Vibration Analysis of Gear Mesh using Finite Element Method

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Vibration Analysis of Gear Mesh Using Finite Element Method

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Declaration of Originality

I, *Mradul Mishra*, Roll Number 214ID1391 hereby declare that this thesis entitled *Vibration Analysis of Gear Mesh using Finite Element Method* presents my original work carried out as a research student of NIT Rourkela and, to the best of my knowledge, contains no material previously published or written by another person, nor any material presented by me for the award of any degree or diploma of NIT Rourkela or any other institution. Any contribution made to this research by others, with whom I have worked at NIT Rourkela or elsewhere, is explicitly acknowledged in the thesis. Works of other authors cited in this thesis have been duly acknowledged under the section “Reference”. I have also submitted my original research records to the scrutiny committee for evaluation of my thesis.

I am fully aware that in the case of any non-compliance detected in future, the Senate of NIT Rourkela may withdraw the degree awarded to me on the basis of the present thesis.

May 20, 2016

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Abstract

Gears are widely used in Machines and Mechanisms. One of the key advantages of gears is having very low fluctuation in speed transmission, but there are several drawbacks also in gears, vibration is one of the main difficulties of gears since load transmission fluctuation is too much high during the meshing of teeth. In this study, motion transfer has been considered between two Involute profile Spur gears mounted on two different shafts. The main objective of the study is to get the frequency of vibration at top node of all four clamps because clamps can be managed by shock-absorbers. One more objective of the study is to obtain the signals of vibration at the top nodes of all four clamps. The analysis is done by using finite element method. “Hunting Tooth Frequency” of gear box has been considered same as rotation frequency of gear box for analysis. Since there are various types of defect also may occur in gear tooth because the tooth is most force bearing elements in the gears, so tooth of gears generally affected by fatigue failures, which generally affect the transmission of motion and power in the gear box. Minor damage in gear teeth profile lead to high chattering, noise and vibration and fluctuation of the load at that particular failure point get increased and at one position, teeth get failed. So for analysis, this case also has been considered for simple spur gear box.

A defected tooth has been considered in driving gear. After that having almost same conditions, the analysis is done by the author for defected gearbox also. In results, the amplitude of acceleration at input shaft and at output shaft shown in both cases (with considering X, Y, Z components also) and the result is compared with using the graph with Amplitude of Acceleration and time. Results are verified by comparing “Hunting Tooth Frequency (HTF)” using the analytical method and single sided amplitude spectrum of the gear box.

Keywords: Hunting Tooth Frequency (HTF); Profile; Defect; Gear box.

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Nomenclature

r_1	Pitch Circle Radius of Pinion
r_2	Pitch Circle Radius of Gear
n_1	Number of teeth on Pinion
n_2	Number of teeth on Gear
N_1	Rotation Speed of Pinion
N_2	Rotation Speed of Gear
Φ	Pressure Angle
k	Fraction of Addendum
n	Number of Teeth
[M]	Mass matrix
[D]	Damping matrix
[K]	Stiffness matrix
$\{\ddot{a}\}$	Nodal acceleration vector
$\{\dot{a}\}$	Nodal velocity vector
$\{a\}$	Nodal displacement vector
$\{F(t)\}$	Load vector

Chapter 1

Introduction

A gear box is a medium of transfer of motion, power and torque etc. In this thesis analysis is done for obtaining the amplitude of vibration of gear mesh using finite element method, so a simple gear box is considered for analysis. The main components of gear box are gears, shafts, and supports. Gearbox consists some other supporting components also like bearings, lubrication oil etc. but these components are just for reducing friction, vibration, and noise. So for analysis ideal conditions are considered by the author. Two spur gears having an involute profile in mesh considered by author and shafts are considered simply supported on clamps for analysis.

1.1 Gear

Gears are widely used in mechanical heavy machinery and mechanisms, main motive of gears to transmit motion and power through parallel, non-parallel, intersecting and non-intersecting shafts. Gears provide speed at output shaft at very high accuracy so gears are mostly used where less factor of safety have to be used. Gears are just a rotating cylinder which has a tooth at the outer surface; the tooth is used for transferring speed and torque. The methodology of motion transfer in gears is similar as motion transfer through belt pulley system but here both pulleys are assumed to be in contact and transferring motion through their profile only. So the direction of rotation is different in gears.

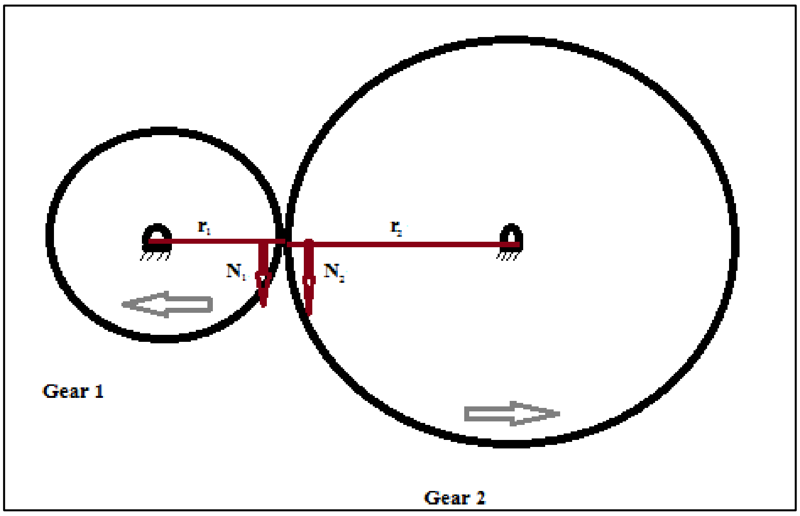


Figure 1.1 Law of Gearing

$$\frac{N_1}{N_2} = \frac{n_2}{n_1} = \frac{r_2}{r_1} \qquad 1.1$$

1.2 Classification of gears

There are various areas for the classifying different type of gears.

1.2.1 According to peripheral speed

1. Low Velocity Gears (peripheral speed range < 3 m/sec)
2. Medium Velocity Gears (peripheral speed range 3 to 15 m/sec)
3. High Velocity Gears (peripheral speed range > 15 m/sec)

1.2.2 According to position of axes of revolution

Spur Gear

Spur Gears are simplest types of gears; the projection of teeth on these gears is redial. These teeth used for torque transmission through parallel shafts.

Helical Gear

These gears are similar to spur gears but leading edge of teeth is not parallel to the axes of revolution, the angle makes the tooth shape a segment of a helix. Helical gears can mesh in parallel or crossed orientations.

Herringbone Gear

A Herringbone gear is similar to double helical gears, for overcoming axial thrust, herringbone gears have used these gears are having V Shape leading edge of teeth. These gears also used for power transmission through parallel shafts.

Bevel Gear

Bevel gears are used for power transmission through intersecting axes shafts, these gears can transmit power at all angles except 0 and 180.

Worm Gear

Worm Gears are used for transferring motion at very high-speed ratio, main motive of using these gears are to achieve higher torque, this type of gears consist a threaded shaft rotating at high speed and a Gear, motion transfer through these types of gears cannot be reverted.

1.2.3 According to profile of teeth

1. Gears with straight teeth
2. Gears with curved teeth
   a. Gears with cycloid profile teeth
   b. Gears with involute profile teeth
3. Gears with inclined teeth

1.2.4 According to Type of Gearing

Internal Gear

These gears are having teeth on the internal side of the periphery. Two internal gears can mesh together and size of external gear must be lower than the size of internal gear.

External Gear

These types of gears have teeth at the outside of peripheral. All above-given examples are examples of external gears.

Rack and Pinion Gear

These gears are used for convert rotational motion into translation motion. An external gear rolls on a rack having infinite no. of teeth.

Sun and Planet Gear

In Sun and Planet gear system, generally one gear is fixed and other gears rotate on it with using an arm or some another gear. In these types of gears, axes of gears also rotate.

1.3 Vibration

When a body is displaced from equilibrium position by application of forces, then elasticity of body forces to bring it in original position, When the body reaches the equilibrium position, the whole of the elastic strain energy converted into kinetic energy due to this body continue to move in opposite direction and at a position came when whole kinetic energy converted into elastic energy and body again start to move into equilibrium position. This motion is called as vibration, and this path of the body is repeated indefinitely. Vibrations can be classified into following types.

1.3.1 According to Force applied

Free Vibration

When a mechanical system displaced from its equilibrium position and allowed to vibrate freely then this type of vibration is called as free vibration or natural vibration. Free vibration can be classified into three types

Forced vibration

When a mechanical system moves displaced from equilibrium position again and again due to an external force applied periodically then this type of vibration is termed as forced vibration.

1.3.2 According to direction of vibration

Longitudinal vibration

When the direction of motion of vibration is in the direction of long axes of the body then this type of vibration is called as longitudinal vibration.

Example- An oscillating spring mass system

Transverse vibration

When the direction of linear vibration is perpendicular to the long axes of the body then this type of vibration is termed as transverse vibration.

Example- An oscillating pendulum

Torsional vibration

When a body starts rotation along its rotation axes, then angular vibration in the rotation is called as torsional vibration.

Example- Cam.

1.4 Objective

There are several problems encountered in gears, the author has considered spur gears and a simple gear box for vibration analysis. Generally, axial vibration encountered in clamps due to gears, but in study linear vibration is considered in X, Y, Z directions. The main objectives of this study are:

1. To study on problems encountered in gears.
2. To carry out signals of vibration on the clamps having one simply supported shafts at each pair and shafts are having two spur gears in mesh on the middle of the shaft.
3. To examine failures in gears.
4. To carry out indications of vibration at clamps due to failure.
5. To verify the results of analysis by comparing tooth hunt frequency obtained analytically and by using amplitudes of acceleration obtained from finite element method.

1.5 Methodology

The overview of the subject and the status of the current trends as well as the availability of recent technology and considered for achieving the broad objectives of the present research work. While the details of the word and method used are present in chapter 3, the following lines attempt to give a brief idea about the methodology.

Optimization of parameters

The first step after the objective is to define parameters, optimization of parameters is necessary for balancing accuracy of results and reducing the size of analysis. This topic has been discussed in details in chapter 3.

Geometry

After completing detail design of parameters, firstly modelling was done in CATIA V5R19 with assembly of 6 bodies considering 4 clamps, 2 shafts with gears, but after facing problem due to inaccuracy of shaft periphery, entire geometry were drawn out in ANSYS 15.0 Geometry Modular using 3D primitives, only teeth profile of gear were drawn by sketching rest all geometry and assembly was drawn out using 3D primitives.

Finite Element Analyses

For finite elements analyses, Mechanical Workbench of ANSYS 15.0 software is used. After applying boundary conditions and proper distribution of nodes and elements, analyses is done with considering steady state of outsource effects. This topic has been discussed in details in chapter 4.

Defected Gear Box Analysis

After completing analysis for gear box without any mechanical/geometrical defect, a defect has been considered for achieving fourth and fifth objectives. This topic has been explained in chapter 3 and chapter 4.

1.6 Motivation

In today’s era of industrialization, the scope of using of gears is increasing at a rapid rate. Gears are nothing but improved power transmission system of belt pulley system, but the main advantage of the use of gears is zero slip. Friction losses also too much less in gears because theoretically there is only line contact during transferring of torque but practically also area of contact is too much less compare to other transmission systems, the other advantage of gears is long life, since gears do not have any flexible elements so generally materials are considered for gears having high hardness and toughness. So the life of gears is also more than other motion-transferring machines. In practical life, since there is variation in the axes of transferring force through teeth so there are several unnecessary forces also get generated due to the reaction, these forces become the cause of vibration in shafts. Since vibration can be reduced but cannot be neglected, estimation of vibration can also reduce the factor of safety during the design of heavy machinery such as aircraft, turbines. So author found it as objective to define indications of vibration in gears.

Since vibration put an effect on the whole dynamic system so there are chances for become several types of failures, fatigue failures are the main problem of gear teeth because teeth are most force bearing elements in the mechanism. Little damage of gear becomes the cause of unnecessary vibration, noise, and chattering. Sometimes gears are used at such places where routine checkup increase the maintenance cost very high. So identification of vibration due to crack can reduce the maintenance cost of a machine consisting various types of gears. So author gets one more objective for research work.

1.7 Summary

It seems a good logic that study of this work should start with the broad introduction of the problem; the chapter 1 in this thesis gives introduction and scope of the problem as well as the need of scope of the need of objective of this work. In chapter 1 brief introduction of the solution is also presented. Gears come in existence for human in the 13th century than after several types of improvements done in gears. Introduction to types of gears and vibrations encountered in gears has been discussed in this chapter and chapter ends with Objective, Motivation, and initiation of theses.

Chapter 2

Literature Review

There were several modifications done in gears in recent years but few problems of gears are yet to be sorted out.

Faker Chaari *et al.* validated formula for time varying stiffness of gears by using FEM analyses and analytical solution [1]. An original analytical modelling of tooth cracks is presented and the gear mesh stiffness reduction due to this fault is quantified. A comparison with finite element model is presented in order to validate the analytical formulation.

Zhonghong Bu *et al*. developed the analytical formula for vibration in a herringbone gear mounted on a bearing having 4 factors of stiffness and he analyzes 6 factors of vibration using the analytical method [2]. The proposed dynamic model and analysis methods can be applied to HPGT with any number of planets. Only when the asymmetric interaction exists in journal bearings, will the dramatic change of mode shape for translational mode occur. The new relations between deflections of planets in translational mode are also derived in this research.

M. Masjedi *et al*. predicted wear rate of spur gears running in steady state [3]. The wear results are also compared to those obtained by an engineering approach that utilizes an estimate of the contact flash temperature with reasonable accuracy. Also investigated is the effect of starvation on the wear rate for applications where the inlet is not fully flooded. The results show that wear rate can be enormously large depending on the degree of starvation.

