Tribology International 101 (2016) 375–382

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

Energy efficiency tests in a full scale wind turbine gearbox

Carlos M.C.G. Fernandes [a,*], Luis Blazquez [b], Jorge Sanesteban [c], Ramiro C. Martins [a], Jorge H.O. Seabra [d]

[a] INEGI, Universidade do Porto, Campus FEUP, Rua Dr. Roberto Frias 400, 4200-465 Porto, Portugal
[b] BP Portugal, Lagoas Park – Edifício 3, 2740-266 Porto Salvo, Portugal
[c] Sincro Mecánica S.L., Ctra. de Cedeira Km. 1,4, 15570 Narón Coruña, Spain
[d] FEUP, Universidade do Porto, Rua Dr. Roberto Frias s/n, 4200-465 Porto, Portugal

ARTICLE INFO

Article history:
Received 14 March 2016
Received in revised form
27 April 2016
Accepted 3 May 2016
Available online 4 May 2016

Keywords:
Wind turbine gearboxes
Gear oils
Additives
Efficiency

ABSTRACT

A 850 kW wind turbine gearbox, widely used in wind farms, was used to perform efficiency tests of different wind turbine gear oil formulations.

One mineral and three polyalpholephin commercially available lubricants were chosen, all them are ISO VG 320 wind turbine gear oils.

Study the influence of different gear oil technologies in the overall gearbox transmission efficiency is the main objective of this work.

The results show that different energetic efficiency and different oil particle counting were obtained even when the same base oil is used.

1. Introduction

Renewable energies, due to concern over the environment, represent a new path into world sustainability [1–4]. Wind is considered to be one of the most effective and one of the world's fastest-growing renewable energy sources [5]. One of the reasons is that wind is an infinite and free source of energy with no harmful waste products.

Wind turbines are those which convert the kinetic energy present in the wind to mechanical energy. The blades of a wind turbine rotate at very low speeds, typically 20 revolutions per minute, which are not suitable for conventional power generation using an electrical generator. This constraint is solved using a multiplying gearbox between the hub and the electrical generator [6,7].

The rapid growth in wind power needs fast improvements in technology, so in order to make wind energy competitive with other power sources in the near future, enhancements on gearbox efficiency, availability, reliability and lifetime are required [8–11].

Car or bus gearbox's efficiency is often considered very high and the power loss problem is mainly focused on the engine and vehicle weight [12,13], however, wind turbine gearboxes handle several megawatt of power and even a small efficiency increase in the gearbox can save energy useful for several more households [14].

In order to increase gearbox efficiency it is important to quantify the main sources of power loss. The most common wind turbine gearboxes have one or two planetary gear stage and two or one parallel stage. The main power losses sources are friction loss between the meshing teeth [15–20]; friction loss in the bearings [15,21–23]; friction loss in the seals [15,23,24]; no-load gear losses [25–30]; air-drag losses [31–33].

Since the total power loss is due to no-load and load losses, it is important to select a lubricant that promotes low friction in the contact of power transmitting components, which will lead to increased rolling bearings, gears and lubricant life. Simultaneously, the lubricants physical properties should also promote low no-load losses which also contributes to decrease the lubricant operating temperature.

The power loss reduction has a direct influence on lubrication quality, i.e. promotes lower heat dissipation and lower oil operating temperature. Lowering the operating temperature minimizes oil oxidation and degradation, which has a large impact on the lubrication quality and consequently on the surface protection against failures. Höhn et al. [34] showed that reducing the oil temperature also reduces the risk of failure. Even in gearboxes without failure problems overtime, the oil replacement will be less frequent contributing for the reduction of all the maintenance costs associated [14].

The author's recent works [30,35–42] focused on the evaluation of the tribological performance of different wind turbine gear oils using several laboratory test rigs to test ball-on-disc contacts, rolling bearings, gears and gearboxes. The results clearly showed

* Corresponding author.
E-mail address: cfernandes@inegi.up.pt (C.M.C.G. Fernandes).

http://dx.doi.org/10.1016/j.triboint.2016.05.001

the influence of the oil formulation and allowed to calibrate a gearbox power loss model [20,23,43,44]. The model was then applied to a 2.5 MW wind turbine gearbox to verify the possible energy savings by change the oil formulation [14]. However, the results presented were not experimentally verified on a full scale gearbox. This study intends to discuss the efficiency of a full scale wind turbine gearbox with 850 kW of power capacity. While the study deals with the gearbox efficiency, the main focus is the gear oil influence on the gearbox efficiency.

2. Wind turbine gear oils

Four ISO VG 320 gear oils, suitable for the lubrication of wind turbine gearboxes, were selected. All wind turbine gear oils selected have the same viscosity grade, ISO VG 320, and are expected to have approximately 320 cSt at 40 °C.

MINS is a mineral base oil with EP additives. PAOF and PAOM are polyalphaolephin base oils with an additive package for heavy duty industrial applications. According to FTIR analysis, PAOF and PAOM include approximately 7% and 11% of ester oil, respectively. The addition of ester oil to a PAO base oil is done with the objective of improve compatibility and dispersion of additives. PAOX is based on polyalphaolephin but with a distinct additivation technology based on PD (plastic deformation) additives and does not include ester oil on the formulation. The lubricants are formulated to achieve full EP performance and meet the requirements of DIN-51517.

Table 1 displays the wind turbine gear oils physical properties as well as their chemical composition (ICP). Among the PAO's, PAOX has clearly a different formulation with much lower sulphur concentration, zinc free and has molybdenum and calcium in high concentrations. The FTIR spectra are presented in Fig. 1 for all wind turbine gear oils.

A reference mineral wind turbine gear oil (MINV) with EP additives and Flender approval was always used to lubricate a driving gearbox and its physical properties are presented in Table 2.

Table 1
Physical and chemical properties of the wind turbine gear oils.

