

Article

Failure Analysis of Wind Turbine Planetary Gear

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Abstract: The article presents selected problems related to an analysis of damage to wind turbine planetary gear. It is the most vital element installed in wind turbines, affecting the operational costs (prolonged downtime), and costs of repairs and servicing including delivery of required components. The authors have analyzed the wear/failure of planetary gear. The process initiating fatigue wear, different from similar devices in other industries, can be easily observed in wind turbine gear. This establishes a specific direction of research into the causes, and early detection of ‘gas spots’ on gear teeth as they seem to trigger fatigue wear.

Keywords: wind turbine gear; failure analysis; damage to planetary gear toothed wheels; surface wear processes



Citation: Bejger, A.; Frank, E.; Bartoszek, P. Failure Analysis of Wind Turbine Planetary Gear. *Energies* **2021**, *14*, 6768. <https://doi.org/10.3390/en14206768>

Academic Editor: Frede Blaabjerg

Received: 13 September 2021

Accepted: 13 October 2021

Published: 17 October 2021

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1. Introduction

On average, it is estimated that the maintenance and operating costs of a wind turbine account for 20% to 30% of the total turbine maintenance costs.

Offshore wind energy plays an important role in meeting Europe’s renewable energy targets. Many other countries and regions outside Europe have also shown interest in offshore wind energy, including China, India, Japan, South Korea, Taiwan, and the United States [1].

The authors conducted a wear/failure analysis of the planetary gearbox of a wind turbine. It turns out that the process that initiates fatigue wear, is different from similar equipment in other industries. In fact, there are different processes in the gearbox of a wind turbine that cause the formation of micro-cracks. This sets a specific direction for research on the causes and early detection of “gas spots” on gear teeth of gears installed in wind turbines (especially offshore). The authors observed that, in combination with stray currents, they are the main cause of electrochemical corrosion on the surface of gears.

The gears are subjected to alternating loads due to wind speed changes and free braking pulses, which makes them one of the most fragile components of a wind turbine with low reliability [2,3]. According to Spinato et al. [4], gearboxes rank second in terms of downtime/failure due to their size and strong connection to other components, which makes access, repair or replacement difficult. Previous studies have shown that gearboxes typically do not reach their design life of 20 years and suffer early failures [5,6]. For onshore wind turbines in Europe, the average downtime/failure is 6–15 days and the failure rate is 0.1–0.15 failures/turbine/year [7]. They require replacement after 6–8 years, which is much shorter compared to their design life [7]. For the gearbox working state, the type of working force needs to be taken into the consideration. Wind speeds play a huge role in the process of power generation. The load of a gearbox and its working conditions are heavily dependent on this parameter and its consequences. To maintain wind turbine gearbox in good working state, the storm control strategy is implemented, depending proportionally on the wind speed and the wind turbine working program characteristic. This design behavior requires implementing a proper storm control strategy, and recognition of its effect on a gearbox and its vulnerable parts, such as planetary gear [8].

To prevent wind turbine breakdowns caused by major or minor failures, it is necessary to implement one of the condition-based turbine monitoring systems. Various monitoring-based methods have been improved to diagnose the performance, wear and operation of moving parts of a turbine. To monitor such performance, an analysis of Turbine Driving Monitoring is implemented based on vibration, using SSD Method [9], MSSD method [9] and Improved Kolomorov Complexity [9]. Another real time condition check method for diagnosing structural health of turbine moving parts is Acoustic Emission monitoring system [10]. Acoustic emission monitoring and strain monitoring systems allow prognosing and detecting real time existence of evolutionary defects of a wind turbine [10].

In addition, studies have been conducted into the seismic behavior of wind turbines affected by sea ice; because such wind turbines are new structural forms, there is a lack of codified seismic design provisions governing their analysis, especially for offshore wind turbines. The ice load induced structural failure is one of the main types of damage of offshore structures [11]. Such conditions can have a huge impact on the process of power and force transmission in a wind turbine gearbox, shafts and all connections. Therefore, finding a reasonable computing method is urgent for seismic design of offshore structures [11].

The paper [12,13] reviews various diagnostic and predictive methods based on implementation strategies, reliability and accuracy. The review also covers the challenges and future scope in conditioning and maintenance monitoring for large-scale FW. The authors have made a comparison of different (presented in the literature) methods for assessing the condition of wind turbines and factors contributing to their reliable operation.

The world's growing demand for renewable energy requires a compromise to be made between wind turbine cost and its efficiency calculated as a green energy profit. Factors to be considered during planning period are: design cost, including steel quality used, its class and project arrangement processes of construction a wind turbine, a choice of type of a structure, water or land base, leading to its calculated efficiency outcome. Some conclusions can be made with reference to Taguchi's Quality Loss Function (QLF) [14]. It states that the means to addressing quality goals are not necessarily by specifying higher quality material, but by controlling the process variations by improved process capability [14].

Therefore, efficient use of steel requires some understanding of how and why steel components fail in service. The purpose of such understanding is to prevent (or delay) the failure by taking certain precautionary and pre-emptive measures that can help to improve performance of the component, and not just opting blindly for enhanced design quality, making its production difficult and expensive [14]. Steel acquired in such a way has a great influence on gearbox components working condition and its reliability.

2. Development Trends in Wind Turbine Construction Elements

Research and development activities aimed to advance the technology and make wind more efficient and cost competitive continue to be conducted. Key approaches to cost reduction include [1]:

- Reduction of the turbine structure/weight;
- New drivetrain to eliminate or reduce the size of gearboxes;
- Turbine load analysis and mitigation, considering the dynamic coupling between translational (surge, sway, and heave), rotational (roll, pitch, and yaw) loading and turbine motions, as well as the dynamic characteristics of mooring lines for floating systems;
- Turbine and rotor designs to increase the efficiency and reliability and reduce weight;
- New generators and power electronics to increase the efficiency and reliability;
- Improving wind farm performance, considering interactions between wind turbines and also wind farms (e.g., improved wake models);
- Advanced maintenance strategies, remote monitoring, diagnostics, prognosis and health-monitoring systems, for improving reliability, reducing turbine down time, and operation and maintenance costs;
- Economic modelling and optimization of the overall wind farm system.

