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헬리컬 기어의 치형최적화에 관한 연구

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A Study on Optimization of Tooth Micro-geometry for a Helical Gear Pair

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ABSTRACT

Nowadays, modern gearboxes are characterized by high torque load demands, low running noise and compact design. Also durability of gearbox is specially a major issue for the industry. For the gearbox which used in wind turbine, gear transmission error(T.E.) is the excitation that leads the tonal noise known as gear whine, and radiated gear whine is also the dominant source of noise in the whole gearbox. In this paper, tooth modification for the high speed stage is used to compensate for the deformation of the teeth due to load and to ensure a proper meshing to achieve an optimized tooth contact pattern. The gearbox is firstly modeled in Romax software, and then the various combination analysis of the tooth modification is presented by using Windows LDP software, and the prediction of transmission error under the loaded torque for the helical gear pair is investigated, the transmission error, contact stress, root stress and load distribution are also calculated and compared before and after tooth modification under one torque condition. The simulation result shows that the transmission error and stress under the loads can be minimized by the appropriate tooth modification.

Key Words : Helical Gear(헬리컬기어), Transmission Error(T.E.)(트랜스 미션 에러), Tooth Modification(치형수정), Contact Stress(접촉응력)

1. Introduction

Wind power is alternative energy source and clean energy in many countries, it become a major part of their plans for sustainable development. According to the global wind energy report, the wind industry has been expanding at an annual growth rate of 28% over the past ten years[1]. And the wind generating system wind turbine, which emits no carbon dioxide, has been widely accepted as the clean and environment friendly machine. Since gearboxes are one of the most expensive components of the wind turbine system, the failure rates are very important to the cost of wind energy. In order to help bring the cost of wind energy back to a decreasing

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trajectory, the technical trend for wind turbines is to increase the gearbox reliability and efficiency instead of reducing the large cost of operation[2,3]. Fig. 1 is the schematic of the helical gear pair under the investigation.

2. Gear Noise and Durability

2.1 Gear Transmission Error

Gear transmission error is expressed as a linear deviation measured at the pitch point and calculated at successive positions of the pinion as it goes through the meshing cycle. It is the result of manufacturing geometry errors, gear tooth, shaft and housing deflections, mesh stiffness variability and gear dynamics. It is calculated using the micro-geometry of the gears and the misalignments in each loaded condition. The accurate calculation of misalignment is important, including all the effects such as shaft stiffness, non-linear bearing stiffness and housing stiffness.

When a pair of perfect gears is meshed under zero loads, theoretically, the involute geometry of drive gear dictates that the driven gear follows exactly the rotation of the driving gear in proportion to the gear ratio. However, the gear tooth geometry contains errors, it results in the driven gear being momentarily ahead or behind its theoretical position. The gear transmission error is classified as positive or negative T.E., for example, a burr on the surface of the tooth profile would cause a positive transmission error, but the elastic deflections of the teeth under the load would cause a negative transmission error.

From measured angle of rotation, the T.E. of a pair of gears can be expressed in terms of a linear discrepancy tangential to pitch circle.

$$TE = \theta_2 r_{b_2} - \theta_1 r_{b_1} \qquad (1)$$

Or an angular discrepancy in position

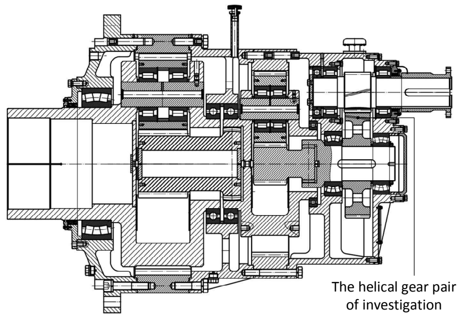


Fig. 1 The schematic of two planetary, one helical wheel stage

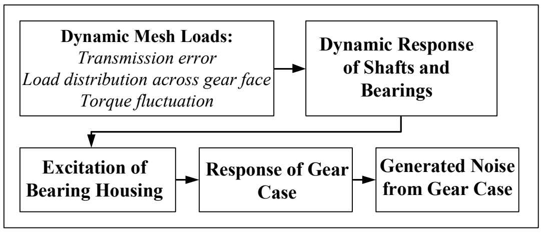


Fig. 2 The schematic of gear noise transmission path

$$TE = \frac{\theta_2 z_2}{z_1} - \theta_1 \qquad (2)$$

where θ is the angle of gear rotation, r_b is the base radius and z is the number of gear teeth. Subscripts 1 and 2 respectively denote the pinion and wheel. Fig. 2 is the schematic of gear noise transmission path.

