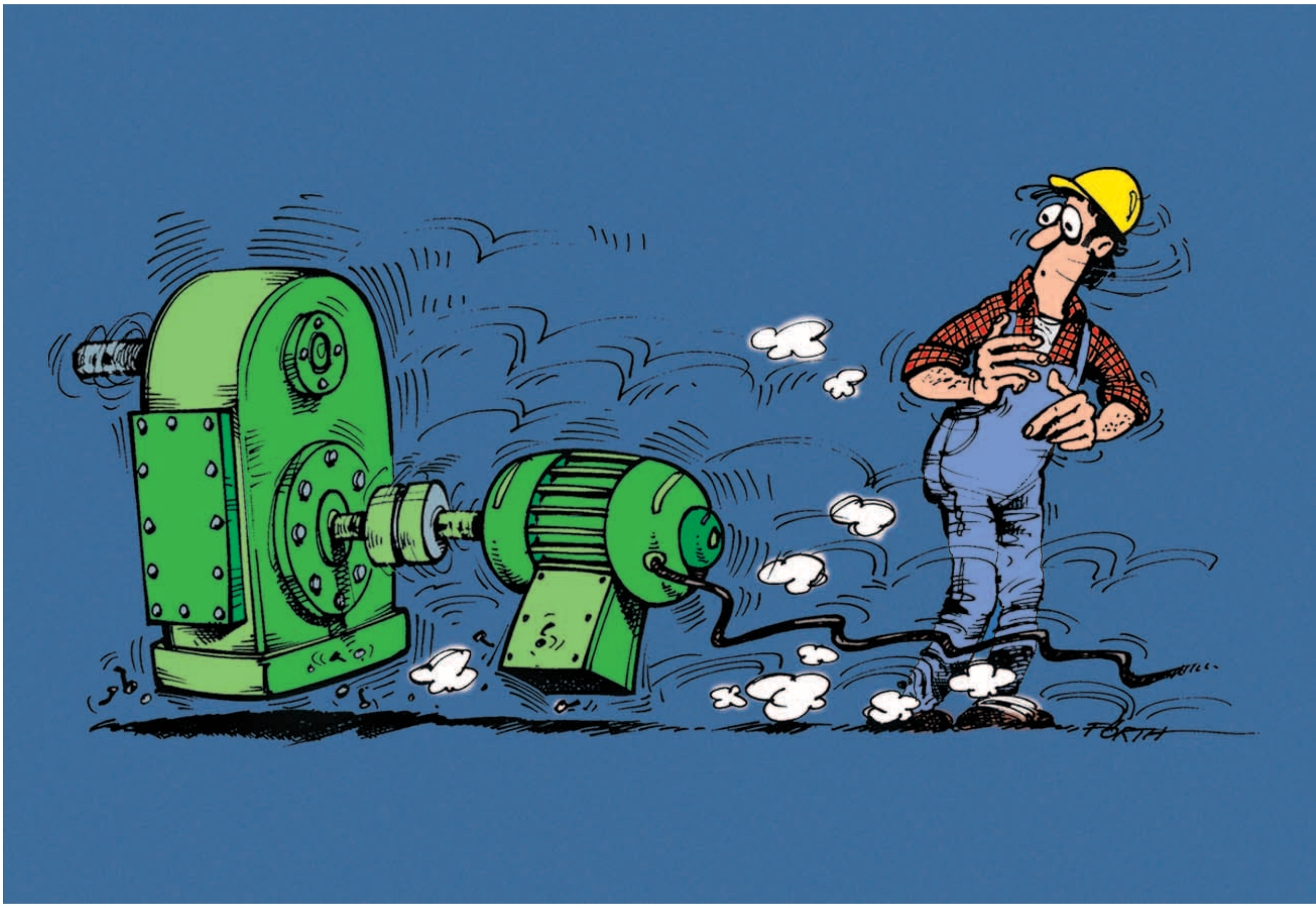
Vibration Handbook

Fundamentals – Transducers – Measurement methods



Foreword

Maintenance plays an increasingly important economic role in modern industry, as managers realize that the key to retaining long-term investment capital value lies in regular service and inspection. In addition, conscientious maintenance practices help avoid unplanned machine stoppage and the resulting production downtime while improving production quality and plant safety.

The German DIN Standard 31 051 defines maintenance as ‘all measures for retention and re-creation of the specification condition as well as for determining and evaluating the actual condition of technical aspects of a system.’ Vibration measurement is one of the primary measurement methods for determining actual machine condition and allows a comprehensive evaluation through the potential of Fourier transformation.

PRÜFTECHNIK has a long history of putting innovative technological ideas into industrial maintenance practice - with the practical advantage for the user always foremost in mind. This guiding principle was first confirmed by OPTALIGN, the world's first laser shaft alignment system.

This handbook is intended to provide you, the user, with a solid basis of vibration knowledge and a useful reference for putting this knowledge into practice. We wish you much enjoyment and success in doing so.

PRÜFTECHNIK AG

Vibration Handbook
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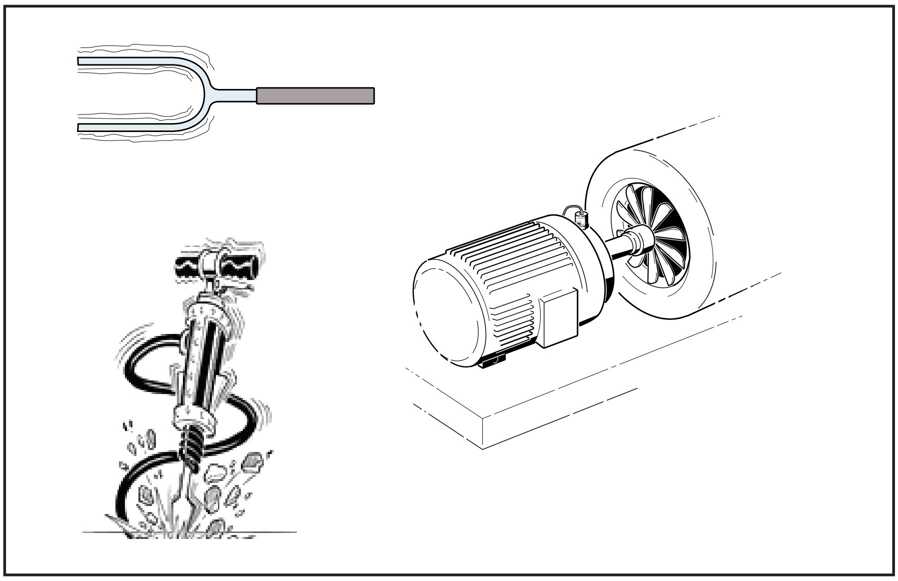
1. Introduction to vibration measurement

Vibration can occur in a multitude of forms. Whether they be desirable (in the form of a tone emitted by a tuning fork) or undesirable (such as the noise and vibration of a jackhammer or the characteristic jolt made by a refrigerator starting up), vibrations are a part of everyday life.



Examples of vibration

In industrial environments, the study of vibrations and their causes and remedies is very important to ensure the smooth operation and long operating life of critical production aggregates. Mechanical vibration serves as an excellent indicator of machine condition, because it includes the influences of dynamic loading, foundation behavior and all external influences. Most rotating equipment vibration occurs due to rotor imbalance or insufficient shaft alignment.
The following pages offer a brief introduction to appropriate theoretical fundamentals, which aid in understanding the subsequent discussion of measurement processes and analysis. This introduction has purposely been kept simple for the novice to the field of vibration measurement; the bibliography lists sources for more in-depth study of these topics.

1.1 Periodic vibration in the time domain

Vibration is generally considered to be the movement of a mass or a body about its position at rest. Periodic vibration occurs when the vibration repeats itself at regular intervals of time (e.g. tuning fork).

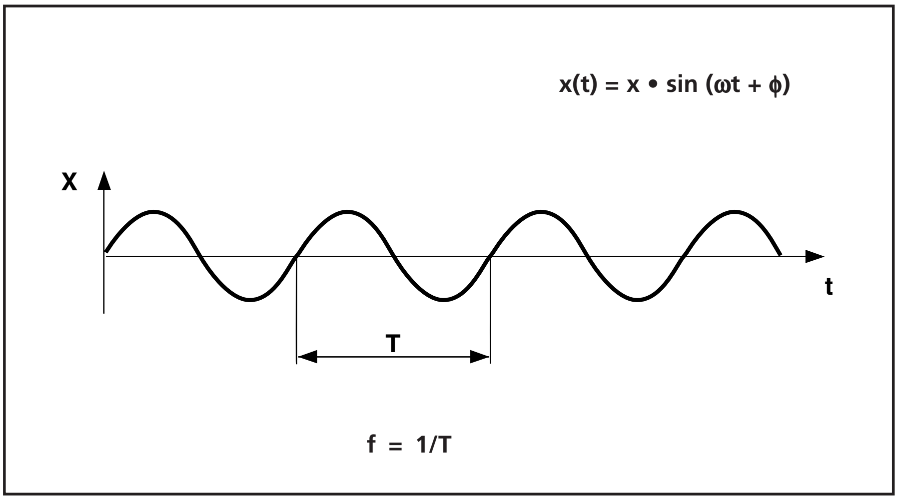


Fig. 1 Periodic vibration

The time length of a single vibration cycle is known as the vibration period T. The reciprocal 1/T is known as the vibration frequency and is expressed in cycles per second, or Hertz (Hz). The following sine function describes the undamped periodic vibration shown above:

$x(t) = \hat{x} \bullet \sin(\omega t + \phi)$;
$x(t)$: Displacement, instantaneous vibration value
$\hat{x}$: Amplitude, maximum instantaneous value
ω: Angular frequency; $\omega = 2 \bullet \pi / T = 2 \bullet \pi \bullet f$
ϕ: Phase angle, initial position of the oscillation at $t = 0$.

The equation indicates the displacement - the vibration - as a function of time.

The fundamental physical source of this vibration and of the recorded signal could be thought of, for example, as follows:

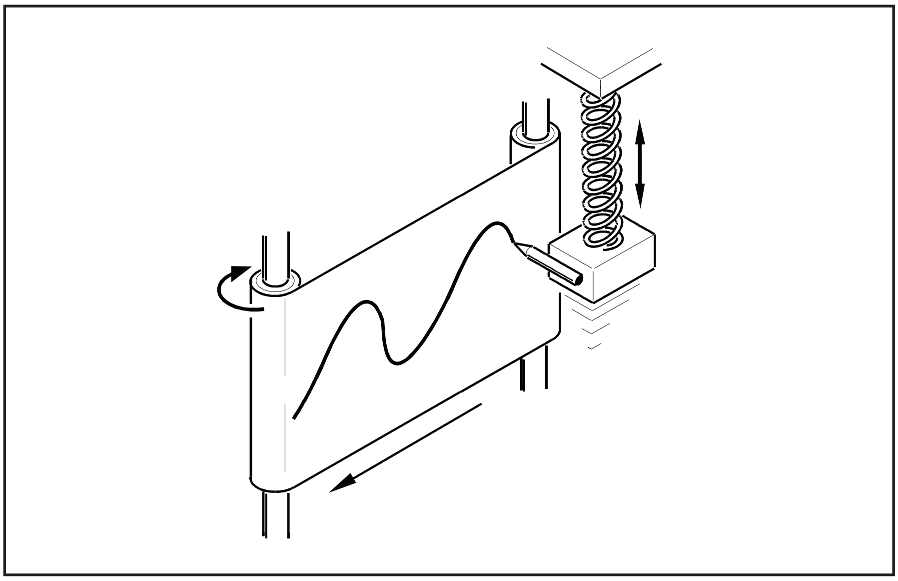


Fig. 2 Mass on spring pendulum

The vibration velocity v is the change in displacement x over a period of time. Its waveform is shifted by 90° from the position of the displacement, i.e. velocity reaches its minimum (v=0) when amplitude is greatest. The velocity attains its maximum when the system oscillates through its zero level:

$$v(t) = x \bullet \omega \bullet \cos(\omega t + \phi);$$

The vibration acceleration a is the change in velocity over time. From a mathematical point of view, it is the second derivative of displacement with respect to time. The acceleration waveform is shifted by 90° from the corresponding velocity wave (and therefore by 180° from the displacement curve). This means that acceleration reaches its maximum when displacement is at its minimum; both pass the zero point at the same time.

$$a(t) = -x \bullet \omega^2 \bullet \sin(\omega t + \phi);$$

The displacement amplitude alone may be considered as follows:

Vibration displacement	x in mm
Vibration velocity	$v = x \bullet \omega$ in mm/s
Vibration acceleration	$a = x \bullet \omega^2$ in mm/s^2

At mid-range frequencies, vibration is normally measured in terms of velocity; high frequencies call for acceleration measurement. These deliver the best possible dynamic performance.

