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INTRODUÇÃO

As caixas de engrenagens têm aumentado a sua capacidade de carga cada vez mais rapidamente ao longo dos anos. Esta evolução tem sido conseguida sobretudo à custa do endurecimento dos aços e de desenvolvimentos na área dos lubrificantes. Os lubrificantes para aplicações industriais podem ter pacotes de aditivos que chegam aos 20% do seu conteúdo, nomeadamente, aditivos de extrema pressão, contra a corrosão, o desgaste ou pitting. Muitos desses aditivos são extremamente nocivos para as pessoas e para o ambiente. Uma nova abordagem para o problema da lubrificação de engrenagens é necessária de modo a oferecer novas combinações de materiais e lubrificantes que consigam o mesmo desempenho com menores custos e menores riscos ambientais.

O presente estudo pretende avaliar e comparar diferentes combinações de materiais e lubrificantes em termos resistência ao desgaste e ao micro-pitting. A comparação terá como referência aquele que é o material mais usual, aço de cementação, lubrificado com um óleo mineral parafínico de referência. Os outros materiais a serem ensaiados são ADI (Austempered Ductile Iron), ferro fundido nodular austemperado, e aço nitrurado com revestimento de superfície. Os diferentes materiais, incluindo o aço de cementação, serão ensaiados com o óleo mineral parafínico de referência e um outro óleo candidato.

Um total de oito ensaios serão efectuados, combinando três materiais com dois lubrificantes (o aço de cementação será ensaiado com dois níveis de carga).

Nesta nova abordagem o óleo candidato é um lubrificante biodegradável combinado com materiais tribo-reactivos (revestimentos de superfície ou ADI) de modo a substituir os aditivos de extrema pressão, anti-desgaste e melhoradores de atrito. No caso dos revestimentos de superfície, o custo adicional do revestimento tem de ser compensado com o aumento de vida do produto, a sua qualidade e com benefícios ambientais.

1. INTRODUCTION

Power transmission in gearboxes has been increasing faster and faster over the years. These improvements have been achieved mostly at the expense of hardening of steels and research in the area of lubricants. Lubricants for industrial applications may have additives packages that sum up to 20% of their content, regarding extreme pressure, corrosion, wear and pitting protection of surfaces. Most of those additives are extremely harmful for people and the environment. A new approach to the problem of lubrication in gear applications is needed in order to provide different combinations of materials and lubricants that are able to perform the same tasks at lower costs and environmental risks.

The present study aims to evaluate and compare different combinations of materials and lubricants in terms of wear and micro-pitting resistance. The comparisons will be established in reference to what is the most common material, case carburised steel, lubricated with a reference paraffinic mineral oil. The other materials to be tested are ADI (Austempered Ductile Iron) and nitrided steel with Zr-N surface coating. These different materials, including case carburised steel, will be tested with the reference paraffinic mineral oil and one other candidate oil (non-toxic and biodegradable).

A total of eight test runs took place, combining three materials with two lubricants (2 different load levels were used for carburised steel tests).

In this new approach the candidate oil is a biodegradable lubricant which is combined with tribo-reactive materials (surface coatings or ADI) in order to replace extreme pressure, anti wear and friction improvers additives. In the case of surface coatings, the additional cost of the coating has to be compensated by increased life time, product quality and environmental benefits.

2. GEAR OILS

Two industrial gear oils were studied: the reference oil, previously used as industrial gear oil in many applications and the candidate lubricant, potential biodegradable non-toxic industrial gear oil. The two oils were chosen so that their kinematic viscosities will be almost the same at 100 °C.

The most significant properties are shown in Table 1. The density and the Viscosity Index of the candidate oil are significantly higher than those of the mineral oil. The thermoviscosity and the piezoviscosity coefficients are smaller for the candidate oil.

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

Figure 1 and Figure 2 show, respectively, the variation of the lubricants viscosity and density with temperature. Between 60°C and 100°C the kinematic viscosities of the two lubricants are very similar. The density of the candidate oil is significantly higher than that of the mineral oil at all temperatures.

Parameter	*Method*	*Desig.*	*Units*	*Lubricating Oils*	
Lubricant reference				**Mineral Oil**	**Candidate Oil**
Oil type		/	/	Industrial gear oil	Industrial gear oil
Base oil	DIN 51451	/	/	paraffinic mineral oil	ester biodegradable
Physical properties [a]					
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 2909		ºC	-21	-42
Viscosity and density equations					
Viscosity [1] $Log\,Log(\nu + a) = n - m * Log(T)$	ASTM D341	m	/	9.1350	7.9757
		n	/	3.5260	3.0763
		a	cSt	0.7	0.7
Thermoviscosity [1] – $\beta = \frac{m}{T} \times \frac{(\nu + a) * Ln(\nu + a)}{\nu}$		β_{40}	cSt/ºC	0.0565	0.0443
		β_{100}	cSt/ºC	0.0267	0.0229
Piezoviscosity [2] – $\alpha(p,T) = \frac{1}{a_1 + a_2 * T + (b_1 + b_2 * T) * p}$		a_1	bar	361	526
		a_2	bar/ºC	2.775	3.845
		b_1	/	6.65×10^{-3}	4.90×10^{-2}
		b_2	1/ºC	3.36×10^{-4}	1.14×10^{-4}
Thermal expansion [1] - $\rho = \rho_{ref}\left[1 - \alpha_v \cdot (T - T_{ref})\right]$		α_v	g/cm^3 ºC	7.16×10^{-4}	8.05×10^{-4}
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	DIN 51347-2	/	N/mm^2	68	37
FZG rating	DIN 513540	K_{FZG}	/	>13	?
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

a) Other kinematic viscosity measurements were made using an Engler viscometer and other density measurements were made using an Anton Paar densimeter – see Appendix A.

Table 1 - Physical and chemical properties of the lubricants.

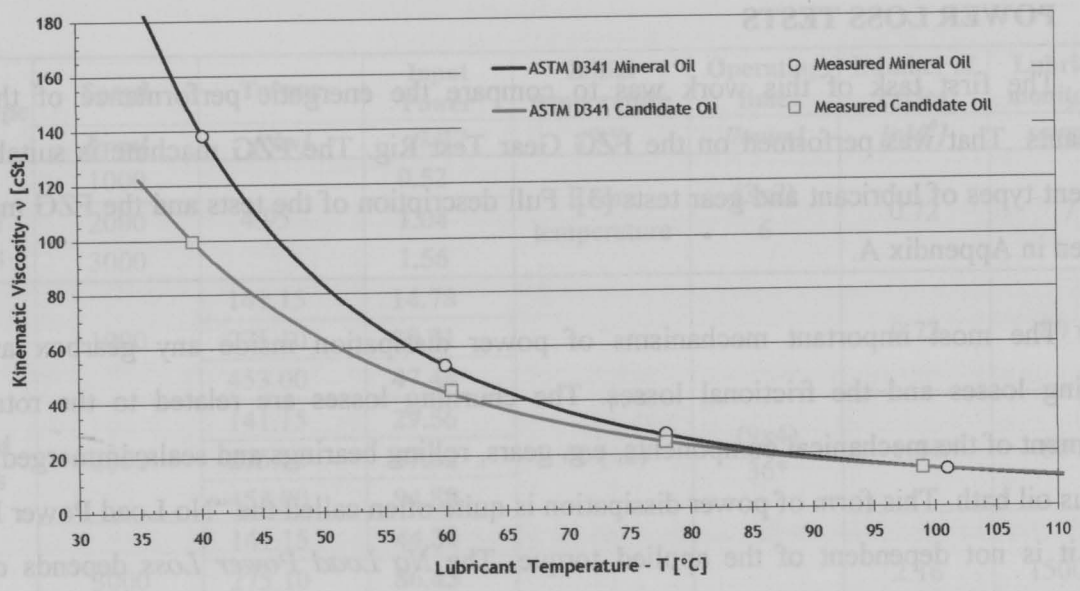


Figure 1 – Variation of the lubricants viscosities with temperature (based on measurements made with an Engler viscometer – Appendix A).

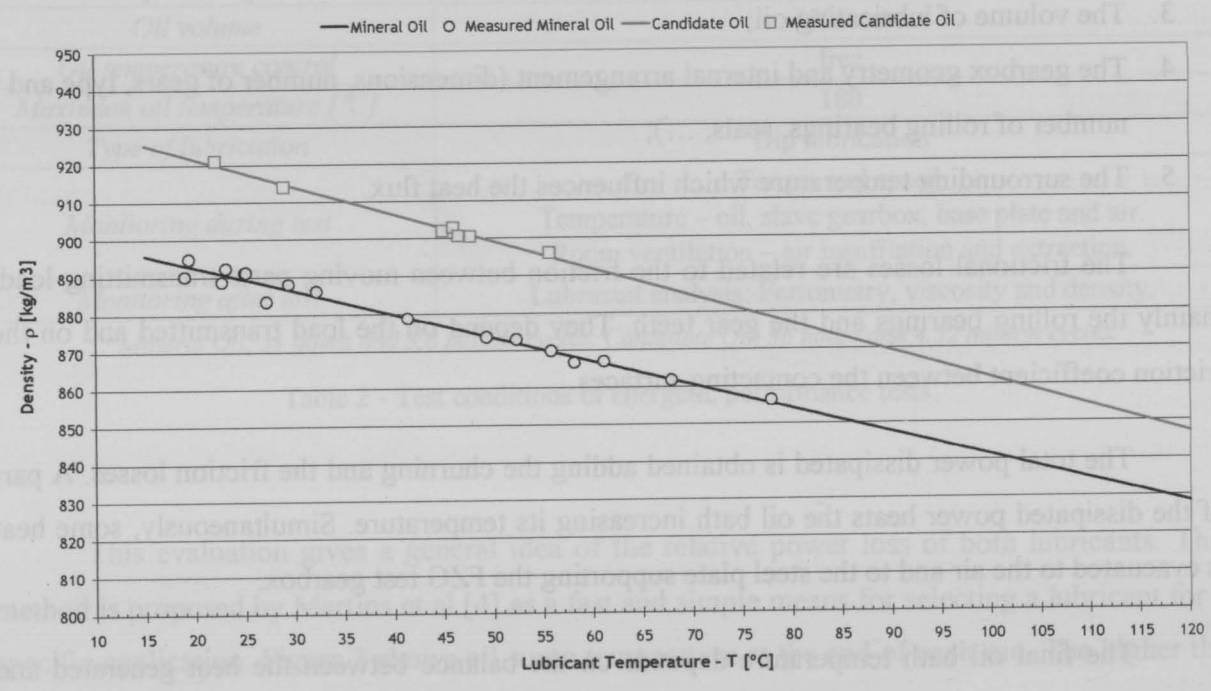


Figure 2 – Variation of the lubricants densities with temperature.

3. POWER LOSS TESTS

The first task of this work was to compare the energetic performance of the two lubricants. That was performed on the FZG Gear Test Rig. The FZG machine is suitable for different types of lubricant and gear tests [3]. Full description of the tests and the FZG machine is given in Appendix A.

The most important mechanisms of power dissipation inside any gearbox are the churning losses and the frictional losses. The churning losses are related to the rotational movement of the mechanical components, e.g. gears, rolling bearings and seals, immerged in the viscous oil bath. This form of power dissipation is quite often called the "No Load Power Loss", since it is not dependent of the applied torque. The *No Load Power Loss* depends on the following parameters:

1. The input rotational speed of the gearbox;
2. The viscosity of the lubricant oil at the operating temperature;
3. The volume of lubricating oil;
4. The gearbox geometry and internal arrangement (dimensions, number of gears, type and number of rolling bearings, seals, …);
5. The surrounding temperature which influences the heat flux.

The frictional losses are related to the friction between moving parts transmitting load, mainly the rolling bearings and the gear teeth. They depend on the load transmitted and on the friction coefficient between the contacting surfaces.

The total power dissipated is obtained adding the churning and the friction losses. A part of the dissipated power heats the oil bath increasing its temperature. Simultaneously, some heat is evacuated to the air and to the steel plate supporting the FZG test gearbox.

The final oil bath temperature depends on the balance between the heat generated and heat evacuated. After a long operating period both the temperature of the oil and of the test rig tend to an equilibrium state. Gear tests were performed in order to evaluate the influence of the lubricants characteristics on the power dissipated.

Test conditions are presented in Table 2.

Test type	Speed	Torque	Input Power	Initial temperature	Operating time	Number of cycles	Lubricant monitoring
	[rpm]	*[Nm]*	*[kW]*	*[ºC]*	*[hours]*	*[x10^6]*	*samples*
No load tests	1000	4.95	**0.52**	Room temperature	(3x2) 6	0.72	/
	2000		**1.04**				
	3000		**1.56**				
Load tests	1000	141.15	**14.78**	40 (±2)	(9x4) 36*	0.72	80 ml
		275.10	**28.81**				
		453.00	**47.44**				
	2000	141.15	**29.56**			1.44	80 ml
		275.10	**57.62**				
		453.00	**94.88**				
	3000	141.15	**44.34**			2.16	1500 ml
		275.10	**86.43**				
		453.00	**142.31**				
Total						**4.32***	

General operating conditions:	
Oil volume	1500 ml
Oil temperature control	free
Maximum oil temperature [ºC]	180
Type of lubrication	Dip lubrication
Monitoring during test	Torque and speed. Temperature – oil, slave gearbox, base plate and air. Room ventilation – air insufflation and extraction.
Monitoring after test	Lubricant analysis: Ferrometry, viscosity and density.

** Mineral Oil, 31 hours and 3.8 million cycles. Candidate Oil, 36 hours and 4.32 million cycles.*

Table 2 - Test conditions of energetic performance tests.

This evaluation gives a general idea of the relative power loss of both lubricants. This method is proposed by Martins et al [4] as a fast and simple means for selecting a lubricant for a specific application. Figure 3 shows oil sump temperature at the end of each run. The higher the equilibrium temperature, the more power is lost in gear meshing.

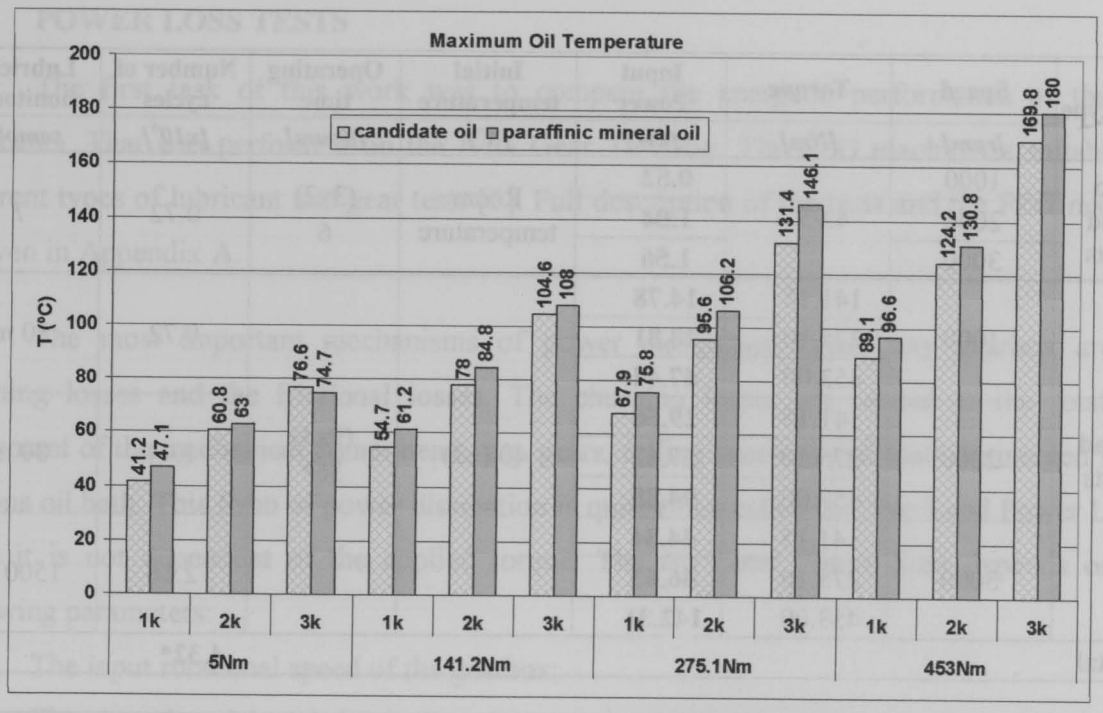


Figure 3 - Maximum oil sump temperature at 1000, 2000 and 3000rpm for wheel torques corresponding to FZG load stages 1, 5, 7 and 9.

As the tests were performed under the same operating conditions, the results show clearly a significant reduction in the operating temperature obtained by the candidate oil. This reduction in the operating temperature is the result of the lower friction coefficient provided by the candidate oil. Using the thermal balance proposed by Martins [5] and described above, the reduction of the friction coefficient provided by the candidate oil when compared with the reference oil is illustrated in Figure 4.

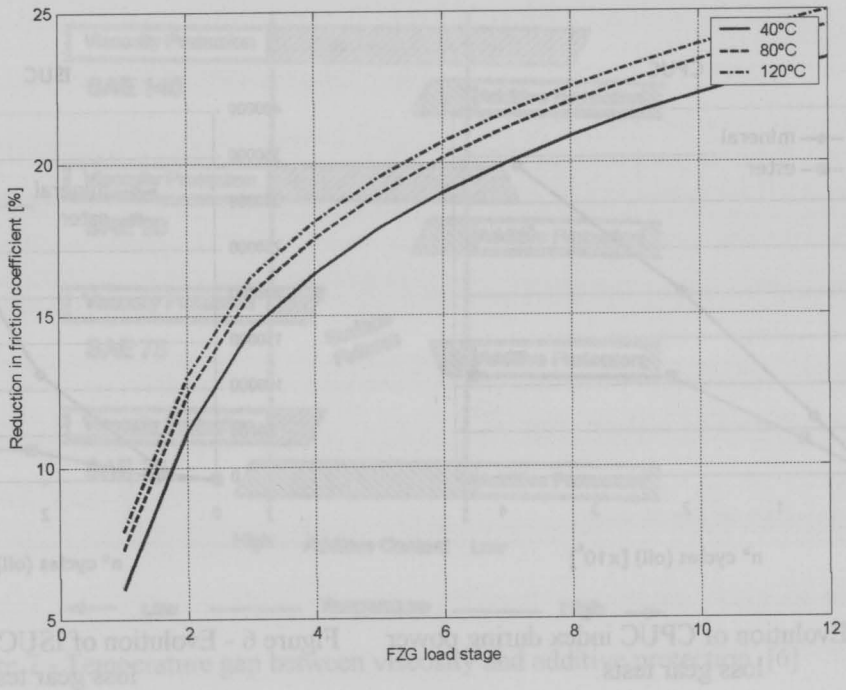


Figure 4 - Reduction in friction coefficient provided by the candidate oil when compared with the reference oil at three different temperatures.

The reduction in friction coefficient increases with increasing input torque. At load stage K7 (T = 183Nm) the reduction is about 20%.

During the power loss tests performed in the FZG machine oil samples were collected at pre-defined moments. Figure 5 and Figure 6 show the evolution of the ferrometric wear particle parameters during the whole test program for the mineral and candidate oils, respectively.

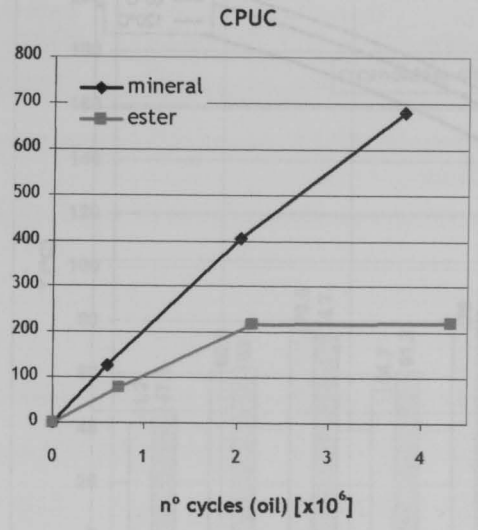


Figure 5 - Evolution of CPUC index during power loss gear tests.

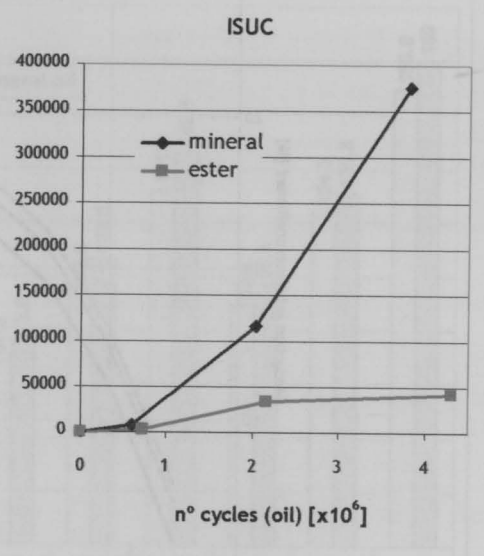


Figure 6 - Evolution of ISUC index during power loss gear tests.

It is clear that the ferrometric wear particle parameters of the mineral oil are higher and grow faster than those corresponding to the candidate oil.

It might also indicate that, inside the contact between the gear teeth, the film thickness is higher and the friction coefficient is lower, when lubricated with the candidate oil.

Temperature is, in fact, an important aspect in gear failure. According to Höhn et al [6], temperature has a decisive influence on gear failure mechanisms. This happens due to the temperatures at which viscosity protection and additive protection are more effective. Rarely temperature at which additives are more effective or the additive package contents are known, causing difficulties in establishing the optimum service temperature for the lubricant. Figure 7 shows different temperature combinations that can be found for several oils.

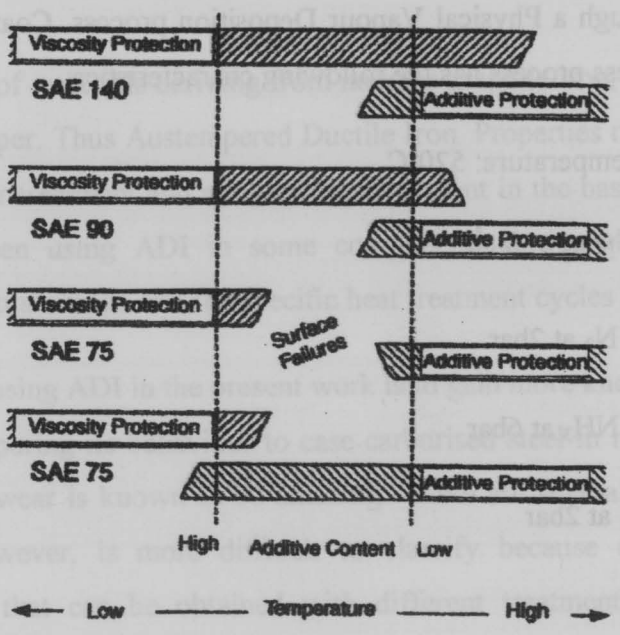


Figure 7 - Temperature gap between viscosity and additive protection. [6]

Therefore, this method is also a simple way to have a general idea of the most suitable temperature for the lubricant in study.

4. SURFACE COATINGS

Surface coatings have also been studied in the scope of this project, regarding the latest developments in the area of materials for gears. Surface coatings have been used in industry in less demanding applications, but have been ruled out of gear applications because they still lack reliability and their cost is too high. Their potential is, nevertheless, very interesting. The materials used in these coatings are usually carbon or molybdenum based, meaning they are self-lubricant. This allows reducing the friction coefficient and prevents excessive wear during start/stop operations. The research focused mainly the work of Amaro [7] and is described in more detail in Appendix B.

The coating used in this project is a Zirconium (Zr) and Nitrogen (N) based coating. It is a bilayer coating with around 3μm thickness. Tests indicated good adherence of the coating.

Coating is applied through a Physical Vapour Deposition process. Coated gears are previously nitrided. Nitriding process process has the following characteristics:

- Process temperature: 520°C
- 60min of NH_3 at 6bar
- 30min of N_2 at 2bar
- 60min of NH_3 at 6bar
- 19h of N_2 at 2bar

5. PERFORMANCE EVALUATION OF A GEARBOX

In the work of Martins [5] it is studied the performance evaluation of a gearbox. That work was carried out on an actual two speed transfer gearbox used in four wheel drive light trucks. That study, along with the power loss tests carried on the FZG machine were the basis of a simulation of the power loss in gearboxes.

The analysis of power losses in gearboxes can be divided in load dependent losses and non load dependent losses. Non load dependent losses only vary with input speed, whereas load dependent losses are highly influenced by the friction coefficient of the contact and by the applied torque. When using dip lubrication, the reduction of oil volume in the box benefits the overall performance. The advantage of reducing oil volume can only be achieved by self lubricant coatings or materials, bringing out the importance of establishing new combinations of materials/lubricants in engineering applications. Further description of the work is given in Appendix C.

6. ADI

ADI is a family of materials deriving from nodular graphite iron, obtained through a heat treatment called austemper. Thus Austempered Ductile Iron. Properties of ADI vary greatly with parameters of austemper treatment and alloy elements present in the base cast iron. Some major manufacturers have been using ADI in some components during the last decades always successfully. But the actual results and the specific heat treatment cycles are not known.

The purpose of using ADI in the present work is to gain more knowledge of the adequacy of its use in gears comparing its behaviour to case carburised steel in micro-pitting conditions. ADI's ability to resist wear is known to be amazing. There are several results to support that. Fatigue resistance, however, is more difficult to classify because of the wide variety of mechanical properties that can be obtained with different treatments. This is always very important to keep in mind when analysing results of ADI tests. Therefore, the results to be obtained in this work will only be reliable for the same base material and the same heat treatment as the ones used in this work. Further information about ADI can be found in the work of Magalhães [8] and in Appendix D. The chemical composition of the ADI used in this work is shown in Table 3.

C	Si	Cu	Mn	P	S	Cr	Mo	Ni	Mg	Others
3.52	2.39	0.89	0.51	0.012	0.012	0.038	0.042	0.026	0.048	–

Table 3 - Nominal chemical composition (weight % ± 10%) of the nodular iron.

The ADI produced for this work was treated in the facilities of FEUP. Figure 8 shows the heat treatment cycle.

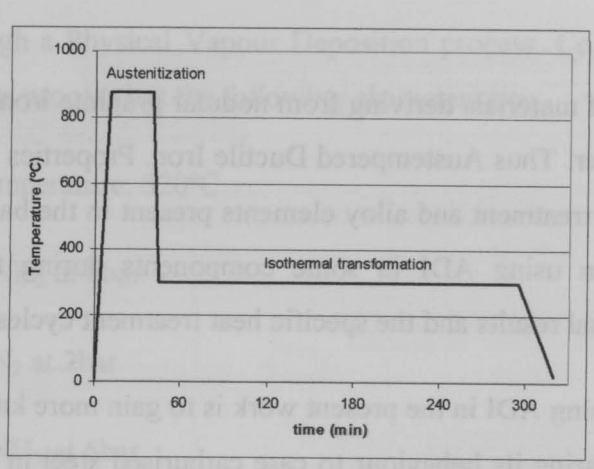


Figure 8 - Heat treatment cycle for single stage ADI heat treatment.

In Table 4 the expected mechanical properties are presented.

Isothermal transformation temperature (ºC)	σr (Mpa)	σ02 (Mpa)	Elongation (%)	Impact toughness (J)	Hardness (HV)
300	1433	1178	6	73	426

Table 4 - Expected properties of ADI for the austempering temperature of 300ºC.

