電機機械原理與智慧監測
Electromechanical Principles and Intelligent Monitoring

國立臺灣科技大 學機械工程系

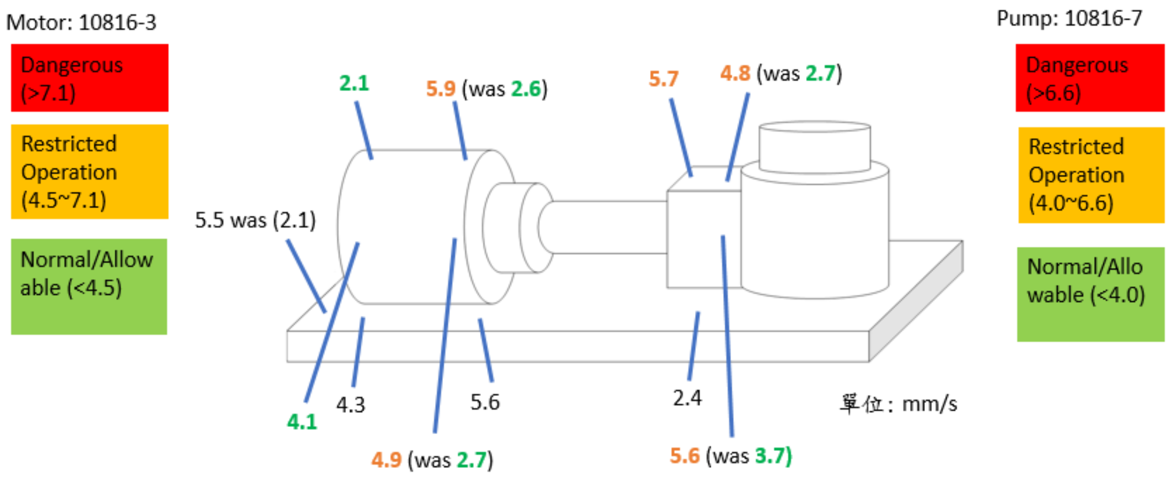
助理教授：藍振洋

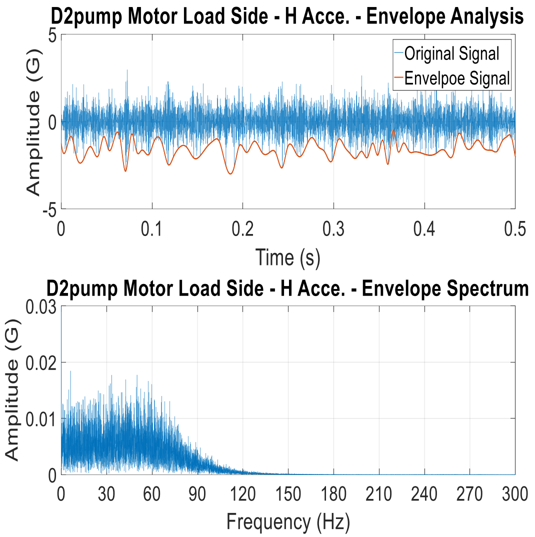
轉動設備狀態監診

狀態監診

Vibration Analysis

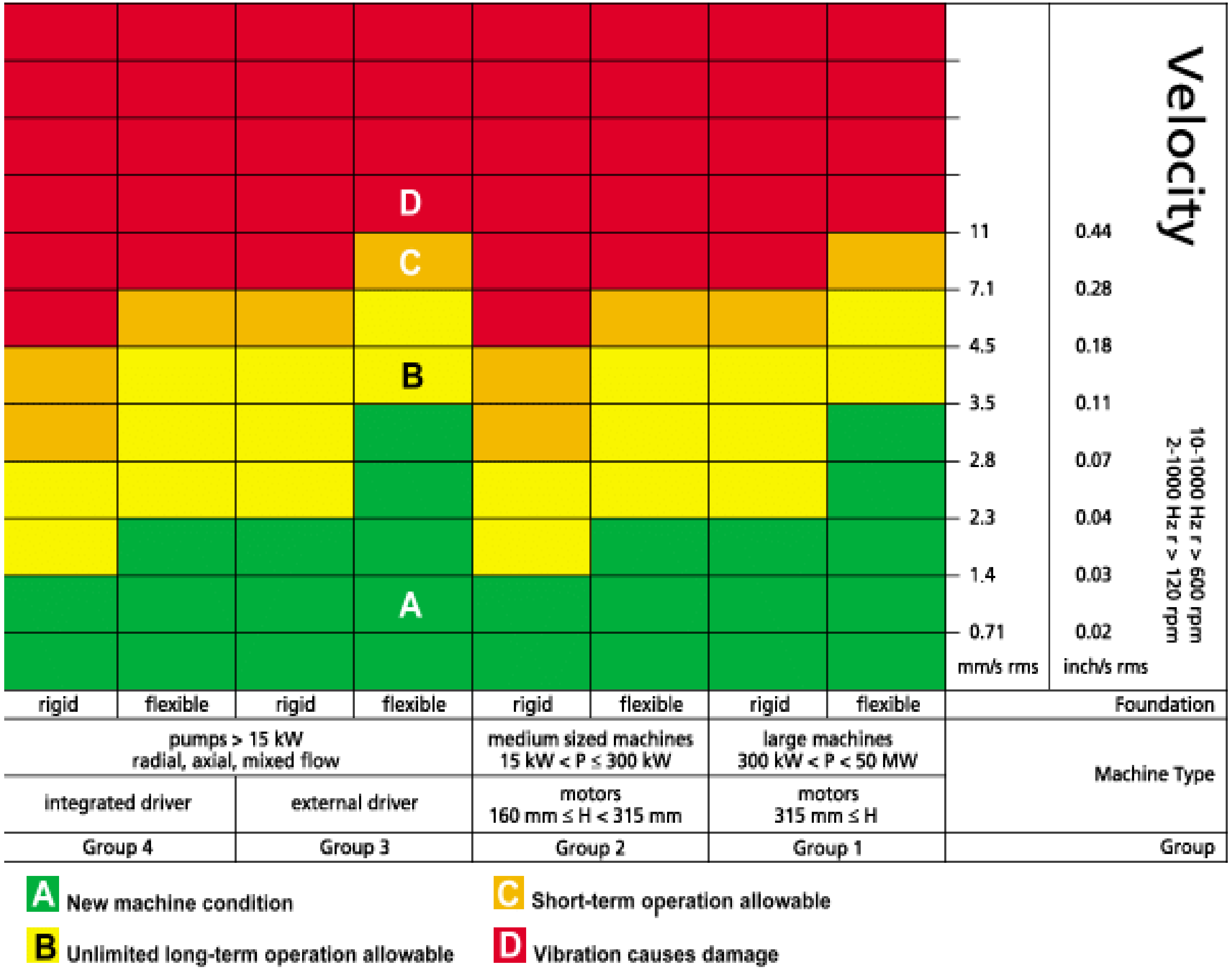
- Vibration Values (ISO-10816)
- Vibration Signatures





Envelope Analysis

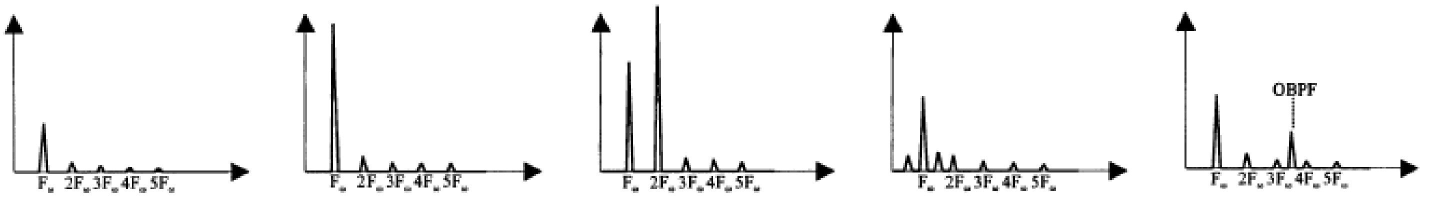
狀態監診



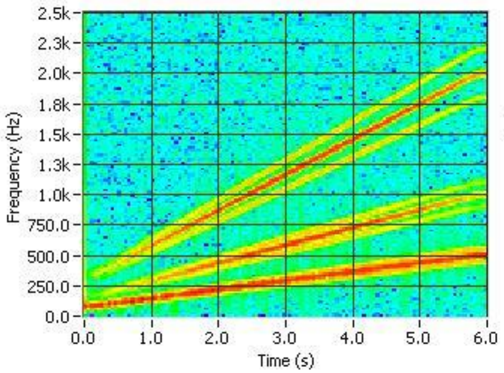
Velocity
D
C
B
A
11
7.1
4.5
3.5
2.8
2.3
1.4
0.71
mm/s rms
0.44
0.28
0.18
0.11
0.07
0.04
0.03
0.02
inch/s rms
10-1000 Hz r > 600 rpm
2-1000 Hz r > 120 rpm
rigid
flexible
Foundation
pumps > 15 kW
radial, axial, mixed flow
medium sized machines
15 kW < P ≤ 300 kW
large machines
300 kW < P < 50 MW
Machine Type
integrated driver
external driver
motors
160 mm ≤ H < 315 mm
motors
315 mm ≤ H
Group 4
Group 3
Group 2
Group 1
Group
A New machine condition
B Unlimited long-term operation allowable
C Short-term operation allowable
D Vibration causes damage

