{5}, 1\frac{2}{5}, 1\frac{3}{5}, 1\frac{4}{5} \quad \text{or} \quad 2\frac{1}{2}, 2\frac{1}{3}, 2\frac{2}{3}, 2\frac{1}{5}, 2\frac{2}{5}, 2\frac{3}{5}, 2\frac{4}{5}$$

If the gears have a common factor that is not a prime number, such as 6, 8, 9, or 12, then fractional gearmesh frequency could occur at the reciprocal of the common factor or any of the prime factors of the common factor, depending upon the eccentricity in the gear.

Often, meshing gears that have a proper ratio generate a fractional gearmesh frequency. This occurs when every Nth tooth has some form of eccentricity. The most common occurrence is when every other tooth is eccentric. When this occurs, $\frac{1}{2}$ or $1\frac{1}{2}$ gearmesh frequency is generated. Every other tooth could be thicker or the pitch line could be higher on every other tooth for a given number of teeth. Please note that $\frac{1}{2}$ or $1\frac{1}{2}$ gearmesh frequency is not generated continually during each revolution of the gear. These fractional gearmesh frequencies are generated once each revolution of the gear only when the eccentric teeth are meshing. The number of eccentric teeth can vary from two to several. However, the number of eccentric teeth can be counted in the time domain signal.

The previous description explains why the amplitude of fractional gearmesh frequency is severely understated. Accurate measurement of fractional gearmesh frequency can be accomplished only by measuring the amplitude of the individual cycles in the time signal. When the number of eccentric teeth on a gear exceeds the common factor, fractional gearmesh frequency related to the common factor may not be generated. In such cases, fractional gearmesh frequency of the eccentric teeth, every other tooth, or every third tooth, etc., can be generated.

From these explanations, it would appear that a gear ratio is undesirable when the number of teeth on the meshing gears and/or gears mounted on a common shaft has a common factor. If such gear ratios cannot be avoided, then extra quality control measures should be used to ensure that eccentric gears are not permitted in the supply system.

Hunting tooth frequency, or HTF, occurs when the same tooth on each gear comes into mesh again. The HTF can be determined by dividing the least common multiple, or LCM, of the teeth on the two gears by the uncommon factor of the gear of interest. The product equals the number of revolutions the gear must make before the HTF occurs. The reciprocal of the gear speed identifies the time for one revolution. To obtain the HTF, multiply the time for one revolution by the number of revolutions, then take the reciprocal of the product.

$$HTF = \frac{1}{\frac{L}{U} \times \frac{1}{G_S}}$$

HTF = *Hunting tooth frequency*

L = *Least common multiple*

U = *Uncommon factor of gear*

G_S = *Speed of gear*

For example, a 43-tooth pinion drives a 99-tooth gear. The pinion must rotate 99 revolutions for a given tooth to mesh with the same tooth on the gear. Likewise, the gear must rotate 43 times for the same teeth to be in mesh again.

$$\frac{43 \times 99}{43} = 99 \quad or \quad \frac{43 \times 99}{99} = 43$$

Since the pinion is rotating at 1776 RPM, it will take 3.34 seconds for the same teeth to mesh again.

$$\frac{L = 43 \times 99}{U = 43} = 99$$

$$\frac{1776}{60} = 29.6\ Hz$$

$$\frac{1}{G_S} = \frac{1}{29.6} = 0.03378\ sec$$

$$0.03378 \text{ sec} \times 99 = 3.34 \text{ sec}$$

$$HTF = \frac{1}{3.34} = 0.299 \; Hz$$

The HTF is normally not measurable because the frequency is so low. However, it can be observed in the time domain when two meshing gears have a broken tooth on each gear. The broken tooth on each gear generates a pulse each time it goes into mesh. When two broken teeth mesh with each other, a major pulse is generated. The frequency of this event is the HTF.

Planetary gears require different formulae for frequency calculations. The gearmesh and hunting tooth frequency calculations stated above are valid only if the centerlines of the gear shafts are not orbiting. When the shaft centerlines move relative to each other, as in the case of a planet gear orbiting around a sun gear, the gearmesh frequency is no longer equal to the gear rotational speed times the number of teeth on the gear. In this case, the relative speed between the planet carrier and the gear with the fixed centerline must be calculated to obtain the gearmesh frequency.

The speed ratio r, for a gear set with fixed centerlines, is equal to the input speed S_i divided by the output speed S_o or:

$$r = \frac{S_i}{S_o} \tag{1}$$

This ratio is also equal to the number of teeth on the output (driven) gear N_o divided by the number of teeth on the input (drive) gears N_i, or:

$$r = \frac{N_o}{N_i} \tag{2}$$

Combining Equations 1 and 2 gives the gear speed ratio as:

$$r = \frac{S_i}{S_o} = \frac{N_o}{N_i} \tag{3}$$

Equation 3 is the basis for determining the speeds of a gear train. However, it cannot be immediately applied to a planetary gear train with moving centerlines.

Fig. 6-2 shows a simple planetary gear train with a sun gear at the center, three planet gears mounted on a carrier, and a ring gear. The centerlines of the sun and ring gears are fixed. The centerlines of the planet gears orbit around the sun gear at the speed of the carrier.

The sun gear, ring gear, and carrier may all be used as either an input or an output. Depending upon the configuration, a planetary gear train may be used as a speed reducer or a speed increaser. For example, if the carrier is stationary with the sun gear as the input and the ring gear as the output, then the gear train is a speed reducer. If the ring gear is stationary with the carrier as the input and the sun gear as the output, then the gear train is a speed increaser.

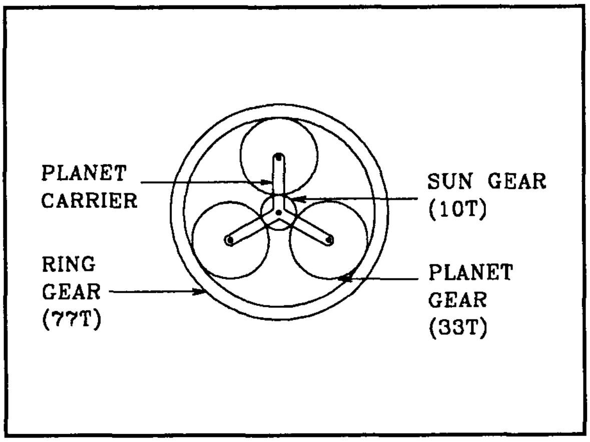

Fig. 6-2. Planetary Gear Geometry.

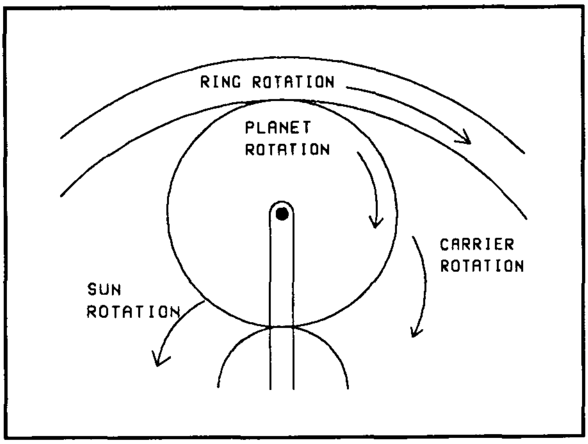

Fig. 6-3. Gear Rotation and Directions.

A basic planetary gear train has four different speed components, with rotational directions as shown in Fig. 6-3. These are the sun gear speed S_s, planet gear speed S_p, ring gear speed S_r, and carrier speed S_c.

Clockwise speeds will be considered a positive direction (+), and counterclockwise speeds will be considered a negative direction (-). The speeds and directions of two of the four components must be known in order to calculate the speeds of the other two. For example, if the rotational speeds of the sun and ring gears are known, the speeds of the planet gears and the carrier can then be determined.

In order to use Equation 3 for a planetary gear train, the relative speeds between the various components must be calculated. The speed of the sun gear relative to the carrier $S_{s/c}$ is equal to the sun speed minus the carrier speed,

$$S_{s/c} = S_s - S_c \tag{4}$$

Similarly, the speed of the ring gear relative to the carrier $S_{r/c}$ is equal to the ring speed minus the carrier speed,

$$S_{r/c} = S_r - S_c \tag{5}$$

Equations 4 and 5 may be substituted into Equation 3 for S_i and S_o respectively. If the sun gear is considered the input and the ring gear is considered the output, then the sun and planets are the driving gears (sun driving the planets, planets driving the ring). In this case the planets and ring are the driven gears. The product of the number of teeth on the driven gears is equal to the number of teeth on the planets N_p times the number of teeth on the ring N_r. The product of the number of teeth on the driving gears is equal to the number of teeth on the sun N_s times the number of teeth on the planets. Finally, Equation 3 can be expressed as:

$$r = \frac{(S_s - S_c)}{(S_r - S_c)} = \frac{(N_p \times N_r)}{(N_s \times N_p)} = \left(\frac{N_r}{N_s}\right) \rightarrow \frac{(S_s - S_c)}{(S_r - S_c)} = \frac{N_r}{N_s} \tag{6}$$

Equation 3 can also be used to calculate the planet speed when expressed as:

$$r = \frac{(S_p - S_c)}{(S_r - S_c)} = \left(\frac{N_r}{N_p}\right) \tag{7}$$

Equations 6 and 7 are used to derive the formulas for speeds of each rotating component (S_s, S_r, S_c, S_p) and gearmesh frequencies between the sun and planet gears ($GM_{s/p}$) and the planet and ring gears ($GM_{p/r}$). The digression below details the derivation of these formulas. Readers not interested in the derivations may skip to the discussion following the digression.

<u>Digression</u>

Let us first derive the formula for speed of the sun gear (S_s) using Equation 6.

$$\frac{(S_s - S_c)}{(S_r - S_c)} = \frac{N_r}{N_s}$$

multiply both sides of the equation by $(S_r - S_c)$

$$(S_r - S_c) \times \frac{(S_s - S_c)}{(S_r - S_c)} = \left(\frac{N_r}{N_s}\right) \times (S_r - S_c)$$

$$\therefore S_s - S_c = \left(\frac{N_r}{N_s}\right)(S_r - S_c)$$

add S_c to both sides of the equation

$$S_s - S_c + S_c = \left(\frac{N_r}{N_s}\right)(S_r - S_c) + S_c$$

$$\therefore S_s = \left(\frac{N_r}{N_s}\right)(S_r - S_c) + S_c \tag{7a}$$

$$= \left(\frac{N_r}{N_s}\right)S_r - \left(\frac{N_r}{N_s}\right)S_c + S_c$$

$$\therefore S_s = S_r\left(\frac{N_r}{N_s}\right) + S_c\left(1 - \frac{N_r}{N_s}\right)$$

Let's now use Equation 7a to derive the formula for the carrier speed.

$$S_s = \left(\frac{N_r}{N_s}\right)(S_r - S_c) + S_c$$

multiply expressions $\left(\frac{N_r}{N_s}\right)$ *and* $(S_r - S_c)$

$$S_s = \left(\frac{N_r}{N_s}\right)S_r - \left(\frac{N_r}{N_s}\right)S_c + S_c$$

$$\left(\frac{N_r}{N_s}\right)S_c - S_c = \left(\frac{N_r}{N_s}\right)S_r - S_s$$

$$S_c\left(\frac{N_r}{N_s} - 1\right) = \left(\frac{N_r}{N_s}\right)S_r - S_s$$

$$\therefore S_c = \frac{\left(\frac{N_r}{N_s}\right)S_r - S_s}{\left(\frac{N_r}{N_s}\right) - 1}$$

Let's now use Equation 6 to derive the formula for ring gear speed.

$$\frac{(S_s - S_c)}{(S_r - S_c)} = \left(\frac{N_r}{N_s}\right)$$

multiply both sides by $(S_r - S_c)$

$$\therefore S_s - S_c = \left(\frac{N_r}{N_s}\right) \times (S_r - S_c)$$

now multiply both sides by $\left(\frac{N_s}{N_r}\right)$

$$\therefore S_r - S_c = \left(\frac{N_s}{N_r}\right) \times (S_s - S_c)$$

add S_c to both sides

$$S_r = \left(\frac{N_s}{N_r}\right) \times (S_s - S_c) + S_c$$

$$= \left(\frac{N_s}{N_r}\right) S_s - \left(\frac{N_s}{N_r}\right) S_c + S_c$$

$$\therefore S_r = S_s \left(\frac{N_s}{N_r}\right) + S_c \left(1 - \frac{N_s}{N_r}\right)$$

Let's now use Equation 7 to derive formula for planet gear speed.

$$\frac{(S_p - S_c)}{(S_r - S_c)} = \left(\frac{N_r}{N_p}\right)$$

multiply both sides by $(S_r - S_c)$

$$(S_r - S_c) \times \frac{(S_p - S_c)}{(S_r - S_c)} = \left(\frac{N_r}{N_p}\right) \times (S_r - S_c)$$

$$S_p - S_c = \left(\frac{N_r}{N_p}\right) \times (S_r - S_c)$$

add S_c to both sides

$$S_p - S_c + S_c = \left(\frac{N_r}{N_p}\right) (S_r - S_c) + S_c$$

$$S_p = \left(\frac{N_r}{N_p}\right) (S_r - S_c) + S_c$$

multiplying the expressions $\left(\frac{N_r}{N_p}\right)$ *and* $(S_r - S_c)$ results in

$$S_p = \left(\frac{N_r}{N_p}\right) S_r - \left(\frac{N_r}{N_p}\right) S_c + S_c$$

$$\therefore S_p = S_r \left(\frac{N_r}{N_p}\right) + S_c \left(1 - \frac{N_r}{N_p}\right)$$

The gearmesh frequency between the sun and planet gears ($GM_{s/p}$) is equal to the sun gear speed minus the carrier speed, times the number of teeth on the sun gear. Therefore

$$GM_{s/p} = |(S_s - S_c)|\, N_s$$

This frequency is also equal to the planet gear speed minus the carrier speed, times the number of teeth on the planet gear.

$$GM_{s/p} = |(S_p - S_c)|\, N_p$$

Similarly, the gearmesh frequency between the planet and ring gears ($GM_{p/r}$) is equal to the ring gear speed minus the carrier speed, times the number of teeth on the ring gear.

$$GM_{p/r} = |(S_r - S_c)|\, N_r$$

<u>End Digression</u>

The formulas for calculations of the speeds of the sun (S_s), planet (S_p), and ring (S_r) gears and the carrier (S_c) and the sun/planet gearmesh ($GM_{s/p}$) and planet/ring gearmesh ($GM_{p/r}$) are listed below. Speeds of two of the rotating components and number of teeth on all gears must be known to calculate the speeds of the other two components and gearmesh frequencies.

$$S_s = S_r\left(\frac{N_r}{N_s}\right) + S_c\left(1 - \frac{N_r}{N_s}\right) \qquad (8)$$

$$S_c = \frac{S_r\left(\frac{N_r}{N_s}\right) - S_s}{\left(\frac{N_r}{N_s}\right) - 1} \qquad (9)$$

$$S_r = S_s\left(\frac{N_s}{N_r}\right) + S_c\left(1 - \frac{N_s}{N_r}\right) \qquad (10)$$

$$S_p = S_r\left(\frac{N_r}{N_p}\right) + S_c\left(1 - \frac{N_r}{N_p}\right) \qquad (11)$$

$$GM_{s/p} = |(S_s - S_c)|\, N_s \qquad (12)$$

$$GM_{s/p} = |(S_p - S_c)|\, N_p \qquad (12a)$$

$$GM_{p/r} = |(S_r - S_c)|\, N_r \qquad (13)$$

The gear ratios $\left(i.e. \ \frac{N_r}{N_s} \right)$ in the equations above may be positive (+) or negative (-) if the gears are rotating either in the same or opposite directions, respectively.

For the gear train shown in Fig. 6-2, if the sun gear rotates counterclockwise at 30 Hz and the ring gear is fixed, the carrier speed is calculated from Equation 9 as :

$$S_c = \frac{S_r \left(\frac{N_r}{N_s} \right) - S_s}{\left(\frac{N_r}{N_s} \right) - 1}$$

$$S_c = \frac{(0) \left(\frac{-77}{10} \right) - (-30)}{\left(\frac{-77}{10} \right) - 1}$$

$$= \frac{0 + 30}{-\frac{87}{10}}$$

$$= \frac{30 \ x \ 10}{-87}$$

$$S_c = -3.45 \ Hz$$

The carrier is rotating at a speed of 3.45 Hz in the counterclockwise direction. The planet speed is then calculated from Equation 11 as:

$$S_p = S_r \left(\frac{N_r}{N_p} \right) + S_c \left(1 - \frac{N_r}{N_p} \right)$$

$$S_p = (0) \left(\frac{77}{33} \right) + (-3.45) \left(1 - \frac{77}{33} \right)$$

$$= 0 + (-3.45) \left(\frac{-44}{33} \right)$$

$$= 3.45 \ x \ \frac{44}{33}$$

$$S_p = 4.6 \ Hz$$

The planets are rotating at a speed of 4.6 Hz in the clockwise direction. The gearmesh frequency between the sun and planet gears can be calculated using Equation 12 as:

$$GM_{s/p} = | (S_s - S_c) | \ N_s$$

$$GM_{s/p} = | (-30) - (-3.45) | \ (10)$$

$$= | -30 + 3.45 | \ (10)$$

$$= | -26.55 | \ (10)$$

$$GM_{s/p} = 265.50 \ Hz$$

or using Equation 12a as:

$$GM_{s/p} = | (S_p - S_c) | \ N_p$$

$$GM_{s/p} = | (4.6) - (-3.45) | \ (33) = | 4.6 + 3.45 | \ (33)$$

$$= GM_{s/p} = 265.65 \ Hz$$

Similarly, the gearmesh frequency between planet and ring gears can be calculated using Equation 13 as:

$$GM_{p_{l_r}} = |(S_r - S_c)| N_r$$

$$GM_{p_{l_r}} = |(0) - (-3.45)| (77)$$

$$= |0 + 3.45| (77)$$

$$GM_{p_{l_r}} = 265.65\ Hz$$

Fig. 6-4 shows a side view of a double reduction planetary gear train. The primary planets, primary ring, and sun gear have the same number of teeth as the gear train shown in Fig. 6-2. The secondary planet gears have 23 teeth and the secondary ring gear has 66 teeth. The primary and secondary planets rotate at the same speed. The secondary ring gear is now the output gear.

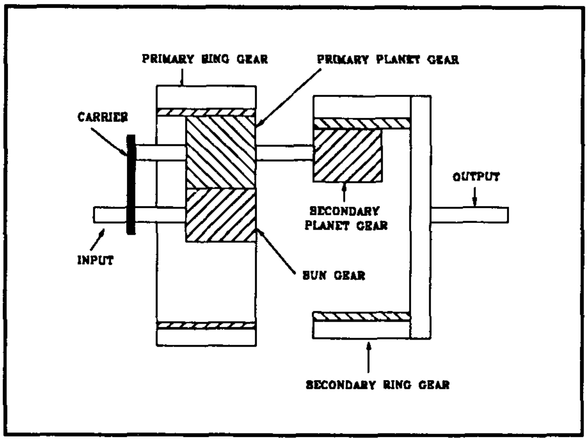

Fig. 6-4. Double Reduction Gear Train.

The gearmesh frequency between the secondary planets and secondary ring can be calculated using Equation 12a. As calculated previously, the planet and carrier speeds are 4.6 Hz and 3.45 Hz, respectively. The secondary gearmesh frequency is then

$$GM_2 = |[(4.6\ Hz) - (-3.45\ Hz)]| (23)$$

$$= 185.15\ Hz$$

The speed of the secondary ring gear (output) may now be calculated using Equation 13:

$$GM_{p_{l_r}} = |(S_r - S_c)| N_r$$

dividing both sides of the equation by N_r yields

$$\frac{GM_{p_{l_r}}}{N_r} = |(S_r - S_c)| N_r \, x \, \frac{1}{N_r}$$

$$\therefore \frac{GM_{p_{l_r}}}{N_r} = S_r - S_c$$

adding S_c to both sides of the equation

$$S_c + \frac{GM_{p_{l_r}}}{N_r} = S_r - S_c + S_c$$

$$\therefore S_r = \frac{GM_{p_{l_r}}}{N_r} + S_c \tag{14}$$

Now, substitute the carrier speed (S_c), number of teeth on secondary ring gear (N_r) and gearmesh frequency between secondary planets and secondary ring ($GM_{p_{l_r}}$) in Equation 14 to calculate the secondary ring speed (output speed)

$$S_r = (-3.45) + \frac{(185.15)}{66}$$

$$= -3.45 + 2.8053$$

$$\therefore S_r = -0.645\ Hz$$

The secondary ring gear is rotating in the counterclockwise direction at 0.645 Hz.

We can use Equation 11 as a check of our calculations.

$$S_p = S_r\left(\frac{N_r}{N_p}\right) + S_c\left(1 - \frac{N_r}{N_p}\right) \tag{11}$$

If we substitute the secondary ring gear (S_r) and carrier speeds (S_c) and number of teeth on the secondary ring gear (N_r) and secondary planet gear (N_p) in Equation 11, we can calculate the speed of the secondary planet gears which should be equal to primary planet gear speed since they are both connected.

$$\therefore S_p = (-0.645)\left(\frac{66}{23}\right) + (-3.45)\left(1 - \frac{66}{23}\right)$$

$$= -1.85 + 6.45$$

$$S_p = 4.6\ Hz$$

Therefore, secondary planet gear speed is 4.6 Hz in the clockwise direction, which is identical to the earlier calculations.

Figs. 6-5 - 6-8 contain data taken with an accelerometer on a planetary gearbox with a configuration identical to that described in the previous examples. The secondary ring gear (output) was rotating at a slightly higher speed than calculated in the example above. Output speed was 0.68 Hz instead of 0.645 Hz. This translates into a secondary gearmesh frequency of about 196 Hz which is seen in Fig. 6-4a (196.25 Hz). Two times secondary

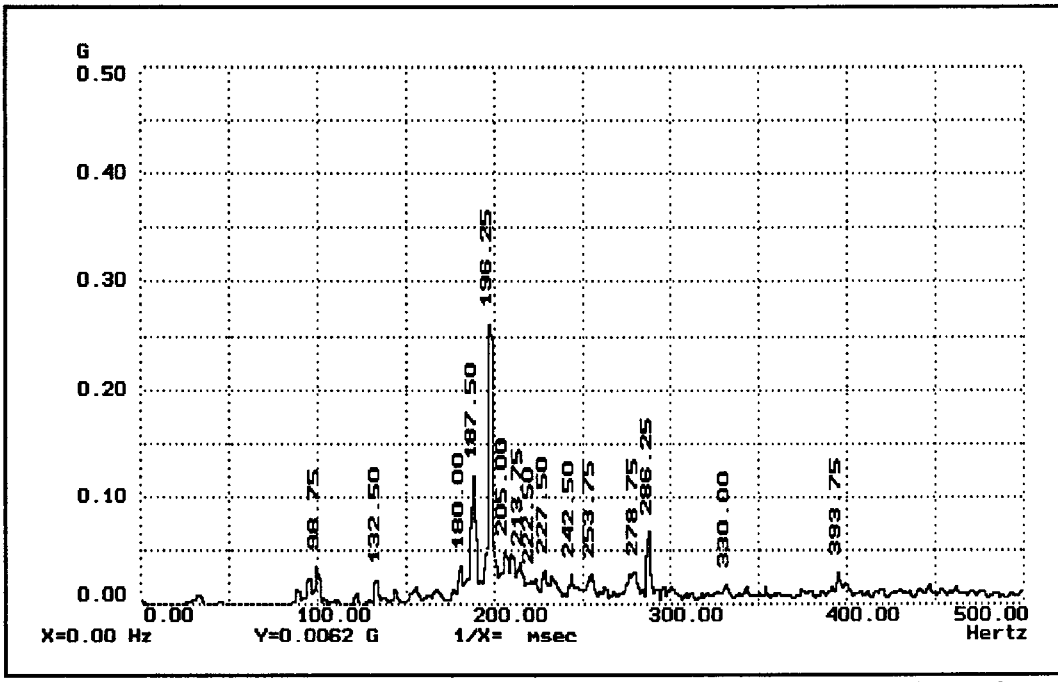

Fig. 6-5. Acceleration Data Showing 1x and 2x Secondary Gearmesh and 1x Primary Gearmesh Frequencies.

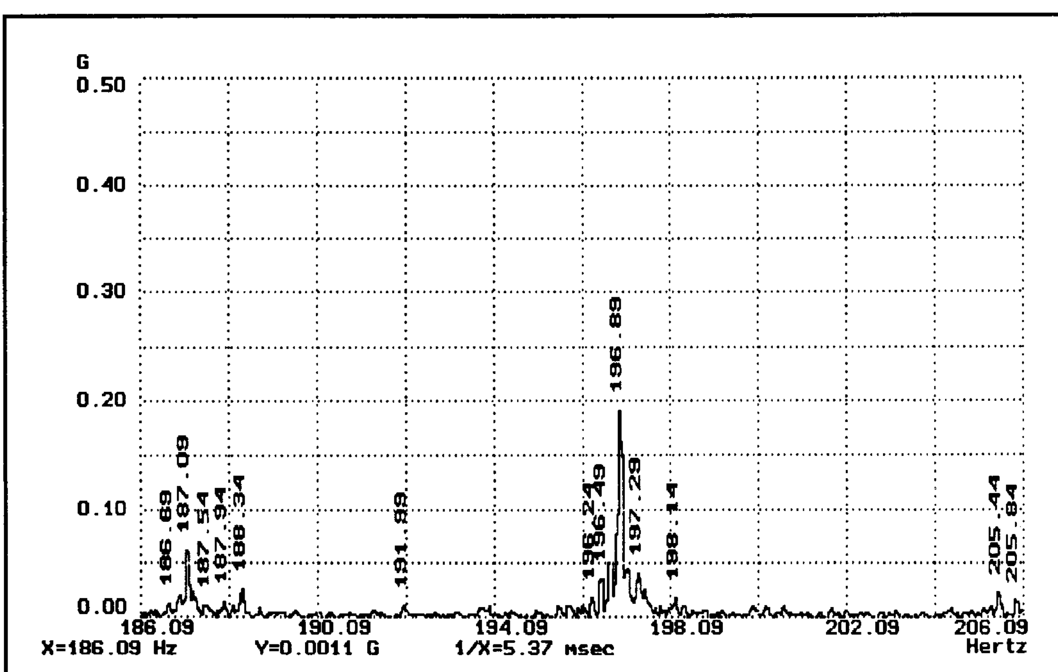

Fig. 6-6. Zoom With 20 Hz Window on Secondary Gearmesh Frequency.

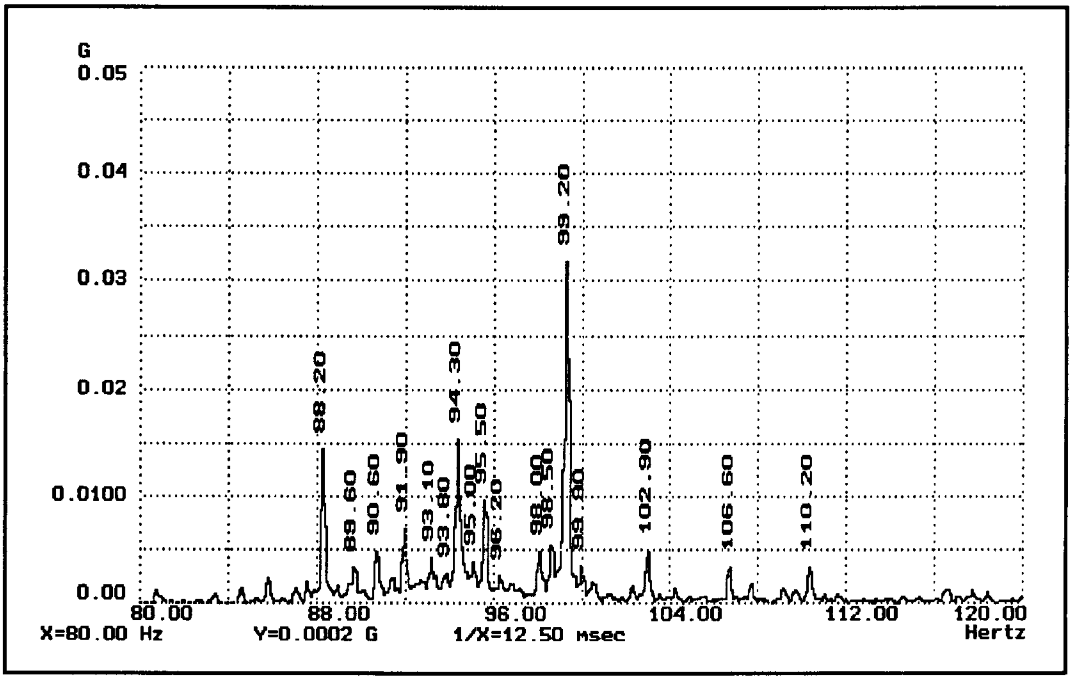

Fig. 6-7. Zoom with 40 Hz Window Showing Multiples of Carrier Speed.

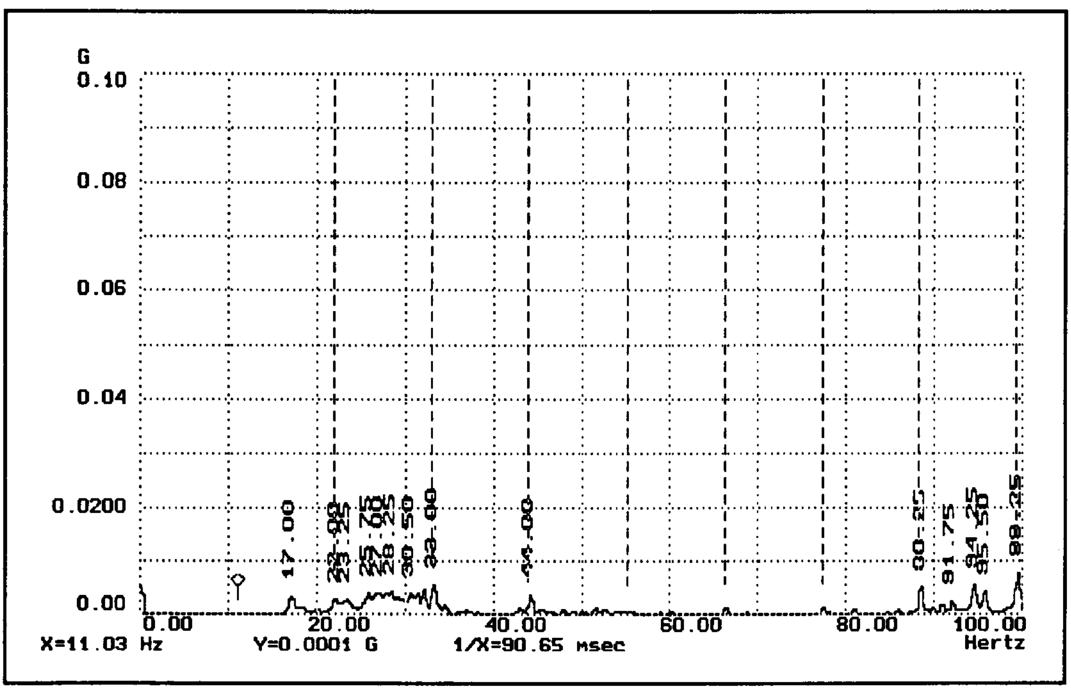

Fig. 6-8. Data Containing 3x Carrier Speed (11 Hz) and Harmonics.

gearmesh frequency (393.75 Hz) is also present in the data. The primary gearmesh frequency of 286.25 Hz is also seen in Fig. 6-5 (instead of 265.65 Hz), calculated earlier.

Fig. 6-6 shows a zoom on the first harmonic of secondary gearmesh frequency (196.89 Hz) with a 20 Hz window. This frequency is modulated by 9.38 Hz which is 2x primary and secondary planet gear speed (4.69 Hz).

Fig. 6-7 shows a high amplitude frequency at 99.2 Hz. This frequency is modulated by 3.67 Hz which is the carrier speed. Now 3x carrier speed is 11.01 Hz. Figs. 6-7 and 6-8 contain harmonics of 11 Hz (22, 33, 44, etc.). 99.2 Hz is the 9th harmonic of 11 Hz. The reason for 3x carrier speed is that the gearbox has three planetary and secondary gears.

This data indicates looseness of the planet carrier. The gearbox was sent out for inspection and the tests confirmed the diagnostics. The difference in amplitudes of the frequencies (88.2 and 99.2 Hz) in Figs. 6-7 and 6-8 is due to data being taken at different points.

Gear life expectancy can be projected based on the common factors. When meshing gears do not have a common factor other than 1, every tooth on one gear must mesh with every tooth on the other gear before the same two teeth are in mesh again. The number of years a good set of gears will last can vary depending on hardness, loading, lubrication, speed, and other variables. A good set should last over 20 years.

If the meshing gears have a common factor, some teeth can wear at an accelerated rate. These worn teeth can cause premature failure of the gears. The realized life of the gear could be the reciprocal of the common factor, as in Table 6-1.

Table 6-1. Life Expectancy.

Common Factor	% of Life Expected
1	100%
2	50%
3	33%
4	25%
5	20%
6	16%
7	14%
8	12%
9	11%
10	10%

However, care should be exercised in drawing conclusions from this table. If gear life for a perfect gear is assumed to be infinity, then 1% of infinity is a long time. Also, if neither gear is eccentric, the gears will last longer. For example, if two gears have a common factor of 5 and one gear is eccentric, a life expectancy of 2.5 years could be reduced to six months. Gear life expectancy in percentage is equal to the inverse of the common factor times 100. For example: The reciprocal of 5 = $\frac{1}{5}$ or 0.2; 2.5 x 0.2 = 0.5 years.

Unknown thousands of dollars are wasted match-marking gears when the common factor is 1. In such cases, match-marking is not required except for timing gears, because every tooth on each gear must mesh with every tooth on the other gear before the same two teeth mesh again. Match-marking of gears may be desirable only when the number of teeth on the two gears has a common factor other than 1. In such cases, deliberate mismatching of the marks may extend gear life.

Amplitude modulation of gearmesh frequency and harmonics reveals useful information about the mesh, for example, misaligned gears, improper backlash, loading, eccentricity, etc. Gearmesh frequency does not usually reveal a broken, cracked, or chipped tooth, except in rare cases when natural frequencies are not measurable. When a gearmesh problem exists, gearmesh frequency is generated. Gearmesh frequency can be modulated by the speed of the problem gear and amplitude modulation can occur. Drosjack and Houser [2] have discussed some forms of frequency modulation when torque loading is present.

A defective tooth on a gear can generate a hit or impact each time the tooth goes into mesh. The repetition rate of this impact is the speed of the gear. The impact excites the axial natural frequencies in helical gears and the radial natural frequencies in spur or straight-cut gears. These frequencies are modulated by the impact repetition rate and tend to be well damped. Each impact is not the same intensity. The resulting frequency spectrum is a well-damped and wide-banded spectral line, normally below gearmesh frequency, and is modulated by the speed of the gear that has the bad tooth.

A set of properly meshing gears may generate a gearmesh frequency of very low amplitude and the spectral line can be narrow banded. Increases in the amplitude of gearmesh frequency can be caused by varying loads, misalignment, improper backlash, and eccentricity in the shaft, teeth, or gear. These events amplitude modulate and/or frequency modulate gearmesh frequency. The Fourier analysis of modulated gearmesh frequency presents a spectral line at gearmesh frequency with plus and/or minus sidebands. The difference frequency between the spectral lines is the speed of the problem gear, assuming one event occurs during each revolution. When two or more events occur during each revolution, the difference frequency will be the number of events times the speed of the problem gear. If both gears have a problem, gearmesh frequency can be modulated by the speed of both gears. In such cases, sidebands can occur at the speed of the two gears and can complicate the analysis process.

Sidebands on the high side of gearmesh frequency indicate eccentric gears. This occurs because there is a constant phase relationship between gearmesh frequency and the speed of the eccentric gear. Therefore, the frequencies add. Sidebands on the low side of gearmesh frequency indicate the gears are loose. This occurs because there is a constantly changing phase relationship between gearmesh frequency and the gear speed. Therefore, the frequencies subtract.

Sidebands present on both sides of gearmesh frequency indicate the gear is eccentric and loose. If the amplitudes of the sidebands are higher on the high side of gearmesh frequency, then eccentricity is the more severe problem. If the amplitudes are higher on the low side of gearmesh frequency, then looseness is the more severe problem. Spectra that have wide-banded noise under the baseline indicate looseness.

In the case of looseness, various combinations of difference frequencies can be present. For example, difference frequencies can occur at G_{s1}, G_{s2}, $2G_{s1} - G_{s2}$, $G_{s1} + G_{s2}$, etc.

The Gears Program can be used to calculate gear frequencies much more efficiently than manual calculations. The only entries required are the number of teeth on each gear, and the speed of one gear or gearmesh frequency. The software provides:

1. Gearmesh frequency plus harmonics
2. Fractional gearmesh frequencies
3. The speed of the gears
4. Hunting tooth frequency
5. The percentage of life expectancy
6. The number of bad teeth on each gear
7. The common factor
8. Recommended AGMA gear quality based on pitch-line velocity

Fig. 6-9 contains the output of the **Gears Program**. Notice all of the previously stated information is presented. Percentage of life expectancy and the recommended AGMA

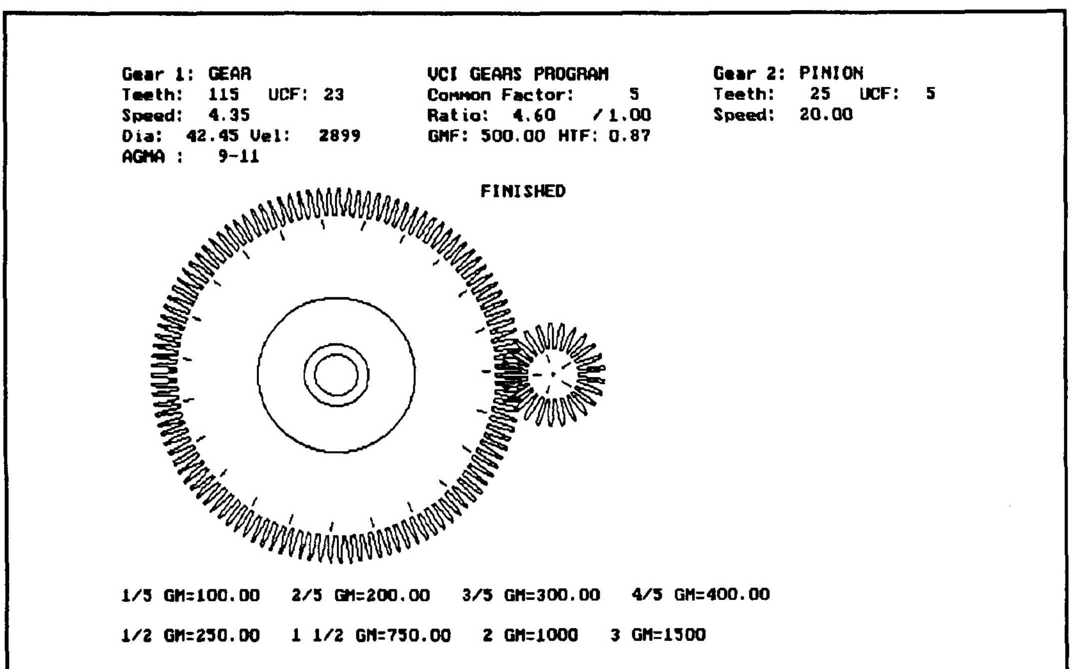

Fig. 6-9. Gears Program Output of 25-Tooth Pinion and 115-Tooth Gear.

gear quality are accessible through pop-up windows.

The American Gear Manufacturers Association, or AGMA, Quality Number System is an established gear quality rating system for specifying gear quality numbers for different pitch line velocities. The gear quality numbers range from 3 to 15, in order of increasing accuracy for spur, helical, bevel, hypoid, rack, and worm gears. Table 6-2 displays a shortcut method of determining the appropriate gear quality numbers for gears.

Table 6-2. Pitch-Line Velocity vs. Recommended AGMA Quality Number [3].

Pitch-line velocity feet per minute	Recommended AGMA quality number
0 - 1000	3 - 8
1,000 - 5,000	9 - 11
5,000 - 10,000	11 - 12
10,000 - 15,000	12 - 13
15,000 - 20,000	13 - 14
20,000 and over	Special control required

Pitch-line velocity of a gear is calculated as [3]:

$$P_v = s\pi P_d \text{, where}$$

$$P_v = \text{pitch line velocity } (FPM)$$

$$s = \text{speed } (RPM)$$

$$\pi = 3.1416$$

$$P_d = \text{pitch diameter } (feet)$$

The pinion speed in Fig. 6-9 is 20 Hz. If an estimated pitch diameter is one foot, the pitch-line velocity is calculated as follows:

$$20\ Hz \times 60 = 1200\ RPM$$

$$P_v = 3.1416\ (1200\ RPM)\ (1\ ft)$$

$$P_v = 3769.92\ FPM$$

The recommended quality number from Table 6-2 is from 9 to 11.

Drago [3] is highly recommended for all personnel involved in vibration analysis, maintenance, or specification and repair of gears.

GEAR PROBLEMS AND CAUSES

We can now examine gear problems encountered in the industry and their possible causes. The major causes of gear problems include: eccentric gears; loose and worn gears; misaligned gears; backlash problems or oscillating gears; and broken, cracked, or chipped teeth. The following examples will be presented to assist in understanding each potential gear problem.

Eccentric gears can take many forms. For explanation purposes, eccentric gears are divided into four broad categories:

1. Meshing gears that have a common factor and one gear is eccentric
2. Gears that do not have a common factor and one or both gears is eccentric
3. Gears that are out-of-round or have several high places
4. Gears installed on a bent shaft

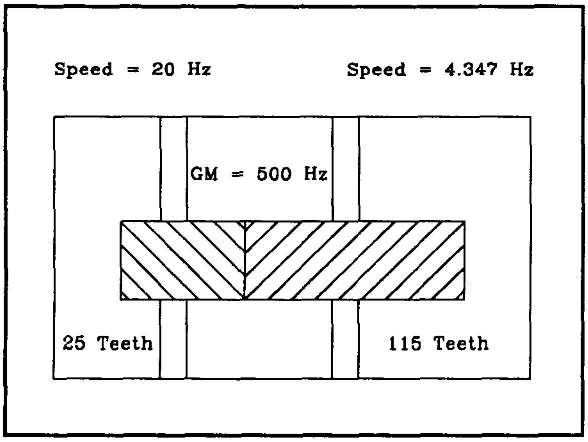

Fig. 6-10. Single Reduction Gearbox.

Meshing gears that have a common factor and one gear is eccentric require further explanation. Fig. 6-10 is an example of a single reduction gearbox. The pinion has 25 teeth and the gear has 115 teeth. Fig. 6-9 contains the completed calculations for this gearbox. The number of teeth on each gear has a common factor of five. If one gear is eccentric, every fifth tooth on the other gear can be imprinted with the eccentricity, whatever form the eccentricity takes.

In some cases, every fifth tooth can become severely worn and can wear every fifth tooth on the other gear. The **Gears Program** in Fig. 6-9 indicates $\frac{1}{5}$, $\frac{2}{5}$, $\frac{3}{5}$, and $\frac{4}{5}$ gearmesh could be present if one gear is eccentric. The data in Table 6-3 indicates the bad place on the eccentric gear will hit the 15th, 5th, 20th, 10th, and zero tooth on the pinion, and repeat this pattern. The final result is every fifth tooth becomes worn or imprinted. The **Gears Program** simulates this process.

Table 6-3. Gear Ratios Evaluated.

