

**The Lifetime Reliability Solutions
Certificate Course in Maintenance and Reliability
Module 2 – Machinery Vibration Analysis Fundamentals**

**Session 7
Vibration Analysis for Fault Diagnosis**

1. Principal Causes of Failure in Machinery.

There are numerous causes that could be listed for machinery failure. A few faults might arise from particularly aggressive service environments or operating conditions. However, it can be argued that most faults are attributable to either **design limitations** in the machine or **inaccurate assembly, installation or alignment**.

It follows therefore that there must be a reason for most failures. It might be said that the most common fault is bearing failure. But why did the bearing fail? Perhaps it was subject to forces beyond the design assumptions arising from misalignment or unbalance. Or perhaps the lubrication arrangements were inadequate or contamination got into the bearing by some means.

Hence the growing interest in the application of Root Cause Failure Analysis and we will deal with that later. In the meantime we will examine the diagnostic processes used to analyse vibration data to find the causes for excessive vibration and therefore the reasons for potential failure in a machine.

These are some of the most common faults typically found as a result of vibration surveys:

- Misalignment
- Unbalance
- Resonance
- Bearings
- Looseness
- Flow-related problems
- Electrical
- Bent Shaft
- Gear Mesh

Of all of these, there are four that comprise more than 90% of all faults reported; **unbalance, misalignment, looseness and bearing failure**. In this session we are going to look in detail at the processes of diagnostic analysis and particularly the **Top 4 Faults** that are generic to most rotating machines.

It must be remembered that the vibration data we are analysing arises from forces generated in the machine. Each force or combination of forces must have an identifiable source. The task of the analyst is to determine the source of those forces and then decide whether the resulting vibration represents a problem.

To complicate things further, the path from the source of the vibration force (eg unbalance) to the point where it is measured has certain mass, damping and stiffness qualities so that the amplitude of the resulting vibration when measured is modified by those characteristics along the way.

The good vibration analyst needs to have a good appreciation of the dynamic response of the machine and structure being analysed if the diagnosis is to be credible.

2. Important Applications for Time Waveform Analysis.

The time waveform, as we have discussed before, is the summation of all the time components measured by the transducer at the particular point of measurement. It is therefore called a ‘complex time waveform’ and, because it is complex, it doesn’t lend itself easily to trending or comparison from one survey to the next. Generally, severities only greater than 2g are considered to need analysis or attention.

However, to the trained eye it does contain vital information for certain applications which we list now.

- Low speed applications (less than 100rpm).
- Indication of true amplitude in situations where impacts occur, such as assessment of rolling element bearing defect severity.
- Gears.
- Sleeve bearing machines with X-Y probes (2 channel orbit analysis).
- Looseness.
- Rubs.
- Beats.
- Cyclic or variable speed equipment.

The kind of vibration data produced by these sorts of forces acting in a machine may not be completely expressed in a frequency spectrum. Therefore it is diagnostically useful to look at the original time waveform to learn more about the nature of the vibration.

3. Guidelines for Domain Selection.

The following table indicates when Frequency, Phase and Time should be examined for diagnostic information.

Application/Problem	Spectrum/FFT	Phase	Time Waveform
Unbalance	X	X	
Misalignment	X	X	
Resonance	X	X	X
Rolling Element Bearing	X		X
Sleeve Bearings	X	X	X
Gears	X		X
Electrical	X		X
Looseness	X	X	X
Flow Problems	X		
Very Low Frequency			X
Cyclic Vibration			X
Variable Speed	X		X

The following notes will also be of value to practitioners using time domain analysis.

- Because of the uncertainty of phase shift in the integration process, do not integrate the time waveform but work in the transducers native units – usually acceleration.
- Overlap averaging should be disabled.

- Windows as required for FFT processing can, in some instruments, corrupt the time data. A Uniform or Rectangular window should be applied.
- For normal “snap shot” time waveforms use 1 Average of the data. If more than one is used it will simply provide the last sample and memory will be wasted.
- To acquire data with sufficient resolution use 4096 samples (equivalent to 1600 lines). However, for very high frequency events the samples may well be reduced to achieve the small time period required.
- The time period sampled should be sufficient to collect 6 – 10 repetitions of the event of interest, or cycles of the shaft under review.
- Synchronous Time Averaging can be used to provide data only for the shaft under review – this is particularly useful in looking for cracked gear teeth, also in reciprocating plant for linking time events to a specific crank angle.
- Transducer Mounting. Hand-held measurements must not even be considered for data to be analysed in the time domain.

Some further notes on Time Waveform Patterns is given as an Appendix at the end of these session notes for those who may wish to pursue this information further.

4. Guidelines for Frequency Domain Set-up.

In Session 1 we looked briefly at the FFT process and some classic frequency conversions. We will not go any further into the detail of how the FFT works. However, one of the important considerations in setting up a database is the resolution of the frequency domain.

As discussed in Session 1, the resolution of the frequency domain is a trade-off between processing time and required accuracy of frequency data. A good choice on a modern machinery analyser is a spectrum resolution of 800 lines. If we use a frequency bandwidth of 800Hz then the resolution of frequency analysis will be around 1Hz. Note that there must also be a correction for a Hanning window.

Guidelines for frequency bandwidth for common machinery are as follows:

- Velocity – 20x the speed of the fastest shaft **if** an acceleration spectrum is also being taken.
- Velocity – 2 kHz **if** there is no acceleration spectrum being taken.
- Acceleration – 50x the speed of the highest shaft **or** 3.2x highest gear mesh frequency **or** 2 kHz – whichever is the greater.

5. The TOP 4 Faults.

The chart on the next page has been drawn up from years of experience in the vibration analysis business. It is essential to keep in mind that the defect characteristics presented in this chart, and indeed any diagnostic chart, are not definitive. The characteristics for a particular defect may vary significantly between different machine types and also be influenced by speed and load. A diagnosis must be supported by as many other verifying tests as is possible.

