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Design and evaluation of two-stage planetary gearbox for special-purpose industrial machinery†

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Abstract

The gearbox, as a traditional machine that changes the transmission ratio and transfers power safely, stably, quietly and efficiently, is an irreplaceable component in the field of machinery design. The planetary gear train is the most widely used of the traditional gearbox designs. Although scholars have carried out a variety of research on gearboxes, gear failures, noise and other issues are still common. This article presents a study involving the initial macro geometry design, gear flank modification, static and dynamic analysis, and experimental verification of a two-stage planetary gearbox for special-purpose industrial machinery. Compared to traditional static transmission error simulation, this study presents a novel analysis of the gearbox overall dynamic transmission error. In this research, the modal flexibility is also analyzed to determine the possibility of resonance in the gearbox. As a result, the strength of this gear system is guaranteed by macro design. PPTE (noise evaluation index) of each gear pair is greatly improved after flank modification. The efficiency of power transmission system is also improved with the improvement of vibration. The result of final bench test of the prototype is also quite satisfactory. This also verified the correctness of the theoretical simulation method presented in this research.

Keywords: Geometry design; Planetary gearbox; Dynamic transmission error; Modal flexibility analysis; Efficiency

1. Introduction

Different types of gear trains have different characteristics. So, different gear trains are used according to different requirements. Planetary gear trains are used in a wide variety of applications such as automobiles, aircraft, wind turbines, industrial machinery etc. because of their desirable characteristics of being light, compact, and having low noise, while giving high performance. In recent years, strength, durability and NVH (noise, vibration, harshness) have become widely considered as important indicators for the evaluation of gearboxes [1-4]. Although gear technology has been around for thousands of years, gear failure and gear noise are still the common problems arising from its design, its manufacturing, or its own structural characteristics. A substantial amount of research has been carried out on the optimization of gearboxes. For gear fault, Han et al. have carried out some novel methods to monitor and diagnose the faults of gear failure [5-7]. Research of dynamic characteristics and vibration of gearbox have been reported by Xu et al. [8-15].

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Many designers try to increase the strength of gears by means of tooth surface modification. However, the strength of gears is greatly dependent on the initial macro geometry design that mainly consists of the definition of number of teeth, module, gear tooth width, pressure angle, helical angle, modification coefficient, gear material etc. To improve NVH performance of gearboxes, designers rely mainly on empirical methods. However, the experimental method can be costly and time consuming as repeated trials and design changes have to be done to meet the final design requirements, and there is not much knowledge crossover between the different gearboxes.

Static analysis is the method of choice of researchers working on the noise problem of gearboxes. However, the consideration of dynamic factors can increase the accuracy of this work. Therefore, in this research modal flexibility calculation which is a dynamic analysis method has been utilized. This novel application of dynamic analysis can help increase the accuracy of the simulation outcomes. This paper presents a study on a two-stage planetary swing gearbox widely used in special-purpose industrial machinery such as excavators, which require high reliability and good NVH performance while performing reciprocating rotational operations. In order to achieve the required performance, the geometry design method was used and a series of evaluation analyses were carried out. In this work, the empirical formulae prescribed by various international gear standards were used to define the macro geometry parameters. Afterwards, the commercially available software RomaxDESIGNER was utilized to model the research object and analyze the strength of the macro geometry design [16]. Gear modification, and results repeatedly observation of static transmission error and load distribution obtained from micro geometry contact analysis were carried out. The advanced dynamic analysis of dynamic transmission and efficiency calculation of the modified model were also implemented in this research. To countercheck the results, bench test was carried out to verify the durability of the optimized design.

