FZG Gearboxes Lubricated with Different Formulations of Polyalphaolefin Wind Turbine Gear Oils

Carlos Fernandes, R. Martins, J. Seabra and L. Blazquez

Five fully formulated wind turbine gear oils were characterized. The gear oils are 320 ISO VG grade and of different formulations, i.e. — four different polyalphaolefins and a mineral. A back-to-back FZG test rig with re-circulating power was used, with a torque-cell included on the rig to measure torque loss. Eight thermocouples monitored temperatures in different locations of the rig. Friction generated between the meshing teeth, shaft seals and rolling bearing losses was predicted.

Introduction

The generation of electricity by wind power is becoming more popular due to the concerns about the effects of global warming (Ref. 1). To make wind energy competitive with other power plants in the near future, enhancements on availability, reliability and lifetime will be required.

In our global economy it is mandatory to increase the efficiency of wind turbines, to reach the highest efficiency of gearbox drives, and their parts and to minimize power loss (Ref. 15). In order to increase gearbox efficiency it is important to quantify the main sources of power loss. The most common wind turbine gearboxes have planetary gears; the main losses occurring are: friction loss between the meshing teeth (Refs. 2-6); friction loss in the bearings (Refs. 2, 7); friction loss in the seals (Refs. 16, 24); no-load gear losses (Refs. 25-29); and energy loss due to air-drag (Ref. 8).

Friction generated between the meshing teeth is the main source of power loss in a gearbox when the torque transmitted is high (Ref. 30). On a gearbox with low transmitted torque, the friction due to viscous forces of the lubricant on the seals, gears and bearings must be accessed in order to correctly predict the power loss. The energy loss due to no-load mechanisms is highly dependent on the lubricant viscosity. The meshing teeth power loss is influenced by the oil formulation and also by their ability to promote a lubricant film while keeping low coefficient of friction.

A back-to-back FZG test rig was used to investigate the torque loss influence of five ISO VG 320, fully formulated, wind turbine gear oils. The operating temperatures of the FZG test rig were monitored in eight different spots with thermocouples. Tests at 1.13, 2.26 and 6.79 m/s (pitch line speeds) were performed for different FZG standard load stages: $K1$, $K5$, $K7$ and $K9$ (arm lever = 0.35 m). Both gearboxes were jet-lubricated with an oil flow of 3l/min. The oil jet input temperature was kept almost constant (80 ± 1°C).

A torque loss model will be presented and the coefficient of friction of each oil formulation will be determined.

Wind Turbine Gear Oils

In order to analyze the different gear oils suitable for the lubrication of wind turbine gearboxes, five fully formulated ISO VG 320 gear oils were selected. In between the selected gear oils, four PAO base oils can be found: PAOR, PAOM, PAOC and PAOX. A mineral-based oil (MINR) was also included as reference.

The FTIR analysis was used in order to identify some of the characteristic peaks of the lubricants (Fig. 1). In the PAO formulations both PAOR and PAOM — which have ester in the formulation to function as comptibilizer for the additives — show a peak in spectra for a wavenumber of 1,800. The PAOX is known to have molybdenum in the formulation while; both PAOC and PAOX are formulated without ester.

