TECHNICAL UNIVERSITY OF LIBEREC
Faculty of Mechanical Engineering

DIPLOMA PROJECT

TECHNICAL DIAGNOSTICS OF SMOKE VENTILATORS

LIBEREC 1996

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TECHNICAL UNIVERSITY OF LIBEREC

Faculty of Mechanical Engineering

Department: Packing and Polygraphic Machines **School Year: 1995/1996**

DIPLOMA PROJECT

Graduate: Essam Hassan Ibrahim Suleiman

Study discipline: (23-21-08) Packing and Polygraphic Machines

According to the Law Digest No. 172/1990 for universities, the Head of the Department of Packing and Polygraphic Machines determines the following topic for your Diploma Project:

Project Heading: TECHNICAL DIAGNOSTICS OF SMOKE VENTILATORS

Content items:

Hot stations and heating plants should fulfill high requirements to get a reliable operation of smoke ventilators, their unplanned breakdown during a sudden failure, always threatens the operation of hot stations or heating plants. For smoke ventilators in the heating plant in Liberec carry out the following :

1. Calculate from the technical documentation the frequencies of the main sources of vibration excitation,
2. Design and determinate the vibration measuring places,
3. Choose the suitable methods for without dismantling diagnostics of bearings,
4. Measure the vibration of three smoke ventilators,
5. From measuring analysis, make a comparison between the machinery technical conditions,
6. According to ISO norms, identify the limiting values of power vibration to judge the condition and the probable operating life,
7. Analyze the possibilities of smoke ventilators' dynamical debugging.

Graphics: reports of measurements

Text: 50 pages (figures including)

References:

1. Research reports in **Packing and Polygraphic Machines.**

2. Brochures from Brüel & Kjaer and SKF companies.

3) Randall, B.: Frequency analysis. Brüel a Kjaer 1987.

4) Jens Trampe Broch: Mechanical vibration and shock measurements. Brüel a Kjaer 1984.

5) Beneš, Š.: Teorie stavby strojů. Skripta TU Liberec, 1986.

6) Beneš, Š., Tomeh, E.: Metody diagnostiky valivých ložisek . Skripta TU Liberec, 1991.

7) Fröhlich, J.: Valivá ložiska ZKL. SNTL Praha 1980.

8) Janoušek, I. a kol.: Technická diagnostika. Praha 1988.

Supervisor: Dr. Ing. Elias Tomeh

Consultant: Ing. Roman Petruch - Heating Plant - Liberec

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Semester: Summer

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Liberec 20.2.1996

TECHNICAL UNIVERSITY OF LIBEREC

FACULTY OF MECHANICAL ENGINEERING
DEPARTMENT OF TECHNICAL DIAGNOSTICS

TECHNICAL DIAGNOSTICS OF SMOKE VENTILATORS

DIPLOMA PROJECT

Study discipline: (23-21-08)

ESSAM HASSAN SULEIMAN

Supervisor: Dr. Eng. Elias Tomeh (DTD, TU of Liberec)
Consultant: Eng. Roman Petruch - Heat Plant - Liberec

Extent of the Diploma Project

Number of pages: 95
Number of figures: 59
Number of tables: 9

Declaration

I declare that I developed my diploma project indipendently with aid of refereed literature under the control of supervisor and consultant.

In Liberec, 20. 5. 1996

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Signature

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THE SYMBOLS USED

a.........acceleration [mm · s^{-2}]

d.........inside diameter of bearing [mm]

D.........outside diameter of bearing [mm]

B.........width of bearing [mm]

d_0........ball or roller diameter of bearing [mm]

d_S........pitch diameter of bearing [mm]

f_i.........bearing frequency of inside diameter [Hz]

f_o.........bearing frequency of outside diameter [Hz]

f_v.........frequency of ball or roller [Hz]

f_k.........frequency of cage [Hz]

f_R.........shaft frequency [Hz]

f_n.........natural frequency [Hz]

BPF.....blade passes frequency [Hz]

CG.....center of gravity

n.........number of rotations [min^{-1}]

q_1, q_2..dimensional coefficients of bearings

t..........time [s]

v_{ef}.......effective value of vibrating velocity [mm · s^{-1}] ?

a_{ef}.......effective value of vibrating acceleration [mm . s^{-2}] ?

N........number of bearing balls or rollers

α.........contact angle [°]

CPM...cycles per minute

RMS...root mean square

RPM...revolutions per minute

U........power vibration [mm · s^{-1}]

gSE.....spike energy

HFD....High Frequency detiction

dB......shock pulse ?

MD.....Measuring direction

TV......Type of ventilator

V........vertical direction

H.........horizontal direction

ISO.....International Standard Organization

1. INTRODUCTION

Useing diagnostics during machine process, it isn't a new method in industry. Initially, the most important factor in diagnostics was the man serving the machine who used his senses, sight, hearing, touch and smell. As a result of his subjective attention and his accumulated experiences, he decided whether the machine needed repairing. But human senses have their limits. The technical development requires higher precision in the quantity and quality.

That is why the use of sensors, that are much more powerful than human senses, together with modern techniques it became inevitable nowadays to respond to the new demands of modern industry.

There are two forms of technical diagnostics:

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

- Immediate diagnostics, which is the subject of my diploma project, it 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.

For the practical use of immediate diagnostics on smoke ventilators, there are two things must look for:

- the quality and reliability of the measuring analyzers and computing technique,

- the actual precise categories of damage situations that has to be memorized in the computer. We will also have to deal with situations that are not categorized. In this case we make a precise analysis and determine the damage with the help of our personal experiences.

This presented diploma work results from theoretical notes in the specialization of vibration measurement, signals analysis and on the basis of dynamic analysis of the smoke ventilator parts. These results are completed by orientational measuring, using the Microlog one channel analyzer and software PRISM2. This ~~which~~ is a knowledge based information system designed to provide an extensive analysis of the vibration information obtained from data collection system.

The Analyzer and software They have been applicated

~~And both of them~~ are from the SKF company (Fig. 1), on the smoke ventilators in the Heat Plant in Liberec.

These meaurements serve as a study of the effects of the smoke ventilator's mechanical vibrations on the efficiency and the machine assembly.

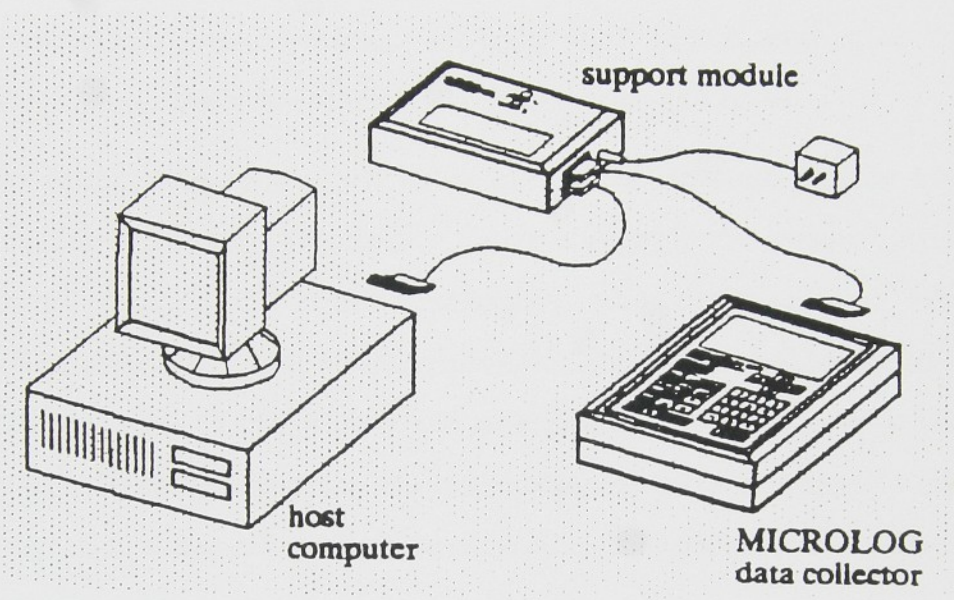


Fig. 1 The MICROLOG/PRISM2 System

The aim of this diploma project is detecting the faults before their happening and the use of continuous monitoring to keep the machine in good condition.

The microlog data collector (Fig. 1), collects machinery vibration, temperature, and other condition monitoring measurements. Together with visual observations it allows for detailed machine condition analyses in a harsh industrial environment.

Microlog performs all the tasks required for machinery predictive (condition) maintenance. It collects both dynamic (vibration) and static measurements from almost any source.

Values read from indicators are entered into Microlog by pressing the appropriate numeric keys on the Microlog keyboard. You can enter your observations in English or as coded notes.

In addition to its function as a data collector, microlog is a powerful analyzer with all the functions and performance, including frequency zoom, to capture and disply high resolution spectra (from 100 to 6400 lines) for detailed analysis.

2. SMOKE VENTILATOR AND VIBRATION

An ideal smoke ventilator would produce no vibration at all because all energy would be channelled into the job of work to be done.

In practice, vibration occurs as by-product of the normal transmission of cyclic forces through the mechanism. Smoke ventilator elements react against each other and energy is dissipated through the structure in the form of vibration, heat and noise.

A good design will produce low levels of inherent vibration. As the machine wears however, foundations settle and parts deform, subtle changes in the dynamic properties of the mechine begin to occur. Shafts become misaligned, parts begin to wear, rotors become unbalanced and clearances increase, coupling problems and mechanical looseness. All of these factors are reflected in a lack of accuracy and an increase in vibration energy, which, as it is dissipated throughout the machine, excites resonances and puts considerable extra dynamic loads on bearings. Cause and effect reinforce each other, and the smoke ventilator progresses towards ultimate breakdown.

It's no longer practicle to relie upon the experienced plant engineer's senses to recognise whether a smoke ventilator was running smoothly or not. That is for at least two reasons:

1. The days of plant attendants with any oily rag and a grease gun at hand who tend "their" machines are gone. The personal relationship between man and machine is not economically feasible and not even required because machines are expected to run automatically with only occasional attention from service personal.

2. Secondly, most modern machinery runs so fast that many tell-tale vibrations occur at such a high frequency that instruments are needed to detect and measure them.

2.1 VIBRATION INDICATES MACHINE CONDITION

The mechanical vibration has a much wider application today in other areas such as the automobile, construction, electronics, and other. The most common uses are for:

1 - Production control

2 - Frequency response/dynamic performance testing

3 - Environmental test

Vibration is normally a destructive by product of the force transmission through a smoke ventilator which provokes wear and accelerates breakdown.

Smoke ventilator elements which constrain these forces, for example bearing housings, are usually accessible from the outside of the smoke ventilator so that vibration resulting from the excitation forces can be measured at these points.

As long as the excitation forces are constant, or vary within certain limits, the vibration level measured will also be constant and vary within similar limits. Furthermore, for most smoke ventilators, the vibration has a typical level and its frequency spectrum has a characteristic shape when the smoke ventilator is in good condition.

This frequency spectrum, a plot of vibration amplitude against frequency, is known as the vibration signature of the smoke ventilator, and is obtained by frequency - analysing the smoke ventilator vibration - signal.

When faults begin to develop, the dynamic processes in the smoke ventilator change and some of the forces acting on machine parts are also changed - thereby influencing the vibration level and the shape of the vibration spectrum.

The fact that vibration signals carry much information relating to running condition of smoke ventilator is the basis for using regular vibration measurement and analysis as an indicator of machine health trends and the need for maintenance.

SMOKE VENTILATOR

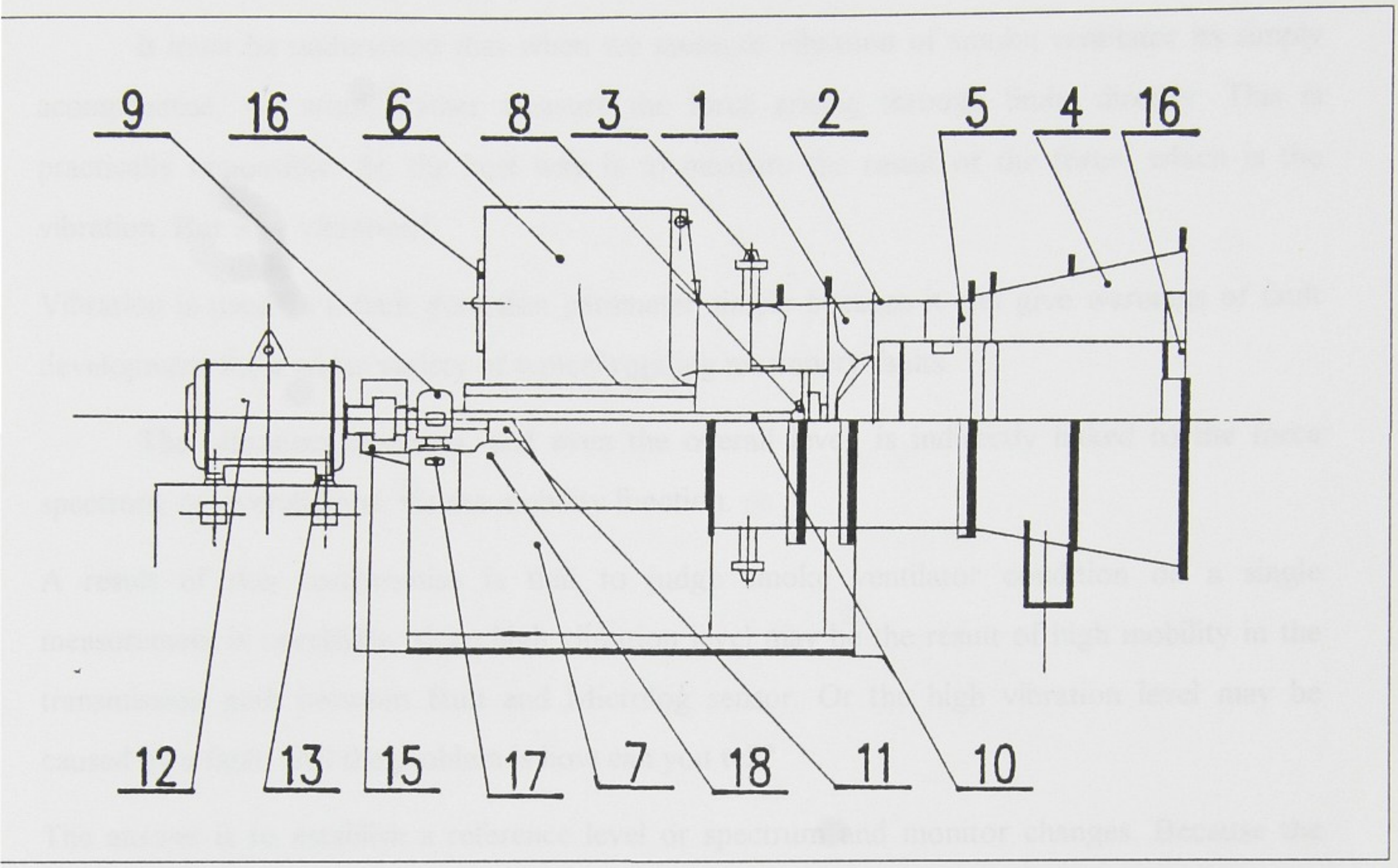


Fig. 2 Smoke ventilator

The basic parts of the smoke ventilator according to ČSN 12 0000.

1 - FAN	**10 - shaft**
2 - Casing	**11 - cooling tube**
3 - control device	**12 - electric motor**
4 - diffuser	**13 - motor stand**
5 - distributing ring	**14 - clutch**
6 - suction chest	**15 - clutch cover**
7 - ventilator bed	**16 - manhole**
8 - inside Bearing	**17 - terminal board of thermometer**
9 - outside Bearing	**18 - lubricating tube**

2.2 THE REASONS OF SMOKE VENTILATOTS VIBRATIONS

It must be understood that when we measure vibration of smoke ventilator its simply acompromise, we would rather measure the force arising through faults directly. This is practically impossible. So the best way is to measure the result of the force, which is the vibration. But why vibration?

Vibration is used as a fault detection parameter simply because it can give warnings of fault development for a wider variety of typical rotating machinery faults.

The vibration spectrum, and even the overall level, is indirectly linked to the force spectrum, or overall level, via the mobility function.

A result of this compromise is that to judge smoke ventilator condition on a single measurement is unreliable, since high vibration level may be the result of high mobility in the transmission path between fault and Micrologʼ sensor. Or the high vibration level may be caused by a fault. But the problem is how can you tell?

The answer is to establish a reference level or spectrum and monitor changes. Because the mobility is a constant, any changes are typically related to faults.

The diagram (Fig. 3) shows an interesting mobility phenomenom. The force spectrum contains a peak at the frequency shown. However, because the mobility has an "anti-resononce" at the same frequency, the vibration spectrum contains no significant peak at that frequency. This shows that its not only the largest peaks in a spectrum that we should monitor. But note that on 8 dB increase in force will still show as an 8 dB increase.

Also, it must be well known that the most basic level on a broadband basis in range of, for example, 10 - 1000 Hz or 10 - 10000 Hz. Such measurements are also relevant with displacement measurements from proximity probes, where the frequency band of interest is usually from about 30% of the running speed up to about 4th harmonic. An increasing vibration level is an indicator of deteriorating machine condition.

Trend analysis involves plotting the vibration level as a function of time, and using this to predict when the machine must be shut down for repair an other way of using the measurements is to compare them with published vibration criteria.

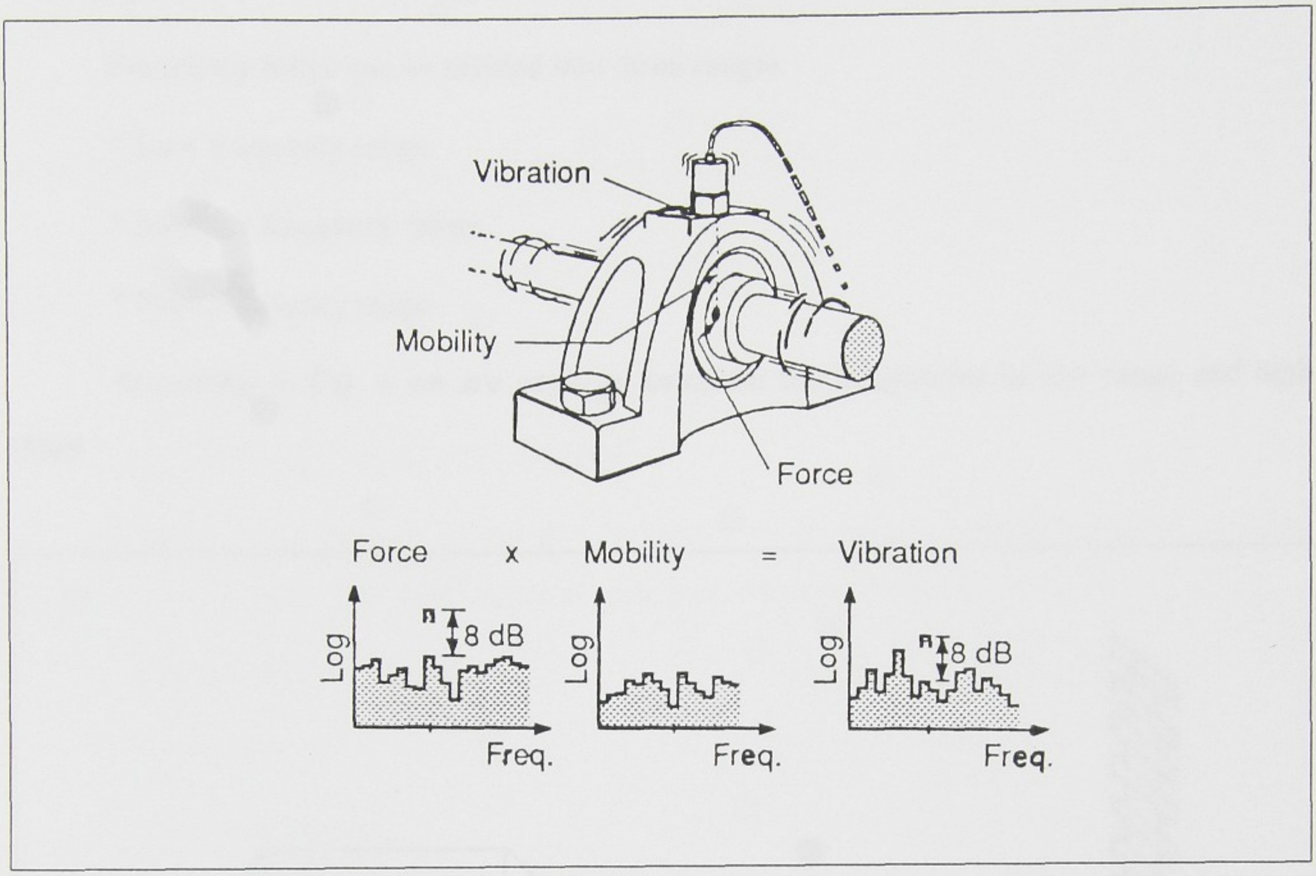


Fig. 3 Transmition of vibration

3. FREQUENCY ANALYSIS OF A SMOKE VENTILATOR

Frequency range can be divided into three ranges

* Low frequency range
* Medium frequency range
* High frequency range

According to Fig. 4 we are going to calculate the frequencies in low range and high range.

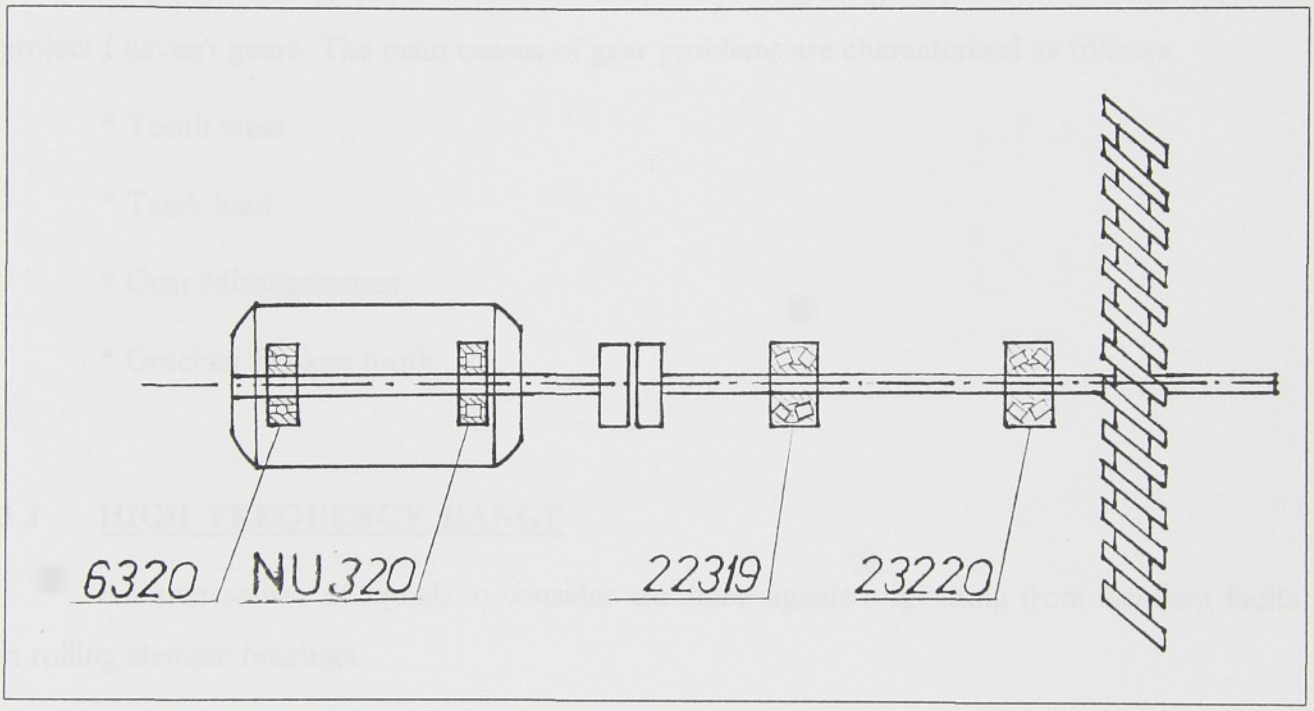


Fig. 4 Kinematic diagram of the smoke Ventilator

3.1 LOW FREQUENCY RANGE

Frequency spectra obtained from measurements made on the motor bearings, or bearings in the smoke ventilator will reveal low frequency Components at shafts revolution speed originating from unbalance, misalignments, bent shafts, etc.

3.1.1 Frequencies of shafts

The frequencies of shafts can be calculated by substituting in the following equation:

$$f_R = \frac{\text{Number of rotation of motor}}{60} = \frac{n_m}{60}$$

by substituting in the above equation we get the following result:

$$f_R = \frac{980}{60} = 16.333 \text{ [Hz]}$$

3.2 MEDIUM FREQUENCY RANGE

The cause of vibration in medium frequency range is gear problems. In my diploma project I haven't gears. The main causes of gear problems are characterised as follows

* Tooth wear

* Teeth load

* Gear Misalignement

* Gracked Broken tooth

3.3 HIGH FREQUENCY RANGE

Another source of signals to consider are those signals originating from incipient faults in rolling element bearings.

An incipient fault in rolling element bearing will typically by acrack or corrosion pit either on the inner race, on the outer race, on the cage or on the rolling element itself. This crack will create small impulses every time one of the rolling elements passes over it, and will transmit energy to the bearing housing, which in turn will vibrate at it's natural frequency (resonance frequency), and decay with the damping in the mechanical structure similar to striking a brass bell.

The ringing frequency of the bell depends only on it's dynamic properties and not on how it is struck or how hard. The structure acts in this way as a mechanical amplifier. If the fault is on the stationary race, assumed here to be the outer race, each pulse will be the same strength. If, however, the fault is on the rotating race, indicated here as the inner race, the pulse will vary with the changes in rolling element load. Fig. 5.

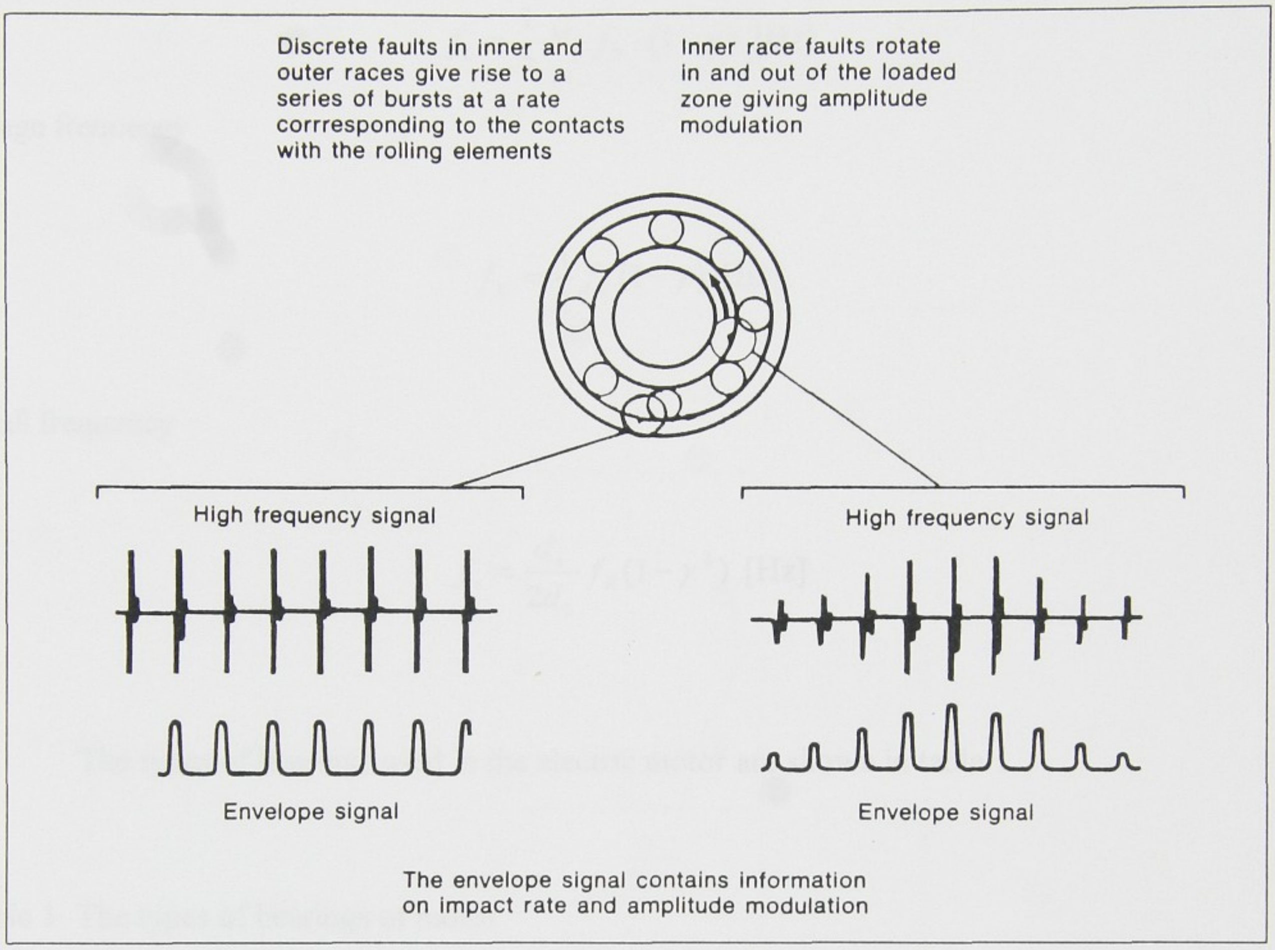


Fig. 5 Discrete faults in inner and outer races cause a series of bursts. A fault on the rotating race causes amplitude modulation.