2. Macro geometry optimal design

As mentioned above, macro geometry is more effective in improving the durability and strength of gear teeth. To meet the gear ratio and output torque requirements, a two stage planetary gear train is utilized in this design. Several methods can be adopted for connecting two planetary gear trains, such as simple epicyclic method, compound epicyclic method, coupled planetary method, differential planetary method etc. [10]. Considering the transmission ratio and strength requirements of this design, the simple epicyclic method was used in this design. As shown in Fig. 1, the power input to the gear train is through the sun gear of the first stage, and output is through the planetary carrier of the second stage. The first and second stages of the gearbox share a fixed ring gear. Power from the planets of the first stage is directly transferred to the sun gear of the second stage. The total gear ratio is calculated using the following equation:

$$i_{total} = \left(\frac{Z_{r1}+Z_{s1}}{Z_{s1}}\right)\times\left(\frac{Z_{r2}+Z_{s2}}{Z_{s2}}\right) \tag{1}$$

The number of gear teeth and module affects the size of the gearbox, however, the appearance size is limited by the space of the whole machinery. Therefore, during the macro geometry design of planetary gear train, in addition to considering the conditions that determine the number of gear teeth, the external limitations must also be taken into account. Considering the gear strength requirement, the module of the entire gearbox was defined as 2.0 mm, and the number of gear teeth were defined according to the geometric relationship. As described in ANSI/AGMA 6123 standard: Assuming deflection or other small errors in tooth action, a non-factorizing system is theoretically smoother and quieter than one that factorizes for torsional vibrations. Therefore, the non-factorizing design was applied in this research.

Specific sliding of gears is defined as the ratio of gear tooth sliding velocity to rolling velocity, which is related to the wear of tooth surface. The specific sliding calculation was con-

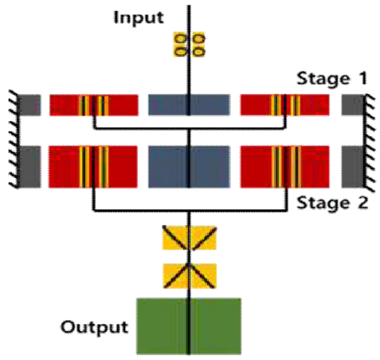
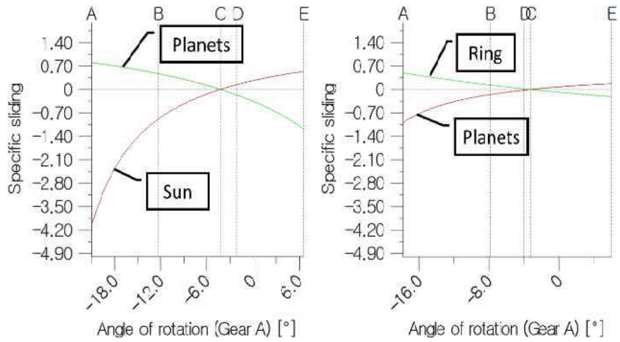
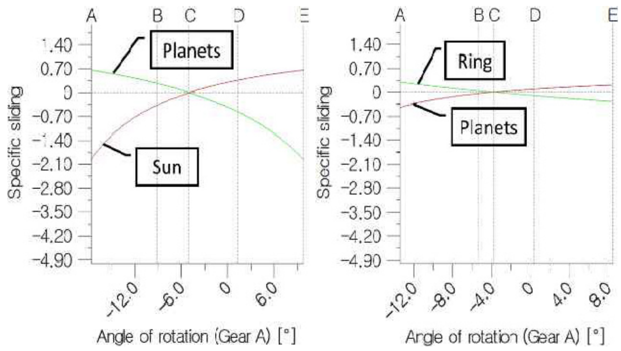


Fig. 1. Constituent parts of the conceptual design.



(a) Sun and planets (b) Planets and ring

Fig. 2. Specific sliding before profile shift modification.



(a) Sun and planets (b) Planets and ring

Fig. 3. Specific sliding after profile shift modification.

ducted using KISSsoft software. As shown in Figs. 2(a) and (b), there is a big difference in specific sliding at each end of the path of contact. Namely, the sliding velocity on the contact surface of meshing gears shows uneven distribution, which may lead to localized wear on the gear tooth surface. The profile shift modification was utilized to balance the specific sliding at each end of the path of contact to minimize wear of gear teeth. From the results of specific sliding ratio as shown in Figs. 3(a) and (b), the sliding velocity has become relatively uniform can be observed.

