International Scientific Journal of Contemporary Research in Engineering, Science and Management (ISJCRESM)
Vol. 1, Issue 1, March-May 2016. ISSN : 2456–1134

Driveline Noise Source Identification for Intermediate Commuter Vehicle

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Abstract: Driveline torsional vibration in vehicle generates the maximum noise. The driveline noise indirectly leads to high fuel consumption. So the noise level has to be reduced. The noise level measurement were taken on driver seat, Commuter seat, engine, clutch, gearbox, silencer, centre bearing, propeller shaft, rear axle differential in static and dynamic conditions. The measurements also has been taken for idle and run-up condition. With help of the measurements, the analysis was carried out and the major source for noise level was predicted and the noise level was ranked based on the dB level. The analytical model was developed to found out the critical frequency is found. The experimentally measured frequency level matches well with analytical frequency

Keywords: Commuter, Torsional vibration, driveline noise

INTRODUCTION

Mounting purchaser demands for quieter vehicles have resulted in the requirement for better understanding of the static dynamic behavior of drivelines and for the decrease of torsional vibrations to an acceptable level. This work is particularly focused on vehicle driveline torsional vibrations wound up by engine torque fluctuations

The work presented is study of torsional vibrations of a driveline under static and dynamic torque and excited by engine torque fluctuations. It includes the design of an experimental set-up and computer software to predict static and dynamic behavior and seek efficient technical solutions. The time response is calculated and a Holzer method is used. An experimental set-up of the whole driveline has been taken from the vehicle and to allow simulation of the basic torsional phenomena occurring on the vehicle driveline under static and run-up conditions to validate numerical models.

The following torsional vibration occurring on the vehicle driveline are concerned in this work.

Idle Gear Rattle: At idle speed, the transmission is in neutral position and the clutch is engaged. Engine torque fluctuations cause collisions between gears and/or spline teeth in the gearbox and the clutch hub. Since backlashes actually exist in all geared systems and, in neutral, all gear and spline meshes are essentially unloaded, thus they are all potential rattle sources.

Drive Rattle : Drive rattle and idle gear rattle are similar; however, resonance frequencies of the driveline can affect the nature of drive rattle. Drive rattle is a phenomenon caused mainly by the non-linear torsional vibration of the driveline. This causes collision between the gear teeth in the transmission which in turn causes excitation forces, making the bearings of the gearbox vibrate. A conventional clutch damper can greatly reduce the drive rattle.

Surging: Surging results from sudden changes of engine torque actuating the driveline. The first mode shape is involved and the vehicle is subjected to damped rocking motion. This phenomenon occurs during sharp accelerations at low engine speed and results from a tip-in or back-out procedure.

Booming Noise and Hammering: Booming noise involves a mode shape concerning transmission drive axle shafts, and hammering is essentially generated by tripod joints.

Source of torsional vibration : Engine torque fluctuations involve engine angular accelerations which are the major cause of driveline torsional vibrations. These torque fluctuations are the sum of discrete torque pulses due to combustion pressure in the cylinders and cyclic inertia torques due to reciprocating pistons. In a four-cylinder four-stroke engine, the flywheel angular velocity is synchronized with the engine combustion which fluctuates repeatedly at a firing pulse frequency of twice per engine revolution. Thus, the dynamic torque is the sum of a constant mean torque and an oscillating torque whose frequency spectrum is essentially composed of the even harmonics, which are preponderant for a four-cylinder engine. Misalignment between the transmission input shaft, clutch disc and crankshaft axis of rotation can generate excitation forces synchronous with the engine speed of rotation. The high inertia of the engine flywheel tends to reduce the magnitude of angular accelerations which are particularly high at idle speed and in diesel engines. Unfortunately, it also reduces the acceleration of the vehicle, resulting in sluggish Performance.