3.3.1 Frequencies of bearings of the electric motor.

The frequencies of bearing can be calculated by substituting in the following equations:

Inner race frequency

$$f_i = \frac{1}{2} N \cdot f_R \cdot (1+\gamma) \text{ [Hz]}$$

Outer race frequency

$$f_0 = \frac{1}{2} N \cdot f_R \cdot (1-\gamma) \text{ [Hz]}$$

Cage frequency

$$f_K = \frac{1}{2} f_R (1-\gamma) \text{ [Hz]}$$

Ball frequency

$$f_v = \frac{d_s}{2d_0} f_R (1-\gamma^2) \text{ [Hz]}$$

The types of bearings used in the electric motor are shown in table 1.

Table 1. The types of bearings of motor

TYPE OF BEARING	d[mm]	D[mm]	B[mm]
6320	100	215	47
NU 320	100	215	47

TYPE 6320

Ball bearing

q_1 and q_2 are the dimensional coefficients of bearings and equal:

$q_1 = 0.33$

$q_2 = 0.95$

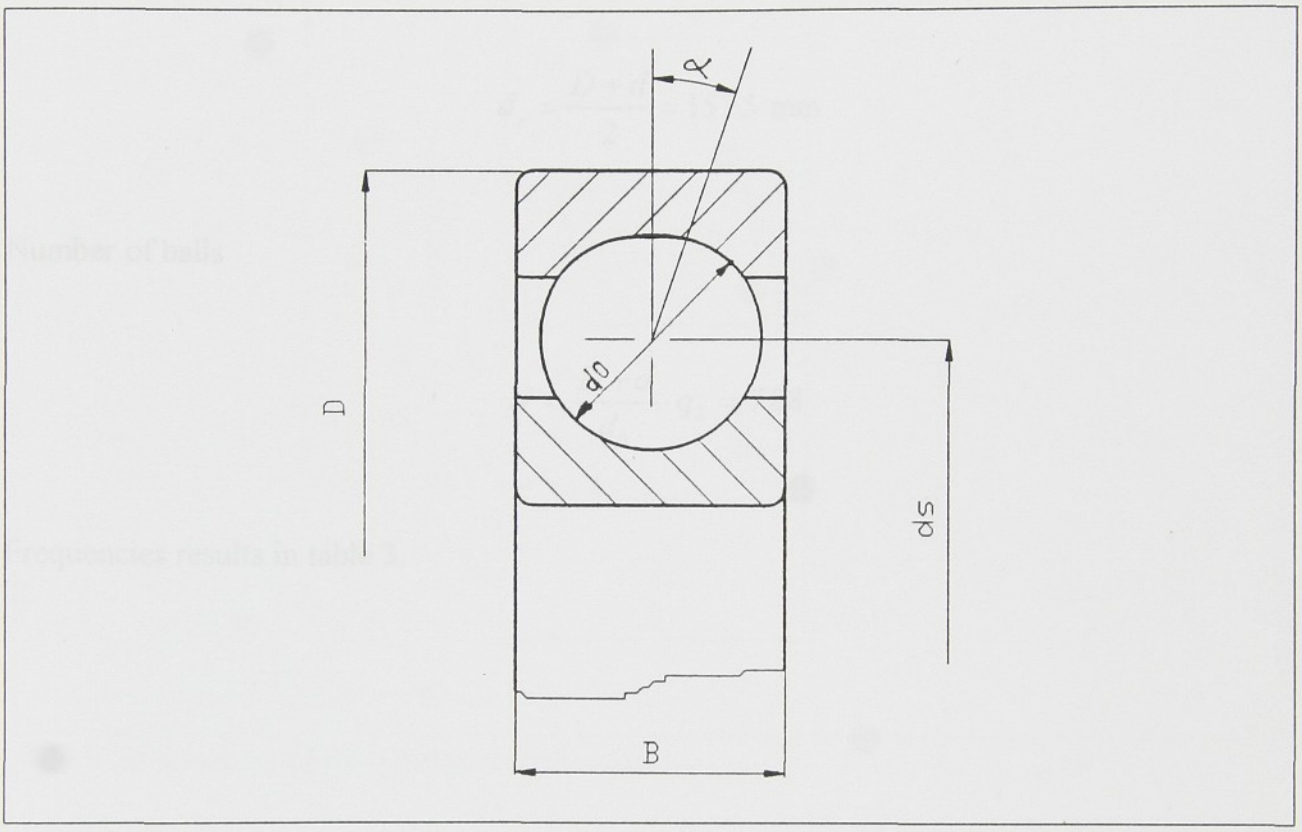


Fig. 6 Dimensions of the ball bearing

By substituting in the following equations we get

$$\gamma = \frac{d_0}{d_s} \cos\alpha = 0.240952$$

Contact angle

$$\alpha = 0$$

Ball diameter

$$d_0 = q_1(D - d) = 37.95 \text{ mm}$$

Pitch diameter

$$d_s = \frac{D + d}{2} = 157.5 \text{ mm}$$

Number of balls

$$N = \frac{D + d}{d_0} \cdot q_2 = 7.88$$

Frequencies results in table 3.

TYPE NU 320

Cylindrical roller bearing

$q_1 = 0.25$

$q_2 = 0.98$

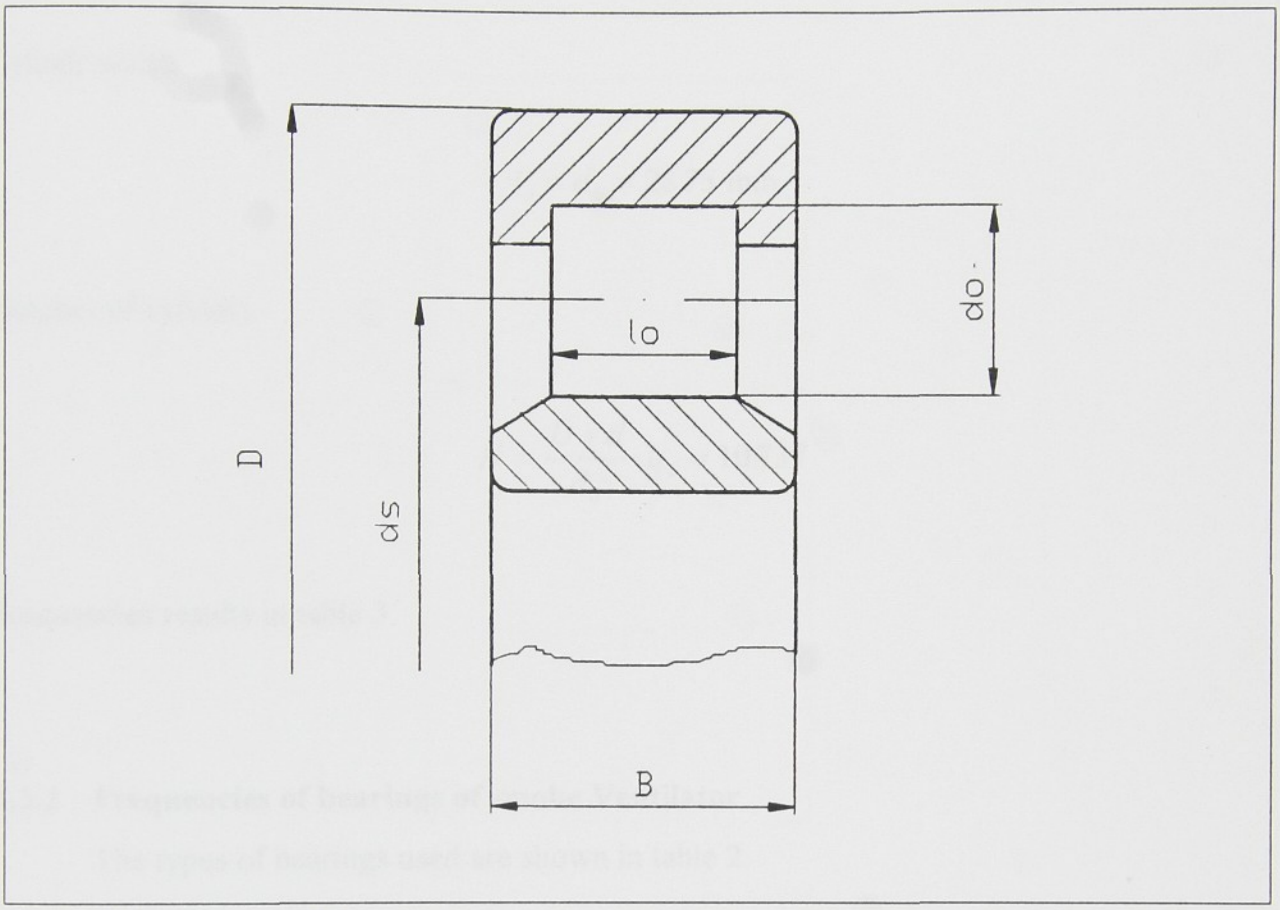


Fig. 7 Dimensions of the cylindrical roller bearing.

$$\gamma = \frac{d_0}{d_s}\cos\alpha = 0.182539$$

Contact angle

$$\alpha = 0$$

Pitch diameter

$$d_s = \frac{(D + d)}{2} = 157.5 \text{ mm}$$

Cylindr diameter

$$d_0 = q_1(D-d) = 28.75 \text{ mm}$$

Cylindr width

$$l_0 = d_0 = 28.75 \text{ mm}$$

Number of cylindrs

$$N = \frac{D+d}{d_0} \cdot q_2 = 10.737$$

Frequencies results in table 3.

3.3.2 Frequencies of bearings of smoke Ventilator

The types of bearings used are shown in table 2.

Table 2

TYPE OF BEARING	d[mm]	D[mm]	B[mm]
22319	95	200	67
23220	100	180	60.3

TYPE 22319

Spherical roller bearing

$q_1 = 0.25$

$q_2 = 1.2$

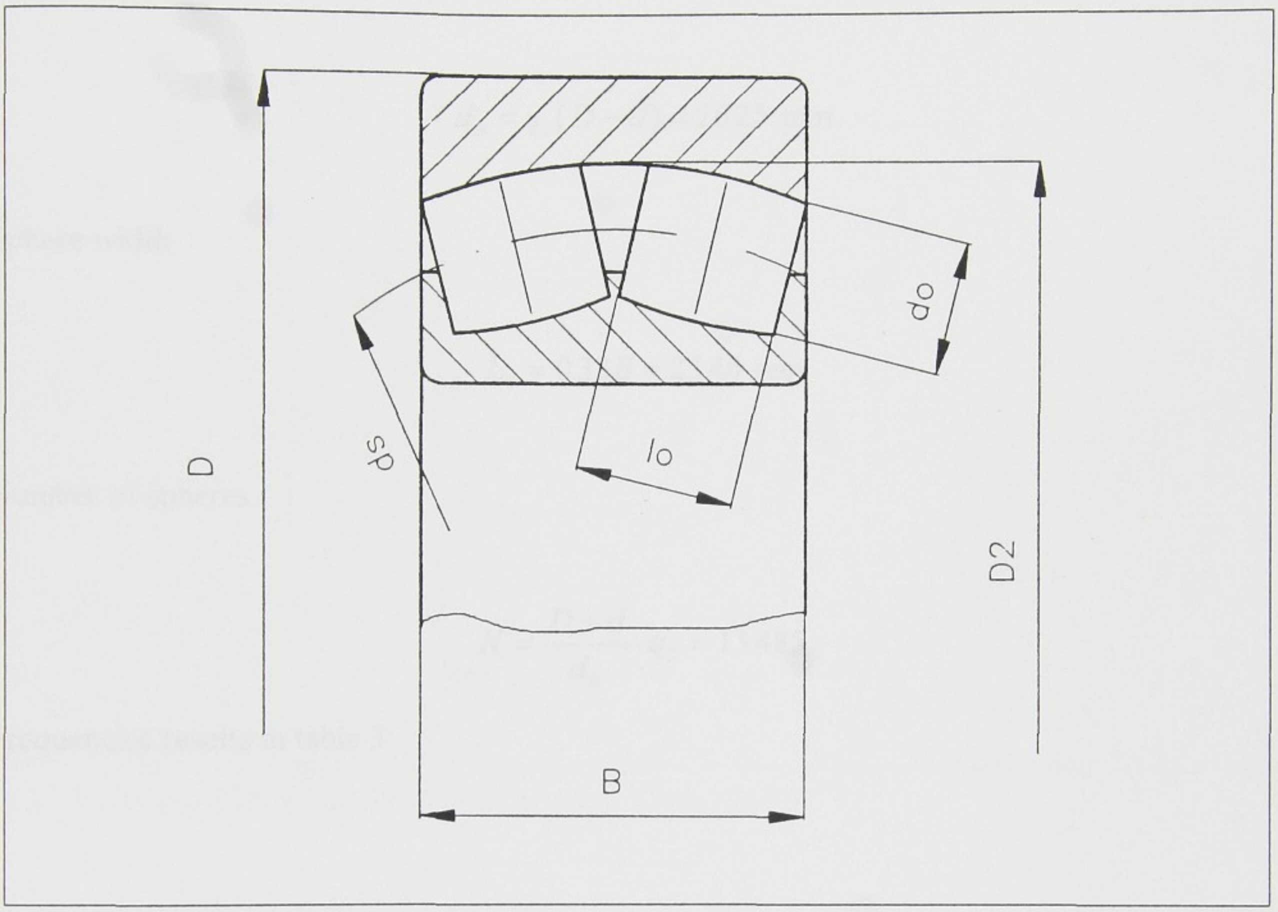


Fig. 8 Dimensions of spherical roller bearing Type 22319.

Contact angle

$$\alpha = 12.6^0$$

$$\gamma = \frac{d_0}{d_s} \cdot \cos\alpha = 0.17368$$

Pitch diameter

$$d_s = D_2 - d_0 = 147.5 \text{ mm}$$

$$D_2 = \frac{(D+d)}{2} + d_0 = 173.75 \text{ mm}$$

Sphere diameter

$$d_0 = q_1(D-d) = 26.25 \text{ mm}$$

Sphere width

$$L_0 = 0.38B = 25.46 \text{ mm}$$

Number of spheres

$$N = \frac{D+d}{d_0} \cdot q_2 = 13.485$$

Frequencies results in table 3.

TYPE 23220

Spherical roller bearing

$q_1 = 0.22$

$q_2 = 1.15$

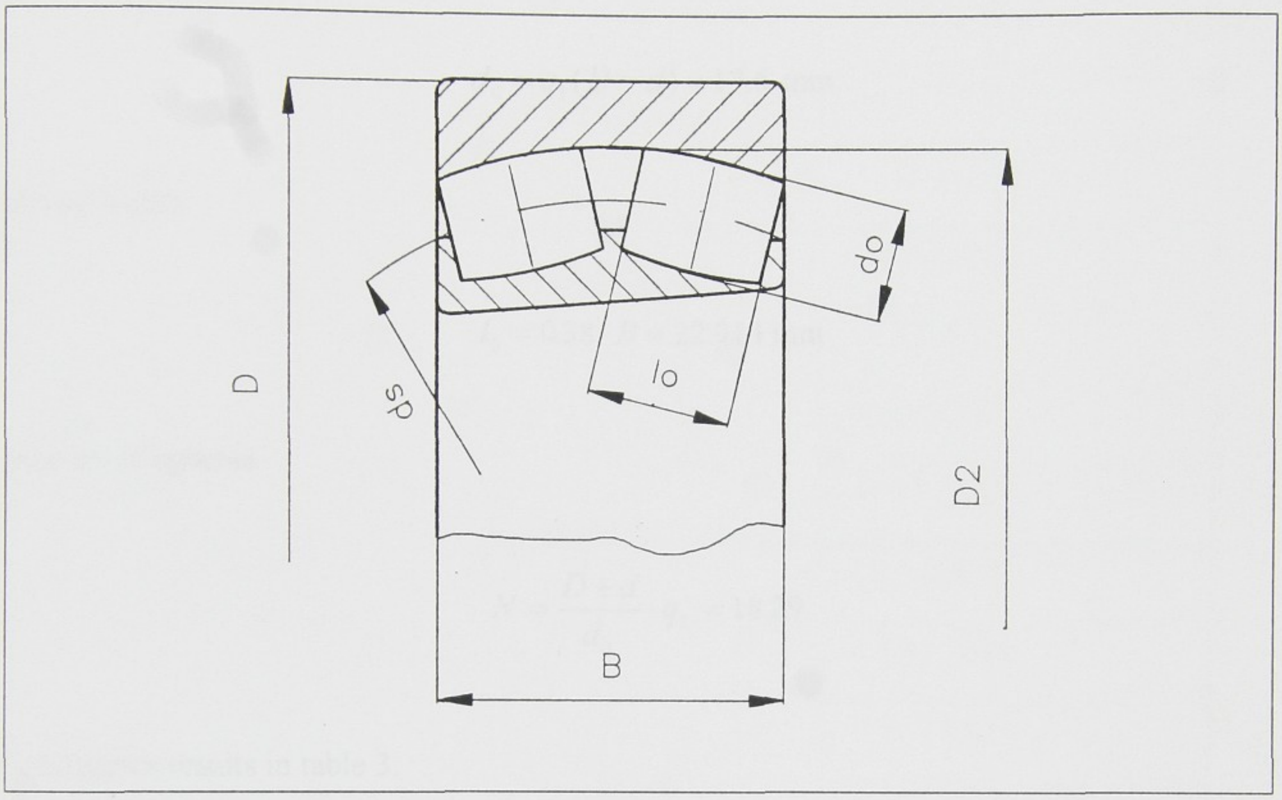


Fig. 9 Dimensions of spherical roller bearing Type 232 20

Contact angle

$$\alpha = 12.6^0$$

$$\gamma = \frac{d_0}{d_s} \cdot \cos\alpha = 0.1226866$$

Pitch diameter

$$d_s = D_2 - d_0 = 140 \text{ mm}$$

$$D_2 = \frac{D+d}{2} + d_0 = 157.6 \text{ mm}$$

Sphere diameter

$$d_0 = q_1(D-d) = 17.6 \text{ mm}$$

Sphere width

$$l_0 = 0.38 \cdot B = 22.914 \text{ mm}$$

Number of spheres

$$N = \frac{D+d}{d_0} \cdot q_2 = 18.29$$

Frequencies results in table 3.

Table 3. The results of bearings frequencies

Type of bearing	Frequencies of bearing [Hz]
6320	$f_i = 79.908$
	$f_0 = 48.877$
	$f_k = 6.19837$
	$f_v = 31.9248$
NU 320	$f_i = 103.689$
	$f_0 = 71.679$
	$f_k = 6.6757$
	$f_v = 43.247$
22319	$f_i = 258.503$
	$f_0 = 181.998$
	$f_k = 6.7481$
	$f_v = 44.503$
23220	$f_i = 335.0985$
	$f_0 = 262.8894$
	$f_k = 7.1645$
	$f_v = 63.983$

4. FAULT DETECTION FOR A SMOKE VENTILATOR

To avoid tedious and expensive signal analysis in day-to-day work, it is essential to use a method of detection that:

1 - Detects the majority of faults as early as possible.

2 - Gives as few false detection as possible.

3 - Is so easy to use that even a layman can carry it out.

4 - Gives enough information to make aqualified judgement about the fault to enable management to evaluate when amore through analysis must be made.

For many years, fault detection has been carried out by comparing RMS readings of vibration velocity with preceding readings, or with established standards. (International standards ISO 2372 and 3945)

The theory is that smoke ventilators, of similar size and grouped after shaft power, will have similar of even the same vibration level, measured in velocity filtered from 10 Hz to 1kHz see Fig. 10.

As has been discussed before, monitoring based on this system will presumably identify unbalance, coarse misaligment and severely bent shafts because of the high energy in these signals.

As low energy signals will be buried in the more powerful vibration signals, incipient bearing faults will not be detected by this method.

At least they will not be detected before they generate a vibration signal higher than the low frequency domain signals.

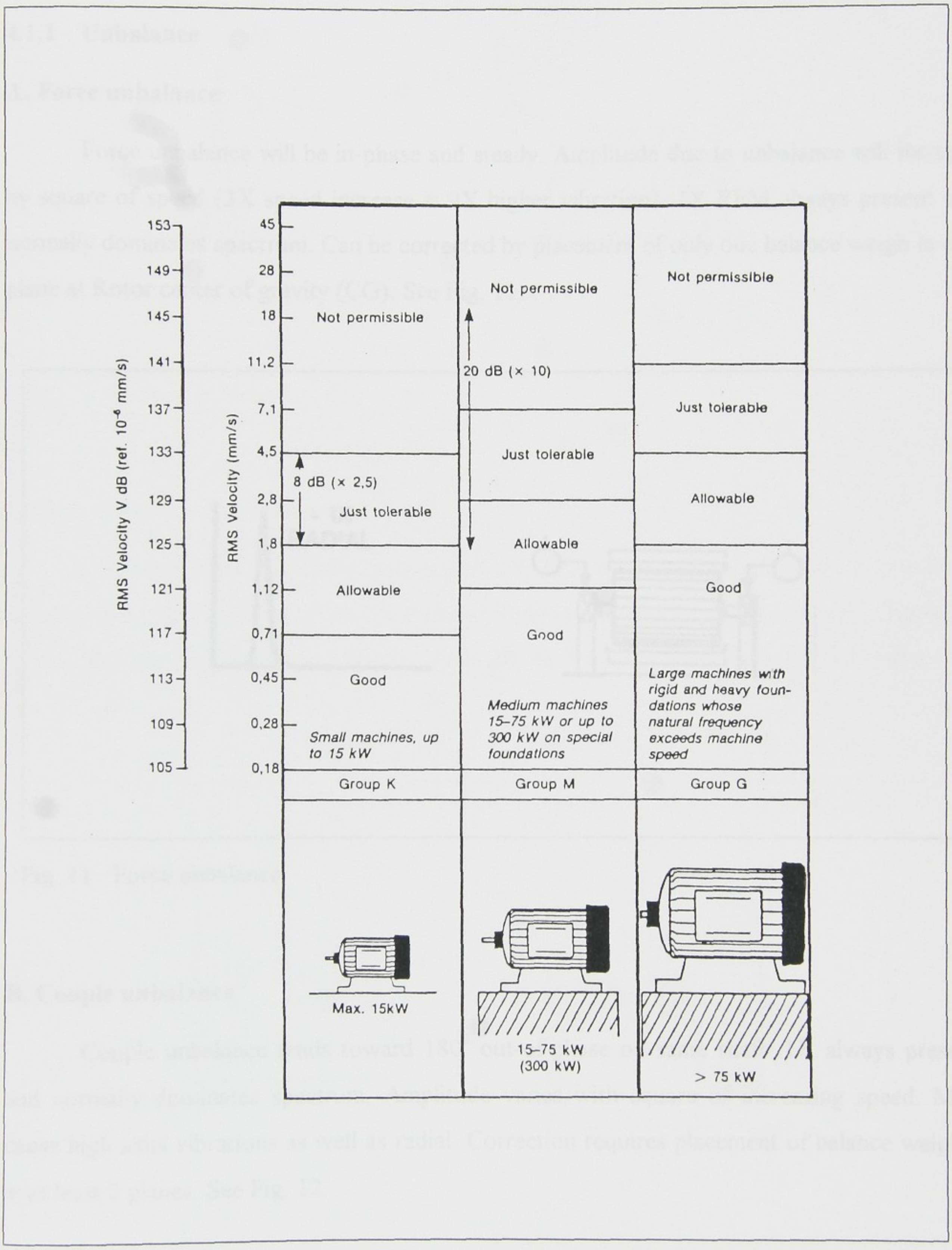


Fig. 10 Classification of machines according to vibration levels.

4.1 FAULTS IN ROTATING SHAFTS

4.1.1 Unbalance

A. Force unbalance

Force unbalance will be in-phase and steady. Amplitude due to unbalance will increase by square of speed (3X speed increase = 9X higher vibration). 1X RPM always present and normally dominates spectrum. Can be corrected by placement of only one balance weigh in one plane at Rotor center of gravity (CG). See Fig. 11.

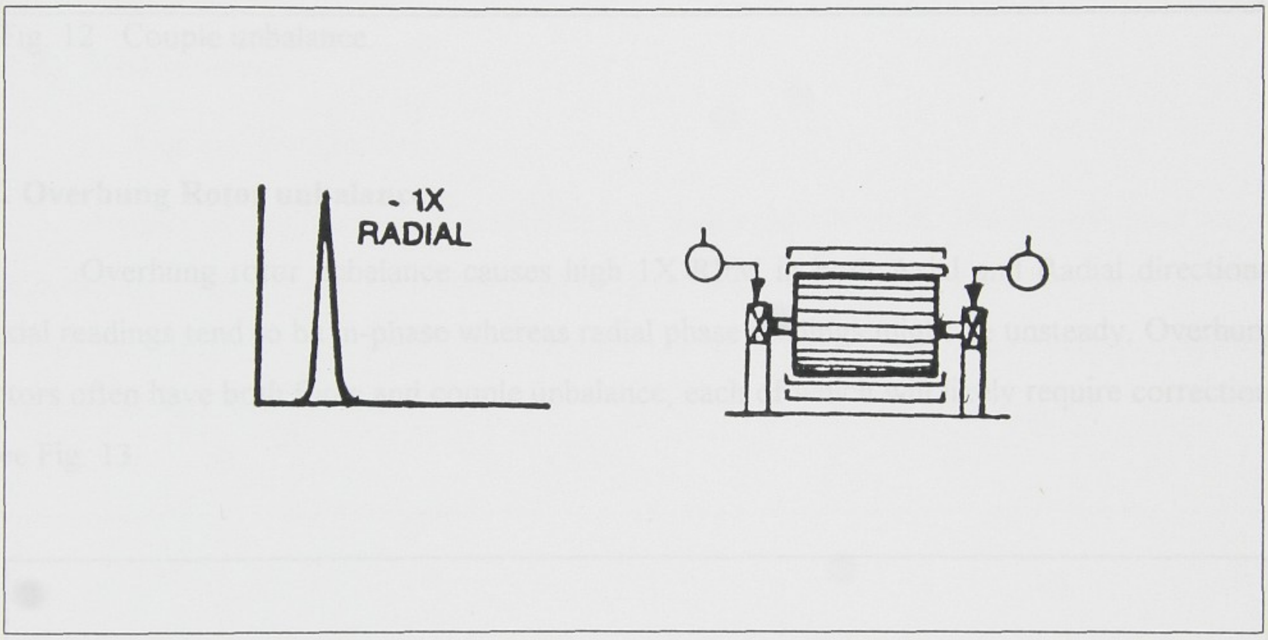


Fig. 11 Force unbalance

B. Couple unbalance

Couple unbalance tends toward 180^0 out-of-phase on same shaft. 1X always present and normally dominates spectrum. Amplitude varies with square of increasing speed. May cause high axial vibrations as well as radial. Correction requires placement of balance weights in at least 2 planes. See Fig. 12.

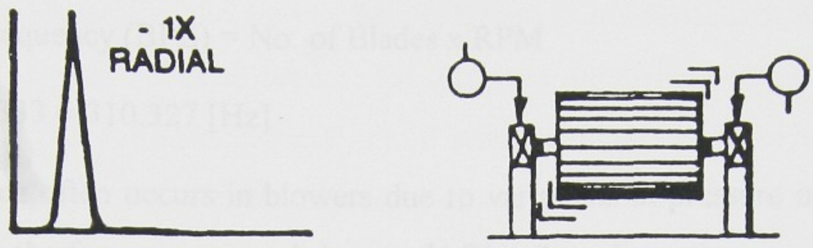


Fig. 12 Couple unbalance.

C. Overhung Rotor unbalance

Overhung rotor unbalance causes high 1X RPM in both Axial and Radial directions. Axial readings tend to be in-phase whereas radial phase readings might be unsteady. Overhung rotors often have both force and couple unbalance, each of which will likely require correction. See Fig. 13.

