International Journal of Precision Engineering and Manufacturing (2019) 20:1969–1977
https://doi.org/10.1007/s12541-019-00170-5

Online ISSN 2005-4602
Print ISSN 2234-7593

REGULAR PAPER

Experimentally Validated Geometry Modification Simulation for Improving Noise Performance of CVT Gearbox for Vehicles

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Received: 18 March 2019 / Revised: 22 May 2019 / Accepted: 3 June 2019 / Published online: 6 July 2019

Abstract
Noise pollution has become one the most important environmental issues in modern life. The automobile, a product of the second industrial revolution, has a very large sales volume of approximately 100 million per year. With the popularity of automobiles, the noise generated by them has become a major cause of noise pollution. This paper presents an experimentally validated geometry optimization method for reducing the noise by a CVT gearbox used in motor vehicles. We used the CAE software RomaxDESIGNER to simulate the transmission error and the load distribution on meshing gear tooth surfaces before and after gearbox optimization. After determining the best modification values, we carried out a series of noise bench comparison tests to verify the simulation results. The test results confirmed that the optimization reduced the fluctuation in noise that occurred during certain speed stages of the gearbox. Due to the optimization, the overall noise level decreased and the noise curve became smoother.

Keywords CVT gearbox · NVH improvement · Gear flank modification · Transmission error · Noise pressure test

1 Introduction

In recent years, noise, vibration, harshness (NVH) has become widely considered as an important indicator for the evaluation of automotive products. Rotating bodies such as engines, gearboxes, alternators etc. generate noise and thus affect passenger ride comfort. Noise is produced due to vibration, it can cause damage to human hearing, induce a variety of diseases and interfere with people's lives and work. When the frequency of the vibration is close to the natural frequency of the system, the resonance phenomenon occurs, this is a key factor affecting the life of the mechanical system. A considerable amount of research has been carried out on the reduction of NVH of gearboxes [1–6].

Continuously variable transmissions (CVTs) are widely used in various types of vehicles because they improve fuel economy and passenger comfort. In a CVT gearbox, the distance of the V-groove formed by the conical discs, controlled by the hydraulic drive mechanism, can be changed according to different loads. With this change, the transmission belt adjusts its radial position under the squeeze of the conical discs, thus achieving the change in transmission ratio. Because of its design, its manufacturing, or its own structural characteristics, the gear mechanism of the CVT still has the problem of gear whine. Micro geometry modification of the meshing gears is the main means of reducing this problem. In the contemporary context, designers rely more on experimental methods to reduce the noise problems of gearboxes [7–10]. This method may have a high cost and require a long test time. To meet the design requirements, repeated trials and design changes have to be done, and there is not much knowledge crossover between the designs of different gearboxes. This article presents a NVH optimized design of the CVT gearbox used in a best-selling SUV. The

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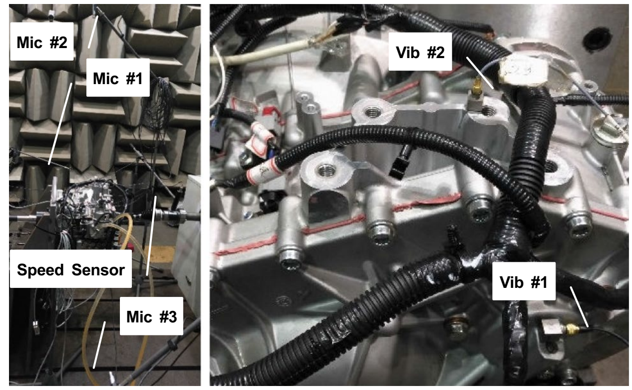
finite element method (FEM), was utilized during the design and development process. [11–14] In order to accurately simulate the contact and to shorten the design time, we used a FEM based CAE software to analyze gear transmission error and load distribution on meshing gears. Based on the simulation results, we improved the micro geometry modification until we achieved the optimal modification value. To countercheck the results, the optimized prototype was bench tested to verify the correctness of the simulation.

2 Background and Discussions

Generally, design process of a gearbox is divided into two stages, the macro geometry basic design stage and the micro geometry modification stage. The macro geometry basic design includes the number of teeth, module, gear tooth width, pressure angle, the helical angle, modification coefficient, gear material etc. After the initial macro geometry design stage, we manufactured a prototype of the system and evaluated it using a noise bench test. From these preliminary test results and the design structure check, we found out that the main noise sources in the gearbox were the second and third gear sets, as shown in Fig. 1. Based on these observations, we performed further noise tests on the second and third gear sets.

