University at Buffalo
The State University of New York

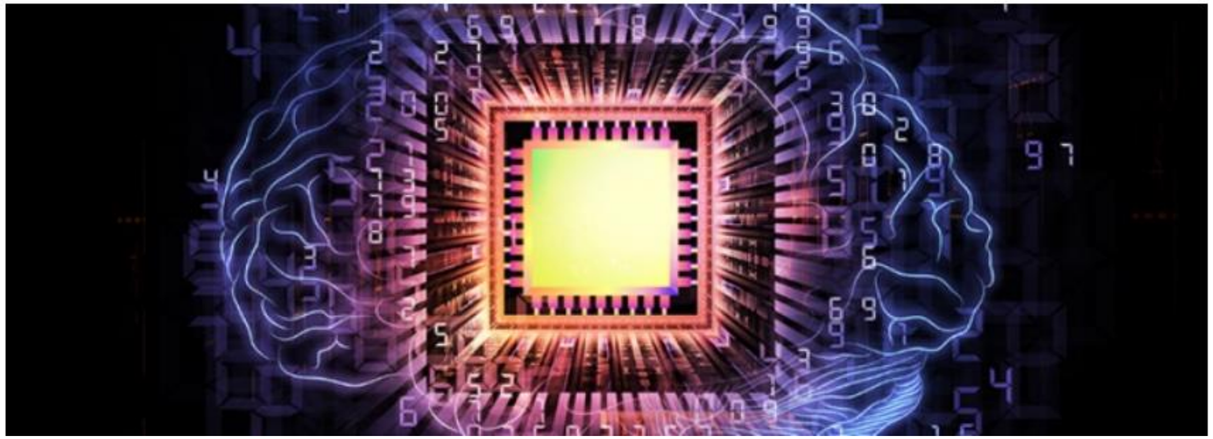
ENGINEERING SCIENCE
ARTIFICIAL INTELLIGENCE MS

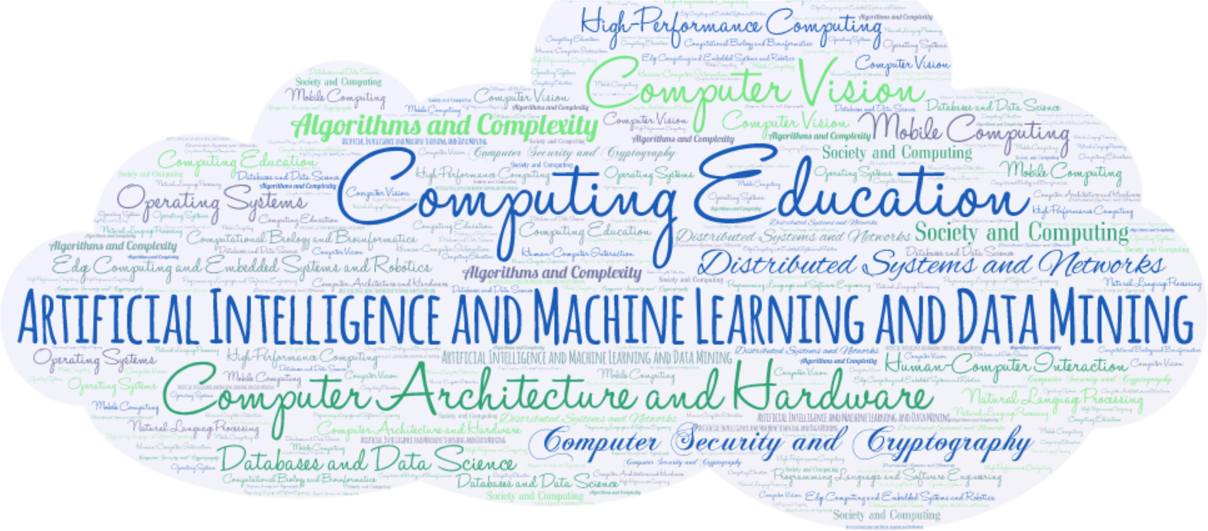
Institute for Artificial Intelligence and Data Science

In Partnership with

MACHINERY MONITORING SYSTEMS, LLC

ENGINEERING SCIENCE (ARTIFICIAL INTELLIGENCE) MS





AI & MACHINE LEARNING Masters Program Course Material

Courtesy of:

Steve Follmar (President, MMS)

John Biondolillo (CTO, MMS)

Enhanced by: MachineryRx

Course Objectives

- Create an automated, rule-based vibration diagnostic model, able to ingest Data and visualize waveforms.
- Include pattern-recognition into the model, based on the fault-frequency lookup library.
- Transform (integrate) incoming waveforms into spectral Data and utilize pattern-recognition to detect faults.
- The model should be “portable” /easily integrated (i.e. Azure, AWS)
- Final app should incorporate rule-based and pattern-recognition diagnostic model along with waveform, FFT and fault overlay visualization.

Please Review the Following Videos

MACHINERY MONITORING SYSTEMS, LLC

https://www.youtube.com/watch?v=w9QeqCQ_EWA

https://www.youtube.com/watch?v=nB-c0in5XIk

https://www.youtube.com/watch?v=BPMjYJ_HoWk&list=PLDNHqPpwBs8O2QIGHdi8Bwu3p-WbTLsXG

https://www.youtube.com/watch?v=r214-8_gbjk

https://www.youtube.com/watch?v=HRjmPe0fncA

MachineryRx®

Class Structure & Materials

Success: Diagnose Defect w/Model

Structure

- Provide materials (Below)
- Build Fault Detection Models, Based on Charts & Fault Frequency Lookup Library
- Simulate Vibration Waveforms, Verify on Rotor Kit
- Use Model to Detect Simulated Faults (Unbalance, Misalignment, etc.)

Materials:

✓ Fault Charts
✓ Fault Frequency Library (Available Online)
- Remote Access to Rotor Kit & VM Module w/Embedded software (As illustrated in VM Rotor Kit Demo Video)
- Development Site Access
- Online Class (1-1.5 Hrs.)
- Rotor Kit
- VM Module

Types of Maintenance

MACHINERY
MONITORING
SYSTEMS, LLC

1- Reactive Maintenance

Repair as needed AKA – Run to Failure (Fix when it breaks)

2- Preventative Maintenance

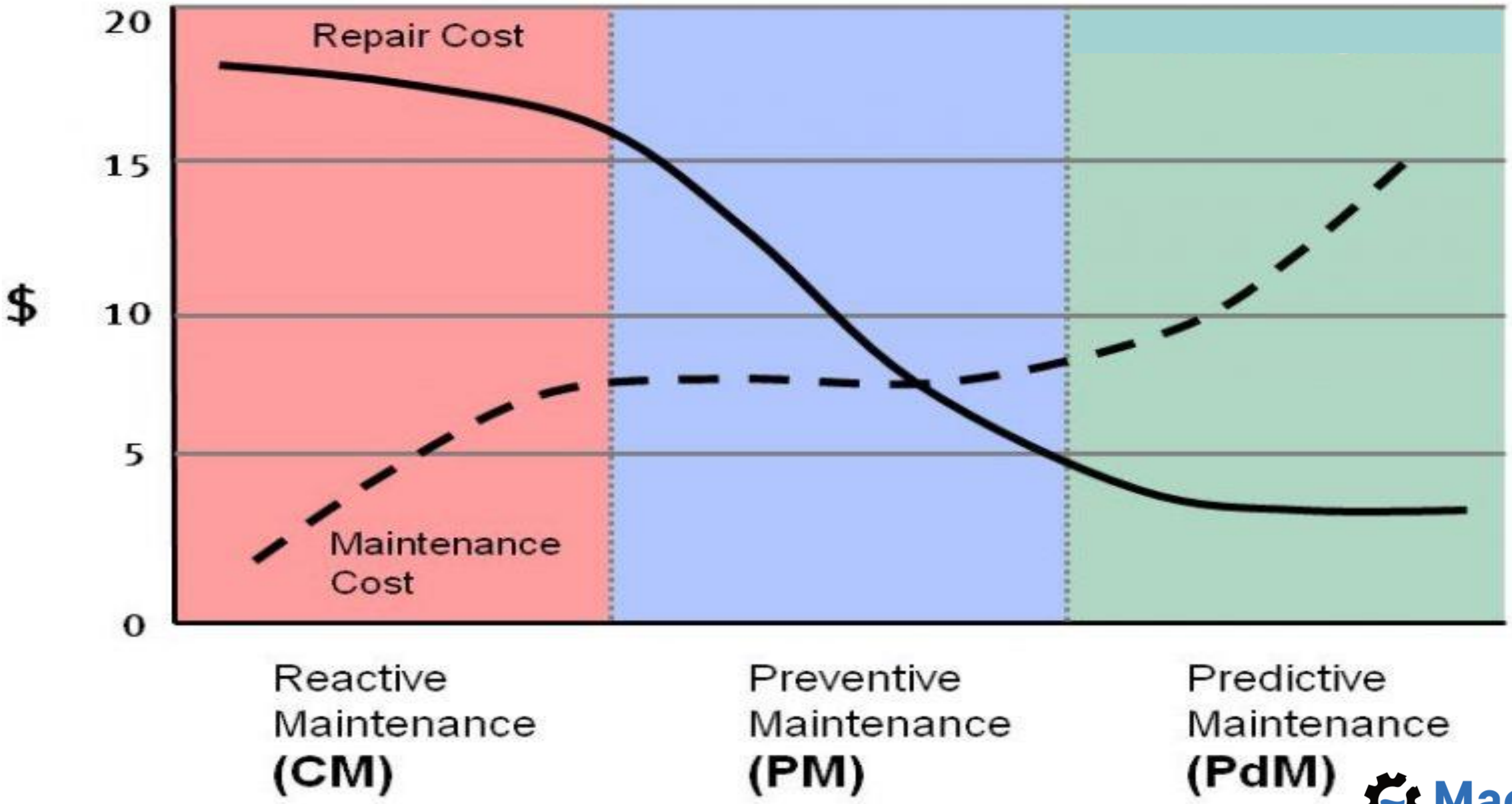
Scheduled/planned maintenance, performed on a regular basis

3-Predictive Maintenance

Condition based maintenance, Data driven, collected through sensors

MachineryRx®

Types of Maintenance vs. Cost



How Do We Utilize Vibration in Predictive Analysis?

- Amplitude

An indication of the **severity**

(amplitude is important ,but change in amplitude is more important)

- Frequency

An indication of the **nature** of the fault

(many tools are available to interpret the pattern)

- Motion

An indication of the **structural** response

(use phase to learn more about the fault and check for resonance)

How Do We Utilize Vibration in Predictive Analysis?

Frequency

- Different components within the machine generate different frequencies :shaft.vanes and blades
- Vibration at those frequencies tells us what part of the machine maybe damaged
- Its hard to look at waveform and relate the vibration to the machine

From Google AI

A Fast Fourier Transform (FFT) is a computational tool that can be used to analyze vibration waveforms by converting time-domain data into the frequency domain. This process allows engineers to see the frequency components of a signal, rather than just the sum of those components.

Here are some things to know about using an FFT to analyze vibration waveforms:

Frequency resolution

The resolution of an FFT is directly related to the signal's length and sample rate. To improve the resolution, the recording time needs to be extended.

Separating signals:

FFT analysis can be used to separate vibration signals from multiple sources into the frequency domain.

Understanding the source of vibration

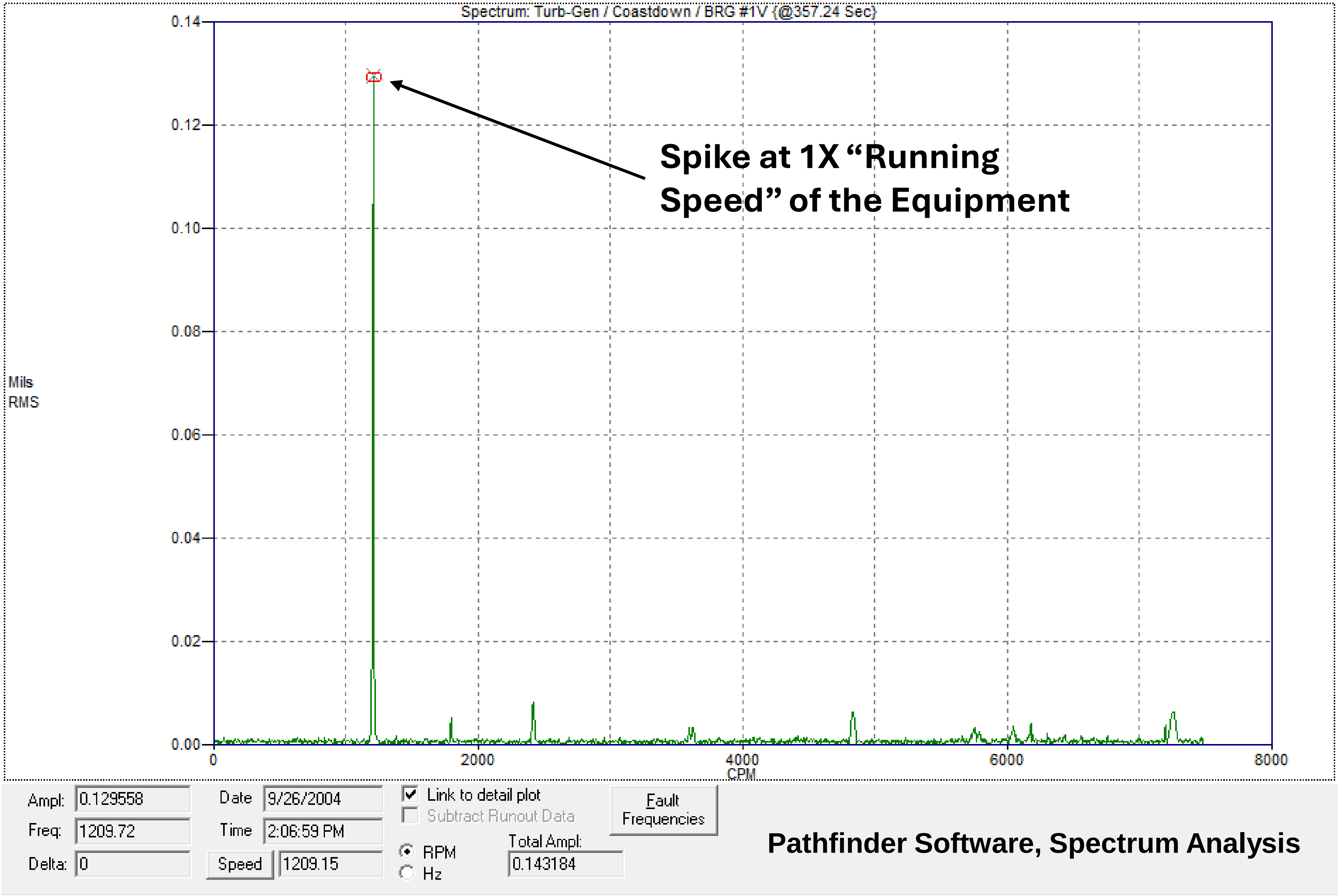
An FFT can help identify the source of vibration, such as an unbalanced mass on a rotating machine.

Understanding the machine's health

Studying vibration data can help determine the overall health of a machine, including how often there are abnormal vibrations and where they occur in the machine's work cycle.

How Do We Utilize Vibration in Predictive Analysis?

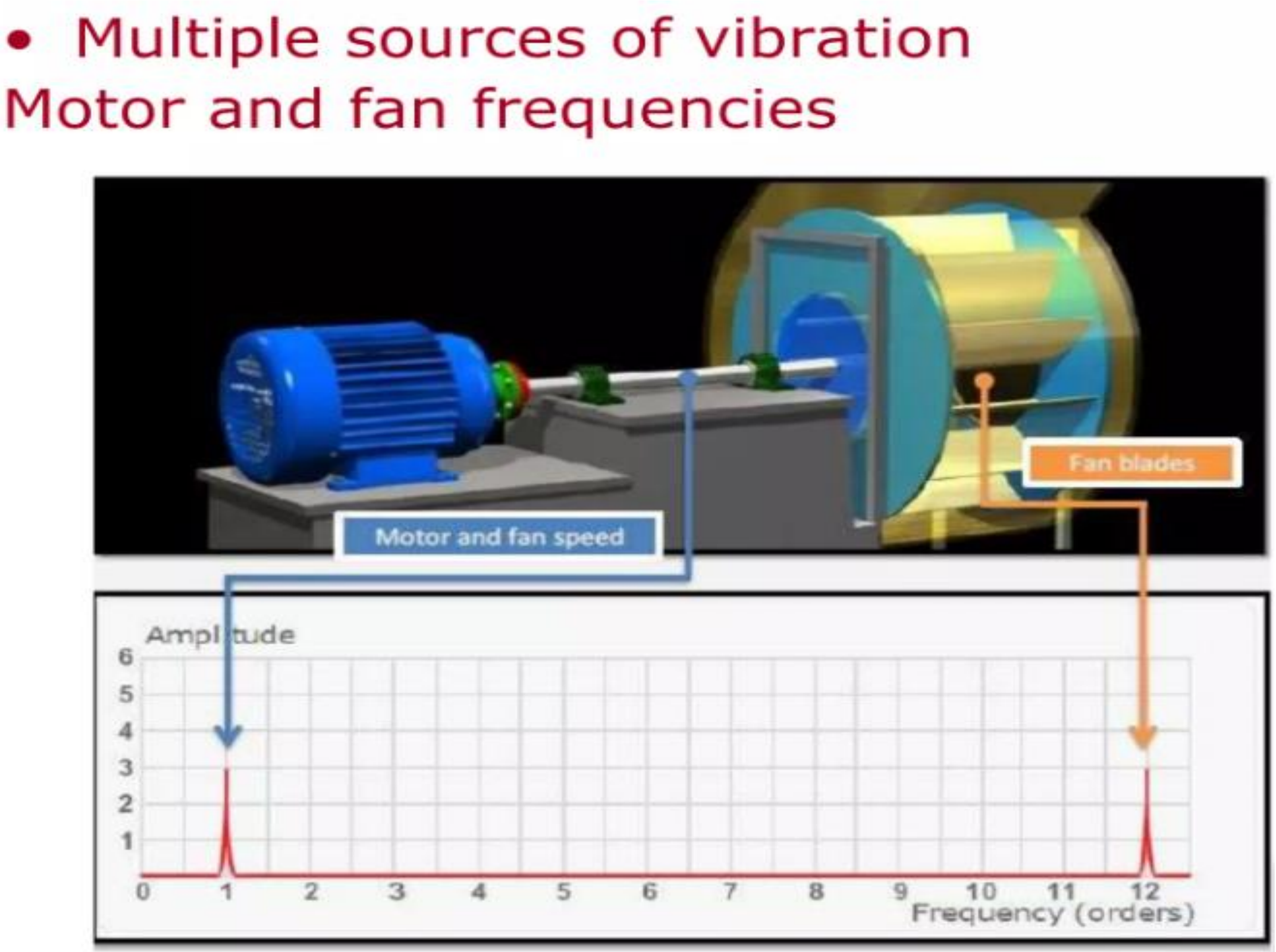
Machinery Monitoring Systems, LLC



Pathfinder Software, Spectrum Analysis

Spectrum Analysis Basics

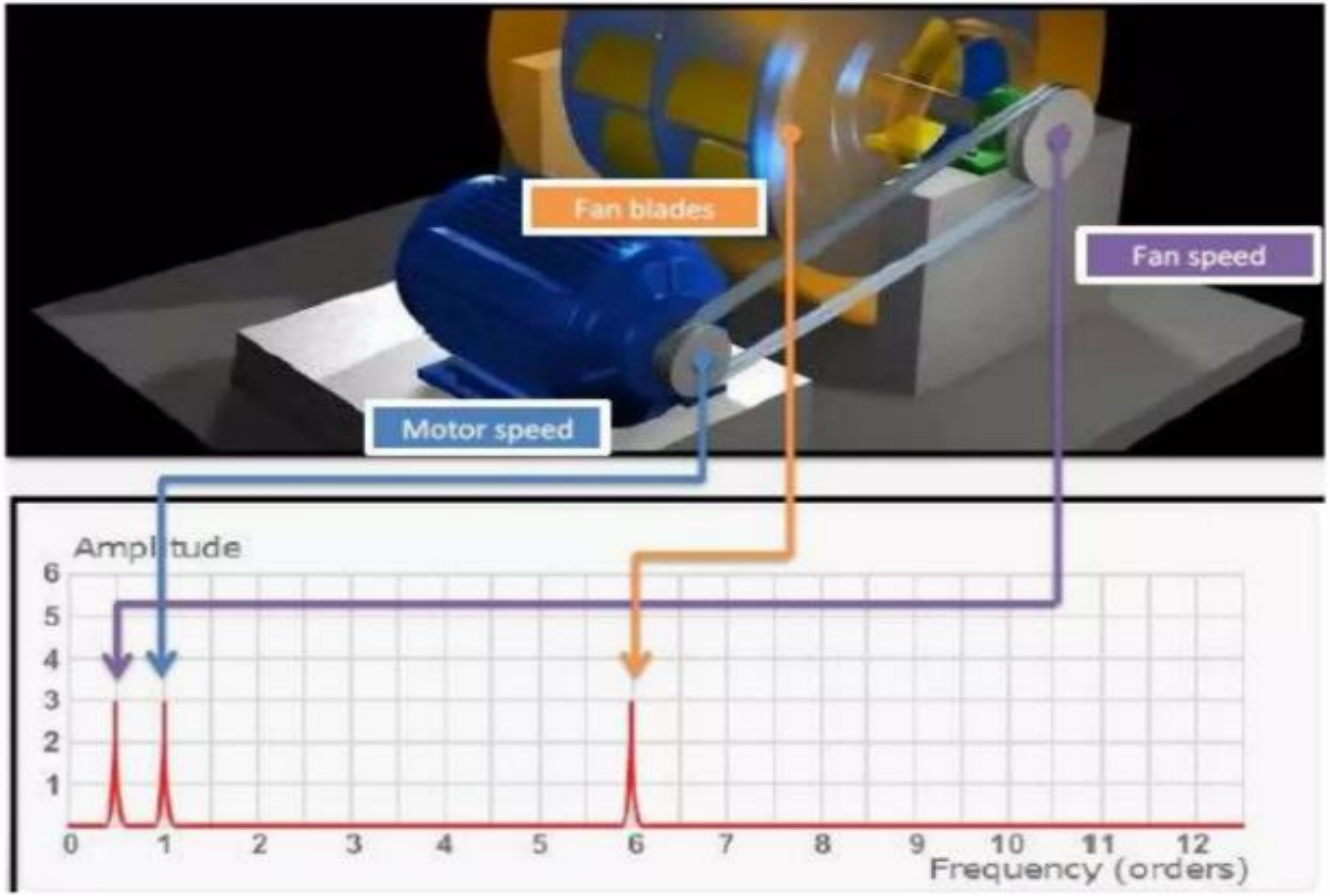
Machinery Monitoring Systems, LLC



MachineryRx®

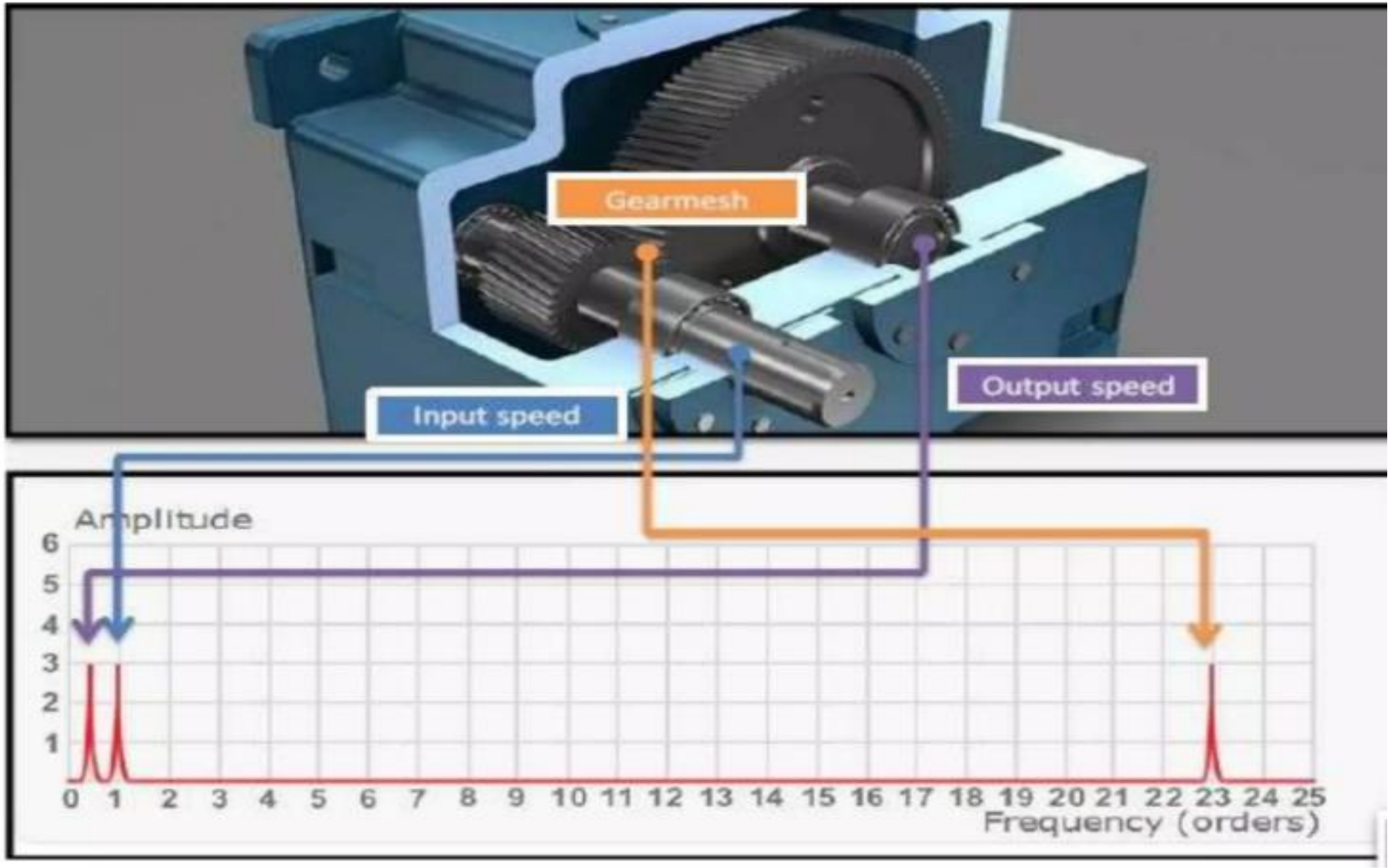
A Belt Drive Halves (1/2) the Speed

Machinery Monitoring Systems, LLC



MachineryRx®

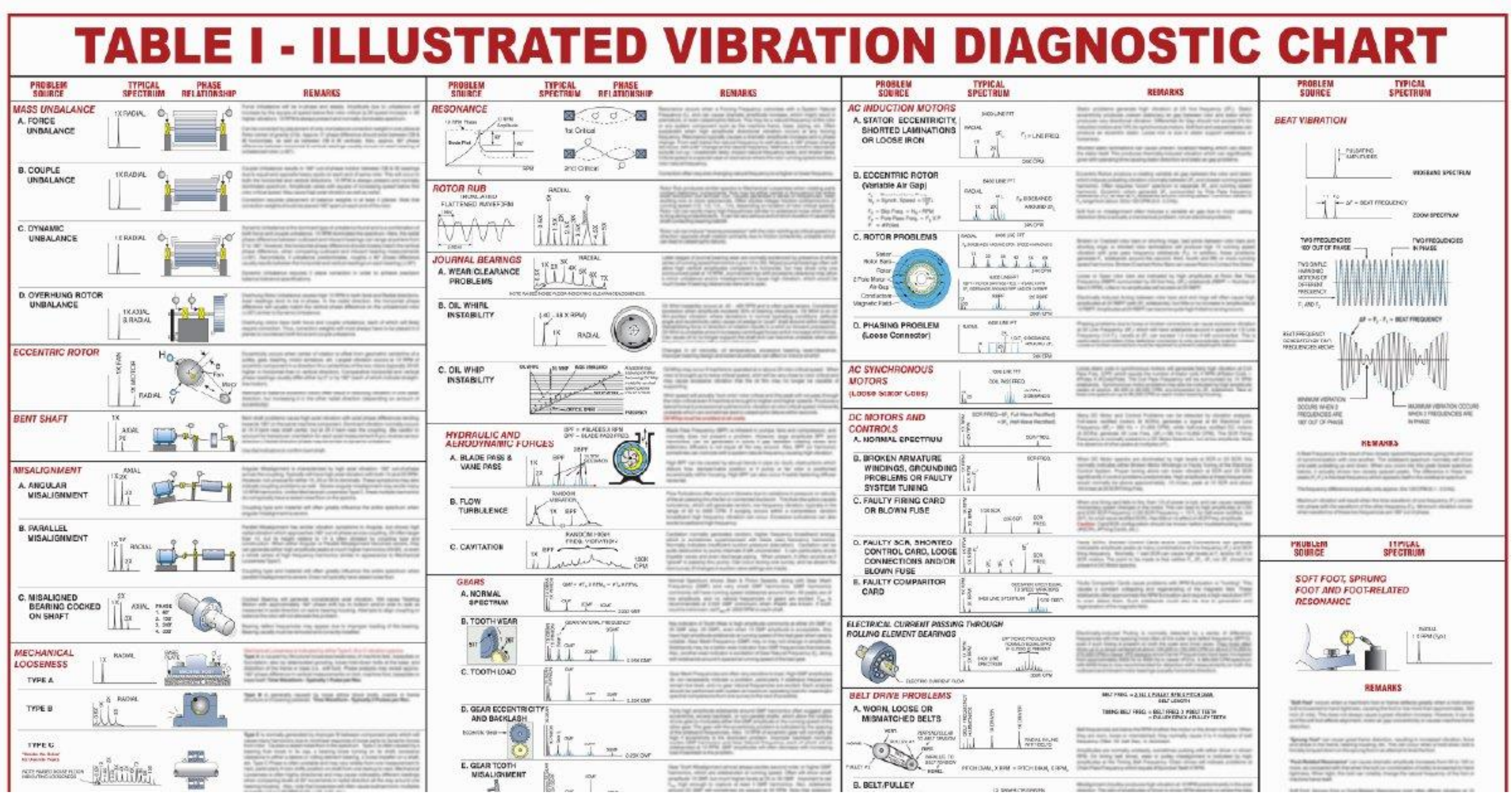
Gearbox Frequencies



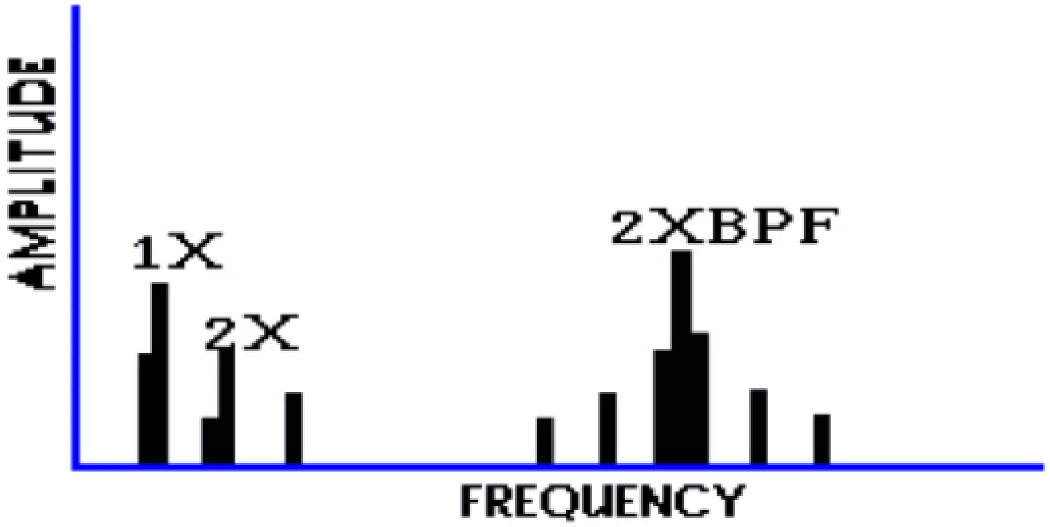
MACHINERY MONITORING SYSTEMS, LLC

MachineryRx

The table below is an example of a standard vibration diagnostic chart from which we could derive our <u>Rule Based Model.</u> The following slides illustrate each segment or failure mode of this chart



