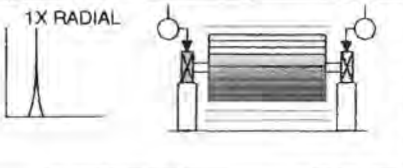
TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE | **TYPICAL SPECTRUM** | **PHASE RELATIONSHIP** | **REMARKS**

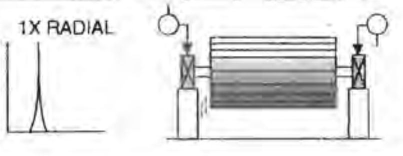
MASS UNBALANCE

A. FORCE UNBALANCE



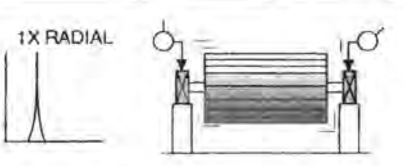
Force Unbalance will be in-phase and steady. Amplitude due to unbalance will increase by the square of speed below first rotor critical (a 3X speed increase = 9X higher vibration). 1X RPM always present and normally dominates spectrum. Can be corrected by placement of only one balance correction weight in one plane at Rotor center of gravity (CG). Approx. 0° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. 90° phase difference between horizontal & vertical readings usually occurs on each bearing of unbalanced rotor (± 30°).

B. COUPLE UNBALANCE



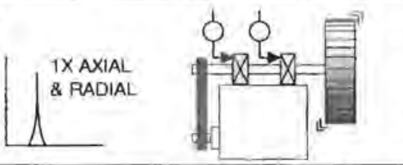
Couple Unbalance results in 180° out-of-phase motion on same shaft. 1X RPM always present and normally dominates spectrum. Amplitude varies with square of increasing speed below first rotor critical speed. May cause high axial vibration as well as radial. Correction requires placement of balance weights in at least 2 planes. Note that approx. 180° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. a 90° difference between the horizontal & vertical phase readings on each bearing usually occurs (± 30°).

C. DYNAMIC UNBALANCE



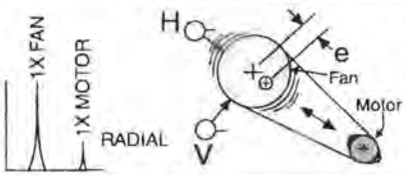
Dynamic Unbalance is the dominant type of unbalance found and is a combination of both force and couple unbalance. 1X RPM dominates the spectrum, and truly requires 2 plane correction. Here, the radial phase difference between outboard and inboard bearings can range anywhere from 0° to 180°. However, the horizontal phase difference should closely match the vertical phase difference, when comparing outboard and inboard bearing measurements (± 30°). Secondarily, if unbalance predominates, roughly a 90° phase difference usually results between the horizontal and vertical readings on each bearing (± 40°).

D. OVERHUNG ROTOR UNBALANCE



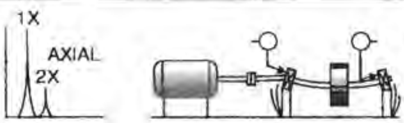
Overhung Rotor Unbalance causes high 1X RPM in both Axial and Radial directions. Axial readings tend to be in-phase whereas radial phase readings might be unsteady. However, the horizontal phase differences will usually match the vertical phase differences on the unbalanced rotor (± 30°). Overhung rotors have both force and couple unbalance, each of which will likely require correction. Thus, correction weights will most always have to be placed in 2 planes to counteract both force and couple unbalance.

ECCENTRIC ROTOR



Eccentricity occurs when center of rotation is offset from geometric centerline of a pulley, gear, bearing, motor armature, etc. Largest vibration occurs at 1X RPM of eccentric component in a direction thru centerlines of the two rotors. Comparative horizontal and vertical phase readings usually differ either by 0° or by 180° (each of which indicate straight-line motion). Attempts to balance eccentric rotors often result in reducing vibration in one radial direction, but increasing it in the other radial direction (depending on amount of eccentricity).

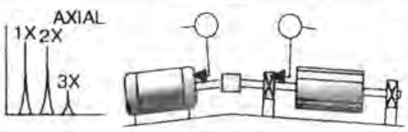
BENT SHAFT



Bent shaft problems cause high axial vibration with axial phase differences tending towards 180° on the same machine component. Dominant vibration normally occurs at 1X if bent near shaft center, but at 2X if bent near the coupling. (Be careful to account for transducer orientation for each axial measurement if you reverse probe direction.) Use dial indicators to confirm bent shaft.

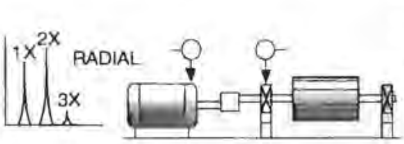
MISALIGNMENT

A. ANGULAR MISALIGNMENT



Angular Misalignment is characterized by high axial vibration, 180° out-of-phase across the coupling. Typically will have high axial vibration with both 1X and 2X RPM. However, not unusual for either 1X, 2X or 3X to dominate. These symptoms may also indicate coupling problems as well. Severe angular misalignment may excite many 1X RPM harmonics. Unlike Mechanical Looseness Type 3, these multiple harmonics do not typically have a raised noise floor on the spectra.