SCOPE OF THE PROJECT

Driveline torsional vibrations fall into two broad categories, namely gear rattle and driveline vibration. Gear rattle is classified as quasi-steady impulsive noise. The basic source is usually a component or sub-system with clearance Non-linearity which includes backlashes, multi-valued stiffness characteristics, dry friction, hysteresis, etc. Driveline vibrations create noises emanating from the driveline when loaded. The resonant driveline behavior can amplify gear motion at certain engine speeds of rotation\ thus requiring frequency analysis.

Both experimental and theoretical researches have been focused on the torsional vibrations of the complete driveline in controlling the natural frequencies of the system. Also analytical and experimental results have demonstrated that a system approach is necessary to understand torsional resonant phenomena occurring within certain ranges of operating speeds. Useful experimental research has been conducted to analyze these phenomena with emphasis placed on the analysis of driveline resonance.

- Quantifying Driveline Noise Levels in Vehicle Aggregates like driver ear level, Commuter ear, engine, gear box, clutch, centre bearing, propeller shaft, rear axle differential and silencer.
- To study the effect of change of vibration level in vehicle on static and dynamic condition.
- To Rank the predominant sources, to generate more noise in steady state and Run-up condition.

EXPERIMENTAL STUDY

An experimental set-up has been designed and built to identify torsional phenomena occurring on a vehicle driveline and validate numerical models. The Figure 1 shows the noise measurement locations on vehicle driveline assembly.

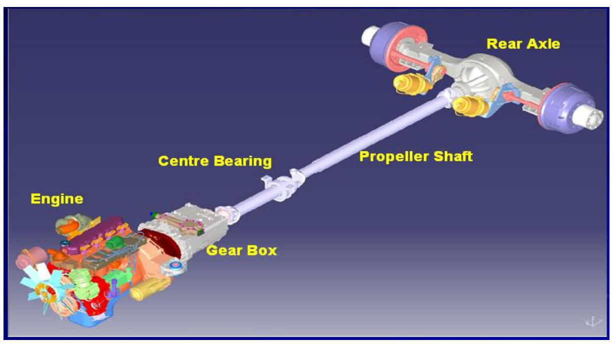


Figure 1 Experimental Set-up

The driveline is composed of the Engine, flywheel, clutch, gear box, transmission drive axle shaft, equipped with a toothed wheel and an inertia corresponding to the rear wheel through propeller shaft. The supporting frame receives mounts driveline components and measurement devices while microphones are inserted between two torsional rigid couplings upstream and downstream from the gearbox. A low-pass mechanical filter is composed of a low stiffness torsional shaft associated with a high value inertia. The noise levels were measured using microphone at specified location as shown in the figure 2 to 3.

Figure 2 : Near Propeller shaft

Figure 3 : Near Centre Bearing

Figure 4 : Driver Ear level

Figure 5 : Commuter Ear level

DATA ACQUISITION MEASUREMENTS AND SIGNAL PROCESSING

All data are conditioned and simultaneously recorded by using high-end data acquisition system at sampling rate 20KHz which is controlled by an IBM computer. Using the analysis software storage and time signal processing were carried out. By using the above acquisition system, universal files are created for each test. The technique developed within the specific signal processing software is based upon frequency demodulation by means of the Hilbert transform combined with numerical filters. The instantaneous phase of the Hilbert transform of a sinusoidal carrier signal is proportional to the rotational angle of the gear.

INSTRUMENTS USED

16 ¼"microphones – To measure the noise levels

- Range 0 to 120 db

Data acquisition system

- 16 channels
- 20460 sampling rate
-

STATIC ACQUISITION: The noise levels were at specified locations in vehicle static and engine idle (600rpm) running condition. The same were measured in engine run up condition i,e engine accelerating from 600 to 2400rpm. Along with noise engine rpm speed variation also recorded.

DYNAMIC ACQUISITION: The noise levels were at specified locations in vehicle running condition at steady speed on asphalt road high way. Steady speeds were maintained as 30,40 & 50 Kmph. The noise levels were measured in vehicle normal accelerating condition also.