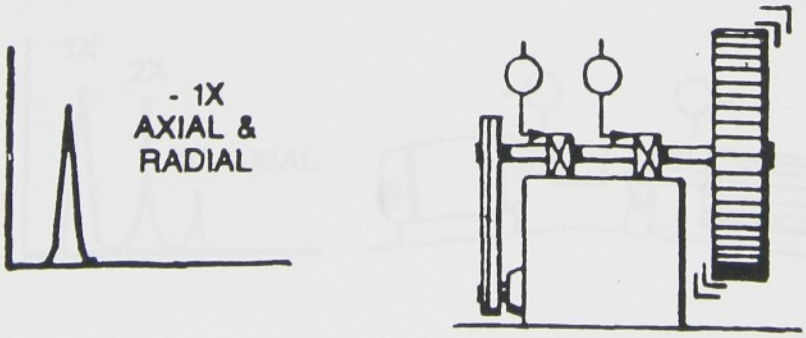


Fig. 13 Overhung Rotor unbalance.

D. Balance of fan and flow turbulence.

In our case of smoke ventilator we have one fan with diameter 1400 mm. To calculate its Balancing we shall use this equation:

Blade Passes Frequency (BPE) = No. of Blades x RPM

BPF = 19 x 16.333 = 310.327 [Hz]

Flow turbulence often occurs in blowers due to variations in pressure or velocity of the air passing through the fan or connected ductwork. This flow disruption causes turbulence which will generate random, low frequency vibration. Typically in the range of 50 to 2000 CPM.

4.1.2 Misalignment:

A. Angular misalignment

Angular misalignment is characterized by high axial vibration, 180 out-of-phase across the coupling. Typically will have high axial vibration with both 1X and 2X RPM. However, not unusual for either 1X, 2X or 3X to dominate. These symptoms may also indicate coupling problems as well. See Fig. 14.

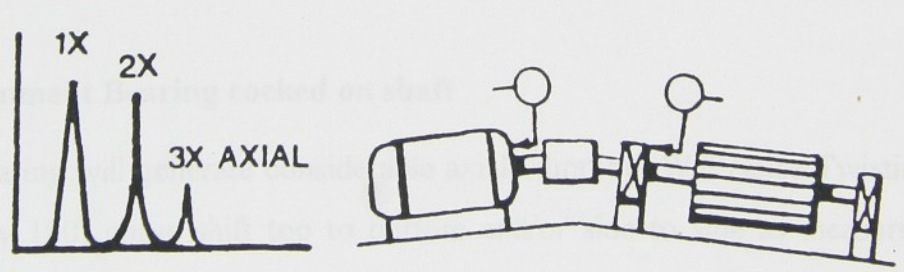


Fig. 14 Angular misalignment.

B. Parallel misalignment

Offset Misalignment has similar vibration symptoms to Angular, but shows high radial vibration which approaches 180° out-of-phase across coupling. 2X often larger than 1X, but its height relative to 1X is often dictated by coupling type and construction. When either Angular or Radial Misalignment becomes severe, can generate either high amplitude peaks at much higher harmonics (4X - 8X) or even a whole series of high frequency harmonics similar in appearance to mechanical looseness. Coupling construction will often greatly influence shape of spectrum when misalignment is severe. See Fig. 15.

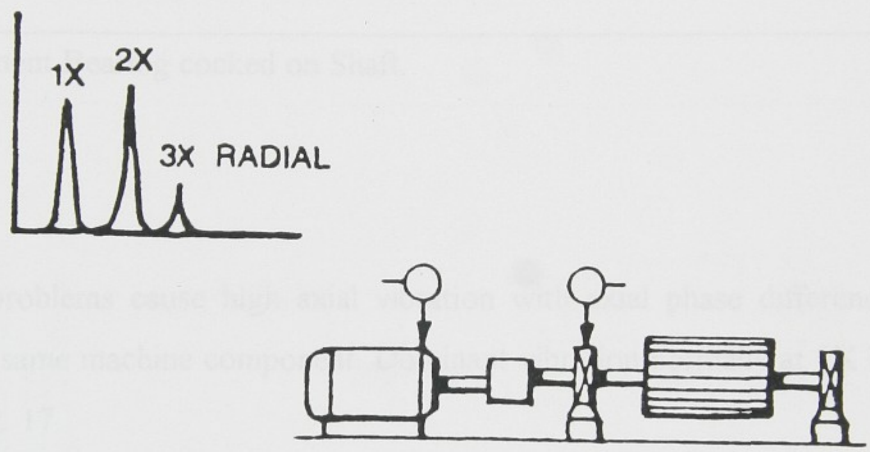


Fig. 15 Parallel misalignment.

C. Misalignment Bearing cocked on shaft

Cocked bearing will generate considerable axial vibration. Will cause Twisting Motion with approximately 180° phase shift top to bottom and/or side to side as measured in axial direction of same bearing housing.Attempts to align coupling or balance the rotor will not alleviate problem. Bearing must be removed and correctly installed. See Fig. 16.

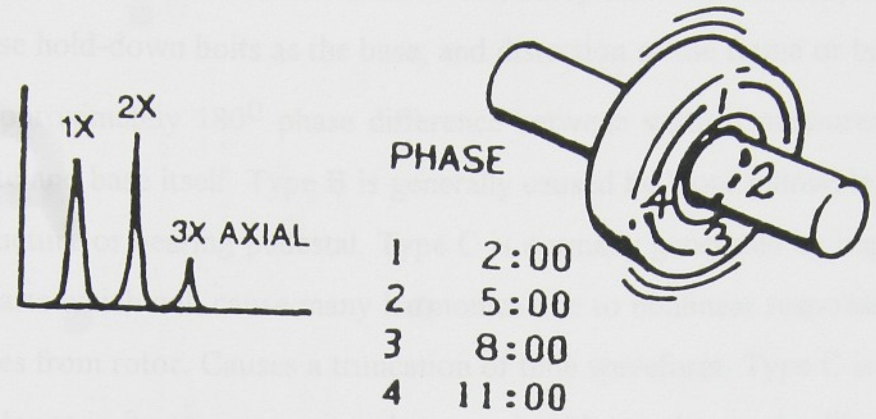


Fig. 16 Misalignment Bearing cocked on Shaft.

4.1.3 Bent shaft

Bent shaft problems cause high axial vibration with axial phase differences tending toward 180° on the same machine component. Dominant vibration normally at 1X if bent near shaft center. See Fig. 17.

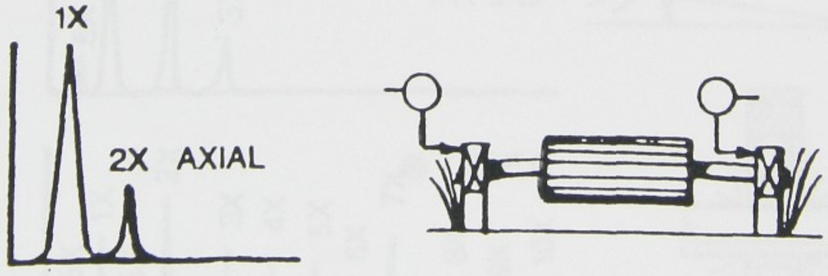


Fig. 17 Bent shaft

4.2 MECHANICAL LOOSENESS

Mechanical Looseness is indicated by either Type A, B or C spectra. Type A is caused by Structural looseness/weakness of machine feet, baseplate or foundation; also by deteriorated grouting, loose hold-down bolts as the base; and distortion of the frame or base. Phase analysis may reveal approximately 180^0 phase difference between vertical measurements on machine foot, baseplate and base itself. Type B is generally caused by loose pillowblock bolts, cracks in the frame structure or bearing pedestal. Type C is normally generated by improper fit between component parts which will cause many harmonics due to nonlinear response of loose parts to dynamic forces from rotor. Causes a truncation of time weveform. Type C is often caused by a bearing liner loose in its cap, excessive clearance in either a sleeve or rolling element bearing, or aloose impeller on a shaft. See Fig. 18 and Fig. 19.

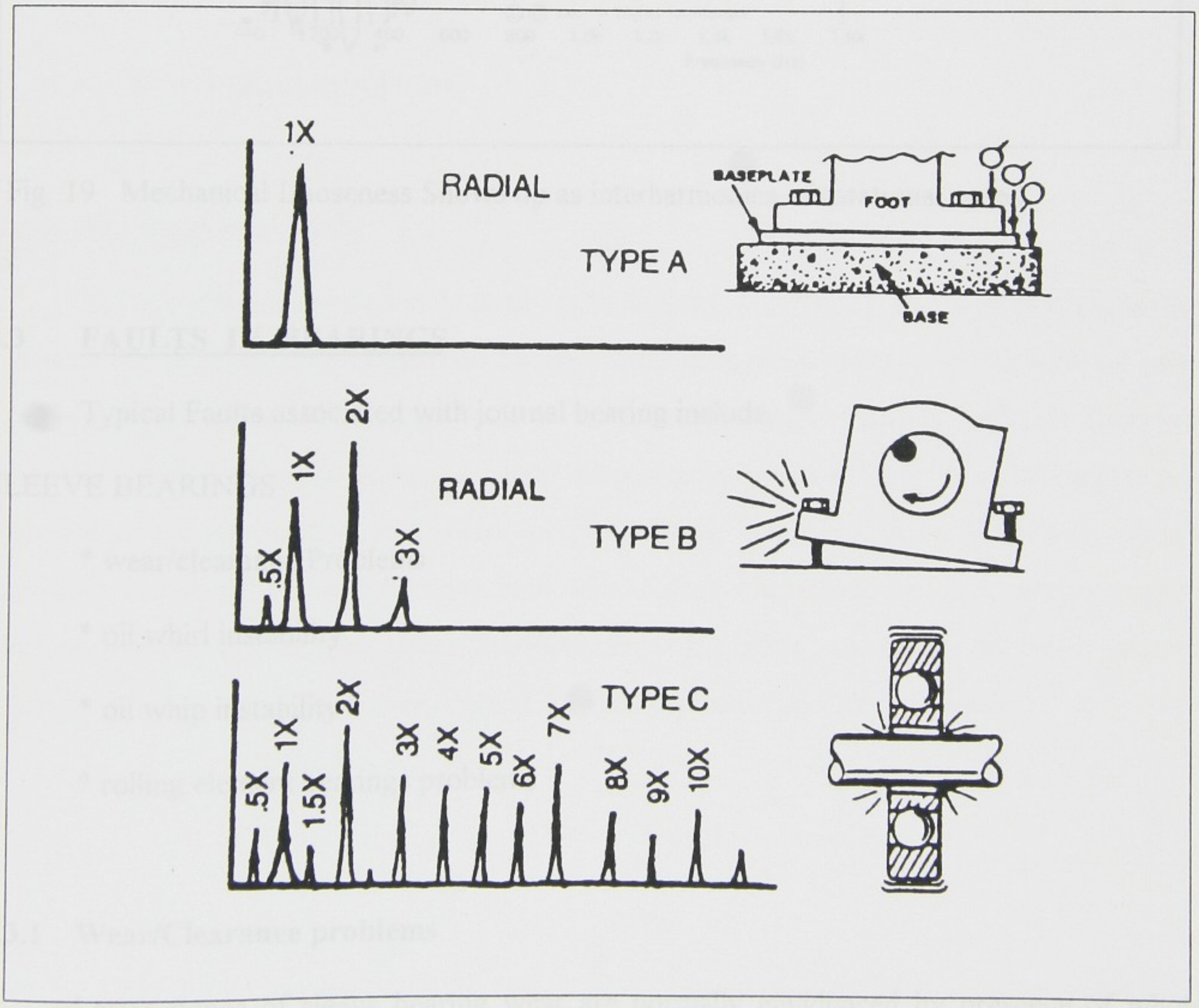


Fig. 18 Mechanical Looseness Types

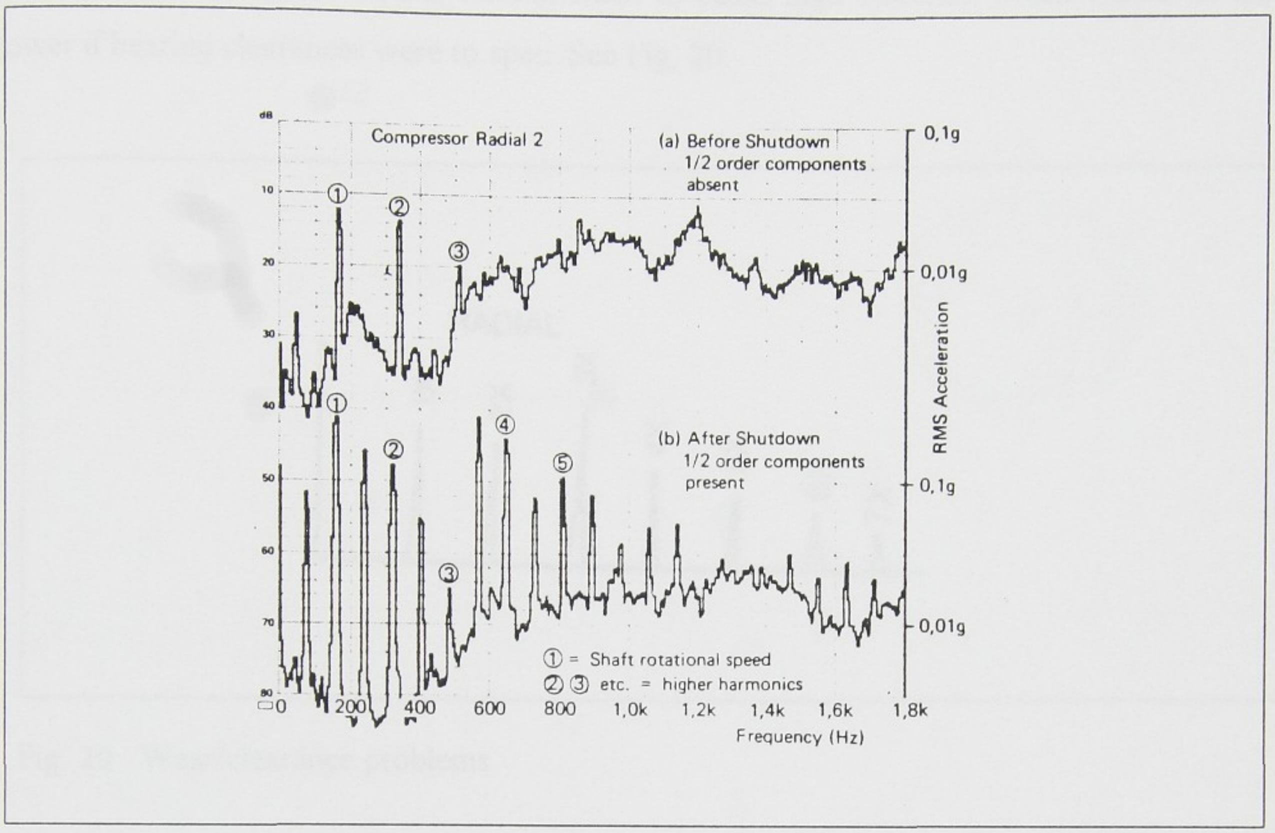


Fig. 19 Mechanical Looseness Shows up as interharmonics of rotational speed

4.3 FAULTS IN BEARINGS

Typical Faults associated with journal bearing include:

SLEEVE BEARINGS

* wear/clearance Problems

* oil whirl instability

* oil whip instability

* rolling element bearings problems

4.3.1 Wear/Clearance problems

Latter stages of sleeve bearing wear are normally envidenced by presence of whole series of running speed harmonics (up to 10 or 20). Wiped sleeve bearings often will allow high vertical amplitudes compared to horizontal. Sleeve bearings with excessive clearance may allow a minor unbalance and/or misalignment to cause high vibration which would be much lower if bearing clearances were to spec. See Fig. 20.

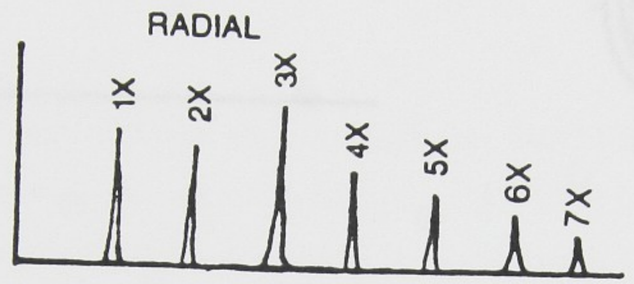


Fig. 20 Wear/clearance problems

4.3.2 Oil whirl instability

Oil Whirl instability occurs at 0.42 - 0.48 X RPM, see Fig. 21, and is often quite severe. Considered excessive when amplitude exceeds 50% of bearing clearances. Oil Whirl is an oil film excited vibration where deviations in normal operating conditions (attitude angle and eccentricity ratio) cause oil wedge to "push" shaft around within bearing. Destabilizing force in direction of rotation results in a whirl (or precession). Whirl is inherently unstable since it increases centrifugal forces which increase whirl forces. Can cause oil to no longer support shaft, or can become unstable when whirl frequency coincides with a rotor natural frequency. Changes in oil viscosity, lube pressure and external preloads can affect oil whirl.

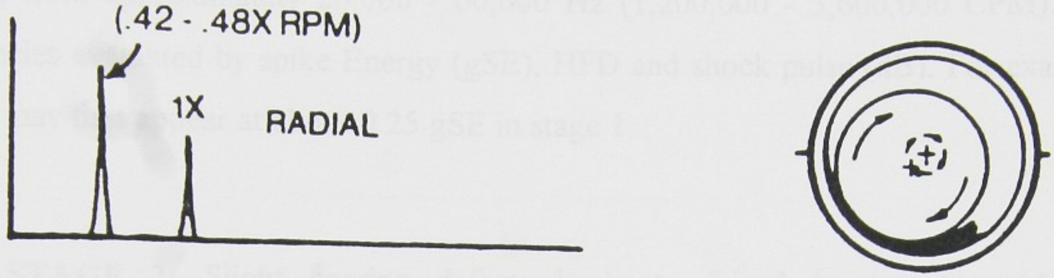


Fig. 21 Oil whirl instability

4.3.3 Oil whip

Oil Whip may occur if machine operated at or above 2X rotor critical frequency. When rotor brought up to twice critical speed, whirl will be very close to rotor critical and may cause excessive vibration that oil film may no longer be capable of supporting. Whirl speed will actually "lock onto" rotor critical and this peak will not pass thru it even if machine brought to higher and higher speeds. Fig. 22.

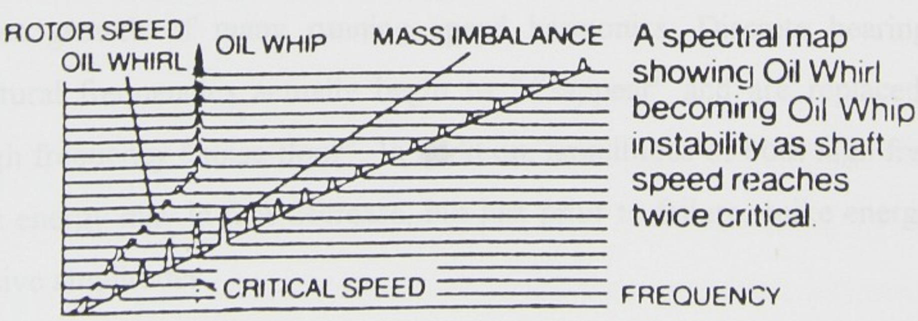


Fig. 22 Oil whip

4.3.4 Rolling element bearings

4 Failure stages:

STAGE 1: Earliest indications of bearing problems appear in ultrasonic frequencies ranging from approximately 20,000 - 60,000 Hz (1,200,000 - 3,600,000 CPM). These are frequencies evaluated by spike Energy (gSE), HFD and shock pulse (dB). For example, spike energy may first appear at about 0.25 gSE in stage 1.

STAGE 2: Slight bearing defects begin to "ring" bearing component natural frequencies (f_n) which predominantly occur in 30K - 120K CPM range. Sideband frequencies appear above and below natural frequency peak at end of stage 2. Spike energy grows (for example, from 0.25 to 0.5 gSE).

STAGE 3: Bearing defect frequencies and harmonics appear when wear progresses, more defect frequency harmonics appear and number of sidebands grow, both around these and around bearing natural frequencies. Spike energy continues to increase (for example, from 0.5 to over 1 gSE). Wear is now usually visible and may extend throughout periphery of bearing, particularly when well formed sidebands accompany bearing defect frequency harmonics.

STAGE 4: Towards the end, amplitude of 1X RPM is even effected. It grows, and normally causes growth of many running speed harmonics. Discrete bearing defect and component natural frequencies actually begin to "disappear" and are replaced by random, broad band high frequency "noise floor". In addition, amplitudes of both high frequency noise floor and spike energy may in fact decrease, but just prior to failure, spike energy will usually grow to excessive amplitudes.

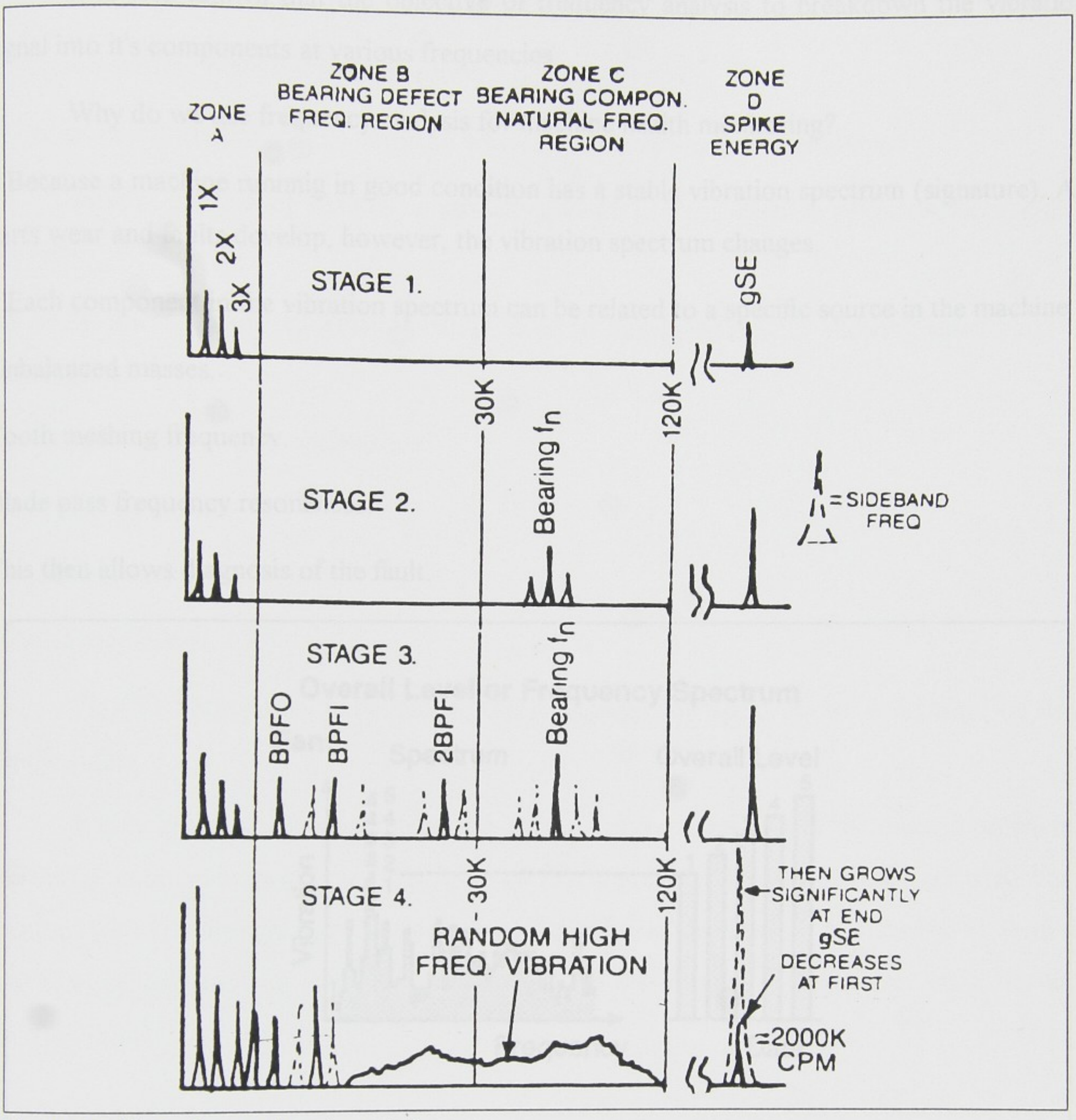


Fig. 23 Rolling element bearings failure stages.

4.4 WHAT DOES A SIGNAL FROM ROTATING MACHINE LOOK LIKE?

The basis of fault diagnosis is that different faults in a machine will manifest themselves at different frequencies in the vibration spectrum. The frequency domain information can then be related to periodic events in bearings, blades or lobes, etc.

Note that fault diagnosis depends on having a knowledge of the machine in question, that is the shaft frequencies, bearing geometries, etc.

It is well known that the objective of frequency analysis to breakdown the vibration signal into it's components at various frequencies.

Why do we use frequency analysis for machine health monitoring?

* Because a machine runnnig in good condition has a stable vibration spectrum (signature). As parts wear and faults develop, however, the vibration spectrum changes.

* Each component in the vibration spectrum can be related to a specific source in the machine.

Unbalanced masses.

Tooth meshing frequency.

Blade pass frequency resonances.

This then allows diagnosis of the fault.

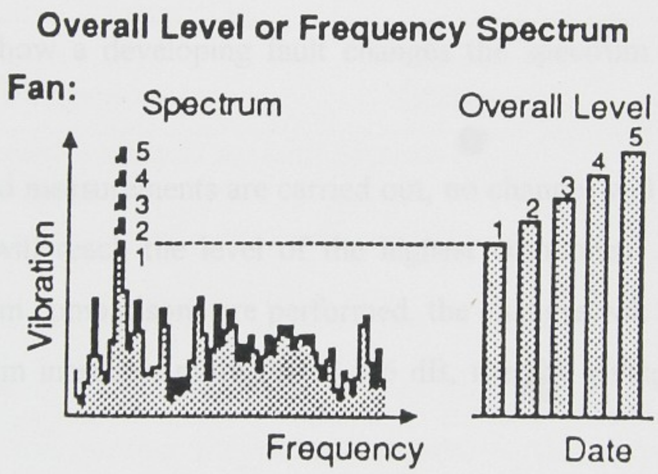


Fig. 24 Overall level or Frequency spectrum

4.5 BENEFITS OF FREQUENCY ANALYSIS FOR FAULT DETECTION

Simple vibration meter measures the overall vibration level over a wide frequency range. The measured level reflects the vibration amplitude of dominating frequency components. Which of course are important to monitor.

Although overall vibration measurement proves:

1. A good starting point for fault detection.

2. Far more information can be obtained when a frequency analysis is employed.

Firstly, a frequency analysis will usually give far earlier indication of the development of the fault.

Secondly, the frequency information can be used to diagnose the fault allowing spare parts to be bought in.

The benefits of frequency analysis are:

- Early fault detection

- Fault diagnosis

- Trend prediction

Fig. 25 Shows how a developing fault changes the spectrum as a function of time (upper right).

If only broadband measurements are carried out, no changes will be detected before the increasing component will reach the level of the highest component in the measured band (bottom left). If spectrum comparisons are performed, the changes will be detected as soon as one band in the spectrum increases more than 3 - 6 dB, thereby giving a very early warning (bottom right).