Revolutions plus No. of Teeth for 25-Tooth Gear	Revolutions for 115-Tooth Gear
4 + 15	1
9 + 5	2
13 + 20	3
18 + 10	4
23 + 0	5

The gears have uncommon factors of 5 and 23. The common factor identifies the number of teeth from one bad tooth to the next; the uncommon factor identifies the number of bad teeth on each gear. The least common multiple is 5 x 23 = 115. The LCM divided by the uncommon factor identifies how many revolutions the gear must make before the same teeth mesh again:

$$\frac{115}{23} = 5 \textit{ and } \frac{115}{5} = 23$$

This means the small gear must rotate 23 times while the large gear rotates five times. The input shaft rotates at 1200 RPM or 20 Hz. Calculations for the HTF would be:

$$\frac{1}{20} \times 23 = 1.15 \text{ sec}, \quad \textit{then } \frac{1}{1.15} = 0.87 \textit{ Hz}$$

Once again, the amplitude of such a signal would not be seen with normal calibration standards. Gearmesh frequency is $20 \times 25 = 500$ *Hz*.

The product of the **Gears Program** for this set of gears contains these frequencies. This program provides a fast and simple solution for these calculations and indicates every fifth tooth may be worn. Also, this set of gears may last only 20% of the expected life.

$$\frac{1}{5} \text{ gearmesh is: } \quad \frac{500}{5} = 100 \textit{ Hz}$$

The theory that frequency equals events times speed still holds true because $\frac{1}{5}$ gearmesh frequency could also be expressed as:

$$5 \times \textit{pinion speed, or } 5 \times 20 = 100 \textit{ Hz}$$

or

$$23 \times \textit{gear speed, or } 23 \times 4.348 = 100 \textit{ Hz}$$

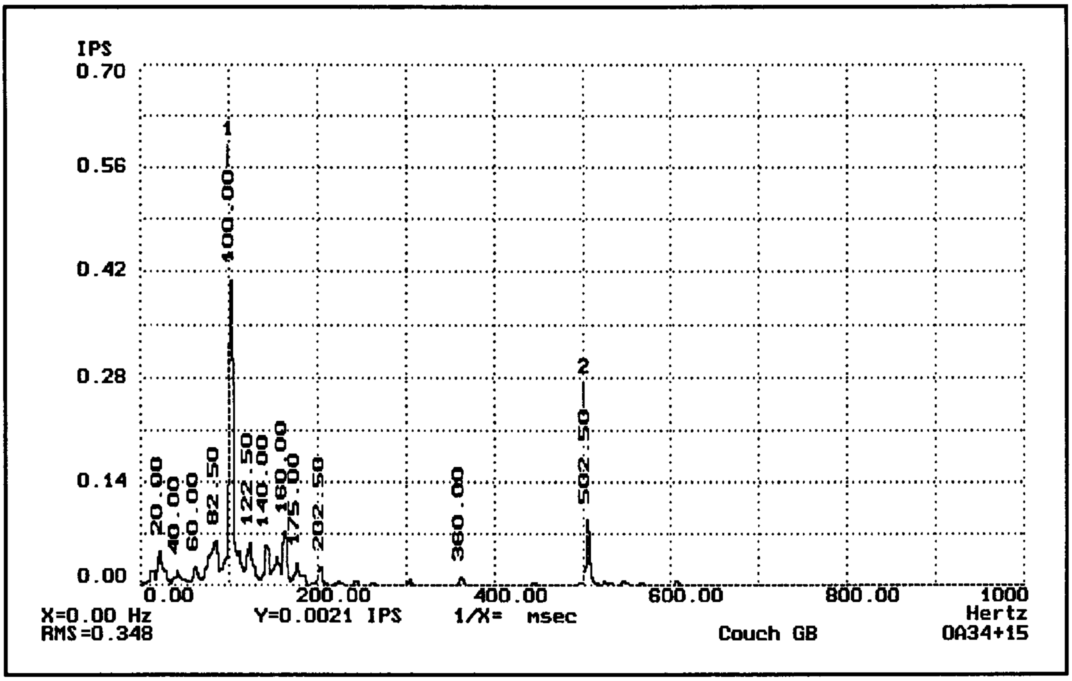

Fig. 6-11. Frequency Spectrum with 1/5 Gearmesh Frequency.

Fig. 6-11 contains the frequency spectrum from the gearbox. Note the high amplitude of the 100 Hz frequency. This means both gears have every fifth tooth worn or imprinted. The spectral line at 100 Hz is modulated. However, we must use a lower frequency range to identify the problem. The spectral line at 502.5 Hz is gearmesh frequency. The modulation around gearmesh frequency is not significant. The amplitude is low and indicates the gears are meshing satisfactorily.

Fig. 6-12 contains the spectrum on a 200 Hz frequency range. The spectral line at 100 Hz is modulated on the high and low sides with difference frequencies of the high and low speed shafts. The higher amplitude modulations on the high side indicate both gears are eccentric, and eccentricity is the more severe problem. The lower amplitudes on the low side of 100 Hz indicate both gears are loose.

Fig. 6-13 contains the time domain signal from this gearbox. The delta time between points 1 and 2 is the pinion speed of 0.05 seconds or $\frac{1}{0.05} = 20\ Hz$. The delta time between each of the five cycles between points 1 and 2 is 0.01 seconds or $\frac{1}{0.01} = 100\ Hz$, which is $\frac{1}{5}$ gearmesh frequency. This frequency could also be expressed as five times pinion speed or 23 times gear speed. Either one equals 100 Hz. The delta time between points 3 and 4 is 0.23 seconds, or $\frac{1}{0.23} = 4.348$ Hz, which is the gear speed. The number of cycles between points 3 and 4 is 23, which is the uncommon factor for the gear. The number of cycles between points 1 and 2 is 5, which is the uncommon factor for the pinion.

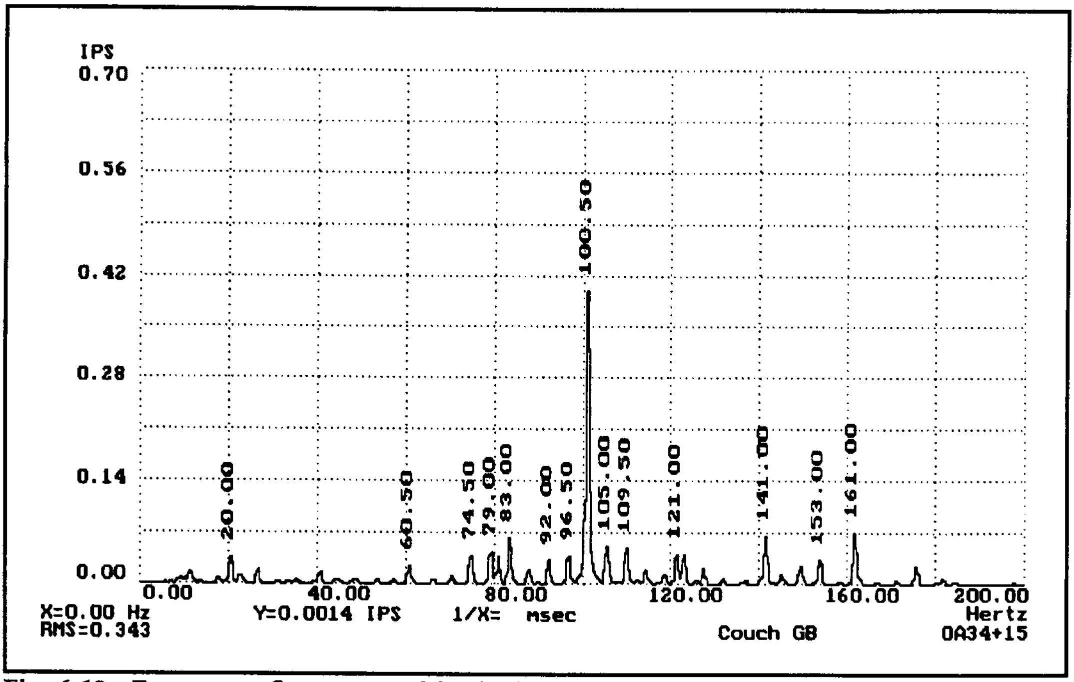

Fig. 6-12. Frequency Spectrum with 1/5 Gearmesh Frequency.

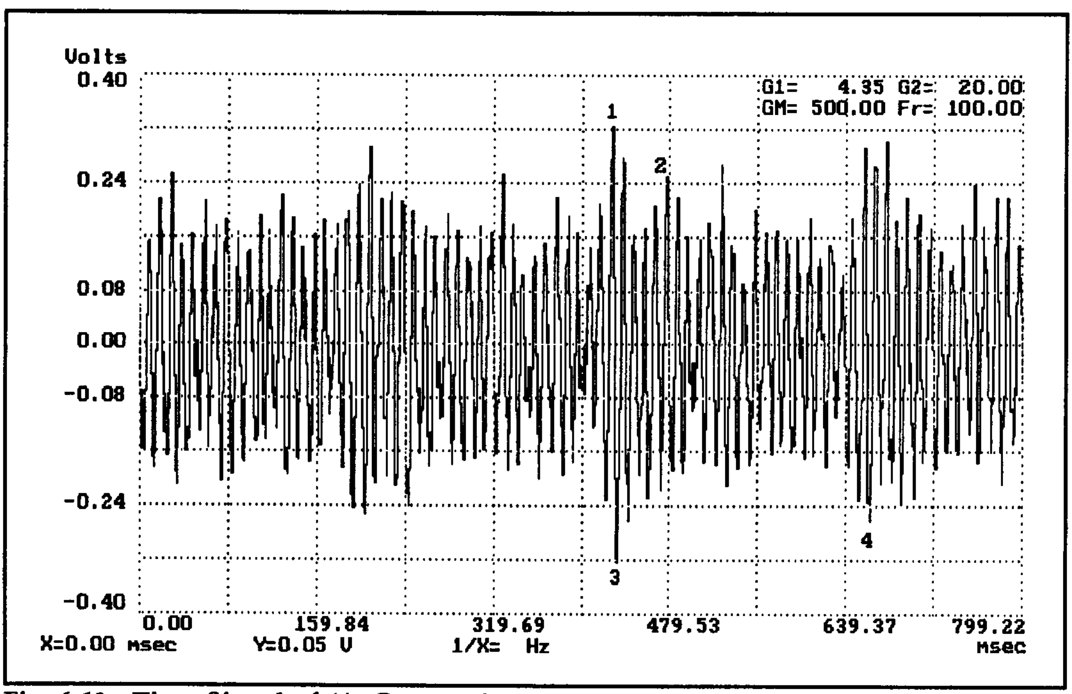

Fig. 6-13. Time Signal of 1/5 Gearmesh Frequency.

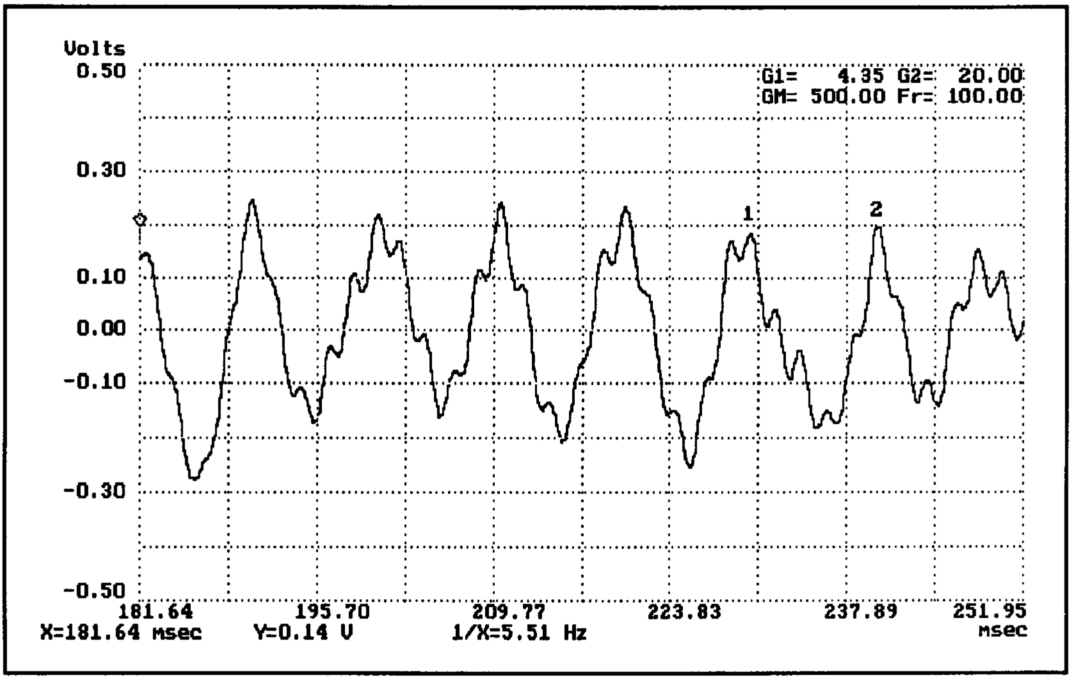

Fig. 6-14. Expanded Time Signal.

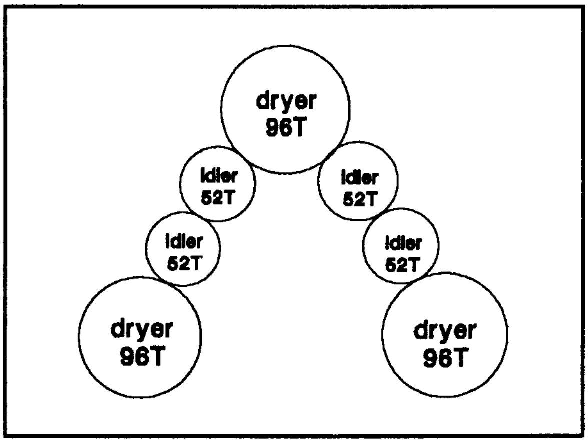

Fig. 6-15. Paper Machine Gear Configuration.

Gearmesh frequency cannot be observed in Fig. 6-13 because more resolution is required. Fig. 6-14 contains the time signal with more resolution. The time period between points 1 and 2 is 0.01 sec, or $\frac{1}{0.01}$ = 100 *Hz*,, which is $\frac{1}{5}$ gearmesh frequency. Each cycle contains five positive peaks. The time period between these peaks is 0.002 seconds, or $\frac{1}{0.002}$ = 500 Hz, which is gearmesh frequency.

It is important to establish that there is no cause-and-effect relationship. The 500 Hz is a high frequency riding the low frequency of 100 Hz. However, the pinion and gear speeds both amplitude modulate the 100 Hz, establishing a cause-and-effect relationship.

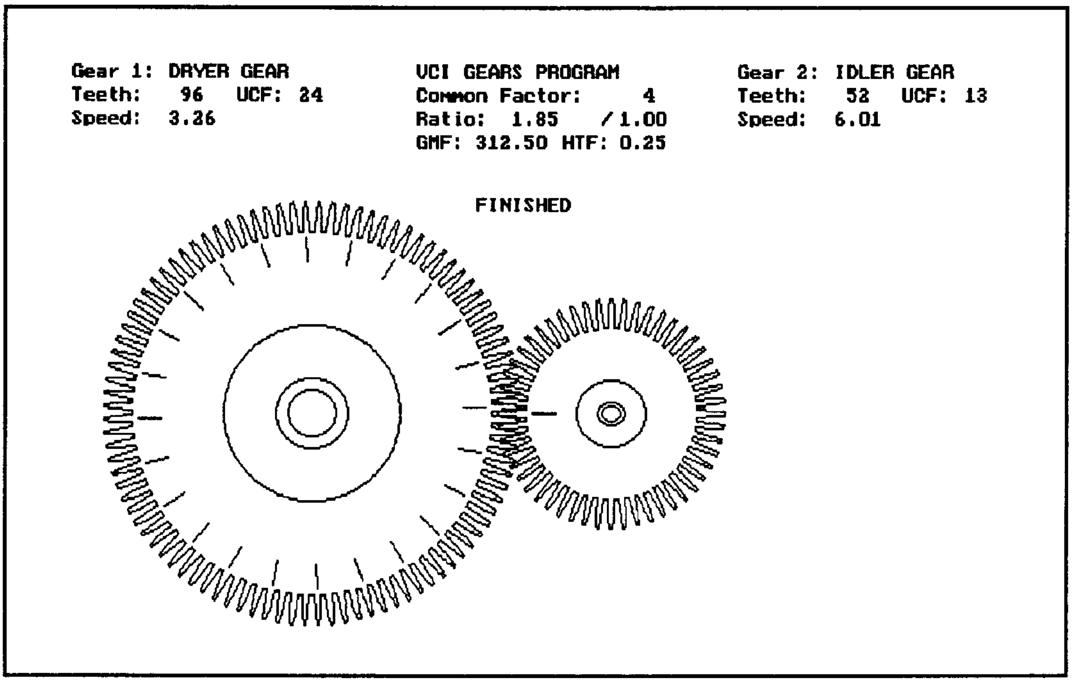

Fig. 6-16. Frequencies Generated by 96- and 52-Tooth Gears.

We cannot tell which gear is eccentric and which gear has worn teeth. However, it may be reasonable to assume the larger gear to be eccentric and both gears worn. In the final analysis, it makes little difference because both gears must be replaced.

The cause of this type of gear failure could be one or more of the following conditions:

1. The gears have an improper ratio because there is a common factor of 5.

2. One or both gears may be eccentric.

3. Use of poor quality gears, i.e. the AGMA gear quality number may be too low.

It is interesting to note that after the gearbox was repaired with a new set of gears, the frequency at 100 Hz was present at startup. The solution to this problem may be to use a different gearbox or to use gears with a higher AGMA gear quality number.

This technique also applies to gears on the drive side of a paper machine. Some of the newer paper machines have a "V" or a "W" gear configuration, as in Fig. 6-15. First we must factor the number of teeth on each gear.

$$\textit{Dryer } 96T = 2 \times 2 \times 2 \times 2 \times 2 \times 3$$

$$\textit{Idler } 52T = 2 \times 2 \times 13$$

The number of teeth on each gear has a common factor of 4, or 2 x 2. This means that

$\frac{1}{4}$, $\frac{1}{2}$, and/or $\frac{3}{4}$ gearmesh frequency could be generated if one gear is eccentric. The numbers in Table 6-4 indicate how the eccentricity in one gear can imprint the teeth on the other gear. Assume the idler gear is eccentric.

Table 6-4. Paper Machine Gear Ratio Evaluation.

Revolutions of Idler Gear	Revolutions of Dryer Gear Plus No. Teeth
1	0 + 52
2	1 + 08
3	1 + 60
4	2 + 16
5	2 + 68
6	3 + 24
7	3 + 76
8	4 + 32
9	4 + 84
10	5 + 40
11	5 + 92
12	6 + 48
13	7 + 04
14	7 + 56
15	8 + 12
16	8 + 64
17	9 + 20
18	9 + 72
19	10 + 28
20	10 + 80
21	11 + 36
22	11 + 88
23	12 + 44
24	13

Please remember it does not matter which gear is eccentric because both or several gears must be replaced, depending on how long the machine runs before the bad gears are replaced.

Analysis of the data above and in Fig. 6-16, from the **Gears Program**, indicates the 52-tooth idler gear must rotate 24 times, while the 96-tooth dryer gear rotates 13 times. If the idler gear is eccentric, then every 4, 8, 12, 16, 20, 24, 28, 32, 36, 40, 44, 48, 52, 56, 60, 64, 68, 72, 76, 80, 84, 88, 92, and 96th tooth becomes worn or imprinted. Then the process starts over again.

Other information provided by the **Gears Program** is included in Table 6-5. Expected life is 25% of normal life. Machine speed is 3681.56 feet per minute (FPM). The dryer cans

Table 6-5. Gears Program Information.

$$common\ factor = 4$$
$$uncommon\ factor\ 96TG = 24$$
$$uncommon\ factor\ 52TG = 13$$
$$dryer\ speed = 3.2552\ Hz$$
$$idler\ speed = 6.0096\ Hz$$
$$GF = 312.50\ Hz$$
$$2 \times GF = 625.0\ Hz$$
$$3 \times GF = 937.5\ Hz$$
$$\frac{1}{4}\ GF = 78.125\ Hz$$
$$\frac{1}{2}\ GF = 156.25\ Hz$$
$$\frac{3}{4}\ GF = 234.375\ Hz$$
$$1\frac{1}{2}\ GF = 468.75\ Hz$$
$$HTF = 0.25\ Hz$$

are 6 feet in diameter. Therefore,

$$D_s = \frac{\frac{FPM}{d \times \pi}}{60}$$

where

D_s = *dryer speed in Hz*

FPM = *machine speed in FPM*

d = *dryer diameter in feet*

π = 3.1416

then,

$$D_s = \frac{\frac{3{,}681.56}{6 \times 3.1416}}{60}$$

$$= 3.2552\ Hz$$

Fig. 6-17 contains the frequency spectrum from the idler gear. The spectral lines at 77.5, 155, and 227.5 are $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$ gearmesh frequency within measurement accuracy on the 1,000 Hz range.

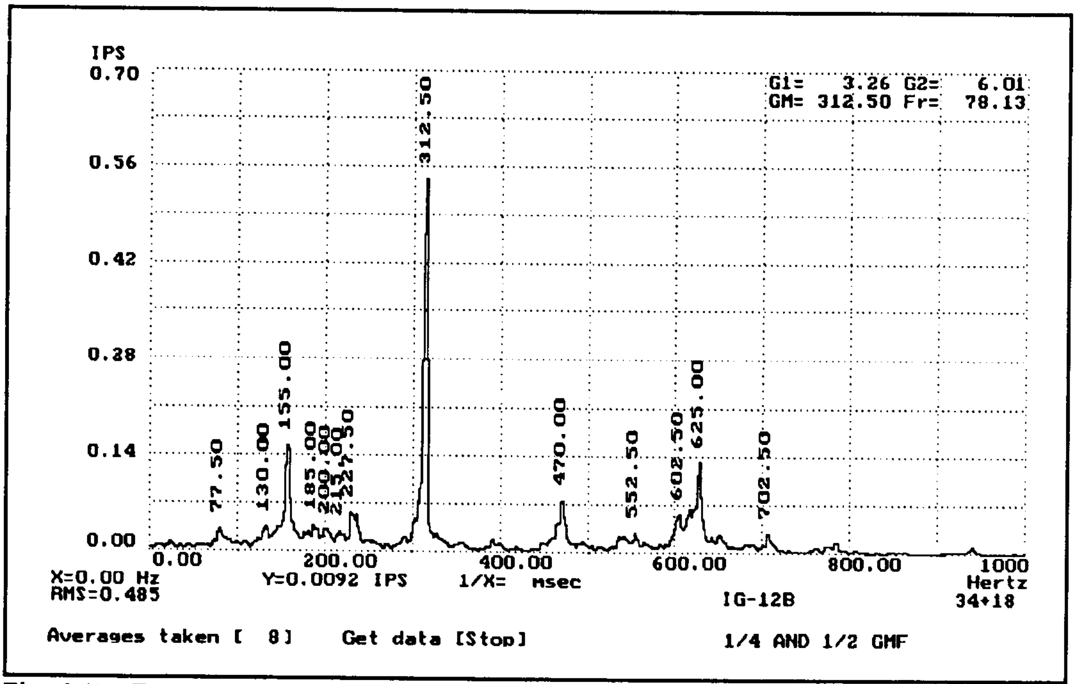

Fig. 6-17. Frequency Spectrum with 1/4 and 1/2 Gearmesh.

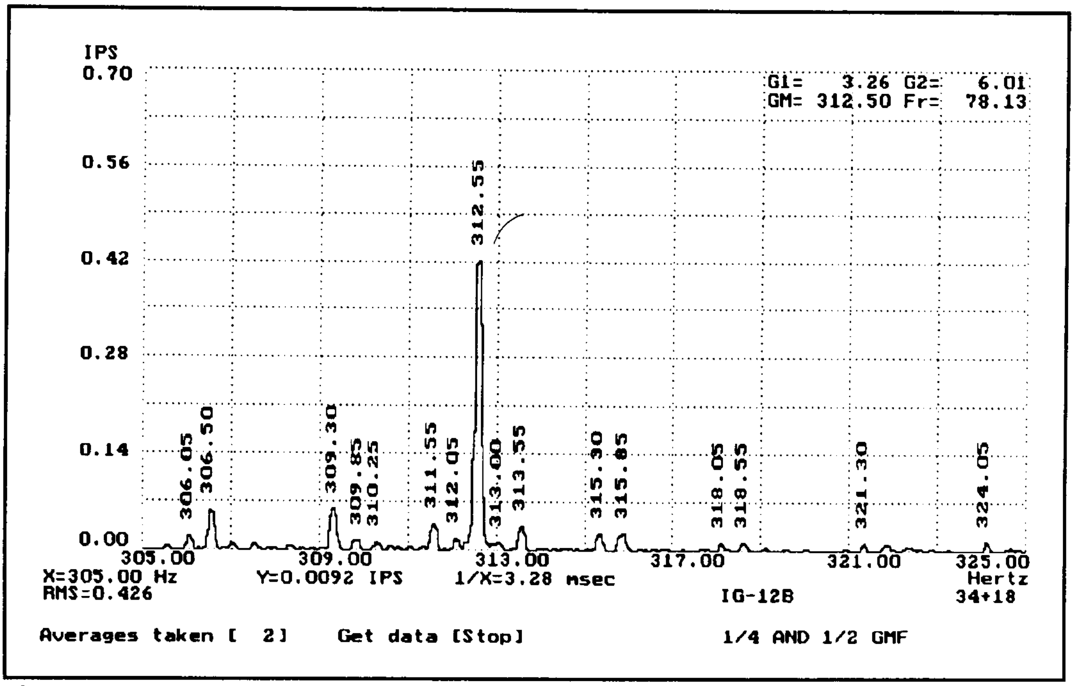

Fig. 6-18. Frequency Spectrum of 20 Hz Window.

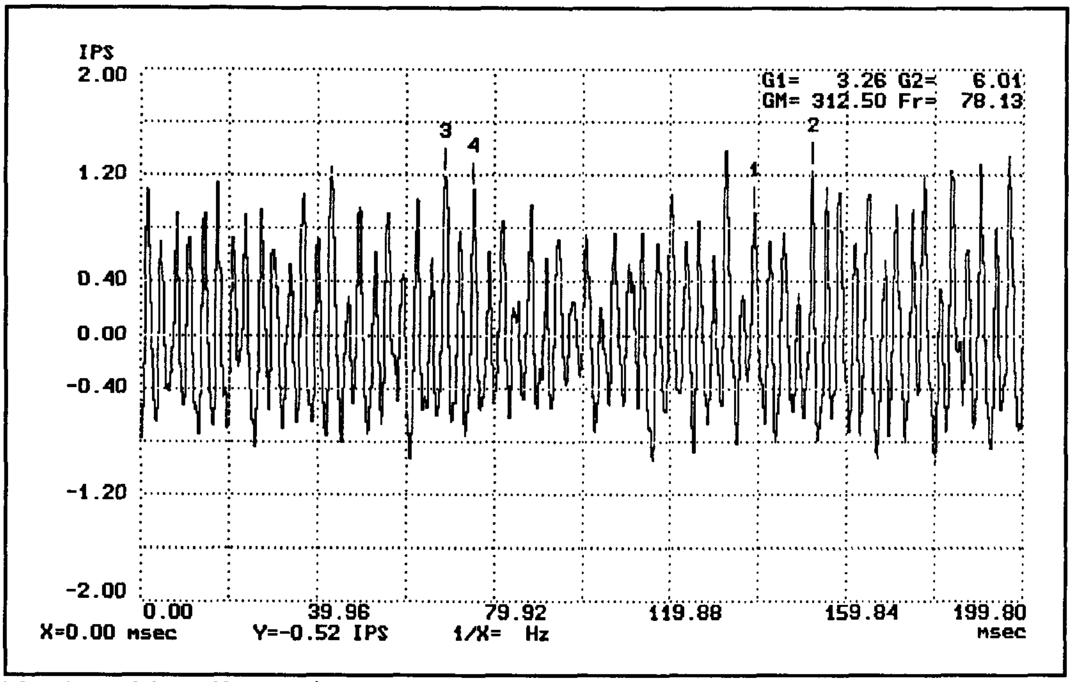

Fig. 6-19. Time Signal of 1/4 and 1/2 Gearmesh Frequency.

Fig. 6-18 is a frequency spectrum of a 20 Hz window around gearmesh frequency. The idler speed is $\frac{312.5}{52}$ = 6.010 Hz. Note the sidebands at gear and idler speeds. The difference frequency of 0.5 Hz is the difference between 2 x main gear speed and 1 x idler speed, or 2 x 3.2550 - 6.010 = 0.50.

The difference frequencies, equal to gear and idler speeds, are higher in amplitude on the low side of gearmesh frequency. This means the gears are loose. The difference frequency of 0.5 Hz is further verification of looseness. These difference frequencies do not occur without considerable distortion in the signal. The broad-banded noise under the spectral lines in Fig. 6-17 is additional evidence of looseness. The difference frequencies, equal to gear and idler speeds on the high side of gearmesh frequency, indicate the gears are eccentric. The fractional gearmesh frequencies in Fig. 6-16 also confirm eccentric gears.

If only the looseness problem is corrected and the eccentric gears are not replaced, the vibration level may increase after the repairs are made.

Fig. 6-19 contains the time signal from the gears. The delta time between points 1 and 2 is 0.01280 seconds and is $\frac{1}{4}$ gearmesh frequency. Also notice the four cycles of gearmesh frequency contained between these points. The delta time between points 3 and 4 is 0.00640 seconds, which is the time period for $\frac{1}{2}$ gearmesh frequency.

The explanation of the $\frac{3}{4}$ gearmesh frequency must be expressed as a function of time.

The time period for each frequency is listed in Table 6-6.

Table 6-6. Time Periods for Frequencies.

$$\frac{1}{GF} = time; \qquad \frac{1}{312.50} = 0.00320 \text{ sec}$$

$$\frac{1}{\frac{3}{4}\,GF} = time; \qquad \frac{1}{234.38} = 0.00427 \text{ sec}$$

$$\frac{1}{\frac{1}{2}\,GF} = time; \qquad \frac{1}{156.25} = 0.00640 \text{ sec}$$

$$\frac{1}{\frac{1}{4}\,GF} = time; \qquad \frac{1}{78.13} = 0.01280 \text{ sec}$$

Analysis of the data indicates the time period for one cycle of gearmesh frequency:

$$\frac{1}{312.50} = 0.00320 \text{ sec.}$$

Four times the time period for one cycle is 4 x 0.00320 = 0.01280. This is the time period for $\frac{1}{4}$ gearmesh frequency. For example, $\frac{1}{78.13}$ = 0.01280.

Also, three times the time period for $\frac{3}{4}$ gearmesh frequency is 3 x 0.00427 = 0.01281 seconds, which is also the time period for $\frac{1}{4}$ gearmesh frequency.

The time period between points 1 and 2 in Fig. 6-20 is 0.0032, or gearmesh frequency. The time period between points 1 & 3 is 0.00640, which is $\frac{1}{2}$ gearmesh frequency. The time period between points 1 and 4 is 0.00427 seconds, which is $\frac{3}{4}$ gearmesh frequency. The difference between the time periods of gearmesh and $\frac{3}{4}$ gearmesh is only 1.07 milliseconds.

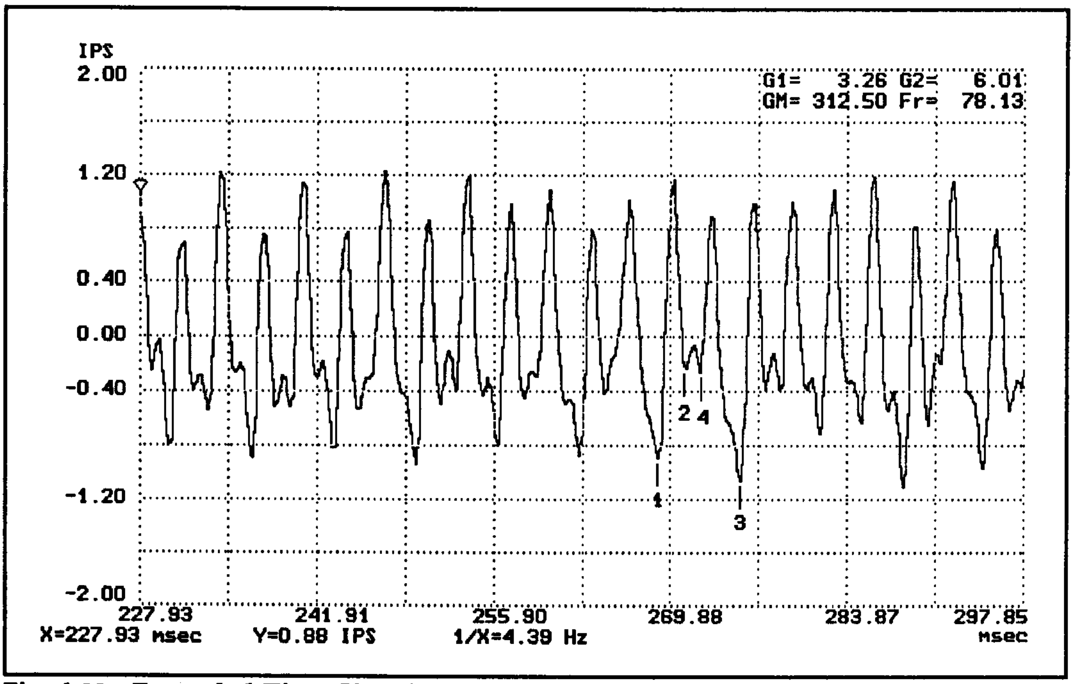

Fig. 6-20. Expanded Time Signal.

How long the gears will last depends on several factors which include:

1. Speed
2. Common factors
3. Amount of eccentricity
4. Number of eccentric teeth

A gear could be within tolerance when measured alone. However, if some point is on the outer edge of tolerance on one or both gears, then thermal expansion could cause an out-of-tolerance condition at operating temperatures. Once the process is set in motion, when a common factor other than 1 is present, the problem increases until it is unbearable or an associated failure occurs. Once again, the length of time is determined by the above factors.

The correction of these problems is not a simple matter. It is impractical to redesign the gears at this point. The only choice left may be to replace the gears with a higher AGMA gear quality number on a systematic basis. In severe cases on paper machines, it may be desirable to remove all idler gears except the input drive idler and drive the machine with the felt and dryer can. Both of these choices have either economic and/or operational impact and must be evaluated locally on a case-by-case basis.

Gears that do not have a common factor and one or both gears are eccentric require a more detailed understanding. When gears have a proper or an improper ratio and one gear is eccentric, fractional gearmesh frequency can be generated. Often, the fractional gearmesh frequency is not what we expect. For example, $\frac{1}{2}$ gearmesh frequency can be generated when every other tooth contains some form of eccentricity. These forms of eccentricity cause every other cycle of gearmesh frequency to be higher in amplitude. When this occurs, $\frac{1}{2}$ gearmesh frequency is generated regardless of the tooth ratio. These eccentric teeth can generate $\frac{1}{2}$, $1\frac{1}{2}$, $\frac{1}{3}$, $\frac{2}{3}$, or $1\frac{1}{3}$, etc., gearmesh frequency. This type of eccentricity normally affects a few teeth in one or more places on a gear. The form of this eccentricity is usually thick teeth, pitch-line runout, or unevenly spaced teeth.

Fig. 6-21 is the **Gears Program** output of a gear and idler gear in mesh. The dryer gear has 126 teeth and the idler gear has 81 teeth. The common factor is 9. Gearmesh frequency is 355 Hz. Fractional gearmesh frequencies, gear speeds, HTF, number of teeth per gear, and number of revolutions per gear are presented.

Since the common factor is 9, every ninth tooth could be imprinted if one tooth is eccentric. The following fractional gearmesh frequencies could be generated:

$$\frac{1}{9}, \frac{2}{9}, \frac{1}{3}, \frac{4}{9}, \frac{5}{9}, \frac{2}{3}, \frac{7}{9}, \text{ and } \frac{8}{9}.$$

Also note that fractional gearmesh frequencies of $\frac{1}{2}$ and $1\frac{1}{2}$ are present.

The result in Fig. 6-21 is when one gear has one eccentric tooth. The result in Fig. 6-22

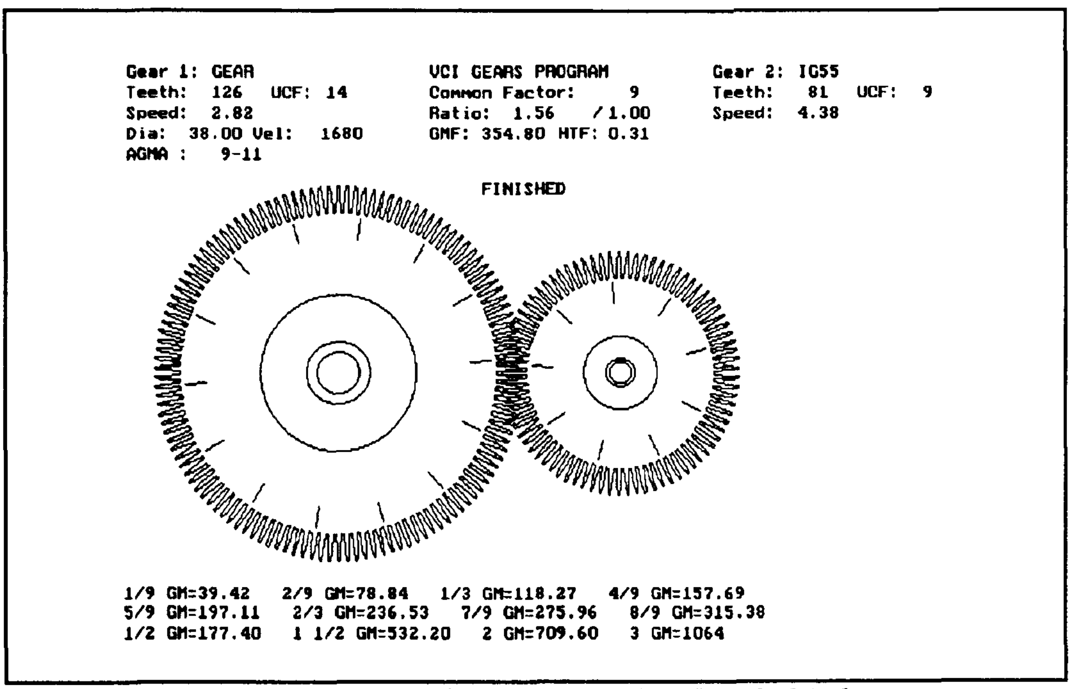

Fig. 6-21. Gears Program Output of a Dryer and Idler Gear in Mesh.

is when one gear has every other tooth eccentric for a total of four eccentric teeth. Each gear is imprinted with groups of eccentric teeth, and every other tooth in the group is eccentric. The large gear has 14 groups, which is the number of revolutions the small gear must make for one HTF to occur. The small gear has nine groups, which is the number of revolutions the large gear must make for one HTF. The marking of the teeth can be simulated with the **Gears Program**. When this type of eccentricity occurs, $\frac{1}{2}$ and/or $1\frac{1}{2}$ gearmesh frequency are the only fractional gearmesh frequencies generated.

Figure 6-23 is a frequency spectrum from the idler gear. Gearmesh frequency is 355 Hz. $\frac{1}{2}$ gearmesh frequency is 177.5 Hz. $1\frac{1}{2}$ gearmesh frequency is 532.5 Hz. The frequencies are contained in Fig. 6-21 and Fig. 6-22. To identify sidebands of gearmesh frequency, it is necessary to zoom in on gearmesh frequency.

Fig. 6-24 is a frequency spectrum of a 40 Hz window around gearmesh frequency. The difference between points 1 & 2 and points 1 & 4 is approximately 2.8 Hz, which is gear speed. The difference between points 1 & 3 and points 1 & 5 is approximately 4.4 Hz, which is idler speed. The difference frequency of gear speed is higher in amplitude on the low side of gearmesh frequency. This indicates the gear is loose. The difference frequencies of gear and idler speeds on the high side of gearmesh frequency indicate the gears are eccentric. In this case, looseness is the more severe problem. If the looseness is removed, the amplitude could increase and eccentricity could become the more severe problem.

Fractional gearmesh frequency is sometimes generated when specific teeth on one gear are in mesh with certain teeth on the other gear. This occurs when some teeth are borderline eccentric, meshing with other teeth that are borderline eccentric. This problem can also occur when some teeth on one gear are eccentric. This problem makes a strong case for viewing long periods of the time domain signal.

Fig. 6-25 is the time domain signal from the idler gear. Every other cycle is higher in amplitude. This is how $\frac{1}{2}$ gearmesh frequency is generated.

Each cycle of the time domain signal is a profile of two teeth meshing. For example, the time period between points 1 and 2 is two cycles, representing two different teeth going into mesh. The delta time between points 1 and 2 is 5.65 ms, or $\frac{1}{2}$ gearmesh frequency (177 Hz). The delta time between points 1 and 3 is 2.8 ms or 355 Hz, which is gearmesh frequency. The amplitude of each cycle indicates the stress on the tooth when it goes into mesh.

Fig. 6-26 is an expanded time domain signal. The time between points 1 and 2 is 2.815 ms, which is 355 Hz, or gearmesh frequency. The time between points 1 and 4 is 5.63 ms or 177.5 Hz, which is one half gearmesh frequency. The time between points 3 and 4 is 1.87 ms or 535 Hz, which is $1\frac{1}{2}$ gearmesh frequency.

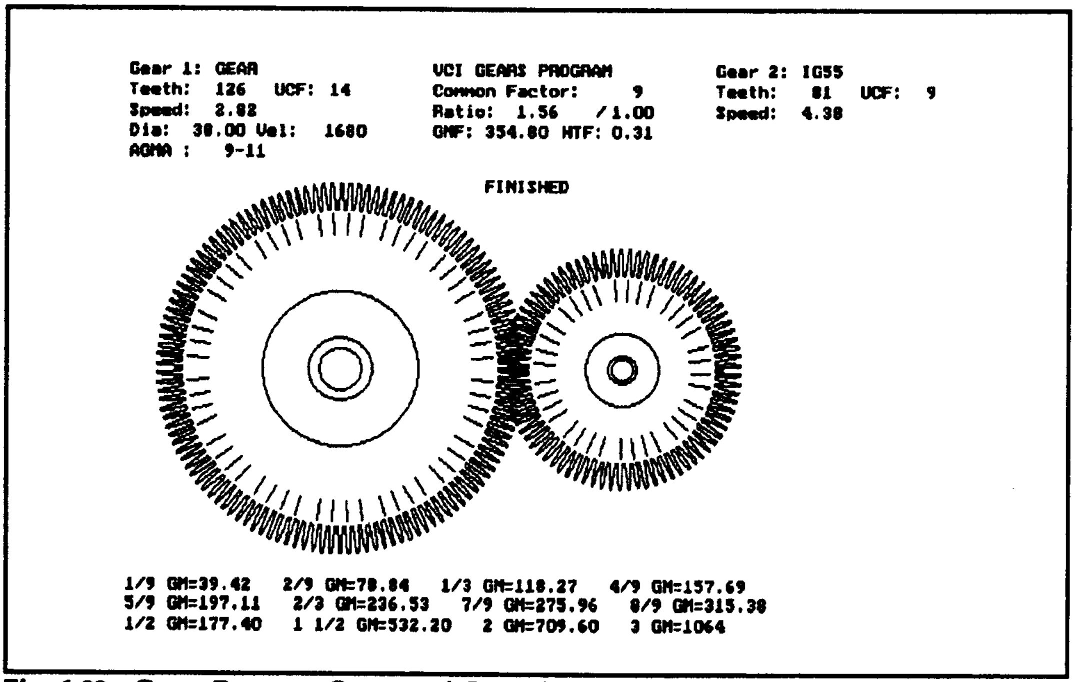

Fig. 6-22. Gears Program Output of Gears in Mesh.