There can be several combinations of macro geometry parameters that satisfy the design conditions. Combined with the

Table 1. Optimal macro geometry design parameters.

Items	1st stage			2nd stage		
	Sun	Planet	Ring	Sun	Planet	Ring
No. of planets	3			3		
Module [mm]	2			2		
No. of teeth	19	26	71	19	26	71
Pressure angle [deg.]	20			20		
Helix angle [deg.]	0 (spur gear)			0 (spur gear)		
Facewidth [mm]	15	13	15	19	18	20
Profile shift coefficient	0.2919	0.1334	-0.5587	0.2919	0.1334	-0.5587
Center distance [mm]	45.8			45.8		
Gear ratio	22.4					
Material & heat treatment	SCM 420H	SCM 420H	GCD 70	SCM 420H	SCM 420H	GCD 70
Gear accuracy	ISO 8	ISO 8	ISO 8	ISO 8	ISO 8	ISO 8

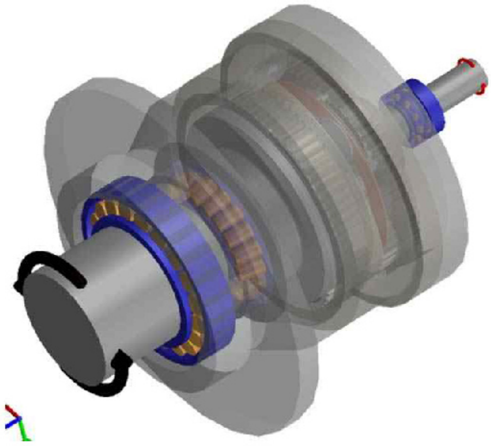


Fig. 4. 3D model of gearbox system.

strength calculations mentioned below, the optimal macro geometry design parameters were summarized as given in Table 1. According to the parameter that was obtained from macro geometry design, the basic modeling was established utilizing RomaxDESIGNER, Fig. 4 graphically shows the basic modeling of the proposed gearbox.

There are many different methods for evaluating the strength of gears. Among these, numerical analysis has nowadays been widely adopted as an accurate and more efficient method. In this research, the gear strength calculations for the proposed gearbox were carried out using numerical analysis software based on standards established by International Organization for Standardization (ISO). Results of the strength analysis are presented in the form of the histogram shown in Fig. 5. It can be seen from the histogram that the safety factor of every gear in each stage is above 1.0 which is the industrial safety requirement. In other words, this result verifies that our macro geometry design meets the specified design targets.

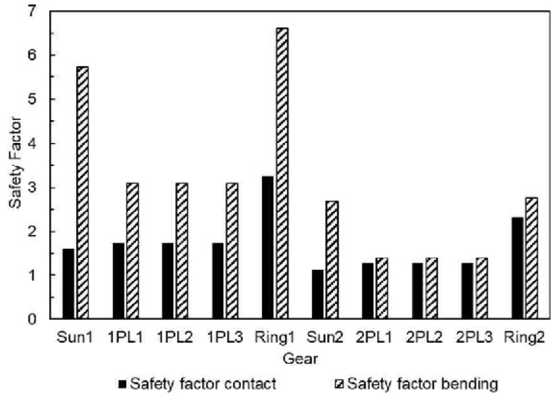


Fig. 5. Duty cycle safety factors of gearbox.

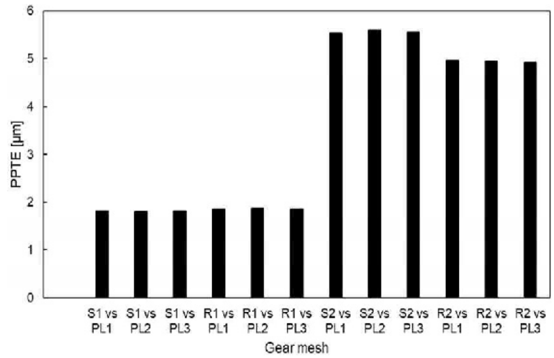


Fig. 6. Results of PPTE before optimization.

