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Gearbox power loss. Part II: Friction losses in gears

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ABSTRACT

The second part of the study presents an extensive campaign of experimental tests in an FZG test rig. An average coefficient of friction between meshing gears was devised from the experimental results. Several aspects regarding the meshing gears power loss are discussed such as gear loss factor, coefficient of friction and the influence of gear oil formulation (wind turbine gear oils).

1. Introduction

In the first part of this work, a rolling bearing torque loss model was calibrated for TBB (thrust ball bearings – 51107) and RTB (cylindrical roller thrust bearings – 81107 TN) and for several wind turbine gear oil formulations. The model was then applied with success to predict the torque loss in gearbox rolling bearings, in particular those used in the FZG machine slave and test gearboxes.

The second part of this work is dedicated to the analysis of the friction torque loss in helical gears lubricated with the same wind turbine gear oils used in Part I, under oil jet lubrication at 80 °C.

In this part, tests performed with a FZG test machine will be presented and discussed. The tests allow to validate the lubricant parameter determined for each wind turbine gear oil and presented in [1].

The authors calculate the meshing gear power loss considering an average coefficient of friction along the path of contact. For such situation the gear loss factor have a significant influence in the quantification of meshing gears power loss. So, the gear loss factor will be discussed and a method validated for a wide range of gear geometries will be also presented and discussed.

2. Gearbox power loss model

The power losses occurring in a gearbox are generated through different mechanical sources [2]. In this work the loss sources that were considered are represented in Fig. 1. Take note that the gears losses are divided into load dependent (P_{VZP}) and load independent losses (P_{VZ0}).

2.1. No-load gears power loss

Depending on the input power and speed, lubricant characteristics, and gearbox design, the no-load gear power losses usually are a very important source of energy dissipation. Due to an almost infinite combination of gearbox design choices and operating conditions, it is very difficult to develop a simple and general formulation to evaluate these power loss mechanisms.

Since the main objective of this work was to measure and predict accurately the friction torque loss in the meshing gears, the no-load gear power loss was determined experimentally for each operating speed and gear oil formulation, at the operating temperature of 80 °C, using a special testing procedure.

The overall torque loss in the slave and test gearboxes of the FZG machine were measured at very low input torque (FZG load stage 1) and for a wide range of operating speeds. Under these conditions the friction power loss in the meshing gears was assumed to be null. Thus, for any input torque (load stage i) the overall power loss is given by the following equation:

$$P_V^i = P_{VZ0}^i + P_{VZP}^i + P_{VL}^i + P_{VD}^i \quad (1)$$

For load stage 1 (low input torque, $T_W = 4.95$ Nm) Eq. (1) becomes

$$P_V^1 = P_{VZ0}^1 + P_{VZP}^1 + P_{VL}^1 + P_{VD}^1 \quad (2)$$

The term P_V^1 is determined experimentally at load stage K1.

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Nomenclature

a	axis distance (mm)
b	tooth face width (m)
H_V	gear loss factor (–)
l^i	length of contact of a tooth (mm)
m	module (m)
p_0	Hertz pressure (centre of the contact) (MPa)
p_b	transverse pitch (mm)
R_a	arithmetic average roughness of pinion and gear (μm)
T_L	torque loss (Nm)
u	gear ratio (z_2/z_1) (–)
v_t	pitch line velocity (m/s)
x	addendum modification (–)
z	number of teeth (–)
α	pressure angle
α_p	piezoviscosity
β	helix angle
ϵ_α	transverse contact ratio (–)
ϵ_β	overlap contact ratio (–)
ν_{oil}	oil kinematic viscosity at operating oil sump temperature (mm^2/s)
μ_{mZ}	coefficient of friction on meshing gears (–)

As the concept suggests, the no-load gears losses are independent of the load which gives the following equation:

$$P^i_{VZ0} = P^1_{VZ0} = P_{VZ0}, \quad \forall i \tag{3}$$

For load stage 1 it was assumed that

$$P^1_{VZP} \approx 0 \tag{4}$$

since the corresponding meshing torque loss (T^1_{VZP}) at the operating speed is lower than the precision of the torque cell (ETH Messtechnik DRDL II) used to measure the overall torque loss in the slave and test gearboxes of the FZG machine.

The power loss in the rolling bearings (P^1_{VL}) of the slave and test gearboxes are calculated using the model developed in Part I of this work.

The power loss in the seals is evaluated using Eq. (5) given in Ref. [4] by Freudenberg and is independent of the load applied:

$$P^i_{VD} = P^1_{VD} = P_{VD}, \quad \forall i \tag{5}$$

Finally, for any load stage 1 Eq. (1) becomes

$$P^1_V = P_{VZ0} + P^1_{VL} + P_{VD} \tag{6}$$

Thus with the Eq. (7) is possible to determine the no-load gears loss (P_{VZ0}),

$$P_{VZ0} = P^1_V - P^1_{VL} - P_{VD}. \tag{7}$$

2.2. Load dependent power loss in meshing gears

Ohlendorf [5] introduced an approach for the load dependent losses of spur gears. The power loss generated between gear tooth contact can be calculated according to Eq. (8),

$$P_{VZP} = P_{IN} H_V \mu \tag{8}$$

where H_V represents the gear loss factor which is determined according to Eq. (9), and assuming that the coefficient of friction (μ_{mZ}) is constant along the path of contact. In fact, this is a simplification of the problem.

