The International Journal Of Engineering And Science (IJES)
||Volume|| 3 ||Issue|| 2 ||Pages|| 18 - 36 || 2014 ||
ISSN (e): 2319 – 1813 ISSN (p): 2319 – 1805

Vibration Analysis for Gearbox Casing Using Finite Element Analysis

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ABSTRACT

This paper contains the study about vibration analysis for gearbox casing using Finite Element Analysis (FEA). The aim of this paper is to apply ANSYS software to determine the natural vibration modes and forced harmonic frequency response for gearbox casing. The important elements in vibration analysis are the modelling of the bolted connections between the upper and lower casing and the modelling of the fixture to the support. This analysis is to find the natural frequency and harmonic frequency response of gearbox casing in order to prevent resonance for gearbox casing. From the result, this analysis can show the range of the frequency that is suitable for gearbox casing which can prevent maximum amplitude.

Date of Submission: 23 December 2013 Date of Acceptance: 15 February 2014

I. INTRODUCTION

Gearbox casing is the shell (metal casing) in which a train of gears is sealed. From the movement of the gear it will produce the vibration to the gearbox casing.

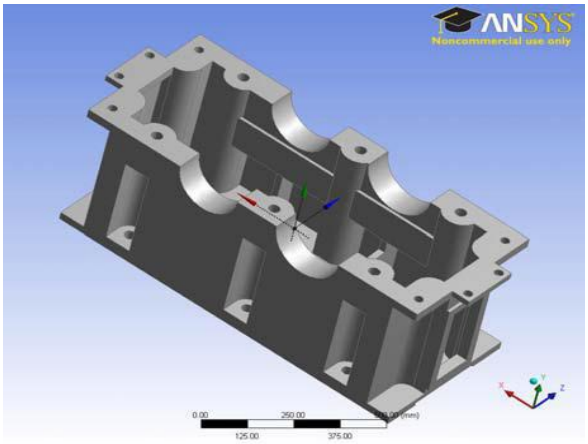

Figure 1. A gearbox casing

Reference [4] show that the study of natural frequency,consider a beam fixed at one end and having a mass attachedto the other, this would be a single degree of freedom (SDoF)oscillator. Once set into motion it will oscillate at its naturalfrequency. For a single degree of freedom oscillator, a systemin which the motion can be described by a single coordinate,the natural frequency depends on two system properties; massand stiffness. The circular natural frequency, ωn, can be foundusing the following equation:

$$\omega_n^2 = k/m \quad (1)$$

Where:
k = stiffness of the beam
m = mass of weight
ωn= circular natural frequency (radians per second)

From the circular frequency, the natural frequency, fn, can befound by simply dividing $\omega n$by 2π. Without first finding thecircular natural frequency, the natural frequency can be founddirectly using:

$$f_n = (1/2\pi)(k/m)^{1/2} \quad (2)$$

Where:
fn= natural frequency in hertz (1/seconds)
k = stiffness of the beam (Newton/Meters or N/m)
m = mass of weight (kg)

For the forced harmonic frequency, the behaviour of thespring mass damper model need to add a harmonic force in theform below. A force of this type could, for example, begenerated by a rotating imbalance.

$$F = F_0 \cos(2\pi f t). \quad (3)$$

Then, the sum the forces on the mass are calculate usingfollowing ordinary differential equation:

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos(2\pi f t). \quad (4)$$

The steady state solution of this problem can be written as:

$$x(t) = X \cos(2\pi f t - \phi). \quad (5)$$

The result states that the mass will oscillate at the samefrequency, f, of the applied force, but with a phase shift φ.The amplitude of the vibration “X” is defined by the followingformula.

$$X = \frac{F_0}{k} \frac{1}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}}. \quad (6)$$

Where “r” is defined as the ratio of the harmonic forcefrequency over the undamped natural frequency of the mass–spring–damper model.

$$r = \frac{f}{f_n}. \quad (7)$$

The phase shift , φ, is defined by following formula. the base.

$$\phi = \arctan\left(\frac{2\zeta r}{1-r^2}\right). \quad (8)$$

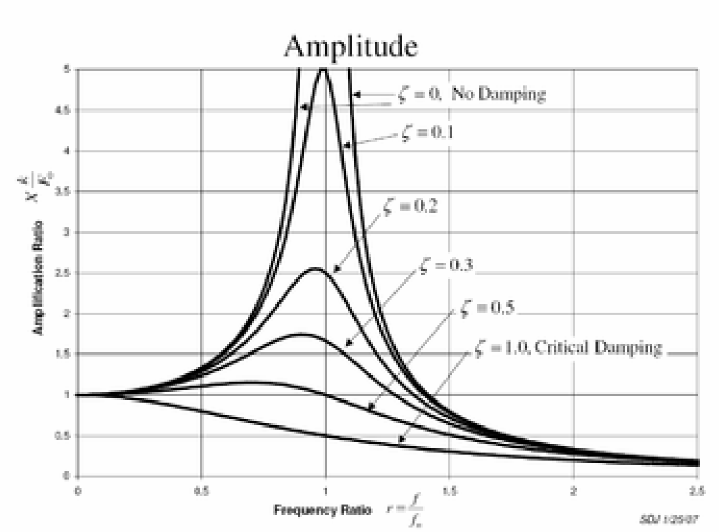

Figure 2. The frequency response of the system

The plot of these functions, called "the frequency response ofthe system", presents one of the most important features inforced vibration. In a lightly damped system when the forcingfrequency nears the natural frequency ( ) the amplitudeof the vibration can get extremely high. This phenomenon iscalled resonance (subsequently the natural frequency of asystem is often referred to as the resonant frequency). In rotorbearing systems any rotational speed that excites a resonantfrequency is referred to as a critical speed.If resonance occurs in a mechanical system it can be veryharmful – leading to eventual failure of the system.Consequently, one of the major reasons for vibration analysisis to predict when this type of resonance may occur and thento determine what steps to take to prevent it from occurring.As the amplitude plot shows, adding damping cansignificantly reduce the magnitude of the vibration. Also, themagnitude can be reduced if the natural frequency can beshifted away from the forcing frequency by changing thestiffness or mass of the system. If the system cannot bechanged, perhaps the forcing frequency can be shifted (forexample, changing the speed of the machine generating theforce).The following are some other points in regards to the forcedvibration shown in the frequency response plots.• At a given frequency ratio, the amplitude of thevibration, X, is directly proportional to the amplitudeof the force $F0$ (e.g. if double the force, the vibrationdoubles)• With little or no damping, the vibration is in phasewith the forcing frequency when the frequency ratio$r < 1$ and 180 degrees out of phase when thefrequency ratio $r > 1$• When r _ 1 the amplitude is just the deflection of thespring under the static force $F0$. This deflection iscalled the static deflection δst. Hence, when r _ 1 theeffects of the damper and the mass are minimal.• When r _ 1 the amplitude of the vibration is actuallyless than the static deflection δst. In this region theforce generated by the mass ($F = ma$) is dominatingbecause the acceleration seen by the mass increaseswith the frequency. Since the deflection seen in thespring, X, is reduced in this region, the forcetransmitted by the spring ($F = kx$) to the base isreduced. Therefore the mass–spring–damper systemis isolating the harmonic force from the mountingbase – referred to as vibration isolation. Interestingly,more damping actually reduces the effects ofvibration isolation when r _ 1 because the dampingforce ($F = cv$) is also transmitted to the base.This analysis is to find the natural frequency and harmonicfrequency response of gearbox casing in order to preventresonance for gearbox casing. From the result, this analysiscan show the range of the frequency that is suitable forgearbox casing which can prevent maximum amplitude.

