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Sensitivity analyses of gear rattle noise using multi body simulation with advanced gear contact model

Saravanan K Bhagate Rajkumar   G Mathan

Abstract

Rattle noise from manual transaxle is always a concern which affects comfort and ride quality. The rattle is influenced by the nonlinear dynamic deformation of multiple pairs of idler gears in a multistage transaxle system. Prediction of overall contribution of gear rattle noise in multistage gear box is very difficult. The drag torque on driven gear, gear backlash, gear inertia, oil level viscosity and the level of torque fluctuation through gear system are influencing on the level of gear rattle. In this work, six speed manual transaxle used to examine for 5^{th} and 6^{th} gear rattle noise. To build mathematical simulation model based on dynamic system combined with AVL Excite power unit® software is used. The vibro – impact by backlash between the meshing gears which leads excessive vibration and noise in many geared rotating systems which is evaluated by using data recovery method to predict overall housing structural NVH which is further used to analyze gear rattle. In this paper gear rattle is analyzed for different backlash such as maximum, nominal, and minimum value used with different second order excitation in gear input shaft. Also, the structural velocity and impulse force from simulation is used to investigate unloaded gear rattle in manual transaxle. Based on the sensitivity analysis, design guidelines are proposed to optimize the gear rattle in the transaxle.

Keywords: *Gear rattle, Finite element model, Excite Power Unit, Surface velocity and Gear housing*

1.Introduction

Gear rattle noise is a common problem in automobile with manual transaxle. At present certain efforts have required to improve sound quality of inner cabin noise especially in the passenger vehicles. The low-level noise inside car gives a perception of good quality of the product as well as human comfort, and this may cause the impact in automobile market. The gear rattle phenomenon has played a significant role for creating undesirable sound in a passenger vehicle and relates to a noise mainly caused by vibration, especially the manual speed gear, by the repeated impact between the teeth of unladen gear pair or unsynchronized gear pair and it is audible at a low frequency. Rattle noise is generated by nonlinear dynamic deformation of multiple gear pairs of unladen gears in a manual transmission. Due to engine torque fluctuation, a series of vibro – impacts induced between the meshing gears which leads excessive vibration and noise in rotating system. For this reason, all automobile manufacturers are concerned about NVH (noise, vibration, and harshness) with the primary aim of analyzing all the source of undesirable excitation and minimizing the acoustics effects which is transmitted inside cars.

Markus Bodden et al.,[1] have elaborated spectral cues to predict gear rattle from an artificial head recording interior vehicle noise with specified driving condition. Its described structure of the sound carries perceptual relevant information. Also, they have elaborated modulation analysis method to investigate time structure of the gear rattle in vehicles. In passenger car, gear rattle occurs in some certain driving condition which is analyzed theoretically as well as detailed studies have carried out to reduce gear rattle by introducing multi-stage torsional vibration damper. It has good control of fluctuation of angular acceleration at input shaft. Also, the gear rattle examined in manual transmission by implementing single and multistage torsional vibration damper [2].

Clutch stiffness has significant role to create ideal gear rattle in manual transmission. But Engine in idle condition driveline vibration mode and natural frequencies of the model has not influence the ideal gear rattle. Also, they have reduced the gear rattle by modifying the clutch stiffness parameter [3]. Xiaona He et al, Explained gear rattle mechanism through simulation model of single stage gear transmission system. And the detailed study of dynamic response of single gear system by using Runge-Kutta algorithm and summarized the way of reducing rattle noise in automotive transmission [4]. The dynamic response of loose gear mainly depends on the design parameters, the engine operating conditions, unknown parameters such as drag torque and the coefficient of restitution [5].

Although, the impacts between loose gears do not change the dynamic behavior of the driveline, it is widely believed that the rattle noise is particularly noticeable in the automotive multistage gear transmission system because of the influence of more nonlinear parameters. For one pair of the idler gear, the key factors affecting rattle noise are mainly gear backlash, moment of inertia of the loose gear, rotation speed of the active gear, and damping coefficient [6, 7]. Although the effort of hydrodynamic lubricant in manual gear transmissions working under low torque level has analyzed with several mathematical formulation. And the effect of lubricant entrance between gear mesh contact area plays a decisive role in the dynamic behavior. As a result, the dynamic forces obtained by mathematical formulation which is used to evaluate the sound level of inside manual transmission.

