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An Approach to Minimization of Drive Train Noise Through Redesign of Engine Mounts

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ABSTRACT:

Drive train noise is more annoying for passenger. For a V-hull base mine protected 4WD defence-vehicle, metallic noise was observed from drive train. To resolve this kind of NVH problem of complicated vehicle like defence required commending knowledge base and experience or say know-how. The critical constraint of subject vehicle was fixed position of engine or engine mounts and transfer case. This causes higher slope equivalent angle of first drive shaft with respect to the installation standard. Noise issue observed as consequence of aforesaid issue which has impact installation parameters of torsional, inertia and secondary coupling excitation frequencies. In this context, a structured methodology has been followed in present work for finding root causes and optimizing the design to come out with optimum solution. Engine mounts have major influence to finalize first drive line slope angle. Hence, re-designing and verification have been done for engine mounts considering its major design parameters and criteria (e.g. shape factors, shore hardness, static deflection and isolation efficiency). At the same time, effects of those changes have been verified theoretically and practically on vehicle. Pass by noise, cabin inside noise, isolation efficiency of front and rear mounts and vehicle floor vibration level were measured. The objective of the exercise is to find out a solution for minimization of drive-train noise.

KEYWORDS:

Engine mounts; Drive train noise; Torsional excitation; Inertia excitation; Secondary coupling

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1. Introduction

Engine mount is the one of most key vibration isolators in vehicles. It transmits engine vibration to chassis or body, and road and chassis vibration to engine. Design of engine mount requires special attention as the nature of vibration isolation from engine-to-chassis is different from that of the road-to-engine-through-chassis. There are a handful of works done on engine mount design. Relevant literatures are covered in following paragraphs. Horovitz [13] focused on details study of three areas: 4-cylinder 4-stroke engine for cars and commercial vehicles; chassis less construction and flexible mounting of oil engines. Also the paper provides formulae for calculating various modes of vibrations and frequencies. Also it emphasizes on area of sandwich rubber mounts. Timpner [3] worked on details design consideration on decouple engine vibration modes of isolation system. He recommended different methods to eliminate vibrations due to internal and external disturbances. The elastic centre of the mounts must coincide with the center of gravity of engine to decouple the modes.

As a consequence, the ideal locations of the mounts are inside the mass engine, which is not feasible. Practically, the mounts are placed outside the engine and still achieve the goal of having the elastic center and the center of gravity coincide. The author discussed three different engine mounts orientations and those are namely two equal mounts symmetrically located, two equal mounts with axes normal to each other and two vertical mounts with different rates respectively. Racca [5] also worked on decoupling engine mounting system design considerations. This exercise is more focused on practical approaches. He recommended mounting location at nodal points. This work also speaks about stability requirement on fore-aft, vertical, lateral and torsion conditions. The author also discussed effective stiffness which is calculated using stiffness of support and mounts. Foumani et al [8] introduced a technique and process for optimizing the characteristics of automobile engine mounting system using an experimental analysis. The proposed technique is independent of the mount type. In the optimization algorithm, rubber, hydraulic or even active mounts can be considered.

The proposed method uses experimental data for optimization and any mathematical model of the vehicle. The experimentations are similar to existing trial-and-error based tests performed on a vehicle for mounts selection. The technique is evaluated experimentally using a quarter car model without separating the system into its sub systems and the results validate the proposed method. Suh et al [9] worked on a multidisciplinary design optimization (MDO) technique to design engine mount system. Forced vibration analysis for vehicle systems and mode decouple analysis for engine mount system is performed to design engine mount system while accounting for driveline. The engine excitation force inertia force, torque due to inertia force, torque due to gas pressure is calculated to obtain the frequency characteristics and vibration level of idle shake. The engine mount system and vehicle systems including body, driveline & exhaust are modelled (FEA) in MDO. The model minimizes not only sum of vertical acceleration response, but also its maximum value of vertical acceleration response of both the steering column and the floor seat in the frequency domain simultaneously to make passengers feel comfortable.

Paliwal et al [10] optimized the power train mounting using design of experiments (DOE). Using MSc. ADMS, a DOE study was conducted by varying the mount stiffness, location & orientation. Static deflection, roll mode frequencies, % kinetic energy distribution were optimized using 8 variables & 2 run. Later one CAE simulation model was verified through actual test data. It holds good correlation between simulations and tests results. Shih et al [14] has worked on trouble shooting methodology of powertrain NVH problem with six different explanations. They considered bending vibration, torsional vibration, neutral idle gear, coating gear rattle, gear noise, transmission shift lever. Even though these literatures provide rich knowledge on engine mounting design consideration and parameters; few trouble shooting issues of powertrain; still, it has few untouched area. Further, none of these papers talked about importance of engine mounts on finalization of drive shafts layout on vehicle. Engine mounts has an influence to finalize drivetrain layout other than vibration isolation of powertrain. This exercise has tried to address the same. The experiments have been carried out on a mine protected 4WD defence-application vehicle. The objective of the exercise has been to minimize the drive-train noise through optimization of engine mount design and to eliminate discomfort level of occupants.

2. Methodology

In general, for commercial vehicle truck, bus, tipper and tractor-trailer except defence-application, equivalent slop angle of drive shafts depends upon engine inclination angle, caster angle of rear axles and rear suspensions geometry only. But, for defence-vehicle, equivalent slope angle of drive shafts depends upon transfer-case or auxiliary gear box inclination angle and front suspensions geometry in addition to the earlier-mentioned ones as these are all wheel drive (AWD) vehicle. The effect of suspension geometry is negligible on equivalent slope angle of first drive shaft. It depends upon engine and transfer case inclination angle only for defence-application. For this typical AWD defence vehicle, positions of engine or engine mounts and transfer case are fixed. It is to be noted that transfer-case position is also fixed by the installation requirement of drive shafts TC to FA and TC to RA. Hence, engine inclination has the key role in determining equivalent slope angle of first drive shaft for this vehicle and it depends upon engine mounts.

Higher equivalent slope angle with respect to installation standard causes higher torsional, inertia and secondary coupling excitation frequencies. This can create vibrational noise from drive shaft as a consequence. Metallic noise has been witnessed from drive train above 57 kmph of the vehicle. To identify probable root causes fish bone diagram Fig. 1 has been prepared for drivetrain noise. After basic analysis of nature of the noise with the help of NVH team and fish bone diagram, power plant disturbance has been eliminated as root cause. Also, two clauses marked using dotted box under driver shaft induced disturbances have been cancelled out. Drive shaft induced disturbances is influenced by drive shaft design parameters and engine RPM, whereas universal joint induced disturbances completely depends upon equivalent slope angle of drive shafts.

