EFFICIENCY TESTS OF A TRANSFER GEARBOX: BIODEGRADABLE NON-TOXIC ESTER VS. MINERAL OIL

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ABSTRACT

Two industrial gear oils, a biodegradable non-toxic ester and a reference parafinic mineral oil, were compared in terms of gearbox wear and efficiency. Transfer gearbox efficiency tests were performed in a back-to-back gearbox test rig, for wide range of the operating conditions. Input and output power, as well as, oil and gearbox temperatures were recorded throughout the tests. Gearbox efficiency was determined and its dependence on input torque, speed and power was analyzed. Lubricant samples, collected during and at the end of the tests, were analyzed by Direct Reading Ferrography (DR3). The wear particles concentration (CPUC) and wear particles severity (ISUC) were determined in order to estimate the wear of the transfer gearbox.

1 INTRODUCTION

Environmental awareness leads to a growing interest in biodegradable non-toxic lubricants. Environmental compatibility is usually viewed in respect to biodegradability and toxicity. While the first issue is reached by using a suitable bio-degradable base fluid, low toxicity requires an additivation that is environmentally friendly, too [1].

However, lubricant performance (friction, wear, lifetime, load bearing, efficiency etc.) has a major impact on its overall environmental compatibility. Premature wear, high energy needs are as well harmful to the environment. So the key aspect for any industrial application is technical performance and technical advantages proved in dedicated tests.

The main objective of this work is to evaluate and compare the energetic efficiency [2-5] of two industrial gear lubricants, a reference ISO VG 150 mineral oil, containing an additive package to improve micropitting resistance, and an ISO VG 100 biodegradable fully saturated ester lubricant with a low toxicity additivation. Besides efficiency, the wear behaviour of the transfer gearbox was also analyzed for both lubricants using Direct Reading Ferrography (DRIII) measurements [6].

2 LUBRICANTS

The reference gear oil is based on a paraffinic mineral oil with significant residual sulphur content. It contains an ashless antiwear additive package based on phosphorous and sulphur chemistry and metal-organic corrosion preventives.

In contrast, the biodegradable product uses a fully saturated ester based on harvestable materials. The absence of unsaturated bonds in this base fluid leads to excellent thermal and oxidative stability. To combine the desired low toxicity with superior gear performance, environmentally compatible, highly efficient additives have been selected. Metal-organic compounds have been completely avoided.

The main properties of the two lubricants are shown in Table 1. The two oils were selected so that their kinematic viscosities are similar at 100 ºC.

The additive content of the mineral oil is considerably higher than that of the ester oil, mainly in what concerns the sulphur compounds.

Standardised, internationally recognized test methods are available for determining the biodegradability and environmental toxicity of lubricants and their components.

The "ultimate" biodegradability of lubricants is best assessed using a "ready" biodegradability test as published by the OECD and adopted by European Union.

The mineral oil didn't match the minimum requirements of 60% biodegradability in 28 days, as show in Table 1. Thus, no toxicity tests were performed for this lubricant.

The ester based oil exceeded the minimum requirements of 60% biodegradability in 28 days and pass both toxicity tests, OECD 201 "Alga growth inhibition test" and OECD 202 "Daphnia Magna acute immobilization" as show in Table 1.

3 GEARBOX

Fig 1 shows a cross section of the two speed gearbox used in this study. It is usually used as a transfer gearbox, mounted after the conventional gearbox of the vehicle, allowing the vehicle to have 2 drive axles (4 wheel drive) and an auxiliary power output.

This transfer gearbox uses 5 gears mounted in three shafts employing gibs. The gears mounted on the input and output shafts are supported by needle roller bearings. The gears are manufactured with DIN 15CrNi6 steel. After machining, the gears are case hardened, quenched in oil and annealed. The geometric character-ristics of the gears are given in Table 2.

The gearbox uses a total of 15 bearings: 4 roller bearings, 2 tapered roller bearings, 2 cylindrical roller bearings and 8 needle roller bearings.

Table 1 – Mineral and ester oil properties.

