Study of Powertrain Isolation of vehicles including Frame Flexibility

M.Tech. Thesis

By

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Study of Powertrain Isolation of vehicles including Frame Flexibility

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of

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Igave Sayali Dinesh

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CANDIDATE'S DECLARATION

I hereby certify that the work which is being presented in the thesis entitled **Study of Powertrain Isolation of vehicles including Frame Flexibility** in the partial fulfillment of the requirements for the award of the degree of **MASTER OF TECHNOLOGY** and submitted in the **DISCIPLINE OF Mechanical System Design, Indian Institute of Technology Indore**, is an authentic record of my own work carried out during the time period from August 2020 to December 2021 under the supervision of Dr. Anand Parey.

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.

22nd May 2022

Signature of the student with date
(Sayali Igave)

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

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DEDICATION

This research is dedicated to my parents for all the inspiration along with my brothers, sisters, and friends for constant support throughout the process.

Abstract

Powertrain in vehicle is a major source of excitation of noise and vibration for a vehicle. It significantly contributes to Noise, Vibration and Harness characteristics of complete vehicle. The powertrain is mounted on the frame through powertrain mounts. These mounts provide support and isolates the vibration of powertrain from rest of the vehicle. The energy from powertrain is transmitted to chassis and surrounding parts through mounts of powertrain.

This paper focus on study of powertrain isolation and its parameters by simplifying powertrain as 6 DOF model mounted on rigid foundation with isolators as one part. In that a simple powertrain model is developed with rigid foundation to study powertrain mounting system. Analytical model is developed to study isolators parameters like stiffness, orientation, and location. This model results are validated using available simulation software. In addition to this paper also include the dynamic interaction between powertrain and remaining vehicle including suspension and frame flexibility. The objective of the paper is defined as to study the effect of suspension system on rigid body modes and study of effect of coupled model of frame flexibility and suspension system on dynamic properties of complete vehicle using analytical method.

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ACRONYMS (if any)

NVH – Noise, Vibration and Harness

PMS – Powertrain Mounting System

TRA – Torque Roll Axis

DOF – Degree of Freedom

CG – Center of Gravity

M – Mass Matrix

K – Stiffness Matrix

Chapter 1

Introduction

For a vehicle, reduced noise and vibration in passenger compartment is one goal for customer comfort. The torque generated in the multi-cylinder engines is source of energy to propel the vehicle with a concentrated mass. Along with road-tire excitation, powertrain is also a major source contributing to noise and vibration. This becomes even more critical in future vehicles which will employ high power density powertrains with lighter and more compact body frames. [1] The powertrain mounting system (PMS) is generally a system with a powertrain along with three or four isolators mounted to chassis or subframe or vehicle body. The PMS also has high influence on ride as it is excited by road bums, type, and irregularity.

For a better NVH characteristics and to comply all vibration targets, optimum design of powertrain mounting system is expected. The functions of isolation mounts are to support the powertrain weight which is undergoing torque load along with road load excitations and provide isolation to the unbalanced engine disturbance force from the vehicle structure to reduce structure born noise and vibration. The important parameters for any PMS designing are to identify the layout and powertrain structure which helps to get number of mounting locations it requires from available space of engine compartment. Also, the mass properties, center of gravity and inertia properties of powertrain are influencing parameters. The isolator information like individual stiffness, damping, and orientation angles are derived from NVH targets for the vehicle. Thus, designing an optimal PMS is challenging task because of packaging constrains, manufacturability, complex structure of powertrain and its properties.

To design a PMS, the system should be decoupled as the coupled body modes of vibration have influence on each other. There are several decoupling methods available to use for optimal designing. Decoupling the engine block rigid body modes aims at separating and tuning the natural frequencies within the acceptance limits. [2]. This is also important to start in early development phase of designing for correct optimization of space and layout of PMS as later lots of constraints of other subsystems starts arising. And change in the optimum design of PMS can also lead to poor vibration performance.

One of the design objectives of a powertrain mounting system is to reduce engine vibration coupling in a certain frequency range with respect to certain engine excitations by properly defining mount locations, orientations and stiffnesses. [11]

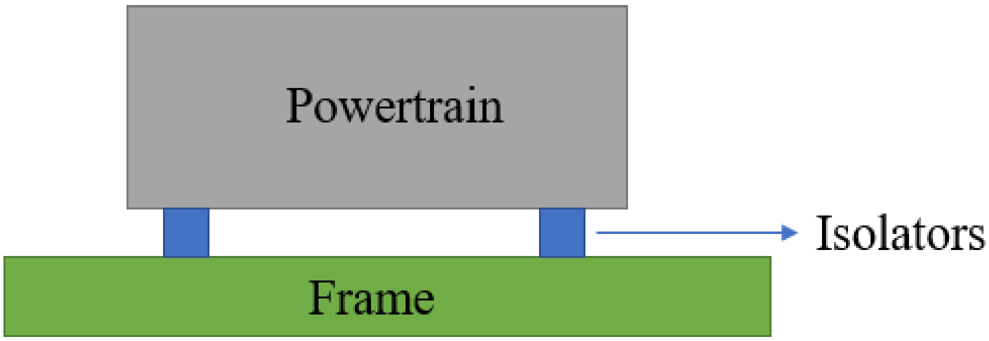

Fig 1. Powertrain Mounting System (PMS)

Torque Roll Axis

The torque roll axis is one of the efficient and popular decoupling method used for to study the PMS analytically. Generally, a powertrain is showed as a six degree of freedom rigid body with three translation and three rotational independent cartesian axis. As shown in below figure, when powertrain is mounted on mounts it has six physical modes i.e. three translational - vertical bounce (wz), longitudinal force-aft (wx), horizontal side motion (wy), and three rotational - roll (hx), pitch (hy), and yaw (hz).

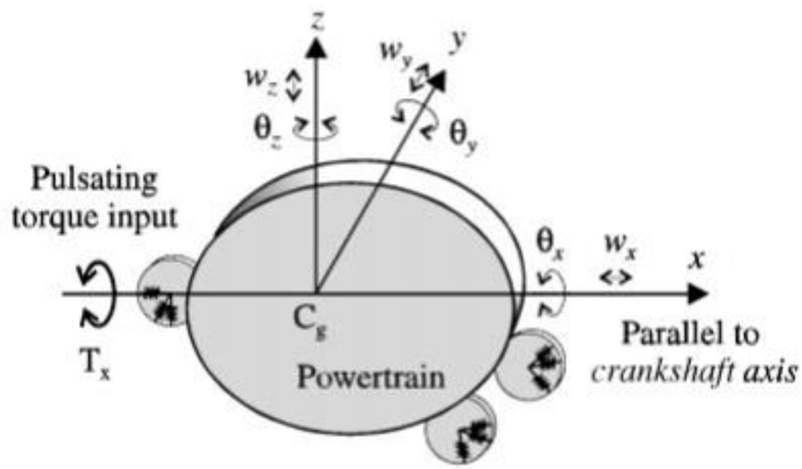

Fig. 2 Powertrain mounting system coupled [1]

Torque Roll Axis is the axis around which pure rotation appears when a pulsating torque is applied on a free rigid body about an arbitrary direction [2]. When torque is applied two forced vibration responses produce. One is decoupled from all other modes i.e. pure rotational in x-direction and other is coupled along various axes. In many practical cases, the TRA does not coincide with any of the principal inertial axes or the crankshaft axis in fact typical deviations may be up to 25 degrees. [1].

