Vibration analysis has rapidly established itself as one of the most useful and important items in the modern mechanical engineering condition-based maintenance tool box. The technique is relatively easy to apply, with appropriate equipment, and the recorded data is quickly available for analysis on the ubiquitous personal computer – but all too often the technique is reserved for monitoring ancillary equipment rather than the vital production drives.

The perceived wisdom, passed down to most vibration analysts during basic training, is that all data should be gathered under repeatable conditions of speed and load, and at speeds in excess of 120 RPM. This ensures that data sets gathered within a routine monitoring programme can be directly compared with each other, with no fear of amplitudes being unduly affected by process changes, and that the vibration data gathered is unlikely to be compromised at the lower limits of an accelerometer’s frequency range.

This approach to condition monitoring allows the use of amplitude trend analysis, both for the assessment of overall parameters and of discrete frequency bands within the gathered data. If repeatability in process conditions is assured it follows that the only variable in the monitoring process will be the machines’ health, so any increase in levels must be due to parameter deterioration. This is a reasonably safe attitude to monitoring, but for these reasons many vibration analysts do not venture into monitoring variable speed or low speed equipment, as it is not considered viable as part of a vibration monitoring programme.

But what if your plant production equipment is operating at speeds below the recognised cut off of 120 RPM, or operates in a variable or intermittent way? For many plant processes operating in today’s environment of maximum return at minimum cost, this widely accepted vibration analysis rule set excludes from the monitoring programme the very items that need to be regularly examined. So where does this machine speed limit come from?

A very high proportion of today’s monitoring programmes will utilise piezo-electric industrial accelerometers as the vibration sensors. These are relatively small, cost effective, solid state, and rugged – and are easy to install as part of a new or retrofit installation – and most modern portable vibration analysers will be supplied with an accelerometer as standard. In addition, the problems associated with the installation of alternative transducers for slow speed applications, such as eddy current probes, or LVDTs (Linear Variable Differential Transformers) – which are technically superior in slow speed applications – require machine-specific mounting arrangements and prepared target areas, leaving the piezo-electric accelerometer as a much more attractive solution on grounds of cost, and thus the sensor of choice for most fixed monitoring installations. However, there are difficulties when it comes to using these devices in slow speed applications –

1. The operation of the piezo-electric accelerometer relies on continuous movement to excite the sensing crystal, with the majority of the general purpose units having a low frequency operational detection limit of around 2 Hz or 120 CPM, before the signal output rolls off in a decaying fashion.

2. If signal processing is required to enable frequency analysis (Fourier Transformation), as opposed to pure signal amplitude monitoring, then the signal processing itself can produce large mathematical errors at the very low frequency end of the spectral range. This can lead to false data representation, which swamps the genuine frequency data and can dramatically alter overall amplitude values.

While accepting the general validity of the usual recommendation regarding vibration analysis data, i.e. that it should be gathered under repeatable conditions of speed and load, and at speeds in excess of 120 RPM, the author nevertheless advocates that experimentation with the application of available techniques to the health monitoring of variable speed or slow speed equipment may sometimes offer beneficial results, particularly when the alternative might be no monitoring at all. He supports his contention with several case studies drawn from his own practice.

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Abstract

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While accepting the general validity of the usual recommendation regarding vibration analysis data, i.e. that it should be gathered under repeatable conditions of speed and load, and at speeds in excess of 120 RPM, the author nevertheless advocates that experimentation with the application of available techniques to the health monitoring of variable speed or slow speed equipment may sometimes offer beneficial results, particularly when the alternative might be no monitoring at all. He supports his contention with several case studies drawn from his own practice.

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It is for the above reasons that the 120 CPM limit is conventionally advised as the lower operating limit for conventional monitoring, following an assumption that the analyst will need to be able to see the machine’ rotational frequencies within the spectra to diagnose the most common vibration related faults, viz. imbalance and misalignment (1 x RPM for the shaft monitored). But is this necessarily the case in all circumstances, or are there other characteristics that could be assessed instead, making the use of industry standard components viable, enabling process monitoring that would otherwise be considered prohibitively expensive?

For many plants and items of equipment the defect which is most likely to stop a machine in its tracks is bearing failure. The detection and resolution of other mechanical conditions, such as imbalance, misalignment, mechanical looseness, resonance, etc. is important, and will have a direct detrimental impact on machine life cycle, but it is a bearing failure that is the likely final consequence.

