Abstract — Vibro-impacts Induced by backlash between meshing gears leads to excessive vibration and noise in many geared rotating system. A five-speed manual transmission of a light commercial vehicle is examined, with focus on the neutral gear rattle problem. An non-linear model of physical system was developed with help of MATLAB-Simulink software. A few rattle criteria proposed in early research are used as a parameter of compression between two states of system. Modified design of clutch is suggested for reducing neutral gear rattle in vehicle. This modification includes addition of spring (pre-damper) in clutch system so as to absorb the vibration in driveline.

Index Terms — Neutral gear rattle, MATLAB-Simulink, pre damper.

I. INTRODUCTION

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. In this work, vehicles with internal combustion engines were studied. This kind of 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 [2]. Engine torque harmonics are transferred to the driveline. The clutch is a key member in driveline which reduces the acyclic behavior of transmitted torque. It has a greater influence in determining torsional vibrations issues such as shuffle, judder and 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. 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 [1]. Rattle has a distinct sound quality that differentiates it from noises produced by other sources in the vehicle [2], while the attenuation of engine noise during the past decades has brought it to the forefront of noise and vibration issues, as a major concern for the automotive industry [3]. Vehicle owners are usually annoyed by this noise and often attribute it to some form of malfunction.

II. RATTLE CRITERIA [3]

Various criteria used for identifying the level of permeable rattle.

Fig. 1: Schematic representation of components in gear rattle problem

Following are some criteria (with reference to fig. 1):

(a) Criterion based on relative displacement between gears ($x_b$):

$$x(t) < x_b : \text{rattle}$$

$$\geq x_b : \text{no rattle.}$$

Where; $x_b = \text{backlash}$

(b) Criterion based on angular acceleration of output gear $\dot{\theta}_i$
Where, \( T_{da} = \) Drag torque at output gear.
(c) Criterion based on angular acceleration of input gear (\( \theta_i \)):
When the gears are in contact, the elastic deformation is very small and therefore it can be approximated as:
\[
\ddot{\theta}_i(t) \approx -(R_i/R_j) \ddot{\theta}_j(t)
\]
(2)
Using equations (1) and (2), one can define the approximate rattle level (\( \beta \)) as,
\[
\beta(t) = (I_iR_i/T_{pa}R_j) \ddot{\theta}_i(t)
\]
(3)
Where, \( R_i \) and \( R_j \) are radius of the input gear and output gear respectively.
Hence, the rattle criterion is approximately given as,
\[
\beta(t) \leq -1 : \text{rattle}
\]
\[
\beta(t) > -1 : \text{no rattle}
\]
For a practical gearbox, \( x_b \) and \( T_{da} \) may not be constants, and the \( \ddot{\theta}_i(t) \) signature may be fairly complicated. Hence, it is more convenient to calculate the root-mean-square (rms) value, \( \beta_{rms} \), which is related to the energy contents of the motion. Assuming a harmonic relationship for \( \ddot{\theta}_i(t) \), the criterion obtained as,
\[
\beta_{rms} \geq 0.707 : \text{rattle}
\]
\[
< 0.707 : \text{no rattle},
\]
where,
\[
\beta_{rms} = \frac{1}{T} \int_{0}^{T} \dot{\theta}_i^2(t) \, dt,
\]
\( T = \) time window
The rattle level \( L_p(t) \) (in dB) and the corresponding rattle criterion are defined as follows:
\[
L_p = 20 \log_{10}(\beta_{rms} / 0.707) \, \text{dB},
\]
\[
L_p \geq 0 \, \text{dB}: \text{rattle}
\]
\[
< 0 \, \text{dB}: \text{no rattle}.
\]

Rattle Index:
A rattle index (RI) based on the value of the unloaded gear acceleration, divided by the acceleration of flywheel;
\[
RI = \frac{\ddot{\theta}_{\text{gear (RMS)}}}{\ddot{\theta}_{\text{Flywheel (RMS)}}}
\]
[9]
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 [9].

III. PROBLEM FORMULATION

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,
\[
T_p(t) = T_m + T_{eb}(t)
\]
The fluctuating torque is vibratory part of the torque that causes rattling phenomenon. \( T_{eb}(t) \) can be written using the Fourier series as follows,
\[
T_{eb}(t) = \sum_{j=1}^{N_e} T_{e_j} \sin(\omega_{e_j} t + \phi_{e_j}),
\]
where, \( \omega_{e_j} = (N_e / 2) \Omega_e \)
\( N_e \) is the number of cylinders (this is for considering the firing order of the engine, as main fluctuations occur due to firing order) and \( \Omega_e \) is engine rotational speed in rad/sec.
This torque is further transmitted to flywheel which has inertia \( I_1 \), then clutch with inertia \( I_2 \) and further to gear box having input and output shaft inertia as \( I_3 \) and \( I_4 \) respectively.

