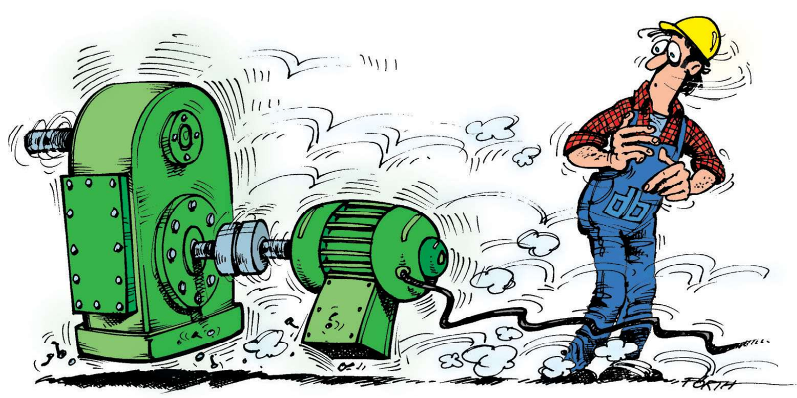
Section 2
Vibration Analysis



Condition Monitoring

Most people involved in plant maintenance have heard of Condition Monitoring (CM). By definition CM means to periodically view machine operating condition and when necessary respond to any changes in machine condition. CM can be carried out by a number of "maintenance functions"; visual inspection, wear debris analysis, thermographic analysis and vibration analysis are the most popular methods. In this handbook we are concerned with the use of vibration analysis to measure, monitor and analyze machine condition.

Later we will look at vibration measurement techniques and explain some of the basic parameters and terminology that is used for machine condition measurement. First, it is useful to briefly look at the benefits that a CM regime can bring to an operating plant.

In 1988 the DTI reported that companies who have implemented a CM program on their plant on average spend 25% less on maintenance of the plant than companies who have no CM program (DTI Boardroom report on maintenance in British Industry 1988).

Given that a moderately sized UK plant will spend £250,000 annually on plant maintenance a saving of £62,000 plus additional savings on production, power and ancillary activities represents a very good return on the investment in a CM program.

If the returns on an investment on a CM program are so good why doesn't every plant have a system in operation? Most often the answer to this lies in a lack of understanding of what is required for CM on the plant, and on a fear that the cost of implementing a system and running it will be more than the return on cost that the system will realize.

Condition monitoring essentially means that the machines on plant get a "regular health check." This is usually taken in the form of periodic vibration measurements. These measurements are compared to a standard or "known" operating condition. In the case of vibration this standard is usually an ISO norm or in some cases an on plant standard or manufacturers recommendations. By making a comparison between current condition and " standard or known" condition an evaluation can be made as to whether the machine operating condition has changed. Depending on the extent of change the machine condition can be further investigated or monitored more frequently to detect further changes. The key tool in this CM concept is to trend collected data and respond to changes in the trend. The objective is to intervene before the machine fails catastrophically.

The "Nuts and Bolts" of a CM system

Having identified the basic principle of a CM regime, what are the costs of getting a program up and running? The answer is, not much. A system can cost as little as a few hundred pounds for a simple portable "point and shoot" product that is touched onto the machine and gives a reading of vibration severity, usually RMS vibration. This reading can be manually recorded on a chart. Readings are then compared over time to identify a change in measured operating condition.

VIBSCANNER – a typical low cost portable CM system

A development of this is to collect machine condition data and input this into a PC program to automatically establish and trend machine condition data. Further developments allow analysis of machine condition via algorithmic calculations (FFT analysis) to determine specific machine condition defects. From portable systems you can progress to on-line monitoring systems which provide round the clock measurement and alarming of machine condition.

As system complexity and analysis capabilities expand so does system cost. Normally the criticality of machines to the operation of the plant will determine the level of monitoring system required on the plant. It does not follow, however, that a company who spends £20,000 on a CM system gets a 20 times better system than a company who spends £1,000, or will get 20 times better CM results.

Take the CM route that is most comfortable. Match the system to the plant and to the skills available. Don't go for the most expensive system option just because a salesman says it is the best, it may be, but, it may not be appropriate to the plant.

If the plant has many process critical machines on-line monitoring may need to be considered. If the process plant is small, start simple; routine visual inspection and simple overall vibration readings will suffice. If in doubt, irrespective of plant size, start simple. Provided the system implemented is capable of expansion and can grow as CM requirements grow, you need not worry about more complex analysis capabilities.

Trend the data collected – most systems (even the most expensive) trend the data and only analyze when a problem occurs. If necessary outsource the analytical expertise required.

Implementing a CM program

First and foremost, implementing a CM regime means that you have to know your plant. A basic understanding of the way the machines behave and the way they should behave is essential. This doesn't mean extensive initial investment in sensors, expert analysis or highly skilled personnel. Information from ISO standards, machine suppliers and past plant operating experience will often provide the information required to initially establish how the plant **should** behave. How the plant **actually** behaves can be established by a combination of techniques including vibration measurement, thermography, oil analysis and operator experience of the plant.

One of the keys to running an effective CM regime is the investment that the plant management is prepared to make in ensuring operators are skilled in using the systems they employ, and that on going training is available to maintain operator skills.

Implementing and maintaining a CM regime doesn't have to be a full time job; but is does require commitment and regular monitoring routines via some form of data acquisition, storage and regular review.

Returns on CM investment

The most effective prevention of machine breakdown is a combination of regular data acquisition, trending analysis, root cause analysis, and machine operator awareness. Attention to changes in operating condition, a leaking seal, an increase in overall vibration, a change in machine operating temperature or even an increase in operating noise will notify an alert operator or engineer of a potential problem. It doesn't mean you shut the plant. What it should mean is that you investigate further, eliminate variable process changes and then increase frequency of monitoring of the machine to establish the rate of change in operating condition. A rapid rise requires intervention quickly before plant failure, a slow rise means that you can plan a convenient future time for intervention. This is one of the principle benefits of implementing CM on your plant.

- It allows your engineers to plan plant shutdown, order spares, and get the right personnel available to carry out the shutdown work.

Everyone who has been involved in maintenance or production using process plant will have experienced a "sudden" plant failure. A bearing on a pump for example seizes; the catastrophic result is a mechanical seal is destroyed; product spills; a shaft or coupling is destroyed. If you can reduce or eliminate these "sudden" failures and intervene before catastrophic failure, only the failing component (in this case the bearing) has to be replaced.

- CM will help prevent ancillary plant damage thus the cost of replacement parts will be lower, and time to effect repair will be less – plant will restart quicker.
- Labor costs can be reduced by focusing the work force on problem areas.
- A proven CM routine should enable you to negotiate better plant insurance rates.
- You can "push" process machines harder to gain extra production if needed whilst monitoring plant condition.
- You can reduce or eliminate routine machine shutdown.
- You can build production reserves prior to a forecast machine shutdown in order to eliminate production losses.

The common denominator of all these benefits is cash. Improved process plant availability, and reduced maintenance costs effectively mean a more profitable production plant.

Having reviewed the options available and the CM strategies that you can adopt, you should sit back and ask what you really want out of a plant improvement scheme. You may just want a quiet life, or to improve machine reliability or you may want to improve plant-operating profitability. What you want will dictate what you are prepared to spend and commit to CM. Whatever the reasons for the investment, CM will repay you long after the cost of equipment has been capitalized and written off in your accounts.

Vibration analysis

Vibration data has high information content
Vibration measurements contain a lot of useful information that will help determine the health of the machine for example:

- It provides information for safe machine operation.
- It can detect that the condition of a machine has changed.
- It can be used to diagnose the cause of change.
- It can be used to classify the condition of a machine.

Vibration measurement is normally a non intrusive measurement procedure and it can be carried out with the machine running in its normal operating condition.

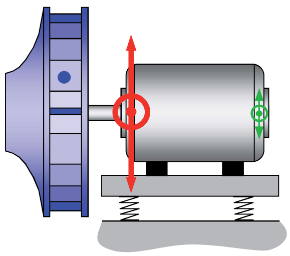
Basic parameters

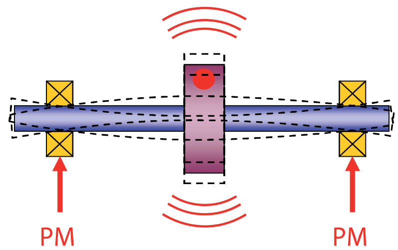
Vibration is an effect caused by machine condition. Vibration is simply the oscillation about a reference point (i.e. a shaft vibrates relative to the casing of a piece of machinery and a bearing vibrates relative to a bearing housing.) Vibration exists when a system responds to some internal or external excitation and can be broken down into 3 basic types.

The amplitude of vibration depends on the magnitude of the excitation force, the mass and stiffness of the system and its damping. Vibration occurs because we are not able either to build a perfect piece of machinery or to install it perfectly. If we could build a perfect piece of machinery, the center of mass of the rotating element would be located exactly at its center of gravity. When the center of mass and center of gravity do not coincide the rotor has a heavy spot and some degree of unbalance. This unbalance produces a vibration proportional to the amount of weight of the heavy spot. Additional sources of vibration are machine tolerances, machine structure, bearing design, loading and lubrication, machine mounting and rolling and rubbing between moving parts.

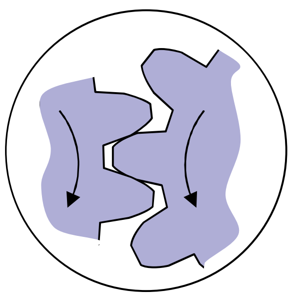
The analysis of vibration requires an understanding of the terminology used to describe the components of vibration.

Free body vibration

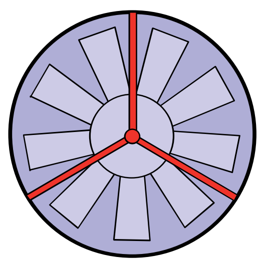




Meshing and passing vibration

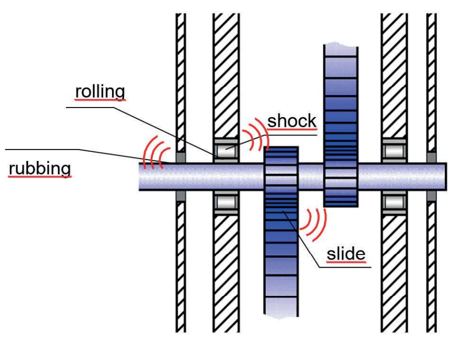


Gear mesh vibration



Blade pass vibration

Frictional vibration



Frequency
Frequency is the cyclic movement in a given unit of time. The units of frequency are:

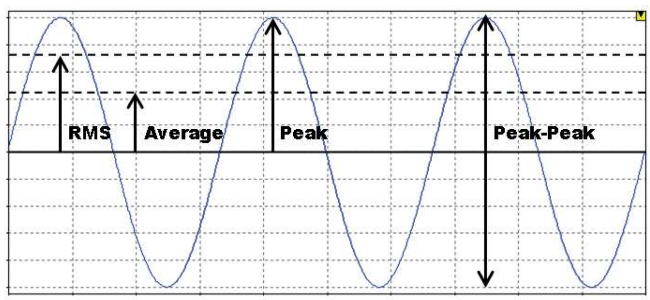
- rpm = revolutions or cycles per minute.
- Hertz (Hz) = revolutions or cycle per second.

These are related by the formula: F = frequency in hertz = rpm/60.

Amplitude
Amplitude is the magnitude of dynamic motion of vibration. It is typically expressed in any of the following terms:

- RMS (Root Mean Square); Zero to Peak; Peak to Peak.

The sketch below illustrates the relationship of these three units of measurement associated with amplitude.

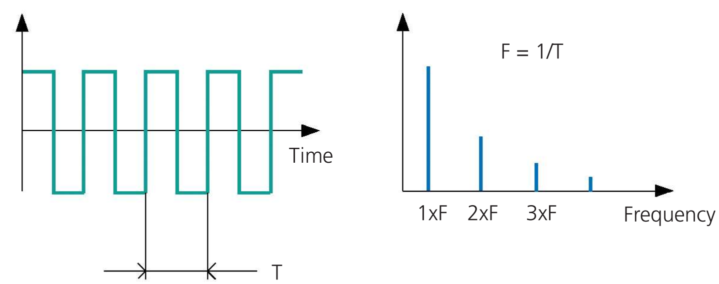


Amplitude, whether expressed in displacement, velocity or acceleration is generally an indicator of severity. Since industrial standards of vibration severity will be expressed in one of these terms, it is necessary to have a clear understanding of their relationship. Care must be exercised to note the "type" of amplitude measurement when comparing machinery vibration to industry standards.

Fundamental Frequency
Fundamental frequency is the primary rotating speed of the machine or shaft being monitored and usually referred to as the running speed of the machine.

You will also see the fundamental frequency referred to as 1 x rpm, or as Hz; using as an example an 1800 rpm motor this would be 30 Hz (1 x 1800/60). The fundamental frequency is important because many machinery faults such as misalignment or unbalance occur at some multiple of the fundamental frequency. For example misalignment at 1 x fundamental frequency.

Harmonics
These are the vibration signals having frequencies that are exact multiples of the fundamental frequency (i.e. 1 x F, 2 x F, 3 x F etc.).



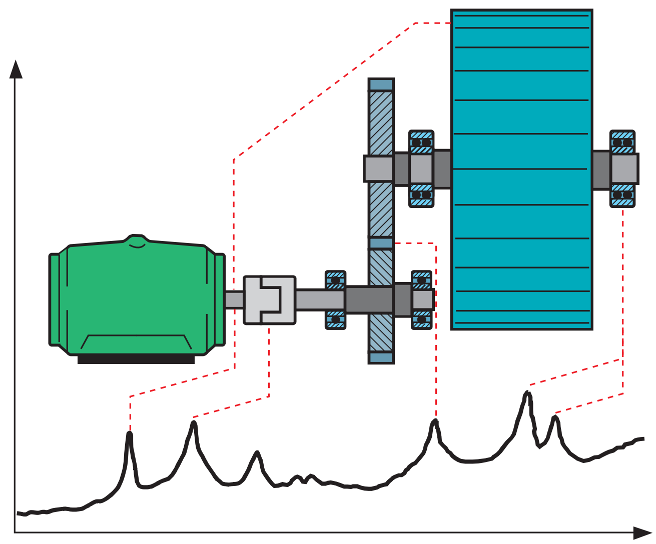
Displacement (D)
Displacement is the actual physical movement of a vibrating surface. Displacement is usually expressed in mils (thousands of an inch) or microns. When measuring displacement, we are interested in the Peak to Peak displacement which is the total distance from the upper limits to the lower limits of travel.

Velocity (V)
Velocity is the speed at which displacement occurs. We define velocity as the rate of change in the relative position. Velocity is usually measured in mm/sec RMS, or inches/sec RMS.