The vibration amplitudes produced by failing rolling element defects can be very low, even for machines operating at ‘normal’ operating speed. A defective bearing operating at 1500 RPM will frequently not generate sufficient energy, at the fault frequencies to breach amplitude limits recommended by international vibration assessment standards, until well into fault progression, dramatically reducing the time interval between point of detection to point of failure. A large percentage of bearings carrying light loads may not trip the alert level limits, and can fail with no warning at all, despite the use of on-line monitoring systems expressly intended to detect this very problem. So if overall amplitudes are not sensitive enough, even at conventional operating speeds, what can be done to detect the bearing deterioration early enough to allow remedial action to be scheduled correctly?

Perhaps the most powerful analytical tool available within the modern maintenance programme, and frequently the most under-rated, is the data analyst. Too much reliance is often placed on a monitoring system’s ability to differentiate between healthy operating symptoms and those which indicate a fault mode developing from infancy to failure. Rather than high amplitudes, it is often the defects frequency patterns, or the changes to a time waveform that give a fault presence away.

I painfully recall being responsible, early in my monitoring career, for monitoring an 11 KV, 11,000 HP steel mill drive gearbox, which failed with no apparent warning of any difficulty despite regular monitoring. A one metre diameter output shaft bearing, supporting a gear wheel five metres in diameter, had disintegrated, damaging the shaft in the process. A check of the overall vibration trend showed no change in amplitudes for the previous twelve months, but when the frequency spectra recorded were checked there in the spectra was the tell-tale pre-defined bearing defect, with a total vibration amplitude of just 0.1mm/sec rms. Consider this in light of the closest international standards recommended fault limit of 7.2mm/sec rms, and the problems with overall amplitude as the sole health indicator start to become clear.

This was my first experience with a slow speed bearing failing catastrophically, but my experiences over the past twenty years of monitoring indicate that bearing failure seldom generates high amplitudes in low speed applications. From a positive perspective post-failure analysis indicated that the symptoms were present, and accordingly the method of analysing the frequency data was altered, in this case, to actively seek symptoms within the recorded frequency spectra, or time waveforms – which simply shouldn’t be there.

In terms of failure mode symptoms, the rolling element bearing is very obliging. It produces a series of pre-definable vibration characteristics which are directly related to the unit’s geometry and speed of rotation. These frequencies are readily available from the bearing manufacturers themselves, as they strive to demonstrate their customer support and awareness, so the identification of the frequency component within a bearing vibration signature isn’t difficult, provided the unit’s relevant speed details for the data are known. This fault signature can be used to accurately pinpoint a faulty unit within an assembly cluster, and can even be subtle enough to enable differentiation between manufacturers of equivalent replacements.

As a bearing fault mode deteriorates it passes through a series of vibration phases, and each can be helpful in flagging, and then tracking, a bearing’s progress from fault inception to final failure. The following description is a simplified account of a common fatigue wear path for a failing rolling element bearing, which can be used by an analyst in determining where a component is in its deterioration progression.

**Stage 1:** A rolling element component fault frequently starts as a crack developing in the sub-surface material. As the rolling elements pass over the defect site, encouraging the crack to develop, the bearing material is excited and produces a defect frequency, which is transmitted through the bearing material, and the support structure of the machine. The action of the succession of rolling elements travelling continuously over the fault zone can produce a very high number of low-amplitude frequency harmonics, extending high into the frequency range. Although low in amplitude the signal carries sufficient energy to excite resonant frequencies within the bearing, the support structure, or even the monitoring sensor, and these resonances can amplify the signals, making them detectable. As the impact forces contained within a bearing fault are one of the few sources which produce sufficient energy to drive these high frequency multiples, many early warning fault detection systems seek to exploit this symptom.

Some of these devices simply give out a derived signal based on the overall amplitude at the high frequency, but as this amplitude is reliant on resonant amplification, and therefore structural design and loading, the signal will only be repeatable for a specific application. Other systems will use signal processing to display frequency content or time wave-form data to enable an analyst to further assess the source of the fundamental.

**Stage 2** (see Figure 1): As the crack progresses, and the bearing’s structural integrity is compromised, the resonant amplification frequencies will alter and migrate the amplification zone to lower frequencies. It is possible to track
this second stage resonant migration. Again, frequency analysis can allow the analyst to detect the discrete fault frequencies being amplified, and so identify the source component.

