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Condition monitoring and diagnostics of machines — Vibration condition monitoring —

Part 2: Processing, analysis and presentation of vibration data

Surveillance et diagnostic d'état des machines — Surveillance des vibrations —

Partie 2: Traitement, analyse et présentation des données vibratoires

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Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the WTO principles in the Technical Barriers to Trade (TBT) see the following URL: Foreword - Supplementary information

The committee responsible for this document is ISO/TC 108, *Mechanical vibration, shock and condition monitoring*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

This second edition cancels and replaces the first edition (ISO 13373-2:2005), which has been editorially revised.

ISO 13373 consists of the following parts, under the general title *Condition monitoring and diagnostics of machines — Vibration condition monitoring*:

— *Part 1: General procedures*

— *Part 2: Processing, analysis and presentation of vibration data*

— *Part 3: Guidelines for vibration diagnosis*

— *Part 9: Diagnostic techniques for electric motors*

Introduction

The purpose of this part of ISO 13373, which covers the area of vibration condition monitoring of machines, is to provide recommended methods and procedures for processing signals and analyzing data obtained from vibration transducers attached to a machine at selected locations for the purpose of monitoring the dynamic behaviour of a machine.

Broadband vibration measurements provide an overview of the severity of machine vibration that can be observed and trended to alert machine users when an abnormal condition exists with a machine. Processing and analyzing these vibration signals further in accordance with the procedures specified in this part of ISO 13373 gives the user an insight into ways of diagnosing the possible cause or causes of the machinery problems, which allows for more focused continued condition monitoring.

The advantages of such a monitoring programme are that machinery operators will not only be made aware that a machine can fail at a certain time, and that maintenance needs to be planned prior to the failure, but that it will provide valuable information regarding what maintenance needs to be planned and performed. The vibrations are manifestations or symptoms of problems such as misalignment, unbalance, accelerated wear, flow and lubrication problems.

ISO 13373-1 contains guidelines for vibration condition monitoring of machines. This part of ISO 13373, however, contains guidelines for the processing, analysis and presentation of the vibration data thus obtained, and that can be used for diagnostics to determine the nature or root causes of problems.

The signal processing, analysis and diagnostic procedures applied to vibration condition monitoring can vary depending on the processes to be monitored, degree of accuracy desired, resources available, etc. A well-conceived and implemented condition monitoring programme will include consideration of many factors, such as process priority, criticality and complexity of the system, cost-effectiveness, probability of various failure mechanisms and identification of incipient failure indicators.

An appropriate process analysis needs to dictate the types of data desired to monitor the machinery condition suitably.

The vibration analyst needs to accumulate as much pertinent information as possible about the machine to be monitored. For example, knowing the vibration resonance frequencies and the excitation frequencies from design and analytical information will provide an insight regarding the vibration frequencies anticipated and, consequently, the frequency range that is to be monitored. Also, knowing the machine's initial condition, the machine's operational history, and its operating conditions provides additional information for the analyst.

Other advantages to this pre-test planning process are that it provides guidance as to what types of transducers are necessary, where they need to be optimally located, what kind of signal conditioning equipment is required, what type of analysis would be most appropriate, and what are the relevant criteria.

Further standards on the subject of machinery condition monitoring and diagnostics are in preparation. These are intended to provide guidance on the overall monitoring of the "health" of machines, including factors such as vibration, oil purity, thermography and performance. Basic techniques for diagnosis are described in ISO 13373-3.

Condition monitoring and diagnostics of machines — Vibration condition monitoring —

Part 2: Processing, analysis and presentation of vibration data

1 Scope

This part of ISO 13373 recommends procedures for processing and presenting vibration data and analyzing vibration signatures for the purpose of monitoring the vibration condition of rotating machinery, and performing diagnostics as appropriate. Different techniques are described for different applications. Signal enhancement techniques and analysis methods used for the investigation of particular machine dynamic phenomena are included. Many of these techniques can be applied to other machine types, including reciprocating machines. Example formats for the parameters that are commonly plotted for evaluation and diagnostic purposes are also given.

This part of ISO 13373 is divided essentially into two basic approaches when analysing vibration signals: the time domain and the frequency domain. Some approaches to the refinement of diagnostic results, by changing the operational conditions, are also covered.

This part of ISO 13373 includes only the most commonly used techniques for the vibration condition monitoring, analysis and diagnostics of machines. There are many other techniques used to determine the behaviour of machines that apply to more in-depth vibration analysis and diagnostic investigations beyond the normal follow-on to machinery condition monitoring. A detailed description of these techniques is beyond the scope of this part of ISO 13373, but some of these more advanced special purpose techniques are listed in Clause 5 for additional information.

For specific machine types and sizes, the ISO 7919 and ISO 10816 series provide guidance for the application of broadband vibration magnitudes for condition monitoring, and other documents such as VDI 3839 provide additional information about machinery-specific problems that can be detected when conducting vibration diagnostics.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

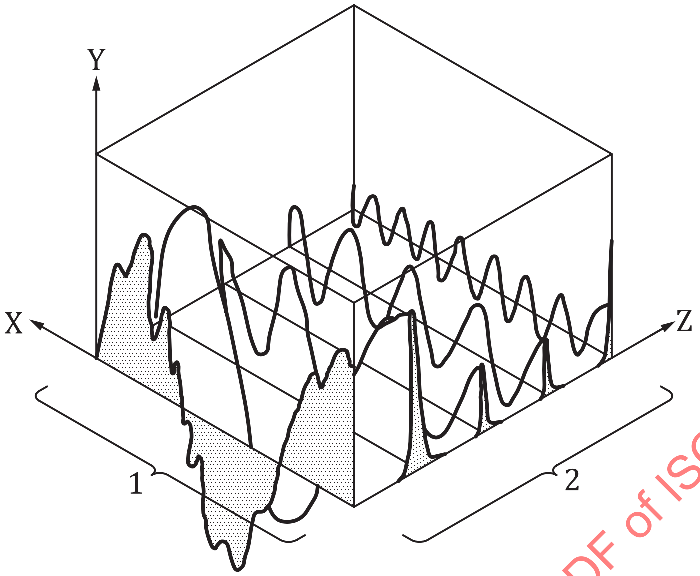
ISO 1683, *Acoustics — Preferred reference values for acoustical and vibratory levels*

3 Signal conditioning

3.1 General

Virtually, all vibration measurements are obtained using a transducer that produces an analogue electrical signal that is proportional to the instantaneous value of the vibratory acceleration, velocity or displacement. This signal can be recorded on a dynamic system analyzer, investigated for later analysis or displayed, for example, on an oscilloscope. To obtain the actual vibration magnitudes, the output voltage is multiplied by a calibration factor that accounts for the transducer sensitivity and the amplifier and recorder gains. Most vibration analysis is carried out in the frequency domain, but there are also useful tools involving the time history of the vibration.

Figure 1 shows the relationship between the vibration signal in the time and frequency domains. In this display, it can be noted that there are four overlapping signals that combine to make up the composite trace as it would be seen on the analyzer screen (grey trace in the XY plane). Through the Fourier process, the analyzer converts this composite signal into the four distinct frequency components shown.



Key

X time

Y amplitude/magnitude

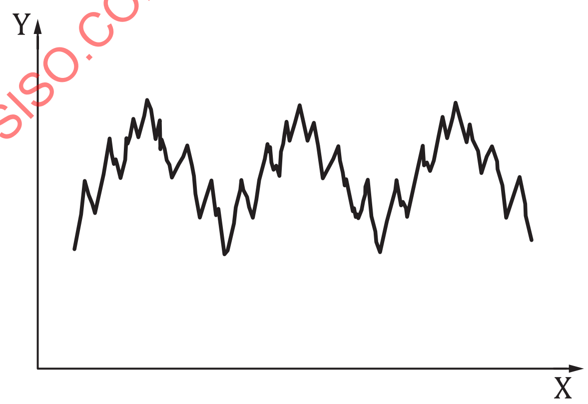
Z frequency

1 time domain oscillogram

2 frequency domain spectrum

Figure 1 — Time and frequency domains

Figure 2 is a simpler example of a composite trace from a single transducer as seen on the analyzer screen. In this case, there are only three overlapping signals, as shown in Figure 3, and their distinct frequencies are included in Figure 4.

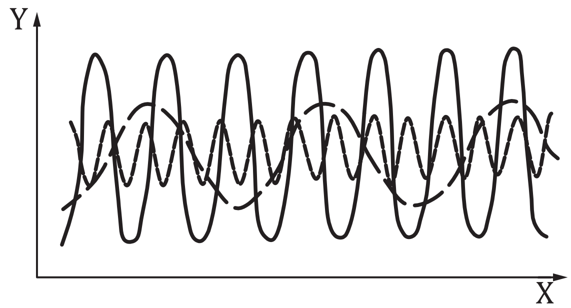


Key

X time

Y amplitude

Figure 2 — Basic spectra composite signal

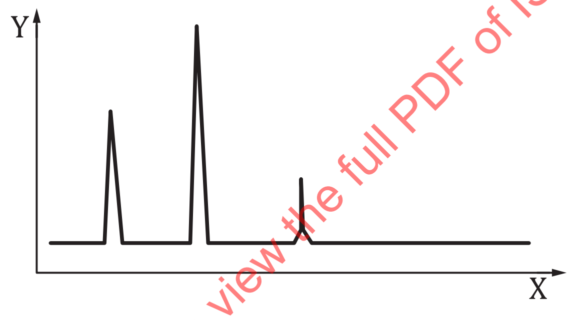


Key

X time

Y amplitude

Figure 3 — Overlapping signals



Key

X frequency

Y amplitude

Figure 4 — Distinct frequencies

For many investigations, the relationship between vibration on different structure points, or different vibration directions, is as important as the individual vibration data themselves. For this reason, multi-channel signal analyzers are available with built-in dual-channel analysis features. When examining signals with this technique, both the amplitude and phase relationships of the vibration signals are important.

