

# Methodology for Analysis and Design for Powertrain Vibration Reduction in Passenger Bus-with Focused Engine and Transmission Mounts

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**ABSTRACT:** Powertrain Vibration is a very critical source of overall vibration, rattling and structural noise in passenger compartment of any vehicle. Not only it leads to discomfort for the commuters but also increased vibration levels lead to reduced durability (or increased weight due to stiffer vehicle structure to tackle this vibration), as well as, it can be potential safety hazard due to fatigue experienced by drivers due to this continuous noise and vibration. This paper focuses on a stepwise approach to as to study, analyse and design mounts for improved isolation of powertrain mounts. Paper is developed around a case of front engine bus with a 4 cylinder engine with focused mounting arrangement and it clearly explains the methodology that involves measurement on real time front engine Bus, followed by data analysis, design, calculation & CAE verification, and then verification on the prototype bus with new design parts. So, complete product development cycle is covered in this approach. This paper will serve as simple consolidated guideline for students and practicing engineers as how to approach a the new Engine /Transmission powertrain mount design and how to handle vibration issues in same to design/optimize the powertrain mounts for improving isolation behaviour.

**KEYWORDS:** Vibration isolation, Transmissibility, Powertrain vibration, mount, Engine mount, focussed mount, passenger bus

## I. INTRODUCTION

Engine is a highly dynamic system which is a source for various frequency vibration inputs to the vehicle. This involves periodic gas forces, rotating and reciprocating mass inertia, frictional/pumping periodic forces and these all are the function of engine RPM, number of cylinders and number of strokes. Through balancing of engines the rotating mass imbalance aspect is taken care in modern engines, refer figure 2 for the details on this aspect for different types of engine configurations. Normally mainly the target is on the Gas fluctuation inside the cylinder. Frequency for same is given by following formula:-

Excitation frequency by firing disturbance is given by

$$f = \frac{2 \times N \times I}{60 \times C} \text{ Hz}$$

Where, N= Engine idle RPM,

I= No of cylinders

C= Engine cycle (4 /2 for stroke 4/ 2 stroke respectively).

While the road, rear axle and tyre wheel vibration area filtered to some extent by suspension properties, the engine vibration are filtered through the rubber mounts used for installations. Based on transmissibility of these rubber mount and suspension the vibration inputs are transferred to the bus frame from where it is transferred to the body welded /mounted to same.

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 4, Issue 7, July 2015

Based on the mass and stiffness property and thus the natural frequency of body plus frame structure and finally the internal/external components mounted on it like seats , windows , hat track and rear view mirrors etc., the vibration level are amplified and thus finally perceived by the Occupant – driver and passenger.

## II. VIBRATION TRANSMISSIBILITY BASICS

Transmissibility in the case of force-excited system is defined as the ratio of the force transmitted to the foundation to that of impressed force upon the system. Consider a system of mass ‘m’ supported on the foundation by means of an isolator having equivalent stiffness and damping coefficient ‘k’ and ‘ce’ respectively as shown in the below figure 1

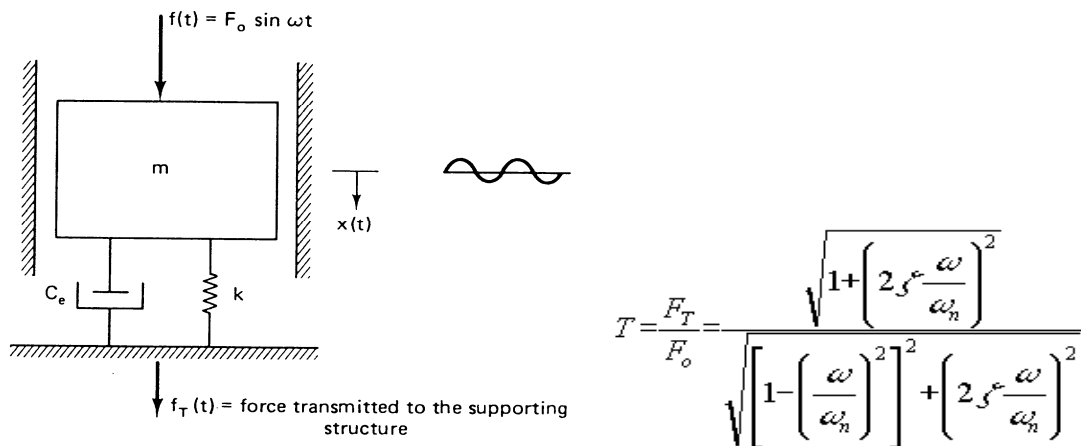


Figure 1- Force Transmissibility to foundation with sinusoidal input on mass

Transmissibility is given by the formula as shown in the figure 1, where f(t) is the harmonic force acting on the system and fT(t) is the force transmitted to the supporting structure, ωn is the natural frequency of the system, ω is the excitation frequency at the supporting structure and ζ is the damping factor. In the region of attenuation, refer isolation instead of transmissibility as a measure of reduction in vibration input and is given as a percentage value occurring for a particular disturbing frequency. Isolation efficiency = % Isolation = [1-T] × 100.

