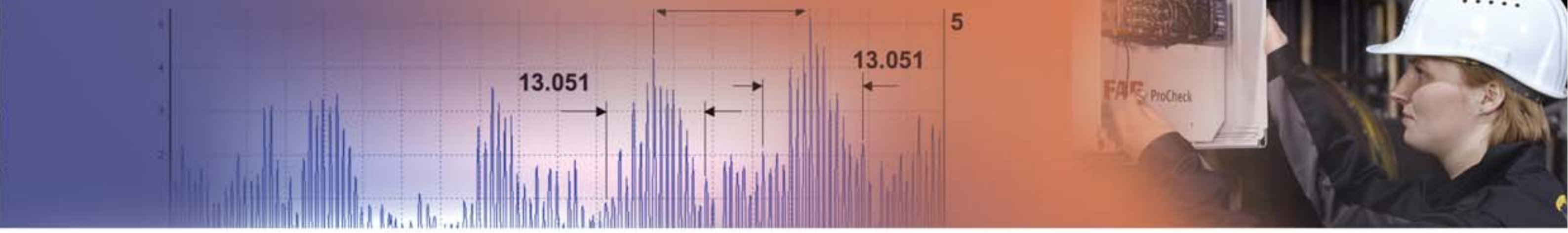

The Role of Vibration Monitoring in Predictive Maintenance

Dr. S. J. Lacey, Engineering Manager Schaeffler (UK) Limited

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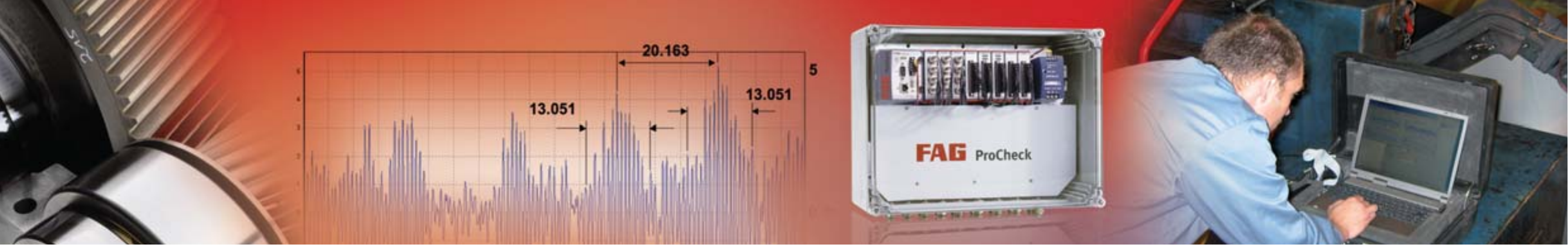

The Role of Vibration Monitoring in Predictive Maintenance

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Summary

Unexpected equipment failures can be expensive and potentially catastrophic, resulting in unplanned production downtime, costly replacement of parts and safety and environmental concerns.
Predictive Maintenance (PdM) is a process for monitoring equipment during operation in order to identify any deterioration, enabling maintenance to be planned and operational costs reduced. Rolling bearings are critical components used extensively in rotating equipment and, if they fail unexpectedly, can result in a catastrophic failure with associated high repair and replacement costs. Vibration based condition monitoring can be used to detect and diagnose machine faults and form the basis of a Predictive Maintenance strategy.

1. Introduction

As greater demands are placed on existing assets in terms of higher output or increased efficiency, the need to understand when things are starting to go wrong is becoming more important. Add to this the increasing complexity and automation of plant and equipment, it becomes more important to have a properly structured and funded maintenance strategy. There is also a need to understand the operation of equipment so that improvements in plant output and efficiency can be realised. In today's increasingly competitive world all of these issues are of key importance and can only be achieved through a properly structured and financed maintenance strategy that meets the business needs.

Maintenance can often be a casualty as businesses seek to save costs. How often have we heard the words "we have had no problems since the equipment was installed so we don't need condition monitoring". This is often borne out of ignorance and not undertaking a proper risk assessment to identify the criticality of existing assets so that the potential return on investment (ROI) of a properly funded maintenance strategy can be determined.

The need to run a plant at a higher efficiency yet often with fewer people puts increasing pressure on all concerned when equipment fails prematurely. When equipment does fail it is often at the most inconvenient time, either in the middle of a key process, at a weekend or in the middle of the night, when obtaining replacement parts may be difficult and labour costs are high due to overtime. While there is never a good time for equipment to fail, the technology available today means that there is simply no excuse for not taking the necessary steps to protect key assets. This can be achieved by minimising the risk of early and unexpected failures through a properly structured and funded maintenance strategy which will ultimately reduce overall operational costs.

The cost of not having a robust maintenance strategy should not be underestimated. It should not be looked at simply as an upfront cost, but viewed as an investment to safeguard and protect key assets, reducing the need for costly repairs and protecting the output of key processes. In some industries, maintenance is now the second largest or even the largest element of operating costs. As a result, it has moved from almost nowhere to the top of the league as a cost control priority in the last two or three decades.

The need to contain costs and run plant for longer more reliably means that there is a growing awareness of the need to prevent unnecessary equipment failures. Central to this is a maintenance strategy which is based on monitoring key assets to detect when things are starting to go wrong, enabling plant outage to be better planned in terms of resource availability, spare components, repairs etc. As a result, the risk of missing important contract deadlines is reduced and customer confidence is improved.

Until recently, many industries have and still do take the reactive approach to maintenance since this has no upfront costs but can result in many hours or days of plant downtime and/or lost production. While this may have been acceptable in the past, the increasing complexity and automation of equipment has meant this is not now a cost-effective option.

Having a clear and robust maintenance strategy fully supported by senior management is becoming more important, particularly in industries where it not only has a major impact on costs but also on the health and safety of employees and in situations where secondary damage and a catastrophic failure may result.

2. Maintenance Approach

Maintenance is traditionally performed in either time based fixed intervals as so-called preventive maintenance, or by corrective maintenance when a breakdown or fault actually occurs. In the latter, it is often necessary to perform the maintenance actions immediately, but in some cases this may be deferred depending on the criticality of the equipment. With predictive maintenance, an advanced warning is given of an impending problem and repairs are only carried out when necessary and can be planned to avoid major disruption.
A summary of all three approaches is given in Figure 1 and discussed briefly below:

Reactive Maintenance	Preventive Maintenance	Predictive Maintenance
DISADVANTAGES		
High risk of catastrophic failure or secondary damage. High repair & replacement costs Loss of key assets due to high downtime. Lost production & missed contract deadlines Inventory - high cost of spare parts or replacement equipment High labour cost – overtime, subcontracting. High cost due to hire of equipment Increased Health & Safety risks	High replacement costs - parts replaced too early Risk of early failure - infant mortality Human error during replacement of repaired or new parts Parts may often have many years of serviceable life remaining	High upfront costs including equipment & training
Environmental concerns		
ADVANTAGES		
No upfront costs, e.g. equipment, training. Seen as an easy option	Maintenance is planned and helps to prevent unplanned breakdowns Fewer catastrophic failures resulting in expensive secondary damage Greater control over inventory	Risk of unexpected breakdowns are reduced Equipment life is extended Reduced inventory & labour costs Maintenance can be planned and carried out when convenient Reduced risk of Health & Safety & environmental incidents Opportunity to understand why equipment has failed and improve efficiency

Figure 1. Comparison of different types of maintenance

2.1. Reactive Maintenance

Reactive maintenance of machinery, often referred to as the "run till failure" approach, involves fixing problems only after they occur. Of course, this is the simplest and cheapest approach in terms of upfront costs for maintenance, but often results in costly secondary damage along with high costs as a result of unplanned downtime and increased labour and parts costs. Since there are no upfront costs, it is often seen as an easy solution to many maintenance strategies – or there is no strategy at all.

In rotating equipment, rolling element bearings are one of the most critical components both in terms of their initial selection and, just as importantly, in how they are maintained.
Bearing manufacturers give detailed guidelines as to what maintenance is required and when which is often overlooked.

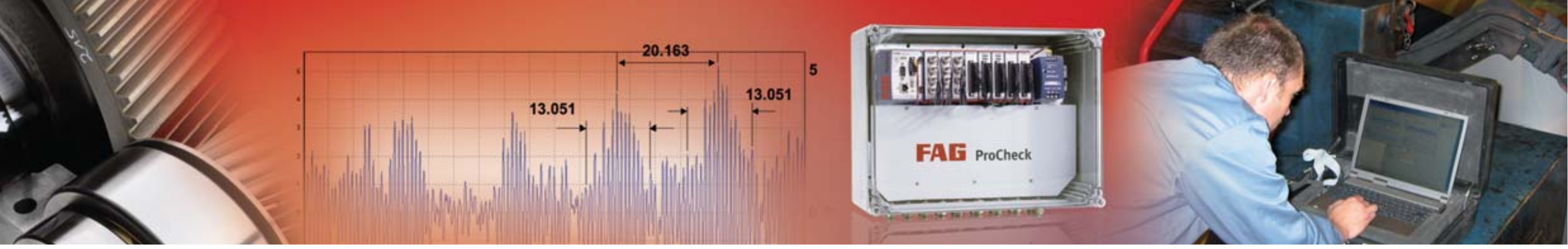
This can have disastrous consequences in terms of poor quality output, reduced plant efficiency or equipment failure. Monitoring the condition of rolling bearings is therefore essential and vibration based monitoring is more likely to detect the early onset of a fault.

2.2 Preventive Maintenance

With Preventive Maintenance (PM), machinery is overhauled on a regular basis regardless of the condition of the parts. This normally involves the scheduling of regular machine/plant shutdowns, whether or not they are required. The process may cut down failures before they happen but it also leads to increased maintenance costs as parts are replaced when this is not necessarily required.

There is also the risk of infant mortality due to human error during the time the asset is taken out of service for repair, adjustment, or installation of replacement parts. Other risks include installing a defective part, incorrectly installing or damaging a replacement part, or incorrectly reassembling parts.

A frequent and direct result of preventive maintenance is that much of the maintenance is carried out when there is nothing wrong in the first place. If the plant can be monitored in such a way as to obtain advance warning of a problem, significant costs savings can be obtained by avoiding unnecessary repair work. Such an approach is known as Predictive Maintenance.



2.3 Predictive Maintenance

Predictive Maintenance (PdM) is the process of monitoring the condition of machinery as it operates in order to predict which parts are likely to fail and when. In this way, maintenance can be planned and there is an opportunity to change only those parts that are showing signs of deterioration or damage.
The basic principle of predictive maintenance is to take measurements that allow for the prediction of which parts will break down and when. These measurements include machine vibration and plant operating data such as flow, temperature, or pressure.