3.2 Analogue and digital systems

3.2.1 General

The analogue signal from a transducer can be processed using analogue or digital systems. Traditionally, analogue systems were used that involved filters, amplifiers, recorders, integrators and other components which modify the signal, but do not change its analogue character. More recently, the advantages of digitizing the signals have become more and more apparent. An analogue-to-digital converter (ADC) repeatedly samples the analogue signal and converts it to a series of numerical values. Mathematical routines on computers can then be used to filter, integrate, find spectra (see 4.3.2), develop histograms or do whatever is required. Of course, the digitized signal may also be plotted as a function of time. The analogue signal, as well as the digitized one, contains the same information on the premises of an appropriate choice of the sampling frequency.

When using either an analogue method or a digital method, it is important to know the sensitivity of the signal to be measured. The sensitivity is the ratio of the actual output voltage value of the signal to the actual magnitude of the parameter measured. To obtain adequate signal definition, the signal of interest should be significantly greater than the ambient noise levels, but not so large that the signal is distorted (e.g. so that the peaks of the signal are clipped).

3.2.2 Digitizing techniques

The most important parameters in the digitizing process are the sampling rate and the resolution. It is important to ensure that no frequencies are present above half the sampling rate. Otherwise, time histories will be distorted or fast Fourier transforms (FFT) will show aliasing components that do not really exist (see 4.3.7 for further information about aliasing). The sampling rate will be determined by the type of analysis to be performed and the anticipated frequency content of the signal. If a plot of vibration versus time is desired, it is recommended that the sampling rate be of about 10 times the highest frequency of interest in the signal. However, if a frequency spectrum is desired, an FFT calculation requires that the sampling rate needs to be greater than two times the highest frequency of interest to be measured. Anti-aliasing filters are used to eliminate any high-frequency noise or other high-frequency components that are above half the sampling rate. When digitizing, the number of bits used to represent each sample shall be sufficient to provide the required accuracy.

3.3 Signal conditioners

3.3.1 General

The vibration signals from transducers usually require some sort of signal conditioning before they are recorded in order to obtain proper voltage levels for recording, or to eliminate noise or other unwanted components. Signal conditioning equipment includes transducer power supplies, pre-amplifiers, amplifiers, integrators and many types of filters. Filtering is discussed further in 3.4.

3.3.2 Integration and differentiation

Vibration records can be in terms of displacement, velocity or acceleration. Usually, one of the parameters is preferred because of the frequency range of interest (low-frequency signals are more apparent when using displacement, and high-frequency signals are more apparent when using acceleration) or because of the applicable criteria. A vibration signal can be converted to a different quantity by means of integration or differentiation. Integrating acceleration with respect to time gives velocity, and integrating velocity gives displacement. Double integration of acceleration will produce displacement directly. Differentiation does the opposite of integration.

Mathematically, for harmonic motion, the following relationships apply:

displacement:

$$x = \int v\mathrm{d}t = \iint \left(a\mathrm{d}t\right)\mathrm{d}t = -\frac{\mathrm{i}}{\omega}v = -\frac{1}{\omega^2}a \tag{1}$$

velocity:

$$v = \frac{\mathrm{d}x}{\mathrm{d}t} = \int a\mathrm{d}t = \mathrm{i}\omega x = -\frac{\mathrm{i}}{\omega}a \tag{2}$$

acceleration:

$$a = \frac{\mathrm{d}v}{\mathrm{d}t} = \frac{\mathrm{d}^2x}{\mathrm{d}t^2} = -\omega^2 x = \mathrm{i}\omega v \tag{3}$$

where ω is the angular frequency of the harmonic vibration with $\omega = 2\pi f$.

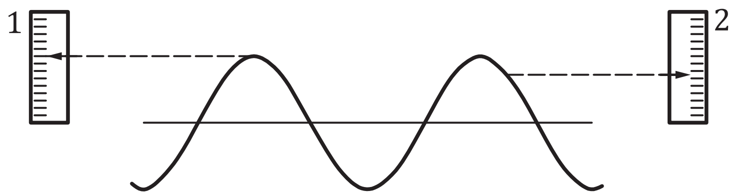
NOTE See also 4.3.12.

A common vibration transducer is the accelerometer, so integration is much more common than differentiation. This is fortunate since differentiation of a signal is more difficult than integration, but special care shall be taken when integrating signals at low frequencies. A high-pass filter should be used to eliminate frequencies lower than those of interest before integrating.

3.3.3 Root-mean-square vibration value

The root-mean-square (r.m.s.) value of the vibration signal is commonly used in vibration evaluation standards. Criteria often apply to r.m.s. vibration values within a certain frequency range. This is the most used quantity of vibration over a given time period. Other measures of a vibration signal can be confusing when there are many frequency components, or when there is modulation, etc. However, the r.m.s. value is a mathematical quantity that can be found for any signal, and most instruments are designed to find that quantity (see Figure 5). Alternatively, the r.m.s. value can be found by using a spectrum analyzer, by integrating the spectrum between the upper and lower frequencies of interest.

A vibration signal may be filtered as required and displayed on an r.m.s. meter if the reading does not change significantly in a short time period. However, if the indicated output varies significantly, an average over a certain period of time shall be obtained. This can be done with an instrument that has a longer time constant.



a) Sinusoidal signal where the r.m.s. value equals 0,707 times the peak value

b) Non-sinusoidal signal

Key

1 peak value

2 r.m.s. value

Figure 5 — Peak and r.m.s. values

3.3.4 Dynamic range

The dynamic range is the ratio between the largest and smallest magnitude signals that a particular analyzer can accommodate simultaneously. The magnitudes of the signals are proportional to the output voltages of the transducers, usually in millivolts.

The dynamic range in analogue systems is usually limited by electrical noise. This is usually not a concern with respect to the transducer itself, but filters, amplifiers, recorders, etc., all add to the noise level, and the result can be surprisingly high.

In digital systems, the dynamic range is dependent on the sampling accuracy, and the sampling rate shall be adequate for the frequencies of concern. The relationship between the number of bits, N, used to sample an analogue signal and the dynamic range D, in decibels, (if one bit is used for the sign) is as follows:

$$6\,(N-1) = D \tag{4}$$

Therefore, a dynamic signal analyzer (DSA) with 16 bits of resolution will have a dynamic range of 90 dB, but any inaccuracies will reduce the dynamic range.

3.3.5 Calibration

The calibration of individual transducers is well covered in the referenced documents (e.g. ISO 16063-21), and is usually carried out in the laboratory before their use *in situ*. It is recommended, however, that a calibration check be carried out for any field installation. The field calibration check normally does not include the calibration of the transducer, but does include the rest of the measuring/recording system, such as amplifiers, filters, integrators and recorders. Most often, it involves the insertion of a known signal into the system to see what output relates to it. The signal can be a d.c. step, a sinusoid or random noise, depending on the type of measurement.

Certain transducers, such as displacement transducers or proximity probes, are pre-calibrated. However, in this case, their calibrations should be checked in the field in conjunction with the surface being measured, since proximity probes are sensitive to shaft metallurgy and finish. Calibration of these probes is carried out in place with micrometre spindles, and the outputs for each are noted.

When checking the calibration of seismic transducers in the field, a shake table is required.

Strain gauges are also often calibrated in the field after they are installed. The most desirable calibration is for a known load to be applied to the component being measured. If that is not practical, a shunt calibration may be made where a calibration resistor is connected in parallel with the strain gauge, thus changing the apparent resistance of the gauge by a known amount, which is equivalent to a certain strain determined by the gauge factor.

3.4 Filtering

There are three basic types of filters available for signal conditioning and analysis:

— low pass;

— high pass;

— bandpass.

Low-pass filters, as the name implies, are transparent only for the low-frequency components of the signal, and they block out the high-frequency components above the filter limiting frequency (cut-off frequency). Examples of application are anti-aliasing filters (see 4.3.7), or filters that exclude high-frequency components that are unwanted for special investigations (e.g. gear meshing components for balancing).

High-pass filters are mainly used to exclude low-frequency transducer noise (thermal noise), or some other unwanted components from the signal, prior to analysis. This can be important since such components, although of no interest, can dramatically reduce the useful dynamic range of the measurement equipment.

Bandpass filters, when included for analysis, are used to isolate distinct frequency bands. Very common bandpass filter types are the octave filters or 1/*n* octave filters, which are especially used to correlate vibration measurements with noise measurements.

