

Machine diagnosis: Quick and easy through FFT analysis



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1. Introduction

Vibration monitoring and vibration diagnosis of machines and aggregates has gained enormous importance during the past several years. Even smaller and medium-sized machines are being included in vibration monitoring strategies with increasing frequency. This is largely due to the fact that vibration measurement equipment has reached a price level that makes application of vibration measurement a viable alternative for these machines as well.

The interest in vibration technology and its successful application in the electrician's trade has also increased dramatically during recent years. On the one hand, machine operators increasingly often demand a 'vibration signature' record following installation or repair, while on the other hand, vibration monitoring and diagnosis offer considerable potential for additional service business, especially through

consultancy to smaller operations that lack the resources to pursue vibration measurement on their own. And of course, vibration diagnosis is a fantastic tool for localization of defects and causes of damage to machines and aggregates, and one which can even be used as an objective defense against unjustified warranty claims.

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2. Vibration spectra of a belt-driven exhaust fan

Let us examine a simple practical example to illustrate the possibilities of vibration analysis: a belt-driven fan unit had failed due to excessive vibration. Since the most severe vibration level was measured on the drive motor, the motor seemed the logical candidate for examination. Vibration analysis showed, however, that the extremely severe vibration (15.2 mm/s) at the motor was occurring primarily at a frequency which was conducted to the motor via the belt drive. When the belt drive wheel on the fan was balanced, vibration decreased to acceptable levels of 2.3 mm/s on the fan and 3.2 mm/s on the drive motor.

This case presents a typical method of operation: a simple measurement of overall vibration level allows the machine condition to be rated as 'good', 'satisfactory', 'unsatisfactory' and 'unacceptable'. In the case of excessive

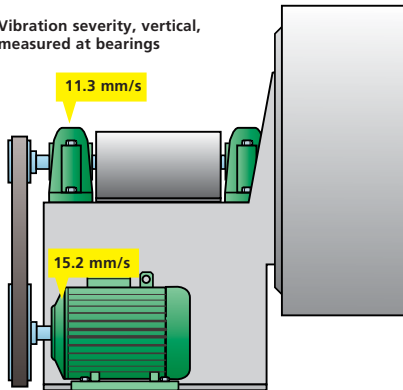
vibration, the root cause - drive belt wheel unbalance - was made clear by checking the frequency peaks in the FFT vibration spectrum.

Vibration spectra of a belt-driven exhaust fan

Paint shop exhaust fan ($P = 37 \text{ kW}$)

1. Parameter measurement

Vibration severity, vertical,
measured at bearings

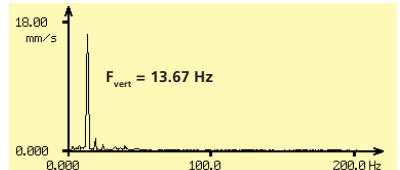


Motor: 1475 rpm = 24.58 Hz

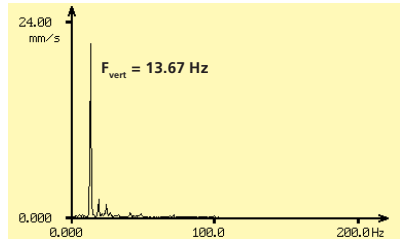
Fan: 820 rpm = 13.67 Hz

2. Signal analysis

FFT spectrum of vibration signal



Fan bearing, radial/vertical

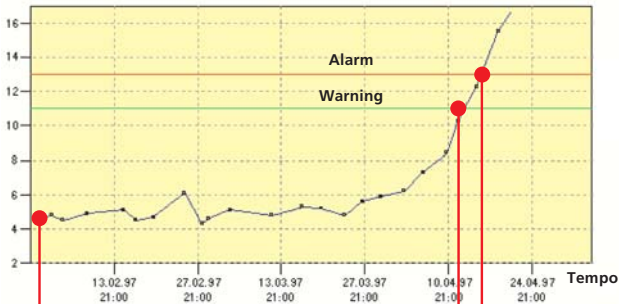


Motor bearing, radial/vertical

3. Machine condition trending

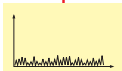
A rational approach to successful and effective condition monitoring is that of trending the development of characteristic overall value measurements of machine condition over time. The trend readings are plotted as shown here and compared with appropriate warning and alarm thresholds. When thresholds are exceeded (and not before then), detailed vibration diagnosis is performed in order to locate the exact source of trouble and to determine the corresponding maintenance remedy. Let us examine, then, the vibration monitoring and diagnosis techniques that hold particular relevance for electric motors.

