

Fundamentals of Vibration Measurement and Analysis Explained

Thanks to Peter Brown for this article.

1. Introduction:

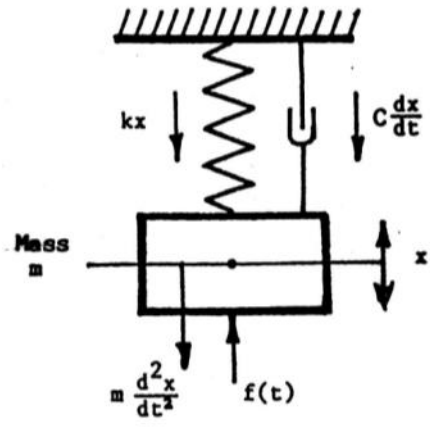
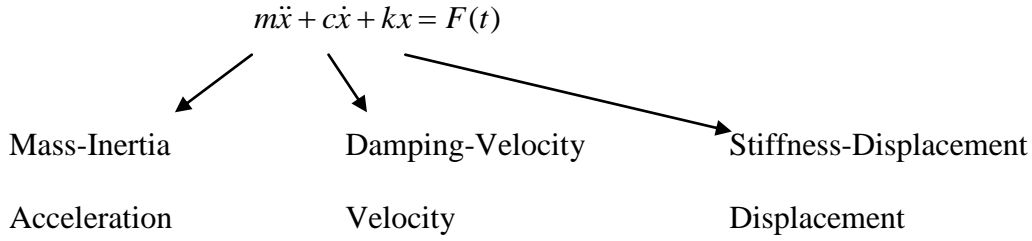
The advent of the microprocessor has enormously advanced the process of vibration data acquisition and analysis in recent years. Measurement tasks that took hours only two decades ago can now be completed in minutes and better decisions made because of better data presentation.

However, the basic processes of measurement and analysis have remained essentially unchanged, just like the machines from which the vibration is measured. The results of the measurement and data analysis need to be compared with known standards or guidelines and decisions made as to whether the machine is acceptable for service or maintenance should be planned. Increasingly these processes are being handled electronically but we are still a long way from replacing the fundamental knowledge and experience of the vibration analyst.

In this article we will review the basic principles of vibration measurement and analysis in order to lay the foundation for capable fault diagnosis to be considered later.

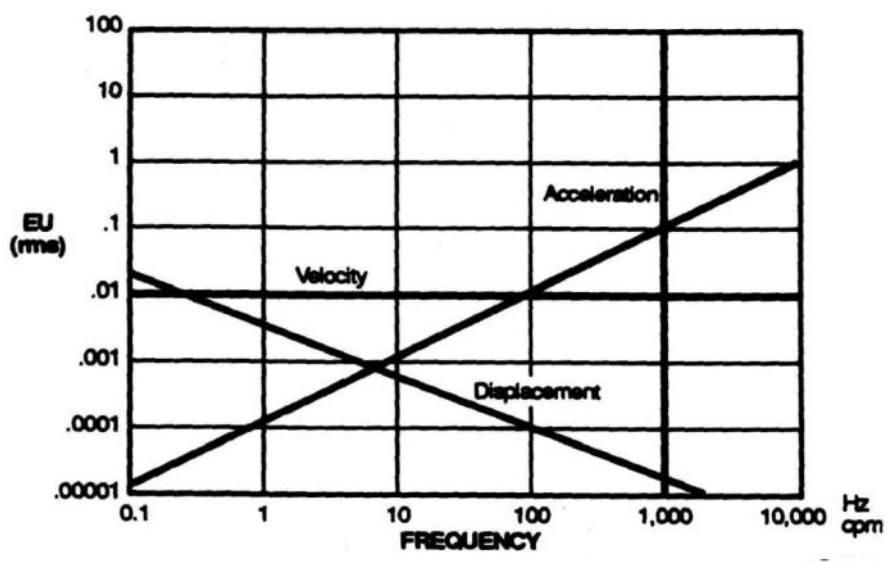
2. Fundamentals of Vibration:

A simple machine may be represented as in the diagram below having mass, stiffness and damping. If we take this simple, single-degree-of-freedom model and excite it with a sinusoidal force $F(t)$ then the distribution of forces generated by the resulting dynamic displacement x may be determined by the following equation:



The above diagram can be easily related to the typical independent front suspension of a vehicle. Consider how the displacement x might change with changes in spring stiffness, shock-absorber damping rate and mass of the wheel and suspension system.

The graph below shows the typical relationship between displacement, velocity and acceleration for a sinusoidal (single frequency) vibration.



Note that **displacement** amplitudes decrease to very small amplitudes above about 100 Hz. For that reason, seismic displacement measurements are rarely used for machine condition monitoring. They are more useful for monitoring structural vibration.

Velocity tends to have a reasonably uniform response over a wide range of machine frequencies and for that reason it is the universal measure of evaluation of machine integrity in relation to balance, tightness, alignment and the like.

