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Fault Diagnosis and Dynamic Simulation of Gear Box Using Finite Element Analysis

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ABSTRACT

In gearboxes, load fluctuations on the gearbox and gear defects are two major sources of vibration. Further, at times, measurement of vibration in the gearbox is not easy because of the inaccessibility in mounting the vibration transducers. Vibration analysis techniques are used for detection of fault in gear system, fluctuation in gear load such as a method for monitoring the evolution of gear faults based on the newly developed time-frequency analysis through FEA, in which analysis is carried out with the decomposed current signal to trace the sidebands of the high frequencies of vibration,. It is also helpful tool for health monitoring of gears.

Acoustic signal can be used effectively along vibration signal to detect the various local faults in gearboxes using the wavelet transform technique. Two commonly encountered local faults, tooth breakage and tooth crack were simulated. In fault simulating, two very similar models of worn gear have been considered with partial difference for evaluating the preciseness of the proposed algorithm. Moreover, the processing of vibration signals has become much more difficult because a full-of-oil complex gearbox system has been considered to record raw vibration signals. Raw vibration signals were segmented into the signals recorded during one complete revolution of the input shaft using tachometer information and then synchronized using piecewise cubic hermite interpolation to construct the sample signals with the same length.

Keywords— Gear Defects, Vibration Analysis, Finite Element Analysis, Acoustic Signal, Tooth Breakage, Raw Vibration Signals.

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I. INTRODUCTION

Gear Boxes constitute a very vital link in the transmission chain of a variety of equipment and machinery. The earliest drives used cylindrical rods inserted radically in one wheel meshing with similar rods mounted axially in another wheel. This type of drive performed satisfactory, though inefficiently, at low speeds and low loads but trouble was encountered as soon as load or speed was raised [1].

The increasing trend towards predictive maintenance has led to the development of a vast number of machine condition monitoring techniques. Of these techniques, vibration analysis and oil analysis are the two distinct and most readily used methods in determining mechanical failures in common components of industrial machinery such as gears and bearings. [2]

Gear mechanisms are an important element in a variety of mechanical systems. For that reason, early fault detection in gears has been the subject of intensive investigation and many methods based on vibration signal analysis have been developed. Conventional methods include crest factor, kurtosis, power spectrum and cepstrum estimation, time-domain averaging and demodulation, which have proved to be effective in fault diagnosis and are now well established [3].

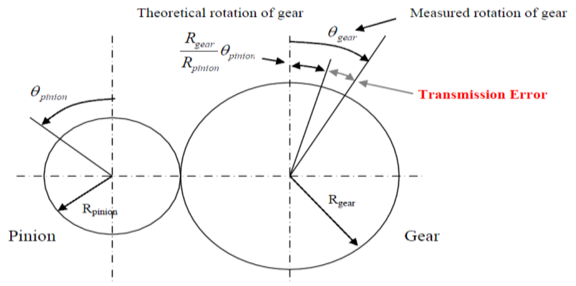
In gearboxes and power drive trains in general, gear damage detection is very critical and its early diagnosis can lead to increased safety in aviation and in various industrial applications. Few research teams have published experimental data coming from long-term testing to study the effect of natural gear pitting mostly upon vibration recordings [5]. The interest for applications of acoustic emission (AE) for condition monitoring in rotating machinery is relatively new and has grown significantly over the last decade. As AE mainly detects high-frequency elastic waves, it is not affected by structural resonances and typical mechanical background noise (under 20 kHz). The AE from helical gears based mainly in the root-mean-square of the recorded signals.AE to spur gears in a gearbox test-rig. They simulated pits of constant depth but variable size and AE parameters such as energy, amplitude and counts were monitored during the test. AE was proved superior over vibration data on early detection of small defects in gears. [1]

A. Problem Description

If the gears were perfectly rigid and no geometrical errors or modifications were present, the gears would transmit the rotational motion perfectly, which means that a constant speed at the input shaft would result in a constant speed at the output shaft. The assumption of no friction leads to that the gears would transmit the torque perfectly which means that a constant torque at the input shaft would result in a constant torque at the output shaft. No force variations would exist and hence no vibrations and no sound (noise) could be created. However, in reality, there are geometrical errors, deflections and friction present, and accordingly, gears, by their inherent nature, cause vibrations due to the large pressure which occurs between the meshing teeth when gears transmit power. Meshing of gears involves changes in the magnitude, the position and the direction of large concentrated loads acting on the contacting gear teeth, which as a result causes vibrations. Extended period of exposure to noise and vibration are the common causes of operational fatigue, communication difficulties and health hazards. Reduction in noise and vibration of operating machines has been an important concern for safer and more efficient machine operations [2].

