

Experimental Study of the Factors Affecting Transfer Case NVH Performance in 4-Lo Operation Mode

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ABSTRACT

In this paper, the results of the experimental study are presented to describe the impact of several gear design and manufacturing related factors on NVH performance of a transfer case operating in 4-Lo mode. The investigated gear design factors include lead crowning and profile crowning of the planet gears. The influence of manufacturing and assembly is investigated by varying carrier pinhole tangential position error and carrier pinhole tangential tilt error. The experimental DOE study is performed on chassis dynamometer by using actual vehicle. The strategically placed accelerometers and microphones are used for data acquisition. The results show that, among the gear design related factors, lead modification has larger influence on the NVH performance than profile modification. The study also shows how manufacturing errors influence NVH performance of the transfer case by causing lead misalignment of gears and unequal planet load sharing.

INTRODUCTION

Planetary gears are extensively used in aerospace and automotive applications. Their major advantages over their parallel axis counterparts are compact size (power density), high efficiency, larger speed reduction and possibility to achieve several speed ratios by using relatively simple controls. They find one of their major applications in Transfer Cases, used in 4×4 and AWD vehicles to divide power between front and rear axles.

Dynamic gear behavior was extensively studied in the past [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20]. Loss of conjugate contact between mating gear teeth and variation in the gear-pair tooth meshing stiffness lead to large

transmission error and dynamic forces, respectively, which are the main reasons for the problems with gear NVH performance [1, 2, 3, 4]. Due to their simple geometry, spur gears are the most often studied kind of gears. Lin et al [5] in their analytical work on spur gear-pairs with linear tip modification and concluded that the profile modification has significant influence on the dynamic and vibration response. They found out that the necessary amount and shape (length) of the profile modification is influenced by the load - the higher the load, the larger the required modification length. Similar results were reached by Kahraman et al [6] by performing experimental work on spur gear pairs with various linear tip modifications. They concluded that a tip modification is not capable of yielding favorable results for a wide range of loads, and proposed using alternate profile modifications in such cases. Houser et al [7] studied large number of spur gear-pairs with different linear tip and root profile modifications and different level of accuracy, and concluded that, when reliable data is not available, it is better to provide more than less modification. On the other hand, in their analytical work on high-contact gears Lee et al [8] found out that providing more than necessary profile modification is more detrimental for dynamic response of a spur gear pair.

Helical gears, having larger contact ratios experience a smaller deviation in the mesh stiffness and for that reason they are generally quieter [3]. Houser et al [9] experimentally investigated the influence of the common errors (pitch error and face width) on dynamic behavior of spur and helical gears. Park et al [10, 11] experimentally confirmed that the NVH performance of helical gears is significantly influenced not only by lead error, but also by its direction.

The dynamics of planetary gears sets is far more complicated than the dynamics of spur and helical gears. It is influenced

not only by the gear modifications and manufacturing and assembly errors, but also by the position and phasing of the planets [12]. A number of experimental studies was performed [13, 14, 15, 16] on dynamic behavior of planetary gear sets. Detailed modeling of the behavior requires a large number of the degrees of freedom. Recently, with the development of the FEA capable of analyzing non-linear deformable body models of the complete planetary assemblies [17, 18, 19] it became possible to predict the NVH performance of a planetary gear set. Unfortunately, most of the models are still limited to 2D cases (spur gear), and it is not possible to study the influence of the lead, or pinion pin slope error.

Narayanaswamy et al [20] investigated the means of predicting vibro-acoustic behavior of a Transfer Case. They used 2D non-linear deformable body model of a planetary gear set to obtain dynamic response of a planetary gear set. The results were then used in commercial FEA software to calculate surface velocities of the Transfer case housing. Finally, they employed Boundary Element Analysis to predict generated noise through sound pressure and sound power levels. While the agreement with the test results was generally good, some discrepancies due to modeling idealizations, boundary condition assumptions and manufacturing and assembly errors were noted.

The major objective of this study is the experimental investigation of the influence of the main tooth modifications

and manufacturing errors on the NVH performance of a Transfer Case operating in 4 -Lo mode of the operation.

PLANETARY GEAR SET DESCRIPTION

Nominal dimensions of the planetary gear set used in the experimental study are summarized in Table 1. Pinion and sidegear were ground to achieve close agreement to the desired profile and lead traces. Ring gear, not being ground after heat treatment, required several iterations to achieve satisfactory maximum lead error of 16 μm . In the course of the iterations, several ring gears with lead error of up to 40 μm were produced. They were used in the course of the experimental study as the representatives of the high lead error ring gears.

Sun and ring gears had only tip relief, while pinion had both, tip and root relief. In addition, only pinions had modification in lead direction (nominally 20 μm).