During testing, we placed the gearbox in a semi-anechoic room with an indoor noise floor of 20 dB. Figure 2a is the installation diagram of the sound sensors. According to the test standard, we ensured that the three microphones were located in the same sectional plane of the center of the gearbox and were 1 m away from the measured part. In order to obtain the vibration signal accurately, two acceleration sensors were placed on the concerned part of the gearbox chassis, as shown in Fig. 2b, where Vib #1 is a X–Y–Z direction three-way sensor and Vib #2 is a Z direction one-way sensor. The test analysis equipment utilized the B&K 3560C data acquisition front end and B&K PULSE software to collect and process data. Transmission oil was cooled by an external water cooling system. We changed the rotation speed at a constant rate of 10 rpm/s. According to the test standard, the input torque was set as 300 Nm, 430 Nm, 560 Nm, and the input rotational speed was set to range from 1000 to 3000 rpm. The order of measurement is determined by the number of meshing gear teeth. Table 1 shows the meshing gear parameters and measuring order of noise.



(a) Sound sensors (b) Acceleration sensors

Fig. 2 Sensor installation diagram

Table 1 Number of gear teeth and measuring noise order

Gear	Number of gear teeth	Order
Secondary gear	31	170.85
Reversing driven gear	51	
Intermediate gear	20	67
Crown wheel	67	

Figure 3a–c show the sound pressure noise curves of the gears for torque values of 300 Nm, 430 Nm, 560 Nm, respectively. Here, it is clearly observable that there is a large fluctuation in the overall noise curve in the input speed band between 600 and 680 rpm under each working condition. After disassembly, it was found that the gear tooth surface was scratched as shown in Fig. 4, it was preliminarily determined that the load distribution on tooth surface was uneven, and the top of the gear tooth had sharp edges, which resulted in tooth surface scratching and gear whine.

Subsequently, we used the RomaxDESIGNER software to simulate and analyze the contacts of the second and third gear sets under various actual load conditions. We repeatedly modified the micro geometry based on the results of the transmission error and the load distribution on each meshing gear tooth surface under various working conditions.

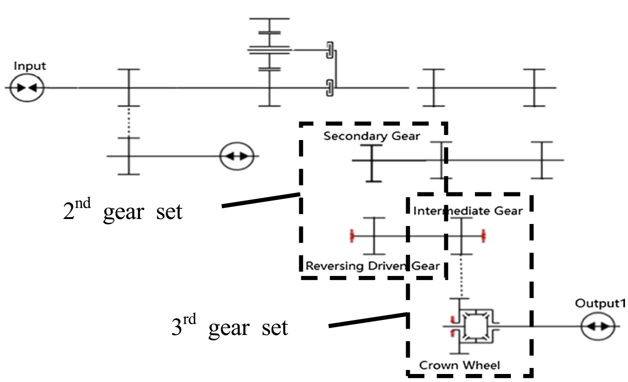
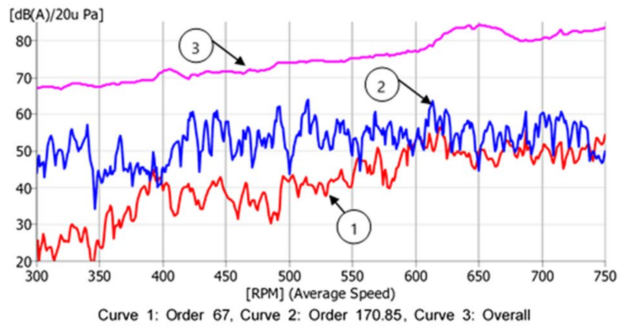
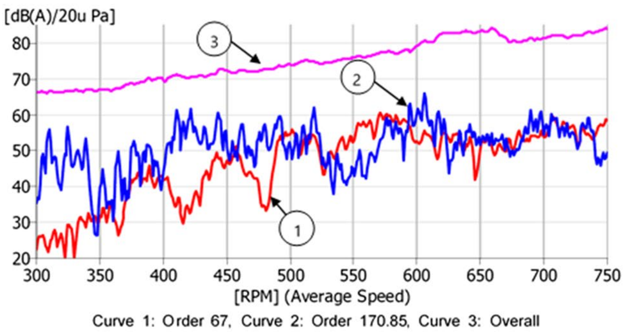