B. PARALLEL MISALIGNMENT

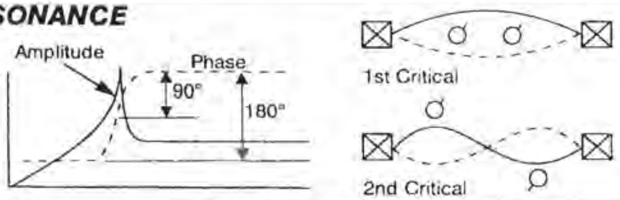


Offset Misalignment has similar vibration symptoms to Angular, but shows high radial vibration which approaches 180° out-of-phase across coupling. 2X often larger than 1X, but its height relative to 1X is often dictated by coupling type and construction. When either Angular or Radial Misalignment becomes severe, they can generate either high amplitude peaks at much higher harmonics (4X-8X), or even a whole series of high frequency harmonics similar in appearance to mechanical looseness. Coupling type and material will often greatly influence the entire spectrum when misalignment is severe. Does not typically have raised noise floor.

C. MISALIGNED BEARING COCKED ON SHAFT

Cocked Bearing will generate considerable axial vibration. Will cause Twisting Motion with approximately 180° phase shift top to bottom and/or side to side as measured in axial direction on same bearing housing. Attempts to align coupling or balance the rotor will not alleviate problem. Bearing usually must be removed and correctly installed.

RESONANCE



Resonance occurs when a Forcing Frequency coincides with a System Natural Frequency, and can cause dramatic amplitude amplification, which might result in premature, or even catastrophic failure. This may be a natural frequency of the rotor, but can often originate from support frame, foundation, gearbox or even drive belts. If a rotor is at or near resonance, it can be almost impossible to balance due to the great phase shift it experiences (90° at resonance; nearly 180° when passes thru). Often requires changing natural frequency to a higher or lower frequency. Natural Frequencies do not generally change with a change in speed which helps facilitate their identification (unless on a large plain bearing machine or on a rotor which has significant overhang).

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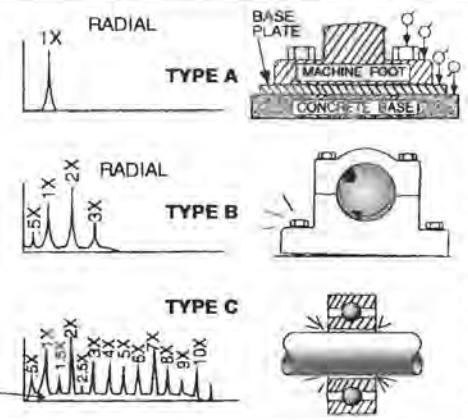
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS

MECHANICAL LOOSENESS



NOTE RAISED NOISE FLOOR INDICATING LOOSENESS

Mechanical Looseness is indicated by either Type A, B or C vibration spectra.

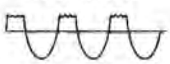
Type A is caused by Structural looseness/weakness of machine feet, baseplate or foundation; also by deteriorated grouting, loose hold-down bolts at the base; and distortion of the frame or base (i.e., soft foot). Phase analysis may reveal approx. 90° to 180° phase difference between vertical measurements on bolt, machine foot, baseplate or base itself.

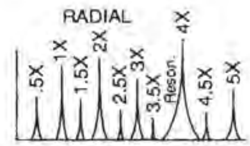
Type B is generally caused by loose pillowblock bolts, cracks in frame structure or in bearing pedestal.

Type C is normally generated by improper fit between component parts which will cause many harmonics due to nonlinear response of loose parts to dynamic forces from rotor. Causes a truncation of time waveform and a raised noise floor in the spectrum. Type C is often caused by a bearing liner loose in its cap, a bearing loose and turning on its shaft, excessive clearance in either a sleeve or rolling element bearing, a loose impeller on a shaft, etc. Type C Phase is often unstable and may vary widely from one measurement to next, particularly if rotor shifts position on shaft from one startup to next. Mechanical Looseness is often highly directional and may cause very different readings when comparing levels at 30° increments in radial direction all the way around one bearing housing. Also, note that looseness will often cause subharmonic multiples at exactly 1/2 or 1/3X RPM (.5X, 1.5X, 2.5X, etc.).

ROTOR RUB

TRUNCATED FLATTENED WAVEFORM

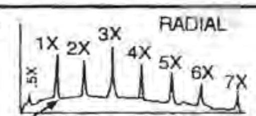




Rotor Rub produces similar spectra to Mechanical Looseness when rotating parts contact stationary components. Rub may be either partial or throughout the entire shaft revolution. Usually generates a series of frequencies, often exciting one or more resonances. Often excites integer fraction subharmonics of running speed (1/2, 1/3, 1/4, 1/5,...1/n), depending on location of rotor natural frequencies. Rotor rub can excite many high frequencies (similar to wide-band noise when chalk is drug along a blackboard). It can be very serious and of short duration if caused by shaft contacting bearing babbitt. A full annular rub throughout an entire shaft revolution can induce "reverse precession" with the rotor whirling at critical speed in a direction opposite shaft rotation (inherently unstable which can lead to catastrophic failure).

