Design and Analysis of Clutch Pre Damper to Reduce Neural Gear Rattle in Automotive Driveline

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***Abstract*— The phenomenon of Gear rattle, caused by the torque fluctuations of the engine, is produced by backlash and vibro-impacts of unloaded gear pair tooth inside the gearbox. Gear rattling noise is one of the major problems facing the industry and the car industry. Gear rattle can be heard during neutral condition of vehicle called as neutral gear rattle or idle rattle. This article contains design and analysis of clutch disk pre damper spring. This work involves modeling of MATLAB Simulink model for clutch system. Noise, Vibration and Harshness (*NVH*), is the study and optimization of the noise and vibration characteristics for improving the system performance and comfort in passenger and heavy duty vehicles. Paper presents NVH experimentation and testing on TATA Ace vehicle.**

Index Terms*—Neutral Gear rattle, clutch disk pre damper spring, MATLAB-Simulink model, NVH etc.

I. INTRODUCTION

The sources of gear rattle in automotive transmission and other components are clearance non-linearities, which include, multi-valued springs, hysteresis, etc. Periodic vibro-impacts are generated because of the single sided or double sided impacts [8]. A vehicle with internal combustion engine generates torque with fluctuations due to its working principle. These fluctuations and also variations on the mean torque generated by the engine induce vibrations into the system, leading to several NVH (Noise, Vibration and Harshness) phenomena [14]. Vehicle power-trains are systems designed to transmit power from the engine, usually an internal combustion engine, to the body of the vehicle, providing motion. This system is usually made of specific components such as the engine itself, a flywheel, a gearbox, clutch systems, differential gear and shafts. The clutch is the element which transfers power from engine to gearbox. Drexl (1988) defined that it enables a vehicle to start from standstill and it also allows interruption of power flow to stop and change gear.[4]

II. LITERATURE REVIEW

Padmanabhan et al.[7], presented a state of the art in the modeling of transmission rattle. Specifically they have developed a step-by-step approach to address the rattle problem. Miyasato et al.[4], Analysed idle rattle with a systemic approach. It was seen that clutch parameters do influence rattle index. Modification on the clutch stiffness parameter had strong influence on the gear rattle intensity. Brancati, et al.[9], The analysis was conducted on an unloaded gear pair subjected to a multi-harmonic excitation had evidenced interesting aspects in relation to the gear rattle phenomenon by examining the gear relative angular motion, both in time and frequency domains.

Research gap found is no experimentation has been done by Miyasato et al. [4]. A test rig was used, actual conditions were not included in paper by Brancati, et al.[9] The main work done till now is in modification of gear geometry, or modifying flywheel i.e., by using a dual mass flywheel. It has also been seen that various clutch parameters do influence gear rattling, but this was not attempted by any researcher. Study of induced fluctuation in drive line is done, which is helpful for modeling of real world rattling problem.

III. GEAR RATTLE

The gear rattle phenomenon is associated with the characteristic noise that unselected impacting gears radiate to the environment. The phenomenon occurs at low impact forces, qualitatively similar to the noise produced, when a marble hits a tin can.

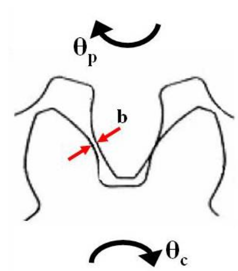

Fig1. Gears in meshing vibrate due to backlash (b)[4]

The problem is induced by engine order vibrations in the presence of backlash in meshing pairs and is particularly troublesome in vehicles with diesel engines, because of higher output torques [14]. Backlash (b) is the technical name given to clearances found between the gear tooth pairs (Fig. 1). It occurs due to the dimensional tolerances chosen during the gear manufacturing stage, allowing also a proper lubrication of the contact surfaces. Low values of backlash are desirable, but it cannot be reduced significantly without component wear degradation.[4] It can be noticed on manual transmission vehicles in neutral condition (idle rattle) related to the engine firing frequency. These collisions result from torque fluctuations transmitted from the engine. The impact force on a driven gear during a collision changes its speed so that a relative motion develops between the mating gears. Rattle is also described in literature referring to a condition where high levels of vibrations are found in the transmission.[2]