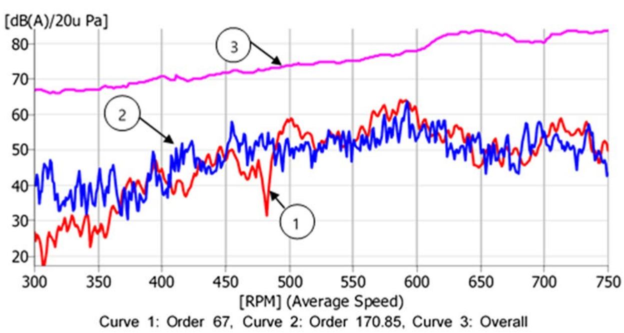
Fig. 1 Structural diagram of a CVT gearbox



(a) Under 300Nm load condition



(b) Under 430Nm load condition



(c) Under 560Nm load condition

Fig. 3 Results of noise sound pressure

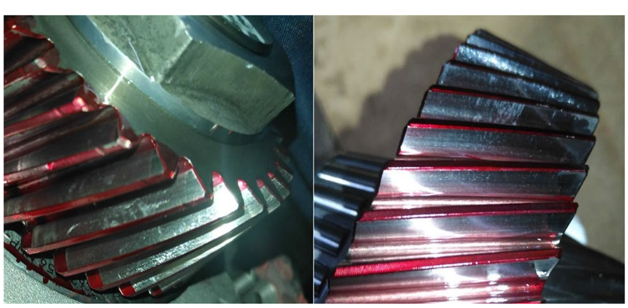


Fig. 4 Disassembled view of meshing gears

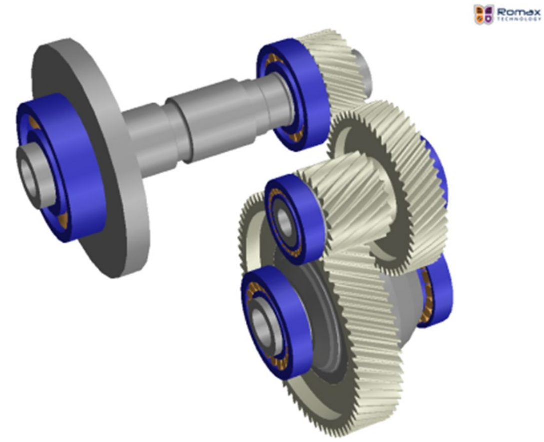


Fig. 5 Model of the gearbox used in this study

Table 2 Load spectra

Load case	Input speed (rpm)	Input torque (Nm)	Duration (h)
LC1-2.631-ELOW	1754	544.617	2.49
LC2-2.304-LOW	1519	599.04	8.37
LC3-1.000-MEDIUM	5000	174	72.41
LC4-0.684-TOP	8041	119.016	94.14
LC5-0.416-OD	8414	108.16	81.4
LC6-0.378-EOD	7516	98.28	103.22

3 Simulation and Discussions

Based on the parameters obtained during the macro geometry design stage, the second and third gear sets of the CVT gearbox were set up as shown in Fig. 5. In this configuration, the power is transmitted from the upper CVT driven shaft to the intermediate gear shaft through the second stage gear pair. It is then transmitted to the differential gear shaft through the intermediate gear and crown wheel of third gear set. According to the actual working conditions of the CVT gearbox, we included the tension of the belt at the input driven shaft in the modeling of the system. We also kept the models and positions of all the bearings the same as in the real product. We developed a more accurate working load cases table according to the actual operation requirements, as shown in Table 2.

Transmission error (T.E.) is the most important factor affecting gear noise and vibration. In reality, the rotation angle of the driven gear tends to deviate from the

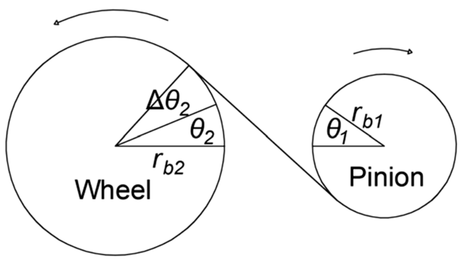


Fig. 6 Schematic diagram of transmission error

theoretical value when gears meshing. Generally, T.E. can be expressed as an angular deviation, or a linear deviation measured at the pitch point and calculated at successive positions of the pinion during gear meshing cycle. Figure 6 shows the schematic diagram of T.E., and the calculation formula is given below:

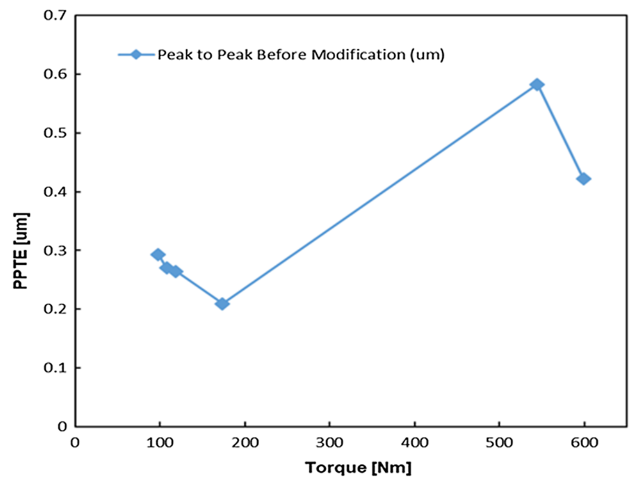
$$\mathrm{TE} = \theta_2' r_{b2} - \theta_1 r_{b1} \tag{1}$$

where

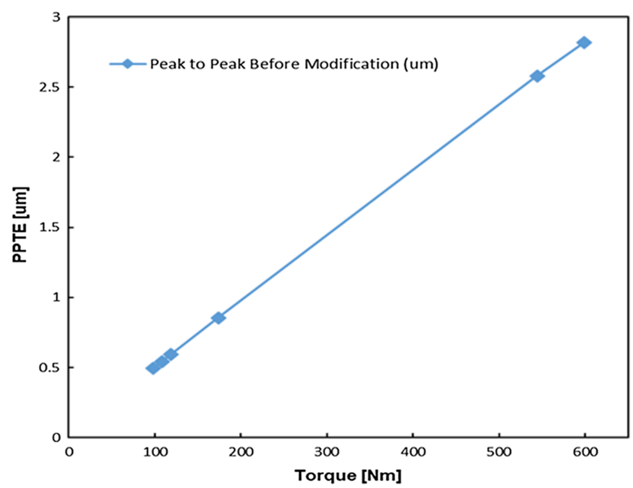
$$\theta_2' = \theta_2 + \Delta\theta_2 \tag{2}$$

Peak-to-peak transmission error (PPTE) is the difference between the maximum transmission error and the minimum transmission error during roll angle from base of one gear tooth. Under certain working conditions, larger PPTE values mean larger amplitude of the gear vibration.

Firstly, we conducted a contact analysis on the model before modification to observe the current transmission error and load distribution status of the meshing gears. The transmission error and stress induced in gear micro geometry are very complex in nature, therefore use of traditional calculation methods may waste a lot of time. The correctness of results obtained using such methods is also difficult to guarantee. In this research, we used the software to analyze the contact status of meshing gears. RomaxDESIGNER is an advanced CAE software used for the design and analysis of mechanical and electromechanical transmissions in the automotive industry. We extracted the PPTE value for every load case from the simulated TE results. Figure 7a is a depiction of the PPTE values of the second gear set under each of the working conditions. As can be seen, the maximum PPTE (0.58292 μm) of the second gear set occurs in the case where 544.617 Nm torque is applied at 1754 rpm rotation speed. Figure 7b is an illustration of the PPTE of the third gear set. The PPTE value is almost proportional to the input torque. Under the LOW load case of gear ratio 2.304, the PPTE reaches the maximum value of 2.82 μm.



