ME maintenance & asset management

An Overview of Bearing Vibration Analysis

Abstract

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Vibration produced by rolling bearings can be complex and can result from geometrical imperfections during the manufacturing process, defects on the rolling surfaces or geometrical errors in associated components. Noise and vibration is becoming more critical in all types of equipment since it is often perceived to be synonymous with quality and often used for predictive maintenance. In this article the different sources of bearing vibration are considered along with some of the characteristic defect frequencies that may be present. Some examples of how vibration analysis can be used to detect deterioration in machine condition are also given.

Keywords: education, maintenance engineering, reliability engineering, off-campus education, distance education, flexible learning, internet.

INTRODUCTION

Rolling contact bearings are used in almost every type of rotating machinery whose successful and reliable operation is very dependent on the type of bearing selected as well as the precision of all associated components, i.e. shaft, housing, spacers, nuts etc. Bearing engineers generally use fatigue as the normal failure mode, on the assumption that the bearings are properly installed, operated and maintained. Today, because of improvements in manufacturing technology and materials, it is generally the case that bearing fatigue life, which is related to sub-surface stresses, is not the limiting factor and probably accounts for less than 3% of failures in service.

Unfortunately though, many bearings fail prematurely in service because of contamination, poor lubrication, temperature extremes, poor fitting/fits, unbalance and misalignment. All these factors lead to an increase in bearing vibration and condition monitoring has been used for many years to detect degrading bearings before they catastrophically fail (with the associated costs of downtime or significant damage to other parts of the machine).

Rolling element bearings are often used in noise sensitive applications, e.g. household appliance electric motors which often use small to medium size bearings. Bearing vibration is therefore becoming increasingly important from both an environmental consideration and because it is synonymous with quality.

It is now generally accepted that quiet running is synonymous with the form and finish of the rolling contact surfaces. As a result, bearing manufacturers have developed vibration tests as an effective method for measuring quality. A common approach is to mount the bearing on a quiet running spindle and measure the radial velocity at a point on the bearing's outer ring and in three frequency bands, viz. 50-300, 300-1800 and

1800-10000 Hz. The bearing must meet RMS velocity limits in all three frequency bands.

Vibration monitoring has now become a well accepted part of many planned maintenance regimes and relies on the well known characteristic vibration signatures which rolling bearings exhibit as the rolling surfaces degrade. However, in most situations bearing vibration cannot be measured directly and so the bearing vibration signature is modified by the machine structure, this situation being further complicated by vibration from other equipment on the machine, i.e. electric motors, gears, belts, hydraulics, structural resonances etc. This often makes the interpretation of vibration data difficult other than by a trained specialist and can in some situations lead to a mis-diagnosis, resulting in unnecessary machine downtime and costs.

In this paper the sources of bearing vibration are discussed along with the characteristic vibration frequencies that are likely to be generated.

An Overview of **Bearing Vibration Analysis**

SOURCES OF VIBRATION

Rolling contact bearings represents a complex vibration system whose components – i.e. rolling elements, inner raceway, outer raceway and cage – interact to generate complex vibration signatures. Although rolling bearings are manufactured using high precision machine tools and under strict cleanliness and quality controls, like any other manufactured part they will have degrees of imperfection and generate vibration as the surfaces interact through a combination of rolling and sliding. Nowadays, although the amplitudes of surface imperfections are in the order of nanometres, significant vibrations can still be produced in the entire audible frequency range (20 Hz – 20 kHz).

The level of the vibration will depend upon many factors, including the energy of the impact, the point at which the vibration is measured and the construction of the bearing.

Variable compliance

Under radial and misaligning loads bearing vibration is an inherent feature of rolling bearings even if the bearing is geometrically perfect and is not therefore indicative of poor quality. This type of vibration is often referred to as variable compliance and occurs because the external load is supported by a discrete number of rolling elements whose position with respect to the line of action of the load continually changes with time (*see Figure 1*).

As the bearing rotates, individual ball loads, hence elastic deflections at the rolling element raceway contacts, change to produce relative movement between the inner and outer rings. The movement takes the form of a locus which under radial load is two dimensional and contained in a radial plane, whilst under misalignment it is three-dimensional. The movement is also periodic with base frequency equal to the rate at which the rolling elements pass through the load zone. Frequency analysis of the movement yields the base frequency and a series of harmonics. For a single row radial ball bearing with an inner ring speed of 1800 rev/min a typical ball pass rate is 100 Hz and significant harmonics to more than 500 Hz can be generated.

Variable compliance vibration is heavily dependent on the number of rolling elements supporting the externally applied load; the greater the number of loaded rolling elements, the less the vibration. For radially loaded or misaligned bearings 'running clearance' determines the extent of the load region, and hence, in general, variable compliance increases with clearance. Running clearance should not be confused with radial internal clearance (RIC), the former normally being lower than the RIC due to interference fit of the rings and differential thermal expansion of the inner and outer rings during operation.

Variable compliance vibration levels can be higher than those produced by roughness and waviness of the rolling surfaces. However, in applications where vibration is critical it can be reduced to a negligible level by using ball bearings with the correct level of axial pre-load.

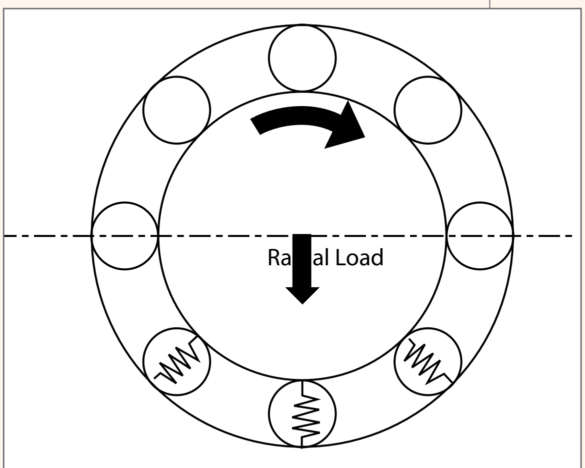


Figure 1 *Simple bearing model*

Geometrical imperfections

Because of the very nature of the manufacturing processes used to produce bearing components geometrical imperfections will always be present to varying degrees depending on the accuracy class of the bearing. For axially loaded ball bearings operating under moderate speeds the form and surface finish of the critical rolling surfaces are generally the largest source of noise and vibration. Controlling component waviness and surface finish during the manufacturing process is therefore critical since it may not only have a significant effect on vibration but also may affect bearing life.

It is convenient to consider geometrical imperfections in terms of wavelength compared with the width of the rolling element-raceway contacts. Surface features of wavelength of the order of the contact width or less are termed roughness, whereas longer wavelength features are termed waviness (see *Figure 2*).

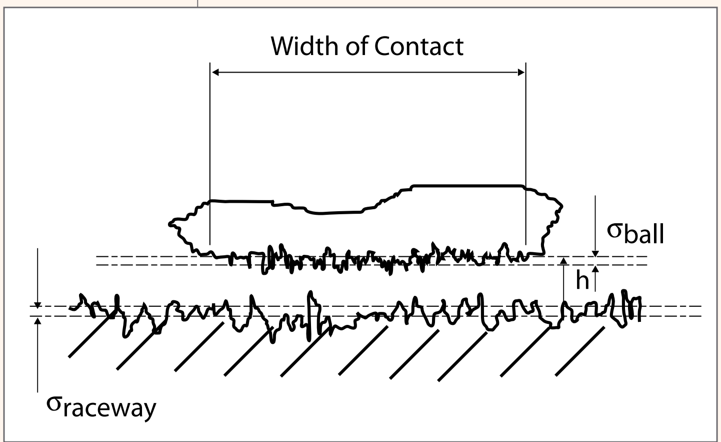


Figure 2 *Waviness and roughness of rolling surfaces*

SURFACE ROUGHNESS

Surface roughness is a significant source of vibration when its level is high compared with the lubricant film thickness generated between the rolling element-raceway contacts (see *Figure 2*). Under this condition surface asperities can break through the lubricant film and interact with the opposing surface, resulting in metal-to-metal contact. The resulting vibration consists of a random sequence of small impulses which excite all the natural modes of the bearing and supporting structure.