2.2.1. Gear loss factor (H_V)

Eq. (8) can be used to calculate the average friction power loss between gear teeth, given the correct gear loss factor H_V. Despite

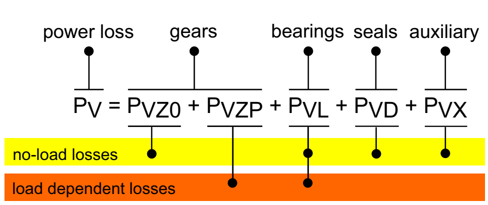

Fig. 1. Power loss contributions [3].

considering β_b, Eq. (9) initially proposed by Ohlendorf [5] is mostly valid for spur gears [1]:

$$H^{ohl}_V = (1+u)\frac{\pi}{z_1}\frac{1}{\cos\,\beta_b}(1-\epsilon_\alpha+\epsilon_1^2+\epsilon_2^2) \tag{9}$$

The load between gear teeth along the meshing line can be calculated just considering that the load per unit of length is given by Eq. (10), the coefficient of friction is assumed constant along the path of contact and the elastic effects were disregarded. With these conditions, the gear loss factor can be obtained by application of Eq. (11) proposed by Wimmer [6]:

$$F_N(x,y) = F_n \cdot \frac{1}{\sum_{i=1}^{n} l^i(x)} \tag{10}$$

$$H^{num}_V = \frac{1}{p_b}\int_0^b \int_A^E \frac{F_N(x,y)}{F_b}\cdot\frac{V_g(x,y)}{V_b}\,dx\,dy \tag{11}$$

Niemann and Winter [7] also proposed a gear loss factor that is shown in the following equation:

$$H^{Nie}_V = (1+u)\frac{\pi}{z_1}\frac{1}{\cos\,\beta_b}\epsilon_\alpha\left(\frac{1}{\epsilon_\alpha}-1+(2k_0^2+2k_0+1)\epsilon_\alpha\right) \tag{12}$$

Buckingham [8] also introduced a formula for the efficiency of a meshing gear pair. A gear loss factor (Eq. (13)) can also be derived from this approach:

$$H^{Buc}_V = (1+u)\frac{\pi}{z_1}\frac{1}{\cos\,\beta_b}\epsilon_\alpha\left(2k_0^2-2k_0+1\right) \tag{13}$$

Velex et al. which did no *a priori* assumption on tooth load distribution by using generalized displacements, in order to calculate the efficiency of a meshing gear pair, obtained a closed form solution for the efficiency of a meshing gear pair (constant coefficient of friction was assumed), as presented in Eqs. (14)–(16).

$$\rho = 1-\mu\cdot(1+u)\cdot\frac{\pi}{z_1}\cdot\frac{1}{\cos\beta_b}\cdot\epsilon_\alpha\cdot\Lambda(\mu) \tag{14}$$

with

$$\Lambda(\mu) = \frac{2k_0^2-2k_0+1}{1-\mu\cdot\left(\dfrac{\tan\alpha_t\cdot(2k_0-1)-\dfrac{\pi}{z_1}\epsilon_\alpha\cdot(2k_0^2-2k_0+1)}{\cos\beta_b}\right)} \tag{15}$$

where

$$k_0 = \frac{z_1}{2\pi\cdot\epsilon_\alpha\cdot u}\left(\left(\left(\frac{ra_2}{rp_2}\right)^2\frac{1}{\cos\alpha_t^2}-1\right)^{1/2}-\tan\alpha_t\right) \tag{16}$$

It turns out that Eq. (13) suggested by Buckingham is an approximation of the one suggested by Velex and Ville [9] when the coefficient of friction $\mu \ll 1$.

The different formulations presented for the gear loss factor can predict significant different values for helical gears with profile shifts. The meshing gears power loss is clearly dependent on the formulation used for the gear loss factor as presented in Eq. (8).

2.2.2. Comparison of gear loss factors

In order to compare the different gear loss factor equations, H_V was calculated for eight different gear geometries, in which, spur, helical and "low" loss gears are included (see Table 1). The gear loss factor was also calculated based on the results obtained with the commercial software "*KissSoft*" which accounts for elastic effects.

Fig. 2 shows the comparison between the different gear geometries as a function of the k_0 parameter defined by Eq. (16). There are clearly two groups of results that diverge when $k_0 \geq 0.75$.

The H501 and H951 geometries were tested for power loss in an FZG test rig and the results are presented in Section 5 [10]. Changing from H501 to H951 resulted in a dramatic power loss reduction, which was attributed to the H951 gear geometry (everything but the gear geometry was kept the same). These experimental results suggest that the gear loss factor of the H951 must be lower that of the H501. The reason for such deviations between different formulas is due to the use of helical gears with profile shift.

The trends shown by the gear loss factors obtained with "*KissSoft*", Ohlendorf equation and Eq. (11) are in agreement with the experimental observations. The gear loss factors obtained with Eq. (11) are close to those obtained from the "*KissSoft*" computations. Aiming for simplicity and fast computing the gear loss factor was calculated using Eq. (11) which is the solution of Wimmer equation disregarding elastic effects.

