ADDIS ABABA UNIVERSITY
ADDIS ABABA INSTITUTE OF THECHNOLOGY
SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Forced Vibration Analysis of Chery 100hp Tractor Transfer Case

A Thesis Submitted to the Partial Fulfillment of the Degree of Masters of Science in Mechanical Engineering

(Mechanical Design Stream)

By

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A thesis submmited to school of graduate study of Addis Ababa university in partial fullfilment of the requirement of the degree masters of science in mechanical engineering (Mechanical design)

By Tsiyon Simeneh

Approved by Board of Directors

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Declaration

This is to certify that, the thesis prepared by Tsiyon Simeneh, with the title of *forced vibration analysis of Chery 100hp Tractor Transfer Case*. This thesis is original work and it has not been submitted for a degree in any university or institution.

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Dedicated

To my

Beloved

Brother

Shemels Gebeyehu

List of Abbreviations

Abbreviations	Definitions
❖ a	Amplitude
❖ f	Frequency
❖ v	Velocity of fluid
❖ k	Stiffness
❖ x	Displacement
❖ $\dot{x}$	Velocity
❖ $\ddot{x}$	Acceleration
❖ C	Damping coefficient
❖ X(t)	External force
❖ F(t)	Harmonic excitation force
❖ W	Weight
❖ C1	Constant 1
❖ C2	Constant 2
❖ $X_p(t)$	Particular solution
❖ $X_h(t)$	Homogeneous solution
❖ ω	Natural frequency
❖ δ_{st}	Deflection
❖ τ	Shear Stress
❖ φ	Phase angle
❖ m	Mass of the transfer case
❖ c_c	Critical damping coefficient
❖ r	Input shaft radius
❖ ξ	Damping factor
❖ b	Base
❖ h	Height
❖ I_f	Moment of inertia for front face of transfer case
❖ I_R	Moment of inertia for rectangular face of
❖ I_c	Moment of inertia for composite face of transfer case

- I_{rl} Moment of inertia for right lateral face of transfer case
- I_{ll} Moment of inertia for left lateral face of transfer case
- I_B Moment of inertia for bottom face of transfer case
- I_T Moment of inertia for left top face of transfer case
- E Young modulus
- g gravity
- L Length of transfer case
- U Velocity of oil distribution
- Y Vertical distance
- dF Infinitesimal force
- dT Infinitesimal torque
- θ Angle of rotation
- ρ Density
- K_v Kinematic Viscosity
- V_{rms} RMS velocity
- AAMI - Adama agricultural machinery industry
- FEM - Finite Element Method
- HP - Horse Power
- ISO- International Standardization Organization
- RMS- Root Mean Square
- PSD- Power Spectrum Density
- ETB- Ethiopian Birr
- PTO- Power Take off

Abstract

The main purpose of this thesis is to analyze the forced damped single degree of freedom vibration of Chery100hp tractor transfercase. The tractor which assembled at AAMI.An appropriate mathematical modeling, field measurement,response on harmonic excitation of the transfer case by means of ANSYS 17.2 were performed to identify the critical frequency, to know whether the natural frequency calculated through mathematical analysis fall on the range of critical frequency or not and to identify the high vibration speed zone on the particular tractor.
The general solution found in mathematical modeling imported to MATLAB to generate results. The recorded value calculated by Random velocity calculation method and compared with ISO General Standard (10816-1:1995).3D modeling of transfer case made by using CATIA V5 and imported to ANSYS 17.2 for modal analysis.
According to the analysis the SAE 140 has significance effect on 100hp but negligence effect on 95 hp. The comparison calculated natural frequency value of the current transfer case with critical frequency analyzed harmonic excitation indicates that the natural frequency found to be 5063.05 rad/sec or 805.8Hzwhile the critical frequency fall on the range between (825-875) Hz. This natural frequency is in the same range (500-3500) Hz as experimental analysis obtained by Jiri Tuma (2009).The measured vibration velocity value of eight speed zone condition was calculated and comparedwith ISO as a result bad vibration zone was identified.

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Chapter One

1. Introduction

Agricultural tractor is one of the class of mobile machines that involves the ‘traction’ process. The word’ traction' and name 'tractor' come from the word to 'draw' or 'pull' so a tractor is basically a machine for pulling; other mobile machines such as locomotives are in the same class. Vehicles like road trucks and even motor cars, which are essentially vehicles for carrying loads, also involve the traction process[1].Tractor represents the most important machine in agriculture and designated to pull and power variety of different agricultural machines and implements applied in complex technological operations of agricultural production. In many countries about 40 to 45 % of the total agricultural machinery investment is related to tractors[2]. Tractors are sometimes designed to perform vibration on field activities intentionally. Passes of vehicles over irregularities of the road surface and presence of offset mass on rotating axes of these machines, cause mechanical vibrations Furthermore, since the majority of light weight agricultural tractors (below 150 hp) haven’t any conventional suspension system, these machines transmit vibrations to the driver more than the vehicles with suspension system [3].

The tractor drive train has three functions. It transmits power from the engine to the wheels, PTO, hydraulic pump, and other auxiliary drives; changes the torque and speed into the torque and speed required by the particular drive; and provides means for operator control with disconnect clutches and speed ratio selection for the wheel and PTO drives. The tractor tractive capability provides a practical limit on the amount of pull that can be generated and thereby limits drive train torques. When a limited number of speeds are available, it will not always be possible to use the maximum engine power available, especially if the load is highly variable.

Transfer cases are used in off road vehicles to divide engine torque between the front and rear driving axles. The transfer case also allows the front driving axle to be disengaged, which is necessary to prevent undue drive line component wear during highway use. Another purpose of the transfer case is to move the drive shaft for the front driving axle off to the side so that it can clear the engine. This arrangement is necessary to allow adequate ground clearance and to allow the body of the vehicle to remain at a practical height. A conventional transfer case is constructed similar to a transmission in that it uses shift forks, splines, gears, shims, bearings, and other components found in manual and automatic transmissions. The trucks with four wheel drive option will be running in rear wheel drives and the front wheels will be rotating freely. In extreme tough driving conditions, the risk for slipping of the rear wheels or getting stopped in mud is high. By using four wheel drives, which makes all the wheels drive the vehicle, the traction and the maneuverability of the vehicle will be high. [4]

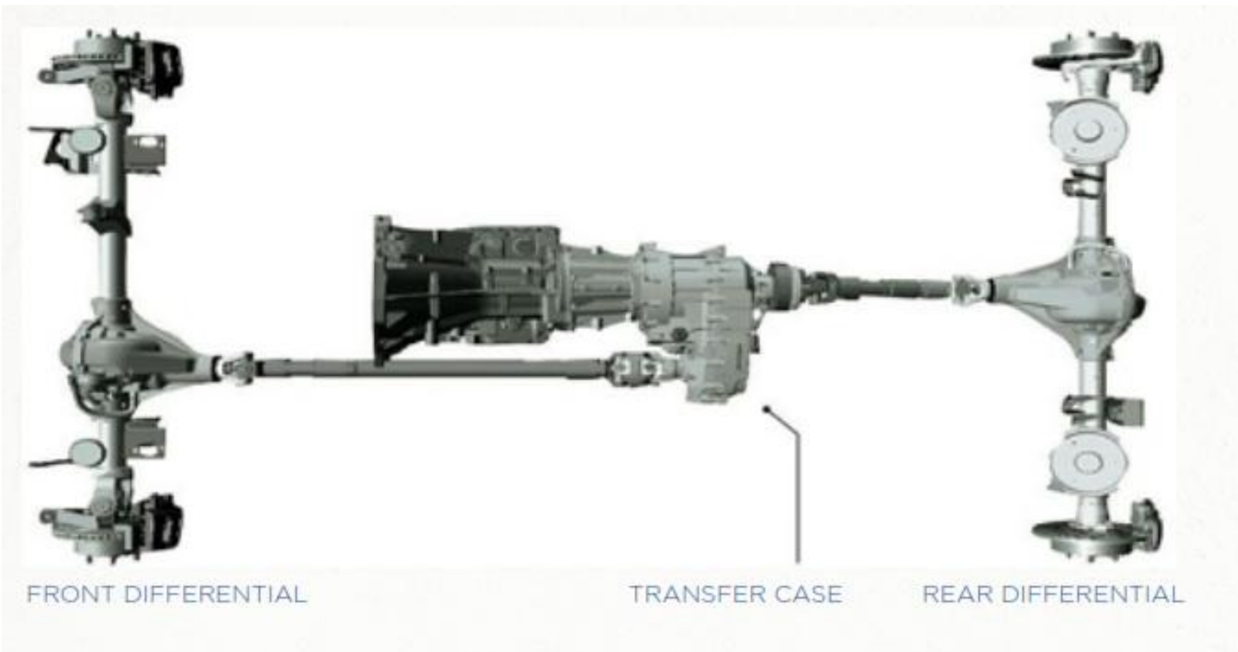


Figure .1 Placement of the transfer case in a drive line

The above picture were taken from Holman Automotive Group, 2015, for the better explanation of the location of transfer case in the drive line.

Gearbox Casing is one of the most critical components of a power transmission system in automobile. It protects components of gearbox by providing the fluid tight casing to hold the lubricants and provides support to moving components [5].

In transmission the power is transmitted in this order engine, clutch, gearbox, prop shaft, differential, half shafts, hubs and tires. During power transmission if there is a slack in drive train it may cause high vibration known as transmission Shock. It is the main reason for gearbox assembly and housing failure [6].

Vibration analysis is used to determine the operating and mechanical condition of equipment. A major advantage is that vibration analysis can identify developing problems before they become too serious and cause unscheduled downtime [7].

Gearbox casing failure is the main problem for the vehicle manufacturer. Noise and vibration are the main reason for failure [8].Heavy vehicle truck transmission gearbox housing is subjected to load fluctuations, harmonic excitation, gear meshing excitation, gear defects, varying speed and torque conditions. Internal excitation is the reason for transmission housing failure. Resonance occurs due to matching of harmonic excitation frequency with natural frequency of gearbox causing excessive vibration and leads to failure of vehicle transmission system[10].The external excitation on transmission housing must be eliminated to prevent the fracture of housing. The reason for the fracture is matching of external excitation frequency to natural frequency of transmission housing. heavy vibration excitation is the main reason for gearbox casing failure. In order to prevent failure the natural frequency and natural mode shapes should be known [5].

The main parts of the gearbox are the gears, shafts, bearings, housing, and the outside of the gearbox clutches and couplings [8].The wrong shifting of gears causes clashing. Clashing is a loud noise produced during collision of gear tooth and this collision leads the transmission failure [6].

The gearbox casing analysis is very essential because to predict casing behavior under different operating conditions and decide exact dimensions, typically casing failure occurs at 800rpm to 3200rpm to damage gearbox[11].

2. Backgroundof the study

Adama Agricultural Machinery industry was established as Nazret Tractor Assembly Plant (NTAP) on 1984. AAMI primarily produce tractors with variety of horse powers besides it produce different type of water pumps irrigation materials and linked implements. Chery 100hp tractor is one of the tractor assembled at AAMI with capability of 100 horse power.

This thesis work consists on the forced vibration analysis of Chery 100 hp tractor transfer case model at AAMI. Those tractors are multi purposed tractor like compacting, plowing, harrowing with flexible steering, great traction and extensive range of usage. When fitted with appropriate agricultural machinery the tractor can be applied tilling, harrowing sowing and harvesting and other operation.

According to the data that have been gather round from marketing sector of AAMI Chery 100 hp model tractor begin to assemble at AAMI since 2017.Those tractors have been sold out to different part of the country. The total amount of tractors traded from the beginning of production are 113 in number. These tractors are distributed to different part of the country specially at Bale, Arsi, Mojo ,Ambo,Southern GujiGambela and others .The unit cost of those tractors are 1,143,012.255 including 15% of tax .

Whenever the tractors got damage on their different spare parts due to many causes the customers have the right to return those tractors for the sake of maintenance with in the allowable warranty time length. When it exceeds the warranty period the expense will be covered by the customers.

According to the information that has been grasped from the operators and chief mechanics who works in the company and sent for trouble shouting those tractors on the field. Almost all of this tractor model fails with transmission housing or gear box problems. Based on evidences from the company within the past two years operators sent to the field to repair 6 transfer case. Throughout their experience they explained that there is no maintenance for transfer case whenever the failure happens as an alternative the whole transfer case will be rejected and replaced by new one from the store or will be buy from markets away. Which is pricey because the failure happens with in the first six month of tractor operation.

The cost for the replacement of the transfer case including labor, transport and spare part cost will be 11,000ETB.

The labor cost is depends on the location of the tractor but the average per dime is 300ETB for chief mechanic and 200ETB for assistance mechanic minimum for ten day field stay during troubleshooting. The data from marketing department told that those tractors are sold to investors, micro enterprise association and farmers. It’s known that the affordable capacity of Ethiopian farmers is low so that most of them took those tractors in a credit to pay back within a certain period of time.

The transfer case mainly used in four wheel drives on the time of operation in order to make the traction force high. Inside transmission housing of this tractor model there is main gear shifting which is controlled by 2 gear levers. The main gear lever can be shifted in to 4 gears (1, 2, 3, 4) and the auxiliary gear lever can be shifted in to 2 speed zone (low speed zone and high speed zone) .When the auxiliary gear lever is pushed from neutral position to the right and to the front the reverse gear will be engaged. When the tractor is driving on the asphalt it will be driven at high speed because no need of traction force. But during field operation the speed should be reduced to get more power or torque in order to increase the traction force and to perform the field operation in depth. The location of the transfer case is inside the transmission housing between the transmission and the rear differential.

As said by the employers of marketing department the company relationship with the customers is quiet good based on business approach and on how they meet the expectation of customers. Even though the amount Chery 100 hp tractors that has been sold out are 113 since 2016 but the amount of tractors assembled are 1000 tractors this shows the negative aspect of development of marketing process on this and will be challenge for sustainability market development and growth.

Cause of the transfer case failure

Transfer Cases are very sensitive to any external changes due to their integration with the restof the powertrain control system in today’s vehicles.Noises, vibrations and other abnormal driving characteristics can help to diagnose a differential or driveline problem.The broken bearings, great noise and vibration often occurs with the transfer case when it starts up working [31].

Gearbox is a source of vibration and, consequently, noise. Except for bearing fatal defects or extreme structure-resonance amplification, gears are the main sources of high frequency vibration and noise, even in newly built units[8]. In a power transmission gear system, the vibrations generated at the gear mesh are transmitted to the gearbox housing through the shafts and bearings[9].After reviewing many literatures related to the main reason for the failure of the transfer case the analysis shifted in to vibration analysis by considering the remaining things to be negligible than vibration.

1.2. Statement of the Problem

Generally Tractors are not manufactured in Ethiopia instead tractor components are imported from foreign markets and assembled in Ethiopian companies. AAMI is one of these companies which assemble tractor by importing components from Chinese company. Chery 100hp is a tractor model assembled at AAMI. The tractor originally designed to overcome 100hp but when it exceeds 95hp the transfer case will be fail. The main reason of this transfer case failure is vibration. Once the vibration is initiated and propagated to every component of the transfer case it begun to wear finally failure or damage happened to the transfer case.

1.3. Objectives

1. 3.1. General objective

The main objective of the thesis is to analyze the forced dampedsingle DOF vibration of Chery 100hp tractor transfer case

1.3.2. Specific objectives

- ✓ To measure the vibration of transfer case using vibration meter.
- ✓ To identify the high vibration speed zone by comparing the result with ISO standards.
- ✓ To make mathematical modeling of the system for 100hp and 95hp.
- ✓ To input general solution in to Matlab software and generating result for different value of damping factor and compare with the oil type used currently.
- ✓ To identify the modal points and critical frequency at 100hp by using Ansys software.

Chapter Two

Literature Review

This chapter will more detail explain the significant theories and reviewed literature on drive line vibration. Initially, the chapter discussed on the characteristics of vibration and the role of transfer case and transmission housing in tractor and vehicles. This is followed by a brief description of the failure of transmission line due to vibration. Then after a revision on the concepts of vibration characteristics and how to record, analyze and interpret different vibration data. Additionally ways mathematical analysis of vibration were also revised. The last section explains some literatures that are related on this thesis.

2.1 Characteristics of vibration

The term vibration is used to include shocks and movements of any kind. They may result from machine and terrain factors. One of the most critical issues in designing and using agricultural tractors is to minimize the vibration level [13].All rotating machines produce vibrations that are a function of the machine dynamics, such as alignment and balance of the rotating parts [7].

In some cases vibration can be harmful and should be avoided, in other cases it can be the crucial element in the success of a particular engineering process. In mechanical engineering, knowledge about vibration is very important and the measurement, analysis, modelling and prediction of vibration have provided engineers with important tools with which to understand engineering devices [14].

Vibration characteristics are determined by a structure's mass and stiffness values, with damping (ability to dissipate vibrational energy) playing an integral Role by controlling amplitudes. Excitation frequency equal to a damped natural frequency is a resonance. Excitation near a resonance can result in large amplitude responses. Response amplitudes are controlled by damping. With enough damping, the response peak can be completely eliminated. The biggest change is natural frequency has a corresponding unique "mode-shape" with different parts of the structure vibrating at different amplitudes and differing phases relative to one another [15].

Two basic approaches have been used for analysis of machinery vibration over the last twentyyears. Firstly, there has been the development of dedicated hardware, (digital signal analyzers), by a number of companies. These systems, many of which are extremely sophisticated, allow the engineer to connect the vibration measurement sensor directly to the hardware, where the signal is then digitized and may be analyzed by a number of sophisticated techniques. The second approach for vibration analysis was used widely prior to the advent of the dedicated vibration signal analyzers, and involved writing software. This approach has become more important recently because of the need to analyze and interpret the measured vibration of complex machinery for condition monitoring [16].

2.2. Vibration in transmission housing and transfer case.

Heavy vehicle transmission systems are subjected to noise and vibration under excitation condition. Internal excitation forces, meshing forces, load and speed variation and gear defects are major sources of excitation.Noise and vibration reduction in heavy vehicle transmission system is a constant development, because noise and vibration are the two reasons for transmission failure [6].

In the power transmission system, the vibrations generated at gear mesh are transmitted to gearbox housing through the shaft and bearings. Gearbox is a very important component in any machine. So as to increase the life of the gears and bearings in the gearbox assembly, vibration reduction in gearbox casing is of prime importance[17].

Truck transmission gearbox casing is subjected to vibration induced by the harmonic excitation, meshing excitation, and load fluctuations, gear defects, varying speed and torque conditions. Noise and vibrations are the reasons for the transmission failure. The transmission gearbox casing vibrations are the resultant of internal excitation caused by the gear meshing process and transmission error. This internal excitation produces a dynamic mesh force, which is transmitted to the casing through the shafts and bearings. Noise and vibration reduction problem in heavy vehicle industry is a continuous development goal. [18].The vibrations generated at the gear mesh are transmitted to the gearbox housing through shafts and bearings. Casing could be a part of gear box, it provides support to shaft, bearing and the gear loading [19].

2.3. Vibration measurement

Measuring the amplitude of vibration at certainfrequencies can provide valuable information about the accuracy of shaft alignment and balance,the condition of bearings or gears, and the effect on the machine due to resonance from thehousings, piping and other structures. Measurements are taken on the structure of a machine for special purposes such as identifying the type and location of a structural failure or in determining the natural frequency of the structure [7].

2.3.1. Location and direction of measurement:-Two important parameters which significantly affect the result of measurement, are the location and direction of the measurement. Usually vibration is measured at the supports of the rotating parts, exactly the casing when possible. This is because that the vibration of the rotating parts is transmitted only through the bearings casing vibration of bearing themselves is measured at their casings [7].

Accelerometers:-Nowadays, an accelerometer is used as a basic vibration sensor, particularly for measuring on stationary parts of the equipment (rotating machinery), If velocity or displacement is needed, this information can be obtained by integrating the signal from the accelerometer [20].

Vibration severity:-It is generally accepted that between 10 Hz and 1000 Hz velocity gives a good indication of the severity of vibration, and above 1000 Hz, acceleration is the only good indicator. Since the majority of general rotating machinery (and their defects) operates in the 10–1000 Hz range, velocity is commonly used for vibration measurement and analysis[7].According to ISO 10816Standard ISO 10816 provides guidance for the assessment of machine condition for Different types of machines based on two criteria: [21].

1. vibration magnitude and
2. Change in vibration magnitude

Criterion 1:

Vibration Magnitude -Standard ISO 10816 is based on measuring the total Rms value of vibration velocity in the frequency range from 10 to 1000 Hz. The highest value of measurements at different locations is called the vibration severity. The standard defines evaluation zone limits of the vibration severity. Based on these limits, a machine can be classified according to its state into one of 4 zones [21].

- Zone A - Vibration of newly commissioned machines would normally fall within this zone.
- Zone B - Machines with vibration within this zone are normally considered acceptable for unrestricted long-time operation.

- Zone C - Machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.
- Zone D - Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine.

  Classification of the machine into the zone would help to decide about the future operation of the machine and to propose necessary actions. Zone limits are indicative values rather than strict ones and may be adjusted on the base of experience of the manufacturer or operator [21].

Criterion 2: Change in Vibration Magnitude -This criterion provides an assessment of a change in vibration magnitude from a previously reference value measured under steady-state operating conditions. To assess a machine according to the criteria 2 requires long-term monitoring. It should be noted that both sudden increase in vibration and its sudden decrease is considered to be a serious change[21].

Table 1 ISO 10816 RMS velocity classification [39]

ISO General Standard (10816-1:1995)				
RMS vibration velocity (mm/s)	Class1	Class2	Class 3	Class 4
0.28	A	A	A	A
0.45	A	A	A	A
0.71	A	A	A	A
1.12	B	A	A	A
1.8	B	B	A	A
2.8	C	B	B	A
4.5	C	C	B	B
7.1	D	C	C	B
11.2	D	D	C	C
18	D	D	D	C
28	D	D	D	D
45	D	D	D	D

Table 2 vibration severity zone and class [39]

Zone A	Newly commissioned machinery excellent condition.
Zone B	Good-Acceptable for Unrestricted, long-term operation.
Zone C	Unsatisfactory for long-term, continuous operations (alert level).
Zone D	Bad. Sufficient severity to cause long long-term damage to machine (alarm level).
Class 1	Very small machinery or parts of machinery (20HP or below).
Class 2	Small machinery (20-100HP) on rigid foundation.
Class 3	Large machinery mounted on rigid &heavy foundations.
Class 4	Large machinery mounted on relative soft foundations.

These standards represent a starting point or rough gauge to apply to most machinery using an absolute vibration reference level by machine condition that is based on both experience & historical data across many industries and machine types [22].According to the above standard the Chery 100Hp falls in range of class 2 because the horse power is in between 20 Hp up to 100 Hp.

2.4. Random vibration or Nondeterministic vibration

If the excitation is non-deterministic or random; the value of excitation at a given time cannot be predicted. In these cases, a large collection of records of the excitation may exhibit some statistical regularity. It is possible to estimate averages such as the mean and mean square values of the excitation[23].

To compute the RMS values there is a special formula for computing the area of triangles on log-log graph paper. The definition of a straight line on log-log graphs between two points (f_1, a_1) and (f_2, a_2) is a power relationship, where the slope is the exponent, and the offset is the multiplicative factor [32].

The slope and offset are computed as follows

$$a = \text{offset} * f^{\text{slope}} \quad ---------------- \quad (2.1)$$

$$\text{slope} = \frac{\log(a2) - \log(a1)}{\log(f2) - \log(f1)} ------------- (2.2)$$

$$\text{offset} = \frac{a1}{f^{\text{slope}}} -------------------- (2.3)$$

Given this slope and offset we can integrate from f_1 to f_2 to compute the area under the line

$$\text{area} = \frac{\text{offset}}{\text{slope} + 1} * (f2^{\text{slope}+1} - f1^{\text{slope}+1}) ------ (2.4)$$

So, for each pair of points we can use equations (2.3) or (2.4) to compute the area under the curve. The total area under the curve will then be the sum of the individual area calculations between each pair of points, and this sum is the mean-square acceleration. We take the square-root of the result to get the RMS acceleration level.

Adding up the area values gives the mean-square acceleration and then taking thesquare root of this result gives the overall RMS value. Which is the acceleration units of this result will be the square root of the acceleration density units. When using density units of $(m/s^2)^2/Hz$, the result will be in m/s^2.Since most curves are plotted as straight lines on log-log paper, calculating the area under the sloped lines is more complicated than for the region of constant PSD[24].

2.3.1. Random velocity calculation

The RMS velocity can be computed in the same manner; however the point numbers need to be converted from acceleration squared/Hz units to velocity squared/Hz units, with appropriate unit conversion, if required. This conversion is done by the relationship between velocity and acceleration for a sine wave of a given frequency [23].

$$\text{velocity} = \text{accelaration} * \frac{1}{2\pi f} ------------ (2.6)$$

As a result the velocity density becomes

$$v = \frac{\text{offset}}{(2\pi)^2} * f^{\text{slope}-2} -------- (2.7)$$

This can be integrated from f1 to f2 to get the area under the velocity line

$$\text{area} = \frac{1}{(2\pi r)^2} * \frac{\text{offset}}{\text{slope} - 1} * f1^{\text{slope}-1} - f2^{\text{slope}-1} ------ (2.8)$$

To get the mean-square velocity the velocity will be added, and take the square root to get an RMS velocity value for the random spectrum. Also, when using acceleration units in G's, a conversion factor needed to be applied to get a suitable velocity unit. [23]

$1 \text{ G} = 386.09 \text{ in/s}^2 = 9.80665 \text{ m/s}^2$

2.4. Mathematical Analysis

2.4.1. Methods for mathematical modeling

The analysis of vibratory system usually involves mathematical modeling derivation of the governing equation, solution of the equation and interpretation of the result.

- Mathematical Modeling**:-**The purpose of mathematical modeling is to represent all the important features of the system for the purpose of deriving the mathematical or analytical equations governing the system's behavior. The mathematical model should include enough detail to allow describing of the system in terms of equation without making it to complex.
- Derivation of governing equation**: -** once the mathematical model is available the principal dynamics is used and the equation that describe the vibration of the system will be drive. The equation of motion can be derived conventionally by drawing the free body diagram of all the mass involved. And indicating all externally applied force, the reactive force, and the inertia force. The equation of motion of vibrating system are usually in the form of a set of ordinary differential for both discreet and partial differential equation for a continuous system. Several approach are commonly used to derive the governing equation. Among them are Newton's second law of motion, D'Alemert's principle, and the principle of conservation of energy.
- Solution of the Governing Equation**: -** The equation of motion must be solved to find the response of the vibration system. Depend on the nature of the problem, the following techniques will use for finding the solution: Standard method of solving differential equation, Laplace transform method, matrix method and numerical method.
- Interpretation of the Results**:-**The solution of the governing equation gives the displacements, velocities, and accelerations of the various masses of the system .these result must be interpreted with a clear view of purpose of the analysis and the possible design implication of the result[25].

