FATIGUE FAILURE ANALYSIS OF A WIND TURBINE GEARBOX CAUSED BY TORSIONAL VIBRATIONS

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Abstract. In recent years, the demand for wind turbines for sustainable development has increased. However, they are still an inefficient mechanism, since almost 60% of their failures occur in the mechanical transmission, known as gearbox, affecting structural integrity, and reducing efficiency. The objective of this paper is to present the fatigue failure analysis of a wind turbine gearbox caused by the torsional vibrations and gear fatigue life prediction, the most common failures. Firstly, the NREL 750 kW wind turbine transmission will be shown in detail and its modelling, considering mesh stiffness, mesh damping and shaft stiffness, the equations of motion will be derived using Lagrange mechanics to calculate bending and contact stresses. Secondly, the gearbox were simulated and the results were obtained using the numerical simulation software Matlab©, and the NREL 750 kW baseline turbine was modelled in Simulink / Simscape. The results showed the transmission is able to operate within the required velocity range out of the resonances. The contact stresses attained higher levels compared to the bending stresses, reducing gearbox life to less than 10 years. The failures most likely to occur were tooth breakage in the planetary gear, pitting and scuffing on pinion-gear pairs.

Keywords: gearbox, vibrations, fatigue, failure analysis

1. INTRODUCTION

The worldwide wind energy sector has been rapidly developing in recent years and the total wind energy capacity, showed in Figure 1, reached 742,7 GW in December/2020, according to the data from Global Wind Energy Capacity (GWEC). However, wind turbine is still an inefficient system, 60% of failures occur in the gearbox, reducing its 20-year-old projected useful life, reaching from 8 to 10 years (OYAGUE, 2009). Predicting the wind speed variations, material properties and imperfections in the system assembly, the gearbox is submitted by torsional vibrations.

According to the figure 2, Sheng (2012) explains the gearbox is responsible for high maintenance and replacement costs, along with the costs caused by production losses due to those components do not work, constitute a large part of the expenses of operating wind farms, resulting in greater downtime and maintenance compared to other wind turbine components.

The purpose of this paper is to obtain the fatigue stresses of wind turbine gears, their useful lives and the predominant failures from modeling subjected to torsional vibrations and simulated in a Matlab/Simulink environment. The paper brings an alternative modelling to determine fatigue stresses and useful lives through torsional vibrations, mainly with the normal forces linking angular displacements obtained from Lagrange method.