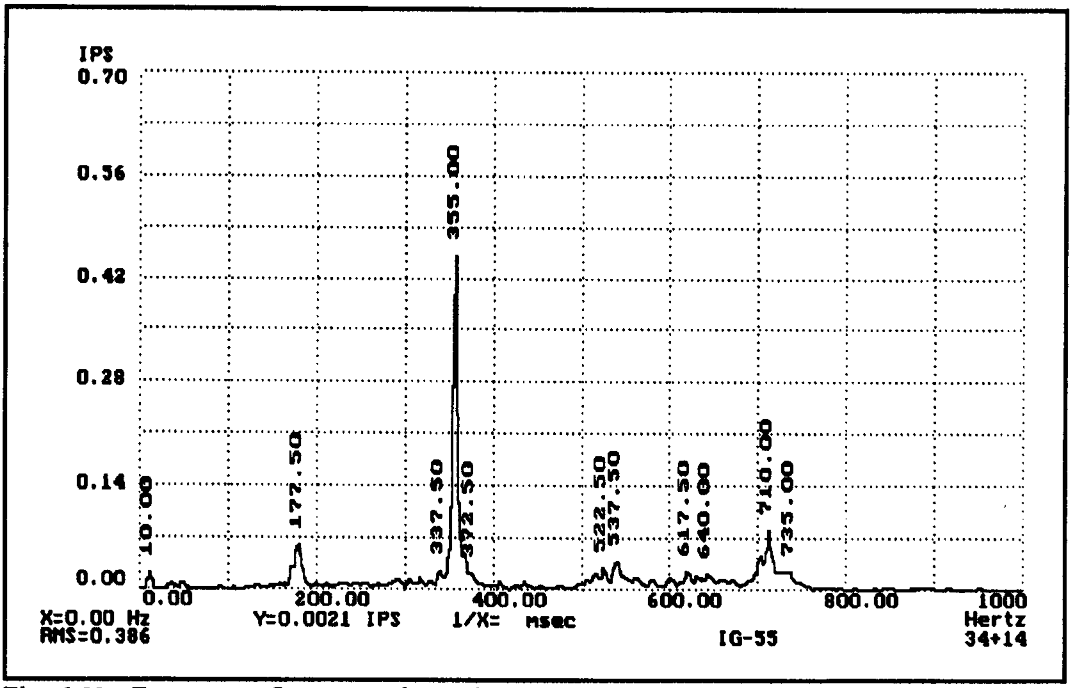

Fig. 6-23. Frequency Spectrum from the Idler Gear.

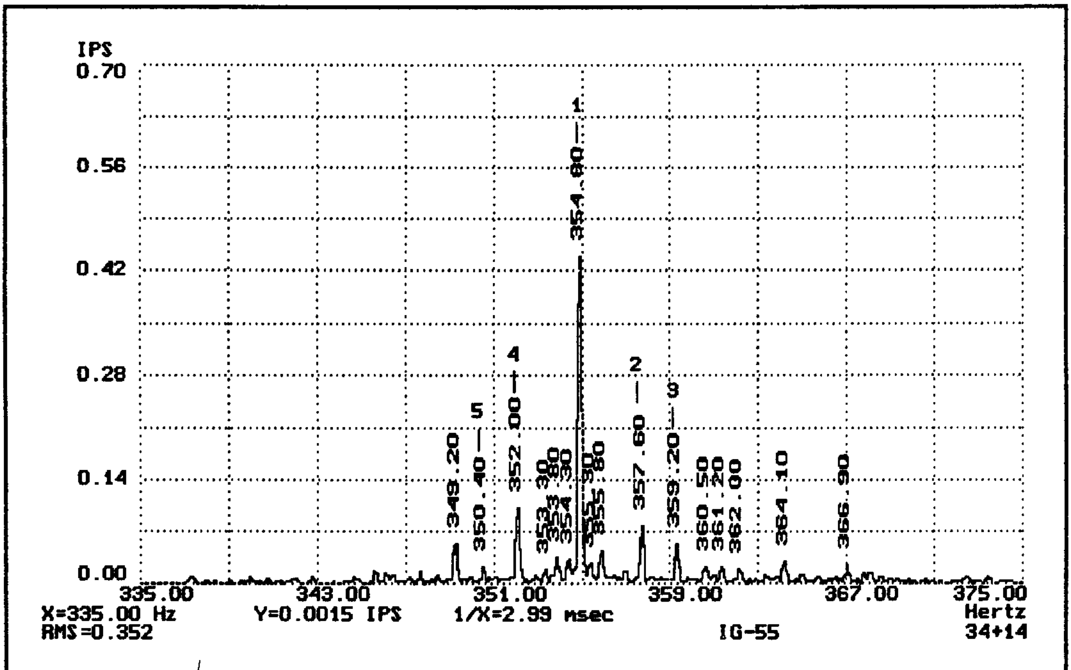

Fig. 6-24. Frequency Spectrum of 40 Hz Window around Gearmesh Frequency.

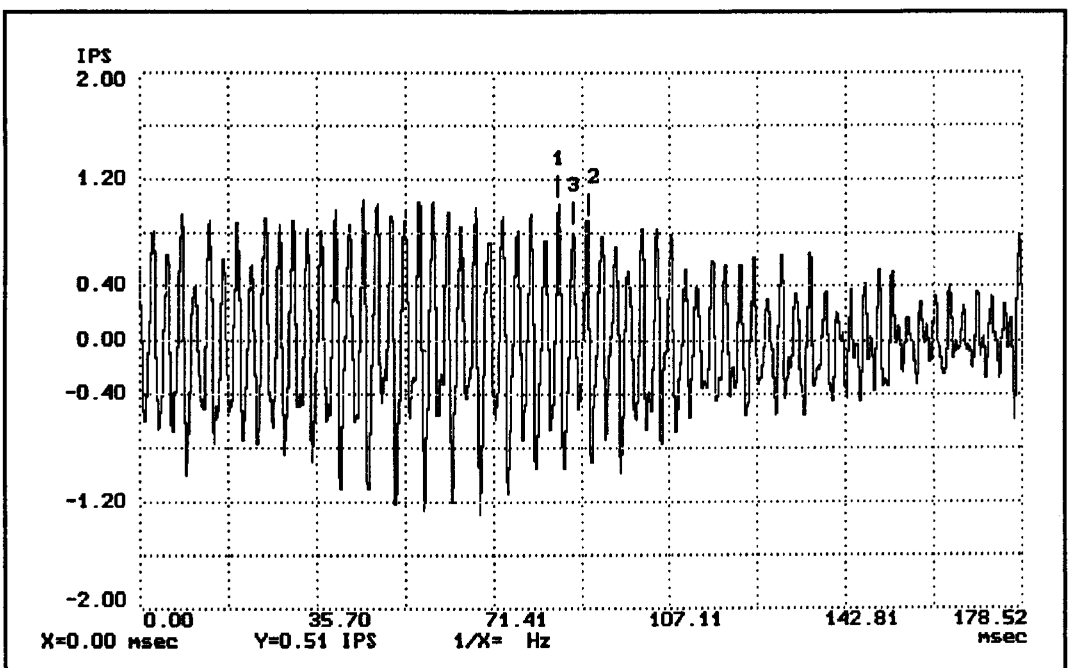

Fig. 6-25. Time Domain Signal for Idler Gear.

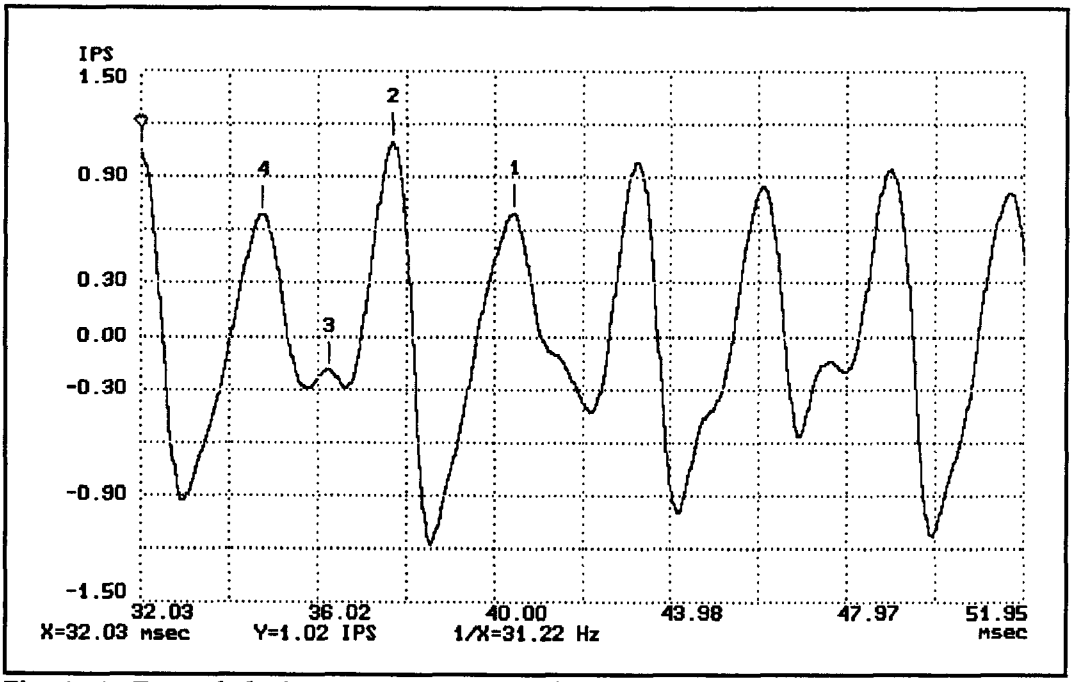

Fig. 6-26. Expanded Time Domain Signal of the Idler Gear.

Every other $\frac{1}{2}$ cycle has a longer time period. There is some speculation that every other tooth is thicker, causing $1\frac{1}{2}$ gearmesh frequency when the teeth go into mesh. Mathematically, this speculation has not been proven and should only be noted as a knowledgeable speculation.

In conclusion, eccentric gears that produce fractional gearmesh frequencies require much detailed analysis to verify the eccentricities. In the above case, the gear seemed to be more loose than eccentric, but still showed characteristics of eccentric gears. To verify the eccentricities of the gear, the time domain signal was required. If the time period for one revolution was displayed with enough resolution, every tooth on the gear (one cycle equaling one tooth going into mesh) could be identified. The **VCI Gears Program** simulates the process, showing why the fractional gearmesh frequencies occur.

Gears that are out-of-round or have several high places are common forms of eccentricity in gears that have spokes. See Fig. 6-27. In some cases, the high places occur at or near the spokes.

The cause of these high places has not been documented. However, it appears reasonable to assume one or more of the following are major contributors:

1. Something could have gone wrong during fabrication.

2. When the gear is heated and installed on a shaft that has a slight interference fit, the gear may not go back to its original shape

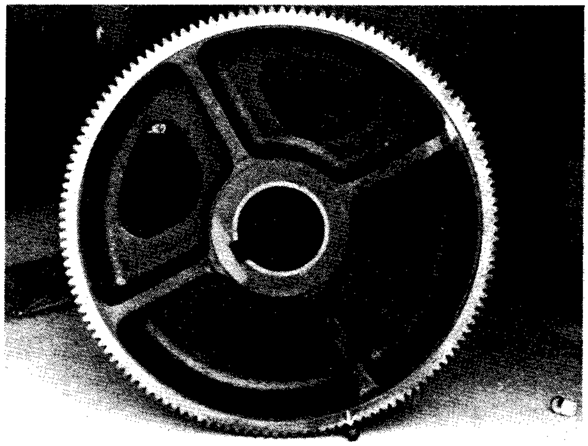

Fig. 6-27. A Gear with Spokes.

after it cools.

3. The coefficient of expansion may be greater at the spokes when the gear reaches operating temperature.

Sometimes the number of high places can be more or less than the number of spokes. These high places could be caused during fabrication, by segments in older gears, by inconsistent hardening of the gear, etc.

Whatever the cause of the high places, vibration data taken from these gears exhibit unique spectral frequency components. A spectral line is generated at events times speed. The number of events is the number of high places on the gear. If there are four high places, the frequency is four times speed. If there are five high places, the frequency is five times speed, etc. A second harmonic of this frequency can also be present, and the frequency can appear as sidebands around the gearmesh frequency. For example, if a gear has four high places, the gearmesh frequency is normally modulated by four times the speed of the eccentric gear. Sidebands with difference frequencies equal to gear speed can also be present, however, the fourth sideband will be higher in amplitude.

Some dryer gears on paper machines have spokes. When these gears are out-of-round or have high places, a spectral line is present at the number of high places times speed. Fig. 6-28 is a frequency spectrum of a gear that has five high places. The gear has 84 teeth and rotates at 2.008 Hz. The gearmesh frequency is 2.008 x 84, or 168.7 Hz. Five times dryer speed is 2.008 x 5 = 10 Hz. Note the spectral line at 10 Hz. Also, note gearmesh frequency has a spectral line on the high side at 178.7, which is 10 Hz above gearmesh frequency. This means the gear is eccentric.

Fig. 6-29 is the time signal from the gear. The amplitude of gearmesh frequency increases five times in one revolution. Each time the amplitude increases, the high place is in mesh. Each revolution is not exactly the same because the high places are meshing against different teeth each revolution and enough time is not presented to see the variation.

Fig. 6-30 is the expanded time signal. The time period between each peak is 0.1 seconds

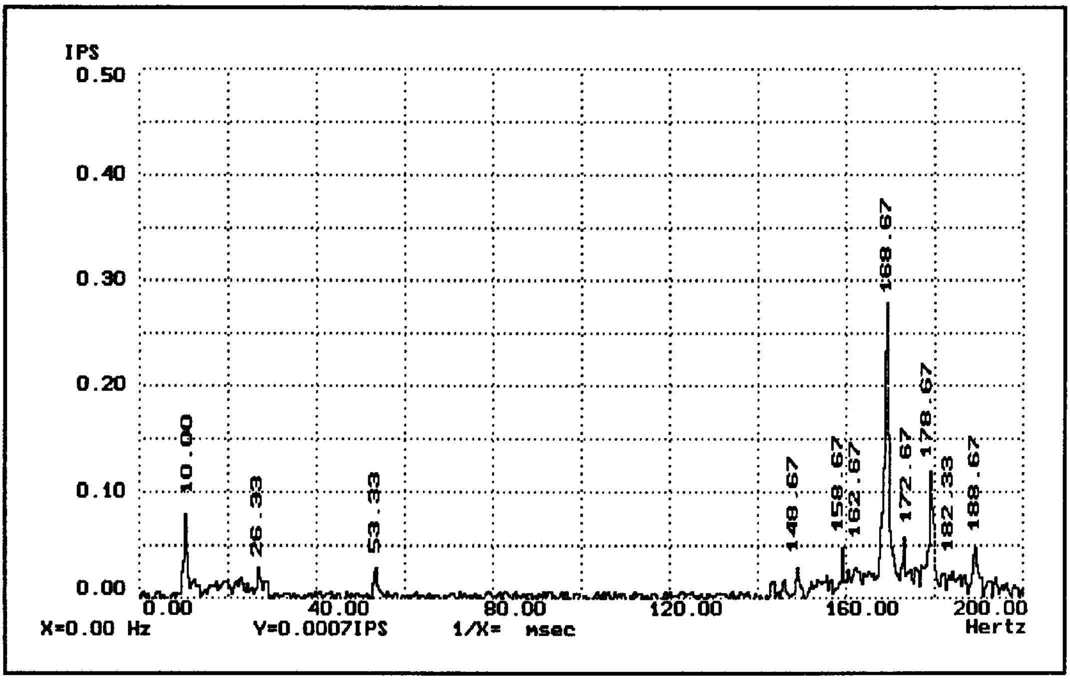

Fig. 6-28. Frequency Spectrum from a Dryer Gear.

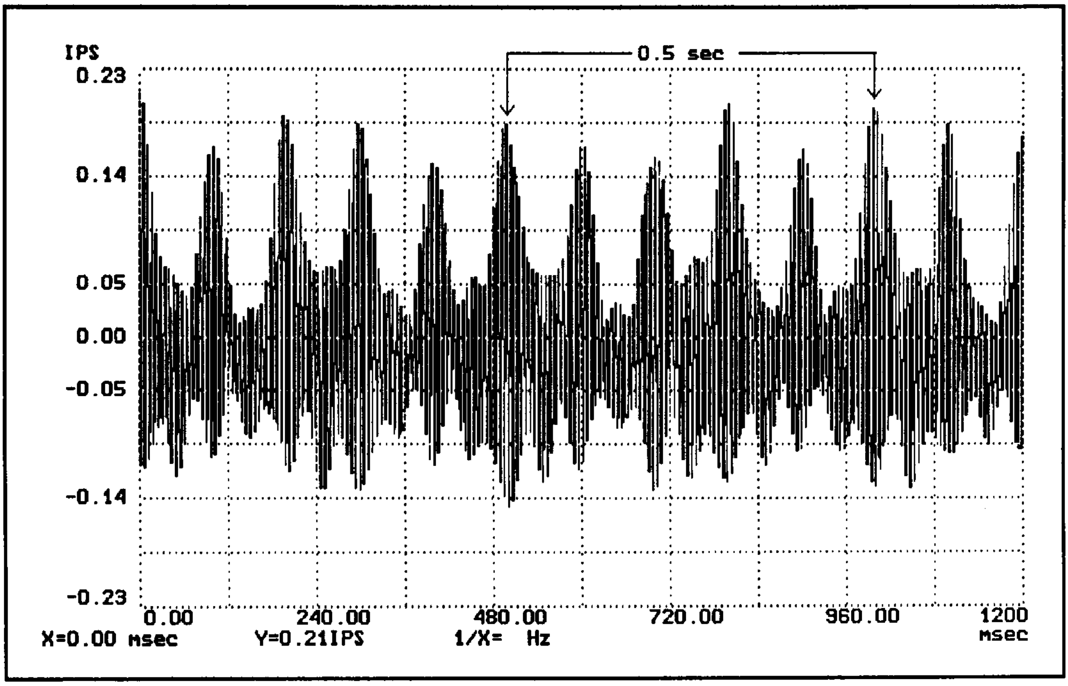

Fig. 6-29. Time Signal from the Dryer Gear.

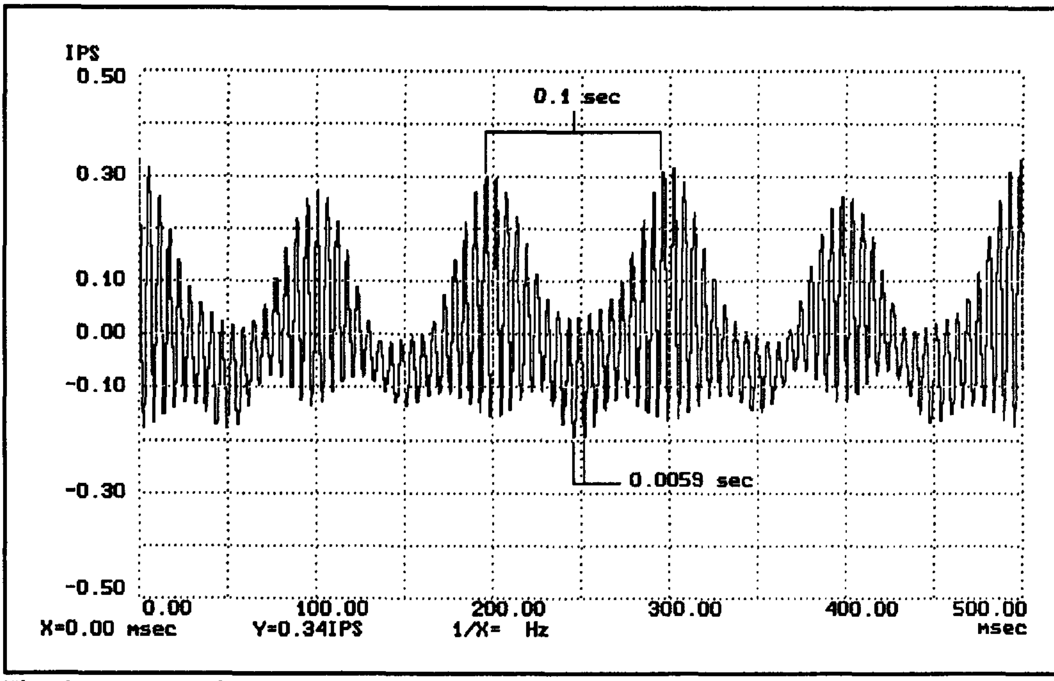

Fig. 6-30. Expanded Time Signal.

or $\frac{1}{0.1}$ = 10 Hz. It is difficult to determine the cause of this problem without access to the gear. However, it may be reasonable to assume thermal expansion caused the high places at the spokes or the gear was manufactured with the high places. The AGMA gear quality number could also be too low. At any rate, this case history dramatically indicates the need to obtain baseline data on all new installations before the machines are accepted.

Gears installed on a bent shaft may or may not generate a high amplitude of gearmesh frequency. A high amplitude of gearmesh frequency is generated when a gearmesh problem exists. Therefore, the shaft must be bent enough to cause a meshing problem. High amplitudes of gearmesh frequency may not be present.

Fig. 6-31 is the **Gears Program** output display for a gear in mesh with an idler gear. Gearmesh frequency is 172.13 Hz. Idler and gear speeds are 3.375 Hz and 2.265 Hz, respectively.

Fig. 6-32 is a frequency spectrum of the velocity data from the gear. Note that the spectral line at 171.67 Hz is gearmesh frequency. The amplitude is very low in this case, and a gearmesh problem does not exist. The frequency spectrum is a little above the baseline, indicating some looseness.

Fig. 6-33 is a frequency spectrum of the idler gear. Idler gear speed is 3.375 Hz. This spectrum contains five multiples of idler speed, which indicates looseness. Also, the frequency spectrum is up off the baseline, again indicating looseness. This data was measured with a velocity transducer that has a frequency response from 10 to 2,000 Hz.

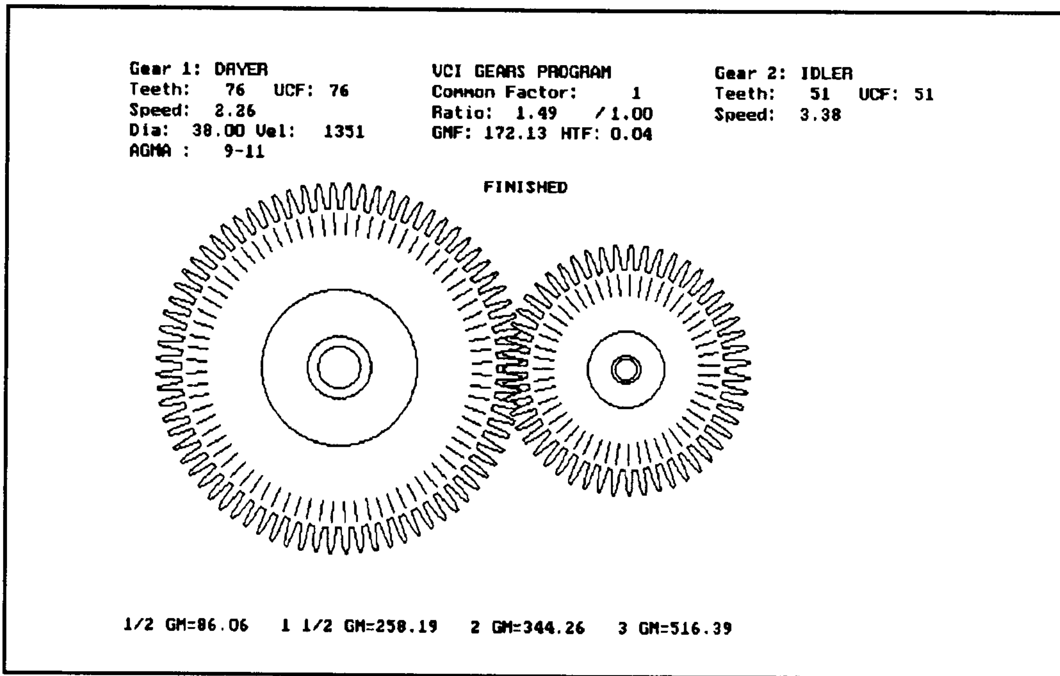

Fig. 6-31. Gears Program Output for Dryer and Idler Gear.

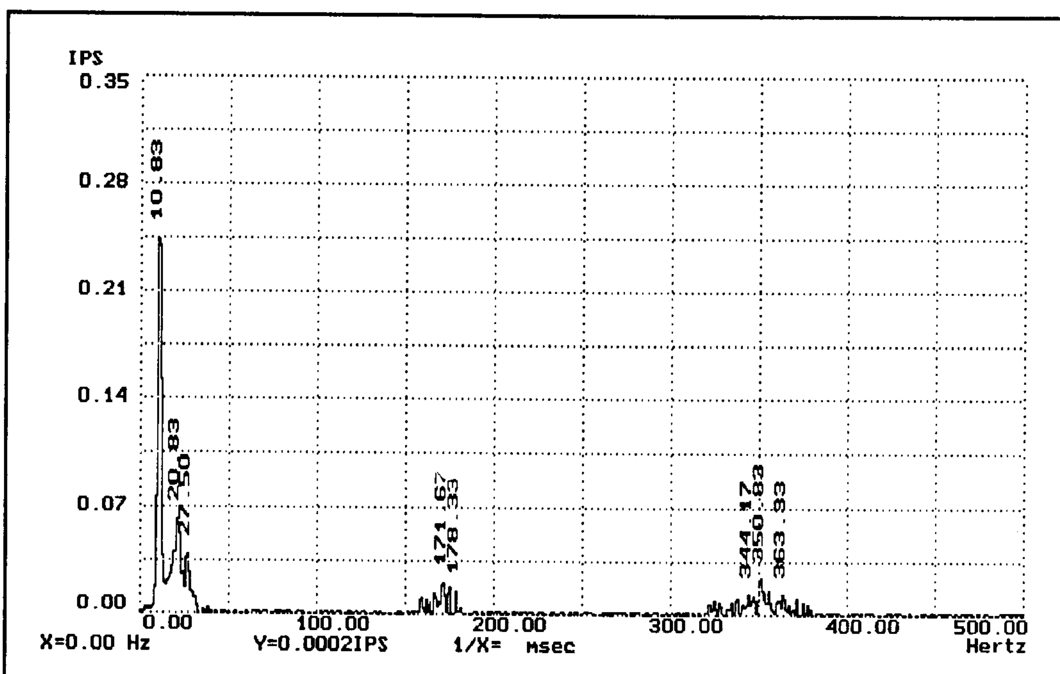

Fig. 6-32. Frequency Spectrum of Dryer Gear.

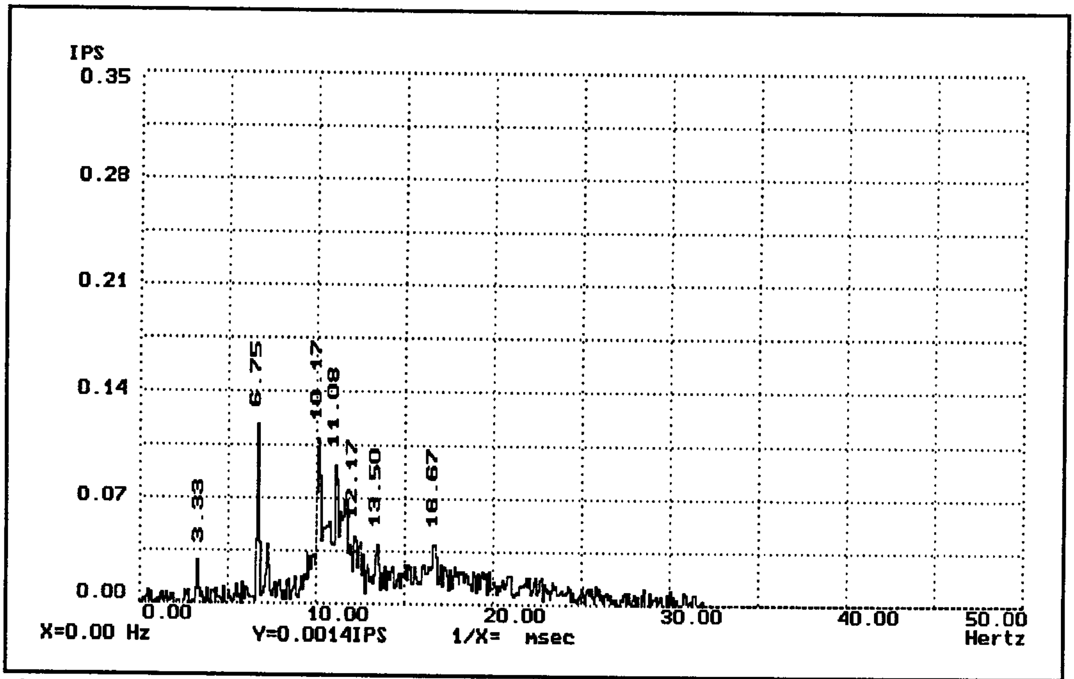

Fig. 6-33. Frequency Spectrum of Idler Gear.

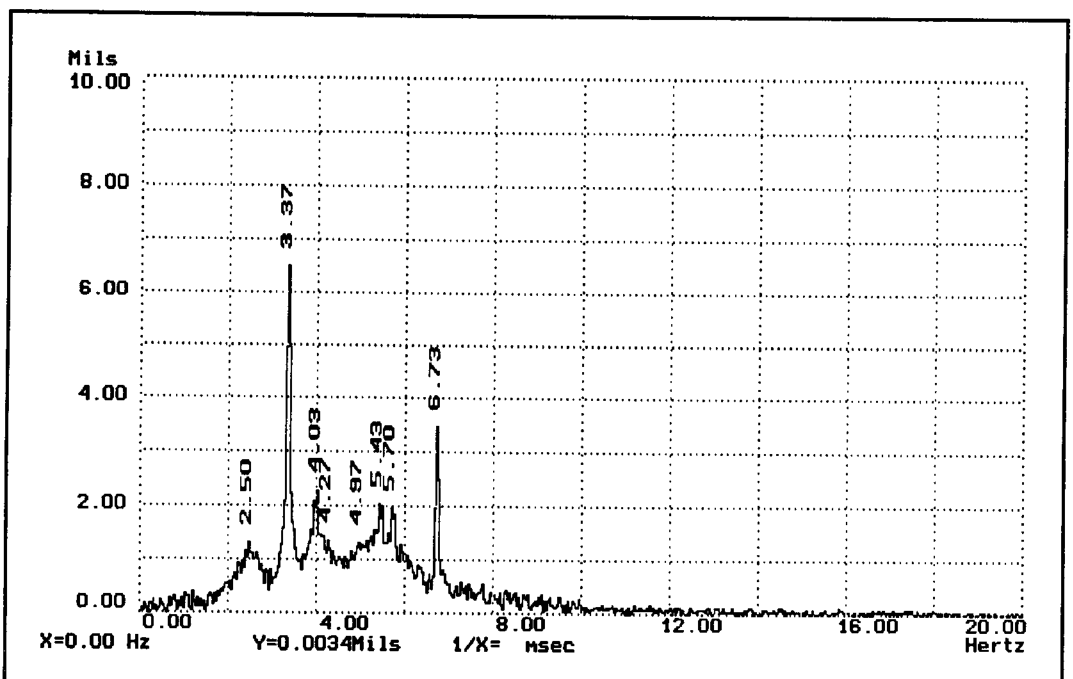

Fig. 6-34. Frequency Spectrum of Dryer Gear.

Therefore, all frequencies below 10 Hz are understated. Please refer to CHAPTER ONE for transducer selection.

Fig. 6-34 is a frequency spectrum of the gear. This data was measured with a displacement transducer that has a frequency response from 0 to 200 Hz. The frequency spectrum is up off the baseline, indicating looseness. However, the first and second harmonics of idler speed are distinctive. This indicates the idler has a bent shaft.

This case proves the need to use two different transducers for accurate analysis. The data in Fig. 6-34 was taken with a contacting displacement transducer on the tending side of the dryer. This data is the relative motion between the dryer bearing housing and the machine frame. The bend in the idler gear shaft was verified by measuring the backlash on the top and bottom of the idler gear, and then rotating the idler gear 180° and making the measurement again. The excessive and inadequate amounts of backlash should change by 180° when the shaft is bent. The bend in this idler gear shaft was caused by a severe paper wrap on the tending side of the dryer.

<u>Loose and worn gears</u> can cause wide-banded white noise up off the baseline. This normally means the gears are moving in an unpredictable manner. Sometimes the noise can have difference frequencies equal to the speed of the loose gear, as in Fig. 6-35. In some installations, when a bearing is loose in the housing, four times running speed can be generated. This frequency can be presented as a discrete frequency, or it can modulate the gearmesh frequency. In such cases, the four times speed will be on the low side of gearmesh frequency, i.e., gearmesh frequency minus four times speed.

Note that if four times speed is added to gearmesh frequency (GF + 4 x speed), the gear is eccentric. If four times speed is subtracted from gearmesh frequency (GF - 4 x speed),

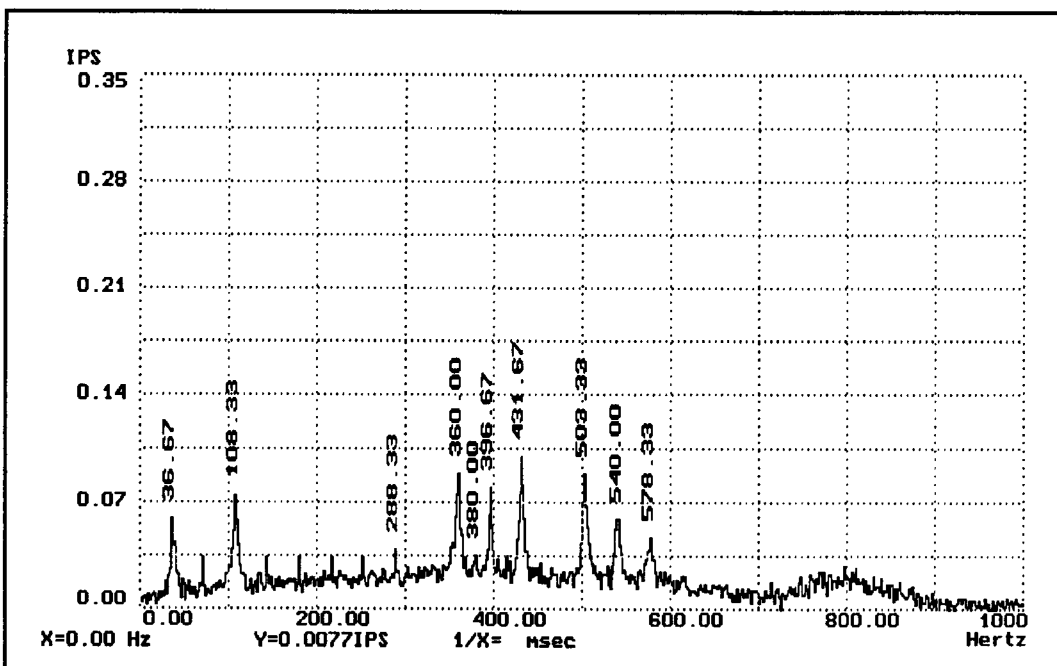

Fig. 6-35. Frequency Spectrum of Severely Worn Gear.

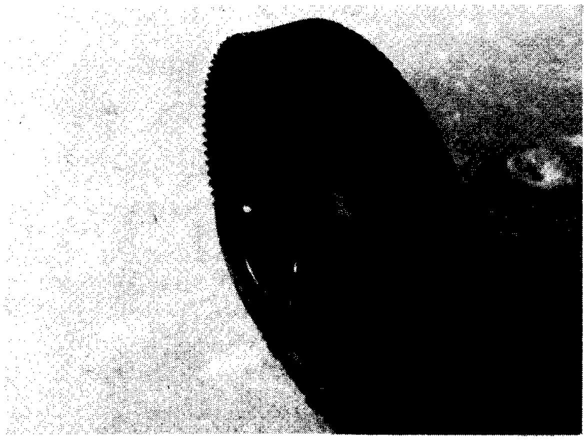

Fig. 6-36. Severely Worn Gear.

the gear is loose. Normally the bearing would be loose in the housing. When the sidebands are on both sides of gearmesh frequency, eccentricity and looseness are present. The underlying theory is that if the gear is eccentric, there is a constant phase relationship between the gearmesh frequency and four times speed. Consequently, the frequencies will add. If the gear is loose, the phase relationship is changing and the frequencies will subtract.

Fig. 6-36 shows a severely worn gear. Severe wear of this nature normally occurs when the gear is too soft. Fig. 6-35 contains the frequency spectrum from the worn gear. The gear speed is 36 Hz and the frequency spectrum is up off the baseline, which normally indicates looseness.

Misaligned gears also generate unique spectra. As in other forms of misalignment, if the first three harmonics of gear-mesh frequency are distinctive, the gears are misaligned.

The distinctive first three harmonics can take several forms. Normally, the fundamental gearmesh frequency is higher in amplitude, and the second and third harmonics are lower.

These downward sloping harmonics can approach a 6 or 12 dB per octave slope. Sometimes the second harmonic is higher or lower than the first or third harmonic. When the second harmonic is higher in amplitude, caution should be used because there could be a backlash-type problem, such as too much backlash, or oscillating gears.

Misaligned gears indicate the gears are not meshing evenly across the pitch line. Fig. 6-37 shows two misaligned gears. Note the wear pattern on only half of the tooth width. Fig. 6-38 is the frequency spectrum from the misaligned gears.

Gears can become misaligned for several reasons, a few of which are listed as follows:

1. Repaired gearboxes are not line-bored correctly.

2. The gear is loose on the shaft or bearings are loose in the housing.

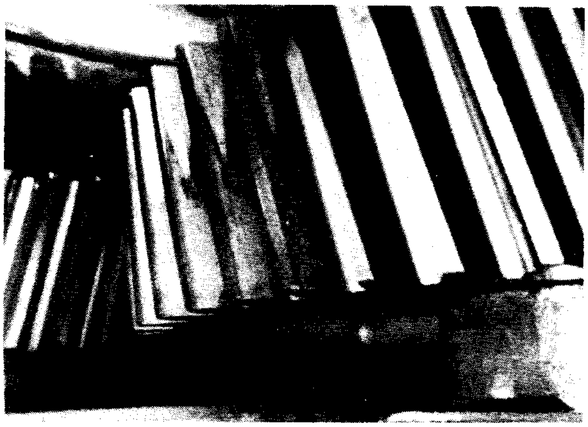

Fig. 6-37. Wear Pattern on Misaligned Gears.

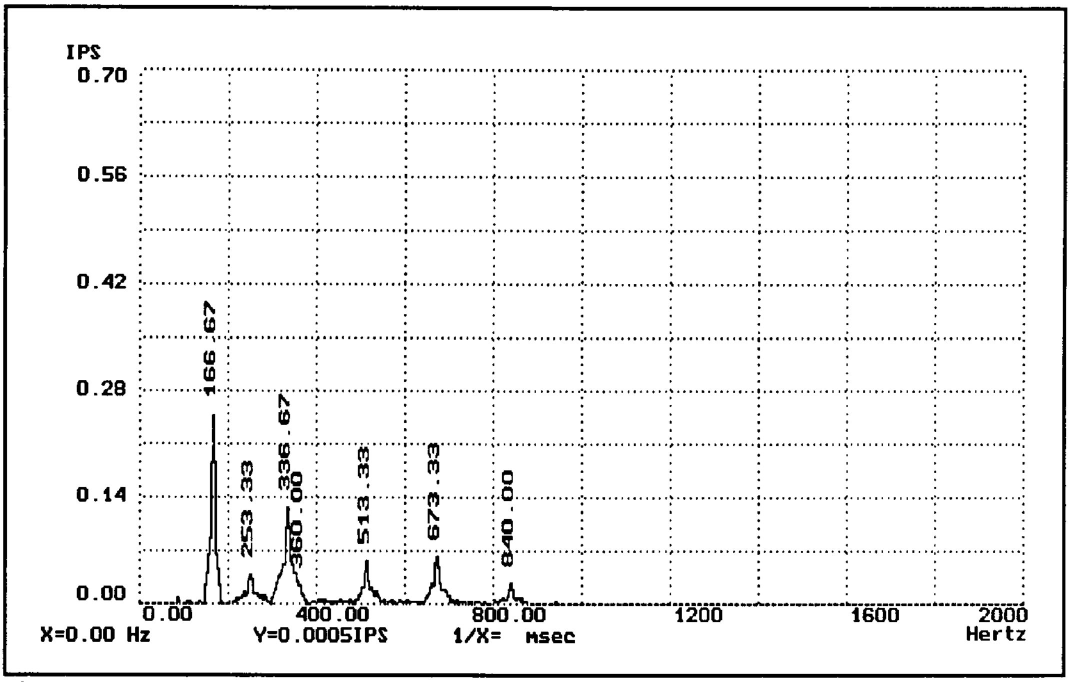

Fig. 6-38. Frequency Spectrum from Misaligned Gears.

3. Gears are not pulled up on a taper correctly.

4. In paper machines, one side of the machine could have settled.

Gears are precision parts and are manufactured to very close tolerances. For example, class eight gears with a diametrical pitch of two in the 20 to 30 inch range can have only 0.0015" to 0.0018" pitch-line runout [3]. In herringbone and other better quality gears, the tolerances are much closer.

Backlash problems or oscillating gears can generate a high second harmonic of gearmesh frequency.

Fig. 6-39 contains an idler gear with 64 teeth and a gear with 128 teeth. Idler gear speed is 5 Hz and gear speed is 2.5 Hz. Gearmesh frequency is 320 Hz and two times gearmesh frequency is 640 Hz. Fractional gearmesh frequencies and HTF have also been calculated.

Fig. 6-40 contains a frequency spectrum of the idler gear. Gearmesh frequency is 320 Hz and two times gearmesh frequency is 641.67 Hz. Both gearmesh and two times gearmesh frequency are modulated by either the dryer or idler speed, but this cannot be seen in this frequency spectrum. It is necessary to zoom in on gearmesh and two times gearmesh frequency to determine the running speeds.

Fig. 6-41 is a 40 Hz window around gearmesh frequency. The difference between points 1 & 2 and points 1 & 3 is approximately 2.5 Hz, which is gear speed. The difference between points 1 & 4 and 1 & 5 is approximately 5 Hz, which is idler gear speed. The higher amplitude of idler speed on the high side of gearmesh frequency indicates the idler is eccentric. The lower amplitude of idler gear speed on the low side of gearmesh frequency indicates looseness. Since the amplitude is higher on the high side than on the low side of gearmesh frequency, the idler gears are more eccentric than loose.

Fig. 6-42 is a 20 Hz window around two times gearmesh frequency. The difference between points 1 & 2 and points 1 & 3 is approximately 2.5 Hz, which is gear speed. The difference between points 1 & 4 and points 1 & 5 is approximately 5 Hz, which is idler gear speed. The higher amplitude of gear speed on the low side of two times gearmesh frequency indicates looseness. The gear is more loose than eccentric. The high amplitude of two times gearmesh frequency is a clear indication of backlash/oscillating gears. Further analysis in the time domain is required.