Surface roughness produces vibration predominantly at frequencies above sixty times the rotational speed of the bearing. Thus the high frequency part of the spectrum usually appears as a series of resonances.

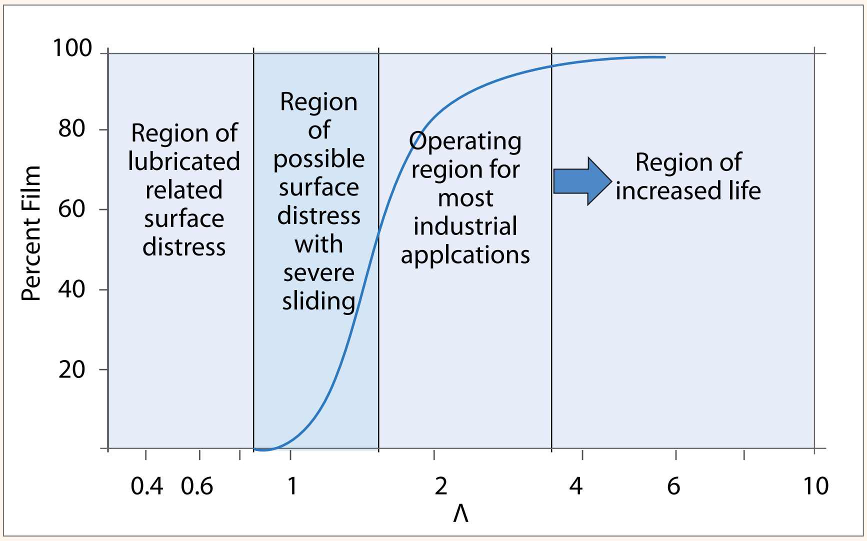


Figure 3 Percent film versus Λ (function of film thickness and surface roughness)

A common parameter used to estimate the degree of asperity interaction is the lambda ratio (Λ). This is the ratio of lubricant film thickness to composite surface roughness and is given by the expression

$$\Lambda = h\,(\sigma_b^2 + \sigma_r^2)^{0.5}$$

where Λ = degree of asperity interaction
h = the lubricant film thickness
σ_b = RMS roughness of the ball
σ_r = RMS roughness of the raceway

If we assume that the surface finish of the raceway is twice that of rolling element, then for a typical lubricant film thickness of 0.3μm surface finishes better than 0.06 μm are required to achieve a Λ value of three and a low incidence of asperity interaction. For a lubricant film thickness of 0.1_m surface finishes better than 0.025 _m are required to achieve Λ=3. The effect of Λ on bearing life is shown in *Figure 3*.

If Λ is less than unity it is unlikely that the bearing will attain its estimated design life because of surface distress, which can lead to a rapid fatigue failure of the rolling surfaces. In general, Λ ratios greater than three indicate complete surface separation. A transition from full EHL (elastohydrodynamic lubrication) to mixed lubrication (partial EHL film with some asperity contact) occurs in the Λ range between 1 and 3.

Waviness

For longer wavelength surface features, peak curvatures are low compared with that of the Hertzian contacts and rolling motion is continuous with the rolling elements following the surface contours. The relationship between surface geometry and vibration level is complex, being dependent upon the bearing and contact geometry as well as conditions of load and speed. Waviness can produce vibration at frequencies up to approximately three hundred times rotational speed but is usually predominant at frequencies below sixty times rotational speed. The upper limit is attributed to the finite area of the rolling element raceway contacts which average out the shorter wavelength features.

In the direction of rolling, elastic deformation at the contact attenuates simple harmonic waveforms over the contact width (see *Figure 4*).

The level of attenuation increases as wavelength decreases until, in the limit, for a wavelength equal to the contact width, waviness amplitude is theoretically zero. The contact length also attenuates short wavelength surface features. Generally poor correlation can exist between parallel surface height profiles taken at different points across the tracks and this averages measured waviness amplitudes to a low level. For typical bearing surfaces poor correlation of parallel surface heights profiles only exists at shorter wavelengths.

Even with modern precision machining technology waviness cannot be eliminated completely and an element of waviness will always exist albeit at relatively low levels. As well as the bearing itself, the quality of the associated components can also affect bearing vibration and any geometrical errors on the outside diameter of the shaft or bore of the housing can be reflected on the bearing raceways with the associated increase in vibration. Therefore, careful attention is required to the form and precision of all associated bearing components.

Discrete defects

Whereas surface roughness and waviness result directly from the bearing component manufacturing processes, discrete defects refer to damage of the rolling surfaces due to assembly, contamination, operation, mounting, poor maintenance etc. These defects can be extremely small and difficult to detect and yet can have a significant impact on vibration-critical equipment or can result in reduced bearing

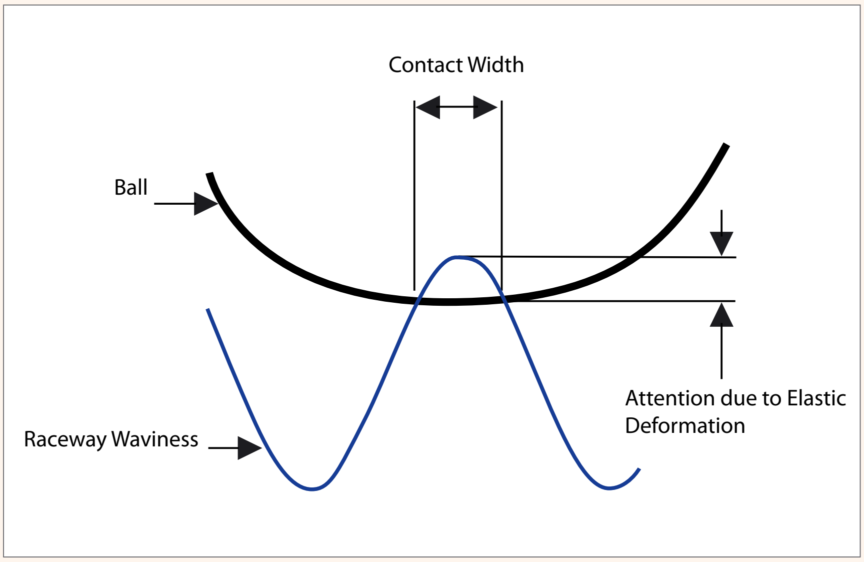


Figure 4 Attenuation due to contact width

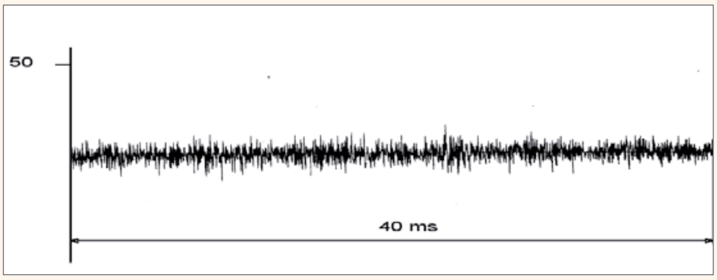


Figure 5(a) Signal from a good bearing

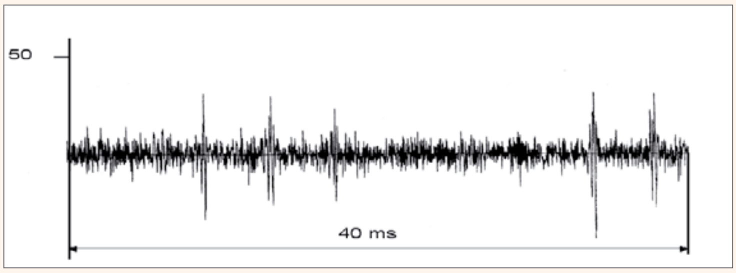


Figure 5(b) Signal from a damaged bearing

life. This type of defect can take a variety of forms, viz. indentations, scratches along and across the rolling surfaces, pits, debris and particles in the lubricant.

