A Study on Gearbox Noise with Induced Tooth Defects

A Dissertation submitted
in partial fulfilment of the requirements
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in

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by

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JULY, 2015

CERTIFICATE

I hereby declare that the thesis entitled "**A Study on Gearbox Noise with Induced Tooth Defects**" is an authentic record of my work carried out as requirements for the award of the degree of **Master of Engineering** in **Production Engineering** at **Thapar University, Patiala** under the supervision of **Dr. S.P. Nigam**, Visiting Professor, Department of Mechanical Engineering, Thapar University, Patiala during July, 2014 to July, 2015. No part of the matter embodied in this report has been submitted to any other university or institute for the award of any degree.

Date: 15/07/2015

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It is certified that the above statement made by the student is correct to the best of my knowledge and belief.

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Dedicated to

FAMILY

Acknowledgement

Though my name appears on the cover of this dissertation, a great many people have contributed to its production. I owe my gratitude to all those people who have made this dissertation possible and because of whom my graduate experience has been one that I will cherish forever.

My deepest gratitude is to my advisor, **Dr. S.P. Nigam**. I have been amazingly fortunate to have an advisor who gave me the freedom to explore on my own and at the same time the guidance to recover when my steps faltered. My mentor taught me how to question thoughts and express ideas. His patience and support helped me overcome many crisis situations and finish this dissertation. I express my thanks to my friends for their support and guidance.

At **Thapar University**, We learned many things like the research work is mainly aimed at enabling the student to apply their theoretical knowledge to practical as "The theory is to know how and practical is to do how" and to appreciate the limitation of knowledge gained in the class room to practical situation and to appreciate the importance of discipline, punctuality, sense of responsibility, money, value of time, dignity of labor.

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Abstract

Rotating machinery plays an important role in industrial applications. When these machines recently are getting more complicated, fault diagnosis techniques have become more and more significant. Gears are used extensively in drive systems. Effective gear fault detection is crucial to ensure safety. In addition, tremendous economic benefits can result from condition based maintenance practices, for which gear fault detection plays an important role.

Over the past 25 years, much research has been devoted to the development of Health and Usage Monitoring systems for gearbox and drivetrain components. In order to keep the machine performing at its best, the principal tools for the diagnosis of rotating machinery problems is the vibration analysis & noise analysis, which can be used to extract the fault features and then identify the fault patterns.

In the majority of cases the source of noise and vibrations is transmission errors introduced during manufacture. These errors can e.g. be geometry inaccuracies and eccentricities which both results in impact noises. Other sources of impact noises can be gear rattle. Gear rattle is caused by a combination of backlash and unloaded gears. Friction, wear, crack and pitting due to gear fatigue are also sources of noise.

Most modern techniques for gear diagnostics are based on the vibration signal picked up by an accelerometer from the gearbox casing. The vibration signal is normally filtered by time synchronous averaging (TSA) and analysed in the frequency domain with methods such as the Wavelet Transform (WT) or the FFT. A similar approach is to use a microphone instead of an accelerometer and record the noise from the gearbox. This method was used to detect faults in industrial gears.

The expected noise spectrum from a gearbox would contain the gear meshing frequencies and integer multiples of it. A different method for measuring the noise from the gearbox with seeded fault is used. Sound level meter is used to measure the SPL of gearbox with different imposed faults (cracks and decrease in tooth height) in the tooth of gear. The results have been compared with the sound pressure of normal gear to extract the fault information.

Key words: Noise Analysis, Sound Pressure Level.

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Nomenclature

AE	≡	Acoustic Emission
AWT	≡	Analytic Wavelet Transform
C1	≡	Crack of depth 1.475mm
C2	≡	Crack of depth 2.395mm
C3	≡	Crack of depth 3.391mm
CF	≡	Crest Factor
dB (A)	≡	A-weighted Decibel
dB	≡	Decibel
EDM	≡	Electro Discharge Machining
EMD	≡	Empirical Mode Decomposition
FFT	≡	Fast Fourier Transformation
Hz	≡	Hertz
IMF	≡	Intrinsic Mode Function
L_{10}	≡	10 Percentile exceeded Sound Level
L_{eq}	≡	Equivalent Continuous Sound Level
MCSA	≡	Motor Current Signature Analysis
PTS	≡	Permanent Threshold Shift
SL	≡	Sound Level
SPL	≡	Sound Pressure Level
TSA	≡	Time Synchronizing Average
TTS	≡	Temporary Threshold Shift
WEDM	≡	Wire Electro Discharge Machining

Chapter 1

Introduction

1.1 Introduction

There is not big difference in sound and noise, the unwanted sound is called noise. In present world, noise pollution is the biggest growing problem. Hearing of loud sound for a long period of time causes permanent loss of hearing and other serious health problems.

There are three issues of noise

- Industrial noise
- Traffic noise
- Community noise

Out of these above issues, traffic noise encounters more noise than the others. It is generated due to large road traffic by vehicles. It produces approximate 70% of noise.

1.2 Sound

It is the tangible disruption in a medium which is caught by ear. Mass and density are very necessary for the movement of sound wave in a medium. It can't go through the vacuum. The fluctuation in the steady atm. pressure (atm. pressure is 10^5 Pa) causes the generation of sound in the air.

1.3 Decibel

Sound is generally expressed as a sound pressure level (SPL). Decibel (dB) is the unit of sound. The human ear can detect the variation in 1 dB of SPL. The human ear detects the sound as doubled when there is an increase in 10 dB in SPL and similarly it detects the sound as half if there is drop of 10 dB in SPL.

1.3.1 Decibel Scale

The logarithmic scale which is used to measure the SPL is called decibel scale. The value of 0 dB which is threshold of hearing is equal to SPL of 20µPa. The following Fig. 1.1 shows the decibel scale.

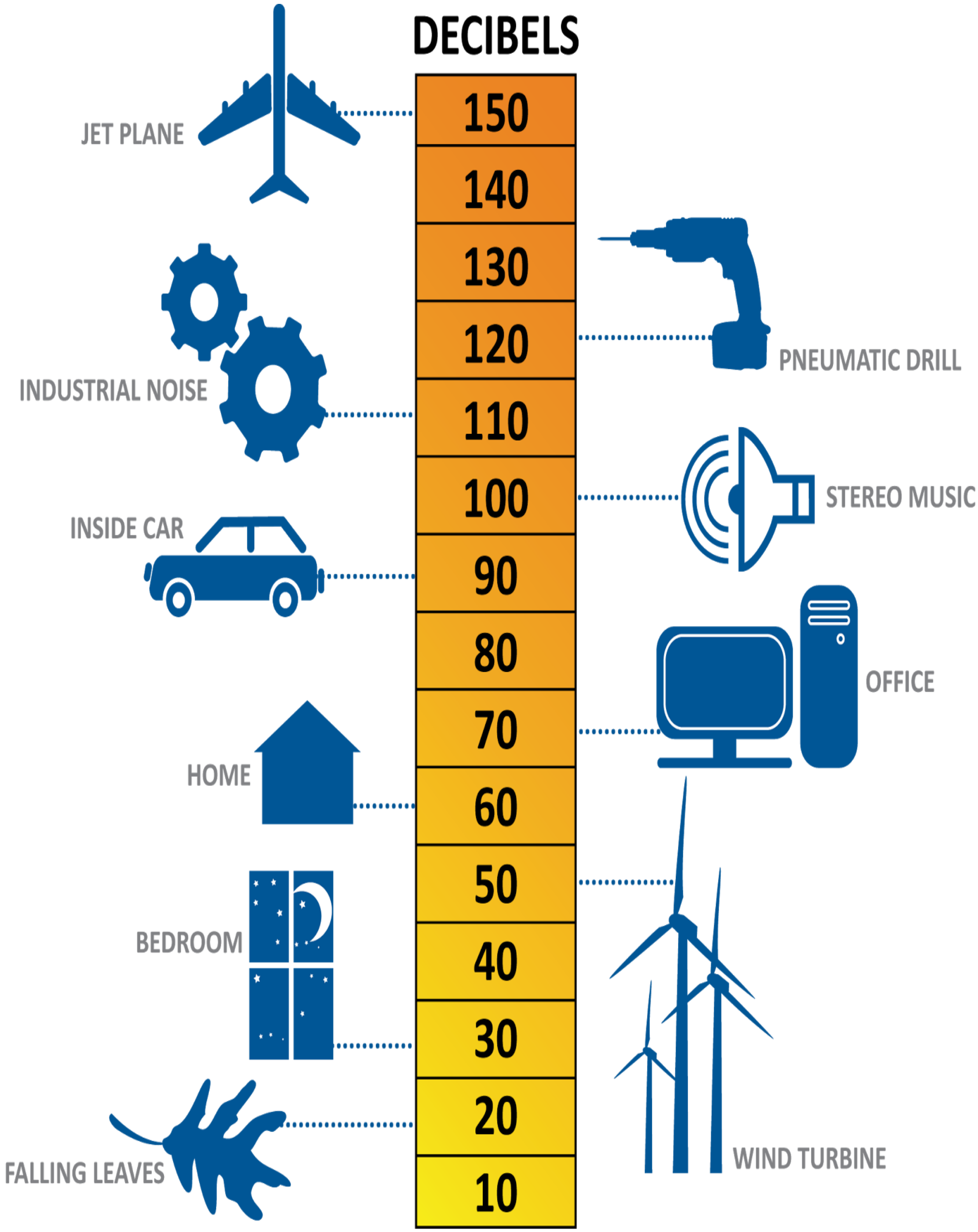

Figure 1.1: Decibel scale [W.1]

1.4 Noise Sources

- Point source
- Line source

1.4.1 Point Sources

A noise source said to be a point source, if its geometry is small with respect to distance to the receiver and it give off same amount of energy in all directions. The industries, aircrafts and road vehicles are represented as point sources. The SPL decreases 6 dB as the gap between receiver and point source is doubled [Harris, 1991].

1.4.2 Line Source

A line source said to be a ceaseless emission, such as pipes carrying fluids, or may be massive number of point sources separated by small distance so that their dissipation considered as ceaseless line. The SPL decreases 3 dB, as the gap of receiver to a line source is doubled.

1.5 Acoustic Notions

1.5.1 Frequency

The sound wave frequency is defined as the number of events repeated per second. It is measured in Hertz (Hz). Sound occurs in a broad frequency span. Normal hearing sound range for all peoples lies between 20 Hz to 20 kHz. Sound below 20 Hz is called infrasound and sound above 20 kHz is called ultrasound.

1.5.2 Amplitude

The amplitude presents the power of sound wave. It is the magnitude of the difference between two adjacent peaks of frequency.

1.5.3 A- Weighting Filter

Weighting filter is used to take sound pressure level measurements. Measurements with this filter are called A-weight of SPL measurements. Because it obeys to the international standard of A-weighting curves, it is called A-weighted filter. It highlights the frequencies between 1 to 6.3 kHz, in an endeavor to duplicate the relative response of the human ear.

1.5.4 Sound Pressure Level

It is the logarithmic ratio of sound pressure amplitude of the measured sound to the reference pressure and is expressed by

$$L_p = 10\log_{10} (p/p_o)^2 \qquad (1.1)$$

Where Reference pressure (P_0) = 20µPa

P is the sound pressure of the measured sound

1.5.5 Equivalent Sound Pressure Level

The sound from noise sources usually vary over a given period of time. The equivalent SPL (L_{eq}) is the average value of SPL for a period of time. It can be expressed as

$$L_{eq} = 10\log_{10}\left[\frac{1}{T}\int_0^T \left(\frac{P_{A(t)}}{p_0}\right)^2 dt\right] \qquad (1.2)$$

Where $P_{A(t)}$ represents A-weighted sound pressure

T denotes the total time

P_0 denotes reference pressure.

1.5.6 Sound Intensity Level

Sound intensity is created by the sound source at some distance from the source, when sound is produced by it. It is measured by amount of power over a certain area at that distance. Sound intensity level (L_I) in decibels is logarithmic ratio of sound intensity (I) to the reference sound intensity (I_o) and is expressed as

$$L_I = 10\log_{10} (I/I_o) \qquad (1.3)$$

Where $I_o = 10^{-12}$ W/m^2

1.5.7 Octave Band Filters

When complex sound is studied, for more detail the frequency span of 20 Hz to 20 kHz is divided into bands. This is done with the help of sound pressure level measuring instrument. These bands generally have a bandwidth of 1/1 Octave or 1/3 Octave. An octave band is a frequency band where the highest frequency is twice the lowest frequency. Any frequencies below and above these limits are rejected.

1.5.8 Background Noise

All the other noises except the sound being measured is included in background noise. It may include some conversation noise, power drive noise or noise produced due to some other instrument. There should be more than 10 dB (A) difference between background noise and the sound being measured if the difference is less than 10 dB then background noise should be corrected.

1.5.9 Correction of Background Noise

When there is less than 10 dB (A) difference between the background noise and the sound being measured then the background noise is corrected. To correct the background noise, following steps are used.

- Measure the combined sound pressure level of both source noise and background noise.
- Measure the background noise by switching off the source noise.
- Calculate the difference between these two variables.
- Read the correct value from the Fig. 1.2, if it exceeds 3 dB (A).
- Subtract the correct value from the value taken out by first step.

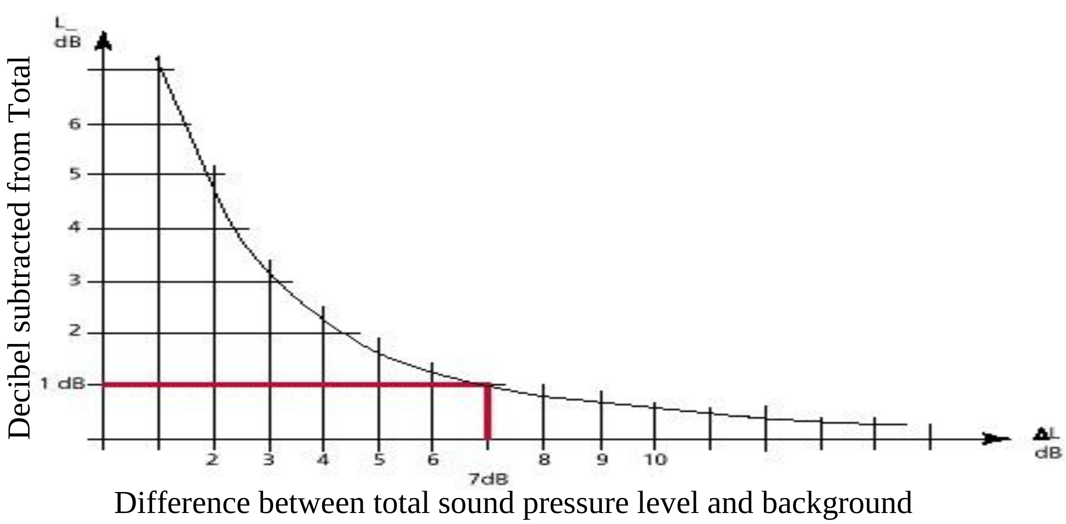

Figure 1.2: Background noise correction [W.2]

For example the total combined SPL of sound source and background is 60 dB (A) and background noise is measured as 53 dB (A). Subtract the background from the combined SPL, the difference is 7 dB (A). From the Fig. 1.2, 1 dB (A) is subtracted from the combined SPL. The corrected SPL is 59 dB (A).

1.6 Applications of Noise

1.6.1 Medical Purpose

To diagnose the health of human, noise of heart beat is used.

1.6.2 Masking Effect

Sometimes it is mandatory that nobody should hear the conversation between two persons. For example doctors wants everything remain confidential about their patients so they put some noisy exhaust fan, which produces noise outside the room. This is called masking effect.

1.7 Need of Diagnostic Analysis

Rotating machinery plays an important role in industrial applications. When these machines recently are getting more complicated, fault diagnosis techniques have become more and more significant.

Gears are used extensively in rotorcraft drive systems. Effective gear fault detection is crucial to ensure safety. In addition, tremendous economic benefits can result from condition based maintenance practices, for which gear fault detection plays an important role. In order to keep the machine performing at its best, the principal tools for the diagnosis of rotating machinery problems is the vibration analysis & noise analysis, which can be used to extract the fault features and then identify the fault patterns.

The expected noise spectrum from a gearbox should contain the gear meshing frequencies and integer multiples of it. There is also common with harmonics and sidebands due to gear eccentricities and geometric errors.

1.8 Fault Detection in a Rotating Machine through Monitoring of Vibration Signal

1.8.1 Permanent Monitoring

- Does not give long term warning
- Damage to machine is of prime concern
- Monitored parameter such as relative shaft vibration & absolute casing vibration

1.8.2 Intermittent Monitoring

- Early warning of incipient fault
- Loss of production is of prime concern
- Monitored parameter such as velocity/acceleration of gears/casing

1.8.3 Run to Breakdown Maintenance

In this method no action takes place until the machine run out completely. Reactive maintenance is depends on the size of job and the number of staff available for service. This method is used when shutdown period is not important. Due to necessity of large number of extra parts and high loss in production rate, it became most costly method.

1.8.4 Protective or Preventive Maintenance

In this method the machine is repaired / maintained at regular interval of time. In this method replacement of parts are done after some interval of time rather than checking their condition. This method has double cost because of replacement of parts and shutdown period. This method is used to prevent the unfortunate failure of machine.

1.8.5 Predictive Maintenance

It is also called condition based maintenance. It is used widely in all the online system maintenance. Its aim is to remove the shutdown period by inspecting the condition of machine, detecting the defects and implementing a correct solution for it to avoid the run out of machine.

1.9 Fault Detection

Overall vibration level measurements in a frequency range 10 Hz to 1 kHz or 10 kHz at certain typical positions. This is compared with available vibration criterion charts.

1.9.1 Trend Analysis

Vibration level vs. Time plot may indicate deteriorating machine condition. Prediction of shut down for repair is possible.