B. Transmission Errors in Gears

Transmission Error (TE) is one of the most important and fundamental concepts that forms the basis of understanding vibrations in gears. The name ‘Transmission Error’ was originally coined by Professor S. L. Harris from Lancaster University, UK and R.G. Munro, his PhD student at the time. They came to the realization that the excitation forces causing the gears to vibrate were dependent on the tooth meshing errors caused by manufacturing and the bending of the teeth under load. TE is defined as the deviation of the angular position of the driven gear from its theoretical position calculated from the gearing ratio and the angular position of the pinion [3].



$$TE = \left(\theta_{gear} - \frac{R_{pinion}}{R_{gear}} \theta_{pinion} \right)$$

Fig.1 Concept of Transmission Error (TE)

C. Factors Affecting Transmission Errors

Usually the teeth mesh frequency and its related harmonics are responsible for the occurence of gear noise [3]

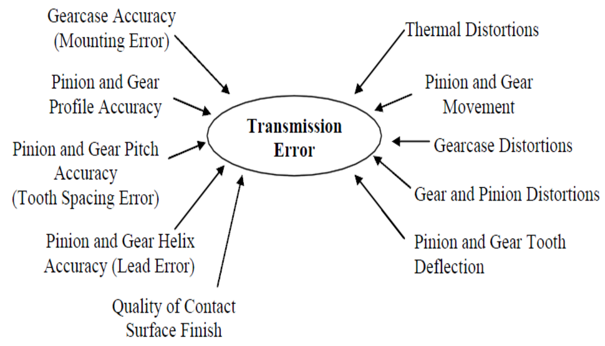


Fig.2 Factors Affecting TE

D. Project Inception

All Naval Warships are fitted with latest state of art transmission systems. The existing defect analysis method using vib-10, SPM and frequency spectrum analysis do not help in identifying defects at embryonic stage. This dissertation attempts to bring a practical solution to identify the gearbox faults at the initial stage.

E. Project Methodology

- To collect vibration signature of good gears and defective gear (missing tooth) from Gearbox Dynamics Simulator (GDS).
- Real time vibration signatures of good and defective gears are to be acquired with the help of accelerometers using Data Acquisition System (DAS).
- Modelling of gear box containing good and defective gears would be carried out using Solid Works.
- The same model will then exported to ANSYS 14 for simulation and further analysis.
- Transient analysis would be carried out initially with only two spur gears, gearbox housing and the two shafts to get feel of simulation and analysis in FEM environment.

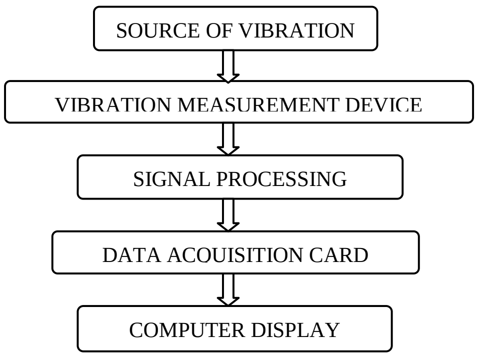


Fig. 3 Steps for detecting the fault in system

II. LITERATRE RIVEW

History

There has been a great deal of research on gear analysis, and a large body of literature on gear modeling has been published. The gear stress analysis, the transmission errors, and the prediction of gear dynamic loads, gear noise, and the optimal design for gear sets are always major concerns in gear design. The first study of transmission error was done by Hiroaki [3]. He showed that the behaviour of spur gears at low speeds can be summarized in a set of static transmission error curves. In later years, Nader [3] analyzed the vibratory excitation of gear systems theoretically. He derived an expression for static transmission error and used it to predict the various components of the static transmission error spectrum from a set of measurements made on mating pair of spur gears. Kohler and Toutountzakis [6] discussed the derivation of gear transmission error from pitch error transformed to the frequency domain. Kubo et al estimated the transmission error of cylindrical gears using a tooth contact pattern. The current literature reviews also attempt to classify gear model into groupings with particular relevance to the research. The following classification seems appropriate [6].

