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STEAM TURBINE CONDITION MONITORING BY VIBRATION ANALYSIS

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Abstract

A description is given of the technique of vibration analysis. Its value as a maintenance tool in the sugar industry is explained with examples of the type of faults which can be detected at an incipient stage without dismantling or taking production machinery off-line.

Introduction

APE Africa have been involved for many years in the supply and installation of steam turbines to the sugar industry and also provide service, repair and overhaul facilities for these machines.

In order to provide a more sophisticated approach to extend these facilities, a decision was taken in 1986 to provide a system of periodic machine condition monitoring to record the vibration signature of the machine, together with the main operational parameters.

An intensive survey was made of all available equipment in order to select the most suitable for the application, bearing in mind the requirements of portability and environmental conditions in a sugar mill.

The concept of vibration analysis monitoring

In the traditional approach to maintenance of steam turbines, the scheduling of repairs is difficult because the need for repair usually cannot be assessed without dismantling the machine. If a problem is serious enough to be readily apparent, damage has probably already occurred.

Without a means to assess machine condition by external examination, scheduling is inaccurate: machines in perfect working order are dismantled, whilst machines on the verge of failure may be ignored.

Modern technology provides a number of methods for externally determining the condition of machinery. The most effective of these is vibration analysis.

Any rotating machine, and especially a steam turbine, has a unique 'vibration signature' and any change in the machine condition will result in a change to this signature. By the regular monitoring of the vibration signature of the turbine, defects can be detected before they have a chance to cause extensive damage or failure. More importantly, the characteristics of the vibration are unique to the specific defect. By analysis of the vibration signal, the nature and location of the defect can often be determined.

The key advantage of this approach is that need for repair and the specific nature of any problems, can be assessed without any dismantling or even taking the machine out of service.

The implementation of machinery vibration analysis has been made practical by the development of devices known as Dynamic Signal Analysers (DSA). Machinery vibration is a complex combination of signals caused by a variety of internal sources of vibration. The power of Dynamic Signal Analysers lies in their ability to reduce these complex signals to their component parts.

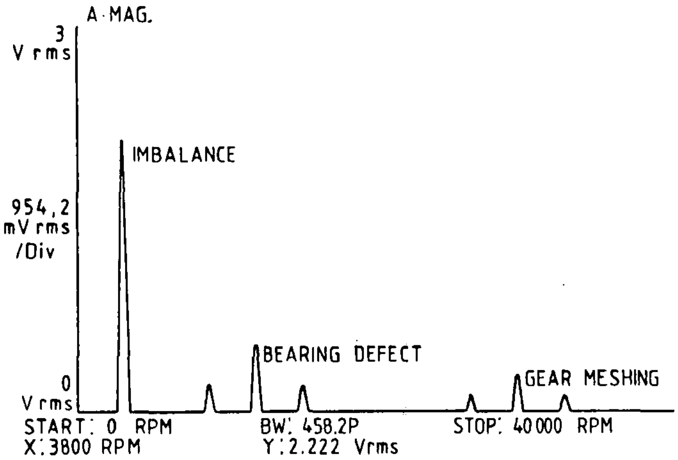

FIGURE 1 Typical complex vibration signal (amplitude against frequency

By displaying vibration amplitude as a function of frequency (the vibration spectrum) the DSA makes it possible to identify the individual sources of vibration. (Fig 1).

Dynamic Signal Analysers can also display vibration amplitude as a function of time, a format that is especially useful for investigating impulsive vibration, for example, from a chipped gear. (Fig. 2).

Another type of display is the spectral map format which adds a third dimension to vibration amplitude versus frequency displays. The third dimension is often rpm but can also be time or load - any variable that changes the vibration characteristics of the machine. (Fig. 3).

The DSA can be interfaced to a computer for automatic data storage and analysis.