Parameter	Method	Desig.	Units	Lubricating Oils	
Base oil	DIN 51451	/	/	paraffinic mineral oil	fully saturated ester
Physical properties					
Density @ 15ºC	DIN 51757	ρ_{15}	g/cm^3	0.897	0.925
Kinematic Viscosity @ 40 ºC	DIN 51562	ν_{40}	cSt	146	99.4
Kinematic Viscosity @ 100 ºC	DIN 51562	ν_{100}	cSt	14.0	14.6
Viscosity Index	DIN ISO 2909	VI	/	92	152
Pour point	DIN ISO 3106		ºC	-21	-42
Wear properties					
KVA weld load	DIN 51350-2	/	N	2200	2200
Kva wear scar (1h/300N)	DIN 51350-3	/	mm	0.32	0.35
Brugger crossed cylinder test	DIN 51347-2	/	N/mm^2	68	37
FZG rating	DIN 513540	K_{FZG}	/	>13	>12
Chemical Content					
Zinc	ASTM D-4927	Zn	ppm	-	-
Calcium	ASTM D-4927	Ca	ppm	40	-
Phosphor	ASTM D-4927	P	ppm	175	146
Sulphur	ASTM D-4927	S	ppm	15040	180
Biodegradability and toxicity properties					
Ready biodegradability	OECD, 301 B		%	<60	≥60
Aquatic toxicity with Daphnia	OECD, 202	EL_{50}	ppm	-	>100
Aquatic toxicity with Alga	OECD, 201	EL_{50}	Ppm	-	>100

The transfer gearbox is filled up with 2.85 litters of lubricant oil, as recommended by the gearbox manufacturer.

4 GEARBOX TEST RIG

The transfer gearbox tests were performed in a back-to-back test rig with power re-circulating. Thus, the driving electrical motor only supplies the power needed to overcome the frictional and inertial losses and to reach and maintain the desired operating speed. Fig 2 shows a schematic view of the test rig.

This test rig was developed to allow the testing of different types of gearboxes. The test and the slave gearboxes are attached to adjustable platforms, and connected by the output torque and speed sensor (*Sen.2*), which is mounted on a mobile platform allowing the adjustment of its height and depth, as shown in Fig 2.

In order to close the loop and allow the re-circulation of power two gear sets are needed at each end connected by a long rear shaft.

The torque is applied by a hydraulic cylinder connected to a helical gear. When the hydraulic cylinder moves forward the helix angle of the helical gear twists the connecting gears and applies the desired torque.

The operating limits allowed by the test rig are up to 4000 rpm and 1400 Nm. Torque and speed applied to the input and output shafts as well as the oil and case temperatures are monitored permanently during the gearbox tests.

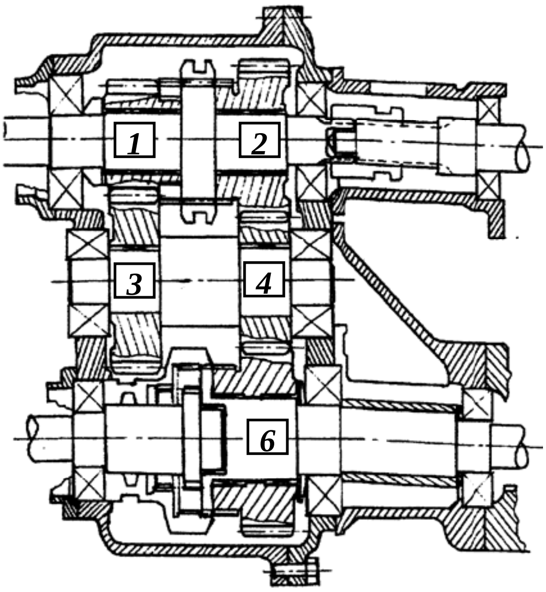


Fig 1 - Cross section of the two speed transfer gearbox.

Table 2 - Geometric characteristics of the transfer gearbox gears.

Parameter [units]	Design.	Gear wheel nº				
		1	3	2	4	6
Module [mm]	m	4	4	3.5	3.5	3.5
Number of teeth [/]	Z	17	28	27	23	32
Profile shift factor [/]	x	0.051	-0.24	0.161	0.415	0.381
Width [mm]	b	35	33.5	35	35	35
Pressure angle [º]	α	20	20	20	20	20
Helix angle [º]	β	20	20	20	20	20
Max. addendum diameter [mm]	da_{max}	80.7	125.2	108.4	95.3	128.6

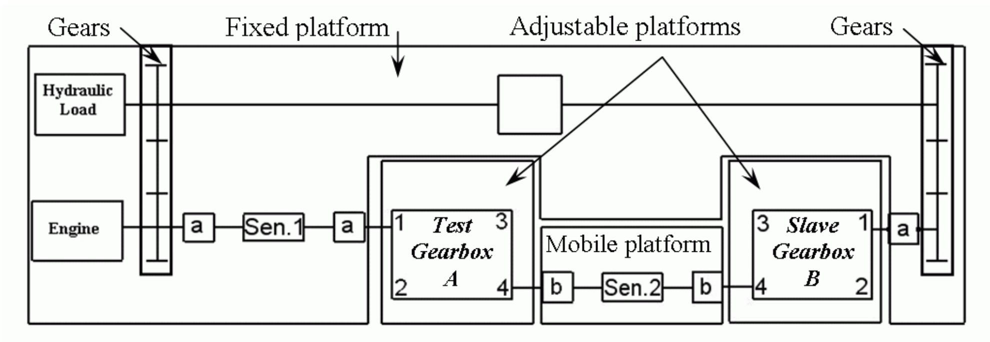


Fig 2 - Schematic view of the gearbox test rig.

