Mild wear in lubricated gear transmissions

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Resumo

O presente trabalho destina-se ao estudo do dano superficial dos flancos dos dentes de engrenagem, em particular, dos danos de desgaste. Em relação a esta temática, várias leis têm sido desenvolvidas, mas o presente trabalho cingir-se-á à lei de Archard, com o objetivo de se perceber como o desgaste (coeficiente de desgaste) varia de acordo com diferentes condições de funcionamento.

Para se efetuar a simulação de desgaste, é necessário que as tensões de contacto que se desenvolvem entre os dentes durante o engrenamento sejam conhecidas com alguma precisão, incluindo os efeitos que a rugosidade das superfícies exerce na lubrificação e nas pressões de contacto, que a literatura considera determinantes para o desgaste.

O presente trabalho constrói-se à volta dos seguintes eixos principais:

- utilizar uma máquina de ensaio de engrenagens para testar engrenagens com diferentes condições de carregamento, espessura de filme e oleo utilizado;
- analisar os teste realizados através dos resultados de análise de perda de massa, análise de oleo, topografias e perda de potência;
- através do modelo desenvolvido por Brandão identificar os diferentes valores de coeficiente de desgaste da lei de Archard para os diferentes ensaios realizados;
- através do modelo desenvolvido por Fernandes quantificar a perda de potência nos diferentes teste realizados.

A análise dos resultados permite obter uma imagem clara da influência dos parâmetros citados acima no processo de degaste normal.
Abstract

The present work is intended to study surface damage on gear tooth flanks, in particular wear damage. To study wear several laws have been developed, but in the present work Archard’s law is the one used to understand how the wear (wear coefficient) varies with the different operating conditions.

The simulation of wear demands that the contact stresses developed between teeth during gear meshing shall be known with some accuracy, including the effects of surface roughness in lubrication and contact pressure, which are known to be extremely important for surface wear.

These are then the principal guidelines around which the present work is constructed:

- use of a gear test rig to test spur gears at different conditions of load, film thickness and lubricant;
- analyse the tests performed in terms of results of the mass loss, oil analysis, roughness measurement and power loss;
- through the model developed by Brandão to identify the different values of coefficient of wear in Archard’s law for the different tests performed;
- through the model developed by Fernandes quantify the power loss values for the different tests performed.

Analysis of these results brings a clear picture of the influence of these operating conditions on the process of mild wear.
Keywords
wear
Spur gears
FZG
Archard’s law

Palavras Chave
Desgaste em engrenagens
Engrenagens cilíndricas
FZG
Lei de Archard
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## Symbols and Abbreviations

**Abbrev.** | **Description** |
---|---|
apc | Above pitch circle |
BAC | Bearing area curve |
bpc | Below pitch circle |
EHD | Elastohydrodynamic lubrication |

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
</table>
a | Hertzian contact half-width | [m] |
b | gear tooth width | [m] |
d | dilution | |
E | Young’s modulus | [Pa] |
$E^*$ | effective Young’s modulus | [Pa] |
f$_A$ | load sharing function in mixed lubrication | |
$F_N$ | normal contact force | [N] |
$F_{N}^{BDR}$ | portion of the contact load borne by direct surface contact | [N] |
$F_{N}^{EHD}$ | portion of the contact load borne by lubricant film | [N] |
G | elastic shear modulus | [Pa] |
h | wear depth | [m] |
h | film thickness | [m] |
H | hardness | [Pa] |
h$_o$ | central film thickness | [m] |
h$_{oc}$ | central film thickness with inlet shear heating corrections | [m] |
h$_c$ | central film thickness | [m] |
$H_V$ | gear loss factor | |
$\kappa$ | Von Mises yield stress in shear | [Pa] |
K | dimensionless wear coefficient | |
$M_l$ | mass loss | [kg] |
n$_1$ | driving gear angular velocity | [rpm] |
n$_2$ | driven gear angular velocity | [rpm] |
p | pressure | [Pa] |
$p_0$ | maximum Hertzian pressure | [Pa] |
$p_{BDR.T}^{BDR}$ | normal contact pressure when supposing that the surfaces are not lubricated and the entirety of the load is borne by direct surface contact | [Pa] |
$p_{BDR.T}^{EHD}$ | normal contact pressure when supposing that the surfaces are ideally smooth and the lubrication is in the full film EHD regime | [Pa] |
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>R</td>
<td>radius</td>
<td>[m]</td>
</tr>
<tr>
<td>$P_{IN}$</td>
<td>gearbox input power</td>
<td>[W]</td>
</tr>
<tr>
<td>$P_{VD}$</td>
<td>shaft seals loss</td>
<td>[W]</td>
</tr>
<tr>
<td>$P_{VL}$</td>
<td>rolling bearings power loss</td>
<td>[W]</td>
</tr>
<tr>
<td>$P_{VZ0}$</td>
<td>load independent meshing gear loss</td>
<td>[W]</td>
</tr>
<tr>
<td>$P_{VZP}$</td>
<td>load dependent meshing gears power loss</td>
<td>[W]</td>
</tr>
<tr>
<td>$P_V$</td>
<td>gearbox total power loss</td>
<td>[W]</td>
</tr>
<tr>
<td>$P_{Vexp}$</td>
<td>gearbox total power loss measured</td>
<td>[W]</td>
</tr>
<tr>
<td>$R^*$</td>
<td>effective radius of curvature</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_a$</td>
<td>radius of the addendum circle of a gear</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_b$</td>
<td>radius of the base circle of a gear</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_p$</td>
<td>radius of the pitch circle of a gear</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_a$</td>
<td>arithmetic average roughness</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_Z$</td>
<td>average roughness height</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_p$</td>
<td>maximum height of peaks</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>reduced peak height</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_v$</td>
<td>maximum depth of valleys of peaks</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_{vk}$</td>
<td>reduced valley depth</td>
<td>[m]</td>
</tr>
<tr>
<td>$R_q$</td>
<td>root mean square deviation (RMS) profile height;</td>
<td>[m]</td>
</tr>
<tr>
<td>$S$</td>
<td>sliding distance</td>
<td>[m]</td>
</tr>
<tr>
<td>SRR</td>
<td>slide-to-roll ratio</td>
<td></td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
<td>[K]</td>
</tr>
<tr>
<td>$T_f^{avg}$</td>
<td>average lubricant temperature within the contact</td>
<td>[k]</td>
</tr>
<tr>
<td>$U_1$</td>
<td>tangential velocity of pinion tooth surface</td>
<td>[m/s]</td>
</tr>
<tr>
<td>$U_2$</td>
<td>tangential velocity of wheel tooth surface</td>
<td>[m/s]</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>pressure-viscosity coefficient</td>
<td>[Pa$^{-1}$]</td>
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<tr>
<td>$\beta$</td>
<td>temperature-viscosity coefficient</td>
<td>[k$^{-1}$]</td>
</tr>
<tr>
<td>$\Delta V$</td>
<td>volume lost by wear</td>
<td></td>
</tr>
<tr>
<td>$\dot{\gamma}$</td>
<td>shear strain rate</td>
<td>[m$^3$/s]</td>
</tr>
<tr>
<td>$\phi_T$</td>
<td>inlet shear heating correction coefficient</td>
<td></td>
</tr>
<tr>
<td>$\phi_R$</td>
<td>inlet roughness correction coefficient</td>
<td></td>
</tr>
<tr>
<td>$\theta_A$</td>
<td>influence of power supply conditions in the converging</td>
<td></td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>contact ratio of spur gears</td>
<td></td>
</tr>
<tr>
<td>$\eta$</td>
<td>dynamic low's hear viscosity</td>
<td>[Pa.s]</td>
</tr>
<tr>
<td>$\eta_0$</td>
<td>dynamic viscosity at reference temperature $T_0$ and atmospheric pressure</td>
<td>[Pa.s]</td>
</tr>
<tr>
<td>$\eta_{100}$</td>
<td>dynamic viscosity at 100°C and atmospheric pressure</td>
<td>[Pa.s]</td>
</tr>
<tr>
<td>$\eta_{40}$</td>
<td>dynamic viscosity at 40°C and atmospheric pressure</td>
<td>[Pa.s]</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>wear coefficient</td>
<td>[Pa$^{-1}$]</td>
</tr>
<tr>
<td>$\Lambda$</td>
<td>specific film thickness</td>
<td></td>
</tr>
<tr>
<td>$\lambda$</td>
<td>thermal conductivity</td>
<td>[W.m$^{-1}.K^{-1}$]</td>
</tr>
<tr>
<td>$\mu$</td>
<td>coefficient of friction</td>
<td>[W]</td>
</tr>
<tr>
<td>$\mu_{BDR}$</td>
<td>coefficient of friction in boundary film lubrication</td>
<td></td>
</tr>
<tr>
<td>$\mu_{EHD}$</td>
<td>coefficient of friction in EHL</td>
<td></td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity</td>
<td>[m$^2$.s$^{-1}$]</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>--------------------------------------------------</td>
<td>--------------</td>
</tr>
<tr>
<td>$\nu_{100}$</td>
<td>kinematic viscosity at 100(^\circ)C</td>
<td>[m(^2)s(^{-1})]</td>
</tr>
<tr>
<td>$\nu_{40}$</td>
<td>kinematic viscosity at 40(^\circ)C</td>
<td>[m(^2)s(^{-1})]</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density</td>
<td>[kg.m(^{-3})]</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>combined root mean square roughness</td>
<td>[m]</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>normal stress</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$\tau$</td>
<td>shear stress</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$\tau_{BDR}$</td>
<td>portion of the tangential contact stress borne by direct surface contact</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$\tau_{EHD}$</td>
<td>portion of the tangential contact stress borne by lubricant film</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$\tau_{MIX}$</td>
<td>tangential contact stress in mixed film lubrication</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$\tau_{\text{max}}$</td>
<td>maximum shear stress of the mesoscopic stresses</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$\tau_{\text{oct}}$</td>
<td>octahedral shear stress</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$\omega$</td>
<td>angular velocity</td>
<td>[rad.s(^{-1})]</td>
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</table>
1. Introduction

A simple definition of wear would be the progressive removal of material from a surface caused by contact. Rolling contact fatigue and scuffing for example are excluded from this types of surface damage. Wear may be classified as sever or mild, depending on its severity. In spur gears severe wear is associated with poor surface treatment or inadequate lubrication. Inadequate lubrication may happen because low gear speed cannot provide enough lubrication film to avoid contact between surfaces. Mild wear is inevitable, even with a perfect lubricating film.

Due to the damage caused by severe wear, many studies have been carried out as severe wear has a catastrophic effect on the gearing quality, causing loss of form and increased vibration. On the other hand, mild wear has not received much attention as it is considered the initial phase of the removal of material from the solid surface. Nonetheless, recent studies have shown a considerable correlation between mild damage and the manner in which surface damage has severe consequences on gear transmissions. In particular, it has been shown that there is an interplay between mild wear and surface contact fatigue, because these two forms of surface damage act in competition with one another [1]. Consequently mild wear has gained importance as it is related to surface damage.

Mild wear has generally been modelled using Archard’s law and its most important parameter is the wear coefficient. Existing studies provide different values for wear coefficient which strongly influence both operating and lubricant conditions.

In gear applications where precipitous tooth failure mode such as scoring or scuffing has been avoided, ‘normal’ wear becomes a life-determining factor. Studies show that most materials are removed from both the addendum and dedendum tooth surfaces, and that the highest wear occurs at the beginning of an engagement. This high wear region corresponds to the root of the driving (pinion) teeth and the tip of the driven gear teeth.
2. Literature Review

Gears have existed and were common since ancient times, used in grain mills, water pumping stations and windmills.

Since the first gears were made no other mechanism has been able to transmit power, provide torque conversion and speed variation as efficiently as the gear, all in a compact format. Gears may be found in all areas, from power generators (1MW) to desktops (1 mW) and cars (1kW).

In the last decades the focus in the development of spurs gears has been to increase the power and the torque transmitted using smaller gears. These improvements have occurred due to improvements in the following areas:

- manufacturing - tooth geometry, improved flanks;
- thermal treatments and materials - use of highly alloyed steels, surface hardening treatments, new materials;
- lubricants - improved anti-wear properties, greater resistance to high temperatures, reducing friction [2].

2.1. Gear material

Material proprieties are very important for surfaces health and longevity. It is possible to obtain identical behaviour even when different materials are used. Combination of the proprieties is more important than the actual proprieties of each one of the materials.

A material with a high modulus of elasticity suffers minimal surface deformations which can causes high stress concentration when in contact. On the other hand materials with low modulus of elasticity are easier to deform and the contact loads are better distributed. Because they are easier to deform, they can not support high loads, due to their tendency to deform plastically.

In dry contacts the coefficient of friction is crucial.

When high loads are required along with high speeds, most contacting parts are made of metal alloys. An alternative to this are polymers which may be used for low loads as they work with a low coefficient of friction. One of their advantages is that there is no need for lubrication, and this is why it is used in domestic appliances. Metal has high stiffness, hardness and greater thermal conductivity. When a metal is required to endure high surface stress, steel and cast iron are the best choices. When a smooth metal is required due to frequent contact, as they form good contact pairs with other metals, then cast iron, steel, bronze is recommended.

An aspect that may not be overlooked is the constitution of the metal. When in contact, if both metals are very similar, they can weld very easily. On the other hand,
2. Literature Review

if the metals are very different then galvanic corrosion may occur. There is also a need to pay attention to metals pairs which are chemically soluble in order to avoid adhesion and seizure.

In many cases surfaces are coated to improve performance when in contact. Coatings may reduce fatigue and lower friction. For example titanium and tungsten coatings are extremely resistant to wear. The critical point of the coating is that it needs to remain firmly attached to the surface in order to provide good performance.

Indeed surface treatments are made to improve surface performance. The manufacturing process and the metal constitution may be optimized intentionally to provide more resistant surfaces with better tribological behavior. Some treatments are made only to the active surface, usually to harden or make it stronger. Carbonizing and nitriding are also hardening techniques, as there is an increased use in the amount of carbon and nitrogen respectively. Rolling and blasting are mechanical treatments which introduce compressive stress which helps to hinder the opening of cracks and fatigue wear [3].

2.2. Gear manufacturing

The orientation of the grooves left by cutting tools is very important, as these can or cannot overlap the surface movement direction changing the friction. Thus, friction depends on relative motion but also on contact surface roughness. Generally, high surface roughness causes high wear, due to high stress and friction coefficient. High roughness can be helpful if the grooves push the lubricant inside the contact. It also can improve the meshing between the surface and the hardened surface layers.

The quality of the cutting tools and the machines are determinant for roughness. Manufacturing processes always leave typical marks on surfaces. Average roughness is between 2 \( \mu \) m to 0.05 \( \mu \) (polished) [3].

2.3. Gear roughness

There are no perfect finished surfaces. The idea of a perfect plane and smooth surfaces is nothing more than a mathematical idealization. Mechanical component surfaces may be polished, but always show surface roughness in the form of small scratches or indentations which are made during the manufacturing process.

Ductile materials are better able to withstand strain efforts, while harder materials can endure higher loads but tend to crack more easily. This is why there must be a balance between stiffness and ductility.

Surfaces change with use, suffering several forms of wear, changing in texture as roughness increases or decreases as time elapses. A profilometer, which is able to obtain a surface image and characterize the level of roughness, is used to measure surface roughness.

Roughness analysis is very important in quality control to prevent damages as waviness and other defects in the mechanical components may be detected [3]. The surfaces found in machines-elements are not perfectly smooth, they have roughness and waviness which is possible to measure.
Roughness parameters can be calculated either two-dimensionally (2D) or three-dimensionally (3D). 3D roughness parameters are calculated for an area of the surface instead of a single line [4].

The main parameters are:

- \( R_a \) - arithmetic average roughness;
- \( R_z \) - average roughness height;
- \( R_p \) - maximum height of peaks;
- \( R_v \) - maximum depth of valleys of peaks;
- \( R_q \) - root mean square deviation (RMS) profile height;

### 2.3.1. Roughness parameters

\( R_a \) is the grandfather of all roughness parameters. It is the average deviation of the profile from the mean line. It is determined by the following equation:

\[
R_a = \frac{1}{l_m} \int_0^{l_m} |y(x)|\,dx
\]  

Where:

- \( l_m \) - length of the profile;
- \( y(x) \) - height variations.

The average height parameter is easy to define, easy to measure and gives a good height variation description. Because the roughness average is measured, defects in the surface do not influence the measured results greatly. Parameter \( R_a \) does not differentiate between peaks and valleys. \( R_a \) is expressed in micrometers (\( \mu m \)).

To determine the parameter, \( R_Z \), average roughness height, the filtered roughness profile is divided into five equal lengths. This parameter can be defined in two ways, according to DIN or ISO international standards. The difference between them is, the German DIN defines \( R_Z \) as average of the summation of the five highest peaks and the five deepest valleys along the assessment length of the profile.

Because \( R_Z \) examines the heights instead of the average as the \( R_a \), \( R_Z \) is more sensitive to changes in surface finish than \( R_a \). The following equations demonstrate the difference between the standard mentioned:

\[
R_{Z,(ISO)} = \frac{1}{n} \left( \sum_{i=1}^{n} p_i - \sum_{i=1}^{n} v_i \right)
\]  

\[
R_{Z,(DIN)} = \frac{1}{2n} \left( \sum_{i=1}^{n} p_i + \sum_{i=1}^{n} v_i \right)
\]

\( R_p \) is defined as the maximum height of the profile above the mean line within the assessment length. \( R_v \) is defined as the maximum depth of the profile below the mean line within the assessment length. In the Figure 2.1 \( R_{p3} \) represents the \( R_p \) parameter as well as \( R_{v4} \) represents the \( R_v \) parameter.
2. Literature Review

\[ R_q = \sqrt{\frac{1}{l_m} \int_0^{l_m} y^2(x) dx} \]  

(2.4)

\( R_q \) is the root mean square of the deviation of the profile from the mean line. If is determined by the following equation:

\( R_q \) is relevant when looking at the surface profile as a statistical function. This parameter is more sensitive than \( R_a \) to large deviation, from the mean line.

Roughness can be classified according to its orientation. Figure 2.2 demonstrates the different types of roughness orientation. This may be longitudinal (a), isotropic(b) or transverse(c). The roughness in gears usually has a transverse orientation.

\[ \sigma = \sqrt{(R_{q1})^2 + (R_{q2})^2} \]  

(2.5)

Where:

- \( R_{q1} \) - root mean square of body number one;
- \( R_{q2} \) - root mean square of body number two.

Table 2.9 displays typical roughness composed values in gear teeth obtained with different types of finishing.
2.3. Gear roughness

Table 2.1.: Typical roughness composed values in different types of finishing gear teeth [5].

<table>
<thead>
<tr>
<th>Types of finish</th>
<th>Roughness composed</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Initial value</td>
</tr>
<tr>
<td>Milling</td>
<td>2.3-4.6</td>
</tr>
<tr>
<td>Milling fine</td>
<td>1.2-2.3</td>
</tr>
<tr>
<td>Schaving</td>
<td>0.7-1.4</td>
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<tr>
<td>Lapping</td>
<td>0.6-1.1</td>
</tr>
<tr>
<td>Polish</td>
<td>0.3-0.6</td>
</tr>
</tbody>
</table>

2.3.3. Abbott-Firestone curve

The Abbott-Firestone curve or bearing area curve (BAC) is defined as the percentage of solid material of the profile lying at a certain heigth. This parameter is a useful indicator of the effective contact area as the surface wear. Figure 2.3 displays a typical Abbott-Firestone curve.

The horizontal axis represents the bearing area lengths as a percent the total assessment length of the profile and the vertical axis represents the heights of the profile.

The interpretation of the curve is that if the surface worn down to a certain height the appropriate figure would represent the fraction of a solid contact at that height. It is commonly used to compare wear, since after running-in a surface shows a marked change in the curve distribution [4].

Figure 2.3.: The Abbott-Firestone curve of a profile. [4].

The $R_k$ parameter has enormous practical significance. It is called the core roughness, because this part of the profile has the greatest increase in the material ratio over a given profile height difference and forms a dense bearing core.
The calculation of the parameter $R_k$ is displayed in Figure 2.4. The first step in the calculation is to find the point of minimum slope (the turning point) on the 'S' shaped material ratio curve. Two points, labelled A and B display in Figure 2.4, are taken on the material ratio curve. A and B are arbitrarily separated by 40% on the horizontal axis. The turning point is located by shifting points A and B along the curve until the vertical distance between them is at a minimum. Vertical distance between points C and D is the core roughness depth.

Parameter $R_{pk}$ is the reduced peak height display in Figure 2.5. The area under the material ratio curve above the roughness core, designated $A_2$, is determined. $R_{pk}$ is the height of a triangle. $R_{vk}$ is the reduced valley depth, is founded a similar way at the lower and oh the material ratio curve.