1.1.1 Composite vibrations

When several periodic vibrations occur simultaneously, a com-

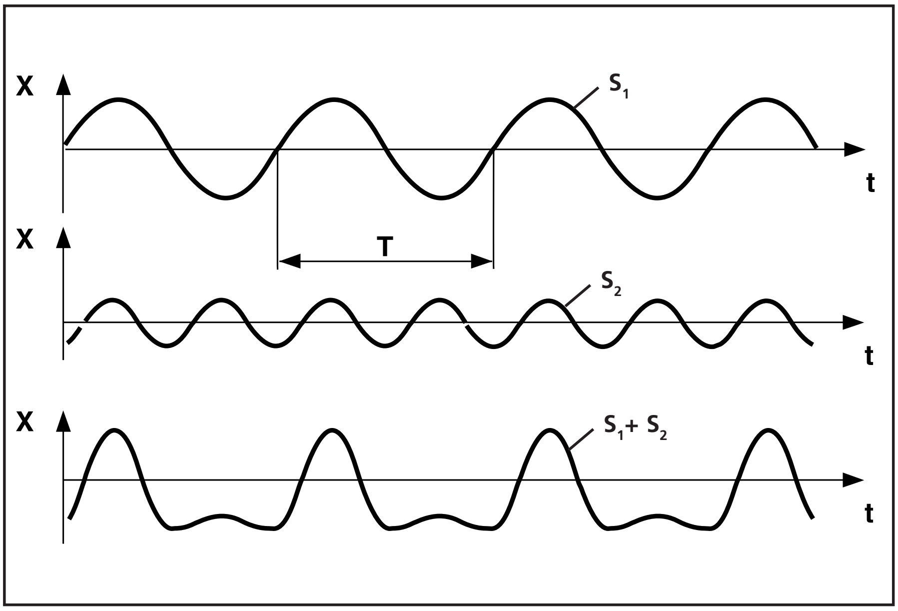


Fig. 3 Superimposed sine waves

posite vibration results which is also periodic. This type of complex vibration is quite normal for practically all machinery.

When vibrations overlap, the time domain depiction (with amplitude plotted along the time axis) cannot show individual wave components of the composite signal.

1.1.2 Vibration parameters

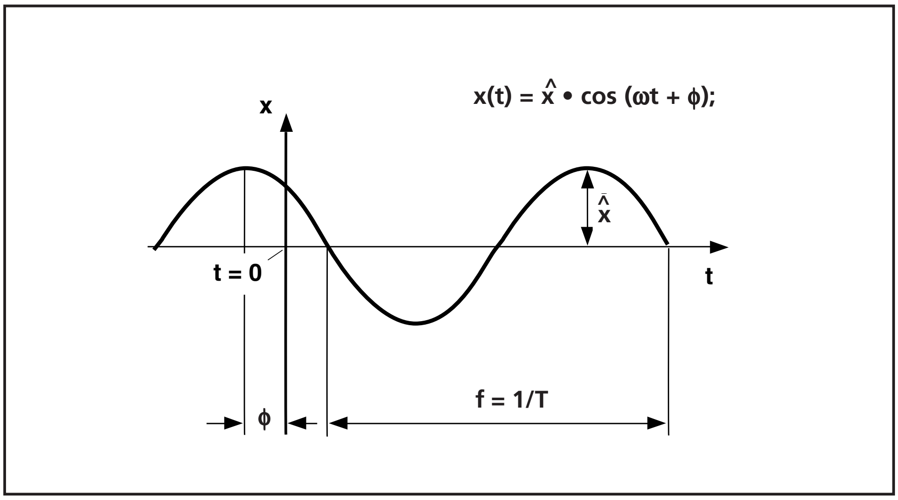


Fig. 4 Parameters of a sine wave

Amplitude, frequency and phase angle determine the exact shape of a sine wave.

The vibration frequency is important information for machine diagnosis, because certain causes of vibration - for example, rotor imbalance - occur at fixed multiples of the speed (frequency) at which massive components rotate.
The phase angle ϕ describes the starting position of the oscillating point at time $t = 0$. This phase angle and the phase difference between two measurement locations is extremely important for balancing and vibration diagnosis.

Standard vibration parameters

Peak-to-peak value:
Difference between the largest and smallest vibration value, abbreviated as x_{pp}

Peak value:
Maximum vibration (also known as amplitude), abbreviated as x_m.

Median value:
Quantifies the vibration strength over time:

$$\text{Definition:} \quad x_{Med} = \frac{1}{T} \quad x\,dt;$$

$$\text{For a sine wave: } x_{Med} = \frac{2}{\pi}\, x_m$$

Effective value:
Quantifies the energy of vibration over time (also known as: RMS = root-mean-square).

$$\text{Definition: } x_{eff} = \sqrt{\frac{1}{T} \quad x^2\,dt}$$

$$\text{For a sine wave: } x_{eff} = \frac{1}{\sqrt{2}}\, x_m$$

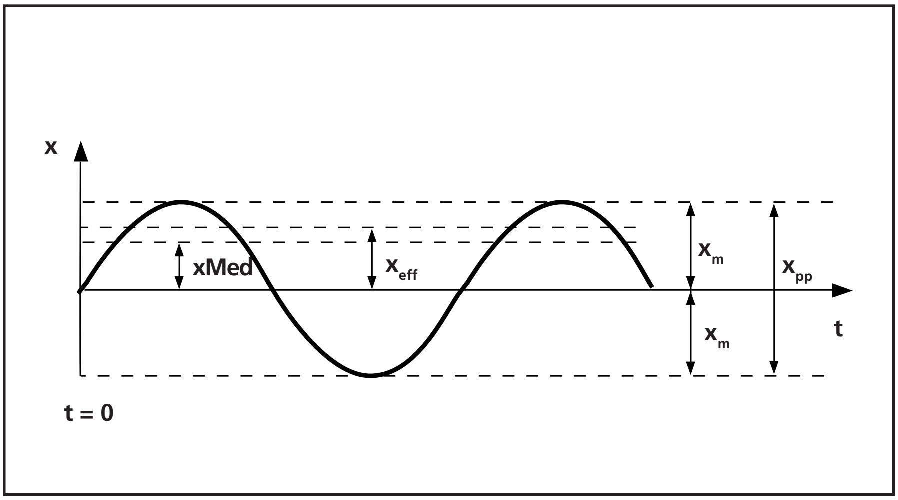


Fig. 5 Parameters of vibration strength measurement

The peak-to-peak value of a vibration is preferable when the maximum displacement is of interest, e.g. for problems related to machine component loading or internal separation distances.
The peak value is best for examination of short-lived phenomena or transient signals.
The effective value (RMS) shows most clearly the energy content of the vibration.

1.1.3 Conversion factors

Conversion factor (from → to)	Peak-to-peak value	Peak value	Median value	Effective value
Peak-to-peak value	**1**	**0.5**	**0.32**	**0.35**
Peak value	**2**	**1**	**0.64**	**0.71**
Median value	**3.14**	**1.57**	**1**	**1.11**
Effective value	**2.83**	**1.41**	**0.90**	**1**

Fig. 6 Conversion factors

1.2 Periodic vibration in the frequency domain

As mentioned earlier, when the vibration signal is composed of several waves occurring simultaneously, the time signal alone does not show much information regarding its individual components. The frequency diagram obtained from the same time signal, however, is generally much more useful because it clearly shows the individual vibrations (frequency components) which make up the complex signal.

1.2.1 Fourier transformation

The French mathematician Jean Baptiste Fourier devised a method for breaking down any waveform into a series of sine wave components, and conversely.
In other words, everyday machine vibrations can be analyzed to determine which frequencies account for what portion of the overall measured vibration. Fourier's formulae supply the mathematical basis for transforming the time signal into the frequency spectrum.

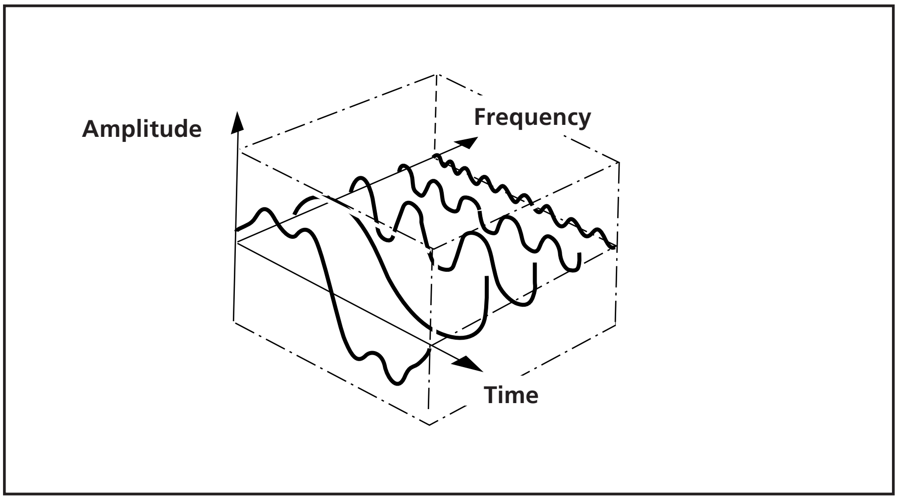


Fig. 7 Vibration in time and frequency domains

The great advantage of displaying the frequency spectrum is that all the individual components of vibration are readily recognizable.

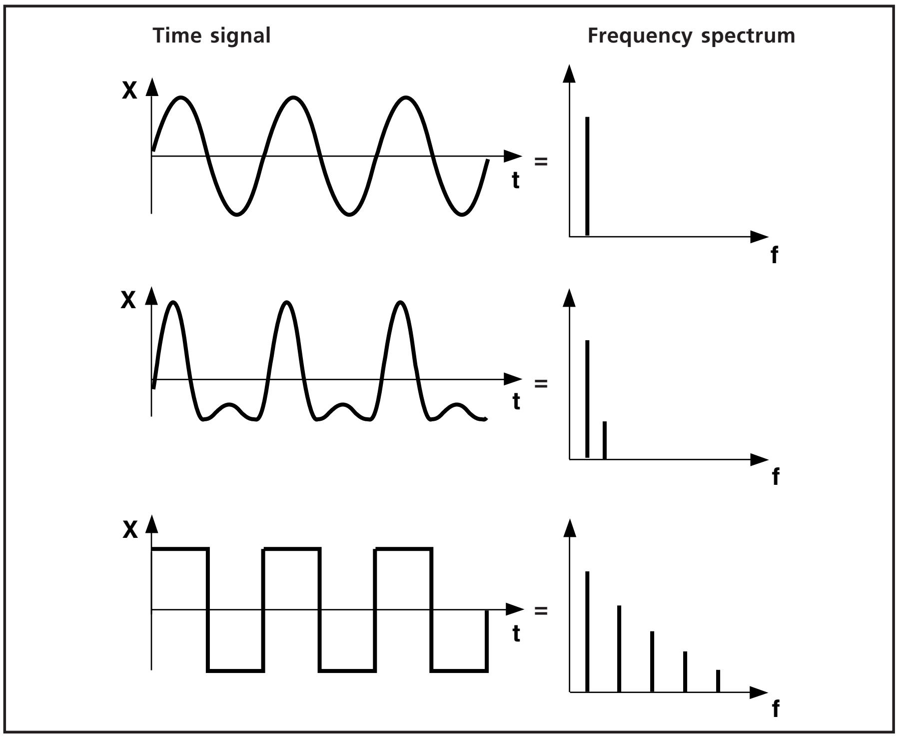


Fig. 8 Time signals and their corresponding frequency spectra

1.2.2 Frequency analysis

The objective of frequency spectrum analysis is to break down a complex vibration signal into its individual frequency components according to their respective power. Experience has shown that the operating condition can be observed very closely, since a machine in good condition exhibits a stable spectrum over extended periods of time.

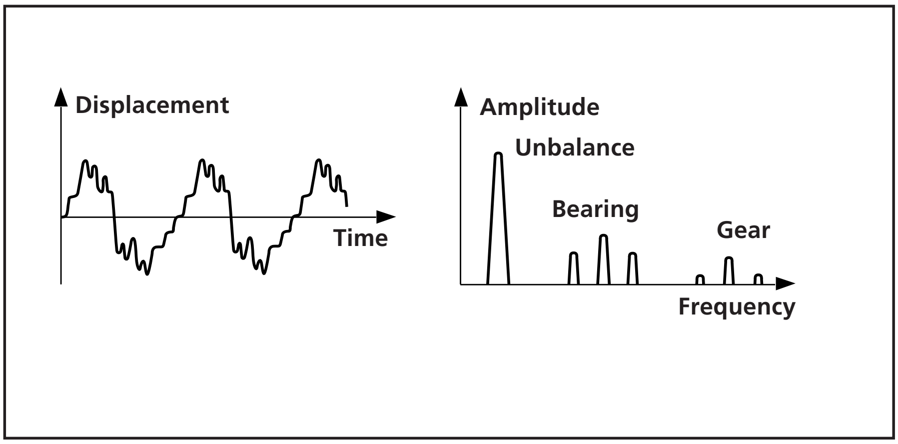


Fig. 9 Comparison of time domain vs. frequency domain

Reliable machine diagnosis is possible when the individual peak frequencies in the vibration spectrum can be traced back to their corresponding machine parts.