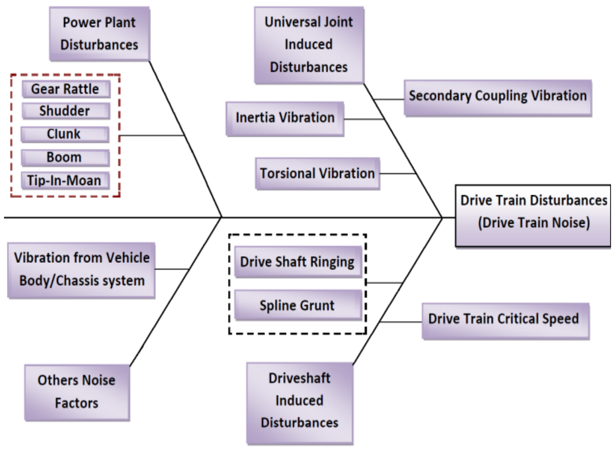


Fig. 1: Fish bone diagram of drivetrain disturbances

From the discussion in the previous paragraph, it can be inferred that engine mount is a key player for afore mentioned disturbances. Vibration from vehicle body/chassis system is considered in external perturbation factor of drive train disturbances. Here also, engine mount plays key role in vibration isolation. Higher is the transmissibility of mount and lower is the isolation efficiency, higher is the vibration transmitted to chassis or to vehicle body from engine. Transmissibility has indirect relation with total or equivalent stiffness of engine mounting system. In the present work, no change has been done on transfer case mounting system due to the constraints on rest to drive shaft layout and packaging constraints. Only few parameters of engine mounts have been optimized to improve equivalent slope angle of drive shaft to minimize vibrational noise. Based upon the fishbone diagram and the discussion in the previous paragraphs, Fig. 2 has been created to find out the options to increase engine inclination angle along with its constraints. Considering the constraints, it has been decided to lift the front mounts without changing the shape factors of rubber pads. Reverse strategy has been adopted for rear mounts. Modified engine inclination angle is 3° considering all constraints. Fig. 3 shows the criteria that are needed to be taken into consideration for design modification of engine mounts. These diagrams are also co-related with previous discussion and fish bone diagram of root causes.

Referring the above three Figs. 2-4, design calculation, verification and experimental works have been initiated. Criteria mentioned in Fig. 5 are measured through experiments. Required input details for calculation are shown in the next section.

Engine Inclination Angle (°) →

Parameters

1. Lift of Front Mounts-
A. change of mounting bracket (shape/height)
B. change of shape factor [<1]

2. Down of Rear Mounts
A. Change of mounting bracket (shape/height)
B. Change of Shape Factor [<1]

3. Mounting position change (distance between front and rear mounts)

Constraints

- *Design Installation constraints*
  a. Max recommended engine inclination angle ≤3°
  b. Principal axis angle w.r.t crank centerline within 8-15°
  c. MRFOB value should be < 1356Nm
- *Packaging constraints –*
  a. Minimum clearance of powertrain with V-Body ≥ 20 mm
  b. Cooling package should have minimum ≥ 25mm clearance with Bonnet/Hood
  c. Transfer case position fixed

Fig. 2: Engine inclination angle - influencing parameters and constraints

Engine mount System →

- **Criteria**
  1. Principal Axis angle **[within 8-15°]**
  2. MRFOB to ensure Engine mounts location**[<1356Nm]**
  3. Each mounts Stiffness & Theoretical Static deflection Calculation **[within 1.5 to 2 mm]**
  4. Transmissibility **[<0.4]** and Isolation Efficiency **[>75%]** verification of mounting system and each mounts.
- **Parameters**
  1. Verification of above criteria 3 and 4 with actual data (Stiffness **[correct shore hardness]** and shape factor **[<1]**) and necessary correction.

Fig. 3: List of criteria and parameters for designing of engine mount system

Drive shaft installation →

1. Torsional Vibration **[≤ 400 rad/ s²]**
2. Inertia Vibration **[≤ 2000 rad/s²]**
3. Secondary Coupling Vibration**[≤ 34.02kg]**
4. Drive shaft Critical Speed **[Ns = 0.75 Nc]**

Fig. 4: Drive shaft installation criteria

Noise Test →

1. Pass By Noise Test **[≤79 dBA]**
2. In Cab Noise Test **[≤ 90 dBA]**

Fig. 5: Noise test criteria

2.1. Initial configuration of powertrain, engine mounting system and drive shaft layout

The required details are mentioned below successively.

2.1.1. Powertrain details

a). Engine
  i. 160 hp TCIC at 2400 rpm
  ii. 550 Nm (Te) at 1400 rpm
  iii. Ideal speed 700-750 rpm
  iv. Fly up speed 2800 ± 50rpm
b). Clutch - single disc dry frictional pull type
c). Gear box
  i. First gear ratio 8.47(G1)
  ii. 6-Speed, direct drive
d). Max speed of vehicle N = 2400 rpm or 88 kmph

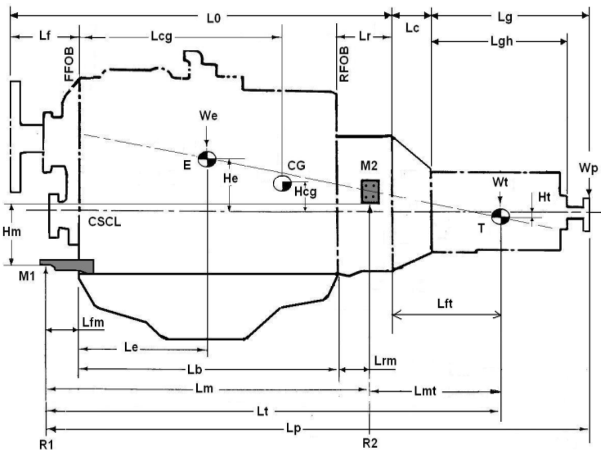
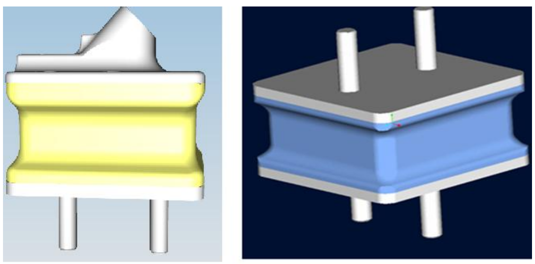