Vibration parameter

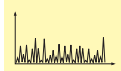


Event-oriented

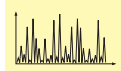
- Parameter trend monitoring
- Alarm notification when tolerances are exceeded
- Reference spectra (good condition)
- Manual in-depth diagnosis / on-site analysis



Offline spectrum
Good condition



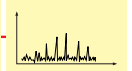
Spectrum
Warn



Spectrum
Alarm



Offline signal analysis
Diagnosis / Analysis



4. Level 1 / Level 2 condition monitoring strategy

Machine condition monitoring calls for measurement of suitable vibration characteristic overall values, which allow the general vibration condition of the machine to be estimated. The trend development of these characteristic overall values points out condition deterioration, i.e. damage progression. This type of overall vibration measurement is characterized as 'Level 1' as shown here. It allows monitoring of many aggregates without imposing high demands in terms of equipment and manpower.

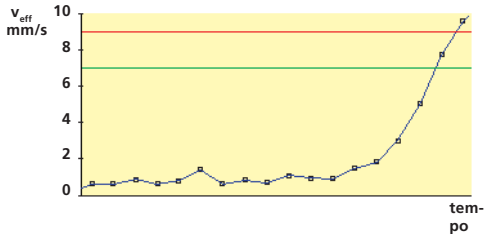
Characteristic overall value (Level 1) measurements such as these, however, are insufficient for precise localization of defects, as this requires closer analysis of the machine spectrum. Most types of damage can be detected by their characteristic frequencies or typical pattern of frequencies. 'Level 2' vibration

diagnosis normally requires measurement of vibration signals using an FFT vibration analyzer by trained personnel who are experienced in interpreting vibration spectra.

Level 1 / Level 2 condition monitoring strategy

Level 1: Parameter trend monitoring

- Comprehensive
- Long-term
- Less-skilled personnel



Machine monitoring

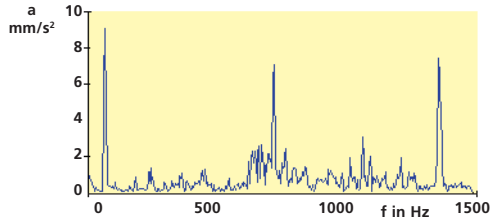
Vibration load
Bearing condition

Parameters

Vibration strength, displacement, acceleration
Shock pulse for bearing evaluation
Temperature
RPM
Pump cavitation

Level 2: Vibration diagnosis following alarm violation

- Isolated
- One-time
- Specialist



Defect localization via spectrum analysis

Rotor unbalance, shaft misalignment, gear damage, turbulence, field faults, bearing diagnosis etc.

Signal analysis

Amplitude spectrum
Envelope spectrum
Time signal
Ordinal analysis
Cepstrum

5. Vibration severity according to ISO standards

DIN ISO 10816-3 plays a very important role for maintenance technicians in the evaluation of machine vibrations. Part 3 of this standard, which is the section that is of relevance to Condition Monitoring, has been revised. Groups 3 and 4 of Part 3, which dealt with pumps, have been removed. Instead, the standard was expanded to include Part 7 – namely, DIN ISO 10816-7. This new part deals entirely with vibrations in centrifugal pumps. The new DIN ISO 10816-7 has been in effect since August 2009.