Filtering is particularly important when analysing signals with large dynamic ranges. If there are frequencies in the spectra with both high and low amplitudes, for instance, they cannot usually be analyzed with the same level of accuracy because of limitations in the dynamic range of the analyzer. In such cases, it can be necessary to filter out the high-amplitude components to examine more closely those of low amplitude.

Filtering is also important for separation of informative signals and disturbances (as electronic noise is in the high-frequency range or seismic waves are in a very low-frequency range).

When filters are used to isolate a particular frequency component to examine the waveform, care shall be taken to ensure that the filter sufficiently excludes any component of frequencies other than those of interest. Simple filters, analogue as well as digital, do not have very sharp cut-off characteristics, because the filter slope outside of the transmission band is poor.

EXAMPLE A particular filter with a 24 dB per octave slope will pass about 15 % of a component with twice the frequency, and about 45 % of a component with 1,5 times the cut-off frequency. To improve the filter's suppression characteristics, several simple filters can be cascaded, or a higher-order filter can be used instead.

4 Data processing and analysis

4.1 General

Data processing consists of raw-data acquisition, filtering out unwanted noise and/or other non-related signals, and formatting the measured signals in the form required for further diagnosis. Therefore, data processing is an important step towards achieving a fruitful and meaningful diagnosis. The device that acquires the vibration signals from the transducer should have adequate resolution in both amplitude and time. If digital data acquisition is utilized, then the amplitude resolution should be high enough for the application. A higher number of bits of resolution provide the ability to obtain greater accuracy and sensitivity, but it typically requires more expensive hardware and greater processing power.

Once the signals are acquired, the next step is to process them and then display the outputs in various useful formats so that the diagnosis is made much easier for the user. Examples of such formats include Nyquist plots, polar plots, Campbell diagrams, cascade and waterfall plots and amplitude decay plots. The objective of this clause, therefore, is to present these various methods of presentation available to the user in order to determine better the conditions of machines.

4.2 Time domain analysis

4.2.1 Time wave forms

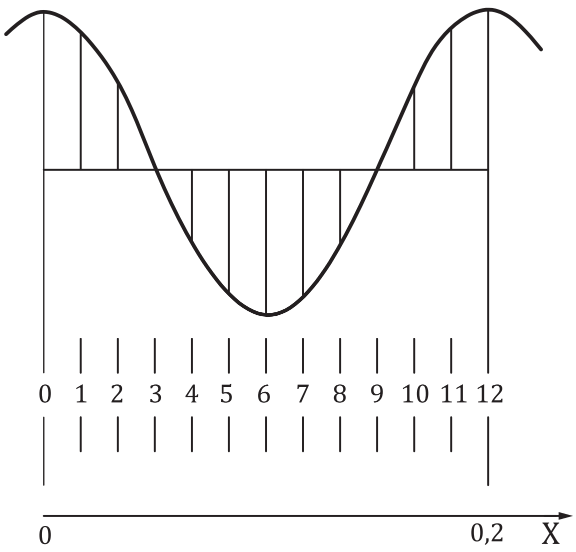
In the past, waveform analysis was the primary method of vibration analysis. An instantaneous vibration versus time strip chart or oscillograph was usually analyzed graphically, and broadband peaks were noted. While these broadband techniques are still being used, it is helpful to look at the waveform with some of the more basic techniques in mind. For example, a scratched journal can be detected by looking at waveform data from displacement transducers, a waveform with a clipped top or bottom can indicate a rub, mechanical looseness, etc.

While these time-domain signatures can portray waveforms that provide basic information regarding the nature of a phenomenon occurring in a machine, the more in-depth frequency analysis techniques described in 4.3 can be required.

The analysis of waveforms is based on the principle that any periodic record can be represented as a superposition of sinusoids having frequencies that are integral multiples of the frequency of the waveform. Figure 6 to Figure 9 show several examples of waveforms.

Figure 6 is essentially a one-cycle sinusoid with a constant amplitude. The double amplitude (or peak-to-peak) of the vibration is obtained by measuring the double amplitude of the trace, and multiplying by the sensitivity of the measuring and recording system, which is found by calibration. The frequency is found by counting the number of cycles in a known time period. The time on an oscillograph is indicated by timing lines, or simply by knowing the paper speed. For the trace shown, there are 60 timing lines per second; therefore, the 12 lines indicate that the fundamental period, T, is 0,2 s, and hence the frequency, $f = 1/T$, is 5 Hz. Accuracy is improved if the number of cycles in a longer section of the record is used.

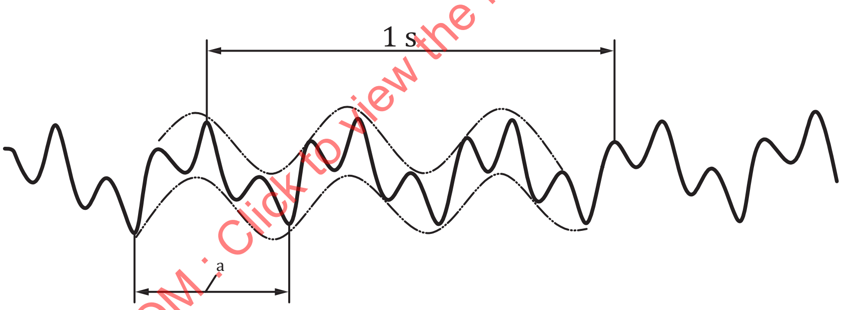
Figure 7 is the superposition of two sinusoids with three cycles of the lowest frequency shown. The components can be separated by drawing sinusoidal envelopes (upper and lower limits) through all the peaks and troughs as shown. The amplitude and frequency of the low-frequency component is that of the resulting envelope. The vertical distance between envelopes indicates the peak-to-peak value of the high-frequency component, and the high frequency can usually be counted. In this example, it can be found that the frequencies differ by a factor of three. When the frequency ratio of two superimposed sinusoids is high, they can be separated as shown; in all other cases, a Fourier analysis is more useful.



Key

X time, s

Figure 6 — Waveform characteristics



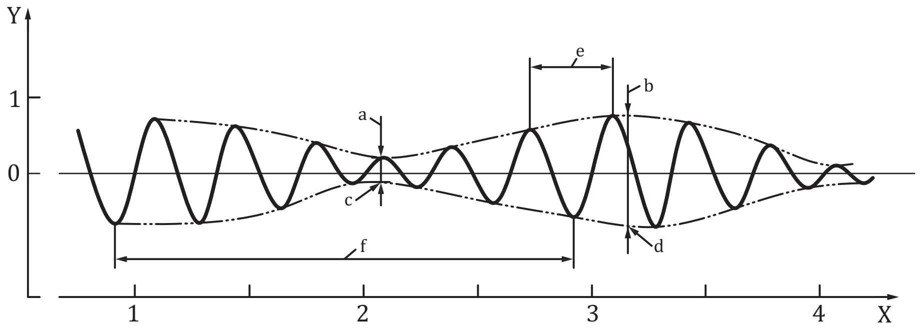
Key

a Cycle.

Figure 7 — Superposition

4.2.2 Beating

Often, signals look like the trace of Figure 8, where the envelopes are out of phase, causing bulges and waists. This signal is caused by two components that are close in frequency and amplitude. This is called beating, which is a special case of superposition. An example of beating is the two blade frequencies of the twin propeller drives of a ship added together. The peaks of the two signals alternately add and subtract. Other characteristics of beating are that the lengths of the beats are about the same, and the spacing between the peaks at the bulges is different than that at the waists. The distances between the envelopes at the bulges and waists represent the sums and differences, respectively, of the peak-to-peak values of the two components. Another example is the vibration that is forced by two coupled machines (compressors or others), driven by asynchronous electrical motors.



Key

X time, s

Y amplitude (arbitrary unit)

a Peak-to-peak value at waist: 0,2.

b Peak-to-peak value at bulge: 0,7.

c Waist.

d Bulge.

e Vibration cycle: 0,33 s corresponds to 3 Hz.

f Beat cycle: 2 s corresponds to 0,5 Hz.

Figure 8 — Example of beating

EXAMPLE If the components' amplitudes are X_m for the major and X_n for the minor, measurements show that $X_m + X_n = 0,7$ and $X_m - X_n = 0,2$, the solution being $X_m = 0,45$ and $X_n = 0,25$. These record amplitudes have to be multiplied by the system sensitivity to get actual amplitudes. The major frequency can be found by counting the number of peaks as described before (in Figure 8, it is 3 Hz). This frequency is also an integral multiple of the beat frequency, in this case six times. The frequency of the minor component is either one more (7) or one less (5) times the beat frequency. The spacing of the peaks at the waist indicates which one it is, since it reflects the major component. In Figure 8, the spacing is narrower so the major component has the higher frequency. In Figure 8, the beat frequency is 0,5 Hz and the minor frequency is five times that, i.e. 2,5 Hz.

It should be noted that the beat frequency is the difference between the frequencies of both components, but the average peak frequency is equal to one-half the sum of both. A simple rule for calculating the frequencies is:

$$f_b = f_m - f_n \tag{5}$$

where

f_b is the beat frequency;

f_m is the frequency of the major component;

f_n is the frequency of the minor component.