In the case of wind turbine operation, the gear and the generator are components most vulnerable to damage (Figure 1). They represent approximately 25% [7] to 35% of all failures.

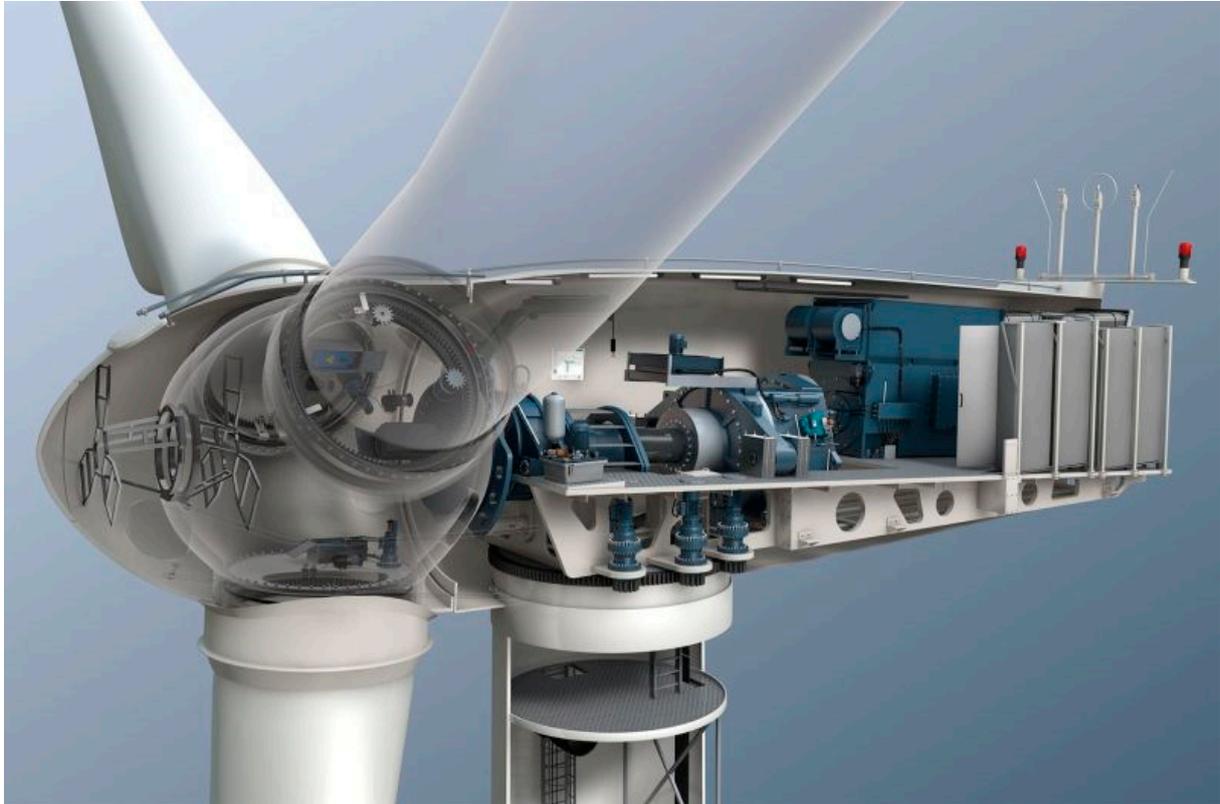


Figure 1. View of the main wind turbine modules (with permission of ifm.com) [15].

3. Operational Problems of Selected Wind Turbine Modules (Mechanisms Causing Damage to Toothed Wheels in Gearbox)

Environmental changes, humidity, temperature fluctuations, loads from high wind power and speed as well as foreign bodies causing mechanical damage and corrosion phenomena, are the reasons resulting in very difficult working conditions of wind turbine gearboxes. After only a few years of operation, repair and reconditioning costs of turbine gearboxes can be even more than USD 300,000. [9]. Turbulence and varying wind forces can cause enormous stress on gears and bearings. This in turn can lead to damage to individual turbine components, resulting in emergency shutdown. The greater the wind force, the more vulnerable the gears are. In particular, offshore turbines are the most susceptible to damage in this case. The gearbox is therefore one of the most maintenance-intensive components of a wind turbine.

The planetary gearbox of a wind turbine is one of the most important and vulnerable components.

Planetary gearboxes are significantly different from fixed-axis gearboxes and the choice of methods to assess their condition is much more difficult. Among others, in works [13,14,16,17] certain characteristic damages of planetary gearboxes were identified and analyzed. Studies on condition monitoring and damage diagnosis of planetary gearboxes based on the methodology adopted by the authors are summarized. Additionally, open problems were discussed and potential research topics were identified.

Condition monitoring and fault diagnosis at this time is quite a challenging task. However, there are few publications that have reviewed suitable fault diagnosis algorithms designed for wind turbine planetary gearboxes. Lei et al. proposed a general overview

of planetary gearbox condition monitoring and fault diagnosis [18], and more broadly, diagnosis methods were summarized and reviewed by Tianyang Wang et al. [12].

The following are most frequent forms of wear and damage to gear teeth: abrasive wear, bending fatigue, fretting corrosion, severe adhesion (scuffing), plastic deformation, tooth shear, contact fatigue (spalling), misalignment reduced efficiency, bending fatigue.

Later in the paper, the authors analyze a failure case of one of the units of the planetary gear module in a wind turbine (Figure 2). The gear toothed wheels were damaged along with the bearing unit (roller bearing cage completely destroyed).