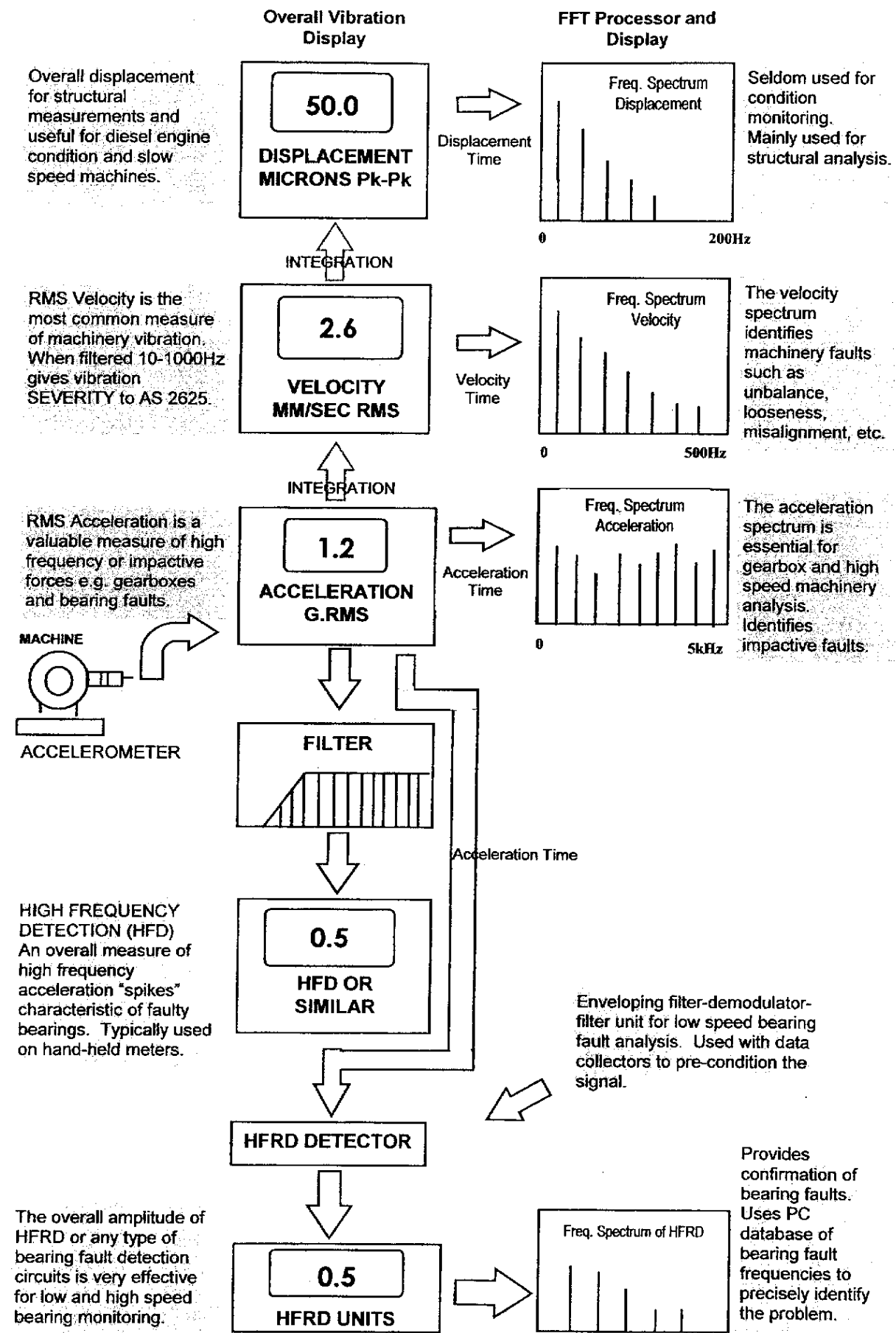
Acceleration increases relative amplitude with increasing frequency and is therefore the logical choice for monitoring those components that generate high frequency vibration such as bearings, gearing and blade-passing (as in screw compressors). As was mentioned in Session 3, the high frequency sensitivity of accelerometers is used to provide measures of rolling-element bearing condition and all these measures are in acceleration units.

3. Measuring Vibration.

The diagram on the opposite page shows diagrammatically the general arrangement for vibration measurements using an accelerometer..

3.1 Overall Vibration Values.

The output of the accelerometer is an AC waveform that exactly reproduces the vibration acceleration. This waveform might be useful for an experienced vibration analyst to use for fault diagnosis but for most situations it is not appropriate or even necessary. It is much more useful to measure the total power of the waveform and express it as a DC value. We call this the **OVERALL VALUE OF THE VIBRATION**.



DIAGRAMMATIC REPRESENTATION OF BASIC VIBRATION MEASUREMENT AND ANALYSIS PROCESSES

To get this overall value we need to do an RMS (Root Mean Square) calculation on the AC signal. Older style vibration meters were able to calculate RMS values using simple analogue circuits. Modern machinery analysers do the same thing via a more complex process of signal digitisation.

Overall vibration measurements, usually expressed as the RMS value (except for displacement where peak-peak values apply), form the basis of condition monitoring measurements and trending, but have limited value for analysis.

The diagram on the previous page showed a typical value for vibration of 1.2 g rms. Acceleration can be expressed in the true values of m/sec² rms or gravitational units ‘g’ where 1g is equal to 9.81 m/sec² approximately.