3. Gear flank modification and results comparison

Most gear failures result due to damaged tooth edges, which are the most fragile parts of gears. Therefore, one aim of the work presented in this section is to distribute the load evenly at the center of every tooth surface by using gear micro geometry modification. Transmission error (T.E.) is the most important factor affecting gear noise and vibration. PPTE (peak-to-peak transmission error) 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. Thus, minimizing the PPTE is the second aim of the work presented in this section.

Firstly, a contact analysis of the model before modification was conducted to observe the transmission error and load distribution status of the meshing gears. The transmission error and load distribution induced in the gear micro geometry are very complex in nature, to reduce the calculation time and guarantee the correctness of results, the commercial software RomaxDESIGNER was used to analyze the contact status of meshing gears. The input load case was defined as 875 rpm and torque 37.8 Nm, based on the actual working conditions. The PPTE value for every gear mesh from the simulated TE results was extracted, as can be seen in Fig. 6. The maximum PPTE (5.61 μm) of the second planetary gear train occurs between the sun gear and planet gear number 2. Fig. 7 shows the load distribution analysis results of the contact between the

Table 2. Optimal modification values of gear flank.

Gear	Flank	Type	Value [μm]
1st sun	Both	Involute barrelling	8
		Involute	9
		Slope	
1st planets		Lead	3
		Crown	
		Involute	5
		Barrelling	
		Involute	8
		Slope	
2nd sun		Lead	3
		Crown	
		Involute	5
		Barrelling	
		Involute	8
		Slope	
2nd planets		Lead	8
		Crown	
		Involute	9
		Barrelling	

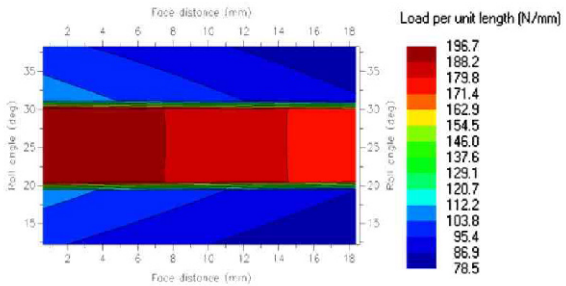


Fig. 7. Load distribution of second stage sun gear and planet gear before modification.

sun gear and planet number 2 of the second planetary gear train. This is where the maximum PPTE occurred. It can be observed that the overall load is in a single shape and the maximum load is located on the left side of the gear tooth surface. This local concentration of load can easily cause tooth surface wear, affecting the endurance of the gear and may 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.

After understanding the problem, empirical gear flank modifications were carried out. In this method, the modification value needs to be repeatedly changed after analyzing the simulation results. Although the modulus and the number of teeth of the gears in both the stages are same, the mesh misalignment is different due to the different face widths and axial positions. Due to this kind of problem, different modification values were given to the first and second stage gears. Theoretically speaking, each contact gear can be given a mi-

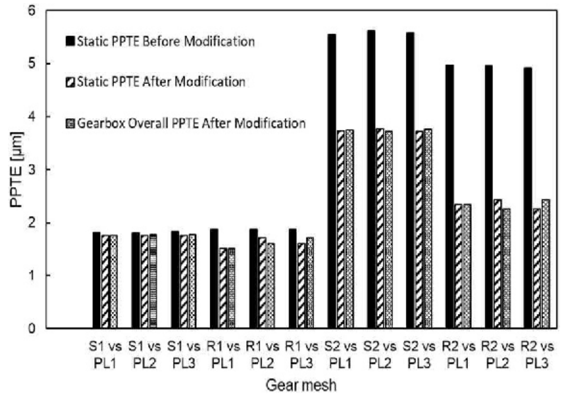


Fig. 8. Comparison of PPTE values before and after optimization.