5 EFFICIENCY TESTS

The main mechanisms of power dissipation inside a gearbox are the churning losses and the frictional losses [3, 7-9]. These are commonly called the no-load dependent losses and load dependent losses, respectively.

The churning losses are related with moving parts immersed in the lubricant and depend on the lubricant properties, speed and parts geometry. The main sources of churning losses are the gears, the bearings and the seals.

The frictional losses are related with contacting bodies in relative motion. The main sources of frictional losses are the gears and the rolling bearings. These losses are mainly dependent of the applied load, rolling and sliding speeds, lubricant properties and friction coefficients.

There are three main heat evacuation mechanisms: conduction, radiation and convection. These forms of heat evacuation depend mainly on the case temperature, the environment temperature, the case geometry and the coefficients that govern each one of these heat exchange mechanisms, such as gearbox surface emissivity, etc.

The thermal equilibrium of a gearbox is reached when the operating temperature stabilizes i.e. when the power dissipated inside the gearbox is equal to the heat evacuated from gearbox to the surrounding environment. The equilibrium temperature is dependent of the gearbox characteristics and of the lubricant properties [3, 7, 9], having the lubricant a capital influence in all kind of power losses, mainly the friction losses.

The higher the power losses the higher will be the equilibrium temperature of the gearbox. A lower stabilization temperature means higher efficiency, lower friction coefficient, smaller oil oxidation and longer oil life [2].

5.1 Test definition

The main objective of this study is to compare the energetic performance of two industrial gear oils considered: The reference mineral oil and a non-toxic biodegradable ester. The efficiency tests were designed for evaluating and compare the performance of two lubricants in near real operating conditions, considering wide ranges of the applied torque, speed and power.

Table 3 shows the operating conditions used for the transfer gearbox efficiency tests with both lubricants. Fig 3 shows the four levels of input power at which the gearbox efficiency tests have been performed in function of the input torque and speed.

The maximum contact pressure (Hertz pressure) attained in the contact between gear pair 1-3 and gear pair 4-6 is shown in Table 3 for all the tests. The maximum Hertz pressure changes from *0.84 GPa* to *1.88 GPa*.

Table 3 – Operating conditions in gear efficiency tests.

Test sequence	Input speed	Input torque	Input Power	**Max. Hertz Pressure** Pair 1-3	**Max. Hertz Pressure** Pair 4-6
[/]	[rpm]	[Nm]	[kW]	[GPa]	[GPa]
1	500	400	20.9	1.54	1.19
2	500	600	31.4	1.88	1.45
3	1000	200	20.9	1.09	0.84
4	1000	300	31.4	1.33	1.03
5	1000	400	41.9	1.54	1.19
6	1500	200	31.4	1.09	0.84
7	1500	267	41.9	1.25	0.97
8	1500	400	62.8	1.54	1.19
9	2000	200	41.9	1.09	0.84
10	2000	300	62.8	1.33	1.03

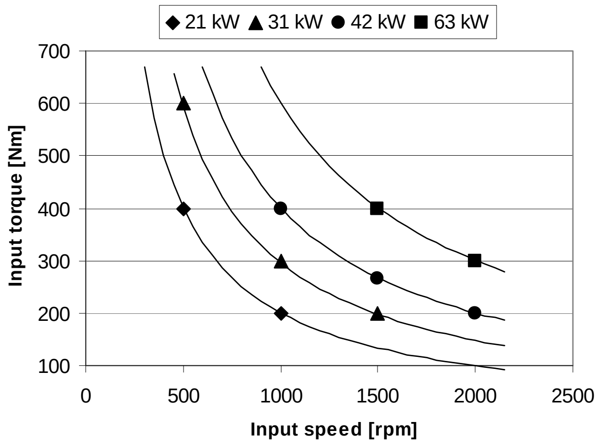


Fig 3 – Graphical representation of efficiency tests operating conditions.

The tests are made at constant operating conditions (input speed and torque) and each test lasts during 4 hours. The tests are initiated with an oil bath temperature near *30ºC* and the oil temperature increases freely along tests. A limit temperature of *120ºC* is established in order to protect gear from eventual failures, being the tests stopped if limit temperature is reached.

The test sequence is represented in Table 3. An oil sample is collected after each group of tests at the same speed. These lubricant samples are analysed by Direct Reading Ferrometry and by Analytical Ferrography, for evaluation of the wear particles produced during the tests.

5.2 Results

The results of these gearbox tests are presented below in terms of stabilization temperature, power losses and efficiency for both lubricants along all tests. The oil analysis results are also presented, showing the influence of each lubricant on gearbox wear.

