Vibration Analysis and Diagnostic Guide

Book · June 2012

Jaafar Alsalaet
University of Basrah

Syllabus

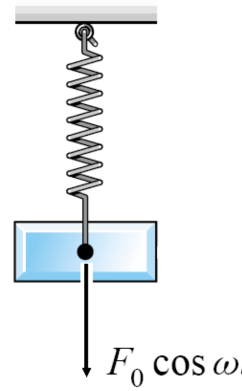
Material	No. of hours
1. Introduction 1.1 Mechanical Vibration 1.2 Machinery Preventive and Predictive Maintenance 1.3 Evolution of maintenance philosophies 1.4 Vibration analysis – a key predictive maintenance technique 1.5 Vibration Analysis and Measurement Equipment	2
2. Measuring Parameters and Vibration Severity Criteria 2.1 Oscillatory Motion 2.2 Acceleration, Velocity and Displacement 2.3 Location and Direction of Measurements 2.4 Common Vibration Severity Charts and Tables	2
3. Vibration Analysis Techniques 3.1 Definitions 3.2 Level Measurement 3.3 Time Waveform 3.4 FFT Spectrum and Phase Analysis 3.5 Orbit 3.6 Bode and Nyquist Plot 3.7 Cepstrum Analysis 3.8 Envelope Analysis	4
4. Diagnosis of Common Vibration Problems 4.1 Unbalance and Bent Shaft 4.2 Bent Shaft 4.3 Eccentric Rotor 4.4 Misalignment 4.5 Mechanical Looseness 4.6 Resonance 4.7 Electrical Problems 4.8 Pumps Related Problems 4.9 Blowers and Fans Related Problems 4.10 Compressors Problems 4.11 Reciprocating Engines Problems	8
5. Diagnosis of Special Parts Problems 5.1 Journal Bearing Problems 5.2 Roller Bearing Problems 5.3 Gear Trains Problems	4
6. Advanced Techniques 6.1 Transient Analysis 6.2 Dual and Multi-Channel Analysis 6.3 Machine Run-up/Coast-down	5
7. Field Balancing 7.1 Introduction 7.2 Procedure of Field Balancing	1
Total	28

1. Introduction

1.1 Mechanical Vibration

What is vibration? simply speaking, it is the motion of a machine or its part back and forth from its position of rest. The most classical example is that of a body with mass M to which a spring with a stiffness k is attached. Until a force is applied to the mass M and causes it to move, there is no vibration.

Mechanical vibration is the term used to describe the movement produced in mechanical parts due to the effect of external or internal forces on that parts. Each part can be considered composed of one or more spring-mass-damper system subjected to an exciting force. The amplitude of vibration is a function of system parameters and severity of the exciting force. When the machine is new, vibration level is low since there is no looseness or wear, i.e., the stiffnesses and damping factors are high. Also, the exciting forces are low in general because there is no mechanical problem yet. As the machine deteriorates, wears and looseness are produced and also, there may be exciting forces produced due to some faults such as unbalance and misalignments. Therefore, mechanical vibration becomes high. Maintenance procedure should be carried out to bring vibration low again and obtain smooth and trouble-free operation.



1.2 Machinery Preventive and Predictive Maintenance

If we were to do a survey of the maintenance philosophies employed by different process plants, we would notice quite a bit of similarity despite the vast variations in the nature of their operations. These maintenance philosophies can usually be divided into four different categories:

- Breakdown or run to failure maintenance
- Preventive or time-based maintenance
- Predictive or condition-based maintenance
- Proactive or prevention maintenance.

1.2.1 Breakdown or run to failure maintenance

The basic philosophy behind breakdown maintenance is to allow the machinery to run to failure and only repair or replace damaged components just before or when the equipment comes to a complete stop. This approach works well if equipment shutdowns do not affect production and if labor and material costs do not matter. The disadvantage is that the maintenance department perpetually operates in an unplanned 'crisis management' mode. When unexpected production interruptions occur, the maintenance activities require a large inventory of spare parts to react immediately. Without a doubt, it is the most inefficient way to maintain a production facility. Futile attempts are made to reduce costs by purchasing cheaper spare parts and hiring casual labor that further aggravates the problem. The personnel generally have a low morale in such cases as they tend to be overworked, arriving at work each day to be confronted with a long list of unfinished work and a set of new emergency jobs that occurred overnight.

1.2.2 Preventive or time-based maintenance

The philosophy behind preventive maintenance is to schedule maintenance activities at predetermined time intervals, based on calendar days or runtime hours of machines. Here the repair or replacement of damaged equipment is carried out before obvious problems occur. This is a good approach for equipment that does not run continuously, and where the personnel have enough skill, knowledge and time to perform the preventive maintenance work. The main disadvantage is that scheduled maintenance can result in performing maintenance tasks too early or too late. Equipment would be taken out for overhaul at a certain number of running hours. It is possible that, without any evidence of functional failure, components are replaced when there is still some residual life left in them. It is therefore quite possible that reduced production could occur due to unnecessary maintenance. In many cases, there is also a possibility of diminished performance due to incorrect repair methods. In some cases, perfectly good machines are disassembled, their good parts removed and discarded, and new parts are improperly installed with troublesome results.

1.2.3 Predictive or condition-based maintenance

This philosophy consists of scheduling maintenance activities only when a functional failure is detected. Mechanical and operational conditions are periodically monitored, and when unhealthy trends are detected, the troublesome parts in the machine are identified and scheduled for maintenance. The machine would then be shut down at a time when it is most convenient, and the damaged components would be replaced. If left unattended, these failures could result in costly secondary failures. One of the advantages of this approach is that the maintenance events can be scheduled in an orderly fashion. It allows for some lead-time to purchase parts for the necessary repair work and thus reducing the need for a large inventory of spares. Since maintenance work is only performed when needed, there is also a possible increase in production capacity. A possible disadvantage is that maintenance work may actually increase due to an incorrect assessment of the deterioration of machines. To track the unhealthy trends in vibration, temperature or lubrication requires the facility to acquire specialized equipment to monitor these parameters and provide training to personnel (or hire skilled personnel). The alternative is to outsource this task to a knowledgeable contractor to perform the machine-monitoring duties. If an organization had been running with a breakdown or preventive maintenance philosophy, the production team and maintenance management must both conform to this new philosophy. It is very important that the management supports the maintenance department by providing the necessary equipment along with adequate training for the personnel. The personnel should be given enough time to collect the necessary data and be permitted to shut down the machinery when problems are identified.

1.2.4 Proactive or prevention maintenance

This philosophy lays primary emphasis on tracing all failures to their root cause. Each failure is analyzed and proactive measures are taken to ensure that they are not repeated. It utilizes all of the predictive/preventive maintenance techniques discussed above in conjunction with root cause failure analysis (RCFA). RCFA detects and pinpoints the problems that cause defects. It ensures that appropriate installation and repair techniques are adopted and implemented. It may also highlight the need for redesign or modification of equipment to avoid recurrence of such problems. As in the predictive-based program, it is possible to schedule maintenance repairs on equipment in an orderly fashion, but additional efforts are required to provide improvements to reduce or eliminate potential problems from occurring repeatedly. Again, the orderly scheduling of maintenance allows lead-time to purchase parts for the necessary repairs. This reduces the need for a large spare parts inventory, because maintenance work is only performed when it is required. Additional efforts are made to thoroughly investigate the cause of the failure and to determine ways to improve the reliability of the machine. All of these aspects lead to a substantial increase in production capacity. The disadvantage is that extremely knowledgeable employees in preventive, predictive and prevention/proactive maintenance practices are required. It is also possible that the work may require outsourcing to knowledgeable contractors who will have to work closely with the maintenance personnel in the RCFA phase. Proactive maintenance also requires procurement of specialized equipment and properly trained personnel to perform all these duties.

1.3 Evolution of maintenance philosophies

Machinery maintenance in industry has evolved from breakdown maintenance to timebased preventive maintenance. Presently, the predictive and proactive maintenance philosophies are the most popular. Breakdown maintenance was practiced in the early days of production technology and was reactive in nature. Equipment was allowed to run until a functional failure occurred. Secondary damage was often observed along with a primary failure. This led to time-based maintenance, also called preventive maintenance. In this case, equipment was taken out of production for overhaul after completing a certain number of running hours, even if there was no evidence of a functional failure. The drawback of this system was that machinery components were being replaced even when there was still some functional lifetime left in them. This approach unfortunately could not assist to reduce maintenance costs. Due to the high maintenance costs when using preventive maintenance, an approach to rather schedule the maintenance or overhaul of equipment based on the condition of the equipment was needed. This led to the evolution of predictive maintenance and its underlying techniques. Predictive maintenance requires continuous monitoring of equipment to detect and diagnose defects. Only when a defect is detected, the maintenance work is planned and executed. Today, predictive maintenance has reached a sophisticated level in industry. Till the early 1980s, justification spreadsheets were used in order to obtain approvals for condition-based maintenance programs. Luckily, this is no longer the case. The advantages of predictive maintenance are accepted in industry today, because the tangible benefits in terms of early warnings about mechanical and structural problems in machinery are clear. The method is now seen as an essential detection and diagnosis tool that has a certain impact in reducing maintenance costs, operational vs repair downtime and inventory hold-up. In the continuous process industry, such as oil and gas, power generation, steel, paper, cement, petrochemicals, textiles, aluminum and others, the penalties of even a small amount of downtime are immense. It is in these cases that the adoption of the predictive maintenance is required above all. Through the years, predictive maintenance has helped improve productivity, product quality, profitability and overall effectiveness of manufacturing plants. Predictive maintenance in the actual sense is a philosophy – an attitude that uses the actual operating conditions of the plant equipment and systems to optimize the total plant operation.

It is generally observed that manufacturers embarking upon a predictive maintenance program become more aware of the specific equipment problems and subsequently try to identify the root causes of failures. This tendency led to an evolved kind of maintenance called proactive maintenance. In this case, the maintenance departments take additional time to carry out precision balancing, more accurate alignments, detune resonating pipes, adhere strictly to oil check/change schedules, etc. This ensures that they eliminate the causes that may give rise to defects in their equipment in the future. This evolution in maintenance philosophy has brought about longer equipment life, higher safety levels, better product quality, lower life cycle costs and reduced emergencies and panic decisions precipitated by major and unforeseen mechanical failures. Putting all this objectively, one can enumerate the benefits in the following way:

- *Increase in machine productivity:* By implementing predictive maintenance, it may be possible to virtually eliminate plant downtime due to unexpected equipment failures.
- *Extend intervals between overhauls:* This maintenance philosophy provides information that allows scheduling maintenance activities on an 'as needed' basis.
- *Minimize the number of 'open, inspect and repair if necessary' overhaul routines:* Predictive maintenance pinpoints specific defects and can thus make maintenance work more focused, rather than investigating all possibilities to detect problems.
- *Improve repair time:* Since the specific equipment problems are known in advance, maintenance work can be scheduled. This makes the maintenance work faster and smoother. As machines are stopped before breakdowns occur, there is virtually no secondary damage, thus reducing repair time.
- *Increase machine life:* A well-maintained machine generally lasts longer.
- *Resources for repair can be properly planned:* Prediction of equipment defects reduces failure detection time, thus also failure reporting time, assigning of personnel, obtaining the correct documentation, securing the necessary spares, tooling and other items required for a repair.
- *Improve product quality:* Often, the overall effect of improved maintenance is improved product quality. For instance, vibration in paper machines has a direct effect on the quality of the paper.
- *Save maintenance costs:* Studies have shown that the implementation of a proper maintenance plan results in average savings of 20–25% in direct maintenance costs in conjunction with twice this value in increased production.

1.4 Vibration analysis – a key predictive maintenance technique

1.4.1 Vibration analysis (detection mode)

Vibration analysis is used to determine the operating and mechanical condition of equipment. A major advantage is that vibration analysis can identify developing problems before they become too serious and cause unscheduled downtime. This can be achieved by conducting regular monitoring of machine vibrations either on continuous basis or at scheduled intervals. Regular vibration monitoring can detect deteriorating or defective bearings, mechanical looseness and worn or broken gears. Vibration analysis can also detect misalignment and unbalance before these conditions result in bearing or shaft deterioration. Trending vibration levels can identify poor maintenance practices, such as improper bearing installation and replacement, inaccurate shaft alignment or imprecise rotor balancing.

All rotating machines produce vibrations that are a function of the machine dynamics, such as the alignment and balance of the rotating parts. Measuring the amplitude of vibration at certain frequencies can provide valuable information about the accuracy of shaft alignment and balance, the condition of bearings or gears, and the effect on the machine due to resonance from the housings, piping and other structures. Vibration measurement is an effective, non-intrusive method to monitor machine condition during start-ups, shutdowns and normal operation. Vibration analysis is used primarily on rotating equipment such as steam and gas turbines, pumps, motors, compressors, paper machines, rolling mills, machine tools and gearboxes. Recent advances in technology allow a limited analysis of reciprocating equipment such as large diesel engines and reciprocating compressors. These machines also need other techniques to fully monitor their operation. A vibration analysis system usually consists of four basic parts:

1. Signal pickup(s), also called a transducer

2. A signal analyzer
3. Analysis software
4. A computer for data analysis and storage.

These basic parts can be configured to form a continuous online system, a periodic analysis system using portable equipment, or a multiplexed system that samples a series of transducers at predetermined time intervals. Hard-wired and multiplexed systems are more expensive per measurement position. The determination of which configuration would be more practical and suitable depends on the critical nature of the equipment, and also on the importance of continuous or semi-continuous measurement data for that particular application.

1.4.2 Vibration analysis (diagnosis mode)

Operators and technicians often detect unusual noises or vibrations on the shop floor or plant where they work on a daily basis. In order to determine if a serious problem actually exists, they could proceed with a vibration analysis. If a problem is indeed detected, additional spectral analyses can be done to accurately define the problem and to estimate how long the machine can continue to run before a serious failure occurs. Vibration measurements in analysis (diagnosis) mode can be cost-effective for less critical equipment, particularly if budgets or manpower are limited. Its effectiveness relies heavily on someone detecting unusual noises or vibration levels. This approach may not be reliable for large or complex machines, or in noisy parts of a plant. Furthermore, by the time a problem is noticed, a considerable amount of deterioration or damage may have occurred.

Another application for vibration analysis is as an acceptance test to verify that a machine repair was done properly. The analysis can verify whether proper maintenance was carried out on bearing or gear installation, or whether alignment or balancing was done to the required tolerances. Additional information can be obtained by monitoring machinery on a periodic basis, for example, once per month or once per quarter. Periodic analysis and trending of vibration levels can provide a more subtle indication of bearing or gear deterioration, allowing personnel to project the machine condition into the foreseeable future. The implication is that equipment repairs can be planned to commence during normal machine shutdowns, rather than after a machine failure has caused unscheduled downtime.

1.4.3 Vibration analysis – benefits

Vibration analysis can identify improper maintenance or repair practices. These can include improper bearing installation and replacement, inaccurate shaft alignment or imprecise rotor balancing. As almost 80% of common rotating equipment problems are related to misalignment and unbalance, vibration analysis is an important tool that can be used to reduce or eliminate recurring machine problems. Trending vibration levels can also identify improper production practices, such as using equipment beyond their design specifications (higher temperatures, speeds or loads). These trends can also be used to compare similar machines from different manufacturers in order to determine if design benefits or flaws are reflected in increased or decreased performance.

Ultimately, vibration analysis can be used as part of an overall program to significantly improve equipment reliability. This can include more precise alignment and balancing, better quality installations and repairs, and continuously lowering the average vibration levels of equipment in the plant.

1.5 Vibration Analysis and Measurement Equipment

1.5.1 Online data acquisition and analysis

Critical machines are almost always provided with continuous online monitoring systems. Here sensors (e.g. Eddy current probes installed in turbomachinery) are permanently installed on the machines at suitable measurement positions and connected to the online data acquisition and analysis equipment. The vibration data are taken automatically for each position and the analysis can be displayed on local monitoring equipment, or can be transferred to a host computer installed with database management software. Because monitoring equipment are permanently connected to the sensors, intervals between measurements can be short and can be considered as continuous. This ability provides early detection of faults and supplies protective action on critical machinery. Protective action taken by online data acquisition and analysis equipment is in the form of providing alarms to warn the operators of an abnormal situation. In cases of serious faults, this protective action can shut down machines automatically to prevent catastrophic failures. Transferring the information to a host computer with database management software enhances the convenience and the power of online data acquisition. It is also possible to connect multiple local monitoring units that can send data from different machines to a central host computer. Thus, machines at various physical locations can be monitored from one location. Also, information can be transferred from the host computer to the local monitoring unit for remote control. Vibration analysis/database management software can also be networked to multiple computers with the local area network (LAN) or a wide area network (WAN) to allow multiple users to perform condition monitoring of the machines.

Advantages

- Performs continuous, online monitoring of critical machinery.
- Measurements are taken automatically without human interference.
- Provides almost instantaneous detection of defects.

Disadvantages

- Reliability of online systems must be at the same level as the machines they monitor.
- Failure can prove to be very expensive.
- Installation and analysis require special skills.
- These are expensive systems.

1.5.2 Portable data collectors/analyzers

Modern data collectors/analyzers can provide information of any vibration characteristics in any desired engineering unit. There are basically two types of data collectors and analyzers, Single channel and Dual channel.

Advantages

- Can collect, record and display vibration data such as FFT spectra, overall trend plots and time domain waveforms.
- Provides orderly collection of data.
- Automatically reports measurements out of pre-established limit thresholds.
- Can perform field vibration analysis.

Disadvantages

- They are relatively expensive.
- Operator must be trained for use.
- Limited memory capability and thus data must be downloaded after collection.

1.5.3 Handheld Vibration Meter

A handheld vibration meter is an inexpensive and simple-to-use instrument that is an essential part of any vibration program. Plant operators and vibration technicians carry handheld meters and analyzers on their routine rounds. When these are held in contact with machinery, they provide a display of vibration levels (either analog or digital). The readout provides immediate information that can be used to determine if the overall vibration levels are normal or abnormal. Handheld vibration meters are typically battery powered and use an accelerometer for sensing. Sometimes a velocity pickup is used. They are small, lightweight and rugged for day-to-day use. Handheld meters can provide the following data (depending on the specific model):

- Acceleration (pk) (g)
- Velocity (pk-rms) (mm/s or in./s)
- Displacement (pk-pk) (microns or mils)
- Bearing condition (discussed later) (gSE, dB and others).

Advantages

- They are convenient and flexible, and require very little skill to use.

- It is an inexpensive starting point for any new condition-monitoring program.

Disadvantages

- Limited in the type of measurements that they can perform.
- They also lack data storage capability (however, some instruments are now available with some limited storage capacity).

2. Measuring Parameters and Vibration Severity Criteria

2.1 Oscillatory Motion

For the oscillatory motion shown in Figure beside, the motion of the mass from its neutral position, to the top limit of travel, back through its neutral position, to the bottom limit of travel and the return to its neutral position, represents one cycle of motion. This one cycle of motion contains all the information necessary to measure the vibration of this system. Continued motion of the mass will simply repeat the same cycle. This motion is called periodic and harmonic, and the relationship between the displacement of the mass and time is expressed in the form of a sinusoidal equation:

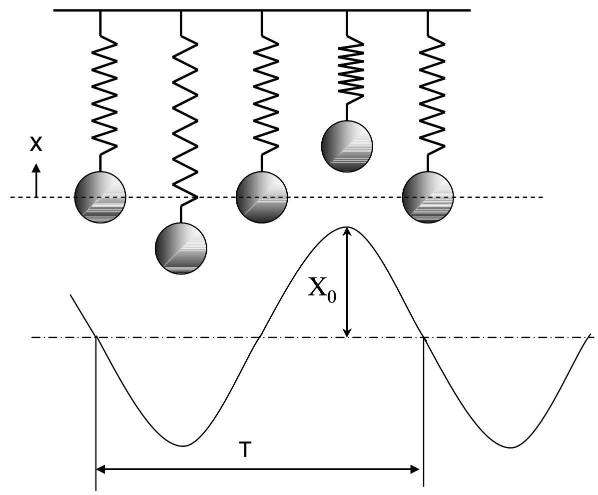


Figure 2.1 Oscillatory Motion

$$x = X_0 \sin \omega t$$

x = displacement at any given instant t;
X_0 = maximum displacement or peak amplitude;
$\omega = 2\,\pi f$, the radian frequency and measured in rad/s
f = frequency (cycles/s – hertz – Hz); t = time (seconds)
In the above Figure, T is the periodic time (period) is seconds, i.e., the time required for complete one period. The frequency of the signal is simply the reciprocal of the periodic time, i.e. $f = \dfrac{1}{T}$.

2.2 Acceleration, Velocity and Displacement

Returning to the Figure above, the velocity of the oscillating mass fluctuate from maximum value at the zero position to minimum value (zero) at the lowest and highest positions. In fact velocity is the derivative of the displacement i.e.:

$$v = \frac{dx}{dt} = X_0 \omega \cos \omega t$$

On the other hand, acceleration of the mass varies from maximum value at the highest and lowest positions to zero at the zero position. In fact acceleration is the derivative of velocity:

$$a = \frac{dv}{dt} = -X_0 \omega^2 \sin \omega t$$

It is clear that when the frequency ω is high, both velocity and acceleration will be high even when the displacement is small. At low frequency, the inverse is true. Determining which of the three parameters, acceleration, velocity or displacement is chosen to measure vibration depends primarily on two factors, the first is the reason of taking the measurement. Are the readings being taken for vibration analysis, periodic check, balancing , etc.. The second factor is the frequency of vibration to be measured, or in other words depending on the running speed and type of machine element such as anti-friction bearing, gear, etc..

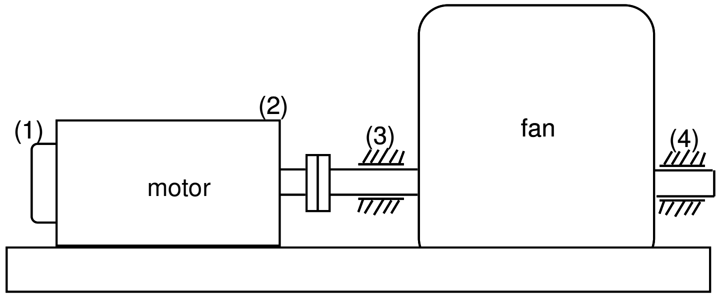
The amount of time required to break a machine part is a function of two parameters, the first is the amount of deformation (displacement) that the part undergoes, and the second is the frequency of deformation. Vibration severity, thus, appears to be a function of displacement and frequency. Since velocity is also a function of these two parameters, it is a direct measure of vibration severity. Displacement and acceleration may also be used to measure vibration severity, however, when these parameters are used it is also necessary to know the frequency of vibration. Displacement may be a better indicator to vibration severity under conditions of dynamic stress where the property of brittleness tends to fasten the failure or when the stress (deformation) reaches a given limit even though it is repeated only a few times. Thus displacement measurement is useful when low frequencies are encountered.