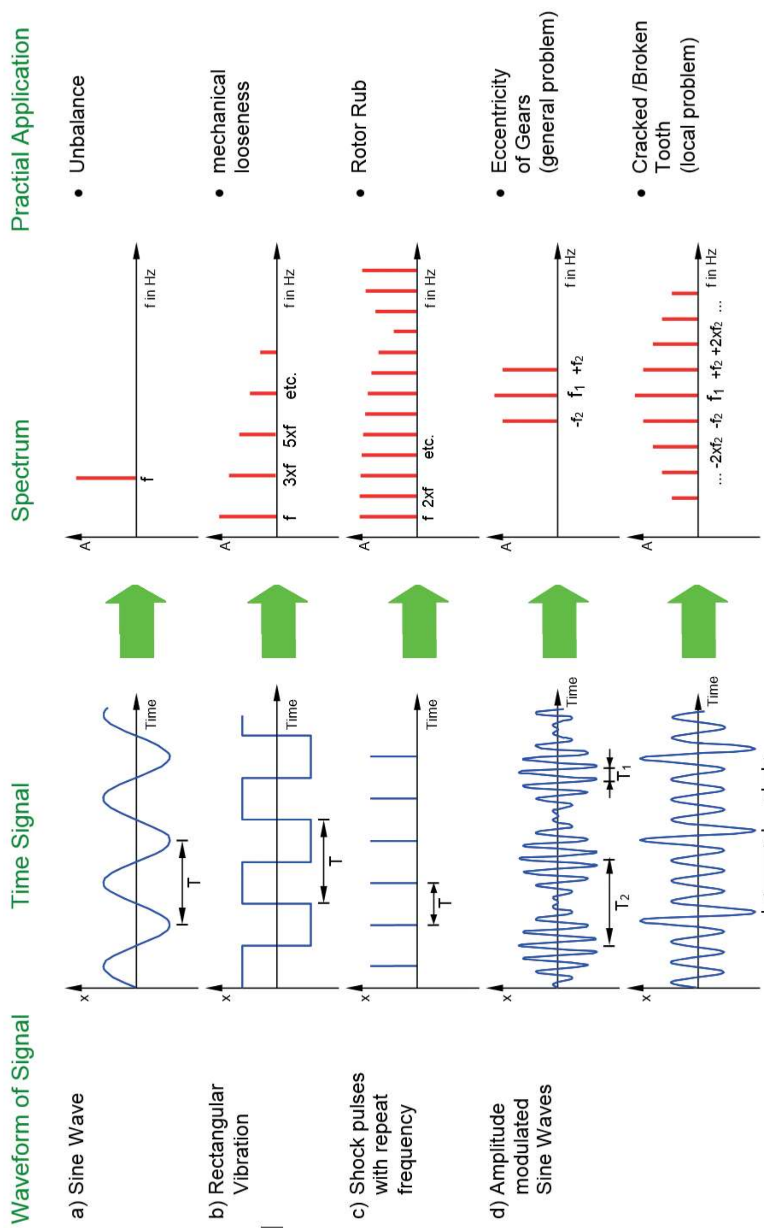
Acceleration (A)
Acceleration is the rate of change of velocity. This we can simply define as the change of velocity in a period of time or change in rate of velocity. Acceleration is usually measured in m/s^2 or in g's of gravitational force.

Vibration frequency spectrum

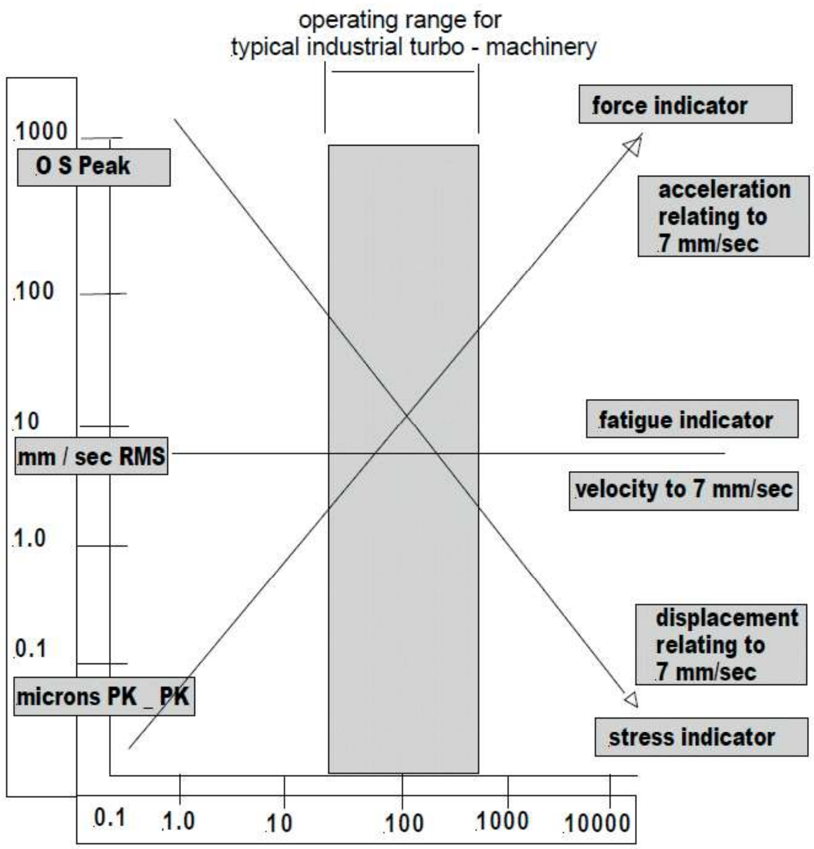
Machinery vibration consists of various frequency components as illustrated below. The amplitude of each frequency components provides an indication of the condition of a particular rotating element within the machine.



The advantage of frequency spectrum analysis is the ability to normalize each vibration component so that the complex machine spectrum can be divided into discrete components. This ability simplifies the analysis of mechanical degradation within the machine. The following chart illustrates typical signal forms for various machine components.


Waveform of Signal
Time Signal
Spectrum
Practial Application
a) Sine Wave
b) Rectangular Vibration
c) Shock pulses with repeat frequency
d) Amplitude modulated Sine Waves
x
Time
T
T1
T2
A
f in Hz
f
3xf
5xf
etc.
2xf
-f2 f1 +f2
... -2xf2 -f2 f1 +f2 +2xf2 ...
• Unbalance
• mechanical looseness
• Rotor Rub
• Eccentricity of Gears (general problem)
• Cracked /Broken Tooth (local problem)

Relationship between displacement, velocity, amplitude, frequency
Variation in the values of velocity and acceleration with frequency is extremely important, for it forms the basis for vibration severity criteria, provides guidelines for selecting the variable which will be most representative for a particular purpose, and explains how failures can occur without warning if the wrong variable is monitored. This variation is best illustrated by plotting displacement and acceleration versus frequency at a constant velocity amplitude of 7 mm/s as shown below.



Note that velocity appears to be a valid indicator of condition across the entire range of frequencies. This is the main reason why vibration is used as the prime indication of mechanical condition.

The relationship between displacement, velocity and acceleration also provides the best indication of which parameter should be measured to assess condition. The diagram clearly shows that when examining the low frequencies around or below the running speed of most machinery, displacement or velocity measurements are likely to produce the best quality signal. On the other hand, phenomena such as bearings resonance's at 5 – 10KHz and above, are best measured in terms of acceleration.

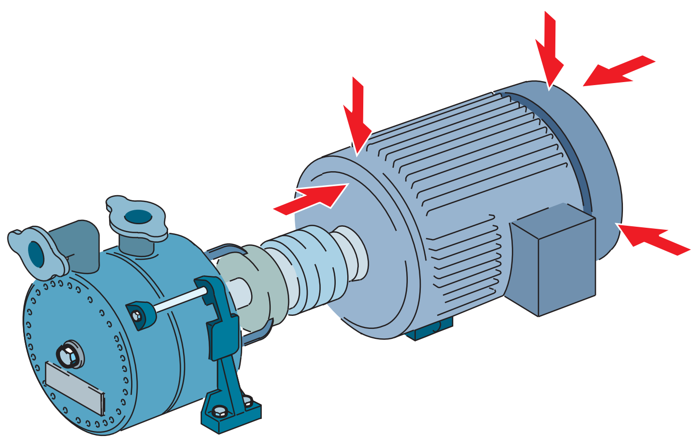
FFT (Fast Fourier Transform)
FFT is predominantly the most used tool in analysis of spectral data with respect to vibration analysis of machine components. Fourier transform is a mathematical operation which decomposes a time domain function into its frequency domain components.

Transducers

Mounting location

The position and the manner in which data is collected is very important to a successful vibration monitoring program. In order to properly diagnose a fault, data must be collected in the right plane and must be repeatable. Some faults show the high amplitude in the radial direction and some in the axial direction.

Measure location should usually be on exposed parts of the machine that are normally accessible and that reflect the vibration of the bearing housing. Vertical and horizontal mounting directions are the most usual transducer locations for horizontally mounted machines, any angular position is acceptable provided that the location reasonably represents the dynamic forces present in the machine. For vertically mounted machines the location giving the maximum vibration reading should be used as a future monitoring reference point



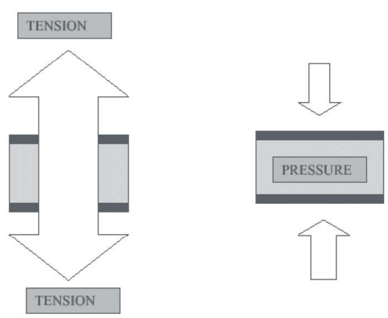
The data collection points should be clearly marked to ensure that data is collected at the same point every time. (Frequently, measurement studs are permanently fixed to the machine ensuring reproducibility of measurement location). When analyzing a machine for changes, the analysis can be inaccurate if data is not collected at the same point each time.

Measurements should be carried out when the rotor and main bearings have reached their normal steady state operating temperature, speed, load, voltage and pressure. Where machine speeds vary measurements should be taken at all conditions at which the machine operates for a prolonged period.

Transducer design

There are a variety of instruments (transducers) that will convert actual mechanical movement (vibration) into electrical energy.

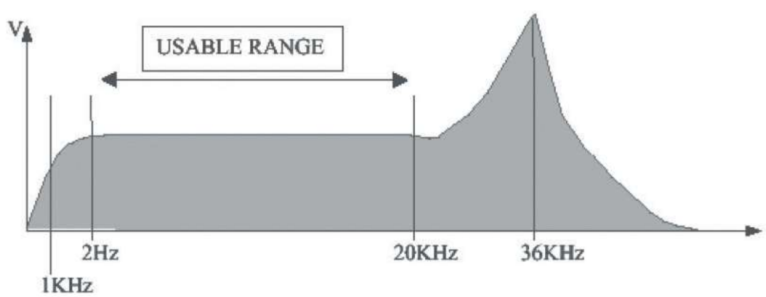
The industrial accelerometer
Accelerometers are the most widely used transducer in routine vibration monitoring programs. A typical accelerometer contains a piezoelectric crystal element which is pre-loaded by a mass of some type and the entire assembly is enclosed in a rugged protective housing. The piezoelectric crystal produces an electrical output when it is physically stressed by either a pressure or tension effect as shown below.



The variable vibration force exerted by the mass on the crystal produces an electrical output proportional to acceleration. Accelerometers have a broad frequency range, typically from 2 Hz to 10 KHz. The accelerometer is also easily mounted using either a stud, a magnet, an adhesive or by hand-holding it onto the machinery surface. Accelerometers also have good temperature and environmental responses and are usually of a rugged construction.

Accelerometer frequency response

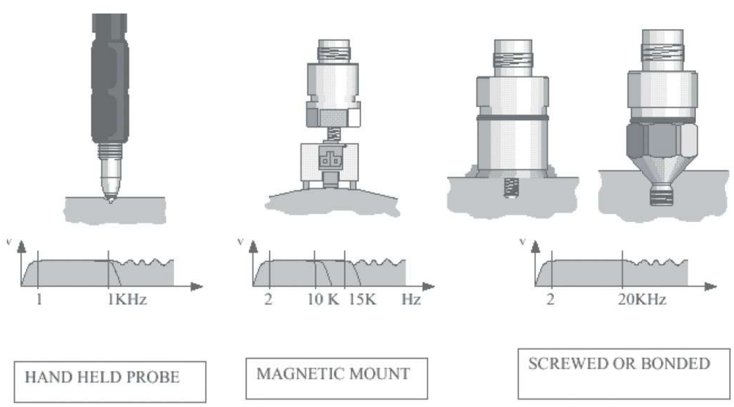
Each accelerometer has a usable frequency range and response curve typically as shown below. If the data to be collected is outside the frequency range shown on the response curve, an accelerometer having the correct response should be chosen. As a general rule, the smaller the accelerometer, the higher the usable frequency range.



Typical accelerometer response for threaded or bonded transducers

Accelerometer mounting

The mounting of an accelerometer plays a significant role in its frequency response. Shown below are four different types of mounting methods for transducers, all of which are used in vibration analysis programs.



The screw or stud mounted unit with the proper accelerometer has a frequency response of around 20 KHz. The epoxy mount (glue mount) has approximately the same response. The permanent magnet mount has a frequency response of approximately 5 KHz, while the hand held unit is typically around 1.5KHz. The more rigid the transducer contact with the machine the better the frequency response and hence the better the reliability of the vibration reading.

Signal processing

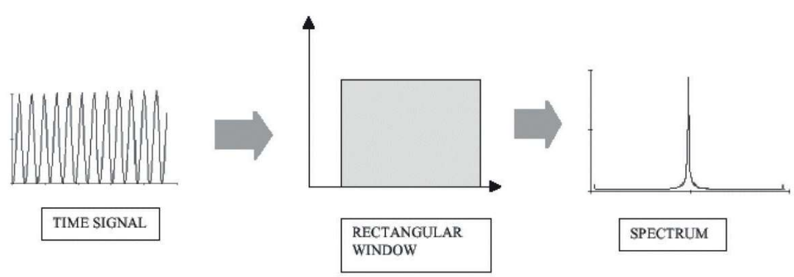
The raw data collected by the transducers must be enhanced to provide useful information. For vibration data this raw data must be "conditioned" to prevent errors. Typically such conditioning includes:

- Filtering to remove unwanted or spurious signals
- Amplifying to enhance the resolution of low energy signals
- Data averaging to remove spurious data
- Conversion to frequency domain (FFT)

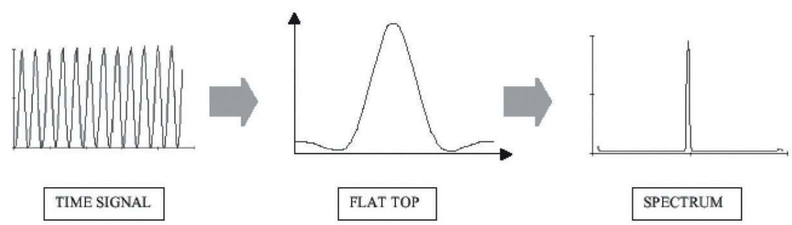
To assist with these filtering techniques many analyzers provide a number of "window" functions which, depending on the type selected, will assist with analysis of data.

Rectangular window

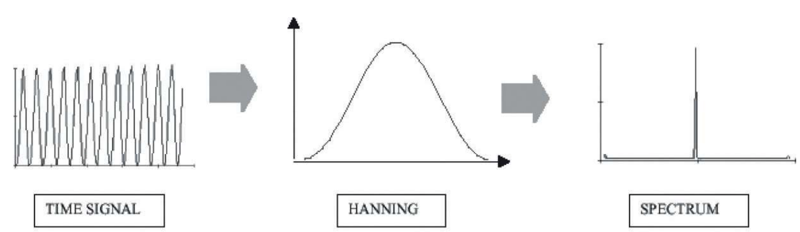
This provides for higher inaccuracy in the amplitude domain but with a greater accuracy in the frequency domain. A practical use for this window is for transient process e.g. bump tests to identify natural component frequencies.



