

**The Lifetime Reliability Solutions  
Certificate Course in Maintenance and Reliability  
Module 2 – Machinery Vibration Analysis Fundamentals**

**Session 9  
Rolling Element Bearing Testing**

**1. Introduction.**

In this session we will consider in more detail the special methods used to provide early detection of bearing faults for the purposes of condition monitoring and machinery acceptance testing.

Vibration measurement, as we have considered it up to this point, assumes an essentially linear relationship between the force developed in the machine or structure and the vibration measurements made. For the purposes of analysis we assume that the machine is acting as a rigid body and the measurements we make have meaningful values because the measuring system is calibrated accordingly. When we come to the world of early bearing fault detection we leave most of that behind!

At very high frequencies (say from 5kHz to 50kHz) vibration no longer manifests in a ‘whole body’ mode but increasingly becomes like shock waves travelling through the body and reflecting randomly at surfaces. Mating faces become significant attenuators of these shock waves making measurements at external locations difficult to evaluate.

To assist with the measurement process, the uncalibrated natural frequency response of the accelerometer is deliberately used to help increase the sensitivity of the measurement system at these high frequencies. Various signal processing tools are then applied to help make sense of the output.

Names like Shock Pulse System (SPM), Spike Energy, High Frequency Resonance Demodulation (HFRD), SEE Technology, Peak-View and HFD are all commercial systems that have been designed to provide measures of bearing condition.

There are no Standards making specific reference to these methods of measurements and a limited amount of guidance is provided by manufacturers to assist in evaluation of the measured values. Note however, under the section on Standards that there is increasing *recognition* of these techniques without giving any specific guidelines for their proper use.

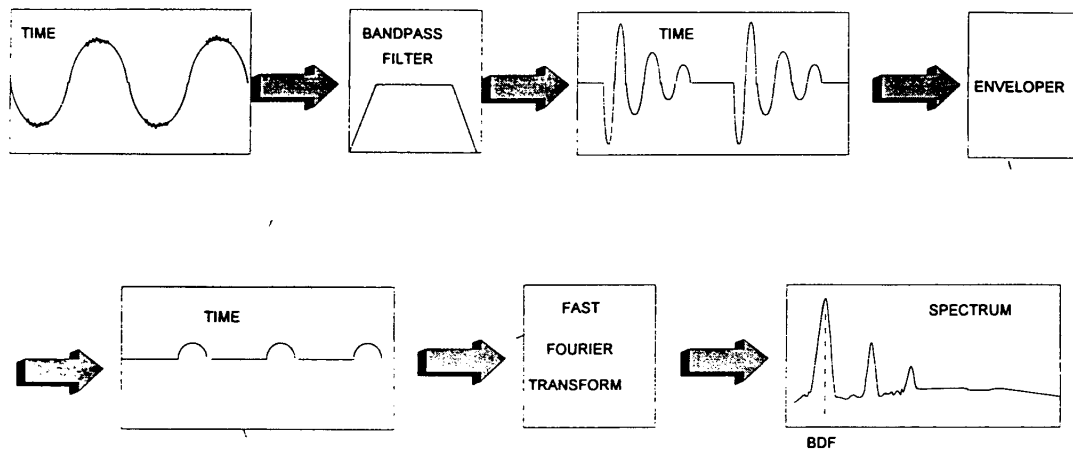
The user is encouraged to look for increases in measured values above what has been determined is a good or acceptable condition for each bearing measured. Experience helps in this situation!

The Shock Pulse Method was the first of these systems in the 1970’s and is a well proven technology. The more recent systems do not appear to offer any significant advantages, if any. It is intended to work with the mounted transducer having a resonance of 36kHz.

**2. How Bearing Fault Detection Systems Work.**

All the proprietary systems named above, and many others, use the same basic principles of detection. The differences are mainly in the methods of data processing and methods of presentation.

A typical block diagram for an Amplitude Demodulation (Envelope Detection) system.



The basic principles are described as follows.

- All bearings – even those in perfect condition – produce measurable ‘noise’ as the elements roll over the raceways and rub against the cage and the flanges.
- This noise is generated at quite high frequencies and low amplitudes. The natural frequencies of the bearing and bearing housing and associated support structures can in fact amplify these very low amplitude signals to the point where a sensitive accelerometer can ‘hear’ them.
- By careful filtering and detection methods these low amplitude signals can be amplified and presented as an overall value and/or time domain or frequency domain data.
- When the elements or raceways produce more noise for a variety of reasons – as discussed below – the systems are able to detect this increase in amplitude and present the data for analysis.
- Most of the faults that occur in rolling element bearings are directly related to either the rolling elements, the inner or outer raceways or the cage. For example, a fault in the outer raceway of a bearing will produce an impact every time an element passes over it – in the load zone. The frequency of this fault has a direct relationship to the geometry of the bearing and the relative speed of rotation of the raceways.
- Software libraries are available that list the ‘passing frequencies’ of just about every bearing ever manufactured. This data can be directly transferred to the spectrum and confirmation of the fault enabled.
- The actual method of presentation of the data varies but most systems are quite effective in presenting an overall value for trending and spectrum information for analysis. There is not a lot to pick between them all.
- The circuits described on the Basic Vibration Measurement and Analysis Chart given in Session 3 are examples of how these systems work and present their data.

### 3. Bearing Failure Modes.

Bearing condition measurement systems are designed to detect the earliest possible signs of distress in a bearing. We will briefly consider the most important of these ‘distress signals’ in the following notes.