It is generally accepted that between 10 Hz (600 cpm) and 1000 Hz (60 kcpm) *velocity* gives a good indication of the severity of vibration, and above 1000 Hz (60 kcpm), acceleration is the only good indicator. Since the majority of general rotating machinery (and their defects) operates in the 10–1000 Hz range, velocity is commonly used for vibration measurement and analysis. Acceleration is closely related to dynamic forces and relatively large forces can occur at higher frequencies even though the displacement and velocity may be small. Thus acceleration measurement is a good indicator to vibration severity for high frequency vibration (above 1000 Hz).

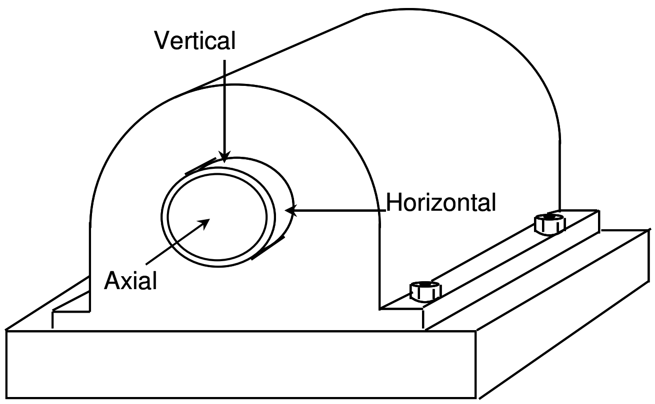
2.3 Location and Direction of Measurements

Two important parameters which significantly affect the result of measurement, are the location and direction of the measurement. Usually vibration is measured at the supports of the rotating parts, exactly the bearing casing when possible. This is because that the vibration of the rotating parts is transmitted only through the bearings. Furthermore, vibration of bearing themselves is measured at their casings. Figure 2.2(a) shows the locations of vibration measurements of a forced draft fan. However, sometimes measurements are taken on the structure of a machine for special purposes such as identifying the type and location of a structural failure or in determining the natural frequency of the structure.

The tri-axial measurements in the horizontal, vertical and axial directions are extremely useful not only in condition monitoring procedures, but also in diagnostic analysis as will be shown later in this chapter. The horizontal and vertical directions have a relative meaning rather than absolute one. They are both perpendicular to the shaft axis, but the difference between them, is that the horizontal direction is parallel to the fixing plane of the machine regardless of the machine alignment, while the vertical direction is perpendicular to it (along the fixing bolts). The axial direction is parallel to the shaft axis. This is illustrated in Figure 2.2(b).



(a) Locations of measurement



(b) Directions of measurement

Figure 2.2 Locations and Directions of Measurements

2.4 Common Vibration Severity Charts and Tables

As mentioned above, vibration amplitude (displacement, velocity or acceleration) is a measure of the severity of the defect in a machine. A common dilemma for vibration analysts is to determine whether the vibrations are acceptable to allow further operation of the machine in a safe manner. To solve this dilemma, it is important to keep in mind that the objective should be to implement regular vibration checks to detect defects at an early stage. The goal is not to determine how much vibration a machine will withstand before failure! The aim should be to obtain a trend in vibration characteristics that can warn of impending trouble, so it can be reacted upon before failure occurs. Absolute vibration tolerances or limits for any given machine are not possible. That is, it is impossible to *fix* a vibration limit that will result in immediate machine failure when exceeded. The developments of mechanical failures are far too complex to establish such limits.

However, it would be also impossible to effectively utilize vibrations as an indicator of machinery condition unless some guidelines are available, and the experiences of those familiar with machinery vibrations have provided us with some realistic guidelines. There are many operational criteria which set out vibration boundary levels for judging a machine condition.

In 1972, the American Gear Manufacturing Association formulated the “AGMA standard Specification for Measurement of Lateral Vibration on High Speed Helical and Herringbone Gear Unite”. This standard is shown in Figure 2.3.

The “IRD Mechanalysis Vibration Acceleration General Severity Chart” shown in Figure 2.4 can be used in cases when machinery vibration is measured in units of acceleration. This chart is useful for evaluating machinery condition for the vibration of relatively high frequencies (above 1000 Hz) such as bearing vibration. It is obvious from the chart that the constant velocity lines are replaced by constant acceleration lines for the frequencies above 1000 Hz (60,000 rpm).

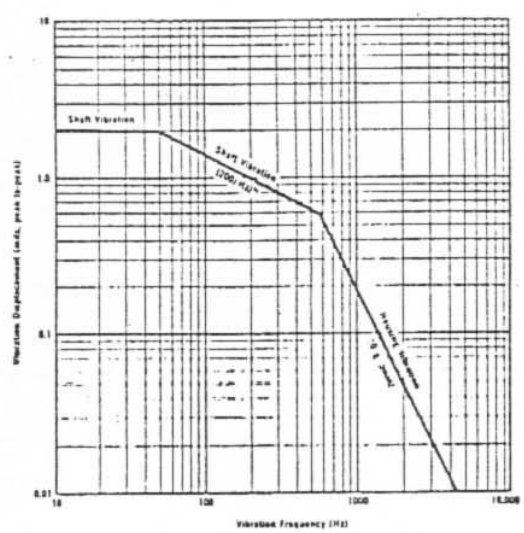


Figure 2.AGMA Vibration Severity Chart

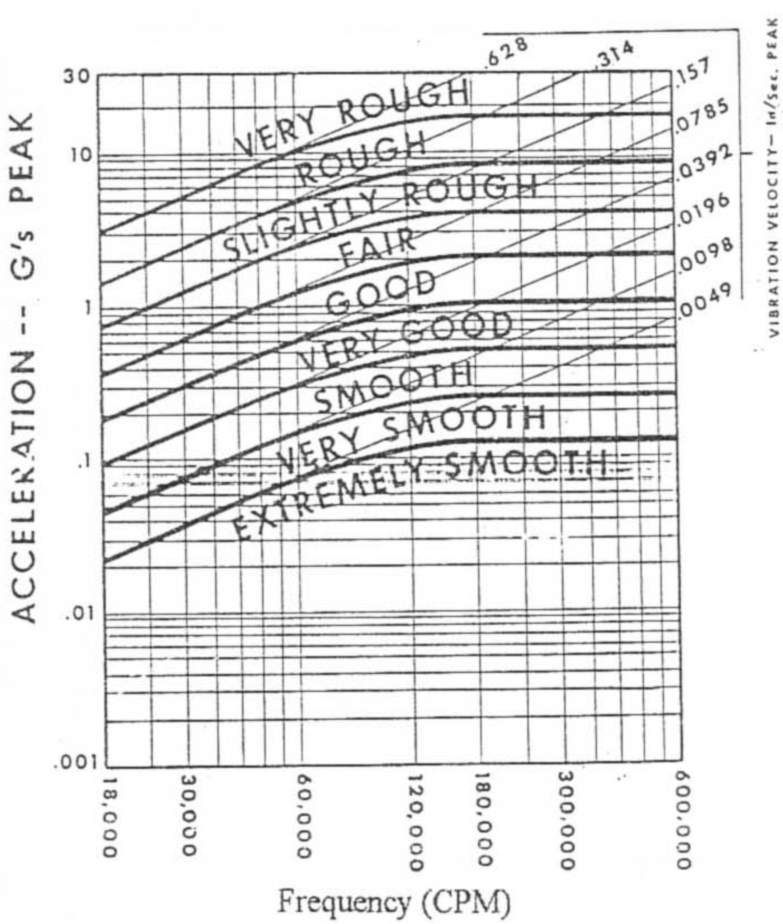


Figure 2.4 IRD Mechanalysis Vibration Acceleration General Severity Chart

Another commonly used severity criterion was by Verein Deutscher Ingenieure (German Engineering Society) published the VDI 2056 Vibration Severity Criteria in 1940. This criteria is based on the RMS values of vibration velocity over the frequency range from 10 Hz to 1000 Hz. The VDI 2056 criteria is somewhat unique, compared to other guidelines presented, in that an attempt to establish allowance for different types of machines and foundations. Examples of machine classification as well as the vibration limits are shown in Figure 2.5. The ISO2372 agree with VDI 2056 criterion.

Another severity chart which is close to VDI 2056 is ISO 10816 Vibration Severity chart which is shown in Figure 2.6. A distinct feature in this chart is that it consider machine speed as factor influencing filter selection. For machines running at speed above 600 RPM, the filter choice must be 10-1000Hz, while for machines running at speed 120 RPM up to 600 RPM filter choice must be 2-1000 Hz.

Ranges of vibration severity		Classes of machines			
Range	RMS velocity (mm/s)	Class I	Class II	Class III	Class IV
0.28		A	A	A	A
0.45	0.28	A	A	A	A
0.71	0.45	A	A	A	A
1.12	0.71	B	A	A	A
1.8	1.12	B	B	A	A
2.8	1.8	C	B	B	A
4.5	2.8	C	C	B	B
7.1	4.5	D	C	C	B
11.2	7.1	D	D	C	C
18	11.2	D	D	D	C
28	18	D	D	D	D
45	28	D	D	D	D
71	45	D	D	D	D

Class I : Individual parts of engines and machines, integrally connected with the complete machine in its normal operating condition. (production electrical motors of up to 15 kW are typical examples of machines in this category).

Class II : Medium sized machines, (typically electrical motors with 15 to 75 kW output) without special foundations, rigidly mounted engines or machines (up to 300 kW) on special foundation.

Class III : Large prime movers and other large machines with rotating masses mounted on rigid and heavy foundations which are relatively stiff in the direction of vibration measurement.

Class IV : Large prime movers and other large machines with rotating masses mounted on rigid and heavy foundations which are relatively soft in the direction of vibration measurement.

A: Good
B: Allowable
C: Just Tolerable
D: Not Permissible

Figure 2.5 VDI 2056 and ISO2372 Vibration Severity Chart

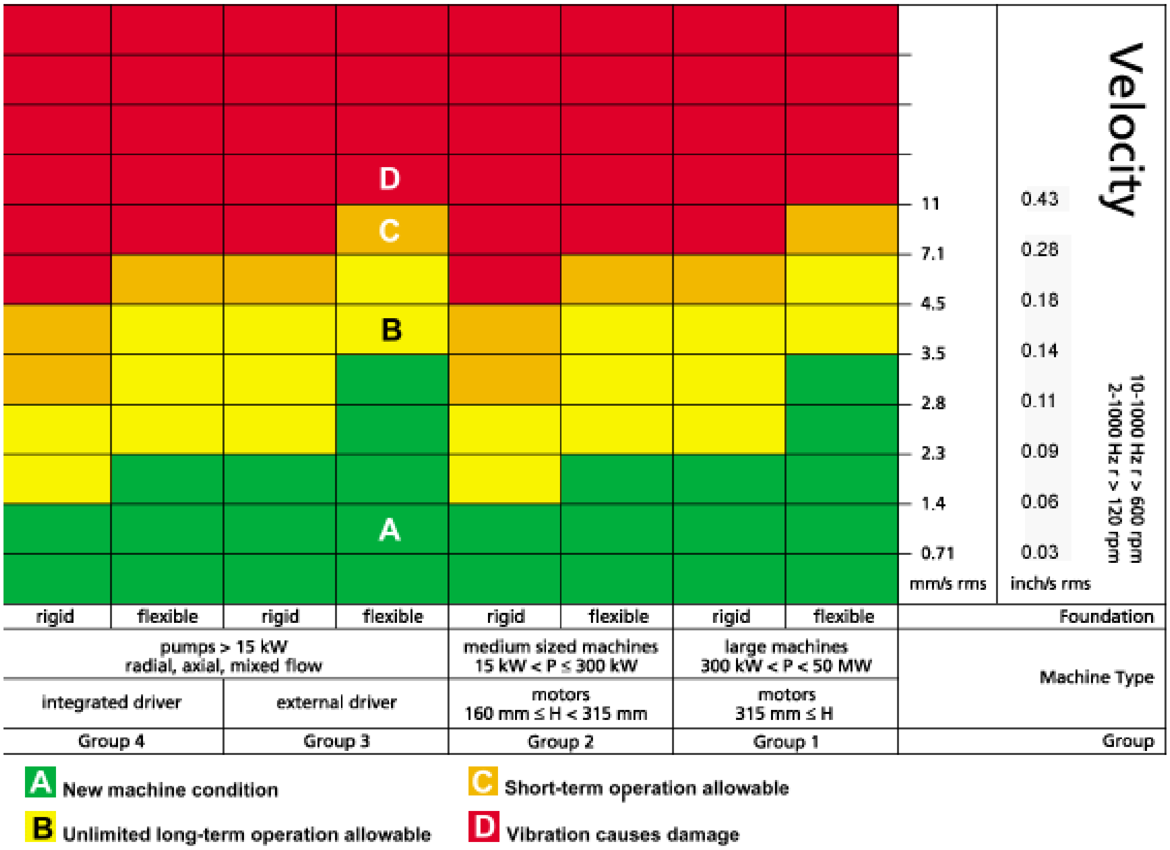


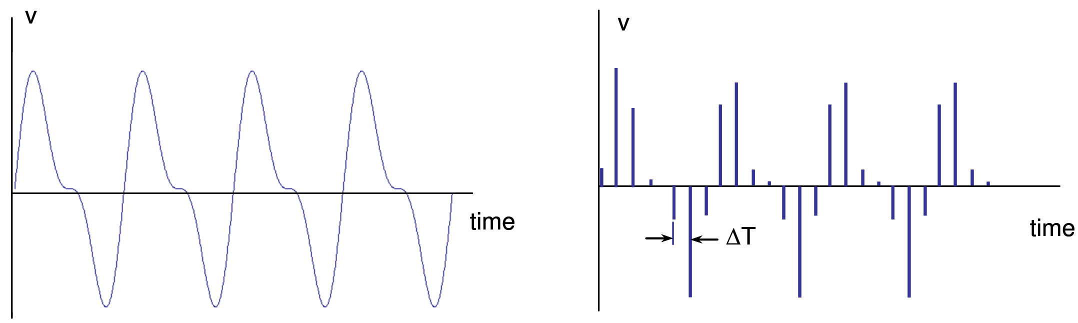
Figure 2.6 ISO 10816 Vibration Severity Chart

3. Vibration Analysis Techniques

3.1 Definitions

Analog Signal: it is simply analog voltage varying with time. Examples are AC main which has frequency of 50 or 60 Hz, microphone output, vibration transducer output … etc. See Figure 3.1 below.

Digital Signal: an analog signal after digitizing process. Digitizing process includes conversion from analog voltage into digital numbers by the Analog to Digital Converter. The sampling time is ΔT and it is fixed in general. The sampling frequency is the reciprocal of ΔT; $f_s = 1/\Delta T$



(a) Analog signal (b) Digital signal

Figure 3.1 Analog and Digital Signals

Analog to Digital Converter: a device used to convert voltage into equivalent integer number. The size of the integer number (number of bits) depends on the resolution of the ADC. For example, an ADC with 16-bit resolution produces an integer value between 0 and 65535 (or -32768 to +32767)

Peak Value: the difference between the signal average and maximum absolute value.

Peak to Peak Value: the difference between the lowest and highest values in the signal.

Stationary Signal: a signal which has the same statistical parameters over time such as signal produced by rotating machinery.

Non-stationary Signal: a signal which has variable parameters over time such as variable frequencies, amplitudes, power content ...etc. Examples are speech signals.

3.2 Level Detection

Signal level is sometimes referred as magnitude or overall value. As related to vibration, level can be detected in terms of Root Mean Squares (defined below), Peak value, Peak-Peak value or simply average value. The RMS, peak and peak to peak detection techniques are mainly used in the determination of vibration severity for condition monitoring of machines. The average value has limited applications and can be used to estimate the average rotor position for example.

The root mean squares value (RMS) is indication for the power content in the signal, or in other words, the effective value. Therefore, it is commonly used in vibration level detection. For analog signals, the RMS can be detected by using RMS detector which is sort of complex circuitry. The voltmeter is a kind of RMS detector. The RMS vlaue is given by;

$RMS = \sqrt{\frac{1}{T}\int_0^T [v(t)]^2 dt}$ where T is the period of the signal or generally record length.

For digital signals, the RMS is given by:

$RMS = \sqrt{\frac{1}{N}\sum_{i=1}^{N} v_i^2}$ where N is the number of points (record length)

Facts about RMS:

- For pure sine wave, the RMS = 0.707A, where A is the amplitude of the sine wave, therefore, the RMS does **not** depend on the frequency of the signal
- The RMS takes into account all the frequencies contained in the signal at equal weight. The phases and frequency ratios between different components have no effect on the RMS value.
- While digital RMS detectors are simpler and more efficient, sampling frequency for digital signals should be sufficiently high (more than at least twice the maximum frequency in the signal) to obtain reliable RMS value from digital signals.

The Peak value can easily be obtained by using a peak detector for analog signals. While for digital signals, it can be obtained by taking the difference between the maximum absolute value and the average. This is in fact referred as True Peak. Another widely used peak value in modern vibration systems is the Scaled Peak which is obtained directly from the RMS value by multiplying be 1.414;

$Scaled\ Peak = 1.414 RMS$

Scaled Peak is often used to obtain an approximate Peak value from a signal which undergoes some processing techniques that modify the original shape such as filters, integrators and differentiators.

The True Peak-Peak value is the difference between the lowest and highest value in the signal and it can easily be obtained by peak-peak detector in the analog signals. For digital signals the Peak-Peak value can easily be estimated by subtracting the minimum value from the maximum value. However, the Scaled Peak-Peak value is commonly used in modern analysis systems since the signal will not be kept at its original shape (for example when the accelerometer signal is double integrated to obtain displacement signal);

$Scaled\ Peak - Peak = 2.828 RMS$

Table 3.1 Conversion between RMS, Peak and Peak-Peak Values

	From			
	Multiply by	Scaled Peak	Scaled P-P	RMS
	Scaled Peak	1	0.5	1.414
To Get	Scaled P-P	2	1	2.828
	RMS	0.707	0.3535	1

3.3 Time Waveform

Time Waveform is simply displaying the signal in the same manner as the oscilloscope plot. It is the amplitude-time plot. The most common use of time waveform data is to compare the waveform pattern of one machine with another obtained from a machine with similar defects. If necessary, the frequency components of the major events in the waveform pattern can be calculated.

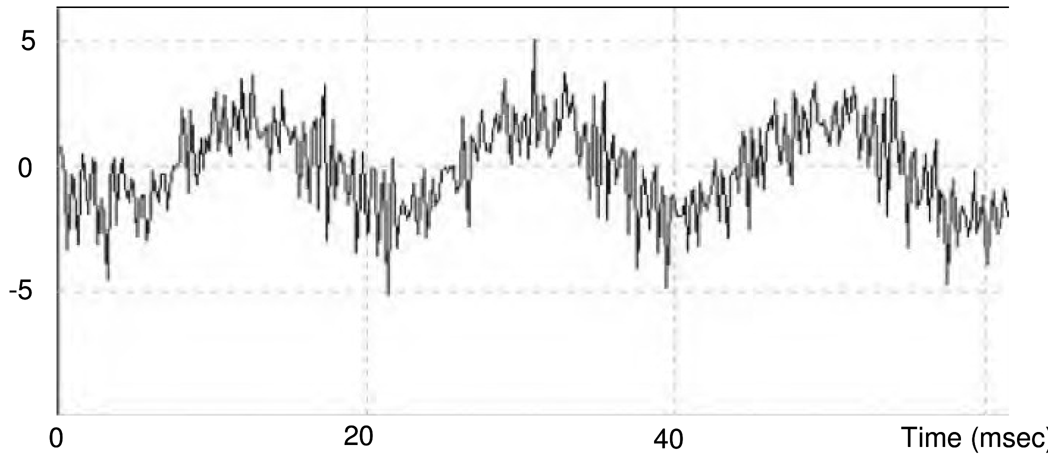


Figure 3.2 Time Waveform of Low and High Frequency

Figure 3.2 shows a waveform collected from a pump with a predominantly 1x RPM waveform on which a high-frequency wave is superimposed.

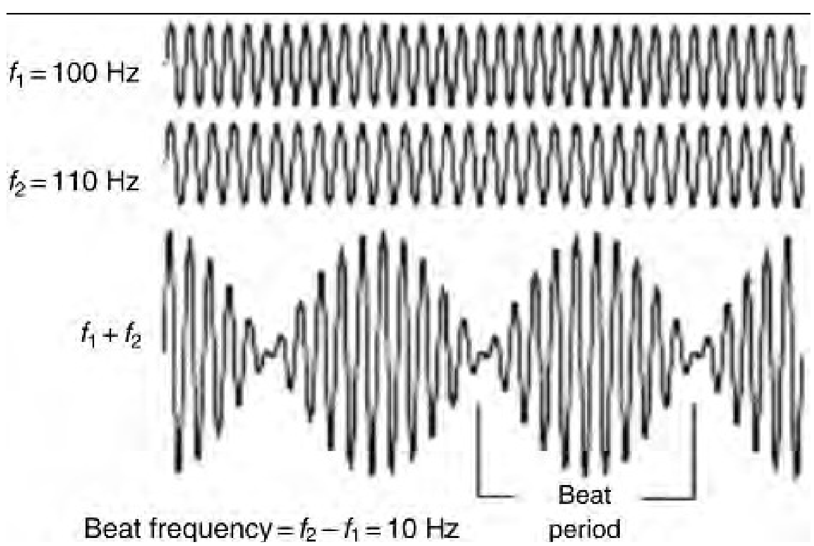


Figure 3.3 Waveform Beats

Figure 3.3 shows a special waveform describing a phenomenon called *beats*. Two waveforms having frequencies separated only slightly and with approximately the same amplitude will produce a beating waveform. These are literally pulses due to alternating reinforcement and cancellation of amplitudes. The amplitude change is called the modulation and has a frequency equal to the difference between the frequencies of the two waveforms. If the difference decreases, the beat frequency will also decrease. Beating waveforms are common at centrifuges that have bowl or scroll at marginally different speeds, and it is very normal to obtain the beat frequency if there is some unbalance in each. In some cases, it might be possible to time the beats to determine the difference between the bowl and scroll speeds. This phenomenon also occurs in motors that have electrical defects. These defects tend to generate a vibration frequency of twice the transmission power line frequency. If the line frequency is 50 Hz (3000 cpm), the defect frequency would be 6000 cpm. Now, if the motor's physical speed were 2980 rpm, then its second harmonic would be 5960 cpm. Thus, the two waveforms of 6000 and 5960 cpm will generate beats and modulation of amplitude.