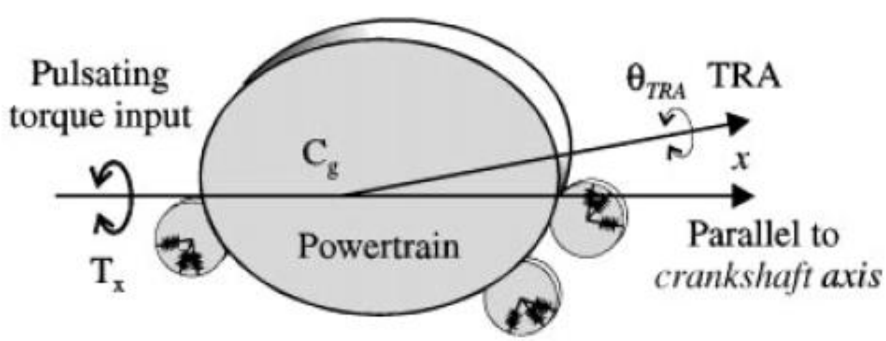

Fig 3 Decoupled TRA mode [1]

For TRA calculation, if a body is unconstrained, the TRA is calculated by Engine torque and inertia properties of the engine. Hence the TRA cannot have unique direction as the TRA is defined by Euler’s equation of motion. But when engine is constrained the second order terms of Euler’s equation are negligible as the rotational displacement of body becomes small. This gives unique direction of TRA. The term &&mode" may refer to the forced response &&mode" along or about a chosen physical co-ordinate axis. [1].

The main purpose of any design is to keep natural frequencies below excitation frequency range. Therefore, a physically decoupled system has a better chance of producing fewer resonances over the operating range. [1]. The decoupling can excite only one mode and can have less chances to have resonance.

Chapter 2

Literature Review

For designing an optimized powertrain mounting system with decoupling method, TRA is one of the best and efficient method to do it. A new axiom for the TRA mode decoupling has been suggested and the necessary mathematical conditions have been established [1]. The paper shows comparison with computer simulation of TRA decoupling engine mounting system that provides a complete decoupling of physical modes for a multi-dimensional mounting system of an automotive engine gearbox system.[1]. Generally, powertrain is known as rigid body with six degree of freedom along with mounts considered as springs with stiffness. TRA method is used to solve the given problem of off-highway vehicle presented in paper [2].

The analytical way of solving the rigid body modes of 6-DOF powertrain mounting system is presented compared with actual test setup example in [3]. Most of the OEMs have developed the application for designers to simply get the mounting configuration early in the design phase with limited input data [4]. So far most of the papers showed the decoupling of powertrain mounting system by considering the rigid foundation and ignoring the connection between other sub-systems.

The paper [5] has showed the effect of presence of compliant base on TRA coupling method. It has analyzed a vehicle with 31 DOF considering the chassis, suspension system and axle-tire subassembly. A new analytical axiom for the TRA decoupling has been proposed in the paper [5]. The major disturbance for a vehicle comes from powertrain, so this paper has demonstrated the interaction between chassis and powertrain explained with case of City Bus. The coupling matrix is introduced using chassis and powertrain general equations. The study showed this method can be applied to complex structures as well [6].

Various paradigms and alternate isolation system design methods along with TRA decoupling paradigms are discussed. The paper mathematically showed that complete decoupling is not possible for practical system and hence the partial decoupling paradigm is pursued [7]. A full vehicle model of heavy-duty truck is modeled which includes frame flexibility by adding chassis and suspension system to model. The reduced frame model is integrated with powertrain and suspension to develop a full vehicle model and solved using ADAMs software to get optimized parameters [8].

The powertrain mounting system has various parameters to study for designing the optimized system. This paper has shown approach of solving rigid body modes for powertrain considering with or without damping along with a development of analytical model of compliant base [9]. The optimization by using constraints and objectives is done using TRA decoupling method where constraints and objectives are improved by six sigma approaches. The paper shows modeling of PMS with or without damping and a compliant base model [10].

The proposed method is that it requires a minimal set of inputs for determining mount topology, orientation, and stiffness properties for decoupling powertrain modes, and as such it can be used at early design stages, unlike the conventional approaches based on analysis and optimization techniques. [11].

The above all papers referred showed that TRA is the effective way to decouple the PMT and to get optimize solutions of parameters. But most of the research has revolved around only powertrain and has not considered the dynamic interaction between other vehicle systems. Some research shows the effect of compliant base on PMS design analytically. This paper represents an off-highway vehicle with chassis and suspension included. Bernard and Starkey [12] and Spiekermann et al. [13] analyzed various optimization algorithms to prevent the resonances of the engine natural rigid-body mode from being excited by engine excitation.

Chapter 3

Mathematical modeling of Powertrain Mounting System

In this paper to study first the PMS, analytical model is developed. This helps to understand contribution of each parameter to calculate the modes of PMS. This shows high confidence for early development process of the vehicle. It gives room to palay with stiffness, locations of the isolator to reach to optimum level. Various codes and programs have been developed on analytical method to give optimized results based on input target. The analytical method is the correlated with simulation of PMS in Hyper mesh Opti struct environment. With that correlation it is easy to have various parameters tested based on vehicle targets. The powertrain is considered as a rigid body because the flexible modes of powertrain have higher natural frequencies than that of mounting system.

The powertrain is mounted on four mounts known as P1, P2, P3 and P4 supported on vehicle frame. Each mount is considered to have three orthogonal springs having stiffness in all u, v, and w direction. The powertrain is developed with origin at center of gravity in XYZ co-ordinate system. The below figure indicates the CG of powertrain and location of mounts along with orthogonal springs.

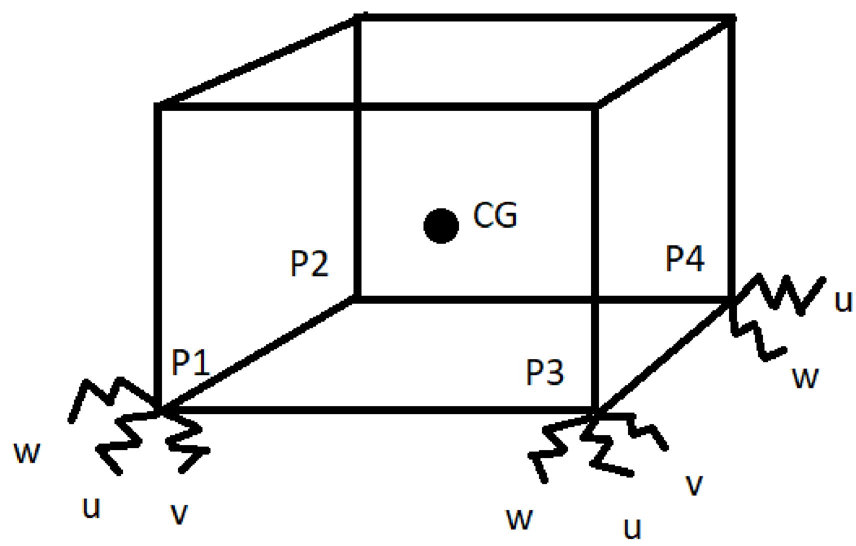

Fig 4. Powertrain mounting system with isolator

To solve the powertrain model analytically, below algorithm is used. This method is used to find out optimal mount configurations of the powertrain mounting system which meets the vibration targets of the vehicle. The analytical model gives you natural frequency of PMS is coded in MATLAB. The results are then validated through hyper mesh model. The MATLAB code works on creation of mass matrix, global stiffness matrix and natural frequencies. The assumptions made in code are as below –

1. Damping of the isolator is not considered
2. Each isolator consists of set of three orthogonal springs.
3. External force can be neglected.