Continuous monitoring detects the onset of component problems in advance, which means that maintenance is performed only when needed. With this type of approach, unplanned downtime is reduced or eliminated and the risk of catastrophic failure is mitigated. It allows parts to be ordered more effectively, thereby minimising inventory items, and manpower can be scheduled, thereby increasing efficiency and reducing the costs of overtime.

The main benefits of PdM are:

- Improved machine reliability through the effective prediction of equipment failures.
- Reduced maintenance costs by minimising downtime through the scheduling of repairs
- Increased production through greater machine availability
- Lower energy consumption
- Extended bearing service life
- Improved product quality

Rolling bearings are often a key element in many different types of plant and equipment spanning all market sectors. On one hand they can be of a standard design, readily available and low cost commodity items costing only a few pounds, such as those in electric motors, fans and gearboxes, while on the other hand they can be of bespoke design with long lead times and cost hundreds of thousands of pounds, as is the case in wind turbines, steelmaking plant etc.

However, they have one thing in common: if they fail unexpectedly, they can result in plant and equipment outage resulting in lost production costing from a few thousand to many millions of pounds. With a Predictive Maintenance strategy, such large costs can be avoided by giving advance warning of a potential problem, enabling remedial action to be planned and taken at a convenient time.

Replacing a bearing in a gearbox is preferable to replacing the whole gearbox, and replacing a motor bearing is better than having to send the motor to a rewinder to make expensive repairs and replace parts.

At the heart of many Predictive Maintenance strategies is Condition Monitoring which detects potential defects in critical components e.g. bearings, gears etc at the early stage thereby enabling the maintenance activity to be planned, saving both time and money and preventing secondary damage to equipment which can often be catastrophic.

3. Identifying Asset Criticality

Rolling bearings are used extensively in almost every type of rotating equipment whose successful and reliable operation is highly dependent on the bearing type, bearing fits and installation and maintenance requirements, such as relubrication.

When rolling bearings deteriorate it can result in expensive equipment failures with high associated costs. Unplanned downtime, the costly replacement of equipment, health and safety issues and environmental concerns are all potential consequences of a maintenance strategy that fails to monitor and predict equipment problems before they escalate into a more serious situation.

Assessing the criticality of an asset to the overall operation of the plant is therefore essential in terms of determining the type of condition monitoring required and whether it is necessary at all. In some cases where a plant has a large number of low cost assets, where replacements are readily available and/or are not deemed critical, a reactive or preventive approach may well be appropriate.

Even if an asset does warrant condition monitoring, a decision must be made not only on the technology but also on whether the asset warrants continuous (online system) or non continuous (patrol) monitoring. To help with this decision, assets are often assigned to one of three categories depending on their criticality, Figure 2[1].

Category	Description	Economics
A	Equipment assets having a large impact on plant output; equipment that represents significant repair costs; equipment with significant health & safety impacts. Failures can occur very suddenly and do not always give advance warning.	Failures are very expensive due to lost production, health & safety impacts or environmental impact. Examples include large horsepower, high energy density machines with very high replacement and maintenance costs. Financial justification: prevention of lost production, reduced maintenance costs, protection of life and environment
B	Equipment assets having a lesser impact on plant output; equipment with moderate repair costs; equipment that can have health & safety implications if failure occurs. Failures can occur relatively quickly, but usually with some advance warning	Similar to economics of critical equipment assets, but of smaller magnitude. Typical examples include medium horsepower machines with moderate replacement and maintenance costs
C	Equipment assets having little or no direct impact on plant output; equipment that represents limited repair costs; equipment that has minor safety ramifications	Typically include smaller assets with small individual replacement & repair costs & little or no costs related to lost production. However, they collectively comprise a large percentage of annual maintenance costs. Small individual repair & replacement costs. Costs of failure do not exceed costs to monitor (or excessive payback periods)

Figure 2. Asset categorisation

Category A assets are deemed to be critical and generally fulfil one or all of the following criteria:

- Failure results in total or major interruption of the process
- Failure represents a significant safety risk, such as fire, toxic leak, or explosion
- Long lead times and/or significant repair costs.

A good example of this type of asset would be the main turbine-generator trains in a large power plant. For such assets, it is the cost of failure that is of primary concern. Other examples would be the main rotor bearing or gearbox bearings in a wind turbine. Due to the generally remote location, failure of the main rotor bearing or gearbox bearings, which may lead, to secondary damage makes replacement costly in terms of replacement parts, hire of equipment and labour.

Category B assets are, on the other hand, essential assets and include, for example, pumps or compressors where, in the event of a fault, a standby unit is available; this may then become a category A asset.

Category C or non-essential assets are at the far end of the spectrum and the reasons for monitoring these, if indeed they are monitored at all, might be to prevent failure by eliminating root cause and allow more effective maintenance planning.

4. Return on Investment (ROI)

As already discussed, equipment failure can be expensive and potentially catastrophic resulting in unplanned downtime, missed customer schedules, costly machine replacements/repairs as well as safety and environmental concerns. By initiating a Predictive Maintenance strategy, unforeseen failures are minimised and this can yield an impressive ROI. Another major benefit of introducing CM as part of a Predictive Maintenance program is that it enables a greater understanding of the equipment critical to the process and also allows more time to be spent on improving the overall condition of the assets and improving the efficiency of key processes.

When justifying PdM, the following should be taken into consideration:

(1) Direct Costs
Labour
- Normal and overtime labour for
  - planned repair activities
  - unplanned repairs

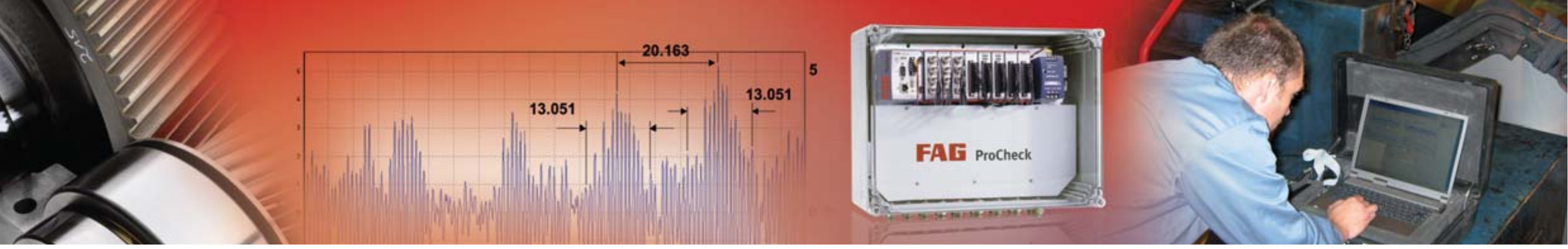
Materials
- Parts replaced
- Machinery replaced

(2) Indirect Costs
- Lost production (£ per hour)
- Outside services
- Insurance costs
- Parts inventory

Total Potential Cost Reduction (1+2)

(3) PdM Program Costs
- Site survey
- Cost of capital equipment
- Cost of any additional labour
- Cost of training
- Initial setup and baseline
- Scheduled data collection (visits per year at £ per visit)

Total PdM Costs for One Year (3)



By contracting out PdM, there will be no capital equipment and training costs and the benefits tend to be more immediate because of the use of highly trained staff. However, it is often more beneficial to keep the activities in-house, which allows greater familiarity with plant, equipment and processes and gives the benefits not only of preventing unplanned downtime but also enables more time to be spent on mitigating potential failure modes and improving process efficiency.

The cement industry is a good example where CM has been implemented and saved money both in terms of repair costs and lost production. In one case, the failure of a large gearbox caused a three week shutdown and extensive repair costs are typically €50,000 to €100,000. To prevent such damage, F'IS (FAG Industrial Services, Schaeffler Group) installed an eight channel FAG DTECT X1 system and trained the customer's staff who received three months' support at a total cost of €18,000. Detecting deterioration of the gearbox early resulted in a repair cost of €5000, saving the customer at least €27,000. More importantly, the company avoided lost production amounting to around €6000/hour.

5. Condition Monitoring

Condition monitoring is a process where the condition of equipment is monitored for early signs of deterioration so that the maintenance activity can be better planned, reducing down time and costs. This is particularly important in continuous process plants, where failure and downtime can be extremely costly.

The monitoring of vibration, temperature, voltage or current and oil analysis are probably the most common. Vibration is the most widely used and not only has the ability to detect and diagnose problems but potentially give a prognosis i.e. the remaining useful life and possible failure mode of the machine. However, prognosis 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.

5.1. Vibration Monitoring

Vibration monitoring is probably the most widely used predictive maintenance technique and, with few exceptions, can be applied to a wide variety of rotating equipment. Since the mass of the rolling elements is generally small compared to that of the machine, the velocities generated are generally small and result in even smaller movements of the bearing housing, making it difficult for the vibration sensor to detect.

Machine vibration comes from many sources e.g. bearings, gears, unbalance etc and even small amplitudes can have a severe effect on the overall machine vibration depending on the transfer function, damping and resonances, Figure 3. Each source of vibration will have its own characteristic frequencies and can manifest itself as a discrete frequency or as a sum and/or difference frequency.

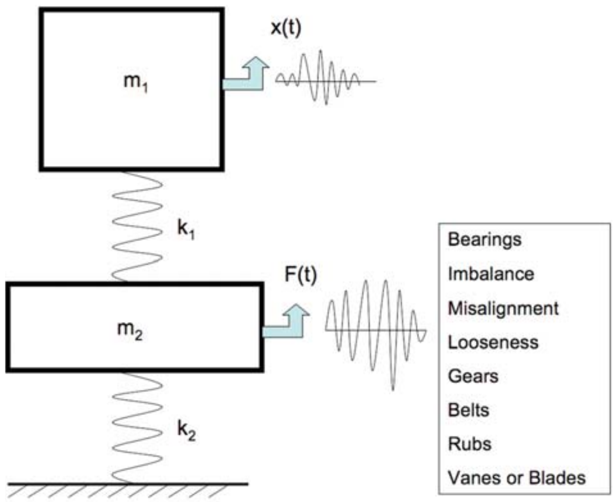


Figure 3. Simple machine model

At low speeds, it is still possible to use vibration but a greater degree of care and experience is required and other techniques such as measuring shaft displacement or Acoustic Emission (AE) may yield more meaningful results although the former is not always easy to apply. Furthermore, AE may detect a change in condition but has limited diagnostic capability. Vibration is used successfully on wind turbines where the main rotor speed is typically between 5 and 30 rpm. In a wind turbine, there are two main groups of vibration frequencies generated - gear mesh and bearing defect frequencies. This can result in complex vibration signals, which can make frequency analysis a formidable task. However, techniques such as enveloping (see section 5.1.3), which has a high sensitivity to faults that cause impacting, can help reduce the complexity of the analysis. Bearing defects can excite higher frequencies, which can be used as a basis for detecting incipient damage.

