VIBRATION CONTROL FOR COMFORTABLE RIDE IN TRACTOR

M.Tech. Thesis

By

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DEPARTMENT OF MECHANICAL ENGINEERING

INDIAN INSTITUTE OF TECHNOLOGY INDORE

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VIBRATION CONTROL FOR COMFORTABLE RIDE IN TRACTOR

A THESIS

Submitted in partial fulfillment of the
requirements for the award of the degree
of
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by

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CANDIDATE'S DECLARATION

I hereby certify that the work which is being presented in the thesis entitled **VIBRATION CONTROL FOR COMFORTABLE RIDE IN TRACTOR** in the partial fulfillment of the requirements for the award of the degree of **MASTER OF TECHNOLOGY** and submitted in the **DEPARTMENT OF MECHANICAL ENGINEERING Indian Institute of Technology Indore**, is an authentic record of my own work carried out during the time period from July 2023 to May 2025 of M.Tech. Thesis submission under the supervision of Prof. Anand Parey, Professor, IIT Indore and Mr. Prashant Bardia, Staff Engineer, John Deere India Pvt Ltd - Pune

The matter presented in this thesis has not been submitted by me for the award of any other degree of this or any other institute.

11/06/2025

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

18/06/25 18/06/25

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Date: 18/06/2025

Convener, DPGC
Date: 23-6-25

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With Regards

MANJUNATHA A V

“Dedicated to my family and friends, who stood by me through every challenge”

Abstract

In modern agriculture, the use of tractors is essential for activities such as cultivation, soil preparation, and pulverization, among others. Farmers increasingly depend on agricultural machinery due to its efficiency and immediate availability. As such, ride comfort is crucial for achieving productivity targets while also ensuring the health and safety of operators. This paper discusses driver comfort and safety in off-road vehicles, along with the vibration measurement processes and practical solutions to enhance the overall experience for tractor operators.

In this report, case study related to enhanced customer comfort by reduction of steering system vibration is presented. Systematic approach is used to understand the steering performance, replication in lab, incorporation of advance testing like ODS (Operational Deflection Shape) and virtual analysis to execute the different iterations. Target location stiffening is adopted based on the modal strain response to achieve the desired results. With modified design, there is significant vibration reduction. By incorporating the ISO standards and appropriate weighting filter, Ride and comfort evaluation process is established in conjunction with subjective evaluation. Quantification of Ride and Comfort parameters for steering system are discussed in this report.

KEYWORDS: Hand Arm Vibration, Steering System, ODS Testing (Operational deflection Shape), Dynamics, Modal Testing, Modal Analysis, Engine Vibrations, Vibration parameter, Vibration limit evaluation

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

ACRONYMS

1. Introduction 1

 1.1. Vibration 2

 1.2. Degree of Freedom 5

 1.3. Components of Mechanical Engineering System 6

 1.4. Accelerometer 9

 1.5. Human Body Vibration 13

2. Literature Review 15

 2.1. Background 15

 2.2. Previous studies 17

3. Methodology 18

 3.1. Modal Analysis 18

 3.2. Operation Deflection Shape Analysis 18

 3.3. Experimental Approach 19

 3.4. Analytical Approaches 21

 3.5. Design Modification 23

 3.6. Comfort Parameter 27

4. Results and Discussion 28

5. Conclusion 31

6. References 33

LIST OF FIGURES

Figure 1.1 Types of Free Vibration 3

Figure 1.2 Periodic Vibration 4

Figure 1.3 Random Vibration 4

Figure 1.4 Spring mass system.. 5

Figure 1.5 Spring mass system with 2D of Freedom system 5

Figure 1.6 Spring mass system with 3D of Freedom system 6

Figure 1.7 Multi degree of freedom 6

Figure 1.8 Piezoelectric effect, basic calculations and Frequency response curve 9

Figure 1.9 Shear Design 10

Figure 1.10 Compression Design 10

Figure 1.11 Bender Design.. 10

Figure 1.12 Accelerometer 11

Figure 1.13 Physical Dimesions... 11

Figure 1.14 Whole Body Vibration 12

Figure 1.15 Hand Arm Vibration... 12

Figure 1.16 Potential health and safety concerns related to whole-body and hand-arm vibration... 13

Figure 1.17 Human Body vibration measurement process......................... 13

Figure 1.18 Accelerometer positions to measure HBV …………………… 14

Figure 3.1 Instrumented Steering Assembly.. 19

Figure 3.2 Baseline Vibration Response...20

Figure 3.3 Steering Wheel Impact Test.. 20

Figure 3.4 Operational Deflection Shape.. 21

Figure 3.5 Baseline Model Deformation Response 22

Figure 3.6 Baseline Modal Strain Energy Response 23

Figure 3.7 Option-1 Design Modification 23

Figure 3.8 Option-1 Modal Deformation Response...................................... 24

Figure 3.9 Option-2 Design Modification 24

Figure 3.10 Option-2 Modal Deformation Response.................................... 25

Figure 3.11 Option-3 Design Modification.. 25

Figure 3.12 Option-3 Modal Deformation Response.................................... 26

Figure 3.13 Option-3 Modal Strain Energy Response.................................. 26

Figure 3.14 Axis direction and measurement... 27

Figure 4.1 Recommended Design Configuration 28

Figure 4.2 Steering Wheel Vibration Response: Baseline vs Modified Design.. 28

LIST OF TABLES

Table 1.1 Steering Wheel Impact Test.. 10

Table 1.2 Steering Wheel Impact Test.. 11

NOMENCLATURE

τ	Time period
f	frequency
m	Mass of the wagon in Kg
k	Stiffness of the suspension system in N /m
c	Damping coefficient of the suspension system in Ns/m
Fd	Damping force in N
Fk	Spring force in N
a	Acceleration in m/s2
r.m.s	Root mean square
a_{hv}	hand arm vibration total value m/s2
a_{hwx}	frequency-weighted r.m.s. acceleration values for the x axes
a_{hwy}	frequency-weighted r.m.s. acceleration values for the y axes
a_{hwz}	frequency-weighted r.m.s. acceleration values for z axes

1 Chapter

Introduction

Comfort and safety important in the vehicle including half road vehicle. Humans can be adversely affected by vibration depending on the amplitude, excitation frequency, and exposure duration. The vibration effects can vary widely: hardly noticeable to annoying to creating health concerns. At high levels of vibration and exposure, health and safety concerns include fatigue, reduced sensitivity, abdominal or chest pain, lower back pains, or even a condition known as white finger syndrome

Based on these concerns, the European Union created directive 2002/44/EC defining standards for human body vibration. Under this directive, it is mandatory for certain products to be certified according to the International Standards Organization (ISO).

These requirements classify specific products that must meet and/or report vibration performance based on specified metrics. Based on their classification, some products cannot be sold within Europe without being certified to the standards.

The main ISO standards under this directive are ISO 2631 and ISO 5349. These standards define how to execute the test procedure including: take proper measurements, mount instrumentation sensors, apply weighting filters, and assess the exposure

The human body is more sensitive to some frequencies than others, and the weighting function takes this sensitivity into consideration. It is applied on the measured accelerometer data which results in the predicted human body response. After the data is analyzed, the standards outline corrective actions to be taken.

While meeting requirements is important, the vibration metrics outlined in the standards has led to engineering the performance of products beyond the prescribed limits. By reducing the vibration levels lower than required, companies can gain a competitive advantage over competitors and create brand identity.

The design of the suspended seat is vital for driver comfort. Research shows that driving without a load on a rough road with a parallel suspension seat requires less energy than using a non-parallel suspension.

Seats that transmit all bumps to the driver are the least comfortable, while the best seats use parallel seat guides with hydraulic dampers to reduce vibrations.

Although suspended seats can significantly decrease vertical vibrations, the levels are still too high due to poor choice of suspension parameters. Various mathematical models have been used to study how seat characteristics affect ride vibrations. Some models simplify the human body as a rigid mass supported by springs and dampers, but this is not entirely accurate since only part of the body weight affects the seat directly.

The goal of this study is to develop a new process to vibration evaluation of a tractor seat and the steering wheel that addresses some of the shortcomings of existing evaluation method. This new method will include cushion parameters and provide a better representation of the human body.

1.1 Vibration

Vibration is the motion of a particle, or a body or system of connected bodies displaced from a position of equilibrium

What is the aim of vibration analysis?

- Vibrations can lead to excessive deflections and failure on the machines and structures.
- To reduce vibration through proper design of machines and their mountings.
- To utilize profitably in several consumer and industrial applications.
- To improve the efficiency of machine and process.
- Cause rapid wear.
- Create excessive noise.
- Health and safety of the consumer
- Resonance – natural frequency of vibration of a machine/structure coincides with the frequency of the external excitation.