The detail of the particular methods used for bearing fault detection and analysis will be dealt with in a later session. This is because special measurement techniques are used to provide early detection of faults long before the results become evident in the whole-body vibration of the machine. A few notes on each of the faults described.

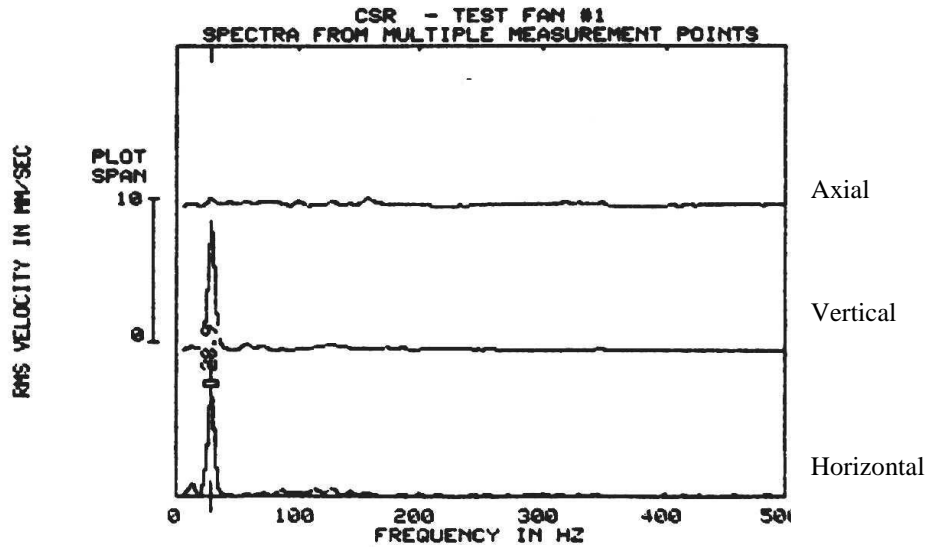
TOP FOUR MACHINE FAULTS

FAULT	MECHANICAL DESCRIPTION	VA IDENTIFIERS	LIKELY CAUSES	POSSIBLE SOLUTIONS
UNBALANCE	Rotor centre of mass is displaced from centre of rotation. Result is a radial force proportional in magnitude to eccentricity and shaft speed squared.	Dominant shaft x1 radial. Phase at x1 rock steady. Horizontal and vertical displacement will usually be similar. Amplitude usually stable. Axial vibration usually low.	<ol style="list-style-type: none"> Rotor was not correctly balanced when new or rebuilt. Rotor wear, corrosion or build-up of solids. Loss of balance weights. Shaft may not be straight. Machine could be operating close to a structural or shaft natural frequency. 	<ol style="list-style-type: none"> In-situ balance, or remove for shop balancing. Clean and rebuild rotor. 'Clock' shaft and straighten or replace. Do impact tests or take amplitude/phase data on rundown.
MISALIGNMENT	In a perfectly aligned and precision made coupling assembly, the transfer of torque from one shaft to the other takes place without any radial or axial forces being developed. This is never realised in practice and the degree in which axial and radial vibration forces are developed is a measure of the errors of machining and alignment.	Identifiers vary, will include: Vibration is directional and Phase measurement may be helpful. Parallel Misalignment Shaft x2 Phase at x1 or x2 is 180° shifted radially across coupling. Angular Misalignment Shaft x1, x3, x5 Axial vibration often higher than radial. Vibration of similar spectral shape is usually observed on BOTH machines.	<ol style="list-style-type: none"> Poor alignment procedure or incorrect tolerances. Poor bedplate or sub-frame condition or design. 'Soft-Foot' conditions may be present. Incorrect allowances for thermal growth. <p>Note: Poor coupling condition or fit on shafts can often give symptoms similar to misalignment.</p>	<ol style="list-style-type: none"> Realign to tolerance with new shims and clean surfaces. Check vibration at mounting points to observe any weakness or looseness. Check alignment immediately after hot shutdown. Then again when cold. Note the differences.
LOOSENESS	Looseness is seen as a non-linear response to the absorption of forces in a machine. Can be 'Gravitational' (such as broken motor foot), or 'Rotational' (such as loose bearing in housing).	'Gravitational' Looseness – Shaft x2 dominant. 'Rotational' Looseness – Shaft x2, x4, Coupling Looseness – Shaft x3, x5, x7. Time Domain truncated or offset. Sub-harmonics may be present. Phase will vary between starts	<ol style="list-style-type: none"> Mountings loose or broken. Shaft loose in bearing, or bearings loose in housing. Coupling looseness can produce a 'square wave' effect, hence the odd orders. 	<ol style="list-style-type: none"> Check and tighten. Use vibration meter while machine is running to find looseness. Do lift test if possible to confirm. Dismantle and inspect. Look for fretted keyways and other signs of 'working' If in doubt re-fit to ensure tightness.
ROLLING ELEMENT BEARING FAILURE	Bearings can fail in numerous ways. The two most common are: Pitting or marking of the rolling elements or raceways is preceded by low amplitude shocks in the bearings. Abrasion of the cage, raceways and elements results in looseness, sometimes without any discernible high frequency vibration or shock.	Best identifier of 'pitting' and similar conditions are proprietary high frequency bearing fault detection systems. Bearing wear is harder to identify and may first manifest as an increase in harmonics of shaft speed. Note that velocity measurements should always be examined for signs of looseness.	<ol style="list-style-type: none"> Pitting may be caused by fatigue, false brinelling or incorrect fitting. Ingress of abrasive material or corrosive liquids into bearing. Bearing overheated. Bearing overloaded. Note: Axial thrust overload is a common cause of failure. 	<ol style="list-style-type: none"> Replace bearing. Conduct Root Cause Analysis on removed bearing. Examine/upgrade seals. Installation or lubrication faults. Study failure mode and re-design bearing system.

5.1 Unbalance.

Note that the word is Unbalance, not Imbalance as is often used in US publications. AS 2606 which defines the vocabulary of shock and vibration requires that the word Unbalance be used.