JOURNAL BEARINGS

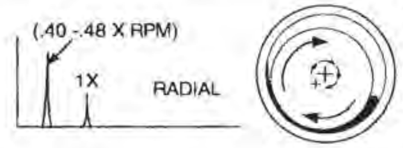
A. WEAR/CLEARANCE PROBLEMS



NOTE RAISED NOISE FLOOR INDICATING CLEARANCE/LOOSENESS.

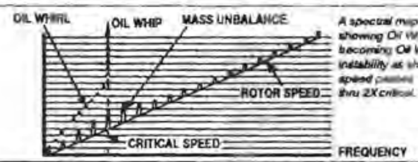
Latter stages of journal bearing wear are normally evidenced by presence of whole series of running speed harmonics (up to 10 or 20). Wiped journal bearings often will allow high vertical amplitudes compared to horizontal, but, may show only one pronounced peak at 1X RPM. Journal bearings with excessive clearance may allow a minor unbalance and/or misalignment to cause high vibration which would be much lower if bearing clearances were set to spec.

B. OIL WHIRL INSTABILITY



Oil Whirl instability occurs at .40 - .48X RPM and is often quite severe. Considered excessive when amplitude exceeds 40% of bearing clearances. Oil Whirl is an oil film excited vibration where deviations in normal operating conditions (attitude angle and eccentricity ratio) cause oil wedge to "push" shaft around within bearing. Destabilizing force in direction of rotation results in a whirl (or forwards precession). Oil Whirl is unstable since it increases centrifugal forces which increase whirl forces. Can cause oil to no longer support shaft and can become unstable when whirl frequency coincides with a rotor natural frequency. Changes in oil viscosity, lube pressure and external preloads can affect oil whirl.

C. OIL WHIP INSTABILITY



A spectral map showing Oil Whirl becoming Oil Whip instability at shaft speed passes thru 2X critical.

Oil Whip may occur if machine operated at or above 2X rotor critical frequency. When rotor brought up to twice critical speed, whirl will be very close to rotor critical and may cause excessive vibration that oil film may no longer be capable of supporting. Whirl speed will actually "lock onto" rotor critical and this peak will not pass through it even if machine is brought to higher and higher speeds. Produces a lateral forward precessional subharmonic vibration at rotor critical frequency. Inherently unstable which can lead to catastrophic failure.

ROLLING ELEMENT BEARINGS
(4 Failure Stages)

f_n = Natural Frequencies of Installed Bearing Components and Support Structure

BEARING DEFECT FREQUENCIES:

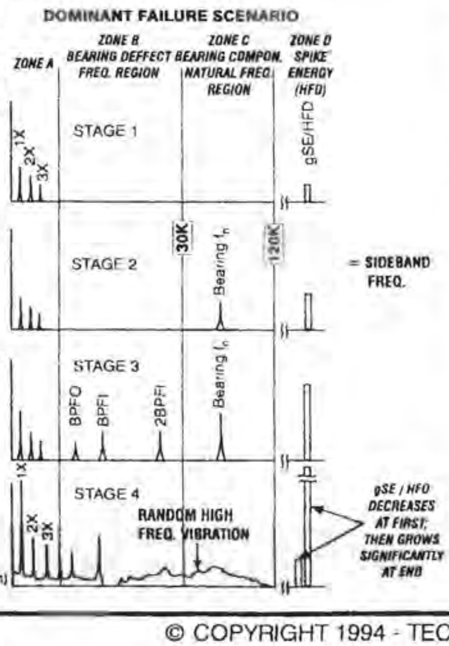
$$BPFI = \frac{N_b}{2}\left(1 + \frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BPFO = \frac{N_b}{2}\left(1 - \frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BSF = \frac{P_d}{2B_d}\left[1 - \left(\frac{B_d}{P_d}\right)^2 (\cos\theta)^2\right] \times RPM$$

$$FTF = \frac{1}{2}\left(1 - \frac{B_d}{P_d}\cos\theta\right) \times RPM$$

Where:
- BPFI = Inner Race Frequency
- BPFO = Outer Race Frequency
- BSF = Ball Spin Frequency
- FTF = Fund. Train (Cage) Freq.
- N_b = Number of Balls or Rollers
- B_d = Ball/Roller Diameter (in or mm)
- P_d = Bearing Pitch Diameter (in or mm)
- θ = Contact Angle (degrees)



4 ROLLING ELEMENT BEARING FAILURE STAGES

STAGE 1: Earliest indications of bearing problems appear in ultrasonic frequencies ranging from about 250,000 - 350,000 Hz; later, as wear increases, usually drops to approximately 20,000 - 60,000 Hz (1,200,000 - 3,600,000 CPM). These are frequencies evaluated by Spike Energy (gSE), HFD(g) and Shock Pulse (dB). For example, spike energy may first appear at about .25 gSE in Stage 1 (actual value depending on measurement location and machine speed). Acquiring high frequency enveloped spectra confirms whether or not bearing is in Failure Stage 1.

