Vibration Analysis of MINIDOR Force Bajaj Engine Mount Using FEA

JatemaMegersa Bane

A Thesis Submitted to the Department of Mechanical Engineering,
College of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfillment of the Requirement for the Degree of Master's in Automotive Engineering

School of Postgraduate Studies
Adama Science and Technology University

May, 2025
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I hereby declare that this Master's Thesis entitled "Vibration Analysis of MINIDOR Force Bajaj Engine Mount Using FEA" is my original work. That is, it has not been submitted for the award of any academic degree, diploma, or certificate in any other university. All sources of materials that were used for this thesis have been duly acknowledged through citation.

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We, the undersigned, members of the Board of Examiners of the thesis by Jatema Megersa Bane, have read and evaluated the thesis entitled "Vibration Analysis of MINIDOR Force Bajaj Engine Mount Using FEA" and examined the candidate during open defense. This is, therefore, to certify that the thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Automotive Engineering.

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Abstract

*Due to low cost and simplicity, many Ethiopian people use MINIDOR Force Bajaj and others Bajaj for transportation. However, these vehicles experience vibrations due to road conditions, engine dynamics, and their overall condition. This thesis investigates the vibration characteristics of the Minidor Force Bajaj engine mount through a combination of Finite Element Analysis (FEA) and experimental testing. The study explores the relationship between engine speed and vibration levels, analyzing data collected across a range of 1200 to 3000 RPM, corresponding to frequencies between 20 Hz and 50 Hz. The findings of this thesis reveal that resonance point is found at 34 Hz, which is aligned with the fundamental frequency of the engine mount system. The results indicate that while the engine mount effectively reduces vibrations within a specific operational range, operating at high RPM leads to exceeding permissible vibration levels outlined in ISO 10816-6 standards. Moreover, the FEA model closely correlates with experimental results, in which the maximum acceleration on th passenger is 0.95m/s2 and maximum acceleration for FEA is 2*10^-004 at frequency of 50Hz.Thus,the maximum permissible acceleration for whole human body from ISO 10816-6 standards is 0.8m/s2 though slight variation highlights the need for refined simulation parameters to account for real-world factors such as damping and material imperfections. It is highly recommended to limit the operational RPM, optimizing material and design of engine mount, and mitigating resonance effects for improving the MINIDOR Force Bajaj vibration.*

Keywords: Vibration Analysis, MINDOR Bajaj, Engine Mount, FEA, Vibration Standard

Contents

List of Figure

List of Table

CHAPTER ONE

INTRODUCTION

1.1 Background of Study

The transportation industry plays a significant role in the economic progress of any nations. Fuel efficiency, stability, collision safety, vibration problems in conventional combustion engines, pollution management, raw material costs, and consumer safety are just a few of the engineering hurdles the car industry has faced recently. Consequently, a wide range of new vehicle models have emerged in the market, aiming to enhance strength, safety, comfort and fuel efficiency (Nishinaka & Matsuoka, 2019).

Among these innovations, MINIDOR Force Bajaj vehicles have become more well-liked for both individual and public transportation, particularly in poor countries like Ethiopia. These vehicles, sometimes referred to as auto rickshaws or Bajaj, are frequently used because of their versatility in a variety of driving situations. In nations like Ethiopia and India, they are very common. Like motorcycles and scooters, the MINIDOR Force Bajaj has a single-wheeled front steering system, and the two rear wheels, which have a differential and suspension system like those in vehicles, serve as the drive wheels. Their road stability and maneuverability are improved by this design. (Thakur & Deep, 2012).

Techniques for reducing vibration are crucial for improving passenger happiness and comfort. Automotive engineers can drastically reduce interior vibration levels by identifying and reducing the main sources of vibration. The need for accurate vibration analysis technology has grown since the introduction of conventional combustion engines and changes to vehicle architectures. The entire user experience, vehicle performance, and passenger comfort can all be negatively impacted by excessive vibrations from conventional combustion engines. (Keshavarz et al., 2011).

Virtual analysis of engine mount stiffness and stopper gap tuning is a method used to improve vehicle vibration performance. Vibration directly impacts the overall comfort and refinement experienced in a vehicle, making the optimization of engine mounts crucial for minimizing the transfer of vibrations to the vehicle's body and passenger compartment(Kamble & Bhalerao, 2016).

The application of FEA studies in the automotive industry enables engineers to validate their designs early in the powertrain development cycle. This approach improves the accuracy of correlations between FEA and test results, facilitating the replacement of physical hardware prototypes with analytical ones. Powertrain design faces numerous challenges, with vibration being a significant concern. Design engineers must evaluate the vibration properties of engine mounts to tackle issues such as weight reduction, vibration control, and mitigation (Kagnici, 2017).

A large segment of the Indian population depends on MINIDOR Force Bajaj for transportation due to challenging economic conditions. These vehicles are subject to vibrations caused by uneven road surfaces, soil conditions, engine dynamics, and overall vehicle wear and tear. Such vibrations adversely affect the health and comfort of both drivers and passengers. The oscillations are transmitted through the body's tissues, organs, and systems, impacting internal structures before being attenuated and dissipated. Among the sources of vibration, engine mounting is a significant issue(Kharat & Dhande, 2015).The repetitive vibrations generated by the engine can lead to the enlargement of holes on the engine mounting bracket, eventually causing the bracket to fail (Kharat & Dhande, 2015).

The engine mount plays a crucial role in enhancing vehicle ride comfort by minimizing vibrations. The primary function of an engine mounting bracket is to ensure the proper balance of the power pack (engine and transmission) on the vehicle chassis, facilitating effective motion control and isolation (Naghate & Patil, 2015).

1.2. Statement of the problem

In Ethiopia, challenging economic conditions have led a significant portion of the population to rely on MINIDOR Force Bajaj as a primary mode of transportation. These vehicles are subjected to various sources of vibration, including rough road surfaces, soil conditions, engine dynamics, and the vehicle's overall state. These vibrations significantly affect the comfort and well-being of both drivers and passengers. The engine of the MINIDOR Force Bajaj consists of reciprocating components that produce unbalanced forces, leading to vibrations that are transmitted to the vehicle's support structures.

The vibrations produced by the MINIDOR Force Bajaj result in mechanical damage, physiological effects, and subjective discomfort in humans. Human engineering involves examining the various impacts of vibrations on different parts of the human body. The repetitive vibrations from the engine can cause the holes in the engine mounting bracket to enlarge over time, ultimately leading to bracket failure (Kharat & Dhande, 2015).Thus, this thesis analyzed the vibration of the MINIDOR Force Bajaj engine mount using FEA and experimental method.

1.3. Objectives

1.3.1. General Objectives

The general objective of this thesis is to experimentally measure the vibration and perform FEA analysis of MINIDOR Force Bajaj engine mounts and compare the result with vibration standards.

1.3.2. Specific Objectives

The specific objectives of the thesis include:

- ❖ To model a MINIDOR Force Bajaj engine mount using CATIA software with existing dimensions.
- ❖ To analyze the vibration of the MINIDOR Force Bajaj engine mounts under different RPM using ANSYS.
- ❖ To measure the vibration of Minidor Force Bajaj engine mount with different RPM.

1.4. Significance of the study

The significance of the study lies in its potential to improve the vibration of MINIDOR Force Bajaj. By employing finite element analysis (FEA) to analyze for the engine mounts.

The study can contribute to enhancing the overall comfort, refinement, and driving experience of these vehicles. The results of the study can provide valuable insights for engineers and manufacturers in developing effective strategies to minimize vibrations transmitted to the vehicle body and passenger compartment, thereby improving the vibration characteristics of MINIDOR Force Bajaj.

1.5. Scope of the study

The scope of the study FEA Analysis of vibration for MINIDOR Force Bajaj Engine Mounts is focused on evaluating vibration analyzing, specifically for the engine mounts of MINIDOR Force Bajaj. The study aims to analyze the dynamic behavior of the engine mounts, assess the effectiveness of different vibration analyzing techniques, optimize design parameters, and validate the results through finite element analysis (FEA) and experimental measurements. The study's scope is limited to the engine mounts and their impact on analyzing vibrations transmitted to the vehicle body and passenger compartment. The study explores different techniques aimed at analyzing vibration in MINIDOR Force Bajaj. These techniques may include adjustments to mount stiffness, damping characteristics, and mounting locations.

1.6. Limitation

This research examines the behavior of the engine mount Minidor Force Bajaj and suggests related activities for enhancing vibration performance, however its scope is limited to improving undesirable vibration caused in the engine mount.

1.7. Organization of the Thesis

This thesis is structured into five chapters. Chapter One introduces the problem investigated, outlining the background, objectives, scope, significance, limitation and overall organization of the thesis. Chapter two presents a review of related literature, exploring previous research studies, including the materials and methodologies employed by other researchers. Chapter Three focuses on the materials and methodology of the vibration analysis of Minidor Force Bajaj Engine Mount and Vibration measurement. Chapter Four deals with the explanation of the experimental and FEA results, and providing discussions based on findings. Finally, Chapter Five concludes the thesis outcomes by presenting key findings from the analysis and experimental results. Moreover, the last chapter presents a recommendation for future work.

CHAPTER TWO

LITERATURE REVIEW

2.1. Introduction

Engine mounts are crucial for reducing vibrations. In internal combustion engines, two primary sources of dynamic disturbances arise: (1) the firing pulse generated by fuel combustion within the cylinder and (2) the inertia forces and torque caused by the rotation and reciprocation of engine components. Research indicates that utilizing engine mounts is one of the most effective methods for dampening these vibrations and transmitting forces between the engine and vehicle structure. A study by Gavade et al. highlights that structural optimization in engine mount design significantly contributes to vibration reduction, emphasizing the importance of addressing these disturbances for improved vehicle performance The firing pulse creates torque along an axis parallel to the crankshaft, while the inertia forces act both along the piston axis and perpendicular to the crankshaft and piston axis (Gavade, S. V. et.al. 2022).

The finite element method (FEM) is instrumental in conducting modal and static structural analysis of engine mount designs. For vibrational analysis using FEA, a 3D model of the mount was created in Solid-Edge V19 software, and Ansys Workbench was utilized to evaluate its natural frequencies and total deformation. FEM helps determine the natural frequencies and vibration modes of an engine mount (Ramachandran, T., 2012).

2.2. Overview of Vibration Analysis and FEA

Modal analysis by FEA detects the vibrational properties of a structure or explicit component through its natural frequencies and mode shapes. Modeling-based analysis plays a crucial role in the development and examination of models, as well as in forecasting various types of vibrations generated by engines. It is also valuable for predicting nodal displacements, stress levels, eigen values in terms of frequencies, and mode shapes. This review primarily highlights finite element analysis (FEA), which is widely documented as a key technique for investigating vibrations educed by engines. Recent studies utilizing FEA simulations have been conducted with tools such as ANSYS, Hyper Mesh, ABAQUS, COMSOL, and Hyper Mesh Opti Struct(H. V. S. Kumar et al., 2022).

Moreover, Ramachandran et al., (2012)stated the analysis of the vibration by FEA as mount was composed of two mild steel plates with a rubber isolator positioned between them. The modeling and assembly of the rubber mount components are carried out using I-DEAS software, while finite element structural analysis and numerical investigations are performed through ABAQUS. The deformations of the rubber mount obtained from the FE analysis are utilized in Minitab to create a mathematical model, with deformations analyzed in relation to various process parameters.

2.3. Vibration Analysis of Engine Mount

The optimization process plays a crucial role in identifying the most suitable material for the engine mounting bracket. Factors such as strength, stiffness, weight, and cost are considered to achieve an optimal design. Finite Element Analysis (FEA) is a powerful tool that enables engineers to simulate and evaluate the structural behavior of the bracket under various loading conditions, providing valuable insights into its performance. A primary objective of Kamble & Bhalerao, (2016) is to perform a comprehensive structural analysis of the engine mounting bracket. This analysis involves assessing the stress distribution, deformation, and failure modes of the bracket under different operating conditions. By employing FEA and optimization techniques, the authors sought to enhance the bracket's performance while ensuring it meets specific requirements. Modal analysis was conducted to examine the natural frequencies and mode shapes of the engine mounting bracket. This analysis allows engineers to understand the bracket's dynamic behavior and potential resonance issues that may arise during operation.

Karagöz & Tuncay, (2020) address the critical role of engine mounts in mitigating vibrations, a key factor influencing vehicle performance, comfort, and passenger health. With trends in lightweight vehicle designs and more powerful engines leading to increased vibration challenges, the researchers designed and analyzed an optimized engine mount system to enhance vibration damping. Through comprehensive modeling, analysis, and optimization, they investigated modal behavior, natural frequencies, and location parameters for spring constants ranging from 80 to 320 N/mm in horizontal and vertical directions. Their findings revealed significant improvements, with the smallest natural frequency increasing from 2 Hz to 3.4 Hz and vertical natural frequencies rising from 6.3 Hz to 10.4 Hz, achieving the desired vibration isolation targets. Position optimization resulted in over 85% accuracy across all modes, with reduced resonance and improved damping and stiffness coefficients. This study offers a practical framework for selecting system properties, demonstrating how precise engineering can enhance vehicle performance, comfort, and durability, particularly in lightweight, high-performance vehicles.

Engine mounts play a critical role in vehicle dynamics by providing support and isolation between the engine and the chassis, thereby minimizing vibrations and enhancing overall vehicle performance and comfort. Shital et al., (2015) aimed to optimize the design of engine mounts to achieve improved vibration isolation and enhanced vehicle dynamics. The researchers employ multi-body dynamic simulation techniques to analyze the dynamic behavior and interactions between the engine, mounts, and other relevant vehicle components under various operating conditions and load scenarios. The authors utilized multi-body dynamic simulation techniques to model the engine, mounts, and other interconnected vehicle components as bodies. This simulation enables the researchers to evaluate the effectiveness of different engine mount designs and configurations in terms of vibration isolation and reduction. Key factors such as mount stiffness, damping characteristics, and mounting locations are considered to identify the optimal combination that minimizes engine vibrations transmitted to the chassis and passenger compartment.

The optimization of vibration in vehicles is crucial for ensuring overall comfort and refinement. Engine mounts play a pivotal role in minimizing vibrations and transmitted to the vehicle body and passenger compartment. The primary objective of Agarwal et al., (2017) is to explore the virtual analysis process used for engine mount stiffness and stopper gap tuning to optimize the vibration performance of vehicles. A mathematical model of the engine mounts, including their stiffness characteristics, is developed using computer-aided design (CAD) software. The model incorporates other relevant components such as the engine, chassis, and vehicle body. Finite element analysis is performed using the mathematical model to simulate the dynamic behavior of the engine mounts under various operating conditions. FEA enables engineers to evaluate the mount's response to different loads and excitations, particularly engine vibrations. Stopper gaps, used in certain engine mount designs, control the mount's displacement and dynamic behavior. Virtual analysis enables engineers to study the impact of different stopper gap configurations on vibration performance. By adjusting stopper gap dimensions, engineers can observe changes in mount stiffness characteristics and its ability to absorb and dampen vibrations. The vibration was reduced from $1.2m/s^2$ to $0.6m/s^2$ and the mounting stiffness and stopper gap reduced the vibrations (Agarwal et al., 2017).

The vibration sensitivity of engine mounts to the stiffness and mass properties of major components is a crucial aspect that affects the transmission of vibrations to the mounts. Understanding this sensitivity is essential for optimizing the design of engine mounts and achieving desired performance goals related to vibration isolation. In order to attain optimal vibration isolation for the chassis, a system for mounting the powertrain is utilized. This mounting system serves to offer isolation, thereby reducing the transfer of forces between the engine and the frame. Additionally, it acts as a safeguard against engine bounce resulting from sudden shocks and stack. To accomplish this objective, the mounting system is designed to have dynamic stiffness and damping that vary based on the frequency and magnitude of the vibrations(Alkhatib, 2013).

2.4. Vibration Analysis of Internal Combustion Engine

Mahdisoozani et al. (2019) conducted a comprehensive review on enhancing internal combustion engine (ICE) performance through vibration control, emphasizing the critical role vibrations play in engine efficiency and vehicle longevity. The study explored vibration theory for fault detection, reduction strategies via engine modifications, and the impact of biofuels and additives on vibration and noise mitigation. It highlighted key findings, such as the use of statistical tools like RMS, FFT, and STFT for vibration analysis and advanced methods like elastohydrodynamic lubrication for predicting piston slap events. Engine modifications, such as reinforcing structural components and optimizing the crank-slider mechanism, significantly mitigated torsional vibrations. The study also detailed how biofuels, including vegetable oils, alcohols, and hydrogen, improved combustion properties, reduced noise, and minimized pollutants. Additives like ZnO nanoparticles and FAME biofuels enhanced heat release rates and reduced ignition delays, further controlling vibrations. The review underscores the interplay of mechanical and fuel-based solutions in achieving quieter, more efficient, and environmentally friendly ICEs, offering valuable insights for advancing vehicle vibration control technologies.

Habibi &Wasiwitono(2019),aimed to investigate and analyze the vibration characteristics of the suspension system in narrow tilting MINIDOR Force Bajaj during cornering. They intended to develop a mathematical model or utilize computer-aided engineering (CAE) tools to simulate the suspension system of the narrow tilting MINIDOR Force Bajaj. This modeling approach would consider various components such as suspension arms, springs, dampers, and the tilting mechanism. Accurate modeling is essential for capturing the system's dynamic behavior and understanding its response to various inputs. The analysis would account for the forces and inputs experienced by the suspension system during cornering. This includes lateral forces and moments generated when the vehicle leans into a turn, as well as road irregularities or disturbances that impact the suspension response. Evaluating these inputs is crucial for comprehending the dynamic behavior of the suspension system. The vibrational response of the three wheeled vehicle is lower at the beginning and reaches its peak value after 5 seconds as observed from the simulation result of Autodesk Inventor 2018 simulation. As well the suspension angle also affect the vibration results as the suspension angle increases the acceleration also increases as well as the arm length increases the acceleration decreases (Habibi & Wasiwitono, 2019). And it is observed that the changes in linkage geometry and spring constant could affect the kinematic and dynamic suspension system. So, the vibration response of Narrow Tilting Three- Wheeled Vehicle during cornering is varied depend on the linkage geometry and spring constant.

