TECHNICAL UNIVERSITY OF LIBEREC

Faculty of Mechanical Engineering

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TECHNICAL DIAGNOSTICS OF EMAG MACHINE TOOLS

Diploma Project

2004

TECHNICAL UNIVERSITY OF LIBEREC

Faculty of Mechanical Engineering

Study program 23-01T - Mechanical Engineering

Study discipline: Manufacturing Systems

Study direction: Flexible Manufacturing Systems for engineering production

Department of Manufacturing Systems

TECHNICAL DIAGNOSTICS OF EMAG MACHINE TOOLS

TECHNICKÁ DIAGNOSTIKA OBRABĚCÍCH STROJŮ EMAG

Abdulhadi A. M. SADEQ

KVS - 146

Supervisor: Dr. Ing. Elias TOMEH – TU Liberec - KSD

Consultant: Dr. Ing. Elias TOMEH – TU Liberec - KSD

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TECHNICAL UNIVERSITY OF LIBEREC

Faculty of Mechanical Engineering

Department of Manufacturing Systems

School Year: 2002/2003

DIPLOMA PROJECT

Graduate name and surname: **ABDULHADI A. M. H. SADEQ**

Study discipline: 2301 T **Manufacturing Systems**

According to the Law Digest No. 1111/1998 for the Universities, the Head of the Department of Manufacturing Systems determines you this Following Topic for your Diploma Project:

Project heading: **Technical Diagnostics of EMMAG Machine Tools**

Thesis Content:

For EMMAG machine tools, which produce gearbox components for automobile ŠKODA Auto Mladá Boleslav, carry out the following:

1. Introduction, analysis of recent state.
2. From the technical documentation, calculate the frequency of the main sources of vibration.
3. Design and indicate vibration measurement places.
4. Choose suitable methods of vibration diagnostics for bearings.
5. Find out the effect of vibration on the components produced, and measure the vibration of EMMAG machine tools.
6. From measuring analysis, compare the technical state of EMMAG machine tools.
7. According to norms ČSN ISO, identify the limit values of vibration severity for the evaluation of the state and probable service life, then suggest the methods of measuring the vibration of machine tools.
8. Summary, conclusions, assessment.

Text: approx. 50 pages

Drawings: as necessary

References:

1) Papers from company Škoda Auto, a.s. Mladá Boleslav.
2) Papers from company Breűl & Keajr and SKF.
3) Randall, B.: Frequency analysis. Breűl & Keajr 1987.
4) Jens Trampe Broch: Mechanical vibration and shock measurements. Breűl & Keajr 1984.
5) Beneš, Š., Tomeh, E.: Metody diagnostiky valivých ložisek. Skripta TU Liberec, 1991.
6) Frohlich, J.: Valivá ložiska ZKL. SNTL Praha 1980.
7) Janoušek, I. a kol.: Technická diagnostika. Praha 1988.

Supervisor: Dr. Ing. Elias Tomeh, TUL, KSD Liberec

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Doc. Ing. Josef Cerha, CSc.
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Dean

Liberec 1.2.2003

Validity of this project task is 15 months from the date above. The terms of submission are given for each academic year by particular study schedule.

ANOTATION

TECHNICAL UNIVERSITY OF LIBEREC

Faculty of Mechanical Engineering

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Study program 23-01T - Mechanical Engineering

Diplomant: Abdulhadi A. M. SADEQ

Theme of diploma project:

TECHNICAL DIAGNOSTICS OF EMAG MACHINE TOOLS

TECHNICKÁ DIAGNOSTIKA OBRABĚCÍCH STROJŮ EMAG

Number of Diploma Project: 146

Supervisor: Dr. Ing. Elias TOMEH – TU Liberec - KSD

Consultant: Dr. Ing. Elias TOMEH – TU Liberec - KSD

Abstract:

The work is focused on the diagnostic of the EMAG Machine Tools, which produce gearbox components for automobile ŠKODA Auto Mladá Boleslav, and it comes up with methods of technical diagnostics. One of them is used for monitoring the technical condition of the EMAG Machine Tools. Measured values are compared with the ISO. Also they specify the actual condition of the EMAG Machine Tools. Also, other advisable actions are designed for monitoring the technical condition of the EMAG Machine Tools in the future.

Stručný výtah:

Tato práce se zabývá diagnostikou obráběcích strojů EMAG, a návrhem metod technické diagnostiky, z nichž je jedna použita k sledování technického stavu obráběcích strojů EMAG. Naměřené hodnoty jsou porovnány s hodnotami ISO. Také je určen součastný stav obráběcích strojů EMAG. Dále jsou zde navrženy další vhodná opatření k sledování technického stavu obráběcích strojů EMAG do budoucna.

Declaration

„I declare that I developed my diploma project independently with the aid of the referred literature under the control of the Supervisor and Consultant.“

In Liberec, 31.5.2004

Signature

ACKNOWLEDGEMENT

At the beginning of my diploma project I would like to thank my parents and family for their support and encouragement during my study.

I also thank my supervisor and consultant Dr. Ing. Elias Tomeh from the Department of Transport Machines of the Technical University in Liberec for his valuable contribution in solving this project.

And special thank to Doc. Ing. Přemysl pokorný, CSc. and every teacher in this University who advised me and helped me accomplish my study at the faculty of Mechanical Engineering in the Technical University of Liberec.

CONTENT

THE USED SYMBOLS

n	$[min^{-1}]$	Number of motor rotation
$n1$	$[min^{-1}]$	Number of spindle rotation
f_r	[Hz]	Rotor frequency of motor
f_r	[Hz]	Rotor frequency of spindle
f_i	[Hz]	Inner ring damage frequency
f_k	[Hz]	Bearing cage damage frequency
f_o	[Hz]	Outside ring damage frequency
f_v	[Hz]	Bearing ball damage frequency
d_o	[mm]	Diameter of bearing ball
e		Bearing coefficient
D	[mm]	Diameter of outside ring
d	[mm]	Diameter of inner ring
d_s	[mm]	Pitch diameter
t	[s]	Time
U	$[mm.s^{-1}]$	Vibration severity
α	[°]	Contact angle
v	$[mm.s^{-1}]$	Velocity of vibration
? v_{ef} = RMS	$[mm.s^{-1}]$	Effective value of vibration velocity (Root mean square value)

$v_{ef} = v_{RMS}$

1. INTRODUCTION

Nowadays, we put heavy demands on production equipment and their maintenance. Thanks to the efforts done to insure the continuity of production, maintenance time is shortened to the minimum. From this point of view, defects prevention itself plays a very important role.

The efforts leading to insure running trim, reliability, and durability of production equipment are urged by huge costs of repairs and by the fear of loss of production that has a decisive importance to insure the economic success of the whole enterprise.

MAINTENANCE has developed together with the development of production equipment. It is all the measures leading to preserve the required condition by means of different service activities as cleaning, conservation, lubrication, filling, replacing operating refill, and additional adjustment or tuning. And of course, there are **REPAIRS** not only of the equipment itself but also by means of renovation, modernization and new setting. As a result to the mentioned increase of production equipment requirements and costs, a new element was added to maintenance and it is **INSPECTION**. It is the measures taken to find out and evaluate the real condition of machines, equipment, construction groups, or construction elements. Inspection is a matter of information obtained by measuring, testing, detecting and controlling. **TECHNICAL DIAGNOSTICS** deal with this particular area and is the subject of my diploma project.

Technical Diagnostics began to develop about 40 years ago, when the diagnosis of machines condition used to be a subjective evaluation. The service personnel used especially touch, hearing and sight. Of course, this was not enough for accurate diagnosis, so, in 1960, a so-called *conventional condition monitoring* began to be carried out. It already used global vibration measurement and compared it to limit values. Even if there was a huge difference between subjective and conventional monitoring of machines condition in machine tool centers, the diagnosis made in the conventional method was inaccurate and insufficient. Therefore, from 1985, diagnostical monitoring of machines condition began to be used. By the help of analytic vibration measurement, and multiparametrical monitoring of limit values relying on operating regime and automatic diagnostics, we get a big amount of information about the present condition, changes in condition and about the operating behaviour of the machine.

Thanks to this information and to the early recognition of defects, the appearance of big resulting damages can be avoided, and the operating personnel and the surrounding environment can be protected. Also, thanks to the continuous machines monitoring, the numbers of reviews, inspections and unplanned breakdowns have decreased. Furthermore, by this method it is possible to reduce the period of review and the costs of repairs, and at the same time to prolong life expectation of equipment.

Technical diagnostics do not only include the analysis of machine tool centers but also the analysis and composition of corresponding mathematical models, and the research and design of concrete diagnostical equipment.

There are two forms of technical diagnostics:

- Limited diagnostics, which ensure, detection of the faults during operation, monitoring some parameters (temperature, pressure, number of rotations, mechanical vibration etc.).

- Immediate diagnostics, which is the subject of my diploma project, consists of detecting the momentary state or a possible future inadmissible state and informs us about the future conditions of the machine. It also informs us about the possible cause and the size of the problem.

2. EMAG MACHINE TOOLS

EMAG machine tools are vertical turret lathes. They serve to produce gearbox part for SKODA automobile. After being machined by these machine tools, the intermediate product is prepared for the teeth milling, drafting inner cuttings and heat treatment. The machines are placed in production halls M2 and M6.

In Skoda Auto plant, we find Turret lathes EMAG in two types:

- one-shaft, marked VSC 200 (figure1),
- and double-shaft.

The double-shaft is also in two different types: in one of them the two shafts rotate independently VSC 200 DUO (figure 2) and simultaneously VSC160 TWIN (figure 3).

Figure 1 One-shaft turret lathes EMAG

Figure 2 Double-shaft turret lathes EMAG DUO

Figure 3 Spindle of type VSC 200 a VSC 200 DUO

Figure 4 Double-shaft turret lathes EMAG

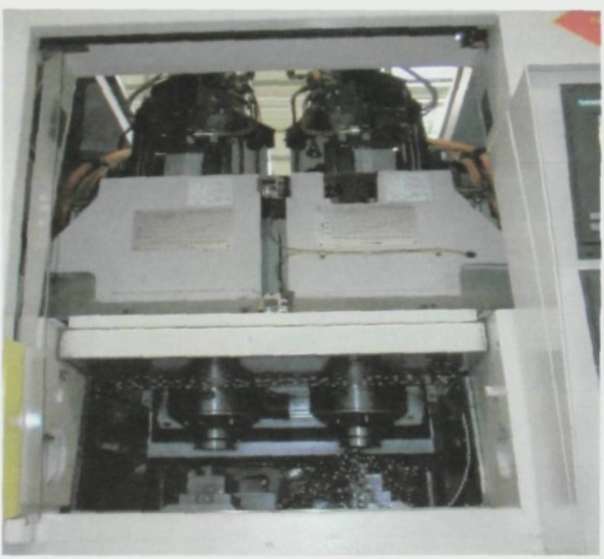

Figure 5 Spindle of type VSC 160 TWIN

2.1 PRODUCTION HALL M2

Production hall M2 is operating from 1970. In the present time, there are produced the four cylinders Skoda motors 1.4 and part of gearbox MQ 200. In this hall there is also the maintenance center and 15 tool machines EMAG (figure 6).

Figure 6 Production Hall M2

2.2 PRODUCTION HALL M6

In the production hall M6, there are 12 tool machines EMAG. In this hall, the production began in the beginning of 2001. In the hall M6, three cylinder motors EA111 1.2 2V/4V are produced from august 2002. Also, production and mounting of gearbox MQ 200 are carried out. The hall M6 is called the new motor hall. (Figure 7)

Figure 7 Production Hall M6

3. THE DRIVE OF EMAG TOOL MACHINES

EMAG tool machines are driven only directly, when the motor is fixly related to the spindle and creates a compact unit (figure 8). In this drive method, the number of motor rotations is the same as the number of spindle rotations.