STAGE 2: Slight bearing defects begin to "ring" bearing component natural frequencies (f_n) which predominantly occur in 30K - 120K CPM range. Such natural frequencies may also be resonances of bearing support structures. Sideband frequencies appear above and below natural frequency peak at end of Stage 2. Overall spike energy grows (for example, from .25 to .50 gSE).

STAGE 3: Bearing defect frequencies and harmonics appear. When wear progresses, more defect frequency harmonics appear and number of sidebands grow, both around these and bearing component natural frequencies. Overall spike energy continues to increase (for example, from .5 to over 1 gSE). Wear is now usually visible and may extend throughout periphery of bearing, particularly when many well formed sidebands accompany bearing defect frequency harmonics. High frequency demodulated and enveloped spectra help confirm Stage III. ***Replace bearings now! (Independent of bearing defect frequency amplitudes in vibration spectra).***

STAGE 4: Towards the end, amplitude of 1X RPM is even effected. It grows, and normally causes growth of many running speed harmonics. Discrete bearing defect and component natural frequencies actually begin to "disappear" and are replaced by random, broadband high frequency "noise floor". In addition, amplitudes of both high frequency noise floor and spike energy may in fact decrease; but just prior to failure, spike energy and HFD will usually grow to excessive amplitudes.

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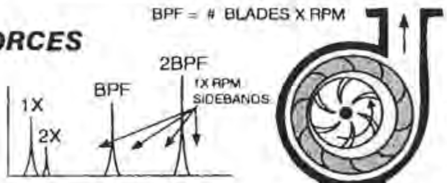
PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS

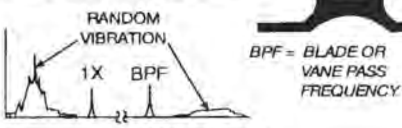
HYDRAULIC AND AERODYNAMIC FORCES

A. BLADE PASS & VANE PASS



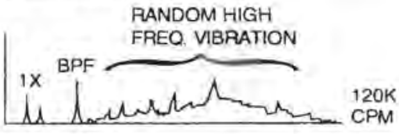
Blade Pass Frequency (BPF) = No. of Blades (or Vanes) X RPM. This frequency is inherent in pumps, fans and compressors, and normally does not present a problem. However, large amplitude BPF (and harmonics) can be generated in pump if gap between rotating vanes and stationary diffusers is not equal all the way around. Also, BPF (or harmonic) sometimes can coincide with a system natural frequency causing high vibration. High BPF can be generated if impeller wear ring seizes on shaft, or if welds fastening diffuser vanes fail. Also, high BPF can be caused by abrupt bends in pipe (or duct), obstructions which disturb flow, damper settings or if pump or fan rotor is positioned eccentrically within housing.

B. FLOW TURBULENCE



Flow Turbulence often occurs in blowers due to variations in pressure or velocity of the air passing thru the fan or connected ductwork. This flow disruption causes turbulence which will generate random, low frequency vibration, typically in the range of 50 to 2000 CPM. If surging occurs within a compressor, random broadband high frequency vibration can occur. Excessive turbulence can also excite broadband high frequency.

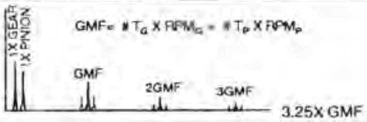
C. CAVITATION



Cavitation normally generates random, higher frequency broadband energy which is sometimes superimposed with blade pass frequency harmonics. Normally indicates insufficient suction pressure (starvation). Cavitation can be quite destructive to pump internals if left uncorrected. It can particularly erode impeller vanes. When present, it often sounds as if "gravel" is passing thru pump. Cavitation is usually caused by insufficient inlet flow. Can occur during one survey, and be absent the next survey (if changes in suction valve settings are made).

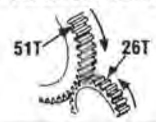
GEARS

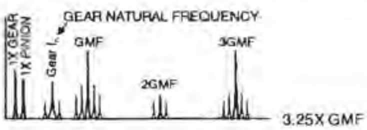
A. NORMAL SPECTRUM



Normal Spectrum shows Gear & Pinion Speeds, along with Gear Mesh Frequency (GMF) and very small GMF harmonics. GMF harmonics commonly will have running speed sidebands around them. All peaks are of low amplitude, and no natural frequencies of gears are excited. F_{MAX} recomended at 3.25X GMF (minimum) when # teeth are known. If tooth count is not known, set F_{MAX} at 200X RPM on each shaft.

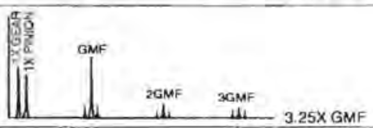
B. TOOTH WEAR





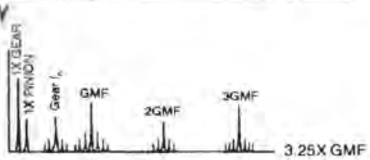
Key indicator of Tooth Wear is excitation of Gear Natural Frequency (f_n), along with sidebands around it spaced at the running speed of the bad gear. Gear Mesh Frequency (GMF) may or may not change in amplitude, although high amplitude sidebands and number of sidebands surrounding GMF usually occur when wear is noticeable. Sidebands may be better wear indicator than GMF frequencies themselves. Also, high amplitudes commonly occur at either 2XGMF or at 3XGMF (esp. 3XGMF), even when GMF amplitude is acceptable.