Nowadays the demands for condition monitoring and vibration analysis are no more limited trying to minimize the consequences of machine failures, but to utilize existing resources more effectively. This paper describes the practical aspect of vibration phenomena and the measurement requirements of a general monitoring system consisting of data collection with data reports in digital manner, followed by the acquisition of the vibration values for faulty and good Gears. [6]

B. Conclusion for Literature Review:

After a brief analysis of some current vibration based techniques used for condition monitoring in geared transmission systems and it is conclude that frequency domain features are generally more consistent in the detection of damage than time domain parameters, so any fault diagnosis method which is based on frequency domain can be selected for project work.

It is also conclude that the envelope analysis and Power Spectrum Density techniques have shown a better representation for fault identification. The Hilbert Transform and PSD techniques are suitable for multiple point defect diagnostics for condition monitoring. Also Resonance Demodulation is a recent technique that can be used for early detection of gear tooth crack but can be effectively used at the early stages of fault generation.

It can be conclude for the above literature review that the envelope analysis or demodulation analysis is most suitable and common for diagnosing the bearing faults and can also be used for the detection of gear faults also.

III. PROJECT METHODOLOGY

A. EXPERIMENTAL SETUP.

The experimental rig consists of two electrical machines, a pair of spur gears, a power supply unit with the necessary speed control electronics and the data acquisition system. Referring to Fig.4, a DC machine of 1.5 KW rotates the pinion. The load is provided by an AC asynchronous machine, which is configured as a brake. The transmission ratio is 35/19=1:842, which means that an increase in rotational speed is achieved. The vibration signal generated by the gearbox was picked up by an accelerometer bolted to the pinion body and the electrical signal was transferred to an external charge amplifier through slip rings. No form of signal averaging was used as the signal to noise ratio has been considered high enough.

The sampling interval was Δt =0:05 ms corresponding to sampling frequency (fs) of 20 kHz. The signal was low pass filtered at 5 kHz through a fourth order Bessel type filter, in order to limit aliasing distortion and retain waveform integrity as much as possible. Data were stored for post processing to a portable PC. The acquisition system was flexible enough to be used in actual industrial conditions.[10]

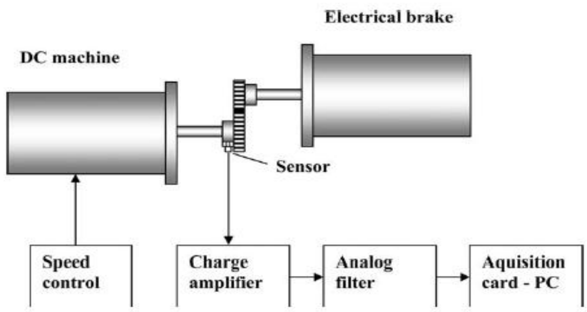


Fig.4 Experimental Test Rig

B. Collection of Vibration Signature using GDS.

Spectra Quest's Gearbox Dynamics Simulator (GDS) has been specifically designed to simulate industrial gearbox for experimental and educational purposes. The gearbox consists of a 2 stage parallel shaft gearbox with rolling bearings and a magnetic brake. All elements of the GDS have been designed to investigate gearbox dynamics and acoustic behaviour, health monitoring, vibration based diagnostic techniques, lubricant conditioning or wear particle analysis. It is robust enough to handle heavy loads and spacious enough for easy gear placement, setup, and installation of monitoring devices. The two-stage parallel shaft gearbox can be configured as to reduce or increase the gear ratio. The GDS is a specially designed platform for studying signatures of common gearing faults. Spur or helical gears can be fitted into a two-stage parallel shaft design. The gearbox is driven by a 3 HP motor with a speed range of 0-3600 rpm. Load can be applied using the magnetic brake. The loading force is controlled by a current source. The gears can be set up at different locations along the shaft in order to alter system stiffness and make room for additional devices. The GDS is also a test bed for analyses in gear noise, loading effect and fault diagnosis techniques.[10]

Salient Features: Main features of the equipment includes:

- Portable, robust, cost-effective, gearbox and bearing vibration trainer.
- Ideal for teaching multi-plane balancing with centre hung / overhung rotors.
- Can be setup to exhibit bearing fault frequencies.
- Develop signal processing techniques to identify bearing faults and Gear faults.