William D. Mark *et al.* formed analytical results for averaging of transducer response of vibration of gear mesh in a time synchronous, so that mating tooth can be eliminated based on results [15]. The tooth-meshing-harmonic contributions of the transmission error are caused by the deviations, from equally spaced conjugate surfaces, of the average elastically deformed working surfaces from each of the two meshing gears, where these averages are formed by the average over all teeth on each gear. The non-tooth meshing rotational harmonic transmission error contributions, from each gear of a meshing pair, are caused by deviations of the individual elastically deformed tooth working surfaces from the above-described average elastically deformed working surface.

Houjun Chen *et al.* compared the equation result with a computerized simulation of hyperboloid-type normal circular arc gears [4]. According to local conjugate theory, the point-contact design method for HNCGs was presented, and the equations of pinion-gear tooth surfaces were deduced. Using the moving frame method, the errors and the variations were quantified, and the error-variation equations were established.

Pedro MT Marques *et al.* carried out a work on the power loss due to friction in Spur gears and Helical Gears using finite element analysis method [10]. The aim of this work is to intro-duce gear load sharing models for spur and helical gears taking into account elastic and frictional effects allowing to doing more refined estimations of gear friction losses.

Siang Yu-Ye *et al.* compared the equation result with a computerized simulation of high contact ratio spur gears in mesh with considering tip contact and shaft misalignment [11]. The contact characteristics of a spur gear pair with a high contact ratio are studied. The gears have a high tooth profile and are paired as a negative profile-shifted drive. The contact stress and the load sharing of the gear pair are analyzed with consideration of the misalignment of the shafts and the flank modification. The flank modification consists of lead crowning and linear tip or root relief.

S. Sjoburg *et al.* worked for calculating mesh efficiency of two spur gears in mesh using experimental method; he also drew the least fit graph between mesh efficiency and pitch speed [12]. In this paper an FZG gear test rig was used to investigate how two running-in loads affect the gear mesh efficiency for two different temperatures. The gear surface roughness was investigated in parallel with testing. Higher efficiency was observed for tests using a high running-in load, and for low lubricant temperatures.

2.1 History

Gears have been used by human's long time ago as a name of a cogwheel, around 200-300 BC, oldest use of gear is preserved, but cogwheel mechanism was invented a long time ago. Around 200 BC Ma Jun invented a wood carriage using gear as a part of this carriage. Around 800 BC, first mechanical clock was invented in China, mechanical clock was having gears as main parts and all adjustment of gears done it, around 1300BC, a mechanical clock invented, which is working till today name of that clock is Salisbury cathedral clock.

In the 17th century, gears were used first time with a conjugated profile for getting the exact ratio. These gears were used in vehicles at that time. In 1835, the first time a gear used in the mechanical machine, in England Gear were used in shaper as a component, then after use of gear boxes increased rapidly. Up to mid of 19th century, gears became part of mostly rotation transfer mechanisms in 1897, Gears were started using with cam also, first time Gear cum Cam arrangement was used in hobbing machine. The hobbing machine was the first machine for cutting teeth of helical and spur gears. Then after in the 20th century, Gears become most reliable part for motion transfer and in 1975 an NC Hobbing the machine was invented in Germany. Then after in 1982, NC machine modified to 6 axes and all types of gears production was started with automation. In recent days, gears are widely used in Mechanical industries, Automobiles, Aircrafts etc.

2.2 Selection of Material

Table 2.1 Selection of materials by authors in Finite Element Analyses

Reference No.	**Author's name**	**Title**	**Material**	**Year**
17	M. Ramesh Kumar	Finite Element Analysis of High Contact Ratio Gear	Structural Steel	2010
8	Ali Kami Zabur	Numerical and Experimental Dynamic Contact of Rotating Spur Gear	Structural Steel	2011
9	Vivek Karaveer	Modeling and Finite Element Analysis of Spur Gear	Grey Cast Iron, Stainless Steel	2013
18	Shai Bharath	Design and Analysis of High-Speed Helical Gear By Using ANSYS	Stainless Steel	2013
19	Sachindra Kumar	Transmission Error Analysis of Involute Gears in Mesh by Fem	Structural Steel	July 2014
20	Singireddy Ravindra	Design and analysis of Gear Shaft	Aluminum Alloy	September 2015

Above mentioned papers are those which were studied by me for considering the material of Gears, Shafts, and clamps. Since most of the authors used Structural Steel as the material of gears because it is default material in ANSYS but few authors used Grey Cast Iron, Aluminum Alloy and Structural Steel for analyses using Finite Element Method by using ANSYS software. After getting data of strength, fatigue and density, it was optimized that best material for analysis will be stainless steel for gears and shaft and clamps will be of Structural Steel.

2.3 Selection of Parameters

Author studied several papers for getting most commonly used parameters for designing of gears, parameters considered in few papers explained below.

M. Rameshkumar *et al.* considered parameters for analyses are standard addendum and dedendum [17]. He considered two same size involute profile gears having a number of teeth are 50 on each He considered face width of 18mm and pressure angle is 20 degree.

Omar D Mohhmed *et al.* considered parameters for analyses are number of teeth on pinion is 25 and on gear is 30, Module for both is 2mm and teeth width is 20mm, contact ratio is 1.63 and pressure angle is 20, Young Modulus of material is 2×10^5 and Poisson ratio is 0.3 .[16]

Vivek Karaveer *et al.* considered some different parameters for analysis he considered a number of teeth on both gears is 20, pitch circle diameter was 127mm, Pressure angle was 20 degree, Addendum Radius was 69.85mm, Deddendum radius is 55.88mm, face width was 25.4mm and shaft diameter was 31.75mm. [9]

These papers give some idea for estimating parameters for design since the problem was different from these papers so these parameters gave just some rough idea about designing of geometry. Actually, the author needs to optimize the accuracy and number of elements also for calculation and analysis. Author mainly adopted addendum fraction and deddendum of gears from these papers.

2.4 Problem Identification

In this section, author has presented problems which were considered by other authors for analysis of gear box.

Ji Wang *et al.* analyzed a gear system having frictional effect as well as backlash [5]. Using this analysis, the physical phenomena associated with the friction and non-friction gear system are compared and the essential differences in the system behavior are examined. Finally, bifurcation, chaos and their corresponding largest exponents of gear system with sliding friction are investigated and the critical parameters are identified.

DP Jena *et al.* analyzed for identifying a defect in already meshed gear and angle between two defected teeth [6]. When defected tooth came in mesh with another tooth, amplitude of acceleration become high, Author presented the experimental result for one rotation of defected gear using angle vs. amplitude of acceleration graph. Angle between two peaks is the actual angle between two defected teeth. Author also presented the results for amplitude of acceleration using time domain and for verifying results, author presented a graph in frequency domain and results were matched with “Hunting Tooth Frequency (HTF)”.

Omar D. Mohammed *et al.* has done experimental analyses for vibration signal due to gear fault diagnosis with various crack progression scenarios [16]. The time-varying gear mesh stiffness has been investigated using the program code developed in this paper, and the crack propagation can be modeled with any of the presented crack propagation scenarios. Dynamic simulation has been performed to obtain the residual signals of all the studied cases for each crack propagation scenario.

Xihui Liang *et al.* has evaluated an analytical formula for dynamic analyses with considering the effect of crack in gear, he compares his result of analytical formula with the result of the experiment [13]. A modified cantilever beam model is used to represent the external gear tooth and derive the analytical equations of the bending, shear and axial compressive stiffness. A crack propagation model is developed and the mesh stiffness reduction is quantified when a crack occurs in the sun gear or the planet gear.

Hui Ma *et al.* has been analyzed on Helical, Spur and planetary gears for observation of crack propagation prediction, time-varying mesh stiffness (TVMS) calculation, and vibration response calculation in various modelling methods named as an analytical method, Finite Element method, and combined analytical-FE approach [7]. The research objects mainly involve the spur gear and planetary gear pair where the dynamic models mostly focus on lumped mass models and finite element models. For the reduction gearbox, the lumped mass model is widely adopted because the support including the shafts and bearings is rigid.

Vincent Savaria *et al.* did experimental analyses of bending fatigue strength of induction hardened aeronautical gear [14]. In the results, he verified that residual stresses can have a significant impact on bending fatigue and that two induction treatments can present very different fatigue resistance even if the shape and depth of the hardened layer is identical in the root. The proposed methodology of author could be adapted to other geometries and surface treatments.

So after studying these papers, Author reached on target that most work is carried out by using the analytical or experimental method. Finite Elements method is used for some analysis but that too used for calculation of static functions like stress, strain etc. For a steady state running gear box, there are several dynamic functions also, so author identified his problem, He found out his goal to determine the acceleration probe at the nodes of Input and Output shafts.

2.5 Research Gap

After studying these journal papers, the author came to be knowledged that most work has been done on gears and gear boxes are using the analytical method only. Several experimental works is also carried out by different authors but a Finite Element method has not used for steady state analysis for calculating some dynamic characteristics of gears. Some authors worked on both experimental and analytical work for defining characteristics of vibration, sound, crack signals etc.

Finite Element method is used for static characteristics only and this is also verified by using other methods by authors.

2.6 Research Objective

Small defects in gears are a normal problem in practice since the involute profile is one of the tough profiles for manufacturing so all gears might not be homogeneous, so there might be some defects like backlash; tearing, interference etc. are possible in the gears. Gears are assumed for one of the accurate speed and power transmission through shafts but these problems create obstructions in the transmission so analysis of a model having one defected teeth is required.

Analysis of defined problem is to be done in ANSYS 15.0 so firstly a model is required in ANSYS, then after element distribution has to be done in proper way on model then after amplitude of acceleration at the supports of input shaft and at the supports of output shaft to be solved out. A defected gear means one tooth of that gear has failed partially or completely. For verification of results, amplitude of acceleration equation in the function of time has to be converted into function of frequency and results will have to be verified using “Hunting Tooth Frequency”.

Chapter 3

Geometrical Modelling

Geometry modelling is one of the important tasks of designing a gear. First thing is to overcome the initial problems of gears then after design for reduction of such characteristics which decreases the durability of gears. Initial problems are those problems which are necessary to be shorted out for revolution of gears, Interference and undercutting are most common problem in these, another think is considered about parameters, which does not put so much effect on experimental or analytical analyses but put too much effect on Finite Element method, so author have to notice this also for geometry modelling.

Selection of software also give important role in geometry modelling, Author thinks about using CATIA V5R19, SOLIDWORKS 12 and ANSYS 15.0 Geometry Modular for designing, CATIA is a good option for visualizing simulation of undercutting and interference because ANSYS 15.0 consume too much time for verification of avoiding interference and undercut. ANSYS 15.0 give more accurate results compare to CATIA V5R19 because CATIA V5R19 give just visualization but ANSYS 15.0 give actual situation so firstly author checked on CATIA with a design for checking Interference and undercut and then after encountering some failures, author matched the analytical results for avoiding interference. Then after same geometry was tried by the author in ANSYS 15.0 but there again failure encountered because IGES file was not accurate.

IGES files have an accuracy of 1mm. So author takes the decision of doing geometry work in ANSYS 15.0 Geometry modular. Since accuracy was need of time in ANSYS also because CATIA gave information of primary problems only. So author used MATLAB R12 for defining coordinates of Involute profile teeth then after Author used 3D primitives only for designing of gear except Involute profile.

3.1 Literature Review

As mentioned in chapter 2, in mostly papers geometry was directly sketch out in ANSYS 15.0/13.0 Geometry modular only. But for static analyses it does not matter so much, even interference or undercut problems do not come behind, so author just skipped to follow these papers and used some analytical solution papers, There It appeared that Involute profile used, for defining parameters, several papers were studied as mentioned in chapter 2.

3.2 Optimization of Parameters

With the help of literature papers, parameters have to be optimized for analysis in a first way because the role of parameters highly depends on calculations as well as for further results; Optimization of parameters is a necessary step for balancing between accuracy of results as well as time and effort in analysis. Standard values also have to be in scope with both conditions so before starting analysis work, the author did optimization of few parameters.

3.2.1 Gear Design

In gear design, parameters are estimated for designing of gears, all parameters are related with teeth profile of gears and these parameters are necessary to be checked for avoiding chances of interference and undercut.

Pitch Circle Radius

Pitch Circle radius directly depend on the size of gears, since the range of size of gears is too much wide, gears used in wrist watch as well as clock tower so Pitch Circle Radius is taken as 12mm. The size of both teeth is assumed same for further analyses.

Number of Teeth

Since the majority of nodes and elements marked in teeth only because teeth are most effective parts of gear so a number of teeth have to be reduced to a least possible value. In past analyses number of teeth was higher but for this analysis author considered just 12 teeth per gear, another reason of less number of teeth is standard value of module of gear, the module is taken as 2 for next step analysis.

Pressure Angle

Pressure angle has a major role in avoiding interference, undercut of teeth, it does not put so much effect on gear size but it plays an important role in backless also. Standard value of pressure angle is 14.5, 18, 20 degree, but sometimes other integer values also used

For avoiding condition of interference and undercut, gear has to fulfill the condition given for two similar gears in mesh,

$$\sin^2 \Phi > \frac{4k(n+k)}{3n^2} \tag{3.1}$$

Here n = 12 and k= 1

After solving above equation minimum value of Φ is 20.3 degree, so for analyses pressure angle is considered as 25 degree

Nose Radius

Theoretically, there is no nose radius for an idle gear but in real life for avoiding fatigue effect on the top notch of gear, a fillet is necessary. Increasing fillet radius means affecting involute profile so fillet radius was taken as 0.25mm.