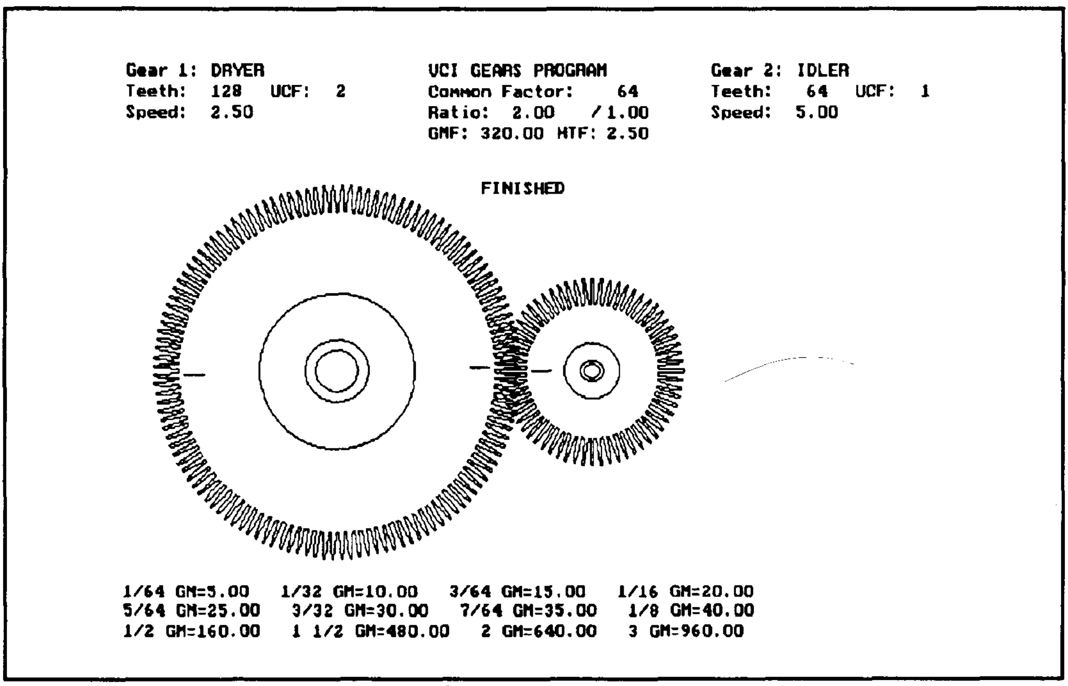

Fig. 6-39. Gears Program Output of an Idler Gear and a Dryer Gear.

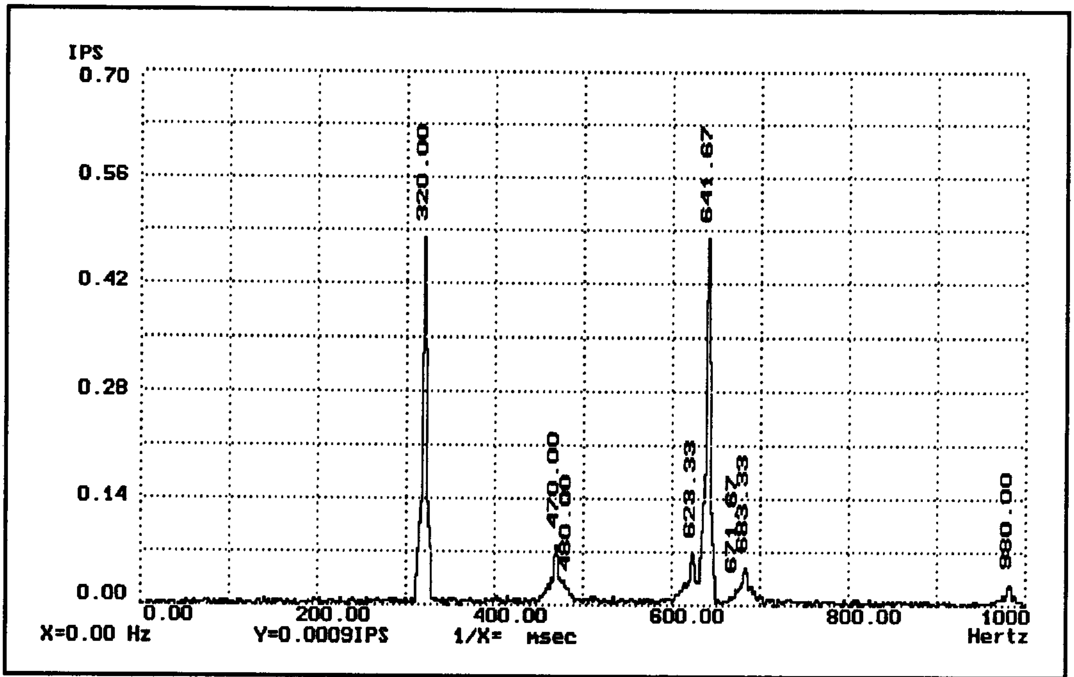

Fig. 6-40. Frequency Spectrum of the Idler Gear.

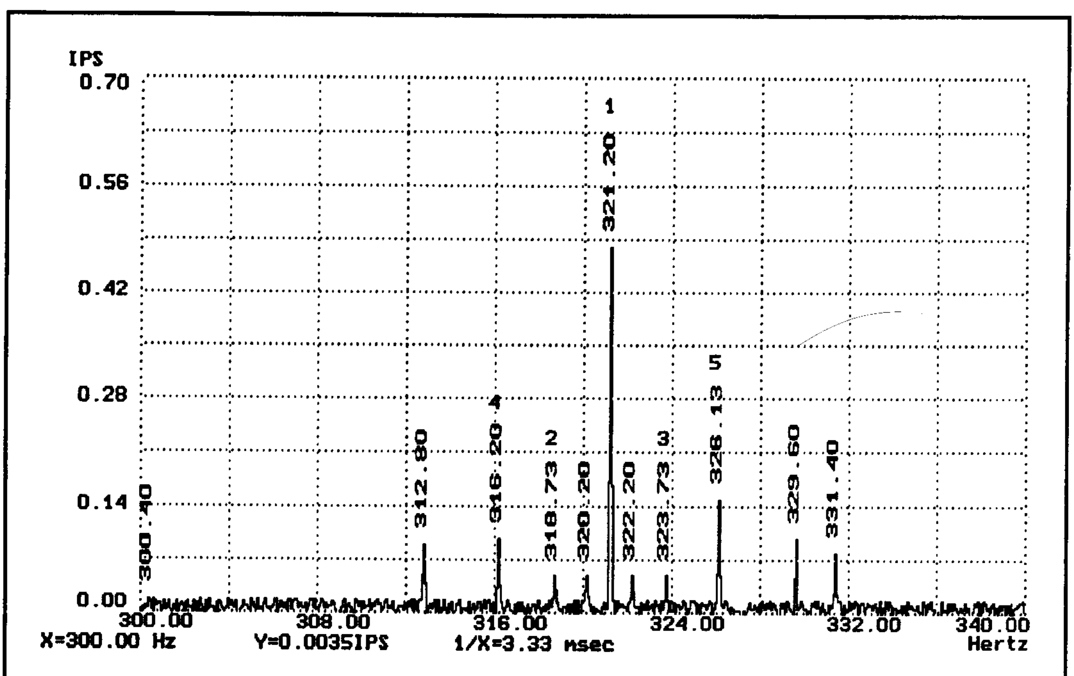

Fig. 6-41. 40 Hz Window around Gearmesh Frequency.

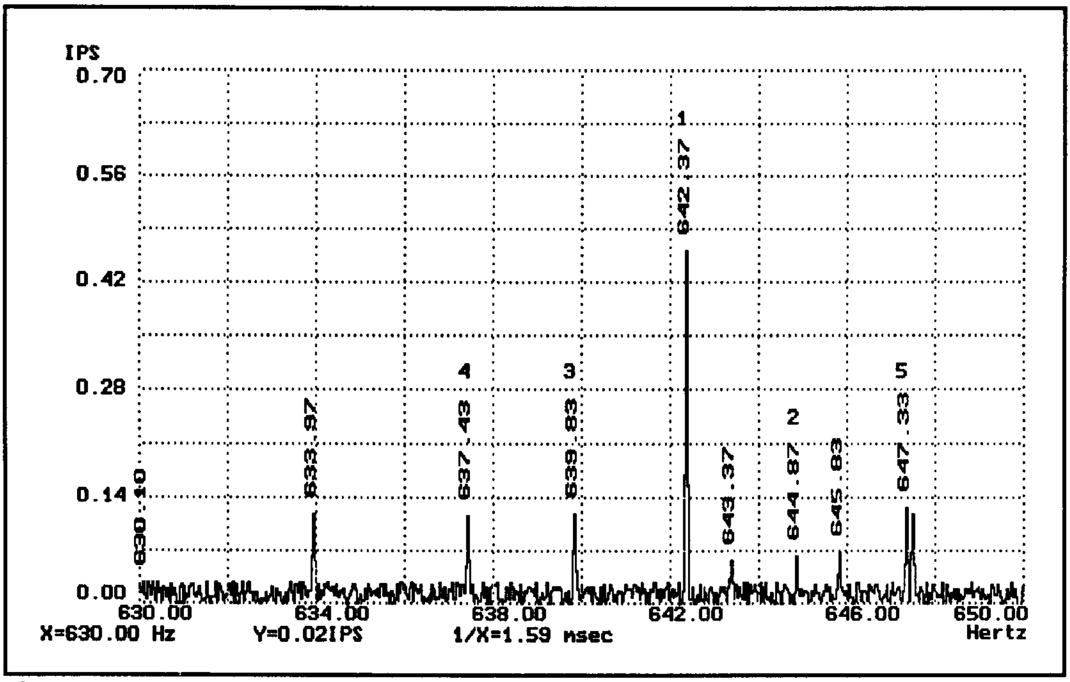

Fig. 6-42. 20 Hz Window around Two Times Gearmesh Frequency.

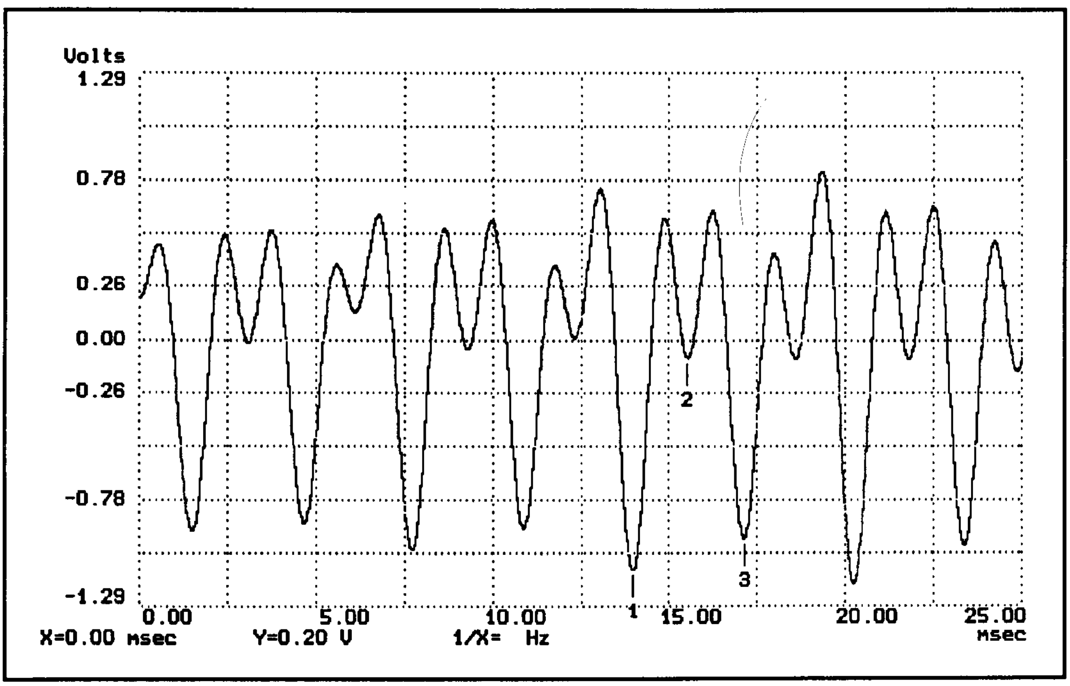

Fig. 6-43. Time Domain Signal of One and Two Times Gearmesh.

Fig. 6-43 is the time domain signal of one and two times gearmesh frequency. The difference between points 1 & 3 is gearmesh frequency of 321.25 Hz. The difference between points 1 and 2 is two times gearmesh frequency, or the second harmonic, which is 642.5 Hz.

Please note the second harmonic is present at the top of the signal. This means the second harmonic is **180°** out of phase with the fundamental. This phase difference indicates a backlash-type problem. If the second harmonic was caused by misalignment, both signals could be in phase and the second harmonic would be at the bottom of the signal. In this case, the out-of-phase condition means the gears are moving back and forth. Inspections for loose gears or looseness associated with the bearings could be uneventful.

The cause of backlash problems and oscillating gears is complex. Following are a few examples of possible sources:

1. Lightly loaded gears tend to oscillate.
2. Inconsistent loading in agitators, digesters, etc. cause oscillation.
3. Draw problems in paper machines can cause both of the above examples.
4. The power source for some electric drives may not be operating properly.
5. Excessive backlash may be a possible source.

This list is not all-inclusive. However, it should provide the general idea.

Broken, cracked, or chipped teeth can generate a well-damped pulse. In Fig. 6-1, each gear has one tooth in mesh, one tooth going out of mesh, and one tooth going into mesh. When the broken tooth goes into mesh, a pulse is generated. The next tooth is good and effectively stops the system from vibrating. This explains why the pulse generated by a broken tooth is well damped and often only $\frac{1}{2}$ cycle wide.

A gear with small spalls below the pitch line may not generate a unique signal. For the purposes of this book, such spalls or irregularities are not considered defects. A spall or chipped tooth at or above the pitch line, a cracked tooth, or a tooth completely broken out, can generate a unique signal in the form of a pulse.

The pulse generated by a defective tooth has four measurable characteristics:

1. Pulse frequency
2. Pulse width
3. Repetition rate
4. Amplitude

The frequency contained in the pulse is often an excited frequency. The source of excitation is the broken tooth. More than one frequency can be present. These pulses are in the form of amplitude and frequency modulation. The pulse width is largely determined by damping and system response.

The pulse repetition rate is the speed of the gear if one tooth is broken. If more than one tooth is defective, a pulse is generated each time a defective tooth goes into mesh. If two defective teeth are 180° apart, the frequency spectra can have difference frequencies of two times shaft speed. If the defective teeth are spaced 90° apart, the difference frequencies can occur at four times speed, etc. If a gear has more than one defective tooth and the defective teeth are not spaced at 45°, 90°, or 180°, the difference frequencies between the spectral lines are gear speed. However, every second, third, fourth, etc., spectral line will be higher in amplitude, depending on the number of defective teeth on the gear.

The amplitude of the pulse is affected by transfer function, resonance, damping, loading, and defect severity. In some cases, the pulse may not be generated each revolution. This can occur when a tooth floats through the mesh unloaded.

On some slow speed machinery the defect can be heard as a dull knock or "clunk," which may occur each revolution of the defective gear. The time signal in Fig. 6-44 is a typical pulse generated by a defective tooth. The Fourier analysis of the pulse is also included. This analysis of the pulse indicates more than one frequency is present and the frequency response is wide-banded. The higher amplitudes of the frequency spectra are about 23 and 57 Hz. The time signal in Fig. 6-45 contains seven pulses. The Fourier analysis presents a series of spectral lines at the basic frequency of the pulse. The difference frequency between the spectral lines is equal to the reciprocal of the pulse repetition rate. The increases in amplitude around 23, 33, and 57 Hz are caused by the higher amplitudes

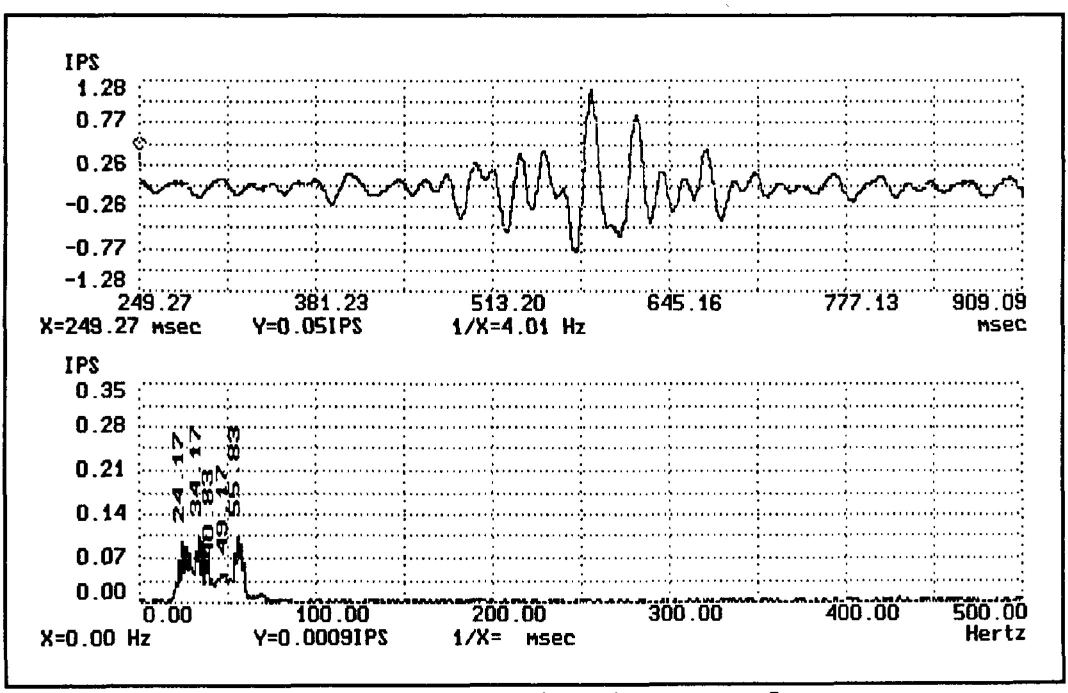

Fig. 6-44. Typical Pulse in the Time Domain and Frequency Spectrum.

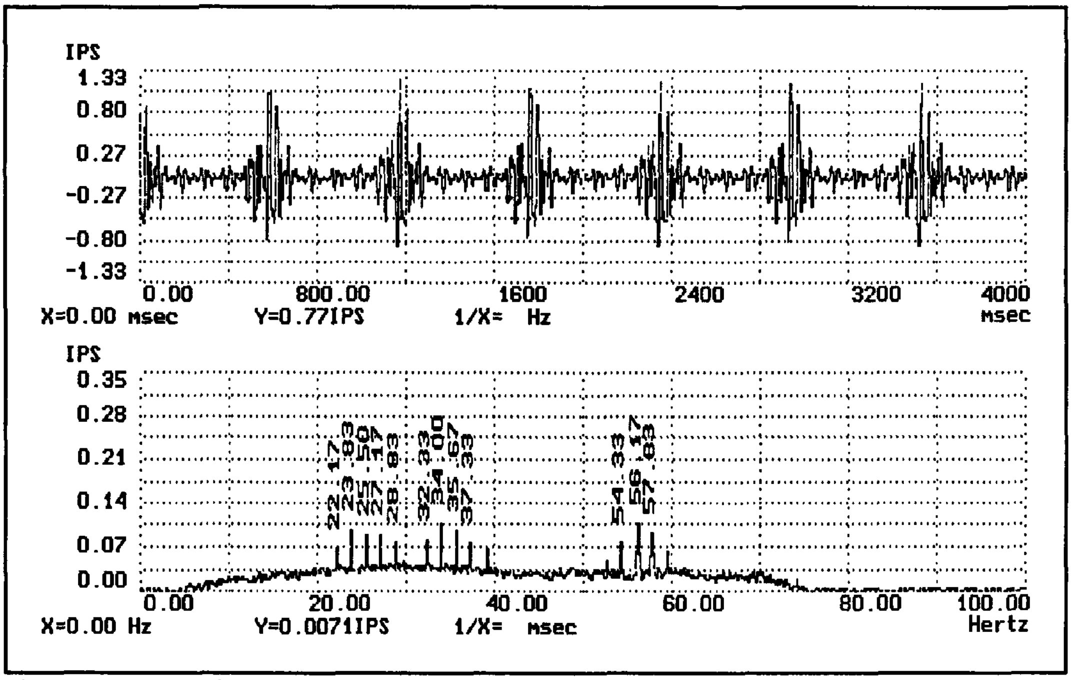

Fig. 6-45. Several Pulses and Frequency Spectrum.

of the basic frequencies in the pulse. The sources of the basic frequencies in the pulse are the excited natural frequencies. Current tests and calculations eliminate the gear and tooth natural frequencies. The present consensus is that the axial natural frequency of the gear and shaft is the most likely source of the basic frequency. The pulse width is very short in duration and often is only $\frac{1}{2}$ cycle.

Empirical data and gear physics indicate the signal is dampened very quickly because of the gearmesh and rotation. When the broken tooth hits the meshing tooth, a pulse is generated. Then the next tooth, which is good, goes into mesh and stops the vibration or dampens any ringing effect resulting from the pulse.

A gear with one defective tooth can generate a pulse once each revolution. If two or more teeth are broken, two or more pulses occur each revolution. These pulses can be observed in the time signal. When several pulses are generated by several broken teeth, the pulses may seem nonrepetitive at first glance. This is true because the number of good teeth between the broken teeth may vary. When this occurs, the pulses appear to be random.

If the frequency spectra from the pulses contain random noise, the pulses are random. Random pulses indicate looseness. However, if the frequency spectra contain spectral lines at discrete sum and difference frequencies, the pulses are not random, and the gear may have more than one broken tooth. In such cases, the difference frequency will be the speed of the gear or gears. The sum frequency can be harmonics of gear speed, or an excited or a generated frequency plus gear speed.

When two teeth are broken, every other spectral line can be higher in amplitude,

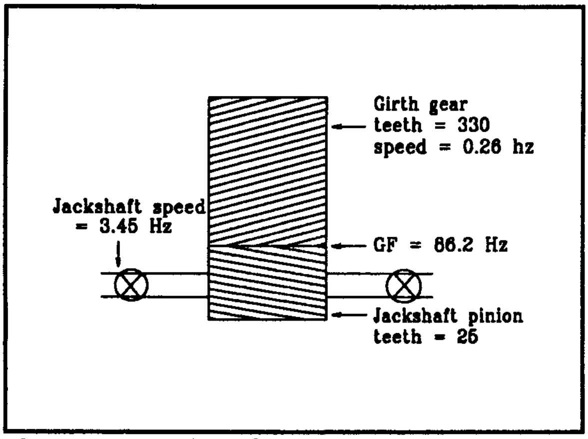

Fig. 6-46. Jackshaft Pinion and Girth Gear Arrangement.

indicating two events per revolution. Every third, fourth, etc., spectral line could be higher if the gear had three, four, etc., broken teeth. Observation of a once-per-revolution marker and the pulse on a dual channel oscilloscope or real-time analyzer can reveal the location of the defective tooth with respect to the marker. When long time periods are necessary, the **Time Plot** software is required for accurate analysis.

Pulse amplitude is related to defect size, loading, system damping, and frequency addition and subtraction. Amplitude is used to describe good and bad conditions and to identify multiple pulses when more than one pulse exists. For example, if the pulse is in phase with a cycle of another frequency, the amplitude of the two signals will add and the pulse amplitude will be overstated. If the pulse is out of phase with a cycle of another signal, the amplitude of the two signals will subtract, and the pulse amplitude will be understated. The net result is that the pulse amplitude may vary in some repeatable pattern, depending on how long it takes for the two signals to go into and out of phase.

Fig. 6-46 is a sketch of a jackshaft and girth gear arrangement. Jackshaft speed is 3.45 Hz, and gearmesh frequency is 86.2 Hz. The girth gear is used to grind phosphate rock. The gears carry a heavy load and the gearmesh frequency is always present. When a tooth broke out of the pinion gear, a sharp smack was heard each revolution of the pinion. However, the vibration amplitude did not increase.

Fig. 6-47 contains the frequency spectrum taken from the jackshaft bearing. The frequencies at 13.8 and 27.6 Hz are generated in the gearbox that drives the jackshaft. The spectral line at 86.2 Hz is gearmesh frequency. The spectral line at 100 Hz is the sum of gearmesh frequency plus 13.8 Hz. The series of spectral lines around 45 and 115 Hz have difference frequencies of 3.45 Hz, which is jackshaft speed. The amplitudes of these spectral lines are displayed at a magnified scale for visibility. Under normal calibration standards, these spectral lines appear as small bumps on the baseline. These spectral lines are the result of an FFT of the pulse generated by the broken tooth on the pinion.

Fig. 6-48 contains the time signal from the jackshaft bearing. The negative-going pulse occurs once each revolution of the pinion and is only $\frac{1}{2}$ cycle wide.

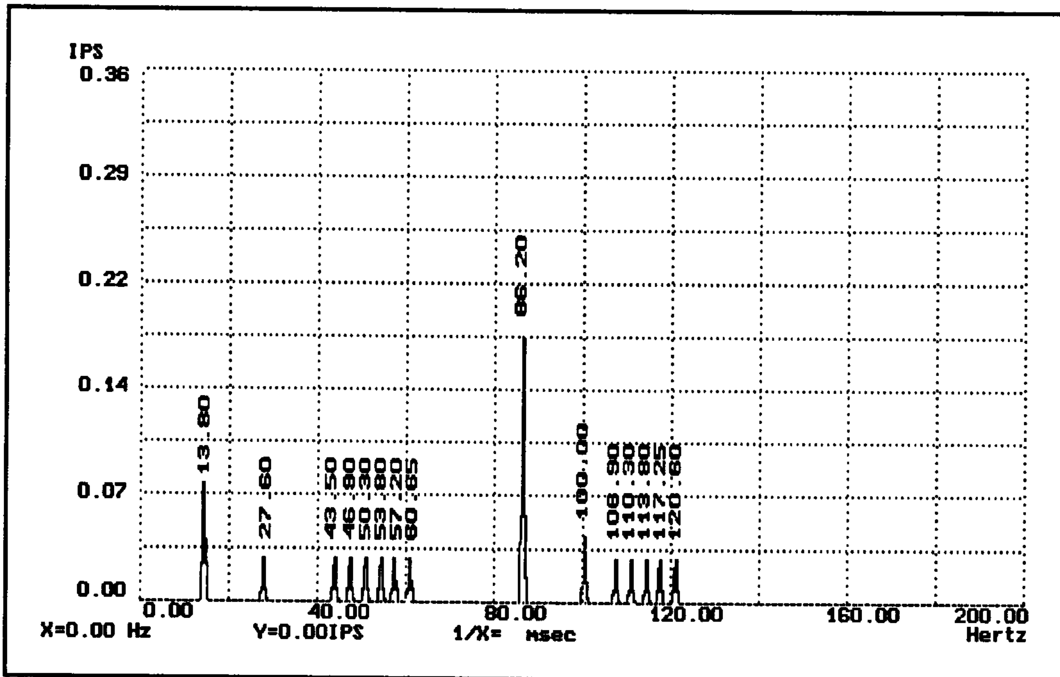

Fig. 6-47. Frequency Spectrum from the Jackshaft Bearing.

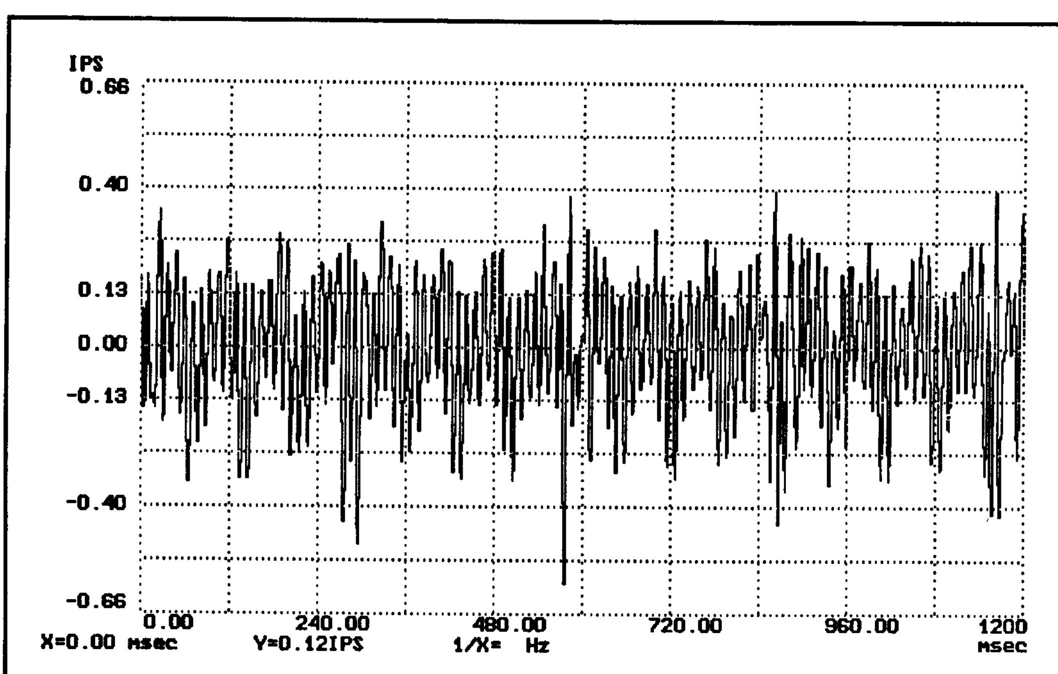

Fig. 6-48. Time Signal from the Jackshaft Bearing.

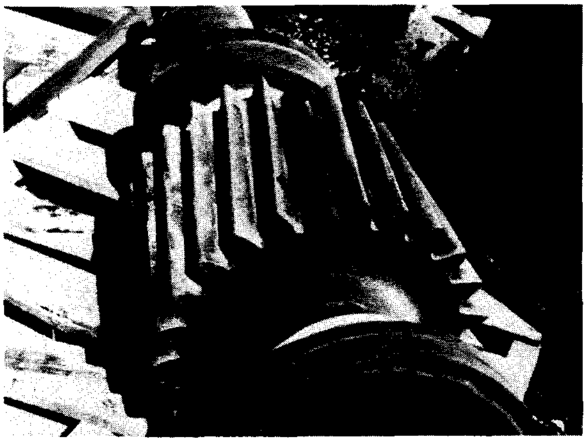

Fig. 6-49. Pinion with the Broken Tooth.

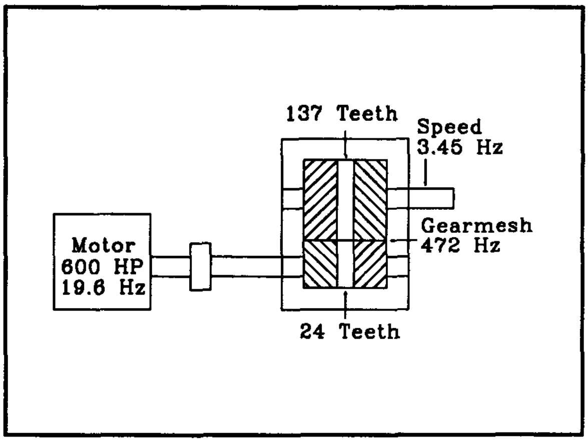

Fig. 6-50. Single Reduction Gearbox.

Fig. 6-49 is a photograph of the pinion with the broken tooth.

Since the frequency spectrum from the pulse has a very low amplitude, it could be, and often is, overlooked. Also, other time signals, such as those containing frequency modulation and some other forms of amplitude modulation, can generate similar spectra. For these reasons, a pulse must be observed in the time signal before diagnosing a broken tooth.

Analysis of spectra generated by defective gears can be difficult because quite often more than 100 spectral lines are generated. Each spectral line can be identified with the sum and difference technology to reveal all the problems in the gearbox. An actual case shall be presented to illustrate and explain this technique.

Fig. 6-50 contains a sketch of a single reduction gearbox driven by a 600 HP electric motor. Shaft speeds, number of teeth on each gear, and gearmesh frequency are included in the sketch. The pinion has one tooth badly chipped and generates a 19.6 Hz pulse. The driven gear has one tooth with the end broken off and one tooth slightly chipped. This gear is generating one pulse at 3.4 Hz. A gearmesh problem exists, caused by four high places on the gear. These high places cause amplitude modulation of the gearmesh frequency. This modulation causes sidebands on each side of the gearmesh frequency.

Fig. 6-51 contains the output from the **Gears Program**. The number of teeth on these gears has a common factor of 1. This indicates a proper ratio. The **Gears Program** indicates $\frac{1}{2}$ and $1\frac{1}{2}$ gearmesh frequency could be present. These frequencies can occur in this case only if some teeth are eccentric. Since these frequencies are not present in the data, this form of eccentricity is not present.

Fig. 6-52 contains the frequency spectra taken from the axial position on the low speed shaft. The bottom spectrum (zero to 500 Hz) indicates all frequencies present. In the second spectrum (zero to 200 Hz), difference frequencies of 19.6 Hz are present. The difference frequencies of the spectral lines between 98 and 117.6 Hz are at 3.46 Hz. The third and fourth spectra are 100 Hz frequency windows, and contain difference frequencies of both shafts. The fifth spectrum is a 100 Hz window containing gearmesh frequency. The spectral line at 470.4 Hz is gearmesh frequency. The spectral line at 484.2 Hz is gearmesh frequency plus four times gear speed. Gearmesh frequency has sidebands of 13.8 Hz on both sides. The higher amplitude on the high side means the gear is eccentric and the four times gear speed means the gear has four high places. The difference frequencies of 3.44 and 19.6 Hz indicate a broken tooth on each gear.

The data in Fig. 6-53 was taken from the axial position on the high speed shaft. The spectrum identifies the 19.6 and 3.4 Hz pulses. These pulses are quite obvious in the time domain signal and verify broken and/or chipped teeth.

Fig. 6-54 shows the pinion. The broken tooth generated the pulse in the above data. Fig. 6-55 shows the gear with two defective teeth. The broken tooth generated the 3.4 Hz pulse. The spalled teeth in Fig. 6-56 are located around one of the high places on the

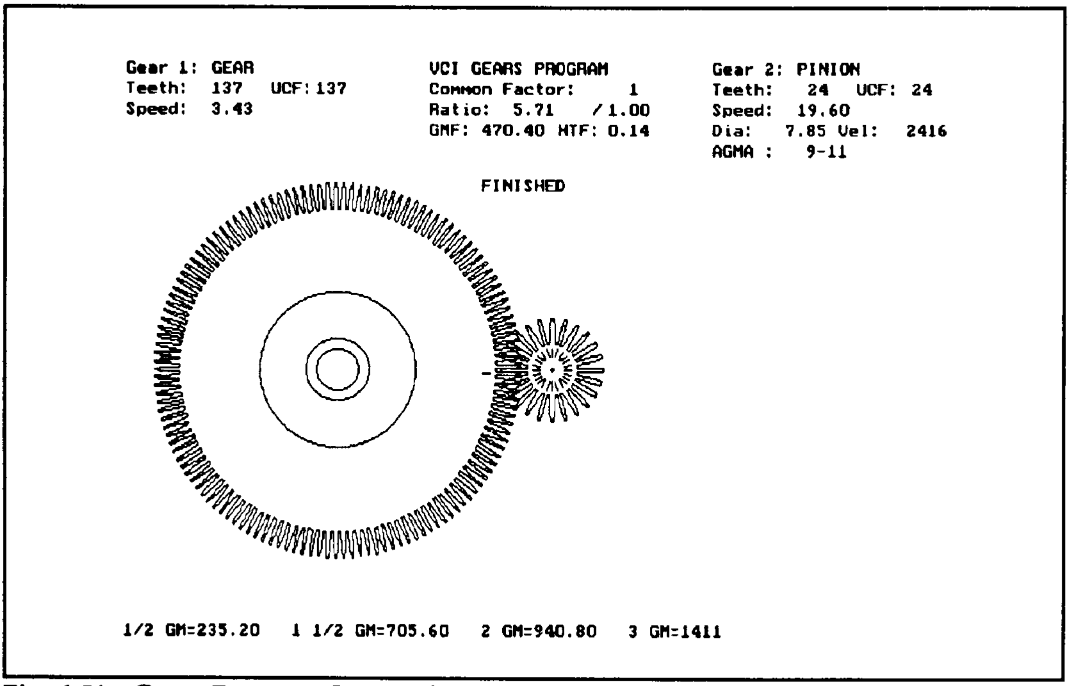

Fig. 6-51. Gears Program Output for Gearbox.

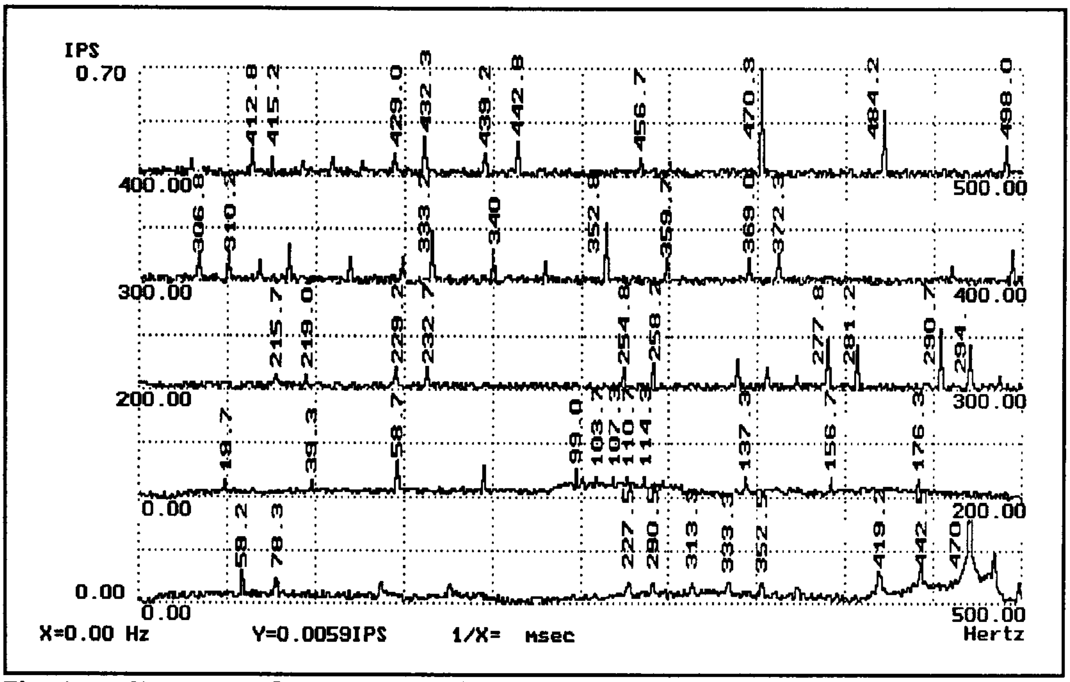

Fig. 6-52. Frequency Spectra Taken from Low Speed Shaft.

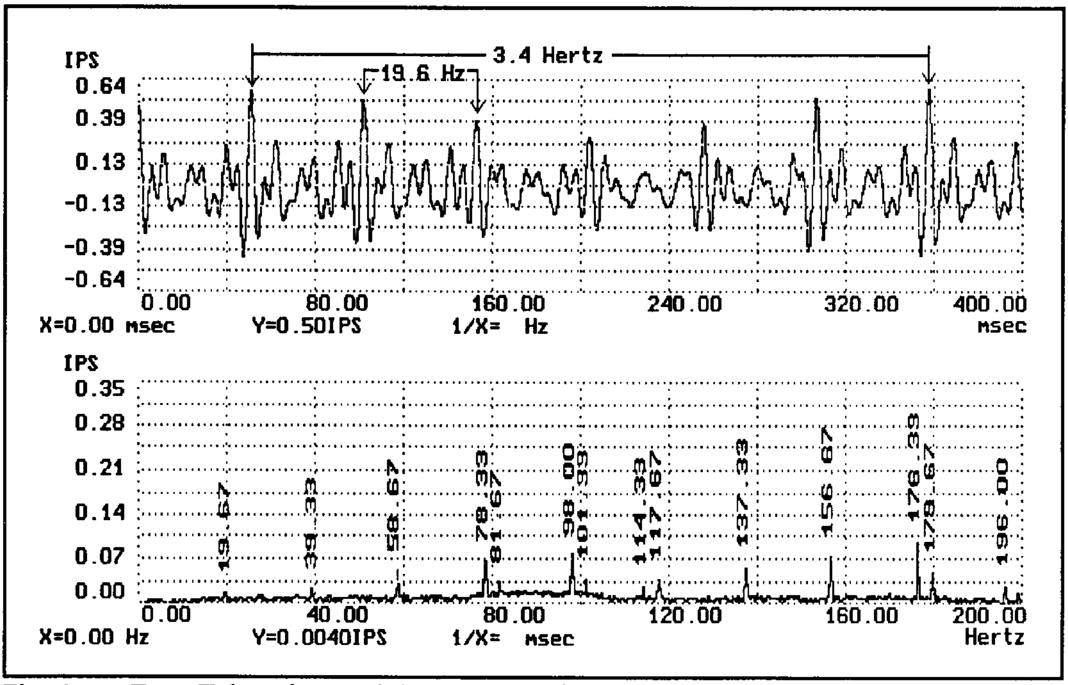

Fig. 6-53. Data Taken from High Speed Shaft.

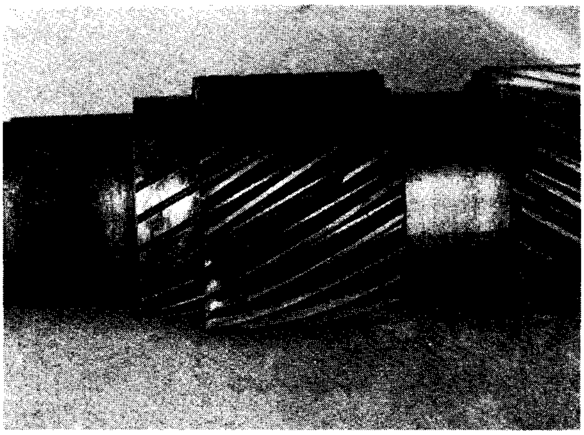

Fig. 6-54. Broken Tooth on Pinion.

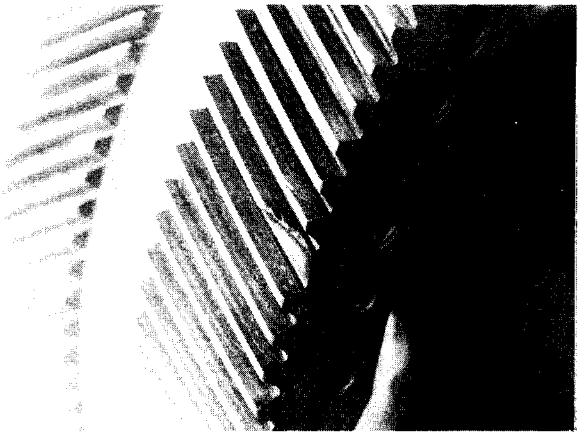

Fig. 6-55. Broken Teeth on Gear.

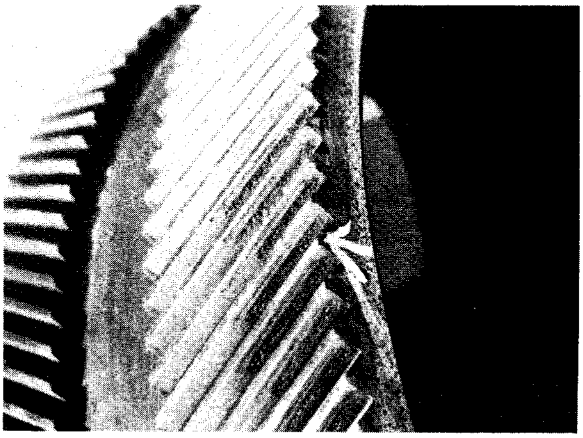

Fig. 6-56. One High Place on Gear.