5.2.1 Stabilization temperature

At constant operating conditions (speed, torque and ambient conditions) the lubricant that promotes the lowest oil bath temperature is the lubricant that also promotes the lowest power losses.

Fig 4 shows the stabilization temperature after each efficiency test for both lubricants. The equilibrium temperature increases if the input torque increases (at constant input speed) and it also increases if the input speed increases (at constant torque).

The ester lubricant displays the lowest stabilization temperature in all the operating conditions tested. This fact is emphasized on tests at *400 Nm* at *1500 rpm* and *300 Nm* at *2000 rpm* where the mineral oil reached the limit temperature of *120ºC* after *77 minutes* after *98 minutes*, respectively, while the ester oil accomplished the predetermined test time (240 minutes) and the oil temperature reached *109ºC* at *1500 rpm* and *115ºC* at *2000 rpm.*

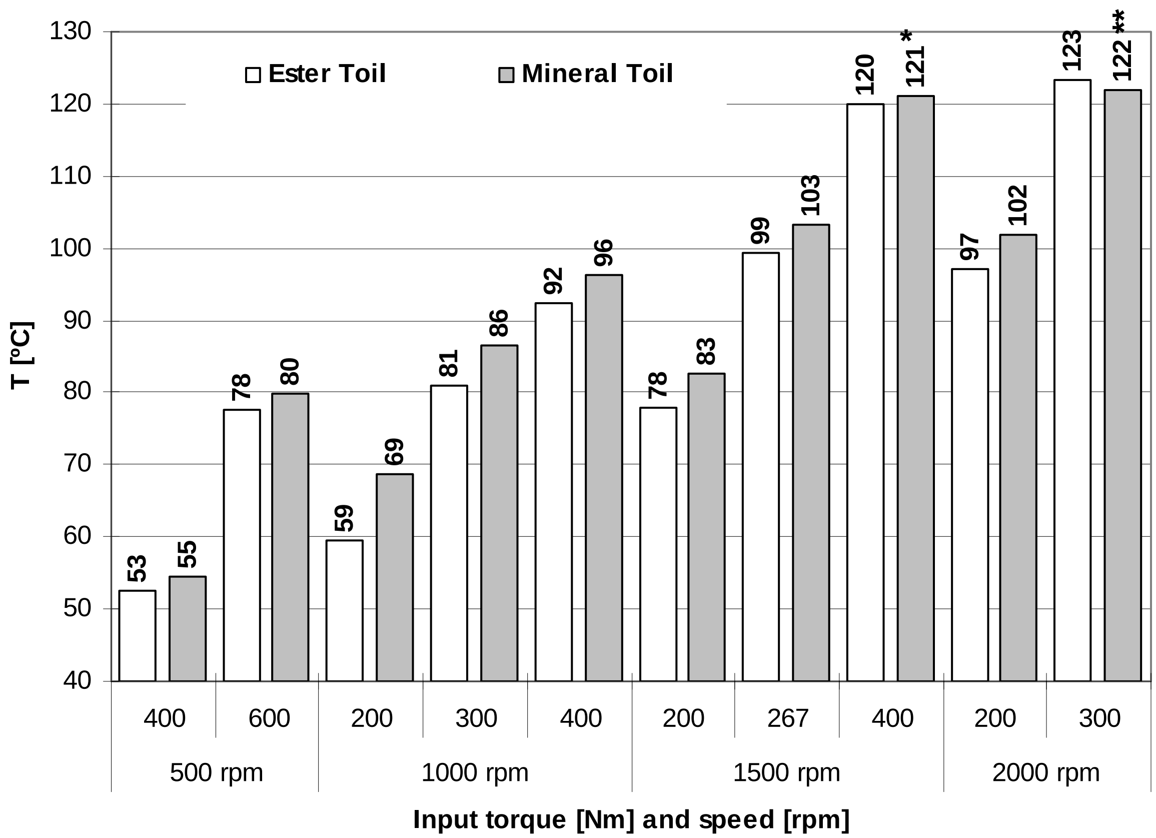


Fig 4 - Stabilization temperature in the efficiency tests for both lubricants.
- Test terminated after 77 minutes; ** - Test terminated after 98 minutes.

The reason for these differences lies on the smaller friction coefficient that is the most relevant factor at low speed and high torque conditions but is also due to the lower viscosity that also influences the churning losses at higher speeds (where its influence is higher) and lower torques that produce higher lubricant film thickness. The two lubricants have a similar viscosity near 90ºC, thus the mineral lubricant promotes higher churning losses at lower temperatures.