The parameters $R_k$, $R_{pk}$ and $R_{vk}$ are determined from a material ratio curve derived from a roughness profile filtered with a valley suppression filter according to DIN 4776 [5].

Tab.2.2 show the selection of roughness values according to function [5].
2.3. Gear roughness

Table 2.2.: Selection of roughness values according to function [5].

<table>
<thead>
<tr>
<th>$R_a(\mu m)$</th>
<th>$R_z(DIN)(\mu m)$</th>
<th>Production method</th>
<th>Function</th>
<th>Example</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>160</td>
<td>Sand casting</td>
<td>Non-functional</td>
<td>Engine block</td>
</tr>
<tr>
<td>25</td>
<td>100</td>
<td>Sawing hot rolling</td>
<td>Fluid friction</td>
<td>Ship hull plate</td>
</tr>
<tr>
<td>12,5</td>
<td>63</td>
<td>Forging shaping</td>
<td>Joint face</td>
<td>Pipe flange</td>
</tr>
<tr>
<td>6,3</td>
<td>40</td>
<td>Turning milling</td>
<td>Assembly face</td>
<td>Gearbox flange</td>
</tr>
<tr>
<td>3,2</td>
<td>25</td>
<td>Extruding EDM</td>
<td>Bearing</td>
<td>Hand crank</td>
</tr>
<tr>
<td>1,6</td>
<td>10</td>
<td>Investment casting</td>
<td>Clearance fit</td>
<td>Bolt shank</td>
</tr>
<tr>
<td>0,8</td>
<td>6,3</td>
<td>Grinding</td>
<td>Lubricated bearing</td>
<td>Gear teeth</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td>Cold rolling</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0,4</td>
<td>2,5</td>
<td>Barrel finishing</td>
<td>Bearing</td>
<td>Engine cylinder</td>
</tr>
<tr>
<td>0,2</td>
<td>1,6</td>
<td>Roller burnishing</td>
<td>Rolling bearing</td>
<td>Valve follower</td>
</tr>
<tr>
<td>0,1</td>
<td>1</td>
<td>Electro-polishing</td>
<td>Measuring reference</td>
<td>Gauge</td>
</tr>
<tr>
<td>0,05</td>
<td>0,63</td>
<td>Polishing</td>
<td>Performance seal</td>
<td>Fuel injector</td>
</tr>
<tr>
<td>0,4</td>
<td></td>
<td>Lapping</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0,025</td>
<td>0,25</td>
<td>Lapping</td>
<td>Wrink surface</td>
<td>Gauge block</td>
</tr>
<tr>
<td>0,16</td>
<td></td>
<td>Spur finishing</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

2.3.4. The surface as a composite

Surface profile characteristics can be broken into six categories. Form, waviness and roughness are the main concerns in manufacturing. The point at which roughness become waviness (the cut-off point) is arbitrary, and related to the manufacturing process and the function of the surface. No absolute definition of what constitutes roughness and when roughness becomes waviness exists. The cut-off between wavelength and roughness is expressed as a length. An excessive cut-off will include waviness in the results, and an insufficient cut-off length will lead to missing data from the surface roughness and, therefore, incorrect roughness parameters. The only way to separate the roughness from waviness is with the use of Gaussian filter according to ISO 11562 [5].

Two profiles are obtained when a surface profile is filtered: waviness and roughness. As previously stated the parameter that determines what is waviness and what is roughness is the cut-off length. Roughness cut-off length selected for a teeth will be different to the cut-off length when measuring a piston.

The cut-off length should be at least 2.5 times the peak-to-peak spacing of the profile roughness. If the surface is random roughness pattern will not exist, and in this case Table 2.3 [5] can be used.

Digital filters represent the profile by a series of numbers or ordinates that describe the profile height at regular intervals instead a smooth curve. To find the profile mean line, the average height of the profile at a given point is calculated as the arithmetic average of the ordinate heights around it.

A variation of the digital filter conceived specifically for asymmetrical surface profiles is the valley suppression filter according to DIN 4476 [5]. The filtering technique takes place in three steps:

- the surface profile is filtered with a PC digital filter to determine the mean line;
2. Literature Review

Table 2.3.: Selection of the cut-off length $\lambda_c$ according to DIN 4768.

<table>
<thead>
<tr>
<th>Cut-off peak spacing of periodic profiles (mm)</th>
<th>Measured roughness $R_a$ ($\mu$m)</th>
<th>$R_Z$(DIN) ($\mu$m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.08</td>
<td>&lt;0.032</td>
<td>-</td>
</tr>
<tr>
<td>0.25</td>
<td>0.032 - 0.1</td>
<td>&lt;0.1</td>
</tr>
<tr>
<td>0.8</td>
<td>0.1 - 0.32</td>
<td>0.1 - 2</td>
</tr>
<tr>
<td>2.5</td>
<td>0.32 - 1</td>
<td>2</td>
</tr>
<tr>
<td>8</td>
<td>&gt;1</td>
<td>&gt;10</td>
</tr>
</tbody>
</table>

• all valleys beneath the mean line are removed. Valleys are treated as a unique and independent part of the profile. The profile is filtered again with a PC digital filter;

• the mean line obtained by the second filter pass is superimposed on the original, unfiltered profile. Filtered roughness profile is the profile relative to the mean line; the mean line is straightened out to obtain the roughness profile [5] [4].

2.4. Contact mechanics

2.4.1. Types of contacts

There are three types of contacts which are defined by surface geometry:

• point contact (sphere/sphere);

• linear contact (cylinder/cylinder parallel axes);

• surface contact (plane/plane);

Hertzian contacts may only be used in point and linear contact. Figure 2.6 shows the contact between two teeth and is an example of line contact [2].
2.4. Contact mechanics

Figure 2.6.: Contact between the surfaces of the teeth of a gear. [2]

2.4.2. Hertz theory

When two elastic solids of revolution touch at a given point or in line, they deform in the neighbouring area of the initial contact, causing a small contact area. These problems are only possible to study as the contact theory allows us to define the contact area of the elliptical shape. The Hertz theory consider that both surfaces are perfectly smooth (surface roughness zero), and its geometry is defined by a mathematical function.

When loaded these two solids deform elastically and a small contact is created symmetrically distributed around the contact center line. During the application of the load the centres are moving along the zz axis and approaching each other. Of particular interest for gears is the application Hertzian theory to the line contacts.

The assumptions in the Hertz theory are:

- the material of the solids have a homogeneous, isotropic and linear elastic behaviour, as Hooke’s law;

- the load is fully normal;

- the dimensions of the contact area to be very small compared to the dimensions of the solids in contact and the dimensions of the equivalent radius of curvature [2].
2. Literature Review

As the load is normal to the tangent plane, there is no tangential force. The pressures within the surface of contact are normal to the surfaces and compressive. Outside the contact area the traction stress is null [2].

Equation 2.6 allows to calculate the half-width of the linear contact:

\[ a = \frac{R_X}{E^*p_0} \]  \hspace{1cm} (2.6)

Where:

- \( R_X \) - equivalent radius;
- \( E^* \) - effective Young’s modulus;
- \( p_0 \) - maximum Hertz pressure.

2.4.3. Influence of roughness on contact surface

The presence of roughness and waviness implies a theoretical reformulation of the problem.

Figure 2.7 show the differences between a Hertzian contact and a real contact. The pressure distribution show several pressure peaks, all higher than the maximum Hertz pressure. The contact surface decrease 40% comparatively to the Hertz contact. There are now several contact surfaces.

![Figure 2.7: Pressure distribution and area of contact between a smooth cylinder and a flat roughened surfaces: a) roughness after manufacturing b) roughness after running-in. [2]](image-url)

Figure 2.7.: Pressure distribution and area of contact between a smooth cylinder and a flat roughened surfaces: a) roughness after manufacturing b) roughness after running-in. [2]
Figure 2.7 show the influence of running-in as beneficial as it causes a substantial decrease in overpressure and an increase of the real contact area.

In summary what may be observed in Figure 2.7: roughness and waviness are factors which increase the severity of the contact as they increase pressure [2].

2.5. Gear lubrication

Osborne Reynolds in 1883 made an important distinction between laminar and turbulent flow based on former works from Beauchamp Tower, and concluded that the flow inside the contact is described by viscous flow laws. In 1886 the Reynolds equation was used universally in the Theory of Hydrodynamic Lubrication after which it was possible to calculate film thickness and load capacity.

Researchers like Pepler and Meldhal considered the elastic deformation of bodies, Gatacombe, Hersey and Lowdenslager studied the piezoviscosity effect on the lubrication film, Petrusevich not only confirmed what Grubin has predicted but also obtained solutions which simultaneously satisfied the equations of hydrodynamics and the elasticity of the surfaces for a wide operating range and two important characteristics in EHD contacts:

- a small restriction of thickness near the exit of the contact;
- an almost perfect Hertzian pressure distribution with a pressure peak at the exit related to the thickness restriction.

The oil which is dragged into the contact area is under heavy stress. The pressure may reach 1 GPa, the temperature may be higher than 100°C and the strain rate may reach $10^{-7} s^{-1}$. These conditions: the different surfaces speed, the load acting on the teeth which is transmitted to the oil, the temperature change justifies all of the changes in the lubricant and the elastic deformations of the solids in contact which are experimentally observed and determined theoretically.

Elastohydrodynamic lubrication (EHD lubrication) allows to assess three major parameters in lubricated contacts:

- determining the thickness of the film between the two surfaces in contact;
- assessing the friction between the surface in contact due to the visco-elastic plastic deformation;
- evaluating the energy balance of the contact, given the power dissipation in the lubricant film due to stresses and heat dissipation through the lubricant.

All this research has helped comprehending how gear lubrication works. [2]

2.5.1. Lubricant oil rheology

Oil rheology, thermal behaviour and film thickness are the main factors inside an EHD contact [2].
2. Literature Review

2.5.1.1. Newtonian viscosity

Rheology is the study of the physical proprieties which affect the transport of momentum in a fluid.

A Newtonian fluid is a fluid whose viscosity is the same for different rates of shear stress even with the variation in time. There is a linear relationship between the shear stress and deformation speed.

Viscosity is the resistance of a fluid to the internal sliding of its molecules.

\[
\tau = \sigma_{xy} = \eta \frac{\partial v}{\partial y}
\]  

(2.7)

In equation 2.7 \( \eta \) is the dynamic viscosity. The proportionality between the shear stress and the gradient of speed is verified experimentally in many Newtonian fluids. The ratio of the dynamic viscosity and the density is kinematic viscosity.

\[
\nu = \frac{\eta}{\rho}
\]  

(2.8)

As one may observe Figure 2.8 there are particles with different speeds.

![Laminar flow of a fluid](image)

Figure 2.8.: Laminar flow of a fluid [2].

Table 2.4.: Viscosity units.

<table>
<thead>
<tr>
<th>Viscosity</th>
<th>Dimension</th>
<th>C.G.S.</th>
<th>S.I.</th>
<th>Correspondence</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \eta ) dynamics</td>
<td>( ML^{-1}T^{-1} )</td>
<td>Poise</td>
<td>Pascal seconds</td>
<td>1cPo=1mPa.s</td>
</tr>
<tr>
<td>( \nu ) kinematics</td>
<td>( L^2T^{-1} )</td>
<td>Stokes</td>
<td>mm²/s</td>
<td>1 cSt = 1 mm²/s</td>
</tr>
</tbody>
</table>

In mineral or synthetic oils viscosity decreases when temperature increases. Figure 2.9 displays viscosity of a paraffinic mineral oil variation as the temperature increases.

There are many expressions to show this relationship, such as Cameron’s, ASTM D341.
2.5. Gear lubrication

Not only does temperature change the fluid viscosity, pressure also may do the same. However when pressure increases viscosity also increases. In gears and bearings this phenomenon has a significant importance because pressure is able to reach $10^9$ Pa.

In Table 2.5 it is possible verify that the variation is of the exponential kind and the behaviour depends on the kind of the oil. Figure 2.10 displays the variation of the logarithm of viscosity with pressure for liquids [2] [1].

<table>
<thead>
<tr>
<th>Pressure (MPa)</th>
<th>Viscosity, Paraffinic oil (Pa s)</th>
<th>Viscosity, Naphthenic oil (Pa s)</th>
<th>Viscosity, Water (Pa s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>0.052</td>
<td>0.055</td>
<td>0.001</td>
</tr>
<tr>
<td>140</td>
<td>0.81</td>
<td>2.2</td>
<td>0.00111</td>
</tr>
<tr>
<td>280</td>
<td>8.7</td>
<td>91</td>
<td>0.00123</td>
</tr>
</tbody>
</table>

![Figure 2.9. Viscosity variation with temperature for a paraffinic mineral oil (ISO VG 32 oil) [2].](image)

![Figure 2.10. Typical curve of the variation of the logarithm of viscosity with pressure for liquids [1].](image)
2. Literature Review

2.5.1.2. Non Newtonian viscosity

In a Newtonian fluid the strain rate do not have any impact on the dynamic viscosity. When it does the fluid is non Newtonian because the relationship between the strain rate and shear stress ceases to be linear. The Newtonian behaviour limit occurs when the stress cease to evolve linearly with the difference in speeds of the surfaces. Figure 2.11 shows the difference between greases, Newtonian fluid and non Newtonian fluid.

![Figure 2.11.: Viscosity variation with strain rate for various lubricants: a) greases b) Newtonian fluid c) non Newtonian fluid [2].](image)

Inside the contact the strain rate can reach $10^6$ to $10^7 \text{s}^{-1}$ and it is possible to observe a decrease in the oil viscosity. This is a feature that must be taken into account when calculating the stresses in the oil.

In this case to establish the rheological model the pressure, temperature and strain rate must be taken into account. The shear rate is an important feature because a linear relationship between shear rate and shear stress ceases to exist [2] [1].

2.5.2. Elastohydrodynamic lubrication (EHD)

To be able to analyse the EHD lubrication problem all the phenomenons involved must be known:

- the Reynolds equation;
- the equation of the elastic displacement of the surfaces;
- the balance equation of forces;
- the lubricant state equations;
- the equilibrium equations of the lubricant film and solids in contact.

The analysis considers that:

- the gearing conditions is continuous;
2.5. Gear lubrication

- the inertia and external forces are negligible;
- the film thickness is negligible when compared to other dimensions in the contact;
- the pressure do not change along the film thickness;
- the speeds of the surfaces do not change from point to point in a given direction;
- there is no slip between the fluid and the contact surfaces [2] [1].

**Reynolds’s equation** Reynolds’s equation was derived due to the in conformal contact which occur in bearings and spur gears. It is used when two faces are parallel with relative motion and there is a film thickness between them. This film thickness changes with position and time. The main goal is to obtain the hydrostatic pressure and lubricant film thickness. This may only be attainable when the surfaces suffer plastic deformation and the pressure gradient in contact is high as viscosity depends on pressure. In lubricated gearing, a surface roughness can be high enough to ensure that lubrication is boundary or mixed film. In the contact between gears it may not be wise to admit that the lubricant behaves like an Newtonian fluid.

The Reynolds’s equation derived from Naiver-Stokes equation and the mass conservation law also need some simplifications:

- the fluid is Newtonian;
- pressure and viscosity are the same in the film thickness;
- the fluid is compressible.

The Reynolds equation with simplifications:

\[ \frac{\partial}{\partial x} \left[ \frac{\rho}{\eta} (H_1 + H_2)^3 \frac{\partial p}{\partial x} \right] + \frac{\partial}{\partial y} \left[ \frac{\rho}{\eta} (H_1 + H_2)^3 \frac{\partial p}{\partial y} \right] = 6(\dot{U}_1 + \dot{U}_2) \frac{\partial}{\partial x} \left[ \rho (H_1 + H_2) \right] - 12p \left[ \dot{U}_1 \frac{\partial \varepsilon_1}{\partial x} + \dot{U}_2 \frac{\partial \varepsilon_2}{\partial x} \right] \] (2.9)

**Elasticity of the surfaces** In EHD lubricants, due the high pressure involved in the contacts, the surfaces deforms changing the lubricant film geometry.

The elastic displacement can be divided in two directions: normal and tangential. The normal displacement has the load direction. The tangential displacement is neglected.

**Balance system force** The system needs to be in balance therefore the pressure in the contact needs to be in balance with the load transmitted by the solid.

**Rheology and lubricant properties** The film thickness, the load capacity, type of flow, etc depend on lubricant behaviour and the properties of operating conditions. The complexity and importance of the rheological laws require a very detailed treatments, which in this work is not the main objective.
Energy balance of the film and solids contact  Temperature in EHD lubricant contacts may be the major influence to film thickness as viscosity is vital to film thickness. As previously stated, the viscosity changes with temperature and that change influences the film thickness.

The energy behaviour of solid/flow is considered:

- steady state;
- no outside heat sources;
- the pressures is the same along the film thickness.

Complexity and importance of heat transfer in contacts and EHD is a complex subject of which detailed study is not the purpose of this work [2].

2.5.2.1. Analytical solution of the EHD lubrication problem

The easiest way to analyse EHD lubrication problems is to divide them into two different problems as far as their features and behaviour. One is about the film thickness and deformed geometry while the other is about the study of rheology, lubricant properties and thermal balance.

These two aspects are independent:

- the film thickness is small compared to the other contact dimensions, so only a small amount of lubricant exits laterally. The rest is forced to cross all the contact and the low lubricant compressibility limits the variation of the film thickness;

- from the reduced influence of lubricant rheology and thermal contact in the formation of the thickness film comes the fact that the formation of the lubricating film is dependent on the geometric and kinematic conditions;

- between the inlet and outlet zone is where lubricants have non-Newtonian behaviour rheology and a high thermal effect, it is the high pressure zone also known as Hertzian zone. This area supports the entire load and corresponds to an area where the film thickness is approximately constant. It is in this area where the friction is expressed more intensely [2].

Figure 2.12 demonstrates the elastohydrodynamic contact.
2.5.2.2. EHD and Hertzian pressure fields

In Figure 2.13 it is possible to see the difference between the EHD and Hertzian pressure fields. The two main differences are in the inlet zone and in the outlet zone. In the first zone the EHD pressure has a more smooth growth, rising quickly until the high pressure zone. In the final zone the EHD pressure fields shows a peak and drops, while the Hertz pressure decreases in a continuous manner.

The peak shown in the EHD pressure field comes from the deformation geometry which leads to the point where the film thickness is minimal as it is possible to see in Figure 2.14 [2].
2. Literature Review

2.5.2.3. Lubricant film thickness

**Linear contacts** The Higginson’s [2] solution of minimum film thickness for linear contacts is:

\[ h_m = 1.325 R_X U^{0.70} G^{0.54} W^{-0.13} \]  \hspace{1cm} (2.10)

For central film thickness formula is given by:

\[ h_{oc} = 0.975 R_X U^{0.727} G^{0.727} W^{-0.091} \]  \hspace{1cm} (2.11)

Where:

- \( R_X \) - equivalent radius;
- \( U \) - speed parameter;
- \( G \) - material parameter;
- \( W \) - load parameter;

The influence of each term in equation 2.10 is explained in the Table 2.6.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Exponent</th>
<th>Variation range</th>
<th>Dependence</th>
</tr>
</thead>
<tbody>
<tr>
<td>( U )</td>
<td>0.700</td>
<td>big</td>
<td>very important</td>
</tr>
<tr>
<td>( R_X )</td>
<td>0.430</td>
<td>big</td>
<td>important</td>
</tr>
<tr>
<td>( E^* )</td>
<td>0.030</td>
<td>small</td>
<td>independent</td>
</tr>
<tr>
<td>( W )</td>
<td>0.130</td>
<td>big</td>
<td>less important</td>
</tr>
</tbody>
</table>

Through the analysis Table 2.6 notes speed is the most important parameter on the minimum film thickness.

Central film thickness must be corrected due to lubricant heating, the supply conditions and the surface roughness. This correction cannot be applied to the minimum
2.5. Gear lubrication

thickness of the lubricant (hm).

\[ h_{oc} = \theta_T \theta_A \theta_R h_0 \]  \hspace{1cm} (2.12)

Where

- \( \phi_T \) - heating influence in the converging;
- \( \phi_A \) - influence of power supply conditions in the converging;
- \( \phi_R \) - influence of roughness on convergent.

Specific film thickness is calculated by equation:

\[ \Lambda = \frac{h_{oc}}{\sigma} \]  \hspace{1cm} (2.13)

**Heating influence in the inlet zone on the contact EHD** Temperature increase occurs in the inlet zone of the contact due to the high shear strain caused by the rolling speed and the pressure gradient. As it is known the viscosity decreases with the increase of temperature which causes a decrease in film thickness.

**Corrections for oil supply in the inlet zone on the contact EHD** Figure 2.15 show the abscissa of the meniscus formation. If this point is closer to the inlet zone the film cannot be formed correctly causing lack of lubricant in the contact center. Despite the existence of corrections to the film thickness which takes into account the abscissa of the meniscus formation this knowledge is only obtained by experience, as the abscissa of the meniscus formation is difficult to predict [2].