1.1.1 Free vibration

When no external force acts on the body, after giving it an initial displacement, then the body is said to be under free or natural vibrations.

The frequency of the free vibration is called free or natural frequency.

Three types of free vibrations

1. Longitudinal vibrations
2. Transverse vibrations
3. Torsional vibrations

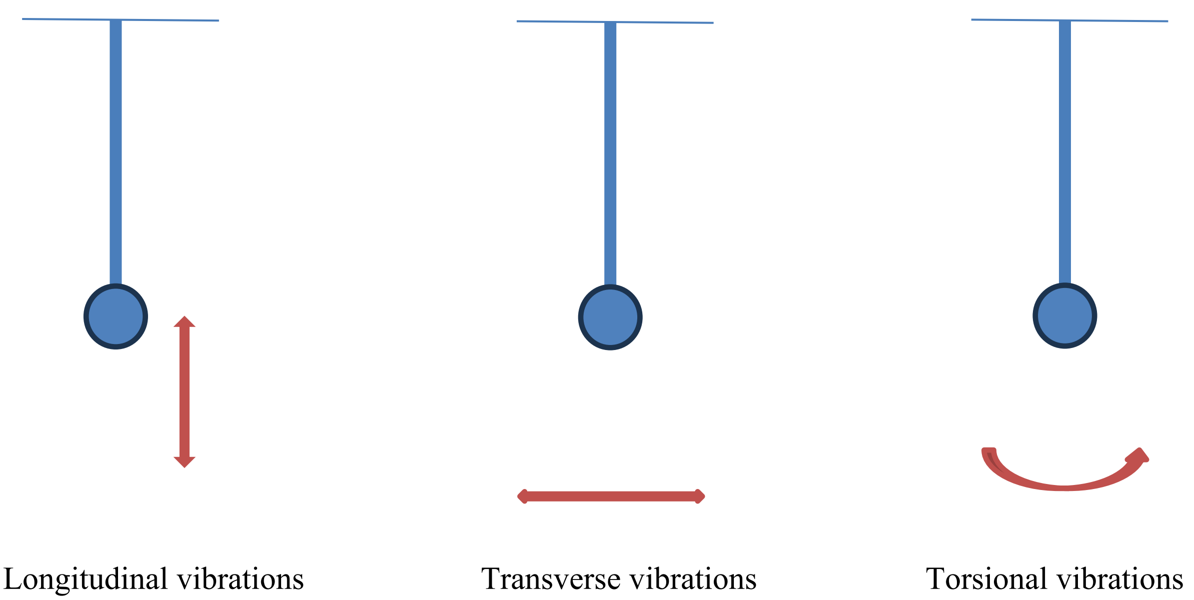


Fig 1.1 Types of Free Vibrations

1.1.2 Forced vibration

When the body vibrates under the influence of external force, then the body is said to be under forced vibrations. The external force applied to the body is a periodic disturbing force created by unbalance. The vibrations have the same frequency as the applied force. When the frequency of the external force is same as that of the natural vibrations, resonance takes place. Resonance occurs when the frequency of the external force coincides with one of the natural frequencies of the system

1.1.3 Damped vibration

When there is a reduction in amplitude over every cycle vibration, the motion is said to be damped vibration. This is due to the fact that a certain amount of energy possessed by the vibration system is always dissipated in overcome friction resistances to the motion.

Linear Vibration: When all basic components of a vibratory system, i.e. the spring, the mass and the damper behave linearly

Nonlinear Vibration: If any of the components behave nonlinearly

Periodic Vibration: If the value or magnitude of the excitation (force or motion) acting on a vibratory system is known at any given time

Random Vibration: When the value of the excitation at a given time cannot be predicted

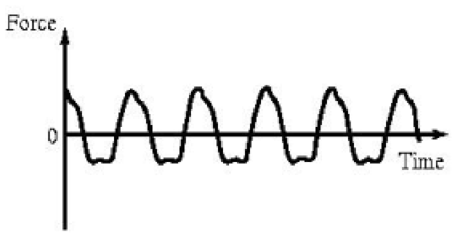


Fig.1.2 Periodic Vibration

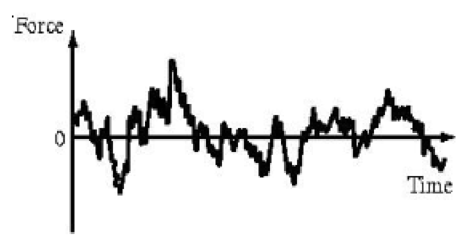


Fig.1.3 Random Vibration

Period: The time taken for one repetition. Period of vibration or time period. It is the time interval after which the motion is repeated itself. The period of vibration is usually expressed in seconds

Period $\tau = \frac{2\pi}{\omega}$ s/cycle

Frequency: it is the reciprocal of period. It is the number of cycles described in one second. The frequency is expressed in hertz

Frequency $f = \frac{1}{\tau}$ hertz

Damping: Damping is the dissipation of energy with time or distance

Viscous damping: The damping provided by fluid friction is known as viscous.

Critical damping: It is the minimum viscous damping that will allow a displaced system to return to its initial position without oscillation.

1.2 Degree of freedom

The minimum number of independent co-ordinates required to define completely the position of all parts of the system at any instance of time. How many mass will be there in a system.

1.2.1 Single degree-of-freedom systems

The number of degrees of freedom of a mechanical system is equal to the minimum number of independent co-ordinates required to define completely the positions of all parts of the system at any instance of time

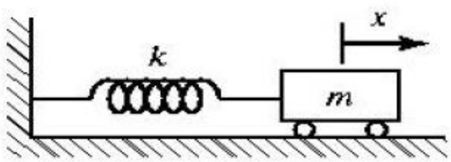


Fig. 1.4 Spring mass system

1.2.2 Two degree-of-freedom systems

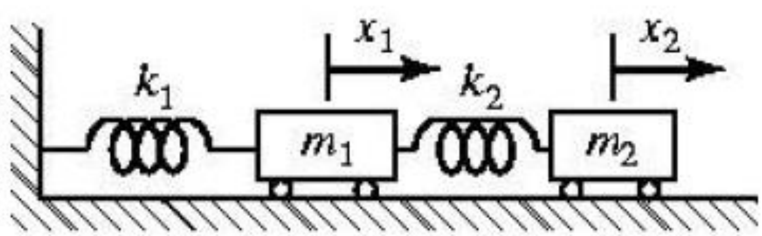


Fig. 1.5 Spring mass system with 2D of Freedom system

1.2.3 Three degree of freedom systems

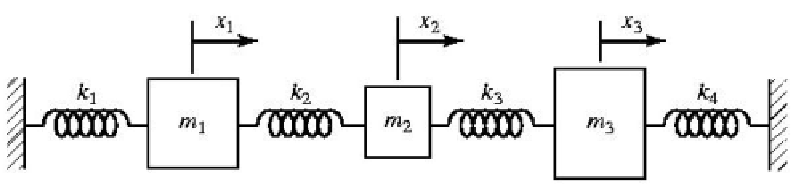


Fig. 1.6 Spring mass system with 3D of Freedom system

1.2.4 Multi-degree of freedom

Infinite number of degrees of freedom system, for which 2 or 3 co-ordinates are required to define completely the position of the system at any instance of time.

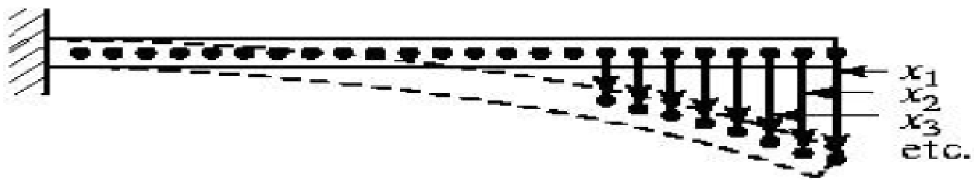
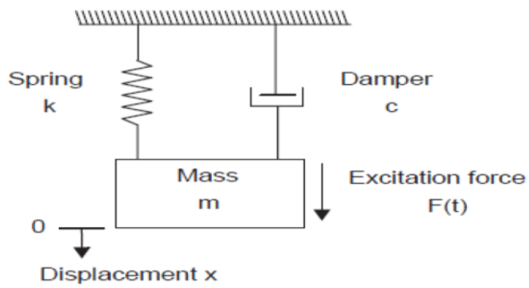


Fig. 1.7 Multi degree of freedom

1.3 Components of Mechanical Vibrating Systems

It consists of mass, spring and damper



Mass Element: The mass provides inertia force to the system.

Spring Elements: Linear spring is a type of mechanical link that is generally assumed to have negligible mass and damping.

Spring force is given by: $F = kx$

F = spring force,

k = spring stiffness or spring constant, and

x = deformation (displacement of one end with respect to the other)

Damping elements: The process of energy dissipation is referred to in the study of vibration as damping. A damper is considered to have neither mass nor elasticity. The three main forms of damping are viscous damping, Coulomb or dry-friction damping, and hysteresis damping.