Flat top window
This has the highest accuracy for the amplitude domain but higher inaccuracy for the frequency resolution domain.



Hanning window
This is the standard window for most vibration analysis, it has the best accuracy for the frequency resolution domain, but with higher inaccuracy in the amplitude domain.



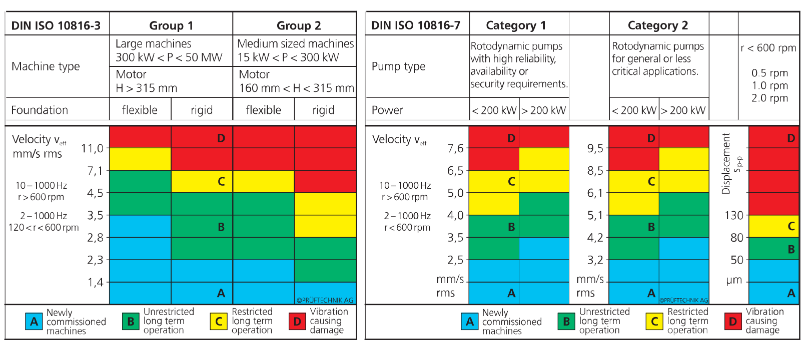
Fault detection

As has been discussed, the main objective of a vibration monitoring program is the detection of incipient machine failures. The methodologies associated with fault prediction usually involve comparing current vibration information with a vibration description of that machine or a similar machine in satisfactory operating condition. This comparison is made by two methods:

- Comparison to industrial standards – ISO 10816-3-7
- Comparison to a previously measured reading

ISO 10816-3-7

The ISO 10816-3-7 is the current standard for the evaluation of "standard" rotating machine operating condition. Issued in 2009 it covers "large and medium size industrial machines with nominal power rating above 15 kW and nominal speeds between 120 rpm and 15000 rpm". In addition pumps are added as a specific category for consideration. This range covers most rotating machines and can therefore be used as a good guide for in-situ operating condition. The chart below illustrates the standard and allows a quick comparison of actual against standard operating condition.



Variations will inevitably occur when comparing these standards to actual machine operating condition. Machines should not however be condemned because of variations in readings without first considering other potential reasons for the difference in readings.

The chart on the previous page shows 4 zones of vibration severity ranging from good to unacceptable.

- Zone A – the vibration of newly commissioned machines would normally fall into this zone.
- Zone B – machines with vibration within this zone are normally considered acceptable.
- Zone C – machines with vibration in this zone are normally considered unsatisfactory for long term continuous operation. Machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial operation. (It is advised to increase the frequency of vibration monitoring during this operating period.)
- Zone D – vibrations within this zone are normally considered to be of sufficient severity to cause damage to the machine.

The use of these zones and the numerical values ascribed to them are not intended as an acceptance standard for machine manufacturers and customers but the values do help to establish alarm and warning criteria for a routine condition monitoring program. If machines are found to be operating at vibration levels consistently above the nominal values shown in the standard, investigation as to the cause should always be carried out.

Comparison to previous readings

This method is the most widely used method of identifying changes in machine operating condition. Most commonly referred to as trending, comparison quickly shows the machine operator or manager if the machine condition has changed, and by how much, and in what period of time. Trended graphical measurement values combined with ISO alarm limits give clear visual warnings of machine condition change and with some software packages, can be used to predict likely run to failure intervals in order to schedule remedial repair. (Graphic predictions of machine trends should always be considered as a guide to likely failure and not as a definitive measurement of machine failure intervals.)

Fault mode analysis

There is a great deal of literature on diagnostic techniques employed for various types of equipment. Although we will not discuss specific diagnostic methods in this handbook, you will find that a great deal of information must be readily available to execute an effective vibration monitoring program. Once the vibration monitoring program flags a machine as potentially having a mechanical problem, the following questions must be answered.

- How severe is the problem?
- What is the problem?
- When must the machine be taken out of service for repair?

Machinery diagnostics using vibration analysis provides information that addresses these questions.

The machinery diagnostics technique viewed here is based on a technique known as "fault mode" analysis. This technique utilizes the fact that specific mechanical events, such as unbalance, misalignment, looseness, bearing defects, aerodynamic and hydraulic problems, and gearbox problems usually generate vibration frequencies in specific patterns. The frequency, amplitude and pattern of the peaks in a vibration spectrum can be a telling indication of the type of problem being experienced by the machine. The principles of "fault mode" analysis include:

- Measurement of mechanical faults such as unbalance and misalignment generate mechanical vibration in a well defined frequency pattern.
- Comparing the vibration levels and vibration spectra on similar types of machines will help establish the severity and cause of a vibration problem.

The following chart summarizes specific machinery faults and their vibration patterns.

Possible cause	Dominant frequency	Direction	Comment
Imbalance	1x rotational frequency	Radial for dynamic imbalance; possible axial	Vibration amplitude proportional to imbalance & rpm – causes severe vibration to occur
Misalignment	2x rotational frequency	Radial & axial	Severe axial vibration 2nd harmonic, best realigned using a laser alignment system.
Bearing defect	High frequency vibration	Radial & axial loaded	Use bearing enveloping diagnosis or shock pulse to determine damage severity
Machine foundations	Typically at one or more natural frequency (transient vibration)	Radial	Natural resonant frequency of foundation or machine base plate
Belt vibrations	Rotational frequency and multiples thereof	Radial	Additionally recommend strobe to combine machine rpm and belt speed to check for belt slippage
Blade pass vibration	Number of vanes or blades x the fundamental frequency	Radial	Vibration frequency represented by the number of blades x the shaft rpm
Electrical	Line frequency, 50 Hz (UK) 60 HZ (USA) and multiples thereof	Radial & axial	Side bands may also occur at multiples of the rotational frequency. Vibration stops when power is turned off
Gear mesh defect	Gear frequency equal to the number of teeth x the rpm of the gear	Radial & axial	Sidebands occur from modulation of the gear teeth meshing vibration at the rpm e.g. the output shaft speeds of the gearbox
Resonance	Natural component frequency	Radial & axial	A components natural frequency coincides with an excitable frequency

Imbalance
Vibration caused by imbalance occurs at a frequency equal to 1 rpm of the imbalanced part, and the amplitude of vibration is proportional to the amount of imbalance present.

Normally, the largest amplitude will be measured in the radial (vertical or horizontal) directions.

Misalignment
Generally, misalignment can exist between shafts that are connected with a coupling, gearbox or other intermediate drives. Three types of misalignment are:

- Angular – where the center line of the two shafts meet at an angle
- Offset – where the shaft center lines are displaced from one another
- A combination of angular and offset misalignment

A bent shaft looks very much like angular displacement, so its vibration characteristics are included with misalignment.

Misalignment, even with flexible couplings, have two forces, axial and radial, which result in axial and radial vibration. The significant characteristics of vibration due to misalignment or a bent shaft is that it will be in both the radial and axial directions. For this reason when axial vibration is greater than one half of the highest radial measurement (horizontal or vertical), then misalignment or a bent shaft should be suspected.

All misalignment conditions will produce vibration at the fundamental (1 x rpm) frequency components since they create an imbalanced condition in the machine. Misalignment will sometimes produce vibration at the second harmonic (2 x rpm).

Loosness
Mechanical looseness can be caused by loose rotating components or loose machine foundations.

Mechanical looseness causes vibration at a frequency of twice the rotating speed (2 x rpm) and higher orders of the loose machine part. In most cases, vibration at the fundamental (1 x rpm) frequency will also be produced.

Bearing problems

One of the results of damage to rolling element bearings is that the natural frequencies of the bearing components are excited by the bearing defect. The resonant vibration or "ringing" occurs at frequencies between 2 KHz and 60 KHz.

This vibration is most effectively measured at a level of acceleration in units of g's Peak. Vibration is measured by the machinery monitoring system as a HFE (High Frequency Energy) measurement and gives an effective indication of the condition of rolling element bearings. Based on field experience, the shock pulse technique works well on motors and other quiet equipment. Care must be taken when using the technique on pumps and gearboxes, where flow, cavitation, and tooth meshing can produce impulses which interfere with and mask the impacts produced by bearing defects.

Rotational frequencies related to the motion of the rolling elements, cage and races are also produced by mechanical degradation of the bearing. These frequencies are dependent on bearing geometry and shaft speed and can be found typically, in the 3 – 10 x rpm range and because of these reasons the "enveloping" method is the most widely adopted method of viewing specific bearing defects. This method of bearing condition evaluation and that of shock pulse analysis of bearing condition are reviewed in the following pages.

Aerodynamic and hydraulic problems

Normally associated with blade or vane machinery such as pumps or compressors, aerodynamic and hydraulic vibration is created by an unstable or unbalanced condition within the machine.

In most cases this will produce a vibration at the fundamental frequency (1 x rpm) of the machine and blade pass/vane frequency components.

Gearbox problems

Gear defects or faulty gears produce low amplitude, high frequency vibration. The vibration is predominantly at gear mesh frequency. Gear mesh frequency is calculated as follows:

Gear mesh frequency (GMF) =
Speed of output gear x
Number of teeth in output gear

Example:
52 tooth gear running at 90 rpm (90/60 = 1.5 Hz).
GMF = 52 x 1.5 = 78 Hz.

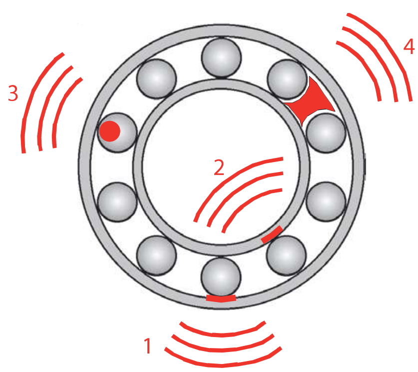
Most gear problems exhibit vibration at the gear mesh frequency, and may be summarized as follows:

- Imbalance – predominant at the 1 x rpm of the gear.
- Misalignment – predominant at the 1 x rpm and 2 X rpm; may excite GMF.
- Pitch line run out – predominant at GMF with 1 x rpm sidebands.
- Faulty gear teeth – predominant at GMF with sidebands at 1 x rpm of faulty gear.

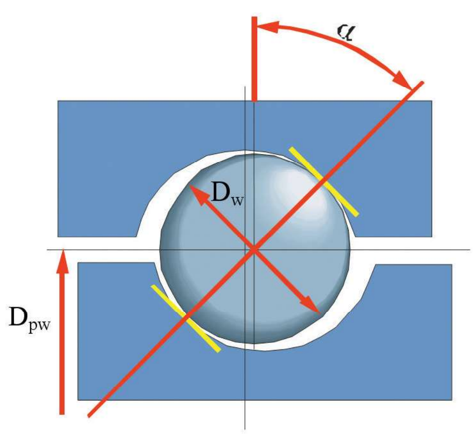
Basic theory of enveloping

When a bearing defect exists in a rolling element bearing the vibration signature will show high frequency vibration generated each time a damaged roller or damaged race make contact. These repetition rates are known as the natural bearing defect frequencies. In any rolling element bearing arrangement there are four types of element defect frequency.

- Ball Pass Frequency Outer Race – BPFO
- Ball Pass Frequency Inner Race – BPFI
- Ball Spin Frequency – BSF (Rolling element defect)
- Fundamental Train Frequency – FTF (Cage defect)



1 = Outer ring defect
2 = Inner ring defect
3 = Rolling element defect
4 = Cage defect



a = contact angle
D_w = Rolling element diameter
D_{pw} = Bearing pitch diameter
Z = Number of rolling elements
n = Shaft speed in rpm

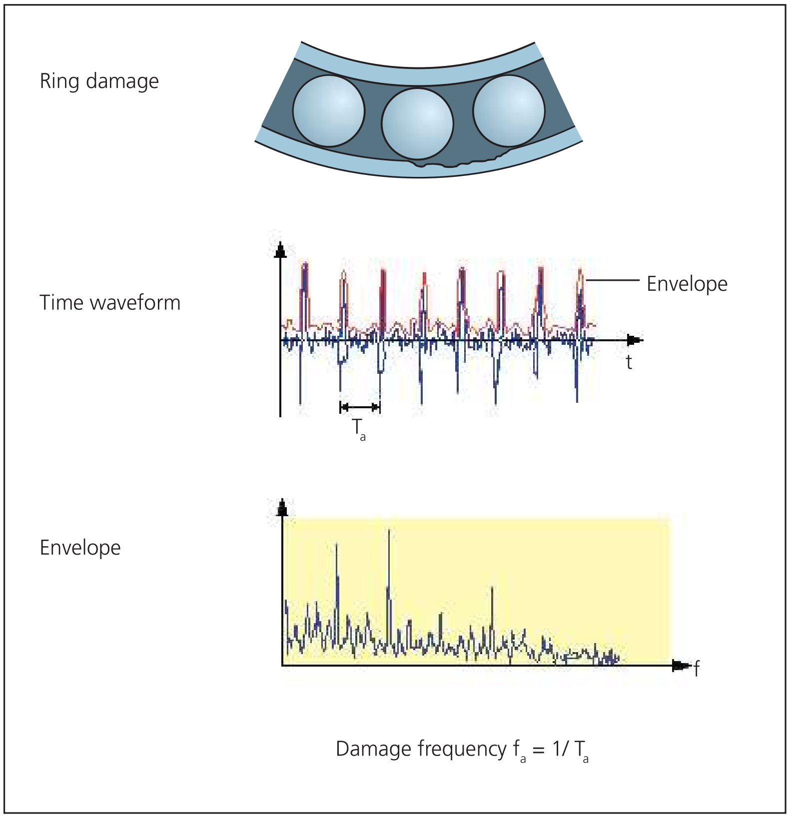
Formula for calculating bearing defect frequencies:

- BPFO = $Z * n / (60 * 2) * (1 - (D_w / D_{pw} * \cos(a))$.
- BPFI = $Z * n / (60 * 2) * (1 + (D_w / D_{pw} * \cos(a))$.
- BSF = $(D_{pw} * n) / (D_w * 60 * 2) * (1 - [D_w / D_{pw} * \cos(a)]^2)$
- FTF = $n / (60 * 2) * (1 - (D_w / D_{pw} * \cos(a))$.

Example: Pass frequencies.
Bearing type SKF 6211, operating speed 2998 rpm

Dimensions	Defect frequencies
D_{pw} = 77.50 mm	BPFO = n / 60 * 4.0781 = 204 Hz
D_w = 14.29 mm	BPFI = n / 60 * 5.922 = 294 Hz
Z = 10	BSF = n / 60 * 5.239 = 264 Hz
a = 0	FTF = n / 60 * 0.4079 = 20 Hz

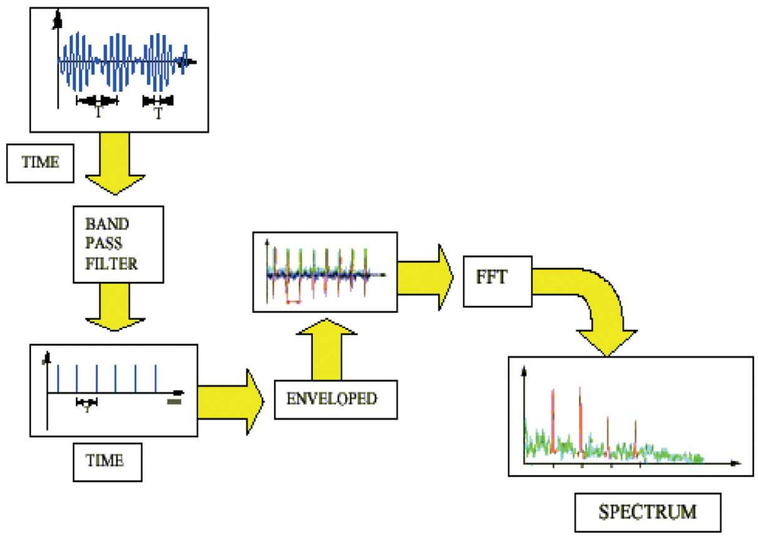
Enveloping is essentially a 2-stage process; the first stage is a band pass filtering of the time waveform. The filtering process results in a series of spiky peaks when enveloping is applied to extract the repetition rate relating to the bearing defect and its harmonics as shown in the following frequency spectra. Since healthy rolling element bearings may exhibit vibration at the natural frequency of the bearing components, it is very important to measure accurately the severity of bearing deterioration.