The dynamic equations of powertrain mounting system can be written as the matrix form:

$$[M]\,[\ddot{q}] + [C]\,[\dot{q}] + [K][q] = [F] \qquad \ldots\ldots\ldots\ldots(1)$$

Where,

[M] is mass matrix

[C] is the damping matrix

[K] is stiffness matrix and

[F] is force and moment matrix of 6x1 degree.

As per assumptions, damping matrix and F matrix can be equaled to zero.

Hence the equation becomes,

$$[M]\,[\ddot{q}] + [K][q] = 0 \qquad \ldots\ldots\ldots\ldots\ldots\ldots(2)$$

The mass matrix can be written as

$$\begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & Ixx & -Ixy & -Ixz \\ 0 & 0 & 0 & -Iyx & Iyy & -Iyz \\ 0 & 0 & 0 & -Izx & -Izy & Izz \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \\ \ddot{z} \\ \ddot{\alpha} \\ \ddot{\beta} \\ \ddot{\gamma} \end{Bmatrix} + \begin{bmatrix} kxx & kxy & kxz & kx\alpha & kx\beta & kx\gamma \\ kyx & kyy & kyz & ky\alpha & ky\beta & ky\gamma \\ kzx & kzy & kzz & kz\alpha & kz\beta & kz\gamma \\ k\alpha x & k\alpha y & k\alpha z & k\alpha\alpha & k\alpha\beta & k\alpha\gamma \\ k\beta x & k\beta y & k\beta z & k\beta\alpha & k\beta\beta & k\beta\gamma \\ k\gamma x & k\gamma y & k\gamma z & k\gamma\alpha & k\gamma\beta & k\gamma\gamma \end{bmatrix} \begin{Bmatrix} \mathrm{x} \\ y \\ z \\ \alpha \\ \beta \\ \gamma \end{Bmatrix}$$
$$= 0 \qquad \ldots\ldots(3)$$

Where,

m is the mass of powertrain and

Iij are moment of inertia of powertrain in all X, Y and Z direction.

The global stiffness matrix [K] can be calculated as follows –

$$[K] = \sum_{i=1}^{n} [Bi]^T [Ti]^T [ki][Ti][Bi] \qquad \text{……………………} (4)$$

Where $[ki]$, [Bi] and [Ti] represents the isolator parameters like stiffness, location, and orientation. Stiffness of the isolator is provided by supplier or added in catalogue from where it can be selected. Location and orientation is decided based on available space and vehicle systems and subsystem designs. Finally, this all parameters contribute in calculation global stiffness matrix of the system.

$[ki]$ is the stiffness matrix of i isolator,

$$\text{Ki} = \begin{bmatrix} Ku & 0 & 0 \\ 0 & Kv & 0 \\ 0 & 0 & Kw \end{bmatrix}$$

$[Bi]$ referes to position matrix

$$\text{Bi} = \begin{bmatrix} 1 & 0 & 0 & 0 & -z & y \\ 0 & 1 & 0 & z & 0 & x \\ 0 & 0 & 1 & -y & -x & 0 \end{bmatrix}$$

$[Ti]$ is the angle matrix betweem mount axis and system co − ordinate axis

$$T_i = \begin{bmatrix} cos\alpha u & cos\beta u & cos\gamma u \\ cos\alpha v & cos\beta v & cos\gamma v \\ cos\alpha w & cos\beta w & cos\gamma w \end{bmatrix}$$

Solving equation 2, solution we get,

$$[q] = [Xi] \sin (wit + \alpha) \qquad \text{………………..} (5)$$

Putting solution back in equation,

$$[K] - wi^2[M] = 0 \qquad \text{......................... (6)}$$

Thus, the natural frequency of system,

$$fi = wi / 2\pi \qquad \text{......................... (7)}$$

from above equations, natural frequency of powertrain system can be calculated.

This natural frequency matrix is 1X6 to give six modes of frequency of PMS. The primary reason for calculating natural frequency of the PMS is to avoid the resonance with the engine excitation frequency which achieved proper attenuation of the system. These frequencies are then compared with engine low idle frequency, at which the fluctuation of torque is lowest. These isolator mounts frequency should be wells separated from torque excitation frequency of the engine. The decoupling of modes is also important to achieve vibrational targets. In case of a decoupled system, when the PMS is excited, the response will occur in only one of the modes, and therefore it is easier to tune the mounting system.[11].

Chapter 4

Simulation of Powertrain Mounting System

Analytical Problem formulation

Consider a powertrain of a vehicle to solve analytically as per above equations.

Input parameters –

Mass of the system – 150 kg

No. of Cylinders – 4

No of Isolators – 4

Low Idle RPM – 1000 rpm

Engine Orientation – Transverse

Moment of Inertia – $kg - mm^2$

Ixx	4.000E+07
Ixy	1.500E+05
Ixz	7.000E+05
Iyy	1.300E+07
Iyz	4.500E+05
Izz	8.000E+06

Table 1 Moment of Inertia of Powertrain

Center of Gravity of Powertrain -

X (mm)	Y (mm)	Z (mm)
-45	450	-1650

Table 2 CG of Powertrain

As seen in above chapter, along with powertrain details, isolator details like stiffness, location and orientation also plays important role in calculating natural frequency of the PMS.

Isolator Details –

Isolator Position with respect to CG of system

	X	Y	Z
Front	-50.00	275.00	-1420.00
Mid	-40.00	275.00	-1750.00
Left Rear	155.00	450.00	-2000.00
Right Rear	-155.00	450.00	-2000.00

Table 3 Isolator Location

Isolator Mount Stiffness (N/mm)

	X	Y	Z
Front	150.00	700.00	150.00
Mid	160.00	710.00	160.00
Left Rear	50.00	300.00	50.00
Right Rear	50.00	300.00	50.00