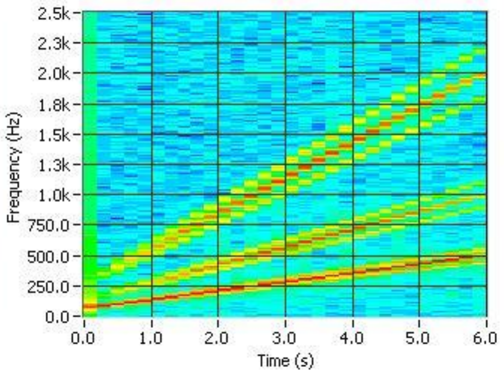
狀態監診

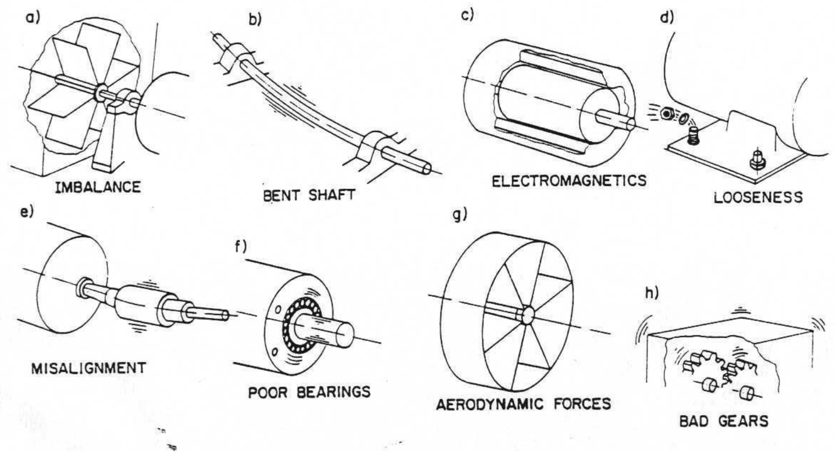
Vibration Analysis



Unfault　Unbalance　Misalignment　Looseness　Outer ball bearing





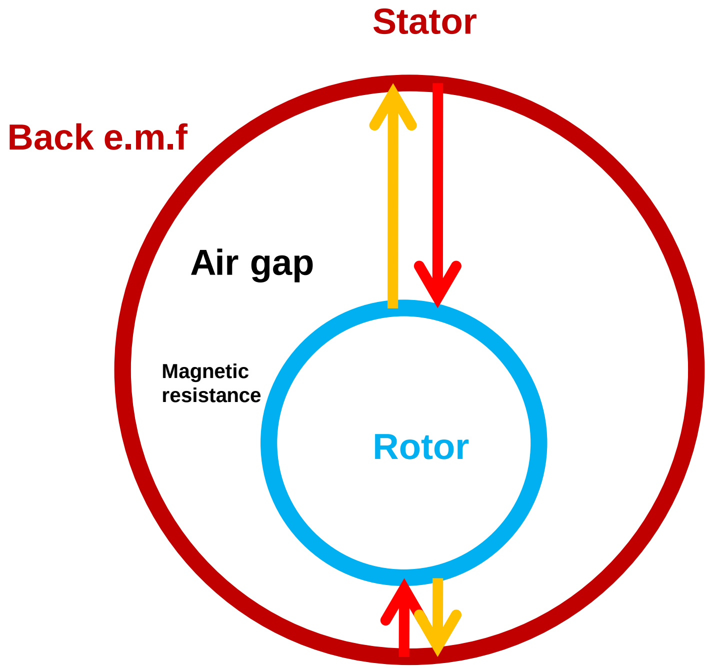


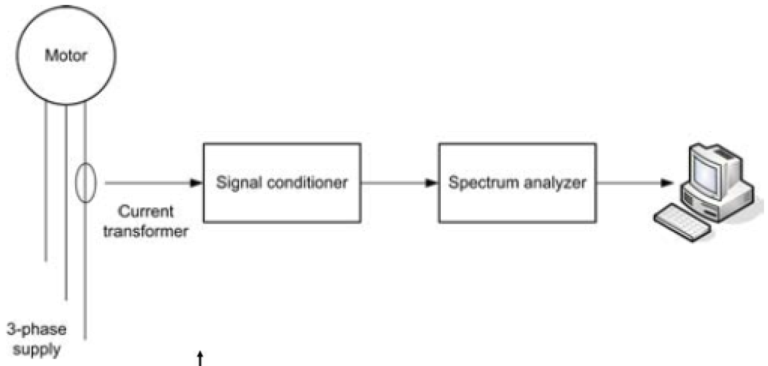
Source:IEEE Transactions on instrumentation and measurement

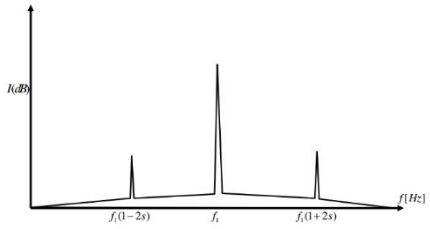
狀態監診

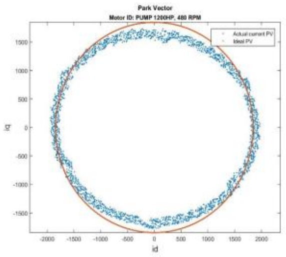
Motor Current Signature Analysis

- Vibration from mechanical parts will affect air gap clearance.
- The magnetic resistance is large for air compared to metal. (~10K:1)
- The back e.m.f will be affected due to rotor vibration. This produces signature in current waveform.
- The analysis on current is thus used to detect faults.

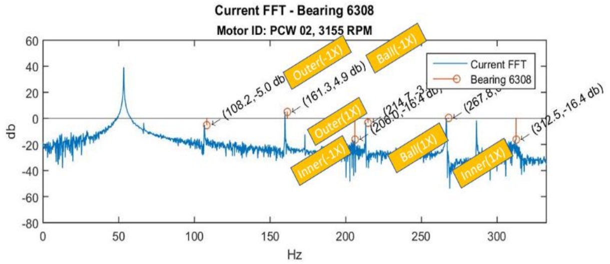








Park‘s Vector Analysis



MCSA

狀態監診

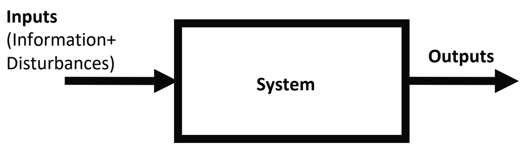
Comparison Between Vibration Analysis and MCSA

Method	Major Advantages	Major Disadvantage
Vibration	➢ Reliable; ➢ Standardized	➢ Expensive; ➢ Intrusive; ➢ Subject to sensor failure; ➢ Noise and disturbance; ➢ Limited performance for low speed rotation; ➢ Accessibility
MCSA	➢ No addition sensor needed; ➢ Inexpensive; ➢ Non-intrusive; ➢ Easy to impalement; ➢ Good sensibility to motor faults; ➢ Standardized	➢ Still in development stage; ➢ Less sensitive to mechanical faults; ➢ Load Requirement

Signal Based and Model Based

Model Based vs Signature based
Differs in design concepts

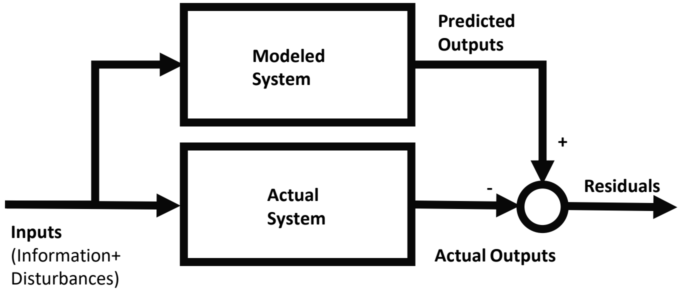
Signal based



Signal based systems:
- Vibration waveform
- Electrical waveform MCSA and ESA

Disturbances as part of measurement inputs.

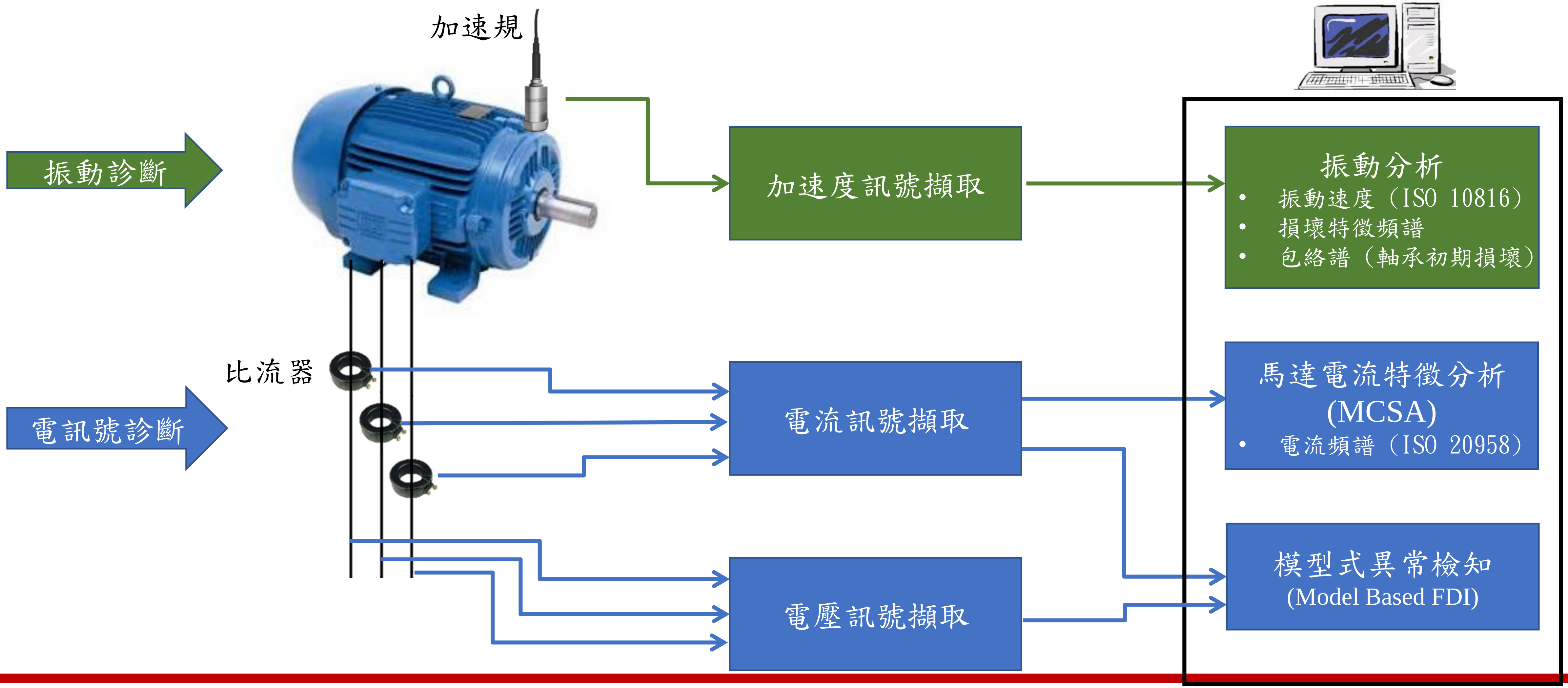
Model based



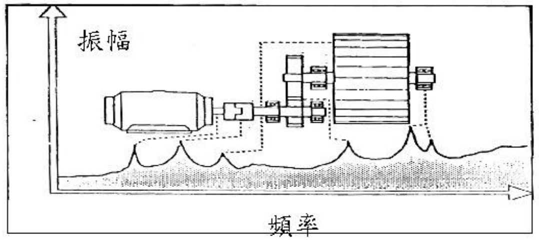
Model based systems takes disturbances into account and utilize the residuals between model predicted outputs with actual outputs.