C. Analysis of Tooth Breakage with Vibration Signal

Fig.5: Gears before removing Teeth.

Fig.6 Sample of Defective Gear: Missing Tooth defect

Fig.7: Gears after removing Teeth.

D. Real Time Vibration signatures of good and defective gears are to be acquired with the help of accelerometers using Data Acquisition System (DAS).

The Fig. 8 and Fig.9 shows the signals in time domain and frequency domain, when the gear is fault free or healthy. The Time domain signal is converted into frequency domain with the help of FFT of the signal.[4]

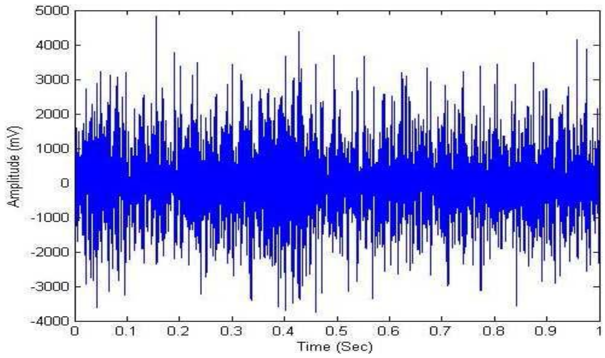


Fig.8 Vibration Signal In Time of Healthy Gear

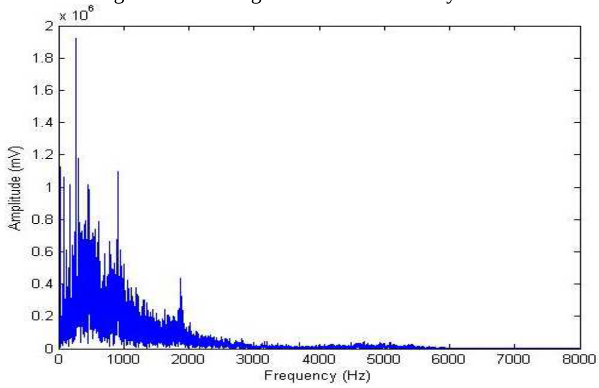


Fig.9 Vibration Signal in Frequency of Healthy Gear

As the Fault is introduced in driven gear, the vibration signals are changed as shown in figure 11. It is clear from the figure 11 that the amplitude is higher at the frequency nearly 5.2 kHz.

From the spectrum in figure 11, one can observe that the vibration bursts due to the defect in the gear tooth generates high frequency components (in the range of 4.5–5.5 kHz). The defect is identified in the spectrum as high intensity stripes. Therefore a frequency band range between 4.5 kHz to 5.5 kHz is selected by applying the filtering of the signal as shown in figure 12. [4]