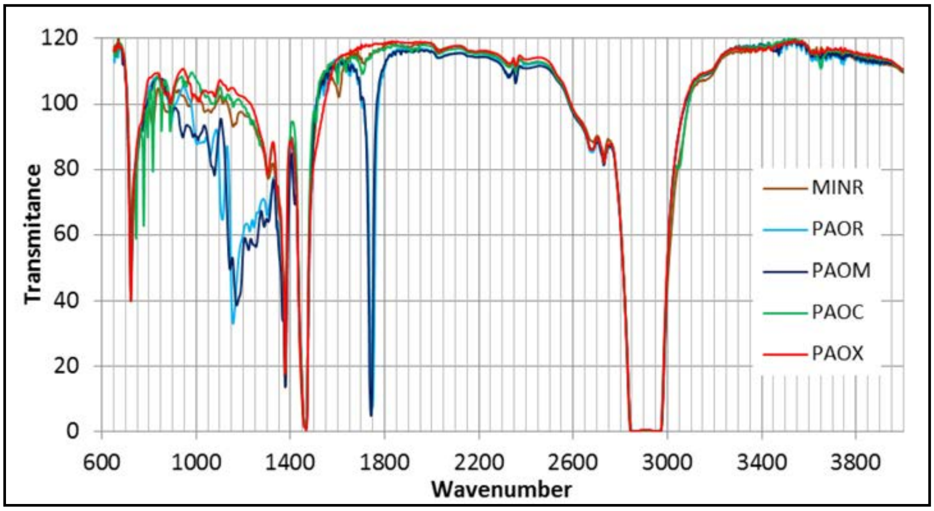

Figure 1 FTIR spectra presenting the transmittance of the wind turbine gear oils.

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Table 1 Physical and chemical characterization of wind turbine gear oils						
Parameter Base Oil	Unit [-]	MINR Mineral	PAOR	PAOM PAO	PAOC	PAOX
Chemical composition						
Zinc (Zn)	[ppm]	0.9	3.5	4	2	<1
Magnesium (Mg)	[ppm]	0.9	0.5	1	<1	<1
Posphorus (P)	[ppm]	354.3	415.9	448	385	400
Calcium (Ca)	[ppm]	2.5	0.5	6	<1	2000
Boron (B)	[ppm]	22.3	28.4		-	
Silicium (Si)	[ppm]	-		6	3	19
Molybdenum (Mo)	[ppm]	-			12	1150
Sulphur (S)	[ppm]	11200	5020	4436	6265	1800
Physical properties						
Density @ 15°C	[g/cm3]	0.902	0.859	0.863	0.861	0.855
Thermal expansion coefficient[/] (at.10-4)		-5.8	-5.5	-7.0	-7.4	-7.5
Viscosity @ 40°C	[cSt]	319.2	313.5	332.65	310.07	307.75
Viscosity @ 80°C	[cSt]	43.9	60.4	91.17	86.31	92.41
Viscosity @ 100°C	[cSt]	22.3	33.3	39.25	31.98	30.50
m	[/]	9.066	7.351	7.134	7.302	7.238
n	[/]	3.473	2.787	2.698	2.767	2.739
VI	[l]	85	150	159	152	150

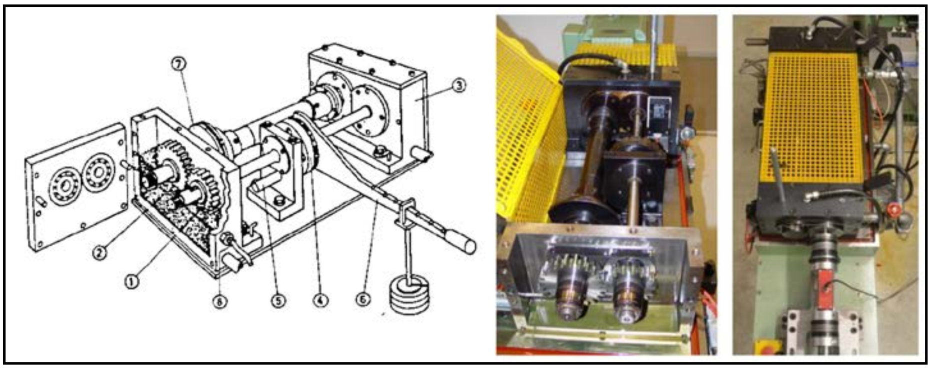

Figure 2 FZG test machine.

Table 1 displays the wind turbine gear oils' physical properties, as well as their chemical composition.

Test Rig

Figure 2 presents the FZG test rig used for this work; the gear test rig uses the recirculating power principle (Ref. 9). Note that the test pinion [1] and wheel [2] are connected to the drive gearbox by two shafts [3]. The shaft connected to the test pinion [1] is divided into two parts by the load clutch [4]. One half of the clutch can be fixed with the locking pin [5], whereas the other can be twisted using the load lever and different weights [6].