5.2.2 Power loss

The power loss figures represent the average power loss values for the last 30 minutes of test, after temperature stabilization (i.e. temperatures represented in Fig 4). The tests with the mineral lubricant at *400 Nm -1500 rpm* and *300 Nm - 2000 rpm* where the limit temperature was reached are also represented and the power loss correspond to the values measured on the last *5 minutes* (meaning that the power loss is lower than the obtained at stabilization temperature and the efficiency is higher).

Fig 5 displays the power loss curves at constant speed as function of input torque and Fig 6 displays the efficiency on the same conditions. These figures show that the ester lubricant always generates a smaller power loss and higher efficiency. For input speeds equal or above *1000 rpm*, the efficiency shows a minimum value for the input torque of *300 Nm*.

Fig 7 shows the power loss at constant torque as function of input speed and Fig 8 shows the efficiency for the same conditions, both clearly showing that the ester lubricant generates a smaller power loss than the mineral lubricant for all the tested conditions. The efficiency at constant torque always decreases when speed increases (the results with mineral oil at *400 Nm -1500 rpm* and *300 Nm - 2000 rpm* are not at stabilized temperature).

The increase of torque at constant speed promotes a fast grown of power losses due to the increase of contact pressure and to the decrease of film thickness. The increase of power losses on his turn increases the operating temperature that promotes a decrease of lubricant viscosity that will produce a decrease of film thickness that has a result an increase of friction.

The increase of speed at constant torque also promotes a fast increase of power losses due to the increase of churning losses besides the increasing of speed cause an increase of film thickness that reduces the friction losses, therefore the churning losses are a very important form of power dissipation on the transfer gearbox.

Fig 9 represents the power loss as function of input power and input speed. The power loss shows an evident increase with the increasing input power, being evident four degrees of power loss, corresponding to the four levels of input power tested.

Fig 10 displays the efficiency at constant input power as function of speed for both lubricants at stabilization conditions.

5.2.3 Oil analysis

The protection against wear provided by the two lubricants is also compared. The analysis of the wear particles contained in the lubricant gives quite good indication about the wear of lubricated parts, since those particles in a closed box have origin in the contacting parts.

The lubricant samples, collected after each group of tests at constant speed (*500 rpm*, *1000 rpm*, *1500 rpm* and *2000 rpm*), were analyzed by Direct Reading Ferrography in order to measure the ferrometric parameters ***Dl*** (large wear particles index) and ***Ds*** (small wear particles index). The values of ***Dl*** and ***Ds*** are used to evaluate the concentration of wear particles index - CPUC and the severity of wear particles index - ISUC, defined as

$$CPUC = \frac{Dl + Ds}{d} \text{ and } ISUC = \frac{Dl^2 - Ds^2}{d^2},$$

where d stands for the oil sample dilution.

The CPUC index grows when the sum of the small and the large particles increase, while the ISUC index grows when the number of particles with a size greater then 5μm (*Dl*) are present in major number than the smaller than 5μm (*Ds*).

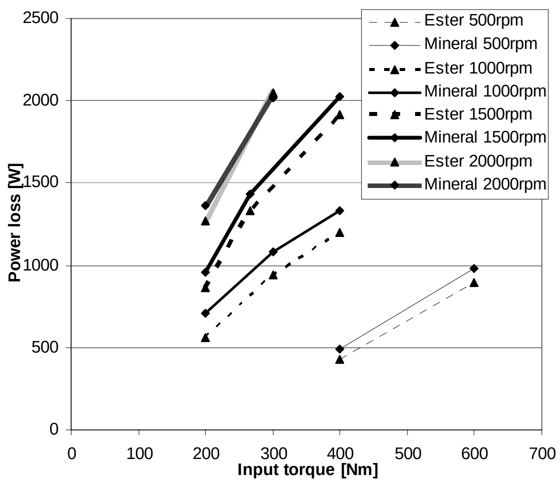


Fig 5 - Power loss curves at constant input speed v.s. input speed for both lubricants.

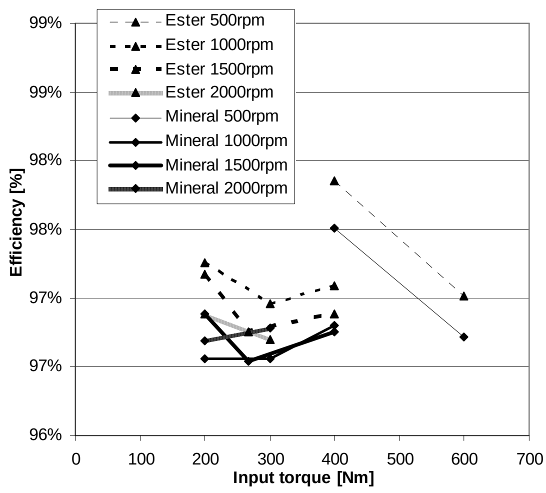


Fig 6 - Efficiency curves at constant input speed v.s. input speed for both lubricants.