Parameter Base oil	Unit [−]	MINS Mineral	PAOF Polialphaolefin	PAOM	PAOX
Physical properties					
Density at 15°C	[g/cm^3]	0.903	0.855	0.860	0.856
Viscosity at 40 °C	[cSt]	320	320	320	320
Viscosity at 100 °C	[cSt]	25	36.6	37.4	34.9
VI	[/]	100	162	166	152
Chemical composition					
Calcium (Ca)	[ppm]	5	31	1	2074
Iron (Fe)	[ppm]	<1	0	3	1
Magnesium (Mg)	[ppm]	1	1	6	8
Molybdenum (Mo)	[ppm]	1	0	2	1059
Sodium (Na)	[ppm]	5	1	2	0
Phosphorus (P)	[ppm]	298	216	494	374
Silicon (Si)	[ppm]	4	13	20	23
Sulfur (S)	[ppm]	19,800	4931	4562	1930
Zinc (Zn)	[ppm]	4	4	23	0
Vanadium (V)	[ppm]	0	0	4	0
Acid Number (AN)	[mg KOH/g]	0.48	0.44	0.82	1.78

3. Test rig

The test rig follows an energy closed-loop principle as shown in Fig. 2. An electric motor is unable to drive directly a gearbox with very high torque and a speed of 20 rpm, so a driving gearbox reducer is necessary to decrease the speed from the electric motor to the input speed of the test gearbox. An electric motor (1) drives the gearbox reducer (2), then the gearbox reducer output (2) is connected to the input shaft of the test gearbox (3). The test gearbox transmits power at desirable rotational speed (usually 1620 rpm) to the electrical generator (4). The electrical generator will feed back the electric motor with the generated energy. The difference between the input power to the electric motor and the power generated by the electrical generator is the power loss of all the system.

3.1. Driving and driven gearboxes

The driving gearbox is oil jet lubricated with MINB oil during all the tests. The driving gearbox works as reducer ($i \approx 1/47$), see Fig. 3. The driving gearbox has a compound planetary architecture.

The test gearbox is also oil jet lubricated with the candidate wind turbine gear oil (MINS, PAOF, PAOM or PAOX). The gearbox has 850 kW of power capacity and operates as multiplier, as presented in Fig. 4. This gearbox has one planetary stage plus two parallel helical gear stages with a ratio of $i \approx 62$. The generator needs to be driven at 1620 rpm that combined with the two gearbox ratios of the test rig, implies that the electric motor input speed must be 1230 rpm.

The driven (or test) gearbox uses 140 l of oil for the lubrication system. The filtration flow is between 70 and 90 l/min. Each gearbox has its own pump, container and circulation system. The

Table 2
Physical properties of wind turbine gear oils used in driving gearbox.

Parameter Base oil	Unit [−]	MINV Mineral
Physical properties		
Density at 15°C	[g/cm^3]	0.904
Viscosity at 40 °C	[cSt]	320
VI	[/]	96

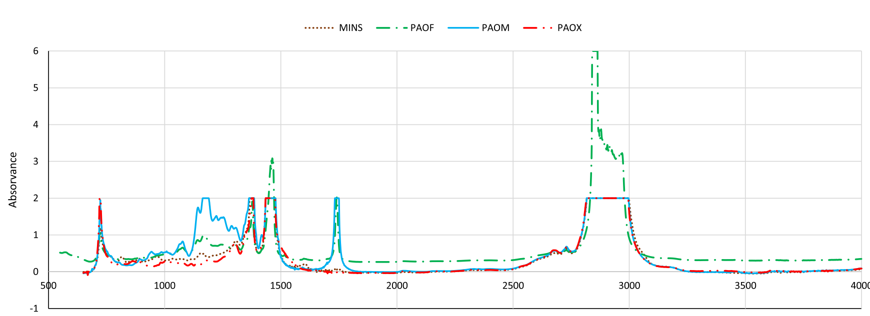


Fig. 1. FTIR spectra for all wind turbine gear oils.

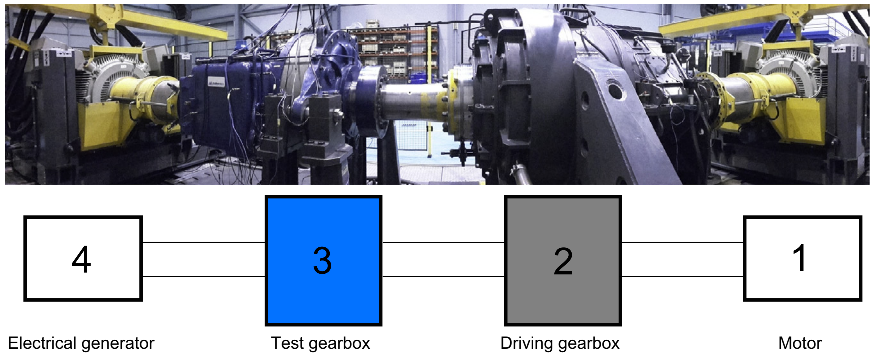


Fig. 2. Gearbox test rig assembly.