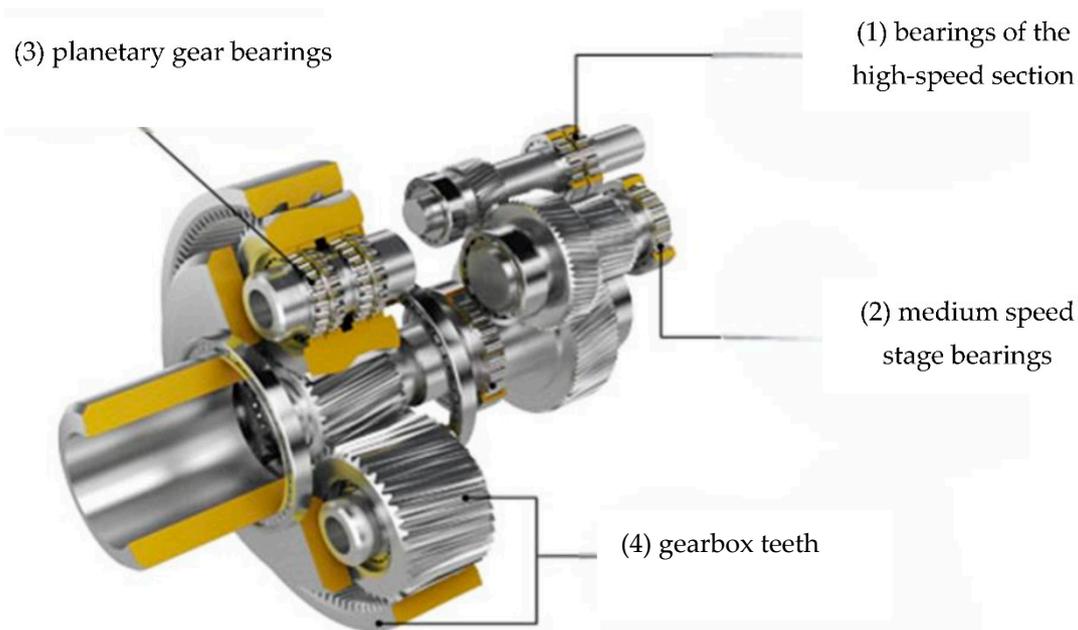


Figure 2. The planetary gear of the wind turbine (1—bearings of the high-speed section; 2—medium speed stage bearings; 3—planetary gear bearings; 4—gearbox teeth) (with permission of Olympus) [19].

3.1. The Forces Acting on the Gear Wheels of the Planetary Gear

Schematically, the rotor from the hub to the planetary stage is shown in Figure 3. Even if we ignore the load transfer through the planetary stage (Figure 3), the mechanical problem is an excessive constraint due to the clearances and elastic properties of the individual components that ultimately determine the load sharing among the bearings. However, we can reduce this problem by considering two limiting cases. Case one, when (PLC-A) has no clearance or compliance and therefore carries the entire lateral load ($F_{PS} = F_{PLC-B} = 0$). Equilibrium will be satisfied when

$$F_{PLC-A} = \frac{M_X}{L_b} - \frac{F_Y \times L_h}{L_b}$$

The loads on PLC-A in this case are in the order of 50% of the fatigue limit (or less) when $0 < m < 1$. PLC-A is therefore very unlikely to accumulate fatigue cycles, which is consistent with the fact that it has not been implicated as a significant contributor to premature gearbox failure [8].

In the most general case, the non-torque loads are responded to by the radial forces of the main bearing, PLC-A and PLC-B, and the planetary bearings, PL-A and PL-B. The distribution of these loads is undefined, so we consider two limiting cases. In case 1, only F_{main} and F_{PLC-A} are non-zero. To determine whether these moments contribute to bearing failure, it would be necessary to determine whether and by how much these moments increase the bearing loads. However, the authors assumed that for specific gearboxes installed in wind turbines, for which the failure-free life should be between 20

(for onshore turbines) and 30 years (for offshore turbines), there is no need to consider limit states for the loads involved. Therefore, in order to analyze the damage, the classical model of contact stress distribution was adopted.

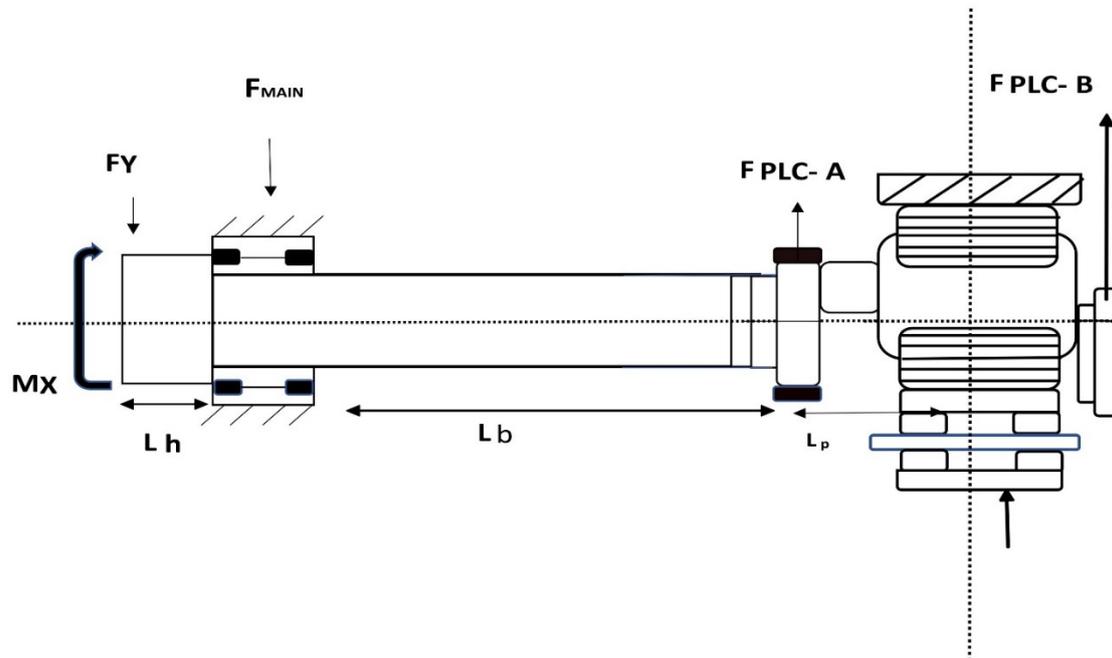


Figure 3. Free body diagram of the system from the hub to the planetary stage own elaboration based on [8,10].