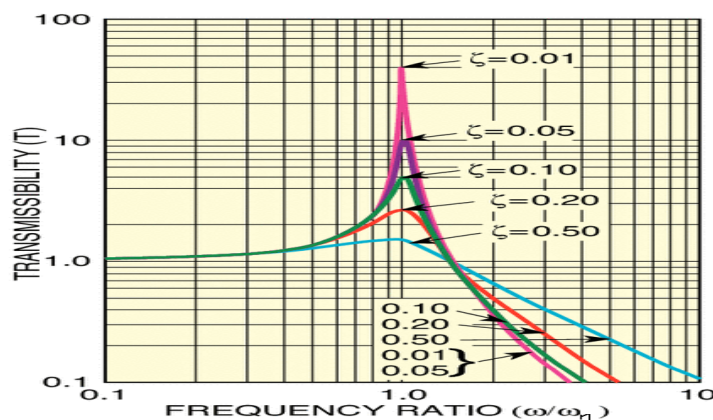


Fig 2-Transmissibility V/s Frequency ratio curve [5]

The curve of transmissibility Vs frequency ratio is the vibration transmission chart. All the curves start from unit value of transmissibility, pass through the unit transmissibility at  $(\omega/\omega_n) = \sqrt{2}$  and after they tend to zero as  $(\omega/\omega_n)$  tends  $\infty$ .

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(An ISO 3297: 2007 Certified Organization)

Vol. 4, Issue 7, July 2015

These curves can be divided into three different frequency regions. The isolation is effective at the higher frequency ratio range i.e  $(\omega/\omega_n) > 1$  and is known as mass controlled region because larger mass gives us the higher frequency ratio. The region with  $(\omega/\omega_n) < 1$  and is known as stiffness controlled region because larger stiffness gives high value of natural frequency and consequently low value of frequency ratio. The middle region is damping controlled region and it involves the frequency ratio of  $(\omega/\omega_n) = 1$ , this is possible if the natural frequency of the mount is close to the frequency of the system, resulting in the resonance and leads to the transmissibility of greater than 100%. This is obviously undesirable.[5]

From the above mentioned points it can be inferred that the effectiveness of the isolator can be improved by selecting the lower stiffness damper and the damping factor has to be selected in such a way that it has to give less value of amplitude at the  $(\omega/\omega_n) = 1$  i.e at resonance condition and even it has to isolate the vibration at higher frequency ranges. Lowest firing frequency which will be at idle RPM is taken as the frequency to be isolated and with transmissibility target  $< 0.35$ . [3] All higher frequencies will automatically get isolated with lesser transmissibility as suggested from above curve in figure 2.

### III. METHODOLOGY

#### A. Parameters to be considered in Design of Engine and / Or Transmission mounts and steps for calculation:-

1. **Centre of Gravity, Moment of inertia and principal axis** of engine and transmission assembly to be estimated to be defined by help of 3D model [1]
2. **Reactions** coming on mounts and force coming on mount in normal and shear direction ,to be calculated for 1 g condition [1]
3. **Torque roll axis** of engine & transmission assembly and elastic centre position to be estimated [2]

