Pitting Analysis Of Wind Mill Gear

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Abstract

A windmill is a machine that is powered by the energy of the wind. To achieve a better availability of wind turbines and to reduce the cost of wind energy, the deployment of new designs with reduced maintenance requirements and increased reliability is an important consideration for future development, especially for offshore wind turbines. The gearbox is one of the important subsystems in an indirect drive wind turbine providing the functions of transferring power from the low speed turbine shaft at high torque to the high speed generator shaft at low torque. One of the defects in the wind mill was that the gear used fails out due to pitting, hence in this project an attempt is made to increase the life of gear used in the wind mill. In our project an attempt is made to design a wind mill gear using Pro- e wild fire 2.0 and the stress analysis of gear is done by using ANSYS Work Bench and the result are calculated. By achieving this computation process of analysis the concurrent engineering concept, the product development cycle time can be reduced considerably for any kind o-f model with required quality. Based on the computation process and the detailed analysis the life of the windmill has increased up to 10% of the desired life. So it tends to reduce the breakdown the windmill failure and also increase the power production range.

Keywords: Pitting, Gear, Pro-e & Ansys.

1. Introduction

Power generation from wind has emerged as one of the most successful renewable energy technologies. Despite the facts of unpredictability and lack of control on wind energy wind mills are preferred as the source of power generation because of growing environmental concerns with respect to use of other conventional fuels and to preserve the finite resources of fossil fuels. The field reliability study of modern wind turbines shows a reduction in failure rates of the gearbox subsystem in comparison with other subsystems.

However, the gearbox has a low availability due to its high downtime per failure and gearbox failure incurs high costs for repair. However, for gear systems used in the wind turbine applications, the demand for gearboxes designed with extremely high gearratios and operating under conditions subject to a broad spectrum of load and speed variations makes the reliability prediction and failure prevention difficult. Current research initiatives in this area include the Gear Reliability Collaborative (GRC) of US NREL/DOE, which have committed research efforts to tackle this challenging problem. An important method in reliability analysis appropriate at the design stage is Failure Mode. It provides benefits to improve designs by identifying weaknesses in subassemblies and components of a complex system. Identification of weak points in design also prompts further improvement in the configuration design of the system.

1.1. Construction of Wind Mill

The wind mill can be divided to two major parts are Mechanical Turbine and Electric Generator. The following section gives a brief description on the Wind Energy System including the main components, design and operation. Among the above the Nacelle is a casing that houses the key components of a windmill such as Gear box, Generator, Electronic Control Unit, Yaw Controller, Brakes, etc. The rotor blades capture the wind's energy and convert it to rotational energy of shaft. The hub in turn transfers the energy to the low speed shaft. Blade designs operate on either the principle of drag or lift. The low speed shaft of the wind turbine connects the rotor hub to the gearbox.

1.2 Gear Box

Most wind turbine installations are equipped with similar gearboxes; typical are three-stage spur wheel gearbox (lower capacity area), one-stage planetary gearbox with two spur wheel stages (middle capacity area) and two-stage planetary gearbox with one spur wheel stage (higher capacity area). In addition to that, there are at present some interesting variants, see the majority of wind turbine installations has one rotor only that generally is up. Likewise, most installations have one generator only i.e, the gearbox has exactly one input and one output. Conceivably and partially implemented, are the four combinations. The several generators or rotors are unusual, the author does not know of any installation with several rotors and several generators. The wind turbine gearbox mission is that of transforming the low speeds and high torques at the gearbox input into higher speeds and lower torques at the output.

2. Literature Survey

A new approach is needed to overcome these barriers and accelerate the development of more robust gearbox designs. The Gearbox Reliability Collaborative initiated at NREL provides a fresh approach toward better gearboxes that combines the resources of key members of the supply chain to investigate design-level root causes of field problems and solutions that will lead to higher gearbox reliability. This survey is taken from the paper in title “Improving Wind Turbine Gearbox Reliability” by W. Musial and S. Butterfield.