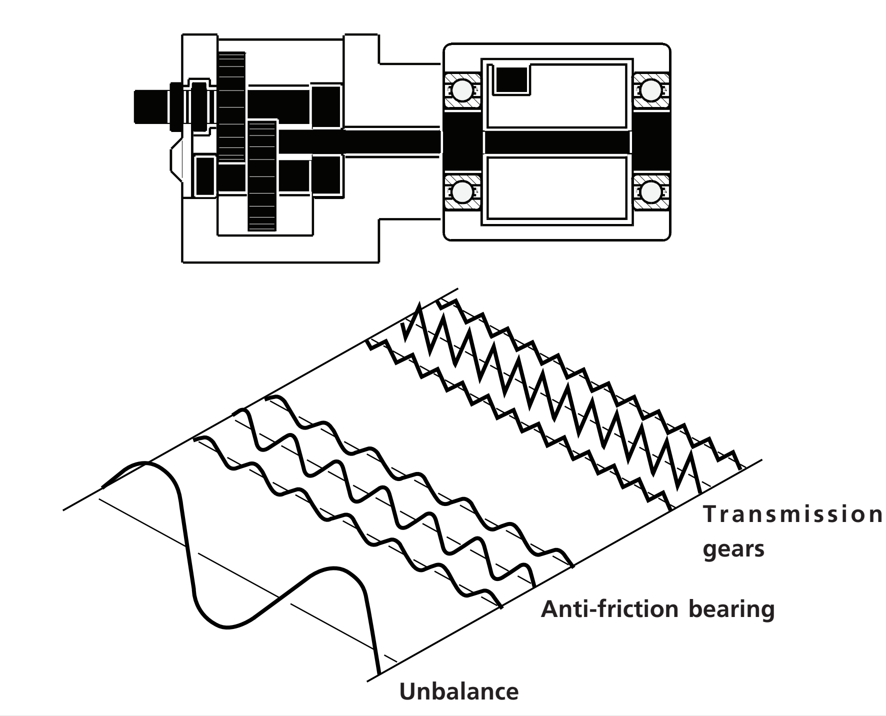


Fig. 10 Machine vibrations

When measuring vibration, you should also consider that the signal measured on the outer surface of the machine housing may not coincide with the original 'inner' signal. The reason for this lies in the transmission path.
Example:
If the inner ring of a bearing is damaged, the vibration must be transmitted from the inner ring via the rolling elements. then the signal is transferred from the outer ring outward via the bearing race and corresponding load-bearing housing portion to the measurement location. Here is where the vibration transducer converts the vibration into a measurable quantity - e.g. voltage. Factors such as mass or rigidity of machine parts and the damping caused by individual parts - and, of course, at component interfaces - can affect the output signal. This means that two identical machines mounted on different foundations under otherwise identical conditions can emit different vibration signals. These conditions must be considered when evaluating measurements.

1.3 Rigidity, mass and damping

A turbocompressor, for example, contains a vast number of the components and interfaces just mentioned. Each of these components can in turn be seen as a separate vibration system unto itself. Detailed information on these components (mass-spring systems) and their vibration behavior is extremely beneficial toward accurate machine vibration analysis - and nearly crucial to explaining and understanding phenomena such as resonance. Machines possess three qualities which determine their vibration behavior - that is, their reaction to excitation forces-
Rigidity ("K" in kg/m),
Mass ("m" in kg),
Damping ("c" in kg•s/m).

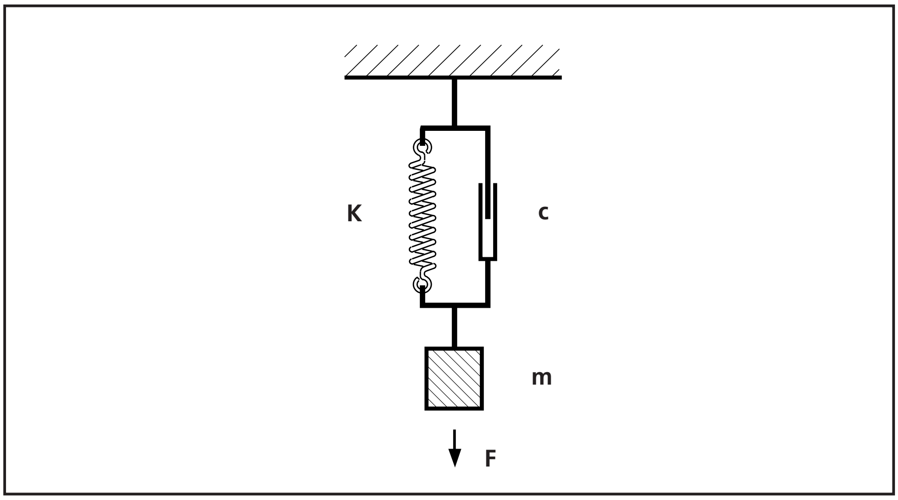


Fig. 11 Mass-spring system

Rigidity is a measure of force needed to bend or to stretch a structure by a certain amount (spring constant). Damping is a measure of the force that works to counteract this.
The relationship can be simplified as follows:
Excitation force = Rigidity 'force' + Mass 'force' + Damping 'force'

In other words: equilibrium always exists between the excitation force and the reaction forces (rigidity, mass and damping).

If any one of the reaction forces is increased, then the vibration amplitude changes accordingly. The value of vibration amplitude is generally dependent on the total influence of the three reaction forces. In practice, however, these do not combine in a linear fashion.
Example: Mass (weight) force is proportional to acceleration. Rigidity, in turn, is proportional to displacement. As a result, these two reaction forces tend to counteract each other, since their phases differ by 180° (just like those of displacement and acceleration.)
The third factor, damping, is proportional to vibration velocity, and its phase is 90° away from those of mass and rigidity.
The vibration frequency must also be considered relative to these opposing forces. While the rigidity remains constant over the entire frequency range, the mass forces increase proportionally to the square of the frequency. At a certain frequency, the mass and rigidity forces are in equilibrium and effectively cancel each other due to their 180° phase difference. The only resistant force remaining is that of damping, and as a result, the entire system vibrates at considerably higher amplitude.

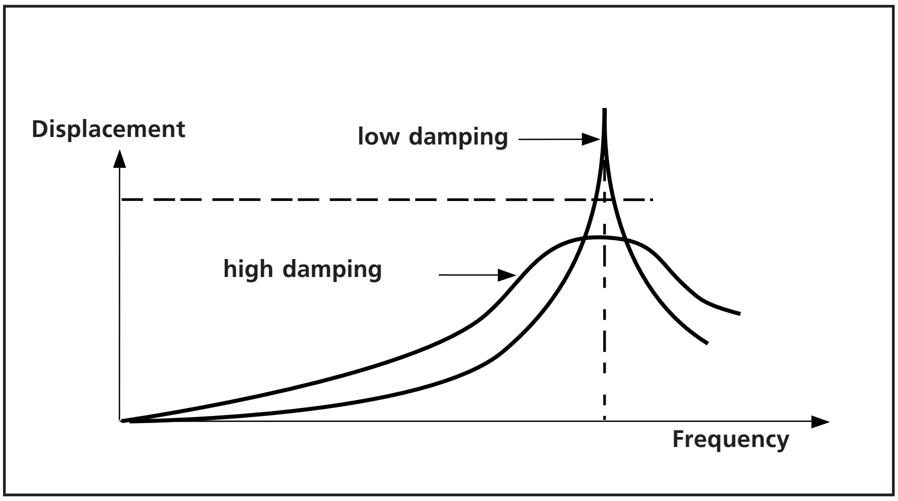


Fig. 12 Resonance behavior

This point is known as the resonance frequency. Here the amplitude can be held low with appropriately high damping.

Another interesting resonance-related phenomenon is the phase shift that occurs between the excitation force and its resultant vibration. The circumstances previously mentioned lead to the conclusion that for frequencies below resonance, rigidity plays the dominant opposing role in phase with the vibration. At higher frequencies, the mass forces take over, with 180° phase difference between the excitation force and the vibration. This means that a 180° phase shift between excitation and vibration takes place as the resonance frequency is passed.

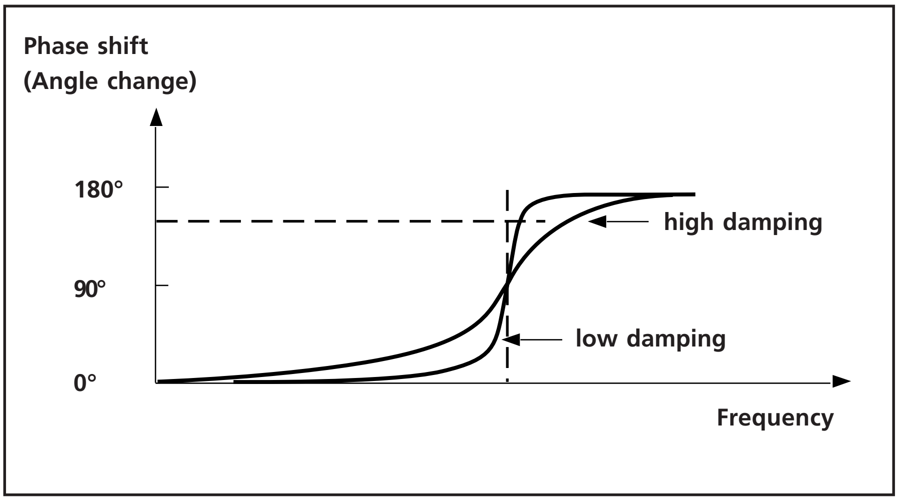


Fig. 13 Phase shift as a function of rotation frequency

The operator or measurement technician must realize that every system - including his favorite vibration transducer, mounting bracket or magnetic fixture - has its own resonance frequency. The resonance characteristics of a vibration transducer mounted with a magnetic fixture can influence high-frequency readings considerably, and the transducer's own resonance frequencies can limit its applicability.
Machines mounted on their foundations must be seen as complex systems composed of many mass-spring units. Each of these individual systems, in turn, has several degrees of freedom and its own resonance frequencies. This means that the same aggregate can exhibit quite different vibration behavior in different operating locations, since the foundation, piping etc. all exert their influence on each individual oscillator.
For example, it is nearly impossible to balance a rotor which operates near its resonance frequency, because then even minor speed deviations cause a phase shift of 50° or 60°. This multitude of potential oscillators, combined with a wide range of possible excitation sources, is quite a common situation in rotating equipment maintenance.

1.4 Summary

Before vibration readings are taken, the rotation frequency and the frequency range of expected vibration should be ascertained. Any variable load characteristics such as pressure, flow rate, temperature etc. (all of which lead to different results) must be considered as well.
A general evaluation of the vibration level can be made by comparing the measured level with accepted standards for the same machine type. ISO Standard 2372 defines these limits for the measured RMS velocity v_{eff}. This value describes only the total vibration severity without regarding the actual cause of vibration at the individual characteristic frequencies.
Frequency analysis is the only method of examining a defined source of vibration: with detailed knowledge of the machine design, the individual frequency components of the measured spectrum can be traced to their respective sources, the offending machine parts. This procedure is especially suitable for condition-based monitoring of machines, allowing the maintenance specialist to intervene when necessary.
Displacement transducers deliver the best results at frequencies below approx. 10 Hz due to the limited dynamics of vibration velocity and acceleration within this range.
Accelerometers are recommended for comprehensive examination over a wide frequency range, including high-frequency vibrations above 1000 Hz (e.g. gears).

2. Vibration transducers for various measurement problems

Vibration transducers transform mechanical movements into corresponding electrical signals. Each type of transducer has its own advantages and disadvantages depending upon the area of application. The first step toward successful analysis is therefore selection of the appropriate transducer. This section explains the three most important types of vibration transducers and the application areas in which each delivers the most accurate and repeatable measurements possible.

2.1 Piezoelectric transducer (Accelerometer)

Piezoelectric transducers can be used to measure nearly all mechanical vibrations that occur in maintenance applications, because they provide accurate measurements over a very wide frequency range. Their resonance frequencies lie above the working measurement range, and their design is especially rugged, yet precise. Since they contain no freely-moving parts, these transducers are virtually invulnerable to wear and tear; their long-term stability is excellent. Moreover, they are simple to mount.
Principle of operation:
When a piezoelectric crystal is placed under a mechanical load, the crystal deforms and emits a voltage directly proportional to the degree of loading. This phenomenon is used with Newton's law that 'force equals mass times acceleration'; the transducer's oscillating seismic mass is known exactly, so the force can be measured in terms of the electrical signal emitted by the piezoelectric crystal.

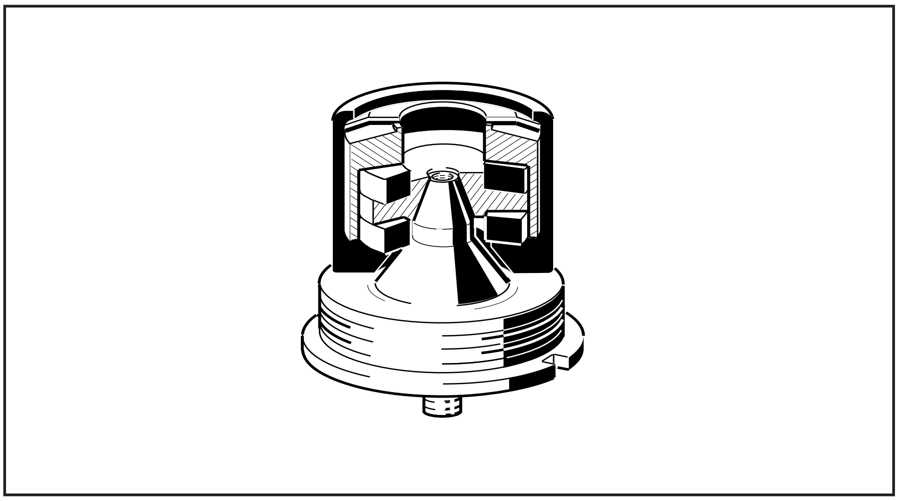


Fig. 14 Tandem-Piezo® transducer

This electrical signal at the transducer output is directly proportional to the acceleration (or force effect) over a very wide frequency range.

