Simulation analysis of the tooth modification about wind power gearbox based on Romax software

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Abstract. At present, the gearboxes are faced with the problems of high failure rate, large vibration and noise, short service life etc. Gear modification is an effective means to reduce vibration and noise, extend life, which has been proved by practical experience at home and abroad for many years. With the introduction of some professional gear software or wind turbine analysis software from abroad, the work efficiency has been improved greatly and the extra cost due to experimental verification has also been reduced. With the use of Romax WIND excellent load distribution analysis function on the basis of the theory of gear modification, the static simulation analysis of the gearbox through the Romax WIND software was carried out under the equivalent loads. According to the load distribution on the tooth surface of gear, transmission errors and other parameters, the length, amount and curve of axial modification and the tooth profile of the high-speed pinion gear were analyzed and determined in the end. Conclusions can provide reference for gear modification.

Keywords. Gearbox; Gear Modification; Romax WIND; Modification Parameters

1. Introduction

The Twelfth Five-year Plan has given unprecedented support to the wind energy industry [1]. According to introduction of related documents, nearly over half of the high-power wind power equipments installed in China all comes from abroad [2]. Gear box is one of the key components of a wind generator and its power rating ranges from several hundred kilowatts to megawatt [3,4]. Therefore it is necessary to continue working on how to improve the performance of gear box and extend its life.

As a professional gear design software, Romax WIND is recognized as the most authoritative load distribution calculation software in the world. The 2.5MW gear box studied in this paper is composed of a first-class planetary gear and two parallel levels. We can use the excellent micro-analysis capability of Romax WIND to obtain the gear modification method suitable for a 2.5MW wind power gear box and its specific length, quantity and curve, thus further improving gear meshing performance,
avoiding undesirable phenomena such as deviation load, enhancing the bearing capacity of gear and prolonging the life of the gear box [5].

2. Gear modification theory

2.1. Tooth profile modification
In the meshing process of single and double teeth of gear, the load distribution of gear teeth has obvious mutation, its elastic deformation changes accordingly. Therefore, the standard involute gear has meshing interference when meshing in or out in actual operation [6]. The tooth profile modification refers to the appropriate cutting off part of the meshing gear teeth of the gear pair in the interfering tooth flank, that is to say, remove part of the teeth near the crest or root so that the original involute tooth profile is no longer involute. It is also called tip or root relief.

In this paper, the equation recommended in the gear manual is adopted for the amount of the modification on the tooth profile.

\[ \Delta_{u} = 5 + 0.04W_{f} \]
\[ \Delta_{a} = 0.04W_{f} \]

Where
\( W_{f} \) — Load per face width, \( W_{f} = F_{f} / b (N) \)
\( F_{f} \) — The circumferential force of gear (N);
\( b \) — the effective width of gear (mm)

According to the method recommended by Tian Junfu [8], the long length of the modification is \( l_{1} = p_{b} (e_{i} - 1) \) and the short one is \( l_{1} = 1/2 p_{b} (e_{i} - 1) \) (unit: mm). The commonly used curve of the modification is a straight line and a parabola.

2.2. Axial modification
The gear modification aims to get a balanced transfer of force, which can make the tooth load distribution tend to be uniform and at the same time there is no stress concentration on the tooth flank. Accordingly, a smooth force transfer is also required at both ends of the teeth.

It is dangerous to extend the contact line of the tooth flank all the way to the end of the gear teeth because it will cause excessive stress in the corners of the teeth, thereby causing spalling or fatigue damage to the tooth flank. So a comprehensive approach [7], end relief + crowning + helix angle modification, was adopted for high-load gear in figure 1.

![Figure 1. Helix angle modification with a drum type.](image)

The quantity of crowning is:

\[ C_{s} = 0.5F_{pe} + \frac{F_{sa}}{C_{b}} \]

The quantity of helix angle modification is:
The end relief can also be modified by straight line or curve. The length of modification recommended in the resources [9] is: \( l = 0.1b \)

The calculation equation recommended in the literature [10] is:

\[
\Delta = 0.5(f_{ma} + 1.5f_{m\beta}) + (5:10)
\]  

In the equation:
- \( f_{ma} \)—the component of meshing longitudinal form error due to the comprehensive deformation;
- \( f_{m\beta} \)—helix bias amount of deflection (not including helix shape deviation, um).