#### 3.1 Lubrication.

Friction and wear behaviour and the attainable life of a rolling element bearing depend directly on the lubrication condition. Therefore the ability to measure less-than-desirable lubrication conditions is vital to the optimisation of bearing life.

Bearing lubrication conditions are generally categorised in three groups:

- **Full Film Lubrication.** This is where the elements and raceways are separated from each other by a film of oil.
- **Mixed Lubrication.** Where the lubricant film is too thin, local metal-to-metal contact occurs, resulting in an increase in friction and heat generation.
- **Boundary Lubrication.** This is an unacceptable condition unless certain extreme-pressure additives have been included in the lubricant. These are fused in place by the high pressures and temperatures and maintain an acceptable lubricating effect.

While full film lubrication is the ideal condition it is probable that most industrial bearings operate in the mixed lubrication regime. This means that most proprietary bearing condition detection systems can readily measure a ‘noise floor’ in the bearing and analysis of these signals can show the presence of passing frequencies at low amplitudes.

One manufacturer goes so far as to claim that their system can actually measure the lubrication film thickness. We interpret that to mean that the noise floor can be measured and determined to be in an acceptable state below certain values.

It is interesting to note that increasing load is not a critical factor for lubrication film thickness. One bearing manufacturer states that, within the design range, every doubling of load decreases film thickness by 5% for point loaded and 9% for line loaded bearings.

### **3.2 Abrasive Particles.**

The biggest enemy of bearing life in the mining industry is the ingress of abrasive particles. When these are bigger than the film thickness – and this is usually the case - then the result is scoring of the surfaces which results in a wear process.

Very fine particles can have the effect of generating significant wear but maintaining very highly polished raceways and elements. This makes detection by the high frequency detection methods very difficult because no passing frequencies are generated. Sometimes clearances open up so significantly that velocity measurements can detect looseness before the bearing condition system has flagged any change at all!

The sealing of bearings is a real challenge. A common method for sealing plummer blocks is to use grease-purged labyrinth seals at the shaft entry or entries. The alert condition monitoring engineer watches closely the condition of bearing seals.

### **3.3 Ingress of Liquids.**

The entry of liquids such as process chemicals and water generally results in a corrosive effect on the bearing and severe reduction in life. (Beware the washdown hose!!). These effects are easily identified at early stages by all bearing fault detection systems as the etched surfaces generate impacts at low amplitudes.

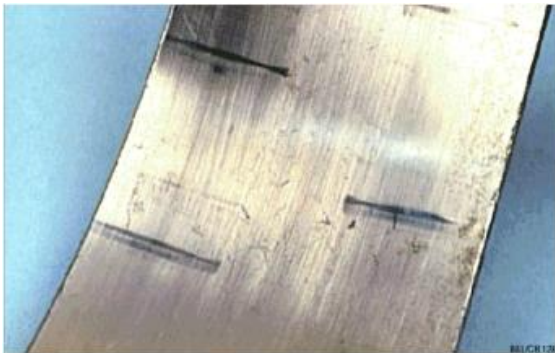
These pollutants can be cleaned out of the bearing by flushing the oil or grease purging and life may be extended as a result.

### 3.4 False Brinelling.

Machines that stand idle on a regular basis, or are transported long distances, can suffer an effect known as False Brinelling.

The name is taken from the Brinell Hardness Test where the measure of indentation is a measure of hardness. False Brinelling occurs when the movement of bearing elements against the raceways of a stationary machine – either by transport or background vibration – causes the lubrication film to break down and the metal surfaces to bind together with resultant etching of the surfaces.

Here is a photo of typical False Brinelling.



False Brinelling is very easily detected as passing signals – usually inner and outer raceways. False Brinelling is much harder to manage or eliminate. Line contact bearings are much more prone to this problem than point contact. Therefore, where deep groove ball bearings can replace cylindrical roller bearings (such as on the DE of direct-coupled electric motors) this should be done. Regular running of stand-by machines is another preventative action.

### 3.5 Fatigue Life Limit.

Not many bearings in the mining industry reach the end of their design life and fail by surface fatigue. Bearings that are removed with pitted raceways that may *appear* to be at the end of their design life may in fact only be a few months or years old. The causes may be related to any of the above conditions that have initiated failure.

Root Cause Analysis can be applied to find out why bearings failed early and appropriate steps taken to minimise these effects. Great care is needed in removal of the bearings to preserve all the needed evidence. This is discussed more fully in Unit 2.

## 4. Bearing Defect Frequencies

Over the years, a series of formulas have been developed which identify specific defect frequencies for rolling element bearings. They can separately detect faults on the inner race, outer race, cage or rolling elements themselves, and are based upon the bearing geometry, the number of rolling elements and the bearing rotational speed. Formulas are provided for each of these four bearing defect frequencies, along with some unique relationships between these frequencies.