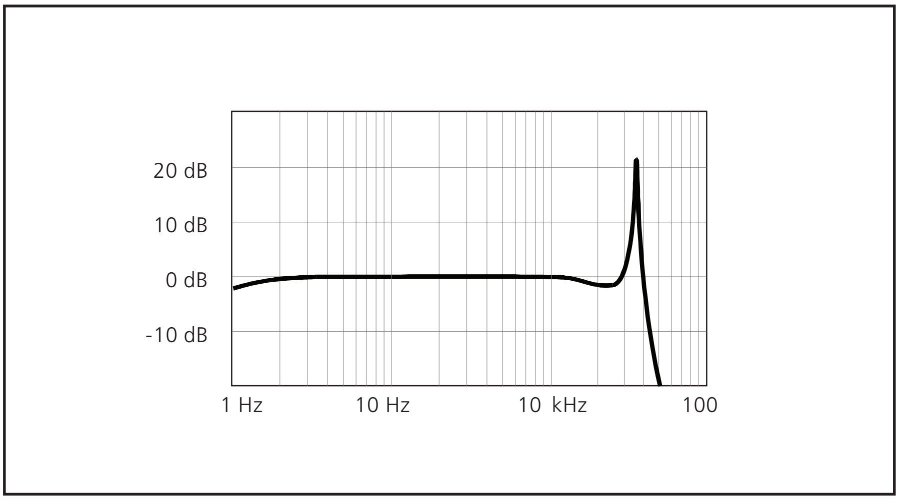


Fig. 15 Accelerometer frequency characteristic

The lower frequency boundary is dependent upon the noise generated by the cable and the preamplifier. The upper boundary depends upon the resonant frequency of the transducer. The range of frequencies to be measured must lie below this resonance frequency. As a rule of thumb, the highest measurement frequency must not exceed 1/3 the resonance frequency, or else distortion may skew the measurement.
Accelerometers typically exhibit low sensitivity and low output levels. This is why a preamplifier often must be used to obtain a usable signal. The Tandem-Piezo® probe shown contains a built-in 'line-drive' stage just for this purpose. Another advantage: no additional amplification is necessary to conduct the measurement signal without distortion, even over long distances.
Accelerometers can easily be used to measure vibration over a frequency range of 10 Hz to 10 kHz, which is sufficient for the vast majority of rotating equipment.

2.2 Induction transducers (velocity probes)

Induction transducers have a very low resonance frequency and are used to measure vibration frequencies above that level.
Operating principle:
There are two main types of velocity probes: with the first, the sensor's permanent magnet is joined via its housing directly to the object to be measured. The magnet delivers a strong, permanent magnetic field and vibrates with the measured object. The coil is suspended so that it can oscillate freely in relative movement about the magnet. The voltage induced in the coil is directly proportional to the actual vibration velocity. (With the second type of probe, the coil remains mounted stationary on the machine housing and the magnet oscillates freely about it.)

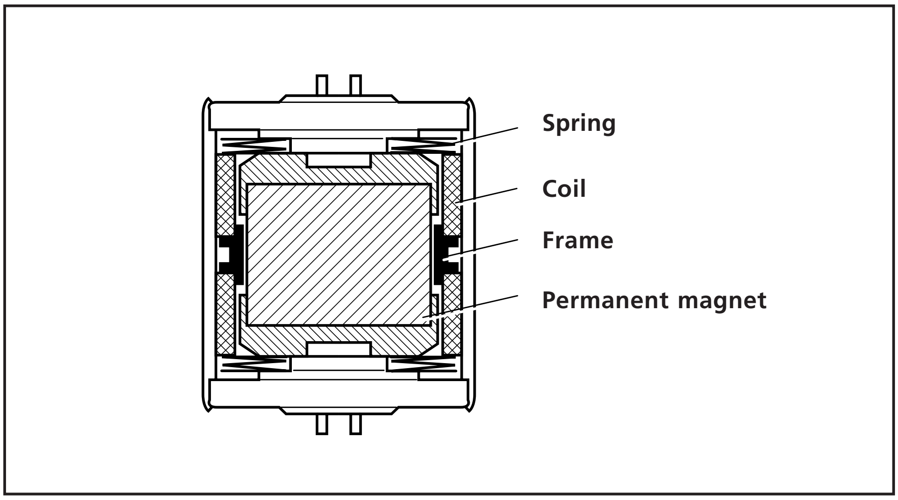


Fig. 16 Induction probe

The usable frequency range of this transducer type is approx. 10 Hz to 2000 Hz. The output signal level is very high, i.e. no further amplification is necessary. However, velocity transducers are sensitive to magnetic field influences and susceptible to wear over time, i.e. the mechanical components (such as springs and suspensions) succumb to fatigue. Transducer size and sensitivity to lateral forces and temperature changes can also become problematic in practical use.

2.3 Proximity probes (Displacement probe)

Large turbines, centrifugal pumps or certain compressors are embedded in heavy, massive machine housings and fixed bearings (often sleeve bearings). Their housings are often so rigid that measurable vibration levels are quite low on the surface, but the shaft can still vibrate severely within the bearing. In that case, the shaft vibration may be measured using permanently-installed induction transducers commonly known as proximity probes.
Operating principle:
This measurement method takes advantage of the eddy currents that flow in an electrically conductive object due to the electromagnetic field created when AC current is passed through a coil wrapped around the object. The input signal passed through the core is changed in direct proportion to the distance (proximity) between the probe tip (coil) and the object to be measured (the shaft, as shown below). This carrier signal modulation can be measured as an AC signal proportional to the peak-to-peak vibration level.

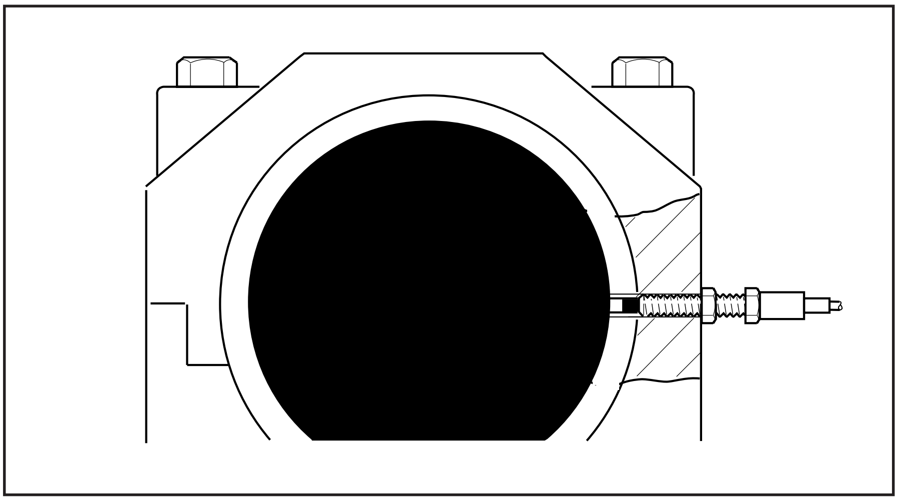


Fig. 17 Proximity probe installation

The usable frequency range extends from 0 Hz to approx. 200 to 300 Hz. Probe installation can often be difficult. The cable length must be matched to the probe by the manufacturer and cannot be varied without recalibration prior to measurement. Proximity probes are used primarily to measure shaft vibration of turbomachinery.

2.4 Transducer mounting

The frequency characteristics and dynamic range of vibration transducers can be influenced by the type and method of mounting. Poor connection leads to excessive damping and to restriction of the frequency range. Moreover, transducer cables should be taped to the machine for safety reasons.
Observe the following points to achieve optimum measurement results.

Threaded attachment:
Threaded attachment affords the sturdiest mounting and the highest accuracy: this method should be used to mount transducers whenever practicable. Recommended tolerances according to ISO Standard 1101 are shown below.

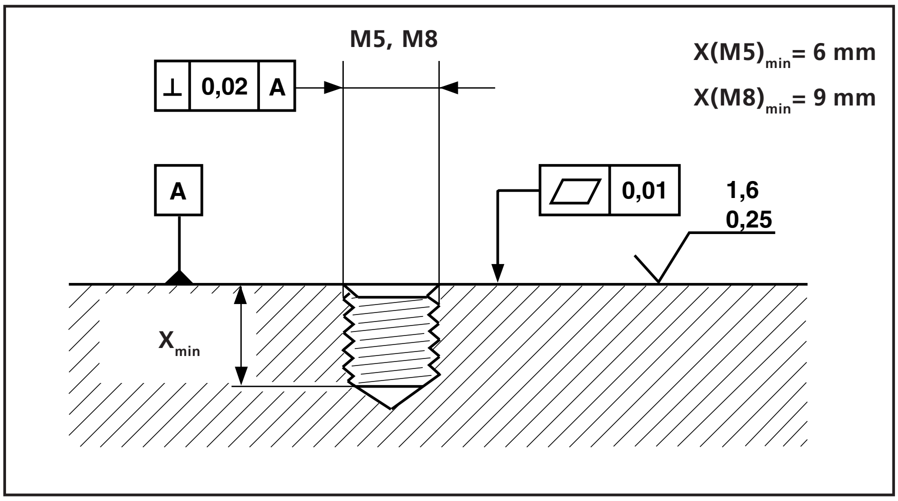


Fig. 18 Tolerances for threaded attachment

Procedure:
- Prepare the mounting surface according to the ISO specification mentioned above. The mounting surface should be flat (within 1/100 mm / 0.4 mil tolerance) and perfectly clean.
- Clean the mounting surface of the transducer.
- Drill the pilot hole and tap the appropriate thread (M5 or M8). Be sure to drill the pilot hole deep enough so that the entire transducer stud can be screwed into the hole.
- Screw the transducer into place and ensure that it is seated properly. A bit of silicone grease on the mounting surfaces of the machine and the transducer will help improve the contact between them.

Adhesive mounting:

Another excellent method for joining the transducer to the measurement object is adhesive bonding.

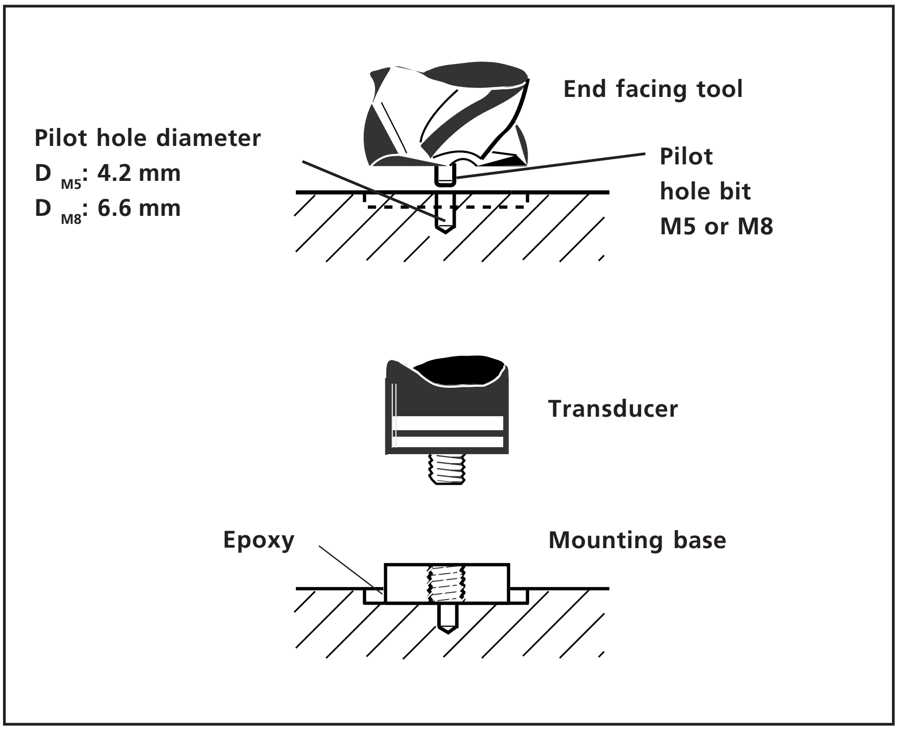


Fig. 19 Adhesive bonding onto mounting base

Procedure:

- Prepare the surface of the measurement object according to ISO Standard 1101: face the indentation for the mounting base. The facing tool should have a diameter of at least 20 mm (13/16") so that the 17 mm (11/16") mounting base fits easily onto the finished surface.
- Bond the mounting base to the prepared surface. Make sure that both surfaces are clean and de-greased. A 2-component epoxy adhesive (such as the Double Bubble brand available under order number VIB 8.474) should be used for best results.
- Allow the adhesive to set, then screw the transducer firmly into place. Be sure not to apply too much torque to the bonding surface.

Hand-held (contact) probes:

If the transducer tip is pressed onto the measurement location, then the rigidity of the contact point and the mass of the transducer create so-called contact resonance, which of course corrupts measurement accuracy. This is why hand-held probes should be avoided for measurement of vibration frequencies in excess of 1000 Hz. Above this level, directional sensitivity may adversely affect measurement reproducibility.

Magnetic mounts:
Permanent magnets offer quick and simple mounting. Advantage: the measurement location may be easily moved. Disadvantage: the measurement frequency range is reduced (to approx. 2000 Hz max.), and the method is limited to ferromagnetic objects. The contact surface should be finished and clean.