The torque loss (T_L) was measured using an ETH Messtechnik DRDL II torque transducer assembled on the FZG test machine. Operating temperatures on several points of the assembly were also measured and recorded using type-K thermocouples. The temperatures were recorded during each test with a sampling rate of 1 Hz.

Table 2 Geometric properties of the gears tested		
Gear Type	Type C40	
	Pinion	Wheel
Number of teeth	16	24
Module [mm]	4.5	
Centre distance [mm]	91.5	
Pressure angle [0]	20	
Face width [mm]	40	
Addendum modification [/]	+0.1817	+0.1715
Addendum diameter [mm]	82.64	118.54
Transverse contact ratio Ea Ul	1.44	
Total contact ratio Ey [/]	1.44	
Material	20 Mn Cr 5	
Ra [um]	0.7	

Drive and Test Gearboxes

Gears. The torque loss tests performed in this work used type-C gears with face width of 40 mm usually assembled on FZG drive gearboxes. Table 2 displays the main geometric properties of the C40 gears. The same C40 gearset was used for testing all the lubricants. To assure that a similar surface finish was used with all lubricants, the C40 gear was run-in during 48 hours under dip lubrication with a PAO 150 gear oil. The surface roughness was evaluated before and after the run-in period and the value of the average roughness can be found in the Table 2.

Rolling bearings and seals. The shafts on the test and slave gearbox are supported with cylindrical roller bearings NJ 406 MA. The rolling bearings have a dynamic load capacity of $C = 60.5$ kN and a static load capacity of $C0 = 53$ kN. The rolling bearings have an internal diameter of 30 mm and an external one of 90 mm.

The gearboxes are sealed with four Viton lip seals with an internal diameter of $d_{sh} = 30$mm. A Viton lip seal is also assembled on the drive gearbox motor shaft — $d_{sh} = 26$mm.

Test Procedure

The operating conditions used in the torque loss tests are displayed in Table 3. The tangential speed, power circulating in the system, tangential force transmitted by the gears, radial forces on the rolling bearings and maximum Hertzian pressure in the gears are also included. The oil volumetric flow was set to 3 l/min at a temperature of 80° C.

The test procedure can be summarized as follows:

1. Run load stage Ki and rotational speed condition (Table 3) during 3h according to test sequence presented

Table 3 Test conditions for tests performed							
		Wheel speed (rpm)					
		200	400	1200	Gears		Rolling bearings
FZG Load Stage	Wheel Torque (T_{s2}) (Nm)	Input power (W)			F_{bn} (N)	P_H (MPa)	F_r (N)
K1	4.95	103.7	207.3	622.0	97.5	179.75	49.5
K5	104.97	2198.5	4396.9	13190.9	2068.6	827.73	1049.1
K7	198.68	4161.2	8322.4	24967.2	3915.4	1138.8	1985.6
K9	323.27	6770.4	13540.9	40622.7	6370.7	1452.6	3230.7

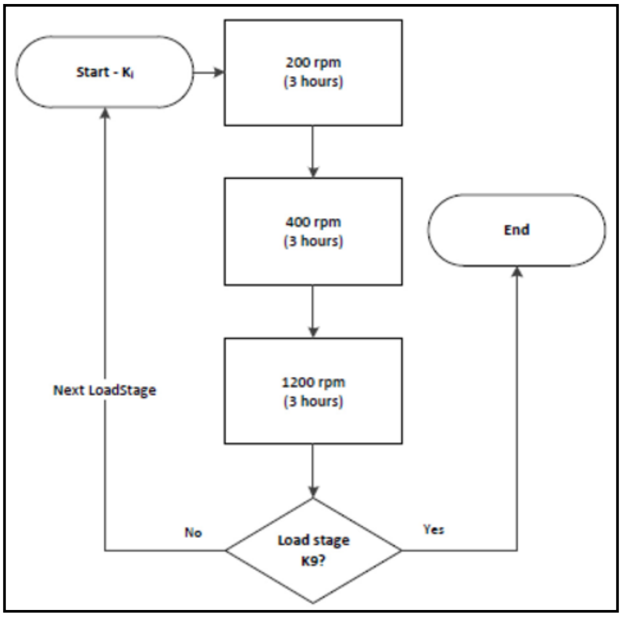

Figure 3 Test sequence.