The creating mechanism of transmission rattle noise and corresponding suppression methods are still not clear so for multistage gear box system. To study dynamic behavior between unloaded gears at the same time to reduce the sound pressure, on one hand, many efforts have focused on accurate description of rotation speed of active gears [8–10]. The input speed fluctuation is normally affected by change of engine power. The research studies are mainly to optimize the transfer function of the torsional excitation between the engine and the gear input shaft of manual transmission. It has more stable to reduce the vibration in manual transmission through the driveline [11, 12]. As of now torsional fluctuation or excitation is often simplified as the second order harmonics excitation in the most studies [13]. On the other hand, some of the research investigation has started from the equivalent dynamic model of impact effects between unloaded gears [14-16].

Rocca and Russo [14] elaborated the influence of gear backlash with periodic fluctuation on rattle noise based on a linear equivalent dynamic model. Fietkau and Bertsche [15] established a simplified model with Kelvin–Voigt method, and the influences of gear backlash and moment of inertia of idler gears on rattle noise were studied. The calculation functions of the resistance moment of lubricating oil on idler gears had been obtained theoretically and experimentally, and the relationship between resistance moment and rattle noise was analyzed in Brancati's [16] research work. Kadmiri et al. [17] and Shangguan et al. [18, 21] studied the mechanism of rattle noise through a gear rattle test rig, and the effects of lubricating oil damping on the gear impact were discussed deeply. Although these studies considered many other factors besides velocity fluctuation of driving gear, they mainly concerned the linear dynamics characteristics of related components. The influences of nonlinear factors between each parameter on rattle dynamic need to be studied in depth. Otherwise, some studies begun to pay attention to the nonlinear factors, but they were mainly based on the nonlinear dynamic model of single-pair gears by the numerical simulation method.

From the past studies, it can be found that sound pressure level of rattle noise is not high compared to other meshing noise in a car, it is more likely to influence the NVH performance. Some nonlinear rattle dynamics can be explained based on linear dynamic of a single pair of gears by the simplified dynamic model. However, the total rattle noise of multistage gears is affected by all other pairs of unloaded gears. In order to evaluate and optimize multiple nonlinear coupling parameters to reduce the total rattle noise for the whole transmission system, an equivalent rattle dynamic model of a single pair of gears is modeled with a help of AVL Excite power unit® regarding to observe the rattle noise in a multistage transmission system in this paper

We intend to identify the nonlinear impact dynamics of multistage gear system and at the same time to evaluate unloaded gear rattle noise in manual gearbox particularly 5[th] and 6[th] gear engaged drive condition. The repeated impact is observed with different second order excitation. The total rattle noise and it is calculated with a help of surface normal velocity. Finally, we summarize the structural velocity and repeated impulse of dominating gear pairs in manual transmission under two different second order excitation as well as changing gear shaft inertia and stiffness.

2.Dynamic modeling of gear pair in Excite power unit

Based on the equivalent principle of inertia [5], rattle behavior of the gear system can be equivalent to the superposition of single gear pair vibration effect. For convenience, a single straight gear pair is explained for the dynamic modeling, the same principle is used to build multistage gear box in multibody dynamic simulation tool called excite power unit [6]. Dynamic model of single gear pair is as shown in figure1.The corresponding dynamic mathematical model is built under the energy conservation law. Considering the meshing error, the force analysis and driven gear is carried out. The equations are as following.

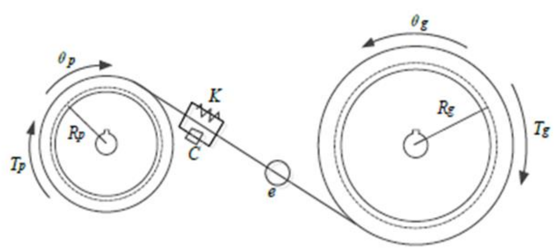


Figure 1. Dynamic model of single gear pair

$$I_p\ddot{\theta}_p + R_pC(R_p\dot{\theta}_p - R_g\dot{\theta}_g - \dot{e}(t) + R_pK(t)(R_p\theta_p - R_g\theta_g - e(t) = T_S + T_v(t) \qquad (1)$$

$$I_g\ddot{\theta}_g - R_gC(R_p\dot{\theta}_p - R_g\dot{\theta}_g - \dot{e}(t) - R_gK(t)(R_p\theta_p - R_g\theta_g - e(t) = T_g(t) \qquad (2)$$