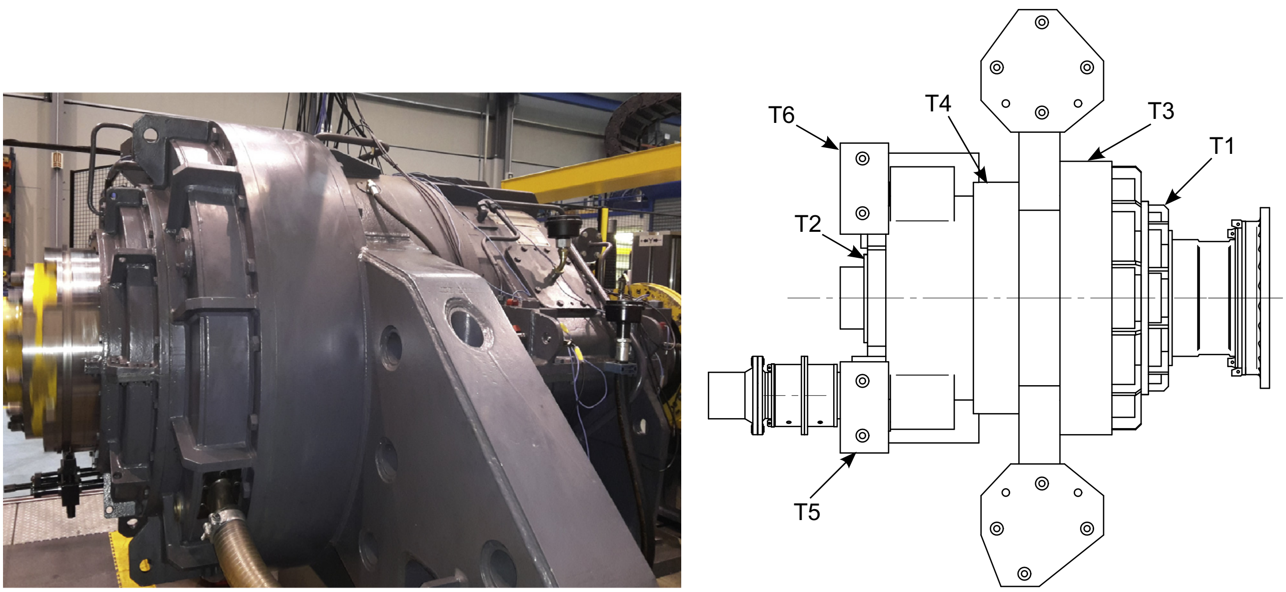


Fig. 3. Driving gearbox working as reducer ($i \approx 1/47$).

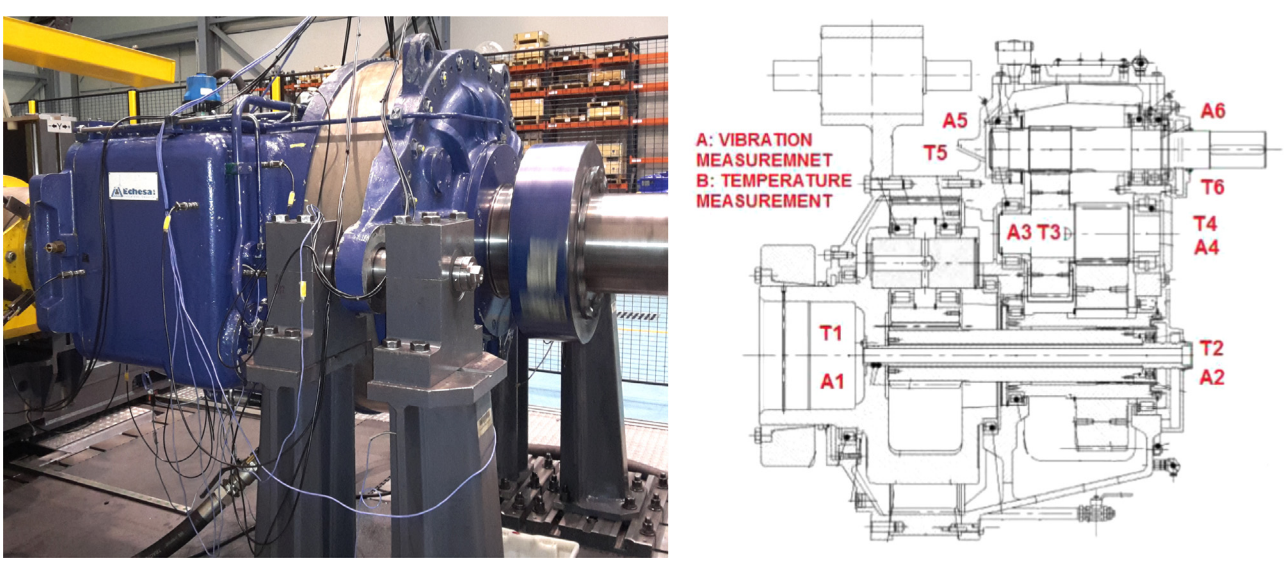


Fig. 4. Test gearbox working as multiplier ($i \approx 62$).

lubrication system has an inline filter of glass fiber with a filtration rating of 5 μm with $\beta \geq 1000$, similar to those used in a wind turbine.

The oil temperature is set free for both gearboxes. The oil operating temperature depends on ambient temperature and oil/gearbox performance [41].

Each gearbox has 8 PT100 in order to monitor its temperature. The measurement locations are displayed in Table 3.

The driven (or test) gearbox has a vibration measurement system as well as online particle counter for oil cleanliness code measurement (according to ISO 4406:99 and AGMA 6006-A03).

4. Test procedure

The operating conditions used in the power loss tests are displayed in Table 4, i.e. the generator rotational speed, the input power in the system and the duration of each test stage.

The test procedure can be summarized as follows:

1. Measure power loss with an output speed of 1620 rpm and input power of:
   - 1 kW (no-load condition) during 60 min;
   - 280 kW during 30 min;
   - 560 kW during 30 min;
   - 850 kW during 90 min.
2. Temperatures and oil cleanliness ISO codes were recorded.

The power loss is recorded each minute, storing the input and output power of the system, i.e. the power consumed by motor (1) and the power delivered by the electrical generator (4), see Fig. 2.

The power loss values presented ahead are the average of the recorded values during each load step, i.e. 280, 560 and 850 kW. The power input values selected for the first and second load steps were 1/3 and 2/3 of full power capacity (850 kW). A wind turbine gearbox operating in field application is most of the time working under 2/3 of full power capacity. So, 560 kW is a quite representative test for the most frequent operating condition in a real wind turbine of this size. The output rotational speed was kept constant, 1620 rpm, to achieve adequate conditions for the electrical generator.