The objective of maintenance is to keep machines running, especially those which are critical to production, for example, the turbo-alternators in a sugar mill. Unexpected catastrophic failures cause both loss of production and large repair bills. The classic maintenance strategy for avoiding such failures is to periodically dismantle critical machines for inspection and rebuilding. This can be an expensive exercise particularly when the machine is found to be in perfect working order and there is always a danger that the perfect order machine may be damaged during dismantling or reassembly.

It is obvious that a much more effective approach is to schedule repairs on the basis of machine condition as determined by vibration analysis supported by an audit of operating parameters, such as oil pressures, temperatures, etc.

This 'predictive' maintenance strategy can be readily applied to steam turbines, which are always equipped with the necessary pressure gauges, thermometers etc.

In a typical programme the overall vibration level is measured regularly and compared to previous readings or to established severity limits. In modern machinery of a critical

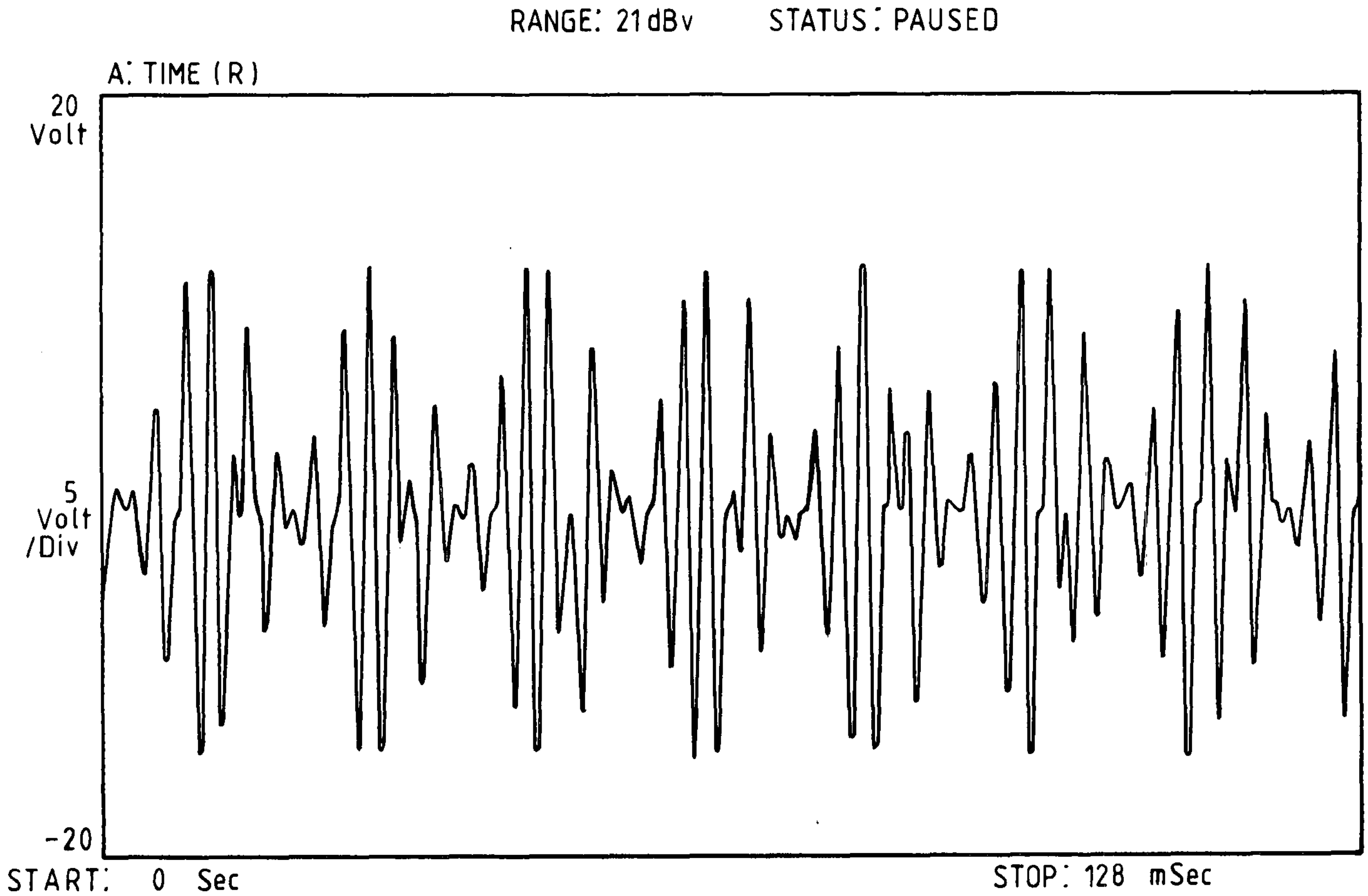

FIGURE 2 Typical complex vibration signal (amplitude against time)