II. DESIGN OF GEARBOX CASING

A. Joint Design

Equivalent bolt radius for bolts connecting gearbox halves is= 3r

When r = 16.5mm (inside radius)
=3×16.5
=49.5mm (outside radius)
When r = 13mm (inside radius)
=3×13
=39mm (outside radius)
Thickness is 1mm.

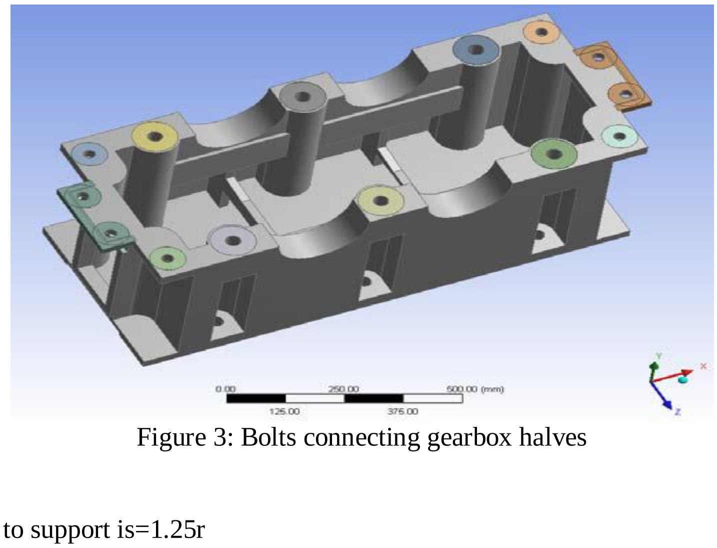

Figure 3: Bolts connecting gearbox halves

B. Supports Design

Equivalent bolt radius to support is=1.25r
When r = 16.5mm (inside radius)
=1.25×16.5
=20.625 mm (outside radius)
Thickness is1mm

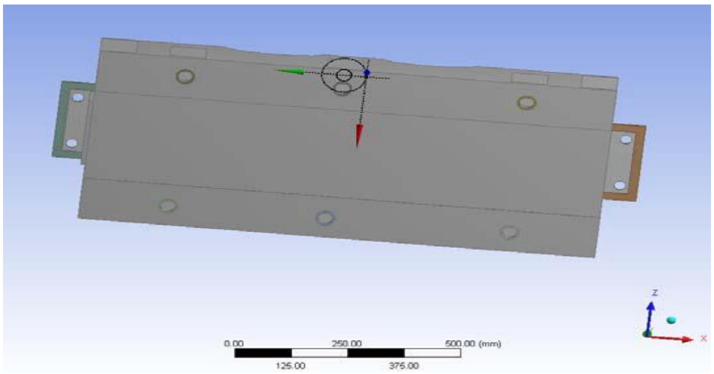

Figure 4: Bolt radius to support (bottom view of gearbox casing)

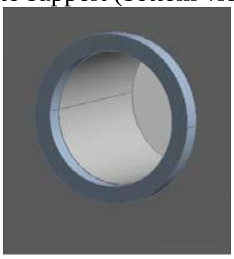

Figure 5: Details of one bolt for support

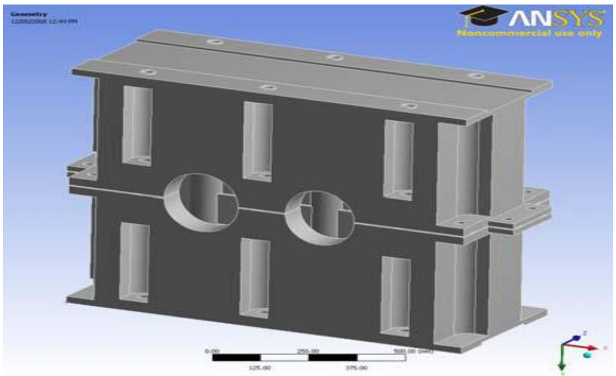

Figure 6: Full box of gearbox casing

III. MESH STRATEGY

The details of mesh strategy are defined in Table 1 andFigure 7.An appropriate esh is selected to make sure thismeshing can solve in 1 hour duration. This mesh is applied towhole object as one body meshing.

Table : Details of meshing strategy

Object Name	*Mesh*
State	Solved
Defaults	
Physics Preference	Mechanical
Relevance	0
Advanced	
Relevance Center	Coarse
Element Size	Default
Shape Checking	Standard Mechanical
Solid Element Midside Nodes	Program Controlled
Straight Sided Elements	No
Initial Size Seed	Active Assembly
Smoothing	Low
Transition	Fast
Statistics	
Nodes	71961
Elements	39946

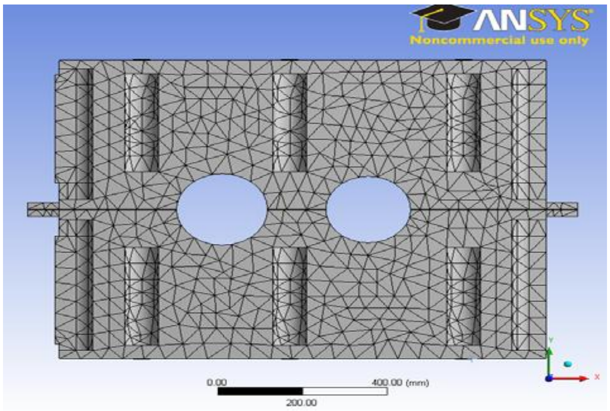

Figure 7: Actual mesh of gearbox casing

IV. BOUNDARY CONDITION AND APPLIED LOAD

This section described the details of applied load andboundary condition of natural vibrations and harmonicanalysis.

Natural Vibration Analysis

A modal analysis is performed with number of modes is10.The details of the support is in Table

Object Name	*Fixed Support*
State	Fully Defined
Scope	
Scoping Method	Geometry Selection
Geometry	6 Faces
Definition	
Type	Fixed Support
Suppressed	No

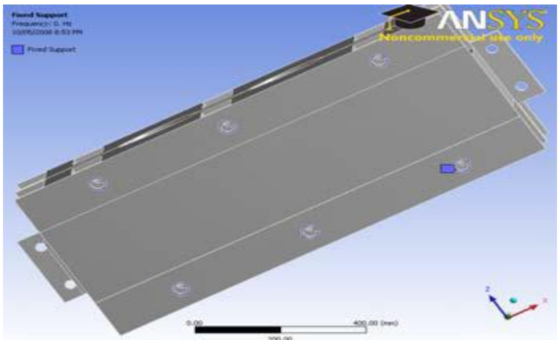

Figure 8: Actual fixed support on bottom created circle surface

Harmonic Frequency Response Analysis

In the harmonic frequency response analysis, the fixedsupport is exactly same condition in Figure 8.In this analysis, 1MPa pressures is applied to the upper halfof the bearings on one side of the gearbox and to the lowerhalf of the other side for a frequency range from zero to 1.2times the frequency of the tenth vibration mode.This 1MPa pressure is applied normal to the surfaceaccording to the Table and Figure 9.

Object Name	*Fixed Support*	*Pressure*	*Pressure 2*	*Pressure 3*	*Pressure 4*
State	Fully Defined				
Scope					
Scoping Method	Geometry Selection				
Geometry	6 Faces	1 Face			
Definition					
Type	Fixed Support	Pressure			
Suppressed	No				
Define By		Normal To			
Magnitude		1. MPa			
Phase Angle		0. °			

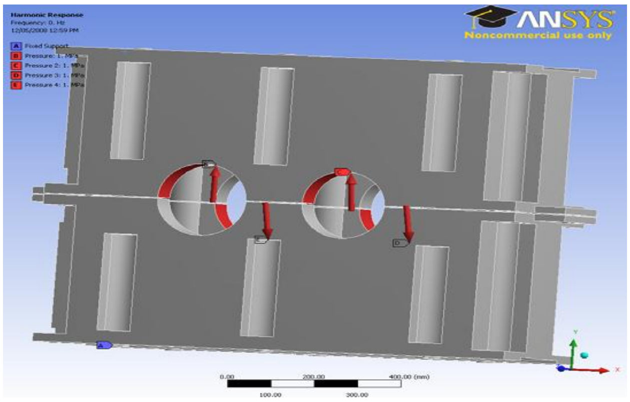

Figure 9: The actual applied load in gearbox casing.