IV. MATHEMATICAL MODELING AND DESIGN

Automotive gear rattle phenomenon belongs to a general class of problems where periodic vibro-impacts are observed in noise and vibration signals. Dynamic excitation is usually at lower frequencies, say engine torque pulsations and the mean load of the system is usually very small. Also in some cases the load reversals may also take place within a cycle thereby introducing conditions for single sided or double sided impacts. Drag or mean friction torque also plays an important role along with dynamic system parameters.

As per the literature available, the rattle phenomenon can be represented in the following manner (as in fig.2),

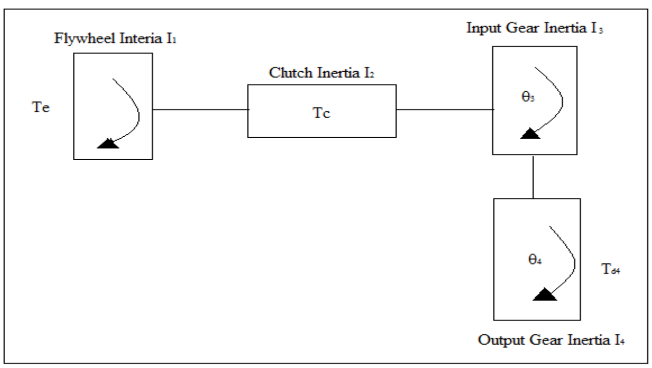

Fig. 2:Schematic representation of components in gear rattle problem

Various criteria used for identifying the level of permeable rattle**.** Following are some criteria (with reference to fig. 2:

(a) Criterion based on relative displacement between gears (x_r) [8]:

$$x_r(t) < x_b : \text{rattle}$$
$$\geq x_b : \text{no rattle.}$$

Where; x_b = backlash

(b) Criterion based on angular acceleration of output gear $\ddot{\theta}_4$

$$T_{d4} - I_4\ddot{\theta}_4(t) \leq 0 : \text{rattle}$$
$$>0 : \text{no rattle} \qquad (1)$$

Where, T_{d4}= Drag torque at output gear.

(c) Criterion based on angular acceleration of input gear (θ_3):

When the gears are in contact, the elastic deformation is very small and therefore it can be approximated as:

$$\ddot{\theta}_4(t) \simeq -(R_3/R_4)\ddot{\theta}_3(t) \qquad (2)$$

Using equations (1) and (2), one can define the approximate rattle level (β) as,

$$\beta(t) \simeq (I_4 R_3 / T_{d4} R_4)\ddot{\theta}_3(t) \qquad (3)$$

Where, R_3 and R_4 are radius of the input gear and output gear respectively.

Hence, the rattle criterion is approximately given as [8],

$$\beta(t) \leq -1 : \text{rattle}$$
$$> -1 : \text{no rattle}$$

For a practical gearbox, x_b and T_{d4} may not be constants, and the $\ddot{\theta}_3(t)$ signature may be fairly complicated. Hence, it is more convenient to calculate the root-mean-square (rms) value, β_{rms}, which is related to the energy contents of the motion. Assuming a harmonic relationship for $\ddot{\theta}_3(t)$, the criterion obtained as,

$$\beta_{rms} \geq 0.707 : \text{rattle}$$
$$< 0.707 : \text{no rattle,}$$

where,

$$\beta_{rms} = \sqrt{\frac{1}{T}\int_0^T \beta^2(t)\,dt} \quad ,$$

T = time window

The rattle level L_β (in dB) and the corresponding rattle criterion are defined as follows:

$$L_\beta = 20\log_{10}(\beta_{rms}/0.707)\ \text{dB},$$

$$L_\beta \geq 0\ \text{dB: rattle}$$
$$< 0\ \text{dB: no rattle.}$$

A. *Rattle Index [8]:*

A rattle index (RI) based on the value of the unloaded gear acceleration, divided by the acceleration of flywheel;

$$RI = \frac{\ddot{\theta}_{\text{gear (RMS)}}}{\ddot{\theta}_{\text{Flywheel (RMS)}}}$$

RI is used for comparison between two different conditions of the system for which it is calculated. The state of system with minimum value of RI should be preferred.