Note: For high precision gears, the amount of modification is usually 60% to 70%.

3. Simulation analysis based on Romax software before the tooth modification

The load distribution on tooth surface in figure 2 and figure 3 and the transmission error variation trend chart in figure 4 are obtained after the gear box is simulated under equivalent load and modelled according to table 1.

| Table 1. Basic parameters of 2.5MW gear box. |
|---------------------------------------------|
| Element          | High speed | Medium speed | Star rating |
| Number of teeth  | 38         | 106          | 24          | 95          | 19           | 47           | 113          |
| Modulus          | 6          | 10           | 16.5        |
| Pressure angle   | 20         | 25           | 27          |
| Helix angle      | 21         | 16           | 3           |
| Center distance  | 465        | 621          | 550         |

Figure 2. Load distribution on gear surface of high speed pinion.
From the load distribution diagrams of gear surface of high-speed large gear and pinion, it can be seen that the high-speed gear has extremely serious misalignment before modification and the lower speed shaft of a high speed gear pair can hardly bear the load, while the higher speed shaft bears too much. That is easy to cause early agglutination on one end of the gear or other faults, thus influencing the gear box performance. And it can be determined from the transmission error curve of gear pair which fluctuates greatly that the serious meshing contact exists.

4. Determination of gear modification parameters

In order to solve the problems in the above simulation analysis, we need to design a set of suitable modification parameters.

Load per unit tooth width of high speed gear pair $W_f=535.026$ N/mm, therefore it can be obtained that $\Delta_{\gamma}=20.382$ μm and $\Delta_{\alpha}=25.382$ μm according to the equation. That is, the modification amount of tooth tip and tooth root is 25 μm and 20 μm respectively.

On the premise of the above determined profile modification, Romax WIND is used to simulate straight line and parabola modification, long modification and short modification and the diagrams on the load distribution and transmission error curve of tooth surface are compared and analyzed. Finally, a parabola is adopted to modify the tooth tip and root of the high speed pinion.
From equation (2), the quantity of crowning can be obtained exactly as \( C_a = 17.02 \, \mu m \) and the decimal is rounded to an integer, that is \( C_a = 17 \, \mu m \). According to equation (3), it is obtained that the quantity of helix angle modification \( C_\rho \) equals to 29 \( \mu m \).

Because the gear accuracy inside the megawatt gear box is 6 and it is relatively high, it can be calculated that the thinning of the tooth end is 12 \( \mu m \).

The above parameters were simulated by Romax WIND, and a certain parameter was adjusted slightly after the simulation results were analyzed. The modified parameters were compared and analyzed when the quantity of crowning was 17 \( \mu m \), 16 \( \mu m \) and 15 \( \mu m \) respectively. When the quantity of crowning is 16\( \mu m \), the transmission error changes smoothly, so the quantity of crowning is set as 16\( \mu m \). Other modifications remain unchanged, and the spiral angle modifications are reduced successively, as shown in table 2.

| Spiral angle modification value | Transmission error | Unit load | Tooth surface maximum stress | Reverse Flash temperature | Maximum Stress | Maximum Temperature |
|---------------------------------|--------------------|----------|------------------------------|--------------------------|-----------------|---------------------|
| 29um                            | 0.4928 897         | 910      | 0.0286                       | 74.9                     |                 |                     |
| 27um                            | 0.4837 896         | 910      | 0.0286                       | 74.9                     |                 |                     |
| 25um                            | 0.46906 895        | 909      | 0.0286                       | 74.8                     |                 |                     |
| 23um                            | 0.46071 894        | 909      | 0.0286                       | 74.7                     |                 |                     |
| 21um                            | 0.45089 894        | 909      | 0.0286                       | 74.7                     |                 |                     |

When the quantity of helix angle modification \( C_\rho \) is 21 \( \mu m \), the maximum load and the maximum stress of the tooth flank are equal to the results when that is 23 \( \mu m \), the transmission error differentials decreases, and the load distribution on the tooth flank has misalignment (as shown in figure 5).