Another application where time waveform is found to be useful is the identification of bearings and gears problems. Pulses or spikes are found whenever a localized defect exists with frequency equal to the number of times that defect excited per a second. For example a pinion gear with broken tooth will exhibit pulses at frequency equal to 1xRPM of the pinion as shown in Figure 3.4. The time between any two successive pulses is the reciprocal of the rotation frequency.

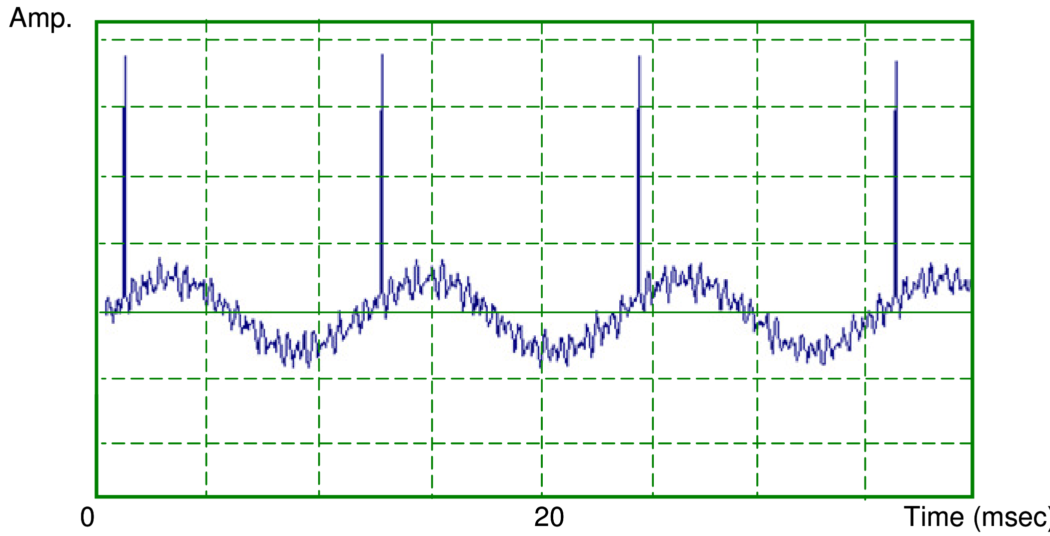


Figure 3.4 Vibration of Pinion Gear with Broken Tooth

Areas where the time waveform can provide additional information to that obtained from FFTs are:

- Low-speed applications (less than 100 rpm)
- Indication of true amplitude in situations where impacts occur, such as assessment of the severity of defects in rolling element bearings and gears
- Looseness
- Rubs
- Beats.

In the case of defects such as unbalance or misalignment, where the time waveform is not too complex, it will not be an advantage to the time waveform for diagnosis.

3.4 FFT Spectrum and Phase Analysis

Fourier theorem states that any time waveform can be reconstructed from a number of harmonically related sine and cosine frequency components. Fourier transform is found to be very efficient and useful tool to analyze vibration signals and to detect most of the common vibration problems. Fourier spectrum is simply the amplitude-frequency plot and can be done through different techniques. Before the revolution of digital computers and related components, Fourier analysis was executed using tunable bandpass filter inline with amplitude or RMS detector.

Recent advances in digital circuit technology, in particular, the develop-ment of large-scale integration (LSI) technology, have caused a revolution in system design. Many functions that only a few years ago were most practically implemented with analog circuitry can now be implemented more practically in digital form. Using the digital approach, the designer no longer has to be concerned with the realizability constraints of analog devices. As a result, significantly more sophisticated algorithms can now be chosen for problem solving. Examples of digital processing operations are digital filtering, integration and differentiation, FFT (spectrum) analysis, processing of speech and images, and many other operations.

The digital systems have several advantages when compared with their analog equivalents. The first of these is that the digital system is inherently more stable, this means that the system is less susceptible to changes in environment. The second advantage is the improved linearity. The only significant source of amplitude non-linearity in digital systems is within the ADC. Likewise, their frequency axes are set by the sampling frequency, which is in turn is referenced to a crystal controlled oscillator. Hence, the sampling frequency can be controlled to a high degree of accuracy, to give an exceptional frequency linearity.

The first digital frequency analysis technique was the digital filtering, but soon it has superseded by the technique of Fast Fourier Transform. The Fast Fourier Transform (FFT) is an algorithm or calculation process for obtaining the Discrete Fourier Transform (DFT) with a greatly reduced number of arithmetic operations compared with a direct evaluation. Fourier showed that any periodic function g(t) with a period T can be represented as a sum of sinusoidal components (or equivalently rotating vectors) at equally spaced frequencies kf_1, with (f_1=1/T). The k*th* component is obtained from the integral,

$$G(f_k) = \frac{1}{T}\int_{-T/2}^{T/2} g(t) e^{-j2\pi f_k t} dt \tag{3.1}$$

and hence,

$$g(t) = \sum_{k=-\infty}^{\infty} G(f_k)\, e^{j2\pi f_k t} \tag{3.2}$$

The series of complex values $G(f_k)$ are known as the spectrum components of g(t).

When g(t) is sampled in time domain, i.e. it is defined at finite number of instants N, then both the time signal and frequency spectrum will be implicitly periodic due to sampling process. This periodicity or circularity leads to some interesting effect. The forward transform for this case takes the form

$$G(f_k) = \frac{1}{N}\sum_{n=0}^{N-1} g(t_n) e^{-j\frac{2\pi kn}{N}} \tag{3.3}$$

and, hence, the inverse transform takes the form

$$g(t_n) = \sum_{k=0}^{N-1} G(f_k) e^{j\frac{2\pi kn}{N}} \tag{3.4}$$

The direct evaluation of eq. (3.3) requires N^2 complex multiplications and additions, and for moderately large N, say N greater than 1000, this direct evaluation is rather costly in computer time. Methods for saving computer time are thus be used. The most efficient algorithm for evaluating DFT is the so called "Fast Fourier Transform". In fact, this algorithm results in dramatic saving in computer time when N is large, however, it can not be directly applied when N is prime. When N is a power of 2, the FFT requires a number of computations proportional to $N \log_2 N$ rather than N^2. Thus for N=1024 this is a computational saving of 99 percent.

The FFT algorithm produces an identical result to direct application of the DFT. Thus any limitations of the FFT process are in fact those of the DFT. These are basically due to the finite (circular) and discrete nature of the DFT algorithm. Thus, regardless of the actual nature of the input signal, the analyzed record and results are a finite number, N, of discrete digital sample. In theory this represents one period of an infinitely long periodic signal. Generally, three problems are associated with the FFT processing, they are "aliasing" , window effect and picket fence effect. These three problems, and how to deal with them, will now be discussed in a little more details.

3.4.1 Definitions

Number of Points (N): total number of samples in a record

Record Length (T): total record length in seconds, equal to number of points multiplied by ΔT

Frequency Resolution Δf: reciprocal of record length or sampling frequency divided by number of points, $\Delta f = f_s / N$

3.4.2 Aliasing

The misinterpretation of high frequencies (above half the sampling frequency) as lower frequencies, as illustrated in Figure 3.5, is termed "aliasing". This is obviously one of the pitfalls to be avoided when digitizing continuous signals.

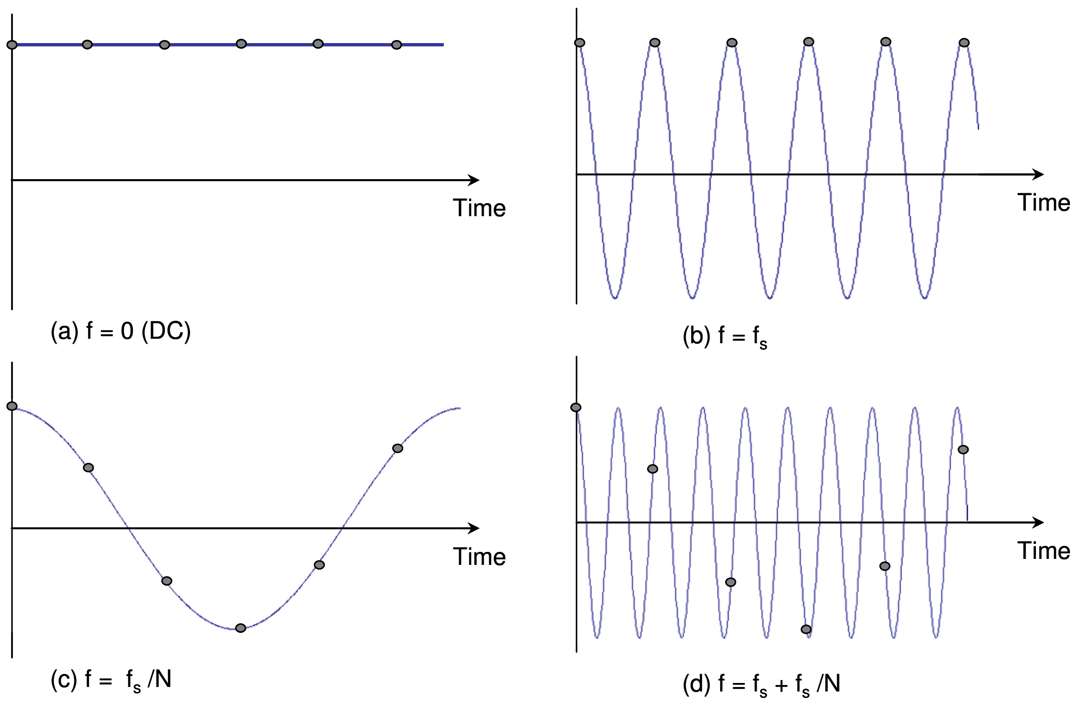


Figure 3.5 Illustration of Aliasing

Aliasing can be avoided by using analog low-pass filter to ensure that the maximum frequency in the passed signal is not greater than on half of the sampling frequency (this is called Nyquist frequency). The low-pass filter should be very steep in order to cancel out high frequency components that may exist. 8^{th} order or higher order Butterworth or Elliptic filter is preferred although 4^{th} order is commonly applied in many digital systems. To overcome the problem of amplitude change near the cut-off frequency, the useful frequency range is chosen to be only 80% (or less) of the Nyquist frequency.

3.4.3 Window Effect

Window effect, also know as leakage effect, results from fitting the time signal in a finite length. Generally when the sampling frequency is not direct multiples of signal frequency, window or power leakage effect will arise.

The window effect results in the sidelobe generation or power leakage from the main frequency components into adjacent band. This is determined by the type of window function used. For a rectangular window function, the sidelobe generation is illustrated in Fig. 3.6. When a sinusoid period or its multiples exactly coincide with the record length, i.e., its frequency coincides with one frequency line in the spectrum, and due to convolution (in frequency domain) between that single line and the spectral function of a rectangular window (sin x /x), the zero of the latter will lie on the frequency lines of the resulting spectrum, and thus, no sidelobe is generated.

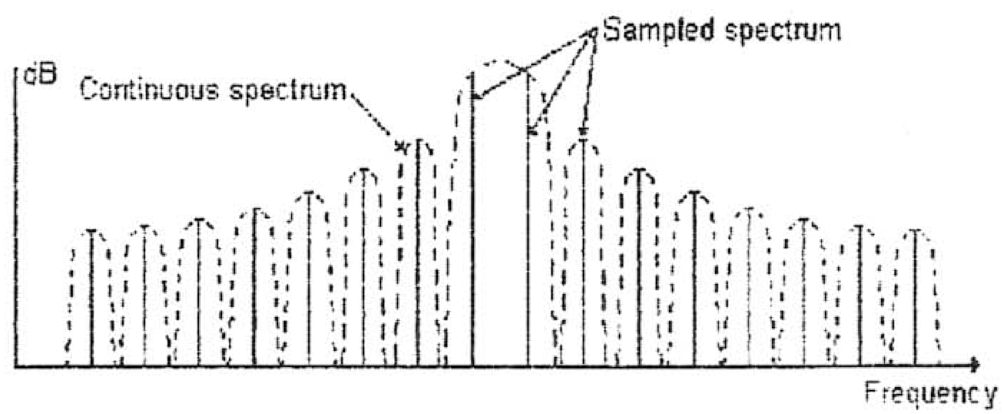


Figure 3.6 Sidelobes Generation for Rectangular Window

To minimize sidelobes generation and amplitude error, a window function having a minimal sidelobes in the frequency domain must be used. An excellent general purpose window function is known as "Hanning", its name being derived from Von Hann, who applied an equivalent process to meteorological data. In the time domain the Hanning window is equivalent to one period of a raised cosine (i.e. cosine squared) function, as illustrated in Fig. 4.9,

$$h(n) = 0.5[1 - \cos(2\pi n / N)] \qquad n=0,1,2,\ldots N-1 \qquad (3.5)$$

Other good window functions are Hamming, Blackman, Flat-top, Kaiser and others. Hamming window has lower sidelobes than Hamming but sidelobe falloff is small. Flat-top window is used to obtain minimum amplitude error when picket fence correction technique is not used.

3.4.4 Picket Fence Effect

The picket fence effect is usually limited to FFT analysis. The picket fence effect results from the fact that there are only specified number of lines to represent the continuous spectrum which have an infinite number of lines. In general, unless a frequency component coincides exactly with an analysis line, there will be an error in both the indicated amplitude and frequency (where the highest line is considered as representing the frequency component) as shown in Figure 3.7. This can be compensated for, provided it is known that one is dealing with a single stable frequency component, by using picket fence correction technique where both the actual frequency and amplitude are retained from the analysis data.

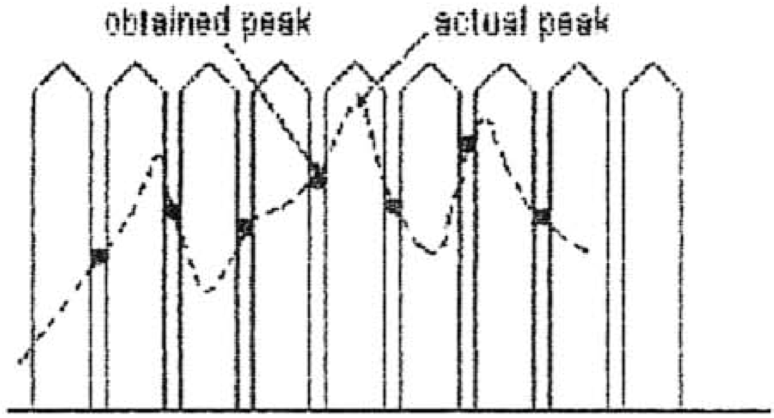


Figure 3.7 Illustration of Picket Fence Effect

3.4.5 Practical Analysis of Vibration Signals

Most of the FFT parameters are previously mentioned, however may be summarized here for practical application as follows.

The *frequency range* for baseband analysis is from zero (DC component) to the Nyquist frequency which is one half of the sampling rate. Thus, in order to avoid aliasing, the highest frequency in the signal to be analyzed must be lower than the Nyquist frequency. To achieve this, a low-pass filter having a cut-off frequency at 80% to 100% of the Nyquist frequency must be applied before digitizing or re-sampling .

The actual *useful frequency range* (or maximum frequency) is the displayed range, which is limited by the anti-aliasing filter. It is from zero to the cut-off frequency.

The *number of lines* for baseband analysis is related to the number of samples in the data record (N) and the useful frequency range, i.e. it is equal to 80% to 100% of N/2, e.g., if N=1024 time samples, and the cut-off frequency is 80% of the Nyquist frequency, then the number of lines would be 409 or 400.

In some cases, it is required to obtain the overall RMS level which is an indication to the total power contained in a signal. This easily done, for rectangular weighting, as follows,

$$\mathrm{RMS} = 0.707\sqrt{|G_1|^2 + |G_2|^2 + |G_3|^2 + \ldots} \tag{3.6}$$

Equation (3.6) can not be generalized for other weighting functions. In fact the result should be corrected for the noise bandwidth (B) inhered to multiplying by the window function. Generally, for any data weighting function, the RMS value is given by

$$\mathrm{RMS} = \frac{1}{B}\left[0.707\sqrt{|G_1|^2 + |G_2|^2 + |G_3|^2 + \ldots}\right] \tag{3.7}$$

Averaging is useful to obtain more reliable data about the analyzed signal. it is important to perform averaging over a number of individual FFT transforms, each transform corresponds to a different time record. Figure 3.8 illustrate averaging for 0% and 50% overlapping.

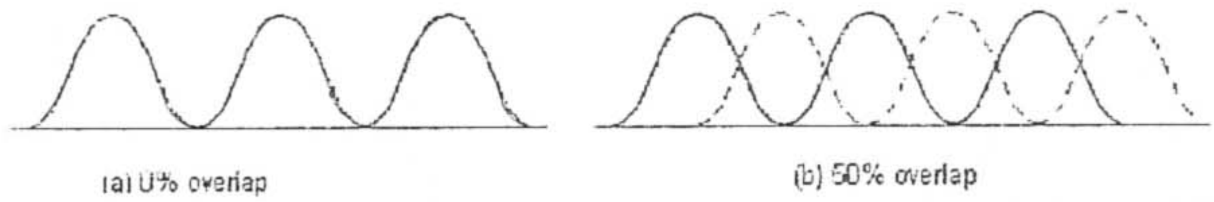


Figure 3.8 Averaging with 0% and 50% Overlapping

3.4.6 Phase Detection from FFT Spectrum

The complex frequency spectrum of a time function g(t) can be rewritten as [32];

$$g(t) = \sum_{k=-\infty}^{\infty} G_k e^{j2\pi f_k t} = |G_0| + \sum_{k=1}^{\infty} 2|G_k| \sin(2\pi f_k t - \phi_k) \tag{3.8}$$

where ϕ_k is the phase angle of the k*th* sinusoid (frequency component) referenced to the starting point of the time record. Therefore, when the reading is trigged by a reference signal, all phase angles will be relative to the reference sensor position;

$$\phi_k = \pi + \tan^{-1} \frac{\mathrm{Re}(G_k)}{\mathrm{Im}(G_k)} \tag{3.9}$$

However, when the actual frequency lies between two analysis lines, k and k+1, the phase indicated by these lines will be determined in terms of the phase of the actual component, as well as the frequency difference between that component and the analysis line. The actual phase can be determined in terms of the phase of the second analysis line and the frequency difference (Δx) which is found by the picket fence correction technique,

$$\phi_{actual} = \begin{cases} \Delta x \cdot \pi + \tan^{-1} \dfrac{\mathrm{Re}(G_{k+1})}{\mathrm{Im}(G_{k+1})} & x > 0 \\ (1 + \Delta x)\pi + \tan^{-1} \dfrac{\mathrm{Re}(G_{k+1})}{\mathrm{Im}(G_{k+1})} & x < 0 \end{cases} \tag{3.10}$$

3.5 Orbit

Orbits are Lissajous patterns of time domain signals that are simultaneously plotted in the X–Y coordinate plane of an oscilloscope or vibration analyzer. In this form of display, it is very difficult to trace the start of the orbit as it appears to be an endless loop. In order for us to determine the direction of rotation, a phase trigger is employed. The trigger will show the direction of rotation by looking at the dot on the orbit as the starting point of 1× RPM and the blank space as the end point.

Orbit analysis is the vibration measure of any rotor system in an X–Y plot (Figure 3.9). In most applications, the unit of measurement is displacement which is measured directly using proximity probes. These types of measurements are relative vibration readings. Relative readings are considered vibration measurements of the shaft with respect to the bearing housing. As the probes are clamped firmly to the housing, there is no relative motion between the probe and the housing. Thus, the orbit is achieved. With that in mind, orbit plots give a visual graph of the actual shaft centerline movement inside the bearing housing. Accelerometers and velocity pickups can also be used to create orbits. These are external transducers, which require mounting on the outside of the bearing housing. These types of measurements are called case orbits. Case orbits are useful to separate shaft and case vibrations. This can provide absolute shaft motion (relative to space). Orbit may be done for the overall signals as measured or it can be done for filtered signals where it is required to show orbit for specific frequency such as the frequency of rotation or its multiples.

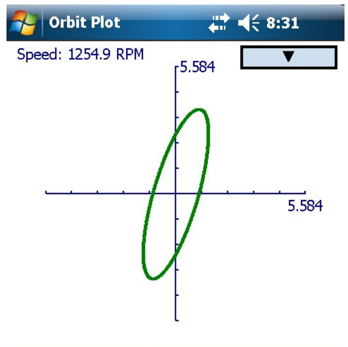


Figure 3.9 Orbit Analysis

To understand orbits, waveforms and their relationship to orbits must be explained. Let us begin with waveforms. The waveform plot shown in Figure 3.10 has two sine waves, Y and X. The Y plot is on the top and the X plot is at the bottom. The waveform signature runs left to right and the amplitudes change from negative to positive, whatever the case may be. The changes in the waveform cause the orbit to form. An orbit is made up of an *X*- and *Y*-axis with zero in the center. Starting from the center, up is positive and down is negative. Right is positive and left is negative. Now that we know waveform and orbit conventions, let us trace the waveforms and create an orbit.

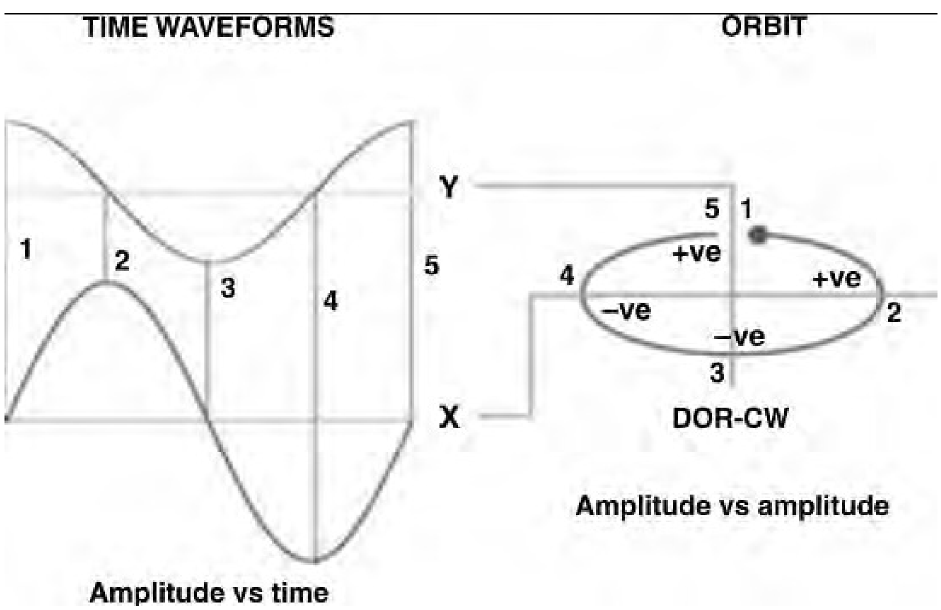


Figure 3.10 Waveforms and their Orbit

3.6 Bode and Nyquist Plots

A Bode plot comprises of two graphs; amplitude vs. machine speed and Phase vs. machine speed. To display a Bode plot, a phase trigger is used to obtain a shaft reference for phase measurement and measure the machine speed. The analyzer triggers and records the amplitude and phase

simultaneously at specific speed intervals (which can be defined by the user), and the two graphs are displayed on top of each other.
In rotor dynamics, the Bode plot is mainly used to determine the critical speed of the rotor. In the plot, the speed at which amplitude of vibration is maximum is noted, and for confirmation the phase graph is checked to see if it differs from the starting value by 90°.

