Investigation on Automotive Driveline Gear Rattle Phenomenon

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ABSTRACT: The vehicle or drive mechanism is responsible for the transfer of energy from the engine to the wheels. Many of its issues in noise, vibration and harshness (NVH) only arise late in the design chain when all components are checked together in a wide range of circumstances. To alleviate the phenomena, few changes can be made without very costly palliative solutions. The only viable alternative is to configure clutch parameters once this engine and transmission are set in early stages to satisfy power and torque modulation requirements. In technical studies, the effect was shown to be highly influential on dynamic drivetrain behaviour, including laminated steepness and hysteresis. The gear shaft is an effect caused by induced noise of the unloaded gear pairs. In low- or idle-speed diesel engines like buses and trucks in traffic jam and in a more specific manner, it is diagnosed with a higher intensity when a resonance is excited for the entire drive system. The creation of low-rattle vehicles also helps to reduce noise pollution in cities in general. In this study the picture of the clutch found in literature will be evaluated and numerical integration evaluates the response of the torsional model. Results of simulations that change clutch parameters are tackled against changing properties of other system elements such as flywheel inertia, gear reaction and gearbox damping parameters so that it can be verified what plan to determine in a lower gear rattle stage.

KEYWORDS: Automotive Systems, Driveline, Gear Rattle, Powertrain, Noise Analysis.

INTRODUCTION

The gearbox is a gear system that permits the adjustment in the gear ratio between input and output that can be manually or automatically made. This component's main role is to optimize the motor power supply. In addition, a clearance between the teeth must be created in order to make a perfect gear mesh (without friction). Impacts between these teeth cause many types of rattles and clunks. The term gear rattle refers to the sound caused by collisions in the transmission between the unloaded mesh pairs. The engine firing frequency is correlated with manual vehicles in neutral state (idle rattle). These collisions are due to the engine's transmission of torque variations. In a collision the impact force on a driven gear shifts its speed, which induces relative movement between the connecting gears. In the literature Rattle is also defined as a condition in which high vibratory rates are found in the move [1].

The appearance of dead spaces and backlashes is due to the rattle of gears. Furthermore, with relative angular rotation, angular backlashes are typically variable, and the occasional fluctuation is a cause for gear rattling. The word rattle refers to the sound caused by the effects in the transmission between unloaded mesh pairs. The neutral (idle rattle) movement of the engine firing frequency can be observed on manual transmission vehicles. Impacts on diesel vehicles become more severe once torque abnormalities with this fuel become increased. In literature on a condition in which high vibration levels are found in transmission (drive rattle), Rattle has also been described. All manually driven car drives operate at high levels of torsional vibration between 1000 and 2000 rpm in linear characterization, related to natural frequencies ranging between 50 Hz and 70 Hz.

Researcher identified the gear rattle as an airborne sound that occurs when the gearbox's torsional movements are passed over the bearings to the airbox. The parcel is also structurally raised, resulting from the contact between the gearbox mounting system and the vehicle frame. The changeover mechanism will often lead directly to the cavity of the passenger to transmit vibrations. Due to its inertial, rigidity and damping effects, a device known as a dual voltage (DMF) has been used in recent years to mitigate these vibrations. Installation of a DMF would increase the inertia in the transmission input shaft, allowing greater vibration isolation in both idle and drive rattle condition, instead of using a single float wheel inert connected to the crest shaft [3]. Nonlinear models for this phenomenon use a not linear dead zone to reflect the backlash of the gear. The estimated helical pairs of helical gear with time-variable mesh steadiness are generated on the basis of experimental results. For this previous work, geometric parameters like overlap ratio (μs) and transverse base pitch (Xz) were incorporated into this formulation, enabling the total mesh stiffness of the number of teeth in touch to be possible discontinuities. For gear rattles, the modified formula has been added.