Before testing a new oil, the gearboxes were flushed with a complete oil charge of the new oil to test in order to avoid any contamination. The oil reservoir and the injection system are completely flushed and drained. A new filter cartridge was installed for each oil tested.

Table 3
PT100 location in the each gearbox.

PT100 code	Position
T_1	Rolling bearing carrier
T_2	Rolling bearing hole shaft
T_3	Rolling bearing ISS
T_4	Rolling bearing ISS
T_5	Rolling bearing HSS
T_6	Rolling bearing HSS
T_7	Oil sump
T_8	Ambient

Table 4
Operating conditions for the torque loss tests performed.

Step	Generator Speed [rpm]	Power [kW]	Step time [min]	Total time [min]
1	1620	1	60	60
2	1620	280	30	90
3	1620	560	30	120
4	1620	850	90	210

5. Experimental results and discussion

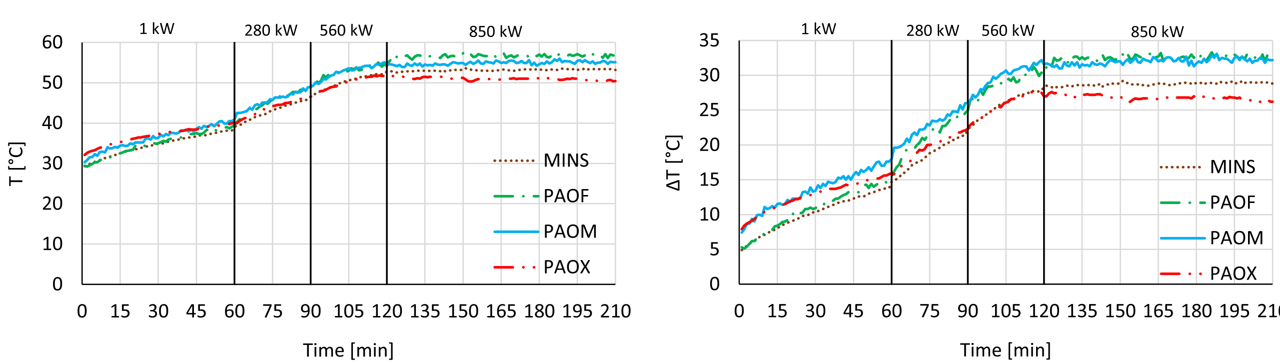
5.1. Power loss

The power loss was measured during 30 min at 280 kW and 560 kW, while at full power, 850 kW, the power loss was measured during 90 min, as described in Table 4. The temperatures measured are displayed in Fig. 5 showing that stabilized operating temperature only occurred under full power operating conditions, i.e. after 120 min of test and 30 min after oil start flowing through the heat exchanger (see Table 4). At the beginning of the 850 kW load stage the oil lubrication circuit is open and the oil starts to flow through a heat exchanger.

The 280 kW and 560 kW test stages were performed in transient conditions, which needs further study in order to understand if the power loss measurements are directly linked with gearbox temperatures.

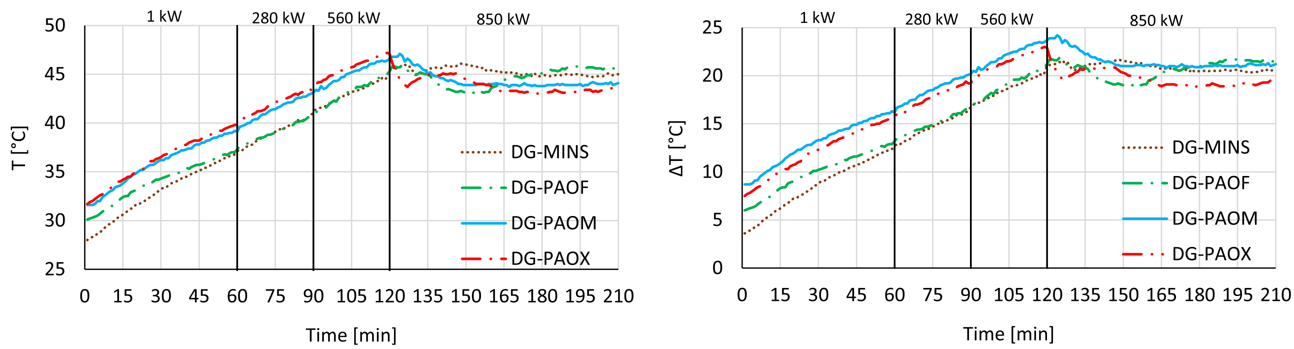
Fig. 6a presents the oil temperature of the driving gearbox during each test performed and it is important to remind that driving gearbox is lubricated with a mineral oil. It is interesting to verify that the oil operating temperature of the driving gearbox under 850 kW tests is similar for all lubricants with a difference smaller than 3 °C which can be explained by measurement precision and differences in heat transfer conditions due to air humidity and ambient temperature. The stabilization temperature presented in Fig. 6b has a similar behaviour.

Fig. 7 shows the difference between the oil temperature of test gearbox (lubricated with different wind turbine gear oils) and the oil temperature of driving gearbox (always lubricated with a reference mineral oil – MINV).



(a) Oil temperature (T_7 sensor). (b) Oil stabilization temperature ($\Delta T = T_7 - T_8$).

Fig. 5. Temperatures recorded with T7 sensor of the test gearbox for each oil formulation.



(a) Oil temperature (T_7 sensor). (b) Oil stabilization temperature ($\Delta T = T_7 - T_8$).

Fig. 6. Temperatures recorded with T7 sensor of the driving gearbox lubricated with MINV.