in Figure 3

- Register the assembly working temperatures
- Continuous torque measurement with a sample rate of 1-measurement-per-second

2. Repeat procedure until the highest load stage

The values presented for torque loss and temperature are the average of the last 30 minutes of operation; i.e. — only the steady state operating conditions are considered for the average calculation. Between each oil test the gearboxes were flushed with solvent, the oil reservoir and the injection system completely drained and cleaned with a solvent.

Experimental Results

This section presents the results for the total torque loss measurements for all test conditions; Table 6 displays the torque loss (TL) measurements for all lubricants and test conditions.

Figure 4a displays the torque loss measured for load stage $K1$ at the input speeds of 200,400 and 1,200 rpm. These test conditions were performed to gather knowledge about the torque loss for a no-load condition, i.e. — the results presented are mainly driven by load-independent losses.

PAOC oil generated the lower friction torque loss when load stage $K5$ was applied — regardless of rotational speed selected. At 200 and 400 rpm the PAOM generated higher torque loss than the other PAOs. At 1,200 rpm the no-load losses of the PAGD oil are higher, resulting in the highest total torque loss generated (Fig. 4b). For the tests performed at load stage $K7$, the higher torque loss is again achieved for the PAOM oil. At low speed (200 and 400 rpm) the torque loss generated by PAOR is lower than other oil formulations. At 1,200 rpm all lubricants increase the torque loss. For load stage $K9$ the PAOR oil generated much lower torque loss than the other oils — mainly at lower speed. As speed increases, differences between the oils decrease. The MINR benefits due to the lubrication regime transition and the PAOs are penalized due to their higher viscosity generating higher no-load losses.

Gearbox Efficiency

The calculation of gearbox efficiency in a closed-loop test rig is a function of the static torque installed in the system and the torque applied by the driving motor (designated as torque loss in this work, T_L). A static torque was applied to the pinion shaft (T_{S1}); as a result, the wheel shaft has a higher torque (T_{S2}) related to the pinion torque by the transmission ratio ($i=Z2/Z1$), as represented by Equation 1. The wheel shaft torque values tested were already presented in Table 3.

$$T_{S2} = iT_{S1} \tag{1}$$

The torque loss (T_L) — or the torque applied by the electric motor — was measured on the wheel shaft; the efficiency of the test rig is given by Equation 2.

$$\eta_{Global} = \frac{T_{S2} - T_L}{T_{S2}} \times 100 \tag{2}$$

The test and slave gearboxes have the same gears, and so it is assumed
that both gearboxes have the same efficiency.Thus efficiency of the drive gearbox (η_D) is equal to the efficiency of

