ENGINE NOISE, VIBRATION, AND HARSHNESS ANALYSIS AT THE DESIGN CONCEPT STAGE

M.Tech. Thesis

By

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DISCIPLINE OF MECHANICAL ENGINEERING

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A THESIS

Submitted in partial fulfillment of the

requirements for the award of the degree

of

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by

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CANDIDATE'S DECLARATION

I hereby certify that the work which is being presented in the thesis entitled **ENGINE NOISE, VIBRATION AND HARSHNESS ANALYSIS AT DESIGN CONCEPT STAGE** in the partial fulfillment of the requirements for the award of the degree of **MASTER OF TECHNOLOGY** and submitted in the **DISCIPLINE OF MECHANICAL ENGINEERING, Indian Institute of Technology Indore**, is an authentic record of my own work carried out during the time period from AUGUST 2022 to MAY 2024 under the supervision of Prof. Pavan Kumar Kankar, System Dynamics Lab, Department of Mechanical Engineering, Indian Institute of Technology Indore and Ashok Kumar Patidar, Deputy General Manager, VE Commercial Vehicles Limited, Pithampur, Indore.

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

25/05/2024

(BIMAL BASHYAL)

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

25/05/2024 25/05/2024

(Prof. Pavan Kumar Kankar) **(Ashok Kumar Patidar)**

BIMAL BASHYAL has successfully given his M.Tech. Oral Examination held on **30th May 2024**.

31/05/2024

Signature(s) of Supervisor(s) of M.Tech. thesis Convener, DPGC

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With Regards,

Bimal Bashyal

ABSTRACT

Vibration and noise are more common challenges in the automobile industry that arise from moving parts and gas pressure within engines. Excessive noise and vibration in an engine have major drawbacks which lead to premature component wear, reduced performance, environmental impact, and a negative image of the vehicle's quality on the customer perception.

This research work was carried out for the prediction of vibration-based noise levels in a complete engine assembly during the design concept stage. Its uniqueness lies in its capacity to bypass the physical measurements for noise and vibration which allows for a comprehensive study of vibration and noise at the early conceptual stage that saves substantial time and cost. The methodology applied in this research work involves the detection and optimization of engine components for their potential for vibration and noise during their design phase by examining the complete engine assembly. This approach is beneficial in mitigating vibrations and enhancing overall component performance in the domain of noise.

The process involves the preparation of finite element models for each engine component using Altair Simlab, and assembling them to form a complete engine model. Into that assembled model excitation is applied via the measured data. Modal and harmonic response analyses were performed and experimentally verified. The results of this analysis were further used to calculate the noise generated from each component.

Individual components responsible for noise generation are topologically optimized by changing their geometry at intense locations. The modified component was assembled back to the initial engine assembly which resulted in the reduction of the engine-generated noise.

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NOMENCLATURE

$[M]$	Mass matrix
$[C]$	Damping matrix
$[K]$	Stiffness matrix
x	Distance
$\dot{x}$	Velocity
$\ddot{x}$	Acceleration
F	Force
u	displacement along the x-axis (global coordinate)
v	displacement along the y-axis (global coordinate)
w	displacement along the z-axis (global coordinate)
r	displacement along the x-axis (local coordinate)
s	displacement along the y-axis (local coordinate)
t	displacement along the z-axis (local coordinate)
ω	angular frequency
n	engine order
X	amplitude of response
Φ	phase angle between the excitation and response

ACRONYMS

AC	Air Conditioner
APDL	Ansys Parametric Design Language
AT	Anti Thrust
ATV	Acoustic Transfer Vector
BDC	Bottom Dead Centre
CAD	Computer Aided Design
CAE	Computer-Aided Engineering
CFD	Computational Fluid Dynamics
CG	Centre of Gravity
CMS	Component Mode Synthesis
DOF	Degree Of Freedom
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Model
FFT	Fast Fourier Transform
FRF	Frequency Response Function
MBA	Multi-body Analysis
MBD	Multi-body Dynamics
MBS	Multi Body System
MDOF	Master degree of freedom
MPC	Multi Point Constraint
NVH	Noise Vibration and Harshness
RAM	Random Access Memory
RBE	Rigid Body Element
RPM	Revolution per Minute
SE	Super Element
TDC	Top Dead Centre

Chapter 1: Introduction

1.1 Overview

Vibration and noise are more typical difficulties in the automotive sector, caused by moving parts and gas pressure within engines. Noise and vibration originating from a vehicle are emanating as an important challenge for the automobile industry. The engine is a major source that contributes for annoying noise perceived within a vehicle. Excessive vibrations may have drawbacks that cause wear of parts and damage the frame and engine mountings due to stress, the consequences of which may be premature failure and a reduction in the service life of the product. Effective NVH development is crucial for optimizing the performance, reliability, and user experience of automotive vehicles.

As demonstrated by studies showing significant improvements in vehicle quality and comfort through NVH analysis in internal combustion engines. To achieve the optimum NVH level for an engine, special attention must be given during the early power-train development process. Objectives must be clearly defined such as the determination of natural frequencies, vibration intensity, and noise emission.

In the field of automobiles, the frequency spectrum is divided into two major ranges: 0 to 30 Hertz, identified as the ride frequency range that includes a secondary ride domain too, and the 30 to 500 Hertz range which is termed as harshness domain that affects the passenger comfort and can induce structure-borne noise.[1]

For NVH development to be effective, a planned integration procedure is needed. CAE calculations must be used right from the beginning, at the concept definition phase of the process. The applicable instruments must be able to replicate the whole noise-generating chain which starts from combustion and concludes with radiated noise. Similarly, an engine component can also be assessed and optimized for its NVH behavior. [2]

1.2 Computer Aided Engineering (CAE)

Computer-aided engineering refers to the implication of current modern tools and software for conducting engineering analysis in various domains. It includes methodologies like finite element analysis (FEA), multi-body dynamics (MBD), and computational fluid dynamics (CFD. CAE can be used for analyzing multiple domains to determine the durability and performance of the components. It incorporates simulation, validation, and optimization of any components used in engineering applications. It incorporates the following:

1. Stress analysis of any components or assemblies.
2. Fluid flow analysis including thermal flow analysis, with the help of CFD.
3. Multi-body dynamics and kinematics calculation and simulation.
4. Optimization of any parts or products based on the design requirements.

1.2.1 CAE in the automotive industry

CAE tools are extensively utilized in the automotive industry, significantly transforming the way vehicles are designed and developed. CAE tools have enabled automakers to cut down on product development costs and time while improving vehicle safety, comfort, and durability. With their advanced predictive capabilities, CAE tools have largely replaced traditional physical prototype testing with sophisticated computer simulations for design verification. This shift not only speeds up the development process but also allows for more thorough testing under a variety of conditions. The dependability of CAE tools hinges on the accurate consideration and input of numerous parameters, ensuring that the simulations closely replicate real-world scenarios.

1.2.2 Finite Element Analysis (FEM)

The finite element method is a numerical analysis technique used to obtain approximate solutions for a wide range of engineering problems. Its versatility and flexibility make it a popular analysis tool across nearly every industry. Increasingly, engineering scenarios require approximate solutions that closely match exact solutions to accurately predict the behavior of components. A computer model of a material or design is created, loaded, and analyzed to generate specific results. The main concept involves performing calculations at a finite number of points and then interpolating these results across the entire domain, whether it be a surface or volume.

1.3 Engine Noise and Vibration History

Vibration results from engine excitation caused by cylinder pressure, while noise arises because of this vibration, impacting both the lifespan of the vehicle and the comfort of its occupants.

Reason for analysis

1. Lightly damped structures may exhibit significant vibration levels even from minor sources if disturbance frequencies align closely with the system's natural frequencies. [3]
2. Even low-level disturbances can pose challenges when integrated into a vehicle.
3. To mitigate these issues, it is important to accurately model the system and assess its response to probable disturbances during the planning phase.

Structure Optimization for reducing noise and vibration.

Improving the structural attenuation of the power will reflect a reduction in the noise radiation properties of the powertrain through usual methods. [4]

1.4 Excitation Mechanisms

The primary sources of noise, vibration, and harshness (NVH) in an engine typically fall into several categories: the force generated during combustion; the reaction forces on main bearings, influenced by factors like mass, damping, and flywheel movement, adjustments made by the damper at the front end; lateral forces on pistons; forces on camshaft bearings; impacts during valve opening and valve closing; forces on the valve train from movements of chains, belts, or gears; forces within the gear train of the transmission; and reaction forces and moments within the drivetrain.

To ensure accurate predictive calculations of noise, vibration, and harshness (NVH), it's crucial to thoroughly examine the behavior of the entire engine, including all accessories, and consider all relevant excitation mechanisms. This entails having a simulation model that encompasses the dynamic characteristics of all rotating components along with the dynamic behavior of housings.

Combining the Finite Element Method (FEM) with Multi-body Analysis (MBA) enables the calculation of engine excitation. FEA models simulate and analyze the behavior of vibrations in all the individual components, while the MBA model simulates the overall movement of all dynamic components in the system.

