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

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The wind industry grew in the first decade of the 21st century at rates consistently above 20% a year. For wind turbine, gearbox failure can be extremely costly in terms of repair costs, replacement parts, and in lost power production due to downtime. For the gearbox 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, gear tooth micro-modification for the high speed stage was 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 was firstly modeled in Romaxdesigner software, and then the various combined tooth modification were presented, and the prediction of transmission under the loaded torque for the helical gear pair was investigated, the transmission error, transmission harmonics, normal load distribution were also obtained and compared before and after tooth modification under one torque. The simulation results showed that the transmission error and normal load distribution under the load can be minimized by the appropriate tooth modification. It is a good approach where the simulated result is used to improve the design before the prototype is available for the test.

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1. Introduction

Wind power is suitable and clean energy, and in many countries, wind power has 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 environmentally 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 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] In this paper, the wind turbine gearbox is modeled in Romaxdesigner software, and the various combined tooth modifications for a helical gear pair are used to investigate the transmission error, transmission harmonics, normal load distribution under the loaded torque which the helical gear mesh misalignment is considered. The simulation results show that the transmission error and normal load distribution under the load can be minimized by the appropriate tooth modification after tooth modification.

2. T.E. Theory and Micro-modification

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

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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.

$$TE = \theta_2 r_{b_2} - \theta_1 r_{b_1} \tag{1}$$

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

2.2 Thin Slice Model

There are various methods of calculating transmission error in spur and helical gears, or combination of them is used. In this paper, the "Thin Slice Method are expressed, this assumes that the gear (either helical or spur) is divided into a number of slices, each acting as a spur gear, parallel to and independent from its neighboring slices. The transmission error can then be calculated for each of these spur gear slices by approximating them as a spring of known stiffness and the sum of the loads equated to the total applied load (Fig. 2 & Fig. 3).[4]

2.3 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. 4 is the definition of profile and lead micro-modifications.[5,6]