In the example shown in Figure 8, by counting the peaks, there are six peaks in 2 s, which means $f_m = 3$ Hz. The beat cycle is 1 cycle in that same time period, which means $f_b = 1/(2\ \text{s}) = 0,5$ Hz. Inversing Formula (5) to become $f_n = f_m - f_b$, yields the frequency of the minor component $f_n = 3\ \text{Hz} - 0,5\ \text{Hz} = 2,5$ Hz.

4.2.3 Modulation

Figure 9 shows the trace of a modulated vibration signal. It looks similar to beating but there is actually only one component whose amplitude is varying with time (modulating). This is distinguishable from beating because the spacing of the peaks is the same at the bulges and the waists. Also, the length of the bulges might not be the same. Gear problems often result in modulation of the gear mesh frequency at the gear rotational frequency.

Unfortunately, many vibration records contain more than two components, and can involve modulation and perhaps beating as well. Such records are extremely difficult to analyze, but the analyst might be able to find sections of the record in which one component is temporarily dominant, and obtain the frequency and amplitude of that component in that section.

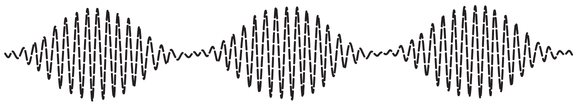


Figure 9 — Modulation

4.2.4 Envelope analysis

Envelope analysis is a process for the demodulation of low-level components in a narrow frequency band, which are obscured by a high-level broadband vibration (impulse-excited free vibration, gear meshing vibration, and others). Envelope detection provides a means for recognizing flaws earlier and with greater reliability. Its most common application is in analysis of gears and rolling element bearings where a low-frequency, generally low-amplitude repetitive event (such as a defective tooth entering mesh or a spalled ball or roller striking a race) excites high-frequency resonance(s), resulting in the high frequency being modulated by the defect frequency. A sample of an envelope trace is shown in Figure 10.

It should be noted that the modulated component needs to be separated previously by narrow band filtering.

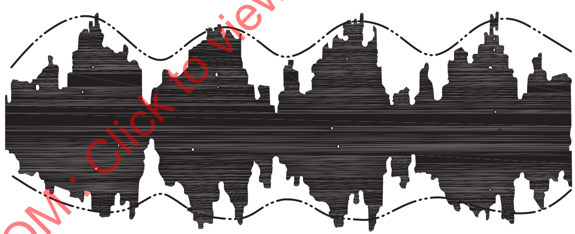


Figure 10 — Envelope analysis

4.2.5 Monitoring of narrow-band frequency spectrum envelope

Monitoring of narrow-band frequency spectrum envelope detects any penetration of an envelope, which is usually an alarm limit, around a reference spectrum. The constant-bandwidth envelope, where the frequency difference is the same number of lines at low and high frequencies, is generally used for machines with constant rotational speed.

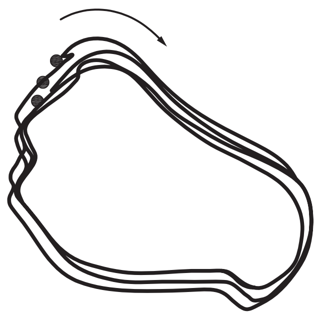
A constant-percentage bandwidth envelope increases the frequency difference (offset) between the envelope and the monitored component proportionally to the increase in frequency. This method has advantages because all harmonic components will remain in the same frequency band over small speed changes.

Amplitude limits for individual frequency components are of two types. A constant-percentage offset is the most commonly used because it is the simplest to calculate and only requires a single reference spectrum.

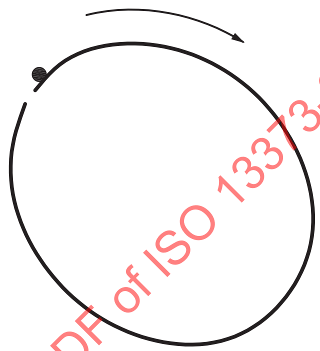
A more representative method is to calculate a statistical mean for each segment in the envelope, and then set the alarm limit 2,5 to 2,8 standard deviations above the mean. The statistical calculation requires 4 high-resolution spectra or 5 high-resolution spectra, and then automatically accounts for normal differences in amplitude variation commonly observed in the machinery spectra.

4.2.6 Shaft orbit

Orbit analysis can be performed on any machine using displacement transducers usually mounted 90° apart. On large rotating machinery with sleeve bearings, it is common practice to use shaft orbit analysis to determine the movement of the shaft within the bearing clearance space. However, care should be taken to ensure that the shaft orbit display is not distorted unnecessarily by the effects of shaft mechanical and electrical run-out. Proper interpretation of the orbit can yield insight into the nature of the forcing function. It is also possible to determine whether the rotor whirl is forwards (in the direction of rotation) or backwards (against the rotation). Orbit presentations are displayed as either unfiltered or filtered signals. Typical broadband (unfiltered) and single-frequency (filtered) orbit plots are shown in Figure 11.



a) Unfiltered



b) Filtered

Figure 11 — Shaft orbits

The synchronous (1×) filtered display is common; however, other harmonics or sub-synchronous frequencies are displayed in an orbit presentation to further describe or solve a problem. A mark (point, highlight, etc.), which provides a shaft reference (e.g. once-per-revolution signal), gives information about the relationship between the vibrational and the rotational frequencies.

The orbit plot presents the dynamic motion of the centre of the rotating shaft at the measurement plane. An orbit is sometimes called a Lissajous presentation. The transducers for the orbits should be the same type and should be mounted orthogonally (90° apart). If the transducers are not orthogonal, the orbit will be skewed. In the case of a notched shaft, the convention is blank–bright. Blank indicates the beginning of the notch, bright indicates the end of the notch. Therefore, in Figure 11 the whirl direction is clockwise.

The direction of shaft rotation, clockwise or counter-clockwise, is independently determined depending upon the view direction. If the whirl direction is the same as the direction of rotation, the vibration is referred to as forward whirl. Backward whirl is when the whirl direction is opposite from the direction of rotation. In Figure 11, since both the rotation and the whirl directions are clockwise, the whirl is forward.

4.2.7 d.c. shaft position

To determine the d.c. shaft position, displacement transducers are frequently used to give indications of the relative loading of sleeve bearings by their eccentricity ratios. The attitudes of the journals within their bearings as measured from the d.c. part of the signal (i.e. the gap) is very useful in monitoring large machines. The d.c. position can validate appropriate bearing lift and correct shaft position. Care should be taken, however, to avoid misrepresentation due to d.c. signal drift over a long period of time.

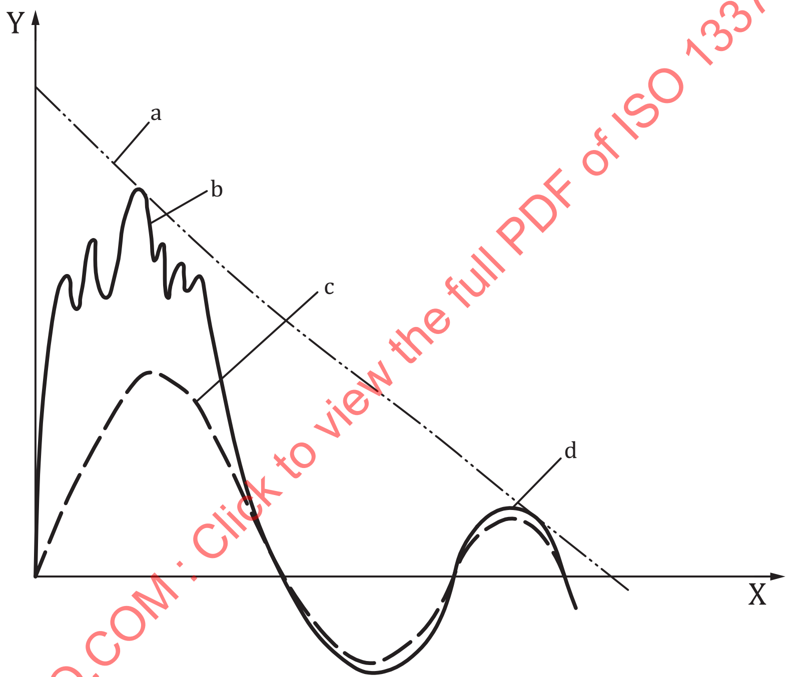
4.2.8 Transient vibration

Transient speed vibration is usually described as the vibration information obtained during the start-up and coast-down conditions of a machine train. The vibration data are usually displayed in presentation formats such as cascade (waterfall) diagrams, Bode plots, polar diagrams (Nyquist diagrams) and Campbell diagrams.

Transient vibration of a structure occurs when it is excited by an instantaneous force. It can be a single pulse or an oscillating excitation of short duration. When the excitation ceases, the structure tends to vibrate at its natural frequencies while the damping in the system causes it to decay exponentially.

Therefore, the time history of the structural response after the force stops is a combination of decreasing sinusoid(s). An example of a damped sinusoid is given in Figure 12. It can be noted that the composite waveform due to superposition of the natural modes of the system is excited simultaneously by the instantaneous forcing. In general, the higher frequency components decay rapidly and the composite waveform progressively degenerates into a damped sinusoidal response of the lowest frequency mode, the higher frequency modes having been damped out.

Faults in rolling element bearings are often detected from repeated high-frequency transient responses to ball or race defects.