Where *p* and *g* are the subscripts which demonstrating driving and driven gears. I_p, I_g are the rotational inertia; $\dot{\theta}_p$ and $\dot{\theta}_g$ are the torsional vibration angular displacement; R_p, and R_g is the base radius; T_p, and T_g is the torque; K, C is the torsional stiffness and damping coefficient respectively. e(t) is the gear dynamic meshing error. T_s, and T_v (t) is the stable component and variation component driving torque. The convenience, the relative vibration displacement between the two meshing teeth is

$$x = R_p\theta_p - R_g\theta_g \qquad (3)$$

Considering the clearance, actual deformation and tooth surface vibration displacement are no longer equal [7-8]. Deformation happens only when the relative displacement is greater than the clearance. Set *b* as teeth clearance on both sides, the actual deformation can be expressed by function $f_r(t)$:

$$f_r(t) = \begin{cases} x - b, & x > b \\ 0\ , & -b \le x \le b \\ x + b, & x < b \end{cases} \tag{4}$$

Considering the equation *(1 to 3)* and dynamic equation is as equation *(5)*

$$m\ddot{x}(t) + c[\dot{x} - \dot{e}(t)] + K(t) f_r\big(x - e(t)\big) = F_t - m\ddot{e}(t) \quad (5)$$

Where m_e is the equivalent mass of gear pair, F_t is the total load of transferring between teeth. The above theory has been used to build complex gear box in excite software.

2.1 Advanced Gear Contact Model

In excite power unit the pinion and gear connection are made by using advance gear joint which is shown in figure2. The resolution of the multi-flank pair contact is the main aim of the gear mesh model presented in this paper. It can be characterized as an advanced force law which gives a relationship between the dynamic motion of the connected gear wheels and the constraining forces/moments acting on the gears. To meet the requirements regarding computational efficiency of the models applies reasonable simplifications wherever this makes sense. More specifically the gear contact is resolved in the following steps:

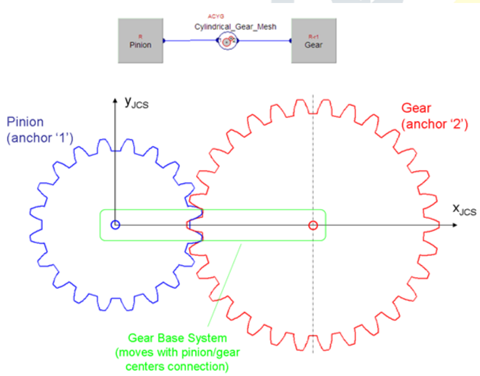


Figure 2: Advanced gear contact model in Excite power unit

Discretization into a series of slices along the gear's face width and detection of contact by intersecting candidate flanks with plane of action which is explained clearly in figure2. Similarly, to create the retainer node for all drive and driven gear which is used to connect the input and output shaft with the help of gear joints. This advanced gear joint contains gear details such as no of teeth, module, helix angle, gear macro details and gear properties.

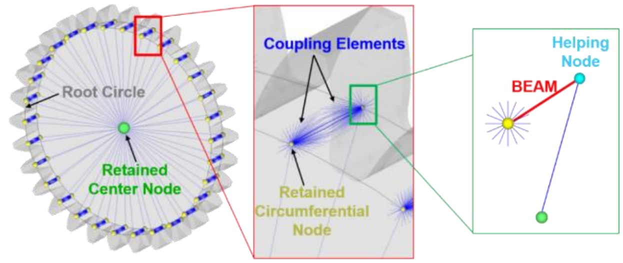


Figure 3: Representation of retained node and coupling elements

Using this joint we built the multistage transaxle in excite power unit which is shown in figure 3. The FEM model is condensed by Nastran solver and it is represented by Excite body. The following gear material properties and design parameter are used for simulation as shown in table1&2