If we want to know the overall velocity of the vibration the instrument can calculate that for us and read it out in mm/sec rms. This is a single integration of the acceleration signal. The diagram shows a typical reading of 2.6 mm/sec rms.

Occasionally it might be useful to know the displacement amplitude. This is a double integration of the acceleration signal and may be a little ‘noisy’ as a result. Displacement units are always measured in microns peak-to-peak. The chart shows a value of 50 microns peak-to-peak.

It is very common to **TREND** the overall values of acceleration or velocity to look for increases or instability in overall values. This is the most basic form of vibration monitoring.

3.2 Calculation of Frequency Spectra.

The right column of the Basic Processes diagram shows that the time waveform can be converted to a frequency spectrum in order to show the analyst where the vibration energy is coming from. Frequency analysis is the essence of vibration analysis and enables the satisfactory resolution of most machine problems.

It is important to understand the relationship between the TIME WAVEFORM and the FREQUENCY SPECTRUM.

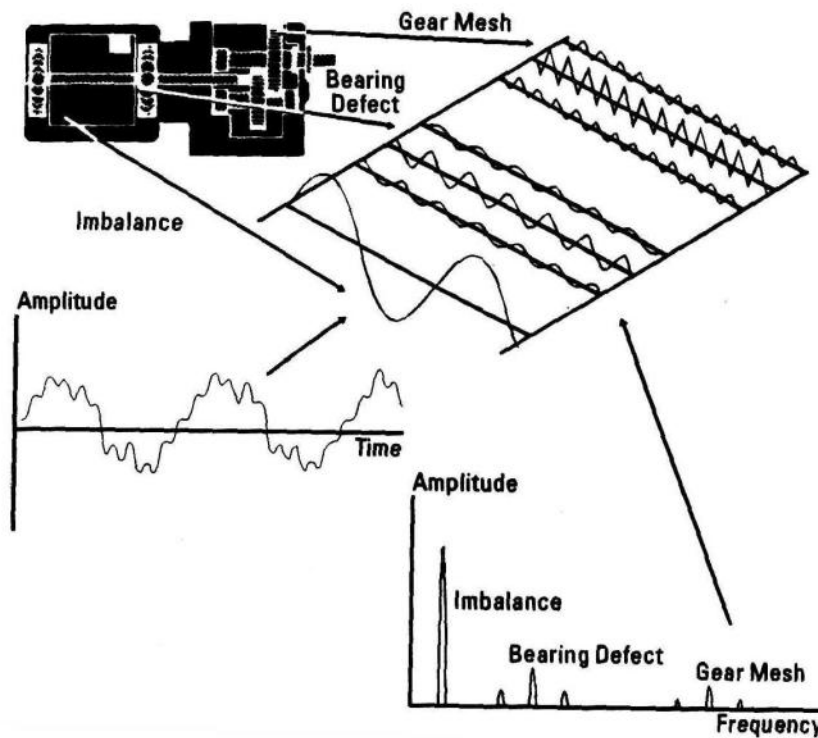
On the following page is a sketch showing a geared motor producing three different forms of vibration. There is vibration from motor unbalance, vibration from gearing and bearings.

An accelerometer placed at any one point on the geared motor will measure a COMPLEX VIBRATION waveform as shown in the time drawing on the left side. This is the summation of all the vibration present at that location.

Spectrum analysis enables us to untangle this complex waveform and make a representation of its original components on a diagram showing frequency on the X-axis and amplitude vertically. This is known as a VIBRATION SPECTRUM and is extremely valuable for fault diagnosis.

The way the diagram has been drawn shows that the process is like looking at the vibration from two directions at right angles to each other. From the time-direction we see the

algebraic summation of all the individual frequencies present. From the spectrum-direction we see the location and amplitude of each frequency component.



The conversion from Time to Frequency is achieved by use of an algorithm called the Fast Fourier Transform (known as an FFT). The underlying principle was first postulated by Baron Fourier in the 19th century who stated that any periodic curve, no matter how complex, can be considered as comprising a number of pure sinusoidal curves with harmonically related frequencies.

3.3 OPTIONAL MATERIAL further explaining the proper use of spectrum analysis.

There are some important technical considerations that need to be understood by the engineer using FFT systems.

- a) **Anti-Aliasing Filters.** The first part of the process is to digitise the data. The fundamental rule of conversion from analogue to digital data is to ensure that a low-pass filter is installed so that *the signal to be digitised contains no frequencies above half the sampling frequency*. This is known as the Nyquist Criteria and ideally the sampling frequency should be no lower than 2.56 times the highest frequency in the data to be processed.

This is very important to be considered also when using data loggers (effectively A-D converters) recording dynamic data. If anti-aliasing filters are not applied there will be spurious low frequency components in the spectra that can be quite misleading.

- b) **Window Function.** The FFT process requires that discrete blocks of the digitised time data are taken and entered into the processor. To avoid errors in the process it is

necessary that the values at the beginning and the end of the sample are compressed to zero. This does not significantly affect the accuracy of the result.

