Addis Ababa University

Addis Ababa Institute of Technology

Department of Mechanical Engineering

EXPERIMENTAL INVESTIGATION OF GEAR FAULT USING VIBRATION SIGNATURE

A thesis submitted to the Mechanical Engineering Department of Addis Ababa University (AAU) in partial fulfillment of the requirement of the degree of Master of Science (M.Sc.) in Mechanical Engineering (Specialization Mechanical Design).

By: - Behailu Mamo

Advisor: - Daniel Tilahun (Dr.)

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Approved by board Of Examiners

Dr. Daniel Tilahun	____________________	____________________
Chairman, SMIE	Signature	Date
Dr. Daniel Tilahun	____________________	____________________
Thesis Advisor	Signature	Date
Dr. Ing. Zewdu Abdi	____________________	____________________
Internal Examiner	Signature	Date
Dr. Ing. Tamirat Tesfaye	____________________	____________________
External Examiner	Signature	Date

Candidate's Declaration

I hereby declare that the work which is being presented in this thesis entitled " EXPERIMENTAL INVESTIGATION OF GEAR FAULT USING VIBRATION SIGNATURE " is original work of my own, has not been presented for a degree in any other university and that all sources of material used for the thesis have been duly acknowledged.

________________________ ________________________

Behailu Mamo (Candidate) Date

This is to certify that the above declaration made by the candidate is correct to the best of my knowledge.

________________________ ________________________

Dr. Daniel Tilahun (Thesis Advisor) Date

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ACRONYMS

FFT	*Fast Fourier Transform*
TSA	*Time synchronous average*
RMS	*Root Mean Square*
DFT	*Discrete Fourier Transform*
IFT	*Inverse Fourier Transform*
ADC	*Analog to Digital converter*
f_m	*Mashing Frequency*
RPM	*Revolution per Minute*
CPM	*Cycle per Minute*
Hz	*Harzi*
Max	*Maximum*
Min	*Minimum*
T	*Length of time Record*
X(f)	*Frequency Domain*
X(t)	*Time Domain*

LIST OF FIGURES

GLOSSARY

Accelerometer: - *A transducer with an electrical output directly proportional to the acceleration of a vibrating point in the direction in which the transducer is attached. The acceleration of a vibrating component is usually measured using a accelerometer.*

Amplitude: - *The magnitude of a signal or periodic motions e.g. the magnitude of the velocity of a vibrating body. Amplitude can be expressed in a variety of ways, the most common amplitude types being peak, peak-peak, and root-mean-square (rms).*

Fast Fourier transform:-*An algorithm for performing the DFT operation efficiently i.e. an algorithm for calculating a discrete spectrum from a discrete waveform. The term 'fast Fourier transform' is often abbreviated as 'FFT'. The FFT algorithm determines the frequencies and the amplitudes corresponding to the frequencies that are present in the waveform. Jean B. J. Fourier was a French mathematician who developed a means of expanding periodic functions in terms of harmonic functions, thereby contributing much to the fields of heat flow and vibration analysis.*

Fault frequency: - *The frequency of repeating forces caused by faulty machine components. Usually, the vibration spectrum shows spectral peaks at the fault frequencies and their harmonics. Some examples of fault frequencies are blade pass frequencies, rotor bar pass frequencies, ball pass frequencies, gear mesh frequencies, and the operating speed of the machine.*

FFT Analyzer: - *A spectrum analyzer that uses the FFT algorithm to calculate spectra from waveforms. Most spectrum analyzers are FFT analyzers.*

Filter: - *A device that allows certain frequency components of a signal to pass through, but blocks other frequency components. See also Band pass filter, high pass filter, and Low pass filter.*

Frequency: - *The number of periodic cycles or oscillations completed per unit time. Frequency is the reciprocal of period, and is usually expressed in Hz (which* *is equivalent to cps or cycles per second), cpm (cycles per minute), rad/s (radians per second), or derivatives of these units.*

Frequency domain: - *That which has a frequency axis as its x axis, or a set of frequency values to which are mapped a set of other values e.g. amplitude. A spectrum is a frequency domain chart i.e. a spectrum has a frequency axis as its x axis (and an amplitude axis as its y axis).*

Gear mesh frequency: - *The rate at which gear teeth contact. This is equal to the number of teeth on the gear multiplied by the rotation speed of the gear. A machine with gears will potentially vibrate at the gear mesh frequency.*

Harmonic: - *A spectral peak at a frequency that is a whole number multiple of the fundamental frequency or of the frequency of any excitation force present. A harmonic of a frequency 'n' times that of the fundamental frequency is called 'nX'. The frequency at which a harmonic occurs may or may not be a whole number multiple of the fundamental frequency e.g. the frequencies of harmonics of the ball pass and ball spin frequencies are not whole number multiples of the fundamental frequency. Most kinds of machine vibration are periodic and can be described as the sum of a series of sinusoids. The harmonics in a spectrum correspond to these sinusoids.*

Linear averaging: - *A commonly used method of averaging spectra or time-synchronized waveforms. The amplitude at each frequency or time value of the 'average' spectrum or waveform is the arithmetic mean of amplitudes of the individual spectra or waveforms at that frequency or time value.*

Machine vibration: - *The reciprocating or back-and-forth movement of a machine or machine component involving a continual interchange of kinetic energy and potential energy.*

Peak amplitude:-*The maximum amplitude attained by a vibrating object in a given time period e.g. the peak velocity amplitude of a vibrating object during a given time period is the maximum velocity achieved by the object during that time period. The terms 'peak amplitude' and 'zero-to-peak amplitude' (Ø-peak) are synonymous.*

Resonance: - *The situation where the vibration amplitude increases rapidly due to the natural frequency of the system is being excited by a periodic force that has a frequency similar to the natural frequency. A machine should never be operated continuously at its natural frequency. If it is necessary for a machine to operate at a frequency higher than its first natural frequency, the speed of the machine should be increased past the natural frequency as quickly as possible.*

Resonant frequency: - *The natural frequency of a system when there is no damping in the system. An n degree-of-freedom system has n resonant frequencies. See also damped natural frequency.*

Sidebands: - *Minor peaks, caused by amplitude or frequency modulation, located symmetrically on either side of spectral peaks.*

ABSTRACT

Gear Fault detection is an important problem associated with many rotating machinery that is coupled with gearbox. A damage not detected in time can result in severe damage to machinery, catastrophic injuries, and substantial financial losses. On the other hand, if a fault is detected in its early stages, corrective and preventive actions can be taken to avoid any significant machine failure. From this perspective; the main objective of this thesis work is to investigate the presence of initial gear flunk fault by using behavioral variation of vibration signature as a tool without shutting down of the gearbox.

This thesis outlines the experimental investigation and vibration signature analysis of gear (Number of teeth is 75) and pinion (Number of teeth is 25) flank fault (specifically on the pitch circle) of a single stage in line gearbox under different working condition i.e. with different input speed (600 RPM, 1000 RPM, and 1400 RPM),and with different external load (0 N-m, 1.5 N-m, 2.5 N-m, and 3.5 N-m) that is applied on the output shaft. The result is collected by using industrial oscilloscope in the form of frequency domain for both gear and pinion assembly with fault and without fault and the analysis is done by comparing the frequency domain result for faulted gear pair with a gear pair without fault.

With respect to this outline; the experimental is done for the above different conditions and the result shows that the presence of the gear flank fault is indicated by the difference in magnitude in the first and second meshing frequency and difference in magnitude and number of side bands around the meshing frequency. So that this thesis work concludes that if there is a fault in gear flank, the presence is identified from magnitude increment in 1st and 2nd meshing frequency of the gear pair and also increment in magnitude and number of the side bands around the 1st and 2nd meshing frequency.

Key words: *Gear fault, External Load, Vibration signature, Fault Indicator, Mashing Frequency, mashing vibration, side band.*

CHAPTER ONE

1. INTRODUCTION

1.1. Background

Gears are the means by which power is transferred from source to application. Gearing and geared transmissions drive the machines of modern industry. Moreover, gears move the wheels and propellers that transport us over the sea, on the land, and in the air. A sizable section of industry and commerce in today's world depends on gearing for its economy, production, and livelihood. [1]

The art and science of gearing have their roots before the Common Era .Yet many engineers and researchers continue to delve into the areas where improvements are necessary, seeking to quantify, establish, and codify methods to make gears meet the ever-widening needs of advancing technology; also to find the different mode of gear failure and its methodology of analysis to find the magnitude, type and position of failure. [1]

Surface wear is considered to be one of the three major failure modes in gear system, the other two being scoring and tooth bending.[2] The impact of wear on operational life of any gear system has far reaching consequences as gears are essential components of almost every power transmission system. Apart from the direct material loss, which leads to functional failure, surface wear also causes the gear system to change its vibration and noise characteristics significantly. [3] Surface wear can also affect the patterns of gear contact in such a way that it can alter stresses and load distribution to accelerate the occurrence of other failure modes.

Fault detection is the major problem associated with rotating machinery like a gear. [4] A damage not detected in time can result in severe damage to machinery, catastrophic injuries, and substantial financial losses. On the other hand, if a fault is detected in its early stages, corrective and preventive actions can be taken to avoid any significant machine failure.

It is well-known that the vibration characteristics of machinery like a gear box in operation are indicative of its mechanical condition. Analysis of this vibration signature provides information to identify and evaluate potential failure in its mechanisms. [5]

With the development of sophisticated and computer based instruments in recent years, the technique of vibration signature analysis has come into prominence. [6, 7] This technology is increasingly finding its use in modern highly stressed high speed machinery. With respect to this; the vibration signature analysis [1]

- ✓ serves as a diagnostic tool for machinery fault detection and prediction,
- ✓ Provides the maintenance engineer with an early warning of failures and helps in maintenance planning,
- ✓ Provides a cost effective non-destructive test technique,
- ✓ is used as a quality control tool for new as well as post refit machinery,
- ✓ helps in extending or/and shorting the time between machinery overhauls with confidence, and
- ✓ Helps in assessing vibratory contribution to airborne noise.

1.2. Statement of problem

The advancement in research has aimed at finding a reliable monitoring strategy for gear transmission systems. This is because the previous method called visual inspection and/ or physical assessment alone no longer provide adequate early warning to any emerging problem in a gear system to restrict down the maintenance time, maintenance cost and propagation of damage. In addition it is quite impossible to examine gear transmissions on-line during operation by using visual inspection method. The visual inspections are used mainly after the machine failure has already been experienced.

On the other hand, some of on-line condition monitoring systems for rotor gear systems often fails to provide sufficient time between warning and failure such that safety procedures can be implemented. [8] At times, a small fault in the gear system can quickly develop into a dangerous failure mode without any notable signs. In addition, inaccurate interpretation of operational conditions may often result in false alarms and unnecessary repairs and downtime. All these needs call for the development of an accurate early fault detection system that permits on-line inspections without costly shutdowns during machine operations.

Vibratory motion is a characteristic of all types of machinery, especially rotating machinery. There is great interest in measuring and quantifying this motion because it is indicative of the state and health of the machinery. Measuring vibrations is one part of a so called Condition Based Maintenance system in which repair and maintenance decisions are based not on machine hours or time but on the condition or state of the machinery. The so called “vibration signature” of the device will tell the operator whether the device is operating properly and can offer an early warning if the machinery is beginning to fail. And vibration signature analysis does not require shutdown of the rotating machinery, and can be carried out on-line by a computer-based machine health monitoring system. The acquired vibration signals are processed by a variety of methods to identify the faults of gears. However, the interpretation of the vibration signals, in some cases, may require extensive experiences and knowledge in vibration signal interpretations for an accurate identification of the faults and their corresponding locations.

Hence in this research work effort is made to generate vibration data from experimental gear system with and without fault (the fault is generated artificially, by considering the most commonly created type of spur gear fault in the practical condition); with external load and without external load and also data obtained is recorded and analyzed to investigate the presence of fault on gear flank. In addition a comparison will be done between different vibration signals to propose the most efficient way for finding gear fault.

1.3. Objectives of the thesis

1.3.1. General Objective

The general objective of this study is to develop a comprehensive experimental procedure to investigate gear fault using a behavioral variation of vibration signature as a tool.

1.3.2. Specific Objective

The specific objectives of the research will be to:

- ✓ conduct an experimental investigation for condition monitoring of a single stage gear box with faulty gear and pinion by using vibration signature.
- ✓ analyze the variation of the vibration signature with faulted gear and external loads as the gear box run with different speed.
- ✓ identify different fault indicators which are used for identification of gear fault within the gearbox.
- ✓ recommend the type of signal which are used to show the presence of gear fault.

1.4. Research methodology

The methods employed to achieve the above objective are

a) Literature Survey

Literature survey of relevant material on different types of a gear fault and vibration signature analysis of faulted gear has been done. The literatures available are from electronic media, journals, and books. In addition, secondary data are referred from previous related research studies, existing statistical data, etc.

b) Data collection from experimental investigation

After the gears manufactured without any fault, the gears is assembled on the main test setup (PT 500) to find the initial data. Then an artificial fault is introduced to investigate the variation of the vibration signature due to this fault with different test conditions (Input speed and external load).

c) **Experimental Data analysis and discussion**

The data collected from the experiments is analyzed using frequency domain analysis techniques.

1.5. Conclusion Recommendation and future work

Finally, conclusion is drawn based on the analysis done on the data and alternatives are forwarded to achieve the goal as a recommendation

CHAPTER TWO

2. LITERATURE REVIEW

2.1. Condition Monitoring Techniques

All machines with moving parts give rise to sound and vibration. Each machine has a specific vibration signature related to the construction and the state of the machine. If the state of the machine changes, the vibration signature will also changes. A change in the vibration signature can be used to detect incipient defects before they become critical. This is the basics of many condition monitoring methods. Condition monitoring can save money through increased maintenance efficiency and by reducing the risk of serious accidents by preventing breakdowns. [9, 10]

Both consciously and subconsciously, operating engineers routinely use at least four of the five human senses to varying degrees to assess the condition of the machinery under their care. Sight, hearing, touch, and (although to a lesser degree) smell are useful in monitoring overall plant status, but sight and smell become virtually useless in evaluating the condition of an individual machine until long after an abnormality, or an abnormal trend, has become quite obvious. [11, 12] Normally they are limited to detecting the existence of a problem which has advanced to a point where some form of corrective action is called for without delay, whether it be as minor as a simple adjustment that may be made with the unit running, or as major as the immediate shutdown of a piece of equipment. On the other hand, hearing and touch are more sensitive to small variations in operating conditions and, with respect to an individual machine, the onset of specific problems such as pump cavitation, bearing defects, drive belt defects, mechanical looseness, and the like, may be detected. In general, though, what is being felt or heard is the *vibration*, or *the sound produced by the vibration*, which results from some specific change which has occurred in either the operating parameters or the material condition of the machine components or their alignment. Unfortunately, even with the significant dynamic range and filtering capabilities of the human ear which enable the selective identification of small signals (sounds) in the presence of large random signals (background noise), the changes so detected may be due to specific maladies or component defects which are well on their way toward necessitating an unscheduled shut down for repairs. [13]

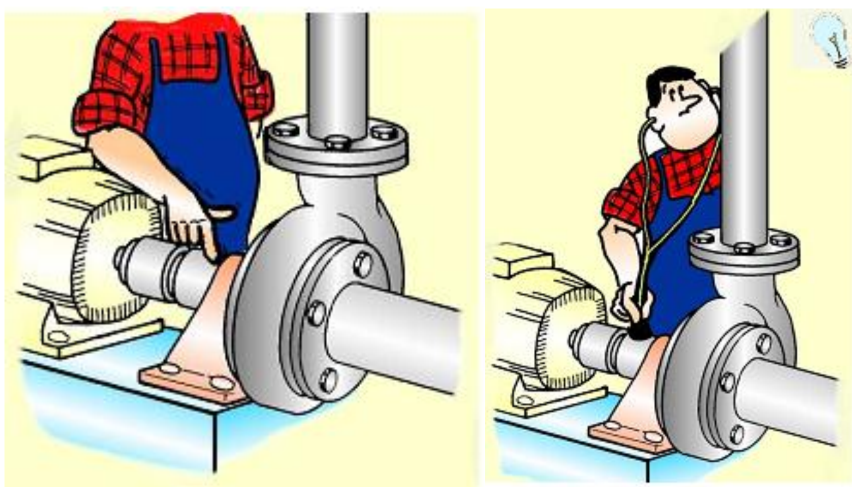

Figure 2.1: - Vibration Measurement in the past (& still today)

The basics of condition monitoring has been known and used in practice for a long time. An experienced operator can monitor the state of a machine by listening to the vibrations via a screwdriver shaft. However, it takes years for an operator to develop the necessary skills and some defects appear very seldom. It would be an advantage if the monitoring process could be automated.

A large number of methods for automated condition monitoring have been developed during the last decades. These methods usually include three basic steps; measurement of a physical quantity, extraction of suitable state features with advanced signal processing techniques and comparison of the state features to reference values. [14]

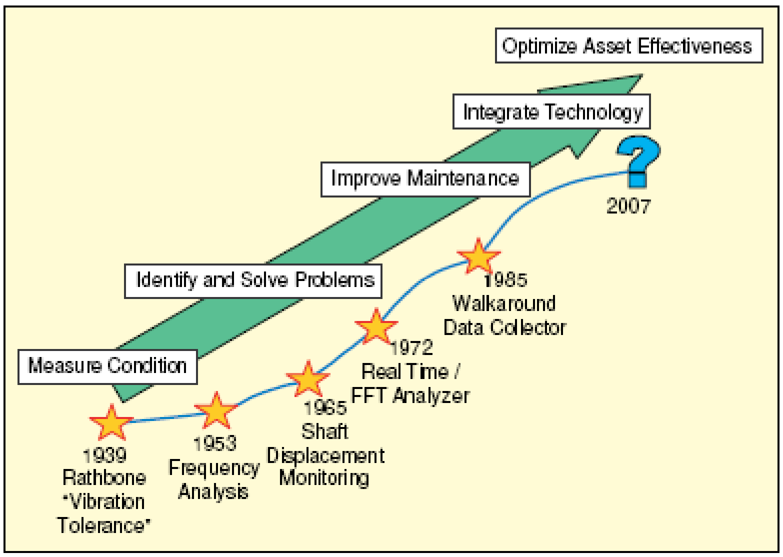

Figure 2.2:- Major developments in the progression of vibration measurement, monitoring and condition assessment.

Gears are among the most important machine components in the vast majority of rotating machines and exigent demands are made upon their carrying capacity and reliability. Generally, gear cannot rotate forever. It often works well in non-ideal conditions, but sometimes minor problems cause the gears to fail quickly and mysteriously without any notable warning. The gear failures are mainly resulted from excessive wear or damage in the gear flank and face. [15]

Excessive wear can be an indication of component failure, and its detection will be a valuable tool in diagnostics and prognostics. Especially for aircraft applications, such as helicopter transmissions and engines, where safety and reliability are crucial, accurate fault detection will be valuable. So in the last two decades, lots of works [16-18] have been done with this strong need for the development of a reliable damage detection procedure that can be effective in long term management in rotating shaft.

Traditionally, optical/visual inspection has been used as one of the basic tools in discovering deterioration and damage exhibited on the surface as well as the inner integrity of the shaft member [19-21]. The most commonly used optical procedure is the visual inspection procedure conducted by maintenance personnel. However, due to the limited capabilities of human eyes, damage cannot be easily detected on-site even with experienced skillful technicians, in addition a complete shutdown and disassembly of the machine is often necessary. At times, for the purpose of double confirmation, material samples are extracted from the structures and tested under laboratory environment to determine the reliability of the inspection. In the meantime, X-ray and thermal imaging procedures have been applied successfully in some smaller shafts [22, 23]. However, the applications of these procedures are very limited and they are very expensive to implement on site.

And in the last twenty years, acoustic emission (AE) procedure has been widely regarded as an effective approach in non-destructive monitoring and damage detection. It has been extensively applied for the location and monitoring of fatigue cracks in a variety of metal structures [24-26]. Acoustic emissions are essentially elastic stress waves, generated by a rapid release of energy from a localized source within a stressed material [27, 28], which can be detected by an arrangement of sensors. The accuracy and the usable range of the sensors depend quite heavily on the applied frequency/wave length and the input/output power of the actuator unit. Although traditional AE monitoring, such as ultrasound, can accurately pinpoint incipient damage in structural members, its application on rotating structures has been limited due to the requirement for a large number of connecting wires onto the rotating shaft.

As of Ruiwei Wu's study [29], another commonly used fault prediction approach based on condition monitoring is the oil-debris monitoring method. Gear and bearing failures in transmissions produce significant wear debris in oil lubrication systems. The debris mass measured by oil debris monitor will show an increase when damage began to occur. A variety of on-line methods are available for oil debris monitoring [30, 31]. Some of the principles of the various methods are Ferrography [32, 33], ultrasonic [34], and X-ray fluorescence (XRF) [35].

With the increasing requirements for long life and safe operation in mechanical systems, signature analysis of machine vibration signals [36-47] is one of the advanced fault identification procedures used in any rotating mechanical systems. The acquired machine vibration/acoustic signature is compared with a signature data bank of the healthy machine allowing the detection of abnormalities in the input signal.

2.2. Vibration analysis techniques

2.2.1. General Concept of Vibration

A concept that is basic to the understanding of vibrations is the concept of source-path-receiver [13]. The oscillatory motion of a machine represents its response to exciting forces whose sources may be internal or external to the machinery unit. Examples of internal sources are rotor imbalance, coupling or bearing misalignment or worn or damaged mating components such as gears; all representatives of material or alignment defects which may be corrected to reduce the amplitude of the forces produced by them. Examples of external sources are load variations, flow conditions, or vibration of adjacent structures or equipment; all representative of problems which cannot be attributed to any physical defect of the machinery unit itself. The transmission of the forces from their sources follows one of two paths of prime interest to the vibration analyst, one in the form of the mechanical transmission of forces from one component to the next, the other in the form of the acoustic emissions emanating from each component which is set into vibratory motion. This describes the two basic types of signals that are measured and analyzed, mechanical and acoustic, which coincides with the two general fields that have developed. The instrumentation used is different for each of these two signal types, but the essentials of the analyses of the signals follow similar paths.

Both the above signals analysis procedure does not require a shutdown of the rotating machinery, and can be used as an on-line diagnostic and trend monitoring tool. And, vibration signal analysis is a logical choice of a field to explore and in which to develop measurement and analysis techniques which can serve as natural extensions of those basic human senses which are inherently more responsive to, hence more informative about, machinery condition [48, 49]. These methods can be classified into time domain analysis and frequency domain analysis.

2.2.2. Time domain analysis

Time domain analysis is used for the early detection of machine faults [50 - 56]. The digitized time domain vibration signal is averaged over a large number of cycles, synchronous with the running speed of a particular machine element. This time synchronous averaging removes the background noise, leaving a more accurate estimate of the vibration signal of the component of interest. This process can be repeated for each machine element in the systems to provide the vibration signal to extract specific fault information. In most machinery, vibration signals are often characterized by a variation of both amplitude and frequency with respect to time. For most mechanical faults, both the amplitude and frequency contents of the signal will shift during the period in which the damage is generated. The nature of these short-term variations in the time signal could prevent an accurate indication of the characteristics of the mechanical fault present.

The time domain methods analyze the amplitude and phase information of the vibration time signal to detect the fault of gear-rotor-bearing system [42, 51, 57-59]. Some are using the difference of vibration amplitude and phase due to the damage of components to detect faults at gears and bearings. McFadden [51] suggested the use of phase and amplitude demodulation of the dominant meshing frequency residual for tooth crack detection, which has proved to be a very successful technique in a number of cases. Time synchronous average (TSA) is a signal averaging process over a large number of cycles, synchronous with the running speed of a specific shaft in the gearbox. Advanced gear tooth or bearing damage can often be identified readily by the direct inspection of the TSA trace [58]. In addition, kurtosis of the phase modulation as well as its derivatives can also be used for gear and/or bearing fault diagnosis. Ismail et al. [59] used kurtosis of beta function to emphasize transients generated by a tooth crack. They also proposed a statistical index to assess gear and bearing damage.

2.2.3. Frequency domain analysis

For frequency analysis, the amplitude spectrum of the measuring vibration signal is calculated and displayed in a continuous manner [60, 61]. This method is particularly powerful in identifying vibrations due to various mechanical elements in the system at their operating frequencies. For example, a gear box will experience vibration at the fundamental and harmonic frequencies of the gear tooth meshing frequencies, which are usually different for each pair of gears in the gearbox. The changes in the frequency spectrum, such as an increase in the vibration amplitude at a particular frequency, or the appearance of modulation sidebands about that frequency, can often be identified and related to a signal element in the complex system. Furthermore, abnormal vibration signal, caused by wear, deformation of the machine element, ‘’ghost component” etc., which are sometimes difficult to quantify in the time domain, can be readily identified in the frequency spectrum. However, the spectral analysis technique is difficult to apply in a highly complex system, such as a helicopter gearbox, which may have as many as 30 gears and 50 bearings [41].The large number of spectral lines which these elements produce during normal operation can make it difficult to even detect changes in the spectrum, let alone to identify the sources by which these changes are generated. In addition, the modulation of the vibration signal of early fatigue cracks in the mechanical elements often yield sidebands which extended over a broad frequency range with low amplitude. Under these circumstances, the background noise level in the spectrum can easily mask the presence of the sidebands, and mechanical defects in the system may go undetected.

Spectral analysis is the classical gear/bearing diagnostic technique. By comparing the spectrum of a damaged gearbox with its reference spectrum in the healthy condition, some gear/bearing faults could be detected [42, 62-66]. The frequency domain methods mainly apply numerical Fast Fourier Transform (FFT) to the vibration signals to obtain the frequency spectrum [49-64]. Others use the difference of power spectral density of the signal due to the fault of gear and bearing to identify the damage of elements [65]. Cepstrum is the inverse Fourier transform of the logarithmic power spectrum [66] & it highlights periodicity in the spectrum; therefore, a periodic signature in the spectrum caused by a gear/bearing fault could be recognized. For complicated gear systems, however, it is difficult to identify faults from the spectrum or the cepstrum because of the large number of components involved.

From the studies of former researches that are discussed above, it can be noticed that there is a limitation of the study on variation of vibration signal due to gear fault and also amplitude type and magnitude which shows the presence of fault clearly within the gear box at the initial stage. Therefore this paper mainly focuses on the effect of gear fault on the vibration signal and comparison of different vibration frequency that is used as an indicator for presence of fault with load and without load within the gear box.

2.3. Review of vibration analysis techniques

2.3.1. Introduction

Gear wear produces changes in the *vibration signatures* measured by accelerometers installed on gearboxes. In practice, direct comparisons of current vibration signatures with previous signatures are not effective, due to large variations. Instead, more useful techniques that involve the extraction of *features* from the vibration signature data are being used, based on some statistical measurement of vibration energy.

Feature extraction is the process of extracting *condition indicators* about the system input which are more informative than evaluating the raw input itself. It is a parameterization process, which often reduces the data volume. Feature extractors output only the information relevant for detecting the failure modes to which the associated components are susceptible.

Before any feature can be calculated on the raw vibration data, the data must be conditioned or preprocessed. Conditioning may range from signal correction, based on the data acquisition unit and amplifiers used, and mean value removal to time-synchronous averaging and filtering. A variety of signal processing techniques are used based on the feature being implemented. From this Variety of signal processing techniques the time-domain analysis and frequency-domain analysis is predominant.

Time domain and frequency domain are two ways of looking at the same dynamic system. They are interchangeable, i.e., no information is lost in changing from one domain to another.

2.3.2. Time domain analysis

2.3.2.1. Waveform analysis

Prior to the commercial availability of spectral analyzers, almost all vibration analysis was performed in the time domain. By studying the time domain waveform using equipment such as oscilloscopes, oscillographs, or 'vibrographs', it was often possible to detect changes in the vibration signature caused by faults. However, diagnosis of faults was a difficult task; relating a change to a particular component required the manual calculation of the repetition frequency based on the time difference observed between feature points.

2.3.2.2. Time domain signal metrics

Although detailed study of the time domain waveform is not generally used today, a number of simple signal metrics based on the time domain waveform still have widespread application in mechanical fault detection; the simplest of these being the peak and RMS value of the signal which are used for overall vibration level measurements.

2.3.2.3. Peak

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

$$peak = \frac{1}{2}\left[\max\left(x(t)\right) - \min\left(x(t)\right)\right] \qquad (2.1)$$

*The **peak-to-peak value** is valuable in that it indicates the maximum excursion of the wave, a useful quantity where, for example, the vibratory displacement of a machine part is critical for maximum stress or mechanical clearance consideration*

2.3.2.4. RMS

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

$$RMS = \sqrt{\frac{1}{T}\int_0^T (x(t) - \bar{x})^2\, dt} \quad \text{(2.2)}$$

Where T is the length of the time record used for the RMS calculation and $\bar{x}$is the mean value of the signal:

$$\bar{x} = \frac{1}{T}\int_0^T x(t)dt \quad \text{(2.3)}$$

For discrete (sampled) signals, the RMS of the signal is defined as:

$$RMS = \sqrt{\frac{1}{N}\sum_{n=0}^{N-1} (x(n) - \bar{x})^2} \quad \text{(2.4)}$$

$$\bar{x} = \frac{1}{N}\sum_{n=0}^{N-1} x(n) \quad \text{(2.5)}$$

The RMS of the signal is commonly used to describe the 'steady-state' or 'continuous' amplitude of a time varying signal.

The root mean square (RMS) is a measure of the power content of the vibration signal. It is a general fault indicator, which provides no information on which component is failing, and shows no appreciable changes in the early stages of gear damage if the analysis is time domain. Alone, it can be effective only in detecting major out-of-balance.

2.3.2.5. Crest Factor

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

$$Crest\ Factor = \frac{peak}{RMS} \quad \text{.......... (2.6)}$$

The crest factor is often used as a measure of the 'spikiness' or impulsive nature of a signal. It will increase in the presence of discrete impulses which are larger in amplitude than the background signal but which do not occur frequently enough to significantly increase the RMS level of the signal.

In early stages of damage, there is no change in the RMS value, while the peak value increases, therefore the Crest Factor increases. As the damage progresses, the RMS value increases and the Crest Factor decreases. It is used to detect changes in the signal pattern due to tooth breakage, but is not considered a very sensitive indicator.

2.3.2.6. Kurtosis

Kurtosis is the normalized fourth statistical moment of the signal. For continuous time signals this is defined as:

$$Kurtosis = \frac{\frac{1}{T}\int_0^T (x(t) - \bar{x})^4 dt}{(RMS)^4} \quad \text{.......... (2.7)}$$

For discrete signals the kurtosis is:

$$Kurtosis = \frac{\frac{1}{T}\sum_{n=0}^{N-1}(x(n)-\bar{x})^4}{(RMS)^4} \quad \text{...} (2.8)$$

The kurtosis level of a signal is used in a similar fashion to the crest factor that is to provide a measure of the impulsive nature of the signal. Raising the signal to the fourth power effectively amplifies isolated peaks in the signal.