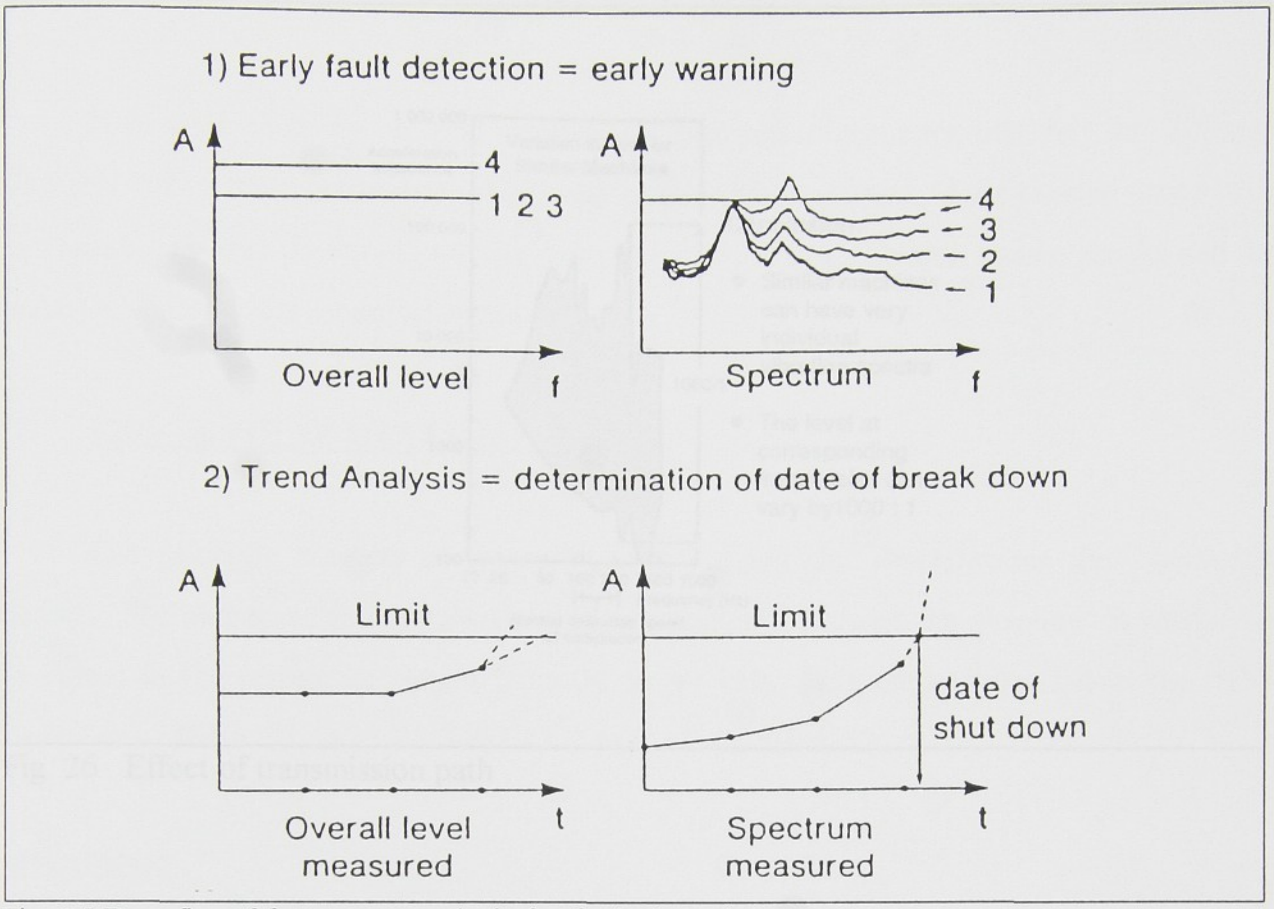


Fig. 25 Benefits of frequency analysis for fault detection

4.6 EFFECT OF TRANSMISSION PATH

Another problem in using fixed standards as limits for the vibration levels is the transmission path from the exciting point (bearing race) to the measurement point. If this transmission path does not have the same attenuation from unit to unit, then the same degree of fault, the same pitting or cracks, can give different readings from the vibration transducer, and consequently give misleading interpretations and conclusions.

From Fig. 26 it can be seen that the transmission path attenuation, here quantified as acceleration impedance, of similar machines can vary considerably at certain frequencies by up to 1:1000 or 60 dB.

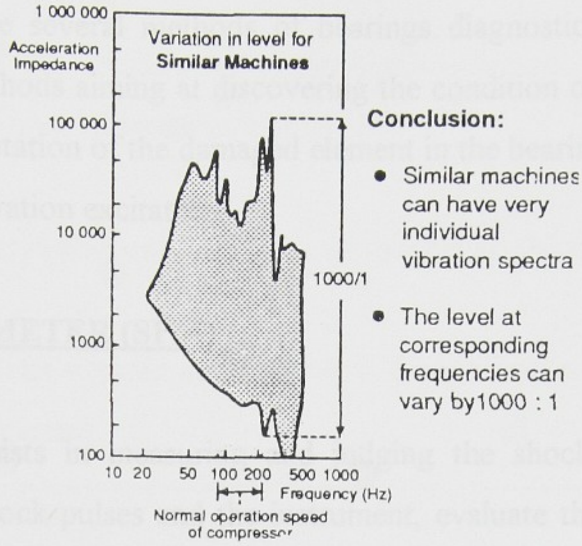


Fig. 26 Effect of transmission path

We may therefore conclude that detection must be carried out by:

- comparing current results with those obtained over a period of time. Readings froM other machines or other measurement points can not give early indication of change in the condition of the rotating machine.

- performing spectrum comparison to detect changes also at lower levels in the spectrum.

5. 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 excitater.

5.1. 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.

Advantages:

- The method is precise, quick, and permit to find out the quality of bearings lubrication.

Disadvantages:

- We have to know exactly the parameters of bearings, and it is exacting in directing accelerometers and in maintaining the measurement conditions.

5.2. 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 - Figure 27. It is obvious from Fig. 27 that 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

Advantages:

- After measuring the bearings absolute vibration, we can define SE as cheap and quick.

Disadvantages:

- In measurements carried out with an accelerometer, the estimation of the damaged bearing depends on the diagnostic person.

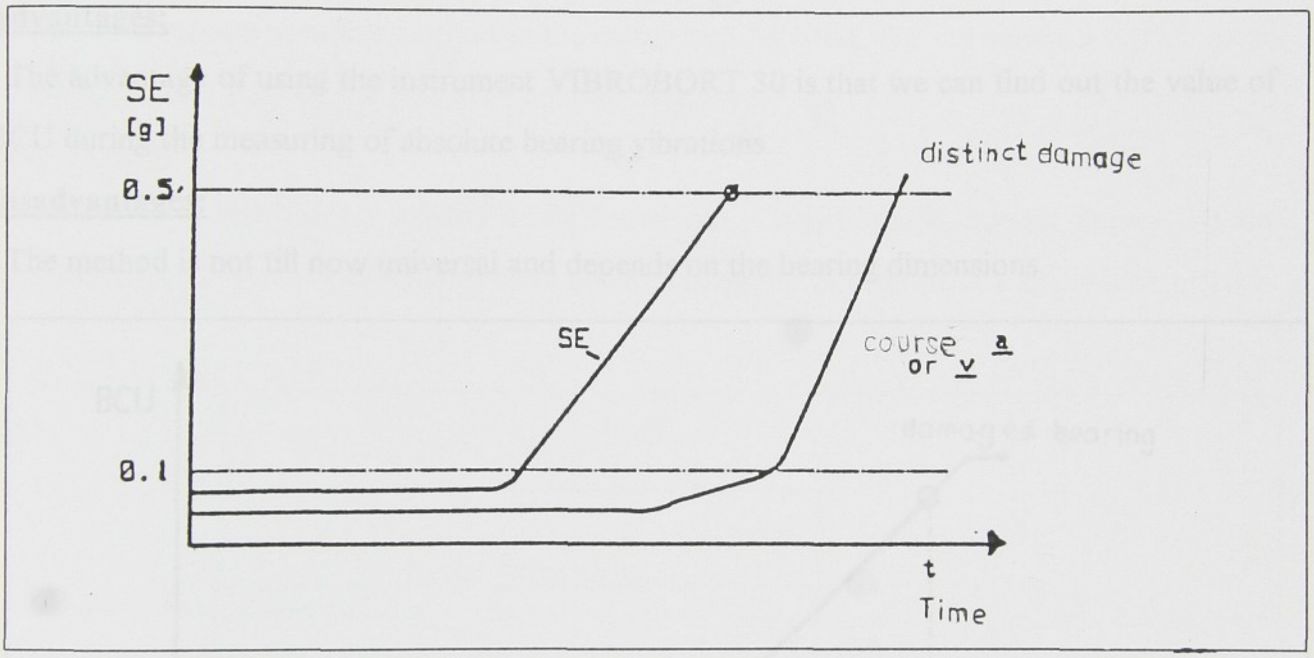


Figure 27. Typical course of spike energy SE , acceleration a or velocity v bearing vibration in relation with time.

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 cavitation) or complex gearboxes (with high frequency gearmesh harmonics).

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

5.3. METHOD BCU (BEARING CONDITION UNIT)

This method is used in the instruments of SCHENCK company. The shock pulses of the damaged bearing excites the vibration in the accelerometer placed on the bearing in the resonance range from 36 to 38 kHz. The principle of this method is used in instrument VIBROCAM 1000 and VIBROBORT 30. Quantity trend BCU is in figure 28.

Advantages:

- The advantage of using the instrument VIBROBORT 30 is that we can find out the value of BCU during the measuring of absolute bearing vibrations.

Disadvantages:

- The method is not till now universal and depends on the bearing dimensions.

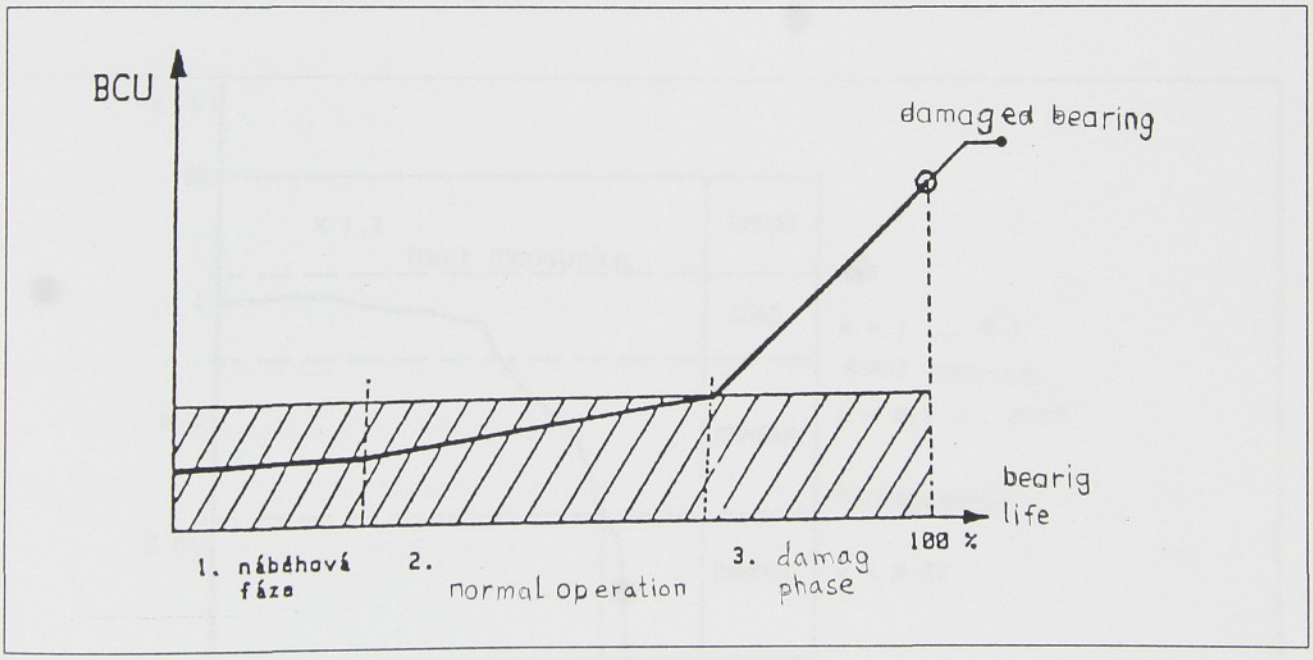


Figure 28. Quantity trend of the bearing condition.

5.4. METHOD K(t) PARAMETER

The method was defined by Prof. Sturmem:

$$K(t) = \frac{a_{ef}(0) \cdot a_v(0)}{a_{ef}(t) \cdot a_v(t)}$$

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. 29 shows a typical cause of the parameter K(t) on the period of bearing operation.

Advantages:

K(t) method is reliable, no validity limits found out, simple, quick, doesn't depend on the measuring direction, the instruments have a wide possibility of use and at a reasonable price.

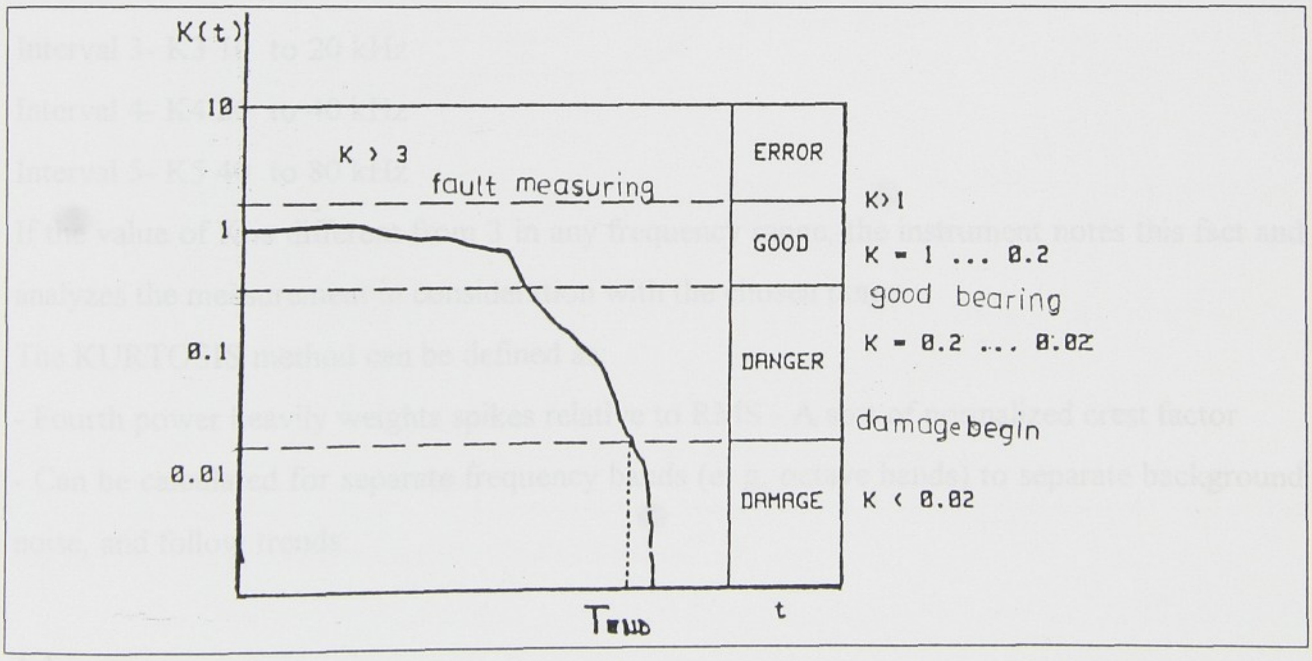


Fig. 29 The dependance of K(t) parameter dimension on the condition of the bearing.

5.5. KURTOSIS METHOD

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

$$Formula\beta_2 = \frac{\int_{-\infty}^{\infty} (x - \bar{x})^4 p(x) dx}{\sigma^4}$$

where:

x = signal amplitude, $\overline{x}$ = 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 an 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 analyzes 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.

Advantages:

- The light portable instrument gives a wider possibility of gaining information about the watched bearing.

Disadvantages:

- Time exacting measurement of gears under 1000 RPM, uncertain estimation of double rows cylinder and spherical bearing, impossibility of measuring "big bearings" as it is written in the instructions of measurement where the dimensions of bearings are not defined.

5.6. CREST FACTOR METHOD

The crest factor K_v is the proportion of the peak to the effective acceleration value. This proportion is evaluated with time sequences in the frequency range 10 Hz to 10 kHz.

$$K_v = \frac{a_v}{a_{ef}}$$

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

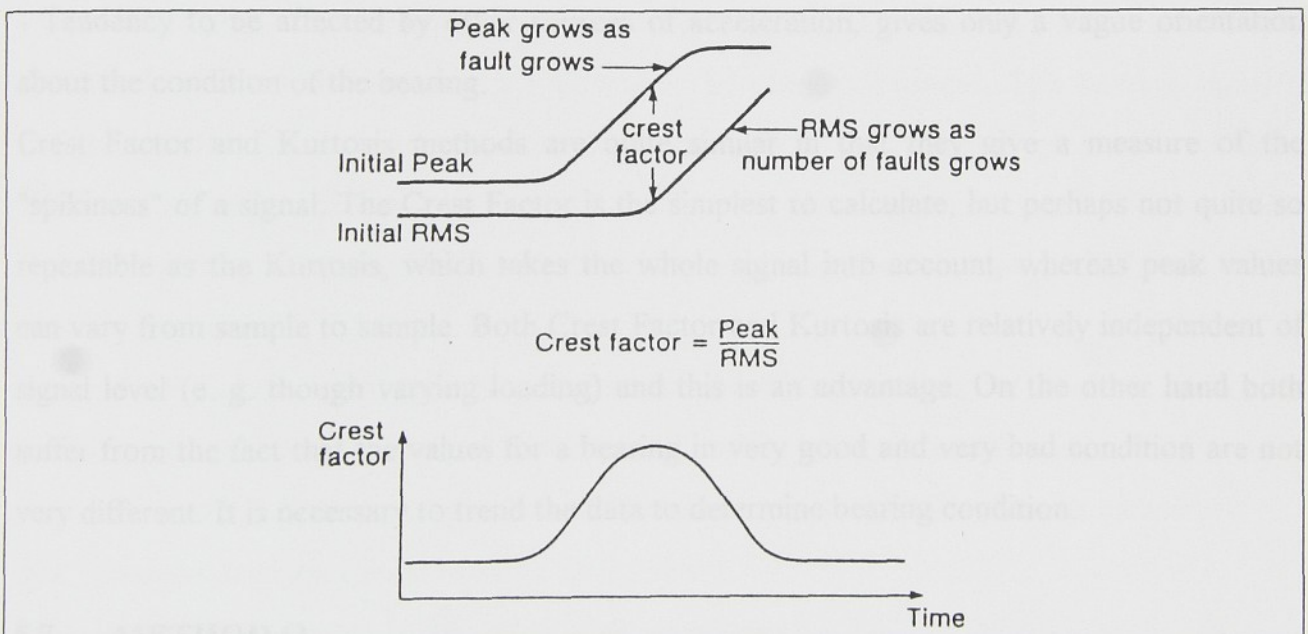


Figure 30. The principle of Crest - Factor method

The curve in Figure 30 shows Crest Factors typical trend as the function of the damaged bearing condition deteriorates.

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.

Advantages:

- Simple method, low charges.

Disadvantages:

- Tendency to be affected by other sources of acceleration, gives only a vague orientation about the condition of the bearing.

Crest Factor and Kurtosis methods are quite similar in that they give a measure of the "spikiness" of a signal. The Crest Factor is the simplest to calculate, but perhaps not quite so repeatable as the Kurtosis, which takes the whole signal into account, whereas peak values can vary from sample to sample. Both Crest Factor and Kurtosis are relatively independent of signal level (e. g. though varying loading) and this is an advantage. On the other hand both suffer from the fact that the values for a bearing in very good and very bad condition are not very different. It is necessary to trend the data to determine bearing condition.

5.7. METHOD Q

It is a new method in evaluating the technical condition of bearings. The instrument Diagnost 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.

Advantages:

- The bearing evaluation doesn't depend on the operational speed, the dimension of the bearing and the load , a quick determination of the bearing condition, a precise diagnosis in seconds.

Disadvantages:

- One - purpose instrument, Produced only in few prototypes (year 1989).

5.8. METHOD ENVELOPE SPECTRUM

The principle of this method is used in the analyzer 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 resonances, 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 resonances. This bearing signal is masked by other background vibrations from the machine, and the basic problem is to find a frequency range where the bearing signal is dominant over the background vibration. Note that the repetition frequency is better indicated by analyzing the envelope of the bursts, rather than the raw signal.

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.

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

Use of an FFT Analyzer 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 - analyze envelope signal as a base band analysis to obtain repetition frequencies

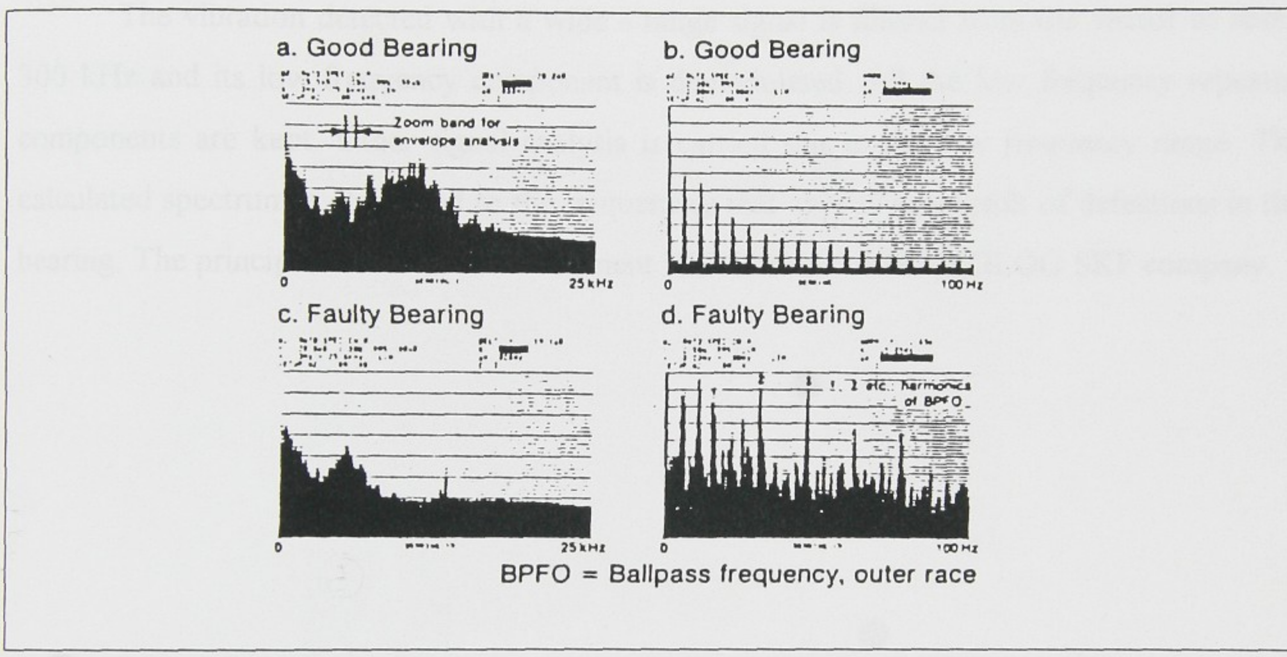


Figure 31. An example of envelope analysis

The Figure 31 shows an example where envelope analysis is far superior to normal spectrum and cepstrum analysis. It is taken from a paper mill with shaft speed 1,4 Hz, but where the resonances excited by the bearing fault are around 5,4 kHz. To complicate the issue, the bearings are lubricated by pneumatic lubricators with a pulse frequency 5,4 - 6,4 Hz. This dominates the major part of the normal spectrum, in particular at Bearing 1 which is in good condition. Even so, an outer race fault is detectable in the envelope analysis for Bearing 19, even in the presence of 2 lubricator frequencies.

Advantages:

- Repetition frequency indicated directly,

- Envelope insensitive to small speed fluctuations, and less sensitive to small variations between the individual impulses, even for high frequency bursts (since the carrier frequency information disappears).

5.9. 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.

6. EVALUTION 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 Fig.32. 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.

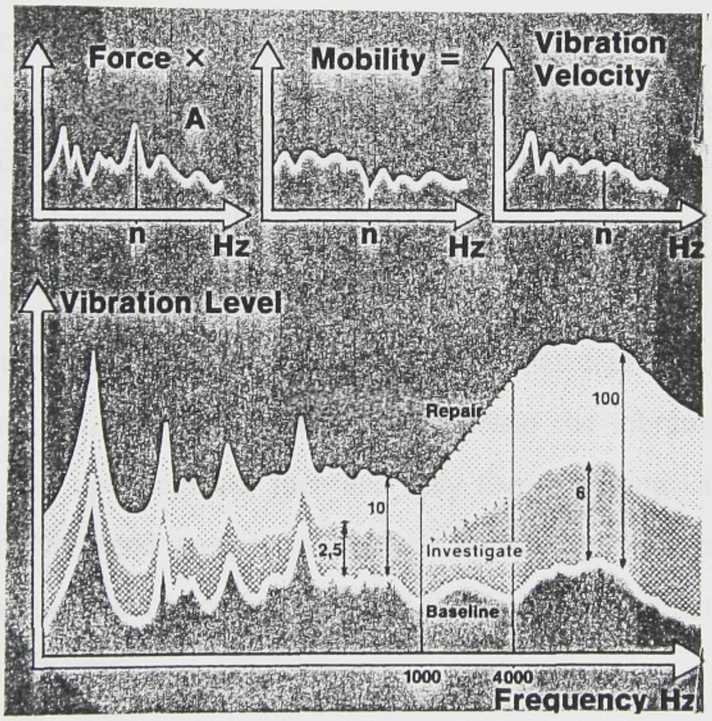


Fig.32 Evaluation of vibration levels

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 concetrating on relative changes, by specifying a reference "baseline" spectrum or level and allowing certain fixed factor changes.

Practice has shown that for frequency components up to 1000 Hz an increase by a factor of 2,5 times, or 8 dB should be considered a significant change in condition warranting investigation, and an increase by a factor of 10 times, or 20 dB from the reference condition signifies the need for repair as suggested by the ISO norms and other criteria. For frequency components above 4000 Hz these factors can be cautiously increased to 6 times, or 18 dB and 100 times, or 40 dB as shown in Fig. 32.

6.1 TRENDING FAULT DEVELOPMENTS

The trending technique is frequently used in monitoring, quite often simply to monitor a development such as oil pressure or cooling water temperature. In the case of vibration monitoring, however, it is a little more complicated. 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.

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 8dB, 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 experiance is built up.

7. MEASURING OF SMOKE VENTILATORS

7.1 USE OF FFT ANALYSERS

Most of today's portable data loggers and machines analyzers 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 analyzer.

However, problems with FFT's which are:

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.

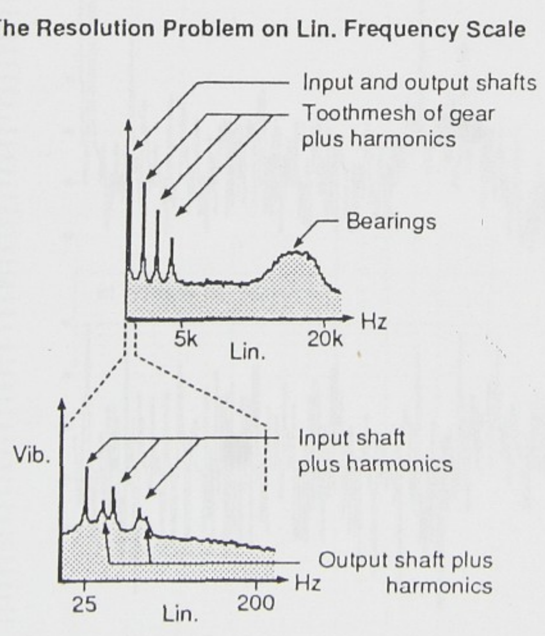


Fig. 33

The example shows (Fig. 33) that in order to be able to show the high frequency bearing components at the same time as the low frequency input and output shafts. The FFT can no longer discriminate between the input and the output shafts, since the resolution is poor due to selecting a high baseband frequency.

If a lower baseband frequency is chosen, this problem can be solved. However, a new problem of "information overKill" arises since at least two spectra will need to be stored and managed for this point. Like wise, if a larger number of points are taken in the transform the resolution may improve at low frequency for high baseband frequencies.

Very small changes in speed will shift the position of the peaks and result in large differences and give exceedance, and therefore false warnings see Fig. 34.