Vibration measurement can generally be characterised as falling into one of three categories – 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, for example, of 10-1000Hz or 10-10000Hz. In machines where there is little vibration other than from the bearings, the spikiness of the vibration signal indicated by the Crest Factor (peak/RMS) may imply incipient defects, whereas the high energy level given by the RMS level may indicate severe defects.

This type of measurement generally gives limited information (other than to an experienced operator) but can be useful for trending, where 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 taken out of service for repair or at least a more in depth survey must be performed.

Another way of using the measurement is to compare the levels with published vibration criteria for different types of equipment.

Although broadband vibration measurement may provide a good starting point for fault detection, it has limited diagnostic capability and, while a fault may be identified, it may not give a reliable indication of where the fault lies, for example in bearing deterioration/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 also its source.

Having detected and diagnosed a fault, it is much more difficult to give a prognosis on the remaining useful life and possible failure mode of the machine or equipment. This often relies on continued monitoring of the fault, to determine a suitable time when the equipment can be taken out of service, and/or on experience with similar problems.

In general, rolling bearings produce very little vibration when they are free of faults and have distinctive characteristic frequencies when faults develop. A fault that begins as a single defect, such as a spall on a raceway, is normally dominated by impulsive events at the raceway pass frequency, resulting in a narrow band frequency spectrum. As the damage increases, 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 may also be difficult to detect. Furthermore, 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.

5.1.1 Overall Vibration Level

This is the simplest way of measuring vibration and usually involves measuring the RMS (Root Mean Square) vibration of the bearing housing or some other point on the machine with the transducer located as close to the bearing as possible. The vibration is measured over a wide frequency range, such as 10-1000Hz or 10-10000Hz.

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 is only really suitable for detecting defects in the advanced condition and has limited diagnostic capability.

It is also easily influenced by other sources of vibration, such as unbalance, misalignment, looseness, electromagnetic vibration etc.

In some situations, the Crest Factor (Peak-to-RMS ratio) of the vibration is capable of giving an earlier warning of bearing defects.

As a local fault develops, this 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 decreases but the RMS vibration increases.

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 are difficult to detect.

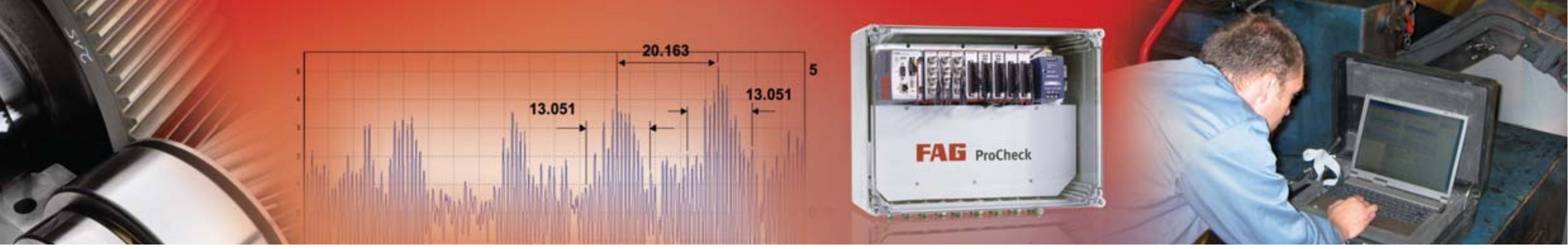
5.1.2 Frequency Spectrum

Frequency analysis plays an important part in the detection and diagnosis of machine faults. In the time domain, the individual contributions such as unbalance, bearings, gears etc 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.

It is not always possible to rely on the amplitude of bearing discrete frequencies to provide defect severity because each machine will have different mass, stiffness and damping properties. Even identical machines can have different system properties and this can affect the amplitudes of bearing defects of similar size.

It is often the pattern of the bearing defect frequencies that is most significant in determining the defect severity.
The number of bearing related harmonic frequencies, frequency sidebands and characteristic features within the time waveform data can be much more reliable than amplitude alone as a method of determining when action needs to be taken.

As 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.



5.1.3 Envelope Spectrum

When a bearing starts to deteriorate, the resulting time signal often exhibits characteristic features that can be used to detect a fault. Furthermore, 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 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 there is a fault on the outer ring, the natural frequency of the ring may be excited as the rolling element hits the fault and this will result in a high frequency burst of energy which decays and is then 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/roller pass frequency of the outer raceway. In practice this vibration will be very small and almost impossible to detect in a base spectrum, so a method of enhancing the signal is required.

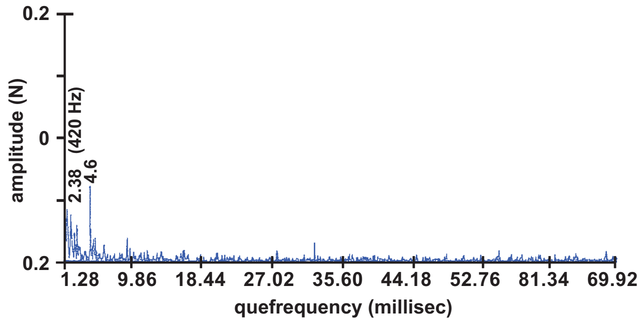
By removing the low frequency components through a suitable high pass filter, rectifying the output and then using a low pass filter, this leaves the envelope of the signal whose frequency corresponds to the repetition rate of the defect. This technique is often used to detect early damage in rolling element bearings and is also often referred to as the High Frequency Resonance Technique (HFRT) or Envelope Spectrum.

5.1.4 Cepstrum Analysis

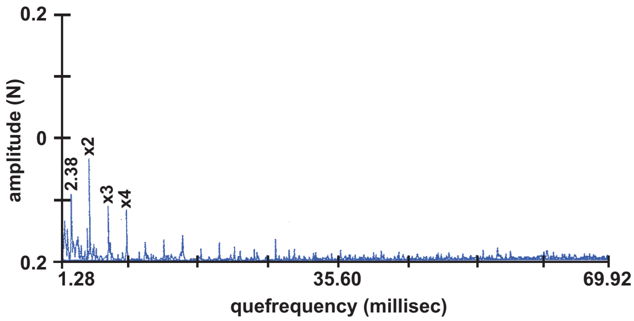
Vibration spectra from rotating machines are often very complex, containing several sets of harmonics and also sidebands as a result of various modulations. When trying to identify and diagnose possible machine faults, a number of characteristics of the vibration signal are considered, including harmonic relationships and the presence of sidebands. Cepstrum analysis can simplify this because single discrete peaks in the cepstrum represent the spacing of harmonics and sidebands in the spectrum i.e. the cepstrum identifies periodicity within the spectrum. Cepstrum analysis converts the spectrum back into the time domain i.e. it plots amplitude versus time (quefrency) and harmonics are known as rhamonics.

Vibration monitoring can also be used to gain valuable information about the condition of machining processes. In the manufacture of rolling bearings, grinding of the raceways is a critical process in terms of achieving a high surface finish and roundness, essential to achieving the required service life.

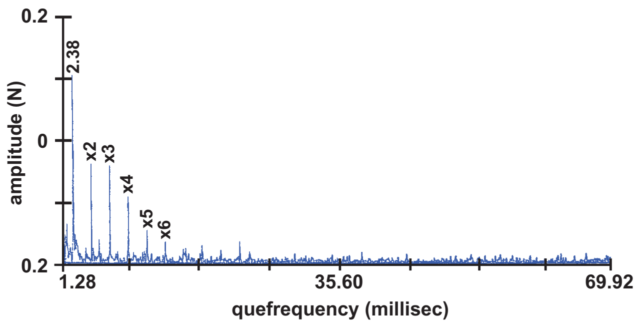
Figure 4 shows cepstra of shoe force obtained during the shoe centreless grinding of bearing outer ring raceways[2].



(a) Diamond infeed 0,01mm/rev



(b) Diamond infeed 0,064mm/rev



(c) Diamond infeed 0,125mm/rev

Figure 4. Cepstra of top shoe force during the internal grinding of bearing outer ring raceways

In this case, as the severity of the dressing process increases i.e. increasing diamond infeed, the amplitude of the first peak at 2.38ms increases along with the number of rhamonics. The quefrency of 2.38ms corresponds to the wheel rotational frequency of 420Hz. This is because, as the severity of the dressing operation increases, it has a significant effect on wheel form, hence workpiece quality, and the vibration signal becomes more highly modulated at wheel rotational speed.

6. Rolling Element Bearings

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 e.g. 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. Thanks to improvements in manufacturing technology and materials, bearing fatigue life, which is related to sub surface stresses, is generally no longer the limiting factor and probably accounts for less than 3% of failures in service.

Unfortunately, many bearings fail prematurely in service due to contamination, poor lubrication, misalignment, 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 of small to medium size are often used in electric motors for noise sensitive applications e.g. household appliances. Bearing vibration is therefore becoming increasingly important from both an environmental perspective and because it is synonymous with quality.

Vibration monitoring has now become a well accepted part of many Predictive Maintenance regimes and relies on the well known characteristic vibration signatures which rolling bearings exhibit as the rolling surfaces degrade. In most situations, however, bearing vibration cannot be measured directly and the bearing vibration signature is modified by the machine structure. This situation is further complicated by vibration from other equipment on the machine, such as electric motors, gears, belts, hydraulics, structural resonances etc (Figure 3).

This often makes interpretation of vibration data difficult other than by a trained specialist and can, in some situations, lead to a misdiagnosis resulting in unnecessary machine downtime and costs.

6.1 Bearing Characteristic Frequencies

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

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

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

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/Roller pass frequency of outer raceway (BPFO)

$f_{b/i}$ = Ball/Roller pass frequency of inner raceway (BPFI)

f_b = Rolling element rotational frequency

D = Pitch circle diameter

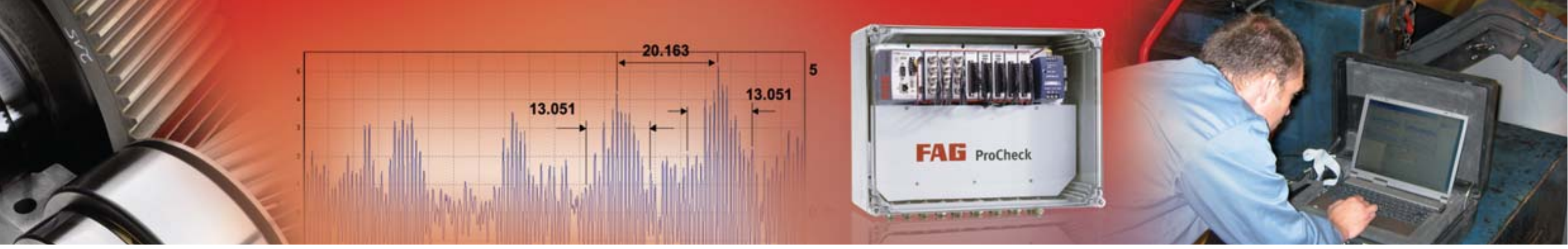
d = Diameter of roller elements

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. In practice, however, this is rarely the case and, due to a number of factors, the rolling elements undergo a combination of rolling and sliding. In addition, the operating contact angle may be different to the nominal value. 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 that data is obtained at identical speeds when comparing vibration signatures. Speed changes will cause shifts in the frequency spectrum, leading to inaccuracies in both amplitude and frequency measurement. In variable speed equipment, spectral orders may sometimes be used where all the frequencies are normalised relative to the fundamental rotational speed. This is generally called “order normalisation”, in which the fundamental frequency of rotation is called the first order.