Because wind turbine gears operate at high speeds (1000 to 1800 RPM for the NREL 750 kW model), this subjects the system to vibrate at high frequencies, resulting in fatigue failures in the system. Two predominant fatigue damage modes are characteristic: tooth bending fatigue and contact fatigue. The most common fatigue failure is contact fatigue, causing pitting and scuffing, normal wear and plastic surface deformation; the other failure is bending fatigue, resulting in the gear teeth breaking (BECKHAM, 2019).
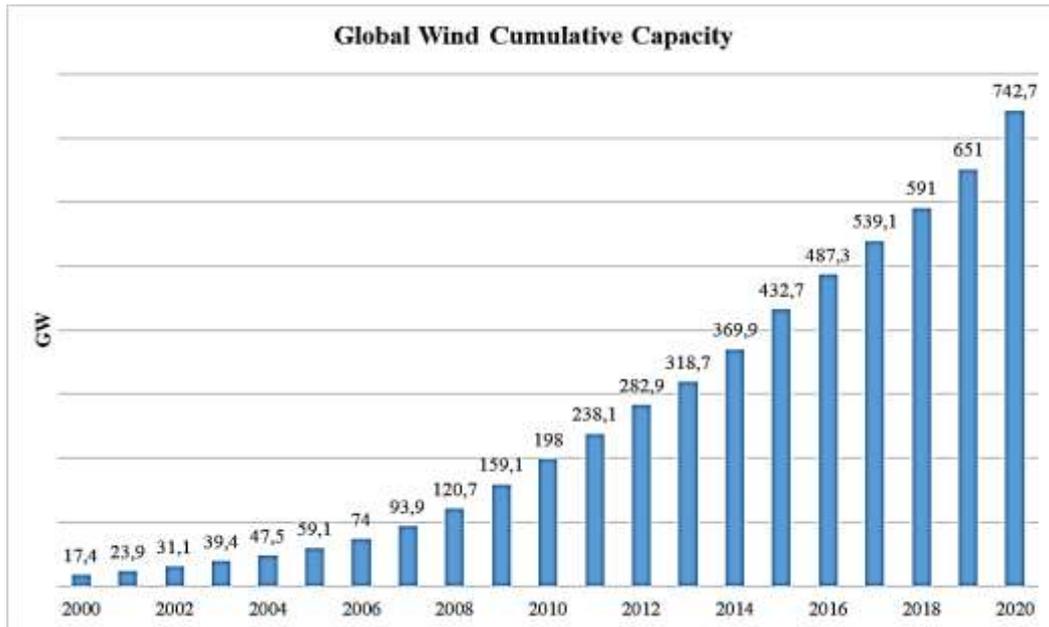
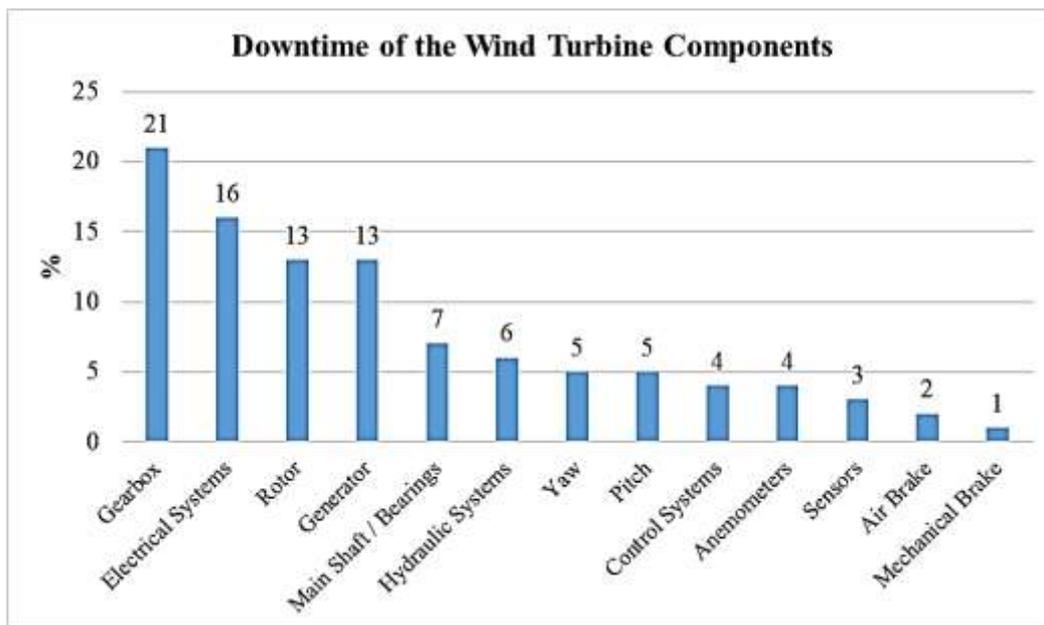Figure 1 – Global wind cumulative capacity over the years.

Figure 2 – Downtime of the wind turbine components.

