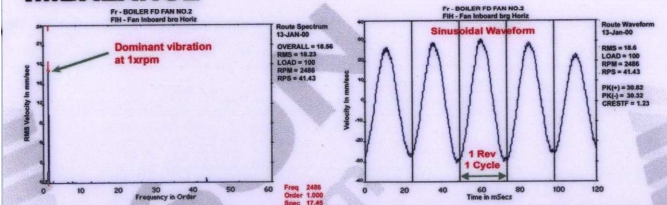
VIBRATION ANALYSIS CHART

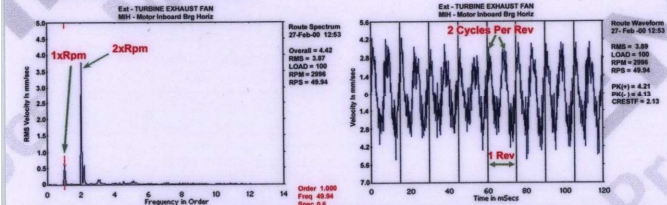
UMAR ALI KHAN

IMBALANCE



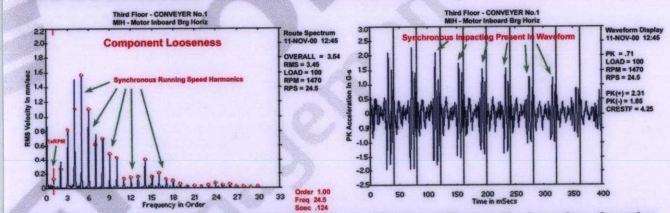
Imbalance occurs when the centre of mass differs from the centre of rotation, resulting in a heavy spot on a rotor. This heavy spot produces a centrifugal force causing high vibration amplitudes at a frequency equal to 1xRPM (1x rotational speed) in spectral data and a sinusoidal waveform in the time domain. On rigidly mounted machines amplitudes will typically be highest in the horizontal direction versus the vertical direction, although raised amplitudes can also exist axially on overhung rotors. The imbalance amplitude will increase by the square of speed (when the running speed is below the 1st rotor critical frequency). Typically a 90° phase difference will exist between horizontal and vertical readings at each bearing with a pure imbalance problem.

MISALIGNMENT



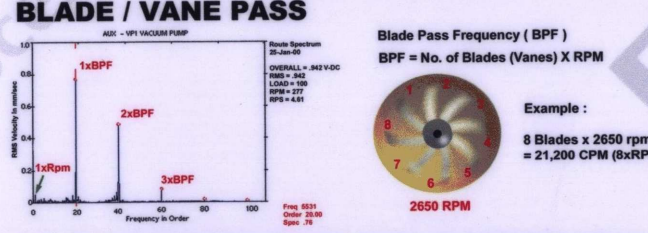
Misalignment occurs when the shaft centrelines of two directly mating components meet at angles and/or are offset from one another. Misalignment of couplings and bearings typically results in high radial and/or axial vibration. Angular misalignment typically produces raised axial amplitudes with a phase difference of 180° axially across couplings. Offset misalignment typically produces high radial amplitudes with a phase difference of 180° radially across couplings. A misalignment problem will typically produce dominant frequencies at 1xRPM and/or 2xRPM, depending upon the degree of angular misalignment versus offset misalignment, as well as the type and design of the couplings. In situations where 1xRPM dominates due to a misalignment problem, phase readings are essential to distinguish from an imbalance problem. Waveforms typically show repeatable, periodic patterns with one or two clear cycles per-shaft revolution.

LOOSENESS *(Structural & Component)*



Mechanical Looseness occurs when rotating components do not fit correctly. A number of running speed harmonics will dominate spectral data (i.e. 1x to 10xRPM). In certain instances and stages of deterioration, spectral data may also show sub-harmonic multiples of ½xRPM. Waveforms typically show high random impacting patterns, and in some instances the waveform may appear truncated. Structural Looseness is the result of loose or distorted machine feet (soft foot), loose foundations, weak grouting, loose bolts or bedplates. These conditions typically produce a frequency at 1xRPM and/or 2xRPM radially in spectral data (vertical amplitudes may be more predominant), depending upon the type of structural problem. Waveform data is typically periodic, with one or two cycles per-shaft revolution.