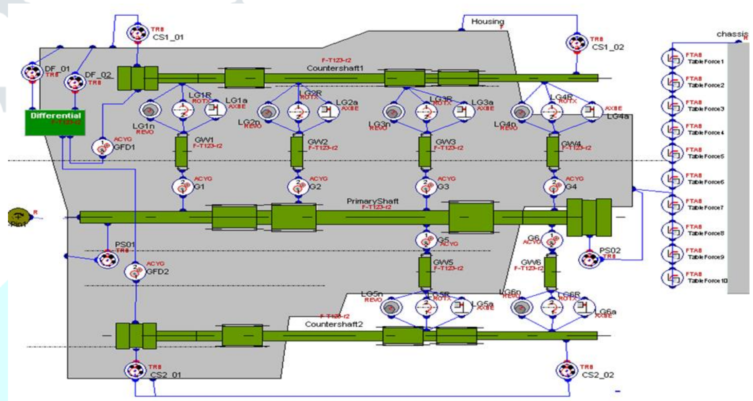


Figure 4: The 2D view of the AVL Excite power unit where y is the rotation axis and z are upward

To determine the displacement field by considering local connection node position and deviation from the ideal involute (shape modification / correction). Computation of total normal contact force (figure 2) considering the deflection induced by flank contact, tooth bending and tooth tilting in wheel body. To evaluate damping force (contact and backlash) and friction force between the drive & driven gear by using normal surface velocity method

2.2 Simulation methodology

The simulation method involves two steps, in the first step the modal vector is calculated block. In the second step, the condensed mode shapes are used in the MBD model. In simulation, the following approaches are used to build the dynamic simulation model toto determine the structural vibration of gear housing.

Table1: Material parameters of all gears

S. No	Gear	Youngs module, N/mm2 (mm)	Poisson ratio	Damping, Ns/m	Drag factor
1	Gear 1	210000	0.3	5240.00	0.0354
2	Gear 2	210000	0.3	589.00	0.0519
3	Gear 3	210000	0.3	1551.00	0.0743
4	Gear 4	210000	0.3	913.00	0.1014
5	Gear 5	210000	0.3	486.50	0.1070
6	Gear 6	210000	0.3	913.00	0.1230
7	GFD1	210000	0.3	1350.00	0.1016
8	GFD2	210000	0.3	1661.00	0.1010

Table2: Design parameter of all gears

S. No	Gear	Modulus, (mm)	Individual gear ratio	Total ratio	Pressure angle,degree	Backlash, mm		
						Maximum	Nominal	Minimum
1	Gear 1	2.037	3.93	17.08	21	0.180	0.106	0.089
2	Gear 2	2.788	2.16	9.39	23	0.161	0.097	0.078
3	Gear 3	2.330	1.34	5.83	21	0.146	0.088	0.070
4	Gear 4	2.150	0.92	4.00	21	0.149	0.089	0.072
5	Gear 5	2.150	0.86	3.08	22	0.138	0.081	0.064
6	Gear 6	2.150	0736	2.63	21	0.145	0.085	0.068
7	GFD1	3.325	4.35	-	21	0.200	0.132	0.114
8	GFD2	3.325	3.58	-	21	0.209	0.137	0.119

3. Simulation methodology

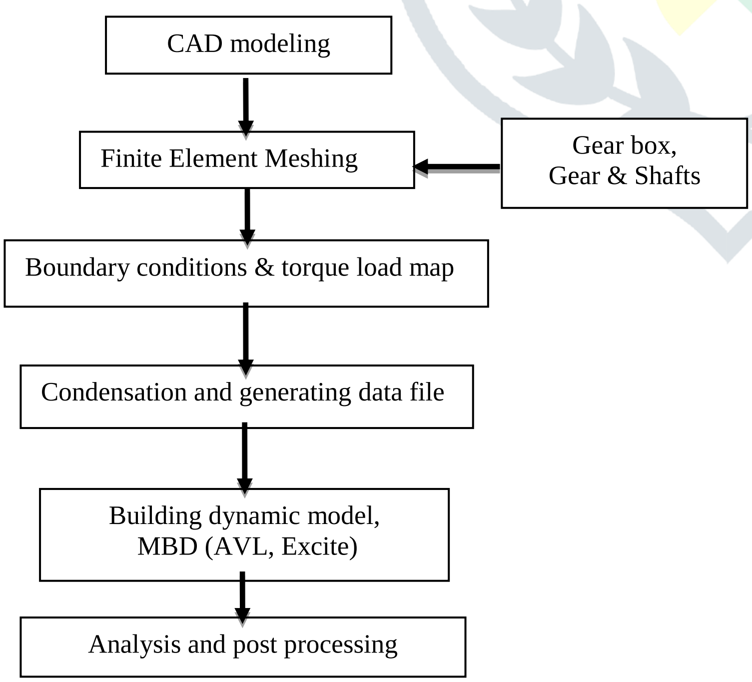


Figure 4. Simulation methodology

The simulation method involves two steps, in the first step the modal vector is calculated block. In the second step, the condensed mode shapes are used in the MBD model. In simulation, the following approaches are used to build the dynamic simulation model (figure 4) to determine gear rattle noise with help of surface normal velocity