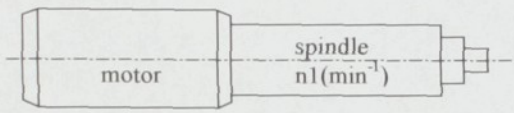

Figure 8 Direct drives of EMAG tool machines

3.1 THE MAIN SOURCE OF EXCITATION

An ideal machine would not cause any vibration. All its energy would be changed into the productive work for which the machine was designed. In practice, nevertheless, Vibration appear as a side product of dynamical forces transfer through the mechanical system. The elements of the machine react to each other and a variance of energy takes place in the system in form of vibration.

Among the main sources of excitation (figure 9):

1) Rotors, 2) belts, 3) gears and 4) bearings

In our case, excitation happens only from bearings and rotors. A screw drive occurs to move the spindle up or down.

– excitations from el. motor (el.-mg. field)

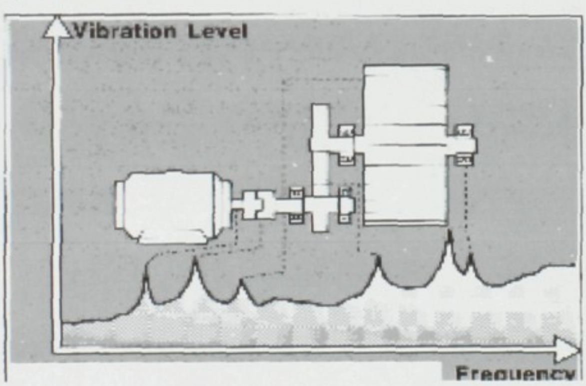

Figure 9 The main sources of excitation

3.1.1 Fault in rotating shafts

Two of the most common faults associated with rotating shafts is unbalance and misalignment. Unbalance produces a component at the rotational frequency of the shaft, mainly in the radial direction. A misaligned coupling, however, will produce a component at the rotational frequency, plus usually its lower harmonics, both in the axial and radial directions.

higher

Misaligned bearings produce a similar symptom, except that the higher harmonics also tend to be excited. A bent shaft is just another form of misalignment, and will produce vibration at the rotation frequency and usually its lower harmonics.

The cracked shaft produces an increase in the vibration at the rotational frequency and the second harmonics.

Phase measurements can be used with rotating shafts to distinguish, between a rocking motion and a bent shaft phase measurement can also be used to distinguish between a force or couple unbalance, by measuring the phase difference between the vibration at the bearings in the radial direction. With a force unbalance, there will be no phase difference at the rotational frequency of the shaft, while with an imbalance, it will be approximately 180°. Q1

3.3.2 Faults in bearings

Discrete faults in ball or roller bearings results in a series of bursts. This illustration shows how a discrete fault causes a series of bursts with a repetition frequency given by the bearing geometry and rotational speed. The frequency content of the bursts is high (dominated by the resonances excited) and the component at the repetition frequency is small. If the envelope of the bursts is formed, however, its frequency spectrum is dominated by the repetition rate (and its harmonics).

It is possible to calculate the repetition frequency of the bursts using simple classical mechanics. However, note that the relationships assume pure rolling motion; while in reality there is some rolling and some sliding motion. Hence the equations should be regarded as approximate. Also amplitude modulations can produce sidebands.

3.2 CALCULATION OF DIRECT DRIVE FREQUENCY

$n = n_1$ *(The number of motor rotations = the number of spindle rotations)*

The rotor frequency of motor = the rotor frequency of spindle

$$f_r = \frac{n}{60} = \frac{n_1}{60}$$

Measurements of vibration will be carried out on all the machine tools for a number of rotation 1000 RPM. Then:

$$f_r = \frac{1000}{60} = 16{,}66 Hz$$

3.3 CALCULATION OF BEARING FREQUENCY

When the inner and outer ring is damaged, the bearing will be a marked vibration excitor. To calculate the damage frequencies of bearing, it is necessary to know the diameter of bearing which can be taken from the bearing catalogue for each type of bearing, and the number of rotations. The producers of bearings provide a program to calculate the damage frequencies of bearing. To calculate these frequencies, it is enough to know the type of bearing and its number of rotations. On the basis of these information the program calculates:

- Inner ring damage frequency $f_i = \frac{n}{2} fr(1 + \frac{d0}{ds}\cos\alpha)$
- Bearing cage damage frequency $f_k = \frac{1}{2} fr(1 - \frac{d0}{ds}\cos\alpha)$
- Outside ring damage frequency $f_o = \frac{n}{2} fr(1 - \frac{d0}{ds}\cos\alpha)$
- Bearing ball damage frequency $f_v = fr\frac{ds}{d0}[1 - (1 + \frac{d0}{ds}\cos\alpha)^2]$

$$ds = \frac{D + d}{2}$$

n .... Number of balls

f_r .... Rotation frequency

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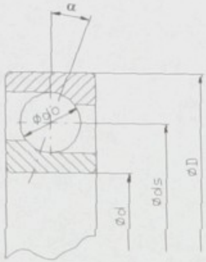


∅D- Diameter of outside ring

∅d- Diameter of inner ring

∅d_s- Pitch diameter

∅d_0- Diameter of bearing ball

α- Contact angle

Figure 10 Inner bearing dimensions

In EMAG machine tools, spindles are of two types. The 200 and 200 DUO have identical spindles and the 160 TWIN have different spindles.

3.3.1 THE CALCULATION OF BEARING FREQUENCY FOR TYPES 200 AND 160

Machine tools EMAG of type 200 use ball bearings double-rows angular-contact B7016E and roller bearing N1011K. In both types of spindles the same bearing is used. In the type 160 TWIN, only in the place of roller bearing, ball bearings B7011E are used. The frequency of every bearing in EMAG machine tools is calculated in table 1.

Table 1 The calculated frequency of bearings in EMAG machine tools

Machine tool	EMAG 200, 200 DUO			EMAG 160 TWIN		
Bearing	B 7016 E	B 71916 E	N 1011K	B 7016 E	B 71916 E	B 7011 E
f_i (Hz)	186,6	218,2	212,8	186,6	218,2	169,4
f_o (Hz)	146,8	181,8	170,5	146,8	181,8	130,7
f_v (Hz)	124,7	164,9	149,2	124,7	164,9	115,2
f_k (Hz)	7,3	7,6	7,4	7,3	7,6	7,3

How it was calculated? - If it is calculated - where are the main sizes of bearings? (mech. parame

4. THE METHODS OF BEARINGS DIAGNOSTICS

Nowadays their are several methods of bearings diagnostics that rely on different principles. All of these methods aiming at discovering the condition of the bearing come from one remark, it is that the rotation of the damaged element in the bearing causes shocks and the bearing is an important vibration excitatory.

4.1. SHOCK PULSE METER (SPM)

The principle consists in measuring and judging the shock pulses that appear in bearings. The sensor of shock pulses and the instrument, evaluate the absolute dimension of the pulses in the resonance ranges from 30 to 40 kHz. This method is used in the SPM Instruments MEPA 10, SPM 21, SPM 43, TMED 1 and BEA-52.

4.2. SPIKE ENERGY METHOD

Spike energy SE - as a method of bearing diagnostic comes from three measurement parameters:

- The mean value of acceleration in the frequency range 5 - 50 kHz,
- Pulses frequency,
- Pulses acceleration peak.

These three measured parameters create then "Spike Energy" that is a reliable indicator of the bearing condition the curve obtained by SE method gives a more timely and objective information about the occurring changes in the bearing than the value of acceleration or vibration velocity.

This principle is used in the instrument Reutlinger IRD 811 and IRD 820. The spike energy method can be resumed as following:

- Transducer resonance 25 - 35 kHz Supplementary high - pass filter 15 kHz
- Meters usually include normal vibration measurement.

Shock Pulse Meter and Spike Energy Method assume that the background noise will be low, and the bearing fault signal high, in the vicinity of the transducer resonance. This is generally true for relatively quiet machines such as electric motors, but often not true for pumps (with high frequency turbulence and cavitations) or complex gearboxes (with high frequency gear mesh harmonics).

Even though all the above techniques work under favourable conditions, a certain percentage of cases will be missed, due to unfortunate combinations of circumstances, because of the fixed transducer resonant frequency.

4.3. METHOD BCU (BEARING CONDITION UNIT)

This method is used in the instruments of SCHENCK Company. The shock pulses of the damaged bearing excite the vibration in the accelerometer placed on the bearing in the resonance range from 36 to 38 kHz.

what is that resonance? (of which ...)

4.4. METHOD K(t) PARAMETER

The method was defined by Prof. Sturme~~m~~:

$$K(t) = \frac{a_{ef(0)} . a_{v(0)}}{a_{ef(t)} . a_{v(t)}}$$

where:

$a_{ef\,(0)}$, $a_{ef\,(t)}$ - the effective value of acceleration in times (0) and (t) [mm.s^{-2}],

$a_{v\,(0)}$, $a_{v\,(t)}$ - acceleration peak in times (0) and (t) [mm.s^{-2}].

Robotron Company uses this method in the instrument M 1302. Fig. 3 shows a typical cause of the parameter K(t) on the period of bearing operation.

4.5. KURTOSIS METHOD

KURTOSIS method is based on finding out the statistic K-factor that is defined:

$$K = \frac{\int_{-\infty}^{\infty} (x - x) P_{(x)} . dx}{\sigma^4}$$

where:

x = signal amplitude, x = mean value of x

p(x) = probability density of x, σ = standard deviation of x

This principle is used in the instrument KURTOSIS K-4100. The instrument gives estimation about the condition of the bearing on the basis of information concerning measured vibrations in five discrete frequency ranges.

Frequency range

Interval 1- K1 2.5 to 5 kHz

Interval 2- K2 2.5 to 10 kHz

Interval 3- K3 10 to 20 kHz

Interval 4- K4 20 to 40 kHz

Interval 5- K5 40 to 80 kHz

If the value of K is different from 3 in any frequency range, the instrument notes this fact and analyses the measurement in consideration with the chosen range.

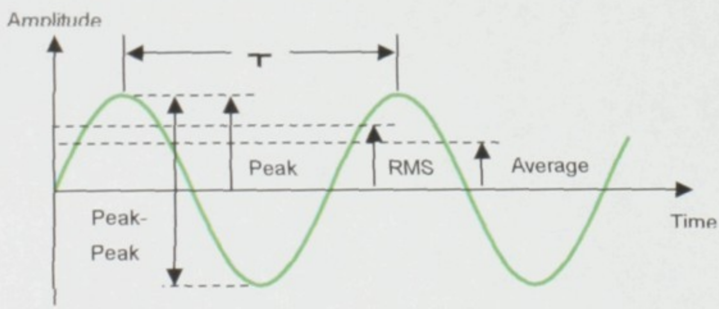
The KURTOSIS method can be defined as:

- Fourth power heavily weights spikes relative to RMS - A sort of normalized crest factor,
- Can be calculated for separate frequency bands (e. g. octave bands) to separate background noise, and follow trends.

4.6. CREST FACTOR METHOD

The crest factor CF is the proportion of the peak to the effective value of vibration RMS. This proportion is evaluated with time sequences in the frequency range 10Hz to 10kHz. The equetion and figure shown below:

$$CF = \frac{PEAK}{RMS}$$



The principle is used in Brüel & Kjaer 2513. We obtain the best results by measuring the acceleration in an interval of higher frequencies (example 1 - 10 kHz).