3. FZG machine and operating conditions

Fig. 3 presents the FZG gear test machine used in this work. This gear test rig uses the re-circulating power principle [11]. 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 overall torque loss (T_L) in the test and slave gearboxes was measured using a ETH Messtechnik DRDL II torque transducer assembled on FZG test machine (see Fig. 3c).

For these tests the FZG slave gearbox was assembled with a C40 gear set and 4 cylindrical roller bearings (NJ 406 MA) while the test gearbox was assembled with a H501 or H951 helical gear set (see Table 1), 2 cylindrical roller bearings (NJ 406 MA) and 2 four-point contact ball bearings (QJ 308 N2MA) (see Table 2).

In a previous work [1], the torque loss of the slave gearbox was measured assembling both gearboxes with C40 gears and 4 cylindrical roller bearings (NJ 406 MA).

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

The test procedure shown in Fig. 4 can be summarized as follows:

1. Run load stage Ki and rotational speed condition (Table 3) during 3 h.
   - Register the assembly working temperatures.
   - Continuous torque measurement with a sample rate of 1 measurement per second.
2. Repeat the procedure till the highest load stage

The values presented for torque loss and temperature are the average of the last 30 minutes of operation, (steady state operating conditions reached after 2.5 h). Between each gear oil formulation tested the gearboxes were flushed with appropriate solvents. The oil reservoir and the injection system were completely drained

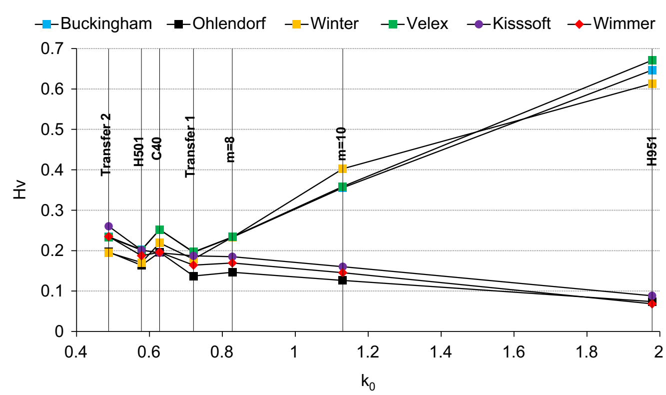

Fig. 2. Gear loss factor comparison with different formulas.

Table 1
Geometrical parameters of the gears.

Gears		Parameters										
		z [/]	m [mm]	a [mm]	α [°]	β [°]	b [mm]	x [/]	d_a [mm]	ϵ_α [/]	ϵ_β [/]	Ra [μm]
FZG C40	Pinion	16	4.5	91.5	20	0	40	+0.1817	82.64	1.44	0	0.7
	Gear	24						+0.1715	115.54			
FZG H 501	Pinion	20	3.5	91.5	20	15	23	+0.1381	80.37	1.45	0.54	0.3
	Gear	30						+0.1319	116.57			
FZG H 951	Pinion	38	1.75	91.5	20	15	23	+1.6915	76.23	0.93	1.08	0.3
	Gear	57						+2.0003	111.73			
Transfer Gearbox 1	Pinion	32	3.5	105.0	20	20	35	+0.3810	128.45	1.32	1.09	0.4
	Gear	23						+0.4150	95.17			
Transfer Gearbox 2	Pinion	28	4	95.0	20	20	33.5	−0.2400	125.22	1.49	0.91	0.4
	Gear	17						+0.0510	80.73			
Gearbox $m=8$	Pinion	17	8	355	20	9	124	+0.4965	160.74	1.40	0.77	–
	Gear	69						+0.3985	580.36			
Gearbox $m=10$	Pinion	19	10	500	20	9	175	+0.6500	222.65	1.32	0.87	–
	Gear	77						+0.8877	814.63			

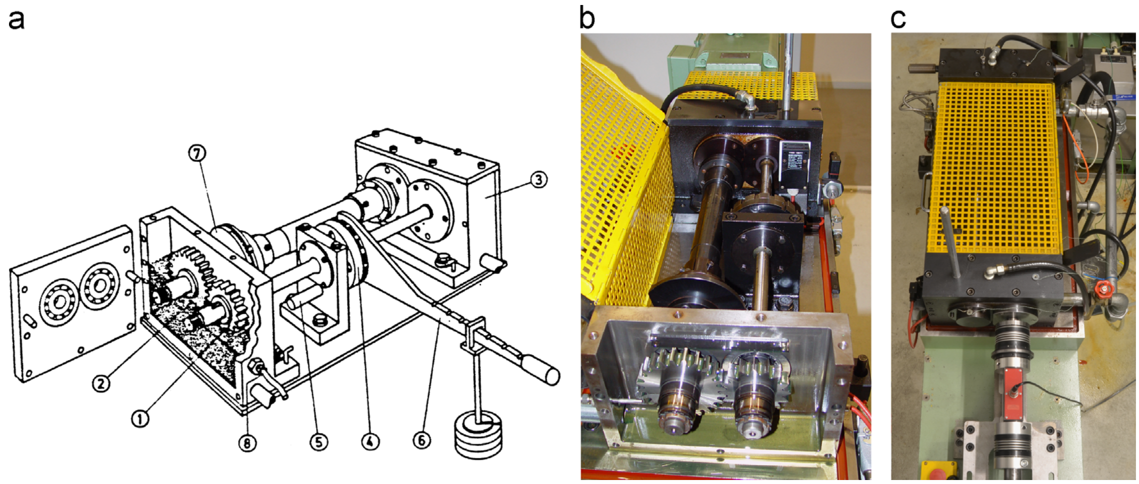

Fig. 3. FZG gear test rig: (a) Schematic view; (b) test gearbox and test gear; and (c) torque transducer.