V. PROBLEM FORMULATION

In automotive driveline, during idling there is a rattling noise of gears which causes irritation to the driver and passengers. Gear rattle may further lead to improper functioning of gears and failure of gears due to repetitive rattle activity. TATA Ace Vehicle facing problem of neutral gear rattle, investigated by R&D Mahinra sona Ltd. NVH testing FFT graphs as shown bellow in figure 3. Frequency observed is 15.5969 and maximum acceleration is 0.09660. Vibration isolation percentage is calculated from testing data it was found -28.23. Negative value of isolation percentage indicates that vibrations coming from engine not isolated but they are integrated.

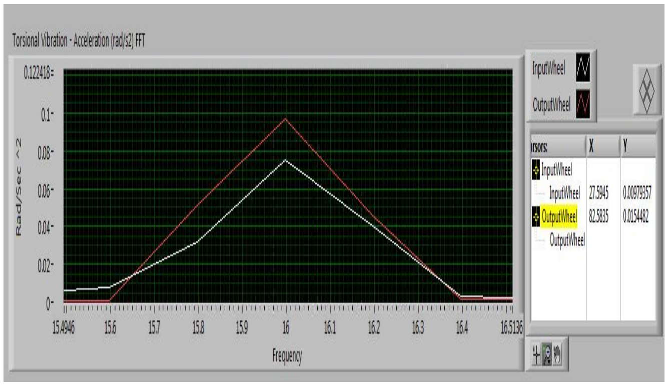

Fig.3 Frequency vs Acceleration graph in FFT testing

In idling condition, the rotating parts are engine, flywheel, clutch and gear-shafts. Torsional vibrations are induced only by intermittent combustion engine, present in vehicle, having total torque T_e. This torque includes, mean torque (T_m) and fluctuating torque (T_p) and is represented by following equation,

$$T_e(t) = T_m + T_p(t)$$

The fluctuating torque is vibratory part of the torque that causes rattling phenomenon T_p. [8]. Therefore, from the literature available following method is used,

i. Formulate a suitable model for simulation (preprocessing stage),
ii. Select a suitable numerical method to obtain solutions (processing stage),
iii. Choose or develop performance indices to evaluate and optimize various design parameters in order to reduce noise and vibration levels (post processing stage).

Although the study of the automotive rattle problem has been divided into three distinct stages so that various issues can be highlighted, it will become clear that these stages are often inter-dependent.

VI. SIMULATION

Software used for simulation is MATLAB- Simulink. Mathematical-model consists of engine which has input parameters like torque and frequency. Block representing flywheel and gearbox consists of inertia I_1 and I_2 respectively. The clutch is represented by a torsional spring, as its stiffness is one of the most important parameter that helps in reduction of rattle phenomenon. Two motion sensors are connected, one at flywheel side and the other at gearbox side so as to measure and monitor the motions before and after clutch. Motion sensors have output as angle, velocity and acceleration**.**

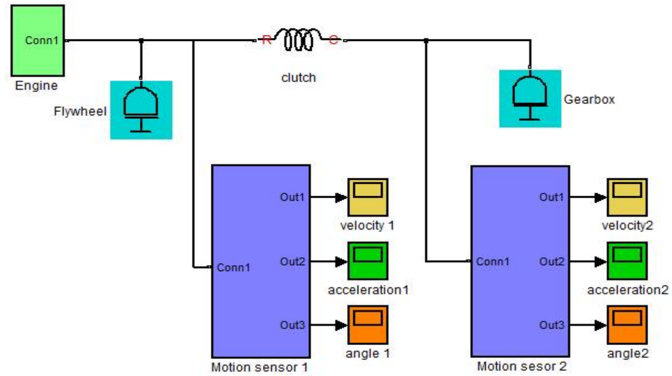

Fig.4 Basic driveline model with motion sensors

After constructing Simulink model as above, only unknown factor remaining is k_t i.e., spring stiffness of the torsional spring. The torsional spring may also be called as “pre-damper spring”, as it will be used to dampen vibration in the idling stage.
For deciding the upper limit of k_t, resonance condition is considered. Resonating spring rate is calculated as[3],