1.9.2 Frequency Analysis

A stable vibration spectrum changes when parts wear out or faults are developed. A component in a vibration spectrum corresponds to a specific source in a machine. Each fault occurs at a particular frequency. Therefore frequency domain analysis is useful for fault detection.

Reasons for Using Frequency Analysis

Frequency analysis is an aid to understanding the vibratory excitation, structural responses, and radiated noise arising from meshing gear pair because

- If the driving gear shaft is rotating at a constant speed and the force transmitted by each gear mesh is nominally constant, then the vibratory excitation arising from each meshing gear pair and the resulting structural responses and radiated noise all are periodic functions of time. Many harmonics can be present in the excitation and response arising from each meshing gear pair. Nevertheless, as the locations and relative and absolute strengths of these harmonics can be very useful in diagnosing the causes of gear noise and in implementing its reduction.
- The above-mentioned attenuating effects of simultaneous multiple tooth contact are most easily described and understood in the frequency domain.
- Structural resonance effects are most easily described and understood in the frequency domain.

1.10 Common Faults with Rotating Shafts

- Unbalance
- Misalignment

1.11 Techniques for Fault Detection

1.11.1 Visual Inspection

This is the most simplest and cost effective technique for fault detection. It consists of visual inspection of machinery or its components using some form of optical based or the unaided eye. The computer system is used for data information logging and analysis. The main advantage of the method is that it is simple and gives the direct information about the defect. It requires very low level skill. If trend is required then data information is more important for which various methods like use of data collectors, still photography, sound tape, video recording, surface printing etc. are used [Gupta].

Some of low cost aids for fault detection are bore scopes, fiberscopes, stroboscopes, dye penetrants etc. bore scopes and fiberscopes are used for complex machinery, which are normally not accessible. Other common techniques for fault detection are dye penetration test for surface cracks, corrosion testing, video imaging, thermal imaging etc.

1.11.2 Infrared Thermography

This technique is practically applied in all engineering systems for fault detection by an unexpected increase or decrease in temperature. In fact temperature changes usually occur much before physical damage to the system. Some examples are overheating of motor bearing and electrical connections. It includes the measurements of temperature of one or few point selected on the system. In the advanced technique it include overall complete thermal imagining of the system and its components.

This thermal imagining is used for detecting faults in overhead power lines along railway tracks. Overheating failure usually occurs in clamped and friction held fittings.

1.11.3 Surface and Internal Defect Detection

The various NDT techniques which can be used to detect surface defects are:

- Eddy current testing
- Electrical resistance testing
- Flux leakage testing
- Magnetic particle testing
- Dye penetrating testing
- Radiographic testing
- Resonance testing

- Thermo graphic testing
- Ultrasonic testing
- Visual examination

1.11.4 Vibration Based Methods

This technique is widely used for fault detection. Complex rotors may have any one of the following faults, the analysis of such these faults may lead to the fault detection.

- Unbalance of rotating parts
- Eccentric component
- Misalignment of coupling
- Bent shafts
- Loose fitting of components
- Damaged gears
- Bad drive belts
- Resonance
- Cracks in shafts
- Torque vibrations
- Electromagnetic, aerodynamic, hydraulic forces

Spectral analysis is performed on vibration signatures to identify the faults. The various faults have frequencies associated with them. Typical plot give the give the frequencies associated with various common faults of shaft rotating speed.

1.11.5 Noise/Acoustic based Method

The spectral and cepstral analysis of noise signals also gives useful information for fault detection. Vibration and noise are interrelated with each other and are combined called as vibroacoustics.

It deals with the dynamic phenomena of the system, such as vibration, noises, pulsation of the working medium in machines, acoustical emission occurring in the frequency range of 0 to 1 MHz or more.

Acoustic emission is a specific technique for fault detection. It is an elastic wave due to a violent release of energy accumulated in the material by the propagating micro damage in the material. During the release of elastic energy, a sequence of decaying waves is emitted. The total number of impulse rate is measured. These measurements are used as the tool for fault detection and internal defects in structure.

1.11.6 Oil and Wear Debris Analysis

Lubricant is used widely in practically all machines. Therefore a periodical examination of wear particles in the lubricant helps in diagnosing faults in the machines. The analysis of wear particles, known more popularly as 'wear debris analysis' essentially comprises of estimating their quantity, size, morphology and composition. Methods of wear debris analysis are as follows:

- Optical method
- Filter blockage
- Ferromagnetic attraction
- Gravimetric
- Ultrasound

1.12 Types of Gears

Meshing gears pairs can be divided into two categories: those that have parallel shafts, and those whose shafts are not parallel. Spur, single-helical, and double-helical, or herringbone, gears are gear types having parallel shafts, whereas straight-bevel, spiral-bevel, hypoid and worm gears are common gear types having nonparallel shafts. Often the term pinion is used to describe the smaller gear of a meshing gear pair. When the smaller gear is described as the pinion, the larger gear sometimes is referred to as the gear [Harris, 1991].

1.12.1 Parallel-Axis Gears

Spur Gears

Spur gears are gears having teeth arranged in a circle with the shaft at the center of the circle and the tooth surfaces parallel to the shaft. When ordinary spur gears are rotating, the number of mating tooth pairs simultaneously in contact alternates between one and two as shown in Fig. 1.3.

Figure 1.3: Spur gears [W.3]

The (transverse) contact ratio Q_t of a meshing pair of spur gears is the time average number of tooth pairs simultaneously in contact. For ordinary spur gears, the contact ratio between 1and2. In the case of high-contact-ratio spur gears, which have long thin teeth, the number of mating tooth pairs simultaneously in contact alternates between two and three; thus, in this case the contact ratio is between 2and3. For given application, high contact ratio spur gears typically generates less noise than ordinary spur gears.

Single-Helical Gears

In contrast to spur gears, helical gears have teeth that are at a specified angle to their shafts. That angle is called the helix angle. In any plane cut normal to the gears shafts, the geometrical meshing action of a pair of helical gears is identical to that of gears. Such a plane cut normal to the shaft of a meshing pair of parallel-axis gears is called the transverse plane.

There are two contact ratio associated with a meshing pair of helical gears. The transverse contact ratio Q_t is the contact ratio defined as for spur gears, based on the geometrical tooth meshing action in the transverse plane. The axial contact ratio Q_a is the time average number of teeth simultaneously in contact along a straight line taken parallel to the gear shafts in the imaginary geometrical surface where tooth contact take place. The axial contact ratio of a meshing pair of a helical gear is positive number that can range from less than unity to about 10, or even larger. For a given application, helical gears typically generate much less noise than spur gears. Fig. 1.4 represents the single helical gear pair.

Figure 1.4: Single helical gears [W.4]

The force transmitted by a pair of teeth acts, essentially, normally to the tooth surfaces. Because of the helix angle, the total force transmitted by all the teeth of a single-helical gear has a component that acts in the direction of the gear axis. In the case of single-helical gears, this axial force must be counteracted by a thrust bearing or a thrust collar.

Double-Helical Gears

Each tooth of double-helical, or herringbone, gears is divided in the middle into two parts having equal helix angles in absolute value but of opposite sign. The axial thrust produced by each of the two halves of the gear is counteracted by the other half; thus thrust bearing or thrust collar are not needed. When the two halves of each tooth are separated by a gap, the gear is referred to as double-helical, whereas when each tooth is continuous so that the two halves meet in the center at a point, the gear is referred as herringbone. Fig. 1.5 represents the double helical gear pair.

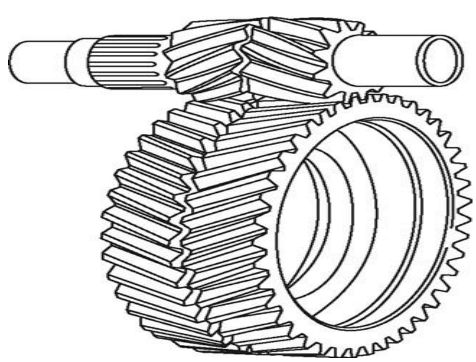

Figure 1.5: Double helical gears [W.5]

1.12.2 Nonparallel-Axis Gears

Bevel Gears

Bevel gears transmit rotational motion between nonparallel shafts that would intersect if extended. Fig. 1.6 represents the bevel gear pair.

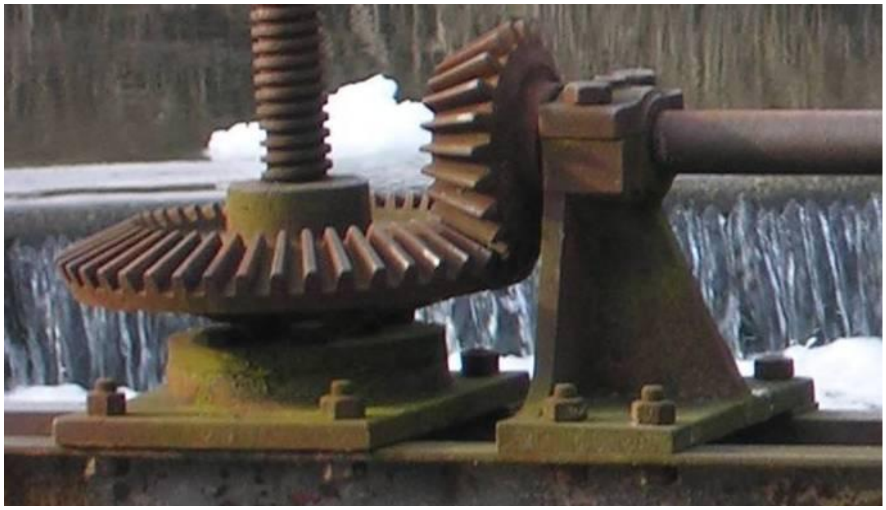

Figure 1.6: Bevel gears [W.6]

A straight bevel gear is the bevel gear counterpart to a parallel axis spur gear. A spiral bevel gear is the bevel gear counterpart to parallel-axis single helical gears. For a given application, spiral bevel gears typically generate less noise than straight bevel gears.

Hypoid gears

Hypoid gears transmit rotational motion between nonparallel, nonintersecting shafts.

Worm Gears

A worm gear pair has the distinguish feature that one gear of the pair, the worm, has one or more teeth wrapped completely around the gear axis and thus has the appearance of the threaded screw without taper. The other gear of the pair is called worm wheel.

1.13 Principal Sources of Various Excitations

1.13.1 Tooth Meshing Harmonics

Elastic deformation of the teeth plus deviation from equispaced, perfect involute surfaces of the unloaded mean running surfaces of the teeth combine to produce the loaded mean running surface deviation, which is the principal source of the tooth meshing harmonics.

1.13.2 Transmission Error

Transmission error is the main cause of agitation in structure for gear noise and vibration. It is defined as the difference of angular location of pinion gear to the real location of output gear, when the pair is running at steady speed.

1.13.3 Tooth Profile and Lead Modifications

As a meshing pair of gears rotates, unmodified involute teeth tend to interfere with one another at the point of contact initiation because of the elastic deformation of the teeth already in contact. Such interference causes very abrupt loading transfer to the teeth coming into contact, and excessive gear noise is a result. Moreover, tooth spacing errors can cause even more interferences at contact initiation. In spur and straight bevel gears, profile tip relief or a combination of tip and root relief is used to prevent interference. If there are no matching errors, journal errors, or alignment errors, it is possible, in principle, to exactly compensate for elastic deformations by using running surface modifications for one prescribed force transmitted by the gear mesh. For moderately highly loaded gears, the maximum amount of modification is about 25 micrometers or slightly more.

1.13.4 Machining Errors

The deviation of the mean running surface of the teeth on a gear element from a perfect involute surface can be different from the intended modification of a surface. This difference between the actual mean surface and the intended one will contribute to the tooth meshing harmonics.

1.13.5 Tooth Wear

Gear teeth can wear, especially when relatively soft steel and high loadings are used. Such wear modifies the mean running surface of the teeth. Significant amount of material can be removed by wear, which can result in significant changes in tooth meshing harmonics levels.

1.13.6 Bearing Misalignment

If the bearing at the two ends of the pinion and gear shaft are not properly located, the pinion and gear shafts will not be parallel. The projection of any such misalignment onto the plane of contact is particularly important. The component of such misalignment in the plane of contact is equivalent to the error in pinion or gear tooth helix angle. The amount of this projected misalignment component with in the tooth contact region should be held to a small fraction of maximum expected elastic deformation of the teeth. In the case of spur or helical gears, if the misalignment results in more than this amount, the teeth at the one end of the gear axis will carry more loading than at the other end of the axis, and the amount of tip modification will not be correct at either end, which, normally, will result in an increase in the tooth meshing harmonic excitation and response level. If the amount of this misalignment component is more than about twice the expected maximum tooth deformation, complete loss of tooth contact can occur at one end of the gear axis, which in the case of helical gears will result in operating value of the contact ratio Q_a being less than the design value. This reduction in axial contact ratio can lead to an additional increase in tooth meshing and rotational harmonics levels as a result of a decrease in attenuating effects of simultaneous multiple tooth contact, as described earlier.

1.13.7 Pinion Body Deformation

In wide face helical gears pinion body torsional, bending and shear deformations can be a significant fraction of tooth elastic deformation. The effects on the tooth meshing harmonics of such pinion body deformations are similar to those from bearing misalignment. Local helix angle correction to pinion teeth generally are made to compensate for pinion body deformations for one loading conditions.

1.13.8 Entrapped Air and Oil

For high speed spur and helical gear, air entrapped in mesh can reach sonic or near sonic velocities when being forced out between mating teeth. When the escaping air reaches sonic velocities, it can be dominant source of gear noise. Increasing backlash and tooth root clearance can enhance the escape of entrapped oil.

1.13.9 Friction

The vibratory excitation caused by sliding friction between mating teeth generally is believed to be small in comparison with that caused by the above-mentioned contribution to the static transmission error tooth meshing harmonics. Fig. 1.7 shows the friction forces.

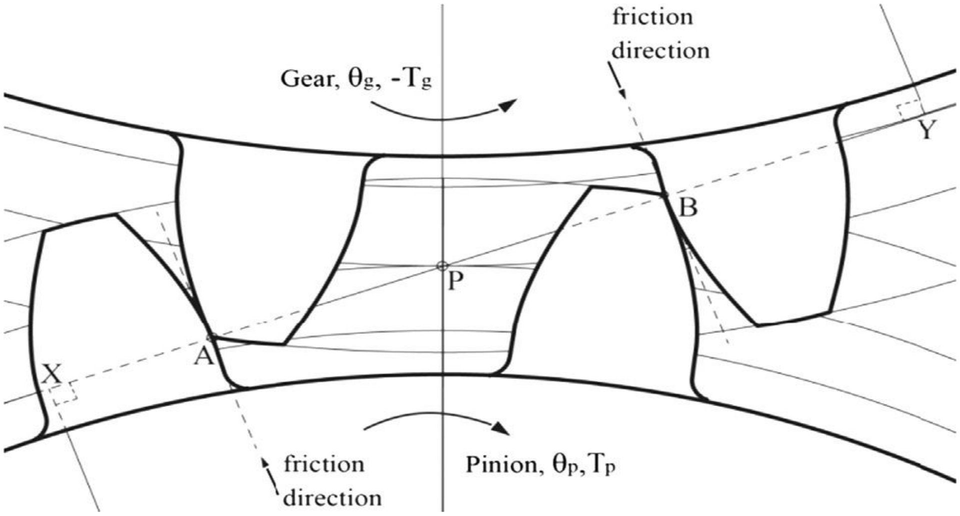

Figure 1.7: Friction forces in meshed teeth [Kumar, 2013]

1.13.10 Pitting

It is a surface defect which occurs on the gear tooth due to greater contact stress than the fatigue tolerance limit. It is shown in the following Fig. 1.8. After some period of time with repeating load, small dents or strike marks occurred on the tooth surface. It tends to grow at an accelerating rate because the un-defected areas which left must support the extra load previously carried by the damaged area.

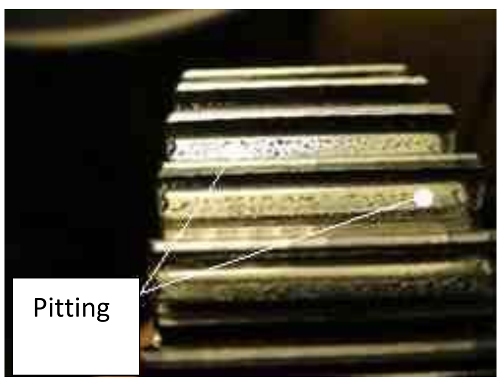

Figure 1.8: Tooth pitting [Al-Arbi, 2012]

1.14 Thesis Organization

The thesis is described in six chapters, the detail of each chapter are given below:

Chapter-2

In this chapter the reviews of literature on fault diagnosis of gear box with various methods have been written. Many techniques have been discussed with results to understand the concept of fault analysis and relevant methods have been discussed to understand the present work in the upcoming chapters.

Chapter-3

This chapter deals with the problem formulation.

Chapter-4

This chapter deals with experimentation investigation done on gears to diagnose fault. For the study of fault diagnosis some artificial defects have been imposed in gears of gear box and sound pressure level of gear box is measured.

Chapter-5

In this chapter results are discussed briefly after the experimentation.

Chapter-6

In this chapter the conclusion is drawn from the results and future scope work is discussed.

Chapter 2

Literature Review

Vibration and noise are interrelated with each other, some vibrations are good for human being for e.g. heartbeat, breathing etc. but most of vibrations are hazardous for human being and machinery. So it is mandatory to avoid vibrations and noise, if the vibration is not reduced then they may cause of failure of machinery and breakdown of production line in an industry.

To avoid this failure due to vibration the diagnostic analysis of gears is done. For the fault detection the knowledge of various sources of vibration and noise in a machine/gear box and how to reduce them, should be required. To understand the phenomena of vibration and noise researches has been carried out. The following are the gist of various researches.