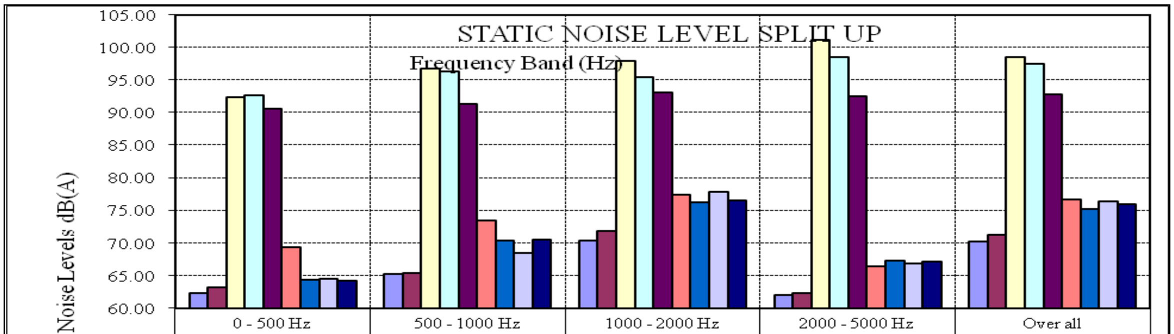
DATA ANALYSIS

STATIC DATA : Acquired Time data This is time data was converted into Frequency domain data using Fast Fourier Transformation. Acquired Time data in vehicle static Engine accelerating was converted into Frequency domain data using Fast Fourier Transformation as water fall analysis. Driver and co driver are mostly disturbed by the noise (around 70dB) in static level, so the noise observed from this position is considered for optimization. This noise is generated around 1000-2000 Hz shown in band split figure 2.20. In the frequency level the maximum noise contributed sources are clutch (97.90 dB), Engine (95.40dB), silencer (93.10dB). The ranking of the noise are follows for driver ear and co-driver ear

- Clutch-I (97.90 dB)
- Engine-II (95.40 dB)

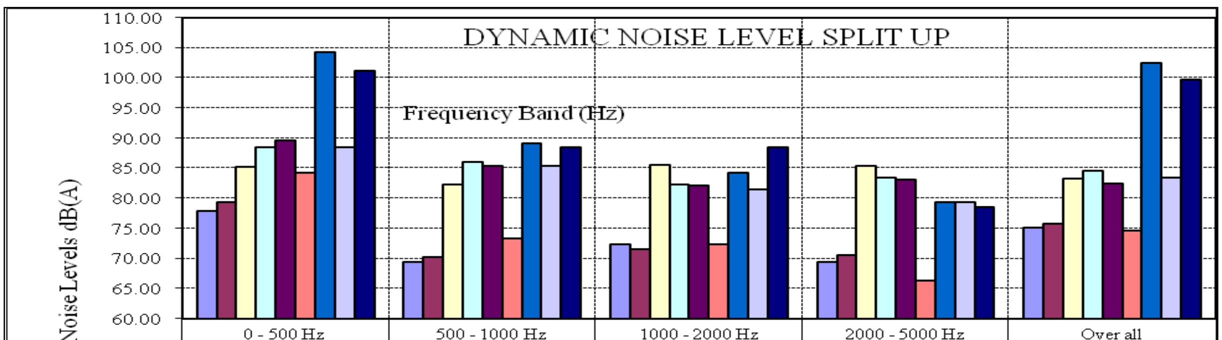
Silencer-III (93.10 dB)

DYNAMIC DATA: Driver and Commuters are mostly disturbed by the noise (around 75dB) in dynamic level, so the noise observed from this position is considered for optimization. This noise is generated around 0-500 Hz shown in band split figure 7. In the frequency level the maximum noise contributed sources are propeller shaft (104.20 dB), Centre Bearing (101.20dB).