BLADE / VANE PASS



Blade Pass Frequency (BPF)

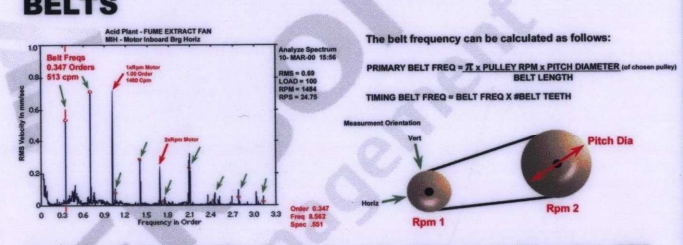
BPF = No. of Blades (Vanes) X RPM

Example :

8 Blades x 2650 rpm
= 21,200 CPM (8xRPM)

Any machines which handle fluids such as water, gases, air and oil will typically produce an inherent vibration frequency (plus potential harmonics) equal to; Number of Blades or Vanes x RPM. This inherent machine characteristic is normally referred to as a Blade or Vane Pass Frequency (BPF or VPF). Excessive BPF or VPF amplitudes can result from uneven or worn distributed gaps between rotating vanes and static diffusers. Should a BPF or VPF coincides with a system natural frequency then extremely high destructive amplitudes will be experienced.

BELTS



The belt frequency can be calculated as follows:

PRIMARY BELT FREQ = $\frac{\pi \times \text{PULLEY RPM} \times \text{PITCH DIAMETER}}{\text{BELT LENGTH}}$ (of chosen pulley)

TIMING BELT FREQ = BELT FREQ X #BELT TEETH

The Primary Belt Frequency (PBF) is equal to the number of times per sec (Hz) or minute (CPM) that a belt rotates around its drive system, which can be calculated from the formula above. Belt problems, which produce PBF and/or harmonics include; wear, cracks, mismatching, incorrect tension, hard or soft spots. It is not unusual to see PBF harmonics in spectral, and often 2x PBF can be the most dominant peak. The highest amplitudes will typically be radial and in-line with belts. Misaligned pulleys/belts normally produce a 1xRPM frequency axially.

ROLLING ELEMENT BEARINGS

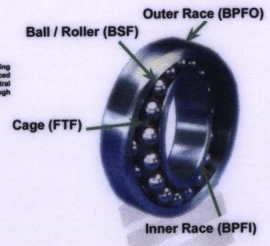
4 Stages to Rolling Element Bearing failure

Mode 1
The first sign of an early bearing problem appears in the ultrasonic range (i.e. up to 20kHz and even higher). A number of ultrasonic measurement techniques are available today, the response and repeatability of which depends upon sensor specification, measurement location, mounting technique, speeds and bearing design.

Mode 2
Very minor defect frequencies begin to "ring" bearing component natural frequencies. Typically higher frequency multiples of the fundamental bearing defect frequencies will also be present. Ultrasonic values also continue to rise.

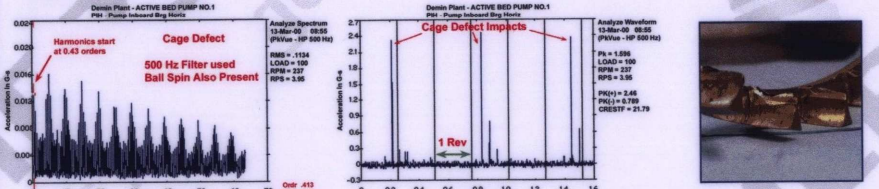
Mode 3
Bearing defect frequencies and related harmonics will then appear. When wear progresses, more defect frequency harmonics appear and the numbers of sidebands grow around bearing defect and natural frequencies. Ultrasonic levels also continue to rise. The bearing should be replaced at this stage.

Mode 4
Bearing clearances then begin to increase potentially causing discrete bearing defects and component natural frequencies to disappear. These are then replaced by random and broadband high frequency noise resulting in a raised spectral noise floor. Ultrasonic levels can actually drop during this failure stage, although with failure eminent they will increase to excessive levels again.