Table 4 Isolator Stiffness

Isolator Mount Orientation

	Z	X	Z
Front	0	-20	0
Mid	0	0	0
Left Rear	0	75	0
Right Rear	0	75	0

Table 5 Isolator Orientation

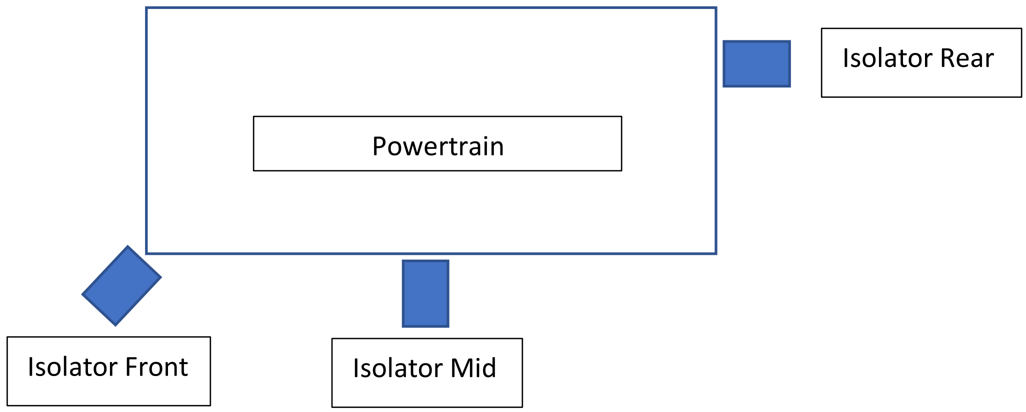

Fig.5 Powertrain Mounting System

Simulation Problem formulation

Above approach was analytical for fast convergence and speedy results for early development phase of vehicle. Once analytical model is developed, similar example is solved in Hyper mesh Opti struct to check the results. In this model, engine is considered as point mass with assigned moment of inertia. Isolators are connected to engine rigidly with stiffness and orientation assigned to it at given location. Solved the same example in Hyper mesh 2019 by Rigid body connections as shown in below image. Both images' shoes one D connections used to solve for natural frequency.

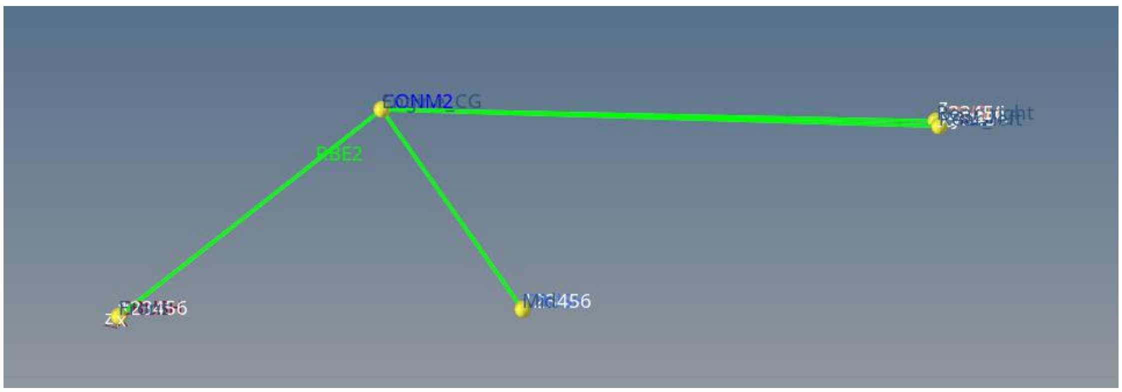

Fig.6 Powertrain Mounting System – Simulation model

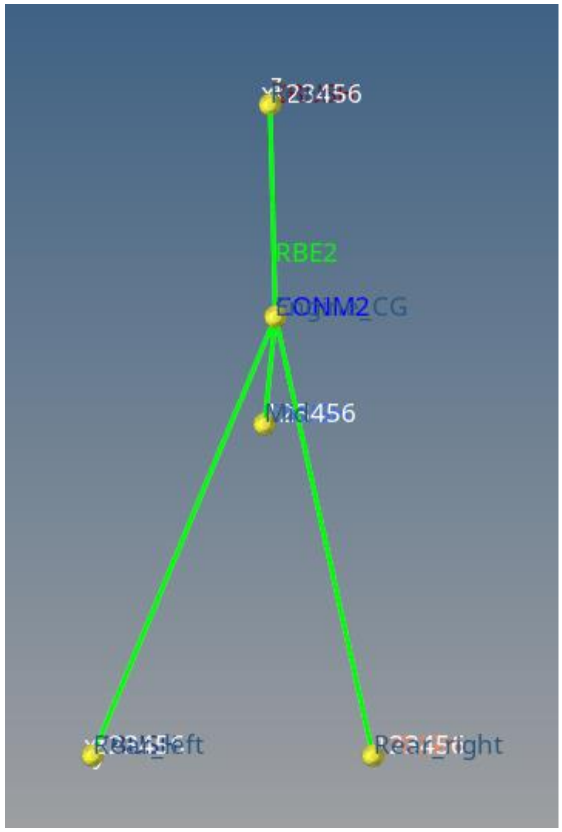

Fig.7 Powertrain Mounting System – Simulation model

Chapter 5

Results and Discussion

In this chapter, the correlation between analytical and simulation model is checked. The analytical model is quick and easy once code is created and various optimization can also be added to try variations. This comparison will validate the analytical model for future use.

Analytical Natural Frequency modes (Hz)	Simulation Natural Frequency Mode (Hz)
3.4	3.36
5.9	5.83
8.2	8.30
10.0	10.5
12.7	12.5
15.8	15.5

Table 6 Natural Frequencies

The rigid body model of powertrain is created in hyper mesh Opti struct with lumped mass at center and assigning stiffness and orientation to four mounts at four ends. The above table shows the correlation between analytical model and simulation model of powertrain mounting system. This result validates simulated design.

Chapter 6

Simulation of Complete Vehicle model

In real life scenario, engine and its mounting system is mounted on chassis. Along with that several other components like suspension, tire is making impact on vibrations in vehicle. This project will include all the components adding flexibility to vehicle to check the effect on natural frequency of the PMS. The project will demonstrate how vehicle side components effects on powertrain isolation.

in PMS, only one stage isolation is used as shown in fig. 8(1). in second step frame is introduced at the where the isolators are getting mounted which is shown by fig. 8(2). The frame here considered as of two types, rigid and flexible. Next flexibility is added by introducing suspension mounted on frame which is shown by fig.8 (3). At the end of suspension, on the wheel hub, tires are mounted which also adds stiffness this is shown by fig.8 (4). Here are the variations of stages which are included in paper –

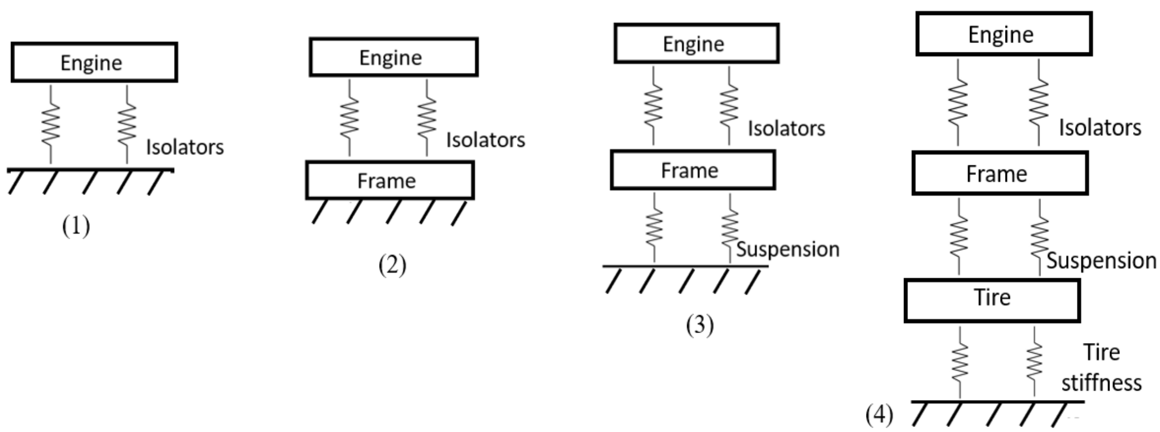

Fig. 8 Different stages under consideration

As shown in fig.8, 1 to 4 shoes different stages of isolation available in typical vehicle. In chapter 4 and 5, single stage of isolation is studied by both analytical and by simulation method. In this chapter, at each step, one component getting added to study the effect on natural frequency of PMS.