DIN ISO 10816-3		Group 1		Group 2	
Machine type		Large machines 300 kW < P < 50 MW		Medium sized machines 15 kW < P < 300 kW	
		Motor H > 315 mm		Motor 160 mm < H < 315 mm	
Foundation		flexible	rigid	flexible	rigid
Velocity v_{eff} mm/s rms	11,0		D		
	7,1				
	4,5		C		
	3,5				
	2,8		B		
	2,3				
	1,4		A		
		 Newly commissioned machines	 Unrestricted long term operation	 Restricted long term operation	 Vibration causing damage

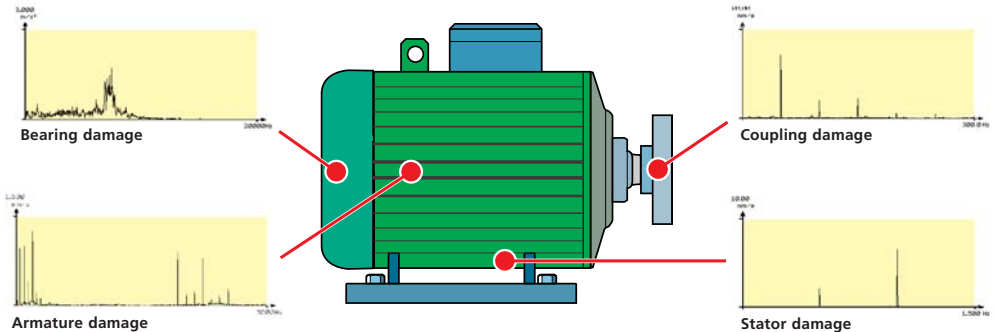
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6. Motor components vulnerable to damage

This illustration gives an overview of the electric motor components most vulnerable to damage. Some types of damage exhibit typical vibration spectra patterns, and each of these phenomena shall now be explained in detail.

Motor components vulnerable to damage



7. Rotor unbalance / Shaft misalignment

Unbalance is understood to be an eccentric distribution of rotor mass. When an unbalanced rotor begins to rotate, the resulting rotating centrifugal force produces additional forces on bearings and rotor vibration at the exact frequency of rotation. This characterizes the spectrum of an unbalanced machine, i.e. the rotation frequency appears as a 'peak' with elevated amplitude, and this can significantly degrade the overall vibration condition of the machine. The necessary redistribution of rotor mass is achieved by balancing the motor rotor either with a balancing machine following disassembly or on-site using a vibration-based balancing instrument. Reference #3 indicates acceptable residual unbalance for rigid rotors.

Shaft misalignment of directly coupled machines results primarily in elevated vibration at

twice the shaft rotation frequency, sometimes with the peak at shaft rotation frequency elevated as well. If the radial misalignment (i.e. shaft offset) dominates, then this peak is most pronounced for measurements taken in radial direction (perpendicular to the shafts). If angular misalignment (coupling gap) is predominant, then vibration elevation will be most noticeable in frequency spectra of axial measurements. Many manufacturers and operators of electric machines have adopted the use of modern laser-optical shaft alignment systems such as OPTALIGN® to correct excessive shaft misalignment. Recommended alignment tolerances are outlined in Note #4.

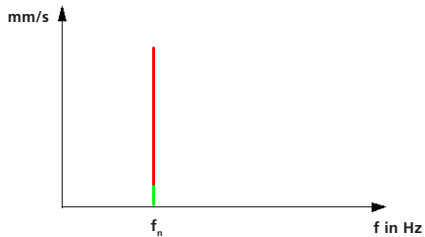
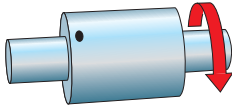
³ ISO 3945

Mechanical vibration of large rotating machines with speed range from 10 to 200 rev/s; Measurement and evaluation of vibration severity in situ, 12/1985

⁴ OPTALIGN® PLUS

Operating instructions and alignment handbook, PRÜFTECHNIK AG, Ismaning, Germany, 03/1997

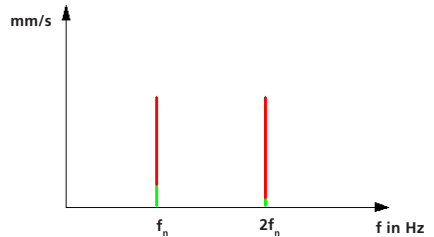
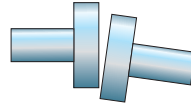
Unbalance