1.5 Novelty and Scope of the project

- The project involves a detailed analysis of all engine components when assembled and their interactions with each other, and their impact on the overall vehicle NVH characteristics.
- This project includes an examination of each component and recognizing their potential to generate noise and vibrations.

1.6 Thesis Outline

The research work is structured under this thesis and consists of six chapters:

Chapter 2. Literature review: Significant advancement has been made over time in the field of NVH. In the past NVH was considered secondary, but along with the advancement of CAE tools and advanced simulation techniques it emerged to predict the issues at an early stage. Although there has been development in this field, there are a few gaps such as integration of excitation to simulate the running engine. This thesis works upon those gaps by developing FE model for all the engine components.

Chapter 3. Methodology: Comprehensive modeling and simulation setup of an engine has been explained along with its assumptions. Various types of mesh and elements used are described, and the process of development of FE model by using software like Altair simlab and APDL has been explained. Likewise, the assembly of those FE model for simulating the complete engine and the application of excitation onto this assembly has been explained which simulates the engine NVH characteristics.

Chapter 4. Results: After the assembly of all FE model to form a complete engine, modal and harmonic analysis has been performed in this model for paving its path towards NVH analysis. The modal frequency of all components is compiled on a table to calculate the modal participation factor. Some of the dominant mode shapes are visualized on a wireframe model, which is later visualized on an actual model. Similarly, the modal frequency for engine mounts and bending modes of an engine was verified. Response analysis was done to depict the variation in frequency of vibration over the surface. Likewise, simulated results of vibration from the oil pan were experimentally correlated with an acceptable range of minor variations.

Chapter 5. Design Optimization: In this chapter to improve the noise and vibration levels of an oil pan, topological optimization process has been followed. The results of various designs is compared with each other and the best feasible design is finalized for achieving noise reduction. This methodology has reduced the noise level of an oil pan by about 5 dB.

Chapter 6. Conclusion and Future Scope: This research work has provided a comprehensive methodology to simulate the vibration and noise level emanating from any arbitrary location of an engine and has been experimentally verified. On the bright side, it can eliminate the need for a physical engine to improve its NVH characteristics and rapidly analyze the change in NVH characteristics due to design changes. In the future, this research work can be further extended by incorporating dynamic effects from other subsystems like air-compressors, AC-compressors, etc. so that more accuracy can be achieved.

Chapter 2: Literature Review

2.1 Introduction

Optimal noise, vibration, and harshness (NVH) qualities in automobiles are critical for their performance, durability, and providing passenger comfort. There have been significant advancements in this domain over the decades which have been driven by the need for quieter and smoother rides. Noise, Vibration & Harshness (NVH) in the automotive industry were often considered secondary to other design priorities till the mid-20th century and it got its importance as consumer expectations for comfort rose, and multiple innovations came into application such as improved insulation and suspension systems.

CAE tools were adopted for modeling and simulating the NVH characteristics in the late 20th century. In the 21st century, with the advancements in simulation tools, prediction and assessment of NVH issues became achievable in the early design stages. This accelerated the engine development procedures because it somehow helped to eradicate the reliance on physical engines for NVH testing.

2.2 Background

There has been development of engine NVH by different researchers by considering different aspects involved and it is briefly described in chronological order as below:

In the 1960s, the framework for current NVH analysis was established with the pioneering work of Hurty[5], who introduced the concept of mode synthesis theory. He found out that large structural systems with complex assembly could be broken down into smaller, manageable substructures and each of them can be analyzed individually in prior to being assembled for understanding the full system's dynamic behavior. This approach was further refined by Craig in 1995[6], who developed methods that can reduce the interface degrees of freedom, enhancing the application of mode synthesis in undamped systems.

Moving into the late 1980s, Agrawal and Kumar[7] came up with the new superelement models for the dynamics analysis of constrained multi-body systems. These models have reduced computational dimensions without sacrificing its accuracy and has significantly sped up the simulation process. By the early 2010s, Lu and colleagues[8] has extended this method to vehicle body dynamics simulation, which demonstrated how superelement modeling could balance computational efficiency and precision which was crucial for optimizing vehicle body structures.

In parallel, the automotive industry continued to evolve its NVH analysis techniques. Liu, in 2008[9]emphasized the dominance of structure-borne noise at low frequencies in three-cylinder passenger vehicles, pointing out the limitations of traditional Fourier transform methods for analyzing the non-stationary signals typical of internal combustion engine vibrations. Instead, he found out that time-frequency distribution techniques are better for handling such kinds of signals.

Throughout the 2000s, the importance of optimizing engine structures to reduce noise became more and more evident. Abbes et al. [10] in 2004 highlighted the importance of optimum engine support for preventing unwanted vibrations from being transferred to the supporting structures, while Kumar's[11] work found out the significance of minimizing noise radiation in frequency ranges which are associated with combustion excitations.

Noise mapping, acoustic holography, and modal analysis are some experimental approaches that have developed significantly during the 2010s. Researchers such as Parizet and colleagues[12] have confirmed effectiveness of these methods of identifying NVH sources and enhancing vehicle design by improving its NVH characteristics. Around the same time, Zhang Junhong and Han Ju[13] showed the importance of computer-aided engineering (CAE) processes for simulating and optimizing engine noise and vibration up to 1 KHz.

Piperpaolo and his team[14] investigated the oil pan as a major contributor to engine noise radiation in 2011. They utilized acoustic transfer vectors (ATVs) to evaluate and analyze radiated acoustic power efficiently across different loading conditions which improved our understanding of how thin-walled structures influence powertrain noise.

The study of NVH issues in automobile engineering has mase substantial over the years. Maria Antonietta Panzaa [15] in 2015 published a comprehensive evaluation of experimental techniques for NVH analysis in commercial vehicles that emphasized the importance of accurate tests for identifying noise and vibration problems. Based on this, He Zhenpeng et al. [16] in 2016 studied the impact of several interaction models between the main bearing and crankshaft of an engine on NVH, emphasizing the role of lubrication models in predicting vibration and noise. Jiachi Yao et al.[17] in 2019 focused on identifying and differentiating noise sources in internal combustion engines that presented methods for analyzing combustion noise and piston slap noise. Xu Zheng et al. [18] in 2020 introduced a coupled engine dynamic model that included the valve train system which demonstrated superior accuracy in vibration simulation and noise prediction. Lastly, Enrico Armentani et al. [19] in 2020 employed topology optimization for reducing engine bracket weight and increasing its natural frequency showcasing the importance of effective design modifications for NVH improvement.

The literature reviewed for this thesis showcases significant advancements in the field of NVH analysis for internal combustion engines that range from novel interaction models to advanced simulation techniques and their experimental methodologies. These studies have collectively contributed to and guided the ongoing efforts to improve the

NVH characteristics of automobile engines which leads to enhanced performance of an engine.

2.3 Research Objectives

There have been significant advancements in NVH analysis and optimization but there remain several key areas for which further research is needed to develop NVH capabilities that can incorporate entire engine systems, including transmission too. One specific gap is the integration and application of excitation from critical engine subsystems such as the gear train, valve train, and crank train. Current methodologies only focus on individual components and use complicated multi-dimensional models that can be computationally intensive and less efficient while simulating a complete engine.

Hence, looking after these research gaps of NVH of an engine, this thesis is developed to address the research gap of developing a complete engine with the incorporation of transmission systems by developing FE model for each component with vibration excitation from the crank train, the valve train, and the gear train.

The objectives are summarized below:

1. To comprehensively understand, analyze, and mitigate noise, vibration, and harshness originating from all aspects of the engine's operation.
2. To perform full engine dynamic analysis to study the complete behavior of the engine on vibration, durability, and noise.
3. Build and validate designs of an engine component which reduces noise and vibration levels.
4. Explore design changes for the engine so that its low-frequency noise and vibration levels will be reduced as compared to the current production mode.

Chapter 3: Methodology

3.1 Overview

Since the engine is composed of multiple parts working harmonically to achieve a common goal of producing power. Each component has its own specific sets of functions, and they are all connected with the help of bolts to form a complete engine. Analyzing the complete engine at once is a cumbersome process, so we go through a specific set of approaches to model the complete engine for its NVH analysis. Its schematic is Figure 3.1 below:

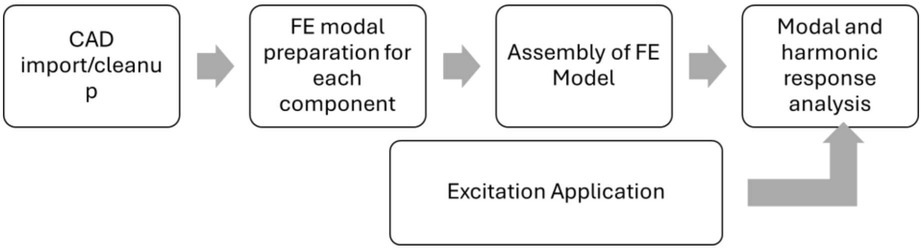

Figure 3.1Schematic approach to model the engine for NVH analysis.