Stage 3 (see Figure 2): As the crack breaks the surface of the race, physical movement is induced within the bearing components as they travel over the defect. At this point in the fault development, detectable defect frequencies within the low frequency spectra will appear, frequently with multiple harmonics. In the early part of this stage it is not uncommon for the fundamental source frequency to remain undetectable.

Stage 4 (see Figure 3): As the fault develops further, stress raisers at the edges of the crack allow damage and particles to chip off, causing the defect to develop into a spall. When this is sufficiently large, the lubrication film separating the bearing components is compromised, allowing metal to metal contact, and frequency noise floors within the spectra start to increase. It is at this point that the bearing’s temperature will increase due to the rising levels of friction and the bearing, if left alone, will fail within a matter of days or weeks.

Of course, the fault progress path described above is simplified, and assumes a fatigue source rather than damage or poor care, but given the generally accepted norms for operation, and correct design and maintenance, this fault progression is reasonably synonymous with most wear-out patterns for rolling element bearings and can be used to assess a bearing’s failure progression and enable maintenance activities to be considered based on the rate of deterioration through these stages. Given a normally loaded bearing, with no mitigating external influences, the fault progression described can take many months, or even years to progress to the point of failure.

So, to summarise: the defect frequency is produced by the load carrying components within a bearing, i.e. the
rolling elements, travelling over the defect – whether it is a sub-surface crack within the race, or component damage. In short, if there is no defect there will be no frequency to detect, or if there is a detectable frequency then damage is already occurring.

A further characteristic of a rolling element defect is that it does not tend to produce single defect frequencies in isolation. Because of the energy involved, and the repetitive nature of successive rolling elements rolling over the damage, bearing faults tend to produce multiple harmonics, and it is frequently these harmonics at higher frequencies that appear before the fundamental defect source appears within the low order spectra.

A useful rule of thumb regarding bearing defects (as a quick glance through a manufacturer’s catalogue will tell you) is that most defect frequencies, by virtue of the complex geometry within the bearing designs, are not synchronous with the shaft carrying the bearing, unlike the symptoms for, say, mechanical looseness or misalignment. Most bearing race defect frequencies fall within the range 3 – 15 x RPM. So if a rotating element bearing is producing a frequency with multiple harmonics spaced at 3.15 x RPM, and is accompanied by multiple harmonics, provided other sources for the frequency have been discounted, there is a very high probability that the bearing displays a Stage 3 race fault.

Slow speed bearing components will give out defect frequency patterns similar to those of higher speed components, provided the units are carrying sufficient load, and are operating in a conventional rotational manner. They will, however, produce much lower amplitudes at the fault frequencies owing to the lower levels of energy being imparted to the bearing fault zone, and this should be considered when evaluating the amplitudes detected.

So, is any of this information of use to the analyst when he is trying to diagnose bearing defect symptoms in a slow speed environment well below the industry’s advisory minimum speed limits? If the data contains a repeatable signal source, with multiple harmonics that exist at a frequency spacing correlating with a non-synchronous order that falls between 3 and 13 x RPM, then there is a significant probability that a bearing fault exists.

If we have used the 120 RPM speed limit because the minimum frequency to be detected is 2 Hz, and we ignore the requirement to monitor the drive’s running speed issues (imbalance, looseness, misalignment, etc.) in favour of detecting bearing failure, then as we are expecting to see defects at 3 x RPM minimum, we can be confident that we should be able to detect faults at speeds down to 40 RPM. If we further consider that we expect to see multiple harmonics of the defect frequency, then we can decrease the speed of the rotating component even further.

When monitoring for these low frequencies it should be remembered that the amplitudes will be extremely low. However, it should be remembered that if the alternative is no monitoring at all because convention says the data won’t be repeatable, what is there to lose (apart from the failure)?

The following are real world examples taken from my experiences in monitoring low, variable and intermittent operation production machinery in service. These cases generally fall into speed and operation categories that conventionally dictate that the units are un-monitorable. However, in each of these cases the fault modes presented themselves readily to either frequency spectra or time waveform analysis, and have been performed as part of routine maintenance programmes at customer request in attempts to avoid causing costly plant stoppages. In each of these cases the concerns over speed and sensor selections were explained to the clients, and each application was trialled first to determine whether the programme was likely to be successful.