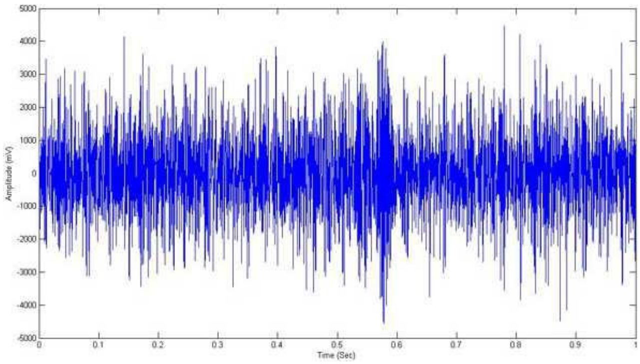


Fig.10 Vibration Signal in Time of Faulty Gear

Fig.11 Vibration Signal in Frequency of Faulty Gear

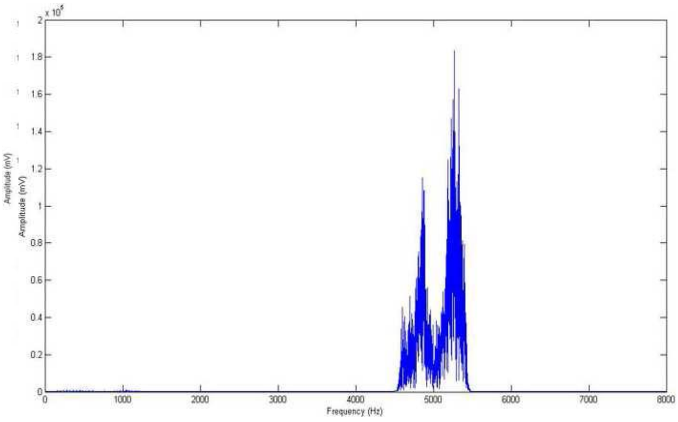


Fig.12 Filtered Signal in Frequency Domain of Faulty Gear

E. Analysis of Spur Gear using ANSYS

Step 1:- Import the solid model from pro e:-File-import-IGES

Step 2:-Preference →Structural→ OK.
Step 3:- Preprocessor Selecting the element type for the gear Preprocessor → Element type →Add/Edit/Delete→ click on Add in the dialogue box that appears → In Library of Element Types select Solid-Tet 10node 187→Ok
Step 4:- Specifying the material properties Choose preprocessor→ Material props→ Material Models→ Double click on Structural→ Linear→ Elastic→ Isotropic. The Young‟s modulus of plain carbon steel is **2.1 E5** and the poisson ratio (PRXY) is **0.3.** Choose preprocessor→ Material props→ Material Models→ Double click on Structural→ Density. Density of plain carbon steel is **6.8e-6**
Step 5:-Meshing the geometry Descretize the model into finite elements. Set the element edge length to 30.
a) Meshing→ Size control→ Manual → Global→ Size→ Enter Element Edge Length as 30 → OK
b) Now mesh all the volumes in the geometry
Step 6:-Applying load and constraints Loads→ Define Loads→ Apply → Structural→ Displacement→ On Areas→ Select the inner surfaces of the key hole→ OK
Step 7:- Solution Solution→ Solve→ Current LS→ In the prompting that appears on the screen click YES
Step 8:-General Postprocessor General Post procedure → Plot Results→ in the Plot Deformed Shape dialogue box select Def + un-deformed → General Post procedure→ Plot Results→ Contour Plot→ Nodal Solu→ Stress→ Von mises Stress→ OK. [10]

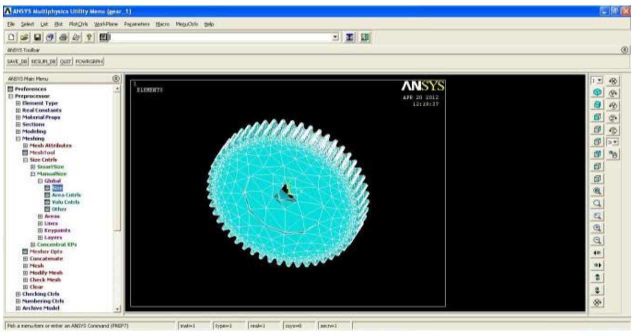


Fig.13 Modelling and Analysis of Spur Gear using ANSYS- (on 22nd April 2014)

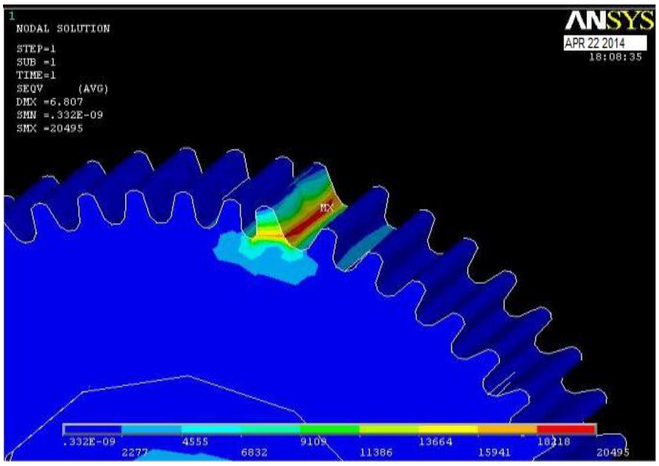


Fig.14 Element Edge - ANSYS- (on 22nd April 2014)