Oswald et al. [1994] conducted an experiment in NASA on gearbox to contrast the noise emitted by spur & helical gears. Using acoustic emission, sound power was calculated from the gearbox top. From the results it was concluded that the noise decreases as the face contact ratio increases. The spur gear with non-involute profile encounters three-four dB noise more than the gear with involute profile and it also examined that the double helical gear encounter four dB more noise than single type gears.

Capdessus et al. [2000] suggested a cyclostationary processes for fault analysis of gear. Synchronized averaging method also derived from cyclostationary first order equation and from cyclostationary's second order equation, the function of spectral correlation is derived. From the results it was concluded that both are very useful for incipient fault detection in gears.

Baydar and Ball [2001] conducted an experiment to check the effectiveness of acoustic emission to sense the defects in gearbox on the basis of pseudo winger-ville distribution. They took three cases of imposed faults, broken tooth, tooth root crack & wear. From the results it was concluded that acoustic emission is more suitable for prediction of faults than the vibration signal method.

Wang et al. [2001] conducted an experiment to check the susceptibility and toughness of phase-amplitude demodulation & kurtosis-wavelet transform methods. They took four cases (healthy, cracked, filed & chipped) defects imposed in gears. The vibration data obtained from the gears box housing is decomposed into signal mean, overall remnant & dominant gear mesh frequency remnant. From the results it was concluded that the beta kurtosis is very good for time domain fault analysis & phase modulation is very responsive to defects.

Dempsey et al. [2002] conducted an experiment for damage indication in spiral bevel gears. They use both oil debris analysis data & vibration analysis data collectively to predict surface damage like pitting on gears. From the results it was concluded that combination of vibration and oil debris analysis technique improves the pitting damage detection on spiral bevel gears.

Zheng et al. [2002] suggested the concept of time average wavelet spectrum to inspect the defects. It gives fine scale resolution for fault extraction. They used two methods spectrum comparison & feature energy using time average wavelet spectrum to inspect the defects. From the results it was concluded that time average wavelet spectrum is suitable for detection of defects.

Baydar and Ball [2003] conducted an experiment to check the effectiveness of acoustic emission over the vibration signal method to sense the defects in the gear box on the basis of wavelet transfer. They took two cases of seeded defects, broken tooth & tooth with root crack for the experiment. From the results it was concluded that acoustic emission is more suitable than the vibration method technique.

Loutridis [2004] suggested a theoretical model of gear pair with imposed crack defect. EMD was implied on the vibration signal, which breaks up it in the form of oscillatory function. Which also known as IMF (intrinsic mode function). From the results it was concluded that the model energy was related to the crack propagation in the gear which can be used for prediction of system run out.

Tan et al. [2005] conducted an experiment on two spur gear pair to check the effectiveness of acoustic emission for fault detection in gears. They took four cases of seeded faults in gears, misalignment, broken tooth, scuffing & worn tooth for the experiments. From the results it was concluded that the acoustic emission techniques used to diagnose the fault detection from non-rotating parts of gear box is still a question on its ability.

Mba and Tan [2005] conducted an experiment to investigate the effectiveness & ability to use in application of acoustic emission to detect the imposed defects and it is compared with the vibration method. From the results it was shown that the acoustic emission occurs due to change in temperature & lubricant film thickness but in case of isothermal conditions, for same oil film thickness, there was no effect on acoustic emission by load. This shows that source of acoustic emission is assign to contact roughness.

Olsson and stolyarchuk [2005] constructed a dynamic model of the gear box & gear vibrations on the force level are obtained from it. These vibrations are used to forecast to the sound recordings with a statistical vibration diagnostic parameter known as crest factor. The result contains the most important mesh frequencies which contains the amplitude increments throughout a spinning period. The CF is able to make the normalized from a low pass filtered noise spectrum that can be useful for the fault monitoring of the gear box.

Mohanty and Kar [2006] conducted an experiment on multi-stage gear box to detect the faults using motor current signature analysis over the old vibration measurement technique. DWT is applied to break up the current signal & fast forrier transformation analysis was done on it to detect the high frequency sidebands. From the results it was concluded that the motor current signature analysis using with DWT is best method on the behalf of old vibration technique.

Blunt and Keller [2006] proposed two new methods of detecting a fatigue crack in a plant carrier of an epicyclic transmission are developed. Vibration data of various number of US-Army UH-60A black hawk helicopter main transmission is used for testing vibration measurements are done of faulted and un-faulted transmission over a span of torque level in controlled conditions like as test cell and on aircraft are used. Both are based on changes to the modification of the fundamental gear mesh vibration brought about by crack. The results showed that the new methods are reliable and detect the crack in these conditions.

Tan et al. [2007] investigated an experiment on spur gear pair for prognosis of gear life. Acoustic emission [AE], vibration & spectrometric oil techniques were used to detect the natural pitting. From the results it was concluded that acoustic emission technique based on rms. value is more efficient than the other methods.

Tuma [2009] reviewed some experimental and methods implemented to gearboxes. According to the author, the noise should be controlled at the source rather than to work on the isolating casing. The author conducted an experiment to reduce the noise of truck transmission unit. For the reduction the modifications of gear design were done. From the results it was concluded that the modification in design is better than using noise isolation housing for reduction of noise. The vibrations & noise are measured by using frequency domain & time domain. By measuring the transmission error, the design accuracy of the gear design can be determined.

Eftekharnejad and Mba [2009] conducted an experiment on two helical gear pairs to inspect the effectiveness of acoustic emission to detect the imposed defects. From the results it was concluded that there was large short acoustic emission rupture occurred rather than the continuous acoustic emission level and it was noted that the acoustic emission rupture occurs at the same tooth where defect was imposed.

Hamzah and Mba [2009] conducted an experiment to inspect the effect of speed and load on acoustic emission generation in spur and helical gears. They presented the numerical model which shows that any change in thickness of lubricant is directly proportional to the load, due to which change in rms. of acoustic emission occurs. From the results it was concluded that, for the effective measurement of fault detection, thermal conditions should be in control.

Kostopoulos et al. [2009] had implemented Acoustic emission [AE] and vibration measurements methods for condition monitoring of single stage gear box. Both methods are used for condition monitoring at specified parameters and it is observed that the AE is the best over vibration measurement and gives the earlier fault detection than the vibration measurements and it is very sensitive to micro scale fault and is non-directional. It is also observed that oil temperature has the considerable effect on the vibration and the recorded AE data.

Combet and Gelman [2009] proposed a methodology for the enhancement of small transients in gears vibration signal in order to detect the local tooth faults like pitting at early stage of damage. They applied the winer (de-noising) filter based on spectral kurtosis. It is applied to gear residual signal which is obtained after mesh harmonics removal in TSA. Advantages of this methodology is reduced interference and improve signal to noise ratio S/N.

Metwalley et al. [2010] conducted an experiment on single stage spur gear box for diagnostic analysis of vehicle gear box. An artificial crack of different depth had been induced with the help of wire-EDM at the root of tooth of spur gear. Advanced signal processing technique had been used for noise measurements. Form the results it was concluded that acoustic emission is very sensitive technique and gives earlier fault detection information than the other classical methods.

Mark et al. [2010] suggested a simple algorithm ALR or its medium component MLR for very early frequency domain detection of damage on teeth of meshing gear. This algorithm utilizes rotational harmonic amplitudes of transducer responses caused by class 3 gear manufacturing errors and potential tooth damage. Ratio of after potential damage and before potential damage rotational harmonic amplitudes are used, which eliminate effects of rotational harmonic amplitude changes caused by transducer of structural path between tooth meshing location and transducer output and minimize the effect of amplitude changes caused by gear meshing due to contribution of manufacturing errors.

Allam et al. [2011] conducted an experiment on single stage helical gearbox and sound pressure level (SPL) was measured. A simple test rig was used with essential instruments to measure SPL. The objective of the author was to modify those parts of gearbox which contributes to the noise without making any patterns. From the results it was concluded that for any methodology used to reduce the noise of gearbox, gear meshing frequencies & shaft frequencies should be reviewed as resonance frequencies.

Karpuschewski et al. [2011] proposed the testing of finished gears in industrial production with non-destructive, ecological and micro-magnetic method called barkhausen noise analysis. It is the method to detect the thermal damage on the work piece. Its strength is influenced by the elastic stresses and the micro structure. So change in residual stress and material structure are measurable and they are indicator for the improvement of grinding process. To control the physical properties is most important for quality assurance. Unfortunately this is offset by the cost of calibration and referencing.

Lewicki et al. [2011] conducted an experiment to show the ability to sense the planetary gear defects in helicopter's rotor transmission using vibration data. Vibration data from the planetary gear arrangement for healthy transmission and for transmission with various imposed artificial defects were extracted. Defect sense algorithm and vibration data were used for defect detection. Cracks and spalls imposed with the help of EDM. For the planet gear, crack and spall were detected using vibration separation technique. For ring gear accelerometers were used to get the clear information about the crack. For sun gear, vibration separation technique was not suitable.

Shao and Chen [2011] suggested an analytical model to forecast the change in the gear mesh stiffness due to crack in root of tooth of the gear. The effect of crack on the dynamics of gear had been stimulated & due to this change in rms. and kurtosis was inspected. From the results it was concluded that, as the crack increases the rms. & kurtosis also increases. From the frequency spectrum it is noticed that the sidebands generated due to crack were more sensitive than the mesh frequency.

Yan et al. [2011] had proposed a fault analysis method by using collectively two techniques bi-spectrum analysis & ANN due to the complexity of marine propulsion system. First the bi-spectrum analysis was implemented to inspect the defects. It gives frequency vs. amplitude plot. After that BPNN & RBFNN is applied to detect the state of gear box. From the results it has been concluded that the faults were extracted by using bi-spectrum successfully & ANN technique sure high detection accuracy.

Ognjanovic and Kostic [2012] studied that the gear box casing has important role in the emission of noise due to striking of gear teeth. An experiment was conducted on the classical gear train to examine the effect of energy absorbed by casing of gearbox generated due to the striking of addendum teeth. This information can be very helpful to fault diagnosis of gearbox. From the theory, numerical evaluation & test analysis it was found that the generated noise from the gearbox is caused by the absorption of disrupting energy in gearbox casing throughout the running operation. This absorbed energy causes the vibrations in the gearbox.

Jena et al. [2013] had the main focus on the burst in vibration signal captured from the experiment. To detect the defect, enveloping technique is quite helpful but is failed to measure the angle between two faulty teeth. To filter the noise and to measure the angle, a signal processing technique is used. In this technique the signal is filtered by un-decimated wavelet transform (UWT). AWT (analytic wavelets transform) is implemented on the signal and time frequency representation is extracted from the filtered signal, TMI graph is used to measure the angle. Kurtosis is very helpful to detect the defect but in multiple damaged teeth case it fails.

Tumalioglu and Tuc [2014] adopted AR chord's wear equation and MATLAB program to solve this modified equation to the internal gears for investigating erode in internal gears. They calculate the erode values in different conditions by using the modified equation. In addition a fatigue and wear test instrument is sketched and fabricated which is similar to the FZG closed circuit power circulation system to investigate erode in internal gears experimentally. Both results indicate that in internal gears, the maximum wear occurs in the region of tooth tip where it begins to mesh with the pinion teeth.

Jena et al. [2014] proposed a new method using an active noise cancellation for removing the noise and improve the signal to noise ratio. This technique is designed with the help of a FIR (Finite Impulse Response) based least mean square LMS adaptive filter. The acoustic signal from the healthy gear mesh has been used as reference signal in adaptive filter. The adaptive wavelet is designed from the signal pattern and used as mother wavelet in continuous wavelet transform. It provides adequate time frequency information in order to analyses the non-stationary acoustic signals.

Ahamed et al. [2014] a multiple pulse individually rescales time synchronous averaging (MIR-TSA) in conjunction with conventional TSA has been purposed. It is efficiently used for fluctuating speeds. A 2D finite element methodology based on linear elastic fracture machine principle is adopted for predicting the crack propagation path at the root of gear tooth. It is produced by WEDM. The vibration signals were recorded for 10%, 20%, 30%, 40% and 50%. Crack length.

Amarnath and Krishna [2014] conducted an experiment on two stage helical gear box to inspect the effectiveness of empirical mode decomposition method for diagnostic analysis. They implemented collectively EMD with vibration signal & acoustic emission to inspect the defects. It breaks up the signal in the form of intrinsic mode function. The vibration and acoustic emission signal segmented to get the high order statistical parameter. From the results it has been concluded that the EMD based method is better for fault analysis than using vibration and acoustic emission techniques alone.

Chapter-3

Problem Formulation

A lot of work had been done on gears in past decades due to the wide use of gears in machine from small scale industry to the large scale industry. These gears are not only the transmission elements of machine but also as source of noise and vibration in the gear box. To show the advantages of various method of fault diagnosis, few research papers have been discussed in previous chapter. To give the high fault detection accuracy in gears has become the challenge. Various method and techniques are proposed in the past few years to diagnose the fault in gears. Acoustic emission is very new technique used for fault diagnosis in gears. Very less work is done to diagnose the fault in gears using AE.

A different technique of measuring the noise for fault analysis is used in which a sound level meter is implemented to diagnose the faults of gearbox. The objective of the present work is to identify the effects of different artificial defects induced in a tooth on the sound pressure level of the gear box. Research has to be done to obtain the sound pressure level of gear with different defects. Further, present study enhances the machine availability, effectiveness and reducing the maintenance cost.

Chapter-4

Experimental Investigation

4.1 Introduction

Rotating element gears are used in every machine from small scale industry to large scale industries. Any failure in these elements can cause shutdown of machine which cause the loss in production. In these rotating elements, fatigue failure is most common cause due to which pitting and crack is propagated on the gear which causes the machine shutdown. Due to cyclic stress on the gear tooth in gear meshing, exceeds the endurance strength, cracks are generated on the surface of tooth. Continuous wearing of gear tooth in gear meshing due to friction also causes its failure. The small defects in the gears of gear box can cause of failure of machine and production loss. Therefore fault analysis becomes very important to run the machine in good condition. In order to keep the machine performing at its best, study of fault analysis of gear box has to be done. For the diagnostic analysis of the gear box to detect the faults several vibration and acoustic methods had been used by the researchers. In the vibration method, the vibration signal is directly obtained from the defected gear which is further processed and gives the information about defects. In the acoustic method noise from the defected gear box is used to find the defects. For the fault analysis of gear box, a healthy gear box of kinetic scooty has been selected and some artificial defects like crack generation and decrease in tooth height have been made with the help of wire-EDM and grinder. Noise analysis, which is one of the principal tools for the diagnosis of the rotating machinery problems, has been done on it.

4.2 Fabrication of Experimental Rig

4.2.1 Mounting Frame

A frame was constructed to mount the coupled motor with gear box. To construct the frame 1.5 inch mild steel angles of 5mm thickness were used and the frame was welded. For the base of motor 1.25 inch thick square wood plate was bolted on the mild steel frame. The Fig. 4.1 shows mounting frame with wooden base plate after welding.

Figure 4.1: Mounting frame

4.2.2 DC Motor

To run the gear box, DC motor of 1HP, 4000rpm has been used. The Fig. 4.2 shows the DC motor.

Figure 4.2: DC motor

4.2.3 Coupling

To connect the motor with gear box, flexible coupling was used. The Fig. 4.3 shows the flexible coupling.

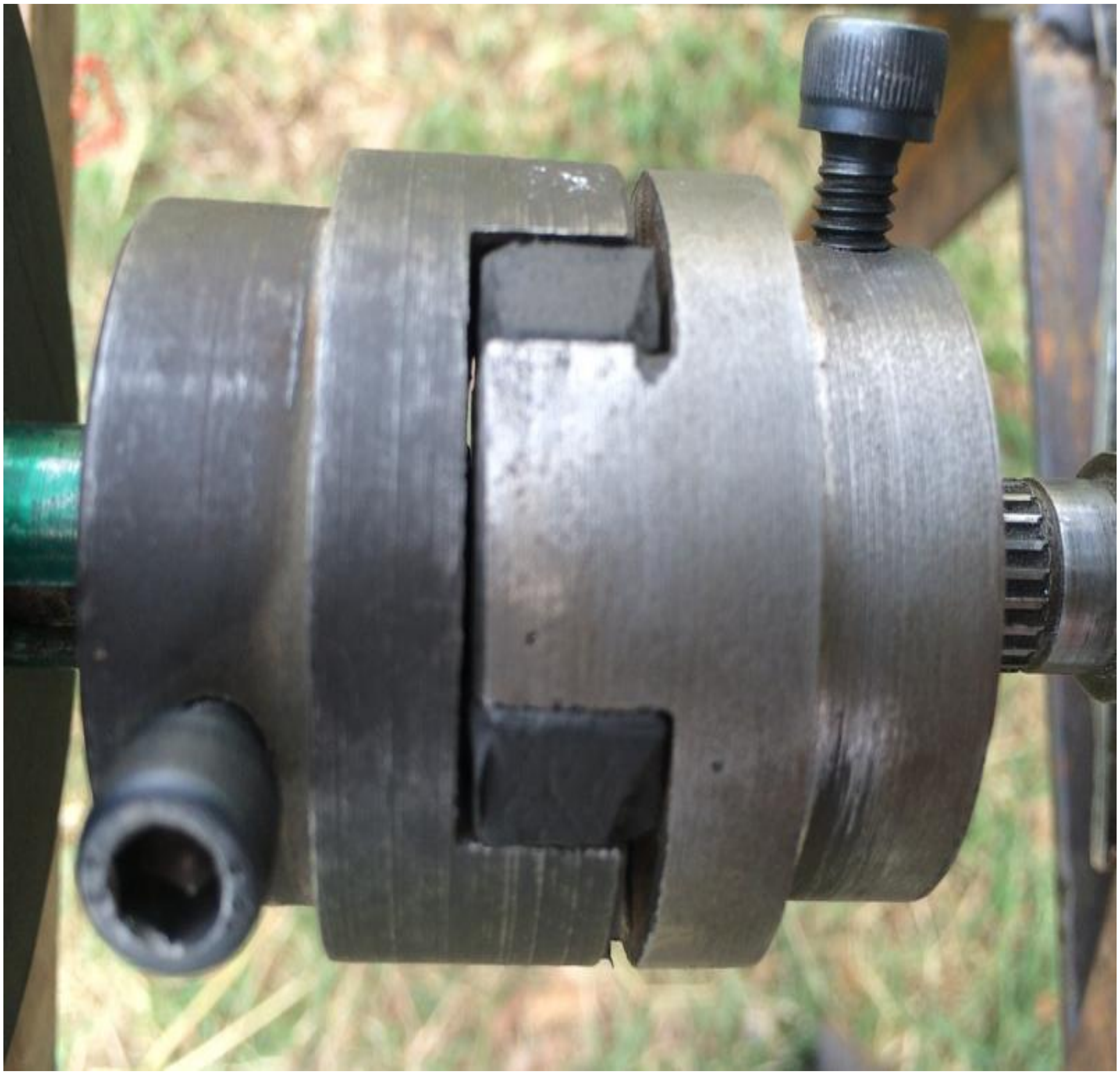

Figure 4.3: Flexible coupling

4.2.4 Gear Box

To perform the fault analysis, gear box of kinetic scooty was used. Gear box consists of one helical gear pair and one spur gear pair. Speed of test gear (big spur gear) was calculated w.r.t input small helical pinion. All gears are made of steel and have 20^0 pressure angle. The picture of kinetic scooty gear box is shown in the Fig. 4.4.