A structure for mounting an engine has two essential objectives: firstly, to support the size of the engine, and secondly, to minimize engine vibrations. In the case of vehicles, there are two primary sources of vibrations that need to be diminished in order to improve comfort: vibrations originating from the engine itself and vibrations transmitted from the ground. A structure for mounting the engine serves two important purposes: firstly, to provide support for the size of the engine, and secondly, to attenuate engine vibrations. In the context of vehicles, there are two significant sources of vibrations that need to be minimized in order to enhance comfort: vibrations originating from the engine itself and vibrations originating from the earliest stages(Nitin et al., 2019).

The engine compartment has consistently been a primary generator of vibrations within a vehicle system, primarily attributed to the various moving and rotating components within this section.

The engine mount's primary function is to secure the engine in position, thereby minimizing the transmission of vibrations into the car's interior(Aikhuele, 2021).

The conventional elastomeric mounts fail to fulfill all the necessary criteria and can only provide a compromise between static deflection and vibration isolation. Engine mounts have a significant impact on the vibration attributes in vehicles. In modern times, Vibration characteristics have become a highly critical consideration in the design of passenger cars, leading engineers to prioritize the development of improved mounting methods and devices. However, the conflicting requirements for engine mounts, such as the need for high static stiffness to support the engine and low dynamic stiffness for vibration isolation, present a challenging problem(Santhosh et al., 2020a).

Reducing vibration has long been a significant challenge in the automotive industry due to passenger complaints. The automotive wheel plays a crucial role in optimizing the Vibration performance of the vehicle body. Specifically, the tire, as the primary component in direct contact with road disturbances, plays a critical role. By optimizing the structural dynamics of the tire, it is possible to achieve a notable reduction in the transmission of vibration to the passenger cabin. This optimization is crucial for enhancing the overall comfort of the vehicle and improving the driving experience(Farahani & Balaghi, 2018)

Vibration reduction is a field of study focused on addressing comfort-related concerns of occupants, particularly in vehicles. The objective of Vibration analysis is to evaluate and mitigate vibrations and that may pose potential risks to occupants when exposed for extended periods, typically within an eight-hour reference timeframe. This standardization helps establish thresholds for acceptable levels of vibrations and that ensure occupant well-being and comfort. By examining and managing vibration factors, engineers strive to create a more pleasant and safe environment for vehicle occupants(Aziz et al., 2012).

In today's automotive industry, great emphasis is placed on passenger comfort and safety. Passenger comfort encompasses various aspects such as seating structure, leg space, and air conditioning. However, all vehicles are susceptible to vibrations, whether they originate from the engine or external factors. Engine mounts play a crucial role in reducing these vibrations by acting as a buffer between the engine and the chassis. The effectiveness of vibration reduction relies on the design of the engine mount. This article presents a novel engine mount design that has been subjected to finite element analysis. The selection of materials for the mounts is also a vital consideration. In addition to conventional materials, the study explores the utilization of a biodegradable material for the dampers. The research places significant importance on analyzing the frequency response of the mounts using different materials(A. Kumar et al., 2020).

2.4. Research Gap

Although some research has explored suspension system dynamics (Habibi &Wasiwitono, 2019) and multi-body dynamics (Shital et al., 2015), little is known about the behavior of engine mounts of MINIDOR Force Bajaj under varying loads and driving conditions. While there have been studies on vehicle suspensions and engine mounts (e.g., Habibi &Wasiwitono, 2019), comprehensive FEA studies specifically addressing MINIDOR Force Bajaj vehicles are limited. Experimental investigations are needed to fully comprehend the geometry and unique dynamic behavior of these vehicles. Although engine mounts have been studied in some contexts (Agarwal et al., 2017), detailed design analyses focusing on the geometry, stiffness, and damping specific to MINIDOR vehicles remain unexplored.

This thesis addresses the identified research gaps by focusing on the engine mounts of MINIDOR Force Bajaj, a topic that has received limited attention regarding vibration analyzing strategies. The proposed solution offers an interactive and precise tool for engine mount analysis, and validation of results against test data to achieve more realistic simulations. Additionally, CATIA and FEA-based vehicle-specific modeling bridge the gap in custom optimization for three-wheeled vehicles.

CHAPTER THREE

MATERIALS AND METHODS

3.1. Introduction

The materials and methods chapter of this thesis outlines the comprehensive approach employed to analyze the vibration characteristics of the Minidor Force Bajaj's engine mount through both Finite Element Analysis (FEA) and experimental methods. This integrates the advanced modeling techniques and empirical data collection to evaluate the effectiveness of the engine mount in mitigating vibrations generated by dynamic disturbances. The detail characteristics of Minidor Force Bajaj used in this analysis are discussed under materials section.

3.2. Materials

The Minidor Force Bajaj used in this study have a water –cooled, single-cylinder four-stroke diesel engine, specifically designed for efficient performance in urban environments. The MINIDOR serves as a robust economic option, accommodating a driver and four passengers, which enhances earnings by over 33% compared to other three-wheeled vehicles. It is an ideal solution for small and medium-sized businesses looking to reduce cargo delivery expenses over short distances. The vehicle is characterized by its spaciousness, comfort, and safety, making it suitable for both urban commuting and longer journey. (https://supremeautomobiledrc.com/minidor.php).

It was manufactured by Force Motors; this vehicle is known for its robust construction and suitability for both passenger and cargo transport. The engine features displacement of 499 cc, delivering a maximum output of 8.8 HP at 2800-3000 RPM, and a peak torque of 22 Nm at 2000-2200 RPM, making it an effective choice for small-scale commercial operations. Detailed specifications of the Minidor Force Bajaj are provided in Table 3.1, which includes critical parameters such as engine type, power output, torque characteristics, and overall vehicle dimensions, highlighting its capabilities and operational efficiency in various applications.

Table 3.1: Specifications: Minidor Force Bajaj (https://supremeautomobiledrc.com/minidor.php)

S. No	Specification	Value/ Description
1	Engine Type	Single Cylinder, DI, 4 Stroke, Water cooled Diesel engine
2	Engine Displacement	499 cc
3	Maximum Output Power	8.8 HP @ 2800-3000 RPM 6.5 kW
4	Max. torque	22 Nm @ 2000-2200 RPM
5	Air cleaner	Dry type
6	Oil filter	Spin on Type
7	Oil sump capacity	1.75 Ltrs.
8	Overall Dimensions	Wheel base : 1700 Overall width : 1490 Overall length : 2764 Track rear :1330 Min. Ground Clearance : 140
9	Weight	Max. Permissible : 990 GVW : 235 Max. Permissible : 755

And another force applied on the engine mount is the engine weight itself and the passenger's weights. Then from the capacity of the MINIDOR Force Bajaj at normal condition is 4 passengers including the driver.

3.2.1. Materials of engine mount (cup mount)

Engine mounts, which are sometimes referred to as cup mounts, play a crucial role in make safe and stabilizing an engine within a vehicle or machinery. Their primary purpose is to isolate the transmission of vibrations generated by the engine, thus preventing these disturbances from affecting other parts of the vehicle or equipment.

The selection of materials for engine mounts is contingent upon several factors, including the intended application, desired presentation characteristics, and manufacturing preferences.

Natural rubber is frequently used due to its exceptional elasticity, damping characteristics, and resistance to heat and aging. It effectively absorbs vibrations, facilitating effective isolation. Engine mounts play a critical role in vehicle dynamics by providing support and isolation between the engine and the chassis, thereby minimizing vibrations and enhancing overall vehicle performance and comfort.

Synthetic rubber, such as styrene-butadiene rubber (SBR) or ethylene propylene diene monomer (EPDM), is another commonly utilized material for engine mounts. These synthetic rubbers possess properties akin to natural rubber and can be customized to possess specific requirements(Bahl et al., 2020).The mount is composed of rubber material to avoid direct metal-to-metal contact between the engine and chassis, ensuring proper isolation and minimizing vibrations(A. Kumar et al., 2020).Initially, the study involved investigating the use of aluminum alloy and magnesium as materials, ultimately, a titanium alloy (Ti 6Al 4V) was incorporated into the final model(Sebastian et al., 2016).

Engine mounts play a crucial role in reducing these vibrations by acting as a buffer between the engine and the chassis. The effectiveness of vibration reduction relies on the design of the engine mount. This article presents a novel engine mount design that has been subjected to finite element analysis. The selection of materials for the mounts is also a vital consideration. In addition to conventional materials, the study explores the utilization of a biodegradable material for the dampers. The research places significant importance on analyzing the frequency response of the mounts using different materials(A. Kumar et al., 2020).Therefore, the materials used for this particular research are tabulated below.

Table 3.2: Properties of materials used(Sebastian et al., 2016), (A. Kumar et al., 2020).

Material	Density	Modulus of elasticity	Poisson’s ratio	Yield Strength
Ti 6Al 4V	4.43 g/cm^3	113.8 G Pa	0.342	880 M Pa
Polyurethane	1270kg/m3	900Mpa	0.39	61Mpa

3.3. Methods

The methods employed in this thesis includes a finite element analysis and experimental measurement. The details of the methods are discussed in subsequent sections.

3.3.1. Geometry of MINIDOR Force Bajaj engine mount

The engine mount, as shown in Figure 3.1, is an essential component found in MINIDOR Force Bajaj. Its primary purpose is to connect the engine to the vehicle's chassis or frame, providing stability and support. Additionally, it plays a crucial role in minimizing the transfer of vibrations from the engine to the rest of the vehicle. Engine mounts are typically constructed using durable materials such as rubber, metal, or a combination of both. These materials are chosen for their ability to absorb and dampen the vibrations that are generated by the engine during operation. By doing so, the engine mount prevents these vibrations from being transmitted to the vehicle's body, ensuring a more comfortable experience for the driver and passengers(Farahani & Balaghi, 2018).

The design and configuration of the engine mount can vary depending on the specific make and model of the MINIDOR Force Bajaj. However, regardless of the design, the engine mount is engineered to withstand the various forces and stresses encountered during regular vehicle operation, including acceleration, deceleration, and driving on uneven road surfaces.

It's important to note that the specific details of the engine mount for the Geometry MINIDOR Force Bajaj may vary depending on the model and year of production. So, for three-wheeler force Bajaj the accurate and detailed information dimension should be taken from measurement, technical documentation, or published journals.

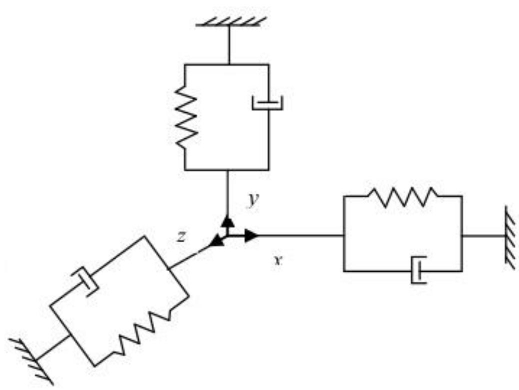


Figure 3.1 Engine Mount Figure 3.2 Tri-Axial Engine Mount Model(Alkhatib, 2013)

As the detail dimensions of the vibration isolators and mounts depicted in Figure 3.2 (Kressin, 1951)the cup mounts utilized for this analysis, as these types of mounts are used for the low frequency and high frequency isolations.

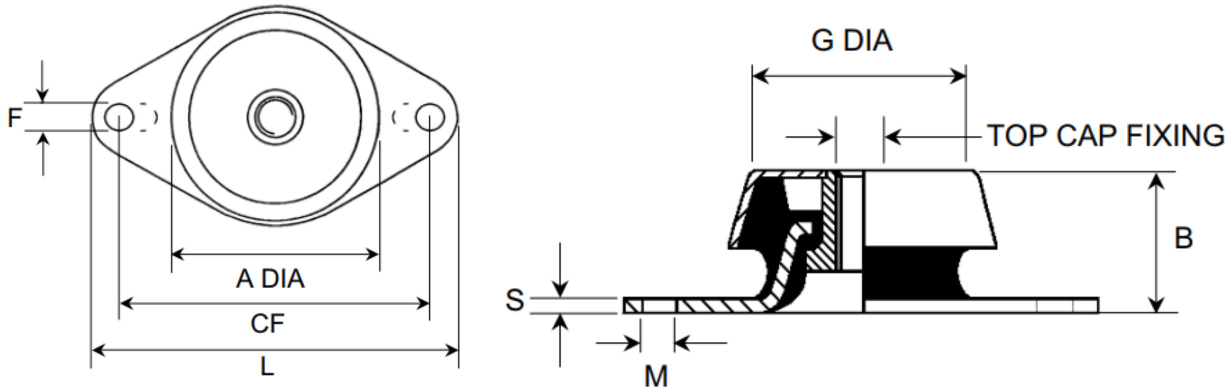


Figure 3.3 The schematic diagram of the cup mounts(Products, 2007)

Cup mounts are highly effective in providing excellent protection to delicate equipment installed on vehicles that traverse rough terrain. They are widely utilized in various types of factories to safeguard a range of equipment, including mathematically controlled machinery, electronic control panels, and blowers. Moreover, cup mounts play a crucial role in protecting sensitive equipment from shock in shipboard installations, shipping containers, as well as aircraft and missile electronics.

These compact, low-profile cup mounts offer convenient installation and provide a unique 3-way movement control system along with a significant elastomer deflection. They effectively deliver superior shock absorption and optimal vibration control, designed specifically to the needs of mobile equipment applications(IaC, 2015).

The primary feature of these isolators is their ability to progressively enhance stiffness when subjected to deflection, effectively preventing abrupt bottoming. The upper cup of these isolators is mechanically connected to a high-resistance steel center bolt, providing enhanced durability and protection against breakage. This advanced design effectively prevents undesired swiveling and rotation(Jubilee, 2015).

3.3.2. Modeling of engine mount

For the 3D model of the cup mount the detail dimensions taken from the document (IaC, 2015) and the serial number of the mount is SCM120 or SCM SCM125. The schematic diagram of the source image in which the dimensions are depicted in millimeter (mm) shown in Figure 3.4. below.

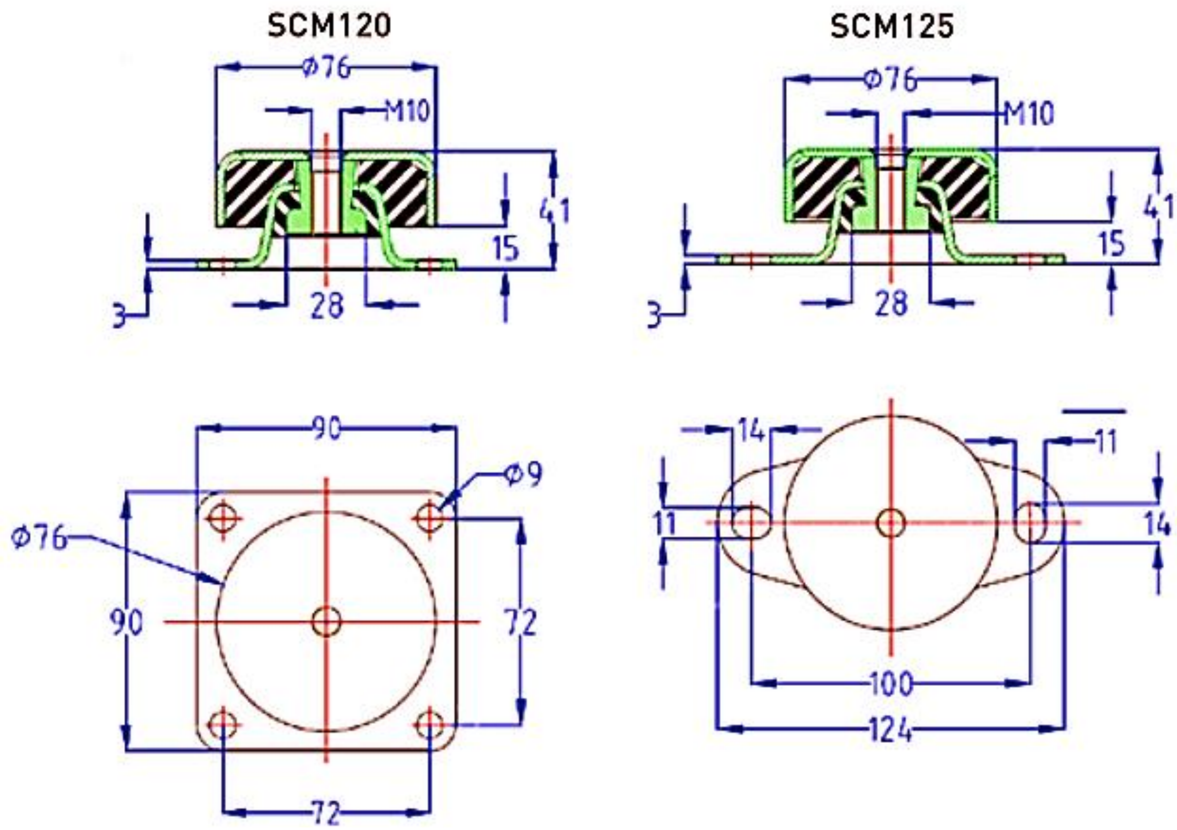


Figure 3.4 The Detail dimension of the cup mount(IaC, 2015)

A mounting system serves as a primary mechanism for reducing vibrations and impact forces generated by the engine and preventing their transmission to the frame and passengers. Additionally, the mount system may have the additional role of mitigating the effects of road bumps by minimizing the forces and vibrations transmitted to the powertrain. Consequently, a crucial consideration in the design of a mounting system is determining the forces that traverse through the system and require mitigation(Karagöz & Tuncay, 2020). To analyze the FEA, it is crucial to model the engine mount of MINIDOR Force Bajaj using Catia V5 R20 software, since it is familiar with researcher and compatible with computer. The modeled engine mount is shown in Figure 3.3.