To measure the severity of a defect in an enveloped spectra the following must be undertaken:

- Measure the amplitude of the specific component in dB (decibels) above the carpet value shown in the spectra.
- The spectra above identifies the carpet and peak values.
- Experience tells us that when a bearing starts to deteriorate a peak to carpet difference of around 10 dB can be seen. This does not mean that bearing breakdown is imminent. As deterioration continues to show a difference of around 15 dB between peak and carpet levels, the bearing should be monitored more closely and preparation made at some point to strip down the machine for repair.

- When the defect amplitude is 20 dB or greater immediate action should be initiated to repair and or replace the bearing.
- The carpet level of the bearing should not be used as a stand alone method of monitoring bearing condition but should be used in conjunction with another trending technique such as shock pulse measurement.



The above diagram shows the steps involved in obtaining an envelope spectra for a bearing.

A common sense approach

Most vibration problems respond well to a logical, systematic approach. A list of suitable steps towards firstly defining and secondly solving problems is given below:

Raw data

- Where is the vibration level highest on the machine and in which direction?
- Is the vibration present in associated machinery and pipework or is it at highest levels on the bearing houses?
- Do changes to the process and lubricating the bearings radically change the vibration response?
- Does the trend show a roughly exponential growth with time?
- How does the machine feel and sound in comparison with similar machines elsewhere?

Diagnostics

- Is this a new machine or one which has recently been worked on? If so, what could have been assembled wrongly and does this tie up with the raw data?
- Which frequencies are dominant on the spectrum? Do these occur at gear meshing frequencies or low order multiples of shaft speed?
- On rolling element bearings and gearboxes, how does the enveloped spectrum appear?
- How quickly is the machine deteriorating and hence how soon does it need to be repaired? (This includes consideration of capital worth of equipment, downtime and maintenance costs.)
- Vibration is usually highest at the point of maximum damage unless a resonant condition exists.
- Vibration is usually the response of a machine to a fault so the only way to stop the vibration is to find the source not the response.
- It is always necessary to build up several items of evidence before diagnosing a fault. The weapons available are HFE, spectra, envelopes, temperature and sound. For each fault the interaction and evidence from these will be different.
- Vibration is a physical phenomenon and as such can be defined by physical means.

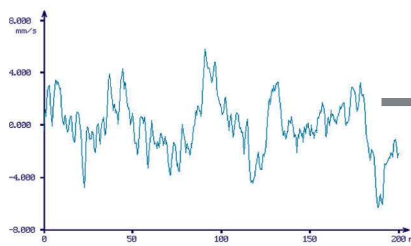
Solutions

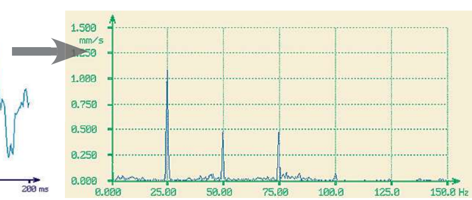
- Where a fault has occurred on previously sound equipment it should be clear from the steps suggested above where the likely problem lies. Having defined the problem, the best course of action should be clear.
- If the fault is on new or recently serviced equipment it may be unclear where the problem lies. Is something resonant? Is there a defect in the installation? Is there a basic design error? The solution to a problem of this type should be achieved in a logical manner. Try one solution at a time (starting with the most likely) taking new sets of data at each step. The best solution will gradually emerge.
- Try to explain all the responses in relation to the damage found once the problem is solved.

FFT analysis

As the name of this short overview suggests the focus of the rest of the section concerns the measurement and subsequent analysis of vibration measurements taken using an FFT analyzer. In effect what we will be doing with this FFT analyzer is taking a time signal analysis and applying a calculation to convert the measured signal into a series of peaks on a standardized graph. Each peak represents an amplitude of vibration and a frequency. Using these two parameters it is possible to see in a relatively simple way the magnitude of vibration and to identify the root cause of the highest vibration signatures.

Frequency analysis





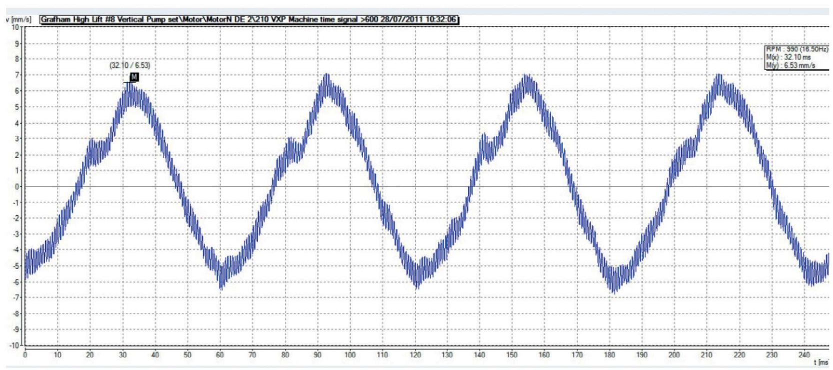
Time waveform

Machine vibration spectrum

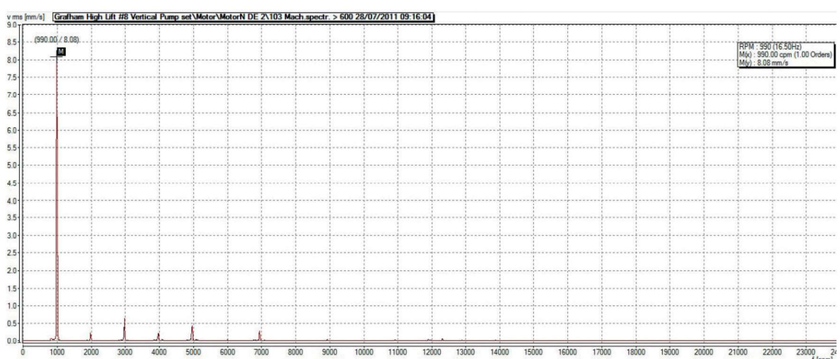
Different factors such as misalignment, unbalance, mechanical looseness can be identified because each factor exhibits its highest vibration amplitude at a different frequency. As an example, misalignment usually exhibits its highest amplitude of vibration at 2 x rotational frequency whereas unbalance exhibits its highest vibration amplitude at 1 x rotational frequency. As we will see later however nothing in the world of vibration analysis is as straightforward as this simple example. However, the principle stands we convert a complex (to understand) time signal analysis into a more understandable FFT vibration plot.

Not only do different factors, such as imbalance or misalignment, exhibit different frequencies of vibration amplitude, so do different components such as bearing or gears. As a general rule (for this initial examination of FFT analysis), machine components have higher frequency vibration characteristics than does the gross mechanical defects such as misalignment, imbalance, looseness or other complete machine operating parameters.

The following two graphs where a time signal for a machine operating in an unbalanced condition is converted to an FFT graph clearly shows the unbalance in an understandable form as a defined frequency – in this case in cycles per minute (cpm) or rotations per minute (rpm) of the machine. An explanation of this important frequency conversion follows.



Time waveform of an unbalanced machine at 1000 rpm.



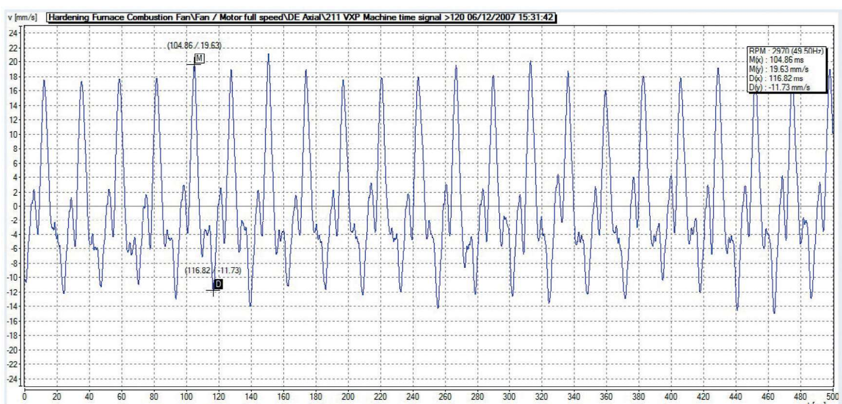
Converted FFT spectrum of the above time waveform.

In the previous time waveform graph the horizontal scale is time in milliseconds and the vertical scale is velocity in mm per second. In the spectrum, the horizontal scale is in cycles per minute (cpm) whilst the vertical scale is again velocity in mm/sec.

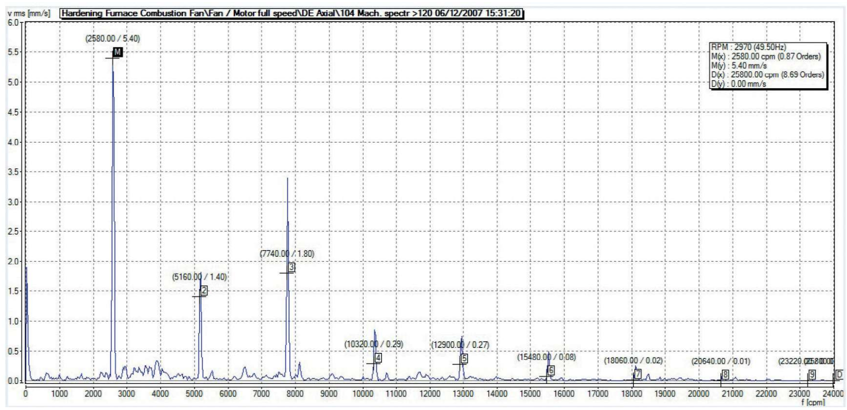
A note about horizontal scaling in FFT graphs will help at this point. There are two conventions that are equally valid in scaling on an FFT graph. Either cycles / rotations per minute (cpm / rpm) or Hertz (Hz) are used. Cpm / rpm is simply a term for the rotating speed of the machine, whilst Hz is the frequency of the machine, simply put it is the cpm / rpm divided by 60.

The spectrum clearly shows a large vibration "spike" at 1000 cpm, which is the fundamental frequency of the machine running at 1000 rpm. This "spike" is described as being at 1x the fundamental frequency of the machine, smaller "spikes" are present at 2x, 3x, 4x and 5x the fundamental frequency. If the horizontal scale had been in Hz the "spikes" would have been in the same position but the scale would have shown the first "spike" at 16.6 Hz (1000/60).

The above is a good example of the ability of an FFT conversion to illustrate specific machine problems in a simplified format. Similarly for another common machine fault (misalignment), the conversion to FFT works just as well (as below).



Time waveform of a misaligned machine set at 1000 rpm.

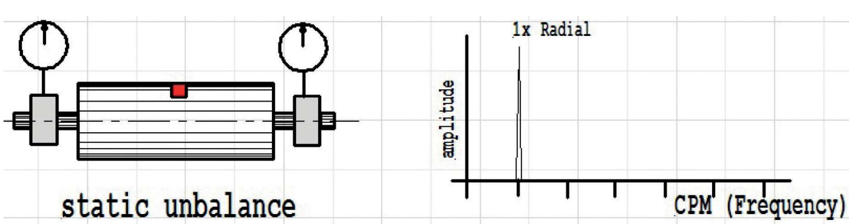


Converted FFT spectrum of the previous time waveform.

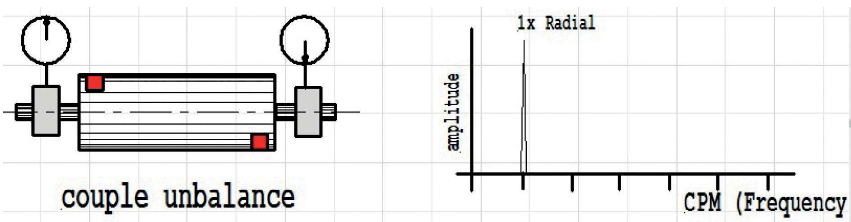
It is perfectly possible to produce a basic reference chart for common machine faults which will help the person new to FFT analysis with diagnosis of machine faults, but at this point it is more helpful for future understanding of the subject that we continue with an explanation of the basics.

For example why does unbalance exhibit its highest frequency at the fundamental frequency? In fact, the explanation is pretty simple: When an out of balance rotor is spinning, the out of balance force is thrown towards the vibration sensor at its greatest velocity once every revolution of the machine.

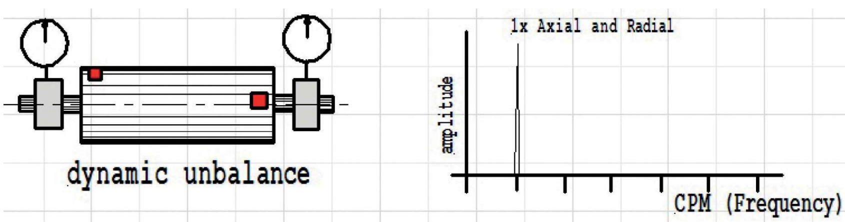
Some commonly encountered types of unbalance
Static unbalance: Also referred to as forced unbalance, this occurs when a heavy spot is located at the mid point between the bearings. This form of unbalance is most common in rotors that have a short length compared to its diameter.



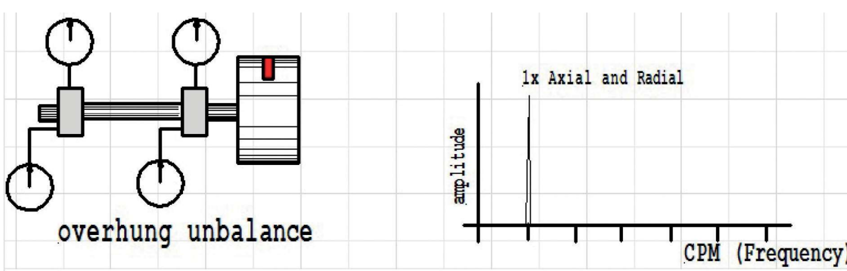
Couple unbalance: Where unbalance forces are 180 degrees out of phase on the same shaft, the 1x frequency spike is always present and dominates the FFT spectrum. Correction requires balance weights to be in at least 2 planes.



Dynamic unbalance: Sometimes referred to as quasi-static unbalance, this is the most commonly encountered form of imbalance. It occurs, when the rotational axis of the shaft and the weight distribution of the rotor do not intersect. This is effectively a combination of static and couple unbalance.



Overhung unbalance: This exhibits radial and axial vibrations – always at 1x frequency, radial signals are the result of influence from shaft bending effects caused by the unbalance load. Axial readings may vary and appear unsteady during the measurement.