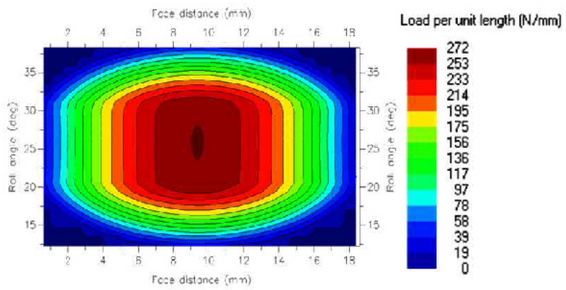


Fig. 9. Load distribution of second stage sun gear and planet gear after modification.

cro geometry modification. However, in practice, due to the production process of planetary gearboxes, the modification of ring gear is very difficult, and its modification will increase the manufacturing cost. Therefore, the sun gears and planet gears were only modified. Since the proposed model is required to perform the special task of swing rotation, the same modifications to both the left and right flanks of each gear were implemented. Table 2 shows the optimal modification value of gear flank form obtained after a substantial number of calculations and comparisons.

The TE was referred to is usually static, based on a single pair of gears. However, when considering the torque sharing of the planets, this will cycle with each tooth pass. The stiffness of the planetary carrier also varies as the carrier rotates, and this variation is significant in a non-symmetric carrier model. Therefore, in this research, a novel analysis of gearbox overall transmission errors caused by all of the loaded gear meshes was carried out. The variation in load sharing among the planets over time was predicted to calculate the resulting excitation forces on the sun, planet and ring gears as output.

Fig. 8 shows a comparison curve obtained by the integration of the PPTE values from before and after the optimal modification calculated by static TE results. In addition, the PPTE values calculated by the gearbox overall transmission error results are also integrated into the chart. It is clearly visible that the PPTE of each gear has significantly improved. Especially for the second stage planetary gear train, the PPTE between ring gear and planet gear mesh has reduced by more than 50 %. The maximum PPTE value of the sun number 2 and planet number 2 gear mesh was reduced from 5.61 μm to 3.78 μm. Compared to the value of static PPTE after optimization, the gearbox overall PPTE value has not changed much, which means that the load sharing of our design is very balanced, this is also the feature of the non-factorizing design has been mentioned earlier. Fig. 9 shows the after modification load distribution on tooth surface of the second stage sun gear and planet gear. Here, it is visible that the maximum contact stress has moved to the center of the tooth surface, which is the best state for gear transmission.

4. Calculation of efficiency analysis

Transfer efficiency is an important indicator of gearbox performance. Efficiency (η) is the ratio between the useful output of an energy conversion machine and the input, which can be accurately measured by experimentation. In this research, a calculation by the numerical method was carried out based on the ISO/TR 14719 standard to validate the efficiency of the proposed model. Many different power loss models are provided in this standard, such as bearing loss model, mesh power loss model etc. In this paper, the bearing loss model, seal drag model, gear mesh drag model and micro geometry based sliding loss model were only considered. Following are the main calculation formulae for gearbox efficiency. These calculations were implemented using RomaxDESIGNER. Influence of load distribution calculated by micro geometry analysis is reflected in the sliding loss calculation of the efficiency analysis.

Overall unit efficiency:

$$\eta = 100 - \frac{P_L + P_N}{P_A} \times 100 \tag{2}$$

Load dependent losses:

$$P_L = \sum P_B + \sum P_M \tag{3}$$

Non-load dependent losses:

$$P_N = \sum P_S + \sum P_W + \sum P_{WB} + \sum P_P \tag{4}$$

Bearing loss model:

$$P_B = \left(T_{VL0} + T_{VLP1} + T_{VLP2}\right) \times Rotation\,Speed \tag{5}$$