![Figure 2.15: Abscissa of the meniscus formation in the convergent on the contact EHD](image)

**Corrections due to the roughness of the surfaces contact EHD** Figure 2.16 displays the surface roughness orientation over the film thickness.
2. Literature Review

2.5.3. Types of lubrication

There are three types of lubrication films (full film, mixed film and boundary film) which often occur in gearing, bearings and camshafts.

In full film \( \Lambda > 2 \) and the lubrication film is still enough to avoid contact between surfaces. The variation of the inlet geometry caused by roughness is not enough to affect the lubrication, film thickness or pressure distribution.

In mixed film \( \Lambda > 0.7 \), the lubricating film is not thick enough to avoid frequent contact between surfaces. Boundary film lubrication is not formed and the loads are sported by the contact between the surfaces.

Table 2.7 displays typical values for \( \Lambda \).
Table 2.7.: Typical values of $\lambda$

<table>
<thead>
<tr>
<th>Lubrication regime EHD</th>
<th>$\Lambda = \frac{h_0}{\tau}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Film</td>
<td>$\Lambda \geq 2$</td>
</tr>
<tr>
<td>Mixed film</td>
<td>$0.7 &lt; \Lambda &lt; 2$</td>
</tr>
<tr>
<td>Boundary film</td>
<td>$\Lambda \leq 0.7$</td>
</tr>
</tbody>
</table>

In gears along the line of engagement film thickness may change and it is possible to find various lubrication regimes. In Figure 2.17, mixed film predominates between A and C points and B and D points. In some cases, a boundary film may occur.

The zone of single contact is between C and D and full film or mixed film predominate, changing with the tangential velocity [2] [1].

![Figure 2.17.](image)

Figure 2.17.: Variation of the lubrication regime along the engagement line due to tangential velocity. FM - mixed film FC - Full film [2].

**Relationship between lubrication system and surface damage** Experimental work shows a relationship between film thickness and the probability of a surface damage (seizure, wear, and contact fatigue). Figure 2.18 shows the specific critical thickness $\Lambda_c$ of the lubricant film due to the tangential velocity ($v_t$) of the primitive to a failure rate of lower than 5%.

The analysis of Figure 2.18 in a certain tangential speed shows that the probability of a breakdown increase as specific thickness decreases. It is also possible to see that when the $\Lambda > 2$ the probability of a failure is always lower than 5%. With regard to what was previously state it is correct to say that the film thickness has an extreme importance in designing lubricated mechanical elements [2].
2. Literature Review

Figure 2.18.: Critical film thickness for a damage probability of 5% [6].

Friction between the teeth of a gear  The Stribeck curve relates the value of the friction($\mu_s$) coefficient with the specific film thickness. Coefficient of friction is defined as:

$$\mu = \mu_s = \frac{F_{Ts}}{F_N}$$  \hspace{1cm} (2.14)

Where, $F_{Ts}$ is the is the sum of two components limit and hydrodynamic (viscous). The Stribeck curve is another away to describe the lubrication regime. Mixed and boundary regime are strongly influenced by oil formulation and surface roughness. The test consists in measuring the friction coefficient, maintaining the load constant and varying only in the rolling speed.

Figure 2.19.: Stribeck’s curve [6].

The analysis of Figure 2.19 may be processed in three manners:
2.5. Gear lubrication

- in boundary film and the coefficient of friction is higher due to the roughness of the surfaces in contact;
- in the middle this is mixed film where the coefficient of friction is lower because of the sliding friction ($\mu_s$) and the hydrodynamic friction ($\mu_{SH}$) take their lower value;
- in full film the coefficient of friction starts to increase because the hydrodynamic friction ($\mu_{SH}$) also increases.

Traction curves relate sliding with the coefficient of friction ($\mu$).

\[
V_e = \frac{|U_1 - U_2|}{|U_1 + U_2|} \quad (2.15)
\]

Figure 2.20 represents the typical traction curve of variation of friction coefficient with the sliding where the curve can be divided in tree zones:

**ZONE I**

- in this zone the friction coefficient varies linearly with sliding;
- the shear rate is small;
- under low pressure, the lubricant has a viscous-linear behaviour, however if the pressure increase its behaviour becomes similar to a linear elastic solid.

**ZONE II**

- in this zone shear rates have an important role;
- the non-linear variation of viscosity $\eta$ with the sliding rate $V_e$ is explained by a non-linear viscous behaviour of the lubricant at low pressures. In high pressures the lubricant can switch to viscoelastic behaviour.
2. Literature Review

Zone I and II are typical of an isothermal contact because the dissipated power is not sufficient to cause the temperature rise.

ZONE III

- in this area thermal effects are preponderant;
- decrease of traction curve is explained because the temperature increase causes a decrease in viscosity and therefore decrease the shear stresses.

2.5.4. Temperatures of tooth flanks and lubricant

The increase of temperature in the lubricant and in the tooth surfaces is caused by higher rate of sliding and loads [6].

**Temperatures of tooth flanks** Breakdowns which occur in the gearing (severe wear, seizure), are related to two possible factors: specific film thickness and the critical temperature. The critical temperature is one of the basis of different criteria for seizure.

The temperature on the surface of teeth may be define as:

\[ T_S = T_M + T_{\text{flash}} \]  \hspace{1cm} (2.16)

Where:

- \( T_M \) - mass gear temperature (a remote point from the contact point);
- \( T_{\text{flash}} \) - localized increase in temperature in each point of the meshing line.

Figures 2.21 and 2.22 show the variation of the flash temperature and the temperature at the contact along the meshing line between two teeth of a gear [6].

![Figure 2.21: Variation the flash temperature on the meshing line [6].](image)
2.5. Gear lubrication

Temperatures of lubricant  Inside the gear contact the lubricant temperature is defined by:

\[ T_F = T_S + \Delta T_F \]  

(2.17)

Where

- \( T_F \) - lubricant temperature inside the contact;
- \( T_S \) - surfaces temperature;
- \( \Delta T_F \) - increase of lubricant temperature above the surface temperature.

![Temperature at the contact between two teeth of a gear][6]

2.5.5. Lubricants: characteristics, properties, and specifications

Lubricants have the ability to minimize wear, evacuate heat and remove particles made in the contact area, avoid corrosion, operate at higher temperatures and more demanding conditions.

For a lubricant to be acceptable it needs to be viscous enough to keep the lubricating film between the contact and it needs to be as fluid as possible to evacuate the heat and minimize the losses due to the viscous drag. Nonetheless to be categorized as a good lubricant more features are required and this explains why additives are used [2].

A lubricant is composed of base fluid and may or may not have additives. The base fluid can be synthetic or mineral.

Mineral lubricants are made mostly by natural hydrocarbons which are obtained by organic waste decomposition. These may be classified into two categories: paraffin based and naphthenic based. Synthetic lubricants are synthesized from hydrocarbons which may come from petroleum or natural products. In the last decades lubricants have evolved greatly due to industry needs for the most demanding lubrication applications.

---

[6]: An image of temperature at the contact between two teeth of a gear.
Additives are used to improve the base lubricant properties but also to give new proprieties which are not in the original lubricants. Additives may improve the viscosity index, anti-wear and they may also work like a detergent, etc [2].

2.6. Surface damage in wear

2.6.1. Propagation of a fatigue crack under cyclic contact loading

Mechanical behaviour of gears are influenced by interaction between contacts and surfaces. Surfaces in rolling and/or sliding contact are exposed to material contact fatigue. Fatigue is the damage which causes changes in the material microstructure which leads to cracks opening.

Fatigue crack life may be divided into two stages: crack initiation and crack propagation. Crack initiation occurs at the location which is under high stress, plastic deformations around inhomogeneous inclusions or other imperfections in or under the contact surface. This is the longest period. Crack propagation causes permanent damage and is influenced by time and rolling/or sliding contact loads. The sum of these two periods leading to collapse of material has the same value as found in the SN curves.

Fatigue cracks initiation represents a very important stage in the pitting process. The position and the mode of fatigue crack initiation depends on each specimen, the type of applied stress and the microstructure of the material.

The crack initiation periods are different and may be initiated on or under the surface, depending on the combination of rolling/sliding contact conditions. Sub-surface pitting usually happens in high quality gears made of alloy steel with the use of good lubrication and where the surfaces contact is smooth. Surface pitting is influenced by roughness and other surface defects (machining marks, inclusions). The thermal treatment caused by residual stress may make surface cracks.

The surface micropitting process does not have a growing period and may remain unaltered for a long period of time even if they are exposed to stress concentration factors, which may cause new fatigue cracks. Otherwise the surface pitting process is also the formation of small surface cracks which extend under repeated contact loading but if these cracks reach a certain size, this may lead to an unstable growth which leads to the breakaway of the surface layer.

The number of stress cycles \( N \) to pitting occurring in a tooth flank is the sum of the number of stress cycles \( N_i \) required for the appearance of initial crack in the material and the number of stress cycles \( N_p \) needed for a cracks to propagate from the initial to the critical crack length [1] [7]

\[
N = N_i + N_p \quad (2.18)
\]

Figure 2.23 show the beginning of crack formation due to the shear stresses located along the surface.
2.6. Surface damage in wear

The orientation of these cracks to the surface depends on the direction of sliding speed which tends to cause metal deformation.

Fatigue cracks are generated under the surface and in surface craters. Even though the surfaces are under a lot of stress, they do not interact much. Shear stress has very high values immediately under the surface which can reach 1.5 GPa. These cracks are made because the metal grains are being deformed and cause the sliding of crystallines plans. These deformations occur during the load cycles and this is how microcracks occur. With the accumulation of cycles the growing cracks tend to head for the surface and eventually small particles are loosened making holes in the surface.

This loosen particles can cause abrasion in 3 bodies and the holes prevent the correct formation of the lubricant film. This wear process may happen even when the materials in contact are hard and well lubricated. Micropitting and pitting are two very similar types of wear small holes invisible to the naked eye, only a dull spot visible. This occurs in well lubricated contacts when the loads are high. In many cases after the running-in the micropitting disappears. Mild wear always occurs, only micropitting may be avoided.

2.6.2. Influence of lubricant on Pitting and Micropitting

Pitting and micropitting are two of the main causes of rolling contact fatigue surface failure which are influenced by the lubricant and lubricant conditions.

Figure 2.24.: Main limits of the load capacity a) through hardened steel b) case carburized steel [9].
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Figure 2.24 displays torque limit with the increase of the tangential speed. Carburized steel has a larger area of non-deterioration than hardened steel and both graphs limits are the micro pitting line which is higher in the right side. Therefore carbonizing treatment is advantageous. Increase of fracture limit can be obtained if the modulus increases and so the limit becomes the pitting line.

In general pitting and micropitting occur in harden surface, and pitting leads to failure. The test conditions (temperature, tangential speed, load), tooth geometry, and the lubricant used determine the stress and the wear in a spur gear.

It is easily verifiable that the use of lubricants with additives deliver different results. Lubricants doped with sulfurphosphorus additives (S-P) are advantageous to micropitting resistance, while on the other hand ZDP-additives are not. The same happens when the gear geometry are the same and the materials are different [9].

2.7. Wear

Wear study is not an exact science, it incorporates many scientific disciplines and principles whose complex inter-relationships may give rise to considerable areas of uncertainty. Many definitions of wear have been proposed; the following is given in specification $DIN^{1979}$ [10];

Wear is the progressive loss of substance from the surface of a solid body caused by mechanical action i.e., contact and relative motion with a solid, liquid or gaseous counter-body

Wear resistance is not an intrinsic material property. Many industrialists hope for a wear test equivalent of the hardness or tensile test and it remains difficult for some to understand why this is not possible. Changes to surface and near surface structures during wear contact normally significantly alter local material properties, both mechanically and chemically and, between different wear situations, so many variables apply [11].

Wear behaviour is normally divided into two time based categories, "running-in" and "steady state". During steady state, wear conditions are relatively stable and can be comparatively examined. During running-in conditions are far more complex and variable, due to work hardening, surface chemistry changes, plastic deformation of roughness and material phase changes.

Surface wear is considered one of the major failure modes in gear systems. Dry or lubricated gears are usually subjected to some mild wear. The general opinion is that mild wear improves the contact conditions such as a post-polishing process of gear flanks. That may be true to some extent but not in general, since the wear of the gear flanks will always be non-uniform due to the varying rolling sliding conditions between the interacting gear teeth. Due to the fact that there are numerous factors affecting wear and that their effects may not be determined exactly, studies in attempting to solve this problem are effectively under way. Study of wear of gear contact is becoming one of the emerging areas in gear technology. Usually, machine components are subjected to fatigue and gears are not different. Pitting, wear and other fatigue effects are often seen on gears specially on single meshing points, as
these are the areas where Hertz surface pressure is maximum. Since the contact is a rolling sliding contact, the profile changes due to wear may be harmful for the contact conditions. Interacting gear teeth roll and slide against each other under high contact pressure. The amount of wear that is acceptable depends on the expected life, noise and vibration of the gear units. Excessive wear is characterized by loss of tooth profile, which results in high loading and loss of tooth thickness, which may cause bending fatigue [12].

2.7. Failure Process

Several theories exist to describe the micro-scale mechanism of wear particle formation in mild wear. The macroscopic circumstances causing mild wear:

- two solid surfaces are in contact under normal load (not fully separated by an intervening fluid film);
- sliding takes place between the surfaces;
- the normal pressure and the sliding velocity in combination are less severe than the threshold condition for severe wear (removal of surface material by the application of tractive forces in multi-asperity dimensions).

It is assumed that the tractive forces operating between asperities sliding over each other are the cause of the particle removal. The wear rate may be speed-independent under conditions where temperature and lubrication parameters do not change with speed [13].

2.7.2. Kinds of wear

Wear may be classified as:

- abrasion

Abrasion is the removal by roughness material. This can be viewed as the removal of material due to the indentation of hard asperities into a softer surface. It occurs when there is relative movement of the contacting surfaces. Tenacity is another property which is important as it prevent cracks opening in the harder solid. Abrasion may be viewed as a primary wear mechanism. The chance of the abrasive wear mechanism occurring in isolation is rarely possible. High temperatures can be generated at the sliding surfaces, thus promoting adhesion and fatigue fracture. Abrasion 3 bodies occurs when the contact have particles inside. This wear process is precipitated by sliding in the contact. The track of the asperity on the opposite surface can often be seen as a scratch designated a kinematic wear mark. In spur gears, marks follow a gear tooth profile line [13]. These particles travel inside the contact and make high stress concentration causing severe wear. These particles also causes scuffing. Filtration system in a circulating oil system will greatly reduce the particles.
2. Literature Review

- erosion

Erosion is the surface attack by particles. This is the removal (pulling out) of small particles from the surface by the impact of repeated flow. The amount of wear is determined by the speed, weight and the attack angle on the surface by the flow. Frontal impact is better supported by ductile surface while harder surface resists better to shear impacts.

- scoring - adhesive wear

Scoring - adhesive wear, is the welding between the contact points. Even polished surface have a degree of roughness and as a result, between most engineering surfaces, there is only asperity contact with the true contact area being far less than apparent and with initial contact stress far higher than apparent. The metal in contacts can warm up due to the free heat caused by friction. If temperatures rise enough, melting may occur and consequently the welding of contacts points. After the welding occurs, that section of the metal is pulled out due to the movement, making a wear particle. If the particle remains on the surface, the lubrication film is affected. Gears operating in low speed and high load are especially prone to adhesive wear due to the lubricant operating in the boundary or mixed lubrication mode.

- fretting

Fretting is sometimes described as a wear "mechanism" whereas it is a wear phenomenon associated with small movement (vibratory) of the contacting bodies. Wear particles remain inside the contact causing 3 bodies abrasion. Wear tends to be highly dependent on this tribochemical mechanism. Recent works, makes distinction between "fretting wear" (as described above) and "fretting fatigue", where cracks are initiated within, or at the edge of, loaded contacts. These forms of damage can co-exist although their origins are different.

- running-in

Running-in is a positive thing to do with low loads as it softens the roughness which increases the lifetime of the material.

- cavitation

Cavitation occurs when air bubbles present in the lubricating film collides with surface teeth.

- overheating

Overheating occurs on the gearing and may cause the heat treatment which prompts a decrease in surface hardness.

- corrosion;

Corrosion is a chemical reaction which happens on surface which can lead to surface damage. It can not be classified as wear [3] [1] [14] [12].
2.7.3. Experimental observations of wear and surface damage

Most wear observations are carried out indirectly (post factum). The rubbing process must be stopped, the worn out elements must be disassembled, and after that the effects of the wear process can be observed. Weighting is the simplest way of detecting wear. It gives the total amount of the removed mass, but the distribution of the wear depth in the contact surface is unknown.

In order to obtain qualitative information on wear, after opening the contact, visual inspection of the worn surface and wear debris is very often used.

The easiest method of the visual inspection of surface damage is to photograph the surface. Furthermore, the worn surfaces can be examined with the aid of optical microscopes and with scanning and transmission electron microscopes. The surface examination by microscope provides a two-dimensional view. To determine how much material had been removed, surface topographic measurements must be preform [14] [11].

It is possible to detect and measure wear in three ways:

- temperature monitoring;
- vibration monitoring;
- oil sample.

Temperature monitoring is an action which helps to avoid machine failure, because if the temperature rises it means the coefficient of friction rises as well. This is possible to monitor before any severe damage is caused. The monitoring of vibration is also very effective as it allows to identify when machine behaviour changes. When the machine is lubricated, it is easy to remove an oil sample and measure how many wear particles are in the sample. This process may indicate before the temperature rise if a breakdown will occur [3].

2.7.4. Evolution of wear

The life of components is limited by wear; even when wear does not exist, fatigue may decrease materials strength. As one may observe in Figure 2.25 the surface wear may be divided into 3 periods. In the first period, the running-in period, the surface roughness may be smoothed and the wear is high. After running-in, in the second period, the wear is low and remains low during the component lifetime. When fatigue is high, any micropitting which may exist interferes with the contact and wear starts to rise again. This is when the end of the lifetime is near and this is the third and last period [3].

Initial wear, running-in, is an expected phenomenon in many gears systems, due to forced sliding of teeth at all locations away from the pitch circle. In the second period the continuing wear damages the geometry and finish of the teeth. It is often abrasive, that is, it brings on three body wear by contamination with extraneous hard particles or wear debris [13].
2. Literature Review

2.7.5. Effects of mild wear

Mild wear has damaging effects on contact components. Damage occurs primarily through the following mechanisms: material removal distorts component geometry. As a result, surface pressure is often redistributed unfavourably. In addition, geometric relationships that influence machine accuracy, backlash (clearance or lost motion in a mechanism caused by gaps between the parts), quietness, may be impaired. Surface finish is modified, which may either improve or destroy as-made surface microgeometry. If material removal is substantial, gears may be impaired. Finally, components which have been surface-treated for wear, fatigue or corrosion resistance, or for solid lubrication, may lose some or all of these layers [13].

2.7.6. Archard’s law

Friction and wear depend as much on sliding conditions (the normal pressure and the sliding speed) as on properties of materials concerned. Normal pressure and sliding action are necessary for wear. Mechanical wear is a result of the mechanical action. Therefore, the wear process discussed in this study results of all on the rubbing process.

The earliest contributions to wear established a relationship for the volume of the material removed by wear ($\Delta V$) in the sliding distance ($s$) and related it to the true area of contact. Archard (1953) formulated the wear equation of the form: the volume of the material removed ($\Delta V$) is directly proportional to the sliding distance ($s$), the normal pressure ($F_N$) and the wear coefficient ($K$), and inversely proportional to the hardness of the surface being worn away ($H$). $K$ is higher for more severe stress conditions, and usually higher for ‘less wear resistant’ materials, rougher surfaces, and higher frictions coefficients. Eq.2.19 is Archard’s law

$$ \frac{\Delta V}{S} = K \frac{1}{H} \times F_N $$  \hspace{1cm} (2.19)

Note that if the sliding distance is the result of sliding at constant velocity $U$, it is then given by:

$$ L = Ut $$  \hspace{1cm} (2.20)
Where:

- \( t \) - sliding time;

Eq. 2.19 can be manipulated by dividing both sides by the nominal (apparent) contact area \( A_n \) and by substituting the sliding distance in terms of sliding velocity and time and by solving for time

\[
t = \frac{dH}{Kp_mU}
\]  

(2.21)

Where:

- \( d \) - is the worn depth;
- \( p_m \) - is the mean or nominal pressure.

This is an indication of the life of a wearing component in terms of the admissible worn depth and the material and process parameter \( H, K, p_m \) and \( U \).