In wind turbines, there is the additional problem of limiting the operating conditions and design space of the gear system. The planetary support is usually a journal bearing. However, due to unavoidable errors, such as (in a lesser case) manufacturing errors and more common assembly errors, there is usually an uneven load problem in the system. This has a negative impact on the load carrying capacity and durability of planetary gear systems. The works [20,21], among others, described the characteristics of journal bearing and analyzed the effects of operating conditions and elastic structure on load sharing characteristics. The results indicate that the load sharing characteristics are related to the journal bearing parameters.

Interesting research has presented in the paper [22], where has described a hybrid dynamic model for the analysis of nonlinear stochastic vibration of a wind turbine's planetary gear train with an elastic ring gear. Moving loads were considered, in which the mesh loads between pairs of planetary gears were taken into account. In addition, the model accounts for both stiffness variation and nonlinearity of the clearance between the meshing gears. The results show that the moving mesh has a strong effect on the planetary gear when the flexibility of the gears is taken into account.

3.2. The Distribution of Forces Acting in the Mating Gear Wheels System

In the Hertz model, the distribution of contact stresses (Figure 4) was determined for an a priori assumed function, which has the shape of a semi-ellipse. The values of stresses are determined taking into account the physical properties of the gear wheel material (mating teeth) i.e., longitudinal elastic moduli and Poisson numbers. The impact of the load will deform the teeth of the planet gear and the ring gear.

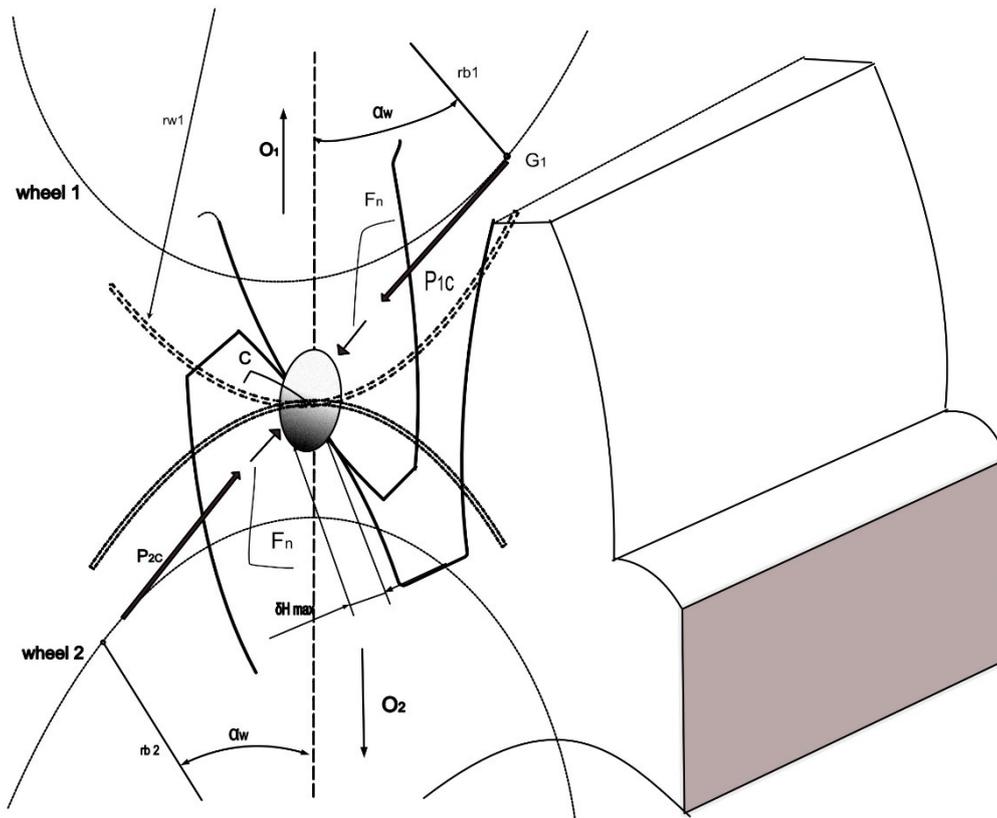


Figure 4. The distribution of contact stresses on the surface of the mating wheels (C—pressure point, σ —stress, ρ —radiuses of the curvature, α_w —angle of the contour, F_n —peripheral force).