This means that if the design parameters of a bearings (pitch diameter, rolling element diameter, number of rolling elements & contact angle) are known, it is possible to detect problems which occur on the races, cage or rolling elements and to track these problems as deterioration continues. In many cases, all the parameters to insert for a particular may not be known, but the bearing manufacturer and model number is known. In these cases, there are several databases published or available on the internet which tabulate each of the four defect frequencies for each of the bearing model numbers. Some of these databases may have up 12,000 or 20,000 different bearings listed. An extract from one such database is shown listing these defect frequencies. Note that this table provides the number of balls ( $N_b$ ), rolling element diameter ( $B_d$ ), bearing pitch diameter ( $P_d$ ), contact angle ( $\theta$ ), outer race defect frequency (BPFO), inner race defect frequency (BPFI), cage defect frequency (FTF) and rolling element defect frequency (BSF). From this the SKF N222 bearing is given as an example. Note that each one of the bearing frequencies are given in terms of running speed orders and, therefore, the frequencies for the SKF N222 bearing show, for example, that the inner race frequency (BPFI) is 9.18 X RPM. Thus, if this bearing were turning at 1000 rpm,(16.67Hz) and in the vibration spectrum a frequency was detected at 9180 cpm (153.0Hz), it could be concluded that this peak is the bearing inner race defect frequency and that a fault is developing there. On the other hand, if a frequency of 6810 cpm (113.5Hz) occurred on this same unit, it would be concluded that this is the outer race defect frequency of this SKF N222 bearing (since BPFO = 6.81 X rpm).

#### 4.2 The Calculation of Rolling Element Bearing Defect Frequencies

These equations assume the inner race is rotating with the shaft while the outer race is fixed (stationary).

$$\text{INNER RACE} \quad = \text{BPFI} \quad = \frac{N_b}{2} \left\{ 1 + \frac{B_d}{P_d} \cos \theta \right\} X \text{ RPM} \quad (1)$$

$$\text{OUTER RACE} \quad = \text{BPFO} \quad = \frac{N_b}{2} \left\{ 1 - \frac{B_d}{P_d} \cos \theta \right\} X \text{ RPM} = N_b X \text{ FTF} \quad (2)$$

$$\text{BALL (OR ROLLER)} = \text{BSF} \quad = \frac{P_d}{2B_d} \left[ \left\{ 1 - \frac{B_d}{P_d} \cos \theta \right\}^2 \right] X \text{ RPM} \quad (3)$$

$$\text{CAGE} \quad = \text{FTF} \quad = \frac{1}{2} \left\{ 1 - \frac{B_d}{P_d} \cos \theta \right\} X \text{ RPM} \approx 0.35 \sim 0.45 X \text{ RPM} \quad (4)$$

NOTE: If the inner ring is fixed (stationary) while the outer race rotates with the shaft, the minus sign must be changed to a plus sign within the parentheses of **Cage** Frequency Equation (4). In this case,  $N_b X \text{ FTF}$  will now equal BPFI rather than BPFO; and  $\text{FTF} = 0.55 \sim 0.65 X \text{ RPM}$ .

(NOTE that  $\text{BPFI} + \text{BPFO} = N_b X \text{ RPM}$ )

where:

$N_b$	Number of Balls or Roller
$B_d$	Ball or Roller Diameter (mm)
$P_d$	Bearing Pitch Diameter (mm)
$\theta$	Contact Angle (degrees)

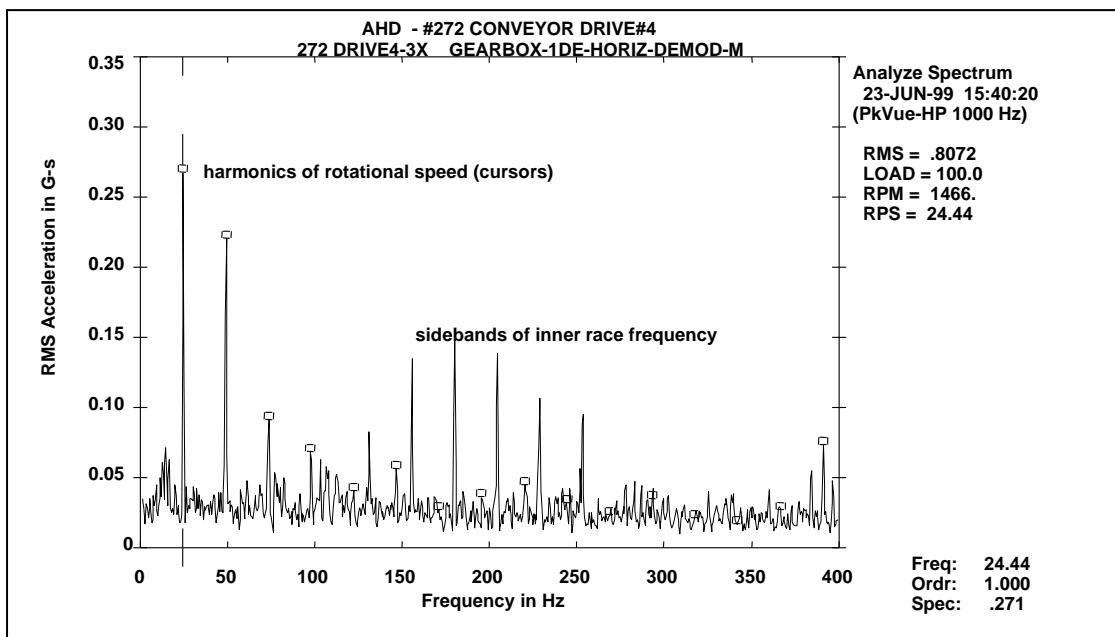
### 4.3 Sample Tabulation of Rolling Element Bearing Defect Frequencies