The most common type of energy-dissipating element used in vibrations study is the viscous damper, which is also referred to as a dashpot.

In viscous damping, the damping force is proportional to the velocity of the body.

Coulomb or dry-friction damping occurs when sliding contact that exists between surfaces in contact are dry or have insufficient lubrication.

In this case, the damping force is constant in magnitude but opposite in direction to that of the motion. In dry friction damping energy is dissipated as heat.

1.3.1 Frequency of Free-undamped vibrations

Consider a constraint (i.e. spring) of negligible mass in an unstrained position

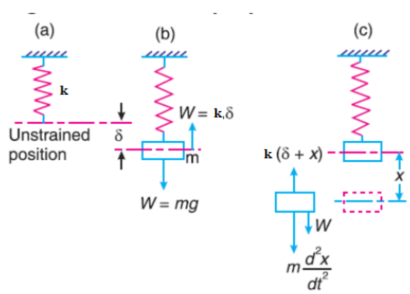
k = Stiffness of the constraint. It is the force required to produce unit displacement in the direction of vibration. It is usually expressed in N/m.

m = Mass of the body suspended from the constraint in kg,

W = Weight of the body in newtons = m.g,

δ = Static deflection of the spring in metres due to weight W newtons, and

x = Displacement given to the body by the external force, in meters



$$m\ddot{x} + kx = 0$$

$$\ddot{x} + \frac{k}{m}x = 0$$

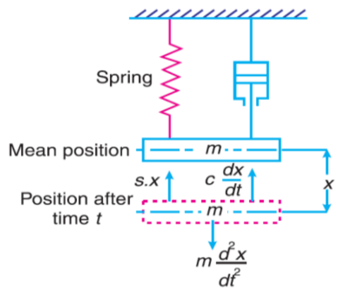
$$\ddot{x} + \omega^2 x = 0 \qquad \therefore \omega = \sqrt{\frac{k}{m}}$$

Time Period, $t_p = \frac{2\pi}{\omega} = 2\pi\sqrt{\frac{m}{k}}$

Natural Frequency $\omega_n = \frac{1}{t_p} = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$

1.3.2 Frequency of Free Damped Vibrations

The motion of a body is resisted by frictional forces, the effect of friction is referred to as damping $\boldsymbol{c}$.



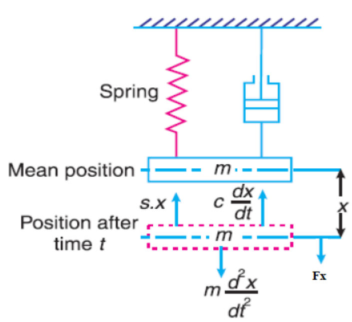
$$m\ddot{x} + c\dot{x} + kx = 0$$

1.3.3 Damping Factor or Damping Ratio

The ratio of the actual damping coefficient (c) to the critical damping coefficient (cc) is known as damping factor or damping ratio.

$$damping\ factor = \frac{c}{c_c} = \frac{c}{2m\omega_n}$$

1.3.4 Forced damped vibration system



$$m\ddot{x} + c\dot{x} + kx = F_x$$

Where $F_x = F\cos\omega t$

1.4 Accelerometer

1.4.1 Why Do We Need Accelerometers?

- Vibration and shock are present in all areas of our daily lives. They may be generated and transmitted by motors, turbines, machine-tools, bridges, towers, and even by the human body.
- While some vibrations are desirable, others may be disturbing or even destructive. Consequently, there is often a need to understand the causes of vibrations and to develop methods to measure and prevent them.
- The accelerometer serves as a link between vibrating structures and electronic measurement equipment.

1.4.2 Operation

The active element of the accelerometer is a piezoelectric material. One side of the piezoelectric material is connected to a rigid post at the sensor base. A so-called seismic mass is attached to the other side. When the accelerometer is subjected to vibration a force is generated which acts on the piezoelectric element. This force is equal to the product of the acceleration and the seismic mass. Due to the piezoelectric effect a charge output proportional to the applied force is generated. Since the seismic mass is constant the charge output signal is proportional to the acceleration of the mass. Over a wide frequency range both sensor base and seismic mass have the same acceleration magnitude hence the sensor measures the acceleration of the test object.

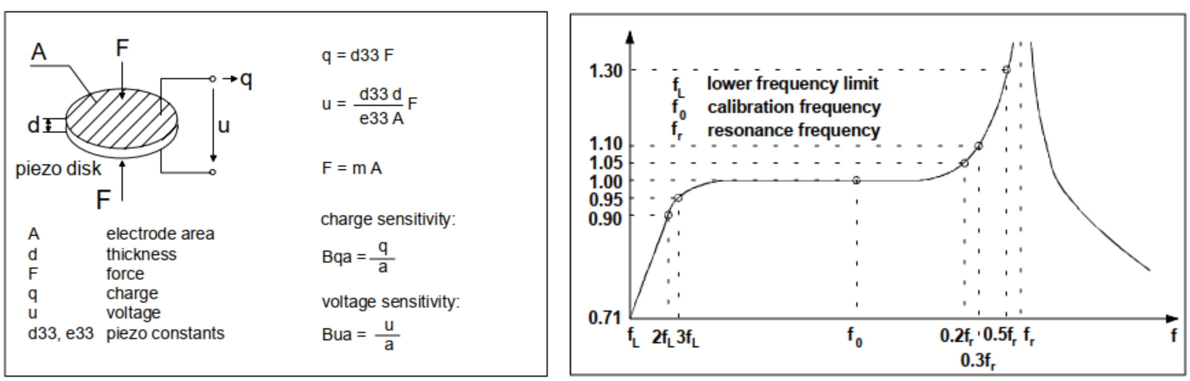


Fig. 1.8 Piezoelectric effect, basic calculations and Frequency response curve

1.4.3 Accelerometer Designs

3 mechanical construction designs:

1. Shear system
2. Compression system
3. Bender system

The reason for using different piezoelectric systems is their individual suitability for various measurement tasks and varying sensitivity to environmental influences

	Shear	Compression	Bender
Advantage	• low temperature transient sensitivity • low base strain sensitivity	• high sensitivity to mass ratio • robustness • technological advantages	• best sensitivity to mass ratio
Drawback	• lower sensitivity to mass ratio	• high temperature transient sensitivity • high base strain sensitivity	• fragile • relatively high temperature transient sensitivity

Table.1.1 Comparison of Shear, Compression and Bender type Accelerometer

Due to its better performance shear design is used in most newly developed accelerometers.

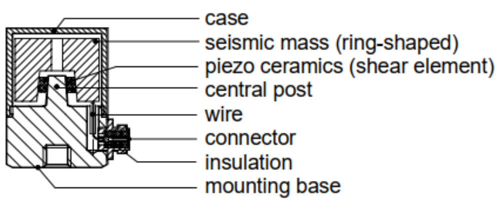


Fig.1.9 Shear Design

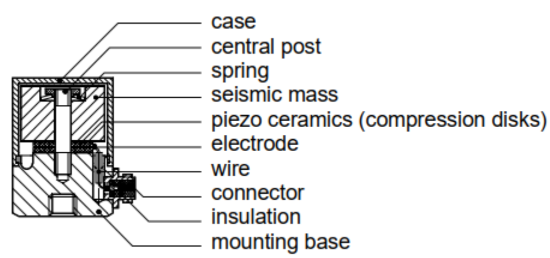


Fig.1.10 Compression Design

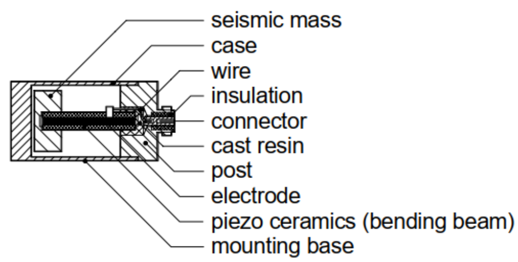


Fig.1.11 Bender Design

1.4.4 Accelerometer properties

Triaxial Delta Tron Accelerometer 4535-B TED

Application	**General Purpose**
Temperature (°C)	-60 to +125
Weight (grams)	6
TEDS	Yes
Frequency Range (Hz)	X: 0.3 to 10.0k Y: 0.3 to 10.0k Z: 0.3 to 10.0k
Mounting	Mounting Clip or adhesive or M3
Sensitivity (mV/ms–2)	1
Isolation	Yes with Insulated adaptor

Table.1.2 Overview of Accelerometer

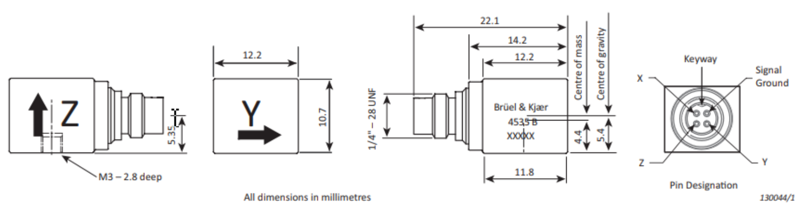


Fig.1.12 Accelerometer

Fig.1.13 Physical dimensions

1.4.5 Why Vibration calculating in g?

- Measuring vibrations in terms of acceleration (g) provides a direct correlation to the forces experienced by the human body and equipment.
- This measurement aligns with human sensitivity to vibrations, making it relevant for assessing comfort and health impacts.
- Acceleration measurements simplify the analysis of dynamic behavior in systems, allowing for easier identification of issues.
- Using acceleration as a standard measurement is consistent with industry guidelines and regulations for evaluating vibration exposure.
- This approach enables engineers and health professionals to effectively assess the impact of vibrations on machinery and individuals, ensuring safety and performance.