The helical equipment pair is included in the model in this work. A systematic approach will be used to evaluate the idle rattle and also to check improvements to other systems components. The linearized system's natural frequencies in idle are measured and compared to the order value of the input torque of the motor [4]. Then, a nicht-linear model with partly linear rigidity and hysteresis which reflect the clutch and time-varying steadfastness of the gear is altered to compare the response of the gear rattle in terms of vibration strength in the clutch, gearing and inertial parameters.

LITERATURE REVIEW

Researcher studied the rattle from an experimental standpoint caused by a multi-harmonic stimulation. Many analysts see a sinusoidal law of speed, but the sum of two harmonic components has been adopted during the analysis as a multiple harmonic excitement. There was an unusual activity in the gears when change was rendered in the second order of the harmonic amplitude of the arousal. In addition to the results of experimental experiments in time and frequency areas in some numerical simulations, the dynamic comfortability was assessed using the test method for unloaded gear pairs. A research on an unloaded gear pair, which experienced a multibaryonic excitement, revealed fascinating aspects of the gear rattle phenomenon, both in time and frequency zones, through observation of the gear's relative angular motion. The rattle frequency is at first equal to the fundamental component of the speed fluctuation when the amplitude of the second harmonic portion of arousal assumes a value equivalent to approximately 70%-70% of the first variable.

The phenomenon of the gear rattle and various techniques for raising gear rattle are understandable from the above literature review carried out. In addition, various methods are studied for modelling the issue of gear rattles. This means that a neutral rattle problem model will be created. The major task so far has been to modify the geometry of the gear or the flywheel by using a dual flywheel. Different clutch parameters have also been found to affect scores. The research is carried out of induced drive fluctuations which are helpful to model the real-world problem. The analysis is carried out. Noise made from diesel components impact the efficiency of engine noise. Getting rattle is one source of consumer complaints. Gear rattles are caused by the effect of gears as a result of insensitive torsional acceleration variations in the drive gears. In previous research in this area, the overall sound quality of diesel engines has been measured without relying explicitly on models for prediction of gear-rack perception. Here they define a method for generating sounds with different rattle rates. The first research was carried out on diesel engine noise records to create the motor speed history and then used to direct gear impact time and produce noise components. The first analysis was completed [2].

The transmission paths were then modified to improve the accuracy of the predictions of gear noise. Currently, the noise simulation tool is used to produce sounds for subjective experiments aimed at quantifying detestability, growth perception and rattle annoyance. The prediction of noise combined with the model for sound quality based on the study of subjective data would provide a forecast of how people interpret gear rattle in order to set component noise goals directly linked to human perception [3]. This form of simulation proved effective in generating realistic sonic times with different equipment rattles. The independent monitoring of the rattles of the gear rattles case would be a useful tool to recognize the thresholds at which the equipment rattles are detected. In the subjective test designed to quantify the detectable levels of rattle gear, the simulations created as described here have been used; the results of such a test shall be reported later. Decisions taken during the development of a method that improved sound from a listening perspective but degraded the measurements of sound quality between the simulated and the actual signals stress the importance of listening to sounds and not relying solely on sound quality metrics during the process. The understanding gained from this simulation phase will contribute to the creation, by noise measurements, of a gear razzle metric. Recent research has shown that gear racket could influence the engine 's operation. The presence of the rattle appears to intensify the sound of the "background" (engine noise not related to rattle impact events). This is also stated earlier. An improved simulation could provide feedback that more precisely simulates how gear rattles impact engine operation.

MODAL ANALYSIS AND DRIVELINE MODEL

Figure 1 shows the typical front wheel drive. Modal analysis can be based on a mass moment of inertia and torsional rigidity by several degrees of freedom. An analysis of the vehicle's driveline torsional vibration using vibration theory. By approximating it to a four-inertia method, the system's natural frequencies were determined. The program for the development of the modal analytical 1D model, shown in Figure 2, is used in this research LMS AMESimv1310. The inertia of the engine crankshaft ensures that the attachments are attached to the single mass flywheel.