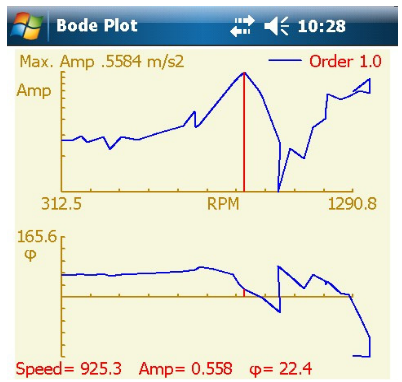


Figure 3.11 Bode Plot

The Bode plot can also be used to determine the amount of runout associated with a proximity probe, the balance condition, system damping and the operational phase angle cum amplitude at various machine speeds.

The polar or Nyquist plot is also a representation of the same three variables as considered in a Bode plot. The variables are plotted on a single circular chart instead of Cartesian axes as shown in Figure 3.12.

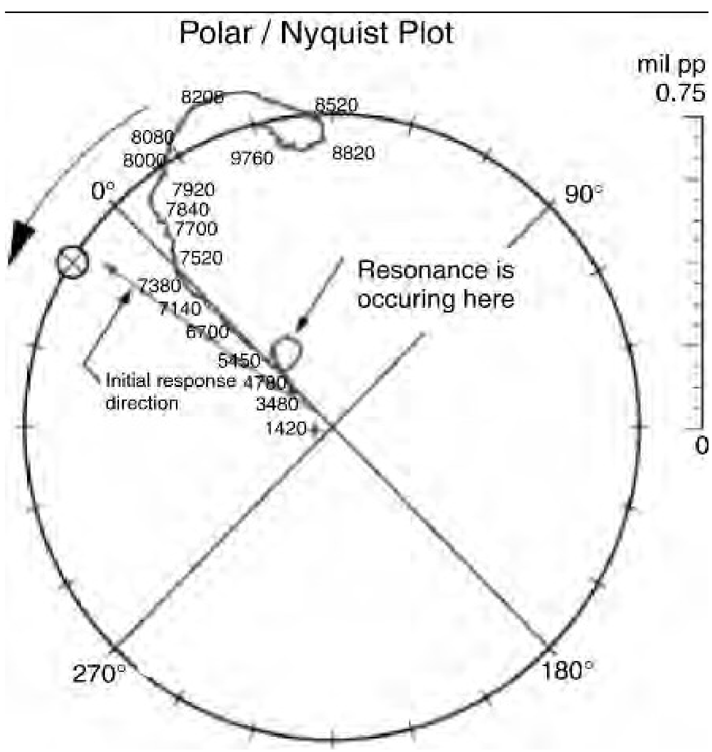


Figure 3.12 Nyquist Plot

3.7 Cepstrum Analysis

Cepstrum analysis is referred to a range of techniques all involving functions which can be considered as "spectrum of the log spectrum" . Presumably, because it was a spectrum of a spectrum, the word "cepstrum" comes from "spectrum" , and likewise the following terms are derived in the same manner;

Quefrency from Frequency

Rahmonics from Harmonics

However, the distinctive feature of the cepstrum is not that it is a spectrum of a spectrum, but rather than the logarithmic conversion of the original spectrum. In fact, the most commonly used definition of the cepstrum nowadays is as the inverse Fourier transform of the logarithmic power spectrum [7]. The last definition of the power cepstrum may be expressed as follows

$$C_{AA}(\tau) = \mathcal{F}^{-1}\{\log\ S_{AA}(f)\} \tag{3.11}$$

in which the two sided power spectrum, $S_{AA}(f)$, of a time signal g(t) is given by:

$$S_{AA}(f) = \overline{\left|\mathcal{F}\{g(t)\}\right|^2} \tag{3.12}$$

Where the bar means averaging over a number of records to improve reliability.

The parameter τ in the definition is actually time, although it is referred to as quefrency. It can better be thought as a "delay time" or "periodic time" rather than absolute time. Since cepstrum can detect harmonic vibration components as single quefrency component, the application of the cepstrum analysis in vibration diagnosis is required for machines with gear mesh, anti-friction bearing and other sources of high frequency vibration containing harmonically related components. Figure 3.13 shows a cepstrum plot. More details about cepstrum will be given in the next chapter.

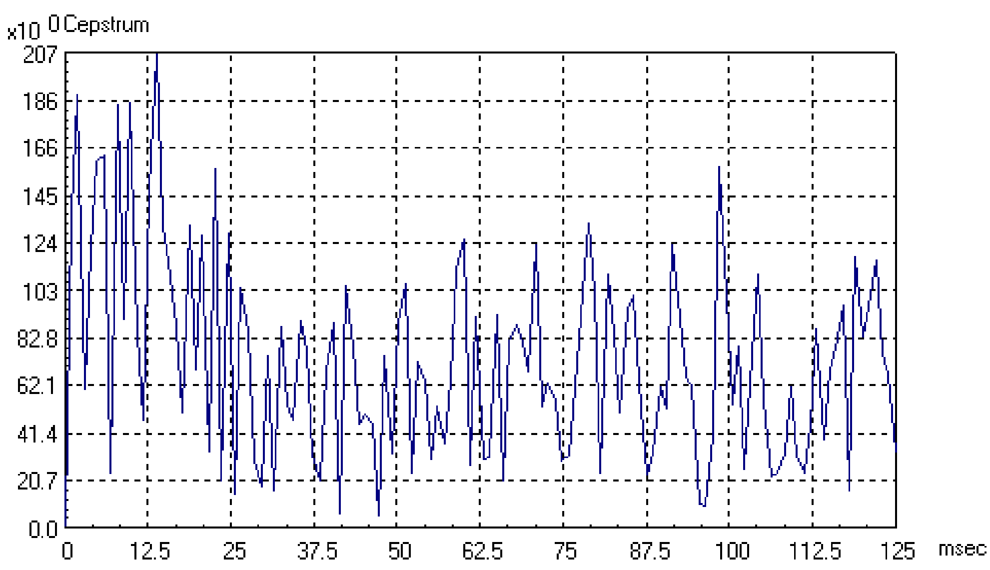


Figure 3.13 Typical Cepstrum Plot

3.8 Envelope Analysis

Envelope detection is widely applied in roller bearing and gearmesh fault analyses. It is a method for intensifying the repetitive components of a dynamic signal. The analysis includes filtration of the signal, squaring the signal and then applying FFT analysis. The low frequency vibration will be rejected during filtration process and only high frequency harmonics are passed. During harmonics-squaring, difference and sum components are created where the difference components fold back in the spectrum while most of the sum components are outside the analysis range.

The envelope method separates a repetitive impulse from a complex vibration signal by using a band pass filter that rejects low frequency components that are synchronous with vibration. Table 3.2 shows filter setting for different machine speeds. Although there are signal enhancements that result from structural resonances, the envelope method is not solely dependent on local resonance to isolate rolling element defect signals. Filter criteria selection is based on suitable rejection of the low frequency sinusoids while optimizing the passband of the defect harmonics

After filtering the vibration signal, the resultant signal is enveloped by means of a circuit that approximately squares the signal as shown in Figure 3.14. The signal is then passed to FFT analyzer to display the FFT spectrum.

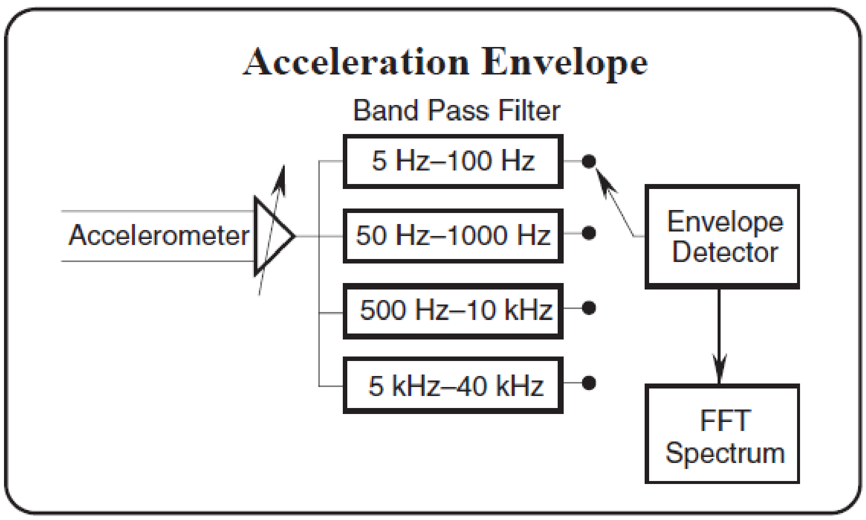


Figure 3.14 Acceleration Enveloping

Table 3.2 Typical Filter Setting for Envelope Analysis

Filters	Frequency Band	Speed Range	Analyzing Range
1	5 – 100 Hz	0 – 50 RPM	0 – 10 Hz
2	50 – 1,000 Hz	25 – 500 RPM	0 – 100 Hz
3	500 – 10,000 Hz	250 – 5,000 RPM	0 – 1,000 Hz
4	5,000 – 40,000 Hz	2,500 – ... RPM	0 – 10,000 Hz

4. Diagnosis of Common Vibration Problems

4.1 Unbalance

Vibration due to unbalance of a rotor is probably the most common machinery defect. It is luckily also very easy to detect and rectify. The International Standards Organization (ISO) define unbalance as:
That condition, which exists in a rotor when vibratory, force or motion is imparted to its bearings as a result of centrifugal forces. It may also be defined as: The *uneven distribution of mass about a rotor's rotating centerline*.
There are two new terminologies used: one is *rotating centerline* and the other is *geometric centerline.* The *rotating centerline* is defined as the axis about which the rotor would rotate if not constrained by its bearings (also called the principle inertia axis or PIA). The *geometric centerline* (GCL) is the physical centerline of the rotor. When the two centerlines are coincident, then the rotor will be in a state of balance. When they are apart, the rotor will be unbalanced. There are three types of unbalance that can be encountered on machines, and these are:

1. Static unbalance (PIA and GCL are parallel)
2. Couple unbalance (PIA and GCL intersect in the center)
3. Dynamic unbalance (PIA and GCL do not touch or coincide).

Static Unbalance

For all types of unbalance, the FFT spectrum will show a predominant 1xRPM frequency of vibration. Vibration amplitude at the 1xRPM frequency will vary proportional to the square of the rotational speed. It is always present and normally dominates the vibration spectrum (Figure 4.1).

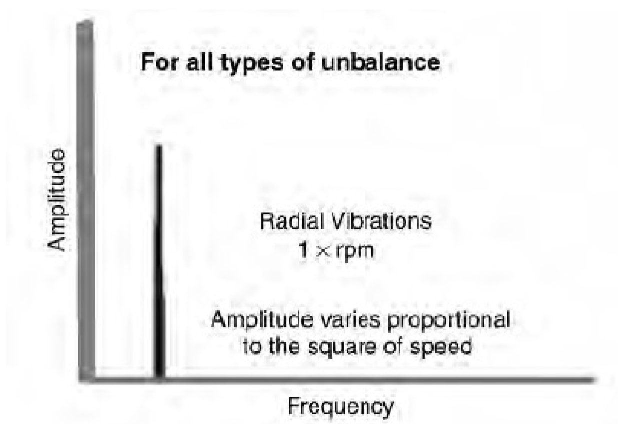


Figure 4.1 FFT Analysis of Unbalance

Static unbalance will be in-phase and steady (15–20°). If the pickup is moved from the vertical (V in the figure) direction to the horizontal (H in the figure) direction, the phase will shift by 90° (±30°). Another test is to move the pickup from one bearing to another in the same plane (vertical or horizontal). The phase will remain the same, if the fault is static unbalance (Figure 4.2).

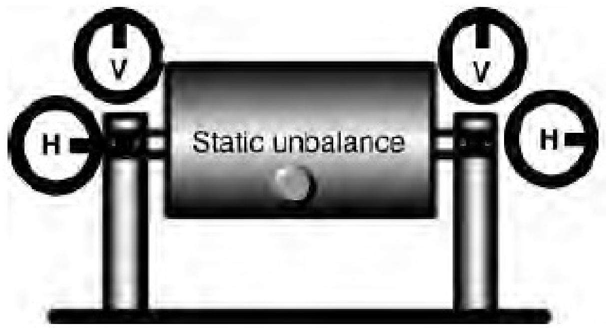


Figure 4.2 Phase Relationship for Static Unbalance

If the machine has no other major defects besides unbalance, the time waveform will be a clean Simple Harmonic Motion waveform with the frequency the same as the running speed.

Couple Unbalance

In a couple unbalance (Figure 4.3) the FFT spectrum again displays a single 1xRPM frequency peak. The amplitude at the 1xRPM varies proportional to the square of speed. This defect may cause high axial and radial vibrations. Couple unbalance tends to be 180° out of phase on the same shaft. Note that almost a 180° phase difference exists between two bearings in the horizontal plane. The same is observed in the vertical plane. It is advisable to perform an operational deflection shape (ODS) analysis to check if couple unbalance is present in a system.

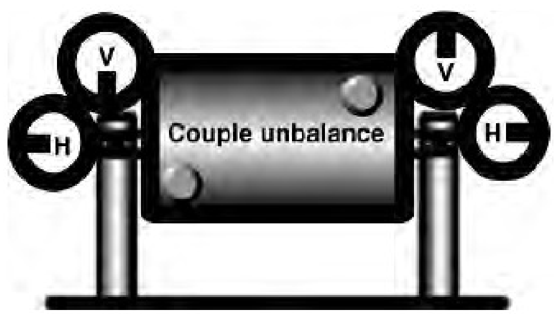


Figure 4.3 Phase relationship – couple unbalance

Unbalance – Overhung Rotors

In this case, the FFT spectrum displays a single 1xRPM peak as well, and the amplitude again varies proportional to the square of the shaft speed. It may cause high axial and radial vibrations. The axial phase on the two bearings will seem to be in phase whereas the radial phase tends to be unsteady. Overhung rotors can have both static and couple unbalance and must be tested and fixed using analyzers or balancing equipment (Figure 4.4).

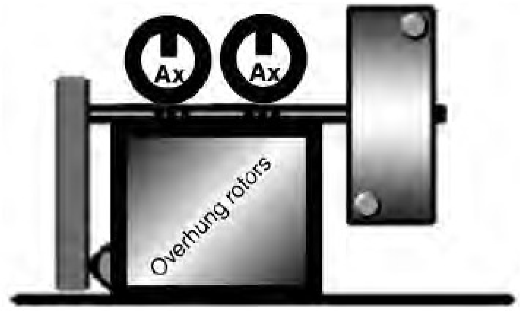


Figure 4.4 A belt-driven fan/blower with an overhung rotor – the phase is measured in the axial direction

4.2 Eccentric Rotor

Eccentricity occurs when the center of rotation is at an offset from the geometric centerline of a sheave, gear, bearing, motor armature or any other rotor. The maximum amplitude occurs at 1xRPM of the eccentric component in a direction through the centers of the two rotors. Here the amplitude varies with the load even at constant speeds as in Figure 4.5.

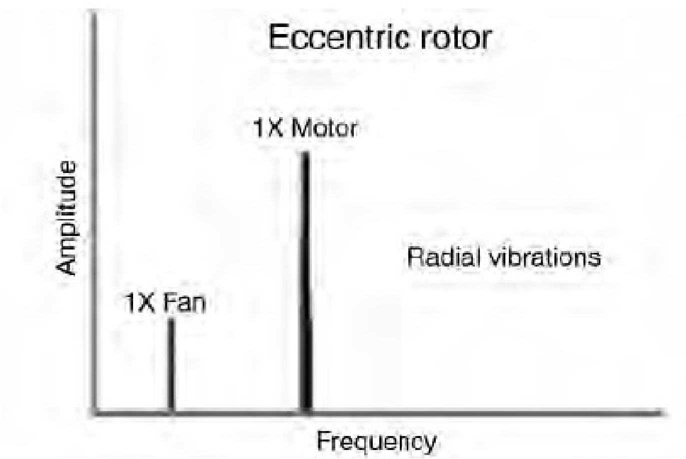


Figure 4.5 A belt-driven fan/blower – vibration graph

In a normal unbalance defect, when the pickup is moved from the vertical to the horizontal direction, a phase shift of 90° will be observed. However in eccentricity, the phase readings differ by 0 or 180° (each indicates straight-line motion) when measured in the horizontal and vertical directions. Attempts to balance an eccentric rotor often result in reducing the vibration in one direction, but increasing it in the other radial direction (depending on the severity of the eccentricity) (Figure 4.6).

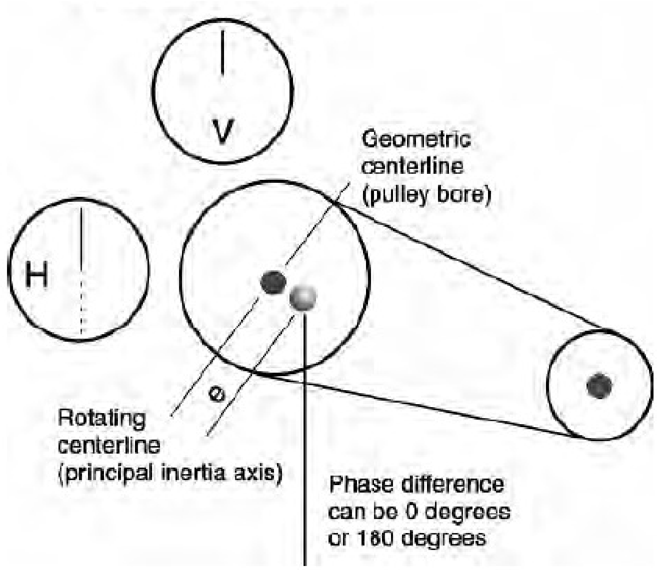


Figure 4.6 Eccentric Rotor

4.3 Bent Shaft

When a bent shaft is encountered, the vibrations in the radial as well as in the axial direction will be high. Axial vibrations may be higher than the radial vibrations. The FFT will normally have 1x and 2x components. If the:

- Amplitude of 1xRPM is dominant then the bend is near the shaft center (Figure 4.7)
- Amplitude of 2xRPM is dominant then the bend is near the shaft end.

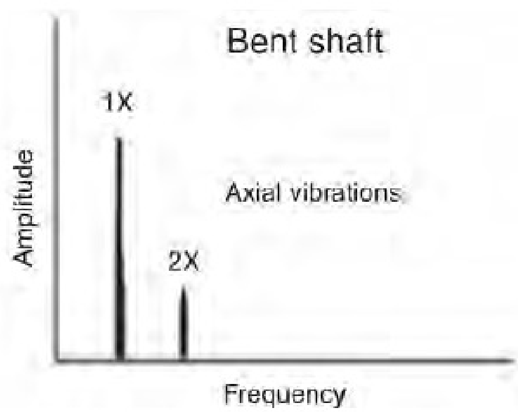


Figure 4.7 An FFT of a bent shaft with bend near the shaft center

The phase of axial vibration for a bent rotor is shown in Figure 4.8. Note that when the probe is moved from vertical plane to horizontal plane, there will be no change in the phase reading (Figure 4.8).

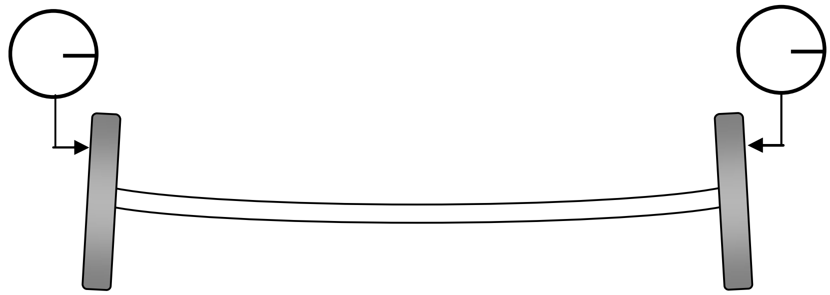


Figure 4.8 Phase of Axial Vibration for a Bent shaft

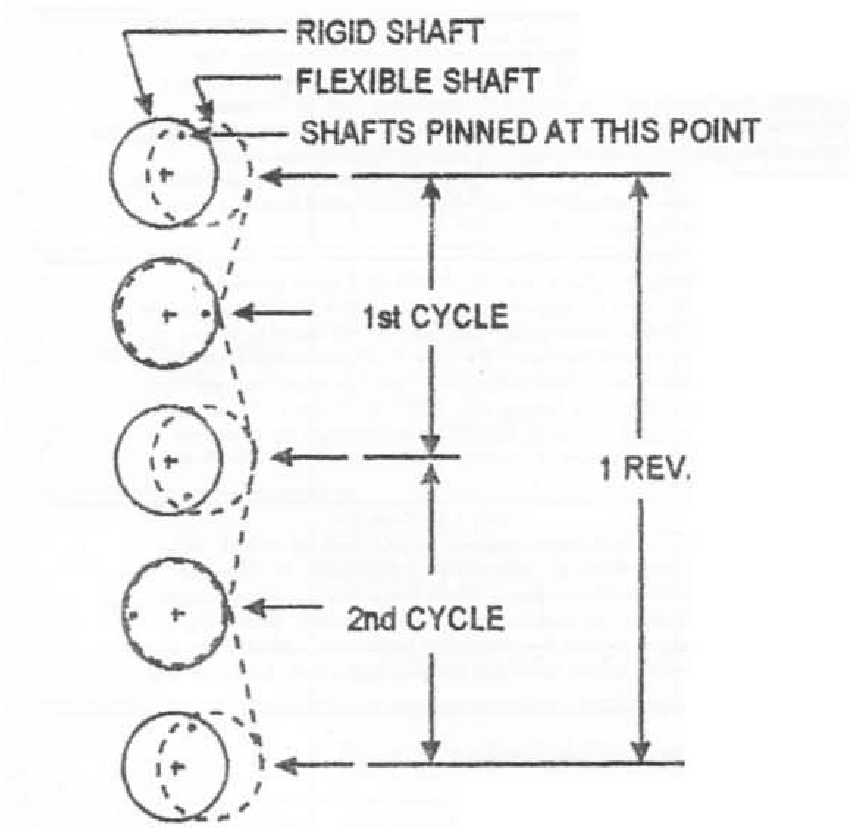
4.4 Misalignment

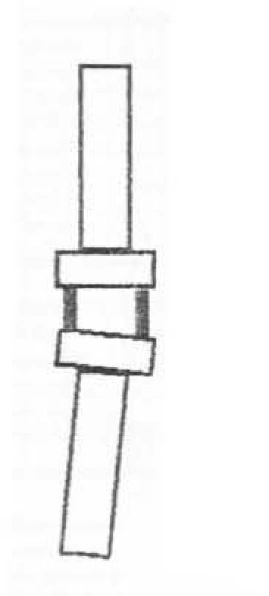
Misalignment, just like unbalance, is a major cause of machinery vibration. Some machines have been incorporated with self-aligning bearings and flexible couplings that can take quite a bit of misalignment. However, despite these, it is not uncommon to come across high vibrations due to misalignment. There are basically two types of misalignment:

1. *Angular misalignment:* the shaft centerline of the two shafts meets at angle with each other
2. *Parallel misalignment:* the shaft centerline of the two machines is parallel to each other and have an offset.