Overview:
After the correct selection of vibration transducer type, the mounting method is of crucial importance to obtaining useful measurements. Prüftechnik transducers offer the following mounting possibilities:

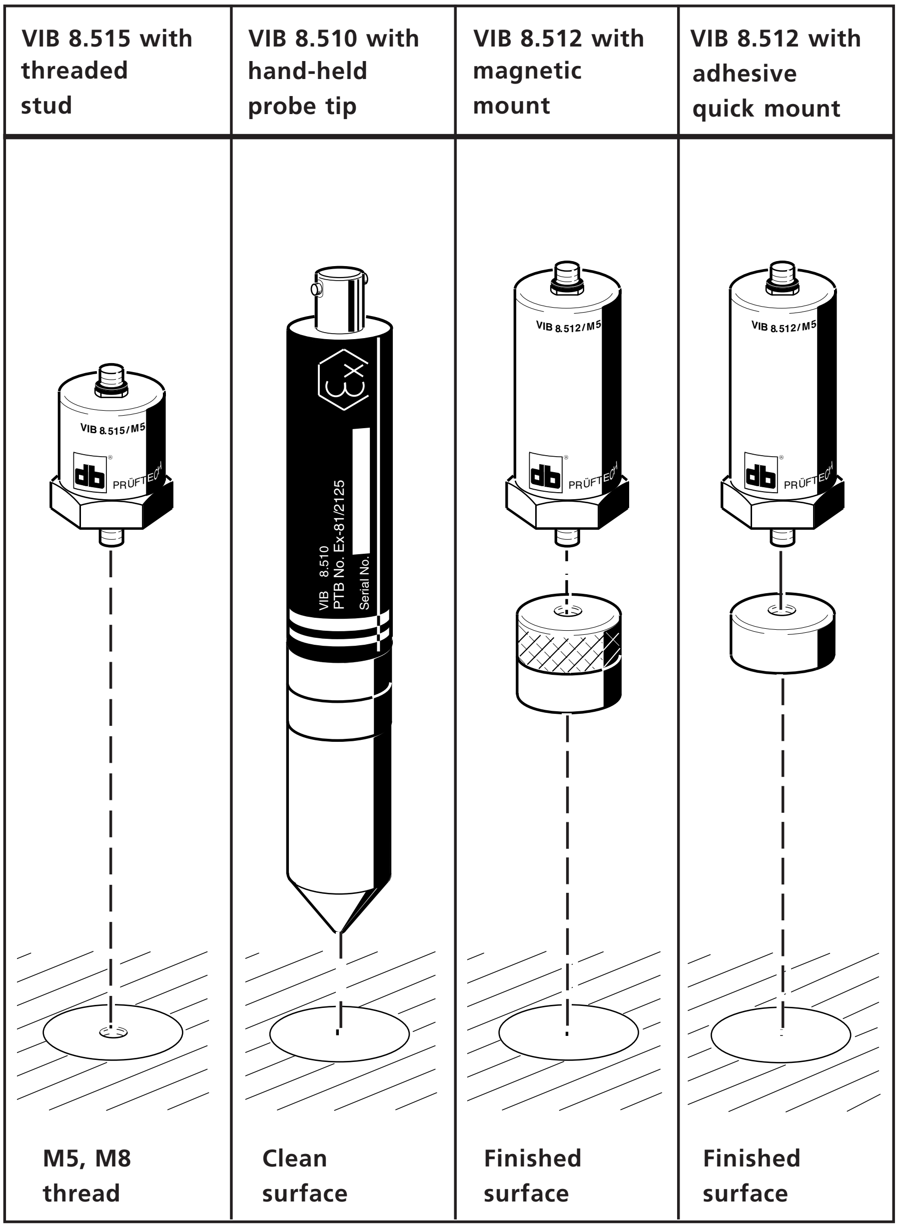


Fig. 20 Transducer mounting

Mounting effects

The figure below makes clear the rapid decline of resonance frequency with decreasing mounting rigidity.

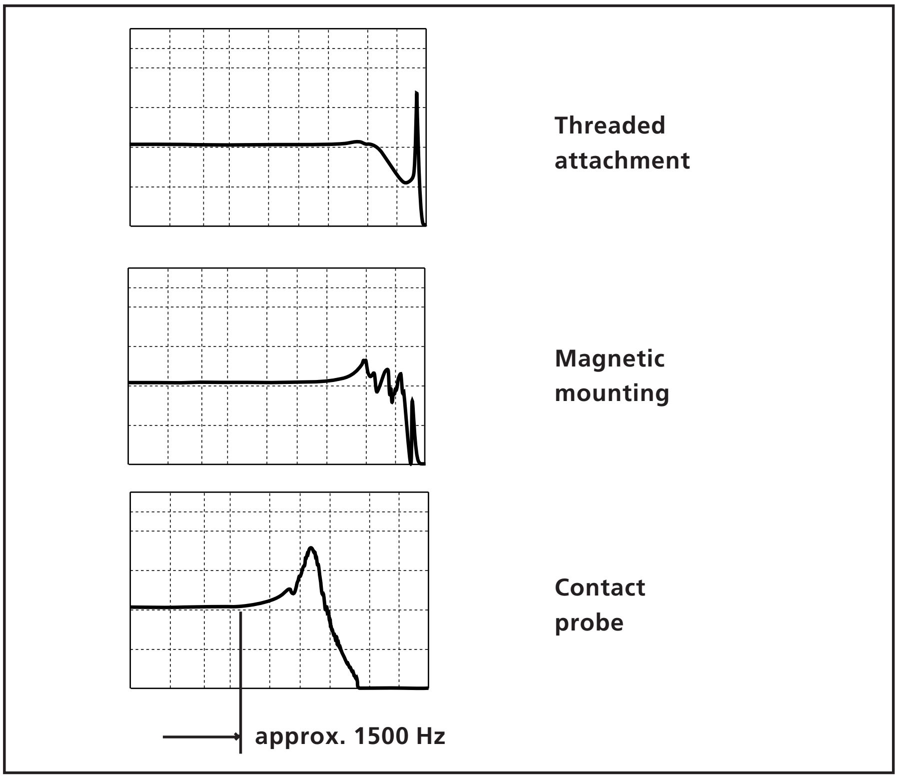


Fig. 21 Influence of mounting on resonance behavior

Measurement locations:

The choice of measurement location depends, of course, on the direction in which the strongest vibrations are expected to occur based upon experience. As a rule, at least one transducer is mounted radially at each bearing location, with one additional transducer per aggregate mounted axially. When anti-friction bearings are to be monitored for wear, a transducer is mounted as close as possible to the bearing. A single measurement point usually suffices for electric motors or fans with rotating unbalance; large turbines, on the other hand, should certainly be measured in all three directions (axially, and radially in vertical and horizontal).

The rigidity of the bearings (or transmission path from the origin of vibration to the measurement location) is critical as well. The transducer should always be mounted at the most rigid location possible: housing covers etc. are not suitable for mounting.

Transducer mass influence:

Through its own mass, the transducer places an additional load on the object to be measured and alters its vibration behavior. This effect is negligible as long as the transducer mass does not exceed 10% of that of the vibrating mass.

3. Evaluating machine condition through vibration analysis

All rotating machines exhibit a characteristic vibration behavior caused by forces which change over time. In the majority of cases, unbalance and shaft misalignment account for these forces, which place the machine under additional load and thus lead to a reduction in operating life.
The goal of condition-based maintenance is to detect these damaging forces before they make repair necessary. This requires an accurate representation of machine condition at all times. Here is where condition monitoring plays a central role. However, real-life machine condition cannot be categorized merely in terms of ‘good’ and ‘bad’; the gradations between these two extremes are many and varied. The maintenance technician should be able to reasonably estimate the probability of failure, for this is the first step toward planning and carrying out timely maintenance and repair work. The production facility operator must rely on the data provided by the proper measurements in order to decide when maintenance is necessary and when production can continue without risk.
The various methods of vibration measurement provide reliable orientation and important machine condition information. Vibration severity measurements according to accepted standards (such as ISO 2372) and frequency analysis allow achievement of the goals outlined above with a high degree of accuracy.

3.1 Vibration severity according to ISO 2372

If no other information is available on the vibration behavior of a particular machine, then the machine condition may be evaluated with the help of accepted industry standards and recommendations.
In 1974, the International Organization for Standardization issued its ISO Standard 2372 (‘Mechanical vibration of machines with operating speeds from 10 to 200 rev/s - Basis for specifying evaluation standards’), which specifies maximum acceptable vibration severity according to machine type and size. Experience has shown that when these values are exceeded for longer periods of time, machine damage (especially bearing damage) must be expected.
Although this ISO standard was one of the very first to appear in industry, it has gained wide recognition and acceptance. As a result, numerous other organizations have devised similar standards based upon this specification.

3.1.1 Definition

ISO 2372 defines vibration strength of a machine as the greatest effective value of vibration velocity occurring at functionally important locations within a frequency range of 10 Hz to 1000 Hz. This effective value is the quadratic mean of vibration velocity.
This value is then compared with tolerances established for six different classes of machines defined as follows:

Class I:	Small machines and individual parts of engines and machines, integrally connected with the complete machine in its normal operating condition. (Electric production motors of up to 15 kW are typical examples of machines in this category.)
Class II:	Medium-sized machines, (typically electric motors with output of 15 kW to 75 kW) without special foundations, rigidly mounted engines or machines (up to 300 kW) on special foundations.
Class III:	Large prime movers and other large machines with rotating masses mounted on rigid and heavy foundations which are relatively stiff in the direction of vibration measurement.
Class IV:	Large prime movers and other large machines with rotating masses mounted on foundations which are relatively soft in the direction of vibration measurement (for example turbo-generator sets, especially those with lightweight substructures).
Class V:	Machines and mechanical drive systems with unbalanceable inertia efforts (due to reciprocating parts), mounted on foundations which are relatively stiff in the direction of vibration measurement.
Class VI:	Machines and mechanical drive systems with unbalanceable inertia effects (due to reciprocating parts), mounted on foundations which are relatively soft in the direction of vibration measurements; machines with rotating slack-coupled masses such as beater shafts in grinding mills; machines, like centrifugal machines, with varying unbalances capable of operating as self-contained units without connecting components; vibrating screens, dynamic fatigue-testing machines and vibration exciters used in processing plants.

Class V and VI describe special machines which are difficult to categorize. For example, a centrifuge must be able to yield to high dynamic forces. This makes it difficult to specify a particular vibration severity tolerance.
The tolerance table is therefore limited to the more commonly-encountered equipment of classes I through IV.

3.1.2 Classification

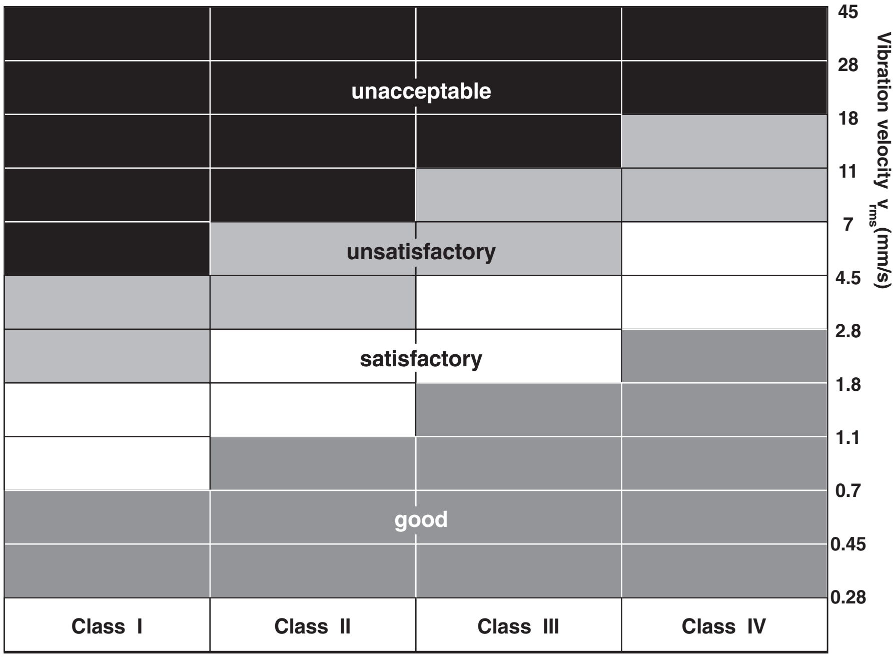


Fig. 22 Vibration severity classification according to ISO 2372

It is entirely possible that in some cases, the categories shown above may need correction to accommodate practical experience previously gained with a particular machine. For example, fan blades usually become encrusted with deposits of dust and grease over time. Eventually, these deposits become so thick that the centrifugal force that occurs during operation causes them to fly off, leading to severe unbalance which takes its toll on the fan shaft, bearings and the entire aggregate. If damage is to be avoided, then the vibration severity tolerance for this machine should be lowered from 7.1 mm/s to, say, 4.5 mm/s.

3.2 Frequency analysis

Frequency analysis offers considerably more detailed information on the condition of an aggregate. In contrast to the evaluation of vibration severity provided by ISO 2372, the vibration signal is filtered and its individual frequency components are evaluated. The main advantage of spectrum analysis lies in its ability to detect deterioration of specific machine elements long before failure - even allowing enough advance warning to procure the necessary spare parts in time. Modern measurement instruments apply the FFT (Fast Fourier Transformation) method; some (such as the hand-held VIBROSPECT®) use selective filtering to sweep through the frequency spectrum.

3.2.1 Fundamentals

When a machine operates smoothly, its vibration spectrum remains stable. When a part becomes worn or broken, changes appear in the spectrum which can be traced back to periodic events in gears, bearings etc. (such as unbalance, meshing of gear teeth or passing of fan or impeller blades). As mentioned in Section 1.2, the individual spectrum peaks can be matched to their respective vibration sources, permitting an exact diagnosis of machine disturbances.