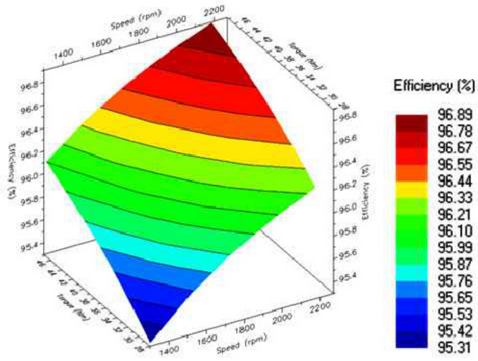
Seal drag model:

$$P_{Si} = \frac{T_S n}{9549} \tag{6}$$

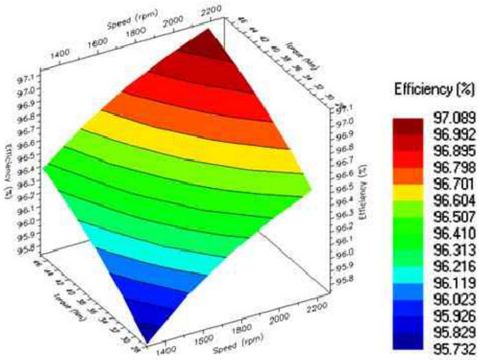
Gear mesh drag model:

$$P_{Mi} = \frac{f_m T_1 n_1 cos^2 \beta_w}{9549 M} \tag{7}$$

Micro geometry based sliding loss calculation ISO 14179-1 (US):



(a) Before micro geometry modification



(b) After micro geometry modification

Fig. 10. Efficiency results of the developed gearbox.

$$\begin{aligned} Power\,loss = \int \big(& normal\,load\,distribution \\ & *sliding\,velocity\,distribution \big) \\ & *\ friction\,coefficient \end{aligned} \tag{8}$$

The efficiency result of the whole gearbox powertrain system obtained after a series of settings and operations is shown in Fig. 10(b). For the sake of comparison, these calculations on the model before micro geometry modification were also carried out, the results are given in Fig. 10(a). The overall transmission efficiency remains at a very high level above 95 % both before and after the micro geometry modification. However, the efficiency of the modified gearbox is 0.3 % higher than that before the micro geometry modification.

5. Evaluation of dynamic transmission error analysis

In this section, the dynamic characteristics of the research object through gear whine dynamic analysis were simulated and observed. Modal flexibility reflects the comprehensive information of the mode shape and frequency, and has a high sensitivity to small changes in structural parameters. Simulation of modal flexibility is widely used in the vibration reliability inspection of bridges, automobiles, aircrafts etc. [3]. Dynamic transmission error is a frequency dependent value that is the actual dynamic displacement between the two gear pair mesh nodes. This is different from the static transmission

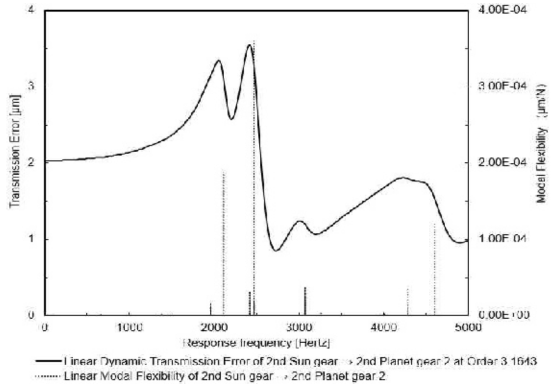


Fig. 11. Dynamic transmission error.

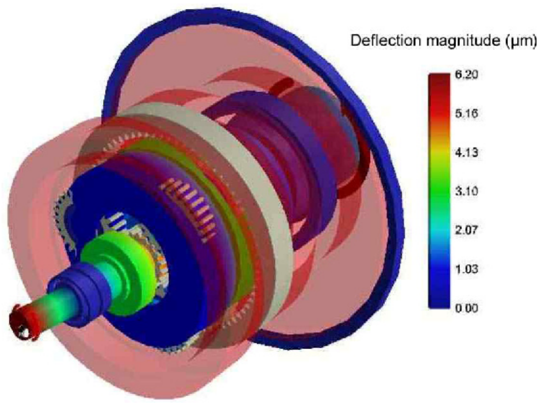


Fig. 12. ODS at frequency of 2473.1 Hz.

error which is the displacement used to excite the model at the gear mesh. In the previous section, the maximum PPTE occurred in the sun gear and planet gear mesh of second stage planetary gear train can be found. The first order harmonic of this gear mesh as the excitation was used to observe the variation of dynamic transmission error and the modal flexibility was used to evaluate the proposed design.