It measures the relative peakeness or flatness of a distribution as compared to a normal distribution. Kurtosis provides a measure of the size of the tails of distribution and is used as an indicator of major peaks in a set of data. As a gear wears and eventually a tooth breaks, this feature should signal a defect due to the increased level of vibration.

2.3.3. Frequency domain analysis

Any real world signal can be broken down into a combination of unique sine waves. Every sine wave separated from the signal appears as a vertical line in the frequency domain. Its height represents its amplitude and its position represents the frequency. The frequency domain completely defines the vibration. Frequency domain analysis not only detects the faults in rotating machinery but also indicates the cause of the defect

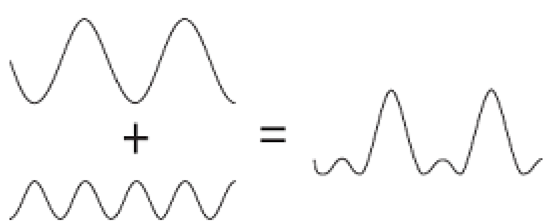

Figure 2.3: The sine component of real world signal

Now view the three-dimensional graph along the time axis, as in figure 3.2. Here we have axes of amplitude versus frequency. This is what is called the frequency domain. Every sine wave we separated from the input appears as a vertical line. Its height represents its amplitude and its position represents its frequency. We know each line represents a sine wave and so we have uniquely characterized our input signal in the frequency domain. This frequency domain representation of our signal is called the spectrum of the signal. Each sine wave line of the spectrum is called a component of the total signal.

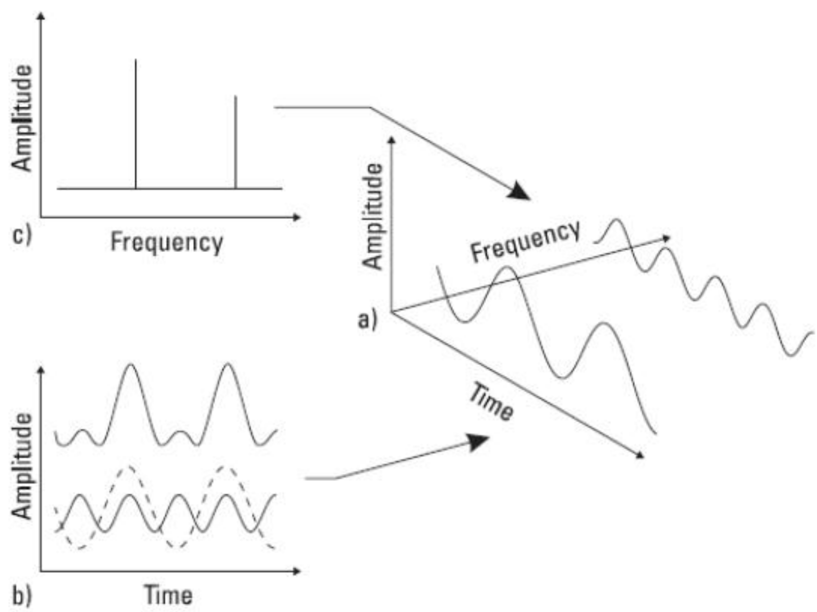

Figure 2.4: - Fourier Transformation of time domain signal to frequency domain.

To analyze the frequency distribution of vibration waveforms it is necessary to transform the time domain signal into frequency domain with the help of Fast Fourier Transformation (FFT)

2.3.4. Spectral analysis

Spectral (or frequency) analysis is a term used to describe the analysis of the frequency domain representation of a signal. Spectral analysis is the most commonly used vibration analysis technique for condition monitoring in geared transmission systems and has proved a valuable tool for detection and basic diagnosis of faults in simple rotating machinery [54]. Whereas the overall vibration level is a measure of the vibration produced over a broad band of frequencies, the spectrum is a measure of the vibrations over a large number of discrete contiguous narrow frequency bands.

The fundamental process common to all spectral analysis techniques is the conversion of a time domain representation of the vibration signal into a frequency domain representation. This can be achieved by the use of narrow band filters or, more commonly in recent times, using the discrete Fourier Transform (DFT) of digitized data. The vibration level at each 'frequency' represents the vibration over a narrow frequency band centered at the designated 'frequency', with a bandwidth determined by the conversion process employed.

2.3.5. Conversion to the frequency domain

The frequency domain representation of a signal can be described by the Fourier Transform [54] of its time domain representation

$$x(f) = \int_{-\infty}^{\infty} x(t) e^{-j2\pi f t}\, dt \qquad \ldots\ldots (2.9)$$

The inverse process (Inverse Fourier Transform [54]) can be used to convert from a frequency domain representation to the time domain

$$x(t) = \int_{-\infty}^{\infty} x(f)e^{j2\pi ft}df \qquad \text{.. (2.10)}$$

2.3.6. FFT Analyzers

Most modern spectrum analyzers use the Fast Fourier Transform (FFT) [55], which is an efficient algorithm for performing a Discrete Fourier Transform (DFT) [53, 54] of discrete sampled data.

The Discrete Fourier Transform is defined as [61]

$$X(m) = \frac{1}{N}\sum_{n=0}^{N-1} x(n)e^{-j2\pi\frac{nm}{N}} \qquad \text{.. (2.11)}$$

And the Inverse Discrete Fourier Transform [61] is

$$X(n) = \sum_{m=0}^{N-1} x(m)e^{j2\pi\frac{nm}{N}} \qquad \text{.. (2.12)}$$

2.3.7. Fast Fourier Transformation (FFT)

The Fast Fourier Transform (FFT) resolves a time waveform into its sinusoidal components. The FFT takes a block of time-domain data and returns the frequency spectrum of the data. The FFT is a digital implementation of the Fourier transform. Thus, the FFT does not yield a continuous spectrum. Instead, the FFT returns a discrete spectrum, in which the frequency content of the waveform is resolved into a finite number of frequency lines, or bins.

2.3.8. The Anatomy of the FFT Analyzer

The FFT Analyzer can be broken down into several pieces which involve the digitization, filtering, transformation and processing of a signal.

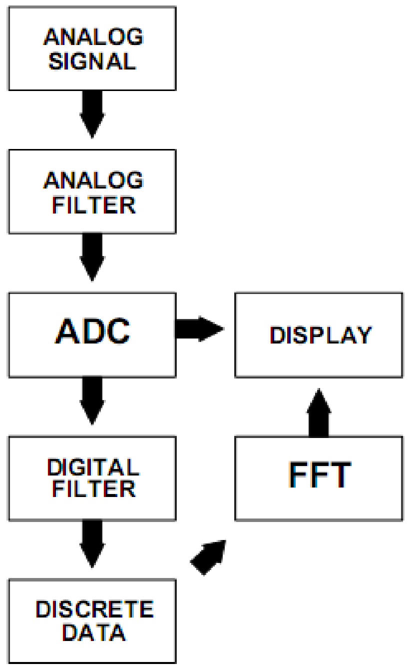

Figure 2.5: - The Anatomy of the FFT Analyzer.

The sampling process used to convert the continuous time signal into a discrete signal can cause some undesirable effects. Some of that is the following:

2.3.8.1. Leakage

When the measured signal is not periodic in the sample interval, incorrect estimates of the amplitude and frequency occur. This error is referred to as *leakage*.

Basically, the actual energy distribution is smeared across the frequency spectrum and energy leaks from a particular Δf into adjacent Δf s. Leakage is probably the most common and most serious digital signal processing error. Unlike aliasing, the effects of leakage cannot be eliminated.

In order to better satisfy the periodicity requirement of the FFT process, time weighting functions, called ***windows***, are used. Essentially, these weighting functions attempt to heavily weight the beginning and end of the sample record to zero - the middle of the sample is heavily weighted towards unity. The three most common type Windows are *Rectangular windows*, *Hanning windows* and *Flattop windows*.

- ✓ **Rectangular Windows** - Unity gain applied to entire sample interval; this window can have up to 36% amplitude error if the signal is not periodic in the sample interval; good for signals that inherently satisfy the periodicity requirement of the FFT process

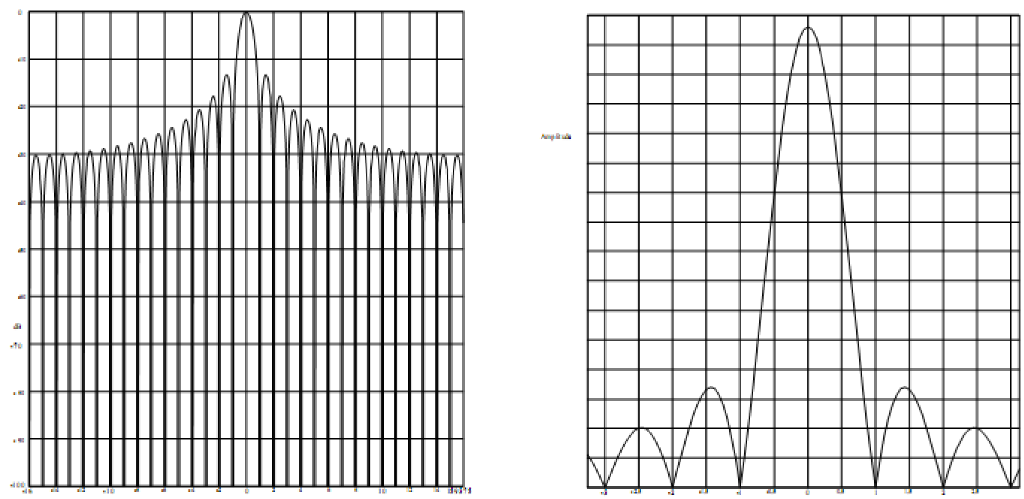

Figure 2.6: - Rectangular Windows.

- ✓ **<u>Hanning Windows</u>**- Cosine bell shaped weighting which heavily weights the beginning and end of the sample interval to zero; this window can have up to 16% amplitude error; the main frequency will show some adjacent side band frequencies but then quickly attenuates; good for general purpose signal applications

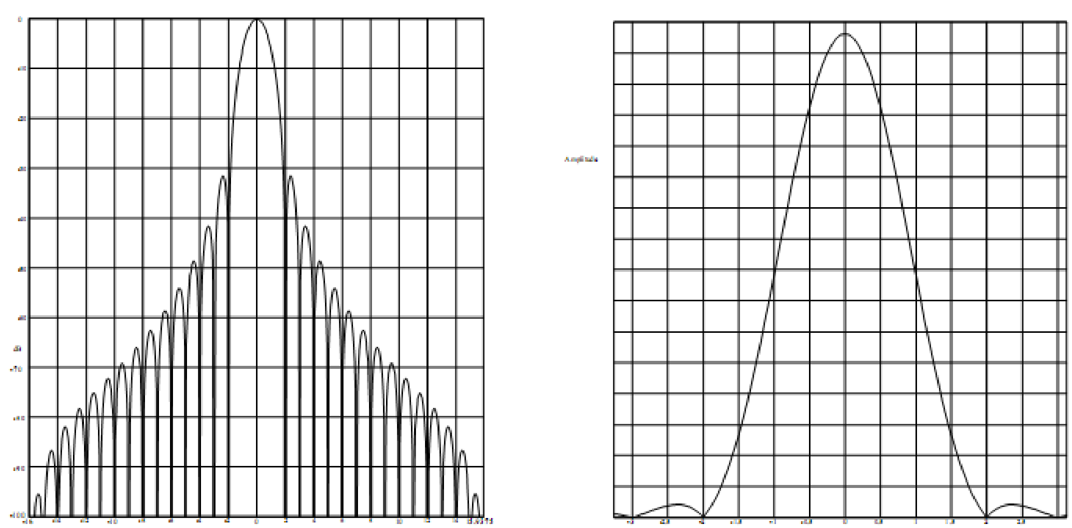

Figure 2.7: - Hanning windows.

- ✓ **<u>Flat Top Windows</u>** - Multi-sine weighting function; this window has excellent amplitude characteristics (0.1% error) but very poor frequency resolution; very good for calibration purposes with discrete sine

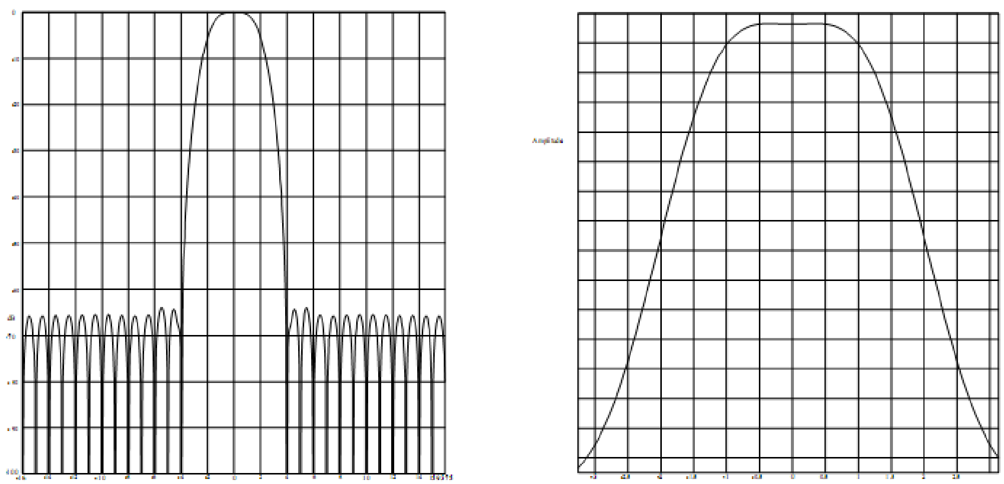

Figure 2.8: - Flattop windows.

2.3.8.2. Aliasing

Frequencies which are greater than half the sampling rate will be aliased to lower frequencies due to the stroboscope effect. To avoid aliasing, an analogue low-pass 'anti-aliasing' filter is used prior to sampling to ensure that there are no frequencies above half the sampling rate.

So that; from all of the above discussion the vibration signature analysis for this study is takes place by converting of the time domain to frequency domain with Fast Fourier Analysis (FFA) by using an Industrial Oscilloscope coupled with spectrum analyzer and anti-aliasing filter with Hanning Windows.

2.4. Gear and gear fault

2.4.1. Gear tooth action

For spur gears, the terminology of gear teeth is given in Figure 2.9. Gear calculations are based on the theoretical *pitch circle*. The *operating pitch circles* of a pair of gears in mesh are tangent to each other. The *clearance circle* is tangent to the addendum circle of the mating gear.

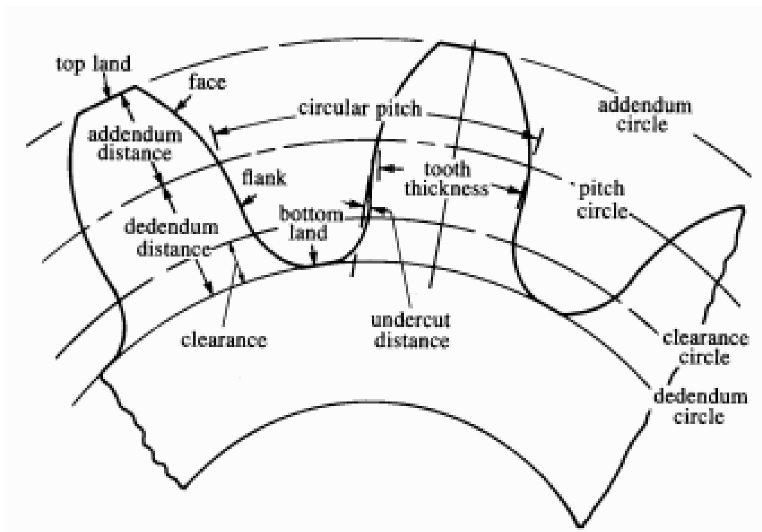

Figure 2.9:- Spur gear terminology

Additional terminology is shown in figure 2.10. Here the *pinion* rotates clockwise and drives a *gear* in a counterclockwise direction. *OP* is the *line of centers*, connecting the rotation axes of the meshing gears. The *pitch circles* are tangent at *P*, the *pitch point*.

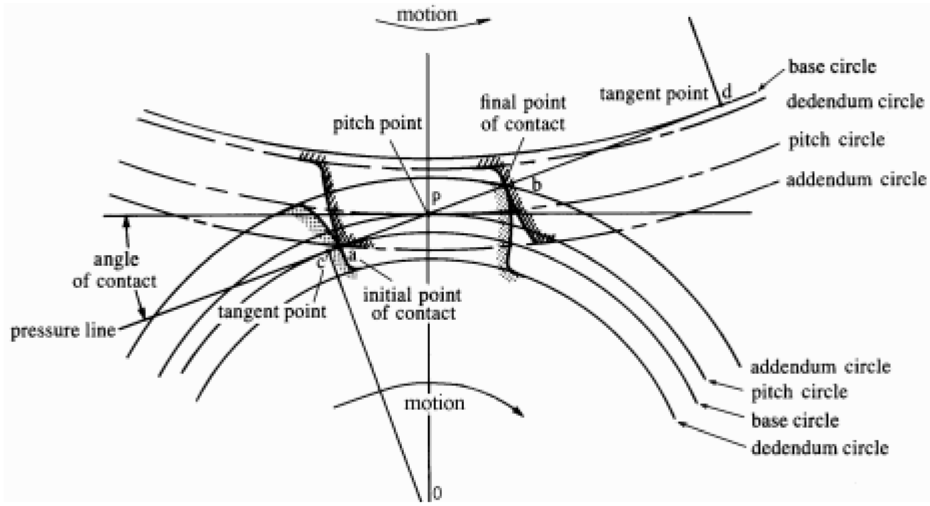

Figure 2.10:- Mashing Gear

The resultant force vector between a pair of meshing gears acts along the *pressure line* (also called *line of action* or *generating line*). The pressure line is tangent at point's 'c' and 'd' to the *base circles*.

The angle between the pressure line and the common tangent to the pitch circles is the *pressure angle*, and it has values of 20 deg. The operating diameters of the pitch circles depend on the center distance used in mounting the gears, but the base circle diameters are constant and depend only on how the tooth forms are generated, because they form the *base* of the starting point on the *involute* profile.

Point '*a'* is the *initial point of contact*, where the flank of the pinion driving tooth just touches the tip of the driven tooth. This point is located at the intersection of the addendum circle of the gear with the pressure line. Should point '*a'* occur on the other side of point '*c'* on the pinion base circle, the pinion flank would be undercut during the generation of the profile.

Point ‘*b’* is the *final point of contact,* when the tip of the driving tooth just leaves the flank of the driven tooth. This point is located at the intersection of the addendum circle of the pinion with the pressure line. For no undercutting of the gear teeth, point ‘*b’* must be located between the pitch point ‘*P’* and the tangent point ‘d’ on the base circle of the gear.

Line *aP* represents the *approach* phase of tooth contact, while line *Pb* is the *recess* phase. Tooth contact throughout the line of action *ab* is by both sliding and rolling, except for an instant at *P* when the contact is pure rolling.

Sliding gives rise to friction forces that vary in magnitude and direction as the teeth go through the meshing cycle. During the approach action, the flank of the pinion tooth is sliding down the face of the gear tooth, producing a frictional force oriented upwards in figure 2.10. During the recess action, the face of the pinion tooth is sliding up the flank of the gear tooth, and the resulting friction force exerted by the pinion against the gear is oriented in opposite direction (downwards in figure 2.10). *Friction forces produce a characteristic type of gear wear.*

The zone of action of a pair of meshing gear teeth is shown in figure 2.11. The *arc of action AB* is the sum of the *arc of approach AP* and the *arc of recess PB.*

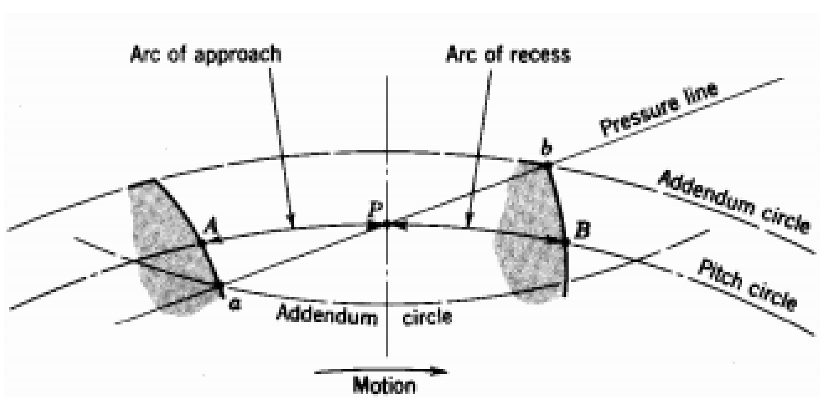

Figure 2.11:- The zone of action of a pair of meshing gear teeth

In the unlikely situation in which the arc of action is exactly equal to the circular pitch, when one pair of teeth are just beginning contact at *a*, the preceding pair will be leaving contact at *b*. Thus, for this special condition, there is never more or less than one pair of teeth in contact.

If the arc of action is greater than the circular pitch (their ratio is called the *contact ratio*) but less than twice as much, then when a pair of teeth come into contact at *a*, another pair of teeth will be still in contact somewhere along the line of action *ab*. Thus, for a short period of time, there will be two pairs of teeth in contact, one near the vicinity of *A* and another near *B*. As the meshing proceeds, the pair near *B* must cease contact, leaving only a single pair of contacting teeth, until the procedure repeats itself. Gears are not generally designed having contact ratios less than 1.20 because inaccuracies in mounting might reduce the contact ratio even more, increasing the possibility of impact between the teeth as well as an increase in the noise level. A contact ratio of 1.2 means 80 percent of the time – single tooth contact, and 20 percent of the time – double tooth contact.

The *contact ratio* is equal to the length of the line of action *ab* divided by the *base pitch*. The *base pitch* is the distance, measured on the line of action, from one involute to the next corresponding involute.

In f*igure 2.12a the* mating teeth of the meshing spur gears are in contact at the *pitch point.* The number of tooth pairs in contact is shown in f*igure 2.12b*. The transition from single to double tooth contact produces variations in the mesh stiffness.

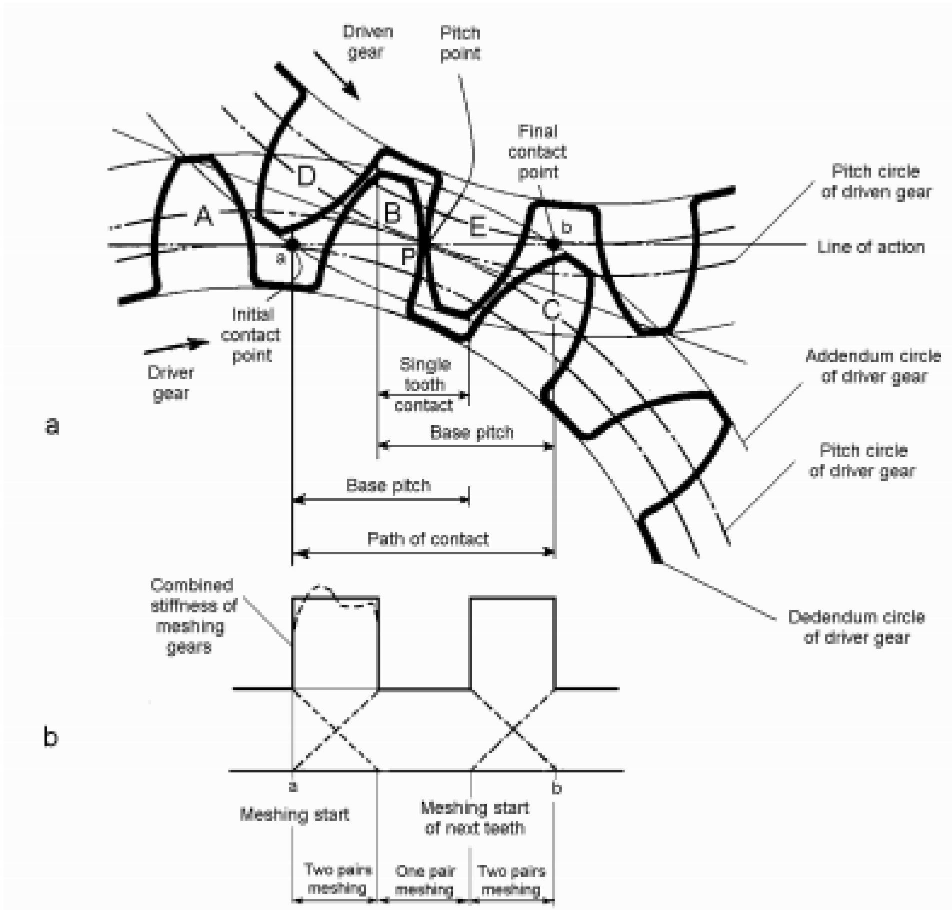

Figure 2.12:- the mating teeth of the meshing spur gears.

The tooth involute profiles are designed to produce a constant angular velocity ratio during meshing. Ideally, when two gears are in mesh, their pitch circles rolls on one another without slipping. Denoting the pitch radii by r_1 and r_2, and the angular velocities by ω_1 and ω_2 the pitch line velocity is

$$v = |r_1\omega_1| = r_2\omega_2 \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (4.1)$$

Thus, the gear ratio is

$$i = \left|\frac{\omega_2}{\omega_1}\right| = \frac{r_1}{r_2} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (4.2)$$

In order to transmit uniform rotary motion during meshing, a pair of gears must meet the following requirements (*Figure 2.12*):

a) the pitch point *P* must remain fixed on the line of centers O_1O_2 ;
b) the lines of action for every instantaneous point of contact *e* must pass through the same point *P ;*
c) the generating (pressure) line must be always tangent to the base circles and normal to the involute profiles at the point of contact *e*.

Deviations from the above requirements produce *transmission errors* giving rise to vibrations.

Changing the center distance, the above requirements are still satisfied, because it has no effect on the base circles used to generate the tooth profiles. Increasing the center distance increases the pressure angle and decreases the length of the line of action, but the teeth are still conjugate, and the angular velocity ratio is not changed. This increase creates two new operating pitch circles having larger pitch diameters but remaining tangent to each other at the pitch point.

Interference might be produced by the contact of portions of tooth profiles which are not conjugate. It is eliminated by *undercutting* (which weakens the teeth), by using a larger pressure angle, or by increasing the number of teeth, hence increasing the pitch line velocity and making the gears noisier, which is an unacceptable solution.

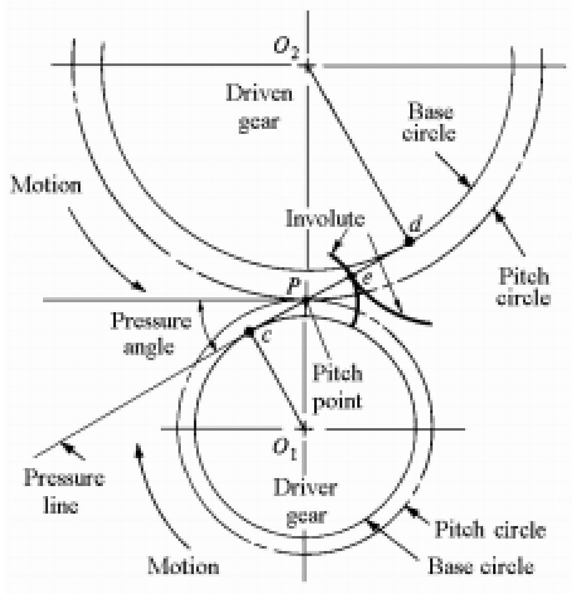

Figure 2.13:- pressure line and involute profile.

2.4.2. Gear vibrations

Rigid and geometrically perfect gears do not produce vibrations. Deviations from the ideal tooth profile and gear geometry generate vibrations whose measurement and analysis can help in diagnosing gearbox faults. The main sources of such deviations are the tooth deflection under load, the wheel distortion during heat treatment or gearbox assembly, and the geometrical errors in the profile itself, resulting from the gear cutting process and wear.

2.4.3. Tooth engagement

Assuming the teeth to be perfectly formed, equally spaced, perfectly smooth, and absolutely rigid, the meshing frequency, f_m, *is* equal to the number of teeth in the wheel, ***N*,** multiplied by the speed of the shaft on which the wheel is mounted, f_s, in RPS or RPM, or Hz

$$f_m = (f_s)(N) \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (4.3)$$

For a pair of spur gears, if f_{s1} and f_{s2} are the rotation frequencies of the two shafts, and N_1 and N_2 are the corresponding number of teeth, the fundamental meshing frequency is the same for both gears in mesh

$$f_m = (f_{s1})(N_1) = (f_{s2})(N_2) = (f_m)_{1x} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (4.4)$$

2.4.4. Effect of tooth deflection

Consider a pair of gears whose teeth are not rigid, but equally spaced, perfectly formed and at constant speed. Since the contact stiffness varies periodically, as shown in the lower part of *Figure 2.14*, with the number of teeth in contact and with the contacting position on the tooth surface, vibration will be excited at the tooth engagement frequency and its harmonics. A typical gear mesh waveform is shown in the lower part of *Figure 2.14*.

In Fig. 2.13, the segment *ab* on the *line of action* denotes the interval of engagement of a pair of gears. At the point *a*, when the flank of the driving tooth A just touches the tip of the driven tooth D, there are two pairs of teeth meshing, each taking a share of the transmitted load. Tooth B will then be relieved of some of its load and will tend to deflect towards its unstressed position, imparting a forward acceleration to tooth E on the driven gear. At the termination of meshing of teeth B and E, only teeth A and D are available to transmit the load, as a result of which tooth A is deflected back further and tooth D will momentarily lag. The final point of contact *b* is where the addendum circle of the driver gear crosses the pressure line.

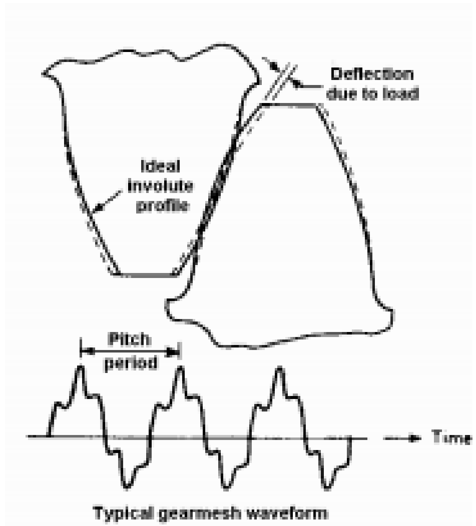

Figure 2.14:-Gear mesh waveform

This tooth deflection is very load dependent. For condition monitoring purposes it is necessary to make measurements always at the same loading, and this loading should be sufficient to ensure that the teeth are permanently in contact, and not able to move into the backlash.

2.4.5. Effect of tooth wear

During the motion of the compliant meshing gears, the wear produced by sliding tends to give the kind of profile deviation indicated in exaggerated form in *Figure 2.15*.