7. MICRO-PITTING TESTS

Micro-pitting is a contact fatigue wear phenomenon that is observed in combined rolling and sliding contacts operating under elastohydrodinamic lubrication (EHL) or mixed EHL/Boundary lubrication conditions. Micro-pitting can be regarded as fatigue failure with a net of cracks close to the surface which typically starts during the first 10^5 to 10^6 stress cycles. [9, 10]

Eventually, after micro-pitting is detected, it can remain unchanged for many millions of cycles. But it is most likely to evolve to more severe damages such as pitting and spalling. In any case, micro-pitting is associated with increase noise and vibration of the gear which can be in itself a source of malfunctioning depending on the type of application. To distinguish micro-pitting from pitting in terms of service conditions and mechanisms that lead to those phenomena

is not very easy. The complex nature of gear contact makes it difficult to separate individual failure modes. [10]

In order to establish the requirements of a micro-pitting test, some aspects of pitting and micro-pitting conditions have to be established. This will help to choose a test procedure and to interpret the results. Table 5 shows the conditions of the two types of failure. [10]

	pitting	Micro-pitting
Depth	Zone of highest subsurface stress; may also appear in surface craters	Typically in the range of 5-10μm or material surface
Size	Visible to the human eye	Invisible to the human eye; grey shaded
Lubrication regime	elastohydrodynamic – full film	Mixed lubrication
Main cause	Contact pressure	Roughness
Other causes	Hardness; speed; temperature	
Lubricant influence	Little	High
Consequences	spalling; 3 body wear; losing of tooth resistance	Losing of geometric accuracy; noise; vibration; may develop pitting
Ways of avoiding	Using lubricants of higher viscosity which don't easily fill surface micro-craters	Increase film thickness; use oils with S-P additives

Table 5 - Pitting and micro-pitting conditions.

Micro-pitting tests included carburized steel, ADI and coated gears. Each material was tested with both lubricants. Appendix E gives a thorough description of micro-pitting phenomenon as well as the tests carried out. Coding of the tests:

Test set 1:

CC1-M – 20MnCr5 carburized steel + mineral oil;

CC1-E – 20MnCr5 carburized steel + ester biodegradable.

Test set 2:

CC2-M – 20MnCr5 carburized steel + mineral oil;

CC2-E – 20MnCr5 carburized steel + ester biodegradable.

Test set 3:

ADI-M – ADI + mineral oil;

ADI-E – ADI + ester biodegradable.

Test set 4:

(Zr+N)-M – nitrided steel + Zr-N coating + mineral oil;

(Zr+N)-E – nitrided steel + Zr-N coating + ester biodegradable

Test conditions are presented in Table 6 and Table 7. In Table 6, 0.35 represents the conditions of test sets 1 and 4. 0.50 represents the conditions of test set 2. ADI conditions are less severe than steel's because ADI cannot work under such high levels of contact pressure.

		K3 – running in		K7		K9-1 / K9-2 b)	
		pinion	wheel	Pinion	wheel	pinion	wheel
Temperature [ºC]		80		90		90	
Torque [Nm] a)	0.35	28.8	43.2	132.5	198.8	215.6	323.4
	0.50	35.3	53	183.4	275.1	302.0	453
Rot speed [1/min]		2250	1500	2250	1500	2250	1500
Vt [m/s]		8.3					
Hertzian Stress [Mpa]	0.35	510		1094		1395	
	0.50	564		1286		1657	
Power [kW]	0.35	6.8		31.2		50.8	
	0.50	8.3		43.2		71.2	
Duration [h] / cycles [10^3]		1 / 135	1 / 90	16 / 2160	16 / 1440	16 / 2160	16 / 1440

a) The same name is given to load stages corresponding to different torques because the dead weights used to create those torques are the same. The difference lies in applying them with a 0.35m lever arm resulting in lower torques or 0.50m lever arm resulting in higher torques.

b) Coated nitrided gear did not ran load stage K9 due to the high grade of deterioration found on load stage K7.

Table 6 - Test conditions of micro-pitting tests for carburized steel and coated gears. [11]

	K3 - running in		K5		K7	
	pinion	wheel	Pinion	wheel	pinion	wheel
Temperature [ºC]	80		90		90	
Torque [Nm]	28.8	43.2	70	105	132.5	198.8
Rot speed [1/min]	2250	1500	2250	1500	2250	1500
Vt [m/s]	8.3					
Hertzian Stress [Mpa]	510		795		1094	
Power [kW]	6.8		16.5		31.2	
Duration [h] / cycles [10^3]	1 / 135	1 / 90	16 / 2160	16 / 1440	16 / 2160	16 / 1440

Table 7 - Test conditions of micro-pitting tests for ADI gears.

Results of micro-pitting tests include micro-pitted area, weight loss, ferrometry and roughness of teeth flanks.

The visual aspect of teeth flanks after tests is shown in Table 8. Appendix E has full details of test results.

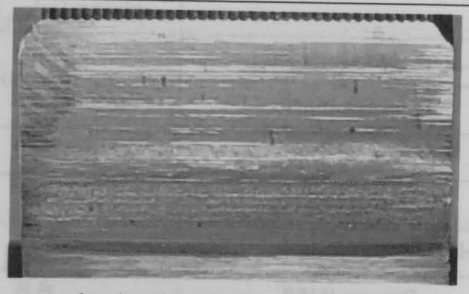
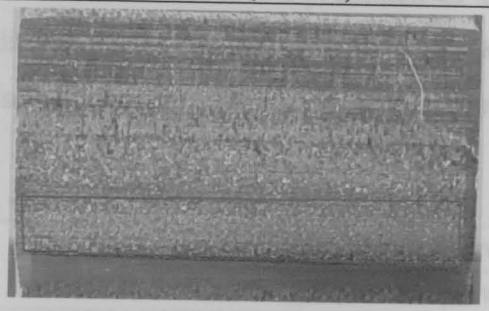
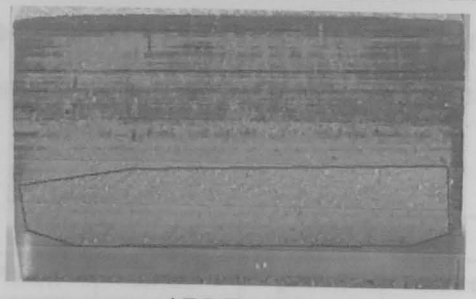
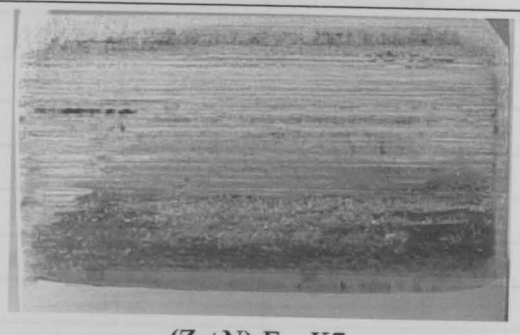
Paraffinic mineral oil	Ester biodegradable
 carburized steel (CC1-M) – K9-2	 carburized steel (CC1-E) – K9-2
 carburized steel (CC2-M) – K9-2	 carburized steel (CC2-E) – K9-2
 ADI-M – K7	 ADI-E – K7
 (Zr+N)-M – K7	 (Zr+N)-E – K7

Table 8 - Pictures of the teeth after test. Teeth width = 14mm

Micro-pitted area is surrounded in red for easier reading. Coated gear presented no micro-piting either with mineral and ester oil. However, tests showed severe pitting around pitch circle. The most important aspect was the destruction of the coating during load stage K7, revealing inadequacy of the coating for gear applications.

Table 9 shows microscope pictures of the teeth flanks below pitch circle.

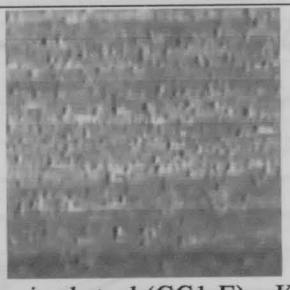
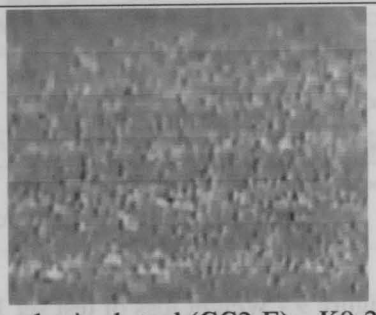
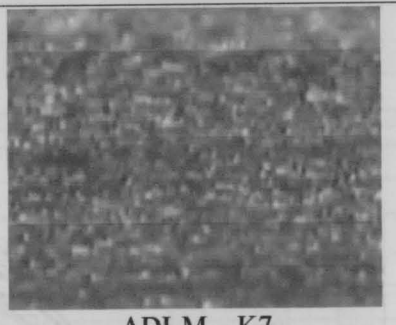
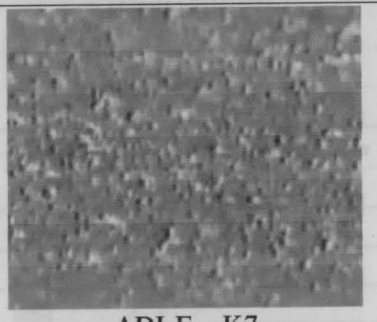
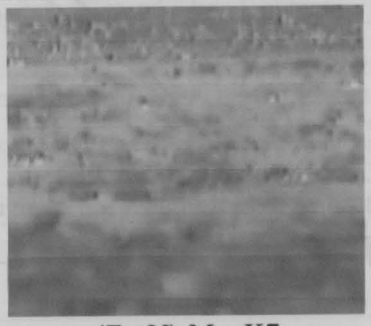
Paraffinic mineral oil	Ester biodegradable
 carburized steel (CC1-M) – K9-2	 carburized steel (CC1-E) – K9-2
 carburized steel (CC2-M) – K9-2	 carburized steel (CC2-E) – K9-2
 ADI-M – K7	 ADI-E – K7
 (Zr+N)-M – K7	 (Zr+N)-E – K7

Table 9 – Video-microscope pictures of teeth flanks after test.

Pits in ADI gears always appear smaller than in steel gears. In steel gears, ester oil seems to develop smaller pits.

Results also include weight loss. The comparison in weight loss between the two lubricants for each material is shown in Figures 9, 11, 13 and 15. Side by that comparison is the cumulative concentration wear particle index.

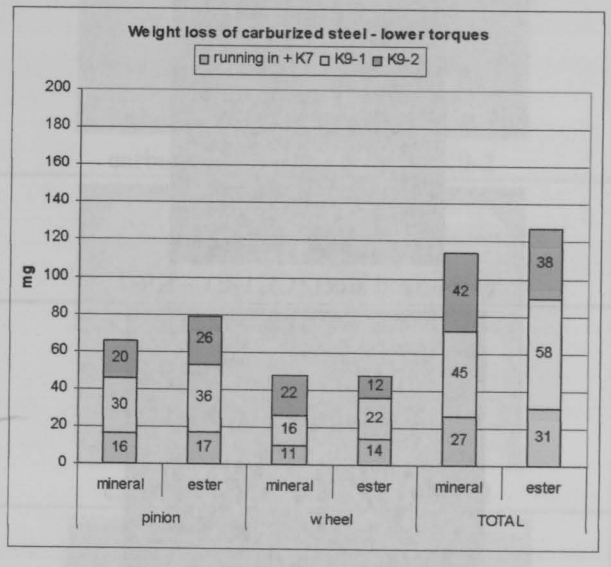


Figure 9 - Weight loss of test set 1.

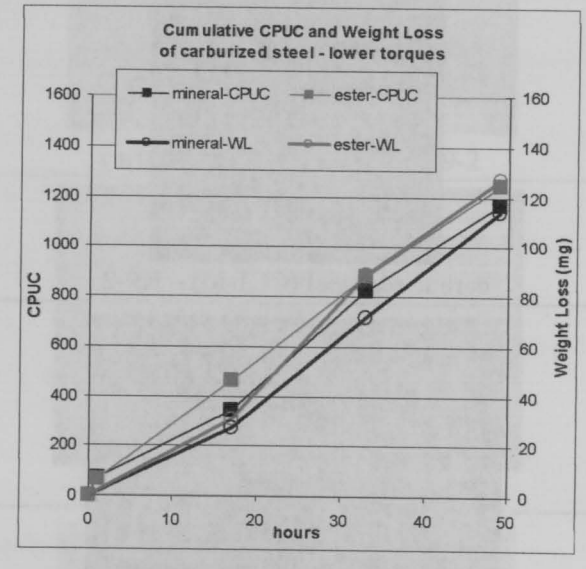


Figure 10 - Wear particle index during test set 1.

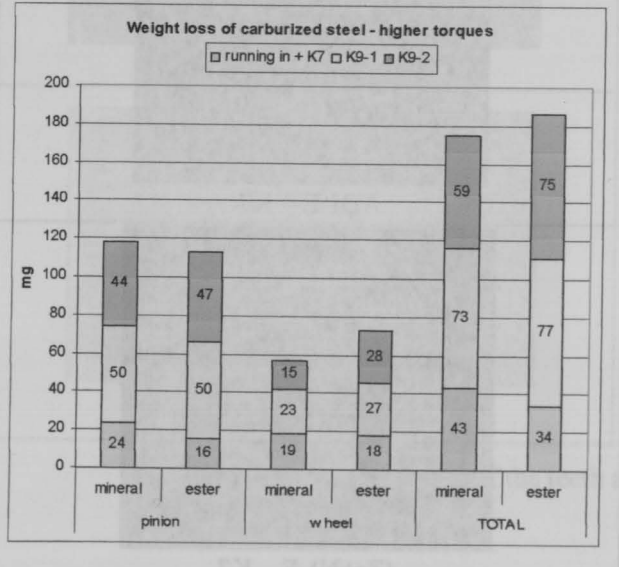


Figure 11 - Weight loss of test set 2.

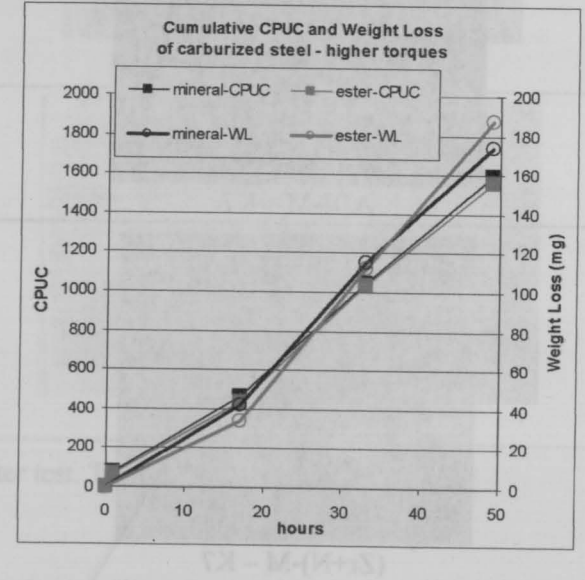


Figure 12 - Wear particle index during test set 2.

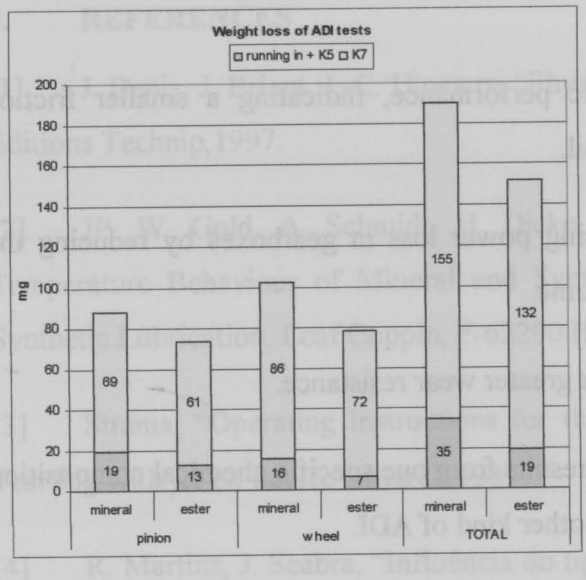


Figure 13 - Weight loss of test set 3.

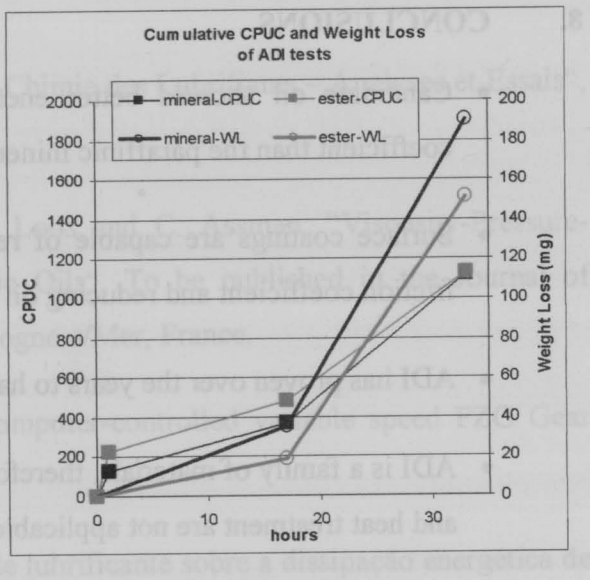


Figure 14 - Wear particle index during test set 3.

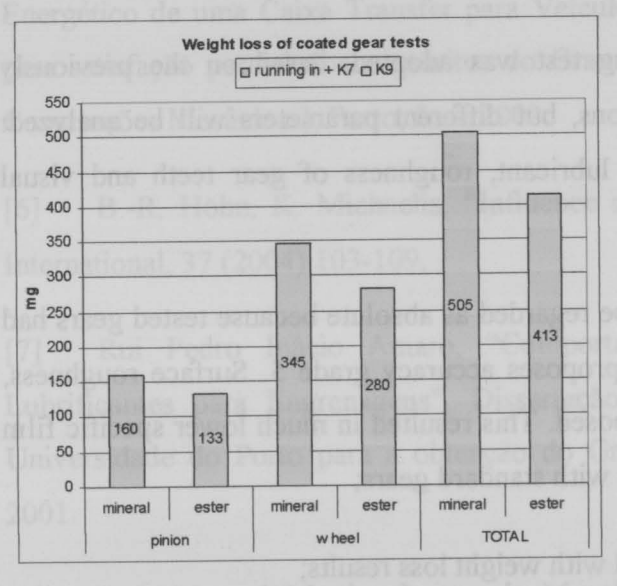


Figure 15 - Weight loss of test set 4.

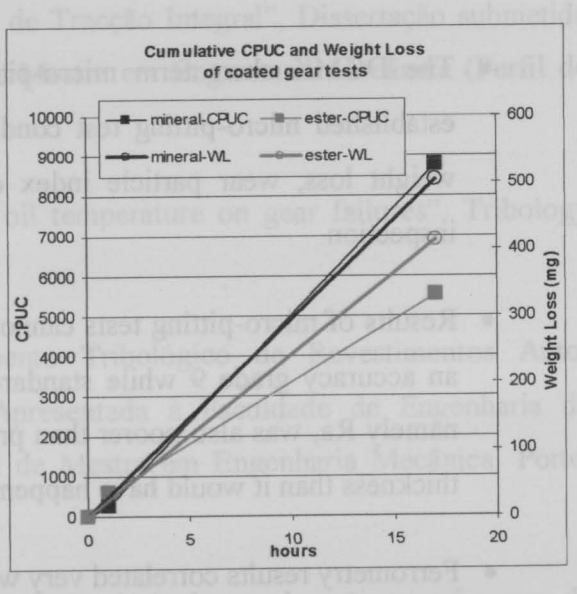


Figure 16 - Wear particle index during test set 4.

Except for test set 1, ester oil always shows less weight loss than mineral oil. In test set 2, only after load stage K9-2 does this tendency alters. Weigth loss and ferrometry correlate very well in all tests. Please refer to Appendix E for complete results.

8. CONCLUSIONS

- Candidate oil showed better energetic performance, indicating a smaller friction coefficient than the paraffinic mineral oil.
- Surface coatings are capable of reducing power loss in gearboxes by reducing the friction coefficient and reducing oil volume.
- ADI has proven over the years to have a greater wear resistance.
- ADI is a family of materials, therefore, results from one specific chemical composition and heat treatment are not applicable to other kind of ADI.
- Micro-pitting tests are usually very long and it is not possible at FEUP to evaluate the mean profile deviation used as criterion of failure in the standard FVA and DGMK tests.
- The DGMK short term micro-pitting test was adopted, based on the previously established micro-pitting test conditions, but different parameters will be analyzed: weight loss, wear particle index of lubricant, roughness of gear teeth and visual inspection.
- Results of micro-pitting tests cannot be regarded as absolute because tested gears had an accuracy grade 9 while standard proposes accuracy grade 5. Surface roughness, namely Ra, was also poorer than proposed. This resulted in much lower specific film thickness than it would have happened with standard gears;
- Ferrometry results correlated very well with weight loss results;
- The very low friction coefficient of candidate oil may be one of the reasons for its good performance in micro-pitting protection.

9. REFERENCES

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APPENDIX A – ENERGETIC PERFORMANCE OF GEAR OILS

INTRODUCTION

The main objective of this study is to compare the energetic performance of two industrial gear oils: a reference mineral oil and a biodegradable lubricant. For that purpose gear tests are performed on the FZG machine using type C gears, and the stabilized oil operating temperature is measured for different values of the input torque and input speed.

The protection against wear given by the two lubricants is also compared. For that purpose lubricant samples are collected during and at the end of the gear tests, and analyzed by Direct Reading Ferrography in order to evaluate the index of wear particles (CPUC and ISUC).

A1. FZG GEAR TEST RIG

The power loss tests were performed on the FZG gear test rig, shown in Figure A1 [A1]. It's a back-to-back spur gear test rig with "power circulation".

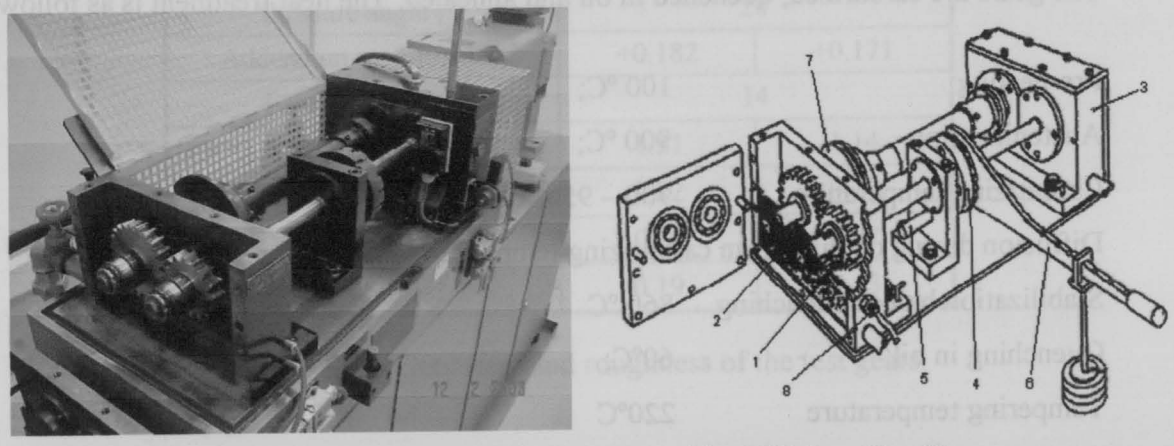


Figure A1: Picture and schematic view of the FZG Gear Test Rig.

The test pinion (1) and the test wheel (2) are connected by two shafts to the driving gears (3). The front shaft is divided in two parts with the load clutch between (4). One half of the load clutch can be fixed to the foundation by a locking pin (5) while the other part can be twisted using a load lever and weights (6). After bolting the clutch together the load can be removed and the shaft unlocked. Now a static torque is applied to the system that can be measured by the

torque measuring clutch (7). The maximum speed of the AC-motor is 3000 rpm. The test gears can be dip lubricated or jet lubricated. When dip lubrication is used, the oil may be heated using the electrical heaters mounted in the test gearbox. The heater and cooling coil allows the settling of a constant oil temperature measured by the temperature sensor (8).

A2. TEST GEARS

Test gears were made of DIN 20 MnCr 5 steel [A1]. The chemical composition of this material is shown in Table A1.

C	Si	Mn	Cr	Mo	Ni	V	W	Others
0.20	0.20	1.25	1.15	-	-	-	-	-

Table A1 - Chemical composition (weight %) of the 20 MnCr 5 carburizing steel

The gears are carburized, quenched in oil and annealed. The heat treatment is as follows:

Pre-heating	100 °C;	
Austenitization	900 °C;	
Carburizing temperature	900 – 950 °C	time - 05:00 h;
Diffusion during cooling from carburizing temperature to 860 °C		time - 02:30 h;
Stabilization before quenching	860 °C	time - 00:30 h;
Quenching in oil:	60°C;	
Tempering temperature	220°C	time - 01:00 h.

This heat treatment provides a surface hardness of the teeth flanks of **58 – 62 HRC** and a carburizing depth of **0.8 mm**.

The most important physical properties (average values) at ambient temperature of the 20 MnCr 5 carburizing steel are:

Modulus of elasticity	[10 exp(3) N/mm^2]	210;
Density	[g/cm^3]	7.85;
Thermal conductivity	[W/m.K]	42.0;
Specific electric resistivity	[Ohm mm^2/m]	0.16;
Specific heat capacity	[J/g.K]	0.46.

The 20 MnCr 5 carburizing steel is applied in highly stressed parts, e.g., gears, crankshafts, bushings.

Both gears were run-in before performing the power loss gear tests. The geometry and micro-geometry (after running-in) of the test gears are shown in Table A2.

Parameter [Units]	*Pinion*	*Wheel*
Number of teeth	16	24
Module [*mm*]	4.5	
Centre distance [*mm*]	91.5	
Addendum diameter [*mm*]	82.45	118.35
Pressure angle [^{o}]	20	
Addendum modification	+0.182	+0.171
Face width [*mm*]	14	
Rz-$_{DIN}$ [μm]	1.31	1.14
Ra [μm]	0.22	0.17
Rq [μm]	0.28	0.22
Rpk [μm]	0.19	0.13

Table A2 – Geometry and roughness of the test gears.

A3. POWER LOSS INSIDE THE FZG SLAVE GEARBOX

The most important mechanisms of power dissipation inside any gearbox are the churning losses and the frictional losses [A2, A3]. The same occurs in the slave gearbox of the FZG gear test rig (see figure A2).

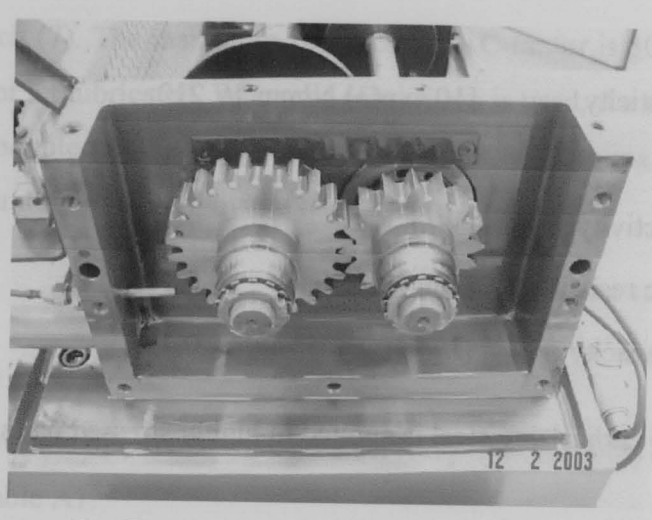


Figure A2 – Test gearbox of the FZG gear test rig (opened).

The churning losses are related to the rotational movement of the mechanical components, e.g. gears, rolling bearings and seals, immerged in the viscous oil bath. This form of power dissipation is quite often called the "No Load Power Loss", since it is independent of the applied torque [A2, A3].

The Churning Losses inside a gearbox have three main origins:

- The churning of the rotating gear set immerged in the oil bath;
- The churning of the rotating rolling bearings immerged in the oil bath;
- Other losses, due to the friction between moving components and the oil bath, e.g. oil seals.

Since there is no torque applied, the friction between the gear teeth is very small and doesn't affect the overall dissipated power. Thus the *No Load Power Loss* depends on the following parameters [A2, A3]:

1. The input rotational speed of the gearbox;
2. The viscosity of the lubricant oil at the operating temperature;

3. The volume of lubricating oil;

4. The gearbox geometry and internal arrangement (dimensions, number of gears, type and number of rolling bearings, seals, …);

5. The surrounding temperature which influences the heat flux.

The frictional losses are related to the friction between moving parts transmitting load, mainly the rolling bearings and the gear teeth. They depend on the load transmitted and on the friction coefficient between the contacting surfaces [A2, A3].

The total power dissipated is obtained adding the churning and the friction losses.

For a given gearbox geometry (for instance the test gearbox of the FZG test rig shown in figure A2), input speed, applied torque and lubricating oil, the power dissipated heats the oil bath increasing its temperature. Simultaneously, some heat is evacuated to the air and to the steel plate supporting the FZG test gearbox.

The final oil bath temperature depends on the balance between the heat generated and heat evacuated. After a long operating period both the temperature of the oil and of the test rig tend to an equilibrium state [A2, A3].