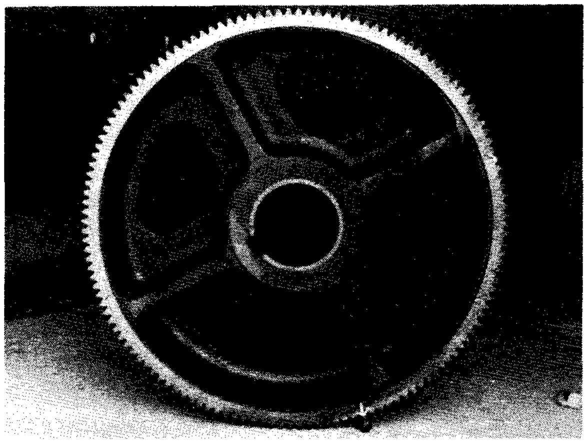

Fig. 6-57. Location of the Four High Places.

gear. Each high place is identified by an arrow in Fig. 6-57. These four high places caused the modulation of the gearmesh frequency, i.e., the sidebands above and below the gearmesh frequency are 13.8 Hz apart.

With most real-time analyzers, the length of time observed on the screen is determined by the frequency range. Often, when the frequency range is reduced to obtain a longer range in the time signal, higher frequencies are filtered out of the signal. Likewise, when the frequency range is increased, the displayed time period is too short to observe repeatable patterns. This problem may be helped somewhat by increasing/ decreasing the lines of resolution. However, there is a limited number of available pixels on a given screen, and only a limited number of lines can be drawn on a piece of standard paper.

In the real world of analysis, the need often arises for the display of very long time periods. Often, time periods in excess of hunting tooth time must be observed to accurately identify two broken teeth. Such long time periods are also required to determine if eccentric teeth are on one gear or if the teeth are eccentric only when meshing with specific teeth on the other gear.

Currently, one of the few RTA's on the market capable of storing and retrieving long time periods of many seconds, with enough resolution to be usable, is the Tektronix TEK 2630 used in conjunction with the **Time Plot** software. It is impossible to present an actual **Time Plot** in this chapter because the printout could cover 20 or more pages, depending on the frequency range and the length of the time period.

A sample of **Time Plot** is contained in Fig. 6-58. The output is printed on a dot matrix printer and contains all information necessary for accurate analysis of the time domain signal. The top of the printout displays the name of the data file stored on disk (IG55.DAT), along with the date and time that the data was acquired (8/8/1990, 12:27). The plot time interval listed indicates the scale of the time data. This scale may be specified to display any desired plot resolution. Tick marks along each edge of the printout represent an inch-ruler scale with a mark every $\frac{1}{10}$ inch. Therefore, each inch of the printout corresponds to a specific period of time data (10.0 ms in this case). Listed next is the number of data points plotted. This number is determined by the plot time interval. It is used to indicate when the data is not printed out at the maximum

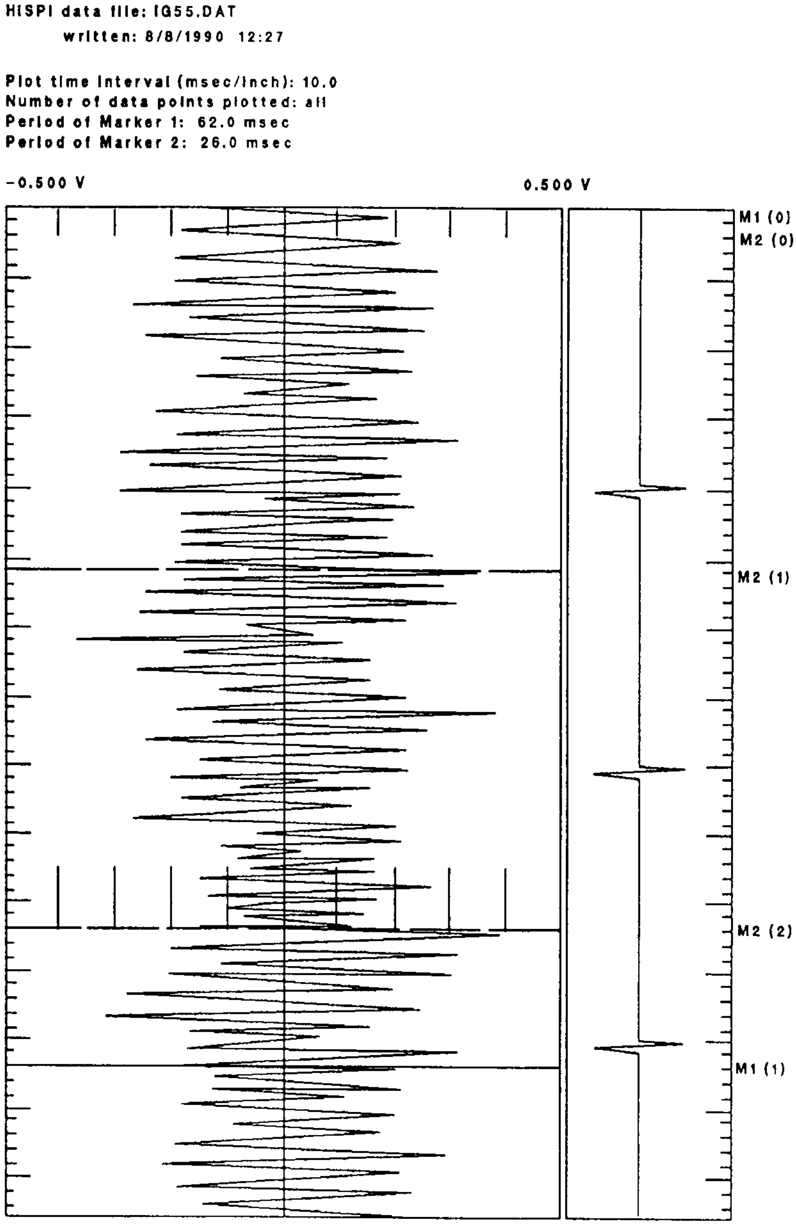

Fig. 6-58. Time Plot Sample.

resolution contained in the file.

Two markers may be used to mark specific time intervals on the printout. The periods of these two markers are listed as "Period of Marker 1" and "Period of Marker 2" (62.0, 26.0 ms). The markers are displayed at the specified time periods as horizontal lines across the data. Marker 1 is represented by a solid line labeled as M1(0), M1(1), M1(2), etc., at the right side of the data. Marker 2 is represented by a dashed line labeled as M2(0), M2(1), M2(2), etc.

Optionally displayed with the time data is a once-per-revolution trigger signal, as shown along the right side of the printout in Fig. 6-58. The displayed trigger pulses are actually a third marker, which can be used to measure repeatable events in the time data. The period of these trigger pulses can be read from the inch-scale marks at the edges of the printout.

The amplitude scale for the time signal is listed at the top edge of the data. This scale is displayed by tick marks along the width of the printout, indicating ten divisions from the minimum voltage scale -0.5 V to the maximum voltage scale 0.5 V. The vertical line along the mid-scale amplitude indicates the zero point.

CONCLUSION

The technology contained in this chapter enables skilled analysts to accurately diagnose problems in gears. The necessary hardware and software are also required. Gear problems can be identified up to six months or more before failure occurs. This long lead time is quite adequate to obtain parts and to plan and schedule the repair.

The technology in this book and in Drago [3] provides useful information for engineers to improve the specifications when installing geared systems. The specifications should include, but not be limited to, proper tooth ratio and AGMA gear quality number. When new geared systems are installed, this technology is quite useful in evaluation of the operating gears.

This technology is also useful for gear manufacturers to evaluate products and quality assurance effectiveness.

CHAPTER SEVEN:
ANALYZING AND SOLVING PRESS ROLL AND NIP PROBLEMS

INTRODUCTION

Rolls in nip are encountered in: rolling mills in the steel and aluminum industries; presses, calender, breaker, and pull stacks in the pulp, paper, tobacco, and printing industries; forming rolls and pattern rolls in the textile industry; plastic industry; and all other rolls that are rolling in nip with another roll or a relatively hard surface.

This chapter describes how to take, process, and analyze data to diagnose problems with rolls in nip. Once the problem has been accurately identified, corrective action can be taken.

Press rolls and other rolls with an overall vibration level of only 1.5 mils zero-to-peak can start a chain of events that can damage and/or destroy other rolls and felts in a matter of days. While overall vibration levels are necessary, other measurements must be made to identify and correct these insidious problems. These measurements are relative motion between the rolls and dynamic measurement of roll shape.

The overall vibration level is normally measured with a velocity transducer or an accelerometer. This overall data is used to identify problems in bearings and gears, to identify resonant frequencies, and can sometimes be used to identify roll cover and felt problems in simple situations. The amplitude is often overstated or understated depending on the phase relationship of the signals. Generally, they are understated because the signals are often out of phase.

The relative motion between two rolls is normally measured with a contacting or noncontacting displacement transducer. These transducers have a good low frequency response and provide much information about the fundamental speed and harmonics of various components, such as press rolls and felts. These signals are complex, and the raw data is often difficult to analyze. The amplitude can also be overstated or understated depending on transfer function (path taken by the vibration signal), cross coupling (algebraic sum of various signals), resonance, and damping.

The technique of synchronous time averaging is used to measure the contribution from each rotating component. This data is excellent for determining the roundness of each rotating unit. However, the exact roll shape and the amplitude of high and low places cannot be determined exactly, except in dynamic measurement of the roll surface.

Another consideration is the hardness and flexibility of the roll. Is a high place on a roll caused by the roll being out-of-round, or is the high place caused by a hard place in the roll, particularly in covered rolls? Flexibility of the roll shell and imbalance are other considerations.

The dynamic shape or roundness of a roll can be measured before the roll is installed or after installation. If the roll is a steel roll, a noncontacting displacement transducer should be used. These transducers are so sensitive that the depth of the finish can also be measured. If the roll is not made of ferromagnetic material, a contacting displacement transducer must be used. Dynamic measurement accurately measures the runout in the roll. Depending upon how a roll is installed, it may introduce natural frequencies in the system, the source being the mounting in some cases. These undesirable frequencies can be removed with the technique of synchronous time averaging.

VIBRATION THEORY OF ROLLS IN NIP

The techniques described in this chapter can accurately identify the following problems:

1. Imbalanced rolls
2. Out-of-round or eccentric rolls
3. Lopsided rolls
4. Corrugated rolls or rolls with bars
5. The cause of bars and/or corrugations
6. Felt problems
7. Rolls with circumferential cracks

The following causes of bars on rolls have been observed: eccentric rolls and rolls that have hard places running against rolls that have an improper ratio of roll sizes, resonant frequencies, and installation of ground rolls when enough material has not been removed during the grind to remove the old bars or get below the set.

In order to pinpoint these problems, use of proper tools is imperative. These tools include both hardware and software required for precise diagnosis of the problem.

Hardware

The following types of transducers are required: displacement, both contacting and noncontacting; velocity transducers with a magnetic base; and accelerometers. The proper selection of these transducers permits measurements of frequencies from DC to 20 KHz and greater. For proper transducer selection, refer to the section titled "Relationship between velocity, displacement, and acceleration" in Chapter One.

Reliable data cannot always be obtained with an accelerometer if it is hand held, or held with a magnetic base. When an accelerometer is used, it should be either glued or screwed down. If the accelerometer is not glued or screwed down, the amplitude of some frequencies can be overstated or understated, and sidebands may not be present when a zoom feature is used.

An optical pickup may be used to obtain a once-per-revolution trigger. These pickups are adequate when the target reflects light. However, it is near impossible to get a trigger from a felt. A transmit-and-receive fiber optic pickup that is capable of triggering on a change in color is a better selection.

A real-time analyzer capable of synchronous time averaging is required. The RTA must be capable of triggering from a pulse in the time signal, have variable lines of resolution, measure delta time, and expand the time domain signal. A high speed dot matrix graphics printer is also required.

Software

Polar Plot Software is needed to view and measure the roundness of a roll. **Roll Ratio** Software is needed to identify various roll sizes that should be avoided. Since the **Roll Ratio** Software can generate large amounts of data, a **Rusch Chart** must be used to identify the various roll sizes that should not cause barring.

Synchronous Time Averaging

The technique of averaging vibration data has been used since the introduction of the real-time analyzer into the rotating machinery industry in the early 1970's. Averaging tends to level the amplitude of the vibration signal and enhance the signal. Leveling the amplitude of the signal occurs because the amplitude can vary well over 100 percent. Signal enhancement is achieved because all frequencies are not present in every average. This also explains one reason why the amplitude is not stable. Such averaging occurs in a random manner and provides signal enhancement for all frequencies. Synchronous time averaging requires a trigger, and data collection is started at a synchronous point for each average. Since each frame of data starts at the same phase relationship of the synchronous roll, and is then algebraically added to the previous data or previously averaged data, the fundamental frequency and all harmonics of the synchronous roll will be in phase and added together. All frequencies that are not synchronous with the fundamental frequency will be out of phase and will be subtracted out of the data. Based on this information, synchronous vibration can be defined as the fundamental frequency of roll speed and all harmonics, including subharmonics of roll speed.

Nonsynchronous vibration is all frequencies that are not equal to some harmonic or subharmonic of roll speed. Some frequencies are out of phase most of the time and average out very quickly. Other frequencies may be very close to a synchronous frequency and may be in phase most of the time. Such frequencies may not average out very quickly. For this reason many averages must be taken to obtain good data.

Synchronous vibration will be defined as the signal and nonsynchronous vibration will be defined as noise. The signal-to-noise ratio improves by the square root of the number of averages:

$$SNR = \sqrt{n}$$

To obtain a signal-to-noise ratio of 10:1, 100 averages must be taken. One hundred averages are satisfactory for most applications, provided the trigger is stable and has remained stable during the averaging.

The following problems can be encountered:

1. Erratic triggers can occur for many reasons. The best indication of an erratic trigger is when the data can average down to zero.

2. If the trigger is erratic part of the time, some data may be present. However, the signal may contain nonsynchronous data.

3. The operator must monitor the data at all times because, if more than one trigger occurs per revolution, the data may be invalid.

The best way to overcome these problems is to:

1. View the trigger in the time domain and ensure it is stable and only one trigger is generated per revolution. Fig. 7-1 contains a trigger. The trigger can be a full cycle going both positive and negative, or it can be only a half cycle and can be either positive or negative. Positive triggers are peaks in the positive direction (above the zero line), and negative triggers are peaks in the negative direction (below the zero line).

2. Obtain an overall vibration signal with the trigger off and attempt to identify the various frequency components.

3. After turning the trigger on again, start the synchronous time averaging and observe the time signal and the frequency spectrum in separate displays. The time screen should contain enough time for at least one revolution of the roll of interest. Two or three revolutions are desirable in most cases. The data should be checked to determine if the frequencies are correct. If the frequency spectrum contains nonsynchronous frequencies, the synchronous data is not valid.

After good synchronous data has been obtained, it is used with the **Polar Plot Program** to display the roll profile. The **Polar Plot** reveals the amplitude of bars (if present) on a particular roll and its total indicated runout. The roll profile displayed by the **Polar Plot Program** is an average profile down the length of the roll. When synchronous time averaging is performed, the time domain signal contains the profile of the roll as its surface goes through nip. See Fig. 7-2. The length of the signal that is equivalent to the time period of one roll revolution is taken and plotted around a circle where a circle represents a perfect roll, as in Fig. 7-3. From this plot, all relevant information about the

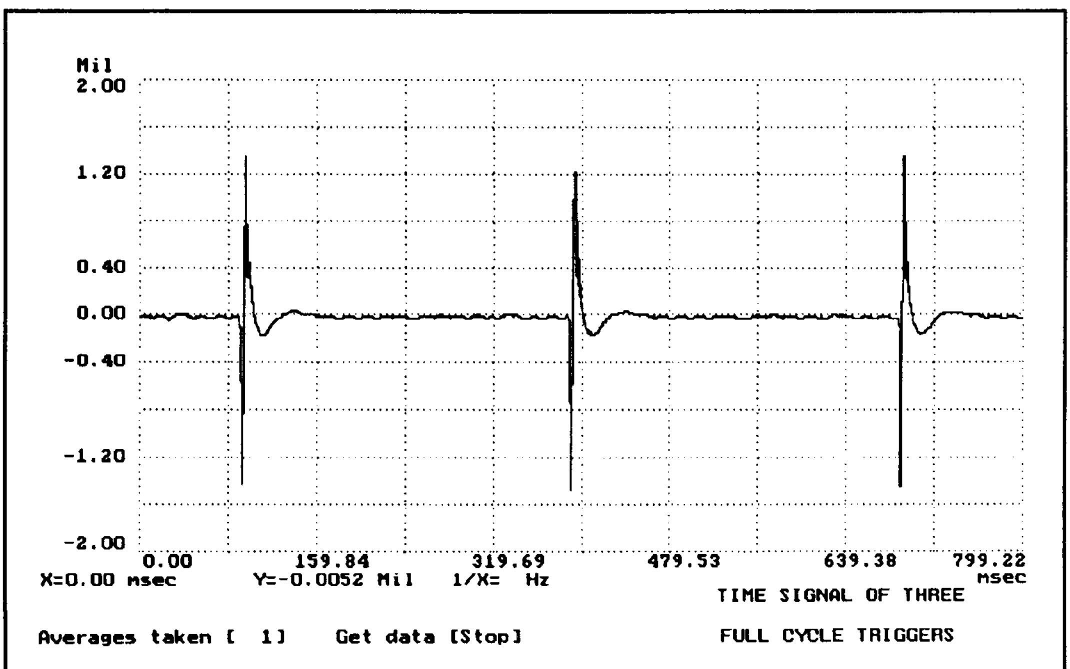

Fig. 7-1. Time Signal of Three Triggers.

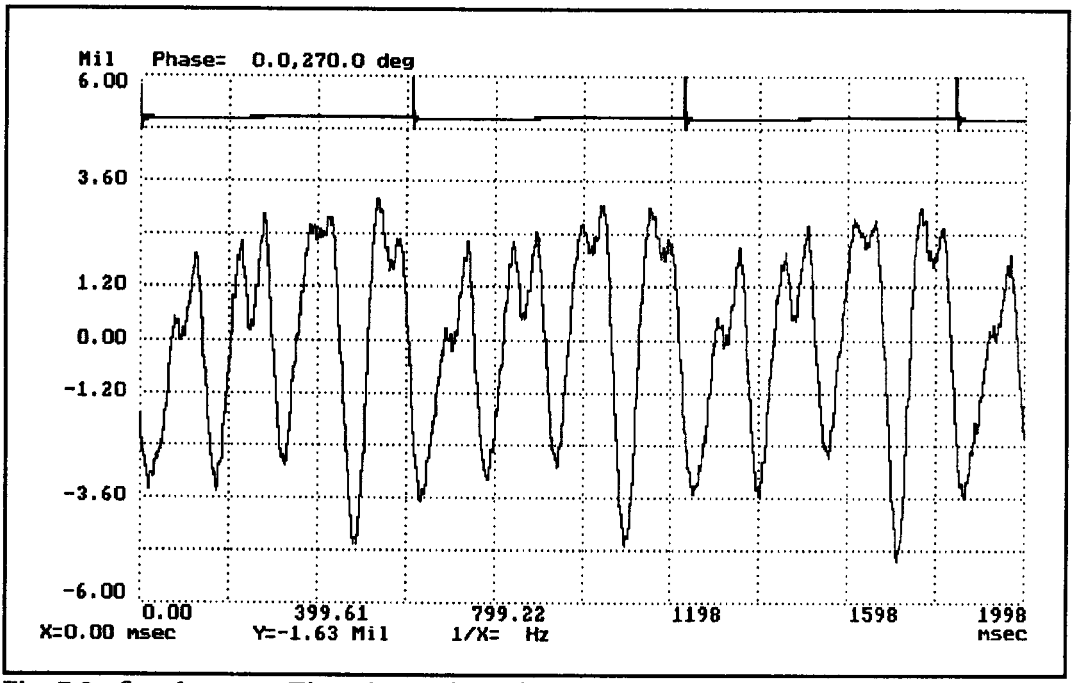

Fig. 7-2. Synchronous Time Averaging of the Roll.

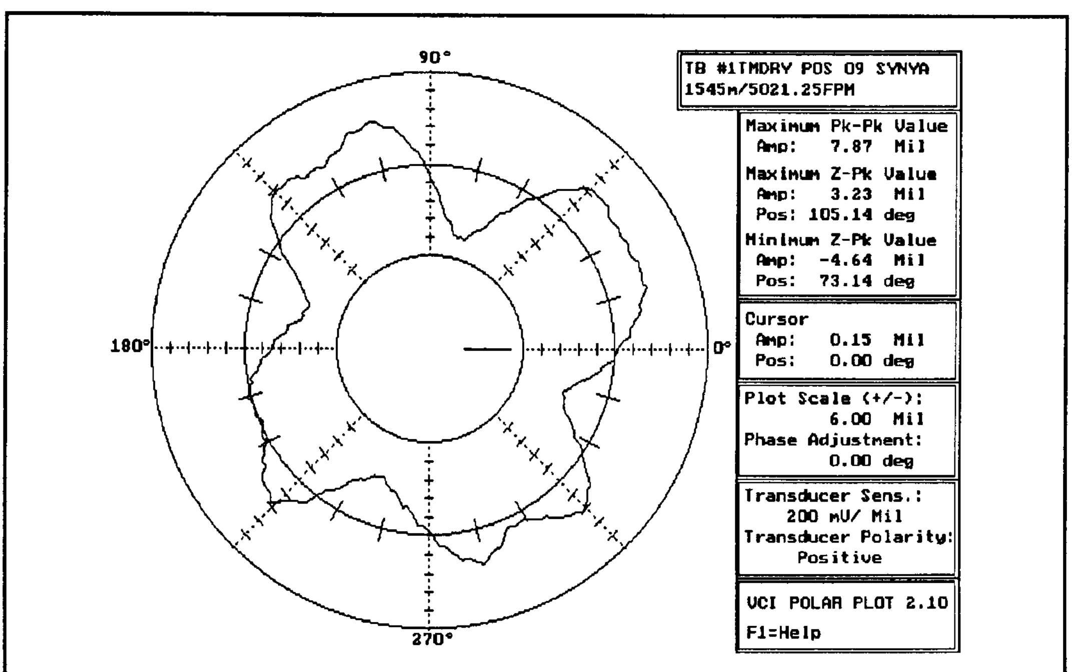

Fig. 7-3. Polar Plot of Synchronous Data.

roll is readily available.

Hardware Setup

When measuring relative motion between two rolls, the transducer base must be attached to one bearing housing and the transducer probe must be placed up against the other bearing housing parallel to the direction of the nip. See Fig. 7-4. When measuring the relative motion between a Yankee Dryer and a Pressure Roll, the machine frame can be used in place of the Yankee Dryer bearing. The transducer base can be attached to the bearing housing or frame with a heavy duty magnetic base. A threaded tab can also be tack welded to the bearing housing and/or frame. The transducer probe should be set near the center of its linear range. This can be accomplished by monitoring the display on the signal conditioner or measuring the gap voltage. Orientation of the transducer must be observed and recorded to determine if the rolls have high places or low places.

The important rule to remember is if the plunger on the contacting displacement transducer goes out, either a positive or negative-going signal is produced. If the plunger is pushed in, an opposite signal is produced. For proximity transducers, either a positive or negative signal is produced when the gap voltage is reduced, and an opposite signal is produced when the gap voltage is increased. The transducer should be positioned to generate a signal of the correct polarity.

When a high or low place, or some other eccentricity is measured, the radial location of the irregularity is often required. Problems associated with identifying the location can be minimized if they are considered before the equipment is set up. For example, if the trigger and vibration transducer are placed at the same radial location, the irregularity can be measured from trigger time. Such installations are often not possible. If the trigger and vibration transducer are installed at different radial locations and the angular distance is not recorded, the location of the irregularity cannot be identified. When the two transducers cannot be placed together, the angle between them must be measured and recorded to identify the location of the irregularity.

For example, if the trigger is placed 90 degrees in the direction of rotation from the displacement transducer, then 90 degrees must be added to the measured angle of the irregularity. If the trigger is placed 81 degrees counter to rotation from the displacement transducer, then 81 degrees must be subtracted from the measured angle to the irregularity. An easier way to remember this rule is: always add the angle in the direction of rotation between the vibration transducer and the trigger to the measured angle of the irregularity. If the sum is greater than 360 degrees, subtract 360 degrees. This procedure will give you the angle between the once-per-revolution marker and the irregularity, measured in the direction opposite to roll rotation.

Assume the once-per-revolution marker is at a point called zero. See Fig. 7-5. The high place on the roll is 90 degrees clockwise from the once-per-revolution marker. Please note that the once-per-revolution marker and the high place are rotating in the clockwise direction.

The displacement transducer is installed at a point where the relative motion can be measured in the direction of the nip. The trigger is installed at a convenient point. In Fig. 7-5, the trigger is installed 60 degrees clockwise from the displacement transducer.

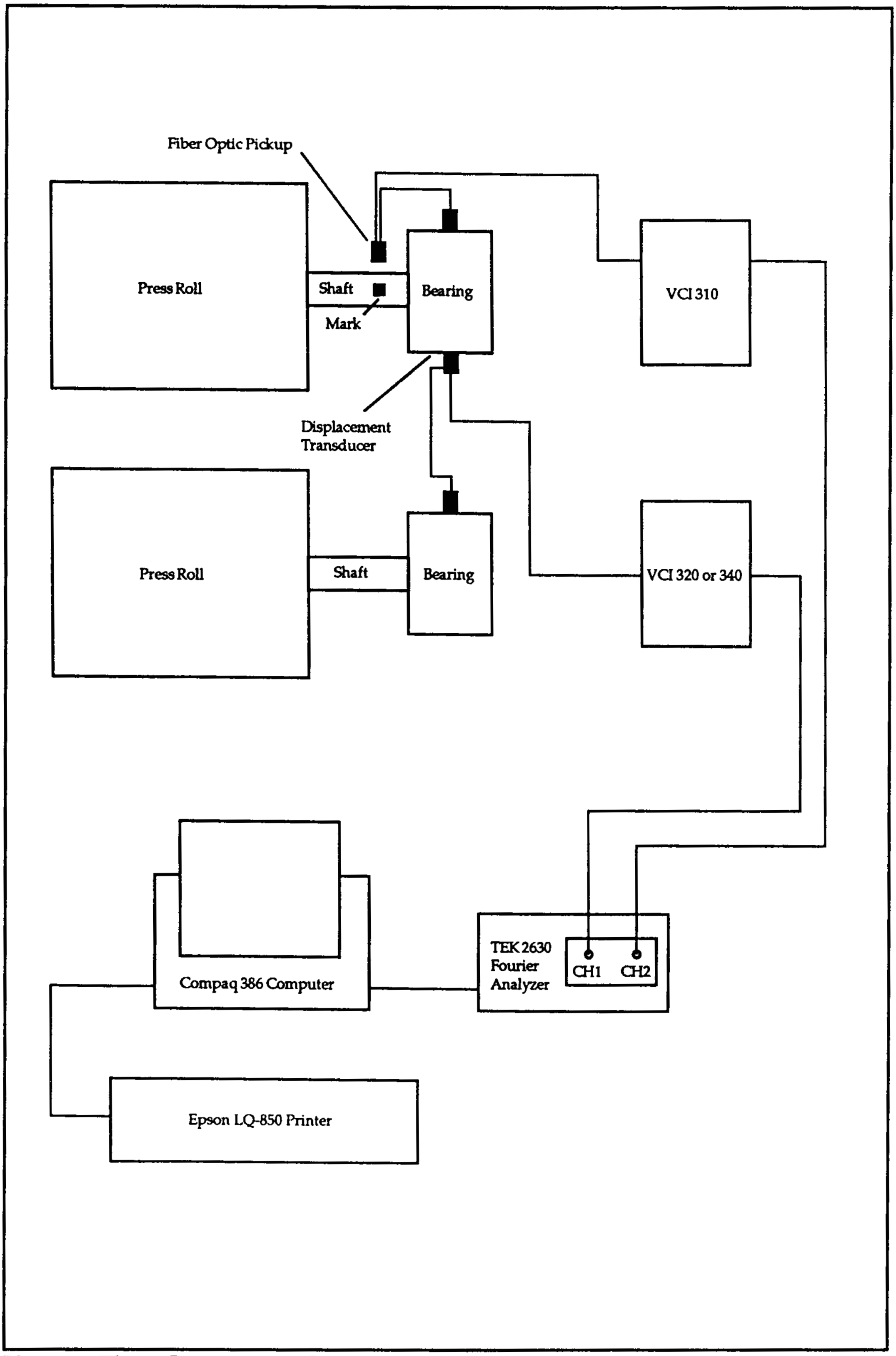

Fig. 7-4. The VCI Hardware System Setup.

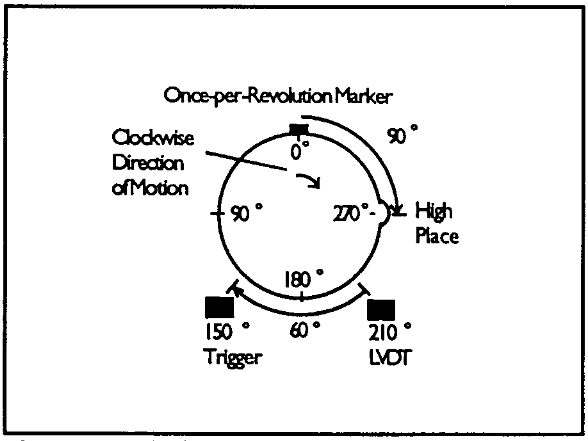

Fig. 7-5. Relationship between the High Place, Once-per-revolution Marker, Trigger, and Displacement Transducer.

When the high place goes through the nip, the press rolls are pushed apart and the LVDT plunger generates a signal that will identify the high place. The once-per-revolution marker passes under the trigger 270 degrees before the high place passes under the trigger. However, the high place passes by the transducer 60 degrees before it passes the trigger. In this case, the high place would be measured at 210 degrees. Using the above described procedure, the actual location of the high place is calculated as $60° + 210° = 270°$. The angular distance between the once-per-revolution marker and the high place is measured in the direction opposite to roll rotation.

Dynamic Measurement of Rolls

Often the requirement exists to measure the dynamic or exact roll circumference or shape. If the roll is accessible during operation, this measurement can be accomplished while the machine is running, while the roll is being ground, or while the machine is down.

Following are some examples of the information expected from such tests:

1. Shape of the roll
2. Total runout and location of the high or low places
3. Depth of the grind or finish on ferromagnetic rolls
4. Evaluation of syphon operation
5. The relative number and depth of porous holes or "cat eyes" on the surface of a Yankee, based on sampling techniques
6. The actual growth or expansion of the roll for a given temperature difference

When performing dynamic measurements, a noncontacting displacement transducer

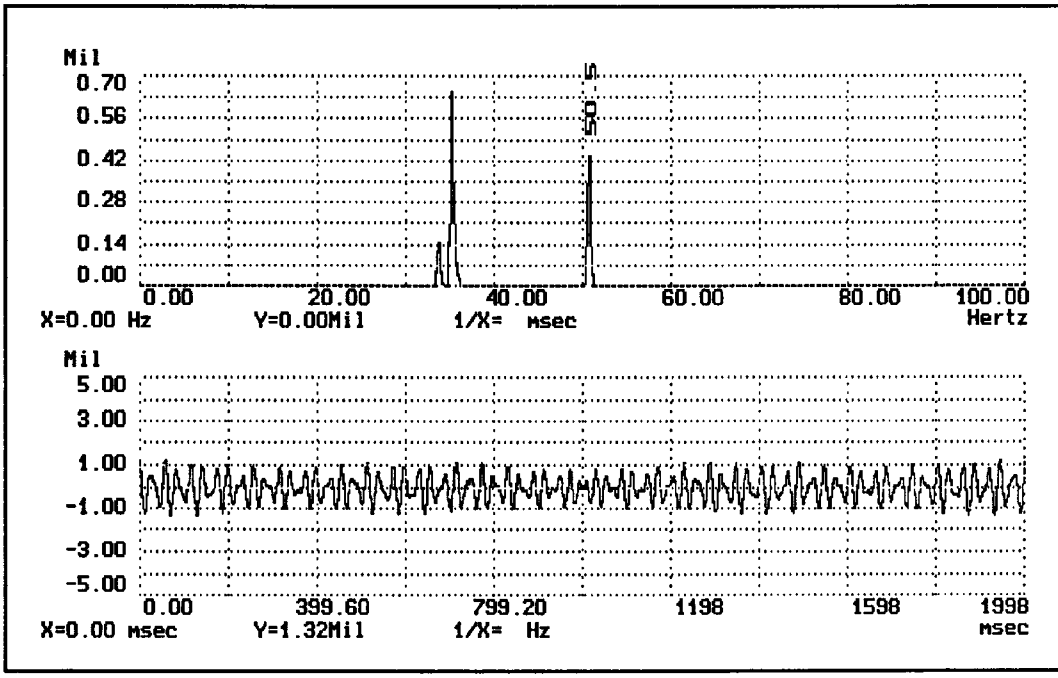

Fig. 7-6. Relative Motion between Transducer and Yankee Dryer.

should be used on ferromagnetic rolls, and a contacting displacement transducer should be used on all other rolls.

Suppose after a Yankee grind, we wanted to know the depth or finish of the grind and how many high and/or low places are on the roll.

When the grinding fixtures have been removed from the machine, the best place to mount the transducer is on an open doctor blade. Using a heavy duty magnetic base with a flex-o-post, the proximity transducer is mounted on the doctor blade and adjusted for the proper gap voltage.

WARNING: The gap voltage must be monitored at all times because if steam is applied during data acquisition, the Yankee can expand enough to destroy the transducer.

The first measurement is made to determine if there is any relative motion between the transducer and the Yankee Dryer while the Yankee is not rotating. The amplitude and frequency of the vibration are both of concern. Fig. 7-6 contains this measurement. The relative motion is 0.65 mils at 35 Hz, and 0.45 mils at 50.5 Hz. Both frequencies are narrow-banded and indicate a relatively undamped resonance. During this test, some multiples of Yankee speed cannot be equal to either of these frequencies.

Fig. 7-7 contains the data from the Yankee while rotating at 7.5 RPM. The spectral line at 0.625 Hz is five times Yankee speed. This frequency is caused by five high places on the Yankee. Since the Yankee has six syphons, this may indicate one syphon may not be working properly. Note that all positive peaks in the time domain are not evenly spaced. The high frequency noise riding the low frequency is the depth of the roll finish. The depth of the finish on this roll varies from about 0.2 to 0.4 mils. At higher speeds, the

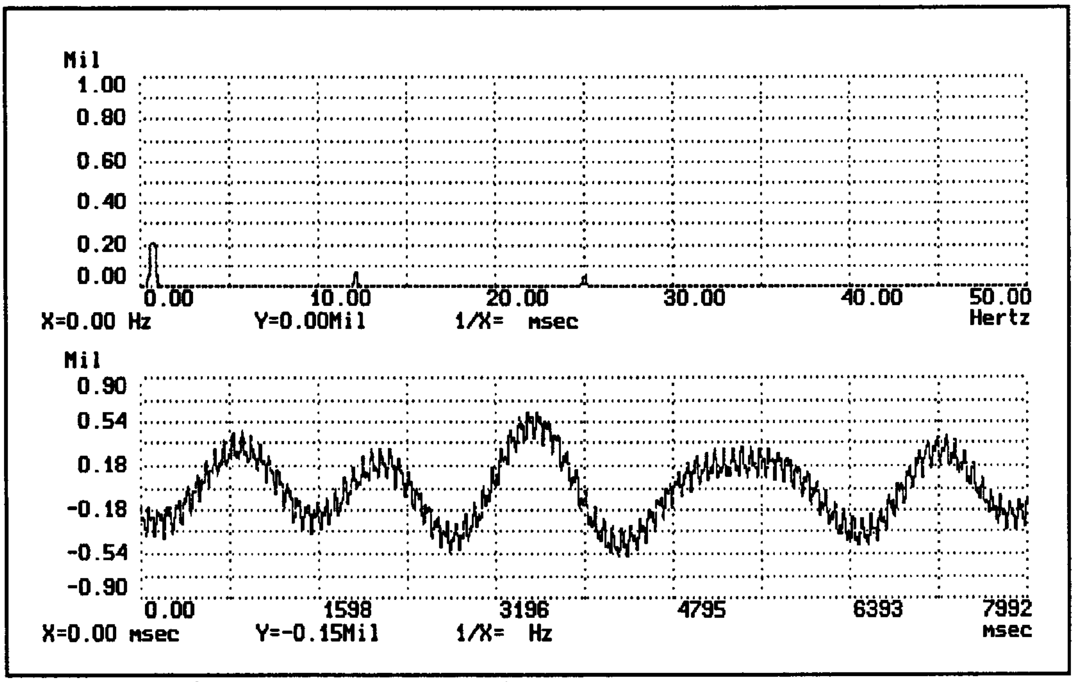

Fig. 7-7. Data from Yankee while Rotating at 7.5 RPM.

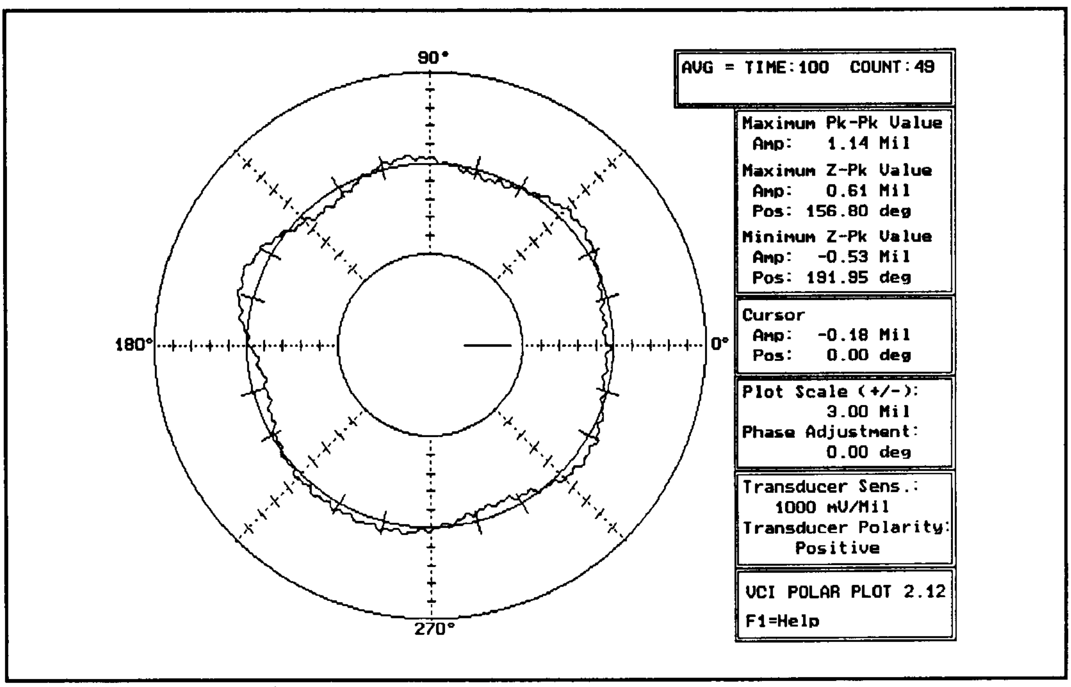

Fig. 7-8. Polar Plot of Yankee Taken at Higher Speed.

high frequency noise is not observed because the high frequency exceeds transducer capability. Please note the resonant frequencies of 35 and 50.5 Hz are not present in the frequency spectrum. These frequencies were eliminated by synchronous time averaging. Also note that only 49 averages, or a signal to noise ratio of 7-to-1, were needed to average out those frequencies. Generally, approximately 100 averages are needed.

Fig. 7-8 is a polar plot of the Yankee taken at a higher speed. This data indicates the five high places are still present. However, the high frequency noise is not present.

If the roll of interest is not ferromagnetic, the shape and/or runout can be measured with a contacting displacement transducer. If bars have been set into the roll and the high places have been ground off leaving some set still in the roll, the set bars may not be measurable until the roll is placed in nip under load. The finish on these rolls cannot be measured accurately with the contacting transducer. However, the finish may not be as important.

PROBLEMS ASSOCIATED WITH ROLLS IN NIP

As stated previously, two or more rolls in nip may give rise to various problems in a given situation.

Eccentric Rolls

An eccentric roll is one that is not round. The physical static shape of the roll could be out-of-round. However, the static shape of the roll while under load could be round, and the dynamic shape could be eccentric because of hard places in the cover and/or flexibility in the roll shell. The homogeneity of the material and thermal expansion or the lack thereof could also cause deformation of the roll.

How much of what kind of eccentricity can be tolerated? The answer to this question is somewhat evasive. It is recommended that rolls be manufactured, covered, and ground with less than 0.0005 inches runout. The technology is available to measure and accomplish this specification. The problem often arises when a roll is installed. Rolls operate under pressures of 500 or more pounds per linear inch (PLI). Roll temperature is often increased considerably and the area of the circumference may not operate at the same temperature. The coefficient of expansion of all materials may not be equal. All of these variables affect the roundness and hardness of a roll. Roundness should be measured both before installation and under operating conditions. Residual imbalance should be less than one mil.

Improper Ratios of Roll Sizes

If the rolls are round, the ratio of the roll sizes is less important. However, if one roll has a high or hard place, the ratio of the roll sizes is critical. Two rolls in nip behave similar to gears, in that one high place on one roll can severely damage the other roll, the felt, and cause quality problems in the product. If the circumferences of the two rolls have a common factor and one roll is eccentric, barring of the rolls can occur. If the felt length contains the same common factor, the felt may get streaks. If the felt length does not contain the common factor, streaks may not occur. The highest common factor of the circumference of the two rolls and the length of the felt equals the distance between the bars or streaks. Once a roll is barred, it can bar the other roll and/or the felt. The uncommon factors of the rolls and felt are the number of bars or streaks that can occur. If the highest common factor is less than two inches, the bars may be so close together that the wear appears even, except for some critical applications.

Factoring the circumferences of two rolls can be more complex than it first appears. For example, if the circumference of one roll is 82 inches and the circumference of the other roll is 94 inches, the factors of the two numbers would be:

$$82 = 2 \times 41 \textit{ and } 94 = 2 \times 47$$

The highest common factor of the two rolls is two, and the uncommon factors of the two rolls are 41 and 47. If both rolls were covered or one roll was a granite roll, and one roll was eccentric, one roll would get 41 bars and the other roll would get 47 bars. The bars on both rolls would be two inches apart.

Seldom are problems so simple. For example, how do you factor a number that contains a decimal point? Using the same rolls after thermal expansion, the circumference of each roll could be 82.2 and 94.2. The best factoring method is to multiply both numbers by 10 to remove the decimal before factoring. After factoring is completed, divide the highest common factor by 10 to arrive at the correct number. For example:

$$82.2 \times 10 = 822 = 2 \times 3 \times 137$$

$$94.2 \times 10 = 942 = 2 \times 3 \times 157$$

The highest common factor is:

$$\frac{2 \times 3}{10} = \frac{6}{10} = 0.6$$

The uncommon factors are 137 and 157. In this case, one roll would get 137 bars, the other would get 157 bars, and the bars would be 0.6 inches apart. Because the bars would be very close together, the rolls would appear evenly worn and may not be considered a problem.

Suppose the circumferences of the two rolls are 78.84 inches and 96.36 inches. Both numbers should be multiplied by 100 before factoring:

$$78.84 \times 100 = 7884 = 2 \times 2 \times 3 \times 3 \times 3 \times 73$$

$$96.36 \times 100 = 9636 = 2 \times 2 \times 3 \times 11 \times 73$$

The highest common factor is:

$$2 \times 2 \times 3 \times 73 = 876$$

$$\frac{876}{100} = 8.76$$

The uncommon factors are 9 and 11. This means one roll can get 9 bars and the other roll can get 11 bars. The bars on each roll will be 8.76 inches apart.