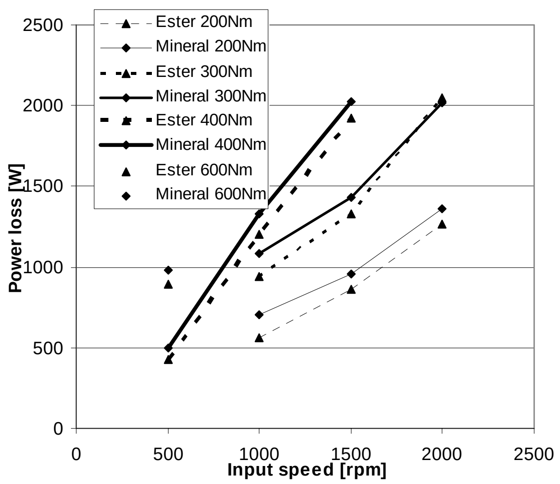


Fig 7 - Power loss curves at constant input torque v.s. input speed for both lubricants.

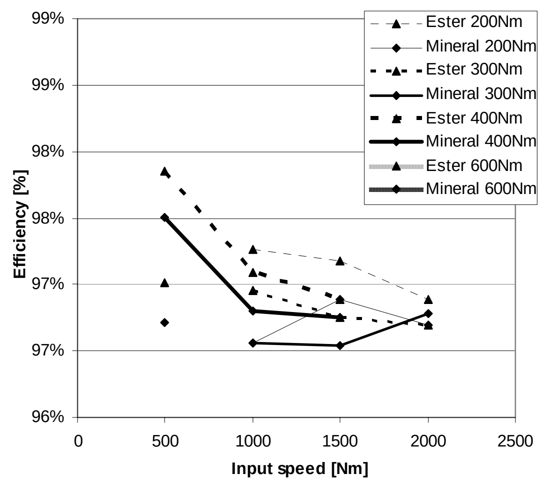


Fig 8 - Efficiency curves at constant input speed v.s. input speed for both lubricants.

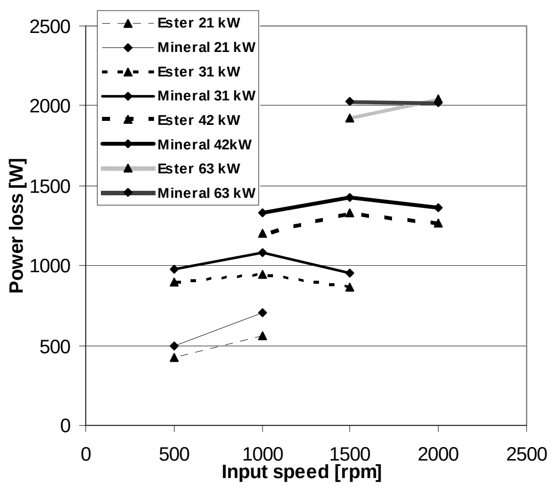


Fig 9 - Power loss curves at constant input power v.s. input speed for both lubricants.

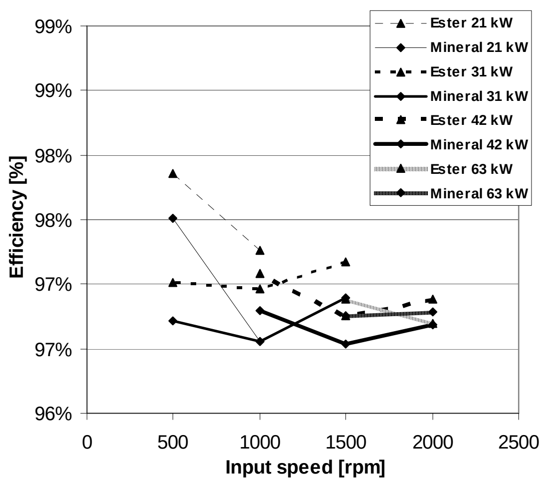


Fig 10 – Efficiency curves at constant input power v.s. input speed for both lubricants.

Fig 11 and Fig 12 show the evolution of ferrometric indexes CPUC and ISUC, respectively, during the gearbox efficiency tests for both lubricants. The sample of the mineral oil after the *500 rpm* tests had a problem and is not represented.

The concentration of wear particles is always larger in the tests with mineral lubricant, indicating a larger amount of particle generation.

The wear severity is also larger for the mineral lubricant, indicating that the wear particles are larger than the generated with ester lubricant. The test at 2000 rpm is an exception since the ester lubricant presents a larger increase of ISUC index. The evolution of both indexes shows a substantial increase on tests at *500 rpm* and *1000 rpm* and on the tests at higher speed the indexes are almost stabilized. This is due to the lowest film thickness on tests at low speed that promote higher contact.