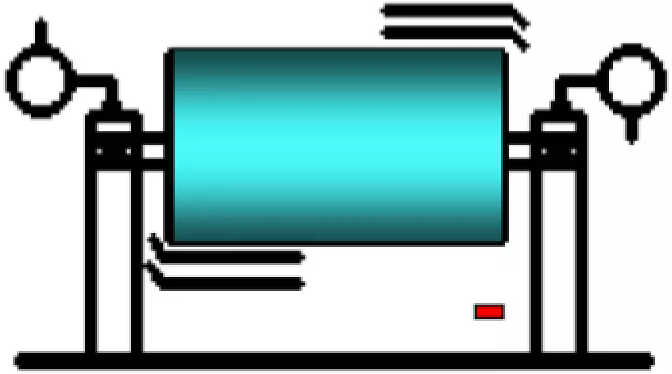
SIGNATURE ANALYSIS

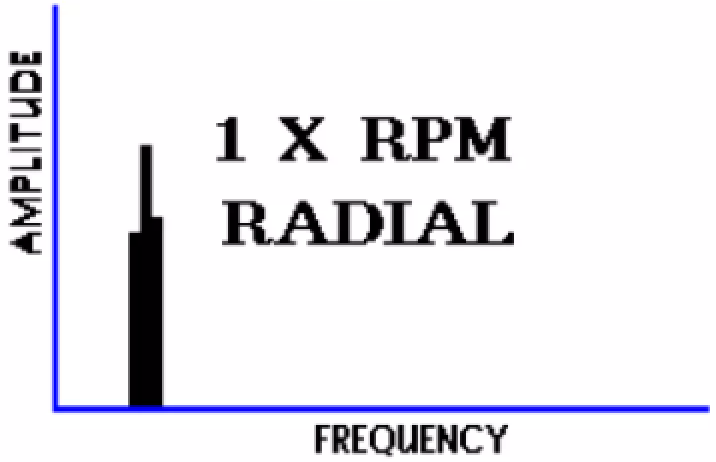


- Which frequencies exist and what are the relationships to the fundamental exciting frequencies.
- What are the amplitudes of each peak
- How do the peaks relate to each other
- If there are significant peaks, what are their source

Machinery Monitoring Systems, LLC

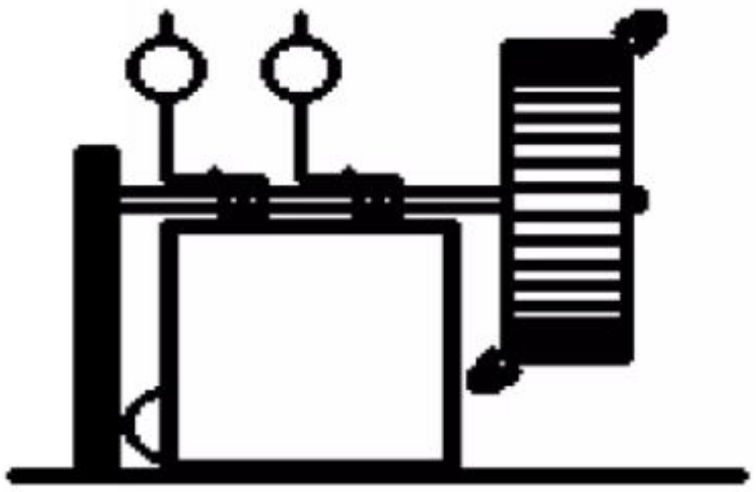
COUPLE UNBALANCE





- 180^0 out of phase on the same shaft
- 1X RPM always present and normally dominates
- Amplitude varies with square of increasing speed
- Can cause high axial as well as radial amplitudes
- Balancing requires Correction in two planes at 180^o

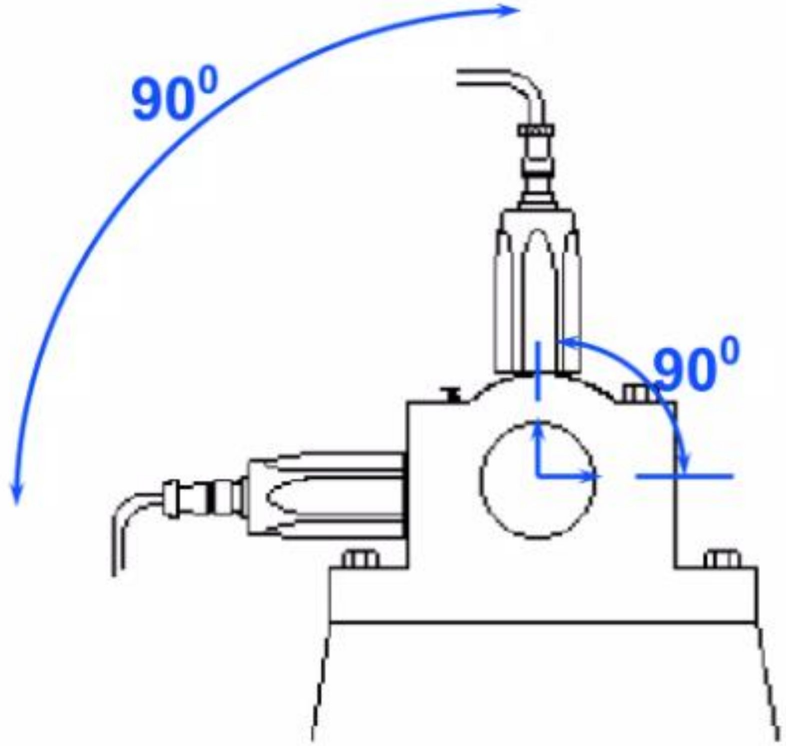
OVERHUNG ROTOR UNBALANCE



- 1X RPM present in radial and axial directions
- Axial readings tend to be in-phase but radial readings might be unsteady
- Overhung rotors often have both force and couple unbalance each of which may require correction

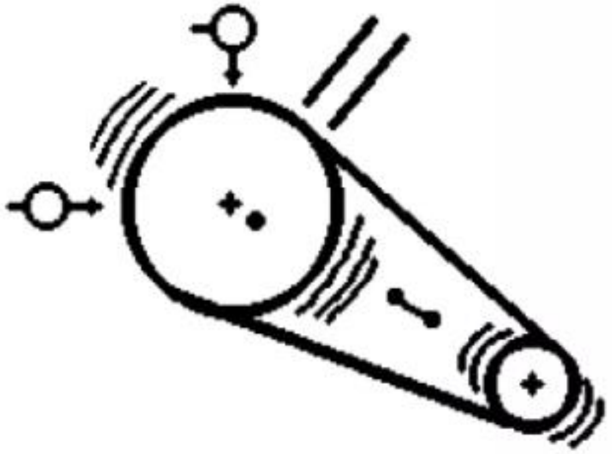
Diagnosing Unbalance

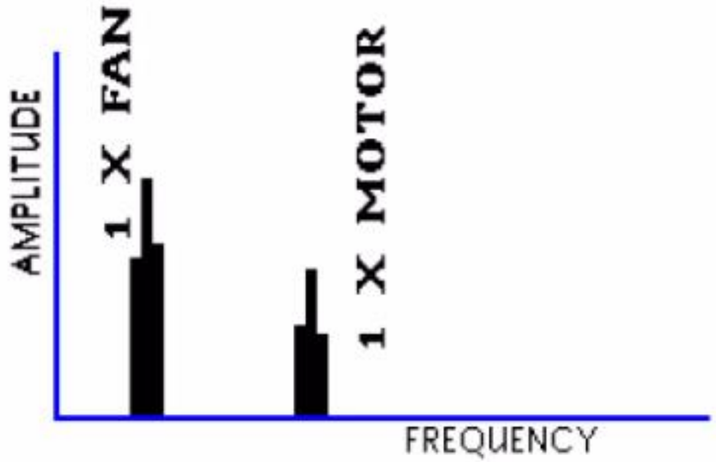
- Vibration frequency equals rotor speed.
- Vibration predominantly RADIAL in direction.
- Stable vibration phase measurement.
- Vibration increases as square of speed.
- Vibration phase shifts in direct proportion to measurement direction.



Machinery Monitoring Systems, LLC

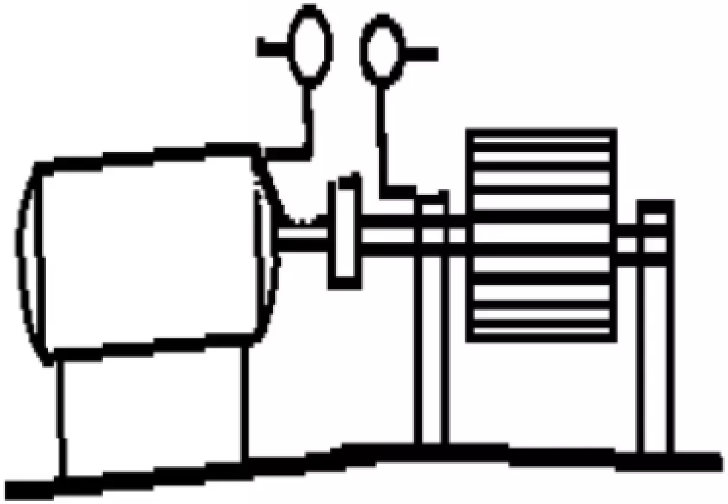
ECCENTRIC ROTOR

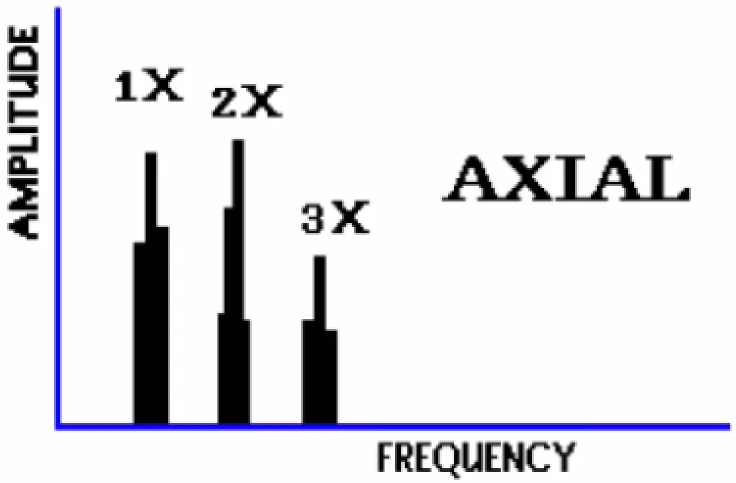




- Largest vibration at 1X RPM in the direction of the centerline of the rotors
- Comparative phase readings differ by 0^0 or 180^0
- Attempts to balance will cause a decrease in amplitude in one direction but an increase may occur in the other direction

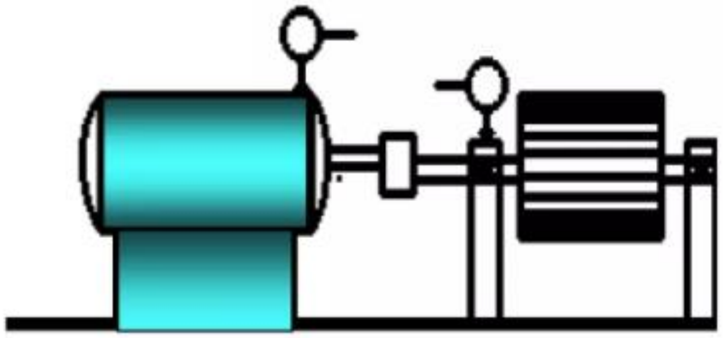
ANGULAR MISALIGNMENT

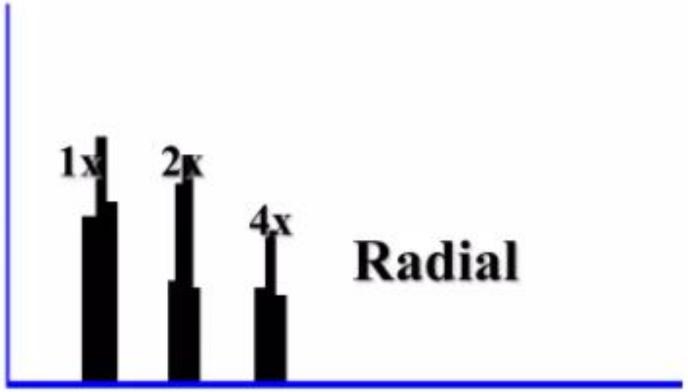




- Characterized by high axial vibration
- 180^0 phase change across the coupling
- Typically high 1 and 2 times axial vibration
- Not unusual for 1, 2 or 3X RPM to dominate
- Symptoms could indicate coupling problems

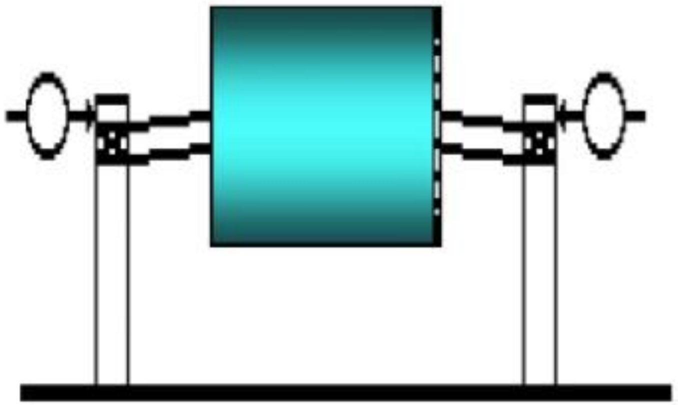
PARALLEL MISALIGNMENT

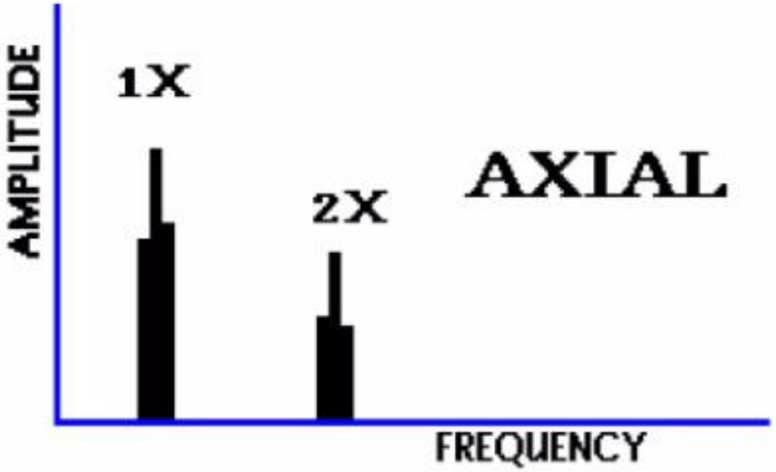




- High radial vibration 180^0 out of phase
- Severe conditions give higher harmonics
- 2X RPM often larger than 1X RPM
- Similar symptoms to angular misalignment
- Coupling design can influence spectrum shape and amplitude

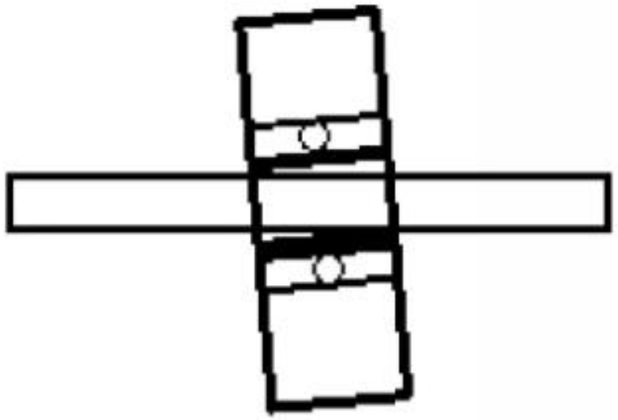
BENT SHAFT

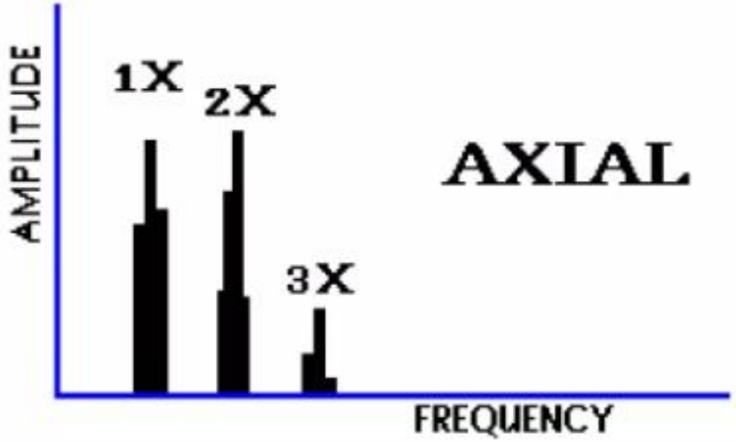




- Bent shaft problems cause high axial vibration
- 1X RPM dominant if bend is near shaft center
- 2X RPM dominant if bend is near shaft ends
- Phase difference in the axial direction will tend towards 180^0 difference

MISALIGNED BEARING





- Vibration symptoms similar to angular misalignment
- Attempts to realign coupling or balance the rotor will not alleviate the problem.
- Will cause a twisting motion with approximately 180^0 phase shift side to side or top to bottom

OTHER SOURCES OF HIGH AXIAL VIBRATION

a. Bent Shafts

b. Shafts in Resonant Whirl

c. Bearings Cocked on the Shaft

d. Resonance of Some Component in the Axial Direction

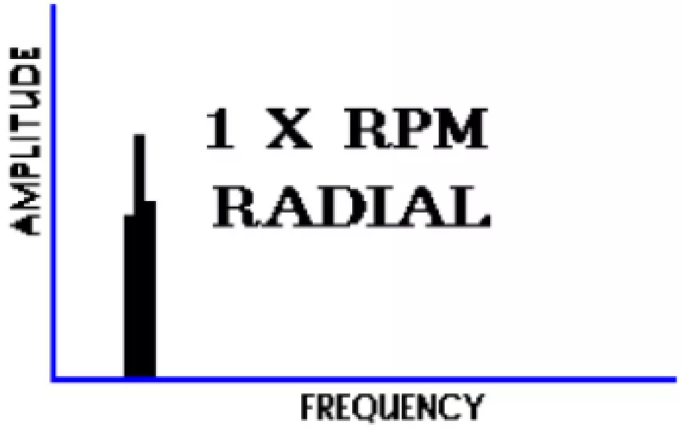
e. Worn Thrust Bearings

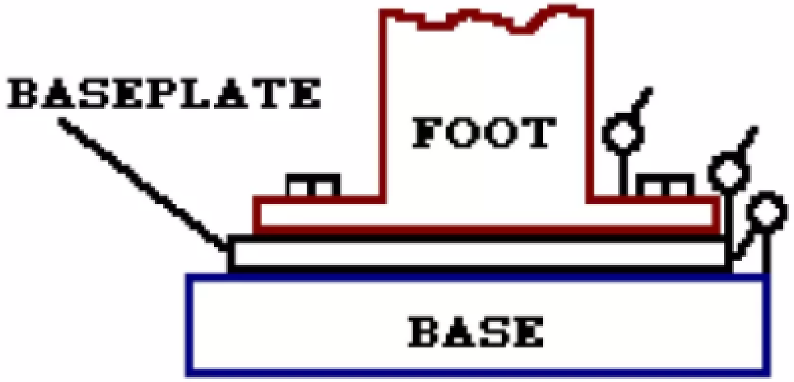
f. Worn Helical or Bevel Gears

g. A Sleeve Bearing Motor Hunting for its Magnetic Center

h. Couple Component of a Dynamic Unbalance

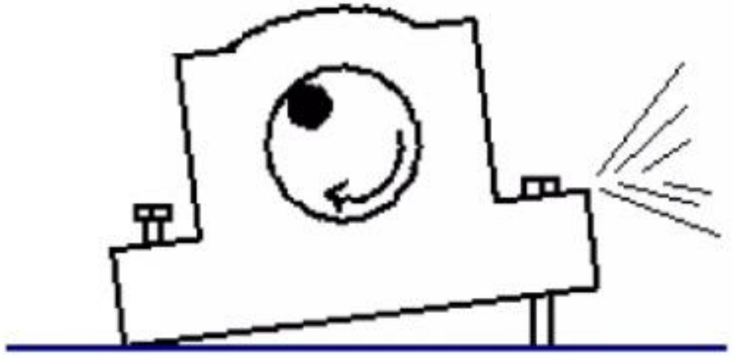
MECHANICAL LOOSENESS (A)

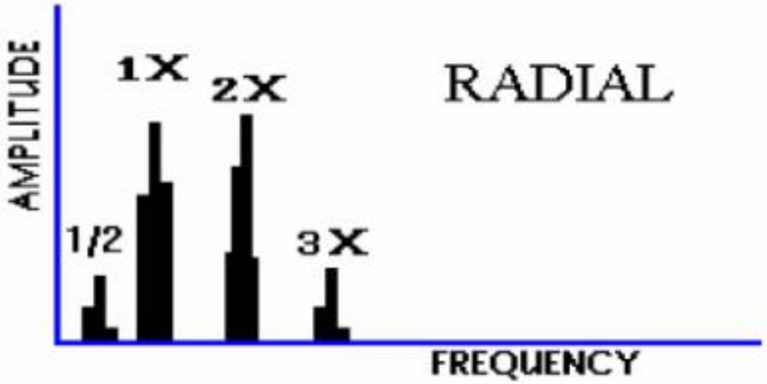




- Caused by structural looseness of machine feet
- Distortion of the base will cause "soft foot" problems
- Phase analysis will reveal aprox 180^0 phase shift in the vertical direction between the baseplate components of the machine

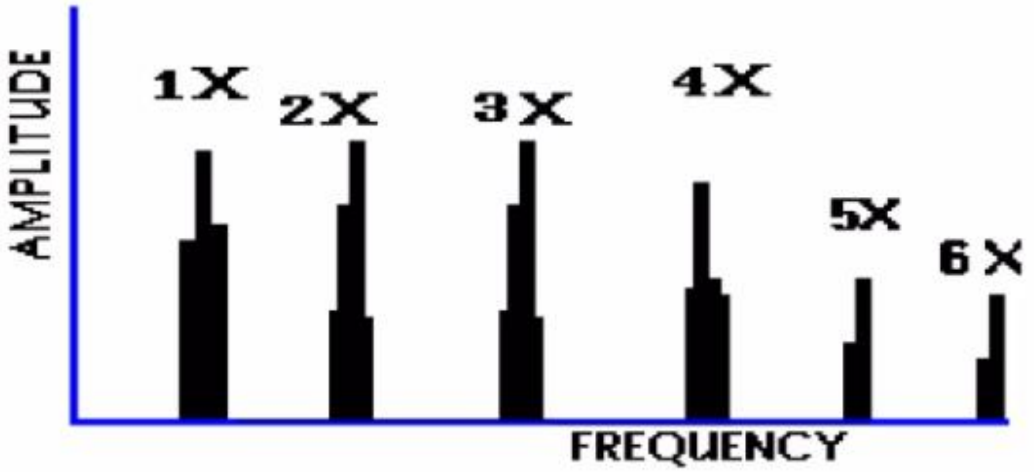
MECHANICAL LOOSENESS (B)





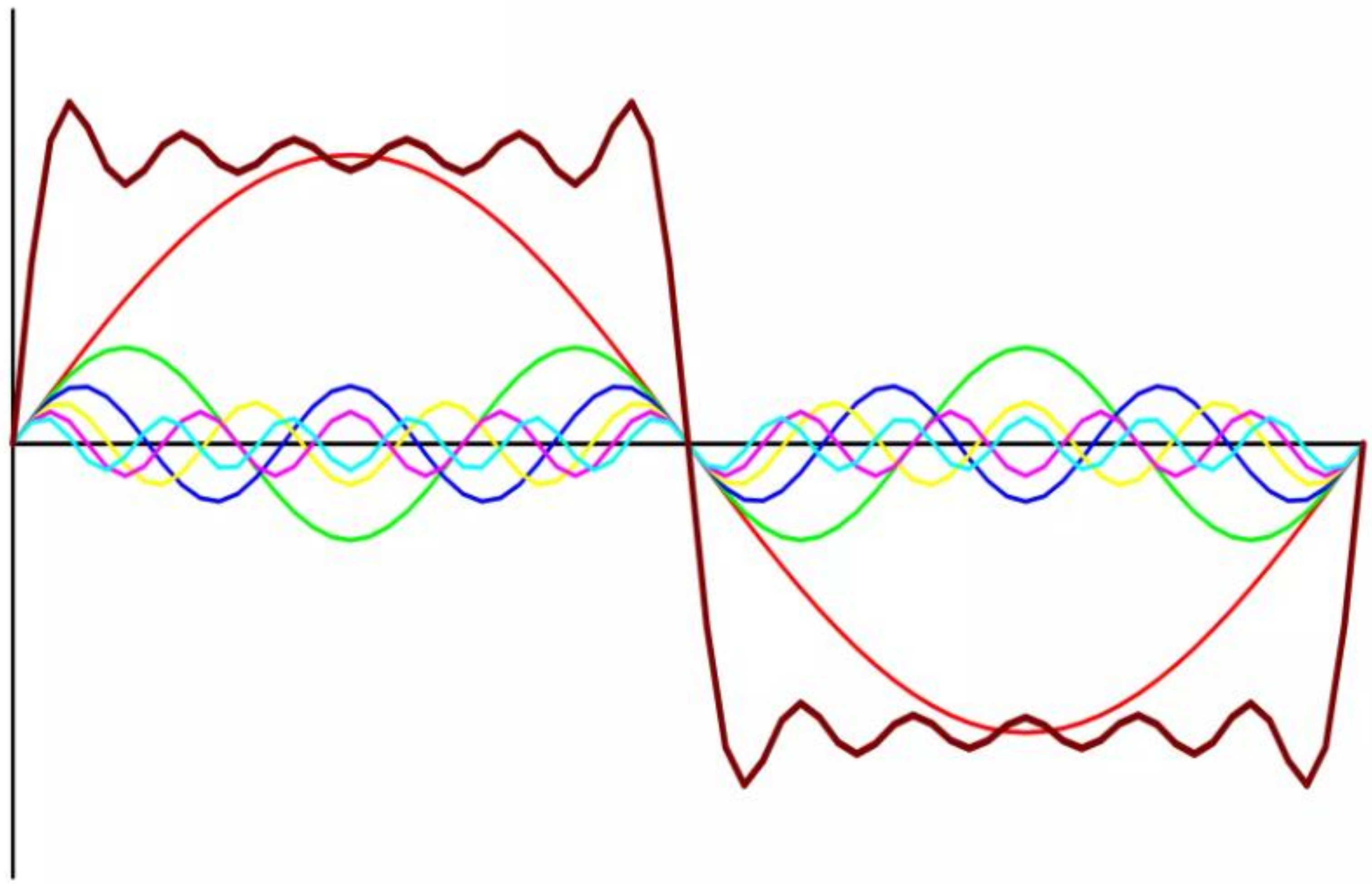
- Caused by loose pillowblock bolts
- Can cause 0.5, 1, 2 and 3X RPM
- Sometimes caused by cracked frame structure or bearing block

SLEEVE BEARING WEAR / CLEARANCE PROBLEMS



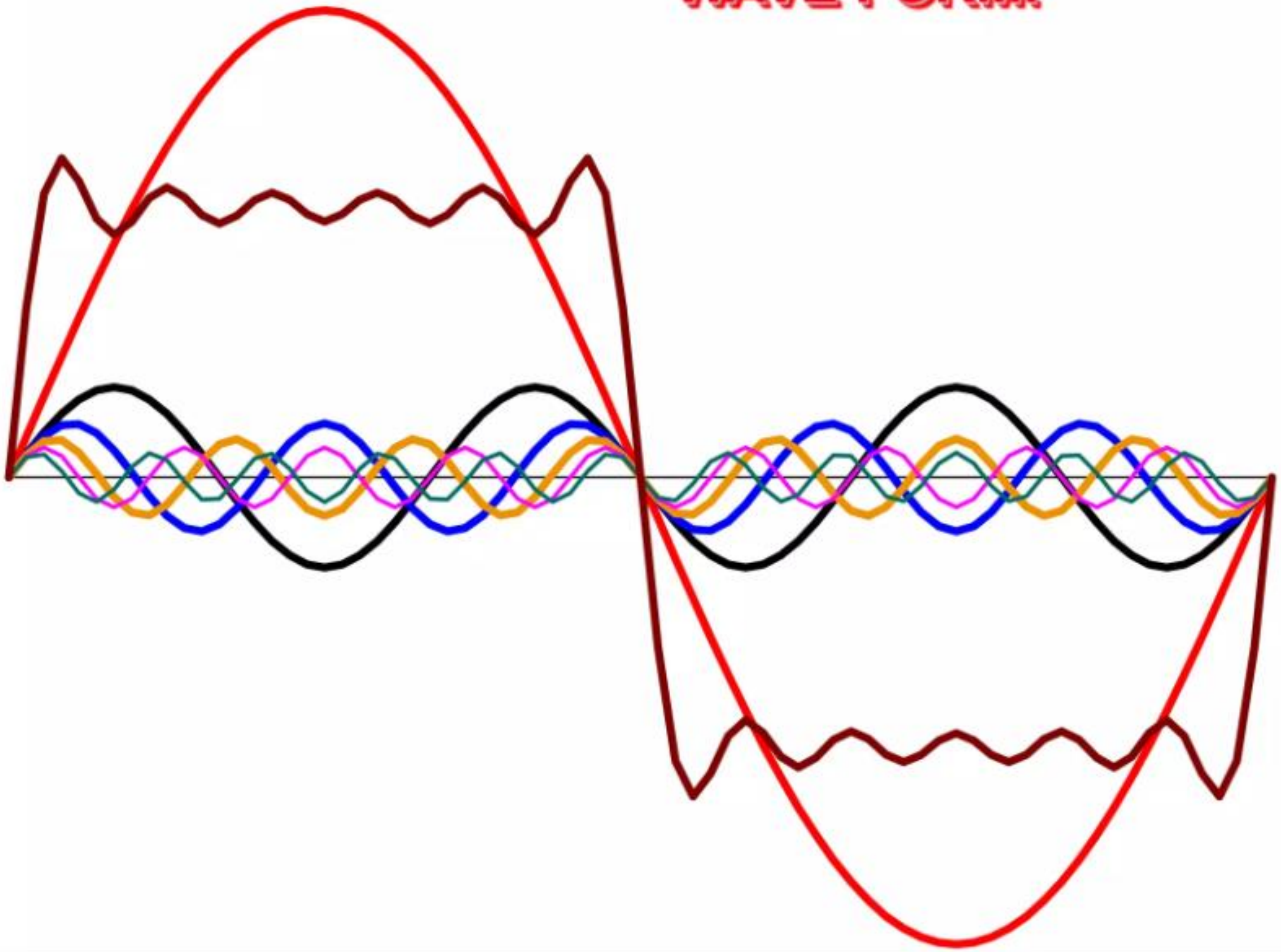
- Later stages of sleeve bearing wear will give a large family of harmonics of running speed
- A minor unbalance or misalignment will cause high amplitudes when excessive bearing clearances are present

COMPONENT FREQUENCIES OF A SQUARE WAVE FORM.



COMPONENT FREQUENCIES OF A SQUARE WAVE FORM.