The different forms of power loss and heat evacuation of the FZG slave gearbox are shown in Figure A3

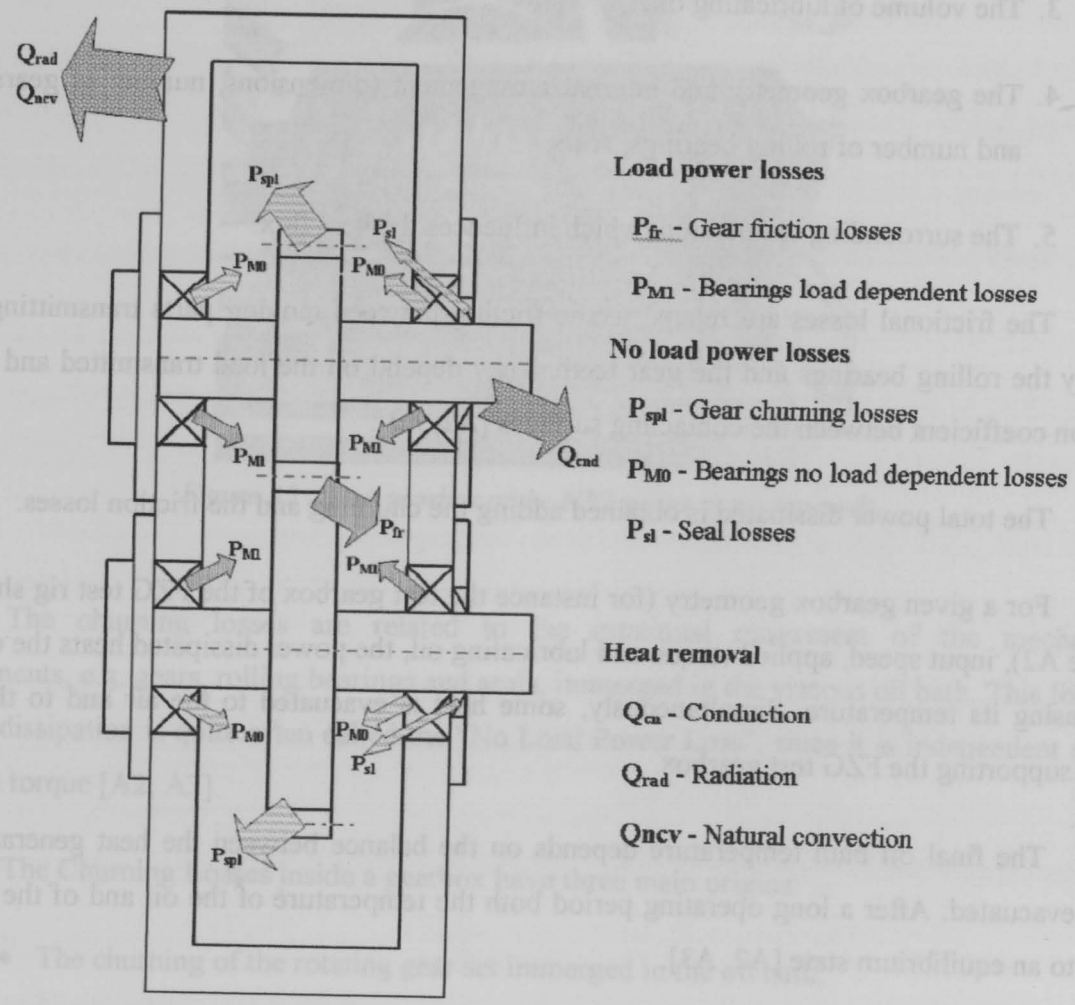


Figure A3 – Forms of power loss and heat evacuation of the FZG slave gearbox.

A4. POWER LOSS TESTS

Gear tests were performed in order to evaluate the influence of the lubricants characteristics on the power dissipated. The gear tests were done on a FZG test rig, using type C gears and dip lubrication and considering wide ranges of the operating conditions. The input powers ranged from 0.5 kW to 142 kW. The operating conditions used are detailed in Table A3.

At the beginning of each test the oil temperature is set at 40 (±2) °C. The test lasted 4 hours, which is the time necessary to reach a stabilized operating temperature. If the oil temperature reaches 180°C the test is stopped for safety reasons (damage of the slave gearbox seals).

Test type	Speed	Torque	Input Power	Initial temperature	Operating time	Number of cycles	Lubricant monitoring
	[rpm]	*[Nm]*	*[kW]*	*[ºC]*	*[hours]*	*[x10^6]*	*samples*
No load tests	1000	4.95	**0.52**	Room temperature	(3x2) 6	0.72	/
	2000		**1.04**				
	3000		**1.56**				
Load tests	1000	141.15	**14.78**	40 (±2)	(9x4) 36*	0.72	80 ml
		275.10	**28.81**				
		453.00	**47.44**				
	2000	141.15	**29.56**			1.44	80 ml
		275.10	**57.62**				
		453.00	**94.88**				
	3000	141.15	**44.34**			2.16	1500 ml
		275.10	**86.43**				
		453.00	**142.31**				
Total						**4.32***	

General operating conditions:	
Oil volume	1500 ml
Oil temperature control	free
Maximum oil temperature [ºC]	180
Type of lubrication	Dip lubrication
Monitoring during test	Torque and speed. Temperature – oil, slave gearbox, base plate and air. Room ventilation – air insufflation and extraction.
Monitoring after test	Lubricant analysis: Ferrometry, viscosity and density.

** Mineral Oil, 31 hours and 3.8 million cycles. Candidate Oil, 36 hours and 4.32 million cycles.*

Table A3 - Operating conditions used in the power loss gear tests (FZG test rig – type C gears).

The tests performed at very low torques (0.5 to 1.5 kW) lasted only 2 hours, which is enough to reach a constant operating temperature. In this case the tests started at ambient temperature (≈20ºC), since, at low speed, the operating temperature is slightly above 40ºC.

During the tests the oil temperature was monitored (but not controlled). The temperatures of the gearbox (outside), of the base plate and of the air were registered. Air insufflation and extraction were used during the tests in order to keep the ambient temperature constant.

During each test the input power remained constant. The torque was imposed at the beginning of the test and the speed was continuously monitored and controlled during the test. Lubricant samples were taken at the end of each test.

The lubricant was not changed from test to test. Two lubricant samples were collected during the tests and all the lubricant was collected at the end of the tests. The Mineral oil worked 34 hours (4.2 million cycles) and the Candidate oil worked 36 hours (4.32 million cycles).

Figure A4 shows the evolution of the oil sump temperature during the tests for the two oils, Candidate oil and Mineral oil.

For the very low torques, designated as "no load tests", the oil sump temperature depends mainly on the oil viscosity at the operating temperature. At low speed, 1000 rpm, the operating temperatures are small (40 °C to 45°C), and thus the Mineral oil heats more than the Candidate oil because it has higher viscosity at those temperatures. At high speed, 3000 rpm, the operating temperatures are very similar (≈75°C), because the two oils have very similar viscosities at those temperatures.

In all the other tests, with input powers ranging from 14.78 kW to 142.31 kW (see Table A3), the operating temperature of the Mineral oil was always higher than the temperature of the Candidate oil, as shown in Figure A4.

This means that the friction power loss in the contact between the gear teeth and the corresponding friction coefficient are considerably smaller in the case of the Candidate oil, taking into consideration that the "no load power losses" are very similar for the two oils.

At high speed (3000 rpm) and high torque (453 Nm), corresponding to an input power of 142.3 kW, the operating temperature of the Mineral oil reaches, very quickly (112 minutes), the limiting temperature admissible for the test rig (180 °C) above which the slave gearbox seals may be damaged. The stabilised temperature will probably be above 220 °C which is not admissible for a mineral oil. In the same conditions, the Candidate oil reaches a stabilised temperature of 170 °C after 4 hours.

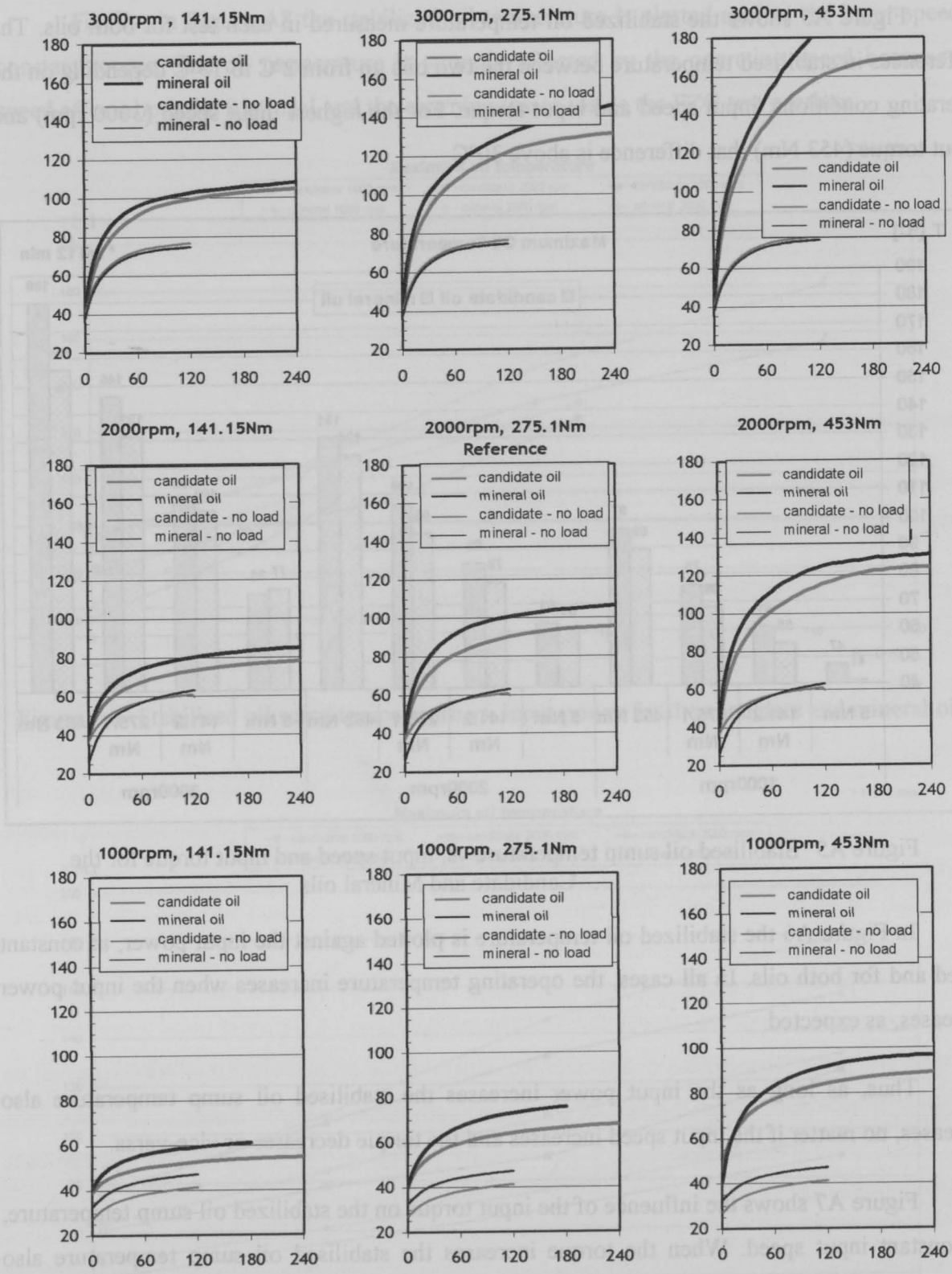


Figure A4 Oil sump temperature during tests. Candidate oil vs. Mineral oil (vertical – temperature in ºC, horizontal - time in minutes, gear wheel speed and torque indicated.

Figure A5 shows the stabilized oil temperature measured in each test for both oils. The differences in stabilised temperature between the two oils go from 2ºC to 15ºC depending on the operating conditions, input speed and input torque. For the highest input speed (3000 rpm) and input torque (453 Nm) that difference is above 30ºC

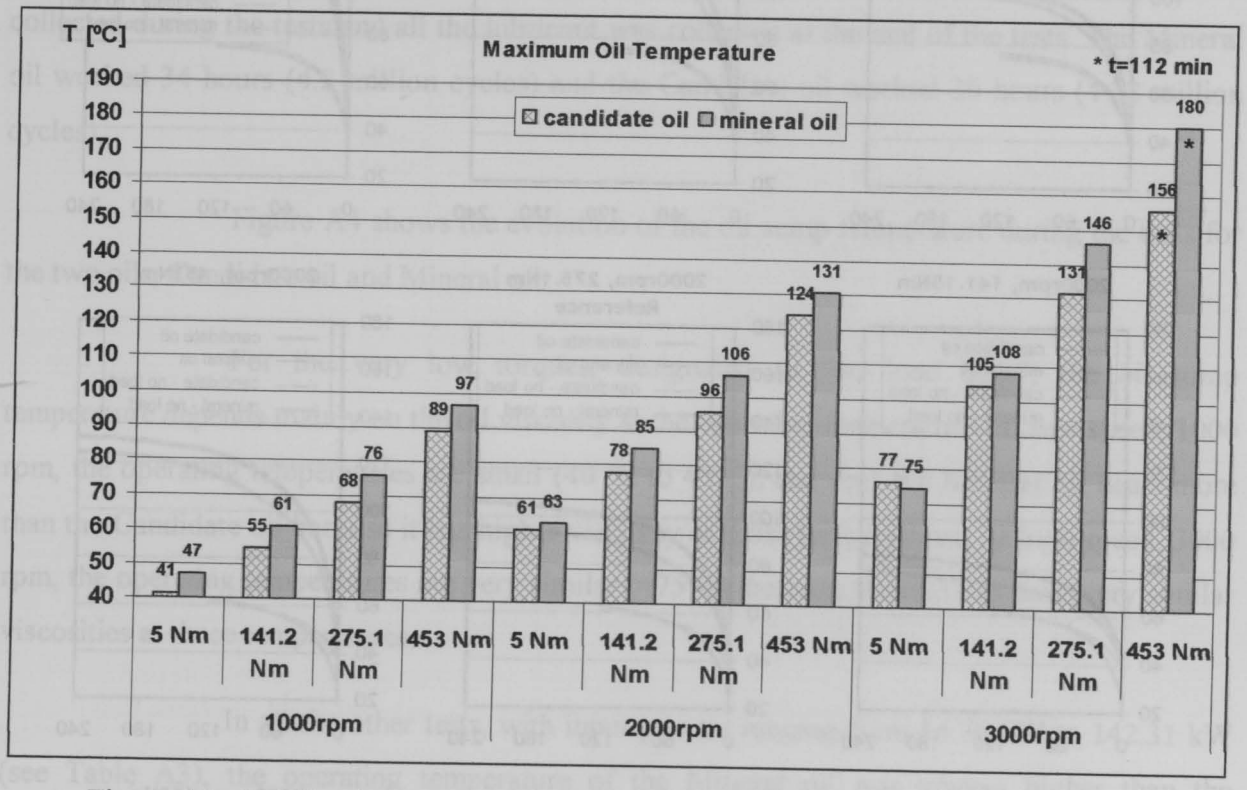


Figure A5 Stabilised oil sump temperature vs. input speed and input torque for the Candidate and Mineral oils.

In Figure A6 the stabilized oil temperature is plotted against the input power, at constant speed and for both oils. In all cases, the operating temperature increases when the input power increases, as expected.

Thus, as long as the input power increases the stabilised oil sump temperature also increases, no matter if the input speed increases and the torque decreases or vice-versa.

Figure A7 shows the influence of the input torque on the stabilized oil sump temperature, at constant input speed. When the torque increases the stabilised oil sump temperature also increases, almost in a linear way. Such increase depends on the input speed, being maximum at 3000 rpm.

Finally, in Figure A8 the stabilized oil temperature is plotted against the input speed at constant torque. The oil temperature is strongly affected by the operating speed because the speed affects both the frictional and the churning losses inside the FZG test gearbox.

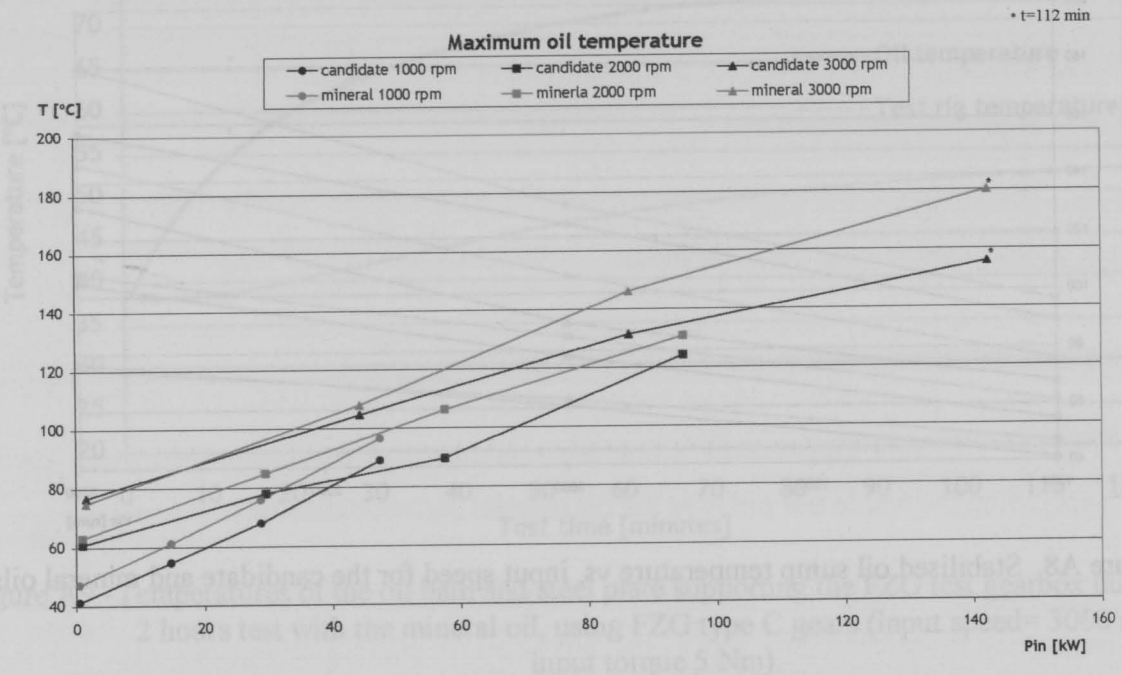


Figure A6 Stabilised oil sump temperature vs. input power for the candidate and mineral oils.

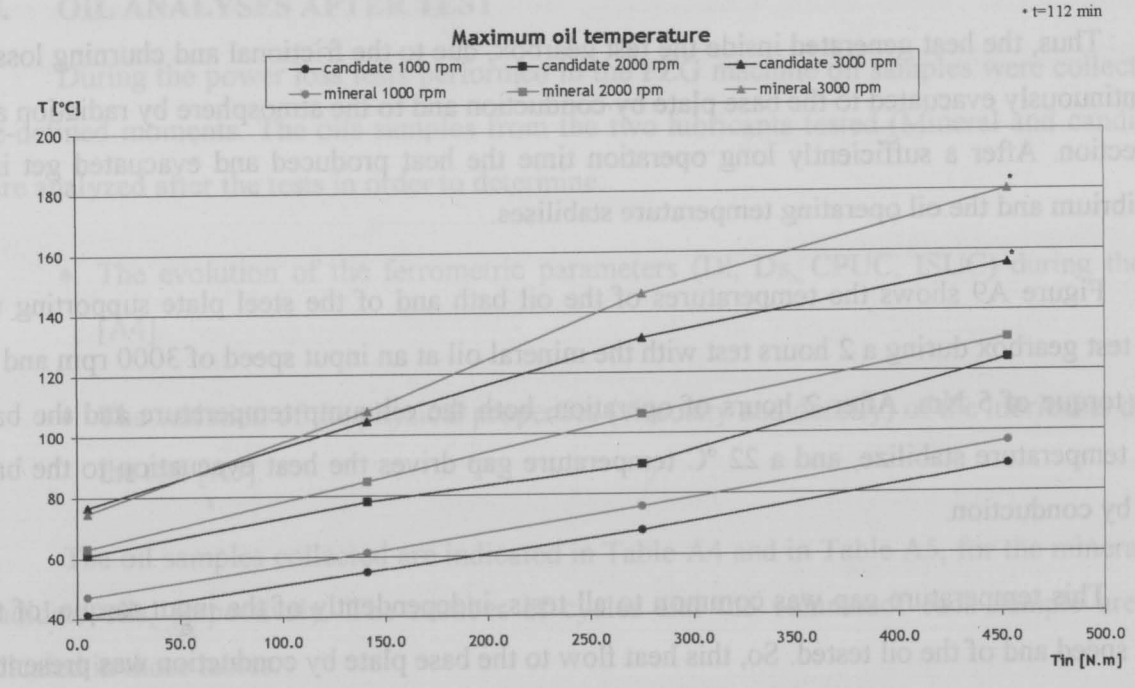


Figure A7 Stabilised oil sump temperature vs. input torque for the candidate and mineral oils.

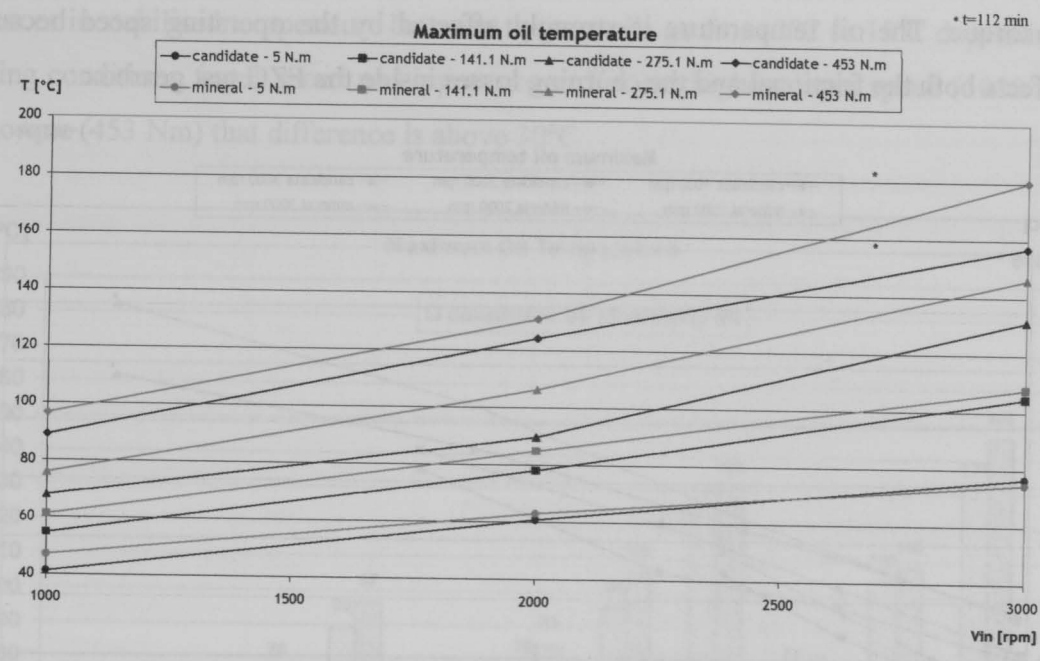


Figure A8 Stabilised oil sump temperature vs. input speed for the candidate and mineral oils.

The oil temperature inside the test gearbox of the FZG machine is always greater than the temperatures of the supporting plate and of the surrounding environment.

Thus, the heat generated inside the test gearbox, due to the frictional and churning losses, is continuously evacuated to the base plate by conduction and to the atmosphere by radiation and convection. After a sufficiently long operation time the heat produced and evacuated get into equilibrium and the oil operating temperature stabilises.

Figure A9 shows the temperatures of the oil bath and of the steel plate supporting the FZG test gearbox during a 2 hours test with the mineral oil at an input speed of 3000 rpm and an input torque of 5 Nm. After 2 hours of operation, both the oil sump temperature and the base plate temperature stabilize, and a 22 °C temperature gap drives the heat evacuation to the base plate by conduction.

This temperature gap was common to all tests, independently of the input torque, of the input speed and of the oil tested. So, this heat flow to the base plate by conduction was present in all tests.

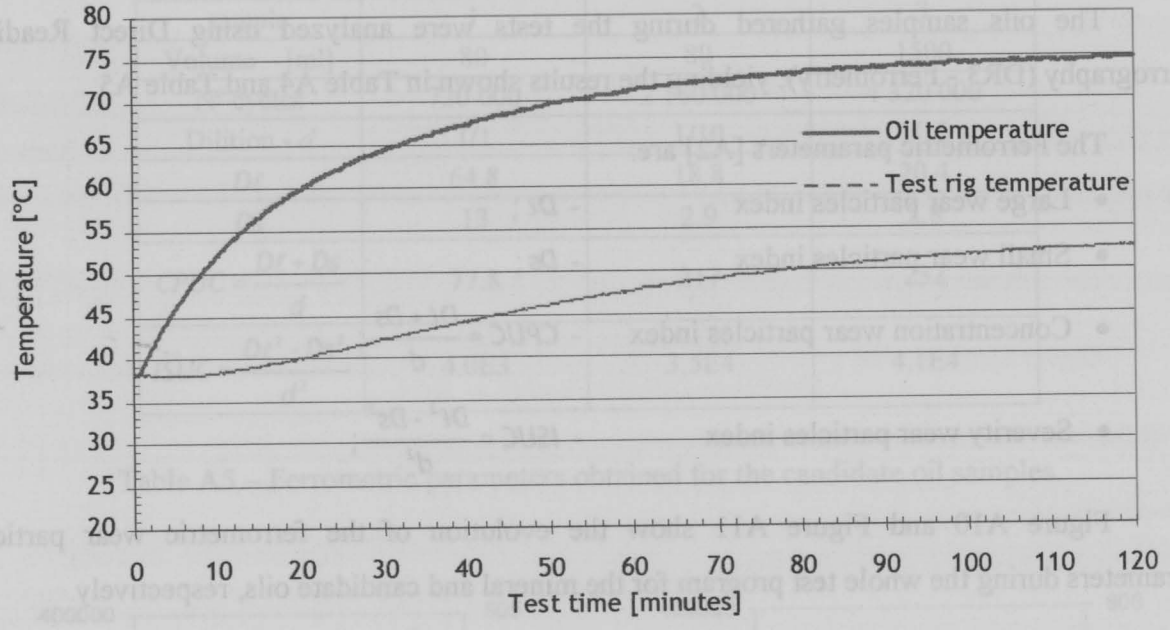


Figure A9 - Temperatures of the oil bath and steel plate supporting the FZG test gearbox during a 2 hours test with the mineral oil, using FZG type C gears (input speed= 3000 rpm, input torque 5 Nm).

A5. OIL ANALYSES AFTER TEST

During the power loss tests performed in the FZG machine oil samples were collected at pre-defined moments. The oils samples from the two lubricants tested (Mineral and candidate) were analyzed after the tests in order to determine:

- The evolution of the ferrometric parameters (Dl, Ds, CPUC, ISUC) during the test [A4].
- The variation of the physical properties (viscosity and density) of the lubricants during the test [A5].

The oil samples collected are indicated in Table A4 and in Table A5, for the mineral and candidate oils, respectively. The number of cycles and the volume of each sample are also indicated in those tables.

Analysis of the oil samples using ferrometry

The oils samples gathered during the tests were analyzed using Direct Reading Ferrography (DR3 - Ferrometry), yielding the results shown in Table A4 and Table A5.

The Ferrometric parameters [A2] are:

- Large wear particles index - $D\ell$;
- Small wear particles index - Ds;
- Concentration wear particles index - $CPUC = \frac{D\ell + Ds}{d}$;
- Severity wear particles index - $ISUC = \frac{D\ell^2 - Ds^2}{d^2}$;

Figure A10 and Figure A11 show the evolution of the ferrometric wear particle parameters during the whole test program for the mineral and candidate oils, respectively.

It is clear that the ferrometric wear particle parameters of the mineral oil are higher and grow faster than those corresponding to the candidate. It might also indicate that, inside the contact between the gear teeth, the film thickness is higher and the friction coefficient is lower, when lubricated with the candidate oil. At 2000rpm and 3000rpm the operating temperature of the mineral oil reached values above 130°C, which mineral oil is not prepared to stand. Consequently, gear wear was very high.