Key

X time

Y amplitude

a Exponential decay of peak amplitude envelope.

b Composite waveform.

c Waveform of lowest frequency mode.

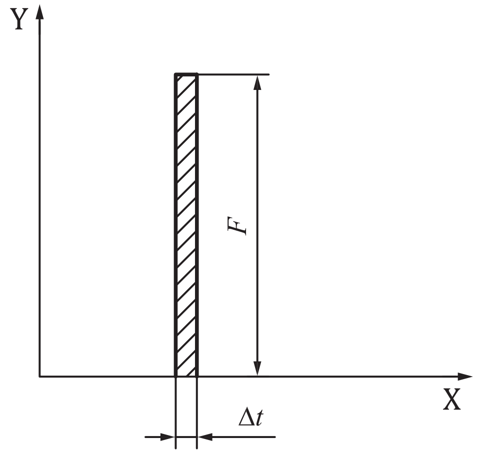
d Degenerated waveform.

Figure 12 — Transient vibration

4.2.9 Impulse

Impulse response is the time history of the vibratory response of a mechanical system to an impulse that can be represented as a force, F, applied over a very small period of time, Δt, where the impulse is the integral of $F\,\mathrm{d}t$ from t to $(t + \Delta t)$, see Figure 13.

In many cases, impulse response is used to identify resonance frequencies in stationary structures.



Key

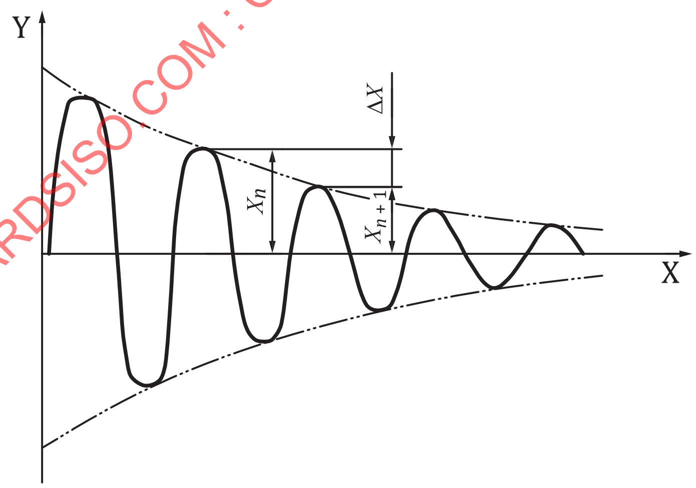
X time

Y force

Figure 13 — Impulse excitation

4.2.10 Damping

Damping is the mechanism by which vibratory motion is converted to other forms of energy, usually heat, resulting in decaying vibration magnitudes. The amount of damping, *c*, is often proportional to the vibratory velocity and, even when it is not, it is often assumed to be for purposes of mathematical analysis. A system has critical damping, c_c, if it has the smallest amount of damping required to return the system to its equilibrium position without oscillation. If the system's damping is less than critical, it will oscillate with decaying amplitudes (see Figure 14 and ISO 2041). For a multi-degree-of-freedom system, some modes can have less than critical damping and some can have more.



Key

X time

Y amplitude

Figure 14 — Decaying amplitudes due to damping

If the amplitude of the decaying vibration of a particular mode, X, is plotted versus the time duration, the logarithmic decrement, d, is:

$$d = 1/n \ln(X_1/X_{n+1}) \quad (6)$$

where

n is the number of cycles for the amplitude to decay from X_1 to X_{n+1}.

The loss factor is a common measure of the relative damping in a system. The logarithmic decrement, d, is related to the loss factor, h, by $h = d/\pi$.

NOTE 1 Typically, the symbols used to denote the loss factor include h, z and η. Those for the logarithmic decrement include α and Λ.

The loss factor can also be found in terms of the decay rate, X', in decibels per second, as follows:

$$h = X' / \left(27{,}3\ f_{\mathrm{n}}\right) \quad (7)$$

where

f_{n} is the natural frequency, in hertz.

The amount of damping, c, in a system is indicated by Q, which is the magnification factor at the undamped natural frequency. The magnification factor is a function of frequency and is the ratio of the system's dynamic displacement amplitude to the static displacement of the system if it were subjected to a constant force of the same magnitude. Provided that there is no significant interaction between the modes, then for a particular mode, Q can be found from:

$$Q = 1/(2c/c_{\mathrm{c}}) \quad (8)$$

From measured response curves, Q can be approximated for a particular mode from the ratio of the resonance frequency, f_{r}, to the difference between the frequencies at the half-power points (0,707 times the maximum amplitude) on each flank of the curve:

$$Q = \frac{f_{\mathrm{r}}}{\Delta f} \quad (9)$$

where

f_{r} is the resonance frequency;

$\Delta f = f_2 - f_1$ with f_1 and f_2 being the half-power points.

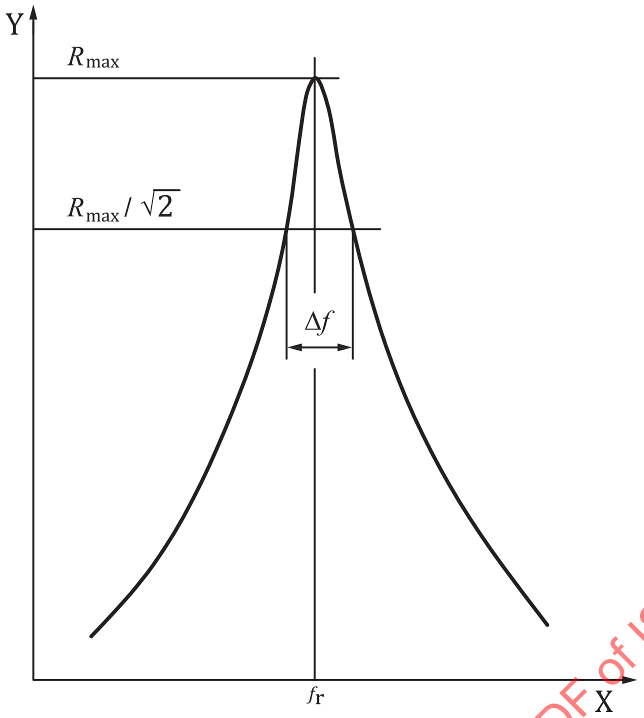
The magnification factor is related to the logarithmic decrement by the following approximation:

$$Q \approx \pi/d \quad (10)$$

NOTE 2 If the damping is small, $Q = 1/h$.

As an example, Figure 15 shows a typical representation of the Q factor derived from a Bode plot. A similar result can be obtained from a polar diagram.

Damping is a useful quantity when investigating the cause and effect of vibration in rotating machinery. A mode near the operating speed can be acceptable as long as it is well damped and therefore not contributing to the response. Alternately, a mode with very little damping can be so sensitive that the machine will respond violently, or might not even be able to pass through a resonance speed.



Key

X frequency

Y response

Figure 15 — *Q* factor

4.2.11 Time domain averaging

Each signal contains components that are synchronous with processes or motions in the monitored machine or equipment, as well as non-synchronous ones (with an origin that is independent of the system under observation). These components can be separated by frequency analysis (see 4.3). Another common technique applied to identify these occurrences is called time domain averaging.

In the time domain averaging process, each data sample is synchronized to different rotating elements via a reference pulse or a trigger. The averaging, which can range from a few samples to more than 200, is computed in the time domain, and a spectrum is obtained only based on the resulting averaged time waveform. Those time signal parts which are non-synchronous with the reference progressively cancel each other. The more averages the better, the number depends on the application.

In time domain averaging, the corresponding samples are actually algebraically added for each record, and then divided by the number of records. The result is that the desired repeating waveform remains intact while all other averages tend toward zero (including other repeating waveforms). The rate at which they decay away equals the square root of the number of averages.

NOTE 100 averages (records) will reduce the unwanted signals by a factor of ten; 10 000 averages will reduce them by a factor of 100.

This technique is very useful for the identification of which rotor in a multiple-rotor machine is the source of a vibration phenomenon. It can be used to detect various faults, such as damaged gears, blades and rolls in paper machines.

EXAMPLE 1 A good example is in the case of a turbine-driven pump with gear drive with different shaft speeds, where there is a once-per-revolution synchronizing trigger on each shaft. The signal from an accelerometer mounted on the gearbox can be analyzed using time domain averaging in which the process is repeated for each synchronizing trigger. Using the turbine shaft trigger, the signal reduces to a sinusoidal waveform indicating the degree of unbalance in the turbine shaft. Using the pump shaft trigger, the signal reduces to a periodic pattern at vane passing frequency, indicating a fixed radial offset of the pump shaft within its housing.

EXAMPLE 2 Another example is where strain gauge bridges are mounted on two large hydroturbine blades, and their signals brought out by means of telemetry. A once-per-revolution trigger is used to synchronize the time domain averaging process. After several averages to reduce the flow noise, an uneven pattern might emerge which is identical for each blade, but offset in time by the rotational angle between the blades. The diagnosis is an uneven flow through the series of wicket gates that feed the turbine. The pattern is then used to re-adjust the gates to even out the flow and reduce the dynamic stresses on the blades.

Although very effective, time domain averaging, by its very nature, cannot show asynchronous events such as antifriction bearing faults.