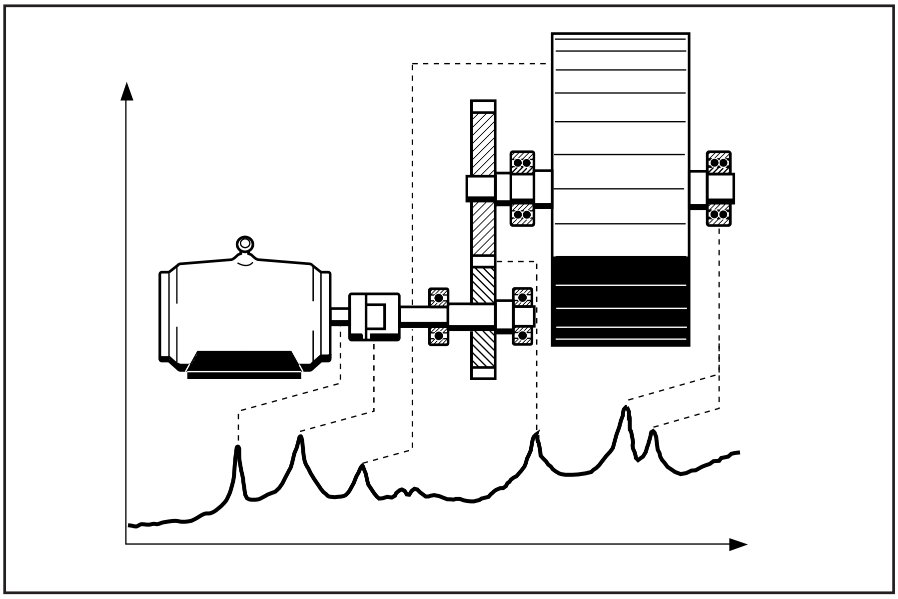


Fig. 23 Vibration spectrum peaks matched to their sources

The signature spectrum (baseline spectrum), a measurement taken when the machine is in good operating condition, is normally dominated by the rotation frequency of the machine. Variations in rotation speed change the spectrum accordingly, making a cursory comparison with the original condition difficult at best.

3.2.2 Terminology

A number of specialized terms are used to describe phenomena related to FFT analysis. The operator should become familiar with the terms explained below in order to better understand digital FFT analyzers and their use.

Aliasing (anti-aliasing filter):
Alias effects occur when the signal sampling rate is too low.

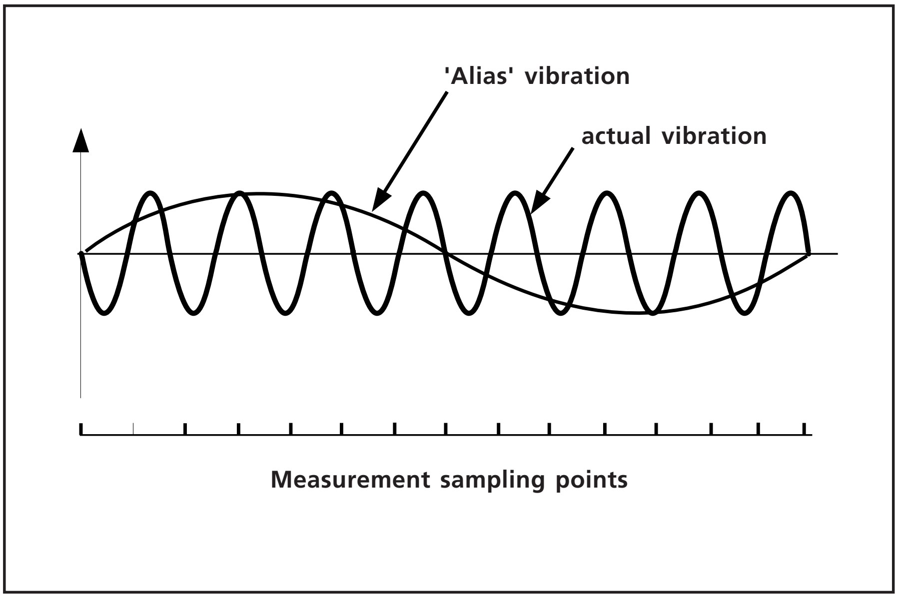


Fig. 24 Alias effect

This low sampling rate does not give sufficient resolution of the frequency spectrum for accurate interpretation. The correct solution is to raise the sampling frequency to at least twice the highest frequency present in the signal to be measured. In practice, however, since it is often difficult to tell whether the measurement signal at hand contains high frequencies, a low-pass (anti-aliasing) filter is used to suppress signal components above this sampling frequency.

Resolution:
The maximum number of lines into which an FFT analyzer can divide a spectrum is known as resolution. The frequency separation between the lines is then calculated from the frequency range and the resolution.
Example:

Resolution:	400 lines
Frequency range:	1000-10.000 Hz
Separation between frequency lines:	22.5 Hz

Bandwidth:
The term ‘bandwidth’ is used in conjunction with filters. The bandwidth is a measure of the separation sharpness of the filter used. With an ideal filter, all frequencies within the bandwidth are admitted and all those outside the bandwidth are filtered out, i.e. suppressed.

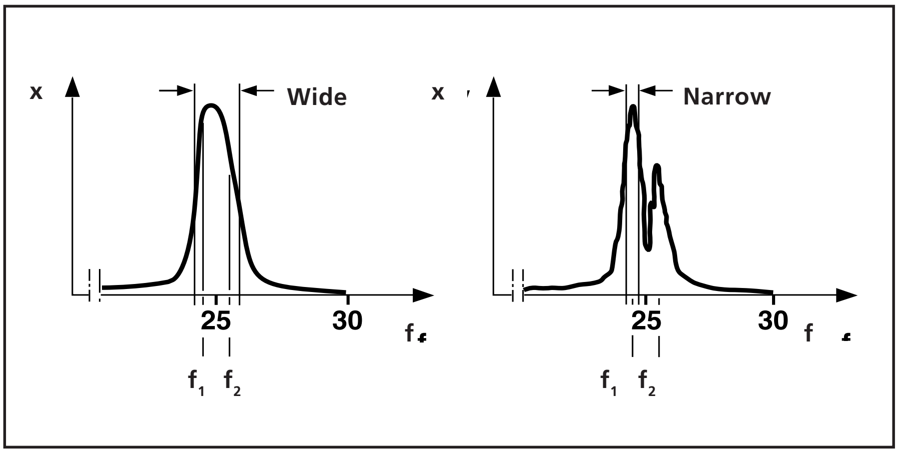


Fig. 25 Signal separation at different bandwidths

The bandwidth of real filters is defined as the separation between the frequencies at which amplification is attenuated (lessened) by 3 dB (or approx. 30% of the maximum).
If two vibrations lie closer together in the frequency spectrum than the filter bandwidth, then they cannot be displayed separately (see Fig. 25, left).

Dynamic range:
(see also 'logarithm')
This value describes the ability of an FFT analyzer to display strong and weak signals simultaneously. A large dynamic range is especially important when weak vibration components occur along with stronger ones such as unbalance. In that case, logarithmic scaling of the display allows clear recognition of the weak signals.

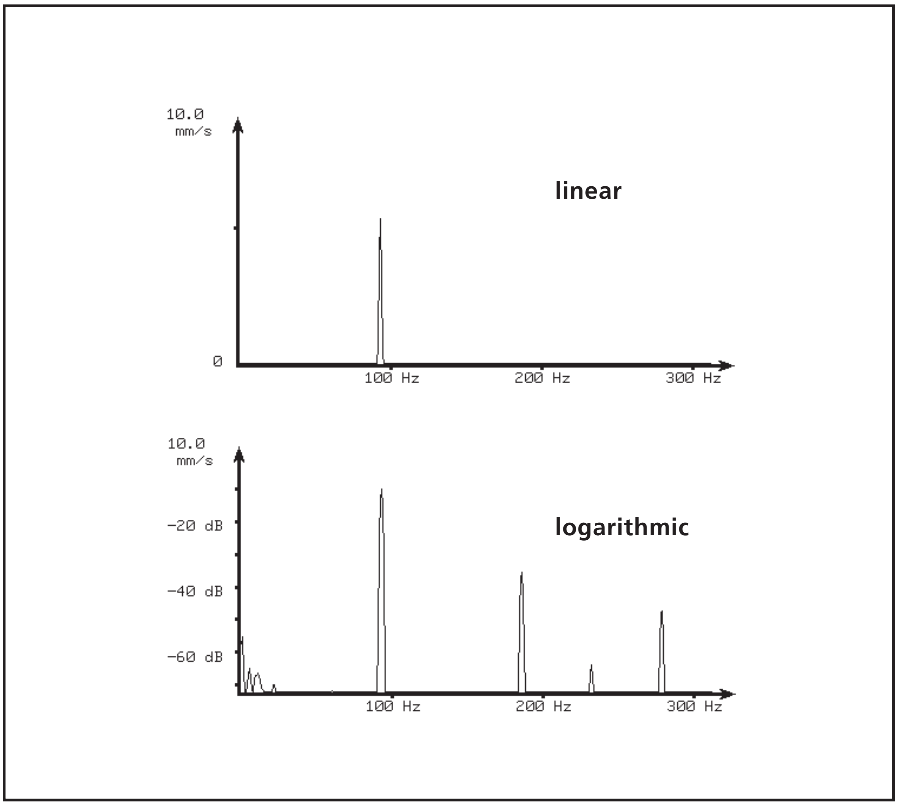


Fig. 26 Dynamic range: linear vs. logarithmic

The dynamic range is defined as the ratio between the greatest and smallest signal amplitudes which can still be displayed at the same time. The 12-bit A/D converters used in modern FFT analyzers give them a dynamic range of 70 dB.

Window function:
One prerequisite to accurate FFT calculation of the vibration spectrum is a periodic (i.e. repetitive) time signal. This means that for a sine wave, the sample period should begin and end at exactly the same point of the waveform, without any leftover portions at the beginning and end of the measurement sample (or, alternatively, the signal could be measured ad infinitum).

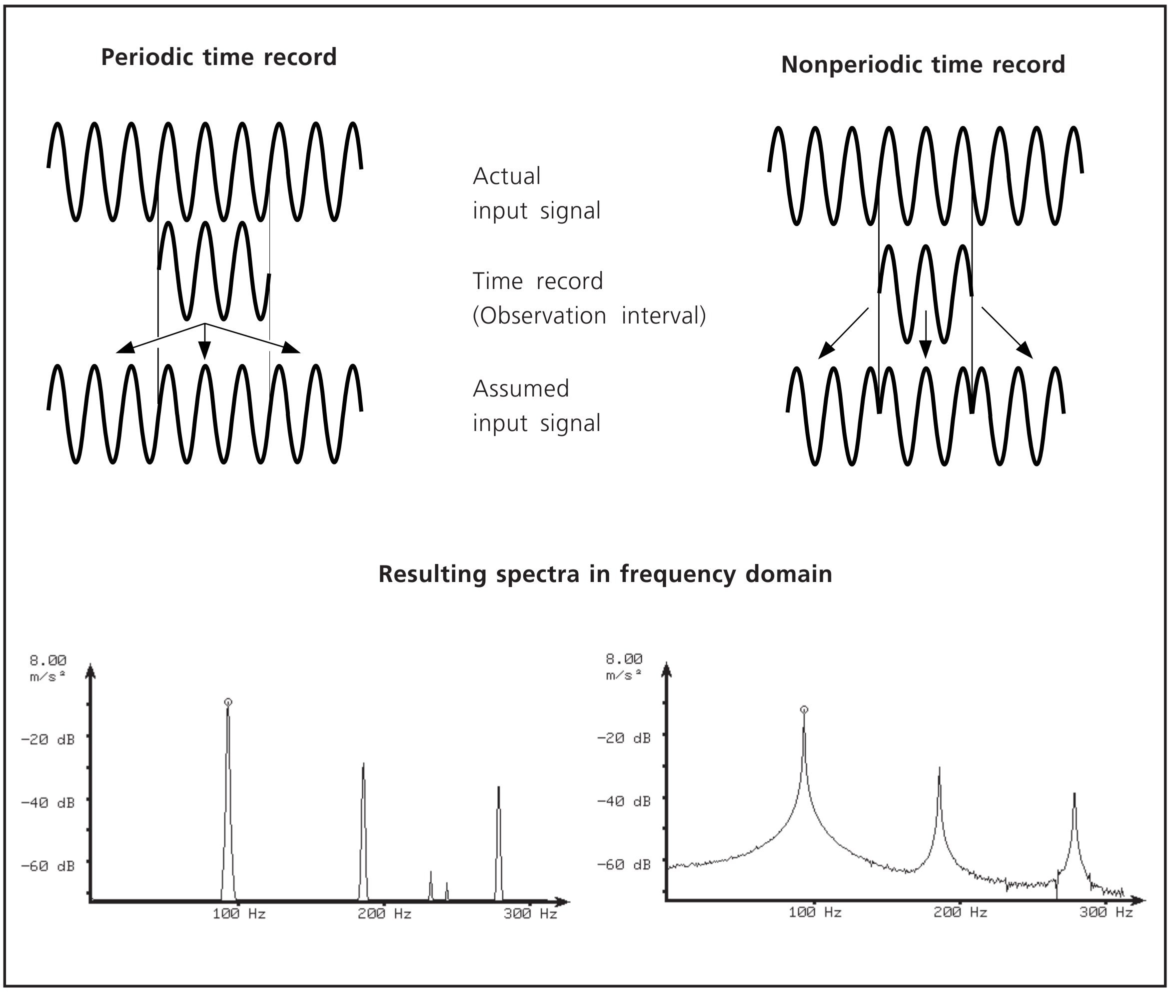


Fig. 27 Periodic vs. nonperiodic time record with resulting frequency spectra

If partial vibration waveforms are included in the time record, the resulting spectrum calculated by the Fourier transform can become considerably distorted; this spectrum-widening phenomenon is commonly known as leakage.
The window functions (or time-limiting functions) suppress the so-called sidebands so that spectral widening is held in check. This is done by superimposing a time weighting function over the time signal so that its beginning and end portions are left out of the spectrum calculation.