Amplitude of f_n too high

- Rotation frequency $f_n = \text{rpm}/60$
- Evaluation standard: ISO 2372, ISO/DIS 10816-3

Shaft misalignment



Twice (2x) rotation frequency $2f_n$

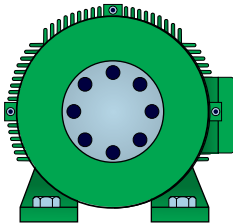
- Radial: radial misalignment
- Axial: axial misalignment

8. Stator field asymmetry

Field asymmetry of electric motors can be caused by stator or rotor (armature) defects. The most common faults are

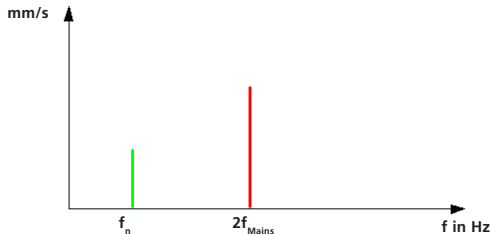
- Motor core short circuiting from armature rubbing or burnout
- Asymmetrical winding
- Asymmetrical power feed and
- Eccentric armature position.

Stator field defects can be recognized in the vibration spectrum as peaks occurring at twice the mains frequency, without sidebands.



Stator field asymmetry

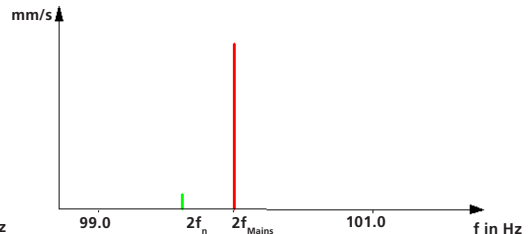
- Core burnout, short circuit
- Eccentric armature position
- Asymmetric power feed
- Asymmetric winding



Twice mains frequency $2f_{Mains}$ visible

Mains frequency $f_{Mains} = 50$ or 60 Hz

Exception: rectifier drives



No sidebands visible around $2f_{Mains}$

2-pole machines:

2x rotation frequency lies just below $2f_{Mains}$

9. Armature field faults

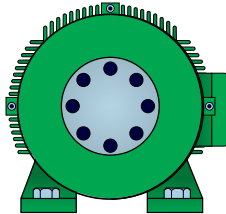
Rotor field asymmetry is caused by:

- Damaged bars (breakage/fracturing, looseness) or
- Short circuited bars or
- Short circuited rings (breakage/fracturing) or
- Short circuited armature packs (e.g. by overloading at excessive speed)

These faults can be detected in the vibration spectrum by the evidence of

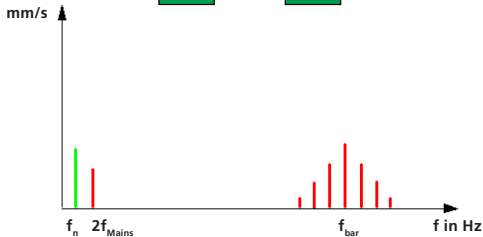
- Bar passing frequency with sidebands at twice the mains frequency and
- Mains frequency with sidebands at slipping frequency.

The only possible remedy here is usually complete replacement of the armature.



Armature field fault

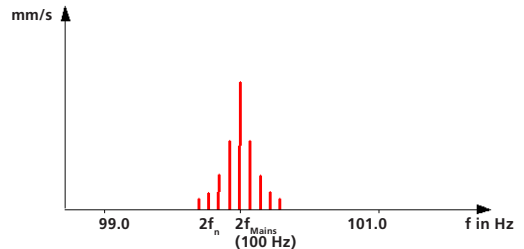
- Bar breakage
- Bar fracture
- Bar looseness



Bar passing frequency f_{bar} with sidebands visible at $2f_{Mains}$ intervals

Bar passing frequency $f_{bar} = f_n \times n_{bar}$ with rotation frequency f_n and n_{bar} = number of armature bars

Mains frequency: $f_{Mains} = 50$ or 60 Hz



Sidebands visible around $2f_{Mains}$ at f_{slip} intervals with slip frequency $f_{slip} = 2f_{Mains}/p - f_n$ and p = number of stator poles

10. Practical vibration diagnosis: Rotor unbalance

The vibration spectrum exhibits a typical unbalance pattern. The levels of vibration severity measured at several locations on the machine point indicate that the source of excitation lies near the coupling. Simple rotor balancing of the brake disk reduced motor vibration to 3.5 mm/s and gearbox vibration to 3.1 mm/s.