2. GEARBOX MODEL

The gearbox used in this paper is the model designed by National Renewable Energy Laboratory (NREL), with a power of 750 kW and generator speed varying from 1200 to 1800 rpm. Figure 3 details this system basically composed of a planetary gear train (PGT), with the annular gear locked in the housing, and two pinion-gear pairs, a low-speed and a high-speed pair.
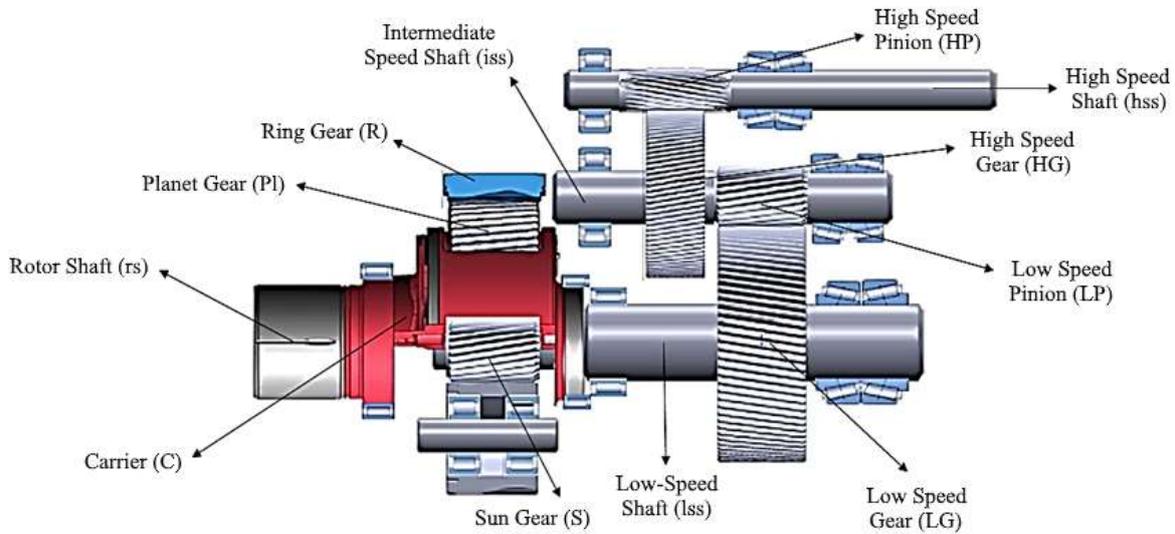
Figure 3 – Representation of the NREL 750 kW gearbox model.

A very important feature to consider is the line of contact, which occurs in the gear teeth, moves along the tooth surface due to the relative motion by the line of action. This relative motion generates friction on the teeth surface, which consequently causes fatigue failure. Furthermore, the transmission error is defined by the difference of the ideal and real angular displacements in rotation caused by the elastic deformation of the teeth, mainly by bending forces and by the existing gap; the latter contributes to the phenomenon known as gear mesh stiffness (GIRSANG *et al.*, 2014). This stiffness is modelled by a linear spring between the gears and its value is obtained experimentally.

According to Zhao and Ji (2016), gear mesh damping has been neglected in many existing works in the study of wind turbine gearbox, however it represents great importance in gear dynamics, because it acts as energy dissipation element to reduce any chance of transmission instability. A linear viscous damping detailed in equation 1 as a function of the mesh stiffness represents the mesh damping. Figure 4 shows the complete representation of the contact between the gears and equation 2 shows the normal force acting on the gear surfaces.
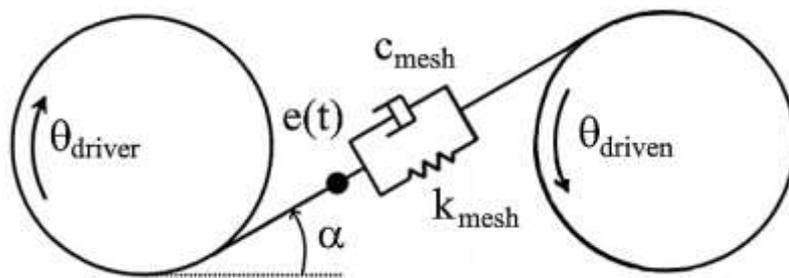

Figure 4 – Representation of the contact model between gears.

$$c_{mesh} = 2\zeta \sqrt{k_{mesh} \frac{J_1 \cdot J_2}{J_1 R_1^2 + J_2 R_2^2}},$$ (1)

$$F_N = k_{mesh}[R_1 \cdot \theta_1 - R_2 \cdot \theta_2 - e(t)] + c_{mesh}[R_1 \cdot \dot{\theta}_1 - R_2 \cdot \dot{\theta}_2 - \dot{e}(t)],$$ (2)

Where ζ is the mesh damping factor, $e(t)$ is the transmission error, R, J, θ and $\dot{\theta}$ are, respectively, pitch radius, polar inertia, displacement and angular velocity of the driver gear (1) and driven gear (2). Zhao and Ji (2016) define ζ as an experimental value varying between 0,03 and 0,17.