Crest factor method can be resumed as following:

- Ratio of peak to RMS value
- Typically acceleration to emphasize high frequency components
- Possibly high - pass filtered 1 kHz to remove background vibration

Initially, there is a relatively constant ration of peak to RMS value. As a localized fault develops, the resulting short bursts increase the peak level substantially, but have little influence on RMS level. The peak level will typically grow to a certain limit.

As the bearing deteriorates, more spikes will be generated per ball-pass, finally influencing RMS levels, even through the individual peak levels are not greater. Towards the end of bearing life, the crest factor may have fallen to its original value, even though both peak and RMS levels have increased considerably. The best way to trend the data is as illustrated: RMS and peak levels on the same graph, with Crest Factor inferred as the difference between the two curves.

4.7. METHOD Q

It is a new method in evaluating the technical condition of bearings. The instrument Diagnostic D 016 FEL uses the new method that evaluates the proportion between the mean amplitude of the ultrasonic signal emitted by the bearing and the peak amplitude. The measurement is carried out in the frequency 40 kHz with range width 10 kHz.

4.8. METHOD ENVELOPE SPECTRUM

The principle of this method is used in the analyser 2515 Brüel & Kjaer 2515. Discrete faults in the elements of a ball or roller bearing give rise to a series of impacts at a frequency determined by the location of the fault: outer race, inner race, etc. The initial impulses are so short, in particular when the faults are still microscopic, that their frequency content extends up to perhaps 300 Hz. The shocks excite structural and other resonance's, including the resonance of piezo-electric transducers used to detect them, and produce a series of bursts, as illustrated, with a frequency content dominated by these resonance's. Other background vibrations from the machine mask this bearing signal, and the basic problem is to find a frequency range where the bearing signal is dominant over the background vibration. Note that analysing the envelope of the bursts, rather than the raw signal better indicates the repetition frequency.

Even though bearing faults can be detected and diagnosed in the spectrum as families of harmonics with a spacing equal to the repetition frequency, it is not known a priori where they will show up. Speed must also be stable to give a separation of high order harmonics.

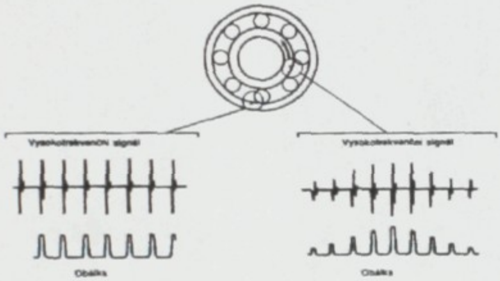


Figure 11 Local damage outside and inside diameter

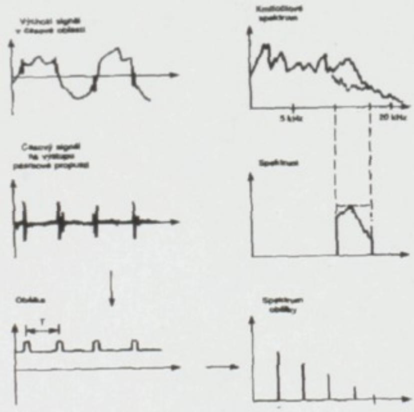


Figure 12 basic principles of envolope detection and analysis

The Envelope Analysis can be resumed as following:

- Discrete faults produce a series of high frequency bursts,
- Burst repetition rate locates source as outer race, inner race etc.,
- Difficulties of extracting repetition rate from raw signal.
    1. Masking by background noise
    2. Very low component at repetition frequency
- Repetition rate to be found as:
    1. Harmonic spacing at high frequencies
    2. Fundamental frequency of envelope signal

very briefly explanation

Use of an FFT Analyser to generate and analyses Envelope Signal

1. Use zoom to band pass filter signal,
2. Resulting complex time signal is analytic i. e. real and imaginary parts related by Hilbert transform,
3. Obtain envelope (amplitude) of analytic time signal,
4. Re - analyse envelope signal as a base band analysis to obtain repetition frequencies.

4.9. METHOD SEE (SPECTRAL EMITTED ENERGY)

The vibration detected with a wide - range signal is filtered from the sensor at about 300 kHz and its low frequency component is demodulated. All the low frequency repeated components are kept. Other signal analysis is carried out in the low frequency range. The calculated spectrum is compared to the frequencies that appear as a result of defections in the bearing. The principle is used in the instrument KIT CMVA 10 MICROLOG SKF Company.

5. THE MECHANICAL VIBRATION

The bearings are from the most strained parts in most rotating machines. The awareness of their condition is necessary because determining their life on the basis of calculations is a very difficult and not always a successful matter. In practice, there are two extremes. The first is to exchange the bearings too early. In this case, breakdown is avoided but the costs of maintenance are very expensive and the maintenance staffs become unnecessary. And the second is to exchange the bearings too late, after failure. In this case, the costs increase because after bearings failure or serious damage, further damage on other parts of the machine occur, and unplanned interruption of production.

These problems disappear by technical diagnostics monitoring of bearings. Bearings will be exchanged in the optimal wear out period. Furthermore, it is possible to determine this moment in advance, as a prognosis and to include it to the stand off plan. Figure 13.

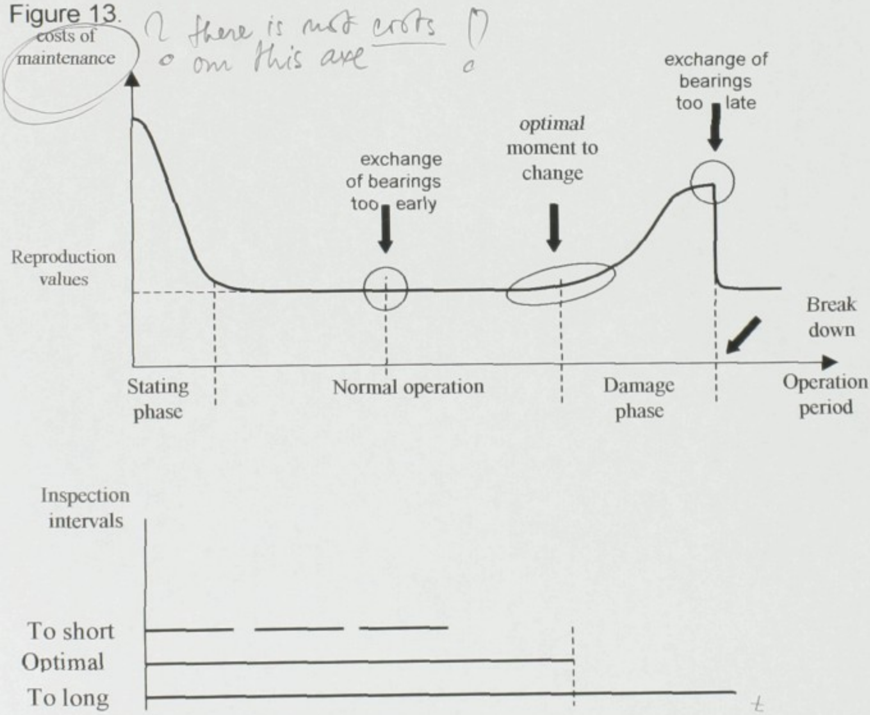


Figure 13 Costs of maintenance during service

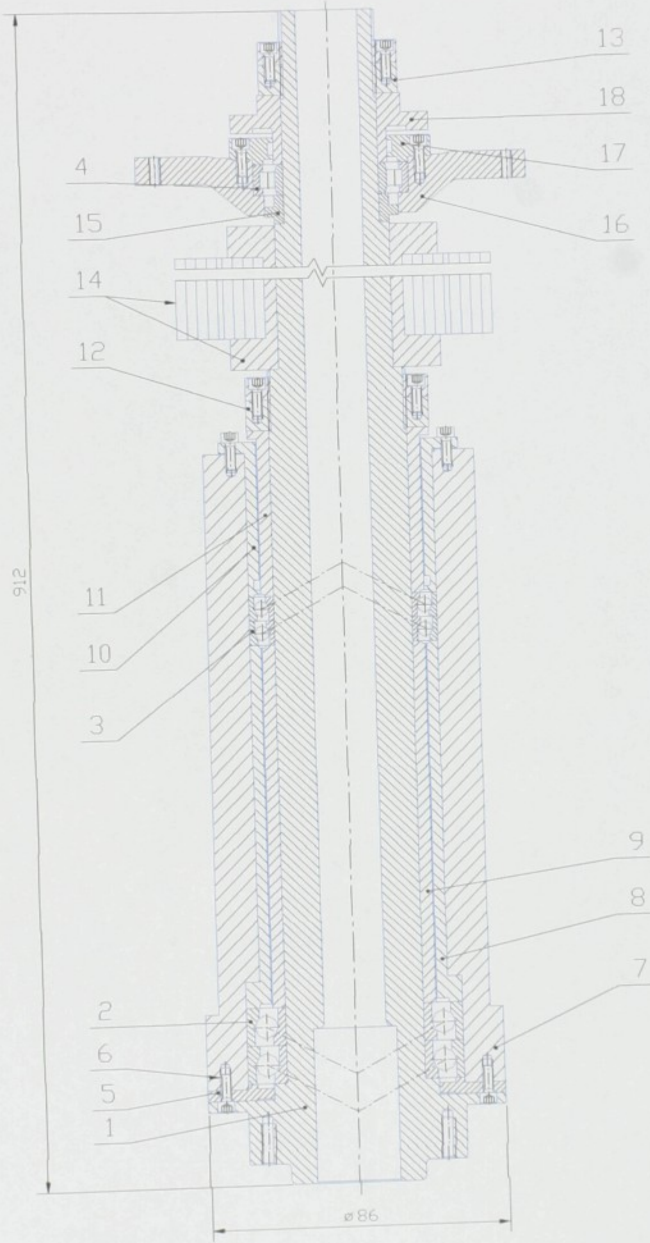


Figure 14 Spindle of type VSC 200 a VSC 200 DUO

Table 2 List of spindle type VSC 200 a VSC 200 DUO

Position	Part	Marking
1	Shaft	
2	Bearing	B 7016 E
3	Bearing	B 71916 E
4	Bearing	N 1011
5	Case of flange	
6	Screw	M6x16 ČSN 02 11 43
7	Strutting cylinder 1	
8	Strutting cylinder 2	
9	Strutting cylinder 3	
10	Collar 1	
11	Collar 2	
12	Set screw	MSR 75x1,5
13	Set screw	MSR 55x1,5
14	Motor	AC1 MB 200 E
15	Casing	81,5x57x18,8
16	disc	
17	Flange 1	
18	Flange2	