1.5 Human Body Vibration

- Human Body Vibration = Whole body vibration + Hand Arm Vibration
- The Human body as a dynamic system of spring/mass/dampers. Each system has a different natural frequency
- The frequency range considered is 0.5 Hz to 80 Hz for health, comfort and perception and 0.1 Hz to 0.5 Hz for motion sickness.

Fig.1.14 Whole Body Vibration

Fig.1.15 Hand Arm Vibration

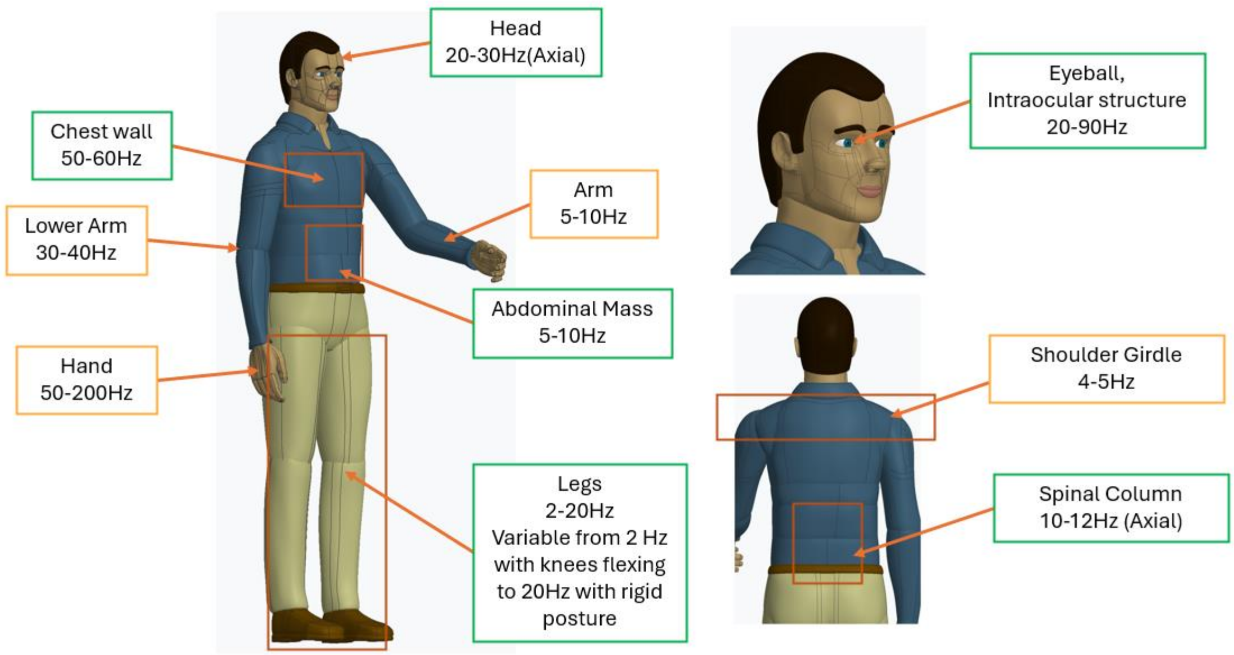


Fig. 1.16

Potential health and safety concerns related to whole-body and hand-arm vibration

1.5.1 Human Body Vibration Measurement Process

Accelerometer mount on the location where vibration need to measure. Time history data obtained from the accelerometer. From the time history, cannot take decision on vibration level. After applying Weighting, weighted time history.

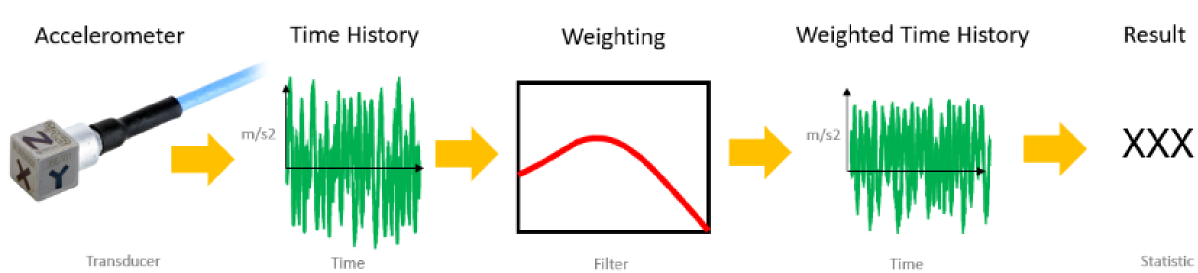


Fig. 1.17 Human Body vibration measurement process

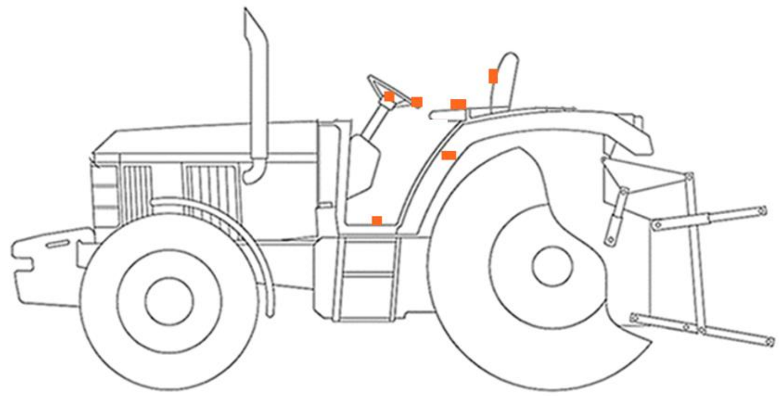


Fig. 1.18 Accelerometer positions to measure HBV

2 Chapter

Literature Review

2.1 Background study

Hand-arm vibration (HAV) has become a focal point in occupational health and safety, particularly among workers who extensively use vibrating machines and tools. The effect of continuous exposure to vibration on the human body, specifically hands and arms, can cause severe health complications, primarily Hand-Arm Vibration Syndrome (HAVS). This syndrome impacts the blood vessels, nerves, and joints, causing symptoms of numbness, tingling, and loss of grip strength. These conditions not only compromise the quality of life for those affected but also present serious challenges to workplace safety and productivity.

The importance of hand-arm vibration study is highlighted by its implications for worker safety and health, regulatory compliance, and economic considerations. Several studies have confirmed a direct link between hand-arm vibration exposure and the development of HAVS. The symptoms usually appear after extended periods of exposure, such that it is important for industries to identify and control these hazards. The World Health Organization and several occupational health agencies have issued guidelines and suggested exposure limits as a way of safeguarding workers against the harmful effects of HAV.

Based on these issues,the establishment and application of accurate measurement methods and evaluation pr ocedures have become highly necessary. Literature presents a number of methods to quantify HAV exposure, such as through the application of accelerometers, which measure the magnitude and frequency of vibrations to which workers are subjected. Additionally, it is essential to apply frequency weighing when measuring vibration exposure because human sensitivity is different across different frequency ranges. Standards like ISO 5349 give detailed guidelines in the measurement and assessment of human exposure to hand-arm vibration, so that there is uniform practice across industries.

Studies in the area of hand-arm vibration also focus on the need for ergonomic tool design and engineering controls to avoid vibration exposure. Studies indicated that tool design has a major role to play in deciding the vibration transfer to the operator. For example, equipment that is fitted with antivibration technology can help eliminate the potential for HAVS development. Moreover, the application of regular training and informing employees about vibration exposure risks can also improve workplace safety protocols.

The economic impact of hand-arm vibration is great. Health-related symptoms caused by HAV can lead to higher medical expenses for employers, days off work, and reduced productivity. Investing in preventive steps and technologies that limit vibration exposure will help shield workers from these injuries and save organizations long-term expenditures on health care and lost productivity.