Types of windows:

Rectangular window: The rectangular window is in principle not a window at all: it admits all portions of the time signal with the same weighting. This makes it ideal for measuring signals whose beginning and ending value is zero (such as shock signals, pulses and noise signals). This window gives the highest amplitude accuracy (provided the beginning and end of the signal are not cut off).

Hanning window: This window gives the highest frequency resolution at the expense of amplitude accuracy, which may be reduced by as much as 15%. The Hanning window is one of the standard window functions.

Flat top window: The flat top window gives a less accurate frequency depiction than that of the Hanning window, but its amplitude accuracy is much higher (only 1% error). This makes the flat top window the most suitable for measuring continuous vibration signals when amplitude must be closely examined.

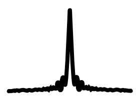
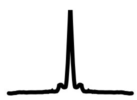
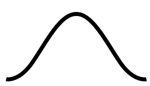
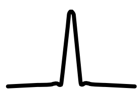
Window function	Damping of highest peak	Spectrum	Damping of secondary peaks
Rectangular 	-		-
Flat top 	10%		0.3 dB
Hanning 	58%		18.4 dB

Fig. 28 Window functions

Frequency resolution capability:
The resolution capability in the frequency domain is determined primarily by the number of intervals (i.e. the number of filters) and the section enlargement (zoom) capability.

Logarithm:
(see also 'dynamic range')
If signals with large and small amplitudes are to be displayed at the same time, the amplitude axis must be set to logarithmic scaling. The chart below shows the relationship between logarithmic decibels (dB), power and voltage:

dB	Relative power	dB	Relative voltage
20	100	40	100
10	10	20	10
3	2	6	2
0	1	0	1
-3	1/2	-6	1/2
-10	1/10	-20	1/10
-20	1/100	-40	1/100

Fig. 29 Logarithmic scaling

The chart clearly shows that a voltage signal of 0.1% (1/1000) of a given signal appears 60 dB weaker. However, if the maximum peak is, say, 80 dB, the much weaker peak will still be shown clearly, filling out 1/4 of the available display height.

Averaging:
Real mechanical vibration signals are often accompanied by large amounts of noise, e.g. from adjacent equipment. The quality of these measurements can be improved by using averaging functions; in particular, time averaging reduces the effects of background noise considerably when a trigger is used to measure in the time domain. Another alternative known as the peak-hold mode can be used to retain in the display the highest levels in either time or frequency domain over a specified number of measurements.

Trigger:
A timing mechanism for starting the measurement. Three different alternatives are generally available: no trigger, internal trigger (based upon the input signal level) and external trigger. The last two may also be set to begin measurement as the measurement signal rises above the desired level or as it falls below it.

Waterfall diagram:
When machine phenomena related to operating speed are to be examined, it may be helpful to take several measurements at different speeds and then to observe all the spectra in the same display. This type of display is also useful for studying resonance behavior since the resonance frequency peak remains unchanged even when the machine speed varies.

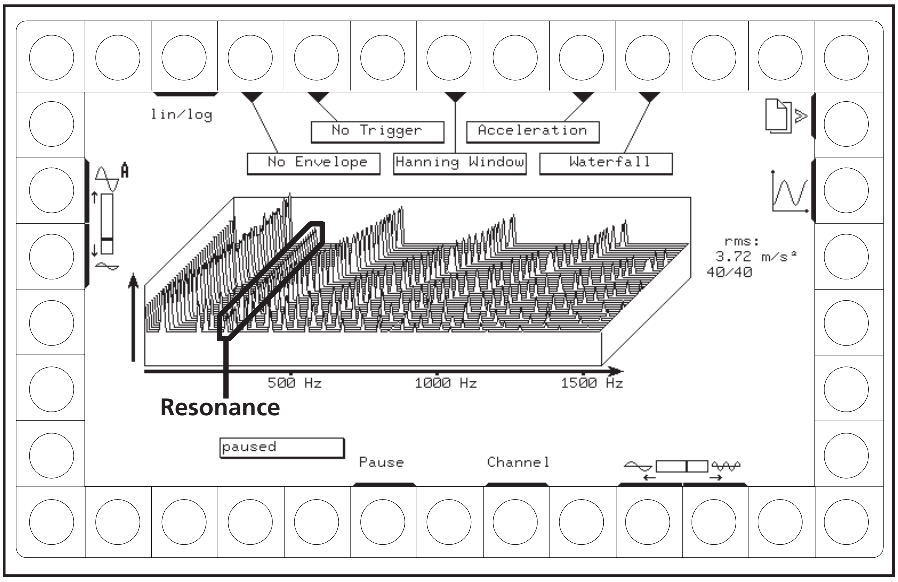


Fig. 30 VIBROSPECT FFT waterfall diagram

Zoom function
Improvement of frequency resolution either by raising the sampling rate or reducing the measurement bandwidth.

3.2.3 Standard vibration spectra

The following signal waveforms appear quite often in practice:

Sine wave:
The spectrum of a classic sine wave consists of one single line.

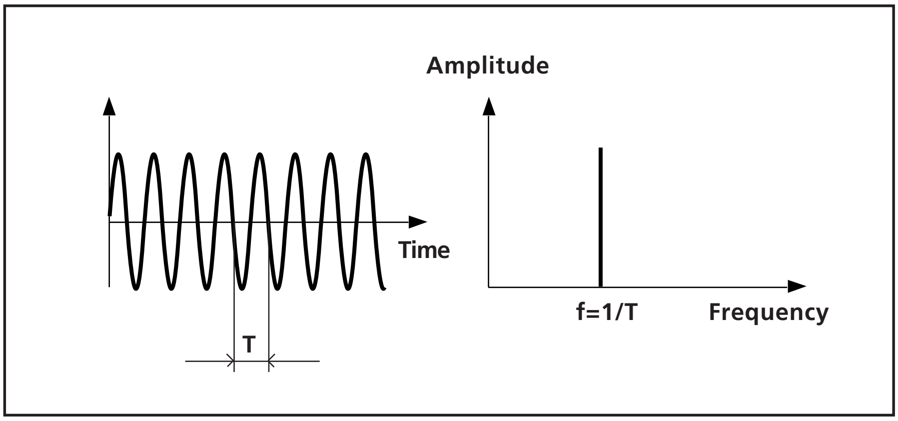


Fig. 31 Sine wave

Square wave:
This vibration form can occur, for example, when a bearing cover becomes loose. The vibration consists of an infinite number of harmonics (multiples of the rotation frequency).

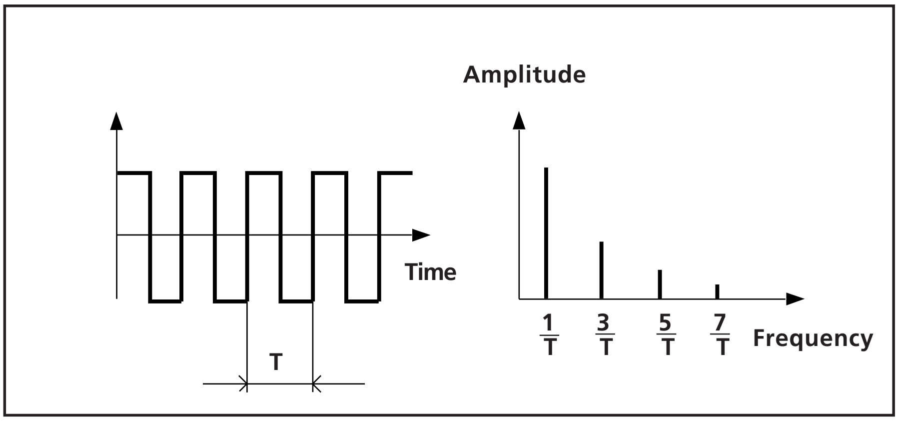


Fig. 32 Square wave

Pulse train:
This vibration form occurs, for example, when bearings or gears are damaged.

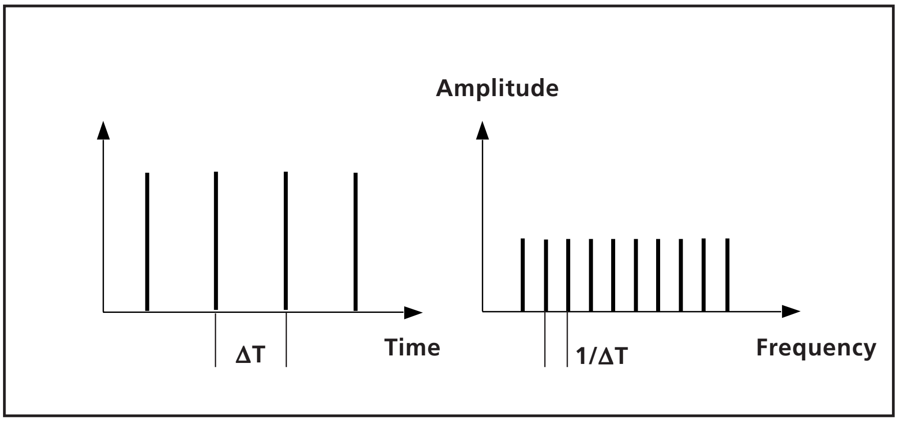


Fig. 33 Pulse train

Amplitude-modulated sine wave
If machine vibrations (for example, from unbalance) are modulated, the corresponding vibration spectrum peak is surrounded by sidebands spaced evenly about the carrier frequency.

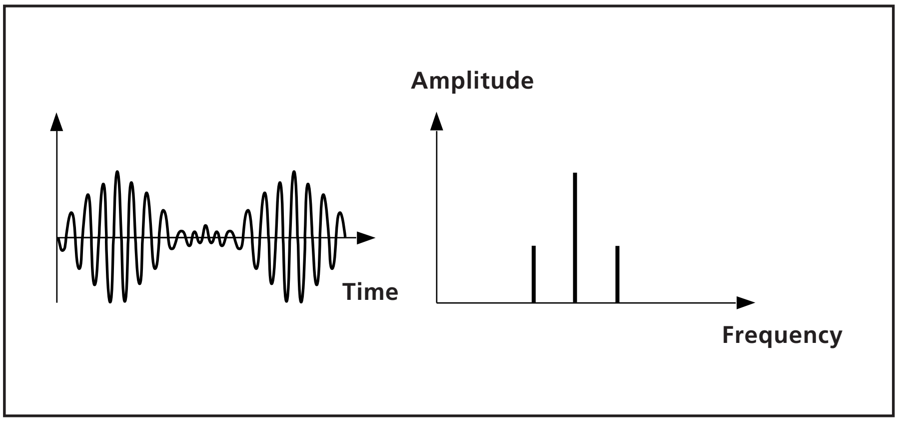


Fig. 34 Amplitude-modulated sine wave

Vibration decay:
Vibrations which decay gradually over time exhibit continuous spectra, because their individual spectrum lines lie very close to one another.

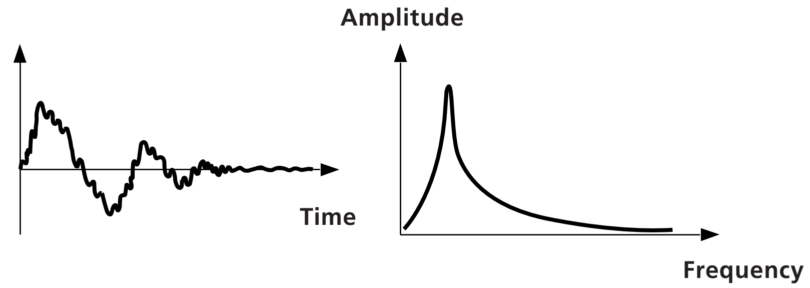


Fig. 35 Vibration decay

3.2.4 Notes for successful frequency analysis

The following points should be observed when using a frequency analyzer to obtain good results:

- For machines with anti-friction bearings, set the lower end of the measurement frequency range to the rotation frequency (or to one-half that amount for machines with sleeve bearings). The upper limit should be the third harmonic of maximum tooth meshing/blade passing/anti-friction bearing frequency.
- When setting the measurement bandwidth, pay attention to the space between component rotation frequencies or sidebands from tooth meshing frequencies, to sufficient separation of signal components and interference at the power supply frequency.
- Use averaged signals: 5 to 10 measurements for RPM-related studies and many more for anti-friction bearings.
- The trigger should be deactivated when turbo machines or electric motors are measured. Transient signals (shocks, pulses, decay) should be measured using internal triggering.
- Make sure the correct window function is activated.