Practical vibration diagnosis: Rotor unbalance

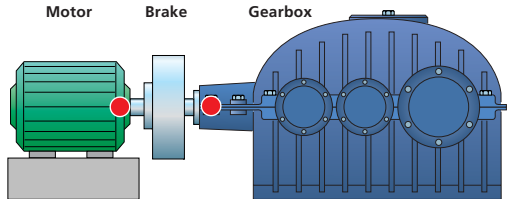
Belt conveyor gearbox

P = 600 kW

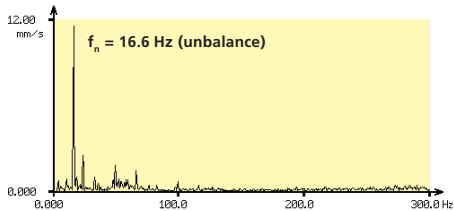
n = 996 rpm ($f_n = 16.6$ Hz)

Vibration severity	Motor	Gearbox
A, RH in mm/s	3.1	-
A, RV	7.8	9.2
A, AX	5.3	6.2
B, RH	4.4	-
B, RV	6.8	-

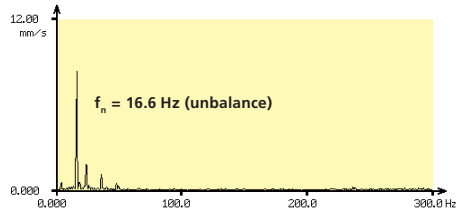
Cause: Brake disk unbalance



Gearbox, inboard bearing, vertical



Gearbox, inboard bearing, axial



11. Practical vibration diagnosis: Shaft misalignment

The vibration spectrum shows a distinct peak at twice the shaft rotation frequency, which clearly indicates shaft misalignment. Following shaft alignment, the peak has disappeared, but the rotor unbalance evident in the previous spectrum remains to be corrected.

Practical vibration diagnosis: Shaft misalignment

Hydroturbine generator

P = 55 kW

n = 1000 rpm ($f_n = 16.67$ Hz)

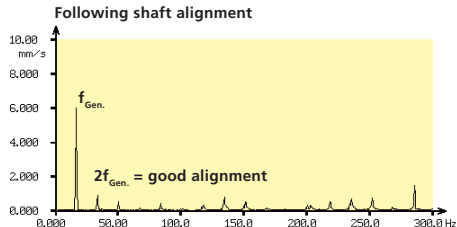
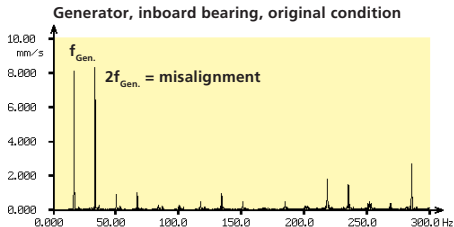
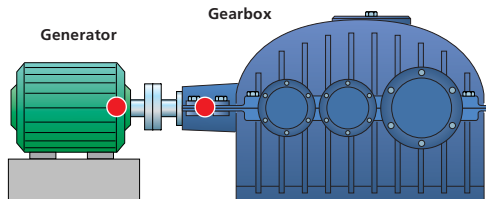
Vibration severity	Generator	Gearbox
--------------------	-----------	---------

inboard, RH	9.5	1.5 mm/s
inboard, RV	4.1	-
inboard, AX	4.4	-

Vertical alignment correction	Before	After
-------------------------------	--------	-------

Angularity ($\varnothing = 170$ mm)	0.42 mm	0.02 mm
Offset	0.44 mm	0.05 mm

Cause: Shaft misalignment



12. Practical vibration diagnosis: Field asymmetry

The motor had drawn attention due to elevated vibration which also occurred with the coupling removed. The unusually high peak at twice the line frequency pointed toward stator damage. Disassembly revealed that the stator packet had burned out due to local short circuiting of the core. The motor had to be completely replaced.