#### SKF Single Row Cylindrical Roller Bearing

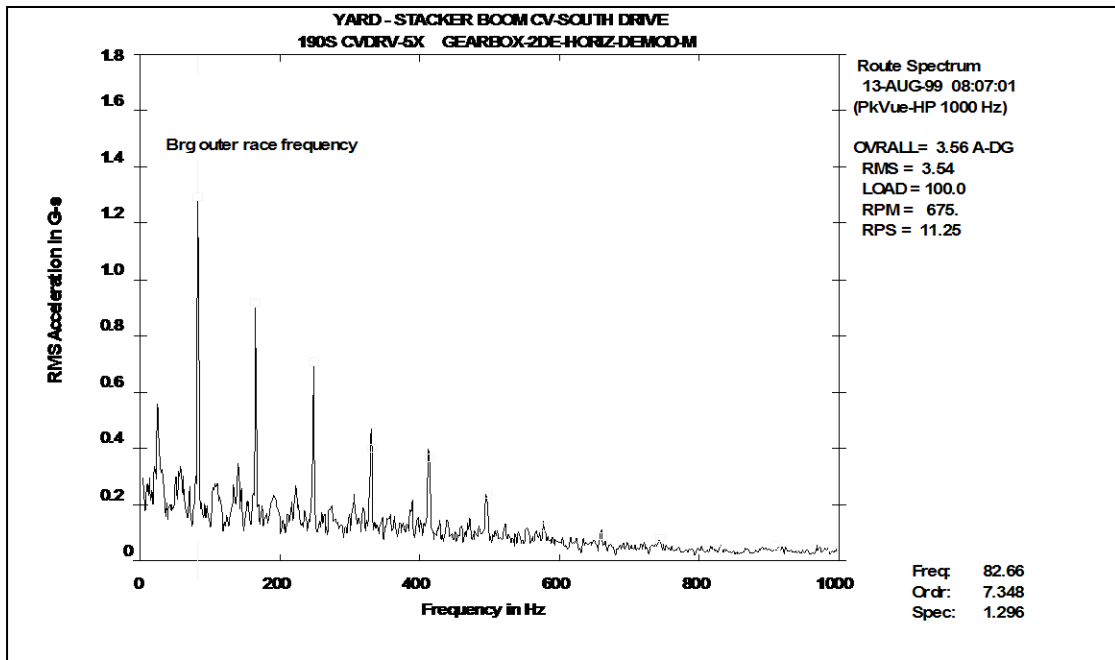
Bearing	Nb	Bd	Pd	$\theta$	BPFO	BPFI	FTF	BSF
N205	12	.256	1.516	0	4.99	7.01	.41	2.87
N206	12	.295	1.811	0	5.00	6.94	.41	2.99
N207	12	.354	2.106	0	4.97	6.98	.41	2.89
N208	13	.394	2.362	0	5.42	7.58	.41	2.92
N209	14	.394	2.559	0	5.97	8.07	.42	3.17
N210	15	.394	2.968	0	6.50	8.48	.43	3.70
N211	15	.433	3.051	0	6.43	8.56	.43	3.45
N212	16	.472	3.476	0	6.92	9.08	.43	3.61
N213	16	.512	3.641	0	6.88	9.11	.43	3.49
N214	16	.512	3.738	0	6.93	9.06	.43	3.68
N215	17	.551	4.035	0	7.34	9.65	.43	3.59
N216	16	.590	4.331	0	6.91	9.08	.43	3.60
N217	16	.620	4.625	0	6.91	9.08	.43	3.60
N218	16	.709	4.921	0	6.84	9.15	.43	3.40
N219	16	.748	5.216	0	6.85	9.14	.43	3.41
N220	16	.787	5.512	0	6.86	9.13	.43	3.43
N222	16	.905	6.102	0	6.81	9.18	.42	5.59
N224	16	.945	6.594	0	6.85	9.14	.42	3.41
N225	18	.945	7.087	0	7.91	10.07	.43	3.32
N228	18	1.024	7.677	0	7.76	10.19	.43	3.68

For Example:

An SKF N222 Bearing has a BPFI = 9.18 X RPM (inner race)  
 BPFO = 6.81 X RPM (outer race)  
 BSF = 5.59 X RPM (rolling elements)  
 FTF = 0.42 X RPM (cage frequency)



Typical Amplitude Demodulated Spectrum of an Bearing Inner Race Defect



Typical Amplitude Demodulated Spectrum of a Bearing Outer Race Defect

**4. Evaluation of Bearing Condition**

Two scenarios, each of four stages will be presented, as ways of tracking the bearing damage development. Also presented is a series of questions aimed at deciding when the bearing should be replaced so that replacement may be safely and effectively managed.

There is no substitute for collecting historical data to build knowledge of the characteristics of your own machines and especially trending the rate of deterioration of bearing defect frequencies in the demodulated spectra.

Bearing failures can be notorious for presenting themselves in ways that are outside of the “normal” patterns. It is, therefore, useful to be able to apply more than one technique when the risk justifies it and to illustrate this the second scenario is presented as a combination of Velocity and Demodulated spectra. The Demodulated spectrum can be used alone to very good effect, and frequently is.

**Caution** The scenarios presented are hypothetical spectra representing the progression of bearing damage and should be used as a guide only. A more comprehensive coverage of the potential scenarios would require a more comprehensive 3 day course.