Here two types of frames are considered, one is rigid and other one is flexible by keeping all other parts and parameters constant. Below is the image shows complete vehicle under simulation. Fig. 9 represents the complete vehicle including frame, suspension, and tires.

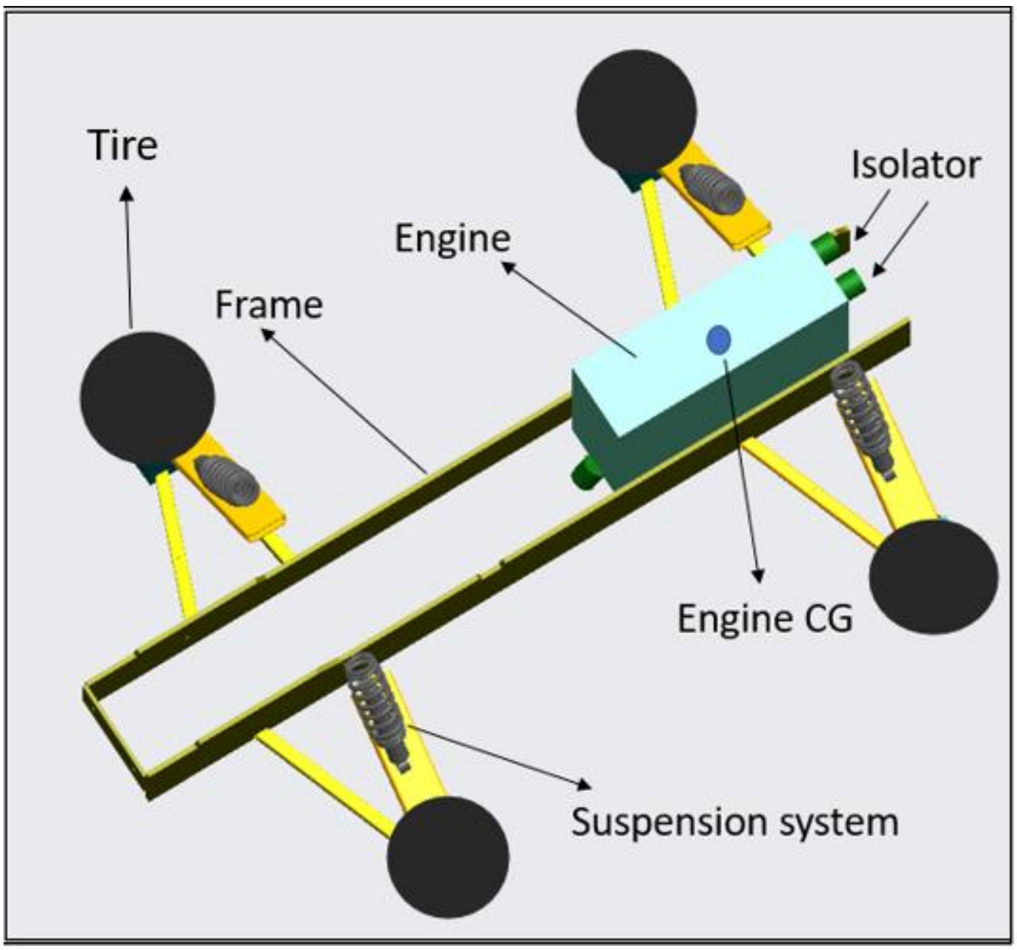

Fig 9. Complete vehicle configuration

For this project, below are the parameters used for various system.

Frame –

- Material – steel
- Density – 7850 kg/m^3
- Poisson's ratio – 0.3
- Young's modulus – 200 GPa

Suspension –

Spring Stiffness

- Rear side – 30 N/mm
- Front side – 25 N/mm

Tire System

- Front tire mass – 12kg
- Rear tire mass – 25kg

Complete vehicle with Engine, isolator, suspension, chassis, and tire is modeled in Hyper-mesh Opti struct environment. Modal analysis is done to understand the behavior of suspension system and chassis flexibility. Suspension and tires are modeled as 1D elements with properties to assign with stiffness and mass. Frame is modeled as shown in fig.9 with change in material for rigid and flexibility. With boundary condition properly given for each run, modal analysis is completed. The output is taken as frequency to see the effect on PMS isolation system.

Modal analysis of complete system gives various modes, along with powertrain isolation modes. Due to added modes, it important to identify only powertrain isolation modes to study. As suspension has modes on lower frequency values like 2-5 Hz. Frame is also having modes starting from 1Hz to high numbers based on components and boundary condition given. Rigid frame shows a smaller number of frequencies as frame modes gets omitted.

Results

Modal analysis is done to study effect of other components on rigid body modes. Here two iterations are given at each step to study different types of frames. Firstly, frame is considered rigid which shows below results.

	Only Engine with Isolator (1)	Engine with isolator with rigid frame (2)	Engine with isolator with rigid frame and suspension (3)	Engine with suspension, rigid frame, and tire (4)
Natural Frequency	3.4	3.78	3.98	3.8
Natural Frequency	5.9	7.5	7.2	7
Natural Frequency	8.2	10	10.3	7.8
Natural Frequency	10.0	11.1	12	9.7
Natural Frequency	12.7	12.8	13	12
Natural Frequency	15.8	16.3	17.8	16

Table 7 Results of Natural Frequencies for Rigid Frame

Graph is created from above data to see the change in natural frequency trends.

