TABLE 3. TRENDS IN EXPORTS OF CPO AND PPO (x 1000 TONNES) (Statistics Dept., K.L.)

	1975	1980	1986p
CPO exports	942	198	113
PPO exports*	212	2,073	4,441
PPO exports (% of total exports)	18.4	91.3	97.5

* excludes oleochemicals but includes PFAD/PAO

p provisional

TABLE 4 : EXPORT MIX OF PPO PRODUCTS (x 1000 TONNES) (Statistics Dept., K.L.)

	1979	1982	1986p
Palm oil related	501	1,142	1,647
Palm olein related	495	1,021	1,849
Palm stearin related	455	467	691
By-products (refining)	118	177	254

P Provisonal

(T Thiagarajan)

• MAINTENANCE

PREDICTIVE MAINTENANCE IN THE PALM OIL INDUSTRY : PART TWO

The series on Predictive Maintenance in the Palm Oil Industry is continued by discussing intermediate and advance level predictive maintenance using narrow band frequency analysis.

INTERMEDIATE AND ADVANCED LEVEL PREDICTIVE MAINTENANCE USING NARROW BAND FREQUENCY ANALYSIS

Intermediate and Advanced Level Instrumentation

Fault detection at an earlier stage together with diagnosis and breakdown prediction become possible when using a system which can perform frequency analysis. Full frequency analysis and spectrum plot-out is performed on the spot for each monitoring point.

Intermediate level instrumentation comprises of a portable vibration analyzer and level recorder. Here current spectra are compared manually with recorded reference spectra to reveal tell-tale increases in the level of individual frequency components.

Advanced level instrumentation comprises of a real-time FFT vibration analyzer, a portable tape recorder and a microcomputer system with machine health condition-monitoring software. Here vibration signals from each machine are collected in the field on an instrumentation tape recorder and automatically compared with reference spectra back at the office. Advanced computer programs aid fault diagnosis and trend monitoring.

Theory of Machine Condition-Monitoring

What is Frequency Analysis?

Single wideband vibration measurements are only a rough indication of machine health. Vibration signals in practice usually comprise of numerous frequencies occuring simultaneously. These frequency components can be revealed by plotting vibration amplitude against frequency. Some frequency components in the vibration spectrum, can be related to shaft rotational speeds, gear tooth-meshing frequencies *etc.* The breaking down of vibration signals into individual frequency components in machine fault diagnosis is called frequency analysis. For this purpose we use a filter which passes those parts of the vibration signal which are contained in a narrow frequency band.

There are two basic types of filters used for frequency analysis - constant bandwidth and constant percentage bandwidth.

Choice of Scale

For predictive maintenance purposes, there is a need for wide dynamic and frequency ranges. Every part of a frequency spectrum is essential for machine condition monitoring and must be measured with adequate accuracy. For fault detection purposes, it is usually best to express the spectrum on a logarithmic frequency scale with constant percentage bandwidth. Logarithmic scale for frequency has the effect of expanding the lower frequencies and compressing the higher frequencies thus giving the same percentage resolution over the whole width of the spectrum plot keeping its size down to reasonable proportions. To detect the spectral changes due to a wide range of differing faults, three decades (1000 : 1) would typically be required for complex machines. Mechanical systems tend to have much of their vibration energy contained in the relatively narrow frequency range of between 10 Hz to 1000 Hz but measurements are often made up to 10 kHz because there are often useful information about machine health at such high frequencies.

For diagnostic purposes, it is most often best to express the spectrum on a linear frequency scale with constant bandwidth, the normal presentation of an FFT analyzer. This facilitates the visualization of harmonic or sideband patterns which are important factors in diagnosis. The best solution of bandwidth and analysis method is that which gives adequate resolution over the whole frequency range and allows analysis in the shortest time.

Both linear and logarithmic amplitude scales are in common use, but only logarithmic amplitude scales allow vibration components within a sufficiently wide range to be measured with adequate accuracy. The ratio between the maximum and minimum voltage levels which can be measured by a spectrum analyser within a single attenuator range, is referred to as the dynamic range. In practice a dynamic range of 300 : 1 to 10 000 : 1, *i.e.* 50 to 80 dB, will accomodate a sufficiently wide range of frequency components for machine condition monitoring. When a frequency spectrum is plotted on a linear amplitude scale only the predominant peaks will usually be within the allowable accuracy. By using decibels, the considerable numerical span of a logarithmic scale is reduced to a concise linear numbering system such that a factor of 10 : 1 on a logarithmic scale is equivalent to a 20 dB interval. In order to quote absolute vibration levels, the reference level must be stated.