The dynamic behavior of mechanical system described by what we call second order ordinary differential equations.In order to be able solve these equations, it is imperative to have a solid background on the solution of homogeneous and nonhomogeneous ordinary differential equations. For homogeneous ordinary differential equations represent the 'Free Vibrations' and the non-homogeneous ordinary differential equations represent 'Forced Vibrations' [26].

Forced vibration: -Ifasystemissubjectedtoanexternalforce(often,arepeatingtype offorce),theresultingvibrationisknownasforcedvibration.Theoscillationthatarisesin machinessuchasdieselenginesisanexampleofforcedvibration.

Ifnoenergyislostordissipatedinfrictionorotherresistanceduringoscillation,thevibrationisknownasun dampedvibration.Ifanyenergyislostinthisway,however,itiscalled dampedvibration.Consideration ofdamping becomesextremelyimportantinanalyzingvibratorysystemsnearresonance [27].

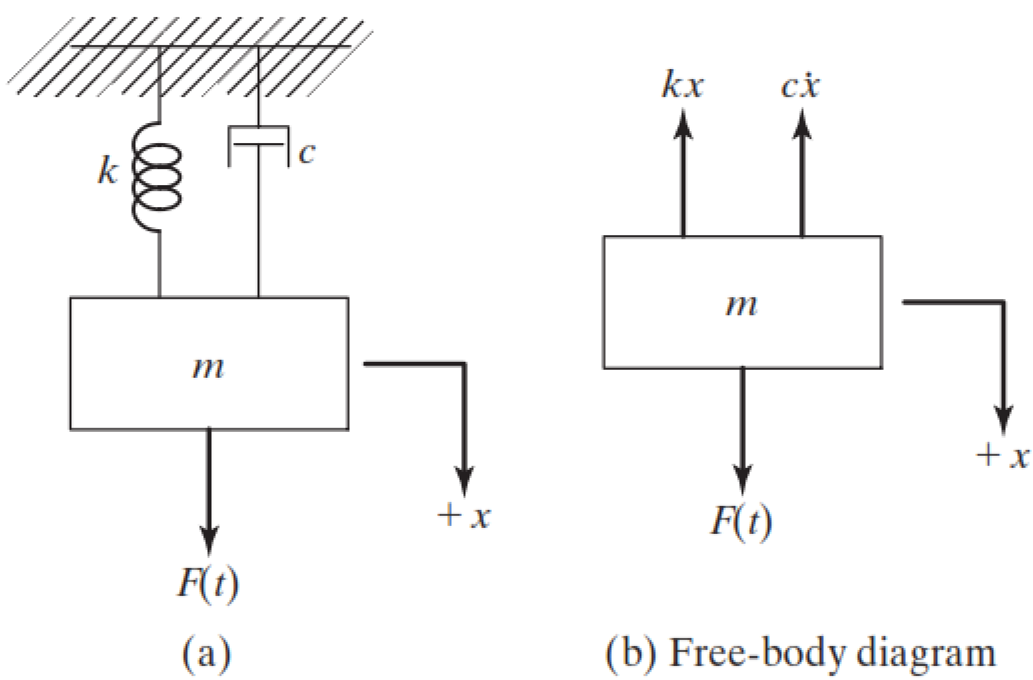


Figure 2 Free body diagram for forced damped SDOF vibration

If a force F (t) acts on a viscously damped spring mass system as shown in figure 2and the equation of motion will be obtained using Newton's second law:

$$m\ddot{x} + c\dot{x} + kx = X(t) - - - - - - - - - (2.14)$$

The above equation is non homogeneous, its general solution x (t) is given by the sum of the homogeneous solution,$x_h(t)$, and the particular solution,$x_p(t)$.the homogeneous solution equation is.

$$m\ddot{x} + c\dot{x} + kx = 0 - - - - - - - - - (2.15)$$

The homogeneous equation represent the free vibration while the former equation is non homogeneous represents forced vibration.

Response of undamped System under harmonic force

If a force$F(t) = F_o cos\omega t$ acts on the mass m of undamped system, the equation of motion reduced to,

$$m\ddot{x} + kx = F_o cos\omega t - - - - - - - - - - - (2.16)$$

Then the homogenous solution of equation becomes

$$x_h(t) = C_1 cos\omega_n t + C_2 sin\omega_n t - - - - - (2.17)$$

$\omega_n = (\frac{k}{m})^{0.5}$The natural frequency of the system. Because the exiting force F (t) is harmonic, the particular solution $x_p(t)$ is also harmonic and has the same frequency.so the solution can be assume in the form of.

$$x_p(t) = Xcos\omega t - - - - - - - - - - - - (2.18)$$

Where X is a constant that denotes the maximum amplitude of $x_p(t)$.

After substituting the last two equation and solving X we obtain

$$X = \frac{F_0}{k - m\omega^2} = \frac{\delta_{st}}{1 - \left(\frac{\omega}{\omega_n}\right)^2} - - - - - - - - - - - - (2.19)$$

$$\delta_{st} = \frac{F_0}{k}$$

Denotes the deflection of the mass under a force F_0 and also called static deflection because F_0 is a constant force .the total solution of becomes

$$x(t) = C_1 cos\omega_n t + C_2 sin\omega_n t + \frac{F_0}{k - m\omega^2} cos\omega t ------(2.20)$$

Using the initial condition x (t=0) =x_0 and $\dot{x}$(t=0) =$\dot{x}$ we find that

$C_1 = x_0 - \frac{F_0}{k-m\omega^2}$, $C_2 = \frac{\dot{x_0}}{\omega_n}$And hence

$$x(t) = \left(x_0 - \frac{F_0}{k - m\omega^2}\right) cos\omega_n t + \left(\frac{\dot{x_o}}{\omega_n}\right) sin\omega_n t + \left(\frac{F_o}{k - m\omega^2}\right) cos\omega t --- (2.21)$$

The maximum amplitude X is expressed as

$$\frac{X}{\delta_{st}} = \frac{1}{1 - (\omega/{\omega_n}^2)} -------------(2.22)$$

The quantity of $\frac{X}{\delta_{st}}$ represent the ratio of dynamic to the static amplitude of motion and is called the magnification factor, amplification factor or amplitude ratio and ω/ω_n is the frequency ratio r

Response of damped system under harmonic force

If the forcing function is given by

$$F(t) = F_0 cos\ \omega t -------------(2.23)$$

The equation of motion becomes

$$m\ddot{x} + c\dot{x} + kx = F_0 cos\ \omega t -----------(2.24)$$

The particular solution is expected to be harmonic; and can be assumed

$$x_p(t) = Xcos(\omega t - \varphi) -------------(2.25)$$

Where X and φ are constants to be determined and they denote amplitude and phase angle by the substituting the above two equations

$$X[(k - m\omega^2)\cos(\omega t - \varphi) - c\omega \sin(\omega t - \varphi)] = F_0 cos\omega t --(2.26)$$

By using the trigonometric relation

$$\cos(\omega t - \varphi) = cos\omega t cos\varphi + sin\omega t sin\varphi = F_0 - -(2.27)$$

$$\sin(\omega t - \varphi) = sin\omega t cos\varphi - cos\omega t sin\varphi$$

Equating the coefficients of $cos\,\omega t\; and\; sin\omega t$ on both sides of the resulting equation, we obtain

$$X[(k - m\omega^2)cos\varphi + c\omega sin\varphi] = F_0 - - - (2.28)$$

$$X[(k - m\omega^2)sin\varphi - c\omega cos\varphi] = F_0 - - - -(2.29)$$

And gives

$$X = \frac{F_0}{\left[(k - m\omega^2)^2 + c^2\omega^{2^{\frac{1}{2}}}\right]} - - - -(2.30)$$

$$\varphi = tan^{-1}(\frac{c\omega}{k - m\omega^2} - - - - - (2.31)$$

By inserting the value of $X\; and\; \varphi$ the particular solution will be found and dividing the above equation by kand making the following substitution.

$$\omega_n = \sqrt{\frac{k}{m}}\; undamped\; natural\; frequency\; - - - -(2.32)$$

$$\xi = \frac{c}{c_c} = \frac{c}{2m\omega_n} = \frac{c}{2\sqrt{mk}}; \frac{c}{m} = 2\xi\omega_n - - - -(2.33)$$

$$\delta_{st} = \frac{F_0}{k} = deflection\; due\; to\; the\; static\; force - - - - - (2.34)$$

$$r = \frac{\omega}{\omega_n} = frequency\; ratio - - - - - -(2.35)$$

Finally

$$\frac{X}{\delta_{st}} = \frac{1}{\left\{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2\xi\frac{\omega}{\omega_n}\right]^2\right\}^{\frac{1}{2}}} = \frac{1}{\sqrt{(1 - r^2)^2 + (2\xi r)^2}} - - - -(2.36)$$

Viscosity and material property

Gear oil is a lubricant prepare mainly for transmission, transfer case in automobile. It is of high viscosity for better protection of gear. The high viscositycertifies transferal of lubricant throughout the gear train.Lubrication oil plays a decisive role to maintain a reliable and efficient operation of gear transmissions. The change of oil viscosity due to oil degradation will cause corresponding changes in both static power consumption and dynamic behavior of a gearbox transmission system.

According to the investigation for the damping capacity data for damping capacity and material microstructure the result indicates that damping in steel and cast iron examined strongly and comparison has made cast and alloy steel cast iron exhibit better damping in most cases[32].

The effect of mechanical properties of materials on natural frequency and mode shapes of heavy vehicle gearbox transmission casing were studied for the vibration response study of casing. Grey cast iron grade FG 260, Structural Steel, Al alloy and Mg Alloy materials were analyzed for different twenty mode. According to the study the frequency Al alloys (1291-3829) band for all materials are grey cast iron (1002.5-2954.8), structural Steel (1306-3879), and Mg Alloys (1273-3784).The research work has concluded that the mechanical properties are directly related with natural frequency and vibration mode shapes of structural rigidity structural steel have excellent properties [28].

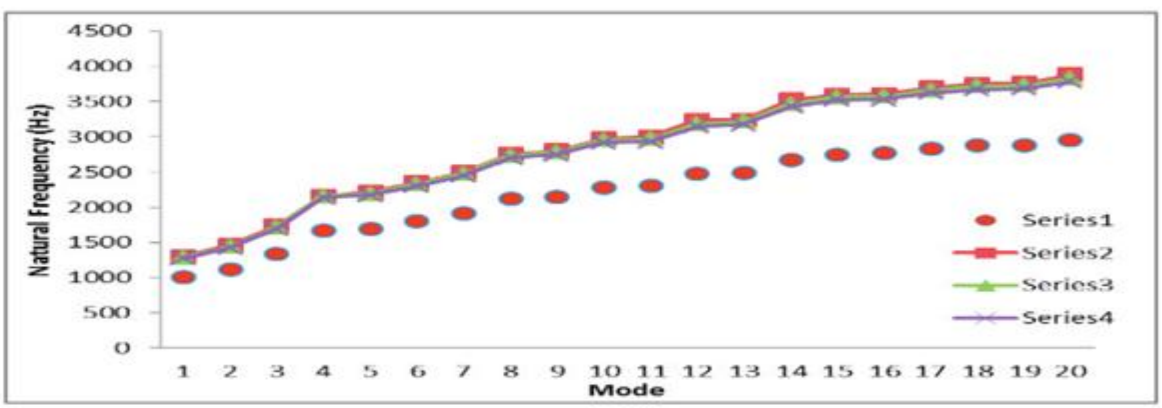


Figure 1.1 Material property versus natural frequency

2.5. General transmission and transfer case vibration research

Andrew J. William et.al (2009), Made a design on transmission housing. They designed gear train and the shafts in accordance with standard design data. The torsional critical speed of the rotor system and the lateral critical speeds of the shafts were obtained using ANSYS. They completed the analysis and they concluded that the housing design plays a very significant role in the industrial scenario.

Christian R et al have made a vibration assessment on tractor trailer and tractor mounted chipper on 2010. They have measured vibration and noise on operator's seat and noise inside the chipper cab during chipping, driving between wood piles, and operational delays separately and analyzed by considering factorial design set up variant which are tractor-trailer and truck-mounted of two chipper models from different manufacturers during chipping of softwood and hardwood tree species. The researchers recorded while the tractor is driving on forest road and during operational delay with chipper and on idling condition. Finally they founded that chipping hard wood produce high vibration than chipping softwood. From the analysis the writers unambiguous about there is no difference between tractors version of both manufacturers but the vibration value of truck mounted truck from manufacturer was significantly higher than Manufacture A. At the end they founded that the difference between the vibration magnitude of hardwood and softwood was 14% on average when chipping were measured below the exposure action value of 0.5 m/s .and from their Statistical analysis they showed that vibration during driving from one wood pile to another exceeded the vibration occurring during chipping and vibration exposure during driving of tractors. For driving and operational delays there was no statistical difference found for the x- and the z-axes. In those cases the least vibration occurred in y-axis.

Shrenik M.et.al (2013) had made a project that targets the study of differential gear box casing of pick up van vehicle modal and stress analysis andto find the set of stiffeners of which the natural frequency coincides with the excited frequency. The analysis of the casing made using die cast ALSi132 material equipped with sets of stiffeners on either side. They create a model in Pro-E Wildfire3.0 and meshed using hyper mesh 11.0. The Stress analysis and natural frequencies of model in free-free conditions are calculated using ANSYS 12.0. They did experimental modal analysis of actual component in free - free condition. Based on the analysis of the project the authors concluded that by the comparison of the natural frequency and gear mesh frequency or operating frequency there is no resonance condition found. The natural frequency of model without stiffeners and with stiffeners had compared. From the resonant point of view, the proposed model is safe from the stress limit because the chance of stress will be high without stiffeners.

Performance Assessment of a Tractor Driver Whole Body Vibration Measuring System (Iman Ahmadi 2013).

In this paper the author made the whole body vibration, fabrication and evaluation in order to record the tractor whole body. The researcher approved testing of the system in static and dynamic condition .during recording he attached the sensor attached to tractor driver in X, Y, and Z axis to measure the layout, longitudinal, transverse and vertical vibrations of the sensor as functions of time. He calculated Parameter by using the measuredvibration time domain data. Therequired program for the calculation of this index waswritten in code writing environment of the Matlabsoftware. AndPractical evaluation of the system was performed by installing the sensor of the system on the seat of an MF285 tractor plowing with moldboard and rotary plows withforward speed of and the measured accelerationswere analyzed. Lastly, using the measured vibration time historiesas input parameter of the conversion program from timedomain to frequency domain and executing the programin the Matlab software environment, dominantfrequencies of the vibration data were taken out. He obtain the graphical result as the dominant frequency rangetransmitted to tractor driver is (0, 3) Hz.

Ashwani. K et.al perform free vibration analysis on truck transmission housing (2014) they used finite element simulation using ANSYS 14.5 software to study the vibration pattern for first twenty modes. They designed the transmission housing by via Solid Edge Software. After the analysis they found that the natural frequency of vibration varies from 1306.3 Hz to 3879 Hz. The geometry of transmission housing is examined using fixed-fixed boundary condition by restricting nodal displacement. Grade FG 260 Cast iron selected for the housing. The analysis outcomes indicated that transmission housing is exposed to Axial bending vibration, torsional vibration and Axial bending with torsional vibration.

Dynamic Vibration Characteristics of truck transmission investigated on 2014 by Ashwani kumara and his friends. They focus on simulation of relation between dynamic vibration of transmission and fixed constraint of vehicle frame 37 connecting bolts. They considered the first ten natural frequencies and their corresponding mode shapes. They used Grey cast iron HT200 for casting material to find the dynamic response. FEA based numerical simulation was used by applying reciprocity principle to determine the main responsible force transmitting to the casing. Solid Edge and Pro-E [14-15] software was used for design .Ansys 14.5 workbench module is used for free vibration simulation. Accordingly the simulation shows that the natural frequency of one bolt unconstraint condition varies from (1637.2 – 2674) Hz. They proved this result with experimental result available on the literature. Two part study has been made the first one is with zero displacement constraint based boundary condition while the second used fixed constraint based boundary condition. Based on the analysis the authors confirmed that there is a variation of (2-6) % fall in natural frequency due to looseness of one connecting bolt from all five positions.

Toshita Dhande1, R. B. Patil (2015) had made a design optimization and analysis of gear box casing in order to reduce wear of gear and bearing due to vibration. Specifically, on the top gear box cover vibration characteristics by both analytical and experimental technique. They produced the model by SOLID EDGE and analyzed using Ansys software for resolving of natural frequency response. The result confirmed by FFT analyzer. Finally, they reached to the conclusion of simulation and experimental result are close agreement with each other.

The vibration analysis of two wheeler gear box casing was made by **Sagar. P** and his friend on August 2015. The finite element simulation was performed using NASTRAN software while the design of the casing was made by CATIA. The objective of the analysis was finding the natural frequency to prevent the resonance. They studied first ten mode of vibration pattern by constraining the displacement of bolt of transmission casing hole. They finally reached to the conclusion of gearbox casing is subjected to Axial bending vibration, torsional vibration and axial bending with torsional vibration.

Sachin S. Pawar et al (2015) made Design & modal analysis of planetary gearbox casing they give more weight on vibration analysis of gear housing of planetary gearbox by determining natural frequencies, mode shapes with the help of modal analysis. Furthermore frequency response measured at different excitations and the responses were analyzed with the help of FFT.

Gravalos.et.al made vibration measurement and analysis of agricultural tractorswhile they are operating on traditional and electronic regulator. They recorded vibration and took measurements from different model of agricultural tractors which are four wheel drive tractors greater than or equals to 70Kw under different operating condition. After the measurement they got the acceleration magnitude in terms of RMS value. An average acceleration of 0.0038m/s2 along in X, Y, and Z direction. They made a comparison between traditional regulator and electronic regulator. They confirmed that electronic regulator cause high acceleration than traditional regulator especially during transportation. Finally they concluded that when tractor aggregated with a plough and it was working with electronic regulator, they observed the average recorded acceleration a little higher than when using a traditional regulator. They observed that electronic regulator can cause higher accelerations over all three axes of the tractor’s driving seat comparing it with the traditional regulator. During its operation, the electronic regulator adjusts the engine speed to stricter upper and lower limits and thus operates faster than the traditional regulator. This results in higher speed adjustment per unit of time, which in effect leads to higher corrective accelerations of the tractor. These accelerations are transmitted through the body of the tractor and induce higher modes of vibrations in all three directions.

Roberto. D and his friends made an assessment Between ISO 5008 and field whole body vibration tractor values. They made frequency analysis on agricultural tractors operated on an artificial test track and on different types of ground. They have conducted field experiment on three brand new tractors direct from the factory. The first tractor which was a two-wheel drive vehicle equipped with radial tires, and the remaining two tractors are four wheel drive equipped with low profile tires. On the test track of grass, harrowed clay, asphalt and farm road. They have made A total of 360 tests were carried out. To improve the accuracy of RMS value analysis, for every test, they recorded and investigated acceleration values. The authors concluded that ISO 5008-smooth track may sometimes reproduce same vibratory condition registered over the other surface in terms of RMS acceleration value and spectral trend. Their final study proved that the smoother ISO track simulate some usual vibratory condition those found when the tractor is run without a trailer across grass farm roadway. According to the outcome found from the first comparison three different tractors running on several agricultural surfaces and on an ISO-smooth track are encouraging.

X. Lai et al 2016 and his associates made finite element transient structure analysis of startup of sugarcane harvester transfer case. In order to simulate the transfer case's startup dynamic process and measure the instantaneous load values they utilize virtual prototype technology. They revealed that from Stress distribution and load variation analysis during startup, the stress concentration of node load increases rapidly with gear speed rising transiently. The writers explored that during startup the impact load of the transfer case increases far greater than in the steady state which deteriorates the structure stress condition seriously. The strength of the transfer case is then amended by structural improvement. To reduce maximum stress by 36.8%, the maximum deviation by 18.5% and the vibration. They recommended by calculating the Engagement Force to 30%, the bearings working reliability will be improved. Finally they improved the transfer case by increasing thickness, welding hydraulic pump directly to the front cover replacing shaft spline key by flat key and reducing the transfer case vibration by damping rubber pad. The result shown that the maximum stress reduced 36.8% and the maximum deformation was 18.5% less benefit for diminishing the vibration and noise. RMS of the vibration acceleration reduced 3% to 30% in three points and the peak vibration acceleration value reduced from 15% to32%. This shows that the strength of the transfer case is definitely improved

Industrial helical gear box casing was analyzed by Balasaheb.S on October, 2016. The content of the research is the design analysis of three stage industrial gear box casing using FEM method and to discover effective design of gear case with relevance cost. The author prepared 3D model by using Pro-E creo2.0 and preprocessed by Hyper mesh. The researcher analyze the gear box housing both statically and dynamically using Ansys 14.5 solver. FEA also be carried out on design of the gearbox housing to check whether the design is safe or not. The casing materials that were selected are FG260 and FG220.After all the author confirmed that the structure safety in a way that the static analysis of the overall quantity of stress and displacement is in permissible limit and by comparing natural frequency with gear mesh frequency thy confirmed there is no resonance condition found so the structure is safe.

Thshar .M and his friend made the static analysis of gear box casing. On the research they try to explain the use of analysis software using in design of gear box casing. The authors mainly focused to analyze and compare the two soft ware’s ALTARI/HYPER MESH/ and ANSYS.

They made a comparison between these soft wares. First authors made a CAD modeling then they imported to the simulation. Proper process plan had been taken to determine boundary condition, study the material property and load pattern by studying assembly system and veichle packing. According to the analysis they considered the Al alloy to be the housing material. They also mention the stress is Von misses stress and compared the two software results which are 119 MP by ANSYS and 115 MP ALTARI software. After the result has gained they did not give any interpretation they just tabulated the outcomes.

Ashwani Kumar et,al (2016):-This paper aims to make weight calculation and modal analysis of gear box housing by picking out four different materials. The authors selected Grey cast iron FG260, Grey cast iron HT200, structural steel and Al alloys for transmission housing. They studied vibration signature patterns for first twenty modes. Throughout this research they neglect oil pressure and sensitivity analysis. They examine free vibration for first twenty inherent natural frequencies and mode shapes by applying free -free boundary condition with no constraint on the moment of transmission housing. The authors confirmed the outcome of the research with existing experimental result obtained in literature. They used Solid Edge and Pro-E software for complex geometric modeling. FEA based software Ansys 14.5 is used for modal analysis. Based on the investigation modal analysis results showed that lateral vibration, axial bending vibration, torsional vibration, and axial bending with torsional vibration. The frequency range for all materials is (1669.5-4655) Hz for all material this range of frequency variation is same (500-5000) Hz as the experimental result obtained by the (24). (2009) concluded that, all materials are suitable for transmission housing manufacturing.

MahaKarray,et.al (2017) the paper aims to investigate the modal properties of gear box that composed of one stage spiral bevel gear and a two stage helical planetary gear used in a bucket wheel excavator gearbox. The authors made a lumped-parameter model to obtain equations of motion and to solve eigenvalue problems. They presented the model in low in low-frequency and high-frequency bands and studied the Distributions of modal kinetic and strain energies. After the investigation they founded that. The representation of helical planetary gear system is three-dimensional lumped-parameter model with six degrees of freedom per gear and the shaft body supported by bearings. They acquired the natural frequencies can be categorized as three main mode classes: coupled modes, bevel gear modes and planetary gear modes. They have seen at the modal kinetic and strain energy distributions, another classification emerges according to the dominant energy in the system for each natural frequency.

Chapter Three

Materials and Methods

3.1Materials

3.1.1. Vibration meter

The vibration meter model is SDL800 vibration meter or data logger which has Basic accuracy of ± (5% + 2 digits); Meets ISO2954 standard the meter shown as on figure This instruments meets the Standard gives an accurate recording of root-mean-square (R.M.S.) vibration displacement, velocity and acceleration value that is defined as a measurement unit with a wide frequency range of 10Hz to 1kHz.

Figure3.1.Vibration meter model SDL800

3.1.2. Transfer case

Cherry 100 hp tractor transfer case is the examined transfer case which is damaged part of the tractor due to vibration. The main purpose of this transfer case is to distribute engine power and transfer it in to all the four wheel by the rear and front axle. During field operation the traction force need to be increased while the speed will be diminished for this purpose the two wheel drive should be changed in to four wheel drive to perform field operation. In this kind of situation this transfer case obtains power from the transmission and transfer it in to four wheel of the tractor this helps the operator to change the tractor from two wheel drive in to four wheel drive by moving the gear selector fork by moving slide. Tractor remains at four wheel drive at the entire field task to increase traction force and with stand off-road condition and it will be turned in to two wheel drive whenever the field operation finished and the tractor leave the field and is driven on asphalt or smooth surface. The transfer case works by using gear driven mechanism these gears are spur gears which are meshed by using constant mesh. The examined transfer case has totally 36 components including every small elements. Shown on figure

Figure 3.2 Components of the transfer casing

3.1.3. Description of the Examined Transfer Case

The transfer case has 36 components including shift fork shaft, intermediate gear shaft, transfer case gear shaft ,bearings, gears both driven and driving gear, sleeve, spacer bush and others shown on figure 5.The gears are spur gears meshed in a constant mesh. The dimension of the transfer case is length (14.5* 12.2*18.5) cm.

Figure 3.3 Chery 100hp failed transfer case

3.2. Methods

3.2.1. Data collection

The data collection process had been planned since the title got acceptance. The collected data includes facts, events, records, observation, and measurements both from internal and external of the study area which is AAMI. Both primary and secondary data gathering techniques were employed. Even if its time consuming and costly.

The data have been gathered by shifting two gear levers main and auxiliary shifting levers starting from one up to four gears at both high and low speed zones. The measured vibration by the vibration meter was transferred to the computer for further software analysis .From the Matlab analysis there is result that gives 8 charts and graphs shows the frequency and amplitude of the imported data. These 8 charts are interpreted so as to identify the main gear zone that can have high frequency.

This data was recorder for four high and low gear and speed zone which are 1^{st} 2^{nd} 3^{rd} 4^{th} low and 1^{st} 2^{nd} 3^{rd} 4^{th} high speed on the field and one idle measurement on asphalt or smooth surface. Totally nine measurements had been recorded. These recorded data were imported in to Matlab in order to extract the high vibration gear zone by interpreting the charts from the output of the software.

3.2.2. Field visit

The total day spent for industry visit is 10 days starting from September 20 up to march 28 the most industry visit is taken place in the research area AAMI the company provided essential and very significant information and data in line with the objective of the thesis. At the beginning different problems projects were visited in order to find the specific problems that are expected to be solved and to start MSc thesis. After discussion of the available problems needed to be solved or studied with an advisors the study was started on the failed Chery 100 hp tractor model transfer case analysis due to frequent damage and repair of this transfer case as per evidence from the operators, chief mechanics and marketing sectors. Once the analysis was stared serious measurements and vibration recording was taken on this particular area.