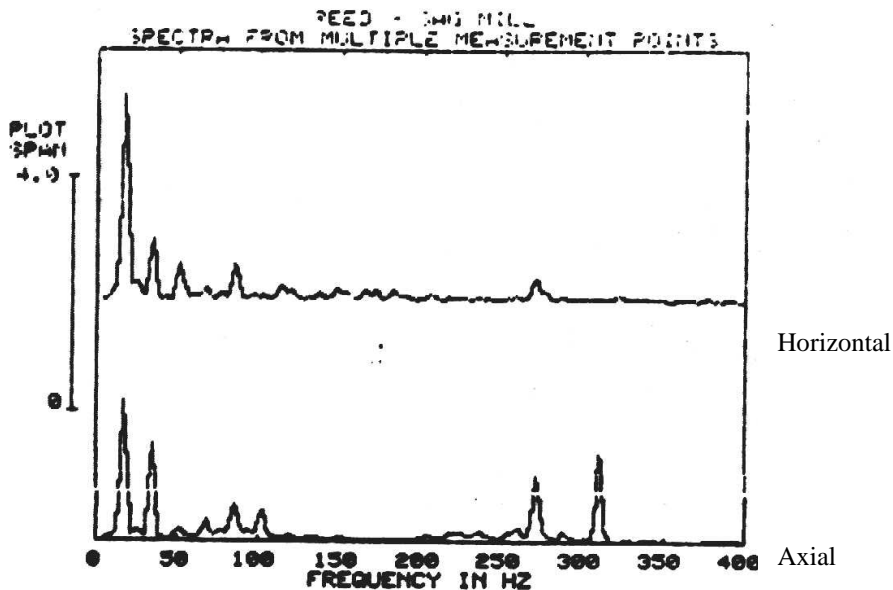
The diagram below shows a typical triaxial spectrum from an unbalanced machine. Note that the vertical and horizontal amplitudes are similar but the axial amplitude is low.



5.2 Misalignment.

Some industry specialists will argue that this is the most common fault to be found. The issue of what are acceptable tolerances for misalignment is considered in another course.

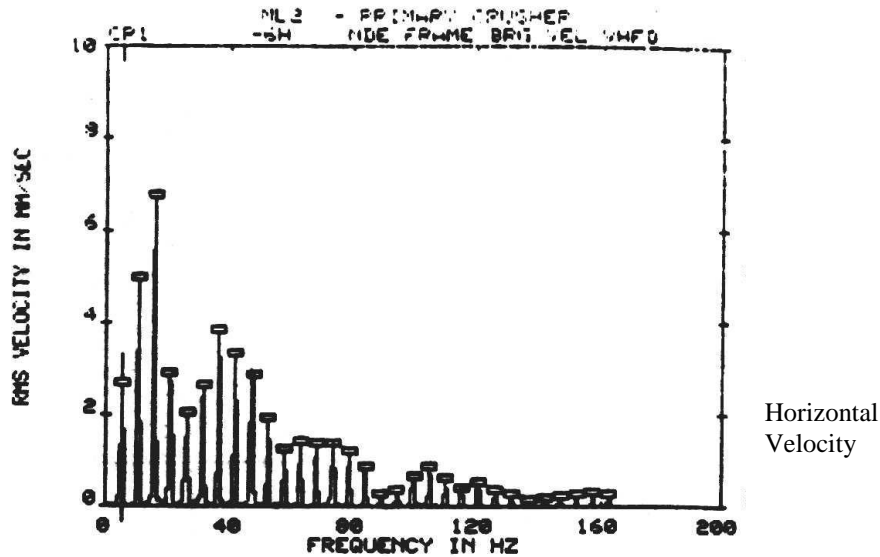
The diagram below shows two traces taken from the bearing of a machine next to the coupling. One is horizontal, one is axial. Note the different harmonic components in each direction and relate this to the information given in the TOP 4 table.



5.3 Looseness.

Looseness can occur in any part of a machine and not necessarily be just associated with the shaft or bearings. For example a loose hold-down bolt could be producing a significant looseness signature at the bearing.

The diagram below shows a typical example of a looseness spectrum.



5.4 Rolling-element Bearing Failure.

This is the fourth of the common machine failure modes listed on the ‘Top Four’ table but we will consider the detection of bearing faults in detail in the next session.

6. Specific Machine Types.

On the following pages are charts and diagnostic information that will assist in the diagnosis of faults on Electric Motors, Pumps, Fans, Gearboxes, Turbo-Machinery and Reciprocating Machinery.

7. Advanced Analysis Techniques.

The following notes will describe some of the more advanced vibration analysis techniques that are available.

7.1 Operating Deflection Shape (ODS).

The purpose of ODS is to display the motion of a machine or structure in slow motion and greatly amplified displacements for the purposes of analysis or corrective action.

Usually the motion is analysed at one selected frequency so that it is necessary to have phase-related data for each measured point. The data can be acquired in a number of ways including the following:

- A single-channel analyser with phase capability. Measurements are taken at the required locations (sufficient to make a recognisable line diagram of the machine) and the same phase reference used. This data is then entered manually into an ODS software package and the result displayed. The output is limited to the frequency chosen for measurements.
- A dual-channel analyser with transfer-function capability. Measurements are made with one transducer in a fixed location that is representative of the machine vibration. The data is stored in the analyser and then downloaded into the ODS package. This method has the advantage of being able to analyse any frequency within the bandwidth selected for measurements. The phase relationships at selected frequencies are automatically compared in the analysis process.
- A multi-channel data acquisition system. Sometimes data is transient and can only be captured in one event. For such analysis it is necessary to capture all data simultaneously and this can be done digitally or using tape recording. As above, the data is transferred to the software and the required frequencies for analysis selected.

Some typical examples of ODS images are shown on the next page. Usually data is supplied on a disc for clients to run in an animated format under Windows.