The averaging of complex frequency spectra of successive realizations normally requires a steady-state vibration condition. If there is an unsteady excitation frequency or a changing rotational speed, the simple time domain averaging does not apply. Instead of this, the signal needs to be sampled in constant intervals of the exciting process (e.g. equidistant rotor angle intervals or other positions; this can be done by means of an encoder). The result of the succeeding frequency transformation is an ordering spectrum instead of a frequency spectrum. For impulse response signals, averaging can be executed in the time domain by event triggering, e.g. a trigger tuned by the excitation impulse.

The source of the trigger is not limited to rotating equipment. Other applications, for instance, are paper machine belts, conveyor belts, etc. In addition, the source of the signal is not limited to vibration. It may be a process signal related to the machine in question that can identify a malfunction or a process parameter that should be monitored for fault development. A frequency multiplier can also be used instead of triggering from different shafts, e.g. in the case of multiple shafts such as in a gear box.

4.3 Frequency domain analysis

4.3.1 General

A great deal of vibration analysis is done in the frequency domain because the various sources of vibration can usually be isolated by the frequencies at which they occur. A single channel analyzed in the frequency domain gives a great deal of information, but often it is important to relate vibration to a second channel as either a phase or amplitude reference, or both.

4.3.2 Fourier transform

The basic technique for converting a broadband time trace to discrete frequencies, or frequency bands, is by the application of the Fourier transform (FT), a mathematical technique that identifies the sinusoidal components that make up the total vibration signal, including any mechanical or electrical noise that might be present. This analysis can be realized with the help of a computer and signal processing software, by special devices (which are usually called a Fourier analyzer), or by hardware microchips (DSPs). More commonly used in analyzers now is a more efficient mathematical routine, the fast Fourier transform (FFT).

The time wave form of a vibration signal is converted into distinct sinusoidal components as a function of frequency by means of the FFT, as shown in Figure 16. There are several important basic factors to take into account when setting up an FFT analyzer to convert a time wave form into a meaningful frequency spectrum. There is a relationship between the bandwidth of the frequency lines (or bins), the frequency span and the length of the time trace. In Figure 16, the bandwidth is 2 Hz, which is

100 frequency lines between 0 Hz and 200 Hz. These parameters should be chosen to optimize the frequency range of interest.

Due to the effects of aliasing (see 4.3.7), higher frequency components can be falsely identified as lower frequency. Anti-aliasing filters should be used in order to avoid this possibility.

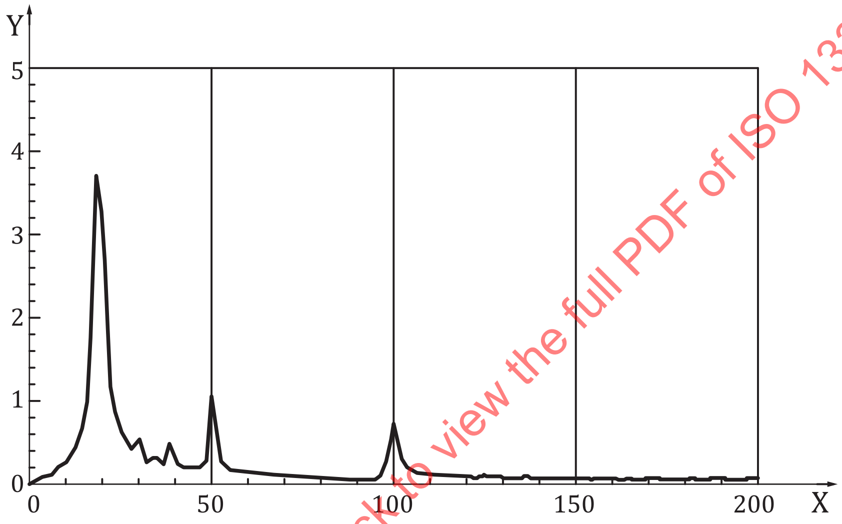
The result of a Fourier transform is a complex spectrum, which can be displayed as

— amplitude and phase, or

— real and imaginary part

of each frequency component.

From a practical viewpoint, the amplitude spectrum (magnitude spectrum) has more information; therefore, the phase spectrum is mostly ignored.



Key

X frequency, Hz

Y amplitude (arbitrary unit)

Figure 16 — Amplitude frequency spectrum

4.3.3 Leakage and windowing

When sampling a waveform, leakage can occur if the sample contains a non-integral number of cycles. The result is smearing of frequency domain peaks because the sample inaccurately represents the waveform from which it was taken. A window function reduces these errors by correcting for this leakage. The Hanning window does an acceptable job for sine waves that are periodic and non-periodic as time records. Although the Hanning window is most commonly used, there are other types of windows that are available and can be used to enhance the signal.

For transient events a Uniform (rectangular) window produces better results. The Hamming window gives a narrower spectral peak than the Hanning window at higher levels, in exchange for flaring skirts further down. The Blackman window, and its derivatives Blackman Exact and Blackman Harris, give a wider peak than the Hanning, but with even lower skirts. The Flat Top window can improve amplitude accuracy over the Hanning window at the expense of being able to resolve small signals that are closely spaced to large ones in the frequency domain. It gives the widest peak, with skirts equivalent to the Hanning, but the top of the peak is flattest for the most accurate level readings with changing frequencies. By correcting sampling bias, windows improve asynchronous waveform plots such as spectrum, cascade and waterfall. The Flat Top window can also be used for calibration.

NOTE Time domain windows for Fourier Transform analysis are described in ISO 18431-2.

4.3.4 Frequency resolution

The mathematics of an FFT requires that the frequency span of interest be divided into a finite number of sections, and the amplitude of vibration within each section is displayed as a vertical line, sometimes referred to as a “baseband” spectrum. The number of sections is referred to as the number of lines of resolution (LOR), N_{LOR}. There can be more than one frequency component at frequencies within a single LOR bin, and the analyzer includes this total energy and displays it as a single line at the centre frequency of the bin.

It is important to have a sufficient number of LOR to distinguish between closely spaced frequency components, and to use a frequency span that includes all frequencies of interest. Normally, at least 400 LOR are used, but many machines require finer resolution than that. The following relationship applies:

$$N_{\mathrm{LOR}} = f_{\mathrm{max}}/B \tag{11}$$

where

N_{LOR} is the number of lines of resolution;

f_{max} is the maximum frequency of interest;

B is the bandwidth (line spacing).

As the relationship shows, for the same frequency range of interest, the finer the resolution, the smaller the bandwidth.

4.3.5 Record length

A single realization of the Fourier transform requires only a short record length, T, and the length of record required for an FFT is dependent on the bandwidth, B, as follows:

$$T = 1/B \tag{12}$$

The length of record available can restrict the resolution. As an example, if a spectrum has a span of 100 Hz and a resolution of 400 lines, the bandwidth shall be 1/4 Hz, and the record length shall be at least 4 s. For the same resolution, if the span increases by a certain factor, the record length decreases by the same factor, and the bandwidth becomes wider by the same factor.

If a machine changes rotational speed slightly during a test, it is important to have the bins wide enough to include each frequency component of interest in a single bin. In the case of a large change in the machine speed, the signal needs to be sampled at constant angular intervals and the succeeding ordering spectrum be processed (see 4.2.11 and 4.3.8).

4.3.6 Amplitude modulation (sidebands)

Amplitude modulation as it is seen in the time domain is shown in 4.2.3. An FFT of a modulating sine wave will show the sine wave’s frequency and sidebands on either side of that frequency, at a distance from it which is equal to the modulating frequency. If the modulation is itself a sine wave, the sidebands will be distinct and only one will appear on each side of the main frequency. This can occur with a gear mesh frequency when one of the gears is eccentric or worn. If the modulation is periodic, such as once per revolution, but not sinusoidal, there will be several distinct sidebands. If the modulation is not periodic, the sidebands will be smeared and indistinct.

The presence of sidebands can be very helpful in the detection of broken rotor bars in large induction motors by measuring the decibel down values. The decibel down, L_D, is equal to 20 times the logarithm of the ratio of the rotor bar fault peak value to the line frequency value. Mathematically, this relationship is:

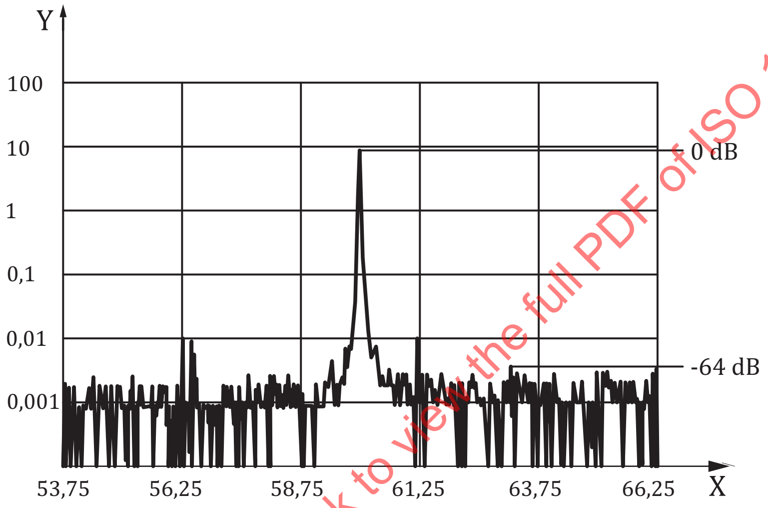
$$L_D = 20 \lg(l_1/l_{ref}) \text{ dB} \tag{13}$$

where

l_1 is the amplitude of the sideband;

l_{ref} is the amplitude at line frequency (50 Hz or 60 Hz).