The literature also contains case studies highlighting effective interventions across different industries, showing the efficacy of vibration reduction measures. Such case studies typically point to joint initiatives among employers, workers, and health professionals in establishing safer workplaces. Through these programs, companies have made considerable progress in limiting the occurrence of HAVS and enhancing the well-being of workers in general.

In sum, hand-arm vibration research is essential for the pursuit of occupational health and safety. As vibration exposure implications are understood, industries can apply countermeasures effectively to safeguard their workforce. This literature review will help add to the discussion concerning best practice on managing hand-arm vibration, supporting ongoing research and innovation in this field, which is paramount. By solving the challenges of HAV, we can encourage a healthier, safer, and more productive workplace for everyone.

2.2 Previous Studies

K.N. Dewangan et al [1] Studied hand tractors serve as a crucial mechanical power source for medium and small farms, playing a significant role in various farming activities. However, one of the primary challenges associated with their operation is the vibration transmitted to the operator through the handles of the tractor. This vibration can lead to discomfort, pain, and premature fatigue, underscoring the need for a thorough evaluation and control of hand-transmitted vibrations. The importance of this research for the agricultural sector is significant. It emphasizes the necessity for effective interventions to minimize hand-transmitted vibration in hand tractors. By enhancing operator comfort and safety, we can not only reduce early fatigue but also promote increased usage of hand tractors among farmers. This study serves as a critical step toward developing strategies that ensure the well-being of operators while maintaining productivity in agricultural practices.

Dynamic response of a structure is determined by external loads and structure dynamic properties like modal frequencies, mode shapes, damping etc. Sugita and Asai [1] demonstrated that the vibration can be reduced by increasing the resonant frequency by different ways like increasing the stiffness, reducing the mass. Ankush Shinde et al [2] used isolator concept to reduce the vibration. They used experimental and analytical method in their work. G Pandiyanayagam et al [3] established the good correlation among the modal parameters. They also demonstrated the method to find out the damping ratio. Yuntao Chen, et al [4] showed good corelation between analysis and test of the electronic module. Storck, H et al [5] presented experimental modal analysis on an exhaust system of an off-road car. In V.K.Tewaria et al [6] work, appropriate isolators were installed in specific locations to reduce the effect of vibration level.

This report focuses on the methodology to understand and predict the dynamic behavior of the system using various mechanical tests and simulations. The modal response of the model is acquired through experiments, and thus, determining its dynamic properties like mode shapes, modal frequencies. Next, the same model is analyzed by finite-element method, determining its dynamic properties. Thus, the results obtained through both the approaches are compared. Further, different designs are analyzed and tested towards vibration reduction. In the present work, the testing and analysis of tractor steering system is presented. The root cause of vibration, identification of path of vibration and effect of engine vibrations are discussed. Operational deflection shape (ODS) Testing is performed to understand the dynamic response in detail. Modal Test response of the steering system is corelated with the virtual modal analysis. Super element technique is used to reduce the solution time of FE Model. Different iterations are executed using virtual analysis. Modal frequency is shifted on higher side considering the engine operating RPM range and the excitation at different RPMs. Finally, significant vibration reduction is achieved with the recommended design and Quantification of Ride and Comfort parameters for steering system.

3 Chapter

Methodology

3.1 Modal Analysis

Modal parameters like frequency, mode shapes etc are properties of the system, and are dependent on the mass, stiffness, damping and boundary conditions of the structure. These properties can be computed analytically using Modal Analysis. Equation 1 is the governing equation of motion.

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = f(t) \qquad (1)$$

where M, C and K are mass, damping and stiffness matrices respectively. Here, inertial force is represented by $M\ddot{x}(t)$, $C\dot{x}(t)$ the damping force, $Kx(t)$ the restoring force and $f(t)$ an externally applied force. The displacement response of the system is represented by variable $x(t)$. In order to determine the inherent structural properties, there is no external force considered in modal analysis. Therefore, by substituting $f(t) = 0$, in Equation (1)

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = 0 \qquad (2)$$

Dynamic characteristics or modal properties can be obtained either through analytical or experimental techniques. These are extremely useful information while designing of almost any structure. There can be two methods to conduct modal analysis. One is the computational method and the other is the experimental method. Often, the results from experiment are used to verify computational results and to validate the computational model.

3.2 Operation Deflection Shape Analysis

An operational deflection shape (ODS) is an animation of the vibration pattern in a structure under various operating conditions. Both the amplitude and phase of vibration measurements are animated. It gives additional insight into vibration response that individual measurements alone do not. It helps to understand the directional behaviour of vibration response and then, to explore the possible design modifications effectively.

To perform an operational deflection shape analysis, three steps are required:

- Geometry: Create a geometry of the test object
- Measurement: Acquire data with consistent phasing at different points
- Analysis: Create animation utilizing the geometry and measurement data

Any type of measurement (order, frequency, time) can be animated. The key is that the phase relationship between all the different channels (corresponding to various points and directions) is preserved during the measurement.

3.3 Experimental Approach

In Physical testing, the steering wheel is instrumented with triaxial accelerometers and engine rpm sensor. A set of accelerometers is installed to capture the detailed vibration behavior in ODS Analysis (Fig.3.1). Data is recorded for engine sweep (from low idle to high idle engine RPM). A sampling rate of 16 kHz is considered. For test data analysis, frequency resolution of 1 Hz with Hanning window is applied.

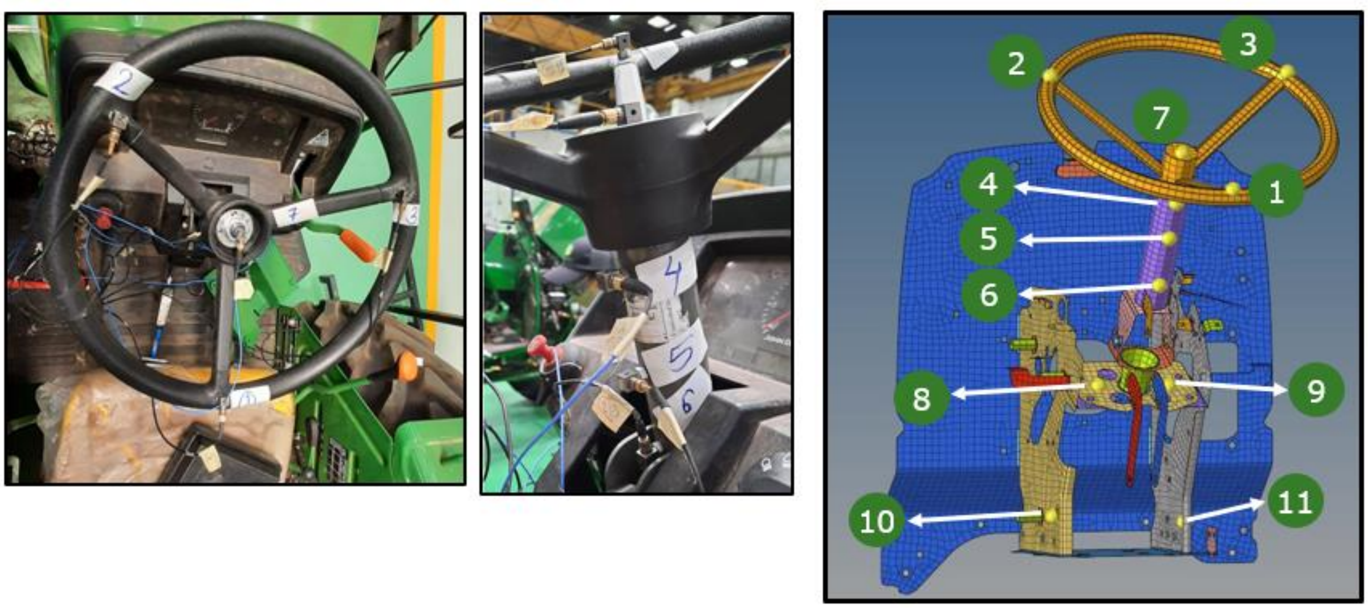


Fig.3.1 Instrumented Steering Assembly

In general, steering wheel vibrates more at its natural frequency, input force frequency and its amplitude. In the present study, higher vibration response is observed at ~45 Hz frequency with associated 1.5^{th} engine order excitation. (Fig. 3.2)

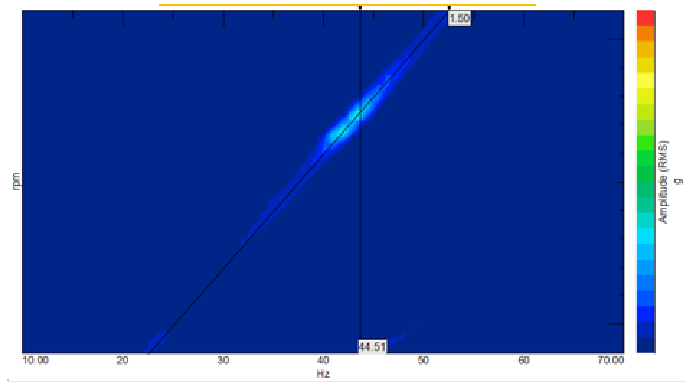


Fig.3.2 Baseline Vibration Response

Impact test is also executed to identify the modal frequency. The modal frequency is ~45 Hz (Fig. 3.3).