	0 - 500 Hz	500 - 1000 Hz	1000 - 2000 Hz	2000 - 5000 Hz	Over all
Driver Ear	62.40	65.20	70.40	62.10	70.20
Co-Driver Ear	63.20	65.40	71.90	62.40	71.20
Clutch	92.28	96.79	97.90	101.10	98.49
Engine	92.58	96.30	95.40	98.45	97.43
Silencer	90.66	91.29	93.10	92.55	92.79
Gear Box	69.40	73.40	77.40	66.40	76.70
Propeller shaft	64.40	70.40	76.30	67.30	75.20
Rear Axle	64.50	68.50	77.90	66.90	76.40
Center Bearing	64.20	70.50	76.60	67.10	75.90

Figure 6 Overall Band Split Data - Static



	0 - 500 Hz	500 - 1000 Hz	1000 - 2000 Hz	2000 - 5000 Hz	Over all
Driver Ear	77.90	69.40	72.30	69.40	75.10
Passenger Ear	79.40	70.20	71.50	70.50	75.80
Clutch	85.20	82.30	85.50	85.40	83.20
Engine	88.40	86.10	82.30	83.40	84.50
Silencer	89.60	85.40	82.20	83.10	82.40
Gear Box	84.20	73.40	72.40	66.40	74.60
Propeller shaft	104.20	89.10	84.20	79.30	102.40
Rear Axle	88.40	85.30	81.40	79.40	83.40
Center Bearing	101.20	88.40	88.50	78.50	99.70

Figure 7 Overall Band Split Data - Dynamic

The ranking of the noise are follows for driver ear and Commuter ear

- Propeller shaft-I (104.20 dB)
- Center Bearing –II (93.10 dB)

ANALYTICAL TORSIONAL MODEL

INTRODUCTION

Since there are a number of driveline configurations resulting from in-line and transverse engines front and rear wheel drive direct and indirect gearboxes etc. computer software is required to analyze a specific driveline configuration and a Holzer approach is used. The driveline is represented as a chain of torsional simple branched systems describing the various components and the model chosen includes linear basic elements and three non-linearities arising from Clearances, non-linear stiffness, and clutch dry friction effects. The representation of the torsional model the driveline is shown in Figure 8. It allows a linear modeling approach in analyzing the natural frequencies and the steady state behavior of the driveline, and in analyzing of the transient phenomena generated by the non-linearities.

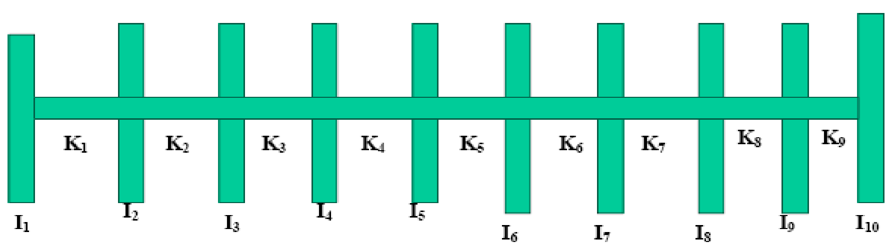


Figure : 8 Torsional Model

I_1	- Fan+ pulley inertia	K_1 to K_5	- Engine shafts
I_2,I_3,I_4, I_5	- Cylinder inertias		
K_6	- Clutch stiffness		
I_6	-Inertia of flywheel+clutch cvr. assy	K_7	- Gear Box stiffness
I_7	- Inertia of clutch		
K_8	- Propeller shaft stiffness		
I_8	- Inertia of Gearbox		
K_9	- Axle Shaft stiffness		