In the first step, complete engine CAD is taken. Each of the components is isolated from the assembly to prepare their Finite Element (FE) model. It is obvious that we go through certain assumptions while modeling the engine components, and it is listed below:

1. **Bolts:** defined as 1D using point mass and beam element. (RBE2 at bolt head)
2. **Fluid mass**: defining the mass point at CG of fluid attached to the surface with a flexible connection. (RBE3)
3. **Gaskets**: neglected, gaps for it will be considered while defining interface nodes.
4. **Flexible members (Pipes, hoses, bellows)**: modeled as a point mass, half the mass is added at each connection point.
5. **Rotating parts**: not modeled.
6. **Internal parts:** mass and CG are calculated, and point mass is defined at the CG.

After the FE model preparation of each component, its reduced order model is generated, which is commonly known as the ‘Super element’ which is a condensed form of the FE model that is composed of modal frequencies and the associated mass & stiffness matrices. The master node at the location of bolt joints is predefined in the FE model, which helps to maintain connectivity and force transfer among the assembled components.

After the FE model of each of the components is developed, they are assembled all together through the predefined master nodes in each of the condensed models. This method is very helpful in building the engine model piece by piece. In addition, it also gives us the flexibility to replace any optimized model of certain components in the main assembly. For example, if we want to replace the air compressor, it will be done by attaching the new air compressor at the predefined attaching point (master nodes*). It generally helps to save computational time while running large models consisting of multiple parts, each with thousands of elements. For the generation of the FE model, Altair Simlab is used as it provides more flexibility for 3D mesh generation. Similarly, for the generation and assembly of FE model, Ansys Parametric Design Language (APDL) is used.

3.2 Simulation Setup

3.2.1 Mesh

For the discretization of the complete engine assembly, individual components are meshed separately. For meshing, the following types of elements are used:

3.2.1.1 SOLID 187: 3D 10-node tetrahedral structure solid

All the solid bodies, having irregular shapes, of an engine component are modeled using a SOLID 187 element, as it can capture the curvatures efficiently. A schematic diagram is given in Figure 3.2 and shape function of this element are given below.

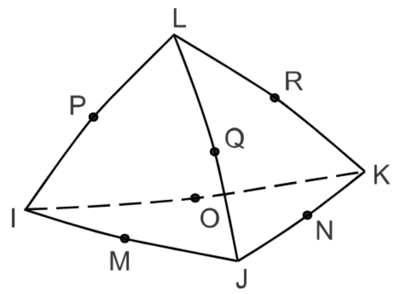

Figure 3.2 10-noded tetrahedral element.

Effective shape function:

It describes how the displacement field varies within an element in relation to the nodal displacements. It interpolates the displacement within an element based on the displacement of the nodes.

$$u = u_I(2L_1 - 1)L_1 + u_J(2L_2 - 1)L_2 + u_K(2L_3 - 1)L_3 + u_L(2L_4 - 1)L_4 + 4u_M L_1 L_2 + 4u_N L_2 L_3 + 4u_O L_1 L_3 + 4u_P L_1 L_4 + 4u_Q L_2 L_4 + 4u_R L_3 L_4$$

$$v = v_I(2L_1 - 1)L_1 + \ldots (analogous\ to\ u)$$

$$w = w_I(2L_1 - 1)L_1 + \ldots (analogous\ to\ u)$$

3.2.1.2 SHELL 181: 4-noded Shell

Bodies that are made from sheet metal such as oil-pan and pipes are modeled using SHELL 181 elements, as it offers flexibility in modeling thin-walled structures with relatively large dimensions. A schematic diagram is given in Figure 3.3 and shape function of this element are given below.

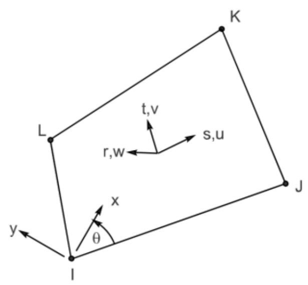

Figure 3.3 4-noded Shell element.

Effective shape function:

$$u = \frac{1}{4}(u_I(1-s)(1-t) + u_J(1+s)(1-t) + u_K(1+s)(1+t) + u_L(1-s)(1+t)$$

$$v = \frac{1}{4}(v_I(1-s)\ldots(analogous\ to\ u)$$

$$w = \frac{1}{4}(w_I(1-s)\ldots(analogous\ to\ u)$$

3.2.1.3 SOLID 185: 3D 8-noded Structural solid

Bodies that are perfectly symmetrical about an axis (main bearing sleeves, liners, etc.) are modeled by using this SOLID 185 element, as it can capture its cylindricity more perfectly. A schematic diagram is given in Figure 3.4 and the shape function of this element is given below.

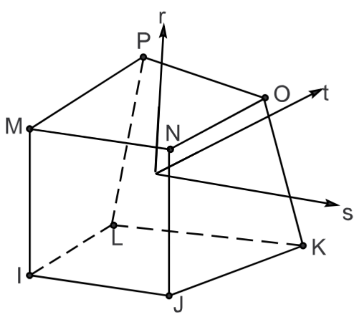

Figure 3.4 8-noded Solid element.

Effective shape function:

$$u = \frac{1}{8}(u_I(1-s)(1-t)(1-r) + u_J(1+s)(1-t)(1-r) + u_K(1+s)(1+t)(1-r) + u_L(1-s)(1+t)(1-r) + u_M(1-s)(1-t)(1+r) + u_N(1+s)(1-t)(1+r) + u_O(1+s)(1+t)(1+r) + u_P(1-s)(1+t)(1+r)$$

$$v = \frac{1}{8}(v_I(1-s)\ldots\ (analogous\ to\ u)$$

$$w = \frac{1}{8}(w_I(1-s)\ldots\ (analogous\ to\ u)$$

3.2.2 Bolts

As in the engine assembly, each of the components is joined together with the help of bolts. To make the simulation simpler and faster, all the bolts are modeled using the 1D formulation by using the BEAM 188 element. A schematic diagram is given in Figure 3.5 and the shape function of this element is given below:

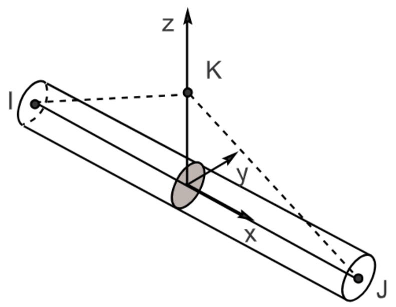

Figure 3.5 2-noded beam element.

Effective shape function:

$$u = \frac{1}{2}(u_I(1-s) + u_J(1+s)$$

$$v = \frac{1}{2}(v_I(1-s) + v_J(1+s)$$

$$w = \frac{1}{2}(w_I(1-s) + w_J(1+s)$$

This BEAM 188 element defines the bolt shank but for modeling of the thread portion and washer area Rigid Body Element 2 (RBE2): kinematic coupling is used, which rigidly connects all the nodes on that region (dependent node) to one central independent node. This central independent node is connected to the BEAM 188 element. For the washer area, there is only one spider web-like RBE2 element on the washer. Similarly, for the thread section, there are multiple spider web-like RBE2 elements along the thread portion.

3.2.3 Fluid mass

For fluid mass such as lubricating oil from an oil pan is modeled by defining point mass at the CG of the fluid. This CG is connected to the contact surface by using the RBE3 element (Distributed coupling), as it provides a distributed connection, which does not influence the local stiffness of the model. Similarly, fluids inside the fuel filter are also modeled.

3.2.4 Contact

For defining the bodies that are in surface-to-surface contact, such as in between liner and cylinder block, block and main bearing cap are defined using CONTACT174 and TARGET170 elements. The schematic contact surface is given in Figure 3.6 below.

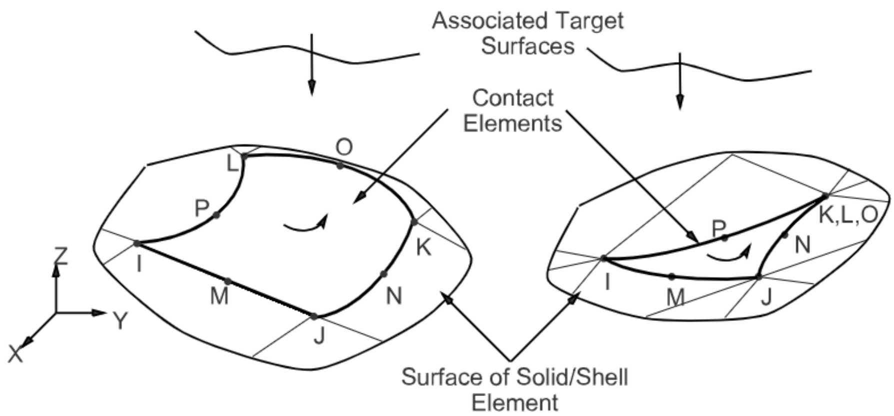

Figure 3.6 Contact Surfaces.