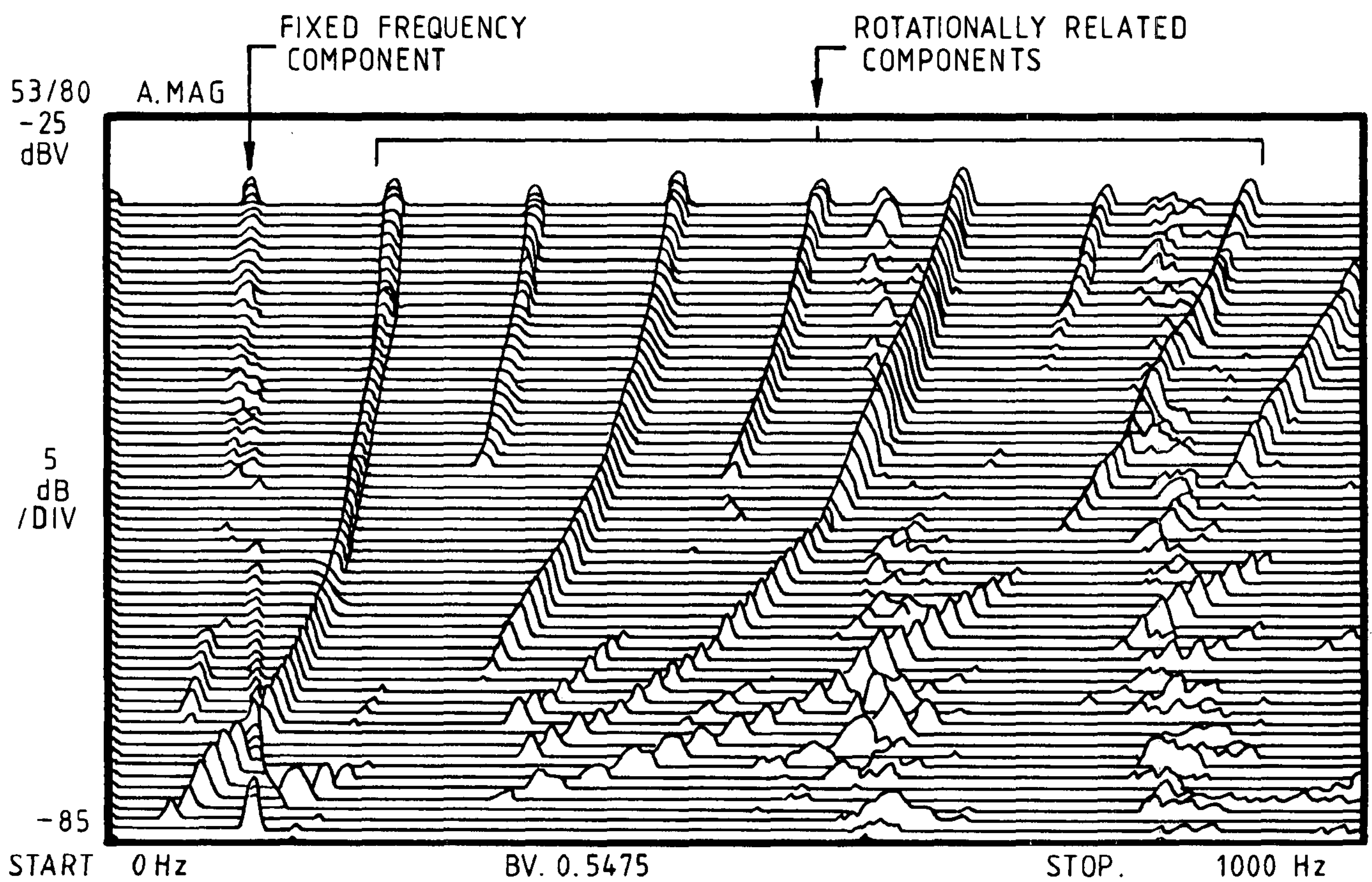

FIGURE 3 The spectral map

nature, the vibration level is often monitored on a continuous basis and compared against pre-set limits. If an excessive level is reached, the Dynamic Signal Analyser is used to determine the exact nature and location of the defect.

The DSA cannot do more than present a display or picture of the vibration characteristic and this display requires skilled interpretation and analysis. This analysis is a critical part of the process for several reasons.

(1) Overall vibration levels can change with load and operating speed, thereby presenting a misleading picture of machine condition. Analysis of the vibration spectrum indicates whether or not a serious problem exists. For instance some designs of steam turbine operate at rotational speeds in excess of the first critical speed. This means that when running-up it is necessary to pass through the critical speed band as quickly as possible.

(2) Taking a machine out of service for repairs during the crop period will inevitably have a major impact on production, so it is important to know how serious the problem actually is. Skilled analysis can assist in making the decision to shutdown immediately, or to take the machine through to the next scheduled maintenance period.

(3) Repair time is minimised because the nature of the problem is known beforehand. Most importantly, the necessary spares can be ordered in advance. This can be very significant in the case of older machines where spares may have to be specially made.

(4) All modern machines are subject to a series of vibration measurement tests at the manufacturer's works. Any abnormalities detected during the subsequent service life of the machine can be transmitted to the manufacturer's for comparison against the original, which can assist in the analysis of more obscure problems.

Equipment

The equipment selected by APE Africa, after lengthy investigation, includes a Dynamic Signal Analyser, Model 3561A by Hewlett Packard and an Olivetti M24 Personal Computer.