Bearing manufacturers have adopted simple vibration measurements on the finished product to detect such defects but these tend to be limited by the type and size of bearing. An example of this type of measurement is shown in *Figures 5(a)* and *5(b)* where, compared to a good bearing, the discrete damage on a bearing outer ring raceway has produced a characteristically impulsive vibration which has a high peak/RMS ratio.

Where a large number of defects occurs individual peaks are not so clearly defined but the RMS vibration level is several times greater than that normally associated with a bearing in good condition.

Bearing characteristic frequencies

Although the fundamental frequencies generated by rolling bearings are expressed by relatively simple formulas they cover a wide frequency range and can interact to give very complex signals. This is often further complicated by the presence on the equipment of other sources of mechanical, structural or electro-mechanical vibration.

For a stationary outer ring and rotating inner ring, the fundamental frequencies are derived from the bearing geometry as follows –

$$f_{c/o} = f_r/2\,[1 - d/D \cos \alpha\,]$$
$$f_{c/i} = f_r/2\,[1 + d/D \cos \alpha\,]$$
$$f_{b/o} = Z\,f_{c/o}$$
$$f_{b/i} = Z\,f_{c/i}$$
$$f_b = D/2d\,f_r\,[1 - (d/D \cos \alpha)^2]$$

where

f_r	=	inner ring rotational frequency
$f_{c/o}$	=	fundamental train (cage) frequency relative to outer ring
$f_{c/i}$	=	fundamental train frequency relative to inner ring
$f_{b/o}$	=	ball pass frequency of outer ring
$f_{b/i}$	=	ball pass frequency of inner ring
f_b	=	rolling element spin frequency
D	=	Pitch circle diameter
d	=	Diameter of roller elements
Z	=	Number of rolling elements
α	=	Contact angle

The bearing equations assume that there is no sliding and that the rolling elements roll over the raceway surfaces. However, in practice this is rarely the case and due to a number of factors the rolling elements undergo a combination of rolling and sliding. As a consequence, the actual characteristic defect frequencies may differ slightly from those predicted, but this is very dependent on the type of bearing, operating conditions and fits. Generally the bearing characteristic frequencies will not be integer multiples of the inner ring rotational frequency which helps to distinguish them from other sources of vibration.

Since most vibration frequencies are proportional to speed it is important when comparing vibration signatures that data is obtained at identical speeds. Speed changes will cause shifts in the frequency spectrum causing inaccuracies in both the amplitude and frequency measurement. Sometimes, in variable speed equipment spectral orders may be used where all the frequencies are normalized relative to the fundamental rotational speed. This is generally called '*order normalisation*', where the fundamental frequency of rotation is called the *first order*.

The bearing speed ratio (ball pass frequency divided by the shaft rotational frequency) is a function of the bearing loads and clearances and can therefore give some indication of the bearing operating performance. If the bearing speed ratio is below predicted values it may indicate insufficient loading, excessive lubrication or insufficient bearing radial internal clearance, which could result in higher operating temperatures and premature failure. Likewise, a higher than predicted bearing speed ratio may indicate excessive loading, excessive bearing radial internal clearance or insufficient lubrication.

A good example of how the bearing speed ratio can be used to identify a potential problem is shown in Figure 6, which shows a vibration acceleration spectrum measured axially on the end cap of a 250 kW electric motor.

In this case the Type 6217 radial ball bearings were experiencing a high axial load as a result of the non-locating bearing failing to slide in the housing (thermal

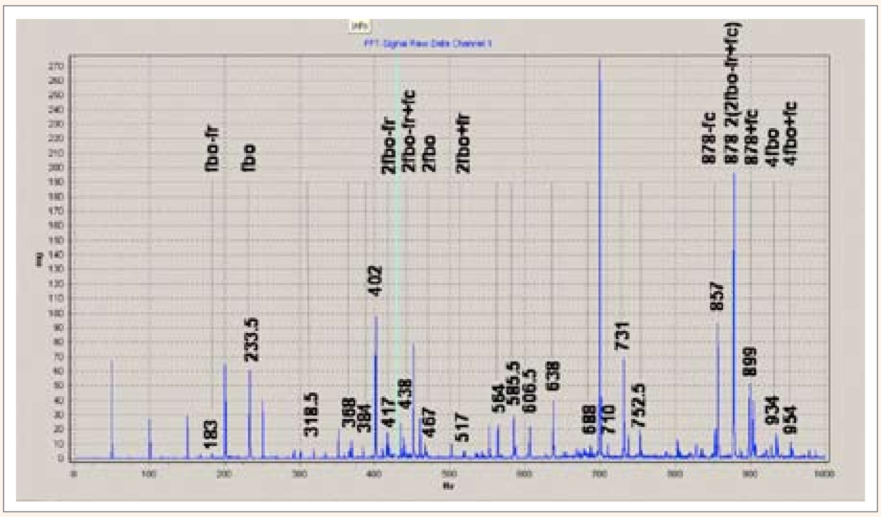


Figure 6 Axial vibration acceleration spectrum on end cap of a 250 kW electric motor

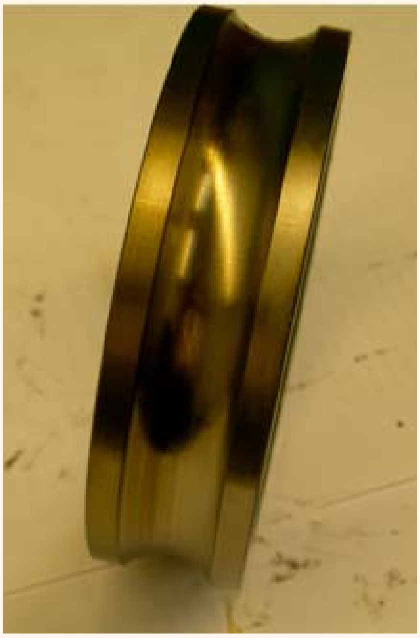


Figure 7 *Photograph of Type 6217 inner ring showing running path offset from centre of raceway*

loading). For a nominal shaft speed of 3000 rev/min the estimated outer ring ball pass frequency, $f_{b/o}$, was 228.8 Hz giving a bearing speed ratio of 4.576. The actual outer ring ball pass frequency was 233.5 Hz giving a ball speed ratio of 4.67, an increase of 2%. A photograph of the inner ring is shown in *Figure 7*, showing the ball running path offset from the centre of the raceway towards the shoulder.

Eventually this motor failed catastrophically and thermal loading (cross location) of the bearings was confirmed. A number of harmonics and sum and difference frequencies are also evident in the spectrum.

Ball pass frequencies can be generated as a result of elastic properties of the raceway materials due to variable compliance or as the rolling elements pass over a defect on the raceways. The frequency generated at the outer and inner ring raceway can be estimated roughly as 40% (0.4) and 60% (0.6) of the inner ring speed times the number of rolling elements respectively.

Unfortunately, bearing vibration signals are rarely straightforward and are further complicated by the interaction of the various component parts, but this can be often used to advantage in order to detect a deterioration or damage to the rolling surfaces.

Imperfections on the surface of raceways and rolling elements, as a result of the manufacturing process, interact to produce other discrete frequencies and sidebands (summarised in *Table 1*).