ω= Forcing Frequency

ω_n = Natural Frequency

$$\omega = \frac{2\Pi N}{60}$$

N = 800 rpm (idling rpm)

$$\therefore \boxed{\omega = 83.776 \text{ rad/s}^2}$$

at resonance

$$\omega = \omega_n$$

$$\therefore \omega = \sqrt{\frac{k_t(I_1 + I_2)}{I_1 I_2}}$$

$$\therefore \boxed{k_t = 0.54 \text{ Nm/}^0}$$

Entering the above value of stiffness k_t in program gives a graph as shown in fig. 5. This shows a resonating nature,

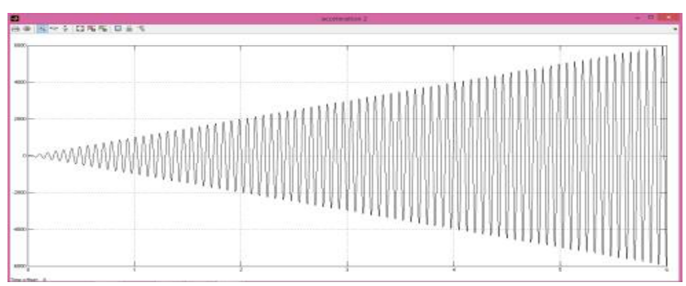

Fig. 5: Final acceleration of drive line showing resonance with spring stiffness (k_t) =0.54 Nm/^{0}

Now, just reducing the spring rate by 0.04 Nm/^{0} and again running the program, graph shown in fig. 6 was obtained.

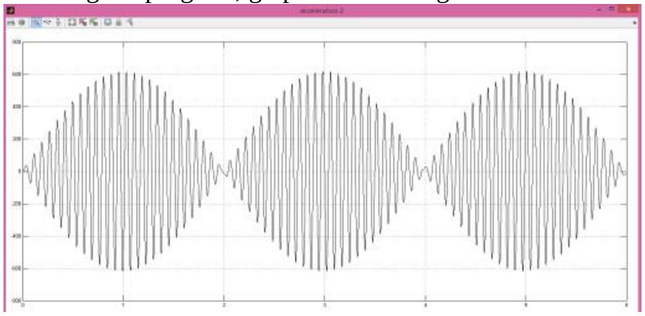

Fig. 6: Final acceleration of drive line showing beating with spring stiffness (k_t) =0.5Nm/^{0}

The above fig. 6 represents the beating phenomenon. This phenomenon has similar effects as that of resonance; hence, this should be avoided. Beating occurs only when natural frequency of system is very near to forcing frequency. Reducing resonating stiffness by a small amount makes the system frequency near to forcing frequency. Now, the spring stiffness was increased drastically i.e, k_t= 100 Nm/^{0}, gives a graph as shown in fig. 7. This graph shows rattling phenomenon as observed in various literature.

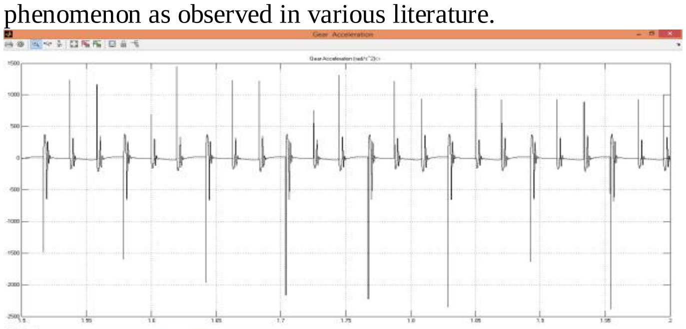

Fig.7: Final acceleration of drive line showing rattling with spring stiffness (k_t) =100Nm/^{0}

Above figures 5, 6 and 7 ensures that Simulink model shows correct results in resonating, beating and rattling conditions. Further Simulink model is modified by adding ratting criteria. This helped to determine the exact spring rate for which there is acceptable rattle condition.