The other industry visit was taken place at is at Akaki basic metals industry this visit was taken three days starting from letter of acceptance for spectrometer testing which is on Tuesday March 21,2018. The sample spectrometer sample measurement of the failed transfer case in order to identify the material type and material composition.

3.3.3. Field measurement

Field measurement taken place around Adama town. Chery 100 hp tractor is taken to the field with disk plow implement. The disk plow was attached on it by using three point linkage and the PTOwas adjusted. After the connection and attachment of the implement get finished the tractor was checked for oil and fuel level, battery, tire and functionality of hand brake.

Figure 3.4 Implement attachment on chery100 hp tractor

The vibration meter SD card were inserted in to the vibration meter and formatted to make sure that the data is going to be gather is the only data off the transfer case. After the SD card was formatted the vibration meter were ready for measurement. Before the measurement were started the vibration meter settled to zero and the magnet tip was attached particularly on the transfer case.

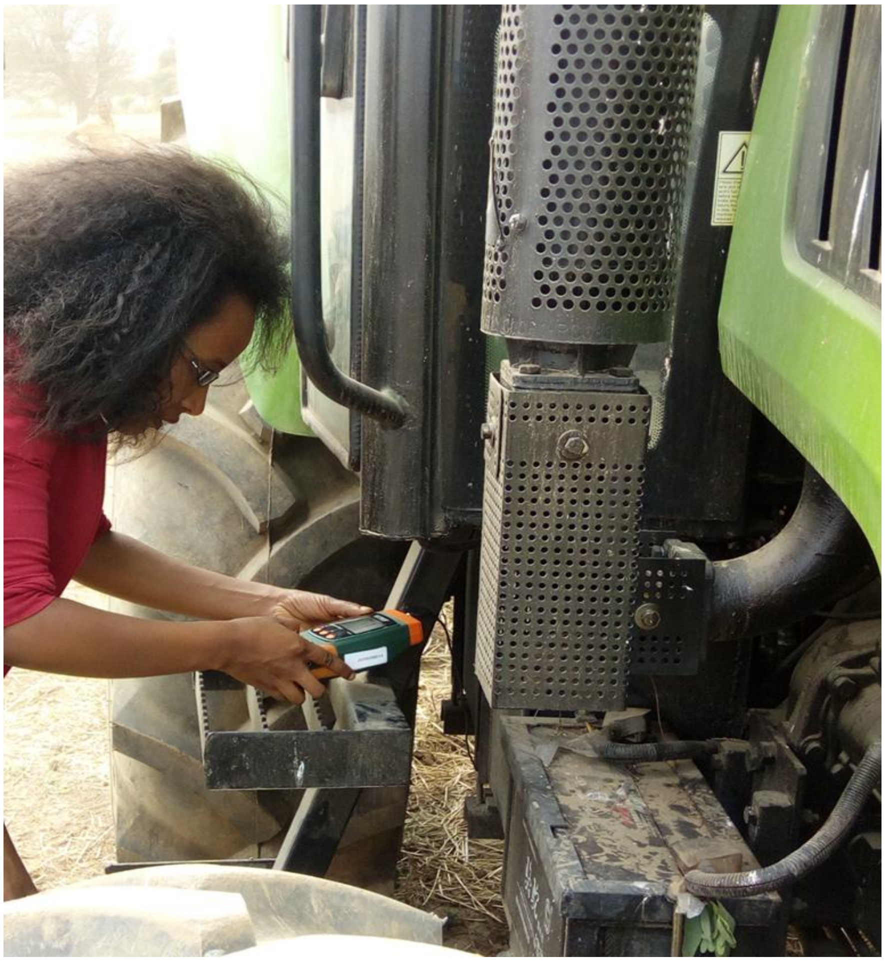


Figure 3.5the vibration meter magnet tip attachment to transfer case

3.3.4. Vibration recording

Test were conducted on Chery 100 hp tractor transfer case tractor test were performed on $1000m^2$ area around Adama town farm field with disk plow implement attached on it for high and low speed zone at 1st 2nd 3rd and 4th gears.

The forward speed of the tractor was set up is changed from 700 rpm up to 2100 rpm .Each measurement was taken in the same position of the truck and two reputation was made to confirm the accuracy of the measurement. To measure the vibration the type of instrument used is Model SDL800 vibration meter has basic accuracy of ± (5% + 2 digits); and Meets ISO2954 standard.

Figure 3.6 Field vibration measurement of Chery 100hp transfer case

After the tractor start plow the land the vibration meter recorded vibration in m/s2 for eight different speed zone and gear rotation. The vibration meter give data in excel format and after recorded each data was transferred in to the computer.

3.3.5. Vibration analyzing

The vibration meter gives a data in excel format these data is the RMS value or root mean square in m/swhich is the velocity value. The data imported in to Matlab software for further analysis.. According to the result the Matlab gives eight charts.

3.4 Technical specification

Technical specification was taken for the sake 3D modeling on Catia soft were each component of the tractor model transfer case disassembled and measured using caliper and meter in order to get proper design.

Figure 3.7 Measurement taken for technical specification

3.5. Spectrometer test

Knowing the material type is mandatory to study vibration the sample of transfer case was taken from AAMI for the purpose test on spectrometer. The spectrometer test were performed at Akaki Basic Metal Industry .The result indicated that the material type was unable to identify from the industry available resource of the company but the composition are identified as follows.

Table 3 Material composition of Chery 100hp transfer case

Minerals	C	Si	N	P	S	Cr	Mo	Ni	Al
Percentage	1.53	1.35	0.75	0.04	>0.096	>0.071	0.00031	0.006	0.003
Minerals	Cu	Ti	V	W	Sn	As	Fe		
Percentage	0.072	>0.026	0.026	<0.04	0.013	0.016	95.9		

Material Type Identification

Material test indicates that the material type cannot be branded from available hard were catalog at Akaki Basic metals industry.With the intention of finding the material type other source should be visited that can almost fulfill the above elementsin percentage. Because of the difficulties of the material composition some approximation has been done in order to stick on a certain material type.

1. Iron = 95.9%

The high amount of iron content on the material indicates that the material is metallic no further exploration is needed to tell the material is metallic. Once the material is metallic the next procedure is what type of metal is this. To identify the metal type other elements of the composition need to be verified with the literatures.

3. **Carbon =1.53 %**

All metals fall within one of two categories either they contain iron and are considered ferrous metals, or they don't have iron and considered nonferrous metals. [29]The above composition indicates that high amount of iron exists which means the material is metal. Cast iron differs from steel principally because its excess amount carbon which exceeds 1.7%.

Items	Percent carbon (%)
Pig Iron	4.5
White cast Iron	3.5
Gray cast Iron	2.5-4.5
Malleable cast iron	2-3.5
Tool steel	0.9-1.7
High carbon	0.5-0.9
Medium carbon steel	0.3-0.5

Cast steel	0.15-0.6
Low carbon steel	Up to 0.3

The amount of carbon exists on the material is 1.53% does not reach or exceeds 1.7 % that indicates the material is steel which is tool steel. Steels can be low alloy or high alloy steel. High alloy steels are (stain less steel and tool steel). As a result tool steel can be categorized as high alloy steel. Thus the material directs to be high alloy steel.

3. Chromium (Cr)-the amount of chromium presents in this material is above 0.071%.

4. Molybdenum (Mo)-the amount of Molybdenum presents in this material is 0.00031%.

5. Nickel (Ni)-the amount of Nickel presents in this material 0.006 %.

6. Vanadium (V) - the amount of Vanadium presents in this material 0.026%.

The above material element value verified with AISI 4140 which has Chromium maximum 0.15, Nickel 0.2 maximum, Molybdenum 0.1max and Vanadium 0.1 max. [30]

The transfer case material type is found that AISI Alloy steel 4140 which has the physical properties are as follows.

Table 4Physical property of AISI Alloy steel 4140

Physical Property of AISI Alloy steel 4140	
Density	7.85g/cm^3
Melting point	1416^{o}C
Tensile strength	655Mpa
Yield strength	415Mpa
Bulk modulus	140Mpa
Shear modulus	80Gpa
Elastic modulus	190-210Gpa
Poisson ratio	0.27-0.30
Elongation	25-75%

Chapter four

Mathematical Modeling

4.1 Physical modeling

Power transmission system of Chery 100 hp tractor works with the same manner of any other tractor or automotive system. During two wheel drive the engine power is on the two tractor wheels and the other two wheels allows to spin. The power came from the engine directly transferred in to clutch to engage the drive line and transfer the power towards the gearbox, then the power transferred in to the gear box for reduction of motor speed in to vehicle speed. Finally the differential allow the rear axle to rotate with respect to each other and power transferred in to wheel. Whenever the traction force needed to be increased the power needed to be split in to four that will give the maximum drive traction and accomplished by engaging the differential gear in to transfer case. From the transfer case the power will be transferred into front axle to make four wheel drive. This will be shown on simple flow chart below.

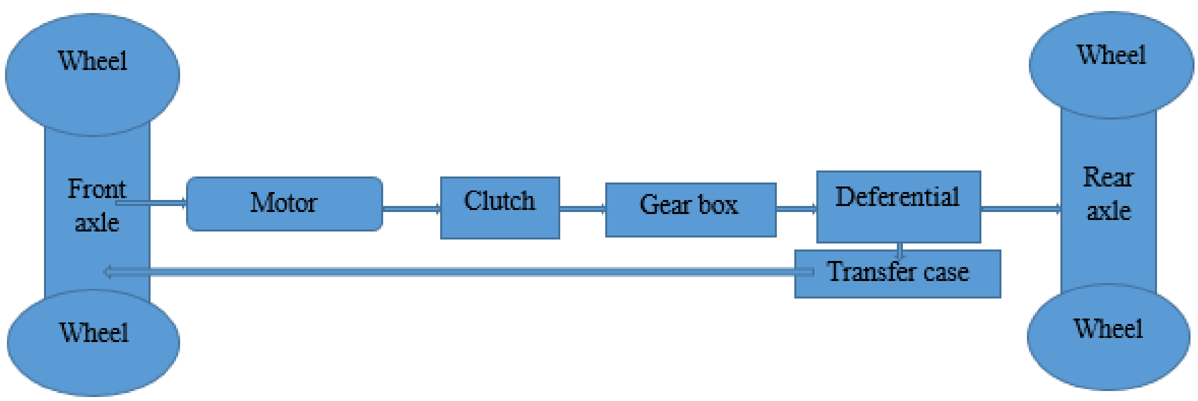


Figure 4.1 position of the transfer case in the tractor

General Assumptions

- ✓ Forced vibration:-The system vibrates under the influence of external force.
- ✓ Single degree of freedom:-The geometrical pattern of transfer case is constrained within 20 bolts and the failure happens in the axial direction.
- ✓ Viscus damping:-The damping system of the entire driveline and gear box and system is viscus damper of oil type SAE140. This can be easily explained on the chart below.

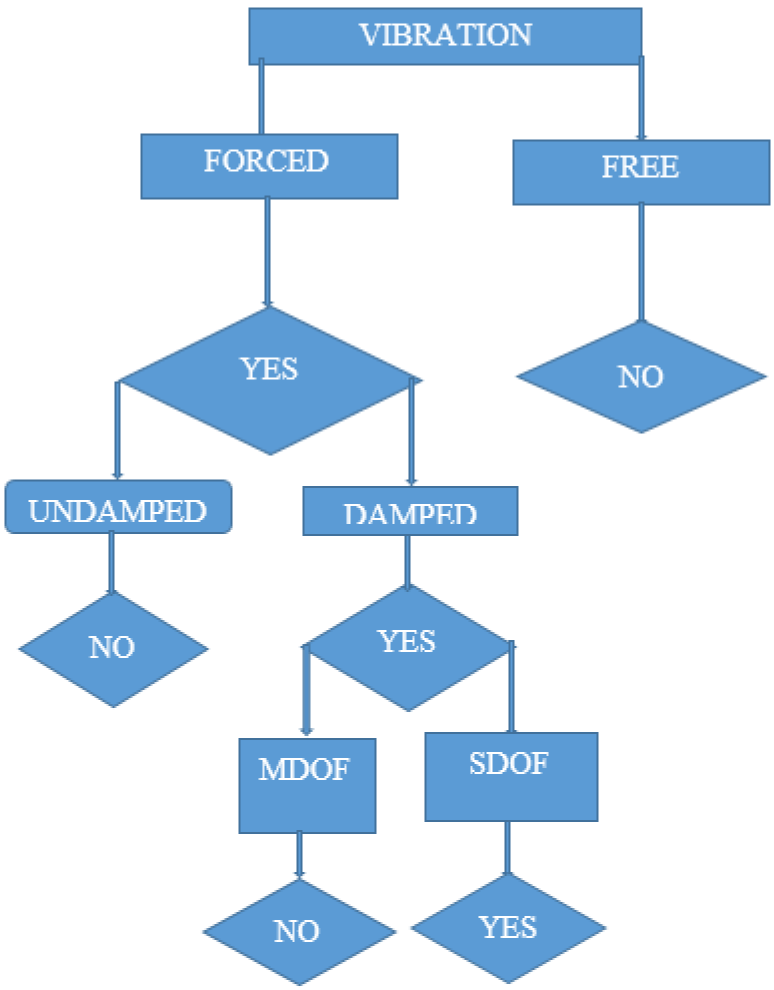


Figure 2.2 Forced vibration damped and SDOF

Degree of freedom

Degree-of-freedom of a general mechanical system is defined as the minimum number of independent variables required to describe its configuration completely. The set of variables (dependent or independent) used to describe a system are termed as the configuration variables. For a mechanism, these can be either Cartesian coordinates of certain points on the mechanism, or the joint angles of the links, or a combination of both.[28] according to geometrical pattern of transfer case the degree of freedom considered as a single degree of freedom.

4.2. Mathematical Analysis

A mass 18.7 kg of transfer case attached in to the differential with a fixed support constrained by 20 bolts. Damping also provided in to the system by SAE140 gear oil which has a physical property of viscosity index 97 and density 0.91kg/l.

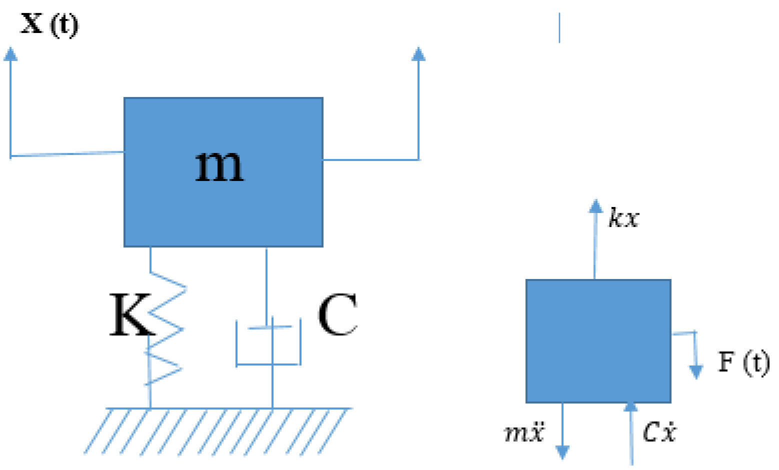
Assumptions

- Lumped mass a system with finite number of degree of freedom are called lumped parameter system. [23](only the mass of the transfer case considered others are considered negligible)
- Stiffness proportional to displacement (according to Hooke's law assume simple harmonic vibration).[12]
- Damping proportional to velocity (Motion is oscillatory motion).

Viscous damping is the most common used damping mechanism in vibration analysis. When mechanical system vibrate in a fluid medium such as air, gas, water and oil, the resistance offered by the fluid to the moving body causes energy to be dissipated. [29]

- Single degree of freedom (assume the transfer case is constrained with bolts but only move on vertical axis

- The system is considered to be mass spring damper system to get the equation of motion of natural vibration.



For the free body diagram of the above figure along y the summation of the force acting on transfer case is not equal to zero since there is forced vibration there is external excitation and the total force acting on the transfer case are damping force, inertia force excitation force and restoring force.

Gravitational force and spring force are negative because the gravitational force is downward force and the spring negative sign indicates the spring pulls when it is stretch and pushes when it is compressed. There is a viscus damper which dissipate the friction.

<u>Newton's law</u>

$$\sum F = ma ------------------ (4.1)$$

$$mg - kx = m\frac{dv}{dt} ----------------- (4.2)$$

$$\frac{d^2x}{dt^2} = m\ddot{x} - kx ---------------- (4.3)$$

$$m\ddot{x} - kx = mg$$

Which is 2nd order homogeneous equation when adding friction dissipation force which is damping force.

$$F_D = -Cv = C\dot{x} - - - - - - - - - - - (4.4)$$

The dissipation force is directly related with velocity v (kind of air dragger the faster it moves the larger the dragger force related to the speed) then importing the friction dissipating force F_D and the excitation force F(t) in to the above equation.

$$ma = -kx - c\dot{x} + m\ddot{x} + F(t - - - - - - - - - (4.5)$$

$$F(t) = F_e = X_0 sin\omega t - - - - - - - - - (4.6)$$

Is the excitation force caused by harmonic vibration and represented by using sine wave equation.

$$kx + c\dot{x} - m\ddot{x} = X_0 sin\omega t = X(t) - - - - - - (4.7)$$

The above solution has homogeneous and particular solution.

$$X(t) = X_{particular} + X_{homogeneous} - - - - - - - - \ (4.8)$$

$$X_{homoginious} \ = kx + c\dot{x} - m\ddot{x} = 0 - - - - - - - - (4.9)$$

$$\ddot{x} + c\dot{x} + kx = 0 - - - - - - - - - (4.10)$$

Re writing the equation

$$\ddot{x} + \frac{c\dot{x}}{m} + \frac{k}{m}x = 0 - - - - - - - - (4.11)$$

$$X = X_0 e^{-\xi \omega t} \sin(\omega_n t + \varphi) - - - (4.12)$$

$$X_h = e^{-\xi \omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2 \omega t] - - (4.13)$$

$$X_{particular} = X_0 sin\omega t - - - - - - - - (4.14)$$

$$X_P = X_0 \sin(\omega t + \varphi) - - - - - - - - - - - (4.15)$$

Whereφ is angle where displacement vector lag to force vector.

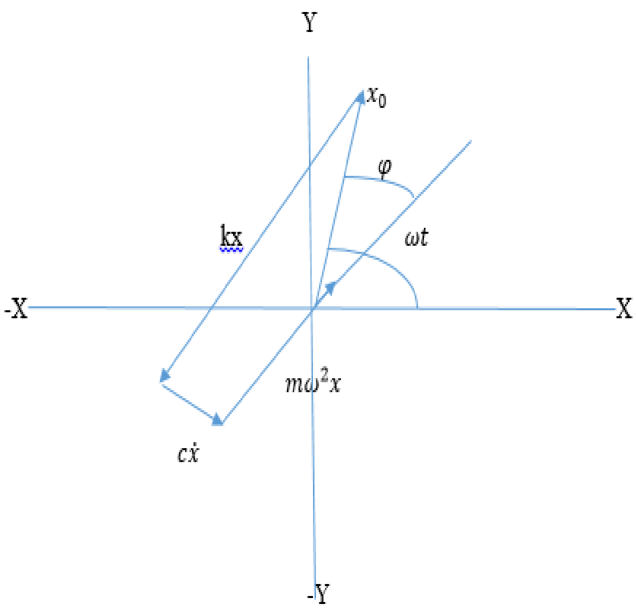
$$\dot{x_p} = \omega x_0 \sin\left(\omega t - \varphi + {}^{\pi}/_{2}\right) - - - - - (4.16)$$

$$\ddot{x_p} = \omega^2 x_0 \sin(\omega t + \varphi + \pi) - - - - - - (4.17)$$

Substituting the above three equation in to general solution

$$-m\omega^2 x_0 \sin(\omega t - \varphi + \pi) + c\omega x_0 \sin\left(\omega t - \varphi + {}^{\pi}/_{2}\right) + kx_0 \sin(\omega t - \varphi) - F_0 sin\omega t = 0 - - - - - - - - - - - - -(4.18)$$

The picture below shows the graphical representation of the above equations.



$$X_0 = {}^{F_0}\Big/_{\sqrt{(k - m\omega^2)^2 + c\omega^2}} - - - - - - - - - - - (4.19)$$

$$tan\varphi = \frac{c\omega x}{kx - m\omega^2} - - - - - - - - - - - - - - - - - - (4.20)$$

$$tan\varphi = \frac{c\omega x/kx}{k(x - mx\omega^2/k)} - - - - -(4.21)$$

$$tan\varphi = \frac{c\omega/k}{1 - m\omega^2/k} - - - - - - (4.22)$$

$$\varphi = tan^{-1}\frac{\frac{c\omega}{k}}{1 - \frac{m\omega^2}{k}} - - - - -(4.23)$$

$$if\ \ \frac{k}{m}=\omega_n{}^2\ then\ \ \frac{m}{k}=\frac{1}{\omega_n^2}\ ,\frac{k}{m}=\omega_n{}^2\ then,\frac{m\omega^2}{k}=\frac{\omega^2}{\omega_n^2}----(4.24)$$

$$\frac{C\omega}{k}=\frac{C}{C_c}*\frac{C_c}{2m}*\frac{2m}{k}*\omega--------(4.25)$$

$$\frac{C}{C_c}=\xi\ ,\frac{C_c}{2m}=\frac{\omega_n m}{k}=\frac{1}{\omega_n^2}------(4.26)$$

$$\frac{c\omega}{k}=\frac{\xi 2\omega}{\omega_n}-----------(4.27)$$

Substituting in to the above equations

$$X_0={}^{F_0/k}\Big/\sqrt{(1-\frac{m\omega^2}{k})^2+\frac{c\omega^2}{k}}-----\quad(4.28)$$

$$\frac{F_0}{k}=X_{st}---------------(4.29)$$

$$X_0={}^{X_{st}}\Big/\sqrt{(1-\frac{\omega^2}{\omega_n^2})^2+2\xi\omega/\omega_n{}^2)}---(4.30)$$

$$\varphi=tan^{-1}\frac{2\xi\ {}^{\omega}/_{\omega_n}}{1-\frac{\omega}{\omega_n}^2}----------(4.31)$$

$$X_p=Xsin(\omega t-\varphi)----------(4.32)$$

After substituting X in the equation

$$X_p=\frac{X_{st}(\omega t-\varphi)}{\sqrt{(1-\frac{\omega}{\omega_n^2}^2)^2+2\xi\frac{\omega}{\omega_n}^2}}-----(4.33)$$

$$X=X_p+X_h------(4.34)$$

Substituting particular and homogeneous solution in to the general solution

$$X = e^{-\xi\omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t-\varphi)}{\sqrt{(1-\frac{\omega}{\omega_n^2}^2)^2 + 2\xi\frac{\omega}{\omega_n}^2}} ----(4.35)$$

The general solution

$$X = e^{-\xi\omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t-\varphi)}{\sqrt{(1-\frac{\omega}{\omega_n^2}^2)^2 + 2\xi\frac{\omega}{\omega_n}^2}} ----(4.35)$$

In damped natural frequencies and the damping ratio are easily calculated from the physical parameters *m*, k and c, so it is quite easy to get an estimate of the behavior of the system.Aand B are constants that will be found from initial condition.

To find the stiffness matrix Kthe modulus elasticity and the moment of inertia should be known.

4.2.1. Moment of inertia of the transfer case

The moment of inertia is a size that characterizes the dispersion of the mass of bodies and serves to study the dynamics of the body and mechanical system. [30]The moment of inertia for the transfer case of the tractor will be difficult to calculate since it has irregular shape and have six faces. Each dimension of the transfer case is directly measured from the real transfer case during technical specification.

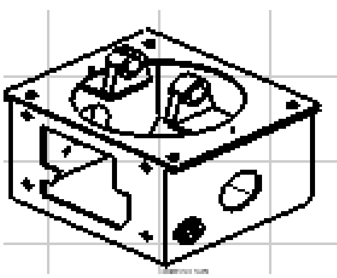
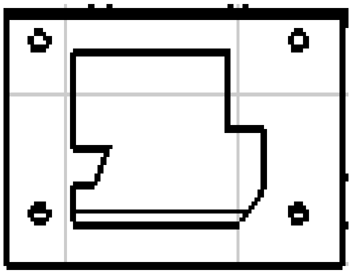


Figure 4.3 Chery 100hp tractor of transfer case

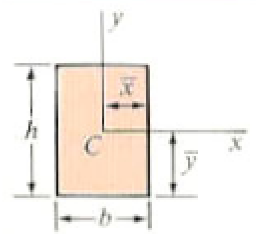
- Front face
- Back face
- Left lateral face
- Right lateral face
- Top face

- Bottom face

Front face It has four hex bolts and one composite shape going to be calculated every shapes and subtract from the entire triangular piece.



The front face of the transfer case has composite shapes and from direct measurement of transfer case it includes four bolts which has a dimension of 12x35, b=125mm where h =145 mm and other dimension. From basic engineering mechanics one concept the moment of inertia of the rectangular shape and the origin of axis at centroid will be calculated using the following formulas.



$$A = bh ----------------------(4.36)$$

$$\bar{x} = \frac{b}{2}, \bar{y} = \frac{h}{2} -------------------- (4.37)$$

$$I_x = \frac{bh^3}{12}, I_y = \frac{hb^3}{12} ----------------- (4.38)$$

This can be calculated by broken up the face in to pieces and subtract from the rectangular shape.

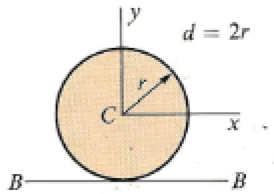
$$A = bh \rightarrow 0.125m * 0.145m = 0.018m^2 -----------(4.39)$$

$$\bar{x} = \frac{b}{2} \rightarrow \frac{0.125m}{2} = 0.0625m, \bar{y} = \frac{h}{2} = \frac{0.145m}{2} = 0.0725 --- (4.40)$$

$$I_y = \frac{0.145 * 0.125^3}{12} = 2.36 * 10^{-5}m^4 ------------ (4.41)$$

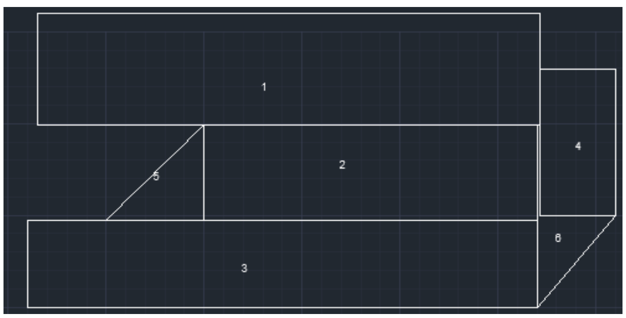
The face is a combination of circle, triangle rectangle shapes these all shapes should be calculated and subtracted from complete rectangle.