Fig. 11 shows the results of dynamic transmission error simulation. The linear dynamic transmission error plot has peaks close to modes with high linear modal flexibility. This is because exciting the system at a natural frequency will cause the system to resonate, and if the modal flexibility is high then the mode can be excited by the static TE. Here, it can be seen that the value of the modal flexibility is much smaller than the value of the transmission error. These results show that our design has no possibility of resonance. For reference, the ODS (operating deflection shapes) at the response frequency of maximum modal flexibility was also simulated as shown in Fig. 12.

6. Experimental verification

Output performance and durability are most important indicators for industrial machinery assessment. In order to verify the durability and output performance of the designed gearbox,

Fig. 13. Performance test equipment.

(a)

(b)

Fig. 14. Disassembled view of components.

a series of bench tests were performed with the optimized prototype. The performance test equipment is shown in Fig. 13. Motors and speed-increase gearbox were utilized to ensure speed-controlled input and torque-controlled output. Speed and torque sensors were placed on both the input and output sides. Durability test uses the same equipment as the performance test. Five randomly selected prototypes passed the durability test successfully. After the durability test, no damage or crack was found on the gears, bearings, shafts or housing as shown in Fig. 14. Slight gear pitting was only observed in the planet gears of the first stage planetary gear train.

7. Conclusions

In this study, geometry method was used to find the optimal macro parameters. Micro geometry modification was then conducted by means of numerical analysis, that evaluated the proposed model using a series of verification simulations. Finally, validation bench test of the reliability of the gearbox was carried out. Through this work can draw the following conclusions:

(1) Macro geometry is a relatively mature design method.

There are many international standards for reference when designing a new planetary gearbox. The profile shift modification is an effective method to balance the specific sliding at each end of the paths of contact to minimize wear of gear teeth. As a result, the safety factor of every gear in each stage is above 1.0, which is the industrial safety requirement.

(2) Through the comparison of before and after gear modification PPTE values of every gear, the PPTE after modification had a significant improvement can be found. The maximum PPTE value (5.61 μm) before modification occurred in sun gear and planet gear number 2 of the second planetary gear train. This was reduced by 32.6 % after modification. The PPTE between ring gear number 2 and planet gear mesh was reduced by more than 50 %. The balanced load sharing design can be verified by the comparison of static and gearbox overall PPTE. The load distribution on meshing gear surfaces was also improved by the empirical modification.

(3) Through numerical analysis and calculation, the transmission efficiency of the entire two stage planetary gearbox was found to be 95 %. In future studies, the loss model should be optimized to ensure the accuracy of the calculation. By comparing the simulation results of the modal flexibility and dynamic transmission error, the possibility of gearbox resonance can be negated. In other words, the NVH characteristics of the developed system are guaranteed.

(4) In order to verify the reliability of the research object and the correctness of the simulation results, a series of bench tests were conducted. All the prototypes passed the test successfully and there was no damage found after post-test disassembly. Through the tests and simulations, the durability of the gearbox has been fully proven.