C. TOOTH LOAD



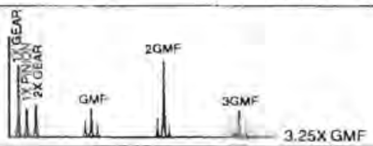
Gear Mesh Frequencies are often very sensitive to load. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low level, and no gear natural frequencies are excited. Each Analysis should be performed with system at maximum operating load for meaningful spectral comparisons.

D. GEAR ECCENTRICITY AND BACKLASH



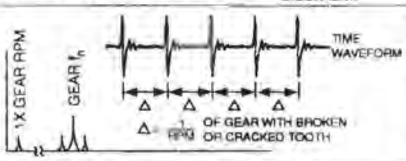
Fairly high amplitude sidebands around GMF harmonics often suggest gear eccentricity, backlash, or non-parallel shafts which allow the rotation of one gear to "modulate" either the GMF amplitude or the running speed of the other gear. The gear with the problem is indicated by the spacing of the sideband frequencies.Also, 1X RPM level of eccentric gear will normally be high if eccentricity is the dominant problem. Improper backlash normally excites GMF harmonics and Gear Natural Frequency, both of which will be sidebanded at 1X RPM. GMF amplitudes will often decrease with increasing load if backlash is the problem.

E. GEAR MISALIGNMENT



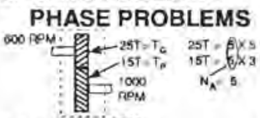
Gear Misalignment almost always excites second order or higher GMF harmonics which are sidebanded at running speed. Often will show only small amplitude 1X GMF, but much higher levels at 2X or 3X GMF. Important to set F_{MAX} high enough to capture at least 3 GMF harmonics. Also, sidebands around 2XGMF will often be spaced at 2X RPM. Note that sideband amplitudes often are not equal on left and right side of GMF and GMF harmonics due to the tooth misalignment. Causes uneven wear pattern.

F. CRACKED/BROKEN TOOTH



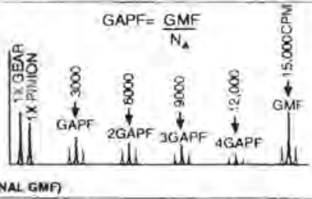
A Cracked or Broken Tooth will generate a high amplitude at 1X RPM of this gear only in the time waveform, plus it will excite gear natural frequency (f_n) sidebanded at its running speed. It is best detected in Time Waveform which will show a pronounced spike every time the problem tooth tries to mesh with teeth on the mating gear. Time between impacts (Δ) will correspond to 1/RPM of gear with the problem. Amplitudes of Impact Spikes in Time Waveform often will be 10X to 20X higher than that at 1X RPM in the FFT!

G. GEAR ASSEMBLY PHASE PROBLEMS



$GAPF = \frac{GMF}{N_A} = \frac{15,000}{5}$

GAPF = 3000 CPM = 0.20X GMF **(FRACTIONAL GMF)**

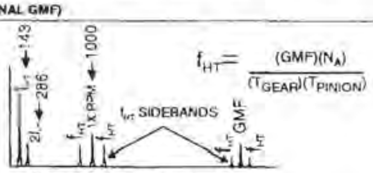


Gear Assembly Phase Freq. (GAPF) can result in Fractional Gear Mesh Frequencies (if $N_A > 1$). It literally means (T_G/N_A) gear teeth will contact (T_P/N_A) pinion teeth and will generate N_A wear patterns, where N_A in a given tooth combination equals the product of prime factors common to the number of teeth on the gear and pinion (N_A = Assembly Phase Factor). GAPF (or harmonics) can show up right from the beginning if there were manufacturing problems. Also, its sudden appearance in a periodic survey spectrum can indicate damage if contaminate particles pass through the mesh, resulting in damage to the teeth in mesh at the time of ingestion just as they enter and leave meshing or that gears have been reoriented.

H. HUNTING TOOTH PROBLEMS

$6 = 1X2X3$
$7 = 1X7$ } $N_A = 1$

$f_{HT} = \frac{(6X1000)(1)}{(6)(7)} = \frac{1000}{7} = 143$ CPM (One Pulse Per 7 Pinion Revolutions)



$f_{HT} = \frac{(GMF)(N_A)}{(T_{GEAR})(T_{PINION})}$

N_A = 1 is the ideal assembly phase factor in gear design

Hunting Tooth Frequency (f_{HT}) occurs when faults are present on both the gear and pinion which might have occurred during the manufacturing process, due to mishandling, or in the field. It can cause quite high vibration, but since it occurs at low frequencies predominately less than 600 CPM, it is often missed. A gear set with this tooth repeat problem normally emits a "growling" sound from the drive. The maximum effect occurs when the faulty pinion and gear teeth both enter mesh at the same time (on some drives, this may occur only 1 of every 10 to 20 revolutions, depending on the f_{HT} formula). Note that T_{GEAR} and T_{PINION} refer to number of teeth on the gear and pinion, respectively. N_A is the Assembly Phase Factor defined above. Will often modulate both GMF and Gear RPM peaks.