3. GEARBOX DYNAMICS

The gearbox dynamic modeling was consolidated according to the works by Girsang *et al.* (2014), Al-Hamadani and Long (2017), considering gearbox, rotor and generator. This paper considered the following hypotheses: there is only one degree of freedom per gear: the polar angular displacement (rotation around the center axis); all gears are coupled to

minimize transmission error; all shafts are aligned and they are not long enough to be bended by vibration caused by spin and whirl, avoiding gyroscope effect and gear misalignment. Shaft stiffness and the electrical resistive torque of the generator, the latter represented by an end-of-course torsion spring, were also considered. Figure 5 details all components.
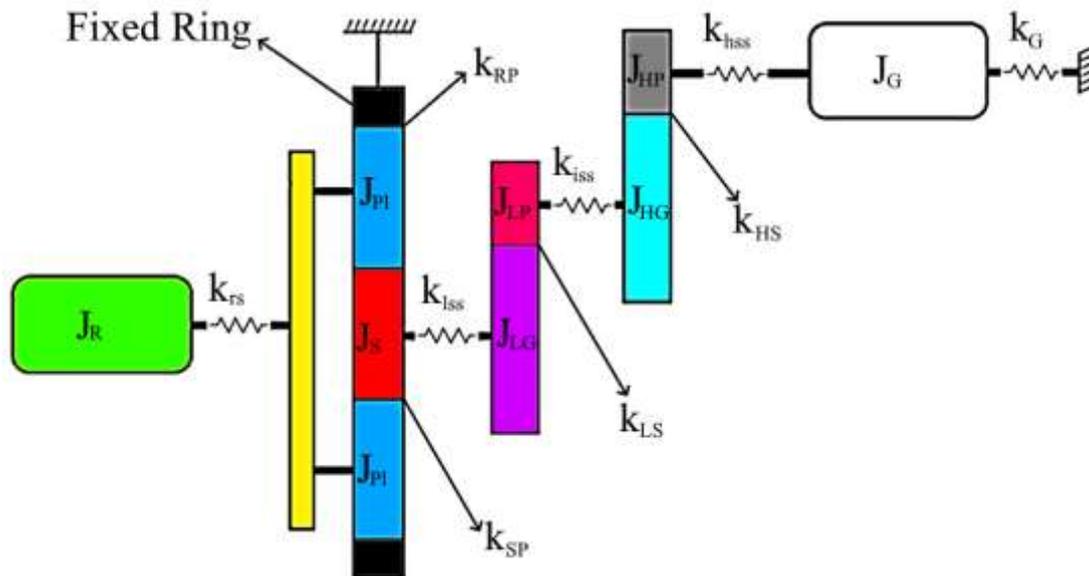

Figure 5 – Topology of the NREL 750 kW gearbox.

Where k are the stiffness of the rotor shaft (rs), low speed shaft (lss), intermediate shaft(iss), high speed shaft (hss), ring-planet mesh (RP), sun-planet mesh (SP)), low speed mesh (LS), high speed mesh (HS) and generator (G). The J are the polar inertia momentums of the rotor (R), planetary gear (Pl), solar gear (S), low speed gear (LG), low speed pinion (LP), high speed gear (HG), high speed pinion (HP) and generator (G). Mesh damping is not expressed to avoid overloading the image with too many parameters. The planetary gears were considered as one for dynamic analysis, but separate for fatigue analysis. Lagrange mechanics is used to obtain the equations of motion, which are expressed in equation 3 by the matrix form.

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{\bar{F}\}, \tag{3}$$

Where $[M]$, $[C]$ e $[K]$ are the mass, damping and stiffness matrices, respectively; ; $\{x\}$, $\{\dot{x}\}$ e $\{\ddot{x}\}$ are, respectively, the displacement, velocity and angular acceleration vectors and $\{\bar{F}\}$ is the generalized force vector.