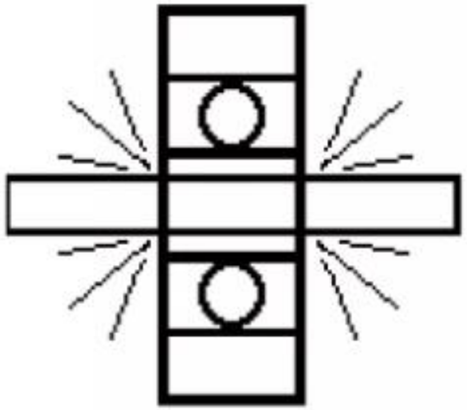
Machinery Monitoring Systems, LLC

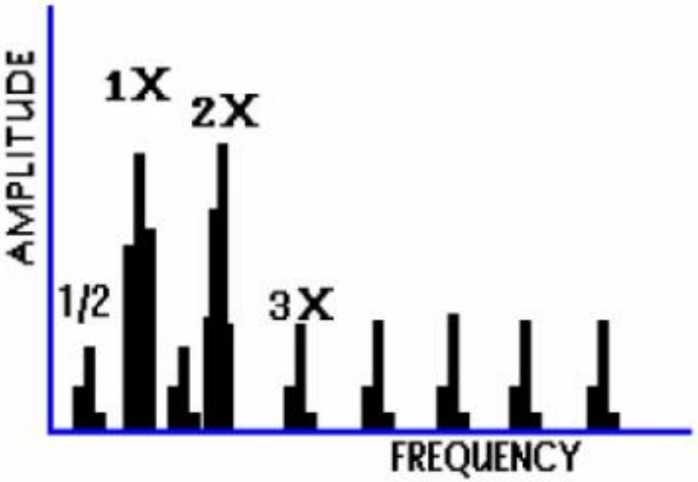


MachineryRx®

Machinery Monitoring Systems, LLC

MECHANICAL LOOSENESS (C)

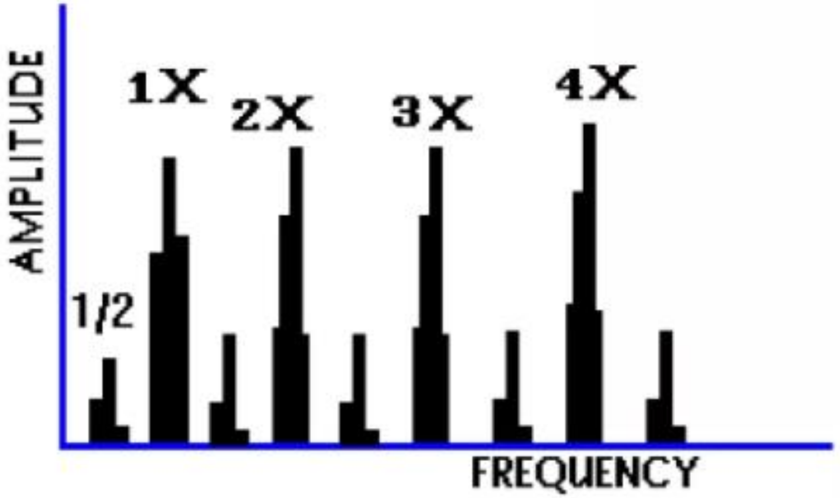


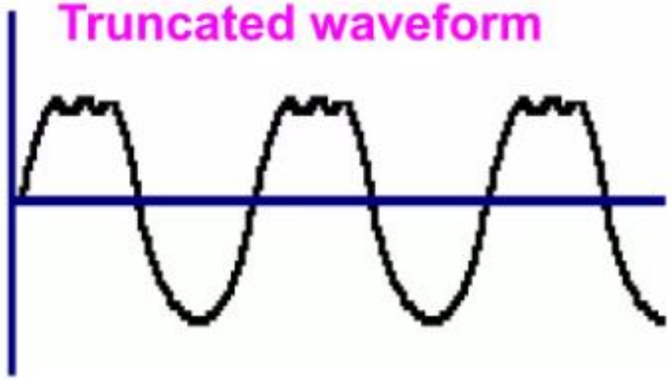


- Phase is often unstable
- Will have many harmonics
- Can be caused by a loose bearing liner, excessive bearing clearance or a loose impeller on a shaft

MachineryRx

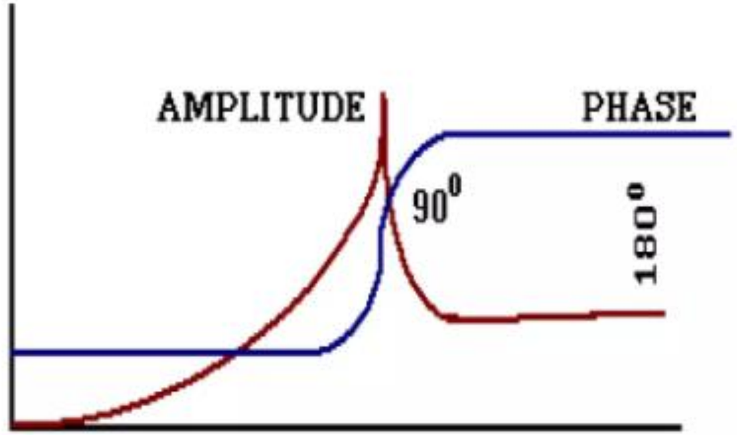
ROTOR RUB

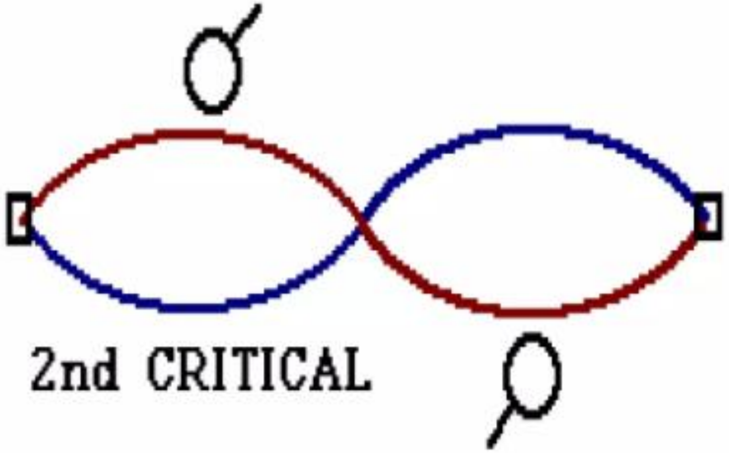




- Similar spectrum to mechanical looseness
- Usually generates a series of frequencies which may excite natural frequencies
- Subharmonic frequencies may be present
- Rub may be partial or through a complete revolution.

RESONANCE

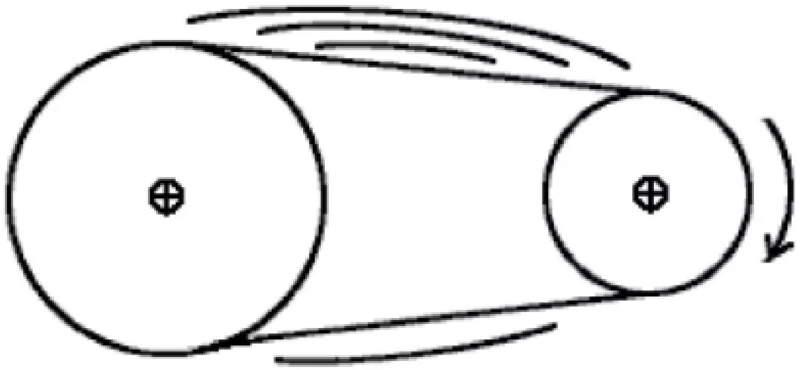


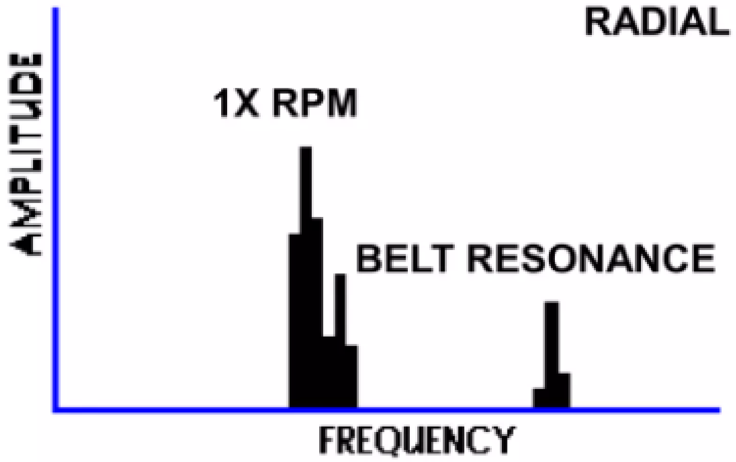


- Resonance occurs when the Forcing Frequency coincides with a Natural Frequency
- 180^0 phase change occurs when shaft speed passes through resonance
- High amplitudes of vibration will be present when a system is in resonance

BELT PROBLEMS (D)

BELT RESONANCE

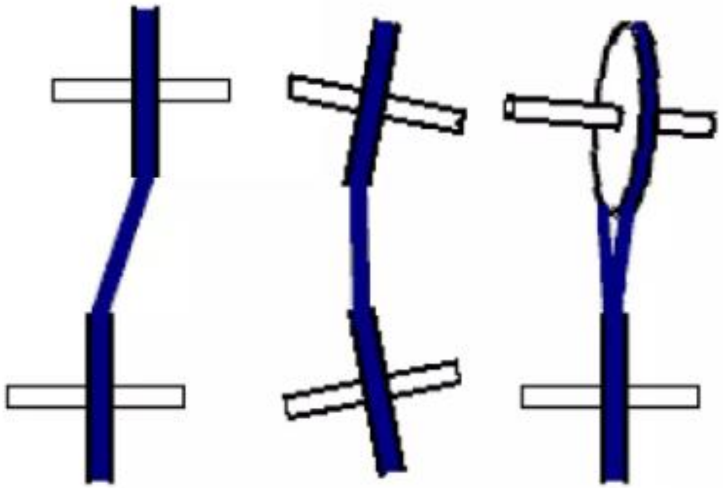


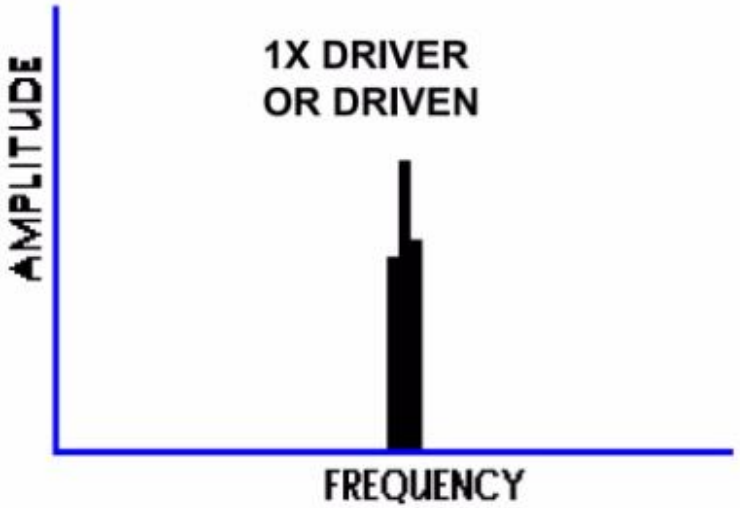


- High amplitudes can be present if the belt natural frequency coincides with driver or driven RPM
- Belt natural frequency can be changed by altering the belt tension

BELT PROBLEMS (B)

BELT / PULLEY MISALIGNMENT

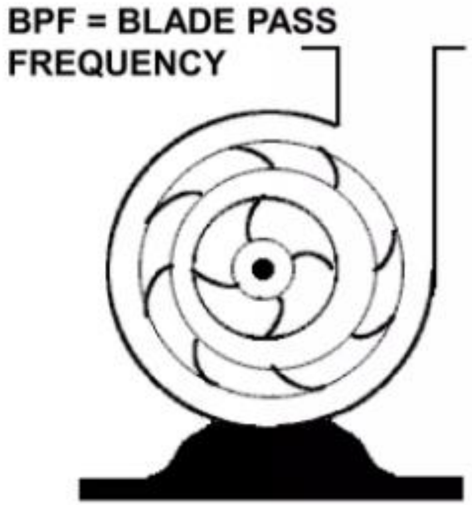


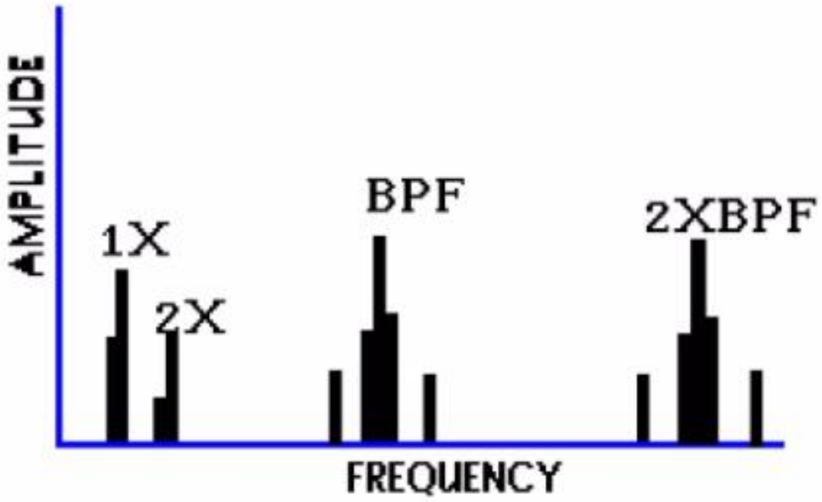


- Pulley misalignment will produce high axial vibration at 1X RPM
- Often the highest amplitude on the motor will be at the fan RPM

HYDRAULIC AND AERODYNAMIC FORCES

MACHINERY MONITORING SYSTEMS, LLC

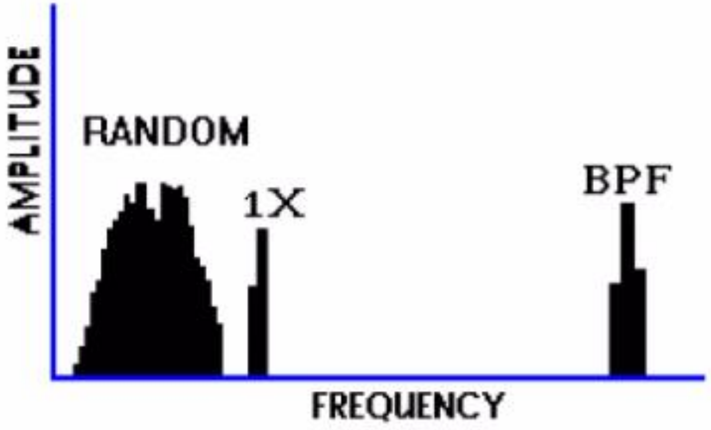




- If gap between vanes and casing is not equal, Blade Pass Frequency may have high amplitude
- High BPF may be present if impeller wear ring seizes on shaft
- Eccentric rotor can cause amplitude at BPF to be excessive

HYDRAULIC AND AERODYNAMIC FORCES

FLOW TURBULENCE

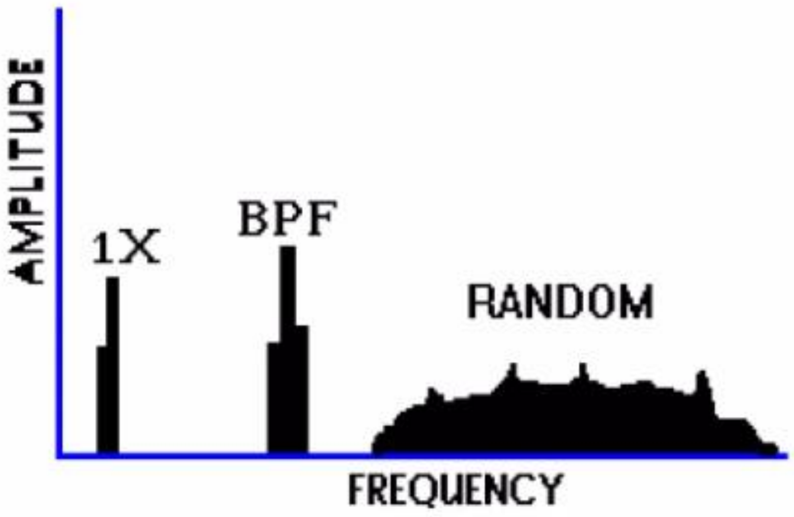


- Flow turbulence often occurs in blowers due to variations in pressure or velocity of air in ducts
- Random low frequency vibration will be generated, possibly in the 50 - 2000 CPM range

HYDRAULIC AND AERODYNAMIC FORCES

Machinery Monitoring Systems, LLC

CAVITATION

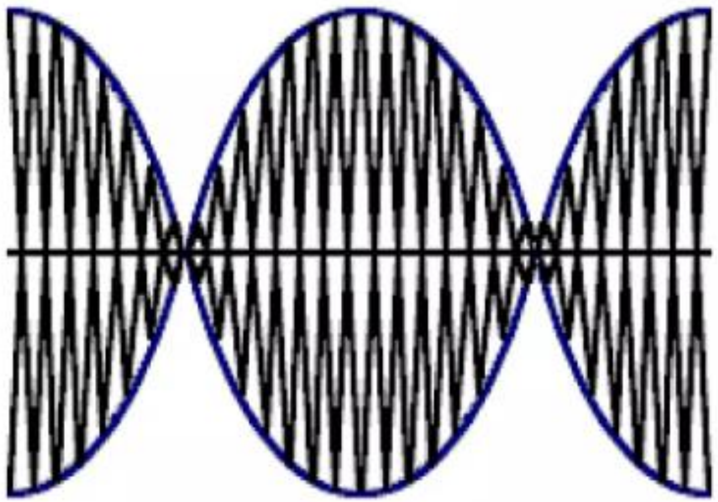


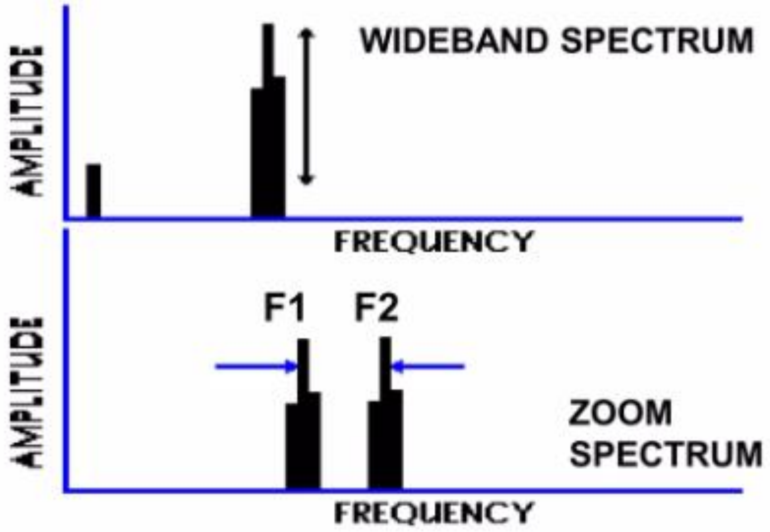
- Cavitation will generate random, high frequency broadband energy superimposed with BPF harmonics
- Normally indicates inadequate suction pressure
- Erosion of impeller vanes and pump casings may occur if left unchecked
- Sounds like gravel passing through pump

MachineryRx®

Machinery Monitoring Systems, LLC

BEAT VIBRATION



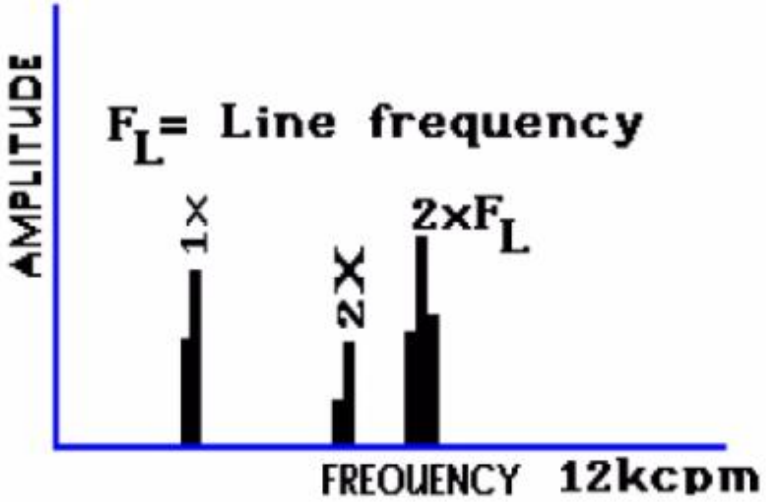


- A beat is the result of two closely spaced frequencies going into and out of phase
- The wideband spectrum will show one peak pulsating up and down
- The difference between the peaks is the beat frequency which itself will be present in the wideband spectrum

MachineryRx

ELECTRICAL PROBLEMS

STATOR ECCENTRICITY
SHORTED LAMINATIONS
AND LOOSE IRON



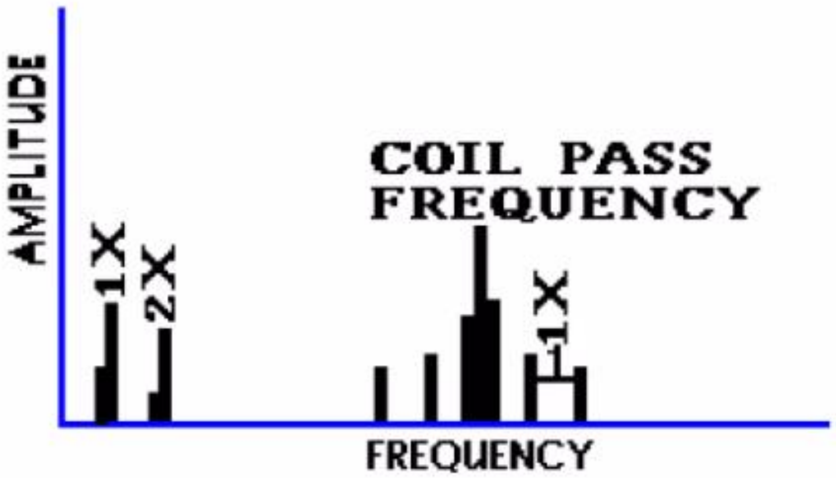
- Stator problems generate high amplitudes at $2F_L$ (2X line frequency)
- Stator eccentricity produces uneven stationary air gap, vibration is very directional
- Soft foot can produce an eccentric stator

FREQUENCIES PRODUCED BY ELECTRICAL MOTORS.

- Electrical line frequency.(FL) = 50Hz = 3000 cpm.
  60HZ = 3600 cpm
- No of poles. (P)
- Rotor Bar Pass Frequency (Fb) = No of rotor bars x Rotor rpm.
- Synchronous speed (Ns) = $\frac{2xFL}{P}$
- Slip frequency (F_S)= Synchronous speed - Rotor rpm.
- Pole pass frequency (F_P)= Slip Frequency x No of Poles.

ELECTRICAL PROBLEMS

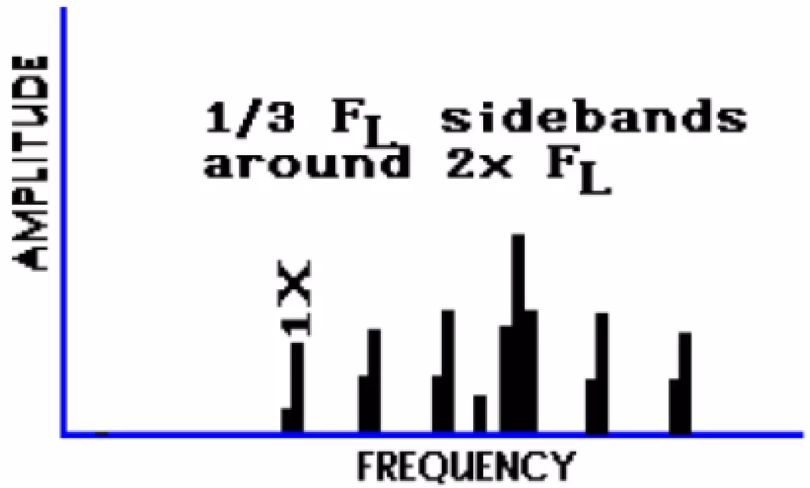
SYNCHRONOUS MOTOR
(Loose Stator Coils)



- Loose stator coils in synchronous motors generate high amplitude at Coil Pass Frequency
- The coil pass frequency will be surrounded by 1X RPM sidebands

ELECTRICAL PROBLEMS

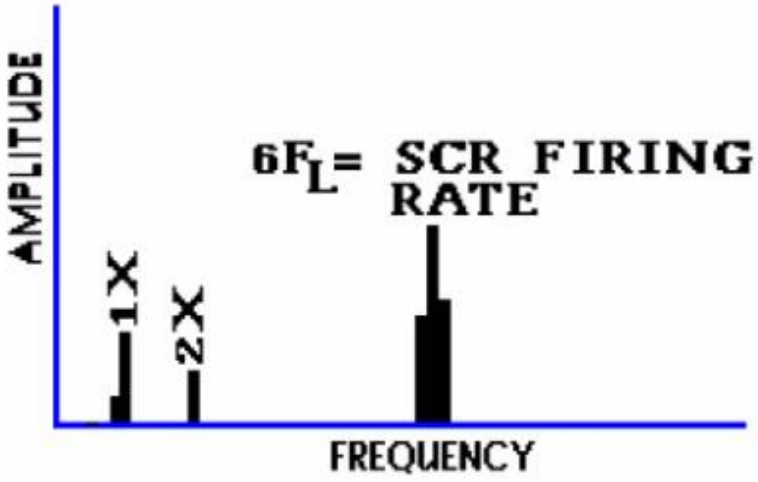
POWER SUPPLY PHASE PROBLEMS
(Loose Connector)



- Phasing problems can cause excessive vibration at $2F_L$ with 1/3 F_L sidebands
- Levels at $2F_L$ can exceed 25 mm/sec if left uncorrected
- Particular problem if the defective connector is only occasionally making contact

ELECTRICAL PROBLEMS

DC MOTOR PROBLEMS

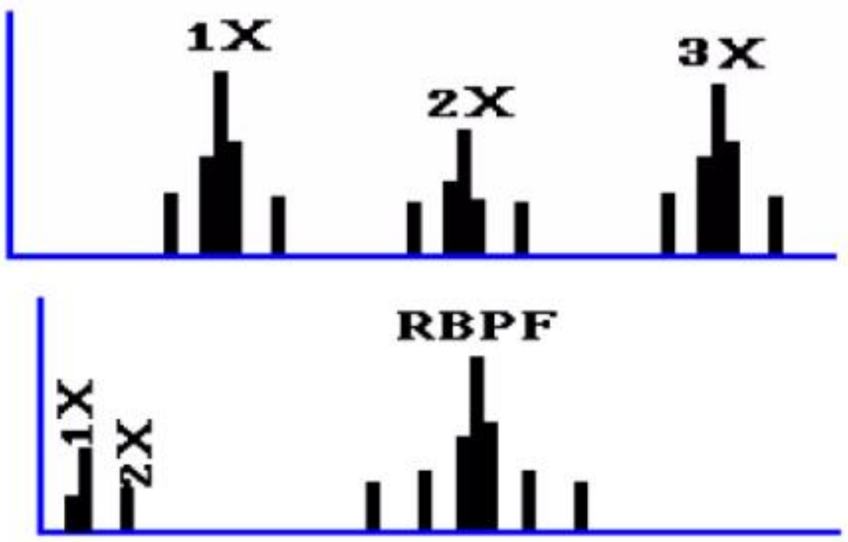


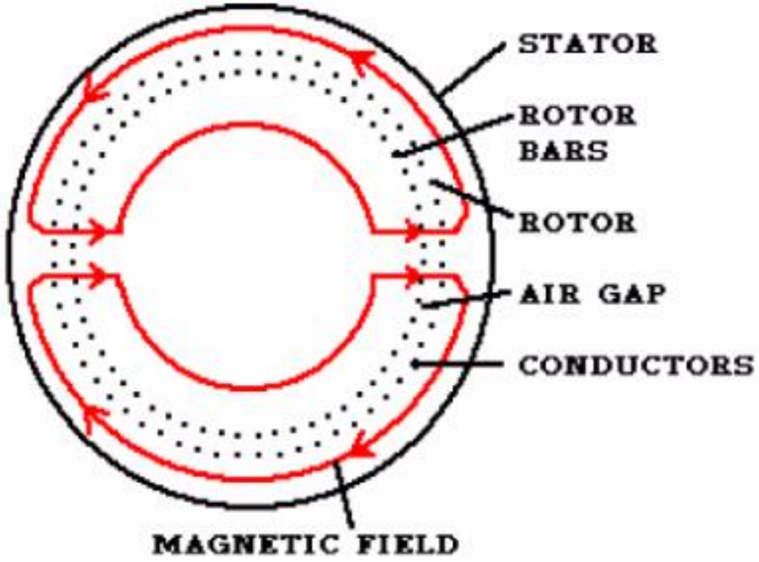
- DC motor problems can be detected by the higher than normal amplitudes at SCR firing rate
- These problems include broken field windings
- Fuse and control card problems can cause high amplitude peaks at frequencies of 1X to 5X Line Frequency

ELECTRICAL PROBLEMS

MACHINERY MONITORING SYSTEMS, LLC

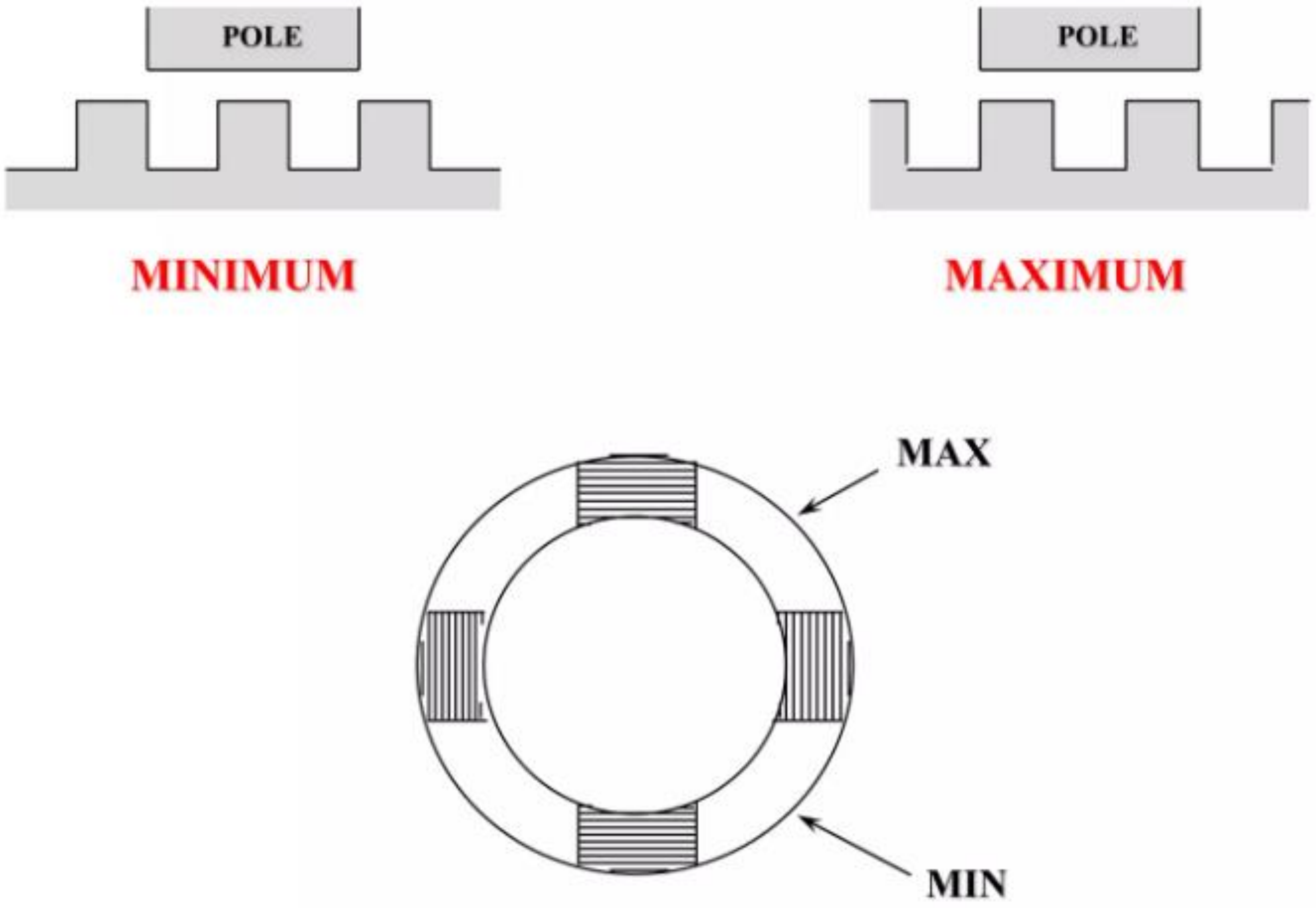
ROTOR PROBLEMS



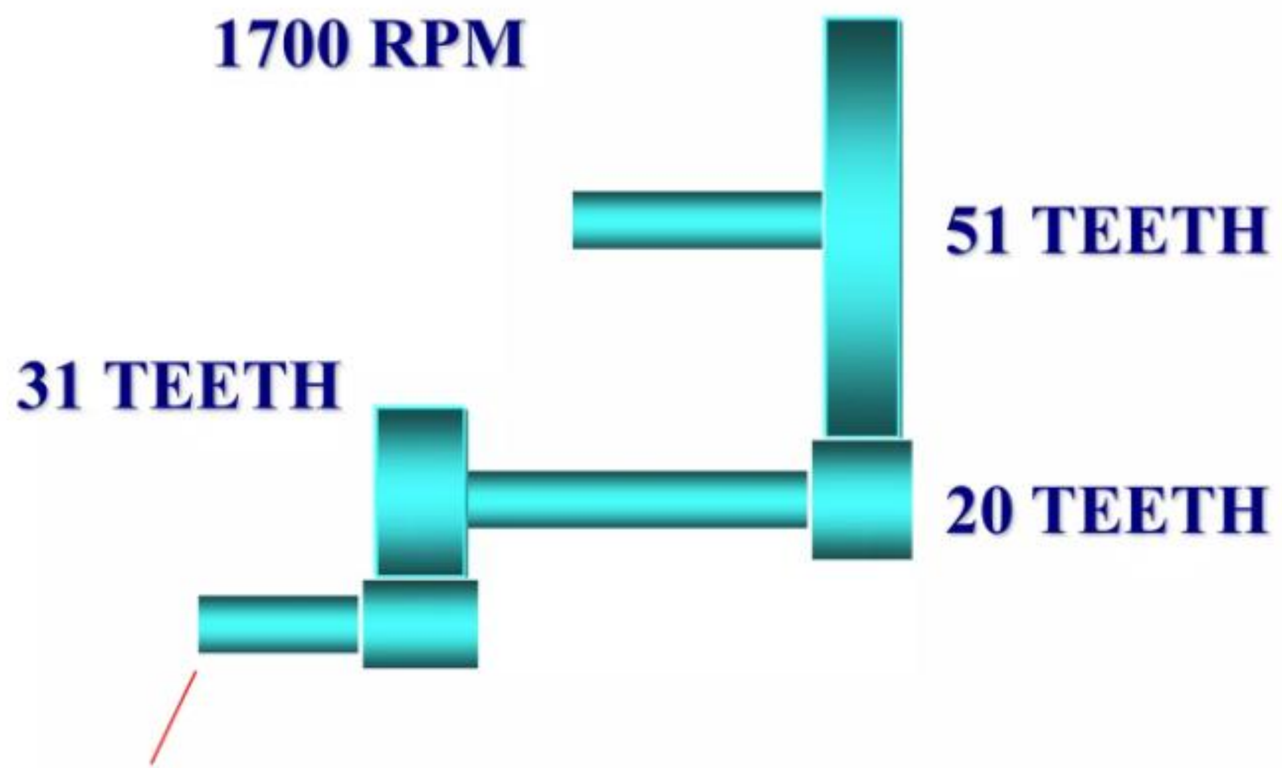


- 1X, 2X, 3X, RPM with pole pass frequency sidebands indicates rotor bar problems.
- 2X line frequency sidebands on rotor bar pass frequency (RBPF) indicates loose rotor bars.
- Often high levels at 2X & 3X rotor bar pass frequency and only low level at 1X rotor bar pass frequency.