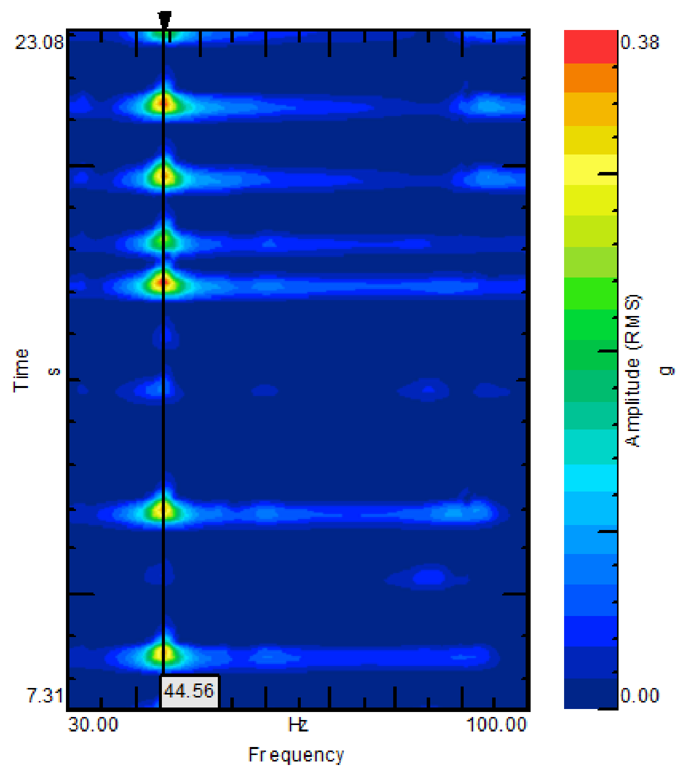


Fig. 3.3 Steering Wheel Impact Test

ODS analysis is executed to understand the vibration pattern. Steering vibration response at ~45 Hz from ODS Analysis is shown in Fig 3.4.

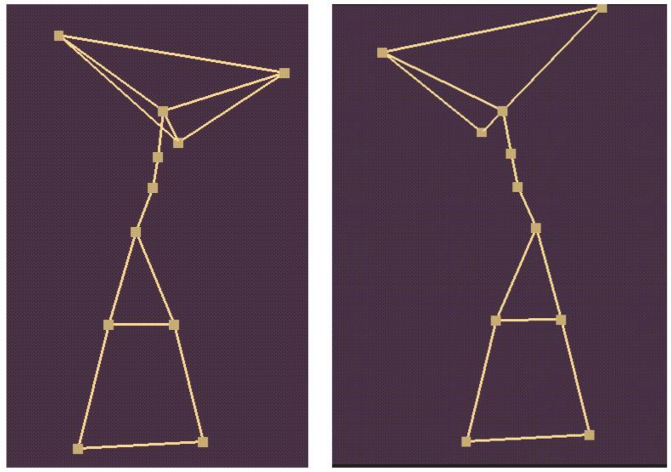


Fig.3.4 Operational deflection shape (ODS)

From impact testing, vibration response, and ODS analysis, it is clear that higher vibration is due to excitation of its natural frequency ~45 Hz because of 1.5th engine order excitation. These details are used for correlating the analysis model later.

3.4 Analytical Approach

In FEA (Finite Element Analysis) model, chassis, fenders, ROPS, platform, and other connected assembly are included (Full Model, as shown in Fig. 3.5). Mass and stiffness of the assembly are captured properly. The FEA model is solved with free-free boundary condition. Modal analysis is performed to identify the frequencies and mode shapes. Modal frequency (~48 Hz) and associated mode shape are shown in Fig. 6.

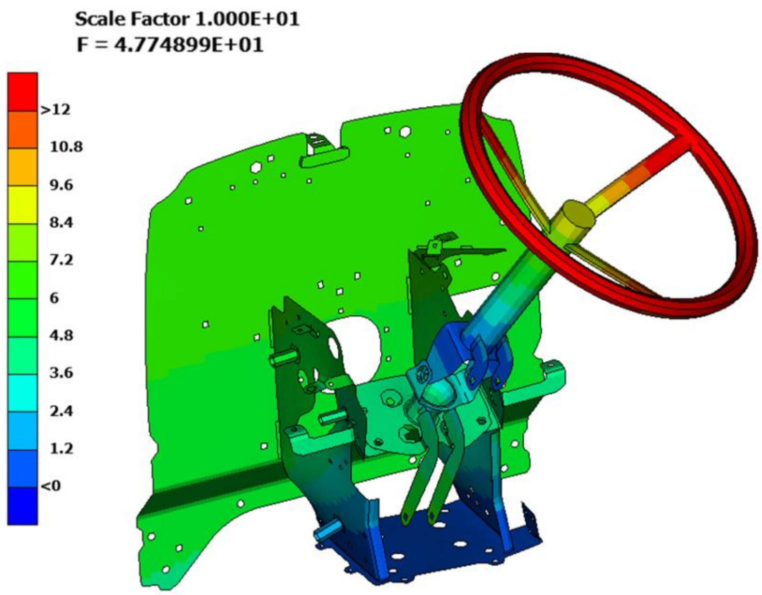


Fig.3.5 Baseline Modal Deformation Response

By comparing it with test results (Fig. 3.4), there is good correlation established between test and analysis.

The engine is a major source of vibration. For maximum engine RPM of 2300 and 1.5th Order engine excitation, frequency comes out to be 57.5 Hz. In baseline configuration, steering assembly mode with natural frequency of ~48Hz is in the engine excitation range. Mode shape from analysis and test shows that it is vibrating laterally and can be excited by rolling of engine. By all these, it can be inferred that lateral mode shape at ~48 Hz is responsible for steering higher vibration.

Considering maximum engine RPM of 2300, 1.5th order engine excitation and 10% variation, steering assembly frequency should be more than 64 Hz to avoid engine excitation frequency range. Corelated model is used for exploring different design modifications towards shifting of natural frequency on higher side, preferably out of engine excitation range.

3.5 Design Modification

Modal Strain energy response plot (as shown in Fig. 3.6) is used to identify weaker section and targeted stiffening (encircled in the Fig 3.6).

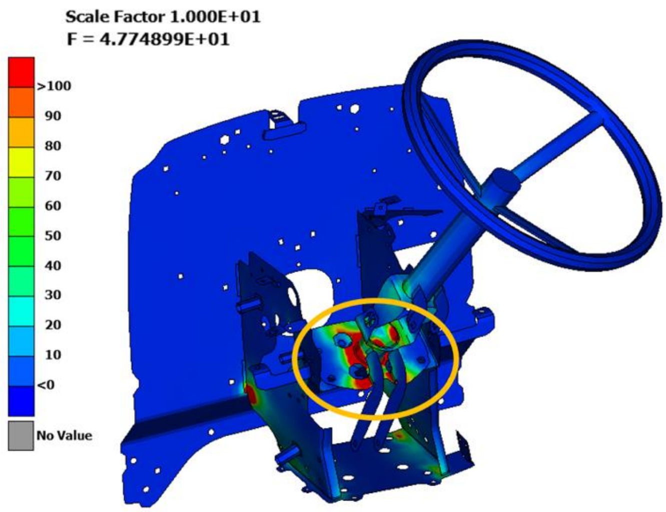


Fig.3.6 Baseline Modal Strain Energy Response

Various design options are analyzed to increase the frequency. In Option-1, plate thickness is increased by 5 mm for red plates as shown in Fig 3.7.

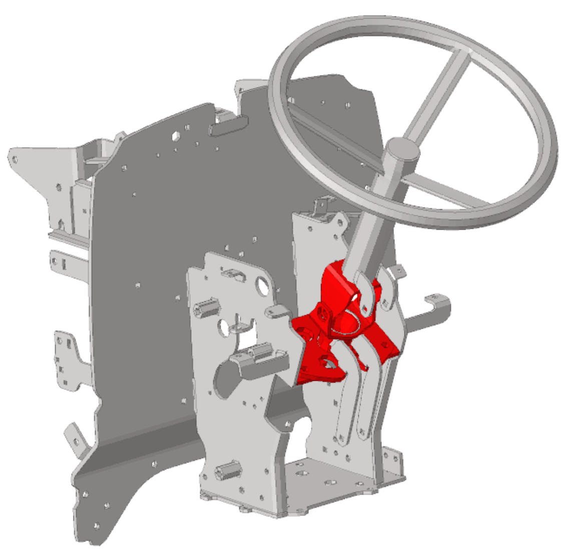


Fig.3.7 Option-1 Design Modification

With this, lateral modal frequency is shifted to 52 Hz (Fig. 3.8), is still in the engine operating range and less than the required limit. Also, there are multiple design assembly and manufacturing constraints for these design changes.