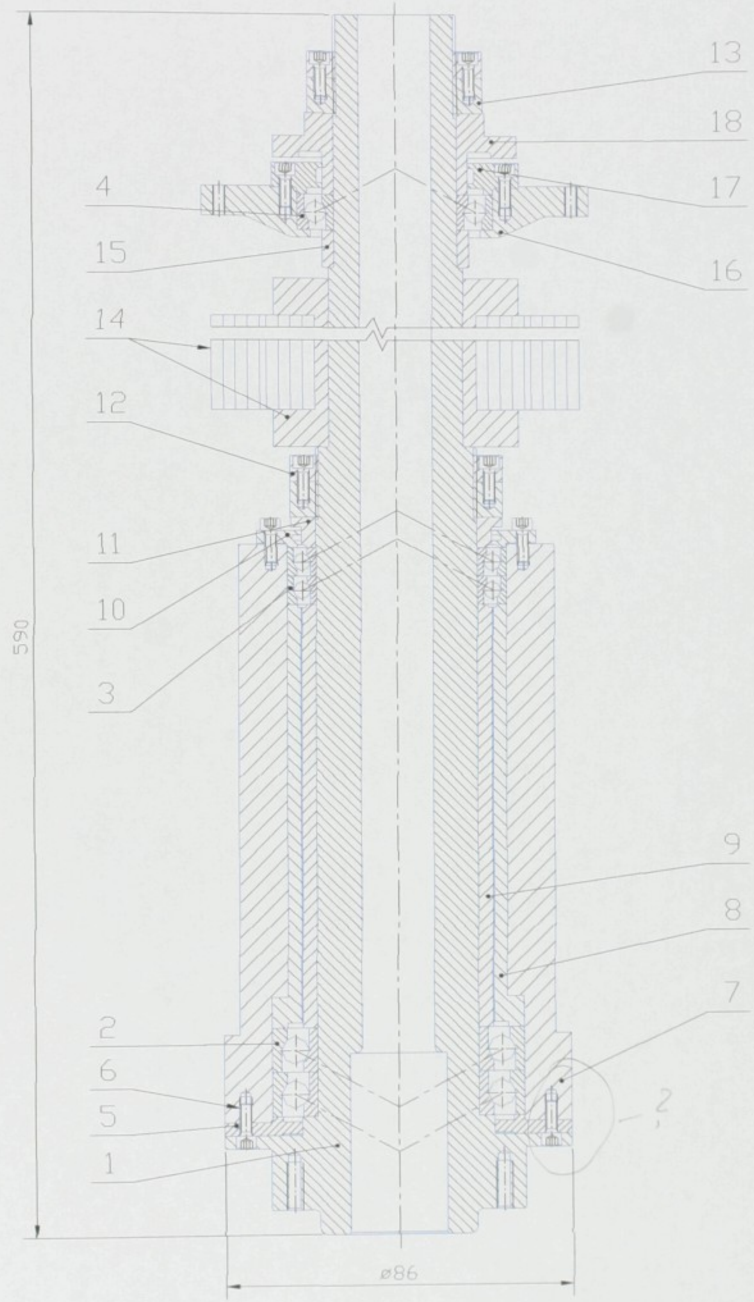


Figure 15 Spindle of type VSC 160 TWIN

Table 3 List of spindle type VSC 160 TWIN

Position	Part	Marking
1	Shaft	
2	Bearing	B 7016 E
3	Bearing	B 71916 E
4	Bearing	B 7011 E
5	Case of flange	
6	Screw	M6x16 ČSN 02 11 43
7	Strutting cylinder 1	
8	Strutting cylinder 2	
9	Strutting cylinder 3	
10	Collar 1	
11	Collar 2	
12	Set screw	MSR 75x1,5
13	Set screw	MSR 55x1,5
14	Motor	AC1 MB 160 N
15	Collar	
16	Disc	
17	Flange 1	
18	Flange 2	

During service every machine emits vibration, which is a very harmful phenomenon. It increases considerably stress on the parts, often to the limit of the material strength, causing shocks that disturb the surroundings and create noise. In EMAG machine tools, vibration interferes disturbingly in the operating process because, apart from the working environment, it affects negatively the quality of the parts of gearbox MQ 200.

The vibration is caused by tolerance and imprecision, that happens during the production and the assembly of each component, deformation under the effect of insufficient solidity, and sometimes-unsuitable operational conditions. So the machine creates a structure succumbing to the deformation process. If there isn't any description of the deformed system, its parameters, its load and motion dependence, and if we need to verify this description practically, we would have to use information that the measurement determines.

5.1 THE CHOICE OF VIBRATION MEASUREMENT PLACES

In the damaged bearing, the strongest impulses appear in the loaded areas and from there they are transferred to the bearing box. That is why we should place the pulses sensor in the most loaded areas of the bearing. For a periodical vibration measurement of the EMAG machine tools, we determined 1 place on the spindle, on the basis of the analysis of the design solution and the knowledge of the machine functions. Fig. 16.

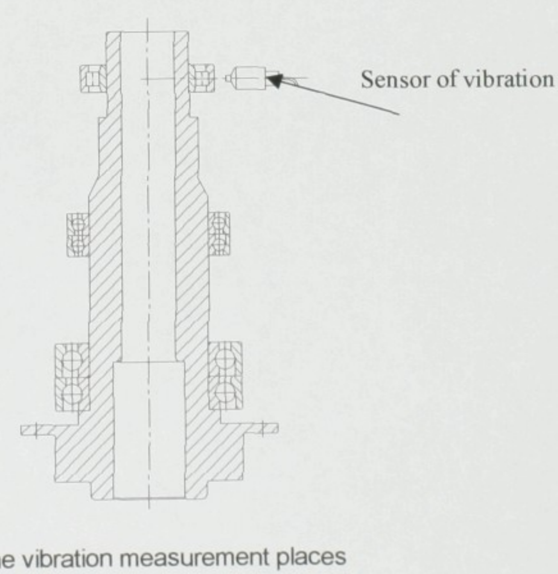


Figure 16 The vibration measurement places

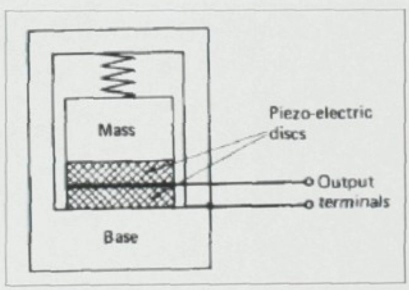
To choose the measurement places, we should apply the following four rules:

1. In the signal path, there can be only one disconnecting material and that, only between the bearing and the bearing box.
2. The direction of the signal path between the bearing and the measurement place must be straight and through. The distance between the point of measurement and the bearing shouldn't be bigger than 75 mm.
3. The measurement place must lie in the loaded area of the bearing.
4. The main axis of the sensor must be pointed at the loaded axis of the bearing and shouldn't deviate more than 5 degrees from this direction.

5.2 THE MEASUREMENT PARAMETERS AND SENSOR

For the periodical diagnostics of EMAG machine tools, the velocity parameter was chosen to measure vibration. This parameter was chosen because it is the only one defined in ISO norms.

Popularly - used seismic transducers are the velocity pick up and the piezoelectric sensor. Piezoelectric sensor have in recent years become by far the most widely - used transducer type for machine vibration measurements because of their much smaller physical dimensions, their long term reliability and their general robustness.



The simplest form of piezoelectric accelerometer the crystal is kept permanently compressed by the seismic mass. Thus the effect of accelerations in alternate directions is an alternating increase and decrease in the compressive force on the crystal. The compressive pre-load is applied by screwing the seismic mass down on to the crystal to a given torque. The electrical connections to the crystal are made by depositing a thin film of metal on the end faces.

One of the difficulties of measuring accelerations on a light, flexible structure is that the mass of the accelerometer may alter the frequency and amplitude of the vibration we are trying to measure. For such applications we need an accelerometer of the smallest and lightest possible type - piezoelectric sheer type. In this type of accelerometer a ring of piezoelectric material is bonded to a central pillar.

The seismic mass is a metal ring bonded to the outside of the piezoelectric material. Acceleration in the direction of the sensing axis causes shear stresses in the piezoelectric material, which is arranged so that corresponding values of electrical charge appear between the central pillar and the seismic ring.

To measure EMAG machine tools, the choice of vibration sensor is very important. We should take into account the characteristics of these machine tools. Among the decisive parameters there is size, mass, sensitivity, and dynamical range. Sensor VIB 6.122 was chosen to measure EMAG machine tools according to the above mentioned. Figure 17.

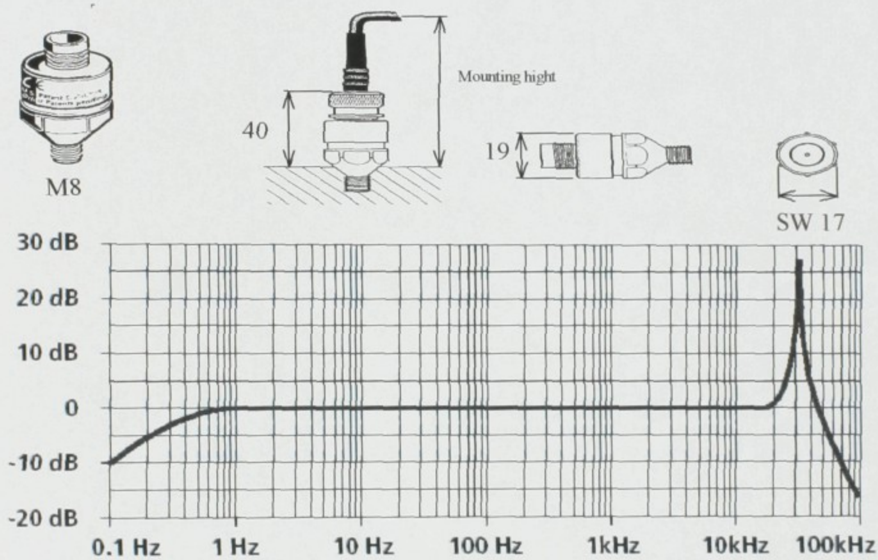


Figure 17 The dimensions of VIB 6.122 sensor and its frequency characteristics

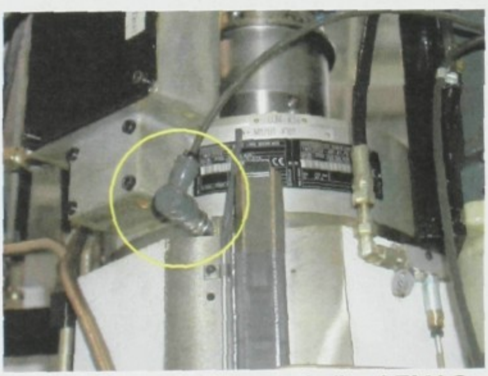


Figure 18 The sensor VIB 6.122 on spindle of EMAG machine tools

5.3 THE USE OF FFT ANALYZERS

Most of today's portable data loggers and machines analysers use the FFT (Fast Fourier Transform) for fault detection. However, FFT spectra cannot be used reliably for spectrum comparison which is needed to obtain the earliest possible warning of a fault developing in a machine, one may be tempted to use the storage and comparison capability of most single channel analyser. However, problems with FFT's which are: Why is only used FFT

1. Too little resolution at the low frequency end if a wide frequency range is used.
2. Too much resolution at the high frequency range.
3. Speed fluctuations cause unequal absolute changes in frequency of the machine vibration components, can be envisaged.

Very small changes in speed will shift the position of the peaks and result in large differences and give false exceedances, and therefore false warnings.

The solution is to perform a kind of data reduction, i.e. group the lines in the FFT spectrum together in larger groups, thus allowing changes inside each of these groups without interfering with the overall level of these groups. The new spectrum created in this way is referred to as synthesized spectrum. ? Which kind of ? CPB ?

Of course, the frequency bands in this synthesized spectrum must be wide enough to absorb the random changes in the signal, but narrow enough to be able defect small changes in the major component of the spectrum.

A problem, however still exists regardless of how wide the frequency band is. If the level of one of the lines of the narrow band FFT spectrum was high, caused by for example a shaft frequency, and was right on the edge of the band limit, it would dominate the level of that band.

Next time fault detection was carried out by spectrum comparison and it the operation speed was slightly higher, (only fractions of a percent), this component would fall in to the next line of the FFT spectrum. It could therefore fall in to the next band in our "group bands" causing this band to rise drastically, and indicate a tale warning.