I_9 - Inertia of Differential + pinion

I_{10} - Inertia of Wheel assembly

HOLZER METHOD

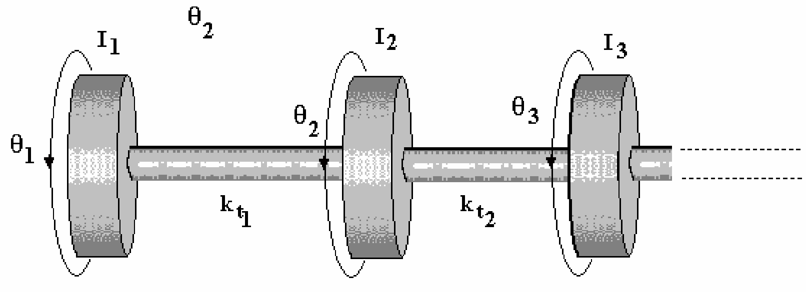


Figure 9 : Torsional Oscillations with Multiple Modes

Torsional Oscillations With Multiple Modes
The toque balance gives
Let disc 1 twist relative to disc 2.
$I1\ \alpha_1 + kt_1(\theta_1 - \theta_2) = 0$
Let disc 2 twist relative to discs 1 and 3.
$I2\ \alpha_2 + kt_1(\theta_2 - \theta_1) + kt_2(\theta_2 - \theta_3) = 0$
Let disc 3 twist relative to disc
$I3\ \alpha_3 + kt_2(\theta_3 - \theta_2) = 0$
For simple harmonic motion we may substitute $\omega^2\theta = -\alpha$
$I_1\ \omega_1^2\ \theta_1 = kt_1(\theta_1 - \theta_2)$
$I_2\ \omega_2^2\ \theta_2 = kt_1(\theta_2 - \theta_1) + kt_2(\theta_2 - \theta_3)$
$I_3\ \omega_3^2\ \theta_3 = kt_2(\theta_3 - \theta_2)$
$I_1\ \omega_1^2\ \theta_1 + I_2\ \omega_2^2\ \theta_2 + I_3\ \omega_3^2\ \theta_3 = 0$
For any number of discs this may be generalized as $\Sigma I\ \omega_2\ \theta = 0$
HOLZER's method of solution proposes that we assume any value of ω and make θ_1= 1 and calculate all the other deflections.
If this was continued the pattern for any number of discs would be as follows.

$$\theta_2 = \theta_1 - \frac{\omega^2}{k_{t1}} I_1\theta_1$$

$$\theta_3 = \theta_2 - \frac{\omega^2}{k_{t2}}(I_1\theta_1 + I_2\theta_2)$$

$$\theta_4 = \theta_3 - \frac{\omega^2}{k_{t3}}(I_1\theta_1 + I_2\theta_2 + I_3\theta_3)$$

And so on for many as exist.
The torque to deflect disc 1 by θ_1 is $\omega^2\ I_1\ \theta_1$
The torque to deflect disc 2 by θ_2 is $\omega^2 I_2\ \theta_2$
The torque to deflect disc 3 by θ_3 is $\omega^2\ I_3\ \theta_3$
And so on for as many shaft section that exist.

Hence

$T_1 = \omega^2 I_1 \theta_1$
$T_2 = T_1 + \omega^2 I_2 \theta_2$
$T_3 = T_2 + \omega^2 I_3 \theta_3$
And so on for as many shaft section that exist.

CALCULATION PROCEDURE

A model of the vehicle drive line was created as Mass-Elastic system in system and solved to get the natural frequencies.

The driveline components are reduced to equivalent lumped masses and shafting to bring out a continuous shaft and lumped mass system. Holzer method has been used in determining the natural frequencies of the system derived representing the entire driveline.

Since here the vehicle combination 5 gear ratios had to be considered and the model studied and system modeling was to be time consuming it was studied through Microsoft-Excel with macro program.

Assumptions
Engine inertia = I Crank/4 +1/2 (Reciprocating mass (M)* crank radius (a))=0.02886Kg.m^2 [4]

I_{Crank}=0.0926 Kg.m2 M=3.578 Kg, a=56.5mm
Propeller shaft inertia is not considered.
Damper pulley inertia, propeller shaft inertia is not considered.