The following variables in Table 3.1 are defined for contact:

Table 3.1 Parameters for contact surface definition.

Contact Algorithm (KEYOPT2)	Multipoint Constraint (MPC)

Initial Penetration/Gap (KEYOPT9)	Excluded
Behavior of contact surface (KEYOPT12)	Bonded (always)

In addition, it excludes any kind of penetration/gap. Also, for MPC, only one equilibrium iteration is required to solve the system of equations.

3.3 Use of SIMLAB for developing FE model

An example showing the preparation of the FE model for the Engine Control Unit (ECU) is described schematically in Figure 3.7 below:

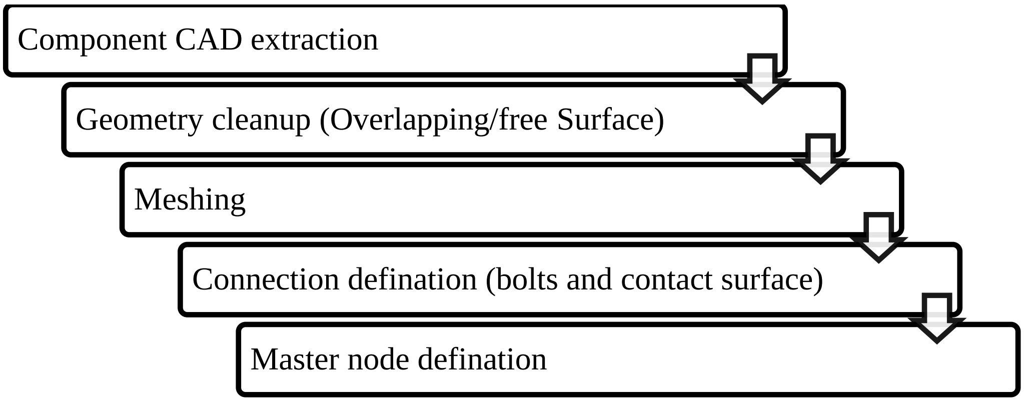

Figure 3.7 Schematic approach to generate FE model.

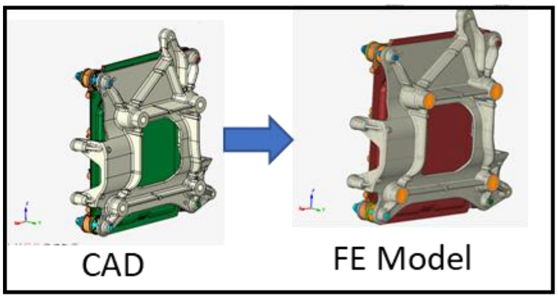

Figure 3.8 Imported CAD model of ECU and its FE model.

Fig 3.8 shows the imported and cleaned CAD model of the ECU bracket which is composed of multiple parts. Each of the parts is meshed separately and the connection for the bolts, contact surface, and spring damper is modeled, which is described below:

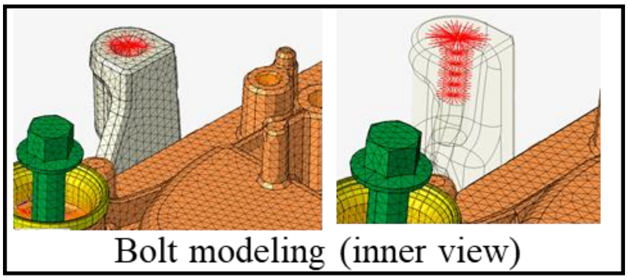

Figure 3.9 Bolt modeling for FE model.

Here, in Figure 3.9 the Physical bolt is modeled as the 1D bolt which maintains the connectivity among the components and is used for the transfer of excitation force all along the engine assembly. Bolt is defined using Spider Element using the Rigid Body Element (RBE2) which rigidly connects the nodes in all 6 directions (3-translational & 3-rotational). Here in a transparent view, we can see different sections of spider elements. Each of them is connected by the help of a bar element, which has the properties of the physical bolt. Sections are defined for assuming the force transmission throughout the thread of the bolt.

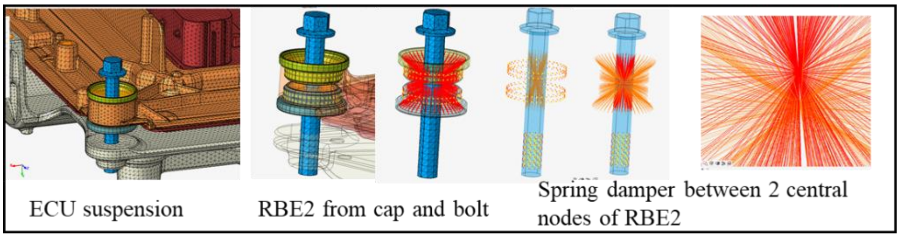

Figure 3.10 Isolation System modeling for ECU

Similarly, ECU has a small isolation system. One bracket is attached to the engine block and over it ECU is isolated using the rubber damper. So, for modeling the rubber isolator a physical bolt is taken onto which the isolator is modeled. Like the bolt modeling, the upper and lower cap of the isolator are connected to one central/independent node with RBE2. Similarly, on the other side to isolate it, another RBE2 with a different central node is created on the bolt itself. Now the spring damper is modeled between the two central/independent nodes of the RBE2 as represented by Figure 3.10.

There are many contact surfaces in the components. Target elements and contact elements are selected by selecting the associated faces. The contact algorithm was set to multi-point constraint (MPC) which is always bonded with the exclusion of any penetration/gap.

For the master node inclusion, the bolts through which it will relate to other components will be noted and an interface node (master node) will be created there. Similarly, the vibration measurement location, master node is created. It is indicated by the red node in the Figure 3.11 given below:

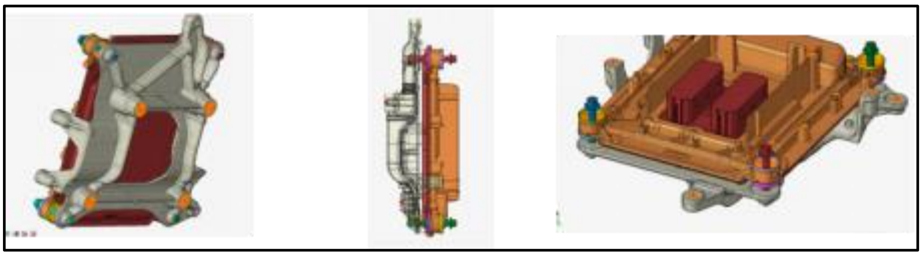

Figure 3.11 Master nodes in the FE model (indicated by red nodes)

Similarly, the FE mode of all the components is prepared separately and exported as a **.cdb** file and. After modeling all the components, all the FE model are assembled to form a full engine model.

3.4 Excitation

Internal combustion engines experience two primary dynamic disturbances: pulses of firing from fuel explosions in the cylinder blocks, along with inertia forces and torque oscillations originating from the movement of rotating and reciprocating components such as the piston, connecting rod, and crankshaft.

To apply the excitation, vibration is measured at various locations within the engines, and it is applied directly to the model.

3.5 Harmonic Response Analysis

In the context of an engine, harmonic analysis refers to the process of studying the vibration and oscillation patterns within the engine system, particularly concerning the reciprocating and rotating components. It involves analyzing the periodic forces and motions generated by the engine's operation, such as piston movement, crankshaft rotation, valve opening and closing, and combustion events. Harmonic analysis is essential for understanding the dynamic behavior of engine systems, including vibration patterns, forces, and motions. By analyzing the frequency content and resonance phenomena, one can optimize engine design, performance, reliability, and NVH issues.

3.5.1 Application of harmonic analysis to an engine:

1. **Vibration Analysis**: Engines produce various vibrations during their operation due to the reciprocating and rotating motion of components like pistons, connecting rods, crankshafts, and camshafts. Harmonic analysis helps in understanding the frequency content of these vibrations, identifying the dominant frequencies, and assessing their impact on engine performance, durability, and comfort.

2. **Forced Vibrations:** Harmonic analysis helps in studying forced vibrations induced by external factors such as imbalances, misalignments, mechanical wear, and combustion forces. These external forces can excite resonant frequencies within the engine system, leading to increased vibration levels, noise, and potential structural damage.

3. **Noise and Vibration Control:** By identifying the dominant frequencies and vibration modes, harmonic analysis assists in designing effective noise and vibration control measures such as

damping systems, isolation mounts, and structural modifications to reduce noise and improve ride comfort in vehicles.

Input for harmonic response analysis (force) will be used from measured data

When the external force is given in terms of rpm and engine orders, it typically implies that the excitation is related to the rotational speed of the engine and its harmonics.