V. RESULT

These results for natural vibration analysis and harmonicfrequency response analysis is done using ANSYS 11.0

Result of Natural Vibration Analysis

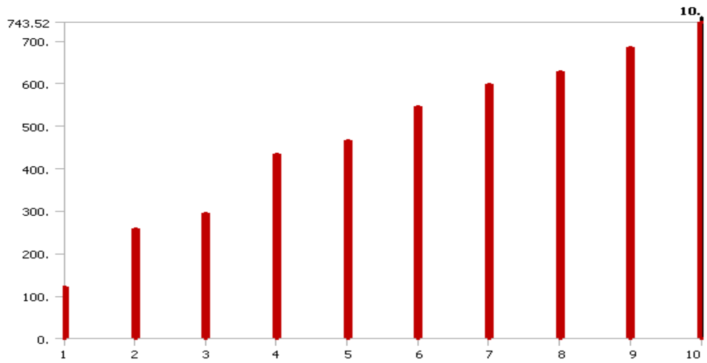

Figure 10: Result of frequency corresponding to 10 modes for normal Vibration analysis.

From these result, 10 lowest vibration frequencies are:

Table 4:10 lowest frequencies for natural vibration analysis

Mode	Frequency [Hz]
1.	120.93
2.	256.71
3.	295.27
4.	434.45
5.	464.22
6.	545.23
7.	598.62
8.	627.11
9.	683.95
10.	743.52

Result of Harmonic Frequency Response Analysis

In this harmonic frequency response analysis, frequencyrange need to be set up from zero to 1.2 times the frequency ofthe tenth vibration mode.In Table 4, tenth vibration mode is743.52 Hz.

1.2×the frequency of the 10 *th*vibration mode

= 1.2× 743.52

= 892.224 Hz

From this result, 0-892 Hz frequency range is applied.

Table 5: Applied frequency in Harmonic Frequency Response Analysis

Object Name	*Analysis Settings*
State	Fully Defined
Options	
Range Minimum	0. Hz
Range Maximum	892. Hz
Solution Intervals	200

All the result is from one vertex as in the Table 5.This point isselected because this point is the maximum total displacementin the Figure 11.

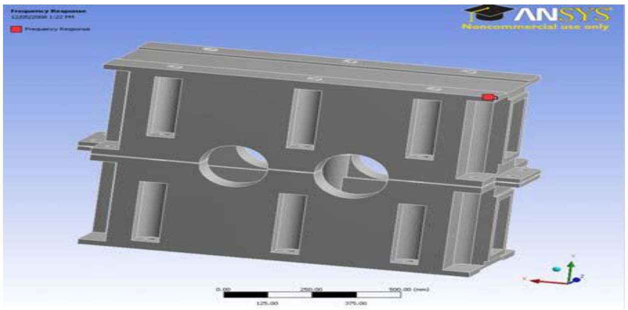

Figure 11: Analysis point

Result of Harmonic Frequency Response Analysis
Y-axis result.

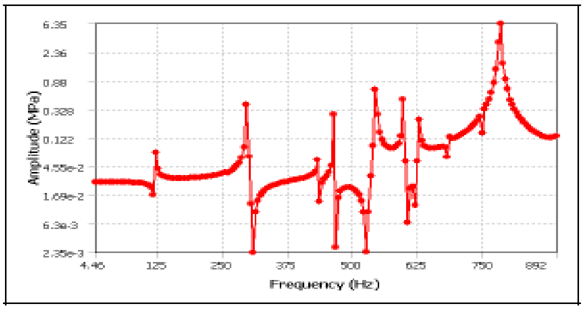

Figure 12: Details of Y-axis result for normal stress

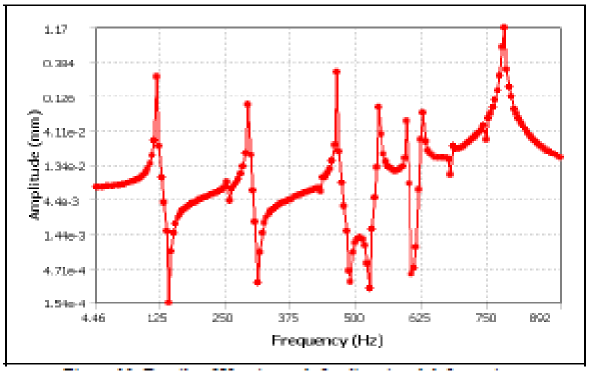

Figure 13: Details of Y-axis result for directional deformation.

X-axis result.

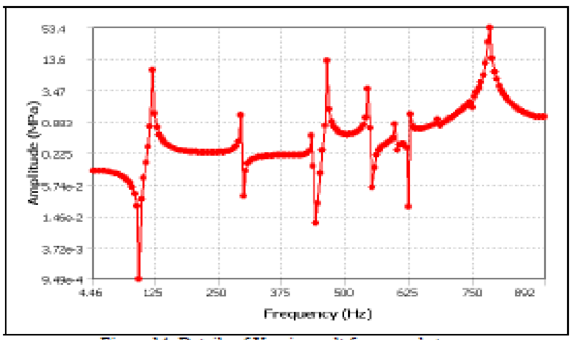

Figure 14: Details of X-axis result for normal stress

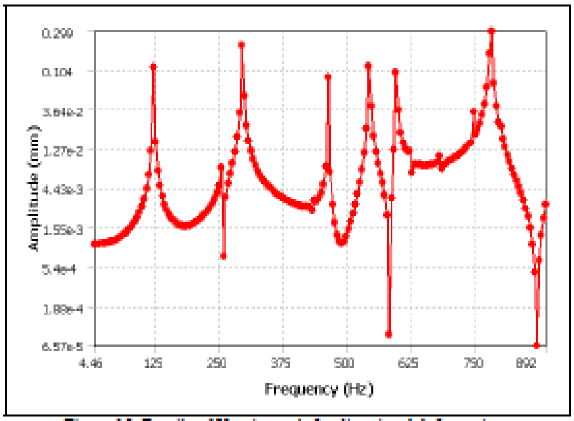

Figure 15: Details of X-axis result for directional deformation

VI. FAULT DETECTION AND DIAGNOSIS FROM VIBRATION ANALYSIS

Diagnostics is understood as identification of a machine's condition/faults on the basis of symptoms. Diagnosis requires a skill in identifying machine's condition from symptoms. The term diagnosis isunderstood here similarly as in medicine. It isgenerally thought that vibration is a symptom of agearbox condition. Vibration generated by gearboxesis complicated in its structure but gives a lot ofinformation. We may say that vibration is a signal ofa gearbox condition. To understand informationcarried by vibration one have to be conscious/ awareof a relation between factors having influence tovibration and a vibration signal. In order to detect(and diagnosis) an impending failure, a goodunderstanding of the evidence relating to the failuremode and methods of collecting and quantifying theevidence is needed. Although many faults may beeasily detectable by physical examination of a component, using techniques such as microscopy, Xray,dye penetrates, magnetic rubber, etc., thesemethods usually cannot be performed withoutremoval of, and in some cases physical damage to,the component. Whilst physical examinationtechniques still play a critical role duringmanufacture, assembly and overhaul, they areimpractical in an operational large transmissionsystem and other (non-intrusive) fault detectionmethods need to be employed for routine monitoringpurposes. Most modern techniques for geardiagnostics are based on the analysis of vibrationsignals picked up from the gearbox casing. Thecommon target is to detect the presence and the typeof fault at an early stage of development and tomonitor its evolution, in order to estimate themachine's residual life and choose an adequate planof maintenance. It is well known that the mostimportant components in gear vibration spectra arethe gear meshing frequency (GMF) and itsharmonics, together with sidebands due tomodulation phenomena. The increment in the numberand amplitude of such sidebands may indicate a faultcondition. Moreover, the spacing of the sidebands isrelated to their source. source identification and faultdetection from vibration signals associated with itemswhich involve rotational motion such as gears, rotorsand shafts, rolling element bearings, journal bearings, flexible couplings, and electrical machines dependupon several factors:

- the rotational speed of the items,
- the background noise and/or vibration level,
- the location of the monitoring transducer,
- the load sharing characteristics of the item, and
- the dynamic interaction between the item and

Other items in contact with it. The main causes ofmechanical vibration are unbalance, misalignment,looseness and distortion, defective bearings, gearingand coupling in accuracies, critical speeds, various form of resonance, bad drive belts, reciprocatingforces, aerodynamic or hydrodynamic forces, oilwhirl, friction whirl, rotor/stator misalignments, bentrotor shafts, defective rotor bars, and so on. Some of the most common faults that can be detected using vibration analysis are summarized in Table 1

Table some typical faults and defects that can be detected with vibration analysis

Item	Fault
Gears	Tooth messing faults, misalignment, cracked and/or worm teeth, eccentric gear
Rotors and shaft	Unbalance Bent shaft Misalignment Eccentric journals Loose components Rubs Critical speed Cracked shaft Blade loss Blade resonance
Rolling element bearings	Pitting of race and ball/roller Spalling Other rolling elements defect
Flexible coupling	Misalignment Unbalance
Electrical machines	Unbalanced magnetic pulls Broken/damaged rotor bars Air gap geometry variations Structural and foundation faults Structural resonance Piping resonance

Ebersbach et al, (2005) [1], has investigated theeffectiveness of combining both vibration analysisand wear debris analysis is an integrated machinecondition monitoring maintenance program. DeckerHarry. J (2002) [2], has proposed two new detectiontechniques. The time synchronous averaging conceptwas extended from revolution-based to toothengagement based. The detection techniques arebased on statistical comparisons among the averages for the individual teeth. These techniques wereapplied to a series of three seeded fault crack propagation tests. Polyshchuk V.V et al (2002) [3],has presents the development of a novel method ingear damage detection using a new gear faultdetection parameter based on the energy change inthe joint time-frequency analysis of the vibrationanalysis of the vibration signal. Choy F. K et al,(2003) [4], demonstrates the use of vibrationsignature analysis procedures for health monitoringand diagnostics of a gear transmission system. Lin J. and Zuo M. J (2003) [5], has introduced an adaptivewavelet filter based on Morlet Wavelet, theparameters in the Morlet wavelet function areoptimized based on the kurtosis maximizationprinciple. The wavelet used is adaptive because theparameters are not fixed. The adaptive wavelet filteris found to be very effective in detection ofsymptoms from vibration signals of a gearbox withearly fatigue tooth crack.

VII. GEARBOX FAILURE AND ITS VIBRATION ANALYSIS TECHNIQUES

The principle causes for gear failure are given here -a) An error of design, b) An application error, c) It is likely that there is a manufacturing error. Designerrors may be due to causes like improper geargeometry, use of wrong materials, quality, lubricationand other confiscations. Application errors can bedue to problems like vibration, mounting andinstallation, cooling and maintenance whilemanufacturing errors can be in the form of mistakesin machining or problems in heat treating. Summaryof safety critical failure modes (Table), severalresearchers worked on the subject of gearbox defectdetection and diagnosis through vibration analysis.Time domain, frequency domain, time frequencydomain based on short time Fourier transform(STFT) and wavelet transform and advanced signalprocessing techniques have been implemented andtested.

Table Safety critical failure modes

Failure	*Failure Mode*	*Cause*	*Contributing factors*
Shaft fracture	Fatigue	Unbalance	
		Misalignment	Coupling
			Bearing failure
		Bent shaft	
	Overload	Interference	Incorrect assembly
			Bearing failure
		Operational	
Gear fracture	Fatigue	Life limit exceeded	
		Surface damage	
	Resonance	Design	
Tooth fracture	Bending fatigue	Life limit exceeded	
		Surface damage	Process related
		Thin tooth	Excessive wear
			Destructive scoring
	Random fracture	Surface damage	Process related
			Foreign object
			Pitting/Spalling
	Overload	Interference	Incorrect assembly
			Bearing failure
		Operational	
Over-heating	Lubrication	Insufficient oil	
		Loss of oil	Oil line failure
			Filter bowl failure
	Insufficient cooling	Cooling fan failure	Shaft/gear fracture

Time Domain Analysis:

The time domainmethods try to analyze the amplitude and phaseinformation of the vibration time signal to detect thefault of gear-rotor-bearing system. The time domainis a perceptive that feels natural, and providesphysical insight into the vibration [6]. It isparticularly useful in analyzing impulsive signalsfrom bearing and gear defects with non-steady andshort transient impulses [7].3.1.1 Time Waveform Analysis: Prior to thecommercial availability of spectral analyzers, almostall vibration analysis was performed in the timedomain. By studying the time domain waveformusing equipment such as oscilloscopes, oscillographs,or 'vibrographs', it was often possible to detectchanges in the vibration signature caused by faults.However, diagnosis of faults was a difficult task; relatinga change to a particular component requiredthe manual calculation of the repetition frequencybased on the time difference observed betweenfeature points. Waveform analysis can also be usefulin identify vibrations that are non-synchronous withshaft speed. In machine cost down analysis waveformcan indicate the occurrence of resonance. A typical vibration waveform is shown in figure-3.1 for agearbox. This waveform shows the anomalousbehavior of the gear after certain intervals with largemagnitude. The peak level, RMS, level, and the crestfactor are often used to quantify the time signal.

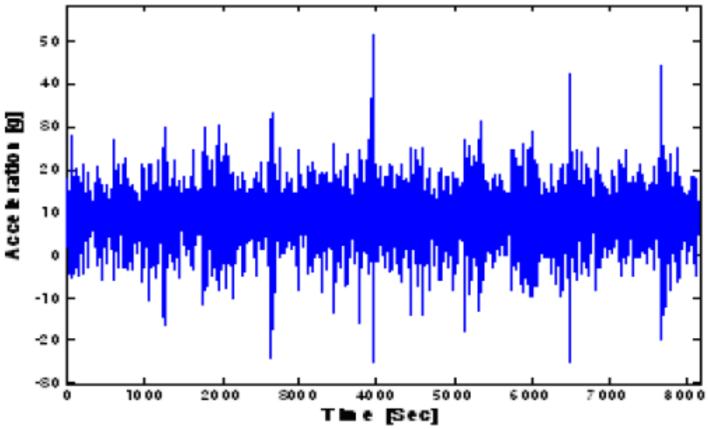

Figure A typical waveform of defected gear vibration signal.

Indices:

Indices have also been used invibration analysis [8, 6]. The peak value, RMS leveland their ratio crest factor are often used to quantifythe time signal. The peak level is not a statistical
Quantity and hence may not be reliable in detectingdamage continuously operating systems. The RMSvalue, however, is more-satisfactory for steady-stateapplications. The crest factor, defined as the ratio ofthe peak value to RMS level, has been proposed as atrending parameter as it includes both parameters.Crest factors are reliable only in the presence of Significant impulsiveness.