Because of the vibrations, the authors implemented a PID feedback control model using the second method of Ziegler-Nichols. Torque is an active external disturbance because it disturbs the rotation (control variable) and ensures the transmission operation beginning, so it is not possible to apply an open loop in the system control. For these reasons, torque regulation is made by an internal feedback to allow greater stability, to ensure the generator operates under certain conditions and to adjust the efforts that act on the gear teeth. Figure 6 details this control model applied to the NREL 750 kW transmission.
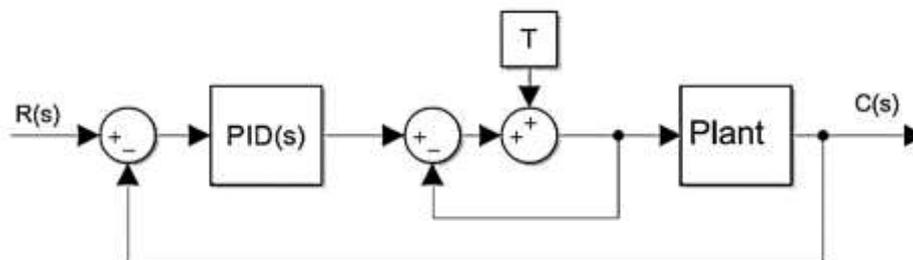

Figure 6 –Representation of the NREL 750 kW transmission control.

To compare with the values obtained from the equation of motion, the NREL 750 kW transmission was also modeled in Simulink/Simscape, seen in the figure 7. The Simulink/Simscape models electrical, mechanical, hydraulics and other elements and allows a closer approximation with physical systems.

Figure 7 – Model of the NREL 750 kW transmission in Simulink/Simscape.

3.1 Fatigue and failure analysis

Once the normal forces were obtained by equation 2, the next step was to determine the bending fatigue (equation 4) and contact fatigue (equation 5) stresses of the gears. As they are oscillating stresses, they were normalized for equivalent stresses (equation 6) by the Goodman criterion.

$$\sigma_F = \frac{F_N.\cos\beta}{m_N.w_F.Y_{GF}}, \tag{4}$$

$$\sigma_H = Z_E\sqrt{\frac{F_N}{Y_{GH}.w_F}\frac{R_1+R_2}{R_1.R_2}}, \tag{5}$$

$$\sigma_i^{eq} = \frac{\bar{\sigma}_i.\sigma_{ia}}{\bar{\sigma}_i-\sigma_{im}}, \; i = F,H, \tag{6}$$

Where β is the helix angle, m_N is the normal modulus, w_F is the tooth face width, Y_{GF} and Y_{GH} are the Lewis factor for bending and contact fatigue, respectively, Z_E is the coefficient of elasticity, in $\sqrt{MPa}$; $\bar{\sigma}$, σ_a, σ_m are the allowable stresses, stress amplitudes and average stresses for bending and contact fatigue. As the planetary gear does two contact points, so they were analyzed with two different stress values for each one, with four stress values in total.

With the equivalent stresses were obtained, the last step was to obtain the useful lives, in cycles (equation 7), by the characteristic equation and, finally, convert them into years (equation 8), according to Rey (2007) and Jelaska (2012).

$$N = \left(\sigma/\sigma_0\right)^m,\tag{7}$$

$$N_{years} = \frac{N}{n_{RPM}.Q_G}\left(\frac{1}{60.24.365}\right),\tag{8}$$

Where σ_0 is the fatigue strength stress for 1 cycle, m is the Basquim exponent, both determined by the material applied to the gears; n_{RPM} is the theoretical rotation, in RPM, in the pitch radius of the gear and Q_G is the number of peripheral gears coupled. The most common gear failures are tooth breakage, pitting and scuffing. Figure 8 details the most common failures it can occur in each gear due to the power flow (in red) along the transmission from the rotor to the generator.
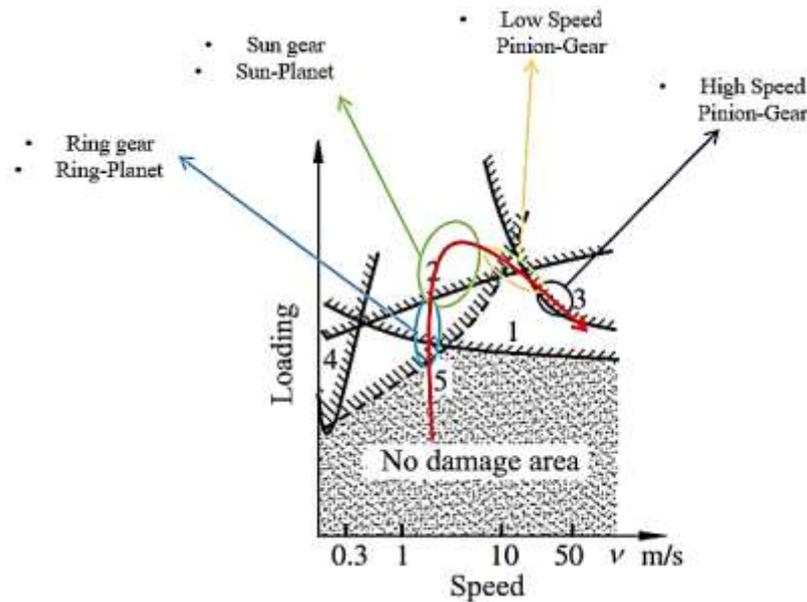