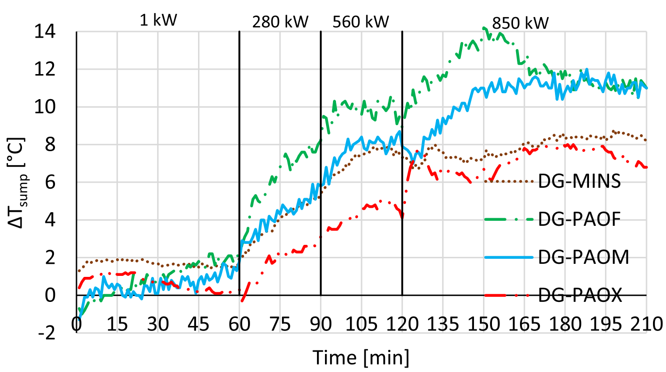


Fig. 7. Oil sump stabilization temperature ($\Delta T = T_7^{test} - T_7^{driving}$) for each oil formulation.

Table 5
Maximum operating temperature [°C].

P_{IN} [kW]	MINS	PAOF	PAOM	PAOX
280	45.8	48.7	48.9	46.3
560	52.2	54.5	55.1	51.7
850	53.4	56.8	55.1	50.4

Based on the maximum operating temperature measurements, displayed in Table 5, it is clear that there are differences up to 6 °C between lubricants at full power (850 kW) operating conditions.

Table 6 displays the operating viscosity at the maximum operating temperature. The differences observed have significant impact on the lubricants viscosity and consequently on churning losses, with PAOX displaying 44.2 cSt larger viscosity than PAOF (22.7%) for the 850 kW operating conditions.

Fig. 8 displays the power loss measured for each lubricant (sum of both gearboxes power losses plus the motor and generator power losses), showing that PAOX promotes about 7% reduction in comparison to PAOF and around 5% in comparison to MINS.

5.2. Test rig efficiency

The test rig efficiency for each tested lubricant is presented in Fig. 9, where a global efficiency between 72% and 93% is observed for the tested operating conditions. The figure clearly shows that for low input power the gearboxes have a very low efficiency due to the low oil operating temperature.

Fig. 10 shows the test rig efficiency variation when replacing the mineral oil (MINS) by the PAO formulations. For the most usual wind

Table 6
Operating kinematic viscosity [cSt].

P_{IN} [kW]	MINS	PAOF	PAOM	PAOX
280	228.2	213.0	212.2	235.5
560	161.8	165.9	163.1	184.2
850	152.1	150.9	152.2	195.1

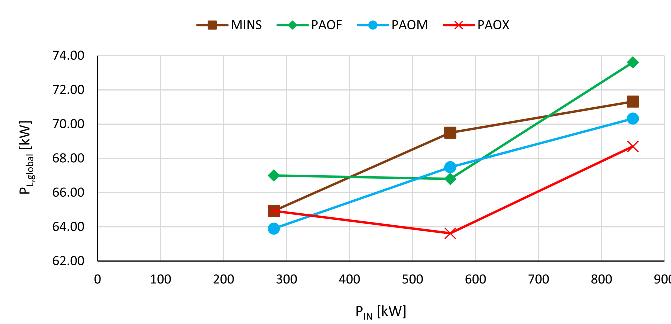


Fig. 8. Test rig power loss for each wind turbine gear oil.

turbine gearbox operating conditions, i.e. 560 kW (2/3 of full power capacity), the PAOX oil has an efficiency 1% higher than MINS.

5.3. Gearboxes efficiency

The global efficiency (η_{global}) was calculated based on the power loss measurements. Since motor (η_{motor}) and electrical generator ($\eta_{generator}$) efficiencies are known, 97.3% and 97.4% respectively, the driving ($\eta_{driving}$) and test gearbox efficiency (η_{test}) can be calculated according to the following equations:

$$\eta_{global} = \eta_{motor} \times \eta_{driving} \times \eta_{test} \times \eta_{generator} \quad (1)$$

$$\eta_{gearboxes} = \eta_{driving} \times \eta_{test} = \frac{\eta_{global}}{\eta_{motor} \times \eta_{generator}} \quad (2)$$

Fig. 8 displays the global power loss and Fig. 11 presents the calculated gearboxes power loss. It is very interesting to verify that the power loss of gearboxes decreases when the input power increases. Comparing with previous works [20,43], such situation can only be explained by a very high influence of the no-load losses, mainly for 280 and 560 kW tests. Since no-load losses are highly affected by oil's viscosity, from 280 kW test to 850 kW test it was observed a viscosity decrease over 30%, as displayed in Table 6. Such results show that PAOF and PAOM formulations promoted lower power loss than MINS for similar operating viscosity. PAOX is expected to be more affected by no-load losses