Angular misalignment

As shown in Figure 4.9, angular misalignment primarily subjects the driver and driven machine shafts to axial vibrations at the 1xRPM frequency. Misalignment is rarely seen just as 1xRPM peak. Typically, there will be high axial vibration with both 1xRPM and 2xRPM. However, it is not unusual for 1x, 2x or 3xRPM to dominate. These symptoms may also indicate coupling problems (e.g. looseness) as well (Figure 4.10).





(a) Offset Misalignment (b) Angular Misalignment

Figure 4.9 Misalignment

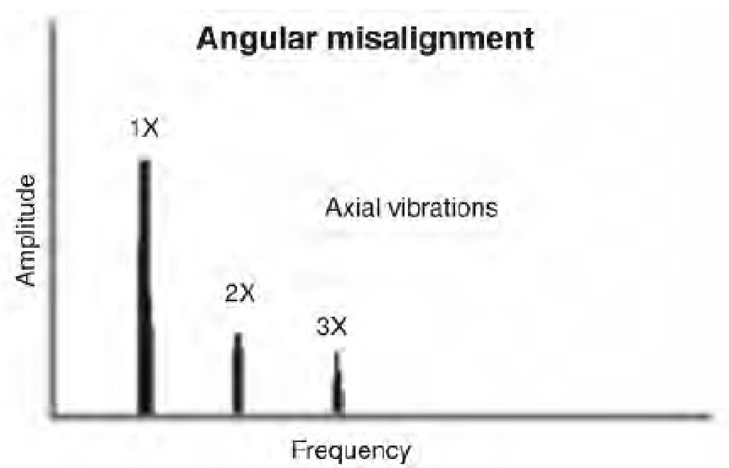


Figure 4.10 FFT Spectrum for Angular Misalignment

Angular misalignment produces in phase axial vibration when measured in two points, on the motor and load across coupling, but considering opposite directions for the pickups during measurements as shown in Figure 4.11.

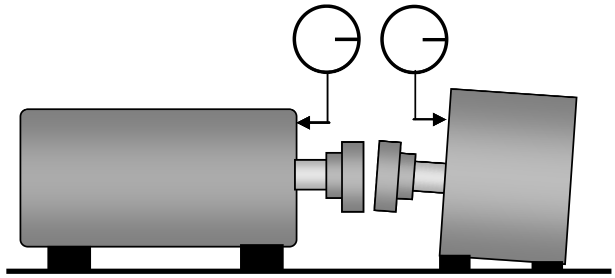


Figure 4.11 Axial Vibration Across Coupling for Angular Misalignment

Parallel Misalignment

Parallel misalignment, as shown in Figure 4.9, results in 2 *hits* per cycle and therefore a 2xRPM vibration in the radial direction. Parallel misalignment has similar vibration symptoms compared to angular misalignment, but shows high radial vibration that approaches a 180° phase difference across the coupling as shown in Figure 4.13. As stated earlier, pure parallel misalignment is rare and is commonly observed to be in conjunction with angular misalignment. Thus, we will see both the 1xRPM and 2xRPM peaks.

When the parallel misalignment is predominant, 2xRPM is often larger than 1xRPM, but its amplitude relative to 1x may often be dictated by the coupling type and its construction. When either angular or parallel misalignment becomes severe, it can generate high amplitude peaks at much higher harmonics (3x to 8x) (Figure 4.12) or even a whole series of high-frequency harmonics. Coupling construction will often significantly influence the shape of the spectrum if misalignment is severe.

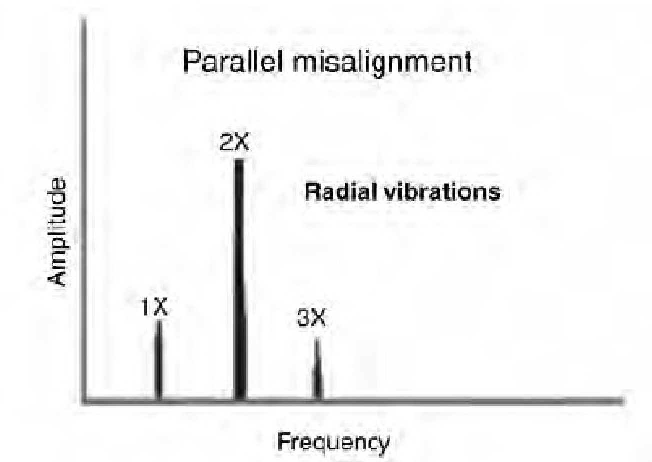


Figure 4.12 FFT Spectrum for Parallel Misalignment

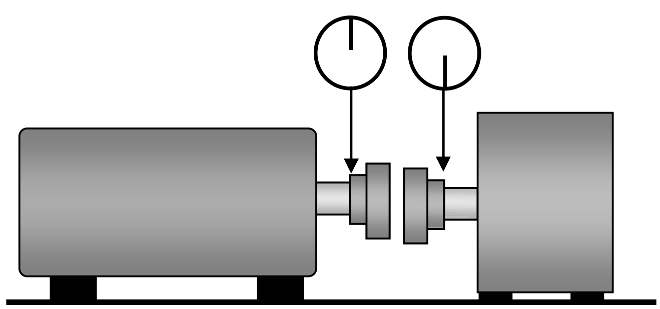


Figure 4.13 Phase Relation of Vertical Vibration Across Coupling for Offset Misalignment

Misalignment vs. Bent Shaft

Often, a bent shaft and dominant angular misalignment give similar FFT spectrums. The vibrations are visible in both the axial and radial vibration measurements. It is only with phase analysis that these problems can be resolved further. In a machine with a bent shaft, a phase difference will be noticed on the two bearings of the same shaft. In the case of misalignment, the phase difference is visible on bearings across the coupling.

4.5 Mechanical Looseness

If we consider any rotating machine, mechanical looseness can occur at three locations:

1. Internal assembly looseness
2. Looseness at machine to base plate interface
3. Structure looseness.

Internal assembly looseness

This category of looseness could be between a bearing liner in its cap, a sleeve or rolling element bearing, or an impeller on a shaft. It is normally caused by an improper fit between component parts, which will produce many harmonics in the FFT due to the nonlinear response of the loose parts to the exciting forces from the rotor. A truncation of the time waveform occurs, causing harmonics. The phase is often unstable and can vary broadly from one measurement to the next, particularly if the rotor alters its position on the shaft from one start-up to the next.

Mechanical looseness is often highly directional and may cause noticeably different readings when they are taken at 30° increments in the radial direction all around the bearing housing. Also note that looseness will often cause sub-harmonic multiples at exactly ½× or ⅓× rpm (e.g. ½×, 1½×, 2½× and further) (Figures 4.14).

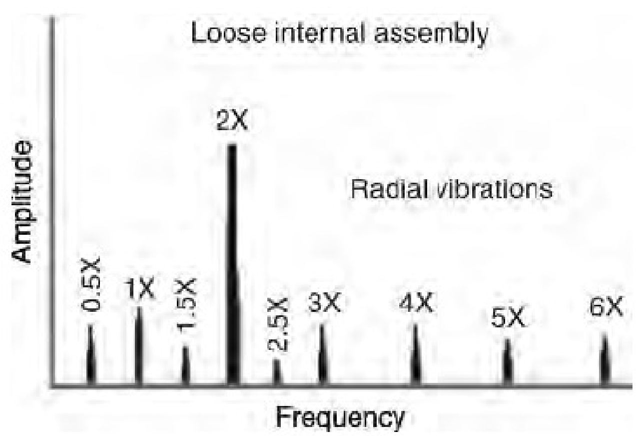


Figure 4.14 FFT of Loose Internal Assembly

Looseness between Machine and Base-plate

This problem is associated with loose pillow-block bolts, cracks in the frame structure or the bearing pedestal. Figures 4.15 and 4.16 make it evident how higher harmonics are generated due to the rocking motion of the pillow block with loose bolts.

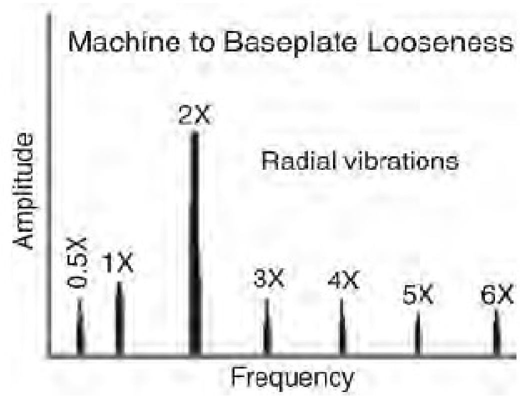


Figure 4.15 FFT of Mechanical Looseness

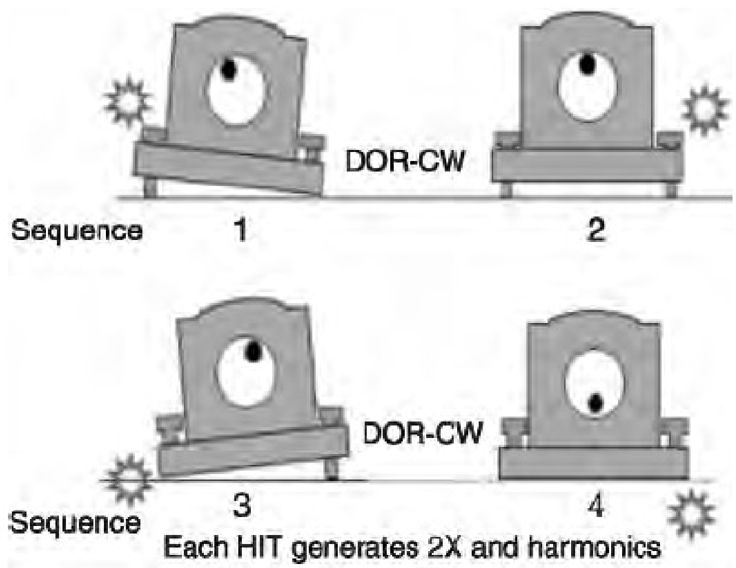


Figure 4.16 Mechanical Looseness Illustrations

Structure Looseness

This type of looseness is caused by structural looseness or weaknesses in the machine's feet, baseplate or foundation. It can also be caused by deteriorated grouting, loose hold-down bolts at the base and distortion of the frame or base (known as 'soft foot') (Figure 4.17). Phase analysis may reveal approximately 180° phase shift between vertical measurements on the machine's foot, baseplate and base itself (Figure 4.17). Soft foot tends to amplify vibration problem due to reduced stiffness as shown in Figure 4.18 for unbalance condition.

When the soft foot condition is suspected, an easy test to confirm for it is to loosen each bolt, one at a time, and see if this brings about significant changes in the vibration. In this case, it might be necessary to re-machine the base or install shims to eliminate the distortion when the mounting bolts are tightened again.

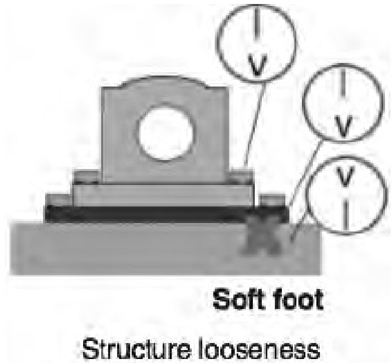


Figure 4.17 Structural Looseness (Soft Foot)

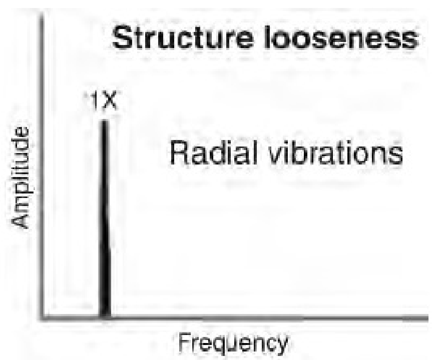


Figure 4.18 Soft Foot Amplifies Vibration Problem

4.6 Resonance

Any object has a natural frequency which is determined by its characteristics of mass, stiffness and damping. If a gong strikes a bell, the bell rings at its own characteristic frequency known as its natural frequency. The gong-striking event is *forced vibration*, whereas the ringing of bell is *free vibration*. A free vibration at a natural frequency is called resonance.

There is a simple method to find the natural frequency of any object or system called the *bump test*. With this method, a vibration sensor is fixed to the body whose natural frequency is required. Using an impact hammer, a blow is struck on the body and the time waveform or FFT is collected. The dominant frequency observed in the two graphs is the natural frequency of the body. Figures 4.19 and 4.20 show the time waveform and the FFT spectrum of a bump test conducted on a metal study table, respectively. As seen in the time waveform, the impact occurs at approximately 100 ms after data collection was initiated. Directly after the impact, the body exhibits free vibrations at its own natural frequency. The amplitude of the vibration reduces logarithmically due to damping effects. The period between 500 ms and 1 s is long enough to count the number of cycles. The calculation indicates that the natural frequency is approximately 990 cpm.

To obtain the FFT, the data collector was reset and another impact was made on the table with a hammer. The collected spectrum shows a dominant peak at 1046 cpm. This is close to the value calculated before with the time waveform. The bump test is simple and used extensively in practice. It is a quick and accurate way of finding the resonance frequencies of structures and casings. It is tempting to use the bump test on a spare pump or other rotors not supported on bearings to obtain an estimate of their critical speeds. Take note that this can be very inaccurate. For example, the critical speed of rotors with impellers in a working fluid and supported by their bearings differs vastly from the critical speed obtained using a bump test off-line on the rotor.

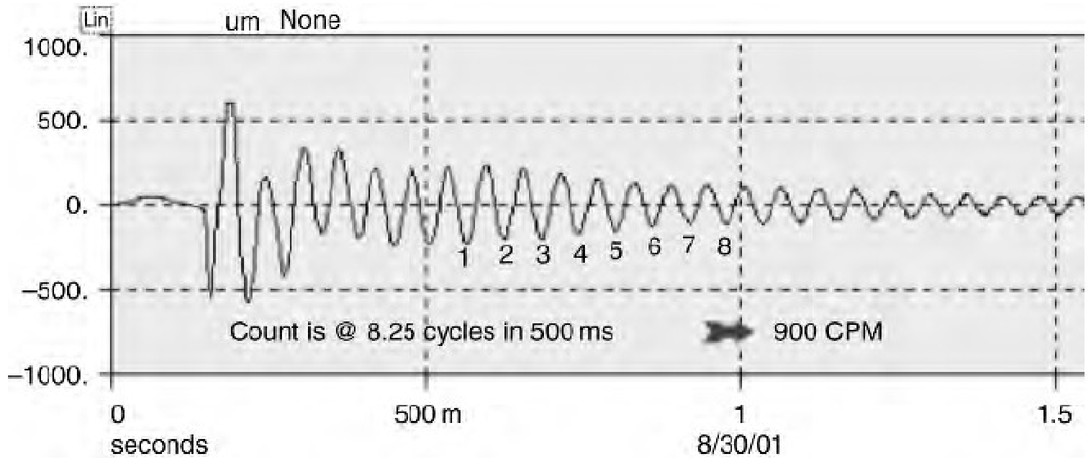


Figure 4.19 Time Waveform of a Bump Test

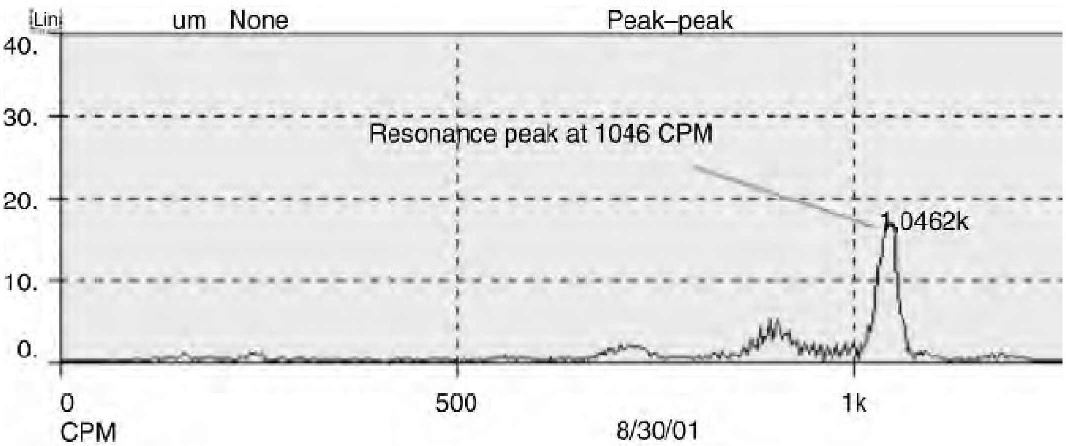


Figure 4.20 FFT Spectrum of a Bump Test

Assume that a multistage pump rotor has a natural frequency of 2500 cpm when pumping a fluid. Assume that the rotor has a slight unbalance, which generates tolerable amplitudes of vibration at 1×RPM. In this example, the unbalance causes the forced vibration frequency at 1× RPM. When the pump is started, the speed begins to increase and along with it also the amplitude and frequency of the vibration due to unbalance. At a particular instant, the forced frequency of vibration due to unbalance will be 2500 cpm. This frequency also happens to be the natural frequency of the rotor.

Whenever the forced vibration frequency matches the natural frequency of a system, the amplitude rises significantly, much higher than expected compared to unbalance effects. This condition is called a *critical speed.* Rotor critical speeds are confirmed using a Bode plot as shown in Figure 4.21. As the rotor approaches its critical speed, the amplitude rises. It reaches a maximum and then drops again. The phase changes steadily as well and the difference is 90° at the critical speed and nearly 180° when it passed through resonance.

The high-vibration amplitudes at critical speeds can be catastrophic for any system and must be avoided at all costs. Besides the example of the natural frequency of a rotor, structural resonance can also originate from support frame foundations, gearboxes or even drive belts. Natural frequencies of a system cannot be eliminated, but can be shifted to some other frequency by various methods. Another characteristic of natural frequencies is that they remain the same regardless of speed, and this helps to facilitate their detection.

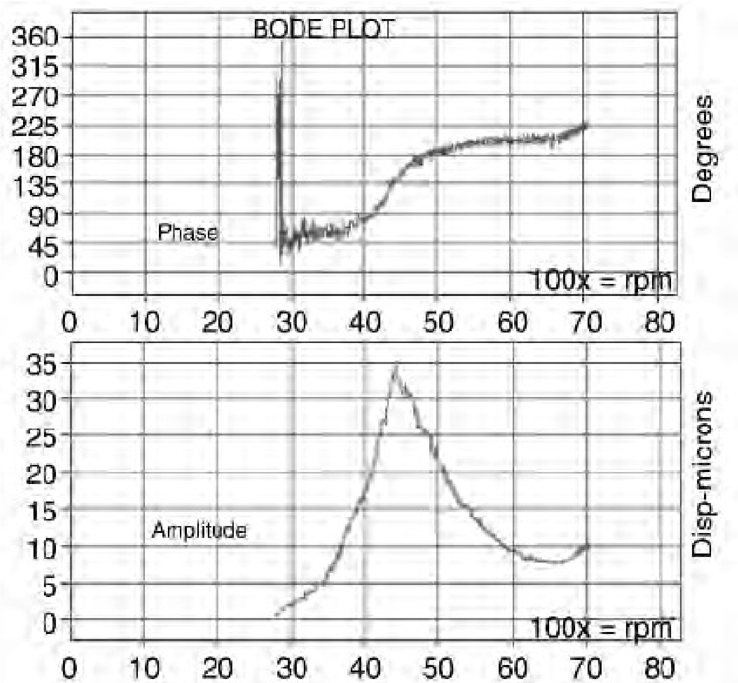


Figure 4.21 Bode Plot for a Rotor Response

4.7 Electrical Problems

Vibrations of electrical machines such as motors, generators and alternators can be either mechanical or electrical in nature. We have discussed most common mechanical problems. Electrical problems also appear in the vibration spectrum and can provide information about the nature of the defects. Electrical problems occur due to unequal magnetic forces acting on the rotor or the stator. These unequal magnetic forces may be due to:

- Open or short windings of rotor or stator
- Broken rotor bar
- Unbalanced phases
- Unequal air gaps.

Generally, the vibration pattern emerging due to the above-mentioned electrical problems will be at 1× RPM and will thus appear similar to unbalance. A customary technique to identify these conditions is to keep the analyzer capturing the FFT spectrum in the *live* mode and then switching off the electrical power. If the peak disappears instantly, the source is electrical in nature. On the other hand, if there is gradual decrease in the 1× amplitude it is more likely to be a mechanical problem. This technique requires caution. If there is a time lag in the analyzer itself, it may delay the drop in vibration amplitude. It is also possible that a resonance frequency may drop quickly as the speed changes.

Induction motors, which have electrical problems, will cause the vibration amplitude to hunt or swing in a cyclic manner. The phase readings will show similar cycles too. Under a stroboscope, the reference mark will move back and forth. The swinging amplitudes in induction motor applications are due to two dominant frequencies that are very close to one another. They continuously add and subtract to one another in a phenomenon known as *beats*. It can also possibly be a single frequency whose amplitude is modulating. In fact, hunting amplitudes are the first indication of a possible electrical problem in the motor. Understanding the nature of these vibrations can assist in identifying the exact defects in an electrical machine. The following are some terms that will be required to understand vibrations due to electrical problems:

$$F_L = Electrical\ Line\ Frequnecy\ (50/60Hz)$$

$$F_S = Slip\ Frequnecy = \frac{2 \times F_L}{P} - RPM$$

$$F_P = Pole\ Pass Frequency\ = P \times F_S$$

Where P is the number of poles.