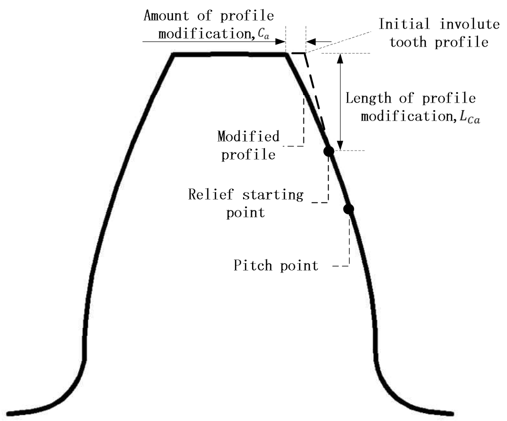
2.2 Gear Tooth Modification

Two main methods currently implemented, in order to reduce the gear noise and vibration responses of the system is by means of macro-geometry and micro-geometry modifications.

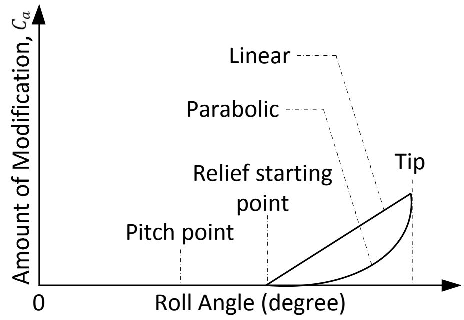
Macro-geometry is defined by gear parameters, such as number of teeth, diameters, pressure angle, backlash and clearance. Macro-geometric modifications involve an important and expensive change of the gear pair as well as other parameters of the gear pair; they are feasible only at the first step of the design process. High quality surface finishing and strict tolerance can lead to excessive manufacturing costs. Moreover, their effect on vibration can be disappointingly small.

Micro-modifications include the intentional removal of material from the gear teeth flanks, so that the shape is no longer a perfect involute. Such modifications compensate teeth deflections under load, and the resulting transmission error is minimized under a specific torque. Therefore in this paper, the micro-geometry modifications will be the focus of the analysis. Micro-geometry modifications can be applied on the involute(or profile) and lead of the gear teeth. In fact, there is a third type of modification, bias, but it is not discussed in this paper.

For involute modifications, modification in the involute direction takes into account of tooth elastic deformation and errors due to casting, heat treatment and assembly. Involute modifications are done for following parameters, involute barreling, root relief start, tip relief end and involute slope. And about lead modifications, lead modifications involve applying lead slope correction and lead crowning to the gears. They are done for following lead parameters, lead crown, lead relief start, lead relief end and lead slope. And Fig 3 is the definition of the profile modifications. Fig.4 is the schematic of typical gear modification[4].



(a) Gear tooth with modified tooth profile



(b) Linear and parabolic tooth modified profile

Fig. 3 Definition of the profile modifications

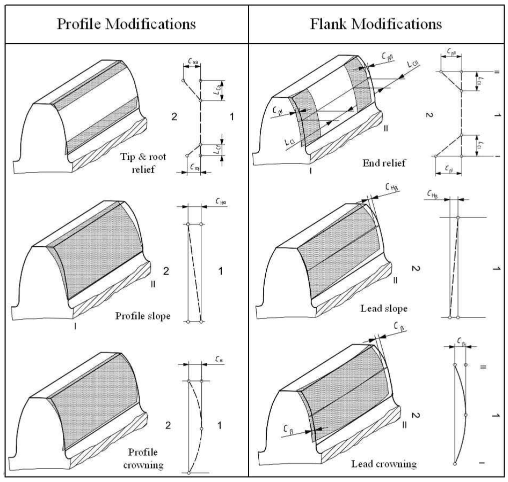


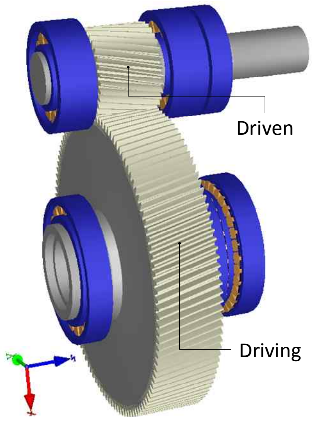
Fig. 4 The typical gear micro-modifications