To incorporate rpm and engine orders into the harmonic response analysis, the following procedures are followed:

1. **Convert RPM to Angular Frequency**: To convert RPM to angular frequency (ω):

$$\omega = \frac{2\pi * rpm}{60}$$

   Here, ω is the angular frequency in radians per second.

2. **Define Engine Orders**: Engine orders are multiples of the fundamental frequency of the engine rotation. For example, the first engine order corresponds to the fundamental frequency (1× RPM), the second engine order corresponds to twice the fundamental frequency (2× RPM), and so on.

3. **Excitation Frequency**: The excitation frequency (ω) in terms of engine order is given by:

$$\omega = n \times \omega_o$$

   Here, n=engine order

   ω_o=angular frequency corresponding to the fundamental frequency of engine rotation

4. **Equation of Motion:** The equation of motion for the system under harmonic excitation becomes:

$$[M]\{\ddot{x}\} + [C]\{\ddot{x}\} + [K]\{x\} = F_o \sin(n \times \omega_o t)$$

5. **Steady-State Response:** The steady-state response of the system can still be represented as:

$$x(t) = X \sin(n \times \omega_o t + \phi)$$

Here, X=amplitude of response

ϕ=phase angle between the excitation and response

6. **Frequency Response Function (FRF)**: The frequency response function (FRF) in terms of engine orders describes the system's response to the excitation at different engine order frequencies.

In summary, when the external force is given in terms of RPM and engine orders, excitation frequency is adjusted accordingly, considering the relationship between RPM, engine orders, and angular frequency. This allows us to perform harmonic response analysis tailored to the rotational dynamics of the engine. The Parameters defined for Harmonic Response Analysis is tabulated in 3.3.

Table 3.2 Harmonic Response Analysis Parameters:

Minimum mode number	1
Maximum mode number	1100
Number of engine orders used	30
Minimum rpm (from load file)	800
Maximum rpm (from load file)	2500
Rpm increment (step size)	100
Constant damping ratio (dmprat)	0.02
Mass matrix damping multiplier (alpha)	30
Stiffness matrix damping multiplier (beta)	2e-06

3.6 Noise Analysis

Noise analysis is done by using the surface level vibration which gives surface velocities for the vibroacoustic model. For this project, Ansys script/macros have been developed to calculate the surface velocity. To achieve this, an acoustic transfer vector (ATV) is utilized. The ATV calculates the frequency-dependent relationship between the normal surface velocity of a radiating surface and the sound pressure at a specific field point, relying solely on the geometry of the radiating surface.

These ATVs are derived by merging two classical steps in boundary element (BE) analysis: determining the pressures and velocities on the radiating surface and propagating the disturbance through the medium to the field points. Once the ATV matrix is computed, evaluating the field pressure vector for each new distribution of surface normal velocity requires only simple matrix multiplication.[14]

3.6.1 Vibration and Noise Study

After the application of excitation force on the static part of the engine, every assembled component will have the influence of excitation. This excitation applied at certain parts of the engine will propagate throughout the engine in such a way that every component will experience harmonic excitation coming from different parts of the engine. As a result, there will be multiple excitations on every component. There will be overlapping of different orders of vibration at each component which helps to understand which component is severely affected by which order of vibration. Similarly, structure-borne noise can also be calculated at the same time. As all the assembled components undergo the same phenomena, components experiencing greater vibrations and noise at certain frequency levels can be detected. Optimization methods will be applied to that component (topological optimization) to reduce the noise and vibration level at that frequency.

MBA model of an engine

In the multibody analysis (MBA) model, the crankshaft, camshaft, and transmission shafts are modeled as flexible bodies because of their significant influence on the drivetrain's dynamic behavior. In contrast, the connecting rod, piston, and valves are treated as rigid components. The timing drive, tooth contacts, and clutch system are simulated through user-defined subroutines. Stiffness, damping, and friction values are typically obtained from extensive benchmarking data. Flexible elements are used to model housings, and hydrodynamic bearings simulate the interaction between the crankshaft and the engine block. For vibroacoustic simulations, both the crank train and engine models are assembled.[13]

Chapter 4: Results and Discussion

4.1 Harmonic Response Analysis

Excitations measured are applied at predefined locations of engine assembly. This propagates throughout the engine, hence exciting all the components. The frequency Response Function (FRF) plot can be plotted at the location of the master node defined during the FE generation. Hence, vibration response at any arbitrary location of an engine can be evaluated by defining the master node at that location.

4.1.1 Response analysis of component: Bracket

Here, the response of the Bracket is calculated as below Figure 4.1

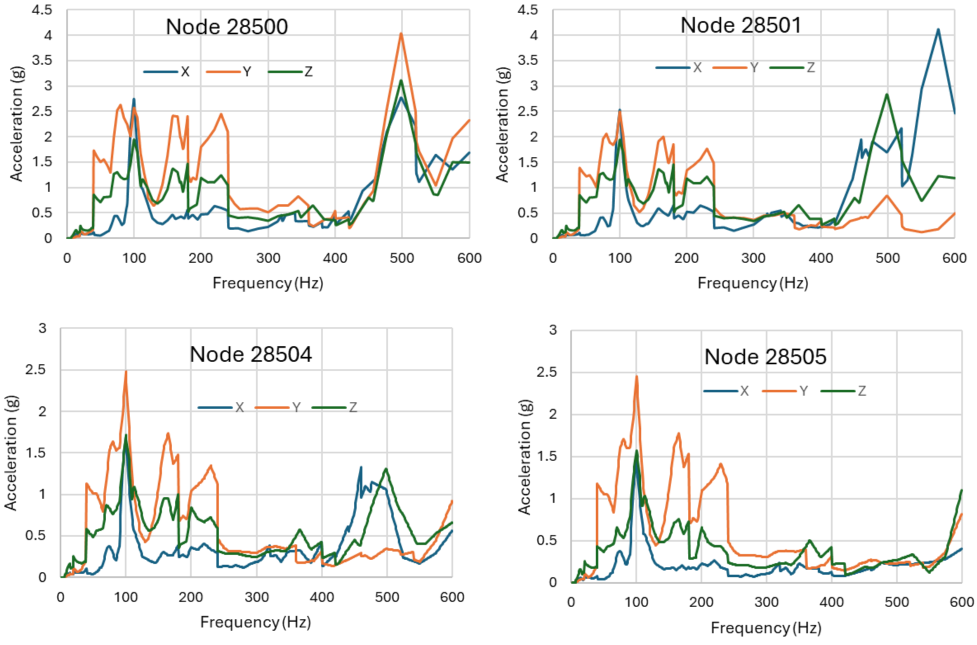

Figure 4.1 Simulated FRF plots at different node locations.

The location of response measurement is varied concerning the distance from the attaching bolt to the cylinder head so that the variation of the FRF is visualized.

The closest location to the bolt is node-28505, from the above Figure 4.1 we can see that lower frequency is more dominating. As we move farther away from the bolt i.e. node-28504, we can see spikes begin to rise at a higher frequency too. Furthermore, if we move to the extreme far of the bracket, i.e. nodes 28500 and 28501, the higher frequency is more dominating than the lower frequency. The reason for shifting of these peaks towards the higher frequency side as we move farther away than the attachment point is attributed below:

1. The structure becomes less rigid as its stiffness decreases, allowing higher-frequency vibrations.
2. Damping may be less effective because of material properties, geometry, mass distribution, etc. as we move away from the attachment point.
3. Near the attachment point, the boundary condition may constrain certain modes of vibration, resulting in a low-frequency response.

4.1.2 Validation of harmonic response analysis

For validating the vibration simulated at any arbitrary location of any component, an oil pan is chosen as the measurement of vibration at the oil pan is a little easier. As mentioned in the above paragraphs about the simulation of response at arbitrary locations, vibration response is first simulated at the predefined locations. Afterward, vibration is physically measured at the same location with the help of an accelerometer. Time series data captured by an accelerometer is converted into a frequency domain with the help of Fast-Fourier Transformation (FFT) transformation.