**4.1 Four Stages of Bearing Failure – Vibration Velocity Spectra**

A typical bearing failure scenario is given at Figure 1. The spectra are in Vibration Velocity linear scaling to a span of 2kHz. Also included are overall values of Spike Energy gSE as an example of the behaviour of an overall bearing condition measurand. The failure mode is an outer race defect. Some commentators recommend the Vibration Velocity scaling be Logarithmic to enhance the low level bearing defect frequencies.

**Stage 1** At this stage the bearing has no defects and is in a satisfactory condition. It may be used a reference.

The spectrum shows the rotational frequency and a few harmonics, a fairly normal situation.

**Stage 2** If examined the bearing would probably have no visible defects. It would be necessary to cut and etch the bearing race to identify the sub-surface defects.

The overall bearing condition reading has increased significantly. This in itself could be due to inadequate or contaminated lubricant.

The Velocity spectrum shows activity at the higher frequency end which may be seen as a discreet frequency or as a featureless hump of energy. A possible further sub-stage might be the presence of rotational frequency sidebands on the discreet frequency.

**Stage 3** This stage breaks into three sub-stages. Bearing damage is visible and the bearing life may run from 5%L10 life down to <1%L10 life. In these stages it is very important to track the rate at which characteristics change to establish the likely remaining life.

**Stage 3A** Damage may now be just visible, depending upon the nature of the failure.

The overall bearing condition reading has continued to increase.

The Velocity spectrum

- now shows a single peak at the outer race defect frequency; this may be at 3x to 20x the rotational frequency depending upon the bearing fitted
- shows some development of the activity at the high frequency end, possibly with the addition of rotational frequency sidebands.

**There is early damage but action would not be justified at this point other than to monitor more closely. There is significant life remaining.**

**Stage 3B** Damage is more fully developed.

The overall bearing condition reading has continued to increase.

The Velocity spectrum

- has developed some harmonics of the outer race defect frequency
- the discreet peak at the high frequency end may have developed more extensive sidebands, or the energy hump may have become more extensive.

**Early planning for replacement is appropriate, bearing in mind production requirements.**

**Stage 3C** Damage will now be quite severe and more extensive.

The overall bearing condition reading has continued to increase.

The Velocity spectrum

- rotational frequency sidebands have developed around the bearing defect frequency and its harmonics
- the discreet peak at the high frequency end may have developed more extensive sidebands, or the energy hump may have become more extensive.
- with the further development of damage other bearing defect frequencies are seen.
- up till now there has been little or no change in the amplitude and harmonics of rotational frequency.

**At this advanced stage there is insufficient bearing life left remaining to justify operating with the increased risk of failure. Replace at the next convenient opportunity, subject to the rate of defect development.**

**Stage 4** If the bearing is still in service at this stage it must be with everyone aware of the risks.

The situation has gone from Management to Russian Roulette.