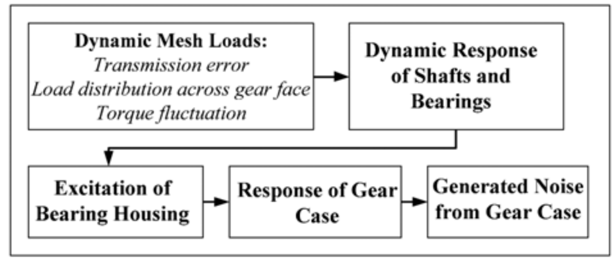


Fig. 1 The schematic of gear noise transmission path

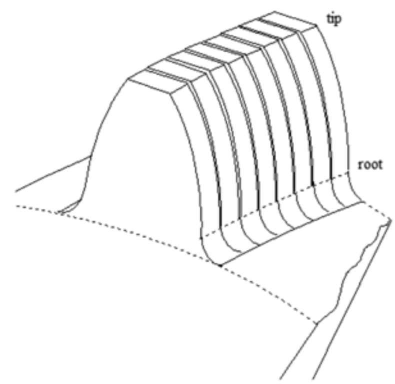


Fig. 2 The thin slice model

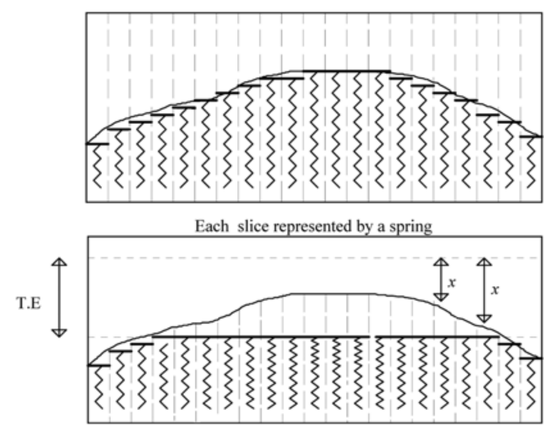


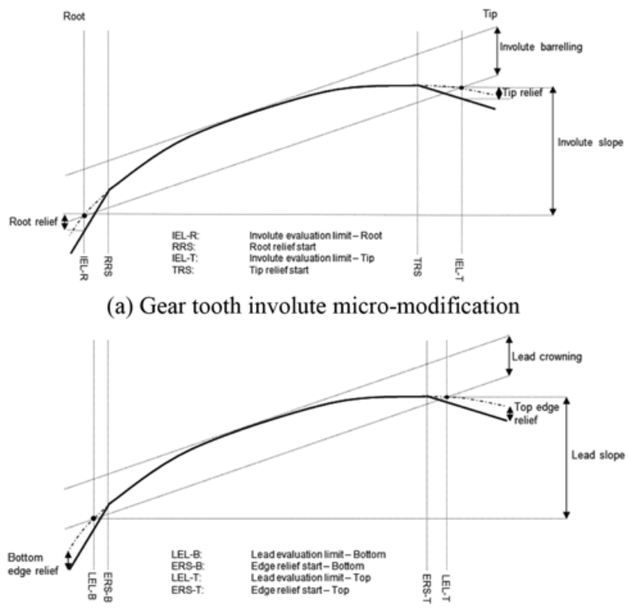
Fig. 3 Representation of a gear as a series of springs, and the deflection caused by TE

2.4 Static Analysis

Before conducting a static analysis, the boundary conditions of the analysis are defined. This entails specifying the input speed and torque (or power) and which clutches or brakes are locked. The convergence scheme is shown in Fig. 5. The non-linear analysis can be completed in a couple minutes. The full six degree of freedom deflection of the system, the rotational speeds and forces acting on all components, gear and bearing misalignment are obtained.

3. Simulation and Discussion

CAE software RomaxDesigner has been applied to a wind turbine gearbox, the gearbox of 2.0 MW can be modeled to reduce the development risks of the full gearbox system. In this paper, the software is only used to analyze the helical gear pair of the third stage



(a) Gear tooth involute micro-modification

(b) Gear tooth lead micro-modification

Fig. 4 Definition of profile and lead micro-modifications

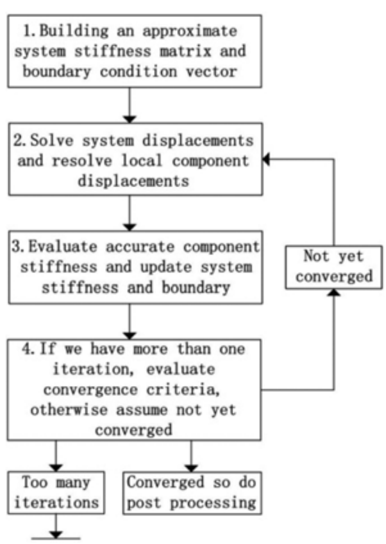
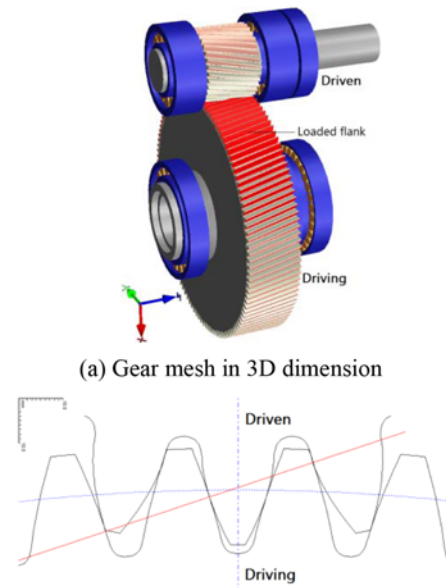


Fig. 5 The static analysis convergence scheme

in the transmission system. From the rotor blades to the 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 has been investigated. And the schematic of the helical wheel stage is shown in Fig. 6, the loaded flank is right flank of gear pair.

And Table 1 is the specification of the helical gear pair. Gear mesh misalignment is considered in this paper (Fig. 7). And the combined defection between two gears is 53.02 um.