TABLE I
GEOMETRY OF THE GEAR SET

Description	Symbol	Formulae	Values
Number of teeth on pinion	zp	-	18
Number of teeth on gear	zg	-	51
Pressure angle	αn	-	200
Module (mm)	m	-	4
Circular pitch (mm)	p	$p=\pi\times m$	12.566
Pitch circle diameter (mm)	dp	dp=m×z	204
Addendum height (mm)	ha	ha=m	4
Dedendum height (mm)	hd	hd=1.25m	5
Addendum circle (mm)	da	da=dp+(2×m)	212
Dedendum circle (mm)	dd	$dd=dp-2+\pi z \times m$	194
Base circle (mm)	db	$db=dp\times\cos\alpha n$	192
Face width (mm)	F	F=10×m	40

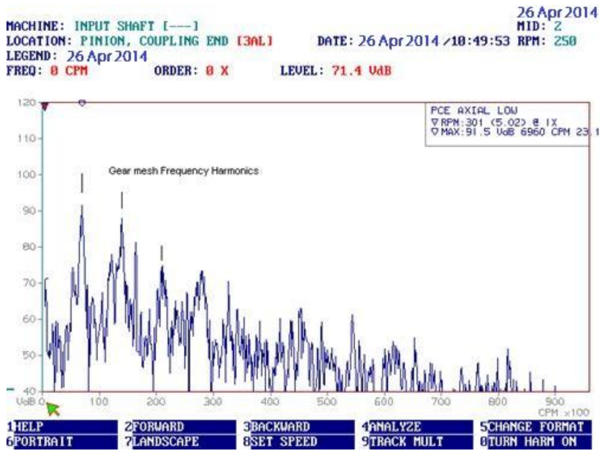


Fig.15 Gear Mesh Frequency Harmonics – ANSYS – (26th April 2015)