It is often the case, that measured worn volumes vary in direct proportion with the total sliding distance. In contrast, while worn volumes often vary in proportion with the applied load over certain load ranges, abrupt changes in wear rates (wear transitions) are observed at specific critical loads. Such changes are the result of the complex interplay between the softening and chemically reacting behaviours of the material induced by high flash temperatures. Abrupt increases in wear rates are commonly found at high loads and these are often associated with welding and seizure. However, in some cases these high wear rates may revert to low values at even higher loads [14], [15].

2.7.7. Summary

Wear cannot be eliminated completely, but it can be reduced. The simplest methods of reduction of friction and wear are as follows: lubrication, formation of sufficiently smooth surfaces, modification of near-surface materials of rubbing components, correct assembling. Friction and wear can be reduced by an optimal choice of structural, kinematical and material parameters of mechanical systems realized by: correct choice of shapes of rubbing elements, forming of loads and motions in adequate limits, correct choice of sliding materials.

Wear in gears can be reduced through the profile shift coefficient (Henriot [2]). To equalize the specific sliding between wheel and pinion. However the profile shift coefficient has advantages and disadvantages such as: a positive profile shift coefficient increase the specific film thickness but also increase the sliding speed. Increase of the specific film thickness reduces the contact between the wheel and the pinion, but increasing the sliding speed causes increased wear. The choice of lubricant with high viscosity is also beneficial to reduce wear as well as the lubricant base.
3. Gear tests for wear characterization

3.1. Introduction

This work focuses on the study of wear in spur gears. To aid in the understanding of what influences the wear on gears three groups of test were made: variation with the load, variation with Λ and variation with lubricant with regard to a central test.

Through experimental results will check the influence of the operating conditions on the wear coefficient (K) which is used in Archard’ law, will be checked in Chapter 5.

In this Chapter are all results obtained experimentally.

3.2. Test equipment

3.2.1. FZG test rig

![FZG gear test rig](image)

Figure 3.1.: FZG gear test rig [16].

The FZG test rig, shown in Figure 3.1 and in more detail in Figure 3.2, is used to test typical gear failure modes: breakage, pitting, micropitting, scoring and wear. This test is standardized in DIN 51 354 as the FZG Test. It is a back-to-back spur gear rig in closed power loop type. The motor is connected to shaft no.2 and it is
3. Gear tests for wear characterization

Figure 3.2.: Left: gears details; Right: torque cell [17].

used to compensate power losses in the system. The shaft no.1 is divided in two parts
with the load clutch between it. It is made with two flanges. When they are loaded
they are twisted in relation to each other and, after the load application are bolted
together. It is possible to use different load levels, by using different weights.

The FZG has two gearboxes: one is a test gearbox and the other a drive gearbox.
In the test gearbox different types of spur gear may be tested: type A and type C.
The test gear type A with high sliding at the pinion tip is used for scoring, wear and
shear stability tests. The test gear type C with balanced sliding at the tip of the
pinion and the wheel is used for pitting and micropitting tests [16]. In every single
test one pair of new spur gear and new lubricant is used. The wheel is placed in shaft
no.2 and the pinion in shaft no.1.

Both gear boxes need to be lubricated. The tests can be performed using dip
lubrication or oil jet lubrication. Under oil jet lubrication conditions, the oil is in a
reservoir with heaters that can increase the temperature of the oil up to the desired
value. After this, the temperature is controlled by the feedback of the temperature
sensor in the tube of the reservoir. The reservoir includes an oil pump to put the oil
into circulation to the gearboxes and it is possible to select the oil flow [18] [16] [17].

The following tests can be performed on the rig:

- FZG gear oil test - to determine the scoring load capacity and also to determine
  the wear characteristics of various gear lubricants with the use of the gravimetric
  method [16];

- FZG pitting test - to determine the influence of different gear oils and additives
  on the pitting load capacity of the gear;

- FZG micropitting test - to determine the influence of different oils and additives
  on micro-pitting [16];

- shear stability test - to determine the shear behaviour of multi-grade gears oils
  with VI- improvements [16];
• investigations on gear damages - tooth breakage, pitting, wear and scoring. S-N curves for pitting resistance of various materials [16].

3.2. Test equipment

3.2.1. FZG measurement equipment

The FZG test rig available at CETRIB has the possibility to measure the torque loss and the operating temperatures. To measure and record the operating temperatures type K thermocouples were used and were assembled at specific points as shown in Figure 3.3. The temperature acquisition was made with a data acquisition board with eight thermocouples and the temperatures that each device measures is presented in Table 3.1.

![Figure 3.3.: FZG with thermocouples.](image)

<table>
<thead>
<tr>
<th>Number</th>
<th>Temperature measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Out oil</td>
</tr>
<tr>
<td>2</td>
<td>Test gearbox bearing case</td>
</tr>
<tr>
<td>3</td>
<td>Test gearbox cover</td>
</tr>
<tr>
<td>4</td>
<td>Oil injection</td>
</tr>
<tr>
<td>5</td>
<td>Room</td>
</tr>
<tr>
<td>6</td>
<td>Drive gearbox bearing case</td>
</tr>
<tr>
<td>7</td>
<td>Drive gearbox cover</td>
</tr>
<tr>
<td>8</td>
<td>Mass temperature</td>
</tr>
</tbody>
</table>

To measure torque loss an ETH Messtechnik DRDL-II torque transducer was used assembled on the FZG test rig between the drive gearbox and the DC motor. A sensor interface (ValueMaster V.2.43) was used to gather data, communicating with a PC via Ethernet.

3.2.1.2. FZG rolling bearings and seals

The FZG machine gearboxes have rolling bearings and seals. The rolling bearings mainly ensure the positioning and loads support of rotating parts, while the seals avoid oil leakage. They are:
3. Gear tests for wear characterization

- two four point contact ball bearings;
- six single row cylindrical roller bearings.

For the tests performed cylindrical roller bearings were used (because there was no axial load). The FZG machine has a total of five seals. The driving gearbox has three seals, while the seal on the motor side is different from the others (26 mm shafts diameter), and the test gearbox has two seals. All seals are made with the same material (viton lip seals) differing only in inner diameter.

3.3. Roughness measurement equipment

3.3.1. 3D Optical Microscopy

3D microscope which allows to analyse and obtain surface roughness. The equipment works with interferometry principle. Additional information about the equipment and the measuring technique can be found in [19].

3.3.2. Profilometer

Profilometer is a surface roughness and topography measurement system that provides standard roughness parameters, surface waviness and irregularities. A diamond stylus is moved horizontally in contact with a sample and then moved laterally across the sample for a specified distance. A profilometer can measure small surface variations in horizontal stylus displacement as a function of position. A typical profilometer can measure small horizontal features ranging in height from 10 nanometres to 1 millimetre. The height position of the diamond stylus generates an analog signal which is converted into a digital signal stored, analyzed and displayed. The radius of diamond stylus ranges from 20 nanometres to 50 µm, and the horizontal resolution is controlled by the scan speed and data signal sampling rate.

3.4. Lubricant analysis equipment

3.4.1. Direct reading ferrography
3.4. Lubricant analysis equipment

![Diagram of ferrography analysis](image)

Figure 3.4.: Representative sketch of direct reading ferrography [20].

Direct reading ferrography analysis is used to quantify the particles in a given volume of lubricant. Figure 3.4 shows the measurement equipment used. For the analysis an oil sample is recovered from the mechanical system in study. 1 ml of the sample crosses a capillary which is under a strong magnetic field and two light beams. Due to the magnetic field the oil particles travel to the bottom of the capillary. The biggest particles go to the bottom first then the smaller ones. The particles density is measured by an optical system, which quantifies the light intensity which crosses the capillary. This light intensity is inversely proportional to the density of deposited particles [20].

3.4.2. Ferrography

![Diagram of ferrography analysis](image)

Figure 3.5.: Representative sketch of ferrography direct analysis [20].

Analytical ferrography, is used to obtain detailed information on lubricant contam-
3. Gear tests for wear characterization

inants. Fig. 3.5 shows the measurement equipment used. The particles are deposited with the same principle of direct reading ferrograph, but in this case on a glass slide - the ferrogram. The larger particles settle to the entrance of the ferrogram and will decrease progressively in size.

Although this technique is more efficient in detecting ferrous particles, the non-ferromagnetic particles, such as copper alloys, aluminum, are also deposited as they acquire some magnetism as a result of friction with steel by being trapped between the filaments of ferrous particles, or simply by sedimentation.

When the ferrogram is finished a microscope is used to see the features of the particles such as:

- dimensions;
- morphology;
- color and brightness;
- type of surface;
- type edges.

Some of these features are associated with different types of wear and other identification of the material that is being worn. Thus, it is possible to determine the wear process (normal fatigue, abrasive, corrosive, etc.) which is being developed in the machine and identify the particular component which is deteriorated.

After observing and photographically recording the most relevant ferrogram zones, the next step is to proceed with the use of a thermal plate a heat treatment. The thermal treatment of ferrogram allows the distinction of various metals, especially in the different leagues of steels (low, medium and high alloy).

3.5. Test samples and materials

3.5.1. Gears Geometries

Type C gears [21] used in this work are balanced with regard to sliding velocity. They have a smaller tooth width compared with type A gears (14 mm-20mm). This is taken into account by the choice of a smaller loading lever arm (0.35 m instead of 0.50 m) when setting the torque so that the Hertzian contact pressure at the pitch point in each load stage during the micropitting test corresponds approximately to the pressure which is achieved with the same numbered load stage in the scuffing test (DIN 51 354).

The test gears are case-carburized (16MnCr5), with a surface hardness 750 HV1, in the area of the tooth flank. Case hardening depth at 550 HV1: 0.8 - 1.0 mm (after grinding). The zone close to the surface shall have no residual austenite content visible in the microscope (< 20%).

A reference value of $R_a = 0.5 \mu m$ is desirable for the mean flank roughness of test gear pairs.

Table 3.2 displays the main geometric properties of the test gears set, shown in Figure 3.6. A set of four gears were used for doing all the tests.
### 3.5. Test samples and materials

Table 3.2.: Geometric properties of the gears [18].

<table>
<thead>
<tr>
<th></th>
<th>FZG C - CF</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Material</strong></td>
<td>16MnCr5</td>
</tr>
<tr>
<td><strong>Treatment</strong></td>
<td>case-carburized</td>
</tr>
<tr>
<td><strong>Surface hardness</strong></td>
<td>750 Hv1</td>
</tr>
<tr>
<td><strong>DIN 3962 grade</strong></td>
<td>5</td>
</tr>
<tr>
<td><strong>a [mm]</strong></td>
<td>91.5</td>
</tr>
<tr>
<td><strong>m [mm]</strong></td>
<td>4.5</td>
</tr>
<tr>
<td><strong>α [°]</strong></td>
<td>20</td>
</tr>
<tr>
<td><strong>Pinion</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Z₁ [/]</strong></td>
<td>16</td>
</tr>
<tr>
<td><strong>b₁ [mm]</strong></td>
<td>14</td>
</tr>
<tr>
<td><strong>x₁ [/]</strong></td>
<td>0.1817</td>
</tr>
<tr>
<td><strong>d₁ₙ₁ [mm]</strong></td>
<td>82.46</td>
</tr>
<tr>
<td><strong>σ₁ [µm]</strong></td>
<td>0.56</td>
</tr>
<tr>
<td><strong>Wheel</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Z₂ [/]</strong></td>
<td>24</td>
</tr>
<tr>
<td><strong>b₂ [mm]</strong></td>
<td>14</td>
</tr>
<tr>
<td><strong>x₂ [/]</strong></td>
<td>0.1715</td>
</tr>
<tr>
<td><strong>d₂ₙ₂ [mm]</strong></td>
<td>118.36</td>
</tr>
<tr>
<td><strong>σ₂ [µm]</strong></td>
<td>0.56</td>
</tr>
<tr>
<td><strong>Profile modif:</strong></td>
<td></td>
</tr>
<tr>
<td>Tip relief</td>
<td>no</td>
</tr>
<tr>
<td>Root relief</td>
<td>no</td>
</tr>
<tr>
<td>Crowning</td>
<td>no</td>
</tr>
</tbody>
</table>

Profile modif:
3. Gear tests for wear characterization

![Figure 3.6.: FZG gear type C [6].](image)

### 3.5.2. Gear oils

For this work two gear oil were tested, one polyalphaolefin base oil, Renolin Unisyn CLP 150 and one mineral oil, Energol GR-XP 150 both presenting a viscosity grade ISO VG 150. The properties of each lubricant are provided by manufacturers. Some of the oils proprieties may be seen in Table 3.3.

<table>
<thead>
<tr>
<th></th>
<th>Energol GR-XP 150</th>
<th>Renolin Unisyn CLP 150</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical content</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Zn (ppm)</td>
<td>0</td>
<td>n/a</td>
</tr>
<tr>
<td>Ca (ppm)</td>
<td>40</td>
<td>n/a</td>
</tr>
<tr>
<td>P (ppm)</td>
<td>175</td>
<td>n/a</td>
</tr>
<tr>
<td>S (ppm)</td>
<td>15040</td>
<td>n/a</td>
</tr>
<tr>
<td>Biodegradability and toxicity (standards OECD 101, 202, 301 F)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ready biodegradability (%)</td>
<td>&lt; 60</td>
<td>n/a</td>
</tr>
<tr>
<td>Aquatic toxicity with Daphnia $EL_{50}$ (mg/l)</td>
<td>&gt; 1000</td>
<td>n/a</td>
</tr>
<tr>
<td>Aquatic toxicity with Daphnia $EL_{50}$ (mg/l)</td>
<td>&gt; 100</td>
<td>n/a</td>
</tr>
<tr>
<td>Density at 15°C $\rho_{15} (kg/m^3)$</td>
<td>897</td>
<td>863</td>
</tr>
<tr>
<td>Kinematic viscosity at 40°C $\nu_{40}$ (cSt)</td>
<td>150</td>
<td>150</td>
</tr>
<tr>
<td>Kinematic viscosity at 100°C $\nu_{100}$ (cSt)</td>
<td>14.6</td>
<td>19.4</td>
</tr>
<tr>
<td>Viscosity index $VI$</td>
<td>96</td>
<td>148</td>
</tr>
</tbody>
</table>

Renolin Unisyn CLP 150 is a commercial polyalphaolefin based oil with significant residual sulphur content. It is formulated with an additive system designed to provide protection against conventional wear modes such as scuffing as well as micropitting fatigue.

Energol GR-XP 150 gear oil range of high quality lubricants are based upon highly refined mineral oil, enhanced with sulphur/phosphorus extreme pressure ad-
3.6. Testing procedure

3.6.1. Pre-test gear measurements and operations

This section describes in detail the way all tests and measurements was performed.

- the first step was to mark the side of the pinion and the wheel with A and B. This was done because each spur gear can be used twice, as only one side is under load, while the other does not suffer any damage. The remaining teeth are noticeable: on the wheel teeth no.1, 9 and 17; on the pinion teeth no.1 and 9, which come into contact with the wheel teeth, and extra tooth no.5 as displayed in Figure 3.7;

![Figure 3.7. Marks in the wheel and in the pinion.](image)

- next, gears were cleaned for five minutes in an ultrasonic bath with petroleum ether;

- next, the pinion was weighted. In the weighing process only the pinion mass was measured in comparison with a standard pattern gear. The mass measurement was done five times for each pinion and the average was retained;

- a topography measurement was carried out in each tooth pinion and wheel mentioned above. A handmade mark was made which served as a reference to compare the surfaces before and after it was used. A roughness measurement was carried out just like the topography measurement was;

- the FZG gearbox was cleaned with petroleum ether;

- the gears were mounted on the shafts with interference;
3. Gear tests for wear characterization

- one litre of lubricant (PAO or Mineral) was added to the test gearbox to lubricate the test gears. Only one litre was added as it was enough to cover the lower tooth of the pinion.

3.6.2. Test operating conditions

Table 3.4 displays all the features and the operating conditions of the seven tests that were performed.

<table>
<thead>
<tr>
<th>ID</th>
<th>oil</th>
<th>pinion</th>
<th>wheel</th>
<th>Temp</th>
<th>n₁ [rpm]</th>
<th>load stage</th>
<th>( \bar{\lambda} )</th>
<th>( \lambda )</th>
<th>N° of cycles</th>
</tr>
</thead>
<tbody>
<tr>
<td>T01</td>
<td>P1</td>
<td>FZG C-CF</td>
<td>FZG C-CF</td>
<td>100</td>
<td>750</td>
<td>K9 (b=0.35m)</td>
<td>0.12</td>
<td>0.12</td>
<td>3600000</td>
</tr>
<tr>
<td>T02</td>
<td>P1</td>
<td>FZG C-CF</td>
<td>FZG C-CF</td>
<td>80</td>
<td>4500</td>
<td>K9 (b=0.35m)</td>
<td>0.46</td>
<td>0.65</td>
<td>3600000</td>
</tr>
<tr>
<td>T03</td>
<td>P1</td>
<td>FZG C-CF</td>
<td>FZG C-CF</td>
<td>90</td>
<td>2550</td>
<td>K9 (b=0.35m)</td>
<td>0.30</td>
<td>0.36</td>
<td>3600000</td>
</tr>
<tr>
<td>T03b</td>
<td>P1</td>
<td>FZG C-CF</td>
<td>FZG C-CF</td>
<td>90</td>
<td>2550</td>
<td>K9 (b=0.35m)</td>
<td>0.30</td>
<td>0.36</td>
<td>3600000</td>
</tr>
<tr>
<td>T04</td>
<td>M1</td>
<td>FZG C-CF</td>
<td>FZG C-CF</td>
<td>100</td>
<td>750</td>
<td>K9 (b=0.35m)</td>
<td>0.12</td>
<td>0.12</td>
<td>3600000</td>
</tr>
</tbody>
</table>

ID in Table 3.4 is the name of the test, temp is the temperature at each test was performed, load stage is the load at each test was performed and \( n_1 \) is the wheel rotation speed. Table 3.5 display the value of each FZG load stage.

<table>
<thead>
<tr>
<th>FZG load stage</th>
<th>Torque on pinion [N.m]</th>
<th>Torque on wheel [N.m]</th>
<th>Contact force [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>K5</td>
<td>69.98</td>
<td>104.98</td>
<td>2069.23</td>
</tr>
<tr>
<td>K7</td>
<td>132.48</td>
<td>198.72</td>
<td>3916.77</td>
</tr>
<tr>
<td>K9</td>
<td>215.57</td>
<td>323.35</td>
<td>6373.24</td>
</tr>
</tbody>
</table>

When controlling the unit, temperature, type of lubrication, speed and numbers of cycles were set. A laptop was used to record temperature and the torque loss.

3.6.3. Post test measurements

When the test is completed an oil sample is taken to be analysed. The cleaning, topography and roughness procedure was are identical to those performed before the test.

The topography and roughness measurement were carried out in the profilometer and in the 3D optical microscopy. In the profilometer the methodology used was to
take roughness measurement, followed by one topography measurement and lastly another roughness measurement.

It should be noted that the positioning of the wheel or pinion, to carry out the topography is a manual process that is prone to errors. Various methods were tried, but the most the most scientific was chosen. It is an automatic process that allows to adjust the surface to involute by an optimization process through least square method. It is not possible to say with 100 % certainty that the profile compared are the same, but it is the most accurate process available at the moment. Through Matlab it was possible to obtain the roughness parameters by applying a Gaussian filter with a cut-off of 0.8 mm.

Topographies were made in T04, T11, T12 and T03b test suffered some disturbance during the measurement. Figure 3.8 displays the disturbance suffered during the measurement. An anomalous and probably fictitious undulation of approximately 6 \( \mu m \) in wavelength is superimposed in the roughness.

Table 3.6 displays the tooth number and which gear was affected.

<table>
<thead>
<tr>
<th></th>
<th>T11</th>
<th>T12</th>
</tr>
</thead>
<tbody>
<tr>
<td>P01</td>
<td>Used</td>
<td>-</td>
</tr>
<tr>
<td>P09</td>
<td>Used</td>
<td>Used</td>
</tr>
<tr>
<td>W01</td>
<td>-</td>
<td>Used</td>
</tr>
<tr>
<td>W09</td>
<td>-</td>
<td>Used</td>
</tr>
<tr>
<td>W17</td>
<td>-</td>
<td>Used</td>
</tr>
</tbody>
</table>

Where:

- P - pinion;
- W - wheel;
- 01- tooth marked as number 1;
- 09 - tooth marked as number 9;
- 17 - tooth marked as number 17;
- Used - only used topographies were affected.

Some tests were done to diagnose the problem but did not reach any conclusion regarding the equipment anomaly. Of the various possibilities to solve this problem, the solution adopted was applying another filter to eliminate this disturbance. The filter used eliminates small wavelength with a cut-off of 13 \( \mu m \).