Mineral Oil			
Oil sample	1	2	3
Volume – [ml]	80	80	1500
N° cycles	600 000	2 040 000	3 816 000
Dilution - d	1/1	1/10	1/10
$D\ell$	96.6	34.6	61.6
Ds	28.2	5.8	6.4
$CPUC = \frac{D\ell + Ds}{d}$	124.8	404	680
$ISUC = \frac{D\ell^2 - Ds^2}{d^2}$	8.5 e3	1.2 e5	3.8 e5

Table A4 – Ferrometric parameters obtained for the mineral oil samples.

Candidate Oil			
sample	1	2	3
Volume – [ml]	80	80	1500
Nº cycles	720 000	2 160 000	4 320 000
Dilition - d	1/1	1/10	1/10
$D\ell$	64.8	18.8	20.4
Ds	13	2.9	2.8
$CPUC = \frac{D\ell + Ds}{d}$	77.8	217	232
$ISUC = \frac{D\ell^2 - Ds^2}{d^2}$	4.0E3	3.5E4	4.1E4

Table A5 – Ferrometric parameters obtained for the candidate oil samples.

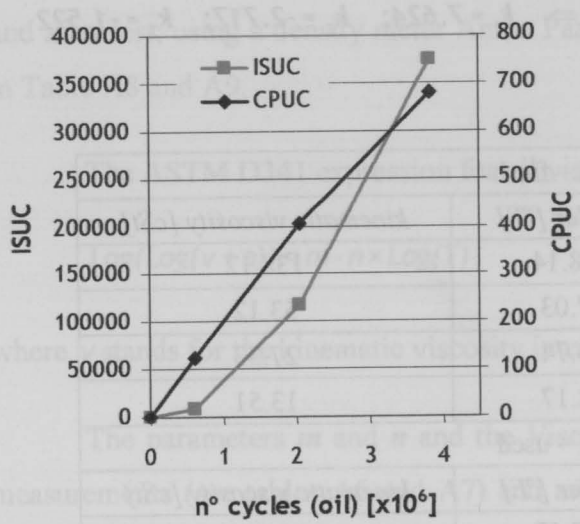


Figure A10 – Evolution of the CPUC index and of the ISUC index during the power loss gear tests with the mineral oil.

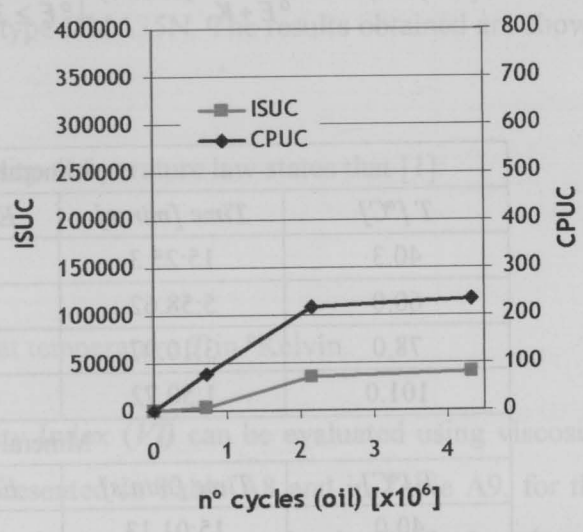


Figure A11 –Evolution of the CPUC index and of the ISUC index during the power loss gear tests with the candidate oil.

A6. LUBRICANTS PROPERTIES AFTER TEST

Before and after the power loss gear tests the lubricants viscosities were measured using an Engler viscometer, in accordance to standard IP 212/92 [A5]. The measuring temperatures, the time measured, the Engler degrees and the corresponding kinematic viscosities of both oils are presented in Table A6 and Table A7.

The relations between Engler degrees and Kinematic viscosity in cSt are as follows [A4]:

$$Engler\ Degrees = {}^{o}E = \frac{flow\ time\ of\ 200ml\ of\ lubricant}{flow\ time\ 200ml\ of\ water\ at\ 20^{o}C}$$,(time measured for water - 51s),

$$\nu(cSt) = k_1 \times {}^{o}E + \frac{k_2}{{}^{o}E + K_3}, \text{ where, } \begin{cases} {}^{o}E < 3 \Rightarrow k_1 = 14.867;\ k_2 = 75.568;\ k_3 = -6.198 \\ {}^{o}E \geq 3 \Rightarrow k_1 = 7.624;\ k_2 = -2.717;\ k_3 = -1.522 \end{cases}$$

Mineral oil – fresh			
T [ºC]	*Time [min:s]*	*Engler [ºE]*	*kinematic viscosity [cSt]*
40.3	15:25.3	18.14	138.12
60.0	5:58.62	7.03	53.12
78.0	3:10.97	3.74	27.33
101.0	1:50.72	2.17	13.51
Mineral oil – used			
T [ºC]	*Time [min:s]*	*Engler [ºE]*	*kinematic viscosity [cSt]*
40.0	15:01.13	17.67	134.54
66.8	4:42.78	5.54	41.60
101.0	1:55.56	2.27	14.47

Table A6 – Viscosity measurements for the mineral oil.

Candidate oil – fresh			
T [ºC]	*Time [min:s]*	*Engler [ºE]*	*Kinematic viscosity cSt*
39.4	11:07.12	13.08	99.49
60.5	4:59.84	5.88	44.20
78.0	2:52.9	3.37	24.26
99.0	1:53.82	2.23	14.13
Candidate oil – used			
T [ºC]	*Time [min:s]*	*Engler [ºE]*	*kinematic viscosity [cSt]*
39.9	10:11.91	12.00	91.22
68.5	3:37.10	4.26	31.46
101.0	1:52.0	2.20	13.77

Table A7 – Viscosity measurements for the candidate oil.

Density measurements of both lubricants were also made at several temperatures, before and after test, using a density meter Anton Paar, type DMA35N. The results obtained are shown in Table A8 and A9.

The ASTM D341 expression for oil viscosity-temperature law states that [1]:

$$Log(Log(v+a)) = m - n \times Log(T),$$

where v stands for the kinematic viscosity in ***cSt*** at temperature T in ºKelvin.

The parameters ***m*** and ***n*** and the Viscosity Index (***VI***) can be evaluated using viscosity measurements (see tables A6 and A7) and are presented in Table A8 and in Table A9, for the mineral and for the candidate oils, respectively, before and after test, that is, for fresh and used oils.

Mineral Oil			
ASTM D341 Parameters, Viscosity Index and density			
	Fresh oil	*Used oil*	*% difference*
m	9.0701	8.60448	/
n	3.5013	3.31626	/
a	0.7000	0.7000	/
ν_{40}	140.44	134.54	-4.20
ν_{100}	13.87	14.84	+6.99
VI	91	109	+19.78
ρ	895.8	898.0	+0.25

Table A8 – mineral oil: ASTM D341 parameters, Viscosity Index and density, before and after test.

Candidate Oil			
ASTM D341 Parameters, Viscosity Index and density			
	Fresh oil	*Used oil*	*% difference*
m	7.9545	7.6442	/
n	3.0678	3.9460	/
a	0.7000	0.7000	/
ν_{40}	96.82	90.64	-6.38
ν_{100}	13.81	14.08	+1.96
VI	141	156	+10.64
ρ	925.2	926.3	+0.12

Table A9 – candidate oil: ASTM D341 parameters, Viscosity Index and density, before and after test.

The viscosity at 40ºC of both lubricants show a small decrease (see Tables A8 and A9), indicating that they didn't suffered significant oxidation during the test time (36 hours), although the maximum temperature reached almost 180 ºC.

The viscosity at 100ºC of both lubricants show a small increase, 7% for the mineral and 2% for the candidate (see tables A8 and A9).

However, the variations in kinematic viscosity are always smaller than ±10%, usually taken as the limiting value.

Both lubricants show an increase of the Viscosity Index, since the viscosity at 40 °C decreased and the viscosity at 100 °C increased, for both oils. The increase in the Viscosity Index is significant, 20% for the mineral and 10% for the candidate (see tables A9 and A10). Density has not changed significantly either for mineral and candidate oils.

CONCLUSIONS

- The kinematic viscosities at 100 °C of the mineral oil and of the candidate oil are very similar. The density of the candidate is considerably larger than that of the mineral oil.

- The operating temperature of the mineral oil is always greater than that of the candidate oil, in power loss gear tests with input powers ranging from 14.78 kW to 142.31 kW. This means that the friction power loss in the contact between the gear teeth and the corresponding friction coefficient are considerably smaller in the case of the candidate oil.

- The type of lubricant, paraffinic mineral oil or biodegradable oil, doesn't influence the heat transfer to the base plate of the FZG machine by conduction and to the atmosphere by irradiation and convection.

- The analysis of oils samples, gathered during the tests using Direct Reading Ferrography (DR3 - Ferrometry), show that the index of wear particles concentration (CPUC) and the index of wear particles severity (ISUC) are significantly higher for the mineral oil than for the biodegradable.

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[A3] Chr. Changenet and M. Pasquier, "Power Losses and Heat Exchange in Reduction Gears: Numerical and Experimental results", VDI-Berichte, NR. 1665, 2002, 603-613.

[A4] T. M. Hunt, "Handbook of Wear Debris Analysis and Particle Detection in Liquids", Elsevier Science, UK, 1993.

[A5] IP 212/92 Standard, "Determination of Viscosity of Bitumen Emulsions – Engler Method", Institute of Petroleum, UK, 1992.

APPENDIX B – TRIBOLOGICAL BEHAVIOUR OF SELF-LUBRICANT GEAR COATINGS

INTRODUCTION

Gears are mechanical components of widely spread use in power and movement transmission. They are vital components of machines and their failure leads to an undesirable stop. Therefore, it is most interesting to extend gear service life, extend their power transmission ability, extend lubricant service life and diminish lubricant properties demands.

The current study focuses on self-lubricant coatings. It is intended to evaluate tribological behaviour of two types of coatings: MoST (MoS2/Titanium) and Graphite-iC (Carbon/Chromium).

B1. SELF-LUBRICANT COATINGS

Self-lubricant coatings are known and used for some time now in several industrial applications. The most important motives not to use those coatings in gears are:

a) possibility of deposition of coatings with poor adhesion;

b) some coatings do not present a friction coefficient low enough and do not provide protection to the contacting surface;

c) high price of coatings.

B2. MECHANICAL AND TRIBOLOGICAL PROPERTIES OF ANALYSED COATINGS

In order to evaluate mechanical and tribological properties of MoST and Graphite-iC coatings, namely, adhesion to the substrate, coating thickness and friction coefficient, Teer Coatings, Ltd (manufacturer of the coatings) performed a series of tests such as ball crater, Rockwell C measurements, scratch adhesion test, reciprocating wear test, pin on disc and Vickers hardness measurements.

These tests confirmed that both coatings have good adhesion to the substrate and low friction coefficients, thus showing their self-lubricant properties. Combining the results from the various tests creates good perspectives for the behaviour of the coatings in elastohydrodynamic service, like gear teeth. The measured friction coefficients were 0.04 for MoST and 0.06 for Graphite-iC.

B3. TWIN-DISC TESTS

These tests have two aims: evaluate the behaviour of the coatings in typical gear service conditions – high contact pressure, high slide to roll ratio, specific film thickness of about 0.5 and additive free lubricants – and identify the failure mechanisms of surface coatings in mixed and boundary lubrication conditions.

The tests revealed some important differences between both coatings and between spherical and cylindrical discs.

Spherical discs showed, for both coatings, greater wear due to having performed many more cycles but most of all, due to their roughness before coating being poorer than that of cylindrical discs.

Wear behaviour is very different from one type of coating to the other. For MoST, and for cylindrical discs, there is an improvement for high Slide To Roll Ratio. Spherical discs show two distinct levels. One of low wear for low Slide To Roll Ratio and another of high wear for high Slide To Roll Ratio.

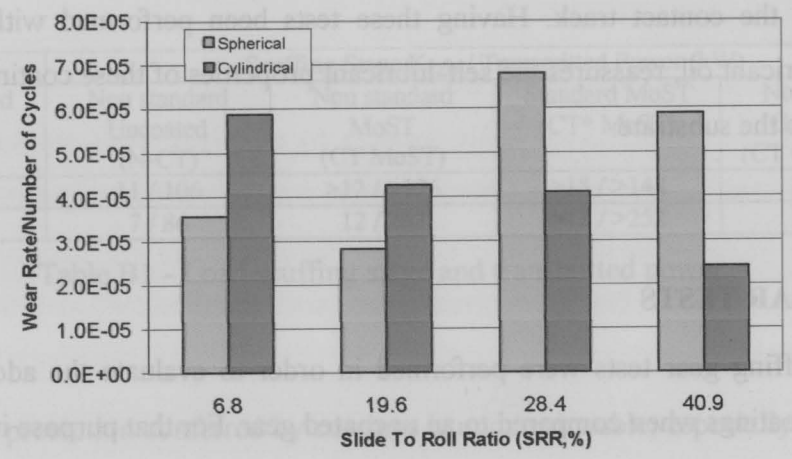


Figure B1 - Percentage of coating wear per cycle vs. slide-to-roll ratio in twin disc tests with MoST coated surfaces

In Graphite-iC discs, wear increases with slide-to-roll ratio up to 28.4% of SRR, decreasing from then on. Spherical discs have the opposite behaviour, with their wear decreasing up to 28.4% and increasing significantly from then on.

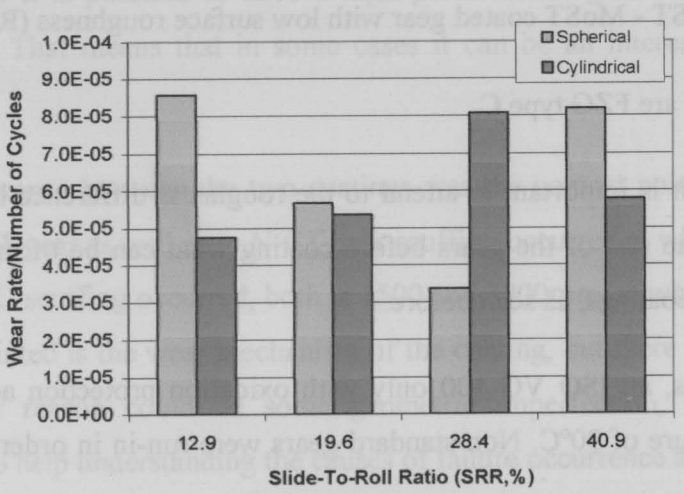


Figure B2 - Percentage of coating wear per cycle vs. slide-to-roll ratio intwin-disc tests with Graphite-iC coated surfaces

Wear mechanisms are very different from MoST coating to Graphite-iC. While in the first one there are localised detachments from the surface, originating small pitts, in the second one the most important mechanism is thin spalls detachment. Localised pitts in MoST, are usually very small and may not reach the substrate. Graphite-iC spalling occurs suddenly and

randomly along the contact track. Having these tests been performed with an ISO VG 22 additive free lubricant oil, reassures the self-lubricant properties of these coatings as well as their good adhesion to the substrate.

B4. FZG GEAR TESTS

FZG scuffing gear tests were performed in order to evaluate the additional protection ensured by the coatings when compared to an uncoated gear. For that purpose it was used:

- N-CT – uncoated gear with high surface roughness (R_a=2.4μm);
- CT MoST – MoST coated gear with high surface roughness (R_a=2.4μm);
- CT Graphite-iC – Graphite-iC coated gear with high surface roughness (R_a=2.4μm);
- CT* MoST - MoST coated gear with low surface roughness (R_a=0.4μm).

All gear sets are FZG type C.

First of all, it is important to attend to the roughness difference between standard gears (CT* MoST) and the rest of the gears before coating what can be of major importance in the performance of the coatings, as seen before.

In these tests, an ISO VG 100 only with oxidation protection additives was used at a monitored temperature of 90°C. Non standard gears were run-in in order to soften their surface roughness.

Tests were taken from stage 7 onwards until scuffing failure occurred. For each gear two tests were performed, one at 1500rpm and another at 3000rpm.

Scuffing stages and transmitted power for each test were as follows:

	Scuffing Stage K_{FZG} / Transmitted Power (kW)			
Rotation Speed (rpm)	Non standard Uncoated (N-CT)	Non standard MoST (CT MoST)	Standard MoST (CT* MoST)	Non standard Graphite-iC (CT Graphite-iC)
1500	11 / 106	>12 / >126	>13 / >148	12 / 126
3000	7 / 86	12 / 252	>12 / >252	12 / 252

Table B1 - Load scuffing stage and transmitted power.

Additional protection conferred by coatings is unquestionable, especially when analyzing the results of 3000rpm tests. Once again, geometric precision and surface roughness importance for the performance of the coatings comes out. CT* MoST gear, being of better quality proved more resistant to scuffing failure and contact fatigue, as only in non standard gears spalls were detached. It is nevertheless remarkable that CT MoST gear only failed in stage 12 at 3000rpm and passed that stage at 1500rpm. Taking into account the very severe conditions of stage 12, these results mean that it is possible to have a high performance gear starting from a lower manufacturing quality. That means that in some cases it can be an interesting solution, even considering the price.

The major difference between the two coatings was the contact point at which scuffing took place. In CT MoST gear as well as in N-CT gear scuffing occurred at wheel tip / pinion root while in CT Graphite-iC scuffing occurred, both at 1500 and 3000rpm, at wheel root / pinion tip. The probable cause pointed is the wear mechanism of the coating, but there is no further data to confirm that. Another reason could be some geometric imperfection, but it would need normalized gear tests to help understanding the causes of failure occurrence at that point.

The results obtained proved the good behaviour of these self-lubricant coatings in severe working conditions such as in elastohydrodynamic contacts – high contact pressures; high rolling speed; high slide-to-roll ratio; partial / boundary lubrication.

B5. FRICTION COEFFICIENTS

Afterwards, friction coefficients were calculated for the various gears using FPI (Friction Power Intensity) criteria and using Kelly formula to calculate N-CT friction coefficient. The result is, as expected, a significant reduction in the friction coefficient value for coated gears, when compared to the uncoated gear. Only that could explain the improvement in scuffing resistance. Improvements in the friction coefficients are about 6 to 19% at 1500rpm and about 42 to 45% at 3000rpm.

B6. Zr-N COATING

The coating used in this project is a Zirconium (Zr) and nitrogen (N) based coating. It is a bilayer coating with around 3μm thickness. Tests indicated good adherence of the coating. Coating is applied through a Physical Vapour Deposition process. Coated gears are previously nitrided. Nitriding process process has the following characteristics:

- Process temperature: 520ºC
- 60min of NH_3 at 6bar
- 30min of N_2 at 2bar
- 60min of NH_3 at 6bar
- 19h of N_2 at 2bar

CONCLUSIONS

These coatings showed good adhesion properties, good self-lubricant properties and good elastohydrodynamic behaviour. These results prove coatings to be an interesting alternative for more demanding gears as it may be less expensive coating them than grinding. An advantage of using coated gears is the additional protection in frequent start/stops, due to their self-lubricating properties. In the future, coating advantages can be of great interest when using biodegradable oils, additive free oils, or even when using less oil than the indicated for uncoated gears. Another set of tests could be carried out using only one wheel of the gear set coated.

APPENDIX C – ENERGETIC PERFORMANCE OF A GEARBOX

INTRODUCTION

Automobiles gearboxes are a well known and very used equipment. Therefore, they have had several improvements over the years, especially during the last decade, in their conception in aspects such as lubricants formulation, materials treatments or the use of surface coatings.

The emphasis of the current work is on these innovative aspects, aiming to evaluate the influence of additives in the energetic performance of lubricants, understand the influence of thin self-lubricant coatings and develop a theoretical model of the energetic performance of a gearbox.

C1. LUBRICANT OIL SELECTION

The selection of the lubricant oil based solely in elastohydrodynamic theory can reveal itself to be misleading regarding the lack of information of the additives contained in the lubricant and the temperature range in which they are effective. We can, thus, select a group of lubricants complying with the demands of EHD theory, although we do not know their energetic performance. In this study a new method, simple and fast, is proposed in order to select the lubricant leading to the lowest power dissipation.

Four lubricant oils have been selected, of very different bases and of very different applications, but all of them complying with the EHD theory demands for the gearbox:

- M1 – mineral oil, EP, for use in industrial gears, with good adhesion properties to metallic surfaces; ISO VG 150; VI 96;
- M2 – mineral oil, EP, for use in industrial gears, with additional protection against micro-pitting, oxidation and corrosion of surfaces; ISO VG 150; VI 95;
- S1 – motor oil, 100% synthetic base with a large additive pack; SAE 15w-50; VI 160;
- S2 – semi-synthetic base oil for use in automobile gears, with high anti-wear and anti-corrosion protection and high thermal stability; SAE 75w-90; VI 190.

The oils have been tested in the FZG Gear Test Rig at three rotational speeds and three load stages, combining nine different input powers from 28.2kW to 251.9kW. Tests were carried out until finishing the preset time or oil temperature reaching 180ºC.

Results showed differences between lubricants M2-S1 and M1-S2. Lubricants M2-S1 allow many more cycles before reaching 180ºC preset temperature and allow more demanding service conditions.

This method proved EHD theory to be insufficient for selecting a lubricant, as lubricants with similar properties and destined for the same kind of applications, like lubricants M1 and M2, may present very different energetic performances.

The most suitable lubricant for the transfer gearbox is, therefore, lubricant M2, as it presents more homogeneous performances either at low and high temperatures, and has the best performances. Lubricant oil S1 (motor oil), despite its good performance lubricating gears is not designed for this application, besides being much more expensive.

C2. GEARBOXES TEST RIG AND TRANSFER GEARBOX

The gearbox test rig used in the study was designed for gearboxes of variable dimensions. It's a back-to-back test rig with power circulation. The rig has various difficulties of operation, namely, in respect with the data acquisition of temperature, torque and speed. Another difficulty, solved during this work, was the impossibility of applying maximum torque due to backlash. The solution lies in applying a very small torque thus eliminating backlash.

The transfer gearbox here studied is used in vehicles with one drive axel allowing it to have two drive axels, plus an auxiliary power output. This transfer gearbox uses 5 gears mounted in three shafts. It has 2 speeds, both speed divisors, thus duplicating the kinematic possibilities of the main gearbox. The low speed range corresponds to the greater division and automatically selects all wheel drive. The high speed range corresponds to the smaller division.

C3. EXPERIMENTAL EVALUATION OF THE ENERGETIC PERFORMANCE OF THE TRANSFER GEARBOX

In order to evaluate the energetic performance of the transfer gearbox, tests were carried out at torques and rotational speeds corresponding to its actual service conditions. Gears used in these tests were coated using three different coatings: MoST, Graphite-iC and TiCx. Results analyses took into account overall performance of the gearbox, oil bath temperature during the tests and visual aspects of gear teeth.

The lubricant used in these tests was, as stated in 1, mineral oil M2.

The torque and rotational speed range were tested selecting 5 input rotational speeds (corresponding, roughly, to the 5 speeds of the main box) and for each of those applied torques of 100% of maximum torque and fractions of 50%, 25%, 15% e 10% of maximum torque. Limitations of the test rig don't allow torques below 100N.m, so some tests could not be performed.

Oil temperature at the beginning of tests was around 40ºC, and its evolution was free, as the rig doesn't allow temperature control. Estimated time for each test was 120 minutes, however, tests were stopped whenever oil temperature rose above 100ºC.

Test results helped understanding the variation of the gearbox performance with rotational speed and input torque and to compare the performance of the three selected coatings.

Results are not significantly different from high speed range to low speed range. Overall performance of the gearbox is high at low speeds (570 and 1400rpm) and is not very influenced by input power. As rotational speed increases, performance drops significantly. Speed at which maximum efficiency occurs increases with input power.

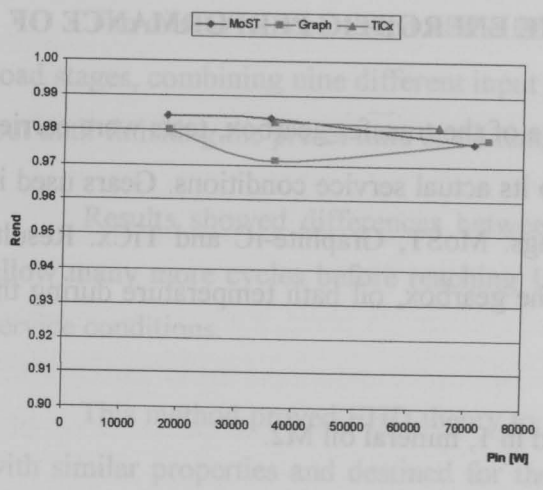


Figure C1 – Performance of transfer gearbox at 570 rpm – high range.

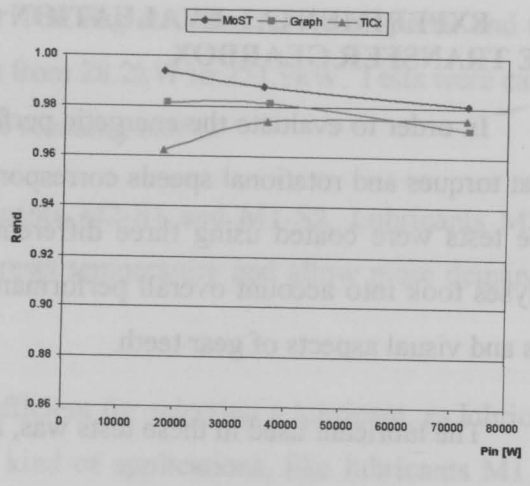


Figure C2 – Performance of transfer gearbox at 1400 rpm – high range.

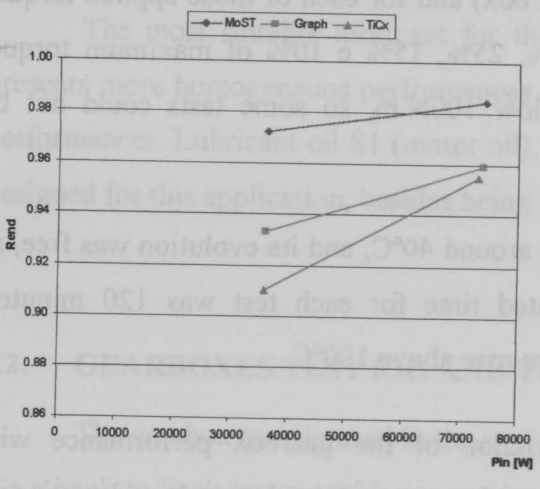


Figure C3 – Performance of transfer gearbox at 2200 rpm – high range.

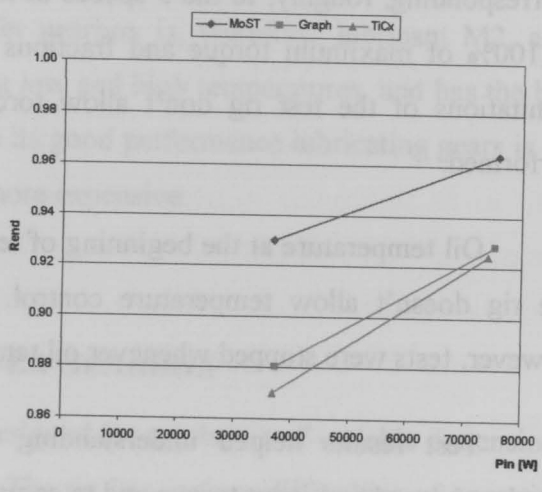


Figure C4 – Performance of transfer gearbox at 3000 rpm – high range.

MoST shows, overall, better behaviour than Graphite-iC and than TiCx. Temperature monitoring during tests yields the same conclusions, as MoST takes many more cycles to reach a certain gearbox service temperature than the other two coatings.

Inspection of gear teeth confirms the better performance of MoST, as they show very slight and uniform wear. Graphite-iC shows slight wear but irregular and TiCx shows severe and irregular wear, not tolerable in this application.