According to the Hertz theory, normal stresses occur on the contact surface of cylinders:

$$\sigma_{Hmax}^2 = \frac{2F_n}{L \times \left(\frac{1}{E_1} + \frac{1}{E_2}\right)} \times \left(\frac{1}{\rho_1} + \frac{1}{\rho_2}\right) \times \frac{1}{2\pi(1 - \nu^2)}$$

If at the central pressure point C (pitch point) we assume the values of the curvature

$$\rho_1 = \frac{d_{w1}}{2} \times \frac{\sin \alpha_{tw}}{\cos \beta_b}, \quad \rho_2 = \frac{d_{w2}}{2} \times \frac{\sin \alpha_{tw}}{\cos \beta_b}$$

and take into account that

$$u = \frac{d_{w2}}{d_{w1}} = \frac{z_2}{z_1}, \quad L = \frac{b}{\cos \beta_b}$$

and denoting σ_{Hmax} as σ_H , we obtain the formula for calculating stresses on the surface of the contacting teeth (including the coefficient of oil viscosity, surface roughness and the peripheral speed)

$$\sigma_H = Z_E \times Z_B \times Z_H \sqrt{\frac{F}{d_1 b} \times \frac{u + 1}{u}} \times \sqrt{K_A K_V K_{H\alpha} K_{H\beta}}$$

where Z_E —material coefficient accounting for the properties of the mating wheels materials (where: ν —Poisson number, E —Young modulus) expressed by the formula

$$Z_E = \sqrt{\frac{1}{\pi \times \left(\frac{1 - \nu_1^2}{E_1}\right) + \left(\frac{1 - \nu_2^2}{E_2}\right)}}$$

Z_B and Z_D for $\alpha \leq 2$ -coefficients of one-pair pressure point of a tooth allow converting contact stresses at the pitch point to the maximum stress in the internal point B of the one-pair teeth of the pinion or internal point D of the one-pair pressure of the wheel tooth. If $Z_B > 1$ or $Z_D > 1$, it is recommended that the coefficient be determined for the wheels if $u < 1.5$. If $u > 1.5$, then generally $M_2 < 1.0$ (2.5) and $Z_D = 1$.

The parameters for determining Z_B and Z_D are calculated from the formulae:

$$M_1 = \frac{\operatorname{tg} \alpha_{wt}}{\sqrt{\left[\sqrt{(d_{a1}/d_{b1})^2 - 1 - \frac{2\pi}{z_1}} \right] \left[\sqrt{(d_{a2}/d_{b2})^2 - 1 - (\varepsilon_\alpha - 1) \frac{2\pi}{z_2}} \right]}}$$

$$M_2 = \frac{\operatorname{tg} \alpha_{wt}}{\sqrt{\left[\sqrt{(d_{a2}/d_{b2})^2 - 1 - \frac{2\pi}{z_2}} \right] \left[\sqrt{(d_{a1}/d_{b1})^2 - 1 - (\varepsilon_\alpha - 1) \frac{2\pi}{z_1}} \right]}}$$

For the cylindrical gear with straight teeth

where: ε_α -transverse contact ratio

$Z_B = 1$, if $M_1 \leq 1$ $Z_D = 1$, if $M_2 \leq 1$

$Z_B = M_1$, if $M_1 > 1$ $Z_D = M_2$, if $M_2 > 1$

Z_H —the coefficient of the contact zone depends on the geometry of the surface of the contacting teeth. It takes into account the impact of the gear tooth sides curvature at the pitch point on contact stresses and conversion of the peripheral force on the tooth reference cylinder to the force normal to the tooth surface on the pitch cylinder and equals

$$Z_H = \frac{1}{\cos \alpha_t} \sqrt{\frac{2 \cos \beta_b}{\operatorname{tg} \alpha_{tw}}}$$

The tooth surface is affected by the pressure of two contacting surfaces, that in the gear used in a wind turbine should be lower than the stresses acceptable on the surface of the teeth contact. The coefficient of surface fatigue strength is calculated for the resistance to an infinite number of cycles, and according to the ISO guidance recommendations the greatest stresses occur in the areas of contact with straight lines forming the 30° angle with the symmetry axis of the tooth. The highest values of contact stresses occur at the points of teeth contact of the two wheels, and at the tooth base. Areas of increased bending stresses at the tooth base occur on the side opposite to its load. Such distribution of stresses is known from the literature [13–15].

3.3. Fatigue Wear Mechanism in Gear Wheels

Analyzing the process of wear of the examined gear (Figure 5), the authors indicate that it occurs mainly due to fatigue wear. A local loss of cohesion occurs, and due to the cyclical impact of contact stresses in the surface layers of the mating friction elements, losses of the material take place.

Geer wheel teeth must have appropriate bending strength as well as wear strength. If two cylinders of elastic material are pressed against each other, they will be flattened along the whole contact width. This force causes internal stresses. The greatest stresses are created in the center of the flattened area. Particularly dangerous is the instant when the teeth contact each other in the area of the pitch circles where no slip occurs, consequently there is no oil film. In general, only one pair of teeth is involved in the work there.

At this point, let us distinguish surface fatigue wear and volume fatigue wear. The effect of surface wear processes is the local losses of surface layer material, while the latter cause fatigue cracks. Phenomena causing gear wheel wear observed in wind turbines are not entirely related to the classical fatigue damage initiating phase, known from industrial machinery. Visible effects have the form of ‘spots’ (Figure 6) that might be due to gases that appear inside the nacelle and settle on the surface of the gear teeth. This,

in turn, triggers micro-damage that changes to well-known pitting, spalling and, eventually, fatigue damage. The formation of a characteristic form of spots can also be the result of stress corrosion. Such corrosion can occur as a result of applied load or residual stress. The wind turbine gear experiences aggressive corrosion environment, with a temperature above 50 degrees Celsius and high concentration of chlorines. One can expect a combined impact of environmental conditions and stresses, which first result in points of pitting corrosion, then cracking begins in places of initiated pitting. It can continue to increase as transcrystalline or intercrystalline propagation.