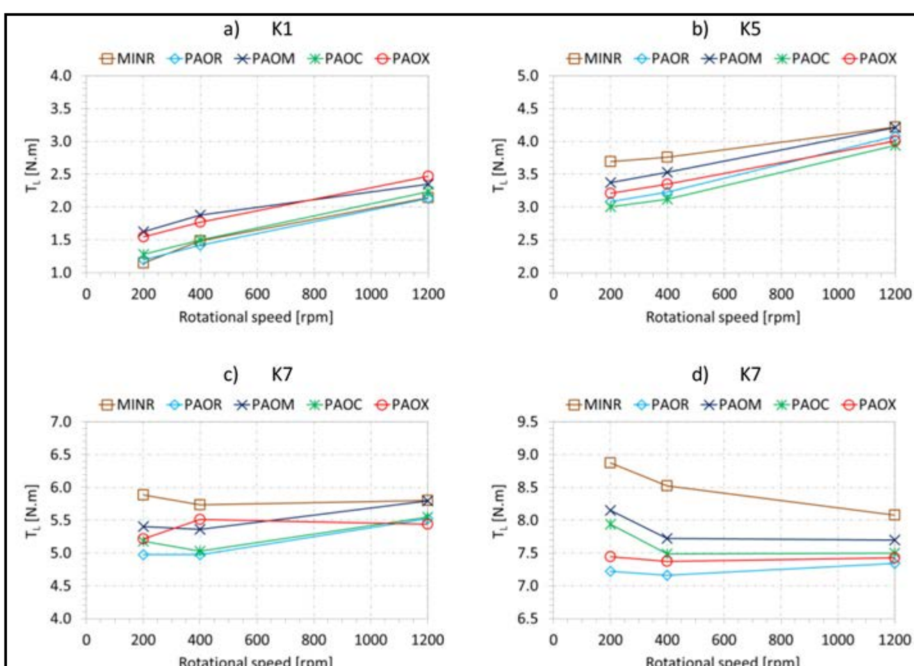

Figure 4 Total torque loss of FZG gearboxes lubricated with different wind turbine gear oils for *K1*, *K5*, *K7* and *K9* load stages.

Table 4 Efficiency values (%) calculated for test gearbox.

		MINR	PAOR	PAOM	PAOC	PAOX
	K1	87.63	87.08	82.00	86.08	82.94
200	K5	98.23	98.52	98.38	98.56	98.46
	K7	98.51	98.74	98.63	98.69	98.68
	K9	98.62	98.88	98.73	98.76	98.84
	K1	83.65	84.43	78.80	83.54	80.13
400	K5	98.19	98.45	98.30	98.50	98.39
	K7	98.55	98.74	98.64	98.73	98.60
	K9	98.67	98.89	98.80	98.84	98.85
	K1	75.30	75.50	72.52	74.10	70.80
1200	KS	97.97	98.04	97.97	98.10	98.07
	K7	98.53	98.60	98.53	98.59	98.62
	K9	98.74	98.86	98.80	98.83	98.84

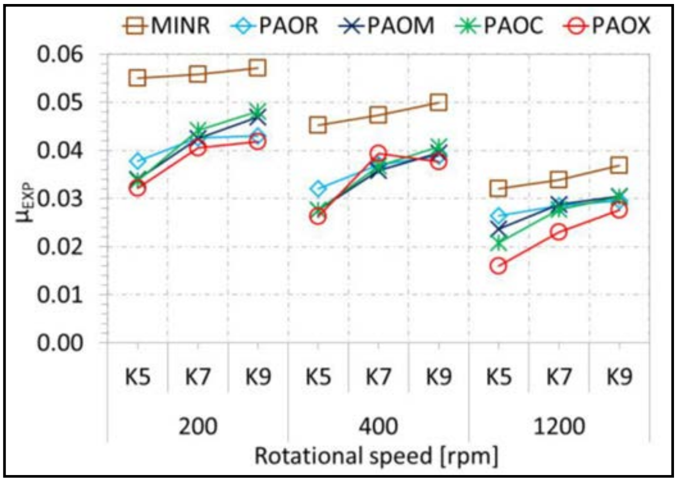

Figure 5 Coefficient of friction on meshing gears.

the test gearbox (η_T)—which is calculated according to Equation 3.

$$\eta_D = \eta_T = \sqrt{\frac{T_{S2} - T_L}{T_{S2}}} \times 100 \tag{3}$$

Table 4 displays the efficiency values calculated for a single gearbox (see Eq. 4 for all the lubricants tested).

Torque Loss Model

The torque loss model presented in this section was used on a previous work (Ref. 10). This model allows determination of the coefficient of friction on the meshing gears using the experimental results; it can be applied to any gear geometry tested in the test rig.

Gear no-load losses. No-load losses were determined for each input speed using the torque loss measured on load stage K1. The no-load losses are calculated subtracting the gear mesh losses, rolling bearing losses and seal losses from the total torque loss on load stage *K*1, as represented in Equation 4. The no-load losses remain equal for higher load stages.

$$T_{VD} = T_L^{K1} - T_{VZP}^{K1} - T_{VL}^{K1} - T_{VD}^{K1} \tag{4}$$

Note that because T_{VZP}^{K1} is very close to zero, the term can be disregarded.