To avoid this, a "broadening" of the bands must be done. The original spectrum is called the reference spectrum, and this is used to create a reference mask, where each line assumes the maximum value of either itself or the two neighbouring bands. This means the changes in the spectrum below this level will not cause an alarm, whereby false alarms from random noise sources will be avoided. The new recordings are then compared with this reference mask.

When comparing new recordings with reference mask, it any levels in the later measurements exceed the mask they may be considered as an indication of a developing fault.

The last problem to consider now is significant speed changes. If a machine changes operation speed more than that allowed keeping the frequency components within one bandwidth, the detection system described up to now will also give false warnings. If the spectrum is synthesized in such away that we obtain a Constant Percentage Bandwidth spectrum, the effect of a speed change will be the same all over the spectrum.

5.4 MEASURING SPECTRUM VIBRATION

5.4.1 The vibration spectrum

For a given machine-element type, there is a characteristic relationship between the rotational frequency of the element and the frequency at which a fault in the element is seen in the spectrum. The table in the illustration gives a general view of machine faults, and their positions in the spectrum. So the EMAG machine tools in a factory SKODA should have its own troubleshooting chart, which lists each machine element, and the frequency at which fault in this element, is seen in the spectrum.

5.4.2 Spectrum comparison

A reference spectrum is established for the machine in good condition, and operating under a well defined set of operating conditions and process parameters.

The latest spectra are taken and compared with this reference. By subtracting identical frequency lines a difference spectrum is obtained. Decision-making can safely start with this difference spectrum.

6. DISIGN OF A MAINTENANCE METHOD FOR EMAG MACHINE TOOLS

For periodical monitoring of EMAG machine tools we design fixed interval measurements, so that developments in the running condition can be followed closely. Mechanical vibration is a very good indicator of EMAG machines running condition. This is the reason why the most common form of EMAG machine Condition Monitoring uses vibration measurements as an indicator. The axiom of On Condition Maintenance is that servicing is permitted only when measurements show it to be necessary.

By means of regular vibration measurements, the onset of fault conditions can be detected and their development followed. Measurements can be extrapolated in order to predict when unacceptable vibration levels will be reached and when the machine must be serviced. This is called Trend Monitoring and it allows the engineer to plan for repairs well in advance.

Where is that?

6.1 THE ECONOMIC BALANCE OF ON - CONDITION MAINTENANCE

On - Condition maintenance based on vibration monitoring has been employed successfully in the continuous - process industries since the early 1970's. Oil and chemical plants quickly adopted the technique and achieved great savings owing to the higher availability of production machinery and the corresponding increase in productivity.

Condition monitoring has since spread rapidly through most industries employing rotating machinery - success stories are numerous.

Huge savings in maintenance work do not necessarily mean that maintenance personnel are thrown out of work. They are employed in performing the condition measurements and probably have time to do a more thorough overhaul and test job on any machine that is taken down for repair, thus contributing even more to the long - term reliability of the machine.

The maintenance engineer will be faced with the task of evaluating the cost/benefits of vibration measurements for on - condition maintenance with respect to his plant.

The number of EMAG machine tools in the factory SKODA Mladá Boleslav is about 50 machines, one-shaft and double-shaft. To demonstrate and explain the analysis of measuring spectra according to ISO norm in this diploma work, I am giving three examples of vibration spectra and their analysis for different EMAG machine tools.

1. One EMAG machine tool without troubles
2. Two different EMAG machine tools with troubles

6.1.1 EMAG machine tool without troubles

To compare all measured spectra in the same measuring place on EMAG machine tool, and to show the increase or decrease of amplitudes, for example spindle amplitude or bearings amplitudes measured on periodic times, the Waterfall diagram can help.

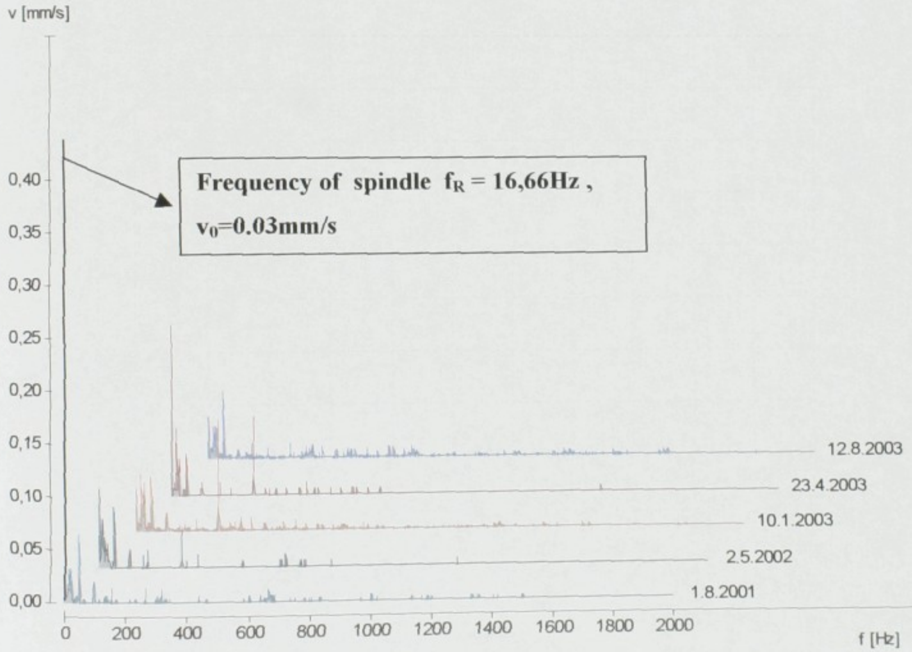


Figure 19 Waterfall diagram for the periodic measurements from 1.8.2001 to 12.8.2003.

All the spectra of EMAG machine tool, that is without trouble, have the same characteristics. The prevailing in these spectra is rotor frequency of spindle 16,66Hz (very small – the balancing of spindle is good), and its harmonic frequency 33.32Hz, 49.98Hz, 66,64 Hz (even smaller – the clearance of spindle is suitable). Bearings frequency amplitude is almost invisible (the bearing of EMAG machine tool spindle is in good condition).

The root mean square values RMS according to ISO norm, and amplitudes of EMAG machine tool spindle, measured on different dates, are shown in table 4.

Table 4 Measured root mean square values, and amplitudes of spindle frequency of the one-shaft EMAG machine tool without troubles.

Date of measurement	**Amplitude of machine tool spindle frequency [mm.s^{-1}]**	**Root mean square value RMS [mm.s^{-1}]**	**State of machine tool according to norm ISO 10816**
1.8.2001	0,030	0,10	Range A - Good
10.11.2001	0.031	0.11	Range A – Good
2.5.2002	0,030	0,10	Range A – Good
11.9.2002	0.060	0.10	Range A – Good
14.10.2002	0.053	0.12	Range A – Good
10.1.2003	0,051	0,10	Range A – Good
14.2.2003	0.058	0.16	Range A – Good
23.4.2003	0.060	0.10	Range A – Good
30.6.2003	0,034	0,14	Range A - Good
12.8.2003	0,035	0,10	Range A - Good
30.9.2003	*0.030*	*0.13*	*Range A - Good*
30.10.2003	0.032	0.11	Range A - Good

A 'Trend' plot is simply a number of amplitude values - snapshots of the total vibration (vibration at all frequencies) - over a period of time. The interval between readings could be from months to weeks depending on the proposed system for Emag machine tool. A trend plot offers limited analysis tools (there is no identification of specific frequencies, for example) but can be an important indicator of developing problems.

As a Trend is amplitude values versus time, a "Spectrum" Plot is amplitude versus frequency. A spectrum, measured by an "FFT", allows to choose severity (with the amplitude) and helps identify the source (with the frequency). This is the most used analysis tool and is usually sufficient protection for general speed machinery. In the following figures, we see trends and spectra of vibration of the EMAG machine tool without troubles.

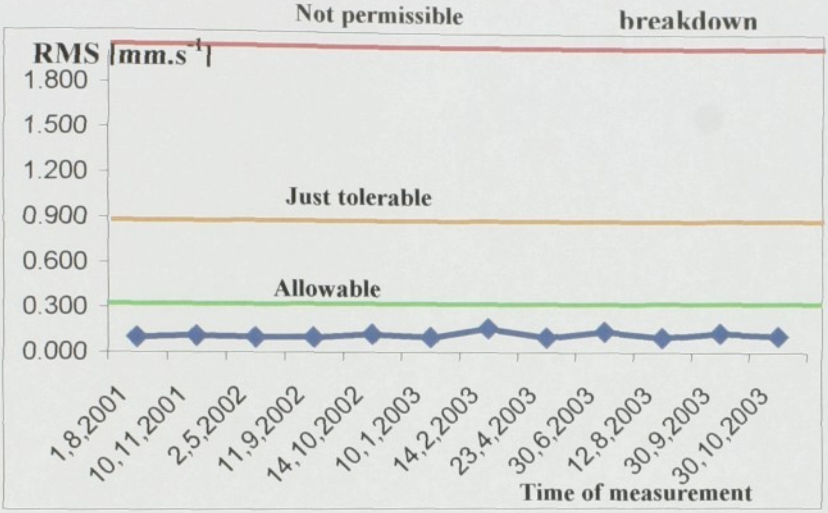


Figure 20 Trend Plot – Root mean square values over a period of time – All of them lying in range A (less than 0.35 mm.s^{-1}, the suggested value in the conclusion), which shows that the machine is in good condition.

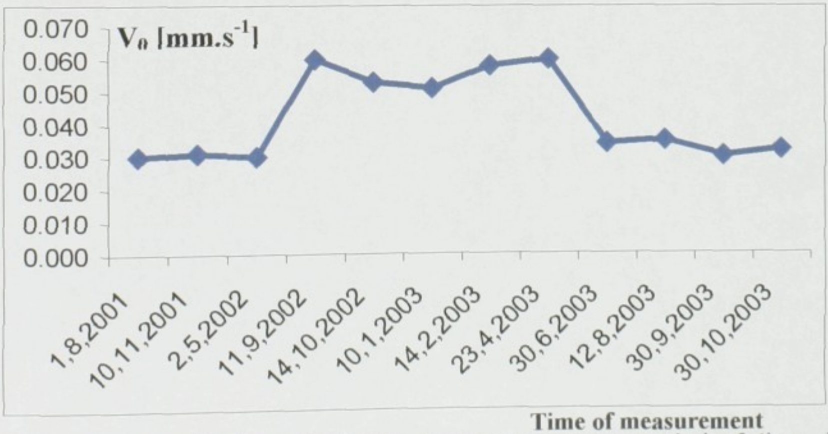


Figure 21 The amplitudes of spindle frequency over a period of time. The amplitudes of measured spindle frequency are very low which shows a good condition of spindle balancing.