ROTOR BAR FREQUENCIES (SLOT NOISE)

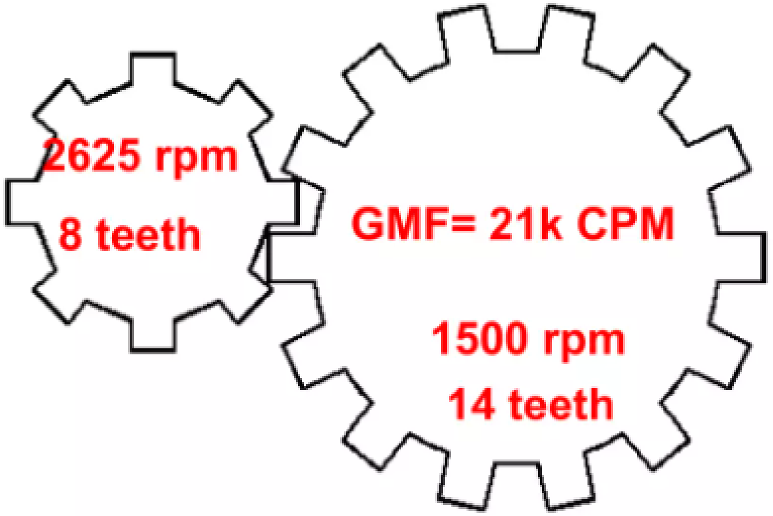


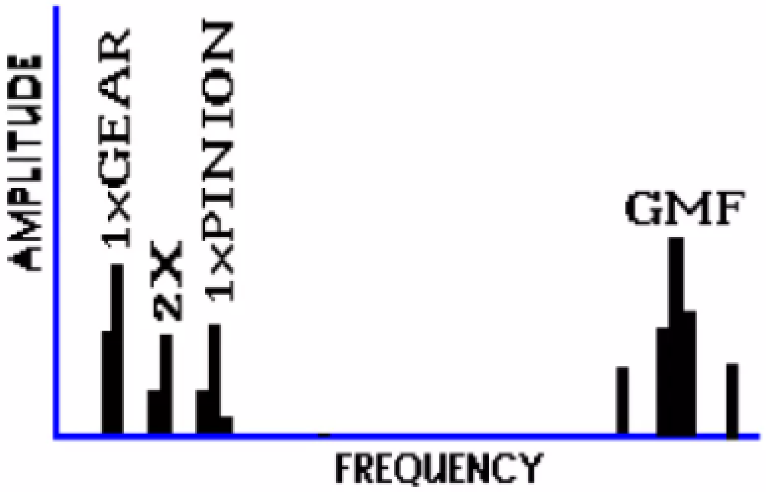
CALCULATION OF GEAR MESH FREQUENCIES



8959 RPM -- HOW MANY TEETH ON THIS GEAR?

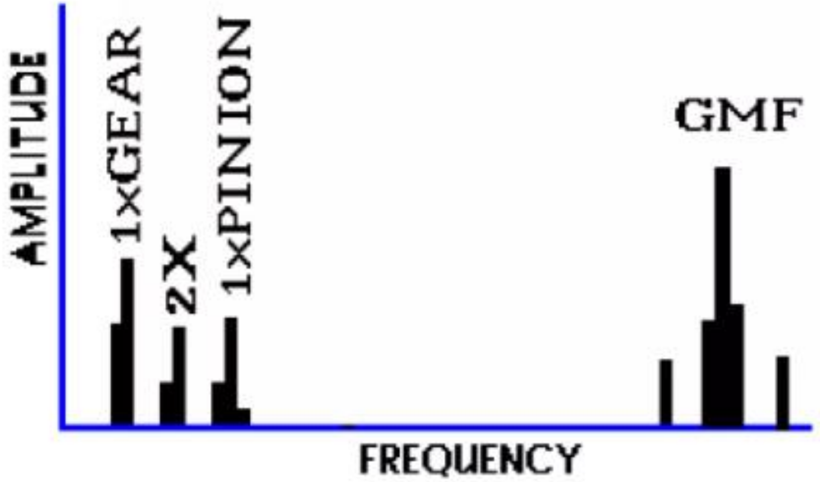
GEARS
NORMAL SPECTRUM





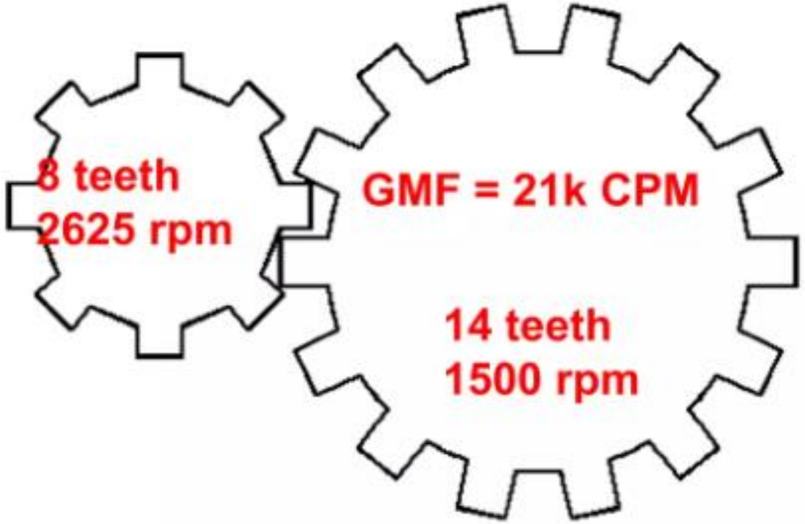
- Normal spectrum shows 1X and 2X and gear mesh frequency GMF
- GMF commonly will have sidebands of running speed
- All peaks are of low amplitude and no natural frequencies are present

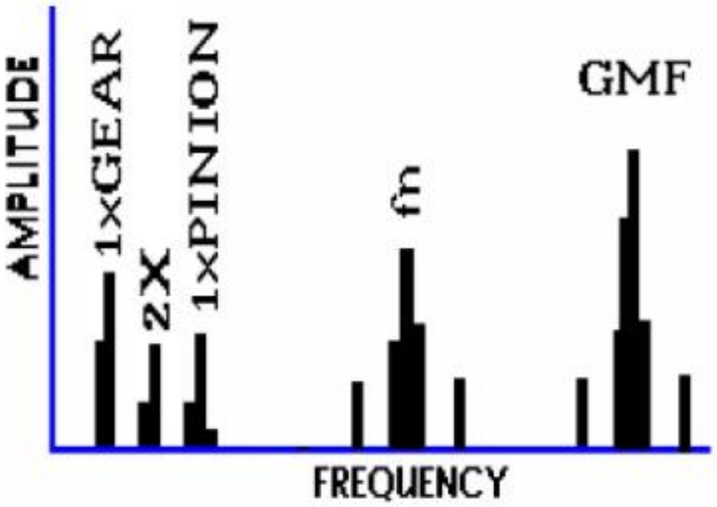
GEARS
TOOTH LOAD



- Gear Mesh Frequencies are often sensitive to load
- High GMF amplitudes do not necessarily indicate a problem
- Each analysis should be performed with the system at maximum load

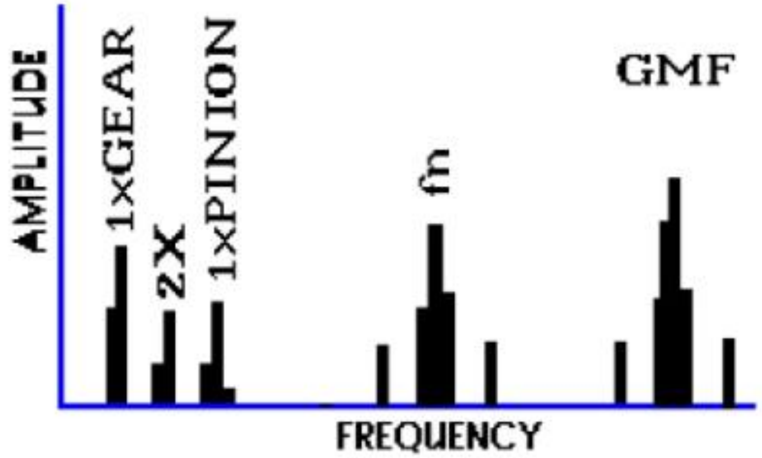
GEARS TOOTH WEAR





- Wear is indicated by excitation of natural frequencies along with sidebands of 1X RPM of the bad gear
- Sidebands are a better wear indicator than the GMF
- GMF may not change in amplitude when wear occurs

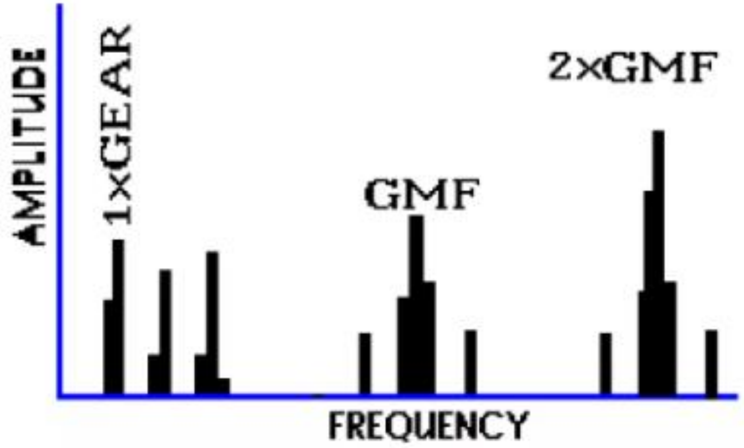
GEARS
GEAR ECCENTRICITY AND BACKLASH



- Fairly high amplitude sidebands around GMF suggest eccentricity, backlash or non parallel shafts
- The problem gear will modulate the sidebands
- Incorrect backlash normally excites gear natural frequency

Machinery Monitoring Systems, LLC

GEARS
GEAR MISALIGNMENT

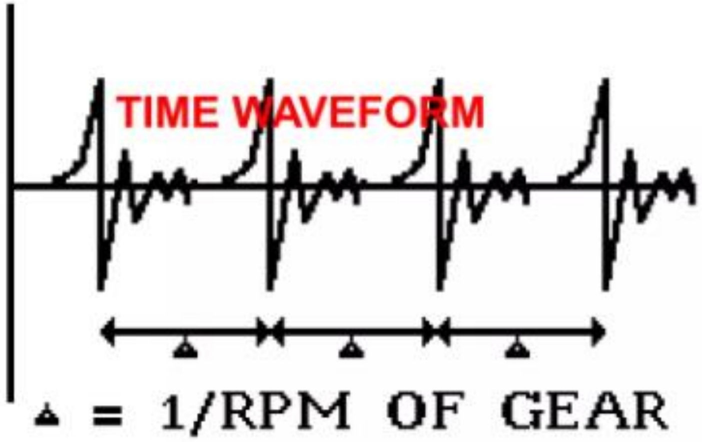


- Gear misalignment almost always excites second order or higher harmonics with sidebands of running speed
- Small amplitude at 1X GMF but higher levels at 2X and 3X GMF
- Important to set Fmax high enough to capture at least 2X GMF

MachineryRx

Machinery Monitoring Systems, LLC

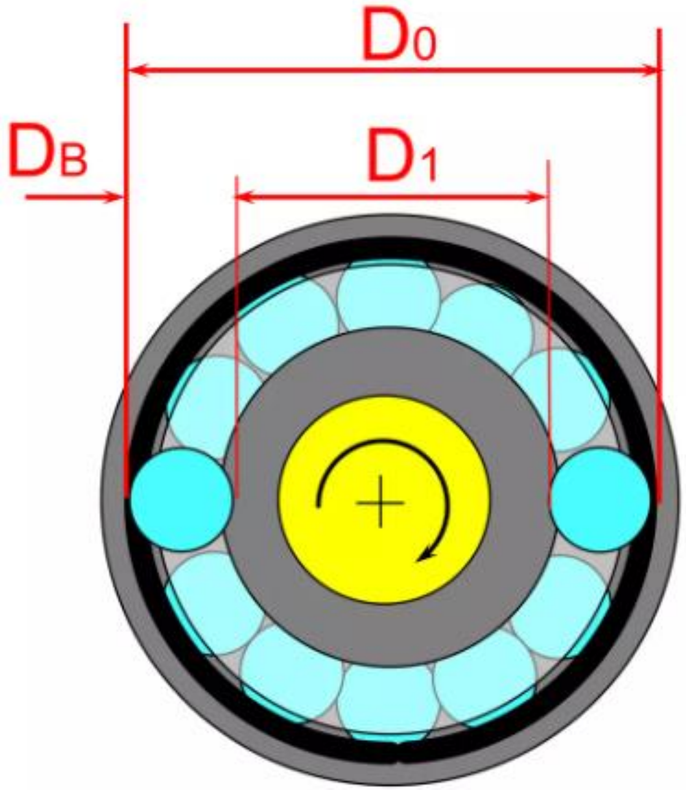
GEARS
CRACKED / BROKEN TOOTH



- **A cracked or broken tooth will generate a high amplitude at 1X RPM of the gear**
- **It will excite the gear natural frequency which will be sidebanded by the running speed fundamental**
- **Best detected using the time waveform**
- **Time interval between impacts will be the reciprocal of the 1X RPM**

MachineryRx

MACHINERY MONITORING SYSTEMS, LLC



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

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

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

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

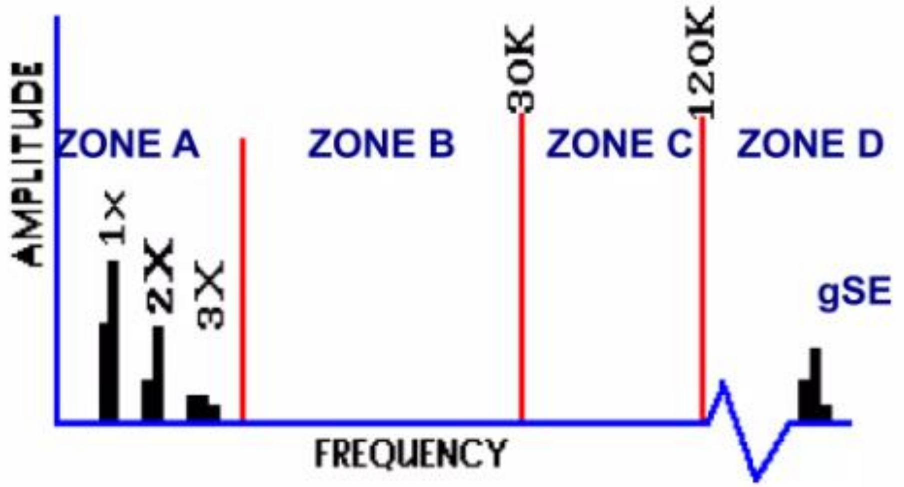
Note : shaft turning
outer race fixed

F = frequency in cpm
N = number of balls

MachineryRx®

MACHINERY MONITORING SYSTEMS, LLC

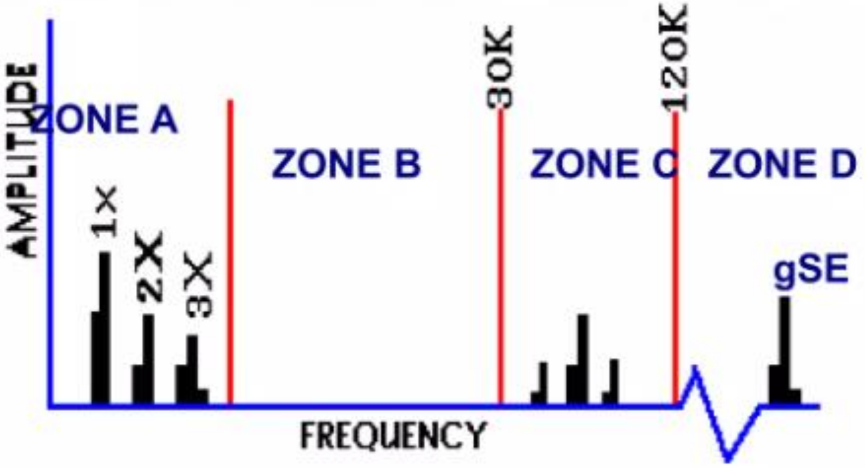
ROLLING ELEMENT BEARINGS STAGE 1 FAILURE MODE



- Earliest indications in the ultrasonic range
- These frequencies evaluated by Spike Energy™ gSE, HFD(g) and Shock Pulse
- Spike Energy may first appear at about 0.25 gSE for this first stage

MachineryRx®

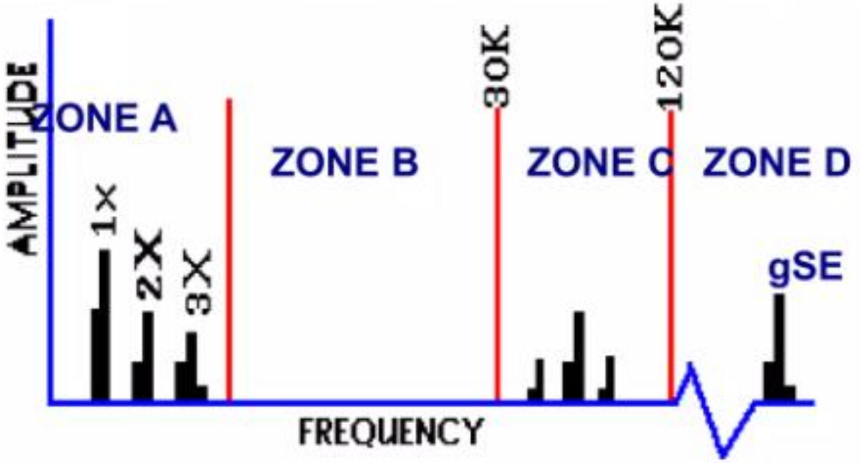
ROLLING ELEMENT BEARINGS
STAGE 2 FAILURE MODE



- Slight defects begin to ring bearing component natural frequencies
- These frequencies occur in the range of 30k-120k CPM
- At the end of Stage 2, sideband frequencies appear above and below natural frequency
- Spike Energy grows e.g. 0.25-0.50gSE

ROLLING ELEMENT BEARINGS
STAGE 2 FAILURE MODE

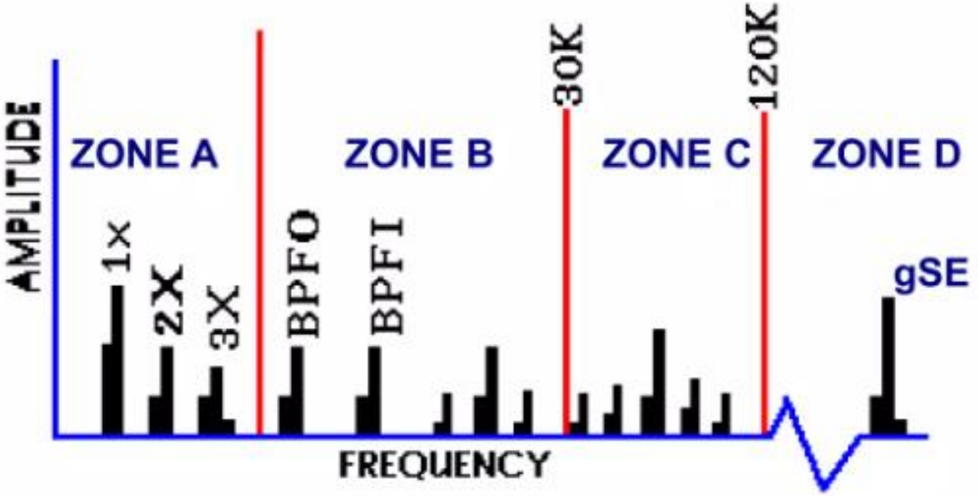
Machinery Monitoring Systems, LLC



- Slight defects begin to ring bearing component natural frequencies
- These frequencies occur in the range of 30k-120k CPM
- At the end of Stage 2, sideband frequencies appear above and below natural frequency
- Spike Energy grows e.g. 0.25-0.50gSE

ROLLING ELEMENT BEARINGS STAGE 3 FAILURE MODE

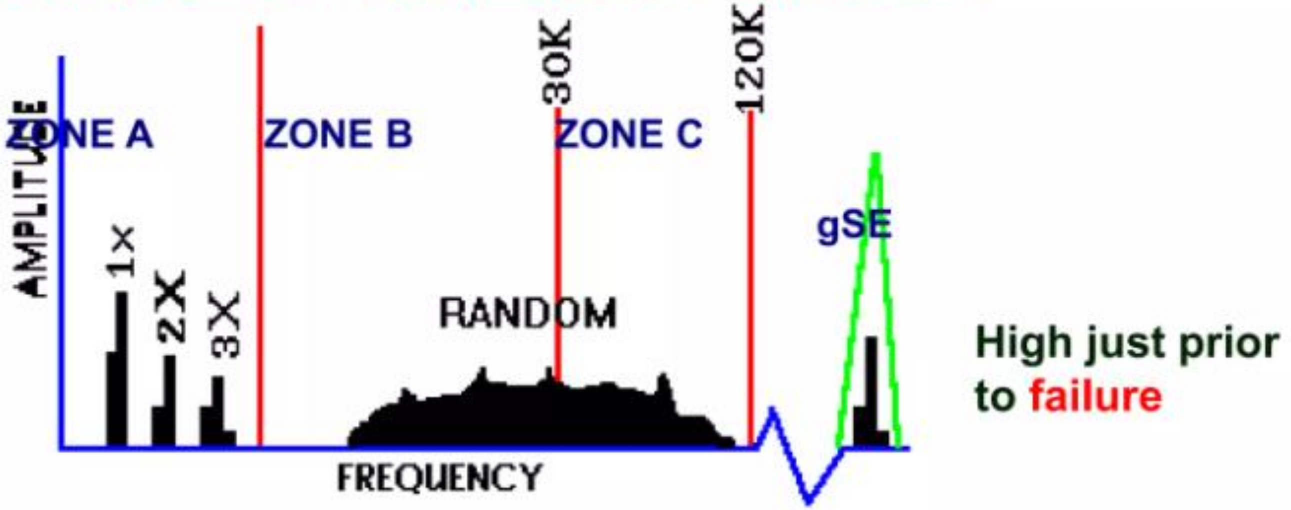
MACHINERY MONITORING SYSTEMS, LLC



- Bearing defect frequencies and harmonics appear
- Many defect frequency harmonics appear with wear the number of sidebands grow
- Wear is now visible and may extend around the periphery of the bearing
- Spike Energy increases to between 0.5 -1.0 gSE

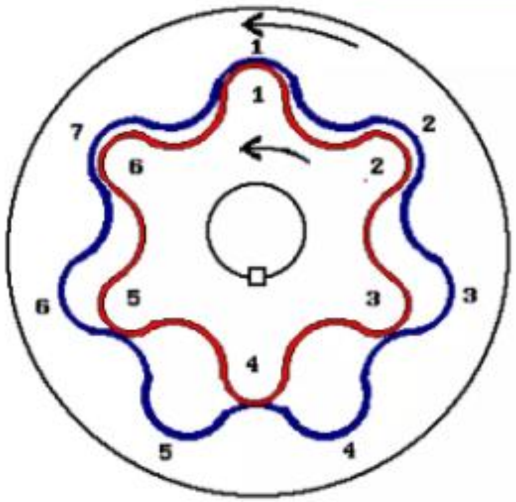
MachineryRx®

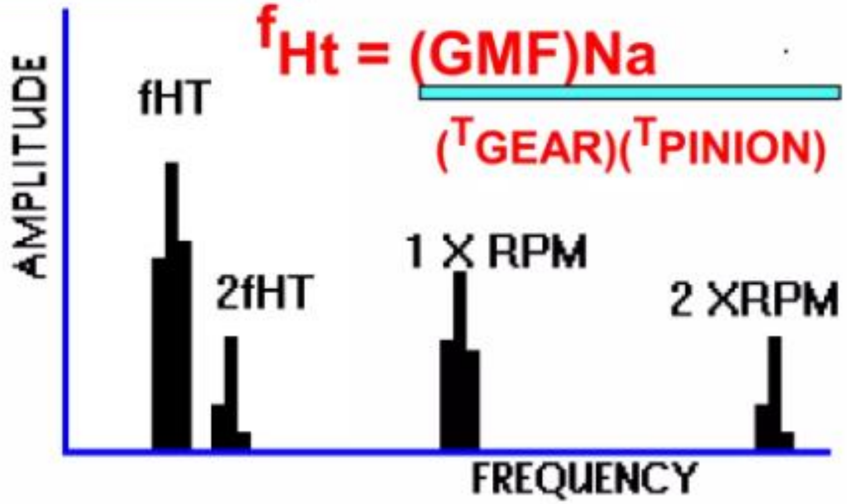
ROLLING ELEMENT BEARINGS STAGE 4 FAILURE MODE



- Discreet bearing defect frequencies disappear and are replaced by random broad band vibration in the form of a noise floor
- Towards the end, even the amplitude at 1 X RPM is effected
- High frequency noise floor amplitudes and Spike Energy may in fact decrease
- Just prior to failure gSE may rise to high levels

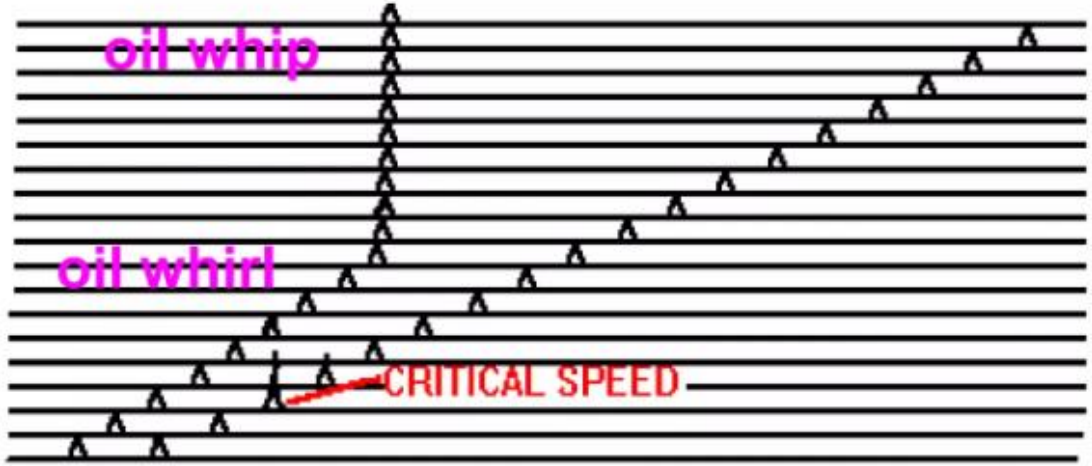
GEARS
HUNTING TOOTH





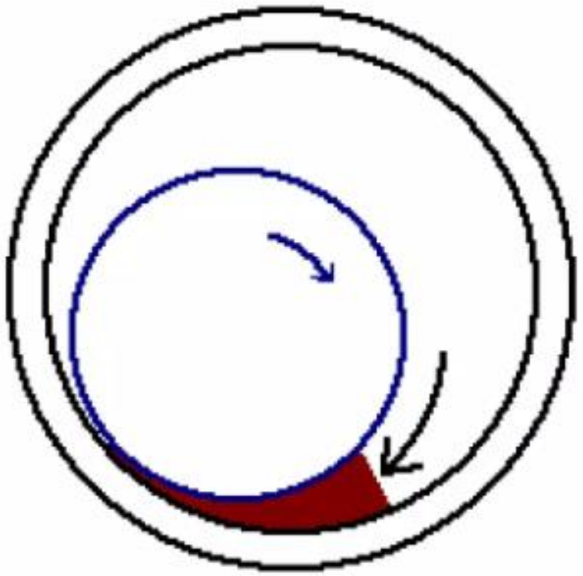
- **Vibration is at low frequency and due to this can often be missed**
- **Synonymous with a growling sound**
- **The effect occurs when the faulty pinion and gear teeth both enter mesh at the same time**
- **Faults may be due to faulty manufacture or mishandling**

OIL WHIP INSTABILITY



- Oil whip may occur if a machine is operated at 2X the rotor critical frequency.
- When the rotor drives up to 2X critical, whirl is close to critical and excessive vibration will stop the oil film from supporting the shaft.
- Whirl speed will lock onto rotor critical. If the speed is increased the whipfrequency will not increase.