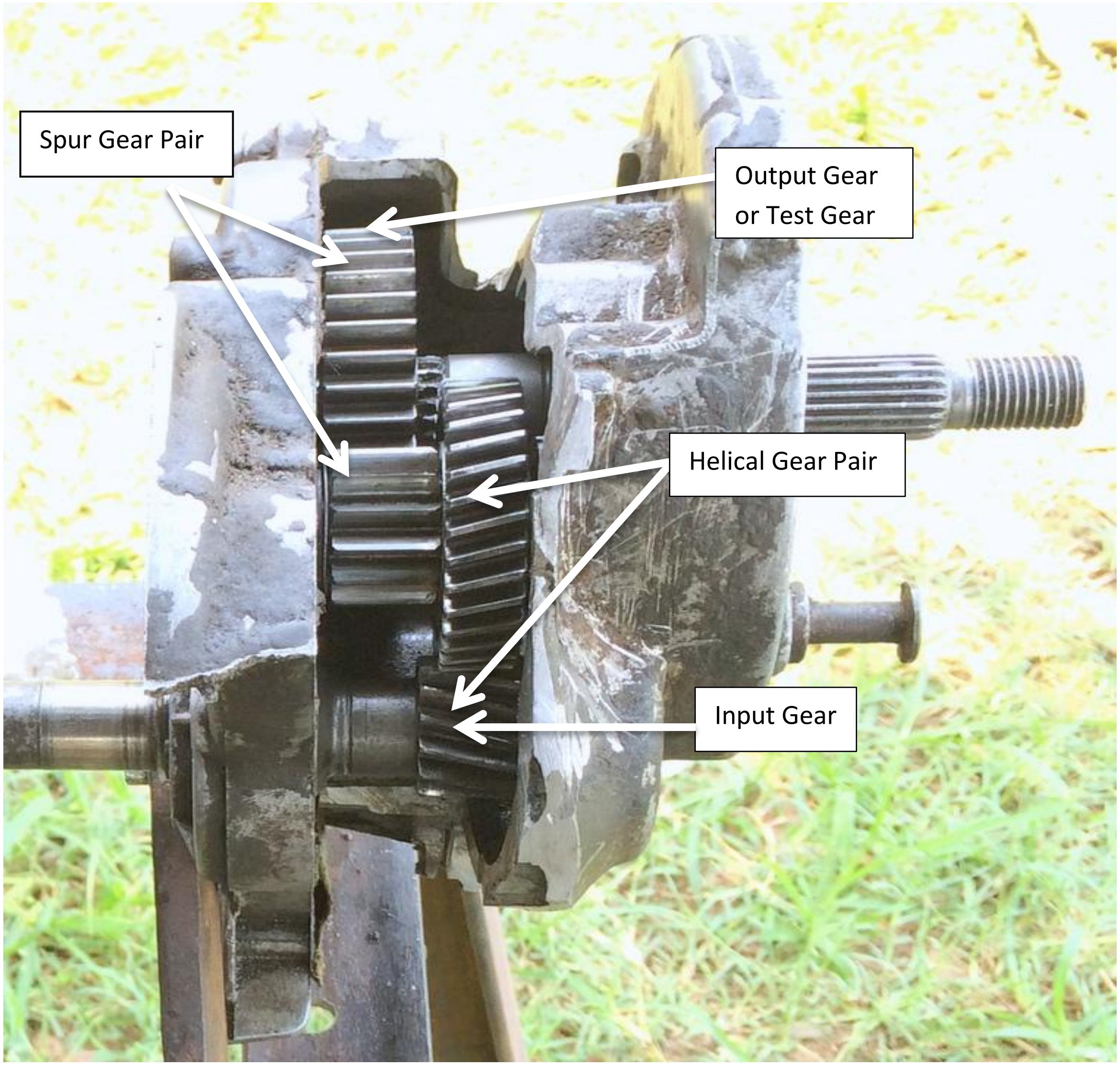

Figure 4.4: Gear box

The gear shafts are supported by ball bearings and proper lubrication has been done. The detail of gears of gear box is given below in Table 4.1.

Table 4.1: Detail of Gear Box

Gear Type	No. of Tooth	Module	PCD	Face Width
Input Helical **Gear (**pinion**)**	14	1.46	20.69mm	13.80mm
Big Helical Gear	33	1.46	48.29mm	11.23mm
Small Spur **Gear (**pinion**)**	13	1.79	23.13mm	17.14mm
Big Spur Gear or Test Gear	36	1.79	64.77mm	14.54mm

4.2.4.1 Test Gear

The big spur gear is taken as test gear. The Fig. 4.5 shows healthy test gear and Fig. 4.6 shows cad model of test gear.

Figure 4.5: Healthy gear

CAD Model

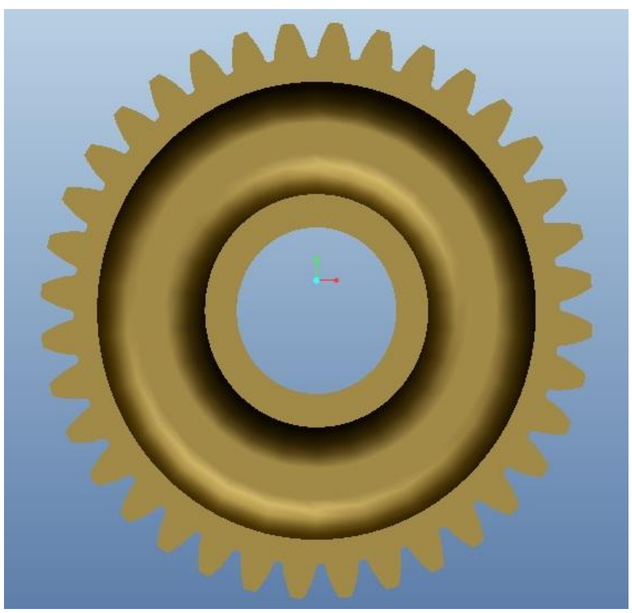

Figure 4.6: CAD Model of healthy gear

4.2.5 Speed Controller

To vary the speed of DC motor, speed controller was used. Main supply was connected to controller and one was made from controller the DC motor's armature. The Fig. 4.7 shows speed controller is given below.

Figure 4.7: Speed controller

4.2.6 Assembly

All the components of the experimental rig were purchased from the market and assembled through nuts and bolts for easy dismantle except mounting frame. The Fig. 4.8 shows the experimental rig with speed controller.

Figure 4.8: Assembly of experimental rig

4.3 Artificial Defects

4.3.1 Crack

For the diagnostic analysis artificial defects have been produced in the healthy output gear. With the help of wire EDM three different dimensions of cut have been produced at the root of one tooth which develops stress concentration which propagates crack. Three different types of cracks were produced with the help of wire EDM as shown below.

Crack C1 is shown in Fig. 4.9. Figure 4.10 shows the length of crack along the face width and Fig. 4.11 shows the CAD Model of crack C1.

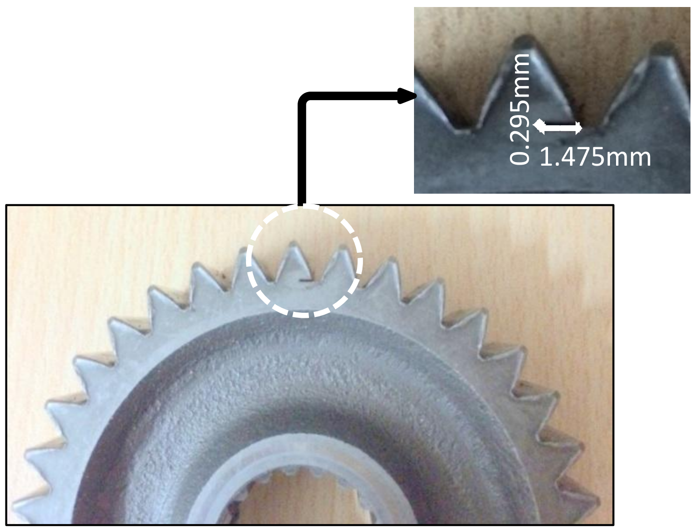

Figure 4.9: Depth and width of crack C1

Isometric view

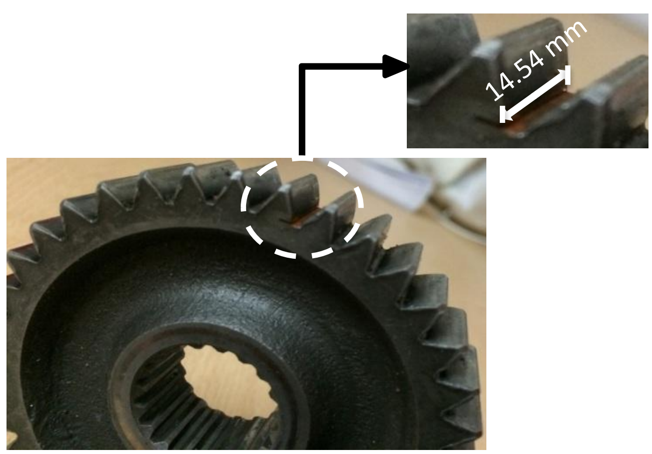

Figure 4.10: Length of crack along the face width of tooth

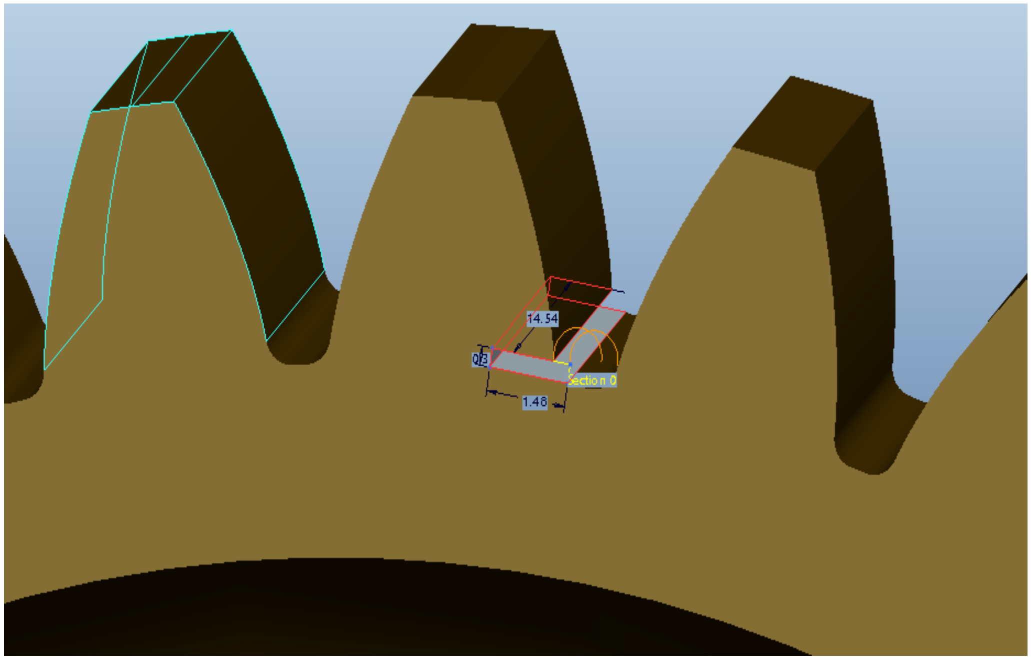

Figure 4.11: CAD Model of test gear with crack C1

Crack C2 is shown in Fig. 4.12 and Fig. 4.13 shows the CAD Model of crack C2.

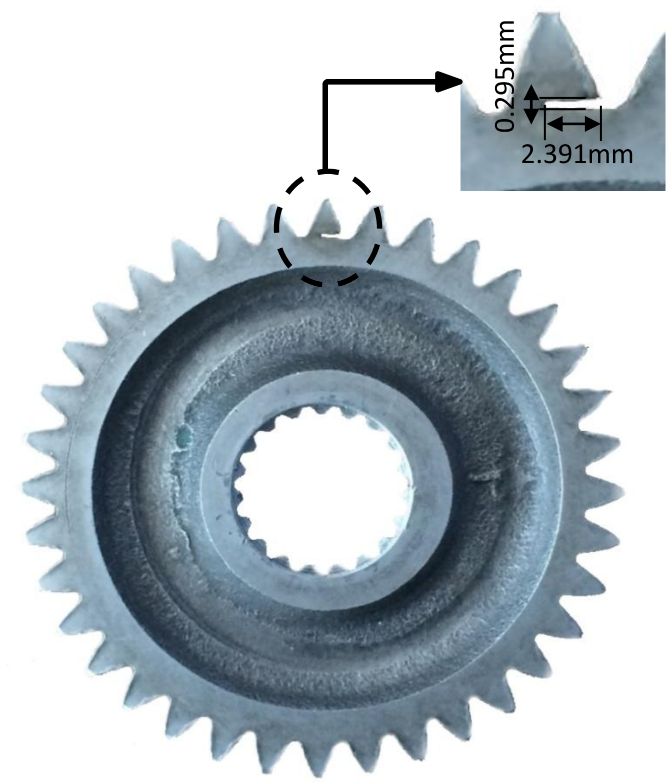

Figure 4.12: Depth and width of crack C2

CAD Model

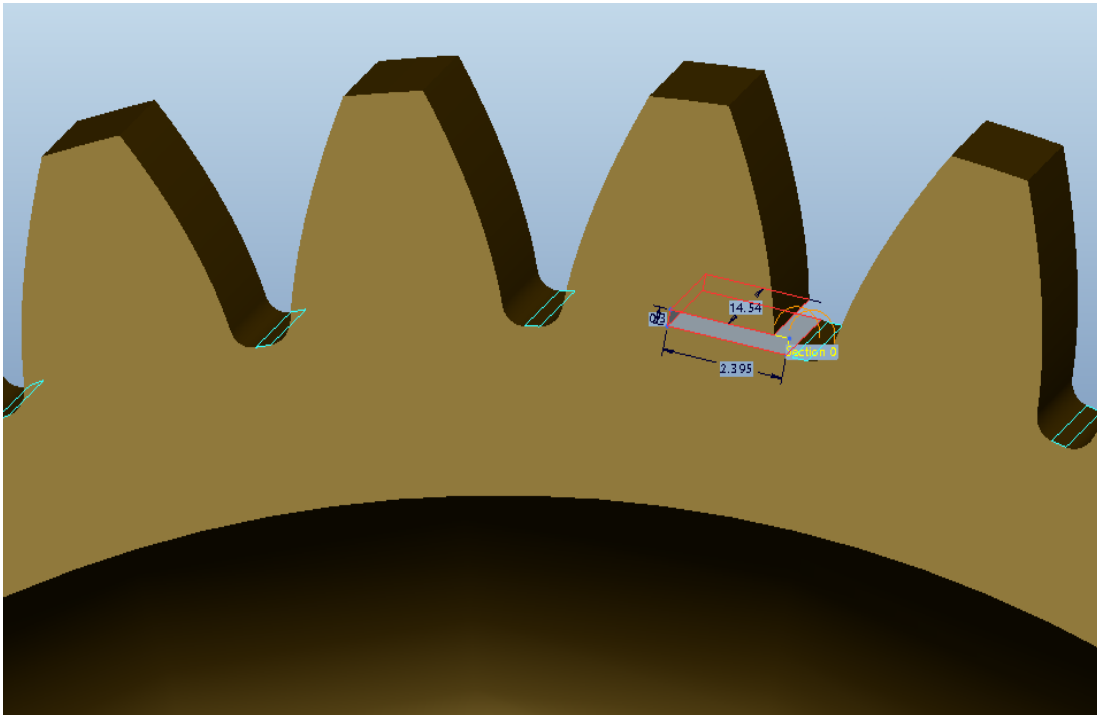

Figure 4.13: CAD Model of test gear with crack C2

Crack C3 is shown in Fig. 4.14 and Fig. 4.15 shows the CAD Model of crack C3.