ODE 45 (Ordinary differential equation solver) was used to solve the model with tolerance 1e^{-9} and sample time 0.0001sec. An idle stage angle of 9^0 is provided by considering the space available for fitment of spring. Hysteresis of 1.2 Nm is provided to limit the amplitude of vibration during starting and stopping of the engine. Backlash between mating gears was measured and found to be 0.18^0. Subsystem and subsystem 1 consist of the mathematical calculations for various rattling criteria. An audio device is connected to the output acceleration of the driveline. With this device, rattling sound can be heard, thus making the model more close to the realistic system. A block is added which contains graph of both input as well as output side acceleration. This helps in comparing both accelerations.

VII. SIMULATION RESULT

Iterations were performed on the above model by changing the spring rate for determining the exact spring rate. Although, value of resonating spring is known but for determining spring rate for minimum rattle condition, iterations are started from 15.44 Nm/^{0}. This is the spring rate of main damper spring present in original clutch. Following table represents value of spring stiffness along with the rattling criteria (rattle index, β*rms and L β*):

TABLE I
Iterations of stiffness

k_t (Nm/^{0})	RI	β *rms	L β * (dB)
15.44	18.22	2.16	9.724
7.5	19.43	2.31	10.29
3	19.8	2.39	10.58
1.5	19.02	2.31	10.3
0.75	18.66	2.27	10.15
0.5	18.62	2.27	10.14
0.4	3.74	0.40	-4.81
0.3	3.41	0.36	-5.63
0.2	3.15	0.34	-6.32
0.1	3.13	0.33	-6.39

By observing table I, selected spring stiffness is 0.2 Nm/^{0}, as it fulfills all the rattling criteria. Lower spring rate can also be selected, but it may create a practical difficulty of spring manufacturing.

Natural frequency of the system by new spring rate is,

$$k_t = 0.2\ \text{Nm}/^0 = 11.46\ \text{Nm/rad}$$

$$\omega_n = \sqrt{\frac{11.46 \times (0.0707 + 0.0047)}{(0.0707 \times 0.0047)}}$$

$$= 20.15\ \text{rad/s}^2$$

Hence, the natural frequency of the system is shifted to 20.15 rad/s^2 from 83.77 rad/s^2.

Following are results obtained by running the program with spring stiffness 0.2 Nm/^{0}, as shown in fig. 8 and fig. 9.

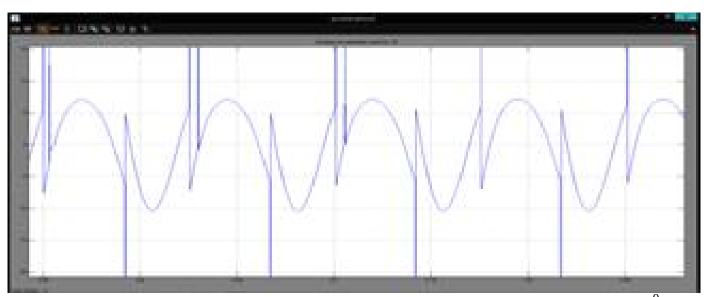

Fig.8: Final acceleration of drive line when spring stiffness (k_t) =0.2 Nm/^{0}

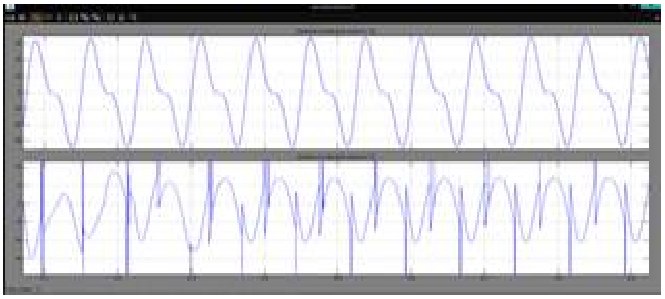

Fig.9: Acceleration of flywheel & Gear box output shaft.