Rotor Problems

Normally, four kinds of problems can occur within the rotor:

1. Broken rotor bars
2. Open or shorted rotor windings
3. Bowed rotor
4. Eccentric rotor.

Along with the stator is a rotor, which is basically an iron following the rotating magnetic field. As the magnetic field sweeps across the conductor, it creates a voltage across the length of the rotor bar. If the bar is open-circuited, no current flows and no forces are generated. When the bar is short-circuited, a current flows. This current is proportional to the speed at which the field cuts through the conductor and the strength of the field. The field interacts with the stator field to generate a force on the rotor bar. If everything else remains the same, an equal and opposite force on the opposite side of the rotor will develop. These two forces generate the torque that drives the load. In case anything disrupts the current or magnetic fields on either side of the rotor, the two forces will become unequal. This results in a radial force, which is the cause for vibration.

A cracked or broken bar can cause this category of unbalanced forces. The forces rotate with the rotor with a constant load plus a load that varies with 2× slip. Therefore, the force acting on the bearings will have frequency components at 1× RPM and 1× RPM ± F_P. Thus, broken or cracked rotor bars or shorting rings, bad joints between rotor bars and shorting rings and shorted rotor laminations will produce high 1× running speed vibration with pole pass frequency sidebands. In addition, cracked rotor bars will often generate F_P sidebands around the 3rd, 4th and 5th running speed harmonics, see Figures 4.22 and 4.23.

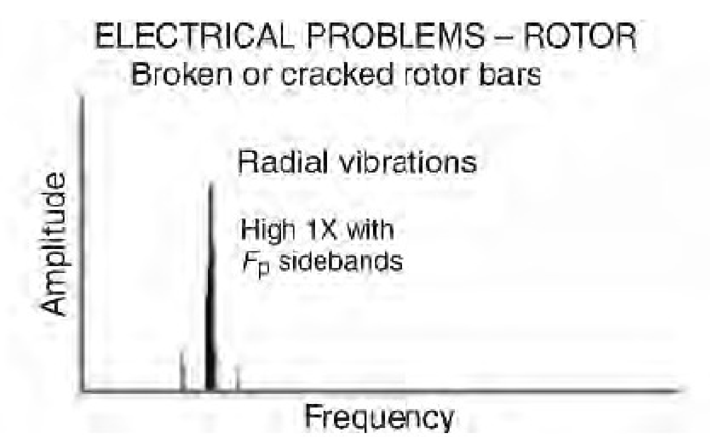


Figure 4.22 Electrical Problem with high 1xRPM and F_P Sidebands

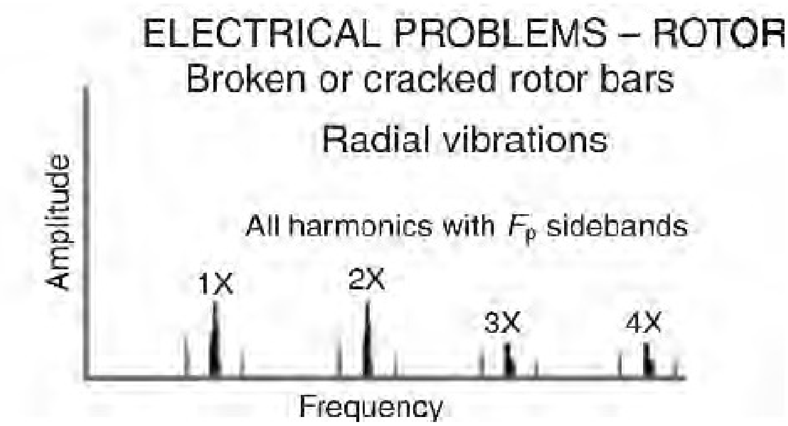


Figure 4.23 1xRPM and Harmonics with F_P Sidebands

Loose rotor bars are indicated by 2× line frequency ($2F_L$) sidebands surrounding the rotor bar pass frequency (RBPF) and/or its harmonics (Figure 4.24).

RBPF = number of rotor bars × RPM

It may often cause high levels at 2× RBPF with only small amplitude at 1× RBPF.

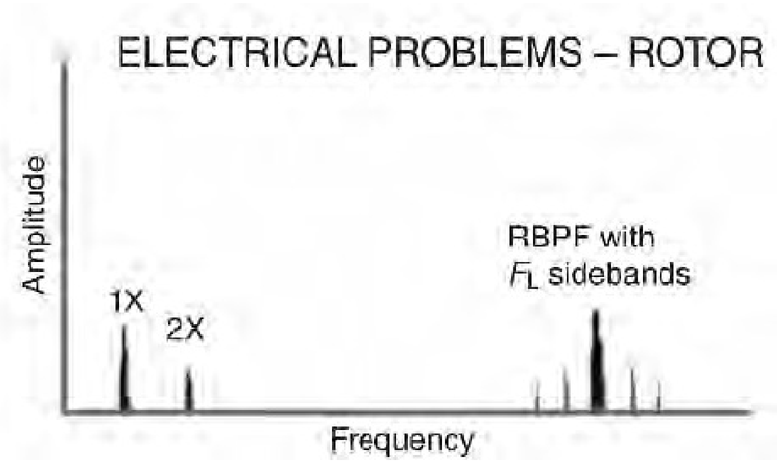


Figure 4.24 Loose Rotor Bars Typical FFT Spectrum

Unequal air gap will produce unbalanced magnetic force which swings in direction twice per line cycle. It is assumed that the eccentricity of the rotor will line up with the magnetic field. The closer side of the rotor will be respectively attracted to the positive pole and to the negative pole; thus the force will vary twice during a single current cycle. This can affect the bearings, and therefore it can modulate any other frequency present in the system. These effects generally cause sidebands of ±2× slip frequency around the 1× RPM frequency caused by unbalance. Eccentric rotors produce a rotating variable air gap between the rotor and stator, which induces pulsating vibrations (it is a beat phenomenon between two frequencies; one is $2F_L$ and is the closest running speed harmonic). This may require a high resolution spectrum to separate the $2F_L$ and the running speed harmonic. Eccentric rotors generate $2F_L$ surrounded by pole pass frequency sidebands (F_P as well as F_P sidebands around 1× RPM). The pole-pass frequency F_P itself appears at a low frequency (Figure 4.25).

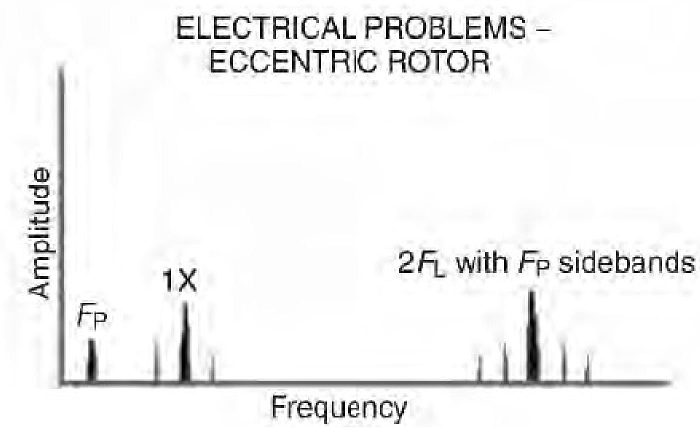


Figure 4.25 Eccentric Rotor

Stator Problems

An induction motor comprises a set of stator coils, which generate a rotating magnetic field. The magnetic field causes alternating forces in the stator. If there is any looseness or a support weakness in the stator, each pole pass gives it a tug. This generates a 2× line frequency (2 F_L) also known as *loose iron*. Shorted stator laminations cause uneven and localized heating, which can significantly grow with time.

Stator problems generate high vibration at $2F_L$. Eccentricity produces uneven stationary air gaps between the rotor and the stator, which produce very directional vibration. Differential air gaps should not exceed 5% for induction motors and 10% for synchronous motors (Figure 4.26).

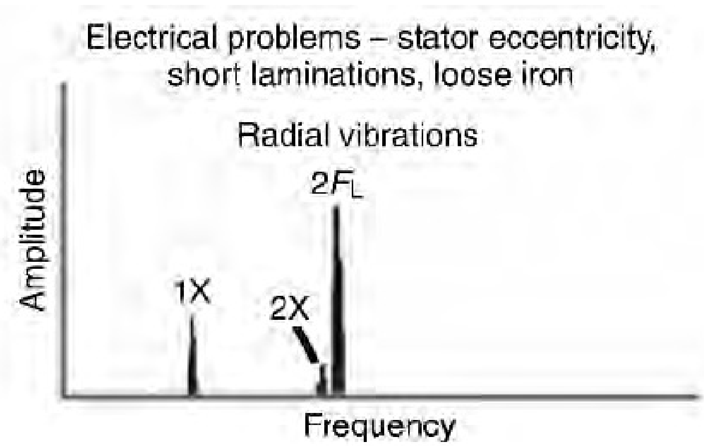


Figure 4.26 Stator Defects

Phasing Problem

Phasing problems due to loose or broken connectors can cause excessive vibration at 2 F_L, which will have sidebands around it spaced at ⅓ rd of the line frequency (⅓F_L). Levels at 2 F_L can exceed 25 mm/s (1.0 in./s) if left uncorrected. This is particularly a problem if the defective connector is sporadically making contact and not periodically (Figure 4.27).

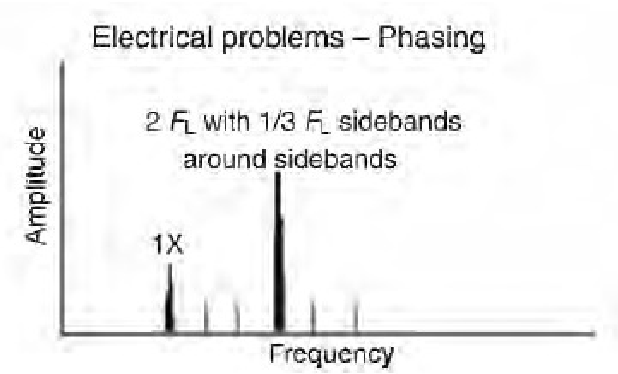


Figure 4.27 FFT Spectrum for Phasing Problem

4.8 Pumps Related Problems

Hydrodynamic Forces (Vanes Pass Frequency)

Blade pass or vane pass frequencies (Figure 4.28) are characteristics of pumps and fans and resulting from hydraulic or hydrodynamic forces. Usually it is not destructive in itself, but can generate a lot of noise and vibration that can be the source of bearing failure and wear of machine components.

Blade (Vanes) pass frequency (BPF, VPF) = number of blades (or vanes) × RPM

This frequency is generated mainly due to the gap problems between the rotor and the stator. Large amplitude BPF (and its harmonics) can be generated in the pump if the gap between the rotating vanes and the stationary diffusers is not kept equal all the way around.

In centrifugal pumps, the gap between the impeller tip and the volute tongue or the diffuser inlet is a certain percentage (in the region of 4–6% of the impeller diameter), depending on the speed of the pump. If the gap is less than the recommended value, it can generate a noise that resembles cavitation. However, an FFT plot will immediately highlight the vane pass frequency of the impeller. Also, the BPF (or its harmonics) sometimes coincides with a system natural frequency, causing high vibrations. A high BPF can be generated if the wear ring seizes on the shaft or if the welds that fasten the diffusers fail. In addition, a high BPF can be caused by abrupt bends in line-work (or duct), obstructions which disturb the flow path, or if the pump or fan rotor is positioned eccentrically within the housing.

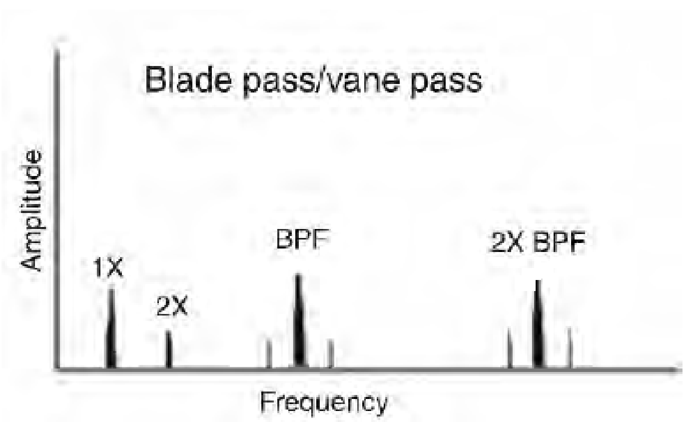


Figure 4.28 Hydrodynamic Forces

Cavitation and Recirculation

Cavitation normally generates random, high-frequency broadband energy, which is sometimes superimposed with the blade pass frequency harmonics. Gases under pressure can dissolve in a liquid. When the pressure is reduced, they bubble out of the liquid. In a similar way, when liquid is sucked into a pump, the liquid's pressure drops. Under conditions when the reduced pressure approaches the vapor pressure of the liquid (even at low temperatures), it causes the liquid to vaporize. As these vapor bubbles travel further into the impeller, the pressure rises again causing the bubbles to collapse or implode.

This implosion has the potential to disturb the pump performance and cause damage to the pump's internal components. This phenomenon is called *cavitation*. Each implosion of a bubble generates a kind of impact, which tends to generate high-frequency random vibrations (Figure 4.29) in the range 9–30 x RPM. Cavitation can be quite destructive to internal pump components if left uncorrected. It is often responsible for the erosion of impeller vanes. Cavitation often sounds like 'gravel' passing through the pump. Measurements to detect cavitation are usually not taken on bearing housings, but rather on the suction piping or pump casing.

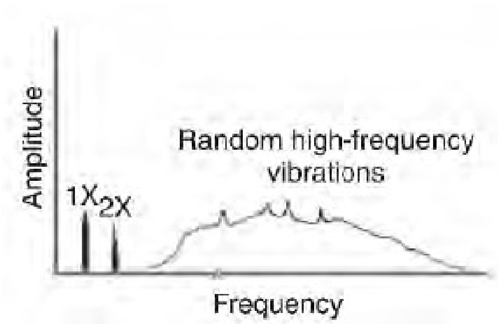


Figure 4.29 Cavitation Problem

4.9 Fans and Blowers

Similar to hydrodynamic forces mentioned above in pumps, blowers and fans may produce aerodynamic forces at frequency equal to Blade Pass Frequency.

In blowers flow turbulence (Figure 4.30) often occurs due to variations in pressure or velocity of the air passing through the fan or connected line-work. In fans, duct-induced vibration due to stack length, ductwork turns, unusual fan inlet configuration and other factors may be a source of low-frequency excitation. This flow disruption causes turbulence, which will generate random, low-frequency vibrations, typically in the range of 20–2000 cpm.

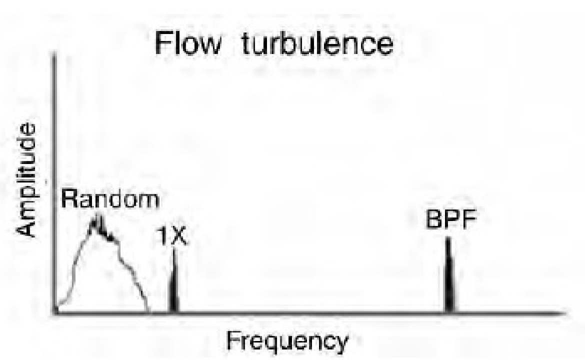


Figure 4.30 Flow Turbulence

4.10 Compressors Problems

In compressors, similar to hydraulic pump cavitation, surge is the rapid backward movement of a specific volume of air through the compressor. Surge will occur when forward flow through the compressor can no longer be maintained due to an increase in pressure across the compressor, and a momentary flow reversal occurs. Surging is most often caused by exceeding the design discharge pressure (high discharge pressure). Choke, on the other hand, is simply a very high flow point on the compressor's map where the total amount of energy available to the impeller is utilized for pumping gas, but at a very low head or pressure ratio. Mild surge will reveal vibration at blade-pass frequency or its multiples sometimes, while severe surge will result in random vibration at broad frequency range which is system and operating conditions dependent. Vibration characteristics of choking will be essentially similar to those encountered in surging.

When a compressor is operated away from its design point, the gas flow into the aerodynamic components (impellers, blades, diffusers, etc.) deviates from its design direction. If the angle of deviation (or incidence angle) is large, flow separation occurs. At higher incidence angles, the flow fully separates at the impeller leading edge or diffuser inlet, and the flow is said to be stalled. The rotating stall is a special form of stall, where one or multiple flow regions in the diffuser (or impeller) are stalled but where other regions of the same impeller or diffuser are not stalled yet. The stall regions usually travel in the direction of the rotation at a speed that is fractionally lower than the rotating speed of the compressor. Stall and flow separation may be precursors to surge, but not necessarily so. Rotating stall events also increase the measured vibration levels, but at a distinct frequency that is lower than the shaft rotating frequency (typically between 10-50%).

4.11 Reciprocating Machines Problems

Reciprocating machines such as IC engines and compressors will have inherent vibrations which are the results of inertia of the reciprocating parts plus varying pressure which causes torque variation. The vibration frequencies encountered are 1x and 2xRPM. However, higher order frequencies are also common with some designs, depending on the number of pistons and relation between them. The following table lists the major vibration problems and their causes.

Table 4.1 Predominant Frequency Components of Reciprocating Machines

Description	Predominant Frequencies	Cause	Remedy
Inertia forces	are *1x* and *2xRPM*	Primary and secondary inertia forces of piston and connecting rod	Proper design and balance masses
Power pulses	$\frac{N}{2} \times RPM$ for 4-stroke engines $N \times RPM$ for 2-stroke engines	Engine power cycle	Increase number of cylinders
Misfiring piston	$0.5 \times RPM$ for 4-stroke engines $1 \times RPM$ for 2-stroke engines	Negative power stroke	Repair engine
Worn Connecting rod bearings	$2 \times RPM$	Bearing impact when piston changes direction	Repair engine
Worn crankshaft main bearings	Same as power pulses	Bearing impact each power stroke	Repair engine

Piston slap	*2 x RPM*	Under heavy load, force component perpendicular to connecting rod	Repair engine
Unbalance inertia forces	*2 x RPM and multiples*	Secondary inertia forces due to improper correction weights	Selection of proper replacement parts

5. Diagnosis of Special Parts Problems

5.1 Journal Bearing Problems

Excessive Bearing Clearance

Late stages of journal bearing wear normally display a whole series of running speed harmonics, which can be up to 10× or 20×RPM. The FFT spectrum looks very much like that of mechanical looseness. Even minor unbalance or misalignment can cause higher vibration amplitudes compared with bearings having a normal clearance with the journal. This is due to a reduction in the oil film stiffness on account of higher clearances (Figure 5.1).

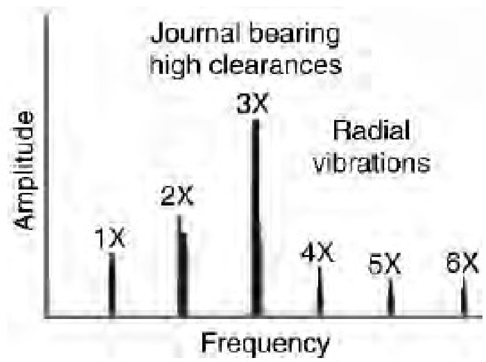


Figure 5.1 Excessive Bearing Clearance

Oil Whirl

Oil whirl is an oil film-excited vibration. It is known to occur on machines equipped with pressure-lubricated journal bearings operating at high speeds (beyond their critical speed). Consider a shaft rotating in a bearing at speed *N*. The bearing speed is zero. The oil film is wedged between the shaft and the bearing and should ideally rotate at a speed of 0.5× RPM. However, some frictional losses cause the oil film to rotate at 0.42–0.48× RPM. Under normal circumstances, the oil film pushes the rotor at an angle (5 o'clock if the shaft is rotating CCW – see Figure 5.2). An eccentric crescent-shaped wedge is created that has sufficient pressure to keep the rotor in the 'lifted' position. Under normal conditions, the system is in equilibrium and there are no vibrations.

Some conditions would tend to generate an oil film pressure in the wedge much higher than required to just hold the shaft. These conditions can cause an increase in bearing wear resulting in the shaft to have lower eccentricity causing a reduction in stiffness, oil pressure or a drop in oil temperature. In these cases, the oil film would push the rotor to another position in the shaft. The process continues over and over and the shaft keeps getting pushed around within the bearing. This phenomenon is called oil whirl. This whirl is inherently unstable since it increases centrifugal forces that will increase the whirl forces.

Oil whirl can be minimized or eliminate4d by changing the oil velocity, lubrication pressure and external pre-loads. Oil whirl instability occurs at 0.42–048× rpm and is often quite severe. It is considered excessive when displacement amplitudes exceed 50% of the bearing clearances.

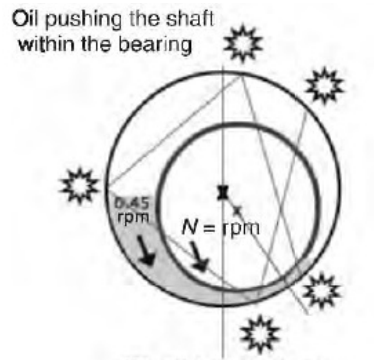


Figure 5.2 Oil Whirl

Dry Whirl

Sometimes inadequate or improper lubrication can also cause vibrations in a bearing. This is because lack of lubrication results in friction between the shaft and the bearing. The friction force will also tend to excite other parts of the machine. This vibration is similar to the experience of moving a moist finger over a glass pane. The vibration caused by this phenomenon is known as dry whirl. The vibration is generally at high frequencies, and harmonics may not be present. Phase will not provide any meaningful information.

5.2 Roller Bearing Problems

A rolling element bearing comprises of inner and outer races, a cage and rolling elements. Defects can occur in any of the parts of the bearing and will cause high-frequency vibrations. In fact, the severity of the wear keeps changing the vibration pattern. In most cases, it is possible to identify the component of the bearing that is defective due to the specific vibration frequencies that are excited. Raceways and rolling element defects are easily detected. However, the same cannot be said for the defects that crop up in bearing cages. Though there are many techniques available to detect where defects are occurring, there are no established techniques to predict when the bearing defect will turn into a functional failure.

In an earlier topic dealing with enveloping/demodulation, we saw how bearing defects generate both the bearing defect frequency and the ringing random vibrations that are the resonant frequencies of the bearing components. Bearing defect frequencies are not integrally harmonic to running speed. However, the following formulas are used to determine bearing defect frequencies. There is also bearings database available in the form of commercial software that readily provides the values upon entering the requisite bearing number.

$$BPFI = \frac{N}{2}\left(1+\frac{D_b}{D_p}\cos\theta\right)\times RPM$$

$$BPFO = \frac{N}{2}\left(1-\frac{D_b}{D_p}\cos\theta\right)\times RPM$$

$$FTF = \frac{1}{2}\left(1-\frac{D_b}{D_p}\cos\theta\right)\times RPM$$

$$BSF = \frac{D_p}{2D_b}\left(1-\left(\frac{D_b}{D_p}\cos\theta\right)^2\right)\times RPM$$

Where D_b : ball or roller diameter

D_p : pitch circle diameter of the bearing

N : number of balls

θ : contact angle

BPFI : Ball Pass Frequency (Inner Race)

BPFO : Ball Pass Frequency (Outer Race)

FTF : Fundamental Train Frequency (Cage)

BSF : Ball Spin Frequency (Rolling Elements)

Bearing deterioration progresses through four stages. During the initial stage, it is just a high-frequency vibration, after which bearing resonance frequencies are observed. During the third stage, discrete frequencies can be seen, and in the final stage high-frequency random noise is observed, which keeps broadening and rising in average amplitude with increased fault severity.