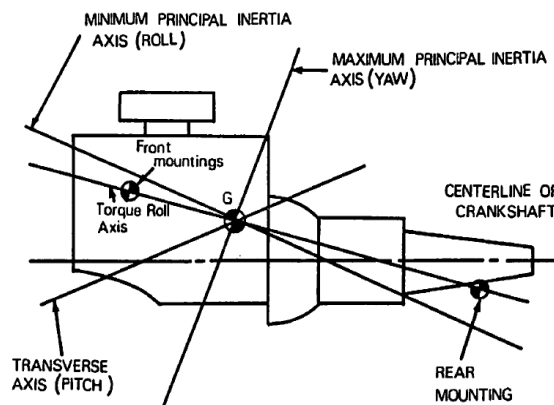


Figure 3 – Axis involved in powertrain mount analysis

4. **Orientation and location of mounts-** mounts to be located as close as possible to the nodes for chassis frame (Modal Analysis to be done in Finite Element model considering various major aggregate connected and with body structure connected). Moreover, for maximum decoupling of bounce and pitch modes each mount to be made center of percussion with respect to other mount location. The following relation to be maintained while placement of mounts along the chassis length-  
 $A \times B = J/M$ , where J mass moment of inertia about principal axis (for pith mode) and M is the engine plus transmission combined mass. A is distance of front mount from center of gravity and B is the distance of rear mount from center of gravity [2]
5. **Transmissibility** is vertical , lateral as well as rotational direction to be limited to less than 0.35 for the highest value of stiffness of mount (considering manufacturing variations.) However hardness of rubber should not be reduced below 40 IHRD from durability point of view . [3] [4]

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Vol. 4, Issue 7, July 2015

6. **Stiffness of mounts & orientation in the selected plane (as per above criteria)** ,to be such that the elastic center lies as close as possible to the intersection of torque roll axis. Also the principal axis intersection point in mount plane will lie above the elastic center , as well as torque roll axis intersection point .This to be ensured for both front and rear mounts. Angular Orientation of mounts to be iterated (refer formula [2] ) to meet above criteria as well as transmissibility and deflection criteria stated in point 5 above and 8 below respectively.
7. **Coupling of modes** and transmissibility to be verified in multi body analysis with design proposed through above steps. [1]
8. **Structure** used for placing the rubber mounts should be designed to have stiffness is respective direction minimum **5 times** the stiffness of rubber mounts to avoid significant effects on transmissibility calculated considering rigid foundations.[6]
9. **Deflection of mounts & strength load cases for mount structures-**  
Mount to be designed considering below criteria in addition to the transmissibility aspect-

Static condition – under static 1g load the deflection should not exceed 3 mm

Dynamic condition – ( based on measurement done on typical 10 ton front engine bus)

- Vertical- 5g load condition
- Lateral- 3g load condition
- Longitudinal – 6g load condition
- Torsional– 2 X (Maximum engine torque) X (1<sup>st</sup> gear ratio) X (Final drive ratio). [6]

All vehicle packaging should have sufficient space considering all these deflections with lowest tolerance limit of mount stiffness (considering manufacturing variations) ,combined with stack up of geometrical tolerance in various mounting and surrounding components.

Most importantly the rubber mount deflection in any condition should be limited to below mentioned criteria in table 1:-

Load type	Continuous Load	Instantaneous Peak load
Normal	<10%	<30%
Shear	<15%	<75%

Table 1- Strain limits of rubber mounts

Both the above criteria can be ensured with using some retainers /interlocking in mounts that limit it motion in these peak load conditions, while also allowing sufficient movement in normal conditions. (Note – above given values for load cases and deflection are guiding reference to initiate the design but values to be confirmed through Road load data acquisition for the particular vehicle application **and also strain limits in rubber to be concluded** through validation.)

#### IV. FLOWCHART

Below mentioned flow chart in figure 4 give methodology to be followed for handling engine vibration transfer issues or designing the new mount system:-.