The compression process is called a ‘window’ and there are a number of different windows that can be used to achieve certain characteristics in the spectra. We will look at only three.

- (i) **Hanning:** Perhaps the most common. Gives very good frequency resolution but relatively poor amplitude accuracy. In most cases good frequency resolution is more useful and the less accurate amplitude errors can be accepted.
 - (ii) **Flat-top.** Poor frequency resolution but highly accurate amplitude values.
 - (iii) **Uniform.** This is ‘no window’ and may be more effective where windowing is not necessarily needed such as for transient data such as impact tests.
- c) **Averaging.** It is common in the process of data sampling to take a number of samples and average them to ensure a more representative result. There are three common averaging options available.
- (i) **RMS:** A normal summation average of consecutive data samples. Random data such as ‘noise’ tends to be averaged out. Typically 8 to 16 samples may be taken for an RMS average and the result displayed and stored.
 - (ii) **Exponential:** This is useful for data that may be slowly varying in frequency or amplitude. Typically 99% of the data is in the last five samples. Only used for special purpose analysis.
 - (iii) **Peak Hold:** this is not really an average at all but a means of storing the highest values over a sampling period. The profile displayed is therefore the maximum amplitudes in each frequency box. This can be useful in run-up/run-down analysis to identify machine or structural resonances.
- d) **Trigger.** The FFT process requires a trigger to initiate the data acquisition. Usually the default setting is an automatic internal trigger. However, for certain purposes it can be operated manually. For example, an external optical tachometer looking at a piece of tape on the machine shaft may be used to provide phase-referenced data and the tachometer pulse will drive the trigger. Also the process of Time Synchronous Averaging (used to enhance the time-domain data by cancelling noise) requires to be driven by the tachometer pulse.
- e) **Bandwidth and Resolution.** The frequency axis of a spectrum display has a certain pre-selected number of digital samples known as ‘boxes’. Normally the frequency resolution for condition monitoring would be set at 400 boxes. However, most machinery analysers have the capability to provide 3200 box resolution or higher. While this would seem to be useful it in fact greatly increases the signal processing matrix and therefore slows the data processing speed enormously. Frequency resolution is therefore a direct trade-off against processing time and digital storage capacity.

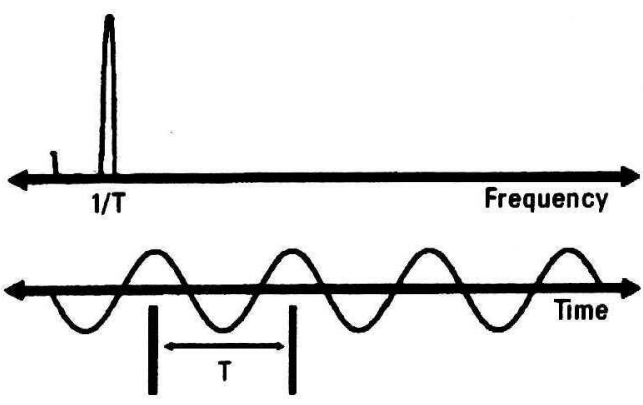
Similarly, bandwidth is a compromise. While it might be useful to have all velocity spectra covering (say) 2000Hz with a 400 box spectra the resolution will be around 5Hz per box. This is quite unacceptable for most purposes. Therefore, a more typical choice for velocity is 200 to 500Hz with a 400 or 800 box resolution to give acceptable definition for condition monitoring purposes.

4. Classic Time and Frequency Relationships.

We have been considering the process of converting time waveforms to frequency spectra. Before moving on it will be useful to consider some classic time-frequency relationships.

4.1 Dominant Single Frequency.

Where the vibration source is largely sinusoidal (single frequency) the time and frequency domains will appear as below.



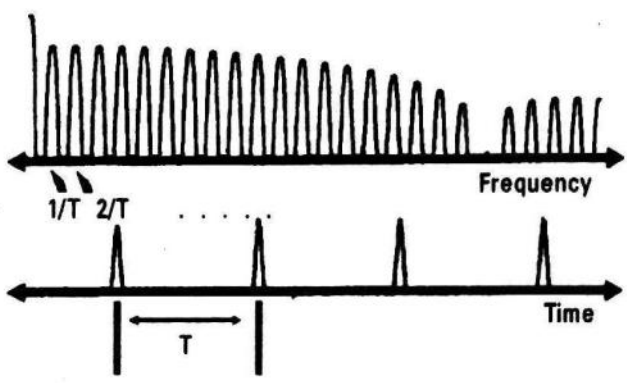
This sort of signal is characteristic of Unbalance or Bent Shaft where the force causing the vibration is simply related to the rotation of the shaft.

4.2 Impactive.

This might be considered as the opposite to a single frequency signal because, to some degree, all frequencies are present.