Table 2
Configuration of FZG gearboxes tested.

Gearbox	Slave	Test
Gears	C40	H501
Rolling bearings	4 NJ 406 MA	2 NJ 406 MA+2 QJ 308 N2MA
Seals	2 viton seals ($d=30$ mm)+1 viton seals ($d=26$ mm)	2 Viton seals ($d=30$ mm)

Table 3
Operating conditions regarding the torque loss tests (H501 gear).

FZG load stage	Gears			Rolling bearings		Wheel speed (rpm)		
						200	400	1200
	Wheel torque (Nm)	F_{bn} (N)	p_H (MPa)	F_r (N)	F_a (N)	Input power (W)		
K1	4.95	100	171	37	24	104	207	622
K5	104.97	2128	787	777	518	2198	4397	13,191
K7	198.68	4027	1083	1471	980	4161	8322	24,967
K9	323.27	6553	1382	2393	1594	6771	13,541	40,623

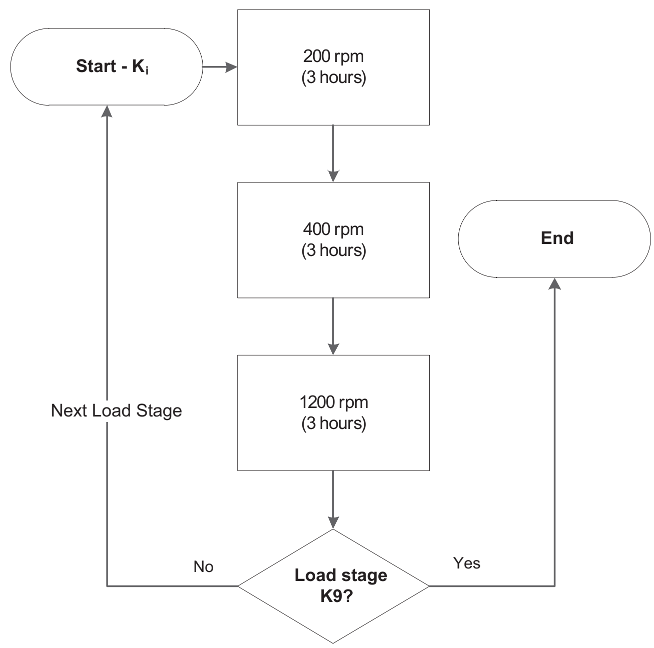

Fig. 4. Test procedure sequence.

and flushed with appropriate solvents (depending on the oil formulation).

4. Power loss with FZG C40/C40 gears

In a previous work [1] the power loss generated in an FZG test rig equipped with C40 gears, both in the driving and in the test gearboxes was characterized. The results are presented in Fig. 5.

4.1. Rolling bearings losses

The rolling bearing model presented in the Part I of this work was applied to the experimental results with C40/C40 gears. Using the corresponding μ_{bl} and μ_{EHD} determined on Part I of this work, the rolling bearing losses are presented in Table 4.

4.2. No-load gears power loss

Using the previous experimental results obtained in load stage 1 it is possible to estimate the no-load losses, according to the following equation:

$$P_{VZ0} = P_V^1 - P_{VL}^1 - P_{VD} \tag{17}$$

The corresponding results are presented in Table 5 for the wind turbine gear oils considered.

4.3. Average coefficient of friction

The friction power loss generated by the meshing gears can be calculated for any load stage according to Eq. (18), that is

$$P_{VZP} = P_V^{EXP} - (P_{VZ0} + P_{VL} + P_{VD}) \tag{18}$$

The friction power loss P_{VZP} generated by the meshing gears (Eq. (18)) inside the gearbox can be used to calculate an experimental average coefficient of friction (μ_{EXP}) for all gear meshes of the gearbox. The average coefficient of friction μ_{EXP} can be calculated according to the following equation:

$$\mu_{EXP} = \frac{P_{VZP}}{P_{IN} \cdot H_V^{num}} = \frac{P_V^{EXP} - (P_{VZ0} + P_{VL} + P_{VD})}{P_{IN} \cdot H_V^{num}} \tag{19}$$

