풍력터빈의 베어링 수명해석 및 계산
Analysis and Calculation of Bearing Life for Wind Turbine

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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 2007 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. The technical trend for wind turbines is to increase their reliability and efficiency instead of reducing the large cost of operation.

For the requirements of Wind Turbine Standards, the bearings, which are one of the most important components for wind turbine gearbox, should be designed to optimize reliability and economic efficiency.[2-3]

2. Bearing Reliability

Bearing technology, as well as the bearing industry, began to develop with the invention of the bicycle in the 1850's. And in 1924, A. Palmgren published a paper outlining his approach to bearing life prediction which was the basis for the basis for the Lundberg-Palmgren life theory.[4]

Lundberg and Palmgren obtained the following additional relation:

$$L_{10} = \left(\frac{C_r}{P_{eq}}\right)^p \qquad (1)$$

Where

L_{10} is 10% life, or operating time exceeded by 90% of a group of bearings;

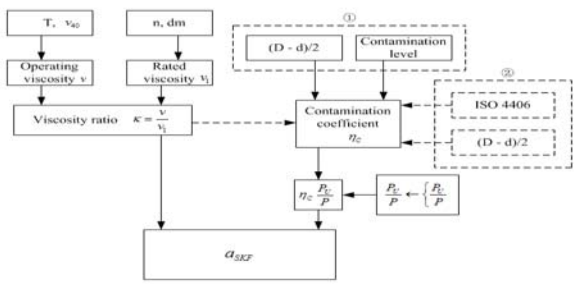
C_r is the basic dynamic load rating;

P_{eq} is the equivalent bearing load;

p is the load-life exponent, Lundberg-Palmgren use 3 for ball bearings and 10/3 for cylindrical roller bearings.

ISO (the International Organization for Standardization) and most bearing manufacturers like SKF, FAG, Timken etc., have adopted these values to the bearing rating life. Simplified analysis of bearing rating life is calculated according to ISO 281 as Basic Rating life calculation or Extended Basic Rating Life calculation which takes account of internal geometry, operation clearance, load sharing and distribution of rolling, lubricant viscosity, cleanliness and other effects. But the calculating procedure of modification factor from the bearing manufacturer is different, the example of modification factor for SKF is shown as follows:

Chart 1 The flowchart of SKF bearing life modification factor



For the flowchart 1, ① and ② are two calculation methods for contamination level factor η_C.

Life rating of the bearings can be used in different methods for the bearing companies. So it is necessary to compare these rating equations to get the right rating life (Tab. 1).

Table 1 Comparison of bearing life rating methods

Methods	Basic Rating Life	Adjusted Rating Life
ISO 281	$L_{10} = (\frac{C}{P})^{\varepsilon}$	$L_{nm} = a_1 a_{ISO} L_{10}$
SKF		$L_{nm} = a_1 a_{SKF} L_{10}$
FAG		$L_{nm} = a_1 a_{23} L_{10}$
NSK		$L_{nm} = a_1 a_{NSK} L_{10}$
Timken		$L_{nm} = a_1 a_2 a_{3d} a_{3h} a_{3k} a_{3l} a_{3m} a_{3p} L_{10}$
Romax Designer		$L_{nm,Adjusted} = f_{lz} L_{10,ISO}$
	$L_{nm,ADV} = \left(\frac{Q_c}{Q_e}\right)^{\varepsilon}$	
$\varepsilon = 3$, for ball bearings; $\varepsilon = 4$, for roller bearings.		

According to AGMA 6006, the calculated rating life may not be less than 130,000 hours. When adjusted methods are used, the result should be compared to DIN ISO 281 and the values produced by Tab. 2.

Table 2 Minimum basic rating life

Bearing position	Required life, L_{h10}, hr
High speed shaft	30,000
High speed intermediate shaft	40,000
Low speed intermediate shaft	80,000
Planet	100,000
Low speed shaft	100,000

Note:
Values in this table are valid for a design life of 20 years and shaft shall be adjusted for designs with different design life.

And the contact stress using the Miner's sum dynamic equivalent bearing load should not exceed the values listed in Tab. 3.

Table 3 Maximum contact stress for rolling element bearing at Miner's sum dynamic equivalent load

Bearing position	Maximum contact stress, P_{max}, MPa
High speed shaft	1300
High speed intermediate shaft	1650
Low speed intermediate shaft	1650
Planet	1450
Low speed shaft	no equivalent load

3. Analysis and Simulation

Gearbox failures continue to be a major source in wind turbines. Due to a variety of different reasons, this paper just takes account of bearing failures. And from the simulation of gearbox, bearing durability analysis including the effect of system misalignment and advanced contact stress analysis can be conducted.