4. Modeling approach

The analyzed problem concerns a numerical study of gear rattle noise and structural vibration of gear housing at 1st and 2nd gear condition, for speed range from 1500rpm to 4000rpm with constant output torque 150Nm. Instead of a complex numerical analysis of the system through a direct FEM, which could involve prohibitive number of degrees of freedom, the authors present a modeling approach exhibiting a gradually increasing complexity. Starting from rigid body analysis and introducing progressively the elastic behavior of the various subsystems, using then a modal FEM analysis, the impact on accuracy of such modeling refinements comes out. The originality of the overall approach is related to the combination of various numerical approaches. The analyses are focused gear rattle impact on 1st and 2nd gear engaged condition with constant output torque by varying gear shaft diameter and length. The time dependent simulation results at the unloaded gear, gear housing is evaluated by using second order excitation at gear input shaft A Multi body dynamic simulation (MBDS) of manual transaxle is used to characterize its dynamic behavior from input shaft to output shaft with two different second order acceleration giving at input shaft. The modeled components were: Gear input shaft, countershaft, gear joints bearings, transaxle mounts ("also supported brackets") between the chassis and gear box as shown in figure4

5. Results and discussion:

The following simulation cases (Table3) have been performed and third octave normal surface velocity results as well as impulsive forces are reported in the present paper.

Table3: Simulation of sensitivity analysis

S. No	Case	Sensitivity analysis of Gear 1st and gear 2nd
1	Case1	Structural velocity of base model at 150Nm with different acceleration
2	Case 2	Impulse force of unloaded gear with variation of backlash values

Case1: Structural velocity of base model at 150Nm

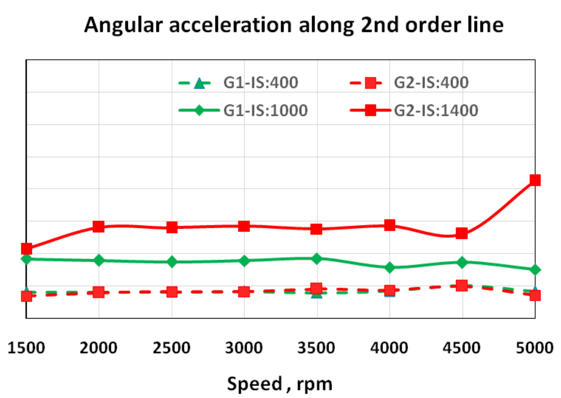


Figure5: Second order inputshaft acceleration of 1st and 2nd gear .

The second order gear input shaft acceleration of 1st and 2nd gear engaged condition as shown in figure 5. First and second gear sum level surface normal velocity of base model with different second order angular excitation such as 400, 1000 and 1400 rad/s^2 is shown in figure 6. Figure 6 clearly indicates the structure borne noise is increased with increasing fluctuation. The surface normal velocity is in- directly proportional to speed. If the speed increases the surface normal velocity decreased. Because the gear rattle always played at low frequency range. Here, the overall structural vibration for common mesh gear pair order, main order and overall level is higher when increasing input shaft excitation which is clearly shown in figure 6