ISO-20958

狀態監診


加速規
振動診斷
比流器
電訊號診斷
加速度訊號擷取
電流訊號擷取
電壓訊號擷取
振動分析
• 振動速度（ISO 10816）
• 損壞特徵頻譜
• 包絡譜（軸承初期損壞）
馬達電流特徵分析
(MCSA)
• 電流頻譜（ISO 20958）
模型式異常檢知
(Model Based FDI)

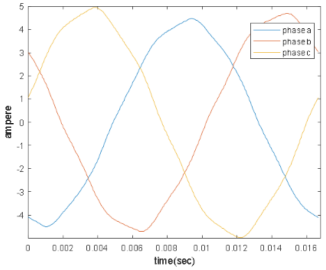
狀態監診



選定設備與安裝感測器

擷取三相電壓、電流與振動訊號

訊號時域分析與頻域分析



建立正常馬達模型

建立異常馬達模型

能耗比較

殘差分析

模型驗證

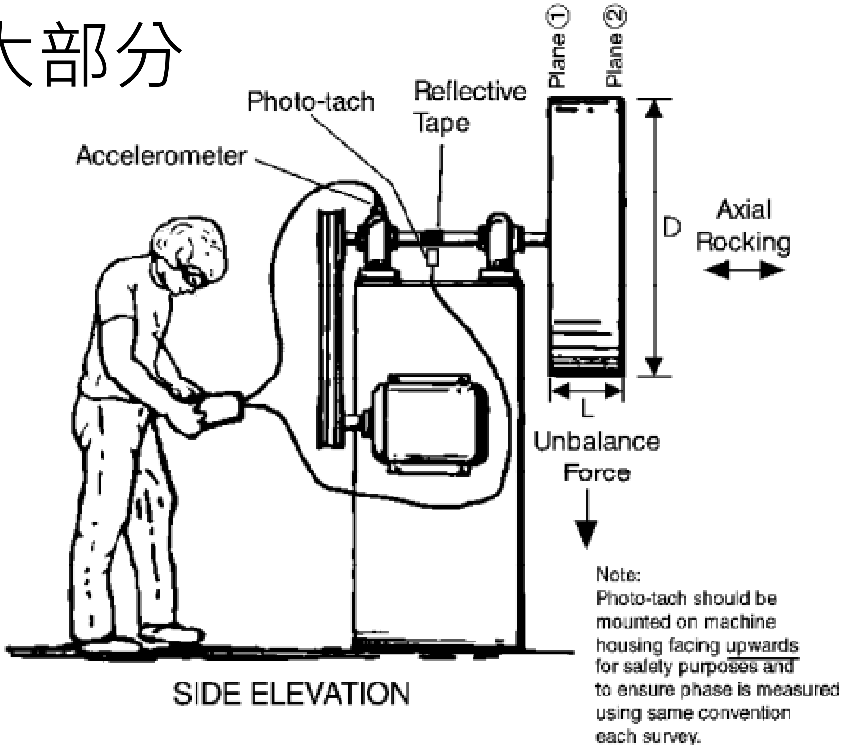
建立原始資料及歷史資料庫

AIOT架設應用

轉動設備狀態監診 － 振動頻譜

振動頻譜分析

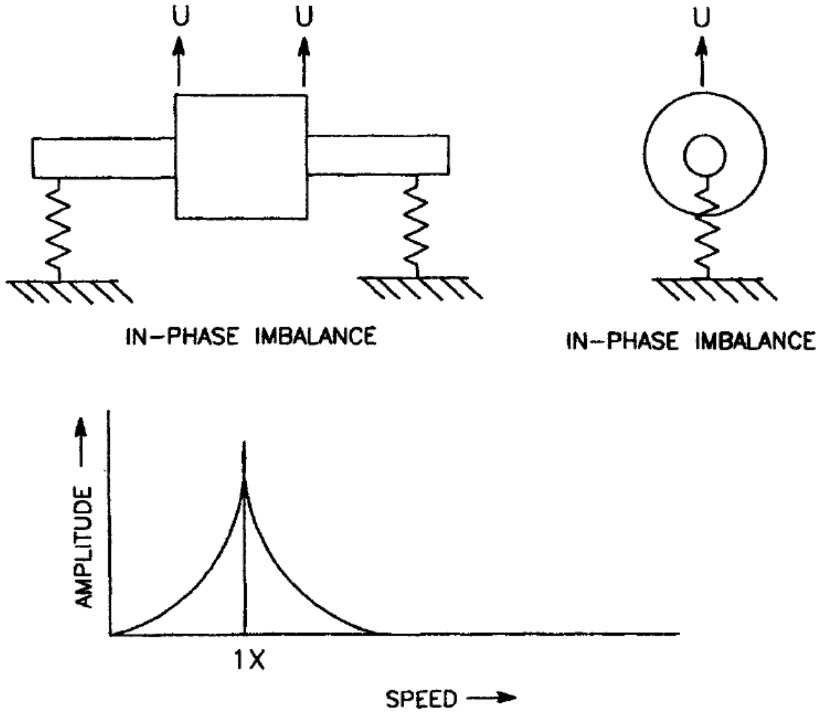
- 搭配在機械方面之旋轉機械中，各種故障因素的**物理現象**。
- 利用訊號分析中**頻率域（frequency domain）訊號**來呈現其現象。
- 有關之旋轉機械的各種故障或振動特徵，大部分皆與馬達旋轉速度有關。



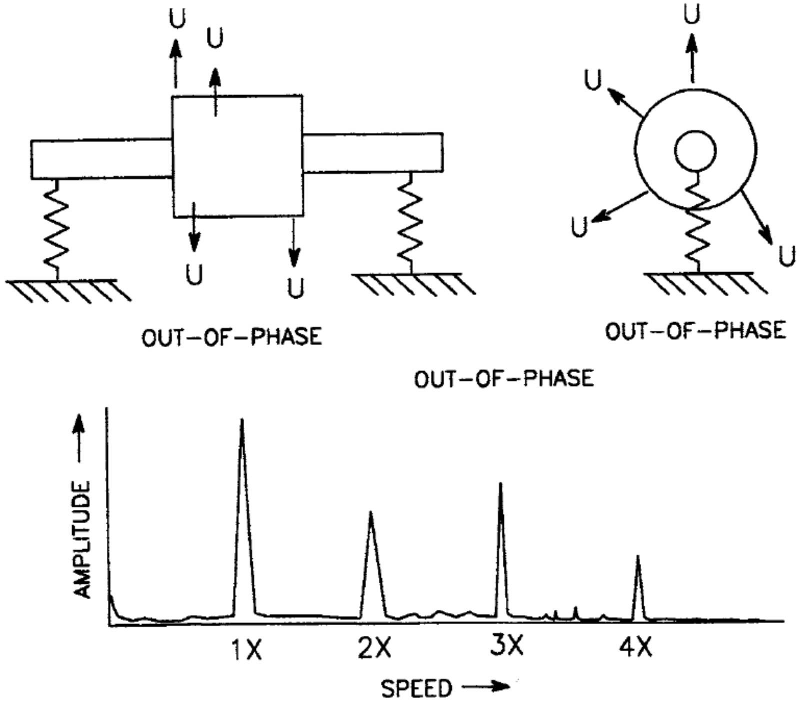
振動頻譜分析 – 不平衡

- 旋轉機械都存在著某種程度的不平衡，且是最普遍存在的異常現象，如轉子結構設計不合理、材質不均勻、人為裝配誤差等。
- 在振動頻率域上會產生馬達真實轉速頻率的一倍頻(1x)。