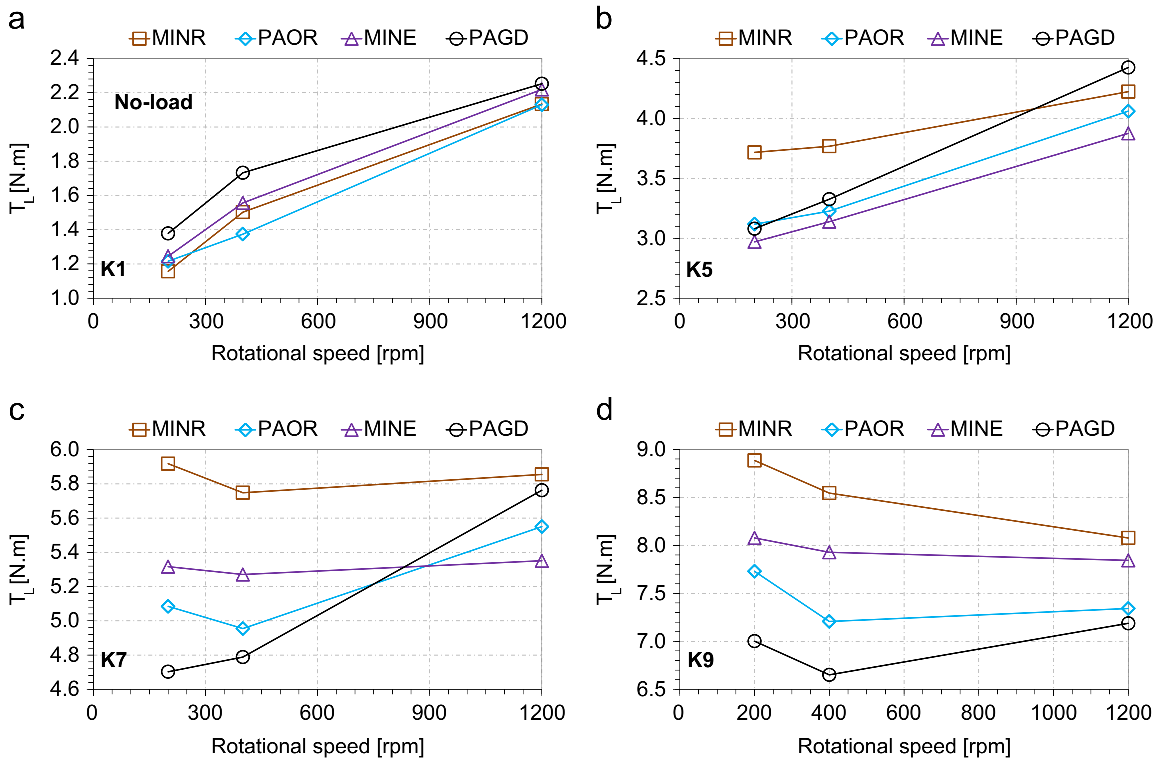

Fig. 5. Total torque loss results, C40/C40 gears: (a) K1; (b) K5; (c) K7; and (d) K9.

Table 4
Rolling bearing power losses [P_{VL}/W] for C40/C40 gears.

Operating condition	MINR	PAOR	MINE	PAGD
200				
K1	4	4	5	5
K5	10	12	12	13
K7	12	14	15	17
K9	15	17	18	19
400				
K1	11	14	14	16
K5	29	34	36	40
K7	36	42	44	49
K9	43	49	52	57
1200				
K1	76	87	90	96
K5	171	197	205	220
K7	206	236	245	265
K9	237	273	284	306

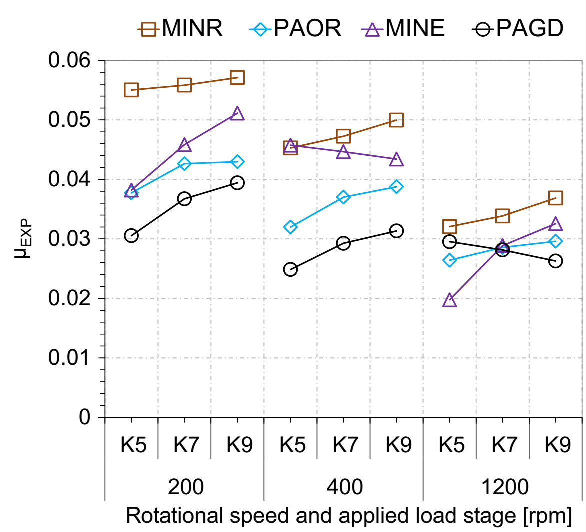

Fig. 6. Experimental coefficient of friction using C40 FZG gears.

Table 5
Overall no-load power loss in the FZG machine [P_{VZ0}/W] for C40/C40 gears.

Rotational speed (rpm)	MINR	PAOR	MINE	PAGD
200	12	13	13	16
400	35	30	35	41
1200	146	132	141	136

Table 6
XL lubricant parameter.

Oil	X_L	X_L [1]
MINR	0.846	0.89
PAOR	0.666	0.65
MINE	0.751	$0.5\left(\frac{F_{bt}/b}{\nu_{\Sigma C}\rho_{redC}}\right)^{0.1}$
PAGD	0.585	$0.5\left(\frac{F_{bt}/b}{\nu_{\Sigma C}\rho_{redC}}\right)^{0.05}$

Fig. 6 shows the coefficient of friction calculated using Eqs. (18), (19) and (11).