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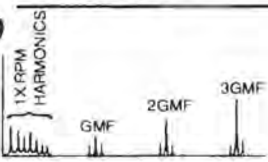
GEARS (CONTINUED)

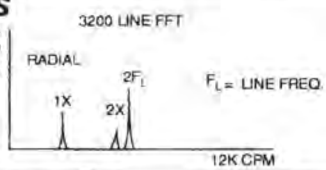
I. LOOSE BEARING FIT



Excessive Clearance of bearings supporting the gears can not only excite many running speed harmonics, but will often cause high amplitude response at GMF, 2GMF and/or 3GMF. These high GMF amplitudes are actually a response to, and not the cause of, looseness within the bearings supporting the gearing. Such excessive clearance can be caused either by extensive bearing wear or by improper bearing fit onto the journal during installation. Left uncorrected, it can cause excessive gear wear and damage to other components.

AC INDUCTION MOTORS

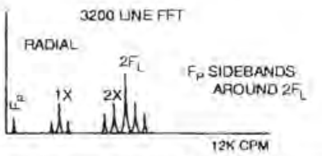
A. STATOR ECCENTRICITY, SHORTED LAMINATIONS OR LOOSE IRON



Stator problems generate high vibration at 2X line frequency ($2F_L$). Stator eccentricity produces uneven stationary air gap between rotor and stator which produces very directional vibration. Differential Air Gap should not exceed 5% for induction motors and 10% for synchronous motors. Soft foot and warped bases can produce an eccentric stator. Loose iron is due to stator support weakness or looseness. Shorted stator laminations can cause uneven, localized heating which can distort the stator itself. This produces thermally-induced vibration which can significantly grow with operating time causing stator distortion and static air gap problems.

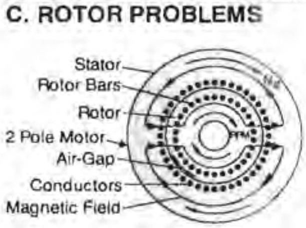
B. ECCENTRIC ROTOR (Variable Air Gap)

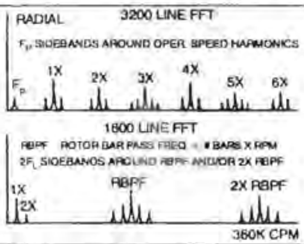
F_L = Electrical Line Freq.
N_s = Synch. Speed = $\frac{120F_L}{P}$
F_s = Slip Freq. = N_s – RPM
F_p = Pole Pass Freq. = F_s X P
P = # Poles



Eccentric Rotors produce a rotating variable air gap between the rotor and stator which induces pulsating vibration (normally between $2F_L$ and closest running speed harmonic). Often requires "zoom" spectrum to separate $2F_L$ and running speed harmonic. Eccentric rotors generate $2F_L$ surrounded by Pole Pass frequency sidebands (F_p), as well as F_p sidebands around running speed. F_p appears itself at low frequency (Pole Pass Frequency = Slip Frequency X # Poles). Common values of F_p range from about 20 to 120 CPM (0.3 - 2.0 Hz). Soft foot or misalignment often induces a variable air gap due to distortion (actually a mechanical problem, not electrical).

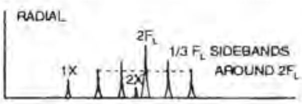
C. ROTOR PROBLEMS





Broken or Cracked rotor bars or shorting rings; bad joints between rotor bars and shorting rings; or shorted rotor laminations will produce high 1X running speed vibration with pole pass frequency sidebands (F_p). In addition, these problems will often generate F_p sidebands around the second, third, fourth and fifth running speed harmonics. Loose or open rotor bars are indicated by 2X line freq. ($2F_L$) sidebands surrounding Rotor Bar Pass Frequency (RBPF) and/or its harmonics (RBPF = Number of Bars X RPM). Often will cause high levels at 2X RBPF, with only a small amplitude at 1X RBPF. Electrically induced arcing between loose rotor bars and end rings will often show high levels at 2X RBPF (with $2F_L$ sidebands), but little or no increase in amplitudes at 1X RBPF.

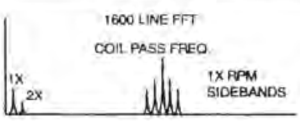
D. PHASING PROBLEM (Loose Connector)



Phasing problems due to loose or broken connectors can cause excessive vibration at 2X Line Freq. ($2F_L$) which will have sidebands around it spaced at 1/3 Line Freq. (1/3 F_L). Levels at $2F_L$ can exceed 1.0 in/sec if left uncorrected. This is particularly a problem if the defective connector is only sporadically making contact. Loose or broken connectors must be repaired to prevent catastrophic failure.