Bolts:-the bolts in this transfer case have 12mm diameter and 35mm in length.



$$A = \pi r^2 = \frac{\pi d^2}{4}, I_x = I_y = \frac{\pi r^4}{4} = \frac{\pi d^4}{64} = \pi * \frac{0.024^4}{64} = 1.628 * 10^{-8} m^2 ----(4.42)$$

The next composite shape in this face have six peace of different shapes



Piece1 Rectangular

Specification b=90mm, h=56mm

$$A = bh = 0.09m * 0.056m = 5.04 * 10^{-3} m^2 ------(4.43)$$

$$\bar{x} = \frac{b}{2} = \frac{0.09}{2} = 0.045m, \bar{y} = \frac{h}{2} = \frac{0.056}{2} = 0.028m --- (4.44)$$

$$I_y = \frac{hb^3}{12} = \frac{0.056 * 0.09^3}{12} = 3.402 * 10^{-6} m^4 ----(4.45)$$

Piece 2 b=54mm, h=20mm

$$A = bh = 0.054m * 0.02m = 1.08 * 10^{-3} m^2 ----- (4.46)$$

$$\bar{x} = \frac{b}{2} = \frac{0.054m}{2} = 0.027m, \bar{y} = \frac{h}{2} = \frac{0.02}{2} = 0.01m --(4.47)$$

$$I_y = \frac{hb^3}{12} = \frac{0.02 * 0.054^3}{12} = 2.67 * 10^{-7} ------(4.48)$$

Piece 3 Rectangular b=90mm, h=56mm

$$A = bh = 0.09m * 0.056m = 5.04 * 10^{-3} m^2 ------(4.49)$$

$$\bar{x}=\frac{b}{2}=\frac{0.09}{2}=0.045m,\bar{y}=\frac{h}{2}=\frac{0.056}{2}=0.02----(4.50)$$

$$I_y=\frac{hb^3}{12}=\frac{0.056*0.09^3}{12}=3.402*10^{-6}m^4----(4.51)$$

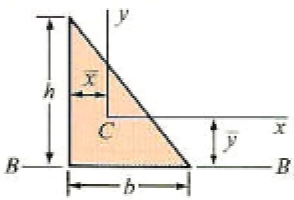
Piece 4 Rectangular b=13mm,h=34mm

$$A=0.013m*0.034m=4.42*10^{-4}m^2-----------(4.52)$$

$$\bar{x}=\frac{b}{2}=\frac{0{,}013}{2}=6.5*10^{-3}m,\bar{y}=\frac{h}{2}=\frac{0.034}{2}=0.01-----(4.53)$$

$$I_y=\frac{hb^3}{12}=\frac{0.034*0.013^3}{12}=6.23*10^{-9}m^4----------(4.54)$$

Piece 5 triangular b=13mm, h=56mm



$$A=\frac{bh}{2}=0.013*\frac{0.056}{2}=3.6410^{-4}m^2--------------(4.55)$$

$$,\bar{x}=\frac{b}{3}=\frac{0.013m}{3}=4.33*10^{-3}m,\bar{y}=\frac{h}{3}=\frac{0.056m}{3}=0.0187m----(4.56)$$

$$,I_y=\frac{hb^3}{36}=\frac{0.056*0.013^3}{36}=3.41*10^{-9}m^4,------------(4.57)$$

Piece 6 triangular h=23mm,b=10mm

$$A=\frac{bh}{2}=\frac{0.01m*0.023m}{2}=1{,}15*10^{-4}m^2-------------(4.58)$$

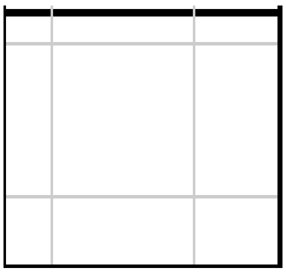
$$\bar{x}=\frac{b}{3}=\frac{0.01}{3}=3.33*10^{-3}m,\bar{y}=\frac{h}{3}=\frac{0.023}{3}=7.67*10^{-3}m----(4.59)$$

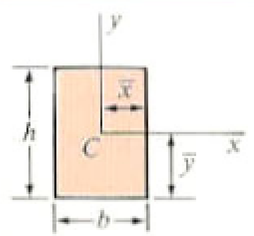
$$I_y=\frac{hb^3}{36}=\frac{0.023*0.01^3}{36}=6.38*10^{-10}m^4------------(4.60)$$

The total moment of inertia for the front face will be founded by adding up the total moment of inertia and subtracting from the rectangular section

$$I_F = I_R - \left(\sum_{1}^{8} I_c\right) = (5.53 * 10^{-5} - 7.12 * 10^{-6})m^4 = 4.81 * 10^{-5}m^4 ------ (4.61)$$

2 Back face-: The back face has only single piece and since it doesn’t have composite shapes it is quite easy to get its moment of inertia.





b=185mm.h=145mm

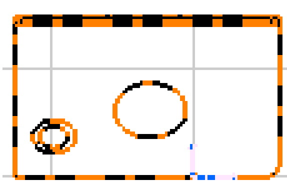
$$A = bh = 0.185m * 0.145m = 0.0268m^2 -----------(4.63)$$

$$\bar{x} = \frac{b}{2} = \frac{0.185}{2} = 0.092m, \bar{y} = \frac{h}{2} = \frac{0.145}{2} = 0.073m -------(4.64)$$

$$I_y = \frac{hb^3}{12} = \frac{0.145m * 0.185m^3}{12} = 7.651m^4 -----------(4.65)$$

3. Left lateral face

The dimension for rectangular portion is b=193mm, h=195mm



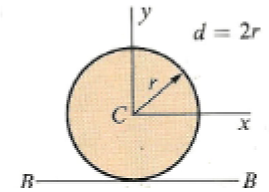
$$A = bh = 0.193m * 0.195m = 0.038m^2 ---------(4.66)$$

$$\bar{x} = \frac{b}{2} = \frac{0.193}{2} = 0.0965m, \bar{y} = \frac{h}{2} = \frac{0.195}{2} = \frac{0.0975}{2} = 0.04875m --- (4.67)$$

$$I_x = \frac{bh^3}{12} = \frac{0.193 * 0.195^3}{12} = 9.937 * 10^{-6}m^4 ---------(4.68)$$

$$I_y = \frac{hb^3}{12} = \frac{0.195 * 0.193^3}{12} = 1.168 * 10^{-4}m^4 -------(4.69)$$

Second piece is circle with 70mm diameter



$$A = \pi r^2 = \frac{\pi d^2}{4} = \frac{\pi * 0.07^2}{4} = 3.848 * 10^{-3} ---------- (4.71)$$

$$I_y = \frac{\pi r^4}{4} = \frac{\pi d^4}{64} = \frac{\pi * 0.07^4}{64} = 1.178 * 10^{-6}m^4 ---------(4.72)$$

The third piece is a circle with 33mm diameter

$$A = \pi r^2 = \frac{\pi d^2}{4} = \frac{\pi * 0.033^2}{4} = 8.55 * 10^{-4}m^2 -------(4.73)$$

$$I_y = \frac{\pi r^4}{4} = \frac{\pi d^4}{64} = \frac{\pi * 0.033^4}{64} = 5.82 * 10^{-8}m^4 -------(4.74)$$

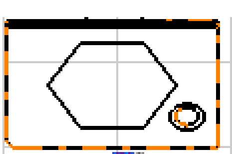
$$I_{ls} = I_R - \left(\sum_{1}^{2} I_{co}\right) -------------------- (4.75)$$

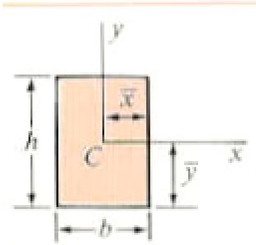
$$I_{ls} = 9.94 * 10^{-6} - 3.066 * 10^{-6} = 6.87 * 10^{-6}m^4 ----(4.76)$$

1. **Right lateral face**

   Dimension for the rectangular section

   h=145mm,b=.185mm



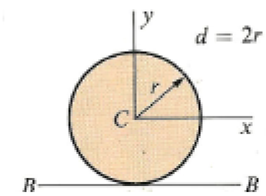


$$A = bh = 0.185m * 0.145m = 0.0268m^2 ------(4.77)$$

$$\bar{x} = \frac{b}{2} = \frac{0.185m}{2} = 0.0925m, \bar{y} = \frac{h}{2} = \frac{0.145}{2} = 0.0725m --- (4.78)$$

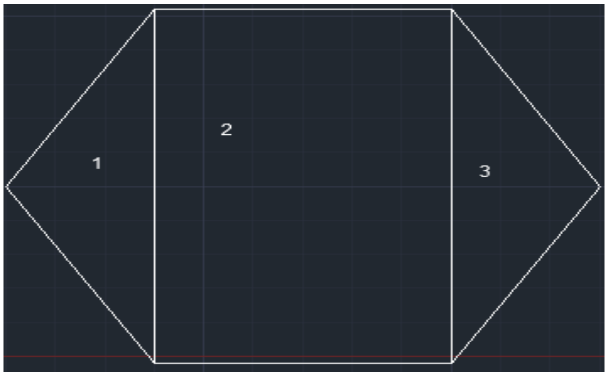
$$I_y = \frac{0.145 * 0.185^3}{12} = 7.65 * 10^{-5}m^4 ------(4.79)$$

The second piece diameter =16mm



$$A = \pi r^2 = \frac{\pi d^2}{4}, I_y = \frac{\pi r^4}{4} = \frac{\pi * 0.016^4}{64} = 3.22 * 10^{-9}m^4 ----(4.80)$$

The geometrical structure of the composite material inside the rectangle is hexagon And classified in to three pieces two triangle and one rectangle.



Piece 1 triangular shape b=100mm, h=20mm

$$I_y = \frac{hb^3}{48} = \frac{0.1 * 0.02^3}{48} = 1.66 * 10^{-8} m^4 - - - - - - - - - (4.81)$$

Piece 2 rectangular shape b=94mm h=100mm

$$, I_y = \frac{hb^3}{12} = \frac{0.1 * 0.094^3}{12} = 9.4 * 10^{-3} m^4 - - - - - - - - - - - - - (4.81)$$

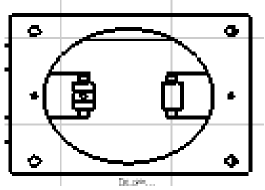
Piece 3 triangular shape h=20mm,b=100mm

$$I_y = \frac{bh^3}{36} = \frac{0.1 * 0.02^3}{36} = 2.22 * 10^{-8} m^4 \quad - - - - - - - - - - - - - (4.82)$$

$$I_{Rl} = I_R - \sum_{1}^{4} I_{co} - (4.83)$$

$$I_{Rl} = (7.65 * 10^{-5} - 7.84 * 10^{-6}) m^4 = 6.86 * 10^{-5} m^4 - - - - - - (4.84)$$

2. **Top face**



Rectangle b=185mm,h=122mm

$$A = bh = 0.185m * 0.122m = 0.02257m^2 - - - - - - - - - - (4.85)$$

$$\bar{x} = \frac{b}{2} = \frac{0.185}{2} = 0.0925m, \bar{y} = \frac{h}{2} = \frac{0.122}{2} = 0.061m - - - - - (4.86)$$

$$I = \frac{0.122 * 0.185}{12}(0.122^2 + 0.185^2) = 9.24 * 10^{-5} m^4 - - (4.87)$$

Circle diameter of 96mm

$$A = \pi r^2 = \frac{\pi d^2}{4} = \frac{\pi * 0.096^2}{4} = 7.2410^{-3} m^2 - - - - - - - - - - - - (4.88)$$

$$I_y = \frac{\pi r^4}{4} = \frac{\pi * 0.096^4}{64} = 4.17 * 10^{-6} m^2 - - - - - - - - - - - - - - (4.89)$$

Hex Bots diameter 12mm

$$A = \pi r^2 = \frac{\pi d^2}{4}, I_y = \frac{\pi r^4}{4} = \frac{\pi d^4}{64} = \frac{\pi * 0.012^4}{64} = 1.017 * 10^{-9} m^4 --(4.90)$$

Bolts diameter =10mm

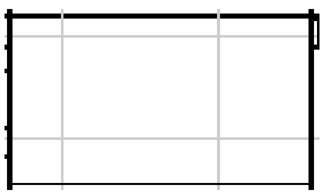
$$I_y = \frac{\pi r^4}{2} = \frac{\pi d^4}{32} = \frac{\pi * 0.01^4}{32} = 9.82 * 10^{-10} m^4 ----- (4.91)$$

$$I_T = I_R - \sum_1^3 I_{Co} ------(4.92)$$

$$I_T = 2.79 * 10^{-5} - 2.08 * 10^{-5} = 7.1 * 10^{-6} m^4 --- (4.93)$$

6. Bottom face

Rectangular shape Dimension h=193mm, b=195mm



Rectangle

$$A = bh = 0.193m * 0.195m = 0.038m^2 ----- (4.94)$$

$$\bar{x} = \frac{b}{2} = \frac{0.193}{2} = 0.096m, \bar{y} = \frac{h}{2} = \frac{0.195}{2} = 0.0975m --- (4.95)$$

$$I_y = \frac{hb^3}{12} = \frac{0.193 * 0.195^3}{12} = 1.193 * 10^{-4} m^4 ----(4.96)$$

$$I_{Bo} = 1.193 * 10^{-4} m^4 -----------(4.97)$$

Faces	Moment of Inertia (m^4)
Top	$7.1 * 10^{-6}$
Bottom	$1.168 * 10^{-4}$
Left lateral	$6.87 * 10^{-6}$
Right lateral	$6.86 * 10^{-5}$
Back	$1.235 * 10^{-4}$
Front	$4.81 * 10^{-5}$

Table 5 Moment of inertia for six face of transfer case

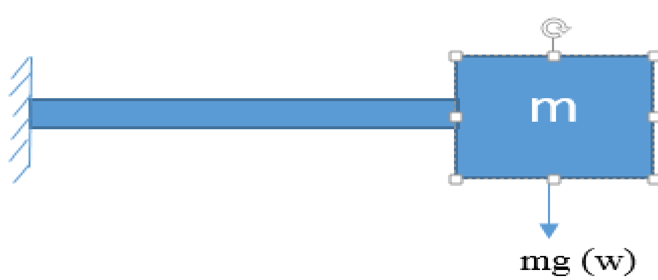
The summation of all moment of inertia of six face equal to the moment of inertia for the transfer case. Whereas the transfer case material type measured by spectrometer from Akaki Basic metal Industry is categorized as AISI Alloy steel 4140.The Physical property of this material indicate the value for elastic modulus is between 190 to 210Gpa.

$$I_T = \sum_{1}^{6} I = 3.71 * 10^{-4} m^4 = 3.71 * 10^{8} mm^4 ----------------(4.98)$$

$$\mathrm{E} = 190 - 210\mathrm{Gpa}$$

The real measurement the transfer case the mass and the length of the transfer case is equal to 18.7kg and 145 mm respectively. These all helps to find the value of stiffness K.

4.2.3 Stiffness(k)



Assumptions

Assuming the elementary beam theory

- The beam is loaded only in the y direction.(because the power transfer from the differential to the transfer case is in a vertical direction)

- Displacement of the beam(shaft) is small in comparison to the transfer case
- The system is considered to be a cantilever beam because the shaft is assumed to be mass less comparing with the transfer case and the mass is concentrated on one end which is the transfer case. From the concept of strength of material,

$$\delta = \frac{wL^3}{3EI} - - - - - - - - - - - -(4.99)$$

$$k = \frac{w}{\delta} \leftrightarrow \frac{3EI}{L^3} - - - - - - - - - -(4.100)$$

$$E = 200Gpa$$

$$I = 3.71 * 10^8$$

$$L = 145mm$$

$$K = \frac{3 * 200Gpa * 3.71 * 10^8 mm^4}{145^3} = 7.30 * 10^9 \, {}^{N}/_{m} - -(4.101)$$

4.2.4. Natural frequency

$$\omega_n = \sqrt[2]{\frac{k}{m}} = \sqrt[2]{\frac{7.3 * 10^9 \, N/m}{18.7kg}} = 40348.08 * 2\pi/60 = \frac{5063.05rad}{sec} - - - - - - - - (4.102)$$

This natural frequency shows that the natural parameter of the system if this frequency falls on the range of critical frequency it indicates the presence of natural frequency which cause machine failure but if the natural frequency is not fall between the critical range shows there is no resonance.

4.2.5 Damping ratio (ξ)

The damping ratio ξ varies for different damping condition cases

- Case 1 ξ=0 undamped
- Case 2 ξ=1 critical damped
- Case 3 ξ>1 over damped
- Case 4 ξ<1 under damped

On this particular condition or thesis the transfer case of Chery 100hp has a viscus damper with oil type of SEA 140.For the current situation the system the model damping ratio ξ calculated from the oil specification follows.

To analyze the situation of resonance on the transfer case due to damping, the basic theory of fluid mechanics is used. For Newtonian fluids (common linear fluids such as water, oil, and alcohols) this shear stress (t) can be expressed as:

$$\tau = \mu \frac{\partial u}{\partial y} ------(4.103)$$

Where u is fluid velocity and y is vertical distance to the flow of fluid. If the flow is induced by relative motion between rotating gears inside the transfer case at a velocity U the velocity distribution assumed to be linear because the velocity of fluid layer is the same as with the velocity between the driving and driven gears. The above equation can be formulated as

$$\tau = \frac{\mu U}{Y} ------- (4.104)$$

Where U is the meshing gears velocity and Y is a distance. By neglecting the shear applied on the top surface and the side wall of the transfer case. A fluid is at the bottom of the transfer case can be considered to find the torque applied to the transfer case from the surrounding fluid shear. Thus:

$$dT = rdF ----- (4.105)$$

Where dT and dF Infinitesimal torque and force applied to the transfer case from the liquid then the force applied to the wall of transfer case can be expressed as.

$$dF = \tau dA ----- (4.106)$$

Where dA is the area of the shaft rotated to splash oil the area become

$$dA = rdrd\theta ----(4.107)$$

Where $d\theta$ and dr and infinitely small angle and distance respectively after combining the above two equations and using the relation linear velocity and angular velocity.

$$U = r\omega - - - - - (4.108)$$

Where ω is the angular velocity of the rotating shaft at specific instant .The torque applied from the fluid to the wall of the transfer case is there for obtained as

$$dT = \frac{ur^3\omega}{Y} drd\theta - - - - - (4.109)$$

Integrating the equation to the entire transfer case the equation for the damping torque applied from the fluid to the transfer case obtained as

$$T = \frac{\pi v R^4}{2Y}\omega - - - - - - - - (4.110)$$

Where R is the input shaft radius. Allowing the equation to be the damping torque applied to the wall of the transfer case. The system damping coefficient easily derived as

$$c = \frac{\pi R^4}{2Y} v - - - - - - - - - (4.111)$$

According to Bernoulli's Equation for fluid velocity

$$P + \rho g h + \frac{1}{2} m v^2 = constant$$

$$P_1 + \rho g h_1 + \frac{1}{2} m v_1^2 = P_2 + \rho g h_2 + \frac{1}{2} m v_2^2$$

Assume constant pressure, mass constant, final velocity and initial velocity zero and

$h_2 = \frac{1}{2} h$, because to splash the oil the shaft need to rotate half of the transfer case.

$$v_2^2 = \left(\rho g \frac{h}{2}\right) 2 = \rho g h$$

$$v = \sqrt{\rho g h} - - - - - - - - (4.112)$$

$$where\ \rho = 91 kg/l\ (specification\ of\ oil\ used\ by\ chery\ 100hp\ transfer\ case\ SEA140)$$

$$g = 9.8 m/s^2$$

$$h = 185mm\ (hight\ of\ the\ transfer\ case\ by\ direct\ measurement$$

$$v = \sqrt{91 * 9.8 * 0.185} = 12.84m/s - - - (4.113)$$

$$where\ v = 12.84mm,$$

$R\ in\ put\ shaft\ radious = 21.5mm\ dia,$ Changing the diameter in to radius

$$r = \frac{d}{2} = \frac{0.0215}{2} = 0.0107mm$$

$$Y\ is\ the\ length\ of\ the\ transfer\ case = 122mm$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 12.84 = 2.16 * 10^{-6} Ns/m$$

The damping co efficient c can be defined as in terms of ξ (Zita) or damping ratio

$$\xi = \frac{c}{2\sqrt{kI}} = \frac{2.16 * 10^{-6}}{2\sqrt{7.3 * 10^9 * 3.71 * 10^8}} = 6.56 * 10^{-16} \approx 0 - - - - - (4.114)$$

The above result shows the damping factor approximates to be zero which shows the system is undamped. Where K and I are the stiffness of the transfer case moment of inertia, the radius of the input shaft found from direct measurement of the shaft. The Phase angle can be expressed by damping co efficient and critical damping co efficient. At ξ=0,

$$\varphi = tan^{-1} \frac{2\xi\ ^{\omega}/_{\omega_n}}{1 - \frac{\omega}{\omega_n}^2} = tan^{-1} 0 = 0 - - - - - - (4.115)$$

The general solution becomes

$$X = e^{-\xi\omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t - \varphi)}{\sqrt{(1 - \frac{\omega}{\omega_n^2}^2)^2 + 2\xi\frac{\omega}{\omega_n}^2}}$$

$$\frac{F_0}{k} = X_{st} - - - - - (4.116)$$

When the tractor works at 100hp

For the maximum power the speed will be minimum during traction process so the minimum speed or ideal speed of the Chery 100hp tractor is 1800rpm.Where r is the input shaft radius that transfer power from the front wheel to the transfer case measured.

$$F_0 = \frac{T}{r} - - - - - - (4.117)$$

Where T is torque and r is the input shaft radius that transfer power to the transfer case. The angular speed taken is the idle speed of Chery 100hp tractor because for the maximum torque the speed will be minimum.

$$1hp = 740watt, 100hp = 74000watt$$

$$T = \frac{power}{speed} = \frac{74000watt}{1800rpm} = 41.11, r = 10.75mm = 0.01m$$

$$F_0 = \frac{41.11}{0.01} = 4111N$$

$$\frac{F_0}{k} = X_{st} = \frac{4111}{7.3 * 10^9} = 5.63 * 10^{-6} - - - - - (4.118)$$

$$X = e^{-\xi\omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t - \varphi)}{\sqrt{(1-\frac{\omega}{\omega_n^2}^2)^2 + 2\xi\frac{\omega}{\omega_n}^2}} - - - -(4.119)$$

$$\dot{X} = -\omega_n Acos\omega_n t + \frac{X_{st}\omega}{(\sqrt{\left(1-\frac{\omega}{\omega_n^2}^2)^2 + 2\xi\frac{\omega}{\omega_n}^2\right)} * X_{st}} - - - -(4.120)$$

$$\ddot{X} = -\omega_n^2 Asin\omega_n t - - - -(4.121)$$

4.2.6Initial conditions

From the general solution A and B are constants that will be found from initial condition.The initial conditions are based on the geometrical pattern of the system .According to basic vibration concept the initial condition for the cantilever beam is.

$$at\ t = 0, X(0) = 0$$

$$at\ t = 0, X(\dot{0}) = 0$$

$$A/\omega_n = \frac{X_{st}\omega_d \Big/ \omega_n - 1/\omega_d^2}{cos\omega_n} - - - - - - - - - - - -(4.122)$$

$\omega_d = \sqrt{1+\xi^2}\omega_n\ at\ \xi = 0\ then\ \omega_n = \omega_d$ Substituting $\omega_n, \omega_d\ and\ \xi = 0$ the value of A becomes 0.011.

$$A = 0.011$$

$$\ddot{X} = -0.011\omega_n^2 sin\omega_n t - - - - - - - - - - - - - - - (4.123)$$

$$at\ t = 0, X(0) = 0$$

$$X = Acos\omega_n t + 0.017B + \frac{X_{st}\omega_d t}{\left(1 - {\omega^2}/{\omega_n{}^2}\right)} - - - - (4.124)$$

$$0 = Acos0 + 0.017B + \frac{X_{st}\omega_d * 0}{\left(1 - {\omega^2}/{\omega_n{}^2}\right)} - - - - - - - - (4.125)$$

$$0 = Acos0 + 0.017B - - - - - - - - - - - - (4.126)$$

Substituting A=0.011 in the above equation

$$0 = 0.011 + 0.017B, B = -0.647 - - - - - - - - - - - (4.127)$$

$$X = 0.011cos\omega_n t - 0.647 - - - - - - - - - - - - - - (4.128)$$

If the value of ξ becomes zero the natural frequency and damped frequency will be similar

$$\omega_d = \sqrt{1 + \xi^2}\omega_n\ at\ \xi = 0\ then\ \omega_n = \omega_d$$

The general solution equation becomes

$$if\ \ \frac{k}{m} = \omega_n{}^2\ then\ \ \frac{m}{k} = \frac{1}{\omega_n^2}, \frac{k}{m} = \omega_n{}^2\ then, \frac{m\omega^2}{k} = \frac{\omega^2}{\omega_n^2}$$

$$X = \ e^{-\xi\omega_n t}[Acos\sqrt{1 - \xi^2}\omega_n t + Bsin(1 - \xi^2\omega t] + \frac{X_{st}(\omega t - \varphi)}{\sqrt{(1 - \frac{m\omega^2}{k})^2 + 2\xi\, \frac{m\omega^2}{k}^2}}$$

At ξ=0 the equation becomes

$$X = \ Acos\ \omega t + 0.017B + \frac{X_{st}(\omega t)}{\sqrt{(1 - \frac{m\omega^2}{k})^2}} - - - - - - - - - - (4.129)$$

After substituting A,B,K,m,X_{st}, ω to the equation the equation becomes

$$X = 0.011cos\ \omega t - 0.0119 + 1.9 * 10^{-13} --------(4.130)$$

The last term approximated to zero and finally the equation turn into

$$X = 0.011cos\ \omega t - 0.0119 ----(4.131)$$

Taking the equation in to Matlab software and plot amplitude versus time graph at ξ=0 it will give the figure below.