OIL WHIRL INSTABILITY



- Usually occurs at 42 - 48 % of running speed
- Vibration amplitudes are sometimes severe
- Whirl is inherently unstable, since it increases centrifugal forces therefore increasing whirl forces

Supplemental & Supporting Material

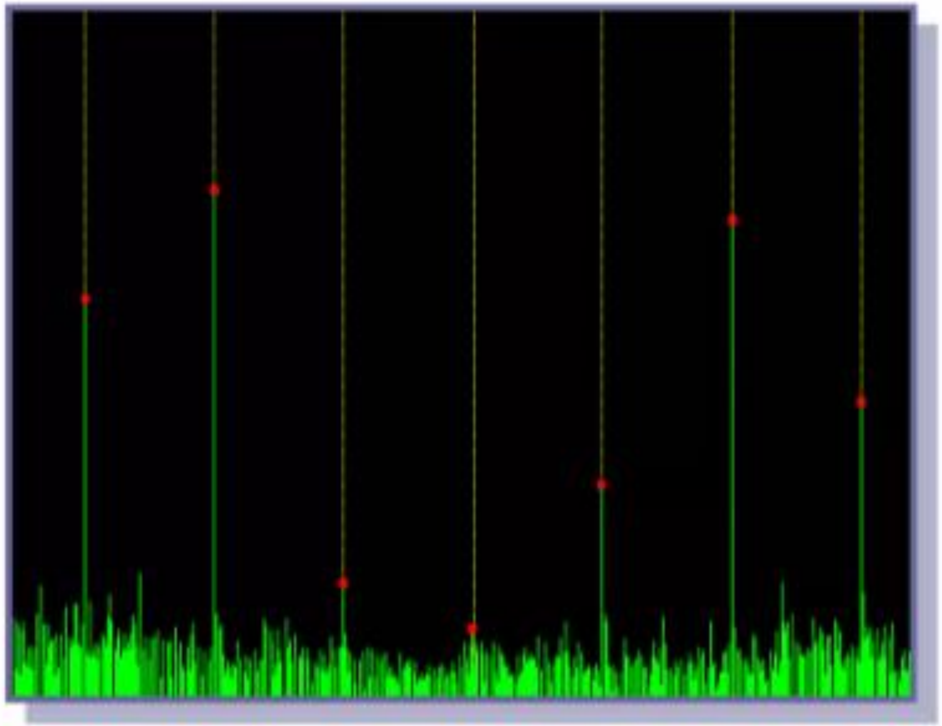
This Slide is Intentionally Blank

FFT or Fast Fourier Transformer

Machinery Monitoring Systems, LLC

Vibration Analysis

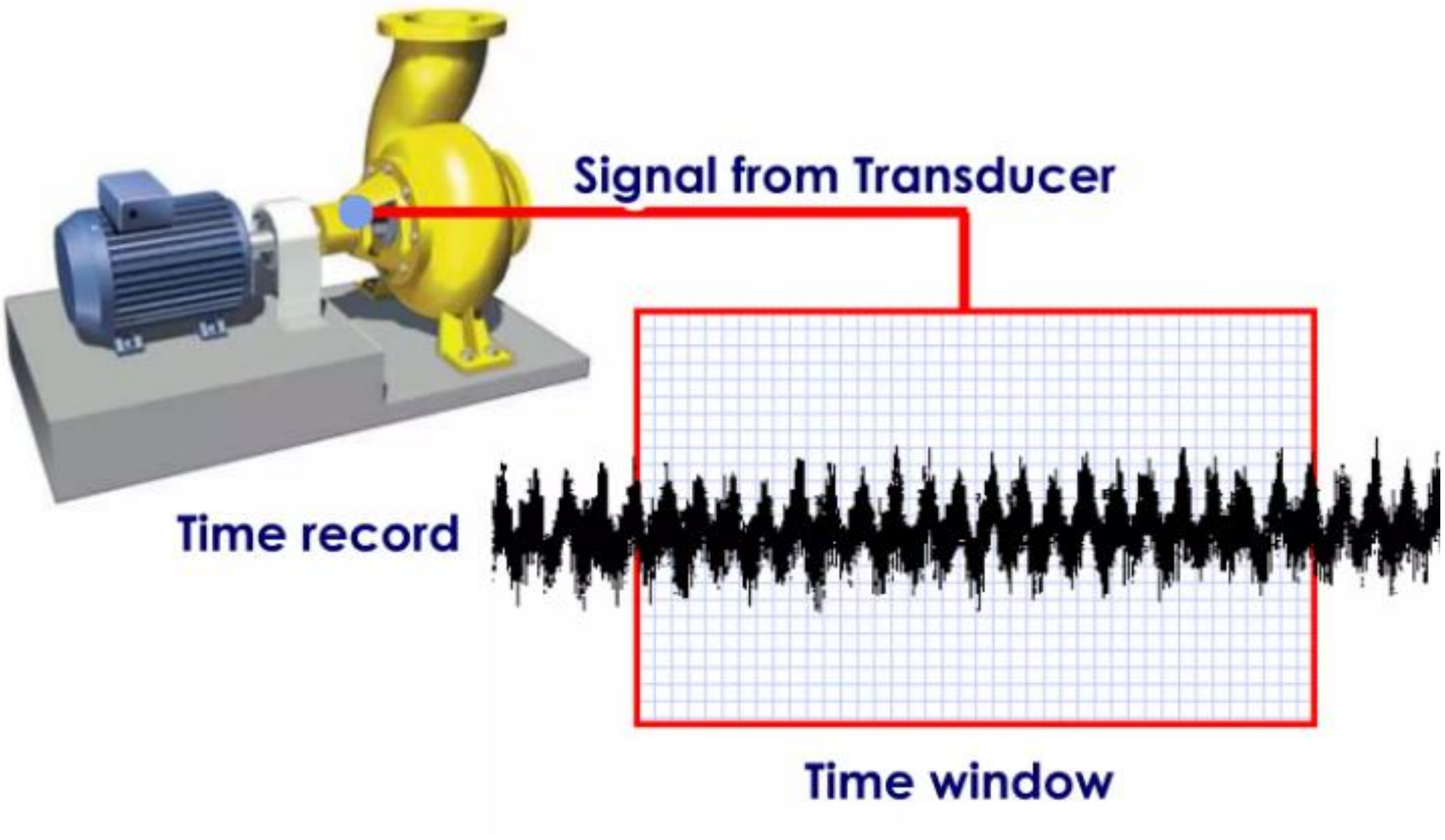
- Uses a FFT to pull data from the time waveform then inputs to a discrete Fourier transform (DFT), a mathematical process or algorithm that transforms a waveform into the components to a frequency spectrum of the input signal from the time waveform.



MachineryRx®

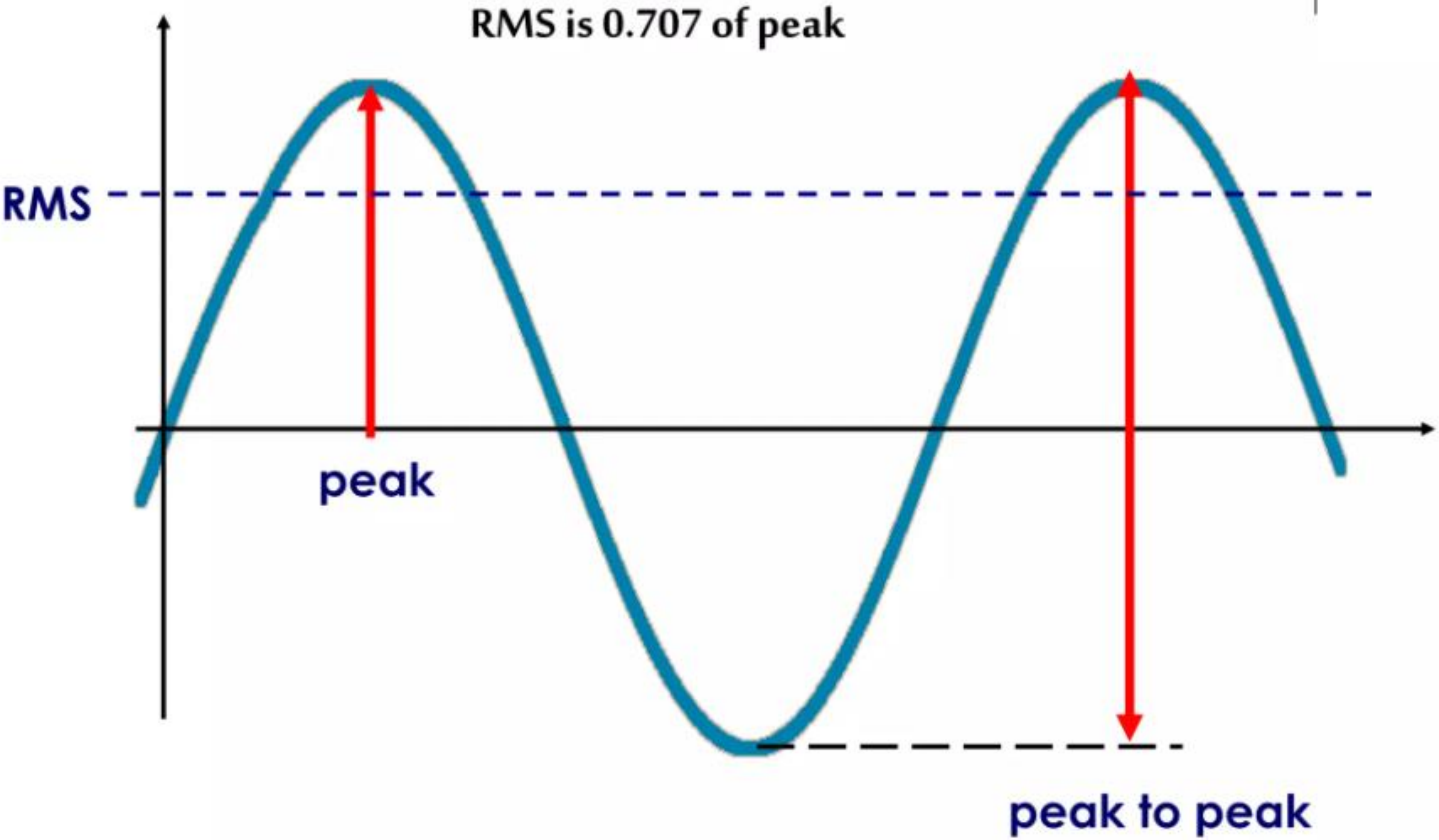
Machine Vibration

Machinery Monitoring Systems, LLC

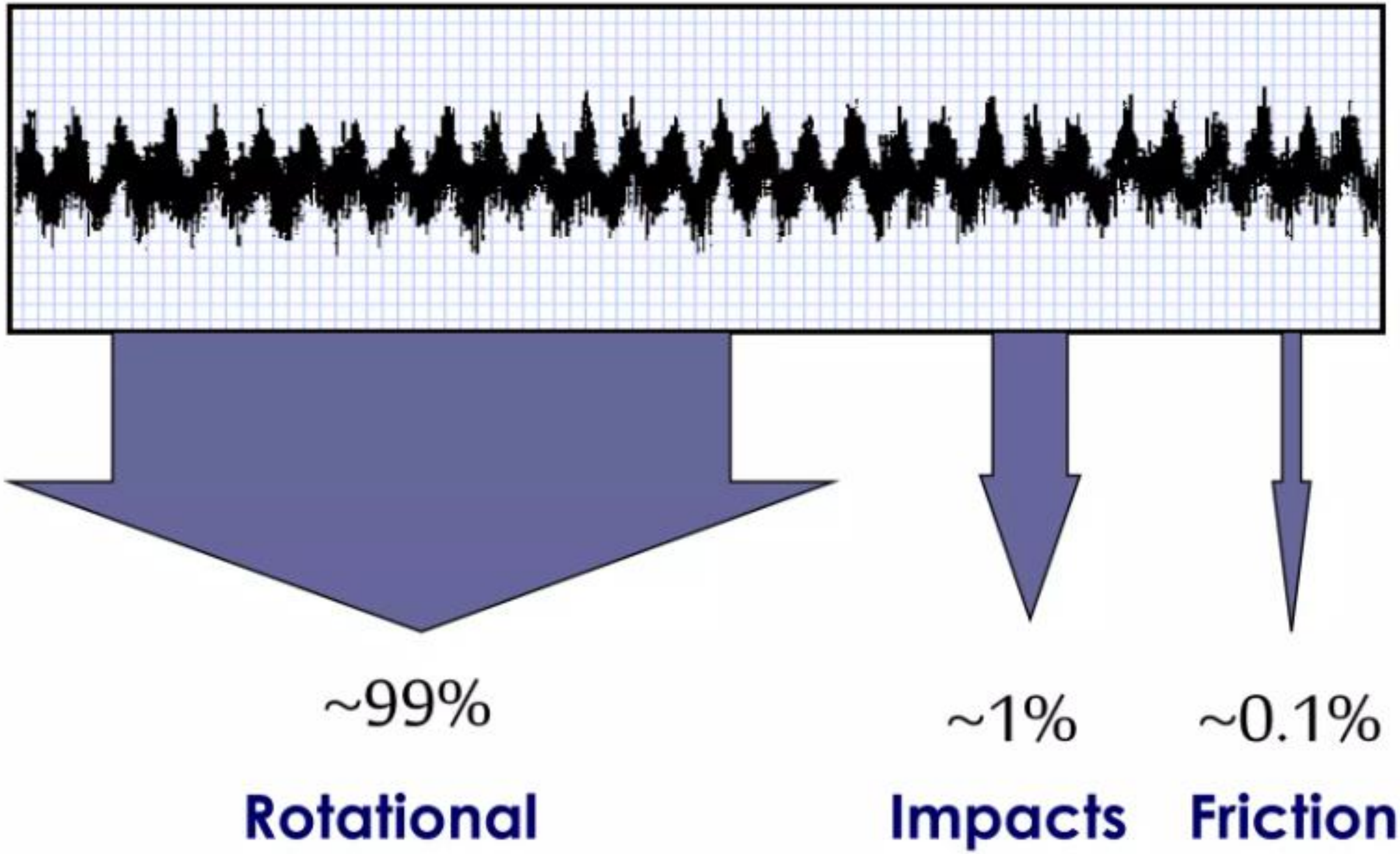


MachineryRx

RMS vs. Peak or Peak to Peak



Forces Causing Vibration



Condition Parameters - Time Domain Calculations

ROTATIONAL Vibration Forces		IMPACTS / SHOCKS Periodic / non-periodic transients			FRICTION Periodic / non-periodic random noise			
ACCELERATION	**VELOCITY**	**CREST**	**KURT**	**SKEW**	**NL1**	**NL2**	**NL3**	**NL4**
ACC Total RMS-value of acceleration	**VEL** Total RMS-value of velocity	Crest factor	Kurtosis	Skewness				

Used to detect and monitor

• Unbalance • Misalignment • Looseness • Wear	• Gear damage • Coupling damage • Other shocks/ impacts	• Scraping parts • Journal bearing

Velocity (VEL)
Rotational Forces

Definition

- RMS amplitude of the vibration velocity over the measured frequency range

Monitors

• Rotational forces	Well
• Impacts/Shocks	Acceptable
• Friction	Unsatisfactory

Comments

- Best used for frequency range < 1,000 Hz

Acceleration (ACC)
Rotational Forces

Definition

- RMS amplitude of the vibration acceleration over the measured frequency range

Monitors

• Rotational forces	Well
• Impacts/Shocks	Acceptable
• Friction	Unsatisfactory

Comments

- Best used for frequency range > 1,000 Hz

Crest
Impacts / Shocks

MACHINERY MONITORING SYSTEMS, LLC

Definition

- The ratio (difference) between vibration peak amplitude and RMS amplitude

Monitors

• Rotational forces	Unsatisfactory
• Impacts/Shocks	Well
• Friction	Unsatisfactory

Comments

- A sine wave has CREST = 1.414
- Normal values is 1.4 - 3.0

MachineryRx®

Kurtosis (KURT)
Impacts / Shocks

Definition

- 4th statistical moment of the vibration signal
- Describes how a signal is grouped around its mean value

Monitors

• Rotational forces	Unsatisfactory
• Impacts/Shocks	Well
• Friction	Unsatisfactory

Comments

- A sine wave has KURT = -1.5
- A signal with transients has KURT > 0.

Skewness (SKEW)
Impacts / Shocks

Definition

- 3rd statistical moment of the vibration signal
- Describes the symmetry of the signal around its mean

Monitors

Rotational forces	Unsatisfactory
Impacts/Shocks	Well
Friction	Unsatisfactory

Comments

- A sine wave has SKEW = 0
- Positive transients give SKEW > 0
- Negative transients give SKEW < 0.

Noise Level (NL1-4)
Friction

Definition

- Random noise in one quarter of the measured frequency range

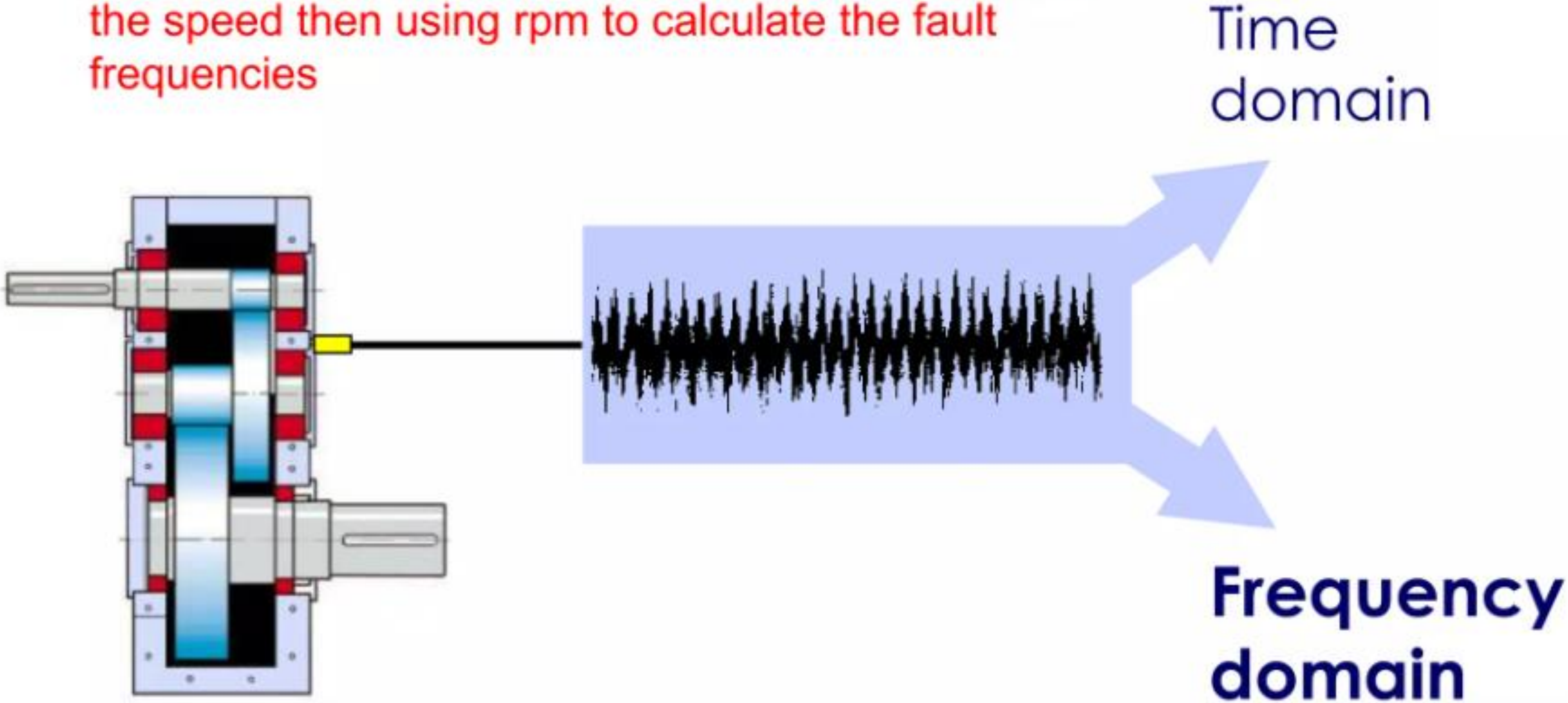
Monitors

• Rotational forces	Unsatisfactory
• Impacts/Shocks	Unsatisfactory
• Friction	Well

Comments

- Describes the broad band noise floor in the spectrum

Extremely important to verify the speed with strobe or laser due to vibration Analysis is based on knowing the speed then using rpm to calculate the fault frequencies



Machinery Monitoring Systems, LLC

Fault Frequencies

Used for fault recognition:

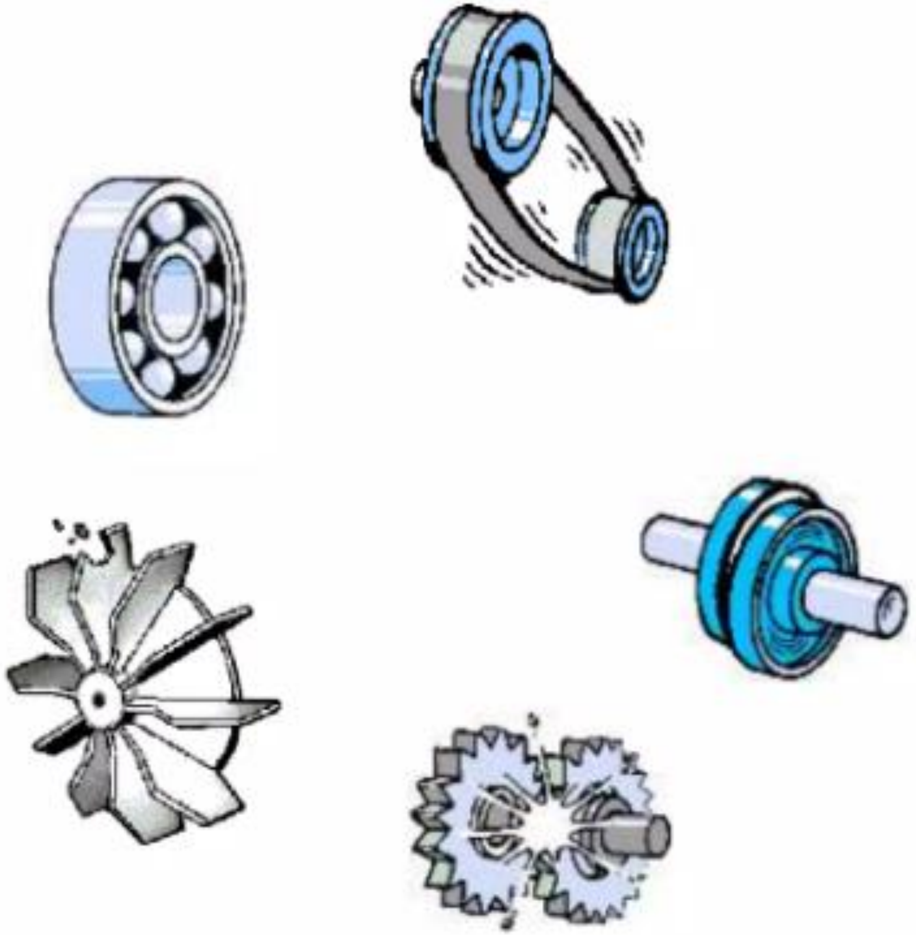
- Moving parts in a rotating machine contribute to the vibration at specific frequencies
- Most symptoms are shown under normal conditions (like gear mesh) but increase with damage
- Some symptoms appear only when the component is damaged (like rotor bar fault)

MachineryRx®

Fault or Forcing Frequencies

Examples

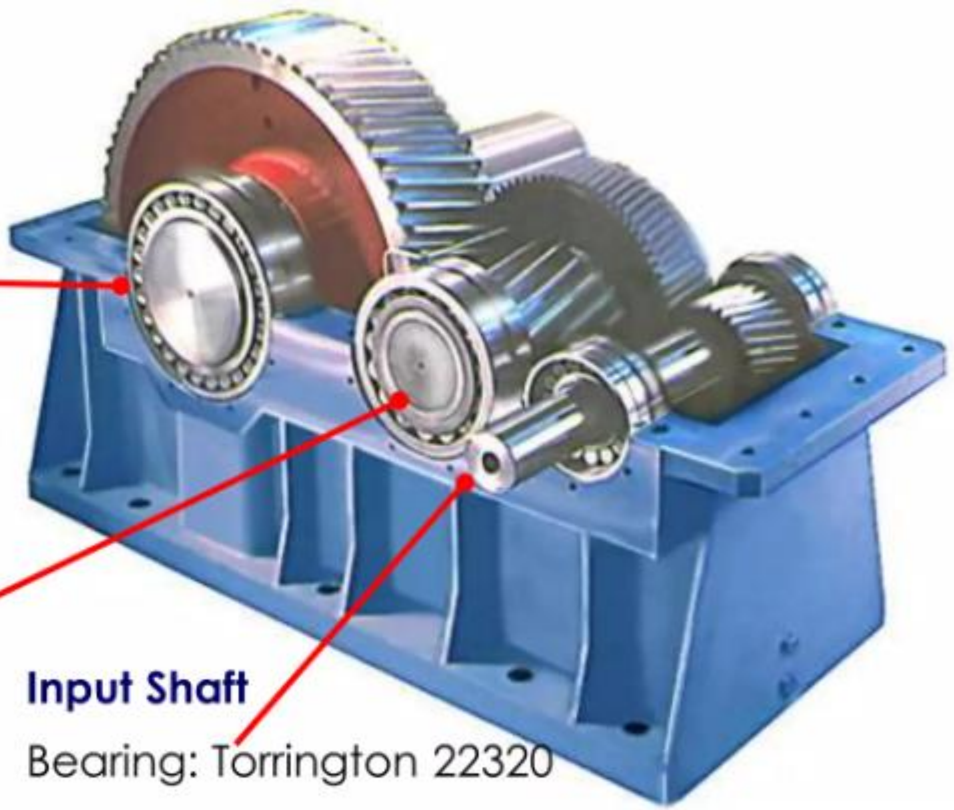
- Unbalance, misalignment, looseness, resonance, gear damage, belt / chain drive faults, fan / blade faults, oil whirl, bearing faults, rotor bars, etc.
- Or, define your own faults if you know what your looking for as to certain Characteristics.



MACHINERY MONITORING SYSTEMS, LLC

Fault Frequency Selection

Example of frequency domain inputs



Output Shaft

Bearing: Torrington 110TDO456

(BPFO, BPFI, FTF, BSF)

63 Gear Teeth * RPM

Intermediate Shaft

Bearing: Torrington 22332

(BPFO, BPFI, FTF, BSF)

13/43 Gear Teeth *RPM

Input Shaft

Bearing: Torrington 22320

(BPFO, BPFI, FTF, BSF)

23 Gear Teeth * RPM

MachineryRx®

Fault Frequency Calculations

ROLLING ELEMENT BEARING FAULTS

Direction: RADIAL, AXIAL

Amplitude: Varied

Frequency: Non-Synchronous w.r.t. 1x RPM

Fundamental Train (Cage) Frequency:

$FTF = (1/2) \times [1 - (B_D/P_D)\cos(\theta)]$ (Inner Race Rotating)

$FTF = (1/2) \times [1 + (B_D/P_D)\cos(\theta)]$ (Outer Race Rotating)

Ball Spin Frequency:

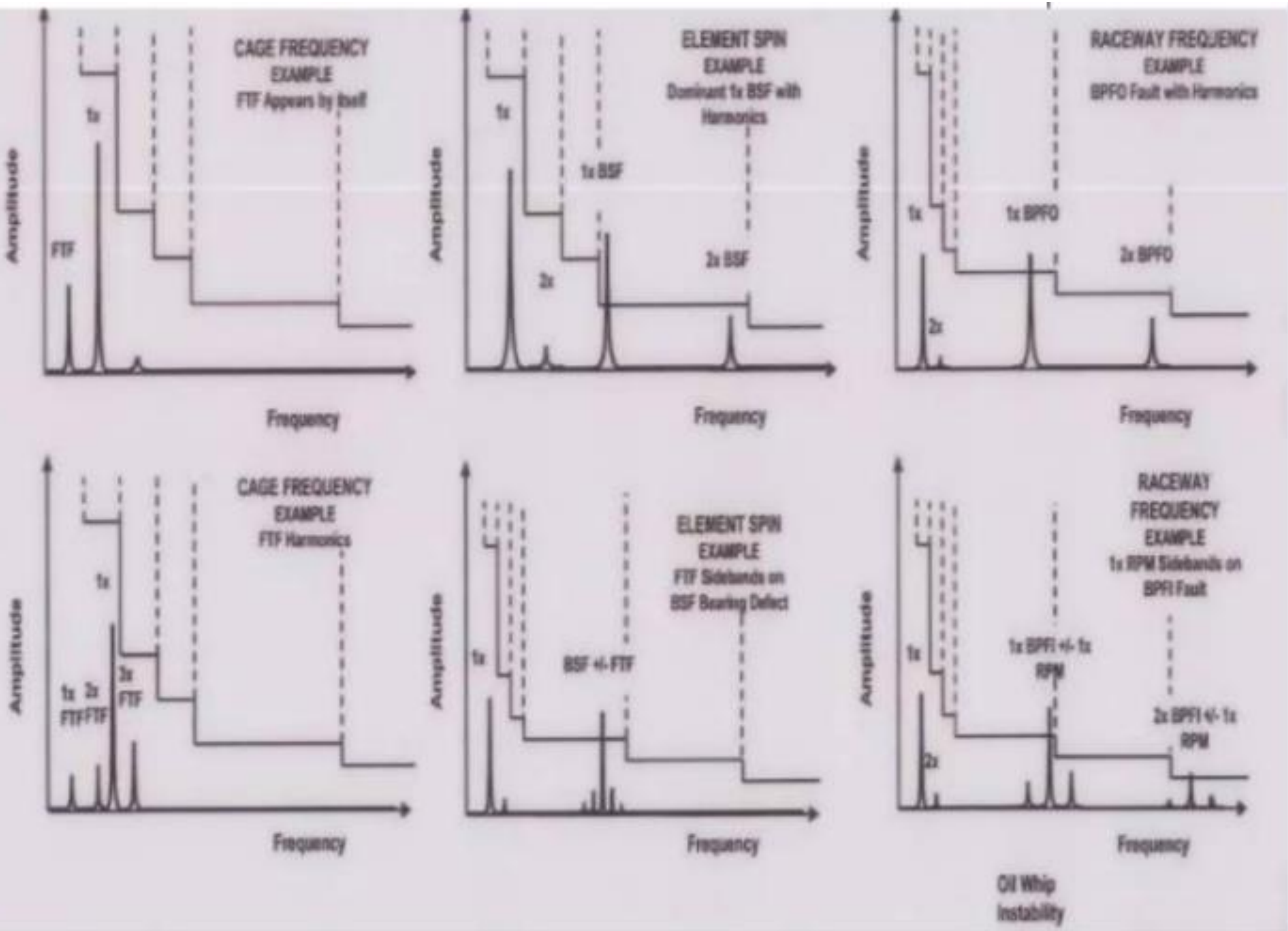
$BSF = [P_D/2B_D] \times [1 - (B_D/P_D)^2 \times (\cos(\theta))^2]$

Ball Pass Frequency Outer / Inner

$BPFO = (N_B/2) \times [1 - (B_D/P_D)\cos(\theta)]$

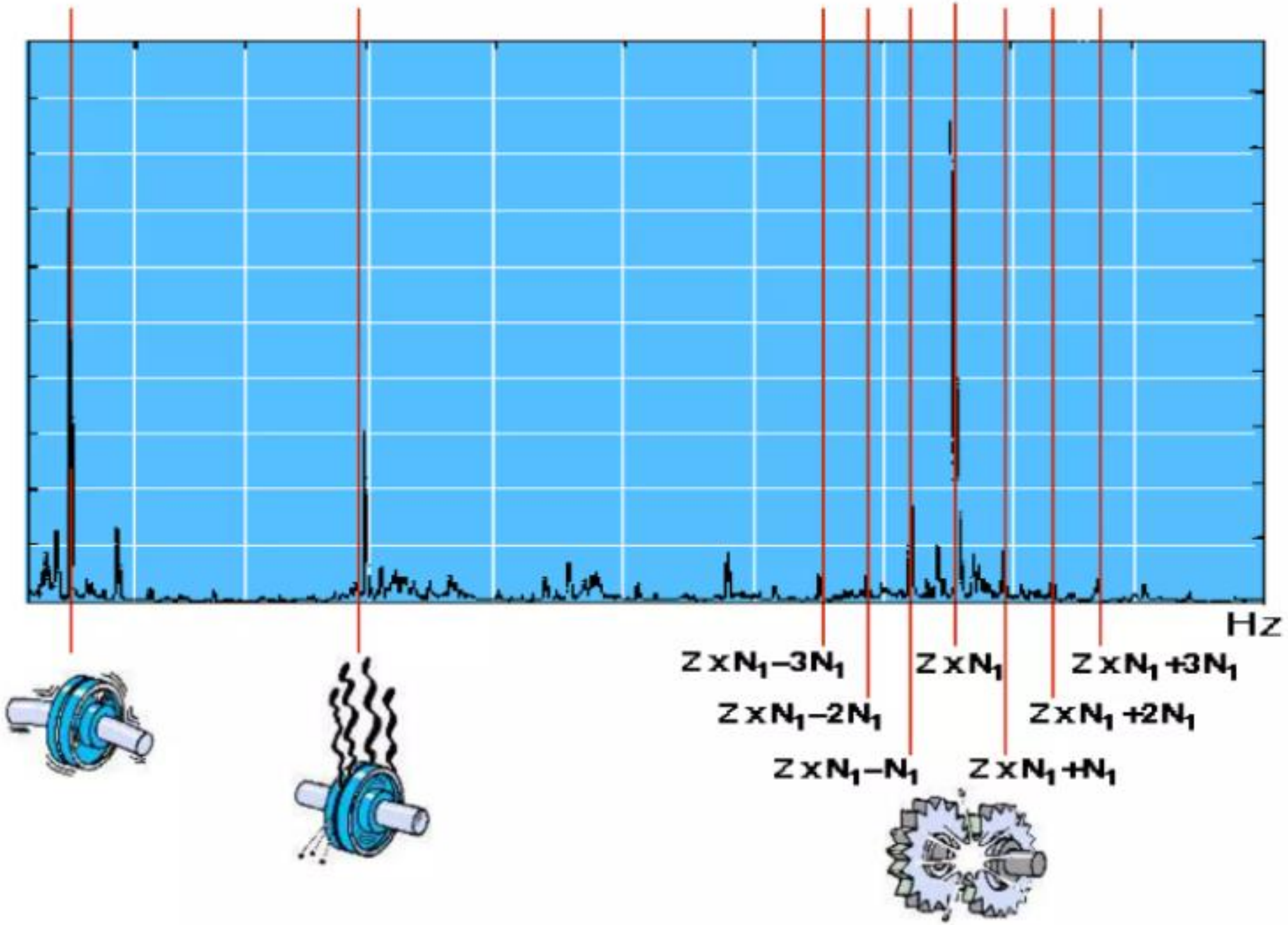
$BPFI = (N_B/2) \times [1 + (B_D/P_D)\cos(\theta)]$