Figure 8 – Gear failure diagram with: 1) Tooth breakage, 2) Macro-Pitting, 3) Scuffing, 4) Wear (Wear), 5) Micro-pitting.

For the PGT gears, the material chosen was 4320H steel, as it guarantees greater torque absorption; For the pinions and gears, 4820H was chosen, as it allows greater resistance to friction due to the high rotations to which these gears are submitted. Table 1 presents the parameters of the gears, which are: polar moment of inertia (J), pressure (α) and helix (β) angles, number of teeth (Z), normal modulus (m_N), face width (w_F), pitch radius (R), theoretical rotation (n_{RPM}) and Lewis factors. These values were obtained from Jelaska (2012) and Juvinall and Marshek (2017)

4. INPUT PARAMETERS

Table 2 shows the properties of the contacts between the gears, including stiffness and damping of the meshes (k_{mesh} e c_{mesh}), fatigue resistance stress (σ_0) and Basque exponents for bending (m_F) and contact (m_H). According to Jorgensen et al. (2014), the fatigue strength stress values should be 10 to 15 times the tensile strength value of the material. The values of stiffness and damping were extracted from studies by Girsang et al. (2014) and Al-Hamadani and Dong (2017).

Table 3 shows the values of the shaft stiffness and the electrical resistive torque of the generator, according to the article by Girsang et al. (2014).

Q_G values are: two for the planetary gear, three for the sun gear, and one for the others.

Table 1 – Properties of the NREL 750 kW gears

Element	J (kg.m2)	α (°)	β (°)	Z (-)	mN (mm)	wF (mm)	R (m)	YGF	YGH
Rotor	998138						35,00		
Carrier	65,2			–			0,30		–
Planet Gear	3,2	20	7,5	39	10	227,5	0,20	0,51	0,08
Sun Gear	1,02	20	7,5	21	10	220	0,11	0,45	0,08
Low Speed Gear	31,72	20	14	82	8,25	170	0,35	0,58	0,08
Low Speed Pinion	0,4	20	14	23	8,25	186	0,98	0,46	0,08
High Speed Gear	3,42	20	14	88	5	110	0,23	0,58	0,08
High Speed Pinion	0,08	20	14	22	5	120	0,06	0,46	0,08
Generator	24			–				–	

Table 2 – Properties of the NREL 750 kW meshes

Contact	k_{mesh} (N.m/rad)	c_{mesh} (kg/s.rad)	σ_0 (MPa)	m_F	m_H
Ring-Planet (RP)	1,22E+10	9,07E+01	15750	5,8	13,2
Sun-Planet (SP)	1,45E+10	8,64E+04	15750	5,8	13,2
Low Speed (HS)	1,63E+09	1,16E+04	17550	5,8	13,2
High Speed (HS)	5,70E+08	4,87E+03	17550	5,8	13,2

Table 3 – Properties of the NREL 750 kW shaft stiffness

Eixo	k (N.m/rad)
Rotor Shaft	3,69E+07
Low Speed Shaft	2,45E+07
Intermediate Speed Shaft	2,70E+08
High Speed Shaft	2,08E+06
Generator Electrical Resistance	2,41E+04

5. RESULTS AND DISCUSSION

First, it was to verify if NREL 750 kW transmission is not operating on resonances. According to Sheng (2012), the generator must operate between 1200 and 1800 *RPM* (125 to 188 *rad/s*) and, based on the table 4 of the natural frequencies obtained both by the eigenvalue method and obtained through the Simscape linearization system, the system operates outside the frequency range.