The lubricant samples collected were also analysed by analytical ferrography, significant pictures are shown in Table 4. The ferrograms after the tests at *500 rpm* aren't displayed due to a sample problem.

After the tests at *1000 rpm* the mineral oil presents larger wear particles than the ester oil, beside it also presents a larger amount of small particles, as confirmed by Direct Ferrography results. The large fatigue particle on mineral lubricant ferrogram has *150x120* μm.

On the ferrograms of lubricant samples collected after the 1500 rpm tests the mineral oil also presents a larger amount of large and small wear particles. Some of those particles

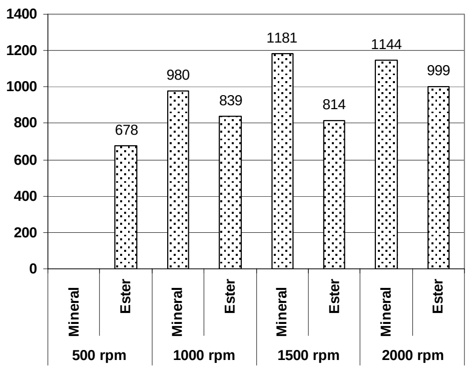


Fig 11 – Evolution of ferrometric index CPUC during efficiency tests for both lubricants.

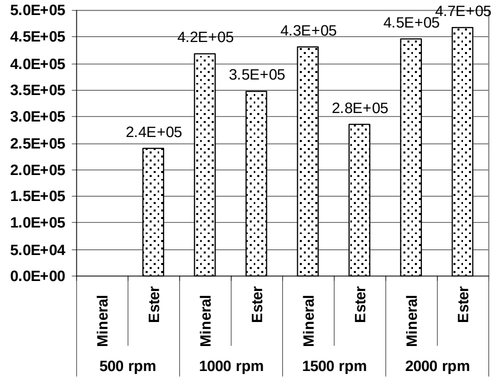


Fig 12 - Evolution of ferrometric index ISUC during efficiency tests for both lubricants.

particles still have very large dimensions, although on the ester oil ferrogram some large wear particles are also found. The ferrograms of lubricant samples collected after the tests at *2000 rpm* are very similar for both lubricants, both presenting some wear particles of large dimensions.

The ferrograms of ester lubricant displayed the presence of some matrix of friction polymers that aggregate very small wear particles. This friction polymers are visible on the ferrogram corresponding to the speed of *1000 rpm*, they have a circular or cylindrical shape and look shading.

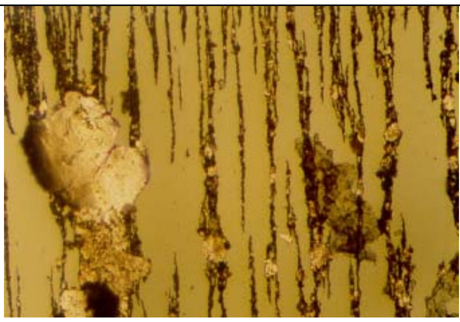
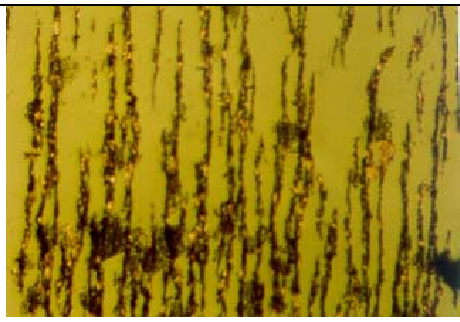
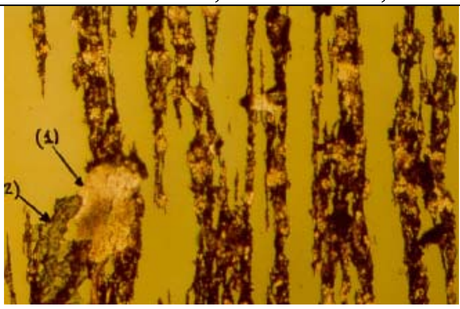
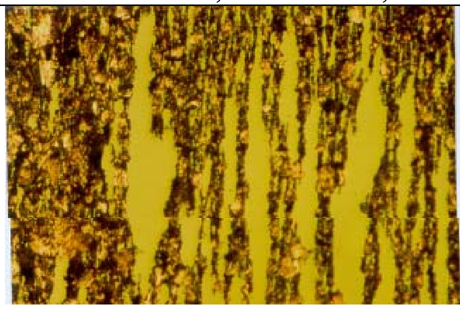
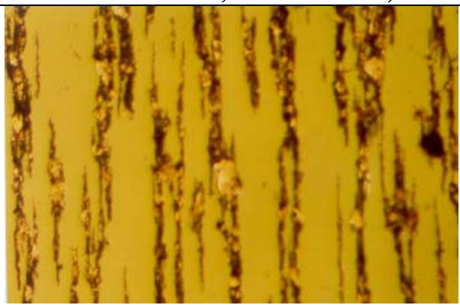
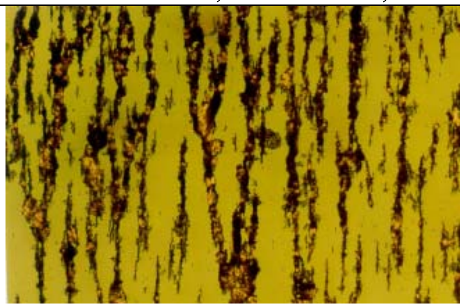
6 DISCUSSION