**ALUMINIUM REEL BEARINGS, VARIABLE SPEED, 300 - 20 RPM**

In this application, the bearings to be monitored supported an aluminium rolling mill re-coiler, which accepts thin rolled sheet material at the output side of a rolling reduction process. The speed of the strip though the mill is an important parameter of the quality of the rolled product, and therefore to maintain the pre-set product pass speed through the mill, the re-coiler mandrel must reduce speed throughout the coil build up, to compensate for the increasing circumference of the built coil. The load on the coil mandrel and its bearings is a function of strip tension and coil mass, the latter increasing as the coil diameter builds. Maximum speed for the re-coiler mandrel is therefore at the commencement of the coiling process, and in this case was no more than 300 RPM at the mill’s highest operating speeds, dropping to 20 RPM as the coil finished.

The application of vibration monitoring was required following catastrophic failure of the mandrel support bearing races, which disintegrated with no significant forewarning to production staff.

Vibration data was gathered using a portable data collector, fixed installation standard accelerometers, and simultaneous speed measurement to speed stamp the gathered vibration data as it was collected. To ensure maximum energy was transmitted from any bearing fault, data was gathered at the highest mandrel speeds possible following the mill’s acceleration to full rolling speed, and spectral collection set to 1 average in an attempt to avoid smearing of the frequency peaks as the re-coiler decelerated to compensate for coil build up. Data was gathered on a routine monthly schedule during normal production, and the subsequent analysis of the data was undertaken manually off site. Each vibration spectrum was manually checked for the presence of abnormal mechanical symptoms, and in particular the presence of bearing defects.

Within two months of the bearing replacement both roll support bearings produced non-synchronous multiple harmonics which correlated directly with the bearing manufacturer’s predicted outer race defect frequencies. As can be seen in Figure 4, the amplitudes were extremely low. The spectra showed, however, that the fundamental defect frequency was evident, indicating that the damage had already breached the bearing surface. As mentioned in the previous discussion, defect frequencies should not be evident at all in a new bearing (split race bearings aside) and, as a consequence, lubrication of the bearings was accelerated and progression of the fault amplitudes and frequency spread monitored for deterioration. This didn’t occur for six months, the increased lubrication managing to hold further deterioration at bay, but eventually increasing amplitudes at the fundamental frequency commenced and the bearings were replaced during a planned maintenance stop.
Examination of the bearing showed a number of radial cracks in the race, mainly clustered within the load zone, with some cracks close to the race edge. The plant were not concerned with further evaluating the fault mode, but as can be seen in the Figure 5 photograph very little wear of the bearing surface had taken place. However, the spacing of the cracks, and their orientation to the rolling plane of the bearing, suggested that they may have been related to damage when the roll assembly was in storage (false brinelling) or have been produced during installation, and that these minor damage sites were aggravated in service by an ingress of the acid-based rolling fluids in use on the mill. It was further surmised that crack propagation of the defects to the edge of the race may have resulted in sudden break up of the outer race with little physical warning.

**Continuous Casting Oscillating Drive Gearbox – 90 RPM**

Continuous casting is a commonplace method of producing metal forms or blanks for process purposes, and has replaced ingot casting routes for the majority of metal processing companies. The process is deceptively simple, with an open ended mould fed continuously from the top with molten material, and a cast form withdrawn from the bottom of the mould at a carefully controlled, and sufficiently low, speed to ensure that the liquid material has a solidified skin before emerging.

To ensure that the molten material does not adhere to the surfaces of the mould, the latter is agitated with a mechanical oscillating drive along the axis of the product withdrawal route. If the oscillation route or process is disrupted, or fails, then the fragile cast skin of the product can rupture, causing the molten product to escape, resulting not only in lost production and raw material for the strand concerned, but also the potential for expensive damage and downtime, depending on how severe the material breakout is.

As a consequence, the mould oscillation can be monitored for evidence of changes or disruptions to what should be a smooth and uniform mechanical motion. The motion of the caster is frequently slow, with mould oscillation varying between 60 and 200 RPM, and is monitored on many plants with a dedicated displacement monitoring device. This frequently forms part of the process monitoring system, and monitors the physical displacement of the mould oscillation as a value, and also the mould’s operating frequency. But these devices are frequently overlooked as a means of monitoring the drive health.

For many smaller casting plants, such devices are not fitted, and in one case we were asked to monitor the drive to assess the reasons for mechanical damage being caused to the oscillation drives. In this case there were no displacement transducers installed, and the plant considered the cost of retro-fitting to be too high due to the limited access available around the machine. We also needed to verify whether the technique of monitoring the oscillation of the drive and its linkage arms would indeed be successful. So, having warned the customer not to expect too much, accelerometers were used with the intention of evaluating the purity of the stroke as a time waveform, and checking the purity, from one side of the table linkage to the other, for discrepancies and differences in amplitude.