Surface Defect		Frequency
Component	**Imperfection**	
Inner Raceway	Eccentricity	f_r
	Waviness	$nZf_{c/i} \pm f_r$
	Discrete Defect	$nZf_{c/i} \pm fr$
Outer Raceway	Waviness	$nZf_{c/o}$
	Discrete Defect	$nZf_{c/}o \pm f_r$; $nZf_{c/o} \pm f_{c/o}$
Rolling Element	Diameter Variation	$Zf_{c/o}$
	Waviness	$2nfb \pm f_{c/o}$
	Discrete Defect	$2nfb \pm f_{c/o}$

Table 1 *Frequencies related to surface imperfections*

Analysis of bearing vibration signals is usually complex and the frequencies generated will add and subtract and are almost always present in bearing vibration spectra. This is particularly true where multiple defects are present. However, depending upon the dynamic range of the equipment, background noise levels and other sources of vibration bearing frequencies can be difficult to detect in the early stages of a defect. However, over the years a number of diagnostic algorithms have been developed to detect bearing faults by measuring the vibration signatures on the bearing housing. Usually, these methods take advantage of both the characteristic frequencies and the 'ringing frequencies' (i.e. natural frequencies) of the bearing (see later).

Raceway defect

A discrete defect on the inner raceway will generate a series of high energy pulses at a rate equal to the ball pass frequency relative to the inner raceway. Because the inner ring is rotating, the defect will enter and leave the load zone causing a variation in the rolling element-raceway contact force, hence deflections. While in the load zone the amplitudes of the pulses will be highest but then reduce as the defect leaves the load zone, resulting in a signal which is amplitude modulated at inner ring rotational frequency. In the frequency domain this not only gives rise to a discrete peak at the carrier frequency (ball pass frequency) but also a pair of sidebands spaced either side of the carrier frequency by an amount equal to the modulating frequency (inner ring rotational frequency) (see *Figure 8*).

Generally, as the level of amplitude modulation increases so will the sidebands. As the defect increases in size more sidebands are generated and at some point the ball pass frequency may no longer be generated, but instead a series of peaks will be generated spaced at the inner ring rotational frequency.

A discrete fault on the outer raceway will generate a series of high energy pulses at a rate equal to the ball pass frequency relative to the outer ring. Because the outer ring is stationary the amplitude of the pulse will remain theoretically the same and hence will appear as a single discrete peak within the frequency domain.

An unbalanced rotor will produce a rotating load, so as with an inner ring defect, the resulting vibration signal can be amplitude modulated at inner ring rotational frequency.

Likewise the ball pass frequency can also be modulated at the fundamental train frequency. If a rolling element has a defect it will enter and leave the load zone at the fundamental train frequency causing amplitude modulation and result in sidebands around the ball pass frequency. Amplitude modulation at the fundamental train frequency can also occur if the cage is located radially on the inner or outer ring.

Although defects on the inner and outer raceways tend to behave in a similar manner, for a given size defect the amplitude of the spectrum of an inner raceway defect is generally much less. The reasons for this might be that a defect on the inner ring raceway only comes into the load zone once per revolution and the signal must travel through more structural interfaces before reaching the transducer location, i.e. rolling element, across an oil film, through the outer ring and through the bearing housing, to the transducer position. The more difficult transmission path for an inner raceway fault probably explains why a fault on the outer raceway tends to be easier to detect.

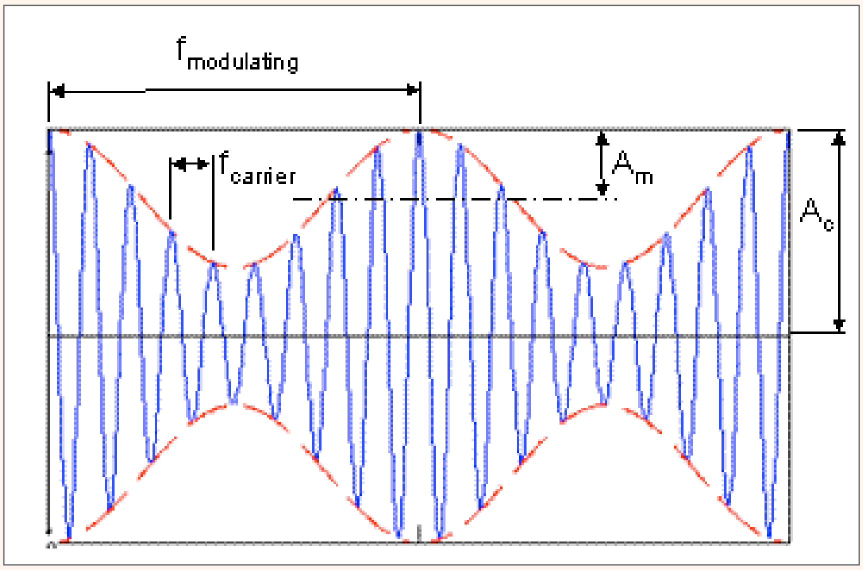


Figure 8 *Amplitude modulation (AM) (a) Amplitude modulated time signal*

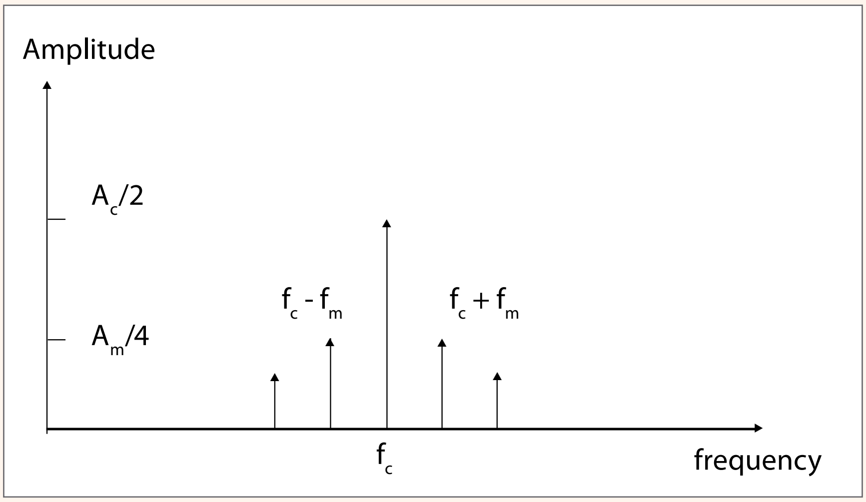


Figure 8 *Amplitude modulation (AM) (b) Spectrum of amplitude modulated signal*

Rolling element defect

Defects on the rolling elements can generate a frequency at twice ball spin frequency, and also harmonics and the fundamental train frequency. Twice the rolling element spin frequency can be generated when the defect strikes both raceways, but sometimes the frequency may not be this high because the ball is not always in the load zone when the defect strikes and energy is lost as the signal passes through other structural interfaces as it strikes the inner raceway. Also, when a defect on a ball is orientated in the axial direction it will not always contact the inner and outer raceway and therefore may be difficult to detect. When more than one rolling element is defective, sums of the ball spin frequency can be generated. If these defects are large enough then vibration at fundamental train frequency can be generated.

Cage defect

As already shown, the cage tends to rotate at typically 0.4 times inner ring speed, generally has a low mass and therefore, unless there is defect from the manufacturing process, is generally not visible.

Unlike raceway defects, cage failures do not usually excite specific ringing frequencies and this limits the effectiveness of the envelope spectrum (see later). In the case of cage failure the signature is likely to have random bursts of vibration as the balls slide and the cage starts to wear or deform and a wide band of frequencies is likely to occur.

As a cage starts to deteriorate, for example from inadequate lubrication, wear can start to occur on the sliding surfaces, i.e. in the cage pocket or in the case of a ring guided cage on the cage guiding surface. This may gives rise to a less stable rotation of the cage or a greater excursion of the rolling elements, resulting in increased sideband activity around the other bearing fundamental frequencies, e.g. the ball spin frequency.

Excessive clearance can cause vibration at the fundamental train frequency (FTF) as the rolling elements accelerate and decelerate through the load zone, which can result in large impact forces between the rolling elements and cage pockets. Also, outer race defects and roller defects can be modulated with the FTF fundamental frequency.