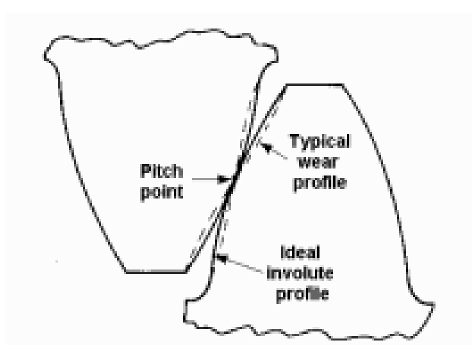

Figure 2.15:-Gear profile deviation

When the point of contact of the engaging teeth reaches the *pitch point*, the direction of sliding reverses, causing a shock – sometimes referred to as the *pitch-circle impulse* – which is perpendicular to the axes of rotation of the two gears. The two shafts are then subjected to bending stress reversals at the rate of the product of shaft speed and number of teeth.

When a new pair of teeth takes part in the transmission of load, the driven gear retrieves its retardation by a renewed forward acceleration. It is subjected to an *engagement shock*, the impulse acting in a tangential direction at a rate of the product of rotational speed and number of teeth. These impulses cause the transmitted torque to fluctuate about a mean level, with variations of the angular velocity, producing a *frequency modulation* of the tooth-meshing frequency.

The pitch circle and engagement vibrations are transmitted through the shaft and bearing housing causing casing vibrations. This vibration can be measured using an accelerometer mounted on the casing.

CHAPTER THREE

3. EXPERIMENTAL CONDITIONS, METHODS AND TEST-RIG

3.1. Test Rig

The main setup for this research work experiment is a Machinery Fault Trainer (PT 500) which is a setup that used to demonstrate a problem within a machine or a mechanism by using vibration signature. The PT 500 machine diagnostic system consists of an aluminum frame with simple profile, on which the experimental gearbox setup, motor and the load unit are mounted using hexagonal socket screws and associated slot nuts.

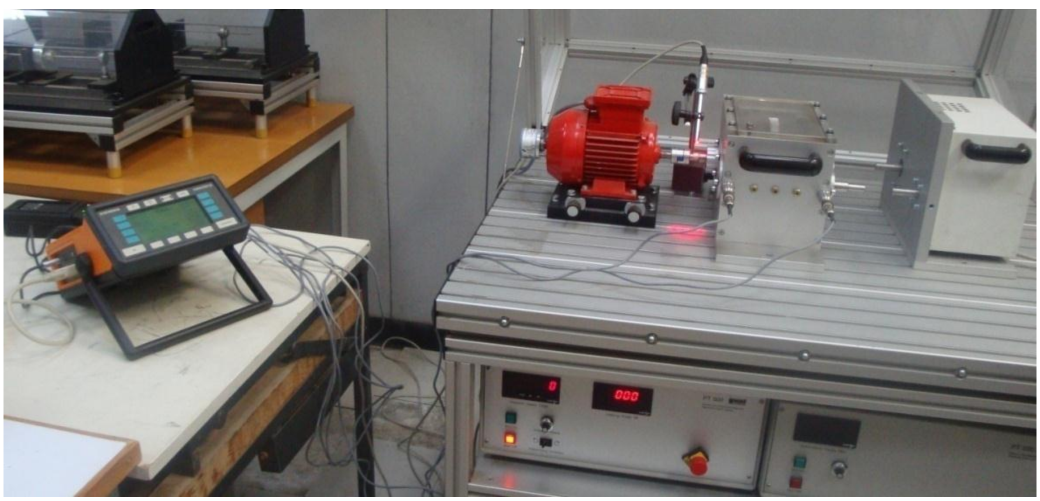

Figure 3.1:- PT 500 Test Setup

The gear box is driven by a three –phase A.C. motor with rotary encoder. The motor is mounted on the base plate with adjusting mechanism and the motor is actuated by the controller. The controller contains a frequency convertor for continues adjustment of the speed. In addition the controller includes a display for the power consumption of the motor and for the speed recorded by the rotary encoder on the motor.

The vibration signal from the gear box is recorded and printed out by an industrial oscilloscope fitted with two acceleration sensors (accelerometer) connected through connecters 1 and 2 (Input 1 and Input 2) and one speed sensor (connected through Reference) . The acceleration sensor is mounted on the required position of the gear box to collect the vibration signal due to the wear (gear fault) and the exact speed of the input shaft is collected from shaft of the A.C. motor by using the speed sensor.

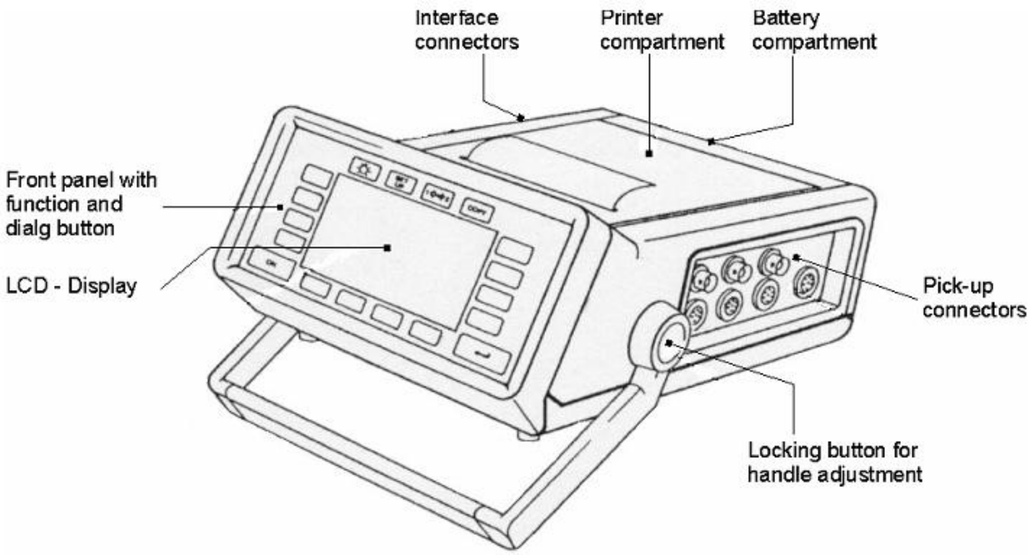

Figure 3.2:- Industrial oscilloscope (VIBROTEST 41)

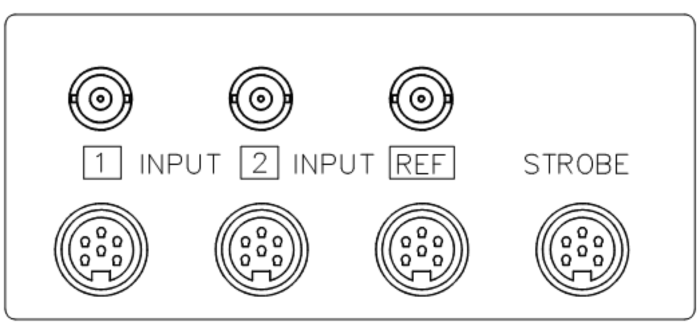

Figure 3.3:- Industrial oscilloscope (VIBROTEST 41) data Pick-up connectors

Finally, to demonstrate the effect of external load on the vibration signature; output shaft of the gear box is directly coupled with the load unit by using flexible coupling.

3.2. Main Component of the Test Rig

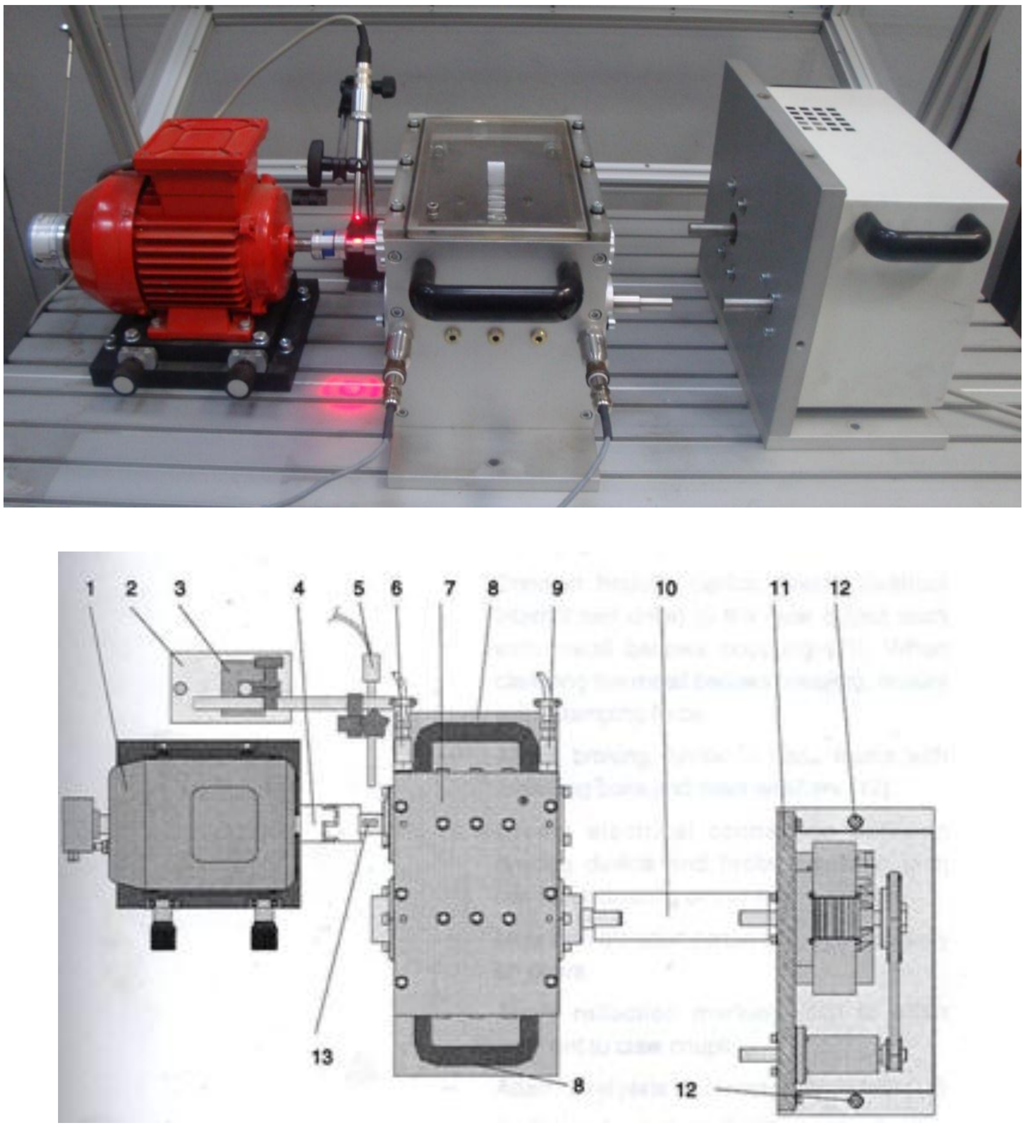

Figure 3.4:- Main component of the test setup

1. Drive unit

2. Steel Plate

3. Magnetic holder

4. Claw Coupling

5. Reference sensor (Speed Sensor)

6. Acceleration Sensor 1

7. Gears

8. Mounting bolt with plain washer

9. Acceleration Sensor 2

10. Metal bellow coupling

11. Braking and load unit

12. Mounting bolt with plain washer

13. Reflection marking

3.3. Gear Box and the Gears

A gear box which is used for this research work is single stage aluminum casing with parallel shaft arrangement (shaft arrangement for the spur gear with simples supported beam and center distance between the shafts are 100 mm) all shaft are supported by two single row deep grove ball bearing.

The gear box have different hole at different position which are used for the connection of the acceleration sensor. The vibration signal which is created due to fault form the gear is directly transferred to the shaft and to the gear box through bearing. So, the data is collected directly from the gear box casing.

Gears are manufactured from C50 tool steel by using convectional milling machine. The teeth number of the gear is 75 and that of the pinion is 25 (*the gear ratio* ***i*** *is 3*). The module and the pressure angle of the gears are 2 and 20^0 respectively. After the reference data collected for all gears and pinion at different speed and loading the gear wear (Gear fault) is introduced at the gear flank.

3.4. Data acquisition

All data is collected from the left and right direction of the gears using two accelerometers which is directly coupled with industrial oscilloscope at the same times (i.e. for every condition two data is collected from the left and right direction of the gears).

The acceleration signal produced by the accelerometer is passed on to the instrument that in turn converts the signal to a velocity signal. Depending on the user's choice, the signal can be displayed as either a velocity waveform or a velocity spectrum. A velocity spectrum is derived from a velocity waveform by means of a mathematical calculation known as the Fast Fourier Transform or FFT.

The filtered out data which is converted in to frequency domain (Velocity spectrum) by using FFT is executed in the form of hard copy for the adjusted frequency and amplitude rage. And also every data which is printed out from the oscilloscope is the average of 16 data's which is collected for the same condition

The data acquisition system (VIBROTEST 41) set up is adjusted as the

following:

Table 3.1:- Setup accelerometer 1 and 2 for frequency spectrum

	Accelerometer 1	***Accelerometer 2***
Input	Active	Active
Sensor type	Acceleration	Acceleration
Type of pickup	Velocity	Velocity
Sensitivity	100 mv/mm/sec	100 mv/mm/sec
Unit	mm/sec rms	mm/sec rms

Amplitude Range	Auto	Auto
Frequency Range	(2 and 5) KHz	(2 and 5) KHz
Number of lines	3200	3200
Window	Hanning	Hanning

3.5. Test Procedure

After the gears manufactured without any required fault coding of the gear and pinion have done (this code is used as a name of the gear and the pinion). Code for the gear is a number 1, 2, 3 and code for the pinion is a letter A, B, C. This all gears and pinion assembled within the gearbox. The assembly is Gear 1 with gear A (Gear pair 1), Gear 2 with pinion B (Gear pair 2), and Gear 3 and pinion C (Gear pair 3). For every test at different speed and load conditions this grouping is respected. For these three groups an artificial fault is introduced. An artificial fault is introduced on the gear and pinion flank (on the pitch circle of the gear and pinion) and finally the vibration signature is collected for every required speed and load conditions.

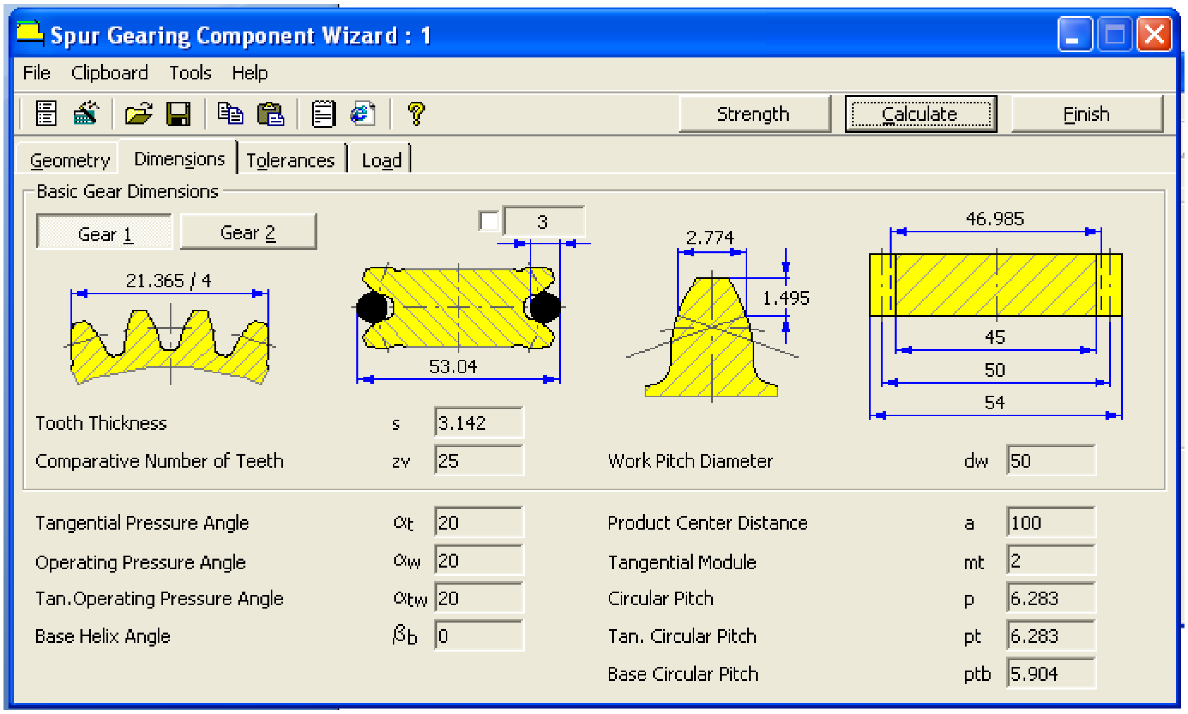

Figure 3.5:- Technical information (basic gear dimension) of the gears generated by mech soft and Solidworks.

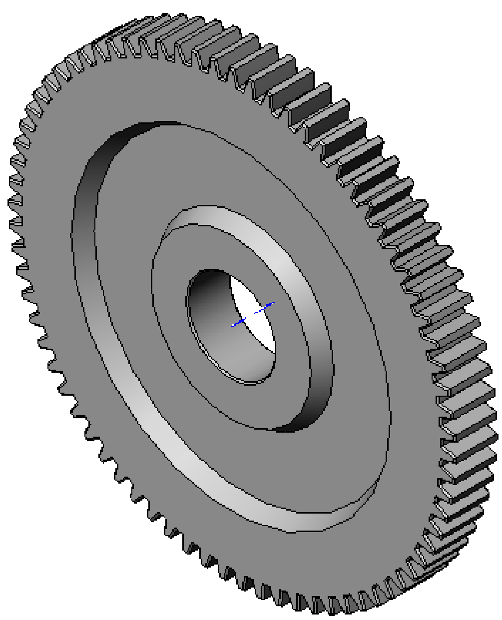

Figure 3.6:- Isometric view of the Pinion (Generated by mech soft and Solidworks)

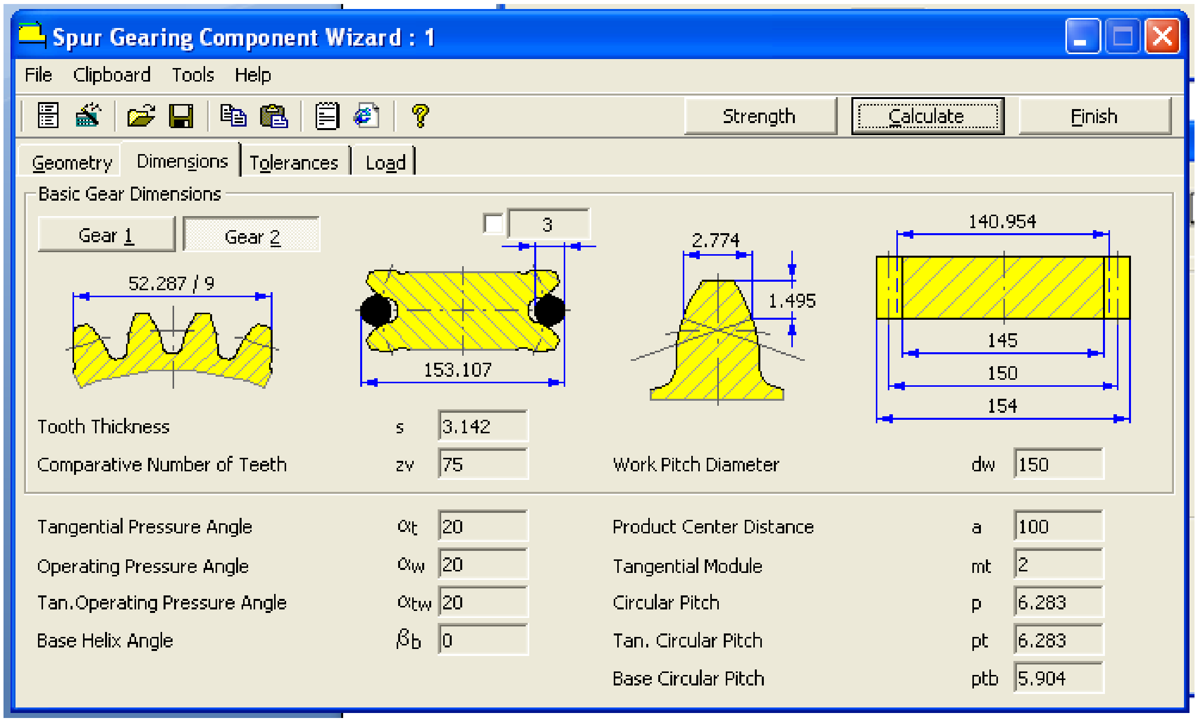

Figure 3.7:- Technical information (basic pinion dimension) of the pinions when the gear designed by the mech soft.

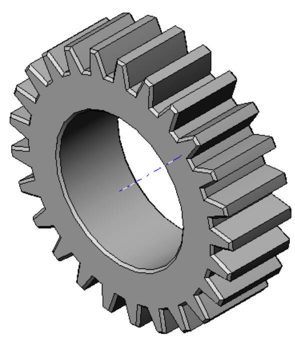

Figure 3.8:- Isometric view of the Pinion (Generated by mech soft and Solidworks)

Figure 3.9:- Manufactured gears and pinion before application of fault

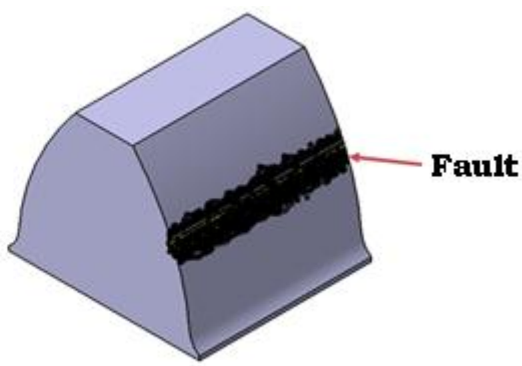

Figure 3.10:- Created artificial fault

3.6. Test Condition

The experiment for this thesis work is carried out in three different test conditions:

1. Gearbox without external load and without fault for different speed
2. Gearbox without external load with fault for different speed
3. Gearbox with external load and fault for different speed

The following table shows all the speed and external load values

Table 4.2:- Experimental conditions

	Ruining speed (Input speed) in RPM	Eternal Load in N-M	Remark
Speed 1	600	0 1.5 2.5 3.5	For all group ***with fault*** and ***without fault***
Speed 2	1000	0 1.5 2.5 3.5	For all group ***with fault*** and ***without fault***
Speed 3	1400	0 1.5 2.5 3.5	For all group ***with fault*** and ***without fault***

For every case of the above experimental conditions; data is collected after the system stabilized the condition (after 3 min) and this data is collected using industrial oscilloscope in the form of hardcopy (both in graph and table format.

Since, the motor is directly coupled with input shaft and the pinion is on the input shaft of the gearbox, the gear box is used as a speed reduction gearbox or a torque multiplier gearbox. The reduction speed (output speed) for the above input speed is given as follows:

Table 4.3:- Input and output speed

	Input Speed (Pinion Speed)		**Output Speed (Gear Speed)**		***Gear ratio***
	CPM / RPM	***Hz***	**CPM / RPM**	***Hz***	
Speed 1	*600*	*10.00*	*200.00*	*3.33*	*3*
Speed 2	*1000*	*16.67*	*333.33*	*5.56*	
Speed 3	*1400*	*23.33*	*466.67*	*7.78*	

With respect to this the harmonic (rotational) frequency and the meshing frequency of the gear and the pinion is given as follows:

Table 4.4:- Harmonic frequency of the gear and pinion

	1st Pinion Harmonic Frequency (Hz)	**1st Gear Harmonic Frequency (Hz)**	**2nd Pinion Harmonic Frequency (Hz)**	**2nd Gear Harmonic Frequency (Hz)**	**3rd Pinion Harmonic Frequency (Hz)**	**3rd Gear Harmonic Frequency (Hz)**
Speed 1	*10.00*	*3.33*	*20.00*	*6.67*	*30*	*9.99*
Speed 2	*16.67*	*5.56*	*33.33*	*11.11*	*50.01*	*16.68*
Speed 3	*23.33*	*7.78*	*46.67*	*15.56*	*69.99*	*23.34*

Table 4.4:- Meshing Frequency

	1st Mesh Frequency (Hz) 1X	2nd Mesh Frequency (Hz) 2X	3rd Mesh Frequency (Hz) 3X	4th Mesh Frequency (Hz) 4X	5th Mesh Frequency (Hz) 5X
Speed 1	*250.00*	500.00	750	1000.00	1250.00
Speed 2	*416.67*	833.33	1250	1666.67	2083.33
Speed 3	*583.33*	1166.67	1750	2333.33	2916.67

Harmonic frequencies of the gear and pinion are totally affected by input speed from the motor and the meshing frequency of the gear is affected by input speed from the motor and the number of teeth of the gears. So that the harmonic frequency of the gear and the pinion is not equal but the meshing frequency is common for the gear and pinion.

For each experimental condition a vibration signature data is collected by taking the averaging of 16 different data. i.e. the oscilloscope collects 16 times to present a single data for a single experimental condition.

CHAPTER FOUR

4. RESULTS AND DISCUSSION

The vibration signature of the gearbox for all experimental condition is presented, analyzed and discussed within this part.

4.1. Vibration signature of the gearbox without gear pair fault

This part of the work shows the vibration signature of the gearbox without any induced gear flank fault (wear) and external load. This result is used as a reference data for the next analysis.

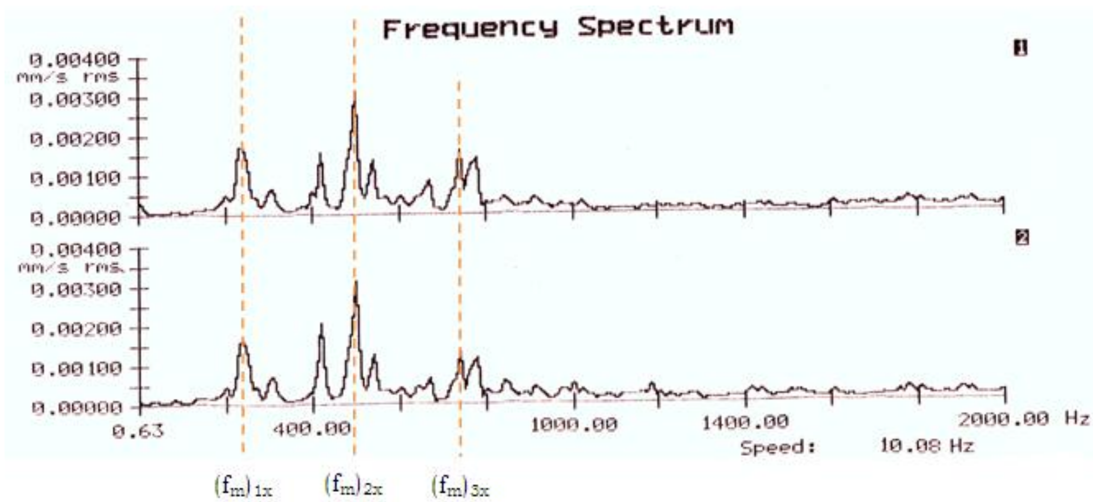

Figure 4.1.:- Frequency spectrum for gear pair 1 without fault at the speed of 10.00 Hz (600RPM)

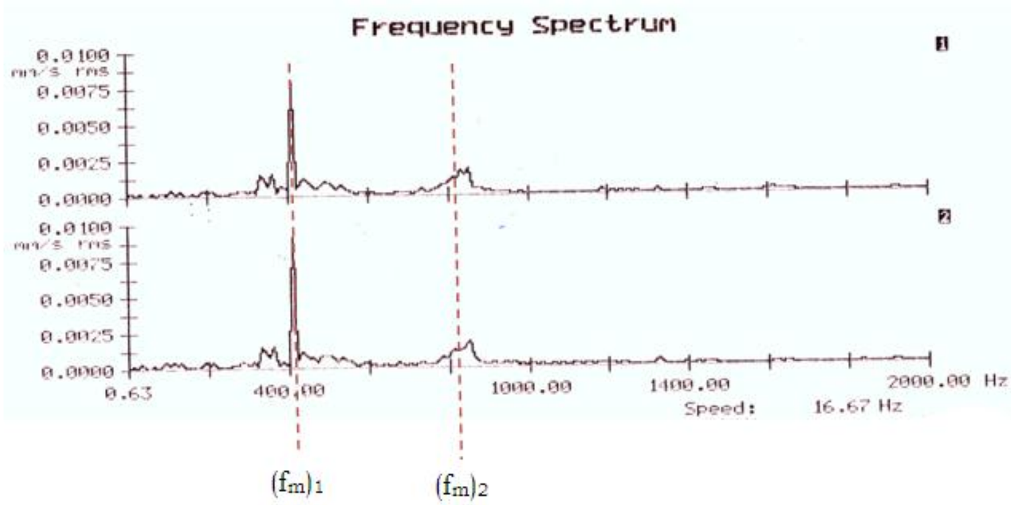

Figure 4.2.:- Frequency spectrum for gear pair 1 without fault at the speed of 16.67 Hz (1000 RPM)

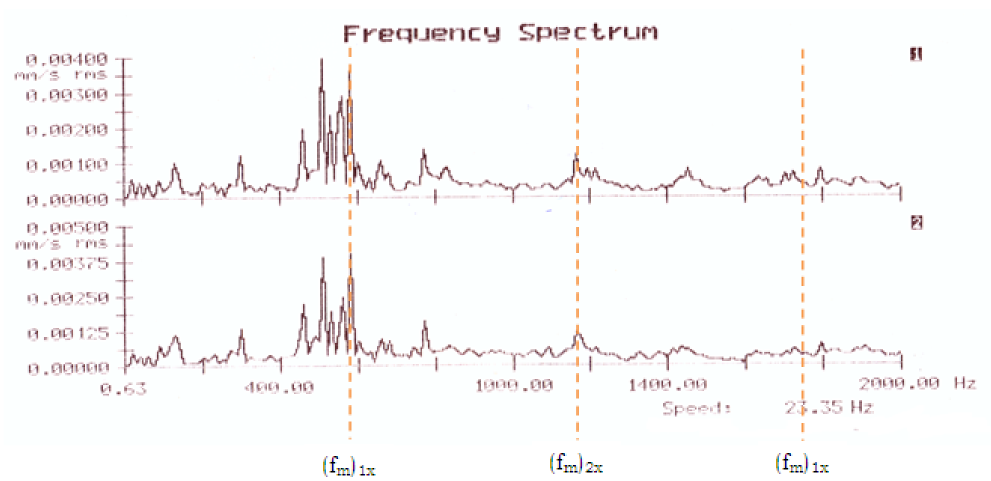

Figure 4.3.:- Frequency spectrum for gear pair 1 without fault at the speed of 23.33 Hz (1400 RPM)

4.2. Effect of gear pair fault (wear) on the vibration signature of the gearbox at different speed

This part of the work shows the variation of the vibration signature with gear flank fault (wear). For this work we used two gears and two pinions (two gear pairs) with fault. In all of the under listed comparison at different speed; the first curve (drawing a's) is a spectrum of the reference gear (gear pair without fault).