Table 4 – Natural frequencies of the NREL 750 kW gearbox by eigenvalue and Simscape.

[rad/s]	Eigenvalue	Simscape
ω1	3,82	5,35
ω2	46,11	34,72
ω3	817,39	1836,00
ω4	2017,80	2396,00
ω5	3056,40	7057,00
ω6	11167,00	22500,00
ω7	12217,00	33400,00
ω8	24823,00	53500,00

Second, it was to verify if the generator could operate within the required rotations. The results showed the generator stabilizes after 1 *s* of operation, proving it can study the gears without problems with the generator.

Errichello (1994) determined some pre-requirements for wind turbine gears:

• Bending stresses should be between 345 and 483 *MPa*.

• Contact voltages must be between 1034 and 2068 *MPa*.

Table 5 presents the results of the final bending and contact stresses for all gears using Goodman criterion.

Table 5 – Bending and contact fatigue stresses of the NREL 750 kW gears.

[MPa]	σ_F	σ_H
Ring-Planet	228,61	355,06
Sun-Planet	430,74	1485,6
Sun Gear	430,74	1429
Low Speed Gear	428,18	2111,9
Low Speed Pinion	428,18	2256,2
High Speed Gear	284,91	3117,7
High Speed Pinion	284,91	3352,6

In general, the contact stresses had higher values in relation to the bending stresses, plus, as the transmission power flow increases, the bending stress values fall, while the contact stress values increase. This happens because of the power equation ($P = T.\omega$), since the focus of the transmission is to reduce torque coming from the rotor and increase the rotation that arrives at the generator. The bending stresses were equal for each geared pair and the explanation for this would be Newton's Third Law (Action-Reaction).

In fact, planetary gears are geometrically larger in relation to the sun gear, but its behavior in each contact was different: as the ring gear is locked, the relative motion of the ring-planet contact prevails only the displacement of the planetary gear, obtaining stresses of less than 300 MPa, while the contact stresses were less than 1000 MPa. However, in the sun-planet contact, the bending stresses were on average 350 MPa and the contact stresses were between 1500 and 1600 MPa; the reason for this is the tangential velocity ends up being the greatest; hence, normal forces will be the greatest.

Analyzing the low speed pinion-gear pair, in relation to the SEP, the bending stresses reduced and the contact stresses increased, however from 100-250 MPa above the desired range. This small difference may have happened during the development of tooth contact modeling. Consequently, this reduces the fatigue life. The high speed pinion-gear pair showed striking results: bending stresses were less than 300 MPa, below the desired range (345 to 483 MPa) and contact stresses were over 3000 MPa, far above the desired range (1024 to 2048 MPa), according to Errichello (1994). While respecting the power equation, high contact stresses values compromise the integrity of the gears, this means that they are more likely to fail quickly due to contact fatigue-related failures with short service life.

5.1 Fatigue and failure analysis

Table 6 shows the values of the fatigue lives of the gears and figure 9 shows those values in a histogram. The ring-planet values are not showed in the histogram due to the higher values.

Table 6 – Bending and contact fatigue lives of the NREL 750 kW gears.