The DSA is readily portable and incorporates a non-volatile bubble memory capable of storing up to 127 separate traces. After visiting site and collecting the data relating to the particular machine, the DSA is returned to the office where the collected data is transferred to a 'floppy disc' for storage and analysis. The computer system also includes a Hewlett Packard plotter which can produce a hard-copy display of the vibration signature.

Once several successive sets of data have been collected on disc, these can be transferred back to the DSA in an overlay display, so that any changes are readily apparent.

Another facility is one enabling data from the storage disc to be replaced in the DSA Bubble memory. In this way, the DSA can be loaded with the previous trace or traces before being taken to the machine. At site, the latest trace can be compared immediately and any deviation immediately noted.

As any user of computers will know, the computer is useless without its associated software, to tell it what to do.

In the present case, several commercially available software packages were examined for viability. These packages were found to be expensive and none was felt to be suitable for the requirements. It was therefore decided to commission a computer scientist to design a suitable package.

This package named Vibration Analysis Survey (VAS) is a storage system for frequency domain traces and DSA set-ups. It maintains sufficient information to adequately identify this data. It does not duplicate any function of the DSA but merely supplements its storage and reporting capabilities.

The current version cannot interpret DSA data and is thus incapable of performing any analysis.

The functions provided by VAS may be catalogued as follows:-

- Management of discs and back-ups
- Management of traces
- Management of set-ups
- Management of identification data
- Creation of spectral maps
- Utilities

VAS considers each machine completely separately and there is thus no limit on the number of machines which may be studied. For each machine VAS maintains a set of discs, consisting of a main and a back-up disc. These are identical and provide protection of data in the event of damage to a particular disc.