Driveline with all inertias and torque can be represented by fig. 1, by applying Newton’s Second law of motion for flywheel,
\[
I_1 \ddot{\theta}_1 + T_c = T_{e}(t)
\]
(5)
Applying for clutch,
\[
I_2 \ddot{\theta}_2 - k_1(\theta_2 - \theta_1) = 0
\]
(6)
Applying for gearbox,
\[
I_3 \ddot{\theta}_3 - k_3(\theta_3 - \theta_2) + k_4 R_y (x_i - x_b) = T_{da}(t)
\]
(7)
\[
I_4 \ddot{\theta}_4 + k_4 R_y (x_i + x_b) = T_{da}(t)
\]
(8)
Where,
\[
T_{da}(t) = -c_d \dot{\theta}_4(t)
\]
\[
T_{da}(t) = -c_d \dot{\theta}_4(t)
\]
y is the dead space function and \( x_i \) is the relative displacement between gears.

The above equations (5), (6), (7) & (8) are a mathematical representation of gear rattle problem. Equations (5), (6), (7) & (8) are non-linear equations and also contain multi-degree of freedom parameters like torsional system incorporating non-linear machine elements such as a multi-stage clutch and/or backlash in one or more gear meshes.
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.

### IV. 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.

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,

\[
\omega = \text{Forcing Frequency} \quad \omega_n = \text{Natural Frequency} \\
\omega = \frac{2\pi N}{60} \\
N = 800 \text{ rpm (idling rpm)} \\
\therefore \omega = 83.776 \text{ rad/s}^2 \\
\text{at resonance} \\
\omega = \omega_n \\
\therefore \omega = \sqrt{\frac{k_t (I_1 + I_2)}{I_1 I_2}} \\
\therefore (83.776)^2 = k_t (0.0707 + 0.0047) \times \frac{\pi}{(0.0707 \times 0.0047) \times 180} \\
\therefore k_t = 0.54 \text{ Nm}^0
\]

Entering the above value of stiffness $k_t$ in program gives a graph as shown in fig. 2. This shows a resonating nature,

![Fig. 2: Final acceleration of drive line showing resonance with spring stiffness ($k_t$) =0.54 Nm$^0$](image)

Now, just reducing the spring rate by 0.04 Nm$^0$ and again running the program, graph shown in fig. 3 was obtained.

![Fig. 3: Final acceleration of drive line showing beating with spring stiffness ($k_t$) =0.5 Nm$^0$](image)

The above fig. 3 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, $k_t= 100 \text{ Nm}^0$, gives a graph as shown in fig. 4. This graph shows rattling phenomenon as observed in various literature.

![Fig. 4: Final acceleration of drive line showing rattling with spring stiffness ($k_t$) =100Nm$^0$](image)

Above figures 2, 3 and 4 ensures that Simulink model shows correct results in resonating, beating and rattling conditions. Further Simulink model is modified by adding rattling 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 $1 \times 10^{-9}$ and sample time 0.0001 sec. An idle stage angle of 90$^\circ$ 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$^\circ$. 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.

V. 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, $\beta$*rms and $L \beta$ *
):

<table>
<thead>
<tr>
<th>$k_t$ (Nm$^0$)</th>
<th>RI</th>
<th>$\beta$ *rms</th>
<th>$L \beta$ * (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15.44</td>
<td>18.22</td>
<td>2.16</td>
<td>9.724</td>
</tr>
<tr>
<td>7.5</td>
<td>19.43</td>
<td>2.31</td>
<td>10.29</td>
</tr>
<tr>
<td>3</td>
<td>19.8</td>
<td>2.39</td>
<td>10.58</td>
</tr>
<tr>
<td>1.5</td>
<td>19.02</td>
<td>2.31</td>
<td>10.3</td>
</tr>
<tr>
<td>0.75</td>
<td>18.66</td>
<td>2.27</td>
<td>10.15</td>
</tr>
<tr>
<td>0.5</td>
<td>18.62</td>
<td>2.27</td>
<td>10.14</td>
</tr>
<tr>
<td>0.4</td>
<td>3.74</td>
<td>0.40</td>
<td>-4.81</td>
</tr>
<tr>
<td>0.3</td>
<td>3.41</td>
<td>0.36</td>
<td>-5.63</td>
</tr>
<tr>
<td>0.2</td>
<td>3.15</td>
<td>0.34</td>
<td>-6.32</td>
</tr>
<tr>
<td>0.1</td>
<td>3.13</td>
<td>0.33</td>
<td>-6.39</td>
</tr>
</tbody>
</table>

By observing table 1, 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 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 result obtained by running the program with spring stiffness 0.2 Nm$^0$, as shown in fig. 5 and fig. 6.