Fig. 6: Power train assembly details

C.G	Centre of gravity of whole assembly (engine + clutch + gear box)
We	Weight of wet engine = 437 kg
Wc	Weight of clutch assembly = 53 kg
Wg	Weight of gear box = 158 kg
Wt	Weight of transmission = Wc+Wg = 211 kg
Wps	Weight of 1st propeller shaft = 30 kg
Wp	40% of Wps = 12 kg [as 40% weight of the 1st drive shaft will come to the engine mounts]
Lf	Distance between fan and FFOB = 203mm
Lr	Length of flywheel housing = 114mm
Lb	Length of engine block = 714 mm
L0	Overall engine length = Lf+ Lb+Lr = 1031 mm
Lc	Length clutch housing = 176 mm
Lgh	Length of gear box housing = 511 mm
Lg	Total length of gear box = 602 mm
Le	Distance of engine C.G(E) from FFOB = 338 mm
Lft	Distance of transmission C.G(T) from front face of clutch housing = 394 mm
Lt	Distance of transmission cg (T) from front mount = Lfm+Lb+Lr+Lft = 1282 mm
He	Height of engine cg (E) from CSCL = 155 mm
Ht	Ht. of transmission C.G(T) from CSCL = -24mm
Lm	Distance between engine mounts = 854mm
Hm	Vertical offset dist. of engine mounts = 84.5 mm
Lfm	Distance of front mount from RFOB = 60 mm
Lrm	Distance of rear mount from RFOB = 80 mm
Lmt	Distance of transmission C.G(T) from rear Mount = Lft + (Lr - Lrm) = 428 mm
Lp	= Lfm+Lb+Lr+Lc+Lg = 1606mm



Front mount Rubber Pad Rear mount Rubber Pad

Fig. 7: Rubber pad of front and rear mounts

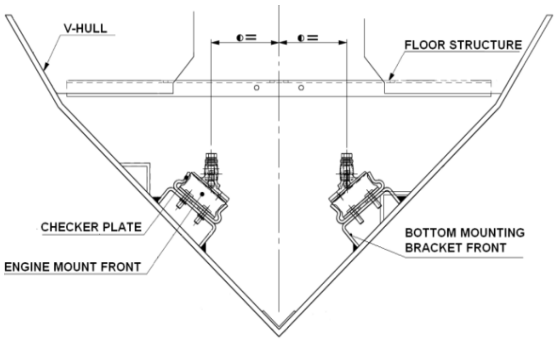


Fig. 8: Information fitment drawing of front engine mounts

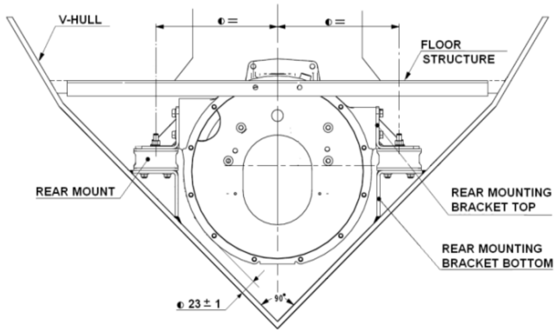


Fig. 9: Information fitment drawing of rear engine mounts

2.1.2. Engine mounting system details

a). Front mounts

i. Shore hardness 65°

ii. l x b x h - 106 x 80 x 44

b). Rear mounts

i. Shore hardness 65°

ii. l x b x h - 105 x 105 x 43

2.1.3. Drive shaft layout details

a) Engine inclination angle 1.67 %
b) Transfer case (TC) inclination angle 3.26 %
c) Frist drive shaft weight (Wps) 30 kg
d) D = Outer diameter of shaft (mm) = 101.6 mm
e) Inner diameter of shaft (mm) = 94.8 mm
f) l_0 = Joint to joint length (mm) = 865 mm

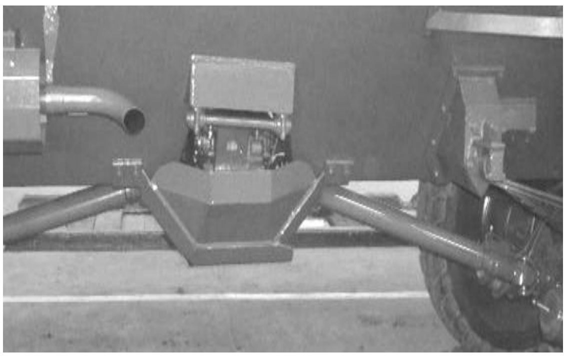


Fig. 10: Drive shafts and transfer case fitment

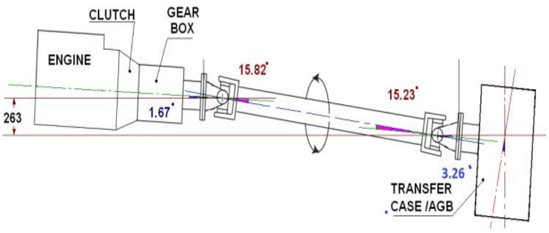


Fig. 11: Schematic layout of first drive shaft (initial case)

3. Design calculation for engine mounts

3.1. Design calculation and verification principal axis angle Ø

From Fig. 5, it is clear that the principal axis angle is,

$$\phi = \tan^{-1}\left(\frac{He + Ht}{Lb - Le + Lr + Lft}\right) = 11.45°$$

This value is within the limit of 8° to 15° [5].

3.2. Design calculation MRFOB and verification engine mounts location

3.2.1. Checking of reaction forces

Referring to Fig. 12, the force equilibrium gives,

$$R1 + R2 = We + Wt + Wp = 660kg$$

Taking moment about R1 (refer Figs. 7 and 15), we get,

$$R2 * 854 = 437 * 398 + 211 * 1282 + 1606 * 12$$
$$\therefore R2 = 542.97kg$$

Substituting R2 in force equilibrium equation as above,

$$R1 = 660 - R2 = 660 - 542.97 = 117.03kg$$

The mounting system is four point symmetrical mounts. Accordingly, reaction on each front mount is (Rfm) = R1/2=58.5 kg. Reaction on each rear mount is (Rrm) R2/2 = 271.48 kg.