Other sources of vibration

Contamination is a very common source of bearing deterioration and premature failure and is due to the ingress of foreign particles, either as a result of poor handling or during operation. By its very nature the magnitude of the vibration caused by contamination will vary and in the early stages may be difficult to detect, but this depends very much on the type and nature of the contaminants. Contamination can cause wear and damage to the rolling contact surfaces and generate vibration across a broad frequency range. In the early stages the crest factor of the time signal will increase, but it is unlikely that this will be detected in the presence of other sources of vibration.

With grease lubricated bearings, vibration may be initially high as the bearing 'works' and distributes the grease. The vibration will generally be irregular but will disappear with running time and generally, for most applications, doesn't present a problem. For noise-critical applications special low-noise-producing greases are often used.

VIBRATION MEASUREMENT

Vibration measurement can be generally characterised as falling into one of three categories, viz. detection, diagnosis and prognosis. Detection generally uses the most basic form of vibration measurement, where the overall vibration level is measured on a broadband basis in a range, say, of 10 to 1000 Hz, or 10 to 10000 Hz. In machines where there is little vibration other than from the bearings, the spikiness of the vibration signal indicated by the Crest Factor (the ratio Peak/RMS) may imply incipient defects, whereas the high energy level given by the RMS level may indicate severe defects.

Generally, other than to the experienced operator this type of measurement gives limited information, but can be useful when used for trending, where an increasing vibration level is an indicator of a 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 taken out of service for repair. Another way of using the measurement is to compare the levels with published vibration criteria for different types of equipment.

Although broadband vibration measurements may provide a good starting point for fault detection it has limited diagnostic capability, and although a fault may be identified it may not give a reliable indication of where the fault is, i.e. bearing deterioration or damage, unbalance, misalignment etc. Where an improved diagnostic capability is required frequency analysis is normally employed, which usually gives a much earlier indication of the development of a fault and, secondly, the source of the fault.

Having detected and diagnosed a fault the prognosis – i.e. what the remaining useful life and possible failure mode of the machine or equipment are likely to be – is much more difficult and often relies on the continued monitoring of the fault to determine a suitable time when the equipment can be taken out of service, or relies on known experience with similar problems.

Generally, rolling bearings produce very little vibration when they are fault free and have distinctive characteristic frequencies when faults develop. A fault that begins as a single defect, e.g. a spall on the raceway, is normally dominated by impulsive events at the raceway pass frequency resulting in a narrow band frequency spectrum. As the damage worsens there is likely to be an increase in the characteristic defect frequencies and sidebands followed by a drop in these amplitudes and an increase in the broadband noise with considerable vibration at shaft rotational frequency. Where machine speeds are very low, the bearings generate low energy signals, which again may be difficult to detect. Also, bearings located within a gearbox can be difficult to monitor because of the high energy at the gear meshing frequencies, which can mask the bearing defect frequencies.

Overall vibration level

This is the simplest way of measuring vibration and usually consists of measuring the Root Mean Square (RMS) vibration of the bearing housing, or of some other point on the machine, with the transducer located as close to the bearing as possible. This technique involves measuring the vibration over a wide frequency range, e.g. 10-1000 Hz or 10-10000 Hz. The measurements can be trended over time and compared with known levels of vibration, or pre-alarm and alarm levels can be set to indicate a change in the machine condition. Alternatively, measurements can be compared with general standards. Although this method represents a quick and low cost method of vibration monitoring, it is less sensitive to incipient defects, i.e. it detects defects in the advanced condition and has a limited diagnostic capability. Also, it is easily influenced by other sources of vibration, e.g. unbalance, misalignment, looseness, electromagnetic vibration etc.

In some situations, the Crest Factor (the ratio Peak/RMS) of the vibration is capable of giving an earlier warning of bearing defects. The development of a local fault produces short bursts of high energy which increase the peak level of the vibration signal, but have little influence on the overall RMS level. As the fault progresses, more peaks will be generated until finally the Crest Factor will reduce but the RMS vibration will increase. The main disadvantage of this method is that in the early stages of a bearing defect the vibration is normally low compared with other sources of vibration present and is therefore easily influenced, so any changes in bearing condition may be difficult to detect.

Frequency spectrum

Frequency analysis plays an important part in the detection and diagnosis of machine faults. In the time domain the individual contributions, e.g. unbalance, to the overall machine vibration are difficult to identify. In the frequency domain they become much easier to identify and can therefore be much more easily related to individual sources of vibration. As we have already discussed, a fault developing in a bearing will show up as increasing vibration at frequencies related to the bearing characteristic frequencies, making detection possible at a much earlier stage than with overall vibration.

Envelope spectrum

When a bearing starts to deteriorate the resulting time signal often exhibits characteristic features which can be used to detect a fault. Also, bearing condition can rapidly progress from a very small defect to complete failure in a relatively short period of time, so early detection requires sensitivity to very small changes in the vibration signature. As we have already discussed, the vibration signal from the early stage of a defective bearing may be masked by machine noise making it difficult to detect the fault by spectrum analysis alone.

The main advantage of envelope analysis is its ability to extract the periodic impacts from the modulated random noise of a deteriorating rolling bearing. This is even possible when the signal from the rolling bearing is relatively low in energy and 'buried' within other vibration from the machine.

Like any other structure with mass and stiffness the bearing inner and outer rings have their own natural frequencies which are often in the kilohertz range. However, it is more likely that the natural frequency of the outer ring will be detected due to the small interference or clearance fit in the housing.

If we consider a fault on the outer ring: as the rolling element hits the fault the natural frequency of the ring will be excited and will result in a high frequency burst of energy which decays and then is excited again as the next rolling element hits the defect. In other words, the resulting time signal will contain a high frequency component amplitude-modulated at the ball pass frequency of the outer ring. In practice, this vibration will be very small and almost impossible to detect in a raw spectrum, so a method to enhance the signal is required.

By removing the low frequency components through a suitable high pass

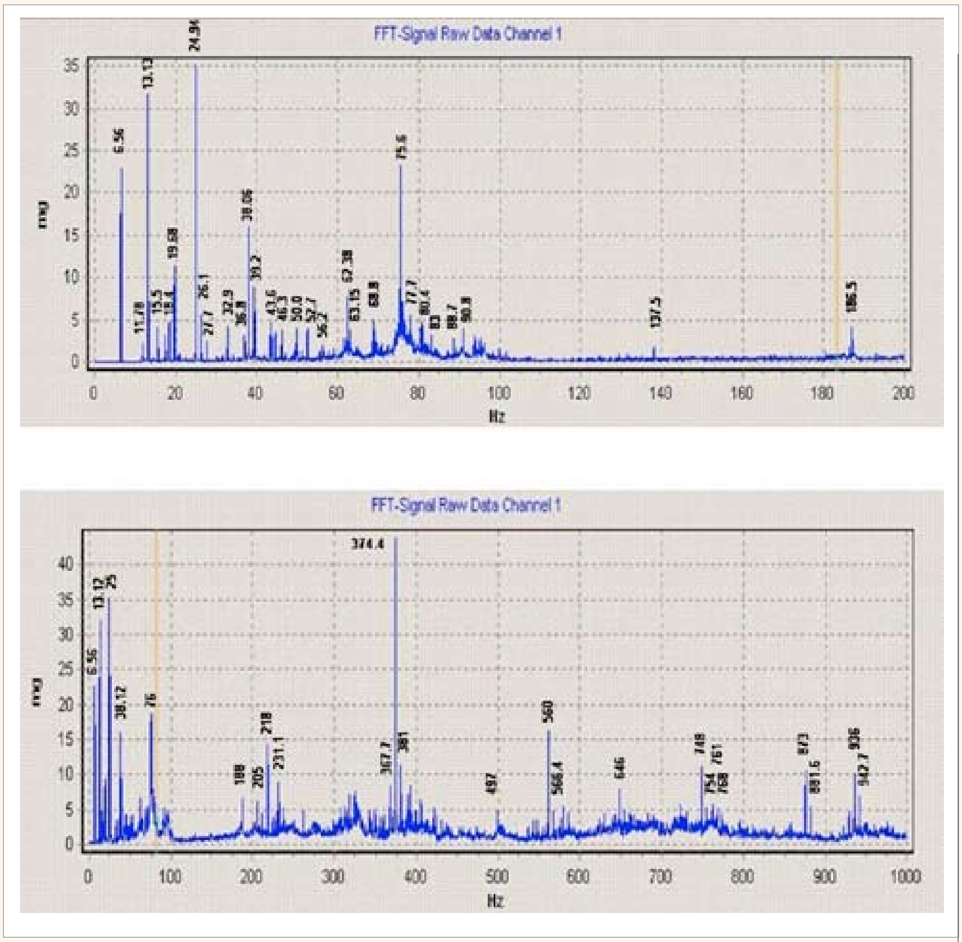


Figure 9 *Spectra obtained from the housing of a taper roller bearing*

filter, rectifying and then using a low pass filter the envelope of the signal is left, the frequency of which corresponds to the repetition rate of the defect. This technique is often used to detect early damage in rolling element bearings and is also often referred to as the High Frequency Resonance Technique (HFRT) or Envelope Spectrum.