3. Analysis and Simulation

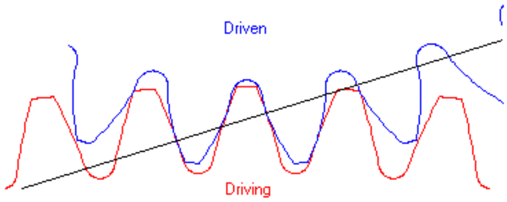
In recent years, there have been considerable developments in computer aided design(CAD) and computer aided engineering(CAE), as a result, engineering can now easily undertake a wide range of design, modeling and analysis in respective research areas. For the Romax Designer software which has been applied firstly to the wind turbine, a wind turbine gearbox of 2.0MW can be modeled to reduce the development risks of the full gearbox system[5]. In this paper, the Windows LDP software is used to analyze the gear pair in the transmission system[6,7]. From the rotor blades to an output shaft for the generator, the speed and angular acceleration create a varying and difficult set of dynamic condition for the output shaft. The model of the helical wheel stage's output shafts which equipped with helical gears producing radial and axial loads is investigated. And the schematic of the helical wheel stage is shown in Fig. 5. And Table 1 is the specification of the helical gear pair.

Gear mesh transmission error is one of the major causes of noise of drive train system. Apparently, one way to reduce the drive line noise is tackle the noise excitation mechanism at its source. And this excitation can be reduced by optimize the gear teeth micro-geometry to minimize the Transmission Error. Therefore, the transmission error can be analyzed and predicted at the stage of transmission design. After the simulation under the torque, the micro-geometry of gearbox third stage is simulated. And Fig. 6 is transmission error, driving gear root stress, contact stress distribution and load distribution without gear modification at the torque 51691Nm. The peak-to-peak transmission error(PPTE) is about 1um, the peak root stress of driving gear is about 110 MPa, the maximum contact stress is about 1476MPa and the maximum load is about 1756N.

The profile parabolic and lead crown modification are investigated that would minimize the peak-to-peak transmission error. For driving gear, the roll angle of EAP is 21degree and the amount of tip parabolic is 45um, lead crown modification is 25um. And for driven gear, the roll angle of EAP is 30degree, and the amount of tip parabolic is 45um, the lead modification is the same as the driving gear. And the simulation result after the gear tooth modification is shown in Fig. 7. By comparing Fig. 6 and Fig. 7, it is observed that the PPTE transmission error is reduced from 1um to 0.3um, and when face width position is 0 or 100%, the driving gear root stress is decreased strictly. The contact stress distribution along



(a) Helical gear mesh in 3D dimension



(b) Helical gear mesh in 2D dimension

Fig. 5 The schematic of helical gears in mesh

Table 1 Helical gear pair specification

	Driving	Driven
Number of teeth	115	31
Module(mm)	8	
Pressure angle(deg.)	17.5	
Helix angle(deg.)	13	
Addendum mod. coeff.	-0.1804	0.2605
Center distance(mm)	600	
Face width(mm)	200	
Outside diameter(mm)	962.594	279.978
Root diameter(mm)	916.022	233.4
Standard pitch diameter(mm)	944.2	254.523
Transverse tooth thickness at SPD(mm)	11.947	14.246
Profile / face contact ratio	2.305	1.79
Total contact ratio	4.095	