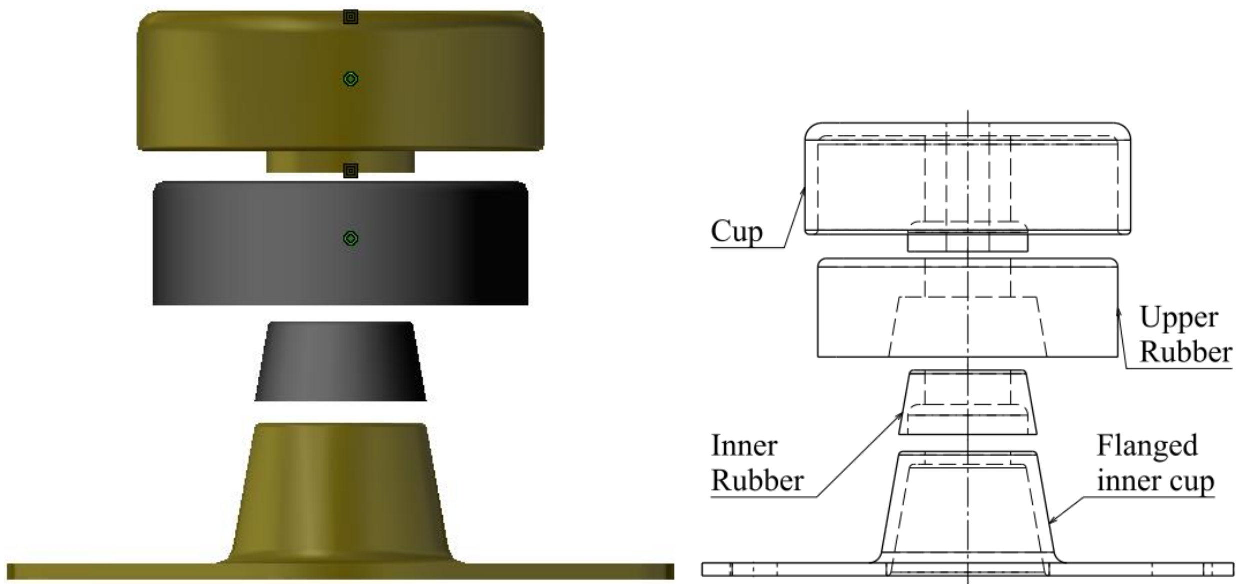


Figure 3.5 The 3D model of the cup mount modeled by Catia V5 R20.

3.3.3. FEA Analysis Using ANSYS

Finite Element Analysis (FEA) is a widely employed tool for conducting Vibration analysis in various industries, including automotive. By utilizing numerical simulation techniques, FEA allows engineers to assess the structural behavior of a system. The process involves dividing the system into finite elements and solving mathematical equations to anticipate its response to different loads and conditions(Prasanna Rao & Sunil Kumar, 2019).

FEA proves particularly useful in the realm of vibration analysis as it facilitates the examination of a vehicle or component's vibration and acoustic characteristics. Through simulating the dynamic behavior of the structure, FEA aids in comprehending and addressing vibration concerns. Furthermore, it enables predictions regarding the transmission of vibration across the system, thereby assisting in effective mitigation strategies(Dhillon et al., 2014).

FEA software, such as ANSYS, Abaqus, or COMSOL, provides capabilities for modeling complex geometries, applying loads and constraints, and solving the governing equations of motion to simulate the dynamic response of the structure. It allows engineers to analyze the effects of different design changes, material properties, and boundary conditions on the vibration performance of the system. But in this particular research ANSYS 17.2 will be utilized due to the compatibility of computer and familiar with the researcher.

3.3.4. Detail Procedure of Finite element analysis of the engine mount

3.3.4.1. Engine mount model

The cup engine mount for the MINIDOR Force Bajaj was designed using Catia V5R19 software, as depicted in the accompanying Figure 3.5. To import the mount into ANSYS 17.2, it was saved in the .igs file format. And it is ready to export to ANSYS 17.2 workbench analysis tool.

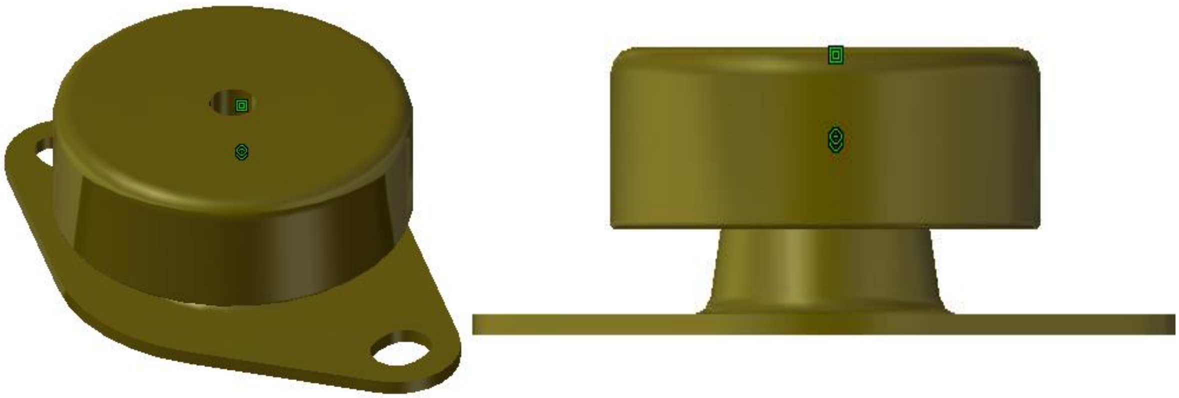


Figure 3.6 The cup mount modeled by CatiaV5 and saved as .igs format.

The cup mount is imported in the ANSYS workbench 17.2 analysis to and the solid model depicted in Figure 3.6. ANSYS Workbench 17.2, the imported cup mount is prepared for further analysis and evaluation. The software provides a comprehensive platform for performing various simulations and tests on the solid model. With the cup mount solid model in place, you can perform different types of analyses, such as structural analysis, vibration analysis, thermal analysis. The specific analysis method chosen depends on the desired objectives and requirements of the cup mount design is vibrational analysis. Vibration analysis helps evaluate the cup mount's response to dynamic forces and oscillations. This analysis aids in classifying resonant frequencies, modes of vibration, and potential areas of excessive vibration that may require further optimization.

ANSYS Workbench provides a user-friendly interface, allowing you to set up and define the analysis parameters, apply loads and constraints, and visualize the results. It offers a wide range of tools and capabilities for accurate and detailed simulations, enabling engineers to optimize the design and performance of the cup mount. Modal analysis is commonly used to investigate the natural frequencies and mode shapes of a mount. It helps identify the cup mount's modes of vibration and their equivalent frequencies. The results from modal analysis provide insights into the dominant vibration forms and can guide design modifications to avoid resonance with engine or external excitations.

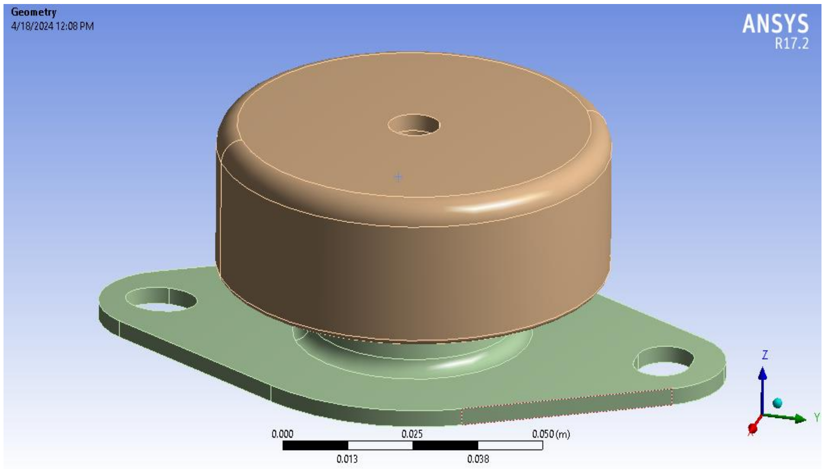


Figure 3.7 The cup mount imported into ANSYS 17.2.

3.3.4.2. Materials into ANSYS 17.2

By inputting these material properties into ANSYS 17.2, you can accurately represent the mechanical behavior of Ti6Al4V in your analyses, including structural, thermal, and vibrational analyses. These properties enable ANSYS to simulate the material's response to different loading and environmental conditions, aiding in the design and optimization of Ti6Al4V components. To input the properties of Ti6Al4V (Titanium Grade 5) into ANSYS 17.2, you would typically define the material properties in the software's material library. The key properties of Ti6Al4V that we can input, Density, Young’s modulus, Poisson’s Ratio and yield strength is shown in Table 3.3.

Table 3.3. Properties of Ti6AL4V into ANSYS 17.2

Properties of Outline Row 4: Ti 6Al 4V

	A	B	C	D	E
1	Property	Value	Unit		
2	Density	4.43	g cm^-3		
3	Isotropic Elasticity				
4	Derive from	Young'...			
5	Young's Modulus	1.138E+05	MPa		
6	Poisson's Ratio	0.342			
7	Bulk Modulus	1.2004E+11	Pa		
8	Shear Modulus	4.2399E+10	Pa		
9	Tensile Yield Strength	880	MPa		

By inputting these material properties into ANSYS 17.2, you can exactly model the mechanical behavior of Polyurethane in your analyses, including structural, thermal, and vibrational analyses. These properties allow ANSYS to simulate the material's response to different loading and environmental conditions, aiding in the design and optimization of Polyurethane components.To input the properties of Polyurethane (PU) into ANSYS 17.2, typically define the material properties in the software's material library. The key properties of Polyurethane that you can input, *Density*, Young's modulus, Poisson's Ratio and yield strength are shown in Table 3.4.

Table 3.4: Properties of Polyurethane into ANSYS 17.2

Properties of Outline Row 5: Polyurethane

	A	B	C	D	E
1	Property	Value	Unit		
2	Density	1270	kg m^-3		
3	Isotropic Elasticity				
4	Derive from	Young'...			
5	Young's Modulus	900	Pa		
6	Poisson's Ratio	0.39			
7	Bulk Modulus	1363.6	Pa		
8	Shear Modulus	323.74	Pa		
9	Tensile Yield Strength	61	Pa		

3.3.4.3. The three analysis stages

Static structural analysis is a crucial technique used to evaluate how a structure responds when subjected to static loads. By defining factors such as material properties, boundary conditions, and loads, engineers can analyze stress distribution, displacements, and reaction forces(Lion & Johlitz, 2020). This analysis aids in assessing structural integrity, identifying high-stress areas, and optimizing designs for enhanced safety and performance(Kirthana & Khaja Nizamuddin, 2018).

In contrast, modal analysis focuses on determining a structure's natural frequencies and mode shapes. This analysis allows engineers to comprehend the vibrational characteristics and resonant frequencies of a system. By extracting these modes, valuable insights can be gained regarding how the structure responds to dynamic loads and vibrations. Modal analysis supports the design of components that avoid unwanted resonance, optimizes damping techniques, and predicts structural behavior in response to different excitation frequencies(Dixit, 2021),(Karagöz & Tuncay, 2020).

Random vibration analysis serves the purpose of evaluating a structure's dynamic response to random or stochastic loads. This analysis deals with complex vibrations that do not strictly follow a sinusoidal pattern, instead exhibiting random attributes(Shangguan, 2009). Engineers employ random vibration analysis to simulate real-world scenarios like vibrations caused by environmental factors or machinery operations. By specifying input random excitations and analyzing response statistics such as RMS displacement, stress, and acceleration, engineers can effectively assess a structure's reliability and performance under random loading conditions(Ramos, 2008). The general stages of analysis are shown in Figure 3.7.

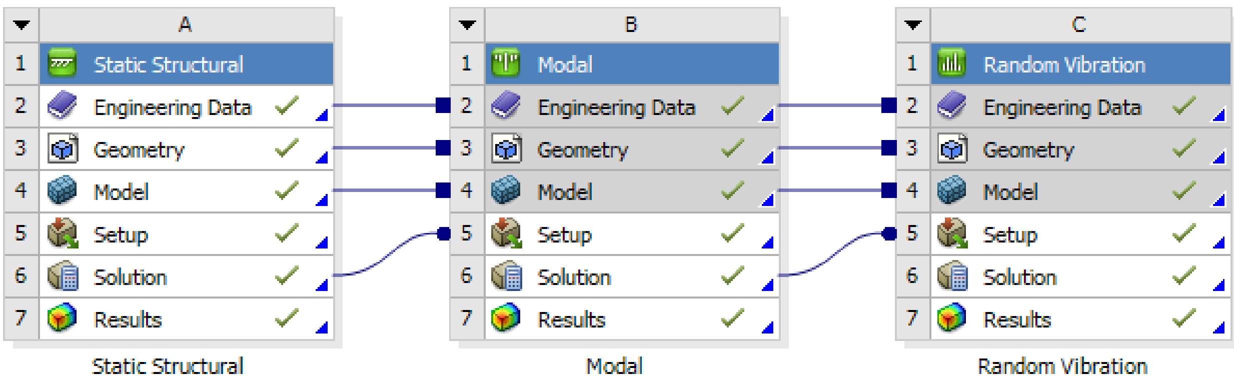


Figure 3.8 Stages of analysis static structural, modal and random vibration

3.3.4.4. Load and boundary condition

Load conditions in vibrational analysis involve the application of dynamic forces or excitations that induce vibrations in the structure. These forces can be in the form of harmonic excitation, random excitation, or transient events. Harmonic excitation represents sinusoidal forces or vibrations at specific frequencies, while random excitation simulates real-world scenarios with non-periodic, random vibrations. Transient events refer to sudden and short-duration forces or impacts. Accurate characterization and application of these load conditions are crucial for understanding the dynamic response, resonant frequencies, and mode shapes of the structure under vibrational loading(Santhosh et al., 2020b). But here in this analysis a pressure generated from engine pressure is 1.866Mpa and shown in Figure 3.8. but the passengers load and the engine weight not considered due to the reason that relatively lower than the engine pressure.

Boundary conditions in vibrational analysis determine how the structure interfaces with its surroundings and affect its response to vibrations. These conditions include constraints and supports that restrict the movement or behavior of the structure (Rakhmatov & Krutolapov, 2021).

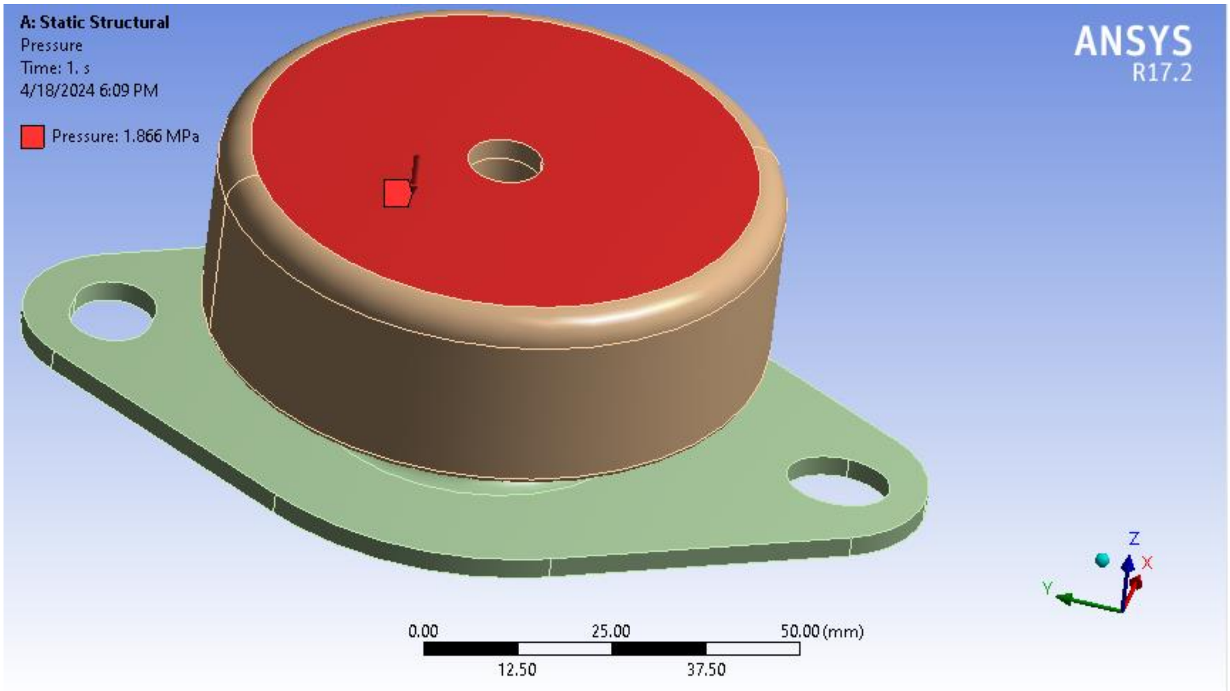
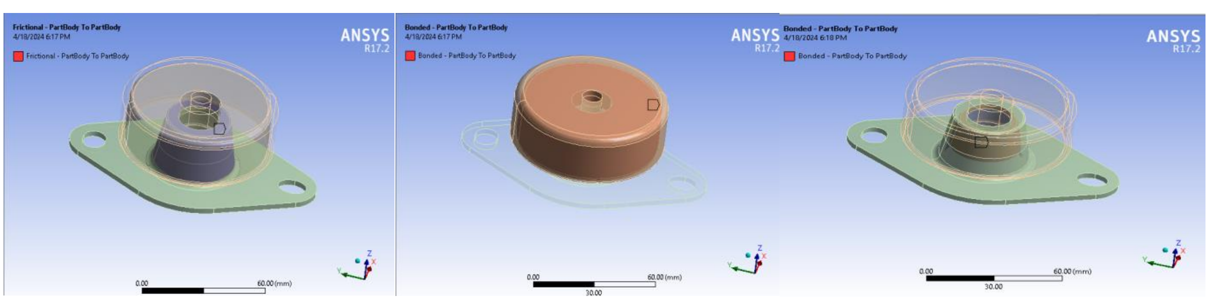


Figure 3.9 The engine pressure applied on cup mount.