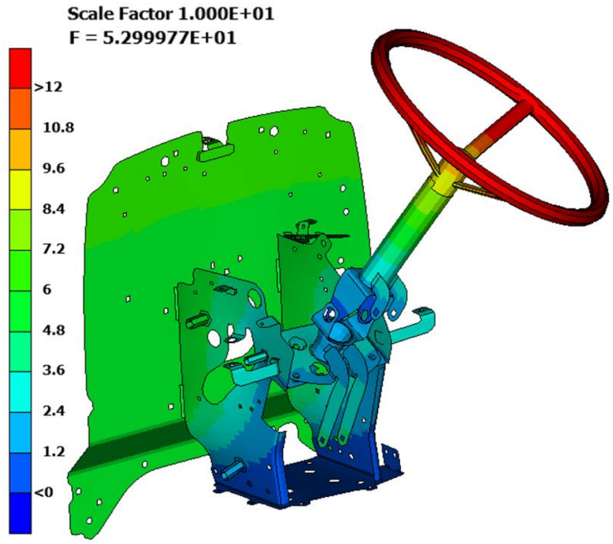


Fig.3.8 Option-1 Modal Deformation Response

In Option-2, gusset is added to cylindrical portion of steering in lateral direction which connects base plate and cylindrical portion as shown in Fig. 3.9.

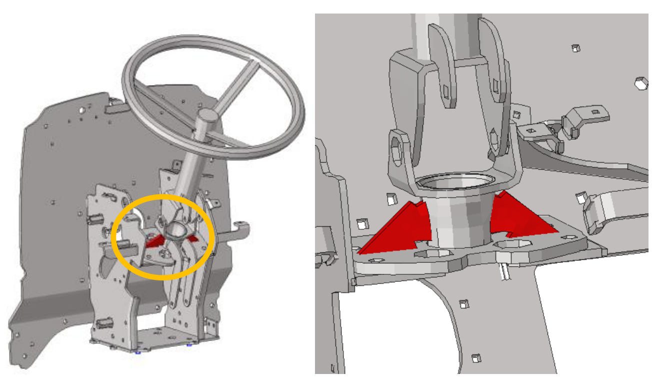


Fig.3.9 Option-2 Design Modification

With this, lateral model frequency is shifted to 58 Hz (Fig. 3.10), which again does not meet the acceptance limit.

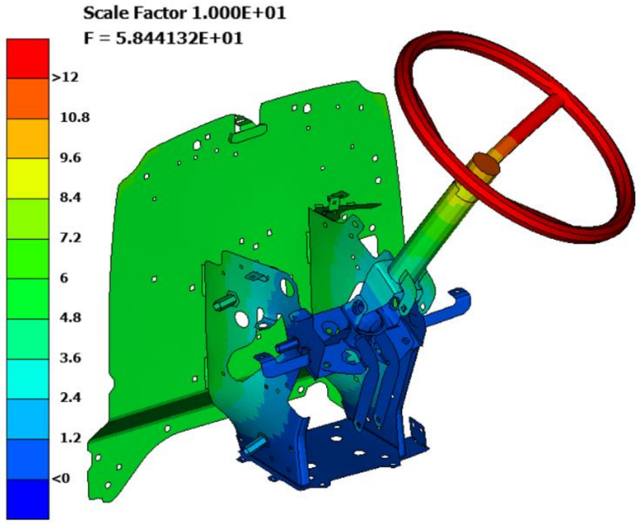


Fig.3.10 Option-2 Modal Deformation Response

With addition of small gusset, there is ~10 Hz change in the frequency. This indicates the sensitivity of the region for further targeted stiffening.

In Option-3, gusset is extended up to u-bend. Now, it connects, base plate, cylindrical portion and U-bend as shown in Fig. 3.11.

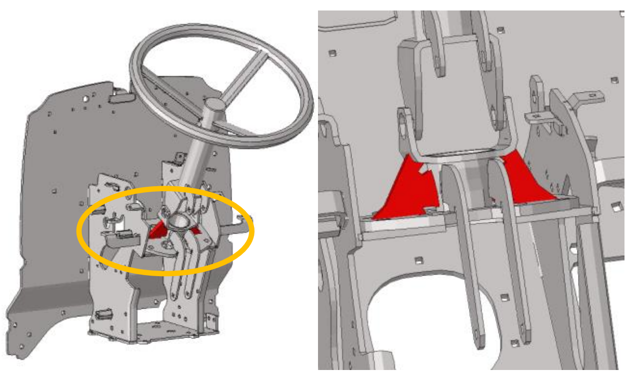


Fig.3.11 Option-3 Design Modification

With this, lateral modal frequency is shifted to 66 Hz (Fig. 3.12), meets acceptance limit. Changed behavior of mode shape is observed with reduced bending of steering at base mounting. This is also indicated by reduced strain values at cylinder location (Fig. 3.13). Considering all these, physical testing is recommended with this modification for verification.

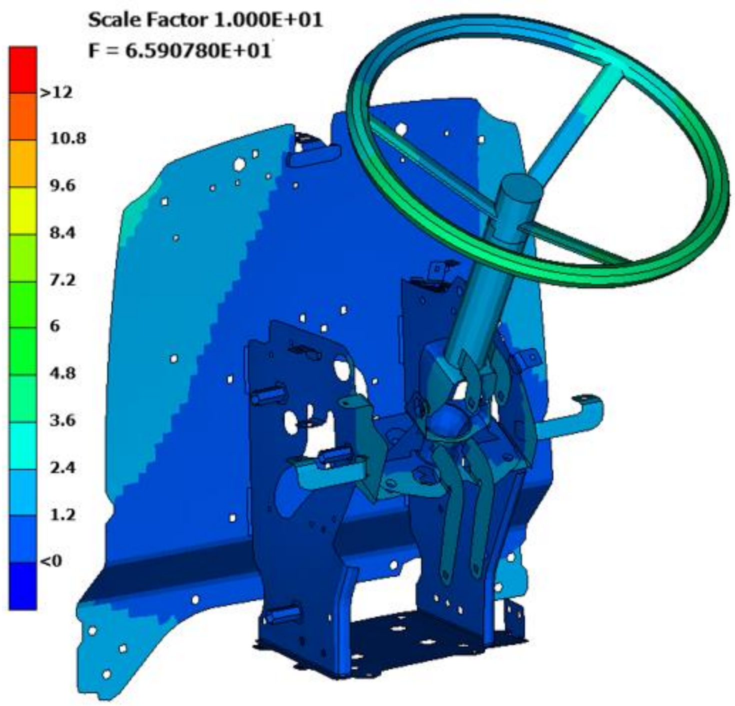


Fig.3.12 Option-3 Modal Deformation Response

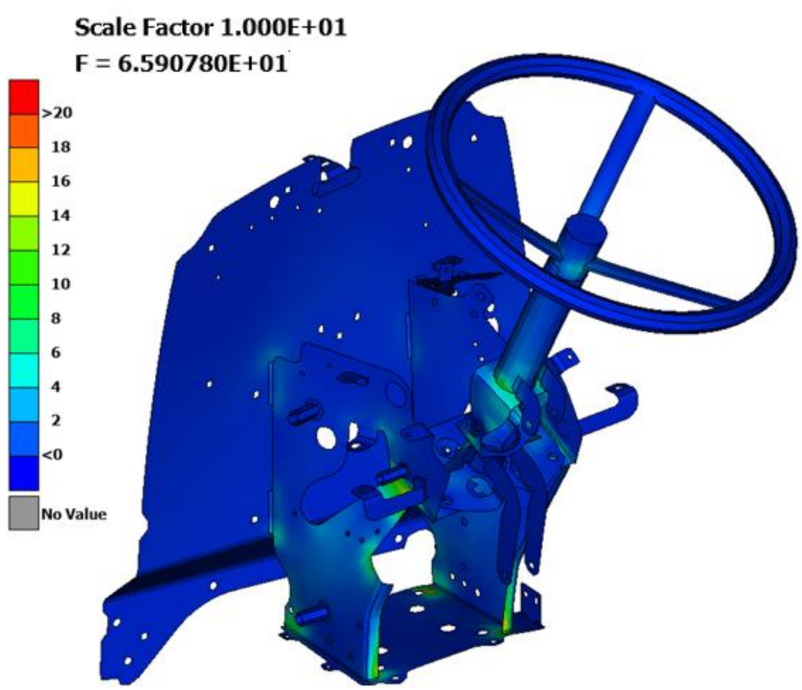


Fig.3.13 Option-3 Modal Strain Energy Response

3.6 Comfort Parameter

Vibration Total Value:

$$a_{hv} = \sqrt{a_{hwx}^2 + a_{hwy}^2 + a_{hwz}^2}$$

Where a_{hv} is hand arm vibration total value

a_{hwx}, a_{hwy} , a_{hwz} are frequency-weighted r.m.s. acceleration values for the x, y and z axes

By evaluating the Vibration Total Value for the actual tractor under worst-case conditions both before and after improvements, assessment of the impact of changes made.