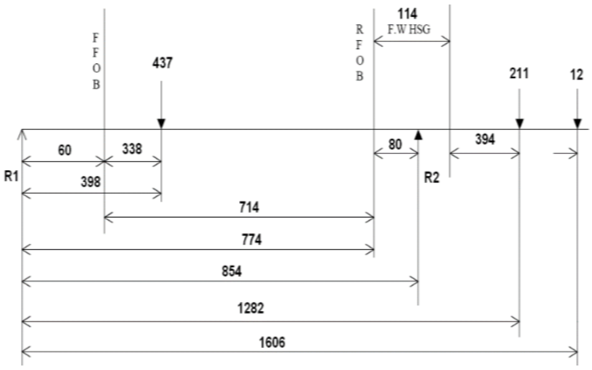


Fig. 12: Reaction force diagram of engine mounting system

3.2.2. Checking of MRFOB and mounts location

As per Cummins recommendations, the MRFOB should not be less than 1356Nm for this engine [4]. The MRFOB is calculated as follows,

$$MRFOB = R1 * 774 - 437 * 376 = -737.3Nm$$

As MRFOB is within limit, the location of the engine mounts is acceptable.

3.3. Theoretical stiffness and static deflection of each mount

In this paper individual stiffness of mounting is calculated using decoupling method only. To find out the static deflection of each mounts it is necessary to calculate the natural frequency of engine and theoretical stiffness value of mounting system. Prior to that, it is required to calculate the C.G of total power train.

3.3.1. Calculation of CG of powertrain (wet)

Total mass of assembly and C.G of power train are derived as follows,

$$Wet = We + Wt = 648kg$$

$$Lcg = (We * Le + Wt * (Lt - Lfm)) / Wet = 625.8mm \quad (1)$$

$$Lcg = (We * He + Wt * Ht) / Wet = 96.7mm \quad (2)$$

3.3.2. Selection of natural frequency

As per engine supplier recommendation, the natural frequency (f_n) for six cylinder engine varies within 8 - 15Hz [4]. For this typical engine, it is f_n = 12Hz.

$$\therefore \omega_n = 2 * \pi * f_n = 75.398\ rad/\sec \quad (3)$$

3.3.3. Theoretical stiffness value of each engine mounts

We know that,

$$\omega_n = \sqrt{k/m} \quad (4)$$

Where, m is the total mass of the system and k is the total stiffness of the mount system. Therefore,

$$k = m\omega_n^2 = 660 * 75.398^2 = 3752N/mm$$

The above is the total stiffness of mounting system. It should be equal to summation of total front mount stiffness and total rear mount stiffness. Let, Kf and Kr be the total front and rear mount stiffness respectively:

$$\therefore k = Kf + Kr \quad (5)$$

Considering decoupling condition, we have,

$$Kf * L1 = Kr * L2 \quad (6)$$

Where, L1 = Distance between front mount and C.G of total assembly = (Lfm+Lcg) = 685.8 mm. L2 = Distance between rear mount and C.G of total assembly = (Lm-L1) = 168.2 mm. Substituting these values in Eqn. (6),

$$Kr = Kf * 685.8/168.2 = 4.08Kf$$
$$k = 5.08Kf = 3752N/mm$$
$$Kf = 738.58N/mm$$
$$Kr = 3752 - 738.58 = 3013.42N/mm$$

Each front and rear mounts are symmetrically placed about the crank shaft axis. Therefore, the stiffness of each front mount (Kvf) = K_f / 2 = 369.29N/mm = 37.6 kg/mm and the stiffness of each rear mount (Kvr) = K_r /2 = 1506.71N/mm =153.6kg/mm.

3.3.4. Calculation of theoretical static deflection

Recommended value from engine manufacturer of static deflection is within 1.5 to 2mm [4].
δf = Front mount static deflection = Rfm/ Kvf = 58.5/37.6 mm = 1.56 mm.
δr = Rear mount static deflection = Rrm/ Kvr = 271.48/153.6 = 1.76 mm.
Both these values are within the limit.

3.4. Verification of stiffness and static deflection with actual inputs and necessary correction

As stated earlier, our objective is to increase the engine inclination angle from 1.67° to 3° to improve equivalent slope angle of first drive shafts with modifications of mounting system. Schematic of these changes are shown in below Fig. For actual case all these values verified using real data of engine mounting pads shore hardness and shape factors. Front mounting system is required to lift by 15.92mm (~16mm) to achieve desired engine inclination. This was done by modifying the bottom mounting bracket fitted on V-Hull. There is no need to change the shape of front mounts. Whereas in rear mounting system required to modify the shape of rear mounts by decreasing its height by 3.9mm (~4mm) because there is no option to down the bottom mounting bracket which is fitted on V-hull further.

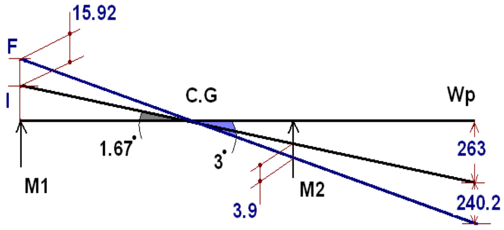


Fig. 13: Schematic layout of engine inclination

3.4.1. Front mounts

3.4.1.1. Shape factor verification

There is no change in shape of front mounts and hence, shape factor will remain unchanged. Shape factor of a mount Ar is the ratio of load carrying area (l x b) and lateral expansion area [2h (l + b)] and it is value should be less than1 [2]. It is expressed as Ar = (l x b) / [2h (l + b)] (7) [2]. Using above equation, we get shape factor Ar for front mount is 0.52 and it is ok.

3.4.1.2. Stiffness value verification

Shore hardness is the parameter of rubber which decides the property and characteristics of it. Initial case, shore hardness (s)° was 65°. The properties of this rubber are as follows [2]. Shear modulus (G) = 13.97kg/cm^2. Young's Modulus (E) = 59.63kg/cm^2. Bulk Modulus (B) = 2334.35kg/cm^2. Non dimensional term (α) = 0.54. The stiffness of a rubber pad in compression (k_c) is given by,

$$\frac{1}{k_c} = \frac{h}{A}\left[\frac{1}{E(1+2\alpha Ar^2)} + \frac{1}{B}\right] \quad (8)$$

Therefore, kc = 1478.05kg/cm = 147.05 kg/mm.