Applying this second filter is not an ideal solution, because some of real roughness will be deleted, but it was the only solution found. Figure 3.8 display a roughness profile affected and the Figure 3.9 display the effects application of the second filter.
Table 3.7 displays the difference in roughness parameters without and with the second filter.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>New</th>
<th>Used</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_{\text{amax}}$</td>
<td>3.217</td>
<td>2.985</td>
</tr>
<tr>
<td>$R_{\text{ZDin}}$</td>
<td>3.092</td>
<td>2.726</td>
</tr>
<tr>
<td>$R_{\text{a}}$</td>
<td>0.405</td>
<td>0.199</td>
</tr>
<tr>
<td>$R_{q}$</td>
<td>0.518</td>
<td>0.451</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.375</td>
<td>0.201</td>
</tr>
<tr>
<td>$R_{k}$</td>
<td>1.347</td>
<td>1.230</td>
</tr>
<tr>
<td>$R_{vk}$</td>
<td>0.685</td>
<td>0.543</td>
</tr>
</tbody>
</table>

In the Table 3.7 a difference in the $R_{vk}$ and $R_{pk}$ was around 0.2/0.3 $\mu m$, which does not really affect the roughness values.

To perform the analysis of the oil collected after each test the analysis may be completed with two methods: direct reading ferrography and ferrography. To perform direct reading ferrography, first it is necessary to dilute, warm and shake (prepare) the sample to be analysed. Once the analysis is finished two values are obtained $D_l$ and $D_s$ representing respectively the quantities of large particles and small particles. Substituting the values obtained in equations 3.1 and 3.2 it is possible to interpret the results.

$$CPUC(\text{Wear Particle Concentration}) = \frac{D_l + D_s}{d}$$  \hspace{1cm} (3.1)
3.7. Test results and discussions

3.7.1. Pinion mass loss

Table 3.8 display the measured pinion mass loss from all tests.

<table>
<thead>
<tr>
<th>ID Test</th>
<th>Mass loss [mg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>T01</td>
<td>31.4</td>
</tr>
<tr>
<td>T02</td>
<td>28.8</td>
</tr>
<tr>
<td>T03</td>
<td>44.2</td>
</tr>
<tr>
<td>T03b</td>
<td>38.8</td>
</tr>
<tr>
<td>T11</td>
<td>14.6</td>
</tr>
<tr>
<td>T12</td>
<td>11.4</td>
</tr>
<tr>
<td>T04</td>
<td>48</td>
</tr>
</tbody>
</table>

Results of T03 and T03b tests for mass loss were not consistent with the Archard wear law, as shown in Table 3.8.

3.7.1.1. Influence of load

Figure 3.10 shows how mass loss during tests T12, T11 and T01. These defer only in the contact force applied. Figure 3.10 shows how loss is attached by load.
3. Gear tests for wear characterization

![Graph showing contact force vs. mass loss](image)

Figure 3.10.: Influence of contact force on mass loss.

Resulting values for mass loss from Table 3.8 are more or less coherent with the Archard wear equation. Least amount of wear loss is associated with the load. However, the variation is not linear with contact force, as would be expected if Archard’s wear coefficient were constant.

### 3.7.1.2. Influence of film thickness

![Graph showing film thickness vs. mass loss](image)

Figure 3.11.: Influence of film thickness on mass loss.

It would be expected that the Figure 3.11 shows a linear behaviour, where in the test with a smallest film thickness had the highest mass loss and the test with highest film thickness had the smallest mass loss. Fig.3.11 displays a peak in test T03 that is hard to explain.
As it is possible to see the Table 3.8 the T03 test was repeated. Despite the fact that mass loss is smaller, for T03 than for T03b (repeat), T03b mass loss is still higher than the T01 test.

Table 3.9 display the ratio between Λ and Λ/Λ₅%. To calculate Λ₅% was through the Figure 2.18 which can be found in Chapter 2. Λ₅% is critical film thickness for a damage probability of 5% and Λ/Λ₅% is the influence of Λ in each test.

Table 3.9.: Ratio between Λ and Λ/Λ₅%.

<table>
<thead>
<tr>
<th>Test</th>
<th>Λ</th>
<th>Λ₅%</th>
<th>Λ/Λ₅%</th>
</tr>
</thead>
<tbody>
<tr>
<td>T01</td>
<td>0.12</td>
<td>0.70</td>
<td>0.17</td>
</tr>
<tr>
<td>T03</td>
<td>0.36</td>
<td>1.10</td>
<td>0.33</td>
</tr>
<tr>
<td>T02</td>
<td>0.65</td>
<td>1.50</td>
<td>0.43</td>
</tr>
</tbody>
</table>

Table 3.9 display that the increase in speed causes the increase of the Λ influence. With analysis of the Table 3.9 and the peak of the Figure 3.11 this may be associated with a change in wear regime.

### 3.7.1.3. Influence of lubricant

Tests T01 and T04 were performed with the same operating conditions but with different oils: a PAO and a mineral oil respectively. Table 3.8 show a considerable difference in mass loss that can be attributed to this change. The higher mass loss is coherent with what was expected, as even though the oils had the same viscosity, the mineral oil presents inferior wear qualities when compared to synthetic oils and thus it leads to a higher wear in tests.

### 3.7.2. Direct reading ferrography results

Before making the analysis of the results, it is noteworthy that the oil sample recovery is a very complex process as the place where the oil sample is recovered is not always exactly the same.

Table 3.10 displays the $D_l$ and $D_s$ experimental results where the ID sample is the test performed.

Table 3.10.: Experimental results of direct reading ferrography.

<table>
<thead>
<tr>
<th>ID sample</th>
<th>Dilution</th>
<th>$D_l$</th>
<th>$D_s$</th>
</tr>
</thead>
<tbody>
<tr>
<td>T01</td>
<td>0.1</td>
<td>17.2</td>
<td>3.1</td>
</tr>
<tr>
<td>T02</td>
<td>0.1</td>
<td>50.2</td>
<td>11.1</td>
</tr>
<tr>
<td>T03</td>
<td>0.1</td>
<td>60.2</td>
<td>13.2</td>
</tr>
<tr>
<td>T03b</td>
<td>0.1</td>
<td>63.7</td>
<td>11.0</td>
</tr>
<tr>
<td>T11</td>
<td>0.1</td>
<td>14.0</td>
<td>2.4</td>
</tr>
<tr>
<td>T12</td>
<td>0.1</td>
<td>5.1</td>
<td>1.1</td>
</tr>
<tr>
<td>T04</td>
<td>0.1</td>
<td>22.4</td>
<td>2.4</td>
</tr>
</tbody>
</table>
3. Gear tests for wear characterization

Using expressions 3.1 and 3.2 to analyse the Table 3.10 it was possible to obtain Table 3.11.

Table 3.11.: Experimental values of ferrography.

<table>
<thead>
<tr>
<th>ID</th>
<th>Sample</th>
<th>Dilution</th>
<th>CPUC</th>
<th>ISUC</th>
</tr>
</thead>
<tbody>
<tr>
<td>T01</td>
<td>0.1</td>
<td>203</td>
<td>28623</td>
<td></td>
</tr>
<tr>
<td>T02</td>
<td>0.1</td>
<td>613</td>
<td>26435</td>
<td></td>
</tr>
<tr>
<td>T03</td>
<td>0.1</td>
<td>734</td>
<td>37982</td>
<td></td>
</tr>
<tr>
<td>T03b</td>
<td>0.1</td>
<td>747</td>
<td>41786</td>
<td></td>
</tr>
<tr>
<td>T11</td>
<td>0.1</td>
<td>164</td>
<td>20176</td>
<td></td>
</tr>
<tr>
<td>T12</td>
<td>0.1</td>
<td>62</td>
<td>2722</td>
<td></td>
</tr>
<tr>
<td>T04</td>
<td>0.1</td>
<td>248</td>
<td>50752</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.10 display the results of direct reading ferrography from all test as the Table 3.11 display the results of ferrography.

3.7.2.1. Influence of load

In the T01, T11 and T12 the same test conditions were maintained except for the load stage, where in T01 the load stage was K9, T11 k7 and T12 k5. As the load stage decreased the wear particle concentration and the severity wear also decrease. Table 3.11 displays wear particle concentration and the severity of wear results from is consistent with the mass loss results.

The Figures 3.12 and 3.13 displays the ferrography results of tests T01 and T02 respectively.
3.7. Test results and discussions

Figure 3.12.: Ferrography of T01 test

(a) Main 1

(b) Main 2

(c) Detail 1

(d) Detail 2
3. Gear tests for wear characterization

Figure 3.13.: Ferrography of T02 test
Making consistent comparison between the test T01 (Figure 3.12) and T02, (Figure 3.13) is impossible to explain why the concentration of T01 (Figure 3.12a) is lower than test T02 (Figure 3.13a). Particles may not reflect the reality of the test T01 because the oil sample was taken twenty minutes after the test finished and the larger particles may have been deposited.

Analysing more carefully the images taken with the microscope:

- 3.12c found contaminant;
- 3.12d typical micropitting particle, it has approximately 30 $\mu m$;
- 3.13c particle that indicates the presence of severe wear due to its size $> 40 \mu m$;
- 3.13d ferrous particle oxide;
- 3.13e this type of image containing particles of this size are typical of corrosion.

### 3.7.2.2. Influence of film thickness

It was necessary to restart tests T03, T02 and T03, for a period of fifteen minutes without a load because they had finished in the middle of the night. This was carried out because the particles deposited at the bottom and for the oil recovery to be similar to the other tests which were preform the particles should be suspension in the oil.

In the T01 test the CPUC and ISUC values may not reflect the reality of the test because, as stated earlier, the oil sample was taken twenty minutes after the test is finished and the larger particles may have been deposited at the bottom. Table 3.11 shows results which are consistent with the tendency found in the mass loss of Table 3.8: greater wear for the intermediate film thickness of T03 compared to the extremes T01 and T02.

Table 3.10 and Table 3.11 shows the same inconsistency observed in mass loss. The Figure 3.14 displays the ferrography results of tests T11.
3. Gear tests for wear characterization

Figure 3.14.: Ferrography of T11 test.
3.7. Test results and discussions

Comparison Test T11 (Figure 3.14) to the central test T01 (Figure 3.12) it is possible to verify there is significantly less wear in the T11 test, because the concentration of particles is smaller. Also the size of the particles is smaller in the T11 test than in the T01 test as it is possible to see in Figures 3.14a and 3.12d. That is another proof that supports the conclusion found.

Analysing more carefully the images taken with the microscope:

- 3.14c fatigue particle with combined wear (bearing and sliding);
- 3.14d the detail highlighted in the image indicates that fatigue occurs, which can be of bearings or gears;
- 3.14e typical cut particle.

3.7.2.3. Influence of lubricant

The difference between the T01 test and the T04 test was the lubricant used. The results of Tables 3.10 and 3.11 is consistent with the mass loss results that confirm that T04 is a more severe test then T01.

The following Figures displays the ferrography results of tests T04.

![Ferrography of T04 test](image)

(a) Main 1  (b) Main 2

Figure 3.15.: Ferrography of T04 test

A comparison of Figures 3.15 and 3.12 reinforces the conclusion achieved in the Tables analysis (3.10, 3.11, 3.8).

3.7.3. Individual roughness profiles

In the analysis that was carried out, the pinion and the wheel were analysed separately as the number of cycles of the pinion is higher to that of the wheel in 1,400,000 cycles. As the direction of the sliding speed is different on the driven and on the driving gear, the resulting damage was different. The analysis of the results will be made through the following division: load, film thickness and lubricant used.

Color code used in roughness profiles (Figures 3.16 to 3.36) always follow the same pattern.
3. Gear tests for wear characterization

- black profiles - new roughness profiles (unused);
- blue profiles - used roughness profiles (after test);
- black circle - identifies the mark that was handmade in the tooth;
- vertical red lines - C: pitch line; between B and D, engagement of a single pair of teeth left off; B: initial engagement of two pair of teeth D: initial engagement of two pairs of teeth.

Before analysing in detail the roughness profiles in a general sense it is possible to verify that there is a strong similarity between the roughness of new and used surfaces. This similarity validates the comparison and analysis as it allows the comparison and analysis of the same surfaces before and after they are tested. In certain cases the similarity may not be as apparent as we may be analysing a test in which wear was so severe that the surface was altered significantly. The use of a second filter (section 3.6.3) removed certain roughness values from the topography identified in Table 3.6, and thus helped to diminish the similarity. As this is a manual process with a certain amount of repetition, the chance of a possible error is inherent.

Through the general analysis of all the roughness profiles it was noticeable that there was a relative smoothing of roughness. When a comparison between the old and new surfaces was carried out, it was visible that the asperities had been removed, which is consistent with the mass loss results.

Figures 3.16 to 3.36 are related to roughness profile that have been extracted from topographies. The tooth number of roughness profile is shown in the legend of the roughness profile. Roughness profiles should be read considering that the zero [mm] refers to the root of the tooth and the seven [mm] refers to the tooth head. As stated above, all the roughness profiles were subjected to Gaussian filter having a cut-off 0.8 [mm].

3.7.3.1. Influence of load

Pinion profiles  Figures 3.16 to 3.18 refer the influence of the load variation on each pinion test.

Figure 3.16.: Roughness profile of the tooth 1 from the pinion of T01 test
3.7. Test results and discussions

Figure 3.17.: Roughness profile of the tooth 1 from the pinion of T11 test

Figure 3.18.: Roughness profile of the tooth 1 from the pinion of T12 test

Table 3.12 displays the roughness parameters of each roughness profile.

<table>
<thead>
<tr>
<th></th>
<th>T01(K9)</th>
<th>T11 (K7)</th>
<th>T12 (K9)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>New</td>
<td>Used</td>
<td>New</td>
</tr>
<tr>
<td>$R_{\text{amax}}$</td>
<td>2.954</td>
<td>4.096</td>
<td>2.350</td>
</tr>
<tr>
<td>$R_{Z \text{Din}}$</td>
<td>2.389</td>
<td>2.862</td>
<td>2.070</td>
</tr>
<tr>
<td>$R_a$</td>
<td>0.355</td>
<td>0.325</td>
<td>0.316</td>
</tr>
<tr>
<td>$R_q$</td>
<td>0.472</td>
<td>0.461</td>
<td>0.424</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.484</td>
<td>0.259</td>
<td>0.485</td>
</tr>
<tr>
<td>$R_k$</td>
<td>1.104</td>
<td>1.050</td>
<td>1.063</td>
</tr>
<tr>
<td>$R_{vk}$</td>
<td>0.754</td>
<td>0.994</td>
<td>0.686</td>
</tr>
</tbody>
</table>

Two facts became apparent when analysing the profiles in Figures 3.16 to 3.18: in the test with the lighter load (T12) the similarities between the new and used surfaces are more visible than the test with higher load (T01). The test with the heavier load (T01) presents the higher percentage of micropitting in the areas where the sliding velocity is greater at the root the tooth. It is fairly easy to understand that in the area of the pitch circle, where the sliding speed is very small, the similarities are evident. The pitch circle is marked with the C letter in the Figures. When analysing the oil used in test T01, certain particles of micropitting were found, resembling of Figures 3.12c in detail, which was expected as the presence of micropitting was evident on the pinion surface. Figure 3.16 shows more micropits in the left area of the pitch circle.

**Wheel profiles** Figures 3.19 to 3.21 refer the influence of the load variation on each wheel test.
3. Gear tests for wear characterization

Figure 3.19.: Roughness profile of the tooth 9 from the wheel of T01 test

Figure 3.20.: Roughness profile of the tooth 9 from the wheel of T11 test

Figure 3.21.: Roughness profile of the tooth 9 from the wheel of T12 test

Table 3.13 displays the roughness parameters of each roughness profile.
3.7. Test results and discussions

Table 3.13.: Comparison of wheel profiles parameters [µm].

<table>
<thead>
<tr>
<th></th>
<th>T01(K9)</th>
<th>T11(K7)</th>
<th>T12(K7)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>New</td>
<td>Used</td>
<td>New</td>
</tr>
<tr>
<td>$R_{amax}$</td>
<td>3.907</td>
<td>3.260</td>
<td>3.903</td>
</tr>
<tr>
<td>$R_{ZDin}$</td>
<td>3.275</td>
<td>2.885</td>
<td>3.205</td>
</tr>
<tr>
<td>$R_a$</td>
<td>0.431</td>
<td>0.352</td>
<td>0.447</td>
</tr>
<tr>
<td>$R_q$</td>
<td>0.561</td>
<td>0.485</td>
<td>0.573</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.646</td>
<td>0.409</td>
<td>0.616</td>
</tr>
<tr>
<td>$R_k$</td>
<td>1.355</td>
<td>1.059</td>
<td>1.417</td>
</tr>
<tr>
<td>$R_{vk}$</td>
<td>0.675</td>
<td>1.223</td>
<td>0.638</td>
</tr>
</tbody>
</table>

When analysing the surfaces of the wheel, it was possible to verify that the wheel surfaces presented a greater wear in the area where the sliding speed was higher and this wear is visible in the form of valleys.

**Tooth surface photographs** The Figures 3.22 to 3.23 correspond to the tooth flank profile of each test.

![Figure 3.22: Figure of roughness profile from T01 test.](image)

(a) Pinion

(b) Wheel

Figure 3.22.: Figure of roughness profile from T01 test.

![Figure 3.23: Figure of roughness profile from T11 test.](image)

(a) Pinion

(b) Wheel

Figure 3.23.: Figure of roughness profile from T11 test.
3. Gear tests for wear characterization

![Image of gear test](image1)

(a) Pinion  
(b) Wheel

Figure 3.24.: Figure of roughness profile from T12 test.

Figure 3.22 is related to the test T01, has a Figure 3.23 is related to the test T11 and Figure 3.24 is related to the test T12. The images illustrate very well the influence of load, because the T01 test, which had a higher load, shows more damage. Wheel of Figure 3.22 may display three body abrasion wear in the tooth’s head.

By analysing the profiles (Figures 3.16 to 3.21) and values of the roughness parameters in the Table 3.13 and Table 3.12 the $R_a$ and $R_{pk}$ values has a small decrease which may indicate that the surface has not suffered very severe damage.

3.7.3.2. Influence of film thickness

**Pinion profiles**  Figures 3.25 to 3.26 refer to the influence of the $\Lambda$ variation on each pinion test.

![Roughness profile of the tooth 1 from the pinion of T01 test](image2)

Figure 3.25.: Roughness profile of the tooth 1 from the pinion of T01 test

![Roughness profile of the tooth 1 from the pinion of T03 test](image3)

Figure 3.26.: Roughness profile of the tooth 1 from the pinion of T03 test

Table 3.13 displays the roughness parameters of each roughness profile.
3.7. Test results and discussions

The analysis of the Figures 3.26 and 3.27 is not an easy task. On the tooth root the surface may appear to have less roughness peaks than in the pitch circle area. This fact is related to the sliding speed, as already stated, it is higher in the root. In the pitch circle, as we have just rolling speed, the “sweeping” effect that exists in the root of the tooth does not exist and therefore there are more roughness peaks conserved.

The roughness parameters of the Table 3.15 show the same behaviour of the parameters of Table 3.14. Despite the high mass loss (Table 3.8) the test was not very severe because the $R_a$ values decreases.

### Table 3.14.: Comparison of pinion profiles parameters [$\mu$m].

<table>
<thead>
<tr>
<th></th>
<th>T01 ($\Lambda=0.12$)</th>
<th>T03 ($\Lambda=0.36$)</th>
<th>T02 ($\Lambda=0.65$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>New</td>
<td>Used</td>
<td>New</td>
</tr>
<tr>
<td>$R_{\text{amax}}$</td>
<td>2.954</td>
<td>4.096</td>
<td>4.250</td>
</tr>
<tr>
<td>$R_{Z\text{Din}}$</td>
<td>2.389</td>
<td>2.862</td>
<td>3.561</td>
</tr>
<tr>
<td>$R_a$</td>
<td>0.355</td>
<td>0.325</td>
<td>0.570</td>
</tr>
<tr>
<td>$R_q$</td>
<td>0.472</td>
<td>0.461</td>
<td>0.725</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.484</td>
<td>0.259</td>
<td>0.658</td>
</tr>
<tr>
<td>$R_k$</td>
<td>1.104</td>
<td>1.050</td>
<td>1.778</td>
</tr>
<tr>
<td>$R_{vk}$</td>
<td>0.754</td>
<td>0.994</td>
<td>0.921</td>
</tr>
</tbody>
</table>

Wheel profiles  Figures 3.28 to 3.29 refer to the influence of the load variation on each wheel test.

![Figure 3.28.](image)

Figure 3.28.: Roughness profile of the tooth 9 from the wheel of T01 test
Figure 3.29.: Roughness profile of the tooth 17 from the wheel of T03 test

Figure 3.30.: Roughness profile of the tooth 17 from the wheel of T02 test

Table 3.15 displays the roughness parameters of each roughness profile.

