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Optimal Design of Wind Turbine Gearbox using Helix Modification*

Young-Jun PARK[*1], Geun-Ho LEE[*2], Jin-Seop SONG[*2], Yong-Yun NAM[*2], Ju-Seok NAM[*3]

Abstract

Planetary gear sets, used commonly in wind turbine gearboxes, should have good load distribution on gears to improve the durability. In this paper, the influences of external loads caused by wind fluctuation on the load distribution on the gear tooth flank are investigated. A state-of-the-art whole system model is developed to analyze the planetary gear set for the WTG. The results of gear mesh misalignment, contact pattern, load distributions are predicted and the gear safety factor is also evaluated. Therefore, the results presented state that if the optimal helix modification is applied, the edge loading of gear tooth ends disappears, contact pattern improves significantly, the face load factor decreases and the gear safety factor increases.

[Keywords] gearbox, load distribution, wind turbine, gear mesh misalignment, contact pattern, helix modification

I Introduction

The wind power generation industry has recently received much attention as a source of alternative energy. The most critical issues in that industry are the performance and service life of a wind turbine system that converts wind energy into electrical energy. The wind turbine system is composed of several blades, a rotor, gearboxes, a generator, a tower, and control systems. The wind turbine gearbox (WTG) is well known as the component with the highest failure rate (Spinato, 2009). Additionally, because the WTG is installed in the narrow space of a nacelle on top of a high tower, the failure of the WTG typically results in long repair time and high repair costs.

The WTG, located between the rotor blades and the generator, converts the input power of the rotor blades and then transmits the power to the generator. The input shaft from the rotor blades has a low rotational speed of 12 - 30 rpm and high torque, while the output shaft to the generator has a high rotational speed of 1000 - 1800 rpm and low torque. Whenever the wind turbine system operates, the fluctuation in wind load applies high dynamic forces and moments to the gears, bearings, and shafts that compose the WTG. This phenomenon has an adverse influence upon the strength and durability of the WTG, perhaps causing the initial failure and reducing the service life. The WTG is required to have a service life of more than 20 years (Lee, 2009). Therefore, the most important concern for the WTG is to improve the durability to guarantee the service life, and it is keenly desired that a WTG design process for high durability be established.

In order to meet the service life of the WTG, the most critical design factors are the load distribution and load sharing among the gears. In the case of a planetary gear set (PGS), loads acting on gears should be distributed evenly over the gear tooth flank and shared equally by several planets to increase the gear life. This study investigates how external loads caused by wind fluctuation influence both the load distribution over the gear tooth flank. A state-of-the-art nonlinear system model is developed to analyze the planetary gearset for the WTG. The model includes the nonlinear mesh stiffness of gears and the nonlinear stiffness of bearings as well as the flexibilities of the housing, planet carriers, and ring gears.

II Materials and Methods

1. Modeling of wind turbine gearbox

(1) Structure of wind turbine gearbox

The WTG used in this study is a conventional gearbox for a 2 MW wind turbine system and is composed of a low-speed planetary gearset (LS PGS), high-speed planetary gearset (HS PGS), and parallel helical gearset, as shown in Fig. 1. The load condition for the two PGSs is that where the input load is on the planet carrier, the ring gear is fixed, and the output load is from the sun gear.

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*1 KSAM Member, Corresponding author, Department of System Reliability, Korea Institute of Machinery & Materials, 171 Jang-dong, Yu-Seong gu, Daejeon, 305-343, Korea; yjpark77@kimm.re.kr
*2 Department of System Reliability, Korea Institute of Machinery & Materials, 171 Jang-dong, Yu-Seong gu, Daejeon, 305-343, Korea
*3 KSAM Member, Department of Biosystems and Biomaterials Science and Engineering, Seoul National University, Daehak-dong, Gwanak-gu, Seoul, 151-742, Korea