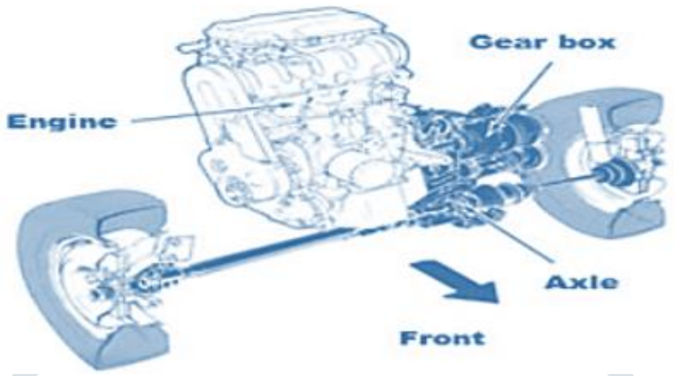

Figure 1: Front Wheel Drive

There is therefore a single mass with all the inertia related to the flywheel. Gearbox is treated as a single mass because the output shaft and its related rotating parts are balanced by the correct gear ratio in the input shaft. The front wheel drive is based on 6 mass systems for the passenger car. Five natural frequencies were obtained with a clutch stiffness of 16.5 Nm/ °. Such five frequencies are suited to each vehicle's phenomenon. The first frequency 7 Hz that shuffles the engine [4]. A reverse torque occurs on the train of a vehicle when this low frequency can be excited. It normally happens during a grip tip-in or tip-out or a static move. Second and third mode leads to the resonance of left and right wheels, resulting in Booming. In modal analysis, the fourth mode is similar to a gearbox, where the gearbox and inertia of the clutch stands opposite to a flywheel. This gearbox resonance mode is the subject of current research.

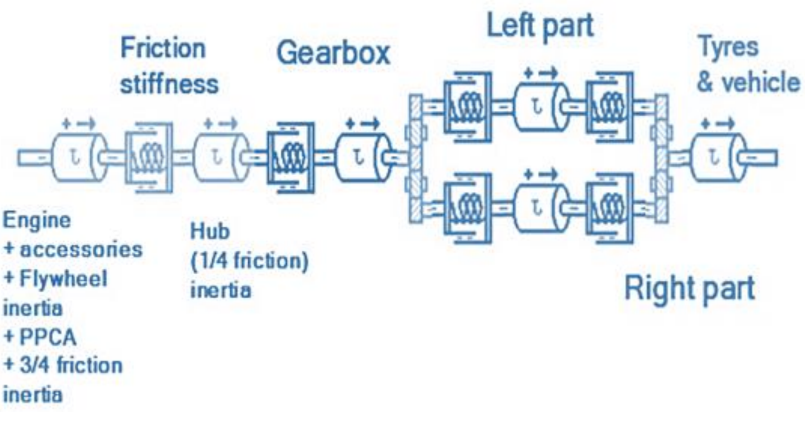

Figure 2: FWD Model Design for the Front Wheel Drive Clutch

METHODOLOGY

The purpose of the work is the temporary FWD drivetrain oscillation. The second consists usually of a manual diaphragm jumper, 6-speed manual gearbox, differential and push-up shaft. Figure 3 displays a schematic diagram of this set up. The sudden involvement of clutches in various driving conditions leads to cases of driving abuse which lead to severely torque fluctuation into the collaborative component. These cases of driving-influences during start-up movement of the vehicle is investigated by application. In model simulations, the following operating conditions:

- The lowest gear selected vehicle at rest;
- Engines acceleration & abrupt embroidery

Throttle is in constant positions (therefore, the clamping force of the clutch plates is considered to rise by the zero towards its max. values within 0.1 sec). These results of model shall be contrasted with the experimental tests carried out on vehicles fitted alongwith similar FWD drive in manoeuvres corresponding to first gear of the vehicle's impact.