These examples indicate that changing the roll circumference by a small amount, 0.1, 0.01, or even 0.001 in some cases, could be a serious problem. The calculations of all the various combinations with a calculator would take a long time. The **VCI Roll Ratio Program** can perform all the calculations and print the results in a few minutes. This program lists the minimum and maximum circumference of each roll, as entered by the operator. The program calculates the uncommon factor, which is the number of bars on each roll. The highest common factor, which is the distance between the bars, is also calculated. The circumferences of the various combinations of roll sizes that can cause a problem are also listed. Table 7-1 contains a single page produced by the **Roll Ratio Program**. The roll sizes listed can generally produce over 25 pages of undesirable combinations. When this much data is generated, it is very difficult to identify a roll combination that is satisfactory.

The next logical step is to utilize the **Rusch Chart**. See Fig. 7-9. The x-axis is the range between the minimum and maximum circumferences of roll 1. The y-axis is the range between the minimum and maximum circumferences of roll 2. The outward and upward sloping lines appear to be parallel. However, they are "sunburst" lines that appear to be parallel because such a small segment of the whole plot is observed. These lines can be analyzed to determine the number of bars each roll can get with certain combinations of roll sizes. More bars can be generated when the lines are closest together. Fewer bars can be generated when there is more distance between the lines. It is best to use a roll combination where there are no lines. The next best choice is to choose the roll combination where the lines are closest together.

It is always best to use round rolls. However, since perfectly round rolls cannot be produced, the **VCI Roll Ratio Program** and the **Rusch Chart** are often required to solve problems with rolls in nip.

Resonant Frequencies

All machines and machine components have one or more resonant frequencies. We must be aware of the presence of natural frequencies and watch for them. A roll will not vibrate at its natural frequency (normally the natural frequency of the loading mechanism) unless it is excited. Natural frequencies can be excited in several ways. An impact can excite a natural frequency if enough energy is introduced by the impact. Therefore, a bump test is often used to determine the natural frequency of various items. The natural frequency can be excited by a discrete frequency such as roll speed, a harmonic, or a subharmonic if one of these frequencies is within the bandwidth of the natural frequency. See Chapter 1, Figs. 1-9 and 1-10, for typical resonant frequency curves for a relatively damped and relatively well-damped frequency, respectively. A natural frequency may not harm a roll if something other than the roll is exciting the frequency and the natural frequency is not a harmonic of roll speed. For example, if roll speed is 7 Hz and the natural frequency is 45 Hz, a problem could not occur because 45 is not a harmonic of 7. The numbers 7 and 45 do not contain a common factor other than one. If viewed from the time domain, the same result would occur:

$$\frac{45}{7} = 6.4286$$

Table 7-1. Roll Ratio Program.

Paper Makers Inc, #3 Paper Machine

Roll 1: 412.00 to 412.70 inches (Yankee Dryer)
Roll 2: 87.00 to 87.50 inches (Pressure Roll)

#BARS	BAR INTERVAL	ROLL 1	ROLL 2
33 / 7	12.48	412.00	87.40
33 / 7	12.49	412.17	87.43
33 / 7	12.49	412.33	87.47
33 / 7	12.50	412.50	87.50
52 / 11	7.92	412.10	87.18
52 / 11	7.93	412.36	87.23
52 / 11	7.93	412.62	87.29
71 / 15	5.80	412.15	87.08
71 / 15	5.81	412.51	87.15
85 / 18	4.85	412.25	87.30
85 / 18	4.85	412.67	87.39
90 / 19	4.58	412.20	87.02
90 / 19	4.58	412.65	87.12
109 / 23	3.78	412.47	87.03
109 / 23	3.78	412.56	87.05
109 / 23	3.79	412.66	87.08
118 / 25	3.49	412.24	87.34
118 / 25	3.49	412.41	87.37
118 / 25	3.50	412.57	87.41
123 / 26	3.35	412.05	87.10

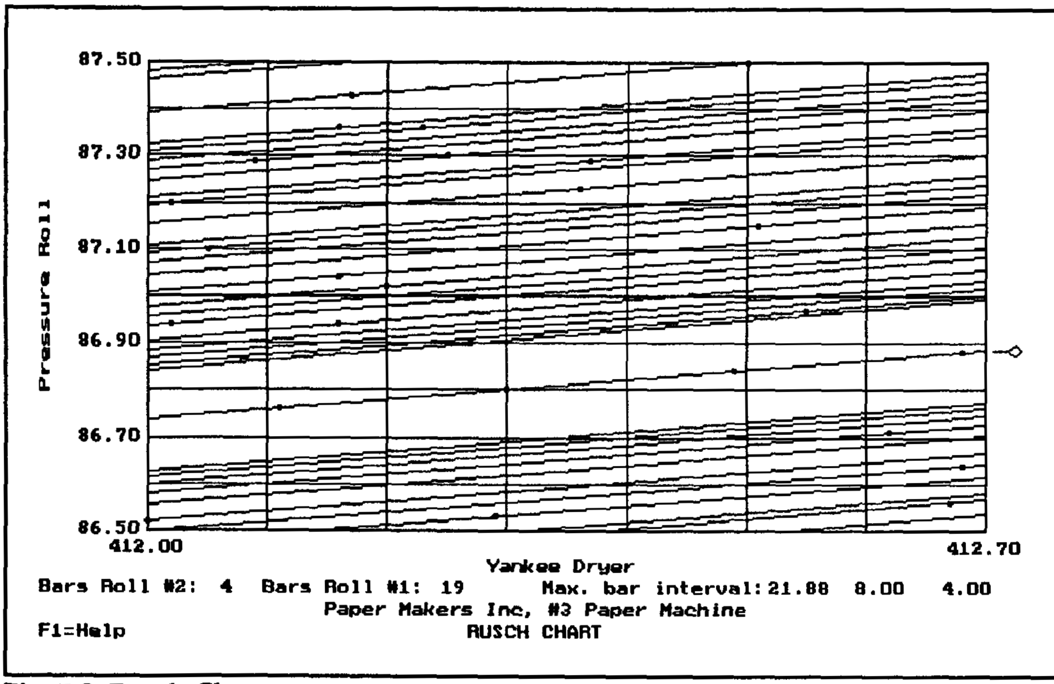

Fig. 7-9. Rusch Chart.

$$\frac{1}{7} = 0.1429$$

$$\frac{1}{45} = 0.0222$$

$$\frac{0.1429}{0.0222} = 6.4369$$

The natural frequency in this case could not cause barring, even if it were excited by some other source.

Now, increase machine speed. Roll speed is 9 Hz and the natural frequency is still 45 Hz. Since 9 and 45 contain a common factor of 9, the roll would get 5 bars, 9 inches apart. The nature of this problem could be quite serious because a regenerative feedback loop is completed. This means as the roll gets barred or deteriorates, more energy is introduced into the system and serves as more excitation for the natural frequency. In cases such as this, serious damage can occur very quickly. The easiest solution in this case is to avoid speeds where roll speed or some harmonic of roll speed equals the natural frequency. If this is not possible, the natural frequency should be changed. The rolls may be round in this case and still get barred.

Installation of Improperly Ground Rolls

Regardless of the material used to make and cover a roll, it may take a set. Meaning, once a roll has been barred, the material at the spacing of the bars has been compressed or expanded. Such rolls can be ground very smooth and round as measured by currently acceptable standards. However, if enough material is not removed to get below the compressed or expanded material, problems can be anticipated when the roll is reinstalled in the machine at startup. Once the roll is installed and placed under load (400 PLI, etc.), the set or hard places can be measured with the system. Various types of hardness testers may be able to identify the problem before installation.

If the cause of the barring has not been found and corrected, the new roll may be barred also. If the installed roll has been barred in the past, and has not been recovered or ground correctly, the bars will reappear very quickly and the roll will have to be replaced again.

DIAGNOSING PROBLEMS

This section describes the techniques of obtaining useful information and proper analysis of this information to reveal the cause and source of a problem.

Data Collection and Analysis of Overall Vibration Data

This data is best for identifying defects in bearings and gears, as applicable. However, this data can also be used to identify some simple felt and roll cover problems. Circumferential cracks in suction rolls can also be identified with overall data. The best signal definition for cover problems is normally in the axial direction.

If the roll has a number of bars, the frequency generated will be the number of bars times the speed of the roll. A spectrum from such a problem can be narrow-banded if the signal is relatively undamped. The spectrum can be wide-banded if the signal is relatively damped. The spectrum can have several spectral lines with the difference frequency between the spectral lines equal to roll speed. These spectral lines must be exactly divisible by roll speed. If they are not, the roll does not have a cover problem. These spectral lines can be between four or five times roll speed to over 50 times roll speed, or more. Fig. 7-10 contains the frequency spectrum from a roll that has a relatively simple cover problem.

Fig. 7-11 contains a 40 Hz frequency window. The average difference frequency between the spectral lines is 2.2 Hz. This frequency is roll speed. The discrete frequency divided by roll speed equals the number of bars. Any one of the four higher peaks could be the discrete frequency. These four peaks probably represent the bandwidth of a natural frequency. The discrete frequency is difficult to identify from this data.

Fig. 7-12 contains the same data after synchronous time averaging was performed. This is the overall data. A trigger was not available. However, the VCI System can perform synchronous time averaging using the frequency data for a trigger if the data is consistent. Also note that only 20 averages were taken because of the short piece of recorded data. However, the data is significantly improved.

Fig. 7-13 contains the expanded time data for one revolution of the press roll. There are 29 cycles occurring during the time period for one revolution of the roll. Also, the

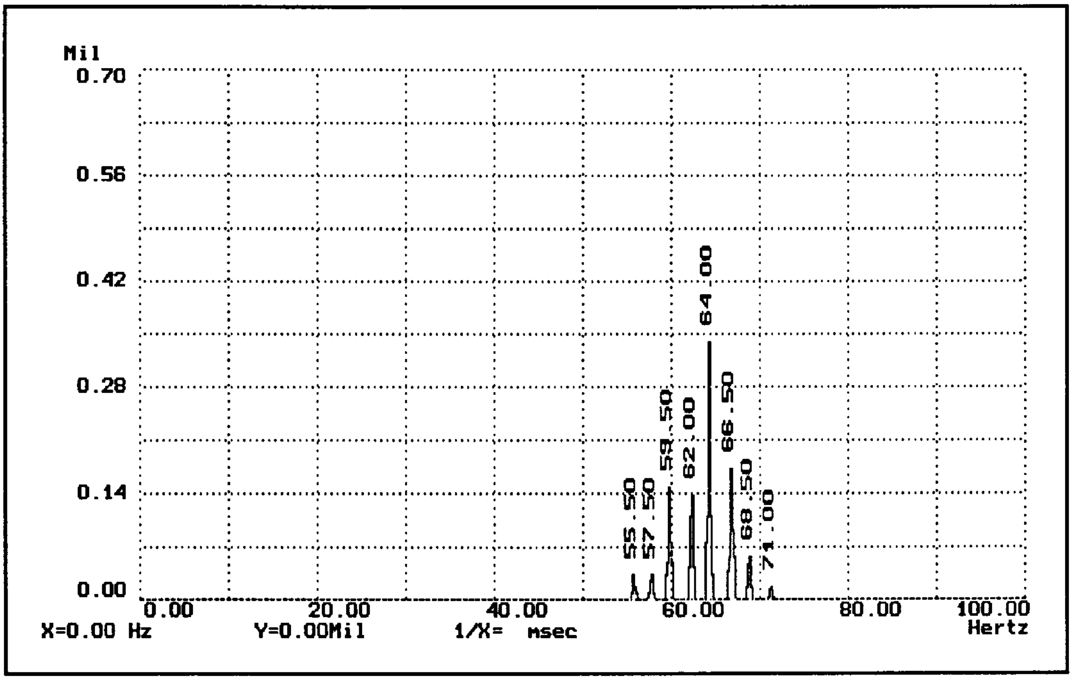

Fig. 7-10. Frequency Spectrum from a Roll with a Cover Problem.

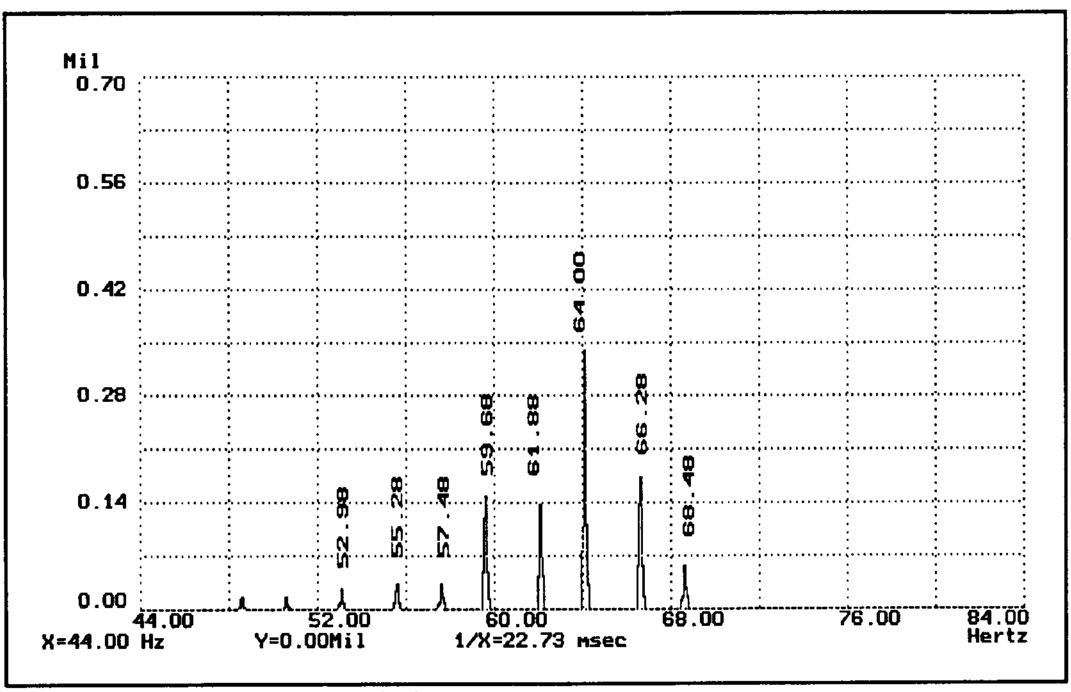

Fig. 7-11. 40 Hz Frequency Window.

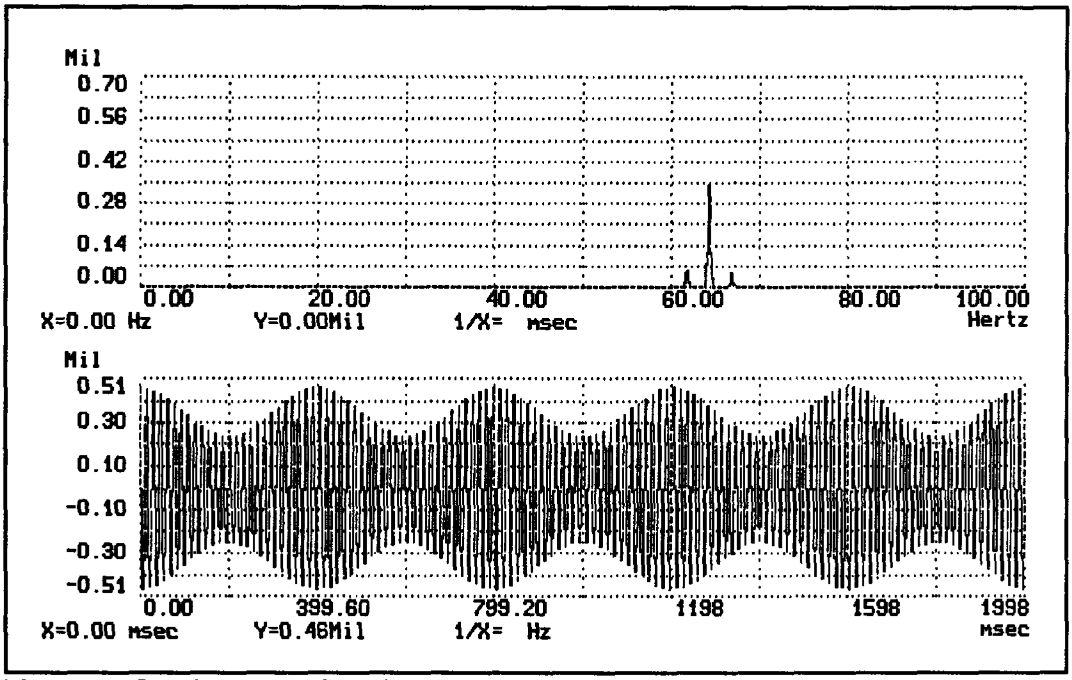

Fig. 7-12. Synchronous Time Averaged Data.

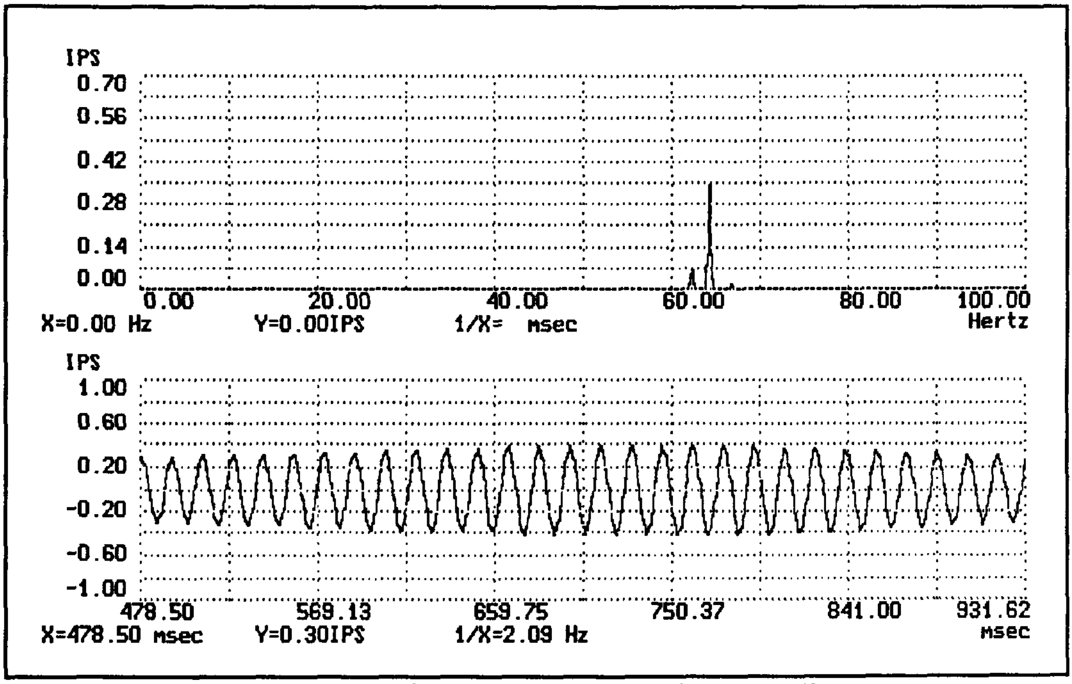

Fig. 7-13. Expanded Time Data for One Revolution of Press Roll.

discrete frequency divided by roll speed is:

$$\frac{64}{2.2} = 29,$$

which is equal to the number of bars on the roll.

Fig. 7-14 is a polar plot from the **Polar Plot Program.** This plot contains the synchronous time averaged signal for one revolution of the roll plotted against a circle containing angle marks and a scale. The angle marks identify the relative location of points of interest and the scale indicates the relative depth of the bars, a little under a mil in this case.

The system equipment setup for this testing is contained in the previous figures. These setups and tests can also be used for other diagnostic procedures such as imbalance, misalignment, bent shaft, and looseness.

Felt problems generate similar spectra, except the frequency is often lower, between 8 and 20 Hz. However, felts on high speed machines can generate frequencies over 100 Hz. The best signal definition is obtained in the radial direction on the moveable press roll. If a low frequency range or narrow zoom range is used, these spectra may have several spectral lines. The difference frequency between the spectral lines is the felt speed. In some cases, the signal may be the natural frequency of the movable loading diaphragm and may be excited by a wad on the felt, a dewatering problem, a manufacturing problem, or streaks in the felt that result from barred rolls. Fig. 7-15 contains the spectrum of a felt problem.

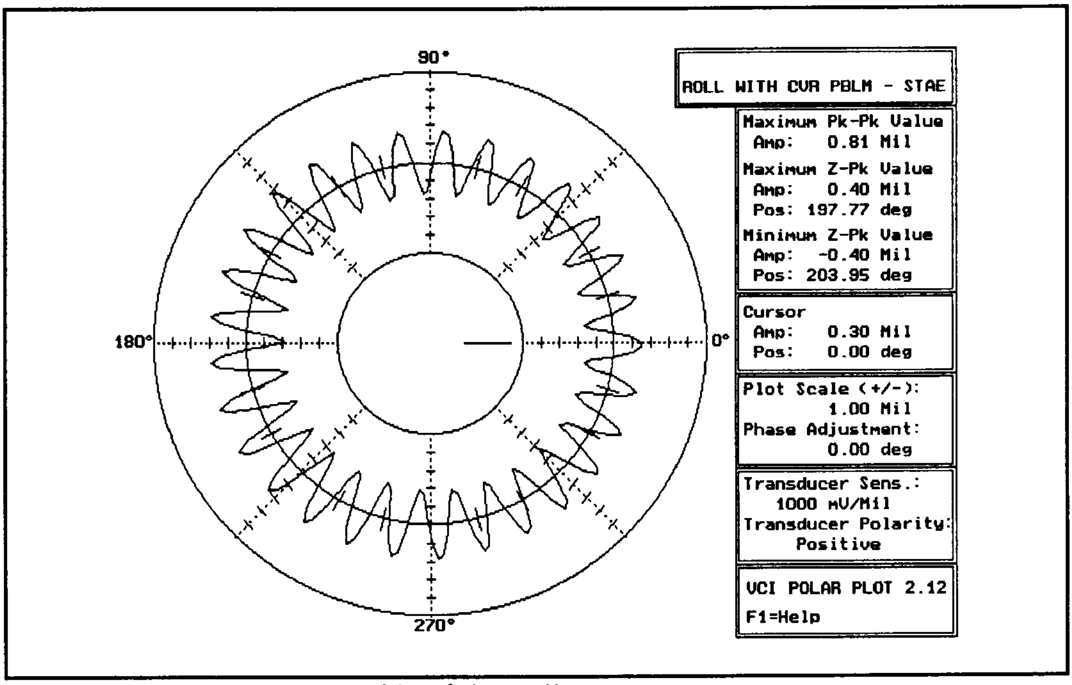

Fig. 7-14. Cross-Sectional Profile of the Roll.

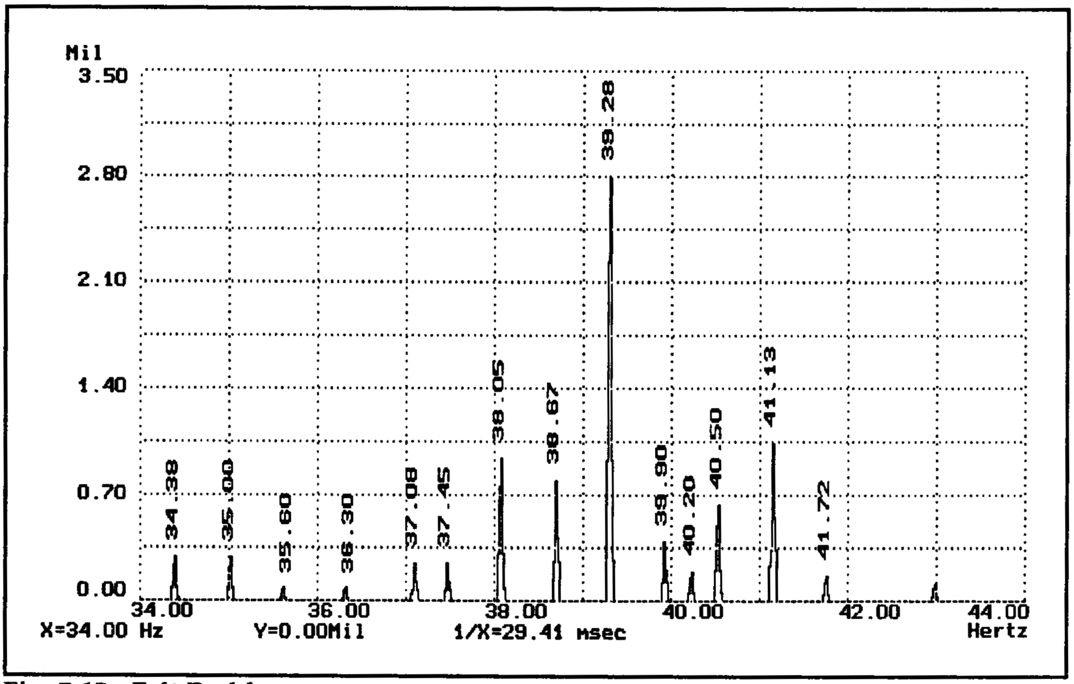

Fig. 7-15. Felt Problem.

$$\textit{Felt Speed} = 0.613\ Hz$$

$$\textit{Spectral line} = 39.275\ Hz$$

$$\frac{39.275}{0.613} = 64\ \textit{streaks in felt}$$

If a felt on a high speed machine has a large number of streaks, the frequency generated can be 100 Hz or more. Suppose a felt is 87 feet long and has 187 streaks 5.6 inches apart. If this felt were installed on a machine running at 3600 FPM, the frequency generated would be:

$$\left(\frac{\frac{3600}{87}}{60}\right) \times 187 = 128.9\ Hz$$

<u>Relative Motion between Rolls</u>

It is often difficult to determine the source of the problem when more than one roll or component is in nip. This can be true for several reasons:

1. The amplitudes of frequencies can add and subtract, depending on their phase relationships. This results in vibration amplitudes being either overstated or understated.

2. More than one component can be very close to the same size. The analyst and/or equipment may not have the capability of separating the two frequencies.

3. X times the speed of one component may equal Y times the speed of another component.

4. The use of the overall time domain signal is diminished. This is true because the overall signal is a composite of all the frequencies, and is often too complex for analysis.

5. It is often impossible to determine if a problem exists until the various frequencies are separated. These insidious problems often cause catastrophic failures. A few examples may provide some clarity.

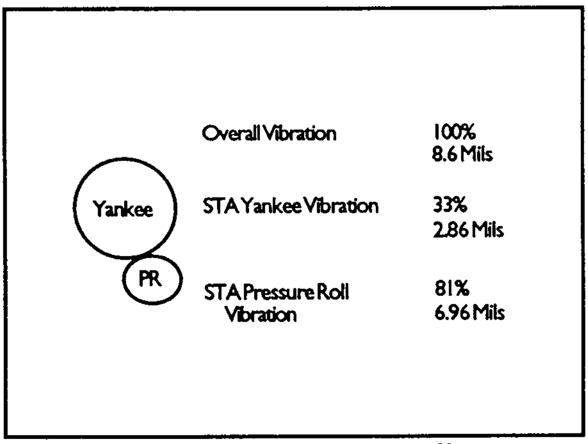

Fig. 7-16. Yankee and Pressure Roll.

Please refer to Figs. 7-16 through 7-23 for the following diagnosis of a Yankee and Pressure Roll problem. The hardware setup should be similar to that shown in Fig. 7-4. The overall time signal indicates the overall relative motion between the two rolls is about 8.6 mils. See Fig. 7-17. This vibration level is high by most standards. The signal is also truncated. Maximum amplitude of the signal in the negative direction is greater than maximum amplitude of the signal in the positive direction, but which roll is causing the truncation?

The vibration transducer should be positioned as it was for the overall measurement. Now, position the optical pickup to obtain a stable trigger from the marker on the Yankee, and record the angle between the transducer and optical pickup in the direction of the Yankee rotation; assume 30 degrees in this case. After taking 100 averages, the signal in Fig. 7-18 is obtained. The envelope of this signal cannot be observed in the overall signal. This signal is not truncated. However, the signal does indicate about eight high places on the Yankee. This signal indicates the Yankee is causing about 33% of the overall vibration.

Next, move the optical pickup to obtain a stable trigger from the Pressure Roll, and record the angle between the transducer and the optical pickup in the direction of rotation of the Pressure Roll. After taking 100 averages, the signal in Fig. 7-19 is obtained. The signal

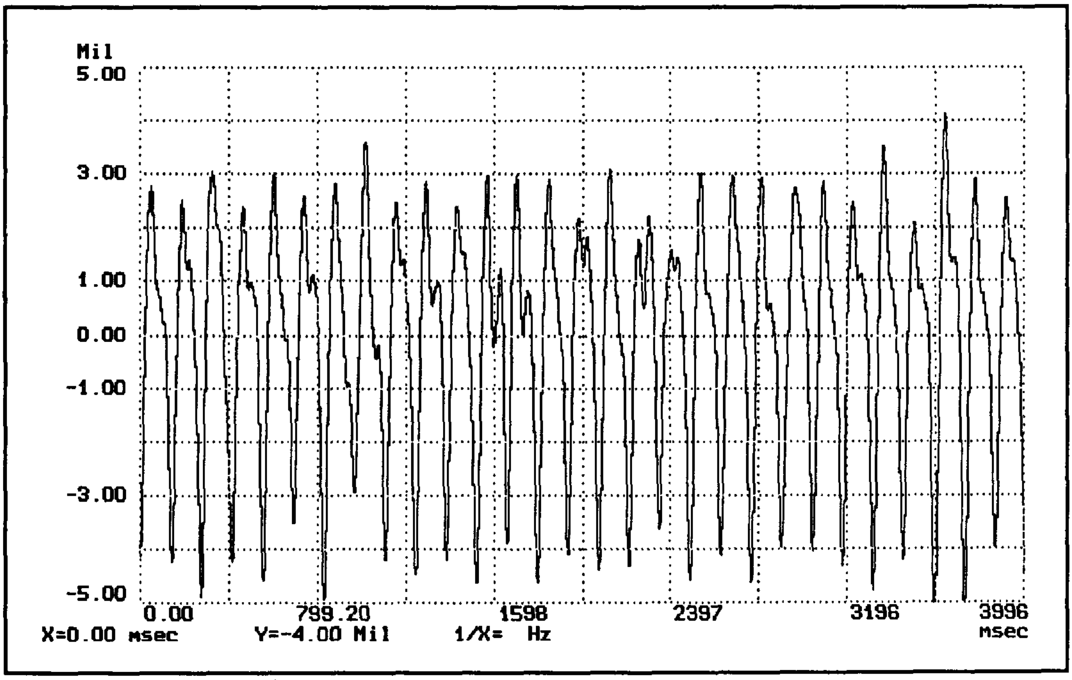

Fig. 7-17. Yankee and Pressure Roll.

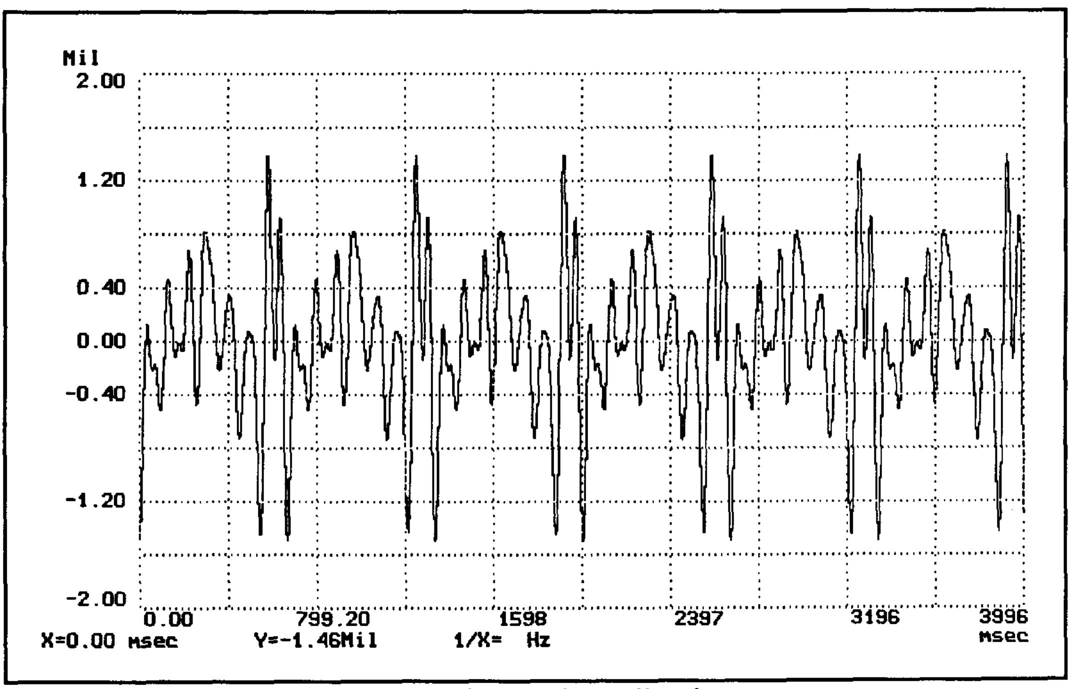

Fig. 7-18. Synchronous Time Averaging Yankee Vibration.

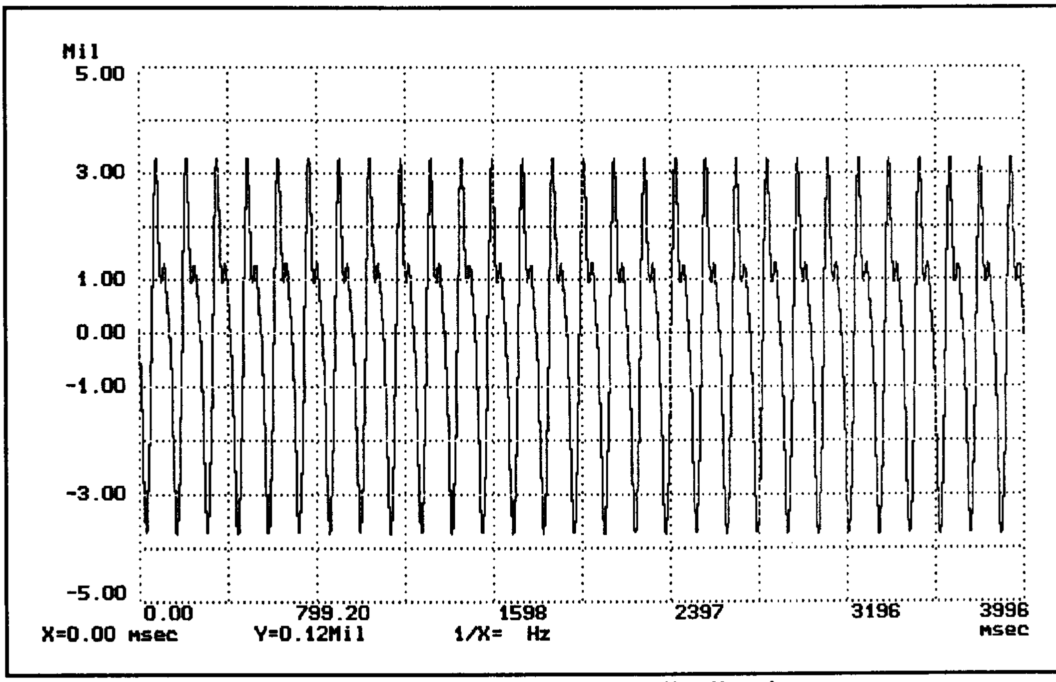

Fig. 7-19. Synchronous Time Averaging Pressure Roll Vibration.

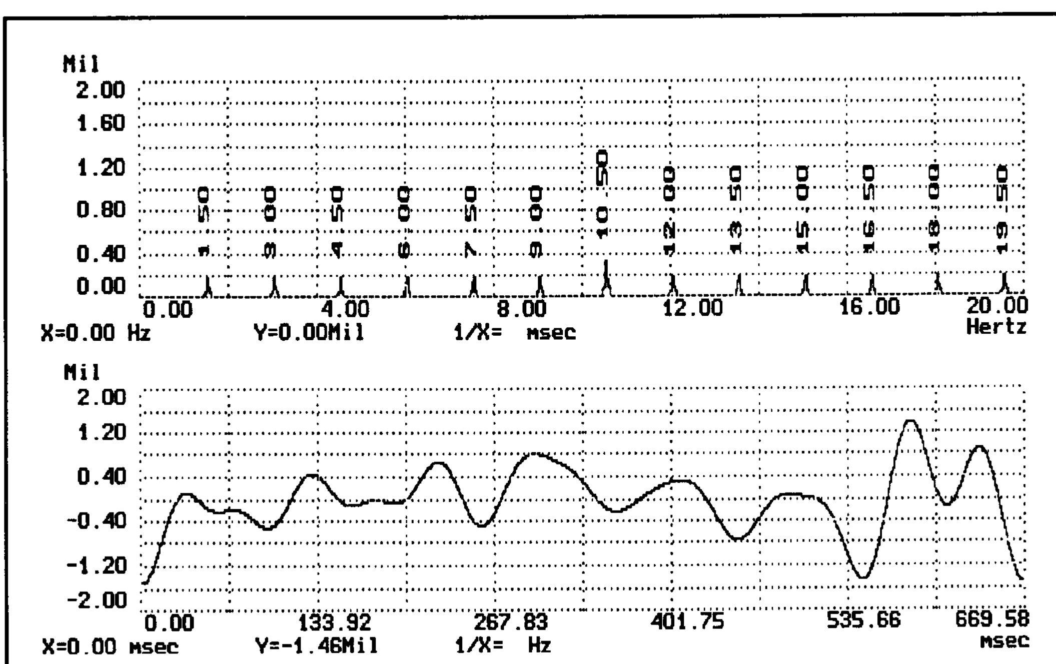

Fig. 7-20. One Revolution of Yankee.

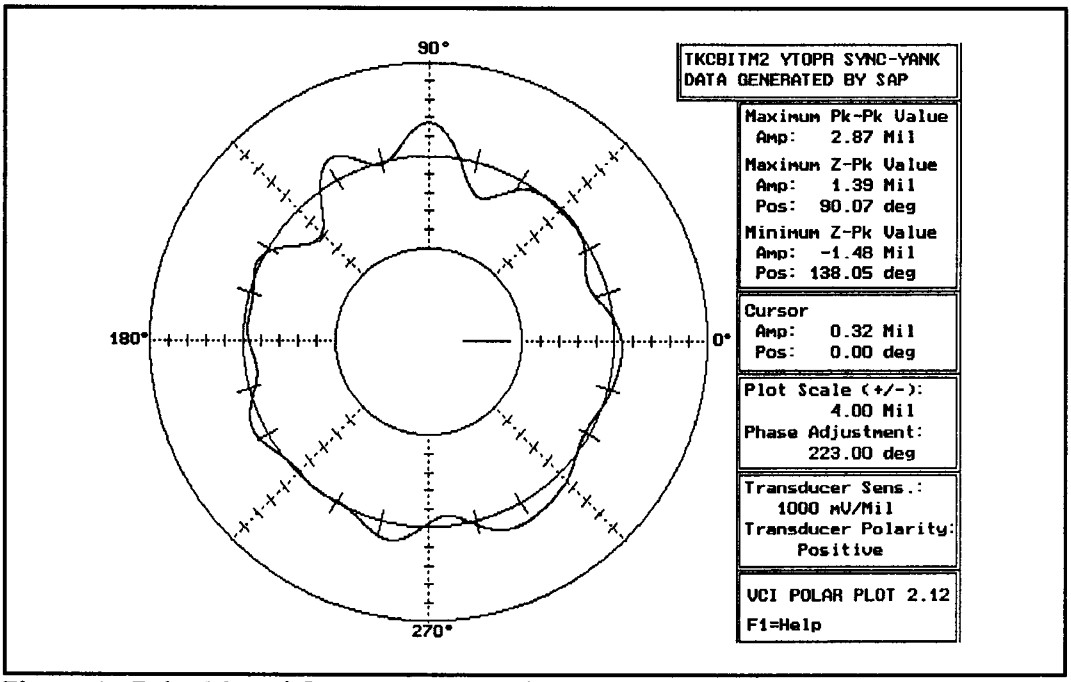

Fig. 7-21. Polar Plot of One Revolution of Yankee.

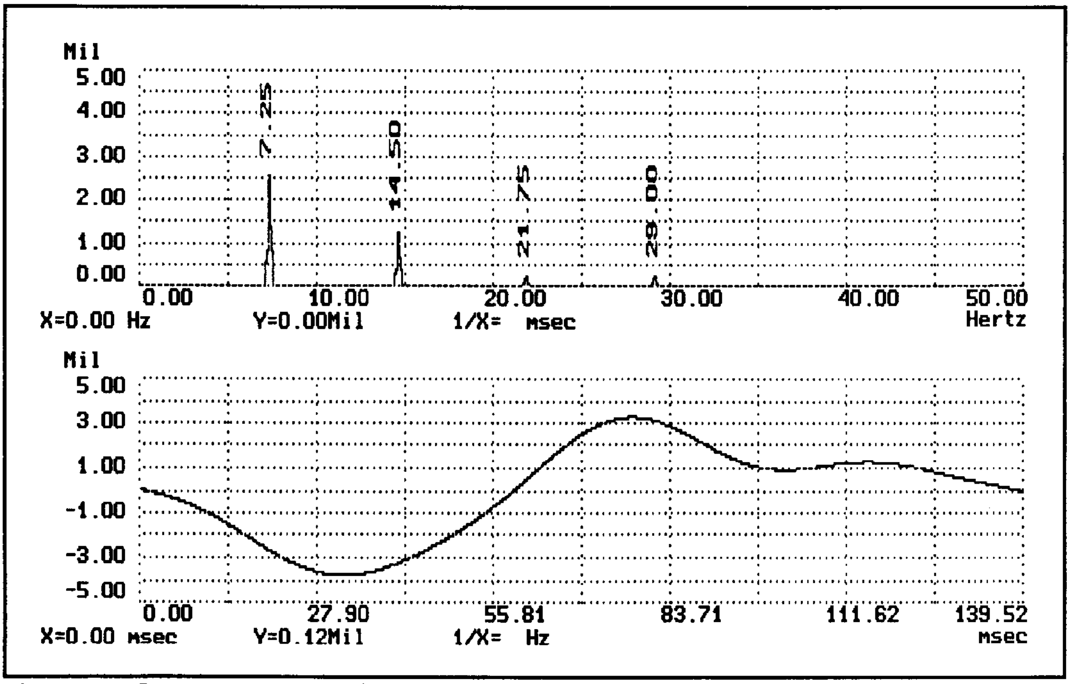

Fig. 7-22. One Revolution of Pressure Roll.

is both truncated and distorted. The Pressure Roll is causing 81% of the overall vibration level. The vibration caused by the felt is negligible in this case. The envelope of this signal can be observed in the overall signal.

This data indicates a definite problem with the Pressure Roll and a possible problem with the Yankee. More analysis is required to determine the nature and extent of the problem.

The time signal in Fig. 7-20 contains one revolution of the Yankee. This time signal indicates the Yankee has eight distinctive high places. The scale is ±2 Engineering Units, and indicates the maximum high place is 1.39 mils, and the peak-to-peak vibration is 2.86 mils.

Fig. 7-21 contains a polar plot of one revolution of the Yankee and provides an indication of the roll shape. The syphons on this Yankee should be checked for proper operation. The dynamic roundness (roll shape while in operation) should also be checked. The signal starts when the optical pickup is triggered. Each division represents 15 degrees. The second high place is 6 divisions or **6** × **15° = 90°** from trigger time. The angle between the vibration transducer and optical pickup was 30 degrees in the direction of rotation. To determine the location of the high place from the once-per-revolution marker, **90° + 30° = 120°**. The high place is 120 degrees from the marker on the shaft in the opposite direction of rotation.