(a) Second gear set



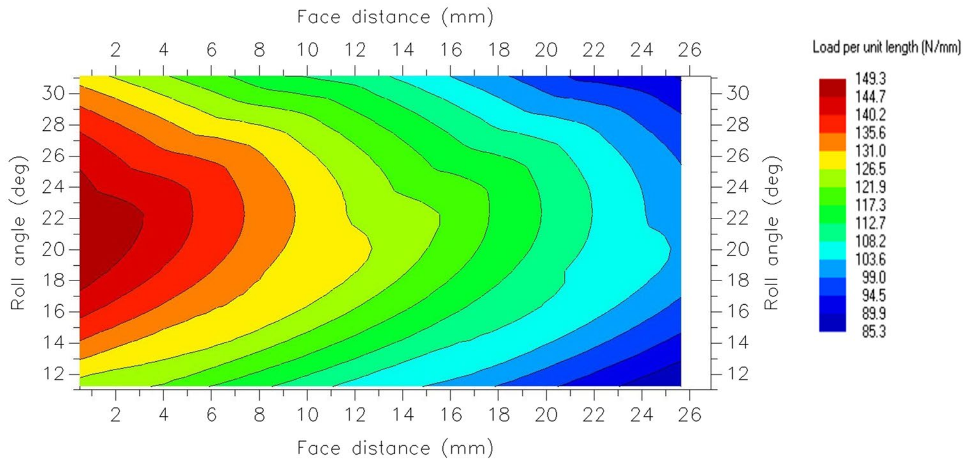
(b) Third gear set

Fig. 7 Results of PPTE before optimization under different load cases

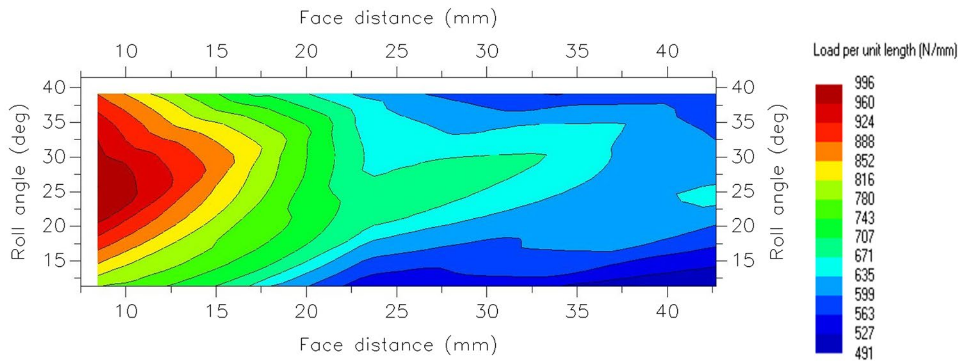
Figure 8a, b respectively show the load distribution analysis results of the contact surface of the second and third gear sets in the MEDIUM load case, which is the most commonly used gear ratio condition. It can be observed that the maximum loads in both the case are located on the left edge of the gear tooth surface. This local concentration of load can easily cause tooth surface wear, affect the endurance of the gear and even lead to gear failure. The purpose of gear micro geometry modification is also to shift the maximum load contact pattern to the center of the tooth surface.

Subsequently, we carried out empirical gear micro geometry modifications. As the name implies, gear micro geometry modification refers to the process of micro-trimming the gear tooth surface to make the gear flank form deviate from

Fig. 8 Results of load distribution analysis before optimization under MEDIUM load case



(a) Second gear set



(b) Third gear set

Table 3 Optimal modification values of gear flank

Gear	Flank	Type	Value (μm)
Secondary gear	Both	Lead crown	1.6
		Lead slope	− 0.3
		Involute barrelling	0.3
		Involute slope	1
		Linear tip relief	11.7
Reversing driven gear	Both	Lead crown	2.9
		Lead slope	1.2
		Involute barrelling	2
		Involute slope	− 1.2
		Linear tip relief	11.3
Intermediate gear	Both	Lead crown	3.8
		Lead slope	4.6
		Involute barrelling	1
		Involute slope	0.6
		Linear tip relief	15.3
Crown wheel	Both	Lead crown	2.7
		Lead slope	3
		Involute barrelling	0.9
		Involute slope	0.7
		Linear tip relief	11.9

the theoretical tooth surface. According to their different objectives, modifications can be divided into two categories: the tooth profile modification and the tooth lead modification. RomaxDESIGNER software provides the design of experiments (DOE) function that calculates the results automatically after defining the variables and target values, but the huge data processing requires a lot of calculation time, therefore, we chose to use the empirical modification method.

In this method, we repeatedly changed the modification value after analyzing the simulation results. We continued this process until we achieved the optimal modification value. In order to eliminate the sharp edge mentioned in the previous section, we also took into account a big end relief during our search for an optimal modification value. Table 3 shows the optimal modification value of gear flank form obtained after a substantial number of calculations and comparisons.