<table>
<thead>
<tr>
<th></th>
<th>T01 (Λ=0.12)</th>
<th>T03 (Λ=0.36)</th>
<th>T02 (Λ=0.65)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>New</td>
<td>Used</td>
<td>New</td>
</tr>
<tr>
<td>( R_{\text{amax}} )</td>
<td>3.907</td>
<td>3.260</td>
<td>5.780</td>
</tr>
<tr>
<td>( R_{ZDin} )</td>
<td>3.275</td>
<td>2.885</td>
<td>4.752</td>
</tr>
<tr>
<td>( R_a )</td>
<td>0.431</td>
<td>0.352</td>
<td>0.543</td>
</tr>
<tr>
<td>( R_q )</td>
<td>0.561</td>
<td>0.485</td>
<td>0.711</td>
</tr>
<tr>
<td>( R_{pk} )</td>
<td>0.646</td>
<td>0.409</td>
<td>0.672</td>
</tr>
<tr>
<td>( R_k )</td>
<td>1.355</td>
<td>1.059</td>
<td>1.722</td>
</tr>
<tr>
<td>( R_{vk} )</td>
<td>0.675</td>
<td>1.223</td>
<td>0.990</td>
</tr>
</tbody>
</table>

What was explained for pinions can also be seen on the wheels, so that in wheel the effect of the sliding speed is noticed.

**Tooth surface photographs**  The Figures 3.31 and 3.32 correspond to the tooth flank profile of each test.
3.7. Test results and discussions

3.7.3.3. Influence of lubricant

Pinion profiles  Figures 3.33 and 3.34 refer the influence of the lubricant variation on each pinion test.

Figure 3.33.: Roughness profile of the tooth 1 from the pinion of T01 test
Figure 3.34.: Roughness profile of the tooth 9 from the pinion of T04 test

Table 3.16 displays the roughness parameters of profile.

Table 3.16.: Comparison of pinion profiles parameters µm.

<table>
<thead>
<tr>
<th></th>
<th>T01 (PAO)</th>
<th>T04 (MIN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>New</td>
<td>Used</td>
<td>New</td>
</tr>
<tr>
<td>$R_{\text{amax}}$</td>
<td>2.954</td>
<td>4.096</td>
</tr>
<tr>
<td>$R_{ZDin}$</td>
<td>2.389</td>
<td>2.862</td>
</tr>
<tr>
<td>$R_a$</td>
<td>0.355</td>
<td>0.325</td>
</tr>
<tr>
<td>$R_q$</td>
<td>0.472</td>
<td>0.461</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.484</td>
<td>0.259</td>
</tr>
<tr>
<td>$R_k$</td>
<td>1.104</td>
<td>1.050</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.754</td>
<td>0.994</td>
</tr>
</tbody>
</table>

Before analysing and interpreting Tables 3.16 and 3.17 which are relevant to the pinion and the wheel, it is important to mention that the initial roughness was high and it decreased during the running-in period. Once the running-in was finished the roughness values increased once again due to the damage which began to appear on the surface.

The possible reason for the appearance of peaks in Figure 3.34 between B and C may be found in the $R_a$ and $R_{pk}$ parameters from the Table 3.16. The value of the $R_a$ parameter at the T01 test decreases from new to used, while in the T04 test $R_a$ increases from new to used. The value of the $R_{pk}$ parameter decreases in both test but the T01 test the reduction was higher because the damage which began to appear on the surface.

**Wheel profiles** Figures 3.35 to 3.36 refer the influence of the load variation on each wheel test.

Figure 3.35.: Roughness profile of the tooth 9 from the wheel of T01 test
3.7. Test results and discussions

Figure 3.36.: Roughness profile of the tooth 1 from the wheel of T04 test

Table 3.17 displays the roughness parameters of profile.
Table 3.17.: Comparison of wheel profiles parameters $\mu m$.

<table>
<thead>
<tr>
<th></th>
<th>T01 (PAO)</th>
<th>T04 (MIN)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>New</td>
<td>Used</td>
</tr>
<tr>
<td>$R_{\text{amax}}$</td>
<td>3.907</td>
<td>3.260</td>
</tr>
<tr>
<td>$R_{ZDin}$</td>
<td>3.275</td>
<td>2.885</td>
</tr>
<tr>
<td>$R_a$</td>
<td>0.431</td>
<td>0.352</td>
</tr>
<tr>
<td>$R_q$</td>
<td>0.561</td>
<td>0.485</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.646</td>
<td>0.409</td>
</tr>
<tr>
<td>$R_k$</td>
<td>1.355</td>
<td>1.059</td>
</tr>
<tr>
<td>$R_{vk}$</td>
<td>0.675</td>
<td>1.223</td>
</tr>
</tbody>
</table>

If there were any doubts in relation to the severity of the test, these disappeared with the visual comparison between Figure 3.36 and Figure 3.35. The clear presence of more valleys when compared to test T01 and the fact that even the handmade mark disappeared show the severity of the test. $R_a$ $R_{pk}$ in Table 3.17 vary in the same manner that the pinion homologous parameters.

When test end and the $R_a$ parameter is lower than the initial one may conclude that the test was not severe and the probability of failure is diminished. In the situation found in the T04 test in which the final $R_a$ is greater than the initial $R_a$.

**Tooth surface photographs** The Figure 3.37 correspond to the tooth flank profile of T04 test. Images display has a direct correlation with the roughness profile, are for the same tooth.

(a) Pinion  
(b) Wheel  

Figure 3.37.: Figure of roughness profile from T04 test.

Figure 3.37 is related to the test T04. Comparing this Figure with Figure 3.22 it is notorious the Figure 3.37 has a higher presence of micropitting. Wheel of Figure 3.37 may display three-body abrasion wear in the tooth’s head.

### 3.7.4. Averaged roughness parameters

In this section the Figures 3.38 to 3.49 was done with the average values of measurements performed. The average values of the wheel and the pinion comprises nine profiles measurements (3 measurements on each tooth). In the analysis that was carried out, the pinion and the wheel were analysed separately as done in 3.7.3
3.7. Test results and discussions

3.7.4.1. Influence of load

Figures 3.38 to 3.40 show how the $R_{pk}$, $R_{vk}$ and $R_a$ of the pinion varies with the contact force. T12 test has lower contact force while the T01 test has the highest contact force. Parameters values presented in Figures are the averaged of each test and are in [$\mu m$].

![Figure 3.38.: Influence of contact force on the pinion average $R_{pk}$ parameter.](image)

![Figure 3.39.: Influence of contact force on pinion average $R_{vk}$ parameter.](image)
The analysis of the Figures 3.38 and 3.40 shows that the final roughness of $R_a$ parameter is very similar. The T12 test, despite having low contact force, showed the greatest decrease in the $R_{pk}$ parameter. This fact is justified because the pinion had a higher initial roughness than others. Figure 3.39 shows that T12 test suffered the greatest increase in $R_{vk}$ parameter.

Figures 3.41 to 3.43 show how the $R_{pk}$, $R_{vk}$ and $R_a$ of the wheel varies with the contact force. T12 test has lower contact force while the T01 test has the highest contact force.
3.7. Test results and discussions

With regards to the wheel, Figures 3.41, 3.42 and 3.43 display an increase, in other words, the used values increased according to the different type of test, from the lighter load to the heavier one. The increase in the $R_{pk}$ and $R_{vk}$ parameters is justifiable with the higher contact force.

3.7.4.2. Influence of film thickness

Figures 3.44 to 3.46 show how the $R_{pk}$, $R_{vk}$ and $R_{a}$ of the pinion varies with the film thickness. T01 test has lower film thickness while the T02 test has the highest film thickness.
3. Gear tests for wear characterization

![Graph showing influence of film thickness on pinion average Rpk parameter.](image)

**Figure 3.44.**: Influence of film thickness on pinion average $R_{pk}$ parameter.

![Graph showing influence of film thickness on pinion average $R_{vk}$ parameter.](image)

**Figure 3.45.**: Influence of film thickness on pinion average $R_{vk}$ parameter.
Figure 3.46.: Influence of film thickness on pinion average $R_a$ parameter

Making a comparison of tests through the values is not an easy task. Fig 3.44 and 3.45 displays used parameters that change in a non-linear fashion with increasing film thickness. Figure 3.44 displays the same shape of the mass loss Figure 3.8. It is not easy to explain why $R_{pk}$ value is so high in T03 test or is so low in T02 test. T02 have high new $R_{pk}$ value.

Figure 3.45 show that with the increasing of the film thickness, $R_{vk}$ used decrease. T03 test have the same value of $R_{vk}$ for new and used.

Fig. 3.46 shows that the used $R_a$ value is lower with higher film thickness.

Figures 3.47 to 3.49 show how the $R_{pk}$, $R_{vk}$ and $R_a$ of the wheel varies with the film thickness. T01 test has lower film thickness while the T02 test has the highest film thickness.

Figure 3.47.: Wheel average $R_{pk}$ parameter vs film thickness
3. Gear tests for wear characterization

The way the line of new and used parameters of the Figures 3.47 and 3.49 vary are similar. $R_{pk}$ and $R_a$ parameters do not considerably change with the variation of film thickness.

### 3.7.4.3. Influence of lubricant

Tables 3.18 and 3.19 show how the $R_{pk}$, $R_{vk}$ and $R_a$ of the pinion and wheel varies with the lubricant used. T01 used a PAO and T04 used a Mineral oil.
3.7. Test results and discussions

Table 3.18.: Averaged pinion test parameters.

<table>
<thead>
<tr>
<th></th>
<th>T01</th>
<th>T04</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>New</td>
<td>Used</td>
</tr>
<tr>
<td>$R_a$</td>
<td>0.364</td>
<td>0.317</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.497</td>
<td>0.263</td>
</tr>
<tr>
<td>$R_{vk}$</td>
<td>0.703</td>
<td>0.892</td>
</tr>
</tbody>
</table>

Table 3.19.: Averaged wheel test parameters.

<table>
<thead>
<tr>
<th></th>
<th>T01</th>
<th>T04</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>New</td>
<td>Used</td>
</tr>
<tr>
<td>$R_a$</td>
<td>0.410</td>
<td>0.336</td>
</tr>
<tr>
<td>$R_{pk}$</td>
<td>0.563</td>
<td>0.384</td>
</tr>
<tr>
<td>$R_{vk}$</td>
<td>0.685</td>
<td>1.109</td>
</tr>
</tbody>
</table>

The analysis of the Tables 3.18 and 3.19 is the same as the analysis of the individual roughness profile. The parameters of both Tables relating to T04 test increased due to test severity.

3.7.5. Power loss

For the analysis of the total power loss was used the same sequence as in the section 3.7.4.

Table 3.20.: Power loss in the tests

<table>
<thead>
<tr>
<th>Test</th>
<th>Power Loss [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>T01</td>
<td>409.4</td>
</tr>
<tr>
<td>T02</td>
<td>2717.7</td>
</tr>
<tr>
<td>T03</td>
<td>1416.9</td>
</tr>
<tr>
<td>T11</td>
<td>264.7</td>
</tr>
<tr>
<td>T12</td>
<td>162.6</td>
</tr>
<tr>
<td>T04</td>
<td>444.2</td>
</tr>
</tbody>
</table>

Table 3.20, displays the power loss of all tests.

3.7.5.1. Influence of load

Figure 3.50 display the power loss behaviour with the load variation.
3. Gear tests for wear characterization

Figure 3.50.: Influence of load variation on the power losses.

Figure 3.50 display the power loss behaviour with the load variation. T12 test have the lowest contact force while T01 test have the highest contact force.

As would expect the increased load causes an increase in power loss as is shown in the Figure 3.50. The Figure also shows that the power loss does not vary linearly with load increasing and this is consistent with the mass loss results.

3.7.5.2. Influence of film thickness

Figure 3.51.: Influence of film thickness variation on the power losses.

Figure 3.51 display the power loss behaviour with the film thickness variation. T01 test have the lowest Λ while T02 test have the highest Λ. The y-axis scale in Figure 3.51 is not the same as the other Figure presented, because the power loss values are quite different.

The Figure 3.51 shows the increase of the power loss with the increase of Λ. This is justify by the increasing in churning losses and the decrease in temperature which
causes an increase in the oil viscosity.

3.7.5.3. Influence of lubricant

Table 3.20 display that the mineral oil (T04 test) causes higher power loss than the PAO oil (T01 test) and it is consistent with the mass loss results.

3.8. Summary

The experimental results show in this Chapter allowed to draw some conclusions as to the influence of load, Λ and lubricant.

- influence of load - wear increases with load non-linearly, as shown in Figure 3.8;
- influence of Λ - wear does not follow a monotonous trend with regard to Λ. It was seen that for all indicators (mass loss, ferrometry, roughness), the tests with extreme Λ (0.1 and 0.65) showed much less wear than a test with intermediate Λ (0.36). This is surprising and may wear indicated a switch in wear mechanism due to temperature. As the oil is strongly additivated and the value of the oil temperature varied that causes a change in the behaviour of additives which may explain the mass loss behaviour;
- influence of lubricant - wear increases with replacing PAO for Mineral oil, as shown in Table 3.8.
4. Simulation of power loss

Prediction of power loss in a gearbox has always been a hard task. Power losses in a gearbox come from gearing, bearing, seals and others (Figure 4.1). Gearing and bearing losses can be separated in no-load losses, which occur without power transmission, and load dependent losses, in the contact of the power transmitting components. No-load losses are mainly related to lubricant viscosity and density as well as immersion depth of the components. Load losses depend on transmitted load, coefficient of friction and sliding velocity in the contact areas of the components [17] [22].

![Power loss contributions](image)

Figure 4.1.: Power loss contributions [23].

Under the nominal torques and rotational speeds usually found in an automotive or industrial gearbox, the load dependent gear losses are the main source of energy dissipation followed by the rolling bearing losses. The load independent losses have usually less influence, but may have a larger influence when the speed is very high and dip lubrication is used [17].

Due to infinite combinations between rolling bearings, gears seals, casing geometries, oil formulations and other variables influencing the power loss, it is very complicated to build a model. To understand and simulate power loss, the model developed by Fernandes et al. [24] was used.

Of particular interest to the present work is the load losses in the gears, which can help illuminate the process of mild wear.
4. Simulation of power loss

4.1. Power Loss Model

The model used, have consider the following sources of the power loss:

- load independent gear losses or spin power losses;
- load dependent gear losses;
- rolling bearing losses;
- shaft seal losses.

4.1.1. Load independent gear losses or spin power losses

No-load gear’s power losses (or spin power losses) usually are a very important source for energy dissipation. As stated above it depends on speed, lubricant characteristics and gearbox design. These are directly related to the type of lubrication method used. Dip lubrication is often used in low to medium speed automotive gearbox or industrial transmission. When operating gear speeds are relatively high, jet lubrication is preferred.

In both types of lubrication methods (dip or oil jet), the spin losses are related to complex hydrodynamic phenomena which are very difficult to describe in analytical formulations.

The no-load losses were determined for each input speed using the torque loss measured on load stage k1. The no-load losses remain similar for higher load stages.

The no load-losses are calculated subtracting the gear mesh losses, the rolling bearing losses and the seal losses to the total torque loss on load stage K1 as represented in eq. 4.1:

\[ P_{VZ0} = P_{k1}^L - P_{VZP}^{k1} - P_{VL}^{k1} - P_{VD}^{k1} \]

Note that \( P_{VZP}^{k1} \) is very close to zero, so this term was disregarded.

4.1.2. Load dependent gear losses

The main source of power loss in gear transmission is the contact between meshing teeth. As stated before, the sliding speed of the surfaces along the path of contact varies, influencing the coefficient of friction.

According to Höhn [25], the coefficient of friction is considered constant along the path of contact because in the pitch point it becomes close to zero and the variation of the coefficient of friction, away from the pitch point, is very small.

Power losses due to meshing gears are calculated with eq. 4.2

\[ P_{VZP} = P_A \cdot \mu_{mz} \cdot H_V \]

Where:

- \( P_A \) - transmitted power;
4.1. Power Loss Model

- \( H_V \) - represents the gear loss factor which is determined according to the below equation;

- \( \mu_{mz} \) - coefficient of friction on meshing gears.

The transmitted power is calculated using equation 4.3:

\[
P_A = F_{bt} \cdot r_b \cdot \omega \tag{4.3}
\]

\[
H_v = \frac{\pi \cdot (i + 1)}{z_1 \cdot i \cdot \cos(\beta_b)} \cdot (1 - \varepsilon_0 + \varepsilon_1^2 + \varepsilon_2^2) \tag{4.4}
\]

Eq. 4.4 assumes that the coefficient of friction (\( \mu_{mz} \)) is constant along the path of contact and that the tangential force (\( F_{bt} \)) is almost constant for the geometry used but in fact this is a simplification of the problem.

4.1.3. Coefficient of friction on meshing gears

Schlenk [25] proposed equation 4.5 for the average coefficient of friction along the path of contact. The equation was derived for a mineral oil without additives considering the lubricant parameter (\( X_L \)) equal to 1. For the mineral and PAO fully formulate oils tested, \( X_L \) is 0.85 and 0.7, respectively by [24].

\[
\mu_{mz} = 0.048 \left( \frac{F_{bt}}{u \Sigma C \cdot \rho C} \right)^{0.2} \cdot \eta_{oil}^{-0.05} \cdot R_a^{0.25} \cdot X_L \tag{4.5}
\]

4.1.4. Rolling bearing losses

All machines have rolling bearings, and like all components that have motion, rolling bearings have losses. Several models are currently used to predict the power loss of rolling bearings. Some models are validated by large number of experimental results as the ones presented by the major rolling bearing manufactures in the word, such as SKF and FAG.

Rolling friction losses in a rolling bearing are identified in literature by the following effects: deformations and elastic hysteresis. Sliding is the major source of friction in rolling bearing, mainly at low speed. The sliding friction occurs due to micro slip and sliding due to rolling motion. Drag friction due to the oil who are not used to cool and lubricate the contact promote contrary motion of the rolling elements.

Rolling bearing power loss was determined using the new SKF friction torque model.

Equation 4.6 was used to calculate the power loss in a bearing, where \( M \) is the frictional moment (N.mm) and \( n \) is the rotational speed of the shaft (rpm).

\[
P_{VL} = M \cdot n \cdot \frac{\pi}{30} \cdot 10^{-3} \tag{4.6}
\]

\[
M = M_{rr} + M_{sl} + M_{seal} + M_{drag} \tag{4.7}
\]

Where:
4. Simulation of power loss

- $M$ is the total frictional moment;
- $M_{rr}$ is the rolling frictional moment;

\[
M_{rr} = \Phi_{ish} + \Phi_{rs} + G_{rr}(\nu \cdot n)^{0.6} \tag{4.8}
\]

Where:

- $M_{rr}$ is the rolling frictional moment;
- $\Phi_{ish}$ is the inlet shear heating reduction factor;
- $\Phi_{rs}$ is the kinematic replenishment/starvation reduction factor;
- $G_{rr}$ is a variable depending on the bearing type, bearing mean diameter ($d_m$ [mm]), the radial force ($F_r$ [N]) and axial load ($F_a$ [N]);
- $n$ is the rotational speed [rpm];
- $\nu$ is the kinematic viscosity at operating temperature of the oil [mm$^2$/s].

- $M_{sl}$ is the sliding frictional moment;

\[
M_{sl} = G_{sl}\mu_{sl} \tag{4.9}
\]

Where:

- $M_{sl}$ is the sliding frictional moment [N.mm];
- $G_{sl}$ is the a variable depending on the bearing type, the mean bearing diameter [mm], the radial load [N] and the axial load [N];
- $\mu_{sl}$ is the sliding frictional coefficient.

\[
\mu_{sl} = \mu_{bl} \cdot \Phi_{bl} + (1 - \Phi_{bl}) \cdot \mu_{EHD} \tag{4.10}
\]

Table 4.1.: Reference values of coefficient of friction for an operating temperature of 80 °C

<table>
<thead>
<tr>
<th></th>
<th>$\mu_{bl}$</th>
<th>$\mu_{EHD}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mineral</td>
<td>0.0339</td>
<td>0.056</td>
</tr>
<tr>
<td>PAO</td>
<td>0.0332</td>
<td>0.037</td>
</tr>
</tbody>
</table>

- $M_{seal}$ is the frictional moment of seals;

There is no seals

- $M_{drag}$ is the frictional moment of drag losses, churning, splashing, etc.
4.1.5. Shaft seal losses

Seal losses are not very well understood yet. Seals power loss is due to friction in the contact zone. The contact zone is very small and the microscopic phenomena is difficult to parametrize.

The friction torque resulting from radial loads only a part of the total friction torque loss at the sealing position. Other contributing factor include the type of medium being sealed, the pressure differential across the seal, the circumferential speed, the ambient temperature, the lubricant and lubrication method and also the condition of the counterface.