Choice of Vibration Measurement Parameters

Most modern vibration meters are equipped to measure all three parameters; displacement, velocity and acceleration. The choice of parameter is that which gives the flattest frequency spectrum so as to best utilize the dynamic range of the instrumentation. Acceleration is used usually for measuring high frequency vibration components while displacement is used for measuring low frequency components.

The best parameter to use for comparison of machine vibration spectra is usually the vibration velocity expressed on a logarithmic or decibel (dB) scale.

Choice of Transducers

Generally, the three types of transducers used have the following dynamic ranges:

The relative displacement probe dynamic range is 100 : 1

The velocity pickup dynamic range is 1 000 1, and

The piezoelectric accelerometer dynamic range is 30 000 000 : 1

As mobility can vary by 1 000 : 1, the only practical transducers for the purpose of predictive maintenance is the piezoelectric accelerometer. The piezoelectric accelerometer has wide frequency and dynamic ranges with good linearity throughout the ranges. It is robust and reliable so that its characteristics remain stable over a long period of time. When the piezoelectric material is mechanically stressed (in tension, compression or shear) it generates an electrical charge across its pole faces which is proportional to the applied force. Accelerometer is a sensor whose electrical output is proportional to acceleration.

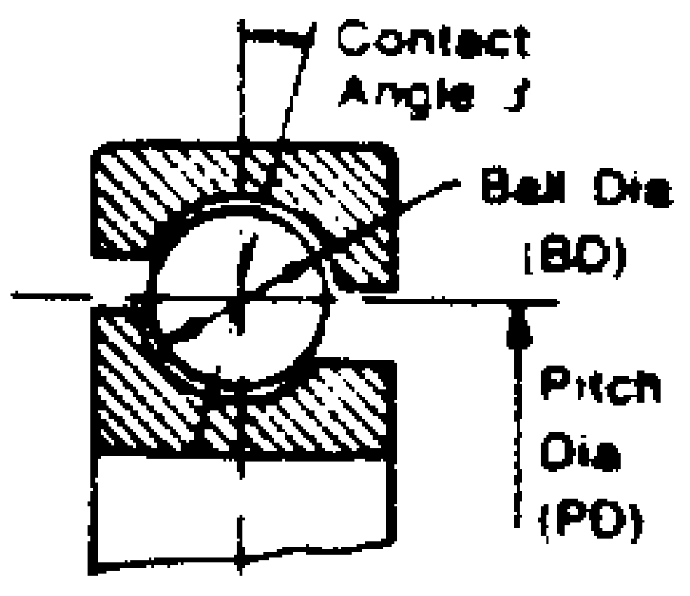
Mounting of a Transducer

This is one of the most critical factor in obtaining reliable and accurate results from practical vibration measurements. The accelerometer should be mounted so that the desired measuring directions coincides with its main sensitivity axis. Sensitivity to vibrations in the transverse direction is slight and can normally be disregarded as it is typicallyless than 1% of the main axis sensitivity. With small accelerometer where the mass is small, the resonant frequency can be as high as 180 kHz, but for the larger general purpose accelerometer resonant frequencies of 20 to 30 kHz are typical.

As a rule of thumb, if we set the upper frequency limit at 1/3 of the accelerometer resonance frequency, vibrations measured at the upper frequency limit would be in error by ± 12%.

For permanent measuring points on machines, mounting studs are usually used. The studs are attached to the measurement points by means of a hard glue *e.g.* epoxy, cyanoarcrylate, *etc.* A general purpose accelerometer using this fixing method will have resonant frequency of almost as high as 30 kHz. A permanent magnet type of attachment will reduce this resonant frequency to about 7 kHz and hence cannot be used for measurements much above 2 kHz. The holding force of the magnet is adequate for vibration levels up to 1 000 to 2 000 ms^{-2} depending on the size of the accelerometer. A hand-held probe with an accelerometer mounted on top is very