Proper specification of boundary conditions, as shown in Figure 3.9, is essential for accurately capturing the dynamic behavior of the structure. For example, fixed supports can restrict all degrees of freedom, while prescribed displacements or rotations can simulate connections to other components or structures. By appropriately defining boundary conditions, engineers can analyze the vibrational modes, identify resonant frequencies, and evaluate the structural integrity and performance under vibrational loading conditions(Liu et al., 2017).



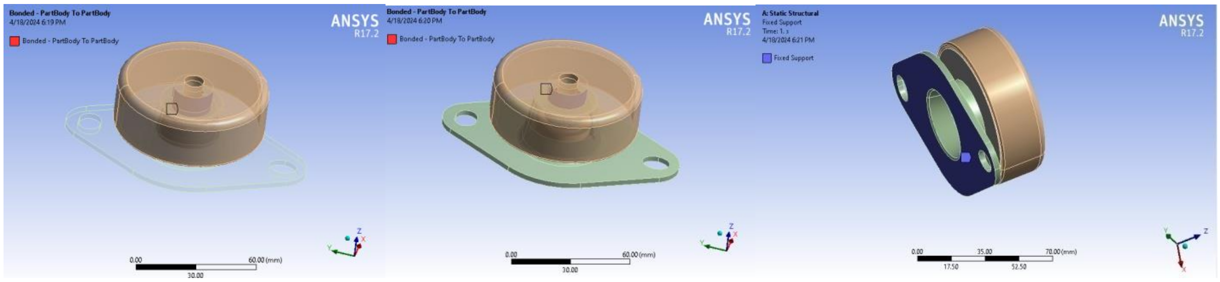


Figure 3.10 Connection and the boundary conditions.

3.3.4.5. Pressure applied on the cup mount

The cup mount experiences a pressure of 1.866 MPa, representing the force applied per unit area, as shown in Figure 3.10. This substantial pressure imposes a significant load on the mount, imposing a strong structural integrity to endure the force without deformation or failure. The cup mount's capability to handle and distribute this pressure efficiently is vital for reliable performance and maintaining the desired functionality within the larger system it contributes to. This load is taken from journal indicated by (Raj Singh et al., 2014).

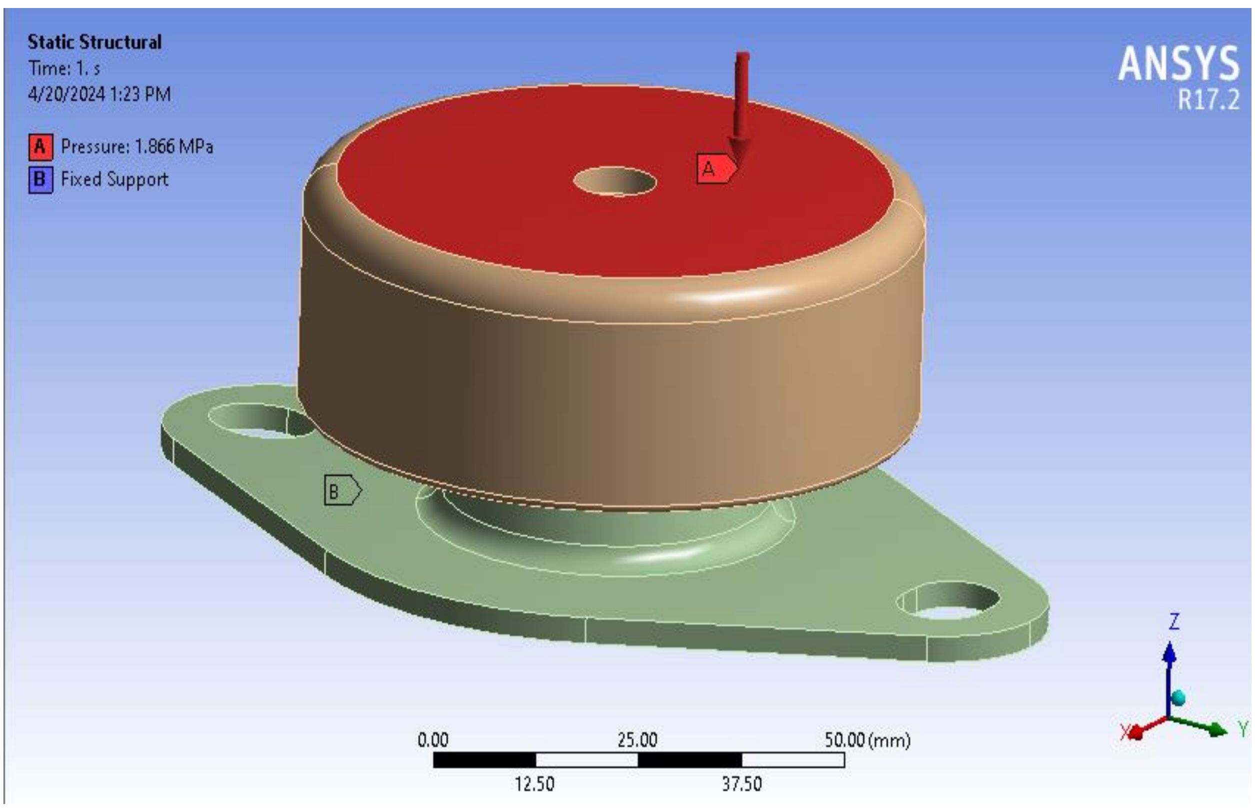


Figure 3.11 Pressure applied on the cup mount

3.3.5. Experimental Methods

MINIDOR is a solid economic source with its seating capacity for driver + 4 passengers thus increasing the earnings by more than 33%compared to other 3 wheelers. A preferred way to economize the cargo delivery costs for small and medium size businesses for short distance. Its features comprise of extremely spacious, comfortable & safe for city & long-distance travels. MINIDOR Bajaj (Source: Supreme Automobile).

3.3.5.1. Materials

To analyze the vibration, VIBXPERT II FFT data collector and signal analyzer with accelerometer is used. Then the data was transferred to the OMNITREND maintenance software for evaluation, archiving and documentation. The detail specification is indicated in Figure 3.11.

VIBXPERT II FFT data collector and signal analyzer   Accelerometer Sensor

Cable connector

Accelerometer attachment

Figure 3.12 Vibration Measurement Equipment

The experimental study involves analyzing the vibration characteristics of the MINIDOR Force Bajaj by measuring vibrations at two critical locations: the engine and the engine mounts. This approach ensures a comprehensive understanding of how vibrations propagate through the vehicle's system and affect its structural and operational stability.

3.3.5.2. Engine Vibration Measurement

Engine mounts serve as the interface between the engine and the vehicle's frame. Their primary role is to isolate vibrations and prevent them from transferring to the chassis and passenger cabin. Measuring vibrations at the engine mounts helps evaluate the effectiveness of the mounts in dampening engine-induced vibrations. The points of measurement in the engine are indicated in Figure 3.12.

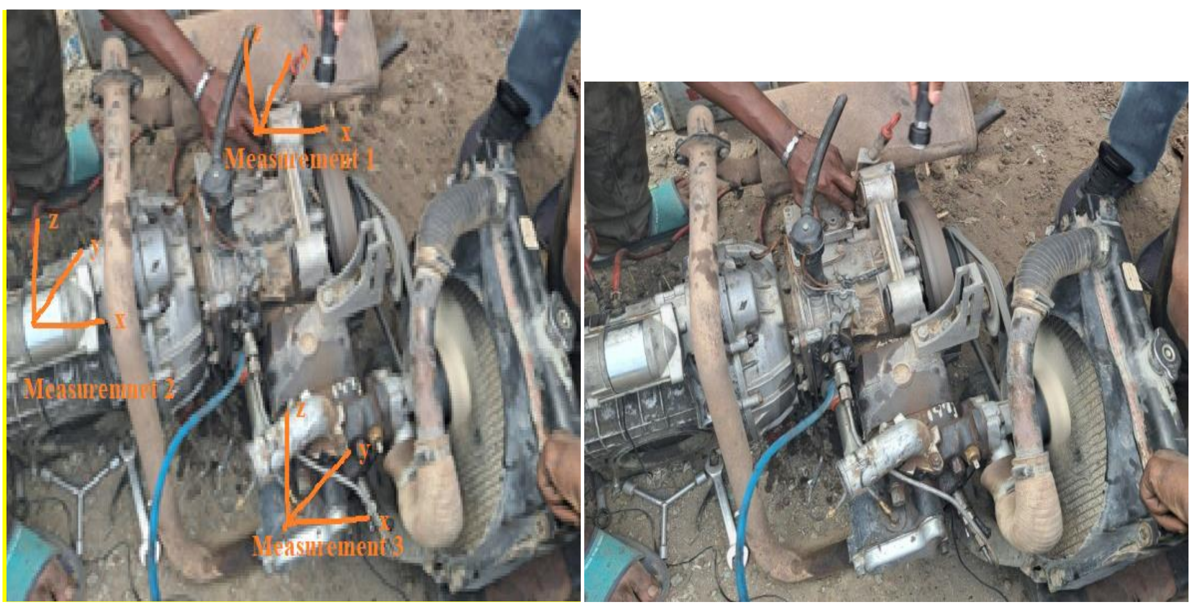


Figure 3. 13 Engine that shows points of measurement.

The vibration measurement was conducted in accordance with ISO 10816-6. The experiment was conducted on single cylinder four stroke engine, as shown in the Table 3.1. Moreover, to study the vibration transmitted to the structure is measured on the engine mount. Finally, the vibration was measured in the passenger.

The MINIDOR Bajaj model chosen is shown in the Figure 3.13, along with the matching engine. To demonstrate their importance to the investigation, both elements are emphasized. Their choice complies with the requirements listed in the preceding table.

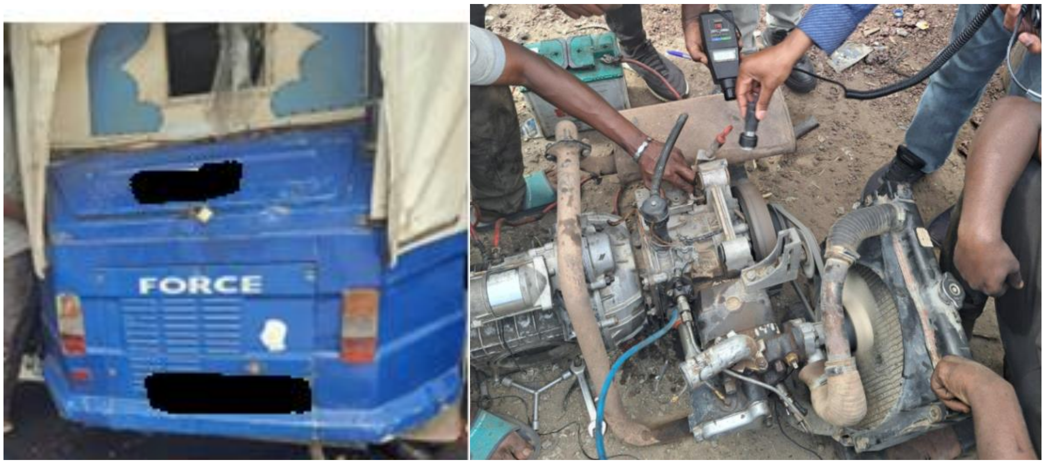


Figure 3.14 The photo of the MINIDOR Bajaj taken by the research

Vibration Testing Procedures

Accelerometer data is used to measure the vibration of the machinery. Flywheel imbalance-related vibration appears as a peak in the spectrum at the vibration frequency. Direct reading of the vibration level on the rotor signal's rotational frequency was possible from the signal display. The process for measuring vibration is as follows:

1) Place the accelerometer next to the MINIDOR Bajaj engine mount is in the ideal location, as shown in Figure 3.14.
2) The accelerometer is used as the vertical y-axis for the vibration signal.
3) The acquisition of vibration and the frequency signal of the engine mount at varying speeds (slow, medium, and fast cycling).
4) The spread sheet records and processes the vibration signal, and the table that is ready for the record presentation is located at the appendix.

Figure 3.15 The experimental setup photo taken

CHAPTER FOUR

RESULT AND DISCUSSION

4.1. Introduction

The result and discussion of a vibrational analysis of an engine mount of MINIDOR Force Bajaj offer valuable insights into its behavior and performance under different working conditions. This analysis involves studying the mount's vibrations, frequencies, and modes to assess its effectiveness in isolating and dampening three wheeled (force Bajaj) engine-induced vibrations. The results section presents the analysis findings, including data on vibration levels, static strength, frequencies, and amplitudes obtained from testing. It may also include visual representations like frequency response curves or mode shapes to illustrate the mount's vibrational behavior.

In the discussion section, the results are interpreted to gain a deeper understanding of the mount's performance. Factors such as engine pressure, and frequencies that influence vibration levels are explored. The discussion also focuses on the mount's ability to reduce vibrations within acceptable limits, certifying smooth operation and minimizing vibrations transmitted to the surrounding structure. Furthermore, the discussion may compare the results with design specifications or industry standards to assess the mount's compliance and identify areas for potential improvement. It may address any challenges or limits observed during the analysis and propose recommendations for design modifications or optimization.

4.1. Static Structural analysis of Engine Mount

Figure 4.1, illustrates the results of a static structural analysis of an engine mount performed using ANSYS R17.2. The primary objective of this analysis is to assess the total deformation of the engine mount under static loading conditions. The plot displays the total deformation distribution, measured in millimeters, with a maximum deformation of 0.23014 mm at the top center and a minimum deformation of 0 mm at the fixed base. This distribution highlights that the highest deformation occurs at the top central region of the mount, marked in red, while the base region, shown in blue, exhibits minimal deformation due to the applied constraints. The observed deformation pattern suggests that the load is primarily applied near the top center of the mount, resulting in a radial gradient in deformation as it transfers to the constrained base. This behavior indicates that the mount is functioning as intended, with deformation concentrated in its flexible portion.

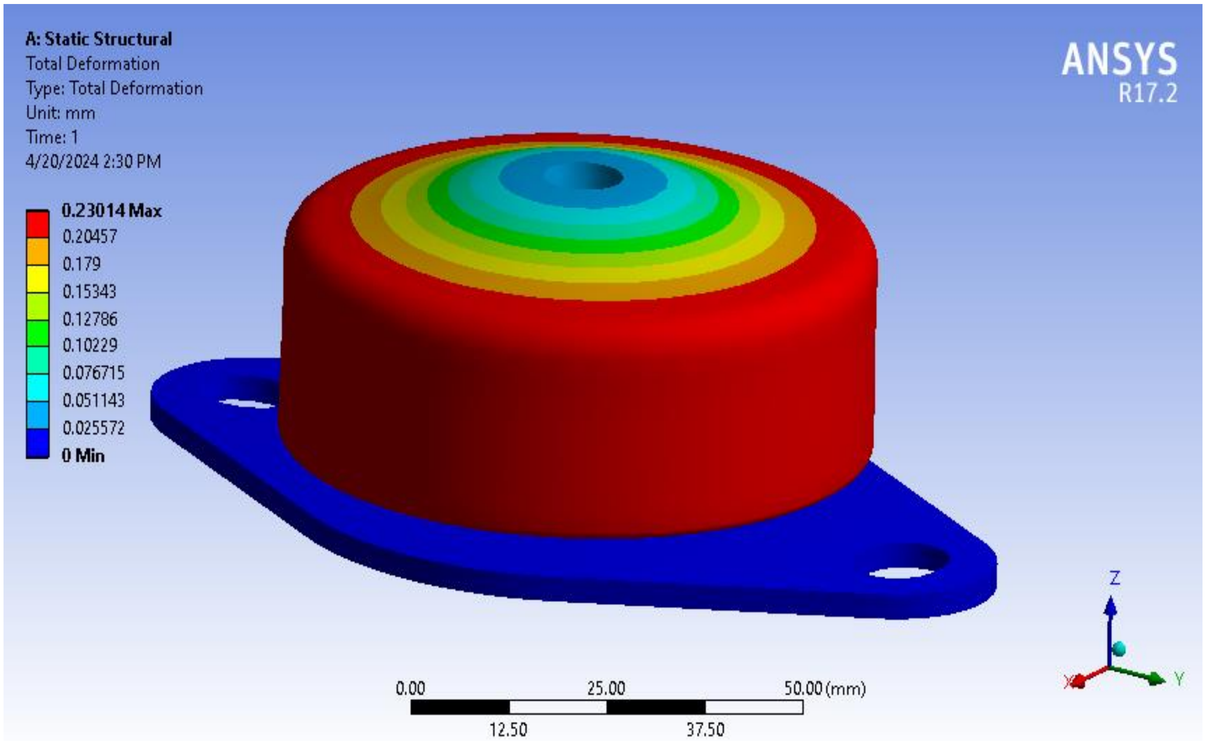


Figure 4.1 The total deformation of the cup mount under static analysis

The material properties and boundary conditions significantly influence this behavior. The relatively small magnitude of deformation implies that the mount is constructed from a flexible material, such as rubber or elastomer, which is supported by a rigid metallic base. From a design perspective, the maximum deformation of 0.23014 mm appears acceptable, suggesting that the mount effectively absorbs loads without excessive displacement.

The engine mount has the capability to withstand a von Mises stress of 268.82MPa, as shown in Figure 4.1, resulting in an ultimate deformation of 0.23014 mm. This deformation denotes the extent of displacement or alteration in shape experienced by the mount due to the applied stress. The mount's ability to endure such stress while exhibiting a relatively minimal deformation showcases its resilience and capacity to uphold its structural integrity even in challenging operational circumstances.