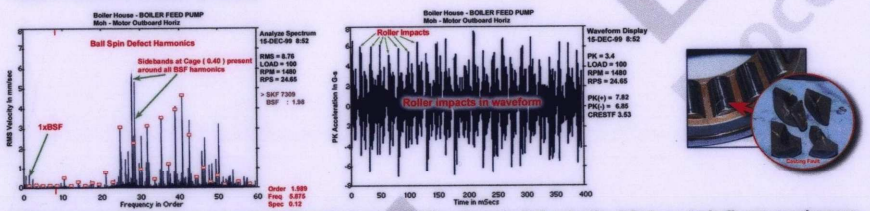
FTF = Fundamental Train Frequency
BSF = Ball Spin Frequency
BPFO = Ball Pass Frequency Outer
BPFI = Ball Pass Frequency Inner

Cage Defects (FTF)



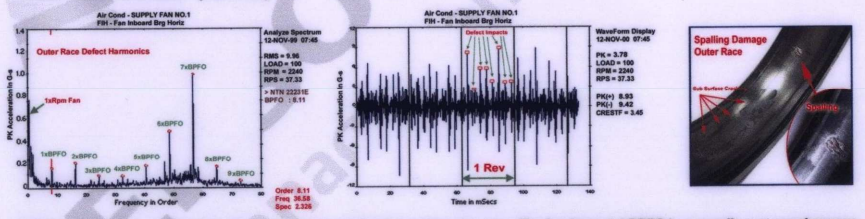
Fundamental cage defect frequencies normally occur between 0.35–0.48xRPM (sub-synchronous energy). Cage defects are usually accompanied by other defective bearing components, such as ball/roller defects and appear as sideband frequencies. Cage defects often cause rapid bearing failures, making them more difficult to detect. Cage defects rarely produce high amplitudes and could potentially be confused with other problems, therefore it is important to capture high resolution conventional data, as well as using advanced supplementary bearing monitoring techniques if available.

Ball / Roller Defects (BSF)



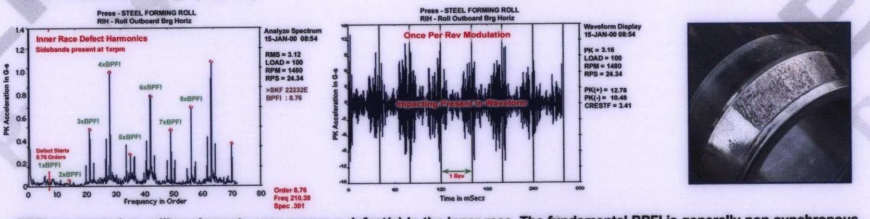
BSF appear when damage occurs to the surface of a bearing's rolling elements. Ball and roller defects are typically non-synchronous frequencies, which can appear at the fundamental BSF or 2xBSF with associated harmonics, that may be larger in amplitude depending upon the number of damaged rolling elements. It is common to see the fundamental BSF or 2xBSF and harmonics surrounded by cage (FTF) sideband frequencies. BSF defects can also be assessed from the time waveform where impacts equal to the fundamental BSF or related predominant BSF harmonic(s) may be present.

Outer Race Defects (BPFO)



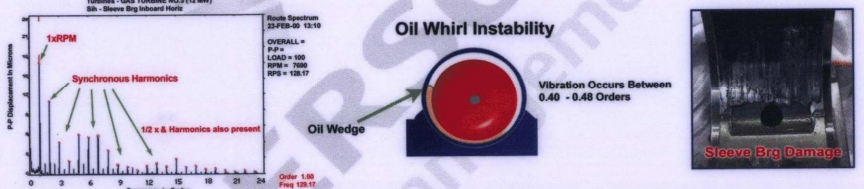
BPFO are created as rolling elements pass across a defect(s) on the outer race. The fundamental BPFO is generally non-synchronous, although on some bearings (depends on geometry) this is close enough to appear as a synchronous frequency. This problem typically shows several harmonics of the fundamental BPFO, which are often higher in amplitude. Where the outer race rotates and inner race remains stationary, BPFO frequencies can be modulated by the shaft speed, resulting in RPM sidebands surrounding BPFO defects. Time waveforms typically show distinctive high impact levels (g's), spaced at the fundamental BPFO or related dominant harmonic(s).

Inner Race Defects (BPFI)



BPFI are created as rolling elements pass across a defect(s) in the inner race. The fundamental BPFI is generally non-synchronous, although on some bearings this is close enough to appear as a synchronous frequency. Spectral and waveform characteristics for BPFI problems are very similar to that defined for BPFO, although a definitive difference is 1xRPM sideband frequencies will typically surround the fundamental BPFI and/or harmonics due modulation by the shaft speed (reverse applies where inner race remains stationary). A once per rev modulation of the time waveform may also occur as a BPFI fault passes through the load zone, making the waveform appear similar to "a school of angel fish".