Examples of vibration spectra

Roller deterioration

Figure 9 is an example of spectra obtained from a taper roller bearing with a 432 mm diameter bore rotating at 394 rev/min.

The shaft was gear driven with a drive shaft speed of 936 rev/min (2.375 reduction) giving a theoretical gear mesh frequency of 374.4 Hz. Vibration at shaft speed 6.56 Hz is clearly evident along with its harmonics. Also evident in the spectra is vibration at 62.4 Hz, which corresponds with twice the rotational frequency of the roller, plus a number of harmonics, i.e. 186.5 (x 3), 497 (× 8), 560 (× 9), 748 (× 12), 873 (× 14) and 936 Hz (× 15).

This would suggest some deterioration in the condition of the roller(s), which was confirmed upon examination of the bearing. The spectrum also shows discrete peaks spaced at cage speed, 2.93 Hz, which again is consistent with deterioration in the condition of the rollers. The 374.4 Hz component is related to the gear mesh frequency, with sidebands at rotational speed, 6.56 Hz.

As previously mentioned, bearing defects normally produce a signal which is amplitude modulated, so by demodulating the signal and analysing the envelope provides a useful technique for early fault detection. *Figure 10* shows the envelope spectrum, where discrete peaks are present at 62.5 Hz, and its harmonics which correspond with the roller defect frequency, clearly showing how demodulation can be used, in some circumstances, to provide a convenient and early detection of deterioration in rolling bearings.

Cage damage

The vibration spectrum shown in *Figure 11* was measured on the spindle housing of an internal grinding machine which was grinding the raceways of bearing outer rings. Although the machine was producing work to the required quality the routine vibration measurement immediately raised some concerns regarding the condition of the spindle.

The spindle was rotating at 19,200 rev/min (320 Hz) and the most unusual aspect of the spectrum is the presence of a large number of discrete peaks spaced at 140 Hz, which related to the fundamental train frequency of the angular contact ball bearings which had a plastic cage and were lubricated with oil mist.

Upon examination of the bearing the

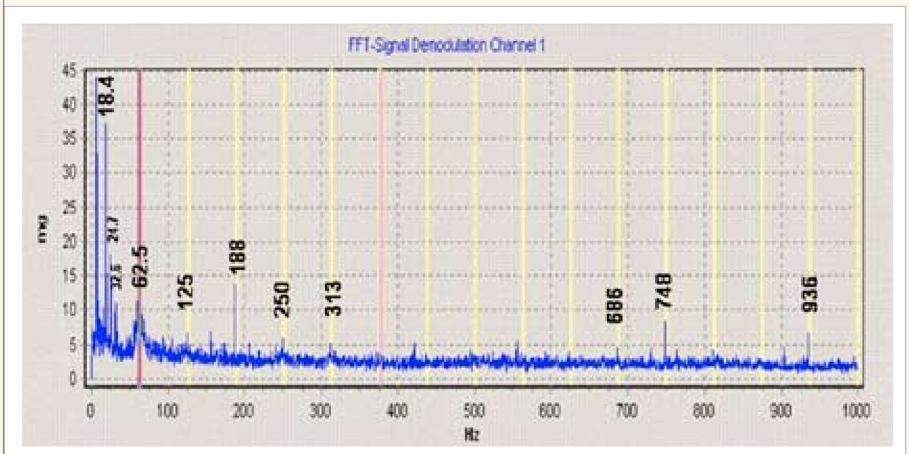


Figure 10 *Envelope spectrum from the housing of a taper roller bearing*

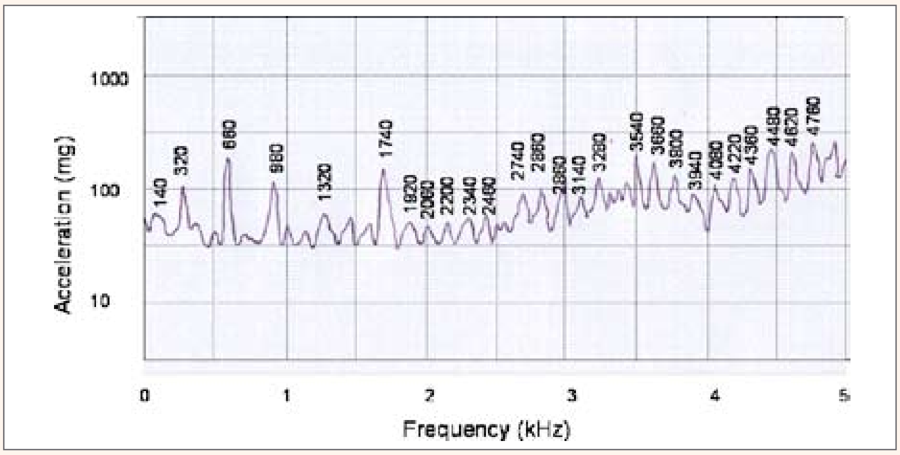


Figure 11 Vibration acceleration measured on the spindle housing of an internal grinding machine.

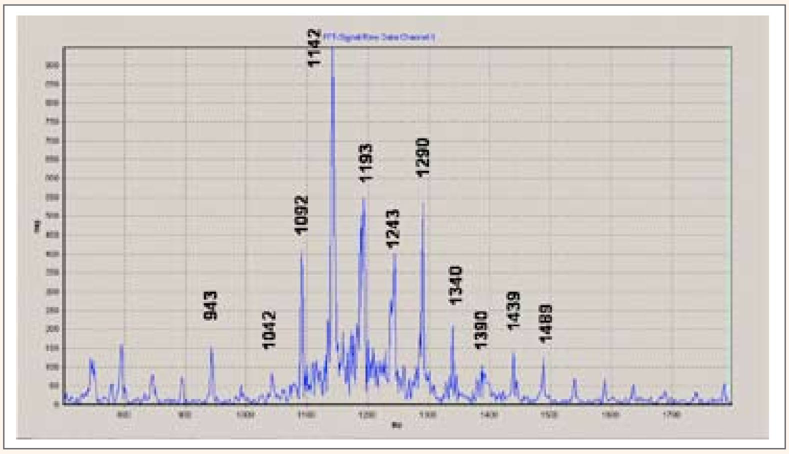


Figure 12 Vibration acceleration measured axially on the DE of a 250 kW electric motor.