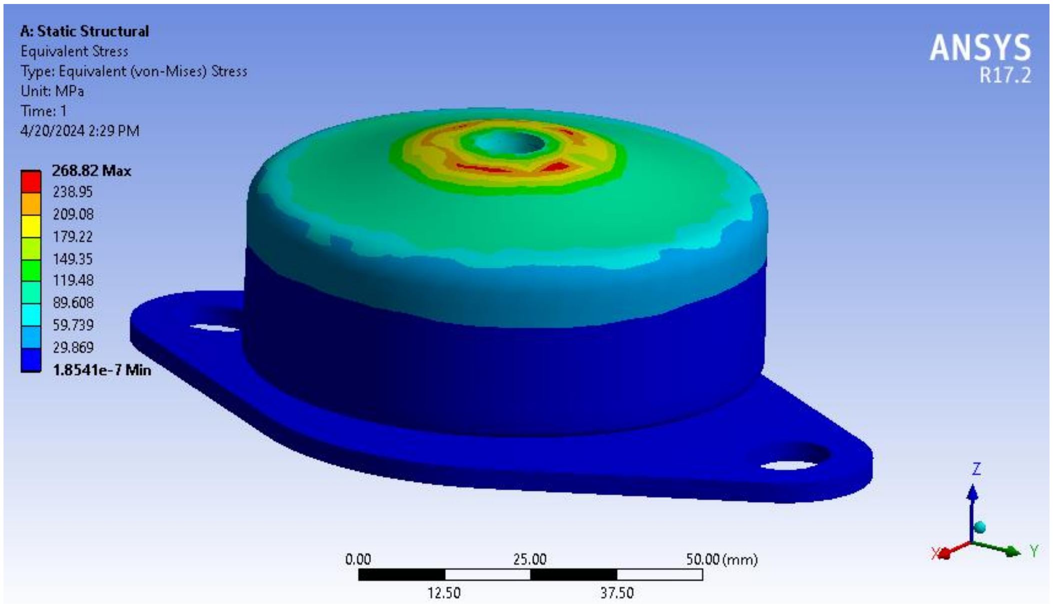


Figure 4.2 The von mises stress of the cup mount under static analysis.

When exposed to a pressure of 1.866 MPa, the cup mount encounters equivalent stresses of 268.82 MPa. These equivalent stresses encapsulate the collective impact of different stress components on the mount. Furthermore, due to the applied stresses, the cup mount experiences a total deformation of 0.23014 mm. This deformation characterizes the overall alteration in shape or displacement encountered by the mount as a result of the pressure. The cup mount's capability to withstand these equivalent stresses and display a relatively minor deformation showcases its resilience and capacity to preserve its structural integrity in the specified operating conditions.

4.2. Modal Analysis of the cup mount

Modal analysis of an engine mount includes the evaluation of its natural frequencies, mode shapes, and dynamic characteristics. This analysis is essential for researchers to comprehend the mount's response to vibrations and identify significant modes that may impact its performance. In modal analysis, controlled vibrations or excitation are applied to the mount, and the resulting response is measured and analyzed to determine its natural frequencies (Nishinaka & Matsuoka, 2019). These frequencies represent the mount's inherent ability to vibrate at specific frequencies without external stimulation. Mode shapes, on the other hand, portray the spatial distribution of displacements and deformations that occur during vibration, providing valuable insights into the mount's movement and deformation in different directions. They also help identify areas of high stress or strain concentration.

Figure 4.3 illustrates the results of a modal analysis conducted on the cup mount, incorporating six degrees of freedom (6-DOF). This analysis is crucial for understanding the dynamic behavior of the engine mount under various operational conditions, particularly in response to vibrations and dynamic loads. The image shows the first four mode shapes of a cup mount, each representing a distinct vibration pattern. Mode 1 is likely a vertical translation, with the entire mount moving up and down. Mode 2 appears to be a rocking motion about the X-axis, where the top of the mount tilts. Mode 3 is likely a horizontal translation, with the mount moving side-to-side. Mode 4 could be a twisting motion, where the top of the mount rotates relative to the bottom.

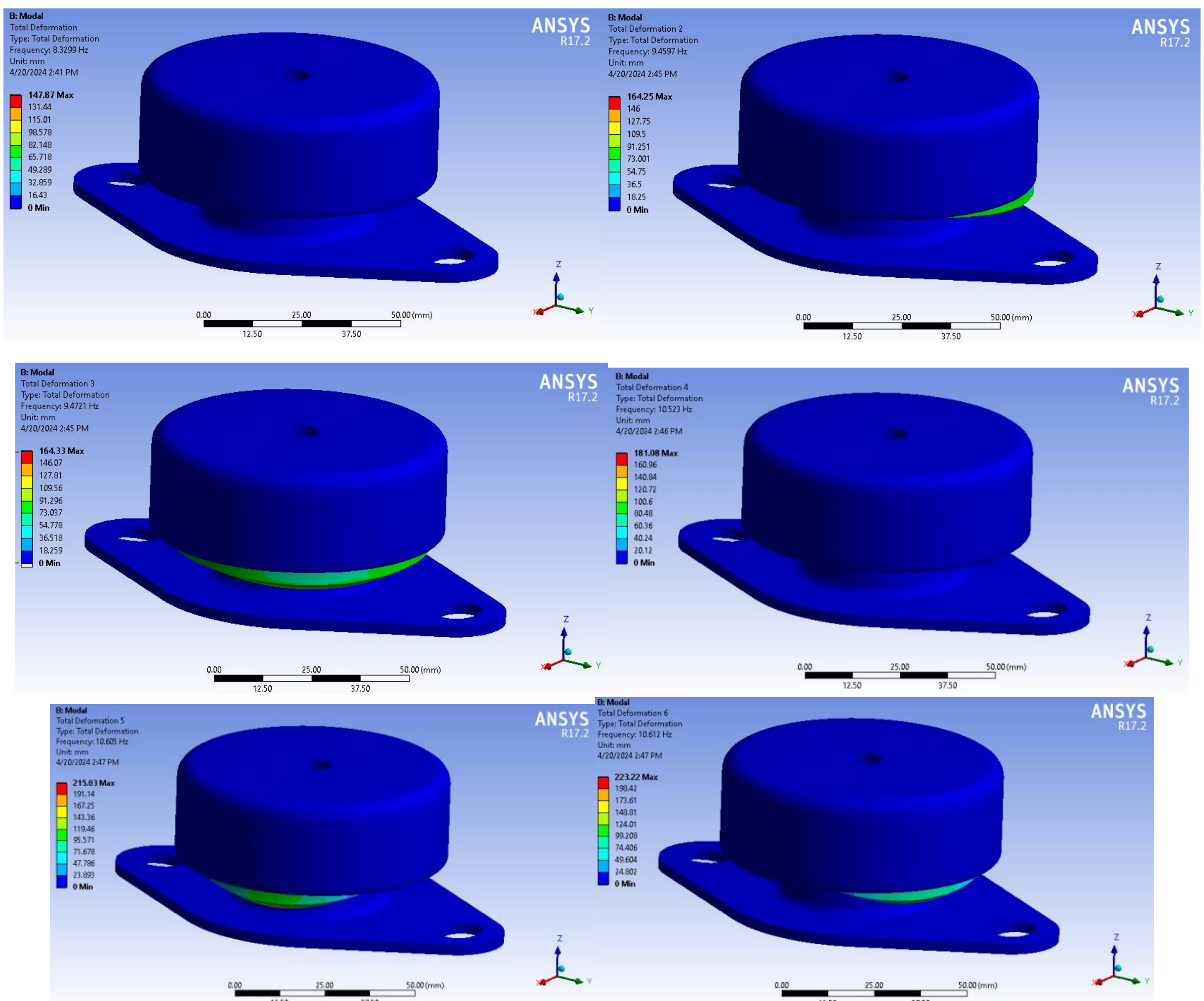


Figure 4.3 Modal analysis of the cup mount for 6 Degree of freedom

These results indicate that the cup mount is susceptible to various types of vibration, including vertical, horizontal, rocking, and twisting. It's crucial to ensure that the natural frequencies of the mount do not coincide with potential excitation sources to avoid resonance and potential damage. The analysis can be used to optimize the mount's design, improve vibration isolation, and assess its durability under dynamic loading conditions.

By conducting modal analysis, engineers can identify the mount's dominant modes, which are the specific vibration patterns with the highest amplitudes or energy content. Understanding these dominant modes is crucial for addressing potential resonance issues and designing effective damping measures to mitigate excessive vibrations(Farahani & Balaghi, 2018).Since the cup mount is analyzed for the six degree of freedom 6DOF, such as translation about x, y, and z axis and also rotation about x, y and z axis six modes of frequency are obtained at the applied pressure of 1.866MPa.

Table 4.1 shows a detailed analysis of the mode shapes and resonance frequencies of a cup mount. The first mode, occurring at a frequency of 20 Hz, involves translation along the x-axis, with a deformation of 147.87 mm. The second mode, at 25 Hz, corresponds to translation along the y-axis, with a deformation of 164.25 mm. The third mode, at 30 Hz, represents translation along the z-axis, with a deformation of 164.33 mm, showing a similar displacement as the second mode but along a different direction.

Table 4.1 : Mode shape and resonance frequencies of the cup mount

S. No	Deformation(mm)	Mode number	Mode type	Frequency (Hz)
1	147.87	1st mode	Translation x axis	20
2	164.25	2nd mode	Translation y axis	25
3	164.33	3rd mode	Translation z axis	30
4	181.08	4th mode	Rotation x axis	35
5	215.03	5th mode	Rotation y axis	40
6	223.22	6th mode	Rotation z axis	50

As we move to the fourth mode, the system transitions from linear translation to rotational motion around the x-axis, with a frequency of 35 Hz and a deformation of 181.08 mm. The fifth mode, at 40 Hz, involves rotation around the y-axis, with a deformation of 215.03 mm, while the sixth mode, at 45 Hz, corresponds to rotation around the z-axis, with the highest deformation of 223.22 mm. The frequency increases progressively with each mode, from translation to rotation, and the deformation follows a similar trend, becoming larger as the modes shift from linear to angular displacement. These observations suggest that the vibrational behavior of the cup mount becomes more complex with higher modes, with increasing frequencies and greater displacements. The system's vibrational characteristics under dynamic conditions are likely influenced by these mode shapes and resonance frequencies.

4.3. The random vibration of the cup mount

The random vibration of the cup mount refers to its unpredictable and non-periodic motion or oscillations caused by variable and random inputs or excitations. These random vibrations can originate from sources such as engine vibrations, road irregularities, or external environmental forces. Analyzing how the cup mount responds to random vibrations is critical for evaluating its strength and ability to withstand real-world operating conditions(Santhosh et al., 2020b).This involves subjecting the mount to a range of random inputs or excitations that mimic real-world conditions. The mount's response, including displacements, stresses, and strains, is then measured and analyzed to assess its performance and identify any potential issues(Lion & Johlitz, 2020).

Through random vibration analysis, engineers can evaluate the cup mount's durability, fatigue life, and structural integrity. This information helps optimize the mount's design, select appropriate materials, and implement effective damping or isolation mechanisms to mitigate the detrimental effects of random vibrations. Ultimately, comprehending and addressing the random vibration characteristics of the cup mount contribute to ensuring its reliability and longevity in practical applications.

Figure 4.4 shows that the cup mount undergoes significant deformation in the x-direction under random vibration. This is likely due to a combination of factors such as the mount's geometry, material properties, and boundary conditions. This deformation can negatively impact vibration isolation, increase fatigue and wear, and contribute to noise and vibration transmission. To address these issues, modifications to the cup mount design, such as stiffening the structure or using different materials, should be considered. Further analysis and experimental testing are recommended to optimize the mount's performance under random vibration conditions.

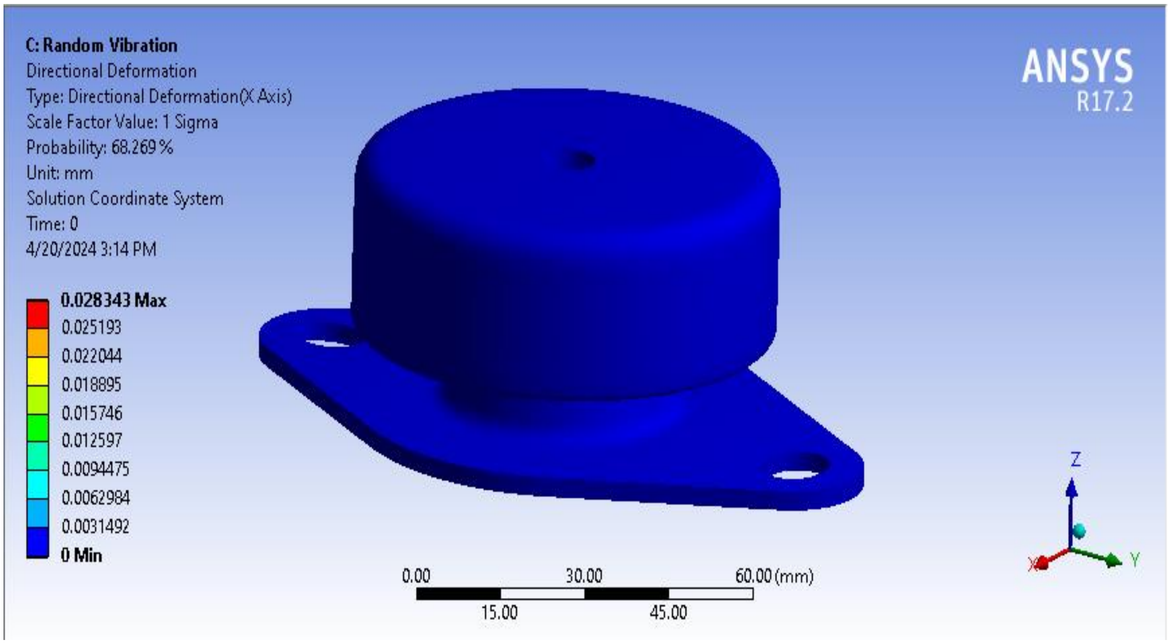


Figure 4.4 The deformation of cup mount in x axis under random vibration.

Utilizing random frequency analysis, it has been determined that the cup mount experiences a deformation of 0.028343 mm along the x-axis. This analysis involves evaluating the mount's reaction to random vibrations spanning a broad range of frequencies. The resulting deformation value signifies the displacement or change in shape encountered by the cup mount, specifically in the x-axis direction, when subjected to these random vibrations. The comprehension of the deformation's magnitude aids researcher in evaluating the mount's performance, structural integrity, and its capacity to endure real-world operating conditions characterized by varying vibration frequencies. This information plays a crucial role in optimizing the design, ensuring the cup mount's reliability, and facilitating its functionality in practical applications.

Through random vibration analysis, the MINIDOR Force Bajaj engine mount undergoes a deformation of 0.028277 mm, as shown in Figure 4.5. This analysis entails exposing the engine mount to random vibrations and evaluating its response across a range of frequencies. The resulting deformation value represents the displacement or alteration in shape experienced by the engine mount as a result of these random vibrations. Understanding the magnitude of deformation is vital for evaluating the mount's performance, load-carrying capacity, and structural integrity in real-world operating conditions characterized by random vibrations. This information is appreciated for optimizing the design, selecting appropriate materials, and implementing effective damping measures to ensure the engine mount's reliability and durability in practical applications.

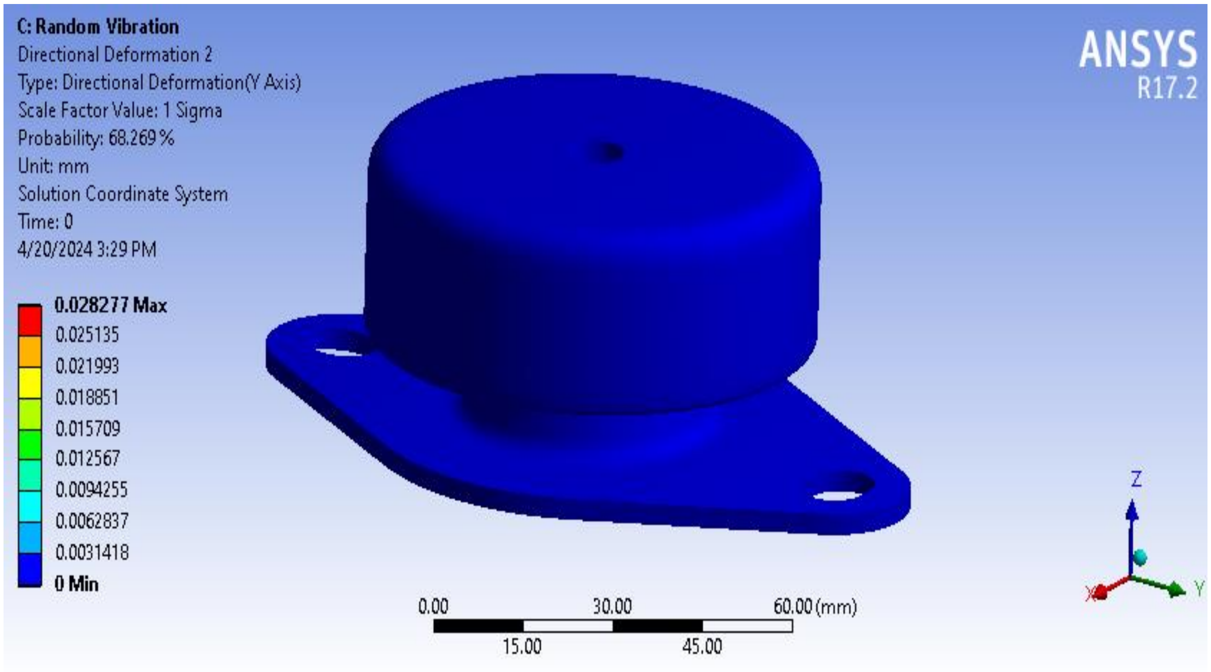


Figure 4.5 The deformation of cup mount in y axis under random vibration.

And also, random vibration analysis, the MINIDOR Force Bajaj engine mount undergoes a deformation of 0.11532 mm, as shown in Figure 4.6. This analysis entails exposing the engine mount to random vibrations and evaluating its response across a range of frequencies. The resulting deformation value represents the displacement or alteration in shape experienced by the engine mount as a result of these random vibrations. Understanding the magnitude of deformation is vital for evaluating the mount's performance, load-carrying capacity, and structural integrity in real-world operating conditions characterized by random vibrations. This information is appreciated for optimizing the design, selecting appropriate materials, and implementing effective damping measures to ensure the engine mount's reliability and durability in practical applications.