Figure 9 shows a comparison curve obtained by the integration of the PPTE values from before and after the optimal modification calculated by TE results. It is clearly visible that the maximum PPTE values of the two gear pairs show significant improvements as compared to those from before

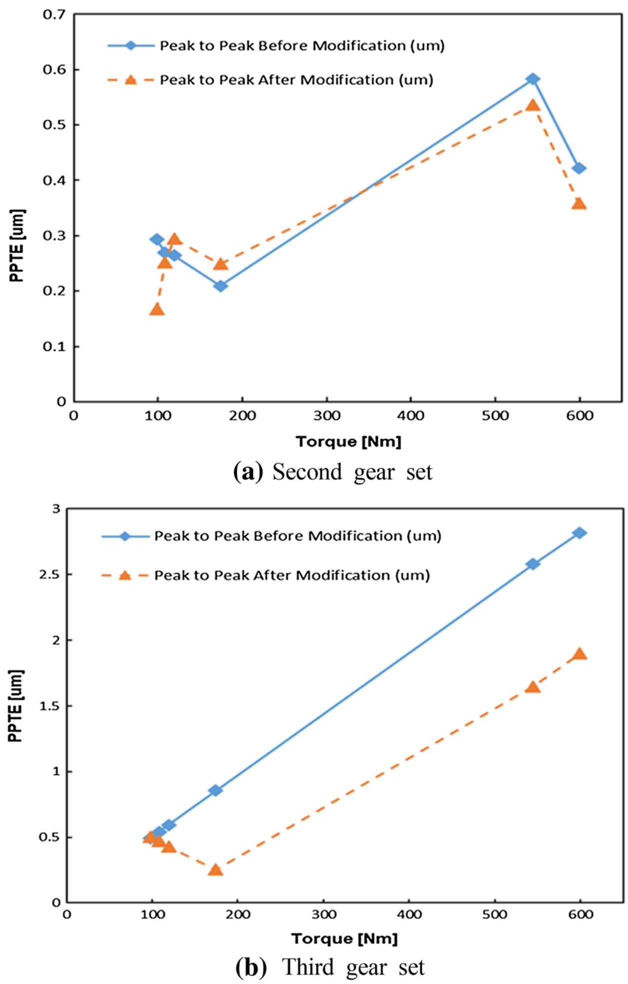


Fig. 9 Comparison of PPTE values before and after optimization under different load cases

the modification. Especially in the third gear set, the maximum PPTE value of 2.82 µm before modification occurred in the LOW condition, this has been reduced by 30% after optimization.

Figure 10a, b respectively show the after modification load distribution on tooth surfaces of the second and third gear sets under the MEDIUM load case. Here, it is visible that the maximum contact stress of both the gear pairs has moved to the center of the tooth surface, which is the best state for gear transmission. Stress concentration at the edge of tooth surface can easily lead to gear failure. Afterwards, we conducted the bending and pitting strength safety factor analysis of each gear. According to the results of this study, shown in Fig. 11, the safety factor of each gear is above 1.0, which is the industry standard requirement. In other words, the results show that the geometry design and optimization modifications are satisfactory.

In the next step, we performed the noise bench test on the modified design to verify the correctness of the modification simulations. Details of this experimental verification are presented in the next section.

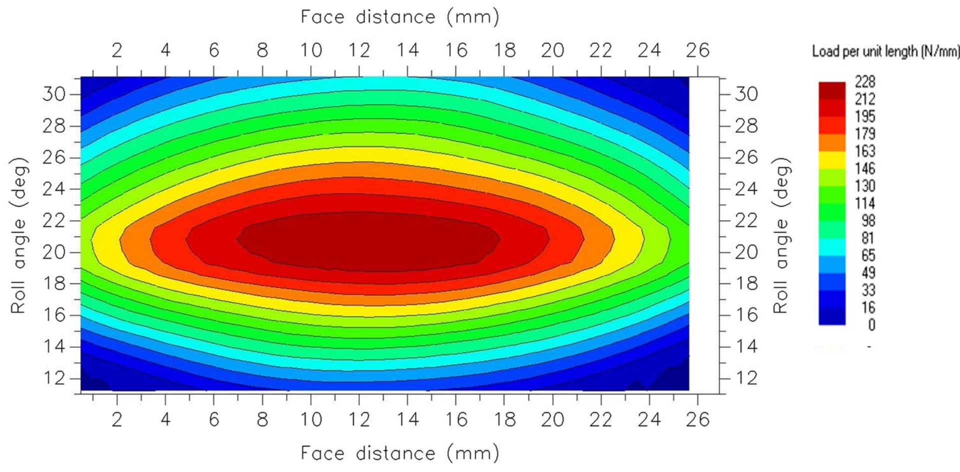
4 Experimental Verification

We carried out the noise bench test on the modified gearbox prototype to observe the noise levels after the micro geometry modifications The test conditions were same as those described in the second section.