A series of complementary tests were carried out with MoST and Graphite-iC coated gears only for the high speed range. Over three million cycles were performed without running-in, although the first stages were performed at low speed and power. In these tests, once again, MoST energetic behaviour was better than Graphite-iC's. However, there was an improvement in Graphite-iC behaviour along the tests. MoST shows, for the same input power and the same oil bath temperature better performance, or, in other words, it takes many more cycles for MoST to reach the same temperature as Graphite-iC. MoST gears were used in another set of tests at rotational speeds of 1000, 2000 and 3000rpm and several input torques, with 30% oil volume reduction. Overall, 810 000 cycles were performed. It is only above 1000rpm that oil volume reduction is noticed, and is characterized by a significant performance improvement at high speeds due to lower viscous power dissipation.

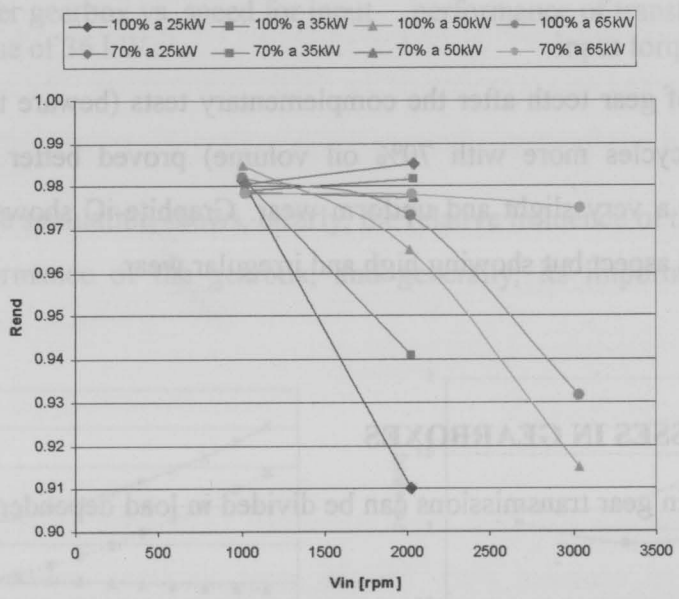


Figure C5 – Performance of transfer gearbox vs. test speed. Influence of oil volume.

The same is concluded if oil temperature variation is considered. For 1000rpm, 100% or 70% oil volume show the same variation. For 2000 and 3000rpm the temperature is significantly different between the two considered volumes, but is very similar for the two speeds.

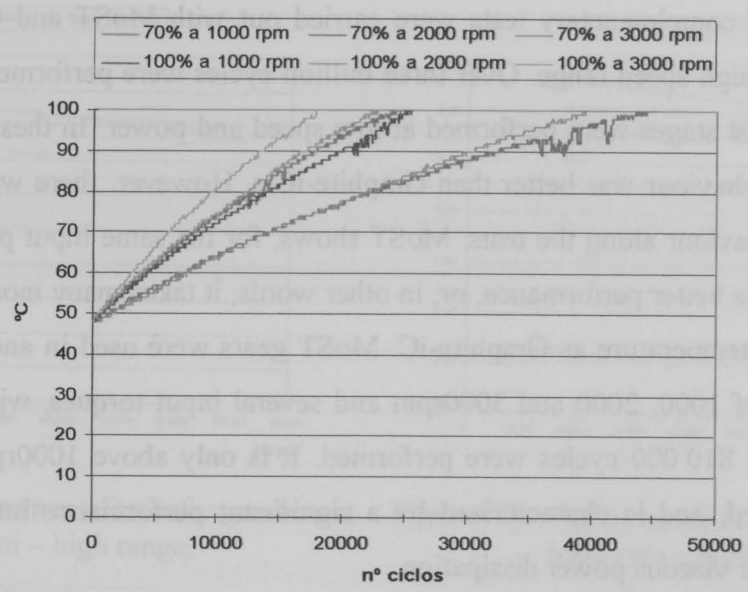


Figure C6 – Evolution of oil sump temperature for input power of 65 kW.

Observation of gear teeth after the complementary tests (beware that MoST gears have performed 810 000 cycles more with 70% oil volume) proved better resistance of MoST, showing this coating a very slight and uniform wear. Graphite-iC shows a good performance overall, with polished aspect but showing high and irregular wear.

C4. POWER LOSSES IN GEARBOXES

Power losses in gear transmissions can be divided in load dependent losses and those not dependent of load.

In the simulation only power losses in the gears due to friction and churning and power losses in the bearings also due to friction and churning were considered

The accuracy of the simulation comparing to the real behaviour of a gear is highly influenced by the method used to compute the friction coefficient. Efficiency values measured and those computed using the simulation are inevitably different, as the friction coefficients for the tested materials are not known. But considering that the efficiency of the various materials

shows greater differences among themselves than that presented between the simulation and some of them, it demonstrates the good prediction of the simulation.

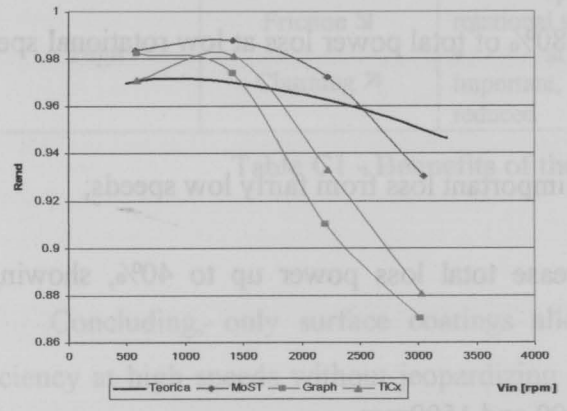


Figure C7 – Theoretical and experimental performance of transfer gearbox vs. speed for input torque of 36 kW.

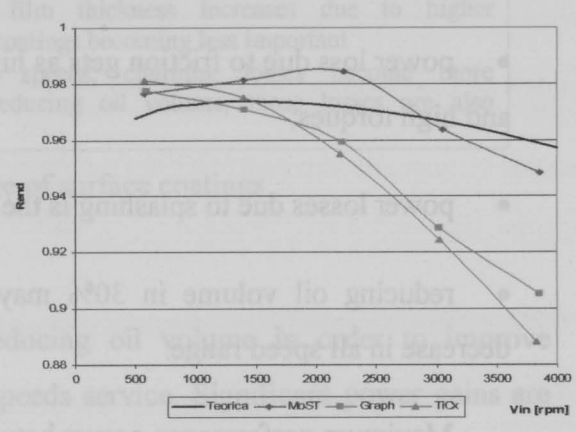


Figure C8 – Theoretical and experimental performance of transfer gearbox vs. speed for input torque of 75 kW.

In any case, the simulation shows, clearly, the relative influence of the various parameters for the overall performance of the gearbox, and generally, its importance in the energetic balance.

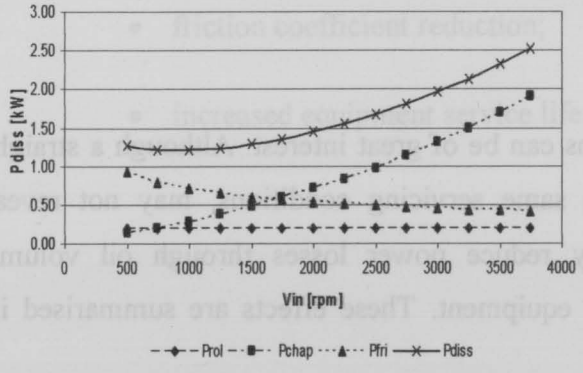


Figure C9 – Theoretical power loss at 40kW.

Prol – power dissipated in rolling bearings (churning + friction)
Pchap – power dissipated in the gears (churning)
Pfri – power dissipated in the gears (friction)
Pdiss – total dissipated power

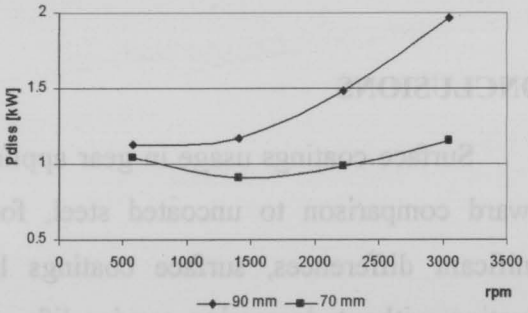


Figure C10 – Oil volume influence at 37 kW.

Thus:

- power loss in the bearings varies between 10 and 15% of total power loss for any given input power;
- power loss due to friction gets as high as 80% of total power loss at low rotational speeds and high torques;
- power losses due to splashing is the most important loss from fairly low speeds;
- reducing oil volume in 30% may decrease total loss power up to 40%, showing a decrease in all speed range.

Maximum performance occurs between 500 and 1500rpm.

Simulation results are affected by increasing of the speed, denoting underestimation of losses due to splashing and low precision of the rig for low torques. The model however, is able to predict experimental results up to 2200rpm with less than 1% error.

It would be necessary to correlate experimental and theoretical results in order to improve the prediction ability of the model.

CONCLUSIONS

Surface coatings usage in gear applications can be of great interest. Although a straight forward comparison to uncoated steel, for the same servicing conditions, may not reveal significant differences, surface coatings largely reduce power losses through oil volume reduction without decreasing service life of the equipment. These effects are summarised in Table C1:

Rotation Speed	Power Losses	Coatings effects
Low	Friction ↗ Churning ↘	• low specific film thickness compensated by the low friction coefficient of coatings • good adhesion of coatings and its good wear resistance stand for equipment functioning with reduced oil volume and service life
High	Friction ↘ Churning ↗	• specific film thickness increases due to higher rotational speed, coatings becoming less important • at high speeds, churning losses become more important, but reducing oil volume, those losses are also reduced

Table C1 - Bennefits of the usage of surface coatings.

Concluding, only surface coatings allow reducing oil volume in order to improve efficiency at high speeds without jeopardizing low speeds service. Significant power gains are achieved thus, using materials which allow:

- reduction of oil volume;
- using lubricants with lower percentage of additives;
- gear protection during start/stop of equipments;
- additional protection when specific film thickness is low;
- friction coefficient reduction;
- increased equipment service life.

APPENDIX D – WEAR AND SCUFFING OF AUSTEMPERED DUCTILE IRON

INTRODUCTION

ADI is a relatively new material which has been studied and used by some manufacturers as a replacement of steels in some applications. Results from these applications are usually not known, which has made that some doubts about its properties remain, and designers don't think of ADI has a reliable and possible material for demanding applications. This study focuses on the ability of ADI replacing steel in gear applications, namely by comparing their scuffing load capacity. ADI is, in fact, a family of materials rather than a material and is obtained through a heat treatment of nodular cast iron.

D1. MATERIAL CHARACTERIZATION

ADI comes from nodular graphite iron by performing a heat treatment called austemper.

The chemical composition of the ADI used in this work is shown in Table D1.

C	Si	Cu	Mn	P	S	Cr	Mo	Ni	Mg	Others
3.52	2.39	0.89	0.51	0.012	0.012	0.038	0.042	0.026	0.048	–

Table D1 - Nominal chemical composition (weight % ± 10%) of the nodular iron.

The treatment can be divided in three phases:

- austenitization;
- rapid cooling followed by isothermal transformation of austenite;
- cooling at ambient temperature.

Figure D1 shows a diagram of a typical heat treatment used for producing ADI.

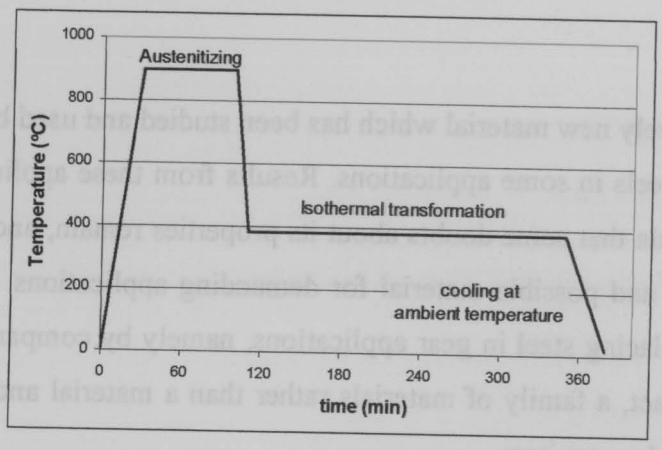


Figure D1 - Typical heat treatment for producing ADI.

As for all heat treatments, temperatures and times have to be carefully chosen to obtain the best properties at the least cost. In ADI treatment the austenitization phase is not the most critical one, especially in terms of the time involved, which can be longer than the necessary for fully transforming the material. Nevertheless, there are some considerations to be taken into account that otherwise will result in poor treatments. Temperature must not be too high as to induce excessive grain growth. Time should not exceed largely that necessary for transforming the whole component. Although it has been stated that time is not critical, exceeding the necessary time does not bring any benefits. In fact, especially if we are thinking at an industrial level, the shorter the time, the better. Besides, prolonging the time raises oxidation and induces grain growth. Temperatures to be used in this stage are from 850ºC to 950ºC, or from 815ºC to 850ºC if components are to be machined afterwards.

The isothermal transformation occurred during austemper is the key part in the whole process. Both temperature and time have a definitive role in the final properties of ADI. It is due to the differences in the properties obtained for different combinations of times and temperatures that ADI is a family of materials rather than a material. Another parameter to take into account is the chemical elements present in SG iron. This transformation is intended to take place in bainitic region, which is formed by two phases: ferrite and austenite.

Typically, there are two ranges of temperatures used in the process:

• from 350°C to 450°C – ADI contains up to 40% stabilized austenite at ambient temperature but if the time of the treatment is exceeded, there may exist carbides in austenite, which decreases mechanical properties;

• below 330-350°C – ADI presents better mechanical resistance but is less ductile.

Alloy elements are often used to improve the properties of ADI. The addition of alloy elements increases the time needed for obtaining similar structures to non alloyed ADI. Any slight variation in the percentage of those elements has a great influence on the final properties. In most cases, performances requested to ADI are only obtained by adding alloy elements and thus, nowadays, 60% to 70% of total ADI produced contains alloy elements.

The effects of some of the most important elements are:

• Molybdenum – allows obtaining bainitic structures in large section components.

• Copper – doesn't prevent the formation of pearlite and needs to be added in more quantities than molybdenum.

• Nickel – prevents the formation of pearlite, but even more quantities than copper are needed to obtain bainitic structures. Also improves components ductility.

• Silicon – improves traction resistance. Prevents the formation of carbides.

• Manganese – reduces ductility when ADI is transformed at high temperatures.

The effect of the austemper temperature on final properties of ADI is:

• yield strength – increases for temperatures up to 330°C and then decreases with increasing temperature;

• elongation – increases for temperatures up to 380°C and then decreases rapidly;

• hardness – decreases with increasing temperature.

Figure D2 shows graphite nodules in the surface of ADI.

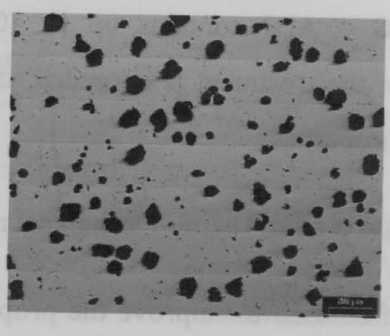


Figure D2 – Graphite nodules in the surface of ADI (x50)

Generally, for the production of ADI with high tenacity and high mechanical resistance, it should be used cast iron containing low quantities of elements that easily form carbides and containing high density of graphite nodules.

ADI has a contact fatigue resistance much superior to any other kind of cast iron, only exceeded by nitrided or case carburized steels. According to some authors, tensile fatigue of ADI may even be greater than that of the named steels.

In general, there are no records of problems or inconvenients in the use of ADI. The results of tests performed in gears are usually better than expected.

D2. TESTS WITH ADI GEARS

Tests with ADI gears followed standard DIN 51534-FZG A/8.3/90, which is used to determine the relative scuffing load capacity of oils. However, the present study intended to compare different materials. So, the same lubricant oil was used with two different kinds of ADI allowing thus to compare the scuffing resistance and the wear behaviour of those two different kinds of ADI.

Table D2 shows the temperatures of the heat treatments.

Phase	ADI 280	ADI 360
Austenitization	860°C	
Austemper	280°C	360°C

Table D2 - temperatures used in the heat treatments of the gears. ADI 280 refers to the gear transformed at 280°C and ADI 360 refers to the gear transformed at 360°C.

The most important properties of the lubricant oil used in all the tests are:

- kinematic viscosity @40°C = 68 mm2/s
- kinematic viscosity @100°C = 8.9 mm2/s
- density = 888 Kg/m3

With these two gears, three tests were performed. Two with ADI 280 and only one with ADI 360 due to the fracture of some teeth during the first test, preventing the use of the reverse tooth flank. Both ADI 280 and ADI 360 were tested with oil temperature uncontrolled and starting at 90°C. The other test with ADI 280 was performed with controlled temperature of 150°C.

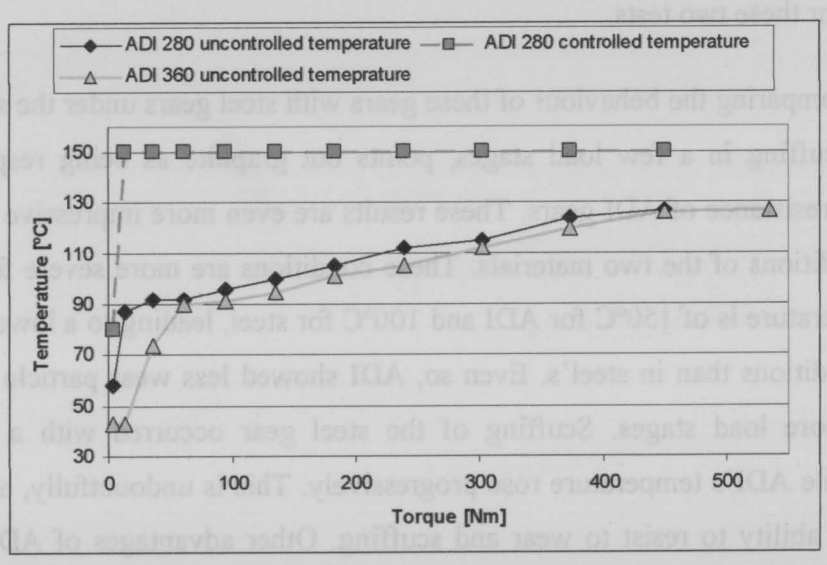


Figure D2 - Oil sump temperature at the end of each load stage.

The temperature at the end of each load stage along with the visual inspection of the gears gives:

- scuffing occurred at load stage 11 for both tests with ADI 280.

• In the uncontrolled temperature test, a slight increase in vibration level was noticed at load stage 10, and scuffing occurred at load stage 11.

• In the controlled temperature test, there was no sign of malfunctioning before load stage 11, at which scuffing occurred.

• Scuffing in the teeth flanks tested at 150ºC was more severe than with uncontrolled temperature.

• Although having broken some teeth, ADI 360 did not suffer scuffing. Some scars could be seen, but scuffing criteria was not reached.

• Temperatures of both tests with uncontrolled temperature were very similar. This could indicate similar load dependent power loss.

• Other parameters, such as film thickness, wear particle index and flash temperature have similar values for these two tests.

When comparing the behaviour of these gears with steel gears under the same conditions, which suffer scuffing in a few load stages, points out graphite as being responsible for the increased wear resistance of ADI gears. These results are even more impressive when analysing the service conditions of the two materials. These conditions are more severe for ADI than for steel. The temperature is of 150ºC for ADI and 100ºC for steel, leading to a lower film thickness in ADI test conditions than in steel's. Even so, ADI showed less wear particle index, and was able to pass more load stages. Scuffing of the steel gear occurred with a sudden rise of temperature while ADI's temperature rose progressively. This is undoubtfully, a very clear sign of this material ability to resist to wear and scuffing. Other advantages of ADI over steel are being 10% lighter and damping vibrations, resulting in less noise level.

After the tests in the FZG test rig, samples of the gears were observed at the microscope. This analysis confirmed the efficiency of the heat treatments. The good quality of the base cast iron was also confirmed. There were found a large number of cracks underneath the surface that grow along the graphite nodules. Despite this large number of cracks, the number of surface

craters was much smaller. The service surfaces that did not experienced scuffing appeared normal and with moderate roughness.

CONCLUSIONS

Being lighter than steel, having a much greater wear resistance, being much cheaper than steel and being possible to obtain in a wide range of mechanical properties, ADI is a very suitable replacement of steel where hardness or tensile strength are not very demanding. In those cases, case carburized steel is the most reliable material.

Further investigation is needed to achieve a good correlation between temperatures, times and properties. Only those tests, performed and confirmed by various investigations, may give designers confidence in using ADI. One thing, though, must be always kept in mind when analyzing any results: each kind of ADI has different mechanical properties.

APPENDIX E – MICRO-PITTING TESTS

INTRODUCTION

Micro-pitting is a contact fatigue wear phenomenon that is observed in combined rolling and sliding contacts operating under elastohydrodinamic lubrication (EHL) or mixed EHL/Boundary lubrication conditions. Micro-pitting can be regarded as fatigue failure with a net of cracks close to the surface which typically starts during the first 10^5 to 10^6 stress cycles. The cracks propagate at a shallow angle to the surface forming micro-pits with characteristic depth in the range of 5-10μm. The micro-pits coalesce to produce a continuous fractured surface with a dull mate appearance. In case the surface cracks propagate deeper into the material, pitting and spalling can be initiated. [E1, E2]

Micro-pitting is known to be influenced by operating conditions such as temperature, load, speed, sliding and specific film thickness and by lubricant additives and surface material as well. Table E1 shows the factors that are reported to influence micro-pitting, their range of influence and suggestions for preventing the problem. [E1, E3]

Table E1 - Factors influencing micro-pitting, range of influence and suggested remedies. [E1, E3]

Influencing Factor	Range of influence	Suggested Remedy
gear surface roughness	1:3 (from 6μm to 3μm)	reduce to 0.3μm
material, heat treatment	1:2.8	retained austenite
lubricant viscosity	1:2	use highest practical viscosity
lubricant additive chemistry	1:2 (equal viscosity of base oil)	use properly selected additives
coefficient of friction	1:1.7	reduce the coefficient of friction
speed	1:1.3	run at high speed
oil temperature	1:1.3 (ΔT=20K)	reduce oil temperature

Not considering lubricant chemistry, the key factors to produce micro-pitting are mixed lubrication and combined rolling and sliding. By a thick EHL film and smooth surfaces micro-pitting can be eluded.

In the case of gears with parallel axis, pure rolling exists only at the pitch point. Above and below this point there is a combination of rolling and sliding, the sliding speed increasing when the contact point moves away from the pitch line. Figure E1 represents two teeth at the

beginning and at the end of the engagement, showing the directions of the rolling speed (R) and sliding speed (S).

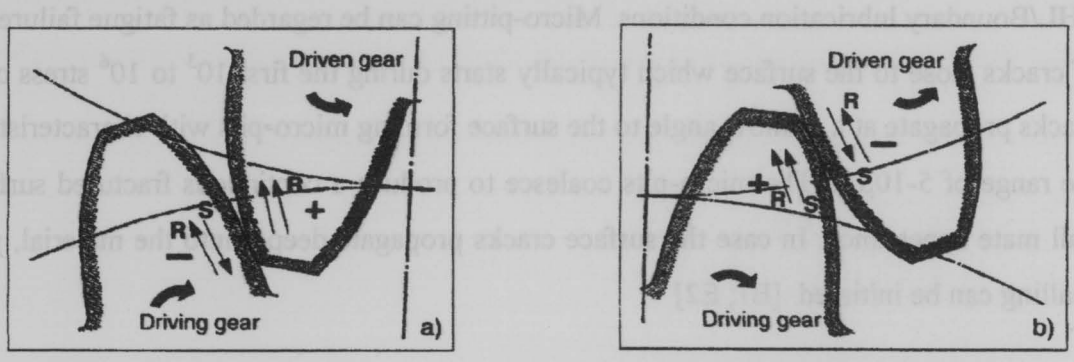


Figure E1 - Rolling and sliding directions. a) Beginning of engagement. b) End of engagement. [E4]

Micro-pitting may occur almost anywhere on the gear tooth. However, the contact condition where sliding and rolling directions are opposite is considered to be the most critical. This contact condition always occurs below pitch circle both for the driving and the driven gears. Micro-pitting is also often found at addendum due to high sliding and at local high spots in the surface topography associated with high stresses.

The progression of micro-pitting may eventually result in macro-pitting. There are also reported cases where the micro-pitting progresses up to a point and stops, sometimes described as a form of running-in or stress relief. Although it may appear innocuous, such loss of metal from the gear surface causes loss of gear accuracy, increased vibration, noise and other related problems. The metal particles released into the oil may be too small to be picked up by filters, but large enough to damage tooth and bearing surfaces.

E1 TEST CONDITIONS

The definition of the test conditions was based on the FVA research project Nr. 54/I-IV [E5] and the DGMK-FZG micro-pitting short test (abbreviated as GFKT-C/8.3/90) [E6]. The GFKT-C/8.3/90 is a short term test that is able to classify candidate lubricants analogous to the

FVA-FZG-micro-pitting test. The later being a well established, standard test but requiring relatively high costs and being quite time consuming.

Four sets of tests were conducted. Two gears of case-carburized steel were used, named C1 and C3. The same gear lubricant was used on both flanks A and B of each gear. Two ADI gears were used for one set of tests and one coated gear was used for another set of tests.

Material used in the case carburized gears is DIN 20Mn Cr5 steel. Gears are carburized, quenched in oil and annealed.

Coding of the tests:

Test set 1:

CC1-M – 20MnCr5 carburized steel + mineral oil;

CC1-E – 20MnCr5 carburized steel + ester biodegradable.

Test set 2:

CC2-M – 20MnCr5 carburized steel + mineral oil;

CC2-E – 20MnCr5 carburized steel + ester biodegradable.

Test set 3:

ADI-M – ADI + mineral oil;

ADI-E – ADI + ester biodegradable.

Test set 4:

(Zr+N)-M – nitrided steel + Zr-N coating + mineral oil;

(Zr+N)-E – nitrided steel + Zr-N coating + ester biodegradable

The first set of tests followed the test conditions proposed by GFKT-C/8.3/90. For the second set of tests the lever arm was changed from 0.35m to 0.50m resulting in higher torques and contact pressures. In these tests, the highest load stage – K9 - was run twice to allow a better understanding of the evolution of micro-pitting. At the end of each load stage, including running-in, an oil sample was collected. The third set of tests was ran under load stages K5 and K7 to better suite ADI's contact pressure resistance. The fourth set of tests did not ran load stage K9

due to the high grade of deterioration of the coating found at the end of load stage K7. At the end of load stages K5, K7, K9-1 and K9-2 gears were unmounted from the shafts for measuring roughness, weighing and photographing. Fresh oil was used after unmounting the gears.

The failure criteria and the comparison of the two lubricants attended the outlines given both in DGMK and FVA micro-pitting tests.

Test conditions are presented in Table E2 and Table E3. In Table E2, 0.35 represents the conditions of test sets 1 and 4. 0.50 represents the conditions of test set 2 conditions are less severe than steel's because ADI cannot work under such high levels of contact pressure.

Table E2 - Test conditions of micro-pitting tests.

		K3 - running in		K7		K9-1 / K9-2 b)	
		pinion	wheel	Pinion	wheel	pinion	wheel
Temperature [°C]		80		90		90	
Torque [Nm] a)	0.35	28.8	43.2	132.5	198.8	215.6	323.4
	0.50	35.3	53	183.4	275.1	302.0	453
Rot speed [1/min]		2250	1500	2250	1500	2250	1500
Vt [m/s]		8.3					
Hertzian Stress [Mpa]	0.35	510		1094		1395	
	0.50	564		1286		1657	
Power [kW]	0.35	6.8		31.2		50.8	
	0.50	8.3		43.2		71.2	
Duration [h]		1	1	16	16	16	16
cycles [10^3]		135	90	2160	1440	2160	1440

a) The same name is given to load stages corresponding to different torques because the dead weights used to create those torques are the same. The difference lies in applying them with a 0.35m lever arm resulting in lower torques or 0.50m lever arm resulting in higher torques.

b) Coated nitrided gear did not ran load stage K9 due to the high grade of deterioration found on load stage K7.

Table E3 - Test confitions of ADI tests.