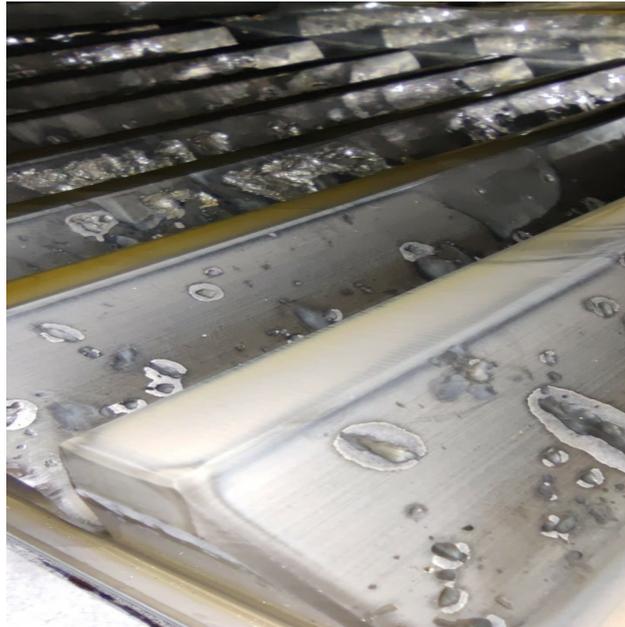


Figure 5. Damage to planetary gear toothed wheels.

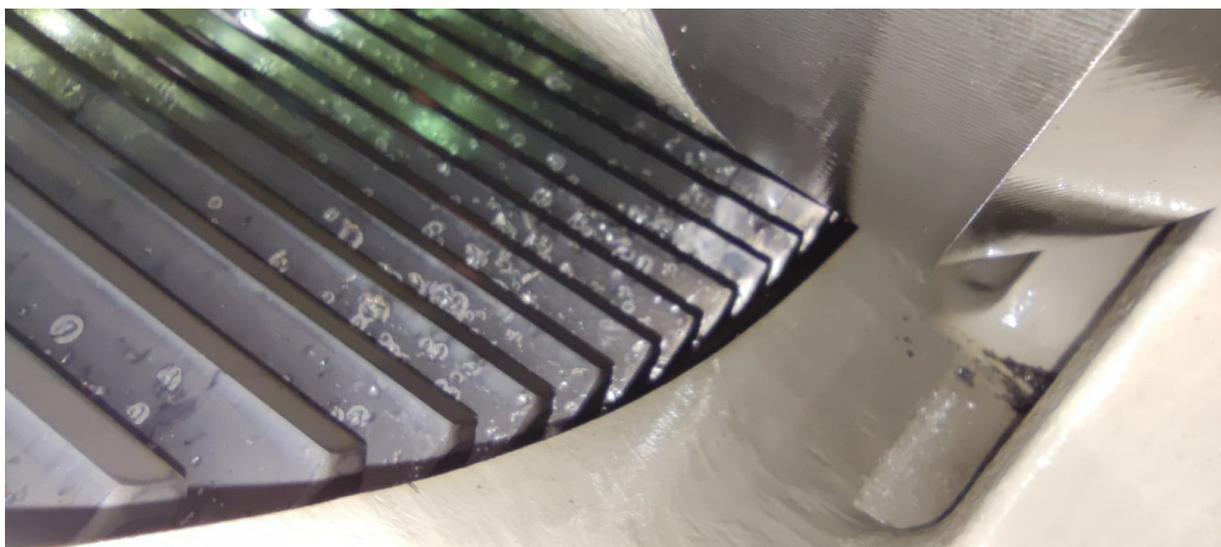


Figure 6. The effect of planetary gear teeth wear in wind turbines in the form of spots caused by gases released inside the nacelle.

In the initial phase pitting wear is the type of damage observed and herein examined. It occurs as a result of cyclical impact of contact stresses created in the surface layers of elements sustaining friction (rolling or rolling with slipping) at the lubricated contact

within the limits of Hertz stresses. Therefore, it is fatigue wear occurring in the presence of oil.

In machine assemblies, pitting occurs in units that are directly in contact with a lubricant (roller bearings, gears, camshaft drives). Damage to the surface layer causes a cavity where the lubricant gets in. The cavity filled with lubricant is subject to force (in this case, pressing oil between the mating gear wheels) which increases the pressure and the scope of damage (Figure 7).

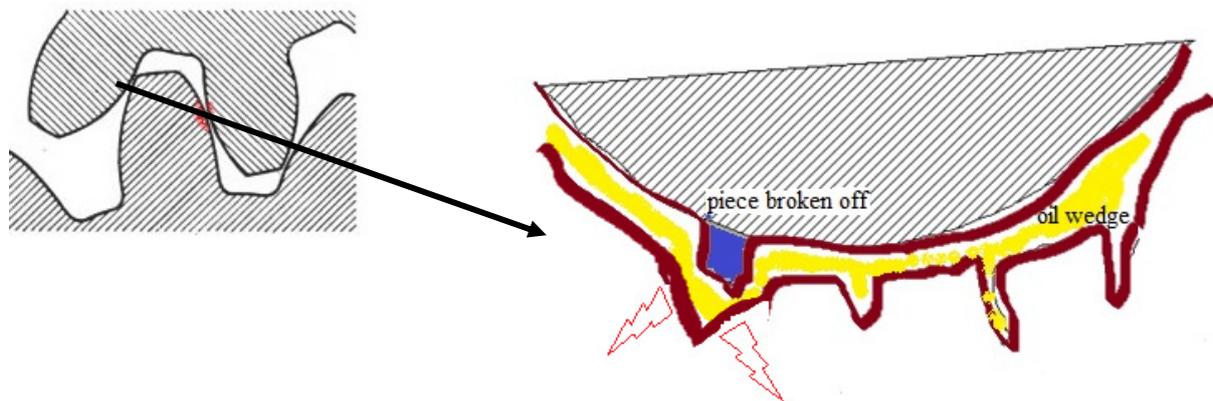


Figure 7. Process of pitting wear in the gear wheel.