Peak: The peak level of the signal is defined simply as half the difference between the maximum and minimum vibration levels:

peak = Max(A)

RMS: The RMS (Root Mean Square) value of the signal is the normalized second statistical moment of the signal (standard deviation):

$$RMS = \sqrt{\frac{\sum_{n=1}^{N}[A(n)]^2}{N}}.$$

Where, A(n) is the amplitude of the nth digitized point in the time domain, and N is the number of point in time domain. The RMS of the signal is commonly used to describe
the ‘steady-state’ or ‘continuous’ amplitude of a time varying signal.

Crest Factor: The crest factor is defined as the ratio of the peak value to the RMS of the signal:

$$Crest\ Factor = \frac{Peak\ Level}{RMS\ Level}..$$

Statistical Methods: Statistical analysis canalso be carried out on time domain data.Kurtosis: Kurtosis is the normalized fourth statisticalmoment of the signal [8]. For continuous time signalsthis is defined as:

$$K = \frac{\sum_{n=1}^{N}[y(n)-\mu]^4}{N \times \left(\sigma^2\right)^2}..$$

Where y(n) is the data; n = 1, 2, 3,…..N; N is the totalnumber of data samples, μ is the mean; and _ is thestandard deviation.

The kurtosis level of a signal is used in a similarfashion to the crest factor that is to provide a measureof the impulsive nature of the signal. Raising thesignal to the fourth power effectively amplifiesisolated peaks in the signal.Skewness: Skewness is a measure of symmetry, ormore precisely, the lack of symmetry. A distributionof data set is symmetric if it looks the same to the left

Skewness:

Skewness is a measure of symmetry, ormore precisely, the lack of symmetry. A distribution of data set is symmetric if it looks the same to the leftand right of the centre point. Equation (3.5) is used tocalculate the values of skewers

$$S - \frac{\sum_{n=1}^{N} [y(n) \quad \mu]^3}{N \times (\sigma)^3}$$

Frequency Domain Analysis:

The frequencydomain methods include Fast Fourier Transform(FFT), Hilbert Transform Method and PowerCepstrum Analysis, etc. They are using the differenceof power spectral density of the signal due to the faultof gear and/or bearing to identify the damage ofelements [8]. Any real world signal can be brokendown into a combination of unique sine waves. Everysine wave separated from the signal appears as avertical line in the frequency domain. Its heightrepresents its amplitude and its position representsthe frequency. The frequency domain representationof the signal is called the signal. The frequencydomain completely defines the vibration. Frequencydomain analysis not only detects the faults in rotating Machinery but also indicates the cause of the defect[6].Theoretically, time domain can be converted intofrequency domain using the Fourier Transforms andvice versa. The Fourier transform is a generalizationof the complex Fourier series in the limit as LLM.Replace the discrete An with the continuous F(k)dkwhile letting n/L_k. Then change the sum to an integral, and the equations become

$$f(x) = \int_{-\infty}^{\infty} F(k) e^{2\pi ikx} dk$$

$$F(k) = \int_{-\infty}^{\infty} f(x) e^{-2\pi ikx} dx$$

Here, equation (3.7) is called forward (-i) FourierTransform and the (3.6) is called the inverse FourierTransform.

Fast Fourier Transformation:

The FastFourier Transform (FFT) is simply a class of specialalgorithms which implement the discrete Fouriertransform with considerable savings in computationaltime. It must be pointed out that the FFT is not adifferent transform from the DFT, but rather just ameans of computing the DFT with a considerablereduction in the number of calculations required. TheFast Fourier transform (FFT) is a discrete Fouriertransform algorithm which reduces the number ofcomputations needed for N points from 2N2 to 2NlgN, where log is the base-2 logarithm.

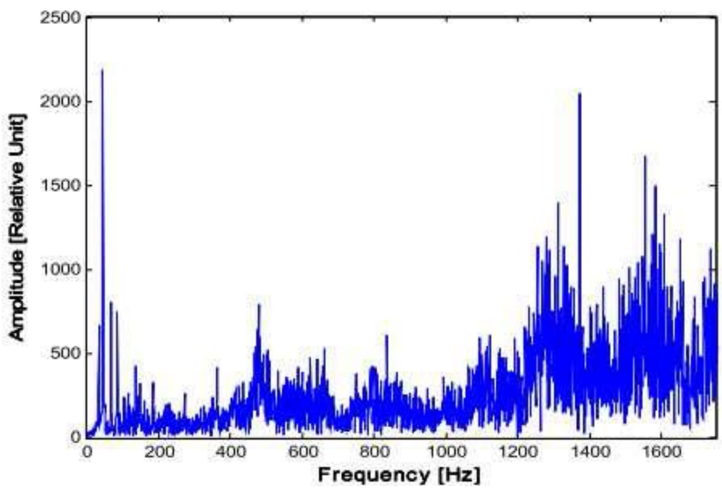

Figure A typical FFT Spectrum of defected

gear vibration signal.The vibration characteristics of any rotating machineare to some extent unique, due to the various transferInternational Journal of Advanced Engineering Technology E-ISSN 0976-3945IJAET/Vol.III/ Issue II/April-June, 2012/04-12characteristics of the machine. In the FFT plotvarious peaks with large and small are presentedcorresponding to characteristic frequencies shows theorigin of defects. Or we can say FFT shows thefrequencies in terms of shaft harmonics. For gearproblems, special attention must be given to gear'sFFT spectrum's bearing defect frequencies. Thespectra of FFT may produce peaks at identified faultfrequencies. These peaks may or may not represent the indicated fault. One must look for harmonics todetermine if the identified frequencies weregenerated from the indicated fault

- If peaks appear at the fundamental fault frequency and also at frequency two times of fundamentalfrequency, it shows strong indication of reality offault.
- If no peak appears at the fundamental faultfrequency but peaks are present at two, three, andmaybe four times of fundamental fault frequency,then this also represents a strong indication thatthe indicated fault is valid.FFT for determination of the severity of the fault
- One way to determine the fault's severity is tocompare its amplitude with the past readingstaken under consistent conditions.
- Another way is to compare the amplitude to theother readings obtained by similar machinesrunning under same conditions. A higher thannormal reading indicates a problem.

Frequency band analysis

Often, the fault detection capability using overallvibration level and/or wave shape metrics can besignificantly improve by dividing the vibration signalinto a number of frequency bands prior to analysis.This can be done with a simple analogue band-passfilter between the vibration sensor and themeasurement device. The rationale behind the use ofband-pass filtering is that, even though a fault maynot cause a significant change in overall vibrationsignal (due to masking by higher energy, non-faultrelated vibrations), it may produce a significantchange in a band of frequencies in which the no-faultrelated vibrations are sufficiently small. For asimple gearbox, with judicious selection of frequencybands, one frequency band may be dominated byshaft vibrations, another by gear tooth-meshingvibrations, and another by excited structuralresonances; providing relatively good coverage of allgearbox components.

Spectral Analysis

Spectral (or frequency) analysis is a term used todescribe the analysis of the frequency domainrepresentation of a signal. Spectral analysis is themost commonly used vibration analysis technique forcondition monitoring in geared transmission systemsand has proved a valuable tool for detection and basicdiagnosis of faults in simple rotating machinery [26].