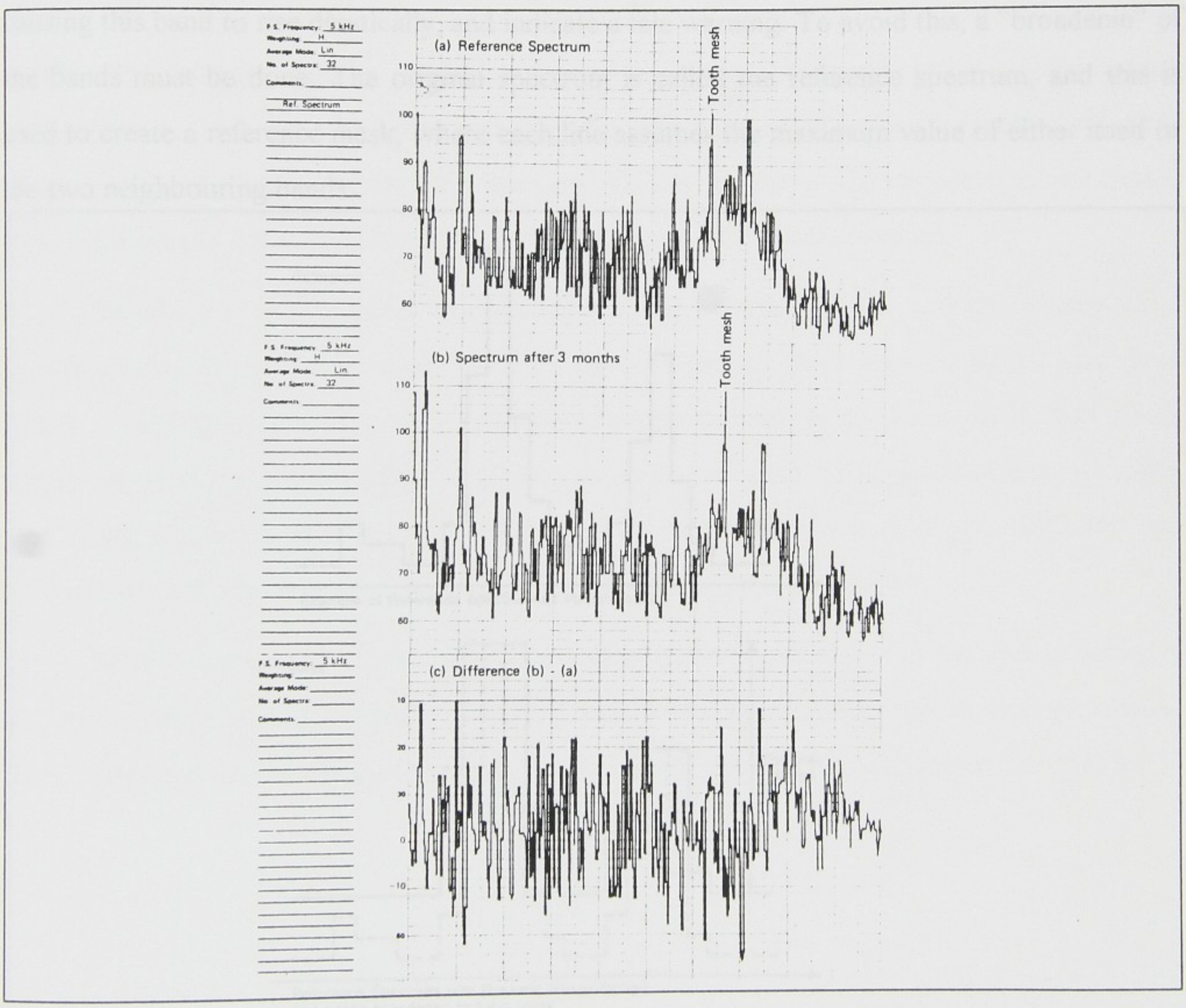


Fig. 34 Comparison of high resolution spectra give large differences, making them unsuitable for early fault detection.

The solution is to perform a kind of data reduction, 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.

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 to detect small changes in the major component of the spectrum.

Next time fault detection was carried out by spectrum comparison and if 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 "broadenin" 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.

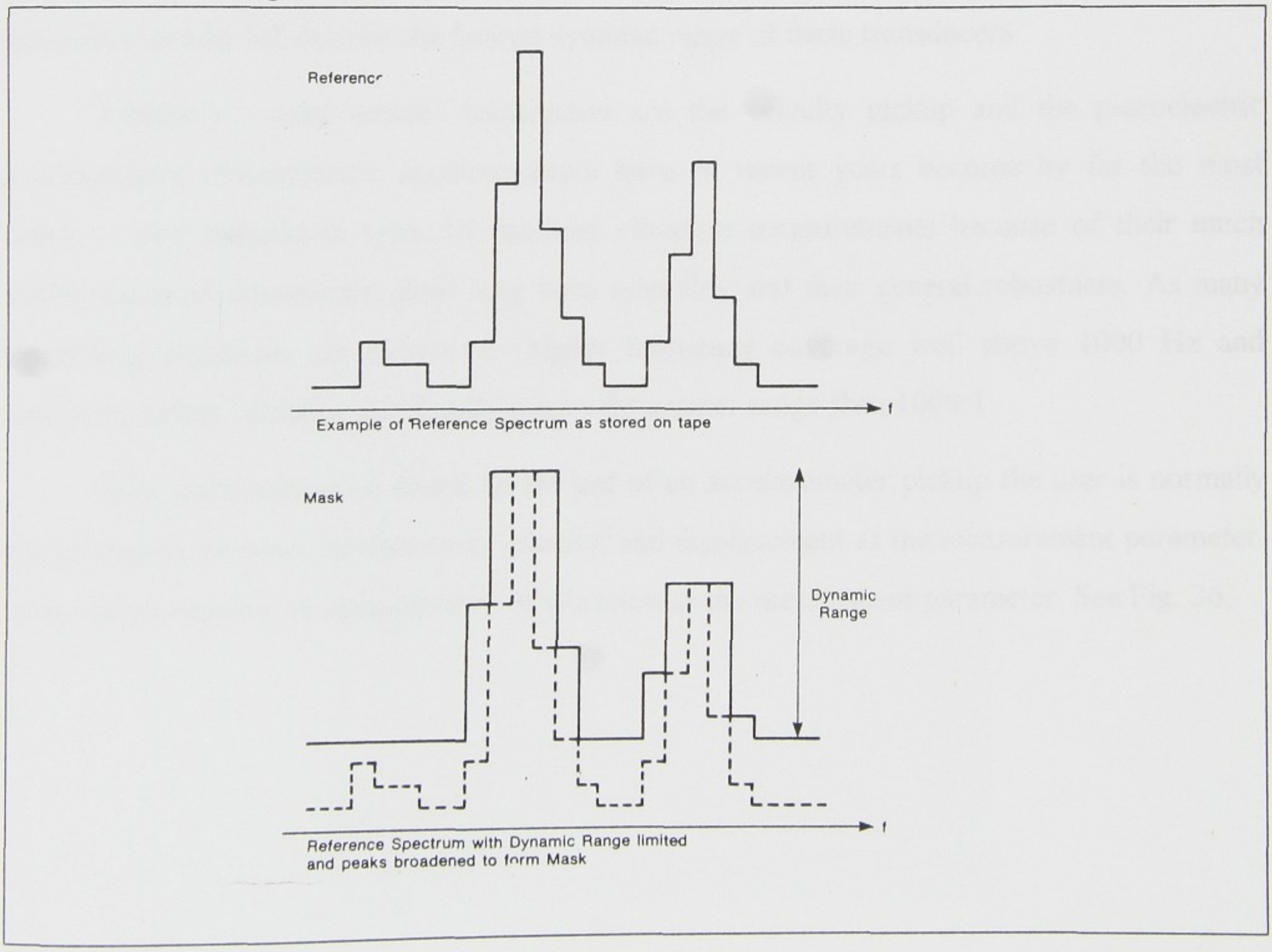


Fig. 35 Reference spectrum and the spectrum with mask

Fig. 35 shows the use of a lower limit to the dynamic range. 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, if any levels in the later measurements exceed the mask they may be considered as an indication of a developing fault.

7.2 VIBRATION TRANSDUCERS AND MEASUREMENT PARAMETERS

While relative displacement transducers are best for some specific shaft monitoring applications, seismic transducers which measure absolute vibration have proved to be far more suitable for general machine condition monitoring duties. Although relative displacement transducers, such as eddy current or proximity probes, have a frequency range that can extend up to 10000 Hz they can only effectively detect low frequency components as the higher harmonics usually fall outside the limited dynamic range of these transducers.

Popularly - used seismic transducers are the velocity pickup and the piezoelectric accelerometer. Piezoelectric accelerometers 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. As many monitoring situations are shown to require frequency coverage well above 1000 Hz and require to defect vibration amplitudes over a far greater range than 1000:1.

With instrumentation based on the use of an accelerometer pickup the user is normally free to choose between acceleration, velocity, and displacement as the measurement parameter. In our measurments we used velocity of vibration as the measurment parameter. See Fig. 36.

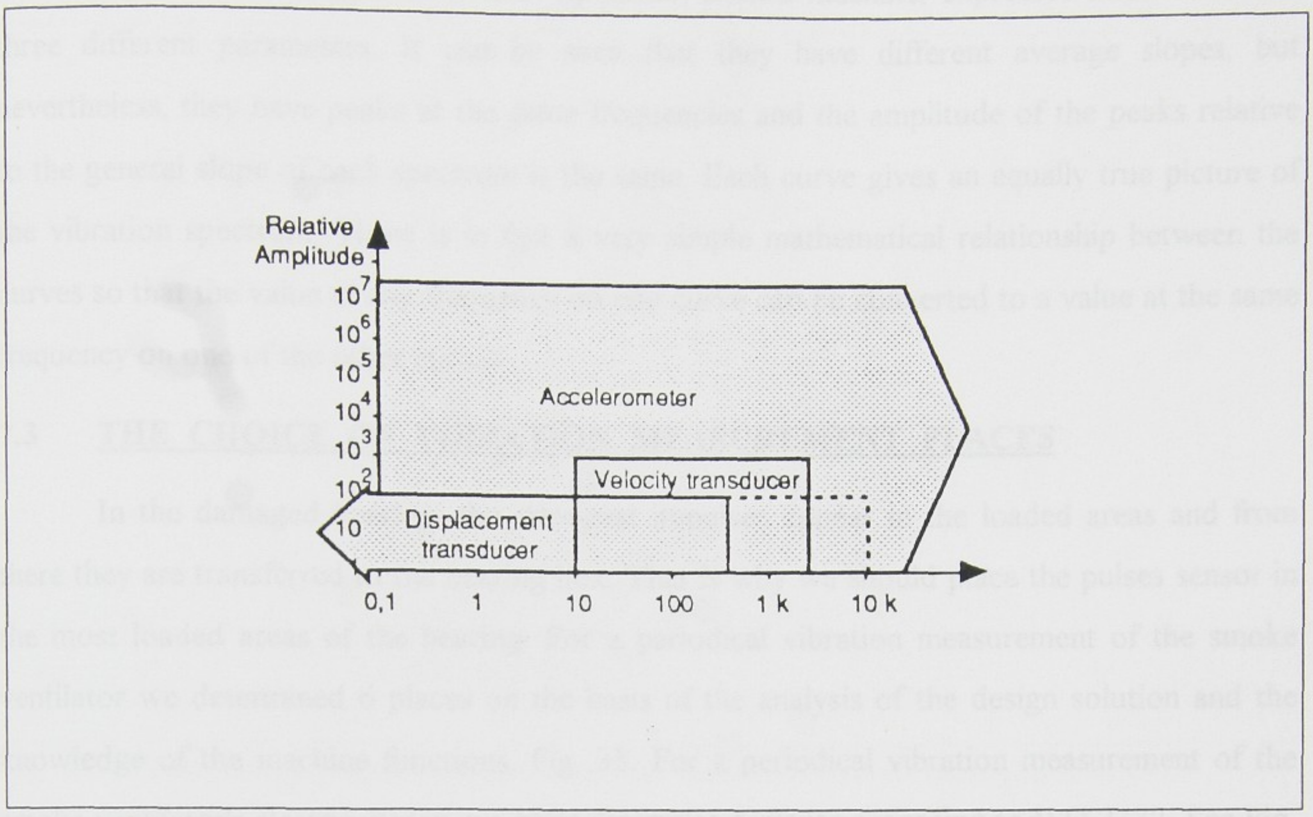


Fig. 36 Vibration transducers

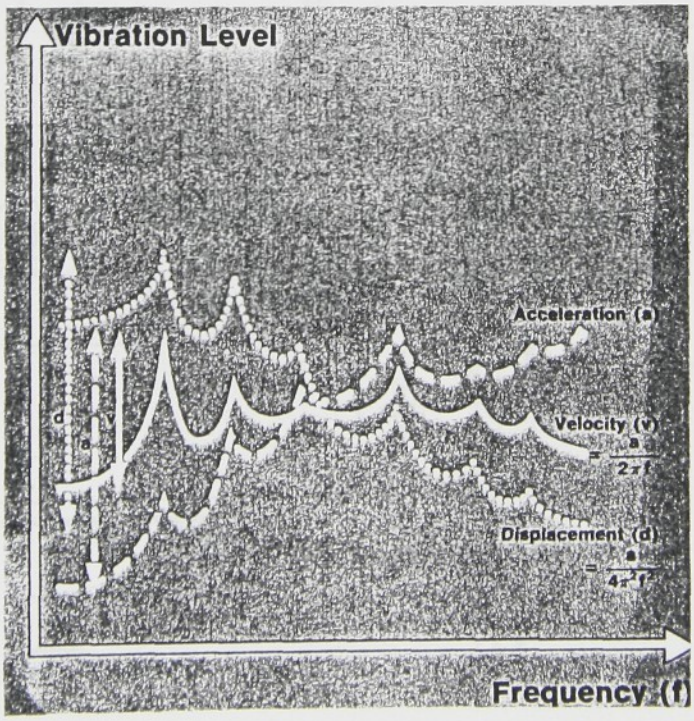


Fig. 37 Measurement parameters

Fig. 37 shows atypical vibration spectrum from a machine, expressed in terms of the three different parameters. It can be seen that they have different average slopes, but nevertheless, they have peaks at the same frequencies and the amplitude of the peaks relative to the general slope of each spectrum is the same. Each curve gives an equally true picture of the vibration spectrum. There is in fact a very simple mathematical relationship between the curves so that the value at any frequency on one curve can be converted to a value at the same frequency on one of the other curves.

7.3 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 smoke ventilator we determined 6 places on the basis of the analysis of the design solution and the knowledge of the machine functions. Fig. 38. For a periodical vibration measurement of the smoke ventilator's electric motor, we must determine 6 places according to ISO 2372. See Fig. 39.

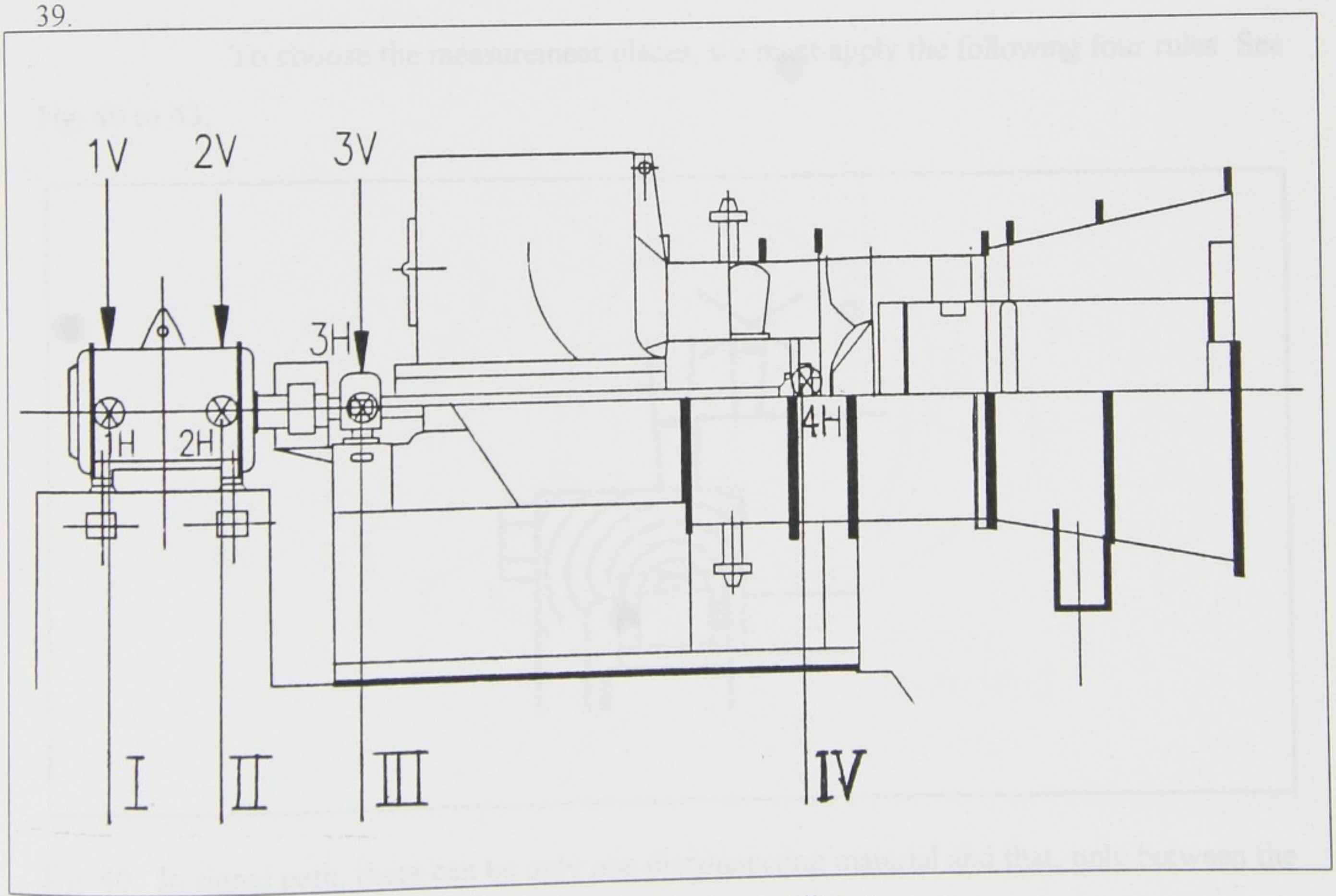


Fig. 38 Designation of measurement places in a smoke ventilator.

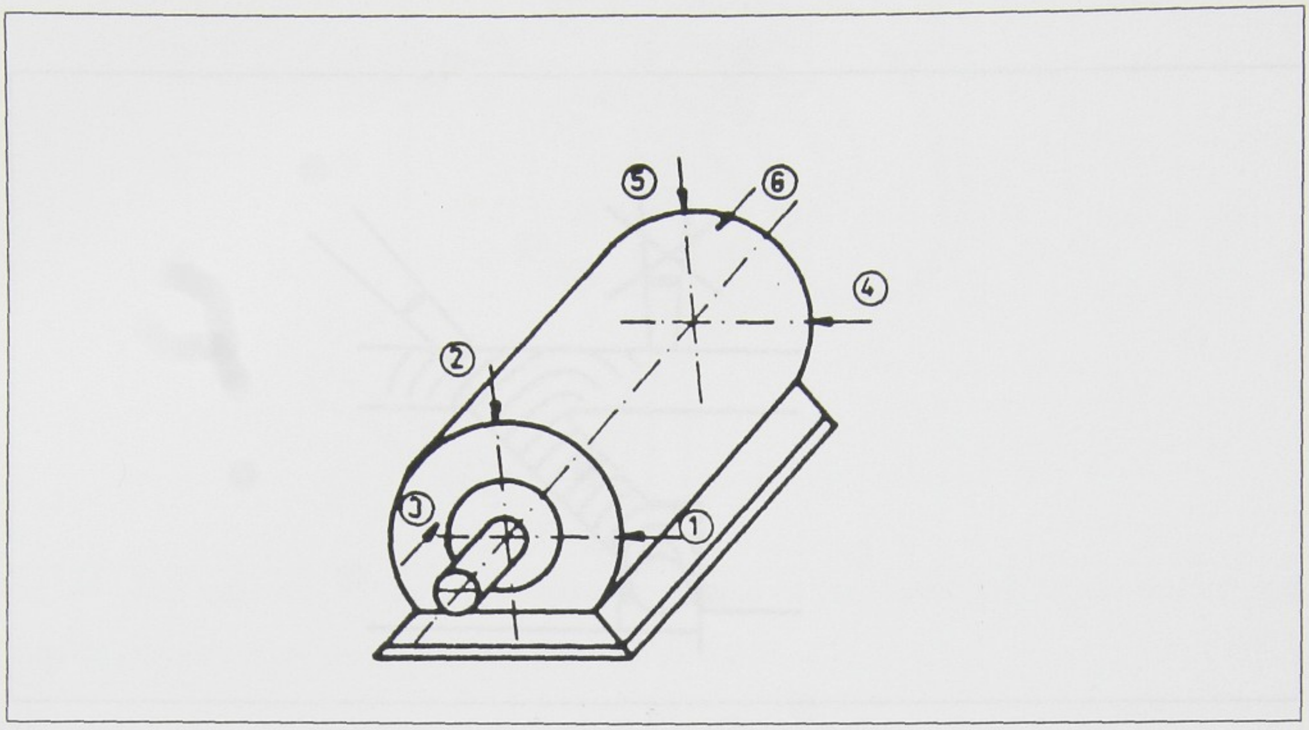


Fig. 39 Measurement places in smoke ventilator electric motor according to ISO 2372.

To choose the measurement places, we must apply the following four rules. See Fig. 40 to 43.

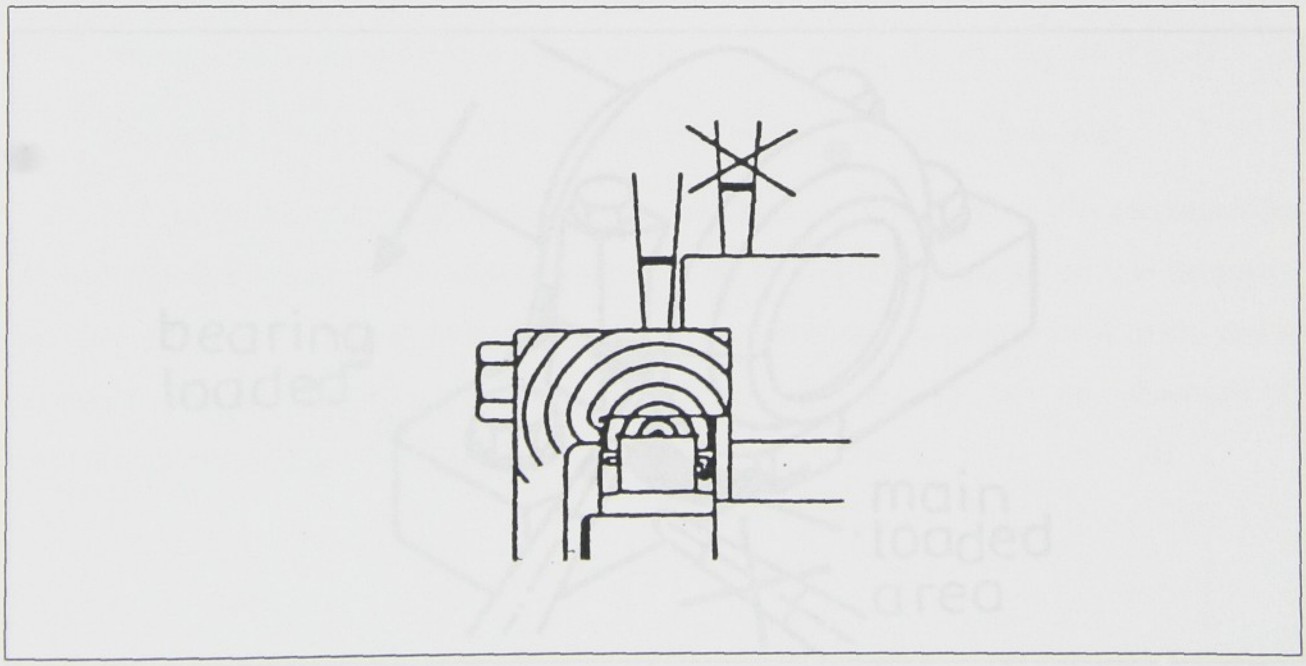


Fig. 40 In signal path, there can be only one disconnecting material and that, only between the bearing and the bearing box.

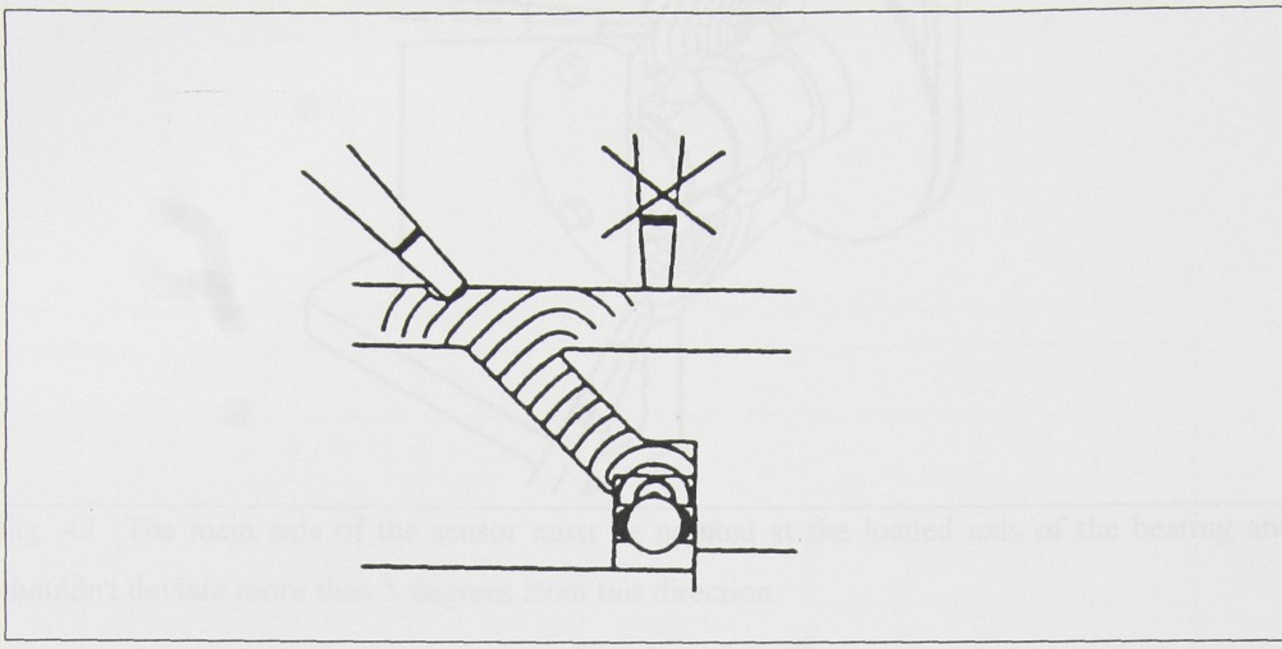


Fig. 41 The direction of the signal path between the bearing and the measurement place must be straight and through. The distance between the point of measurement and the bearing shouldn't be bigger than 75 mm.

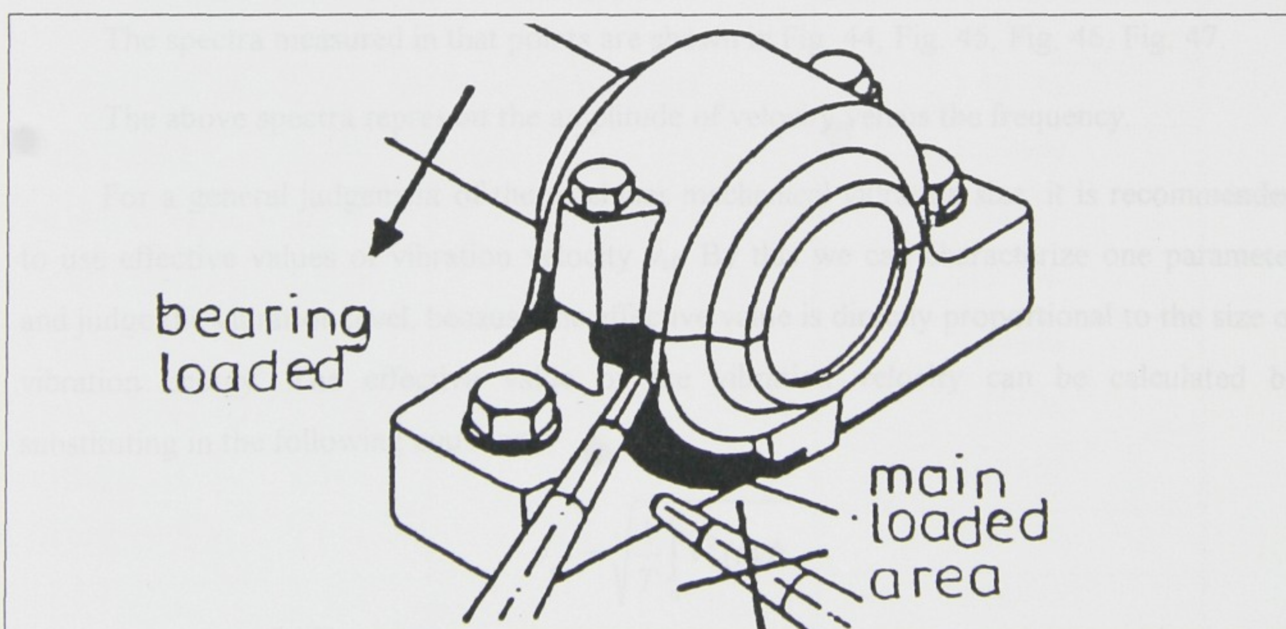


Fig. 42 The measurement place must lay in the loaded area of the bearing.