單面機械不平衡與頻域特徵

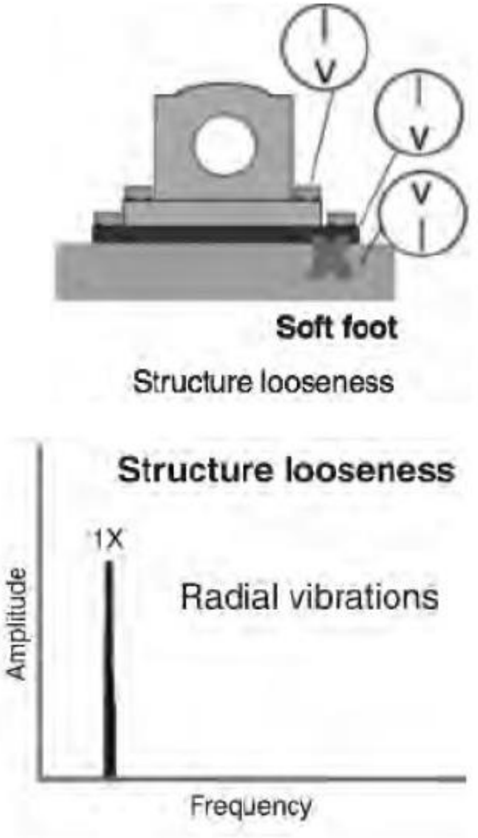


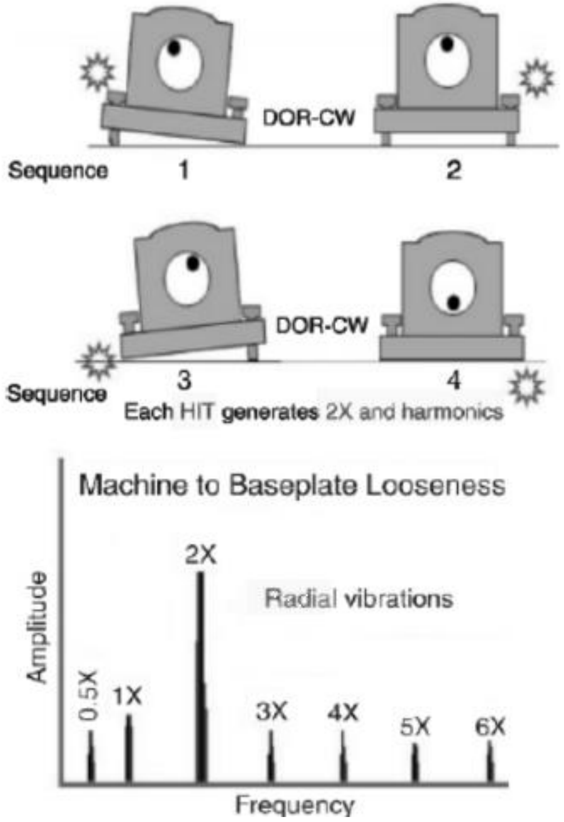
多面機械不平衡與頻域特徵

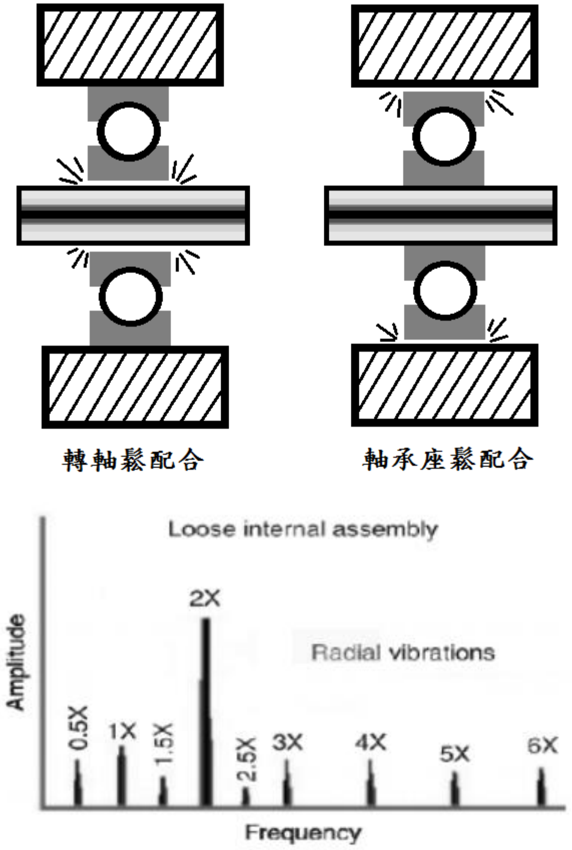


振動頻譜分析 – 機械鬆動

- 旋轉機械中主要有三種類型的機械鬆動。其振動通常在徑向較大。主要是由於**旋轉的離心力**使結構脫離固定端的緣故。

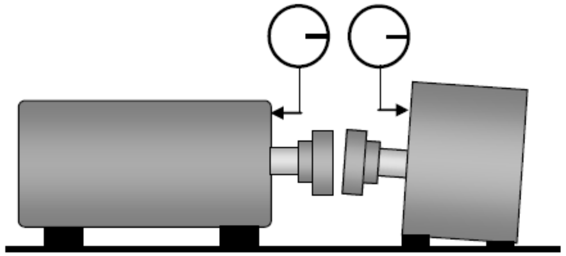


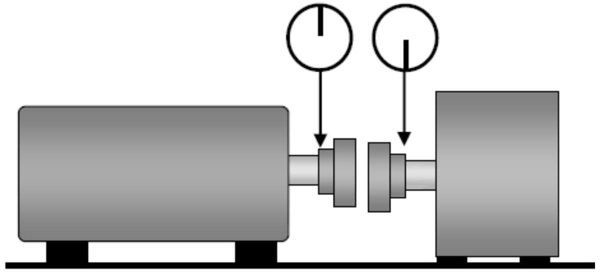


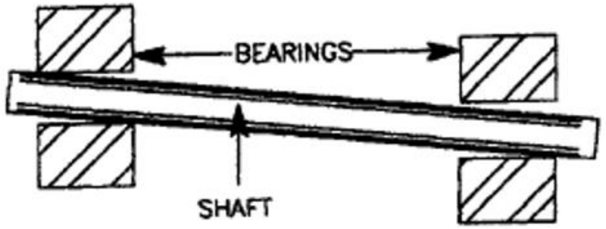


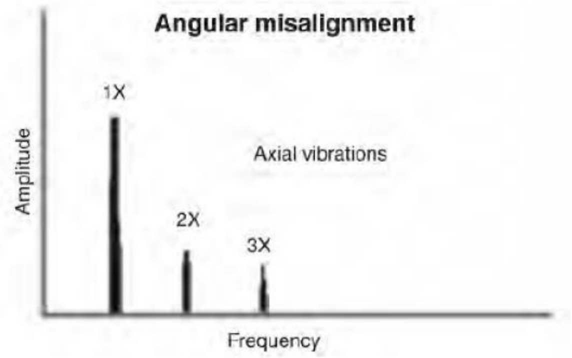
振動頻譜分析 – 軸不對中

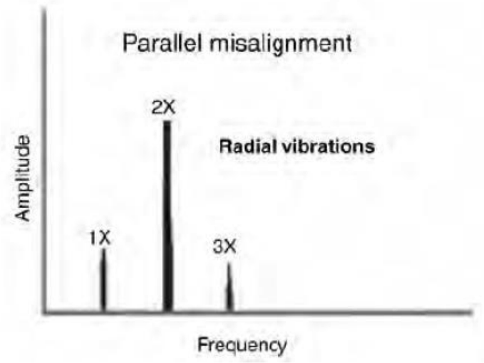
- 軸不對中（misalignment）是指兩相臨旋轉軸的軸心線因各種因素影響而產生不共線（non-collinear）的現象。











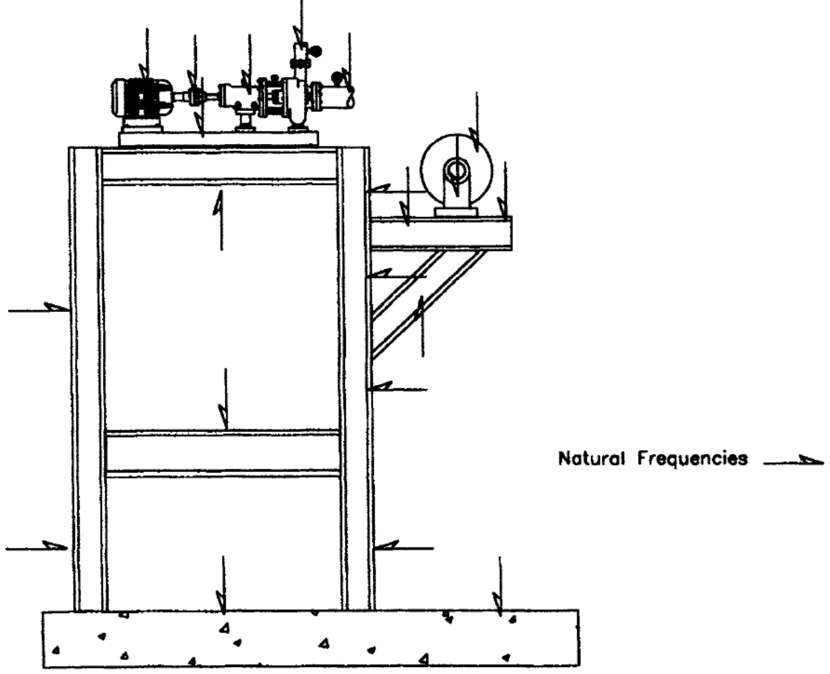
角度不對中使聯軸器附加一彎矩（bending moment），故也造成強大的軸向振動。
角偏移會主要在軸向產生1倍頻訊號，徑向也看得到。

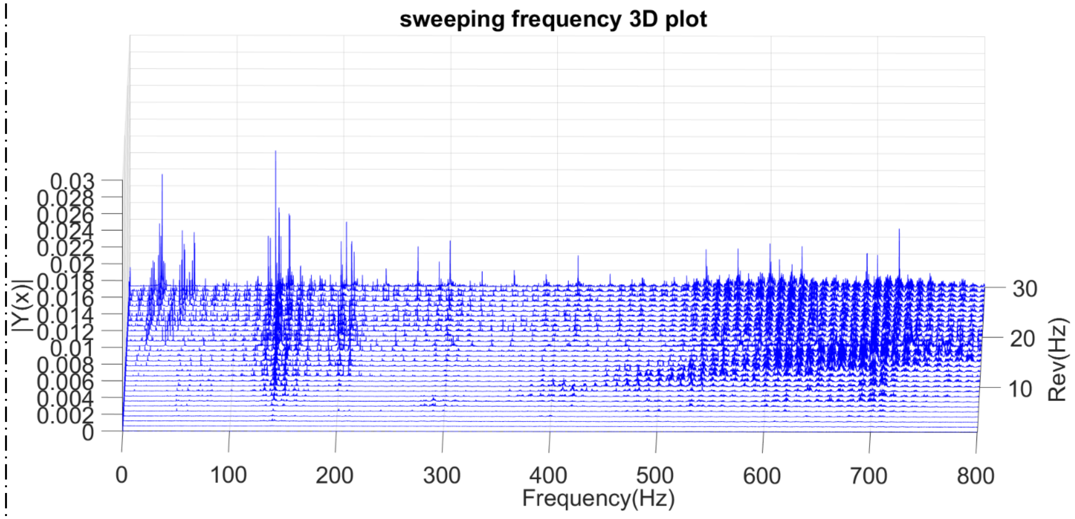
平行不對中與角度不對中也可能同時產生，且多數情況下，二倍頻（2x）能量會超過一倍頻（1x）能量。