As shown in Figure 17, the spectrum for a motor with no problems consists of a clear peak at the line frequency and sidebands equally spaced on either side. The magnitude of the sidebands can be over 60 dB down from the magnitude of the line frequency (60 Hz in this case).



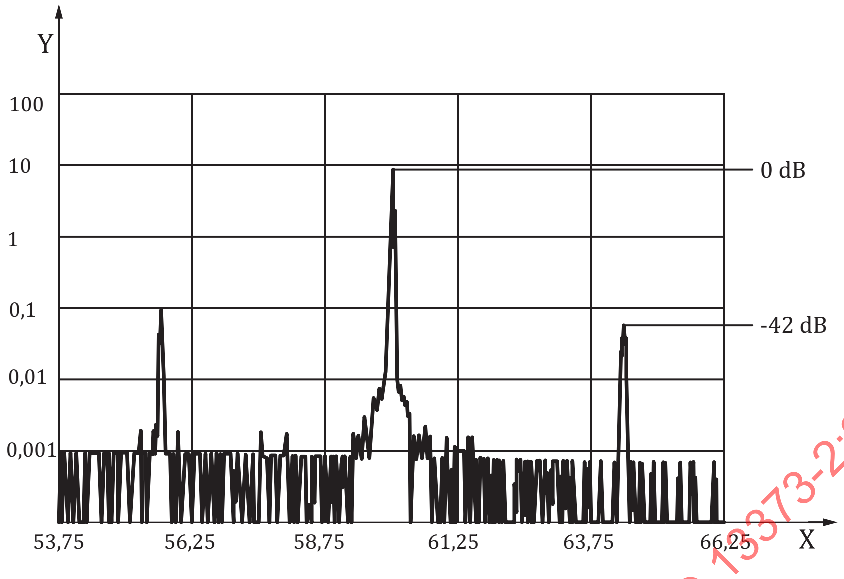
Key

X frequency, Hz

Y vibration magnitude (arbitrary unit)

Figure 17 — Motor with no problems

Figure 18 is the spectrum for a motor with a fault. In this case, there is a distinct peak at line frequency and elevated sidebands at the rotor bar fault frequencies.



Key

X frequency, Hz

Y vibration magnitude (arbitrary unit)

Figure 18 — Motor with a fault

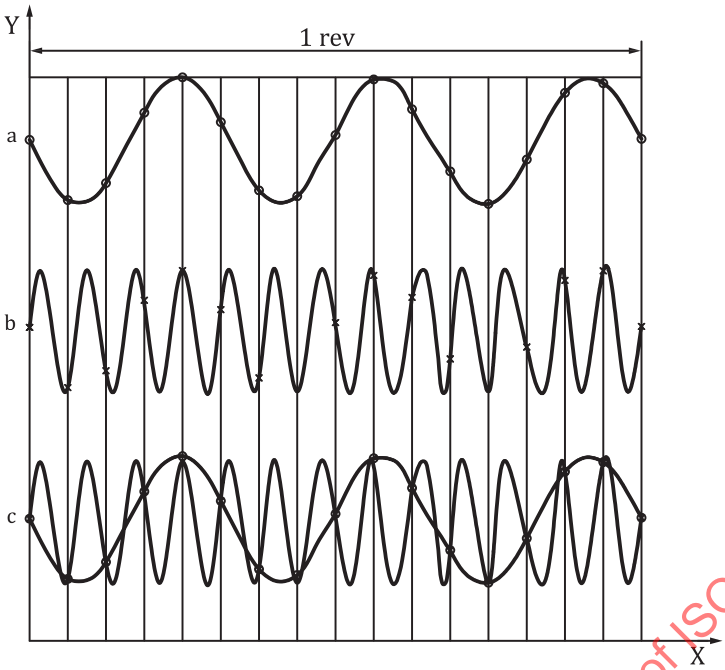
It can be noted that the structure of the sidebands in the frequency domain has the same information as the envelope spectrum in the time domain.

4.3.7 Aliasing

Aliasing is a false representation of a frequency that can result when the sampling rate of a digital analyzer is too low to describe that frequency adequately. It is much the same as when a point on a disc appears fixed if the sampling frequency from a strobe is exactly coincident with the disc rotational frequency. However, if the frequencies are not exactly synchronized, the disc will appear to be rotating slowly. Similarly, if a sine wave is sampled too slowly, it will appear to be a lower frequency. This is eliminated by low-pass filtering the signal before sampling to ensure that it contains no frequency components above half the sampling frequency. This is clearly presented in Figure 19. By comparing the frequency of the high-frequency sine wave and the sampling interval, it can be shown that the sampling frequency is lower than one-half the signal frequency. Therefore, the low-frequency signal will be analyzed as an aliasing signal instead of the actually measured one. This is why the sampled amplitudes (the marked joint points in both curves) correspond to the measured high-frequency signal, as well as to the low-frequency aliasing signal.

When the sampling rate is set exactly at two times the maximum expected frequency, this is known as the Nyquist frequency. In practice, most sampling rates are set at greater than two times the maximum frequency (about 2,56 times) to allow for a low-pass filter without a sharp cut-off.

Digital analyzers today use anti-aliasing filters that remove all frequencies above 40 % of the sampling rate, before the time data are sampled and converted to digital data; therefore, with most digital analyzers, aliasing is no longer a problem. However, the analyst should confirm this before analyzing the data.



Key

X time

Y excitation

a 3/rev excitation

b 13/rev excitation

c 3 and 13 excitation

Figure 19 — Aliasing

4.3.8 Synchronous sampling

Rather than sampling at a fixed rate with respect to time, an external signal can be used in many analyzers to control the sampling rate. Normally, the sampling rate will be some multiple of the external signal frequency. This is most often used with rotating machinery, where a revolution marker is used to determine the sample rate. The sample rate should be greater than two times the highest-order vibration of interest. There are four major advantages to this procedure, as follows.

a) If the rotational speed of the machine changes, most frequency components which are related to the rotational frequency (blade, vane, gear mesh, etc.) will stay in the same frequency bin, rather than spreading the energy over more than one bin.

b) All orders of vibration are in the centre of a frequency bin where its amplitude is measured more accurately.

c) It is possible to average the series of digitized measuring values without consideration of changes in the rotational speed.

d) All orders of vibration will maintain the same phase angle with respect to the external signal. This means that the spectra can be averaged vectorially, reinforcing the pertinent orders of vibration, but causing other signals not associated with that rotational speed, including most noise signals, to average to zero.

The result of the Fourier transform of a synchronously sampled signal is the ordering spectrum $X(n)$. The order $n = 1$ corresponds with one vibration period per one rotor revolution.

It is noted that digital order tracking is an approach used in practice (see 4.6).

When performing synchronous averaging, care should also be taken to avoid averaging out any non-harmonic signals of significance (e.g. bearing instability).

4.3.9 Spectrum averaging

Depending on the component frequencies of the signal, a single FFT requires only a fraction of a second or a few seconds of record. However, a modulating signal can require a longer time period to establish a stable average amplitude. Therefore, averaging successive FFTs is a very important function of analyzers. If only one channel is available, the absolute amplitudes in each bin are averaged without regard to the phase. Averaging of the complete spectrum (real and imaginary part) requires a synchronization of each successive spectrum by a process-dependent trigger signal.

There are other averaging techniques that can be applied, such as frequency domain averaging, but this technique quickly becomes very complicated and is therefore used only for special applications.

Many analyzers do, however, perform exponential averaging, which weights the FFTs with an exponentially increasing function, thereby weighting the signal in favour of the most recently recorded data. This technique is often used for studies of transient vibration in which the amplitudes are exponentially decreasing.

Another type of averaging found on analyzers is peak averaging. This finds the maximum amplitude during a given time period of all the FFTs in each of the frequency bins and displays those peaks. Note that each peak is the average amplitude within its own time record.

4.3.10 Logarithmic plots (with dB references)

With vibration records, there are usually many frequency components with greatly varying amplitudes. Many of the components with small amplitudes are important but, when plotted on a linear scale, can hardly be seen. A logarithmic plot, which compresses the large components and enlarges the small, shows all the significant components, as well as the level of noise present. The amplitude, X, is plotted as level, L, in decibels:

$$L = 20 \lg(X/X_{ref}) \text{ dB} \tag{14}$$

where

X_{ref} is a reference value.

Sometimes, the frequency axis is also displayed in a logarithmic scale for better recognition, or for separation of the low-frequency components. On the abscissa, the decibel unit is not used (see 4.7).

Differences in decibels are equivalent to ratios, examples of which are shown in Table 1.

Table 1 — Differences in decibels and equivalent ratios

Difference dB	**Ratio**
0	1
6	2
20	10
26	20
40	100
60	1 000

Ratios smaller than one are reflected by negative decibel values, e.g. a ratio of 1/2 is −6 dB.