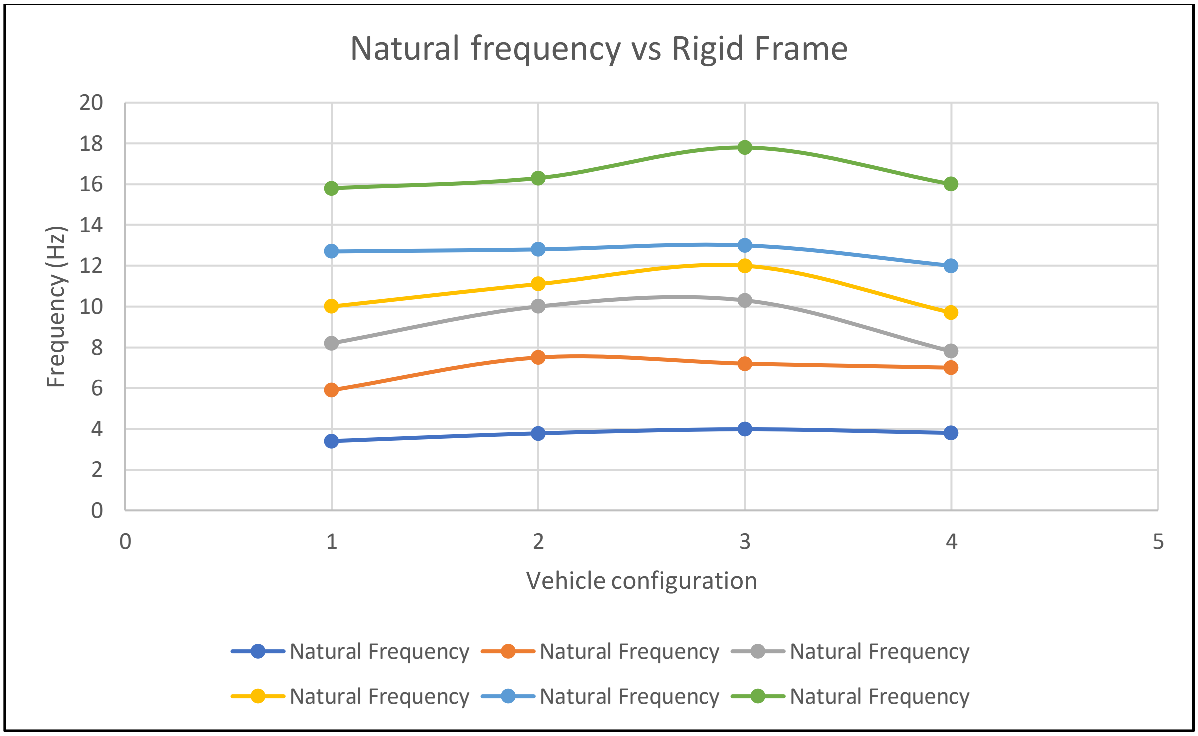

Fig 10. Graphical representation of rigid frame variation

From graph it is seen that, frequency increases when chassis considered as rigid with suspension. When added suspension and tire stiffness, brings system close to PMS system in both frame conditions

Similar approach used for flexible frame condition of vehicle to study effect on modes.

	Only Engine with Isolator (1)	Engine with isolator with flexible frame (2)	Engine with isolator with flexible frame and suspension (3)	Engine with suspension, flexible frame, and tire (4)
Natural Frequency	3.4	2.8	3.1	3.9
Natural Frequency	5.9	6.3	6.5	5.1
Natural Frequency	8.2	8.9	10	8
Natural Frequency	10.0	10.9	11.8	10.2
Natural Frequency	12.7	11.1	11.5	12.5
Natural Frequency	15.8	12.5	12.9	15.9

Table 8 Results of Natural Frequencies for Flexible Frame

Graph is created from above data to see the change in natural frequency trends.

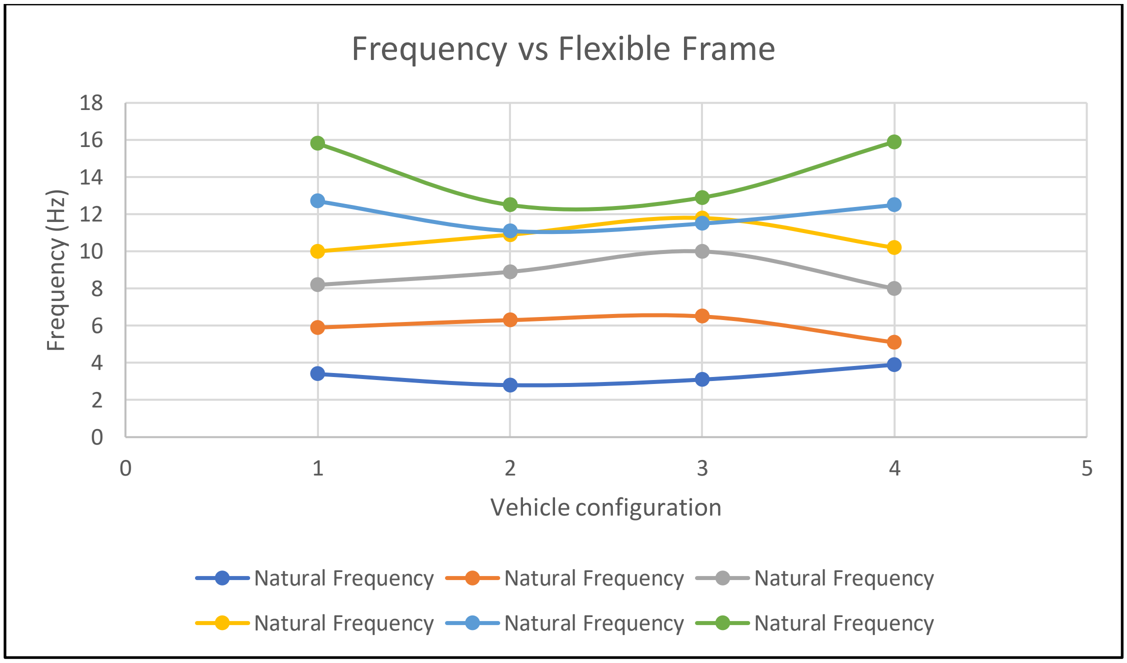

Fig 11. Graphical representation of Flexible frame variation

From graph it is seen that, when chassis flexibility is introduced along with suspension, frequencies moved to lower values. Tire mass and stiffness addition brings system close to PMS system in both frame conditions.

From above two graphs it can be concluded that, when considered complete vehicle, frequencies are very close to that of PMS. The values are coming close to that of PMS with slight shift in higher frequencies as at lower frequencies we see combined effect of other components. Dynamic stiffness ratio of structure to that of isolator should be 5X for effective isolation. Here as the effect is very less it is considered that stiffness ratio of structure to isolator is very high. As structure is rigid, the frequencies moved to higher values and when stiffness of suspension and tire is introduced, the frequencies start coming down like PMS frequencies. When frame is also made flexible, the frequencies moved to lower values for only frame addition. Once the suspension and tire stiffness are added the frequencies moved back near to PMS frequencies.

Chapter 7

Transmissibility

Transmissibility is the ratio of transmitted force to excited force. The transmissibility of isolator is studied that how much force it is transferring through its stiffness when force is applied at Engine CG. In this case to study the isolation force is given as input to engine CG to check the transfer of force through all the isolators at its ends.

To study the effect of other systems on PMS on transmissibility, force of 10N is applied at the CG of Engine in vertically downward direction for a complete vehicle setup including frame, suspension, and tire. It also included both variants i.e., rigid frame and flexible frame.

Below image shows the application of force at CG of engine.