Consequently, stresses gradually increase in the surface layers of the mating elements (rolling with slipping and breaking off of the oil film) at the dry contact within the limits of Hertz stresses as the result of cyclic impact of contact stresses. This leads to the appearance and propagation of microcracks, causing particles of the material to break off from the substrate. The phenomenon is particularly intensive in insufficiently lubricated elements. The resultant phenomenon of spalling manifests itself as local losses of material in the shape of scales separated from the substrate during friction. Spalling wear is created two to four times faster compared to pitting wear, and the material losses are much larger. Spalling wear depends only on the increase of stresses inside the surface layer due to the direct contact of friction elements, whereas as a result of growing stresses, developing microcracks reach the surface, resulting in the total loss of metal particles cohesion with the substrate. The general regularity of spalling wear is as follows: the process of spalling wear, manifested as fatigue-based cracks in metal surface layers, occurs in rolling friction under the impact of loads causing plastic deformation of metal micro-volumes in the contact zone. Quantitative changes of basic characteristics of spalling wear depend on the values of unit pressure at the contact, the number and frequency of load cycles, dimensions and shape of friction elements, and on the mechanical properties of metals. It is then apparent that in the case of heavy load wind turbine modules, this type of wear spreads intensively in planetary gears.

The process of spalling wear has three phases:

1. Creation of microgaps due to fatigue of the material and propagation of cracks;
2. Splitting microgaps by pressed-in lubricant during the rolling contact of teeth (growth and propagation of cracks is due to the oil wedge effect)—Figure 8;
3. Breaking off material particles from the surface layer (oil breaks off particles of metal which weakened or lost cohesion with the native material)—Figure 9.

Two basic stages can be distinguished: initial pitting and progressive pitting.

From the practical point of view, it should be noted that any surface defects and structure discontinuities can initiate fatigue cracks. During the first stage, oil hampers the progress of fatigue processes, as it reduces unit pressures in the contact zone, which positively affects the surface strength of the material. Consequently, fatigue wear in the presence of oil occurs significantly later than in the case of wear of non-lubricated mating

elements. In the second stage, fatigue cracks get larger and comprise increasingly greater parts of the material.

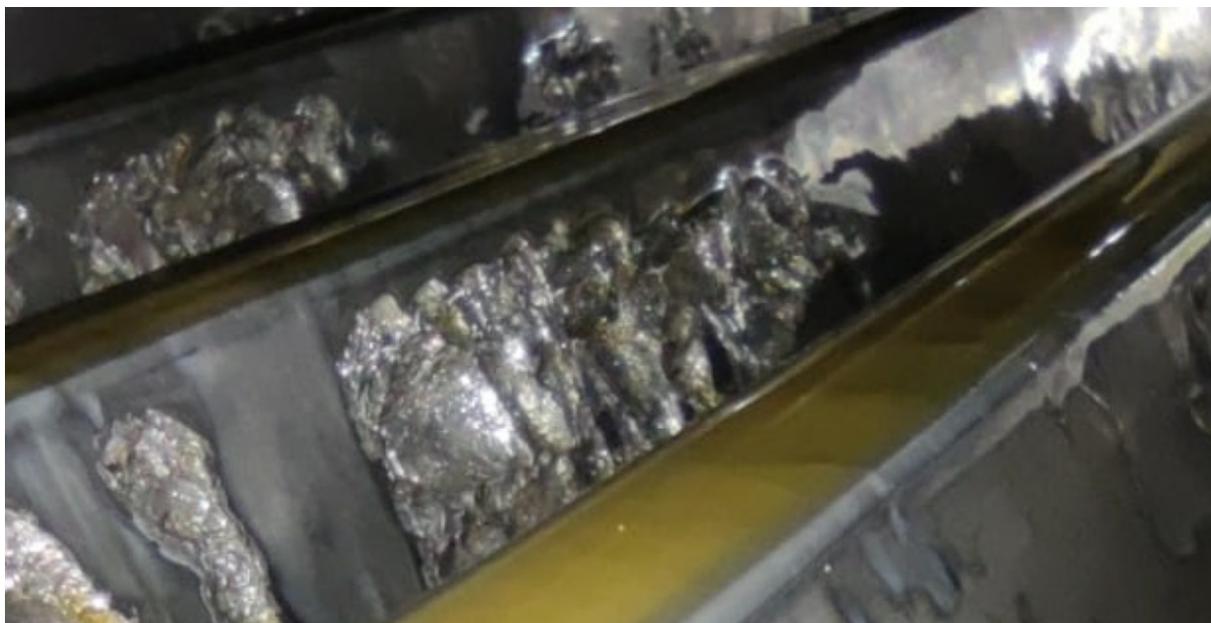


Figure 8. Spalling wear of gear wheels of the wind turbine planetary gear due to the cyclical impact of contact stresses.

Oil plays a major and adverse role in the propagation of cracks. The oil found in the contact zone within the boundaries of Hertz stresses is subject to high unit pressures and penetrates the gaps on the surface. The penetrating capacity of oil increases along with decreasing oil viscosity and surface tension. Highly refined oils used in wind turbines, containing dispersant and detergent additives, have a high penetrating capability. Oil is incompressible, so when pushed under high pressure into a crack, it acts as a wedge and increases crack size. Refining oil additives with high capacity for physical adsorption and chemisorption, sorbing on the internal surfaces of the cracks, lead to reduction of the surface energy and material cohesion inside the cracks, the process caused by adhesive forces. Therefore, oils with surface-active additives have a strong wedging action, thus contributing to the propagation of cracks.

In the third stage, the surface layer of a friction element already has a developed subtle network of gaps and cracks. Particles of the lubricant filling the space between two friction surfaces, particularly lubricant additives, form boundary layers on the surfaces. When two surfaces roll on each other, these layers are subject to compression and stretching. Stretched particles, strongly attached to the substrate, tear off some material that due to fatigue cracks have lost or weakened the cohesion with the native material.

Finally, fragments of metal get torn off from the core (Figure 10). With such impact lasting for a long time, depending on the parameters of the friction process, its nature and the magnitude of external loads, only macro-deformations occur instead of metal losses. After crossing the limit number of load cycles and reaching the fatigue limit by individual micro-volumes of metal in the surface layer, losses of the material mass occur, eventually leading to completely broken off gear wheel tooth.