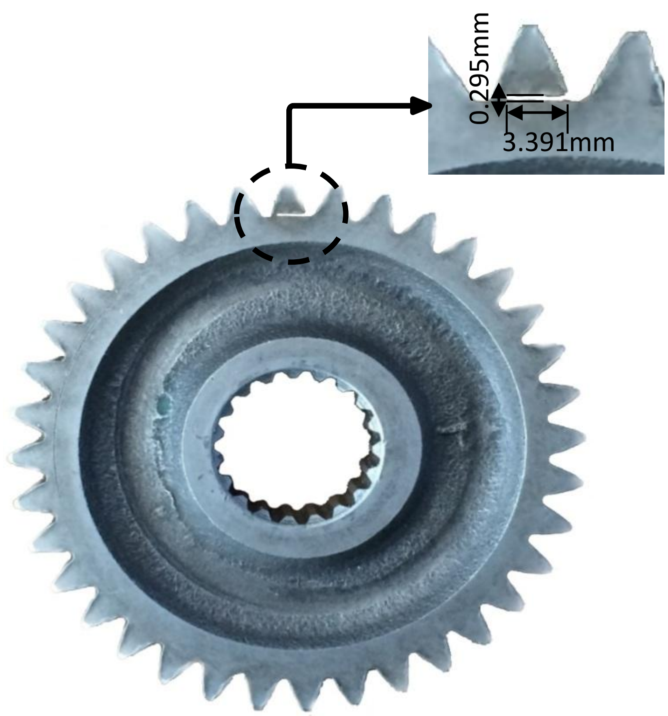

Figure 4.14: Depth and width of crack C3

CAD Model

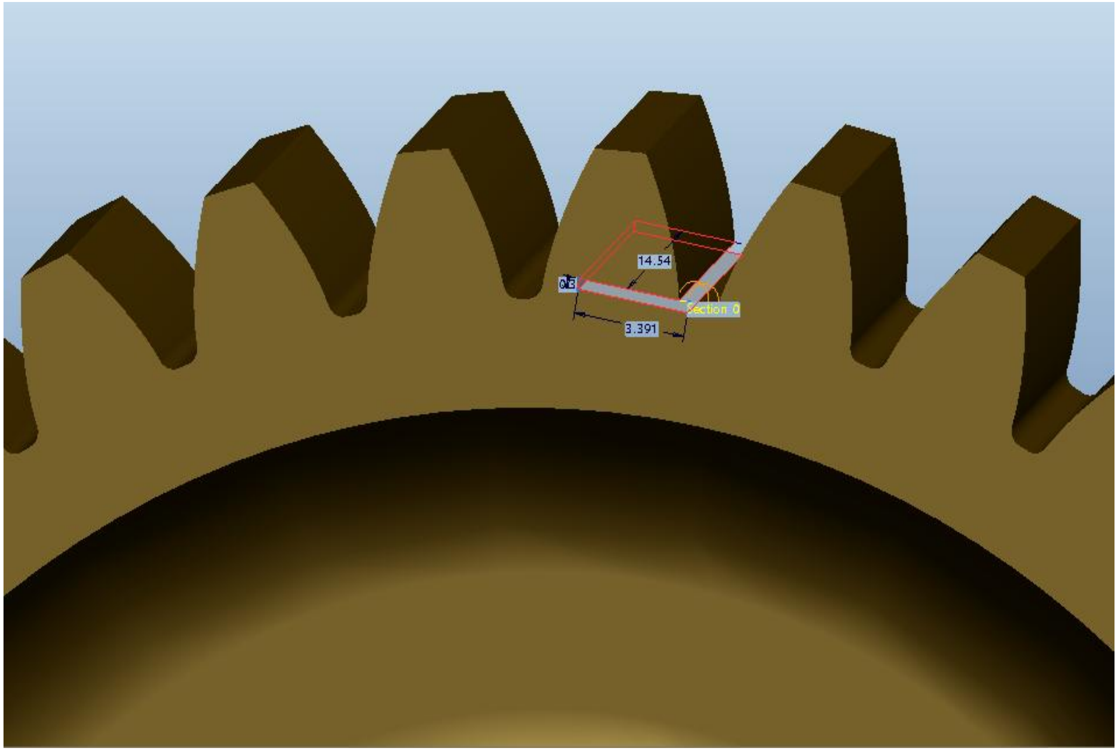

Figure 4.15: CAD Model of test gear with crack C3

Details of crack C1, C2 and C3 are given below in Table 4.2.

Table 4.2: Details of all cracks

Crack Types	Depth	Width	Length
C1	1.475mm	0.295mm	14.54mm
C2	2.395mm	0.295mm	14.54mm
C3	3.391mm	0.295mm	14.54mm

4.3.2 Decrease in Tooth Height

This is the second artificial defect which was produced with the help of grinder at the output gear. In this defect height of only one tooth was decreased. Three types of stubbed tooth were produced with the help of grinder as shown below.

40% Tooth Miss is shown in Fig. 4.16 and its CAD Model is shown in Fig. 4.17.

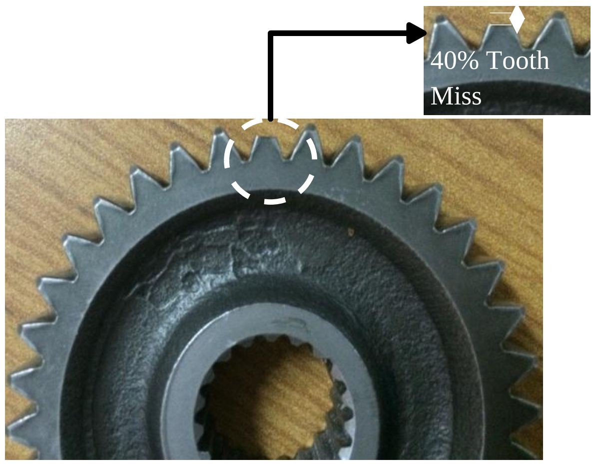

Figure 4.16: 40% Tooth miss

CAD Model

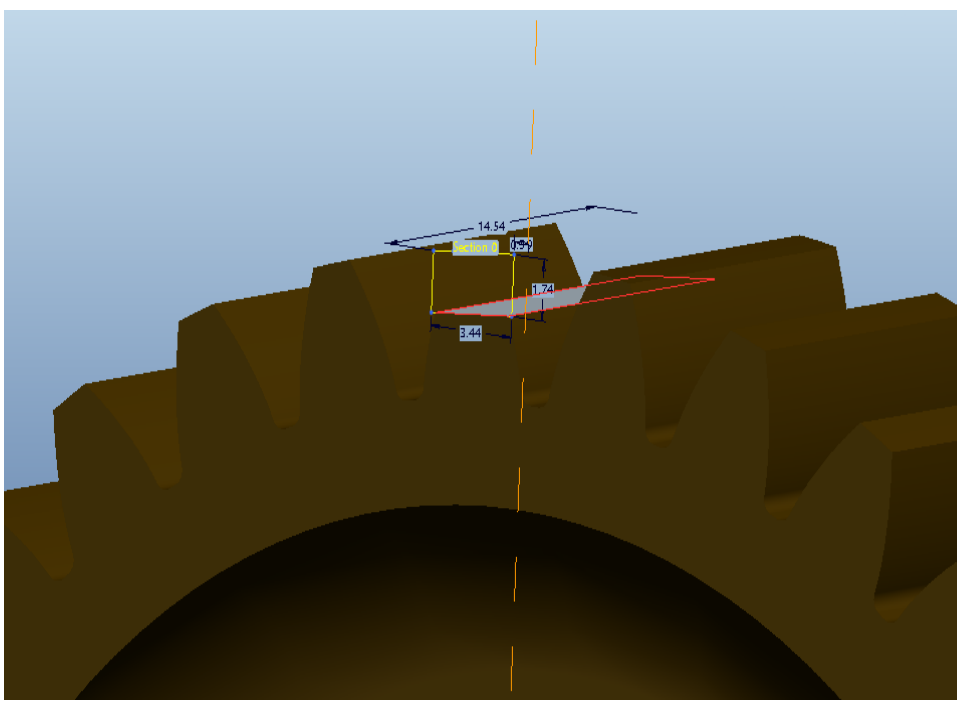

Figure 4.17: CAD Model of test gear with 40% tooth miss

75% Tooth Miss is shown in Fig. 4.18 and its CAD Model is shown in Fig. 4.19.

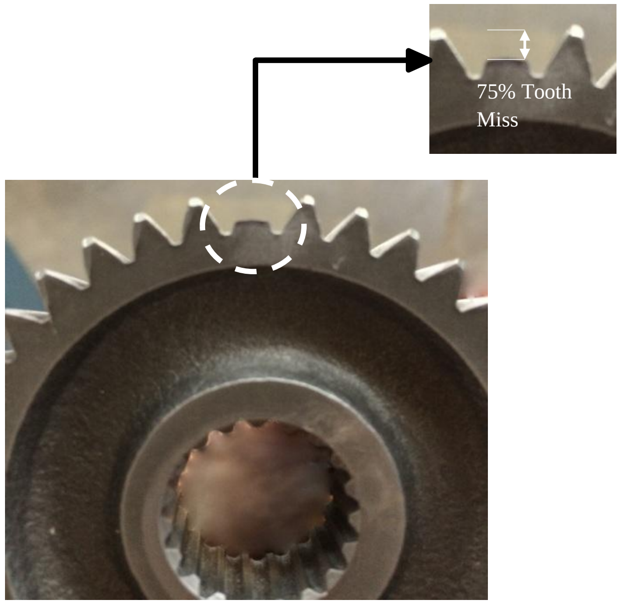

Figure 4.18: 75% Tooth miss

CAD Model

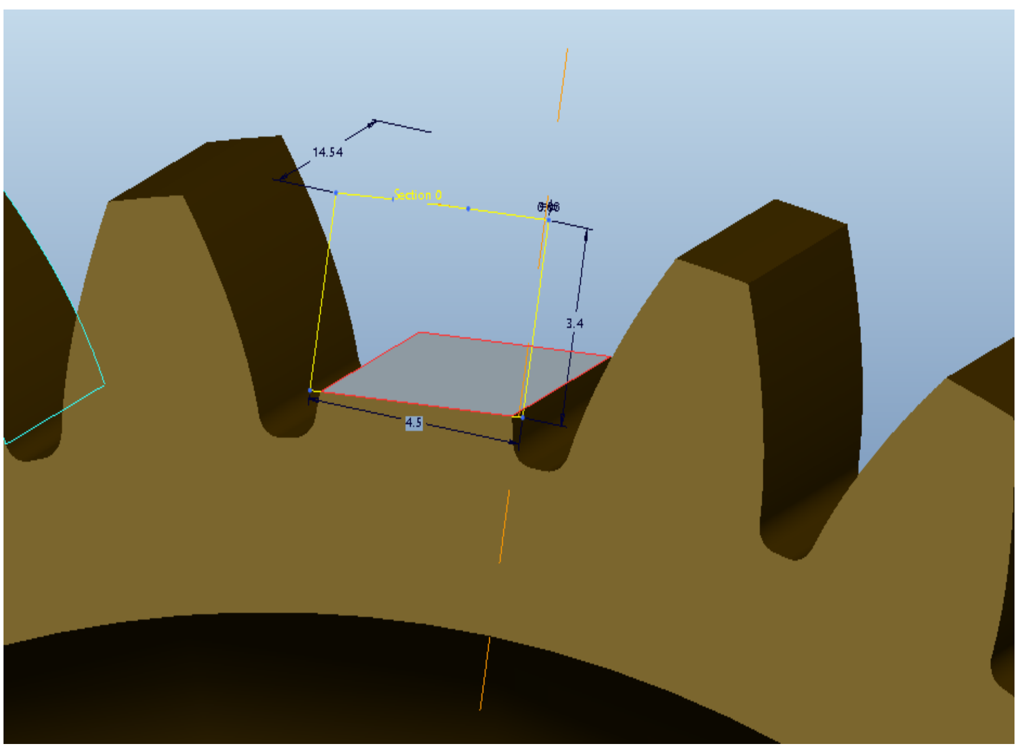

Figure 4.19: CAD Model of test gear with 75% tooth miss

100% Tooth Miss is shown in Fig. 4.20.

Figure 4.20: 100% Tooth miss

In the above defect, a complete tooth was removed from the gear without making any damages to other teeth of the gear. Detail of decrease in tooth height is given below in Table 4.3.

Table 4.3: Details of decrease in tooth height defect

Type of Defects	Decrease in Tooth Height	Total height of Tooth
40% Tooth Miss	1.742mm	4.390mm
75% Tooth Miss	3.292mm	4.390mm
100% Tooth Miss	4.390mm	4.390mm

4.4 Measuring Instrument

4.4.1 Profile Projector

All the measurements of crack and decrease in tooth height defects were measured by using Nikon V10 profile projector. The least count of profile projector is 0.001mm. It can also give measurements in inches. It has movable bed and X-Y scale is shown on its screen. To measuring the distance between two lines, the one line is repositioned according to the X-Y scale to zero set as reference and table is moved until the X axis line or Y axis line reaches to the second measuring line, the measured distance is shown in different digital box. The profile projector is shown below in Fig. 4.21.

Figure 4.21: Nikon V10 profile projector

4.4.2 Tachometer

Beetech HTM-560 Contact type digital tachometer is used to measure the speed of DC motor. The digital contact type tachometer is shown below in Fig. 4.22.

Figure 4.22: Beetech HTM-560 Contact type digital tachometer [W.7]

4.4.3 Sound Level Meter

There are various types of equipment available in market to measure the sound pressure level (SPL). The sound level meter is very common among all of them to measure frequency weighted and time average sound pressure level. It is portable hand held analyzer. The Bruel and Kjaer hand-held analyzer 2250 is shown below in Fig. 4.23.

Figure 4.23: Bruel and Kjaer hand-held analyzer 2250

4.4.3.1 Microphone

It transforms the fluctuation in pressure of sound waves into time based electrical signal. Various types of microphones are written below. Fig. 4.24 shows the sectional view of it.

- Condenser type microphone
- Repolarized type microphone
- Piezoelectric type microphone

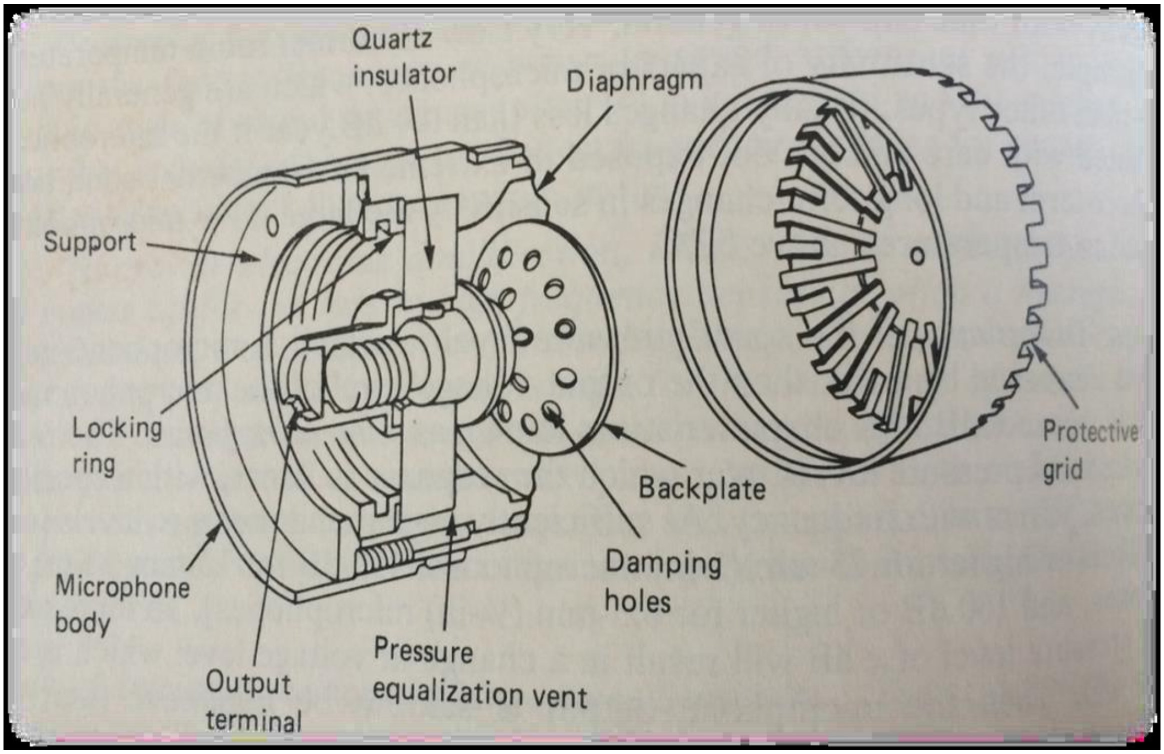

Figure 4.24: Sectional view of microphone

4.4.3.2 Amplifier

The sound receive by microphone is amplified in amplifier to measure the low sound pressure level. It can amplify over a wide range of span.

4.4.3.3 Rectifier

It transforms the electrical signal into digital form.

4.4.3.4 Operating Procedure of Sound Level Meter

- Calibrate the instrument before taking measurements to check the sensitivity of microphone.
- Measure the background noise and correct it with the background correction factor.

- Measure the sound pressure level of desired source.

4.5 Experimental Procedure

Figure 4.25 shows the experimental rig. The gear box was run by DC motor and its speed was measured by tachometer. Bruel and Kjaer (B and K) portable hand-held analyzer type 2250 with microphone and windscreen was positioned at 1.0 meter away at 45^0 angle. The B and K frequency analysis software BZ5503 is used to analyze the result. First the test gear of the healthy gear box is run at five different speeds and the sound pressure level was measured in 1-1 octave band at five different speeds after that two type of artificial defects, cut at the root of tooth which led to the propagation of crack in tooth and second was the decrease in height of gear, were produced at the test gear. In the first case faulty crack C1 is produced at the test gear and sound pressure level was measured at five different speeds then second faulty crack C2 is generated with the help of wire EDM and sound pressure level was measured at five different speeds. Further faulty crack C3 was generated in same way and sound pressure level was measured at five different speeds. The crack depths are ranged from 1.475 to 3.391 mm with thickness of almost 0.295 mm. Each measurement reading of sound pressure level takes 1.0 minute. Total 15 measurement of sound pressure level were measured for faulty crack case.