振動頻譜分析 – 共振

- 共振（Resonance）是指一物體受到強迫振動（Forced Vibration）的頻率和其自然頻率（Natural Frequency）相等時，所產生的現象。
- 共振又可分為靜態共振與動態共振。

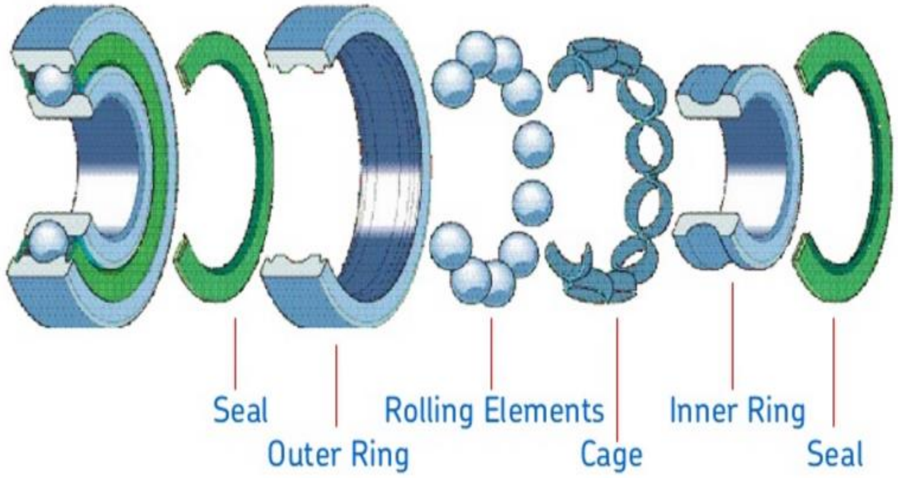
靜態共振是指當一個穩態（Stationary）的結構之自然頻率的能量被激發，將產生共振。
動態共振是指當旋轉機械裡有關旋轉的、動態的（Dynamic）結構，如在風扇內的轉子部件之自然頻率被激發，則旋轉元件產生的共振。其特徵頻率會發生在轉速的一倍頻上（1x），或是其諧波（1x、2x、3x...nx），所以在分析故障特徵訊號時，動態共振通常會與軸不對中、鬆脫現象及不平衡的特徵頻率混在一起，造成不正確的診斷。



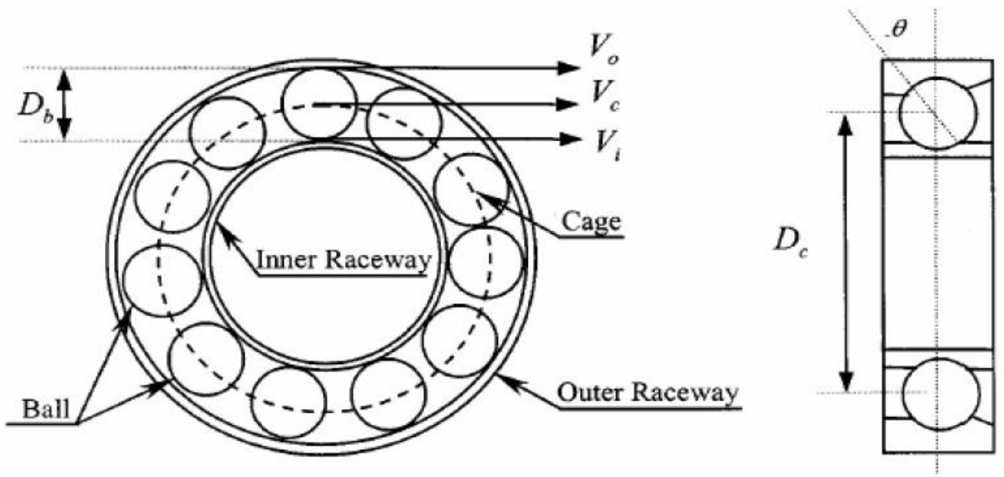


振動頻譜分析 – 軸承

- 當旋轉機構發生異常振動時，有一大部分的原因是由於軸承的損壞造成。
- 軸承損壞形式可經由其幾何尺寸和旋轉速度推導出損壞頻率。



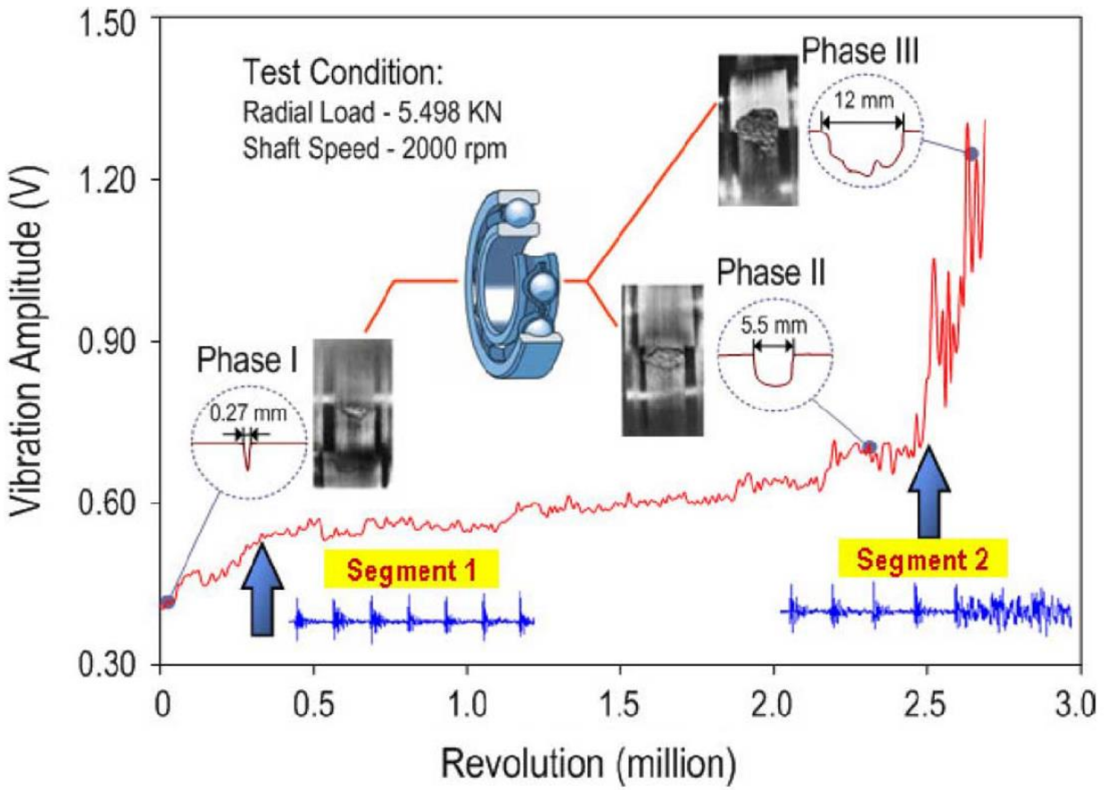
http://www.slideshare.net/NaushadAhamed/bearing-basics-skf



LI, Bo, et al. Neural-network-based motor rolling bearing fault diagnosis. IEEE transactions on industrial electronics, 2000, 47.5: 1060-1069.

振動頻譜分析 – 軸承

- 為何要探討軸承損壞頻率及初期損壞?



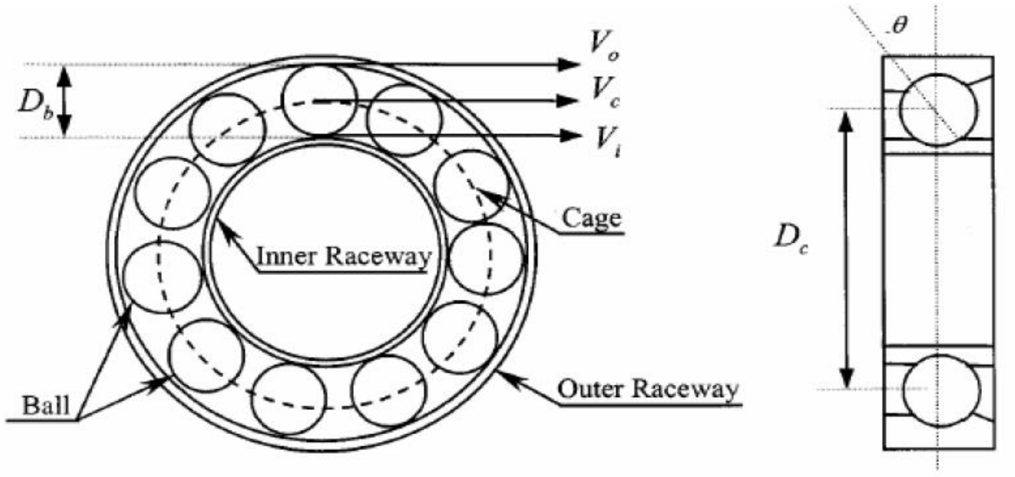
旋轉機械軸承振動趨勢圖

YAN, Ruqiang; GAO, Robert X. Hilbert–Huang transform-based vibration signal analysis for machine health monitoring. *IEEE Transactions on instrumentation and measurement*, 2006, 55.6: 2320-2329.