	K3 - running in		K5		K7	
	pinion	wheel	Pinion	wheel	pinion	wheel
Temperature [ºC]	80		90		90	
Torque [Nm]	28.8	43.2	70	105	132.5	198.8
Rot speed [1/min]	2250	1500	2250	1500	2250	1500
Vt [m/s]	8.3					
Hertzian Stress [Mpa]	510		795		1094	
Power [kW]	6.8		16.5		31.2	
Duration [h] / cycles [10^3]	1 / 135	1 / 90	16 / 2160	16 / 1440	16 / 2160	16 / 1440

FGZ Gear Test Rig and test lubricants are described in Appendix A.

E2 TEST GEARS AND LUBRICATION REGIME

The gears used in these tests are not of the same gear accuracy grade as proposed in the standards [E5, E6]. Therefore, lubricants cannot be classified in GF-Classes proposed by FVA. However, the main objective of the tests is to compare the two lubricants with one another rather than to classify them in those classes. As the lubricants will be tested with similar gears, the results are reliable.

Test gears available differ from standard gears in quality yielding different surface roughness and specific film thickness. Table E4 shows the differences of gear quality and roughness and Table E5 shows differences in specific film thickness.

Table E4 - Gear quality grade and surface roughness.

	Gear quality grade	Roughness Ra of tooth flanks
Standard gear	5	0.5 ± 0.1 μm
20MnCr5 case carburized	9	0.96 ± 0.06 μm
	9	1.10 ± 0.08 μm
	9	0.93 ± 0.05 μm
	9	0.81 ± 0.04 μm
ADI austempered at 300°C	9	1.02 ± 0.02 μm
	9	0.62 ± 0.07 μm
Nitrided steel + Zr-N coating	9	1.13 ± 0.07 μm
	9	0.95 ± 0.03 μm

Table E5 - Specific film thickness for standard test and carried out tests.

		$\frac{R_{a1}+R_{a2}}{2}$ [μm]	R_q [μm]	ν @ 90°C [cSt]	T1 [Nm]	p_c [Mpa]	Λ [a)]			
							A	C2	I	dif(%)
Mineral oil	Std gear	0.50	0.990	18.56	28.8	510	0.230	0.319	0.385	
	CC1-M	0.96	1.748	18.56	28.8	510	0.131	0.181	0.218	-43.4
	CC2-M	0.93	1.628	18.56	35.3	564	0.138	0.190	0.230	-40.3
	ADI-M	0.9	1.64	18.56	28.8	510	0.139	0.193	0.233	-39.5
	(Zr+N)-M	1.21	2.11	18.56	28.8	510	0.108	0.150	0.181	-53.0
Ester oil	Std gear	0.50	0.990	18.57	28.8	510	0.170	0.233	0.280	
	CC1-E	1.10	1.952	18.57	28.8	510	0.086	0.118	0.142	-49.3
	CC2-E	0.81	1.487	18.57	35.3	564	0.111	0.153	0.183	-34.6
	ADI-E	1.0	1.75	18.57	28.8	510	0.096	0.132	0.159	-43.2
	(Zr-N)-E	0.9	1.64	18.57	28.8	510	0.103	0.141	0.169	-39.7

a) Λ – specific film thickness evaluated in:

A – beginning of engagement;

C2 – lowest point of single tooth contact;

I – pitch point

The values shown in Table E5 refer to the roughness at the beginning of the test and torques of running in stage. At the beginning of load stage 7 (after running in), the conditions are a bit more favourable. Nevertheless, the specific film thickness is much lower than it would be if using standard gears. Specific film thickness is different for the two oils because of the effect of piezoviscosity. The differences shown refer to the respective specific film thickness with

standard gears. Only points below pitch line are presented, since it is the area where micro-pitting is more prone to appear.

We are working at boundary lubrication, what can carry some problems with wear and adhesion, moreover for higher torques.

E3 FAILURE CRITERIA

Both in DGMK and FVA micro-pitting tests, failure criterion is assessed by the average profile deviation. In the FVA test, other parameters, namely, micro-pitted area and weight loss are used to assess failure load stage. In the scope of the present tests, there is no equipment for measuring the profile deviation. Therefore, micro-pitted area and weight loss along with ferrometry and roughness of tooth flanks were used for comparing the two lubricants. A thorough visual inspection of the teeth flanks was also carried, with the help of a camera and microscope. Results were compared with FVA failure criteria for comprehension of the differences in the test conditions.

As it was said, FVA proposes a method for converting test results, which is shown in Table E6, but test gears have much higher surface roughness and much lower gear quality, thus not giving results comparable with standard, even attending to this conversion. It is interesting to understand, once again, the effect of surface roughness and lubricant viscosity on test results and, more important, on the overall gear service.

Table E6 – Approximate values for converting test results in the micro-pitting test. [E5]

Surface roughness Ra	Lubricant: ISO VG				
	32	46	100	220	460
0.3	+2…+3	+2	+1	+0.5…+1	+0.5
0.7	-2…-2	-2	-1	-0.5…-1	-0.5

Example: for a lubricant of viscosity grade ISO VG 46, a micro-pitting test with test gears with a flank roughness of Ra = 0.3 μm determines a failure load stage which is on average about two load stages higher than in a micro-pitting test with test gears with a flank roughness Ra = 0.5 μm.

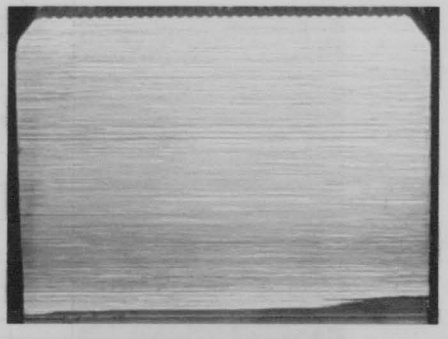
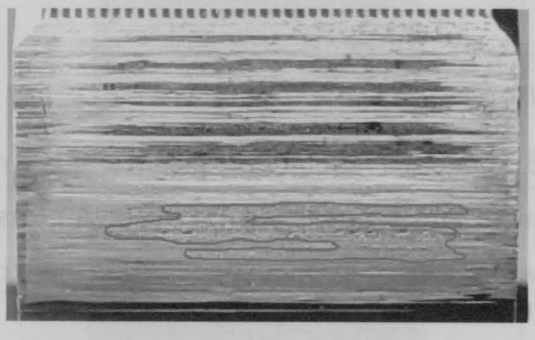
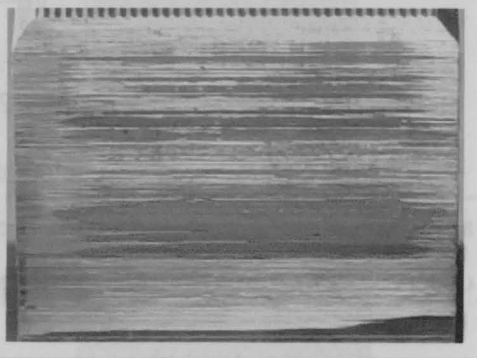
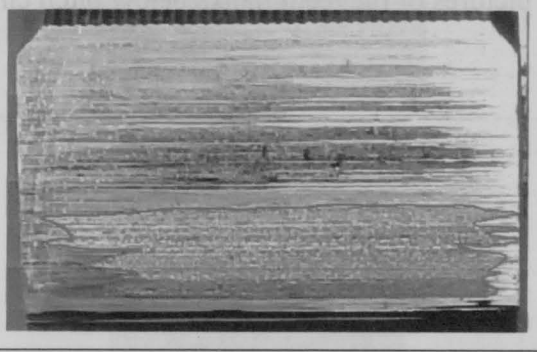
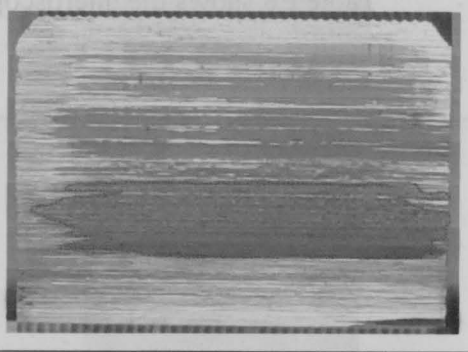
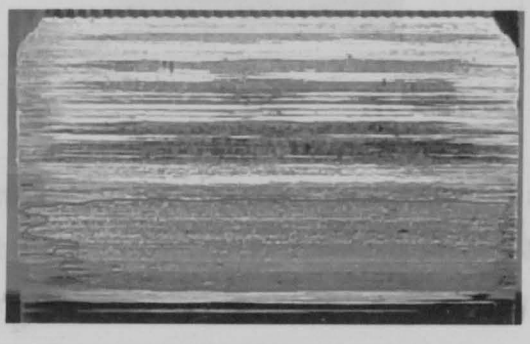
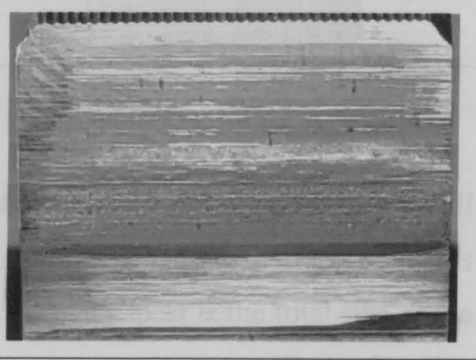
When analyzing the results, the pinion is the focus element, since it performs 1/3 more stress cycles than the wheel. For that reason, micro-pitting appears more rapidly in the pinion. The results presented, namely, photographs, micro-pitted area and surface roughness reports to pinion. Weight loss is presented both for pinion and wheel, especially when comparing with ferrometry. Ferrometry, obviously reports to the gear pair.

E4 TEST RESULTS OF TEST SET 1

Visual aspect of teeth flanks

Assessment of micro-pitting failure is a difficult task to perform by visual inspection. It is often reported that with the gears mounted on the shafts, it is almost impossible to detect micro-pitting. Light of high intensity is needed to distinguish the grey stained area that micro-pitting resembles. In these tests, that same difficulty was experienced. To overcome the problem, a camera and a microscope were used to photograph the teeth. Table E7 shows the pictures of the teeth before test and after each load stage.

Table E7 - Pictures of pinion teeth before test and after each load stage. (20MnCr5 case carburized lubricated with mineral and ester oils.)

	CC1-M	CC1-E
ORIGINAL		
K7 T1 = 132.5Nm n1 = 2250rpm P = 31.2kW		
K9-1 T1 = 215.6Nm n1 = 2250rpm P = 50.8kW		
K9-2 T1 = 215.6Nm n1 = 2250rpm P = 50.8kW		

Micro-pitted area is surrounded in red for easier reading. Figure E2 shows the percentage of micro-pitted area relative to the total active flank area.

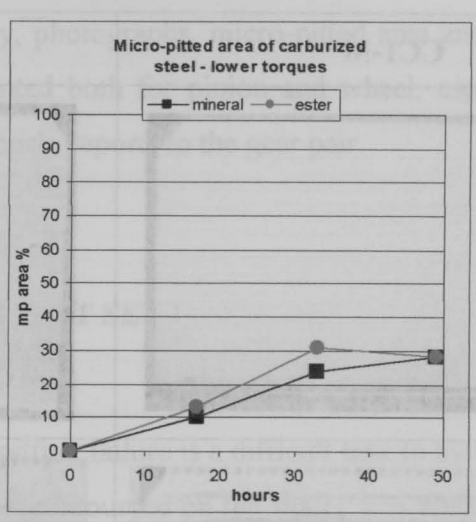
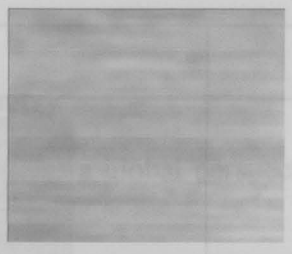
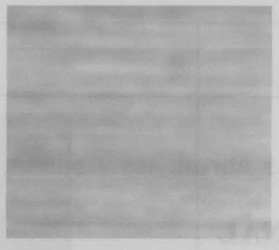
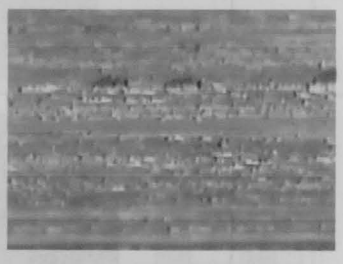
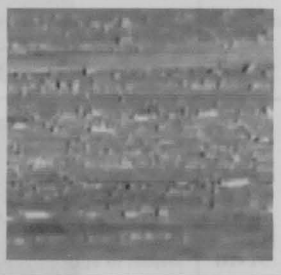
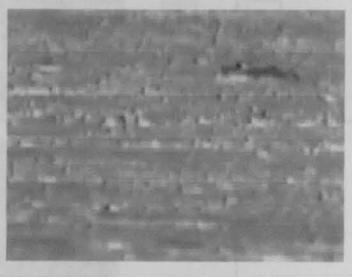
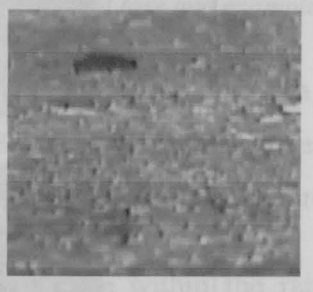
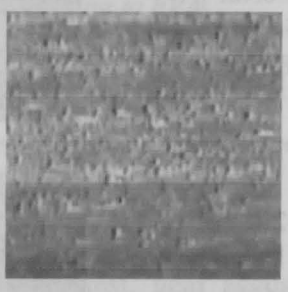


Figure E2 - Percentage of micro-pitted area.

Micro-pitted area was calculated from observation of the pictures. It is a rough, yet valid, approximation. The evolution of the micro-pitted area is similar for both gears. The tendency observed is for micro-pitting to fill the whole area of the tooth flank below pitch circle.

Table E8 shows the microscope pictures of the teeth flanks below pitch circle.

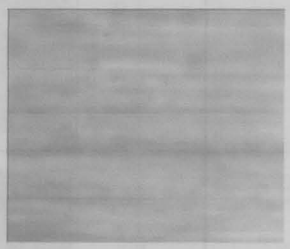
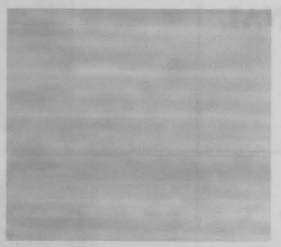
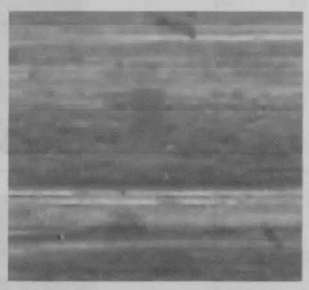
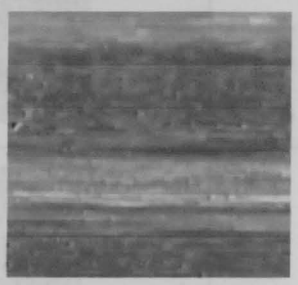
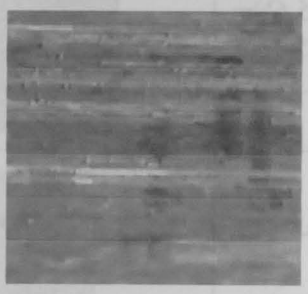
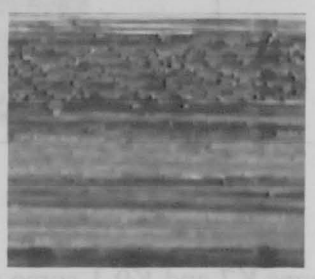
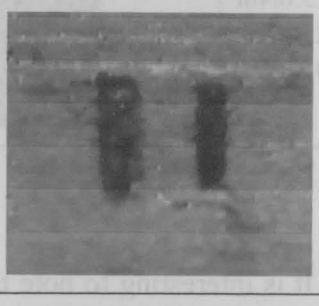
Table E8 – Video-microscope pictures of the teeth flank below pitch circle. Magnification 100x. (20MnCr5 case carburized lubricated with mineral and ester oils.)

	CC1-M	CC1-E
ORIGINAL		
K7 T1 = 132.5Nm n1 = 2250rpm P = 31.2kW		
K9-1 T1 = 215.6Nm n1 = 2250rpm P = 50.8kW		
K9-2 T1 = 215.6Nm n1 = 2250rpm P = 50.8kW		

It is interesting to note that after load stages K7 and K9-1 some grinding marks are still visible within the micro-pitting area. Some pits in CC1-M are larger than those in CC1-E. This could indicate more severe wear in CC1-M, but the most important aspect is the total size of the area where micro-pitting appears. If the pits don't grow into a macro pit or a spall, the fact that they are bigger is not very important.

The next table shows the microscope pictures of the teeth flanks above pitch circle.

Table E9 – Video-microscope pictures of the teeth flanks above pitch circle. Magnification 100x. (20MnCr5 case carburized lubricated with mineral and ester oils.)

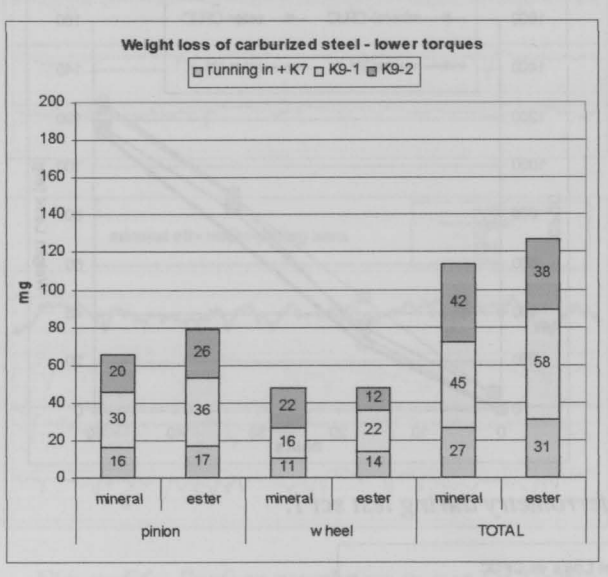
	CC1-M	CC1-E
ORIGINAL		
K7 T1 = 132.5Nm n1 = 2250rpm P = 31.2kW		
K9-1 T1 = 215.6Nm n1 = 2250rpm P = 50.8kW		
K9-2 T1 = 215.6Nm n1 = 2250rpm P = 50.8kW		

The pictures above pitch circle don't contribute to assess micro-pitting. Rather, it is used as a means of examining the wear experienced by the gears and of understanding the weight loss of each gear. In a standard test, no marks in the direction of the sliding speed would be tolerable. In these tests that is acceptable, because there is no intention of classifying the oils in GF-classes.

These observations show that candidate oil does not offer the same protection against wear and adhesion resulting in very clear marks above pitch circle. This fact will also reflect in the weight loss.

Weight loss and ferrometry

The weight loss gives a global perspective of the severity of the wear during each load stage.



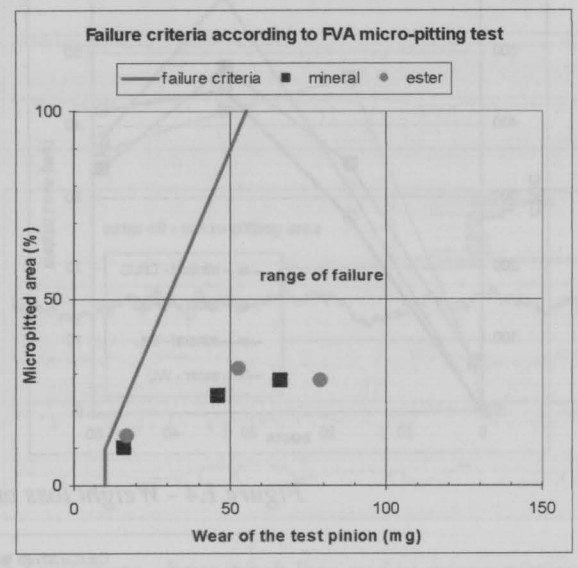


Figure E3 - Weight loss and example of failure criteria according to FVA.

The weight loss is very similar for both oils, despite a slight advantage for CC1-M. The weight loss helps to understand the observations of the teeth flanks. While CC1-M showed larger pits, CC1-E showed more severe wear above pitch circle.

Although all results are within the range of failure of FVA test, gear accuracy and roughness does not allow classifying oils directly by that criterion. Still, the results of load stage K7 are very close to comply with the criterion, confirming the good performance of the oils. This can be explained by the microscope pictures, where grinding marks could still be seen, predicting very low micro-pitting wear.

The oil samples were analysed using Direct Reading Ferrography (DR3 - Ferrometry) and the following ferrometric parameters were calculated:

- Large wear particles index - $D\ell$;
- Small wear particles index - Ds ;
- Concentration wear particles index - $CPUC = \frac{D\ell + Ds}{d}$;

Figure E4 shows the evolution of concentration wear particle index per load stage and cumulative and respective weight loss.

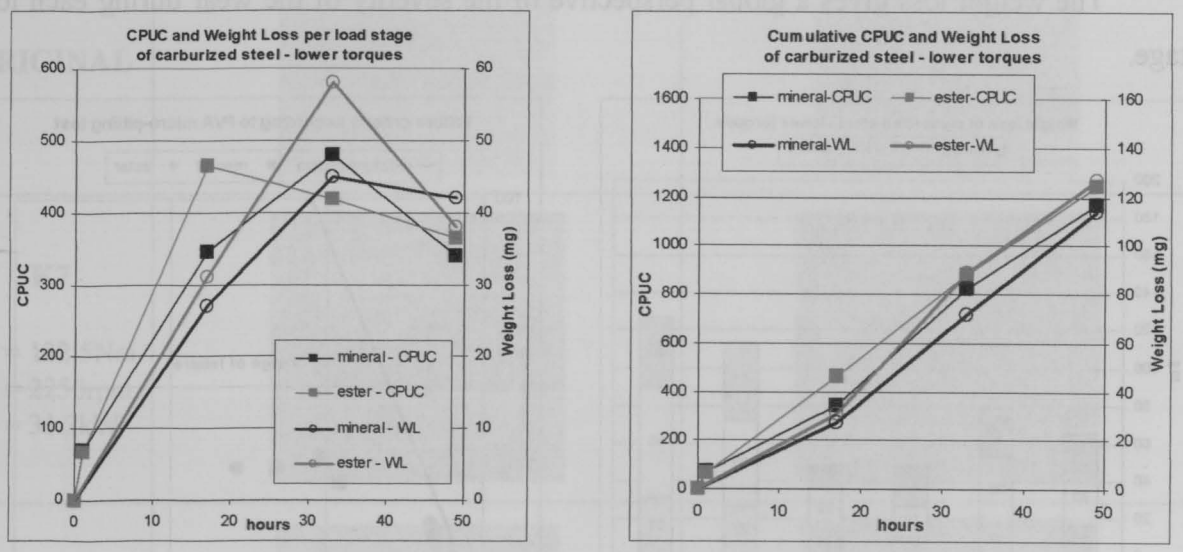


Figure E4 - Weight loss and ferrometry during test set 1.

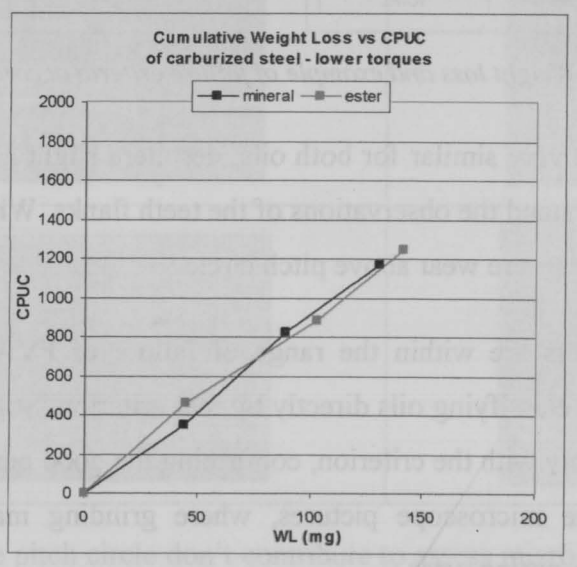


Figure E5 - Particle index vs weight loss.

The weight loss and ferrometry correlate very well, indicating reliable oil samples and analysis. Again, very similar results are found for both oils.

Surface roughness

Surface roughness was evaluated above and below pitch line. These two measures help to confirm the observations of the surface condition above and below pitch line.

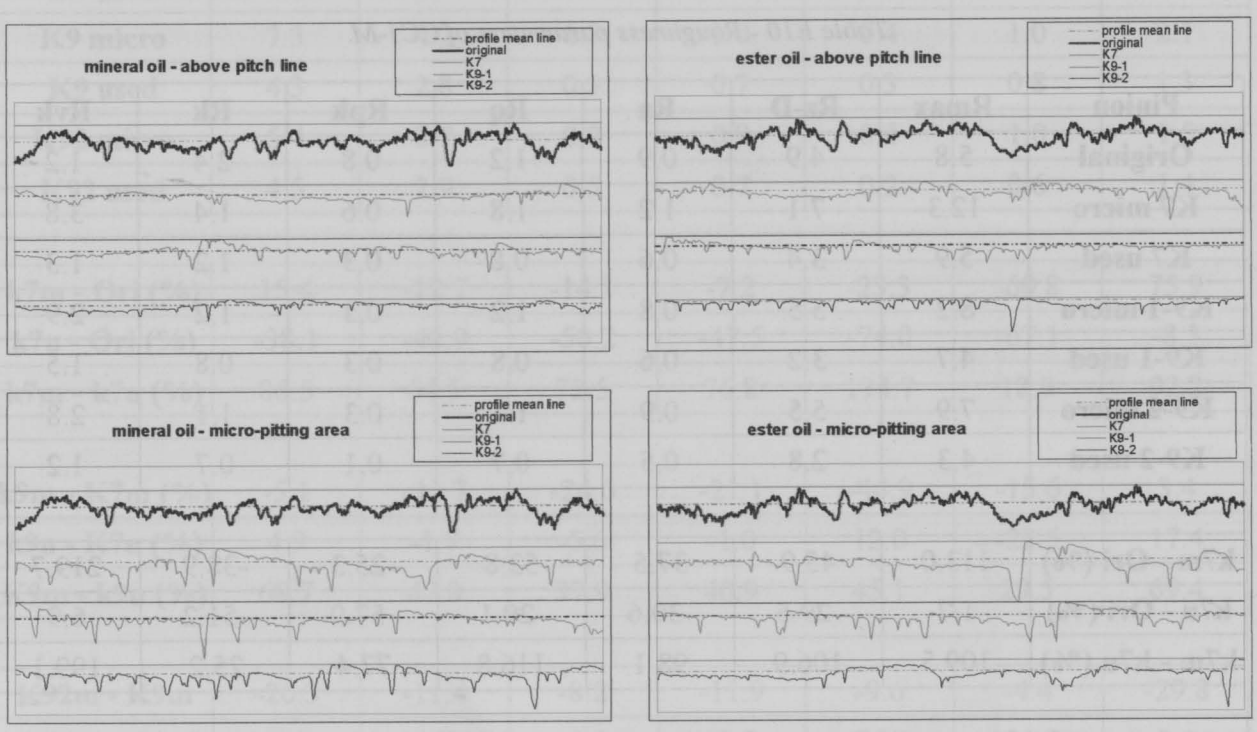


Figure E6 - Surface roughness in test set 1 after each load stage above pitch line and in micro-pitting area.

Above pitch line, the profiles of both pinion surfaces are very similar. The surface becomes much more flat over time, although after load stage K9-2, CC1-E appears to have more valleys than CC1-M. Despite these valleys being shallow, this could indicate that wear and adhesion are more significant for the gear lubricated with ester. The opposite behaviour is found below pitch line, where micro-pitting occurs. In this area, the profile of CC1-M agrees with microscope observations of larger pits found in the gear lubricated with mineral oil.

Roughness parameters help to quantify these evolutions. The removal of the highest peaks is indicated by the decreasing of Rpk. Increasing of Rvk indicates the formation of pits. In CC1-M, roughness parameters such as Ra, Rq or RzDIN are much lower above pitch circle than below pitch circle. They are also lower than the original surface, while below pitch circle they

remain almost the same. In CC1-E those parameters are more similar in the whole flank and smaller than in the original surface. This is in accordance with the roughness profiles shown above. Table E10 and Table E11 show a complete list of these parameters. In the tables, *original* means original surface, *micro* means below pitch line and *used* means above pitch line.

Table E10 - Roughness parameters of CC1-M.