Figure 4.25: Measuring sound pressure level of gear box

In same manner second type of defect, decrease in height of tooth up to 40% of the total height of tooth was produced with the help of grinder and sound pressure level was measured in 1-1 octave band at five different speeds then height of tooth was further decreased up to 75% and sound pressure level was measured at five different speeds after that complete one tooth was removed from the gear box and run at five different speeds to measure the sound pressure level. Each measurement reading of sound pressure level takes 1.0 minute. Total 15 measurement of sound pressure level were measured for decrease in height of tooth case. Measurements of sound pressure level were carefully done for the sake of every small and increasing change in the responses.

Chapter-5

Results and Discussion

There were two artificial defects imposed in the gear to study the gear box noise for fault analysis. Experiments were conducted with healthy gear, faulty gear with different crack C1, C2, C3, decrease in tooth height by 40%, 75% and 100%, each at test gear speed 180, 360, 540, 725 and 900 rpm respectively. For both case results are discussed below.

5.1 Case-I [crack]

Normal Gear

From Fig. 5.1 it can be seen that, there is gradual increase in sound pressure level (SPL) as speed of gear increases. As the speed increases from 180 to 900 rpm, the SPL of healthy gear also increases from 59.1 dB (A) to 74.7 dB (A). The regression equation for normal gear is sound pressure level=4.03*rpm + 55.31.

Faulty Crack C1

From Fig. 5.1 it can be seen that, there is increase in SPL due to crack defect and as the speed of gear increases the SPL also increases. At 180 rpm, SPL is 63.2 dB (A) and at 900 rpm, SPL is 77.7 dB (A) was recorded for crack C1. The regression equation for crack C1 is sound pressure level =3.71*rpm + 59.51.

Faulty Crack C2

From Fig. 5.1 it can be seen that, there is more increase in SPL as compared to the C1. The SPL is increases as the speed of gear increases. The SPL is increases from 64.4 dB (A) at 180 rpm to 80.8 dB (A) at 900 rpm. The regression equation for C2 is sound pressure level =4.09*rpm + 60.37.

Faulty Crack C3

From Fig. 5.1 it can be seen that, there is further increase in SPL due to the increase in crack depth. At 180 rpm, SPL was 65.4 dB (A) and at 900 rpm, SPL was 82.1 dB (A). The regression equation for C3 is sound pressure level =4.23*rpm + 60.09. There is more than 14 dB (A) difference between speed 180 rpm to 900 rpm in each case.

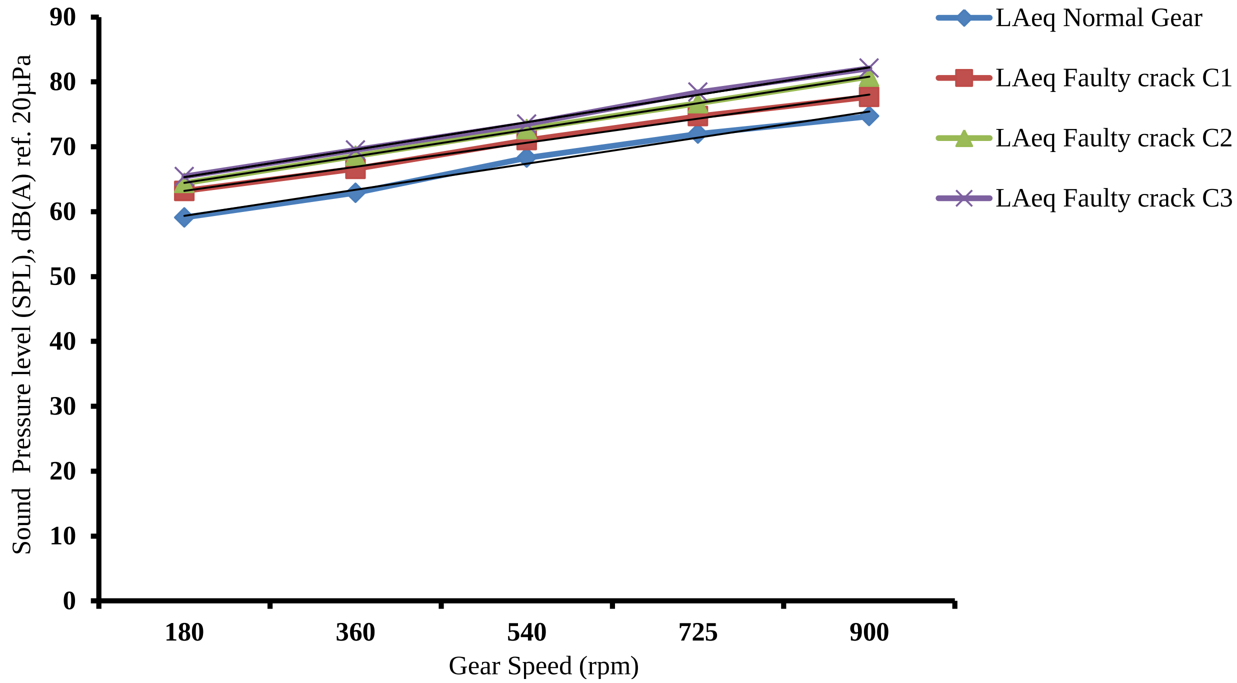

Figure 5.1: sound pressure level vs. Speed with cracks

As speed increases it is seen that peak sound level increases. The range is as shown below.

Table 5.1: Peak sound pressure level in dB (A) vs. speed with crack

Motor speed (input)	**500 rpm**	**1000 rpm**	**1500 rpm**	**2000 rpm**	**2500 rpm**
Test gear speed (output)	**180 rpm**	**360 rpm**	**540 rpm**	**725 rpm**	**900 rpm**
Normal Gear	59.1	62.9	68.3	72.0	74.7
Crack C1	63.2	66.6	71.0	74.7	77.7
Percentage increase in C1	6.93%	5.58%	3.95%	3.75%	4.01%
Crack C2	64.4	68.6	72.7	76.7	80.8
Percentage increase in C2	8.69%	9.06%	6.44%	6.52%	8.16%
Crack C3	65.4	69.5	73.5	78.4	82.1
Percentage increase in C3	10.65%	10.49%	7.61%	8.88%	9.90%

Summarizing, the peak sound level increases with the speed and with crack depth as well. At higher speed percentage increase in SPL is less.

5.1.1 Frequency Spectra

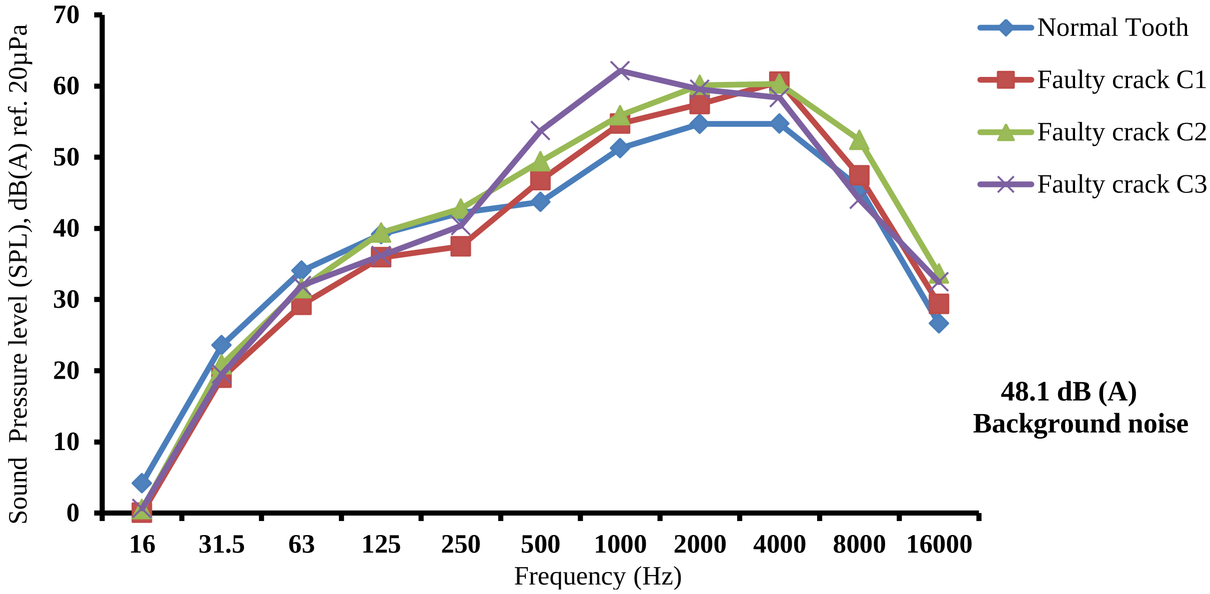

Figure 5.2: sound pressure level vs. Frequency at 180 rpm

At test gear speed 180 rpm, the sound pressure level (SPL) was measured at a distance of 1 meter away from the gear box casing at 45^0 angles with wind screen in frequency domain. All the four cases, SPL are measured at 180 rpm.

Normal Gear from Fig. 5.2, it can be seen that high noise level of frequency ranges are 2000 to 4000 Hz. In this graph, SPL at different frequencies of healthy gear box is shown. The SPL peak occurs at frequency range from 2000 to 4000 Hz is annoying for human hearing. At the peak the SPL is 54.7 dB (A).

Faulty Crack C1 from Fig. 5.2 it can be seen that SPL increases as the fault is localized in the gearbox. The peak occurs at 4000 Hz with 60.6 dB (A) noise level.

Faulty Crack C2 from Fig. 5.2 it can be seen that SPL increases with increase in depth of crack in gear. The peak occurs at frequency range 2000 to 4000 Hz with 60.6 dB (A) noise level.

Faulty Crack C3 from Fig. 5.2 it can be seen that SPL increases more as further increase in depth of crack in gear. The peak occurs at frequency 1000 Hz with 62.1 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 250 Hz after that there is sudden increase in SPL up to 1000 Hz, after that graph remains almost flat from the frequency range 1000 Hz to 4000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at low frequency, when defect is more. The combination of tooth mesh frequencies occurs at 108 Hz and 116.66 Hz causes the sudden increase of noise level after 250 Hz. There is 7.4 dB (A) difference at peak when compared the faulty crack C3 with healthy gear.

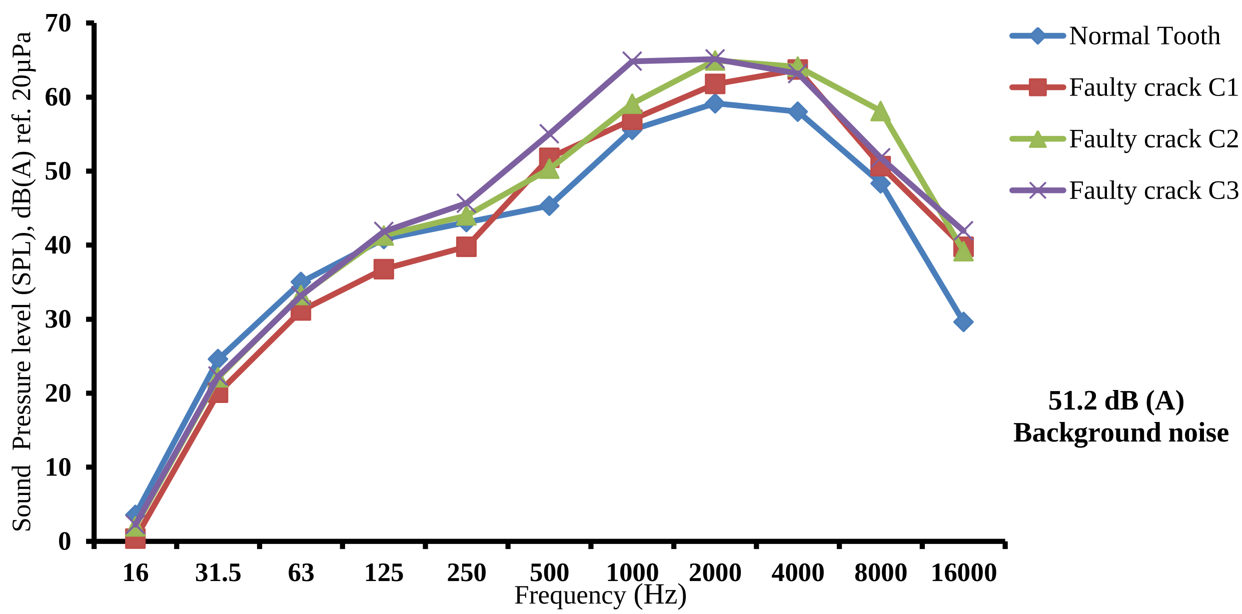

Figure 5.3: sound pressure level vs. Frequency at 360 rpm

Normal Gear from Fig. 5.3 it can be seen that at test gear speed 360 rpm, the SPL peak occurs at 2000 Hz with noise level of 59.1 dB(A). Fig. 3 indicates high noise level occurs at frequency range of 2000 to 6000 Hz. Peak at frequency 2000 Hz is annoying for human hearing.

Faulty Crack C1 from Fig. 5.3 it can be seen that SPL increases as the fault is localized in the gearbox. The peak occurs at 4000 Hz with 63.7 dB (A) noise level.

Faulty Crack C2 from Fig. 5.3 it can be seen that SPL increases with increase in depth of crack in gear. The peak occurs at 2000 Hz with 64.9 dB (A) noise level.

Faulty Crack C3 from Fig. 5.3 it can be seen that SPL increases with increase in depth of crack in gear. The peak occurs at frequency range 1000 to 2000 Hz with 65.1 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 250 Hz after that there is sudden increase in SPL up to 1000 Hz, a graph remain almost flat from frequency range between 1000 Hz to 4000 Hz after which there is sudden decrease in SPL is observed due to the influence of structure resonance frequencies. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at low frequency, when defect is more. Tooth mesh frequencies occurs at 217.27 Hz and 233.33 Hz which causes the sudden increase in noise level after 250 Hz. There is 6 dB (A) difference in the peak when compared the faulty crack C3 with healthy gear.

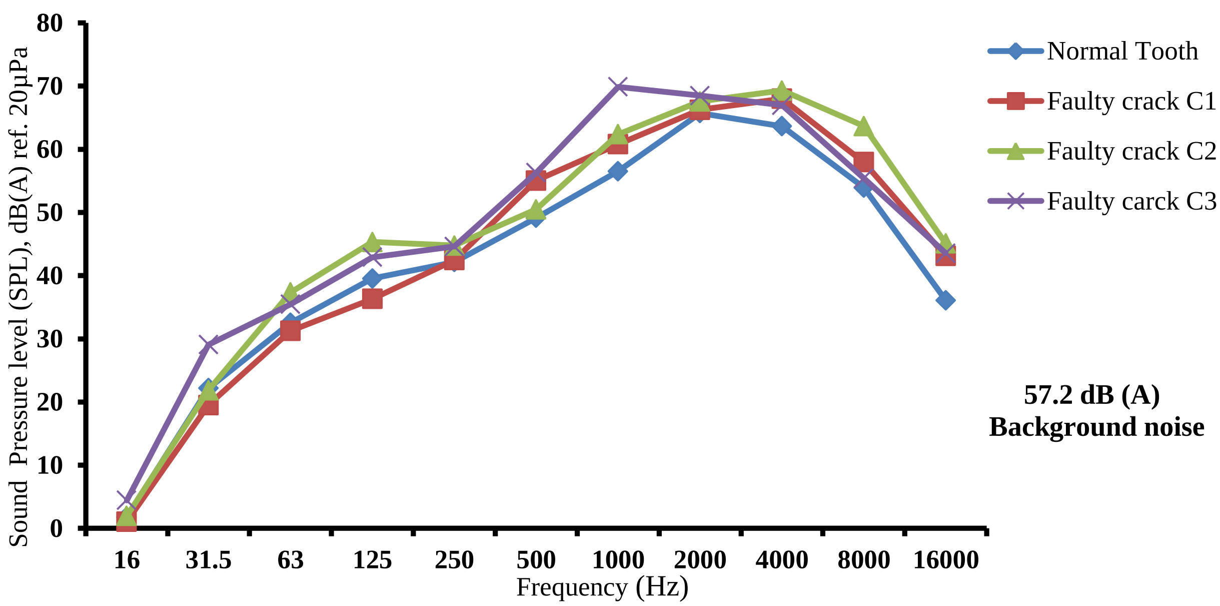

Figure 5.4: sound pressure level vs. Frequency at 540 rpm

Normal Gear from Fig. 5.4 it can be seen that at test gear speed 540 rpm, the SPL peak occurs at 2000 Hz with noise level of 65.7 dB (A). Fig. 4 indicates high noise level occur at frequency range 2000 to 4000 Hz. Peak at frequency 2000 Hz is annoying for human hearing.

Faulty Crack C1 from Fig. 5.4 it can be seen that SPL increases as the fault is localized in the gearbox. The peak occurs at 4000 Hz with 68 dB (A) noise level.

Faulty Crack C2 from Fig. 5.4 it can be seen that SPL increases as the further increase in depth of crack in gear. The peak occurs at frequency 4000 Hz with 69.3 dB (A) noise level.

Faulty Crack C3 from Fig. 5.4 it can be seen that SPL increases more as further increase in depth of crack in gear. The peak occurs at frequency 1000 Hz with 69.9 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 350 Hz after that there is sudden increase in SPL up to 1000 Hz, after that graph remains almost flat from frequency range 1000 to 4000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at low frequency, when defect is more. Tooth mesh frequencies occur at 325.91 Hz and 350 Hz which causes the sudden increase of noise level after 350 Hz. There is 4.2 dB (A) difference in the peak when compared the faulty crack C3 with healthy gear.