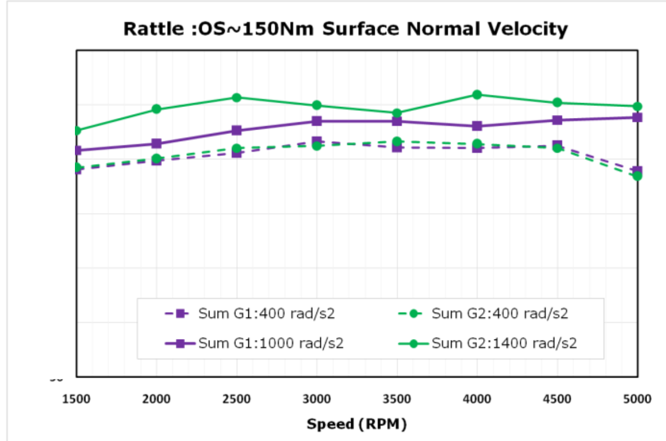
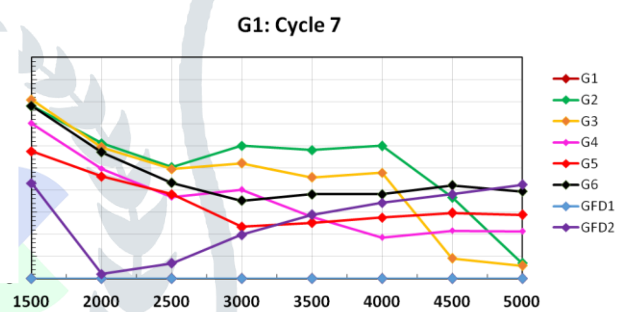
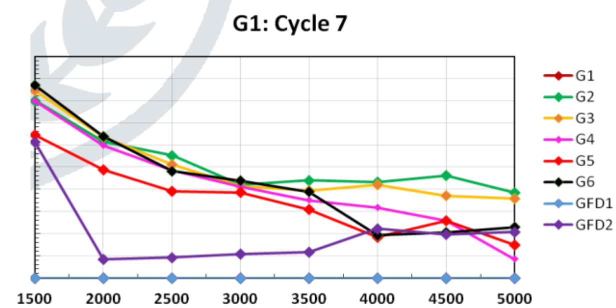


Figure 6: Average surface normal velocity of 1st and 2nd gear sum level

The repeated impulse force of unloaded gear with nominal condition is shown in figure 7. Normally, the repeated impact is happed unloaded gear. In figure 7 clearly indicated impulsive force results on 7th revolution gear with 400 rad/s2 and 1000 rad/s2 inputs haft acceleration.



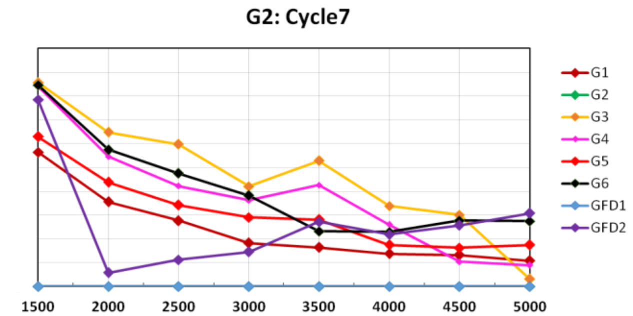
a) Impulse force at 400 rad/s^2



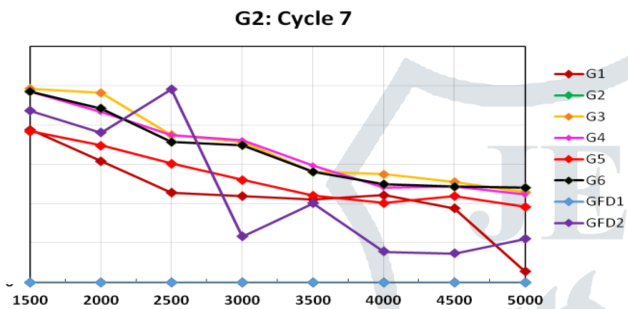
a) Impulse force at 1000 rad/s^2

Figure 7: Imupulse force of unloaded gear at first gear engaged condtion. a) second order excitation 400rad/s2 b) second order excitation 1400 rad/s2

At first gear engaged condition, the impulsive force of third, fifth and 6th gear having more repetead hitting force at lower speed which is clearly observed in figure 7