Two additional levels of the pinion profile and lead modifications were manufactured in order to study their influence on the NVH performance of the Transfer Case. Summary of the modifications is given in Table 2. Correspondingly, four possible combinations of the modifications were used in the experimental study. Only the

Table 1. Planetary gear set dimensions

	Sun	Pinion	Ring
Number of teeth	55	17	89
Module		1.59	
Helix angle, [deg]		18.25	
Pressure angle, [deg]		20	
Outside diameter, [mm]	95.77	33.25	-
Root diameter, [mm]	88.32	26.1	-
Minor diameter, [mm]	-	-	149.77
Center distance, [mm]		122.63	-
Face width, [mm]	35	35	35

Table 2. Pinion lead and profile modifications

Lead modification (parabolic crowning)		Profile modifications (tip and root relief)	
High [μm]	Low [μm]	High (tip/ root) [μm]	Low (tip/ root) [μm]
20	10	30/ 20	15/ 10

magnitude of the profile (tip and root) relief on the pinion was varied, while the start was kept at the approximately same point (roll angle).

Planetary gear set was designed to carry six equally spaced pinions, but only 4 pinions were used. Consequently, pinions were not equally spaced, the angle between them being 60° or 120° (X-configuration). In order to study the influence of planet load sharing and pinion pin slope error, four carriers were modified to include pinion tangential position error, pinhole tilt error in tangential direction and their combination. The summary of the induced carrier manufacturing errors is given in [Figure 1](#). Carrier #1 had all of its pinholes placed at their perfect (design) position. Carrier #2 had its pinholes shifted in tangential direction by 75 µm and then only the bottom holes were shifted in the opposite direction by 80 microns. The tangential shift from true position (TP) of 75 µm was made in such manner ([Figure 1b](#)) that the worst expected pinion load sharing would be achieved. [Figure 1c](#) shows the Carrier #3 with pinions at their design positions (TP=0), and bottom and top pinholes pushed by 40 µm in opposite directions along tangent to the pinhole circle. Such misalignment acts as 80 µm pin tilt (slope) error. Finally, [Figure 1d](#) shows the Carrier #4 with both (top and bottom) pinholes pushed by 75 µm in tangential direction in such way to cause the most damage to pinion load sharing. The inspection of the carriers showed that the intended misalignments were closely matched by the manufacturing process.

EXPERIMENTAL STUDY

PRELIMINARY EXPERIMENTAL WORK

Prior to the actual experimental study an attempt was made to determine the influence of pinion tangential position error, lead modification amount and ring gear lead error on static transmission error (TE) of the planetary gear set. This attempt was not successful and no influence of these factors on TE was detected. There are several possible reasons for such a behavior. First, slow gear rotation (limited by the motor used) was possibly not enough for proper positioning of the floating members. In addition, it was noticed that starting position of the gears in the assembly had influence on the results. Much longer data acquisition procedure could be used to diminish the influence of the starting gear position. The system with the gears specified in the [Table 1](#) would require 935 rotations of the carrier which would require approximately 6 hours of data acquisition with the current setup.

EXPERIMENTAL SETUP

For the experimental study, the transfer cases were installed in a MULE level light duty 4WD pickup using the standard vehicle hardware (no special fixturing was used). The vehicle was then installed on a four roll chassis dynamometer (CHD) housed in a semi-anechoic chamber. For drive (positive) speed sweeps, the vehicle engine was used to regulate speed while the dyno was used to control torque. For coast (negative) speed sweeps, the rolls were used to regulate the speed, with the vehicle drivetrain regulating torque.

The vehicle was instrumented to acquire rpm, vibration, and sound pressure data. The rpm was measured at the front output and rear output of the transfer case (on the driveshafts). Interior sound pressure was measured at the Driver's Right and Left Ear as well as the rear right passengers left ear. In addition, near field microphones were placed at the front and rear output of the transfer case along with directly over the ring gear. All vibration measurements were taken on the transfer case housing at the following locations: Above the rear output, below the front output, on the rear bearing of the front output, and alongside the ring gear of the transfer case. An example of the last location can be seen in [Figure 2](#). The active and passive side responses of the transmission mount were also measured.

EXPERIMENTAL PROCEDURE

For all build iterations, the transfer case housing, ring gear, chain, and output shafts were constant. There were a total of four carriers and four sets of pinions as described above. The press pins and needle bearings were replaced for each build. The lubrication was reused, with additional lubrication being added as required.

The testing focused on drive loading, as coast loading events would most likely be brief in real-world applications. A constant output sum torque of 130 Nm was applied during speed sweeps from 20-80 kph with a ramp rate of 0.35 kph/s a total of three times for each build iteration. As the vehicle was a manual, the transmission gear was held constant. The gear was selected to allow the entire speed range to be tested without shifting.

Data was acquired with a maximum frequency of 10,000 Hz with 6401 spectral lines (25.6 kHz sampling frequency). The first two mesh orders were calculated with constant order slice width and 20 Hz smoothing with frequency abscissa, calculated relative to the rear output tachometer. An amplitude average of the three runs was calculated for each of the mesh orders, with particular focus on first mesh. An example of the mesh order results can be seen in [Figure 3](#).