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 in approximate terms as 40% (0.4) and 60% (0.6) respectively of the inner ring speed multiplied by the number of rolling elements.

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 in order to detect a deterioration of or damage to the rolling surfaces.

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. Depending upon the dynamic range of the equipment, however, background noise levels and other sources of vibration bearing frequencies can be difficult to detect in the early stages of a defect.

Over the years, however, a number of diagnostic algorithms have been developed to detect bearing faults by measuring the vibration signatures on the bearing housing. These methods usually take advantage of both the characteristic frequencies and the "ringing frequencies" (i.e. natural frequencies) of the bearing.

By measuring the frequencies generated by a bearing, it is often possible to identify not only the existence of a problem but also its cause. While it may be only be necessary to identify that a bearing is starting to deteriorate and plan when it should be changed, a more detailed analysis of the vibration can often give some vital clues as to what caused the problem in the first place. This can be further enhanced by inspecting the bearing after removal from the equipment, especially if the fault has been identified at an early stage.

6.2 Bearing Defects

Rolling contact bearings represent a complex vibration system whose components e.g. rolling elements, inner raceway, outer raceway and cage interact to generate complex vibration signatures(3). Although rolling bearings are manufactured using high precision machine tools and under strict cleanliness and quality controls, they have degrees of imperfection like any other manufactured part and generate vibration as the surfaces interact through a combination of rolling and sliding. Although the amplitudes of surface imperfections are now of the order of nanometres, vibrations can still be produced in the entire audible frequency range (20Hz-20kHz).

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, yet they can have a significant impact on vibration critical equipment or can result in reduced bearing life. This type of defect can take a variety of forms: indentations, scratches along and across the rolling surfaces, pits, debris and particles in the lubricant. During the early development of the fault, the vibration tends to be impulsive but changes as the defect progresses and becomes larger.

The type of vibration signal generated depends on many factors including the loads, internal clearance, lubrication, installation and type of bearing. Since defects on the inner ring raceway must travel across a number of interfaces, such as the lubricant film between the inner ring raceway and the rolling elements, between the rolling elements and the outer ring raceway and between the outer ring and the housing, they tend to be more attenuated than outer ring defects and can therefore sometimes be more difficult to detect.

When a defect starts, a single spectral line can be generated at the ball pass frequency and, as the defect becomes larger, it allows movement of the rotating shaft and the ball pass frequency becomes modulated at shaft rotational speed.
This modulation generates a sideband at shaft speed. As the defect increases in size, more sidebands may be generated, until at some point the ball pass frequency may no longer be generated, but a series of spectral lines spaced at shaft rotational speed occurs.

A defective rolling element may generate vibration at twice the rotational speed as the defect strikes the inner and outer raceways. The vibration produced by a defective ball may not be very high, or may not be generated at all, as it is not always in the load zone when the defect strikes the raceway.
As the defect contacts the cage, it can often modulate other frequencies i.e. ball defect frequency, ball pass frequency or shaft rotational frequency and show up as a sideband.
The cage rotational frequency can be generated in a badly worn or damaged cage. In a ball bearing, the rolling elements may never generate ball rotational frequency or twice the ball rotational frequency due to the combination of rolling and sliding and the constant changing of the ball rotational axis.
In cylindrical roller bearings, the damage often occurs all the way around the majority of the rolling element surface, so the rolling element rotational frequency may never be generated.

6.3 Variable Compliance

This occurs under radial or misaligned loads, is an inherent feature of rolling bearings and is completely independent of quality. Radial or misaligned loads are supported by a few rolling elements confined to a narrow region and the radial position of the inner ring with respect to the outer ring depends on the elastic deflections at the rolling element/raceway contacts, Figure 5. The outer ring of the bearing is usually supported by a flexible housing which generally has asymmetric stiffness properties described by the linear springs of varying stiffness.

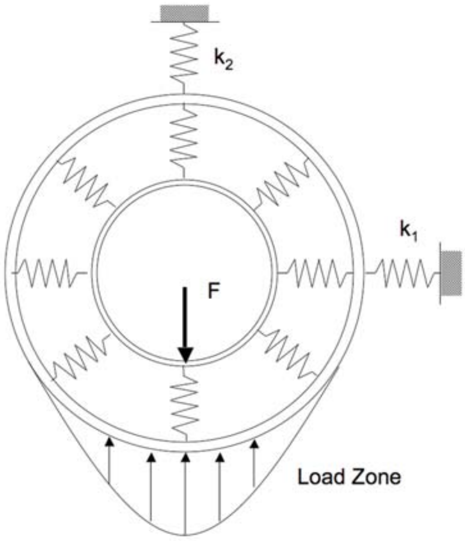


Figure 5. Simple model of a bearing under radial load

As the bearing rotates, the individual ball loads and hence the elastic deflections change to produce a relative movement between the inner and outer rings. The movement takes the form of a locus which is two dimensional and contained in a radial plane under radial load, while it is three dimensional under misalignment. The movement is also periodic, with a 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. So even a geometrically perfect bearing will produce vibration because of the relative periodic movement between the inner and outer rings due to raceway elastic deflections.

Variable compliance vibration is heavily dependent on the number of rolling elements supporting the externally applied loads; 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, hence variable compliance generally increases with radial internal clearance. A distinction is made between "running clearance" and radial internal clearance (RIC). When fitted to a machine, the former is normally smaller than the RIC due to differential thermal expansion and interference fit of the rings. In high speed applications, the effect of centrifugal force should also be considered.

Variable compliance vibration levels can exceed those produced by roughness and waviness of the rolling surfaces. In applications where vibration is critical, however, it can be reduced to a negligible level by using ball bearings with the correct level of axial preload.

6.4 Bearing Speed Ratio

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.

When abnormal or unsatisfactory lubrication conditions are encountered, or when skidding occurs, the bearing speed ratio will deviate from the normal or predicted values. If the bearing speed ratio is below predicted values, this may indicate insufficient loading, excessive lubrication or insufficient bearing radial internal clearance, which could result in higher operating temperatures and premature failure. Conversely, a higher than predicted bearing speed ratio may indicate excessive loading, excessive bearing radial internal clearance or insufficient lubrication.

For an experienced analyst, vibration can be used not only to detect deterioration in bearing condition but also to make an initial assessment of whether the equipment is operating satisfactorily at initial start-up.

In electrical machines, two deep groove radial ball bearings are commonly used to support the shaft; one is a locating bearing while the other is a non-locating bearing that can be displaced in the housing to compensate for axial thermal expansion of the shaft. It is not unusual for bearings to fail catastrophically due to thermal preloading or cross-location where there is insufficient clearance between the bearing outer ring and housing resulting in the non-locating or "floating" bearing failing to move in the housing i.e. the bearings become axially loaded.

The effect of this axial load is to increase the operating contact angle, which in turn increases the BPFO. For a ball bearing, the contact angle can be estimated as follows:

$$\alpha = \text{Cos-1}\ [1 - \text{RIC} / [(2\ (r_o + r_i - D)]\ ]$$

α = Contact angle

RIC = Radial internal clearance

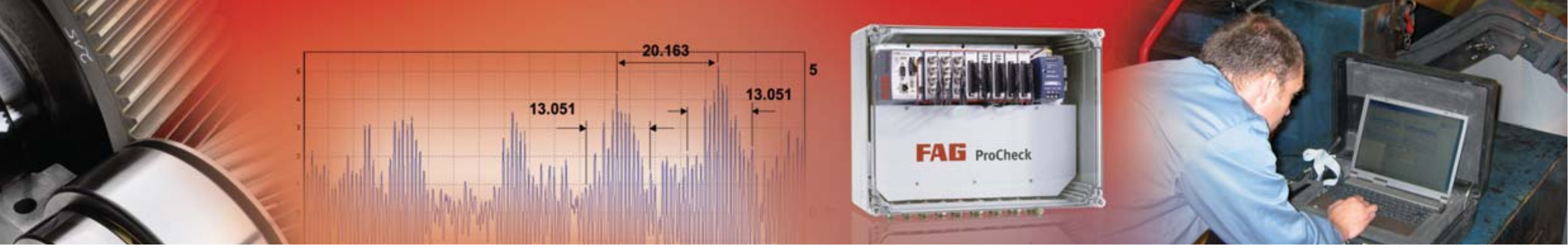
r_o = Raceway groove radius of outer ring

r_i = Raceway groove radius of inner ring

D = Ball diameter

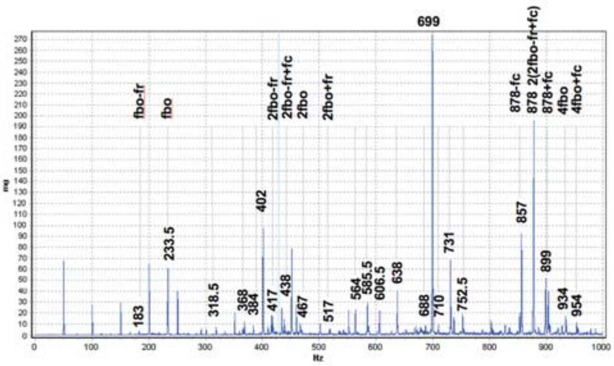
Since a deep groove ball bearing is designed to have a radial internal clearance in the unloaded condition, it can also experience axial play. Under an axial load, this results in the ball/raceway contact having an angle other than zero. As the bearing radial internal clearance and thus the axial play increases, so does the contact angle. For a correctly assembled motor under pure radial load, the contact angle will be zero and the BPFO will be given by:

$$f_{b/o} = Zf_r/2\ [1 - d/D\ ]$$

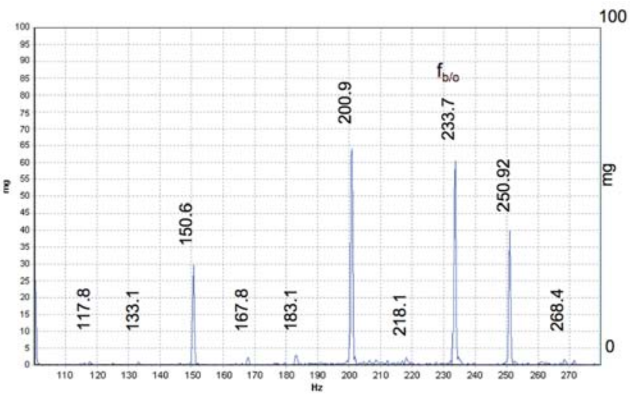


On the other hand, if cross-location occurs (the outer ring cannot move axially in the housing) the bearing radial internal clearance will be lost by the relative axial movement between the inner and outer rings, the bearings become axially loaded and the BPFO will increase due to the increase in contact angle. The amplitude of BPFO is likely to be small until the bearing becomes distressed and it may not always be possible to detect the BPFO, particularly if using a linear amplitude scale. A log or dB amplitude scale may be better, but care should also be exercised here because there may be other frequencies that may be close to the BPFO.