Seal losses was determined using the Freudenberg equation 4.11.

\[ P_{VD} = 7.69 \cdot 10^{-6} d_{sh}^2 \cdot n \]  

(4.11)

Where:

- \( d_{sh} \) is the mean diameter of the seal [mm];
- \( n \) is the rotational speed of the shaft [rpm];

Gearbox power loss model is established according to eq. 4.12 [17].

\[ P_V^i = \underbrace{P_{VZ0}}_{P_{Vexp} \cdot P_{VL} \cdot P_{VD}} + \underbrace{P_{VZP}}_{P_{IN} \cdot H_V \cdot \mu_\text{mix}} + \underbrace{P_{VL}}_{\text{New SKF Model}} + \underbrace{P_{VD}}_{\text{Simrit Equation}} \]  

(4.12)

4.1.6. Needed modifications to the model

In order to tune the model described above for the test performed two simplifications were introduced.

Calculation of \( P_{VZ0} \), load independent meshing gears power loss, is done by the sum of the \( P_{VZ0} \) slave gearbox with \( P_{VZ0} \) the test gearbox. Since the operating temperature and viscosity change when the applied torque increases, the no-load losses will change accordingly on test gearbox. No-load gear losses predictions found in literature are, in general, related to the lubricant Reynolds number. In this way the no-load losses for each rotational speed of the test gearbox can be estimated taking the measurement at load stage K1 as reference.

In the slave gearbox the value of \( P_{VZ0} \) was obtained by the equation 4.13.

\[ P_{VZ0}^{K1} = 0.1105 \cdot \omega^{1.2871} \]  

(4.13)

To calculate the \( P_{VZ0}^1 \) depends on the oil and temperature, which was used.

For PAO oil:

\[ P_{VZ0}^{K1} = 0.1155 \cdot \omega^{1.3117} \]  

(4.14)

For mineral oil:

\[ P_{VZ0}^{K1} = 0.1 \cdot \omega^{1.3681} \]  

(4.15)
4. Simulation of power loss

To calculate the Reynolds’s number, $Re^{i-0.21}$ and $Re^{1-0.21}$, the following equation 4.16 was used:

$$Re = \frac{\omega \cdot d^2}{v}$$

(4.16)

The difference between the calculation of $Re^{i-0.21}$ and $Re^{1-0.21}$ is the oil viscosity $v$. The eq. 4.14 and 4.15 were derived for a constant temperature or constant viscosity. In this way we need to adjust the equation for different viscosities according to eq. 4.17

$$P_{VZ0}^i = P_{VZ0}^{K1} \cdot \frac{Re^{i-0.21}}{Re^{1-0.21}}$$

(4.17)

To calculated $Re^{1-0.21}$ the $v$ may have two values: for PAO is 60.4 and for the MIN oil is 43.9. In $Re^{i-0.21}$, the $v$ changes with the test conditions.

4.2. Simulation results and discussions

Figures 4.2 to 4.5 were obtained applying the model described above, in the previous section.

The symbols in the legends of the Figures 4.2 to 4.5 means:

- $C^{14}$ - load losses due to meshing gears on the test gearbox;
- $C^{40}$ - load losses due to meshing gears on the slave gearbox;
- $P_{VZ0}$ - no-load losses due to spin power;
- $P_{VL}$ - power losses in rolling bearings;
- $P_{VL}$ - power losses due to shaft seal.
4.2. Simulation results and discussions

Figure 4.2.: Influence of the variation of load on power loss (T=100°).

Figure 4.3.: Influence of variation of Λ on power loss.
4. Simulation of power loss

Figure 4.4.: Influence of the variation of Λ on \((C_{14})/\) total power loss ratio.

Figure 4.5.: Influence of the variation of lubricant on power loss.

Figure 4.2 shows the influence of the load variation on power loss. T01 test has the higher contact force while T12 has the lower contact force.

Figure 4.3 shows the influence film thickness variation on power loss. It compares the partial model results with the power loss measurement. T01 test has the low Λ while T02 test have the higher Λ.

Figure 4.4 display the influence of the variation of Λ on \((C_{14})/\) total power loss ratio.

Figure 4.5 shows the influence of the lubricant variation on power loss. It compares the partial model results with the power loss measurement. T04 test was done with the mineral lubricant.

The y-axis scale in Figure 4.3 is not the same as the other Figures presented, because the power loss values are quite different.
4.3. Summary

Scale used may lead to a bad judgement when analysing the difference between the power loss obtained experimentally to the power loss predicted by the model. To clarify and show the error values for the different tests are shown in Table 4.2.

<table>
<thead>
<tr>
<th>Test</th>
<th>Error[%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>T01</td>
<td>8.9</td>
</tr>
<tr>
<td>T02</td>
<td>14</td>
</tr>
<tr>
<td>T03</td>
<td>5.8</td>
</tr>
<tr>
<td>T04</td>
<td>0.2</td>
</tr>
<tr>
<td>T11</td>
<td>28.7</td>
</tr>
<tr>
<td>T12</td>
<td>5.6</td>
</tr>
</tbody>
</table>

Analysing Table 4.2, the results obtained are good taking into account that no no-load tests were performed, especially for T04. T02, despite having an error of 14 \% it is also good, because the model used was never tested at high speeds. Power loss prediction for the test T11 has an error near to 30 \%, which, when compared to the other values, is very high. This can be due to some inaccuracy of the measurement or model.

Because the errors in Table 4.2 are small, the partial results obtained by the model become valid. In particular, the results for load losses in the test gear ($C_{14}$) can be used to analyse wear in the test gears.

With the information displayed in Figures 4.2 to 4.5, the power loss concerning to the seals only changes with the speed. $P_{VZ0}$ varies with the speed and the lubricant used. Losses in bearings varies with the load and speed, but it is visible that they are more sensible to the speed variation.

Power loss due the to meshing gears in the test gearbox ($C_{14}$) of Figure 4.2 has the same behaviour that seen in mass loss (Table 3.8). This is consistent with the usual observation that increasing force leads to increase in wear.

Two conclusions might be take from Figure 4.3. Power loss due to the meshing gears in the test gearbox ($C_{14}$) increase with the increase of the sliding speed. Load losses in the test gears ($C_{14}$) increase with $\Lambda$, in an approximately linear manner. This is not consistent with the measures indicators of wear that showed greater wear in test T03 than in test T01 and T02. However, an interesting fact is that the proportion of the power loss due to $C_{14}$ load losses in the total power loss diminished gradually with increases of $\Lambda$, as shown in Figure 4.4. This may explain the diminution of mass loss from T03 to T02.

In the test gearbox, power loss of ($C_{14}$) on Figure 4.5 has the same behaviour to that seen in mass loss (Table 3.8). This is consistent with all previous results for the variation of the lubricant.

4.3. Summary

A model for power loss prediction was presented, with a few specific alterations. The model was shown to be accurate in predicting the experimental values. This
accuracy leads to the conviction that the evaluation of the partial contribution of machine elements to the power loss are also accurate. In particular, the load loss in the test gears can be used to contribute to the analysis of wear in these gears.

The modifications that were introduced in the model become quite useful because not only show the good relationship between the model and the experimental values, but also allows the use of partial model values.

The experimental results show in this Chapter allowed to draw some conclusions as to the influence of load, $\Lambda$ and lubricant.

- Influence of load - power loss due the to meshing gears in the test gearbox ($C_{14}$) (Figure 4.2) increases with the load;

- Influence of $\Lambda$ - Contrary to what happened with the mass loss, power loss due the to meshing gears in the test gearbox ($C_{14}$) (Figure 4.3) increases with the increase of the $\Lambda$. The contribution of ($C_{14}$) in the total power loss decreases with increasing speed, as shown in Figure 4.4;

- Influence of lubricant - wear increases with lubricant, as shown in Figure 3.8;

- Power loss - power loss due the to meshing gears in the test gearbox ($C_{14}$) increase with the load and $\Lambda$ increase.

Justify why the mass loss in the T03 test was so high stills difficult to explain.
5. Simulation of wear of the gear tests

5.1. Introduction

To simulate the wear in each gear tooth flank during contact, the model needs to consider the type of lubrication that was used and evaluate the normal pressure and tangential traction distributions within the contact to be able to evaluate the load transited to the tooth gear flank surface.

The numerical model can be divided into two main parts:

- mixed film lubrication;
- wear on spur gear teeth.

In the first part of the model, the geometry of the contact (including the tooth flank roughness) and the contact force are determined. The contact loads between the driving and driven gear teeth are obtained through the application of a sub-model, which deals with mixed film lubrication. This procedure must be repeated for every instant of the loading cycle in order to obtain the stresses at each point.

In the second model the prediction of profile variation on gear tooth flanks caused by mild wear.

5.2. Numerical models

5.2.1. Spur gears kinematics

The aim of this work was to test, simulate and understand how wear occurs in gears. To simulate wear it was used the model developed by Brandão et al. [1].

Before describing the model used to perform the simulation of wear in gears it is important to realize that the analysis of the geometry kinematics and loading on spur gears will be restricted to involute spur gears.

Fig. 5.1 presents the schematic of the spur gears used in the tests and only those teeth which actually participate at an instant when two pairs of teeth are meshing are shown.
5. Simulation of wear of the gear tests

The characteristic dimensions and notable points are:

- $O_1$ and $O_2$ - are the centres of the pinion and of the tooth respectively;
- $\alpha'$ - is the operating pressure angle;
- $a'$ - is the operating centre distance;
- $R_{b1}, R_{p1}, R_{a1}$ - are the radii of the base circle, the operating pitch circle and the addendum circle of the pinion respectively;
- $R_{b2}, R_{p2}, R_{a2}$ - are the radii of the base circle, the operating pitch circle and the addendum circle of the wheel respectively;
- $w_1$ and $w_2$ - are respectively the angular velocity of the pinion and of the wheel;
- $T_1$ and $T_2$ - are the extremities of the contact line of length $T_1 T_2$;
- $P$ and $P'$ - are the contact points of two consecutive pair of teeth and lie on the contact line $T_1 T_2$.

As the gears are tested, gears with a contact ratio $\varepsilon$ such that $1 < \varepsilon < 2$ the gear meshing occurs at times between two pairs of teeth and at other times between one pair.

Figure 5.2 displays notable moments during meshing of a pair of teeth.
5.2. Numerical models

In 5.2a the pair initiates its contacts while another is already in contact on the left-hand side; in 5.2b the pair now bears the contact alone; in 5.2c the pair is in pure rolling; in 5.2d a new pair initiates contact on the right-hand side; and in 5.2e the pair ceases its contact.

The pinion receives a driving torque $T$ and transmits it to the wheel as a normal contact force equal $T/R_{bl}$. This contact force is shared between all pairs of teeth simultaneously in contact.

Determination the distribution of contact load between pairs of teeth is a difficult problem as it may be dependent on: the flexibility of the teeth in bending, the bending and torsion flexibility of the axles which the gears are mounted, the position of the wheels along the axles, the clearances within machine elements and finally on dynamic loads and vibrations. To simplify an empirical formula is used to determine the contact force $F_N$ between a pair of teeth. This assumption only observe the position that the contact point $P$ occupies along the contact line.

Figure 5.3 displays notable moments of the meshing of a pair of teeth. Not only does it display the consecutive position of a pair of contacting teeth are superimposed, but also how the normal load borne by this pair of teeth is shared as a function of the contact position along the contact line.

This is equivalent to the formula 5.1 [1]:

$$F_N = \frac{T}{R_{bl}} \left\{ \begin{array}{c} \frac{1}{3} + \frac{1}{3} \frac{AP}{AB} \quad A < P < B \\ \frac{1}{3} + \frac{1}{3} \frac{BP}{BD} \quad B < P < D \\ \frac{2}{3} + \frac{1}{3} \frac{DP}{DE} \quad D < P < A \end{array} \right. \quad (5.1)$$
5. Simulation of wear of the gear tests

Figure 5.3.: Notable moments of the meshing of a pair of teeth [1].

The velocity of any material point has two components: the normal velocity, component along the contact line and the tangential velocity, component along its perpendicular. The sum of these components equals the velocity at any point. The difference between the two is that tangential velocity of contacting teeth is the sliding velocity and it causes a friction force to appear.

The tangential velocity of a material point of a pinion tooth is given by:

\[ U_1 = \omega_1 T_1 P \]  

(5.2)

The tangential velocity of a material point of a wheel the expression is similarly.

\[ U_2 = \omega_2 T_2 P \]  

(5.3)

Where:

- \( \omega \) - angular speed;
- \( T \) - radius.

The rolling speed velocity \( U \) is defined as the average of the tangential velocities:

\[ U = \frac{U_2 + U_1}{2} \]  

(5.4)

The sliding speed velocity is defined as:

\[ U_2 - U_1 \]  

(5.5)

In the pitch line, point \( C \), only has rolling speed. As Figure 5.4 displays, when the contact is inside the pitch circle, \( U_2 \) is faster than \( U_1 \), the opposite is true when the contact is outside the pitch circle of the pinion.

5.2.2. Mixed film lubrication model

Normal contact pressure  The specific film thickness (\( \Lambda \)) regime mentioned in 2.5.2.3 and 2.5.3 is defined by:

\[ \Lambda = \frac{h_0}{\sigma} \]  

(5.6)
5.2. Numerical models

In the present case \( \sigma \), is defined by equation 2.5 (Chapter 2) as the combined roughness RMS of the pinion and the wheel tooth flank profiles. However, Brandão used an alternative definition for the combined roughness, a pseudo- RMS roughness \( R_{q1} \) of the pinion tooth profile is defined as:

\[
R_{q1}^2 = \frac{1}{x_b - x_a} \int_{x_a}^{x_b} (y - \bar{y})^2 dx
\] (5.7)

The pseudo-RMS roughness has a similar definition as that of the pinion. The “local composite roughness” \( \sigma \) is then computed in the usual manner:

\[
\sigma = \sqrt{(R_{q1})^2 + (R_{q2})^2}
\] (5.8)

The limits of integration are the contact area of each instant.

The shear of the contact load borne by lubricant depends in \( \Lambda \) and receives the designation of load sharing function \( f_\Lambda(\Lambda) \). The relations between full film EHD, boundary lubrication and overall normal contact load can then be expressed as follows:

\[
F_N(t) = F^{EHD}(t) + F^{BDR}(t)
\] (5.9)

\[
F^{EHD}(t) = f_\Lambda(\Lambda) \cdot F_N(t)
\] (5.10)

\[
F^{BDR}(t) = (1 - f_\Lambda(\Lambda)) \cdot F_N(t)
\] (5.11)

Where:

- \( F_N \) - is the normal contact load expressed as force per unit;
- \( F^{EHD} \) - is the portion of the \( F_N \) borne by the lubricating film;
- \( F^{BDR} \) - is the portion borne by direct contact between the surfaces.

According to Brandão et al., the main idea of the model is: that the contact pressure distribution may be conceived to be an interpolation between the individual solutions of the perfectly smooth EHD problem and the rough boundary lubrication.

![Figure 5.4.: Direction of sliding on a tooth surface [1].](image-url)
5. Simulation of wear of the gear tests

(BDR) and that each separate problem should contribute a fraction of the total normal pressure distribution [26].

In order to obtain $P^{MIX}$ (mixed film lubrication normal contact pressure), one must know the $P^{EHD,T}$ (normal contact pressure of the ideally smooth EHD contact pressure) and $P^{BDR,T}$ (normal contact pressure of the rough boundary lubrication problem), which are calculated, as well as $f_\Lambda$, which must be obtained from traction tests.

$$P^{MIX}(x, t) = g(x, t) \cdot P^{EHD,T} + (1 - g(x, t)) \cdot P^{BDR,T}$$  \hspace{1cm} (5.12)

$$0 \leq g(x, t) \leq 1$$  \hspace{1cm} (5.13)

For the present purpose it is sufficiently accurate to allow $g$ only to depend on time, indeed that it ultimately depends only on $\Lambda$, the oil properties and the type of roughness of the surfaces. $g(\Lambda)$ must obey the same general restrictions that $f_\Lambda(\Lambda)$ must:

$$\lim_{\Lambda \to 0} g = 0$$  \hspace{1cm} (5.14)

$$\lim_{\Lambda \to \infty} g = 1$$  \hspace{1cm} (5.15)

The procedure, illustrated in Figures 5.5 to 5.7 is thus, at each instant in time:

![Figure 5.5.: Mixed film lubrication: calculation of the EHD portion of the normal pressure [26].](image-url)
5.2. Numerical models

Figure 5.6.: Mixed film lubrication: calculation of the BDR portion of the normal pressure [26].

Figure 5.7.: Mixed film lubrication: calculation of the normal pressure [26].

- calculate $P^{EHD,T}(x,t)$;
- calculate $\Lambda$;
- calculate $P^{BDR,T}(x,t)$;
- define the fraction of the mixed film lubrication pressure contributed by $P^{EHD,T}(x,t)$:
  $P^{EHD,T}(x,t) = f_{\Lambda} \cdot P^{EHD}(x,t)$;
- define the fraction of the mixed film lubrication pressure contributed by $P^{BDR,T}$:
  $P^{BDR,T}(x,t) = (1 - f_{\Lambda}(\Lambda)) \cdot P^{BDR}(x,t)$;
- sum the fractions (EHD and boundary) to obtain $P^{MIX}$:
  $P^{MIX} = P^{EHD}(x,t) + P^{BDR}$

From the procedure outlined and the approximate equality of $g_{\Lambda}$ and $f_{\Lambda}(\Lambda)$ it is easy to deduce that $P^{EHD}$ can be taken as an approximation to the actual portion of the mixed film contact pressure borne by lubricating film and $P^{BDR}$ as approximation to the portion borne by the direct contact between the surfaces [26].
5. Simulation of wear of the gear tests

**Smooth EHD lubrication normal contact pressure** For the model to be fast and simple, Brandão et al. employed the Grubin solution instead of the Reynolds equations. That is how Brandão et al. was able to obtain $P_{EHD,t}^{EHD}$. In addition, the Grubin solution is sufficiently close to the Hertzian pressure distribution. Thus $P_{EHD,t}^{EHD}$:

$$P_{EHD} = p_0 \sqrt{1 - \left(\frac{x}{a}\right)^2} \quad (5.16)$$

where

- $p_0$ - is the Hertzian maximum pressure;
- $a$ - is the Hertzian half-width of the contact.

**Rough boundary lubrication normal contact pressure** To obtain $P_{BDR,T}^{BDR}$ it was necessary to solve the rough dry contact problem at each instant $t$:

$$h(x,t) = h_0(x,t) - \frac{2}{\pi E^*} \int_{-\infty}^{\infty} \ln \left| \frac{x - x'}{L(t)} \right| P_{BDR,T}^{BDR}(x,t) dx' \quad (5.17)$$

$$\int_{-\infty}^{\infty} P_{BDR,T}^{BDR}(x,t) dx = F(t) \quad (5.18)$$

$$\forall x,t : h(x,t) \geq 0 \land P_{BDR,T}^{BDR}(x,t) \geq 0 \quad (5.19)$$

Where:

- $h(x,t)$, the separation between the surfaces;
- $P_{BDR,T}^{BDR}(x,t)$, the contact pressure;
- $L(t)$, an integration constant, which may be interpreted as the distance from an infinitesimal line load at which the vertical displacement vanishes;
- $h_0(x,t)$, the undeformed separation between the surfaces;
- $E^*$ the equivalent Young’s modulus of the surfaces.

The algorithms used to solve this problems were proposed by Polonsky and Keer [27].

**Contact tangential stress** In order to determine the contact tangential stress, one must:

- determine the smooth EHD tangential contact traction $\tau_{EHD}^{EHD}(x,t)$;
- determine the rough boundary tangential contact traction $\tau_{BDR}^{BDR}(x,t)$;
- add the two parts to obtain the mixed film lubrication contact stress:
  $$\tau_{MIX}^{MIX}(x,t) = \tau_{EHD}^{EHD}(x,t) + \tau_{BDR}^{BDR}(x,t).$$
5.2. Numerical models

Smooth EHD lubrication contact tangential stress  The friction force within the contact can be determine with the thermal equations. The thermal solution of the EHD lubrication contact presented here is derived fro Tevaarvek’s [28] shear plane hypothesis.