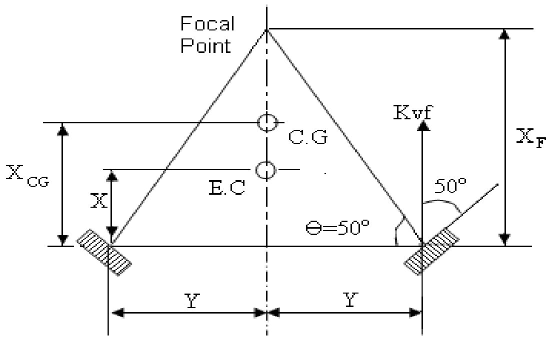


Fig. 14: Schematic of mount orientation

The shear stiffness of the rubber is expressed as,

$$ks = GA/h \quad (9)$$

Where A = cross-sectional area. Therefore ks = 269.24 kg/cm = 26.9 kg/mm. We know [3],

$$Kvf = kc * \sin^2\Theta + ks * \cos^2\Theta \quad (10)$$

Using kc, ks and Θ = 50° in above equation, we get Kvf = 97.45 kg/mm. But, theoretical and required Kvf is near 37.6 kg/mm. So, this rubber is harder than the requirement, and it will transmit more vibration to the vehicle chassis. Hence, we have chosen softer rubber & it has shore hardness (s) of 50°. The properties of this rubber are as below [2]:

a. Shear modulus (G) = 6.52kg/cm^2

b. Young's Modulus (E) = 22.43kg/cm^2
c. Bulk Modulus (B) = 10499.49kg/cm^2
d. Non dimensional term (α) = 0.73.

Using Eqns. (8)-(10), we have Kvf = 40.51kg/mm. This value is nearly matching with the theoretical Kvf. Hence, this rubber of shore hardness 50° is considered for final design of front mounts.

3.4.1.3. Permissible static deflection of front mounts:
Maximum permissible deflection (δ) [2] is given by,

$$\delta = 0.2h \tag{11}$$

Where, h is the height of the mount pad rubber. For front mount this deflection is a combination of compressive and shear deflection. Resultant deflection is given by,

$$\delta f = (\delta fc2 + \delta fs2)^{0.5} \tag{12}$$

δfc (max) = 0.2 x 44 = 8.8mm

δfs (max) = 0.4 x 8.8 = 3.52mm

Hence, δf (max) = (8.8^2 + 17.6^2)$^{0.5}$ = 9.47mm

The static deflection (1.56mm) of front mounts which is less than the maximum permissible deflection (9.47mm) which is desirable.

3.4.2. Rear Mounts

3.4.2.1. Shape factor verification:
As mentioned, shape factor of rear mounts changed by reducing the height of rear mounts by 4mm in place of 3.9mm to achieve engine inclination angle of 3%. Initial height of the mount = 43mm. Modified height of the mount = 39mm. Using Eqn. 7, we get shape factor of initial and modified cases are 0.61 and 0.67. Both cases the values of shape factor are within recommendation limit [2]. So, we considered modified rear mounts in shore hardness selection and permissible static deflection calculation in next sections.

3.4.2.2. Stiffness value verification:
For rear mounts effect of shear stress is negligible because the rear mount axis is perpendicular to the crank shaft axis [2] [4]. Hence, only required calculate the compressive stress of rear mounts (rubber pad). Accordingly, for rear mounts,

$$k_c = K_{vr} \tag{13}$$

Using Eqn. (8), we get initial case, k_c = 218.67 kg/mm = K_{vr} modified case, k_c = 2411.748 kg/cm = 241.1 kg/mm = K_{vr}. This rubber is too harder than the requirement. Hence, we considered nearby two shore hardness (softer side) for verification.

3.4.2.3. Shore hardness (s) = 55°
For the above rubber the design properties are as follows
a. Shear modulus (G) = 8.26kg/cm^2
b. Young's Modulus (E) = 33.13kg/cm^2
c. Bulk Modulus (B) = 11111.11kg/cm^2
d. Non dimensional term (α) = 0.64

The stiffness of rear mount in compression will be k_c = 1467.98kg/cm = 146.79 kg/mm = K_{vr}. It is lesser than the required K_{vr} but very near to requirement so we have to choose comparatively the harder rubber for the rear mounts.

3.4.2.4. Shore hardness(s) = 60°
The properties of this rubber are as follows [2],
a. Shear modulus (G) = 10.81kg/cm^2
b. Young's Modulus (E) = 45.36kg/cm^2
c. Bulk Modulus (B) = 11722.73kg/cm^2
d. Non dimensional term (α) = 0.57

Again, using equation 9 we have, k_c = 1927.45kg/cm = 192.74 kg/mm = K_{vr}. Above cases concludes that shore hardness of the rear mount should be greater than 55° but lesser than 60°. It should be in between 55° to 60° shore hardness but considering factor of safety (FS) and commercially availability, rubber pad of 60° shore hardness selected for rear mount.

3.4.2.5. Permissible static deflection of rear mounts
From the previous section, it is concluded that shore hardness of rear mounts is in between 55° and 60°. Here, it is not necessary to find out the shear stress for rear mounts as it is completely subjected to compressive load only. δrc = 0.2 x 39 = 7.8mm. This value calculated from mount property is greater than the design deflection (1.76mm) which is desirable.

3.5. Transmissibility and isolation efficiency verification of mounting system

3.5.1. Transmissibility verification of mounting system

Transmissibility is the amount of engine vibration which is transmitted through the mounting system to the vehicle structure/body. A transmissibility of 0.4 or less of engine idle speed is necessary for a good mounting system [4]. Inline to this rubber mounts typically have a damping factor of about 0.1. Transmission ratio for a single degree of freedom, spring mass system with damping (ζ) is given by [1]

$$T(\varsigma) = \sqrt{\frac{1 + (2x\varsigma\omega/\omega_n)^2}{(1-(\omega/\omega_n)^2)^2 + (2x\varsigma\omega/\omega_n)^2}} \tag{14}$$

Where, T(ζ) is the transmission ratio, ζ is the damping factor = 0.1, ω = ωm = Exciting frequency, and ω_n = System natural frequency.

3.5.1.1. Calculation of frequency ratio (ωm / ωn)
This actually indicates the ratio of engine's vertical natural frequency to firing frequency. The exciting/firing frequency is expressed as,

$$f_m = 2NI/60C \tag{15}$$

Where, f_m = Exciting frequency of the m/c, N = Idle RPM = 720rpm, I = No of cylinder = 6, C = No of stroke = 4. f_m = 2 x 720 x 6 / (60 x 4) = 36Hz. Therefore,

$$\omega_m = 2\pi f_m = 226.08 rad/\sec \tag{16}$$

From section 3.2, we get natural frequency, f_n = 12Hz and ω_n = 2π f_n = 75.398 rad/sec. To avoid resonance the frequency ratio (f_m/f_n) should be greater than √2 [1]. For better isolation engine manufacturer recommendation is that the ratio should be greater than 2 [4]. Here frequency ratio is (f_m/f_n) = (ω_m/ω_n) = 36/12 = 3. This frequency ratio satisfies the above two condition. If the frequency ratio not satisfies the above two condition then we have to change the idle rpm of the engine.