振動頻譜分析 – 軸承

- 保持器旋轉頻率F_C（Fundamental Cage Frequency）
  - $F_c = \frac{V_c}{r_c} = \frac{V_i+V_o}{D_c}$
  - $= \frac{V_i+V_o}{D_c} = \frac{F_i r_i+F_o r_o}{D_c}$
  - $= \frac{1}{D_c}\left(F_i \frac{D_c-D_b\cos\theta}{2} + F_o \frac{D_c+D_d\cos\theta}{2}\right)$
  - $= \frac{1}{2}F_r\left(1-\frac{D_b\cos\theta}{D_c}\right)$

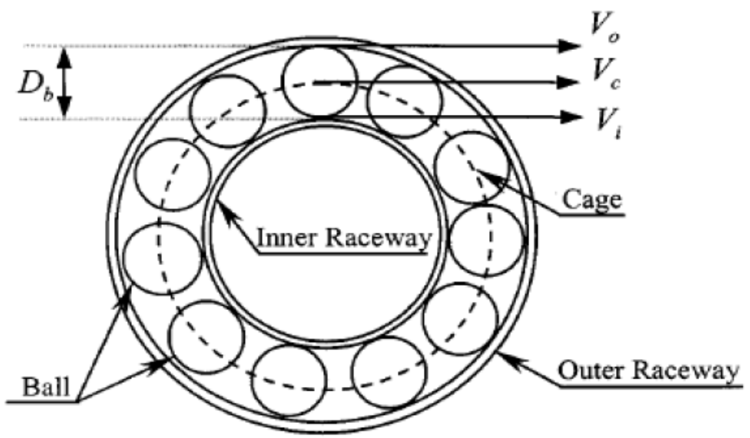
n=馬達旋轉軸的轉速(rpm=轉/min)
$F_r = F_i = \frac{n}{60}(Hz)$
內環半徑$r_i = r_c - (D_b\cos\theta/2)$
外環半徑$r_o = r_c + (D_b\cos\theta/2)$

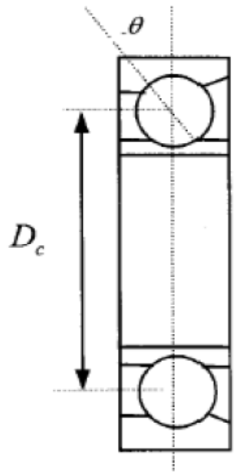


振動頻譜分析 – 軸承

- 內環損壞頻率F_{BPI}（Ball pass inner raceway frequency）
  - $F_{BPI} = N_B|F_c - F_i|$
  - $= N_B\left|\frac{F_i r_i + F_o r_o}{D_c} - F_i\right|$
  - $= N_B\left|\frac{F_i\left(r_c - \frac{D_b\cos\theta}{2}\right) + F_o\left(r_c + \frac{D_b\cos\theta}{2}\right)}{D_c} - F_i\right|$
  - $= \frac{N_B}{2}\left|(F_i - F_o)\left(1 + \frac{D_b\cos\theta}{D_c}\right)\right|$
  - $= \frac{N_B}{2}F_r\left(1 + \frac{D_b\cos\theta}{D_c}\right)$

n=馬達旋轉軸的轉速(rpm=轉/min)
$F_r = F_i = \frac{n}{60}(Hz)$
內環半徑$r_i = r_c - (D_b\cos\theta/2)$
外環半徑$r_o = r_c + (D_b\cos\theta/2)$

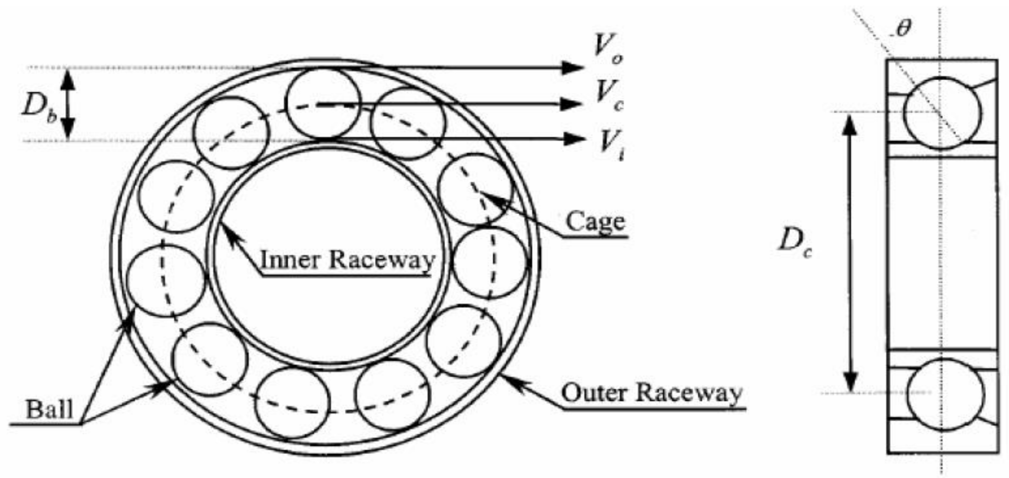




振動頻譜分析 – 軸承

- 外環損壞頻率F_{BPO}（Ball pass outer raceway frequency）
  - $F_{BPO} = N_B|F_c - F_o|$
  - $= N_B\left|\frac{F_i r_i + F_o r_o}{D_c} - F_o\right|$
  - $= N_B\left|\frac{F_i\left(r_c - \frac{D_b\cos\theta}{2}\right) + F_o\left(r_c + \frac{D_b\cos\theta}{2}\right)}{D_c} - F_o\right|$
  - $= \frac{N_B}{2}\left|(F_i - F_o)\left(1 - \frac{D_b\cos\theta}{D_c}\right)\right|$
  - $= \frac{N_B}{2}F_r\left(1 - \frac{D_b\cos\theta}{D_c}\right)$

n=馬達旋轉軸的轉速(rpm=轉/min)
$F_r = F_i = \frac{n}{60}(Hz)$
內環半徑$r_i = r_c - (D_b\cos\theta/2)$
外環半徑$r_o = r_c + (D_b\cos\theta/2)$



振動頻譜分析 – 軸承

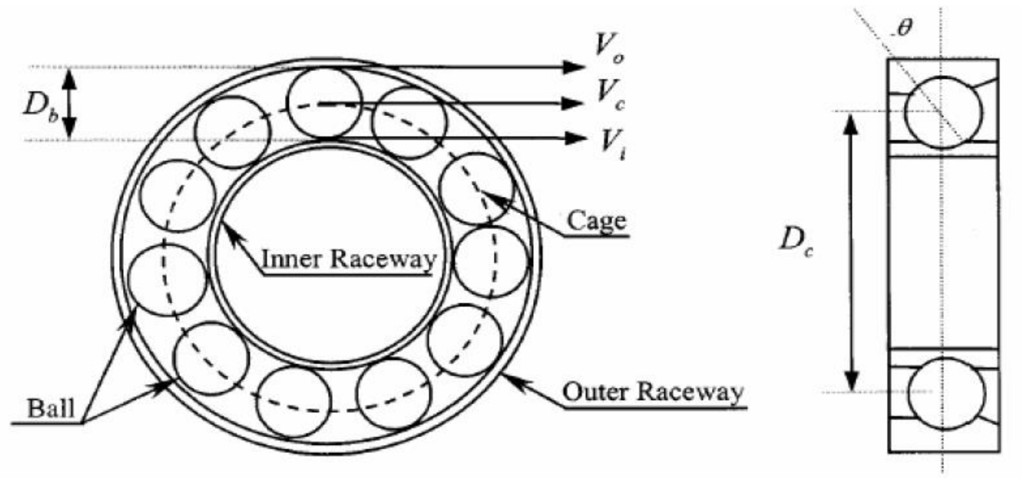
- 滾珠自旋頻率F_B（Ball Rotational Frequency）
  - $F_B = \left|(F_i - F_c)\frac{r_i}{r_b}\right| = \left|(F_o - F_c)\frac{r_o}{r_b}\right|$
  - $= \frac{D_c}{2D_b}\left|(F_i - F_o)\left(1 - \frac{D_b^2 \cos^2\theta}{D_c^2}\right)\right|$
  - $= \frac{D_c}{2D_b}F_r\left(1 - \frac{D_b^2 \cos^2\theta}{D_c^2}\right)$

n=馬達旋轉軸的轉速(rpm=轉/min)

$F_r = F_i = \frac{n}{60}(\mathrm{Hz})$

內環半徑$r_i = r_c - (D_b \cos\theta / 2)$

外環半徑$r_o = r_c + (D_b \cos\theta / 2)$



振動頻譜分析 – 軸承

ROLLING ELEMENT BEARINGS
(4 Failure Stages)

f_n = Natural Frequencies of Installed Bearing Components and Support Structure

BEARING DEFECT FREQUENCIES:

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

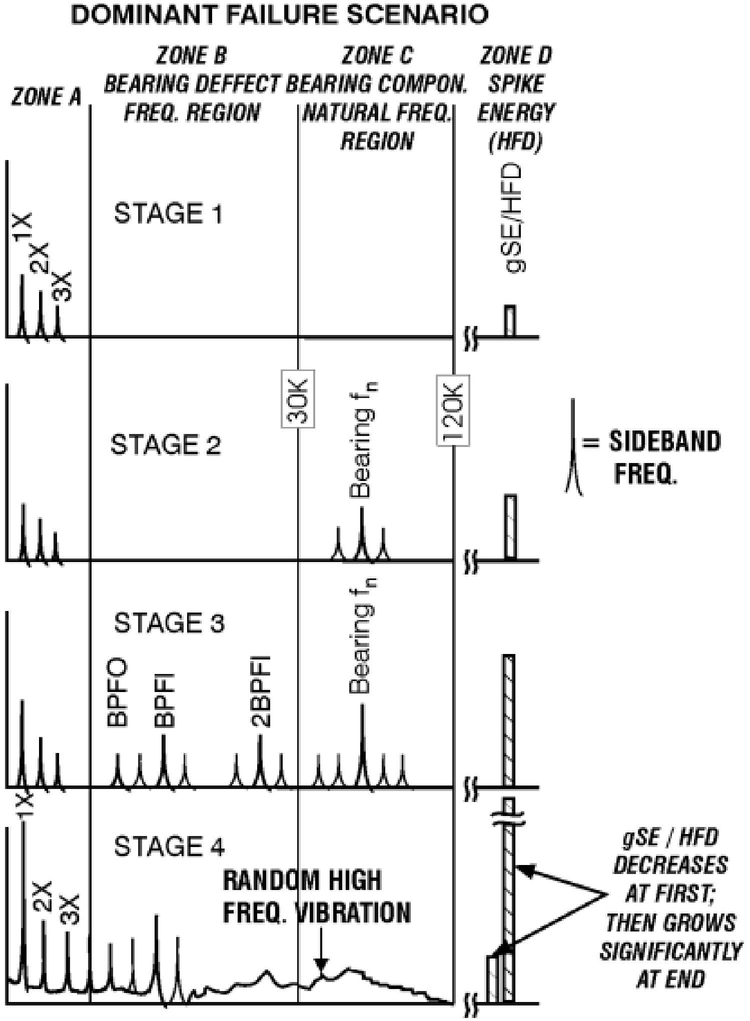
$$BPFO = \frac{N_b}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