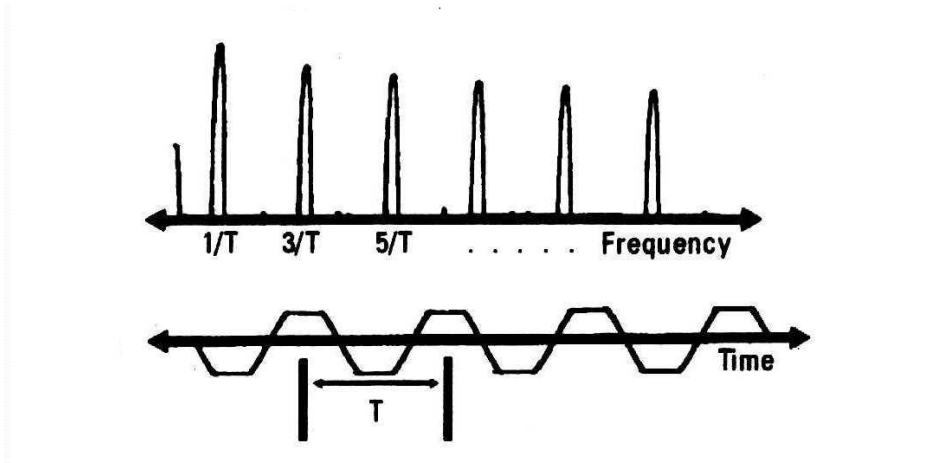
There may be several variations in character. A 'perfect' impact will produce energy right across the spectrum as in the first diagram below. This is the principle of the 'bump test' used to excite and identify structural resonances.

In machinery, impacts are usually repetitive and periodic such as arising from looseness or a broken gear tooth. In this case the spectrum will be rich in harmonics spaced at the repetition frequency as seen in the second diagram shown below.



4.3 Square Wave.

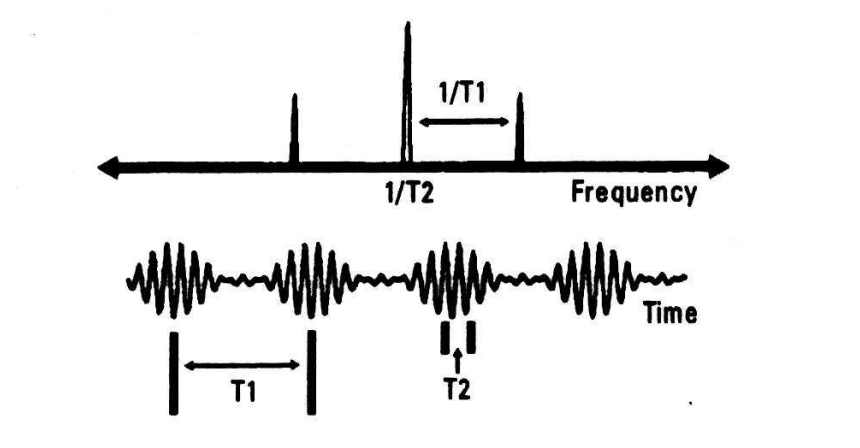
Somewhere between the sine wave and the repetitive impact lies the Square Wave. This is an idealistic concept that rarely is approached in the real world. The characteristic pattern is odd orders in the spectrum, diminishing in amplitude.



This pattern may be seen in some forms of misalignment and looseness and it will usually be very obvious in the time domain.

4.4 Modulation.

It is often found that a single frequency (known as the carrier frequency) will be modulated in amplitude at a lower frequency. An example might be a bent shaft in a gearbox causing the mesh frequency to be amplitude modulated at the rotation frequency of the shaft.



The carrier frequency is seen as a sine wave (single spectrum component) but it has lesser components on both sides known as Sidebands. The spacing between the carrier and the sidebands, and between each of the sidebands, is the frequency corresponding to the time period of the modulation.

This is a very powerful diagnostic tool for gear mesh analysis.

5. Resonance in Dynamic Systems.

5.1 The Theory of Resonance.

The vibration levels arising from forced vibrations on machines that are well designed, manufactured, balanced, assembled and aligned are usually quite low and a long, safe service life may be expected from a mechanical perspective.

However, certain situations arise where the properties of mass, stiffness and damping can produce very undesirable results.

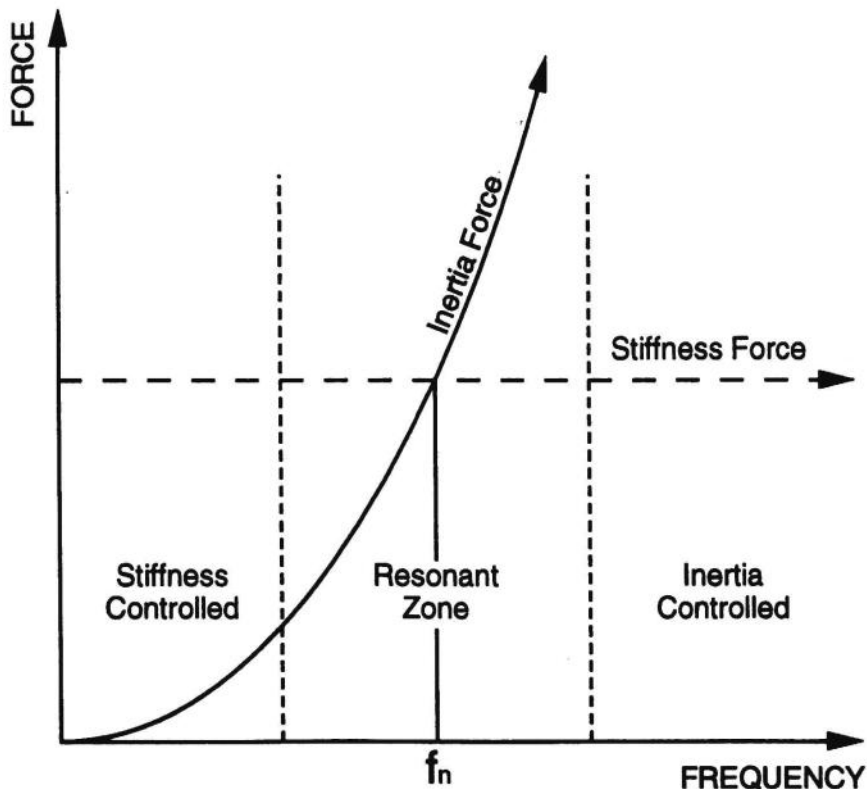
If we take the equation we looked at before and remove any external excitation we have this situation. $m\ddot{x} + c\dot{x} + kx = 0$