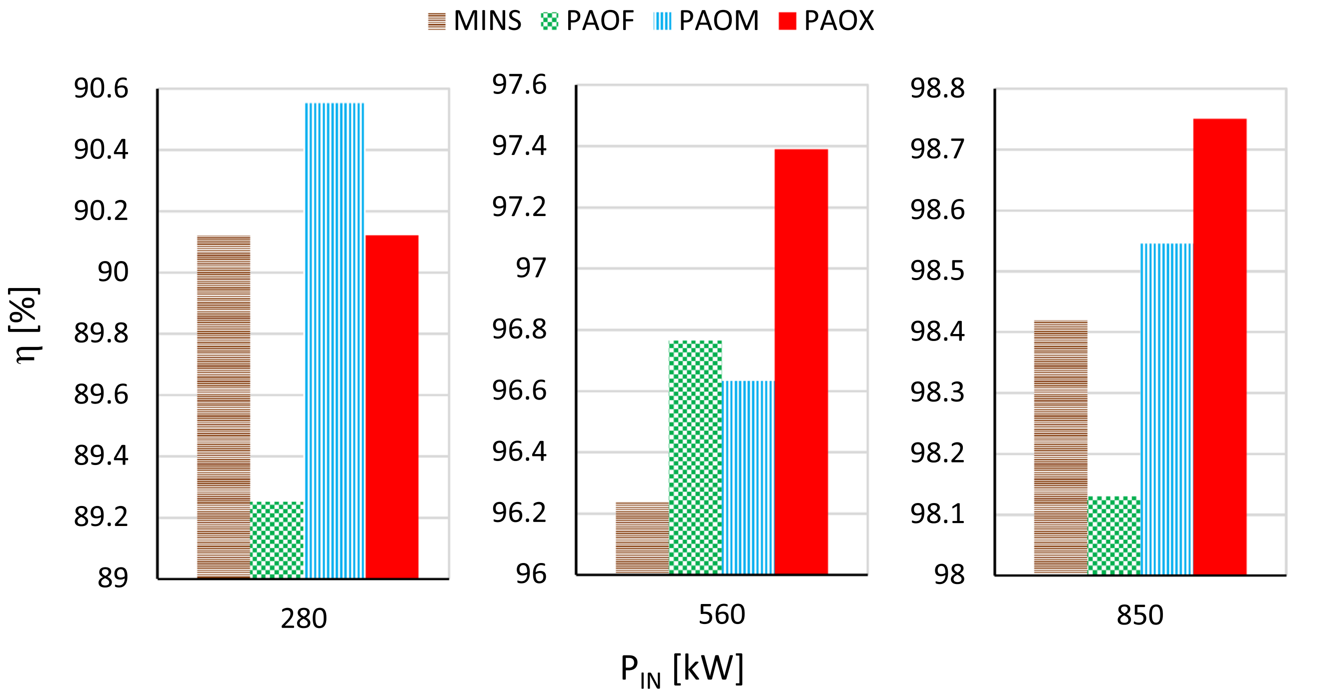


Fig. 9. Test rig efficiency with each wind turbine gear oil.

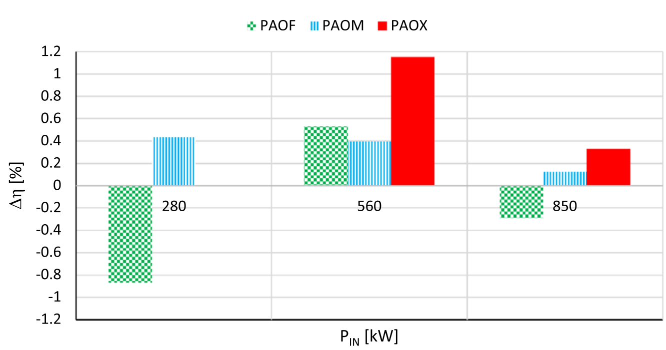


Fig. 10. Test rig efficiency variation by replacing the mineral oil by a PAO.

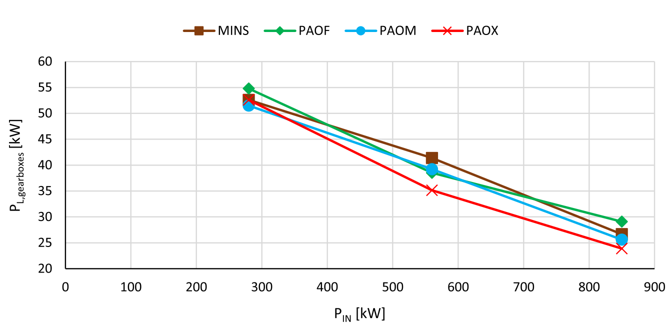


Fig. 11. Gearboxes power loss (driving and test gearbox) for each wind turbine gear oil.

due to its highest viscosity for all operating conditions, however, it presented the lowest power losses due to its expected lower coefficient of friction on the meshing gears, as shown previously in [39]. PAOM and PAOX were submitted to FZG efficiency tests in a previous work [39] which allowed to conclude that two distinct PAO formulations can promote quite different power loss performance. The results here presented show that PAOX has lower power loss than PAOM in almost every tested condition.

Table 7
Total energy consumption [kWh].

Energy source	MINS	PAOF	PAOM	PAOX
Test rig	482.5	486.3	482.0	473.6
Oil pump	19.2	18.3	17.36	17.8

The results presented in Fig. 11 show that oil temperature selection is of fundamental importance for the efficiency of a wind turbine gearbox. Lower oil operating temperature can increase oil's life but it promotes higher no-load losses. So, to overcome the high no-load losses the selection of an adequate oil operating temperature (viscosity) is mandatory.

5.4. *Total energy consumption*

During each test the energy provided to overcome the power losses of the test rig system was measured on the electricity meter. The energy dissipated by the oil pumps was provided by other source being measured in other electricity meter. The measurements are presented in Table 7. It is clear that MINS and PAOM had similar power consumption. PAOX energy consumption was the lowest one, 1.8% lower than for MINS and 2.6% lower than PAOF.

6. Condition monitoring

6.1. *Particle counting*

The particle counting analysis was performed on collected lubricant samples, of fresh lubricant and used lubricant collected after the test, show relevant differences between lubricants. PAOX, promoted the lowest particle quantity, no matter the particle size considered as presented in Table 8. The results show that particle concentration is 20 times higher for PAOF and 10 times higher for PAOM than for PAOX.

Table 8
Particle counting for each wind turbine gear oil before and after test (not available for MINS).

Particle size	Unit	PAOF		PAOM		PAOX	
		New	Used	New	Used	New	Used
> 4 μm	[ppm]	1175	21,354	886	11,776	1245	1602
> 6 μm	[ppm]	385	5154	234	2017	358	487
> 14 μm	[ppm]	48	124	27	63	37	57
> 21 μm	[ppm]	21	25	11	19	21	13
> 38 μm	[ppm]	0	0	0	0	0	0

Table 9
Inline average oil cleanliness during each load step, according to ISO 4406.

Oil	280 kW	560 kW	850 kW
MINS	24/19/10	23/17/10	20/14/9
PAOF	18/16/12	18/16/12	18/15/11
PAOM	19/17/13	18/16/13	17/16/12
PAOX	14/11/7	14/12/7	15/12/8

Lower particle concentration will reduce the bearing and gear indentations and will contribute to increase the reliability of the gearbox.