7.2 Multi-channel Data Acquisition.

As noted above, some events are transient and must be captured in one sequence for later analysis. This is often required for purposes other than ODS.

For example, the periodic condition monitoring of a steam turbo-generator set may include a run-down test and it may well take an hour from shut-off of steam to have the machine at slow-roll speed. This data is usually captured by either connecting into the proximity probe system installed on the machine or placing seismic transducers at each bearing. The output may be tape recorded or digitally stored. A once-per-turn tacho pulse is also recorded to provide phase referencing for analysis purposes.

Digital data capture and storage is becoming more popular because it is so compact and reliable. However, frequency limitations are invariably imposed because of the compromise between sampling rate and available storage. For example, if data is captured at 100 samples per second and is to be post-processed by FFT, a low-pass filter will need to be installed at 50Hz to eliminate aliasing.

7.3 Structural Measurements.

A vibration amplitude and mode study for a large structure presents some technical challenges but useful results can be achieved with care.

Displacements at frequencies as low as 0.1Hz may need to be measured because most structural engineers are only interested in displacement. If accelerometers are to be used – and there are few options – then large high sensitivity devices will be selected, typically 500mV/g. This allows sufficient output at low frequencies to be integrated twice to displacement to achieve the required measurements with a minimum of noise error. ODS is a valuable technique for structural analysis and therefore one of the methods described above will need to be used to present the data in the required format.

8. Special Cases

8.1 Gearboxes

Gearboxes provide the means of transmitting torque and changing direction or speed.

They are complex machines and do not make for easy condition monitoring or diagnostic analysis.

Gear mesh frequency (GMF) is the most commonly observed vibration and the degree to which the mesh is worn, misaligned or damaged will determine the strength of harmonics in the spectrum.

It is often very useful to trend the growth of harmonic amplitudes, in particular the second mesh (GMF x 2), which is an indicator of incorrect backlash and profile wear.

Eccentricity of gears or bent shafts are identified by studying the side bands around GMF and its harmonics (refer to the notes on FFT analysis in Chapter 3). This technique can allow positive identification of the particular shaft and gearset causing the problem in a complex gearbox.

It is very useful to study the time domain in gear analysis, especially if a cracked tooth is suspected. Time synchronous averaging using a tacho pulse from the shaft carrying the faulty gear greatly enhances the fault amplitude above the general noise.

Time domain analysis is essential on slow speed gears such as mill pinions, slew drives and the like where the impact of the faulty tooth may occur so infrequently as to not be captured in the time samples for normal frequency analysis.

Mutual Mesh Frequency (MMF) is worth calculating where two faulty teeth in meshing gears cause a modulation of the fault impact amplitude. The formula for MMF is as follows.

$$MMF = \frac{\text{Speed of Gear 1}}{\text{LCM of Mating Gear 2 Teeth}}$$

The monitoring of bearings in gearboxes calls for special techniques of which HFRD is the simplest. It is essential to be able to positively identify fault frequencies in the midst of all the other high frequency vibration activity.

Countless papers have been written by academics and researchers on other techniques such as the Hilbert Transform and Cepstrum to name just two. We have not found it necessary to use these techniques providing HFRD is carefully applied.

As a final comment, gearbox fault analysis is usually well supported by wear debris analysis and again, providing the sampling is well done, can provide good confirmation of a vibration diagnosed fault.

8.2 Diagnostic Notes For Centrifugal Fans

The following notes relate specifically to centrifugal fans with backward sloping blades but have general application to any centrifugal or axial fans,

UNBALANCE:)
MISALIGNMENT:) **Refer to TOP 4 DIAGNOSTIC CHART**
LOOSENESS:)
FAULTY R/E BEARINGS:)

SURGE: Typically produces vibration at 0.1 to 0.5 x Shaft Speed. Amplitudes are unstable and parallel changes in fan noise may be heard. Check fan operating conditions against makers curves.

RESONANCE:

(a) Structure. Fan structures are often lacking in required stiffness. Resonance will produce very directional vibrations. Check phase between horizontal and vertical on bearings. Axial resonance on bearing supports is common. That is, the 'rocking' stiffness of the bearing assemblies is insufficient. Check with bump test.

(b) Shaft. A shaft natural frequency near operating frequency will cause whirl and very high shaft x 1 vibration. Variable speed fans are very prone to this problem- Fine trim balancing in-situ can reduce vibration to acceptable levels.

***COCKED BEARING:** Inner race alignment to shaft is critical to a smooth running fan because of the high axial thrust. The combination of axial thrust and inner-race wobble will severely reduce bearing life.

RUB: Rubs will usually be heard. Vibration data will most-likely show rotational harmonics (x 1, x2, x3 etc) and possibly sub-orders and half-orders (x 0.5, x 1.5, x 2.5 etc).

UNEVEN AERODYNAMIC LOADING. Eccentric or wobbly runner can generate internal pressure waves, usually at Shaft x 1. Minor run-out errors are not usually significant.

BLADE-PASSING VIBRATION. Vibration generated at Shaft x n fan blades. Not usually a problem unless there are either structural or duct resonances present. Duct resonance can be determined by measuring the straight lengths of the duct **and** relating these dimensions to the celerity of sound and quarter, half **and** full wave reflection lengths.

8.5 Diagnostic Notes For Reciprocating Machines.

Important: Vibration measurement of reciprocating machines can be a useful indicator of condition but it is important to remember that there are many failure modes that are not readily indicated by vibration. For all rotating machines periodic oil sampling and analysis is essential and thermography can be useful.

Reciprocating Engines: Vibration measurements can be useful for monitoring balance and alignment conditions and possibly some aspects of looseness. However, vibration alone is of limited value for true condition monitoring. It is much better to take a 'Performance Monitoring' approach to engines and measure parameters that will give **an** indication of things such as loss of compression, faulty injectors, changes in temperature, etc.