Above figure represents acceleration of flywheel and gearbox output shaft, it can be clearly seen that value of acceleration is reduced to 25 rad/sec^2. This represents the reduction of vibration at output.

VIII. NVH EXPERIMENTATION:

An experiment was conducted on Tata Ace (light commercial vehicle), by measuring acceleration near flywheel and gearbox input shaft during idling; fig. 10 is the conventional clutch plate sample. Pre Damper clutch disk sample shown in fig.11

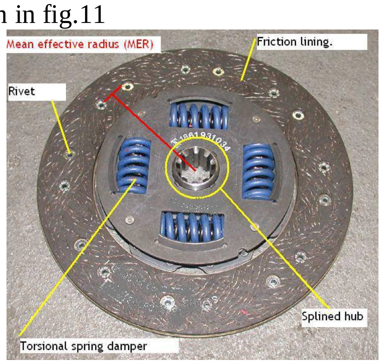

Fig.10 Conventional clutch disk

Experiment was carried on both conventional as well as pre-damper clutch plate. Pre-damper spring with dimension mentioned in table 2 was manufactured and incorporated in clutch assembly as shown in fig. 7.[10,11,12]

TABLE II
Final damper spring diamensions

d (mm)	Do (mm)	Di (mm)	Lf (mm)	Ls (mm)	La (mm)	Δmin (mm)
1.1	6.4	3.5	12.75	8.3	11.7	1.05
C	**D (mm)**	**Nt**	**Na**	**k_t(Nm/^{0})**	**K (Nm)**	**Δmax (mm)**
4.81	5.3	8	6	0.25	1.8	3.65

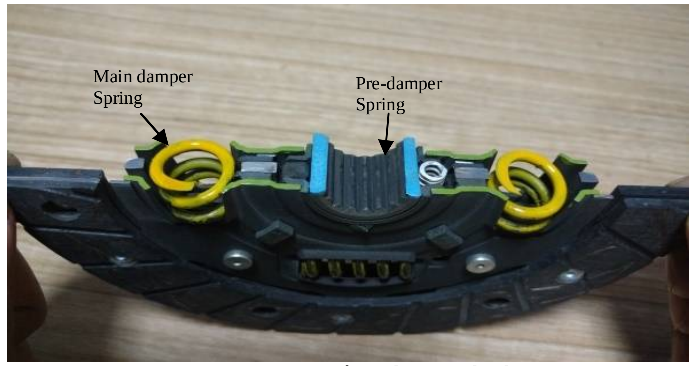

Figure 11: Cross-section view of Pre-damper Clutch (Prototype)

For measurement of acceleration, two proximity sensors were placed near flywheel and gearbox input shaft respectively. The output of these sensors was taken by a data acquisition system which was displayed in form of FFT.

Following fig 12 shows pictorial representation of the experimental set-up.

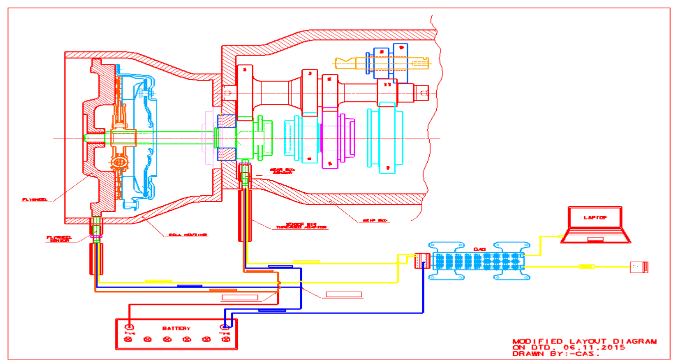

Fig.12 Representation of Experimental Setup

IX. RESULT AND DISCUSSION

A. FFT Result of NVH testing on Convention DP:

FFT of the system was obtained by experiment. Fig 13 shows FFT result obtained by experiment respectively for conventional clutch plate. It was observed that, nature of FFT graphs is similar for flywheel and gearbox. The peak of 1st and 2nd harmonic for flywheel and gear box are almost same. This means that, transmission takes place without any isolation.