Seal losses. The torque loss of the shaft seals is due to the friction between sealing lip and rotating shaft. Equation 5, proposed by Freudenberg (Ref. 11), was used. Note that this equation only accounts for shaft diameter, and so is independent of the oil used.

$$T_{VD} = 7.69.10^{-6} d_{sh}^{\,2} \cdot \frac{30}{\pi} \tag{5}$$

Rolling bearing losses. In order to understand the torque loss behavior of the rolling bearings, the model proposed by SKF (Ref. 7) was used. The total friction torque is the sum of four different physical sources of torque loss, represented by Equation 6.

$$T_{VL} = M'_{rr} + M_{S1} + M_{drag} + M_{seals} \tag{6}$$

The four sources of torque loss considered by the model are rolling torque (M'_{rr}), sliding torque (M_{sl}), drag torque (M_{drag}) and seals (M_{seals}). The rolling bearings used, NJ 406 MA, do not have seals, so seal losses were not a factor (see Refs. 10, 12 for further explanation).

The sliding coefficient of friction (μ_{sl}) used for each lubricant was determined experimentally, with the results published (Ref. 12) and presented in Table 5.

Table 5 Coefficient of friction for roller bearings

Oil	μ_{bl}	μ_{EHD}
MINR	0.035	0.018
PAO's	0.039	0.010

Meshing gears losses. Ohlendorf (Ref. 13) was first to introduce an approach for the load-dependent losses of spur gears. The torque loss generated between gear tooth contact can be calculated using Equation 7.

$$T_{VZP} = T_{IN} H_V \mu_{mZ} \tag{7}$$

H_V represents the gear loss factor that is calculated according to Equation 8 for spur gears.

$$H_V = \frac{\pi(u+1)}{Z_1 u \cos\beta_b}(1 - \varepsilon_\alpha - \varepsilon_1^2 - \varepsilon_2^2) \tag{8}$$

This formula assumes that the coefficient of friction (μ_{mZ}) is constant along the path of contact. The gear load losses for any load stage and input speed are calculated according to Equation 9.

$$T_{VZP}^{Ki} = T_L^{Ki} - T_{VL}^{Ki} - T_{VD}^{Ki} - T_{VZ0}^{Ki} \tag{9}$$

Coefficient of Friction in Meshing Gears

Schlenk (Ref. 14) proposed Equation 10 for the average coefficient of friction along path of contact. The lubricant parameter *XL* is equal to 1 for non-additivated (no additives) mineral oils.

$$\mu_{mZ} = 0.048\left(\frac{\frac{F_{bn}}{b}}{v_{\Sigma C}\rho_{redC}}\right)^{0.2} \eta^{-0.05} Ra^{0.25} X_L \tag{10}$$

The coefficient of friction was also derived from experimental results using Equation 11. Figure 5 shows the coefficient of friction for different operating conditions and different gear oils.

$$\mu_{EXP}^{Ki} = \frac{T_{VZP}^{Ki}}{T_{IN} H_V} \tag{11}$$

The lubricant parameter for each wind turbine gear oil was calculated and is presented in Table 6. The experimental coefficient of friction was used to calculate the lubricant parameter of Schlenck's Equation 10. The lubricant parameter is the value that minimizes the error between the experimental coefficient of friction calculated and the values achieved with the Schlenck equation.

Table 6 Lubricant parameter (*XL*) determined for each wind turbine gear oil

Oil	XL
MINR	0.858
PAOR	0.666
PAOM	0.680
PAOC	0.701
PAOX	0.628

Conclusions

- The results achieved showed that a PAO can reduce more than 10% the torque loss generated with a mineral oil (MINR).
- The model implemented suggests a better behavior of the PAOX on the meshing gears torque loss, which can be verified by the coefficient of friction generated.
- For example, the lubricant parameter used to score the oils in Schlenk equation, can have differences higher than 7% when compared with two different PAO, PAOC and PAOX.

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