A good example of how the bearing speed ratio can be used to identify a potential problem is given in Figure 6, which shows a vibration acceleration spectrum measured axially at the drive end (DE) on the end cap of a 250kW electric motor. The measurements were obtained during a “run-up” test prior to installation in the plant.



(a) Base spectrum



(b) Base spectrum with zoomed amplitude and frequency scale

Figure 6. Axial vibration acceleration spectrum at the DE on the end cap of a 250kW electric motor

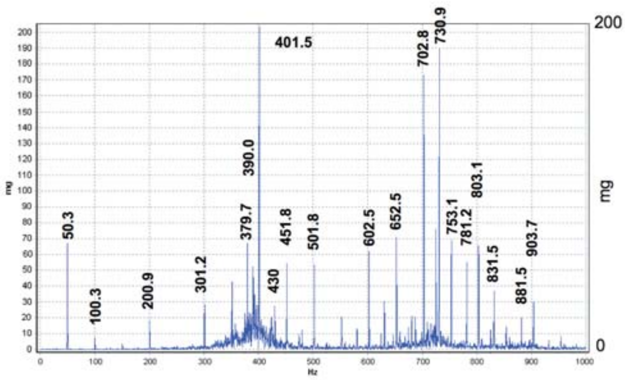
For a nominal shaft speed of 3000 rpm, the calculated BPFO was 228.8Hz, giving a bearing speed ratio of 4.576. The measured BPFO was 233.5Hz (Figure 6) giving a bearing speed ratio of 4.67, an increase of 2%. The BPFO of 233.5Hz corresponds to a contact angle of 25° which strongly suggested that that the type 6217 bearing was subjected to a high axial load.

The most probable reason was that the bearing had been installed too tightly and could not move in the housing as the shaft of the motor expanded and contracted.

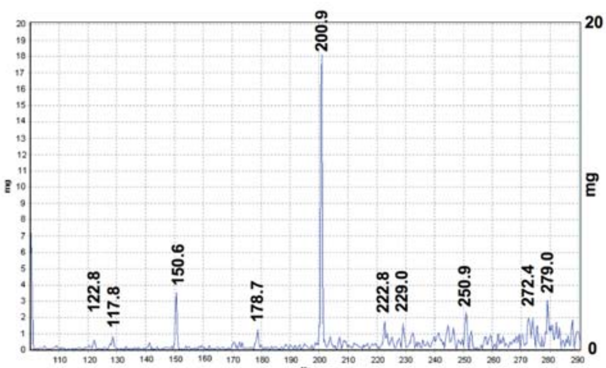
Shortly after installation, the motor failed catastrophically. A photograph of the inner ring in Figure 11 shows the ball running path offset from the centre of the raceway towards the shoulder. After a thorough investigation of all the bearing fits, it was confirmed that there was insufficient clearance between the outer ring and the housing of the non locating bearing, resulting in cross-location (thermal loading) which was consistent with the vibration measurements taken prior to installation.

A number of harmonics and sum and difference frequencies relating to the BPFO (233.5Hz), cage rotational frequency (21Hz) and inner ring rotational frequency are also evident in the spectrum, Figure 6.

Once the motor had been rebuilt with new bearings and the correct bearing fits, the “run-up” test was repeated prior to installation, Figure 7.



(a) Base spectrum



(b) Base spectrum with zoomed amplitude and frequency scale

Figure 7. Axial vibration acceleration spectrum at the DE on the end cap of a 250kW electric motor after fitting with new bearings

The base spectrum shows no characteristic bearing frequencies but, when both the amplitude and frequency scales are expanded, a discrete peak at 229Hz becomes evident, Figure 6(b), which matches very closely with the predicted BPFO, $f_{b/o}$, of 228.8Hz. This motor went on to operate successfully.

7. Examples of Vibration Monitoring

In this section, some examples are given of how vibration can be used to detect and diagnose problems on rotating equipment, ranging from electric motors to large crushing machines used for mining and processing. Examples are also taken from the FAG WiPro Condition Monitoring System used for monitoring the condition of wind turbine drive trains.

7.1 Electric Motor

An example of a vibration spectra measured axially on the DE of a 250kW electric motor is shown in Figure 8.

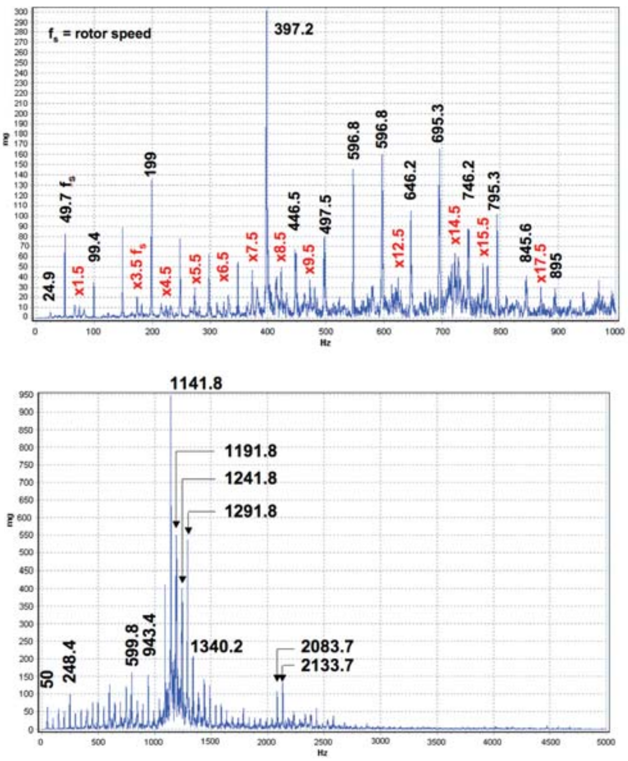


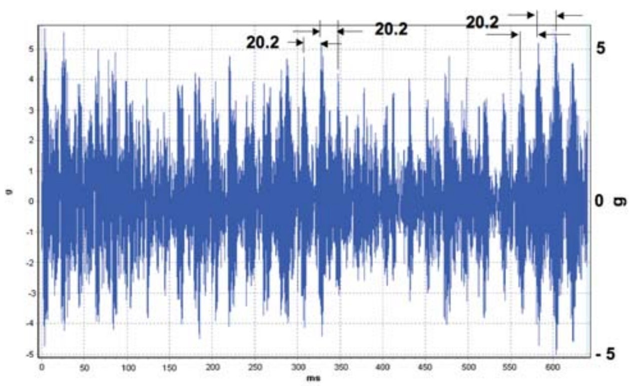
Figure 8. Vibration acceleration spectra measured axially on the DE of a 250kW electric motor

The nominal rotational speed was 3000 rpm and the rotor was supported by two type 6217 C4 deep groove ball bearings (85mm bore) with grease lubrication. The vibration spectra are dominated by vibration at both harmonics and sub harmonics of the rotor speed (49.7Hz). The spectrum 0-1kHz shows a number of harmonics and sub harmonics of the rotor speed with no bearing characteristic frequencies being evident.

In the 0-5kHz spectrum there is a dominant discrete peak at 1141.8Hz which neither corresponds with a harmonic of the rotor speed i.e. 1141.8/49.98 = 22.84 nor with any of the bearing generated frequencies. On either side of 1141.8Hz peak are sidebands spaced at the rotor speed (49.98Hz) i.e. the 1141.8Hz frequency is amplitude modulated at the rotor speed.

This is shown more clearly in Figure 9(a), which shows that in the range 0-650ms the signal is amplitude modulated at 20.2ms which, within the measurement accuracy, corresponds to 49.98Hz, i.e. the rotor speed. Expanding the time scale from 500-600ms, Figure 9(b), shows that the time between peaks is 0.87ms i.e. 13.051ms divided by 15 cycles which corresponds to a carrier frequency of approximately 1149Hz. Within the measurement accuracy of 0.0796ms, this corresponds to the frequency of 1141.8Hz (0.876ms) shown in Figure 8.

Dividing 1141.8Hz by the rotational speed of 49.98Hz gives 22.85, which is not close enough for the frequency to be a harmonic of the rotational speed. One of the extensional vibration modes of the outer ring was estimated to be 1158Hz, which is very close to the measured value of 1141.8Hz.
One possible explanation is that the discrete peak at 1141.8Hz is an excited natural frequency of the outer ring.



(a) Vibration acceleration 0-650ms

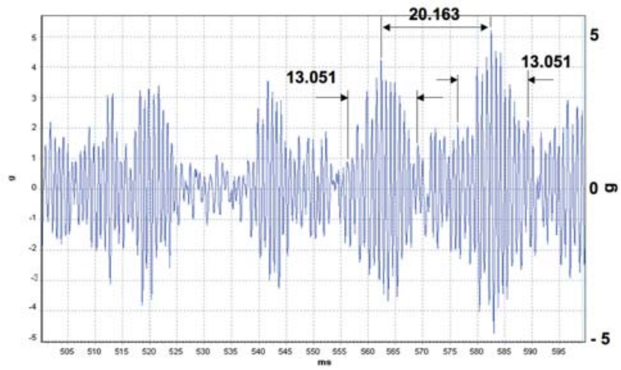
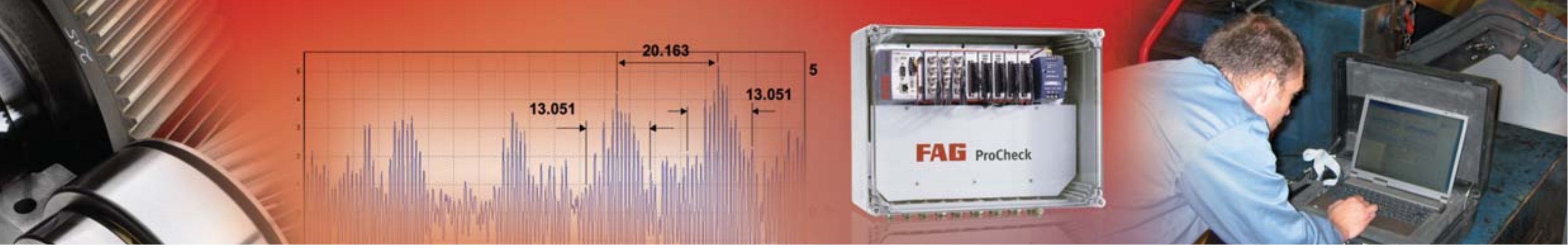


Figure 9. Time signals of vibration acceleration measured axially on the DE of a 250kW electric motor.