Stage 1 of Bearing Defect

The FFT spectrum for bearing defects can be split into four zones (A, B, C and D), where we will note the changes as bearing wear progresses. These zones are described as (Figure 5.3):

Zone A: machine rpm and harmonics zone
Zone B: bearing defect frequencies zone (5–30 kcpm)
Zone C: bearing component natural frequencies zone (30–120 kcpm)
Zone D: high-frequency-detection (HFD) zone (beyond 120 kcpm or 20kHz).

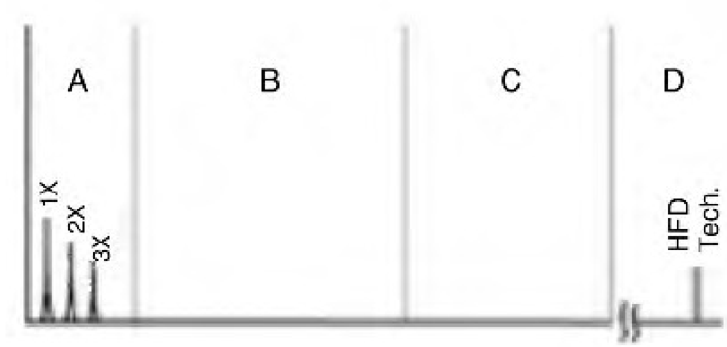


Figure 5.3 High HFD due to Bearing Defect in Stage 1

The first indications of bearing wear show up in the ultrasonic frequency ranges from approximately 2–60 kHz (120–3600 kcpm). These are frequencies that are evaluated by high-frequency detection techniques such as gSE (Spike Energy), SEE, PeakVue, SPM and others. The raceways or rolling elements of the bearing do not have any visible defects during the first stage. The raceways may no longer have the shine of a new bearing and may appear dull gray.

Stage 2 of Bearing Defect

In the following stage, the fatigued raceways begin to develop minute pits. Rolling elements passing over these pits start to generate the ringing or the bearing component natural frequencies that predominantly occur in the 30–120 kcpm range (500 – 2000 Hz). Depending on the severity, it is possible that the sideband frequencies (bearing defect frequency ± rpm) appear above and below the natural frequency peak at the end of stage two. The high-frequency detection (HFD) techniques may double in amplitude compared to the readings during stage one. See Figure 5.4.

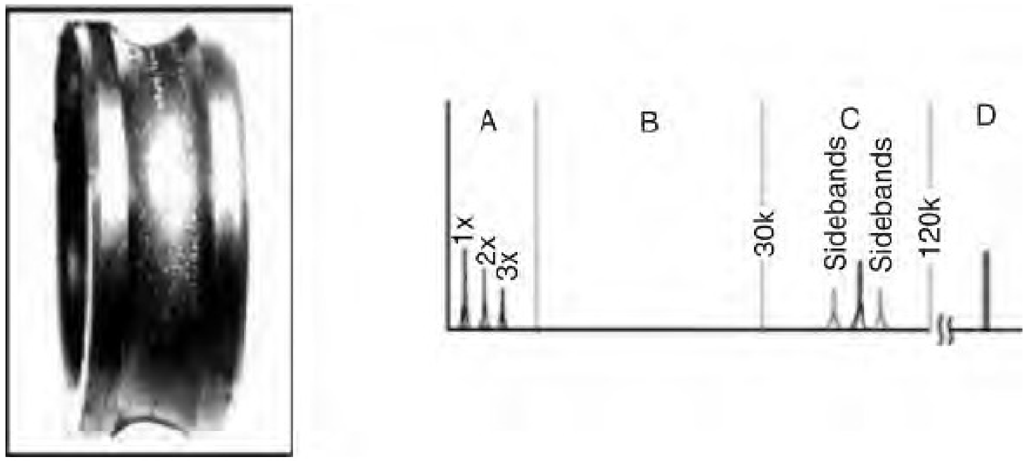


Figure 5.4 Bearing Defect in Stage 2, Components Natural Frequencies plus HFD

Stage 3 of Bearing Defect

As we enter the third stage (Figure 5.5), the discrete bearing frequencies and harmonics are visible in the FFT. These may appear with a number of sidebands. Wear is usually now visible on the bearing and may expand through to the edge of the bearing raceway. The minute pits of the earlier stage are now developing into bigger pits and their numbers also increase. When well-formed sidebands accompany any bearing defect frequency or its harmonics, the HFD components have again almost doubled compared to stage two. It is usually advised to replace the bearing at this stage. Some studies indicate that after the third stage, the remaining bearing life can be 1 h to 1% of its average life.

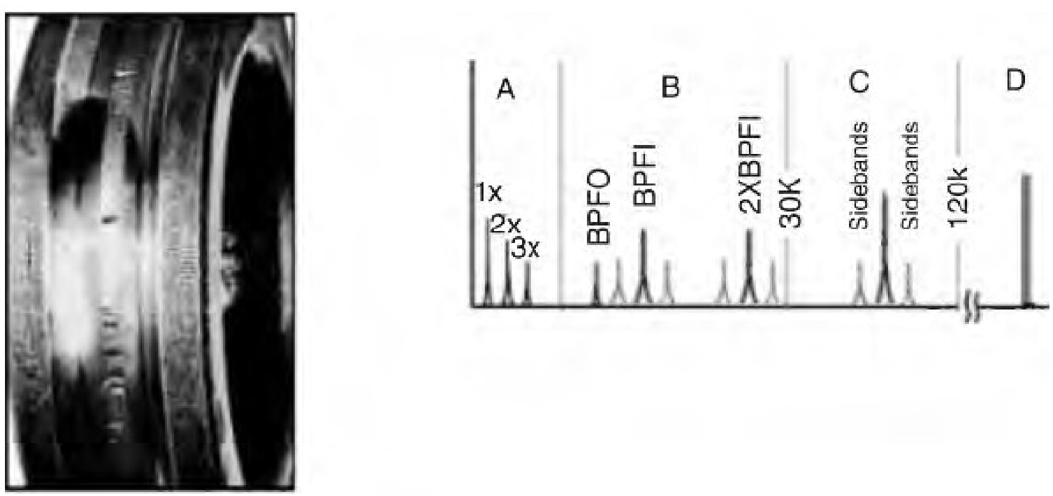


Figure 5.5 Bearing Defect in Stage 3, Defect, Components Natural and HFD Frequencies

Stage 4 of Bearing Defect

In the final phase (Figure 5.6), the pits merge with each other, creating rough tracks and spalling of the bearing raceways or/and rolling elements. The bearing is in a severely damaged condition now. Even the amplitude of the 1× RPM component will rise. As it grows, it may also cause growth of many running speed harmonics. It can be visualized as higher clearances in the bearings allowing a higher displacement of the rotor. Discrete bearing defect frequencies and bearing component natural frequencies actually begin to merge into a random, broadband high-frequency 'noise floor'. Initially, the average amplitude of the broad noise may be large. However, it will drop and the width of the noise will increase. In the final stage, the amplitude will rise again and the span of the noise floor also increases.

By this time, the bearing will be vibrating excessively; it will be hot and making lots of noise. If it is allowed to run further, the cage will break and the rolling elements will go loose. The elements may then run into each other, twisting, turning and welded to one another, until the machine will hopefully trip on overload. In all probability, there will be serious damage to the shaft area under the bearing.

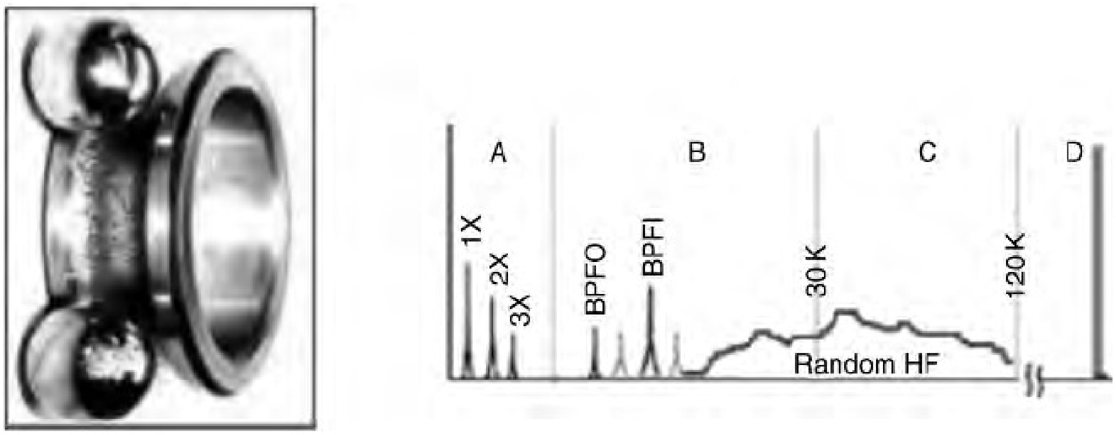


Figure 5.6 Stage 4 Shows Merged Defect and Components Natural Frequencies and High HFD

5.3 Gears Problems

A gearbox is a piece of rotating equipment that can cause the normal low-frequency harmonics in the vibration spectrum, but also show a lot of activity in the high frequency region due to gear teeth and bearing impacts. The spectrum of any gearbox shows the 1× and 2× rpm, along with the gear mesh frequency (GMF). The GMF is calculated by the product of the number of teeth of a pinion or a gear, and its respective running speed:

GMF = number of teeth on pinion × pinion rpm

The GMF will have running speed sidebands relative to the shaft speed to which the gear is attached. Gearbox spectrums contain a range of frequencies due to the different GMFs and their harmonics. All peaks have low amplitudes and no natural gear frequencies are excited if the gearbox is still in a good condition. Sidebands around the GMF and its harmonics are quite common. These contain information about gearbox faults (Figure 5.7).

Tooth wear and backlash can excite gear natural frequencies along with the gear mesh frequencies and their sidebands. Signal enhancement analysis enables the collection of vibrations from a single shaft inside a gearbox. Cepstrum analysis is an excellent tool for analyzing the power in each sideband family. The use of cepstrum analysis in conjunction with order analysis and time domain averaging can eliminate the 'smearing' of the many frequency components due to small speed variations.

As a general rule, distributed faults such as eccentricity and gear misalignment will produce sidebands and harmonics that have high amplitude close to the tooth-mesh frequency. Localized faults such as a cracked tooth produce sidebands that are spread more widely across the spectrum.

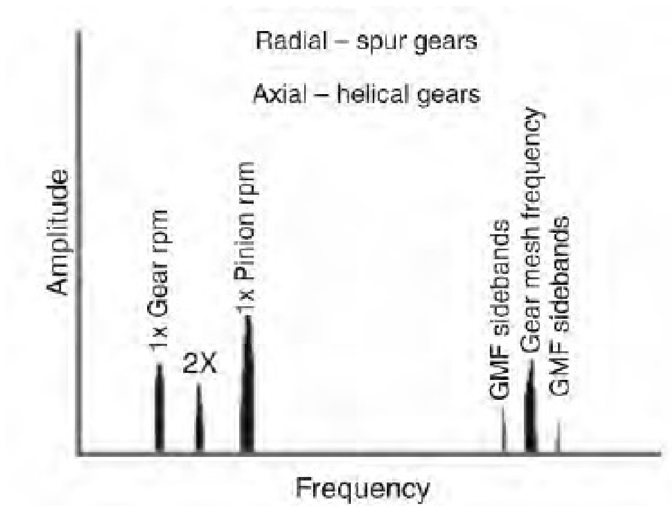


Figure 5.7 Vibration of Gears

Tooth Wear

An important characteristic of gear tooth wear is that gear natural frequencies are excited with sidebands around them. These are spaced with the running speed of the bad gear. The GMF may or may not change in amplitude, although high-amplitude sidebands surrounding the GMF usually occur when wear is present. Sidebands are a better wear indicator than the GMF itself (Figure 5.8).

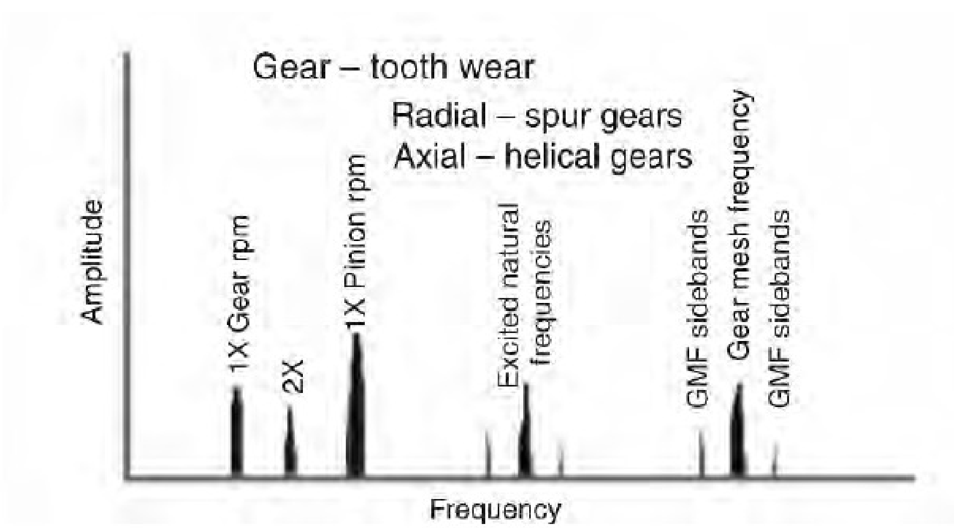


Figure 5.8 Tooth Wear Vibration Problem

Tooth Load

As the load on a gearbox increases, the GMF amplitude may also increase. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low and no gear natural frequencies are excited. It is advised that vibration analysis on a gearbox be conducted when the gearbox is transmitting maximum power (Figure 5.9).

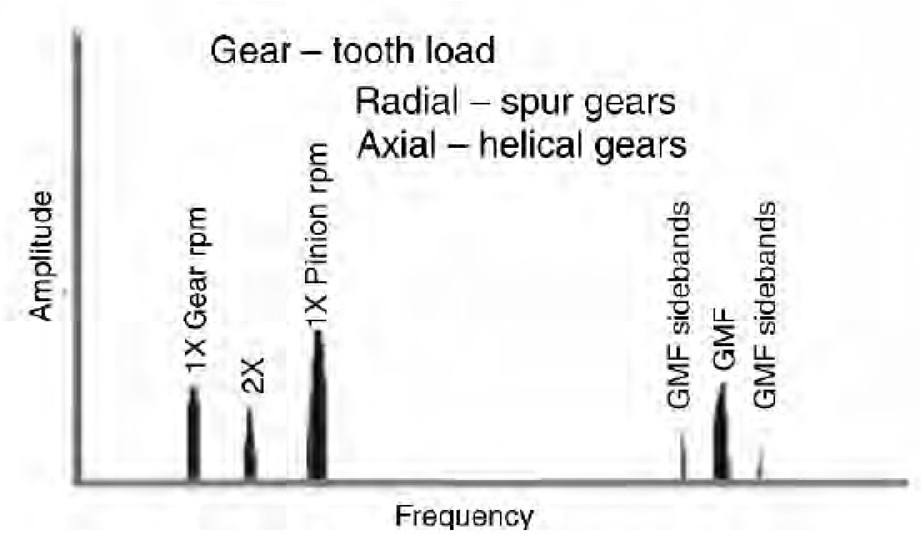


Figure 5.9 Vibration due to Excessive Tooth Load

Gear Eccentricity and Backlash

Fairly high amplitude sidebands around the GMF often suggest gear eccentricity, backlash or non-parallel shafts. In these cases, the rotation of one gear may cause the amplitude of gear vibration to *modulate* at the running speed of the other. This can be seen in the time domain waveform. The spacing of the sideband frequencies indicates the gear with the problem. Improper backlash normally excites the GMF and gear natural frequencies. Both will have sidebands at 1× rpm. The GMF amplitudes will often decrease with increasing load if backlash is the problem (Figure 5.10).

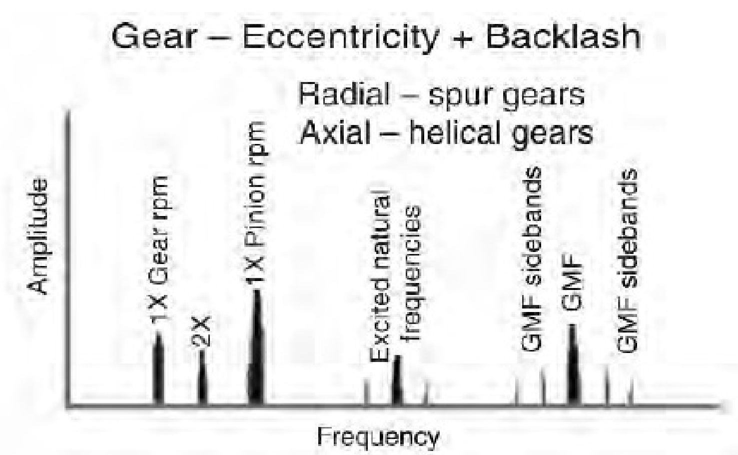


Figure 5.10 Vibration due to Gear Eccentricity/Backlash

Gear Misalignment

Gear misalignment almost always excites second order or higher GMF harmonics, which will have sidebands spaced with the running speed. It will often show only small amplitudes at 1× GMF, but much higher levels at 2× or 3× GMF. It is important to set the F-max of the FFT spectrum to more than 3× GMF (Figure 5.11).

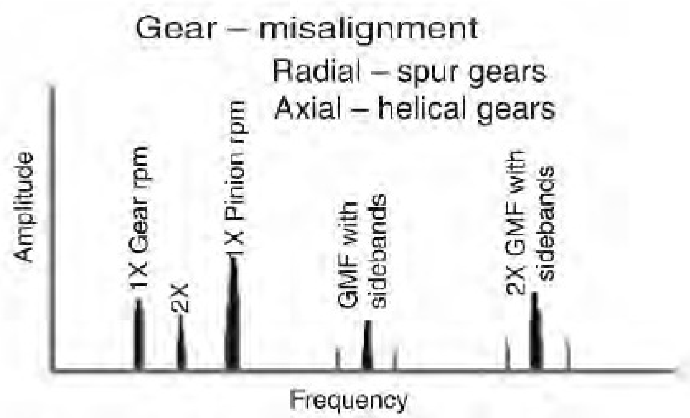


Figure 5.11 Gear Misalignment Vibration Problem

Cracked or Broken Tooth

A cracked or broken gear tooth will generate high amplitude at 1× rpm of this gear, plus it will excite the gear natural frequency with sidebands spaced with its running speed. It is best detected in the time domain, which will show a pronounced spike every time the problematic tooth tries to mesh with teeth on the mating gear. The time between impacts will correspond to 1/speed of the gear with the broken tooth. The amplitude the impact spike in the time waveform will often be much higher than that of the 1× gear rpm in the FFT spectrum (Figure 5.12).

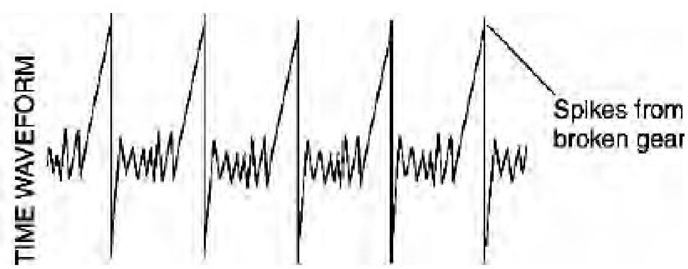


Figure 5.12 Time Waveform for a Cracked or Broken Gear Tooth

Gear Hunting Tooth Problem

The gear hunting tooth frequency is particularly effective for detecting faults on both the gear and the pinion that might have occurred during the manufacturing process or due to mishandling. It can cause quite high vibrations, but since it occurs at low frequencies, predominantly less than 600 cpm, it is often missed during vibration analysis. The hunting tooth frequency is calculated with:

$$Hunting\ Tooth Frequency = \frac{GMF \times N}{No.of\ \ teeth\ in\ pinion \times No.of\ \ teeth\ in wheel}$$

In the above equation, N is known as the *assembly phase factor*, also referred to as the lowest common integer multiple between the number of teeth on the pinion and gear. This hunting tooth frequency is usually very low. For assembly phase factors ($N > 1$), every gear tooth will not mesh with every pinion tooth. If $N = 3$, teeth numbers 1, 4, 7, etc. will mesh with one another (however, gear tooth 1 will not mesh with pinion teeth 2 or 3; instead, it will mesh with 1, 4, 7, etc.). For example, a gear with 98 teeth is running at 5528 rpm and is meshing with a pinion with 65 teeth and running 8334 rpm. The assembly phase factor is $N = 1$. The hunting tooth frequency (F_{ht}) can be calculated as follows:

$$F_{ht} = \frac{(98 \times 5528) \times 1}{65 \times 98} = 85\, cpm\ or\ 1.42\, Hz$$

Another formula is the rpm of the gear divided by the number of pinion teeth (5528/65 = 85 cpm). This is a special case and applies to a hunting tooth combination only when $N = 1$. If the tooth repeat frequency is a problem (Figure 5.13), one can usually audibly hear it since it is a beat frequency. A gear set with a tooth repeat problem normally emits a 'growling' sound from the driven end.