Inertia variation of reciprocating masses is not considered.
Modulus of rigidity G=79 MPa
Damping is neglected.

MODEL CREATION
The driveline is modeled as a mass elastic system (refer:Figure.9) consisting of a series of masses connected by shafts. An input file is to create with the values of moment of inertia and Torsional stiffness of the masses and shafts respectively. The holzer program is then run and from the output file frequency values different orders are noted down.

INPUT DATA

Table 1 Engine

	Moment of inertia (kg/m^2)	**Stiffness of the shaft (N m)**
cy1	0.02886	6.48E+06
cy2	0.02886	6.48E+06

cy3	0.02886	6.48E+06
cy4	0.02886	6.48E+06
Flywheel	0.307	1.21E+04

Table 2 Clutch

Clutch	**Moment of inertia (kg/m^2)**	**Stiffness of the shaft (N m)**
G1	0.375	6.17E+06
G2	0.375	6.38E+06
G3	0.375	7.01E+06
G4	0.375	8.74E+06
G5	0.375	1.33E+07

RESIDUAL TORQUE

Torque balance equations gives.

$I_1 \omega_1^2 \theta_1 + I_2 \omega_2^2 \theta_2 + I_3 \omega_3^2 \theta_3 = 0$

For any number of discs this may be generalized as $\Sigma I \omega_2 \theta = 0$

-refer Holzer, method.

$$T = I \omega^2 \theta$$

$$T_{actual} = T_{translated} + T_{applied}$$

$$\theta_2 = \theta_1 - \frac{\omega^2}{k_{t1}} I_1\theta_1$$

(Since $\theta_3 = \theta_2 - \frac{\omega^2}{k_{t2}}(I_1\theta_1 + I_2\theta_2)$

$$\theta_4 = \theta_3 - \frac{\omega^2}{k_{t3}}(I_1\theta_1 + I_2\theta_2 + I_3\theta_3)$$

Initially by setting the value of ω and θ as 1 by using the holzer's method the initial torque is calculated. Gradually by incrementing the value of ω by 1 the corresponding θ value is taken and each torque value calculated till maximum engine operating speed. ω value taken upto 299. It covers the maximum engine operating speed.(600-2400rpm). Example Calculation for forth gear shown below. The Table 3 shows the calculation; Figure 10 plots the curve for residual torque for speed increment. Table 5 gives the fundamental torsional natural frequency of the system in the forth gear.

Table 3 Gear Engagement IV

	$\omega (rad/s)$	I (kg/m^2)	$I\omega^2$ (Nm)	k_t (Nm)	θ (rad)	$I\omega^2\theta$ (Nm)	$\sum I\omega^2\theta$ (Nm)	$(\sum I\omega^2\theta)/k_t$ (Nm)
cy1	299	0.02886	2580.113	6477979	1	2580.113	2580.113	0.000398
cy2		0.02886	2580.113	6477979	0.999602	2579.085	5159.198	0.000796
cy3		0.02886	2580.113	6477979	0.998805	2577.03	7736.228	0.001194
cy4		0.02886	2580.113	6477979	0.997611	2573.949	10310.18	0.001592
flywheel		0.307	27446.11	12128.38	0.996019	27336.86	37647.03	3.104045
clutch		0.375	33525.38	8743857	-2.10803	-70672.3	-33025.3	-0.00378
G.B		0.0254	2270.785	8753.183	-2.10425	-4778.3	-37803.6	-4.31884
Diff+pinion		0.01061	948.5319	515.142	2.214591	2100.61	-35703	-69.3071
Wheel		0.104385	9332.13		71.52166	667449.5	631746.5	