AC SYNCHRONOUS MOTORS

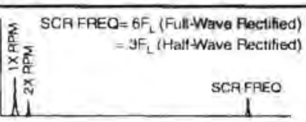
(Loose Stator Coils)



Loose stator coils in synchronous motors will generate fairly high vibration at Coil Pass Freq. (CPF) which equals the number of stator coils X RPM (# Stator Coils = # Poles X # Coils/Pole). The Coil Pass Frequency will be surrounded by 1X RPM sidebands. Synchronous motor problems may also be indicated by high amplitude peaks at approx. 60,000 to 90,000 CPM, accompanied by $2F_L$ sidebands. Take at least one spectrum up to 90,000 CPM on each motor bearing housing.

DC MOTORS AND CONTROLS

A. NORMAL SPECTRUM

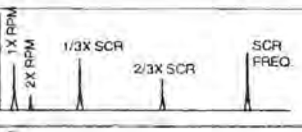


Many DC Motor and Control Problems can be detected by vibration analysis. Full-wave rectified motors (6 SCR's) generate a signal at 6X Line Frequency ($6F_L$ = 360 Hz = 21,600 CPM); while half-wave rectified DC motors (3 SCR's) generate 3X Line Freq. ($3F_L$ = 180 Hz = 10,800 CPM). The SCR firing Frequency is normally present in a DC Motor Spectrum, but at low amplitude. Note the absence of other peaks at multiples of F_L.

B. BROKEN ARMATURE WINDINGS, GROUNDING PROBLEMS OR FAULTY SYSTEM TUNING

When DC Motor spectra are dominated by high levels at SCR or 2X SCR, this normally indicates either Broken Motor Windings or Faulty Tuning of the Electrical Control System. Proper tuning alone can lower vibration at SCR and 2X SCR significantly if control problems predominate. High amplitudes at these frequencies would normally be above approximately .10 in/sec, peak at 1 X SCR and about .04 in/sec at 2 X SCR Firing Freq.

C. FAULTY FIRING CARD OR BLOWN FUSE



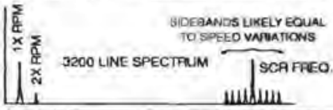
When one firing card fails to fire, then 1/3 of power is lost, and can cause repeated momentary speed changes in the motor. This can lead to high amplitudes at 1/3X and 2/3X SCR Frequency (1/3X SCR Freq. = 1X F_L for half-wave rectified, but 2X F_L for a full-wave rectified SCR).

Caution: Card/SCR configuration should be known before troubleshooting motor (# SCR's, # Firing Cards, etc.).

D. FAULTY SCR, SHORTED CONTROL CARD, LOOSE CONNECTIONS AND/OR BLOWN FUSE

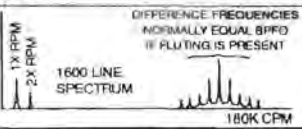
Faulty SCR's, Shorted Control Cards and/or Loose Connections can generate noticeable amplitude peaks at many combinations of line frequency (F_L) and SCR firing frequency. Normally, 1 bad SCR can cause high levels at F_L and/or $5F_L$ in 6 SCR motors. The point to be made is that neither F_L, $2F_L$, $4F_L$ nor $5F_L$ should be present in DC Motor spectra.

E. FAULTY COMPARITOR CARD



Faulty Comparitor Cards cause problems with RPM fluctuation or "hunting". This causes a constant collapsing and regenerating of the magnetic field. These sidebands often approximate the RPM fluctuation and require a high resolution FFT to even detect them. Such sidebands could also be due to generation and regeneration of the magnetic field.

F. ELECTRICAL CURRENT PASSAGE THRU DC MOTOR BEARINGS



Electrically-induced Fluting is normally detected by a series of difference frequencies with the spacing most often at the outer race defect frequency (BPFO), even if such fluting is present on both the outer and inner races. They most often show up in a range centered at about 100,000 to 150,000 CPM. A 180K CPM spectrum with 1600 lines is recommended for detection with measurements on both the OB and IB DC motor bearings.

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TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

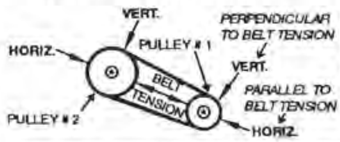
PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS

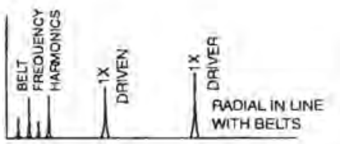
BELT DRIVE PROBLEMS

$$\text{BELT FREQ.} = \frac{3.142 \times \text{PULLEY RPM} \times \text{PITCH DIAM.}}{\text{BELT LENGTH}}$$

$$\text{TIMING BELT FREQ.} = \text{BELT FREQ.} \times \#\ \text{BELT TEETH} = \text{PULLEY RPM} \times \#\ \text{PULLEY TEETH}$$

A. WORN, LOOSE OR MISMATCHED BELTS





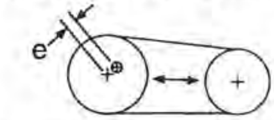
$\text{PITCH DIAM}_1 \times \text{RPM}_1 = \text{PITCH DIAM}_2 \times \text{RPM}_2$

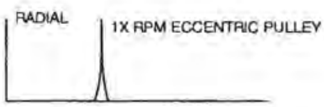
Belt frequencies are below the RPM of either the motor or the driven machine. When they are worn, loose or mismatched, they normally cause 3 to 4 multiples of belt frequency. Often 2X belt freq. is the dominant peak. Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM. On timing belt drives, wear or pulley misalignment is indicated by high amplitudes at the Timing Belt Frequency. Chain drives will indicate problems at Chain Pass Frequency which equals # Sprocket Teeth X RPM.