Due to added deformations that result from superimposed natural stresses and those created by external loads, surface fatigue microcracks occur in areas of accumulated stresses, changing later into macrocracks, finally ending in the total damage to mating elements (Figure 11).



Figure 9. Breaking off material particles from the surface layer (arrow).



Figure 10. Broken off fragment of a planet wheel tooth in the planetary gear.



Figure 11. Total break-off of toothed wheel fragments in the wind turbine planetary gear.

Disturbed friction of mating elements also results in accelerated wear of bearings. In the case under consideration, the impact of additional loads in the bearing system caused the discontinuity of lubricant layers in the first stage. The initial form of wear was related to the adhesive wear of the first type in micro-areas of plastic deformation of the surface layer, particularly the highest roughness asperities. Local metallic bonds of friction surfaces occur along with the destruction of these joints combined with breaking off of metal particles or its smearing on the friction surfaces. The occurrence of increasingly larger unit pressures in the areas of the actual contact surface caused the mating surfaces to become closer and reach the range of molecular forces action. In the next stage, there is intense destruction of metal surface during slide friction, caused by the heating of friction zones to the temperature of metal softening, resulting in the formation of thermal-adhesion micro-bonds and the smearing of metal on the surface of friction elements (second type adhesive wear). In the softened metal, structures of crystallization, tempering and secondary hardening are formed in an uncontrolled manner, showing reduced resistance to wear. Consequently, the gap usually decreases between the friction surfaces, leading to the 'avalanche' increase of micro-bonds and seizure (Figure 12). Overheated zones of the bearing elements are clearly visible.

The present analysis of damage occurring in wind turbines reveals essential indicators that should be borne in mind during maintenance. From the practical point of view, appearing red and brown corrosion spots (or any deposits on the pitch elements or races), moisture from condensation (due to excessive temperature changes) may cause corrosion, which significantly increases the wear of the mating elements. Besides, the occurrence of vibration or increased radial clearance of bearing arrangements also affect the durability of the tested elements.



Figure 12. Bearing elements with visible heat-adhesion damage (arrow).

4. Conclusions

The authors propose extending research to identify sources of and analyze gases emitted in the area of the gear (focus on the concentrations of hydrogen and/or oxygen), and the impact of lubricant type, including oils, on possible increase of released gases that can be a source of micro-cracks triggering fatigue wear processes. In the case of the wind turbine, oxygen depolarization may occur on the surface of the mating gear wheels. It takes place in a neutral and alkaline environment and free air access. In many cases of electrochemical corrosion, both types of depolarization occur. Products created in the cathode and anode processes interact, which is visible as the emerging spots. Observations of actual gear damage let us conclude that there is not one rule for determining the probability of gear wear rate. If we assume that the first symptoms of gear wheels' wear appear due to environmental and corrosion factors, and that they become visible particularly early in the presence of current flow (corrosive cell), then it is likely that the time of their occurrence depends on the extent to which stray currents are present in the gear. Where such phenomena occur, the gear will suffer greatly accelerated wear. The cathode and anode processes are closely interrelated. The reception of electrons on the cathode facilitates the process of anode reaction, and when electrons from the cathode are not collected, the anode reaction will be hampered. Oxygen dissolved in lubricating oil in places of high concentration (good aeration) creates a self-contained oxygen electrode, with positive potential, becoming a cathode within some area of the metal. Other areas of the surface in contact with less aerated solution have lower potential and make up an anode area. A corrosive microcell is thus formed, the basis for subsequent wear processes, as described in the article, eventually ending in total damage. The authors suggest that during gear

servicing the tests for stray currents within the gear be performed. Stray currents are often generated due to contamination, e.g., on electric contact points, which combined with atmospheric moisture produces stray currents, spreading throughout the whole unit. Stray currents do not directly lead to electric damage, but as the article attempts to demonstrate, they may have a vital role in accelerating the wear process in wind turbine gears.

Highlights

1. Observations of actual gear damage allow us to conclude that there is no single rule determining the probability of gear wear rate;
2. The authors indicate that it is necessary to identify the sources and analyze the gases emitted in the gear wheels region;
3. Oxygen depolarization may occur on the surface of mating gears in a wind turbine, which is the cause of corrosion “foci”;
4. If we assume that the first symptoms of gear wear appear under the influence of environmental and corrosive factors, and that they reveal themselves particularly early in the presence of current flow (corrosion cell), it is likely that the time of their occurrence depends on the frequency and value of stray currents in the gearbox;
5. The occurrence of stray currents does not directly lead to electrical failures, but can have a significant impact on accelerating the wear process of wind turbine gearboxes—they are the cause of fatigue corrosion.

Author Contributions: Conceptualization, A.B. and E.F.; methodology A.B., E.F., P.B.; validation, A.B., E.F. and P.B.; formal analysis, A.B.; investigation, A.B. and E.F.; resources, A.B. and E.F.; writing—original draft preparation, A.B.; writing—review and editing, A.B.; supervision, A.B. All authors have read and agreed to the published version of the manuscript.

Funding: This research received no external funding.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Not applicable.

Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

The following abbreviations are used in this manuscript:

PLC-A and PLC-B	the radial forces of the main bearing
PL-A and PL-B	the radial forces of the planetary bearings
σ	stress
ρ	radiuses of the curvature
α_w	angle of the contour
ε_α	transverse contact ratio
F_n	peripheral force
Z_E	material coefficient accounting for the properties of the mating wheels materials
ν	Poisson number
E	Young modulus
Z_B and Z_D	coefficients of one-pair tooth point pressure at the inner point B of the one-pair pinion tooth pressure or the inner point D of the one-pair wheel tooth pressure, respectively
Z_H	the coefficient of the contact zone

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