Each set of discs is given a unique identity number and there is provision to record the machine serial number, the location in the plant, name of the owner and a comment. Each of these may be up to 80 characters in length. This information has no meaning to the VAS program and is purely to provide a means of identification to the various reports generated.

VAS provides facilities to create, modify, re-label and plot spectral maps which are actually drawn by the DSA. There are four ways to specify a map:-

- by giving individual trace numbers
- by giving a range of trace numbers
- by giving a range of dates
- by giving a range of configuration ID numbers

Another facility is that known as SNAP. The DSA has only a limited storage facility and a situation could arise where the bubble memory is fully loaded and the devise is needed to record more data. It is possible to snap an exact copy of the DSA bubble memory on to a floppy disc. The bubble memory can then be erased ready for further use.

To analyse the traces, the snap is re-written back to the DSA.

It must be emphasised that the VAS programme is not designed to carry out any analysis of data. Commercial programmes are available to carry out this analysis but these are very expensive.

It was decided to make use of the finest computer in existence to carry out the critical operation of analysing the data. This computer is the brain of a skilled and experienced turbine engineer.

Method

It is apparent from the previous comments that the whole process depends on electrical signals so that the first operation must be to convert the vibration of the machine to this electrical signal. This conversion is carried out by a device known as a transducer and these are defined by the parameter measured, namely displacement, velocity or acceleration.

It must be realised that the transducer cannot measure the vibration forces themselves but only the response of the machine to these forces. The mechanical impedances of the machine parts will determine how they respond to the vibration forces and this can alter the characteristics of the signal being measured.

Another influencing factor is the natural frequency of the structural parts of the machine under study.

It is not intended to enter into a highly technical discussion on these aspects in this paper and it is sufficient to say that an acceleration transducer or accelerometer was selected as the signal converter. The accelerometer is typically more rugged and offers a wider range of frequency response than other types. It is sensitive to mounting and should never be hand held.

In order to get consistent and repeatable readings it is essential that vibration readings be taken at the same position on the machine in consecutive measurements.

The method adopted is to bolt to the machine a number of machined cubes of steel to which the transducer may be attached by a high-strength magnetic base. These attachment points are located at each of the bearing housings of the machine.

The signal received from the vibration transducer is a complex combination of responses to multiple internal (and sometimes external) forces. In order to effectively analyse this complex signal it is necessary to reduce it to its individual components, each of which can then be correlated with its source.

Two analysis components are available for determining the components of vibration:

(1) The time domain view of vibration amplitude versus time, and

(2) the frequency domain view of vibration amplitude versus frequency.

Whilst the time domain provides insight into the physical nature of the vibration, the frequency domain is ideally suited to identifying its components.

An inherent advantage of the DSA is its ability to work in both domains.

When using the time domain and, in the case of a simple vibration, say a heavy spot in a wheel on a spindle running in two bearings, the result would be displayed as a sine wave. If this simple case becomes more complex, further vibration components will become apparent, also as sine waves. These sine waves will add and the result will be displayed as a complex wave in which the components are hidden. (Fig. 4).

When changing to the frequency domain, the various components of the complex vibration are separated and by analysis of the separate signals may be traced to their source.

A major advantage of frequency domain is that low level signals are easy to see, even in the presence of signals 1 000 times larger. The key to this capability lies in the use of a logarithmic (or decibel) scale for amplitude. This technique enables the detection of defects at an early stage, for example, the initiation of failure of a rolling element bearing.

Another analysis tool is the spectral map (Fig 3). The vibration characteristics of a machine depend on its dynamics and the nature of the defect. The change of these characteristics with machine speed has two implications for analysis (1) the vibration resulting from a defect may not appear in all speed ranges (2) insight into the nature of a defect may be obtained from observing the change in vibration with speed. Spectral maps are three dimensional displays that effectively show variation in the vibration spectrum with machine speed.