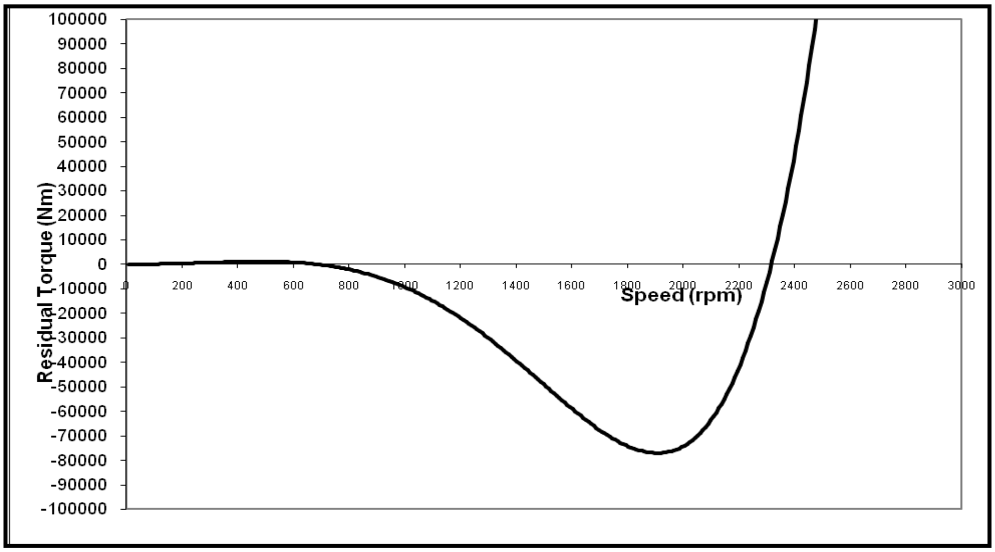


Figure 10 Torsional Natural Frequency Gear IV

Table 5 Natural Frequency Gear IV

S.no	Rpm	Nm	Mean Rpm
1	687.2727	5.305308	
	696.8182	-120.77	692.045
2	2310	-3041.9	
	2319.545	1316.138	2314.77

DISCUSSIONS

The natural frequencies of the driveline system in different gear ratios for our system are tabulated below based on the above caluacations. Refer table 6.

Table 6 Natural frequencies of the system

1st gear	rad/s	rpm	Hz
$\omega_{\square}$	68.44	653.864	10.9

ω_2	245.3	2343.41	39.06
2^{nd} Gear			
$\omega_{\square}$	68.44	653.864	10.9
ω_2	244.3	2333.86	38.9
3^{rd} Gear			
$\omega_{\square}$	69.44	663.409	11.06
ω_2	244.3	2333.86	38.9
4th Gear			
$\omega_{\square}$	72.43	692.045	11.53
ω_2	242.3	2314.77	38.58
5th Gear			
$\omega_{\square}$	77.43	739.773	12.33
ω_2	236.3	2257.5	37.63

$$\text{Frequncy} = \frac{\omega}{2\pi}(\text{hz})$$

$$\omega = \frac{2\pi N}{60}(\text{rad}/\text{s})$$

CONCLUSION

The Noise levels were measured in static and dynamic condition. Water fall and band analysis were carried out on static noise level and found that at the noise level rankings at Driver & Co-driver was around 70db and at Commuter ear level the noise in around 65db. From frequency analysis we inferred that the Noise from Clutch assembly is the major contributor at Driver Ear level. These have been observed in all engine speeds (600 to 2600 rpm).

Based on dynamic noise data frequency analysis the noise at driver ear level are high at 4^{th} gear around 40Kmph during acceleration. The major contributors for this noise levels are Clutch, Engine and Silencer.

The analytical model was developed based on Holzer method, and the critical frequencies were plotted for each gear. Based on gear duty levels it was found that the most of the time the vehicle will be operating in 40kmph with 4^{th} gear. Considering these two results we conclude the

4th gear around 2200~2400 rpm has higher torsional frequencies which produce higher noise levels.

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