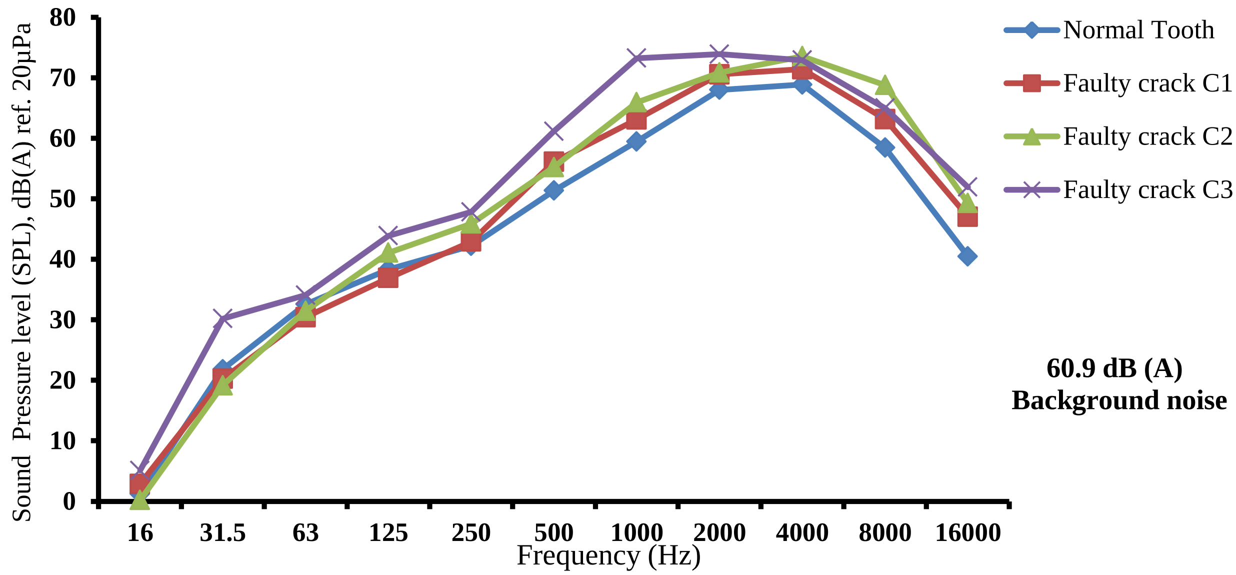

Figure 5.5: sound pressure level vs. Frequency at 725 rpm

Normal Gear from Fig. 5.5 it can be seen that at test gear speed 725 rpm, the SPL peak occurs at 4000 Hz with noise level of 68.9 dB(A). Fig. 5 indicates high noise level occurs at frequency range 2000 to 4000 Hz. Peak at frequency 4000 Hz is annoying for human hearing.

Faulty Crack C1 from Fig. 5.5 it can be seen that SPL increases as the fault is localized in the gearbox. The peak occurs at 4000 Hz with 71.4 dB (A) noise level.

Faulty Crack C2 from Fig. 5.5 it can be seen that SPL increases as the further increase in depth of crack in gear. The peak occurs at frequency 4000 Hz with 73.6 dB (A) noise level.

Faulty Crack C3 from Fig. 5.5 it can be seen that SPL increases more as further increase in depth of crack in gear. The peak occurs at frequency 2000 Hz with 73.9 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 400 Hz after that there is sudden increase in SPL up to 2000 Hz, after that graph remains almost flat from the frequency range 2000 to 5000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at low frequency, when defect is more. Tooth mesh frequencies occurs at 434.55 Hz and 466.66 Hz which causes the sudden increase of noise level after 400 Hz. There is 5 dB (A) difference in the peak when compared the faulty crack C3 with healthy gear.

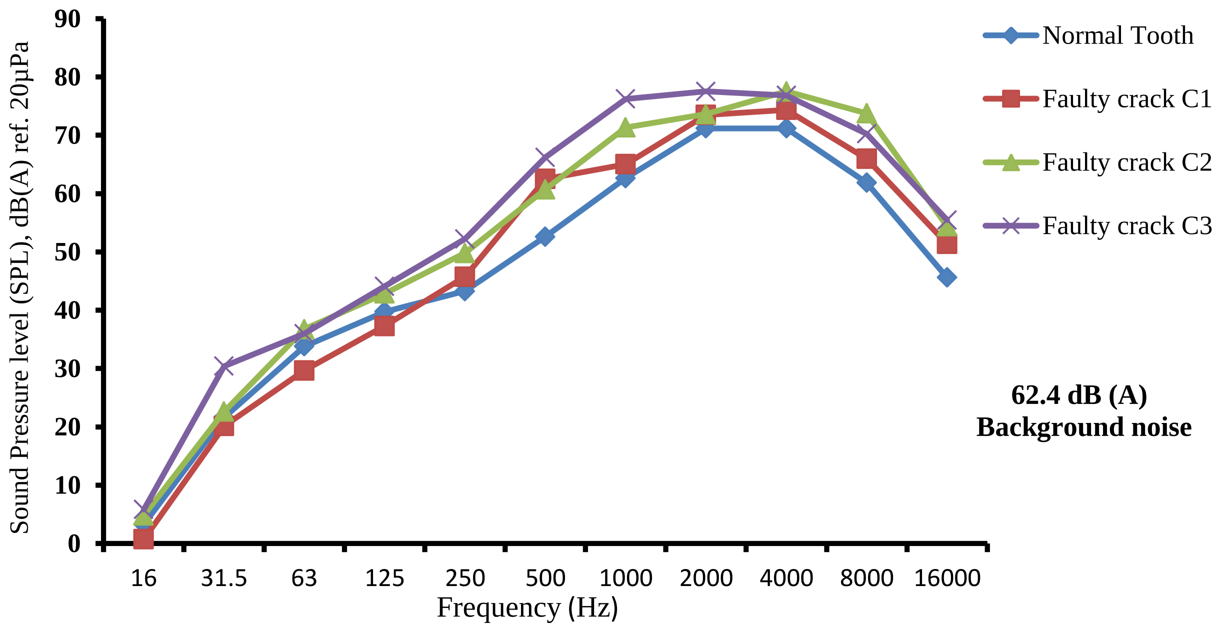

Figure 5.6: sound pressure level vs. Frequency at 900 rpm

Normal Gear from Fig. 5.6 it can be seen that at test gear speed 900 rpm, the SPL peak occurs at frequency range 2000 to 4000 Hz with noise level of 71.2 dB (A). Fig. 6 indicates high noise level at frequency range 2000 to 4000 Hz.

Faulty Crack C1 from Fig. 5.6 it can be seen that SPL increases as the fault is localized in the gearbox. The peak occurs at 4000 Hz with 74.4 dB (A) noise level.

Faulty Crack C2 from Fig. 5.6 it can be seen that SPL increases as the further increase in depth of crack in gear. The peak occurs at frequency 4000 Hz with 77.5 dB (A) noise level.

Faulty Crack C3 from Fig. 5.6 it can be seen that SPL increases more as further increase in depth of crack in gear. The peak occurs at frequency 2000 Hz with 77.5 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 475 Hz after that there is sudden increase in SPL up to 2000 Hz, after that graph remains almost flat from the frequency range 2000 to 5000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at low frequency, when defect is more. Tooth mesh frequencies occurs at 543.19 Hz and 583.33 Hz which causes the sudden increase of noise level after 400 Hz. There is 6.3 dB (A) difference in the peak when compared the faulty crack C3 with healthy gear.

The SPL peak at frequencies w.r.t. speed is shown below in table.

Table 5.2: peak sound pressure level at frequencies w.r.t. speed

Test gear speed		**180 rpm**	**360 rpm**	**540 rpm**	**725 rpm**	**900 rpm**
Normal Gear	1KHz					
	2KHz	54.7 dB (A)	59.1 dB (A)	65.7 dB (A)		71.2 dB (A)
	4KHz	54.7 dB (A)	59.1 dB (A)		68.9 dB (A)	71.2 dB (A)
Faulty crack C1	1KHz					
	2KHz					
	4KHz	60.6 dB (A)	63.7 dB (A)	68.0 dB (A)	71.4 dB (A)	74.4 dB (A)
Faulty crack C2	1KHz					
	2KHz	60.6 dB (A)	64.9 dB (A)			
	4KHz	60.6 dB (A)		69.3 dB (A)	73.6 dB (A)	77.5 dB (A)
Faulty crack C3	1KHz	62.1 dB (A)	65.1 dB (A)	69.9 dB (A)		
	2KHz		65.1 dB (A)		73.9 dB (A)	77.5 dB (A)
	4KHz					

In all cases peak noise level occurs in mid frequency range due to the combination of gear mesh frequencies and their harmonics. There is a lower shift in frequency corresponding to peak noise level as crack size increases and an increase in peak noise level as crack increases.

5.2 Case-II [Decrease in Tooth Height]

Normal Gear

From Fig. 5.7 it can be seen that, there is gradual increase in sound pressure level as speed of gear increases. As the speed increases from 180 to 900 rpm, the SPL of healthy gear also increases from 59.1 dB (A) to 74.7 dB (A). The regression equation for normal gear is sound pressure level =4.03*rpm + 55.31.

40% Decrease in Height

From Fig. 5.7 it can be seen that, there is increase in SPL due to defected tooth and as the speed of gear increases the SPL also increases. At 180 rpm, SPL is 62.5 dB (A) which gradually increases up to 76.9 dB (A) at 900 rpm. The regression equation for 40% decrease in tooth height is sound pressure level =3.85*rpm + 58.09.

75% Decrease in Height

From Fig. 5.7 it can be seen that, there is more increase in SPL as compared to the 40% tooth missing case. The SPL 64.2 dB (A) at 180 rpm is increases up to 78.0 dB (A) at 900 rpm. The regression equation for 75% decrease in tooth height is sound pressure level =3.52*rpm + 60.6.

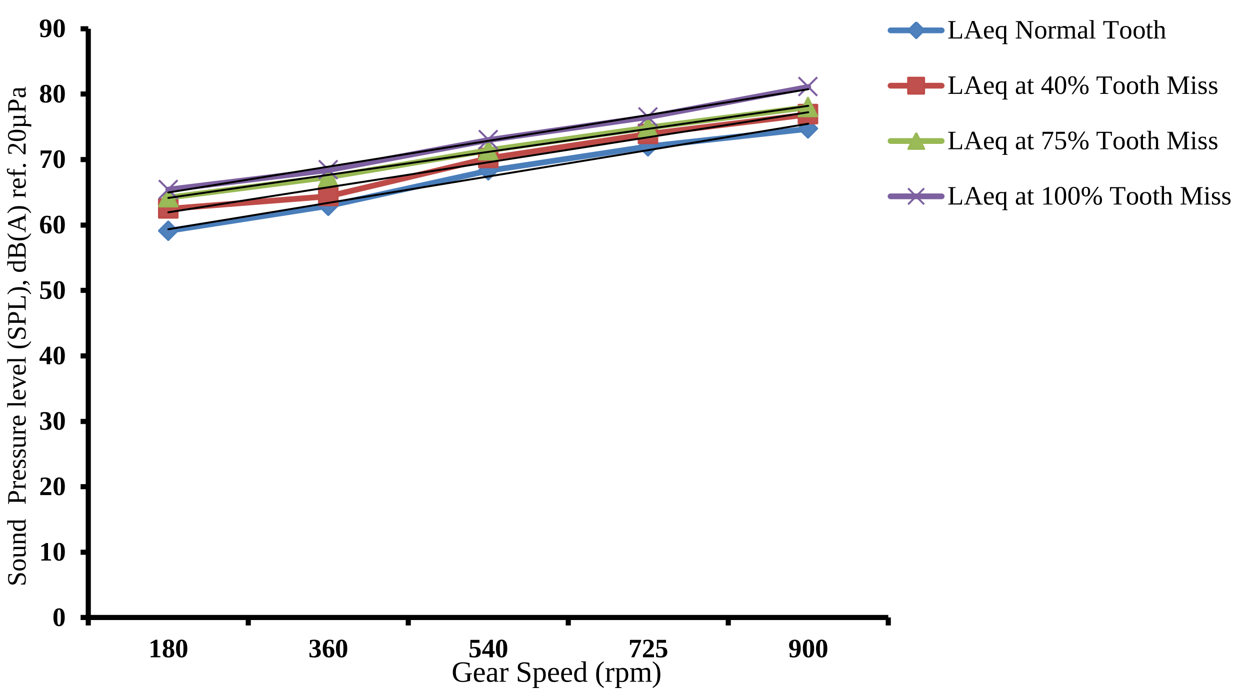

Figure 5.7: sound pressure level vs. speed with decrease in tooth height

100% Tooth Removed

From Fig. 5.7 it can be seen that, there is further increase in SPL due to tooth missing. At 180 rpm, SPL is 65.4 dB (A) which increases up to 81.1 dB (A) at 900 rpm. The regression equation for the complete tooth miss is sound pressure level =3.95*rpm + 61.03. There is more than 14 dB (A) difference between 180 rpm to 900 rpm in each case.

As speed and fault increases it is seen that peak sound level increases. They range is as shown below.

Table 5.3: Peak sound level vs. speed with decrease in tooth height

Test Gear speed	180 rpm	360 rpm	540 rpm	725 rpm	900 rpm
Normal Tooth	59.1 dB (A)	62.9 dB (A)	68.3 dB (A)	72.0 dB (A)	74.7 dB (A)
40% Tooth Miss	62.5 dB (A)	64.4 dB (A)	70.2 dB (A)	73.9 dB (A)	76.9 dB (A)
Percentage increase in 40% Tooth Miss	5.75%	2.38%	2.78%	2.63%	2.94%
75% Tooth Miss	64.2 dB (A)	67.3 dB (A)	71.4 dB (A)	74.9 dB (A)	78.0 dB (A)
Percentage increase in 40% Tooth Miss	8.62%	6.99%	4.53%	4.02%	4.41%
100% Tooth Miss	65.4 dB (A)	68.4 dB (A)	73 dB (A)	76.5 dB (A)	81.1 dB (A)
Percentage increase in 100% Tooth Miss	10.65%	8.74%	6.88%	6.25%	8.56%

All the cases is linearly varying with spur gear speed and more is percentage of tooth removed the noise is more. However, the percentage increase in SPL decreases as the speed increases.

5.2.1 Frequency Spectra

Normal Gear From Fig. 5.8 it can be seen that at motor speed of 500 rpm, test gear measured speed 180 rpm, the sound pressure level (SPL) is 54.7 dB (A) at the peak From Fig. 8, it can be seen that high noise level of frequency ranges are 2000 to 4000 Hz. The SPL peak occurs at frequency range from 2000 to 4000 Hz is annoying for human hearing.

40% Decrease in Height from Fig. 5.8 it can be seen that, the SPL increases as the fault is localized in the gearbox. The peak occurs at 1000 Hz with 57.7 dB (A) noise level.

75% Decrease in Height from Fig. 5.8 it can be seen that, the SPL increases as the further decrease in height of tooth in gear. The peak occurs at frequency 2000 Hz with 60.0 dB (A) noise level.

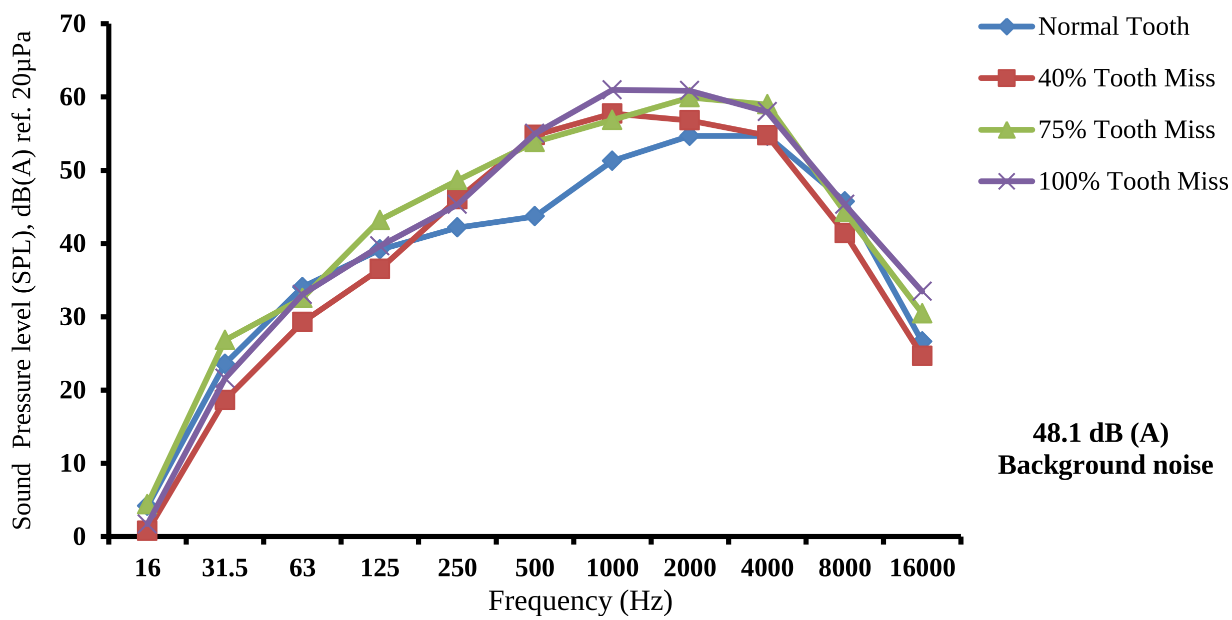

Figure 5.8: sound pressure level vs. Frequency at 180 rpm

100% Tooth Removed from Fig. 5.8 it can be seen that, the SPL increases more as tooth is removed from the gear. The peak occurs at frequency 1000 Hz with 61.0 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 250 Hz after that there is sudden increase in SPL up to 1000 Hz, after that graph remains almost flat from the frequency range 1000 to 4000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. Collaboration of tooth mesh frequencies occurs at 108 Hz and 116.66 Hz causes the rise in noise level of mid frequencies. There is not too much difference in the SPL at low frequencies of healthy gear and faulty gear but there is significant increase in SPL due to higher frequency contributions. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at higher frequency, when defect is more. There is 6.4 dB (A) difference at peak when compared the faulty 100% complete tooth miss with healthy gear.