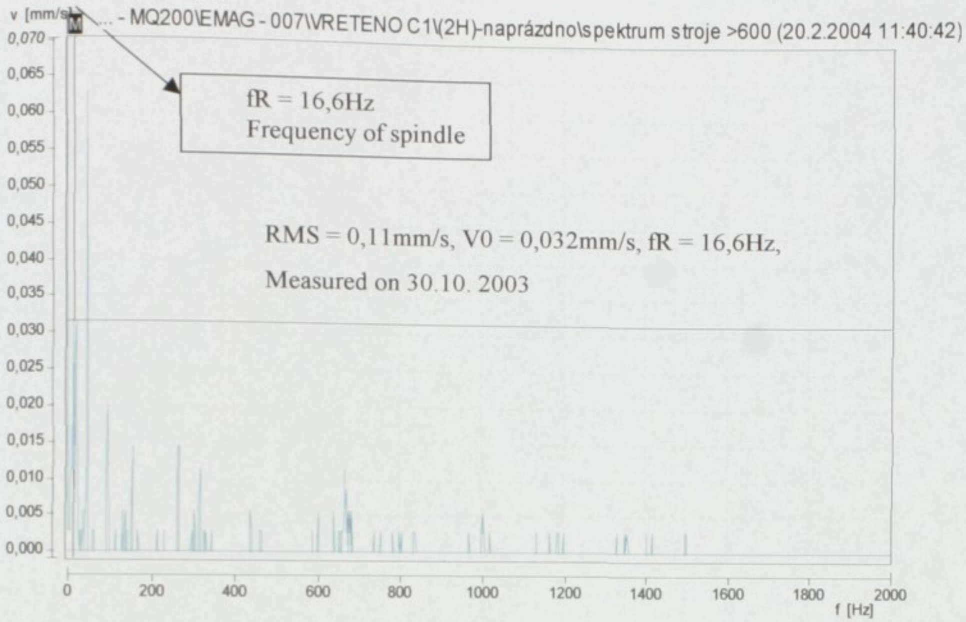


Figure 22 Vibration spectra of EMAG machine tool, measured on 30.10.2003. The frequency of spindle is 16.6 Hz. The amplitude of frequency of spindle is 0.032 [mm.s-1] (the balancing of the machine tool spindle is in a very good condition), root mean square value of machine tool is 0.11 [mm.s-1] (lying in range A - according to norm ISO 10816, the machine tool is in good condition). The harmonic frequencies of spindle are very small and unimportant. The bearings are in good condition too (bearing frequencies - inner ring damage frequency, bearing cage damage frequency, outside ring damage frequency and bearing ball damage frequency) did not appear in the vibration spectrum.

- Harmonic Frequencies - Activating this tool creates additional frequencies (as many as are required) that appear at integer multiples of the base frequency. If the base frequency is located at 1x rpm, harmonics will appear at 2x, 3x, 4x, etc. This is the most important analysis tool available.

- Sideband Frequencies - Activating this tool creates additional frequencies to either side of the base frequency. If the 1st (closest) sideband frequencies are located 50 cpm to either side of the base frequency, additional sideband frequencies (as many as required) will each be located an additional 50 cpm away. For example, the 2nd sideband frequencies will be 100 cpm away from the base frequency, the 3rd will be 150 cpm away, etc.

6.1.2 EMAG machine tool number 1 with troubles

Waterfall diagram for the periodic measurements from 21.1.2002 to 5.8.2003 is shown in the figure 23.

The spectra of EMAG machine tool, that is with troubles, have different characteristics as shown in figure 23. The amplitude of rotor frequency of spindle (16,3Hz) in the beginning of periodic measuring is very small (0.01 $mm.s^{-1}$ – the balancing of spindle is good). This value increased with time until it reached 0.023 $mm.s^{-1}$ on 10.12.2002. The bearing began to show signs of damage on 11.9.2002 and the damage became more serious so that the bearing had to be changed on 10.12.2002. See figure 24 and 25. After its change, the amplitude of rotor frequency of spindle decreased again to the standard value 0.01 $mm.s^{-1}$ (in range A). Bearings frequency amplitude appeared (B71916E with outside frequency 178,1 Hz) in the spectrum of vibration, and sideband frequency also began to show (frequency of spindle 16.3 Hz) - the bearing B71916E of EMAG machine tool spindle is damage).

The root mean square values RMS according to ISO norm, and amplitudes of EMAG machine tool spindle, measured on different dates, are showed in table 5.

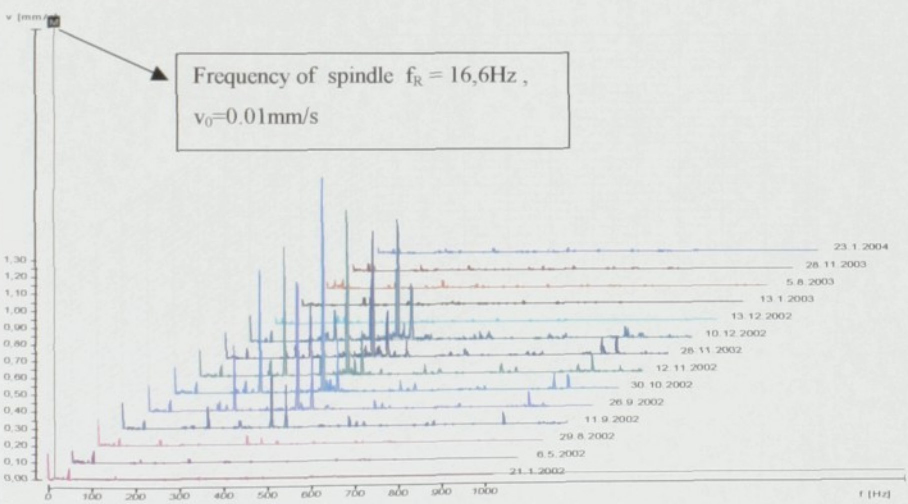
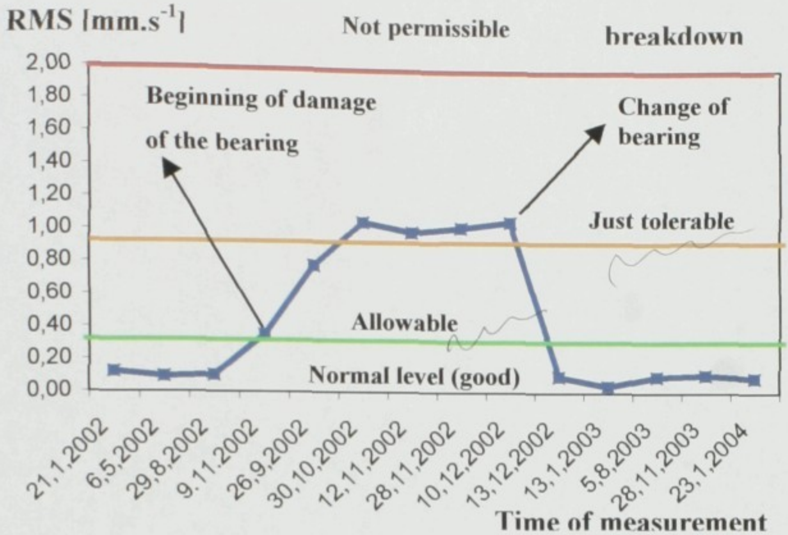


Figure 23 Waterfall diagram for the periodic measurements from 21.1.2002 to 5.8.2003. The spectrum of vibration measured on 11.9.2002, showed an increase in the amplitude of machine tools spindle from 0.01 $mm.s^{-1}$ to 0.02 $mm.s^{-1}$ (increase 2x) and an increase in the root mean square value RMS from 0.1 $mm.s^{-1}$ to 0.35 $mm.s^{-1}$ (3.5x).

Table 5 Measured root mean square values, and amplitudes of spindle frequency of the one-shaft EMAG machine tool number 1 with troubles.

Date of measurement	Amplitude of machine tool spindle frequency [mm.s^{-1}]	Root mean square value RMS [mm.s^{-1}]	State of machine tool according to norm ISO 10816
21.1.2002	0,010	0,12	Range A - Good
6.5.2002	0,010	0,09	Range A - Good
29.8.2002	0,010	0,10	Range A - Good
11.9.2002	0,020	0,35	Range A - Good – beginning of damage of the bearing B71916E with frequency 178,1 Hz
26.9.2002	0,021	0,78	Range B - Allowable
30.10.2002	0,023	1,05	Range B – Allowable
12.11.2002	0,020	0,99	Range B – Allowable
28.11.2002	0.021	1.02	Range B - Allowable
10.12.2002	0.023	1.06	Range B - Allowable - change of damage bearing
13.12.2002	0,010	0,10	Range A - Good
13.1.2003	0,010	0,04	Range A - Good
5.8.2003	0,12	0,10	Range A - Good
28.11.2003	0.011	0.11	Range A - Good
23.1.2004	0.011	0.09	Range A - Good

? see fig. 24!!
Just Tolerable?



Where i the ISO standard ?

Figure 24 Trend Plot – root mean square values over a period of time – At the beginning of measurement they were lying in range A (less than 0.35 mm.s^{-1}), then they increased until they reached 1.06 mm.s^{-1} which is higher than the tolerable value of 0.7. After the change of bearing, the RMS decreased again to the standard value 0.1 mm.s^{-1} (in range A).

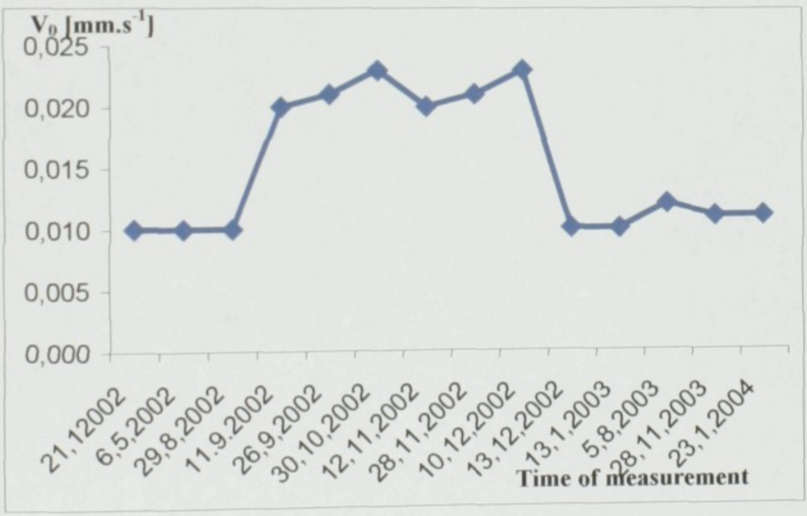


Figure 25 The amplitudes of spindle frequency over a period of time. At the beginning of measurement they were small, then they increased until they reached 0.023 mm.s^{-1}. After the change of bearing, the amplitudes of spindle frequency decreased again to the standard value 0.01 mm.s^{-1}. The amplitudes of spindle frequency grew also together with the growth of RMS.

