APPENDIX- B
VIBRATION CHARTS

TABLE 6.0 - ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS
MASS UNBALANCE A. FORCE UNBALANCE	1X RADIAL		Force Unbalance will be in-phase and steady. Amplitude due to unbalance will increase by the square of speed (2X speed increase = 9X higher vibration). 1X RPM always present and normally dominates spectrum. Can be corrected by placement of only one balance weight in one plane at rotor center of gravity (CG).
B. COUPLE UNBALANCE	1X RADIAL		Couple Unbalance tends toward 180° out-of-phase on same shaft. 1X always present and normally dominates spectrum. Amplitude varies with square of increasing speed. May cause high axial vibrations 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 OB & IB verticals.
C. OVERHUNG ROTOR UNBALANCE	1X AXIAL & RADIAL		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. Overhung rotors often have both force and couple unbalance, each of which will likely require correction.
ECCENTRIC ROTOR	1X FAN, 1X MOTOR, RADIAL		Eccentricity occurs when center of rotation is offset from geometric centerline of a sheave, gear, bearing, motor armature, etc. Largest vibration occurs at 1X RPM of eccentric component in a direction thru centers 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 rotor often result in reducing vibration in one direction, but increasing it in the other radial direction (depending on amount of eccentricity).
BENT SHAFT	1X, AXIAL, 2X		Bent shaft problems cause high axial vibration with axial phase differences tending toward 180° on the same machine component. Dominant vibration normally 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.)
MISALIGNMENT A. ANGULAR MISALIGNMENT	1X, 2X, AXIAL, 3X		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.
B. PARALLEL MISALIGNMENT	1X, 2X, RADIAL, 3X		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, can generate either high amplitude peaks at much higher harmonics (4X,5X) or even a whole series of high frequency harmonics similar in appearance to mechanical looseness. Coupling construction will often greatly influence shape of spectrum when misalignment is severe.
C. MISALIGNED BEARING COCKED ON SHAFT	1X, 2X, AXIAL, 3X, PHASE		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 of same bearing housing. Attempts to align coupling or balance the rotor will not alleviate problem. Bearing must be removed and correctly installed.
RESONANCE	Amplitude, Phase, 90°, 180°, 2nd Critical		Resonance occurs when a Forcing Frequency coincides with a System Natural Frequency, and can cause dramatic amplitude amplification, which can 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 will be almost impossible to balance due to the great phase shift it experiences (90° at rotor speed, nearly 180° when passes thru). Often requires changing natural frequency location. Natural Frequencies do not change with a change in speed which helps facilitate their identification.
MECHANICAL LOOSENESS	1X RADIAL TYPE A; 1X, 2X, 3X RADIAL TYPE B; TYPE C	BASE PLATE	Mechanical Looseness is indicated by either Type A, B or C 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. 180° phase difference between vertical measurements on machine feet, baseplate and base road. Type B is generally caused by loose pillowblock bolts, cracks in frame structure or bearing pedestal. Type C is normally generated by improper fit between component parts to dynamic forces from rotor. Causes a truncated or flat wavefORM. Type C is often caused by a bearing liner loose in its cap, excessive clearance in either a sleeve or rolling element bearing, or a loose impeller on a shaft. Type C2 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 display noticeably different readings if compare levels at 30° increments in radial direction all the way around one bearing housing. Also, note that looseness will often cause subharmonic multiples of exactly 1/2 or 1/3 RPM (.5X, 1.5X, 2.5X, etc.).

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TABLE 6.0 - ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
ROTOR RUB	TRUNCATED FLATTENED WAVEFORM; RADIAL .5X, 1X, 1.5X, 2X, 2.5X, 3X, 3.5X, Random, 4X, 4.5X, 5X	Rotor Rub produces similar spectra to Mechanical Looseness when rotating parts contact stationary components. Rub may be either partial or throughout the whole 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 in internal wide-band noise (where chain is drug along a blackboard). It can be very serious and of short duration if caused by shaft contacting bearing babbitt, but less serious when shaft rubbing a seal, an agitator blade rubbing the wall of a tank, or a coupling guard pressing against a shaft.
SLEEVE BEARINGS A. WEAR/CLEARANCE PROBLEMS	RADIAL 1X, 2X, 3X, 4X, 5X, 6X, 7X	Later stages of sleeve bearing wear are normally evidenced by presence of whole series of running speed harmonics (up to 10 or 20). Worn sleeve bearings often will allow high vertical amplitudes compared to horizontal. Sleeve 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 to spec.
B. OIL WHIRL INSTABILITY	(.42 - .48 X RPM), 1X, RADIAL	Oil Whirl instability occurs at .42 - .48X RPM and is often quite severe. Considered excessive when amplitude exceeds 50% of bearing clearance. Oil Whirl is an oil film induced 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 precession). Whirl is inherently unstable since it increases centrifugal forces which increase whirl forces. Can cause oil to no longer support shaft, or can become unstable when whirl frequency coincides with a rotor natural frequency. Changes in oil viscosity, lube pressure and external pressure can affect oil whirl.
C. OIL WHIP INSTABILITY	OIL WHIP, MASS UNBALANCE, CRITICAL SPEED, FREQUENCY	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 the peak will not pass thru it even if machine brought to higher and higher speeds.
ROLLING ELEMENT BEARINGS (4 Failure Phases)	DOMINANT FAILURE SCENARIO; ZONE A, ZONE B, ZONE C, ZONE D; STAGE 1, STAGE 2, STAGE 3, STAGE 4; RANDOM HIGH FREQ VIBRATION; 2000K CPM	*4 ROLLING ELEMENT BEARING FAILURE STAGES* STAGE 1: Earliest indications of bearing problems appear in ultrasonic frequencies ranging from 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). STAGE 2: Slight bearing defects begin to "ring" bearing component natural frequencies (f_n) which predominantly occur in 36K - 120K CPM range. Sideband frequencies appear above and below natural frequency peak at end of Stage 2. Spike energy grows (for example, from .25 to .50 gSE). STAGE 3: Bearing defect frequencies and harmonics appear (see page entitled "Rolling Element Bearing Defect Frequencies"). When wear progresses, more defect frequency harmonics appear and number of sidebands grow, both around these and around bearing natural frequencies. 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 well formed sidebands accompany bearing defect frequency - harmonics. Replace bearings now. STAGE 4: Towards the end, amplitude of 1X RPM is even affected. 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, broad band 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 will usually grow to excessive amplitudes.
HYDRAULIC AND AERODYNAMIC FORCES A. BLADE PASS & VANE PASS	BPF = # BLADES X RPM; 1X, 2X, BPF, 2BPF	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 kept equal all way around. Also, BPF (or harmonics) sometimes can coincide with a system natural frequency causing high vibration. High BPF can be generated if impeller vanes bring blades on shaft or if vanes fastening diffusers fail. Also, high BPF can be caused by abrupt bends in pipe or duct, obstructions which disturb flow, or if pump or fan rotor is positioned eccentrically within housing.
B. FLOW TURBULENCE	RANDOM, 1X, BPF; BPF = BLADE OR VANE PASS FREQUENCY	Flow Turbulence often appears in blowers due to variations in pressure or velocity of the air passing thru the fan or connected ductwork. This flow deviation causes turbulence which will generate random, low frequency vibration (typically in the range of 50 to 2000 CPM).
C. CAVITATION	RANDOM HIGH FREQ VIBRATION; 1X, BPF; 120K CPM	Cavitation normally generates random, higher frequency broadband energy which is sometimes accompanied with blade pass frequency harmonics. Normally indicates insufficient suction pressure or elevation. 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.

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