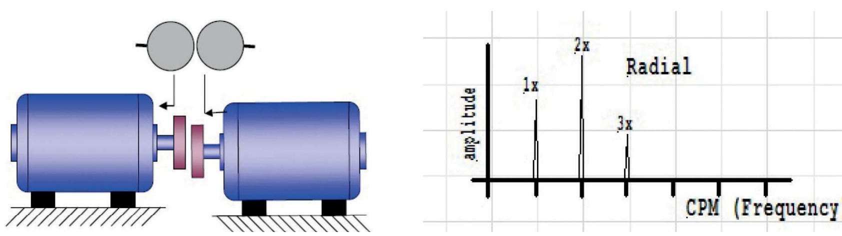


Misalignment

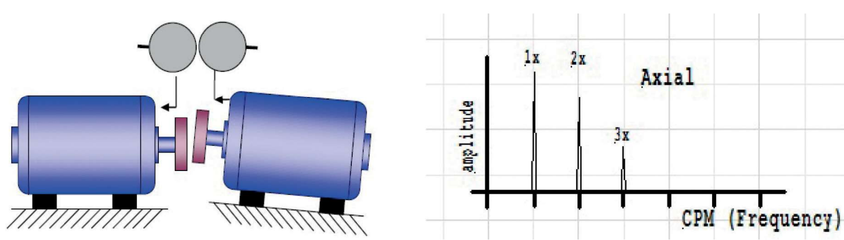
Misalignment alongside unbalance is perhaps one of the most commonly found causes of high vibration in coupled rotating machines such as pumps and other standard machine trains. Unlike unbalance it does not offer a clear frequency spike at one single frequency. Instead misalignment can be identified by having its highest frequency amplitudes at 1x and 2x cpm and with smaller harmonic frequency spikes at 3x cpm up to and including 7x cpm.

The commonly encountered parameters of misalignment

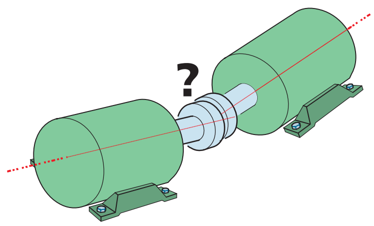
Parallel offset: This exhibits high radial vibration, 180 degrees out of phase with 1x, 2x, 3x and 4x radial vibration prominent with 2x being the dominant vibration amplitude. Vertical and horizontal parallel offset exhibit the same frequency pattern.

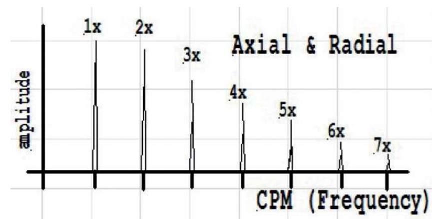


Angular misalignment: This exhibits high axial vibration, 180 degrees phase change across the coupling, 1x, 2x, 3x axial vibration with 1x and 2x dominant (either can be the dominant amplitude). Vertical and horizontal angularity exhibit the same frequency pattern.



General misalignment: This is the most commonly encountered misalignment, it is a combination of parallel and angular misalignment. Axial and radial measurements both show the major frequency components with 1x and 2x the dominant frequencies with harmonics showing up to 7x cpm.





Rolling element bearings

Of all components in a rotating machine probably the most common component associated with vibration analysis is that of the rolling element bearing. The basic construction of the bearing comprising four components; namely an **outer race**, a ball or roller **cage**, **balls** or **rollers** and an i**nner race**, means that its fault diagnosis is considerably more complex than that of many components. Each component has a distinct fault signature on the FFT spectrum. In order to identify the signature faults emanating from the bearing, it is necessary to indulge in a series of mathematical equations to establish the fundamental frequencies of each component as follows.

Inner Race Ball Pass Frequency:
- BPFI $= Z * n / (60 * 2) * (1 + (D_w / D_{pw} *\cos(a))$

Outer Race Ball Pass Frequency:
- BPFO $= Z * n / (60 * 2) * (1 - (D_w / D_{pw} *\cos(a))$.

Ball Spin Frequency:
- BSF $= (D_{pw} * n) / (D_w * 60 * 2) * (1- [D_w / D_{pw} * \cos(a)]^2)$

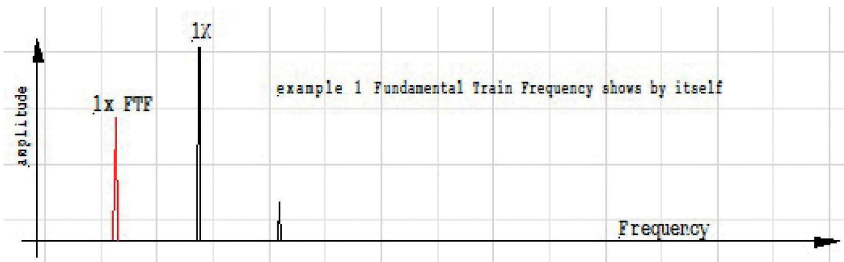
Fundamental Train Frequency:
- FTF $= n / (60 * 2) * (1 - (D_w / D_{pw} *\cos(a))$

where
a = contact angle
D_w = Rolling element diameter
D_{pw} = Bearing pitch diameter
Z = Number of rolling elements
n = Shaft speed in r.p.m.

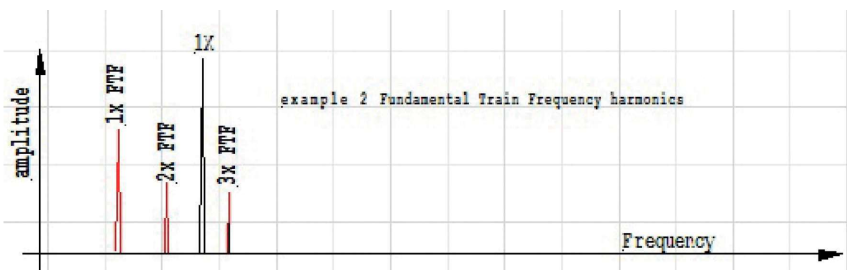
The number of balls and dimensions of the bearing can be obtained from the manufacturers catalogue of the bearing. Additionally, advanced FFT analyzers, such as the VIBXPERT II, have built into the support software the fundamental frequencies of the bearings. This is obtained by simply typing into the software the manufacturers bearing model number. Needless to say, this saves considerable time in setting up and analyzing bearing condition. There is no shortcut, however this information is required in order to properly diagnose bearing condition using FFT spectra.

Cage fault components in velocity spectrum

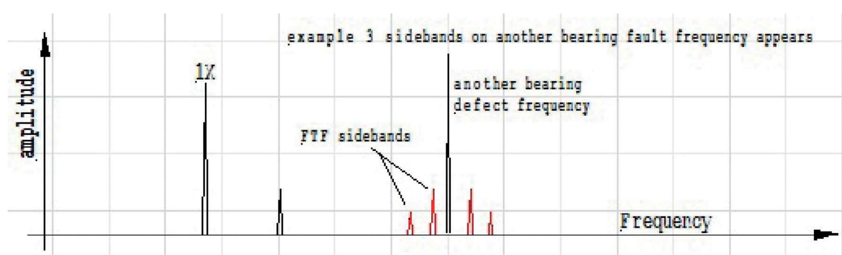
To separate the rolling elements is the chief function of the bearing cage, permitting safe operation at a variety of operating speeds. The cage reduces rolling element sliding, contact and wear. Cages facilitate uniform load distribution by the elements in the bearing but carry no load. Cage faults can appear in the velocity spectrum in different forms based upon the bearing fault condition.



Fundamental frequency of cage alone



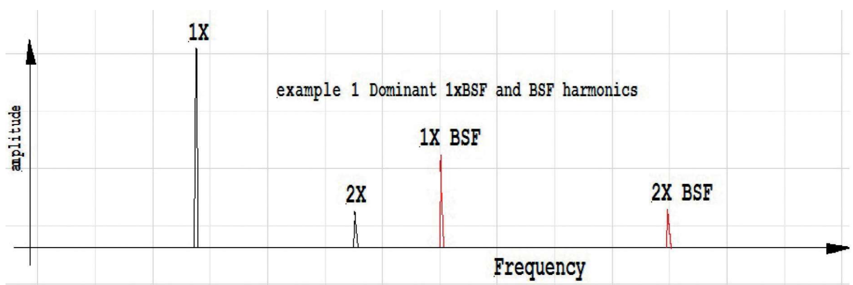
Fundamental train frequency and harmonics



Another bearing fault frequency plus sidebands thereof

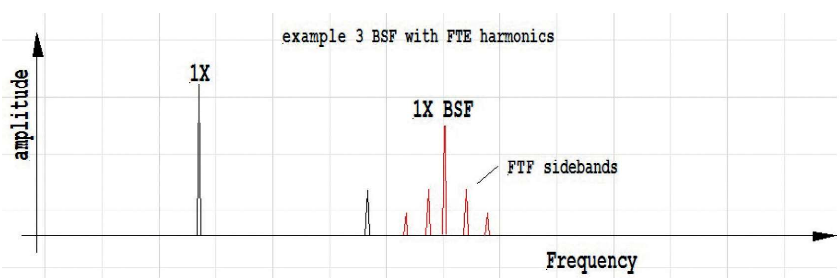
Roller / Ball fault components in velocity spectrum

The function of a ball bearing is to connect two machine members that move relative to one another in such a manner that the frictional resistance to motion is minimal. In most applications one of the members is a rotating shaft and the other a fixed housing. Separating these are the balls or rollers which are in effect the load carrying component of the bearing. Defects or damage to these rolling elements are shown in the FFT spectra.



Dominant ball spin frequency and harmonics (2x BSF).

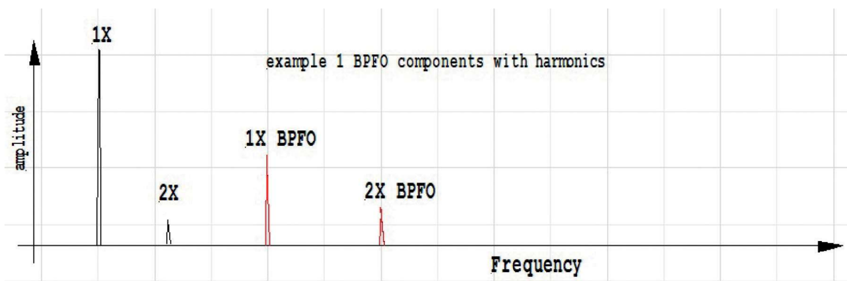
Dominant ball spin harmonics (2x BSF).



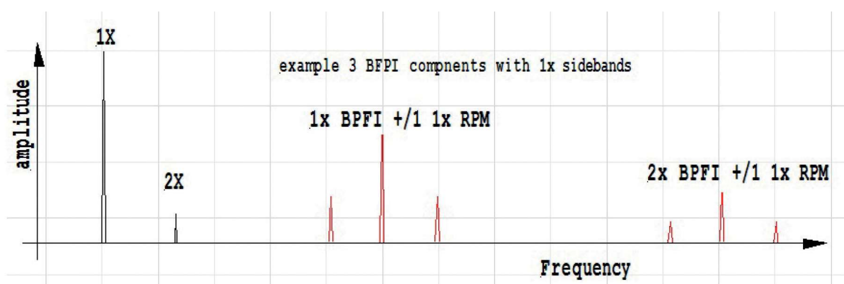
Ball spin frequency with fundamental train frequency sidebands.

Inner and outer race fault components in velocity spectrum

The inner and outer races of bearings are of course two separate bearing components. Each race is a ring with a groove where the balls rest. The groove is usually shaped, so the ball is a slightly loose fit in the groove. Thus, the ball contacts each race at a single point. However, a load on a small point would cause extremely high contact pressure. In practice, the ball deforms (flattens) slightly where it contacts each race, and the race also dents slightly where each ball presses on it. Depending on load, fitting, ingress of wear debris particles and lubrication, the inner or outer races can become permanently damaged. This damage shows up in the FFT spectra adjacent to the fundamental component frequencies calculated earlier.



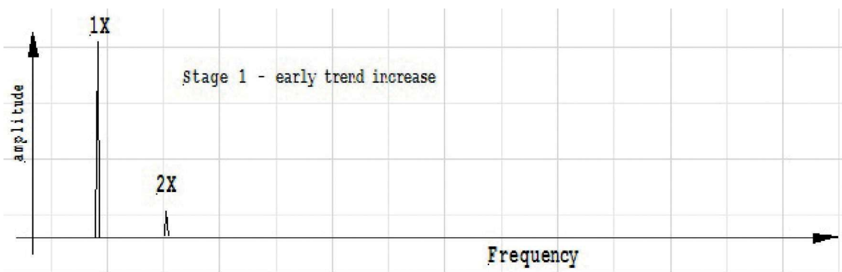
Outer race damage spectrum with harmonics.



Inner race damage spectrum with sidebands.

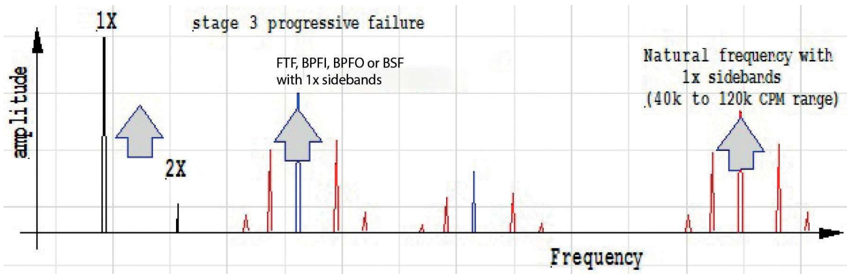
Bearing fault progress stages

The foregoing examples of bearing component failure are an example of the type of spectra that you can expect to find on a damaged bearing. It is more probable however that when analyzing a bearing there will be progressive damage occurring which progresses over time to a position where intervention is necessary. As discussed earlier, progressive damage is highlighted in a trend graph of the measurements. It is helpful, however, to be able during this trending to identify cause of damage and implement corrective action in order to extend the life of the bearing. At stages in the trend graph it is therefore useful to view the bearing FFT spectra. Four typical failure stages are illustrated in the following examples.

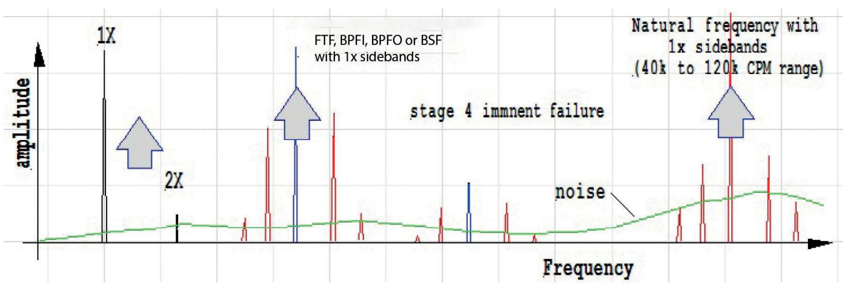


Early trend increase spectrum.

Continued trend increase spectrum: intervention stage.



Quickly increasing trend damage: intervention recommended.



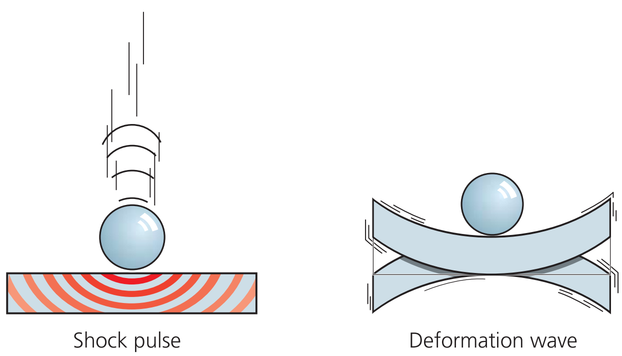
Rapidly increasing trend: failure imminent.