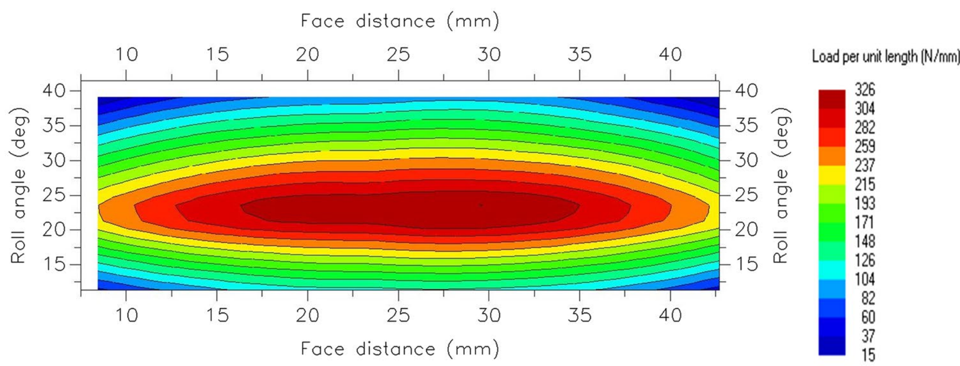
Figure 12 shows the overall noise pressure and 170.85 order noise pressure from before and after the gear modifications under three load cases of 300 Nm, 430 Nm and 560 Nm. Curve 1 is the overall noise before optimization, curve 2 is the overall noise after trimming, and curve 3 and curve 4 are respectively the 170.85 order noise values from before and after modifications. These plots show that the large fluctuation in the overall noise value between the input shaft speeds of 600–680 rpm under three load conditions mentioned in the second section has disappeared after micro geometry modification. It is also observable here that the curve of overall noise level is now much smoother as compared to the pre-modification curve. The 170.85 order noise results also confirm that the noise after modification has been reduced. In summation, the noise test results after gear flank modification are satisfactory, and the correctness of simulation results has also been verified (Fig. 13).

5 Conclusions

In this study, we performed a series of simulations and comparison bench tests to verify the correctness of the empirical modification values. Considering the micro geometry



(a) Second gear set



(b) Third gear set

Fig. 10 Results of load distribution analysis after optimization under MEDIUM load case

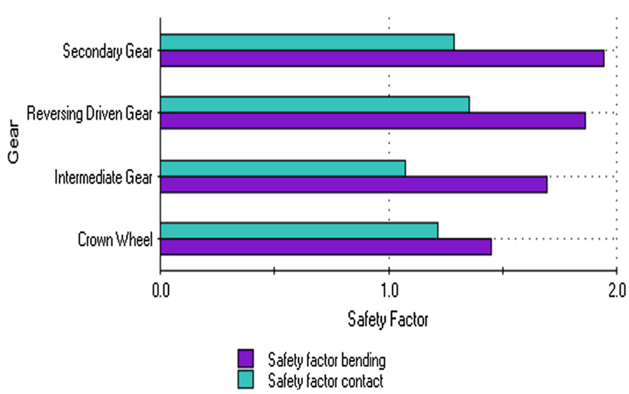
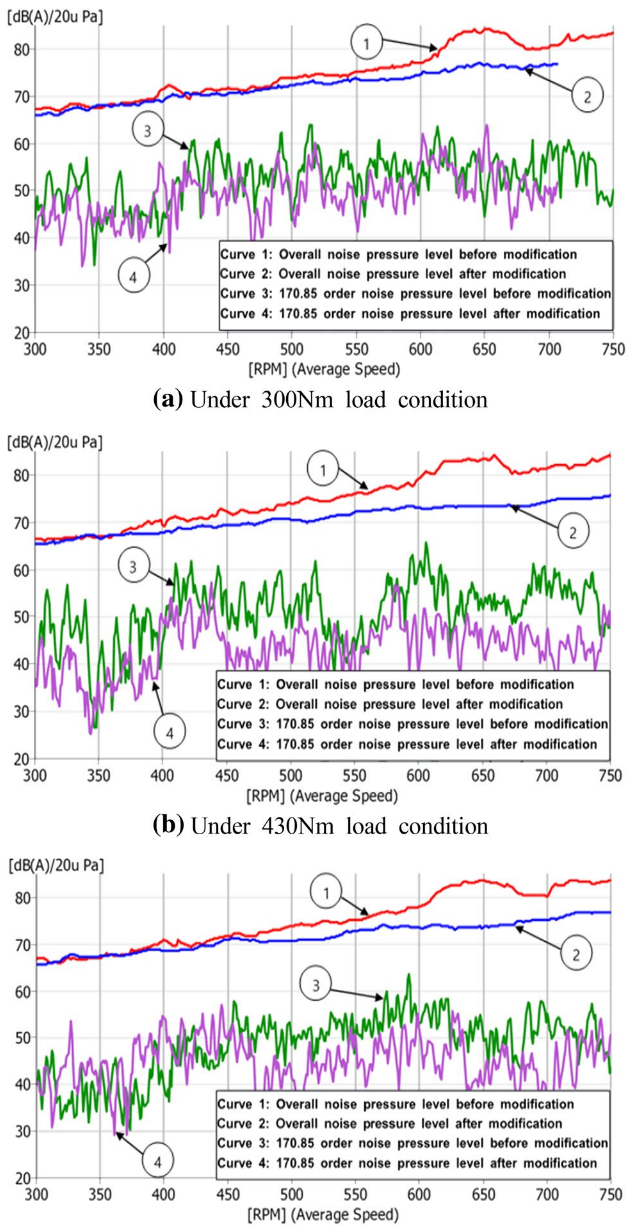