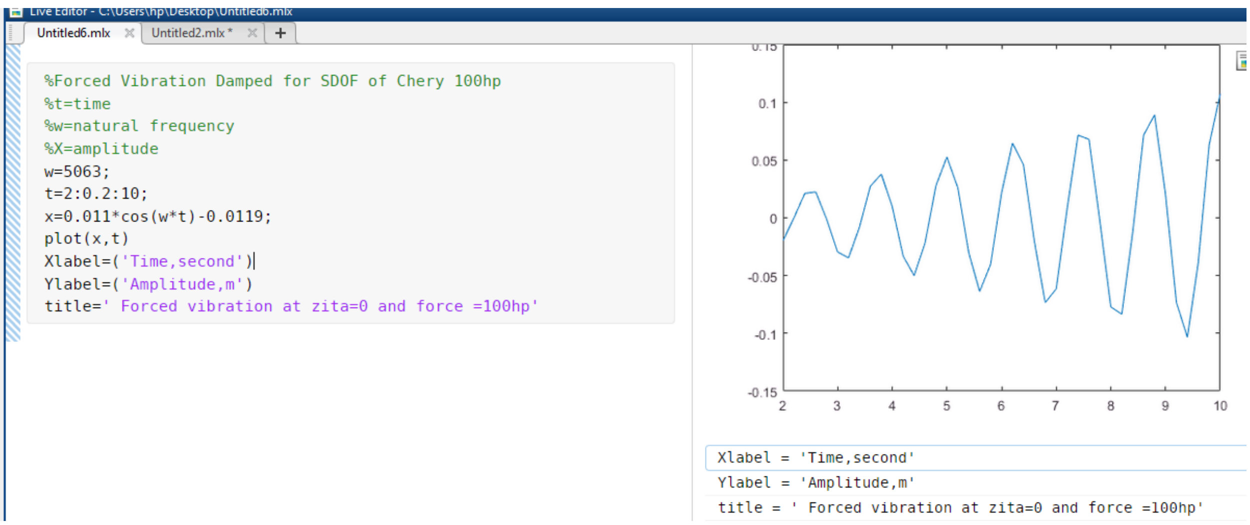


Figure 4.4 Amplitude versus time plot at Zita =0 and 100hp

The above plot shows that as the time increase the amplitude increase this is the clear indication of the system is not damped.

Amplitude versus time graph at ξ=1

$$X = e^{-\xi\omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t-\varphi)}{\sqrt{(1-\frac{\omega}{\omega_n^2}^2)^2 + 2\xi\frac{\omega}{\omega_n}^2}}$$

$$\omega_d = \sqrt{1+\xi^2}\omega_n\ at\ \xi = 1\ then \frac{\omega_n}{\sqrt{2}} = \omega_d = 5063 * \sqrt{2} = 7160 rad/sec$$

$$\varphi = tan^{-1}\frac{2\xi\ ^{\omega}/_{\omega_n}}{1-\frac{\omega}{\omega_n}^2} = tan^{-1}\frac{2 * 1 * {}^{5063}/_{7160}}{1-({}^{5063}/_{7160})^2} = -70.52 ------- (4.132)$$

$$X = e^{-5063t}(0.001cos(2069t) + 0.01sin(1 - \omega t] + 0.0058t - - - - - (4.133)$$

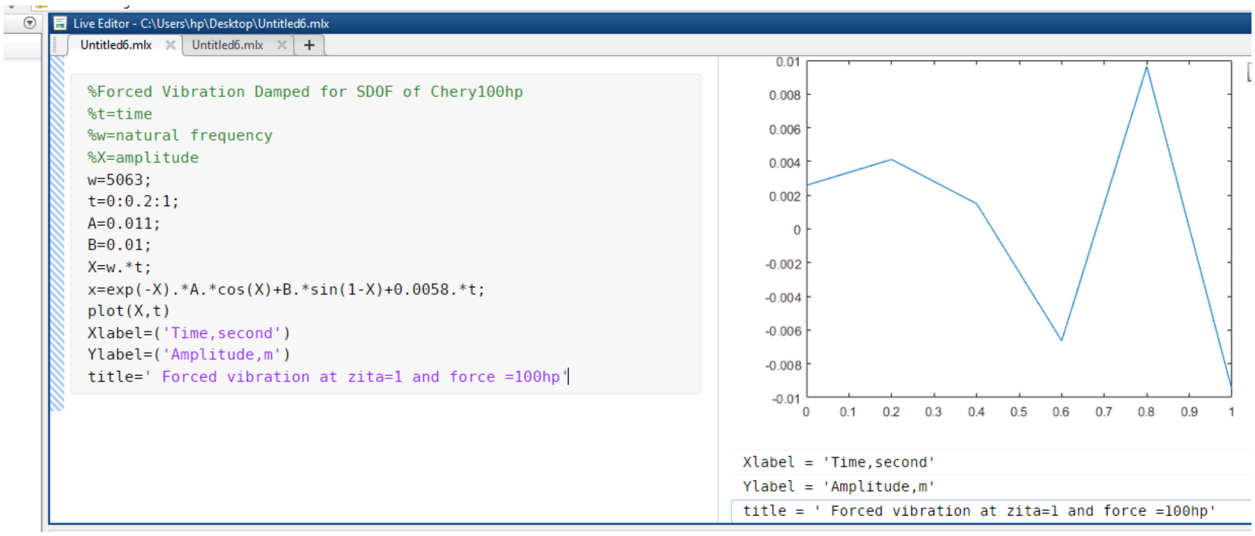


Figure 4.5Plot of amplitude versus time graph at Zita=1and 100hp

The plots specifies that the increment on the damping from 0 to 1 has a basic difference for a system implies theamplitude decreased and the system is damped.That implies one of the failure reason for the system is damping oil type.

When the tractor works at 95hp

$$F_0 = T/r - - - - - - - - -(4.135)$$

Where T is torque and r is the input shaft radius

$1hp = 740watt, 95hp = 70300watt$ The maximum speed of cherry 100hp tractor were used to get the minimum torque because at maximum speed the torque will be minimum.

$$T = \frac{power}{speed} = \frac{70300watt}{2100rpm} = 33.476, r = 10.75mm = 0.01m$$

$$F_0 = \frac{33.476}{0.01} = 3347N$$

$$\frac{F_0}{k} = X_{st} = \frac{3347}{7.3 * 10^9} = 4.58 * 10^{-6} - - - - - - - - -(4.136)$$

The general solution At ξ=0 the becomes

$$if\ \frac{k}{m} = \omega_n{}^2\ then\ \frac{m}{k} = \frac{1}{\omega_n^2}, \frac{k}{m} = \omega_n{}^2\ then, \frac{m\omega^2}{k} = \frac{\omega^2}{\omega_n^2}$$

$$X = e^{-\xi\omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t - \varphi)}{\sqrt{(1-\frac{m\omega^2}{k})^2 + 2\xi\frac{m\omega^2}{k}^2}}$$

$$X = [Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t - \varphi)}{\sqrt{(1-\frac{m\omega^2}{k})^2 + 2\xi\frac{m\omega^2}{k}^2}} - - - - - - -(4.136)$$

After substituting A, B, K, m,X_{st} ω to the equation the equation becomes

$$X = 0.011cos\ \omega t - 0.0119 + 0.0096t - - - - - (4.137)$$

Taking the equation in to Matlab software and plot amplitude versus time graph at ξ=0 it will give the figure below.

Amplitude versus time graph at ξ=1

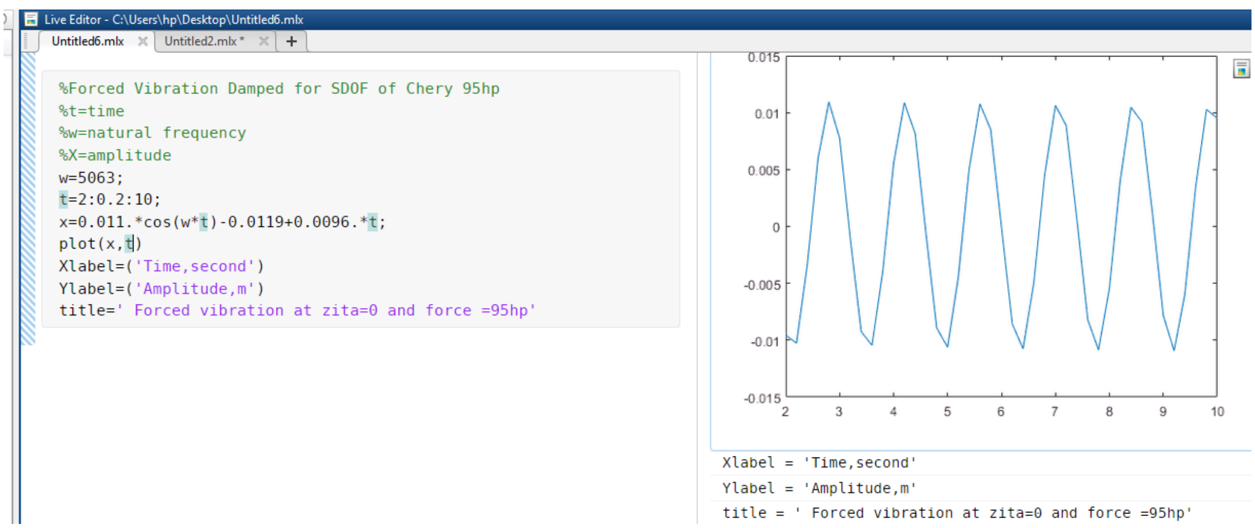


Figure 4.6 Plot of amplitude versus time graph at Zita =0 and 95hp

$$X = e^{-\xi\omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t - \varphi)}{\sqrt{(1-\frac{m\omega^2}{k})^2 + 2\xi\frac{m\omega^2}{k}}}$$

$$\varphi = tan^{-1}\frac{2\xi \, {}^{\omega}/_{\omega_n}}{1-\frac{\omega}{\omega_n}^{2}} = tan^{-1}\frac{2*1*{}^{5063}/_{7160}}{1-({}^{5063}/_{7160})^{2}} = -70.52 ----(4.138)$$

$$X = \; e^{-\omega t}(0.001cos(\omega t) - 0.0647sin(1-\omega t] + 0.0053t + 0.0018 --(4.139)$$

Taking the equation in to Matlab software and plot amplitude versus time graph at ξ=0 it will give the figure below

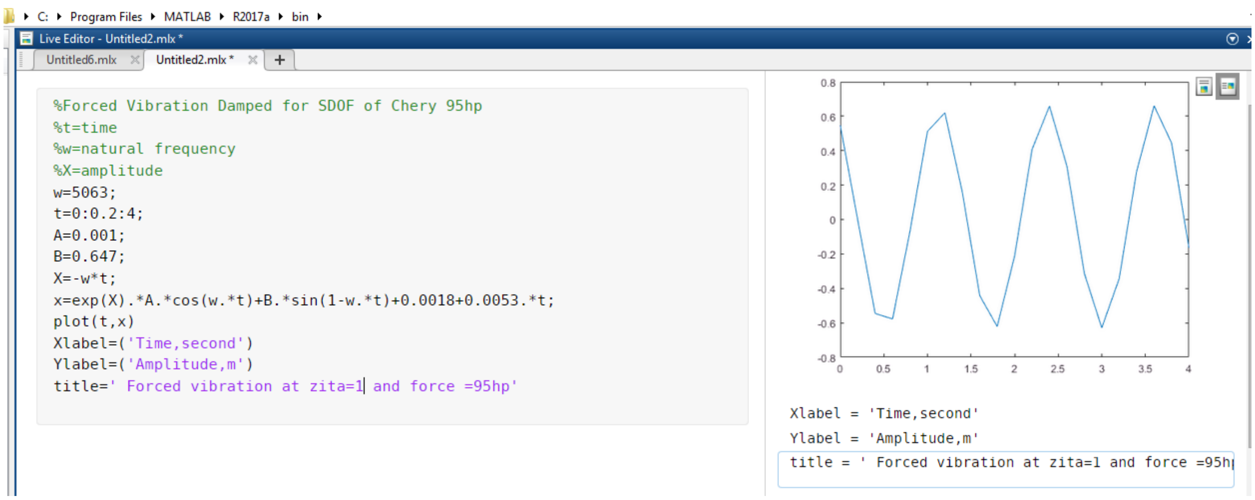


Figure 4.7 Plot of amplitude versus time graph at Zita=1 and 95hp

Table 6Physical properties of SEA 140 oil specification

Viscosity @100^{o}c	28-33
Viscosity index	90
Pour point ^{o}c	-3
Flash point^{o}c	190
Density @15^{o}c kg/l	91

The damping factor for the existing oil specification is calculated and found to be zero with this damping factor the general solution is plotted for amplitude versus time graph for both 95hp and 10 hp excitations. Which indicates that the graph the system remain undamped with this damping factor which is ξ=0.

The amount of damping factor chosen to be ξ=1 which is the critical damping factor of forced vibration system. Again the amount of damping factor inserted in to the general solution to plot whether the system response is damped or not. Based on the plot the system response showed that at the value of ξ=1 the system damped. In order to proceed with this damping factor the oil specification should be chosen from available literature result.

The value of damping factor varied from one type of oil to the other type of oil this leads to know the better oil type that has good damping factor. The following gear oil specification has taken from literatures for better result of damping factor.

Oil type	Kv @T 100c	V_{Index}	ρ@15.6 c	FP	PP
SAE J306	445 mm^2/s	2270	1298Kg/l	92	97
SAE 90	15.5 mm^2/s	90	0.9084Kg/l	482	10
SAE 140	27.5 mm^2/s	93	0.9165Kg/l	486	10
SAE 250	46 mm^2/s	119	0.9157Kg/l	496	16
ISO 100	100 mm^2/s	98	$0.880*10^3$	250	-24
ISO220	233 mm^2/s	96	$0.889*10^3$	496	-18
ISO 150	14.8 mm^2/s	98	$0.884*10^3$	256	-24
ISO68	8.6 mm^2/s	97	$0.878*10^3$	242	-30
SAE 85W-140	30.3 mm^2/s	103	$0.901*10^3$	200	-12

Table 7 Test data made for top damping gear oil

SAE 90 where the density is equals to 0.9084 kg/m^3

$$v = \sqrt{\rho g h}$$

$$v = \sqrt{91 * 9.8 * 0.9084} = 28.46 m/s$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 28.46 = 3.92 Ns/m$$

$$\xi = \frac{c}{2\sqrt{km}} = \frac{3.92 Ns/m}{2\sqrt{7.3 * 10^9 * 18}} = 5.29 * 10^{-6}$$

SAEJ306 where the density is equals to 1298 kg/m^3

$$v = \sqrt{\rho g h}$$

$$v = \sqrt{91 * 9.8 * 1298} = 1075.8m/s$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 1075.8 = 200.22.22Ns/m$$

$$\xi = \frac{c}{2\sqrt{km}} = \frac{200.22Ns/m}{2\sqrt{7.3 * 10^8 * 18.7}} = 0.027$$

SAE 250 where the density is equals to 0.9157 kg/m^3

$$v = \sqrt{\rho g h}$$

$$v = \sqrt{91 * 9.8 * 0.9157} = 28.57m/s$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 28.57 = 3.93Ns/m$$

$$\xi = \frac{c}{2\sqrt{km}} = \frac{3.92Ns/m}{2\sqrt{7.3 * 10^9 * 18.7}} = 5.3 * 10^{-6}$$

ISO 100 where the density is equals to 0.880*10^3 kg/m^3

$$v = \sqrt{\rho g h}$$

$$v = \sqrt{880 * 9.8 * 0.185} = 39.94m/s$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 39.94 = 5.51Ns/m$$

$$\xi = \frac{c}{2\sqrt{km}} = \frac{5.50Ns/m}{2\sqrt{7.3 * 10^9 * 18}} = 7.44 * 10^{-6}$$

ISO 220 where the density is equals to0.889 *10^3 kg/m^3

$$v = \sqrt{\rho g h}$$

$$v = \sqrt{889 * 9.8 * 0.185} = 40.15m/s$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 40.15 = 5.53Ns/m$$

$$\xi = \frac{c}{2\sqrt{km}} = \frac{5.53Ns/m}{2\sqrt{7.3 * 10^9 * 18}} = 5.68 * 10^{-11}$$

ISO 150 where the density is equals to0.884 *10^3 kg/m^3

$$v = \sqrt{\rho g h}$$

$$v = \sqrt{884 * 9.8 * 0.185} = 40.03 m/s$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 40.03 = 5.53 Ns/m$$

$$\xi = \frac{c}{2\sqrt{km}} = \frac{5.53 Ns/m}{2\sqrt{7.3 * 10^9 * 18}} = 5.35 * 10^{-3}$$

ISO68 where the density is equals to0.878 *10³ kg/m^3

$$v = \sqrt{\rho g h}$$

$$v = \sqrt{878 * 9.8 * 0.185} = 39.89 m/s$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 39.89 = 3.49 Ns/m$$

$$\xi = \frac{c}{2\sqrt{km}} = \frac{3.49 Ns/m}{2\sqrt{7.3 * 10^9 * 18}} = 4.81 * 10^{-6}$$

SAE 85W-140 where the density is equals tokg/m^3

$$v = \sqrt{\rho g h}$$

$$v = \sqrt{901 * 9.8 * 0.185} = 40.41 m/s$$

$$c = \frac{\pi * 0.0107}{2 * 0.122} * 40.41 = 5.56 Ns/m$$

$$\xi = \frac{c}{2\sqrt{km}} = \frac{5.56 Ns/m}{2\sqrt{7.3 * 10^9 * 18}} = 7.68 * 10^{-6}$$

The value for each damping oil factor are summarize as follows

Gear Oil type	Zita values
SAE J306	$5.29 * 10^{-6}$
SAE 90	0.027
SAE 140	$5.3 * 10^{-6}$
SAE 250	$4.81 * 10^{-6}$
ISO 100	$7.44 * 10^{-6}$
ISO220	$5.68 * 10^{-11}$
ISO 150	$5.35 * 10^{-3}$

ISO68	$4.81 * 10^{-6}$
SAE 85W-140	$7.68 * 10^{-6}$

Table 8 calculated value of Zita for different damping oil

According to the calculated value of damping factor the better number that can neighboring to one(ξ=1) comparing to the others is SAE 90 gear oil type for this system.

4.2.7 .Natural frequency values for different type of materials

To calculate the natural frequency value for the materials below the modulus of elasticity of the materials should be Known and these were found from the material type specification.

Material	Modulus of elasticity (Gpa)
Aluminum Alloy 1100	69
Copper Alloy	115
Gray cast Iron	169
Steel Alloy	200
Carbon and low steel alloy (ASTM)	103

Figure 3Modulous of elasticity for different metals

1. Aluminum Alloy 1100(69Gpa)

$$k = \frac{w}{\delta} \leftrightarrow \frac{3EI}{L^3}$$

$$E = 69Gpa$$

$$I = 3.71 * 10^8$$

$$L = 145mm$$

$$K = \frac{3 * 69Gpa * 3.71 * 10^8 mm^4}{145^3} = 25 * 10^3 \, {}^{N}/_{m}$$

$$\omega_n = \sqrt[2]{\frac{k}{m}} = \sqrt[2]{\frac{25 * 10^3 \, {}^{N}/_{m}}{18.7kg}} = 1347.09rad/sec$$

2. Copper Alloy (115Gpa)

$$k = \frac{w}{\delta} \leftrightarrow \frac{3EI}{L^3}$$

$$E = 115Gpa$$

$$I = 3.71 * 10^8$$

$$L = 145mm$$

$$K = \frac{3 * 115Gpa * 3.71 * 10^8 mm^4}{145^3} = 41.98 * 10^6\, {}^{N}/_{m} ------------(4.101)$$

$$\omega_n = \sqrt[2]{\frac{k}{m}} = \sqrt[2]{\frac{41.98 * 10^6\, N/m}{18.7kg}} = 2769rad/sec$$

3 Gray cast Iron (169)

$$k = \frac{w}{\delta} \leftrightarrow \frac{3EI}{L^3} -----------------------------(4.100)$$

$$E = 169Gpa$$

$$I = 3.71 * 10^8$$

$$L = 145mm$$

$$K = \frac{3 * 169Gpa * 3.71 * 10^8 mm^4}{145^3} = 34.54 * 10^7\, {}^{N}/_{m} ------------- (4.101)$$

$$\omega_n = \sqrt[2]{\frac{k}{m}} = \sqrt[2]{\frac{34.54 * 10^7\, {}^{N}/_{m}}{18.7kg}} = 4297.74rad/sec$$

4Steel Alloy 200

$$k = \frac{w}{\delta} \leftrightarrow \frac{3EI}{L^3} -----------------------------(4.100)$$

$$E = 200Gpa$$

$$I = 3.71 * 10^8$$

$$L = 145mm$$

$$K = \frac{3 * 200Gpa * 3.71 * 10^8 mm^4}{145^3} = 7.30 * 10^9 \, {}^{N}/_{m} ----------------(4.101)$$

$$\omega_n = \sqrt[2]{\frac{k}{m}} = \sqrt[2]{\frac{7.3 * 10^9 \, N/m}{18.7kg}} = \frac{5063.05rad}{sec}$$

5Carbon and low steel alloy (ASTM)(103Gpa)

$$k = \frac{w}{\delta} \leftrightarrow \frac{3EI}{L^3} ----(4.100)$$

$$E = 103Gpa$$

$$I = 3.71 * 10^8$$

$$L = 145mm$$

$$K = \frac{3 * 103Gpa * 3.71 * 10^8 mm^4}{145^3} = 3.78 * 10^9 \, {}^{N}/_{m} --- (4.101)$$

$\omega_n = \sqrt[2]{\frac{k}{m}} = \sqrt[2]{\frac{3.78*10^9 \, {}^{N}/_{m}}{18.7kg}} = 3091$Rad/sec

Material type	Natural frequency (rad/sec)	Natural frequency (Hz)
Aluminum Alloy 1100	1347.09	214.38
Copper Alloy	2769.5	440.70
Gray cast Iron	4297.74	684.00
Steel Alloy	5063	805.80
Carbon and low steel alloy (ASTM)	3091	491.94

Chapter five

Experimental Analysis

After vibration recording finished by using the vibration meter the data was transferred from data logger 2 GB SD card in to the computer and gives the data in excel format. Vibration recording result is for high speed zone of 1st, 2nd, 3rd and 4th gear.

Any vibration is described by the time history of motion, where the amplitude of the motion is expressed in terms of displacement, velocity or acceleration. Sinusoidal vibration is the simplest motion, and can be fully described by straightforward mathematical equations.[31]

If the random vibration is considered as an infinite number of infinitesimal shocks, the overall G_{rms} may result in a fatigue related structural failure of a component due to the intermittent shocks associated with the random excitation. In this case, the problem might be corrected by reducing the overall G_{rms} or by increasing the strength of the component [31].

To compute the RMS values from these points we need to compute the area under the curve defined by the breakpoints [32]. From the points concidered in this region curve the area under this curve will be concidered as the curve of the peak amplitude.

The filtered data of highest value of amplitude relative to other amplitude value are 16 values from the total of 64 recording values . While The frequency value is calculated from the equation.

$$frequency = \frac{1}{period} \text{-------------------------------} (5.1)$$

By using equations random vibration equatons and formulla the following tables ware prepared in order to find the single RMS velocity.

5.1.Plots of vibration measurments by using Matlab

5.1.1. Fristgear high speed zone random velocity calculation

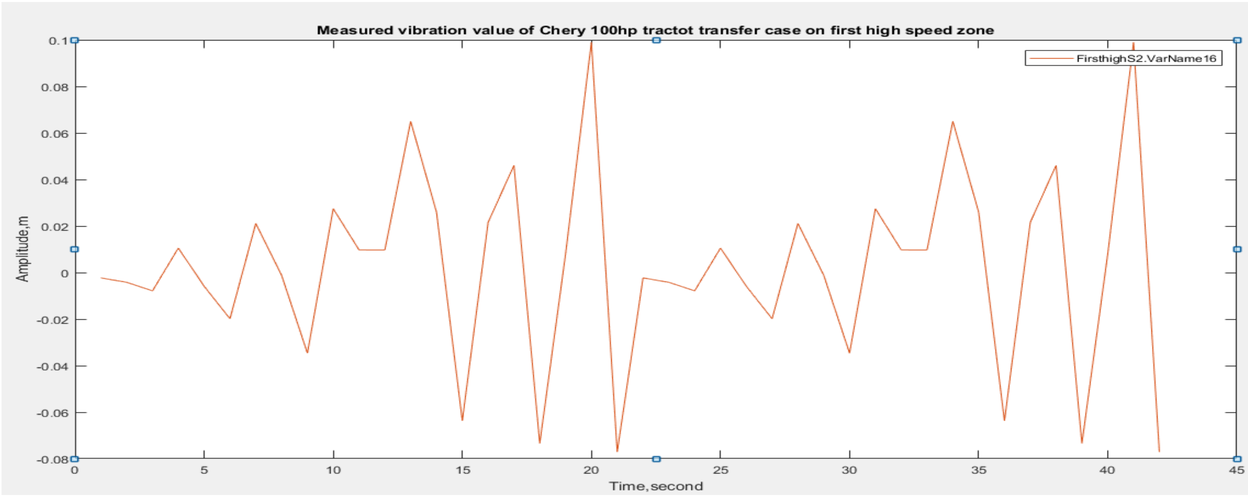


Figure 5 measured vibration value for first gear high speed zone vibration

Freq(Hz)	P(sec)	Amp(m)	Log (Freq)	Log(A)	$\log_{fn+1} - \log_{fn}$	$\log_{an+1} - \log_{an}$	Slope
1.39738	0.715625	25.7	0.145314	1.409933	-7.28243E-06	-0.19774551	3.6873E-05
1.39756	0.715637	16.3	0.145307	1.212188	-6.67545E-06	-0.03900133	0.00017116
1.39735	0.715648	14.9	0.145301	1.173186	-7.2822E-06	-0.03330718	0.00028637
1.39732	0.71566	13.8	0.145293	1.139879	-6.67524E-06	-0.01930515	0.00035775
1.39729	0.715671	13.2	0.145287	1.120574	-7.28196E-06	0.01296497	-0.00056166
1.39767	0.715683	13.6	0.145279	1.133539	-6.67503E-06	-0.03662895	0.00018234
1.39745	0.715694	12.5	0.14573	1.09691	-7.28173E-06	0.055378331	-0.00013149
1.39722	0.715706	14.2	0.145265	1.152288	-7.28161E-06	0.135513386	-5.3733E-05
1.39798	0.715718	19.4	0.145258	1.287802	-6.6747E-06	-7.28161E-06	0.91665938
1.39777	0.715729	22.6	0.145251	1.354108	-7.28137E-06	-0.0383992	0.00089891
1.39753	0.715741	21.3	0.145244	1.32838	-6.67449E-06	-0.03834992	0.00017064
1.39732	0.715752	19.5	0.145237	1.290035	-7.28114E-06	0.025935734	-0.00028074
1.39709	0.715764	20.7	0.14523	1.31597	-6.67427E-06	-0.00630178	0.00152695
1.39787	0.715775	20.4	0.145223	1.30963	-7.28091E-06	-0.119708	6.5697E-05
1.39706	0.715787	15.8	0.145216	1.198657	-7.28078E-06	-0.0372890	0.00019525
1.39704	0.715799	14.5	0.145209	1.161368			

Slope	Offset	$f\hat{}(slope-4)$	$\frac{offset}{(2\pi f)^4}$	$v=\frac{offset}{(2\pi)^2}*\bar{f}^{slope-2}$
3.68273E-05	25.69968	0.262269	0.004333	0.1137
0.00017116	16.29907	0.262299	0.002748	0.0721
0.000218637	14.89891	0.262319	0.002513	0.0659
0.000345775	13.7984	0.262348	0.002327	0.0611
-0.00056166	13.20248	0.262284	0.002227	0.0584
0.000182234	13.59917	0.262367	0.002294	0.0602
-0.00013149	12.50055	0.262356	0.002109	0.0553
-5.3733E-05	14.20026	0.26238	0.002396	0.0629
0.916651938	14.27745	0.356549	0.002409	0.0859
0.000189891	22.59856	0.262435	0.003813	0.1001
0.000174064	21.29876	0.262451	0.003594	0.0943
-0.00028074	19.50183	0.262428	0.003291	0.0864
0.001052695	20.69271	0.262562	0.003492	0.0917
6.56097E-05	20.39955	0.262492	0.003443	0.0904
0.000195252	15.79897	0.262521	0.002666	0.07

Table 9 Random Velocity of first high speed zone

$$\sum_{1}^{15} velocity = \frac{1.1684m}{s} = \frac{1168.4mm}{s}$$

$$\sqrt{velocity} = \sqrt{11168.4mm/s} = 34.18mm/s$$

$$G = 386.09\frac{in}{s} = 9.8066\frac{m}{s}, v_{rms} = \frac{34.18}{9.866} = 3.48mm/sec$$

According to the ISO standard the velocity of first gear high speed zone falls on zone C which is unsatisfactory for long-term, continuous operations (alert level).Then we can sum the areas as before to get the mean-square velocity, and take the square root to get an RMS velocity value for the random spectrum. Also, when using acceleration units in G's, it needs to be apply a conversion factor to get a suitable velocity unit. Common conversions are to convert G's to inches/s2 or to m/s G = 386.09 in/s^2 = 9.80665 m/s^2[32].