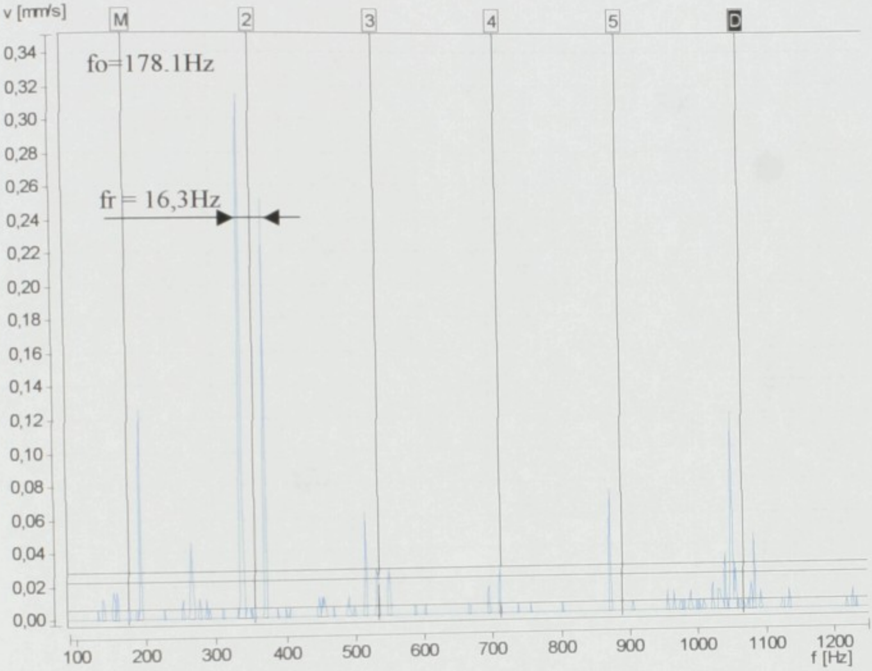
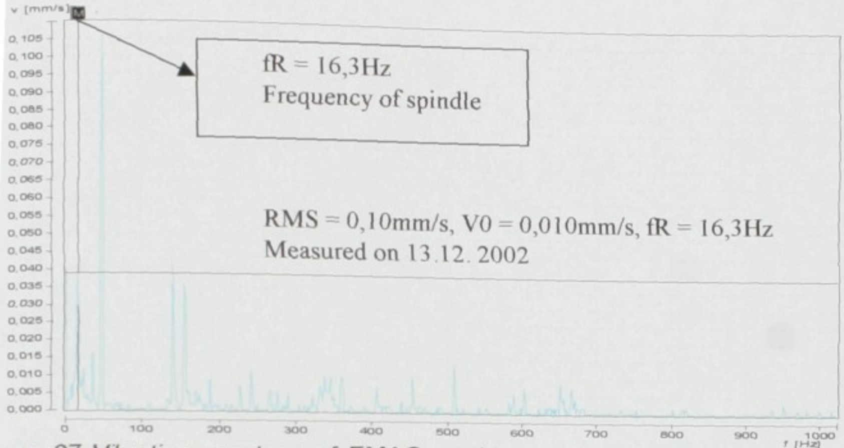


Figure 26 shows the spectrum of vibration measured on 30.10.2002. it shows the damage of the bearing outside ring B 71916 E with frequency f_o = 178.1Hz. The sideband with frequency 16.3Hz (spindle RPM of machine tool number 1 is 978min^{-1}) appeared around the frequency of the outside ring bearing and around his harmonic frequencies. The growth of spindle frequency amplitudes from 0.010 mm.s^{-1} to 0.023 mm.s^{-1} (damage of bearing B71916E with frequency 178,1 Hz) and the growth of root mean square values of machine tool from 0.10 mm.s^{-1} to 1.06 mm.s^{-1} confirm the increase of vibration levels (Figure 33).



igure 27 Vibration spectrum of EMAG machine tool measured on 13.12.2002, after ne change of damaged bearing B 71916 E. The spindle frequency is 16.3Hz, the mplitude of spindle frequency of machine tool is 0.010 $mm.s^{-1}$ (the balancing of the nachine tool spindle is in a very good condition), root mean square value of machine ool is 0.10 [$mm.s^{-1}$] (lying in range A - according to norm ISO 10816, the machine ool is in good condition). The harmonic frequencies of spindle are very small and nimportant for their analysis.

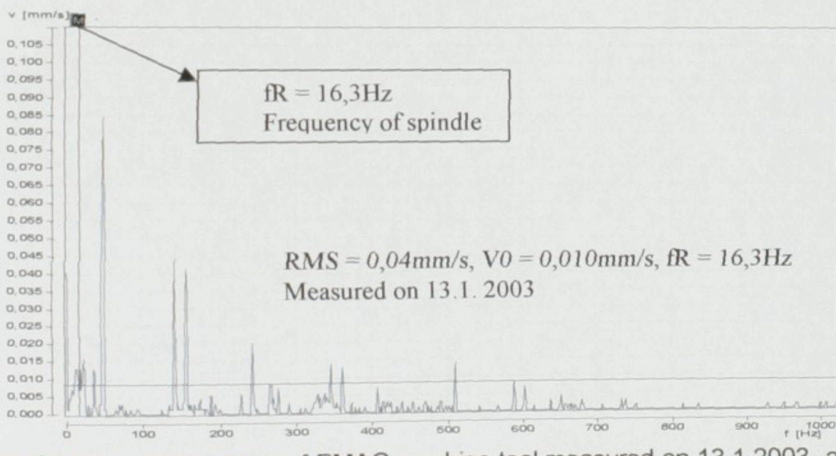


Figure 28 Vibration spectrum of EMAG machine tool measured on 13.1.2003, after the change of damaged bearing B 71916 E. The spindle frequency is 16.3Hz, the amplitude of spindle frequency of machine tool is 0.010 $mm.s^{-1}$ (the balancing of the machine tool spindle is in a very good condition), root mean square value of machine tool is 0.04 [$mm.s^{-1}$] (lying in range A - according to norm ISO 10816, the machine tool is in good condition). The harmonic frequencies of spindle are very small and unimportant for their analysis.

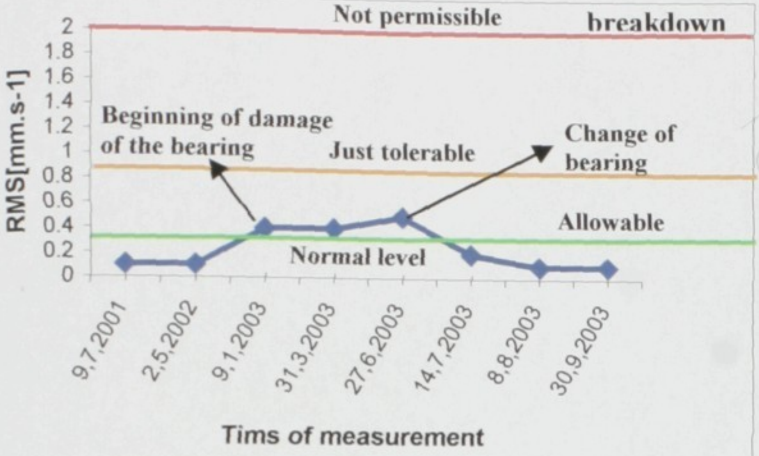
Other spectra, measured on 5.8.2003, 5.9.2003 and 30.10.2003, have the same characteristics as in the mentioned spectra (measured on 13.12.2002 and 13.1.2003). They have no changes concerning the balance of spindle and the state of bearings. See figure 23 and table 5.

6.1.3 EMAG machine tool number 2 with troubles

The spectra of vibration for EMAG machine tool number 2, that is with troubles, were measured from 9.7.2001 to 30.9.2003, too. See table 6.

Table 6 Measured root mean square values, and amplitudes of spindle frequency of the one-shaft EMAG machine tool number 2 with troubles.

Date of measurement	**Amplitude of machine tool spindle frequency [mm.s^{-1}]**	**Root mean square value RMS [mm.s^{-1}]**	**State of machine tool according to norm ISO 10816**
9.7.2001	0,01	0,1	Range A - Good
2.5.2002	0,01	0,1	Range A - Good
9.1.2003	0,02	0,4	Range A - Good beginning of damage of the bearing B71916E with frequency 226,8 Hz
31.3.2003	0,02	0,4	Range A - Good
27.6.2003	0,02	0,5	Range A - Good change of damage bearing
14.7.2003	0,01	0,2	Range A – Good
8.8.2003	0,01	0,1	Range A – Good
30.9.2003	0.01	0.1	Range A - Good



Same as previous!
Where is the ISO?

Figure 29 Trend Plot – root mean square values over a period of time – At the beginning of measurement they were lying in range A (less than 0.35 $mm.s^{-1}$), then they increased until they reached 0.5 $mm.s^{-1}$. The RMS value measured on 9.1.2003, increased 4 times, this is a reason why we should analyze the measured spectrum accurately. After changing the bearing, the RMS decreased again to the standard value 0.1 $mm.s^{-1}$ (in range A).

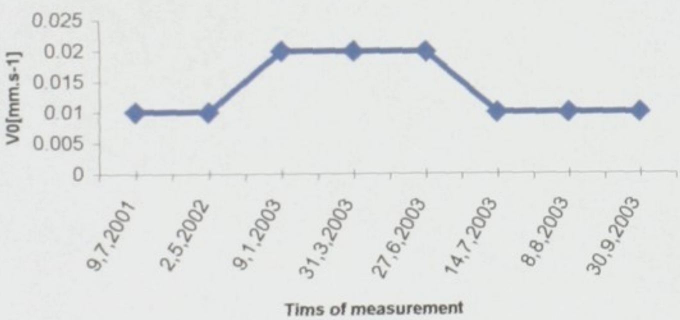


Figure 30 The amplitudes of spindle frequency over a period of time. At the beginning of measurement they were small, then they increased until they reached 0.02 $mm.s^{-1}$. After the change of bearing, the amplitudes of spindle frequency decreased again to the standard value 0.01 $mm.s^{-1}$. The amplitudes of spindle frequency grew also together with the growth of RMS.

The following spectrum of vibration measured on 27.6.2003 represents the damage of innerside ring of the bearing B71916E with frequency 226,8 Hz.

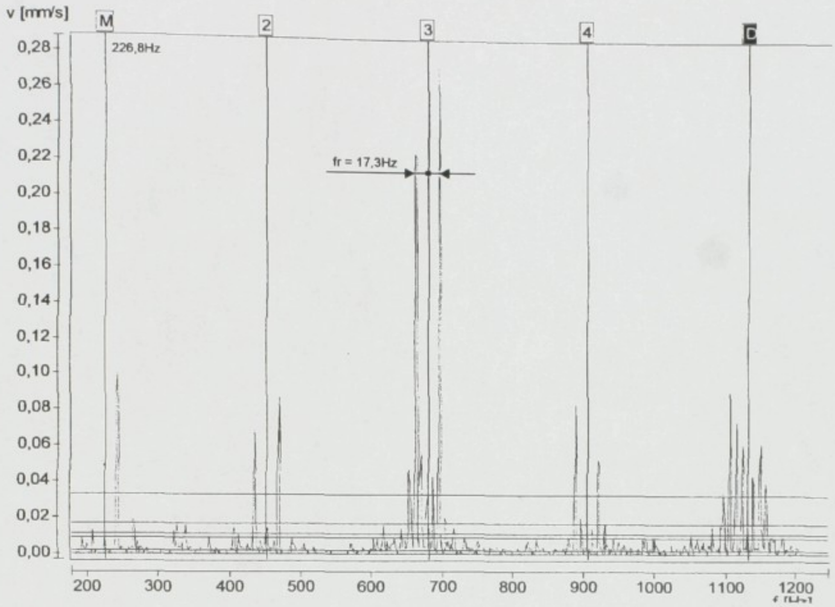


Figure 31 The spectrum of vibration measured on 27.6.2003. It shows the damage of the bearing innerside ring B 71916 E with frequency f_i = 226.8 Hz. The sideband with frequency 17.32 Hz (spindle RPM of machine tool number 2 is 1039.2 min^{-1}) appeared around the frequency of the innerside bearing and around his harmonic frequencies 2x, 3x, 4x. The growth of spindle frequency amplitudes from 0.01 $mm.s^{-1}$ to 0.02 $mm.s^{-1}$ (damage of bearing B71916E with frequency 226.8 Hz) and the growth of root mean square values of machine tool from 0.1 $mm.s^{-1}$ to 0.5 $mm.s^{-1}$ confirm the increase of vibration levels (Figure 33).