Acknowledgments

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Nomenclature

M : Mesh mechanical advantage
n : Shaft speed
n_1 : Pinion rotational speed
P_N : Non-load dependent losses
P_L : Load dependent losses
P_A : Transmitted power
P_B : Total bearing losses (all bearings)
P_M : Total gear mesh losses (all meshes)
P_{Mi} : Mesh power loss
P_S : Total oil seal losses (all seals)
P_W : Total combined windage and churning losses
P_{WB} : Oil churning losses, bearings (all bearings)
P_P : Total oil pump power required (all pumps)
P_{Si} : Power loss for each individual oil seal
T_{VL0} : Viscous friction
T_{VLP1} : Load dependent drag
T_{VLP2} : Cylindrical roller rib friction
T_S : Oil seal torque
T_1 : Pinion torque
Z_r : Teeth number of ring gear
Z_s : Teeth number of sun gear
β_w : Operating helix angle
η : Efficiency

References

[1] X. Chen, C. Song, C. Zhu, J. Tan and N. Ullah, Effects of macro-parameters on vibration and radiation noise for high speed wheel gear transmission in electric vehicles, *Journal of Mechanical Science and Technology*, 32 (9) (2018) 4153-4164.

[2] Y. J. Park, J. G. Kim, G. H. Lee and S. B. Shim, Load sharing and distributed on the gear flank of wind turbine planetary gearbox, *Journal of Mechanical Science and Technology*, 29 (1) (2015) 309-316.

[3] Z. Qin, H. I. Son and S. K. Lyu, Design of anti-vibration mounting for 140a class alternator for vehicles, *Journal of Mechanical Science and Technology*, 32 (11) (2018) 5233-5239.

[4] S. Azadi, M. Azadi and F. Zahedi, NVH analysis and improvement of a vehicle body structure using DOE method, *Journal of Mechanical Science and Technology*, 23 (11) (2010) 2980-2989.

[5] D. Han, N. Zhao and P. Shi, Gear fault feature extraction and diagnosis method under different load excitation based on EMD, PSO-SVM and fractal box dimension, *Journal of Mechanical Science and Technology*, 33 (2) (2019) 487-494.

[6] J. Son, D. Kang, D. Boo and K. Ko, An experimental study on the fault diagnosis of wind turbines through a condition monitoring system, *Journal of Mechanical Science and Technology*, 32 (12) (2018) 5573-5582.

[7] G. Li, X. Liang and F. Li, Model-based analysis and fault diagnosis of a compound planetary gear set with damaged sun gear, *Journal of Mechanical Science and Technology*, 32 (7) (2018) 3081-3096.

[8] L. Xu, C. Zhu, H. Liu, G. Chen and W. Long, Dynamic characteristics and experimental study on a wind turbine gearbox, *Journal of Mechanical Science and Technology*, 33 (1) (2019) 393-402.

[9] H. He, H. Liu, C. Zhu and L. Yuan, Shakedown analysis of a wind turbine gear considering strain-hardening and the initial residual stress, *Journal of Mechanical Science and Technology*, 32 (11) (2018) 5241-5250.

[10] Z. Qin, Y. T. Wu and S. K. Lyu, A review of recent advances in design optimization of gearbox, *Int. J. Precis. Eng. Manuf.*, 19 (11) (2018) 1753-1762.

[11] W. Shi, Y. Park, H. Park and D. Ning, Dynamic analysis of the wind turbine drivetrain considering shaft bending effect, *Journal of Mechanical Science and Technology*, 32 (7)

(2018) 3065-3072.
[12] F. Wang, Y. Lu, H. P. Lee and X. Ma, Vibration and noise attenuation performance of compounded periodic struts for helicopter gearbox system, *Journal of Sound and Vibration* (2019).
[13] M. Zhang, K. S. Wang, D. D. Wei and M. J. Zuo, Amplitudes of characteristic frequencies for fault diagnosis of planetary gearbox, *Journal of Sound and Vibration*, 432 (2018) 119-132.
[14] Q. Wang, Z. Li, H. Ma and B. Wen, Effects of different coupling models of a helical gear system on vibration characteristics, *Journal of Mechanical Science and Technology*, 31 (5) (2017) 2143-2154.
[15] D. H. Lee, K. H. Moon and W. Y. Lee, Characteristics of transmission error and vibration of broken tooth contact, *Journal of Mechanical Science and Technology*, 30 (12) (2016) 5547-5553.
[16] Romax Tech. Ltd., *Romax Designer Software Manual* (2003).

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