Pinion	Rmax	Rz-D	Ra	Rq	Rpk	Rk	Rvk
Original	5.8	4.9	0.9	1.2	0.8	2.4	1.2
K7 micro	12.3	7.1	1.2	1.8	0.6	1.4	3.8
K7 used	5.9	3.4	0.6	0.8	0.3	1.2	1.3
K9-1 micro	8.2	5.5	0.8	1.2	0.3	1.2	2.9
K9-1 used	4.7	3.2	0.6	0.8	0.3	0.8	1.5
K9-2 micro	7.9	5.5	0.9	1.3	0.3	1.1	2.8
K9-2 used	4.3	2.8	0.5	0.7	0.1	0.7	1.2
k7m - Ori (%)	113.0	45.9	37.5	53.8	-25.3	-38.9	219.7
k7u - Ori (%)	1.7	-29.5	-30.6	-29.1	-57.9	-51.2	6.9
k7m - k7u (%)	109.5	106.9	98.1	116.8	77.4	25.2	199.1
k9m - K7m (%)	-33.4	-22.2	-31.9	-33.5	-49.9	-19.8	-24.6
k9u - K7u (%)	-20.1	-5.4	-7.8	-9.7	-23.6	-31.7	17.9
K9m - k9u (%)	74.8	70.1	46.4	59.9	16.4	47.0	91.2
K92m - K9m	-3.7	0.5	6.8	3.8	-1.6	-4.0	-4.2
K92u - K9u	-8.4	-12.3	-7.2	-9.5	-48.7	-17.2	-18.7
K92m - K92u	83.6	94.9	68.4	83.3	123.3	70.3	125.5

Table E11 - Roughness parameters of CC1-E.

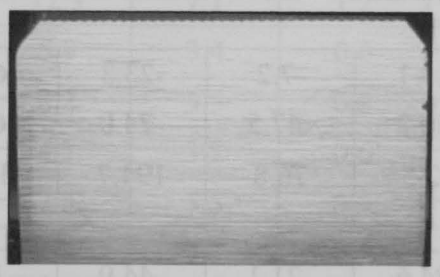
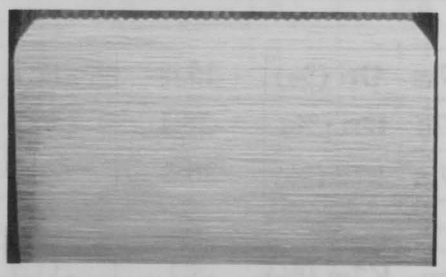
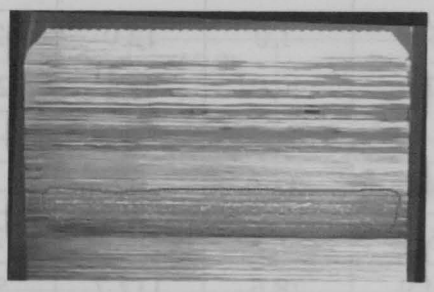
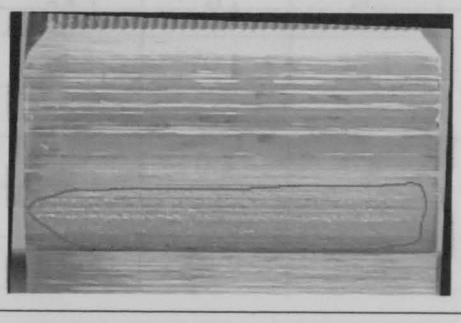
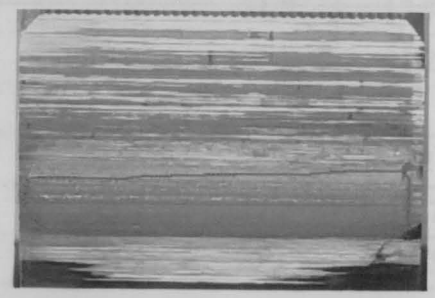
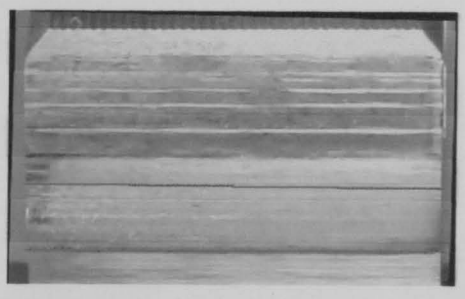
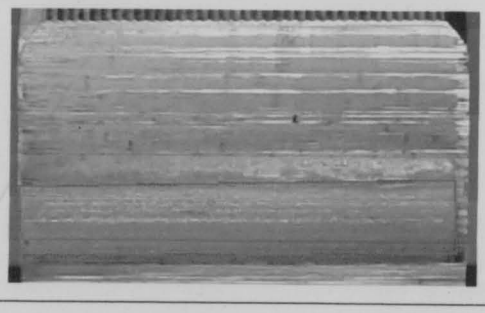
Pinion	Rmax	Rz-D	Ra	Rq	Rpk	Rk	Rvk
Original	6.6	5.4	1.1	1.4	1.0	3.0	1.2
K7 micro	7.7	4.8	1.0	1.3	0.7	1.2	2.1
K7 used	4.1	2.9	0.6	0.7	0.3	1.0	1.1
K9 micro	7.3	4.2	0.7	1.0	0.4	1.0	2.1
K9 used	4.3	2.8	0.5	0.7	0.3	0.8	1.3
K92 micro	5.4	3.7	0.7	0.9	0.4	1.0	1.5
K92 used	4.5	2.9	0.5	0.7	0.2	0.6	1.4
k7m - Ori (%)	15.4	-12.7	-14.1	-7.2	-23.3	-60.8	75.9
k7u - Ori (%)	-38.1	-46.9	-50.2	-47.5	-74.0	-67.1	-8.5
k7m - k7u (%)	86.5	64.5	72.5	76.8	194.7	18.9	92.2
k9m - K7m (%)	-5.1	-11.7	-24.6	-21.1	-44.9	-15.6	3.4
k9u - K7u (%)	4.9	-1.7	-5.6	-1.0	12.0	-22.5	17.4
K9m - k9u (%)	68.7	47.8	37.9	40.9	45.1	29.5	69.4
K92m - K9m	-26.3	-11.4	-8.2	-11.9	-9.6	-4.4	-29.8
K92u - K9u	4.1	3.7	-6.2	-3.5	-39.8	-21.2	9.1
K92m - K92u	19.5	26.3	34.9	28.6	117.7	57.2	9.0

E5 TEST RESULTS OF TEST SET 2

Visual aspect of teeth flanks

For test set 2 the same data was collected. The teeth flanks aspect is shown in Table E12.

Table E12 - Pictures of pinion teeth before test and after each load stage. (20MnCr5 case carburized lubricated with mineral and ester oils.)

	CC2-M	CC2-E
ORIGINAL		
K7 T1 = 183.4Nm n1 = 2250rpm P = 43.2kW		
K9-1 T1 = 302.0Nm n1 = 2250rpm P = 71.2kW		
K9-2 T1 = 302.0Nm n1 = 2250rpm P = 71.2kW		

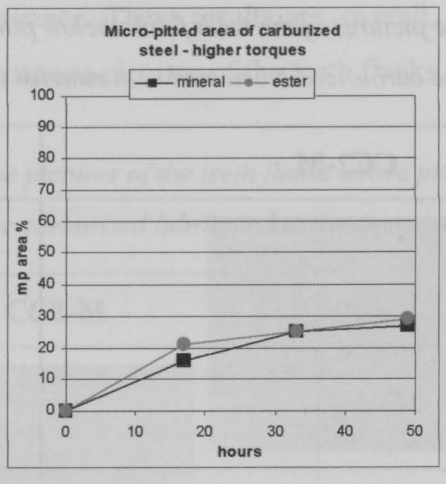
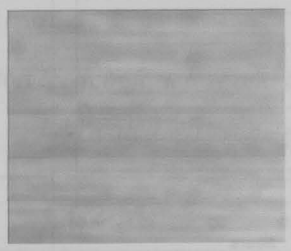
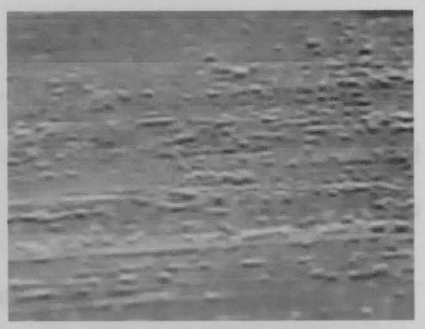
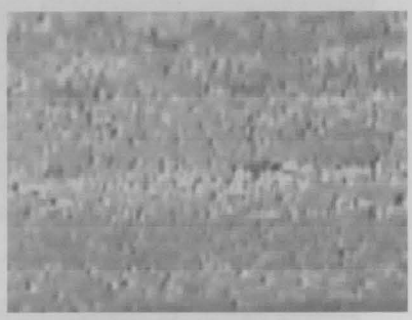
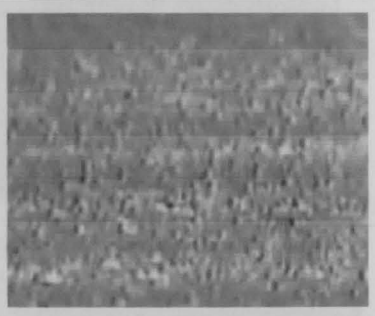


Figure E7 – Percentage of micro-pitted area.

In these tests, the micro-pitting area grows more rapidly, but in the end, the percentage of the tooth flank affected by micro-pitting is the same as in the previous tests, the area below pitch circle.

Table E13 shows the microscope pictures of the teeth flanks below pitch circle.

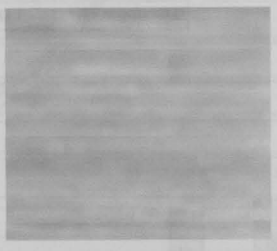
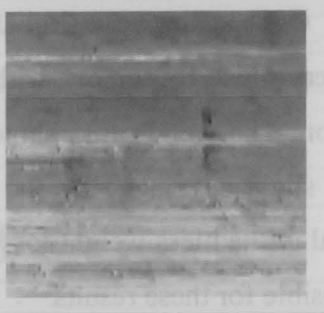
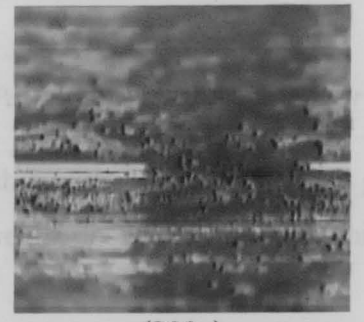
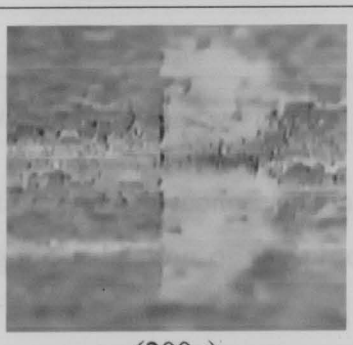
Table E13 – Video-microscope pictures of the teeth flanks below pitch circle. Magnification 100x. (20MnCr5 case carburized lubricated with mineral and ester oils.)

	CC2-M	CC2-E
ORIGINAL		
K7 T1 = 183.4Nm n1 = 2250rpm P = 43.2kW		
K9-1 T1 = 302.0Nm n1 = 2250rpm P = 71.2kW		
K9-2 T1 = 302.0Nm n1 = 2250rpm P = 71.2kW		

In load stage K9-1 a macro pit was formed in CC2-M which could be seen at the bear eye. This pit didn't grow in load stage K9-2 and the rest of the flanks appeared slightly better than before. This improvement in the flank shape also happened in CC2-E. As only one macro

pit was formed, it has little relevance. These results are, overall, very similar with the previous tests. Table E14 shows the microscope pictures of the teeth flanks above pitch circle.

Table E14 – Video-microscope pictures of the teeth flanks above pitch circle. Magnification 50x. (20MnCr5 case carburized lubricated with mineral and ester oils.)

	CC2-M	CC2-E
ORIGINAL		
K7 T1 = 183.4Nm n1 = 2250rpm P = 43.2kW		
K9-1 T1 = 302.0Nm n1 = 2250rpm P = 71.2kW		
K9-2 T1 = 302.0Nm n2 = 1500rpm P = 71.2kW	 (200x)	 (200x)

For higher contact pressures, the differences between the two gears were similar to the lower contact pressures.

Weight loss and ferrometry

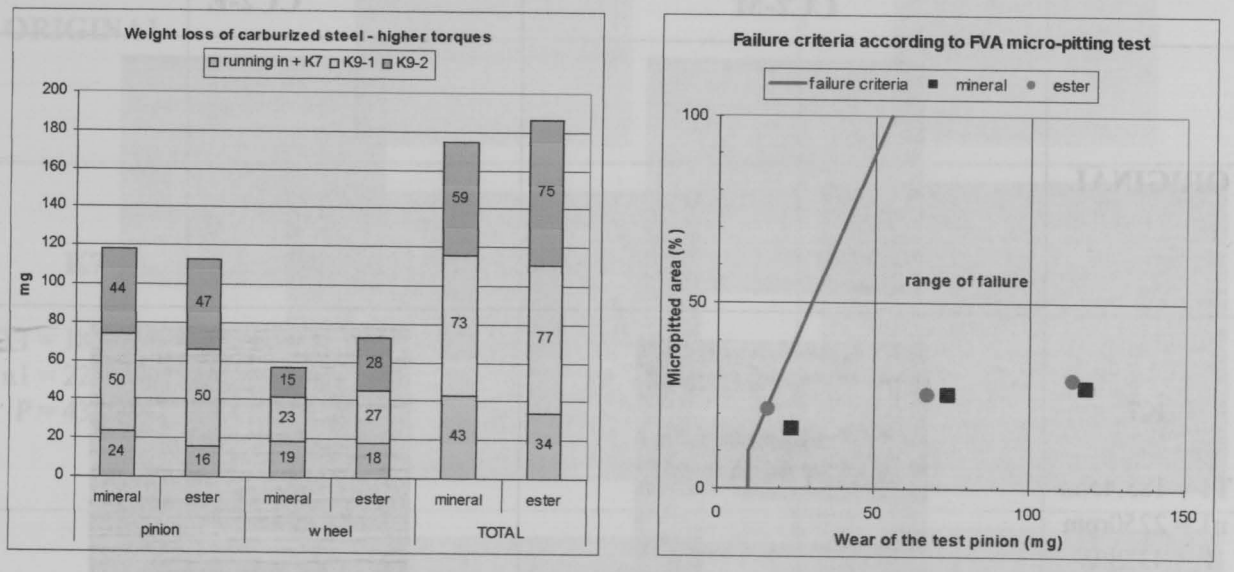


Figure E8 - Weight loss and example of failure criteria according to FVA.

Once again, very similar results are obtained by the two lubricants. Also, load stage K7 was very close to comply with FVA criterion, with ester having a result on the edge of the criterion. The weight loss shows an opposite result from the previous tests, with ester having now a slight advantage for the pinion, but in the total weight loss there was an inversion, and mineral oil had less weight loss in the end of the test. Wear above pitch circle may be the major responsible for these results.

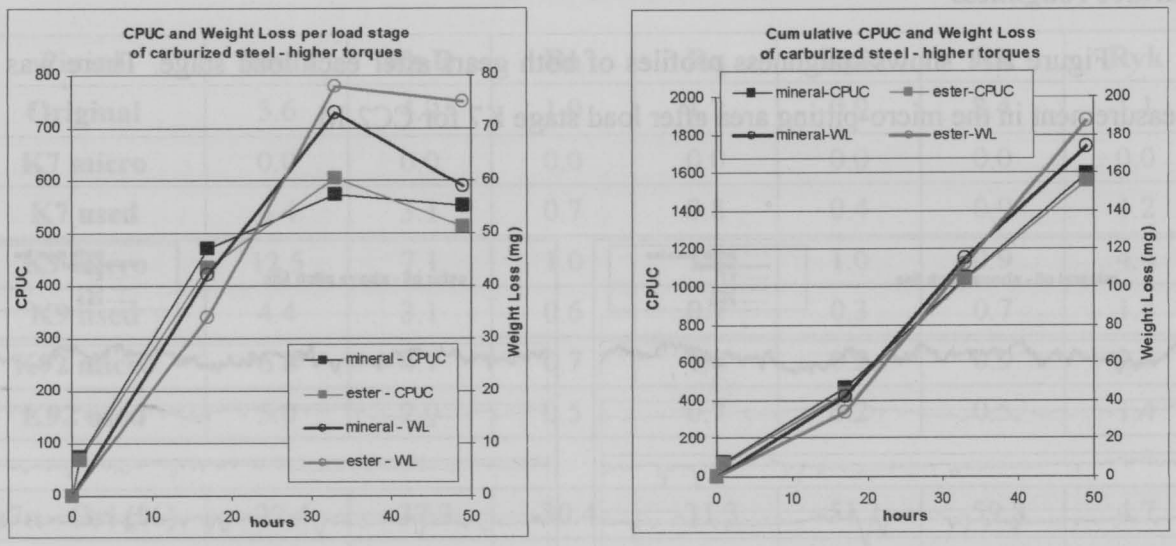


Figure E9 - Weight loss and ferrometry during the tests.

Results of weight loss and particle index are obviously higher than in the previous tests, but are still very similar for both lubricants.

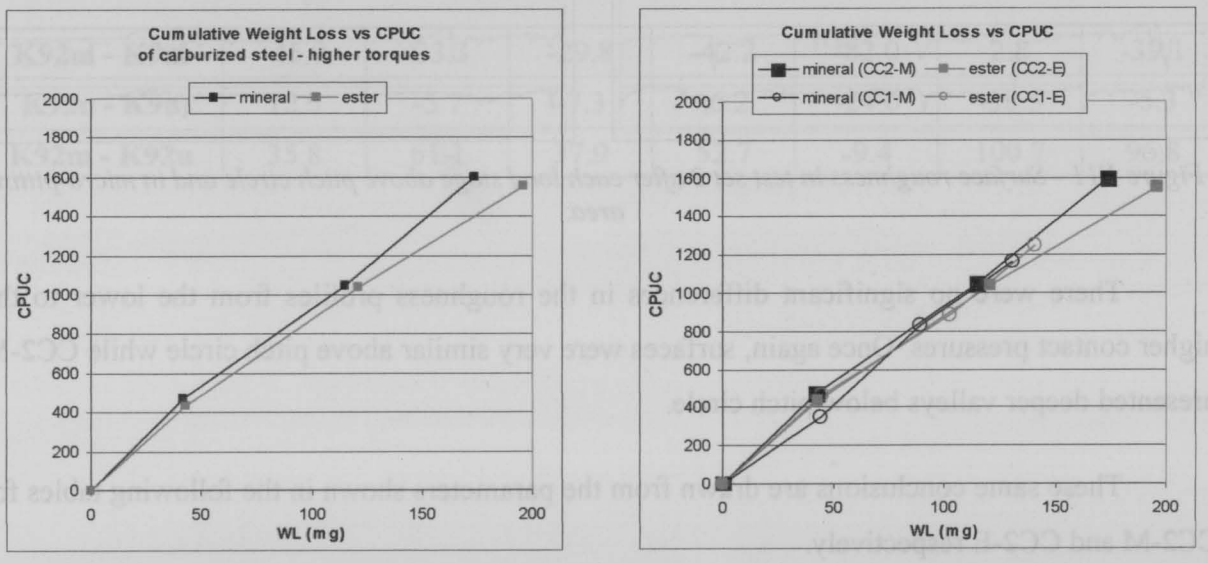


Figure E10 - Particle index vs weight loss for test set 2 and for all carburized steel tests.

Figure E10 shows that there is a very good correlation between concentration wear particle index and weight loss, as expected. This is a very good validation of the procedure adopted.

Surface roughness

Figure E11 shows roughness profiles of both gears after each load stage. There was no measurement in the micro-pitting area after load stage K7 for CC2-M.

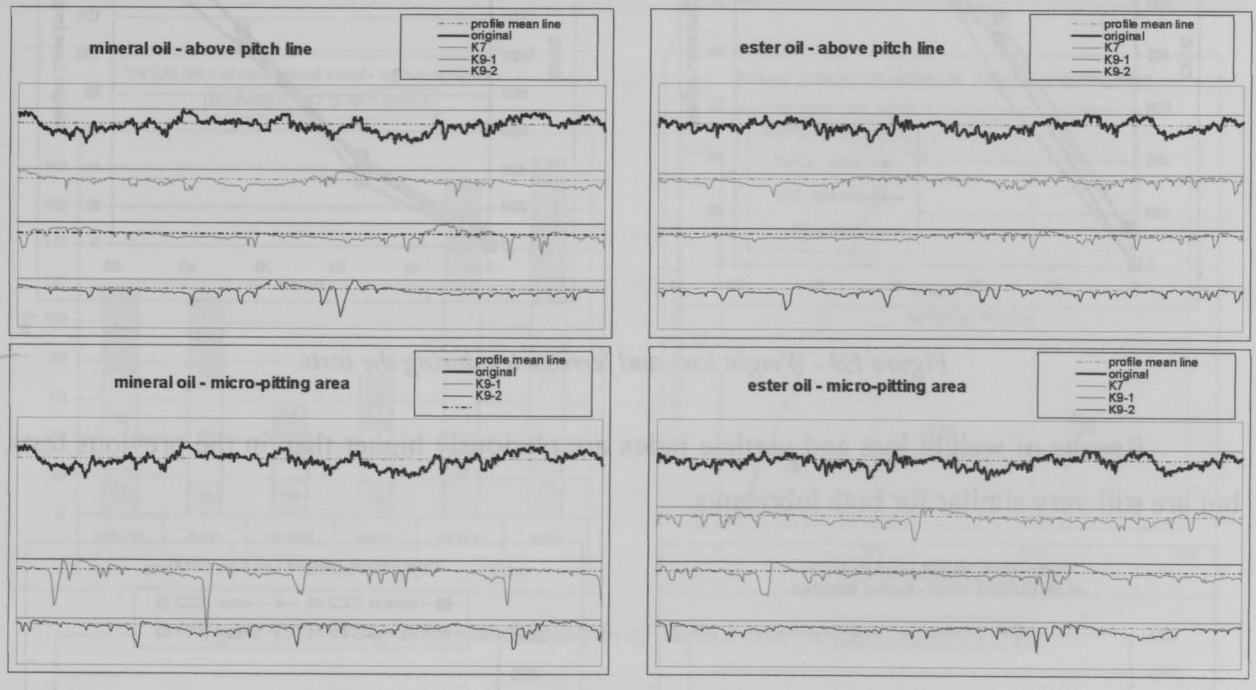


Figure E11 - Surface roughness in test set 2 after each load stage above pitch circle and in micro-pitting area.

There were no significant differences in the roughness profiles from the lower to the higher contact pressures. Once again, surfaces were very similar above pitch circle while CC2-M presented deeper valleys below pitch circle.

These same conclusions are drawn from the parameters shown in the following tables for CC2-M and CC2-E respectively.

Table E15 - Roughness parameters of CC2-M.

Pinion	Rmax	Rz-D	Ra	Rq	Rpk	Rk	Rvk
Original	5.6	4.9	1.0	1.2	0.9	2.4	1.1
K7 micro	0.0	0.0	0.0	0.0	0.0	0.0	0.0
K7 used	4.4	3.1	0.7	0.8	0.4	0.9	1.2
K9 micro	12.5	7.1	1.0	1.8	1.0	0.9	4.4
K9 used	4.4	3.1	0.6	0.7	0.3	0.7	1.4
K92 micro	6.8	4.7	0.7	1.0	0.2	0.9	2.7
K92 used	5.0	2.9	0.5	0.7	0.2	0.5	1.4
k7u - Ori (%)	-22.4	-37.3	-30.4	-31.3	-51.1	-59.8	4.7
k7m - k7u (%)	-100.0	-100.0	-100.0	-100.0	-100.0	-100.0	-100.0
k9u - K7u (%)	1.2	2.0	-16.7	-11.5	-42.5	-21.5	24.6
K9m - k9u (%)	182.2	127.7	82.2	147.7	283.4	20.0	205.9
K92m - K9m	-45.9	-33.3	-29.8	-42.2	-82.0	2.8	-39.1
K92u - K9u	12.5	-5.7	-7.3	-6.2	-24.0	-38.5	-5.3
K92m - K92u	35.8	61.1	37.9	52.7	-9.4	100.7	96.8

Table E16 - Roughness parameters of CC2-E.

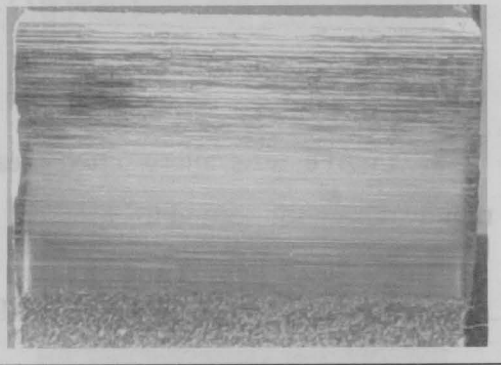
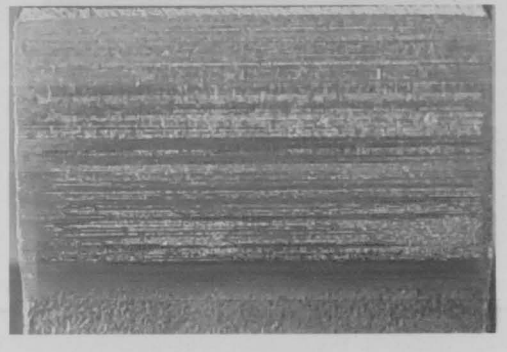
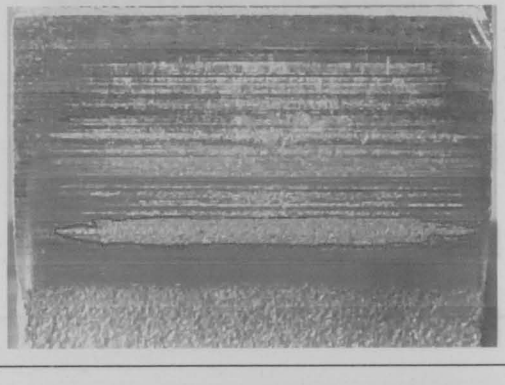
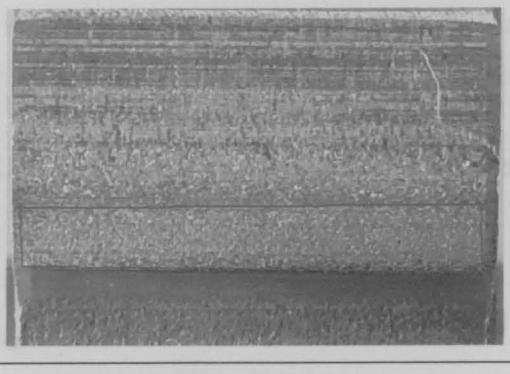
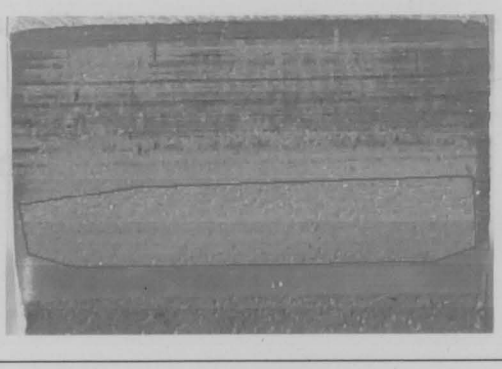
Pinion	Rmax	Rz-D	Ra	Rq	Rpk	Rk	Rvk
Original	4.6	4.0	0.8	1.0	0.8	2.1	0.9
K7 micro	6.4	4.1	0.7	1.0	0.3	0.9	1.9
K7 used	3.6	2.8	0.6	0.7	0.6	1.0	0.9
K9 micro	6.8	4.1	0.6	0.9	0.3	0.8	1.8
K9 used	4.3	2.9	0.5	0.7	0.2	0.8	1.1
K92 micro	4.9	3.2	0.5	0.7	0.1	0.8	1.5
K92 used	3.9	3.0	0.4	0.6	0.2	0.6	1.2
k7m - Ori (%)	40.1	4.7	-11.6	-1.7	-63.5	-54.4	106.0
k7u - Ori (%)	-21.8	-28.8	-30.9	-30.8	-28.2	-50.6	3.4
k7m - k7u (%)	79.1	47.2	28.1	42.1	-49.1	-7.6	99.2
k9m - K7m (%)	7.1	-0.6	-10.0	-4.4	12.5	-17.9	-2.7
k9u - K7u (%)	21.0	3.3	-8.0	-2.1	-62.1	-19.6	16.1
K9m - k9u (%)	58.5	41.5	25.2	38.7	50.9	-5.6	66.9
K92m - K9m	-27.7	-21.5	-23.0	-24.5	-56.2	8.0	-16.4
K92u - K9u	-9.6	1.6	-13.9	-9.7	3.0	-23.2	7.5
K92m - K92u	26.7	9.3	12.0	15.9	-35.7	32.7	29.9

E6 TEST RESULTS OF TEST SET 3

Visual aspect of teeth flanks

Table E17 shows visual aspect of teeth flanks after each load stage.