a) Impulse force at 400 rad/s^2

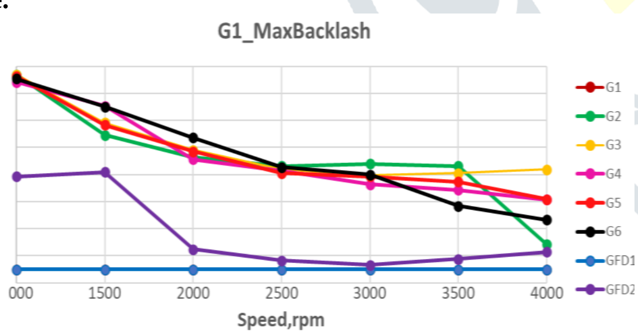


b) Impulse force at 1400 rad/s^2

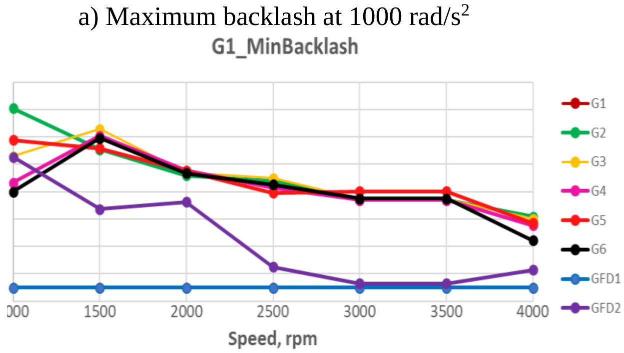
Figure 8: Imupulse force of unloaded gear at 2nd gear engaged condtion. a) second order excitation 400rad/s2 b) second order excitation 1400 rad/s2

At second gear engaged condition, the impulsive force of third, fourth and 6th gear having more repetead hitting force at lower speed which is shown in figure 8.

Case2: Impulse force of unloaded gear with maximum and minimum backlash value.



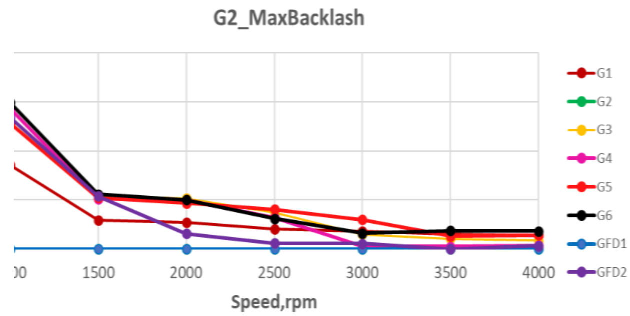
a) Maximum backlash at 1000 rad/s^2



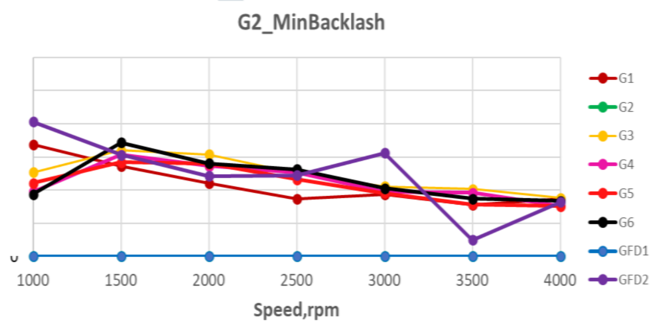
a) Minimum backlash at 1000 rad/s^2

Figure 9: Total impulsive force of first gear engaged condition a) Maximum backlash b) Minimum backlash

Impulsive force of 6th, 5th and 2nd gears are more dominating at lower speed for both maximum and minimum backlash which is clearly observed in figure 9.



a) Impulse force at 1400 rad/s^2



b) Impulse force at 1400 rad/s^2

Figure 10: Total impulsive force of second gear engaged condition a) Maximum backlash b) Minimum backlash

Although , if the second gear engaged condition the repeteated impact on the loose gear forces are shown in figure 10. Here, clearly noticed the 5th and 6th gear impulsive force are higher. Simillarly ,if the shaft angular acceleration increased the rpeted impulsive force on 6th and 5th gears also increased.

6. Summary/Conclusions

A Multibody simulation has been performed in manual transaxle at 1st and 2nd gear engaged condition with different amplitude of second order excitation and variation of backlash values are used in AVL Excite power unit® and following pattern is observed.

- Gear rattle noise is directly proportional to second order gear input shaft fluctuation, and it is predicting higher rattling noise at 1400 rad/s^2 of input shaft angular acceleration.

- Repeated impact force on 5th and 6th gears are observed high compared to other loose gear for both first and second gear engaged condition. From the analysis the rattle force over drive gear comparatively higher for all backlash such as maximum, minimum, and nominal.

Finally, in simulation fifth and sixth gears are creating tattle noise at lower speed. After, modifying backlash value slightly reducing repeated impact in the loose gear. But we recommended to modify the backlash value the input and countershaft bearing stiffness to reduce the rattling effect at lower speed

7. Reference

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