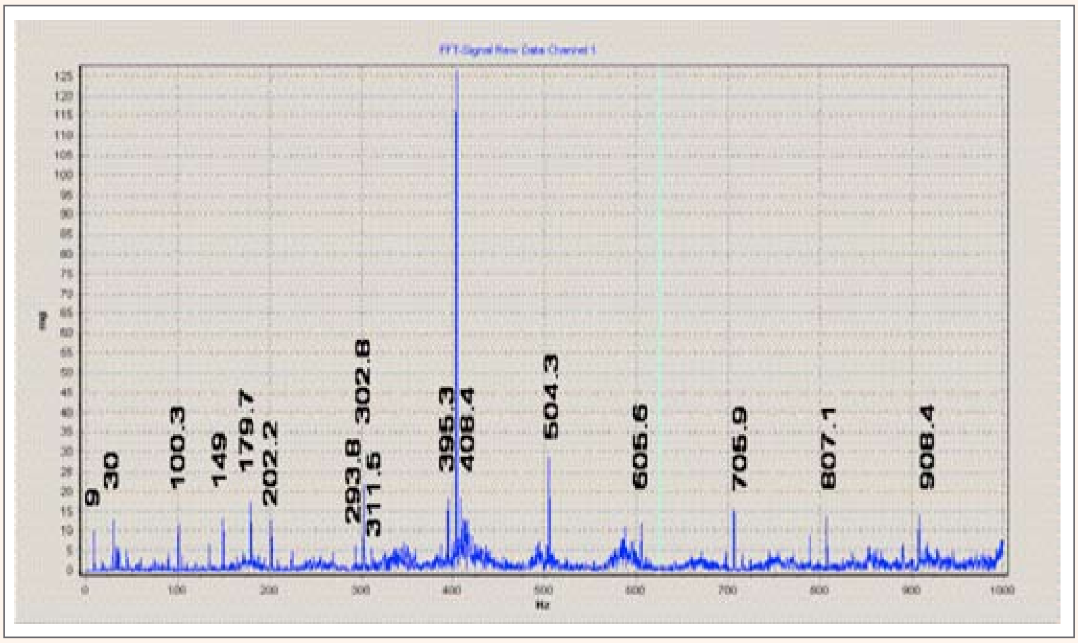


Figure 13 Vibration acceleration measured radially on the housing of a Type 23036 spherical roller bearing

cage outer diameter showed clear signs of damage with some fragments of plastic material which had broken away, but was still attached to the outer diameter. As a result, the spectrum had sum and difference frequencies related to the shaft (fr) and cage (fc), e.g. 1740 Hz (5fr+fc).

As already discussed, the deterioration of rolling element bearings will not necessarily show at the bearing characteristic frequencies, but the vibration signals are complex and produce sum and difference frequencies which are almost always present in the spectra.

Raceway damage

High axial load

An example of a vibration spectrum measured axially on the drive side end cap of a 250 kW electric motor is shown in *Figure 12*.

The rotational speed was approximately 3000 rev/min (50 Hz) and the rotor was supported by two type-6217- C4 (85 mm bore) radial ball bearings, grease lubricated. The vibration spectrum shows dominant peaks between 1 kHz and 1.5 kHz, which can be related to the outer raceway ball pass frequency. The calculated outer raceway ball pass frequency, $f_{b/o}$, is 229 Hz and the frequency of 1142 Hz relates to $5f_{b/o}$ with a number of sidebands at rotational frequency, fr.

When the bearings were removed from the motor and examined the ball running path was offset from the centre of the raceways towards the shoulders of the both the inner and outer rings, indicative of high axial loads. The cause of the failure was thermal pre-loading as a result of the non-locating bearing not sliding in the housing to compensate for axial thermal expansion of the shaft; this is often referred to as *'cross location'*. The non-drive end bearing had severe damage to the raceways and the rolling elements which was consistent with the highly modulated signal and high amplitude of vibration at $5f_{b/o}$. The overall RMS vibration level of the motor increased from typically 0.22g to 1.64g.

Another example of a vibration acceleration spectrum obtained from the housing of a Type 23036 (180 mm bore) spherical roller bearing, located on the main drive shaft of an impact crusher, is shown in *Figure 13*. The spectrum shows a number of harmonics of the outer raceway ball pass frequency, 101 Hz, with a dominant peak at 404 Hz ($4f_{b/o}$) with sidebands at shaft rotational frequency, 9 Hz.

When the bearing was removed from the machine and examined one part of the outer raceway had black corrosion stain (see *Figure 14*).

Figure 14 *Type 23036 spherical roller bearing outer ring raceway showing black corrosion stains*

Also a number of the rollers had black corrosion stains, which was consistent with the vibration at cage rotational frequency, fc=4 Hz, in the envelope spectrum (*see Figure 15*).

The modulation of the time signal at cage rotational frequency can be clearly seen in the time signal, *Figure 16*.

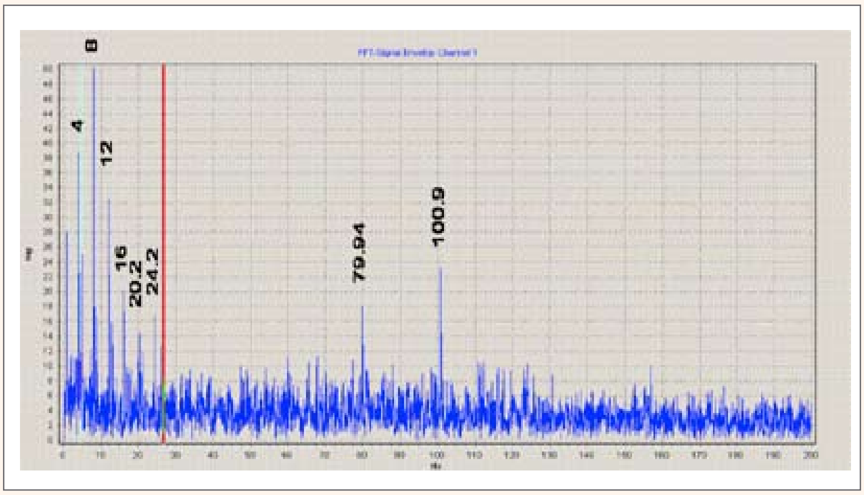


Figure 15 *Envelope spectrum of the Type 23036 spherical roller bearing*

Effect of bearing vibration on component quality

Even low levels of vibration can have a significant impact on critical equipment, such as machine tools that are required to produce components whose surface finish and form are critical. A good example of this is during the manufacture of bearing inner and outer rings. One of the most critical operations is grinding of the bearing raceways which have to meet very tight tolerances of roundness and surface finish, and any increase in machine vibration can result in a severe deterioration in workpiece quality.

Figure 17, which shows the vibration acceleration spectrum, 0-500 Hz, measured on the spindle housing of an external shoe centreless grinding machine during the grinding of an inner ring raceway, where the typical values for out-of-roundness and surface roughness were >4 μm and 0.3 μmRa respectively. The most distinctive feature on the finished raceway was the presence of 21 lobes which, when multiplied by the workpiece rotational speed (370 rev/min or 6.2 Hz), corresponded to a frequency of 129.5 Hz. This was very close to the 126 Hz component in the spectrum which was associated with the ball pass frequency relative to outer raceway of a ball bearing in the drive head motor. Also present are harmonics at 256 and 380 Hz. The discrete peaks at 38, 116 and 190 Hz correspond to the spindle rotational speed and its harmonics.

Figure 18 shows that after replacing the motor bearings the vibration at 126 Hz reduced from 0.012g to 0.00032g and the associated harmonics were no longer dominant. This resulted in a dramatic improvement in workpiece out-of-roundness of <0.4 μm and the surface finish improved to 0.19 μmRa. This demonstrates that with some critical equipment such as machine tools it is possible to assess directly the condition of the machine by measuring the resultant workpiece quality [2, 3].