Whereas the overall vibration level is a measure ofthe vibration produced over a broad band offrequencies, the spectrum is a measure of thevibrations over a large number of discrete contiguousnarrow frequency bands. The fundamental processcommon to all spectral analysis techniques is theconversion of a time domain representation of thevibration signal into a frequency domainrepresentation. This can be achieved by the use ofnarrow band filters or, more commonly in recenttimes, using the discrete Fourier Transform (DFT) of digitized data. The vibration level at each 'frequency'represents the vibration over a narrow frequencyband centered at the designated 'frequency', with abandwidth determined by the conversion processemployed. For machines operating at a knownconstant speed, the frequencies of the vibrationsproduced by the various machine components can beestimated therefore, a change in vibration levelwithin a particular frequency band can usually be
Associated with a particular machine component.Analysis of the relative vibration levels at differentfrequency bands can often give an indication of thenature of a fault, providing some diagnosticcapabilities. The frequency domain spectrum of thevibration signal reveals frequency characteristics ofvibrations if the frequencies of the impulseoccurrence are close to one of the gear characteristicfrequencies, such as gear frequency, pinionfrequency, gear mesh frequency, as shown inequations (3.7 to 3.9). Then it may indicate a defectrelated fault in the gearbox.The Gear frequency (Frg) is given by

$$Frg = Rg / 60 \text{ (Hz)}$$

The Pinion frequency (Frp) is given by

$$Frp = Rp / 60 \text{ (Hz)}$$

The Tooth Mesh Frequency (Frm) is given by

$$Frm = Frp \times Np \text{ (Hz) or } Frg \times Ng \text{ (Hz)}$$

Where: Rg is the speed of gear in rpm, Rp is the speedof pinion in rpm, Np is the number of teeth on thepinion and Ng is the number of teeth on gear

3.2.4 Conversion to the frequency domain

The frequency domain representation of a signal canbe described by the Fourier Transform [9] of its timedomain representation

$$X(f) = \int_{-\infty}^{\infty} x(t) e^{-j2\pi ft} dt.$$

Where x(t) is the original function in time domain,X(f) is the Fourier transform of the function x(t).The inverse process (Inverse Fourier Transform [9])can be used to convert from a frequency domainrepresentation to the time domain

$$x(t) = \int_{-\infty}^{\infty} X(f) e^{-j2\pi ft} df.$$

Where j is the square root of -1 and e denotes thenatural exponent. In the above equation, t stands fortime, f stands for frequency, and x denotes the signalin frequency domain.There are a number of limitations inherent in theprocess of converting vibration data from the time
domain to the frequency domain.

FFT Analyzers

Most modern spectrum analyzers use the Fast FourierTransform (FFT) [10], which is an efficient algorithm ,for performing a Discrete Fourier Transform (DFT)of discrete sampled data.The Discrete Fourier Transform is defined as [27]

$$X(m) = \frac{1}{N} \sum_{n=0}^{N-1} x(n) e^{-j2\pi \frac{mn}{N}}$$

and the Inverse Discrete Fourier Transform [36] is

$$x(n) = \sum_{m=0}^{N-1} X(m) e^{j2\pi \frac{mn}{N}}$$

The sampling process used to convert the continuoustime signal into a discrete signal can cause someundesirable effects.

Order Analysis

Order analysis is a technique for analyzing noise andvibration signals in rotating or reciprocatingmachinery. Some examples of rotating orreciprocating machinery include aircraft andautomotive engines, compressors, turbines, andpumps. Such machinery typically has a variety ofmechanical parts such as a shaft, bearing, gearbox,blade, coupling, and belt. Each mechanical partgenerates unique noise and vibration patterns as themachine operates. Each mechanical part contributes aunique component to the overall machine noise and vibration. When performing vibration analysis manysound and vibration signal features are directlyrelated to the running speed of a motor or machinesuch as imbalance, misalignment, gear mesh, andbearing defects. Order analysis is a type of analysisgeared specifically towards the analysis of rotatingmachinery and how frequencies change as therotational speed of the machine changes. Itresamples raw signals from the time domain into theangular domain, aligning the signal with the angularposition of the machine. This negates the effect ofchanging frequencies on the FFT algorithm, whichnormally cannot handle such phenomena.

Noise or Vibration Characteristics of mechanical faults

Mechanical faults	Vibration components
Imbalance	1x component
Misalignment	1x and 2x components.
Mechanical looseness	Harmonics of 1x and 0.5x components
Resonance	High vibration amplitude and large phase change at certain speed range
Gear defect	Gear mesh *nx* components (n is the number of gear teeth); usually modulated by rotational speed components.
Rolling-element bearing defect	Non-synchronous vibration components, usually modulated by rotational speed components.

Note: 1x means 1st order component and nx means nth ordercomponent.

- A common order analysis application is usuallycomprised of 5 steps:
- Acquire noise or vibration signals andtachometer signal.
- Pre-process the noise or vibration signals.
- Process the tachometer signal to get therotational speed profile.
- Perform order analysis with the noise orvibration signals and speed profile.
- Display the analysis results in differentformats.

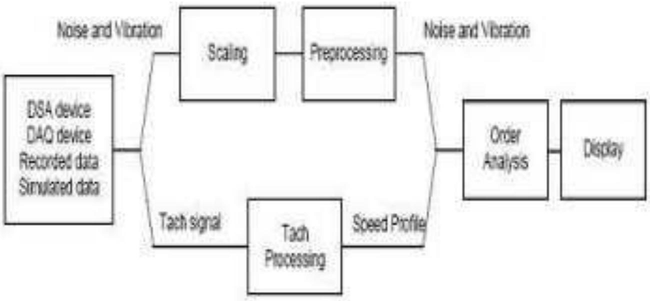

Figure : Common Order Analysis ApplicationProcess

Time synchronous Averaging

Stewart [11] showed that with ‘time synchronousaveraging’ the complex time-domain vibration signalfrom a transmission could be reduced to estimates ofthe vibration for individual shafts and their associatedgears. The synchronous average for a shaft is then treated as if it were a time domain vibration signal forone revolution of an individual, isolated shaft withattached gears. Time Synchronous Averaging (TSA)is a fundamentally different process than the usualspectrum averaging that is generally used in FFTanalysis. While the concept is similar, TSA results ina time domain signal with lower noise than wouldresult with a single sample. An FFT can then becomputed from the averaged time signal. The signalis sampled using a trigger that is synchronized withthe signal. The averaging process graduallyeliminates random noise because the random noise isnot coherent with the trigger. Only the signal that issynchronous and coherent with the trigger will persistin the averaged calculation, as shown below.Traditional spectrum based averaging records a frameof data in the time domain, computes the FFT andthen adds the FFT spectrum to the averagedspectrum. The time signal is discarded and then theprocess is repeated until the averaging number iscomplete. The result is a spectrum with very low noise, but if you examine each time record that isused to compute the FFT spectra, each time recordwill include the signal of interest plus random noisebecause the averaging is performed in the frequencydomain, not the time domain. Another importantapplication of time synchronous averaging is in thewaveform analysis of machine vibration, especially in the case of gear drives. In this case, the trigger isderived from a tachometer that provides one pulseper revolution of a gear in the machine. This way, thetime samples are synchronized in that they all beginat the same exact point related to the angular position of the gear. After performing a sufficient number ofaverages, spectrum peaks that are harmonics of thegear rotating speed will remain while nonsynchronouspeaks will be averaged out from thespectrum. Two kinds of time synchronous average:time synchronous linear average and timesynchronous exponential average.

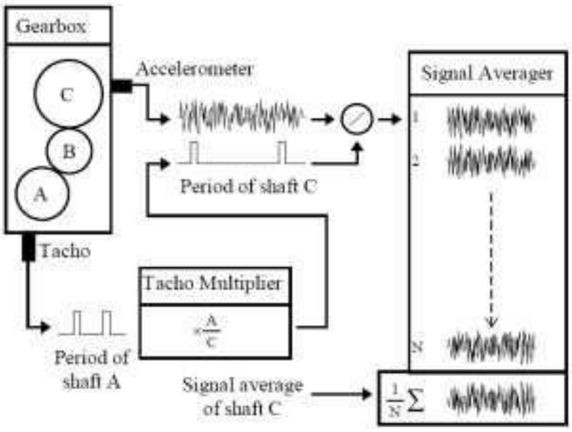

Figure Synchronous signal averaging

For time synchronous linear average the spectrumwill stop updating when the average number isreached.