The location of vibration simulation is represented in Figure 4.2 and measurement is given below along with their responses:

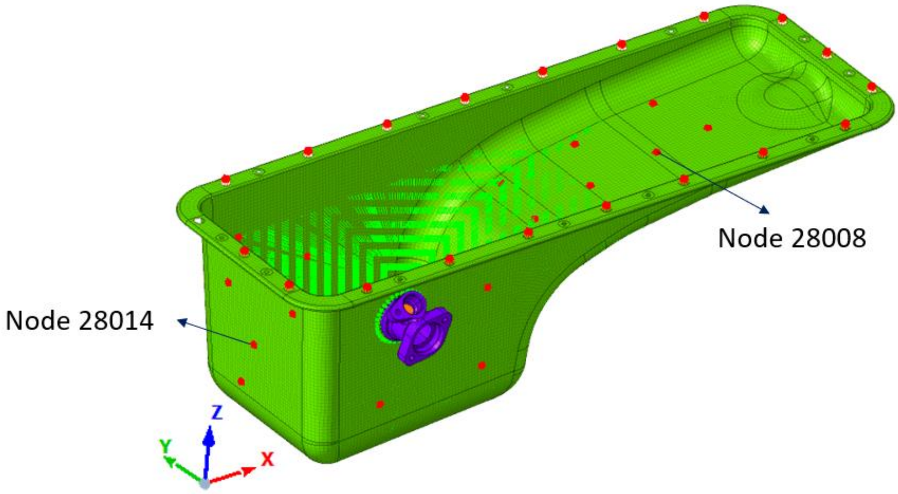

Figure 4.2 Oil Pan and its vibration measurement locations

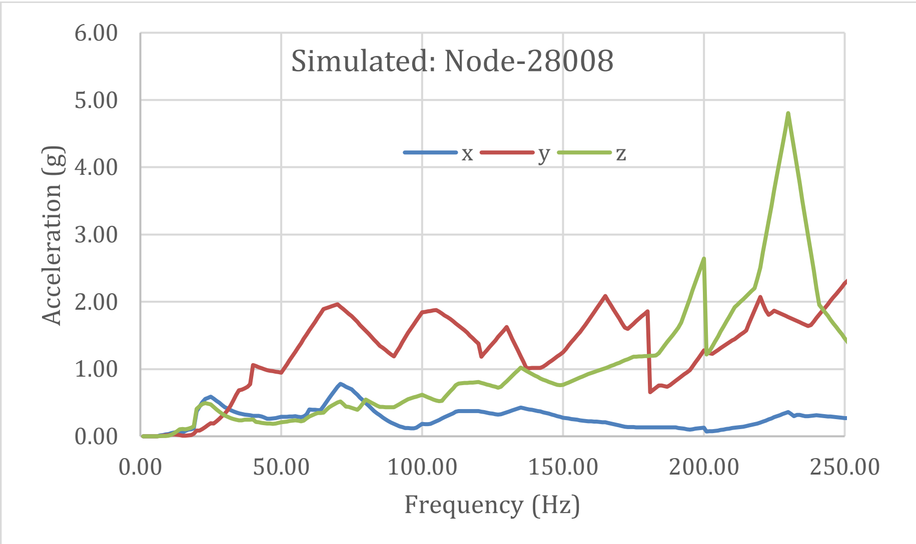

Figure 4.3 Simulated vibration level at node 28008.

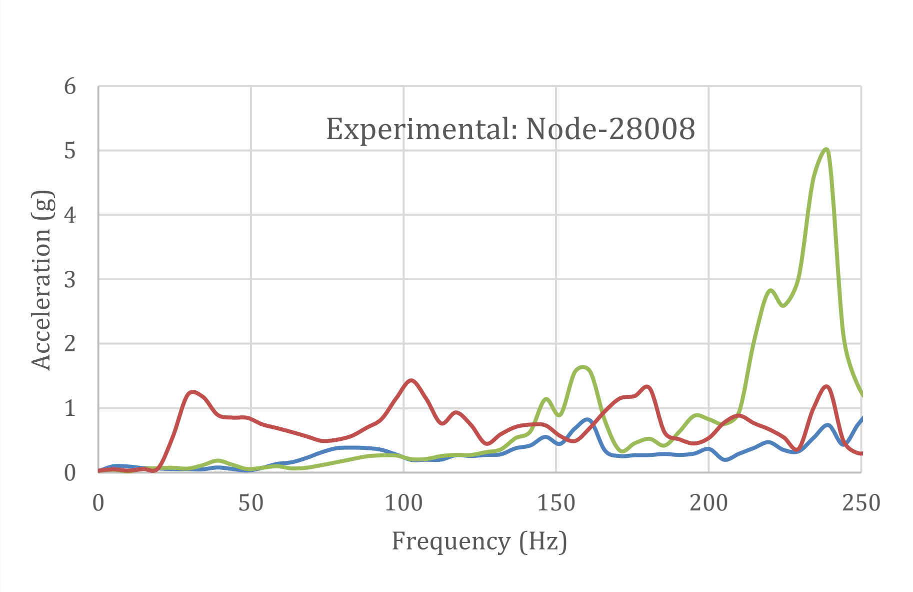

Figure 4.4 Experimental vibration level at node 28008.

The node 28008, which is in the XY-plane, its vibration amplitude will dominate along the Z-axis, it is captured well in both simulated as represented by Figure 4.3, and experimental plots as represented by Figure 4.4. The highest peak is at 231 Hz with an acceleration of 4.55g in the simulated result whereas it is at 239 Hz with 4.96 g in the experimental result. Apart from this peak, the vibration signature is almost similar for all axes. Thus, with this marginal error of 3.34 percent in terms of frequency and 8.26 percent in terms of peak acceleration, it can be said that the result has been physically correlated.

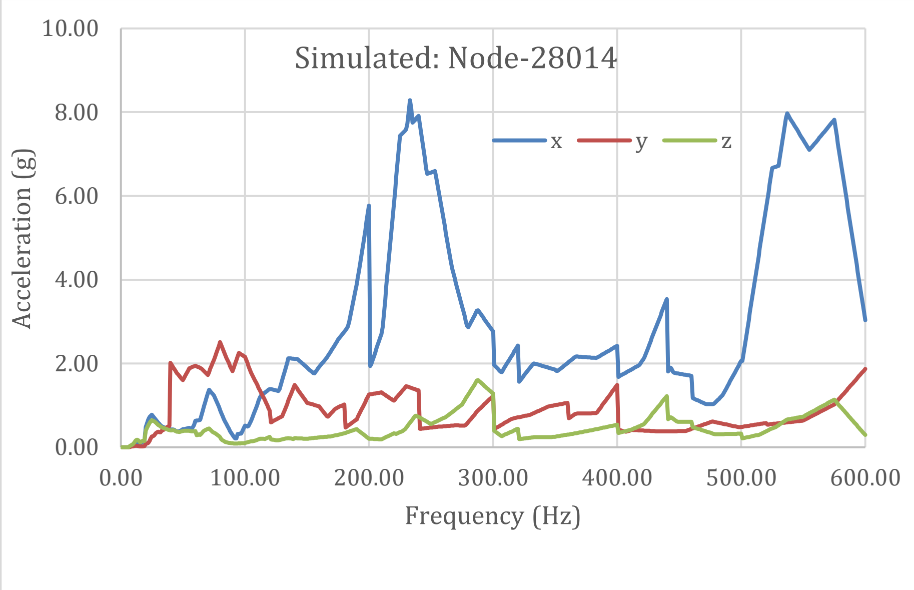

Figure 4.5 Simulated vibration level at node 28014

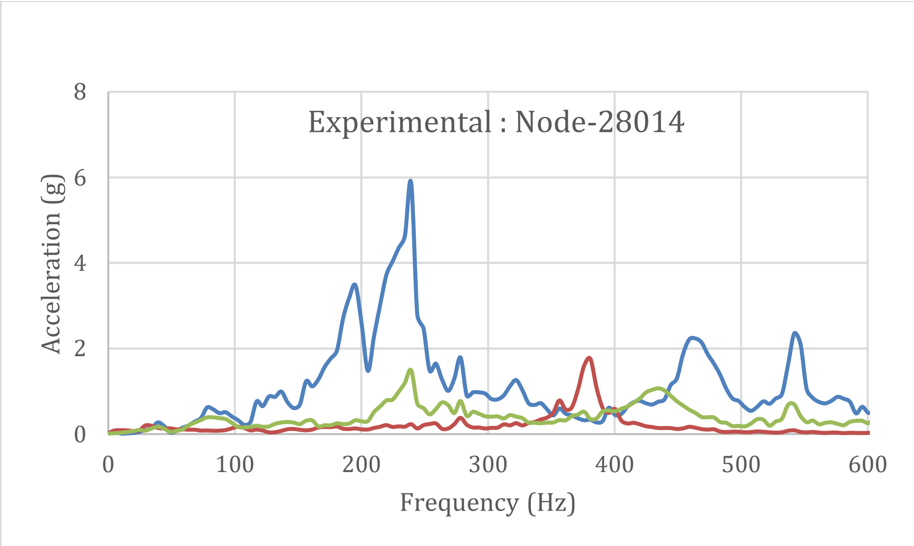

Figure 4.6 Experimental vibration level at node 28014

The node 28014, which is in the YZ-plane, its vibration amplitude will dominate along the X-axis, it is captured well in both simulated as represented by Figure 4.5, and experimental plots as represented by Figure 4.6. The vibration signature is almost similar at the lower frequency, but it has shown some ambiguity at the higher frequency side. This might be because of the effect of the oil inside the measurement location, as the dynamic effect of the oil has not been considered, instead point mass of oil has been distributed over the wet surface of the oil pan. There is a higher amplitude of vibration at the higher frequencies in the simulated result, which is dampened out by the dynamic effect of the oil in the experimental result. Hence, in terms of the nature of spikes in the acceleration level, it can be said that the harmonic analysis has been correlated.