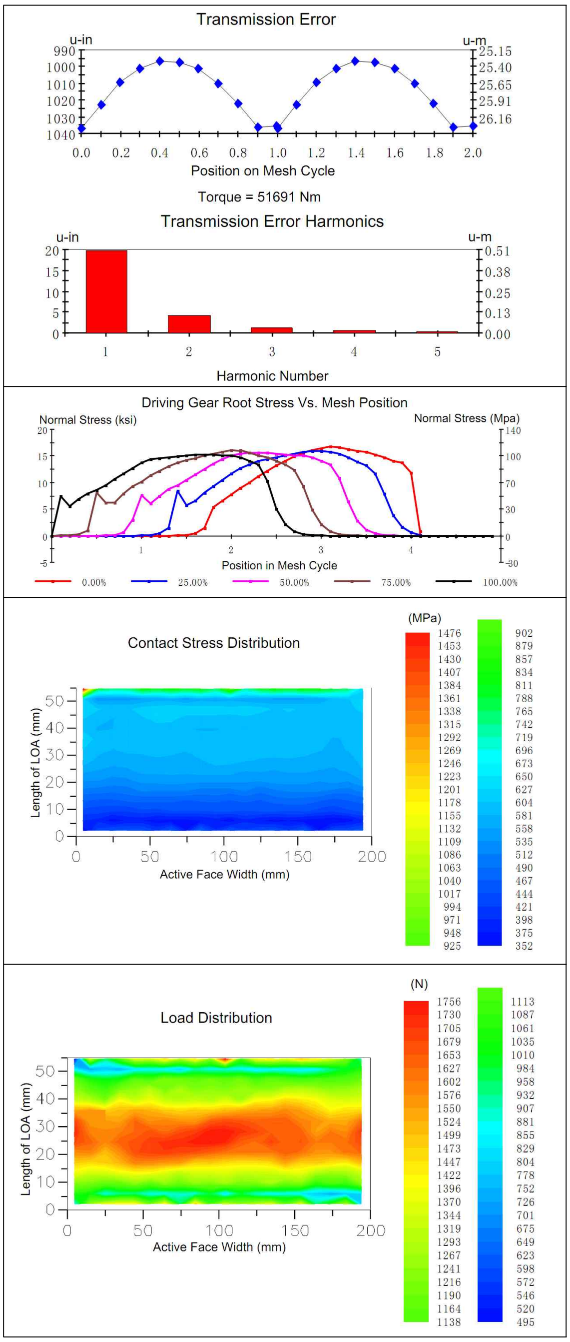


Fig. 6 Transmission error, driving gear root stress, contact stress distribution and load distribution without gear tooth modification at the torque 51691Nm

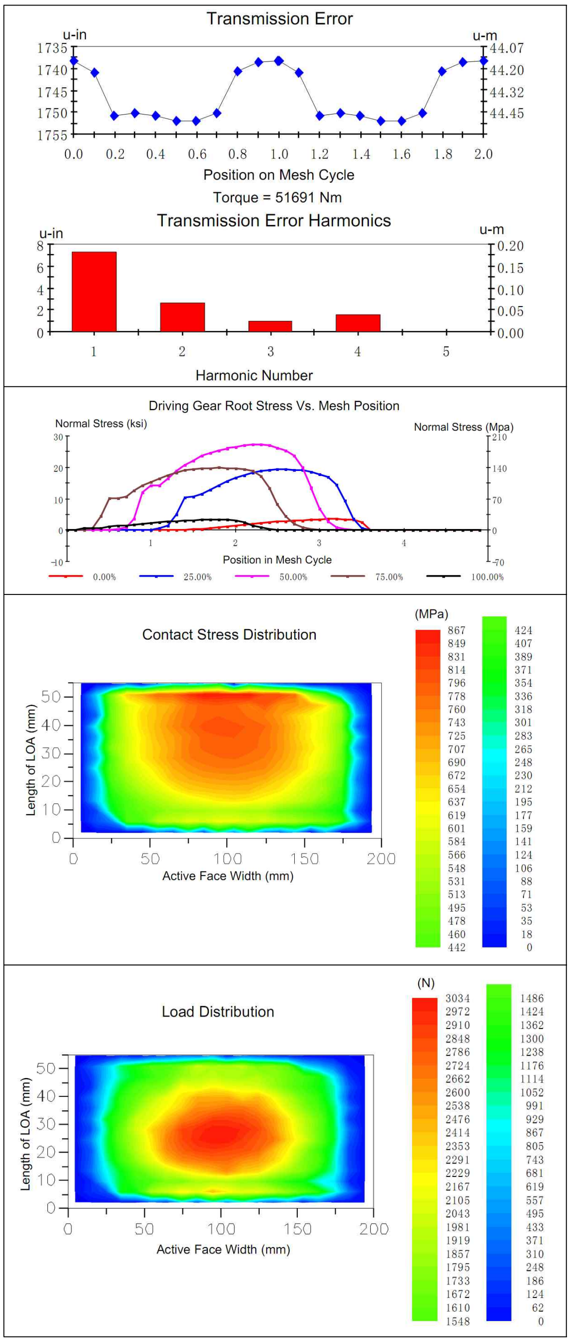


Fig. 7 Transmission error, driving gear root stress, contact stress distribution and load distribution after gear tooth modification at the torque 51691Nm

the face is improved, which is minimized from 1476MPa to 867MPa, and the load distribution is shifted to middle region of the tooth flank which is up to 3034N.

Therefore, involute and lead micro-geometry modifications are often used to adjust the gear mesh misalignment. Thus, contact stress and bending stress can be minimized, leading to noise reduction and improved gear performance.

4. Conclusion

In this paper, the helical gear pair of the third stage is modeled and analyzed. By comparing the simulation results, the optimal tip parabolic and lead crown tooth micro-modification can obtain a good result in gear PPTE transmission error, root stress, contact stress distribution and load distribution. Therefore, it is a good tool for the designer to get the optimal gear tooth micro-geometry before it is available for the test rig.

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