The overall bearing condition reading

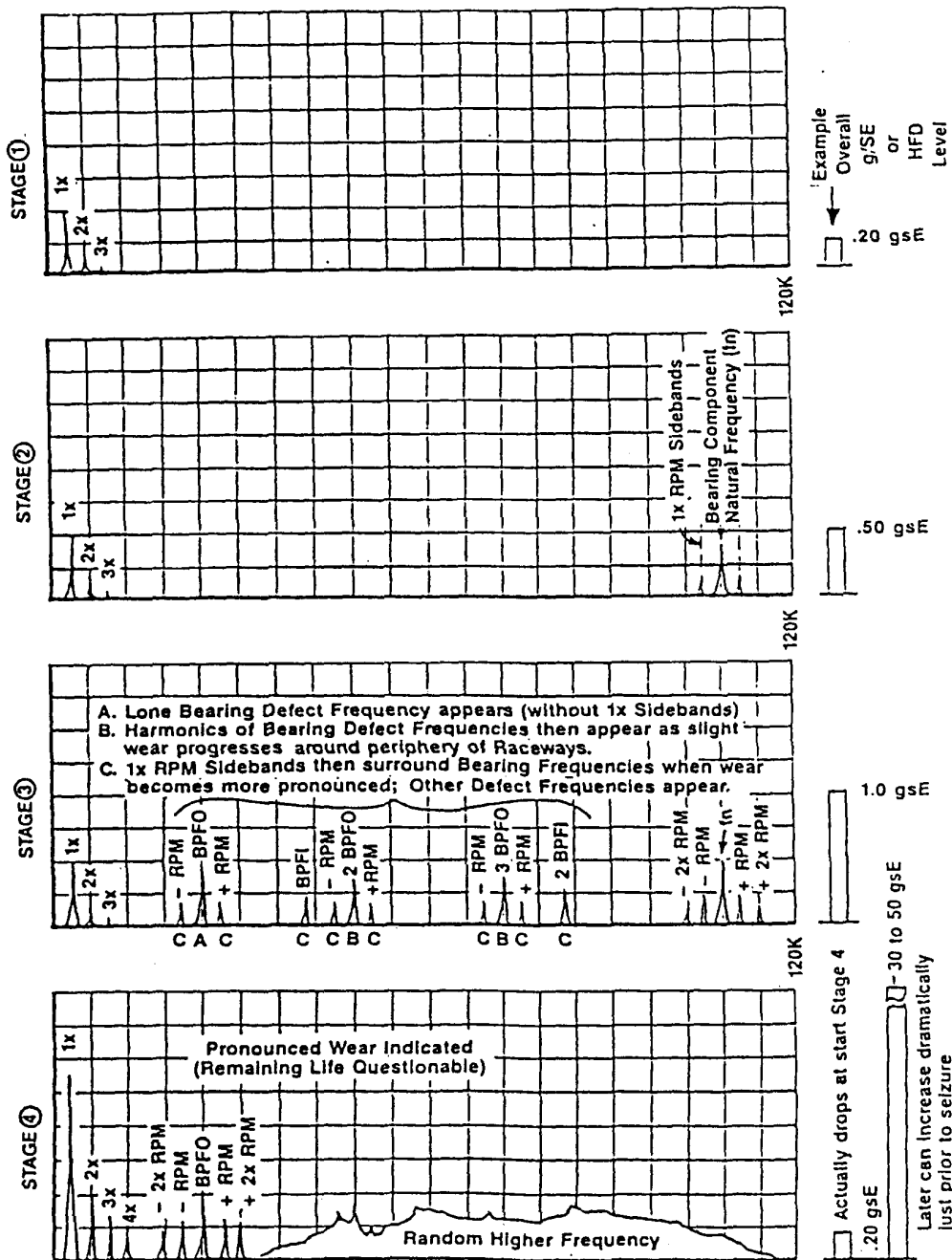


- may actually reduce for a period and then increase again to give very high and erratic values.
- this transition is an important identifier in the defect development.

The Velocity spectrum

- the hump of broadband higher frequencies have extended well down into the spectrum and obscured some of the bearing defect frequency harmonics.
- There is a greater range of rotational frequency sidebands on the bearing defect frequency
- there are now increases in the amplitude and harmonics of rotational frequency.

**Remaining life may now be a matter of days, or even hours, and an opportunity should be made for immediate replacement.**



**4.2 Four Stages of Bearing Failure – Demodulated Spectra**

A hypothetical inner race failure scenario is given at figure 3. Both spectra are given with the velocity in Logarithmic scaling (Velocity dB)

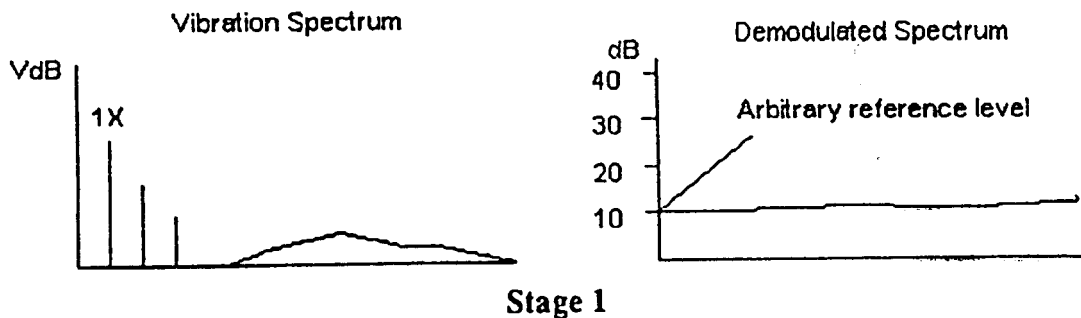
**Stage 1** Is a normal condition with no defects and is taken as the reference condition.

The Velocity spectrum shows

- normal rotational frequency and harmonics
- a clean and low level noise floor

The Demodulated spectrum shows

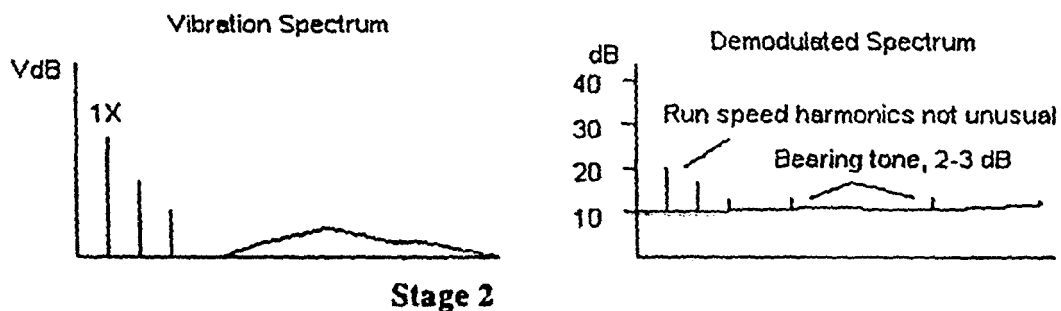
- a clean and low level noise floor
- a noise floor level which is taken as a reference level



**Stage 2** The first stage of bearing degradation has taken place, a tiny inner race flaw. The velocity spectrum shows no change. The Demodulated spectrum shows

- bearing defect frequency and possibly harmonics at 2 to 3 dB above the noise floor.
- rotational frequency and harmonics may appear at this stage depending upon the bearing inner clearances.
- the noise floor remains unchanged