The most common method for this mapping is to measure successive spectra while the machine is coasting down or running up to speed. In addition to showing how vibration changes with speed, spectral maps quickly indicate which components are related to rotational speed. These components will move across the map as speed changes while fixed frequency components move straight up the map. This en-

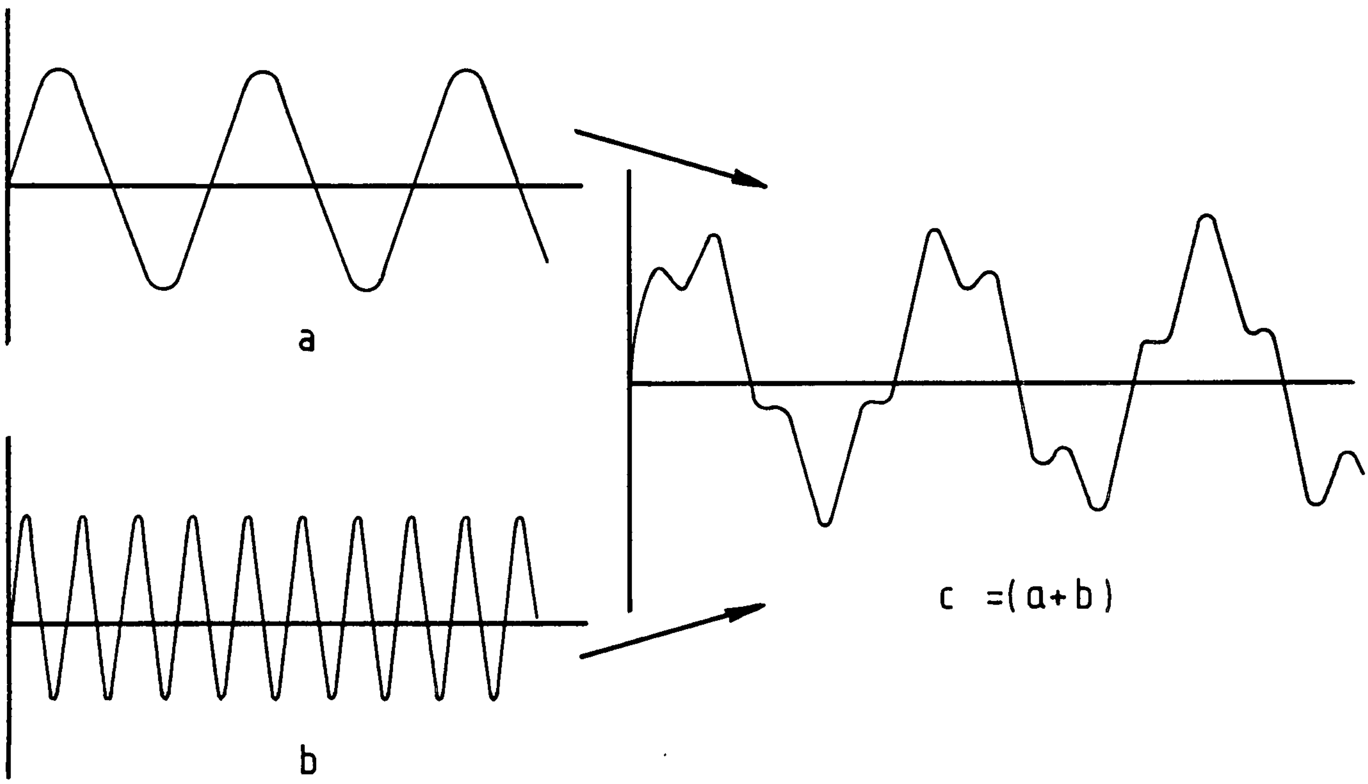

FIGURE 4 More complex imbalance (addition of sine waves)

ables the identification of machine resonances, which occur at fixed frequencies.