With this correlation in the results of vibration analysis, we can proceed towards noise analysis, as noise is the induced effect of vibration itself.

Chapter 5: Noise Analysis and Design Optimization.

5.1 Overview

The design is continuously checked in terms of its results for identifying potential acoustic susceptible areas, particularly concerning crucial acoustic interactions among individual powertrain components. In addition to optimizing each component structurally, comprehensive simulations of the entire powertrain are necessary to identify which of the components is responsible for the generation of the noise. For this, we look directly into the modal participation factor table, which gives a general overview of components that can resonate at multiple frequencies. From the table block, transmission, and oil pan are found to be resonating at multiple frequencies. For this thesis, the oil pan is chosen as it has fewer design variables and can be easily implemented in the engine without thinking of the other major parameters like thermo-mechanical fatigue, which is influenced by slight design changes.

5.2 Oil pan Noise

One of the largest contributors to engine-radiated noise is the oil pan. Typically, powertrain noise radiation is dominated by thin-walled structures, including the head and timing drive covers, the intake manifold, and the oil pan.[14]

This project does not consider the fluid-structure interaction between the oil pan and the oil. Instead, the noise radiation is driven solely by the vibration level from the bottom part of the engine block, which excites the oil pan.

5.3 Oil pan noise simulation

To simulate the noise of any component, the following steps are followed:

1. An expansion pass is done for that component.
2. Surface element (SHELL 181) is created on the solid surface, and only shell element will be used.
3. Elements attached to the master nodes will be selected.
4. Results from the harmonic analysis at full rpm will be imported at master nodes.
5. A-weighted rms sound intensity level normal to surface with respect to rpm is calculated.

In the case of an oil pan, it is already defined with the SHELL181 element, so there is no need to generate a surface element over the surface. The sound intensity level at the surface at 2500 rpm is directly simulated by using the APDL script as shown in Figure 5.1 below.

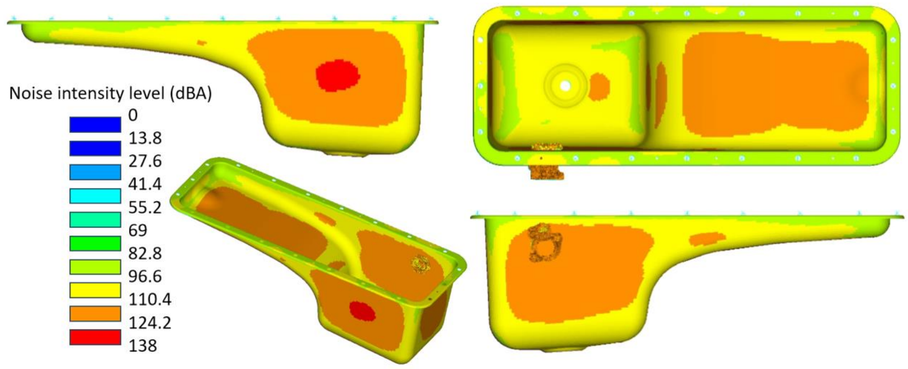

Figure 5.1 Sound Intensity level (dBA) for the initial design.

The initial design of the oil pan is made by using a steel sheet metal of 2 mm thickness. An oil pan has large flat surfaces all along its topology which favors noise emission. Being made up of sheet metal and being connected to the block, the vibration transfer from the block to the oil pan will be intensified because of the transfer of energy from a larger mass to a comparatively lighter mass.

Now, from the contour plot generated for the sound intensity level at the oil pan surface for 2500 rpm, the noise level is found to be higher at the central portion of all the flat surfaces. Meanwhile, curved surfaces generate comparatively less noise. This can be correlated to the vibration directly as noise is just the induced effect of vibration. Curved surfaces are generally stiffer than flat surfaces which means that the amplitude of vibration will be comparatively less resulting in lesser noise. Each color band depicts the range of sound intensity level (dBA) ranging from 0 to 138 dBA with an interval of 13.8 dBA.

5.4 Oil pan optimization

Structural optimization is used to minimize the transfer of vibration and hence noise radiation through the inner path of the engine structure. To improve the structural vibration transfer, stiffening measures are implemented, such as additional ribs or localized wall thickness as well as contouring and eliminating structure-borne transfer path.

To reduce the noise level of the oil pan surfaces, the following iteration has been done:

5.4.1 Introduction of beads on a flat surface.

The addition of beads on a flat surface will stiffen the surface hindering the transfer of vibration through it. As a result, the frequency of vibration of the beaded surface will be lesser as compared to a flat surface. Hence there will be relatively less fluctuation of surface about the mean position. It means that its sound-generating potential will be abruptly reduced.

Cross beads are given at the front and side surfaces, and parallel beads are given at the lower flat surface of an oil pan. To simulate the noise, the whole process is repeated with the changed design of an oil pan. FE model of the newly designed oil pan is prepared, it is replaced by an iterated oil pan in the engine assembly. Then, modal and harmonic response analysis is done on this new assembly. Afterward, an expansion pass of the newly designed oil pan is done to simulate the noise level. The simulated results are shown in Figure 5.2 below:

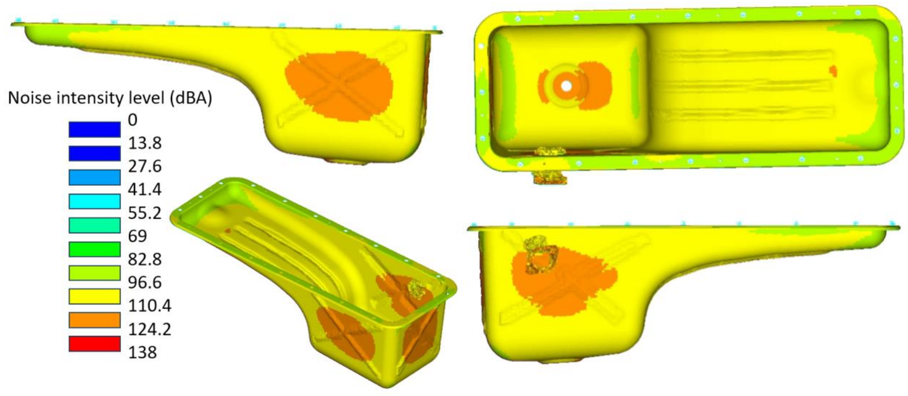

Figure 5.2 Sound Intensity level (dBA) for Iteration 01

Comparing this iterated model of an oil pan with the initial design, it is found that the noise level at the previously flat surface has been greatly reduced with the addition of the beads. Here, two different arrangements of beads were given: one parallel bead and another cross bead. From the obtained result from the simulation, the parallel bead arrangement has been found more effective than the cross beads, as it has completely reduced the noise level over the surface, unlike the cross-bead arrangement which only reduced the noise level in the outer periphery of the beads. Hence, in general, the overall noise intensity level is found to have decreased.

5.4.2 Increment in oil pan thickness

By increasing the thickness of an oil pan, physical properties will be altered, which leads to reduction in vibration level as well as noise levels. An increment in the thickness improves the damping characteristics, which helps to absorb and dissipate the vibrational energy by shifting the natural frequency towards the higher side which will also be away from the engine's dominant excitation frequencies. Vibration amplitudes decrease due to increased stiffness, which resists deformation more effectively, and higher natural frequencies, which reduce resonance with engine vibrations. The added mass also contributes to higher inertia, lowering the vibration response.

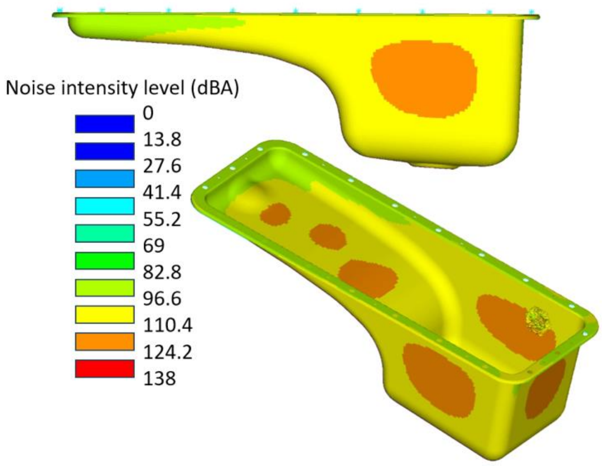

Figure 5.3 Sound Intensity level (dBA) for Iteration 02

From this obtained contour plot in Figure 5.3 we can see that the noise intensity level has been significantly lowered over all surfaces, only the patches of higher noise levels are seen in the flat surfaces, which is comparatively lower than the initial design.