For the RomaxDesigner software which has been applied to the wind turbine, a wind turbine gearbox of 2.0 MW can be modeled to reduce the development risks of the full gearbox system.[5] The schematic figure of two planetary gear stages and one helical wheel stage is shown as follows (Fig. 1). And the model of helical wheel stage is simulated (Fig. 2).

And by using the GH Bladed Software, the wind turbine model of turbulent wind simulations of the complete wind turbine lifetime can be obtained. And a LDD was used to analysis the gearbox.[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 are generally parallel shafts equipped with helical gears producing radial and axial loads that must be supported by the bearing system. But sun pinions should be designed without bearings to achieve load sharing. Types of bearing for this model are shown as follows (Tab. 4):

For Romax adjusted life, load zone factor is used to the ISO life for misalignment, clearances and other effects, but for advanced life, the bearing capacity and contact conditions are calculated from detailed bearing geometry.

After the simulation of load cycles, the modified life and the percentage damage for bearings are obtained:

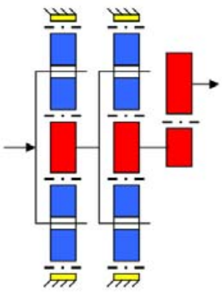


Fig. 1 Schematic figure of two planetary gear stages, one helical wheel stage (Courtesy of KISSsoft)

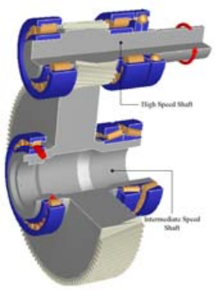


Fig. 2 The model of the helical wheel stage for 2MW Wind Turbine

Table 4 Types of bearings for gearbox components

Arrangement Position	Bearing Types	
Intermediate Speed Shaft	Free End	SKF Cylindrical Roller
	Fixed End	FAG Taper Roller
High Speed Shaft	Free End	SKF Cylindrical Roller
	Fixed End	SKF Cyl. Roller
		FAG 4 Point Contact Ball

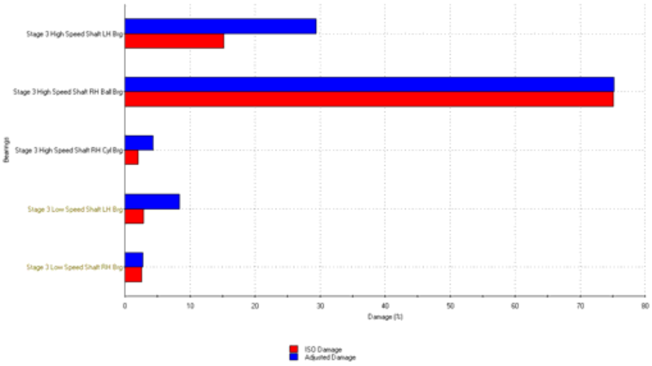


Fig.3 The percentage damage of bearing against cumulative damage duty cycle

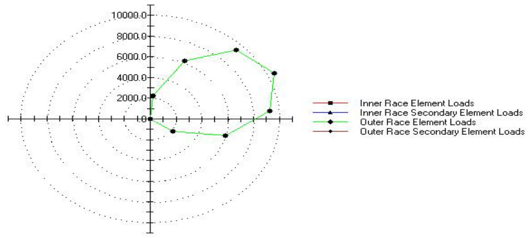


Fig. 4 The radar chart of the ball bearing for high speed shaft

Comparing the requirement of minimum basic rating life, in this case, all bearings survive during the duty cycle, but the right ball bearing of high speed shaft, the percentage damage is over 70% which is the highest one of these bearings, should be adjusted when it can not meet the demand of the whole system estimation (Fig. 3). And for the ball bearing of the high speed shaft, the radar chart shows that there are eight rollers under 9^{th} load cycle condition, and the maximum stress for one roller outer race is above 10000N (Fig. 4).

4. Conclusions

This paper simply analyses the bearing rating life methods of the Standard and different bearing companies. And by using RomaxDesigner software for the modeling and analysis of the helical wheel stage, the bearings used in this model survive in the load cycles.

Acknowledgements

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