A further tool is the phase spectrum. The complete frequency domain representation of a signal consists of an amplitude spectrum and a phase spectrum. In machinery vibration analysis, phase is required for most balancing operations. With the addition of a Keyphaser TM, the DSA is able to determine all information required for balancing.

Vibration characteristics of common faults

Rotor imbalance

Rotor imbalance exists to some degree in all machines and is characterised by sinusoidal vibration of once per revolution. In the absence of high resolution analysis equipment, imbalance is usually blamed for any excessive once per revolution vibration - vibration that can be caused by several different faults. More detailed analysis can differentiate between these faults and eliminate unnecessary and expensive balancing operations.

A state of imbalance occurs when the centre of mass of a rotating system does not coincide with the centre of rotation. The imbalance can be in a single plane (static imbalance) or multiple planes (couple imbalance). In either case, the result is a vector that rotates with the shaft, producing the classic once per revolution vibration characteristic.

The key characteristics of the vibration caused by imbalance are (1) it is sinusoidal at a frequency of once per revolution (2) it is a rotating vector and (3) amplitude increases with speed. These characteristics assist in differentiating imbalance from faults that produce similar vibration.

The vibration caused by pure imbalance is a once per revolution sine wave sometimes accompanied by low-level harmonics. The faults commonly mistaken for imbalance usually produce high-level harmonics or occur at higher frequency. In general, if the signal has harmonics above once per revolution, the fault is not imbalance. However, high level harmonics can occur with large imbalance forces, or when horizontal and vertical support stiffnessess differ by a large amount.

Other faults are often mistaken for imbalance because they result in increased levels of vibration at running speed. Such faults are Misalignment, Load variation, Mechanical looseness, Resonance and Excessive clearance in fluid film bearings. Each of these does, however, have distinctive characteristics which can be used for identification (see below).

Rolling element bearings

Rolling element or anti-friction bearings are the most common cause of small machinery failure and overall vibration level changes are virtually undetectable in the early stages of deterioration. However, the unique vibration characteristics of rolling element bearings make vibration analysis an effective tool for both detection and analysis.

The major problem in detecting the early stages of failure in rolling element bearings is that the resulting vibration is low level and often masked by higher level vibration. If monitoring is performed with a simple vibration meter (or in the time domain) these low levels will not be detected and unpredicted failures are inevitable. A good solution is regular monitoring of critical machinery with a DSA, since the high resolution and dynamic range can reveal components as small as 1/1 000 the amplitude of higher level vibration.

An added advantage of early detection is that indications of the cause of failure, which may be obliterated in later stages, are still visible. An example of this would be false brinelling caused by excessive vibration on a stationary machine. By understanding the cause of problems such as this, the source of chronic failures can be determined.

Oil whirl in fluid film bearings

Rotors supported by fluid film bearings are subject to instabilities not experienced with rolling element bearings.

A basic difference exists between vibration due to oil film instability and vibration due to other faults, such as imbalance. The latter is a forced response to the imbalance force, occurs at the same frequency and is proportional to the size of the force. Oil film instability, on the other hand is a self-excited vibration that draws energy into vibratory motion that is relatively independent of rotational frequency.

This is a highly complex study and further comment is outside the scope of a general paper.

Misalignment

Vibration due to misalignment is usually characterised by a 2 times running speed component and high axial levels. Misalignment takes two basic forms, (1) preload from a bent shaft or improperly seated bearings and (2) offset of the shaft centre-lines of machines or parts of machines in the same train. Flexible couplings increase the ability of the train to tolerate misalignment but are not a cure for serious alignment problems.

Mechanical looseness

Mechanical looseness usually involves bearing mounting housings or bearing caps, and almost always results in a large number of harmonics in the vibration sprectrum. Components at integer fractions of running speed may also occur. Looseness tends to produce vibration that is directional, a characteristic that is useful in separating looseness from rotational defects such as imbalance. Measured vibration will be highest in the direction and vicinity of the looseness. One characteristic of looseness is that the basic sinusoidal waveform is truncated or flat-topped, where the looseness is restricted and taken up at the end of travel.