Fig. 7-22 contains one revolution of the Pressure Roll. This data indicates a serious problem with the roll. The truncation and high amplitude are quite evident.

Fig. 7-23 is a polar plot of the Pressure Roll. This roll is severely lopsided.

This data indicates two distinctive and unrelated problems.

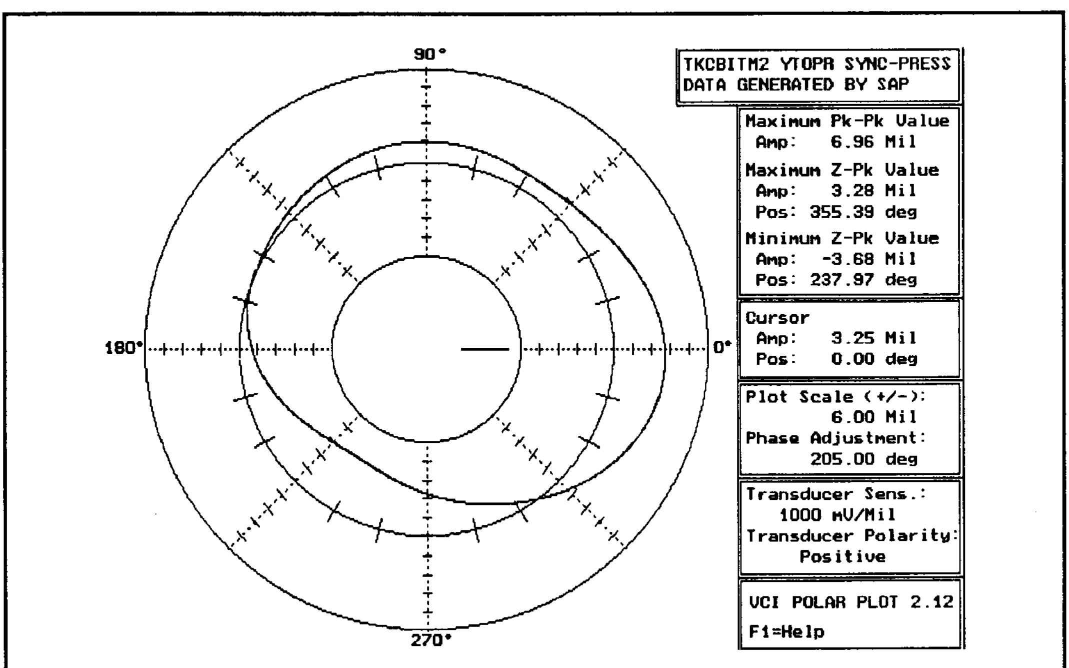

Fig. 7-23. Pressure Roll Polar Plot.

The Yankee Dryer has several high places. The syphons and the dynamic roundness of the roll must be checked. The lopsided Pressure Roll must be replaced. Corrective action can now be made based upon valid information.

Presently, we do not know the cause of lopsided rolls. They are relatively common and may be caused by the covering, curing, and/or grinding process.

One other possible reason for lopsided rubber rolls may be the way they are stored. It should be noted that rubber is not classified as being solid, but rather as non-Newtonian fluid. Some non-Newtonian fluids flow at extremely slow rates. When rubber rolls are stored for any extended period of time, rubber starts to flow to the bottom of the roll. This gradual migration of rubber results in a lopsided roll. The longer a roll sits, the more lopsided it becomes. A simple way to avoid this problem is to rotate the rolls when they are in storage.

This data establishes the requirement to measure the rolls before they are installed, after a grind, and at startup.

Figs. 7-24 through 7-31 contain the data from a simple two roll Second Press. The overall signal in Fig. 7-25 has an amplitude of about 7.1 mils. However, it is difficult to determine the condition of the rolls from this data. The time signal in Fig. 7-26 is the synchronous time averaged data from the top roll. The amplitude is quite low, and this signal is not observable in the overall data.

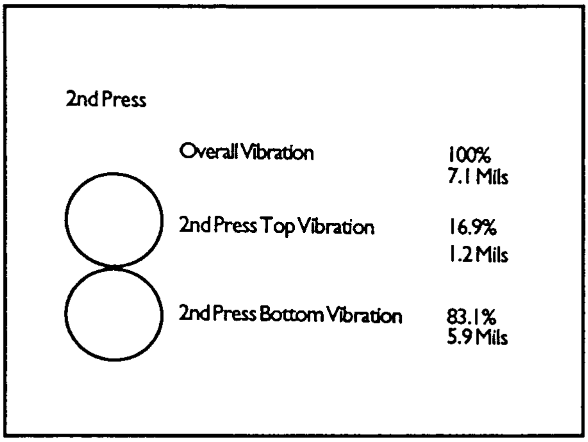

Fig. 7-24. Overall and Synchronous Data from the Second Press.

Fig. 7-27 contains one revolution of synchronous data from the Top Roll. The amplitude is about 1.2 mils, and this roll has the beginning of a serious problem: it has five bars.

Fig. 7-28 contains the polar plot from the Second Press Top Roll. The bars are quite visible. These bars may be residual from a previous grind, or they may be caused by a resonant frequency. It may be difficult to determine at this point.

Fig. 7-29 contains a synchronous time signal from the Bottom Press Roll. The peak-to-peak amplitude is about 5.9 mils and indicates a serious problem.

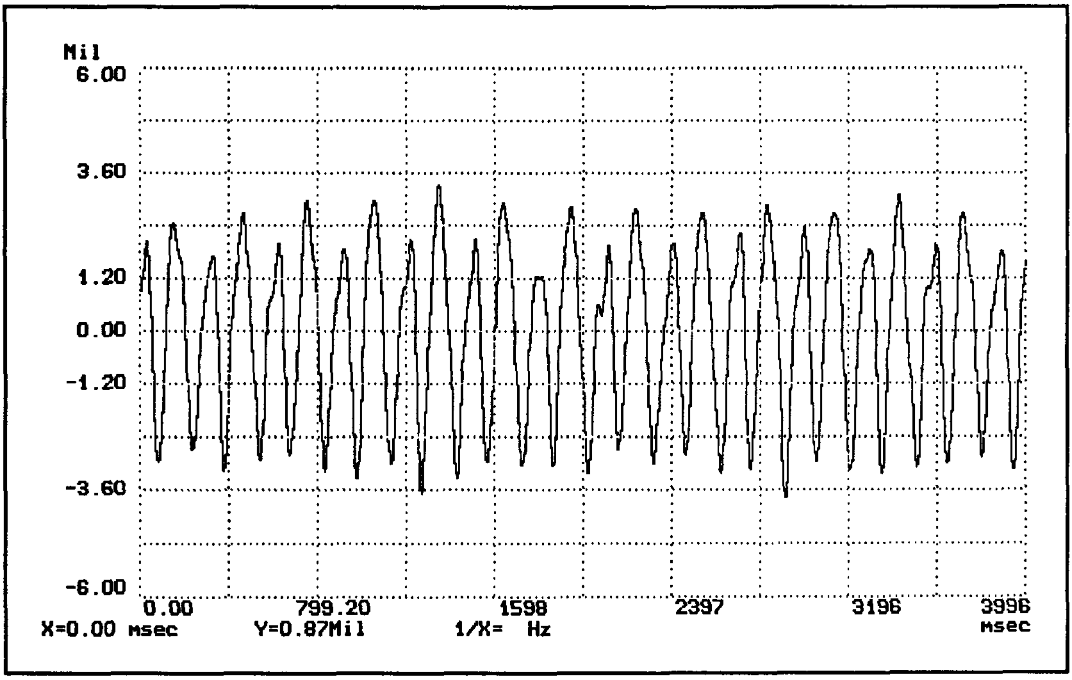

Fig. 7-25. Overall Data from the Second Press.

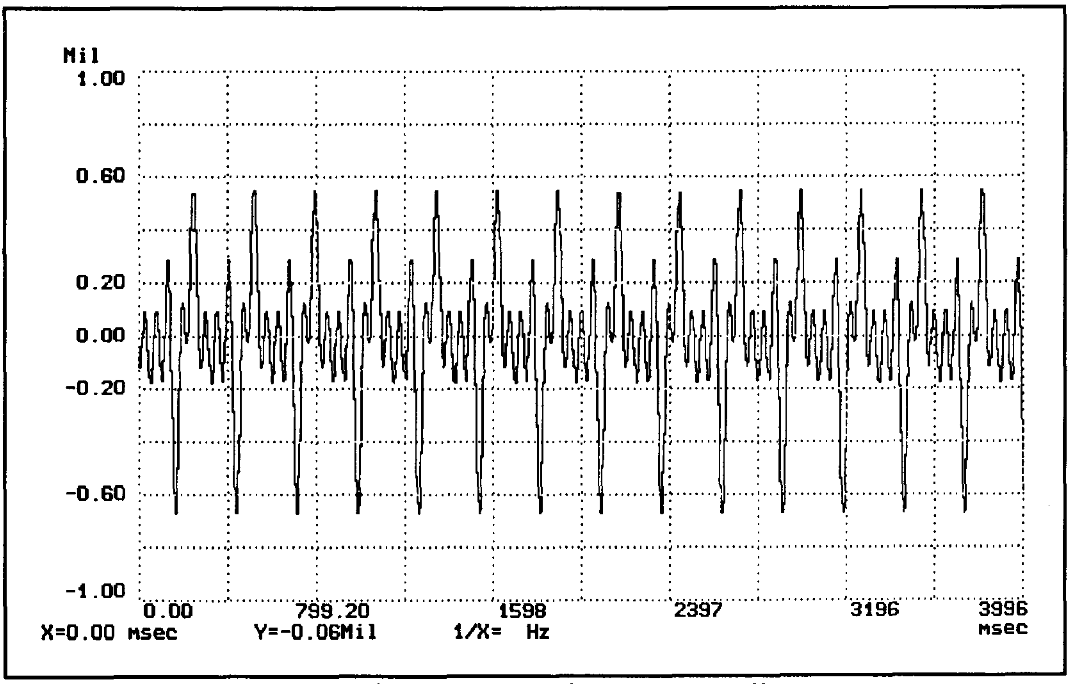

Fig. 7-26. Synchronous Data from the Second Press Top Roll.

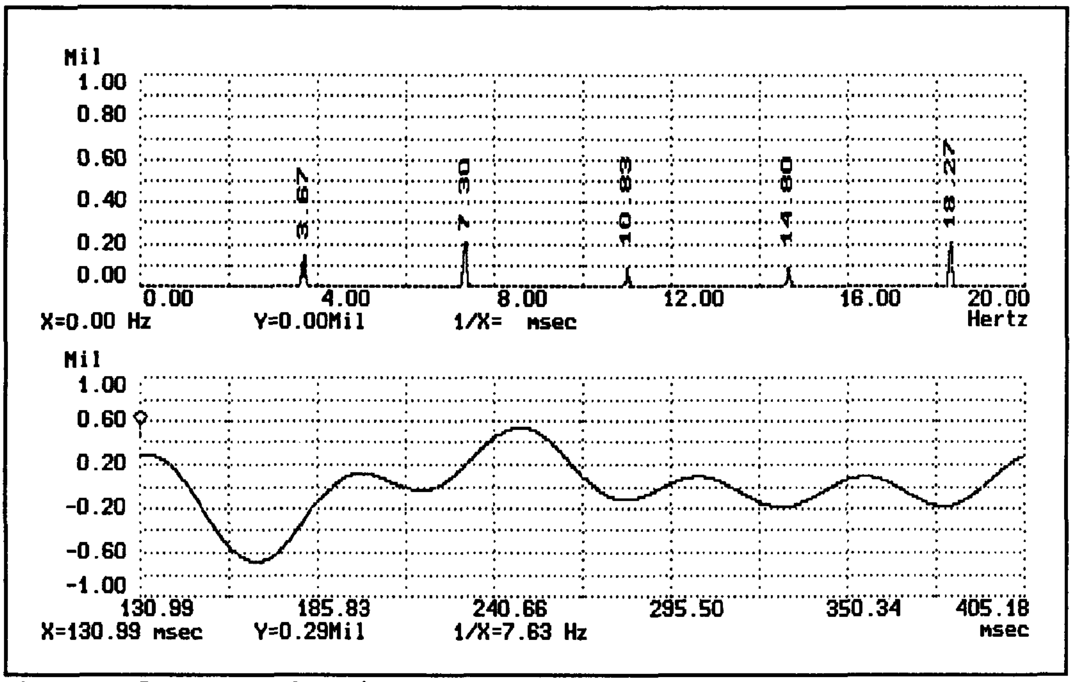

Fig. 7-27. One Revolution of the Top Roll.

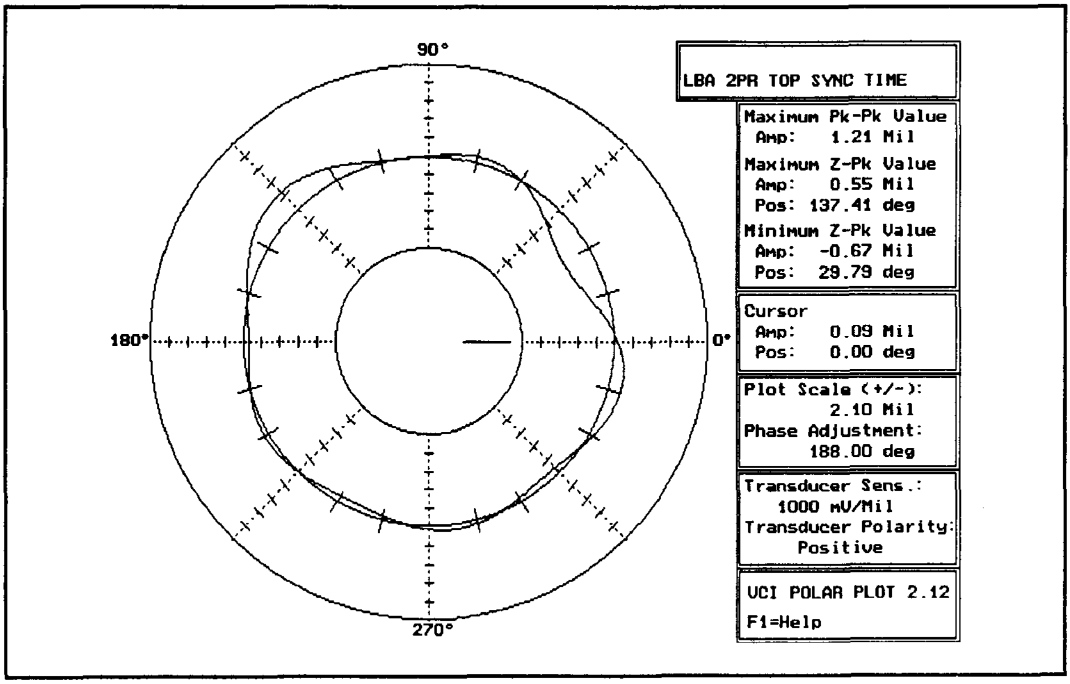

Fig. 7-28. Top Press Polar Plot.

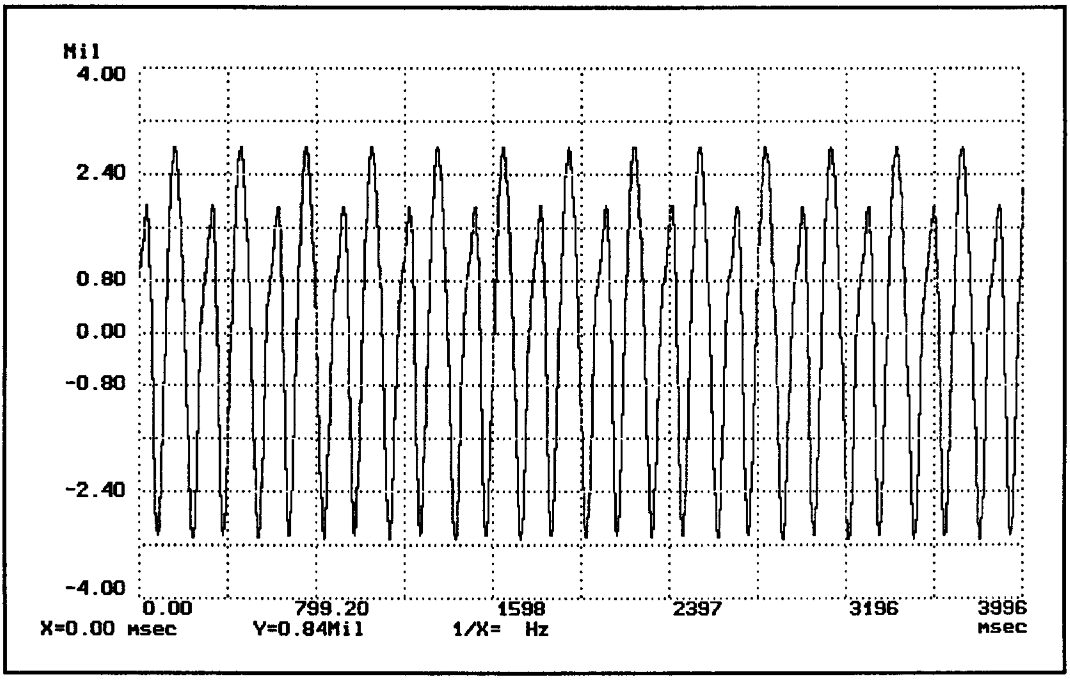

Fig. 7-29. Synchronous Data from the Second Press Bottom Roll.

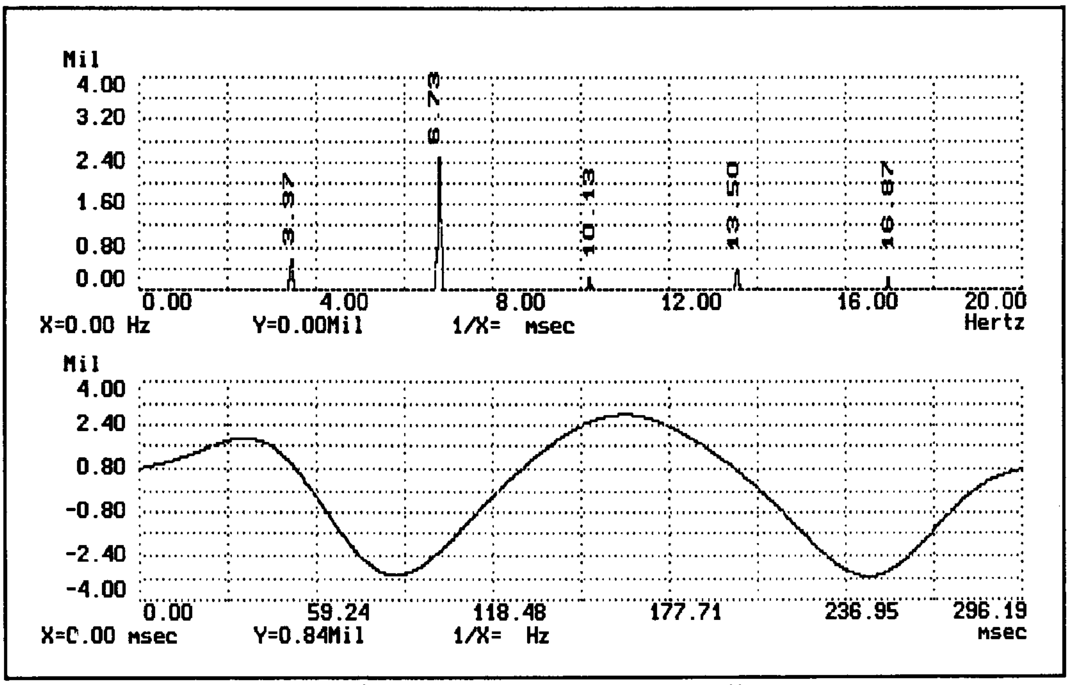

Fig. 7-30. One Revolution of the Second Press Bottom Roll.

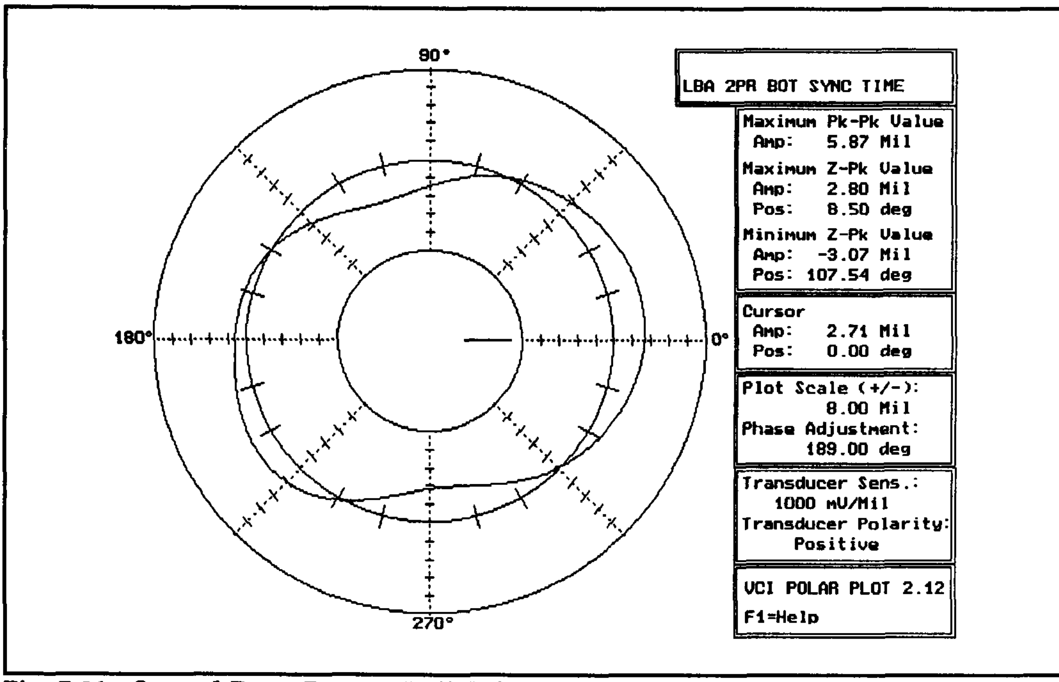

Fig. 7-31. Second Press Bottom Roll Polar Plot.

Fig. 7-30 contains one revolution of the Bottom Press Roll. This signal has two high places. The Fourier analysis of this signal would yield a spectral line at the first and second harmonic of roll speed. However, the critical nature of this problem cannot be recognized without the polar plot. Fig. 7-31 contains the polar plot of the Bottom Roll. The flat sides of the roll appear to be bending in, or concave. The exact cause of this problem is not clear. However, the two sides of the roll may be weak and may be depressing as the weak sides roll through the nip. The roll cover may not be the same hardness and may be compressing more on the soft sides. The roll could have been ground egg shaped. The grinder operator should have identified the problem during the grind. The following discussion may be helpful.

It is an established fact that a few large bearings have reached the mills with an oversized roller in the bearing. If these bearings are not inspected and such a bearing is installed on a roll during grinding, the oversized roller can pass through the load zone (area of contact between the rollers and inner or outer race) every two and one half revolutions of the roll. If the roller is 0.001 inch oversized, more material can be ground off on two places on the roll, about 180 degrees apart. For explanation purposes, assume 0.001 inch is ground off the roll at about 90 degrees from the load zone. Then, when the roll is installed, the two flat places cause a total of 0.002 inch movement. When the roller goes through the load zone every two and one half revolutions, the high places on the roll can move it another 0.002 inch for a conceivable total of 0.004 inch relative motion between the two rolls.

Once again, the need is established to measure rolls before they are installed, after installation at startup, on a monthly basis, and as other conditions indicate.

The following data is required for the next complex example:

1. Machine speed is about 3610 feet per minute.

2. Felt speed is about 0.62 Hz or 37 RPM, and felt length is about 95.3 feet or 1144 inches.

3. Yankee speed is 1.28 Hz or 76.8 RPM and Yankee circumference is about 46.9 feet or 563.2 inches.

4. Second Press speed is 8.2 Hz or 492 RPM and second press circumference is about 7.33 feet or 88 inches.

First, calculate roll and felt ratios by factoring:

Yankee circumference

$$= 563.2 \times 10 = 5632 = 2 \times 2 \times 2 \times 2 \times 2 \times 2 \times 2 \times 2 \times 2 \times 11$$

Second Press circumference

$$= 88 \times 10 = 880 = 2 \times 2 \times 2 \times 2 \times 5 \times 11$$

Felt

$$= 1144.0 \times 10 = 11{,}440 = 2 \times 2 \times 2 \times 2 \times 5 \times 11 \times 13$$

Each of these numbers has a highest common factor of:

$$2 \times 2 \times 2 \times 2 \times 11 = \frac{176}{10} = 17.6$$

The Yankee has an uncommon factor of 32 and the Second Press has an uncommon factor of 5. The felt has an uncommon factor of 65.

Now let's examine the data presented in Figs. 7-32 through 7-39. Fig. 7-32 contains the overall relative motion between the Yankee and the Second Press.

Fig. 7-33 contains an overall frequency spectrum from 0 to 50 Hz.

Fig. 7-34 contains an overall frequency spectrum from 0 to 20 Hz.

Fig. 7-35 contains an overall frequency spectrum from 34 to 44 Hz. These spectra are complex. They contain the above frequencies (Figs. 7-32 through 7-35), and harmonics. The nature and cause of the problem cannot be determined without synchronous time averaging.

Fig. 7-36 contains the synchronous time averaged signal and frequency spectrum. This data indicates the Second Press has five bars. This problem could not have been identified from overall data.

Fig. 7-37 contains the polar plot for the Second Press.

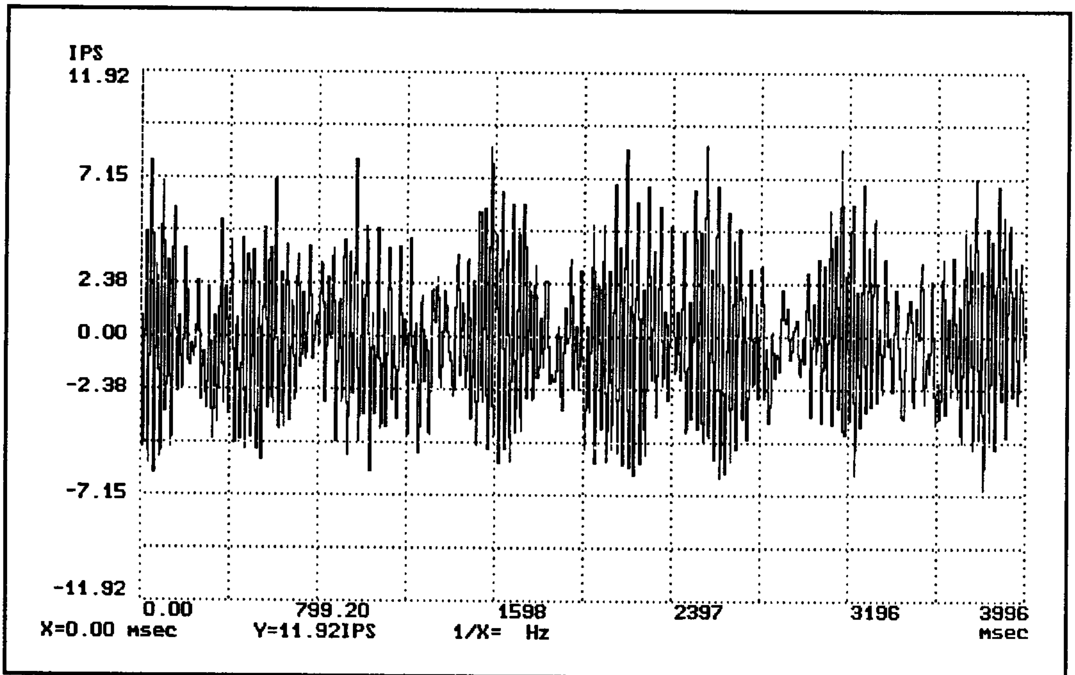

Fig. 7-32. Overall Relative Motion between Yankee and Second Press.

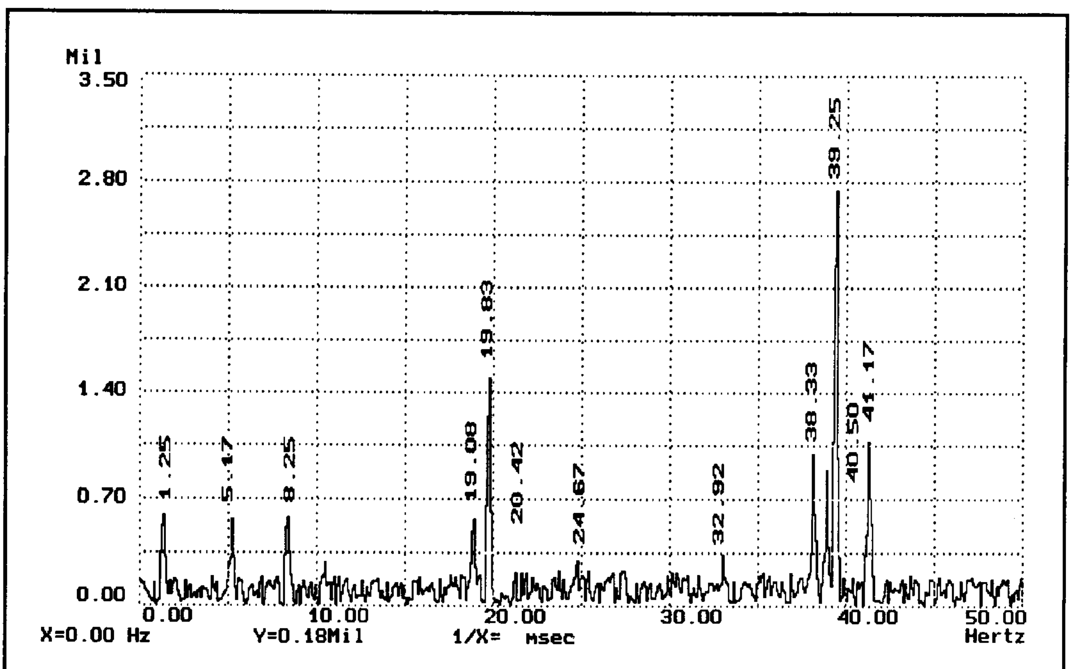

Fig. 7-33. Overall Frequency Spectrum From 0 to 50 Hz.

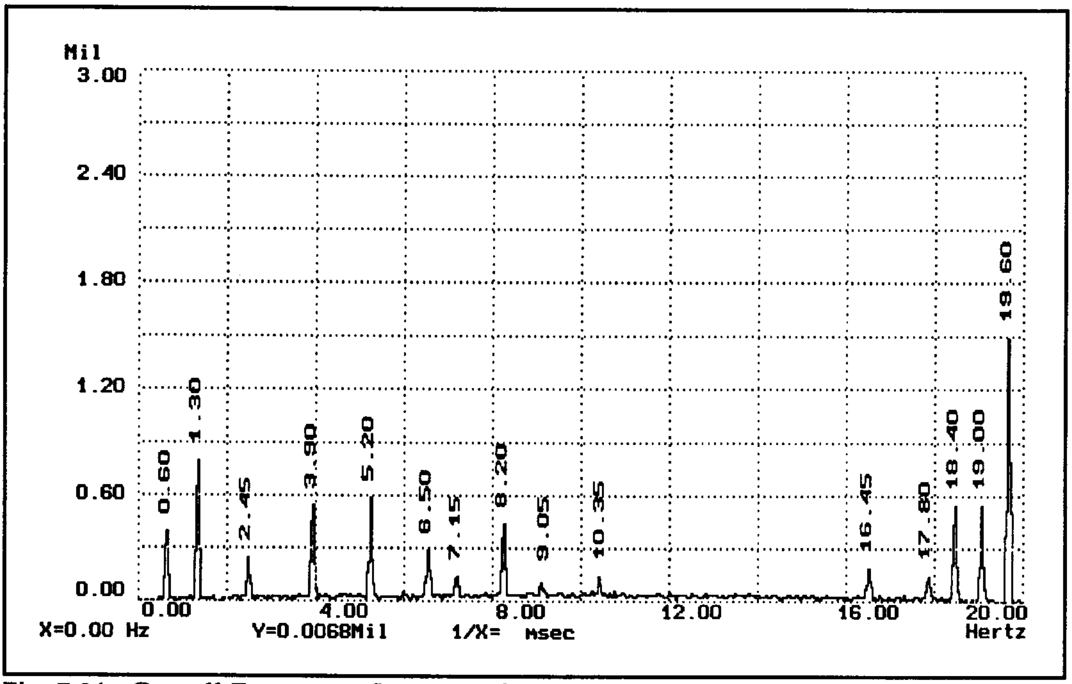

Fig. 7-34. Overall Frequency Spectrum from 0 to 20 Hz.

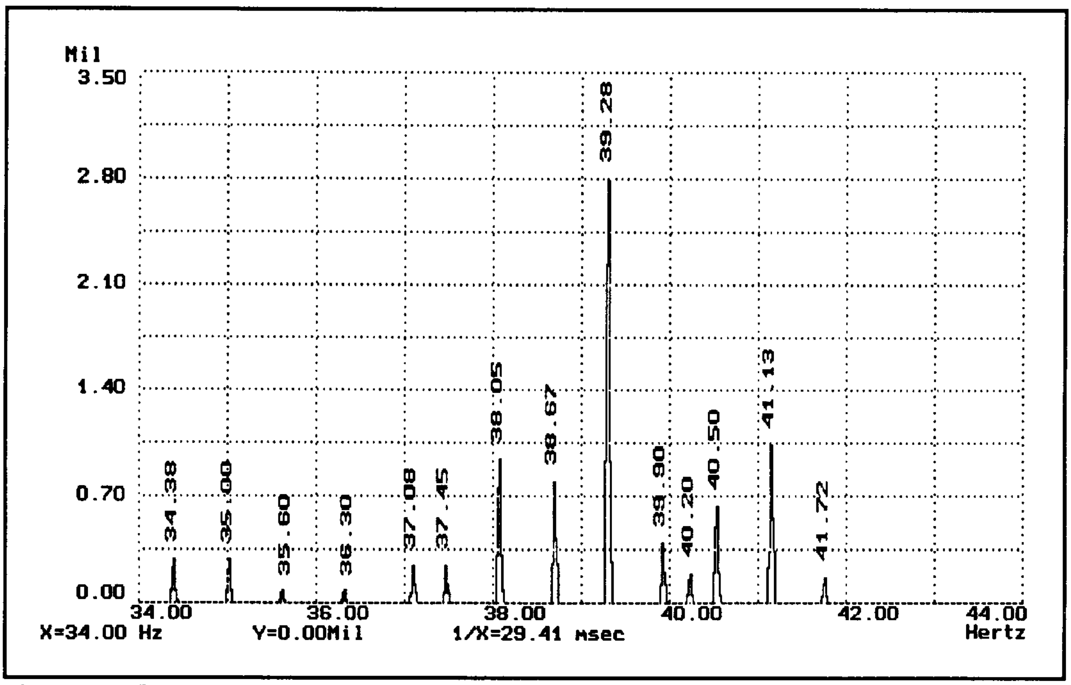

Fig. 7-35. Overall Frequency Spectrum from 34 - 44 Hz.

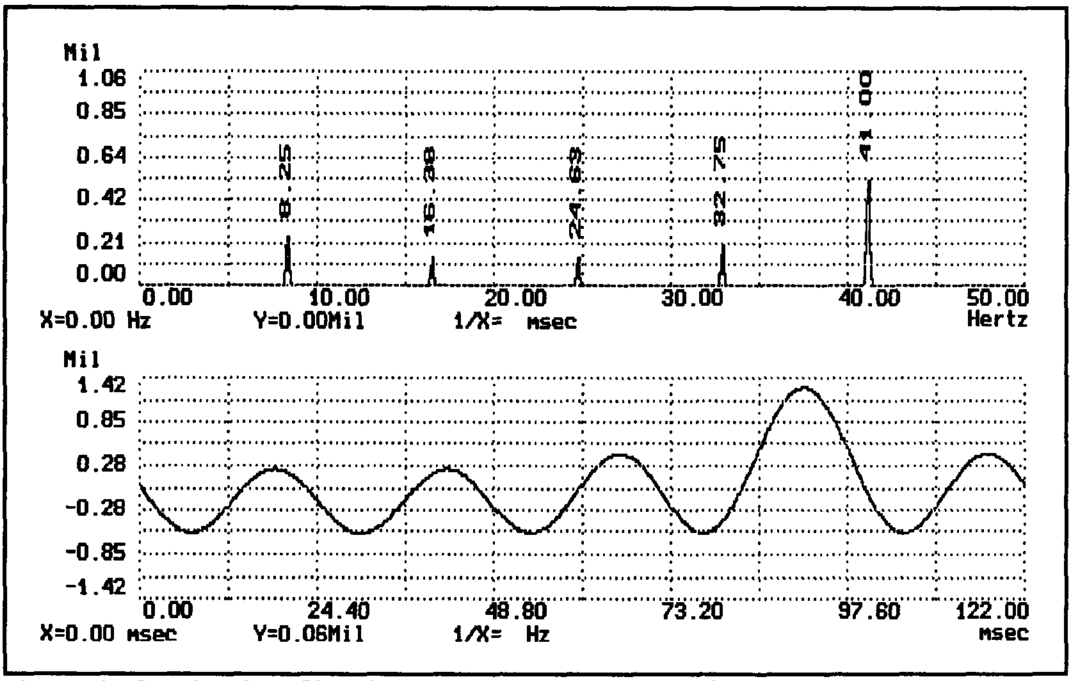

Fig. 7-36. Synch. Time Signal and Frequency Spectrum from Second Press.

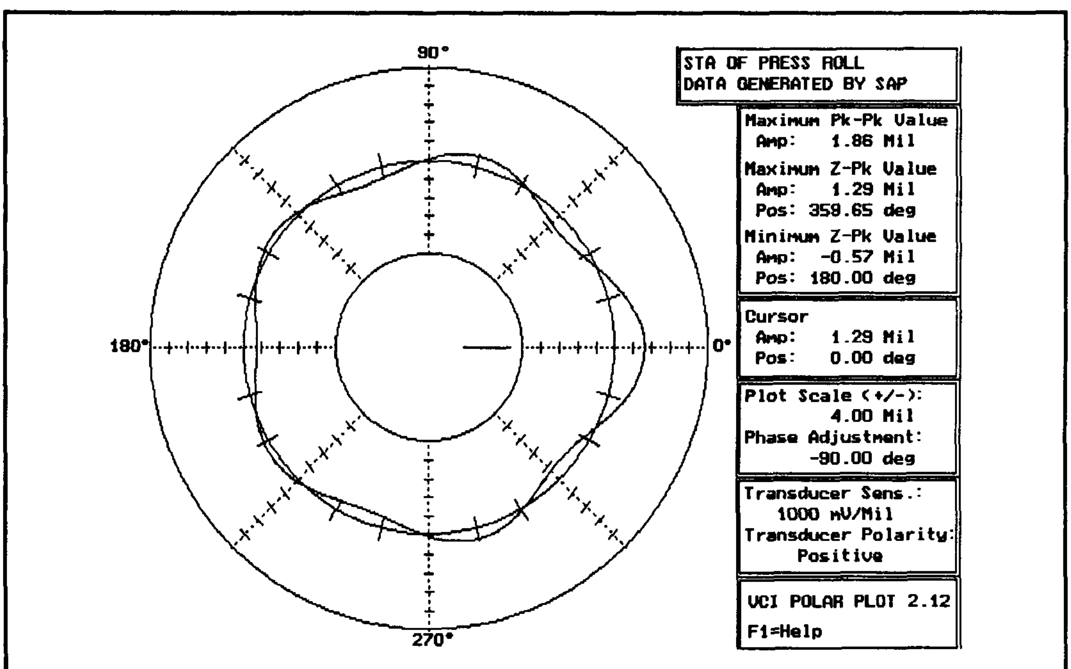

Fig. 7-37. Polar Plot for the Second Press.

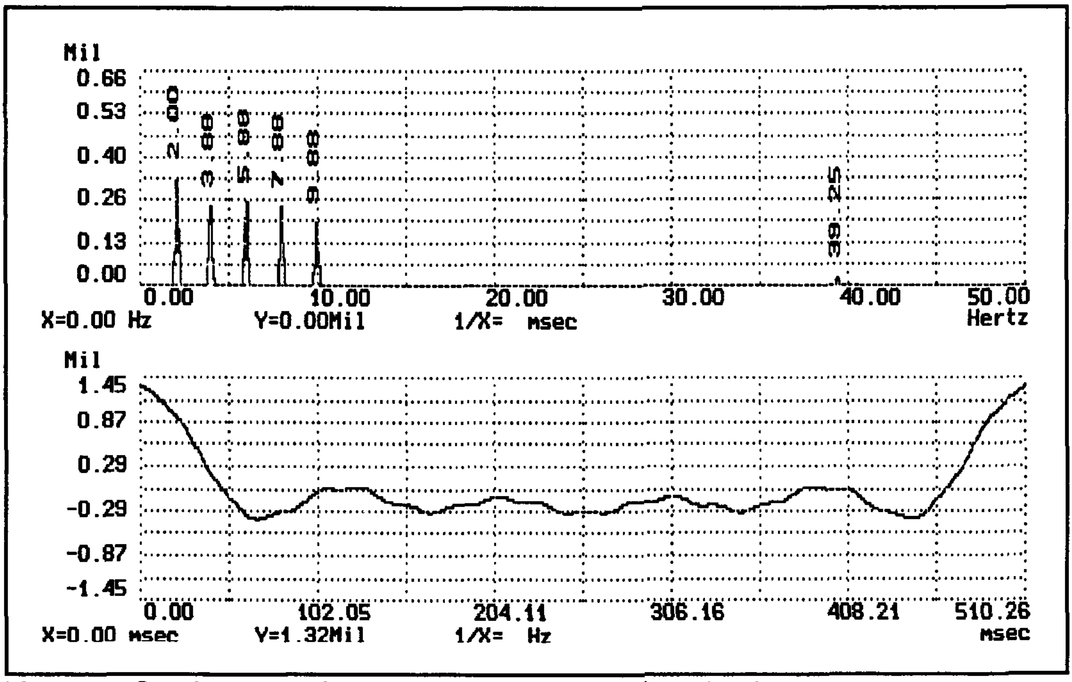

Fig. 7-38. Synchronous Time and Frequency Data from Yankee.

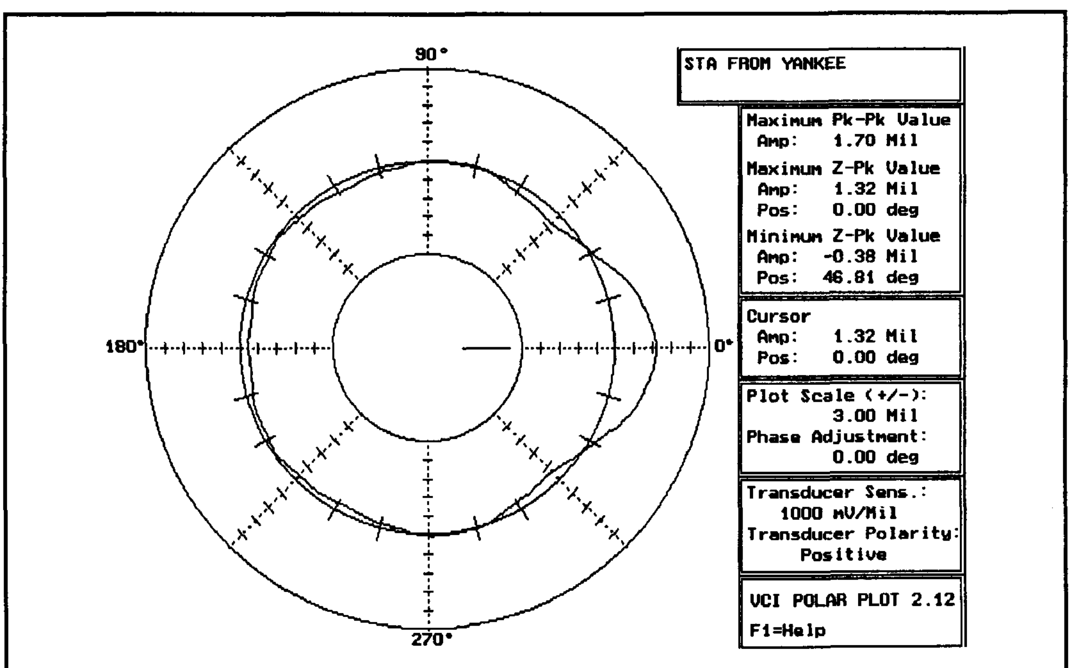

Fig. 7-39. Polar Plot from Yankee.