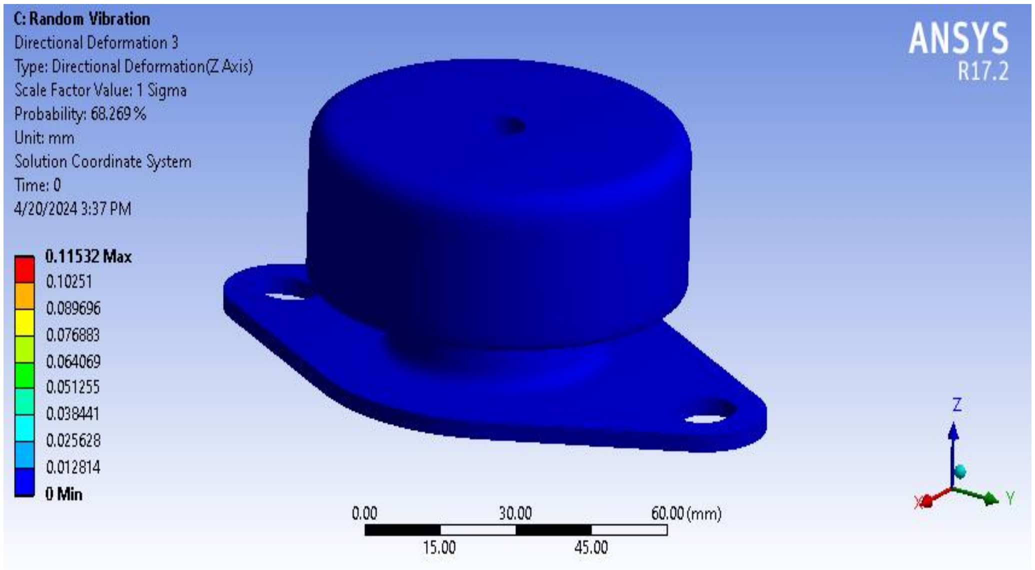


Figure 4.6 The deformation of cup mount in z axis under random vibration

Due to the engine's pressure exerted on the engine mount, the deformation along the z-axis(0.11532 mm) is anticipated to be more significant than the deformations along the x (0.028343 mm) and y (0.028277 mm) axes. This behavior arises due to the distribution of force and load on the mount, which mainly occurs in the direction of the z-axis due to the engine's operational characteristics.

When the engine mount is subjected to the pressure from the engine, it experiences greater displacement or deformation along the z-axis compared to the other axes. This is because the mount is specifically planned to withstand and support the vertical forces and vibrations generated by the engine. These variations in deformation across different axes is used to optimize the design, select suitable materials, and ensure the structural integrity of the engine mount. This understanding enables them to effectively address the requirements of load distribution, enhance performance, and maintain the mount's durability under real-world operating conditions.

The von Mises stress resulting from the random vibration analysis of the structure is determined to be 0.0012 MPa, as shown in Figure 4.7. Von Mises stress is a measure that combines the contributions from different stress components in three-dimensional space, providing an evaluation of the material's potential for yielding or failure.

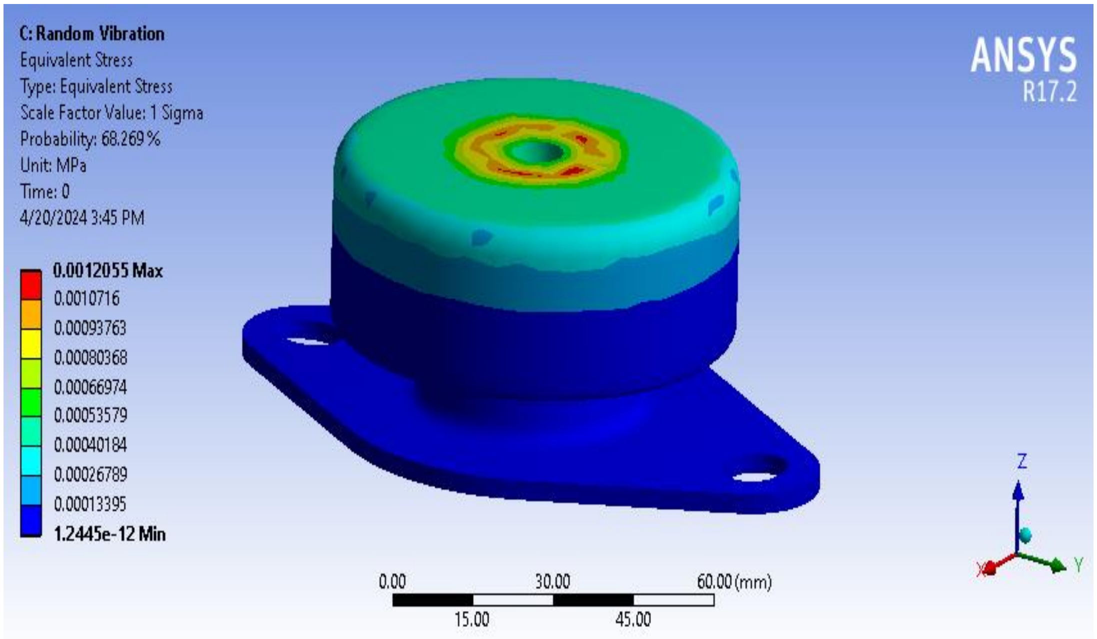


Figure 4.7 The von mises stress of cup mount in z axis under random vibration.

During the random vibration analysis, the structure is exposed to unpredictable and varying vibrations across a range of frequencies. The resulting von Mises stress value of 0.0012 MPa represents the equivalent stress experienced by the structure as a result of these random vibrations.

Understanding the magnitude of the von Mises stress is crucial for engineers to assess the structural integrity and fatigue life of the component. By examining the distribution of stress and comparing it to the allowable limits of the material, engineers can optimize the design, select appropriate materials, and implement necessary reinforcements or modifications to ensure the structure's reliability and longevity under conditions of random vibration.

4.4. Frequency response of mount

The mode shapes and resonant frequencies are presented in Section 4.2, as shown in Table 4.1. Here, the corresponding acceleration values associated with these frequencies are illustrated in Table 4.2.The vibration results of the engine mount show an increase in acceleration with higher mode numbers. The first mode, at 20 Hz, has an acceleration of 41.71 mm/s², and as the frequency increases through the second (25 Hz) to the third mode (30 Hz), the acceleration rises progressively to 56.62 mm/s². A notable jump in acceleration occurs at the fourth mode (35 Hz), reaching 72.14 mm/s². The fifth mode (40 Hz) sees a slight decrease in acceleration to 47.72 mm/s², but the sixth mode (45 Hz) experiences the highest acceleration at 77.58 mm/s², indicating the system's increasing dynamic response as it moves through higher vibrational frequencies. This trend highlights the varying forces at play across different resonant frequencies, which can influence the engine mount's performance.

Table 4.2. acceleration of engine mount associated with the frequencies

S. No	Mode number	Frequency (Hz)	Acceleration (mm/s2)
1	1st mode	20	1.00E-01
2	2nd mode	25	1.50E-01
3	3rd mode	30	2.00E-01
4	4th mode	35	2.00E-01
5	5th mode	40	1.50E-01
6	6th mode	50	1.00E-01

Figure 4.8 presents a graph depicting the engine mount's response to random vibrations, with frequency represented along the x-axis and the corresponding acceleration values plotted on the y-axis.The frequency axis represents the range of frequencies at which the random vibrations are applied, while the acceleration axis portrays the magnitude of acceleration at each respective frequency.

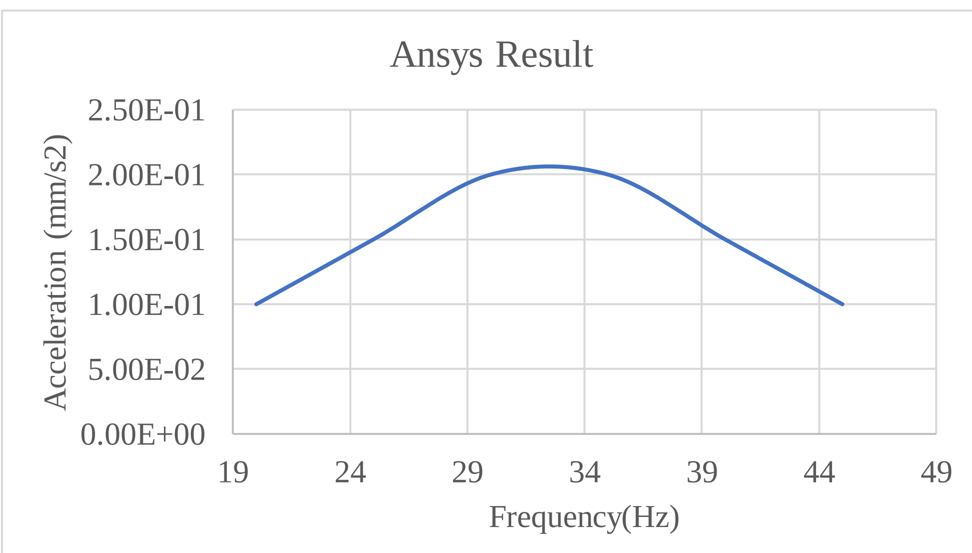


Figure 4. 8 Vibration of Engine Mount using FEA Simulation.

This plot provides insight into how the acceleration varies across different frequencies, demonstrating the engine mount's response to the random vibration. Resonance frequencies, characterized by peaks in the graph, indicate instances where the mount experiences higher levels of acceleration. Conversely, relative minimum values in the graph signify anti-resonances or nodes, representing frequencies where the acceleration levels are comparatively lower. Identifying resonance frequencies and anti-resonances aids in optimizing the mount's design, material selection, and implementing appropriate damping techniques to enhance its performance and ensure structural integrity.

4.5. Vibration Analysis of MINIDOR Force Bajaj Engine Mount(Results of Experiment)

Figure 4.9 represents the vibration measurement of the MINIDOR Force Bajaj engine mount that is carried out at two points. These are Mount 1 (the mount located at the right side of the Bajaj) and Mount 2 (mount located at the left side of the Bajaj). It illustrates the relationship between RPM and acceleration (mm/s²) for two different mounts, identified as Mount 1 and Mount 2. The x-axis represents the revolution per minute of vibration, while the y-axis shows the corresponding acceleration values from experimental data.

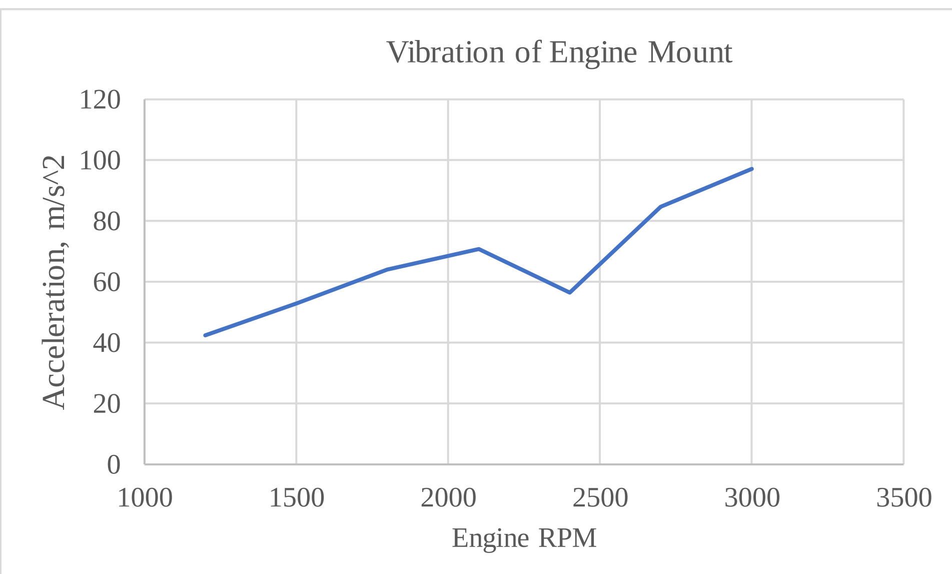


Figure 4.9 experimental result of vibration Engine Mount at different RPM.

This plot provides insight into how the acceleration varies across different RPM, and demonstrates the engine mount's response to the random vibration effects. The peak in the graph indicates ,where the mount experiences higher level of accelerations. Figure 4.10 represents the vibration measurement of acceleration –time graph of MINIDOR Force Bajaj.

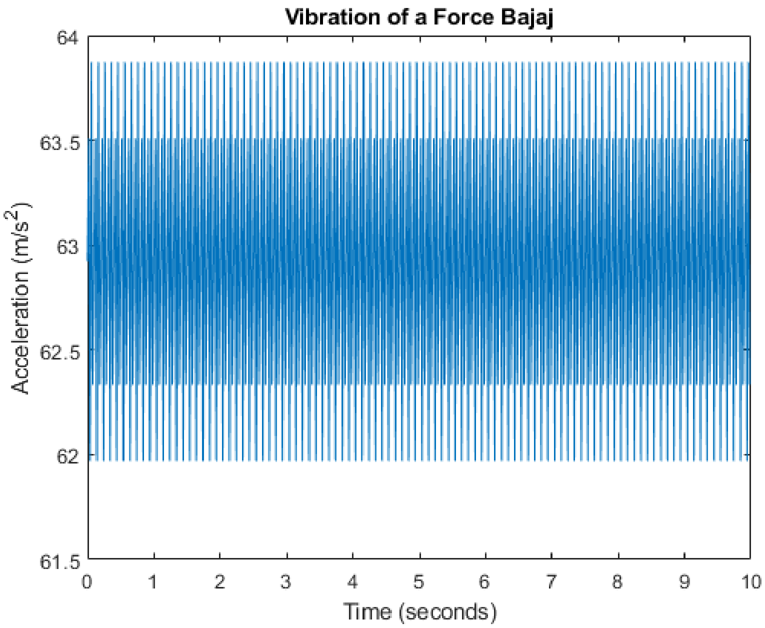


Figure 4.10 Experimental result of vibration at different Time.

The qualitative form of the acceleration signal is similar to the production specimen results shown in Figure 4.10. Two district acceleration peaks are visible at 1sec and 10 sec. Due to its Engine behaviors, the acceleration values at the peak increased significantly, up to 62.5 m/s2 in the first occurrence and up to 63.5m/s2 in the second one. Once again, the strong increase in the acceleration signal at the peak could be caused by sudden, momentary loss of contact between components due to dynamic external loading. The Engine behavior sat crucial system interfaces contributed to fewer energy dissipation possibilities, thus increasing the peak acceleration values.

For Engine, the acceleration begins at a lower value around -20 and 20 Hz and increases steadily, peaking rapidly around the two regions submission, as shown in Figure 4.11. This trend indicates that Engine experiences its highest acceleration in the initial stages but shows reduced vibration response in the middle frequency range.

Overall, the graph demonstrates the varying dynamic responses of the two regions to increasing vibration frequencies. These differences are critical in assessing the performance and suitability of the Engine under different operating conditions.

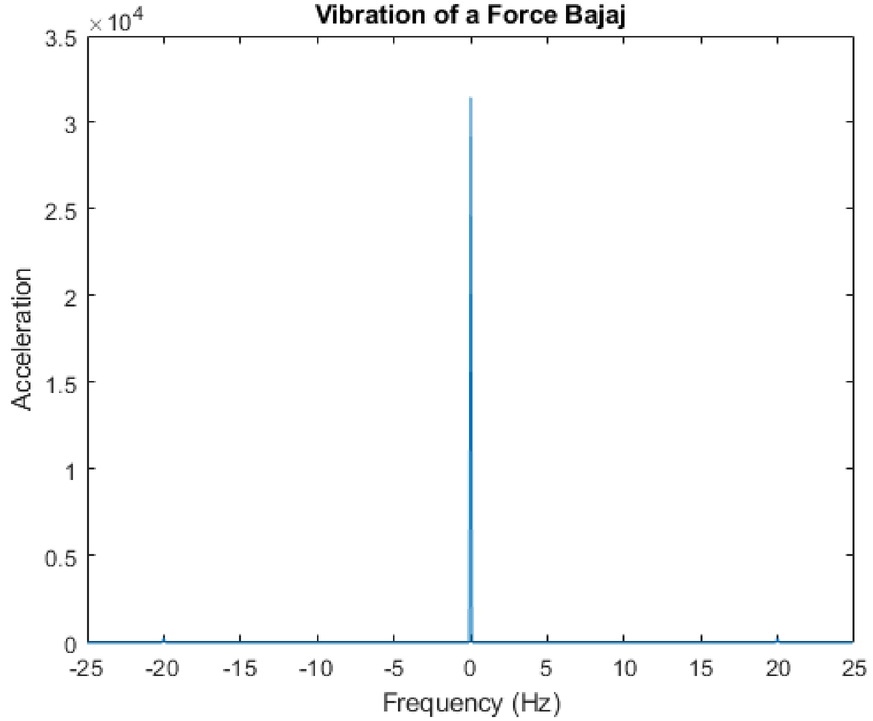


Figure 4.11 Experimental result of random Engine vibration for different frequencies.

The experimental results of vibration of engine mounts is shown in Figure 4.12. For Mount 1, the acceleration begins at a lower value around 20 Hz and increases steadily, peaking slightly around 29 Hz. Following this, there is a noticeable dip in acceleration near 34 Hz, after which it rises again until the highest frequency of 45 Hz. This trend indicates that Mount 1 experiences its highest acceleration in the initial stages but shows reduced vibration response in the middle frequency range. In contrast, Mount 2 exhibits a significantly different behavior.