IV. SAMPLE READING

A. Spectrums before Removing Teeth

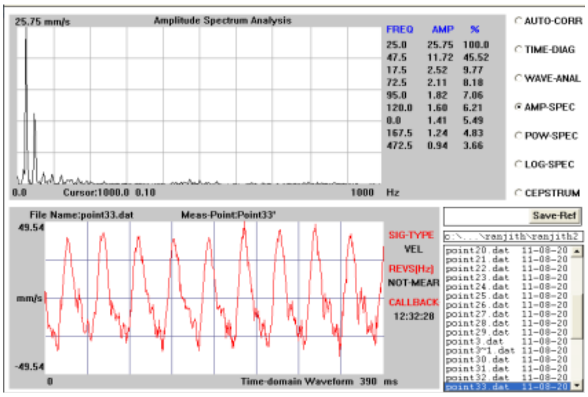


Fig.16. 1st Gear GNDE hor

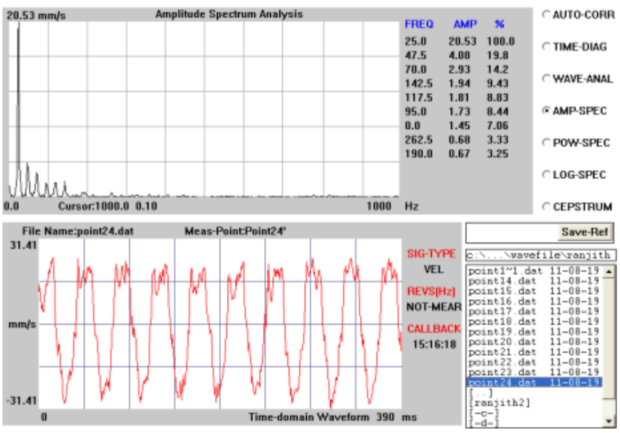


Fig.17. 2nd Gear GNDE hor

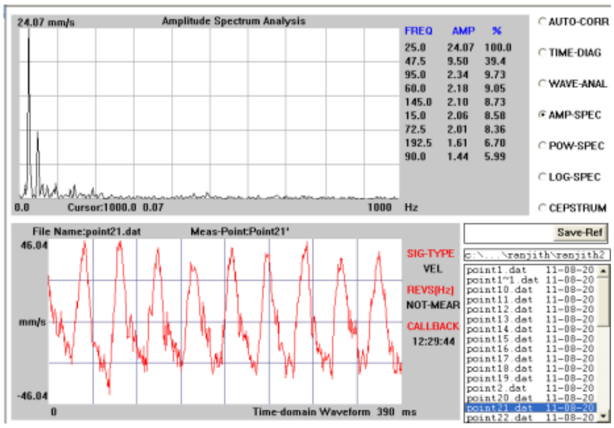


Fig.18. 3rd Gear GNDE hor

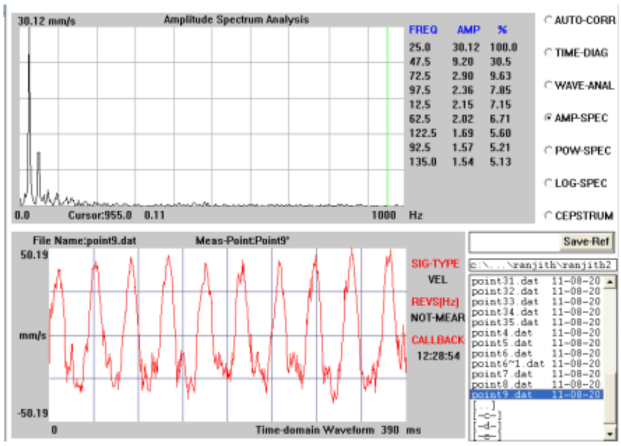


Fig.19. 4th Gear GNDE hor

B. Spectrums before Removing Teeth

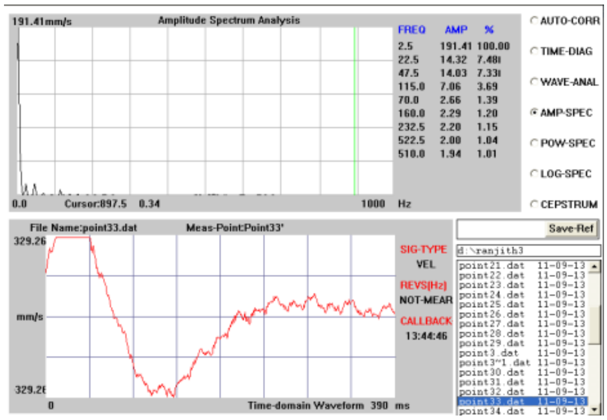


Fig.20. 1st Gear Hor GNDE

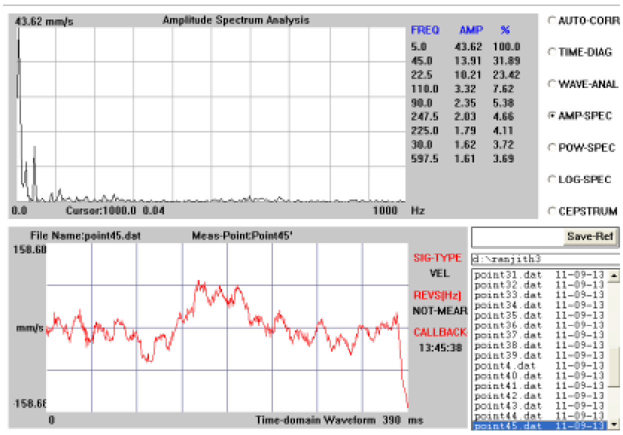


Fig.21. 2nd Gear Hor GNDE

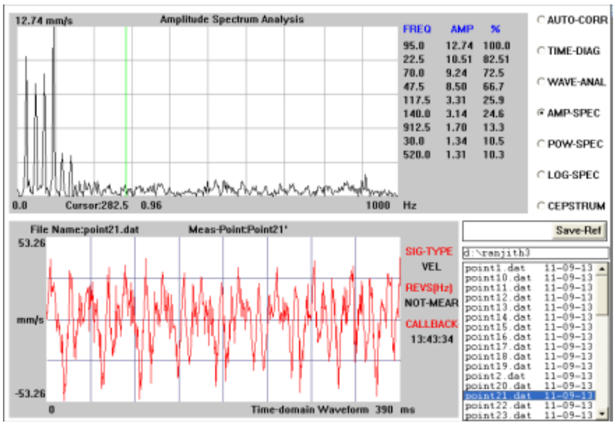


Fig.22. 3rd Gear Hor GNDE

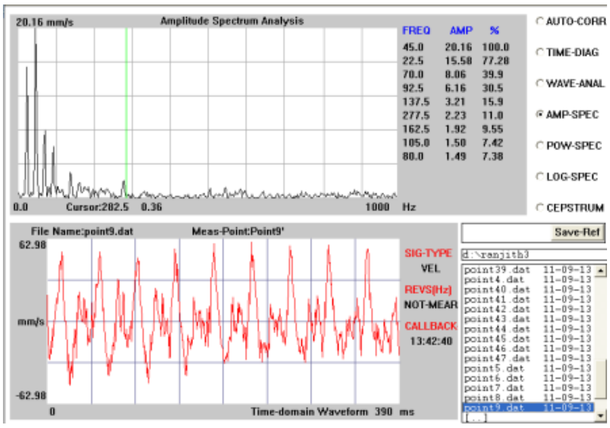


Fig.23. 4th Gear Hor GNDE

C. Observations from Spectrums

- Gearbox is located at GNDE (2nd Gear). By observing the spectrums at GNDE before breaking the peak amplitudes are at 1x.
- Since the Gear shaft is rotated approximately around 1500rpm the harmonics is 1500/60=25cps.
- By observing the spectrums at GNDE after removing Gear teeth the peaks are present at sub harmonics and multiples of frequencies. The cause of presenting the sub-harmonics &multiples of frequencies is due to the presence of fault in the Gearbox.
- The vibrations increased from 5times to 8times after observing the spectrums.

Sr. No.	2nd gear Non Drive End				
	Direction	Harmonic Frequency		Amplitude	
		B.B	A.B	B.B	A.B
1	H	1X	1/5 X	20.53	43.62
2	V	1X	1/10 X	36.77	319.61
3	A	1X	Approx 1X	27.53	15.41

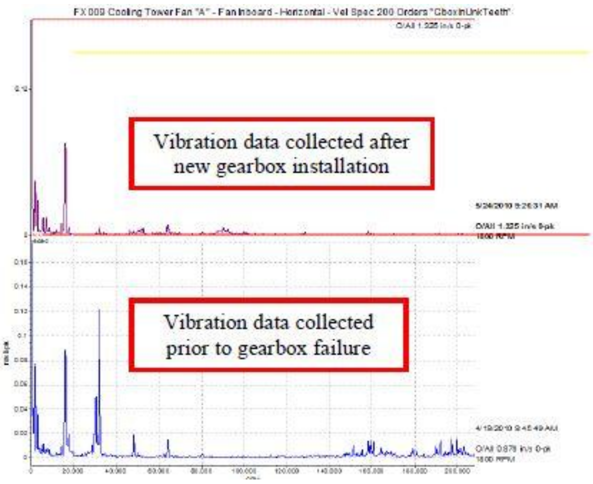


Fig.24. Spectrum Analysis Complied Result

D. Results from Spectrums

TABLE II
COMPARISON OF GEAR SPECTRUMS

Sr. No.	2nd gear Non Drive End			
	Direction	Parameters	GNDE	
			B.B	A.B
1	H	Acceleration	42.9	197