The dominance of vibration at rotor speed and the absence of any frequencies related to the rolling bearings suggest that the bearings have experienced such severe damage to the rolling contact surfaces that this has resulted in an increase in radial internal clearance, allowing significant radial movement of the rotor.

The envelope spectrum, Figure 10, shows a dominance of peaks related to the rotor speed with no evidence of any bearing characteristic frequencies.

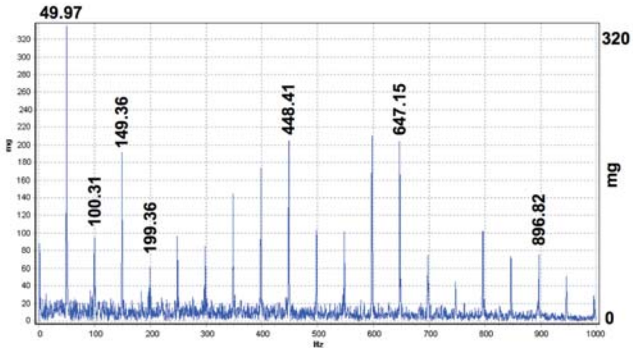


Figure 10. Envelope spectrum of vibration acceleration measured axially on the DE of a 250kW electric motor

When the bearings were removed from the motor and examined the NDE bearing had a ball running path offset from the centre of the raceway towards the shoulder, Figure 11.

Offset running band

Figure 11. Photograph of type 6217 inner ring showing running path offset from centre of raceway

The DE bearing had significant damage all around both raceways and the rolling elements showed signs of severe distress. It was clear from the NDE bearing, however, that the cause of the failure was too tight a fit between the outer ring and housing. This resulted in the bearing being unable to move axially in the housing and compensate for axial thermal expansion of the rotor, leading to a high axial load.

During a "run up" test prior to installation in the plant, the RMS vibration level of the motor in the frequency range 0-1kHz before and after fitting the new bearings was 0.304g and 0.335g respectively.

7.2 Impact Crusher

Figure 12 shows another example of a vibration acceleration spectrum obtained from the housing of a type 23036 (180mm bore) spherical roller bearing located on the main drive shaft of an impact crusher. The spectrum shows a number of harmonics of the BPFO, 101Hz, with a dominant peak at 404Hz ($4f_{b/o}$) and sidebands at the shaft rotational frequency, 9Hz.

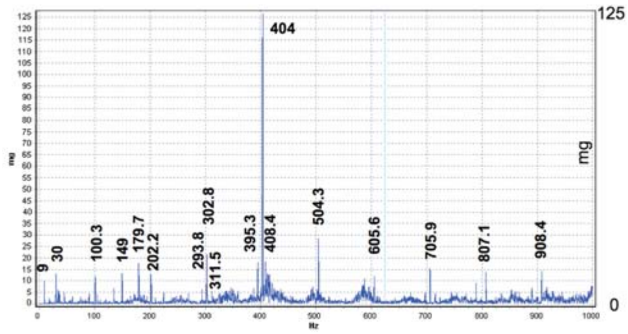


Figure 12. Vibration acceleration measured radially on the housing of a type 23036 spherical roller bearing

When the bearing was removed from the machine and examined, one part of the outer raceway had black corrosion stains as a result of water ingress which had occurred during external storage of the machine, Figure 13.

Figure 13. Type 23036 spherical roller bearing outer ring raceway showing black corrosion stains

A number of the rollers also had black corrosion stains which was consistent with the vibration at the cage rotational frequency, f_c=4Hz, in the envelope spectrum, Figure 14.

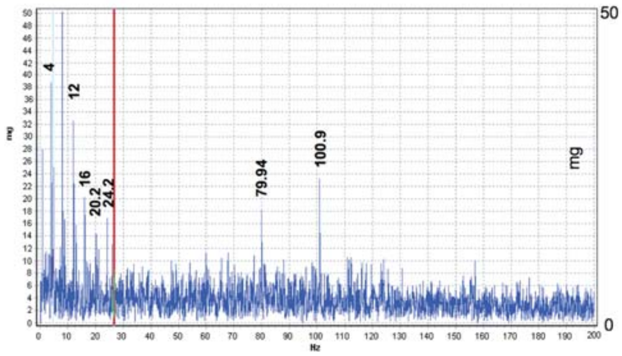


Figure 14. Envelope spectrum of the type 23036 spherical roller bearing (suspect machine)

The modulation of the time signal at the cage rotational frequency can be clearly seen in the time signal, Figure 15.

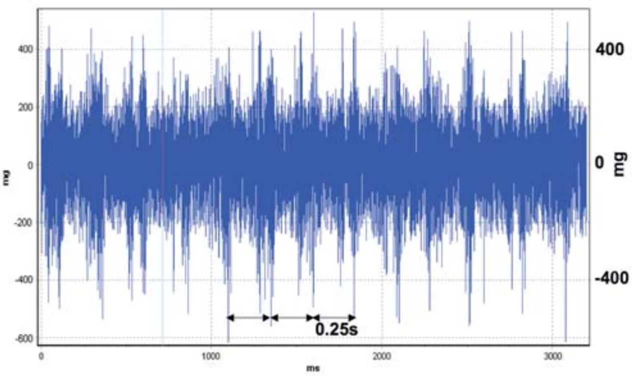
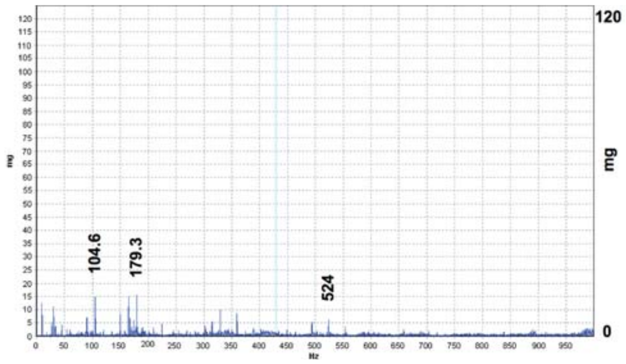
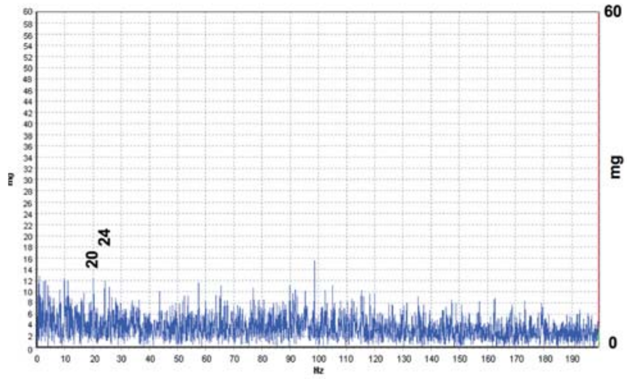


Figure 15. Acceleration time signal of the type 23036 spherical roller bearing (suspect machine)

Figure 16. shows vibration measurements obtained from an identical machine considered to be operating satisfactorily. Both the base and envelope spectrum show no indication of any vibration related to the type 23036 spherical roller bearing.



(a) Base spectrum



(b) Envelope spectrum

Figure 16. Vibration acceleration measured radially on the housing of a type 23036 spherical roller bearing (good machine)

7.3 Generator

During the initial running-in phase of a 2MW generator on a test bed, an intermittent rattling noise was evident.
The generator was fitted with a type 6232 deep groove ball bearing at the DE and a cylindrical roller bearing at the non-drive end (NDE). Both bearings were grease lubricated. The initial suspicion was that the rattling noise was related to the cage because it was intermittent and became worse as the bearings reached operating temperature.

Vibration measurements obtained from the DE of the generator are shown in Figure 17.

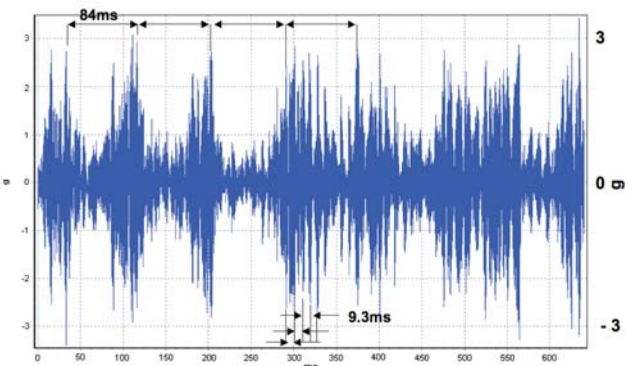
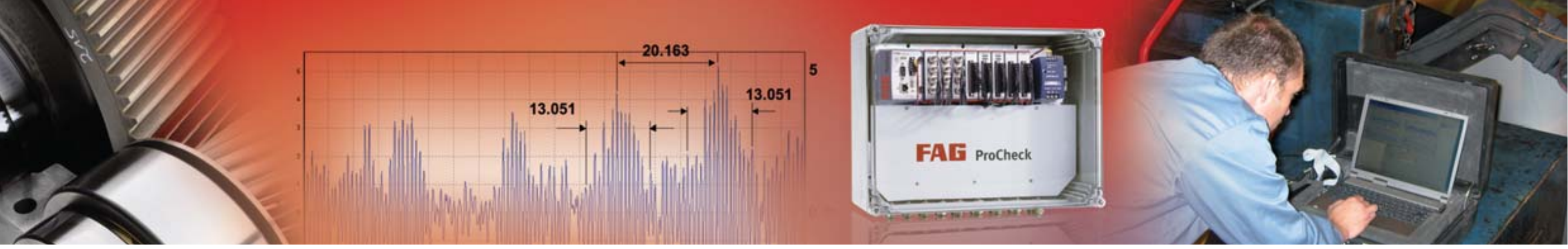
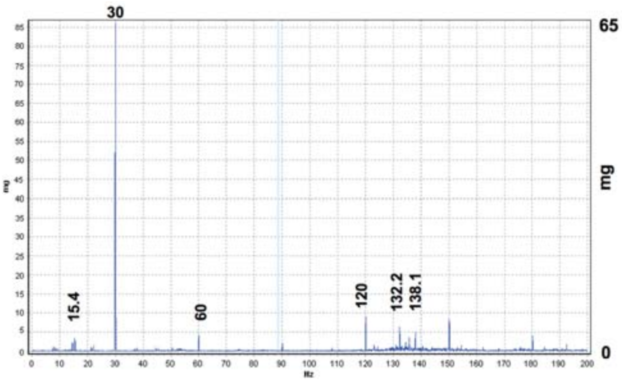


Figure 17. Radial vibration acceleration measured at the DE end cap

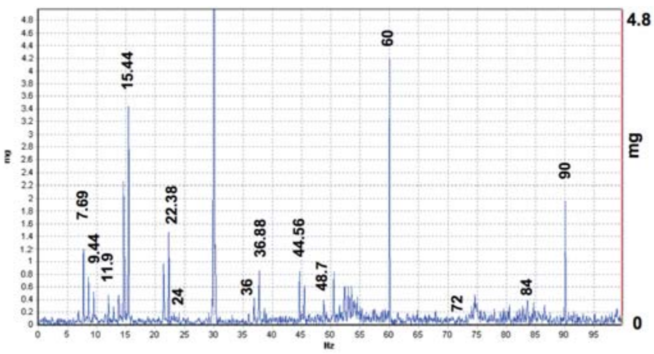
The acceleration time signal shows what appears to be random bursts of high frequency vibration but, on closer inspection, this was in fact modulation at the cage rotational frequency.
The time period between the pulses corresponds to the revolution of the cage, 84ms ($f_{c/o}$=11.9Hz). Also present are pulses spaced at 9.3ms which correspond to the BPFO ($f_{b/o}$ =107.9Hz) of the type 6232 deep groove ball bearing. Dividing the time period for one revolution of the cage, 84ms, by 9.3ms gives the number of rolling elements i.e. 84/9.3=9.