Breakdown risk is one of the most common risk which results in mechanical damage to the windmill. A study carried out on the international and Indian windmill claims gives the following figures:

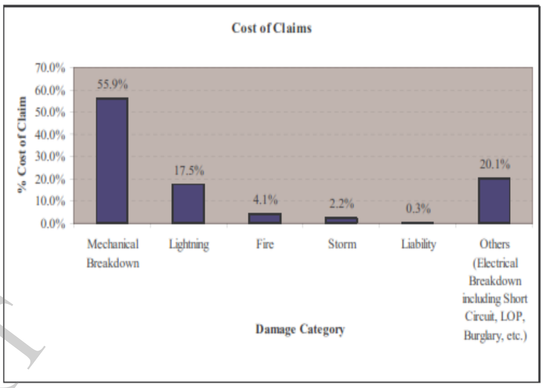

Figure 2.1: Cost of Claims

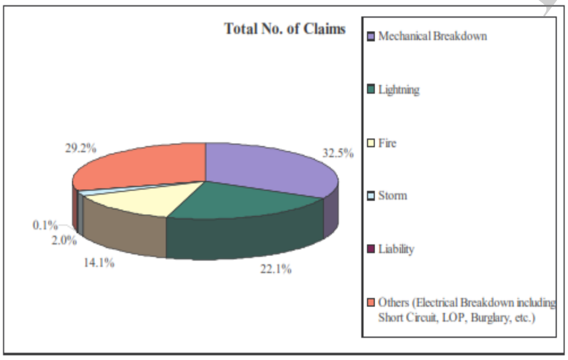

Figure 2.2 No of Claims

This survey is taken from “Breakdown Risks In Wind Energy Turbines” by J. Ramesh Babu and S.V. Jithesh, Solamandalam MS Risk service.

3. Problem Formation

Sixty – five percentage gears are failed by pitting fatigue failure, the gear is fully failed, and the gear is fully failed within 60 thousand hours. The new gear box is designed by replacing the gear material and new gear geometry are introduced.

3.1 Pitting

Pitting is a form of surface fatigue which may occur soon after operation begins and it is due to over loading, improper lubrication; sudden starting and stopping may lead to micro pitting. Due to the higher friction force, a shear stress develops on the tooth surface that can already exceed a critical value even for an uncritical torque for the building of micropitting. The resulting profile irregularities lead to a higher tooth load and it may be of three types: Initial Pitting, Destructive Pitting & Normal Pitting.

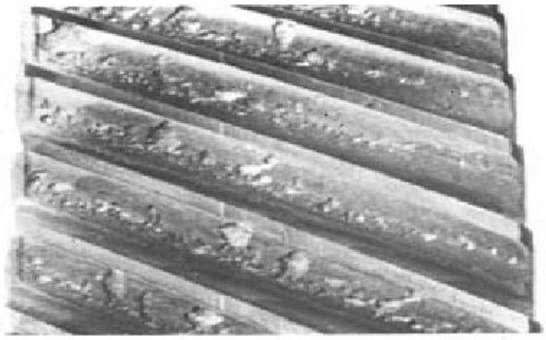

Figure 3.1: Pitting

3.2 Region of Pitting

- ✓ It occurs due to repetition of contact stresses.
- ✓ It appears in the dedendum section of the gear teeth.
- ✓ When the pitting resulted, the tooth is destroyed.

4. Methodology

The first step in this exercise was to design the anticipated gear sets. As gear ratings are parametric in nature, the approximate tooth size needed to carry specific load is to selected. As the dynamic factor decreases as size decreases, the rating slightly higher (< 10%) than the minimum acceptable values. Required gear sets were designed by Pro-E wild fire and then the simulation is carried out by ANSYS WORKBENCH.

Eliminating methods:

i. Replacing the gear material to have higher hardness.
ii. Redesigning the gear geometry.
iii. Eliminating axial force.