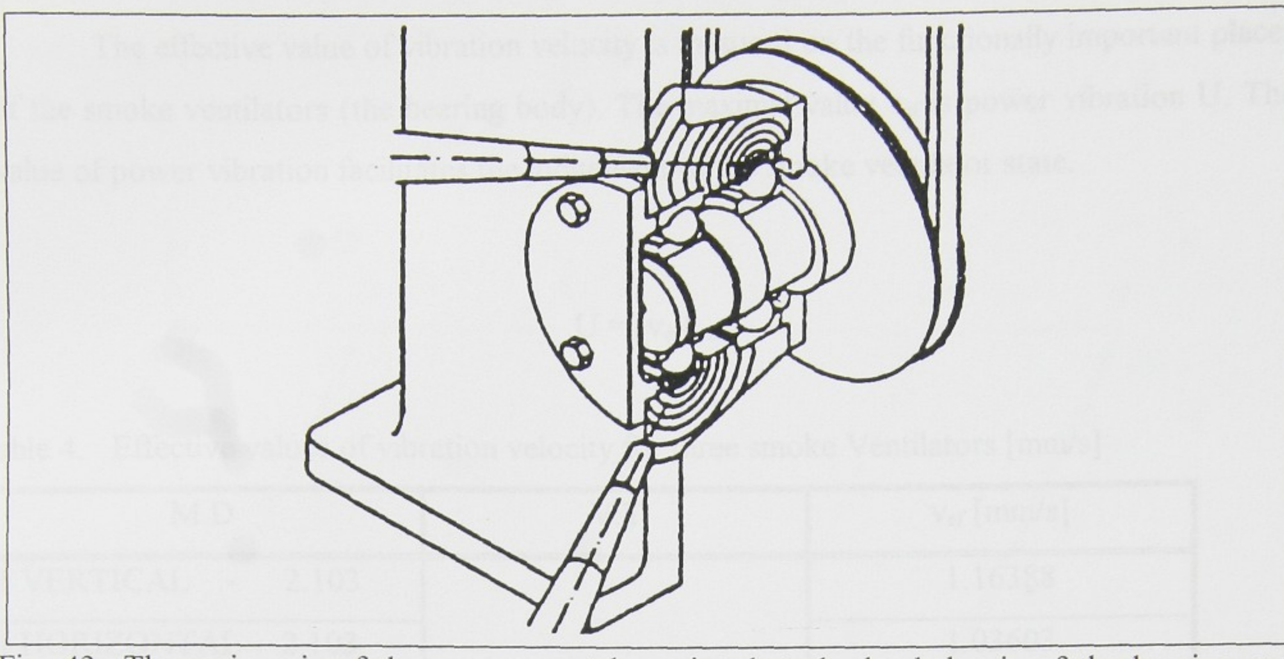


Fig. 43 The main axis of the sensor must be pointed at the loaded axis of the bearing and shouldn't deviate more than 5 degrees from this direction.

7.4 MEASUREMENT RESULTS

The measurement has been done by using the diagnostic device (the Microlog data collector) on four different planes (each plane has two different points) in the vertical and horizontal direction. See Fig. 38.

The spectra measured in that points are shown in Fig. 44, Fig. 45, Fig. 46, Fig. 47.

The above spectra represent the amplitude of velocity versus the frequency.

For a general judgement of the machines mechanical vibration size, it is recommended to use effective values of vibration velocity v_{ef}. By this we can characterize one parameter and judge the vibration level, because the effective value is directly proportional to the size of vibration energy. The effective value of the vibration velocity can be calculated by substituting in the following equation.

$$v_{ef} = \sqrt{\frac{1}{T}\int_0^T v^2(t)dt}$$

The results of effective values of vibration velocity v_{ef} for the smoke ventilators, when the rotational speed was 980 RPM are shown in Tab. 4.

The effective value of vibration velocity is mesured on the functionally important places of the smoke ventilators (the bearing body). The maximal value v_{ef} is power vibration U. The value of power vibration facilitates the judgement of the smoke ventilator state.

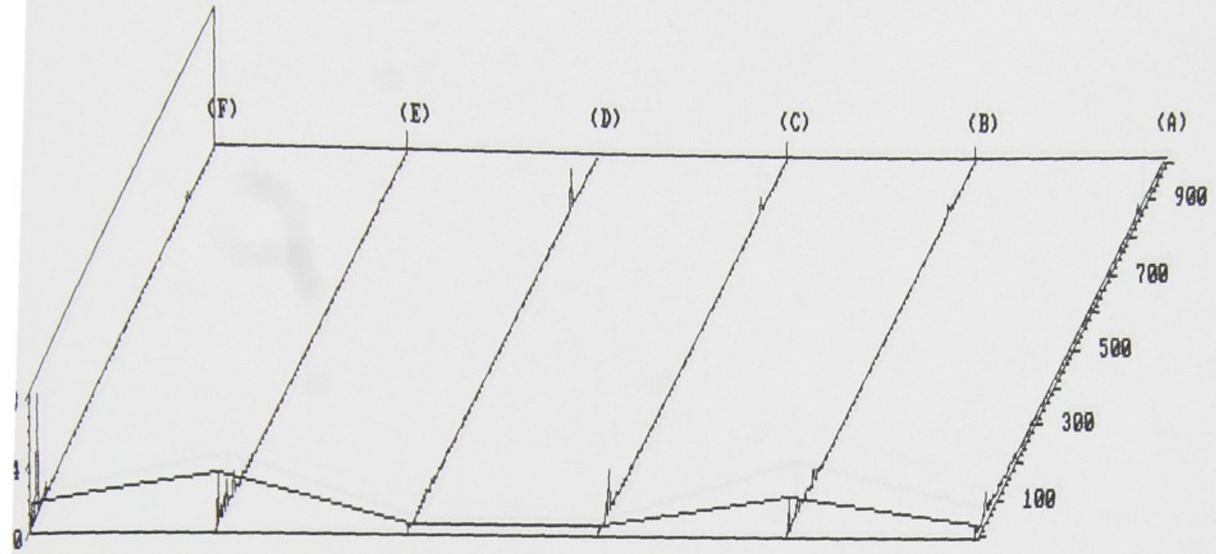
$$U = (v_{ef})_{max}.$$

Table 4. Effective values of vibration velocity for three smoke Ventilators [mm/s]

M.D	V.T	v_{ef} [mm/s]
1 VERTICAL - 2.103	KV - 2.103	1.16388
1 HORIZONTAL - 2.103		1.03603
2 VERTICAL - 2.103		0.825125
2 HORIZONTAL - 2.103		1.29609
3 VERTICAL - 2.103		1.02806
3 HORIZONTAL - 2.103		1.48804
4 HORIZONTAL - 2.103		1.02368
1 VERTICAL - 2.203	KV - 2.203	1.55769
1 HORIZONTAL - 2.203		1.41786
2 VERTICAL - 2.203		1.79649
2 HORIZONTAL - 2.203		1.67188
3 VERTICAL - 2.203		3.25279
3 HORIZONTAL - 2.203		3.52052
4 HORIZONTAL - 2.203		2.69458
1 VERTICAL - 2.104	KV - 2.204	2.64846
1 HORIZONTAL - 2.104		1.25536
2 VERTICAL - 2.104		2.63852
2 HORIZONTAL - 2.104		1.92866
3 VERTICAL - 2.104		1.16413
3 HORIZONTAL - 2.104		1.60070
4 HORIZONTAL - 2.104		1.99741

PALOGRAM SPECTRAL PLOT

FREQ: 16.25



F)0.4885 (E)1.0752 (D)0.1026 (C)0.056 (B)0.677
A)0.1623

) SET: KV-2.103
ID: 1HORIZONTAL-2.103
DATE: 06-MAY-96 09:35:26
FREQ. RNG: 0 - 1000 Hz

DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 0.5 mm/sec
FFT LINES: 800 SPEED: 16.67

) SET: KV-2.203
ID: 1HORIZONTAL-2.203
DATE: 06-MAY-96 09:47:54
FREQ. RNG: 0 - 1000 Hz

DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 5 mm/sec
FFT LINES: 800 SPEED: 16.67

) SET: KV-2.204
ID: 1HORIZONTAL-2.204
DATE: 06-MAY-96 09:42:52
FREQ. RNG: 0 - 1000 Hz

DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 1 mm/sec
FFT LINES: 800 SPEED: 16.67

)) SET: KV-2.103
ID: 1VERTICAL-2.103
DATE: 06-MAY-96 09:37:14
FREQ. RNG: 0 - 1000 Hz

DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 1 mm/sec
FFT LINES: 800 SPEED: 16.67

E) SET: KV-2.203
ID: IVERTICAL-2.203
DATE: 06-MAY-96 09:48:28
FREQ. RNG: 0 - 1000 Hz

DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 2 mm/sec
FFT LINES: 800 SPEED: 16.67

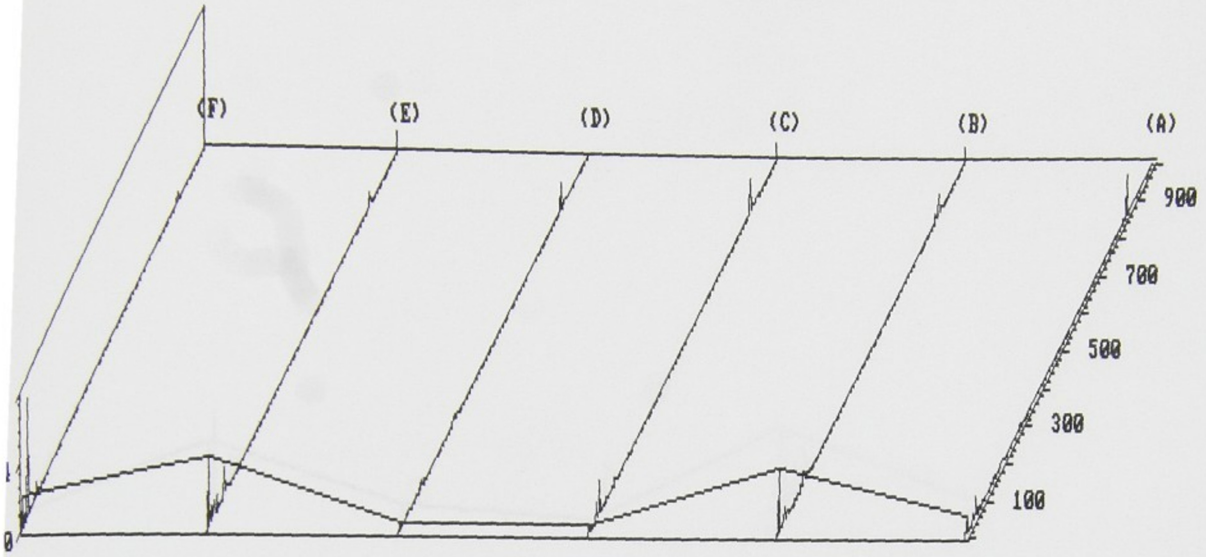
F) SET: KV-2.204
ID: 1VERTICAL-2.204
DATE: 06-MAY-96 09:43:52
FREQ. RNG: 0 - 1000 Hz

DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 5 mm/sec
FFT LINES: 800 SPEED: 16.67

Fig. 44 PALOGRAM SPECTRAL PLOT for three smoke ventilators (2.103), (2.203), (2.204) in vertical and horizontal directions in plane I.

PALOGRAM SPECTRAL PLOT

FREQ: 16.25



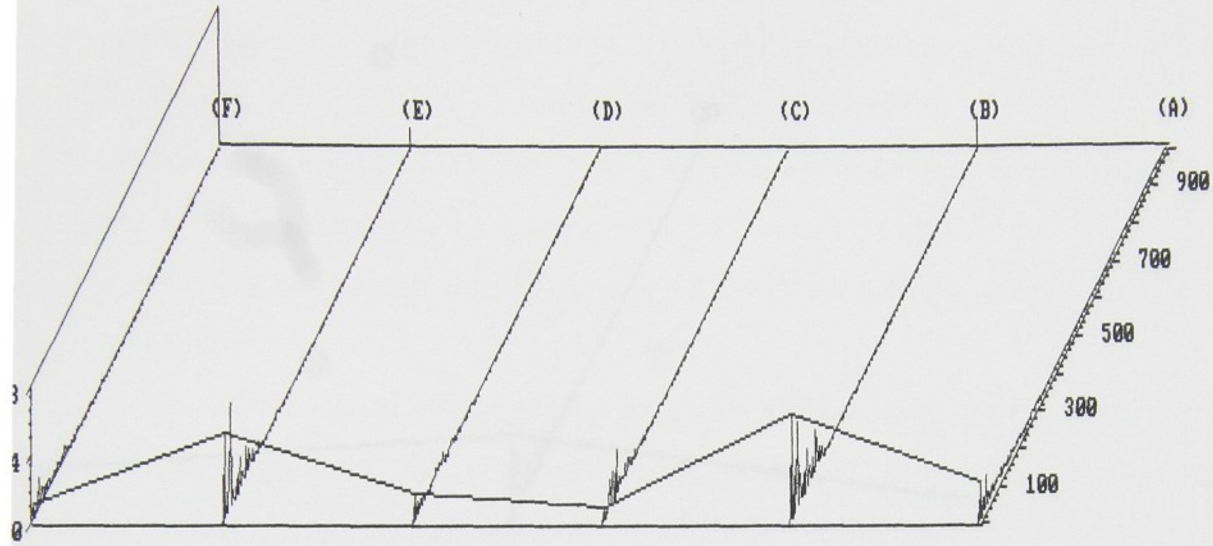
(F)0.6623 (E)1.4086 (D)0.1642 (C)0.1511 (B)1.3302
(A)0.3382

) SET: KV-2.103	DESC: diploma work
ID: 2HORIZONTAL-2.103	DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:37:52	FULL SCALE: 1 mm/sec
FREQ. RNG: 0 - 1000 Hz	FFT LINES: 800 SPEED: 16.67
) SET: KV-2.203	DESC: diploma work
ID: 2HORIZONTAL-2.203	DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:49:04	FULL SCALE: 2 mm/sec
FREQ. RNG: 0 - 1000 Hz	FFT LINES: 800 SPEED: 16.67
) SET: KV-2.204	DESC: diploma work
ID: 2HORIZONTAL-2.204	DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:44:28	FULL SCALE: 5 mm/sec
FREQ. RNG: 0 - 1000 Hz	FFT LINES: 800 SPEED: 16.67
) SET: KV-2.103	DESC: diploma work
ID: 2VERTICAL-2.103	DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:38:30	FULL SCALE: 1 mm/sec
FREQ. RNG: 0 - 1000 Hz	FFT LINES: 800 SPEED: 16.67
E) SET: KV-2.203	DESC: diploma work
ID: 2VERTICAL-2.203	DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:49:46	FULL SCALE: 2 mm/sec
FREQ. RNG: 0 - 1000 Hz	FFT LINES: 800 SPEED: 16.67
F) SET: KV-2.204	DESC: diploma work
ID: 2VERTICAL-2.204	DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:45:06	FULL SCALE: 5 mm/sec
FREQ. RNG: 0 - 1000 Hz	FFT LINES: 800 SPEED: 16.67

Fig. 45 PALOGRAM SPECTRAL PLOT for three smoke ventilators (2.103), (2.203), (2.204) in vertical and horizontal directions in plane II.

PALOGRAM SPECTRAL PLOT

FREQ: 16.25



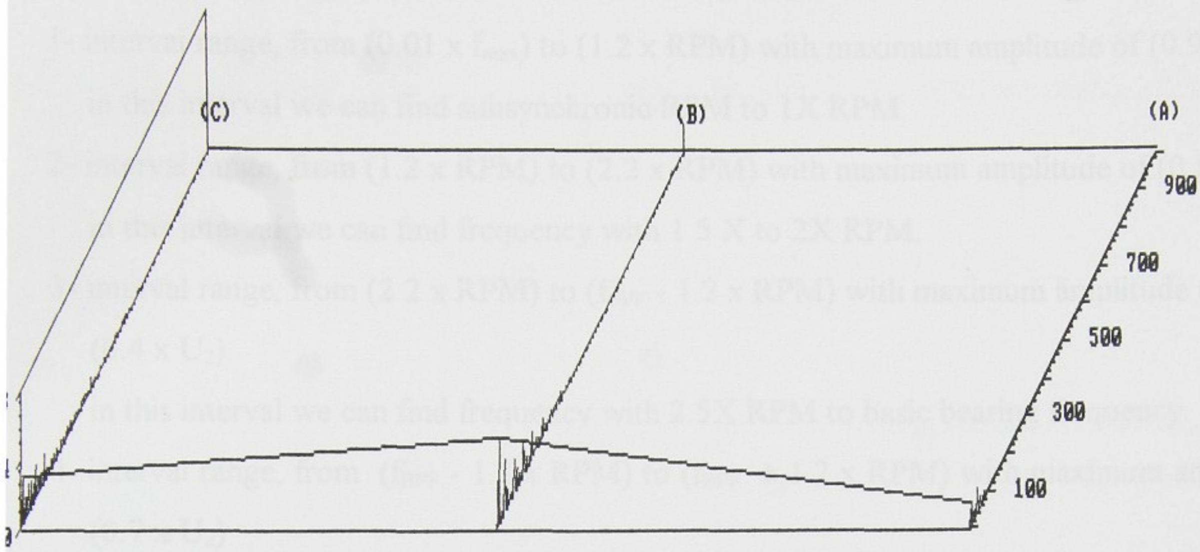
F)0.3469 (E)1.7896 (D)0.5265 (C)0.2336 (B)2.1284
A)0.7742

```
SET: KV-2.103                     DESC: diploma work
ID: 3HORIZONTAL-2.103             DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:39:08          FULL SCALE: 1 mm/sec
FREQ. RNG: 0 - 1000 Hz            FFT LINES: 800    SPEED: 16.67

) SET: KV-2.203                   DESC: diploma work
ID: 3HORIZONTAL-2.203             DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:50:24          FULL SCALE: 5 mm/sec
FREQ. RNG: 0 - 1000 Hz            FFT LINES: 800    SPEED: 16.67

) SET: KV-2.204                   DESC: diploma work
ID: 3HORIZONTAL-2.204             DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:45:42          FULL SCALE: 1 mm/sec
FREQ. RNG: 0 - 1000 Hz            FFT LINES: 800    SPEED: 16.67

) SET: KV-2.103                   DESC: diploma work
ID: 3VERTICAL-2.103               DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:39:42          FULL SCALE: 1 mm/sec
FREQ. RNG: 0 - 1000 Hz            FFT LINES: 800    SPEED: 16.67

) SET: KV-2.203                   DESC: diploma work
ID: 3VERTICAL-2.203               DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:51:00          FULL SCALE: 5 mm/sec
FREQ. RNG: 0 - 1000 Hz            FFT LINES: 800    SPEED: 16.67

) SET: KV-2.204                   DESC: diploma work
ID: 3VERTIKAL-2.204               DESC: smoke ventilator-Liberec
DATE: 06-MAY-96 09:46:24          FULL SCALE: 1 mm/sec
FREQ. RNG: 0 - 1000 Hz            FFT LINES: 800    SPEED: 16.67
```

Fig. 46 PALOGRAM SPECTRAL PLOT for three smoke ventilators (2.103), (2.203), (2.204) in vertical and horizontal directions in plane III.

PALOGRAM SPECTRAL PLOT

FREQ: 16.25



C)0.9913 (B)1.7473 (A)0.4371

SET: KV-2.103
ID: 4HORIZONTAL-2.103
DATE: 06-MAY-96 09:40:32
FREQ. RNG: 0 - 1000 Hz
DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 1 mm/sec
FFT LINES: 800 SPEED: 16.67

SET: KV-2.203
ID: 4HORIZONTAL-2.203
DATE: 06-MAY-96 09:51:46
FREQ. RNG: 0 - 1000 Hz
DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 2 mm/sec
FFT LINES: 800 SPEED: 16.67

SET: KV-2.204
ID: 4HORIZONTAL-2.204
DATE: 06-MAY-96 09:47:02
FREQ. RNG: 0 - 1000 Hz
DESC: diploma work
DESC: smoke ventilator-Liberec
FULL SCALE: 1 mm/sec
FFT LINES: 800 SPEED: 16.67

Fig. 47 PALOGRAM SPECTRAL PLOT for three smoke ventilators (2.103), (2.203), (2.204) in horizontal directions in plane IV.

To determine the increases and changes of amplitude in the spectrum to detect machine condition.

The frequency scale is divided into six intervals / 9/ as the following:

1- interval range, from ($0.01 \times f_{max}$) to ($1.2 \times RPM$) with maximum amplitude of ($0.9 \times U_2$)
in this interval we can find subsynchronic RPM to 1X RPM.

2- interval range, from ($1.2 \times RPM$) to ($2.2 \times RPM$) with maximum amplitude of ($0.5 \times U_2$)
in this interval we can find frequency with 1.5 X to 2X RPM.

3- interval range, from ($2.2 \times RPM$) to ($f_{BPF} - 1.2 \times RPM$) with maximum amplitude of ($0.4 \times U_2$)
in this interval we can find frequency with 2.5X RPM to basic bearing frequency.

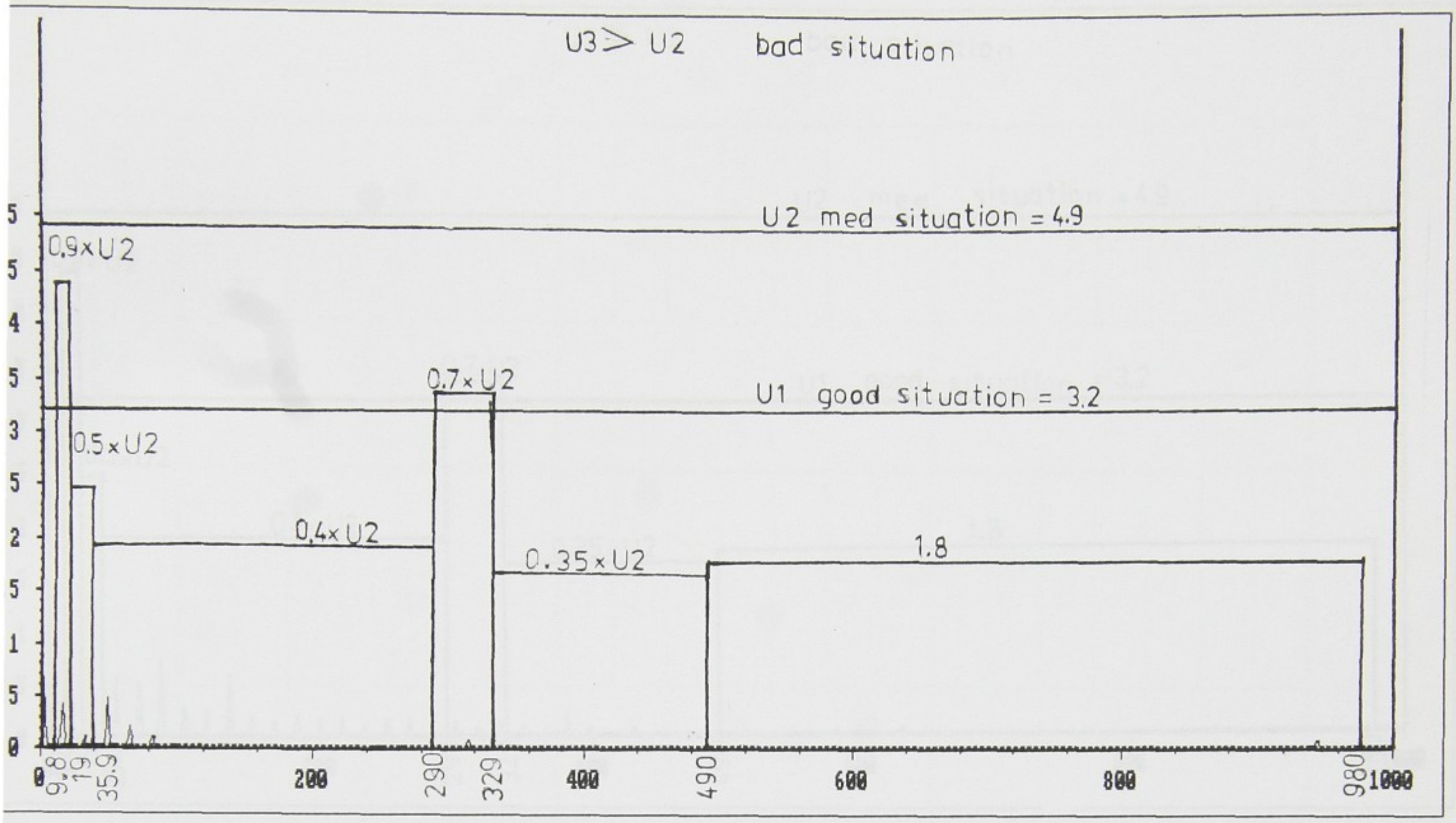
4- interval range, from ($f_{BPF} - 1.2 \times RPM$) to ($f_{BPF} + 1.2 \times RPM$) with maximum amplitude of ($0.7 \times U_2$)
in this interval we can find blade passes frequency and one time modulation.

5- interval range, from ($f_{BPF} + 1.2 \times RPM$) to ($0.5 \times f_{max}$) with maximum amplitude of ($0.35 \times U_2$)
in this interval we can find lower harmonic bearing frequency and blade passes harmonic frequency.

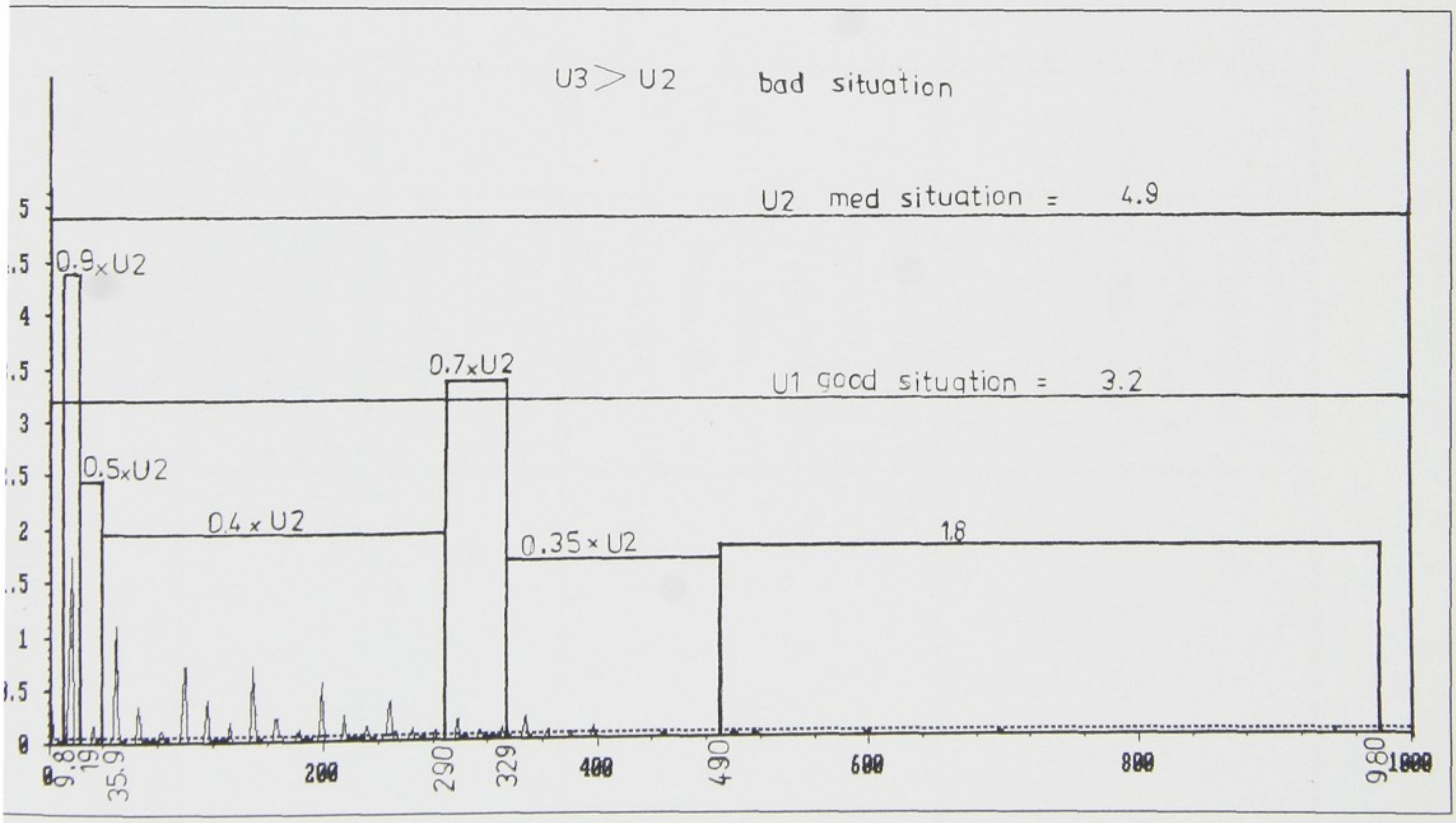
6- interval range, from ($0.5 \times f_{max}$) to (f_{max}) with maximum amplitude of (1.8)
in this interval we can find higher harmonic bearing frequency and self frequency of bearing.