Clearance Radius

Clearance fillet is done on the flank of teeth and deddendum of teeth so without affecting involute profile, Clearance radius does not put so much effect on the rotation of gear main phenomena of clearance radius is to provide bending strength to teeth. Clearance radius was taken as 0.75mm.

Gear Width

Gear width is a parameter that directly proportional to its weight. Increasing gear width will increase the number of nodes and elements but too much less gear width will become the cause of low strength of gear so for stainless steel material Gear Width is considered as 5mm.

3.2.2 Assembly Design

Assembly design consists the location and designing of other components of the gear box. After completing designing of gears, other remaining components are shafts and clamps, shafts are considered attached with gears in analysis but the actually geometry of shafts have been optimized separately because the function of the shaft is different from the function of gear.

Shaft Diameter

Shaft diameter should be higher for avoiding bow effect in the shaft but should be less for reduction of nodes and elements, so shaft diameter is taken as 8mm.

Distance of clamps from gears

More distance means higher chances of bending of the shaft so just 5 mm distance is taken between gear and clamps. The clearance between the hole of clamp and shaft was taken as 0.

Distance between gear centers

Since author assumed a frictional contact between teeth of delivering and receiving gears so some clearance is necessary between both gears, the author considered 0.01mm clearance between both gears so the total distance between centers of gears is 24.01mm.

3.2.3 Defect Drawing

As mentioned in literature papers, here author drawn out a defect in gear, in practice material of gears is not homogeneous so there are chances of some small cracks near the tip of the tooth. Due to fatigue effect, this little crack become increase after cycles, this defect increases and become the cause of failure of the tooth. If one tooth gets failed partially or completely then it simply becomes a cause of noise and high amplitude vibration. So author tried for initiate signals of defect as vibration indications. A partially failed tooth is considered which comes in contact with other teeth. This crack was drawn out by using 4 lines and cut material command in ANSYS 15.0 Geometry Modular.

3.3 Drawing

Sketching is carried out for teeth profile only. For getting coordinates of the involute curve, MATLAB R12 software was used, then after using ANSYS 15.0 Geometry modular, 44 points were pointed out and most fit curve passed by these points for getting an involute profile of tooth. In figure 3.1, Involute curve for teeth profiles compiled in MATLAB with using parameters optimized. Distance on both axes has been taken from perpendicularly from the center of the gear. Figure 3.2 is presented here for a comparison of teeth profile because, in Figure 3.1, the involute curve has drawn out analytically using an infinite number of points but Figure 3.2 is drawn out manually using 44 points, and analysis highly depends on the profile of gears. So it is necessary to a reduction in error in the involute profile of gear.

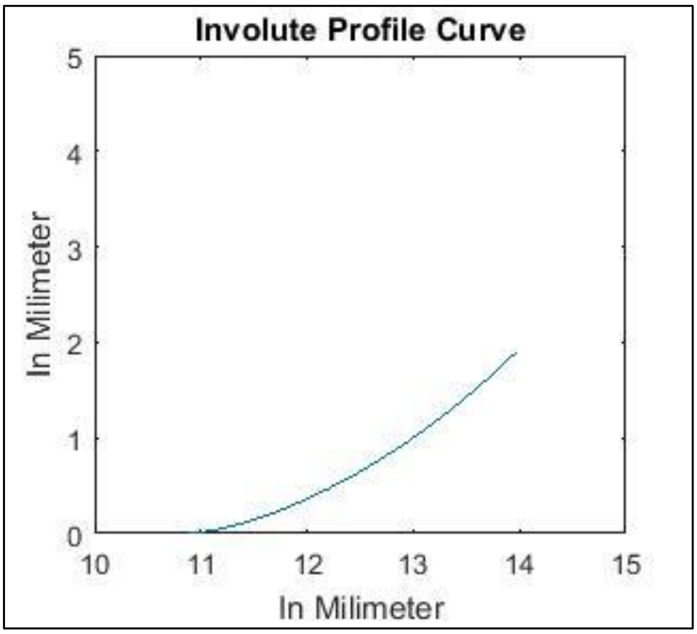


Figure 3.1 Involute Profile in MATLAB

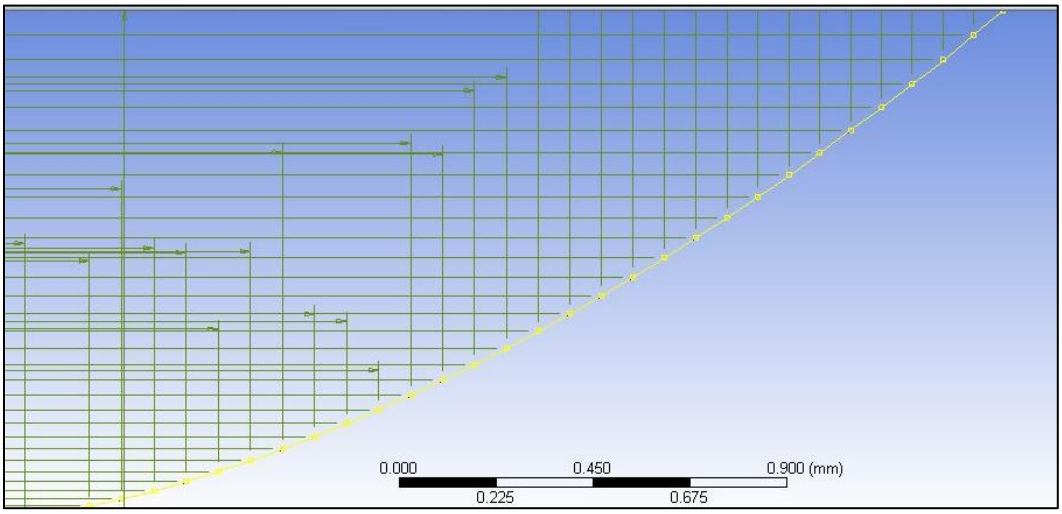


Figure 3.2 Profile of Gear teeth in ANSYS 15.0 Geometry Modular

Then after with the help of sketching, a close loop was created for half tooth, a half face of teeth is extruded to 0.5 cm because the width of gear is 0.5 cm only.

1. Then after teeth are mirrored through a flat surface.
2. Then after it is rotated to 12 patterns using a center radius of 14 mm from out edge of teeth.
3. Then after center plate cum cylinder to be carried out.
4. Then after the shaft is drawn out using again 3D primitive.
5. After that 1 clamp on the end of the shaft mounted using a thin cylinder and box 3D primitives.

3.4 Assembly

1. A duplicate gear is carried out on 24.02 mm difference. 0.02 mm is clearance taken out.
2. After that clamp was selected and rectangular pattern of clamp created so that now both gears with shafts can be simply supported on clamps.
3. Then after the second gear was rotated by some angle for avoiding overlapping with gear.

In Figure 3.3, a picture of gear box shown. Left side gear has been considered as driving gear and that gear having a defected tooth (2^{nd} just before mesh if the gear is rotating clockwise). In Figure 3.4, front view of gear box has shown which is not having any defect.

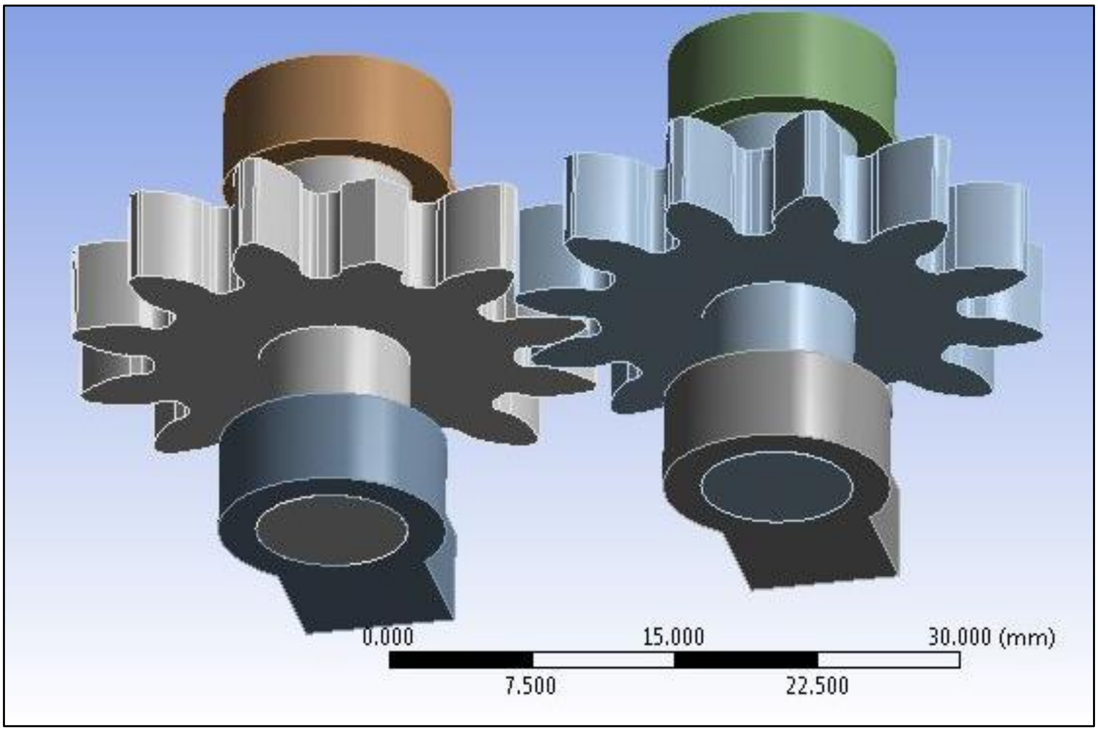


Figure 3.3 Assembly of gears with defect

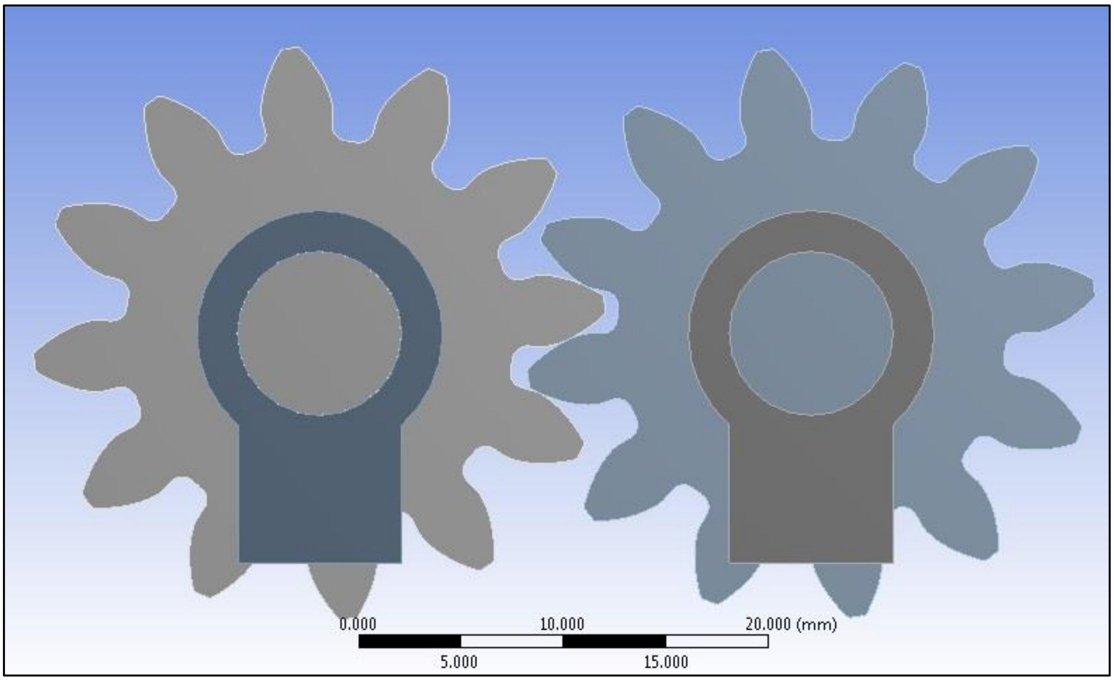


Figure 3.4 Front view gear box without defect

In Figure 3.5, side view of gears tilted a little bit so the arrangement of clamps can be seen easily. In figure 3.6, teeth are shown in mesh, the effect of clearance of 0.02mm can be easily seen in Figure 3.6, Figure 3.7 is showing the profile of partially failed teeth, one tooth is considered as partially failed for one analysis and it will not mesh completely with another tooth.