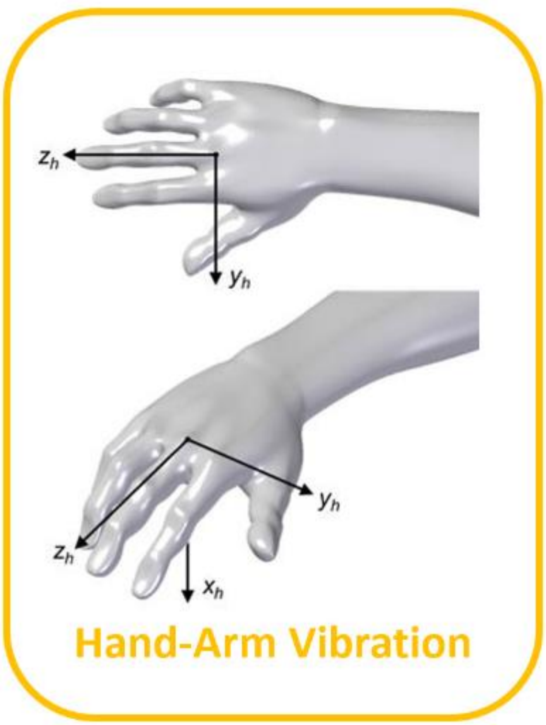


Fig.3.14 Axis direction and measurement

4 Chapter

Result And Discussion

Manufacturing feasibility of incorporating the recommended modification is checked and confirmed. Modified Design Model and Physical Assembly are shown in Fig. 4.1. For physical testing, steering wheel is again instrumented with same set of triaxial accelerometers and engine rpm sensor. Data is recorded for engine sweep (from low idle to high idle engine RPM).

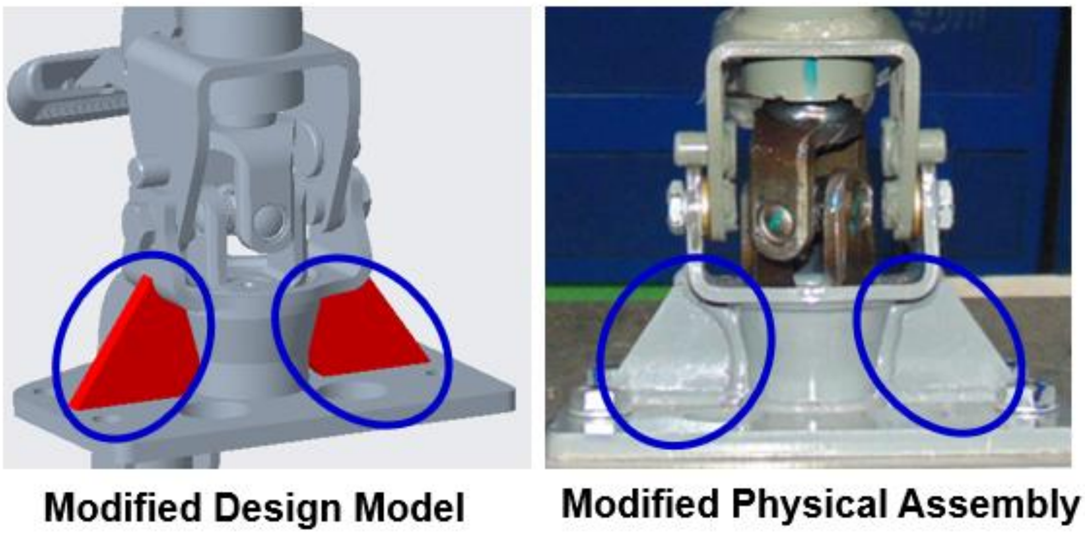


Fig.4.1 Recommended Design Configuration

The results of vibration level for baseline and modified assembly are shown in Fig.4.2. With modified design, there is significant vibration reduction. Same is confirmed with subjective evaluation also. Based on all these, recommended design modification is implemented. This resulted in enhanced customer comfort.

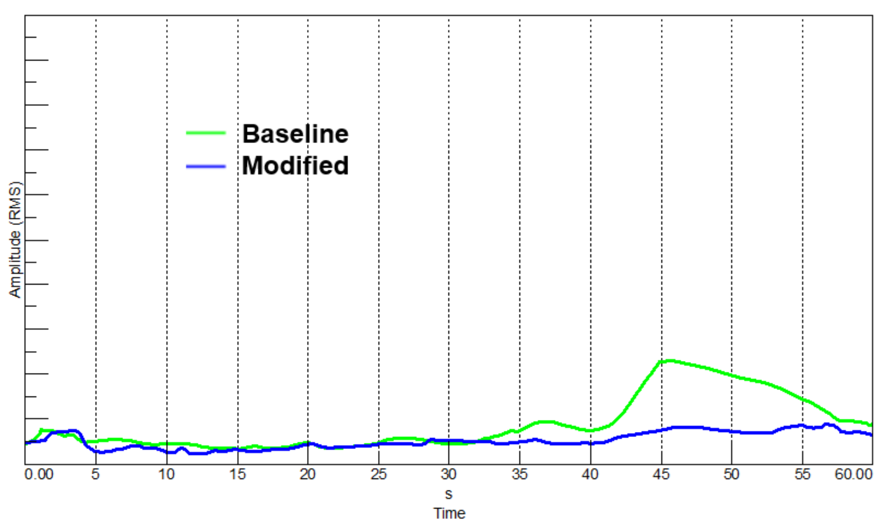


Fig.4.2 Steering Wheel Vibration Response: Baseline vs Modified Design

The $a_{\mathrm{h}v}$ value of the tractor before improvements considered as the baseline, 100%. After the improvements, the $a_{\mathrm{h}v}$ value indicating a reduction of 46% compared to the baseline state. Good correlation achieved with customer feedback and ride comfort

5 Chapter

Conclusion

In pursuit of enhanced vehicle performance and driver comfort, the incorporation of analysis-driven design practices has been a determinant measure to minimize vibrations in steering systems. In this paper, it is described how systematic testing and evaluation have made substantial breakthroughs in steering performance, leading to a better user experience.

The process started with a clear comprehension of steering dynamics.
A controlled method was used to understand the dynamics of steering performance, with real-world conditions replicated in the laboratory environment. New testing techniques, like Operational Deflection Shape (ODS) testing and virtual analysis, were integrated to offer a complete perspective of steering system behavior under different conditions. ODS testing permitted visualization of the dynamic response of the steering parts, allowing engineers to pinpoint where the vibrations were most intense.

One of the most important parts of this analysis was to correlate modal frequencies and shapes based on both analytical models and ODS test data. Through knowledge of the relationship between these modal properties and physical behavior of the steering system, engineers could identify specific areas for optimisation. This correlation provided information on how the steering wheel and column reacted to vibrational forces, giving a foundation to stiffening strategies to eliminate unwanted vibration.

Targeted stiffening in accordance with the modal strain response of the column and steering wheel was enacted as an analysis recommendation. By stiffening the structure where the vibrations were worst, the design team was able to reduce significantly the transmission of vibrations to the operator. Enactment of this improved design yielded a significant reduction of the hand-arm vibration total value by 46% compared to baseline tests.

Apart from the technical measurements, ride comfort was also assessed alongside customer opinion. This was a key step in determining the degree to which the modifications in the steering system impacted levels of user satisfaction. By relating the findings of the vibration tests against the views of the operators, the design team could determine the impact of their changes and confirm that the adjustments they made not only minimized vibrations but also enhanced the driving experience.

In summary, the use of an analysis-driven design methodology has effectively resulted in lower steering system vibrations by combining systematic testing, sophisticated analytical techniques, and controlled design changes. By addressing the technical issues and concerns along with user feedback, this research emphasizes the significance of taking an integrated approach to engineering problems, ultimately optimizing operator comfort and vehicle performance. The dramatic decrease in hand-arm vibration levels highlights the worth of coupling analytical findings with functional design solutions, opening the door to subsequent enhancements in automobile ergonomics and safety.

APPENDIX-A

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