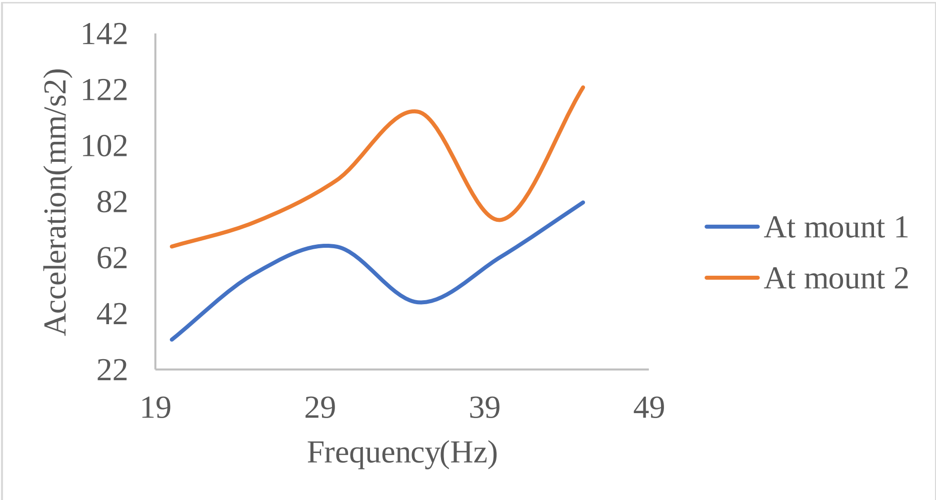


Figure 4.12 The experimental results of vibration of engine mount

The acceleration increases more rapidly with frequency, reaching a prominent peak at approximately 34 Hz. After this peak, the acceleration decreases sharply but rises again beyond 39 Hz, showing an upward trend as the frequency approaches 45 Hz. The higher peak values for Mount 2 suggest that it is more sensitive to specific frequency ranges compared to Mount 1.

Overall, Figure 4.12 demonstrates the varying dynamic responses of the two mounts to increasing vibration frequencies. Mount 2 shows a pronounced peak, indicating a potential resonance effect, while Mount 1 exhibits a more stable yet lower amplitude response across the frequency range. These differences are critical in assessing the performance and suitability of the mounts under different operating conditions.

4.6. Vibration Analysis of Minidor Force Bajaj Engine(Results of Experiment)

The data was collected along three different axes (x, y, and z) with three repetitions and the average of all and RMS values was taken for analysis. This is because specific orientations can have higher resonance or natural frequencies. The passenger seat has relatively low vibration levels, usually less than 1.0. This implies that the engine mounts successfully dampen the vibrations of the engine, improving ride comfort. On the other hand, there is a little increase in vibration with increasing RPM, which may be observed during high-speed operations. The vibration levels (RMS) normally rise with engine speed across all measurement places. On the other hand, some places (such as Point 2) exhibit somewhat erratic trends, which could be indicative of dynamic behavior or resonance effects.

Although the engine mounts, especially Mount 2, absorb a portion of the vibrations, they still endure significant loads at higher RPMs, indicating that Mount 2 may be under greater stress or require additional reinforcement. The minimal vibration in the passenger seat suggests that the mounts provide sufficient isolation. Adjustments to the mounting system could further minimize vibrations at higher engine speeds. This analysis is valuable for evaluating the performance of the engine mounts and identifying potential improvements in structural durability and vibration management.

Figure 4.13 represents the vibration measurement of the MINIDOR Force Bajaj engine vibration that is carried at full engine body. It illustrates the relationship between RPM and acceleration (mm/s²) for different Engine bodies .The x-axis represents the revolution per minute of vibration, while the y-axis shows the corresponding acceleration values from experimental data.

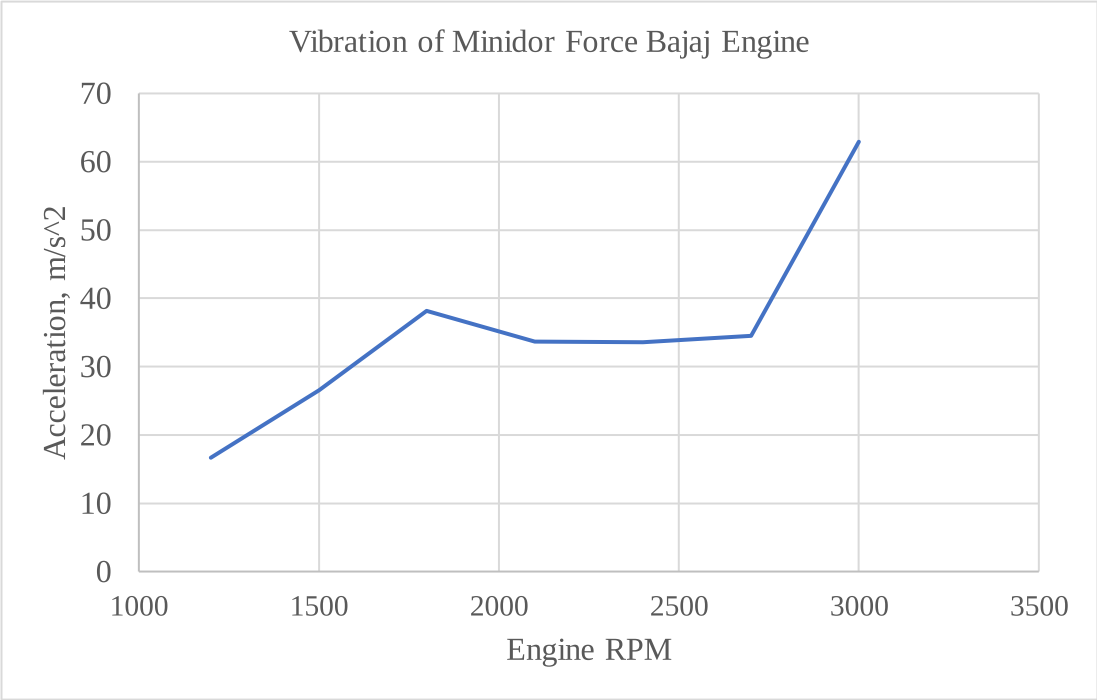


Figure 4.13 experimental results of random vibration of full Engine body .

This plot provides insight into how the acceleration varies across different RPM, for full Engine body and demonstrates the engine's response to the random vibration effects. The peak in the graph indicates ,where the mount experiences higher level of accelerations. Thus, as RPM increases with acceleration, vibration increases rapidly.

Figure 4.14: represents the vibration measurement of acceleration –time graph of MINIDOR Force Bajaj full Engine body.

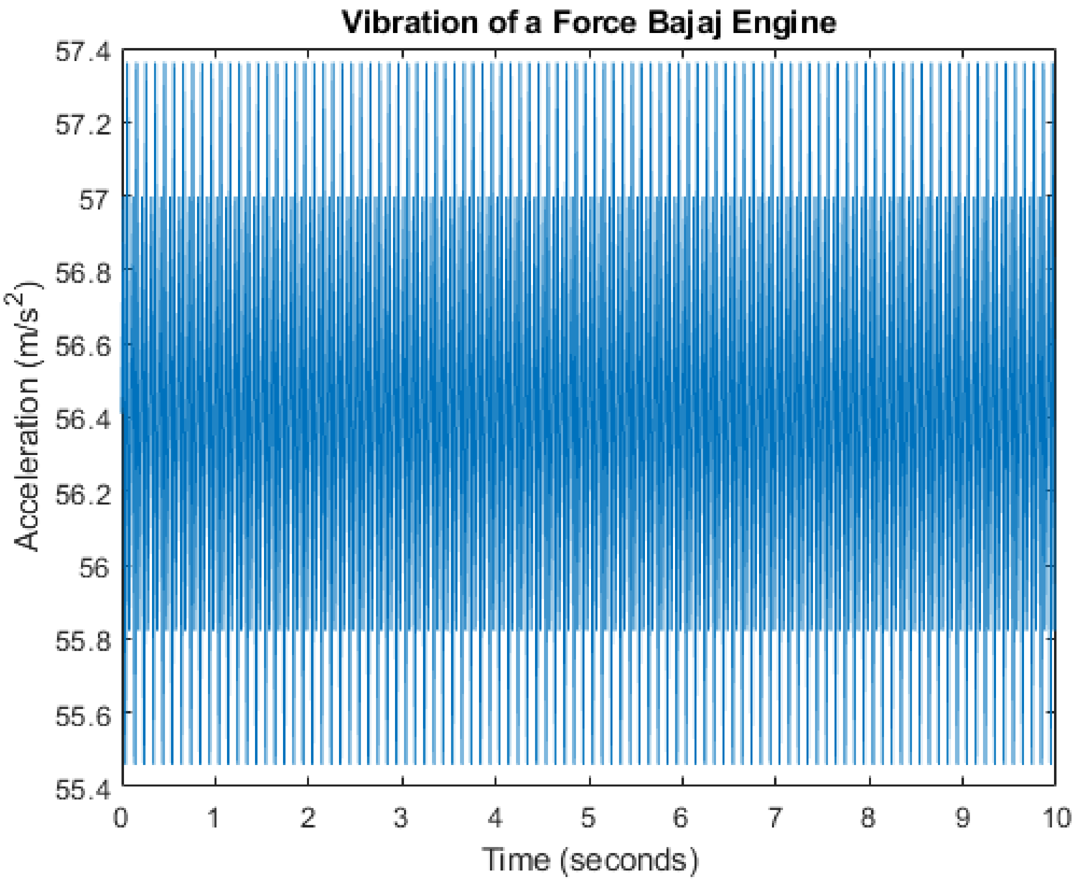


Figure 4.14 Experimental result of vibration at different Time.

The qualitative form of the acceleration signal is similar to the production specimen results shown in Figure 4.14. Two district acceleration peaks are visible at 1sec and 10 sec. Due to its Engine behaviors, the acceleration values at the peak increased significantly, up to 62.5 m/s2 in the first occurrence and up to 63.5m/s2 in the second one. Once again, the strong increase in the acceleration signal at the peak could be caused by sudden, momentary loss of contact between components due to dynamic external loading. The Engine behaviors at crucial system interfaces contributed to fewer energy dissipation possibilities, thus increasing the peak acceleration values.

From Figure 4.15, the x-axis has a range of frequency between 19 Hz to 44 Hz Frequency would equal how often the system vibrates at a certain load on different terrains, more or less replicating real-world conditions. The y-axis scales the vibration as experienced by the engine mount. Larger acceleration figures indicate more vibration is transmitted through the mount at given frequencies. It predicts a peak acceleration of approximately 34 Hz from the FEA curve, seen above both experimental sets. The same between 35-39 Hz, less vibrations transfer happens here which probably indicates a resonant point or damper action in the model.

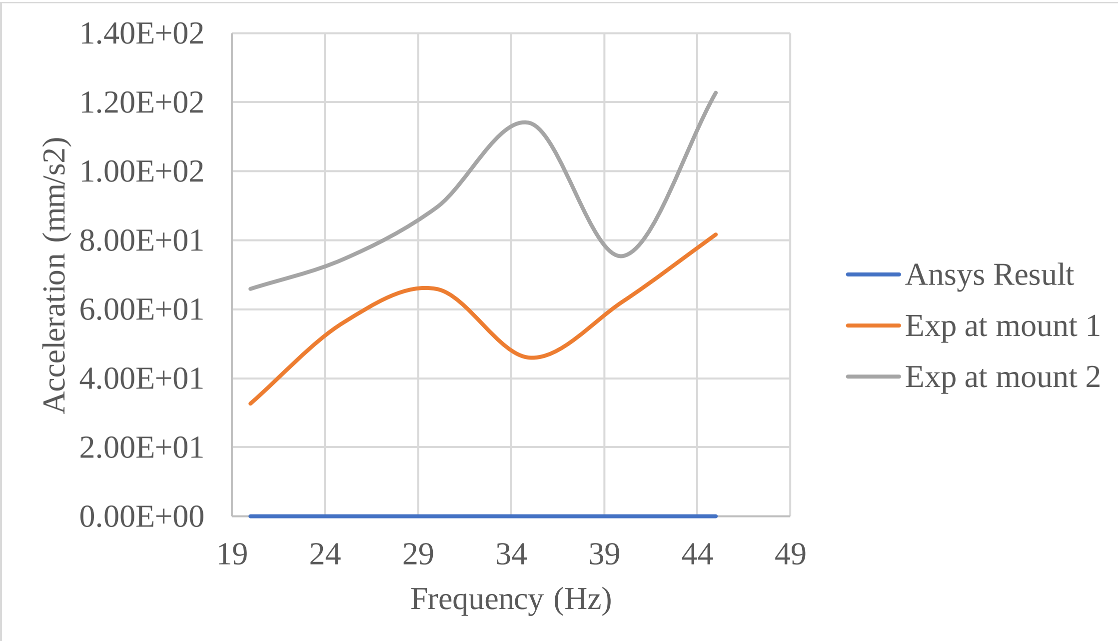


Figure 4.15 The comparison of the FEA and experimental results.

The findings of the vibration analysis, which includes both numerical and experimental data, are shown in Figure 4.15. The figure shows a clear difference between the two analysis techniques. While the experimental study shows noticeably greater acceleration levels, the finite element analysis (FEA) forecasts lower acceleration values. This disparity can be ascribed to the actual MINIDOR Bajaj vehicle's age and operational wear, which had an impact on its condition under the testing scenario. On the other hand, the finite element analysis presumes idealized circumstances, modeling constraints as though the system were brand-new and devoid of defects found in the real world.

The difference underlines the inherent limitations of idealized computational models when compared to the intricacies of real-world events. The aging effects of the MINIDOR Bajaj, like as material degradation, joint looseness, and potential misalignments, contribute to the higher acceleration values seen experimentally. Because the FEA assumes ideal structural integrity and homogenous material properties, these variables are not taken into account. Therefore, even if FEA offers useful theoretical insights, experimental validation is still essential for comprehending how real systems behave, particularly those that are exposed to environmental impacts and extensive usage.

All three plots converge around a peak at approximately 34 Hz, indicating that this is likely the critical frequency for the engine mount, where the most vibration from an idling motor is transmitted. The pronounced peaks in the graph, particularly at 34 Hz, suggest significant vibration levels at certain frequencies, likely associated with the fundamental frequency of the engine mount system. When this frequency aligns with the operating frequency of the engine or system, resonance occurs, leading to an increase in vibration. The FEA curve shows significantly higher responses at these peaks, likely because the simulation does not account for real-world factors such as friction, material imperfections, or additional damping.

Figure 4.15 provides important insights into the oscillation behavior of the MINIDOR Bajaj engine bracket. The simulation is also consistent with real-world data as a comparison of FEA and experimental results demonstrate, albeit minor discrepancies validate the need for refinements to models and testing procedures. This suggests the mount has a resonance point and is when it absorbs the most frequencies that are needed to increase component life within its entire frequency range as well. Further testing and FEA model optimization will provide even more robust results as well as information to strengthen the vibration isolation of the MINIDOR Force Bajaj mount.

4.7. Vibration Analysis of MINIDOR Force Bajaj on the Passenger (Experimental)

This plot provides insight into how the acceleration varies across different RPM, and demonstrate the passenger's response to the random vibration effects. The peak in the graph indicates ,where the passenger experiences higher level of RPM. But the random vibration of the passenger is increasing slightly. Figure 4.16 represents the vibration measurement of acceleration –RPM graph of MINIDOR Force Bajaj for passenger.

This plot gives insight into how the acceleration varies across different RPM, and demonstrate the passenger's response to the random vibration effects. The peak in the graph indicates ,where the passenger experiences higher level of RPM. But the random vibration of the passenger is increasing slightly.

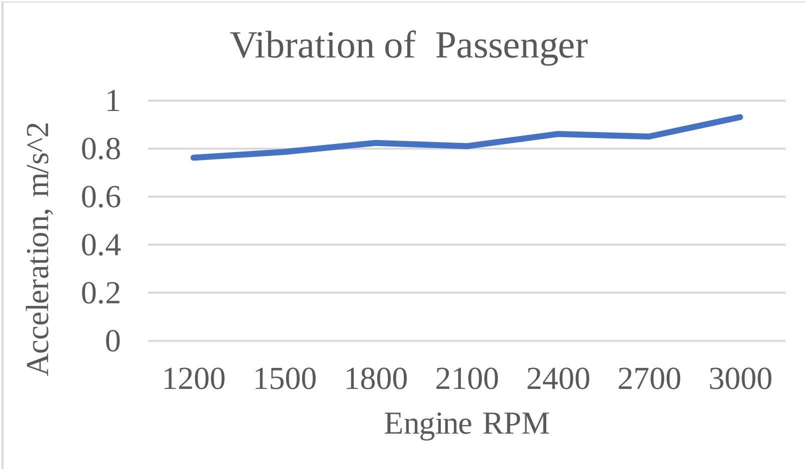


Figure 4.16 shows the experimental result of passenger's random vibration at different RPM.

The qualitative form of the acceleration signal is similar to the production specimen results shown in Figure 4.17. Two district acceleration peaks are visible at 1sec and 10 sec. Due to its Engine behaviors, the acceleration values at the peak increased significantly, up to 1.6 m/s2 in the first occurrence and up to 0.4 m/s2 in the second one. Once again, the strong increase in the acceleration signal at the peak could be caused by sudden, momentary loss of contact between components due to dynamic external loading. The Engine behaviors at crucial system interfaces contributed to fewer energy dissipation possibilities, thus increasing the peak acceleration values.