Fig. 11 Results of strength analysis after modification

analysis and the noise pressure tests, we have drawn the following conclusions.

1. Through the comparison of before and after gear modification PPTE values under various working conditions, we can see that the PPTE after modification of the second gear set under the EOD load case with input torque 98.28 Nm, was reduced by 43% as compared to that prior to gear flank modification. The maximum PPTE value for the 544.617 Nm under ELOW condition before modification was 0.583, which was reduced by 7% after modification. On the other hand, after modification of the third gear set, PPTE was reduced by 70% under the MEDIUM load case of torque (174 Nm). The maximum PPTE value of 2.82 μm before optimization appeared in the LOW condition, which was reduced by 30% after modification. It is evident here that the empirical modification values play a direct role in improving PPTE.
2. To avoid gear failure caused by the fragile edges of the gear flank, sufficient sampling data was used in finding the optimal modification value. The results graphically show that the contact pattern of the gears was shifted to the center of the tooth surface. The load on each edge is now almost zero, and the maximum load now applied



(a) Under 300Nm load condition

(b) Under 430Nm load condition

(c) Under 560Nm load condition

Fig. 12 Comparison of noise sound pressure under different load cases before and after the gear modifications

at the center and decreases progressively in all directions. In other words, the optimal modification has also improved the load distribution on meshing gear surface.

3. It can be seen from the noise test results that the noise fluctuations in the speed band between 600 and 680 rpm disappeared after empirical modification, and the overall noise curve also improved significantly. This means that the results of the noise test verified the correctness of the gear geometry modification simulation.

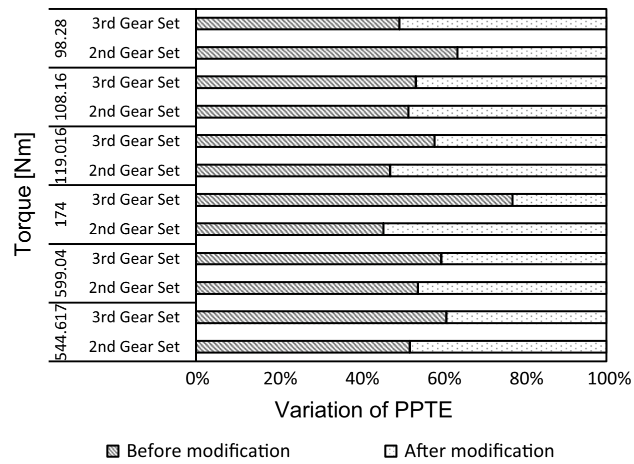


Fig. 13 Comparison of PPTE before and after modification

Acknowledgements This study was supported by the Basic Science Research Program through the NRF of Korea (NRF) funded by the MEST (NRF-2016R1D1A1A09920105) and RomaxDesigner of Romax Technology.

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