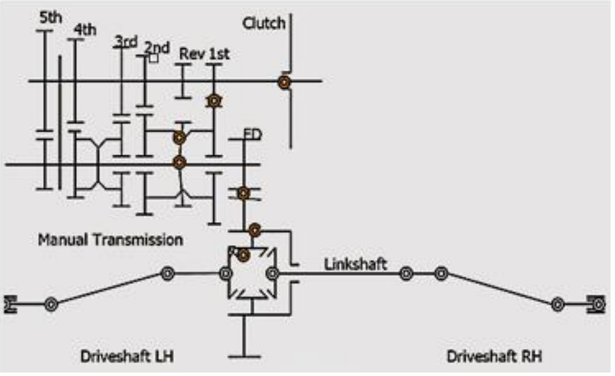

Figure 3: The Configuration of the Control Train

1. *Experimental protocol adopted by industry:*

- The automobile will rest first; the brakes will be released; the first gear will be picked up and the clutch removed.
- The engine is speeded up to 2400 rpm.
- From the clutch pedal the driver foot slides sideway.

During above case, the torque's data are registered with Kistler Roadyn wheel force transducers and related directly data acquisitions soft wares for after-processing purpose (mounted on axle, as indicated in Figure 4. The forces used are therefore determined on four charging cells (and onto the vehicle's coordinate) & converted in torque for axles [4].

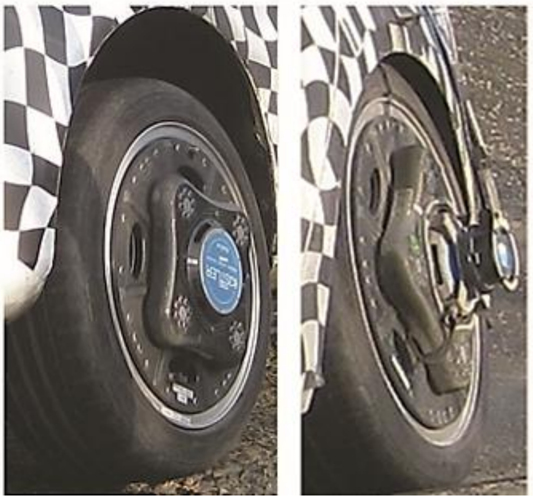

Figure 4: Typical Experimental Design for Measuring Wheel Torque

2. *Modelling system's dynamic :*

As the emphasis is onto transient's torsional oscillation of trains, physics of problem is only defined in terms of the rotary degree of the freedom for interacting component. The device is therefore divided basically in number for key Discrete's parameters (lump parameters models), along with rigidity of material and the hysteretic behaviour. Matrix-form movement equations are expressed:

$$[J]\{\ddot{\phi}\} + [C]\{\dot{\phi}\} + \left\{ \begin{array}{ll} K_n(\phi_i - \phi_j - b_{i,j}), & \text{if } \phi_i - \phi_j > b_{i,j} \\ 0, & \text{if} |\phi_i - \phi_j| < b_{i,j} \\ K_n(\phi_i - \phi_j + b_{i,j}), & \text{if } \phi_i - \phi_j < -b_{i,j} \end{array} \right\} = [\vec{T}]$$

Where [J] are masses matrix, [C] a damping matrices, K_n, nth spring element's rigidity coefficient (available in the rigidity pattern [K]), [T], externally arouse vectors {f} are system independent liberty-grade vector & bi, j half backlashes among I & j matrix. These retaliation areas introducing local nonlinearities into system & modelled using linearly function into part. The gear tooth operation was modelled by taking into account the constant properties of teeth meshing rigidity; due to the duration of intermittent events of the drive train, this simplistic approach was adopted.