6.2 A BASIC SYSTEM FOR FREQUENCY ANALYSIS

Systems, which apply frequency analysis to EMAG machine tools condition monitoring, are available in several configurations. SKODA Auto plant has started with a portable vibration analyser set and level recorder. This set has good all - round facilities and plots out a narrow - band frequency spectrogram on the spot at each monitoring point in turn. The system is easy to use when the plant engineer has previously determined the measurement parameters. We have adopted the norm ISO 10816-1 for analyse the spectra of vibration of EMAG machine tools. Figure 32 and table 7.

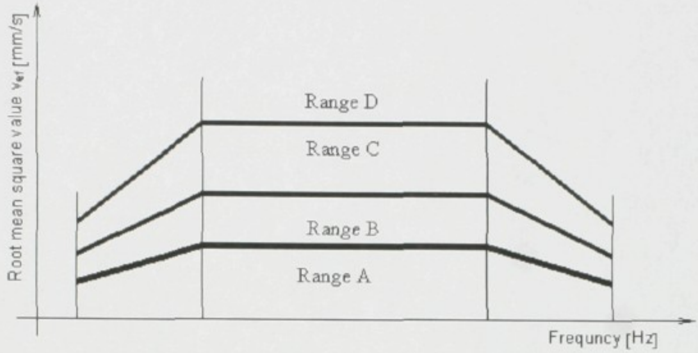


Figure 32 The criteria of admissible vibration velocity

What does it mean? Groups?

Table 7 The limit ranges

<table>
<tr><th>Root mean square value v_{ef} [mm/s]</th><th>Group I</th><th>Group II</th><th>Group III</th><th>Group IV</th></tr>
<tr><td>0,28</td><td rowspan="3">A</td><td rowspan="4">A</td><td rowspan="5">A</td><td rowspan="6">A</td></tr>
<tr><td>0,45</td></tr>
<tr><td>0,71</td></tr>
<tr><td>1,12</td><td rowspan="2">B</td></tr>
<tr><td>1,8</td><td rowspan="2">B</td></tr>
<tr><td>2,8</td><td rowspan="2">C</td><td rowspan="2">B</td></tr>
<tr><td>4,5</td><td rowspan="2">C</td><td rowspan="2">B</td></tr>
<tr><td>7,1</td><td rowspan="5">D</td><td rowspan="2">C</td></tr>
<tr><td>11,2</td><td rowspan="4">D</td><td rowspan="2">C</td></tr>
<tr><td>18</td><td rowspan="3">D</td></tr>
<tr><td>28</td><td rowspan="2">D</td></tr>
<tr><td>45</td></tr>
</table>

Reference spectra are recorded for each monitoring point. We compare subsequent spectra with reference spectra, any differences will be immediately apparent. As the levels of certain frequency components begin to grow, they are plotted on a level versus time chart so that future trends can be predicted. This enables the fault to be diagnosed, spare parts ordered and the repair to be scheduled to a convenient time.

We can perform simple wide band monitoring on a regular basis and apply the frequency analyser only when significant vibration level changes are noted. While analysis at this stage will aid diagnosis of the developing fault, the early warning and trend benefits of regular analysis are not in this case utilised.

6.3 EVALUATION OF VIBRATION LEVELS

Vibration measurements on the surface of machine elements reflect the cyclic forces being transmitted at that point. The actual vibration velocity measured is proportional not only to the forces involved but also the mobility of the structure at that point. Mobility is a measure of the structure willingness to be set into motion. The relationship between Force, Mobility and the resulting vibration velocity, with respect to frequency, is illustrated in Figure 33. Using logarithmic scales one can add force and mobility spectra to get a resultant vibration spectrum. Note that the high force component (A) at frequency (n) is content by a low mobility at frequency (n) so that no special peak is noted in the vibration spectrum.

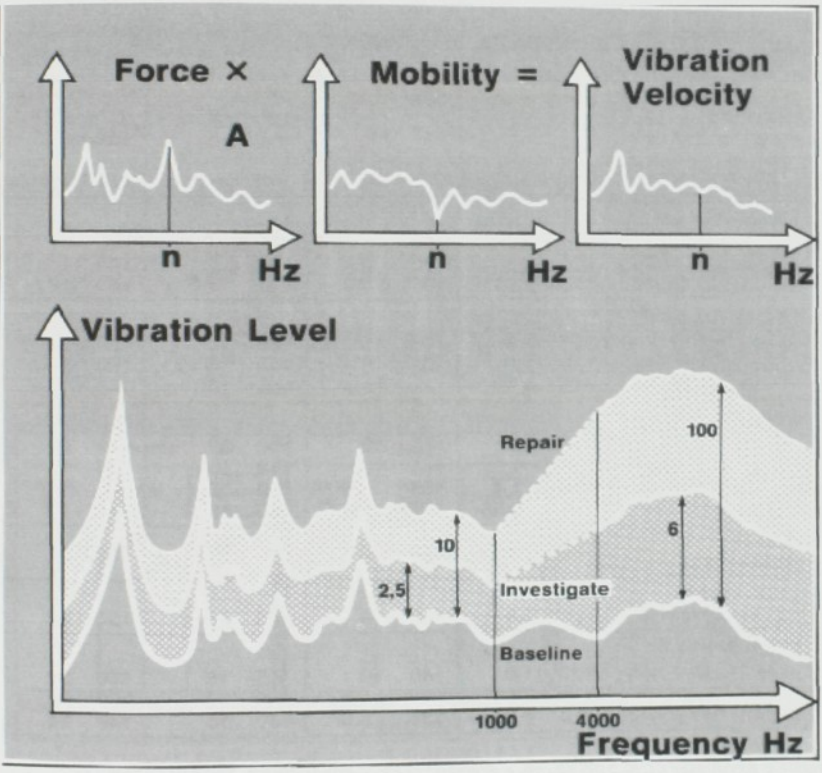


Figure 33 The increase of vibration levels

It is therefore unwise only to look for high level peaks in the vibration spectrum, low values may also contain information about important force changes. Mobility characteristics of machinery do not usually change significantly with time so one can safely assume that if the vibration level at a monitoring point doubles, the force level has also doubled. A much more reliable indication of machine condition is obtained by concentrating on relative changes.

Practice has shown that for frequency components up to 1000 Hz an increase by a factor of 2,5 (8 dB) should be considered a significant change in condition warranting investigation, and an increase by a factor of 10 (20 dB) from the reference condition signifies the need for repair -(as suggested). For frequency components above 4000 Hz these factors can be cautiously increased to 6 (18 dB) and 100 (40 dB) as shown in Figure 33.

6.4 TRENDING FAULT DEVELOPMENTS

Ordinary day-to-day monitoring with no fault developments will give a series of spectra with "no significant changes". An incipient fault will cause a group of lines to suddenly increase. Then, as the fault slowly spreads out, the vibration level will gradually rise. This would be the case with an incipient rolling-element bearing fault, or the effect of ordinary wear.

However, if the detection of a fault leads to a proper diagnosis, from which it can be assumed that the fault will develop gradually, and hence the vibration signal will develop gradually in parallel and not stepwise, then and only then, trending can be carried out. The frequency range for the trend analysis should be chosen on the frequency bands where increases are detected to give a good indication of the fault development and to determine the remaining safe operating time.

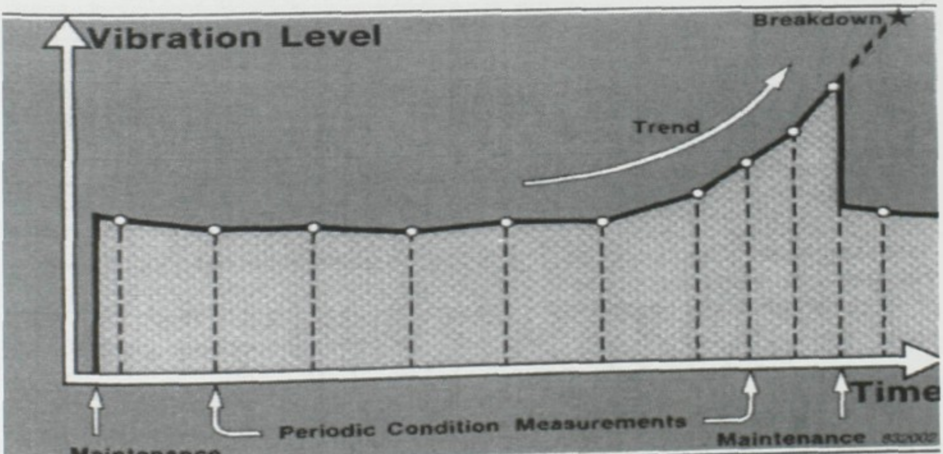


Figure 34 Periodical monitoring of EMAG machine tools operational condition

This type of trending can be done with the computer, which already handles the spectrum comparison. If the spectra which exceed the prescribed levels are stored together with the reference spectra, a trending program included in the software will only need to have the information of how high an increase in level the maintenance engineer is willing to accept.

They indicate that an increase in vibration of 2,5 times, or 8 dB, warrants a further analysis, and an increase of 10 times, or 20 dB, demands immediate action. These values are, however, conservative and reasonable to use as default values until ones own experience is built up. Figure 34.

7. CONCLUSION

The aim of my diploma work was to work out the method of without dismantling watching of the drive operation ability in EMAG machine tools.

In my work, I described the goal of mechanical vibration analysis, its characterising parameters, the methods of bearings diagnostics, the measurement method and the method of choosing measurement places. Then, I described the main sources of excitation and the way of calculating their frequency.

And on the basis of the dynamical analysis of the machine, a method was worked out for the next procedure in the application of technical diagnostics.

The work quality of the whole machine and then even the economical advantage of its mounting depend on the drive condition. Without dismantling diagnostics enable realizing a system of maintenance and reparation of machines according to their momentary state. This means in practice a meaningful economical advantage because the appearance of important damages is avoided, and at the same time, the change of parts still having operation ability does not happen. The expenses of providing measuring and evaluating technical are compensated in a short time by sparing the expenses of maintenance of the machines.

What is the real economy effects?

In modern manufacturing machines, strict requirements and high working values, as higher output and higher accuracy, are expected.

Therefore, EMAG machines tools require better maintenance to reduce the number of breakdowns. The ideal way is to applicate technical diagnostics by measuring vibration. As the measured machine is run 24 hours a day, I suggest that the measurement should be carried out once every two months and compared to the previous measurement. In my diploma work, I have presented 3 different examples of EMAG machine tools, measured periodicaly for a long period of time. The first one is without troubles, its RMS was always under 0.35 $mm.s^{-1}$. The second is with troubles (damage of outside ring bearing), its RMS is between 0.7 and 1.5 $mm.s^{-1}$. The third EMAG machine tool is with troubles too (damage of innerside ring bearing), its RMS is between 0.35 and 0.7 $mm.s^{-1}$.

The state of the EMAG machine tools number 1 and 2 with troubles was always within range A - good state - RMS lower than 0,71 $mm.s^{-1}$ according to ISO norm 10816, knowing that in both cases, the analysis of vibration spectra detected bearing damage.

For these reasons, we advise to lower the value of the ISO norm to 50% for EMAG machine tools to detect possible damage in their parts and to reduce the negative effects on SKODA gearbox parts quality.

According to the obtained results in this diploma work, the ISO norm 10816 for EMAG machine tools becomes as following:

- Max range A: 0.35 $mm.s^{-1}$
- Max range B: 0.90 $mm.s^{-1}$
- Max range C: 2.25 $mm.s^{-1}$
- Max range D: 22.5 $mm.s^{-1}$

When machines break down, everybody suffers. Sudden shutdown affects both labor and management.
Correcting problems before they happen means you can cut overall machine stress, reduce your spare parts inventory, place personnel where they're needed most, and improve your reputation for reliability.

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Datum: 30. 5. 2004

Podpis: ....