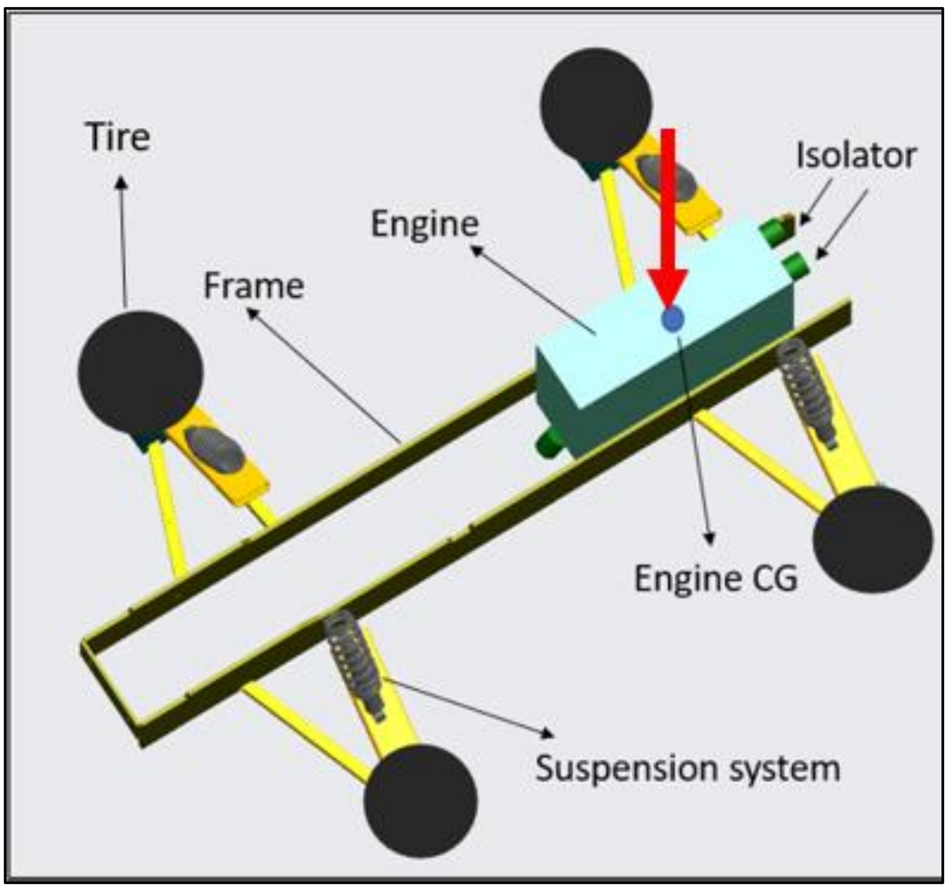

Fig 12. Ten Newton force applied on engine CG to check the transmissibility in the vehicle.

Above shown analysis in fig. 12 is completed in Hyper works, Opti Struct environment to analyze the displacement of the isolators top and bottom nodes against the frequency for both frame variations. A transmissibility is a ratio, here the ratio of displacement of receiver node to source node of isolator is taken as transmissibility. Results are plotted of displacement ratio of receiver to source of force nodes vs frequency to see the transmission of force in the vehicle.

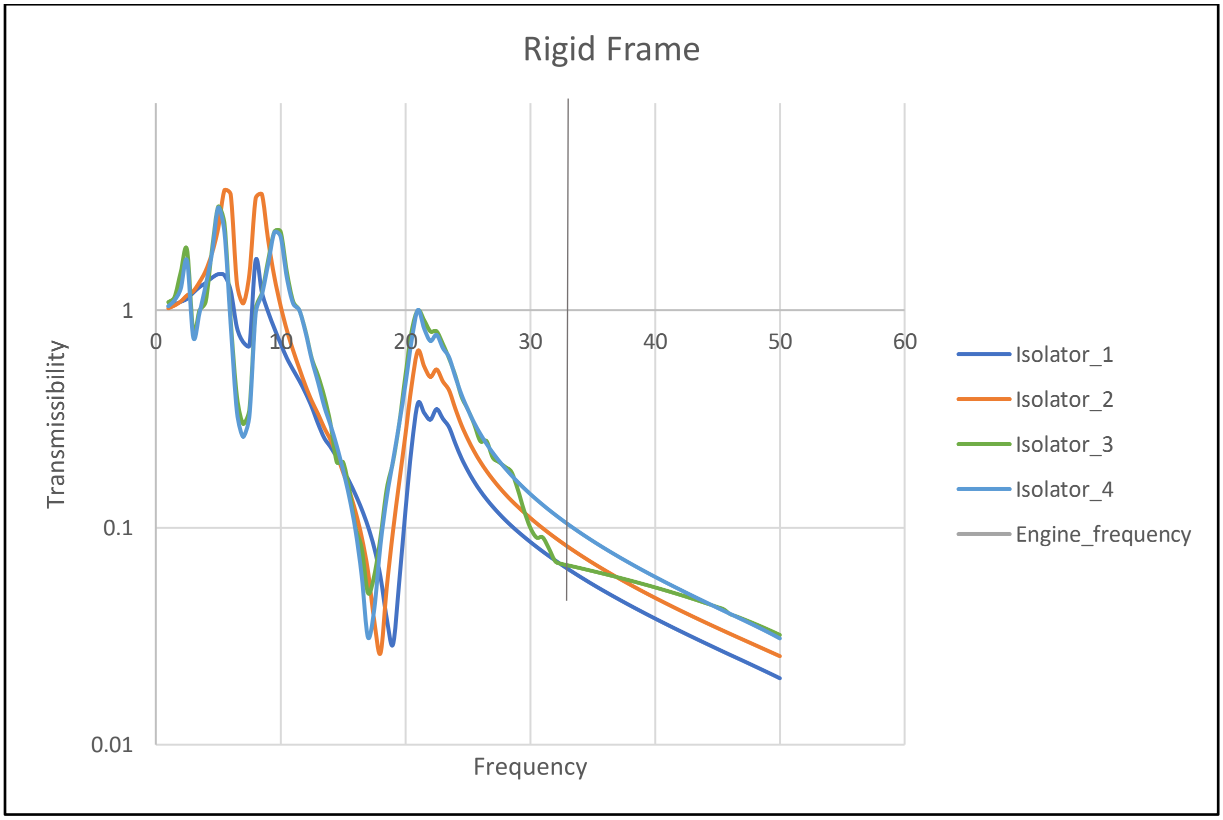

Fig 13 – Transmissibility graph of rigid frame complete vehicle.