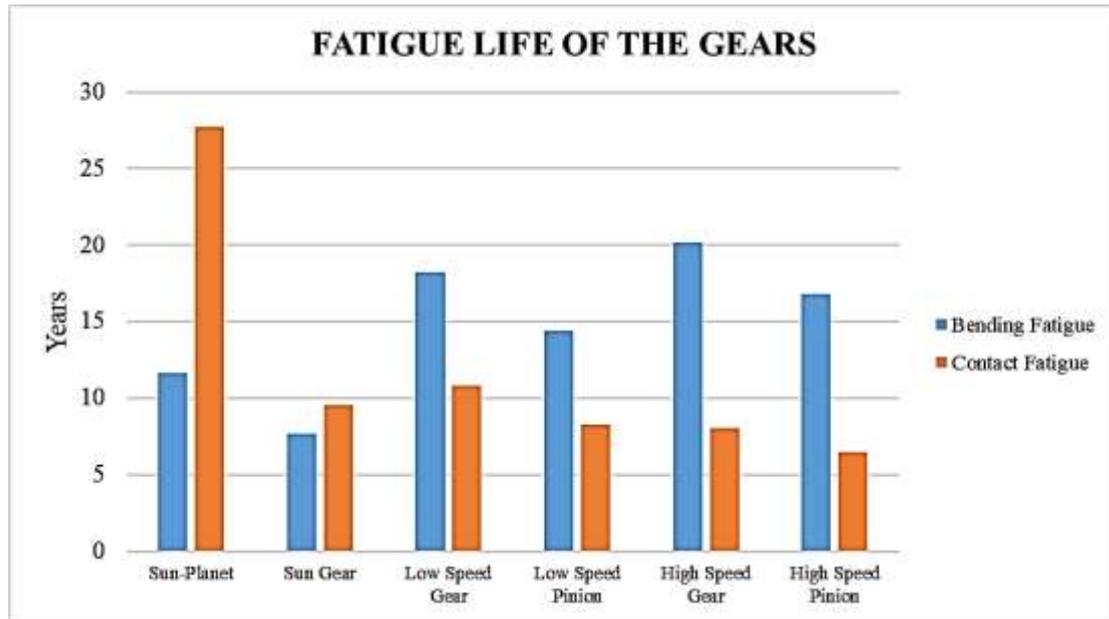| [years]	N_F	N_H
Ring-Planet	9,18E+04	4,76E+08
Sun-Planet	11,65	27,72
Sun Gear	7,64	9,53
Low Speed Gear	18,25	10,84
Low Speed Pinion	14,39	8,27
High Speed Gear	20,18	8,04
High Speed Pinion	16,82	6,45

Figure 9 – Histogram of the Fatigue Life of the Gears.

Where N_F is the bending fatigue life and N_H is the contact fatigue life.

Sun gear showed interesting results: its lives were between 7-10 years. This makes sense as the sun gear makes three couplings with the planetary gears, which greatly reduces its useful life. It concluded that simplifying planetary gears can be useful in a dynamic analysis, but it is not realistic for a fatigue study.

In the study of bending fatigue, the sun gear has the shortest service life in the system, which was expected, because it is the gear with the least amount of teeth coupled to three planetary gears subjected to low rotations, so the applied torque will be the higher. Gears had the longest service lives. The detail about this study is, more detailed the model, shorter the useful life, with the exception of the sun gear. The reason would be the shaft stiffness reduce the efforts, therefore they reduce the power flow that reaches the generator, so the pinion-gear pairs need to increase the efforts to compensate this energy absorbed by the shafts.

The planetary gear, coupled with the sun gear, obtained useful lives above 25 years for bending fatigue life, however obtained lives at least 15 years for contact fatigue life; this is due to the same fact that occurs in bending fatigue, since the solar gear has more gears coupled, so the chance of failure triples. The high-speed pinion, on the other hand, had the shortest useful life of the system, failing in an average of 6 years and the reason would be the equivalent high stresses submitted. The same happens with the high-speed gear, which had identical life as the low-speed pinion instead of the low-speed gear. Fortunately, the life was not shorter, because the high-speed gear has larger diameter and less rotation in relation to the pinion. Rey (2007) and Errichello (1994) explain contact stresses that are far above the limit excessively compromise the gear and allow it to fail earlier than expected.

The planetary gear, in contact with the ring gear, had very high fatigue lives (in the order of 10^4 to 10^8) for all models. The explanation would be, since the ring gear is fixed to the housing and the internal gearing allows for a greater distribution of tension, all efforts are minimized and end up being compensated in contact with the sun gear. The most important feature of this study is the NREL 750 kW gearbox will have a maximum service life of 7 years.

6. CONCLUSIONS

The planetary gears simplification influenced this study, because, from the dynamic point of view, the gears had the same behavior as one and separately. However, in terms of structural integrity, they cannot be simplified, as this reduces the amount of contact points that the sun gear does, allowing longer service lives on models where the system has been simplified.

The bending and contact stresses respected the transmission ratios, demonstrating the model developed in Simscape reached the expectations for fatigue analysis, as well as the control model, essential for this system, although it was very simple for a very complex system for a wind turbine gearbox.

The results are also related to the predominant failures of the gears. PGT gears have tendency to fail by breaking teeth because of the high torques, while the pinion-gear pairs have tendency to fail by pitting and scuffing because of the high rotations.

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