Reciprocating Compressors: Again, vibration is of limited value but worth including as one of several measured parameters. Most common failure modes are valve leakage, cross-head wear, loss of compression due to ring wear, seal leaks and the like. Few of these will be seen in vibration frequency data but if measurements are carefully done they can be seen in the time domain.

8.6 Turbo-Machinery

High-speed turbo machines are generally very reliable and maintenance is usually only required for seal wear, blade fouling and the like.

Most faults are generally bearing-related and require design changes. Oil whirl, oil whip, bearing looseness and the like can usually be designed out and never seen again.

Bearing cap data can be very useful for condition monitoring but is less useful for fault analysis. Obviously direct shaft position measurement via proximity probes and data acquisition systems provides for much more powerful diagnostics.

The diagnostic routines for machine fitted with proximity systems are will developed and readily available for reference.

Most data collector/machinery analysers are able to do single channel Bode and Nyquist plots which are different ways of displaying the amplitude/phase vectors as the machine runs up or down.

Orbit analysis (two channels plus phase) is most commonly applied to steady state conditions where the shape of the orbit gives vital information about alignment, balance, rubs, etc.

Shaft couplings are often gear-type and oil lubricated. They are a common source of problem due to poor capacity for axial float. Tooth wear or binding will often be evident on careful examination. Replacement is the only fix.

9. Understanding Time Wave Form Patterns

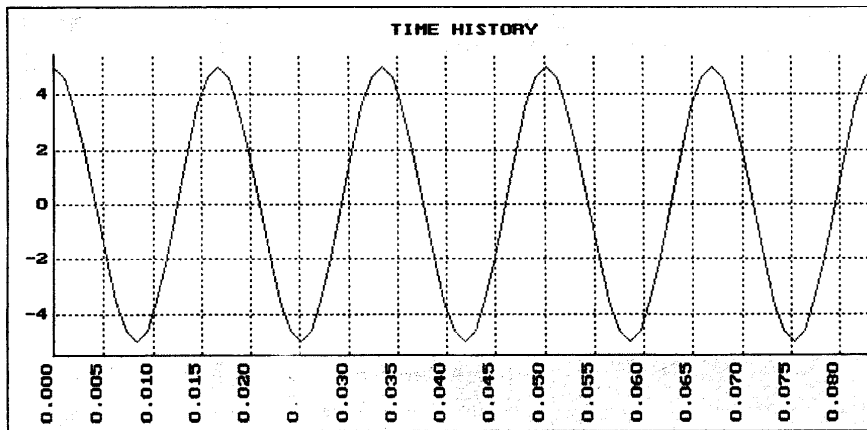
Basic Time Waveform Patterns

The basic elements of the TWF are:

- Sine Waves and their summation
- Symmetry of Amplitude scale and Truncation
- Symmetry in the Time scale
- Phase relationships in the Time scale
- Transient or Impactive signals

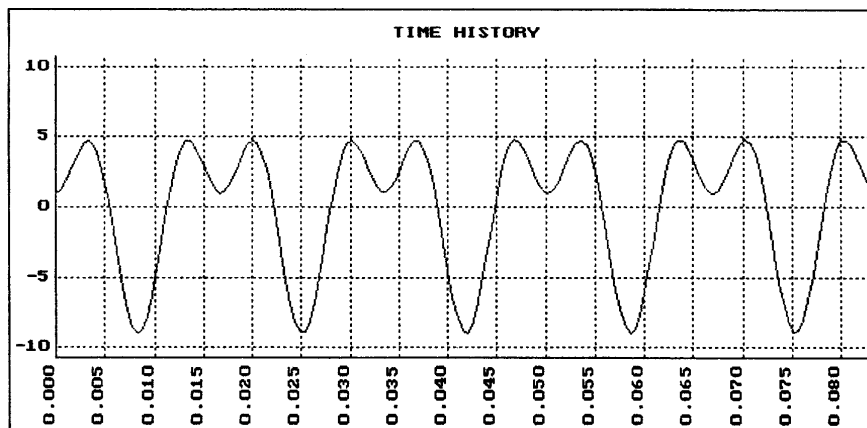
9.1 Sine Waves

Wave Form Summation

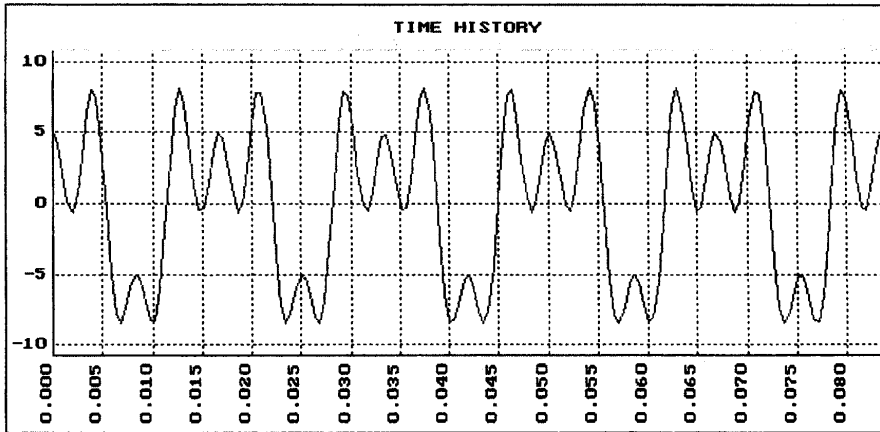


The above wave form is a pure sine wave of period 0.0168 sec and amplitude 5 units; it could be from a machine rotating at 3585rpm