3.5.1.2. Transmissibility of overall all mounting system
Using Eqn. (14), we get T(ζ) = 0.145. The value is meeting the requirement of good mounting system [4].

3.5.1.3. Transmissibility of front mounts

$$\omega_m = \sqrt{k.g/m} = \sqrt{kvf.g/(R1/2)} \tag{17}$$

a. Initial case - shore hardness 65°. Using R1 = 117.03 kg & Kvf = 97.45 kg/mm, we get ωn = 127.83 rad/sec and accordingly frequency ratio (ωm / ωn) = 1.768. Using Eqn. 22, we get T (ζ) = 0.49. This Value is not meeting the requirement [4].
b. Final case - shore hardness 50°. Using R1 = 117.03 kg & Kvf = 40.51kg/mm, we get ω_n = 82.42 rad/sec and accordingly frequency ratio (ω_m / ω_n) = 2.74. Using equation 22, we get T (ζ) = 0.174. This value is meeting the requirement [4].

3.5.1.4. Transmissibility of rear mounts

$$\omega_n = \sqrt{k.g/m} = \sqrt{kvr.g/(R2/2)} \quad (18)$$

a. Initial case - shore hardness 65° and shape factor 0.61. Using R2 = 542.97 kg & Kvr = 218.67 kg/mm, we get ω_n = 88.89 rad/sec and accordingly frequency ratio (ω_m/ω_n) = 2.543. Using equation 22, we get T (ζ) = 0.204.
b. Final case- shore hardness 60° and shape factor 0.67. Using R2 = 542.97 kg & Kvr = 192.74 kg/mm, we get ω_n = 83.454 rad/sec and accordingly frequency ratio (ω_m/ω_n) = 2.709. Using equation 22, we get T (ζ) = 0.178. Both these values are within the recommended limit [4].

3.5.2. Isolation efficiency of mounting system

Isolation efficiency is a percentage value occurring for a particular disturbing frequency. It can be refer more vividly than transmissibility for better understanding mounts isolation effectiveness. Isolation efficiency is defined by,

$$\%Isolation = [1 - T(\varsigma)]*100 \quad (19)$$

To calculate the isolation efficiency of engine mounting system, it is required to determine the frequency ratio of engine mounting system.

3.5.2.1. Transmissibility of overall all mounting system

Isolation efficiency of mounting system. Using Eqn. 19 and T (ζ) = 0.145, we get isolation efficiency 85.5%. Isolation efficiency of overall mounting system is ok.

3.5.2.2. Transmissibility of front mounts

a. Initial case - shore hardness 65°. Using Eqn. (19) and T (ζ) = 0.49, we get isolation efficiency 51%. This rubber mounting is not ok.
b. Final case - shore hardness 50°. Using Eqn. (19) and T (ζ) = 0.174, we get isolation efficiency 82.6%. Isolation efficiency of this mounting is ok.

3.5.2.3. Transmissibility of Rear mounts

a. Initial case - shore hardness 65° and shape factor 0.61. Using Eqn. (19) and T (ζ) = 0.204, we get isolation efficiency 79.6%.
b. Final case - shore hardness 60° and shape factor 0.67. Using Eqn. (19) and T (ζ) = 0.178, we get isolation efficiency 82.2%. Isolation efficiency of final case is better than initial for rear mounts.

4. Design verification for first drive shaft

In previous section, we have discussed the modification engine mounts to improve equivalent slope angle of first drive shaft. Fig. 15 shows the final condition of drive shaft layout post modification of engine mounts.

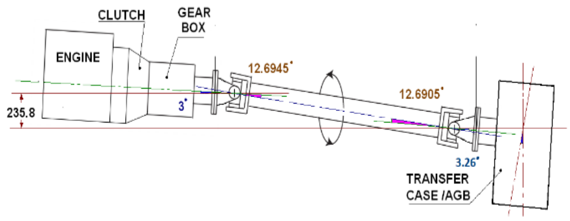


Fig. 15: Schematic final lay out of first drive shaft

4.1. Torsional vibration

We know, $\alpha_{max} = \beta e^2\ \omega^2 \leq 400$ rad/s² [6] (20). Where, βe - torsional equivalent angle (rad) ω - Input angular velocity of drive yoke. Torsional equivalent angle is expressed as,

$$\beta e = \sqrt{\pm\beta_1^2 \pm \beta_2^2 \pm \beta_3^2 ... \pm \beta_n^2} \leq 3^o \quad (21)$$

Generally, equivalent beta angle of 4° is followed for heavy vehicles. There is no phase difference in drive shaft and Eqn. (20) is simplified as,

$$\beta e = \sqrt{\beta_1^2 - \beta_2^2} \quad (22)$$

Where, β_1 and β_2 are the equivalent beta angle of joint 1 and respectively, and are calculated using,

$$\tan\beta = \sqrt{\tan^2\beta_h + \tan^2\beta_v} \quad (23)$$

For the first drive shaft β_{v1} and β_{v2} are zero.

a. Torsional vibration at initial case: Referring Fig. 11 and using Eqns. (23) and (22), we get β1 = 15.82º, β2 = 15.23° and βe = 4.28° = 0.07466 rad. ω = 2ΠN/60 = 251.2 rad/s. ∴α_{max} = βe2 ω2 = 351.735 rad/s². It is within the recommended value [6].
b. Torsional vibration at final case: Referring Fig. 15 and above Eqns., we get β1 = 12.6945º, β2 = 12.6905° and βe = 0.3186° = 0.0055rad.

We know, ω = 2ΠN/60 = 251.2 rad/s. ∴$\alpha_{max} = \beta e^2\ \omega^2$ = 35.198 rad/s² ≈ 35.2 rad/s. Both cases, torsional excitation frequency values are within the recommended limit but final case shows improvement over initial case.

4.2. Inertia vibration

Max. angular frequency for inertia drive and cost [6] is given by,

$$\alpha_{\max} = \theta^2\omega^2 \leq 1000 rad/s^2 \quad (24)$$

This value is considered as 2000 rad/s² for heavy trucks and accordingly this value is applicable for current this typical vehicle. For inertia drive condition θ is represented by θ_D and it is expressed as,

$$\theta_D = \sqrt{(n-1)\theta_1^2 \pm (n-2)\theta_2^2 \pm (n-3)\theta_3^2 \pm ... \pm (1)\theta_{n-1}^2 \pm (0)\theta_n^2} \quad (25)$$

Similarly, for inertia coast condition θ is represented by θ_C and it is expressed by,

$$\theta_c = \sqrt{(n-1)\theta_n^2 \pm (n-2)\theta_{n-1}^2 \pm (n-3)\theta_{n-2}^2 \pm ... \pm (0)\theta_1^2} \quad (26)$$

Here, n indicates the number of joints in both Eqns. (25) and (26).

a) Inertia excitation at initial case
   - Inertia drive equivalent angle. $\theta_D = \beta_1$=15.82º = 0.276 rad. Accordingly, α_{max} = 4806.81 rad/s².
   - Inertia coast equivalent angle. $\theta_C = \beta_1$ = 15.23º = 0.266 rad. Accordingly, α_{max} = 4464.8 rad/s².