Time Waveform: Possible Amplitude Modulation



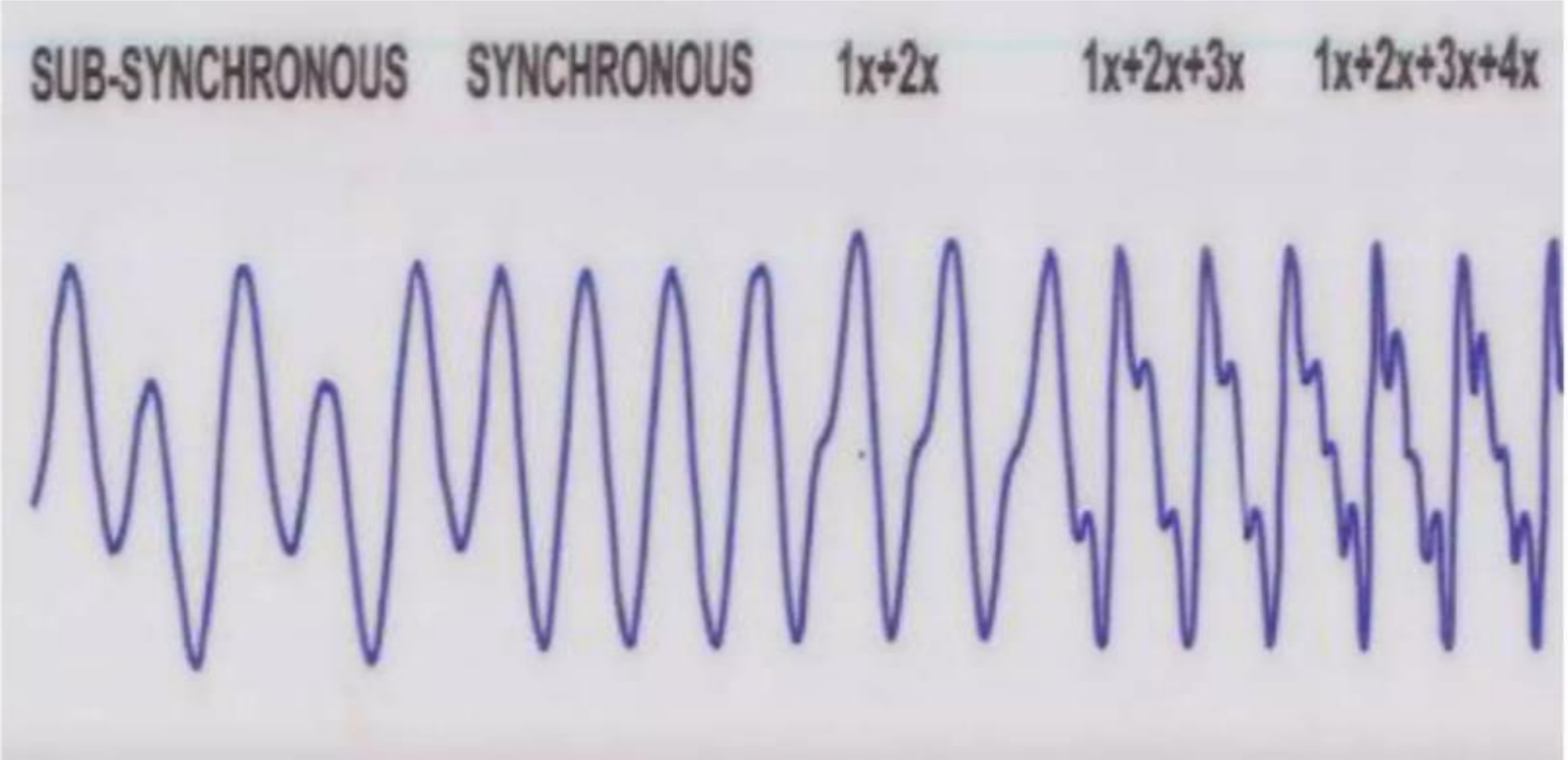
OIL WHIRL – OIL WHIP INSTABILITY

Fault Frequencies Matches



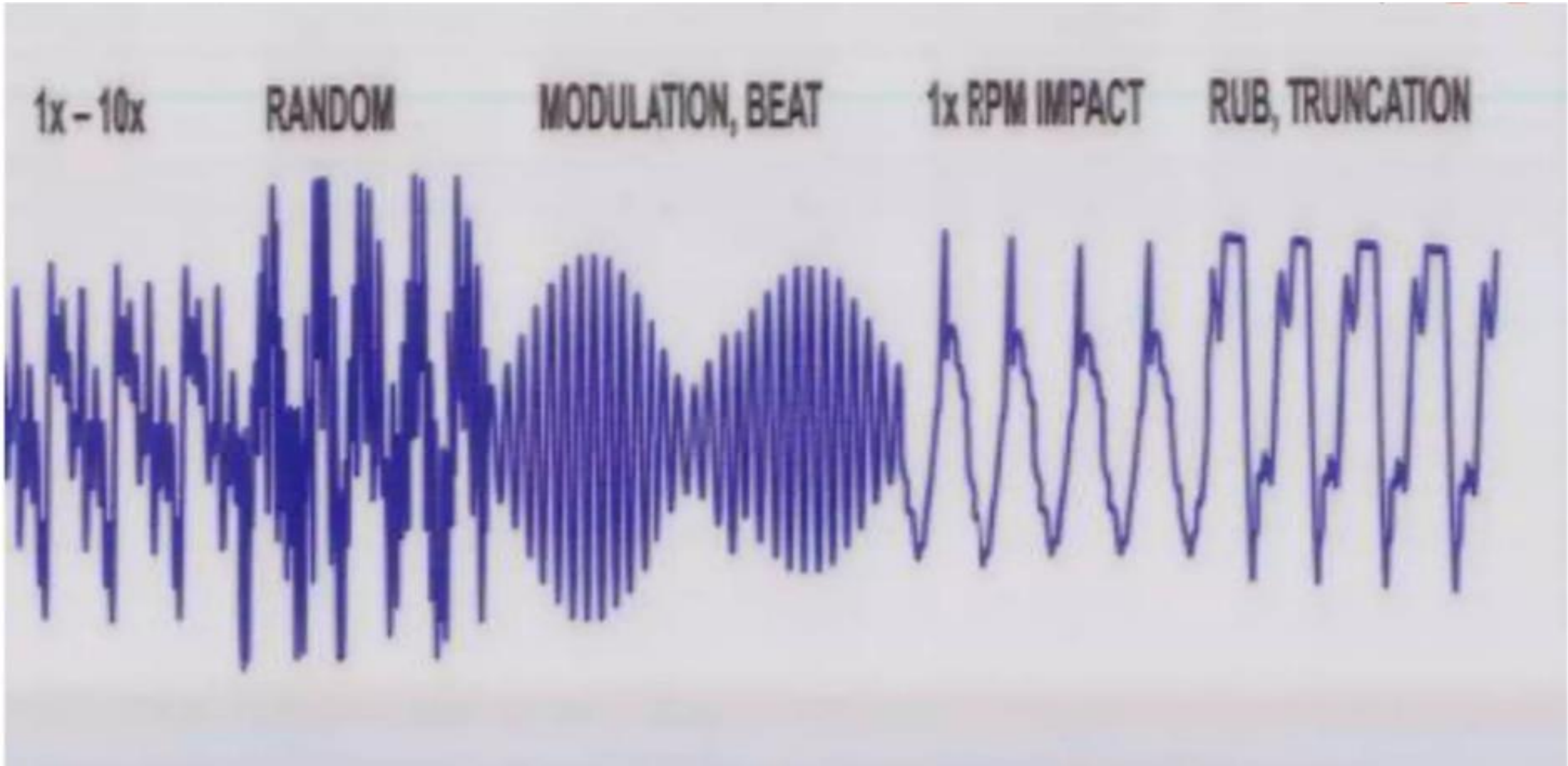
Time Waveform Analysis

Machinery Monitoring Systems, LLC

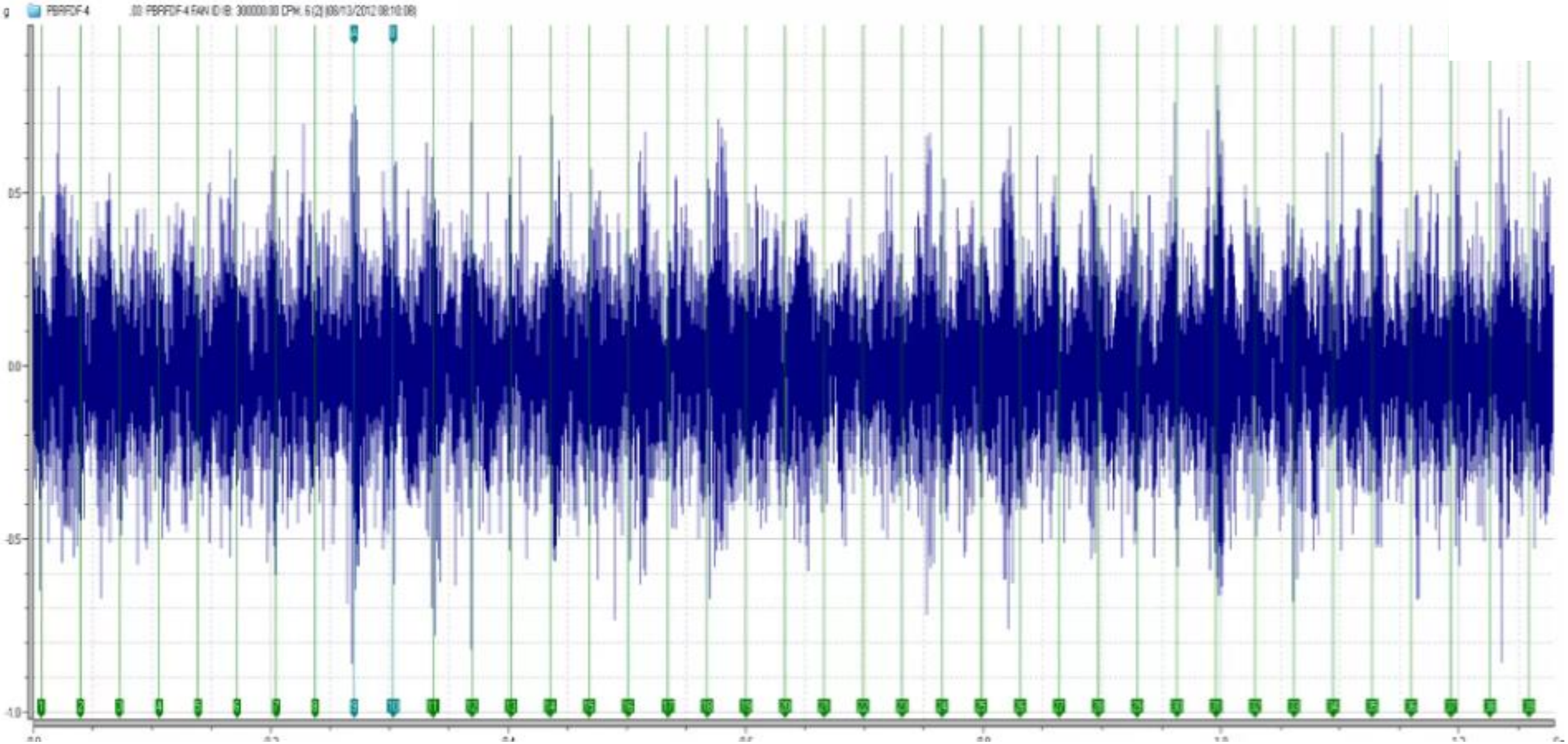


MachineryRx®

Time Waveform Analysis



Time Waveform Analysis

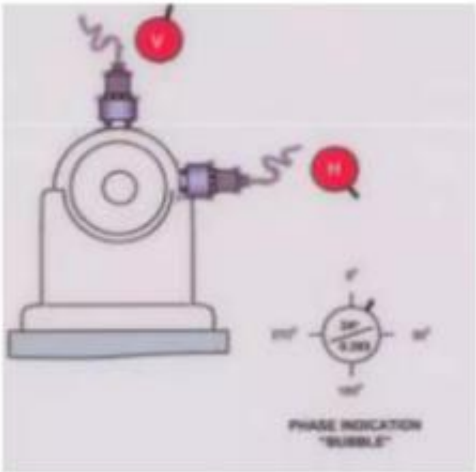


Machinery Monitoring Systems, LLC

MachineryRx®

Machinery Monitoring Systems, LLC

Phase readings



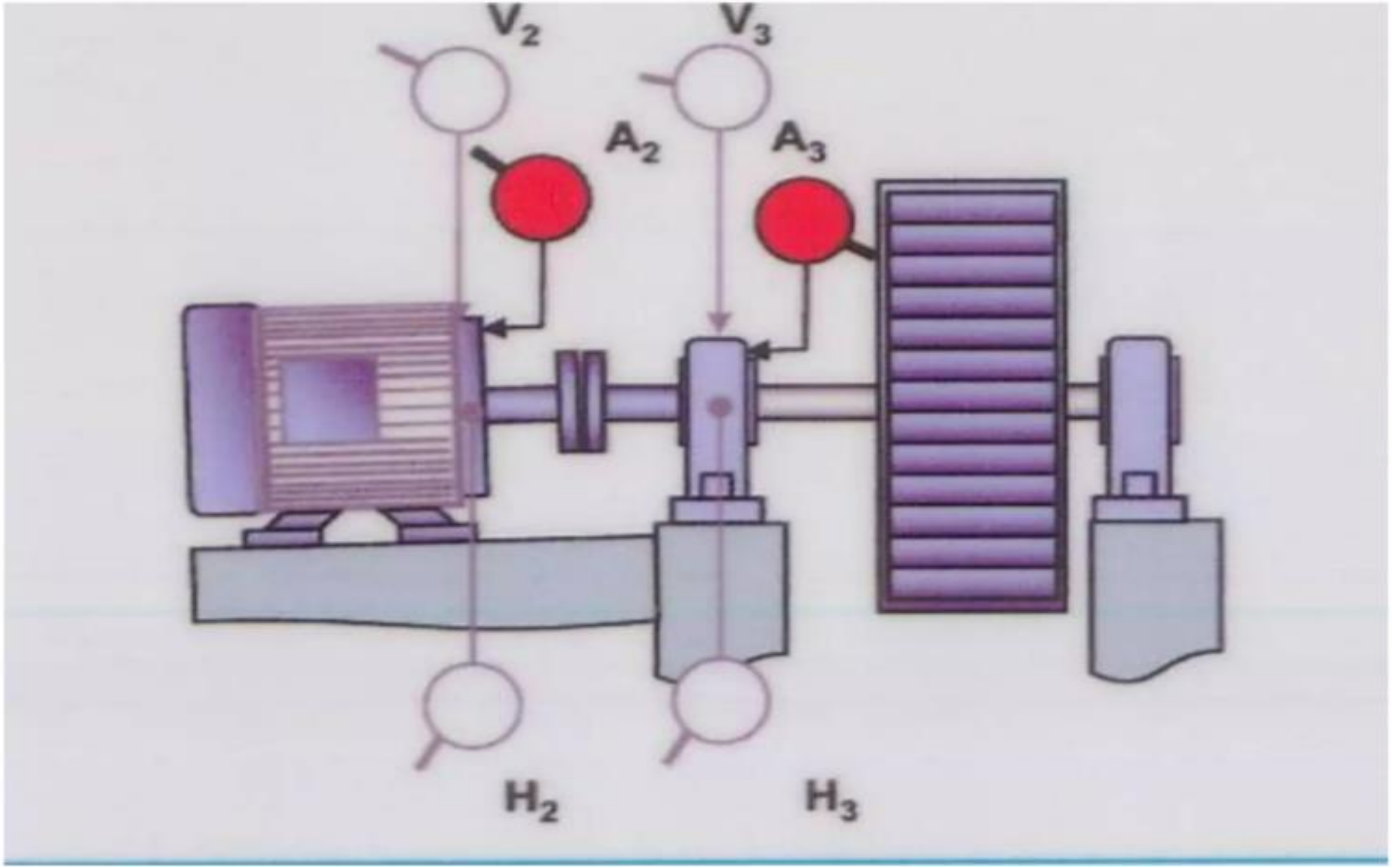
Used for fault recognition:

- Moving parts in a rotating machine contribute to the vibration which can be verified by taking phase readings.
- Unbalance will show phase readings with a 90 degree phase shift from the vertical reading to the horizontal reading.
- Misalignment will show a phase shift of 180 degrees across the shaft in the axial direction between machine components.

MachineryRx®

Misalignment has two types, Angular & Parallel Offset. Below is Angular

MACHINERY MONITORING SYSTEMS, LLC



MachineryRx®

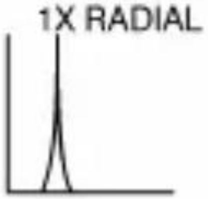
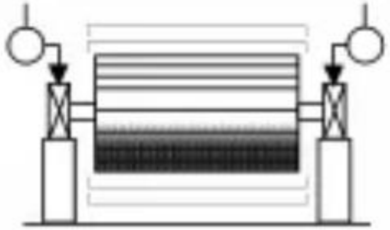
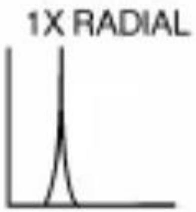
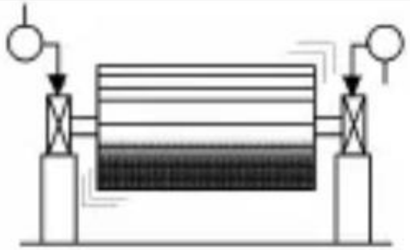
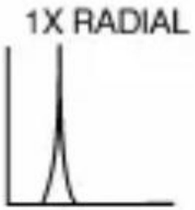
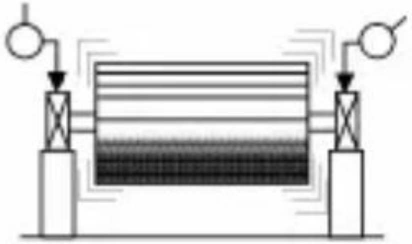
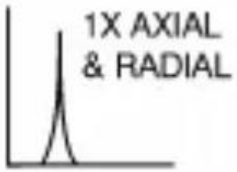
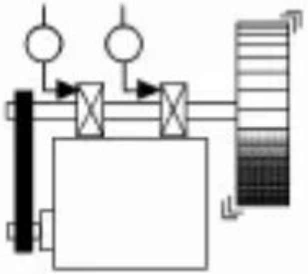
Machinery Monitoring Systems, LLC

Vibration Analysis is extremely helpful to detect machine condition and when conducting readings make sure to take vertical, horizontal and axial readings.

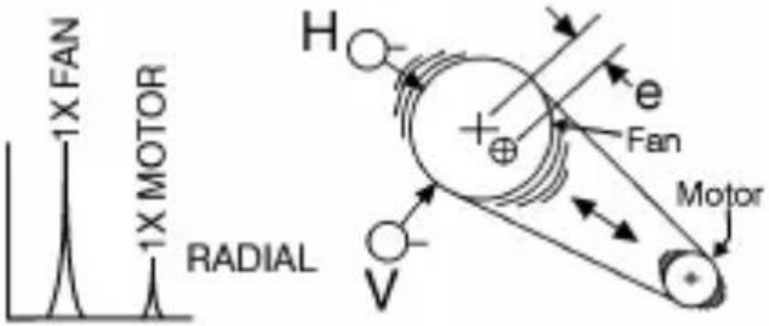
Readings should never be taken on guards or false housing as well again verify rpm. When conducting readings on roller bearings ensure the axial reading is in the load zone.

MachineryRx®

Appendix

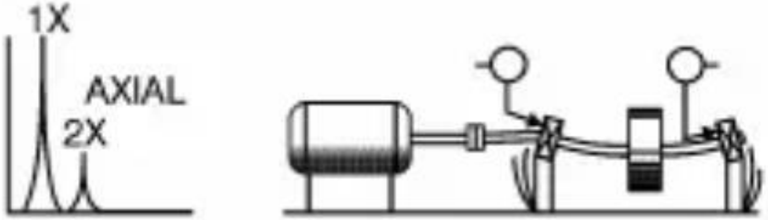
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS
MASS UNBALANCE **A. FORCE UNBALANCE**			Force Unbalance will be in-phase and steady. Amplitude due to unbalance will increase by the square of speed below first rotor critical (a 3X speed increase = 9X higher vibration). 1X RPM always present 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).

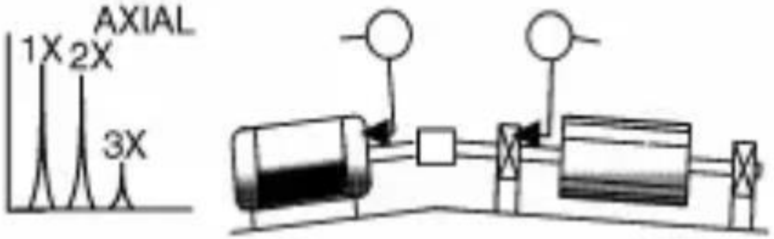
BENT SHAFT



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.

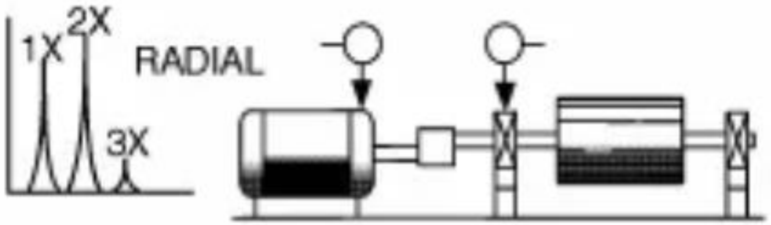
MISALIGNMENT

A. ANGULAR MISALIGNMENT



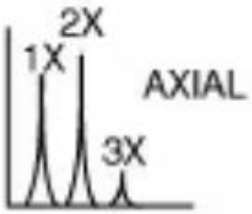
Angular Misalignment is characterized by high axial vibration, 180° out-of-phase across the coupling. Typically will have high axial vibration with both 1X and 2X RPM. However, not unusual for either 1X, 2X or 3X to dominate. These symptoms may also indicate coupling problems as well. Severe angular 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.

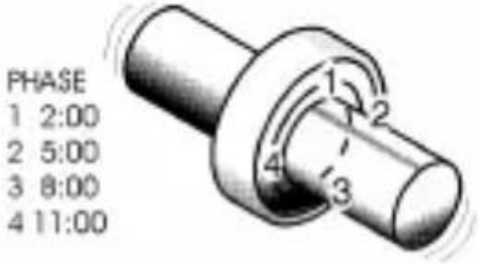
B. PARALLEL MISALIGNMENT



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.

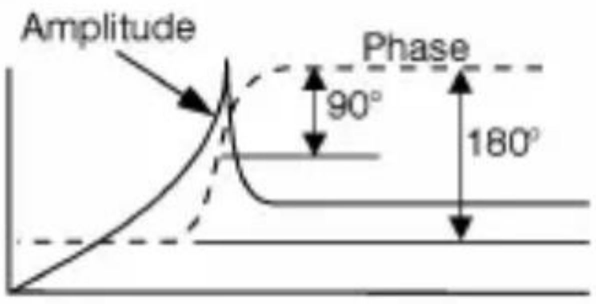
C. MISALIGNED BEARING COCKED ON SHAFT

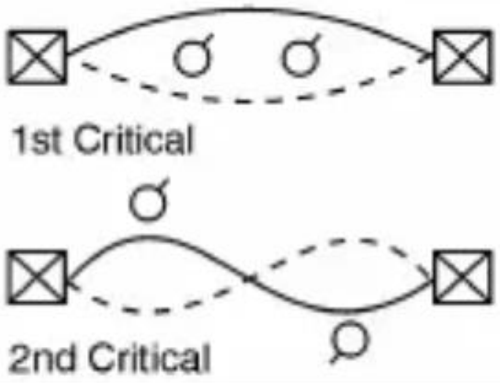




Cocked Bearing will generate considerable axial vibration. Will cause Twisting Motion with approximately 180° phase shift top to bottom and/or side to side as measured in axial direction on same bearing housing. Attempts to align coupling or balance the rotor will not alleviate problem. Bearing usually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

RESONANCE

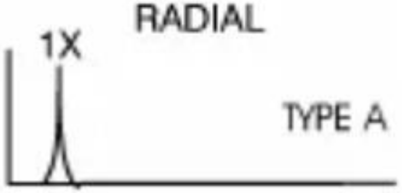


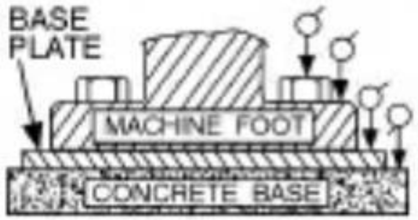


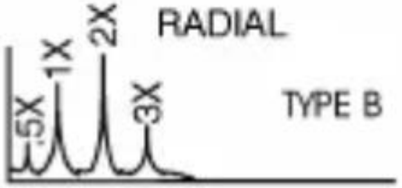
Resonance occurs when a Forcing Frequency coincides with a System Natural Frequency, and can cause dramatic amplitude amplification, which might result in premature, or even catastrophic failure. This may be a natural frequency of the rotor, but can often originate from support frame, foundation, gearbox 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).

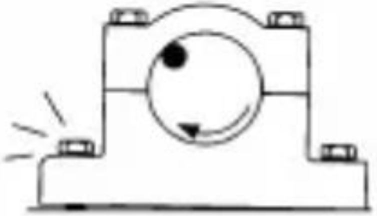
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS

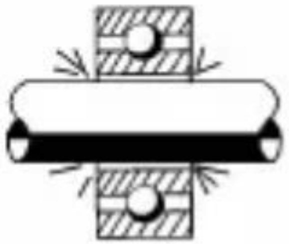
MECHANICAL LOOSENESS







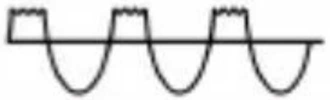


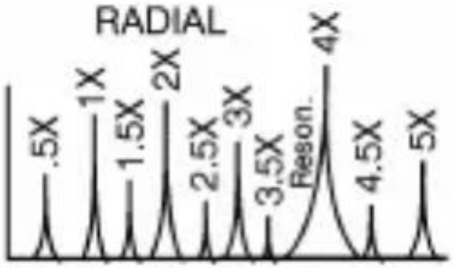


Mechanical Looseness is indicated by either Type A, B or C vibration spectra. ***Type A*** is caused by Structural looseness/weakness of machine feet, baseplate or foundation; also by deteriorated grouting, loose hold-down bolts at the base; and distortion of the frame or base (i.e., soft foot). Phase 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

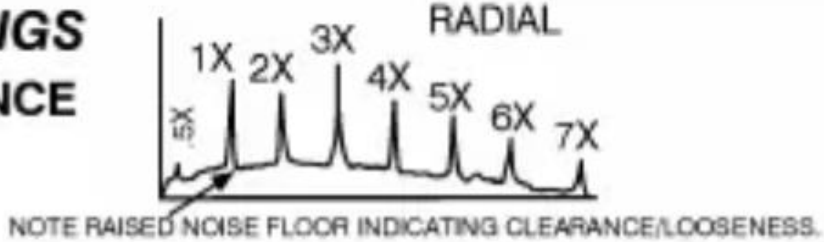




Rotor Rub produces similar spectra to Mechanical Looseness when rotating parts contact stationary components. Rub may be either partial or throughout the entire shaft revolution. Usually generates a series of frequencies, often exciting one or more resonances. Often excites integer fraction subharmonics 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).

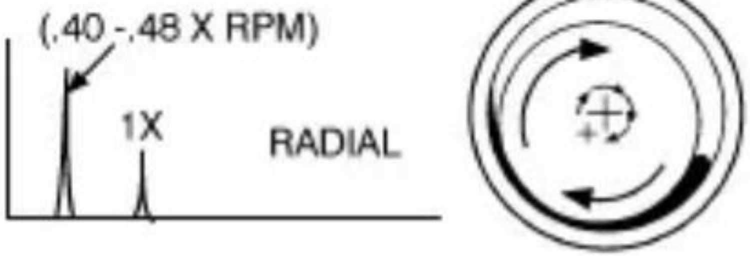
JOURNAL BEARINGS

A. WEAR/CLEARANCE PROBLEMS



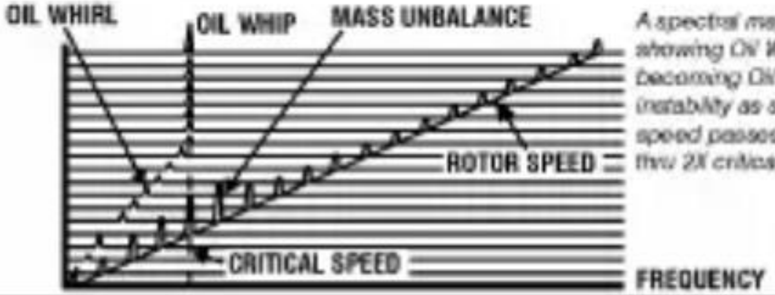
Latter stages of journal bearing wear are normally evidenced by presence of whole series of running speed harmonics (up to 10 or 20). Wiped journal bearings often will allow high vertical amplitudes compared to horizontal, but, may show only one pronounced peak at 1X RPM. Journal bearings with excessive 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



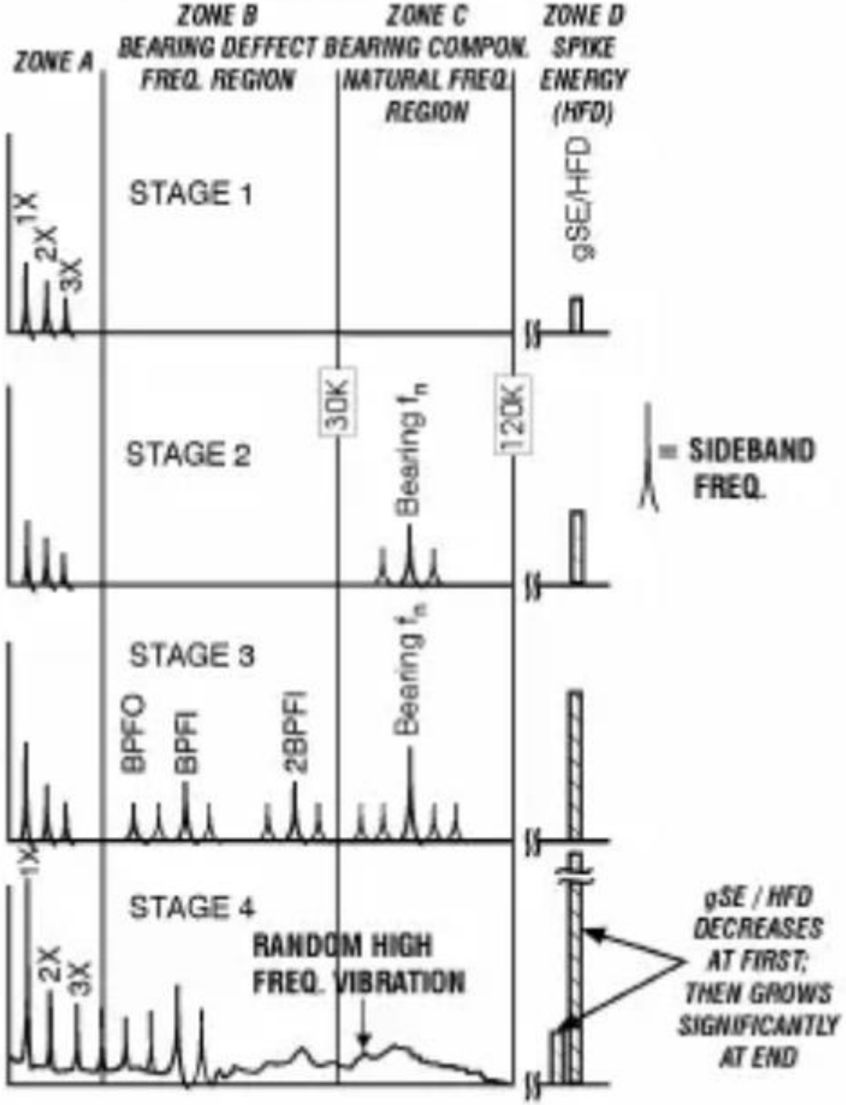
A spectral map showing Oil Whirl becoming Oil Whip instability as shaft speed passes thru 2X critical.