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

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

Where:

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



4 ROLLING ELEMENT BEARING FAILURE STAGES

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

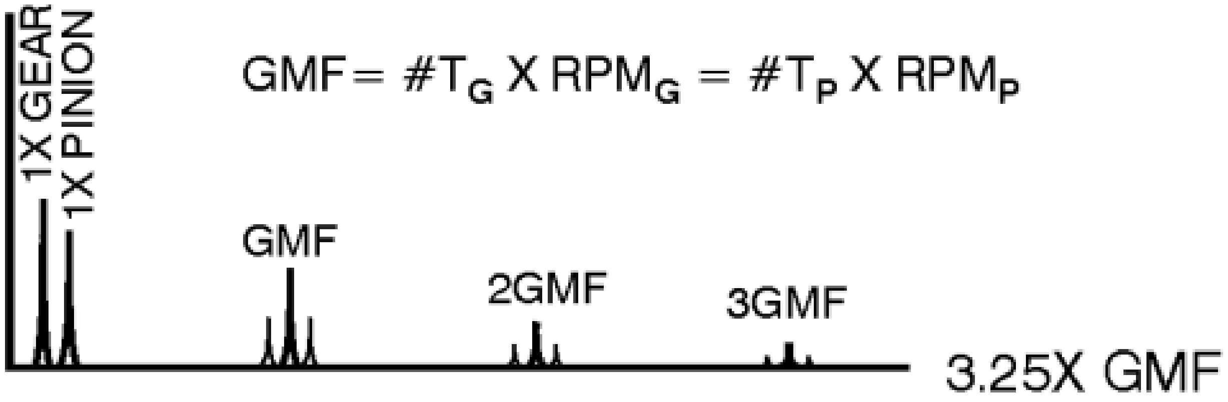
STAGE 2: Slight bearing defects begin to "ring" bearing component natural frequencies (f_n) which predominantly occur in 30K - 120K CPM range. Such natural frequencies may also be resonances of bearing support structures. Sideband frequencies appear above and below natural frequency peak at end of Stage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

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

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

振動頻譜分析 – 齒輪

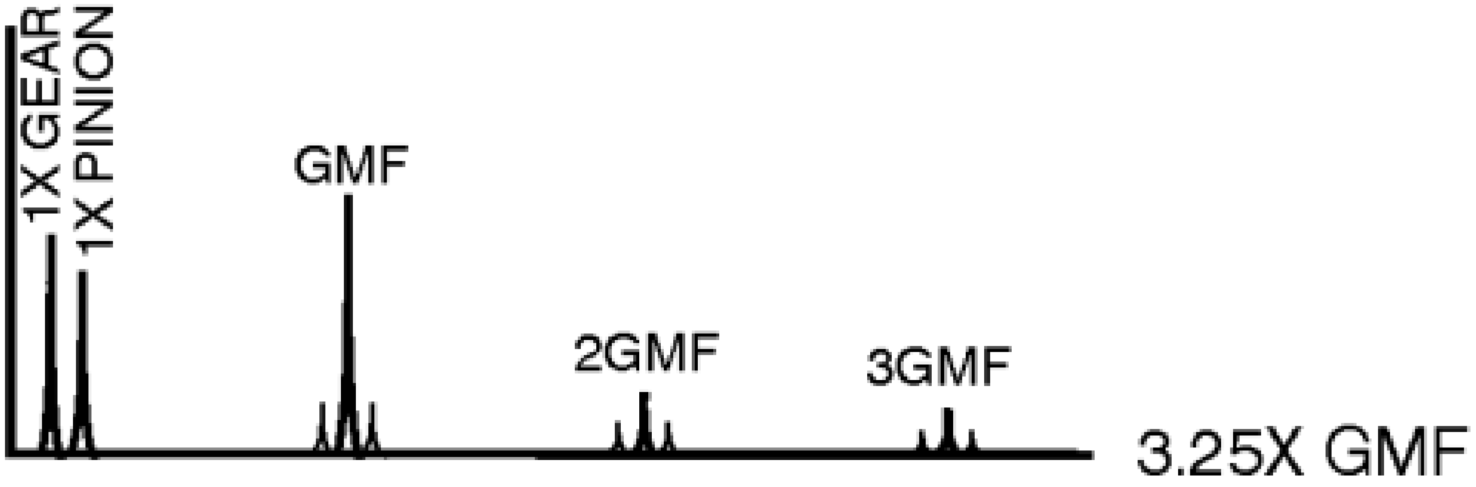
- 正常頻譜



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.

振動頻譜分析 – 齒輪

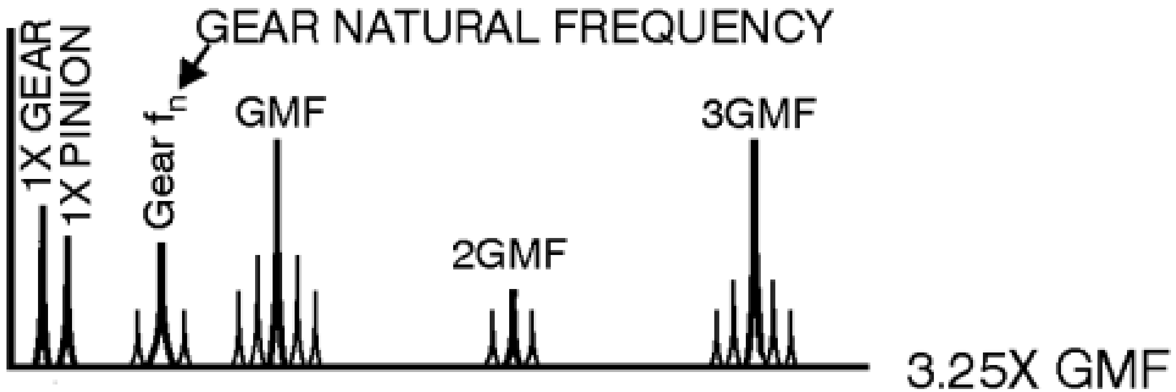
- 負載影響



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.

振動頻譜分析 – 齒輪

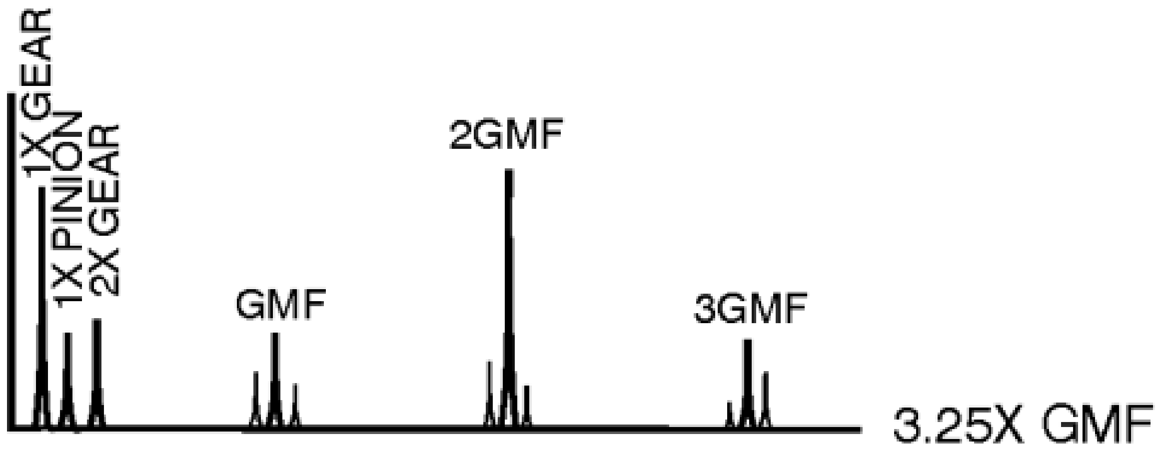
- 齒輪磨耗



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.

振動頻譜分析 – 齒輪

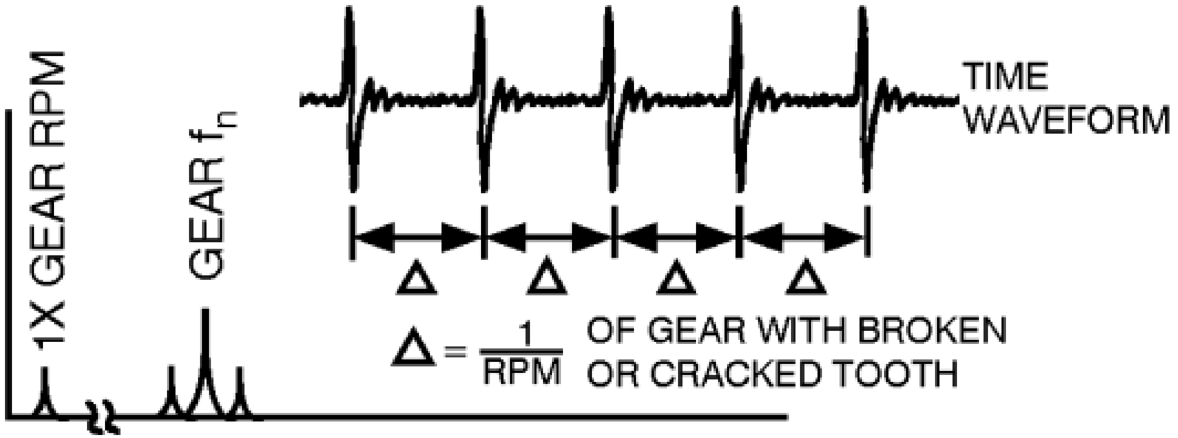
- 齒輪不對正



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.