$f_{max} = 60 \times 16.333 = 980$ [Hz]

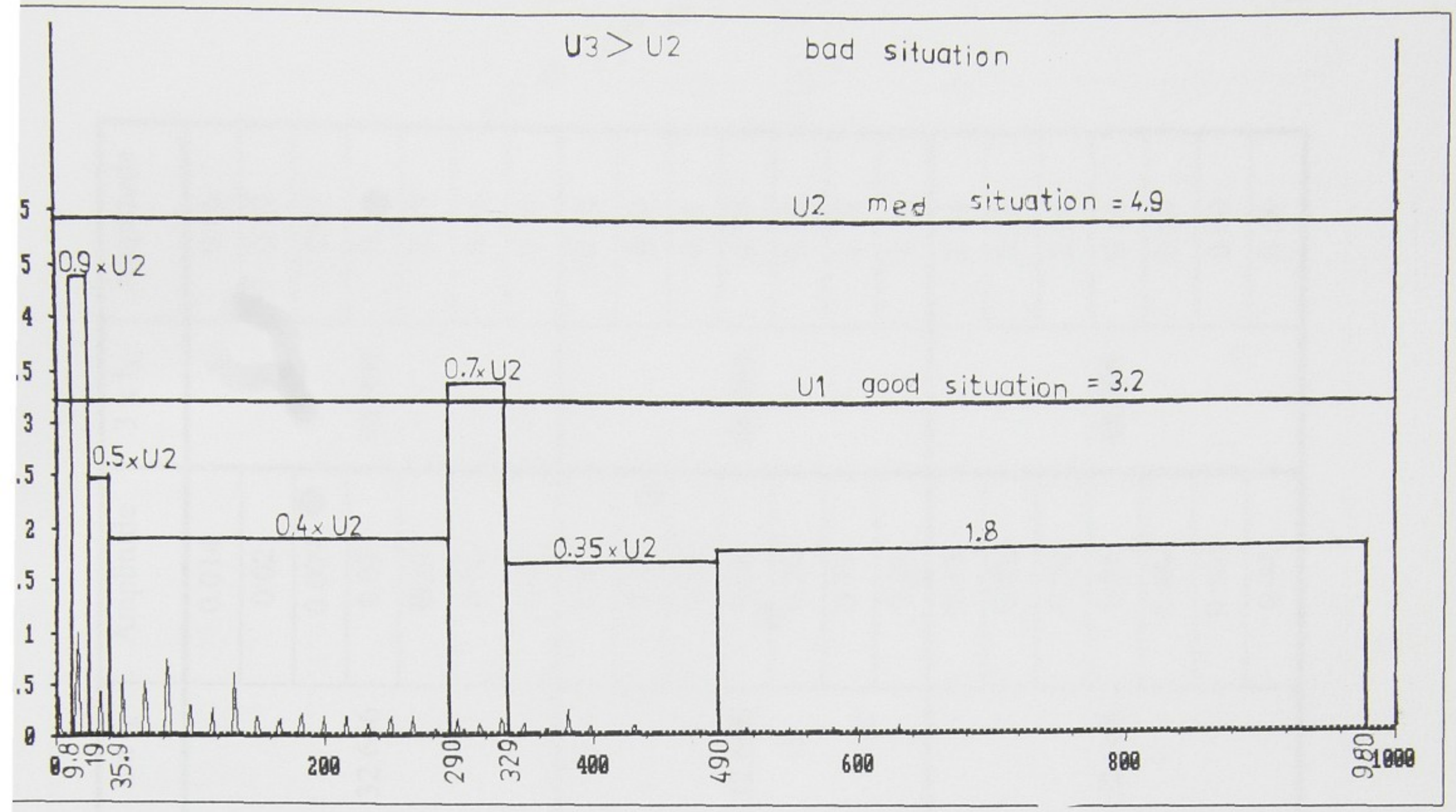
As shown in Fig. 48, Fig. 49 , Fig 50.



. 48 The limit values of velocity amplituds recommended not to be exceeded (4 HORIZONTAL - 03)



49 The limit values of velocity amplituds recommended not to be exceeded (4 HORIZONTAL - 03)



g 50 The limit values of velocity amplituds recommended not to be exceeded (4 HORIZONTAL - 204)

Table 5. of The results of Amplitude shaft frequency and it's higher harmonics

TYPE OF SMOKE VENTILATOR	Mesuring Direction	$1 \times f_R$	Amplitude	$2 \times f_R$	Amplitude	$3 \times f_R$	Amplitude
2.103	1. VERTICAL	16.333	0.10	32,666	0.014	48.999	0.06
	1. HORIZONTAL		0.16		0.02		0.12
	2. VERTICAL		0.16		0.009		0.07
	2. HORIZONTAL		0.34		0.03		0.20
	3. VERTICAL		0.52		0.23		0.22
	3. HORIZONTAL		0.77		0.30		0.66
	4. HORIZONTAL		0.43		0.13		0.50
2.203	1. VERTICAL	16.333	1.07	32.666	0.34	48.999	0.38
	1. HORIZONTAL		0.67		0.15		0.16
	2. VERTICAL		1.40		0.33		0.31
	2. HORIZONTAL		1.33		0.06		0.30
	3. VERTICAL		1.78		0.20		2.16
	3. HORIZONTAL		2.10		0.77		1.78
	4. HORIZONTAL		1.74		0.16		1.10
2.204	1. VERTICAL	16.333	0.48	32.666	0.16	48.999	2.44
	1. HORIZONTAL		0.056		0.02		0.30
	2. VERTICAL		0.66		0.20		2.38
	2. HORIZONTAL		0.15		0.04		0.34
	3. VERTICAL		0.34		0.064		0.60
	3. HORIZONTAL		0.23		0.163		0.60
	4. HORIZONTAL		0.99		0.44		0.59

7.5 MEASUREMENT RESULTS OF BEARINGS BY ENVELOPE METHOD

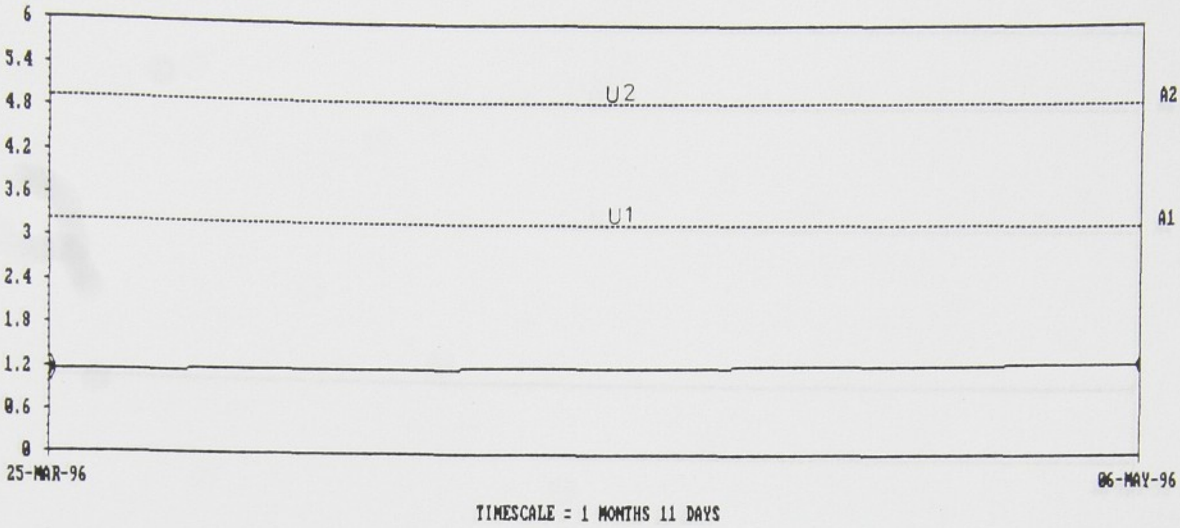
Table 6. Effective values of vibration acceleration a_{ef} for three smoke ventilators by invelope method [mm/s^2]

M.D	T.V	25 - 3 - 1996	6 - 5 - 1996	% CHANGING
3V. OUTSIDE BEARING		9.2772	9.736635	5
4H. INSIDE BEARING	KV - 2.103	7.9394	6.4573	-19
3V. OUTSIDE BEARING		11.3822	12.3838	9
4H. INSIDE BEARING	KV - 2.203	12.6959	18.6059	47
2V. BEARING		1.0123	0.786719	-22
3V. OUTSIDE BEARING		9.9877	10.0852	1
4H. INSIDE BEARING	KV - 2.204	17.4427	17.3608	0
2V. BEARING		0.80203	0.862619	8

OVERALL VALUE VERSUS TIME TREND

SET ID: KV-2.203
POINT ID: 3V.OUT.BEARING 2.203
ALARM TYPE: LEVEL
ALARM1: 3.2 ALARM2: 4.9
DATE: 25-MAR-96 09:28:24

SET DESC:diploma work
POINT DESC: smoke ventilator-Envelope method
UNITS: Gs Env
STD: 0.0708284 MEAN:1.1883
VALUE: 1.138



OVERALL TREND VALUES

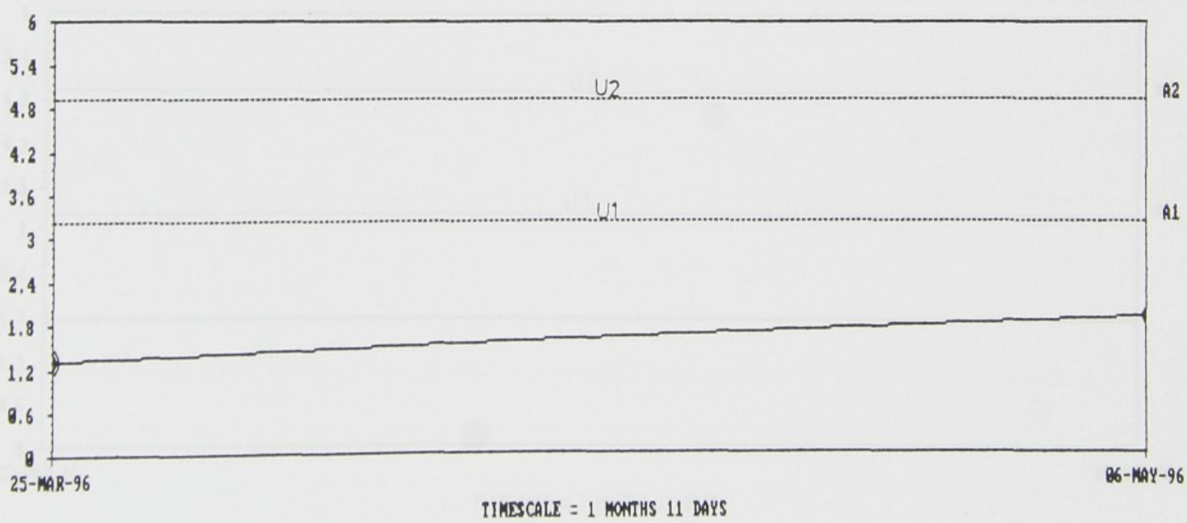
NO.	DATE	TIME	VALUE	INCL	NO.	DATE	TIME	VALUE	INCL
1.	25-MAR-96	09:28:24	1.138	Y	2.	06-MAY-96	09:21:14	1.238	Y

SKF 20-MAY-96 STANDARD DATABASE Page 1

OVERALL VALUE VERSUS TIME TREND

SET ID: KV-2.203
POINT ID: 4H.IN.BEARING-2.203
ALARM TYPE: LEVEL
ALARM1: 3.2 ALARM2: 4.9
DATE: 25-MAR-96 09:30:58

SET DESC:diploma work
POINT DESC: smoke ventilator-Envelope method
UNITS: Gs Env
STD: 0.417899 MEAN:1.56509
VALUE: 1.269



OVERALL TREND VALUES

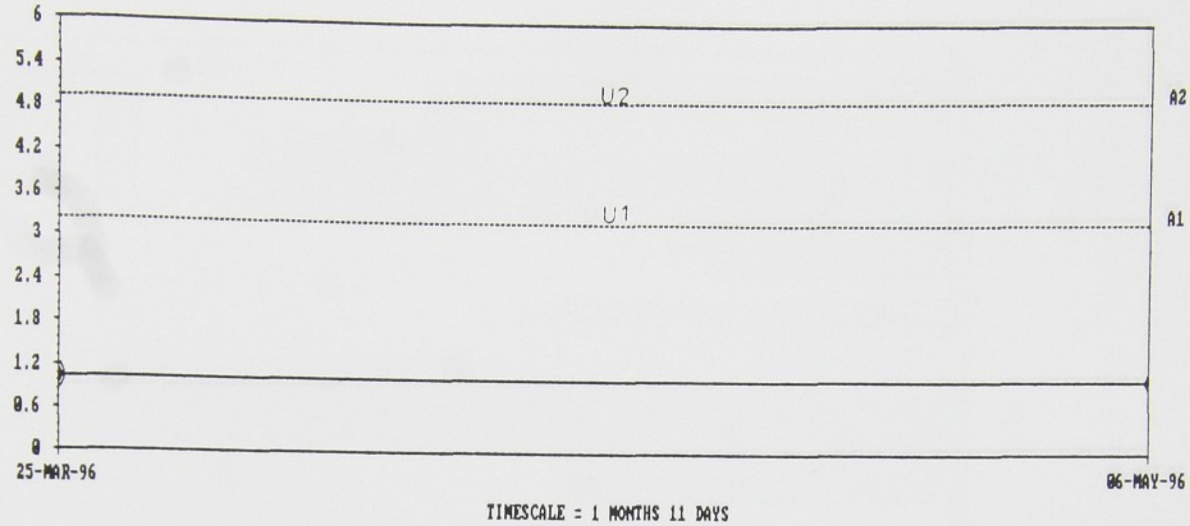
NO.	DATE	TIME	VALUE	INCL	NO.	DATE	TIME	VALUE	INCL
1.	25-MAR-96	09:30:58	1.27	Y	2.	06-MAY-96	09:23:00	1.861	Y

Fig. 51 The overall value versus time trend for smoke ventilator 2.203, "Up" trend is for outside bearing, the effective value of vibration acceleration increase from 1.138 to 1.238 [$mm.s^{-2}$].

„Down" trend is for inside bearing, the effective value of vibration acceleration increase from 1.27 to 1.861 [$mm . s^{-2}$].

OVERALL VALUE VERSUS TIME TREND

SET ID: KV-2.204 SET DESC:diploma work
POINT ID: 3V.OUT.BEARING-2.204 POINT DESC: smoke ventilator-Envelope method
ALARM TYPE: LEVEL UNITS: Gs Env
ALARM1: 3.2 ALARM2: 4.9 STD: 0.0068937 MEAN:1.00365
DATE: 25-MAR-96 09:25:10 VALUE: 0.998



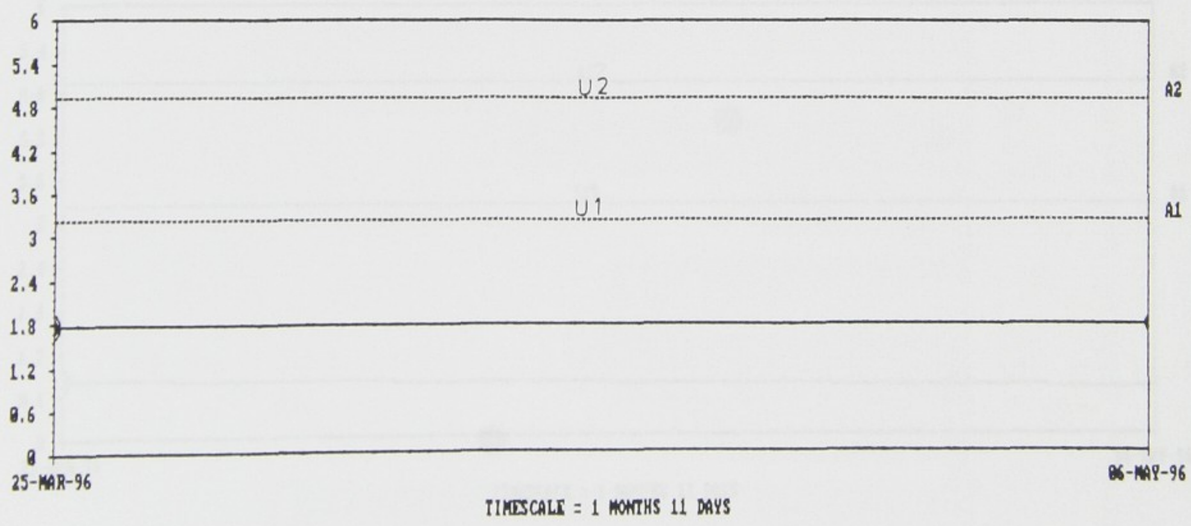
OVERALL TREND VALUES

NO.	DATE	TIME	VALUE	INCL	NO.	DATE	TIME	VALUE	INCL
1.	25-MAR-96	09:25:10	0.9988	Y	2.	06-MAY-96	09:08:08	1.009	Y

SKF 20-MAY-96 STANDARD DATABASE Page 1

OVERALL VALUE VERSUS TIME TREND

SET ID: KV-2.204 SET DESC:diploma work
POINT ID: 4H.INS.BEARING-2.204 POINT DESC: smoke ventilator-Envelope method
ALARM TYPE: LEVEL UNITS: Gs Env
ALARM1: 3.2 ALARM2: 4.9 STD: 0.0057972 MEAN:1.74018
DATE: 25-MAR-96 09:27:08 VALUE: 1.744



OVERALL TREND VALUES

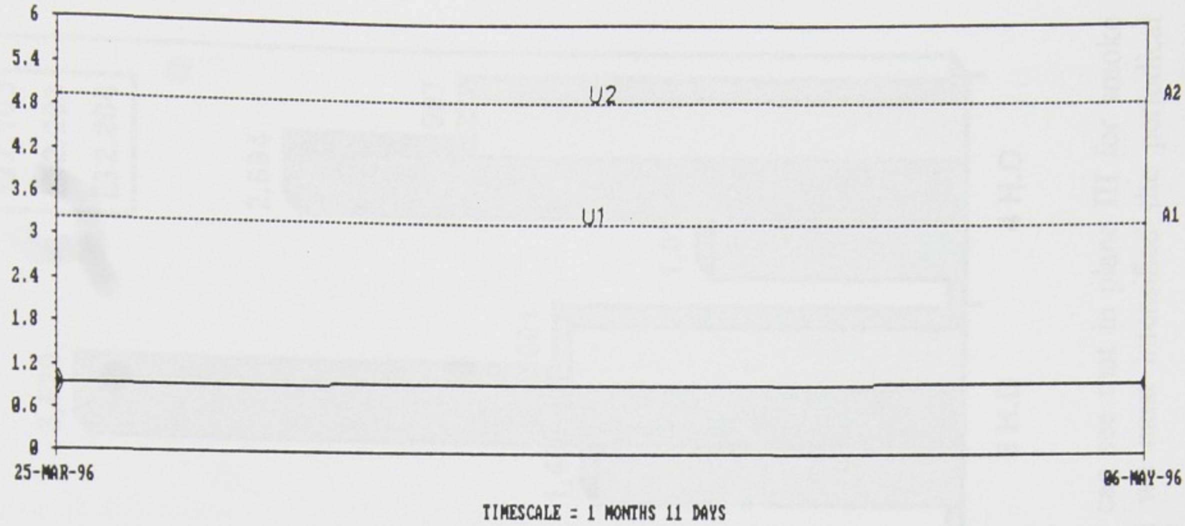
NO.	DATE	TIME	VALUE	INCL	NO.	DATE	TIME	VALUE	INCL
1.	25-MAR-96	09:27:08	1.744	Y	2.	06-MAY-96	09:09:58	1.736	Y

Fig. 52 The overall value versus time trend for smoke ventilator 2.204, „up“ trend is for outside bearing, the effective value of vibration acceleration increase from 0.9988 to 1.009 [mm . s^{-2}].

„Down“ trend is for inside bearing, the effective value of vibration acceleration decrease from 1.744 to 1.736 [mm . s^{-2}].

OVERALL VALUE VERSUS TIME TREND

SET ID: KV-2.103 SET DESC:diploma work
POINT ID: 3V.OUT.BEARING 2.103 POINT DESC: smoke ventilator-Envelope method
ALARM TYPE: LEVEL UNITS: Gs Env
ALARM1: 3.2 ALARM2: 4.9 STD: 0.0324665 MEAN:0.950678
DATE: 25-MAR-96 09:23:16 VALUE: 0.927



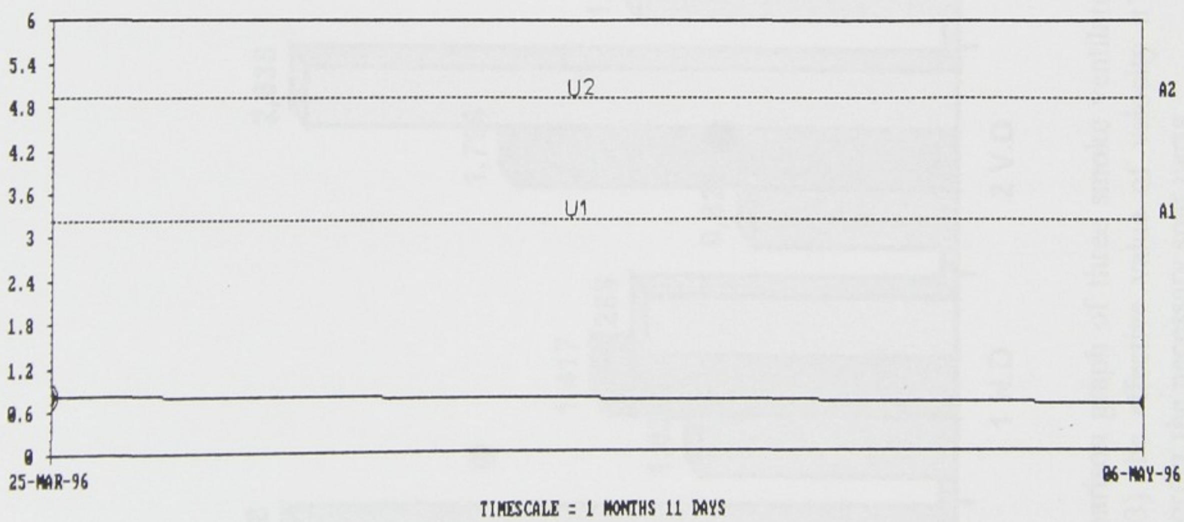
OVERALL TREND VALUES

NO.	DATE	TIME	VALUE	INCL	NO.	DATE	TIME	VALUE	INCL
1.	25-MAR-96	09:23:16	0.9277	Y	2.	06-MAY-96	09:32:46	0.9736	Y

SKF 20-MAY-96 STANDARD DATABASE Page 1

OVERALL VALUE VERSUS TIME TREND

SET ID: KV-2.103 SET DESC:diploma work
POINT ID: 4 H.IN.BEARING-2.103 POINT DESC: smoke ventilator-Envelope method
ALARM TYPE: LEVEL UNITS: Gs Env
ALARM1: 3.2 ALARM2: 4.9 STD: 0.104798 MEAN:0.719837
DATE: 25-MAR-96 09:21:46 VALUE: 0.793



OVERALL TREND VALUES

NO.	DATE	TIME	VALUE	INCL	NO.	DATE	TIME	VALUE	INCL
1.	25-MAR-96	09:21:46	0.7939	Y	2.	06-MAY-96	09:31:48	0.6457	Y

Fig. 53 The overall value versus time trend for smoke ventilator 2.103, „up“ trend is for outside bearing, the effective value of vibration acceleration increase from 0.9277 to 0.9736 [mm . s^{-2}].

„Down“ trend is for inside bearing, the effective value of vibration acceleration dcrease from 0.7939 to 0.6457 [mm . s^{-2}].

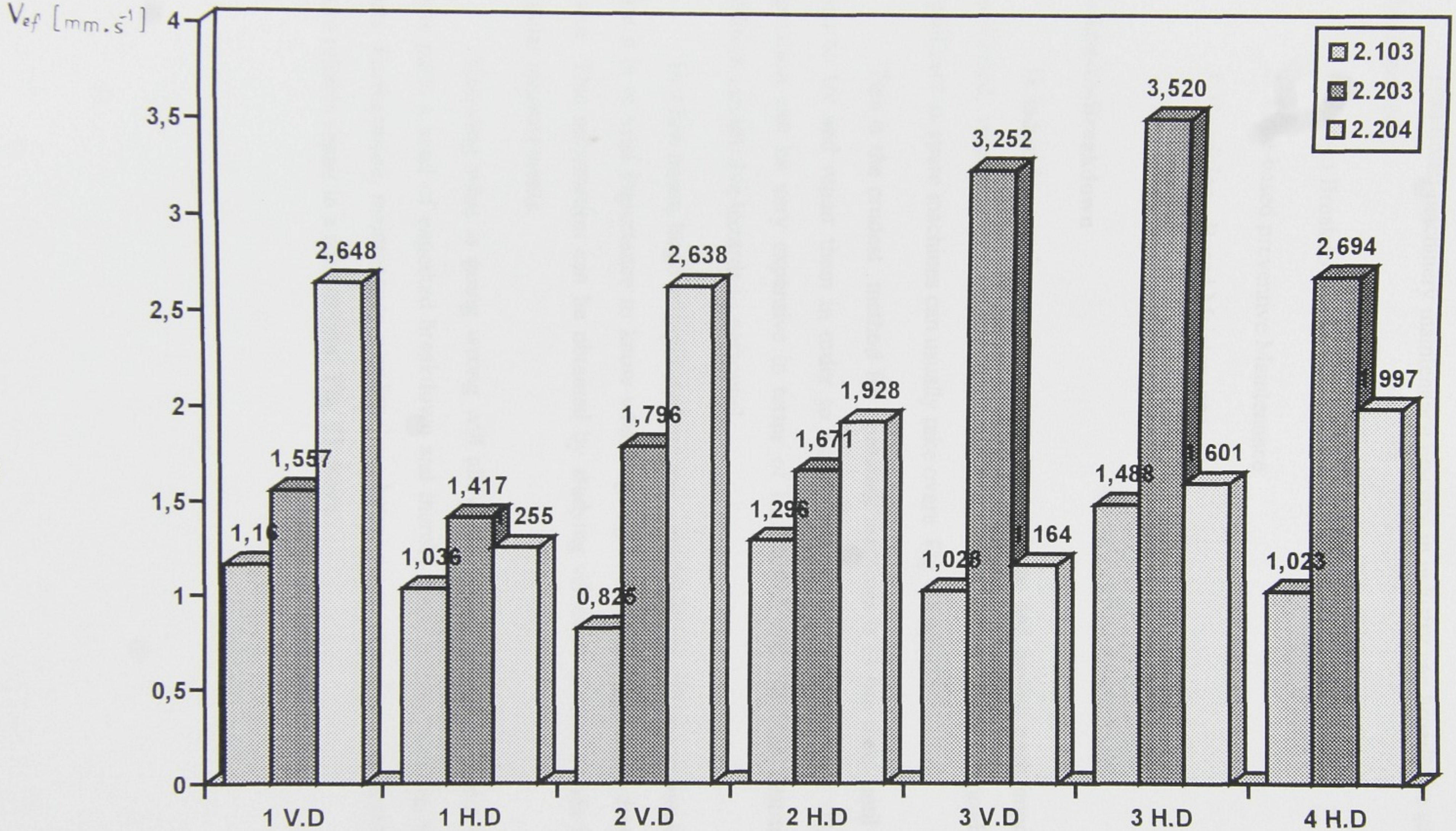


Fig. 54 Comparison graph of three smoke ventilators (2.103, 2.203, 2.203), we can see that in plane III for smoke ventilator (2.203) the effective value of velocity it's above U_1, from this result we must intensified the periodical monitoring and order the necessary spar parts.

8. MAINTENANCE METHODS AND ECONOMICS

Traditional machinery maintenance practice in industry can broadly be categorized into three methods:

1. Run-to Breakdown

2. Time-based preventive Maintenance

3. Condition - Based Maintenance.

1-Run-to-Breakdown

In industries running many inexpensive machines and having each important process duplicated, machines are usually run until they break down. Loss of production is not significant as apare machines can usually take overe. Fig. 55 up.