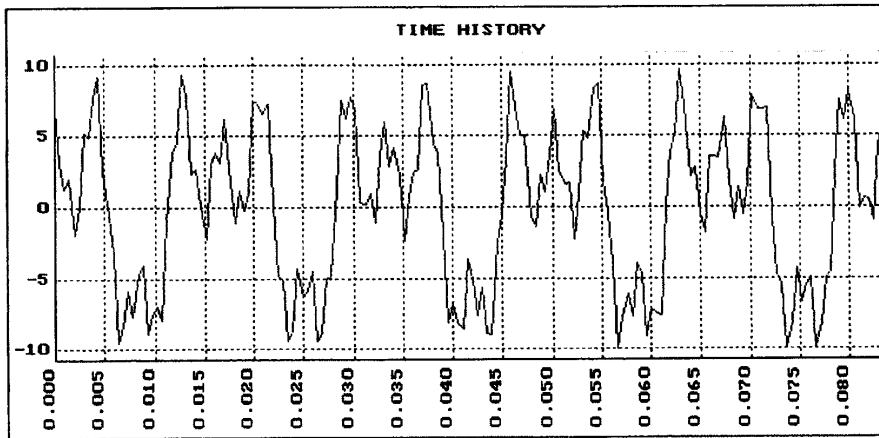
The wave form below is the result of adding a second harmonic, 2 X rpm; it subtracts from the fundamental frequency in the positive half and adds to the fundamental in the negative half. Note the change of vertical symmetry.



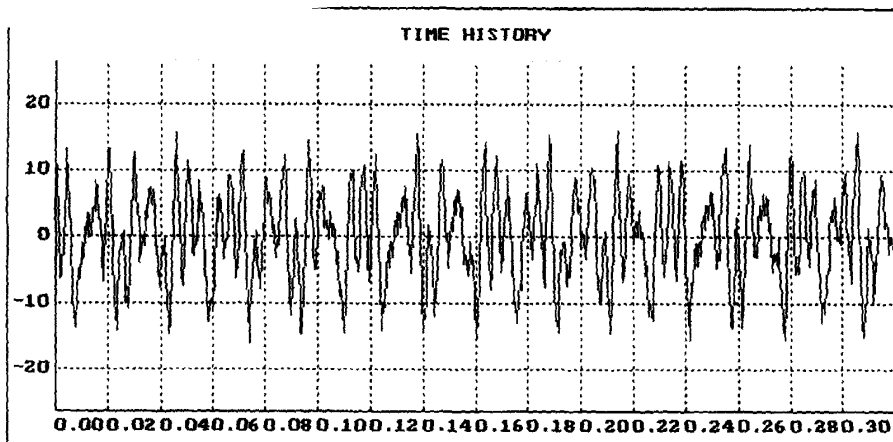
By adding the 4th harmonic (4 X rpm) the following wave form presents;



And some high frequency component representing bearing frequency increases the complexity of the wave form;

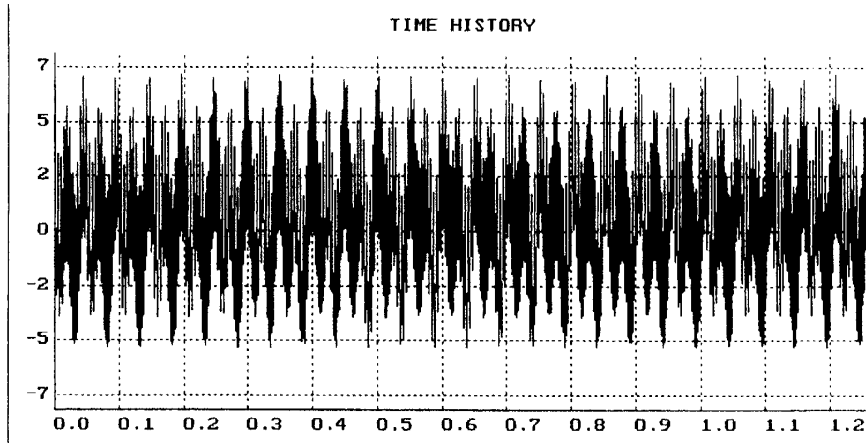


Finally, by adding some very high frequency to represent gear meshing, a more typical wave form is seen;

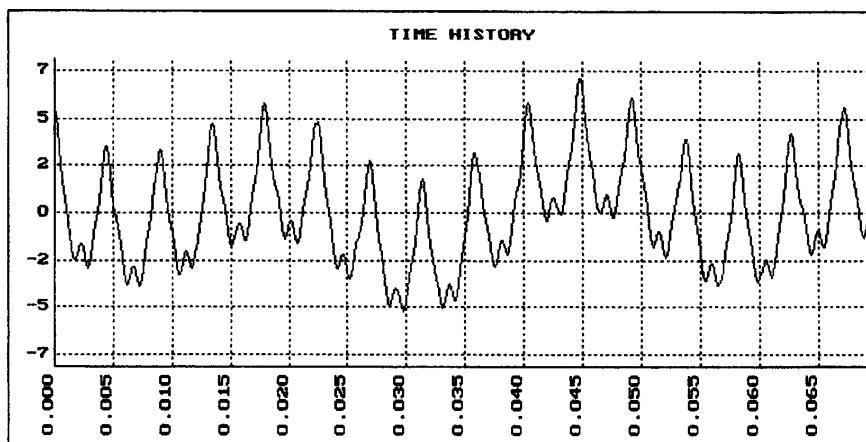


9.2 Importance of Resolution and Time Scale

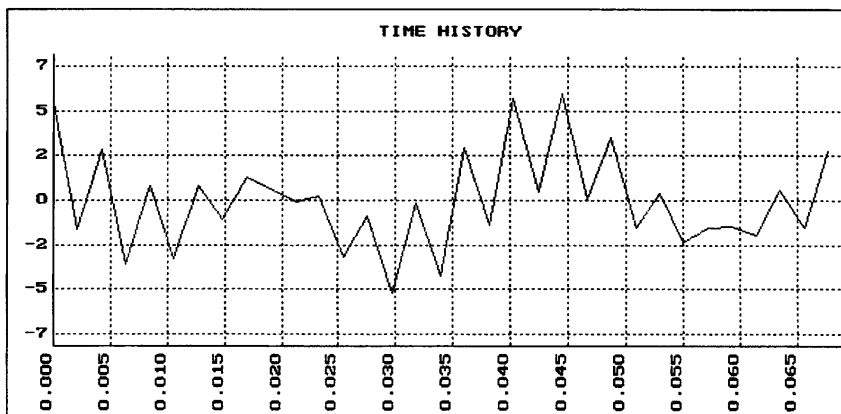
Since most measurements will be made with accelerometers and will be sensitive to the high frequency content it is unlikely that many waveforms will be seen in the real world that look sinusoidal in character. The wave form below is typical, note the time span, 1.2 sec. The wave form does not look particularly sinusoidal and does not appear very informative.