振動頻譜分析 – 齒輪

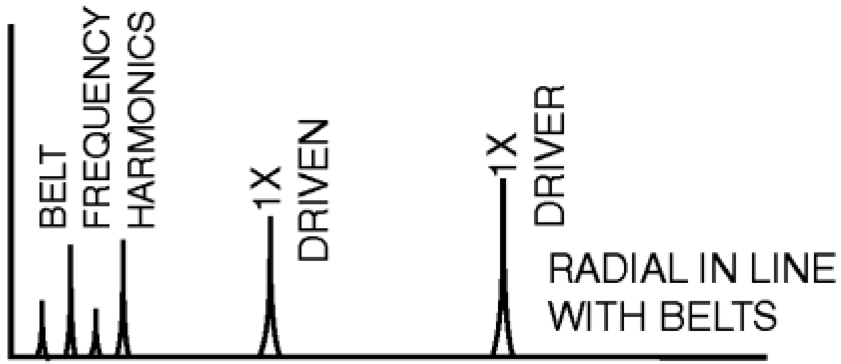
- 齒輪裂斷齒

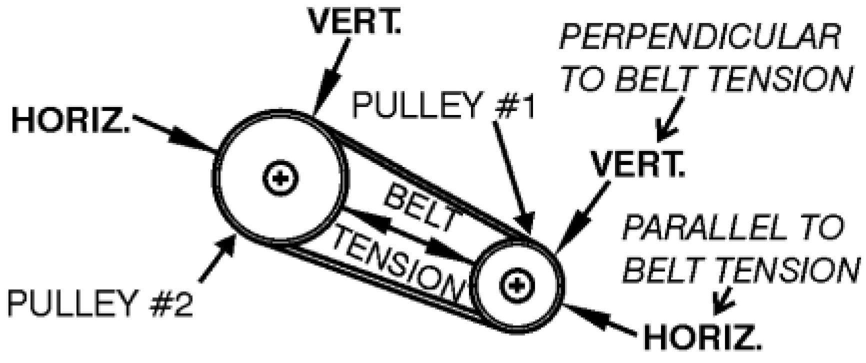


A Cracked or Broken Tooth will generate a high amplitude at 1X RPM of this gear only in the time waveform, plus it will excite gear natural frequency (f_n) sidebanded at its running speed. It is best detected in Time Waveform which will show a pronounced spike every time the problem tooth tries to mesh 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 !

振動頻譜分析 – 皮帶

- 皮帶磨耗、鬆動





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

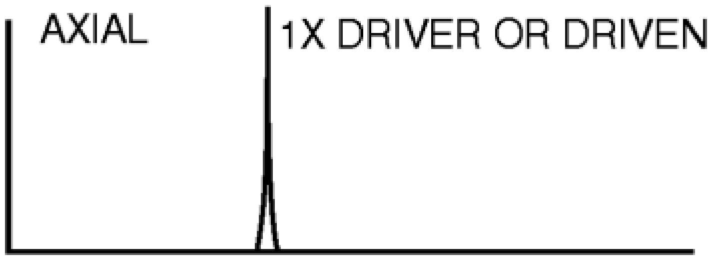
$$\text{BELT FREQ.} = \frac{3.142 \times \text{PULLEY RPM} \times \text{PITCH DIAM.}}{\text{BELT LENGTH}}$$

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

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.

振動頻譜分析 – 皮帶

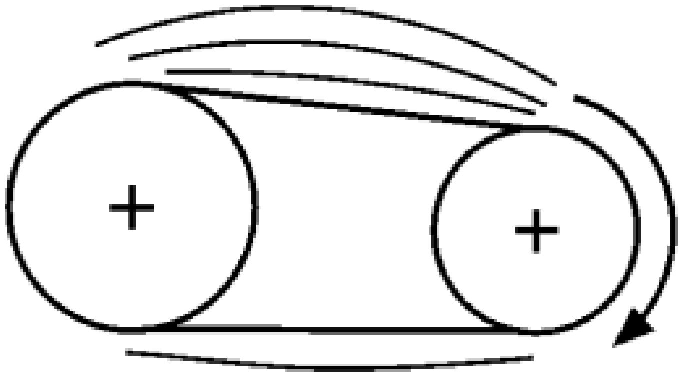
- 皮帶滑輪不對正

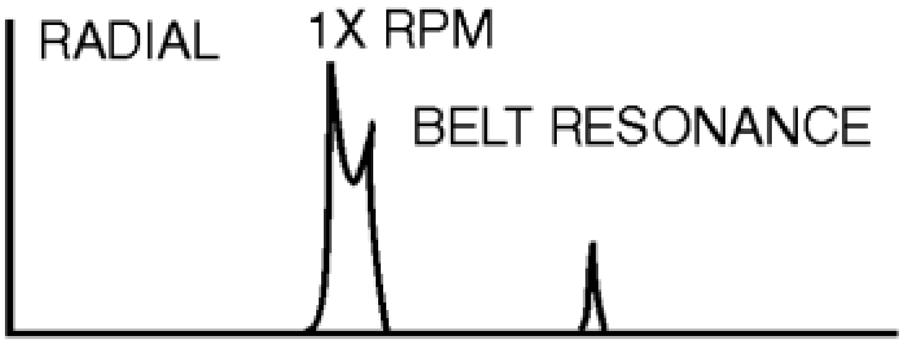


Misalignment of pulley produces high vibration at 1X RPM predominantly in the axial direction. The ratio of amplitudes of driver to driven RPM depends on where the data is taken, as well as on relative mass and frame stiffness. Often with pulley misalignment, the highest axial vibration on the motor 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.

振動頻譜分析 – 皮帶

- 皮帶共振

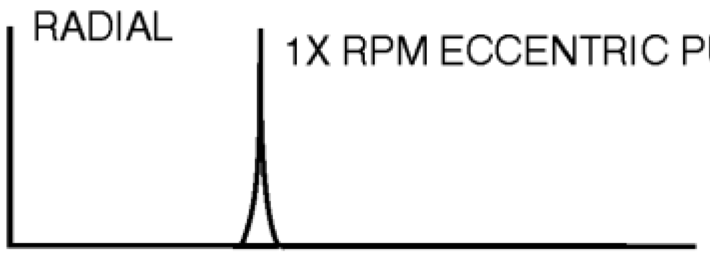


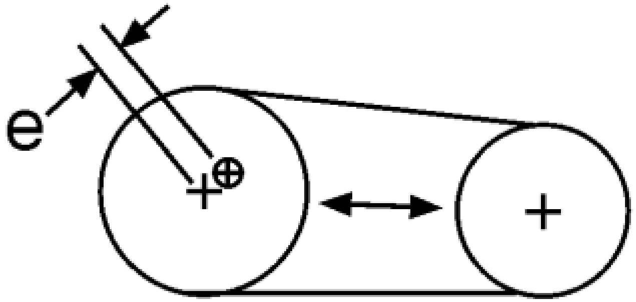


Belt Resonance can cause high amplitudes if the belt natural frequency should happen to approach, or coincide with, either the motor or driven RPM. Belt natural frequency can be altered by changing either the belt tension, belt length or cross section. Can be detected by tensioning and then releasing 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.

振動頻譜分析 – 皮帶

- 皮帶輪偏心





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°.

傳動異常電流特徵

主動輪特徵頻率

$$f_m = f_c \pm k\frac{f_c}{p}$$

從動輪特徵頻率

$$f_l = f_c \pm k\frac{f_c}{n_r \times p}$$

皮帶特徵頻率

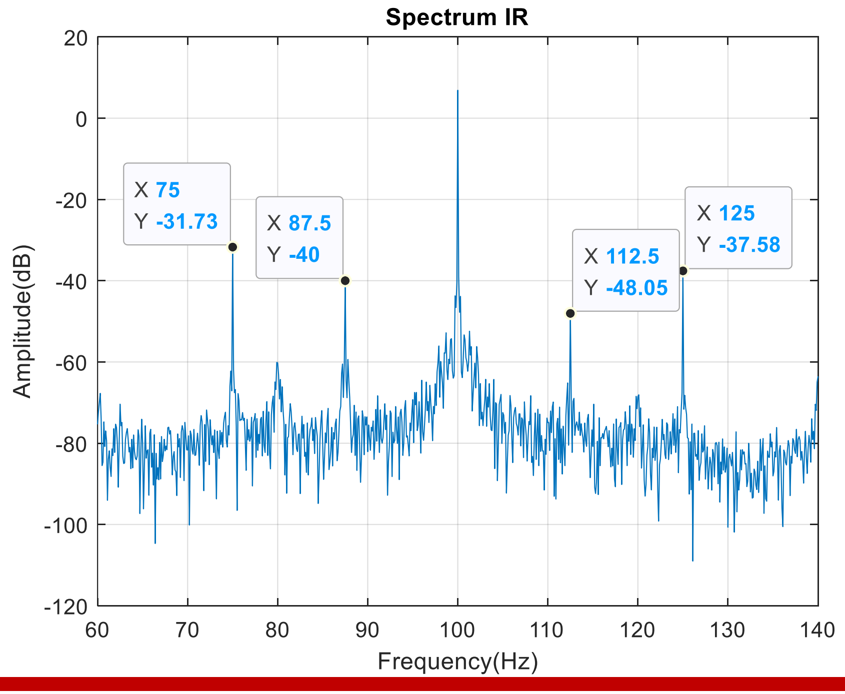
$$f_b = f_c \pm k\frac{2\pi r_m f_r}{l_b}$$

鏈條特徵頻率

$$f_{chain} = f_c \pm k\frac{2\pi r_m f_r}{l_c}$$

傳動異常電流特徵頻率表

	電流(Hz)
齒輪	75、87.5、112.5、125
皮帶	54.6、56.2、61.4、65.6、68.2、81.8、84.4、88.6、93.8、95.4
鏈輪	74.4、75、83、87.5、91.5、108.5、112.5、117、125、125.6



f_c:電壓電流之主頻
n_r:減速比
p:馬達的極對
k=1,2,3…
r_m:主動輪半徑
f_r:為主動輪旋轉頻率
l_b:為皮帶長度
l_c:為鏈條長度