Both of these values are more than twice of recommended value of 2000 rad/s^2.

b) Inertia excitation at final case

Further, using Eqns. (25) and (26), we get

- Inertia drive equivalent angle. $\theta_D = \beta_1$ = 12.6945° = 0.2214 rad. Accordingly, α_{max} = 3094.459 rad/s^2.
- Inertia coast equivalent angle. $\theta_C = \beta_1$ = 12.6905° = 0.22138 rad. Accordingly, α_{max} = 3092.545 rad/s^2.

Both of these values are more than one half times of recommended value of 2000 rad/s^2. But it shows significant improvement over initial condition.

4.3. Secondary coupling vibration

Maximum secondary coupling force on drive yoke is expressed as,

$$F = T.\sin\theta \tag{27}$$

Where, T = Input torque in drive yoke = Te x G1 = 4658.5 Nm, θ = βe = torsional equivalent angle (º).

a. Secondary coupling force at initial case: Using Eqn. 27, we have F = 347.67 N = 35.44 kg. This value is marginally crossing the recommendation standard value of 34.02 kg.
b. Secondary coupling force at final case: Using Eqn. 27, we have F = 2.64 kg. This value is within the recommendation limit.

4.4. Drive shaft critical or safe operating speed

It is the speed which ensures there should not be any transverse vibration in whirling. Permissible or safe operating speed of drive shaft is expressed as.

$$Ns = Fs * Nc \tag{28}$$

Where, Nc = critical speed of drive shaft (rpm), Fs = factor of safety = 0.75 (for M & HCV), Ns = safe operating speed of drive shaft (rpm). Critical speed [7] of drive shaft Nc is derived as,

$$Nc = 119507000\sqrt{D^2 + d^2} / I_o^2 \tag{29}$$

Where, D = outer diameter of shaft (mm) = 101.6mm, d = inner diameter of shaft (mm) = 94.8mm, l_0 = joint to joint length (mm) = 865mm. Using Eqn. 28 and 29, we get N_c = 22195rpm and Ns = 16646rpm [6].

5. Results and discussions

All design verification and calculated data are presented in Tables 1 to 3. Static deflection parameters indirectly help to choose correct stiffness of mounting system. Front mounts rubber pad should be softer than rear one (Refer Table 1).Shape factor has major contribution to engine inclination variation (Refer Table 3).

Table 1: Engine mounts parameters

Mounting properties		Initial case	Final case	Remarks
Front mount	Shape factor	0.52	0.52	This is comparatively difficult to change and develop new front mount with different shape factor
	Shore hardness	65	50	Initial pad was too hard
	Mounting bracket (height /shape)		Height increase 15.92 mm (or ≈ 16 mm)	Instead of shape factor bracket height change (fitted with V-hull body)
Rear mount	Shape factor	0.61	0.67	Shape factor change was favourable w.r.t bracket change.
	Shore hardness	65	60	Initial rubber pad was harder. It should be within this range
	Mounting bracket (height /shape)		Unchanged	Refer packaging constraint (a) in Fig. 2 and Fig. 9

Table 2: Transmissibility and isolation efficiency

Mounting properties		Initial case	Final case
Front mounts	Transmissibility	0.49	0.174
	Isolation efficiency (%)	51	82.6
Rear mounts	Transmissibility	0.204	0.178
	Isolation efficiency (%)	79.6	82.2

Table 3: Inclination / equivalent angles and drive - shaft performance

S. No	Parameters	Initial case shape factor-0.61(rear mount)	Final case shape factor-0.67(rear mount)
1. Inclination angles or equivalent angles	Engine inclination angle	1.67°	3°
	Transfer case inclination angle	3.26°	unchanged
	Principle axis angle	11.45°	unchanged
	MRFOB	737.3 Nm	unchanged
	Torsional equivalent beta angle	4.28°	0.3186°
	Inertia drive equivalent angle	15.82°	12.9°
	Inertia coast equivalent angle	15.23°	12.69°
2. Drive shaft performance	Torsional excitation frequency	351.735 rad/s^2	35.2rad/ s^2
	Inertia excitation on drive/coast frequency	4806.81/4464.8 (rad/s^2)	3195.45/3081.94 (rad/s^2)
	Secondary coupling excitation force	35.44 kg	2.64 kg
	Safe operating speed	16646 rpm	unchanged

Pass by noise has been measured as per the IS: 3028 [11] and the data is given in Table 4. The vehicle is belonging to N3G category and accordingly limiting noise values are as mentioned in Fig. 5. The vehicle inside have the single space and there is no partition between driver, co-driver and passenger area. In-cab noise has indirectly been measured with the help of vehicle inside noise as per AIS: 020. The in-cab noise test results for the steady speed mode and stationary mode and are given in Table 5 and Table 6 respectively. It is difficult to run the vehicle in full throttle acceleration mode in available test track. Hence, the test could not be conducted but, it is likely to meet the in-cab noise norms based on our previous experience and considering other two test condition measurement data.

Table 4: Pass by noise test data of noise tests

Gear No	Approx. engine RPM		Noise level (dBA) (Avg - LHS and RHS)	
	At AA'	At BB'	Initial case	Final case
3	1875	2780	78.4	78
4	1875	2480	78.6	78.1
5	1875	2220	78.9	78.3

Table 5: In-cab noise test at steady speed

Microphone position	Speed km/h at 6th gear			
	40 km/h		60km/h	
	Initial case	Final case	Initial case	Final case
Driver ear level (dBA)	86.1	84.6	90.5	88.3
Co-driver ear level (dBA)	86.4	84.8	90.7	88.4

Table 6: In-cab noise test at stationary mode

S. No	Microphone position	Max noise level	
		Initial case	Final case
Idling condition	Driver ear level (dBA)	78.7	78.4
	Co-driver ear level (dBA)	78.6	78.2
Fully open throttle for 5s	Driver ear level (dBA)	89.9	87.8
	Co-driver ear level (dBA)	89.8	87.5

The vibration levels have been measured on both the engine mounts for different speeds ranging from idling to top speed to understand the isolation at different engine speeds starting from idling rpm. Isolation efficiency data have been measured to verify with theoretical data and presented in Figs. 17 and 18. Speed sensor output has been taken from batteries and taken into data acquisition system to measure the engine speed. The accelerometer has been connected through the amplifier to the data acquisition system. The test results for the third gear are discussed because frequency in this gear is predominant considering engine performance curves; transmission configuration and pass by noise/brake performance testing standards. Third gear uses optimum torque and full rpm range, which is very useful. Further, vibration data captured on vehicle floor level at different speed ranges as vibration feeling has been observed on vehicle floor level in addition to noise. Shore hardness improves transmissibility and isolation efficiency of engine mounting system. It also indirectly helps to reduce the noise (Refer Table 2, 5, 6 and Figs. 17, 18).