4.1 Specification of Present Double Stage Gear:

1. Power of the gearbox (P) =500KW

2. Type of gear: Helical gear.

3. The gearbox material is Chromium-Vanadium.

4. Rockwell hardness of the Material (HRC) = 41

5. Ultimate Tensile Strength of the Material = 13700kgf/cm^2

6. Lubrication- Flash type.

7. Type of bearing: Roller bearing.

4.2 Design Calculation for New Gear (Herring Bone Gear):

Calculation for two stage Herring bone gear:

First stage gear:

Material-Alloy steel (40 Ni 2 Cr 1 Mo 28)

Pinion (N_1)	Main wheel (N_2)
N_2 = 15 rpm	P = 500 kw
HRC = 57	i = 10

Design surface contact stress: $[\sigma_c] = 933.66 *10^6$ N/m^2

Design bending stress: $[\sigma_b] = 309.16*10^6$ N/m^2

Calculation of initial design torque (M_t):

$[M_t]$ = 21107.58 N-m

Calculation of centre distance: a = 1.04211 m

Calculation of normal module (M_n): $M_n = 7.98*10^{-3}$ m

Calculation of number of pinion and gear teeth (Z_1 and Z_2):

Z_1 = 20 (Assume) [Z_1=20] & Z_2 = 200

Calculation of pitch circle diameter of pinion and gear:

d_1 = 0.208865 m & d_2 = 2.08865 m

Revised Calculation:

Calculation of normal module (M_n):

$M_n = 7.25*10^{-3}$ m

Calculation of centre distance (a):

a = 1.14875 m

Calculation of design torque $[M_t]$:

$[M_t]$ = 22211.66 N-m

Checking design calculation:

$\sigma_c = 827.56 *10^6$ N/m^2

$[\sigma_c] > \sigma_c$, Design is Safe.

$\sigma_b = 122.61*10^6$ N/m^2

$[\sigma_b] > \sigma_b$, Design is Safe.

Permissible surface fatigue stress:

$(\sigma_{cp}) = 1425 *10^6$ N/m^2 : $(\sigma_{cp}) > \sigma_{c}$, Design is Safe.

Surface wear strength condition is satisfied.

Second stage gear:

N_2 = 150 rpm & N1 = 1500 rpm

$[M_t]$ = 2110.7666 N-m

Where,

$[\sigma_c] = 933.66*10^6$ N/m^2

$[\sigma_b] = 309.16 *10^6$ N/m^2

Calculation of centre distance (a):

a = 0.483709 m

Calculation of normal module:

$M_n = 3.67 \approx 4*10^{-3}$ m

Revised Calculation:

Calculation of normal module: $M_n = 3.86 \approx 4*10^{-3}$ m

Calculation of centre distance: a = 0.574375 m

Calculation of design torque: $[M_t]$ = 22221.1 N-m

Check calculation: $\sigma_c = 740.16 *10^6$ N/m^2

$[\sigma_c] > \sigma_c$, Design is Safe.

$\sigma_b = 91.72 *10^6$ N/m^2 $[\sigma_b] > \sigma_b$, Design is Safe.

Permissible surface fatigue stress:

$(\sigma_{cp}) = 1425*10^6$ N/m^2

$[\sigma_c] = 933.66 *10^6$ N/m^2: $(\sigma_{cp}) > \sigma_c$.

The condition is satisfied.

4. 3 Comparison Table for Present & New Gear

Table 4.1: Present and New Gear

Title	Present Gear (Helical)	New Gear (Herring bone)
Material	Chrome-vanadium	Nickel-chromium
HRC	41	57
Ultimate tensile strength	$\sigma_u = 1370*10^6$ N/m^2	$\sigma_u = 1550*10^6$ N/m^2
Surface contact stress	$\sigma_c = 635.60*10^6$ N/m^2	$\sigma_c = 933.66*10^6$ N/m^2
Bending stress	$\sigma_b = 279.7*10^6$ N/m^2	$\sigma_b = 309.66*10^6$ N/m^2
Module	First Stage gear Mn = $13*10^{-3}$m Second Stage gear Mn = $6*10^{-3}$m	First Stage gear Mn = $8*10^{-3}$m Second Stage gear Mn = $4*10^{-3}$m
Power	P = 500KW	P = 500KW
Gear ratio	1:10	1:10
Gear type	Helical gears	Herring bone gears

5. RESULT AND DISCUSSION

Pitting is maximum eliminated by the use of material and gear change, there by the best suited gear to be used in wind mill is herringbone gear.