Two potential situations occur during clutch engagement. The first scenario is that the clutch and the flywheel have relative speed, which leads to the use of the film coefficient. The second scenario involves sticking (i.e. when the friction mechanism is not fairly fast, suggesting that the coefficient of static friction may be used). Only as much torque can be transferred as necessary in this latter way for keeping two of the component moving at same velocities; if much of the torques are applied, surfaces can simplified slide back. This difficulty of model above-mention slips behaviours (immediate exchange among the two of the state) increases.

RESULTS AND DISCUSSION

The linear set of motion equations is initially used to determine the natural frequencies and mode forms by solving the problem identified by equation. Throughout this study the clutched frictions interfaces isn't considered; therefore, the matrix forms of equation for the motions inertia of the clutches frictions plates J2 treated an unmodified device. Table 1 displays the normal frequencies.

A free body (unconstrained system) is indicated by the null natural frequency. No natural frequencies are below the motor frequencies of the 40 Hz's (there should be no resonance effect).

Table 1: Naturally Frequency of Drive lines

Natural frequency (Hz)
10,656
9394
2592
1523.7
1068
559.4
212.7
64.95
5.24
0.97
0.00

The non-linearly models simulated after the introduction of the damping matrix. In order for solving the non-linear secondary motion different equation, numerical integrations schemes Runge-Kutta have used to MATLAB. A time stage of 10ms shall be used for the simulation, which has a total length of 5s. The speed of the vehicle is increased fast for approximately 17 rad/sec and matches motor speed of the 2400 rpm. This time step is an important factor in order to accurately reflect transient happenings. The speed then slightly oscillates around this value [5].

A comparative displacement of the two adjacent inertia can provide useful information on the events in the correct lashing field. The latter is compared alongwith the experiment information obtained with wheel forces transducer, which are attached to the two front wheels in car equipped alongwith similar drive trains, to validate methodology and simulation results. Studied 'fast start' clutch misuse case was carried out on the floor of the car. The clutch was disengaged and the first equipment was selected. The engine is held over 2400 rpm & driver sides tacks seizure so that the entire vehicle licks. The history of load time is replicated multiple times in the case, so that the maximum minimum and average peak charges are correctly reflected.

The experimental data are 9 Hz, while the simulations predict a 6 Hz frequency that is relatively similar to the system's third (524 Hz) natural frequency. The variations between observed and expected frequencies are possibly the result of a failure to fit the real values for assuming steepness, damping and inertia. The result may also be caused by the slip of the contact patch that should much pronounced in LH (that is stiff as compared RH drive shaft) for the snapping-start trials for type [12]. These differences also explain how the peak torques is different. Although the numerical results are assumed to be not wholly compatible alongwith experiments evidence based on assumption used & on predicted device variable by quantitatively perspective, simulations predict result which are relatively qualitatively appropriate. Moreover, the general form for predict times history well into line alongwith experimental observations, that facilitate further analysis of the essential driving parameters in transient behaviour.

CONCLUSION

Hence, this may seemed which extend clutched clamps loads engagement timings are beneficially for vehicles into the term for protecting drive lines component by excessively loading while abusively driving scenario, it may detriment affected for clutched discs life due to the increasing into engagement slips time & thermally building-up. Comparing the number result alongwith experiment information by similarly FWD system confirms validity for proposed approaches, but for the reasons already mentioned there appear some quantitative discrepancies. Nevertheless, between the model and the tests, there is very strong qualitative agreement. In order for determining the effect onto the peaks torque value, a preliminary analysis with main clutch parameters was performed. In general, decreasing into engagement torques during first stage of manoeuvre induces decrease into drive shafts' maximum torques, increasing time for taking to achieve the peak values & vice versa. Increased clamp’s time dropped almost 17 percent of the maximum torque. The parameter appearing for having the greatest effects onto system is clamping loading profiles. Although it might appear that extending the cluster loading time to protect the driving components from unnecessary load during aggressive driving situations is advantageous to a car, this may have adverse effects on the cluster disk life due to can clutch slip times and heat build-up.

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