**Bearing damage is minor and no action is justified other than to monitor closely.**

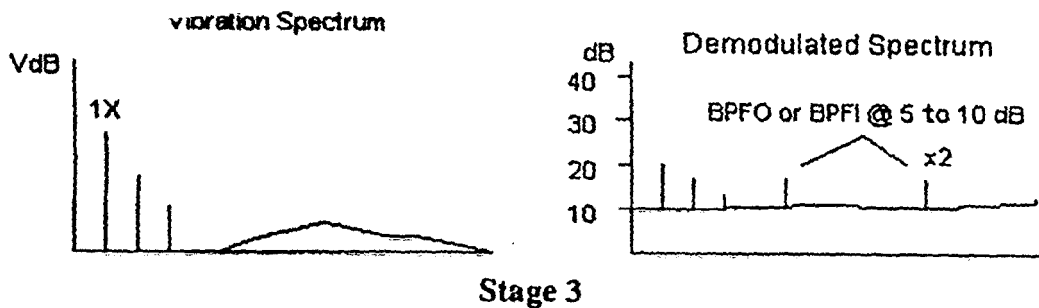


**Stage 3** Damage is now well established and evident.

The velocity spectrum has characteristics similar to Stages 3A & 3B of the Velocity scenario. The Demodulated spectrum shows

- an increase in the bearing defect frequencies to 5 to 10dB above the noise floor
- little or no change in the rotational frequency and harmonics.
- the noise floor remains unchanged

**The bearing is in poor condition but may still have significant service life left. Initiate planning replacement.**

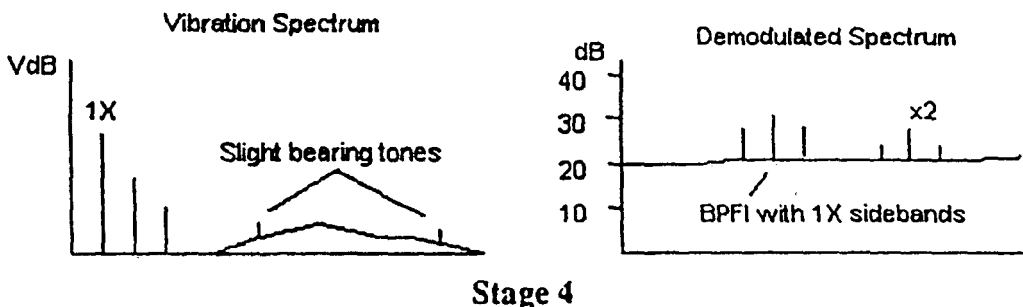


**Stage 4** Damage is now extensive and severe.

The Velocity spectrum has characteristics similar to those of Stages 3B and going into Stage 4 of the Velocity scenario. The Demodulated spectrum shows:

- an increasing noise floor rising up to 10 to 15dB above the reference level.
- the bearing defect frequency and its harmonics develop rotational frequency sidebands.
- As the noise floor rises the amplitude of the defect frequencies appear to decrease but then increase again to 10 to 15dB above the noise floor

**At this advanced stage there is insufficient bearing life left remaining to justify operating with the increased risk of failure. Replace at the next convenient opportunity, subject to the rate of defect development.**



### **4.3 Summary of Characteristics Identifying Failure Development**

- A “hump”, or cluster of peaks, at the higher frequency end of a 2kHz velocity spectrum,
- Harmonics of Bearing Defect Frequencies,
- Rotational frequency sidebands on Bearing Defect Frequencies, or other “odd” peaks,
- Lifting of the noise floor, mainly in the Demodulated spectrum, by 10dB or more,
- For low speed (<100rpm) look in the time domain for repetitive spikes of appropriate periodicity.

The procedures outlined above can provide a lot of information on the condition of a bearing and the nature of the fault and the rate at which it is developing.

However, the procedure for estimating the safe remaining service life is far from straightforward and requires a lot of judgement and experience. In the Workshop Session we will explore the methods and processes available to assist in making these quality judgements.

## **ESTIMATING REMAINING SAFE BEARING LIFE**

### **1. Introduction**

“When is my bearing going to fail?”, is the classic question asked by maintenance managers of their condition monitoring technician. This is one of more difficult problems addressed in condition monitoring, particularly as a bearing has numerous potential failure modes – inner race, outer race, cage, lubrication etc.

The simplest form of bearing measurement, trending the overall value of a bearing condition parameter, can be quite effective but is not sufficiently so where there is significant risk. To gain the higher level of confidence needed, more sophisticated diagnostic and signal processing techniques are required and, being relatively expensive to apply, are normally only used where the risk justifies it.

Having determined that a bearing is in a failure mode there may be pressure by production to keep it running “as long as possible”, or to keep going to await supply of replacement parts, or a scheduled maintenance window. In such a situation there are practices that may help. It is also useful to understand the factors that cause a bearing to deteriorate more quickly.

### **2. Deciding on when a Rolling Element Bearing Should be Replaced.**

The use of the bearing failure scenario does provide a better insight into answering that classic question, "When is the bearing going to fail ?" More correctly the question should be, "When should the bearing be replaced ?" This is not a question that the vibration analyst can answer alone. It is a collective decision with maintenance and production. Some of the questions that must be answered are;

- How critical is this machine ?
- What are the costs for it to be down ?
- How available are the replacement bearings and at what price ?
- Is the bearing condition affecting machine productivity or quality ?
- If not replaced now, what are the consequences of catastrophic failure ?
- Is there a backup for this machine in the event of premature failure ?
- Are there any maintenance or production windows soon ? Will it survive till then ?
- Are we quite confident that this bearing has a problem ?