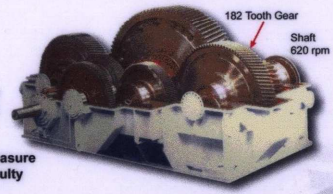
SLEEVE BEARINGS



Sleeve bearings with excessive clearance allow small excitation forces, such as imbalance, to cause significant vibration amplitudes. Spectral data typically shows predominant synchronous frequencies between approximately 1x to 10xRPM, depending on problem severity, bearing design and application (in some cases ½xRPM and harmonics may also apply). Wiped sleeve bearings can reveal higher vertical amplitudes compared with horizontal, and may show a pronounced 1xRPM. Oil whirl occurs when the oil film in pressure-lubricated systems exerts a force that pushes the shaft around within the bearing. This problem typically occurs between 0.4x to 0.48xRPM, and may result from improper bearing design, oil viscosity or pressure.

GEAR DEFECTS

- Gearmesh Frequency :
  No. of Teeth X Shaft Turning Speed = GMF
- One GMF per pair of gears
- Gearmesh Freq will always be present regardless of gear condition. Amplitude may increase due to load or gear condition.
- Sidebands are the key to pinpointing gear faults, measure the spacing of sidebands to show which shaft the faulty gear is on.



Gearmesh Example:

To calculate the gearmesh for a pair of mating gears:

GMF = 620 rpm x 182 teeth
= 112,840 CPM (1880 Hz)

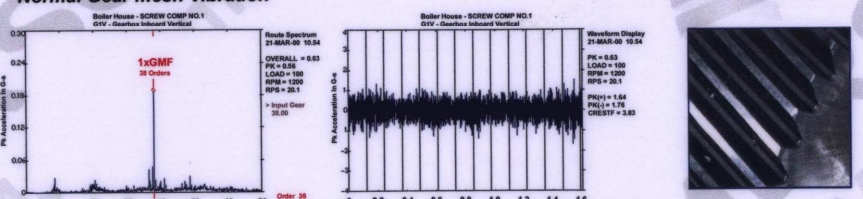
GMF x 1 = 112,840 CPM (1880 Hz)

GMF x 2 = 225,689 CPM (3761 Hz)

GMF x 3 = 338,529 CPM (5642 Hz)

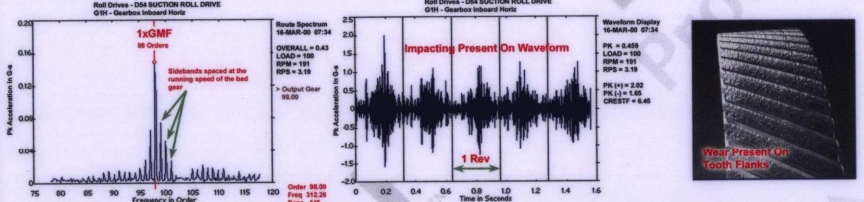
CPM: (Cycles Per Minute)
Hz: (Cycles Per Second)

Normal Gear mesh Vibration



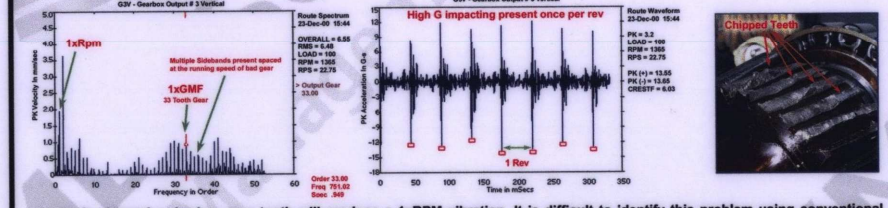
Fundamental gear mesh frequencies (GMF) are calculated by # Gear Teeth x Shaft RPM (apply the same calculation to every shaft in the gear train). However, the fundamental GMF by itself doesn't normally represent a problem, due to the inherent nature of this frequency being present in virtually all gear systems. Changes in fundamental GMF amplitudes may typically be the result of varying loads. Waveforms should appear relatively steady with little sign of impacting, although amplitudes will vary with load. It is typical to set the spectral frequency range to at least 3.5x the fundamental GMF (where high speed shafts and high GMF exist, ideally use maximum resolution).