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

DOMINANT FAILURE SCENARIO



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.

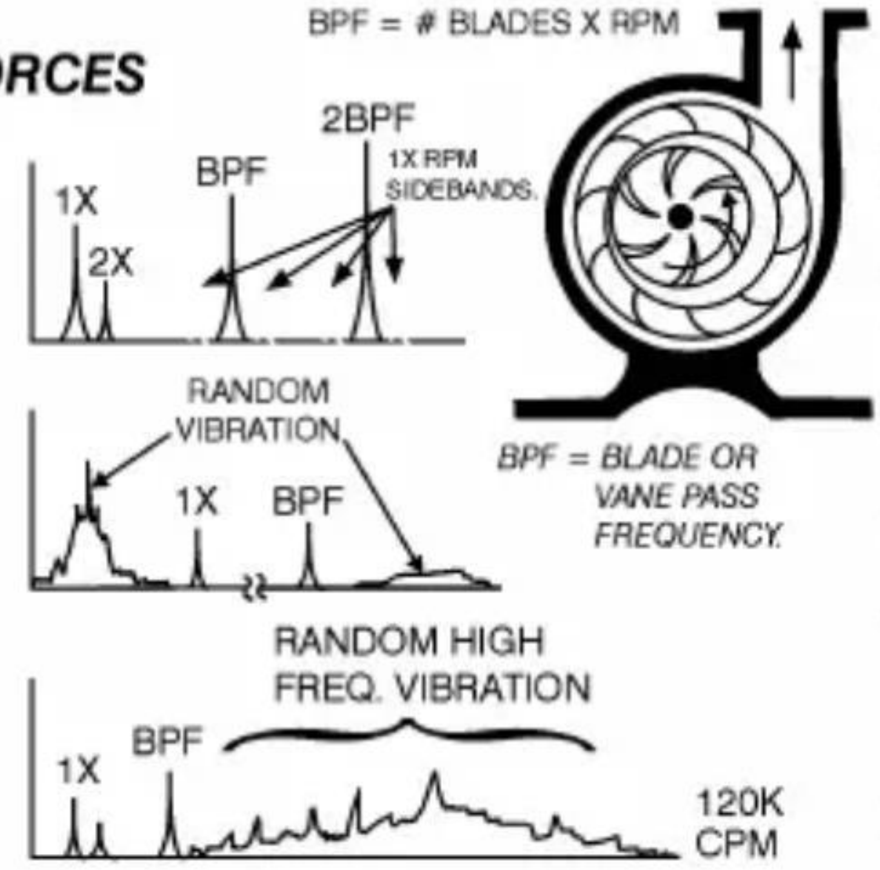
PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS

HYDRAULIC AND AERODYNAMIC FORCES

A. BLADE PASS & VANE PASS

B. FLOW TURBULENCE

C. CAVITATION



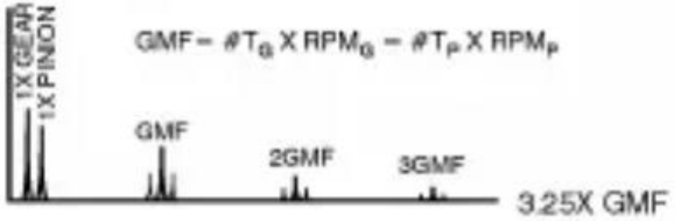
Blade Pass Frequency (BPF) = No. of Blades (or Vanes) X RPM. This frequency is inherent in pumps, fans and compressors, and normally does not present a problem. However, large amplitude BPF (and harmonics) can be generated in pump if gap between rotating vanes and stationary diffusers is not equal all 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.

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.

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).

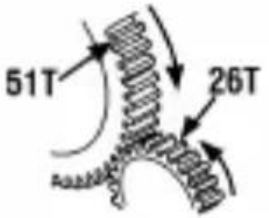
GEARS

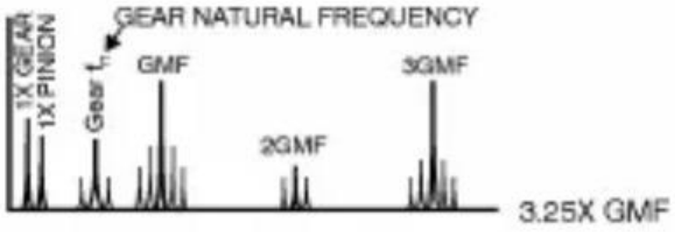
A. NORMAL SPECTRUM



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.

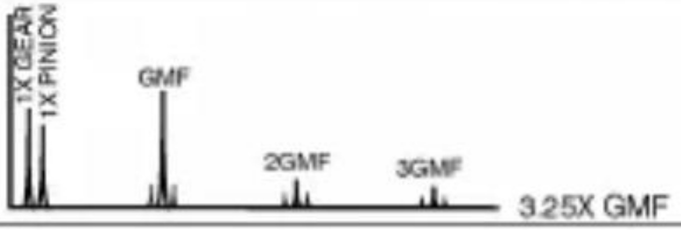
B. TOOTH WEAR





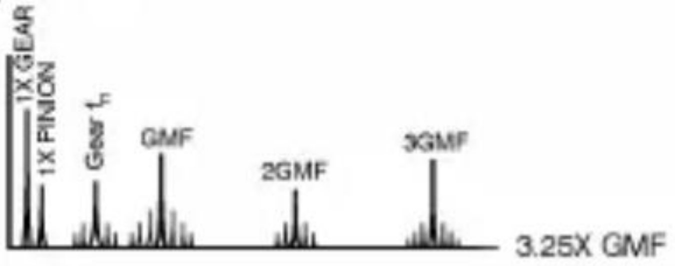
Key indicator of Tooth Wear is excitation of Gear Natural Frequency (f_n), along with sidebands around it spaced at the running speed of the bad gear. Gear Mesh Frequency (GMF) may or may not change in amplitude, although high amplitude sidebands and number of sidebands surrounding GMF usually occur 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



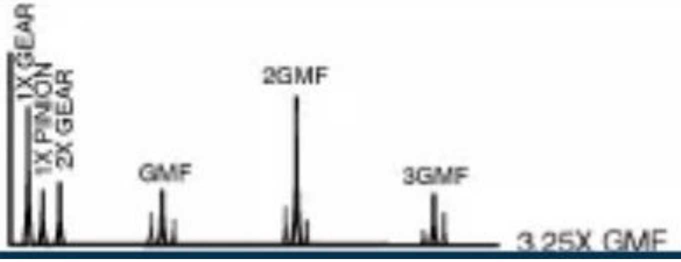
Gear Mesh Frequencies are often very sensitive to load. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low level, and no gear natural frequencies are excited. Each Analysis should be performed with system at maximum operating load for meaningful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

D. GEAR ECCENTRICITY AND BACKLASH



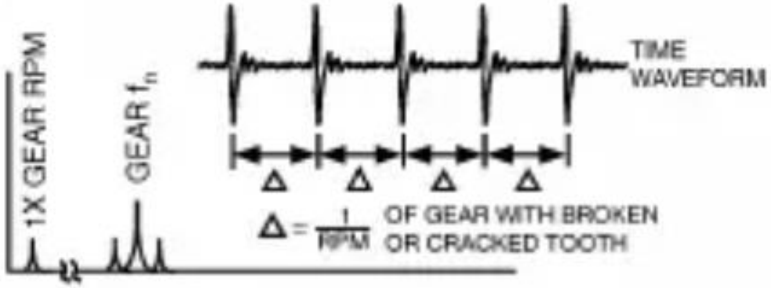
Fairly high amplitude sidebands around GMF harmonics often suggest gear eccentricity, backlash, or non-parallel shafts which allow the rotation of one gear to "modulate" either the GMF amplitude or the running speed of the other gear. The gear with the problem is indicated by the spacing of the sideband 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



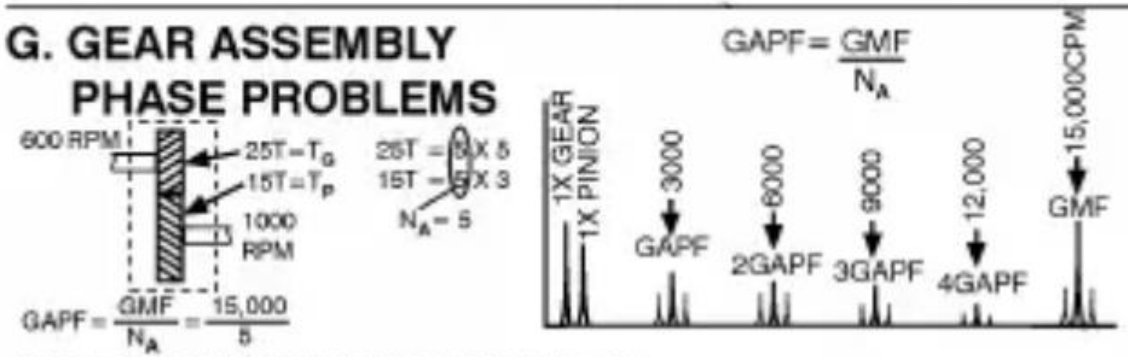
Gear Misalignment almost always excites second order or higher GMF harmonics which are sidebanded at running speed. Often will show only small amplitude 1X GMF, but much higher levels at 2X or 3X GMF. Important to set F_{MAX} high enough to capture at least 3 GMF harmonics. Also, sidebands around 2XGMF 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

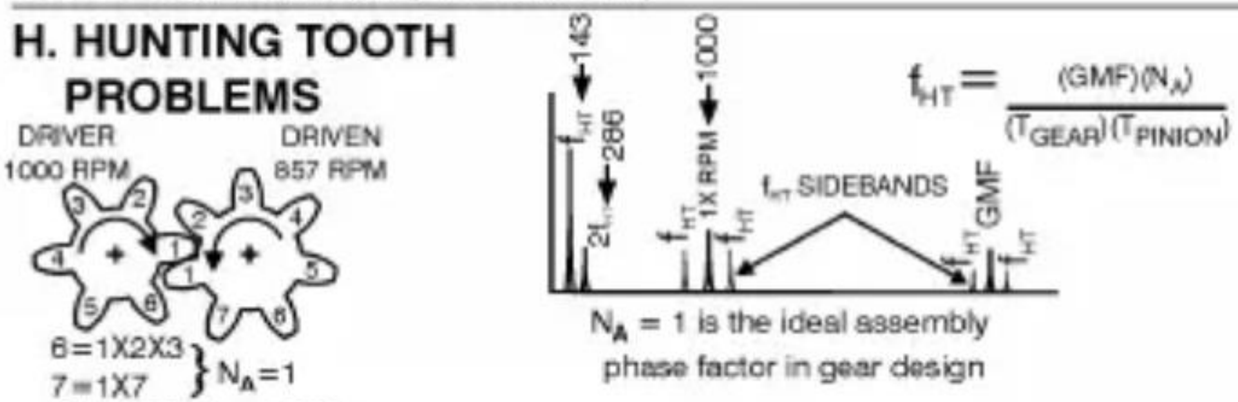


$GAPF = \frac{GMF}{N_A} = \frac{15,000}{5}$

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.

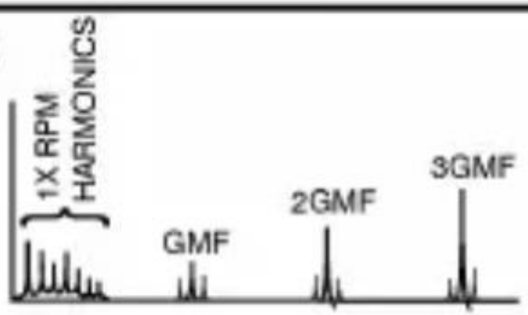
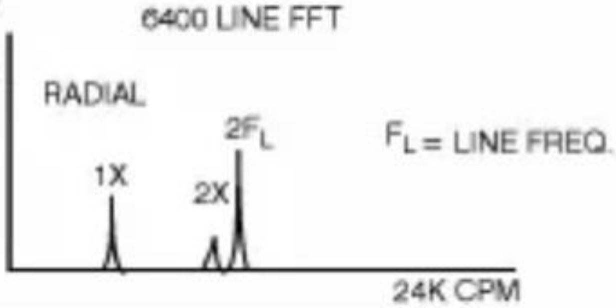
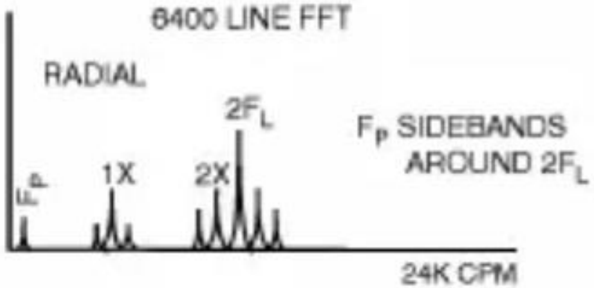
H. HUNTING TOOTH PROBLEMS



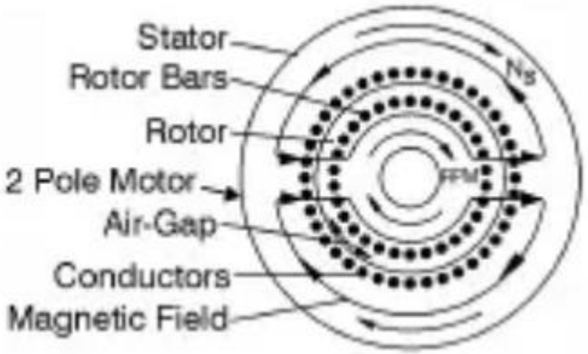
N_A = 1 is the ideal assembly phase factor in gear design

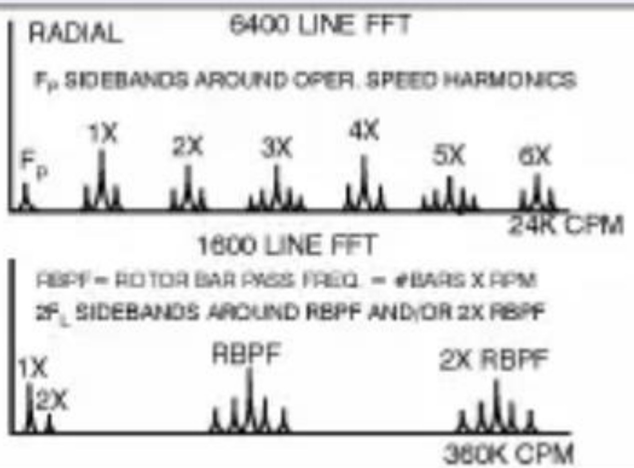
$f_{HT} = \frac{(6X\ 1000)(1)}{(6)(7)} = \frac{1000}{7}$ = 143 CPM (One Pulse Per 7 Pinion Revolutions)

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.

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
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).

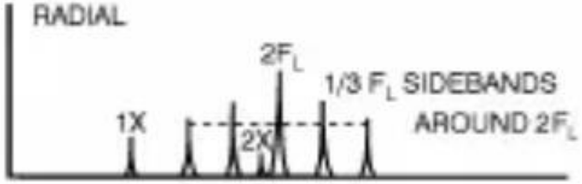
C. ROTOR PROBLEMS





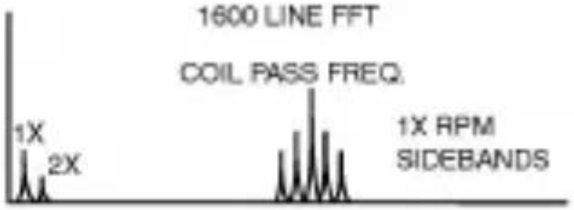
Broken or Cracked rotor bars or shorting rings; bad joints between rotor bars and shorting rings; or shorted rotor laminations will produce high 1X running speed vibration with pole pass frequency sidebands (F_P). In addition, these problems will often generate F_P sidebands around the second, third, 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.

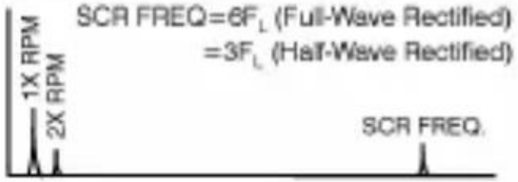
AC SYNCHRONOUS MOTORS (Loose Stator Coils)



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

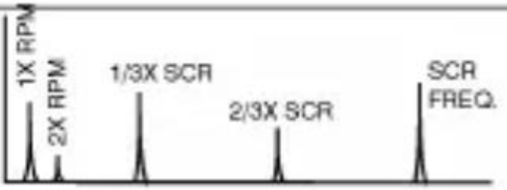


Many DC Motor and Control Problems can be detected by vibration analysis. Full-wave rectified, motors (6 SCR's) generate a signal at 6X Line Frequency ($6F_L$=360 Hz=21,600 CPM); while half-wave rectified DC motors (3 SCR's) generate 3X Line Freq. ($3F_L$=180 Hz=10,800 CPM). The SCR firing Frequency 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.

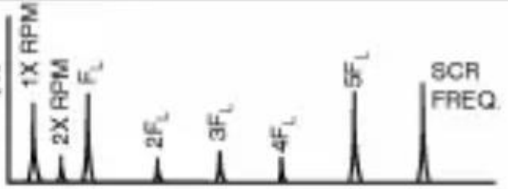
C. FAULTY FIRING CARD OR BLOWN FUSE



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).

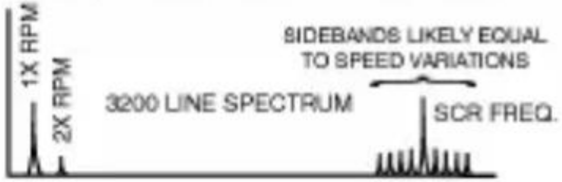
Caution: Card/SCR configuration should be known before troubleshooting motor (#SCR's, #Firing Cards, etc.).

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



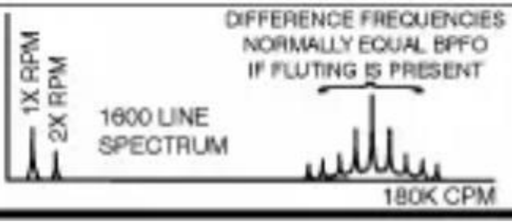
Faulty SCR's, Shorted Control Cards and/or Loose Connections can generate noticeable amplitude peaks at many combinations of line frequency (F_L) and SCR firing frequency. Normally, 1 bad SCR can cause high levels at F_L and/or $5F_L$ in 6 SCR motors. The point to be made is that neither F_L, $2F_L$,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

E. FAULTY COMPARITOR CARD

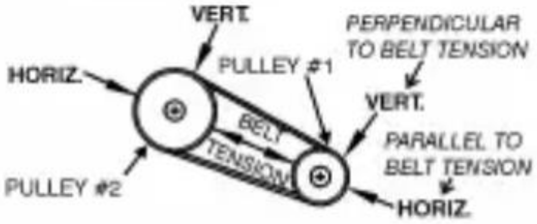
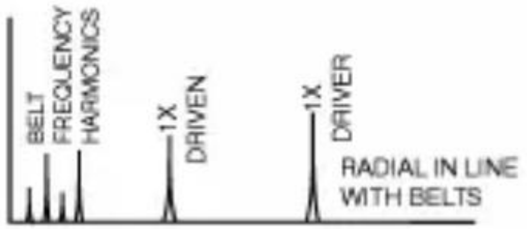
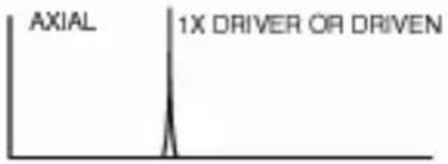
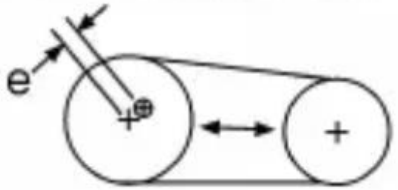
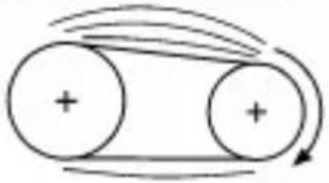
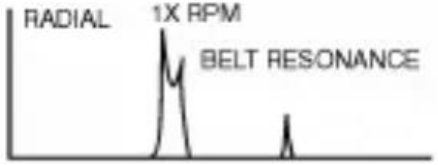


Faulty Comparitor Cards cause problems with RPM fluctuation or "hunting". This causes a constant collapsing and regenerating of the magnetic field. These sidebands often approximate the RPM fluctuation and require a high resolution FFT to even detect them. Such sidebands could also be due to generation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

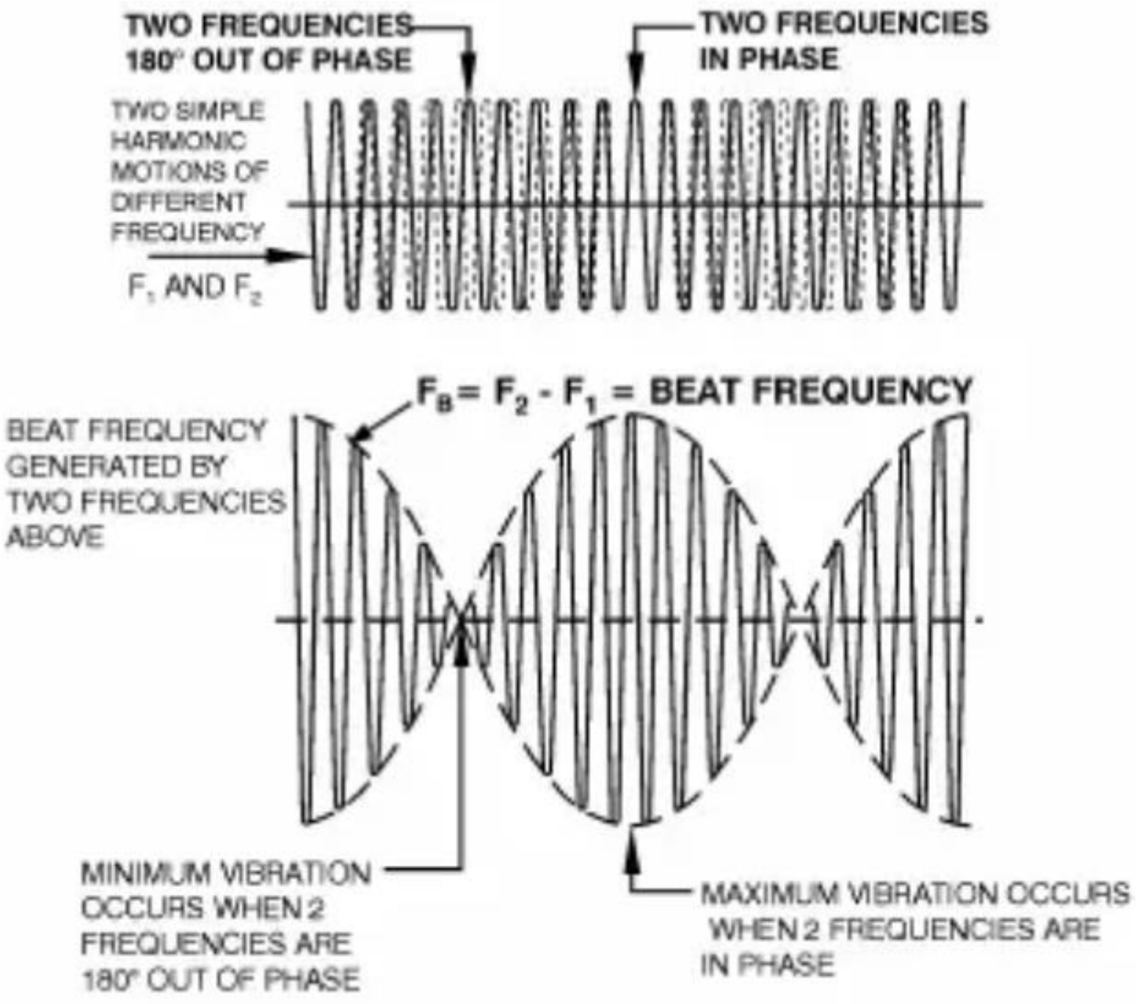
F. ELECTRICAL CURRENT PASSAGE THRU DC MOTOR BEARINGS



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.

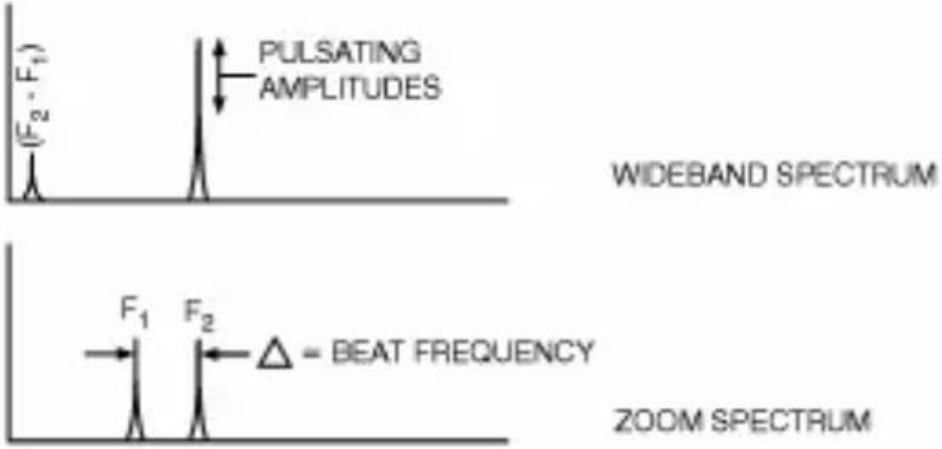
PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
BELT DRIVE PROBLEMS **A. WORN, LOOSE OR MISMATCHED BELTS** 	 $\text{PITCH DIAM}_1 \times \text{RPM}_1 = \text{PITCH DIAM}_2 \times \text{RPM}_2$	$\text{BELT FREQ.} = \frac{3.142 \times \text{PULLEY RPM} \times \text{PITCH DIAM.}}{\text{BELT LENGTH}}$ $\text{TIMING BELT FREQ.} = \text{BELT FREQ.} \times \text{\#BELT TEETH} = \text{PULLEY RPM} \times \text{\#PULLEY TEETH}$ 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

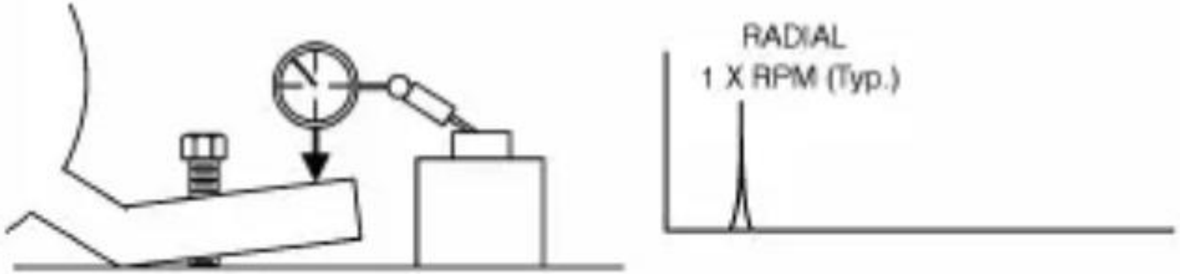


A Beat Frequency is the result of two closely spaced frequencies going into and out of synchronization with one another. The wideband spectrum normally will show one peak pulsating up and down. When you zoom into this peak (lower spectrum below), it actually shows two closely spaced peaks. The difference 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



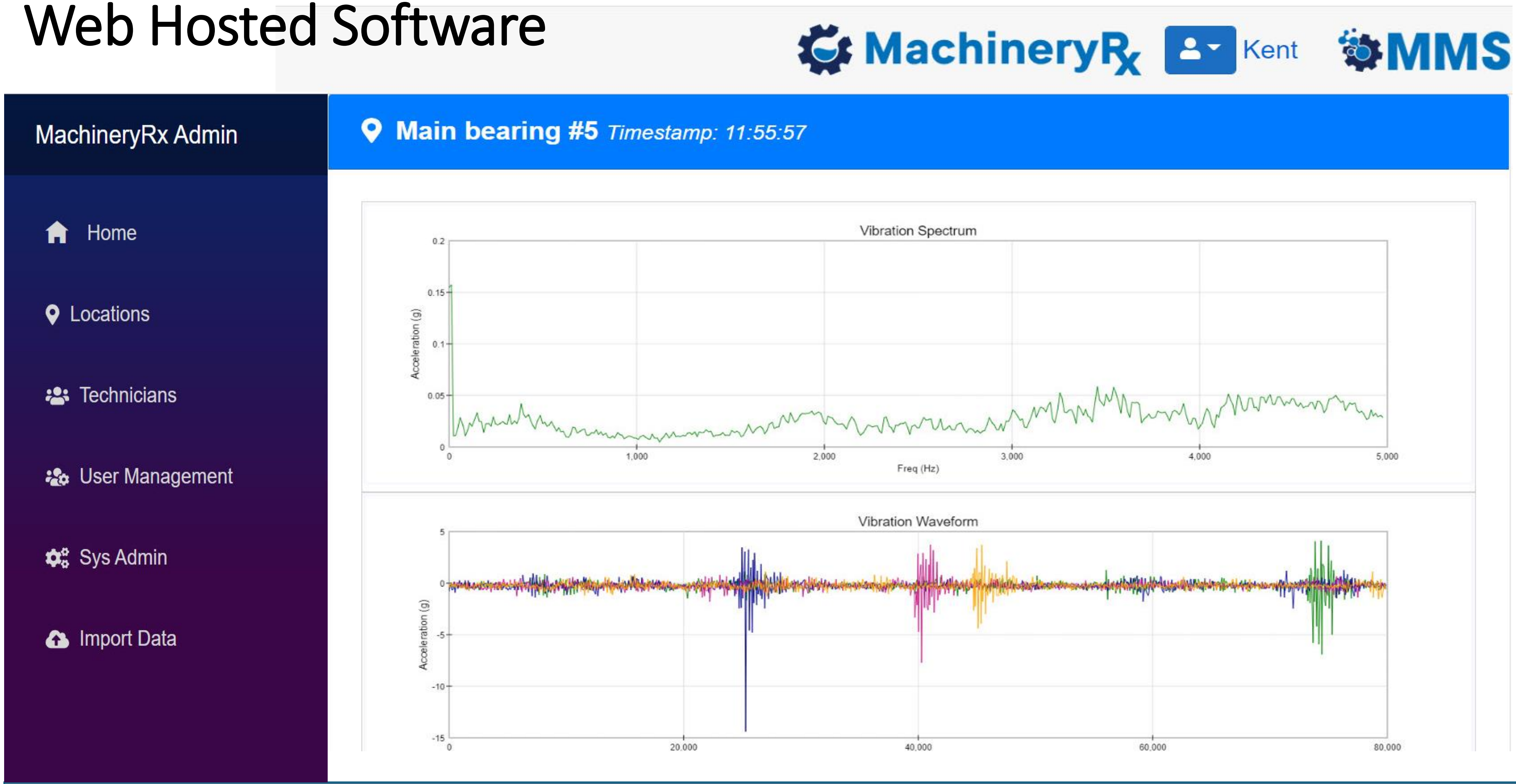
"***Soft Foot***" occurs when a machine's foot or frame deflects greatly when a hold-down bolt is loosened to hand tightness, causing the foot to rise more than approximately .002 - .003 inch. This does not always cause a great vibration increase. However, it can do so if the soft foot affects alignment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