6.2. Oil cleanliness

The oil cleanliness was measured inline according to ISO 4406 during each test and the results are presented in Table 9. The oil cleanliness correlates very well with particle counting of each lubricant presented previously. No matter the test condition, PAOX presented the better oil cleanliness, that is lower than the required value during service specified by ISO 4406, i.e. 18/16/13 [45]. The other PAO formulations still meet the ISO 4406 specification but MINS does not. The filtering capacity of PAOX was far better than the other tested products.

7. Conclusions

The results achieved with this work showed that:

- The total power loss is lower for PAOX, a polyaplhaolephin base oil with plastic deformation additives.
- The power loss measurements are consistent with oil temperature measurements, i.e. PAOX promoted also the lowest oil temperature under stabilized conditions.
- Under conventional operating conditions, i.e. 2/3 of wind turbine full power capacity, changing from a mineral oil to a PAO increases the efficiency by more than 0.3%. If PAOX is used the gearbox efficiency increase surpasses 1%.
- No-load losses are dominant at low operating temperatures (45–55 °C).
- The use of a PAO formulation, besides promoting lower power losses also promotes lower particle counting and better oil cleanliness ISO codes.
- The oil analysis is in agreement with inline oil cleanliness measurements. Regarding oil performance, PAOX promotes lower quantity and smaller wear particles.
- The results show that, despite the bigger price of a PAO in comparison to a mineral oil, both efficiency and oil cleanliness are improved using the PAO oil.

Acknowledgements

The authors gratefully acknowledge the funding supported by:

- National Funds through Fundação para a Ciência e Tecnologia (FCT), under the project EXCL-II/SEM-PRO/0103/2012;
- NORTE-01-0145-FEDER-000022 – SciTech – Science and Technology for Competitive and Sustainable Industries, cofinanced by Programa Operacional Regional do Norte (NORTE2020), through Fundo Europeu de Desenvolvimento Regional (FEDER);
- LAETA under the project UID/EMS/50022/2013;
- Iberdrola Renovables, Spain for providing the test gearbox.

References

[1] High efficiency planetary gearboxes for eco-power. World Pumps 2001;2001 (419):34–6. http://dx.doi.org/10.1016/S0262-1762(01)80327-8.
[2] Crawford R. Life cycle energy and greenhouse emissions analysis of wind turbines and the effect of size on energy yield. Renew Sustain Energy Rev 2009;13(9):2653–60. http://dx.doi.org/10.1016/j.rser.2009.07.008.
[3] Mostafaeipour A. Productivity and development issues of global wind turbine industry. Renew Sustain Energy Rev 2010;14(3):1048–58. http://dx.doi.org/10.1016/j.rser.2009.10.031.
[4] Lin Y, Tu L, Liu H, Li W. Fault analysis of wind turbines in China. Renew Sustain Energy Rev 2016;55:482–90. http://dx.doi.org/10.1016/j.rser.2015.10.149.
[5] Ozgener O, Ozgener L. Exergy and reliability analysis of wind turbine systems: a case study. Renew Sustain Energy Rev 2007;11(8):1811–26. http://dx.doi.org/10.1016/j.rser.2006.03.004.
[6] Hur S, Leithead W. Collective control strategy for a cluster of stall-regulated offshore wind turbines. Renew Energy 2016;85:1260–70. http://dx.doi.org/10.1016/j.renene.2015.07.087.
[7] Sarker BR, Faiz TI. Minimizing maintenance cost for offshore wind turbines following multi-level opportunistic preventive strategy. Renew Energy 2016;85:104–13. http://dx.doi.org/10.1016/j.renene.2015.06.030.
[8] McNiff BP, Musial WD, Errichello R. Variations in gear fatigue life for different wind turbine braking strategies. In: AWEA wind power 1990, Washington; 1990 (10 pp.).
[9] Lukyanov V. Surface roughness and parameters. Precis Eng 1983;5(3):99–100. http://dx.doi.org/10.1016/0141-6359(83)90001-6.
[10] McDade M. Gearbox reliability collaborative (GRC) failure database. In: NREL wind turbine tribology seminar; 2011.
[11] Jungk M. Update on the development of a full life wind turbine gear box lubricating fluid. In: NREL wind turbine tribology seminar; 2011.
[12] Sprei F, Karlsson S. Energy efficiency versus gains in consumer amenities – an example from new cars sold in Sweden. Energy Policy 2013;53:490–9. http://dx.doi.org/10.1016/j.enpol.2012.11.017.
[13] Team UK. UK team creates Ech2o fuel cell car for efficiency record bid. Fuel Cells Bull 2005;2005(9):7–8. http://dx.doi.org/10.1016/S1464-2859(05)70746-X.
[14] Fernandes CM, Hammami M, Martins RC, Seabra JH. Power loss prediction: application to a 2.5 MW wind turbine gearbox. Proc Inst Mech Eng Part J: J Eng Tribol 2015.http://dx.doi.org/10.1177/1350650115622362 URL: 〈http://pij.sagepub.com/content/early/2015/12/16/1350650115622362.abstract〉.
[15] Hohn BR, Michaelis K, Vollner T. Thermal rating of gear drives balance between power loss and heat dissipation; 1996.
[16] Martins R, Cardoso N, Seabra J. Gear power loss performance of biodegradable low-toxicity ester-based oils, Proc Inst Mech Eng Part J: J Eng Tribol 2008;222 (3):431–40.http://dx.doi.org/10.1243/13506501JET345.
[17] Martins R, Seabra J, Brito a, Seyfert C, Luther R, Igartua a. Friction coefficient in FZG gears lubricated with industrial gear oils: biodegradable ester vs. mineral oil. Tribol Int 2006;39(6):512–21. http://dx.doi.org/10.1016/j.triboint.2005.03.021.
[18] Martins R, Moura P, Seabra J. Power loss in FZG gears: mineral oil vs. biodegradable ester and carburizedsteel vs. austempered ductile iron vs. MoS2-Ti coated steel. VDI Berichte 2005;1904.2(1904 II):1467–86.
[19] Magalhes L, Martins R, Locateli C, Seabra J. Influence of tooth profile and oil formulation on gear power loss. Tribol Int 2010;43(10):1861–71. http://dx.doi.org/10.1016/j.triboint.2009.10.001.
[20] Fernandes CM, Marques PM, Martins RC, Seabra JH. Gearbox power loss. Part II: friction losses in gears. Tribol Int 2015;88:309–16. http://dx.doi.org/10.1016/j.triboint.2014.12.004.
[21] Eschmann P, Hasbargen L, Weigand K. Ball and roller bearings. Theory, design, and application. Munich: John Wiley and Sons; 1985.
[22] Skf. SKF general catalogue. SKF; 2003.
[23] Fernandes CMCG, Marques PMT, Martins RC, Seabra JHO. Gearbox power loss. Part i: losses in rolling bearings. Tribol Int 2014. http://dx.doi.org/10.1016/j.triboint.2014.11.017.
[24] Linke H. Stirnradverzahnung. Munich: Hanser Verlag; 1996.
[25] Ariura Y, Ueno T. The lubricant churning loss and its behavior in gearbox in cylindrical gear systems. J Jpn Soc Lubr Eng 1975;20(3).
[26] Mauz W. Hydraulische Verlute von Stirnradgetrieben bei Umfangsgeschwindigkeiten bis 60 m/s [Ph.D. thesis]. Dissertation Uni Stuttgart; 1987.