It more or less goes without saying, that you should try and intervene before you arrive at a stage 4 spectra. Of course it is not always possible to shut down production critical equipment. In such circumstances you should rely on the rate of increase and the amplitude of the trend graph as well as the measured spectra. BUT always be aware that catastrophic bearing failure usually does not begin and end with just the bearing failure. Bent shafts, coupling damage and a whole range of additional major component failures could result, in failure to intervene when an obviously damaged bearing has been identified.

Shock pulse evaluation

There is one non-FFT method widely used to assess the operating condition of anti-friction bearings. It is one of the most successful and popular techniques available and deserves mention in this FFT section, if only for the fact that it is a valuable first line of analysis for small bearings, where overall condition is used as the defining criteria for stopping a machine and changing bearings. It is that of **shock pulse evaluation**.

Shock pulses are a special type of vibration which must be clearly distinguished from ordinary machine vibrations. The actual shock pulse is the pressure wave generated at the moment when one metallic object strikes another. The bulk of the impact momentum, however, acts to deform the target object, which then oscillates at its natural frequency. This vibration ultimately dissipates, primarily as heat, due to internal friction (material damping).

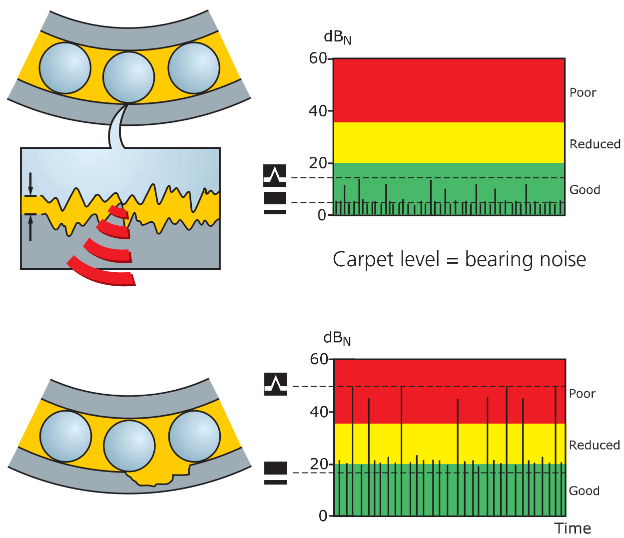


Shock pulses occur during bearing operation when a rolling element passes over an irregularity in the surface of the bearing race. Of course, there is no such thing as a perfectly smooth surface in real life. Even new bearings emit a signal of weak shock pulses in rapid succession. This 'carpet level' rises when the lubrication film between rolling elements and their races becomes depleted.

A defect (pit or crack) on the surface of a rolling element or bearing race produces a strong shock pulse with up to 1000 times the intensity of the carpet level. These irregular peaks (the 'maximum value'), which stand out clearly from the background level, are ideal indicators of bearing damage.

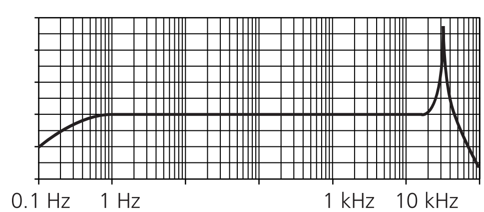
Shock pulses propagate within a much higher frequency range than that of ordinary machine vibration, and their energy content is much weaker. Therefore, the accelerometer used for shock pulse measurement has a resonance frequency (approx. 36 kHz) that lies precisely within this range.

Shock pulse diagrams for good and damaged bearings



Signal peaks above carpet level. Signal strength is shown in decibels (logarithmic scale) for clarity. 60 dB represents a change of 1000x.

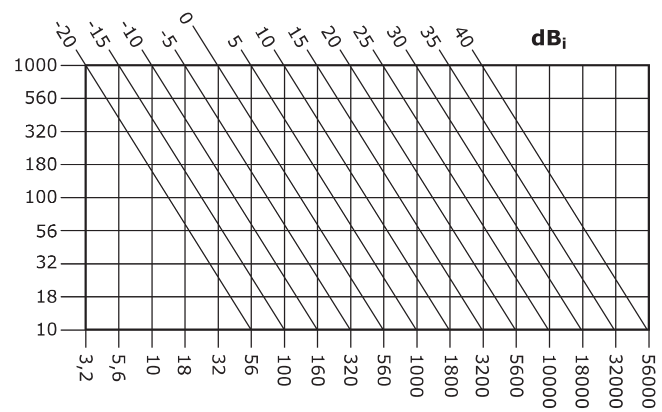
This means, that in this high frequency range of particular interest for bearing condition evaluation, the transducer is especially sensitive to the shock pulse signal – even when far more energetic machine vibration occurs at lower frequencies (for example, due to unbalance or shaft misalignment) or from adjacent machines. And since high frequency signals tend to dissipate rapidly, very little interference is encountered from adjacent bearings.



Evaluating bearing condition

Just as with other condition evaluation methods, the shock pulse technique reaches its conclusions via certain defined parameters. These are influenced by factors such as bearing size, rpm, signal damping and lubrication. Shock pulse readings generally should be compared with 'signature' readings, taken when condition is known to be good, or normalized to take these factors into account.

Over the years reliable normalization methods have been developed based upon extensive measurements, to calculate the effect of bearing size and rpm on shock pulse readings of new, perfect bearings. The normalized signal level (dB_n) calculated for an actual bearing allows its condition to be rated directly as 'good', 'reduced' or 'poor'.



Normalization graph plots bearing diameter over machine speed

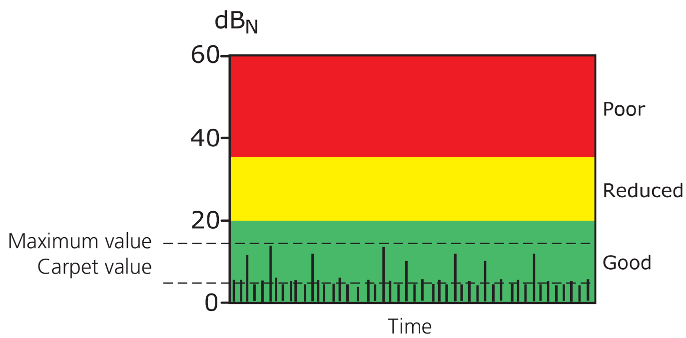
Two normalized parameters are used to determine bearing condition.

- The **carpet value** indicates deteriorating or poor operating condition (e.g. caused by insufficient lubrication, shaft misalignment or improper installation).

Damaged bearing elements, in contrast, generate individual shock pulses of greater intensity.

- The resulting **maximum value** is a direct indication of bearing operating condition.

Specific types of damage can be recognized not only from the absolute signal amplitude, but also from the difference to the carpet level and the pattern of pulses. Comparison with typical shock pulse diagrams often shows clearly where the problem lies (e.g. 'lubricant contamination').

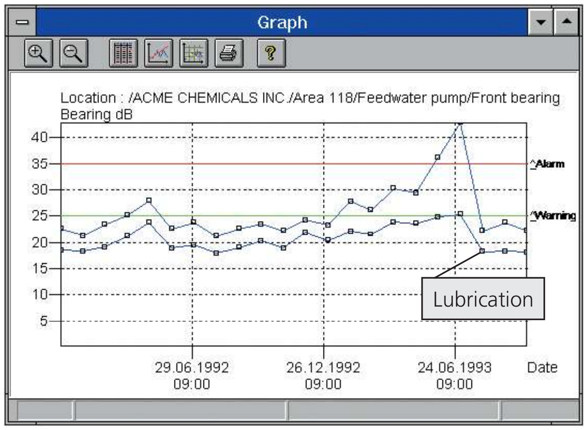


Shock pulse comparison chart

The above diagram shows a typical shock pulse graph where bearing condition over time has changed. The upper line of the graph is the normalized overall dB_n measurement of bearing condition and the lower line is the carpet level measurement dB_c. As can be seen, normalized and carpet levels trend much the same in a good-operating-condition bearing. A developing problem is indicated when the level trends begin to separate, for example by an increase in the dB_n (upper) trend. An increase in the dB_c (carpet) level indicates a potential lubricating film breakdown which can often be resolved by lubrication.

The dB_c trend provides important information regarding lubrication, mounting and loading condition of the bearing. It is related directly to the fluid film thickness at the rolling component interface.

The dB_n trend gives information on irregularities in the bearing surface which give rise to single shock pulses at random intervals. These high values give a good indication of damage already done to the bearing and the overall condition.



Typical shock pulse trend graph

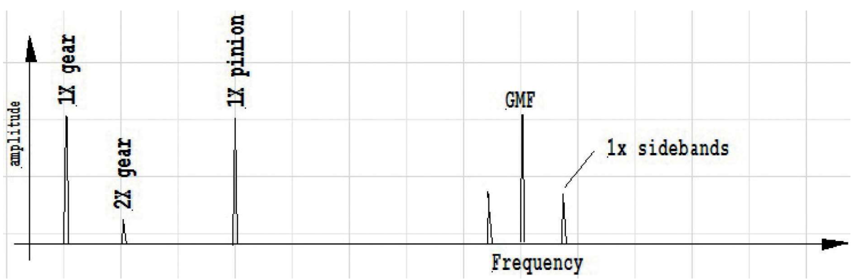
If after lubrication, the dB_c falls below alarm levels and stays there, it is a good indication that intervention in the form of lubrication has brought the bearing operation back within acceptable operating limits. An increase in dB_n over dB_c trends indicates a more fundamental problem unlikely to be improved by lubrication.

Gear and gearbox faults

Gearboxes and more specifically gear meshing and wear problems occupy a significant section in the analysis of rotating machinery. Complex gearbox design, planetary gear systems and the fault analysis thereof can be a daunting task for the condition monitoring engineer no matter what level of training or experience. It is not the plan of this publication to delve too deeply into the analysis of gears and the complex fault analysis that can be required. Instead we will look at a typical spectrum of a gear system in a good condition and at three common and relatively easy to detect problems:

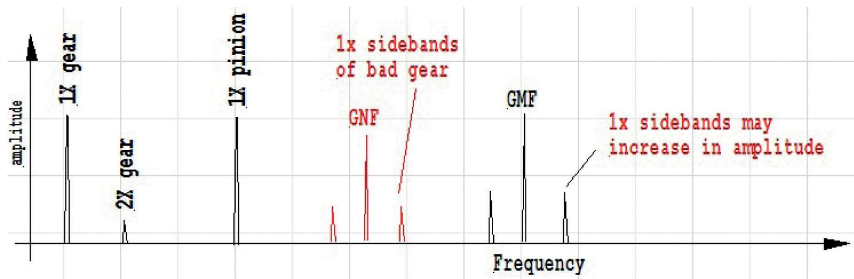
- Excessive gear tooth wear
- Excessive loading on gear teeth
- Mechanical misalignment between gears

The following four spectra graphs illustrate the basic FFT spectra you would expect to see given these problems.



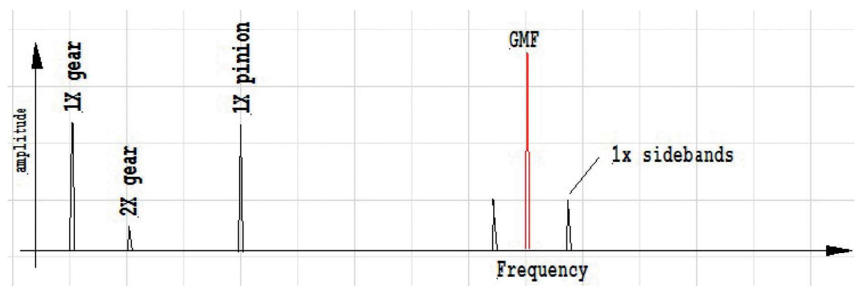
Expected spectrum of a gear in a good condition

This spectrum is showing 1x and 2x frequencies of the gear plus 1x the frequency of the pinion and the Gear Mesh frequency (GMF) with 1x sidebands. There are no gear natural frequencies showing and all peaks are of low amplitude.



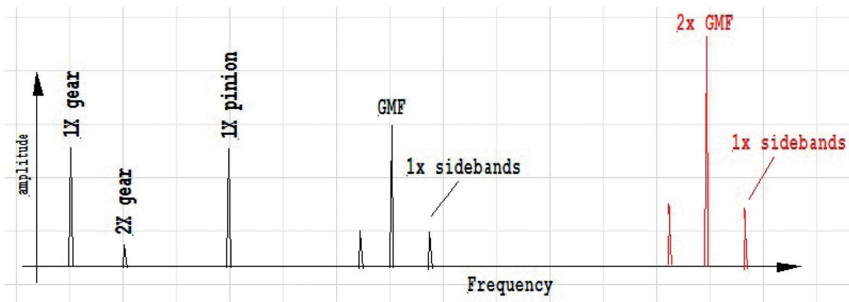
Spectrum of a gear assembly with worn teeth

The above spectrum shows tooth wear which excites the gear natural frequency with 1x sidebands of the bad gear. Gear Natural Frequency (GNF) sidebands may also increase in amplitude. Other frequency components are similar to what you would see in a good gear spectrum.



Spectrum of a gear assembly with excessive loading

High loading of gear teeth will show an increase in the Gear Mesh Frequency amplitude (GMF) and very little change in sideband amplitude.

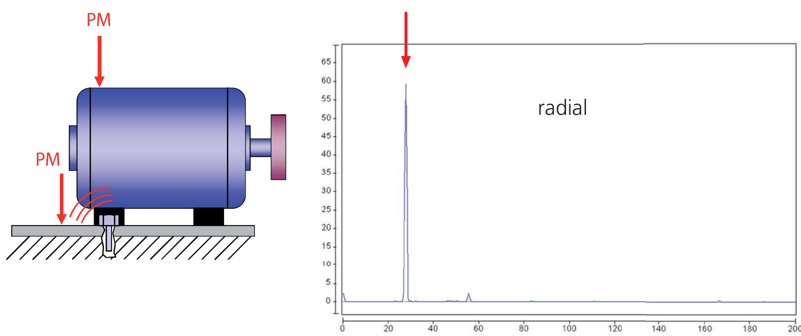


Spectrum of a gear assembly with excessive misalignment

In this example, gear misalignment excite 2nd harmonic of the Gear Mesh Frequency (GMF) and even 3rd harmonic in some cases. The frequencies 1x and 2x are lower compared to 2x GMF.

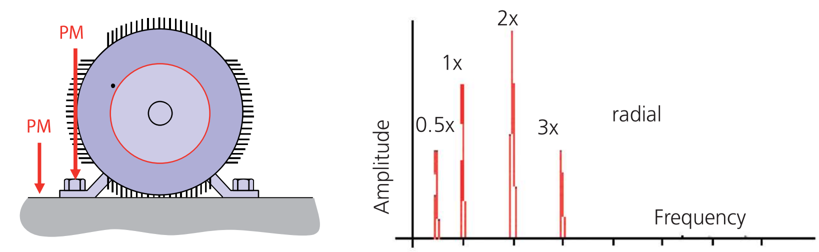
Looseness fault analysis

Looseness of components, poor structural integrity, cracked or broken holding down bolts and poor foundation condition can cause untold problems for the engineer analyzing FFT data. It is often difficult to establish from a spectra since it often manifests itself as another common fault such as unbalance or even misalignment. The users EYES are often the key to unlocking looseness FFT diagnosis. Always look for any obvious problems such as a broken foot. This makes the diagnosis much easier. Below are three typical looseness spectra.