Gears

Gear problems are characterised by vibration spectra that are easy to recognise but difficult to interpret. This difficulty is due to two factors, (1) it is often difficult to mount the transducer close to the problem and (2) the number of vibration sources in a multi-gear drive result in a complex assortment of gear mesh, modulation and running frequencies. It is helpful to detect problems early through regular monitoring, since the advanced stages of gear defects are often difficult to detect. Baseline vibration spectra are helpful in analysis because high level components are common, even in new gearboxes. Baseline spectra taken when the gearbox is in good condition make it easier to identify new vibration components or components that have changed significantly in level.

Blades and vanes

Problems with blades and vanes are usually characterised by high fundamental vibration or a large number of harmonics near the blade passing frequency. Some components of blade passing frequency (number of blades × rpm) are always present and levels can vary markedly with load. This is especially true for high speed turbo-machinery and makes recording of operating parameters for historical data critical. It is important to establish baseline spectra for several operating levels.

If a blade is missing, the result will be simple imbalance, resulting in high levels of one per revolution vibrations. Detection of cracked blades is more difficult as blade vibration cannot be measured directly. A cracked blade is often characterised by a large number of harmonics around blade passing frequency.

Resonance

Problems with resonance occur when natural frequencies of the shaft, machine housing or attached structures are excited by running speed or harmonics of running speed. These problems are usually easy to identify because levels drop appreciably when running speed is raised or lowered. Spectral maps are especially useful in detecting resonance vibration because the strong dependence on running speed is readily apparent. Piping is one of the most common sources of resonance problems. When running speed coincides with a natural frequency, the resulting vibration will be excessive and strain on the pipe and machine will be excessive and can lead to early failure.

The remedy can often be simple by adding stiffness in the form of a support thus changing the natural frequency.

The same effect can be apparent in any attached structure, for example, the support structure for the machine. If this is a fabricated steel structure, its natural frequency can be readily changed by adding stiffeners. In the case of a concrete foundation, the problem is much more complex and any design for a turbine block should always be examined for possible resonance conditions. It is also possible to encounter problems of torsional resonance in shafts and this should be examined when coupling drivers and driven machines from two different suppliers.

Electric motors

Excessive vibration in electric motors can be caused by either mechanical, or electro-magnetic defects. The latter can be quickly isolated by removing power when vibration caused by electrical and magnetic defects will disappear.

Documentation

Much of the basic analysis of vibration spectra can be performed by simple comparison between historical baseline data and current data. Thorough documentation therefore provides, in many cases, the information required to successfully analyse a vibration problem.

Complete documentation will include baseline data maintenance history and engineering data. The baseline data should preferably be established on a new machine or where this is impossible on the machine after its complete overhaul, when it condition is accurately known.

When taking measurements to establish this baseline data, the tests should be as complete as possible and should ideally be repeated several times over a period of time and the results subjected to a statistical analysis to yield mean and standard deviations. This will provide a representative average level and also provide a quantitative basis for determining whether a change in level is significant. The accuracy of these statistics can be improved by updating with data from regular vibration monitoring and especially after major repairs.

The use of the computer for data storage and manipulation aids tremendously in carrying out this documentation but the quality and effectiveness of the overall vibration analysis programme is determined primarily by the calibre of the people involved.

Conclusion

In conclusion APE Africa are confident that this powerful technique for steam turbine condition monitoring with the most sophisticated equipment and skilled turbine engineers will offer the sugar industry a new approach to turbine maintenance and result in longer periods between major maintenance overhauls. Regular Machine Condition Monitoring (MCM) will also simplify the scheduling of planned maintenance, provide advance information to enable ordering of spares and to reduce the overall level of risk to the machine. In respect of this last comment, approaches to a major insurance broker have produced guarded reaction that whilst regular MCM may not result in reduced insurance premiums, it may delay the introduction of increases in premium.

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