This is the crudest method for operating machines is to run them until they fail, and then to try and repair them in order to make them fit for further service. This method of operation can be very expensive in terms of lost output and machine destruction, and in addition can involve hazards to personnel.

In a few cases, large unduplicated process - machines are still run to breakdown. In this case it is of vital importance to know what is going wrong and when breakdown is likely to occur. This information can be obtained by studying vibration spectra trends built up from regular measurements.

Knowing what is going wrong will allow the plant engineer to order the necessary spare parts a head of expected breakdown and thereby avoid a large standing stock of spare parts. Furthermore, maintenance personnel are better prepared and can be expected to plan a more reliable repair in a shorter time. Fig. 55 down.

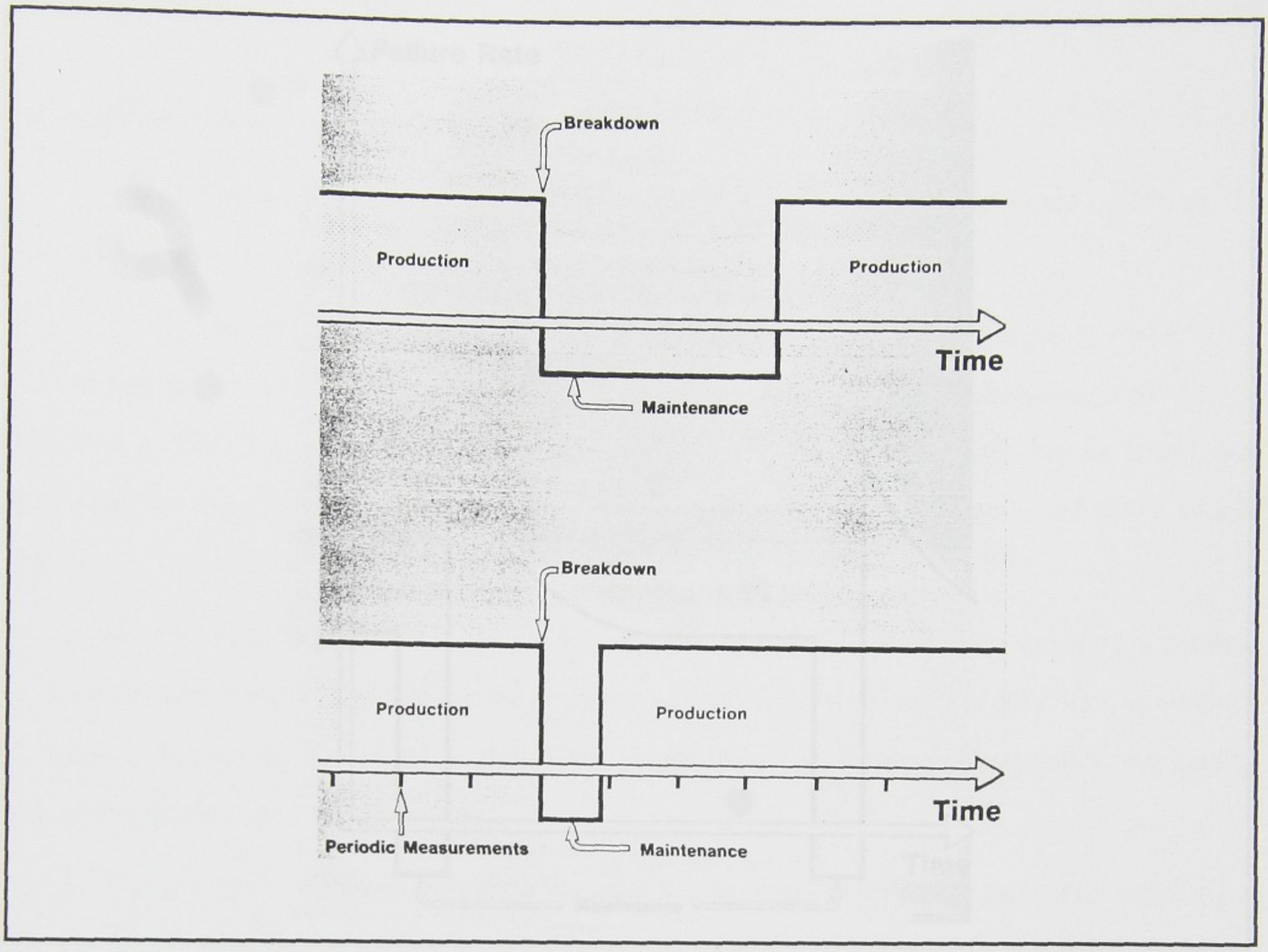


Fig. 55 Run to breakdown

2-Time - based Preventive Maintenance

Where important machines are not fully duplicated or where unscheduled production stops can result in large losses, maintenance operations are often performed at fixed intervals such as every 3000 operating hours or once per year.

This system is therefore called **Preventive Maintanence**, or more correctly, **Time - based Preventive Maintenance**.

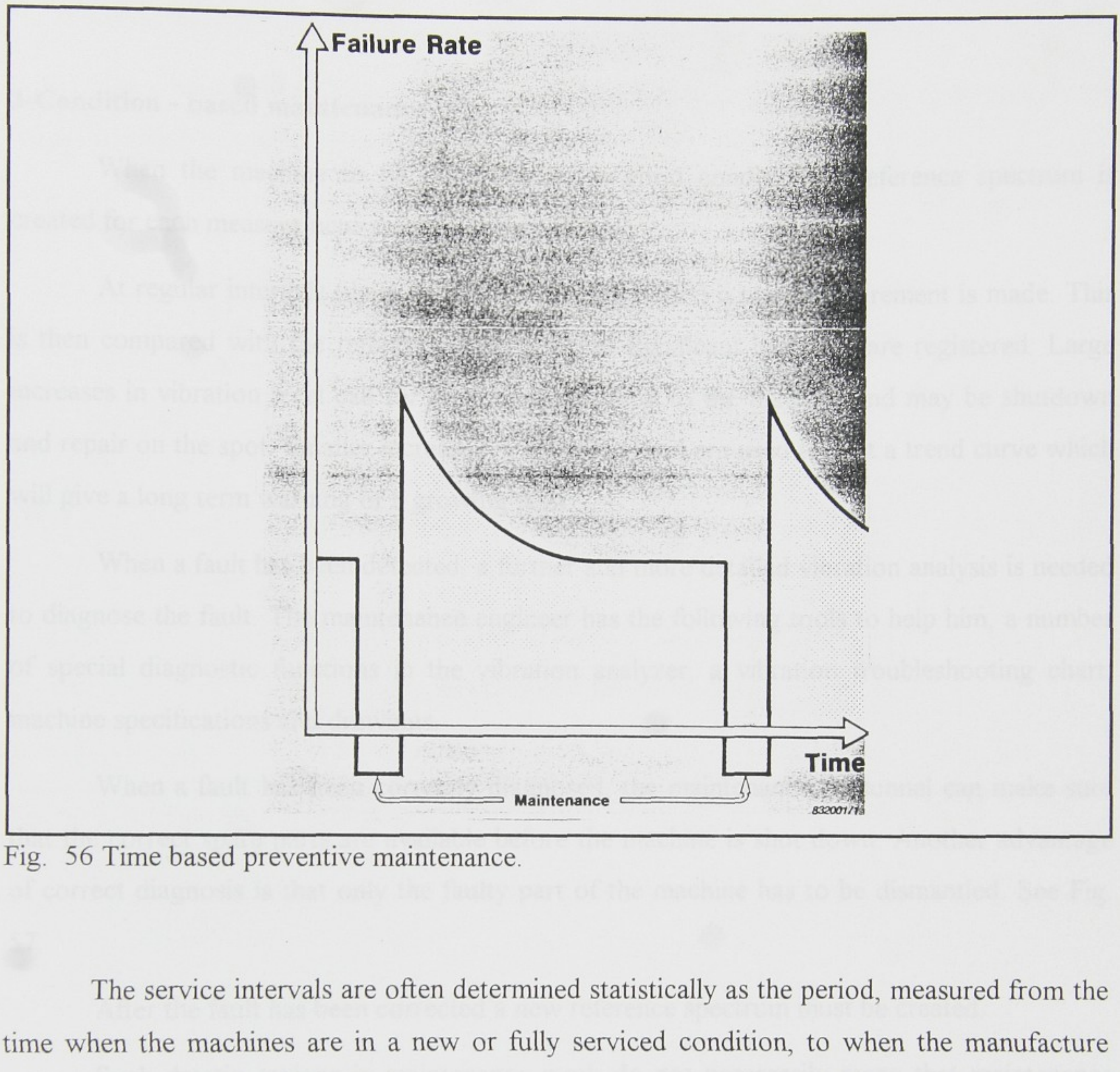


Fig. 56 Time based preventive maintenance.

The service intervals are often determined statistically as the period, measured from the time when the machines are in a new or fully serviced condition, to when the manufacture expects no more than 2% of the machine population to have faild. By servicing at these intervals it is generally believed that as 98% of the machines should survive the running period, failure should be a rare occurrence. Fig. 56.

Experience has shown that in the vast majority of cases, time-based preventive maintenance is uneconomical. A significant fact is that the failure rate of many machines is not improved by replacing wearing parts regularly. On the contrary, the reliability of newly-serviced machines is often reduced temporarily by human interference. As the actual failure pattern for each individual machine cannot be predicted, time-based preventive maintenance cannot be efficiently applied. An individual approach is needed, and this is the axiom of condition-based maintenance.

3-Condition - based maintenance

When the machine to be monitored is in good condition, a reference spectrum is created for each measurement point on the machine.

At regular intervals (days, weeks, or running hours) a new measurement is made. This is then compared with the reference spectrum and significant increases are registered. Large increases in vibration level call for immediate attention to the machine and may be shutdown and repair on the spot. Smaller increases in vibration level are used to plot a trend curve which will give a long term warning of a growing fault.

When a fault has been detected, a further and more detailed vibration analysis is needed to diagnose the fault. The maintenance engineer has the following tools to help him, a number of special diagnostic functions in the vibration analyzer, a vibration troubleshooting chart, machine specifications and drawings.

When a fault has been correctly diagnosed, the maintenance personnel can make sure that the correct spare parts are available before the machine is shut down. Another advantage of correct diagnosis is that only the faulty part of the machine has to be dismantled. See Fig. 57.

After the fault has been corrected a new reference spectrum must be created.

Such drastic 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 condition-based maintenance with respect to his plant.

The economics of condition monitoring and the main savings which can be made by the application of condition monitoring on industrial machinery arise by avoiding losses of out-put due to the break down of machinery, and by reducing the costs of maintenance.

The diagram in Fig.58 shows the procedure required to establish and operate an condition-based maintenance programme.

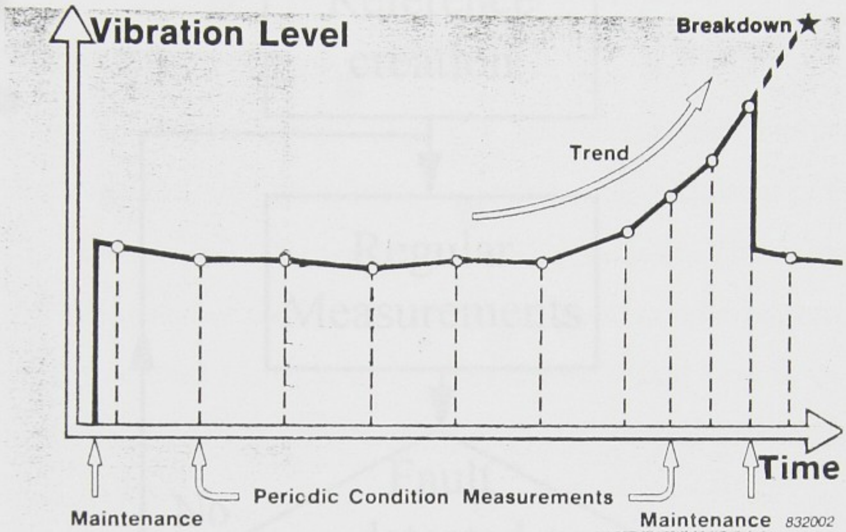


Fig. 57 Condition-Based Maintenence

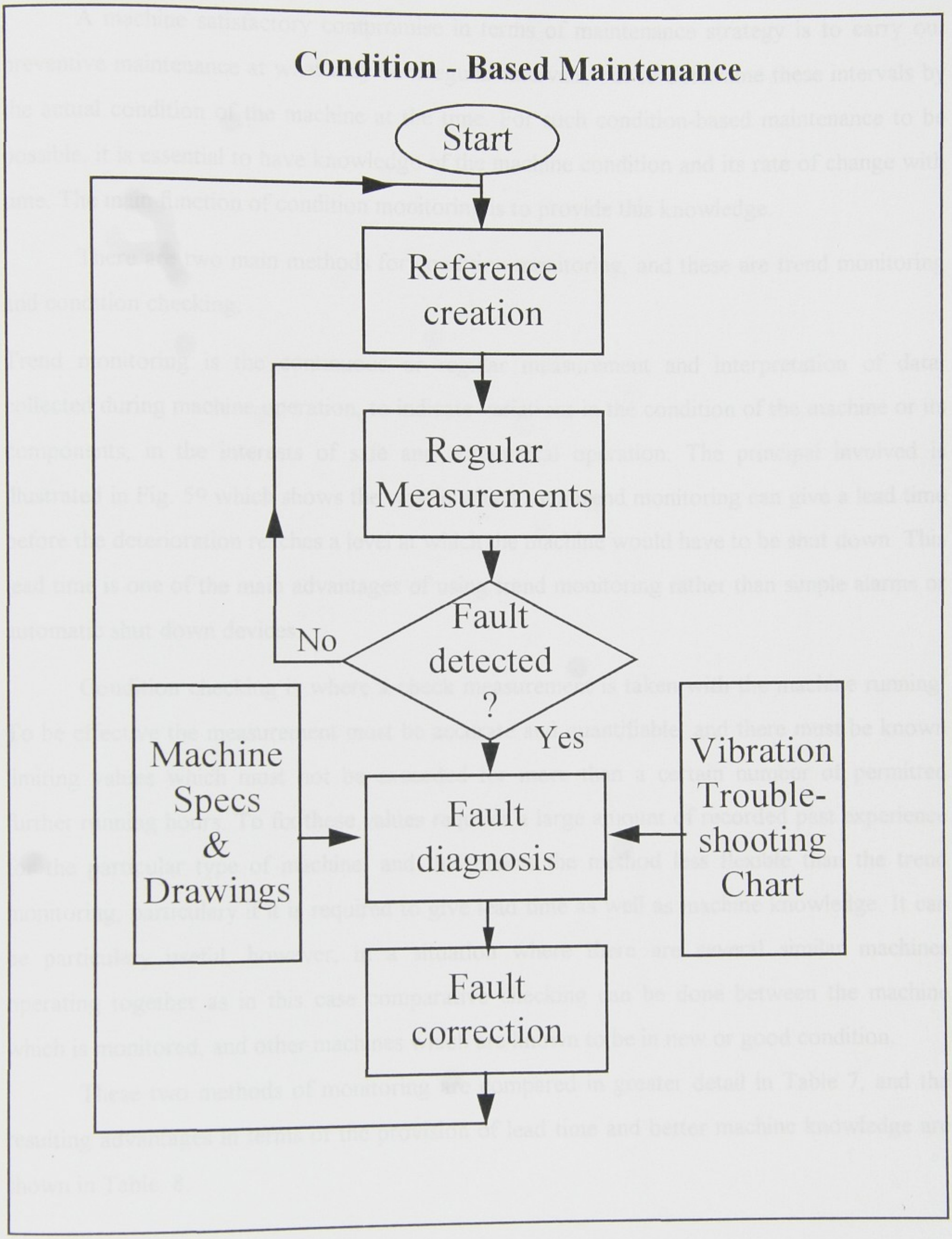

Condition - Based Maintenance
Start
Reference creation
Regular Measurements
Fault detected ?
No
Yes
Machine Specs & Drawings
Fault diagnosis
Vibration Trouble-shooting Chart
Fault correction

Fig. 58

8.1 CONDITION MONITORNG

A machine satisfactory compromise in terms of maintenance strategy is to carry out preventive maintenance at what may be irregular intervals, but to determine these intervals by the actual condition of the machine at the time. For such condition-based maintenance to be possible, it is essential to have knowledge of the machine condition and its rate of change with time. The main function of condition monitoring is to provide this knowledge.

There are two main methods for condition monitoring, and these are trend monitoring and condition checking.

Trend monitoring is the continuous or regular measurement and interpretation of data, collected during machine operation, to indicate variations in the condition of the machine or its components, in the interests of safe and economical operation. The principal involved is illustrated in Fig. 59 which shows the way in which such trend monitoring can give a lead time before the deterioration reaches a level at which the machine would have to be shut down. This lead time is one of the main advantages of using trend monitoring rather than simple alarms or automatic shut down devices.

Condition checking is where a check measurement is taken with the machine running. To be effective the measurement must be accurate and quantifiable, and there must be known limiting values which must not be exceeded for more than a certain number of permitted further running hours. To fix these values requires a large amount of recorded past experience for the particular type of machine, and this makes the method less flexible than the trend monitoring, particulary if it is required to give lead time as well as machine knowledge. It can be particulary useful, however, in a situation where there are several similar machines operating together as in this case comparative checking can be done between the machine which is monitored, and other machines which are known to be in new or good condition.

These two methods of monitoring are compared in greater detail in Table 7, and the resulting advantages in terms of the provision of lead time and better machine knowledge are shown in Table. 8.

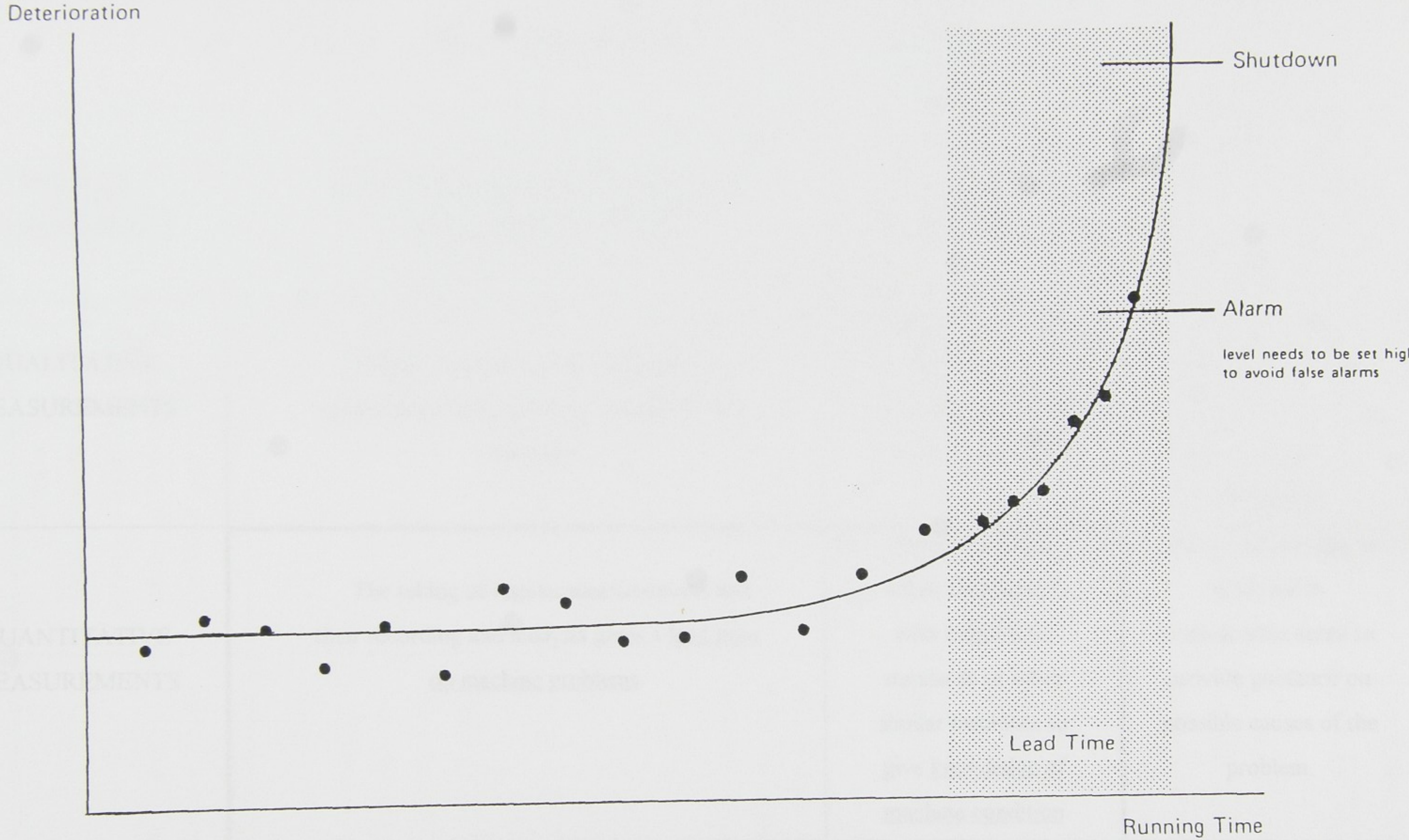


Fig. 59 The regular Monitoring of Deterioration to give advanced warning of failure

	TREND MONITORING	CONDITION CHECKING	PROBLEM OR FAILURE DIAGNOSIS
TIMING OF MEASUREMENTS	Readings taken at regular time intervals while the machine is running.	Readings taken at one time while the machine is running.	When the problem has become manifest or after failure has occurred.
QUALITATIVE MEASUREMENTS	Skilled operators can do subjective trend monitoring if they are close enough to their machines.	Typical activity of an engineer when checking a machine during operation.	When machine is stopped, inspection of components can indicate the cause of the problem.
QUANTITATIVE MEASUREMENTS	The taking of regular measurements and their recording and analysis gives a lead time on machine problems.	Numerate values allow comparison with established standards or other similar machines to give knowledge of machine condition	Measurements may be analysed in considerable detail to provide guidance on possible causes of the problem.
	CONDITION	**MONITORING**	

Tab 7 Comparison of Methods of Condition Monitoring and of Failure Diagnosis

Tab. 8 The advantages obtained by the use of Condition Monitoring

	ADVANTAGES OBTAINED		METHODS WHICH CONDITION MONITORING GIVES ADVANTAGES	
			Lead Time	Better Machine Knowledge
SAFETY	Reduced Injuries and Fatal Accidents to Personnal caused by machinery.		Enables plant to be stopped safely when instant shut down is not permissible.	Machine condition, as indicated by an alarm, is adequate if instant shut down is permitted.
OUTPUT	Increased Machine Availability	More Running Time	Enables machine shut down for maintenance to be related to required production or service, and various consequential losses from unexpected shut downs to be avoided.	Allows time between planned machine overhauls to be maximised and, if necessary, allows a machine to be nursed through to the next planned overhaul.
		Less Maintenance Time	Enables machine to be shut down without destruction or major damage requiring a long repair time. Enables the maintenance team to be ready, with spare parts, to start work as soon as machine is shut down.	Reduces inspection time after shut down and speeds up the start of correct remedial action.
	Increased Rate of Net Output			Allows some types of machine to be run at increased load and/or speed. Can detect reductions in machine effeciency or increased energyu consumption.
	Improved Quality of Product or Service		Allows advanced planning to reduce the effect of impending breakdowns on the customer for the product or service, and thereby enhances company reputation	Can be used to reduce the amount of product or service produced at sub-standard quality levels.

8.2 PERMANENT VIBRATION MONITORING OF MACHINERY (ON LINE SYSTEM)

A technique which is closely related to the methods of machine monitoring so far discussed is permanent vibration monitoring. As the name suggest, permanent monitoring is permanently employed on specific machines to continuously survey their condition. Its main purpose is to give early warning of incipient faults and immediate warning of sudden changes in the condition of expensive non-dupliceted machinery whose continuous operation is vital to the production process. Fault conditions are detected immediately, or within seconds of occurrence, and tigger alert or alarm signals in the control room allowing the appropriate measures to be taken before catastrophic failure occurs. Such systems are widely used in the power-generation, pulp and paper and petro-chemical industries on turbines, paper machines, feed pumps, gas compressors etc.

What Instruments are Required?

A prime requirement of all permanent monitoring systems is high operational-reliability, long-term stability and immunity to adverse environmental conditions and irregularities which can cause false alarms. They should be of sturdy mechanical design, capable of operating in humid and dust-laden conditions. Special "ruggedized" front-end ancillaries such as industrial accelerometers, cables and junction boxes, which can work at high temperatures, are essential requirements. Further, these systems should include an automatic test system to allow the plant operator to immediately check whether the instrumentation is functioning correctly in the event of an alarm.

What Should a Permanent Monitoring System Do?

Ideally, it should provide the maximum possible protection for the monitored machine, continuously monitoring each each measurement point's overall vibration level and generating alarms when this level exceeds pre-set limits. This should be combined with fully automatic or operator directed cyclical spectrum comparison for each measurement point, generating printed warnings on detection of unacceptable increases within frequency bands.

Overall vibration levels (broadband levels) are continuously monitired over a user-specified frequency range. If preset limits are breached (for example Minimum, Alert, and Alarm), the system can trigger visual or audible alarms. Economical coverage of up to sixteen monitoring points can be achieved by multiplexing the inputs through a single monitor module. The Multiplexer continuously steps through the chosen channels, dwelling at each channel for a precelected period before automatically moving to the next. In lager systems, these monitor/multiplexer blocks are connected to a serial data-bus - the system being controlled by an on-line computer. The computer, in conjunction with a vibration analyzer can cyclically access each vibration channel, record a frequency spectrum, compare the recorded spectrum with a stored reference spectrum, issue a printed warning on detection of component increases and then step to the next channel, continuously repeating the measurements. Special systems are also available for explosive-risk and radioactive areas.

8.3 THE LOSSES DUE TO A VENTILATOR BREAKDOWN IN THE HEAT PLANT

In The Liberec heat plant where the measurements have been performed, there is a great dependency on two unduplicated smoke ventilators. Each of them is depended upon the production of 4.5 MW.

The plant is in contract to supply its customers with 8.5 MW. This value varies according to the time of year, Table 9.

1	2	3	4	5	6	7	8	9	10	11	12
8.5	8.5	8.5	8.5	6	0	0	0	0	8.5	8.5	8.5

Table 9. The heat plant's supply time table

Each 1MW is sold for the price of 308,800 Kč. This implies that the average supply of 8.5 MW ($P_{theoretical}$) gives the monthly income of 2,624,800 Kč. Should the plant fall behind the production plan and manages to supply 85% (7.2 MW), the income would be 2,231,080 Kč. This would mean that the estimated losses would be around 390,000 Kč.

$$[308\ 800 \times (P_{theoretical} - P_{real})]$$

The contract states that should the average supply fall bellow the 85 % mark, there would be a 15 % fine on the total income. For a 70 % supply the losses are estimated using the following formula

$$(P_{theoretical} \times 308\ 800 - P_{real} \times 308\ 800(1 - 0.15))$$

This would give us the estimated losses value of 1,063,000 Kč from the 100 % supply. Should the monthly average supply fall below the 70 % mark then the customers are no longer obliged to pay their months consumption.

This contract applies mostly in The critical hours which are between 07:00 to 15:00 of the day. Should a breakdown occur during these critical hours, and the supply ark would fall below 70%.

The total losses then would be as follows:

$$8.5\ \text{MW} \times 308\ 800\ \text{Kč} = 2\ 600\ 000\ \text{Kč}$$

It is seen that the losses are very great. Luckly no breakdown has occured during the critical hours before, still the chances are there.

There are two ways to prevent this form occuring. Firstly, Permanent vibration monitoring (on line system). The equipment cost for this method is ranging from 1 500 000 Kč.[1]

Secondly Preventive maintenance (off line system). The equipment for this method is much less than the one stated above (500 000)[1] , but its capability to give a prompt warning is not as effective as permanent vibration monitoring.

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[1] These prices have been provided by the Brüel & Kjaer comp.

9. CONCLUSION

From this diploma project it shown in the begining the relation between the ventilator and vibration and the method of calculating the theoretical frequencies. Fault detection was later explained. From the results of measurements and calculations made, the methods of maintenance available have been explained.

Through the measurements which have been done in the heat plant it has been found that should the velocity amplitude be below the U_1 (= 3.2 mm/s) level, then the machine part of the ventilator is in good state.

Above the U_2 (= 4.94) level, then it's considered critical and maintenance should take place.

From the measurement and the current state, the ventilators are in, it is advised to proceed with an OFF-LINE system of monitoring every 4 weeks.

Taking into consideration that the ventilators are unduplicated in the heat plant, it is recommended to install an ON-LINE system to monitor the ventilators if it's within the plant's economical capability.

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