Tn = nth frame of the time block signal
An = nth average of the time block signal

N = average number given
For n = 1~N, A1 = T1.
An = (An-1 *(n-1) + Tn)/n

nth frame of the spectrum is calculated from An.When the average number N is reached, the averagedtime block signal is

AN = (AN-1 *(N-1) + TN)/N = (A1 + A2 + A3 + …AN-1 + AN)/N

The averaged spectrum is calculated from AN.For time synchronous exponential average: spectrumkeeps updating and never stops.

P = 1/N = inverse of the average number N
Tcur = current frame of the time block signal
Acur = current average of the time block signal
Apre = previous average of the time block signal

The averaged time block signal is

Acur = (1- _)*Apre + _ * Tcur

The averaged spectrum is calculated from Acur.Stewart developed a number of non-dimensionalparameters based on the synchronous signal average,which he termed 'Figures of Merit' [38]. These wereoriginally defined as a hierarchical group, with whichStewart described a procedure for the detection andpartial diagnosis of faults.

FM0

The parameter FM0 was developed by Stewart in1977 as a robust indicator of major faults in a gearmesh [11]. Major changes in the meshing pattern aredetected by comparing the maximum peak-to-peakamplitude of the signal to the sum of the amplitudesof the mesh frequencies and their harmonics. FM0 isgiven as

$$FM0 = \frac{PP_x}{\sum_{n=0}^{H} P_n}$$

wherePPx is the maximum peak-to-peak amplitudeof the signal x; Pn is the amplitude of the nthharmonic, and H is the total number of harmonics inthe frequency range. Notice that in cases where PPxincreases while Pn remains relatively constant, FM0increases. Also, if Pn decreases while PPx remainsconstant, FM0 also increases.

FM4

Developed by Stewart in 1977, the parameter FM4was designed to complement FM0 by detecting faultsisolated to only a limited number of teeth [11]. Thisis accomplished by first constructing the differencesignal, d; given in Eq. (5). The normalized kurtosis of d is then computed. FM4 is given as

$$FM4 = \frac{N\sum_{i=1}^{N}(d_i - \bar{d})^4}{\left[\sum_{i=1}^{N}(d_i - \bar{d})^2\right]^2}$$

Where d is the mean of the difference signal, and Nis the total number of data points in the time signal.FM4 is no dimensional and designed to have anominal value of 3 if d is purely Gaussian. Whenhigher-order sidebands appear in the vibration signal,FM4 will deviate from this value.

NA4

The parameter NA4 was developed in 1993 byZakrajsek, Townsend, and Decker at the NASALewis Research Center as a general fault indicatorwhich reacts not only to the onset of damage as FM4does, but also to the continuing growth of the fault[12]. The residual signal r; given in Eq. (6), is firstconstructed. The quasi-normalized kurtosis of theresidual signal is then computed by dividing thefourth moment of the residual signal by the square of its run time averaged variance. The run time averagedvariance is the average of the residual signal overeach time signal in the run ensemble up to the pointat which NA4 is currently being calculated. NA4 isgiven as

$$NA4(M) = \frac{N\sum_{i=1}^{N}(r_{iM} - \bar{r}_M)^4}{\left\{\frac{1}{M}\sum_{j=1}^{M}\left[\sum_{i=1}^{N}(r_{ij} - \bar{r}_j)^2\right]\right\}^2}$$

Where r is the mean of the residual signal, N is thetotal number of data points in the time signal, M isthe number of the current time signal, and j is theindex of the time signal in the run ensemble. LikeFM4, NA4 is nondimensional and designed to have anominal value of 3 if r is purely Gaussian.

M6A

The parameter M6A was proposed by Martin in 1989as an indicator of surface damage on machinerycomponents [13]. The underlying theory is the sameas that of FM4. However, it is expected that M6Awill be more sensitive to peaks in the differencesignal due to the use of the sixth moment. M6A isgiven as

$$M6A = \frac{N^2\sum_{i=1}^{N}(d_i - \bar{d})^6}{\left[\sum_{i=1}^{N}(d_i - \bar{d})^2\right]^3}.$$

Note that in this case, the moment is normalized bythe cube of the variance.

M8A

The parameter M8A, also proposed by Martin in1989, is designed to be yet more sensitive than M6Ato peaks in the difference signal [13]. M8A uses theeighth moment normalized by the variance to thefourth power and is given as

$$M8A = \frac{N^3\sum_{i=1}^{N}(d_i - \bar{d})^8}{\left[\sum_{i=1}^{N}(d_i - \bar{d})^2\right]^4}.$$

The parameter NB4 was developed by Zakrajsek,Handschuh and Decker in 1994 as an indicator oflocalized gear tooth damage [14]. The theory behindNB4 is that damage on just a few teeth will causetransient load fluctuations different from those loadfluctuations caused by healthy teeth, and that this canbe seen in the envelope of the signal. As with NA4,

NB4 uses the quasi-normalized kurtosis. However,instead of the difference signal, NB4 uses theenvelope of the signal band-pass filtered about themesh frequency. The envelope, s is computed usingthe Hilbert transform and is given by

$$s(t) = [b(t) + i[H(b(t))]]$$

Where b(t) is the signal band-pass filtered about themesh frequency, H(b(t)) is the Hilbert transform ofb(t); and i is the sample index.

NA4*

The parameter NA4* was developed in 1994 byDecker, Handschuh and Zakrajsek as an enhancementto NA4 [15]. In this case, the denominator of NA4 isstatistically modified, i.e. when the variance of theresidual signal exceeds a certain statisticallydetermined value, the averaging stops and thedenominator is locked. This modification was madebased on the observation that as damage progressesfrom localized to distributed, the variance of the signal increases significantly, causing the kurtosis tosettle back to nominal values after the initialindication of the onset of damage. By normalizing thefourth moment by the variance of a baseline signalfrom the transmission operating under nominalconditions, NA4* is provided with enhanced trendingcapabilities. Since it was observed that the varianceof a damaged transmission signal is greater than thatof a healthy transmission signal, the decision to lockthe denominator is made based on an upper limit, L;given by

$$L = \bar{v} + \frac{Z}{\sqrt{N}}\sigma$$

Where v is the mean value of previous variances, Z isthe probability coefficient usually chosen for anormal distribution, s is the standard deviation of theprevious variances, and N is the number of samples.Z for a normal distribution can be found in anyintroductory statistics text. However, the actualchoice of Z should be made based onexperimentation as too small a value could lead to anoverabundance of false alarms.

Demodulation

The original observation made by Stewart [11] thatgear tooth damage causes an increase in theamplitude of the sidebands about the regular meshingcomponents led to further investigations into thenature of the amplitude and phase modulationfunctions. It was proposed that the vibration signalcould be demodulated to obtain separateapproximations of the amplitude and phasemodulation functions and that these approximationscould subsequently be inspected to find earlyindications of gear damage [16, 17]. This work was further refined by Blunt and Forrester [18] to producea useful damage indicator referred to as a bulls-eyeplot which indicates both amplitude and phasedemodulations simultaneously.

VIII. ADVANCED SIGNAL PROCESSINGTECHNIQUES IN VIBRATION ANALYSIS

An overall schema for intellectual diagnostics ispresented in Figure 5. Intelligent diagnosis beginswith the act of data collection which is followed byfeature extraction usually employing the frequencyspectra. Feature extraction techniques are widespreadand can range from statistical to model basedtechniques and comprises a variety of signalprocessing algorithms which includes wavelettransforms. Fault detection and identification is asubsequent step and is further classified in thisreview into the four categories shown in the figure -these will now be treated separately.