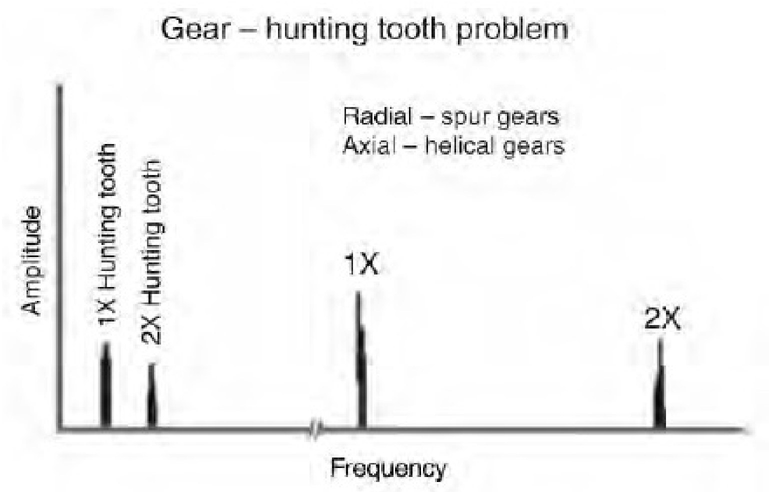


Figure 5.13 Gear Hunting Tooth Problem

6. Advanced Techniques

6.1 Transient Analysis

Transient analysis is similar in someway to startup and shutdown analysis since both of them deal with a transient vibration. However, during machine startup/shutdown test, the overall collection time is much longer and the speed signal is also acquired since it is a parameter of analysis. Transient analysis is very useful to identify resonance frequency or to estimate Frequency Response Function (FRF) of a structure. For example, during impact test, the structure or machine is bumped by a hummer and the corresponding vibration is measured and either analyzed online or saved as time signal and analyzed later to display FFT waterfall. More than one impact can be executed during data collection and averaging process is performed over the overall record length with or without overlapping. In most cases, 50% overlapping is good choice to cover all signal details as shown in Figure 6.1. Logarithmic averaging may be applied where the log of the amplitude is used instead of the direct amplitude. Also, the peak amplitude is sometimes used instead of the averaged amplitude for better identification of the resonance frequencies, but this is applied when the noise level is low during data acquisition. Figure 6.2 shows an FFT waterfall for a transient vibration. During transient analysis, integrators or differentiators of the collection device must be off to eliminate any delay and modification to the measured signal.

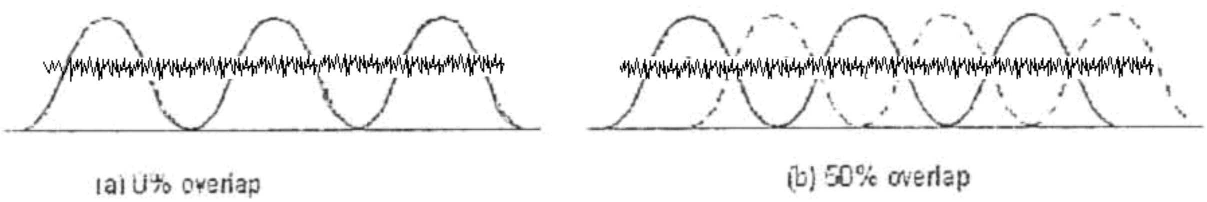


Figure 6.1 Overlapping During FFT Analysis of a Transient Signal

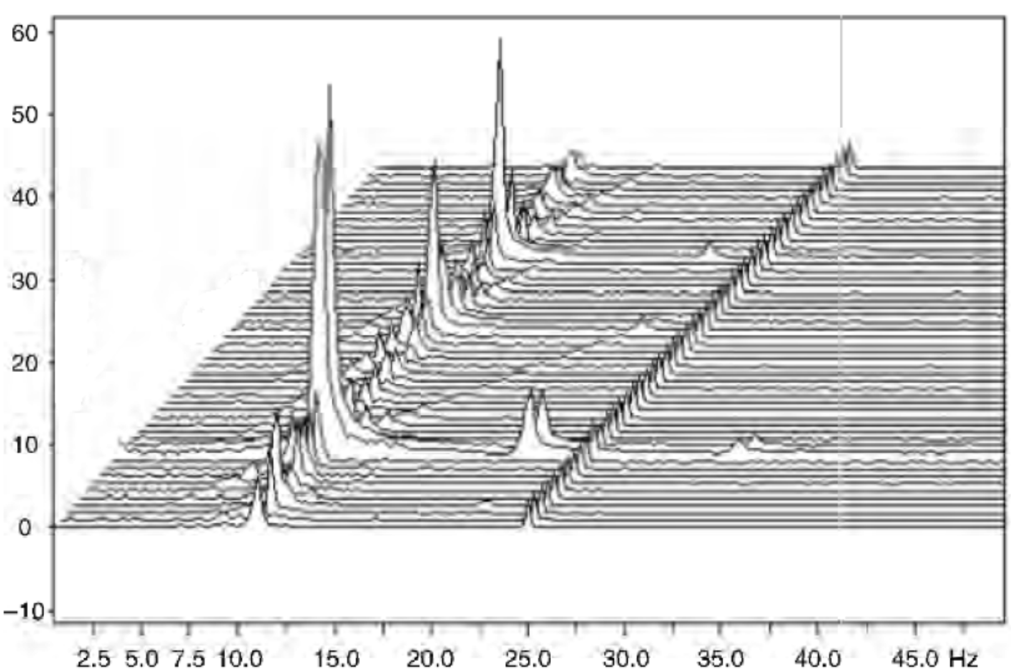


Figure 6.2 FFT Waterfall for a Transient Vibration

6.2 Dual and Multi-Channel Analysis

Dual channel analysis requires simultaneous sampling of two channels. A single channel analyzer can only accept an input from one accelerometer at a time, whereas a dual channel instrument can accept inputs from two accelerometers simultaneously from different locations on the machine. Thus, two vibration waveforms can be collected from a machine and analyzed. This can provide very meaningful vibration data. The biggest advantage is that there is no need for reference marks on the shaft. As a result, there is no need to shut down the machine to provide the marks. The phase differences obtained are very accurate. It can provide phase differences at any frequency.

6.2.1 Orbit Analysis

As described in sec. 3.5, orbits are Lissajous patterns of time domain signals that are simultaneously plotted in the X–Y coordinate plane of an oscilloscope or vibration analyzer. In most applications, the unit of measurement is displacement which is measured directly using proximity probes 90 apart as shown in Figure 6.3. These types of measurements are relative vibration readings. Relative readings are considered vibration measurements of the shaft with respect to the bearing housing.

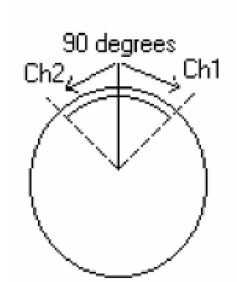


Figure 6.3 Orientation of Proximity Probes

The slight elliptical shape of the 1X orbit plot shown in Figure 6.4 shows a small unbalance condition. In this case, a 1X filter was applied to the measurement for isolating potential running speed related issues. It is important to note, however, that other non-1X related frequencies may be present which are missed if a filtered orbit is used.

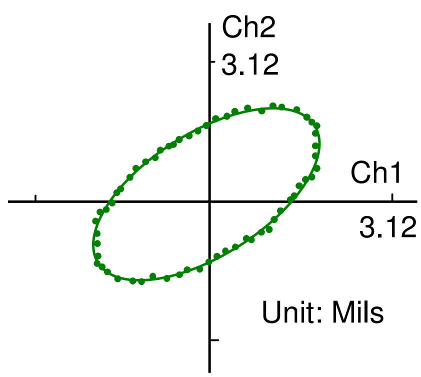


Figure 6.4 Orbit for Unbalance Condition

Inner loops presented in a Lissajous orbit plot can indicate a "hit and bounce" type condition. This phenomenon occurs when the shaft comes in contact with the bearing surface and "bounces" off. In early stages of contact, a "flat" spot in the orbit plot will be presented. As the condition becomes more severe, the number of loops present will increase and become tighter. Detailed analysis of the orbit plot can indicate exactly where the shaft is coming into contact with the bearing in question. Figure 6.5 shows a "non-filtered" orbit plot of the same shaft as discussed above.

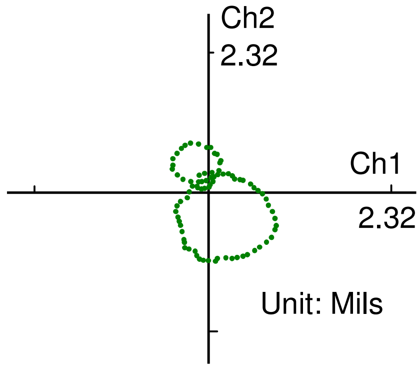


Figure 6.5 Hit and Bounce Orbit Analysis

Misalignment tends to add significant levels of vibration to the shaft at 1X, but also at higher orders of running speed. 1X component from misalignment will not be in phase with the unbalance component. The effect is to make the orbit plot less circular and more elliptical or even non-elliptical (i.e., banana-shape or figure-eight pattern). See Figure 6.6. Resonance and excessive bearing wear also tend to produce elliptical orbits. If the contribution is due to resonance, this condition can be pinpointed if the orbit changes noticeably with changes in running speed.

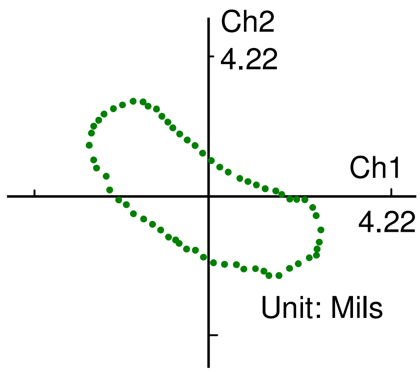


Figure 6.6 Typical Orbit for Misalignment Condition

Mechanical looseness caused by excessive wear tends to produce an orbit similar to that of a rub. Subsynchronous effects show up as secondary loops. However, in this case, shaft movement may be in a forward direction, rather than the reverse direction that is typical of a rub. Figure 6.7 shows a typical Orbit Plot in the Emonitor. See the Emonitor online help for more information.

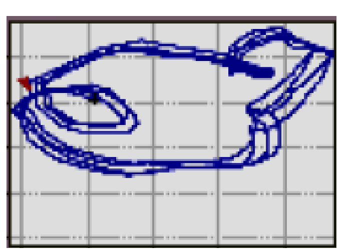


Figure 6.7 Orbit for Mechanical Looseness Condition

6.2.2 Relative Phase Measurement

One useful application for dual channel analysis is the detection of relative phase between two vibration signals. This is very useful for the diagnosis of machinery problems depending on the phase relation between two locations or points as explained in chapter 4. Relative phase of overall vibration or filtered components can be used but only the filtered components will reveal meaningful and stable data especially when the signal contains many frequency components. The machine speed must be known or it can be collected during dual channel measurement to filter the components synchronously. Most vibration data analyzers contain a facility to acquire the machine speed either from a high resolution FFT analysis or from a reference pickup. Table 6.1 shows a typical relative phase data between two channels.

Table 6.1 Typical Table for Relative Phase Data

Order	**Ch1 Amp.**	**Ch2 Amp**	**Relative Phase**
1X	5.21	2.45	**87°**
2X	3.21	2.12	**123°**
3X	1.23	0.565	**165°**
…			

6.2.3 Simulation of Shaft Movement

Some advanced multi-channel systems can simulate actual shaft movement by measuring shaft vibration in two or more locations with two directions in each location as shown in Figure 6.8. This can be useful in the diagnosis of some vibration problems or study of rotor dynamics.

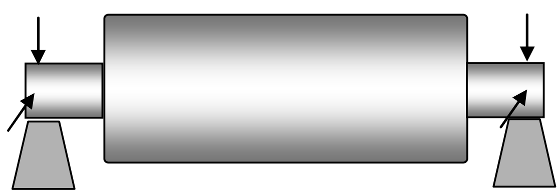


Figure 6.8 Multi-Channel Measurement

6.2.4 Synchronous Data Acquisition

In some applications, it is required to sample more than one channel at the same time for synchronous data sampling. Examples are in the study of Frequency Response of structures and mechanical parts where two ore more pickups are attached to the system. This is useful to maintain the relative phase between signals and also to reduce acquisition time by acquiring the signals at the same time.

6.3 Run-up and Coast-down Test

Of particular interest is the analysis of the vibrations during a run-up or a coast-down of a machine in which case the structural resonances are excited by the fundamental or the harmonics of the rotational frequencies in the mechanical system. Determination of the critical speeds, where the normal modes of the rotating shaft are excited, is very important on large machines such as turbines and generators.

The run-up and coast-down tests are executed by analyzing vibration signals that vary with the speed or time as the machine speed goes up or down. One of the most common applications is detecting resonance related problems in rotors and structures. Not only the resonance frequency can be detected, but also the order at which it occurs. The results can be displayed as Bode, Nyquist, FFT waterfall and Spectrogram. During the test, the collecting device can either be manually or automatically (depending on speed) trigged to start and stop collecting data from the attached vibration and reference pickups. The data can be saved as time waveform and can be processed to later to display various analyses. The reference signal is required to measure running speed and to provide the phase angle information. The overall data record is divided into a number of analysis records to trace machine speed variation. The analysis record can be set in terms of time span, number of revolutions or both (whichever first satisfied), speed change or other parameter.

6.3.1 Bode Plot

Overall vibration and/or filtered components can be traced in Bode plot. The order of a filtered component can be integer or fractional multiples of the running speed and up to 6 orders can be traced on the same plot excluding the overall trace. To extract the order amplitude and phase, the time signal for each cycle (one period of the running speed) is multiplied by sine and cosine functions whose frequency is the order frequency;

$$
\begin{aligned}
a &= \frac{1}{N}\sum_{i=0}^{N-1} g(t_i)\cos\left(\frac{2\pi \times order \times i}{N}\right) \\
b &= \frac{1}{N}\sum_{i=0}^{N-1} g(t_i)\sin\left(\frac{2\pi \times order \times i}{N}\right) \qquad (1) \\
A &= \sqrt{a^2+b^2}, \ \phi = \tan^{-1}\frac{b}{a}
\end{aligned}
$$

Where N is the number of points in one cycle. All other orders will cancel out during summation process. If the analysis record possesses more than one cycle, averaging process will be applied over the number of records available. Figure 6.9 shows a Bode Plot for machine run-up test.

Figure 6.9 Bode Plot for a Run-up Test

6.3.2 FFT Waterfall

FFT waterfall is simply a number of FFT spectra drawn constitutively. When the analysis record contains many FFT records, averaging can be done with or without overlapping, but when the analysis record is shorter than FFT record (depending on FFT options and analysis setting), zero padding is applied to maintain the number of lines fixed for all spectra. The user can optionally decrease the number of FFT lines or increase analysis record to overcome this problem. The resonance frequency and its order can readily be known by inspecting the waterfall, particularly when the nxRPM (or order) is chosen as frequency unit. Figure 6.10 shows a typical FFT Waterfall.

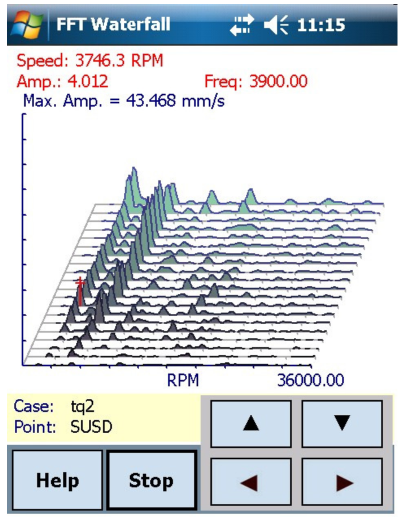


Figure 6.10 FFT Waterfall

6.3.3 Spectrogram

Another more powerful technique is to display the Spectrogram, a diagram having frequency in the x-axis and machine speed or time in the y-axis with amplitude represented by color coding. Any high amplitude can be easily seen at the corresponding machine speed and frequency. When the machine speed is used as parameter in the y-axis, as shown in Figure 6.11, the harmonic components appear on radial lines through the point (0Hz, 0 RPM) while structural resonances appear on vertical straight lines (constant frequency lines). Thus such a plot can be very useful. The smearing of the components, which appears because the time window used for the individual spectra represents a certain sweep in the speed, is however, a disadvantage. The power of the components becomes spread over several lines. In particular, high frequency components in the spectrum, such as toothmesh frequencies, might be smeared so much that details in sideband structures are lost in the analysis. This is the main reason why order analysis is used instead.

Figure 6.12 shows a spectrogram using order tracking (tracking analysis) in which the sampling rate is synchronized with machine speed. The vibration signal at the generator bearing contained among other components a significant 37th harmonic and harmonics of this. A 3-dimensional plot of a 400 line order analysis up to the 40th harmonic during coast-down is shown. This component was found to be caused by a fan with 37 blades in the generator cooling system. Some peaks are easily seen. No sideband structure is seen around this component, indicating that it is a rather pure blade-passing frequency without modulation. This indicates that the increases are not due to structural resonances, but might be caused by increased turbulence in the blower at different speeds.

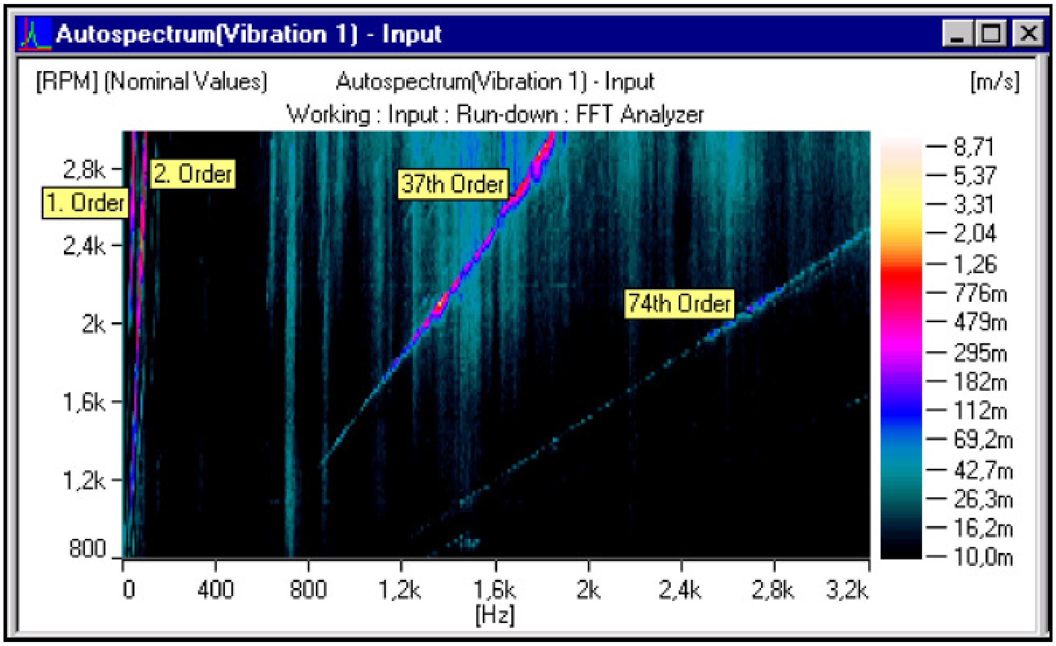


Figure 6.11 Spectrogram using Fixed Sampling Rate

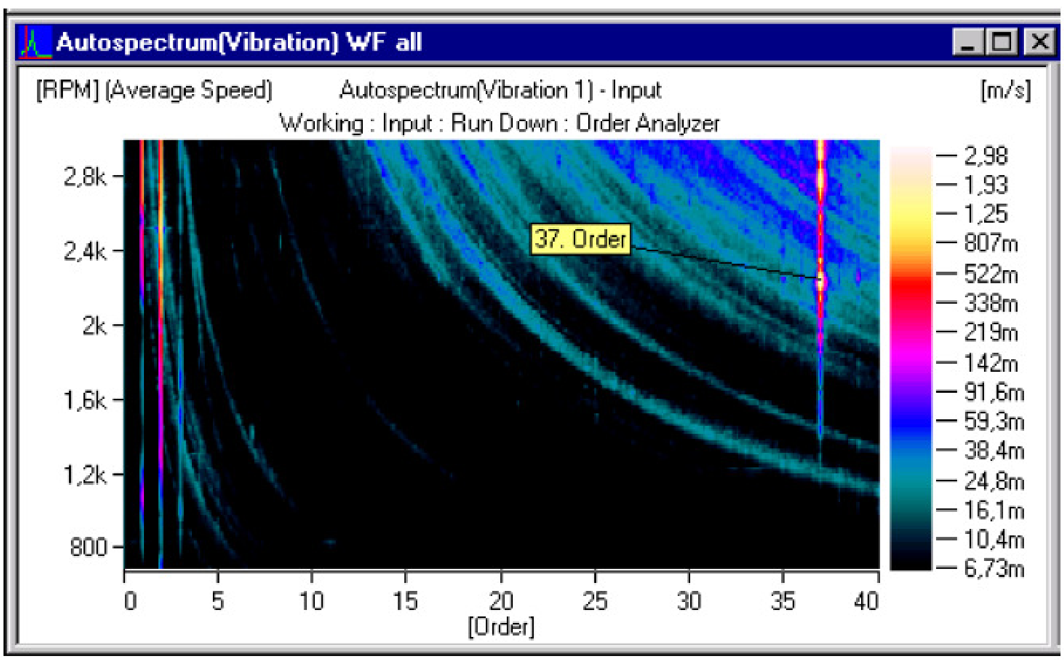


Figure 6.12 Spectrogram using Order Tracking

7. Field Balancing

7.1 Introduction

One of the most important features of a portable vibration analyzer is the field balancing capability. Some equipment have the possibility to do in-place balancing such as fans, blowers, shafts, couplings, large turbo-machinery and others. Field balancing eliminates the need to dismantle the machine to perform workshop balancing. Also, field balancing can take into consideration the effects of the attached parts and thus producing good overall balancing condition. Many vibration analyzers incorporate one-plane and two-plane balancing scheme based on influence coefficients method. To perform balancing procedure, a reflecting tape must be attached at some angle to produce reference mark to measure angles. Vibration pickups of any type can be attached at the bearings housing to measure vibration signals. The machine is operated at first to measure initial vibration, then stopped to add trial mass at the first correction plane and operated again to measure vibration due to the first trial mass. A second trial mass is added at the second correction plane and the vibration is measured at the third run. The device will calculate the influence coefficients and calculate the initial unbalance. The influence coefficients are saved for future balancing procedures where a single run can be executed to measure unbalance

A powerful field balancer must take into accounts the following points:

1. Trial mass may be kept during and after balancing procedure to speed up the process.
2. Setup time of the analyzer must be as short as possible to avoid running for long period of time at high vibration.
3. The correction masses may be divided in case of fans or other discontinuous parts when the angle of correction lies in the blank section.
4. The data collector must be capable of saving the influence coefficients to reduce balancing time in the next maintenance procedure.

7.2 Procedure of Field Balancing

The following procedure is used in two-plane balancing:

1. Stop the machine to insert a reflecting tape on the rotating shaft to represent the reference point for phase measurement.
2. Attach vibration and reference pickups to the machine and start it again.
3. Start your data collector to measure vibration due to initial unbalance.
4. Stop the machine again; add the first trial mass (more than the permissible mass) to the first correction plane.
5. Start the machine and let the data collector measure vibration due to the first trial mass.
6. Stop the machine to add the second trial mass at the second correction plane, the first trial mass can optionally be removed or kept.
7. Start the machine again and let the data collector measure vibration after adding the second trial mass.
8. The data collector will calculate the influence coefficients and accordingly will calculate the initial unbalance and the required correction masses in any of the following cases; the first trial mass is kept or removed, the second trial mass is kept or removed, both trial masses are kept or removed.