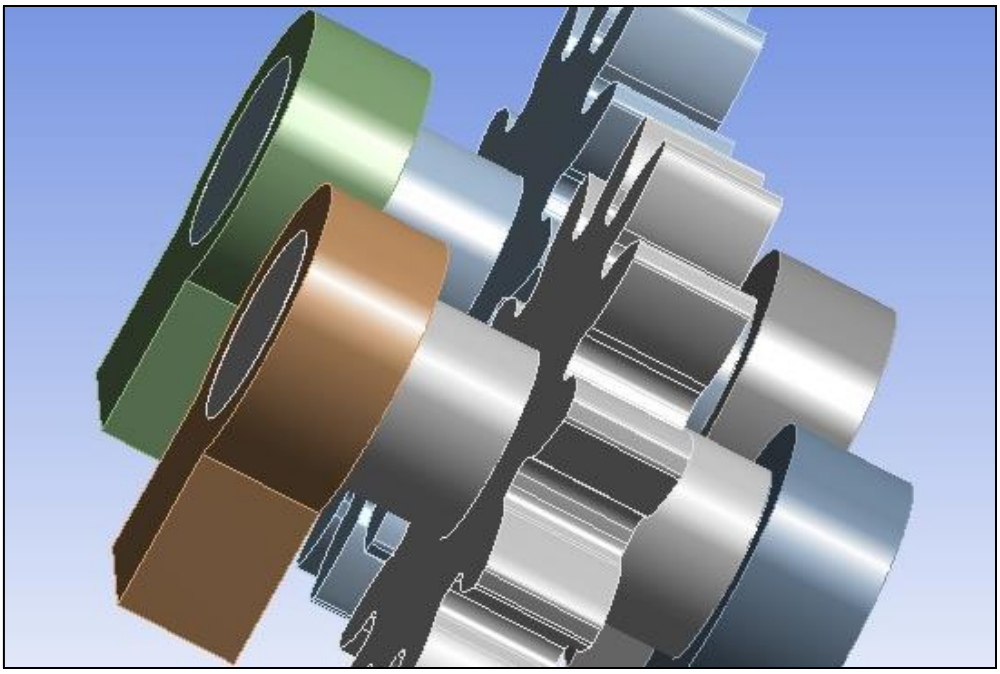


Figure 3.5 Clamps of arranged gear system

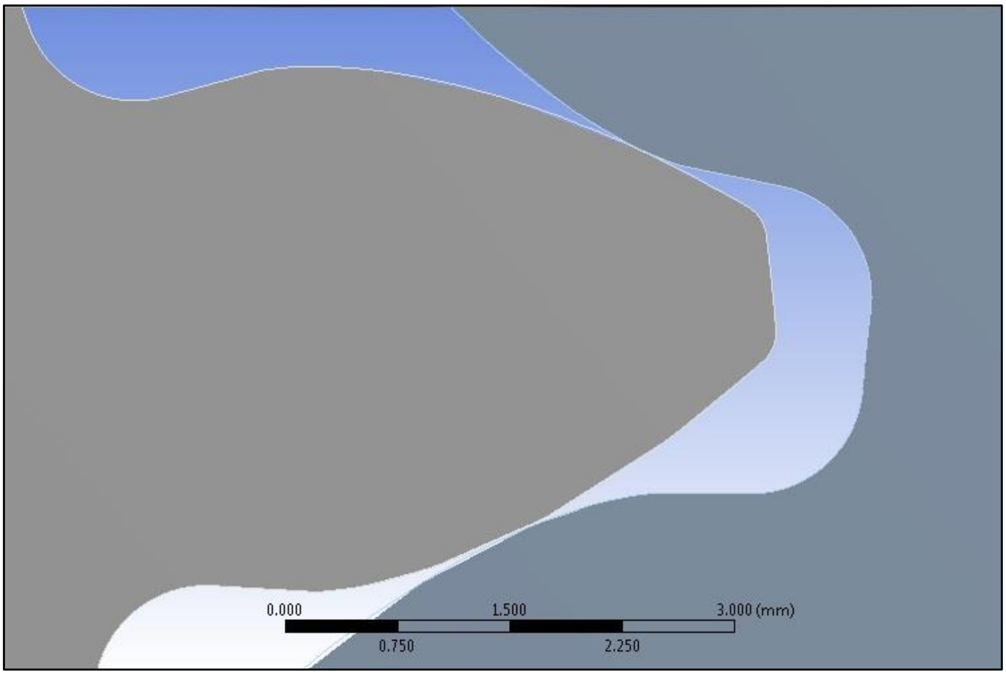
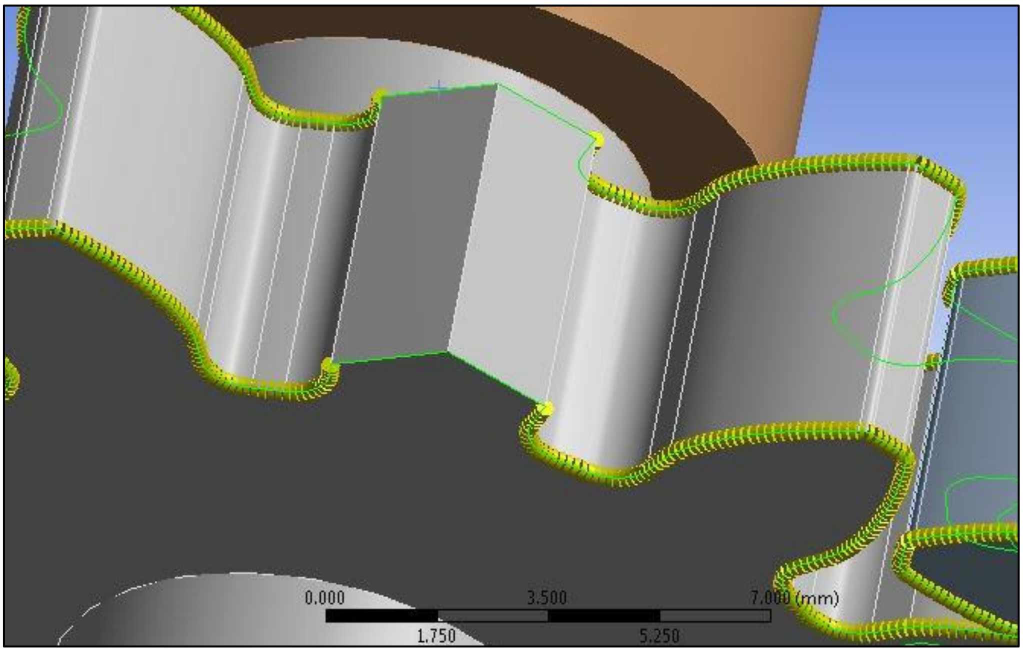


Figure 3.6 Profile of teeth after assembly



0

Figure 3.7 Failure is shown in defected teeth

3.5 Boundary Conditions

After completion of a mechanical model, elements distribution is to be done in an optimized way so that accuracy of the involute profile can be used. After that, input speed is provided on the shaft of gear 1.

Contact between gears is considered with a coefficient of friction as 0.005, Shaft and clamps contact region were assumed as a frictionless contact region.

3.6 Conclusion

So in this chapter modelling of the system is completed using ANSYS 15.0 Geometry Modular. MATLAB program data increased the accuracy of the geometry since this geometry was also not 100% perfect but in practice, these parameters are considered for higher accuracy of the involute profile, till now no machine has been invented for giving exact involute curve for gear profile. This chapter consists optimization of design parameters, the methodology of designing and this chapter ends with assembly and contact region properties of all component of gear box system.

Chapter 4

Finite Element Analyses

In this chapter, Mechanical Model of ANSYS 15.0 Workbench was used for analysis. Firstly degree of freedom needs to be defined, for a simple gear box mechanism; the degree of freedom must be one. So firstly author considered for the degree of freedom then after author focused on the mesh, Mesh plays an important role for Mechanical Modelling, so author complete optimized mesh with minimizing unnecessary nodes and elements and an increasing number of elements for improving the accuracy of results. The author also considers the significant of results, so author considered for results and accuracy of results also defined in this chapter.

4.1 Degree of Freedom

For this work, a simple gear box is considered which is having just 2 gears. So the degree of freedom must be only one. The degree of freedom being less than one makes it a structure and more than one degree of freedom means more than one output on one input. For making the complete set as one degree of freedom, there were several joints, fixed link have to be chosen and if the degree of freedom become less than one then joints, fixed links have to be reduced.

4.1.1 Analyses of Degree of Freedom

The geometry setup which has drawn out for analysis is having 6 independent parts, the system consists 2 gears (with shafts) and 4 clamps. Every part have 6 degrees of freedom so the degree of freedom of complete system is 36 if there is no constraint or joint. But clamps are assumed to be fixed with the ground so 24 Degree of Freedom restricted by clamps itself. Then after clamps are having cylindrical joints with shafts but here 2 cylindrical joints are redundant, every cylindrical joint constraint 4 Degree of Freedom so remaining degree of freedom is just 4.

4.1.2 Restriction of Degree of Freedom

So Geometry model is having 4 degree of freedom, 1 DOF is restricted due to higher pair between both gears and 2 DOF have to be restricted be providing a load on shafts.

4.2 Transient Dynamic Analysis

The transient dynamic analysis is a technique used to determine the dynamic response of a structure under the action of any general time-dependent loads. This analysis is also known as time-history analysis. This analysis can be used for determining the displacements, strains, stresses, and forces varying with time in a structure as it responds to any combination of static, transient, and harmonic loads. The time scale of the loading considers the effect of the inertia or damping.

The basic equation of motion solved by a transient dynamic analysis is

$$[M].\{\ddot{a}\}+[C].\{\dot{a}\}+[K].\{a\}=\{F(t)\} \qquad 4.1$$

The procedure for a full transient dynamic analysis (available in the ANSYS/Mechanical, and ANSYS/Structural products) consists of three main steps:

1. Build the model.

2. Apply loads and obtain the solution.

3. Review the results.

For analysis of transient response damping matrix may be used. It can be represented in the following manner:

$$[C]=\alpha[M]+\beta[K]+\sum_{i=1}^{N_{ma}}\alpha_i^m[M_i]+\sum_{i=1}^{N_{ma}^{MD}}\sum_{k=1}^{N_{sa}}\alpha_p[M_k]_i+\sum_{j=1}^{N_{mb}}\beta_j^m[K_j]$$

$$+\sum_{j=1}^{N_{ma}^{MD}}\sum_{n=1}^{N_{sb}}\beta_q[K_n]_j+\sum_{k=1}^{N_e}[C_k]+\sum_{l=1}^{N_g}\beta_q[G_l] \qquad 4.2$$

Where,

$[C]$ = Structural damping matrix

α = Mass matrix multiplier

$[M]$ = Structural mass matrix

β = Stiffness matrix multiplier

$[K]$ = Structural stiffness matrix

N_{ma} = Number of materials with MP, ALPD input

α_i^m = Number of materials

$[M_i]$ = A portion of structural mass matrix based on material *i*

N_{ma}^{MD} = Number of elements with mass proportional material damping input

N_{sa} = Number of sections in an element with mass proportional material damping input

α_p = Mass proportional material damping factor for section point *K* with material *P*

$[M_k]_i$ = A portion of element *i* structural mass matrix based on section *K*

N_{mb} = The number of materials with MP, BETD input

β_j^m = Stiffness matrix multiplier for material *j* (input as BETD on the MP)

$[K_j]$ = A portion of the structural stiffness matrix based on material *j*

N_{ma}^{MD} = The number of elements with stiffness proportional material damping input

N_{sb} = The number of sections in an element with stiffness proportional material damping input

β_q = Stiffness proportional material damping factor for section point *n* with material

$[K_n]_j$ =A portion of element *j* structural stiffness matrix based on section *n*

N_e = The number of elements with specified damping

$[C_k]$ = Element damping matrix

N_g = The number of elements with Coriolis or gyroscopic damping

$[G_l]$ = Element Coriolis or gyroscopic damping matrix

4.3 Mesh Details

Mesh is nothing but pointing out nodes and elements for analysis, less number of nodes and elements will be reason of less accurate results but high number of elements will consume too much time for calculation so for an optimized mesh, more elements should be near result affecting points and less no. of elements should be at other faces of body.

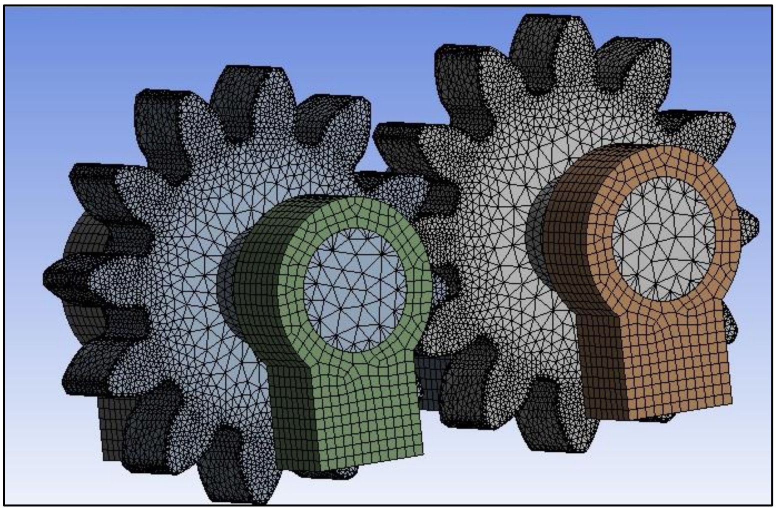


Figure 4.1 Mesh of gear box without defect for analysis

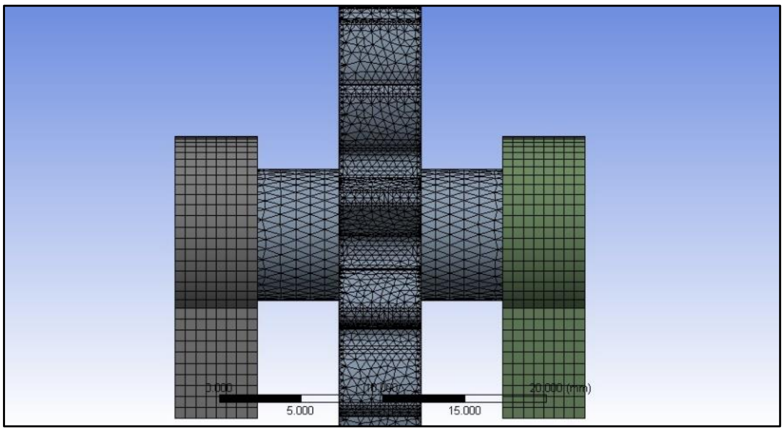


Figure 4.2 Side view of mesh

As shown in Figure 4.1, Element size is less near teeth of gears and element size is high near the center of the gear. On the shaft, regular interval elements arranged for maintaining its cylindrical profile. In clamps number of elements does not so much because clamps do not consider complex geometries. More than 75% elements and nodes are located at teeth only because the profile of teeth is most important for the accuracy of results. In Figure 4.2, Side view of the distribution of nodes and elements is shown.