5.4.3 Selection of oil pan design

Among the 3 designs presented, including the initial design, extensive study is necessary to be done for the final selection of the oil pan design. The overall sound intensity level of an oil pan with respect to the frequency at max rpm i.e. 2500 rpm for all three designs is plotted by Figure 5.4 below:

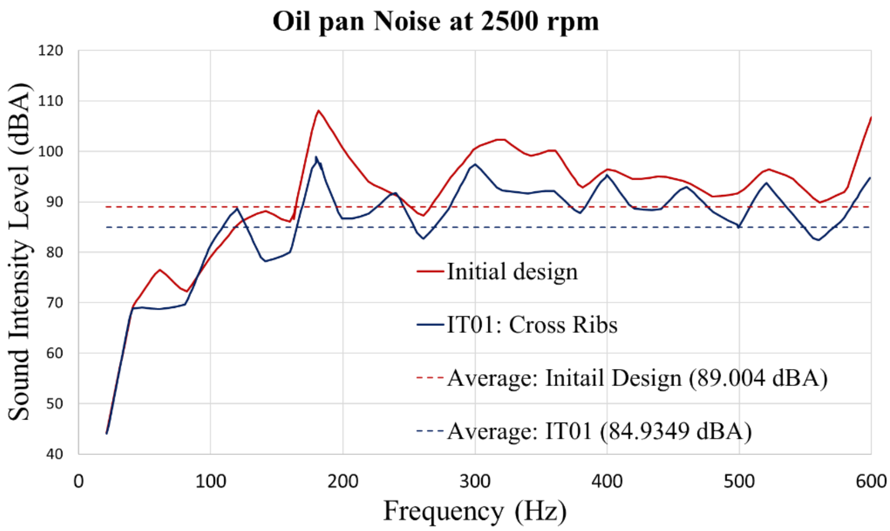

Figure 5.4 Comparison of noise level of oil pan for different design.

Observations:

1. The initial design has higher sound intensity levels throughout the frequency range.
2. The design with cross ribs has a reduced sound intensity level around the peaks when compared to the initial design.
3. Design with 3mm thickness has the lowest sound intensity levels of all.
4. Design with 3mm thickness has better reduction among all designs.

Since we have 2 different iterations for the same component, we will be comparing the feasibility of those iterations with the initial design as tabulated by Table 5.1 below:

Table 5.1 Selection Criteria for initial and beaded design of an oil pan.

Aspects	**Initial design**	**Beaded design**	**Thickened Design**
Mass	10.019 Kg	10.14 Kg	15.0285 Kg
Noise level	89.004 dBA	84.9347 dBA	82.80 dBA
Cost	Neutral	Slightly more	High

From the above table, for selecting one design, a trade-off is necessary between mass and noise level as the noise intensity level is decreased in both cases. In terms of total engine weight, an increment of the weight of certain components by 50% is clearly an unfeasible option. Not only it increases the cost of manufacturing, but it also acts as a dead load on the vehicle, which reduces the vehicle's performance. Although the thickened design has a much-improved noise level it will be traded off by choosing the design with a lighter weight (beaded design) which has a slightly higher noise level.

Now, the sound power level of an oil pan for both initial and modified (beaded) design is compared over the rpm range of 800 - 2500 rpm and is plotted in Figure 5.5 below:

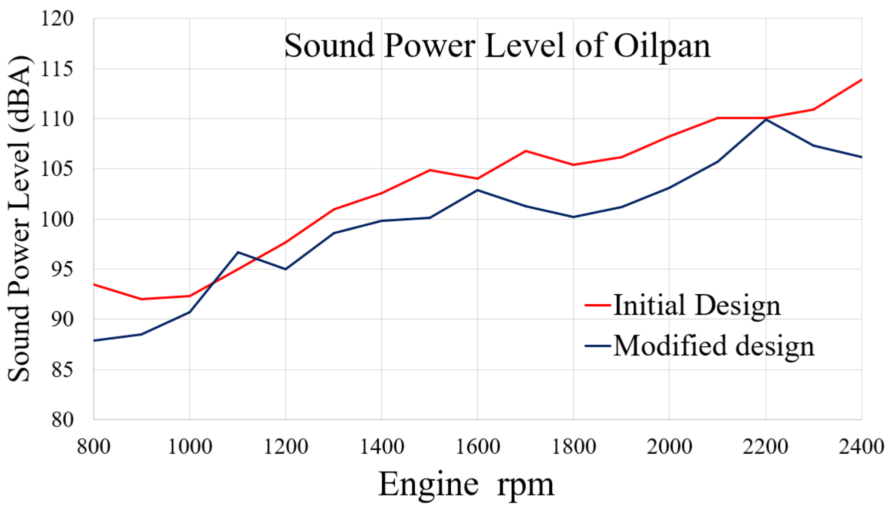

Figure 5.5 The sound Power level of the oil pan for the initial and modified design.

From the above plot, the sound power level of a modified oil pan is lesser at all rpm except around 1100 rpm. Hence, it is concluded that the overall sound power level is found to be reduced with the modified design.

Different aspects of the initial design and final design are tabulated by Table 5.2 below:

Table 5.2 Comparison of initial and beaded design.

Aspects	**Initial Design**	**Beaded Design**	**Observation**
Overall Noise Intensity	Higher overall noise levels	Lower overall noise levels	Significant reduction in overall noise levels
Peak Noise Levels (dBA)	Up to 138 dBA	Up to 124.2 dBA	Peak noise level reduced by ~14 dBA
High Noise Intensity Areas	- Large red regions (124.2 - 138 dBA)	- Red regions are minimized	High-noise areas are reduced in size and intensity

Medium Noise Intensity Areas	- Some green areas (82.8 - 96.6 dBA)	- Expanded green regions	Increase in medium noise intensity areas
Bottom of Oil Pan	Large red region indicating very high noise	Reduced to yellow and green regions	Noise levels significantly reduced
Middle Section	Significant orange and yellow regions	Reduced to green and blue regions	Noise levels significantly reduced
Side Areas	High noise intensities present	Lower noise intensities	Expanded areas of lower noise levels
Noise Distribution	More areas with noise levels above 96.6 dBA	Larger areas with noise levels below 96.6 dBA	Favourable shift towards lower noise levels

5.5 Summary

This chapter explains the simulation of noise levels of a particular component of an engine i.e. oil pan. Various iterations of an oil pan are done for the comparative analysis of noise level. Their sound intensity level over the frequency range at 2500 rpm is plotted together. A comparison of those iterations for the best selection of design is done based on parameters of weight and noise attenuation. An oil pan with a beaded design is selected because of its lighter weight and sufficient noise reduction. The sound power level over the rpm range is also plotted for the initial and final design. To conclude, noise analysis of any component and selection of the best design in terms of noise is done in a similar way as done for an oil pan.

Chapter 6: Conclusion and Research Scope

6.1 Introduction

A full-engine model along with a transmission system is used for the development of the NVH capabilities by incorporating various dynamic excitations coming from a valve train, crank train, and gear train. A research study is done to develop and analyze the complete engine at once. This research meets its objectives to identify and optimize the component in terms of its noise and vibration levels. In addition, it has also developed the following capabilities:

1. Precise calculation of structure-borne noise and vibration of the engine at any arbitrary location in different steps of the engine development process.
2. Structural optimization based on the NVH characteristic during the initial design concept.
3. Multiple feasible design modifications and their comparison can be done which demonstrate the reduction in structure-borne noise of various components.

6.2 Conclusion

The findings of this research work is on the NVH simulation capability development for a complete engine. This research visualized and validated the modal frequencies of engine mounts and engine bending modes. Similarly, it also demonstrated how the nature of vibration changes with the distance from attachment points. Likewise, simulated data is correlated with the experimental data which provided a solid foundation for the further proceedings in this research. One of the components, i.e. an oil pan is optimized by modifying its structure such as adding beads on its surface and increasing its thickness. Both designs were compared with the initial design in terms of noise reduction and weight. One with a beaded design is found to be more feasible.

The advantages of this research are summarized below:

1. This development will eliminate the need for a physical engine for vibrational and noise study, which will be much more economical and timesaving for the development of new engines.
2. Components will be more precisely analyzed for vibration and noise because of combined excitation into it. Currently, the component is analyzed standalone with few assumptions.
3. This method developed will allow us to simulate vibration at any arbitrary point of any components of the engine.
4. It also gives the flexibility to rapidly analyze the effects of change of any components in the engine assembly in a relatively shorter time. (Example: if we just want to replace the air compressor from the assembly, we can just attach the new compressor in the previous master nodes).
5. Optimized engine design can be prepared for NVH in a short duration of time.

6.3 Future scope and development

Based on the findings of this research, it has more potential to extend its coverage area apart from NVH. Since the development of the simulation of the FRF plots at any arbitrary locations has been experimentally verified, it opens many doors for its usage. Besides NVH, vibrational stress can be calculated which can be further extended for the development of Fatigue calculation from the very same model. As NVH capability is developed for the complete engine although with few assumptions, it can be more tuned in the future with the integration of dynamics from other components such as air-compressor, AC-compressor, oil-pump, and so on. A similar methodology can be used for any kind of commercial vehicle apart from internal combustion engine vehicles such as hydrogen engine-based vehicles. The similarity lies in the model development and the key difference is the source of excitation. For hydrogen engines, the excitation can be calculated similarly.

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