### **3. Other Factors for Consideration**

- The urgency of any action is relative to the rate of deterioration. The longer a bearing has been in service the slower the rate of deterioration is likely to be. If the bearing is relatively new the rate is likely to be much faster.
- The higher the rotational speed of the bearing the faster the rate of deterioration.
- The higher the loading of the bearing the faster the rate of deterioration. Figure 4
- Tapered Spherical Roller Bearings will usually produce their best data when measured in the axial plane.
- Inadequacy of Lubricant, or contamination of it, remains the greatest cause of bearing failure. The Demodulated spectrum can give warning of this by showing an elevated noise floor with no discreet frequencies. Prompt attention to this has the potential to extend bearing life by maintaining good lubrication. If the problem was simply lubrication then, after regreasing, the

characteristics will disappear (including overall bearing condition readings) and not reappear after 24 hours.

- Mounting of the transducer is critical for repeatable and meaningful results – do not use stingers, particularly for Demodulated readings.
- Sliding bearings can be a major problem. Sliding is particularly prevalent where the bearing is underloaded or oversized for its duty and is seen more often with parallel cylindrical rollers. In the Demodulated spectrum it may show up as outer race defect frequency and harmonics with normal noise floor.
- The Demodulation technique is extremely sensitive and used alone, or without looking for the patterns of damage development, has been responsible for many bearings being removed before damage is evident. Figure 5.
- Temperature is not usually a helpful indicator of bearing condition. However, in an oil lubricated bearing a loss of oil supply will produce an almost immediate increase in temperature with little response to vibration or shock readings, until damage is established. See figure 6.

Figure 2 shows typical damage development as a function of L10 life since damage inception.

8: Development of fatigue damage on the inner ring raceway of an angular contact ball bearing. The periodic intervals between inspections from damage begin on, are given in percentage of the nominal life  $L_{10}$ .

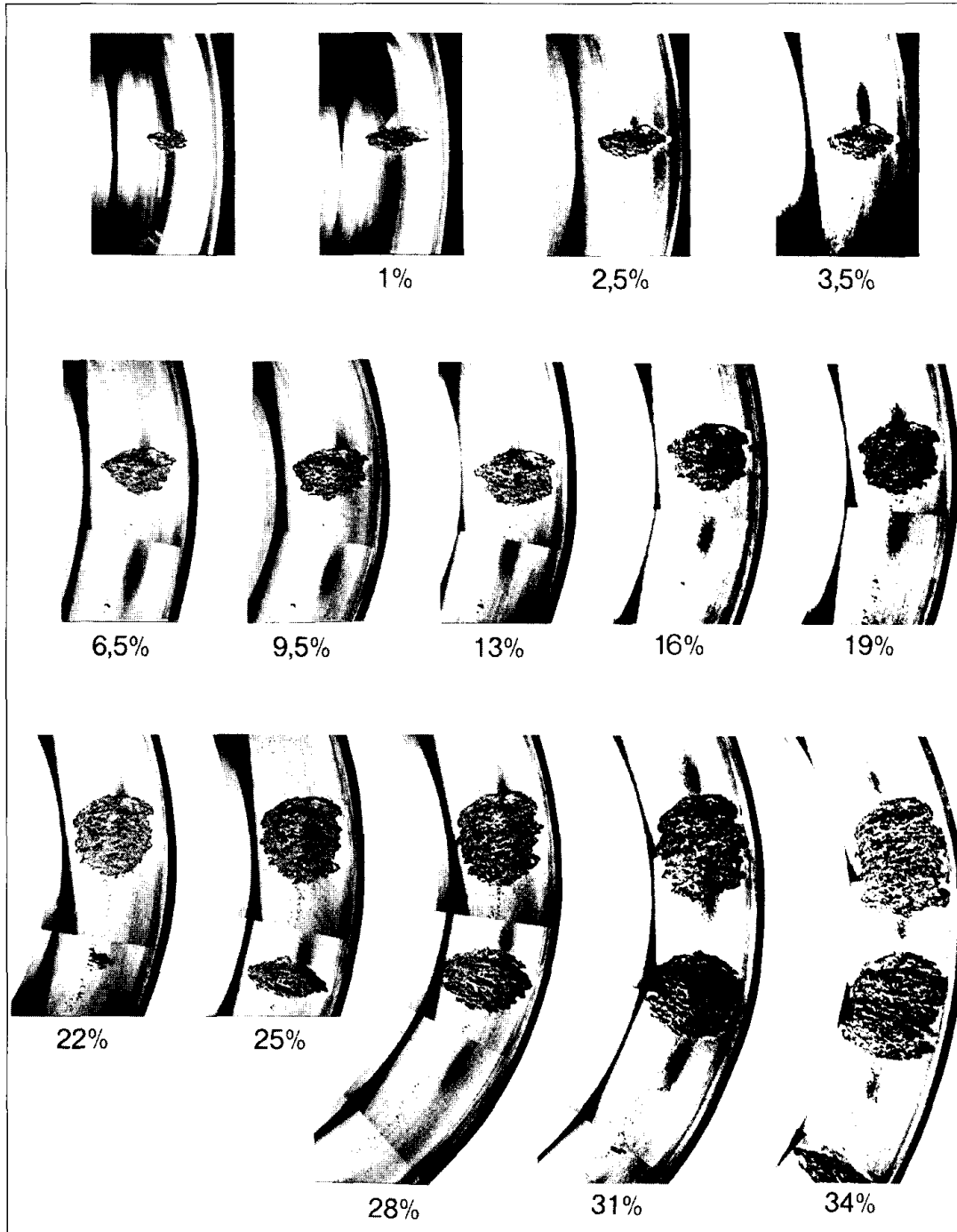


Figure 2