By zooming in on a smaller time span within the same block of data the following is revealed and is more informative. It actually comprises 5 sine waves added together.

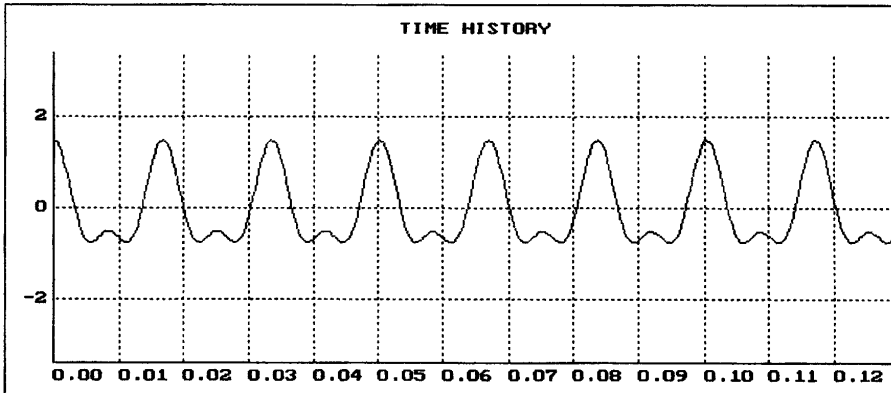


The above pattern is only revealed if sufficient lines of resolution are used. The wave form below is the same data with insufficient resolution.

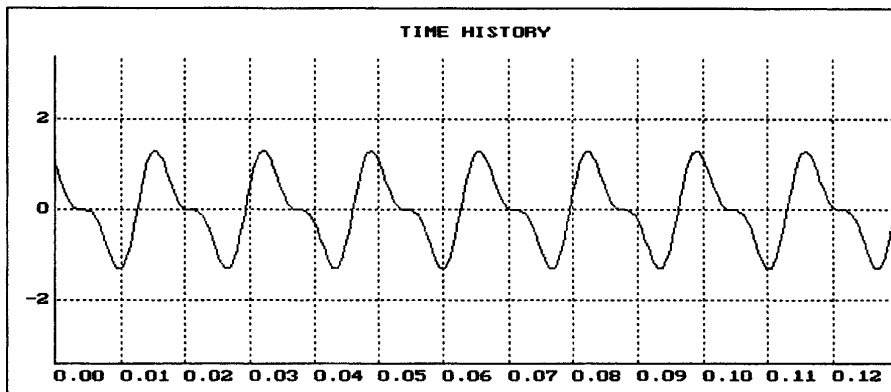


9.3 The Effects of Phase Change on Sine Wave Summation

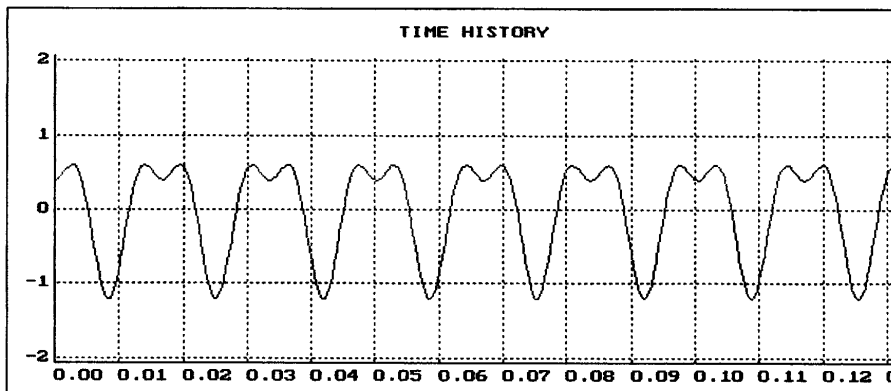
The wave form below comprises a 1 X rpm and a 2 X rpm with a 0° phase difference



The 2 X rpm is now shifted 90° in relation to the 1 X rpm;



And changing the phase relationship to 180° changes the pattern yet again;



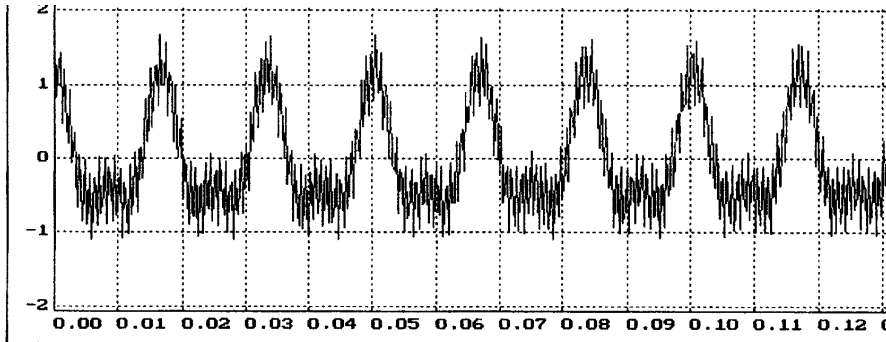
9.4 Symmetry of Amplitude Scale and Truncation

Note in the trace above the change in amplitude symmetry. Truncation of the signal can come from such as rubs but this is more commonly seen from proximity probes. Truncation can also be seen as