Although vibration at cage speed is evident in the time signal, there are no peaks evident in the spectrum at cage speed, Figure 18(a). This is because the energy produced by the cage is very small and evidence of any vibration related to the cage is contained within the overall carpet levels of the spectrum. By reducing the amplitude scale, Figure 18(b), some evidence of cage vibration starts to appear with discrete peaks becoming just noticeable at 11.9, 24, 36Hz i.e. the first three harmonics of the cage speed. The 6th (72Hz) and 7th (84Hz) harmonics of the cage speed are also evident.



(a) Base spectrum



(b) Base spectrum with zoomed amplitude

Figure 18. Vibration acceleration measured radially at the DE of a 2MW generator

While the base spectrum shows evidence of cage vibration, on closer examination vibration at the cage speed is readily seen in the envelope spectrum at 10.9Hz along with the BPFO, which was not evident in the base spectrum, Figure 19. The envelope spectrum was obtained by using a higher sampling frequency giving a frequency resolution of 1.56Hz and the cage frequency of 10,9Hz is within the measurement accuracy.

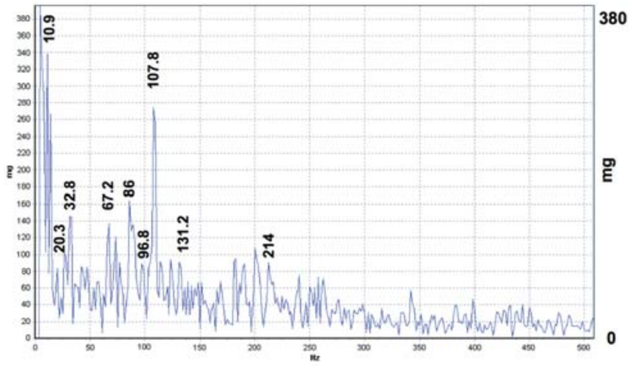


Figure 19. Envelope spectrum obtained from the DE end cap of a 2MW generator with rattling noise present

Vibration measurements were also obtained when the rattling noise was absent and vibration at both the cage rotational frequency and BPFO were not evident in the envelope spectrum, Figure 20.

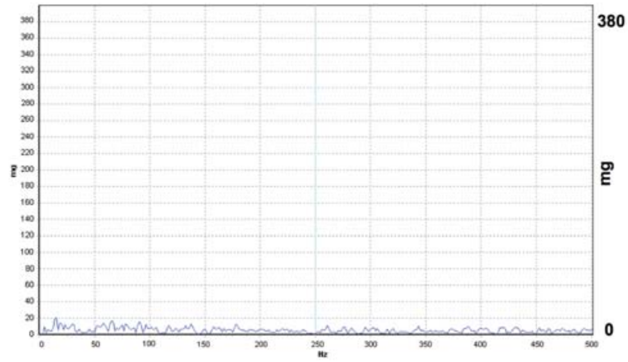


Figure 20. Envelope spectrum obtained from the DE end cap of a 2MW generator when the rattling noise was absent

The presence of vibration at the cage speed and BPFO does not necessarily mean the bearing is in distress. Even a geometrically perfect bearing will generate vibration (3).

Cage noise, which can be loosely described as rattling, is not uncommon in ball bearings fitted with pressed steel cages. This is particularly true under minimal lubrication conditions, where the lubricant cannot provide sufficient damping as the cage interacts with the rolling elements and, in the case of ring guided cages, with the cage guiding surface as the rolling elements speed up and slow down when entering and leaving the load zone. The cage motion is often erratic; the cage may rise and fall in slow running bearings while it may run eccentrically in high speed bearings due to the effects of centrifugal force. The first bending mode of the cage may also be excited giving rise to a squeal or squeak which may be in the low kilohertz range for a 25mm bore bearing.

Cage noise is not uncommon especially in grease lubricated bearings and is often symptomatic of the running-in process as the grease is worked or “milled” and disperses itself within the bearing.

Similarly, the presence of vibration at the BPFO does not necessarily indicate a problem and may be a result of variable compliance (section 6.3).

7.4 Vertical Impact Crusher

A vibration assessment was made on a vertical impact crusher prior to undergoing field trials. The main aim was to verify that the new bearing arrangement, comprising a type NU2230E cylindrical roller bearing and a type QJ326 duplex bearing at the DE and a type NU2230E cylindrical roller bearing at the NDE, was operating satisfactorily. The shaft rotational speed was 1750 rpm and it was driven by a pair of bevel gears with a ratio of 1:1 (36 teeth), giving a gear mesh frequency of 1050Hz.

Vibration acceleration was measured radially on the rotor gear drive housing, Figure 21.

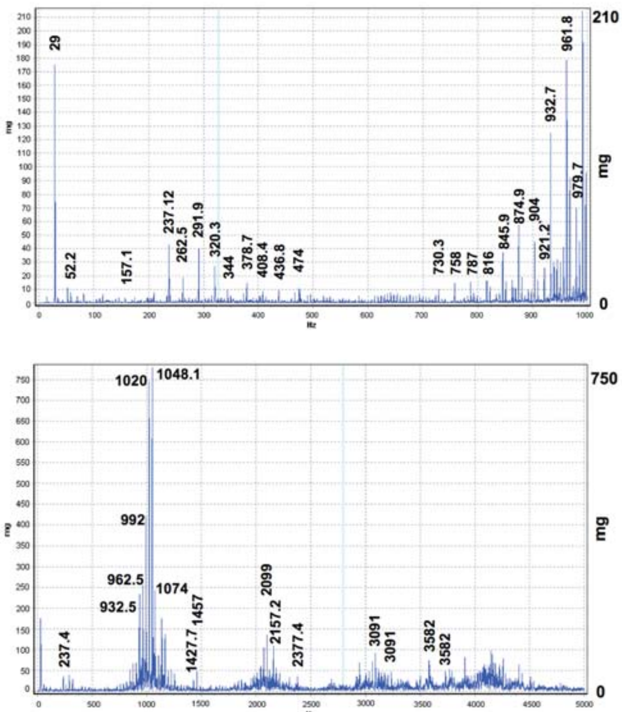


Figure 21. Radial vibration acceleration spectrum on the housing of a vertical impact crusher

Vibration at shaft rotational frequency (29.2Hz) is evident along with a number of harmonics. Vibration is also present at 237Hz, which corresponds to the BPFO of the cylindrical roller bearing, along with harmonics at 474Hz and 711Hz which are just evident on the linear amplitude scale.

The predominant vibration is at the gear mesh frequency, f_{gm}, of 1048Hz, along with a number of sidebands at the shaft rotational frequency, f_s. The presence of sidebands at rotational frequency is not unusual, especially in the case of sidebands at $f_{gm} \pm f_s$.

As more sidebands appear at higher amplitude, however, this is normally an indication of gear eccentricity or backlash. It was therefore decided to remove the drive shaft, inspect the bearings and adjust the gear backlash.

All the bearings appeared in generally good condition, although it should be emphasised that because the bearings were not removed from the housing it was not possible to inspect the outer ring raceways, especially those of the cylindrical roller bearings where vibration at $f_{b/o}$ had been detected albeit at a relatively low amplitude.

Due to variable compliance effects, bearings will always exhibit vibration at their characteristic frequencies, so the detection of a discrete peak is not necessarily an indication of a problem. Conversely, a bearing in an advanced failure condition will not necessarily generate vibration at the characteristic frequencies. It is therefore important to interpret vibration data with a great deal of caution until experience has been built up.

After reassembly, the vibration measurements were repeated and the results are shown in Figure 22.

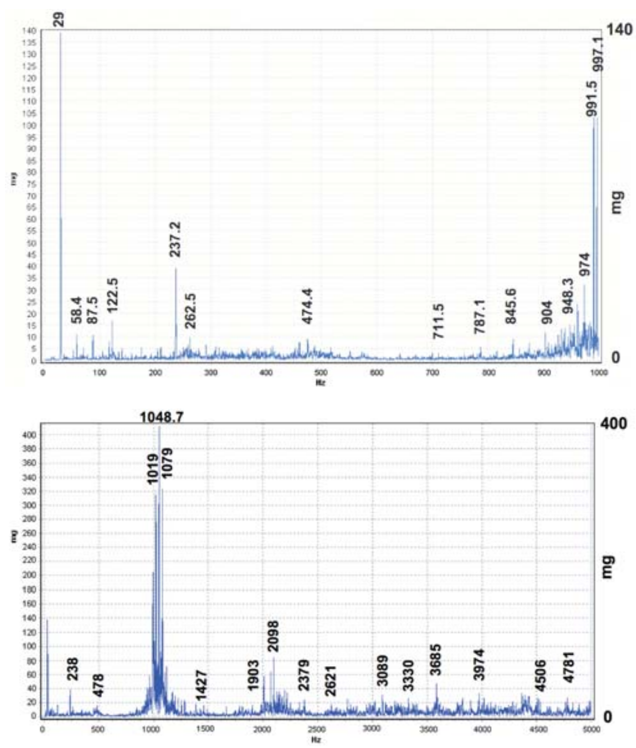
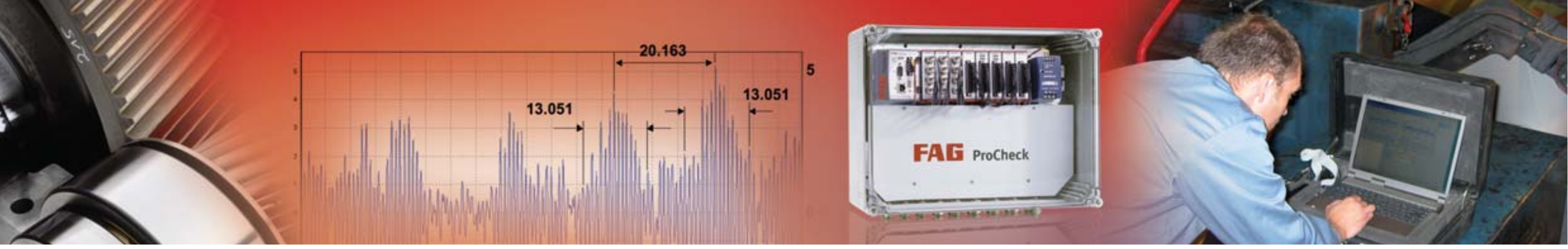
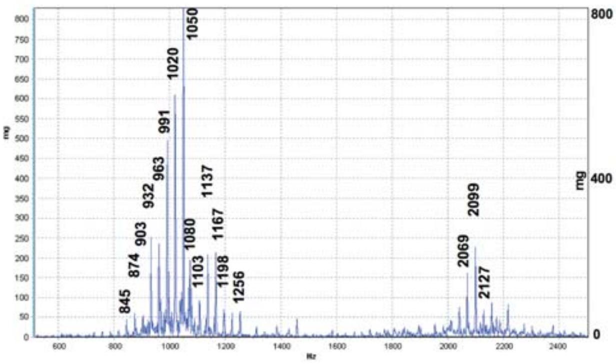