Worn Gears



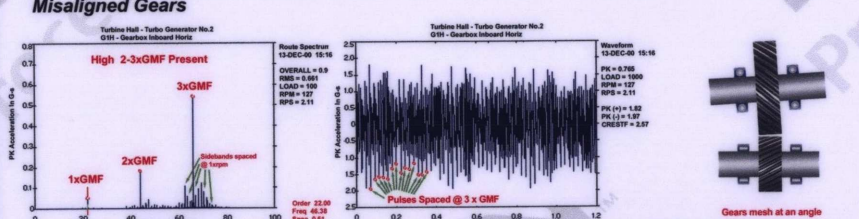
A key characteristic of worn gear(s) is the presence of sidebands, which show up as frequencies equally spaced either side of the GMF or related harmonics. These sideband frequencies will be equal to the rotational speed(s) of the shaft(s) with the defective gear(s) on. Not only is the presence of sidebands surrounding the GMF important when analysing worn gears, but the amplitudes of these sidebands in relation to the GMF. The higher the sideband amplitudes, the more severe the gear wear is likely to be. Worn gears can also excite the natural frequencies of the gears and/or casing.

Chipped / Broken Gear Teeth



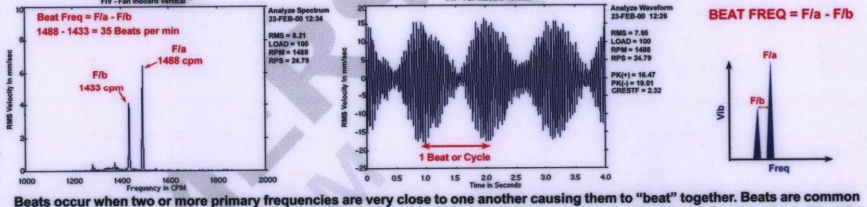
A single chipped or broken gear tooth will produce a 1xRPM vibration. It is difficult to identify this problem using conventional spectral data alone, as the characteristic frequency can easily be confused with other problems occurring at 1xRPM. The best way of identifying this problem is by using the time domain, where a once per revolution (1xRPM) impact will be generated followed by a ring down from the shaft with the defective gear tooth on. Due to the impacting nature of this fault the time waveform is typically taken in acceleration, ensuring that the time period set-up captures as many shaft revolutions as possible.

Misaligned Gears



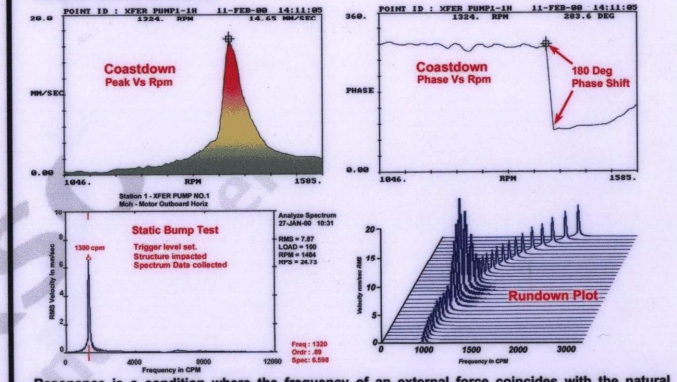
Misaligned gears will generally produce raised vibration amplitudes at 2xGMF and 3xGMF, which will typically be sidebanded at 1xRPM or 2xRPM. Therefore, it is essential to always capture at least 3.5xGMF when either troubleshooting or taking routine vibration data. In general terms viewing both the time waveform and spectral data in acceleration provides the best representation of gear misalignment problem, particularly where the 1xGMF is greater than 1.5kHz (90,000cpm). Where gear misalignment exists this is likely to be accompanied by a degree of gear wear, especially if the problem has been present for sometime.

BEAT VIBRATION



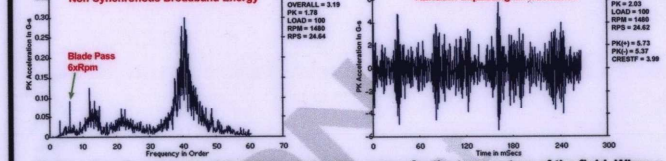
Beats occur when two or more primary frequencies are very close to one another causing them to "beat" together. Beats are common where rotational speeds are running close together, such as a belt driven motor and pump set with a 1:1 ratio, where the only difference between component speeds is slip. The beat frequency itself is the difference between these primary frequencies. Beats usually cause vibration amplitudes to increase and decrease in an equally timed rhythm, and maximum amplitudes occur when these primary frequencies come into phase with one another. When a beat is suspected it is important to capture a high-resolution or true zoom spectrum to confirm the problem.