This equation defines the free motion of the mass as a self-contained system and the solution to the equation for time t will define the natural frequency of the system.

A useful way to understand the system is to provide a graphical solution. If we say that the damping coefficient c is so small as to be ignored then we can re-state the equation as:

$$m\ddot{x} = -kx$$

If we make a graph using the values from the previous exercise we have the following situation. (Note that negative and positive values have been superimposed for simplicity).



It is very obvious that at the point where the values are equal and opposite there is effectively no restraint on the system! Any force applied at that frequency will cause motion of the system at infinite amplitude – and it would self-destruct! This is known as the **Natural Frequency** of the system and the coincidence of an excitation force with the natural frequency is known as **Resonance**.

Of course we have assumed no damping, and this is where damping comes to the rescue. As velocity levels increase even quite low damping coefficients have a significant energy absorption effect and systems do not normally self-destruct.

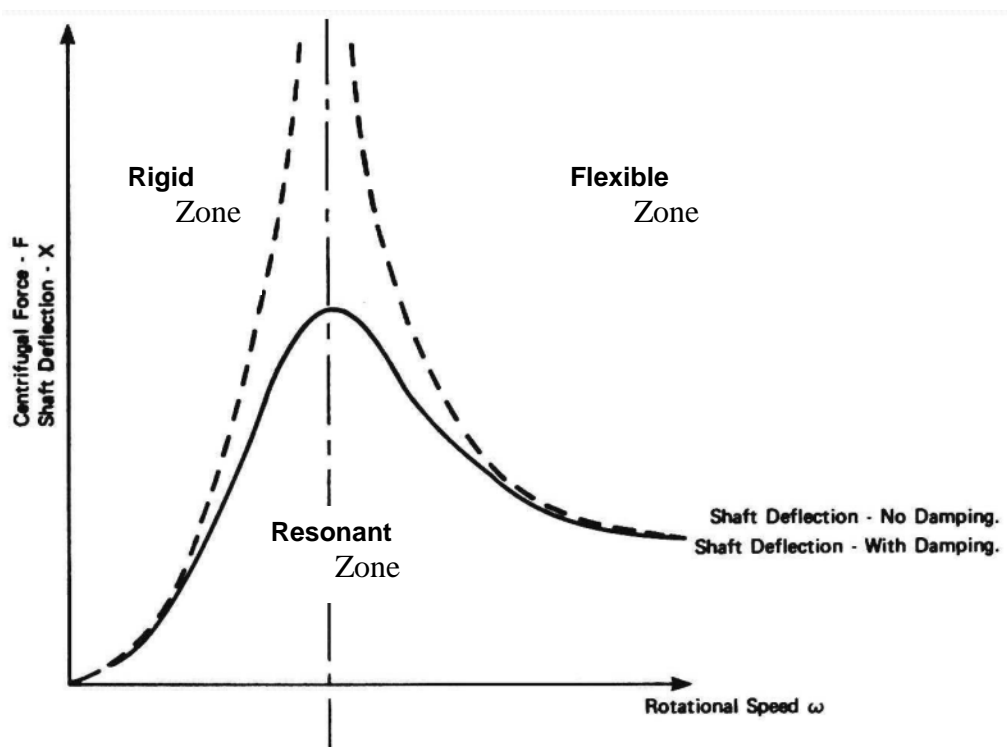
The vibration that occurs at the natural frequency is called **Free Vibration** and it has its valuable uses of course in things like bells, musical instruments and – at much higher frequencies - radio transmissions.

All engineering students will have seen pictures of the Tacoma Bridge disaster in Washington State many years ago. This was the definitive case of excitation corresponding with poorly damped natural frequencies producing destructive amplitudes of vibration.

All engineers will also have come across problems of vibration amplification due to resonance in the machines or structures under their care. It is very common – too common – and is often a reflection of poor design considerations. Vibration consultants earn a lot of money fixing up machine and structural problems arising from resonance!