The efficiency tests showed that the mineral oil promotes a higher oil bath operating temperature for the same operating conditions. This difference in oil bath stabilization changes from *3%* to *14%*, i.e. between *2ºC* to *9ºC*. This difference is very important, since the mineral oil can not operate at so high temperatures as the ester lubricant, because it is very sensitive to oxidation at those temperatures, decreasing the oil life.

In what concerns power loss, the ester oil promotes a reduction between *70W* to *146W*, in comparison to the mineral oil, for the same operating conditions. This corresponds to a power loss reduction from *7%* to *21%*, in comparison to mineral lubricant.

The oil analysis performed show that the wear protection is similar within both lubricants, although the ester oil promotes a slightly smaller particles generation, especially on the larger particles.

Table 4 – Ferrogram pictures after each group of tests at constant speed for both lubricants.

	Mineral Oil	Ester Oil
1000 rpm (200x)	 d = 0.1; Dl = 70.4; Ds = 27.6; CPUC = 980; ISUC = 4.2E5;	 d = 0.1; Dl = 62.7; Ds = 21.2; CPUC = 839; ISUC = 3.5E5;
1500 rpm (200x)	 d = 0.1; Dl = 77.3; Ds = 40.8; CPUC = 1181; ISUC = 4.3E5;	 d = 0.1; Dl = 58.2; Ds = 23.2; CPUC = 814; ISUC = 2.8E5;
2000 rpm (200x)	 d = 0.1; Dl = 76.7; Ds = 37.7; CPUC = 1144; ISUC = 4.5E5;	 d = 0.1; Dl = 73.4; Ds = 26.5; CPUC = 999; ISUC = 4.7E5;

7 CONCLUSIONS

This work allows the establishment of the following conclusions:

1. The ester biodegradable lubricant promotes a lower operating temperature in comparison with the mineral reference lubricant tested.

2. The ester lubricant allows a substantial reduction in power loss, attaining 21%.

3. The overall efficiency is increased around 0.5%.

4. The wear protection of ester lubricant is slightly better than the conferred by the mineral lubricant.

5. The mineral lubricant might be replaced by a biodegradable non-toxic ester lubricant with advantage.

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REFERENCES

[1]. O. Hurasky-Schonwerth and Ch. Bugiel M. Weck, *Service Behaviour of PVD-Coated Gearing Lubricated with Biodegradable Synthetic Ester Oils.* VDI-Berichte, 2002(Nr1665).

[2]. K. Michaelis and A. Doleschel B.-R. Höhn, *Frictional behavior of synthetic*

gear lubricants. Tribology research: From model experiment to Industrial Problem, 2001.

[3]. K. Michaelis and T. Vollmer B.-R. Höhn, *Thermal Rating of Gear Drives: Balance Between Power Loss and Heat Dissipation.* AGMA Technical Paper, 1996.

[4]. B.-R. Hohn K. Michaelis, *Influence of lubricants on power loss of cylindrical gears.* Tribology Transactions, 1994. **37**.

[5]. J. Seabra R. Martins, Ch. Seyfert, R. Luther, A. Igartua and A. Brito. *Power loss in FZG gears lubricated with industrial gear oils: biodegradable ester vs. mineral oil*. in *31th "Leeds-Lyon Symposium" on Tribology*. 2004 (to be published).

[6]. T. Hunt, *Handbook of Wear Debris Analysis and Particle Detection in Liquids,* ed. E. Science. 1993.

[7]. M. Pasquier C. Changenet, *Power Losses and Heat Exchange in Reduction Gears: Numerical and Experimental results.* VDI-Berichte, 2002(1665).

[8]. P. Eschmann, *Ball and roller bearings - Theory, design and applications,* ed. Wiley. 1985.

[9]. K. Michaelis H. Winter. *Investigations on the thermal balance of gear drives*. in *Fifth World Congress on Theory of Machines and Mechanisms*. 1979: American Society of Mechanical Engineers.