B. BELT/PULLEY MISALIGNMENT

Misalignment of pulley produces high vibration at 1X RPM predominantly in the axial direction. The ratio of amplitudes of driver to driven RPM depends on where the data is taken, as well as on relative mass and frame stiffness. Often with pulley misalignment, the highest axial vibration on the motor will be at fan RPM, or vice versa. Can be confirmed by phase measurements by setting Phase Filter at RPM of pulley with highest axial amplitude; then compare phase at this particular frequency on each rotor in the axial direction.

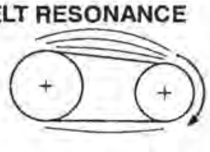
C. ECCENTRIC PULLEYS

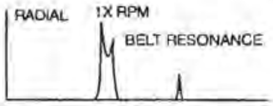




Eccentric pulleys cause high vibration at 1X RPM of the eccentric pulley. The amplitude is normally highest in line with the belts, and should show up on both driver and driven bearings. It is sometimes possible to balance eccentric pulleys by attaching washers to taper-lock bolts. However, even if balanced, the eccentricity will still induce vibration and reversible fatigue stresses in the belt. Pulley eccentricity can be confirmed by phase analysis showing horizontal & vertical phase differences of nearly 0° or 180°.

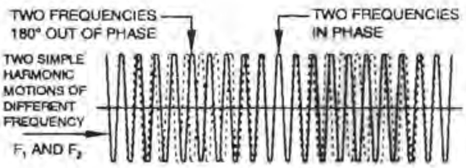
D. BELT RESONANCE

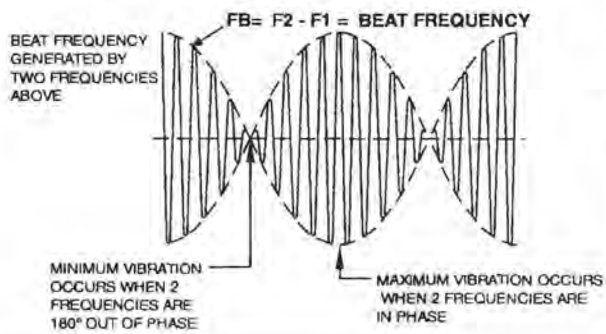




Belt Resonance can cause high amplitudes if the belt natural frequency should happen to approach, or coincide with, either the motor or driven RPM. Belt natural frequency can be altered by changing either the belt tension, belt length or cross section. Can be detected by tensioning and then releasing belt while measuring the response on pulleys or bearings. However, when operating, belt natural frequencies will tend to be slightly higher on the tight side and lower on the slack side.

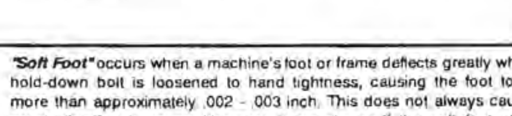
BEAT VIBRATION

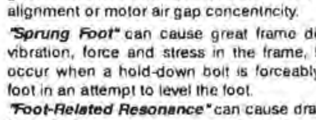




A Beat Frequency is the result of two closely spaced frequencies going into and out of synchronization with one another. The wideband spectrum normally will show one peak pulsating up and down. When you zoom into this peak (lower spectrum below), it actually shows two closely spaced peaks. The difference in these two peaks (F_2-F_1) is the beat frequency which appears itself in the wideband spectrum. The beat frequency is not commonly seen in normal frequency range measurements since it is inherently low frequency, usually ranging from only approximately 5 to 100 CPM.

Maximum vibration will result when the time waveform of one frequency (F_1) comes into phase with the waveform of the other frequency (F_2). Minimum vibration occurs when waveforms of these two frequencies line up 180° out of phase.





SOFT FOOT, SPRUNG FOOT AND FOOT-RELATED RESONANCE

"Soft Foot" occurs when a machine's foot or frame deflects greatly when a hold-down bolt is loosened to hand tightness, causing the foot to rise more than approximately .002 - .003 inch. This does not always cause a great vibration increase. However, it can do so if the soft foot affects alignment or motor air gap concentricity.

"Sprung Foot" can cause great frame distortion, resulting in increased vibration, force and stress in the frame, bearing housing, etc. This can occur when a hold-down bolt is forceably torqued down on the sprung foot in an attempt to level the foot.

"Foot-Related Resonance" can cause dramatic amplitude increases from 5X to 15X or more, as compared with that when the bolt (or combination of bolts) is loosened to hand tightness. When tight, this bolt can notably change the natural frequency of the foot or machine frame itself.

Soft Foot, Sprung Foot or Foot-Related Resonance most often affects vibration at 1X RPM, but can also do so at 2X RPM, 3X RPM, 2X line frequency, blade pass frequency, etc. (particularly Foot-Related Resonance).

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