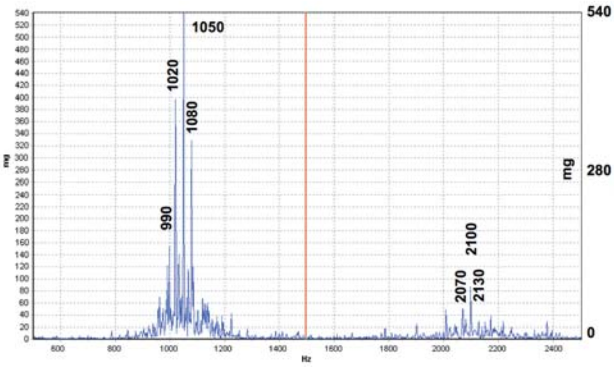
Figure 22. Radial vibration acceleration spectrum on the housing of a vertical impact crusher after adjustment of gear backlash



After resetting of the machine, the gear backlash was reduced and the running speed sidebands around the gear mesh frequency were significantly reduced in both number and amplitude, Figure 23.



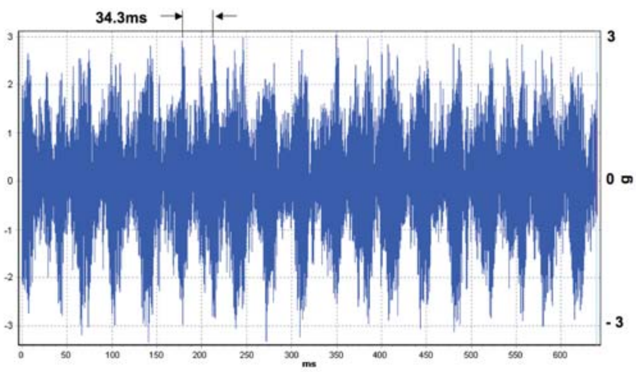
(a) Before adjustment of gear backlash



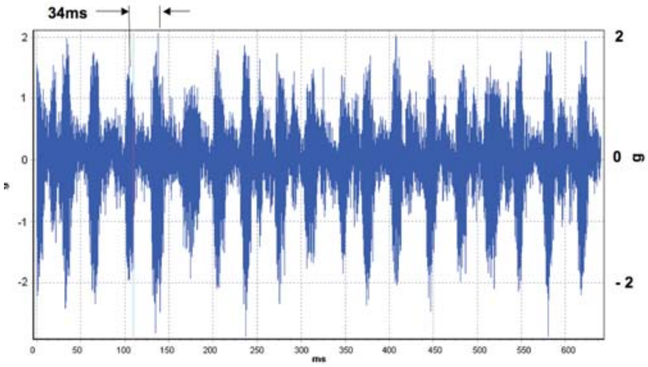
(b) After adjustment of gear backlash

Figure 23. Comparison of sidebands around gear mesh frequency

Figure 24 shows the corresponding time signals, both modulated at the rotational frequency, 29Hz (34.4ms); the RMS of the raw signal and enveloped signal decreased from 1.08g and 1.59g to 0.70g and 1.06g respectively after resetting of the machine.



(a) Before resetting of gear backlash



(b) After resetting of gear backlash

Figure 24. Acceleration time signal on the housing before and after resetting of the gear backlash

7.5 Wind Turbines

Wind power is a rapidly growing form of renewable energy in many parts of the world. As an established source of renewable electricity generation, they are set to play an important role in future energy supply around the world. In the UK, there is increasing interest in placing wind turbines offshore, which offers a number of advantages including improved wind conditions and reduced planning restrictions. However, the environment in which offshore wind turbines must operate is more demanding and often extreme, demanding a higher degree of integrity and reliability if costs are to be minimised.

Due to the remote location and poor accessibility of wind turbines, it is important that faults are detected early and consequential damage reduced or avoided and repair costs minimised. This will lead to shorter downtimes and reduced loss of revenue. Detecting bearing damage early could mean the difference between replacing the gearbox at a cost of around €250,000 and replacing the bearing at a cost of €5000.

Wind turbine gearboxes are subject to high dynamic loads and, due to changing wind conditions, the load spectrum varies greatly and includes high peak loads and low load operating conditions. The high static safety required for maximum load means that bearings with high load carrying capacity are required. When there is little wind, however, loads are low and this can lead to damage due to sliding of the rolling element set. As a result, many field operating failures are a consequence of gearbox bearing failure. Misalignment, poor lubrication and maintenance also contribute towards this trend.

Figure 25 shows the spectrum from a gearbox output shaft where the BPFO, 183Hz, and the harmonics are clearly evident. Sidebands at the rotational speed, 18.7Hz, are also present.

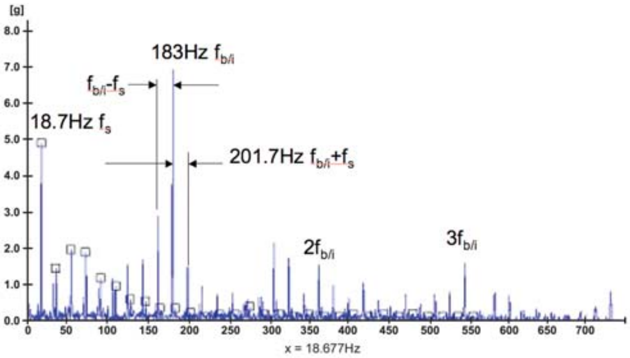


Figure 25. Frequency spectrum from gearbox output shaft

The gearbox was taken out of service for inspection and a photograph of the damaged inner ring raceway of the cylindrical roller bearing located on the high speed shaft is shown in Figure 26.

Figure 26. Damage on the bearing inner ring raceway of a gearbox output shaft

An example of an envelope spectrum obtained from a wind turbine gearbox is shown in Figure 27. Vibration at 227.1Hz, which corresponded to the BPFI of the type NU2326 cylindrical roller bearing located on the gearbox output shaft, is clearly evident along with sidebands at the shaft rotational speed. Inspection of the bearing revealed an inner ring raceway defect.

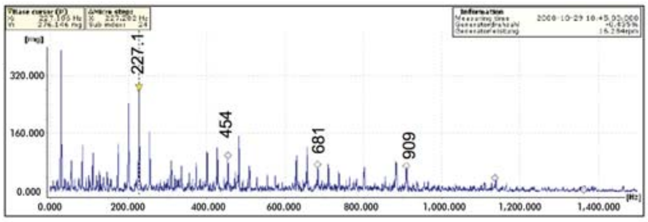


Figure 27. Envelope spectrum obtained from the gearbox output shaft of a wind turbine

This data was obtained from the FAG WiPro online Condition Monitoring system which was monitoring a VESTAS V90 turbine.

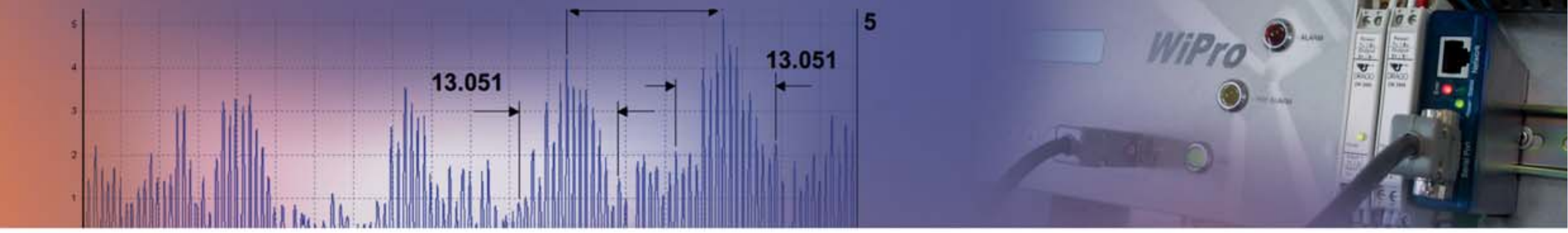
8. Summary

In some industries, maintenance is the second largest or even the largest element of operating costs and as such as becomes a cost control priority. Equipment failure not only affects plant availability but also safety, the environment and product quality. It can also impact on customer service in terms of missed deadlines and loss of confidence.

The complexity and cost of modern day plant and equipment means that plant condition monitoring is now becoming a much more cost-effective option. Although many industries have and still do take a reactive approach to maintenance, since there are no upfront costs, they pay the price in terms of increased plant downtime or lost production.

Vibration monitoring is still probably the most widely used predictive maintenance technique and, with few exceptions, can be applied to a wide variety of rotating equipment. Vibration monitoring allows the condition of machinery to be determined as it operates and detects those elements which start to show signs of deterioration before they actually fail, sometimes catastrophically. With this type of approach, unplanned downtime is reduced or eliminated, thereby increasing plant availability and efficiency and reducing costs.

Rolling bearings are a critical element in many rotating machines and generate characteristic vibration frequencies which can combine to give complex vibration spectra which at times may be difficult to interpret other than by an experienced vibration analyst. In the case of rolling bearings, however, characteristic vibration signatures are often generated in the form of modulation of the fundamental bearing frequencies. This can be used to advantage and vibration condition monitoring software is often designed to identify these characteristic features and provide early warning of an impending problem. This usually takes the form of signal demodulation and the envelope spectrum which indicates early deterioration of the rolling/sliding contact surfaces.



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