Reference values for logarithmic levels are specified in ISO 1683. For vibration analyses, the values given in Table 2 should be used.

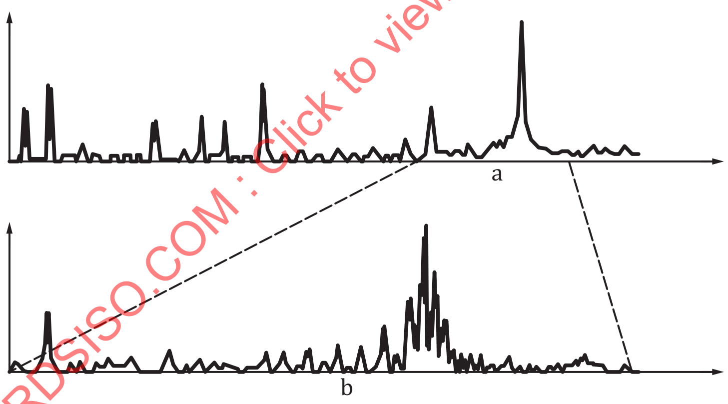
Table 2 — Reference values for logarithmic levels

Quantity	Reference value
Acceleration	10^{-6} m/s^2
Velocity	10^{-9} m/s
Displacement	10^{-12} m
Power	10^{-12} W

4.3.11 Zoom analysis

Often, frequency components are too close together to distinguish between them on a normal FFT, which generally consists of 400 lines (base band); however, others exist. Some analyzers have higher resolution, but often zoom spectra are used to get better resolution. A zoom analysis creates a spectrum with a frequency scale that does not start at zero but at another free eligible frequency, so that the selected number of lines are utilized to expand the frequency range of interest. The bandwidth is correspondingly narrower; however, the record length will still be related to the bandwidth. One problem in using the zoom spectra is that the frequencies must be more stable because of the narrower bandwidth.

An example of the use of zoom spectra is in gear fault analysis. When applied, a fault will result in sidebands of the gear mesh frequency and the spacing of the sidebands will indicate the faulted wheel. A similar zoom approach can also be useful to identify faults in rolling element bearings. Figure 20 shows the advantages of performing zoom analysis. Note that the frequency components not visible in the original zoomed spectrum are now visible.



Key

a section of original spectrum

b higher resolution translated spectrum

Figure 20 — Zoom analysis

4.3.12 Differentiation and integration

Differentiation and integration are important in vibration analysis when signals shall be converted between displacement, velocity and acceleration. For rotating machinery, the vibration signal is often dominated by the synchronous component, and can therefore be harmonic motion. The corresponding formulae then take on the following appearance in the time domain (see also 3.3.2):

displacement:

$$x = \hat{x} \cdot \sin\omega t \tag{15}$$

velocity:

$$v = \omega\hat{x} \cdot \cos\omega t = \hat{v} \cdot \cos\omega t \tag{16}$$

acceleration:

$$a = -\omega^2\hat{x} \cdot \sin\omega t = -\omega\hat{v} \cdot \sin\omega t = -\hat{a} \cdot \sin\omega t \tag{17}$$

and

acceleration:

$$a = \hat{a} \cdot \sin\omega t \tag{18}$$

velocity:

$$v = -\frac{\hat{a}}{\omega} \cdot \cos\omega t = -\hat{v} \cdot \cos\omega t \tag{19}$$

displacement:

$$x = -\frac{\hat{a}}{\omega^2} \cdot \sin\omega t = -\frac{\hat{v}}{\omega} \cdot \sin\omega t = -\hat{x} \cdot \sin\omega t \tag{20}$$

The displacement lags the velocity by 90° and the velocity lags the acceleration by 90°. To convert between quantities in the frequency domain, both differentiation and integration can be carried out by dividing or multiplying, respectively, each component by its angular frequency. Most analyzers include these functions for the frequency domain.

It is stressed that to utilize accurately the integration and differentiation formulae, the vibration signal must be predominantly synchronous. It is necessary to check if the 1× component is greater than 90 % of the unfiltered, or direct, signal. Otherwise, each spectral frequency shall be converted separately.

4.4 Display of results during operational changes

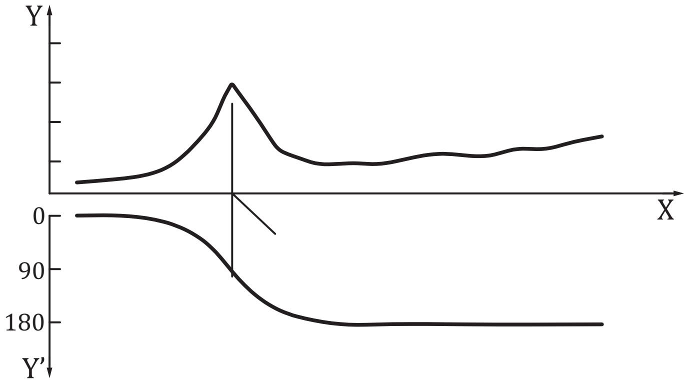
4.4.1 Amplitude and phase (Bode plot)

When a harmonic vibration signal is expressed in terms of an amplitude and phase, a second signal is required as a reference for the phase. It can be a shaft revolution marker, the vibration at a different location or direction, a measured force or some other appropriate reference. The frequency(ies) of the second signal shall be considered in relation to the frequencies of interest. For example, a shaft revolution marker could be used as a phase reference for rotational frequency or any of the higher harmonics of rotational frequency.

The phase may be expressed as between 0° and 360°, or ± 180°.

When the two signals represent different quantities (e.g. force, velocity, acceleration), care shall be taken to interpret the physical significance properly. Note that, for any sine wave, the displacement lags the velocity by 90°, and the velocity lags the acceleration by 90°. Very often, signal-conditioning equipment changes the phases of the signals, and differences between channels shall be compensated for.

The amplitude and phase of a sine wave can be plotted as a function of time. However, when the amplitude and phase of a machine vibration is plotted against the machine rotational speed, it becomes a Bode plot, as shown in Figure 21.



Key

X rotational speed

Y amplitude

Y' phase, degrees

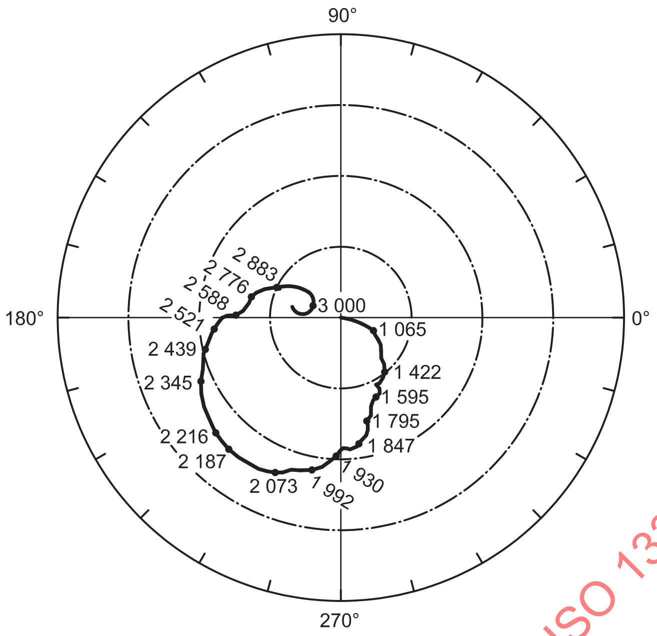
[a] resonance

Figure 21 — Amplitude and phase (Bode plot)

4.4.2 Polar diagram (Nyquist diagram)

In a polar diagram, each point represents an amplitude/phase vector for a discrete frequency as shown in Figure 22. If the diagram includes several vectors for different rotational speeds, or other parameters, by showing only the connecting line between their tips, it is known as a Nyquist diagram.

A polar diagram shall have a phase reference, such as a shaft revolution marker, that indicates each 360° rotation of the shaft. Polar diagram (and/or Bode plots) are used to identify accurately the location (rotational speed) of any resonances of the rotor/bearing/support system.



NOTE The parameter is the rotor rotational speed (r/min).

Figure 22 — Polar diagram (Nyquist diagram)

4.4.3 Cascade (waterfall) diagram

The cascade or waterfall diagram provides a simple comparison of several frequency analyses. It is a three-dimensional form of spectra display that clearly shows vibration signal changes related to another parameter (such as rotational speed, load, temperature, time) taken for specified parameter values, such as time.

The sample cascade spectrum of Figure 23 is an overall picture of many vibration spectra for a machine in the start-up or coast-down region. Normally, the cascade spectrum display provides frequency (Hz or orders) versus machine rotational speed and vibration amplitude of the discrete frequency components. In some cases, however, the machine speed may be substituted by another variable (e.g. time, load), in which case it is then called a waterfall diagram. When using machine speed for this display, it is necessary to record a rotor speed/phase reference signal.

The cascade spectrum of Figure 24 shows the fundamental rotor speed (1×) and any other significant harmonic. It also shows the presence of rotor resonance speeds, if in the transient speed range.

The shape of the plot will vary, depending on the type of machine and the operation. For example, Figure 24 is a cascade plot of a 3 000 r/min (50 Hz) steam turbine during start-up and coast-down.