5.1.2 Second gear high speed zone random velocity calculation

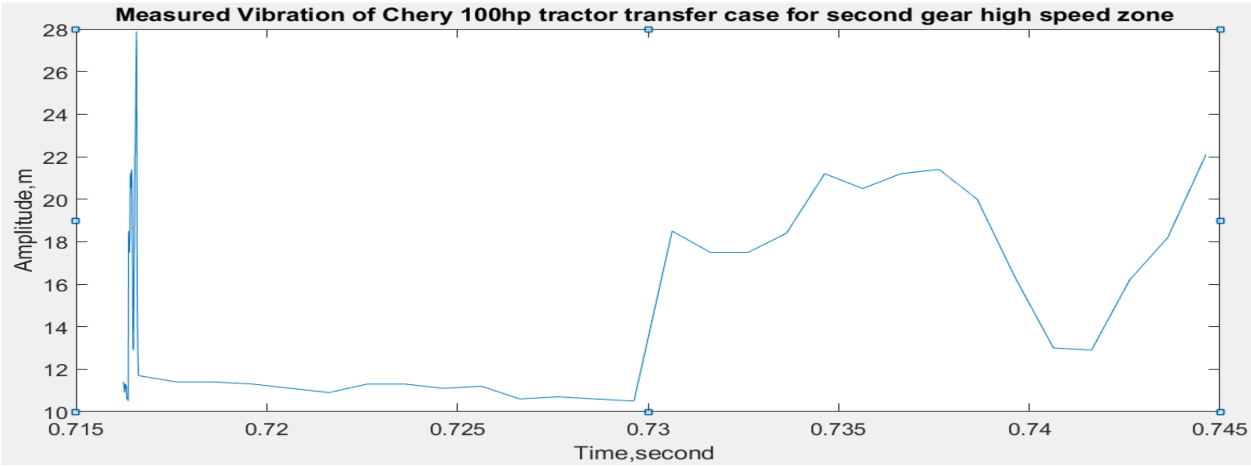


Figure 5.1 Measured vibration value for second gear high speed zone vibration

Freq(Hz)	Period(sec)	Amp(m)	Log (Freq)	Log(amp)	$\log_{fn+1} - \log_{fn}$	$\log_{an+1} - \log_{an}$	Slope
1.395912	0.716377	18.5	0.144858	1.267172	-7.01658E-6	-0.02413	0.00021
1.39589	0.716389	17.5	0.144851	1.243038	-7.01646E-6	0	_
1.395867	0.7164	17.5	0.144844	1.243038	-7.01635E-6	0.02178	-0.00032
1.395845	0.716412	18.4	0.144837	1.264818	7.01624E-6	0.061518	-0.00011
1.395822	0.716424	21.2	0.14483	1.326336	-7.01612E-6	-0.01458	0.000481
1.3958	0.716435	20.5	0.144823	1.311754	-7.01601E-6	0.014582	-0.00048
1.395777	0.716447	21.2	0.144816	1.326336	-7.0159E-06	0.004078	-0.00172
1.395755	0.716458	21.4	0.144809	1.330414	-7.01578E-6	-0.02938	0.000239
1.395732	0.71647	20	0.144802	1.30103	-7.01567E-6	-0.08884	7.9E-05
1.395709	0.716481	16.3	0.144795	1.212188	-7.01544E-6	-0.09824	7.14E-05
1.395687	0.716493	13	0.144788	1.113943	-7.01544E-6	-0.00335	0.002092
1.395664	0.716505	12.9	0.144781	1.11059	-7.01533E-6	0.098925	-7.1E-05
1.395642	0.716516	16.2	0.144774	1.209515	-7.01522E-6	0.050556	-0.00014
1.395619	0.716528	18.2	0.144767	1.260071	-7.0151E-06	0.084321	-8.3E-05
1.395597	0.716539	22.1	0.14476	1.344392	-7.01499E-6	0	0
1.395574	0.716551	22.1	0.144753	1.344392	7.01488E-6	0.032185	-0.00022
1.395552	0.716563	23.8	0.144746	1.376577	-7.01476E-6	0.046669	-0.00015
1.395529	0.716574	26.5	0.144739	1.423246	-7.01465E-6	0.022358	-0.00031

Table 10 Random Velocity of second high speed zone

Slope	Offset	$f^{\wedge}(slope-2)$	$\frac{offset}{(2\pi f)^2}$	$v=\frac{offset}{(2\pi)^2}*f^{slope-2}$
0.00021	1.000097	0.513246227	0.025358	0.013015
-0.00032	0.999893	0.513174484	0.025353	0.013011
-0.00011	0.999962	0.513226685	0.025355	0.013013
0.000481	1.00016	0.513345153	0.02536	0.013018
-0.00048	0.99984	0.513197028	0.025352	0.013011
-0.00172	0.999426	0.51300157	0.025341	0.013
0.000239	1.00008	0.513353411	0.025358	0.013018
7.9E-05	1.000026	0.513342644	0.025357	0.013017
7.14E-05	1.000024	0.513357935	0.025357	0.013017
0.002092	1.000698	0.513720446	0.025374	0.013035
-7.1E-05	0.999976	0.513366748	0.025355	0.013017
-0.00014	0.999954	0.513371723	0.025355	0.013016
-8.3E-05	0.999972	0.513397818	0.025355	0.013017
-0.00022	0.999927	0.513407931	0.025354	0.013017
-0.00015	0.99995	0.513436	0.025355	0.013018
-0.00031	0.999895	0.513424717	0.025353	0.013017

$$\sum_{1}^{15} velocity = \frac{0.2082m}{s} = \frac{208.2mm}{s}$$

$$\sqrt{velocity} = \sqrt{208.2mm/s} = 14.42mm/s$$

$$G = 386.09\frac{in}{s} = 9.8066\frac{m}{s}, v_{rms} = \frac{14.42}{9.866} = 1.472mm/sec$$

According to the ISO standard the velocity of second gear high speed zone falls on zone B which isacceptable condition

5.1.3. Third gear high speed zone random velocity calculation

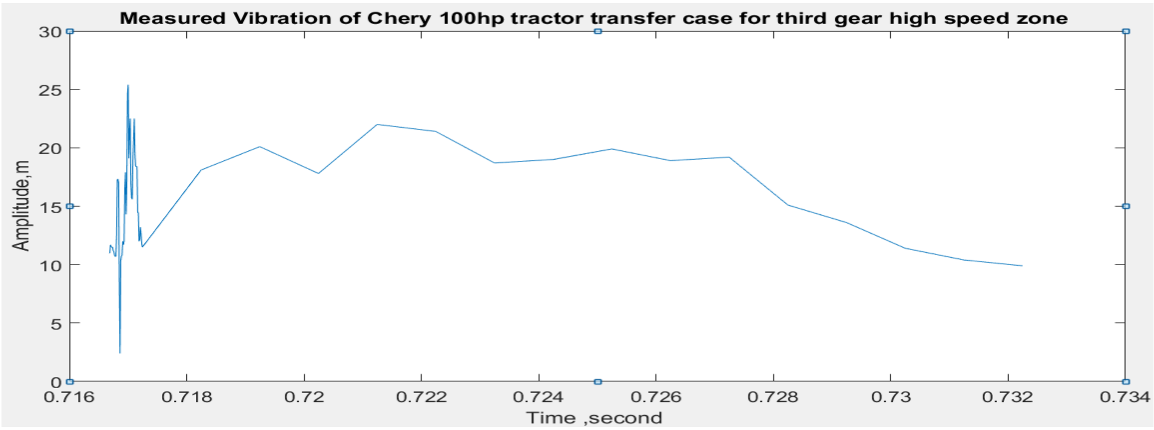


Figure 5.2 Measured vibration value for third gear high speed zone vibration

Freq(Hz)	Period(sec)	Amp(m)	Log(Freq)	Log(amp)	$\log_{fn+1} - \log_{fn}$	$\log_{an+1} - \log_{an}$	Slope
1.394808	0.716944	16.5	0.144514	1.217484	-2.25169E-05	0.035369	-0.00064
1.394786	0.716956	17.9	0.144507	1.252853	-2.25162E-05	-0.09752	0.000231
1.394763	0.716968	14.3	0.1445	1.155336	0	0.075113	-0.0003
1.394741	0.716979	17	0.144493	1.230449	-2.25154E-05	0.160486	-0.00014
1.394718	0.716991	24.6	0.144486	1.390935	-2.25147E-05	0.013899	-0.00162
1.394696	0.717002	25.4	0.144479	1.404834	-2.2514E-05	-0.1238	0.000182
1.394673	0.717014	19.1	0.144472	1.281033	-2.25132E-05	0.04938	-0.00046
1.394651	0.717025	21.4	0.144465	1.330414	-2.25125E-05	0.021769	-0.00032
1.394628	0.717037	22.5	0.144458	1.352183	-17.6053719	-0.12173	0.000185
1.394606	0.717049	17	0.144451	1.230449	-2.25111E-05	-0.03455	0.000652
1.394583	0.71706	15.7	0.144444	1.1959	-2.25103E-05	-0.00278	0.008111
1.394561	0.717072	15.6	0.144437	1.193125	-2.25096E-05	0.085629	-0.00026
1.394538	0.717083	19	0.14443	1.278754	-2.25089E-05	0.037217	-0.0006
1.394516	0.717095	20.7	0.144423	1.31597	-2.25082E-05	0.036212	-0.00062
1.394493	0.717106	22.5	0.144416	1.352183	-2.25074E-05	-0.06215	0.000362
1.394471	0.717118	19.5	0.144409	1.290035	-2.25067E-05		

Slope	Offset	f^(slope − 2)	$\frac{\text{offset}}{(2\pi)^2}$	$v = \frac{\text{offset}}{(2\pi)^2} * f^{\text{slope}-2}$
-0.00064	0.999788	0.514049	0.418463	0.21511
0.000231	1.000077	0.513975	0.453837	0.233261
-0.0003	0.9999	0.514019	0.362627	0.186397
-0.00014	0.999953	0.513782	0.431072	0.221477
-0.00162	0.999461	0.514107	0.624094	0.320851
0.000182	1.00006	0.514014	0.644003	0.331027
-0.00046	0.999848	0.514054	0.484373	0.248994
-0.00032	0.999893	0.514157	0.542676	0.279021
0.000185	1.000062	0.514254	0.570475	0.293369
0.000652	1.000217	0.515548	0.430959	0.22218
0.008111	1.002701	0.51413	0.397017	0.204118
-0.00026	0.999913	0.514088	0.395588	0.203367
-0.0006	0.999799	0.514102	0.481861	0.247726
-0.00062	0.999793	0.514287	0.524978	0.269989
0.000362	1.00012	1	0.570441	0.570441

Table 11 Random Velocity of third high speed zone

$$\sum_{1}^{15} velocity \; = \frac{4.047m}{s} = \frac{4047mm}{s}$$

$$\sqrt{velocity} = \sqrt{\frac{4047mm}{s}} = 63.37mm/mm/s$$

$$G = 386.09\frac{in}{s} = 9.8066\frac{m}{s}, v_{rms} = \frac{63.37}{9.866} = 6.46mm/sec$$

According to the ISO standard the velocity of third gear high speed zone falls on zone C which is unsatisfactory for long term operation.

5.1.4 Forth gear high speed zone random velocity calculation

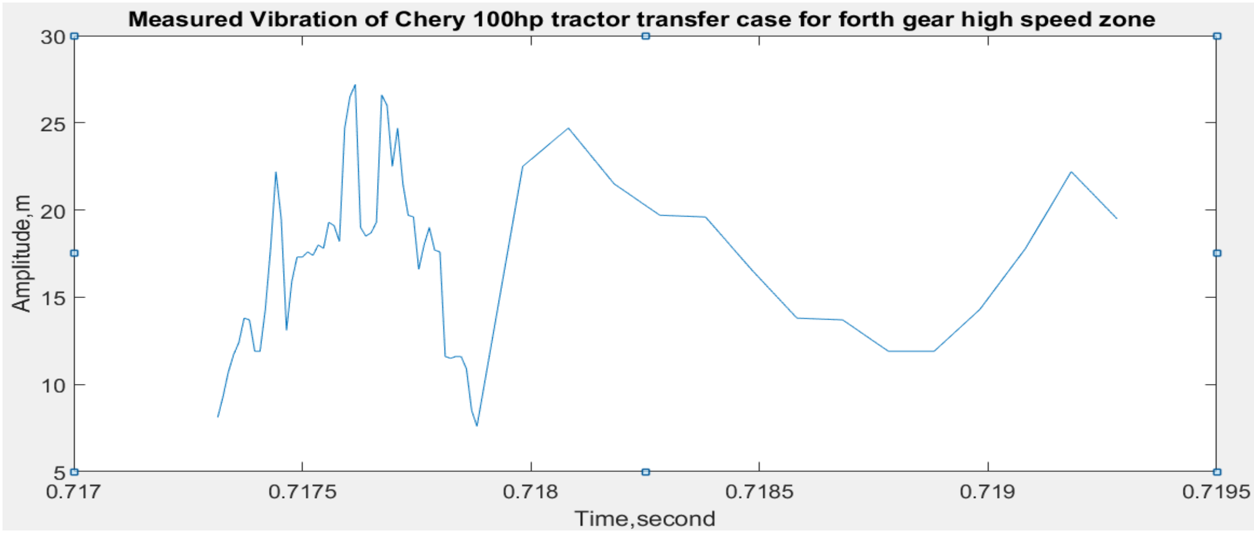


Figure 5.3 Measured vibration value for forth gear high speed zone vibration

Freq(Hz)	Period(sec)	Amp(m)	Log (freq)	Log(amp)	$\log_{fn+1} - \log_{fn}$	$\log_{an+1} - \log_{an}$	Slope
1.393593	0.717569	19.1	0.144136	1.281033	-2.24777E-05	-0.02096	0.001072
1.393571	0.717581	18.2	0.144129	1.260071	-7.00481E-06	0.132626	-5.3E-05
1.393548	0.717593	24.7	0.144122	1.392697	7.00458E-06	0.030549	0.000229
1.393526	0.717604	26.5	0.144115	1.423246	-7.00458E-06	0.011323	-0.00062
1.393503	0.717616	27.2	0.144108	1.434569	-7.00447E-06	-0.15582	4.5E-05
1.393481	0.717627	19	0.144101	1.278754	-7.00435E-06	-0.01158	0.000605
1.393458	0.717639	18.5	0.144094	1.267172	-7.00424E-06	0.00467	-0.0015
1.393436	0.71765	18.7	0.144087	1.271842	-7.00413E-06	0.013716	-0.00051
1.393414	0.717662	19.3	0.14408	1.285557	-7.00402E-06	0.139324	-5E-05
1.393391	0.717674	26.6	0.144073	1.424882	-7.0039E-06	-0.00991	0.000707
1.393369	0.717685	26	0.144066	1.414973	-7.00379E-06	-0.06279	0.000112
1.393346	0.717697	22.5	0.144059	1.352183	-7.00368E-06	0.040514	-0.00017
1.393324	0.717708	24.7	0.144052	1.392697	-7.00356E-06	-0.06026	0.000116
1.393301	0.71772	21.5	0.144045	1.332438	-7.00345E-06	-0.03797	0.000184
1.393279	0.717731	19.7	0.144038	1.294466	-7.00334E-06	-0.00221	0.003169

Table 12 Random Velocity of forth high speed zone

Slope	Offset	$f^\wedge(\text{slope}-2)$	$\frac{\text{offset}}{(2\pi)^2}$	$v = \frac{\text{offset}}{(2\pi)^2} * f^{\text{slope}-2}$
0.001072	19.10033	0.515089	0.484308	0.249377
-5.3E-05	18.19862	0.514913	0.461444	0.237634
0.000229	24.70507	0.514978	0.626422	0.322513
-0.00062	26.4996	0.51485	0.671924	0.346027
4.5E-05	27.19454	0.51498	0.689545	0.355179
0.000605	19.00946	0.515092	0.482004	0.248111
-0.0015	18.50313	0.514749	0.469165	0.24159
-0.00051	18.70031	0.514935	0.474165	0.244209
-5E-05	19.29547	0.51503	0.489256	0.252053
0.000707	26.59902	0.515176	0.674445	0.3474
0.000112	26.00149	0.515091	0.659294	0.339575
-0.00017	22.49913	0.515059	0.570488	0.293873
0.000116	24.69849	0.515125	0.626255	0.322617
0.000184	21.47742	0.515153	0.544581	0.28083
0.003169	19.7	0.51568	0.499513	0

$$\sum_{1}^{15} velocity = \frac{4.060m}{s} = \frac{4060mm}{s}$$

$$\sqrt{velocity} = \sqrt{\frac{4060mm}{s}} = 63.88mm/s$$

$$G = 386.09\frac{in}{s} = 9.8066\frac{m}{s}, v_{rms} = \frac{63.88}{9.866} = 6.51mm/sec$$

According to the ISO standard the velocity of forth gear high speed zone falls on zone C which is unsatisfactory for long term operation.

5.1.5. First gear lowspeed zone random velocity calculation

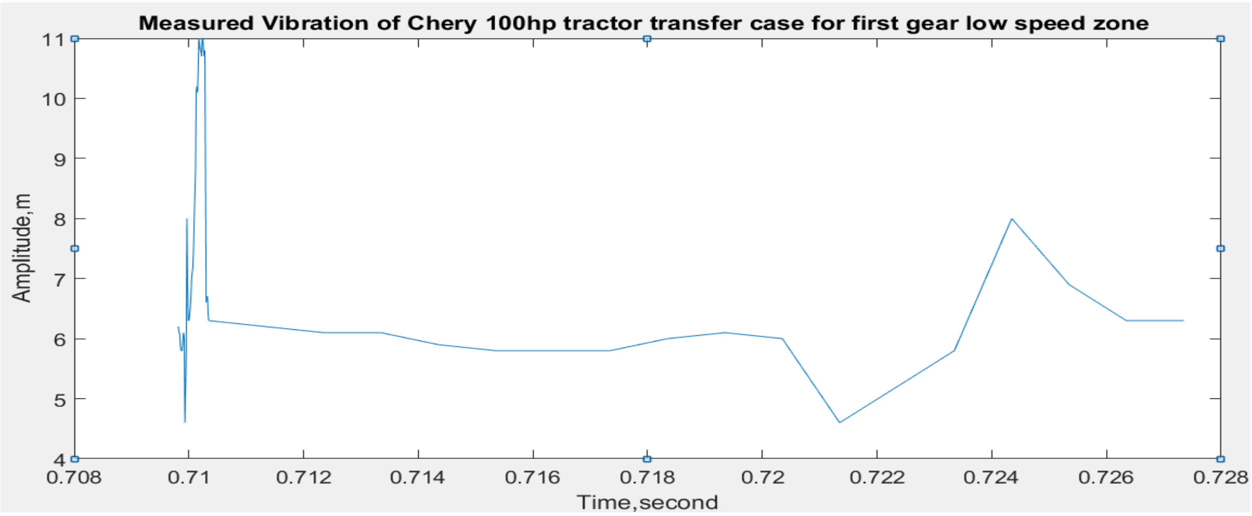


Figure 5.4 Measured vibration value for first gear lows peed zone vibration

Freq(Hz)	Period(sec)	Amp(m)	Log (frq)	Log(amp)	$\log_{fn+1} - \log_{fn}$	$\log_{an+1} - \log_{an}$	Slope
1.408244	0.710104	8.4	0.148678	0.924279	0.020203	-7.07856E-06	-0.0035
1.408221	0.710116	8.8	0.148671	0.944483	0.059839	-7.07845E-06	-0.0012
1.408198	0.710127	10.1	0.148664	1.004321	0.004279	-7.07833E-06	-0.0165
1.408175	0.710139	10.2	0.148657	1.0086	-0.00428	-7.07821E-06	0.01654
1.408152	0.71015	10.1	0.14865	1.004321	0.016868	-7.0781E-06	-0.0042
1.408129	0.710162	10.5	0.148643	1.021189	0.020203	-7.07798E-06	-0.0035
1.408106	0.710174	11	0.148635	1.041393	-0.00397	-7.07787E-06	0.01785
1.408083	0.710185	10.9	0.148628	1.037426	-0.004	-7.07775E-06	0.01768
1.40806	0.710197	10.8	0.148621	1.033424	0	-7.07764E-06	0
1.408038	0.710208	10.8	0.148614	1.033424	-0.00404	-7.07752E-06	0.01752
1.408015	0.71022	10.7	0.148607	1.029384	0.012009	-7.07741E-06	-0.0059
1.407992	0.710231	11	0.1486	1.041393	0	-7.07729E-06	0
1.407969	0.710243	11	0.148593	1.041393	-0.00797	-7.07718E-06	0.00888
1.407946	0.710255	11	0.148586	1.033424	-0.00404	-7.07706E-06	0.01752
1.407923	0.710266	10.8	0.148579	1.029384	0.00404	-7.07695E-06	-0.0175
1.4079	0.710278	10.7	0.148572	1.033424	-0.0939	7.0768E-06	7.54E-05
1.407877	0.710289	10.8	0.148565	0.939519			

Slope	Offset	f^(slope − 2)	$\frac{\text{offset}}{(2\pi)^2}$	$v = \frac{\text{offset}}{(2\pi)^2} * f^{\text{slope}-2}$
-0.00035	8.401008	0.504187	0.213016	1.789548
-0.00012	8.800356	0.504244	0.223142	1.963728
-0.00165	10.10572	0.503995	0.256241	2.589496
0.001654	10.19423	0.504583	0.258485	2.635052
-0.00042	10.10145	0.504241	0.256132	2.587308
-0.00035	10.50126	0.50427	0.26627	2.796169
0.001785	10.99328	0.504655	0.278746	3.06433
0.001768	10.89341	0.504668	0.276213	3.008902
0	10.8	0.504379	0.273845	2.957524
0.001752	10.79353	0.504698	0.273681	2.95398
-0.00059	10.70216	0.504311	0.271364	2.904179
0	11	0.504429	0.278916	3.068076
0.000888	10.99666	0.504599	0.278831	3.066212
0.001752	10.79353	0.504764	0.273681	2.953981
-0.00175	10.70641	0.504176	0.271472	2.90649
7.54E-05	10.79972	0.504508	0.273838	2.957371

Table 13 Random Velocity of first low speed zone

$$\sum_{1}^{15} velocity = \frac{44.20235m}{s} = \frac{44000mm}{s}$$

$$\sqrt{velocity} = \sqrt{44000mm} = 209.7mm/s$$

$$G = 386.09\frac{in}{s} = 9.8066\frac{m}{s}, v_{rms} = \frac{209.7}{9.866} = 21.39mm/sec$$

According to the ISO standard the velocity of first gear low speed zone falls on zone D which is bad condition cause long term damage to the machine.