In Table, a number of nodes and elements are presented in the gear box without defect.

Table 4.1 Distribution of Nodes and Elements in Parts of gear box without defect

Body	**Number of Nodes**	**Number of Elements**	**Type of Element**
Clamps	2619*4= 10476	1896*4= 7584	Multiple
Driving Gear	13501	41700	Tetrahedrons
Driven Gear	13408	40928	Tetrahedrons
Total	37384	90212	Multiple

The Same method is used for the meshing of the defected gear drive gear box. But the here number of elements and nodes have to be increased in driving gear because driving gear have a defect, due to defect unnecessary vibration will generate. Less number of elements and nodes near crack means less effectiveness of crack, but in practice, crack gives proper effectiveness on the gear box vibration, several authors used experimental method for getting the result and in signals effect of crack is clearly visible. So around 1500 nodes and 5000 elements increased in driving gear for analysis of defected arrangement, some nodes were reduced from driven gear for overcoming from time consumption of extra nodes and elements.

In this table, no. of nodes and elements are presented in defected gear system.

Table 4.2 Distribution of Nodes and Elements in parts in defected gear box

Body	**Number of Nodes**	**Number of Elements**	**Type of Element**
Clamps	2619*4= 10476	1896*4= 7584	Multiple
Driving Gear	13105	40355	Tetrahedrons
Driven Gear	13408	40928	Tetrahedrons
Total	36989	88867	Multiple

In Figure 4.3, front view of distribution of nodes and elements in defected gearbox have been shown, distribution is almost similar as the distribution of nodes and elements in gearbox without any defect.

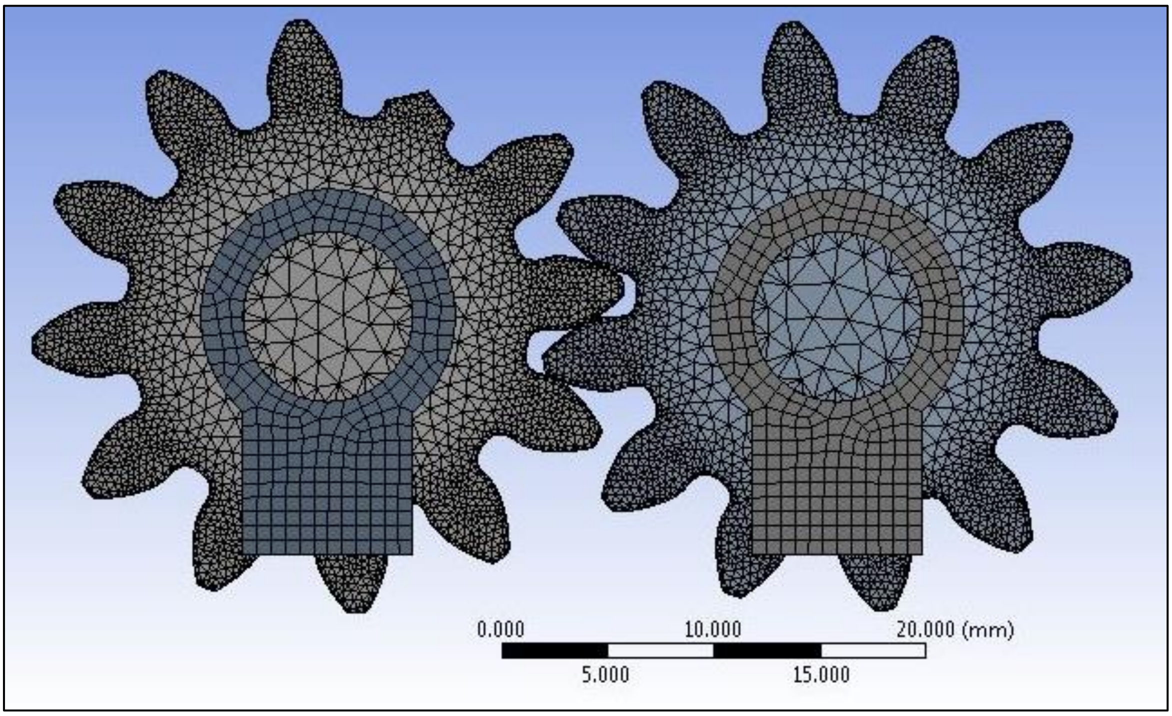


Figure 4.3 Front view of distribution of elements in defected gear system

4.4 Analysis Considerations

Speed range of gears is too much high, the author assumed here a medium speed range gears. The rotation speed of gears is inversely proportional to a life of gears. In this analysis author considered a rotation speed of Gears is 3000RPM since both gears are having the same size so both gears assumed to be rotating at 3000RPM. Driving gear is assumed to be rotating in clockwise direction.

4.4.1 Gear Tooth Frequency

Gear Tooth Frequency also called as gear mesh frequency, GMF of any gear in the mesh is calculated as total n. of teeth meshed in 1 Second. In other words, we can say that Gear Tooth Frequency of gear is equal to the multiplication of frequency of rotation of the gear and total no. of teeth on gear. In the present design, GMF is considered as 600Hz.

GMF= Rotation Speed of Gear × Number of teeth on gears.

Since rotation speed of both gears is 3000RPM

4.4.2 Hunting Tooth Frequency

Hunting tooth frequency is the number of meshing of two same teeth in a gear box in one second. In other words, Hunting Tooth frequency of a gear simple gear box can be defined as the division of Gear Tooth frequency by lowest common multiplication number of teeth on both gears. In the analyzing design, hunting tooth frequency is 50. If hunting tooth frequency is same as rotation frequency then chances of wear and tear increases because if a gear gets failed then that gear will mesh with same gear in every rotation and tends to more wear and tear in another gear also. Hunting Tooth Frequency also known as **Gear Hunt Frequency (GHF).**

$$Hunting\ Tooth\ Frequency = \frac{Gear\ Mesh\ Frequency}{LCM\ of\ number\ of\ teeth\ on\ both\ gears}$$

Here,

$$GHF = \frac{600}{12} = 50Hz \qquad 4.3$$

4.5 Result Considerations

All parameters are defined and all boundary conditions applied on the system. Now results have to be obtained in the forms of signals of vibration. Vibration is a function of acceleration so Author considered four nodes on four clamps (one node on every clamp). Acceleration Probe applied by the author on selected nodes. Then after for time consideration in steady state analysis, Author tried to get data of 0.5 seconds so gear can rotate by 25 rotations in that time, Author considered time gap between results is 5E(-5), so in 25 rotation, around 10,000 results can be obtained out. For analyses of cracked gear system, same time limits considered.

4.6 Conclusion

So in this chapter conditions applied on the system were discussed. The degree of freedom was necessary to be restricted to one. There are several methods for improving mesh quality and increasing the number of elements but increasing number of elements will consume more time of the system to solve the problem. Also, a higher number of elements consume more system memory. So author shortened mesh elements and nodes by using a different type of elements. This chapter starts with an introduction to how to apply boundary conditions and the first application of boundary condition is used for restriction of the degree of freedom. Then after, the mesh has been optimized and this chapter ends with analysis and result considerations. This chapter is last of application of boundary conditions results and more details are presented in chapter 5.

Chapter 5

Results and Discussion

In this chapter, results obtained from Finite Element Method are shown. Signals obtained from ANSYS 15.0 Mechanical Modal are showing linear vibration signals along with X, Y, Z directions, the Total magnitude of vibration is obtained by the square root of the square sum of all components of vibration. In all graphs, Vertical axes show acceleration due to vibration in millimeter/Second^2 and horizontal axes shows time in the multiplication of $5\times10^{(-5)}$ seconds for both gear box without defect and for gear box with defect. For increase, the accuracy of results, the numbers of node and elements have to be increased or step time between two consequent sub-steps has to be reduced. In the present results mesh nodes, elements and step time is considered as discussed in chapter 4.

5.1 Amplitude of Acceleration (without any defect)

In graphs Horizontal axes represent the time. For gear box without defect, 1 second is equal to 20,000 Sub-steps so it can be said that 1000 sub-steps are equal to 0.05 seconds. The rotation speed of gears is constant so 1000 substeps also represent 15.708 radians or 1800 degrees of rotation. On Y- Axis, Y axis representing the signal of vibration, vertical Axis is representing the amplitude of acceleration.

5.1.1 Input Shaft

Ideally signals of vibration should be similar for all teeth in mesh but due to clearance of 0.002mm, sometimes backlash effect also becomes the cause of vibration. In graphs, a sudden rise in the amplitude of vibration is nothing but a signal of backlash effect in gears in mesh.

Sometimes vibration is increased to 30 mm/sec^2 that is nothing but backlash effect, due to friction 0.002mm clearance was taken during geometry designing so that clearance become the cause of vibration due to the creation of backlash. Figure 5.1 present amplitude of acceleration for X component vs. time curve. The damping effect was the cause of high amplitude initially so for presenting signal closely, data of initial 0.01 seconds has been hidden.

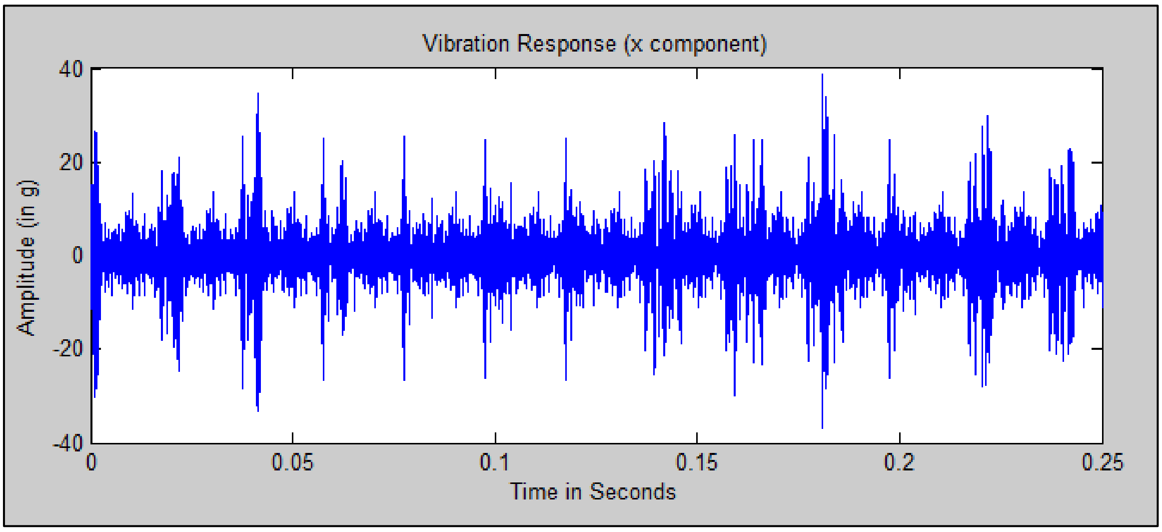


Figure 5.1 without defect input shaft Acceleration (X component) vs. Time

A similar effect on Y component also, Figure 5.2 showing the amplitude of acceleration in Y component.

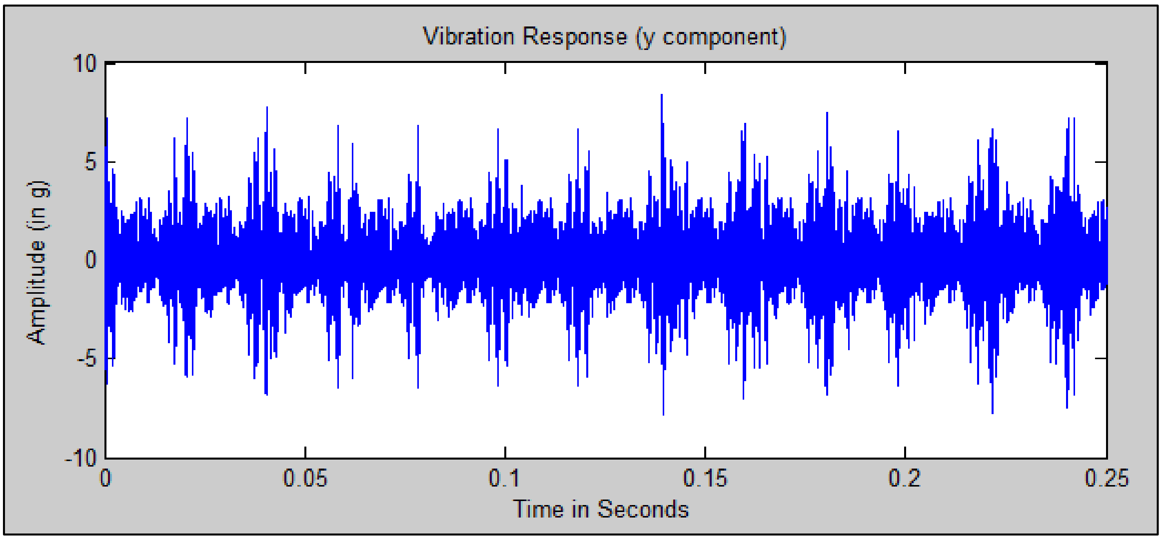


Figure 5.2 without defect input shaft Acceleration (Y component) vs. Time

Backlash directly become the reason of vibration in X and Y components only but due to Torque generation, Backlash becomes the cause of vibration in Z direction also. Figure 5.3 present the amplitude of acceleration in Z component.