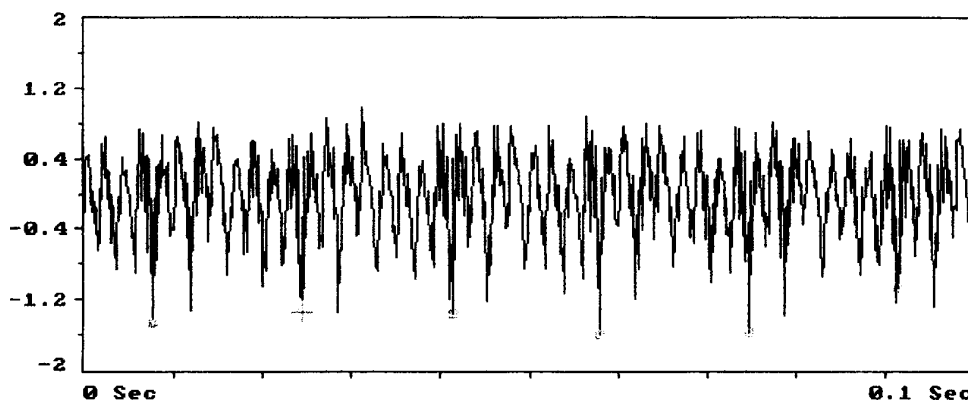
a consequence of overloading the instrument input and causing “clipping”; this usually seen as a quite clean truncation.

The pattern seen above is more typically seen as a consequence of misalignment, the more common cause.

The trace below is similar, but inverted, and carries some high frequency components, making it more typical of a bearing case accelerometer signal for misalignment.

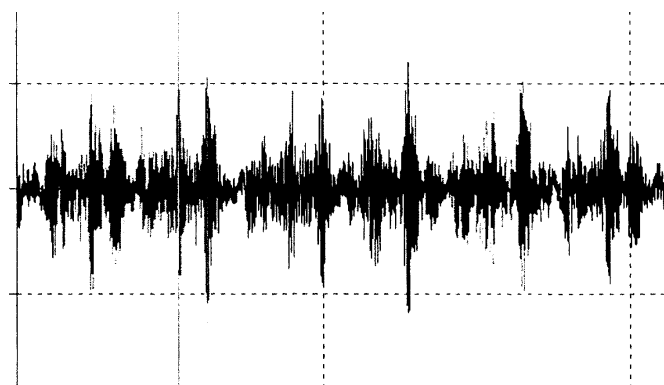


The following trace is from an actual machine misalignment in the real world; note the amplitude symmetry and the machine speed markers.



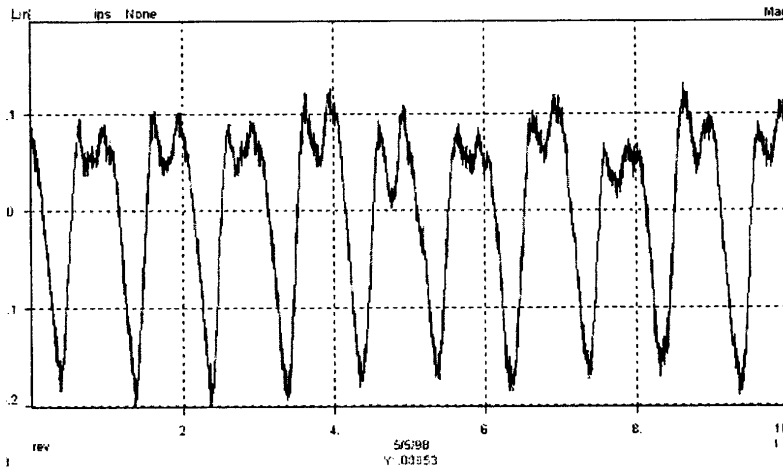
9.5 Symmetry in the Time Scale

This trace is from a bearing defect, showing a series of repetitive impacts with an amplitude of around 10g but varying. The spacing is constant and at a rate revealing it as from the outer race. An FFT of this signal would reveal an amplitude possibly less than 1g.

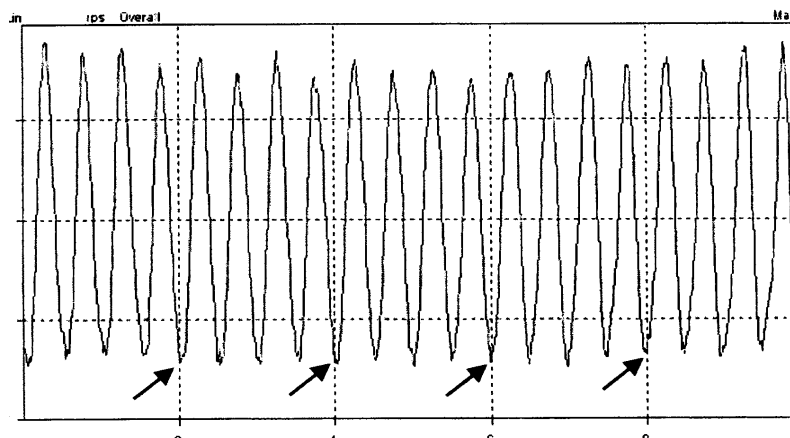


9.6 Synchronous and Non-synchronous Wave Forms.

Instruments which offer a 1X rpm marker, or grid, offer a distinct advantage when determining whether a vibration source is synchronous or not. In the wave form below the horizontal axis is in rpm and the peaks are seen to occur at the same point within each cycle. This indicates the vibration is synchronous with the shaft rpm.



The next wave form shows a 2 pole motor with a dominant vibration at 2 X rpm. Close examination reveals subtle changes in the position of the time wave form relative to the rpm markers. The source of this vibration is in fact electrical hum; there are very low levels of 1X and 2 X rpm, the mechanical vibration. The cause of this problem is motor soft foot.



The wave form above shows a lower amplitude but higher frequency signal “riding” on a higher amplitude but lower frequency signal – in fact, a 1X rpm with a twice Line Frequency. As each cycle progresses the higher frequency component shifts slightly causing gradual changes to the pattern.