3.3 Vibration causes and their characteristic frequencies

Possible cause	Dominant frequency	Direction	Comments
Imbalance	1x rotation frequency	Radial for dynamic imb., possibly axial	Amplitude proportional to imbalance and RPM; causes severe vibration to occur
Misalignment; bent shaft	1x rotation frequency often 2x and higher multiples	Radial and axial	Severe axial vibration and 2nd harmonic; best realigned with OPTALIGN® V or SYSTEM 2 TURBALIGN®
Bearing damage	High-frequency vibration	Radial and axial	May be diagnosed from vibration only through use of diagnostic functions or shock pulse analysis (SPM)
Sleeve bearing play	Subharmonic, exactly 1/2 or 1/3 of rotation frequency	Radial	Usually dependent upon RPM and operating temperature
Oil film whirl or whip (sleeve bearings)	40% - 50% of rotation frequency	Radial	Occurs with high-speed machines; phase fluctuates.
Hysteresis whirl	Critical shaft rotation frequency	Radial	Vibrations are excited as machine climbs through critical RPM and remain at higher speeds. Remedy: Rotor must be reworked (mounting improved).
Gear tooth damage	Tooth mesh frequency and multiples thereof with sidebands located at multiples of rotation frequency	Radial and axial	Sidebands occur from modulation of tooth mesh vibration at rotation frequency; difficult to isolate due to superimposition.
Belt drive damage	Rotation frequency and multiples thereof	Radial	Additionally recommended: combined RPM and belt speed measurements to check for belt slippage.
Turbulence; cavitation	Blade/vane passing frequency	Radial and axial	Additionally recommended for pumps: shock pulse measurement at the pump housing.
Electrically induced vibration	Rotation frequency, 2x line frequency	Radial and axial	Sidebands may also occur located at multiples of the rotation frequency; vibration ceases when power is cut off.

3.4 Case histories of machine damage

In order to ascertain the exact cause of vibration spectrum peaks, the configuration of the machine to be examined must be known in detail: RPM, bearing type and geometry, coupling design, load angle, number of rolling elements, pitch diameter, tooth mesh frequency, resonance frequency, number of teeth or blades etc. must be known exactly. These data allow calculation of the individual frequencies which are characteristic of the corresponding individual machine elements. These frequencies and their multiples are then compiled into a list of potential 'trouble' frequencies. Consider for example a compressor coupled to a reduction gear: important machine dimensions include the RPM of the drive shaft (which determines the main component of vibration), the coupling type and the tooth mesh frequency of the gear.
Additional control measurements are often necessary due to the vast number of possible influences which can occur simultaneously. This also limits the discussion of case histories to specific examples, which should not be construed to be generally valid: the same machine can exhibit completely different vibration spectra depending upon its foundation. Evaluation is greatly facilitated when orderly records of previous measurements and maintenance work are available. Reliable condition diagnosis is impossible without a complete machine history (trend record).

3.4.1 Radial compressor coupling damage

Technical data

1) AEG motor
   Power: 1550 kW
   Drive shaft RPM: 24.7 Hz (1484 RPM)
2) Flender SEG 280 gearbox
   Tooth mesh frequency: 2338 Hz
   (sidebands possible from both couplings)
3) Schiele radial compressor
   Compressor shaft RPM: 86.4 Hz (5184 RPM)
4) Flender Ruppex claw coupling
5) Flender Zappex gear coupling

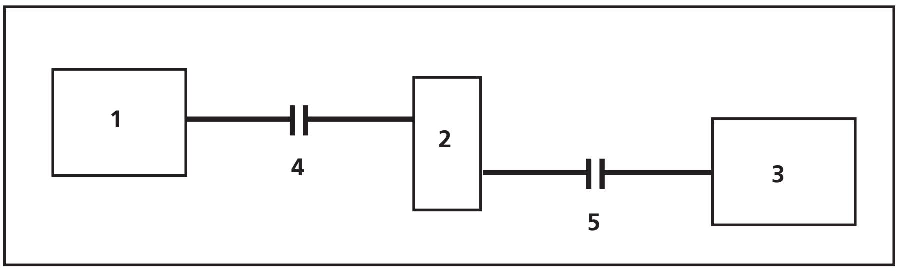


Fig. 36 Radial compressor aggregate

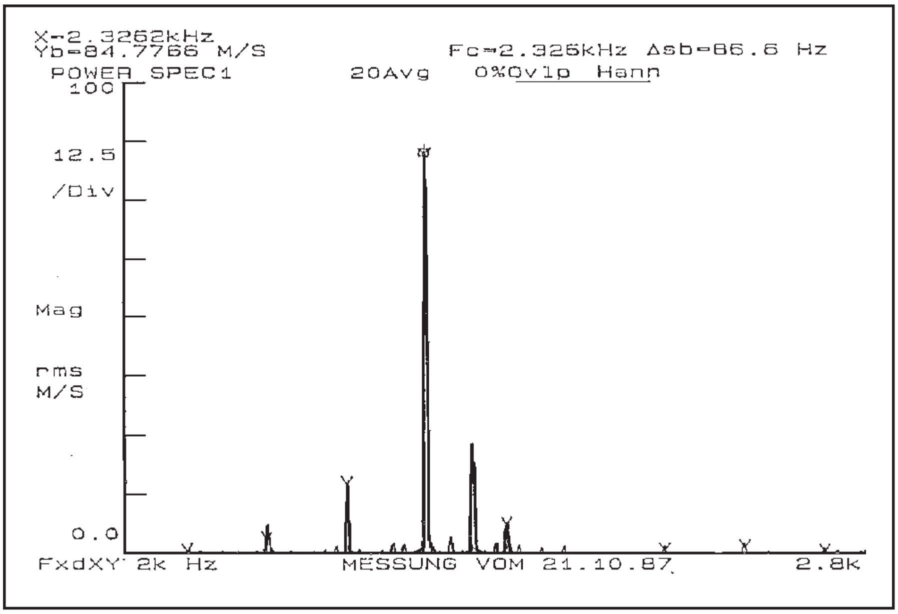


Fig. 37 Turbocompressor vibration spectrum

Measurement location: gear shaft (compressor side), near bearing
Discussion:
Over the course of several machine measurements, the vibration peak at 2326 Hz (the gear tooth mesh frequency) climbed markedly, At the same time, sidebands appeared 86.6 Hz on either side; their amplitudes remained nearly constant. The frequency peak at the tooth mesh frequency combined with sidebands can point toward two possible causes:
1) Early gear damage
However, this tooth mesh frequency could not be measured on the low-speed gear shaft, so the probability of gear damage is low.
2) Gear coupling damage
The sidebands occurring at intervals of 86.6 Hz indicate modulation of the tooth mesh frequency, or carrier frequency, by the shaft (and coupling) rotation frequency. This is clearly due to the gear coupling. Disassembly of the coupling showed that gross axial misalignment had prevented correct centering of the coupling shell, so that the teeth could not mesh properly. The resulting increase in local stress concentration led to gear tooth damage.

3.4.2 Chiller misalignment

Technical data:
1) Motor drive power 900 kW
2) Linde compressor
   Compressor screw divided into 4 portions
   Rotation frequency 50 Hz (2950 RPM)

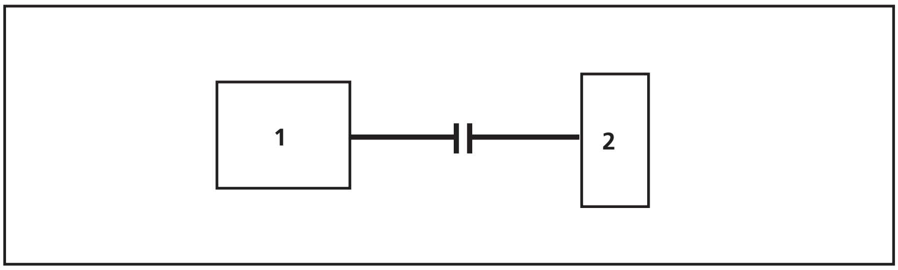


Fig. 38 Chiller aggregate

Measurement location: Motor (1), axial and radial, near bearing.
Despite installation and alignment by specialists, the machine had always exhibited high vibration - even though the prescribed alignment targets had been fulfilled. At four times the

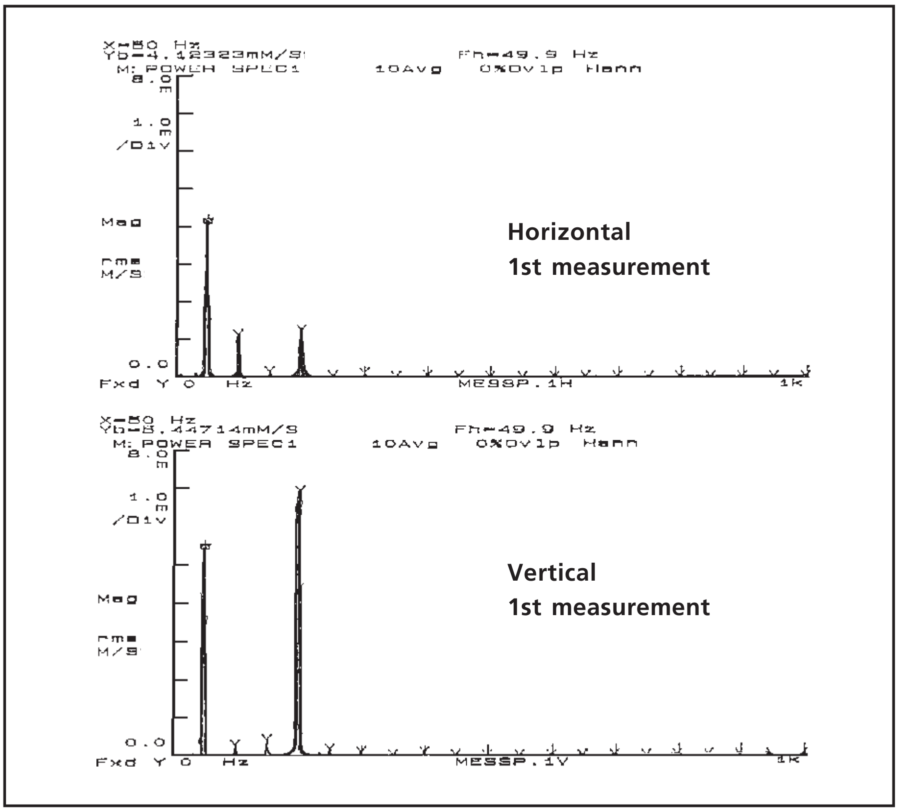


Fig. 39 Chiller vibration spectrum

The first axial measurement shows extremely high vibration of up to 10 mm/s.

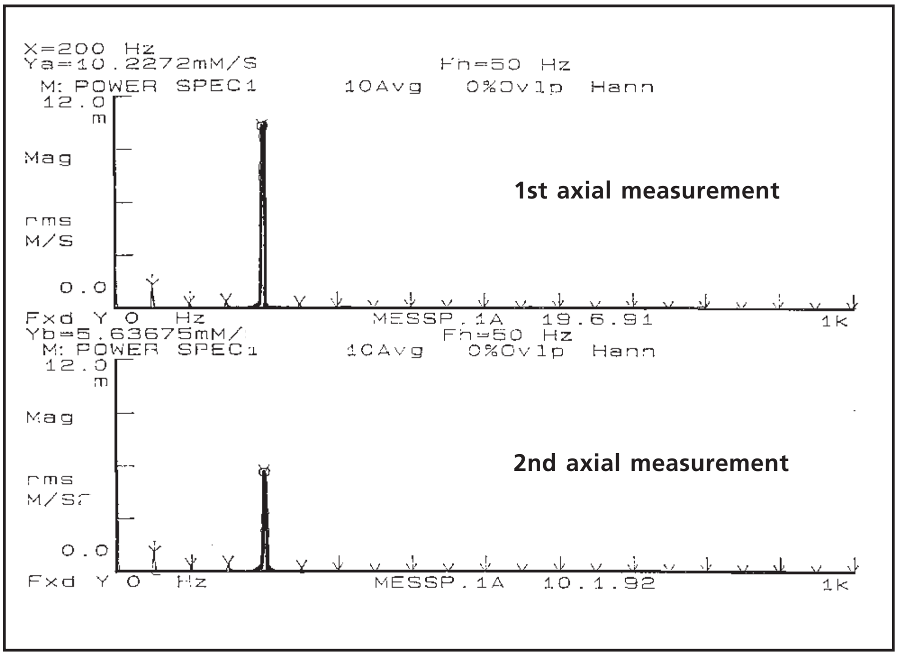


Fig. 40 Control measurement, vibration spectra

rotation frequency, vibration in vertical direction was nearly 7 mm/s.

Discussion:

The cause of high axial vibration is not immediately clear, since the motor has no axial bearings. The measured frequency is equal to the compressor emission frequency, since the compressor screw is divided into 4 portions. Comparative measurement on the compressor itself, however, showed only lower vibration levels than those of the motor bearing.

Following a check of manufacturer-specified thermal growth values with the PERMALIGN® laser-optic coordinate measurement system, it could be shown that the motor should be repositioned to stand 1 mm higher and 0.2 mm to one side. Since the vertical value seemed surprisingly high, it was decided to set vertical offset to a conservative value of 0.7 mm. Vibration results were checked with a follow-up measurement:

The improvement in amplitude at 4 times the rotation frequency is clearly visible (2nd measurement). This confirms that the direction of adjustment was correct; the full amount of 1 mm indicated by PERMALIGN® is to be implemented at the earliest available opportunity.

3.4.3 Summary

Frequency analysis is a very exact ‘tool’ for achieving the goals of condition-based machine maintenance. When sufficient details of the machine design are known, the spectrum allows an exact diagnosis of the vibration source.

4. Index

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