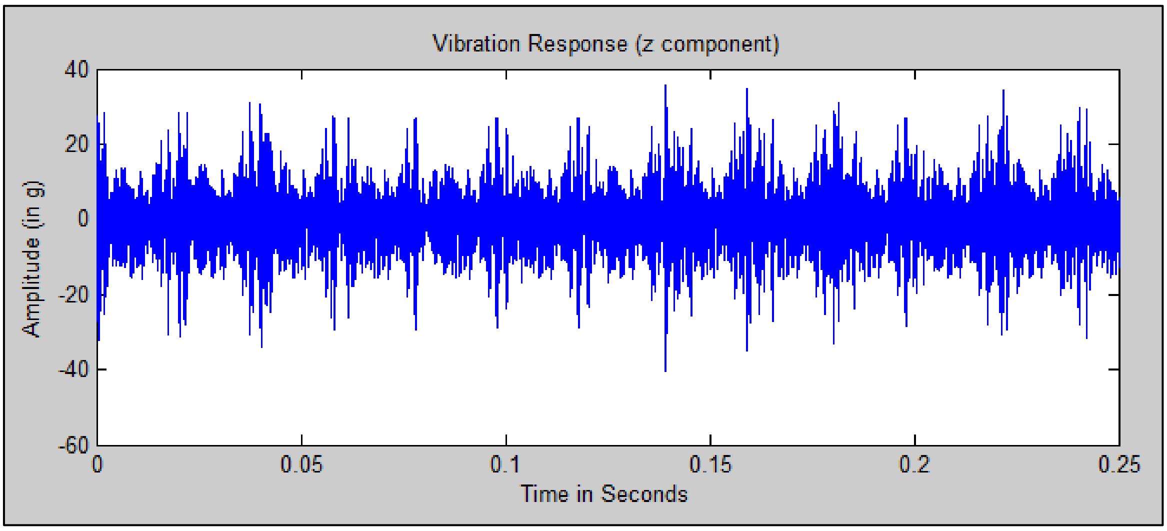


Figure 5.3 without defect input shaft Acceleration (Z component) vs. Time

In final Curve of the amplitude of acceleration, all values are greater than zero because it is nothing but the square sum of all three components. Figure 5.4 presenting the final curve of the amplitude of acceleration at node 1, which is located at clamp 1.

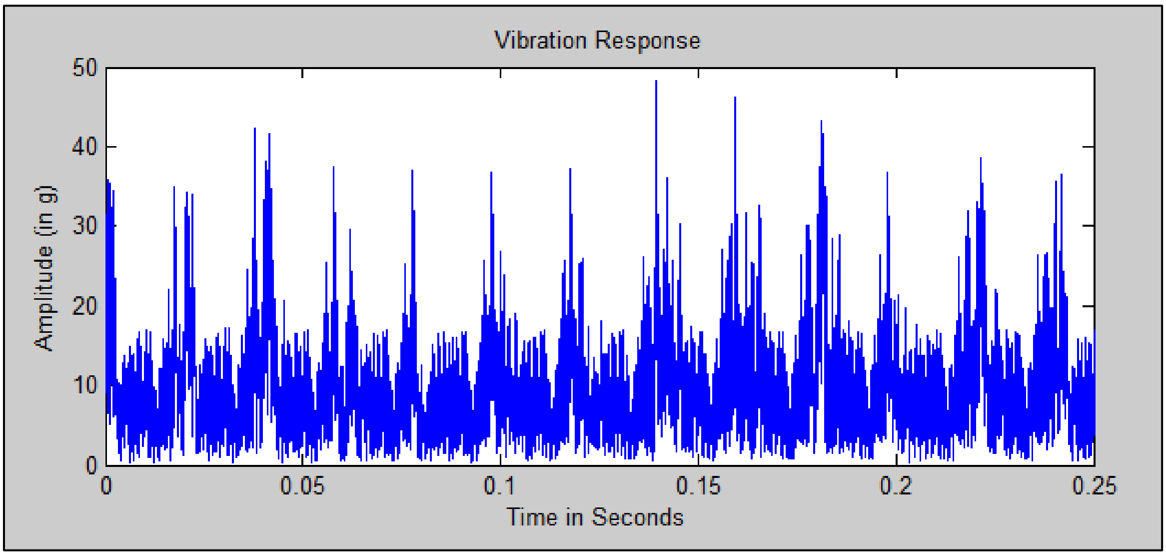


Figure 5.4 without defect input shaft Acceleration vs. Time

5.1.2 Output Shaft

Profile of vibration response on the output shaft is also somehow similar as the input shaft. Because outside conditions and gear profile for both driver and driven gear is similar but the amplitude of vibration on the output shaft is too much less than the amplitude of vibration on the input shaft. This happens due to inertia force and clearance between meshing of teeth. On output shaft, results are compiled in X, Y, Z components and their final response is obtained with the square sum of all three components. In Figure 5.5, X component of amplitude of acceleration vs. time curve is shown, conditions are similar to Figure 5.1.

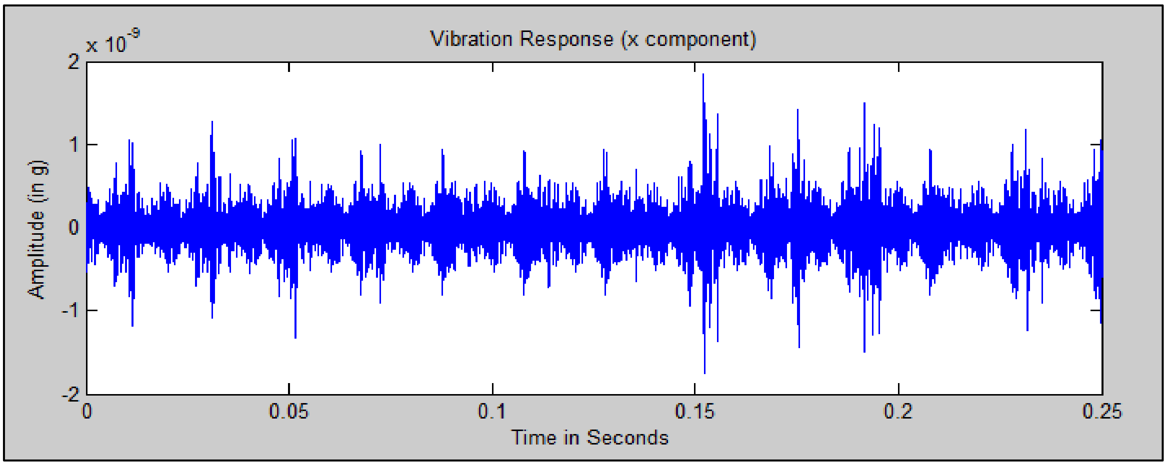


Figure 5.5 without defect output shaft Acceleration (X component) vs. Time

Figure 5.6 presenting the Y component of amplitude of acceleration vs. time curve at node 3 at the output shaft.

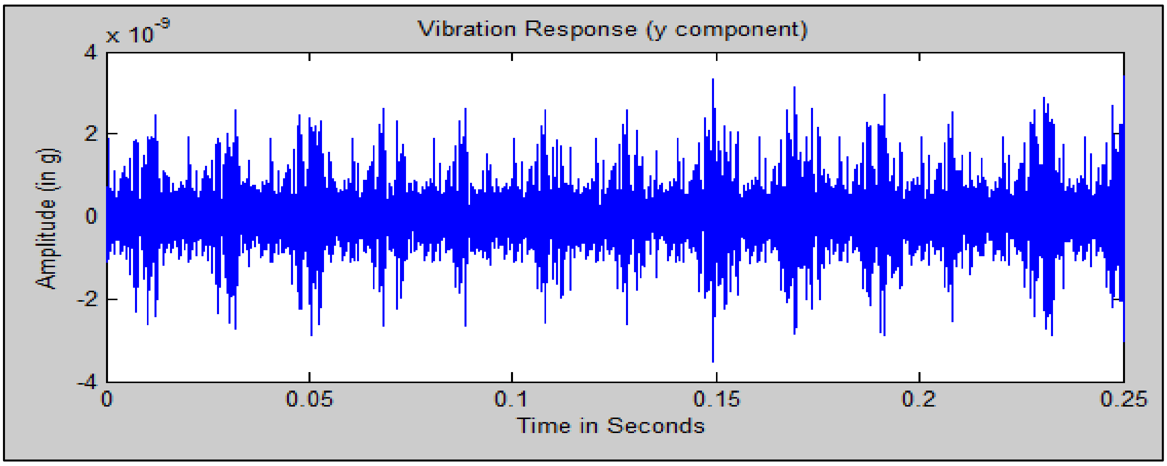


Figure 5.6 without defect output shaft Acceleration (Y component) vs. Time

Figure 5.7 showing the Z component of amplitude of acceleration vs. time curve at the output shaft. Figure 5.8 shows the overall amplitude of vibration at the output shaft. Overall amplitude of vibration is a scalar value and it has no direction because it is the square sum of all three components X,Y and Z so the amplitude of acceleration remain positive at all time in Figure 5.8.

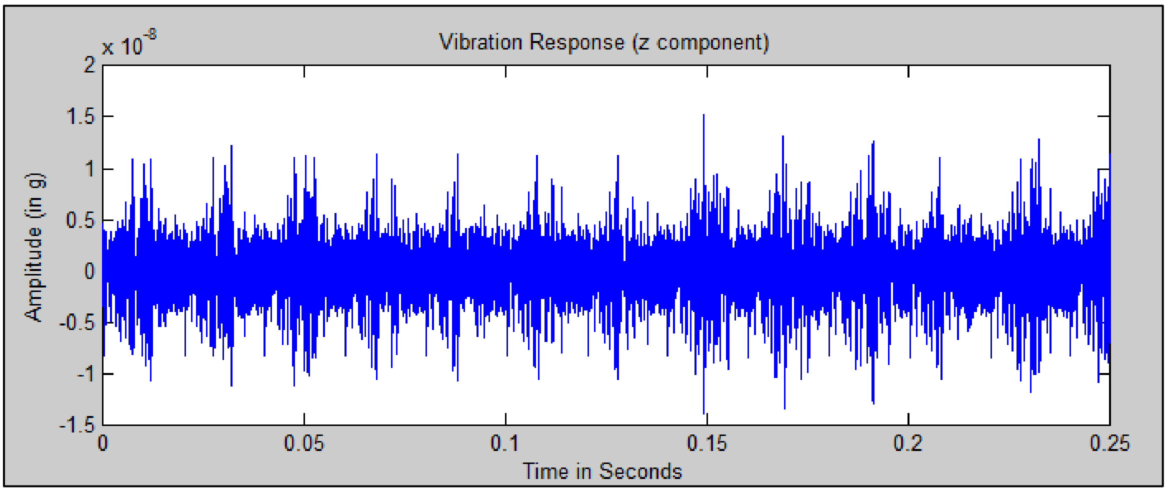


Figure 5.7 without defect output shaft Acceleration (Z component) vs. Time

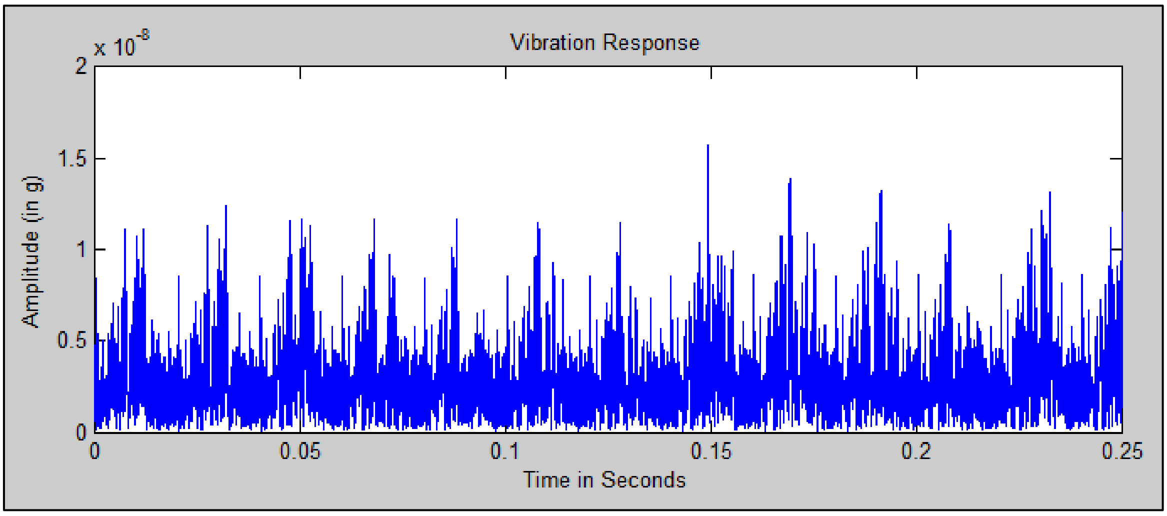


Figure 5.8 without defect output shaft Acceleration vs. Time

5.2 Amplitude of Acceleration (with a defected tooth)

Here also considerations are nearby similar as analysis of not defeated gear box in graphs Horizontal axes represents the time. For gear box without defect, 1 second is equal to 20,000 Sub-steps so it can be said that 1000 sub-steps are equal to 0.05 seconds. The rotation speed of gears is constant so 1000 sub-steps also represent 15.708 radians or 1800 degrees of rotation.

5.2.1 Input Shaft

On Y- Axis, Y axis representing the signal of vibration, vertical Axis is representing the amplitude of acceleration. Figure 5.9, Figure 5.10, Figure 5.11 consequently present amplitude of acceleration with considering only X-component, Y- Component and Z-component consequently. Y- Axis of all curves present their amplitude of acceleration and X- Axis present the time. Here also the amplitude of acceleration for initial 0.01 seconds is not presented due to the higher value of the amplitude of acceleration due to inertia and damping.