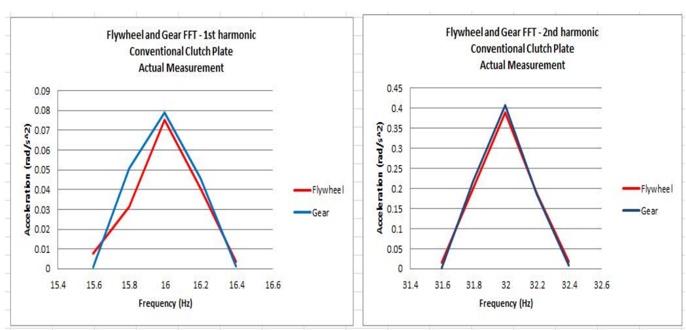

Figure13 Actual measurement for Conventional DP

B. FFT Result of NVH testing on Pre Damper DP:

Fig.14, represents FFT of prototype i.e., pre-damper clutch plate. It was observed in experimental result, that there is reduction of peak for both harmonic. This confirms isolation done by pre-damper spring.

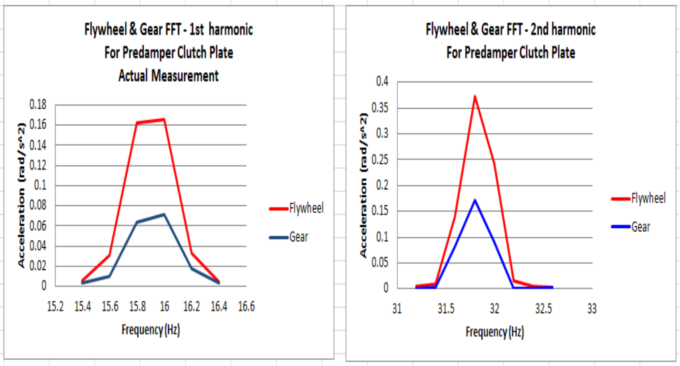

Figure 14: Actual measurement for Pre-damper DP

IX.CONCLUSION

Referring to transmissibility versus frequency ratio graph from Mechanical Vibrations by Grahm Kelly, the range where $r<\sqrt{2}$ is amplification range and range. Where $r>\sqrt{2}$ is called as isolation range. Vibrations can be isolated only when transmissibility T<1. This occurs when $r>\sqrt{2}$ [15]. For conventional clutch plate, frequency ratio, r, is 0.18, which is in the amplification range and hence, vibrations cannot be isolated. For effective isolation, natural frequency of the system is shifted from 548 rad/s^2 to 62 rad/s^2 by changing the torsional stiffness from 15.44 Nm/^{0} to 0.2 Nm/^{0}. Thus, frequency ratio, r, is changed from 0.18 to 1.61, which is in the isolation range.

Actual measurement on vehicle shows that for conventional clutch plate, there is neutral rattle, and when it is replaced by pre-damper clutch plate neutral rattle vanishes.

Comparison of results from actual measurement is as follows:

TABLE III
Comparison of Vibration Isolation

Vibration Isolation %	**1st Harmonic**	**2nd Harmonic**
Conventional DP	-4.62%	-4.76
Pre Damper DP	56.44%	63.05%

In conventional DP Vibration isolation % is negative value that menace there is no isolation of vibrations. Vibration isolation result obtain from Pre damper DP shows that in 1st and 2nd harmonic 56.44% and 63.05% isolation of vibrations takes place respectively.

X. ACKNOWLEDGMENTS

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