5.1.6. Second gearlow speed zone random velocity calculation

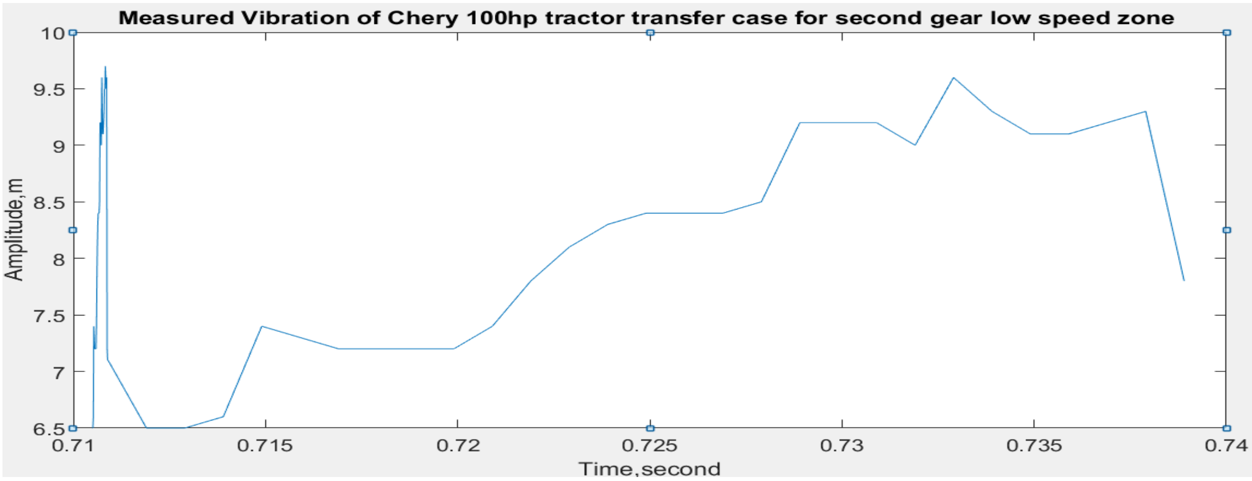


Figure 5.5 measured vibration value for second gear low speed zone vibration

Freq(Hz)	Period(sec)	Amp(m)	Log (Frq)	Log(amp)	$\log_{fn+1} - \log_{fn}$	$\log_{an+1} - \log_{an}$	Slope
1.407212	0.710625	8.1	0.14836	0.908485	-2.29191E-05	-7.07337E-06	0.308623
1.407189	0.710637	8.3	0.148352	0.919078	-2.29184E-05	-7.07326E-06	0.308628
1.407166	0.710648	8.4	0.148345	0.924279	-2.29176E-05	-7.07314E-06	0.308633
1.407143	0.71066	8.4	0.148338	0.924279	-2.29169E-05	-7.07303E-06	0.308638
1.40712	0.710671	8.4	0.148331	0.924279	-2.29161E-05	-7.07291E-06	0.308643
1.407097	0.710683	8.5	0.148324	0.929419	-2.29154E-05	-7.0728E-06	0.308648
1.407074	0.710694	9.2	0.148317	0.963788	-2.29147E-05	-7.07268E-06	0.308653
1.407052	0.710706	9.2	0.14831	0.963788	-2.29139E-05	-7.07257E-06	0.308658
1.407029	0.710718	9.2	0.148303	0.963788	-2.29132E-05	-7.07245E-06	0.308663
1.407006	0.710729	9	0.148296	0.954243	-2.29124E-05	-7.07234E-06	0.308668
1.406983	0.710741	9.6	0.148289	0.982271	-2.29117E-05	-7.07222E-06	0.308673
1.40696	0.710752	9.3	0.148282	0.968483	-2.29109E-05	-7.07211E-06	0.308678
1.406937	0.710764	9.1	0.148275	0.959041	-2.29102E-05	-7.07199E-06	0.308683
1.406914	0.710775	9.1	0.148268	0.959041	-2.29094E-05	-7.07188E-06	0.308688
1.406891	0.710787	9.2	0.148261	0.963788	-2.29087E-05	-7.07176E-06	0.308693
1.406868	0.710799	9.3	0.148253	0.968483	-2.29079E-05	-7.07165E-06	0.308698

Table 14 Random Velocity of second low speed zone

Slope	Offset	$f\hat{}(slope-2)$	$\frac{offset}{(2\pi)^2}$	$v=\frac{offset}{(2\pi)^2}*f^{slope-2}$
0.308623	1.111187	0.561136	0.028175	0.01581
0.308628	1.111183	0.561152	0.028175	0.015811
0.308633	1.11118	0.561169	0.028175	0.015811
0.308638	1.111176	0.561185	0.028175	0.015811
0.308643	1.111172	0.561202	0.028175	0.015812
0.308648	1.111169	0.561218	0.028175	0.015812
0.308653	1.111165	0.561235	0.028175	0.015813
0.308658	1.111161	0.561251	0.028175	0.015813
0.308663	1.111158	0.561267	0.028175	0.015813
0.308668	1.111154	0.561284	0.028174	0.015814
0.308673	1.11115	0.5613	0.028174	0.015814
0.308678	1.111147	0.561317	0.028174	0.015815
0.308683	1.111143	0.561333	0.028174	0.015815
0.308688	1.111139	0.561349	0.028174	0.015815
0.308693	1.111136	0.561366	0.028174	0.015816
0.308698	1.111132	0.561382	0.028174	0.015816

$$\sum_{1}^{15} velocity \;=\frac{0.253m}{s}=\frac{253mm}{s}$$

$$\sqrt{velocity}=\sqrt{253mm/s}=15.9mm/s$$

$$G=386.09\frac{in}{s}=9.8066\frac{m}{s},v_{rms}=\frac{15.9}{9.866}=1.622mm/sec$$

According to the ISO standard the velocity of second gear low speed zone falls on zone Awhich is excellent.

5.1.7. Third gear low speed zone random velocity calculation

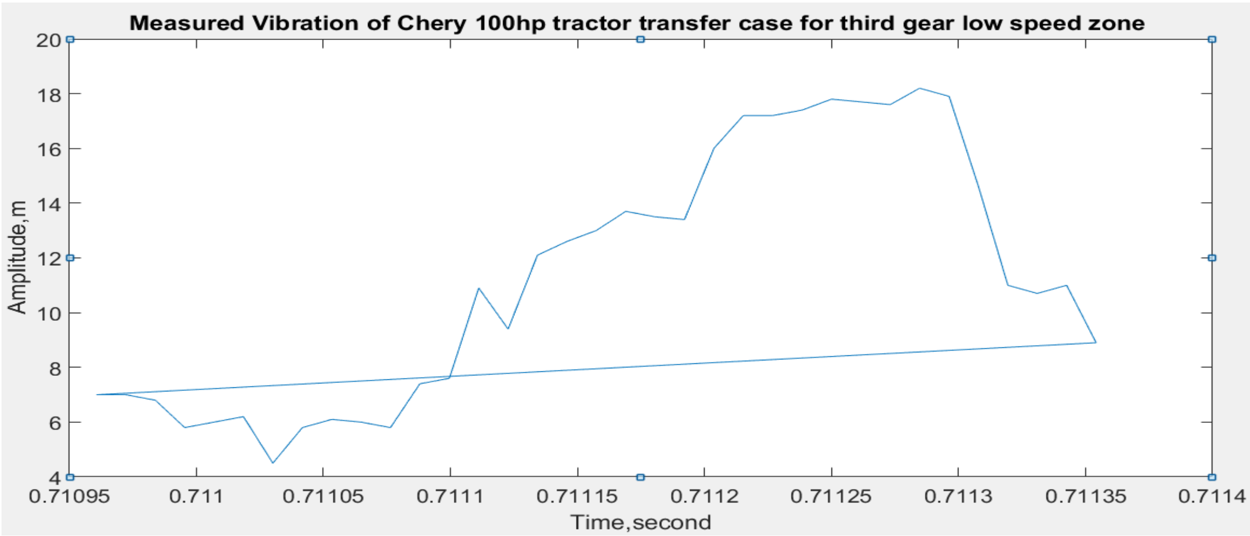


Figure 5.6Measured vibration value for third gear low speed zone vibration

Freq(Hz)	Period(sec)	Amp(m)	Log (Frq)	Log(amp)	$\log_{fn+1} - \log_{fn}$	$\log_{an+1} - \log_{an}$	Slope
1.406204	0.711134	12.1	0.148048	1.082785	0.017585	-7.06831E-06	-0.0004
1.406181	0.711146	12.6	0.148041	1.100371	0.013573	-7.06819E-06	-0.00052
1.406158	0.711157	13	0.148034	1.113943	0.022777	-7.06808E-06	-0.00031
1.406136	0.711169	13.7	0.148027	1.136721	-0.00639	-7.06796E-06	0.001107
1.406113	0.711181	13.5	0.14802	1.130334	-0.00323	-7.06785E-06	0.002189
1.40609	0.711192	13.4	0.148013	1.127105	0.077015	-7.06773E-06	-9.2E-05
1.406067	0.711204	16	0.148006	1.20412	0.031408	-7.06762E-06	-0.00023
1.406044	0.711215	17.2	0.147999	1.235528	0	-7.0675E-06	0
1.406021	0.711227	17.2	0.147992	1.235528	0.005021	-7.06739E-06	-0.00141
1.405998	0.711238	17.4	0.147985	1.240549	0.009871	-7.06727E-06	-0.00072
1.405975	0.71125	17.8	0.147978	1.25042	-0.00245	-7.06716E-06	0.002888
1.405953	0.711262	17.7	0.147971	1.247973	-0.00246	-7.06704E-06	0.002872
1.40593	0.711273	17.6	0.147964	1.245513	0.014559	-7.06693E-06	-0.00049
1.405907	0.711285	18.2	0.147957	1.260071	-0.00722	-7.06681E-06	0.000979
1.405884	0.711296	17.9	0.147949	1.252853	-0.0885	-7.0667E-06	7.98E-05
1.405861	0.711308	14.6	0.147942	1.164353			

Slope	Offset	$\mathrm{f}^\wedge(\mathrm{slope}-2)$	$\frac{\mathrm{offset}}{(2\pi)^2}$	$\mathrm{v}=\frac{\mathrm{offset}}{(2\pi)^2} * \mathrm{f}^{\mathrm{slope}-2}$
-0.0004	12.10166	0.505659	0.30685	0.155161
-0.00052	12.60224	0.505655	0.319542	0.161578
-0.00031	13.00138	0.505708	0.329663	0.166713
0.001107	13.69483	0.505969	0.347246	0.175696
0.002189	13.48993	0.506172	0.342051	0.173136
-9.2E-05	13.40042	0.505795	0.339781	0.171859
-0.00023	16.00123	0.505788	0.405727	0.205212
0	17.2	0.505844	0.436123	0.22061
-0.00141	17.20825	0.505618	0.436332	0.220617
-0.00072	17.40425	0.505753	0.441302	0.22319
0.002888	17.78249	0.506391	0.450893	0.228328
0.002872	17.68269	0.506405	0.448362	0.227053
-0.00049	17.60291	0.505842	0.446339	0.225777
0.000979	18.19393	0.506111	0.461325	0.233482
7.98E-05	17.89951	0.505973	0.45386	0.229641

Table 15 Random velocity of third low speed zone

$$\sum_{1}^{15} velocity = \frac{3.018m}{s} = \frac{3018mm}{s}$$

$$\sqrt{velocity} = \sqrt{3018mm/s} = 54.94mm/s$$

$$G = 386.09\frac{in}{s} = 9.8066\frac{m}{s}, v_{rms} = \frac{54.94}{9.866} = 5.602mm/sec$$

According to the ISO standard the velocity of third gear low speed zone falls on zone C which isUnsatisfactory for long term or continuous operation.

5.1.8 Forth gear low speed zone random velocity calculation

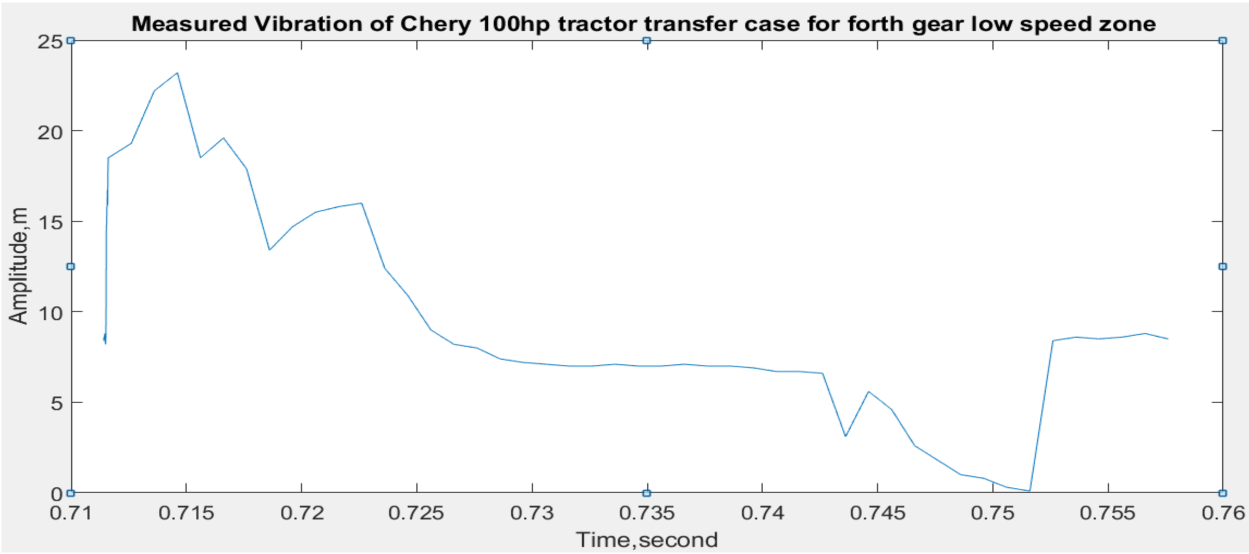


Figure 5.4 Measured vibration value for forth gear low speed zone vibration

Freq(Hz)	Period(sec)	Amp(m)	Log (Frq)	Log(amp)	$\log_{fn+1} - \log_{fn}$	$\log_{an+1} - \log_{an}$	Slope
1.405427	0.711528	14.8	0.147808	1.170262	0	-7.0644E-06	0
1.405404	0.711539	14.8	0.147801	1.170262	0.028395	-7.06428E-06	-0.00025
1.405381	0.711551	15.8	0.147794	1.198657	0	-7.06417E-06	0
1.405358	0.711563	15.8	0.147787	1.198657	0.024059	-7.06405E-06	-0.00029
1.405335	0.711574	16.7	0.14778	1.222716	-0.02132	-7.06394E-06	0.000331
1.405312	0.711586	15.9	0.147773	1.201397	0.046576	-7.06382E-06	-0.00015
1.405289	0.711597	17.7	0.147766	1.247973	0.019198	-7.06371E-06	-0.00037
1.405266	0.711609	18.5	0.147759	1.267172	0.018386	-0.000609871	-0.03317
1.403294	0.712609	19.3	0.147149	1.285557	0.060796	-0.000609016	-0.01002
1.401328	0.713609	22.2	0.14654	1.346353	0.019135	-0.000608163	-0.03178
1.399367	0.714609	23.2	0.145932	1.365488	-0.09832	-0.000607313	0.006177
1.397412	0.715609	18.5	0.145324	1.267172	0.025084	-0.000606464	-0.02418
1.395462	0.716609	19.6	0.144718	1.292256	-0.0394	-0.000605619	0.01537
1.393517	0.717609	17.9	0.144112	1.252853	-0.12575	-0.000604775	0.004809
1.391578	0.718609	13.4	0.143507	1.127105			

Slope	Offset	$f\hat{}(slop - 2)$	$\frac{offset}{(2\pi)^2}$	$v = \frac{offset}{(2\pi)^2} * f^{slope-2}$
0	14.8	0.506272	0.375269	0.189988
-0.00025	14.80125	0.506245	0.375301	0.189994
0	15.8	0.506305	0.400625	0.202838
-0.00029	15.80158	0.506271	0.400665	0.202845
0.000331	16.69812	0.506395	0.423397	0.214406
-0.00015	15.90082	0.506328	0.403181	0.204142
-0.00037	17.70222	0.506307	0.448857	0.22726
-0.03317	18.70997	0.500704	0.47441	0.237539
-0.01002	19.36562	0.506091	0.491035	0.248508
-0.03178	22.43936	0.503806	0.568972	0.286651
0.006177	23.1519	0.511727	0.587039	0.300404
-0.02418	18.65027	0.50797	0.472896	0.240217
0.01537	19.49987	0.516165	0.494439	0.255212
0.004809	17.87146	0.515785	0.453149	0.233727

Table 16 Random Velocity of forth low speed zone

$$\sum_{1}^{15} velocity = \frac{3.234m}{s} = \frac{3234mm}{s}$$

$$\sqrt{velocity} = \sqrt{3234mm/s} = 56.86mm/s$$

$$G = 386.09\frac{in}{s} = 9.8066\frac{m}{s}, v_{rms} = \frac{56.86}{9.866} = 5.799mm/sec$$

According to the ISO standard the velocity of forth gear low speed zone falls on zone C which is Unsatisfactory for long term or continuous operation.

Chapter six

Harmonic Response Analysis using ANSYS

Harmonic excitation refers to a sinusoidal external force of a certain frequency applied to a system. The response of a system to harmonic excitation is a very important topic because it is encountered very commonly and also covers the concept of resonance. Resonance occurs when the external excitation has the same frequency as the natural frequency of the system. It leads to large displacements and can cause a system to exceed its elastic range and fail structurally. At or near the natural frequency of a mode, the overall vibration shape ("operating deflection shape") of a structure will tend to be dominated by the mode shape of the resonance.(Singiresu.S, 2004)

6.1. CAD Modeling and Meshing

The Figure shows representation of Chery 100hp tractor transfer case .The CAD Model of transfer case has specification of length 145mm, Width 122mm and height 185mm.The CAD model is imported wit IGS format to the FEM design software ANSYS 17.2. Mesh model is prepared by using ANSYS 17.2 software meshing is carried out on all the outer and inner surfaces of the geometry. The transfer case is meshed with about 41046 nodes 20343 elements.

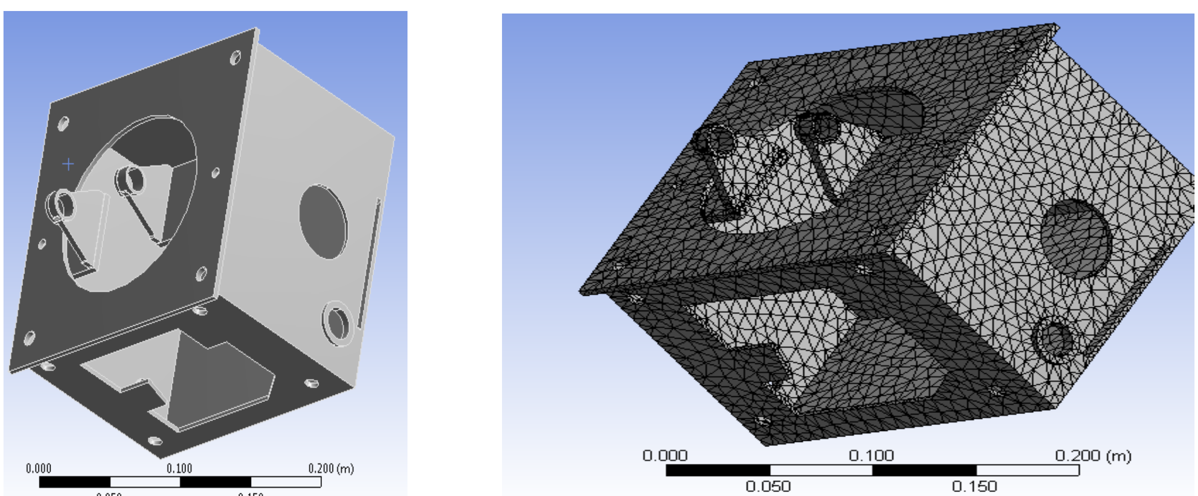


Figure 6 modeled and meshed transfer case

6.2. Boundary condition and material property

Chery 100hp tractor transmission is manufactured by casting process this transfer case is mounted below the differential by using bolts. Based on the laboratory spectrometer t test of the sample of the transfer case the material type is identified to be steel alloy .The Mechanical properties (Elastic modulus, Poisson ratio and density) are required for force vibration or harmonic response vibration analysis The material properties selected for the study of the transfer case are Elastic modulus – 2.0ePa, Poisson ratio- 0.30, density-7850 kg/m3 found from spectrometer test of sample.

After selecting the material the next step is boundary conditions. There are two predefined boundary condition in ANSYS for vibration analysis. These are free-free and fixed-fixed boundary conditions. In free-free-boundary conditions all degree of freedom are unconstraint.Fixed- fixed boundary condition is suitable for the transfer case analysis because transfer case is constraint by bolts.

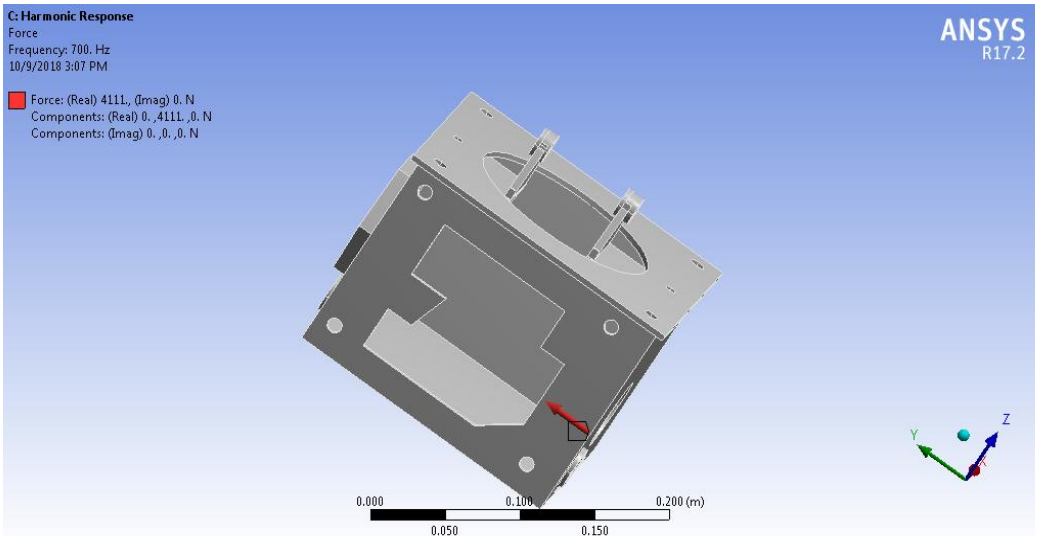
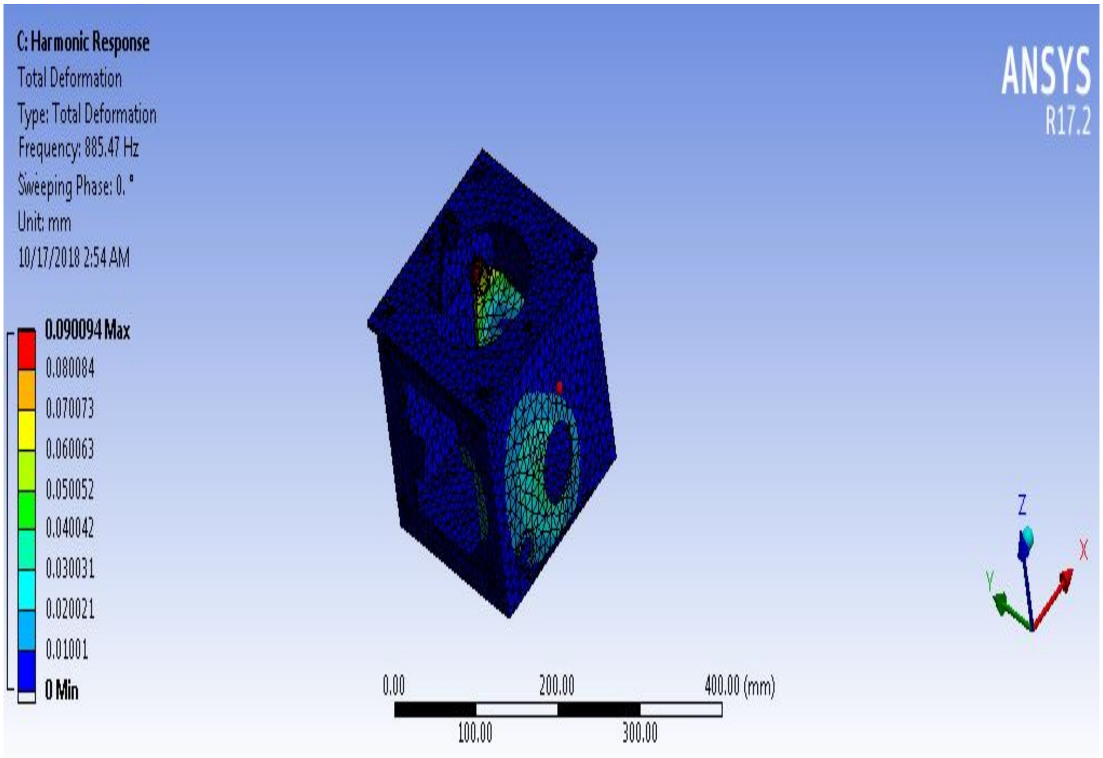


Figure 4 load application on transfer case


C: Harmonic Response
Total Deformation
Type: Total Deformation
Frequency: 885.47 Hz
Sweeping Phase: 0. °
Unit: mm
10/17/2018 2:54 AM
0.090094 Max
0.080084
0.070073
0.060063
0.050052
0.040042
0.030031
0.020021
0.01001
0 Min
ANSYS
R17.2
0.00
200.00
400.00 (mm)
100.00
300.00
Z
Y
X

Chapter Seven

Result and Discussion

7.1. Result from vibration recording

After vibration recording finished by using the vibration meter the data was transferred from data logger 2 GB SD card in to the computer and gives the data in excel format. Vibration recording result is for high speed zone of 1st, 2nd, 3rd and 4th gear. The Recording result statistical data of average, minimum, maximum, median, mode, standard deviation, and variance is tabulated as follows.

Amplitude recorded value of transfer case in four high and low speed zone in (m)							
Gear	Average	Variance	Median	Mode	S.D	Max	Min
1st	7.3323	3.759	6.1	5.8	1.9388	11	4.6
2nd	8.2539	1.0308	8.4	7.2	1.027	9.7	6.5
3rd	11.254	20.469	11	5.8	4.524	18.2	4.5
4th	9.9578	28.9515	8.5	7	5.383	23.2	0.7
1st	13.197	25.26	11.8	7.2	5.23	25.7	4.1
2nd	15.955	25.083	15.2	11.3	5.008	27.9	10.5
3rd	14.624	20.20	12.9	11.5	4.494	25.4	2.4
4th	16.9	25.155	17.65	11.6	4.885	27.2	7.6

Table 17 Statistical value of measured vibration

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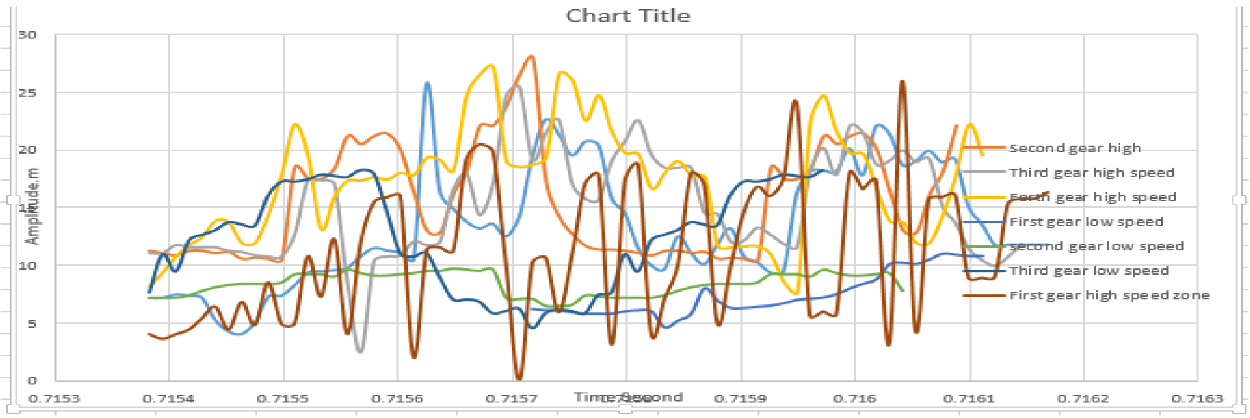

Figure 7Plots of eight speed zone vibration measurement

7.2. Physical interpretations of graphs

The main aim of the experimental study is to confirm the appropriateness of mathematical modeling and to identify the severity vibration speed zone by comparing the results with ISO standards.