VI. EXPERIMENTATION:

An experiment was conducted on Tata Ace (light commercial vehicle), by measuring acceleration near flywheel and gearbox input shaft during idling, as shown in fig. 8. The difference between acceleration will give isolation of vibration done by clutch. 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.

<table>
<thead>
<tr>
<th>$d$ (mm)</th>
<th>$D_0$ (mm)</th>
<th>$D$ (mm)</th>
<th>$L_f$ (mm)</th>
<th>$L_s$ (mm)</th>
<th>$L_a$ (mm)</th>
<th>$\Delta_{min}$ (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1</td>
<td>6.4</td>
<td>3.5</td>
<td>12.75</td>
<td>8.3</td>
<td>11.7</td>
<td>1.05</td>
</tr>
<tr>
<td>$C$</td>
<td>$D$ (mm)</td>
<td>$N_t$</td>
<td>$N_a$</td>
<td>$k_t$(Nm$^0$)</td>
<td>$k$ (Nm)</td>
<td>$\Delta_{max}$ (mm)</td>
</tr>
<tr>
<td>4.81</td>
<td>5.3</td>
<td>8</td>
<td>6</td>
<td>0.25</td>
<td>1.8</td>
<td>3.65</td>
</tr>
</tbody>
</table>

Figure 7: Cross-section view of 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 8 shows pictorial representation of the experimental set-up.

Figure 8: Representation of Experimental Setup

VII. RESULT AND DISCUSSION

FFT of the system was obtained by experiment and also by using Simulink model. It was observed that, nature of both FFT are similar, which confirms that earlier estimation done by Simulink model is correct.

The peak of 1st and 2nd harmonic for flywheel and gear box are almost same. This means that, transmission takes place without any isolation.

Fig.11 and 12, represents FFT of prototype, i.e., pre-damper clutch plate. It was observed in both Simulink model as well as experimental result, that, there is reduction of peak for both harmonic. This confirms isolation done by pre-damper spring.

Peak values observed in FFT are as shown in table 2 for pre-damper clutch plate. The isolation obtained is approximately of 60% and accuracy of estimation of Simulink model is 95.53%.

Table 4. FFT Results

<table>
<thead>
<tr>
<th>Simulink FFT Results</th>
<th>Actual FFT Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>PSD(watts)</td>
<td>Accl. (rad/s²)</td>
</tr>
<tr>
<td>Fly-wheel</td>
<td>Gear</td>
</tr>
<tr>
<td>Fly-wheel</td>
<td>Gear</td>
</tr>
<tr>
<td>926</td>
<td>380</td>
</tr>
<tr>
<td>0.1652</td>
<td>0.072</td>
</tr>
</tbody>
</table>

Rattle parameters i.e. $\beta^*$, $L\beta^*$ and RI obtained during simulation are mentioned in Table 3. It can be observed that there is drastic reduction of RI and also other two parameters, hence, showing isolation of vibration and reduction of neutral gear rattle.

Table 5. Comparison of Rattle Parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Conventional</th>
<th>Pre-damper</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\beta^*$</td>
<td>2.166</td>
<td>0.341</td>
</tr>
<tr>
<td>$L\beta^*$</td>
<td>9.724</td>
<td>-6.321</td>
</tr>
<tr>
<td>RI</td>
<td>18.22</td>
<td>3.159</td>
</tr>
</tbody>
</table>

VIII. CONCLUSION

For effective isolation, natural frequency of the system is shifted from 548 rad/s² to 62 rad/s² by changing the torsional stiffness from 15.44 Nm/° to 0.2 Nm/°. Actual measurement on vehicle also shows that for conventional clutch plate, there is neutral rattle, and when it is replaced by pre-damper clutch plate neutral rattle vanishes. Mathematical model constructed in MATLAB-Simulink also helps to visualize and confirm above results even prior to actual measurement or at the design stage itself. It can be concluded that on an average 70% reduction has taken place.

IX. ACKNOWLEDGMENTS

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