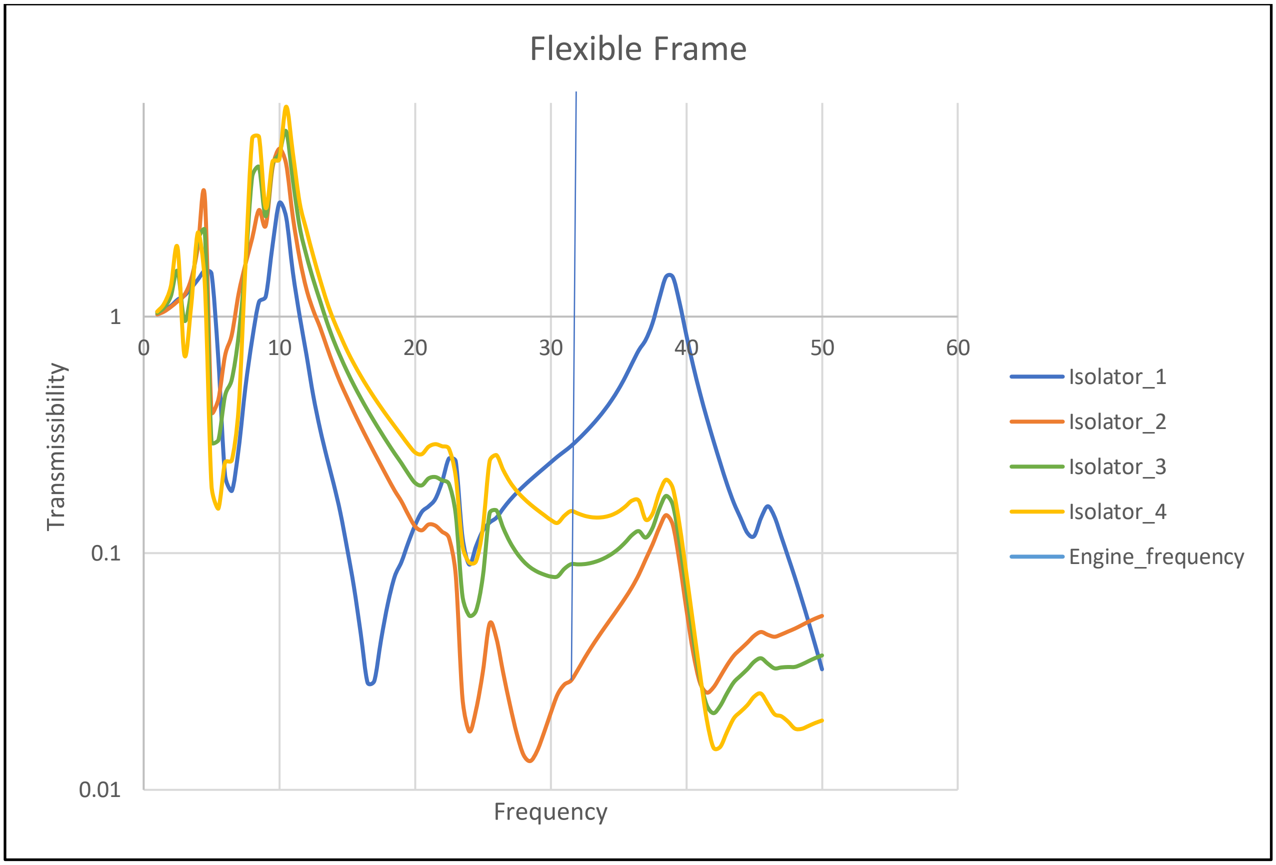

Fig 14 – Transmissibility graph of flexible frame complete vehicle.

In fig. 13 and 14 engine low idle frequency is plotted along with isolator transmissibility which can be calculated from Engine low idle rpm and number of cylinders.

In this case, 4-cylinder engine of 4 strokes is analyzed with low idle frequency is 1000 RPM.

Engine frequency = Engine Rated RPS x No. of firing pulses in one revolution

$$F = \frac{1000}{60} * 4 * \frac{1}{2} = 33.33\ Hz$$

The above graphs show that for rigid frame the transmissibility curves are smooth and below 1 in engine operating range. The acceptable criteria of engine isolation are the highest mode frequency should be <=0.707*low idle engine frequency. In this case the, the highest mode frequency is 23.5Hz and all the natural frequencies of the isolation are coming below 20Hz. It can be concluded that this isolation is under acceptable criteria.

In flexible frame case, the graph is not smooth as that of rigid frame condition due addition of frame modes in between the frequencies of engine modes. First isolator shows peak in transmissibility near 39Hz. This is due to mounting part of the isolator is on the frame which has mode shape at 39Hz. This is showing the affect of flexibility of frame on isolation of the engine. The location of the isolators is one of the important criteria while designing the isolation system and to check the mode of the location with complete vehicle to have smooth transmissibility.

Chapter 8

Conclusion

The focus of the paper was to study the flexibility of frame and its effect on powertrain isolation. The powertrain isolation considered to be one of the important criteria to meet NVH targets of the vehicle and as it directly connected to operator comfort it is given high priority in vehicle design process. So far, many techniques developed to design powertrain isolation of vehicle. Torque Roll Analysis is one of the most effective techniques. Currently more focus is shifted on optimization of the parameters and reaching to optimized solution to design PMS. In this paper PMS is analyzed along with other vehicle systems like frame, suspension, and tire.

Most of the organizations have developed calculator to achieve powertrain isolation by just inputting the vibrational targets, engine, and isolator parameters. In this paper, similar approach is used to develop a MATLAB code with few inputs to get frequency output for isolation system. Similarly same example is solved using Hyper-mesh Opti struct to review the results. The engine isolation and powertrain mounting system is studied by both analytical and simulation method. Both the methods showed high correlation between results for one stage isolation system. This gives high confidence in analytical method to go forward for calculator.

As discussed, the focus was to study PMS along with other vehicle systems, the approach selected by adding one stage isolation at a time. In this paper, two types of frames are considered to get distinctive effect of each system on isolation frequency. When considered complete vehicle including frame, suspension, and tires along with powertrain mounting system, the results showed variation in natural frequency modes of the engine. For rigid frame consideration, the modes showed slight rise in the graph for two stage isolation system, as a result the complete system moved towards rigid side. But upon addition of suspension and tire stiffness, the system shows values close to the PMS system. The slight difference was seen for high frequency (last mode) which was increased by small value than that of PMS alone. Here rigidity of structure played important role.

When considered flexible frame, frame itself added some stiffness that that of rigid ones, which lowered the values of natural frequency modes in the beginning of two stage isolation system. The system become more flexible and operating zone of frequency moved to lower values. As the stiffness goes on increasing when added suspension stiffness and tires stiffness, the value of natural frequency came close to that of PMS in three stage isolation system. The addition of frame flexibility showed minimal affect on natural frequencies shown that of PMS alone. This is due to stiffness ratio of structure to that of isolator is very high.

When checked for natural frequency, it showed minimal effect. So, to check frame modes effect, transmissibility is added to structure. It was to find how much displacement the points show upon adding force on engine CG. The force transfer to isolation end points and to frame side is studied. For transmissibility of rigid frame, the curves of transmissibility of isolator shows smooth flow after the low idle engine frequency. After the engine low idle frequency, it is considered as engine operating zone. And the transmissibility is seen less than one in the engine operating zone.

Similar runs were given with flexible frame as well. But in flexible frame condition, the transmissibility graph showed wavy curves. One peak is observed in engine operating zone particularly for isolator one. As frame is flexible, it is having its own frequency modes which are moving isolation mounting brackets in high frequency zone, which causes the peak in graph. This shows that location of the isolator is one of the important parameters to be considered while designing the PMS system.

Future Scope

In this paper the study of powertrain isolation is studied independently and along with other systems of vehicle like frame, suspension, and tires. Transmissibility showed significant effect of systems and subsystems on PMS system. But this is done only for single vehicle with single parameter values. Similar exercise can be applied to various types of vehicles with varying parameters to check the effect. It is also important to see similar results in off road vehicle where suspension availability is different and tire stiffness is also different based on applications.

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