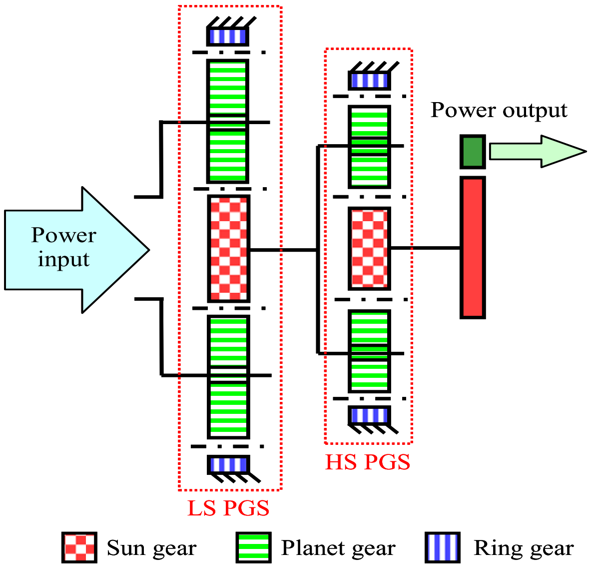

Fig. 1 Structure of 2MW WTG used in this study

(2) Physical model of wind turbine gearbox

The entire 2 MW WTG is modeled in detail. The model is allowed to carry out a quasi-static analysis of a gearbox by linking the shafts, bearings, gears, planet carriers, and housing. Because a whole system model of the WTG is made by considering non-linear components such as bearing stiffness and gear mesh stiffness, the system deflections such as gear mesh misalignment, bearing deformations, and deflections of shafts, planet carriers, and housing can be analyzed simultaneously.

The behavior of gear mesh is non-linear. The gear contact analysis includes a consideration of the calculated gear mesh misalignment, a non-linear tooth stiffness as well as the detailed micro-geometry surface definition. To analyze the gear contact, the thin slice model can be used. This assumes that the gear (either helical or spur) is divided into a number of slices across the face width, each acting as a spur gear, parallel to and independent from its neighboring slices. The gear contact can then be analyzed for each of these spur gear slices by approximating them as a spring of known stiffness and the sum of the loads equates the total applied load (Pears, 2005).

Rolling element bearings allow for the non-linear stiffness to be determined for any load condition. The stiffness matrix for each bearing is obtained by linearizing the non-linear behavior of the bearing close to the operating condition and as the slope of the force versus deflection curve at the bearing's operating displacements. The non-linear stiffness matrix, linking the displacements and tilts of the inner and outer raceway geometric centers, includes the contacts of rolling elements with raceways, non-linear effects of internal clearance and preload (Pears, 2007).

In order to include the stiffness properties and flexibility of non-uniform components such as the housing, ring gears and planet carriers, those parts are imported as meshed finite element models (FEM). These finite element models are condensed to reduced stiffness matrices and coupled with the internal gearbox components through the bearing, shaft and ring gear nodes. Shafts are also modeled as flexible beam elements (Kamaya, 2008).

The model can also predict gear life and bearing life by using system deflection results. Fig. 2 shows an entire 2 MW WTG model used in this study.

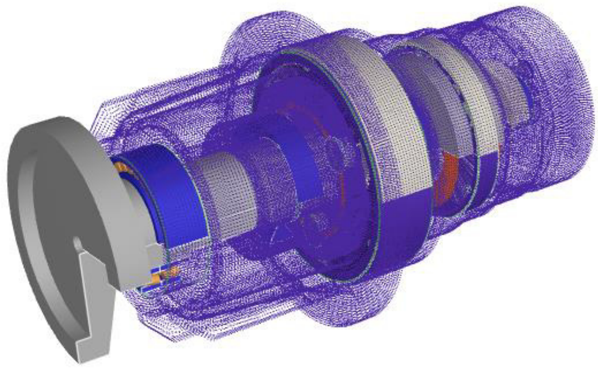

Fig. 2 Entire model of 2MW WTG used in this study

2. Face load factor for contact stress

The face load factor for contact stress, called the load distribution factor, takes into account the effect on the contact stress by the load distribution over the face width and is defined by Eq. (1) (ISO, 2007a).

$$K_{H\beta} = \frac{\text{maximum load per unit face width}}{\text{average load per unit face width}} = \frac{(F/b)_{\max}}{F_m/b} \qquad (1)$$

where F is the load on arbitrary position of tooth flank (N), F_m is the mean load at the reference circle (N) and b is the gear face width (mm).

Uneven load distribution along the face width is caused by manufacturing errors and thermal distortions as well as gear mesh misalignment in the plane of action, comprising load-induced elastic deflection of gears, flexibility of housing and planet carriers, and deformation of bearings. Crowning and lead slope modification may be used to compensate for the detrimental effects of these deviations. The face load factor is the most important for gear strength rating and is greater than or equal to unity. It is equal to unity when the loads acting on a gear tooth flank are assumed to be distributed evenly. It becomes greater than unity when the load distribution on a gear tooth flank is assumed to be more uneven.