5.2 Managing Resonance.

The classic resonant system curve is shown on the following diagram. As a forcing frequency approaches the system natural frequency the vibration amplitude increases. This zone is generally known as the **Rigid Zone** and describes the behaviour of most machines bolted firmly to large concrete foundations. The forces transmitted to the foundation correspond roughly to the size of the forces generated in the machine – less the forces dissipated in the machine itself.



If we go to the other side of the diagram we see an area known as the **Flexible Zone**. In this region the forces generated in a typical machine are not transmitted to the foundation but are largely dissipated in the machine itself. This is the principle of Vibration Isolation that we will look at later in the Course.

The region of concern is the **Resonant Zone** where vibration is amplified and problems of fatigue or dynamic overload are almost certain to occur.

Note that the amplitude of any vibration frequency component is proportional to where it lies on the response curve for the system. At the point of maximum resonance any change in frequency up or down will reduce the amplitude of vibration (that is in fact the definition of resonance). As the excitation moves away from the natural frequency the overall vibration diminishes.

This curve also shows that it is not possible to excite a natural frequency by forcing frequencies half or double the natural frequency – unless there are harmonics that correspond with the natural frequency.

The solution to resonance problems is usually stiffening of the structure or machine and this has the effect of increasing the natural frequency. Occasionally it may be practical to add mass to push the natural frequency downwards.

The optimum test and measurement methods for identification and correction of natural frequency problems are these:

- **Bump Test** – to identify simple natural frequencies.
- **Operating Deflection Shape (ODS)** tests to study mode shapes
- **Finite Element Analysis** – to compute natural frequencies and compare with measurements.
- **Run-up/run-down tests** – to study phase and amplitude changes.

6. Understanding Phase.

In vibration language **Phase** describes the relationships in time between a point on a shaft and the waveform of shaft rotation. It can also be a relationship in time between two or more single frequency waveforms.

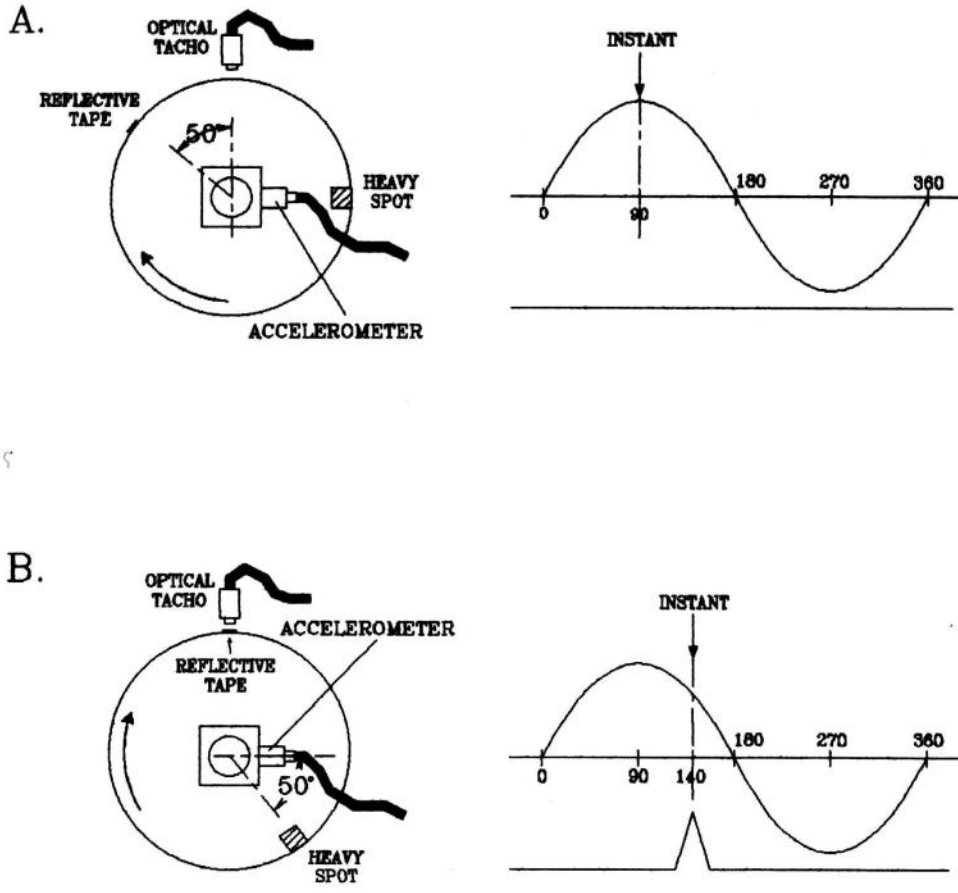
The following diagram shows the principles of how phase is measured using a single channel vibration analyser on a simple machine.

Note that the heavy spot defines the ‘high spot’ on the vibration waveform and comparing this to an optical tacho pulse from reflective tape installed anywhere on the shaft provides a phase reference.

This is the basis for the procedure of in-situ balancing. Applying a ‘trial weight’ to the rotor changes the location of the ‘heavy spot’ and, using simple vector calculations, the true location of the heavy spot can be found and a correction mass applied or mass removed.