4.4. Lubricant parameter XL

Schlenk [12] proposed a formula for the coefficient of friction between gear teeth Eq. (20). This equation was derived from experiments in twin disk machines for non-additivated mineral oils. When different gear oil formulations are considered the X_L coefficient must be adjusted to account for the influence of

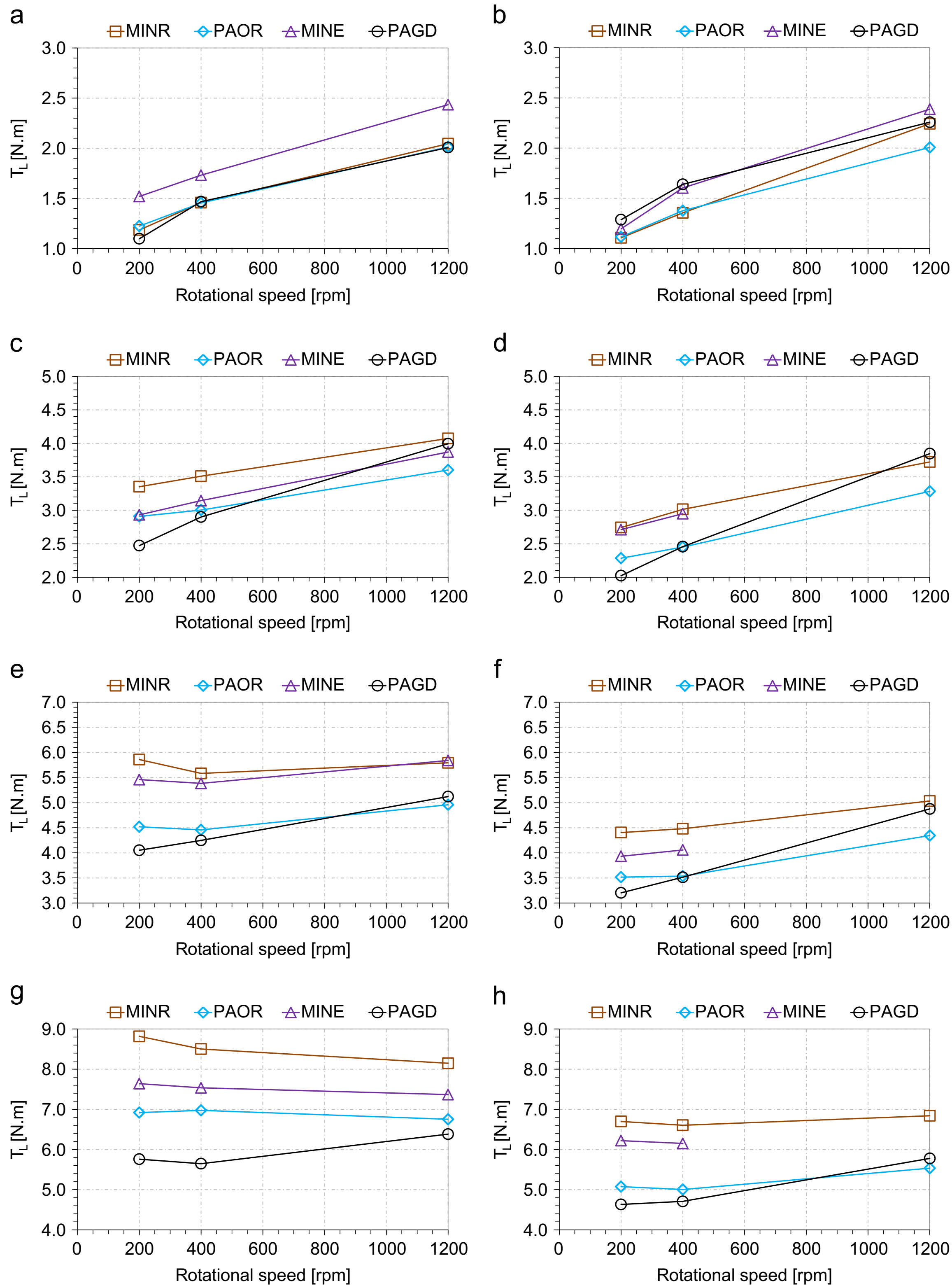

Fig. 7. Experimental results for K1 up to K9 load stages on FZG test machine with 501 and 951 low loss gears: (a) K1; (b) K1; (c) K5; (d) K5; (e) K7; (f) K7; (g) K9; and (h) K9.

different base oils and or additive packages:

$$\mu_{mZ} = 0.048\left(\frac{F_{bt}/b}{\nu_{\Sigma C}\rho_{redC}}\right)^{0.2}\eta^{-0.05}\,Ra^{0.25}\,X_L \tag{20}$$

The lubricant factor X_L was adjusted in such a way that the difference between the experimental and model values is minimized. Table 6 shows the adjusted lubricant factor for each one of the tested gear oils. These results are slightly different from the ones published before [1], because the values of μ_{bl} and μ_{EHD} for the calculation of the rolling bearing power loss were less accurate than the ones presented in Part I of this work.

Knowing the torque loss performance of the FZG drive gearbox standard gears allows the study of the torque loss performance for any configuration of the FZG test gearbox, including helical gears and/or angular contact rolling bearings.