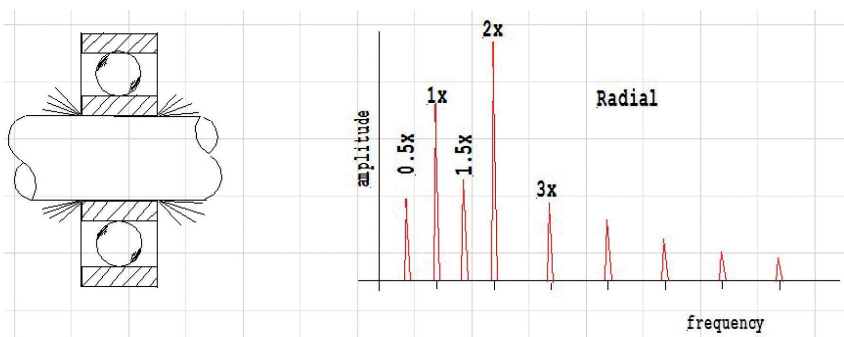
Structural looseness

Structural looseness is caused by a weakness of machine components, typically machine soft foot, baseplate or foundation distortion, or even poor baseplate design creating flexing of the baseplate. This manifests itself as a strong 1x component measured in the radial direction and is therefore frequently misdiagnosed as a form of static unbalance.



Mechanical looseness caused by loose holding down bolts

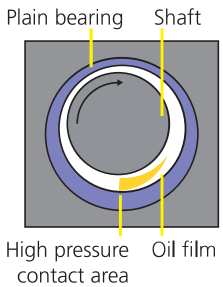
This type of looseness is often easily identified by observation, loose bolts or cracked or broken machine feet, identified by 0.5x 1x, 2x, and 3x cpm, measured in the radial direction.

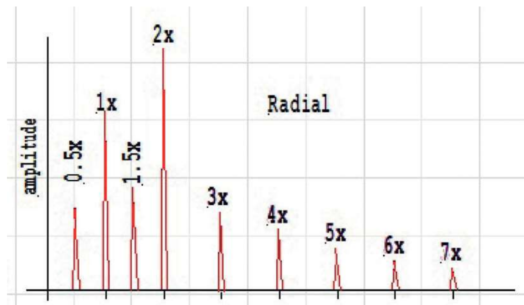


Looseness caused by poor fit components

Component fit looseness can be caused by poor fitting of bearings or excessive clearance of fan impellers on the shaft. There may be a phase change from one measurement to another, generating numerous harmonics 1x, 2x, 3x; and 0.5x, 1.5x, 2.5x may also be present in the spectra.

Journal bearing wear will also show up as a form of mechanical looseness.



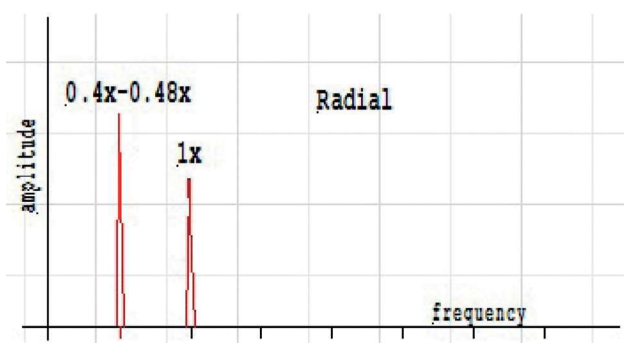


Journal bearing wear

Vibration readings to establish journal bearing wear can be taken from the shaft or the bearing housing. The FFT spectrum generated is very similar to component looseness with 1x component plus many harmonics in the radial vibration spectrum. In very severe cases, peaks may also show at 0.5x, 1.5x etc. The most definitive way to establish actual journal wear is via oil analysis and wear particle debris analysis.

Journal bearing wear is frequently caused by **oil whirl** which occurs when a lubrication wedge cannot form in the high pressure contact areas of the assembly but instead whirls around the bearing. This leads to direct metal to metal contact between shaft and bearing which quickly wears out the bearing.

Sub-synchronous components between 0.4x and 0.48x appears in the spectrum with an unstable amplitude.



Journal bearing oil whirl

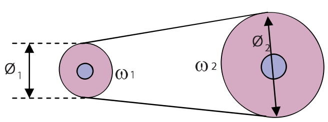
Belts and pulleys

During operation a flexible belt experiences three types of tension as it rotates around a pulley.

- Working tension (tight side – slack side)
- Bending tension
- Centrifugal tension

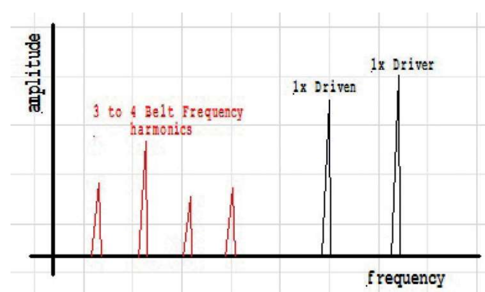
Belts are designed to withstand these working operation states, provided that pre-selection of the belt meets the operating criteria. The design life of the belt will be met and usually exceeded provided that no other forces other than the above act upon the belt during its operating life.

Forces, such as misalignment and loose or over tight belt tension, are killers of flexible belt drives. Poor pulley alignment can reduce useful operating life by as much 80%. Furthermore, not paying adequate attention to basic installation requirements leads not only to belt wear but may also damage pulleys, bearings and seals. FFT analysis can identify many of the "belt drive killers". The following example provide an insight into what you should look for when analyzing belt drive spectra.

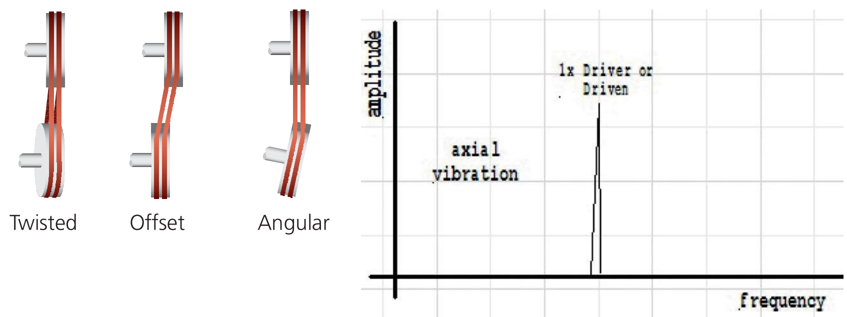


$\omega_1 * \Phi_1 = \omega_2 * \Phi_2$
$f_B = \pi * \omega_1 * \Phi_1 / l_B$, where

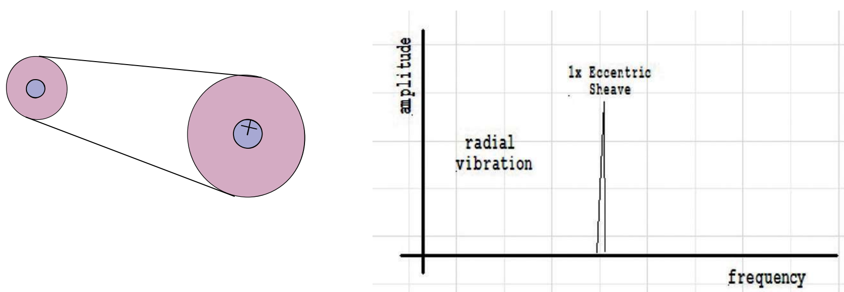
f_B: Belt frequency
l_B: Belt length



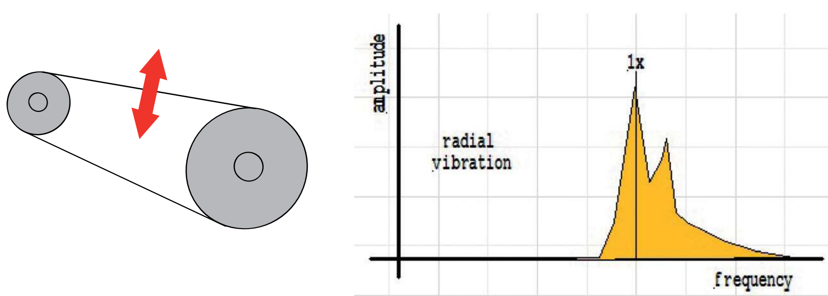
Worn, loose or mismatched belts



Misaligned belts



Eccentric sheave



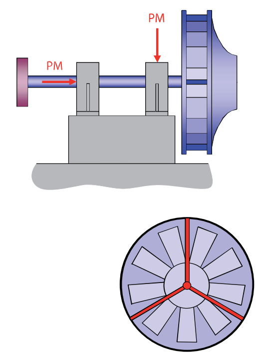
Belt resonance

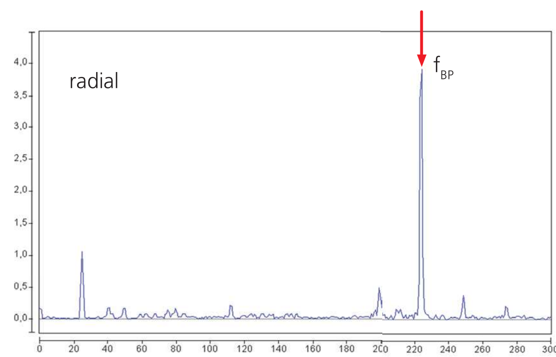
Belt resonance occurs when the belt natural frequency approaches or co-incides with the drive or driven frequency. To correct this, change the belt tension to change the belt natural frequency.

Fans

Most fans we encounter in standard process applications are either axial-flow propeller type or centrifugal. Such fans are prone to an uneven build up of debris on the blades, particularly when handling particle laden air in applications such as car manufacturing paint shops. Particle build up leads to fan unbalance. FFT analysis will quickly show up such operational problems. Fans are also frequently belt driven which also gives rise to typical belt drive FFT characteristics as described in the previous pages.

If however during the course of analysis, unbalance and misalignment can be eliminated from elevated FFT readings on a fan, then the likelihood is that there is some form of mechanical damage that the fan has suffered. This could vary from an extreme problem such as a missing blade to a cracked or chipped blade tip. FFT investigation of this problem requires that the analyst knows the blade pass frequency (f_{BP}) of the fan. It is not a complicated calculation, it is simply the number of blades multiplied by the rpm of the fan itself.





$$f_{BP} = B_n * N$$

B_n = Number of blades or vanes
N = Rotor speed in rpm

Example calculation of a fan with 9 blades and 3 support struts on the intake housing:

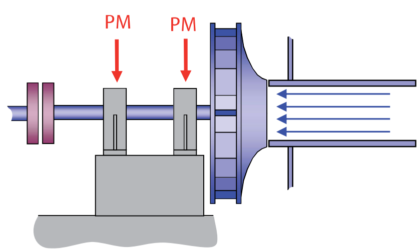
3 struts on intake	$x = 3$
9 blades	$B_n = 9$
Rotor speed	$N = 600$ rpm

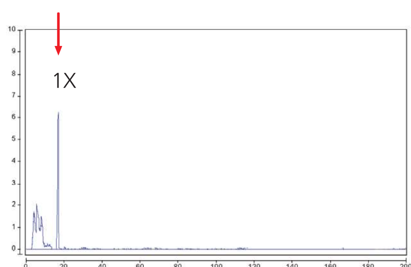
$$f_{BP} * x = N * B_n * x = 600 * 9 * 3$$

Characteristic frequency = 16,200 cpm

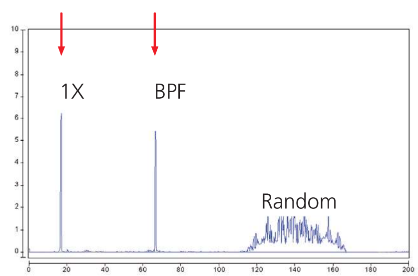
Aerodynamic and hydraulic forces

This is a problem that can affect both fan and pumps assemblies and is associated with fluid and air movement through the structure. The problems can be classified as either **cavitation** or **turbulence**. As can be seen in the following examples, both offer distinct FFT spectra.





Turbulence FFT spectrum



Cavitation FFT spectrum

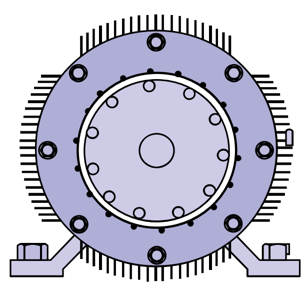
Electric motors

Probably the most common equipment in any process plant is the electric motor. Almost everything is driven by them, and therefore, as a distinct item of equipment, we should consider the electromechanical faults that can arise from their operation. Of course, since they are invariably coupled to another component (fan, pump, etc.), they are subject to the same basic component faults such as unbalance, misalignment, looseness, bearing problems etc,(covered earlier in this section). However, electric motors are rather complex and suffer from electromechanical problems which are distinctly unique to themselves.

Essentially motors consist of a stator (laminated steel sections wound by copper wire or bars) and a rotor (again a laminated steel section wound by copper wire or bars). The rotor is supported on bearings and is separated from the stator by an air gap of a given dimension. Two main types of motors are synchronous and induction motors. They differ in that the synchronous motor has a permanently magnetized rotor which is rotated by the stator “dragging” it round by magnetic attraction. The induction motor is different in that the rotor is not a permanent magnet but an electromagnet. This is determined by the design of the rotor with rotor bars embedded into the laminations. The rotor bars are connected to each other at each end with a continuous copper ring. The induction motor works by magnetic repulsion rather than attraction as with the synchronous motor. Both however experience similar electromagnetic faults in operation.

If you suspect a motor has electromagnetic problems a first useful step is to disconnect it from the driven component and carry out an FFT analysis with the motor running alone. Electric motor problems can be roughly classified as below.

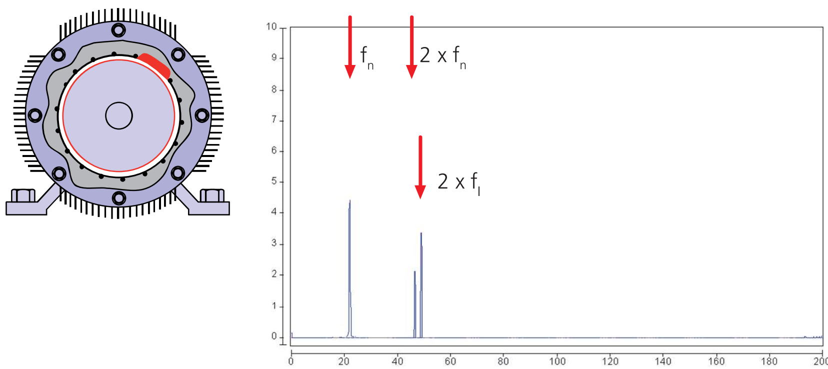
- Stator eccentricity
- Rotor eccentricity
- Rotor problems
- Loose connections



Stator eccentricity

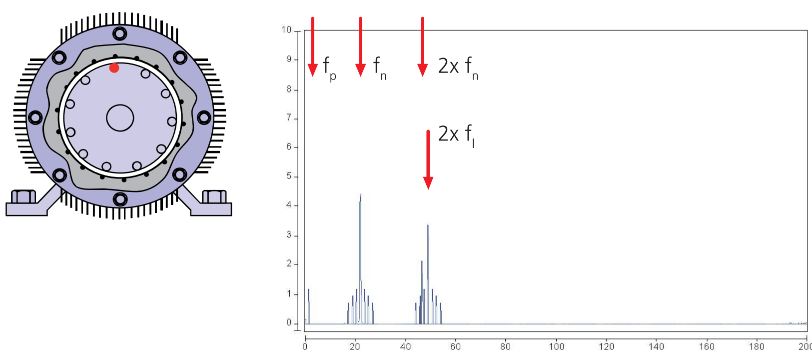
Caused by loose iron (laminations), shorted stator laminations, and soft foot.