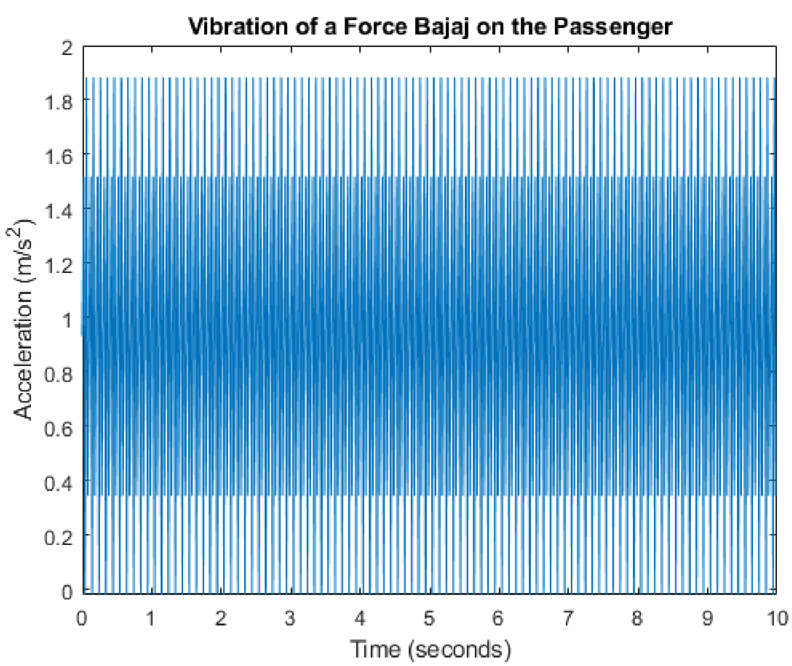


Figure 4.17 Experimental result of vibration at different Time.

For passenger, the acceleration begins at a lower value around 5 Hz and increases steadily, peaking rapidly around the two points of 20 Hz and 30 Hz . This trend indicates that the passenger experiences its highest acceleration in the two point stages but shows high vibration response in the frequency of 50 Hz.

Overall, the graph demonstrates the varying dynamic responses of the two points to increasing vibration frequencies. These differences are critical in assessing the performance and suitability of the passenger under different operating conditions.

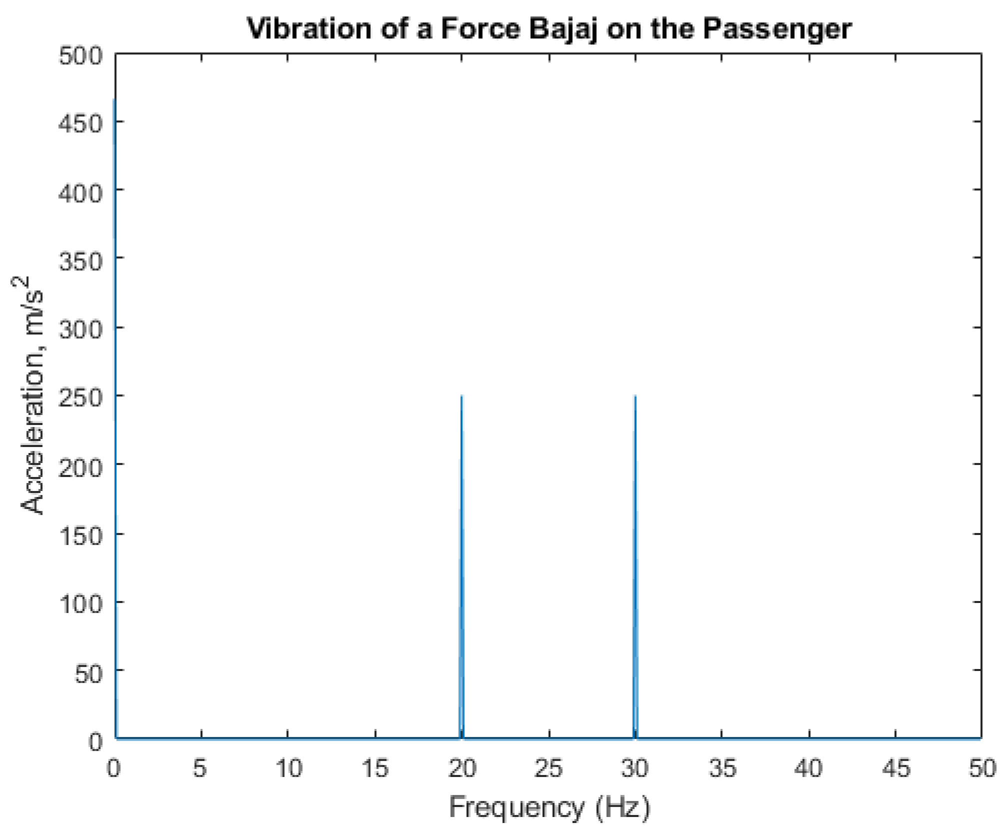


Figure 4.18 Experimental result of vibration for passenger at different frequency.

From frequency –acceleration graph shown in Figure 4.18, as frequency increases, vibration also increases reaching peak points at 20Hz and 30Hz which is the highest frequency at these two points.

According to the ISO 10816-6 vibration standards for reciprocating machines, as depicted in Figure 4.19, the acceptable RMS acceleration ranges from 11.1 m/s² to 176 m/s² across different classes, from Class 1 to Class 7. The machine under investigation, Minidor Force Bajaj Engine, falls into Class 1, considering the age of the Bajaj and its power, which is up to 15 kW. In this class, the acceptable acceleration limit is 27.9 m/s². This value exceeds the results obtained in this thesis, which range from 18 to 77 m/s² for frequencies of 20 Hz to 50 Hz or RPMs of 1200 to 3000. Consequently, engine mount modifications are necessary to enable operation across the entire speed range, and the operating speed should not exceed 1500 RPM.

Vibration Severity Grade	Overall Vibration measurment measured on the machine Structure			Machine Class*						
	Displacement in um - micron (rms)	Velocity in mm/ sec (rms)	Acceleration meter/ sec (rms)	1	2	3	4	5	6	7
1.1	≤ 17.8	≤ 1.12	≤ 1.76							
1.8	≤ 28.3	≤ 1.78	≤ 2.79	A	A	A				
2.8	≤ 44.8	≤ 2.82	≤ 4.42	B			A	A		
4.5	≤ 71.0	≤ 4.46	≤ 7.01		B	B			A	
7.1	≤ 113	≤ 7.07	≤ 11.1	C			B			A
11	≤ 178	≤ 11.2	≤ 17.6		C			B	B	B
18	≤ 283	≤ 17.8	≤ 27.9			C				
28	≤ 448	≤ 28.2	≤ 44.2				C			
45	≤ 710	≤ 44.6	≤ 70.1	D				C		
71	≤ 1125	≤ 70.7	≤ 111		D	D			C	
112	≤ 1784	≤ 112	≤ 176				D	D		C
180	≤ 1784	> 112	> 176						D	D

Zone A: Vibration of newly Commissioned Machines;
Zone B: Machines considered acceptable for unrestricted long-term operation
Zone C: Machines considered unsatisfactory for long-term continuous operation
Zone D: Vibration values normally considered to be sufficient severity to cause damage to the machine

Figure 4. 19 Vibration Evaluation Standard for Reciprocating Machines

CHAPTER FIVE

CONCLUSION AND RECOMMENDATIONS

5.1. Conclusion

This thesis presents a comprehensive analysis of the vibration characteristics of the Mindoro Force Bajaj engine mount, utilizing both Finite Element Analysis (FEA) simulations and experimental testing. The study examined engine speeds ranging from 1200 to 2700 RPM, corresponding to frequencies between 19 Hz and 44 Hz. Based on the analysis, the following conclusions were drawn:

- As expected, total vibration levels (RMS) increase with RPMs, fluctuating with engine speed. This trend indicates that higher RPMs generate more torque, leading to greater vibrations.
- The FEA simulation model predicted the highest acceleration at 34 Hz, closely aligning with experimental results, though slightly higher. It also identified a frequency range (35–39 Hz) where vibrations concavity, suggesting the presence of a resonance point or damping effect in the engine mount reducing vibration transfer.
- Engine mounts effectively minimize vibrations, with most low vibration levels recorded on the passenger seat side (typically below 1.0), contributing to improved ride comfort.
- While the engine mount is effective at absorbing vibrations within a certain RPM range, sustained operation at high RPMs exceeds its capacity, necessitating design modifications to handle such conditions.
- According to ISO 10816-6, the permissible acceleration limit for this machine category (Class 1, up to 8 kW output) is 27.9 mm/s^2. Measured acceleration values ranged from 18 mm/s^2 to over 77 mm/s^2 at various frequencies, with some exceeding the standard. This suggests the need to either restrict engine speed to 1500 RPM or redesign the mount to support long-term operation at higher speeds.

5.2. Recommendation

The mount can effectively damp the vibrations, but its effectiveness is limited within the RPM range. Operating at a high RPM for an extended period of time will inevitably push you beyond what is feasible without requiring design modifications.

Based on the findings, the following recommendations are proposed to address the vibration challenges and enhance the performance of the engine mount system:

- ✓ To ensure compliance with ISO 10816-6 standards and prevent excessive vibrations, limit the engine's operational speed to a maximum of 1500 RPM. This will help reduce the risk of exceeding permissible acceleration levels.
- ✓ Consider redesigning the engine mount to handle higher RPMs effectively. This could involve use of advanced materials with higher damping properties to absorb vibrations more efficiently. Adjust the mount's geometry to optimize load distribution and minimize resonance effects.
- ✓ Conduct additional experimental testing across a broader frequency range to validate and refine the FEA simulation model. This will help align simulated predictions with real-world performance.
- ✓ Address the identified resonance point around 34 Hz by incorporating tuned mass dampers or adjusting the stiffness and damping characteristics of the mount. This will reduce vibration transmission at critical frequencies.
- ✓ Implement a routine inspection and maintenance schedule to ensure the engine mount remains in optimal condition and continues to perform as designed over time.

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Appendix

A) The spreadsheet for the experimental results

The output of the measurement was recorded with the format indicated in the table.

Measurement at point 1

S.No.	Frequency (RPM/Minute)	ax	ay	az	RMS
1	20				
2	25				
3	30				
4	35				
5	40				
6	45				
7	*50*				

Measurement at point 2

S.No.	Frequency (RPM/Minute)	ax	ay	az	RMS
1	20				
2	25				
3	30				
4	35				
5	40				
6	45				
7	*50*				

The output of the measurement was recorded with the format indicated in the table.

Lab1

Measurement at point 1

S.NO	*RPM*	Frequency (RPM/Minute)	*Rms1*	*Rms2*	*Rms3*	*RMS*
1	*1200*	20	*8.378*	*10.149*	*8.736*	*9.088*
2	*1500*	25	*10.250*	*14.083*	*13.505*	*12.613*
3	*1800*	30	*16.576*	*17.458*	*17.345*	*17.1263*
4	*2100*	35	*17.846*	*19.315*	*18.630*	*18.664*
5	*2400*	40	*19.254*	*17.030*	*19.008*	*18.4307*
6	*3000*	45	*18.646*	*17.815*	*20.554*	*19.005*

Measurement at point 2

S.NO	*RPM*	Frequency (RPM/Minute)	*Rms1*	*Rms2*	*Rms3*	*RMS*
1	*1200*	20	*11.528*	*7.912*	*9.378*	*9.606*
2	*1500*	25	*17.999*	*15.445*	*15.191*	*16.2127*
3	*1800*	30	*26.334*	*25.726*	*24.025*	*25.362*
4	*2100*	35	*20.667*	*20.146*	*20.316*	*20.37633*
5	*2400*	40	*20.754*	*20.978*	*21.378*	*21.0366*
6	*30*	50	*22.332*	*22.024*	*20.00*	*21.452*

Measurement at point 3

S.NO	*RPM*	Frequency (RPM/Minute)	*Rms1*	*Rms2*	*Rms3*	*RMS*
1	*1200*	20	*10.400*	*10.373*	*9.669*	*10.14733*
2	*1500*	25	*15.560*	*16.991*	*17.707*	*16.753*
3	*1800*	30	*22.955*	*22.878*	*22.463*	*22.762*
4	*2100*	35	*19.654*	*18.754*	*19.323*	*19.24367*
5	*2400*	40	*20.431*	*17.453*	*18.076*	*18.65333*
6	*3000*	45	*19.985*	*19.138*	*18.415*	*19.179333*

Lab2

Measurement at Engine mount1

S.NO	*RPM*	Frequency (RPM/Minute)	*ax*	*ay*	*az*	*RMS*
1	*1200*	20	*19.769*	*17.961*	*18.765*	*18.83166*
2	*1500*	25	*44.818*	*22.555*	*25.210*	*30.861*
3	*1800*	30	*40.993*	*36.9915*	*35.930*	*37.9715*
4	*2100*	35	*25.025*	*27.826*	*26.695*	*26.51533*
5	*2400*	40	*35.376*	*36.284*	*36.107*	*35.92233*
6	*3000*	50	*44.810*	*56.022*	*38.992*	*46.608*

Measurement at Engine mount2

S.NO	*RPM*	Frequency (RPM/Minute)	*ax*	*ay*	*az*	*RMS*
1	*1200*	20	*36.299*	*38.061*	*39.693*	*38.017667*
2	*1500*	25	*40.308*	*43.009*	*45.603*	*42.97333*
3	*1800*	30	*49.881*	*55.068*	*49.881*	*51.61*
4	*2100*	35	*61.871*	*68.058*	*67.327*	*65.752*
5	*2400*	40	*41.752*	*42.548*	*46.188*	*43.496*
6	*3000*	50	*69.758*	*65.534*	*76.854*	*70.71533*

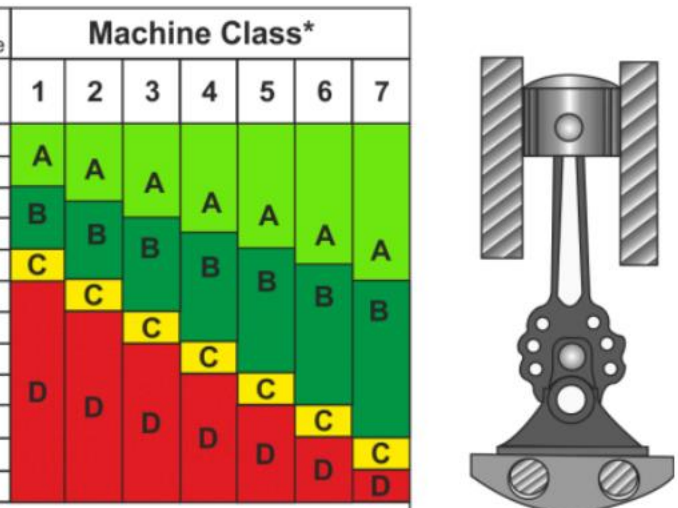
Measurement on passenger

S.NO	RPM	Frequency (RPM/Minute)	ax	ay	az	RMS
1	*1200*	20	*0.775*	*0.742*	*0.771*	*0.74233*
2	*1500*	25	*0.779*	*0.778*	*0.801*	*0.786*
3	*1800*	30	*0.7906*	*0.7902*	*0.891*	*0.82394*
4	*2100*	35	*0.802*	*0.81*	*0.82*	*0.81066*
5	*2400*	40	*0.856*	*0.910*	*0.816*	*0.86066*
6	*2700*	45	*0.867*	*0.8601*	*0.823*	*0.850033*

B) ISO10816-6 vibration standards

Vibration Evaluation Standard - Reciprocating machine

Vibration Severity Grade	Overall Vibration measurment measured on the machine Structure: Displacement in um - micron (rms)	Velocity in mm/ sec (rms)	Acceleration meter/ sec (rms)
1.1	≤ 17.8	≤ 1.12	≤ 1.76
1.8	≤ 28.3	≤ 1.78	≤ 2.79
2.8	≤ 44.8	≤ 2.82	≤ 4.42
4.5	≤ 71.0	≤ 4.46	≤ 7.01
7.1	≤ 113	≤ 7.07	≤ 11.1
11	≤ 178	≤ 11.2	≤ 17.6
18	≤ 283	≤ 17.8	≤ 27.9
28	≤ 448	≤ 28.2	≤ 44.2
45	≤ 710	≤ 44.6	≤ 70.1
71	≤ 1125	≤ 70.7	≤ 111
112	≤ 1784	≤ 112	≤ 176
180	≤ 1784	> 112	> 176



Zone A: Vibration of newly Commissioned Machines;
Zone B: Machines considered acceptable for unrestricted long-term operation
Zone C: Machines considered unsatisfactory for long-term continuous operation
Zone D: Vibration values normally considered to be sufficient severity to cause damage to the machine

Fourier Transforms

The Fourier transform is a mathematical formula that relates a signal sampled in time or space to the same signal sampled in frequency. In signal processing, the Fourier transform can reveal important characteristics of a signal, namely, its frequency components.

The Fourier transform is defined for a vector x with n uniformly sampled points by

$$y_{k+1} = \sum_{j=0}^{n-1} \omega^{jk} x_{j+1}.$$

$\omega = e^{-2\pi i/n}$ is one of *n* complex roots of unity where *i* is the imaginary unit. For *x* and *y*, the indices *j* and *k* range from 0 to $n-1$.

The fft function in MATLAB® uses a fast Fourier transform algorithm to compute the Fourier transform of data. Consider a sinusoidal signal x that is a function of time t with frequency components of 20 Hz and 50 Hz. Use a time vector sampled in increments of $\frac{1}{50}$ of a second over a period of 10 seconds.

```
Ts = 1/50;
t = 0:Ts:10-Ts;
x = 0.9317+sin(2*pi*20*t) + sin(2*pi*50*t);
plot(t,x)
xlabel('Time (seconds)')
ylabel('Acceleration (m/s^2)')
title('Vibration of a Force Bajaj on the Engine mount')
```

Compute the Fourier transform of the signal, and create the vector f that corresponds to the signal's sampling in frequency space.

```
y = fft(x);
fs = 1/Ts;
f = (0:length(y)-1)*fs/length(y);
```

When you plot the magnitude of the signal as a function of frequency, the spikes in magnitude correspond to the signal's frequency components of 20 Hz and 50 Hz.

```
plot(f,abs(y))
xlabel('Frequency (Hz)')
ylabel('Acceleration, m/s^2')
title('Vibration of a Force Bajaj on the Engine ')
```

The transform also produces a mirror copy of the spikes, which correspond to the signal's negative frequencies. To better visualize this periodicity, you can use the fftshift function, which performs a zero-centered, circular shift on the transform.

```
n = length(x);
fshift = (-n/2:n/2-1)*(fs/n);
yshift = fftshift(y);
plot(fshift,abs(yshift))
xlabel('Frequency (Hz)')
ylabel('Acceleration')
title('Vibration of a Force Bajaj on the Passanger')
```