4.2.1. Frquency Spectrum of the gear pairs for 10.00Hz (600RPM)

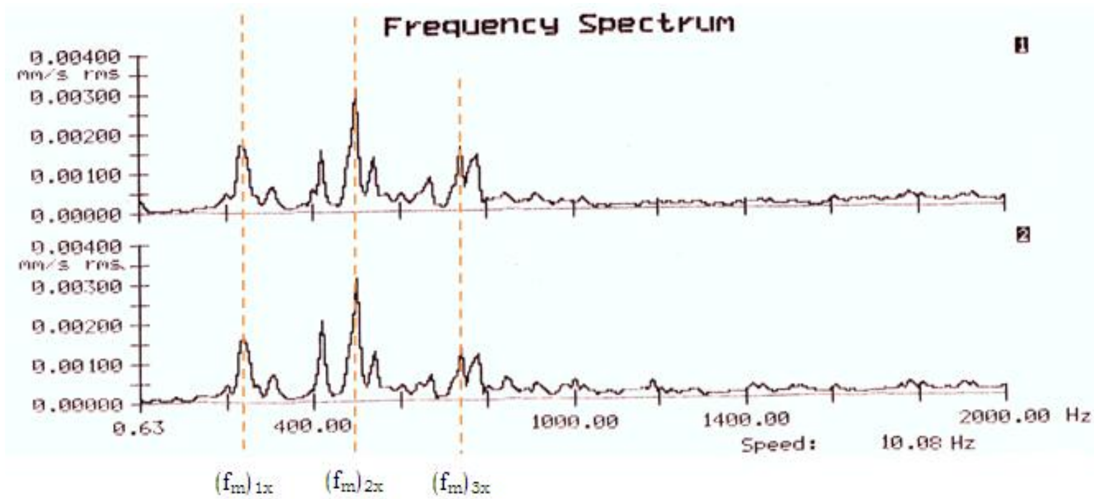

Figure 4.4.:- Frequency spectrum for gear pair 1 without fault at the speed of 10.00 Hz (600RPM)

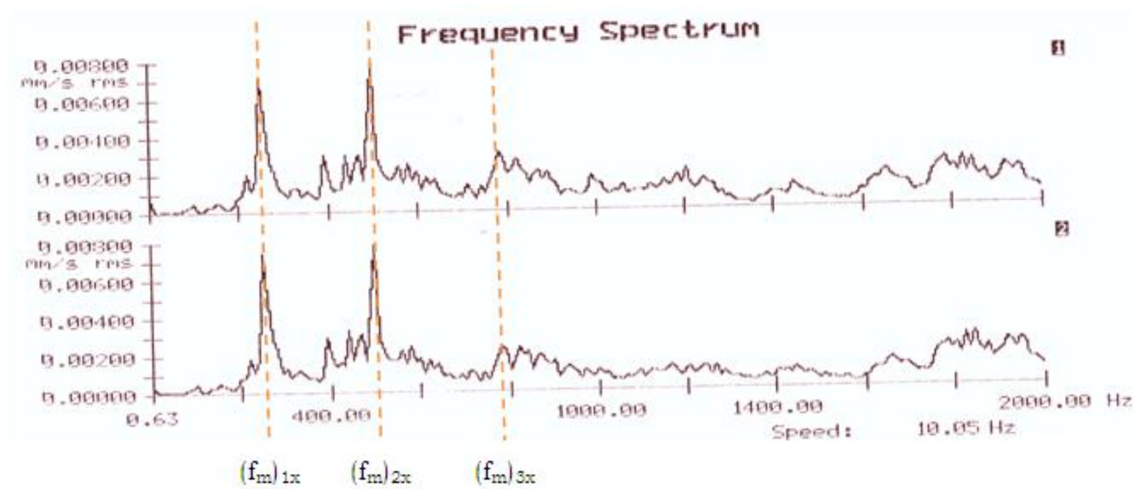

Figure 4.5: - Frequency spectrum for gear pair 2 with fault at the speed of 10.00 Hz (600RPM)

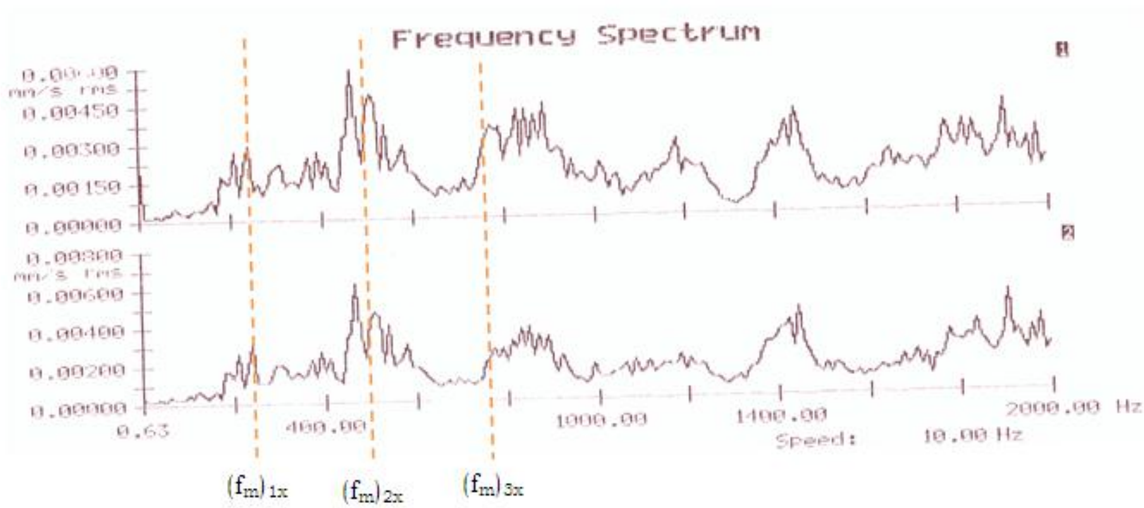

Figure 4.6: - Frequency spectrum for gear pair 3 with fault at the speed of 10.00 Hz (600RPM)

Table 4.1 :- pick list of the three gear pairs at the speed of 10.00 Hz (600 RPM)

signal	*Gear pair 1*		*Gear pair 2*		*Gear pair 3*	
	Hz	mm/s rms	Hz	mm/s rms	Hz	mm/s rms
1	232.085	0.002	251.362	0.007	472.642	0.004
	242.230	0.002	261.396	0.005	476.651	0.006
	252.324	0.001	392.072	0.003	509.816	0.004
	423.821	0.002	445.724	0.003	513.118	0.005
	484.413	0.001	472.662	0.003	522.768	0.005
	494.432	0.002	492.648	0.004	834.163	0.004
	504.554	0.003	495.815	0.005	854.158	0.004
	544.918	0.001	502.715	0.008	892.660	0.004
	746.750	0.002	516.037	0.003	898.240	0.004
	786.932	0.001	790.133	0.003	1908.172	0.004

	Hz	mm/s rms	Hz	mm/s rms	Hz	mm/s rms
2	232.085	0.002	251.362	0.007	472.646	0.005
	242.233	0.002	261.395	0.005	476.669	0.006
	252.321	0.001	271.435	0.003	509.815	0.004
	423.816	0.002	392.090	0.003	513.143	0.005
	484.403	0.001	442.405	0.003	522.820	0.004
	494.433	0.002	445.725	0.003	543.087	0.004
	504.557	0.003	472.666	0.003	1426.984	0.004
	544.921	0.001	492.632	0.005	1442.024	0.005
	746.756	0.001	495.810	0.005	1908.097	0.005
	786.946	0.001	502.714	0.008	1971.861	0.004

Discussion

From all of the above result no resonance signal is created at the harmonic frequency. Almost the signal is zero up to the 20^{th} pinion harmonic frequency (200Hz). But all pick values (resonance signal) is around and /or at the mashing frequency (see the vibration amplitude at; $(f_m)_{1X}$ = 250 Hz, $(f_m)_{2X}$ = 500 Hz, $(f_m)_{3X}$ = 750 Hz). And also if we compared the result of the reference gear data(gear pair 1; which is gear without fault) with the other two gear pairs result (gear pair 2 and 3 ; which are gears with fault) ; as a fault induced on the gear pairs; the vibration amplitude around the mashing frequency increased in magnitude (see table 4.1). In addition to this for all case the largest meshing vibration in magnitude is at the 2^{nd} meshing frequency.

For the gear which is free from gear pair fault the amplitude is almost zero after the third mashing frequency but for the gears which are with fault; there is an increment on the vibration amplitude after the third mashing frequency also.

The vibration spectrum also shows a rise in vibration amplitude (mashing vibration) with simultaneous increment of side bands around the mashing frequencies as fault is introduced.

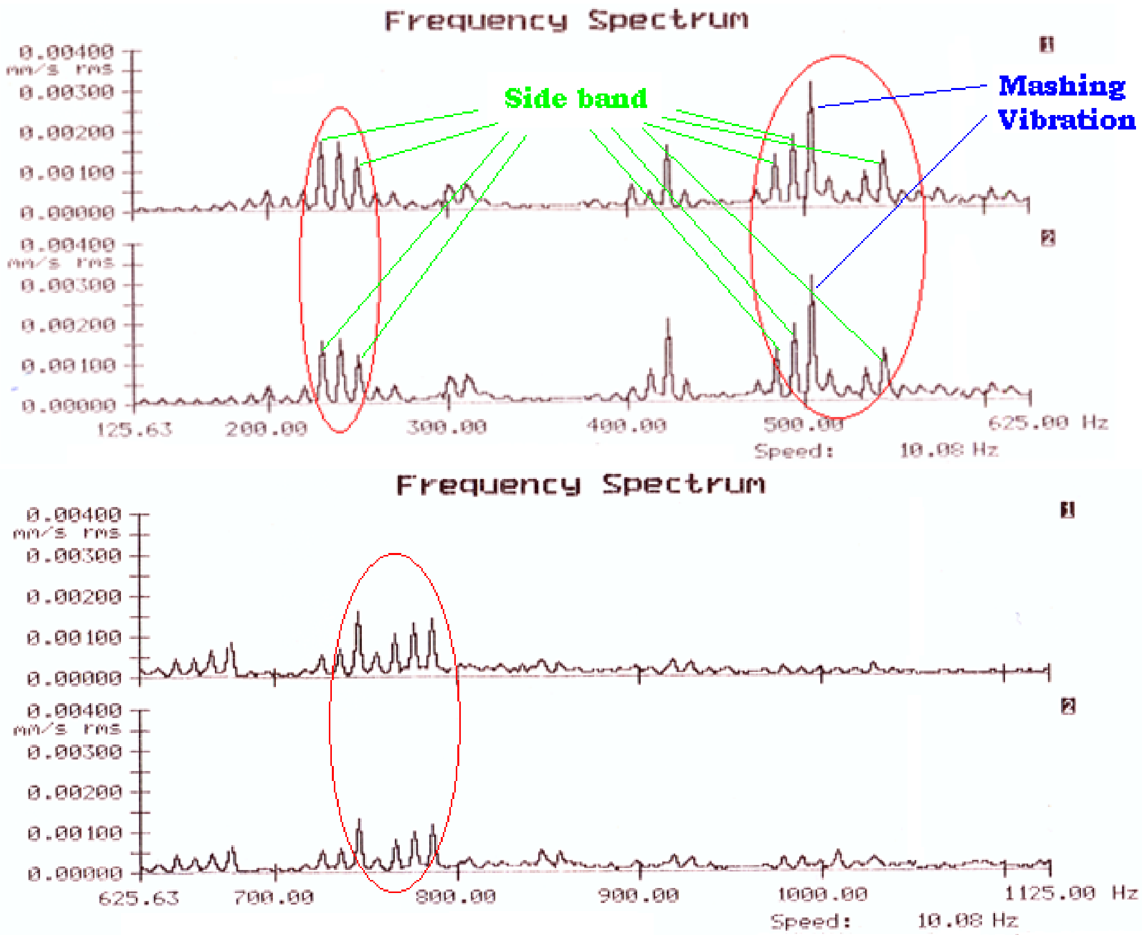

Figure 4.7:- Side bands around the mashing frequency [magnified Frequency spectrum for gear pair 1 without fault at speed 10Hz (600 RPM) (Magnification = 4X)]

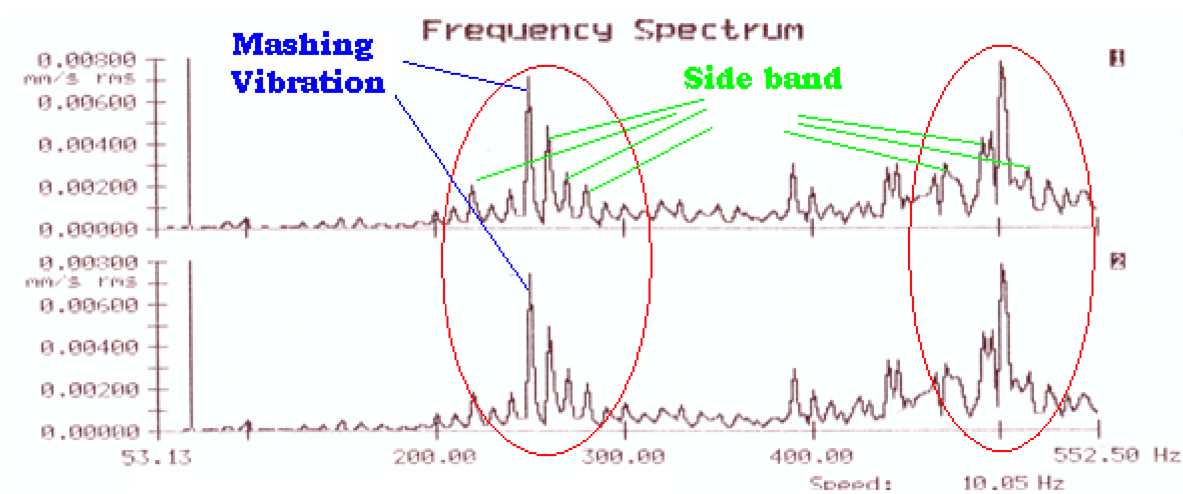

Figure 4.8:- Side bands around the mashing frequency [magnified Frequency spectrum for gear pair 2 with fault at speed 10Hz (600 RPM) (4X)]

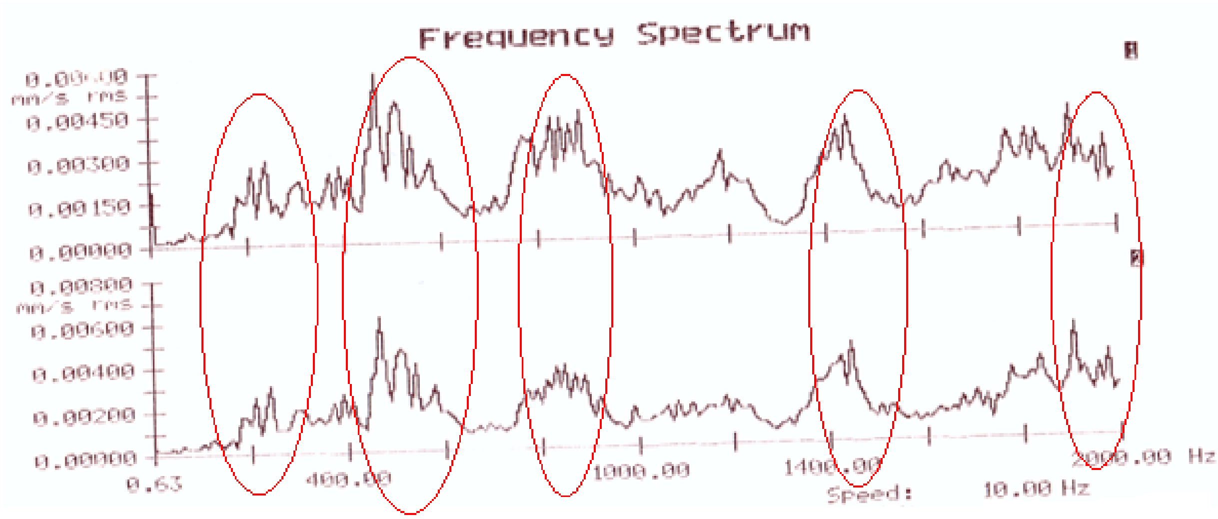

Figure 4.9:- Side bands around the mashing frequency [Frequency spectrum for gear pair 3 with fault at speed 10Hz (600 RPM) (1X)]

In all of the above cases (figure 4.7, 4.8, and 4.9) there are side bands around the mashing frequency but as fault induced the side bands increased both in magnitude and number. And also at some point the side band magnitude is greater than the magnitude of the mashing vibration. (See figure 4.9 around the 8^{th} mashing frequency (2000 Hz))

4.2.2. Frquency Spectrum of the Gearbox for 16.67Hz (1000 RPM)

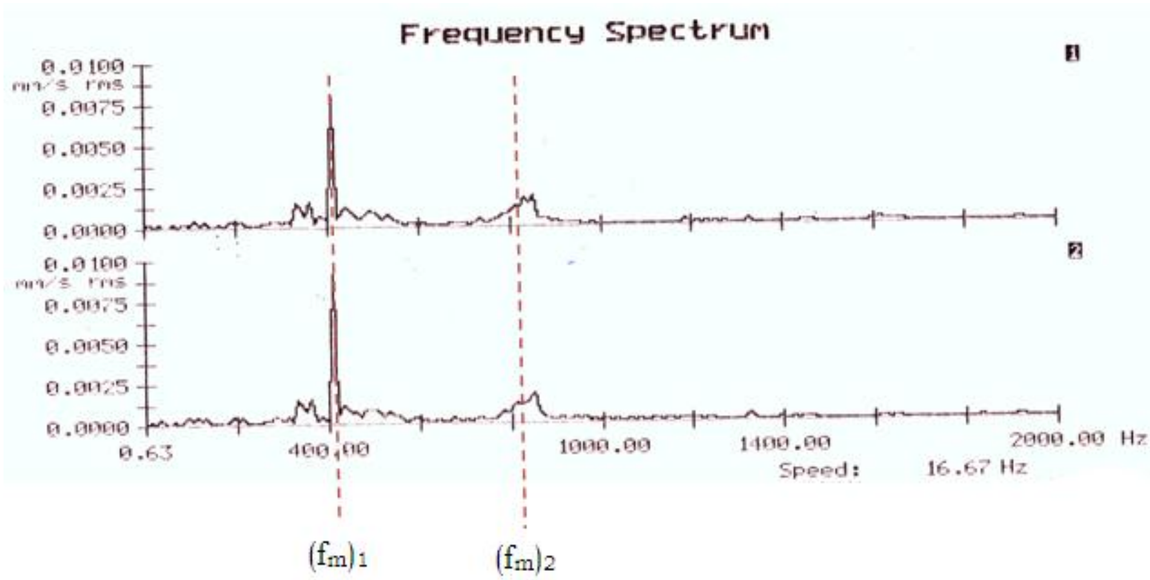

Figure 4.10: - Frequency spectrum for gear pair 1 without fault at the speed of 16.67 Hz (1000 RPM)

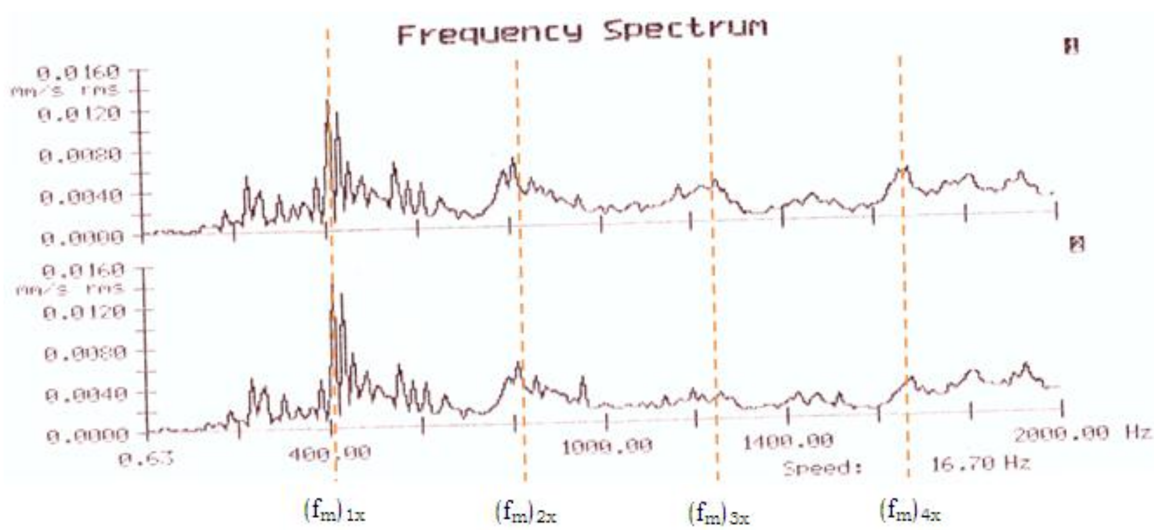

Figure 4.11: - Frequency spectrum for gear pair 2 with fault at the speed of 16.67 Hz (1000 RPM)

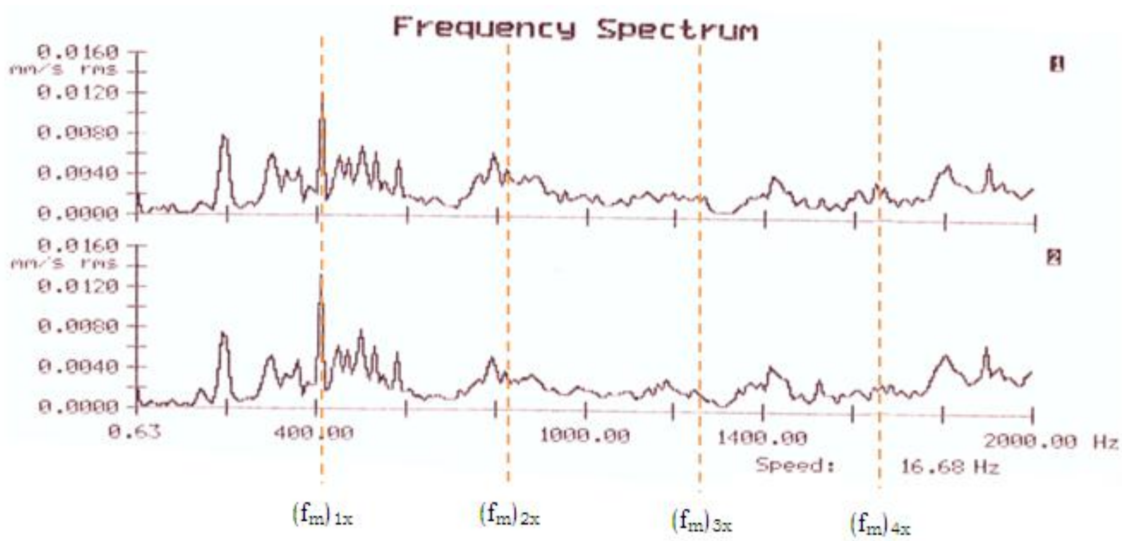

Figure 4.12: - Frequency spectrum for gear pair 3 with fault at the speed of 16.67 Hz (1000 RPM)

Table 4.2:- pick list of the three gear pairs at the speed of 16.67 Hz (1000 RPM)

signal	*Gear pair 1*		*Gear pair 2*		*Gear pair 3*	
	Hz	mm/s rms	Hz	mm/s rms	Hz	mm/s rms
1	333.497	0.002	233.933	0.005	200.259	0.008
	350.073	0.001	384.353	0.005	300.401	0.006
	366.794	0.002	417.857	0.013	417.105	0.012
	416.785	0.008	434.500	0.011	450.506	0.006
	450.026	0.001	451.332	0.007	472.866	0.006
	817.363	0.001	484.547	0.005	500.562	0.007
	827.407	0.001	551.556	0.006	533.794	0.006
	832.042	0.002	796.416	0.005	798.143	0.006
	834.036	0.002	813.308	0.005	1804.349	0.006
	850.599	0.002	819.163	0.007	1896.312	0.006
	Hz	mm/s rms	Hz	mm/s rms	Hz	mm/s rms
2	333.497	0.002	233.939	0.005	200.249	0.008
	350.066	0.001	384.356	0.005	417.119	0.013
	366.792	0.002	417.855	0.014	450.502	0.006
	416.787	0.009	434.499	0.013	472.884	0.006
	433.525	0.001	451.337	0.007	500.562	0.008
	449.977	0.001	484.543	0.005	533.797	0.006
	817.348	0.001	551.557	0.006	584.203	0.006
	827.513	0.001	796.302	0.005	1804.394	0.006
	833.747	0.001	819.109	0.006	1894.330	0.006
	850.636	0.002	1921.803	0.005	1896.319	0.007

Discussion

In all of the above result no resonance signal is created at the harmonic frequency. Almost the signal is around zero mashing vibration up to the 20^{th} pinion harmonic frequency (350 Hz) for gear pair 1 (Gears without fault). But if fault is introduced (Gear pair 2 and 3) the vibration amplitude increased in magnitude after 200 Hz. In addition to this all pick values (resonance signal) is around and /or at the mashing frequency (see the vibration amplitude at; $(f_m)_{1X}$ = 416.67 Hz, $(f_m)_{2X}$ = 833.33 Hz, $(f_m)_{3X}$ = 1250.00 Hz and $(f_m)_{4X}$ = 1666.67.00 Hz). If we compared the result of the reference gear data (gear pair 1) with the other two gear pairs result (gear pair 2 and 3); as a fault induced on the gear pairs; the vibration amplitude around the mashing frequency increased in magnitude (see table 4.2). For all case the largest meshing vibration is at the 1^{st} meshing frequency

For the gear which is free from gear pair fault the amplitude is almost zero after the second mashing frequency but for the gear which is with fault; there is an increment on the vibration amplitude after the second mashing frequency also.

As similar to the privies case there is a formation of side bands around the mashing frequency

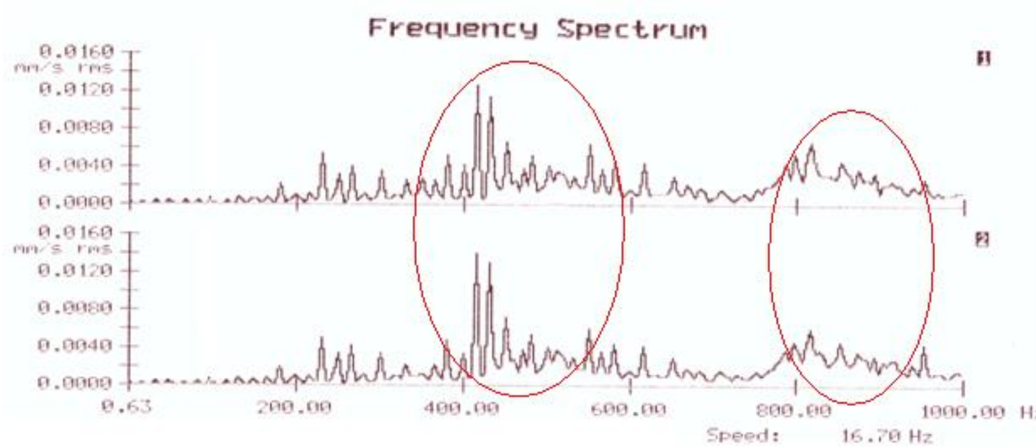

Figure 4.13: Side bands around the mashing frequency [magnified Frequency spectrum for gear pair 2 with fault at speed 16.67 Hz (1000 RPM) (2X)]

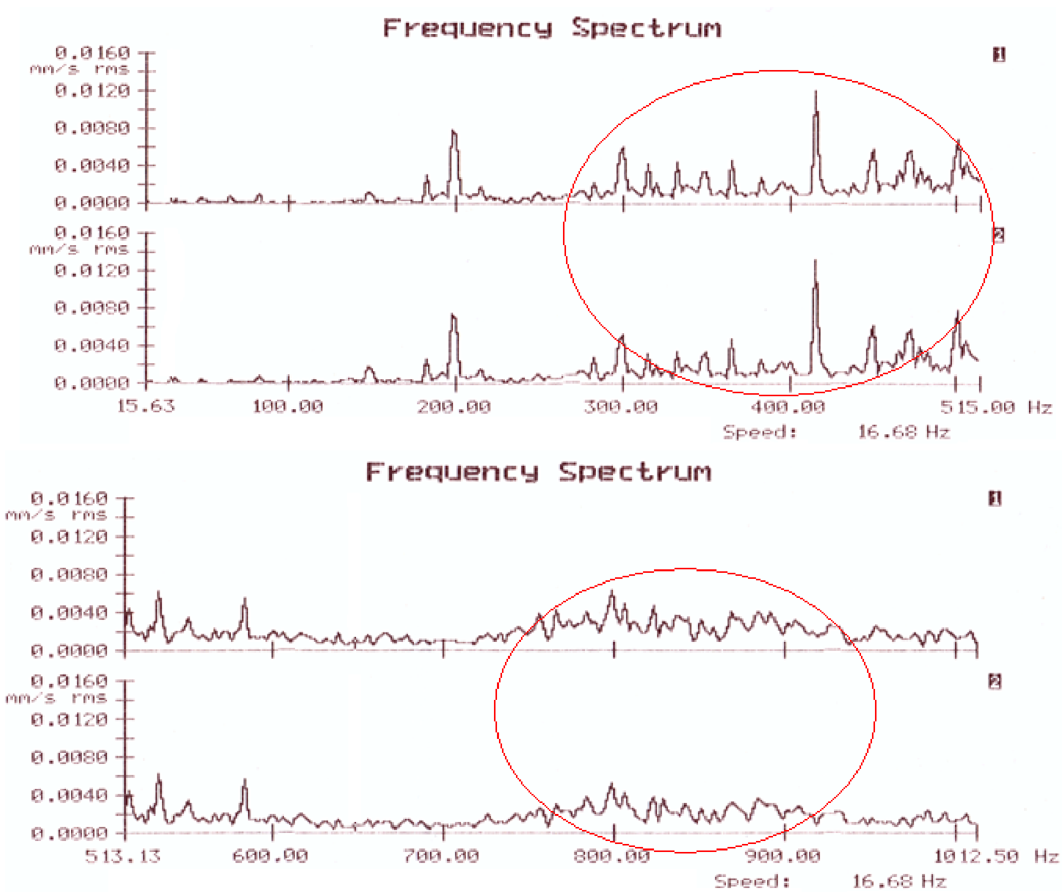

Figure 4.14:- Side bands around the mashing frequency [magnified Frequency spectrum for gear pair 3 with fault at speed 16.67 Hz (1000 RPM) (4X)]

In all of the above cases (figure 4.13, and 4.14) the side bands around the mashing frequency increased both in magnitude and number.