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 optimizing the gear tooth micro-geometry to minimize the transmission error which can be analyzed and predicted at the stage of design by the software. After the simulation



(a) Gear mesh in 3D dimension

(b) Gear mesh in 2D dimension

Fig. 6 The schematic of helical gear pair 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
Reference pitch diameter (mm)	944.2	254.523
Transverse tooth thickness(mm)	11.947	14.246
Transverse / axial contact ratio	2.305	1.79
Total contact ratio	4.095	

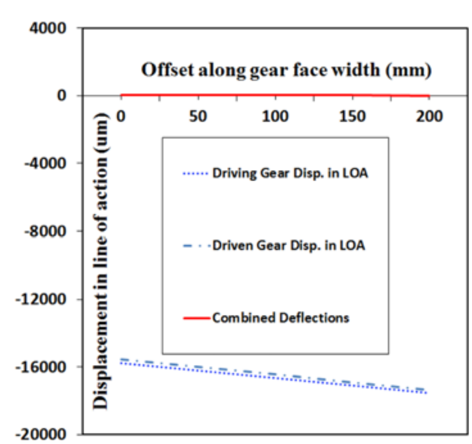
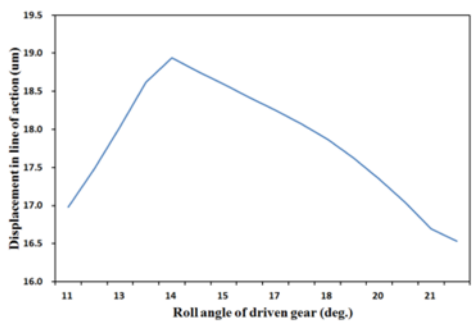


Fig. 7 Mesh misalignment for the helical gear pair

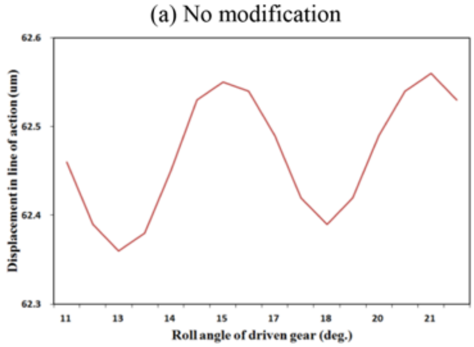
under the specified torque, the micro-geometry of gearbox third stage is simulated. And Table 2 is the proposal of helical gear pair micro-modification. For P1, no modification for two gears, P2, driven gear lead crown is 27.6 um, driven gear lead slope is -53 um in software because of tooth number of driven gear, for P3, linear tip modification

Table 2 Helical gear pair micro-modification

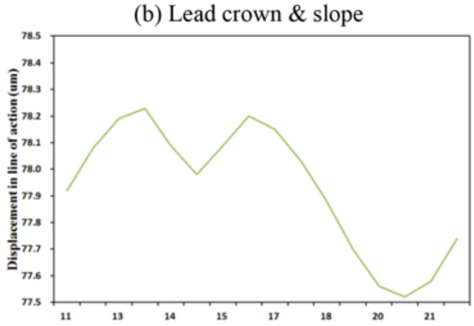
P1	No modification		
P2	Lead crown	27.6 um	
	Lead slope	-53 um	
P3	Driving tip linear	48 um	19.3 deg.
	Driven tip linear	48 um	22.9 deg.
	Lead crown	27.6 um	
	Lead slope	-53 um	
P4	Driven tip linear	48 um	22.9 deg.
	Driven root linear	48 um	16.8 deg.
	Lead crown	27.6 um	
	Lead slope	-53 um	