High 1x f_n and 2x f_n signals; 2x f_l (twice line frequency) without sidebands. Radial measurements predominant. High resolution settings should be used for measurement.



Rotor eccentricity

Caused by rotor offset, misalignment and poor base; f_p, 1x, 2x and 2f_l signals; 1x and 2f_l with sidebands at f_p (pole pass frequency). Radial measurements predominant. High resolution settings should be used for measurement.



Key parameters

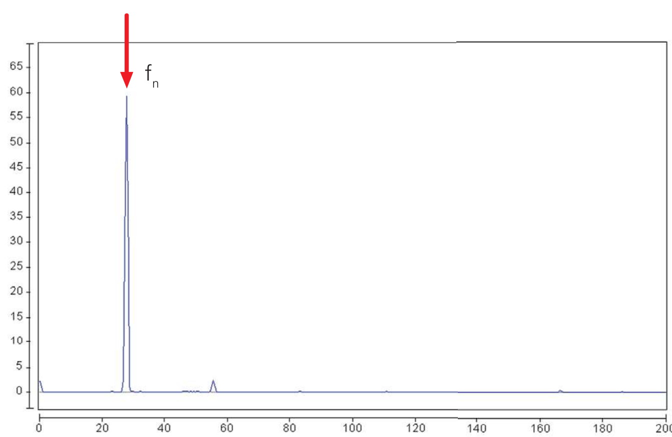
Twice line frequency vibration:	$2*f_l$
Bar meshing frequency	$f_{bar} = f_n * n_{bar}$
Synchronous frequency:	$f_{syn} = 2*f_l / p,$
Slip frequency:	$f_{slip} = f_{syn} - f_n$
Pole pass frequency:	$f_p = p * f_{slip}$

where

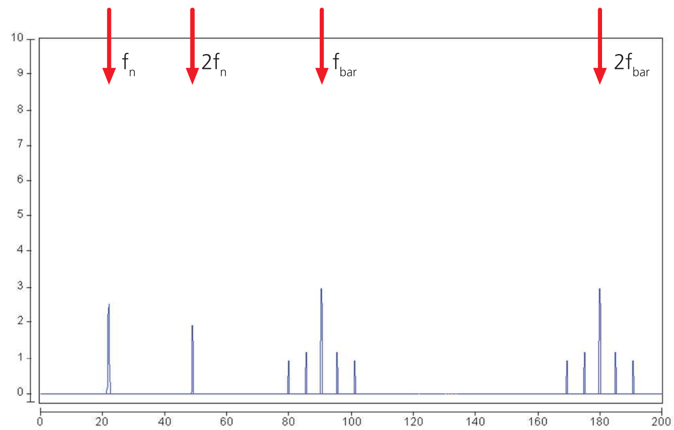
f_n :	rotational frequency
n_{bar} :	number of rotor bars
p :	number of poles

Other electromechanical faults

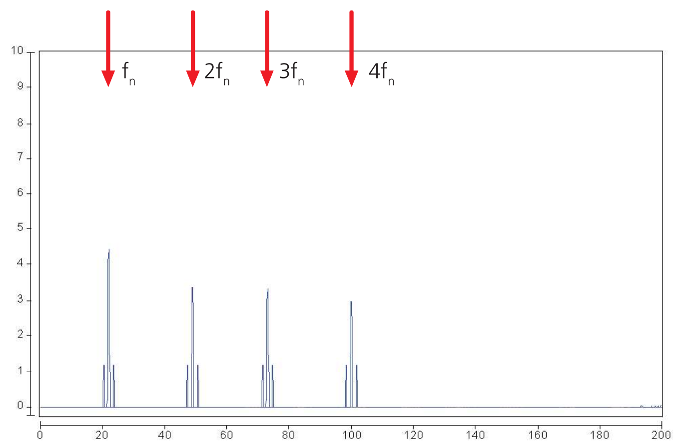
Uneven heating of the rotor due to unbalanced bar currents will cause the rotor to warp (bow). This causes unbalance with the characteristic FFT signature. It can be identified because the symptoms disappear when the motor is cold.



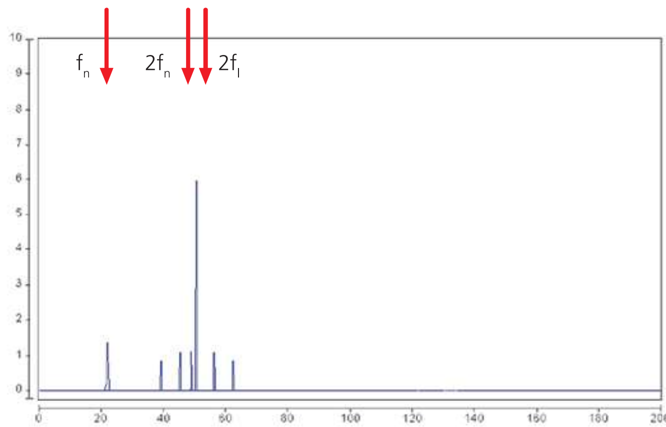
Rotor bow



Loose rotor bars: f_{bar} and $2f_{bar}$ with $2f_l$ sidebands;
$2f_{bar}$ can be higher; 1x and 2x can be present



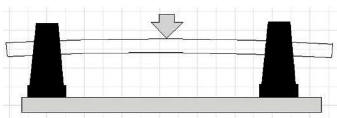
Cracked rotor bars: 1x and harmonics with sidebands at f_p;
high resolution spectrum required; possible beating signal

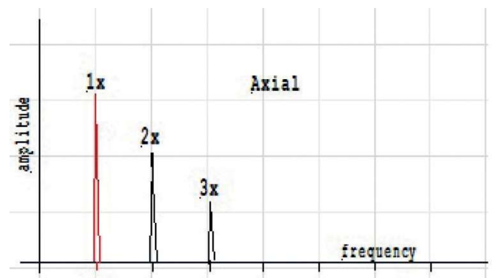


Loose connections: $2f_l$ excessive signal;
electrical phase problems; correction must be immediate

Bent rotor or shaft

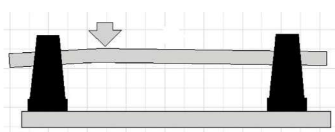
Bent shaft at center

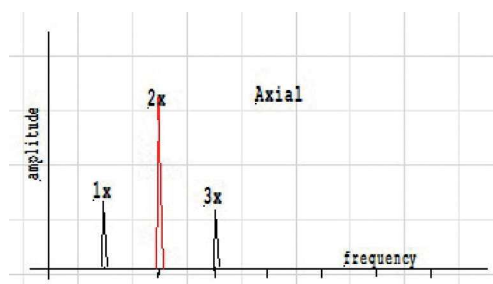




Bent shafts generates high radial and axial loads. Axial vibration shows up at 1x, 2x and 3x components. 1x is dominant if the bend is near the shaft center; and a 180 degree phase shift in the axial direction. Phase measurements are essential in this diagnosis.

Bent shaft at one end





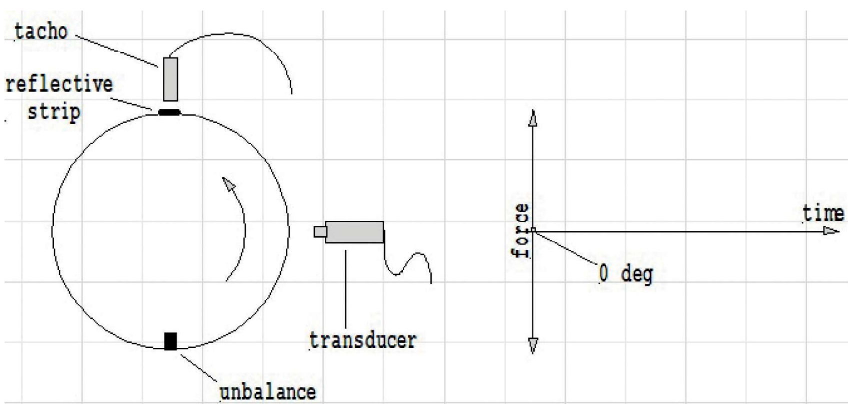
Bent shafts generates high radial and axial loads. Axial vibration shows up at 1x, 2x and 3x components. 2x is dominant if the bend is near the shaft center; and 180 degree phase shift in the axial direction. Phase measurements are essential in this diagnosis.

Phase

Phase measurements are a very useful tool for diagnosis of a number of common rotating machine conditions, such as misalignment, unbalance or bent shafts. It is therefore necessary to have an understanding of the measurement technique and what in fact you are measuring. Phase is a measure of the time difference between two sine waves.

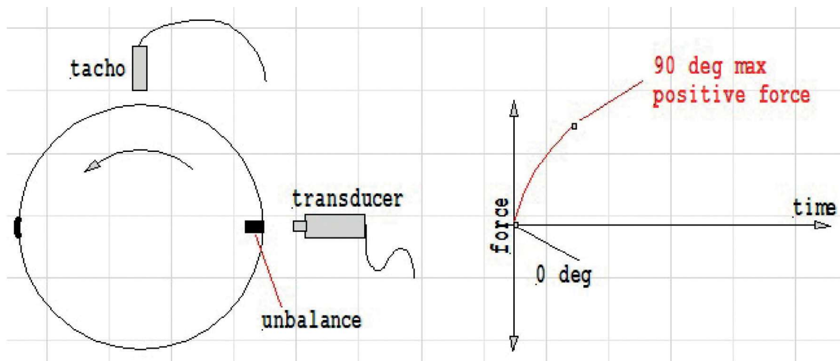
For clarity of explanation, we use in the following examples a vibration transducer to sense the imbalance force and a light sensitive tachometer ("tacho") and a reflective strip attached to the shaft to sense shaft position.

The phase angle is the angle in degrees that the shaft travels from the start of data collection to the position when the vibration transducer measures the maximum positive force of imbalance.



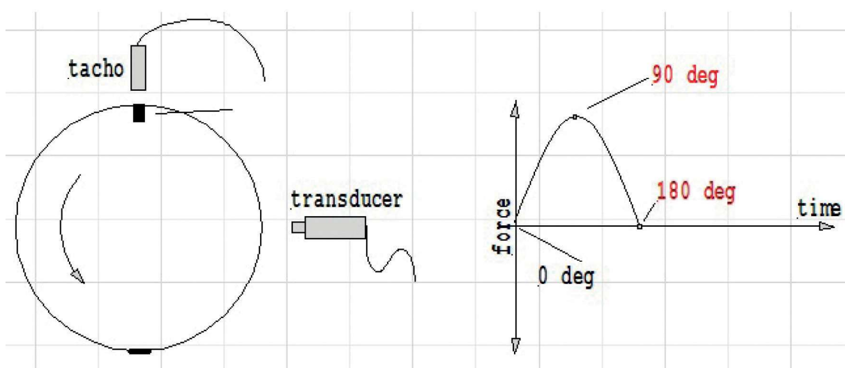
Shaft at start

The tacho senses the reflective strip and starts the data collection. At this point phase = 0.



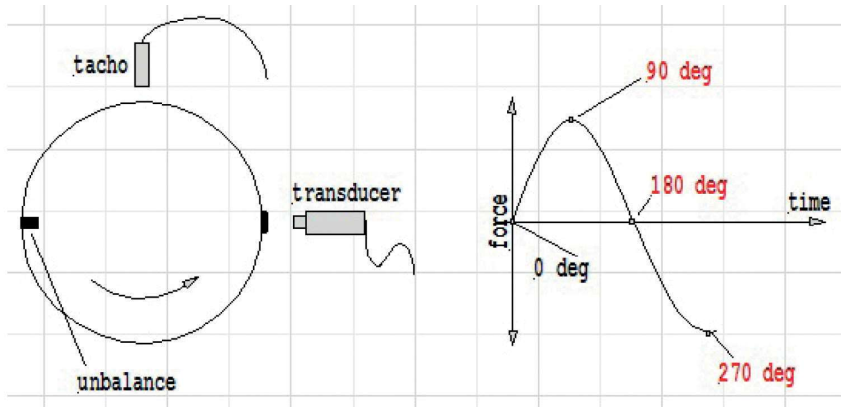
Shat rotated through 90 degrees

The imbalance force has rotated through 90 degrees. At this point the imbalance force produces the highest positive reading at the transducer. As the imbalance is traveling towards the transducer, its force is considered to be in a positive direction.



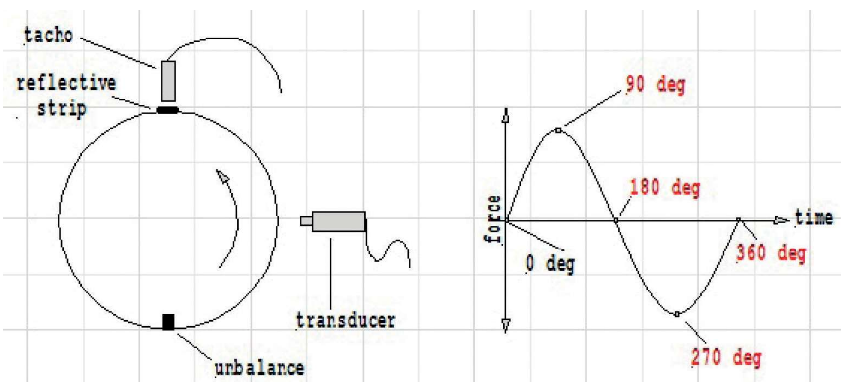
Imbalance travels a further 90 degrees

The force experienced by the transducer is zero.



Imbalance travels a further 90 degrees

The imbalance is now opposite the transducer. At this point the force produced is at its highest negative reading from the transducer. Force is considered to be in the negative direction.



Imbalance travels a further 90 degrees

The imbalance force has completed its 360 rotation and the force experienced by the transducer is again zero.

In the preceding examples the mounting angle between the transducer and tacho for simplicity is shown as 90 degrees. This is not an absolute necessity. They can be mounted in the same plane or indeed 180 degrees apart. The key is the use of the tacho and transducer together to initialize and measure the phase shift of the machine. The examples also use a simple static unbalance to explain the principle of phase measurement.

Phase is a key component of FFT diagnosis. Without carrying out phase measurements, it is often impossible to accurately distinguish between faults such as imbalance, misalignment, bent shafts or other low frequency problems, which manifest themselves in 1x, 2x or 3x of the machine fundamental frequency.

PRUFTECHNIK data collectors and vibration analyzers

The vibration analysis functions and methods described in the previous pages can all be performed by the PRUFTECHNIK **VIBXPERT® II** system. Essentially VIBXPERT II is a modular system which can be configured to perform a number of vibration related tasks including the following:

- Route-based data collection
- Dual channel FFT analysis
- Time waveform analysis
- Dynamic Balancing