Phase can also be measured between two or more simultaneous vibration waveforms using multi-channel vibration analysis equipment. This technique is used to produce phase-referenced vibration amplitude diagrams from which Operating Deflection Shape models can be computed.

Phase Lag on Rotating Shafts.



The natural frequency of a rotating shaft assembly is called the **Critical Frequency** and excessive deflections at the critical frequency are known as **Shaft Whirl**.

The critical frequency of a shaft is not easily determined by a bump test because of speed-dependent stiffness characteristics. If oil film bearings are fitted the running speed stiffness will be quite different to the stationary stiffness.

At critical speeds the amplitude of vibration builds up to the level determined by the amount of damping present. The lack of damping can sometimes be potentially destructive so critical speeds are traversed quickly.

Some high-speed machines may traverse up to three critical modes of the same shaft.

The diagram on the next page shows how the phase lag on a rotating shaft changes by 180 degrees as it moves from the **Rigid Zone** (stiffness controlled) to the **Flexible Zone** (inertia controlled). The shaft moves from rotating about its centre of geometry constrained by the bearings to rotating about its centre of mass. This changes the 'high spot' on the shaft by 180 degrees.

A shaft running well above its critical speed is known as a **Flexible Rotor** because it will take up a deflected shape according to the location of its principal planes of residual unbalance.

To ensure that a high-speed rotor operates with minimum shaft deflection (known as **Eccentricity**) the balancing process is very laborious and thorough. For example, the rotor of a five stage centrifugal pump would be balanced like this:

- a) The shaft would be balanced to tolerance by itself.
- b) Each individual impeller would be balanced to tolerance.
- c) The first impeller would be fitted to the shaft and the assembly balanced again, with trim corrections being made on the impeller.
- d) The second impeller would be fitted and the assembly balanced with corrections being made on the second impeller.
- e) This would be repeated five times. On the final, fully-assembled balance the rotor would be corrected for static unbalance equally along its length and for dynamic unbalance at the end rotors.

The result should be a rotor that runs very smoothly with minimum deflections. A rotor balanced like this will traverse shaft a critical speed with negligible increase in vibration.

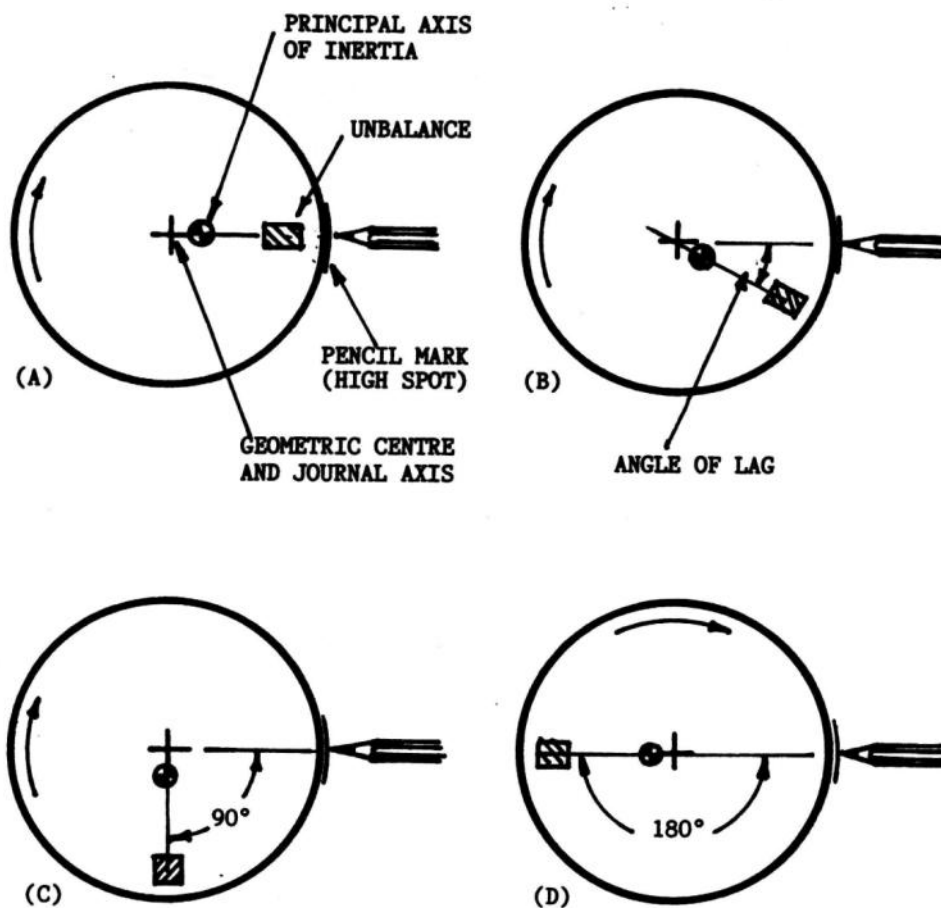


Diagram A. Rigid mode – the shaft is rotating about its geometric centre.

Diagram B. Speed increases. A phase lag is introduced between heavy spot and high spot.

Diagram C. At resonance. The heavy spot is 90 past the high spot. System is unstable.

Diagram D. Flexible mode. The shaft is rotating about its mass centre.