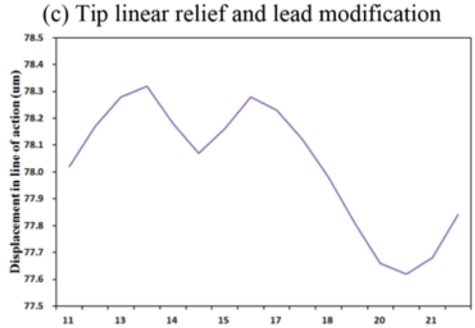
(a) No modification



(b) Lead crown & slope



(c) Tip linear relief and lead modification



(d) Tip&root linear relief and lead modification

Fig. 8 Comparison of transmission error at the torque 51691 Nm

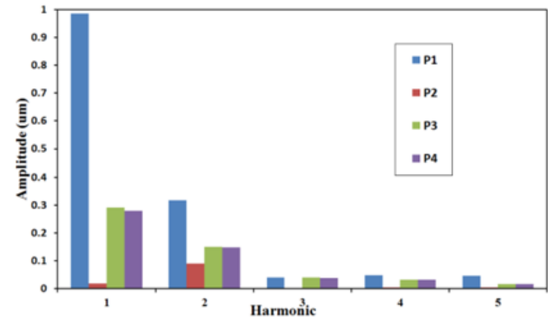


Fig. 9 Comparison of transmission error harmonics at the torque 51691 Nm

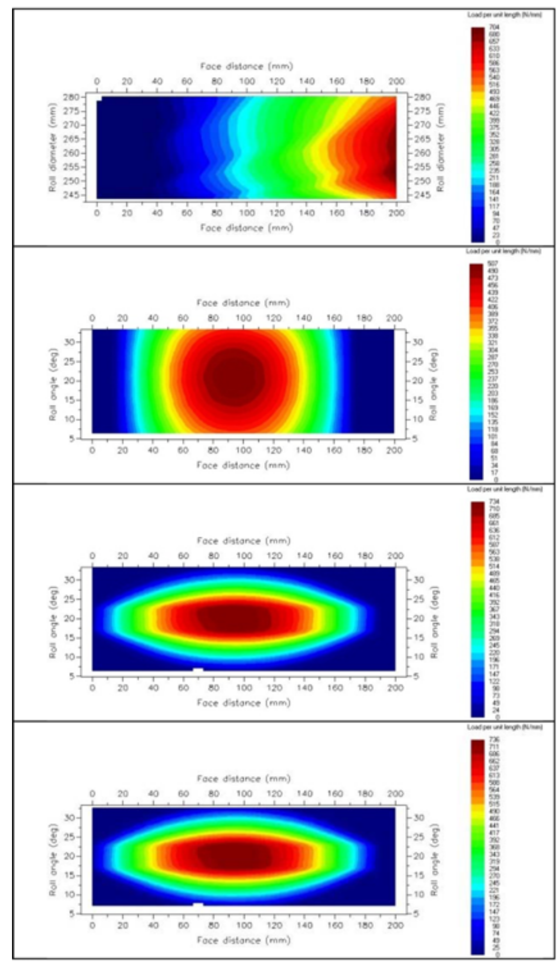


Fig. 10 Comparison of driven gear normal load distribution at the torque 51691 Nm

for two gears are investigated, but P4, only driven gear tip and root modification.

Fig. 8 is the comparison of transmission error with four kinds of modifications at the torque 51691 Nm. The peak-to-peak transmission error (PPTE) of P1 is about 2.4 um, P2 is 0.2 um, P3 and P4 are similar, 0.7 um including the similar waveform of TE. By comparing these four modifications, P2 is a better choice. Fig. 9 is the comparison of transmission error harmonics at the torque 51691 Nm, by comparing Fig. 8 and Fig. 9, the harmonics of P1 are higher, but the waveform of P2 is better which means there will be smaller harmonics in P2, P3 and P4 with the similar modification also have the similar harmonics.

Fig. 10 is the comparison of driven gear normal load distribution at the torque 51691 Nm. Without modification, the normal load is misaligned, and the peak value is 704 N/mm, with driven gear lead micro-modification, the normal load is almost distributed in the middle region of the tooth flank face width, and the peak value is 507 N/mm. With driving gear (though the number of teeth is numerous) and driven gear tip linear modification and lead modification, the normal load is also shifted to the middle region of tooth flank, and the peak value is 734 N/mm, but with driven gear tip and root linear modification and lead modification, the simulation result is similar an d the peak value is 736 N/mm. Therefore, involute and lead micro-geometry modifications are often used to adjust the gear mesh misalignment. Thus, transmission error and harmonics can be minimized, leading to noise reduction and improved gear performance.

4. Conclusions

This paper has simply shown how a software-based modeling can be used to predict gear misalignment. Using this approach, an engineer can investigate the effects of transmission error about the whole transmission system. In this paper, the helical gear pair of the third stage is modeled and analyzed by Romaxdesigner. By comparing the simulation results, the profile and lead tooth micro-modification are compared to obtain a good result in gear PPTE transmission error, harmonic and normal load distribution. Therefore, it is a good example to show on how to get the optimal gear tooth micro-geometry before it is available for the test rig.

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