RESONANCE



Resonance is a condition where the frequency of an external force coincides with the natural (resonant) frequency of a system, resulting in dramatic (often disastrous) amplitudes. Using peak/phase analysis a 180° phase shift can be observed as a system approaches, enters (90° shift here) and passes through a resonance. Taking 2/3-D run-up/coast downs also allow resonant frequencies to be analysed. Static impact testing allows resonant frequencies to be identified, by impacting a system component(s) and measuring the resultant frequency response (use negative linear averaging when machine cannot be stopped). When resonance is suspected, look for dramatically higher amplitudes in one axis versus others. Changing a systems damping, mass or stiffness will alter its natural frequencies, alternatively move operational speeds away from resonant frequency regions. The speed should ideally be 30% higher or lower than the critical speed.

CAVITATION / RECIRCULATION



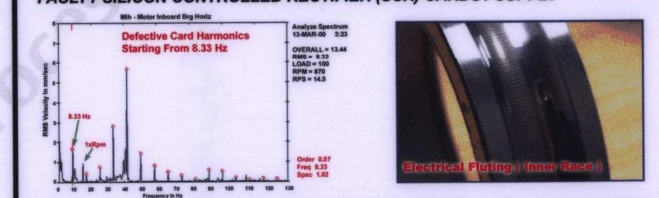
Occurs when the pressure of fluid drops below the vapour for the temperature of the fluid. When a system or localised pressure drop occurs, voids or cavities (bubbles) will form in the liquid. These bubbles implode or collapse when the fluid moves through the impellor to the high pressure side of the pump, causing the impellor to erode. Spectra is typically dominated by high frequency broadband energy present above the impellor blade pass frequency, with random impacting in the time waveform. Recirculation occurs in the presences of a high suction pressure that actually pulls the fluid from the output back through the pump. Recirculation is a hydraulic source of 1xRPM, as well as sub-synchronous broadband noise.

ELECTRICAL RELATED FAULTS

GENERALISED STATOR FAULTS

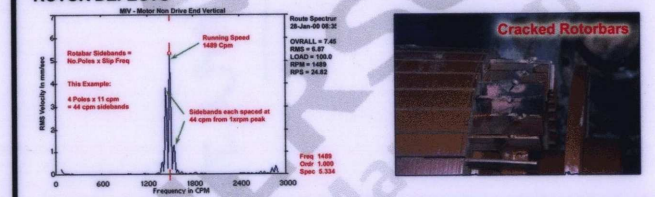
Presence of low amplitude electrically related line frequencies (Lf) are not uncommon in vibration data, particularly at 100Hz (2xLf) and occasionally at 50Hz (1xLf) (120Hz or 60Hz in some countries). However, where raised amplitudes exist due to an electrical problem spectral and waveform can be used to identify the cause. Electrical problems occurring at 2xLf include; stator eccentricity, lamination shorting, loose stator cores, loose laminations, loose coils, loose connections and soft foot resulting in a distorted stator air gap. Where an electrical frequency could be confused with a running speed or related harmonic frequency, then monitoring the waveform in real-time should see the vibration disappear immediately the supply is switched off if the problem is electrically related.

FAULTY SILICON CONTROLLED RECTIFIER (SCR) CARDS / SUPPLY



SCR's will inherently generate low amplitude vibration equal to the number of related cards in a firing cycle x Lf. A DC full-wave rectified unit with 6 SCR's will typically generate a vibration at 6xLf, whereas a half-wave rectified unit with 3 SCR's typically generates 3xLf. When a SCR unit develops a fault due to a defect card(s), loose connections, shorts or open circuits this can produce many harmonics of both the Lf and SCR firing frequency. Where one SCR card or particular side of the circuit malfunctions this can also result in 1/6x (full-wave rectified) or 1/3x (half-wave rectified) and harmonics of the SCR firing rate. Electrical Erosion is another problem experienced on variable speed drives, and is due to passage of electrical currents causing sparks between rotating bearing elements. Fluting is a form of advanced electrical erosion where the rolling surfaces give a "fluted" or "washboard" appearance. Fitting insulated bearings or using special greases can solve electrical erosion problems.

ROTOR DEFECTS



On standard AC induction motors (particularly where the duty is cyclic, namely compressors and cranes), rotors can develop problems with eccentricity and shorted, broken, cracked or loose connections occurring in the rotor bars, end rings and/or laminations. These problems can result in a predominant 1xRPM frequency surrounded by sidebands equal to the number of motor poles x slip frequency (Sf). Rotor bar problems can also be detected using current signature analysis, where sidebands equal to the number of motor poles x Sf, will surround the mains frequency (Mf) at 50Hz or 60Hz. With the Mf normalised to 60dB, these sidebands amplitudes will be 6dB or greater when rotor related problems exists.