When the helix angle modification value \( C_\rho \) is 23 \( \mu m \), the maximum load on the tooth flank and the transmission error differentials are the smallest compared with other terms, and the tooth flank load is uniformly distributetnd and symmetric in the direction of the tooth (as figure 6), and the transmission error variation trend is smooth. Therefore, it can be concluded that under the action of equivalent load, the optimal helix angle modification value is 23 \( \mu m \). At the same time, the torsional displacement obtained from the data in table 2 is independent of the helix angle modification value. The helix angle modification is conducive to reducing the maximum load on the tooth flank, the maximum Hertzain stress, the flash temperature and the transmission error. The helix angle modification value has an obvious influence on the transmission error, and the smaller the helix angle modification value is, the smaller the transmission error will be.
After simulation analysis, the modification parameters are finally determined as shown in Table 3. The load distribution of tooth surface after modification is shown in Figure 7 and Figure 8, and the variation trend of transmission error is shown in Figure 9.

**Table 3.** Modification amount of high speed pinion gear.

| Tooth shape modification amount | Root shape modification amount | Drum shape modification amount | Helix angle modification amount | Tooth end thinning modification amount |
|--------------------------------|-------------------------------|-------------------------------|--------------------------------|----------------------------------------|
| Gear                           | 25 μm                        | 2 μm                         | 16 μm                         | 23 μm                                  | 12 μm                                  |

**Figure 6.** Tooth load distribution when $C_p = 23 \, \mu m$.

**Figure 7.** Load distribution of pinion tooth surface after modification.

**Figure 8.** Load distribution on tooth surface of large gear after modification.
5. Comparative analysis of gear before and after modification

By comparing the load distribution diagram of tooth surface before and after modification, it can be seen that the problem of deflected load in gear meshing process is effectively solved after modification. As for the load distribution along the tooth width, the gear load is decreasing from the middle to both sides because of the drum modification. The maximum load on the tooth surface of the gear pair is 894 N/mm, which is 24 N/mm lower than that on the tooth surface of the pinion before modification, and 27 N/mm lower than that on the tooth surface of the larger gear. Obviously, after gear modification, the load value per unit length decreases, while in the direction of gear width, the area under load increases, thereby improving the bearing capacity of gear.

Transmission error curve of gear is an important index reflecting the dynamic capability of gear system [11]. By comparing the transmission error variation curve of the gear pair before and after modification, it can be seen that the curve of transmission error after modification eliminates the fluctuation and sharp change before modification and effectively reduces the meshing impact. And we know that the transmission error is reduced by 0.36385 μm and 44% after modification. Although the fluctuation is not eliminated in the figure, the fluctuation range of transmission error between teeth is also greatly reduced and the transmission error changes smoothly after the modification, which indicates that the modification greatly improves the dynamic performance of the gear.

6. Conclusion

- After analyzing the basic principle and calculation equation of gear profile modification and gear alignment modification, it is found that there are many methods and functions of gear alignment modification. In this paper, aiming at high-power and high-precision gear, and considering the various deformation of gear meshing, it is proposed that the axial modification should adopt the Drum-shaped spiral Angle modification with end relief and the simulation analysis by Romax WIND shows that this modification method increases the length of the tooth meshing line and improves the tooth surface meshing better.
- Combined with the actual assembly of gear box, in this study, a multibody dynamics model of a 2.5 MW gear box is established by Romax WIND software. Through the simulation analysis of gear modification, various performance parameters such as the load distribution of gear tooth surface and the transmission error variation of gear pair before and after modification can be easily obtained.
- In the case of equivalent load, Romax WIND software is used to simulate the operation of gearbox, and the modification parameters of a 2.5 MW high speed pinion of wind power gearbox are determined. The off-load and impact in gear meshing process and other problems are effectively solved by gear modification An easy way to comply with the Scopus indexed
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