From here on the meshing gears power loss can be calculated using Eq. (21), where the coefficient of friction is only dependent on the test conditions and lubricant parameter (XL) using the Schlenck formula to calculate the average coefficient of friction along the path of contact (Eq. (20)). The gear loss factor is the one calculated with Eq. (11):

$$P_{VZP} = P_{IN} H_V^{num} \mu_{mZ}. \quad (21)$$

5. Experimental torque loss results for helical gears

The experimental torque loss results are presented in Fig. 7 for gears H501 and H951, for load stages 1, 5, 7 and 9, and for several different wind turbine gear oil formulations.

It is important to note that all the oils operated with different kinematic viscosities at 80 °C. The oil properties can be found in the Part I of this work. For load stage K1, which was performed to gather knowledge about the no-load losses of the oils, the lubricants show similar results.

From load stages K5 up to K9, the torque loss at low speed is clearly influenced by the coefficient of friction of each oil. Under low speeds, the PAGD oil can reduce the torque loss in comparison with the other formulations. When the speed increase, the PAGD oil has a different behaviour, since its higher viscosity index seems to affect the high speed behaviour, i.e higher speed promoted a reduction of the effect of the coefficient of friction by means of no-load losses generated.

MINR, with much lower operating kinematic viscosity, generated much higher torque loss at lower rotational speeds. However, when speed increases the difference in performance in comparison with other formulations is much lower. The mineral oil with PAMA thickener always generated lower torque loss than the MINR oil.

The oil formulation behaviour seems to be the same described before, no-matter what is the gear geometry. In this particular, the gear geometry strongly influences the load dependent losses as can be stated by comparing Fig. 7c with d for K5, Fig. 7e with f for K7 and Fig. 7g with h for K9. The H951 gear was developed to promote lower meshing gear losses, reducing the module and increasing the number of teeth as just presented in other works [13,14]. The results show that the gear geometry can increase the gearbox efficiency.

It is very interesting to compare Fig. 7g and h. If we look for the MINR behaviour in Fig. 7g and the MINR behaviour in Fig. 7h, the torque loss at 200 rpm is reduced by 16%. If we do the same comparison for the MINR with H501 and PAGD with H951 the torque loss reduction is 29%.

6. Application of the model to helical gears

The model will be applied to predict the torque loss generated by the test gearboxes for the experimental results presented in

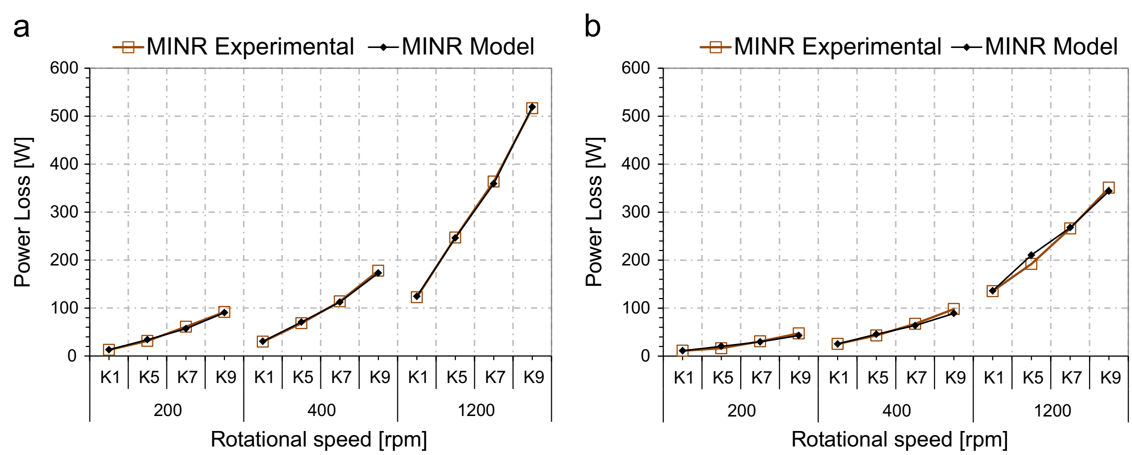

Fig. 8. Experimental vs. model results for K1 up to K9 load stages on FZG test machine with 501 and 951 low loss gears and MINR oil: (a) H501 and (b) H951.

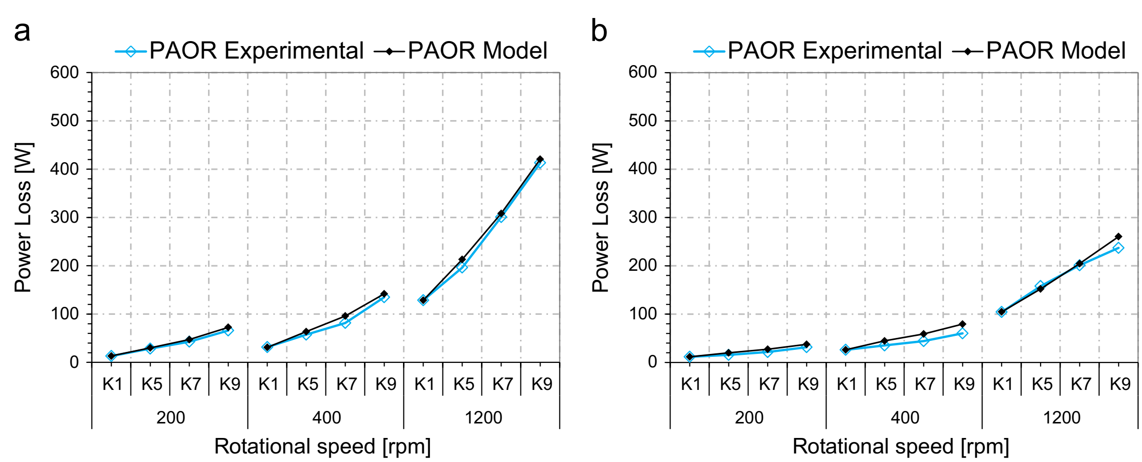

Fig. 9. Experimental vs. Model results for K1 up to K9 load stages on FZG test machine with 501 and 951 low loss gears and PAOR oil: (a) H501 and (b) H951.