Fig. 7-38 contains the synchronous time averaged signal and spectrum for the Yankee Dryer. The problem is not clearly defined.

Fig. 7-39 contains the polar plot for the Yankee. The polar plot clearly identifies the problem as a high place on the Yankee. The time signal indicates it is about 1.3 mils high, zero-to-peak. This is somewhat startling, particularly in view of various vibration standards.

The previous data indicate the following problems:

1. The Yankee has one high place about 1.3 mils high.

2. The Second Press has five bars. One of the bars could be about two mils high.

3. The overall data indicate a serious felt problem.

Now let's compare the results of the data collected with the results of our calculations. Factoring showed that the Yankee, the Second Press Roll, and the felt have a common factor of 17.6. This implies that the distance between the bars is 17.6 inches. This indicates that the Yankee will get 32 bars, 17.6 inches apart. This is very unlikely, due to the construction of the Yankee. The Second Press has an uncommon factor of 5, which means it could get 5 bars, 17.6 inches apart. The felt has an uncommon factor of 65, implying that it has 65 bars.

The Second Press Roll and the felt have a common factor of 88, which is the size of the Second Press. This is a serious situation and can cause the felt to fail very quickly. A general rule to remember is: the higher the common factor, the faster the failure rate.

The insidious nature of the problem and the rapid failure rate require some explanation. It is insidious in the beginning because the high place on the Yankee is only about 1.3 mils. This might not be considered or recognized as a problem. The deterioration starts very slowly because the Yankee rotates 563.2 inches before the high place hits the felt and the Press Roll. During this time, the Press Roll has made six revolutions plus 35.2 inches. Please note that 35.2 inches = 2 x 17.6 inches. In the next revolution of the Yankee, the felt has moved 2 x 563.2 inches = 1126.4 inches, which is 17.6 inches short of the felt length. During this time, the press has made another 6.4 revolutions or a total of 1126.4 inches, which is 17.6 inches from the starting point. This process continues until the Press Roll becomes barred. Then the barring of the felt is accelerated because the Press Roll is barring the felt every 17.6 inches. This is a constant pounding every 17.6 inches. Meanwhile, the problem worsens because of the high place on the Yankee. Therefore, it is easy to visualize the slow insidious beginning and then the rapid acceleration of the failure.

This example is one of the most severe because all three components have a common factor. As the common factor becomes higher, failure becomes more rapid.

Some liberties have been taken in rounding some numbers because machine speed cannot be measured accurately. Also, the exact size of the rolls cannot be determined. For these reasons, the **Roll Ratio Program** must be used, along with the **Rusch Chart**. When these tools are used, the variables can be determined and plotted on the Rusch Chart for more precise analysis.

The last two examples demonstrate the importance of measuring relative motion between the rolls and performing synchronous time averaging. In some instances, the measurements described above are not adequate, and dynamic measurements must be made. Consider the following example:

Figs. 7-40 through 7-44 show data from a Yankee and Pressure Roll. Fig. 7-40 shows relative motion between the Yankee and Pressure Roll. The overall vibration is about 0.8 mils. Most of the vibration is caused by the Yankee and Pressure Roll. Very little vibration is caused by the felt. Fig. 7-41 contains synchronous time averaged data from the Yankee. The total vibration is about 0.8 mils. Fig. 7-42 shows the polar plot of the Yankee and indicates that it is slightly out-of-round. Figs. 7-43 and 7-44 show synchronous time averaged data and polar plot, respectively from the Pressure Roll. The total vibration is about 0.5 mils and the data indicate that the Pressure Roll is slightly out-of-round.

Figs. 7-45 through 7-48 show dynamic measurements taken on the Yankee at two different locations. Figs. 7-45 and 7-46 show measurements taken on the front side of the Yankee, along with the polar plot. The peak-to-peak vibration is 2.0 mils (up from 0.8 mils obtained from previous measurements) and the presence of bars is more evident. Figs. 7-47 and 7-48 show measurements taken at a position down the length of the roll close to the center. The peak-to-peak vibration is 6.8 mils and the amplitudes of the bars are much higher than they were on the front side of the roll.

This example demonstrates the inadequacy of only taking overall vibration measurements. The dynamic measurement of rolls is essential if detailed information is needed, which in this instance, was not furnished by overall measurements and synchronous time

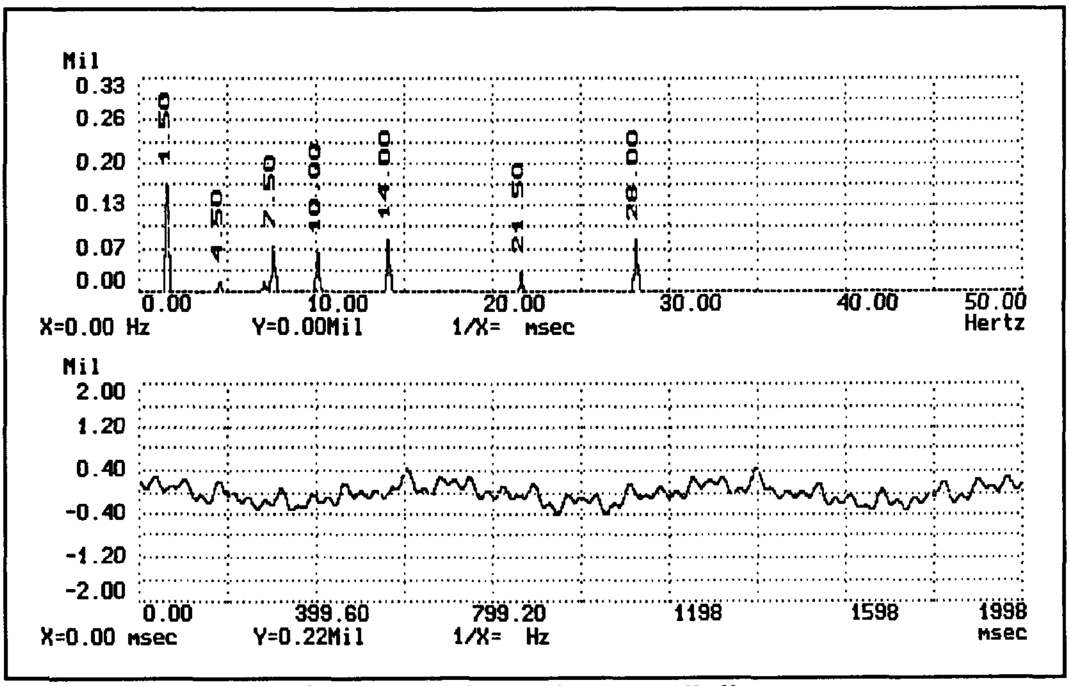

Fig. 7-40. Overall Data from the Yankee and Pressure Roll.

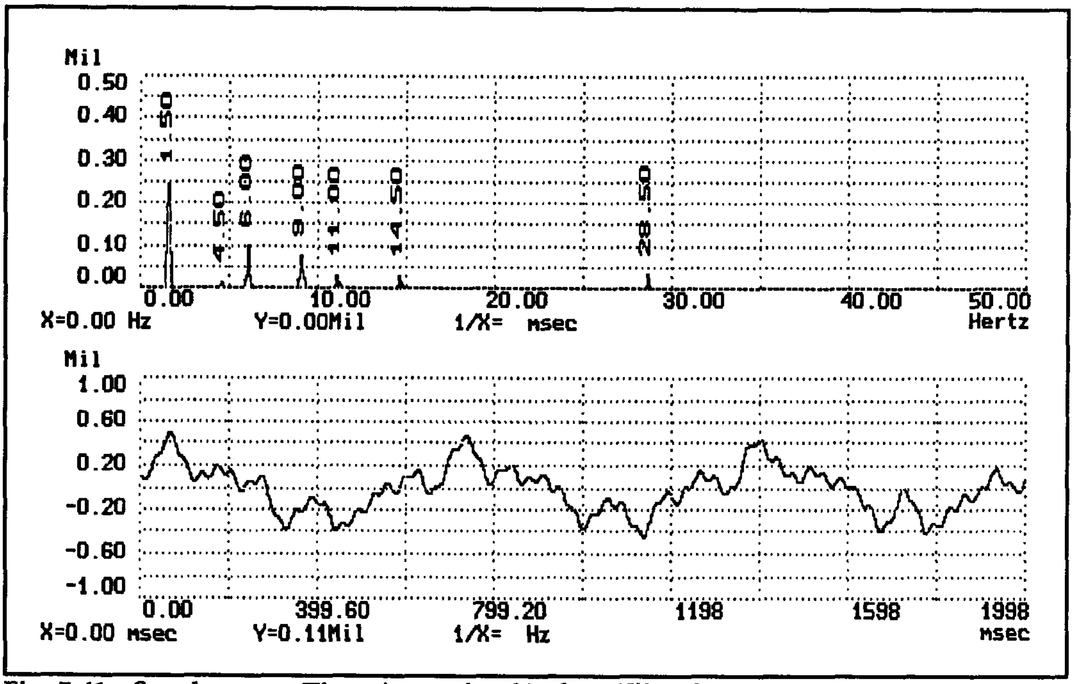

Fig. 7-41. Synchronous Time Averaging Yankee Vibration.

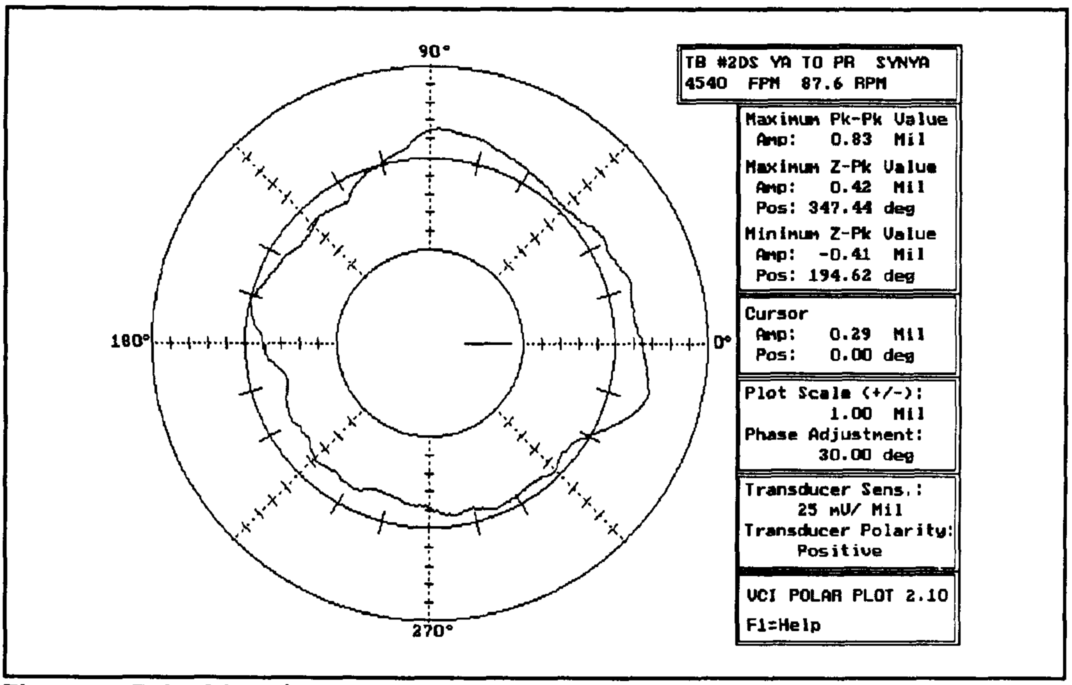

Fig. 7-42. Polar Plot of One Revolution of the Yankee.

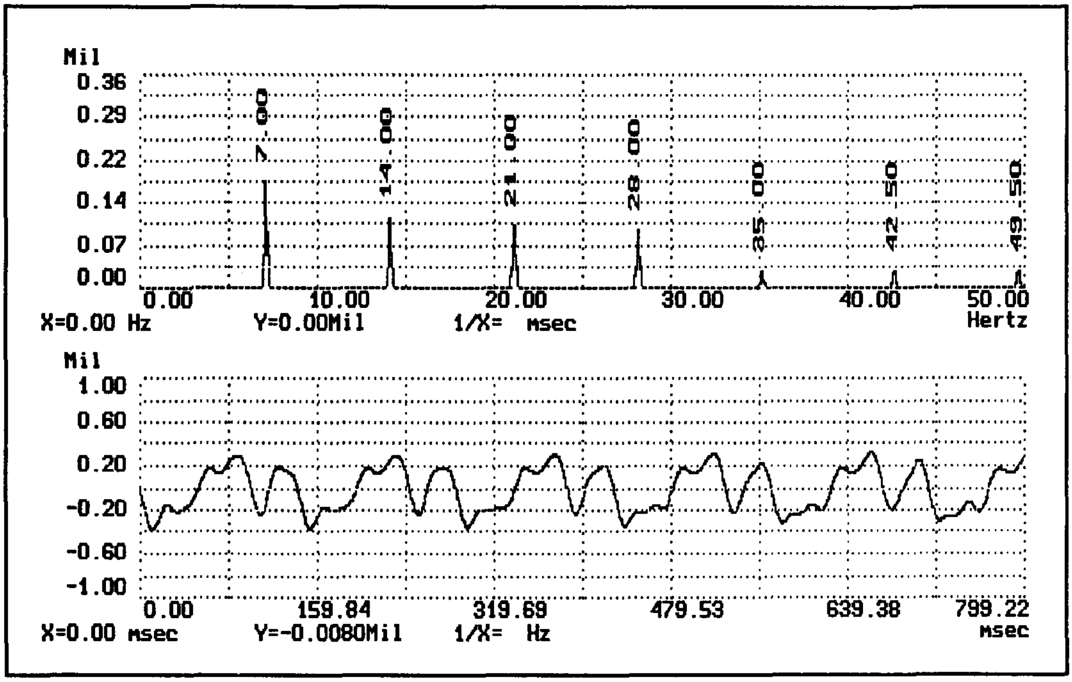

Fig. 7-43. Synchronous Time Averaging Pressure Roll Vibration.

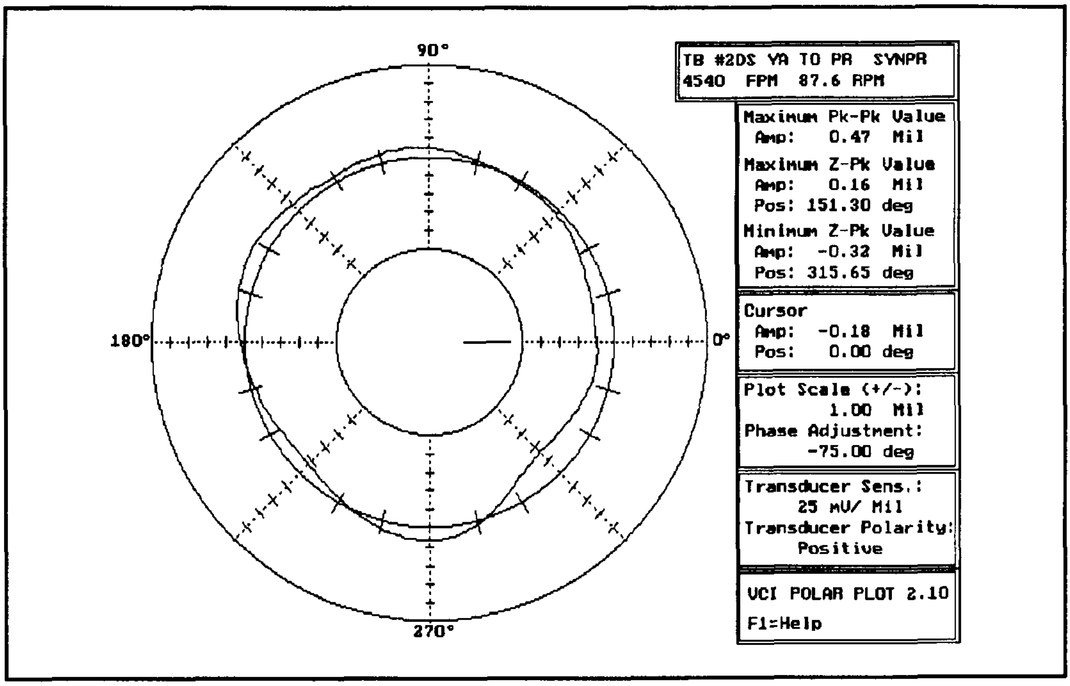

Fig. 7-44. Pressure Roll Polar Plot.

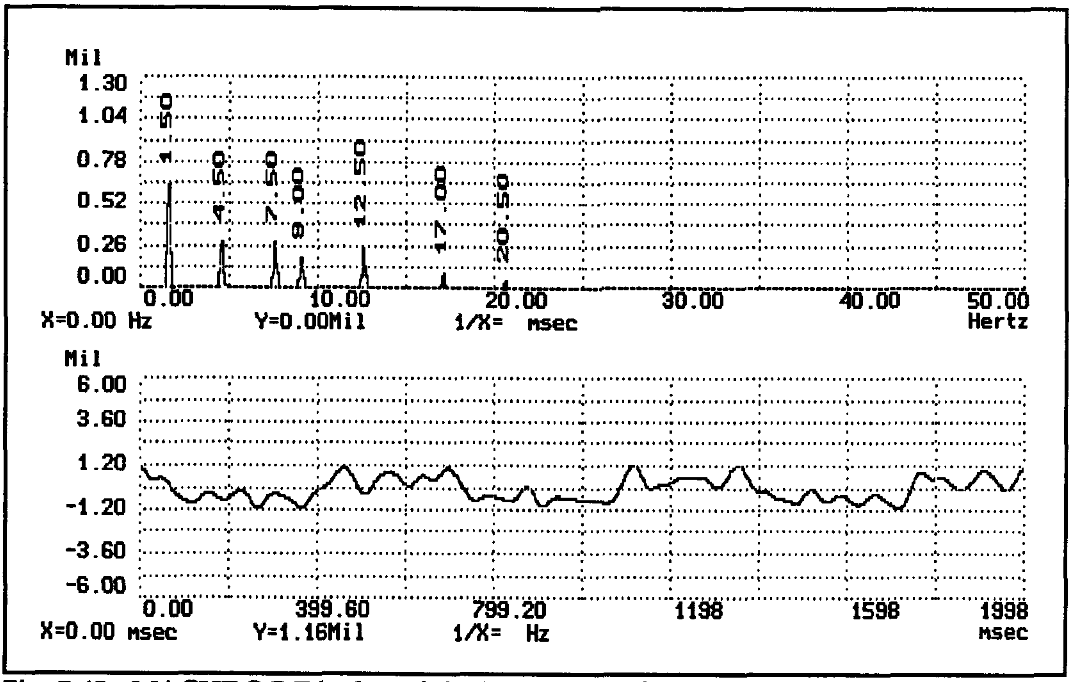

Fig. 7-45. MACHDOC Display of the Front Side of the Yankee.

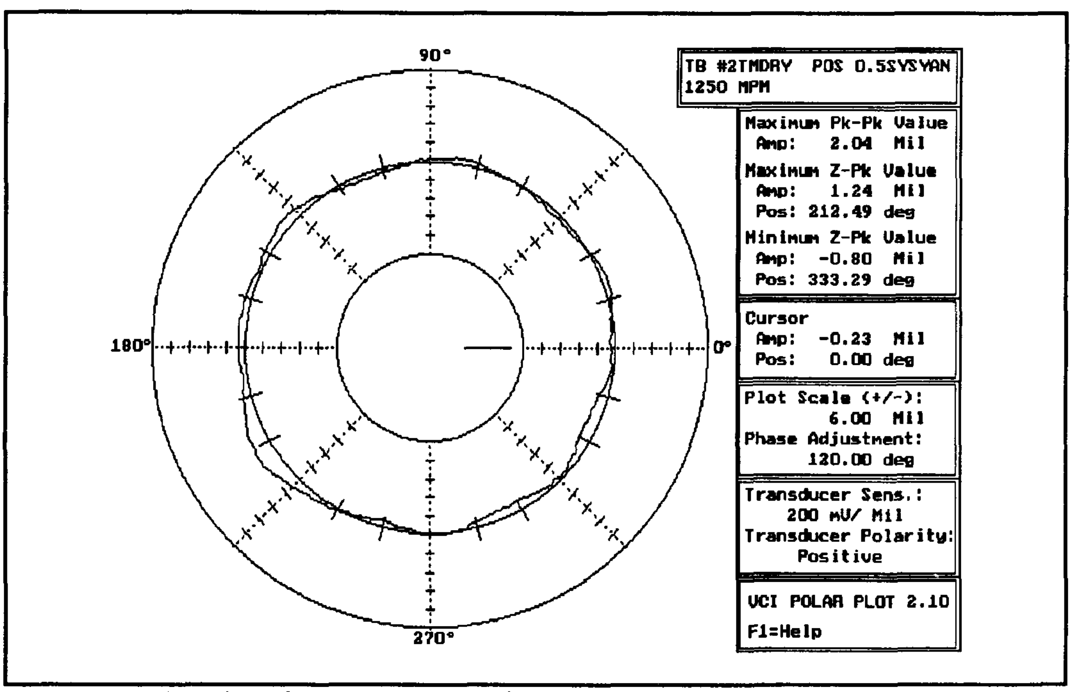

Fig. 7-46. Polar Plot of the Front Side of the Yankee.

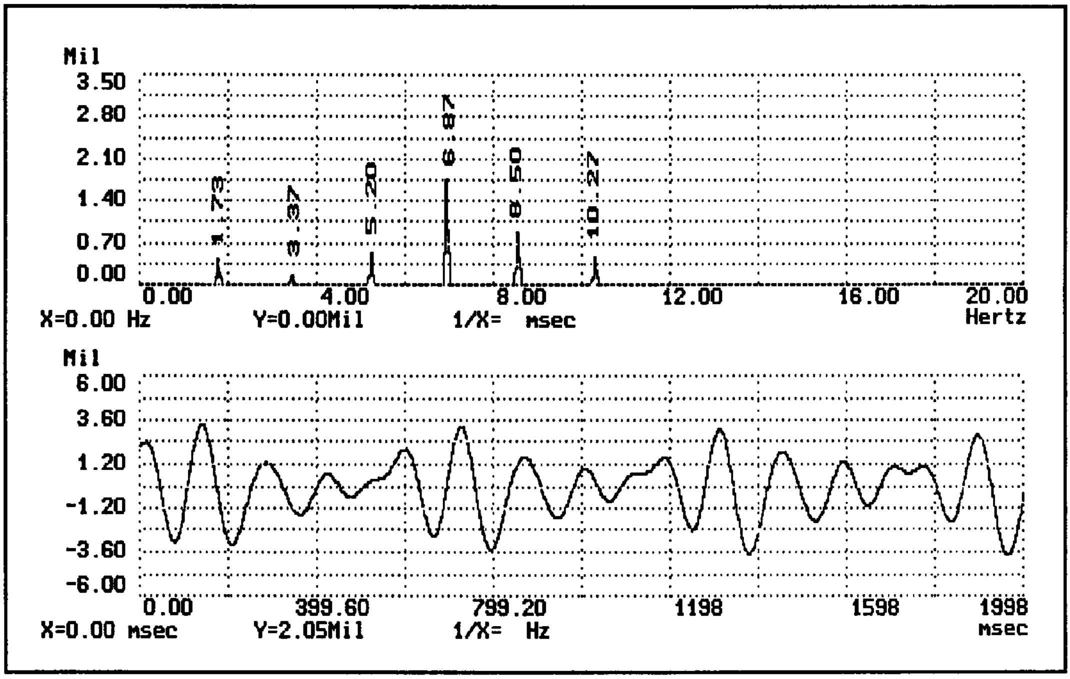

Fig. 7-47. Data Taken on Length of Roll Close to the Center.

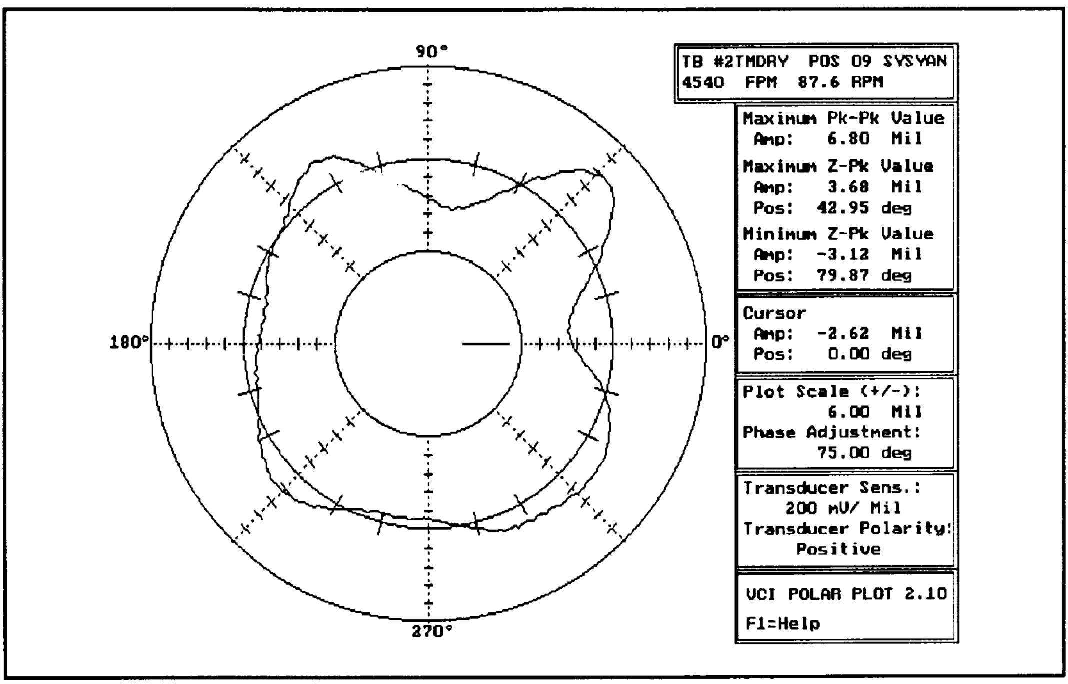

Fig. 7-48. Data Taken on Length of Roll Close to the Center.

averaging alone. Dynamic measurements result in a detailed roll profile at a particular location on the roll giving true amplitude of the bars (if any are present) and true value of total indicated runout, whereas synchronous time averaging yields only an "average" of roll profile. Effects of pressure or absence of the sheet and being in and out of nip with the pressure roll on the Yankee surface may be studied using dynamic measurements. Data accumulated by carrying out these measurements reveal the true severity of the problem and allow for intelligent decision making to correct the problem.

CONCLUSIONS AND RECOMMENDATIONS

After studying the facts presented in this chapter, it should be noted that rolls in nip with other rolls may suffer from some of the problems discussed. In order to correct these problems, knowledge of vibration theory and utilization of proper tools are essential. It is, however, just as important to inspect these rolls for roundness and other such parameters before they go into operation. The **Roll Quality Assurance Program (RQAP)** presents an accurate representation of the surface of a roll and generates detailed reports of its diameter, cross section, and crown at various locations along the length of the roll. This is a superb tool for preinstallation roll analysis, as it provides valuable information as to how much a roll should be ground before it is installed. Improperly ground rolls can cause very high maintenance expenses because they not only damage themselves, but also the rolls with which they are in nip.

The **Polar Plot Program** is an excellent tool for obtaining accurate cross-sectional views of a roll while it is in operation. This program provides vital information about vibration generated by each roll in nip: roundness of the rolls, number of corrugations, and the depth of the corrugations on each roll.

Finally, the **Roll Ratio Program** and correlating **Rusch Chart** provide important information about rolls in nip, such as whether rolls with certain diameter combinations should be used. These programs also furnish information about how much rolls in nip must be ground to avoid barring problems, etc.

The conclusion in this process is to run a predictive maintenance program with maximum efficiency and as economically as possible. These tools aid in achieving these goals and provide valuable information in the decision making process.

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[2] M. J. Drosjack and D. R. Houser, "An Experimental and Theoretical Study of the Effects of Simulated Pitch Line Pitting on the Vibration of a Geared System." ASME (pub. 77-DET-123).

[3] R. J. Drago, Fundamentals of Gear Design. Published by Butterworth Publishers, 1988.

GLOSSARY

A

AC (Alternating Current) An electric current that reverses direction in a circuit at regular intervals.

Acceleration The rate of change of velocity with respect to time.

Accelerometer Any of various devices used to measure acceleration.

AC-Coupling A type of input that blocks the DC portion of the signal.

AC voltage measurements Peak-to-peak is the value of the total swing of the wave. Peak is one-half of peak-to-peak. Average is 0.637 x peak. RMS is 0.707 x peak. These values hold true for a true sine wave only. The more the signal deviates from a true sine wave, the greater the error in these calculations.

AGMA gear quality number An established gear quality rating system for specifying gear quality numbers for different pitch line velocities (in feet/minute).

Amplitude The maximum value of a periodically varying quantity.

Analyzer The hardware unit used to analyze data.

Analytical technique Solution of shock and vibration problems using mathematical analysis.

Angular Velocity A vector quantity describing rotational motion, the magnitude of which is the time rate of change of angle and the direction of which is along the axis of rotation.

Auto Abbreviation used for automatic.

Average The arithmetic mean, as determined by the summation of the x's over n.

$$\Sigma \frac{x}{n}$$

B

Balls The balls in a bearing made from hardened metal alloys, sometimes used interchangeably with rollers when calculating bearing frequencies.

Ball Pass Frequency The frequency balls or rollers pass over a single point on the inner or outer raceway of the bearing.

Ball Pass Frequency of the Inner Race (BPFI) The frequency the balls or rollers pass over a single point on the inner raceway of a bearing.

Ball Pass Frequency of the Outer Race (BPFO) The frequency the balls or rollers pass over a single point on the outer raceway of a bearing.

Ball Spin Frequency (BSF) The number of revolutions per second made by a ball or roller in an antifriction bearing.

Bandwidth refers to the overall range of frequencies, also refers to the range of frequencies between the half power points.

Band-pass filter An electronic device used to filter all signals in a predetermined frequency range.

Baseline Data The first or initial vibration data taken from a machine system.

Beat The process of two frequencies adding and subtracting as the signals go in and out of phase.

Bump Test Measures the response of a component, machine, or structure when enough energy is introduced to excite the natural frequencies.

C

Calibration The process of measuring the accuracy of an instrument.

Cavitation When a pump is not operating at the correct point on the pump curve resulting in restricted suction intake, and the liquid tends to vaporize while coming off the impeller.

Circumferential Crack A crack around the circumference of a roll.

Clipping The truncation or flattening of the positive and/or negative portions of the signal, normally caused by overloading electronic circuits and machinery problems.

Coastdown Data Starts the process of measuring the amplitude of vibration at all speeds from zero to operating speed.

Contacting Displacement Transducer Also referred to as a LVDT (Linear Variable Differential Transformer). A transducer that is used to measure relative motion or displacement in the frequency range of 0 to 200 Hz.

Crest factor One parameter used to describe the dynamic range of a voltmeter's amplifiers. Refers to a wave form and is the ratio of the peak to the RMS voltage with the dc component removed. The crest factor of a square wave is 1, while that for a sine wave is 1.414. A pulse can have a crest factor of more than 9.

Critical Damping The minimum viscous damping that allows a displaced system to return to its initial position.

$$CF = \frac{P - DC}{RMS}$$

Critical speed Rotor speed at which local maximum amplitude occurs. A critical speed occurs whenever the rotor speed coincides with a system's natural frequency.

Cycle A time interval in which a characteristic, especially a regularly repeated, event or sequence of events occurs.

D

Damping A factor that causes a resonance or oscillation to decay over time.

Data Acquisition The collection and processing of data.

DC (direct current) A unidirectional current in which the changes in value are either zero or are so small that they may be ignored.

Decibel (dB) Unit for measuring relative loudness of sound and electronic signals. The logarithmic expression of ratios, equal to one-tenth of a Bell. The voltage ratio is:

$$20 \log \left(\frac{voltage\ output}{voltage\ input}\right)$$

The power ratio is:

$$10 \log \left(\frac{power\ output}{power\ input}\right)$$

where zero dB is always equal to one millivolt or one milliwatt.

DC Coupling Uses a choke input that blocks the ac component and passes the dc component.

Degrees of freedom The minimum number of independent coordinates required to define completely the positions of all parts of a system at any instant of time.

Diameter The width of a circular object.

Disk A wheel, usually solid and axially slim, on which mechanical work is performed, or from which work is extracted. Examples: Turbine disk, compressor wheel.

Displacement Refers to the measurement of the distance an object moves.

Duty cycle The pulse width divided by the pulse recurrence frequency or repetition rate, used in calculating the crest factor.

Dynamic A state in which one or more quantities exhibits appreciable change within an arbitrarily short time interval.

Dynamic Measurement An accurate measurement of a component while all the operational forces are applied. i.e. measuring the roundness of an operating roll or shaft.

Dynamic Shape The resulting shape with all or some of the operational forces applied.

E

Eccentric Gear A gear that does not conform to AGMA Quality Gear Standards.

Eccentric Roll A roll that is not round.

Empirical technique The measured response of a known defect.

Engineering Units (EU) General term for the type of units used: G's, mils, IPS, etc. Used when the type of units does not matter; for explanation or description.

Excited Frequencies Natural or resonant frequencies excited by some source.

Expanded Mode When only a portion of a signal is displayed on the grid, it is said to be expanded.

F

Factor An integer that can be divided evenly into the number of interest and the quotient is an integer.

Fast Fourier Transform A numerical calculation performed on a digitized time signal that creates a limited Fourier series of cosines and phase angles of each cosine.

Fatigue Failure Failure of a mechanical component or system as a result of repeated stress cycles.

Felt A fabric that supports the paper in a paper machine to assist the paper making process.

Fiber Optic Sensor A device that uses a fiber optic light emitter and a fiber optic light sensor that can detect a difference between light and dark surfaces.

Fluting Arcing from the inner race to the outer race through the rolling elements in a bearing which creates defects on the bearing raceways.

Fractional Gearmesh Frequency A fraction of gearmesh frequency caused by eccentric gears and gears with improper ratios.

Frequency The number of cycles present in a specific time period, normally one second. Measured in Hertz, which stands for cycles per second. Frequency is usually represented by the letter "F." The time period is represented by the Greek letter "tau" (τ). The formulas are:

$$F = \frac{1}{\tau} \text{ and } \tau = \frac{1}{F}$$

Frequency Domain A term used to describe the magnitudes and frequencies of sinusoidal functions that once summed together make a time domain signal.

Frequency Modulation Periodic changing of the frequency of a sinusoidal function over time.

Frequency Range A specifically designated part of the frequency spectrum.

Frequency Response The response (i.e., displacement, velocity, or acceleration) of a system described in terms of frequency.

Forcing function The excitation of the vibration; the problem that is causing the vibration.

Foundation Machine support. May be rigid or possess mass-elastic properties.

Fourier series A mathematical description of a nonharmonic periodic function using a linear combination of sine and cosine functions.

Full Scale The largest value indicated on the scale or in the case of instruments having their zero between the ends of the scale, the full-scale value.

Fundamental Frequency The first harmonics or base frequency, such as gearmesh frequency, ball pass frequency, etc.

Fundamental Train Frequency The angular velocity of the individual ball centers. Cage frequency.

G

G Unit of measure used for acceleration measurements.

Gear Life Expectancy The gear's life expectancy is reduced by the inverse of the common factor of the gears in mesh.

Gearmesh frequency The angular speed of the rotating gear times the number of teeth on the gear.

Generated Frequencies All frequencies that can be calculated and vary as a fuction of machine speed.

H

Harmonic Exact multiples of a fundamental frequency.

Harmonic analysis Analysis of a periodic complex wave form using sine and cosine functions.

Harmonic Marker Lines displayed on the data used to identify the harmonic multiples (harmonics).

Harmonic Motion The vibration motion of the projection upon a straight line of a point moving uniformly along a circumference in the same plane.

Hertz The unit of frequency, one cycle per second.

Housing A casing used to enclose a piece of machinery such as a motor or bearing.

Hunting Tooth Frequency (HTF) The reciprocal of the time period one gear makes with a tooth on the other gear a second time.

I

Imbalance A condition where there is more mass on one side of a rotating device than the other causing an outward radial force in the direction of the additional mass when the object is rotated.

IPS (inches per second) Units used when measuring vibration in velocity.

J

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

L

Lambda the greek letter λ symbol for wavelength.

M

Mesh The fitting of gears together, similar to fitting the fingers of two hands together.

Mil Unit of measure, equal to a one thousandth of a second.

Misalignment When two machines are coupled together and their axes of rotation are not aligned.

Misaligned Gears Gears that are not meshing evenly across the pitch line.

Msec An abbreviation for milliseconds or thousandths of a second.

Modal Analysis A vibration response analysis that uses a unique combination of previously determined mode shapes for its mathematical description.

Model A mathematical or experimental simulation of a component system.

Modulation The act of mixing two or more frequencies. Amplitude modulation is a time varying amplitude. Frequency modulation is a time varying frequency.

Motion Description of the displacement, velocity, or acceleration of a system as a function of time.

Mv Abbreviation for millivolts, thousandths of a Volt.

N

Natural Frequency The reciprocal of the natural period of a system.

Nip The point of contact when two rolls are rotating in contact with each other.

Noise Any disturbance, especially a random and persistent disturbance, that reduces the clarity or quality of a signal.

Noncontacting Displacement Transducer A transducer that measures motions without contacting the target, the common name is proximity probe.

O

Oil Whirl The shaft ridge on an oil wedge that whirls while it rotates; occurs in some lightly loaded, plain journal bearings.

Oscillating Gears A condition where gears move from a point of contact between teeth in a direction of advance then bounce back in the opposite direction.

Out-of-Round Rolls A roll whose center of rotation is not the geometric center of the roll.

P

p-p (peak to peak) Signal amplitude measurement, the maximum value of one cycle.

Periodic Motion Oscillatory, periodic or repeating motion.

Pitch-line Velocity The tangential velocity of a rotating gear or bearing at the pitch-line.

Planetary Gears A system of gears where a central, sun gear rotates meshing with and turning a planetary gear whose axis of rotation moves around the sun gear. The planetary gear meshes with a ring gear which surrounds the planetary gears and the sun gear.

Polar Plotting format which is circular. Measurements are in amplitude and degrees.

Proximity Probe Ferromagnetic dynamic displacement transducer. Nonconducting displacement transducer used for measuring relative motion.

Pseudo RMS Technique used by most analog meters to measure RMS. These meters multiply average by 1.11 to present RMS measurements. This works for pure sine waves only.

Pulse A transient amplification or intensification of a characteristic of a system, especially of a wave characteristic, followed by return to equilibrium or steady state.

R

Radial Load A load that is directed toward the center of the axis of rotation.

Random Motion Motion that is not repetitive in magnitude or frequency of occurrence.

Range The set of values lying between the upper and lower limits.

Real-time Pertaining to the actual time during which a physical process transpires.

Relative Motion The motion of one body with respect to another disregarding any motion relative to a third point or reference.

Relative Motion Measurement The measurement of the motion of one body with respect to another usually performed with a contacting or non-contacting displacement transducer.

Resonance The enhancement of the response of an electric or mechanical system to a periodic driving force when the driving frequency is equal to the natural undamped frequency of the system.

Resonant Frequency The frequency of the resonance.

Root Mean Square (RMS) A method of measuring the true energy under the curve. The half power point of a sinusoid or .707 times the peak value of a pure sinusoid.

$$\sqrt{\frac{(P_1^2 + P_2^2 + \ldots + P_N^2)}{N}}$$

Rotor Bars The ferromagnetic bars in the rotor of an electric motor.

RPM An abbreviation used for Revolutions Per Minute.

S

Shallow Flaking A defect in a bearing raceway where the bearing raceway is missing shallow flakes of metal.

Shock A nonperiodic excitation of a mechanical system characterized by sudden loading.

Shock absorber A device that dissipates energy in order to modify the response of a mechanical system to applied shock.

Shock isolater A resilient support that isolates a system from a shock loading.

Shock pulse A substantial disturbance characterized by rise and decay of acceleration in a short period.

Shock spectrum The maximum response (acceleration, velocity, or displacement) of a series of damped or undamped single-degree-of-freedom systems resulting from a specific shock excitation. An independent mass-spring-damper system is associated with each frequency.

Signal-to-noise Ratio The ratio of signal levels to noise level.

Sinusoidal A description for a phenomenon that follows a sine function or a cosine function.

Siren Effect A generated frequency equal to the number of bars/slots in a rotor times the rotating speed.

+slope Refers to the portion above zero in the y-axis direction.

-slope Refers to the portion below zero in the y-axis direction.

Spectra Plural of spectrum.

Spectral lines An isolated peak of intensity in a spectrum.

Spectrum The distribution of amplitude as a function of frequency.

Speed The distance covered by a point divided by the time required to cover that distance defined for an instant in time.

Square Wave A rectangular wave or near rectangular wave that rises to a positive level flat peak for a period of time then falls to a negative level flat peak for a period of time and repeats itself. Contains odd harmonics sometimes called odd fractions.

Startup Data Vibration data taken on a machine as the machine goes from zero rotational velocity up to operating speed.

Starvation This occurs in pumps when there is not enough liquid present to fill each vane on the impeller every revolution.

Stiffness The description of the elastic properties of a system given in terms of pounds force per inch of deflection.

Synchronous Time Averaging Method of time averaging to average out non-synchronous vibration.

T

Thrust Load A load on a machine that is in the axial direction.

Time Delay The time interval between the starting point of a signal and the detection of the trigger.

Time Domain The signal level with respect to time.

Transducer An electronic device that converts a mechanical vibration or motion into an electronic signal.

Transducer Sensitivity Used to transform the voltage output of a transducer to the appropriate engineering units (V/E).

Trigger A pulse or signal used to initiate data collection. A triggered sweep or delay ramp.

Truncation Signals that have the peaks cut off at some level.

Time The inverse of frequency. The continuous passage measured in seconds, minutes, hours.

V

Vane pass frequency The frequency at which the blades of a pump pass a particular point.

Velocity The speed or how fast an object is moving.

Velocity Transducer Unit used to measure the vibration velocity of an object.

Vibration The physical motion of a rotating machine.

Volt The difference of electric potential between two points of a conductor carrying a constant current of one ampere, when the power dissipated between these points is equal to one watt.

Voltage The dot product line integral of the electric field strength along its path. Measured in volts.

W

Wavelength In a periodic wave, the distance between two points of corresponding phase in consecutive cycles.

X

X-axis The horizontal direction on the grid.

X-scale The minimum and maximum scaled values used on the x-axis of the grid.

Y

Y-axis The vertical direction on the grid.

Y-scale The minimum and maximum scale values used on the y-axis of the grid.

Z

Z-p (zero to peak) Signal amplitude measurement from the zero reference to the maximum or minimum value of the signal.

Zoom Also called frequency translation. When data is collected on a range other than between zero and the bandwidth, it is called a zoom. A zoom is different from an expand. The zoom mode increases the resolution, where the expanded mode does not.