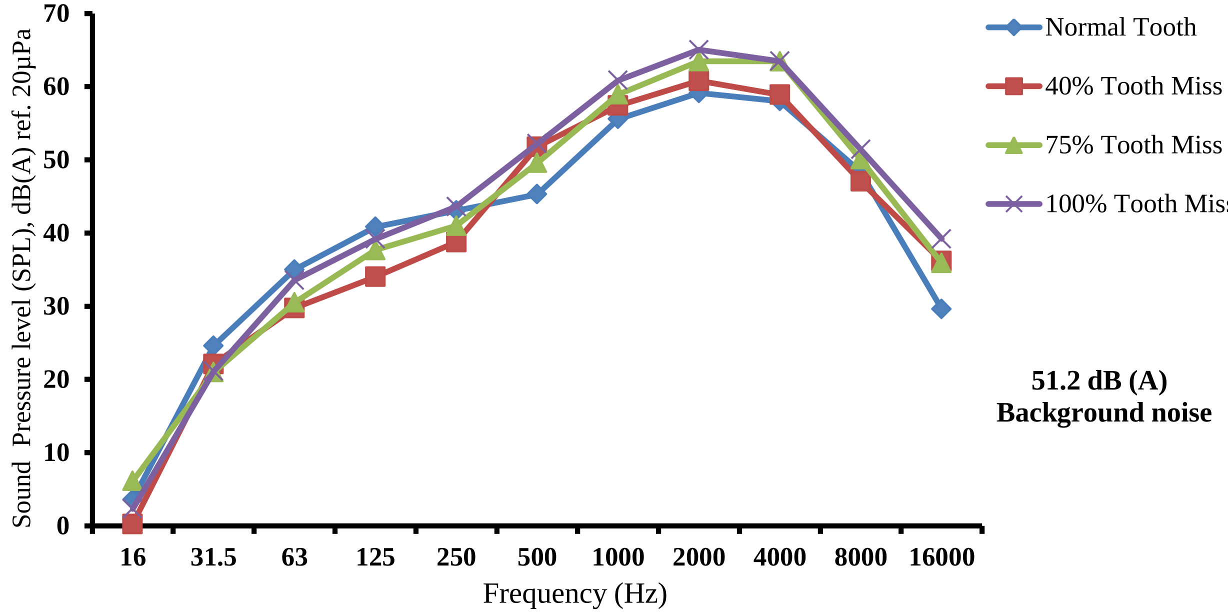

Figure 5.9: sound pressure level vs. Frequency at 360 rpm

Normal Gear from Fig. 5.9 it can be seen that at test gear speed 360 rpm, the SPL peak occurs at 2000 Hz with noise level of 59.1 dB(A). Fig. 9 indicates high noise level occurs at frequency range of 2000 to 6000 Hz. Peak at frequency 2000 Hz is annoying for human hearing.

40% Decrease in Height from Fig. 5.9 it can be seen that, the SPL increases as the fault is localized in the gearbox. The peak occurs at 2000 Hz with 60.8 dB (A) noise level.

75% Decrease in Height from Fig. 5.9 it can be seen that, the SPL increases as the further decrease in height of tooth in gear. The peak occurs at frequency range 2000 to 4000 Hz with 63.5 dB (A) noise level.

100% Tooth Removed from Fig. 5.9 it can be seen that, the SPL increases more as tooth is removed from the gear. The peak occurs at frequency 2000 Hz with 65.0 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 250 Hz after that there is sudden increase in SPL up to 1000 Hz, after that graph remains almost flat from the frequency range 1000 to 4000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. Tooth mesh frequencies occur at 217.27 Hz and 233.33 Hz which causes the sudden increase in graph after 250 Hz. There is not too much difference in the SPL at low frequencies of healthy gear and faulty gear but there is significant increase in SPL due to higher frequency contributions. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at higher frequency, when defect is more. There is 5.9 dB (A) difference at peak when compared the faulty 100% complete tooth miss with healthy gear.

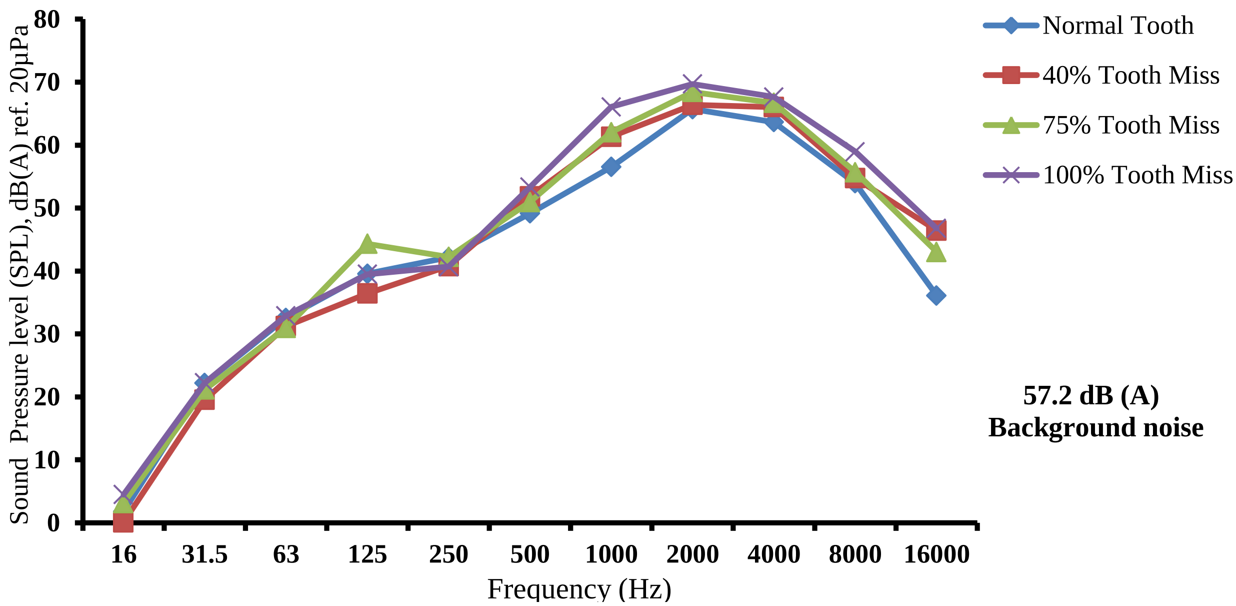

Figure 5.10: sound pressure level vs. Frequency at 540 rpm

Normal Gear from Fig. 5.10 it can be seen that at test gear speed 540 rpm, the SPL peak occurs at 2000 Hz with noise level of 65.7 dB (A). Fig. 10 indicates high noise level occur at frequency range 2000 to 4000 Hz. Peak at frequency 2000 Hz is annoying for human hearing.

40% Decrease in Height from Fig. 5.10 it can be seen that, the SPL increases as the fault is localized in the gearbox. The peak occurs at 2000 Hz with 66.4 dB (A) noise level.

75% Decrease in Height from Fig. 5.10 it can be seen that, the SPL increases as the further decrease in height of tooth in gear. The peak occurs at frequency 2000 Hz with 68.4 dB (A) noise level.

100% Tooth Removed from Fig. 5.10 it can be seen that, the SPL increases more as tooth is removed from the gear. The peak occurs at frequency 2000 Hz with 69.7 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 340 Hz after that there is sudden increase in SPL up to 1000 Hz, after that graph remains almost flat from the frequency range 1000 to 4000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. Tooth mesh frequencies occur at 325.91 Hz and 350 Hz which causes the sudden increase in graph after 340 Hz. There is not too much difference in the SPL at low frequencies of healthy gear and faulty gear but there is significant increase in SPL due to higher frequency contributions. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at higher frequency, when defect is more. There is 4 dB (A) difference at peak when compared the faulty 100% complete tooth miss with healthy gear.

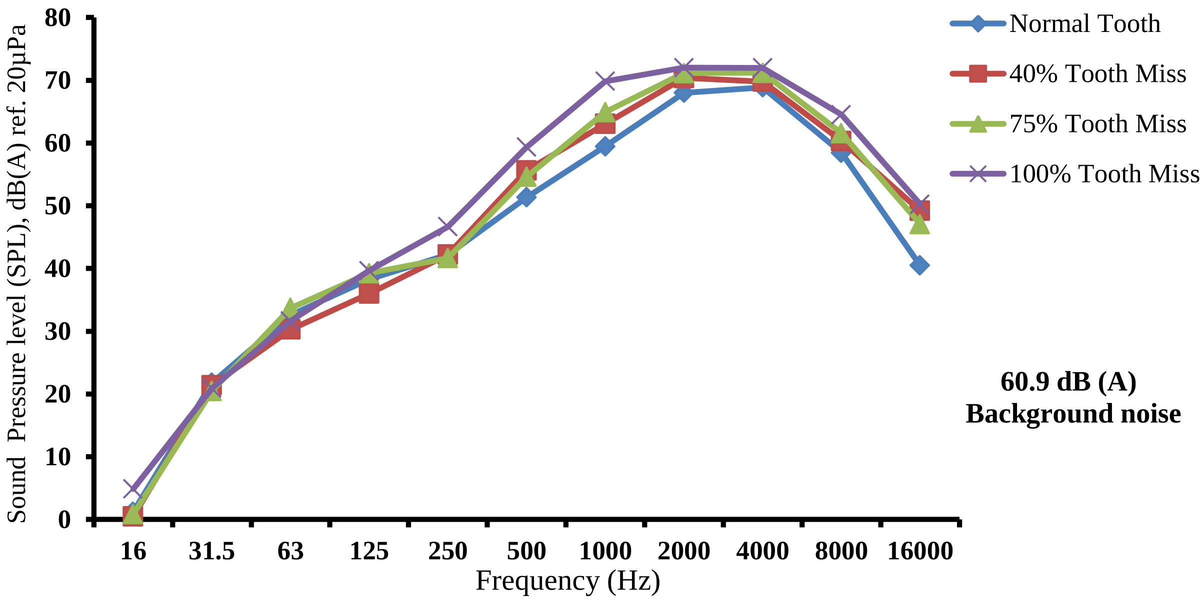

Figure 5.11: sound pressure level vs. Frequency at 725 rpm

Normal Gear from Fig. 5.11 it can be seen that at test gear speed 725 rpm, the SPL peak occurs at 4000 Hz with noise level of 68.9 dB(A). Fig. 11 indicates high noise level occurs at frequency range 2000 to 4000 Hz. Peak at frequency 4000 Hz is annoying for human hearing.

40% Decrease in Height from Fig. 5.11 it can be seen that, the SPL increases as the fault is localized in the gearbox. The peak occurs at 2000 Hz with 70.4 dB (A) noise level.

75% Decrease in Height from Fig. 5.11 it can be seen that, the SPL increases as the further decrease in height of tooth in gear. The peak occurs at frequency range 2000 to 4000 Hz with 71.2 dB (A) noise level.

100% Tooth Removed from Fig. 5.11 it can be seen that, the SPL increases more as tooth is removed from the gear. The peak occurs at frequency 2000 Hz with 72.0 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 400 Hz after that there is sudden increase in SPL up to 1000 Hz, after that graph remains almost flat from the frequency range 1000 to 4000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. Tooth mesh frequencies occur at 434.55 Hz and 466.66 Hz which causes the sudden increase in graph after 400 Hz. There is not too much difference in the SPL at low frequencies of healthy gear and faulty gear but there is significant increase in SPL due to higher frequency contributions. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at higher frequency, when defect is more. There is 3.1 dB (A) difference at peak when compared the faulty 100% complete tooth miss with healthy gear.

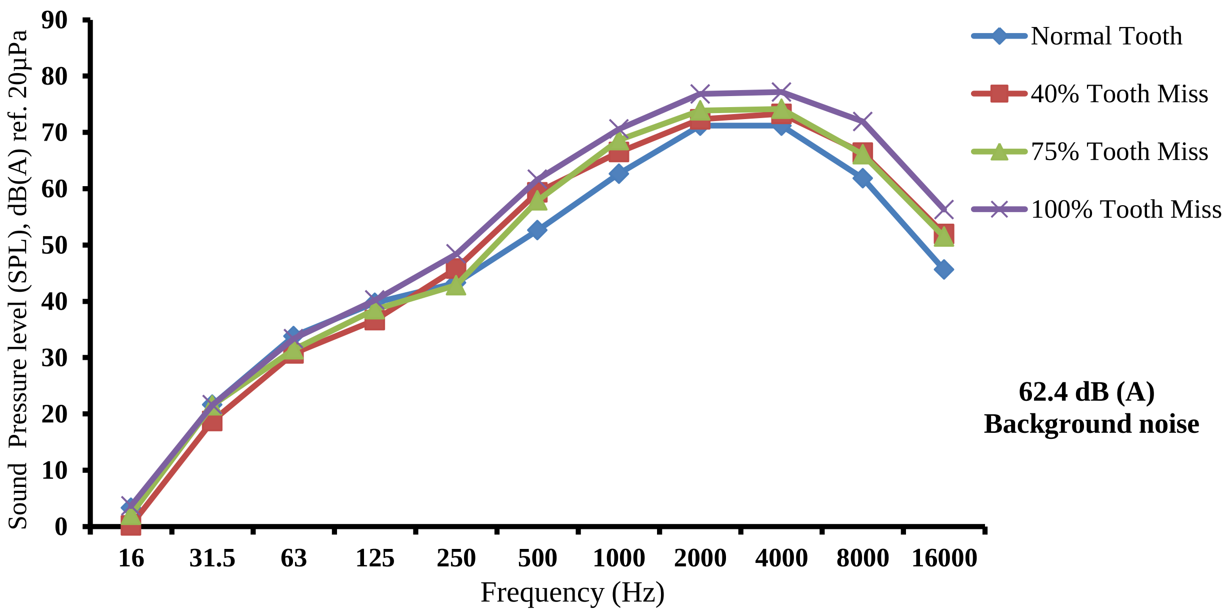

Figure 5.12: sound pressure level vs. Frequency at 900 rpm

Normal Gear from Fig. 5.12 it can be seen that at test gear speed 900 rpm, the SPL peak occurs at frequency range 2000 to 4000 Hz with noise level of 71.2 dB (A). Fig. 12 indicates high noise level at frequency range 2000 to 4000 Hz. Peak at frequency 2000 to 4000 Hz is annoying for human hearing.

40% Decrease in Height from Fig. 5.12 it can be seen that, the SPL increases as the fault is localized in the gearbox. The peak occurs at 4000 Hz with 73.3 dB (A) noise level.

75% Decrease in Height from Fig. 5.12 it can be seen that, the SPL increases as the further decrease in height of tooth in gear. The peak occurs at frequency 4000 Hz with 74.2 dB (A) noise level.

100% Tooth Removed from Fig. 5.12 it can be seen that, the SPL increases more as tooth is removed from the gear. The peak occurs at frequency 4000 Hz with 77.2 dB (A) noise level.

There is gradual increase in SPL from low frequency (16 Hz) to 475 Hz after that there is sudden increase in SPL up to 2000 Hz, after that graph remains almost flat from the frequency range 2000 to 4000 Hz. After this the level starts falling down in the higher frequency range due to the influence of structure resonance frequencies. Tooth mesh frequencies occur at 543.19 Hz and 583.33 Hz which causes the sudden increase in graph after 475 Hz. There is not too much difference in the SPL at low frequencies of healthy gear and faulty gear but there is significant increase in SPL due to higher frequency contributions. This shows that SPL increases with increase in fault dimension, which indicates max. sound level occurs at higher frequency, when defect is more. There is 6 dB (A) difference at peak when compared the faulty 100% complete tooth miss with healthy gear.

In all cases peak noise level occurs at frequency range of 1000 Hz to 4000 Hz. There is lower shift in frequency corresponding to peak noise level as crack size increases and an increase in peak noise level as tooth height decreases.

Chapter-6

Conclusions

In various researches, the fault diagnosis of rotating component gears using vibration measuring technique is the common old technique for detecting the defects in the gears of gear box. Acoustic emission is new technique and is growing considerably. It is more effective than the old classical vibration method techniques. Very few researches have been taken out on fault diagnosis of gears using microphone technique to measure sound pressure level. In the present study, effect of speed and localized defects in a gear tooth on the sound pressure level using sound level meter has been studied. On the basis of present study, following important conclusions have been made:

- The experimental procedure used in the present work is capable of fault diagnosis of gear box. A periodical impulse signal generated due to tooth root crack shows in frequency domain and in 1-1 octave band gives information about defect.
- Fault of gear box are identified by measuring the gear box noise in sound pressure level. It gives the actual state of gear box and gives the reliable results. The peak noise level at mid frequencies shows that as the crack depth increases the sound pressure level increases. Using this method and observing the graphs obtained after the study forecast of defects can be done.
- To study the propagation or development of defects, artificial defects of different types were imposed and noise was measured in term of sound pressure level. By comparing the healthy gears sound pressure level to the defected gear’s sound pressure level, information about defects can be extracted.
- As the height of tooth of gear decrease the sound pressure level increases.
- Sound pressure level also increases as the speed of gear increase.

6.1 Future Scope

- In the present study only two artificial defects (crack and decrease in tooth height) have been consider. Natural pitting, bending and wear can also be considered for further study.

- In the present study only three parameters speed, crack size and decrease in tooth height have been varied. For further study load variation, lubricating oil and multi hour test running can also be considered.
- In the present study no further advanced signal processing technique has been used. For the future work FFT and rms. value can also be considered.

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