Web Hosted Software



Easy User Interface

MachineryRx Admin

- Home
- Locations
- Technicians
- User Management
- Sys Admin
- Import Data

MachineryRx Kent MMS

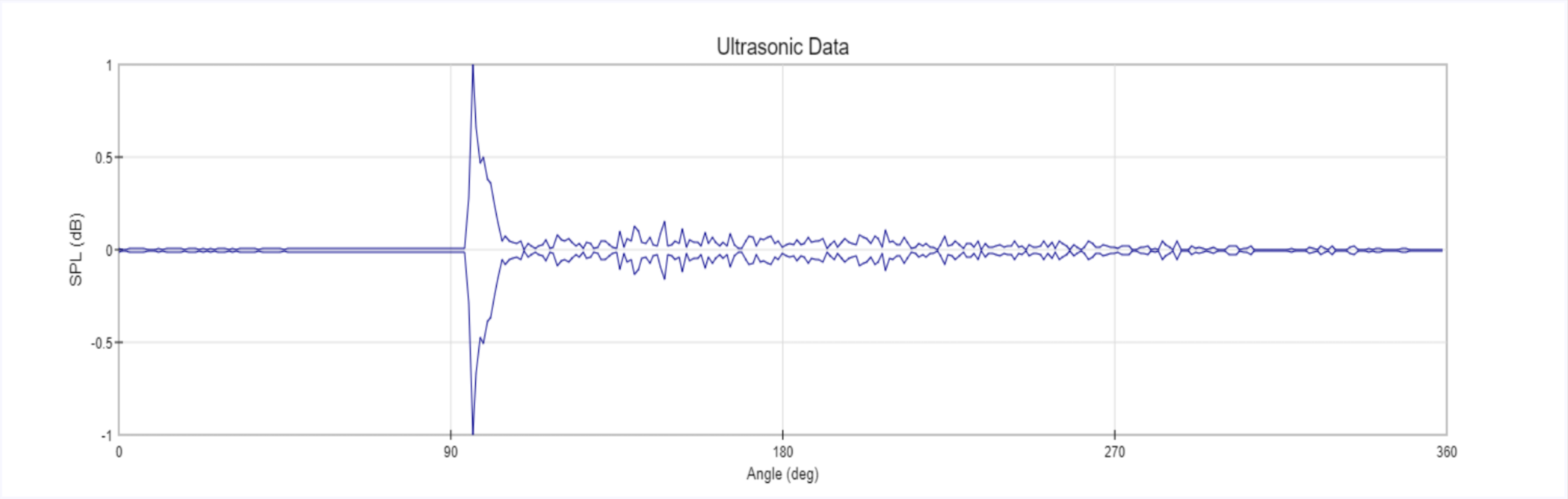
MachineryRx™

Moore Gas Pipeline

Balance Job Status

	Summary Level	Total Machines	Total Balances	Balanced Early	Balanced Late	Avg Hours Between Balances	Avg Minutes to Balance	Percent Left Good	Personnel Involved	Total Rated Power
↓	Division 1	5	8	100.0%	0.0%	305.6	56.5	100.0%	2	10,550
↓	Division 9	9	44	0.0%	2.0%	5067.0	1858.3	27%	3	19,400
		14	**52**	**15.4%**	**1.92%**	**2686.3 hrs**	**1421.5 min**	**38.5%**	**3**	**29,950**

Sample *Timestamp: 14:38:47*



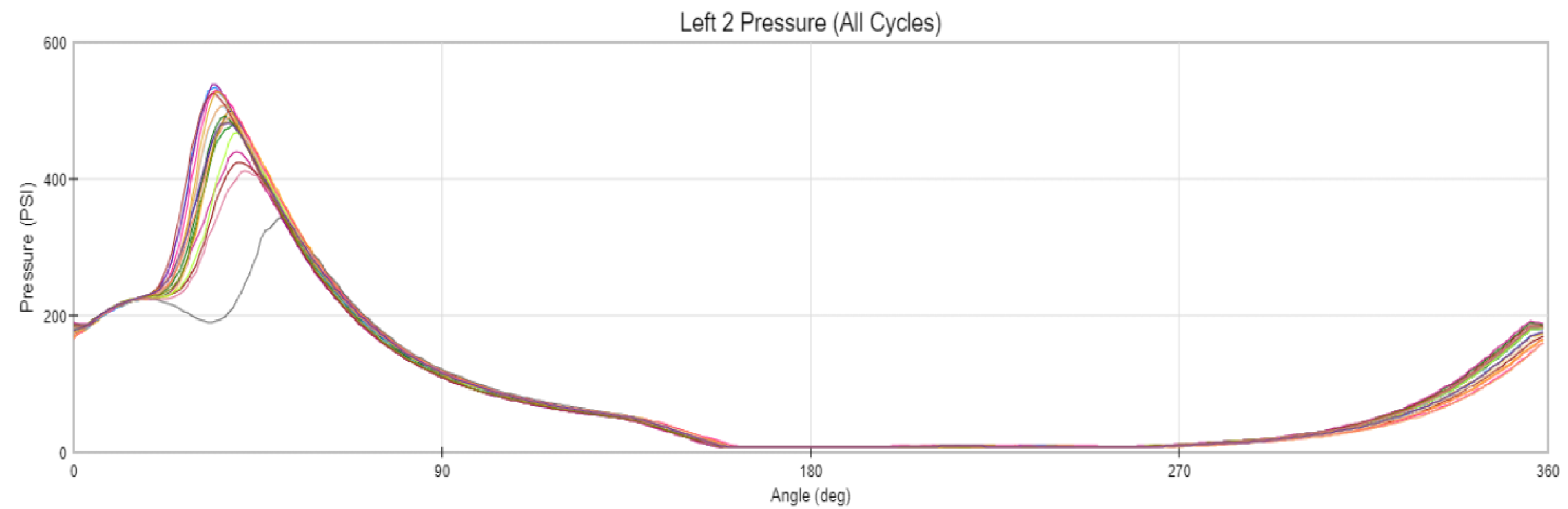
MachineryRx Admin

- Home
- Locations
- Technicians
- User Management
- Sys Admin
- Import Data

MachineryRx Kent MMS

Show	Left 3 +	407.6 PSI	227.9 PSI	531.5 PSI	95.39 PSI
Show	Left 4 +	468.8 PSI	394.2 PSI	629.9 PSI	45.42 PSI
Show	Left 5 +	454.2 PSI	349.0 PSI	538.7 PSI	48.15 PSI
Show	Right 1 +	423.6 PSI	216.4 PSI	523.6 PSI	84.35 PSI
Show	Right 2 +	457.0 PSI	339.3 PSI	518.8 PSI	41.09 PSI
Show	Right 3 +	472.0 PSI	380.1 PSI	590.8 PSI	49.45 PSI
Show	Right 4 +	451.5 PSI	385.6 PSI	535.9 PSI	34.54 PSI
Show	Right 5 +	460.4 PSI	384.9 PSI	523.4 PSI	40.59 PSI

Smoothing: 3 | All | Step | Mean Rep | Min/Max Reps | All Reps | Show Stats | Plot PFP



MachineryRx Admin

- Home
- Locations
- Technicians
- User Management
- Sys Admin
- Import Data

MachineryRx MMS Kent

MachineryRx™

Moore Gas Pipeline

Balance Job Status

	Summary Level	Total Machines	Total Balances	Balanced Early	Balanced Late	Avg Hours Between Balances	Avg Minutes to Balance	Percent Left Good	Personnel Involved	Total Rated Power
	Division 1	5	8	100.0%	0.0%	305.6	56.5	100.0%	2	10,550
	Division 9	9	44	0.0%	2.0%	5067.0	1858.3	27%	3	19,400
		14	**52**	**15.4%**	**1.92%**	**2686.3 hrs**	**1421.5 min**	**38.5%**	**3**	**29,950**

Recent Balance Jobs

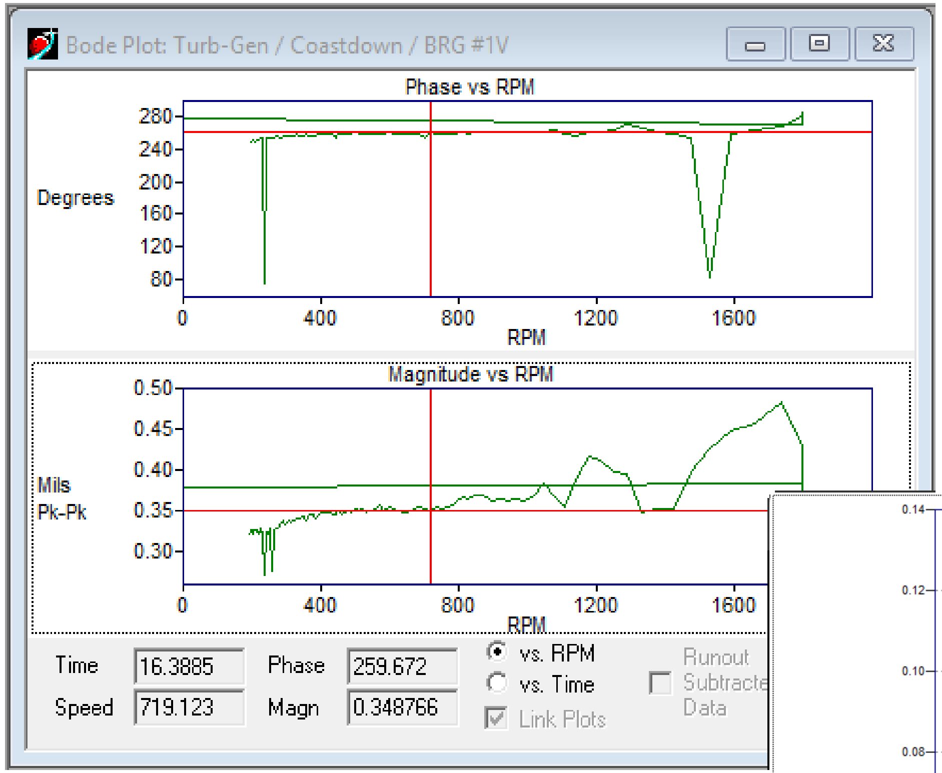
	Location	Rated Power	Model Number	Manufacturer	Balanced By	Balance Date	Time To Balance (Minutes)	As Found Balance	As Found Condition	As Left Balance	As Left Condition
	Unit 1	1500	410-KVR	Ingersoll Rand	PaulW	Sep 28, 2021	56 min	?	Fair	3.0%	Good
	Unit 3A	1650	HBA-6T	Clark	Kent	Sep 24, 2021	0 min	?	Unknown	n/a	Unknown
	Test 4 Cylinder	1000	Model Number	Cooper	Kent	Sep 22, 2021	0 min	2.6%	Good	n/a	n/a
	Test 4 Cylinder	1000	Model Number	Cooper	Kent	Sep 22, 2021	6 min	2.6%	Good	2.6%	Good
	Test 4 Cylinder	1000	Model Number	Cooper	Kent	Sep 10, 2021	0 min	2.6%	Good	n/a	Unknown

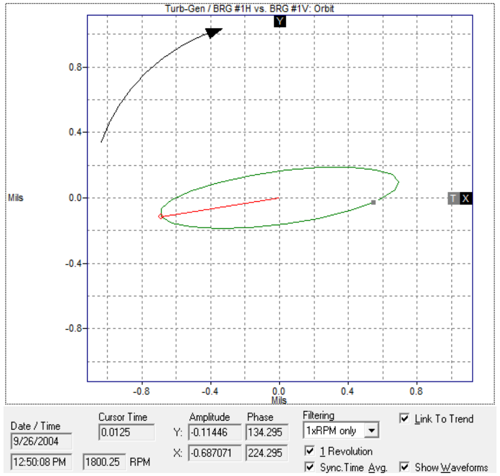
+10 Jobs

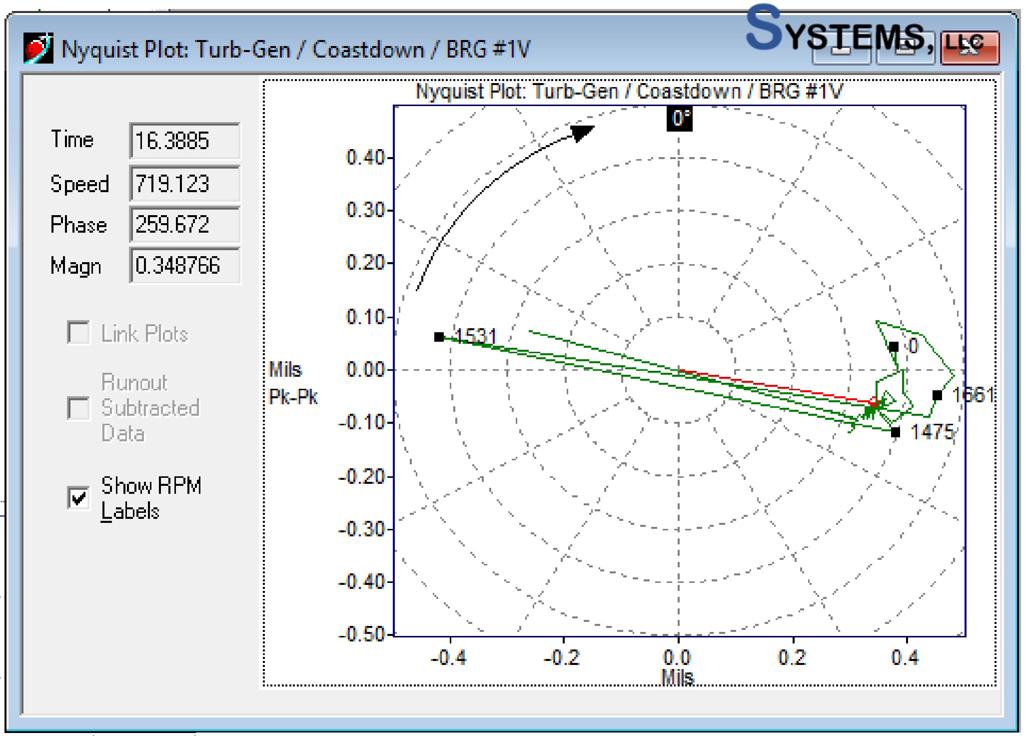
MachineryRx Admin
Home
Locations
Technicians
User Management
Sys Admin
Import Data
MachineryRx
Kent
MMS
Main bearing #5 Timestamp: 11:55:57
Vibration Spectrum
Acceleration (g)
Freq (Hz)
0
0.05
0.1
0.15
0.2
1,000
2,000
3,000
4,000
5,000
Vibration Waveform
Acceleration (g)
5
-5
-10
-15
20,000
40,000
60,000
80,000

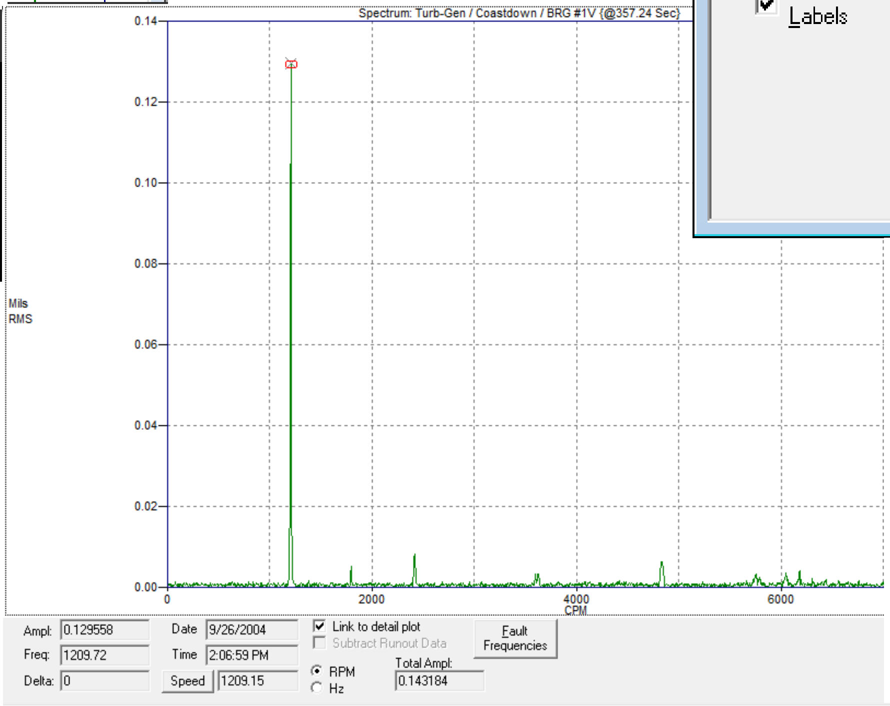
Rotating Diagnostics

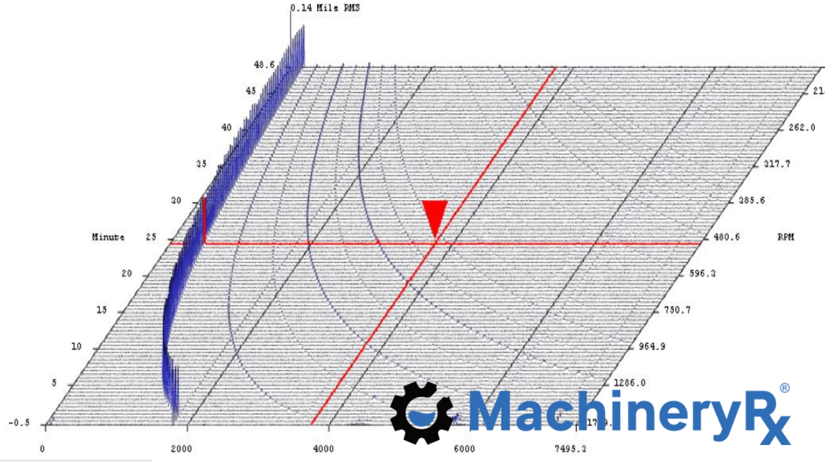
MACHINERY MONITORING SYSTEMS, LLC









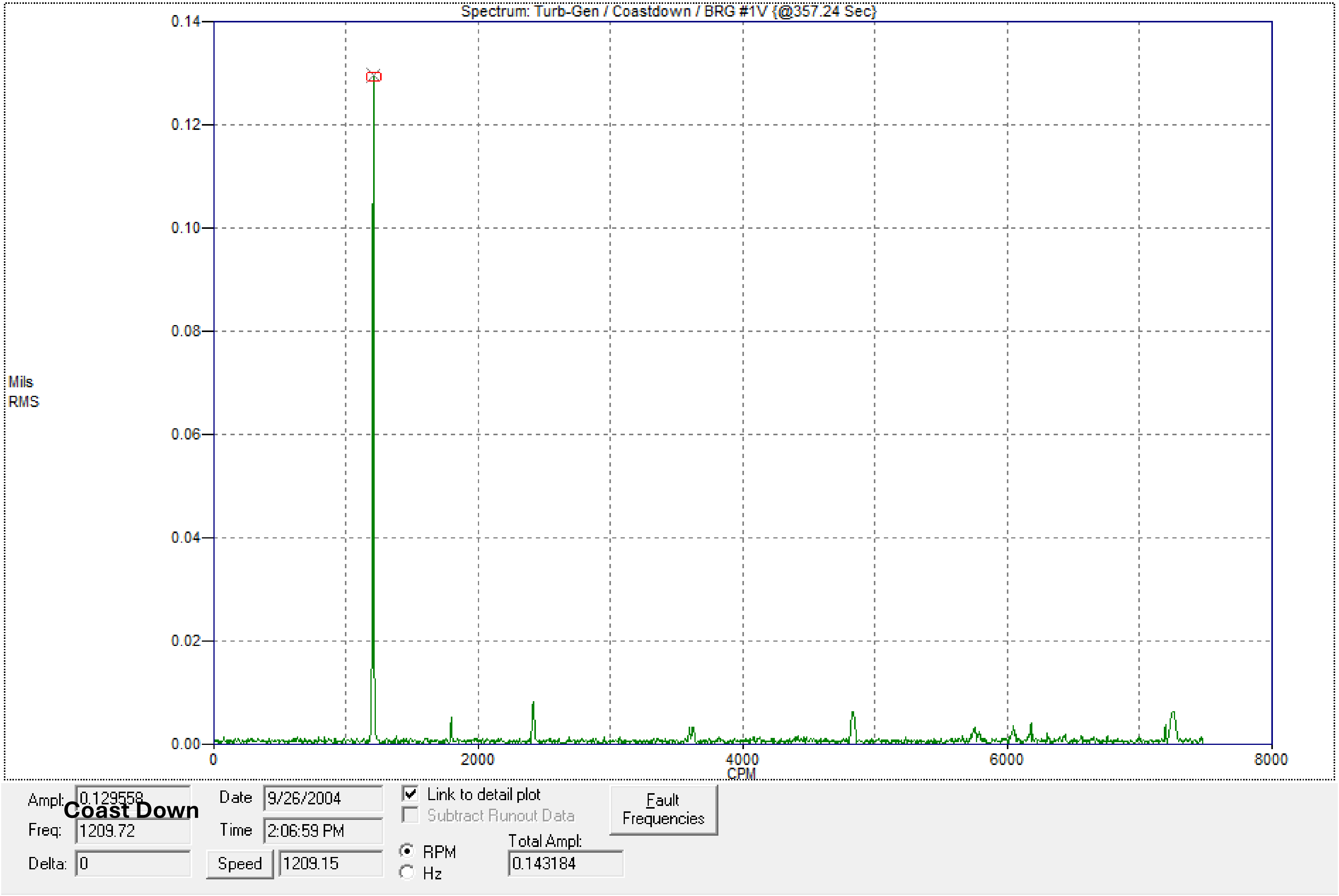


Full diagnostic suite for rotating equipment including bearing fault and gear-mesh frequency libraries

MachineryRx

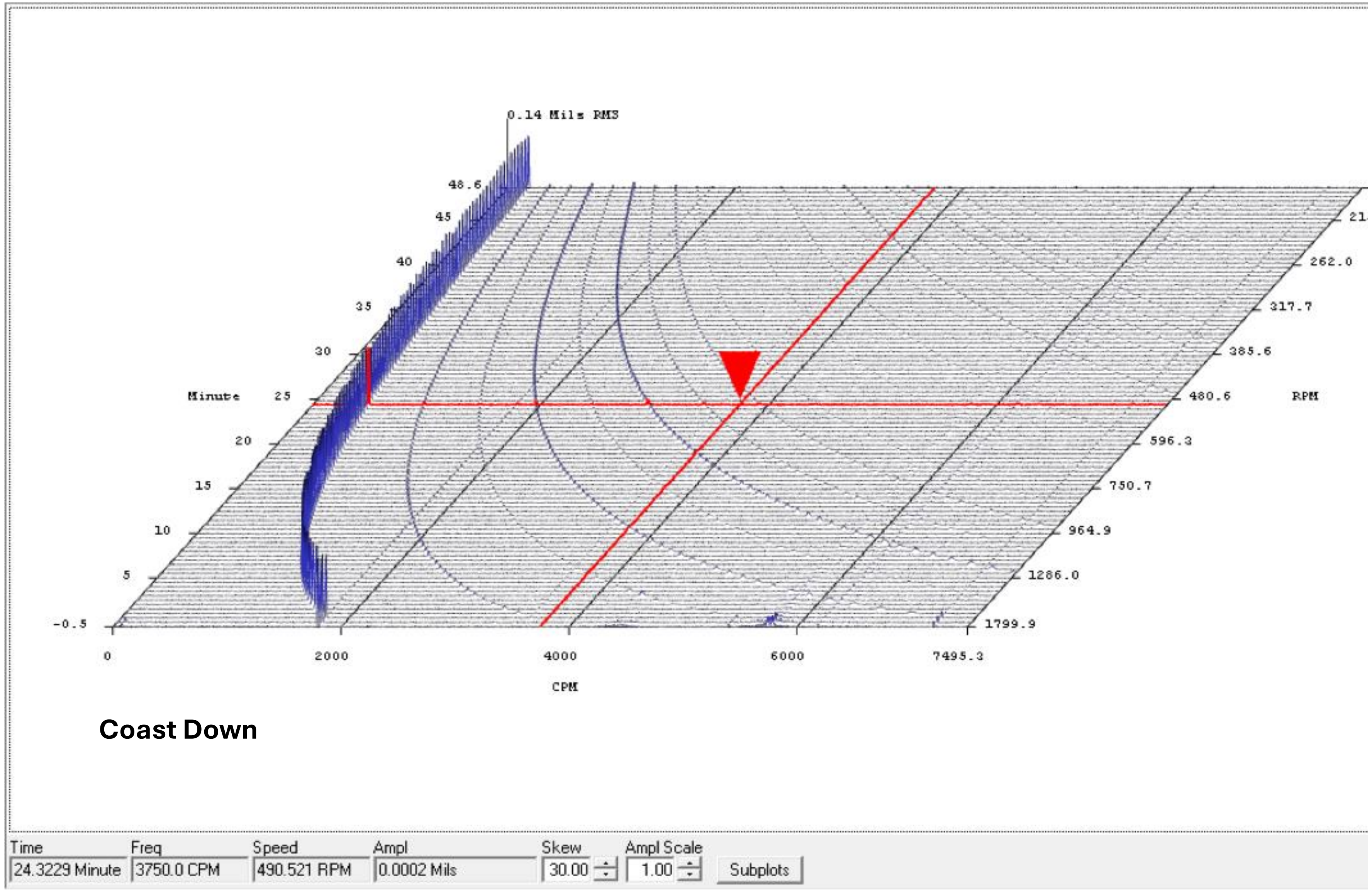
Pathfinder Software, Spectrum Analysis

Machinery Monitoring Systems, LLC



MachineryRx®

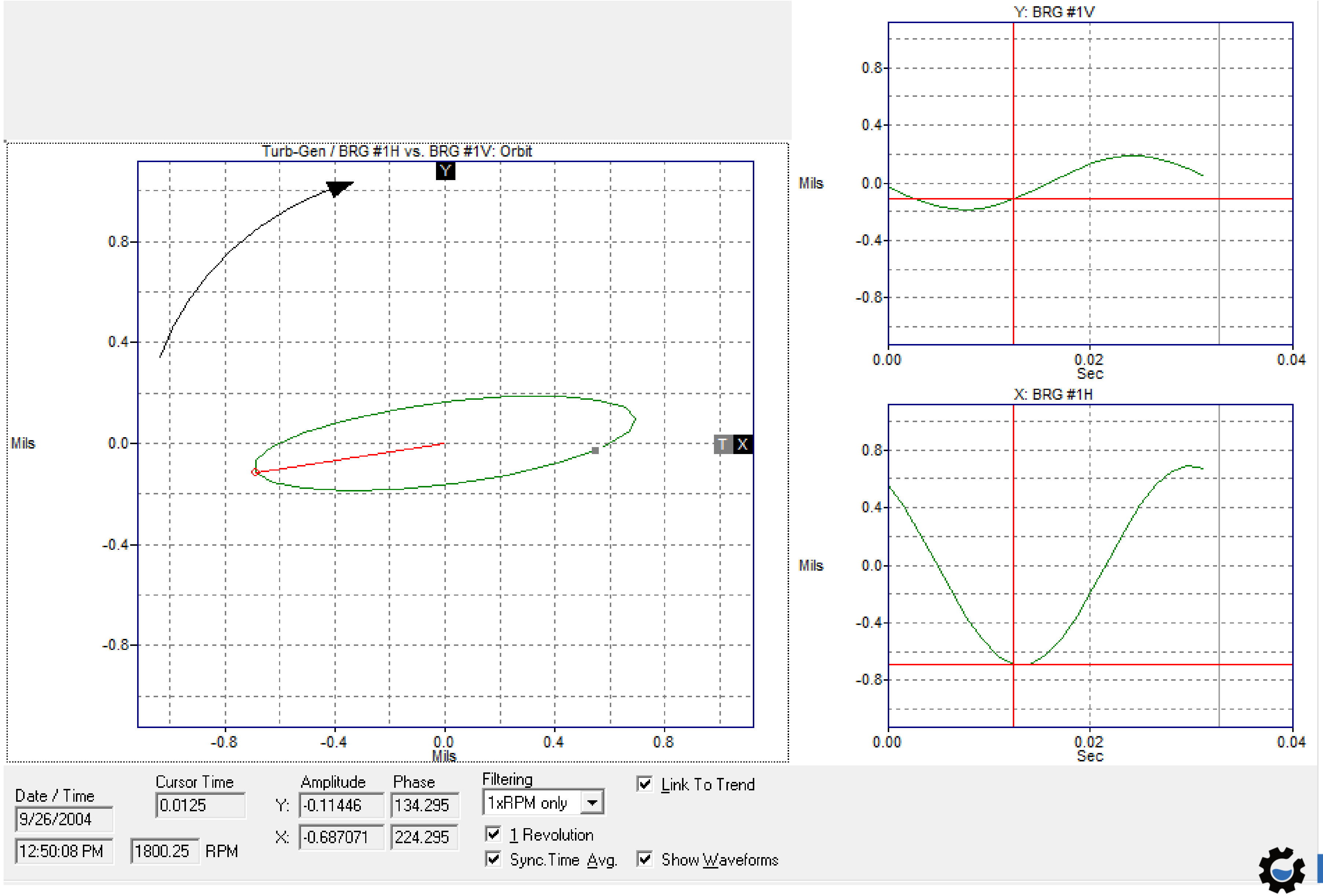
Pathfinder Software, Cascade Plot



Machinery Monitoring Systems, LLC

MachineryRx®

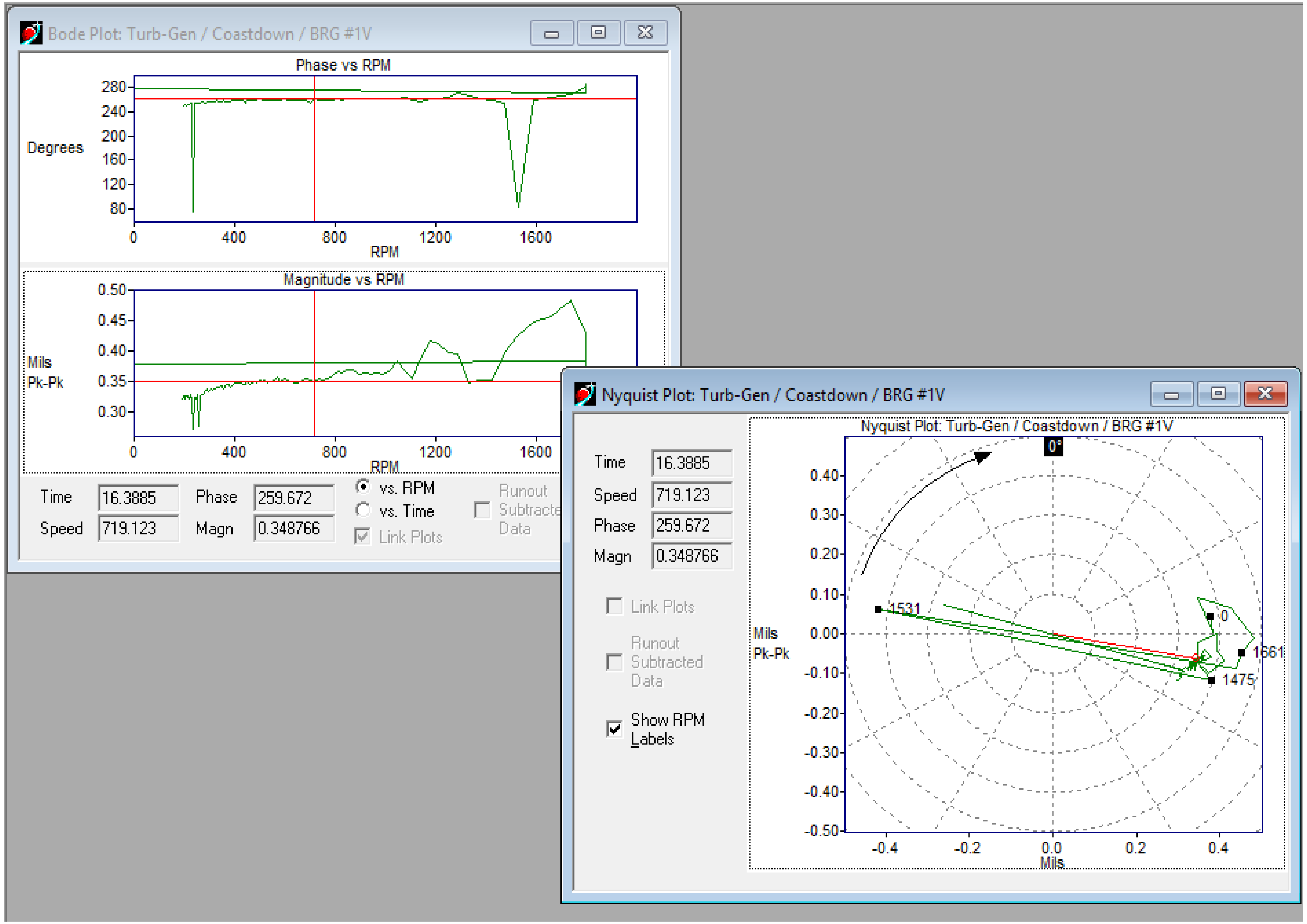
Pathfinder Software, Orbital Plots



MACHINERY MONITORING SYSTEMS, LLC

MachineryRx®

Pathfinder Software, Nyquist/Bode Plots



MACHINERY MONITORING SYSTEMS, LLC

MachineryRx