Table E17 - Pictures of pinion teeth before test and after each load stage. (ADI lubricated with mineral and ester oils.)

	ADI-M	ADI-E
ORIGINAL		
K5 T1 = 70Nm n1 = 2250rpm P = 16.5kW		
K7 T1 = 132.5Nm n1 = 2250rpm P = 31.2kW		

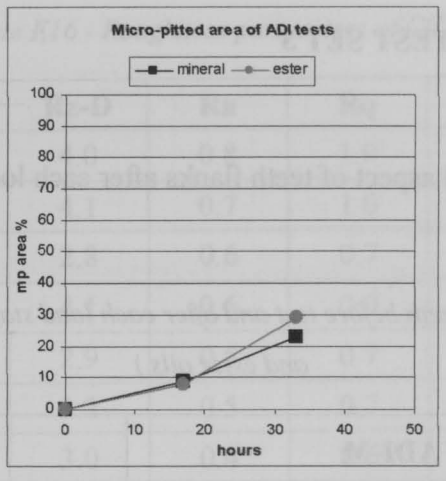
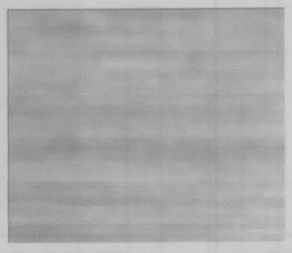
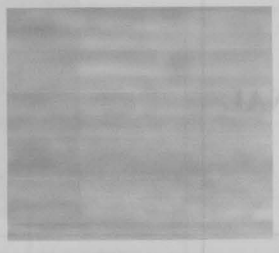
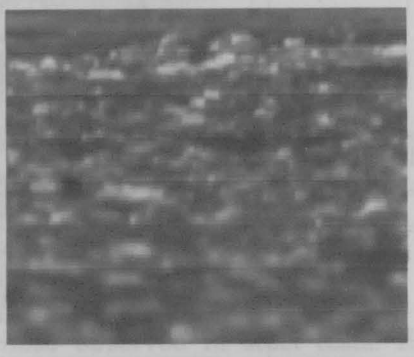
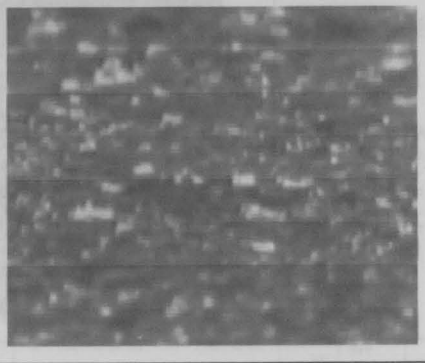


Figure E12 - Percentage of micro-pitted area.

In ADI tests micro-pitted area is very small after the first load stage but grows very rapidly during the second load stage.

Table E18 shows the microscope pictures of the teeth flanks below pitch circle.

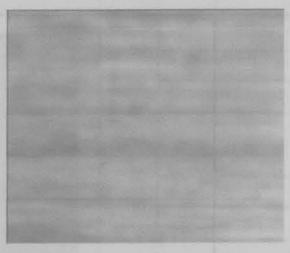
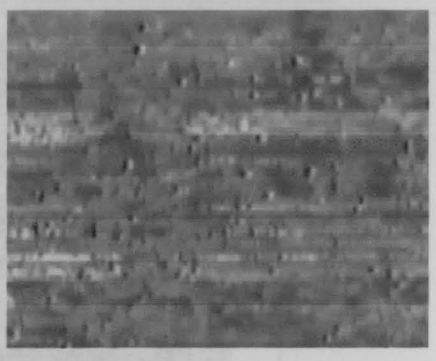
Table E18 – Video-microscope pictures of the teeth flanks below pitch circle. Magnification 100x. (ADI lubricated with mineral and ester oils.)

	ADI-M	ADI-E
ORIGINAL		
K5 T1 = 70Nm n1 = 2250rpm P = 16.5kW		
K7 T1 = 132.5Nm n1 = 2250rpm P = 31.2kW		

Aspect is not very different between the two oils, with pits evenly distributed over the whole surface.

Table E19 shows microscope pictures of the teeth flanks above pitch circle.

Table E19 – Video-microscope pictures of the teeth flanks above pitch circle. Magnification: 100x. (ADI lubricated with mineral and ester oils.)

	ADI-M	ADI-E
ORIGINAL		
K5 T1 = 70Nm n1 = 2250rpm P = 16.5kW		
K7 T1 = 132.5Nm n1 = 2250rpm P = 31.2kW		

The surfaces above pitch circle appear smooth with both lubricants, as expected for this material.

Weight loss and ferrometry

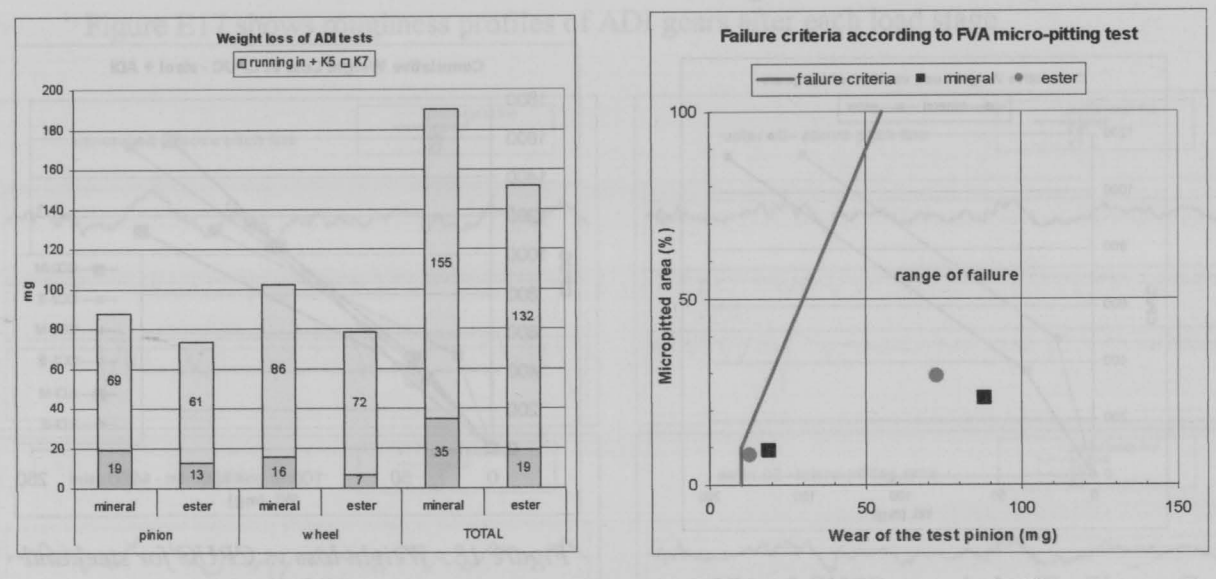


Figure E13 - Weight loss and example of failure criteria according to FVA.

The values of weight loss are much higher than those observed in steel tests. Ester biodegradable showed less weight loss both for pinion and wheel and for the two load stages. This is due to the lower friction coefficient and probably indicates a better combination of the pair oil/material. In fact, it is often reported that additivated mineral oils are not as effective with ADI as they are with steel.

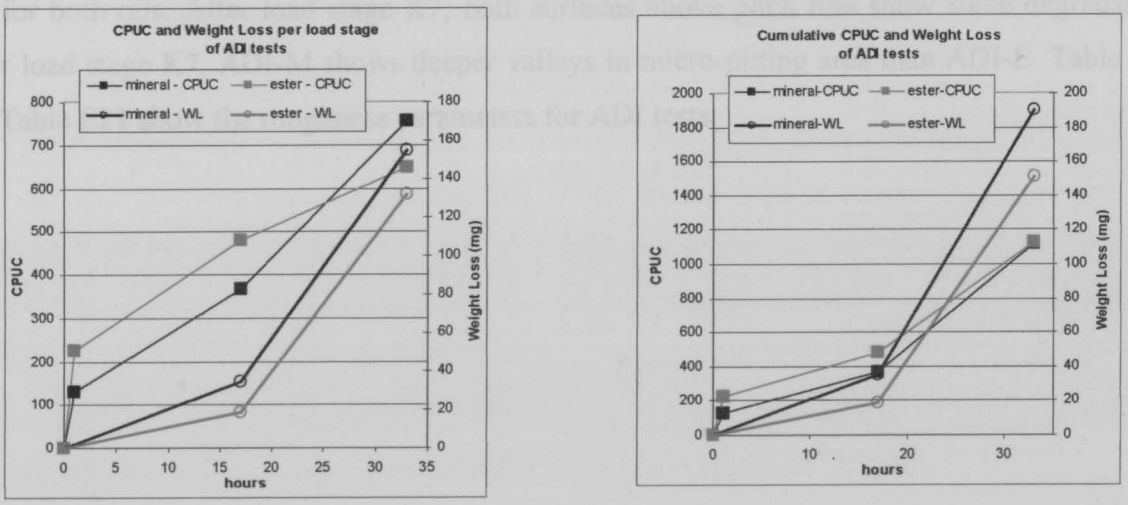


Figure E14 - Weight loss and ferrometry during the tests.

The cumulative CPUC and weight loss clearly shows the tendency of much higher weight loss in load stage K7 than in load stage K5.

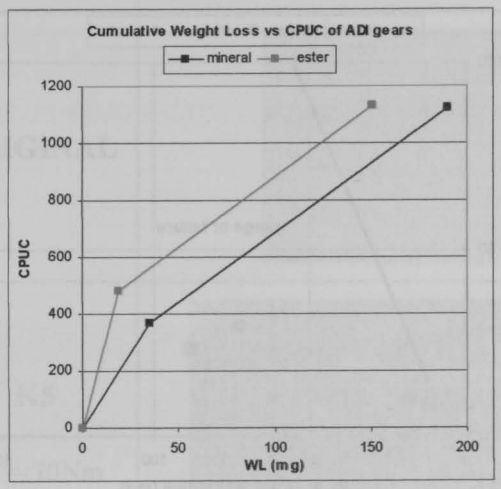


Figure 17 - Weight loss vs CPUC for ADI tests.

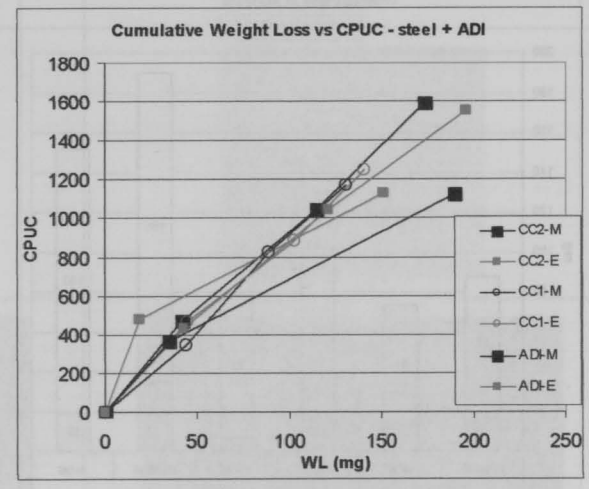


Figure 18 - Weight loss vs CPUC for steel and ADI tests.

ADI also shows a very good correlation between weight loss and CPUC, as Figure E19 shows. When plotting ADI and steel together, in Figure E20, it shows different slopes for the correlation of weight loss/CPUC which is due to the difference of specific weights of the two materials.

Surface roughness

Figure E17 shows roughness profiles of ADI gears after each load stage.

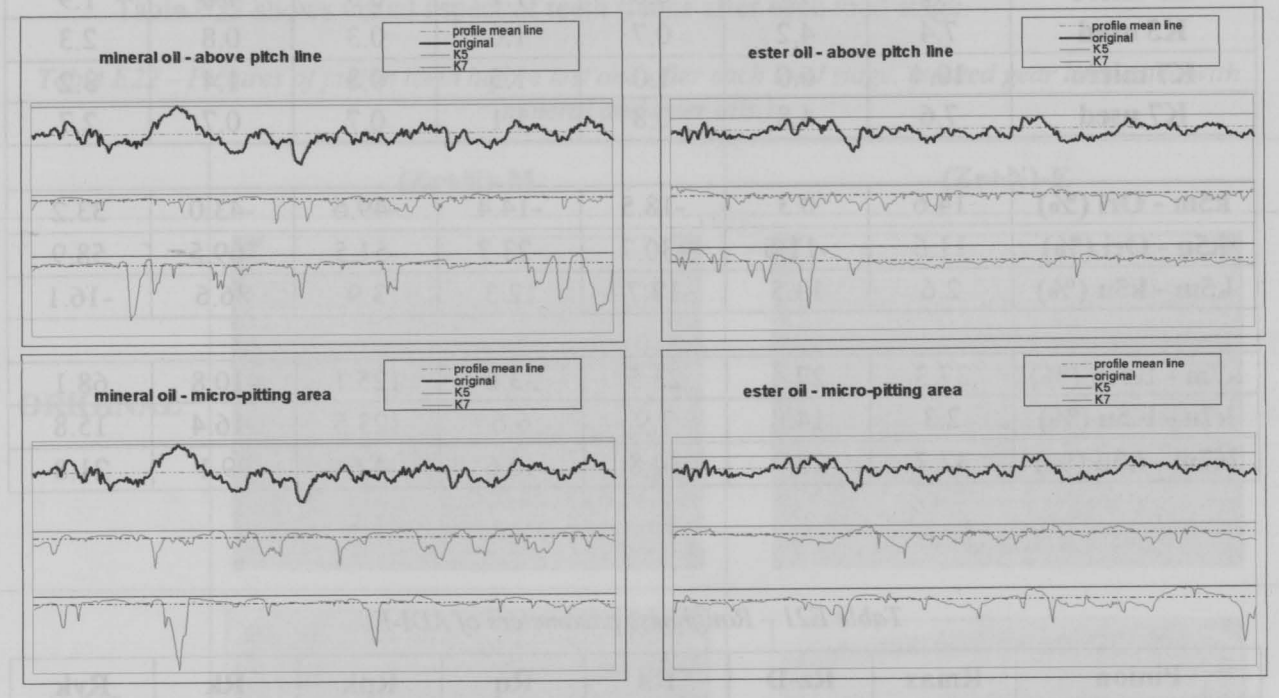


Figure E17 - Surface roughness of ADI gears after each load stage above pitch circle and in micro-pitting area.

There is a clear improvement in surface state after load stage K5 over the whole flank and for both oils. After load stage K7, both surfaces above pitch line show some degradation. After load stage K7, ADI-M shows deeper valleys in micro-pitting area than ADI-E. Table E20 and Table E21 show the roughness parameters for ADI tests.

Table E20 – Roughness parameters of ADI-M.

Pinion	Rmax	Rz-D	Ra	Rq	Rpk	Rk	Rvk
Original	6.6	4.7	1.0	1.3	0.7	2.8	1.5
K5 micro	7.6	4.7	0.8	1.1	0.3	1.6	1.9
K5 used	7.4	4.2	0.7	1.0	0.3	0.8	2.3
K7 micro	10.4	6.0	1.0	1.5	0.8	1.4	3.2
K7 used	7.6	4.8	0.8	1.1	0.7	0.7	2.7
k5m - Ori (%)	14.6	0.3	-18.5	-14.4	-49.6	-43.0	33.2
k5u - Ori (%)	11.6	-11.6	-30.7	-23.7	-51.5	-69.5	58.9
k5m - k5u (%)	2.6	13.5	17.7	12.3	3.9	86.6	-16.1
k7m - K5m (%)	37.3	27.4	23.5	33.6	125.1	-10.8	68.1
k7u - K5u (%)	2.3	14.8	7.9	6.6	123.5	-16.4	15.8
K7m - k5u (%)	37.7	25.9	34.8	40.6	4.6	99.1	21.8

Table E21 – Roughness parameters of ADI-E.

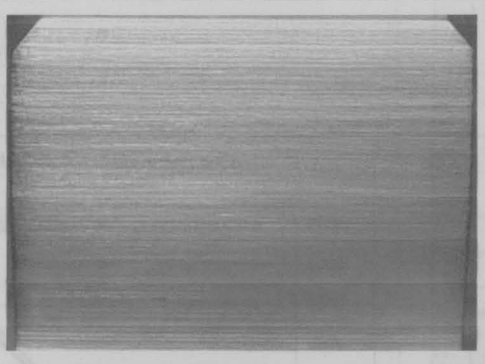
Pinion	Rmax	Rz-D	Ra	Rq	Rpk	Rk	Rvk
Original	7.5	5.7	1.2	1.5	1.3	3.3	1.1
K5 micro	4.7	3.4	0.6	0.8	0.5	1.5	1.3
K5 used	7.2	3.5	0.6	0.8	0.3	0.8	1.5
K7 micro	10.1	6.0	0.9	1.3	0.7	1.0	2.8
K7 used	7.2	4.5	0.6	0.9	0.2	0.6	2.5
K5m - Ori (%)	-37.4	-39.5	-46.4	-43.6	-63.5	-53.9	18.8
K5u - Ori (%)	-4.8	-39.2	-52.6	-45.1	-76.1	-75.4	38.4
K5m - K5u (%)	-34.3	-0.6	13.2	2.7	52.5	87.1	-14.2
K7m - K5m (%)	113.9	74.6	41.2	62.6	44.2	-36.6	124.4
K7u - K5u (%)	0.9	29.7	12.1	17.9	-23.5	-22.0	67.9
K7m - K7u (%)	39.3	33.8	42.5	41.6	187.4	52.2	14.7

E7 TEST RESULTS OF TEST SET 4

Visual aspect of teeth flanks

Table E22 shows visual aspect of teeth flanks after each load stage.

Table E22 – Pictures of pinion teeth before test and after each load stage. (coated gear lubricated with mineral and ester oils.)

	(Zr+N)-M	(Zr+N)-E
ORIGINAL		
K7 T1 = 132.5Nm n1 = 2250rpm P = 31.2kW		

No micro-pitting was found on coated gears either with mineral or ester oil. What was observed was the destruction of the coating, which started even at running in, as Figure E18 indicates. This happened with both oils. This means that this coating is not suitable for gear applications. Furthermore, severe pitting can be found at pitch circle, which means that the substrate – nitrided steel – is not hard enough to support the contact pressure of a gear teeth contact. We have, thus, an inefficient combination of base material and surface coating. The results from the test lose significance, once the coating did not prove suitable.

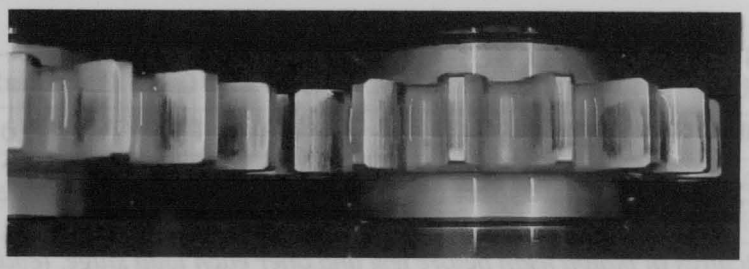


Figure E18 - Gear teeth after running-in.

Weight loss and ferrometry

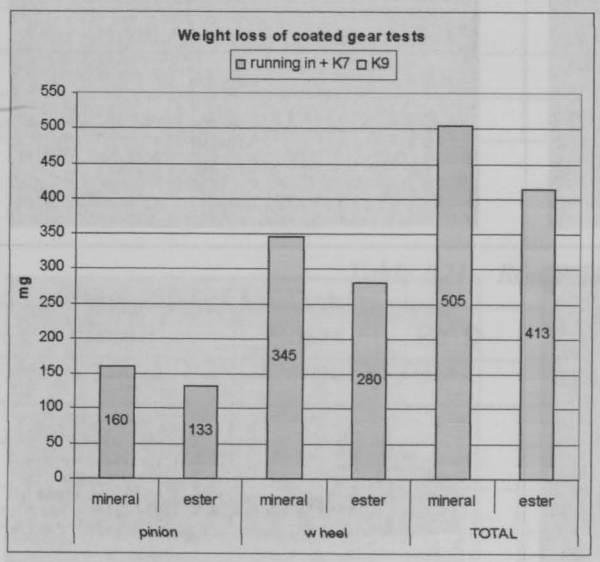


Figure E17 - Weight loss of test set 4.

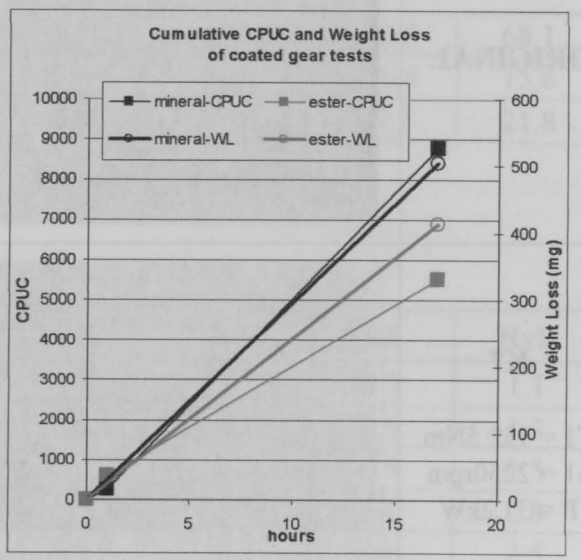


Figure E18 - Wear particle index of test.

The values of weight loss are about eight times higher than carburized steel for the same load stage. But in this case, the weight loss is due to the removal of coating and pitting. The specific weight of the coating is not known, but its destruction, verified in the oil sample is in accordance with these values. Nevertheless, it is interesting to notice the difference between both oils. Once again, candidate oil proved the advantage of the lower friction coefficient with less weight loss than mineral oil.

Surface roughness

Figure E19 shows roughness profiles of coated gear after each load stage.

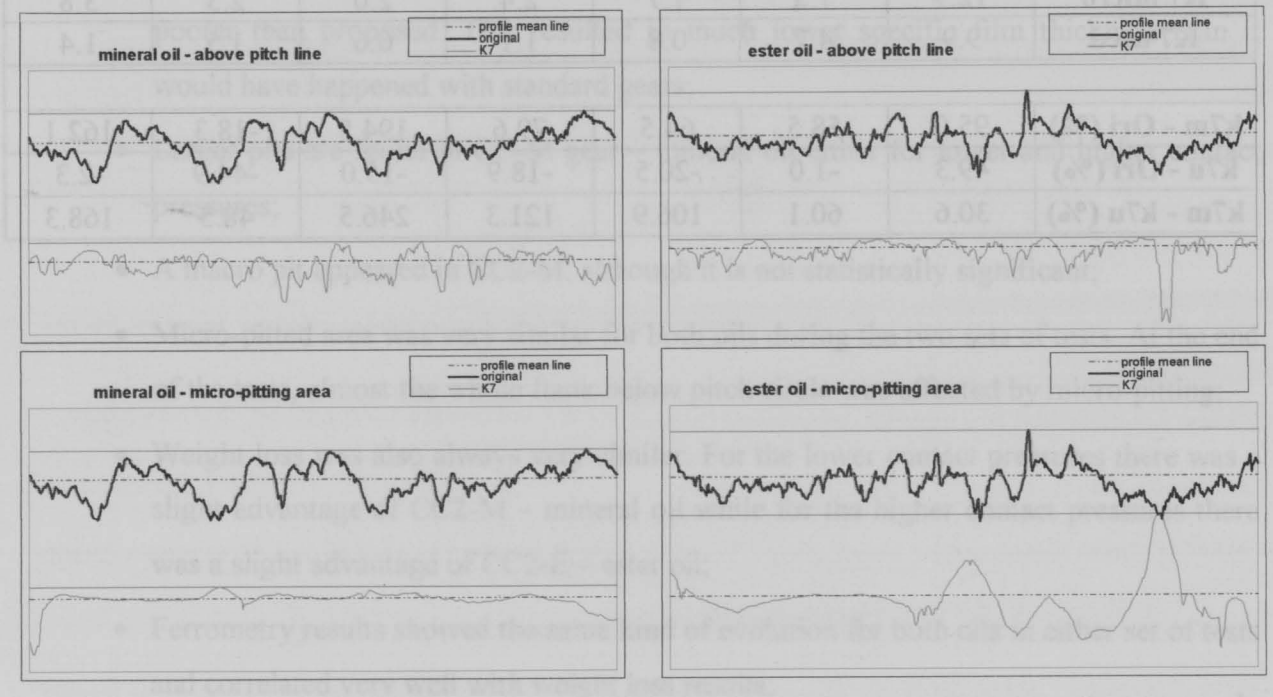


Figure E19 - Surface roughness of coated gear tests after each load stage above pitch circle and in micro-pitting area.

Roughness profiles confirm that there is no micro-pitting in either test. The destruction of the coating prevents taking definitive conclusions from these profiles. Table E23 and Table E24 show the roughness parameters of these tests.

Table E23 – Roughness parameters of (Zr+N)-M.

Pinion	Rmax	Rz-D	Ra	Rq	Rpk	Rk	Rvk
Original	6.0	4.9	1.1	1.4	0.8	2.6	2.1
K7 micro	16.5	8.4	1.8	2.7	1.5	1.6	5.6
K7 used	4.6	3.7	0.9	1.1	0.4	1.8	1.3
k7m - Ori (%)	174.2	71.8	59.7	94.2	76.6	-39.2	168.7
k7u - Ori (%)	-23.2	-25.2	-22.0	-22.5	-47.5	-29.8	-36.4
k7m - k7u (%)	257.0	129.6	104.6	150.5	236.2	-13.4	322.5

Table E24 – Roughness parameters of (Zr+N)-E.

Pinion	Rmax	Rz-D	Ra	Rq	Rpk	Rk	Rvk
Original	6.6	4.7	1.0	1.3	0.7	2.8	1.5
K7 micro	12.9	7.5	1.7	2.4	2.0	2.3	3.8
K7 used	9.9	4.7	0.8	1.1	0.6	1.5	1.4
k7m - Ori (%)	95.0	58.5	64.5	79.6	194.5	-18.3	162.1
k7u - Ori (%)	49.3	-1.0	-20.5	-18.9	-15.0	-44.9	-2.3
k7m - k7u (%)	30.6	60.1	106.9	121.3	246.5	48.5	168.3

CONCLUSIONS

- Results cannot be regarded as absolute because tested gears had an accuracy grade 9 while standard proposes accuracy grade 5. Surface roughness, namely Ra, was also poorer than proposed. This resulted in much lower specific film thickness than it would have happened with standard gears;
- Larger pits are found in CC-M gear – mineral oil either for lower and higher contact pressures;
- A macro pit appeared in CC2-M, although it is not statistically significant;
- Micro-pitted area was very similar for both oils during the two sets of tests. At the end of the tests, almost the whole flank below pitch circle was affected by micro-pitting;
- Weight loss was also always very similar. For the lower contact pressures there was a slight advantage of CC2-M – mineral oil while for the higher contact pressures there was a slight advantage of CC2-E – ester oil;
- Ferrometry results showed the same kind of evolution for both oils in either set of tests and correlated very well with weight loss results;
- Surface roughness improved above pitch circle for both gears in both tests;
- Below pitch circle, surface roughness was always better for CC-E – ester oil;
- The very low friction coefficient of ester oil may be one the reasons for its good performance in micro-pitting protection;
- Mineral oil appeared to offer better protection of the surfaces against wear and adhesion, but that was not within the scope of these tests;

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