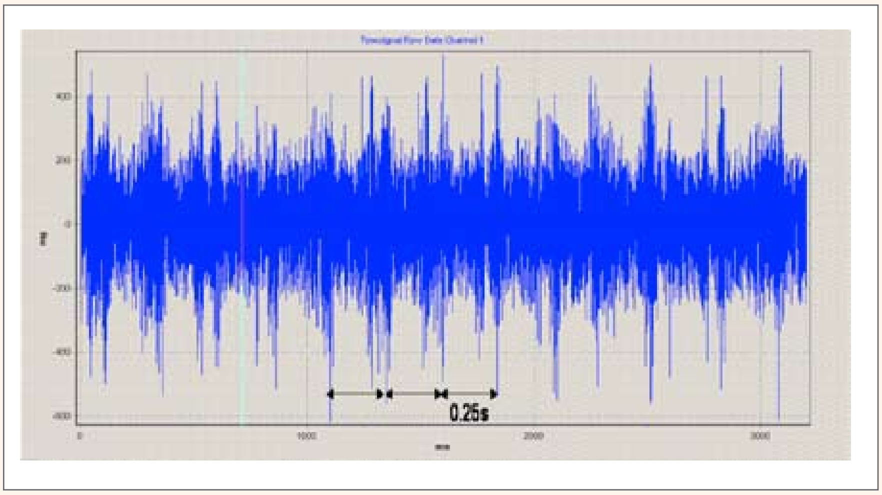
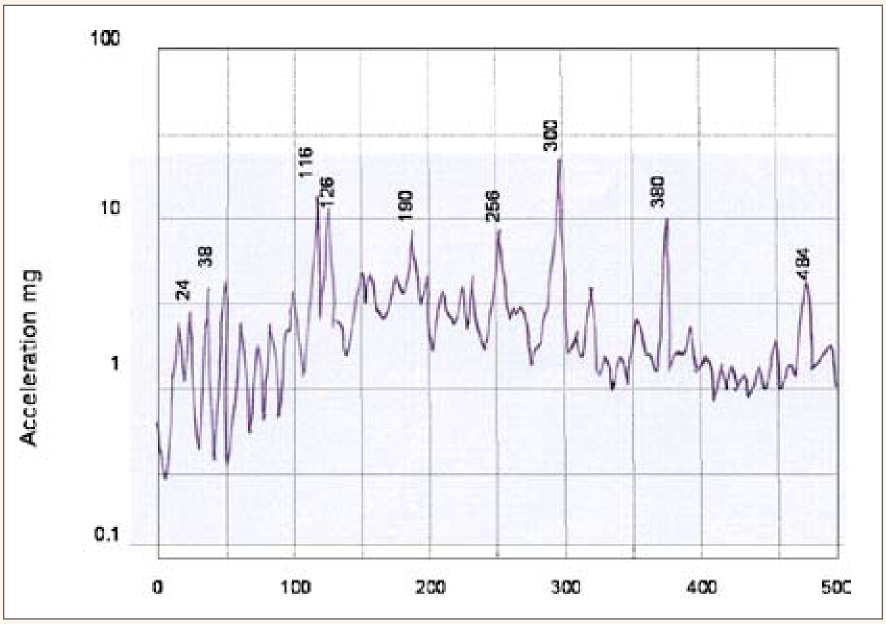


Figure 16 *Acceleration time signal of the Type 23036 spherical roller bearing*



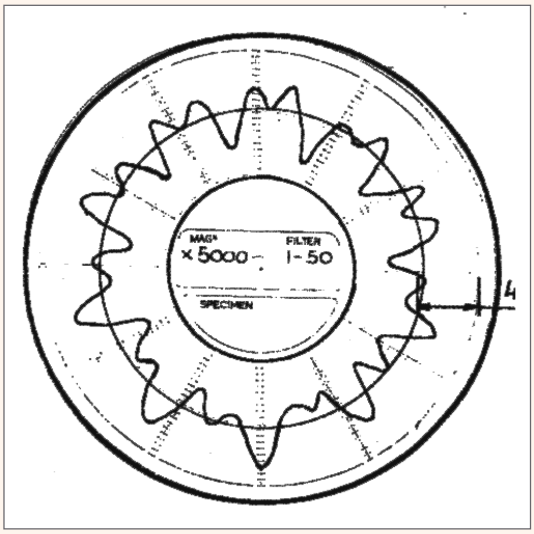
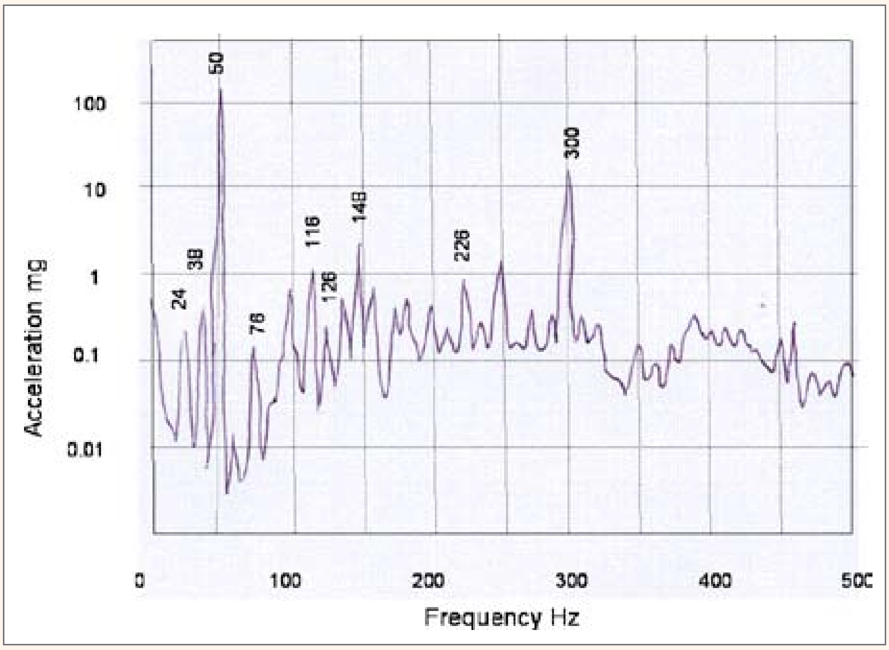


Figure 17 *Vibration spectrum and roundness before replacing wheel head drive motor bearings (a) Vibration spectrum on spindle housing (b) Roundness of raceway*



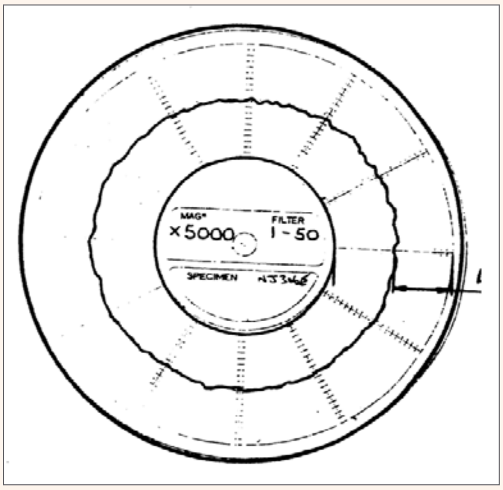


Figure 18 *Vibration spectrum and roundness after replacing wheel head drive motor bearings (a) Vibration spectrum measured on spindle housing (b) Roundness of raceway*

CONCLUSIONS

The various sources of bearing vibration have been discussed, also how each such source can generate characteristic vibration frequencies which can combine to give complex vibration spectra, which at times may be difficult to interpret other than by the experienced vibration analyst. However, with rolling bearings, characteristic vibration signatures are often generated, usually in the form of modulation of the fundamental bearing frequencies. This can be used to advantage, and vibration conditioning monitoring software is often designed to identify these characteristic features and provide early detection of an impending problem. This usually takes the form of signal de-modulation and establishment of the envelope spectrum, where the early indications of sideband activity, and hence bearing deterioration, can be more easily detected.

As long as there are natural frequencies of the bearing and its nearby structures – which occur in the case of a localized defect on the outer raceway, or on the inner raceway, or on a rolling element – the envelope spectrum works well. However, cage failures do not usually excite specific natural frequencies. The focus of demodulation is on the 'ringing' frequency (the carrier frequency) and the rate it is being excited (the modulating frequency).

Simple broad band vibration measurements also have their place, but offer a very limited diagnostic capability, and will generally not give an early warning of incipient damage or deterioration.

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