Practical vibration diagnosis: Field asymmetry

Steel mill exhaust blower

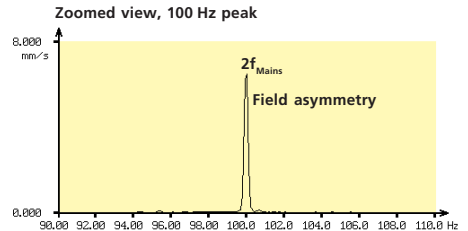
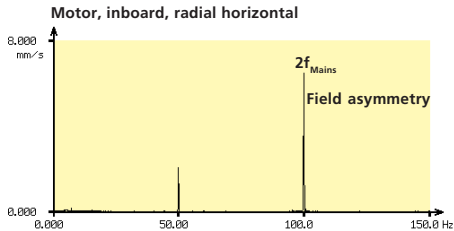
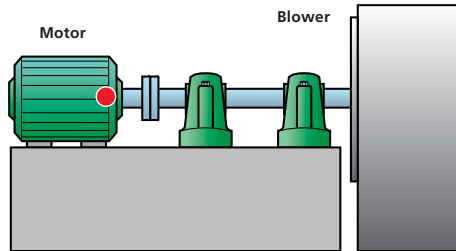
P = 250 kW

n = 2999 rpm ($f_n = 50$ Hz)

Vibration severity

Motor inboard, RH 4.8 mm/s

Cause: Stator core burnout



13. Practical vibration diagnosis: Loose belt drive wheel

A press drive motor had developed severe vibration and was producing unusual noises that had become more pronounced from one day to the next. In stark contrast to the usual vibration spectrum, the rotation frequency was hardly visible at all, but multiples of the rotation frequency were quite obvious. These symptoms remained unchanged when the drive belt was removed from the motor. The source was found to be loose mounting of the belt drive wheel on the motor shaft. The problem was resolved by remachining the motor shaft and reattaching the belt drive wheel.

Practical vibration diagnosis: Loose belt drive wheel

Press drive

P = 200 kW

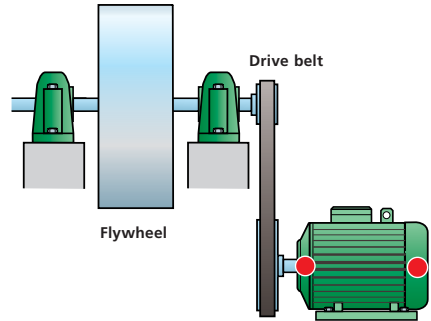
Motor: 1486 rpm = 24.77 Hz

Vibration severity

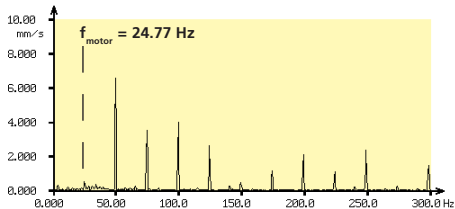
Motor inboard 6.9 mm/s

Motor outboard 7.1 mm/s

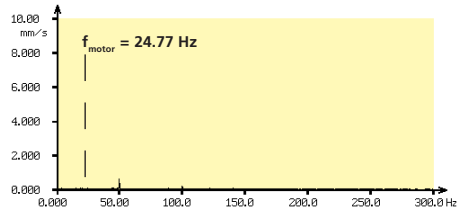
Cause: Excessive play in motor shaft
belt drive wheel



Motor inboard, before repair



Following repair



14. Bearing evaluation characteristic overall values

As a rule, bearing race damage cannot be detected by elevated levels of low-frequency vibration parameters until damage is quite severe. The reason for this is that when the rolling elements pass over a damaged area of the race, a shock pulse is created that can be detected only in the high-frequency range at first. This is why special bearing characteristic overall values were developed for anti-friction bearing monitoring; there is no internationally-accepted standard for these so far, and so a variety of different characteristic overall values can be found in use today.