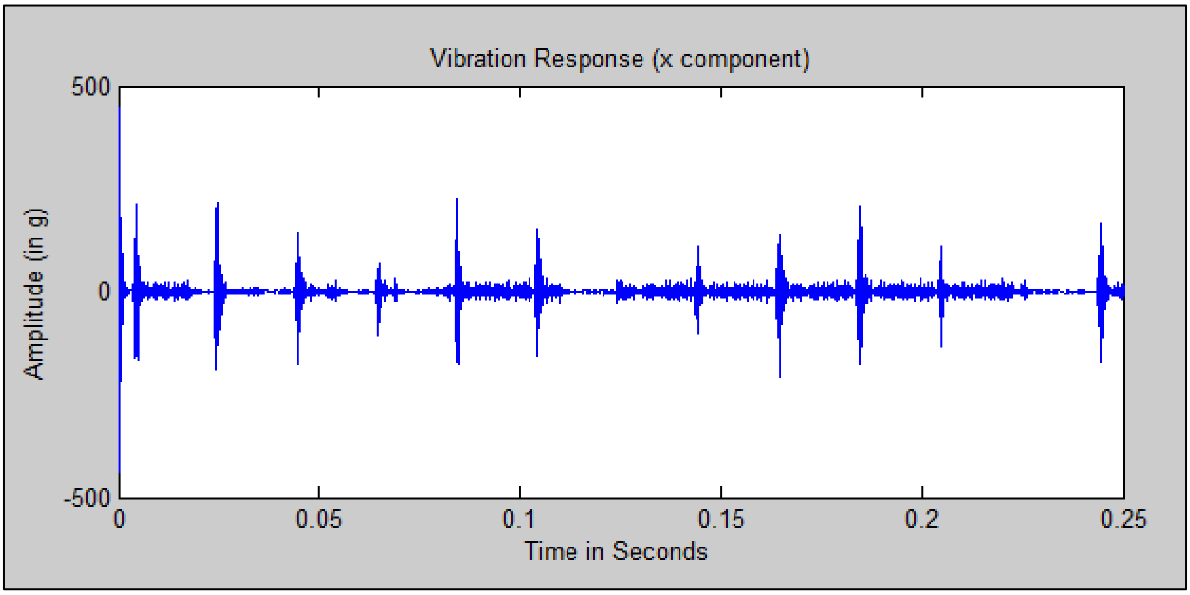


Figure 5.9 gear box with defect input shaft Acceleration (X component) vs. Time

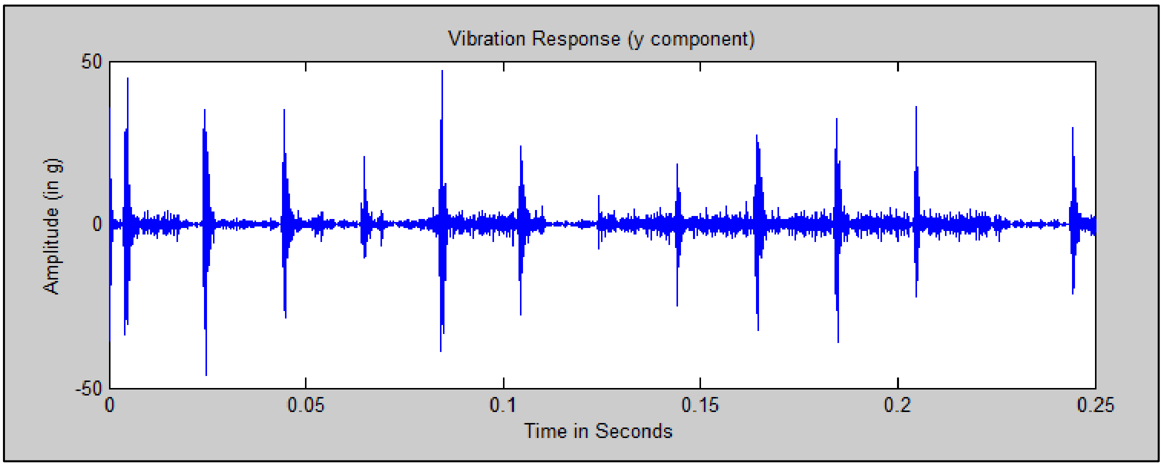


Figure 5.10 gear box with defect input shaft Acceleration (Y component) vs. Time

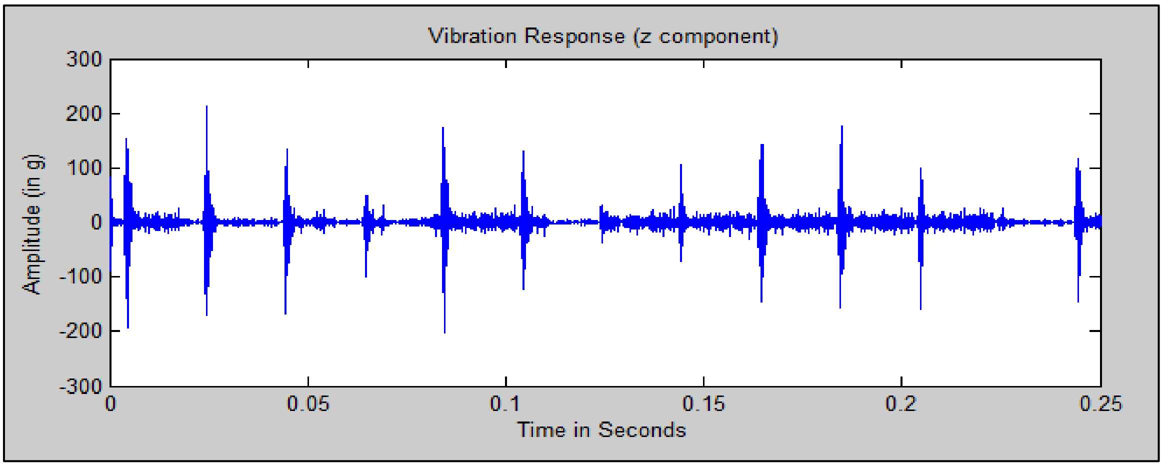


Figure 5.11 gear box with defect input shaft Acceleration (Z component) vs. Time

Figure 5.12 Overall amplitude of vibration is shown. It is a scalar value and it has no direction because it is the square sum of all three components X,Y and Z.

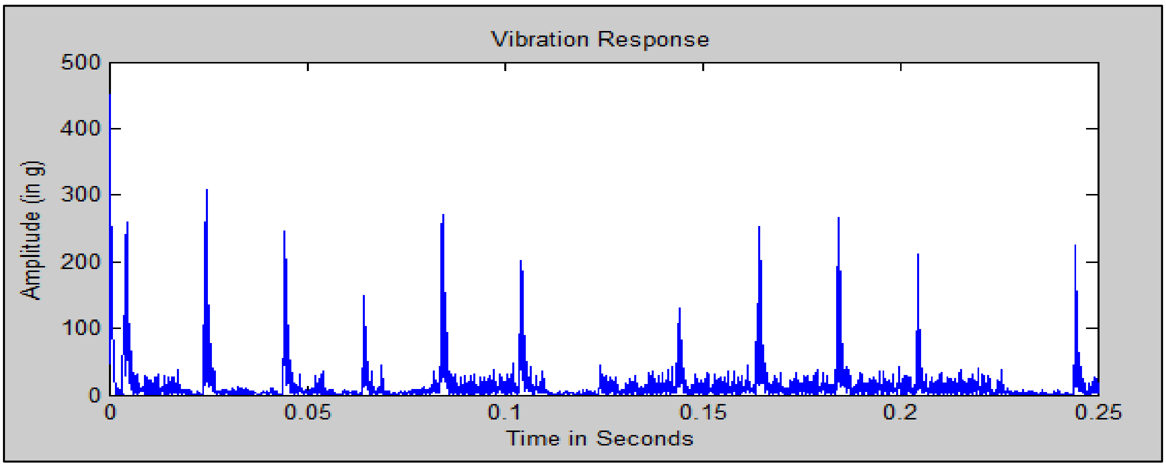


Figure 5.12 gear box with defect input shaft Acceleration vs. Time

5.2.2 Output Shaft

On output shaft, amplitude is too much different from input shaft because gear which is mounted on output shaft is assumed to be an ideal gear and no defect is considered in driven gear, according to design defect is in a teeth of gear 1 only, but backlash effect will surely be too much less on gear 2 because it is a derived gear and own inertia of output shaft and gear is enough for overcome due to backlash action. Figure 5.13, Figure 5.14 and Figure 5.15 consequently present amplitude of acceleration with considering only X-component, Y- Component and Z-component consequently. Y- Axis of all curves present their amplitude of acceleration and X- Axis present the time. Here also the amplitude of acceleration for initial 0.01 seconds is not presented due to the higher value of the amplitude of acceleration due to inertia and damping.

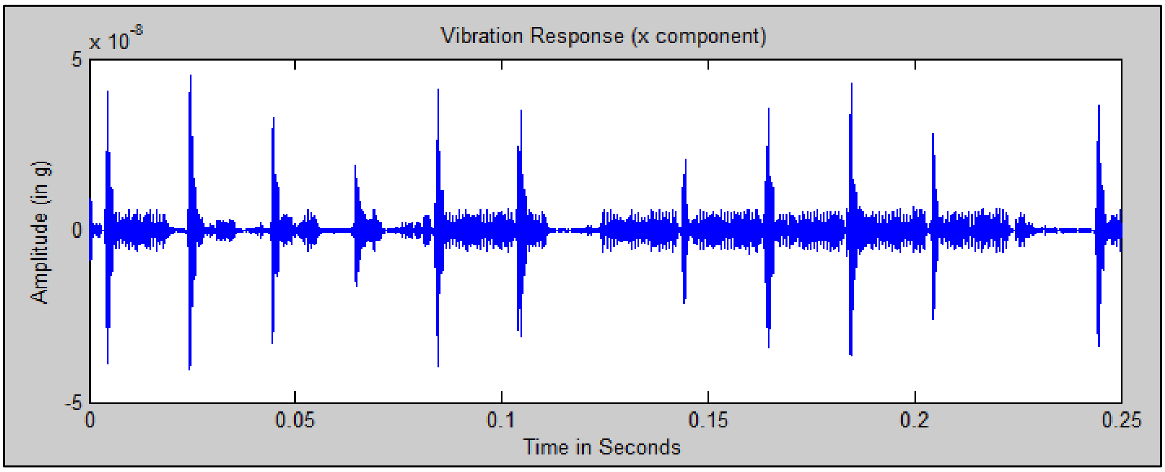


Figure 5.13 gear box with defect output shaft Acceleration (X component) vs. Time

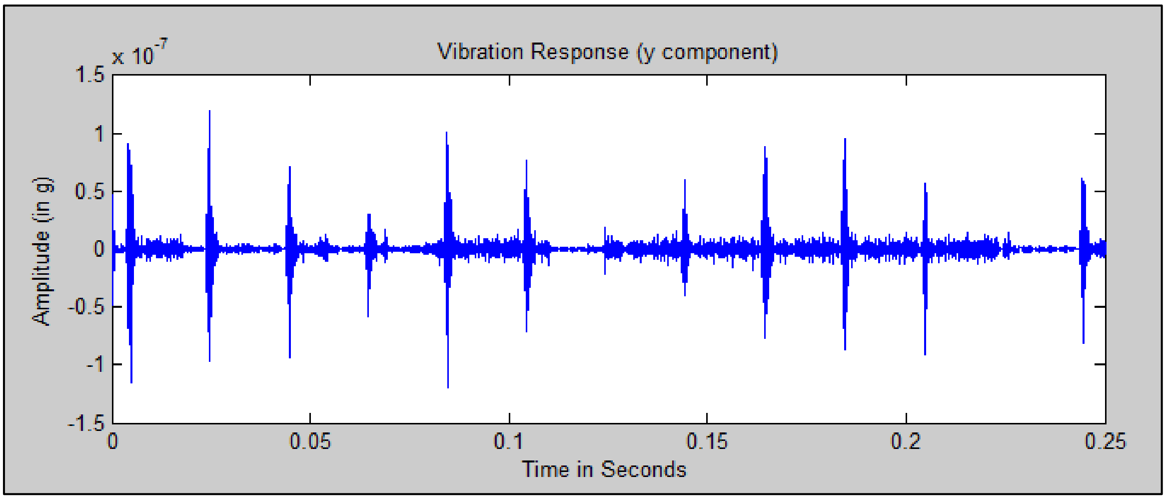


Figure 5.14 gear box with defect output shaft Acceleration (Y component) vs. Time

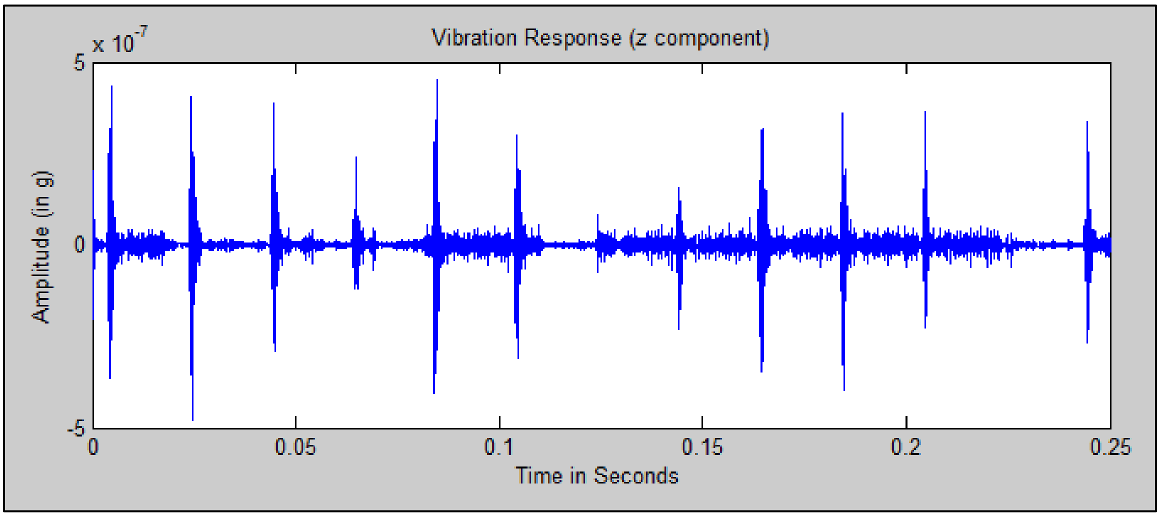


Figure 5.15 gear box with defect output shaft Acceleration (Z component) vs. Time

Figure 5.16 Overall amplitude of vibration is shown. It is a scalar value and it has no direction because it is the square sum of all three components X, Y and Z.