		Velocity	14.1	207
		Displacement	327	2501
2	V	Acceleration	35.2	275
		Velocity	17.2	89.5
		Displacement	291	8426
3	A	Acceleration	172	67.7
		Velocity	27.9	21.5
		Displacement	528	140

B.B- Before Breaking A.B.- After Breaking

TABLE III
COMPARISON OF RESULTS

Size of Element	Solution by conventional method	ANSYS result	% Accuracy
30	11827.2	11777	99.57
40		11386	96.26
50		13179	89.74

V. CONCLUDING REMARKS

1) It is observed that vibrational analysis is better compared to other monitoring techniques.

2) To reduce costs and facilitate diagnosis, vibrational analysis is a very popular technique.
3) When the meshing characteristics are disturbed by a defective tooth, this will change the noise content of the acoustic signal when compared with the sound of healthy operation.
4) Further increases in the fault conditions will not only cause low-frequency components in the wavelet representation, but also the meshing frequency should change and gradually becomes discontinuous. This indicates that the characteristics of the acoustic signals vary due to the fault conditions.
5) It provides a highly sensitive, selective, and co-effective means for online monitoring of a wide variety of heavy industrial machinery.
6) This paper has investigated the detection of Gear fault using vibration monitoring.
7) An experimental study has been conducted on motor with Gearbox, measured quantities such as frequency and amplitudes.
8) The peaks are present at sub-harmonics and multiples of frequencies. The cause of presenting the sub harmonics & multiples of frequencies is due to the presence of fault in the Gearbox.

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