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Vol. 4, Issue 7, July 2015

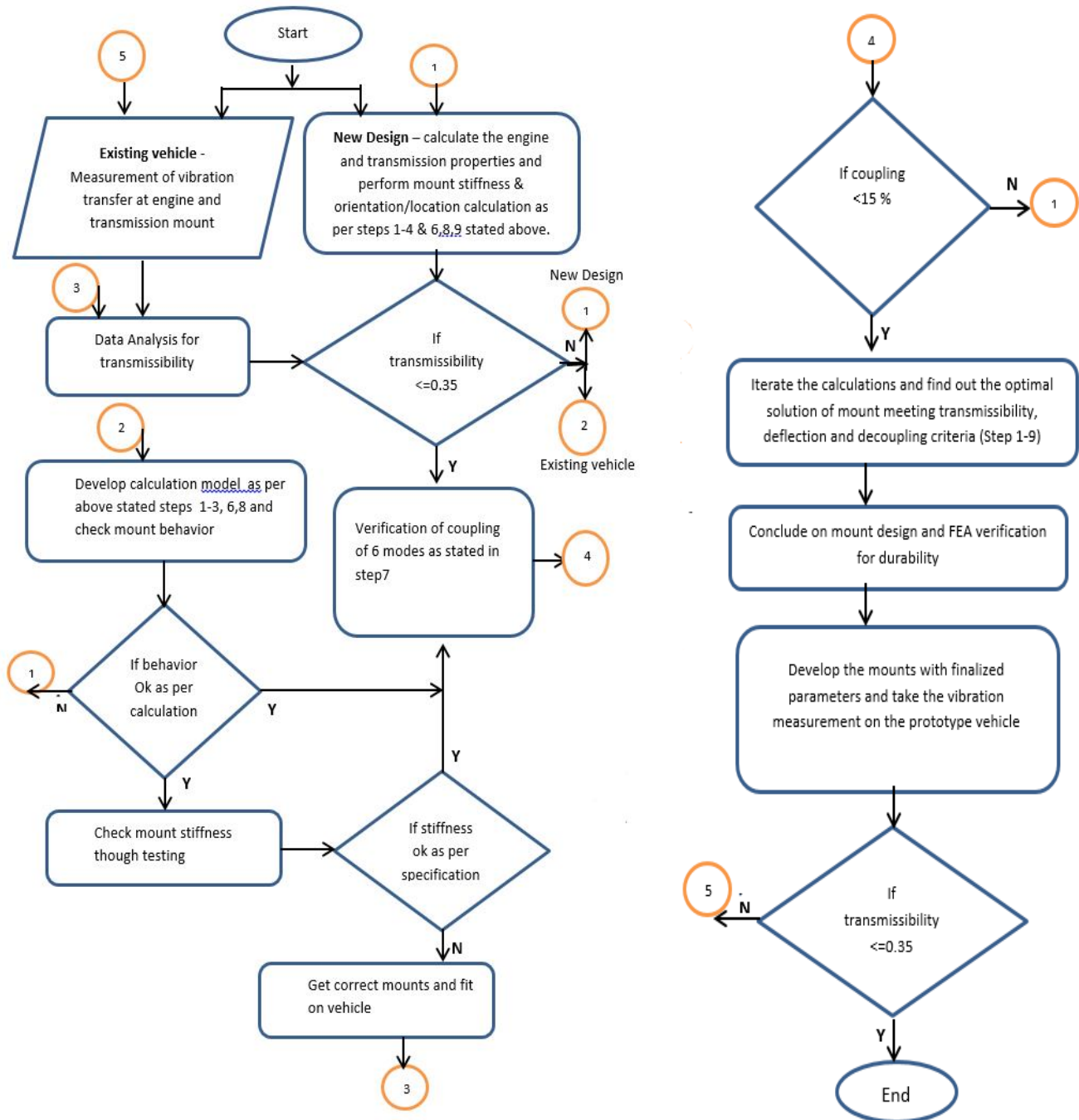


Figure 4-Flowchart powertrain mount design and analysis

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## V. RESULT

### A. Virtual verification of Design and stiffness calculated on Multi body model for decoupling in multi degree system

The calculation done in steps 1-9 except step 7, are based on lumped mass approach and the coupling of 6 modes is not verified in these steps, although checkpoints are provided to avoid coupling. So a multi-body model is to be made and analyzed for all modes. Typical example for a 4 cylinder engine system in front engine 10 T buses is done through CAE ADAMS V13.0 model to understand the extent of coupling in different modes and predict exact behavior. Table 2 below gives the results summary for same for various modes details.[1]

Mode no.	Mode shape	Frequency (Hz)	Target Frequency (Hz)	% Kinetic energy distribution	Target % Kinetic energy distribution
1	long	4.24		98.06	75%
2	roll	5.31	<11.67 @ idle RPM 700	88.31	85%
3	lateral	6.16	<7.52 as wheel hop freq =10.64 & engine idle RPM 700	87.87	75%
4	vertical	7.4		89.51	85%
5	pitch	12.4		72.45	85%
6	yaw	12.83		91.11	75%

Table 2- Result summary for Multi body analysis in ADAMS V13.0 [1]

## VI. CONCLUSION

This paper gives a brief guideline for new design as well as for addressing the powertrain vibration isolation issues in existing vehicle. All critical parameters and steps are listed followed by flow chart giving complete methodology covered all stages for design cycle. This approach was used on a typical 10 Tonner front engine bus with around 110-120 Hp. The transmissibility was approaching to 0.98 when measured [7] and on investigation it was found that stiffness of rubber mounts was very high [10]. Complete design was studied and analyzed. New mounts were designed and implemented on bus to address the issue successfully.

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