TABLE 1 : DIAGNOSTIC VIBRATION TROUBLE-SHOOTING CHART

Nature of Fault	**Frequency of Dominant Vibration (Hz=rpm/60)**	**Direction**	**Remarks**
Rotating Members out of Balance	1 × rpm	Radial	A common cause of excess vibration in machinery
Misalignment & Bent Shaft	Usually 1 × rpm Often 2 × rpm Sometimes 3&4 × rpm	Radial & Axial	A common fault
Damaged Rolling Element Bearings (Ball, Roller, etc.)	Impact rates for the individual bearing component* Also vibrations at high frequencies (2 to 60 kHz) often related to radial resonances in bearings	Radial & Axial	Uneven vibration levels, often with shocks. * Impact-Rates:  **Impact Rates f (Hz)** For Outer Race Defect $f(Hz) = \frac{n}{2} f_r \left(1 - \frac{BD}{PD} \cos \beta\right)$ For Inner Race Defect $f(Hz) = \frac{n}{2} f_r \left(1 + \frac{BD}{PD} \cos \beta\right)$ For Ball Defect $f(Hz) = \frac{PD}{BD} f_r \left[1 - \left(\frac{BD}{PD} \cos \beta\right)^2\right]$ n = number of balls or rollers f_r = relative rev./s between inner & outer races
Journal Bearings Loose in Housing	Sub-harmonics of shaft rpm, exactly 1/2 or 1/3 × rpm	Primarily Radial	Looseness may only develop at operating speed and temperature (e.g. turbomachines).
Oil Film Whirl or Whip in Journal Bearings	Slightly less than half shaft speed (42% to 48%)	Primarily Radial	Applicable to high-speed (e.g. turbo) machines.
Hysteresis Whirl	Shaft critical speed	Primarily Radial	Vibrations excited when passing through critical shaft speed are maintained at higher shaft speeds. Can sometimes be cured by checking tightness of rotor components.
Damaged or Worn gears	Tooth meshing frequencies (shaft rpm × number of teeth) and harmonics	Radial & Axial	Sidebands around tooth meshing frequencies indicate modulation (e.g. eccentricity) at frequency corresponding to sideband spacings. Normally only detectable with very narrow-band analysis and cepstrum
Mechanical Looseness	2 × rpm		Also sub- and interharmonics, as for loose Journal bearings
Faulty Belt Drive	1, 2, 3 & 4 × rpm of belt	Radial	The precise problem can usually be identified visually with the help of a stroboscope
Unbalanced Reciprocating Forces and Couples	1 × rpm and/or multiples for higher order unbalance	Primarily Radial	
Increased Turbulence	Blade & Vane passing frequencies and harmonics	Radial & Axial	Increasing levels indicate increasing turbulence
Electrically Induced Vibrations	1 × rpm or 1 or 2 times synchronous frequency	Radial & Axial	Should disappear when turning off the power

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convenient for a quick-look survey but a low pass filter should be used to limit its measuring range at about 1 000 kHz. Repeatable results cannot be expected with hand-held probes.

As a general rule, the accelerometer mass should be no more than 1/10th of the dynamic mass of the vibrating part on to which it is attached. For measurement of dynamic range, the lower limit is determined by electrical noise from connecting cables and amplifier circuitry while the upper limit is determined by the accelerometer structural strength. A typical general purpose accelerometer is linear up to 50 000 to 100 000 ms^{-2} well into a range of mechanical shocks.

Application of Frequency Analysis in Machine Condition-Monitoring

Simple vibration meters measure the overall vibration level over a wide frequency range which reflects the vibration levels of predominant frequency components. Although such overall vibration levels are important to monitor, wide-band measurements are rather limited with respect to early fault detection, diagnosis and breakdown prediction when compared to frequency analysis.

Fault Detection in the Early Stage of Development

The detection of important individual frequency components in a spectrum at an early stage would require frequency analysis. As the analysis bandwidth is reduced a more detailed spectrum which separates individual frequency component peaks is obtained. In general, the narrower the analysis bandwidth, the earlier can developing faults be detected but, on the other hand, the longer will be the analysis time unless more sophisticated instrumentation is used.

It is often found that the higher frequencies contain information on faults developing well before they influence the actual ability of the machine to do its job, whereas lower frequencies show the faults when they have occurred. To predict breakdowns, the higher frequencies therefore become very important.

Fault Diagnosis

The types of faults which can be diagnosed from a frequency analysis range from fairly simple, such as unbalance, to fairly complex, such as local tooth faults in gearboxes. It is sometimes found from experience that certain faults in individual machines show up in a specific way in the vibration spectrum, but it is difficult to write anything general about these cases.

Not only do vibration level increases in frequency components give an early indication of fault development, but the specific frequency at which they occur indicated which type of faults or machine parts are deteriorating. Table 1 is a diagnostic vibration trouble-shooting chart which gives the most important characteristics of some typical types of machine faults. However, because of the brevity of the information in the chart provided, it is important to realise that the situation is not quite as clear as might appear.

Breakdown Prediction

The increase in vibration level of one or more individual frequency components recorded over a number of periodic measurements when plotted allows the trend in the development of faults to be followed. Extrapolation forward in time will indicate when the machine condition will reach a critical limit and breakdown becomes imminent. Maintenance can thus be scheduled for a convenient time.

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