Section 5. The power loss generated by the rolling bearings was calculated as presented in the first part of this work. The no-load losses were determined using Eq. (17).

The application of the model aims to assess the validity of the coefficient of friction that was determined (Eq. (19)), as well as the gear loss factor proposed by the authors. In the same way the prediction of rolling bearings torque loss is verified. In Eq. (20), the gear tooth width, *b*, was replaced by the minimum of the sum of the lengths of the contacting lines for the helical gears along the path of contact.

The model can accurately predict the power loss of the MINR oil, no matter the helical gear used, H 501 or H 951, with the results presented in Figs. 8 and 9 for PAOR.

The application of the model for the two different geometries shows the effect of the gear loss factor used, assuming that the rolling bearings power loss as well as the other sources are well estimated. Under this assumption the gear loss factor is reliable. The lubricant parameter (X_L) is also reliable to predict the actual power loss of the gears meshing teeth.

7. Conclusions

The FZG tests allow to draw the following conclusions regarding the oil formulations: PAGD generated the lowest torque loss no matter the load and the rotational speed tested and MINR generated the highest total torque loss. This behaviour was also observed in previous works [1,10].

The influence of the geometry was verified through tests with very different geometries in terms of module and the number of teeth for the same axis distance. H 951 with much higher number of teeth and consequent lower module generated much lower torque loss.

Combining oil formulation and gear geometry can promote very good savings of energy due to the increment of efficiency that can be up to 1%.

The power loss model that was proposed was used to predict the losses promoted by the tested gear oils at different load stages. The prediction is in agreement with the experimental results.

The model presented puts in evidence the importance of good prediction of rolling bearings power loss as discussed in the first part of this work.

Acknowledgments

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References

[1] Fernandes CM, Martins RC, Seabra JH. Torque loss of type c40 fzg gears lubricated with wind turbine gear oils. Tribol Int 2013; (0). doi:http://dx.doi.org/10.1016/j.triboint.2013.10.003.
[2] Höhn B-R, Michaelis K, Vollmer T. Thermal rating of gear drives: balance between power loss and heat dissipation. AGMA Technical Paper; 1996.
[3] Höhn B-R, Michaelis K, Hinterstoißer M. Optimization of gearbox efficiency. Goriva i Maziva 2009;48(4):462–80.
[4] Simrit: Radialwellendichtringe, Katalog Nr. 100.
[5] Ohlendorf H. Verlustleistung und Erwärmung von Stirnrädern [Ph.D. thesis], Dissertation. TU München; 1958.
[6] Wimmer AJ. Lastverluste von stirnradverzahnungen, Konstruktive einfüsse, wirkungsgradmaximierung, tribologie, Fakultät für Maschinenwesen der Technischen Universität München; 2006.
[7] Niemann G, Winter H. Maschinenelemente: Band 2: Getriebe allgemein, Zahnradgetriebe – Grundlagen, Stirnradgetriebe, Maschinenelemente/Gustav Niemann. Berlin, Heidelberg, New York: Springer; 1989.
[8] Buckingham E. Analytical mechanics of gears. New York: McGraw-Hill Book Co.; 1949.
[9] Velex P, Ville F. An analytical approach to tooth friction losses in spur and helical gears-influence of profile modifications. J Mech Des, Trans ASME 2009;131(10):1010081–10 [Cited By (since 1996) 11].
[10] Fernandes C, Martins R, Seabra J. Torque loss of spur and helical fzg gears lubricated with wind turbine gear oils. In: International conference on gears, Escuela Superior de Ciencias Experimentales y Tecnología. Madrid, Spain: Campus de Móstoles de la Universidad Rey Juan Carlos; 2013.
[11] Winter H, Michaelis K. Fzg gear test rig – description and possibilities. In: Coordinate European council second international symposium on the performance evaluation of automotive fuels and lubricants; 1985. p. 29–42.
[12] Schlenk L. Unterscuchungen zur Fresstragfähigkeit von Grozahnrädern [Ph.D. thesis]. Dissertation TU München; 1994.
[13] Höhn B, Michaelis K, Wimmer A. Low loss gears. AGMA Fall Technical Meeting Paper N 05FTM11; 2005.
[14] Aes LM, Martins R, Locateli C, Seabra J. Influence of tooth profile and oil formulation on gear power loss. Tribol Int 2010;43(10):1861–71 36th Leeds Lyon symposium special issue: multi-facets of tribology. http://dx.doi.org/10.1016/j.triboint.2009.10.001.