4.2.3. Frquency Spectrum of the Gearbox for 23.33Hz (1400 RPM)

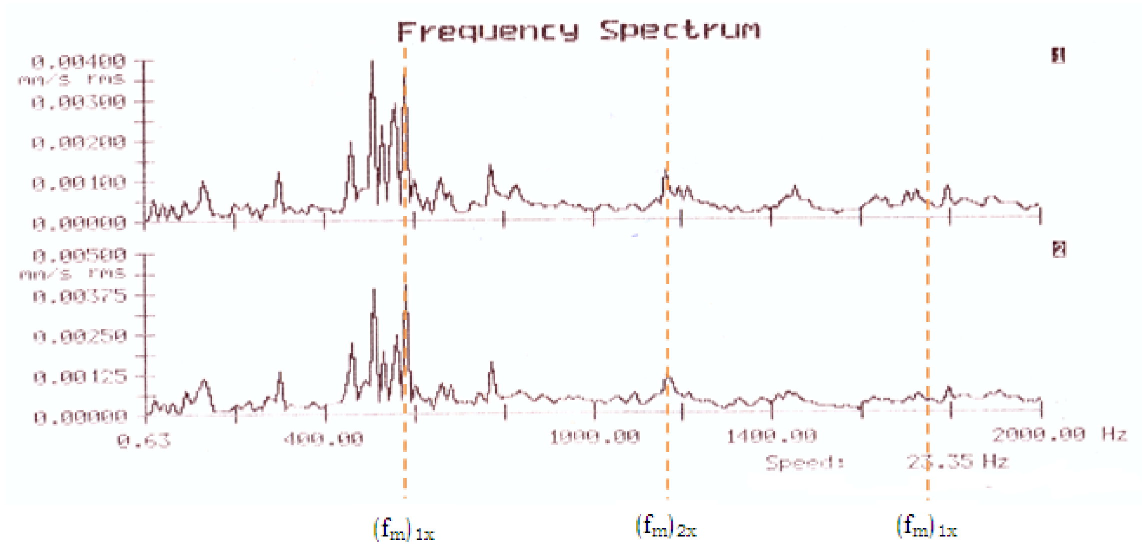

Figure 4.15: - Frequency spectrum for gear pair 1 without fault at the speed of 23.33 Hz (1400 RPM)

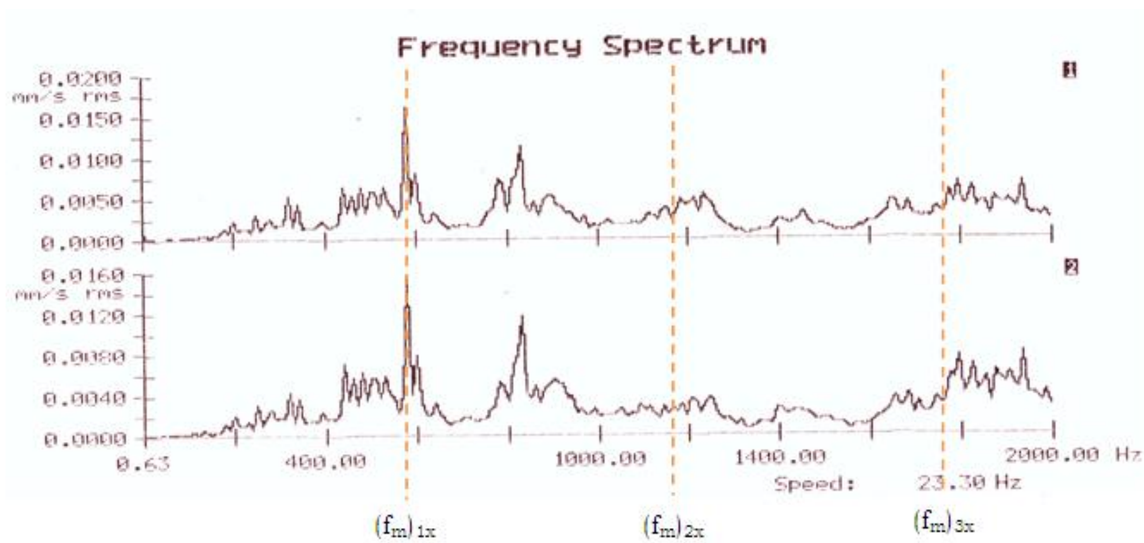

Figure 4.16: - Frequency spectrum for gear pair 2 with fault at the speed of 23.33 Hz (1400 RPM)

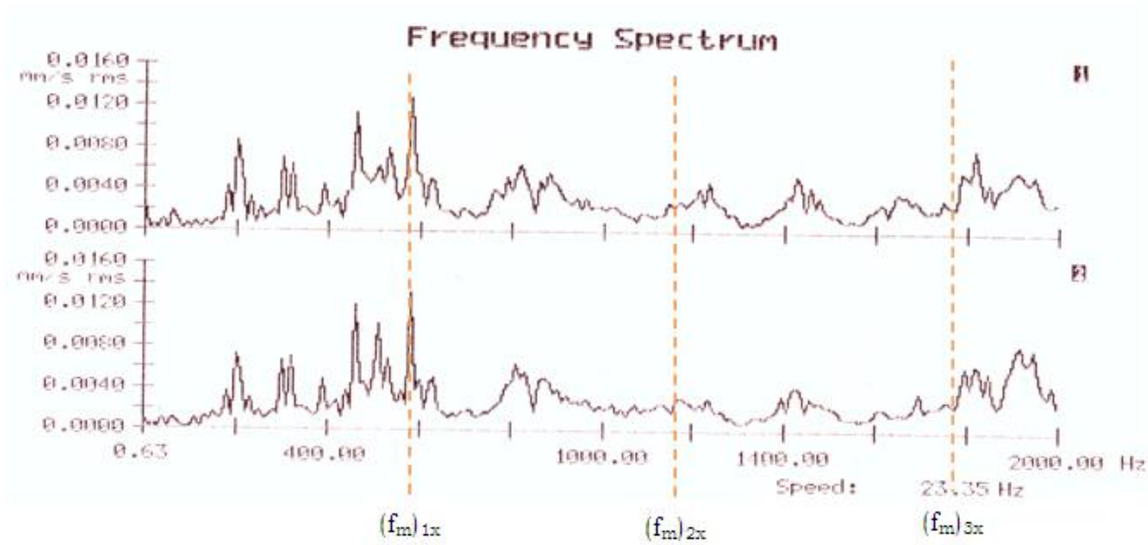

Figure 4.17: - Frequency spectrum for gear pair 3 with fault at the speed of 23.33 Hz (1400 RPM)

Table 4.3:- pick list of the three gear pairs at the speed of 23.33 Hz (1400 RPM)

	gear1 and Pinion A		*gear2 and Pinion B*		*gear3 and Pinion C*	
1	Hz	mm/s rms	Hz	mm/s rms	Hz	mm/s rms
	140.093	0.001	489.335	0.006	210.094	0.009
	303.697	0.001	582.407	0.016	303.516	0.007
	467.099	0.002	605.686	0.008	326.958	0.006
	513.862	0.004	786.858	0.007	466.820	0.011
	537.285	0.002	793.811	0.006	513.538	0.006
	560.524	0.003	799.857	0.007	536.903	0.008
	583.630	0.004	817.100	0.007	583.559	0.013
	606.458	0.001	830.915	0.011	821.109	0.006
	660.596	0.001	1798.254	0.007	1815.583	0.008
	777.399	0.001	1939.333	0.007	1900.744	0.006
2	Hz	mm/s rms	Hz	mm/s rms	Hz	mm/s rms
	140.104	0.001	442.706	0.007	210.134	0.007
	303.716	0.001	489.338	0.006	326.946	0.007
	467.131	0.002	582.403	0.015	466.952	0.012
	490.472	0.001	605.685	0.008	513.677	0.010
	513.916	0.004	817.330	0.007	537.134	0.007
	537.281	0.002	830.957	0.012	583.637	0.013
	560.639	0.002	1798.234	0.008	1900.734	0.008
	584.035	0.004	1822.269	0.007	1915.017	0.008
	777.388	0.002	1876.442	0.006	1937.135	0.007
	1167.520	0.001	1939.348	0.008	1949.348	0.008

Discussion

In all of the above result no resonance signal is created at the harmonic frequency. Pick values (resonance signal) is around and /or at the first mashing frequency (see the vibration amplitude at; $(f_m)_{1X}$ = 583.33 Hz,). But for the 2nd and 3rd mashing frequency for all pair gear; the pick values is shifted from the mashing frequency by some frequency. As a fault induced on the gear pairs; the vibration amplitude around the mashing frequency increased in magnitude (see table 4.3). For all case the largest meshing vibration is at the 1st meshing frequency

For the gear which is free from gear pair fault the amplitude is declining in magnitude after the second mashing frequency but for the gear which is with fault; there is an increment on the vibration amplitude after the second mashing frequency also.

As similar to the privies case there is a formation of side bands around the mashing frequency

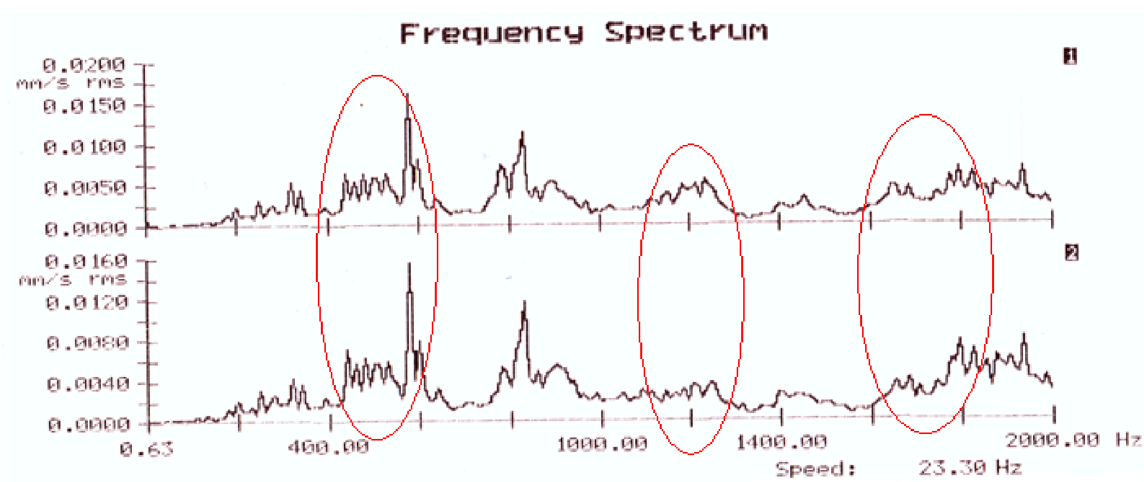

Figure 4.18:- Side bands around the mashing frequency [Frequency spectrum for gear pair 2 with fault at speed 23.33 Hz (1400 RPM) (1X)]

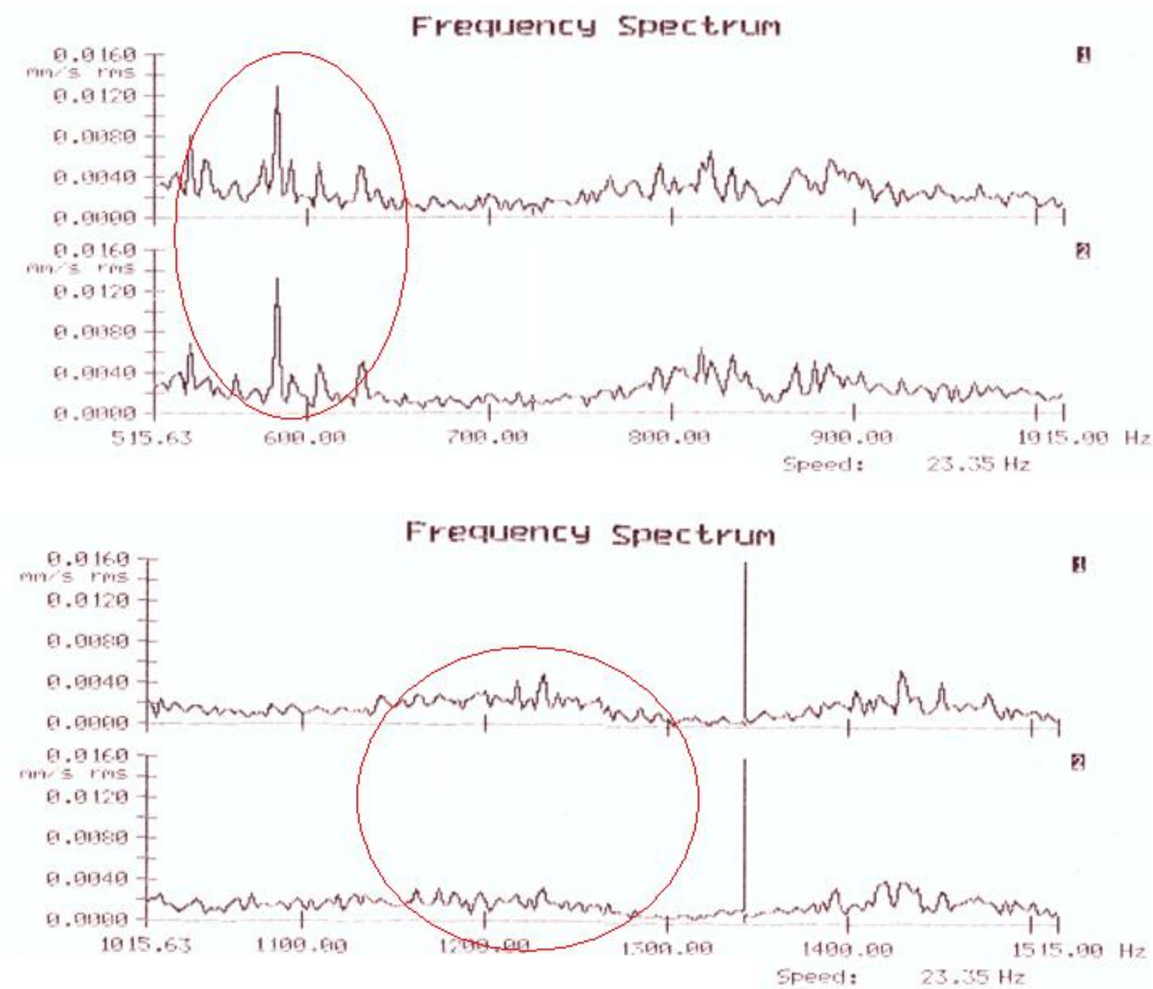

Figure 4.19:- Side bands around the mashing frequency [Frequency spectrum for gear pair 2 with fault at speed 23.33 Hz (1400 RPM) (4X)]

In all of the above cases (figure 4.18, and 4.19) the side bands around the mashing frequency increased both in magnitude and number.

Conclusion

The general conclusion from these three results is as follows:-

i. If a fault is induced on a gear flank the magnitude of the mashing vibration (vibration amplitude) increased around the mashing frequency.

    Mashing amplitude (Vibration Amplitude) shows the total mashing energy which is used to drive the gear by the pinion. That means if fault is induced in a gear flank, it needs additional energy to drive the gear system with its fault. So that increment of this mashing energy shows the presence of gear fault within the gear flank.

ii. The gear and pinion harmonic Frequency is not used as a fault indicator if the frequency spectrum is velocity Vs Frequency. Because there is no change in magnitude or other case in vibration signature with and without fault at all.

iii. If a fault is induced on a gear flank there is a formation of side bands (secondary bands) around the mashing frequency.

    Fault within the gear flank create local vibration between mashing teeth of the gear and pinion. And this vibration is not died out; depending on the magnitude of the vibration amplitude; at the end of each mashing. It transmits to the next mashing teeth. So that if there is hared mashing (a mash with transferred vibration from one mashing teeth to the other) there is a creation of secondary band (side bands) around the mashing frequency.

iv. From the other mashing frequency which is created within the mashing curve the first and the second mashing frequency is good fault indicator. Because this two values is not died out for any condition of the gear.

4.3. Effects of external load on the vibration signature of the gearbox at different speed.

This study is done for three different external loads (1.5 N-m, 2.5 N-m and 3.5 N-m. and the gear pairs run with speeds 10.00Hz (600 RPM). And the basic objective of this experiment is to investigate the variation of gear vibration signature as the gear box run with external loads. The comparison is done with the same gear pairs with different external loads.

4.3.1. For Gear pair 1

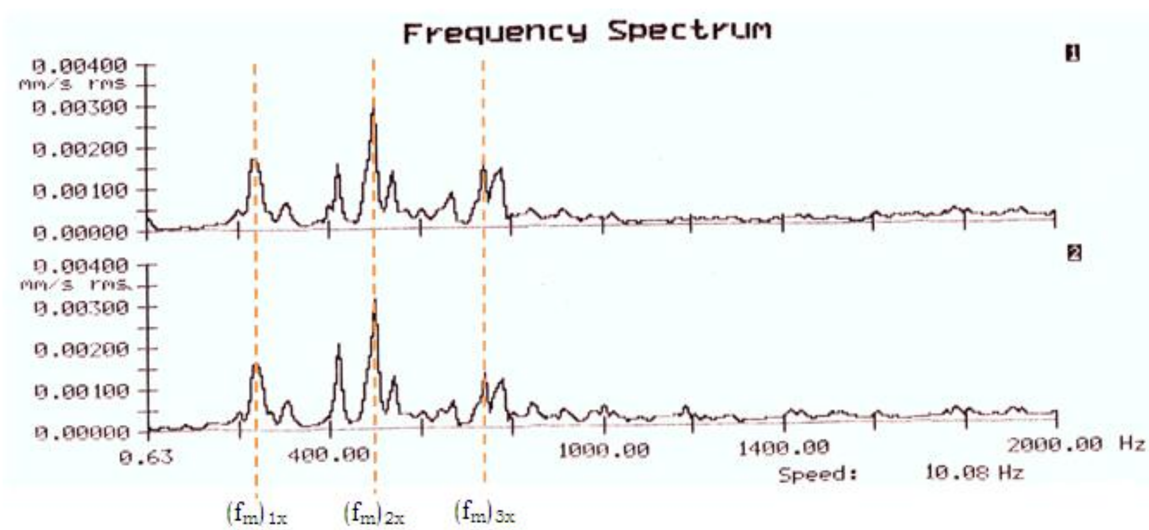

Figure 4.20:-Frequency spectrum for gear pair 1 without fault; with T = 0 N-m at the speed of 10.00 Hz (600RPM)

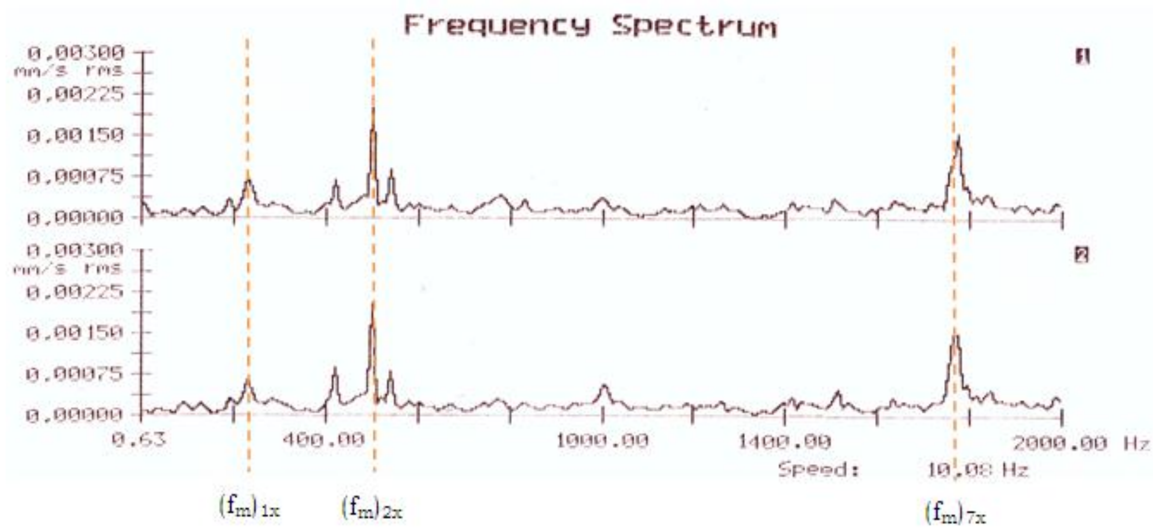

Figure 4.21:-Frequency spectrum for gear pair 1 without fault; with T = 1.5 N-m at the speed of 10.00 Hz (600RPM)

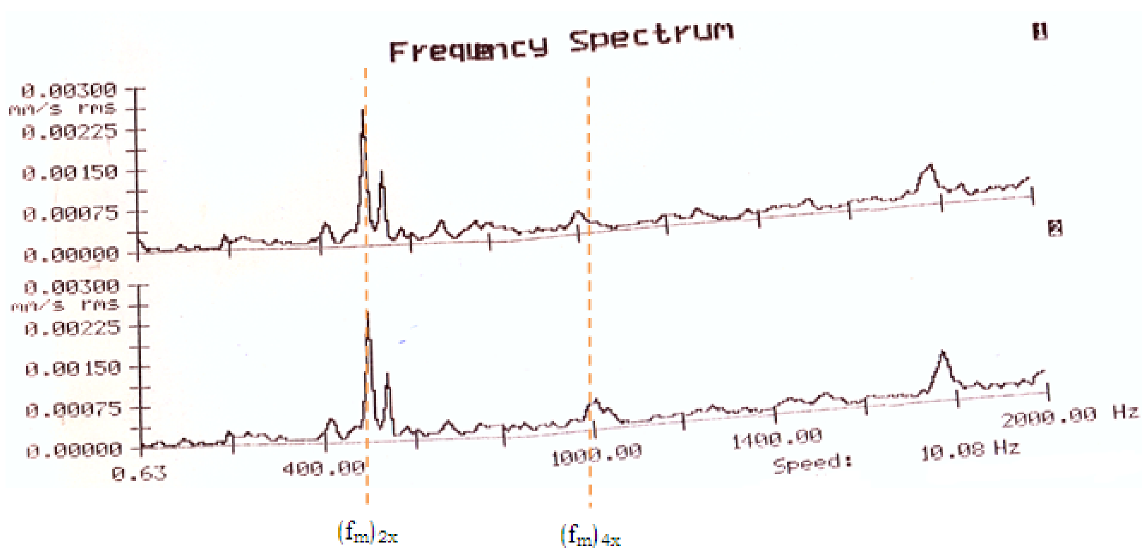

Figure 4.22:-Frequency spectrum for gear pair 1 without fault; with T = 2.5 N-m at the speed of 10.00 Hz (600RPM)

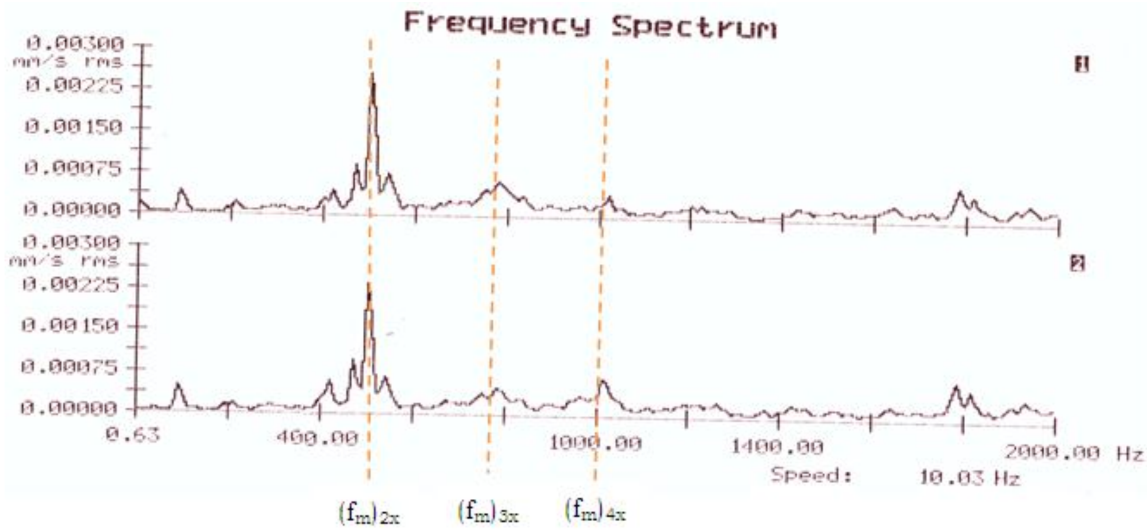

Figure 4.23:-Frequency spectrum for gear pair 1 without fault; with T = 3.5 N-m at the speed of 10.00 Hz (600RPM)

Table 4.4:- pick list of frequency spectrum for gear pair 1 at the speed of 10.00 Hz (600 RPM) with different external load

External Load (T)	*signal*			
	1		*2*	
	Hz	mm/s rms	Hz	mm/s rms
0 N-m	232.085	0.002	232.085	0.002
	242.230	0.002	242.233	0.002
	252.324	0.001	252.321	0.001
	423.821	0.002	423.816	0.002
	484.413	0.001	484.403	0.001
	494.432	0.002	494.433	0.002
	504.554	0.003	504.557	0.003
	544.918	0.001	544.921	0.001
	746.750	0.002	746.756	0.001
	786.932	0.001	786.946	0.001

1.5 N-m	Hz mm/s rms 232.105 0.001 424.059 0.001 503.664 0.001 545.404 0.001 1757.880 0.001 1764.695 0.001 1766.774 0.001 1774.775 0.001 1777.956 0.002 1798.265 0.001	Hz mm/s rms 232.106 0.001 424.047 0.001 503.736 0.001 545.417 0.001 1758.000 0.001 1764.643 0.001 1766.763 0.001 1774.746 0.001 1777.998 0.002 1798.240 0.001
2.5 N-m	Hz mm/s rms 500.693 0.001 506.765 0.002 509.449 0.001 516.725 0.000 540.039 0.000 547.067 0.001 549.444 0.000 1765.507 0.001 1770.659 0.001 1781.024 0.001	Hz mm/s rms 500.677 0.001 506.769 0.002 509.468 0.001 547.014 0.001 1005.420 0.000 1013.833 0.001 1765.486 0.001 1770.716 0.001 1777.346 0.001 1781.114 0.001
3.5 N-m	Hz mm/s rms 475.809 0.001 479.520 0.001 502.647 0.001 505.679 0.001 510.033 0.003 518.024 0.001 543.202 0.001 546.437 0.001 784.306 0.001 1788.871 0.001	Hz mm/s rms 475.769 0.001 479.476 0.001 502.671 0.001 505.676 0.001 510.038 0.002 518.037 0.001 546.435 0.001 1010.657 0.001 1020.576 0.001 1788.945 0.001

Discussion

As an external force applied to the gear box; the resonance value decreased in magnitude specially the mashing vibration (vibration amplitude) or vibration pick value. Generally the total vibration amplitude decreased as an external load applied to the gearbox output shaft increased in magnitude. And also from all of the above result no resonance signal is created at the harmonic frequency. Almost the signal is zero up to the 20th pinion harmonic frequency

(200Hz). But all pick values (resonance signal) is around and /or at the mashing frequency.

All the mashing vibration is canceled out due to the application of external load except the secondary mashing vibration (there is a decrement in magnitude but it is not canceled out (see the above figure 5.23)). But the mashing vibration which is with small magnitude initially, aggravated by the application of an external load (see the 7th mashing frequency in figure 4.21 and 4.22)

As the external load applied to the gearbox the magnitude and number of side bands (secondary band) around the meshing vibration decreased.

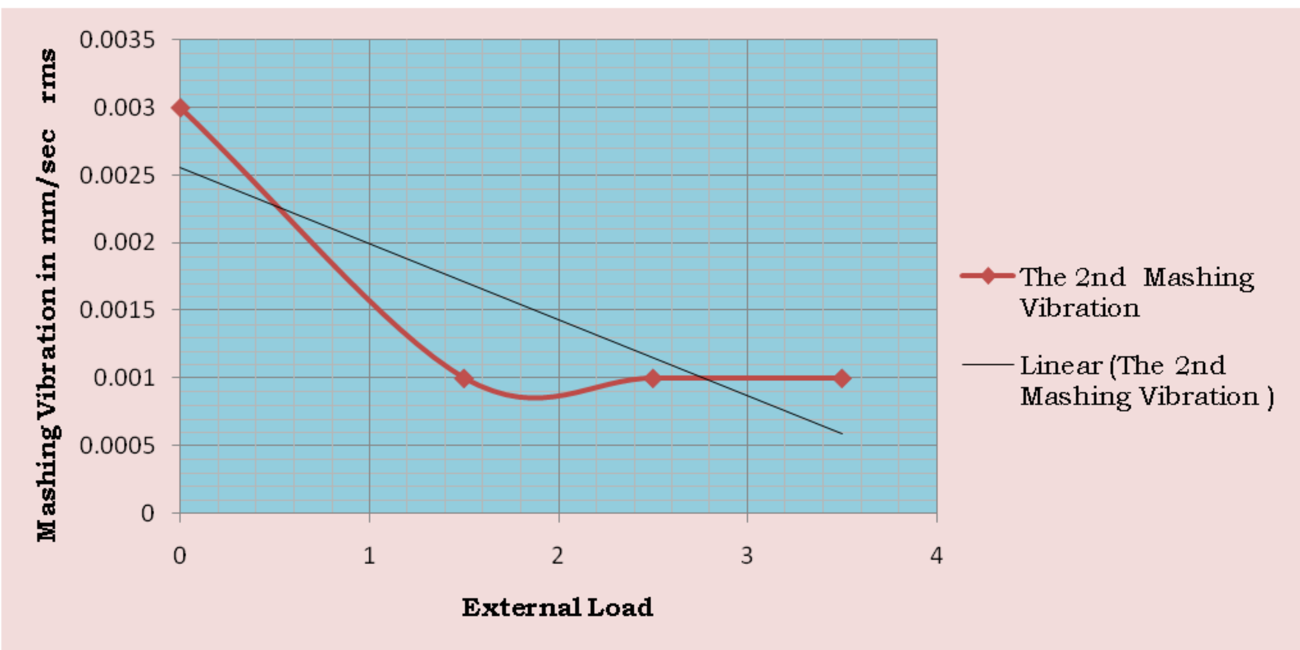

Figure 4.24: - variation of the second mashing vibration with respect to external load for gear pair 1.