Table 5.1: Analyzed values

Particulars	Helical gear	Herringbone gear
Stress deformation	$453.24*10^{-6}$ N/m^2	$554.97*10^{-6}$ N/m^2
Strain deformation	$0.0028328*10^{-3}$ m	$0.0022652*10^{-3}$ m
Total deformation	$0.0001382*10^{-3}$m	$0.0008076*10^{-3}$m

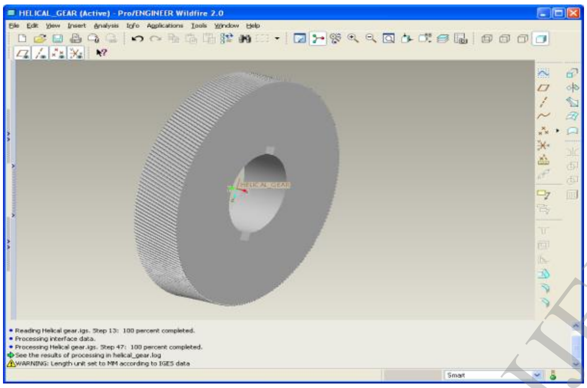

Figure 5.1: Helical gear design

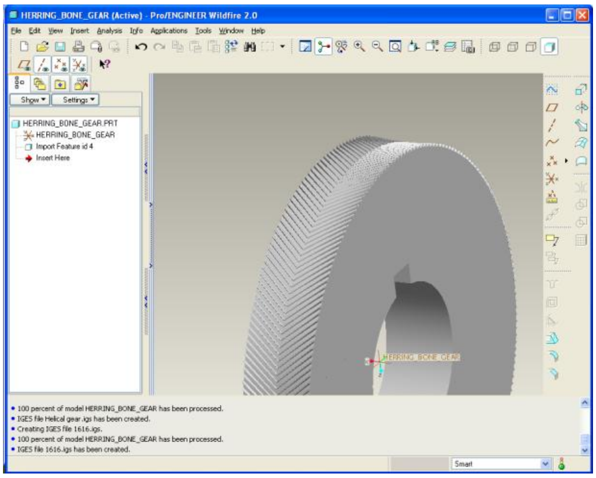

Figure 5.2: Herring bone gear design

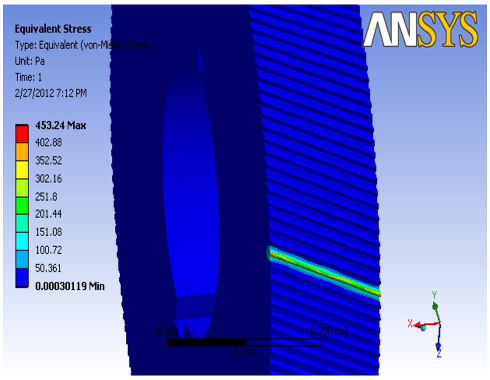

Figure 5.3: Helical gear stress distribution

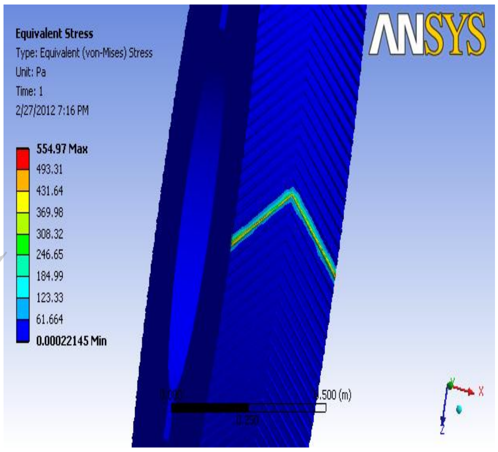

Figure 5.4: Herring bone gear stress distribution

6. Conclusion

Wind mills have been around for thousand of year. They almost resulted in failure due to gear. In this project it was found finally that herringbone gear is best suitable gear to be used in the wind mill. And also lifetime of gear increased almost to 85,000 hrs at full load. The present scenario is all toward wind energy, hence this project would hold up good regarding the construction of wind mill. Through this project it is concluded that herringbone would be the better choice to be used in windmill. By achieving this computation process of analysis the concurrent engineering concept, the product development cycle time can be reduced considerably for any kind of model with required needs.

7. References

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