[27] Maurer J. Ventilationsverluste. FVA Forschungsvorhaben Nr. 44/VI Heft 1994;432.

[28] Changenet C, Velex P. A model for the prediction of churning losses in geared transmissions – preliminary results. J Mech Des 2007;129(1):128–33. http://dx.doi.org/10.1115/1.2403727.

[29] Changenet C, Leprince G, Ville F, Velex P. A note on flow regimes and churning loss modeling. J Mech Des 2011;133(12):121009. http://dx.doi.org/10.1115/1.4005330.

[30] Marques PMT, Fernandes CMCG, Martins RC, Seabra JHO. Power losses at low speed in a gearbox lubricated with wind turbine gear oils with special focus on churning losses. Tribol Int 2013;62:186–97. http://dx.doi.org/10.1016/j.triboint.2013.02.026.

[31] Mare B. Tooth friction loss in simple planetary gears. Technology 2007:153–60.

[32] Csobán A, Kozma M. Performance of heavy-duty planetary gears 73–8, 6th International Multidisciplinary Conference – Scientific Bulletin 6th edition ISSN-1224-3264, Baia Mare, Románia, 9-20 May 2006, pp. 73–79.

[33] Csobán A, Kozma M. Influence of the oil churning, the bearing and the tooth friction losses on the efficiency of planetary gears. Strojniski Vestn/J Mech Eng 2010;56(4):231–8.

[34] Höhn BR, Michaelis K. Influence of oil temperature on gear failures. Tribol Int 2004;37(2):103–9. http://dx.doi.org/10.1016/S0301-679X(03)00047-1.

[35] Fernandes CM, Martins RC, Seabra JH. Friction torque of cylindrical roller thrust bearings lubricated with wind turbine gear oils. Tribol Int 2013;59 (81107):121–8. http://dx.doi.org/10.1016/j.triboint.2012.05.030.

[36] Fernandes CMCG, Martins RC, Seabra JHO. Friction torque of thrust ball bearings lubricated with wind turbine gear oils. Tribol Int 2013;58:47–54. http://dx.doi.org/10.1016/j.triboint.2012.09.005.

[37] Fernandes CM, Amaro PM, Martins RC, Seabra JH. Torque loss in thrust ball bearings lubricated with wind turbine gear oils at constant temperature. Tribol Int 2013;66:194–202. http://dx.doi.org/10.1016/j.triboint.2013.05.002.

[38] Fernandes CMCG, Amaro PMP, Martins RC, Seabra JHO. Torque loss in cylindrical roller thrust bearings lubricated with wind turbine gear oils at constant temperature. Tribol Int 2013;67:72–80. http://dx.doi.org/10.1016/j.triboint.2013.06.016.

[39] Fernandes C, Martins R, Seabra J, Blazquez L. FZG gearboxes lubricated with different formulations of polyalphaolefin wind turbine gear oils. In: Velex P, editor. International gear conference 2014: 26th–28th August 2014, Lyon. Oxford: Chandos Publishing; 2014. p. 699–708. http://dx.doi.org/10.1533/9781782421955.699.

[40] Fernandes CMCG, Martins RC, Seabra JHO. Torque loss of type C40 FZG gears lubricated with wind turbine gear oils. Tribol Int 2014;70:83–93. http://dx.doi.org/10.1016/j.triboint.2013.10.003.

[41] Marques PMT, Fernandes CMCG, Martins RC, Seabra JHO. Efficiency of a gearbox lubricated with wind turbine gear oils. Tribol Int 2014;71:7–16. http://dx.doi.org/10.1016/j.triboint.2013.10.017.

[42] Fernandes CM, Marques PM, Martins RC, Seabra JH. Film thickness and traction curves of wind turbine gear oils. Tribol Int 2015;86:1–9. http://dx.doi.org/10.1016/j.triboint.2015.01.014.

[43] Fernandes CM, Marques PM, Martins RC, Seabra JH. Gearbox power loss. Part iii: application to a parallel axis and a planetary gearbox. Tribol Int 2015;88:317–26. http://dx.doi.org/10.1016/j.triboint.2015.03.029.

[44] Fernandes CMCG, Marques PMT, Martins RC, Seabra JHO. Influence of gear loss factor on the power loss prediction. Mech Sci 2015;6(2):81–8. http://dx.doi.org/10.5194/ms-6-81-2015.

[45] ISO, Hydraulic fluid power – fluids – method for coding the level of contamination by solid particles. ISO 4406:1999; 1999.