4.3.2. For Gear pair 2

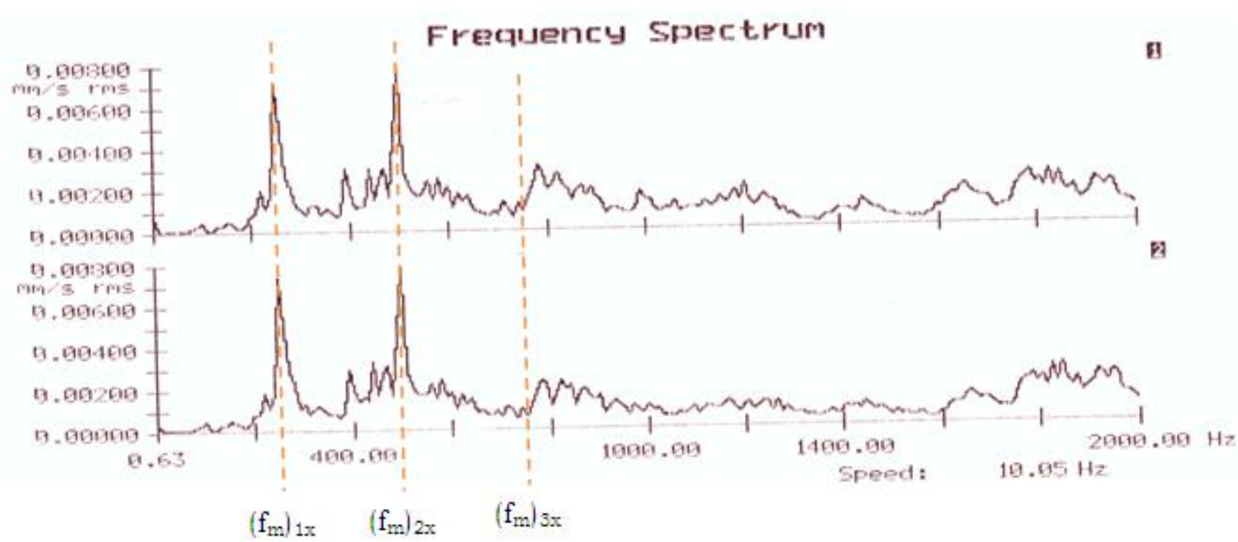

Figure 5.25: - Frequency spectrum for gear pair 2 with fault; with T = 0 N-m at the speed of 10.00 Hz (600RPM)

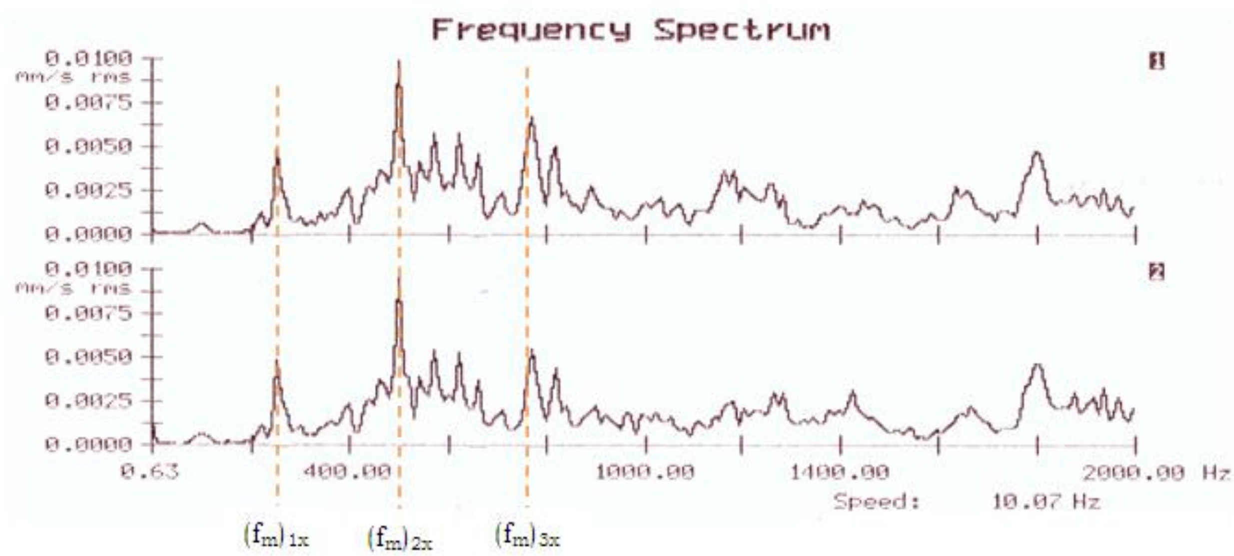

Figure 4.26: - Frequency spectrum for gear pair 2 with fault; with T = 1.5 N-m at the speed of 10.00 Hz (600RPM)

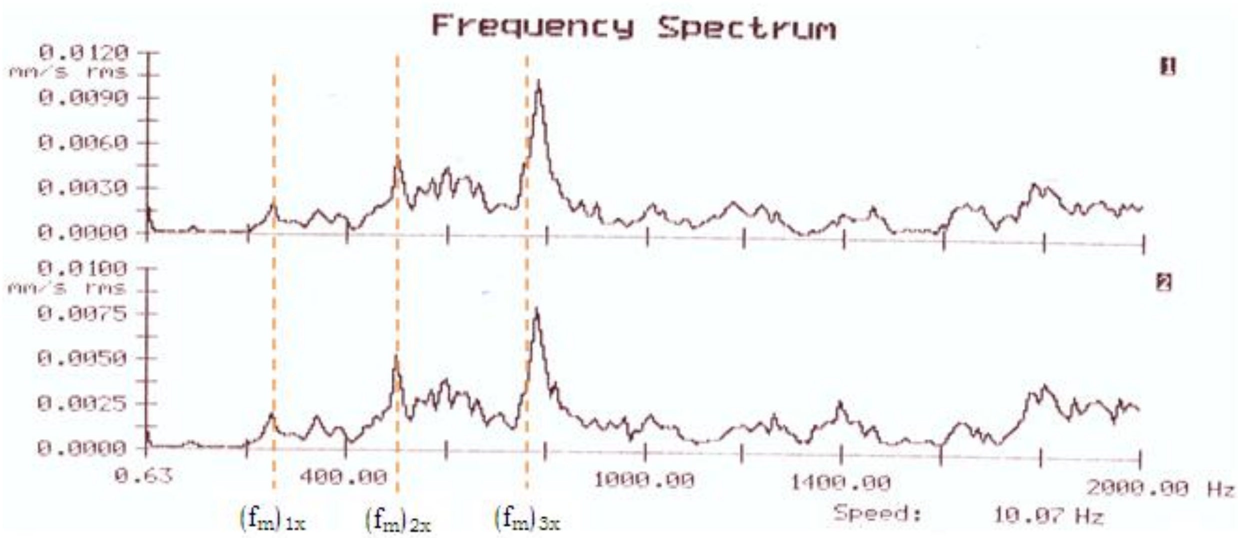

Figure 4.27: - Frequency spectrum for gear pair 2 with fault; with T = 2.5 N-m at the speed of 10.00 Hz (600RPM)

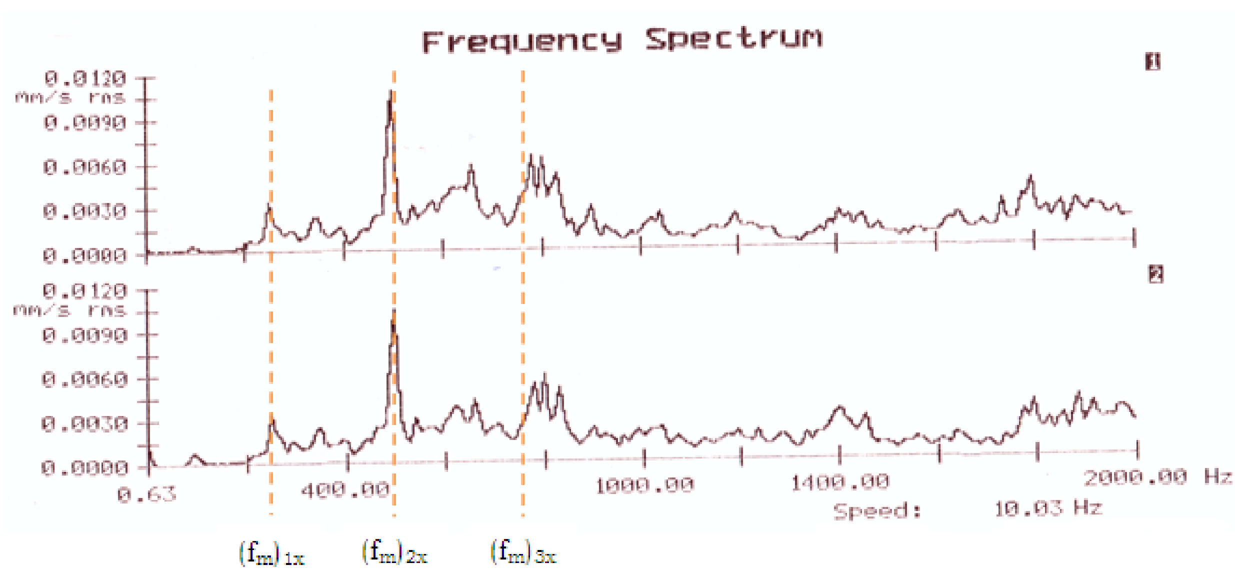

Figure 4.28: - Frequency spectrum for gear pair 2 with fault; with T = 3.5 N-m at the speed of 10.00 Hz (600RPM)

Table 4.5: - pick list of frequency spectrum for gear pair 2 at the speed of 10.00 Hz (600 RPM) with different external load

External Load (T)	*signal*			
	1		*2*	
	Hz	mm/s rms	Hz	mm/s rms
0 N-m	251.362	0.007	251.362	0.007
	261.396	0.005	261.395	0.005
	392.072	0.003	271.435	0.003
	445.724	0.003	392.090	0.003
	472.662	0.003	442.405	0.003
	492.648	0.004	445.725	0.003
	495.815	0.005	472.666	0.003
	502.715	0.008	492.632	0.005
	516.037	0.003	495.810	0.005
	790.133	0.003	502.714	0.008
1.5 N-m	251.397	0.005	251.400	0.005
	502.858	0.010	497.225	0.004
	574.432	0.006	502.809	0.010
	624.992	0.006	513.871	0.004
	665.151	0.005	574.438	0.005
	765.716	0.005	624.988	0.005
	775.686	0.007	775.710	0.005
	785.553	0.005	825.741	0.004
	825.727	0.005	1791.820	0.005
	1791.818	0.005	1801.899	0.005
2.5 N-m	501.989	0.004	501.975	0.004
	508.319	0.005	508.328	0.005
	604.918	0.004	771.104	0.006
	758.726	0.005	778.733	0.004
	760.759	0.004	781.158	0.008
	771.094	0.007	783.503	0.005
	781.187	0.010	786.949	0.004
	786.962	0.005	791.318	0.006
	791.389	0.008	801.691	0.005
	801.683	0.006	1808.356	0.004

3.5 N-m	Hz / mm/s rms 498.962 / 0.008 504.402 / 0.006 508.877 / 0.011 625.830 / 0.004 667.643 / 0.006 783.783 / 0.006 802.637 / 0.006 822.245 / 0.00[illegible] 833.104 / 0.005 1797.432 / 0.004	Hz / mm/s rms 498.962 / 0.008 504.402 / 0.006 508.877 / 0.011 625.830 / 0.004 667.643 / 0.006 783.783 / 0.006 802.637 / 0.006 822.245 / 0.00[illegible] 833.104 / 0.005 1797.432 / 0.004

Discussion

Similar to the case with gear pair one; as an external load increased in magnitude which is applied to the gear box; the meshing frequency decreased gradually. No change on the harmonic frequency with the application of an external load.

The change in mashing vibration with respect to the external load is not uniform. For some case the amplitude increased not for the other. But as a general discussion; the linear approximate shows that the variation is decreasing; because Only one of the second mashing vibration signal (at external load = 1.5 N-m) is greater than the initial values

The variation of the 2nd mashing vibration with external load is shown with the following curve.

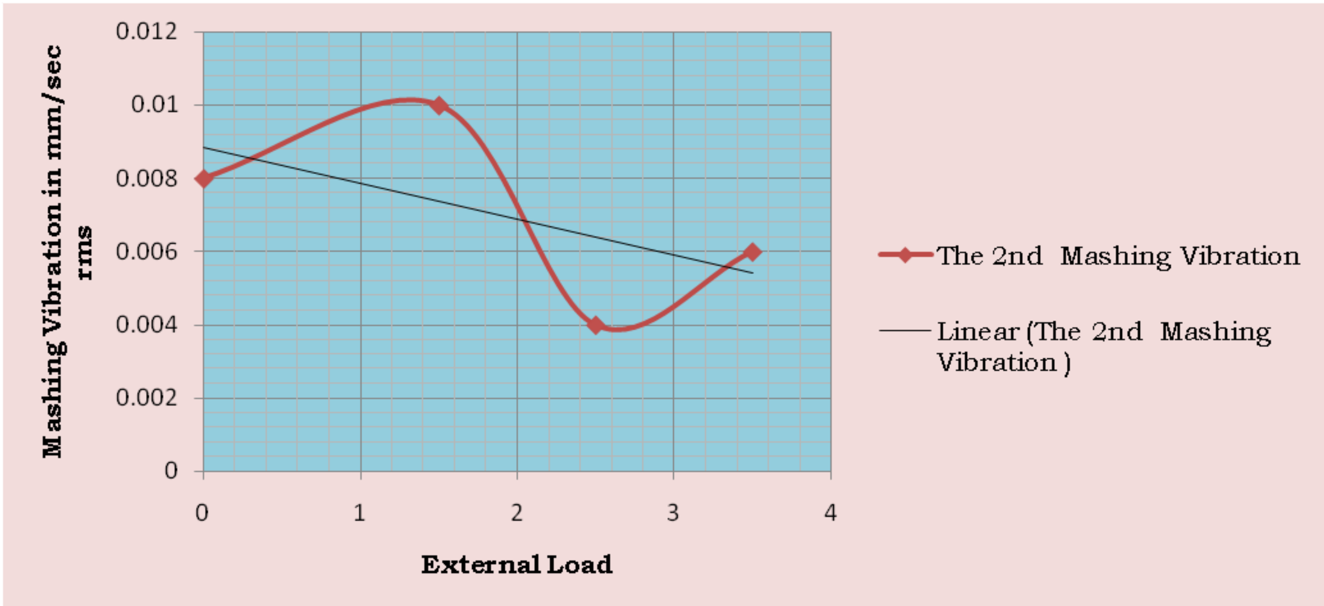

Figure 4.29: - variation of the second mashing vibration with respect to external load for gear pair 2

4.3.3. For Gear pair 3

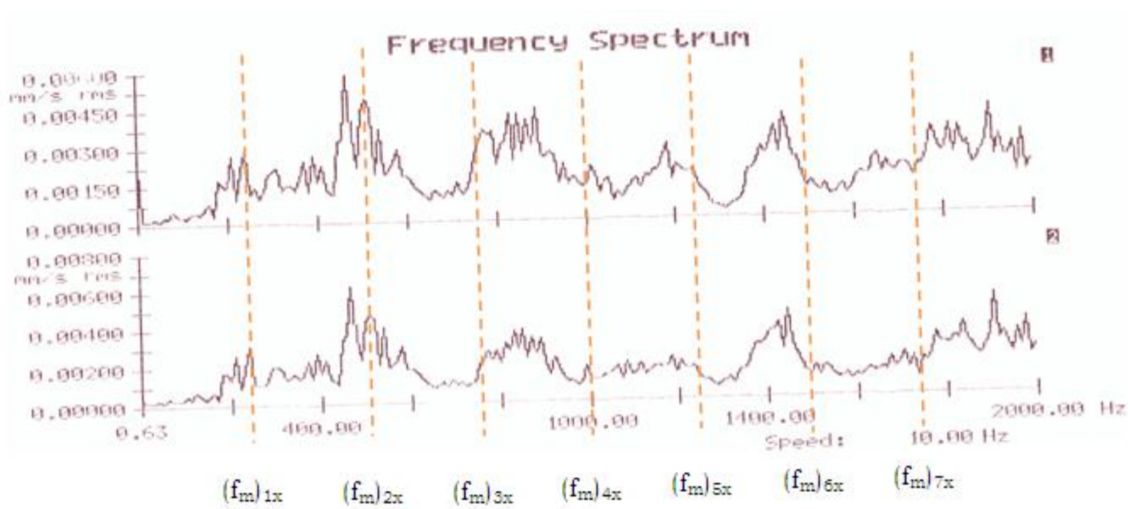

Figure 4.30: - Frequency spectrum for gear pair 3 with fault; with T = 0 N-m at the speed of 10.00 Hz (600RPM)

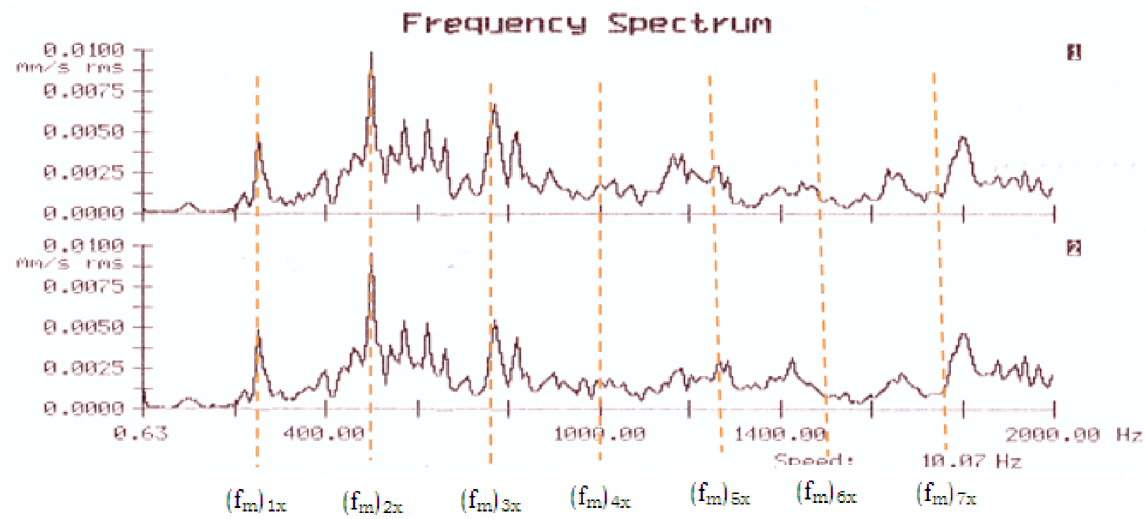

Figure 4.31: - Frequency spectrum for gear pair 3 with fault; with T = 1.5 N-m at the speed of 10.00 Hz (600RPM)

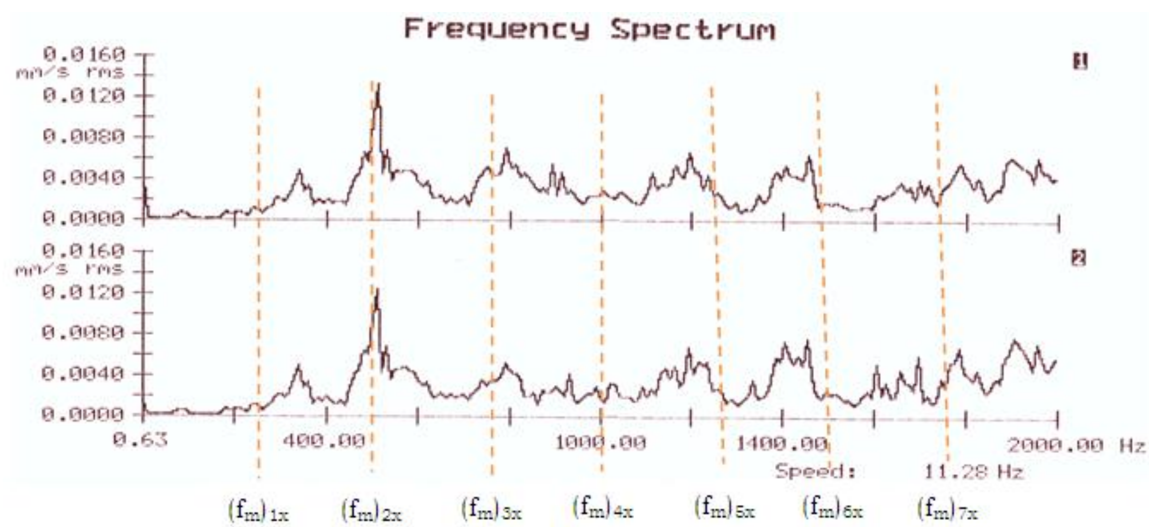

Figure 4.32: - Frequency spectrum for gear pair 3 with fault; with T = 2.5 N-m at the speed of 10.00 Hz (600RPM)

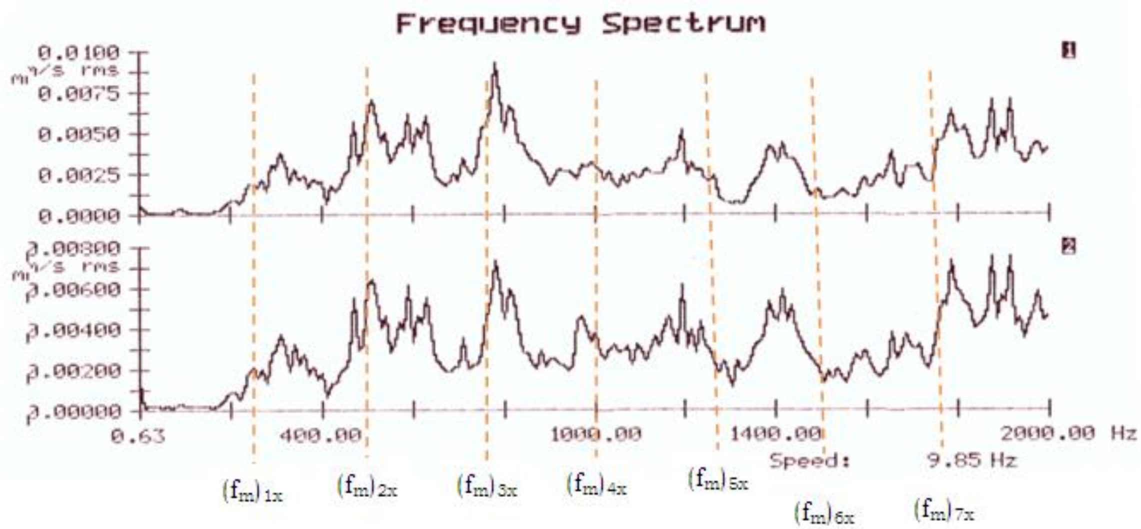

Figure 4.33: - Frequency spectrum for gear pair 3 with fault; with T = 3.5 N-m at the speed of 10.00 Hz (600RPM)

Table 4.6 :- pick list of frequency spectrum for gear pair 3 at the speed of 10.00 Hz (600 RPM) with different external load

External Load (T)	*signal*			
	1		*2*	
	Hz	mm/s rms	Hz	mm/s rms
0 N-m	472.642	0.004	472.646	0.005
	476.651	0.006	476.669	0.006
	509.816	0.004	509.815	0.004
	513.118	0.005	513.143	0.005
	522.768	0.005	522.820	0.004
	834.163	0.004	543.087	0.004
	854.158	0.004	1426.984	0.004
	892.660	0.004	1442.024	0.005
	898.240	0.004	1908.097	0.005
	1908.172	0.004	1971.861	0.004

1.5 N-m	Hz mm/s rms 300.647 0.029 317.362 0.031 334.036 0.033 367.450 0.024 384.173 0.023 417.529 0.050 484.310 0.030 501.080 0.036 534.405 0.040 584.456 0.022	Hz mm/s rms 300.647 0.027 317.364 0.029 334.042 0.032 367.449 0.022 384.161 0.020 417.530 0.046 484.309 0.029 501.083 0.036 534.402 0.040 584.455 0.022
2.5 N-m	Hz mm/s rms 483.925 0.007 500.684 0.008 508.612 0.010 510.677 0.013 530.592 0.007 794.809 0.007 1195.798 0.007 1456.993 0.006 1896.461 0.006 1953.182 0.006	Hz mm/s rms 500.670 0.008 508.595 0.009 510.681 0.012 530.604 0.007 1196.043 0.007 1400.978 0.007 1403.113 0.007 1457.041 0.008 1916.223 0.007 1953.193 0.007
3.5 N-m	Hz mm/s rms 510.874 0.007 513.188 0.006 779.261 0.006 781.759 0.009 796.322 0.007 815.691 0.007 822.131 0.006 1785.037 0.006 1872.435 0.007 1914.438 0.007	Hz mm/s rms 510.874 0.007 513.188 0.006 779.261 0.006 781.759 0.009 796.322 0.007 815.691 0.007 822.131 0.006 1785.037 0.006 1872.435 0.007 1914.438 0.007

Discussion

The change in mashing vibration with respect to the external load is not uniform. Because as we applied an external load the vibration amplitude increased up to 1.5 N-m torque; then it starts to decline in magnitude as the force increased. But it is not continued by increasing in magnitude. The curve is sinusoidal type.

The variation of the 2nd mashing vibration with external load is shown with the following curve.

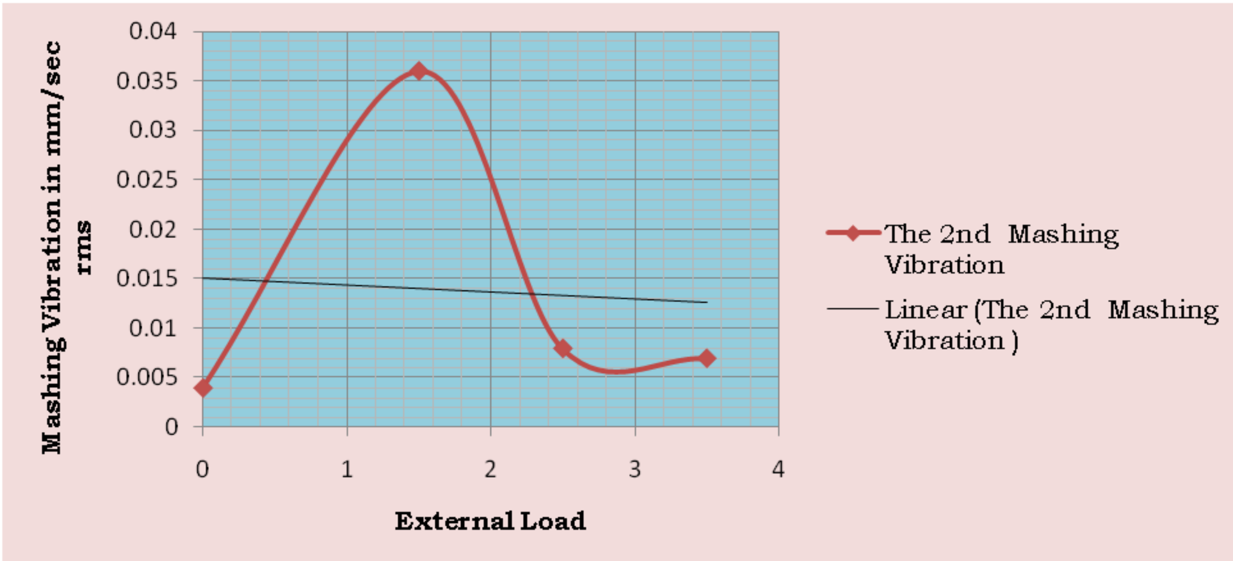

Figure 4.34: - variation of the second mashing vibration with respect to external load for gear pair 2

Conclusion

The general conclusion from these three results is as follows:-

i. As the magnitude of an external load increased the magnitude of vibration amplitude on the mashing frequency decreased.

ii. The overall vibration amplitude is totally affected by the presence of external load both in positive and negative direction. *i.e.:*

   - ✓ *For the gear which is free from gear fault the vibration amplitude is decreasing as the external load is applied. And as this load increasing the vibration amplitude change goes constant.*
   - ✓ *But for the case of gear which is with gear fault the vibration amplitude magnitude increased up to 1.5 N-m torque and then the value start to decline up to the 2.5 N-m torque. Finally the gear vibration amplitude increase in magnitude with external load.*

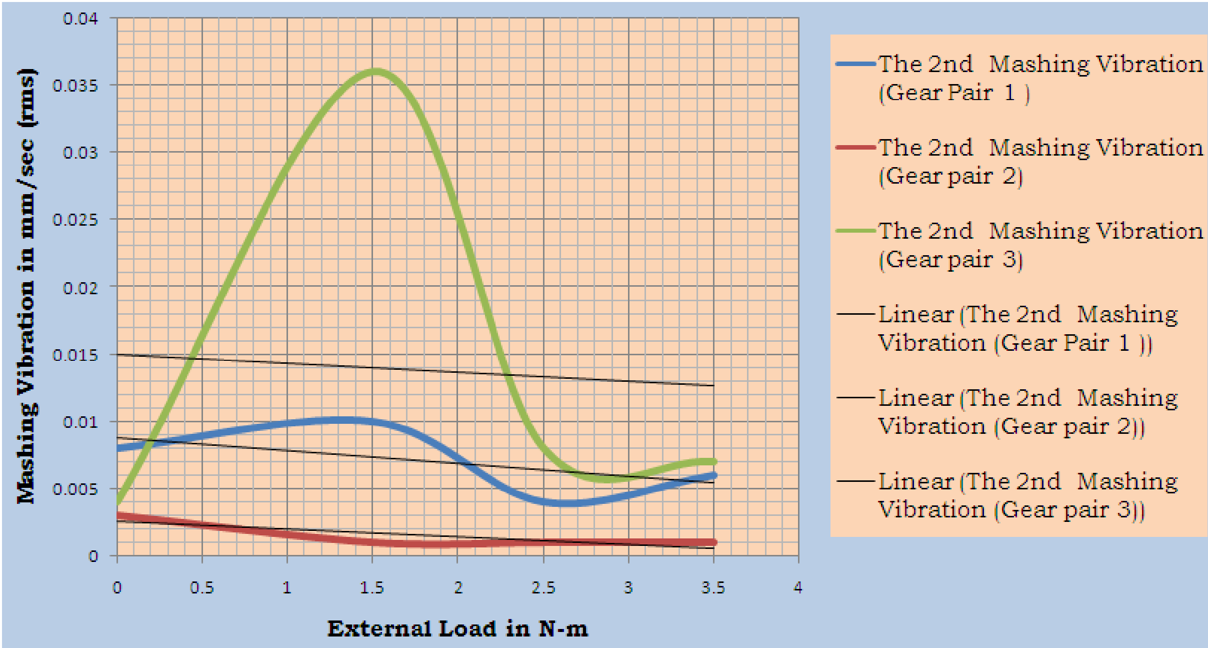

Figure 4.35: - variation of the second mashing vibration with respect to external load for all gear pair