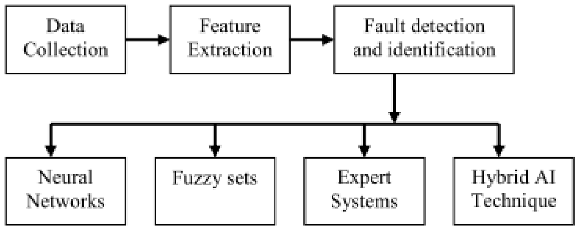

Figure Intelligent fault diagnosis

With the development of soft computing techniquessuch as artificial neural network (ANN) and fuzzylogic, there is a growing interest in applying theseapproaches to the different areas of engineering.Artificial Neural Networks (ANN) have become theoutstanding method in the recent decades exploitingtheir non-linear pattern classification properties,offering advantages for automatic detection andidentification of gearbox failure conditions, whereas they do not require an in-depth knowledge of thebehavior of the system. Recent systems have reliedon artificial intelligence techniques to strengthen therobustness of diagnostics systems. Four artificialtechniques have been widely applied as expertsystem, neural networks, fuzzy logic, and modelbasedsystems [16]. Different kinds of artificialintelligence method have become common in faultdiagnosis and condition monitoring. For example, fuzzy logic and neural networks have been used inmodeling and decision making in diagnosticsschemes. Neural networks-based classifications areused in diagnosis of gearbox. Rafiee. J et al, proposedfault detection and identification of gearboxes using anew feature vector extracted from standard deviationof wavelet packet coefficients of vibration signals of various faultless and faulty conditions of a gearboxusing ANN. Over and above the structure of ANN, anappropriate feature vector plays a vital role intraining a high performance ANN. Ultimately a MLP(Multi-Layer Perceptron) network with a 16:20:5structure has been used that not only is small in sizebut also with a 100% perfect accuracy andperformance to identify gear failures and detectbearing defects [19].ANN-based research to carry out the task can becategorized into two distinct groups: faultidentification systems with low efficiency which waspresented by Kazlas et al [20] to recognize gears andbearings failures of a helicopter gearbox and faultdetection systems with high efficiency which isillustrated by Samanta et al. [52] to detect rollerbearingelements defects. Precisely speaking, fault identification proves effective in the case ofparticular fault classification systems, whereas thismay be in conflict with a situation that there is arequirement to a comprehensive fault detectionsystem to provide accordingly precision andpromptness. The objective of this research was todevelop an ANN-based system with high efficiencyand the lowest erroneous outcome to identify faultygears and detect faulty bearing of a gearbox whichhas a lot of applications for preventing from fatalbreakdowns in rotary machineries. Zhenya et al. proposed a multilayer feed forward network-basedmachine state identification method. They representcertain fuzzy relationship between the faultsymptoms and causes, with highly nonlinearitybetween the input and the output of the network [21].Fuzzy logic-based fault diagnosis methods have theadvantages of embedded linguistic knowledge andapproximate reasoning capability. The Fuzzy logicproposed by Zadeh [22] performs well at qualitative description of knowledge. However, the design ofsuch a system depends heavily on the intuitiveexperience acquired from practicing operators thusresulting in subjectivity of diagnosed faults. fuzzy membership function and fuzzy rules cannot beguaranteed to be optimal in any case. Furthermore,fuzzy logic systems lack the ability of self learning,which is compulsory in some highly demanding realtimefault diagnosis cases [25]. Rough set based intelligence diagnostic systems have been constructedand used in diagnosing valves in three-cylinderreciprocating pumps [24] and turbo generators [25].Intelligent systems cover a wide range of techniquesrelated to hard science such as modeling and controltheory, and soft science such as the artificialintelligence. Intellectual systems, including neuralnetworks, fuzzy logic, and hybrid techniques, utilizethe concepts of biological systems and human cognitive capabilities. These three systems have beenrecognized as a robust and alternative to some of theclassical modeling and control methods [24].

CONCLUSION

1) In this paper, authors have been presented a briefreview of some current vibration based techniquesused for condition monitoring in geared transmissionsystems. After the review of literature on gear faultanalysis, the following points are concluded.
2) Gearbox vibration signals are usually periodicand noisy. Time-frequency domain averagetechnique successfully removes the noise fromthe signal and captures the dynamics of oneperiod of the signals.
3) Time domain techniques for vibration signalanalysis as waveform generation, Indices (RMS value, Peak Level value, and crestfactor) and overall vibration level do notprovide any diagnostic information but mayhave limited application in fault detection insimple safety critical accessory components.The statistical moment as kurtosis is capable toidentify the fault condition but skewness trendhas not shown any effective faultcategorization ability in this present gear faultcondition.
4) Spectral analysis may be useful in thedetection and diagnosis of shaft faults.
5) In frequency domain analysis, it is concludedthat FFT is not a suitable technique for faultdiagnosis if multiple defects are presents ongearbox. The envelope analysis and PowerSpectrum Density techniques have shown abetter representation for fault identification.The Hilbert Transform and PSD techniques aresuitable for multiple point defect diagnosticsfor condition monitoring.
6) Synchronous signal averaging has the potentialof greatly simplifying the diagnosis of shaftand gear faults (i.e., the safety critical failures)by providing significant attenuation of nonsynchronousvibrations and signals on whichideal filtering can be used. Furtherdevelopment needs to done on theimplementation of synchronous averagingtechniques and the analysis of results.
7) Expert system based on ANN and fuzzy logiccan be developed for robust faultcategorization with the use of extractedfeatures from vibration signal.
8) The results further show that the waveformgeneration in case of multiple faults at gearcontact surfaces is only useful to find thehealthy or faulty condition but not capable toidentify the categories of fault.
9) These conclusions motivate further research toincorporate other parameters and symptoms withvibration features to develop more robust expertsystems for diagnose the problem of gear faultssignature analysis.It has been shown that using these ways of vibrationsignal analysis there are possibilities to detect signalfaults and distributed faults in gearboxes. A signalfault is caused by a tooth crack/fracture and breakage,a spall in a gearing or in an inner or outer race of abearing, a spall on a rolling element of a bearing;distributed faults are caused by uneven wear (pitting, scuffing, abrasion, erosion).In this analysis, pressure is applied to surface as in Figure 9as a normal to that surface. This is meaning that force ismainly applied to X-axis and Y-axis. Due to this reason,only result for Y-axis and X-axis is more considerable inthis harmonic analysis. For the Y-axis and X-axis, the first maximum amplitudefor normal stress and directional deformation are happen at124.8 Hz. At this frequency, the resonance is occurred.
10) In this analysis analysis, pressure is applied to surface as in Figure 9 as a normal to that surface. This is meaning that force is mainly applied analysis.
11) For the Y-axis and X-axis, the first maximum amplitudefor normal stress and directional deformation are happen at 124.8 Hz. At this frequency, the resonance is occurred.
12) In this analysis, first resonance is happen when the ratio ofharmonic forced frequency over natural frequency isr = first resonance in harmonic forced frequency/firstmodal natural frequency

$= 124.8/120.93$
$= 1.032 \approx 1$

13) In order to prevent the resonance, frequency ratio need tobe setup to be less than 1.When $r<<1$ the amplitude is justthe deflection of the spring under the static force $F0$. Thisdeflection is called the static deflection δst. Hence, when$r<<1$ the effects of the damper and the mass are minimal.The magnitude can be reduced if the natural frequency canbe shifted away from the forcing frequency by changingthe stiffness or mass of the system. If the system cannot bechanged, perhaps the forcing frequency can be shifted.
14) In this study, frequency ratio can set to 0.25 from the firstmodal natural frequency analysis in order to preventresonance.

Forced frequency = 0.25× natural frequency
$= 0.25\times120.93 = 30.2325$ Hz
Static deflection can be achieved if forced frequency isfrom 0 Hz to 30.2325 Hz.

ACKNOWLEDGMENT

A special thanks to Dr.Yadagiri sir for their advices and guidance in finishing thispaper.

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