Initial traces were encouraging, with newly overhauled tables producing sinusoidal motions at oscillating frequency, accompanied by a visible impact in the waveform as the drive backlash was excited just after the bottom of the table stroke (see Figure 6). This is produced as the friction between the cast slab and the mould results in a steady downward force on the drive table and linkages – but at the reverse
of the stroke at the bottom of the travel, gear backlash in the drive unit is excited. In addition, waveforms taken from both sides of a table operating correctly should produce symmetrical wave forms with uniform displacement values.

The most common problems found during the monitoring programme were the development of impact forces immediately prior to the table reaching the bottom of the stroke (see Figure 7) or disrupted and distorted wave forms, sometimes to one side of the table, producing asymmetric results (see Figure 8). In most cases this was produced by debris build up on the table’s lower bump stops, or around the table’s edges and skirts, although failing linkage bearings could produce similar symptoms.

As a result of the work done to date, debris cleaning operations around the casting tables have been altered, and targeted at specific areas found to be prone to slag build up, and as a consequence table operation has improved in reliability.

**CONTINUOUS CASTER STRAIGHTENER SECTION ROLL BEARINGS**

As an example of why monitoring should be experimented with, this final example just shouldn’t lend itself to monitoring using conventional techniques at all. A client asked us to develop a monitoring approach for their straightener section roll bearings, which were failing after only six months of service. The roll turns at just 4 RPM, creating concerns over whether the components would be operating as conventional rolling elements. The rolls involved transport cast billets from a continuous casting machine to the cooling beds for distribution. The rolls are water cooled but the bearings regularly experience ambient temperatures of over 100 degrees. Cost was an issue due to the number of sensors required for full coverage, and their required locations meant that accelerometers would be the only cost effective measure, so a trial survey was arranged during the plant’s maintenance period.

With the rolls turning at only 4 RPM, analysis of the spectra was considered a tall order, but there was some hope that any significant defect within the bearing might produce either peak amplitude changes in the time waveforms, or overall
rises in carpet level for the time waveform. Accelerometers were attached using magnetic mounts, and as the data collector being used is primarily set for spectral analysis, the Fmax was set to 150 Hz at 3200 Lines, to produce a time waveform of 20 seconds. This was not ideal, as a longer time waveform would be required to confirm repetitious anomalies in the time waveforms, however the monitoring procedure would cause planning issues for the maintenance stop, and time was at a premium to survey the full set of rolls targeted. However, if we did spot something meriting further investigation we would be able to spend the time gathering much longer data sets as required.

I was therefore very surprised when I examined the data from one of the rolls to find a series of harmonic frequencies at very low frequency (see Figure 9). Once back on the computer the frequency matched the predicted defect frequencies for the bearing outer race, and arrangements were made to replace the bearing at a suitable stop to validate the data.

Inspection revealed that the outer race was spalled, with some cracking (see Figure 10), indicating that the survey had been able to detect an early Stage 3 bearing defect at frequencies well below the accepted limits for the sensor and the data collector. At present this survey is being conducted on a routine basis during maintenance stoppages.

**CONCLUSIONS**

I am not advising that the recommendations of sensor and monitoring equipment manufacturers are ignored or discounted, as the wisdom passed down through training has solid grounding in the physics of vibration monitoring and equipment limitations. I am also not suggesting that conventional early warning bearing techniques should not be used in conjunction with spectral and time waveform analysis, but I am advocating experimentation with available techniques, particularly when the equipment to be monitored is vital to manufacturing and is not monitored by other more viable techniques and systems.

However, once the initial investment in monitoring equipment has been made, there is little to be lost in trying to push the limits as far as possible, particularly when the alternative might be no monitoring at all. In the examples discussed above, I have simply looked for frequencies or anomalies which shouldn’t be present in a healthy machine, and taken little consideration of the overall amplitudes as a measure of health.

As ever, failure investigation to evaluate the success of the programme must take place. If the component is replaced too early in its fault progression it is still a fault saved, and the information gleaned lends support to the concept that the fault is detectable. This enables the analyst to assess the symptoms and allow for further service life at a subsequent detection. If the fault is missed, and cannot be seen in the data even with hindsight, then if the alternative was not to apply monitoring, nothing was lost in the effort. Nothing ventured, nothing gained!

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