The vibration on Chery 10hp tractor transfer case is measured by using vibration meter model and type is SDL800 vibration meter or data logger which has Basic accuracy of ± (5% + 2 digits); Meets ISO2954 standard .This instruments meets the Standard gives an accurate recording of root-mean-square (R.M.S.).After recording the results are plotted by using MATLAB software from the experimental recording of Chery 100hp tractor transfer case on different available speed zone on this tractor.

The general description of the above graphs shows the magnitude of vibration plotted as amplitude versus time plot. These amplitude versus time graph indicates that the motion is not repeat itself at a fixed periodic time accordingly motion is not periodic and sinusoidal. But the state of motion change with time and there exist the back and forth oscillation. The space between each cycle is not equal or proportional one is wide and the other is narrowed this leads to the vibration is random vibration .Any vibration that has the nature non repetition within a certain period of time is called random vibration or non-deterministic type of vibration. In 1st low,2nd high and 3rd high speed zone graphsshows there exist a sharp peak at the beginning of time history implies the maximum value of amplitude initially that means a minimum damping factor then after the amplitude dies out instantly and shows an increment again. All plots indicated that practical circumstances of the system shows there is an increment and decrement of the amplitude is neither proportional with time nor steady. Which specifies that the system is not damped.On the plot of 1st high, 3rd high and 1st low the value of amplitude gradually approach to the smaller but this amplitude decay is not rapid.The existence of high vibration displayed by a gradual increment of amplitude and takes long time to relax. High amplitude initially each to the peak value which is shows high severity of vibration and damped rapidly and slows down and energy dissipation for wider range of time. In plot of 3rd low speed zone the Initial amplitude is almost zero value but at the time increase the amplitude increasewith none uniformly but at the end time history the amplitude does not reach to peak value implies good situation of this speed zone.

7.3. Calculatedand measured RMS velocity comparison with ISO standards

The RMS velocity calculated by using random vibration method because of the vibration collected is not deterministic type of data.

Zone	Velocity(RMS) in mm/s	Condition according to ISO Standard
1st low	21.39mm/s	D (Bad. Sufficient severity for long term damage)
2nd low	1.622 mm/s	A (excellent condition)
3rd low	5.602 mm/s	C (unsatisfactory for long time operation alert level)
4th low	5.799 mm/s	C (unsatisfactory for long time operation alert level)
1st high	3.56mm/s	C (unsatisfactory for long time operation alert level)
2nd high	1.472mm/s	B (Good-Acceptable for long term operation)
3rd high	6.46 mm/s	C (unsatisfactory for long time operation alert level)
4th high	6.51 mm/s	C (unsatisfactory for long time operation alert level)

Table 18 Comparison of RMS velocity value of transfer case with ISO standards

7.4 comparison of measured vibration value with calculated by mathematical modeling

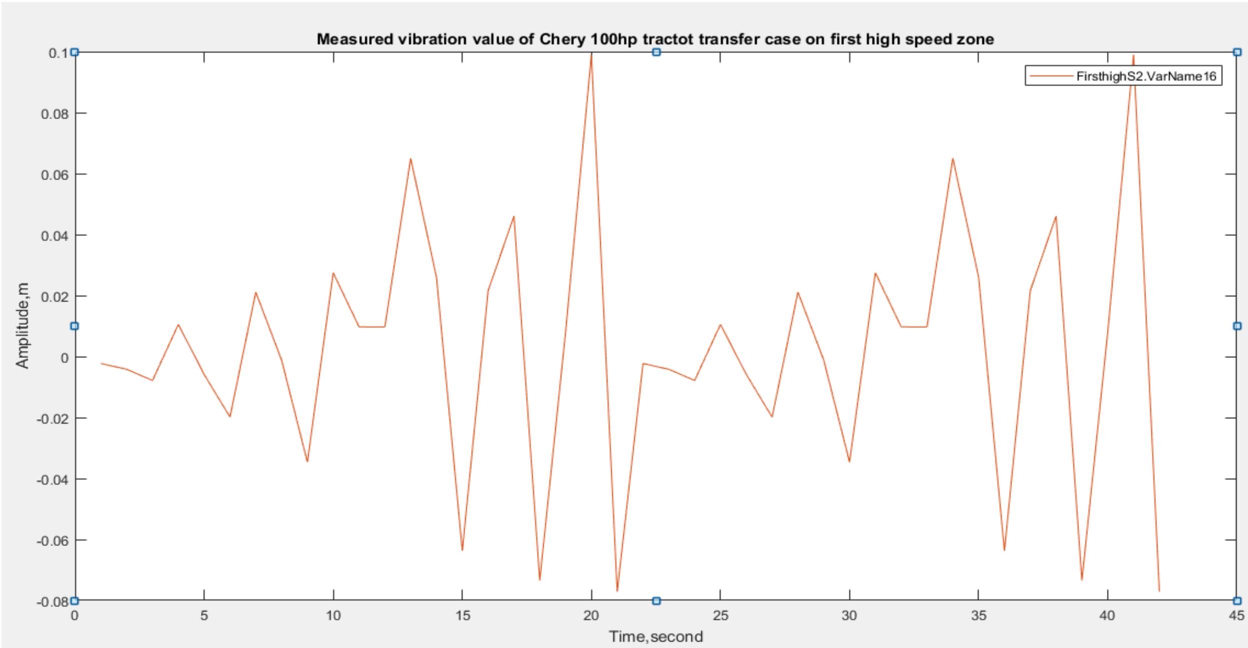

Figure 51 measured vibration value of transfer case at 1st high speed zone

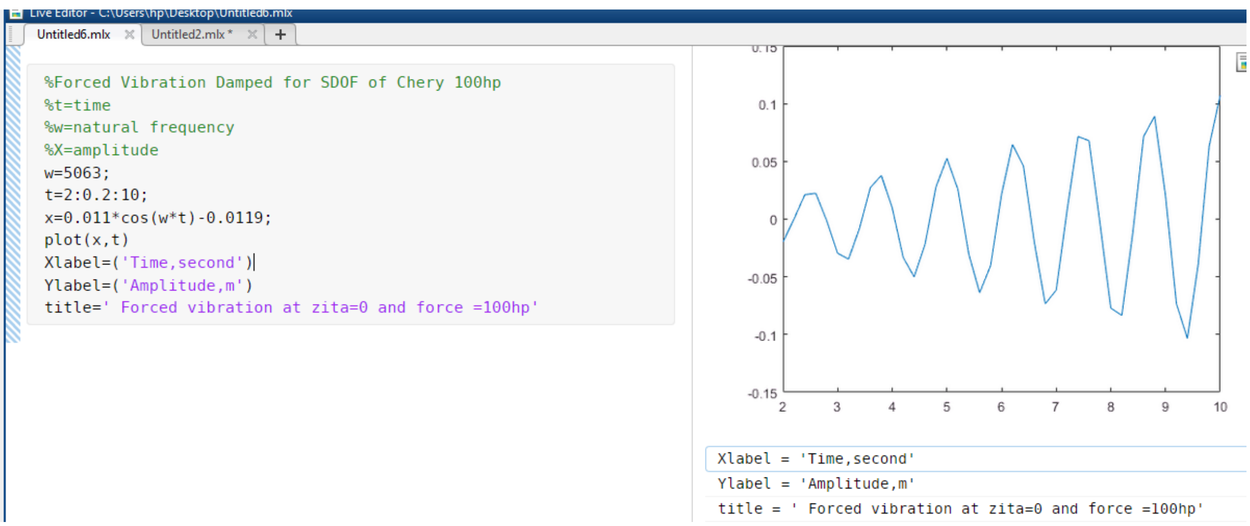

Figure 7.3 Result from mathematical modeling at the existing damping value

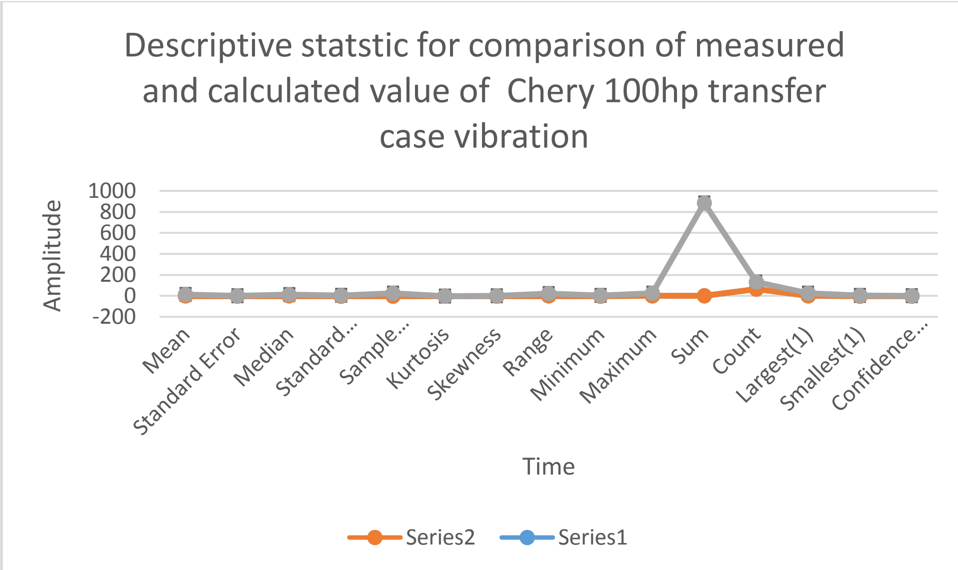

Figure 6 descriptive static for comparison of measured and calculated value of vibration

The graph shows the descriptive statistics comparison of experimental and mathematical results. Based on the results the basic statistics value mean, standard error, range and confidence are almost similar. In contrast, the maximum, sum and count differ because of the field measurement conducted with in a small period of time compared with mathematical.

7.5. Results of mathematical modeling generated from Matlab software

The graph shows the comparison experimental analysis with mathematical analysis. The plot by mathematical modeling at 100hp and calculated value of damping factor generated from Matlab software agree with the experimental analysis of first high gear speed zone plot. This implies the exiting force that cause vibration is due to the first high gear speed zone and the mathematical modelling is straightforward.

The following equation is the general solution for the system after introducing the constants A, B, natural frequency and deflection to the general solution the following graphs are plotted.

$$X = e^{-\xi\omega_n t}[Acos\sqrt{1-\xi^2}\omega_n t + Bsin(1-\xi^2\omega t] + \frac{X_{st}(\omega t-\varphi)}{\sqrt{(1-\frac{\omega^2}{\omega_n^2})^2 + 2\xi\frac{\omega}{\omega_n}^2}}$$

Taking the exiting force 95hp at calculated damping factor $\boldsymbol{\xi} = 6.56 * 10^{-16} \approx 0$

$$X = 0.011cos\ \omega t - 0.0119 + 0.0096t$$

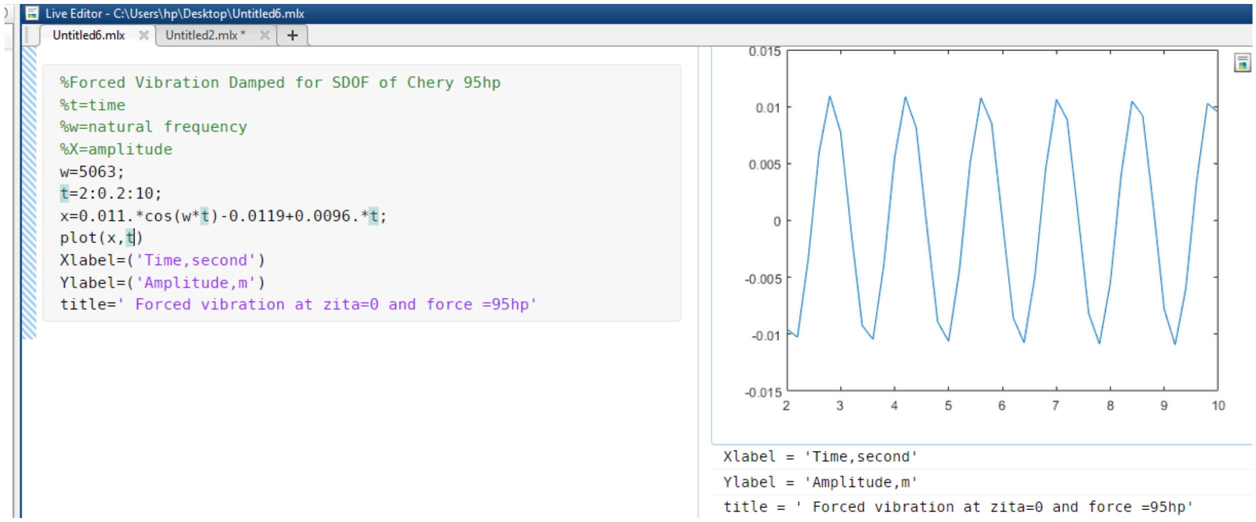

Figure 7 Forced vibration of the system at 95hp and existing damping factor

Its observed that the system is not disturbed on the available oil damper when the excitation force is 95 hp regardless of the oil type this indicate that SAE 140 oil type is not the failure problem on the 95hp excitation force**.**

Talking the exiting force 95hp at critical damping factor (ξ=1)

$$X = e^{-\omega t}(0.001cos(\omega t) - 0.0647sin(1 - \omega t] + 0.0053t + 0.0018$$

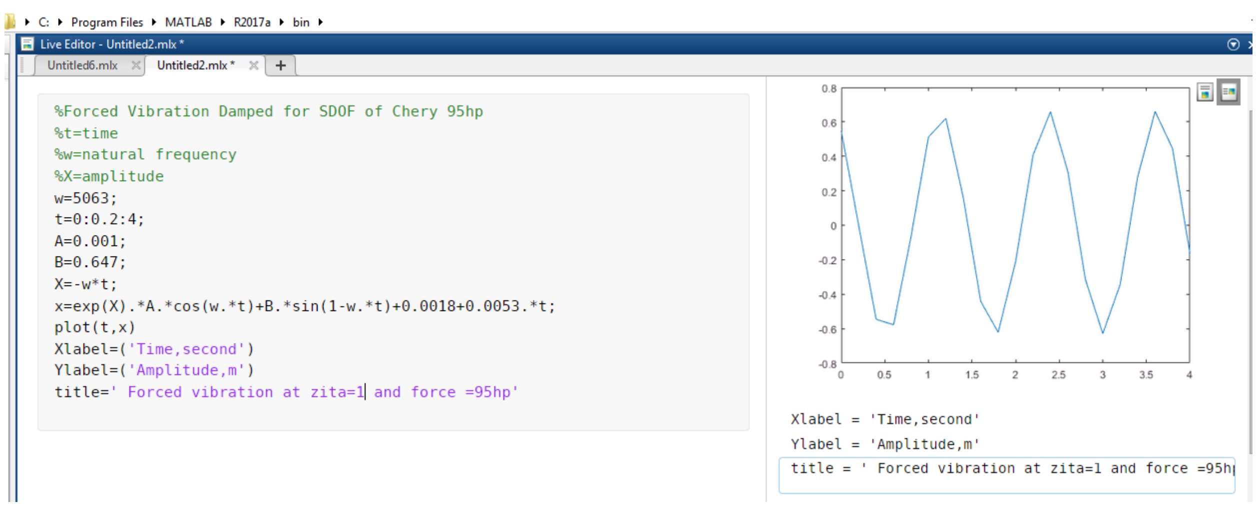

Figure 7.6 Forced vibration of the system at 95hp and 1 damping factor

The plot above indicates that the forced vibration result at the excitation torque is 95 hp but the damping factor is 1 which is critically damped. The plot specify that the system is damped and the amplitude approaches to zero at time increase which means the system is okay when the input torque is 95 hp.

Talking the exiting force or torque is 100hp at calculated damping factor $\xi = 6.56 * 10^{-16} \approx 0$

$$X = 0.011cos\ \omega t - 0.0119 + 1.9 * 10^{-13}t$$

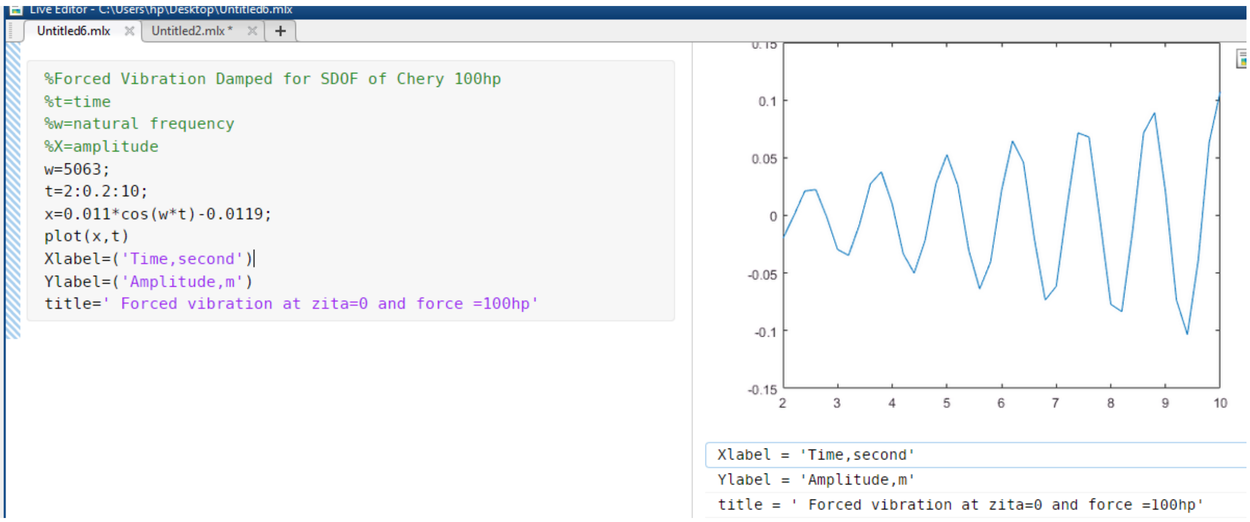

Figure 7.7 Forced vibration of the system at 100hp and existing damping factor

As it can be seen from the plot the graph looks undamped not sinusoidal and unsteady this confirms there is a clear problem on the transfer case when the amount of torque is 100hp and uses the existing damping coefficient.

Talking the exiting force 100hphp at critical damping factor (ξ=1)

$$X = e^{-2069t}(0.001cos(2069t) + 0.01sin(1 - \omega t] + 0.0058t$$

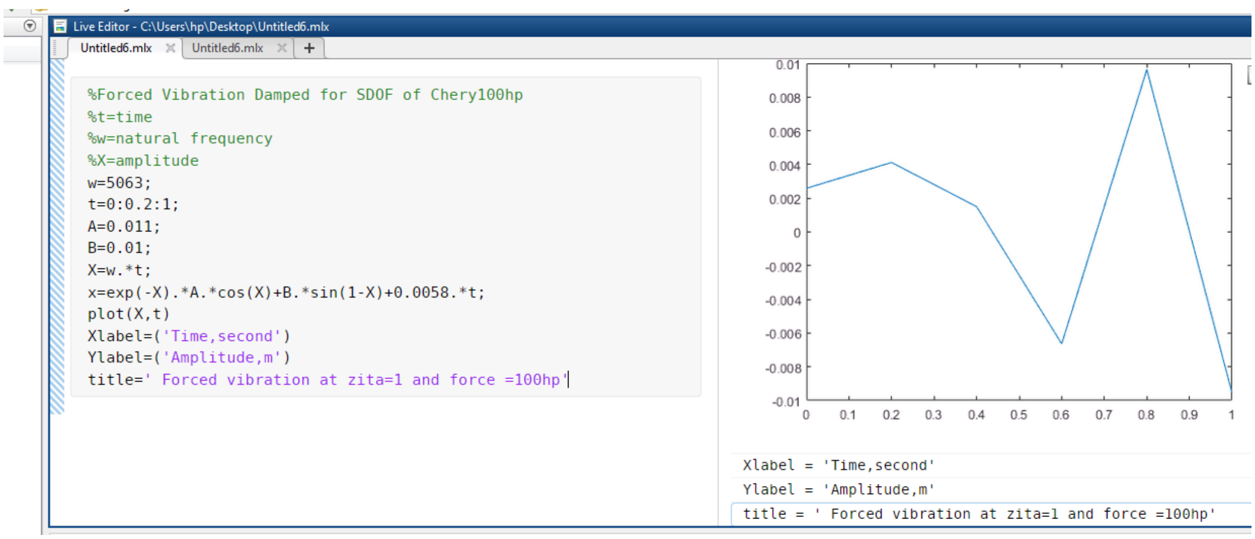

Figure 8 Forced vibration of the system at 95hp and 1 damping factor

The above graph shows the result from a mathematical model on the forced vibration analysis at the value of damping coefficient =1 or at a critically damping value. According to the plot the graph shows the amplitude becomes zero at time increase this indicates the system will be damped by changing oil type.

7.6. Result from ANSYS software.

ANSYS solver is used to calculate the modal and harmonic response evaluate the mode shape of the transfer case the simulation is performed for fixed – fixed boundary conditions. In forced vibration analysis using ANSYS the load calculated from the working power torque and speed of Chery 100hp tractor and applied.

Mode	Frequency [Hz]
1	468.26
2	562.75
3	632.68
4	828.21
5	909.69
6	1173.2
7	1237.3
8	1272.3
9	1286.9
10	1371.3

Table 19 Modal values of transfer case

Frequency response for harmonic excitation

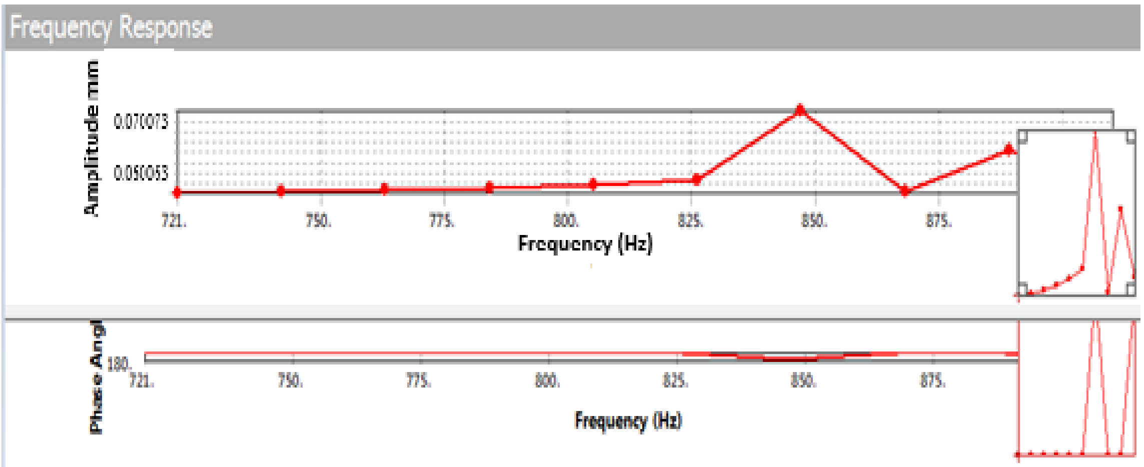

Figure 9 Critical frequency of Cherry 100hp tractor transfer case

A force of 4011N can make a deformation about 0.080084mm in between 825 Hz to 875 Hz this is the place where the resonance happen. The calculated value natural frequency is equal to 805.8Hz that falls on the critical frequency range.as a result the system harmonic analysis shows that there is a resonance that cause failure of the transfer case.

Chapter Eight

Conclusions, Recommendation and future work

The main purpose of the thesis is to make an appropriate vibration analysis of Chery 100hp tractor transfer case by using mathematical analysis, field recording and finite analysis method. The recorded value vibration was analyzed with random vibration approach. Fifteen peak amplitude values are considered to calculate the RMS velocity A methodology was developed to analyze harmonic excitation of the system by designing a transfer case by Catia V5 software and the result has been obtained using Ansys 17.2 The load applied to the system is at 100hp which is with maximum value of power and with minimum value of engine speed which is an idle rpm of tractor that is 1800rpm. According to the analysis the following conclusions are stated.

Comparison of the calculated data at 100hp with the experimental data or measured value of first gear high speed zone are plotted and implies that the acceptability of the mathematical model. That will allow to carry out further studies on the other effect of Chery 100hp tractor transfer case using computational method which has the number of advantage.

The developed mathematical model of the transfer case reflect the behavior of vibration between 95hp and 100hp.The calculated damping ratio of the existing damper of the transfer case ($\xi = 6.56 * 10^{-16} \approx 0$) value shows that the system is almost undamped. To bring the system to a critically damped situation the damping factor chosen to be $\xi = 1$ and showed appropriate change on the result of the mathematical modeling.

The critical frequency obtained from harmonic response has a range between 825 Hz to 875 Hz while the natural frequency found by mathematical modeling fall on the range bet ween the critical frequency that shows that there is a failure due to the alignment of the natural frequency with the critical frequency. The calculation made for other material type indicates that all five materials natural frequency is not in the range of critical frequency found by harmonic analysis this indicates that the materials can be used for the transfer case except steel alloy which has 805.80Hz natural frequency that can be nearer to the critical frequency fond by ANSYS.

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Natural frequency and damping factor calculated for different material type and oil specification based on the result SAE 90 with the value of damping factor $\xi=0.027$ is found to be better oil specification.

The vibration was measured by using Model SDL800 vibration meter which has basic accuracy of ± (5% + 2 digits); and Meets ISO2954 standard. The result were compared with ISO General Standard (10816-1:1995).Based on the comparison: 3rd and 4th from low speed and 1st, 3rd and 4th high speed zone are fall on zone C that is unsatisfactory for long time operation alert level. While 1st low speed found to be on zone D which is bad and sufficient severity for long term damage. Whereas the 2nd low speed zone fall on zone A shows an excellent condition. The left behind is 2nd high speed categorized at zone B that is Good-Acceptable for long term operation.

Recommendations and Future work

The major recommendations that can be made for future work in this thesis are mainly based on the continuation of the current analysis. The current analysis of the transfer case parameters can be further studied to perform better failure analysis and to achieve a more realistic representation.

Finite element analysis carried out in this thesis were solely based on the harmonic analysis but other analysis can be carried out like gear mesh excitation analysis and backlash between two driven and driving spur gears that exist in transfer case can be studied.

Based on mathematical analysis the current oil that is used for Cherry 100hp tractor transfer case SAE 140 shows almost minimum damping capability so that the oil should be changed and this can be further recommended to be SAE40 oil has better damping ability irrespective of other parameters. Also this can be left as a future work to explore on the effect of SAE40 oil type other than damping ability.

The material identified is tool steel but a Cast iron offers basic strength. It's rare to break one of these cases so the material type need to be changed in order to eliminate the failure.

Other factors that are neglected on this analysis need to be considered are the driving skill of farmers and the performance of every material inside the tractor need to be inspected in a very careful way before any distribution has made.

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