4.4. Effect of input speed on the vibration signature of the gearbox

4.4.1. Gear Pair 1

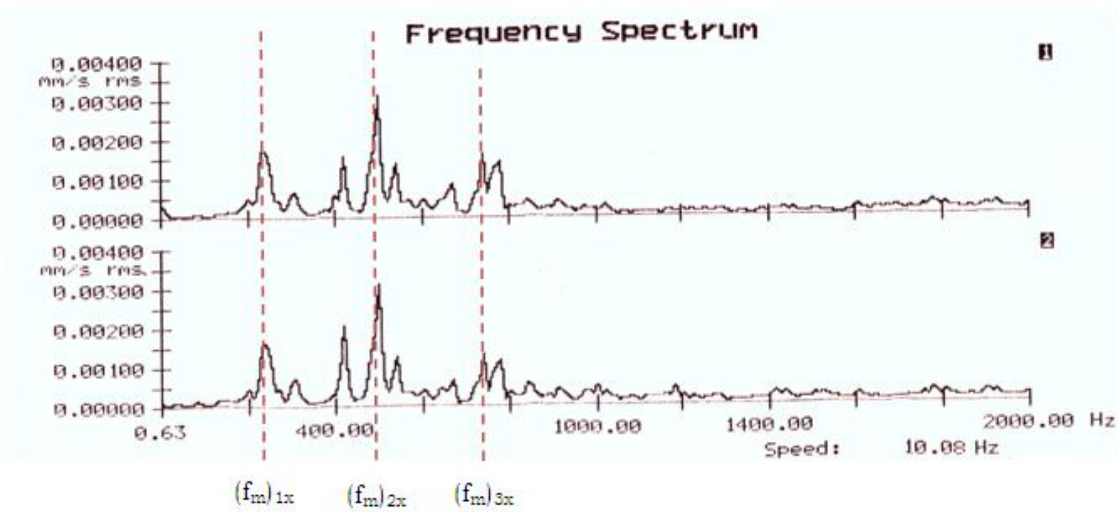

Figure 4.36: - Frequency spectrum for gear pair 1 with fault at the speed of 600 RPM (10.00Hz)

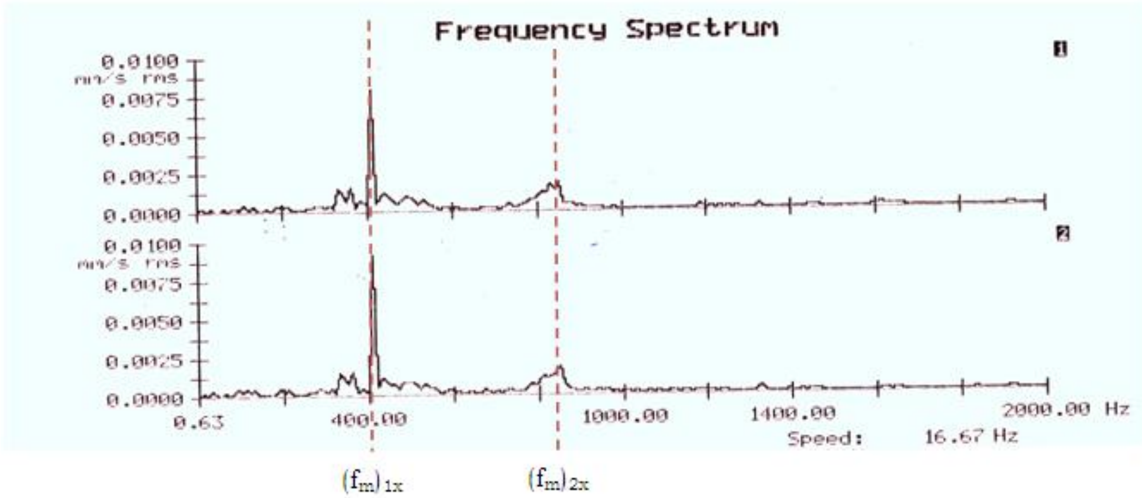

Figure 4.37: - Frequency spectrum for gear pair 1 with fault at the speed of 1000 RPM (16.67Hz)

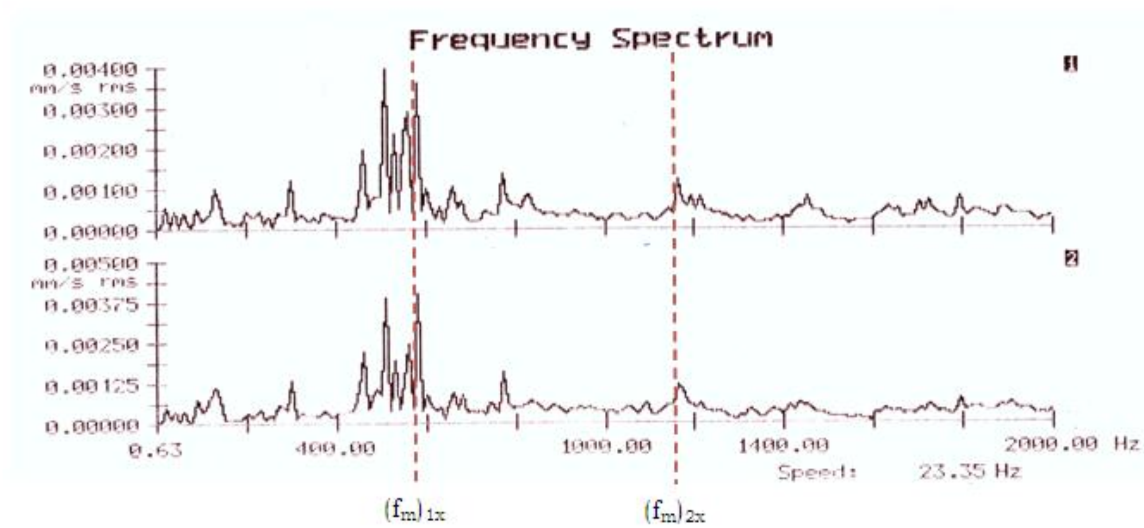

Figure 4.38: - Frequency spectrum for gear pair 1 with fault at the speed of 1400 RPM (23.33 Hz)

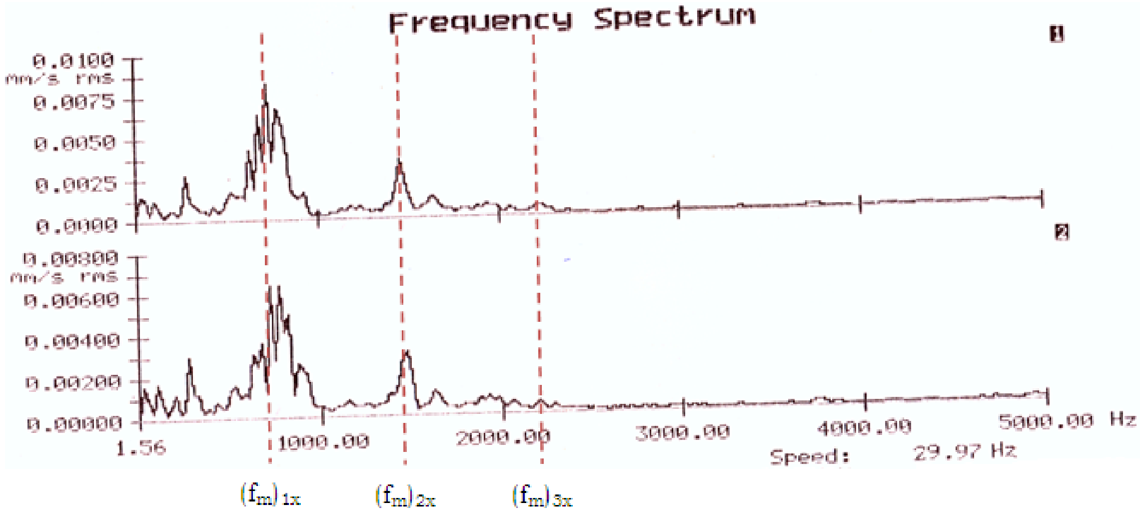

Figure 4.39: - Frequency spectrum for gear pair 1 with fault at the speed of 1800 RPM (30.00 Hz)

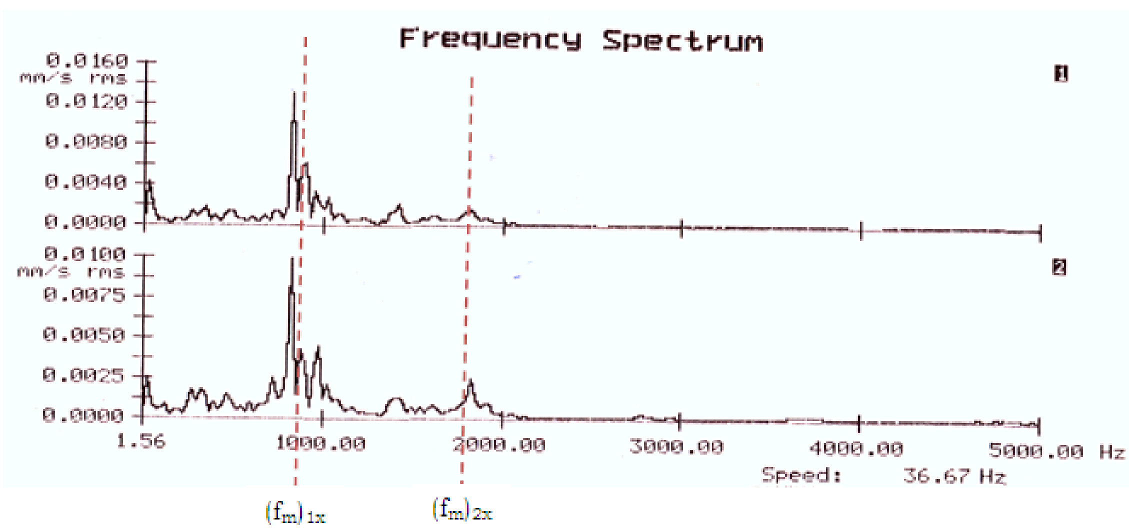

Figure 4.40: - Frequency spectrum for gear pair 1 with fault at the speed of 2200 RPM (36.67 Hz)

Tabel 4.7:- pick list of frequency spectrum for gear pair 1 at the speed of 600 RPM (10.00 Hz), 1000 RPM (16.67Hz), 1400 RPM (23.33 Hz), 1800 RPM (30.00 Hz), 2200 RPM (36.67 Hz)

External Load (T)	*signal*	
	1	*2*
600 RPM (10 Hz)	Hz mm/s rms 232.085 0.002 242.230 0.002 252.324 0.001 423.821 0.002 484.413 0.001 494.432 0.002 504.554 0.003 544.918 0.001 746.750 0.002 786.932 0.001	Hz mm/s rms 232.085 0.002 242.233 0.002 252.321 0.001 423.816 0.002 484.403 0.001 494.433 0.002 504.557 0.003 544.921 0.001 746.756 0.001 786.946 0.001
1000 RPM (16.67)	Hz mm/s rms 333.497 0.002 350.073 0.001 366.794 0.002 416.785 0.008 450.026 0.001 817.363 0.001 827.407 0.001 832.042 0.002 834.036 0.002 850.599 0.002	Hz mm/s rms 333.497 0.002 350.066 0.001 366.792 0.002 416.787 0.009 433.525 0.001 449.977 0.001 817.348 0.001 827.513 0.001 833.747 0.001 850.636 0.002
1400 RPM (23.33)	Hz mm/s rms 140.093 0.001 303.697 0.001 467.099 0.002 513.862 0.004 537.285 0.002 560.524 0.003 583.630 0.004 606.458 0.001 660.596 0.001 777.399 0.001	Hz mm/s rms 140.104 0.001 303.716 0.001 467.131 0.002 490.472 0.001 513.916 0.004 537.281 0.002 560.639 0.002 584.035 0.004 777.388 0.002 1167.520 0.001

1800 RPM (30.00 Hz)	Hz — mm/s rms 299.911 — 0.003 629.766 — 0.004 689.949 — 0.006 720.035 — 0.003 749.831 — 0.008 779.889 — 0.007 809.831 — 0.006 840.248 — 0.005 1470.187 — 0.004 1500.167 — 0.002	Hz — mm/s rms 299.903 — 0.003 629.747 — 0.003 689.998 — 0.004 749.840 — 0.006 779.836 — 0.006 809.845 — 0.004 840.178 — 0.005 900.511 — 0.003 1470.229 — 0.003 1500.234 — 0.003
2200 RPM (36.67Hz)	Hz — mm/s rms 36.637 — 0.004 73.886 — 0.002 806.506 — 0.002 843.589 — 0.013 880.404 — 0.006 917.010 — 0.006 954.245 — 0.003 990.501 — 0.002 1027.089 — 0.003 1430.896 — 0.002	Hz — mm/s rms 36.624 — 0.003 733.322 — 0.002 806.650 — 0.003 843.533 — 0.010 880.191 — 0.004 917.315 — 0.003 953.689 — 0.003 990.312 — 0.005 1026.854 — 0.002 1832.505 — 0.003

Discussion

The curve and table presented above shows as the vibration signature is affected by the magnitude of input speed variation. As the input speed increased in magnitude the vibration amplitude around the 1st, 2nd, and 3rd meshing frequency increased in magnitude. But after the 3rd meshing frequency there is no any change in magnitude. There is no change in the harmonic frequency with the variation of the input speed of the gear box.

The side band around meshing frequency increased in number and magnitude as we increased the input speed.

4.4.2. Gear Pair 2

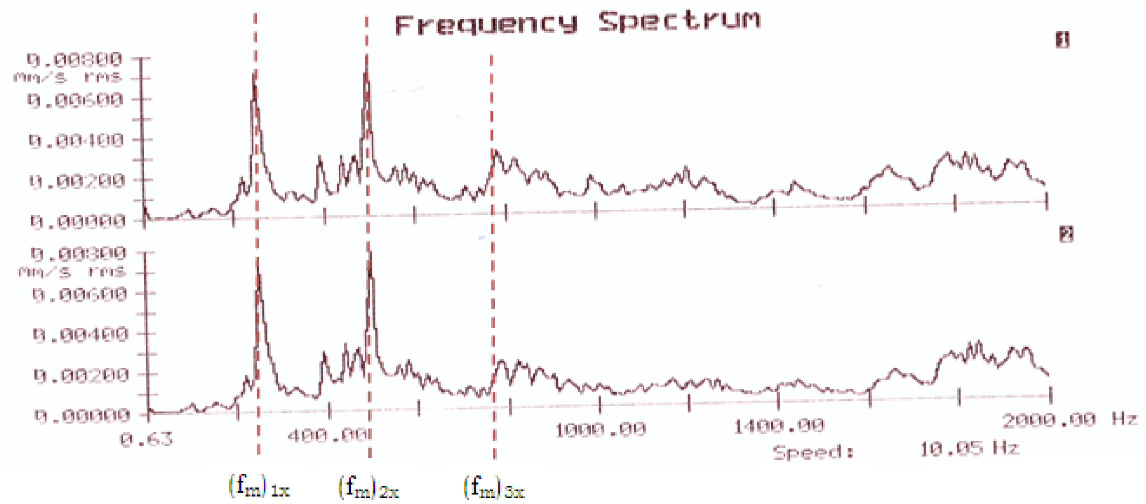

Figure 4.41: - Frequency spectrum for gear pair 2 without fault at the speed of 600 RPM (10.00Hz)

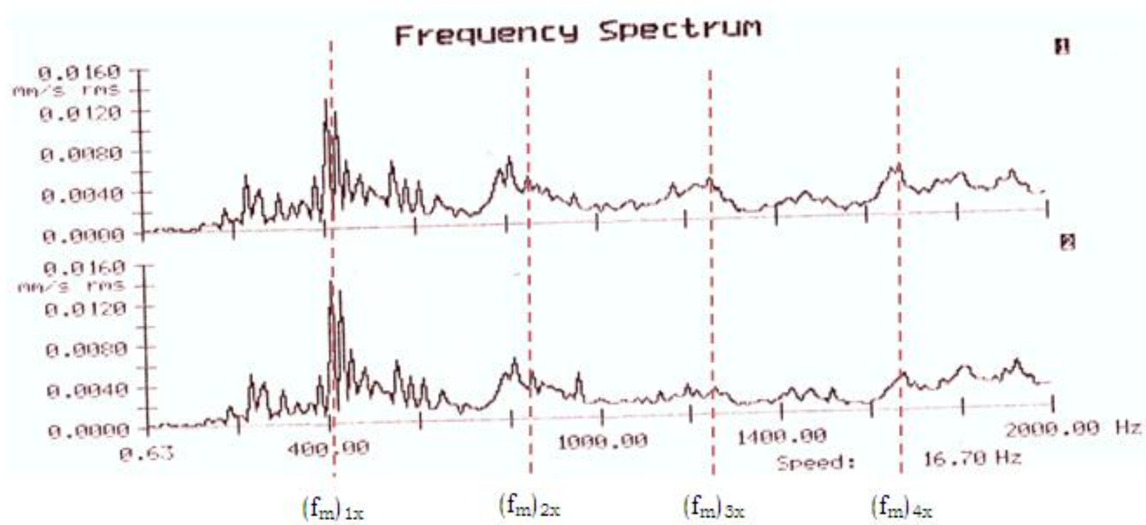

Figure 4.42: - Frequency spectrum for gear pair 2 without fault at the speed of 1000 RPM (16.67Hz)

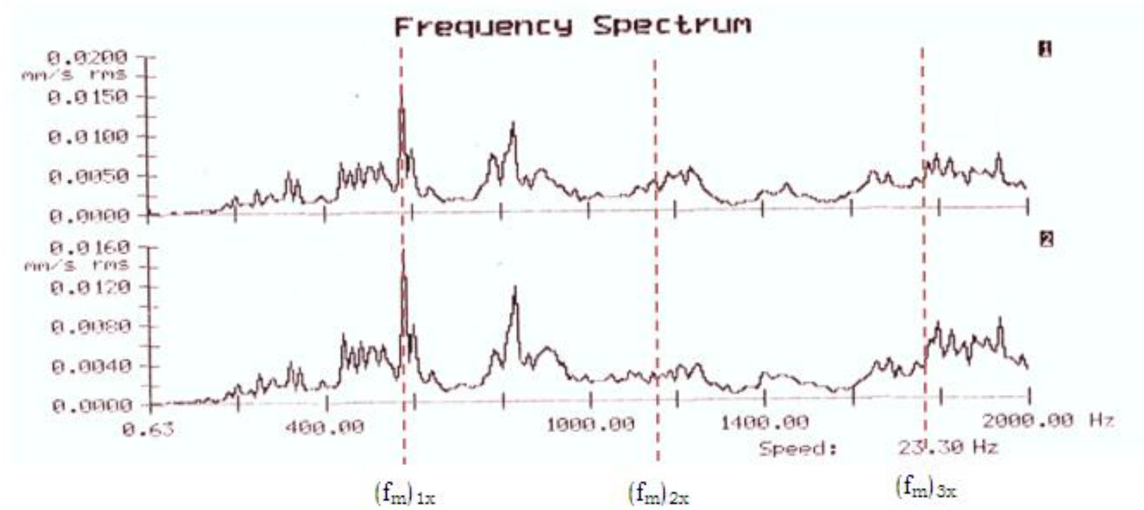

Figure 4.43: - Frequency spectrum for gear pair 2 without fault at the speed of 1400 RPM (23.33 Hz)

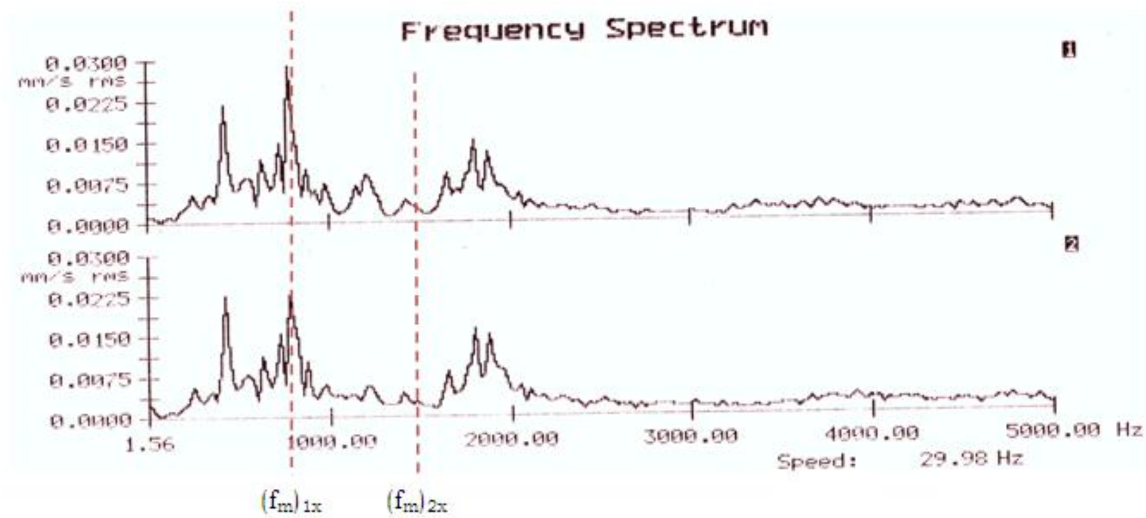

Figure 4.44: - Frequency spectrum for gear pair 2 without fault at the speed of 1800 RPM (30.00 Hz)

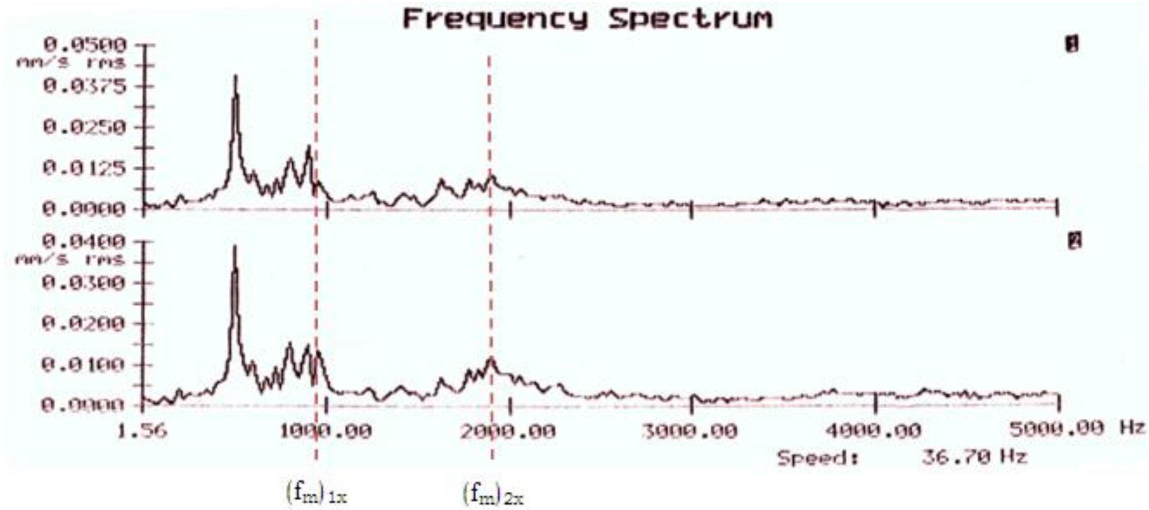

Figure 4.45: - Frequency spectrum for gear pair 2 without fault at the speed of 2200 RPM (36.67 Hz)

Tabel 4.8:- pick list of frequency spectrum for gear pair 2 at the speed of 600 RPM (10.00 Hz), 1000 RPM (16.67Hz), 1400 RPM (23.33 Hz), 1800 RPM (30.00 Hz), 2200 RPM (36.67 Hz)

External Load (T)	*signal*			
	1		*2*	
	Hz	mm/s rms	Hz	mm/s rms
600 RPM (10 Hz)	251.362 261.396 392.072 445.724 472.662 492.648 495.815 502.715 516.037 790.133	0.007 0.005 0.003 0.003 0.003 0.004 0.005 0.008 0.003 0.003	251.362 261.395 271.435 392.090 442.405 445.725 472.666 492.632 495.810 502.714	0.007 0.005 0.003 0.003 0.003 0.003 0.003 0.005 0.005 0.008

1000 RPM (16.67)	Hz 233.933 384.353 417.857 434.500 451.332 484.547 551.556 796.416 813.308 819.163	mm/s rms 0.005 0.005 0.013 0.011 0.007 0.005 0.006 0.005 0.005 0.007	Hz 233.939 384.356 417.855 434.499 451.337 484.543 551.557 796.302 819.109 1921.803	mm/s rms 0.005 0.005 0.014 0.013 0.007 0.005 0.006 0.005 0.006 0.005
1400 RPM (23.33)	Hz 489.335 582.407 605.686 786.858 793.811 799.857 817.100 830.915 1798.254 1939.333	mm/s rms 0.006 0.016 0.008 0.007 0.006 0.007 0.007 0.011 0.007 0.007	Hz 442.706 489.338 582.403 605.685 817.330 830.957 1798.234 1822.269 1876.442 1939.348	mm/s rms 0.007 0.006 0.015 0.008 0.007 0.012 0.008 0.007 0.006 0.008
1800 RPM (30.00 Hz)	Hz 449.817 629.645 749.864 779.640 809.615 819.026 840.071 1804.614 1816.021 1892.338	mm/s rms 0.022 0.012 0.014 0.029 0.019 0.011 0.013 0.015 0.010 0.013	Hz 449.802 629.624 749.816 779.629 809.628 839.864 1804.523 1816.158 1880.980 1892.269	mm/s rms 0.023 0.011 0.016 0.023 0.018 0.014 0.016 0.011 0.012 0.015
2200 RPM (36.67Hz)	Hz 476.806 513.657 550.066 623.633 788.543 806.768 815.459 880.492 917.050 1923.413	mm/s rms 0.014 0.041 0.017 0.012 0.012 0.016 0.014 0.012 0.019 0.011	Hz 476.806 513.669 550.087 623.675 806.788 815.528 880.454 917.079 954.167 1923.394	mm/s rms 0.013 0.039 0.017 0.011 0.015 0.016 0.012 0.015 0.014 0.012

Discussion

As the input speed increased in magnitude the vibration amplitude around the meshing frequency increased in magnitude. But for some the case the vibration amplitude is shifted to the left and right direction of the meshing frequency (see the curve for 2200 RPM).

The side band around meshing frequency increased in number and magnitude as we increased the input speed.

4.4.3. Gear Pair 3

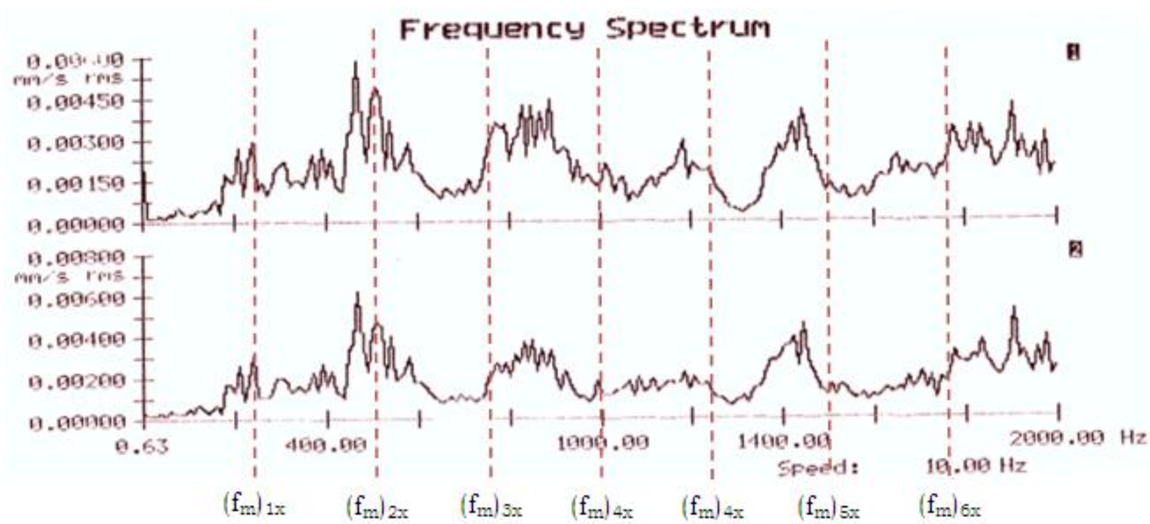

Figure 4.46: - Frequency spectrum for gear pair 3 without fault at the speed of 600 RPM (10.00Hz)

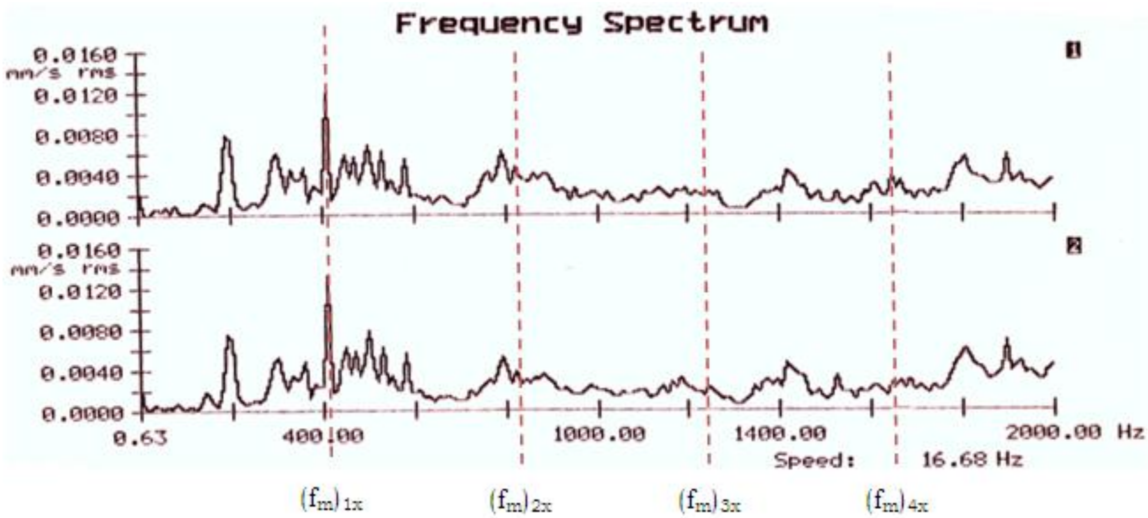

Figure 4.47: - Frequency spectrum for gear pair 3 without fault at the speed of 1000 RPM (16.67Hz)

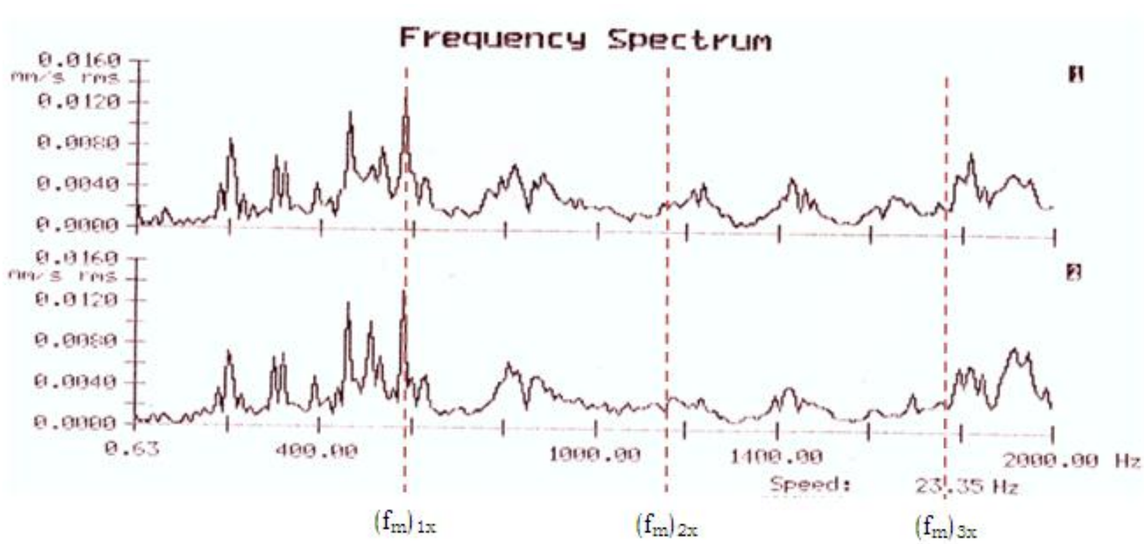

Figure 4.48: - Frequency spectrum for gear pair 3 without fault at the speed of 1400 RPM (23.33 Hz)