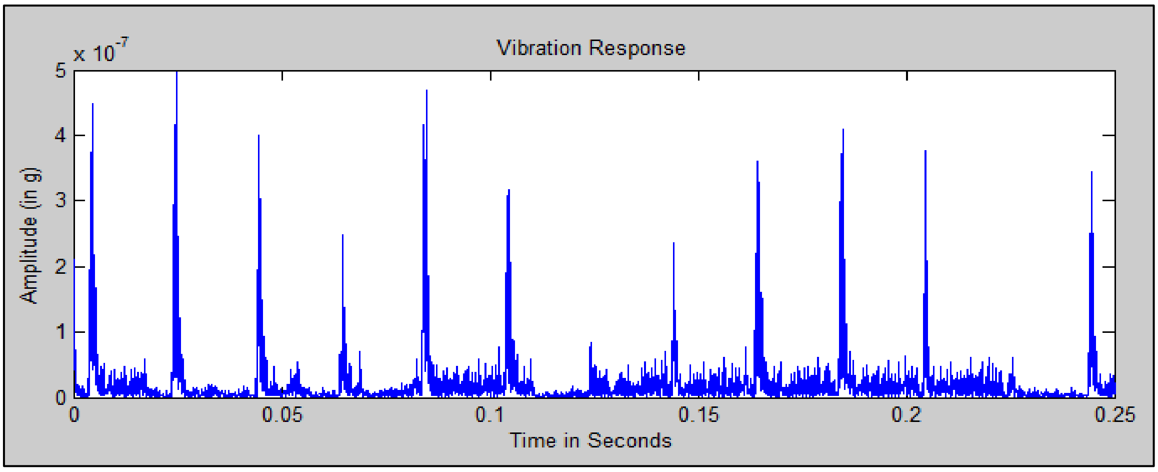


Figure 5.16 gear box with defect output shaft Acceleration vs. Time

5.3 Single-Sided Amplitude Spectrum

Figure 5.17, Figure 5.18 and Figure 5.19 consequently showing the single sided amplitude spectrum of Figure 5.1, Figure 5.2 and Figure 5.3 consequently. These figures are for verifying the results. In chapter 4, it has been proved analytically that hunting tooth frequency of designed gear box is 50Hz, in all three curves, the effect of hunting tooth frequency is visible clearly. These curves are nothing but fast Fourier transformation of those Figures for seeing the results in the frequency domain on the place of the time domain.

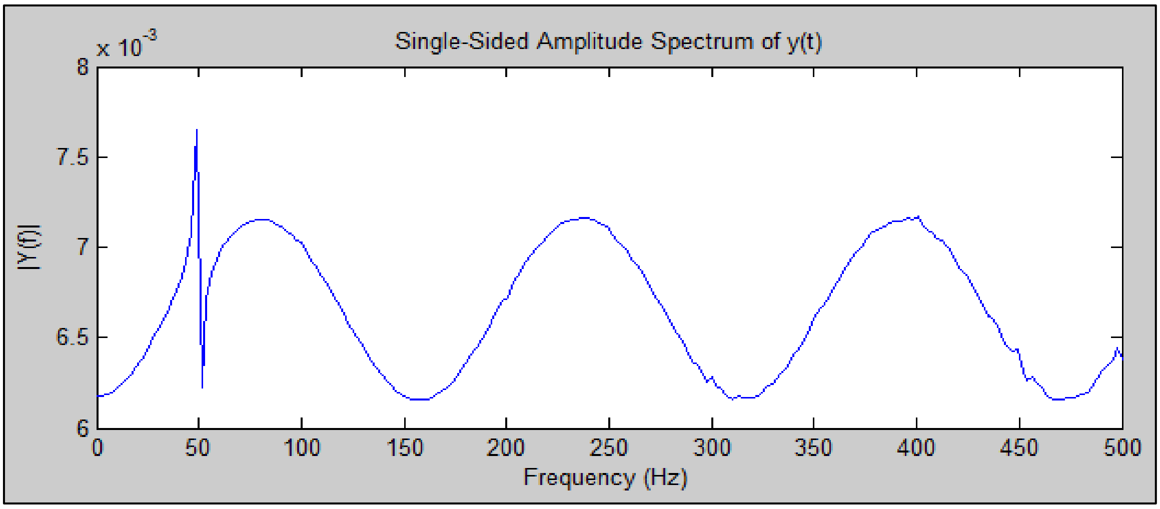


Figure 5.17 Single -sided amplitude spectrum for X component without defect input shaft

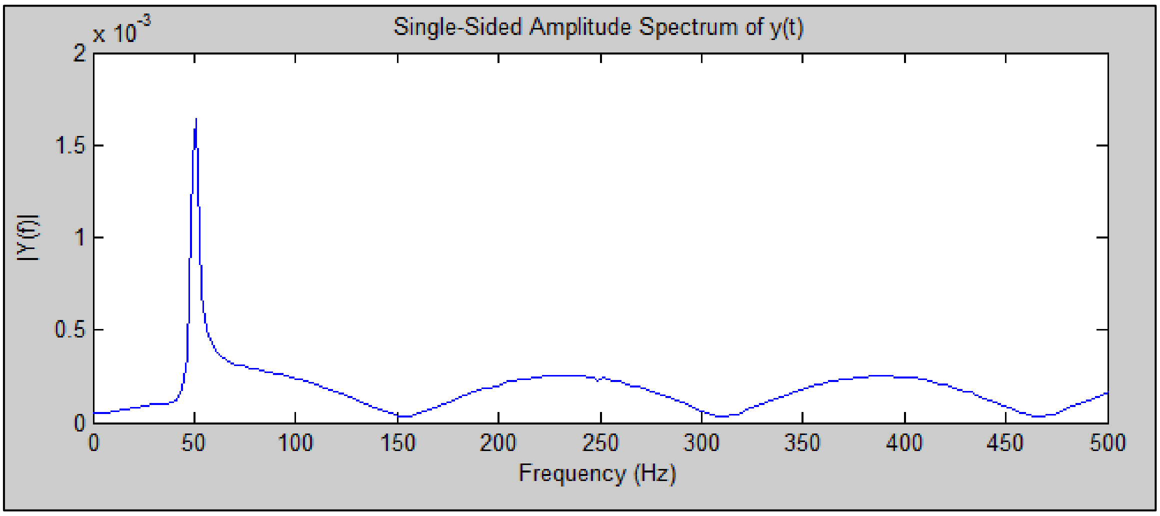


Figure 5.18 Single -sided amplitude spectrum for Y component without defect input shaft

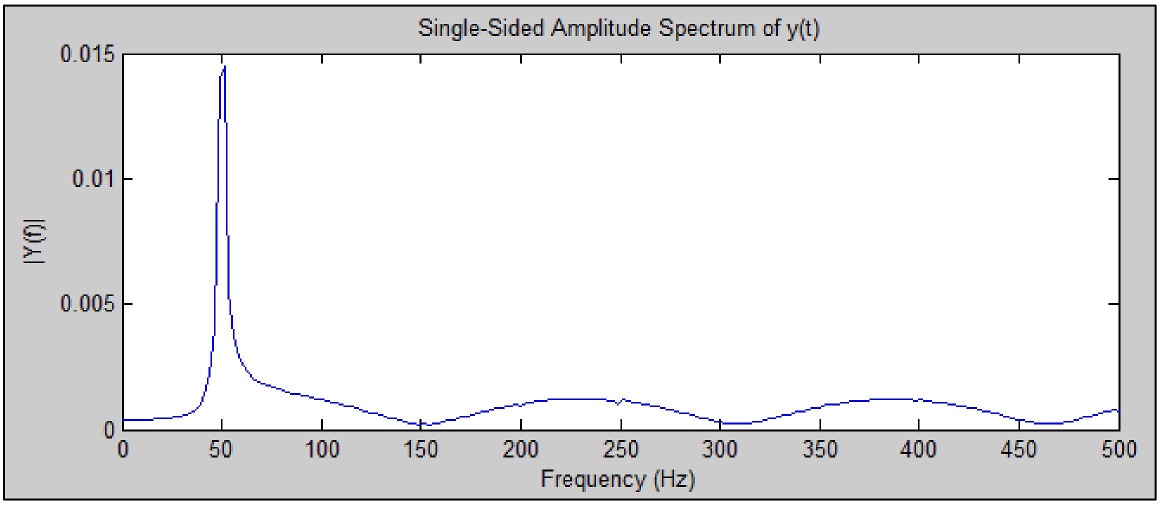


Figure 5- 19 Single -sided amplitude spectrum for Z component without defect input shaft

5.4 Discussion

For clamp input shaft, ideally signals of vibration should be similar for all teeth in mesh but due to clearance of 0.01mm, sometimes backlash effect also become the cause of vibration. This is easily visible in forms of the amplitude of vibration in figure 5.1 to figure 5.8. In graphs, a sudden rise in the amplitude of vibration is nothing but a signal of backlash effect in gears in mesh. A similar case in input shaft acceleration Y Component also, here also backlash effect is visible, Backlash directly becomes the reason of vibration in X and Y components only but due to Torque generation, Backlash becomes the cause of vibration in Z direction also. In final Curve of the amplitude of acceleration at input shaft, all values are greater than zero because the overall amplitude of acceleration is the square sum of all X, Y and Z components and it's a scalar value.

For defected gear, the amplitude of acceleration changed suddenly because defected teeth come in the mesh. In all 4 graphs (from figure 5.9 to figure 5.12) effect of vibration is easily visible in forms of higher values of amplitude of acceleration.

On output shaft, amplitude is too much different from input shaft because gear which is mounted on output shaft is assumed to be an ideal gear and no defect is considered in driven gear, according to design defect is in a teeth of gear 1 only, but backlash effect will surely be too much less on gear 2 because it is a derived gear and own inertia of output shaft and gear is enough for overcome due to backlash action. Profile of amplitude of vibration is somehow similar to the profile of amplitude of vibration at input shaft in this case also. But amplitude of acceleration is too much less here comparing to the amplitude of acceleration at input shaft.

As examined in chapter 4, equation 4.2 verify that tooth hunt frequency of gear box was 50Hz, so for verifying results, amplitude vectors of without defect gear box input shaft were changed into frequency domain from the time domain using fast Fourier transformation. After changing in the time domain to frequency domain, curves of frequency vs. single- sided amplitude spectrum were compiled using MATLAB. Here in curves of input shaft X, Y and Z components, the effect of change in curve at hunting tooth frequency is easily visible. In X component, Single -sided amplitude spectrum get sudden changed as frequency reached at 50 Hz, on another side in Y and Z components, the value of single -sided amplitude spectrum reaches at its maximum at 50 Hz, which is equal to hunting tooth frequency of gear box.

Chapter 6

Conclusion

6.1 Review of Thesis

So Author successfully achieved his goal, his goal to obtain the signal of vibrations in gears using Finite Element Analysis. In chapter 5, results are shown in graphical forms; results are obtained at input shaft and output shaft. On both shafts, there were 2 clamps at each end, but at one end there were just a simply support only but the joint load was applied on the second end, so the amplitude of vibration signals was shown in results for the locations of joint loads in input and output shaft. Firstly in the introduction, author give brief information about the type of gears and vibration because the author has to study on these also for getting his problem, then after author shows that what he have learned from journals and literature papers in chapter 2, In chapter3, the author presented about his design of his problem. In chapter 3, the author also discussed requirement of software for achieving his goal. In chapter 4, details of analysis were written. Chapter 5 start with graphs which were indicating the initial goals of the author and finally chapter 5 ends with verification of results by the author. So the author has completed his thesis based on own work in a sequential manner.

6.2 Future Work

Since it was initial stage and crack is increased by time due to vibration and other material properties so these results can be considered as a conclusion. In future some more analysis can be done on the given setup in few more ways explained bellow-

1. Drawing wear and Tear on the teeth on the place of crack.
2. Internal hole in the gear material for reducing its homogeneous behavior
3. Crack in the gear in the place of teeth.
4. Change in the contact region of the mesh.
5. Replacing cylindrical joints with ball bearings.

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Annexure-A

MATLAB Code for Involute Profile Coordinates

```
s= 10.875
r= 10.875:0.075:14.1
k= ((r.^2-s.^2).^0.5)/s
m= k-atan(k)
x= r.*cos(m)
y= r.*sin(m)
plot(x,y)
xlim([10 15])
ylim([0 5])
title('Involute Profile Curve')
xlabel('In Milimeter')
ylabel('In Milimeter')
```