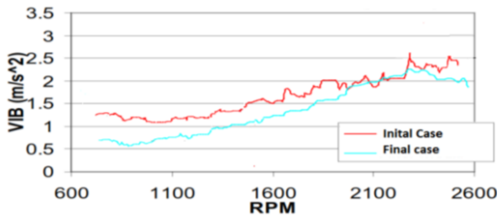


Fig. 16: Vehicle floor vibration v/s RPM

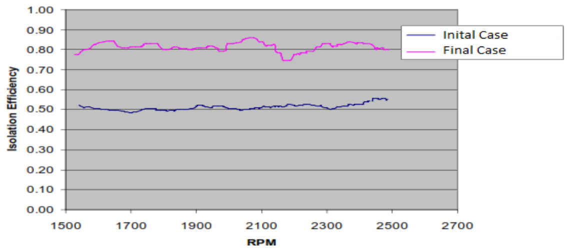


Fig. 17: Isolation efficiency vs. RPM of front mounts

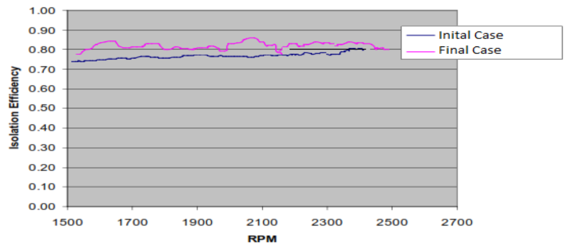


Fig. 18: Isolation efficiency vs. RPM of rear mounts

Shore hardness of both the mounts has improved the isolation efficiency a lot; especially for front mounts (Refer Table 1, 2 and Figs. 17, 18). Transmissibility and isolation factors indirectly influence the vehicle floor vibration level (Refer Table 2 and Fig. 16). Pass by noise level is within the limit for both initial and final parameter combination, but marginal improvement has been observed in the final case (Refer Table 4). In-cab noise and vehicle floor vibration results convey the impression of existence of drive shaft disturbances. Still, significant improvements have been observed in both In-cab noise and vibration level at later case. Final case has satisfied the noise criteria of In-cab noise as per the standard (refer Table 5, 6 and Fig. 16). Except inertia excitation frequency all design recommended standards for drive shafts performance are met with modified design (refer Table 3).

6. Conclusion

From the exercise it can be concluded that engine mounts have key influence on front/first drive shaft lay out and its performance; especially for AWD defense-application. In addition, the work gives an idea of how isolation efficiency changes while different parameters, such as shore hardness of front and rear mount, are tweaked simultaneously, considering the binding constraints. The solution is near-optimum considering all the constraints. It satisfies all design-recommended standards for drive shafts performance except inertia excitation frequency. Also, it meets the criteria of noise tests and engine mounts installation. Because of higher inertia excitation frequency of first driveshaft, drive train noise persists at very high (above 60 km/h) speed of vehicle. As an extension of the work, the exact speed after which drive train noise persists can be identified. Also, for complete resolution of aforementioned trouble, V-hull (body) can be modified to adjust transfer case position marginally so that inertia excitation frequency value lies within the limit by improved inclination angle. This will have impact on layout of other two drive shafts also. Another option is the use of speed limiter considering its application (off-on-road type mine protected/ambushed vehicle). This is the kind of vehicle that hardly runs at speed that is more than 65km/h.

REFERENCES:

[1] W.T. Thomson. 1982. *Theory of Vibration with Application*, Second Edition, Prentice Hall of India Private Ltd, New Delhi.

[2] P. Srinivasulu and C.V. Vaidyanathan. 2009. *Hand book of Machine Foundation*, Chapter - 7, 23rd reprint, Tata McGraw Hill publication, New Delhi

[3] F.F. Timpner. Design consideration in engine mounting, SAE Technical Paper 650093.

[4] K. Mcintire. 2006. *Installation Requirements-Engine Mounting*, AEB 21.36.

[5] R Racca. 2007. *Engine Mounting Design Considerations*, Barry Wright Corp.

[6] *Universal Joint and Driveshaft Design Manual*, AE-7, SAE Int.

[7] *Drive Shaft Installation*, Spicer, J3311-1- HVTSS, Feb 2005.

[8] M.S. Foumani, A. Khajepore and M. Durali. 2003. Optimization of engine mount characteristics using experimental/ numerical Analysis, *J. Vibration and Control*, 1121-1139. https://doi.org/10.1177/1077546030 30697.

[9] M.W. Suh, M.B. Shim, M.S. Kim and S.K. Homg. 2003. Multidisciplinary design optimization of engine mounts with consideration of the driveline, *Proc. IMechE, Part D: J. Automobile Engg*., 217, 107-114.

[10] G. Paliwal, N. Sukumar, U. Gupta and S. Roy. 2015. Design optimization of powertrain mounts for vibration isolation on heavy commercial vehicle (HCV) six cylinder inline diesel engine, *SAE Tech. Paper*, 2015-26-0127. https://doi.org/10.4271/2015-26-0127.

[11] *Automotive Vehicles - Noise Emitted by Moving Vehicles-Method of Measurement*, IS: 3028, 2007, Ed 3.2, Bureau of Indian Standards.

[12] AIS-020, *Automotive Vehicles - Interior Noise - Method of Measurement and Requirements*, India 2004.

[13] M. Horovitz. 1957-58. Suspension of internal combustion engines in vehicles, *Proc. IMechE, Part D: J. Automobile Engg*., 1, 17-35.

[14] S. Shih, J. Yruma and P. Kittredge. 2001. Drivetrain noise and vibration troubleshooting, *SAE Tech. Paper*, 2001-01-2809. https://doi.org/10.4271/2001-01-2809.