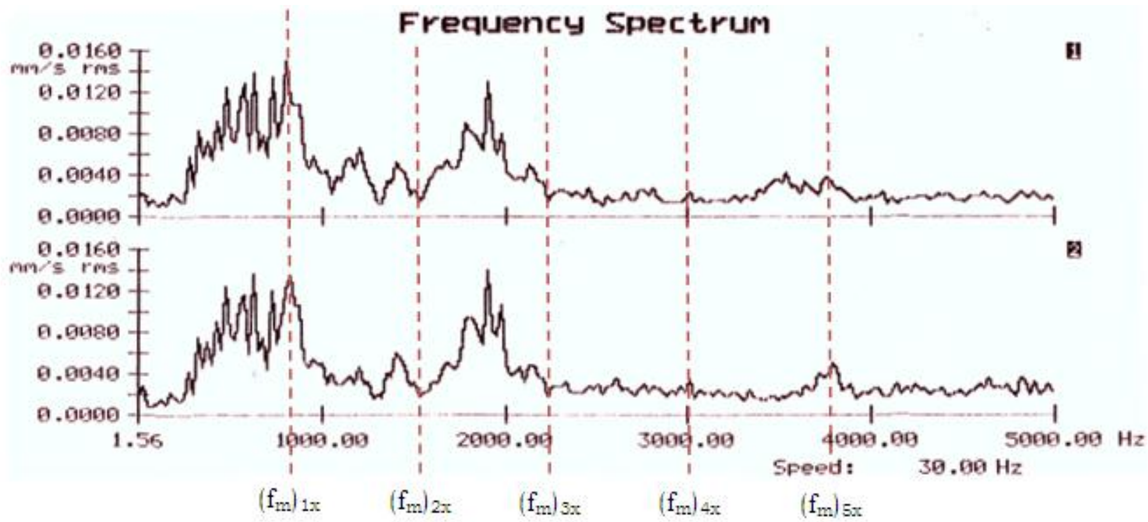

Figure 4.49: - Frequency spectrum for gear pair 3 without fault at the speed of 1800 RPM (30.00 Hz)

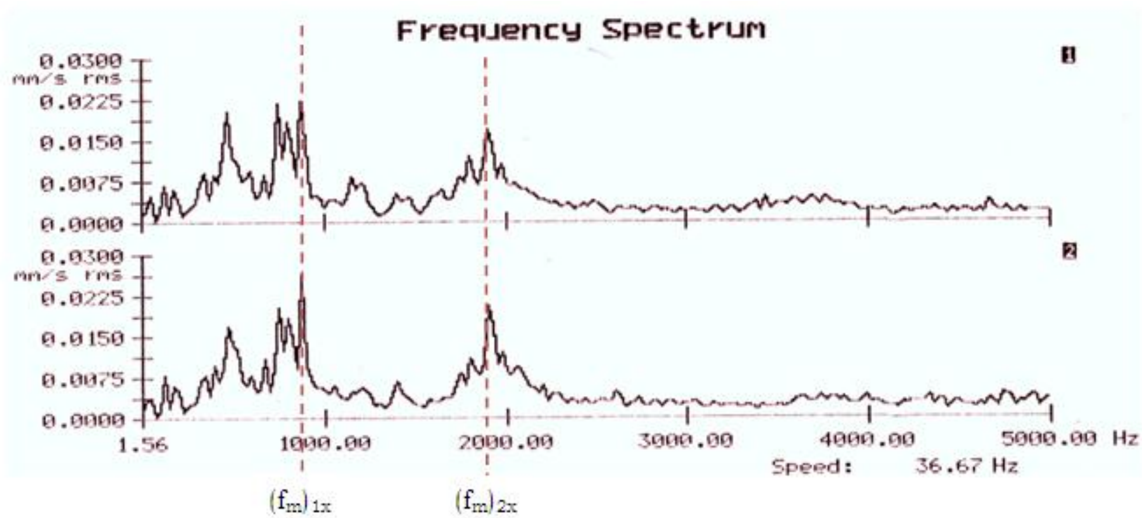

Figure 4.50: - Frequency spectrum for gear pair 3 without fault at the speed of 2200 RPM (36.67 Hz)

Tabel 4.9:- pick list of frequency spectrum for gear pair 3 at the speed of 600 RPM (10.00 Hz), 1000 RPM (16.67Hz), 1400 RPM (23.33 Hz), 1800 RPM (30.00 Hz), 2200 RPM (36.67 Hz)

External Load (T)	*signal*			
	1		*2*	
	Hz	mm/s rms	Hz	mm/s rms
600 RPM (10 Hz)	472.642	0.004	472.646	0.005
	476.651	0.006	476.669	0.006
	509.816	0.004	509.815	0.004
	513.118	0.005	513.143	0.005
	522.768	0.005	522.820	0.004
	834.163	0.004	543.087	0.004
	854.158	0.004	1426.984	0.004
	892.660	0.004	1442.024	0.005
	898.240	0.004	1908.097	0.005
	1908.172	0.004	1971.861	0.004
1000 RPM (16.67)	200.259	0.008	200.249	0.008
	300.401	0.006	417.119	0.013
	417.105	0.012	450.502	0.006
	450.506	0.006	472.884	0.006
	472.866	0.006	500.562	0.008
	500.562	0.007	533.797	0.006
	533.794	0.006	584.203	0.006
	798.143	0.006	1804.394	0.006
	1804.349	0.006	1894.330	0.006
	1896.312	0.006	1896.319	0.007
1400 RPM (23.33)	210.094	0.009	210.134	0.007
	303.516	0.007	326.946	0.007
	326.958	0.006	466.952	0.012
	466.820	0.011	513.677	0.010
	513.538	0.006	537.134	0.007
	536.903	0.008	583.637	0.013
	583.559	0.013	1900.734	0.008
	821.109	0.006	1915.017	0.008
	1815.583	0.008	1937.135	0.007
	1900.744	0.006	1949.348	0.008

	Hz	mm/s rms	Hz	mm/s rms
1800 RPM (30.00 Hz)	479.827	0.012	479.825	0.012
	569.890	0.011	569.892	0.011
	599.619	0.013	599.573	0.012
	629.390	0.014	629.434	0.014
	749.925	0.013	749.901	0.012
	810.432	0.015	810.325	0.012
	816.934	0.012	816.948	0.012
	838.983	0.011	839.124	0.013
	869.147	0.011	1909.294	0.014
	1909.318	0.013	1993.904	0.011
	Hz	mm/s rms	Hz	mm/s rms
2200 RPM (36.67Hz)	476.599	0.020	476.719	0.017
	503.592	0.012	503.160	0.013
	770.536	0.022	538.975	0.012
	807.248	0.019	770.513	0.020
	844.157	0.014	783.208	0.013
	880.549	0.022	807.323	0.018
	917.208	0.014	844.193	0.014
	1802.650	0.012	880.509	0.027
	1915.684	0.017	1915.436	0.020
	1941.006	0.014	1940.826	0.017

Discussion

The curve and table presented above shows as the vibration signature is affected by the magnitude of input speed variation. As the input speed increased in magnitude the vibration amplitude around the 1st, 2nd, and 3rd meshing frequency increased in magnitude. Specially for the last case (speed = 36.67 Hz) the vibration amplitude increased in large magnitude

The side band around meshing frequency increased in number and magnitude as we increased the input speed.

<u>Conclusion</u>

The general conclusion from these three results is; as the magnitude of an input speed increased the magnitude of vibration amplitude around the mashing frequency increased. But the variation of vibration amplitude is not having a sensible difference for the frequency greater than the 3rd meshing frequency.

CHAPTER FIVE

5. CONCLUSIONS AND FUTURE WORK

5.1. Conclusion

This thesis is based on experimental gear transmission systems for detecting fault within gear flank in a gearbox. The study includes the applications and developments of methodology for the detection of the gear tooth damage from the vibration signatures in on-line Health Monitoring system. The experimental test rig for detecting the damages of gear flank in a gearbox was established. The vibration signals for damaged gear tooth were acquired. Experimental results were presented in the frequency domain format. The presence of the tooth damage was identified and quantified from the acquired vibration signature for both the reference gear and faulted gears. As main objective of this work the type of signals which are used as fault indicators was generated for both with external load and without external load conditions.

As discussed in chapter six, if gear flank fault induced within the gear box; there is a variation in magnitude and also profile change in a vibration signature (frequency spectrum data). And as a final conclusion; the presence of gear flank fault is investigated by the following fault indicators:-

1. Magnitude change within the mashing frequency
   a) Gear box without external load
      - ✓ The gearbox with gear flank fault shows increment in magnitude of the vibration amplitude around/at the mashing frequency, special the first and the second mashing.

b) Gear box with external load
    - ✓ The presence of an external load affects the magnitude of the 1st mashing vibration magnitude. As the magnitude of an external load increased the magnitude of the 1st mashing vibration decreased.

c) Gearbox input speed variation
    - ✓ As the input speed increased in magnitude the magnitude of vibration amplitude increased. So that the investigation of the fault within the gearbox is must be within the same speed. Because the variation (the increment/decrement) is may be generated due the variation of the speed.

So that the 1st and the 2nd mashing vibration is good fault indicators of gear flank fault.

2. Formation of side bands (secondary bands)
    - ✓ Gear box with gear flank fault (with and without external load) there is a formation of sidebands around the mashing frequency.
    - ✓ This side bands increase in number and magnitude if the gearbox with fault.

Therefore, sidebands are the other good fault indicators of gear flank fault.

But for both condition the harmonic frequency is not used as a fault indicator.

5.2. Future work

In this thesis work mainly addressed the variation of vibration signature with respect to gear fault on the gear flank only (with and without load, for single stag gear box). But there are different gear fault which can be created on the gears, for different stage of gear box. So that to fully diagnosis any type of fault for any type of gearbox, the following additional work is suggested:-

1. Conduct an experiment for the other type of gear fault and generate a fault indicator for each case.
2. Conduct an experiment for a gearbox that have more than one gear and one pinion (gearbox with n-stage , where n is greater than one)
3. Generate the effect of lubrication type and amount within the gearbox on the created vibration signature.
4. Conduct the same experiment on automobile gearbox to identify a gear problem by using vibration signature analysis.

REFERENCE

[1]. John J. Coy, Dennies P. Townsend and Erwin V. Zaretsky ,“Gearing”, NASA Reference Publication , Lwis Research Center , Cleveland, Ohio, 2011.

[2]. Mircea Rades, “Dynamics of Machinery III”, Editura PRINTECH, 2008.

[3]. B. David Forrester, “Advanced Vibration Analysis Techniques for Fault Detection and Diagnosis in Geared Transmission Systems”, PHD Thesis, Swinburne University of Technology, 1996.

[4]. Pratesh Jayaswal, A. K. Wadhwani, and K . B. M ulchandani, “Machine Fault Signature Analysis”, International Journal of Rotating Machinery, Hindawi Publishing Corporation, 2008.

[5]. V.V. Polyshchuk, F.K. Choy, and M.J. Braun, “Gear fault detection with time-frequency based parameter NP4”, International Journal of Rotating Machinery, 2002.

[6]. Rusmir Bajric, Denijal Sprecic, Ninoslav Zuber, “Review of vibration signal processing techniques towards gear pairs damage identification”, International Journal of Engineering & Technology IJET-IJENS Vol: 11 No: 04 , 2011

[7]. P. K. Chakravorty, V. Bhujanga Rao, K. V. V. S. S. Murthy & T. Unnikrishnan, “Vibration Signature Analysis of Shipboard Machinery”, Naval Science & Technological Laboratory, Visakhapatnam-530006, Def Sci J, Vol 32, No 1, 1982.

[8]. Ruiwei Wu, “Identification of bearing and gear tooth damages from experimental vibration signatures”, Master Thesis, University of Akron, 2007

[9]. Johan Wändell, “Multistage gearboxes: vibration based quality control”, Licentiate Thesis, Royal Institute of Technology, 2006

[10]. Ming Yang, “Early Fault Detection Scheme and Optimized CBM Strategy for Gear Transmission System Operating under Varying Loads”, PHD Thesis, University of Toronto Department of Mechanical and Industrial Engineering, Toronto, 2011.

[11]. Commtest Instruments Ltd , “Beginner's Guide to Machine Vibration”, New Zealand,2006

[12]. Jonathan C. Young, “vibration analysis using a MEMS accelerometer”, Master Thesis, naval postgraduate school, Monterey, California, December 2006.

[13]. Dimarogonas, A.D., Papadopolous, C.A. "Crack detection in turbine rotors", Dynamics of Rotating Machinery, Hemisphere Publishing Corp, New York, NY, 1988, pp.227-39.

[14]. Imam, I., Azzaro, S.H., Bankert, R.J., Scheibel, J. "Development of an online rotor crack detection and monitoring system", Journal of Vibration, Acoustics, Stress, and Reliability in Design, Vol. 111, 1989, pp.241-250.

[15]. Prabhakar, S.; Sekhar, A. S.; Mohanty, A. R., "Detection and monitoring of cracks using mechanical impedance of rotor-bearing system", The Journal of the Acoustical Society of America, Vol. 110, Issue: 5, November 2001. pp. 2351-2359.

[16]. Gounaris, G.D., Papadopoulos, C.A. "Crack identification in rotating shafts coupled response measurements", Engineering Fracture Mechanics, Vol. 69, 2002, pp.339-352.

[17]. Zakrajsek, James J., "An Investigation of Gear Mesh Failure Prediction Techniques", NASA/Technical Memorandum 102340, November 1989.

[18]. Decker, Harry J.; Lewicki, David G., "Spiral Bevel Pinion Crack Detection in a Helicopter Gearbox", NASA/TM—2003-212327, June 2003.

[19]. McEvily, A.J., "Failures in inspection procedures: case studies", Engineering Failure Analysis, Vol. 11, Issue: 2, April 2004, pp. 167-176.

[20]. Abeele, K.V.; Carmeliet, J.; Wevers, M., "Quantification of micro-damage in slate tiles: comparison of nonlinear acoustics resonance experiments with visual and x-ray diagnosis", ISNA Gottingen, 1999, pp.1-4.

[21]. Inagaki, Terumi; Ishii, Toshimitsu; Iwamoto, Toshikatsu, "On the NDT and E for the Diagnosis of Defects Using Infrared Thermography", NDT & E International, Vol. 32, and Issue: 5, July 1999, pp. 247-257.

[22]. McBride, S.L.; Hong, Y.; Pollard, M., "Enhanced fatigue crack detection in ageing aircraft using continuous acoustic emission". Review of Progress in Quantitative Non-destructive Evaluation, Vol. 12, 1993, pp. 2191–2197.

[23]. Roberts, T.M.; Talebzadeh, M., "Acoustic emission monitoring of fatigue crack propagation", Journal of Constructional Steel Research, Vol. 59, Issue: 6, June 2003, pp. 695-712.

[24]. Fang, D; Berkovits, A.; "Fatigue design model based on damage mechanisms revealed by acoustic emission measurements", Journal of Engineering Materials and Technology, Transactions ASME, Vol. 117, 1995, pp. 201–208.

[25]. Sundaresan, Mannur J; Ghoshal, Anindya; Schulz, Mark J, "A continuous sensor for damage detection in bars", Smart Materials and Structures, Vol. 11, Issue: 4, Aug. 2002, pp. 475–488.

[26]. Prabhakaran, Anand; Jagga, C.R., "Condition monitoring of steam turbine-generator through contamination analysis of used lubricating oil", Tribology International, Vol. 32, Issue: 3, March 1999, pp. 145-152.

[27]. Peng, Z.; Kessissoglou, N.J.; Cox, M., "A study of the effect of contaminant particles in lubricants using wear debris and vibration condition monitoring techniques", Wear, Vol. 258, Issue: 11-12, June 2005, pp. 1651-1662.

[28]. Cheiky-Zelina, M., “Ferrography: A Successful Preventive Maintenance Program for Gearboxes”, Proceedings of the 3rd International Machinery Monitoring and Diagnostic Conference, Society for Experimental Mechanics, Inc., Bethel, CT, 1991, pp. 9-13.

[29]. Jones, William R.; Jr.; Loewenthal, Stuart H., “Ferrographic Analysis of Wear Debris from Full-Scale Bearing Fatigue Tests”, NASA Technical Paper 1511 c.1, Sept. 1979.

[30]. Nemarich, C. P., H.K. Whitesel, and A. Sarkady, “On-Line Wear Particle Monitoring Based on Ultrasonic Detection and Discrimination”, US Navy DTRCPAS-89-7, May 1989.

[31]. Whitlock, Robert R., “X-Ray Methods for Monitoring Machinery Condition”, JCPDS-International Centre for Diffraction Data, 1997.

[32]. Lewicki, David G.; Blanchette, D.M.; Biron, G., “Evaluation of an Oil-Debris Monitoring Device for Use in Helicopter Transmissions”, NASA TM-105830, August 1992.

[33]. Dempsey, Paula J., “A Comparison of Vibration and Oil Debris Gear Damage Detection Methods Applied to Pitting Damage”, NASA/TM-210371, Sept. 2000.

[34]. Dempsey, Paula J., “Gear Damage Detection Using Oil Debris Analysis”, NASA TM–210936, Sept. 2001.

[35]. Dempsey, Paula J.; Lewicki, David G.; Decker, Harry J., “Investigation of Gear and Bearing Fatigue Damage Using Debris Particle Distributions”, NASA/TM—2004-212883, May 2004.

[36]. DiPasquale, F., “Field Experience with Quantitative Debris Monitoring”, Aircraft Gas Turbine Engine Monitoring Systems: An Update, SAE, Warrendale, PA, 1988, pp. 113-120 (Also, SAE Paper 871736, 1987).

[37]. Campbell, P., “On-Line Monitoring of Ferromagnetic Debris Concentration”, Current Practices and Trends in Mechanical Failure Prevention: Proceedings of the 44th Meeting of the Mechanical Failures Prevention Group, Vibration Institute, Willowbrook, IL, 1990, pp.131-139.

[38]. Choy, F.K.; Braun, M.J.; Polyshchuk, V.; Zakrajsek, J.J.; Handschuh, R.F.; and Townsend, D.P., “Analytical and Experimental Vibration Analysis of a Damaged Gear System”. Presented in the 1994 AGMA Fall Technical Meeting, St. Louis, Missouri, October 24-26, 1994.

[39]. Choy, F.K.; Huang, S.; Zakrajsek, J.J.; Handschuh, R.F.; Townsend, D.P., “Vibration Signature Analysis of a faulted Gear Transmission System”, Paper Presented in the 30th AIAA Joint Propulsion Conference, Indianapolis, Indiana, June 27-29, 1994.

[40]. Wang, Wilson Q.; Ismail, Fathy; Golnaraghi, M. Farid, “Assessment of Gear Damage Monitoring Techniques using Vibration Measurements”, Mechanical Systems and Signal Processing, Vol. 15, Issue: 5, September 2001, pp. 905-922.

[41]. Polyshchuk, V., “Detection and Quantification of the Gear Tooth Damage from the Vibration and Acoustic Signatures”, PhD Dissertation, the University of Akron, Akron, Ohio, May 1999.

[42]. Choy, F.K., Mugler, D.H., and Zhou, J., “Damage Identification of a Gear Transmission Using Vibration Signatures”, Journal of Mechanical Design, Trans. ASME, Vol. 125, June 2003, pp. 394-402.

[43]. Seibold, S.; Weinert, K., “A Time Domain Method for the Localization of Cracks in

Rotors”, Journal of Sound and Vibration, Vol. 195, Issue: 1, Aug. 1996, pp. 57-73.

[44]. McFadden, P. D., “Detecting fatigue cracks in gears by amplitude and phase demodulation of the meshing vibration”, Journal of Vibration, Acoustics, Stress, and Reliability in Design, Vol. 108, 1986, pp. 165-170.

[45]. McFadden, P. D., “Interpolation techniques for time domain averaging of gear vibration”, Mechanical Systems and Signal Processing, Vol. 3, 1989, pp. 87-97.

[46]. Ismail, F.; Martin, H.; Omar, F., “A statistical index for monitoring tooth cracks in a gearbox”, Proceedings of the ASME Biennial Conference on Vibration and Noise, Boston, MA, Vol. DE-84-1, 1995, pp. 1413-1418.

[47]. Stearns, S. D., Hush, D. R., “Digital Signal Analysis”, Prentice-Hall, 1990

[48]. Warner, Rebecca M, “Spectral analysis of time-series data”, The Guilford Press, 1998

[49]. Robert W. Ramirez, “The FFT Fundamentals and Concepts”, Prentice-Hall, 1985

[50]. Li, b.; Goddu, G.; Chow, M., “Detection of Common Bearing Faults Using Frequency-Domain Vibration Signals and a Neural Network Based Approach”, Proceedings of the American Control Conference, Philadelphia, Pennsylvania, June 1998, pp. 2032-2036.

[51]. Tang, H.; Cha, J.; Wang, Y., “The principle of cepstrum and its application in quantitative fault diagnostics of gears”, ASME Design Technical Conferences, Miami, FL, 1991, De-Vol. 38, pp. 141-144.

[52]. Wang, W. J.; McFadden, P. D., “Early detection of gear failure by vibration analysis – I. calculation of the time-frequency distribution”, Mechanical Systems and Signal Processing, Vol. 7, Issue: 3, May 1993, pp. 193-203.

[53]. Kar, Chinmaya; Mohanty, A. R., “Technical Note: Gearbox Health Monitoring through Multi-resolution Fourier Transform of Vibration and Current Signals”, Structural Health Monitoring, Vol. 5, Issue: 2, June 2006, pp. 195-200.

[54]. Mcfadden, P. D., and Wang, W. J., “Time Frequency Domain Analysis of Vibration Signal for Machinery Diagnostics (II) the weighted Wigner-Ville Distribution”, University of Oxford, Report No.: OUEL-1891, 1991.

[55]. Choy, F. K.; Chen, H.; Zhou, J., “Identification of Single and Multiple Teeth Damage in a Gear Transmission System”, Journal of Tribology Transactions, Vol. 49, 2006, pp. 297-304.

[56]. Neild, S.A.; McFadden, P.D.; Williams, M.S., “A review of time-frequency methods for structural vibration analysis”, Engineering Structures, Vol. 25, Issue 6, May 2003, pp.713-728.

[57]. Boashash, B.; Black, P. J., “An Efficient Real Time Implementation of the Winger-Ville Distribution”, IEEE Trans. on Acoustics, Speech, and Signal Processing, Vol. ASSP35, No.11, November 1987.

[58]. Shin, Y. S., Jeon, J. J., “Pseudo Wigner-Ville Time-Frequency Distribution and its Application to Machinery Condition Monitoring”, Journal of Shock and Vibration, Vol.1, Issue: 1, 1993, pp.65-76.

[59]. Wang, W.J., and Mcfadden, P. D., “Application of Wavelet to Gearbox Vibration Signals for Fault Detection”, Journal of Sound and Vibration, Vol. 192, Issue: 5, May 1996, pp. 927-939.

[60]. Qian, S., "Introduction to time-frequency and wavelet transforms", Prentice Hall PTR, 2002.

[61]. Ehrich, F.F., “Some Observations of Chaotic Vibration Phenomena in High-Speed Rotor dynamics,” J. of Vibrations and Acoustics, Trans. ASME, Vol. 113, January 1991, pp. 50-57.

[62]. Ehrich, F.F., “Observations of subcritical superharmonic and chaotic response in rotor dynamics”, ASME J. of Vibrations and Acoustics, Vol. 114, 1992, pp. 93–100.

[63]. Moon, F.C. “Chaotic Vibrations,” John Wiley and Sons, Inc., 1987.

[64]. Choy, F. K, Zhou, J., Braun, M. J, and Wang. L, “Vibration Monitoring and Damage Quantification of Faulty Ball Bearing”, ASME Journal of Lubrication Technology, Vol. 127, October 2005, pp. 776-783.

[65]. Farmer, J.D., Ott, E. and Yorke, J.A., “The Dimension of Chaotic Attractors,” Phsica, 7D, 1983, pp.153-180.

[66]. Oppenheim, A.V. and Schafer, R.W., Digital Signal Processing, Prentice Hall International, 1975.

APPENDICES

Gear and pinion design Information

Spur Gearing Component Wizard (Version 4.2.1031)

--- Guide

External Gearing - ISO

Calculation of geometry: Calculates the center distance according to the module, number of teeth, correction and helix direction

Distribution of Correction: In Reverse Ratio

Load calculation: Calculates the torque according to the power and speed

Strength calculation: Strength check calculation

--- Basic Parameters

Desired Gear Ratio = 3

Pressure Angle alpha = 20°

Addendum a* = 1 (= 2 mm)

Clearance c* = 0.25 (= 0.5 mm)

Root Fillet = 0.38 (= 0.76 mm)

Addendum of Basic Rack = 1.25 (= 2.5 mm)

Helix Angle beta = 0°

Module m = 2 mm

Center Distance aw = 100 mm

Product Center Distance a = 100 mm

Total Unit Correction = 0

Operating Pressure Angle alphaw = 20°

Circular Pitch p = 6.283 mm

Base Circular Pitch ptb = 5.904 mm

Contact Ratio = 1.7144 (1.7144 + 0)

Precision Specification 6

Limit Deviation of Helix Angle Fb = 0.01 mm

Limit Deviation of Axis Parallelity fx = 0.01 mm

Limit Deviation of Axis Parallelity fy = 0.005 mm

--- Gear 1

Number of Teeth = 25

Unit Correction = 0 (= 0 mm)

Pitch Diameter d = 50 mm

Base Circle Diameter db = 46.985 mm

Outside Diameter do = 54 mm

Root Diameter df = 45 mm

Work Pitch Diameter dw = 50 mm

Tooth Thickness s = 3.142 mm

Outside Tooth Thickness = 0.7198 (= 1.44 mm)

Facewidth = 15 mm

Facewidth Ratio = 0.3

Chordal Thickness T = 2.774 mm

Chordal Thickness Height ht = 1.495 mm

Chordal Dimension M = 21.365 mm / 4

Dimension Over (Between) Wires M = 53.04 mm

Wire Diameter dw = 3 mm

Limit Circumferential Run-out Fr = 0.016 mm

Limit Deviation of Axial Pitch fpt = ±0.007 mm

Limit Deviation of Basic Pitch fpb = ±0.0066 mm

--- Gear 2

Number of Teeth = 75

Unit Correction = 0 (= 0 mm)

Pitch Diameter d = 150 mm

Base Circle Diameter db = 140.954 mm

Outside Diameter do = 154 mm

Root Diameter df = 145 mm

Work Pitch Diameter dw = 150 mm

Tooth Thickness s = 3.142 mm

Outside Tooth Thickness = 0.7962 (= 1.592 mm)

Facewidth = 15 mm

Facewidth Ratio = 0.1

Chordal Thickness T = 2.774 mm

Chordal Thickness Height ht = 1.495 mm

Chordal Dimension M = 52.287 mm / 9

Dimension Over (Between) Wires M = 153.107 mm

Wire Diameter dw = 3 mm

Limit Circumferential Run-out Fr = 0.028 mm

Limit Deviation of Axial Pitch fpt = ±0.0085 mm

Limit Deviation of Basic Pitch fpb = ±0.008 mm

--- Load (Gear 1; Gear 2)

Power P = 1;   0.97 kW

Efficiency = 0.97

Speed n = 1000;   333.3333 rpm

Torque Mk = 9.5493;   27.7885 Nm

Tangential Force Ft = 381.9719 N

Radial Force Fr = 139.0264 N

Axial Force Fa = 0 N

Normal Force Fn = 406.486 N

Circumferential Velocity v = 2.618 m/s

Resonance Speed nE1 = 19960.53 rpm

 Strength Check According to ISO 6336:1996

Durability Lh = 10000 hour

--- Material Values

Tensile Strength = 640;   640 MPa

Yield Point in Tensile = 390;   390 MPa

Contact Fatigue Limit SigmaHlim = 1140;   1140 MPa

Bending Fatigue Limit SigmaFlim = 605;   605 MPa

Hardness in Tooth Core = 200;   200 HV

Hardness in Tooth Side = 600;   600 HV

Base Number of Load Cycles in Contact [10^6] = 100;   100

Base Number of Load Cycles in Bending [10^6] = 3;   3

Wöhler Curve Exponent for Contact = 10;   10

Wöhler Curve Exponent for Bending = 9;   9

Modulus of Elasticity in Tension [10^3] = 206;   206 MPa

Poisson's Ratio = 0.3;   0.3

Type of Treatment = 4;   4

--- Factors for Contact

Application Factor KA = 1.2

Dynamic Factor KHv = 1.104

Face Load Factor KHb = 1.92

Transverse Load Factor KHa = 1.313

Total KH = 3.339

One-time Overloading Factor KAS = 1

Elasticity Factor Ze = 189.81

Zone Factor Zh = 2.495

Contact Ratio Factor Zeps = 0.873

Single Pair Tooth Contact Factor ZB = 1.051; 1

Life Factor Zn = 1; 1

Lubricant Factor Zl = 0.962

Roughness Factor Zr = 1

Velocity Factor Zv = 0.963

Helix Angle Factor Zb = 1

Size Factor Zx = 1; 1

Work Hardening Factor Zw = 1

--- Factors for Bending

Application Factor KA = 1.2

Dynamic Factor KFv = 1.104

Face Load Factor KFb = 1.628

Transverse Load Factor KFa = 1.385

Total KF = 2.988

One-time Overloading Factor KAS = 1

Form Factor YFa = 2.633; 2.24

Stress Correction Factor YSa = 1.55; 1.665

Teeth with Grinding Notches Factor YSag = 1;   1

Helix Angle Factor Yb = 1

Contact Ratio Factor Yeps = 0.687

Alternating Load Factor Ya = 1;   1

Production Technology Factor Yt = 1;   1

Life Factor Yn = 1;   1

Notch Sensitivity Factor Yd = 1.283;   1.306

Size Factor Yx = 1;   1

Tooth Root Surface Factor Yr = 1

--- Results

Factor of Safety from Pitting SH = 1.613;   1.696

Factor of Safety from Tooth Breakage SF = 7.273;   8.101

Static Safety in Contact SHst = 3.668;   3.856

Static Safety in Bending SFst = 14.173;   15.506

Strength Check - True