This illustration lists the most well-known of these bearing parameters. In Germany, for example, the shock pulse method has established itself over the past 25 years as an easy-to-use and reliable measurement technique for monitoring anti-friction bearings. In contrast to all other bearing parameters, this

method uses two parameters for evaluation. The shock pulse maximum value dBm, which indicates the severity of shocks in the rolling behavior of the bearing, is useful in detecting initial damage to bearing races. The 'carpet level' of shock pulses, dBc, indicates the base noise level of the bearing, which increases primarily due to lubrication problems, general wear of races, insufficient bearing clearance or residual stress due to improper installation.

One typical characteristic of all anti-friction bearing parameters is the dependency of their levels upon various influences such as rolling velocity, i.e. bearing size and rpm, signal damping, bearing load and lubrication. This is why it is practically always necessary to take a comparative measurement in good condition or to normalize readings relative to good condition.

■ Shock pulse

■ $K(t)$ method

■ Spike energy

■ BCU value

■ Kurtosis factor

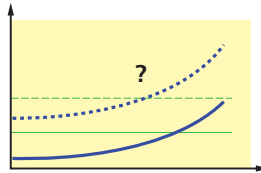
■ GSE factor

■ SEE factor

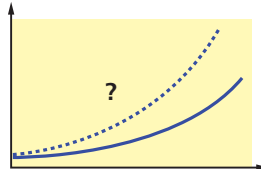
■ Accel. crest factor

Regardless of characteristic overall value:
reliable condition evaluation still requires

Initial value? Tolerances?



Rate of increase over time?



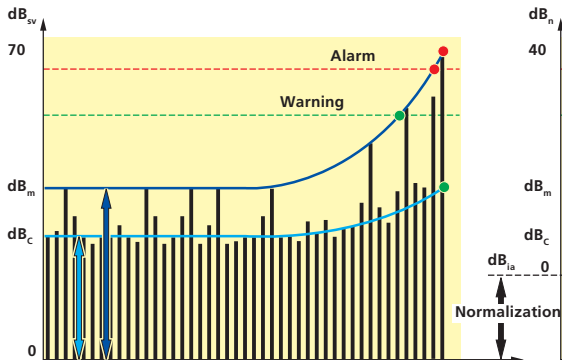
15. Normalization of shock pulse measurement

This illustration shows the normalization procedure that PRÜFTECHNIK instruments use during shock pulse measurement to compensate the influence of rolling velocity differences. The initial level, and in turn the adjusted initial value dBia, are determined by taking a comparative measurement in good condition. This serves as the reference for relative level measurement of maximum shock pulse value dBm and the shock pulse carpet value dBc. This procedure allows measurements from different bearings to be compared using the same level scale so that tolerances do not have to be set individually for every single measurement location.

Normalization of shock pulse measurement

Non-normalized measurement

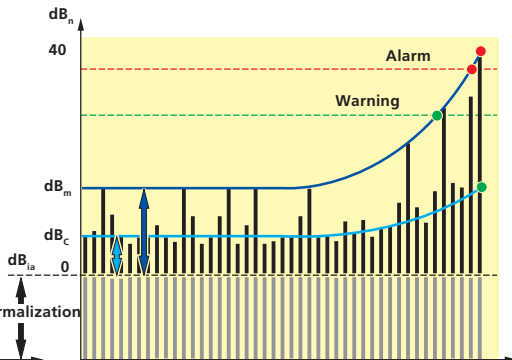
Shock pulse peak value dB_m and carpet value dB_c as absolute level in dB_{sv}



- Threshold (limit) values set individually for every single location

Normalized measurement

Shock pulse max. value dB_m and carpet value dB_c as relative level in dB_{sv} referenced to dB_{ia} value



- dB_{ia} value includes influence factors such as rolling velocity, signal damping, bearing load
- Different threshold limits are linked to the setup dB_{ia} value; the same predefined threshold values are used for all locations

16. Anti-friction bearing damage diagnosis

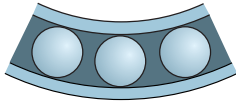
Similarly to vibration diagnosis via frequency spectrum measurement, in-depth diagnosis of anti-friction bearings may be performed through analysis of the signal 'envelope'.

The illustrations here explain the envelope analysis procedure, which begins with filtering out the appropriate range of frequencies that contain the signal emitted by the bearing during operation. This signal component is examined for the pulses that arise when bearing elements roll over damaged locations. Demodulation is used to calculate a curve that 'envelops' the bearing signal. If the time interval between periodically-occurring peaks in the envelope curve match one of the critical frequencies characteristic of bearing damage, then the corresponding bearing component can be assumed to be damaged.

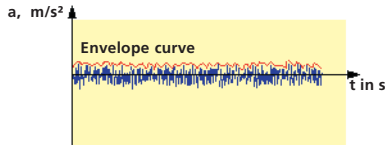
This procedure allows extremely accurate diagnosis of damage to anti-friction bearings, even in cases where extraneous signal components such as gear meshing noise tend to cover up the actual bearing signal. It does require knowledge of certain geometric data of the bearing, including the bearing diameter, the number and diameter of rolling elements, the load angle and the operating speed.

Anti-friction bearing damage diagnosis

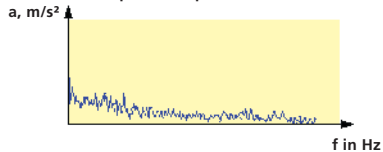
No damage



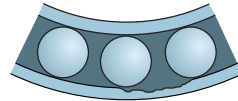
Time signal



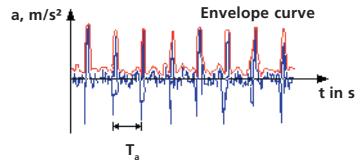
Envelope curve spectrum



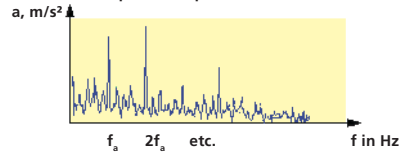
Damage



Time signal



Envelope curve spectrum



Damage frequency $f_a = 1/T_a$

17. Practical bearing diagnosis: inner race damage

This shows an example of advanced damage to the inner race. The great increase in shock pulse levels, especially that of peak value dB_m from 18 to 48 dBsv, signified serious bearing damage. Envelope spectrum analysis indicated a pattern typical of inner race damage, which was then confirmed following bearing replacement: one of the two races of the inner ring already exhibited a damaged surface area of about 15 x 15 mm / 5/8" x 5/8".

Practical bearing diagnosis: inner race damage

Paint shop exhaust fan

P = 110 kW

Motor: 1307 rpm = 21.78 Hz

Fan: 908 rpm = 35.75 Hz

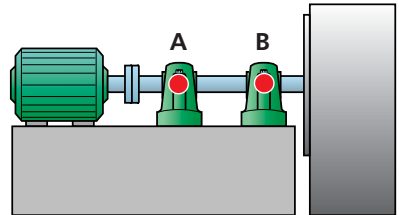
Bearing: 22218 tapered roller bearing

Shock pulse readings dB_m dB_c

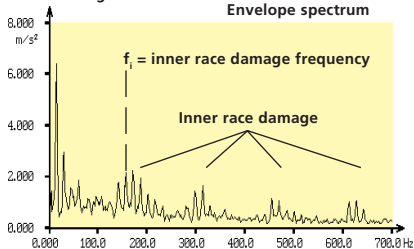
Inboard bearing A 48 29 dB_{SV}

Outboard bearing B 18 7 dB_{SV}

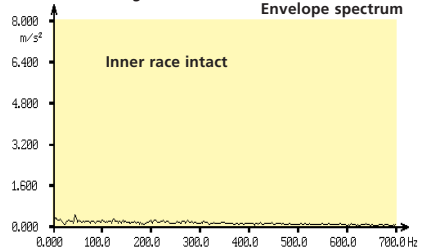
Cause: Severe inner race damage on inboard bearing



Inboard bearing A



Outboard bearing B



Productive maintenance technology



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