The results relevant to the present work are:

\[ \Delta T_{s}^{\text{max}} = \frac{1}{\sqrt{2\pi}} \frac{\mu F_{EHD}}{PT} \frac{U_1 - U_2}{\sqrt{a}} \left( \frac{1}{\sqrt{U_1}} + \frac{1}{\sqrt{U_2}} \right) \]  
\[ \Delta T_{f}^{\text{max}} = \frac{1}{\beta} \ln \left( \frac{\beta (U_1 + U_2)^2 \eta}{8K_f} + 1 \right) \]  
\[ \Delta T_{f}^{\text{avg}} = T_0 + \frac{4}{\pi} (\Delta T_{s}^{\text{max}} + \Delta T_{f}^{\text{max}}) \]

Where:

- \( T_{s}^{\text{max}} \) - is the greatest rise of the temperature of the surfaces above that of the inlet;
- \( T_{f}^{\text{max}} \) - is the greatest rise of the temperature of the lubricant;
- \( T_{f}^{\text{avg}} \) - is the average lubricant temperature in the contact;
- \( \mu \) - is the average coefficient of friction within the contact.

\[ PT = \sqrt{\rho_s C_s K_s} \]  

Where:

- \( \rho_s \) - is the volumic mass of the surfaces;
- \( C_s \) - is the specific heat of the surfaces;
- \( K_s \) - is the thermal conductivity of the surfaces;
- \( \mu \) - is the average coefficient of friction within the contact;
- \( \eta \) - is the viscosity of the lubricant at the average surface temperature and pressure conditions within the contact;
- \( \beta \) - is the thermoviscosity coefficient of the lubricant at the average surface temperature and pressure conditions within the contact.

The lubricant temperature \( T_{f}^{\text{avg}} \) and coefficient of friction \( \mu \) are mutually dependent and are not known. To unlock this deadlock in the model the Bair and Winer viscoplasticity equation was used.

\[ \dot{\gamma} = \frac{\tau_L}{\eta} \ln \left( 1 - \frac{\tau}{\tau_L} \right) \]  

Where:
5. Simulation of wear of the gear tests

- $\dot{\gamma}$ - is the in-plane shear strain rate;
- $\tau$ - is the in-plane shear stress;
- $K_s$ - is the thermal conductivity of the surfaces;
- $\tau_L$ - is the limiting shear stress.

The coefficient of friction between the gear teeth is:

$$\mu_{EDH} = \frac{\tau_{EHD}}{F_{EN}} \cdot 2a$$  \hspace{1cm} (5.25)

Where:

- $\tau_{EHD}$ - is the average shear stress within the lubricant oil

The following equation was obtained by the manipulation of the equation 5.24

$$\mu_{EHD} = \frac{2a\tau_L}{F_{EN}} \left[ 1 - \exp \left( -\frac{\eta \dot{\gamma}}{\tau_L} \right) \right]$$  \hspace{1cm} (5.26)

Where:

- $\eta$ is determined at the average lubricant temperature $T_f^{max}$ and the average Hertezian pressure $(\pi/4)p_0$.

The temperature and the average coefficient of the friction were not used as they were complicate the computation and would not bring any relevant improvements. Equation 5.21 conjointly with equation 5.22 are enough to determine the coefficient of friction within the contact using an iterative scheme of computation.

Finally $\tau_{EHD}$ can be obtained as follows:

$$\tau_{EHD}(x,t) = \mu_{EHD} \cdot \rho_{EHD}(x,t)$$  \hspace{1cm} (5.27)

Rough boundary lubrication contact tangential stress  The boundary coefficient of friction is essentially a property of the lubricant oil and of the roughness of the surfaces and must be determined experimentally. It has been found that the resulting coefficient of friction $\mu_{BDR}$ is constant over variations of geometry, load and velocities. Typically ranging in values from: 0.08-0.15.

Thus the rough boundary lubrication contact tangential stress $\tau_{BDR}$ [26].

$$\tau_{BDR}(x,t) = \mu_{BDR}(t) \cdot \rho_{BDR}(x,t)$$  \hspace{1cm} (5.28)

Figure 5.8 displays the diagram of the numerical model.
5.3. Model of wear in spur gear teeth

As previously stated wear is one of the forms of surface damage that causes a variation on the gear tooth flanks. The model uses the following assumptions:

- the problem is two-dimensional and the teeth are in plane strain;
- one tooth is representative of all other teeth in the same gear;
- the pressure distribution history of one meshing may be used for all subsequent meshings [29].

Archard’s law, which describes the wear volume loss due to the sliding contact between flat surfaces:

$$\frac{\Delta V}{S} = \frac{K}{H} F_N$$  \hspace{1cm} (5.29)

Where:

- $\Delta V$ - is the volume loss;
- $S$ - is the sliding distance;
- $K$ - is the dimensionless wear coefficient;
- $H$ - is the softer surfacer’s hardness;
- $F_N$ - normal contact load.

In contact between gear teeth the Archard’s wear law must be written in a more complex form:

$$\frac{dh(x, t)}{dt} = kp(x, t)|U_2(t) - U_1(t)|$$  \hspace{1cm} (5.30)

Where:
5. Simulation of wear of the gear tests

- $h$ - is the wear depth;
- $p$ - is the contact pressure;
- $k$ - is the wear coefficient (with units of $Pa^{-1}$);
- $x$ - is the position on the surface of the tooth;
- $t$ - is the time coordinate;
- $U_1$ and $U_2$ - is the tangential velocity of the pinion and wheel tooth respectively.

The $k$ coefficient is presumed to be constant in time and position. To know it the height diminished during one full turn of the pinion tooth of a point situated at coordinate $x$, the following calculation is used:

$$\Delta h(x) = \int_{t_A}^{t_E} kp(x,t)|U_2(t) - U_1(t)|dt$$

Where:

- $t_A$ - is the instant when the tooth first comes in contact with is counterpart on the wheel (point A in Figure 5.2);
- $t_E$ - is the instant when the tooth ceases contact (point E in Figure 5.2).

The depth worn during $N_{\text{turns}}$ turns of the pinion will be:

$$h(x) = N_{\text{turns}} \Delta h(x)$$

Therefore, the volume lost by wear $\Delta V$ on all pinion teeth during $N_{\text{turns}}$ turn of the pinion is:

$$\Delta V = Z_1 b \int_{x_a}^{x_A} h(x)dx$$

The following expression is a simplification:

$$\Delta V = N_{\text{turns}} b Z_1 \int_{t_e}^{t_A} F_N(t)|U_2(t) - U_1(t)|dx$$

Figure 5.9 displays how the simulation of one meshing between a pair of teeth is performed and the numerical model illustrated by equations 5.29 -5.34. Note that the distribution of load between simultaneously contacting pairs of teeth is more important than the precise distribution of pressure in the contact between a pair of teeth because it has no influence on the overall wear mass loss.
5.3. Model of wear in spur gear teeth

In order to find the surface pressure distribution it is necessary to use the mixed lubrication model described in Figure 5.2.2. Equation 5.30 the increment in wear depth to be added in the running total. After the disengagement of the pinion tooth under study (instant \( t_E \)), the wear depth distribution on one pinion tooth after one full turn of the pinion \( \Delta h(x) \) has been obtained. The total wear depth distribution \( h(x) \) is computed by equation 5.32.

It is possible to obtain the wear volume \( \Delta V \) through two methods:

- by the quadrature of wear depth distribution, equation 5.33;
- by applying equation 5.34 in which case here is no need to perform the discrete computations described, since the wear volume can be computed analytically.

As seen in the Chapter 2, in the Archard’s law the wear coefficient ”k” is the main difficulty. In literature, there is a little agreement about the value to be used in cases of lubricated contact, with authors using wear coefficients ranging from \( 10^{-18} \) to \( 10^{-12} \) [1].

If it is accepted that the density \( \rho = 7850 \text{kg/m}^3 \) of the gears steel remains constants during the test, the volume loss \( \Delta V \) is easily computed:

\[
\Delta V = \frac{M_l}{\rho} \quad (5.35)
\]

The following equation relation between k and \( \Delta V \) which can be written, after appropriate manipulation of the expressions:

\[
k = \frac{M_l}{\rho N_{\text{turns}} b Z_1 \int_{t_A}^{t_E} F_N(t) |U_2(t) - U_1(t)| \, dt} \quad (5.36)
\]

Where:

- \( k \) - wear coefficient;
- \( M_l \) - mass loss;
5. Simulation of wear of the gear tests

- \( \rho \) - density of the gears steel;
- \( N_{\text{turns}} \) - number of revolutions;
- \( b \) - tooth flank length;
- \( Z_1 \) - number of teeth of the pinion;
- \( F_N \) - normal contact force;
- \( U_2 \) - tangential velocity of wheel tooth surface;
- \( U_1 \) - tangential velocity of pinion tooth surface.

5.4. Simulation results and discussions

To calculate the pinion wear’s coefficient, was measured the pinion’s mass in the beginning and after each test. The difference between these two measurements is the mass loss. Wear coefficient, \( k \), is obtained from equation 5.36 in \( m^2/N \) (Pa\(^{-1}\)).

All data presented in this section refer to the pinion tooth number 1.

Table 5.1.: Mass loss and wear coefficient

<table>
<thead>
<tr>
<th>Test</th>
<th>Load Stage</th>
<th>Mass Loss [mg]</th>
<th>Mass loss rate</th>
<th>( k ) [10(^{-15}) m(^2)/N]</th>
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<tr>
<td>T01</td>
<td>K9</td>
<td>31.4</td>
<td>17.187</td>
<td>25.941</td>
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<tr>
<td>T02</td>
<td>0.65</td>
<td>28.8</td>
<td>15.803</td>
<td>23.926</td>
</tr>
<tr>
<td>T03</td>
<td>0.36</td>
<td>44.2</td>
<td>24.259</td>
<td>36.726</td>
</tr>
<tr>
<td>T04</td>
<td>MIN</td>
<td>48.0</td>
<td>26.326</td>
<td>39.839</td>
</tr>
</tbody>
</table>

Table 5.1 gives the wear coefficient that lead to each wear depth curve, along with the corresponding mass loss and rate of mass loss. Each line gives this information for a specific test. For example, the first line shows that test T01 lost 31.4 mg to wear, which correspond to a mass loss rate of 17.187 \( \times 10^{-3} \) mg for each mm\(^2\) of active tooth surface; or 25.941 \( \times 10^{-3} \) mg/mm\(^2\) below the pitch point and 12.984 \( \times 10^{-3} \) mg/mm\(^2\) above the pitch point. This mass loss is compatible with a wear rate \( k = 3.07 \times 10^{-15} \) Pa\(^{-1}\).

5.4.1. Influence of load
5.4. Simulation results and discussions

Figure 5.10.: Depth of wear along a pinion tooth flank of test T12 (K5)

Figure 5.11.: Depth of wear along a pinion tooth flank of test T11 (K7)

Figure 5.12.: Depth of wear along a pinion tooth flank of test T01 (K9)
Figures 5.10 to 5.18 display the wear depth along tooth flank. The zero [mm] in the Figures correspond to the root tooth, while the six [mm] correspond the area near the head of the tooth. The Figures were placed in ascending order, where Figure 5.10 has the K5 load stage, Figure 5.11 has the K7 load stage and Figure 5.18 has the K9 load stage.

Figure 5.13 display the influence of the variation of load on wear coefficient. These profiles that reflect wear depth, were obtained by subtracting the initial roughness profiles from the roughness profiles obtained by the model described above. Notice how a basic wear depth curve serves as baseline for peaks of wear. The base curve corresponds to the EHL (smooth) portion of the contact pressure and the wear peaks to the pressure spikes of the boundary lubrication part, coinciding with strong interactions between roughness features on the surface of the pinion and wheel tooth.

Analysing the all depth profiles, all show higher wear in the root of the tooth. This is explained by the fact that sliding speed is higher in the root of the tooth.

Figures 5.10 to 5.18 display the influence of load variation. Visually the increase of the load causes an increased in wear. Figure 5.18 show maximum wear depth of seven [mm] while Figure 5.10 display maximum wear depth of two [mm]. Through Table 5.1 could confirm the mass loss values above and below the pitch circle are according to each figure. The wear below the pitch circle is higher than above.

Figure 5.13 surprisingly shows that the wear coefficient (k) varies with the increase of load. It is consistent with the Archard wear law that the load increase is not causes such a notorious variation in friction coefficient, as shown in Figure 5.13.

The micropitting distribution in each test seems to be uniform along all the tooth flank.

5.4.2. Influence of film thickness
5.4. Simulation results and discussions

Figure 5.14.: Depth of wear along a pinion tooth flank of test T01 ($\Lambda=0.12$)

Figure 5.15.: Depth of wear along a pinion tooth flank of test T03 ($\Lambda=0.36$)

Figure 5.16.: Depth of wear along a pinion tooth flank of test T02 ($\Lambda=0.65$)
5. Simulation of wear of the gear tests

Figures 5.16 to 5.15 displays the wear evolution along tooth flank. The zero [mm] in the Figures correspond to the root tooth, while the six [mm] correspond the area near the head of the tooth. The Figures were placed in ascending order, where Figure 5.16 has the 0.36 Λ and Figure 5.15 has 0.65 Λ.

Figure 5.17 display the influence of the variation of Λ on wear coefficient. In contrast what happened in the load variation, visually, the increase of the Λ do not cause an increase in wear. Figure 5.15 show maximum wear depth of nine [mm] while Figure 5.16 display maximum wear depth of six [mm]. As in the tests in which the load varies, the greater wear is found on the tooth root as shown in Figure 5.1.

Figure 5.17 shows that the wear coefficient (k) varies with the increase of Λ as expected. The fact that the T03 test has greater mass loss and wear coefficient can be justified by the fact has occurred a switch in wear mechanism.

Also with the variation of Λ the micropitting distribution in each test seems to be uniform along all the tooth flank.

5.4.3. Influence of lubricant

Figure 5.18.: Depth of wear along a pinion tooth flank of test T01 (PAO)
5.5. Comparison of simulation results with gears tests and discussions

Comparing the test where the influence of lubricant (Figure 5.18 and Figure 5.19) occurred an increase of wear depth, wear coefficient and micropitting.

All the roughness profiles was filtered by means of a Gaussian filter and have an cut-of of 0.8[mm].

5.5. Comparison of simulation results with gears tests and discussions

The same color code used in chapter 3 is used in this section.

- black profiles - new roughness profiles (unused);
- blue profiles - used roughness profiles (after test);
- black circle - identifies the risk that was handmade in the tooth;
- red profiles - predicted roughness profiles;
- vertical red lines - C: pitch line; between B and D, engagement of a single pair of teeth left off; B: initial engagement of two pair of teeth D: initial engagement of two pairs of teeth.

The Figures 5.20 to 5.22 displays the measured initial roughness profile (black), measured final roughness profile (blue) and predicted final roughness profile (red)

Figure 5.19.: Depth of wear along a pinion tooth flank during test T04 (MIN)

Figure 5.20.: Roughness profiles of T01 test: measured initial roughness profile (black), measured final roughness profile (blue) and predicted final roughness profile (red)
5. Simulation of wear of the gear tests

Figure 5.21.: Roughness profiles of T11 test: measured initial roughness profile (black), measured final roughness profile (blue) and predicted final roughness profile (red)

Figure 5.22.: Roughness profiles of T12 test: measured initial roughness profile (black), measured final roughness profile (blue) and predicted final roughness profile (red)

Figure 5.23.: Roughness profiles of T02 test: measured initial roughness profile (black), measured final roughness profile (blue) and predicted final roughness profile (red)
At this point it is useful to explain why the position of the gears in the profilometer is a difficulty in this work. The profiles were measured in the wheel and pinion were positioned manually. Consequently the orientation and the position of the measured device with regard to the profilometer changed from measurement to measurement. This is significant because a gear tooth surface has a large curvature that must be filtered out by means of a Gaussian filter.

With the method described in Chapter 3, the handmade mark on the tooth flank it has become very useful as it served as a reference. With is handmade mark managed to locate the same area in the new and used topography and extract the same roughness profile. It would expect the overall shape of the predicted roughness profile to be similar to that of the measured one.

Seems to be remarkable agreement between the predicted and the measured profiles. Despite all predicted roughness profile has slightly more micropitting, particularly in the tooth root. Model seems to fail when increasing the severity of the test, Figures 5.24 and 5.25

5.6. Sumary

A model of simulation of wear of the gear tests was present in detail. The model was shown remarkable agreement between the predicted and the measured profiles.

- Influence of load - wear coefficient (k) was not constant with the increase of the load, as shown in Figure 5.13;
5. Simulation of wear of the gear tests

- Influence of Λ - wear coefficient varies non-linearly manner with the increase of the Λ, as shown in Figure 5.17;

- Influence of lubricant - wear coefficient (k) increases with the variation of the lubricant, as shown in Table 5.1.

In all test the wear is higher at the root of the pinion as described in the literature.
6. General conclusion and future work

6.1. Conclusion

Chapter 2 attempted to present a literature review on the several fields which are necessary to understand the work developed.

In Chapter 3 the experimental works presented was:

- 7 tests divided in 3 groups: load variation, Λ variation and substitution of a PAO oil by a Mineral oil;
- fourteen weighing procedures;
- more than sixty topographies;
- ferrography;
- power loss analysis.

Experiments results in mass loss, ferrography and roughness show the same evidence:

- an increase of wear with load;
- an increase of wear when replacing PAO with Mineral oil;
- similar wear at Λ = 0.12 and Λ = 0.65 but higher wear at intermediate Λ = 0.36 which may be associated with a wear mechanism transition due to temperature. As the oil is strongly additivated and the value of the oil temperature varied that causes a change in the behaviour of additives which may explain the mass loss behaviour;
- power loss results are somewhat different, because input power increases with load and velocity, which corresponds roughly to increases in $F_N$, Λ and a decrease temperature in our tests. Consequently one expects total power loss to increase similarly. Such in the case in the tests. However, tests performed with the same operating conditions but different oils shows clearly a correspondence of higher wear with higher power loss.
6. General conclusion and future work

In Chapter 4, which shows power loss simulation results the adjustments made to the power loss model allowed to achieve very good results. An interesting fact is that the proportion of the power loss due to \( C_{14} \) load losses in the total power loss diminished gradually with increases of \( \Lambda \). This may explain the diminution of the mass loss from T03 to T02. Power loss increase with the substitution of a PAO oil by a Mineral oil.

In Chapter 5, which contains a numerical wear simulation, the simulation showed significantly more intense wear below the pitch circle than above on the pinion. Surprisingly the wear coefficient \( k \) varies with the increase of load. Wear coefficient have a greater value for the intermediate film thickness (T03) compared to the extremes T01 and T02. There seemed to be remarkable agreement between the predicted and the measured profiles regarding the general shape of the profile.

6.2. Future work

To improve the understanding of wear, some or all of the tests could be repeated, only with a significant alterations: instead of only performing measurements at begins and end of test, each test could be interrupted at certain intervals to perform intermediate measurements. This would give a clear picture of the evaluation of wear as test progresses. Another useful extensions of this work would be to study the influence of other operating conditions: profile shift, profile modifications, initial roughness and switched form spur to helical gears.
Bibliography


Bibliography


A. Appendix

A.1. Tests, ferrography results, roughness profiles and tests pictures

A.1.1. T01 test

A.1.1.1. Test sheet
TEST DATA SHEET

Test Ref.: T01  
Start Date: 02/03/2015
Gear Ref: 3700  
Finish Date: 07/03/2015
Side: A
Material: 16MnCr5 case-carburized

TEST FEATURES

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MASS

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Notes:
Oil sample was taken after re-launch the test without load.
A.1.1.2. Ferrography results
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**LEGENDA**
- DL - Índice de partículas grandes
- DS - Índice de partículas pequenas
- CPUC - Concentração part. de desgaste
- ISUC - Índice Severidade de Desgaste

**CLIENTE:** INEGI  
**MÁQUINA:** Máquina de Ensaios FZG  
**MORADA:** Porto  
**Ref. ÓLEO:** Renolin Unisyn CLP 150  
**Teste Ref.:** T01
CLIENTE: INEGI
MÁQUINA: Máquina de Ensaios FZG
MORADA: Porto
Ref. ÓLEO: Renolin Unisyn CLP 150
Teste Ref.: T01

Ampliação: x 200 Diluição: 0,1
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Observações: Ampliação da Fotografia 1.
Partículas ferrosas, algumas de grandes dimensões.

Ampliação: x 1000 Diluição: 0,1
Localização: Núcleo Luz: Branca / Verde
Observações: Ampliação da Fotografia 1.
Presença de partículas ferrosas de grandes dimensões, típicas de desgaste de micropitting.
Ampliação: x 200 Diluição: 0,1
Localização: Núcleo Luz: Branca / Verde
Observações: Ampliação da Fotografia 1.
Presença de partículas ferrosas de grandes dimensões, típicas de desgaste de micropitting.

Ampliação: x 1000 Diluição: 0,1
Localização: Núcleo Luz: Branca / Verde
Observações: Ampliação da Fotografia 1.
Presença de partículas ferrosas de grandes dimensões, típicas de desgaste de micropitting.
A.1.3. Roughness profiles
A. Appendix

Figure A.1.: Roughness profile of the tooth 1 from the pinion of T01 test

Figure A.2.: Roughness profile of the tooth 9 from the pinion of T01 test

Figure A.3.: Roughness profile of the tooth 1 from the wheel of T01 test

Figure A.4.: Roughness profile of the tooth 9 from the wheel of T01 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.5.: Roughness profile of the tooth 17 from the wheel of T01 test

Table A.1.: Average parameters on each tooth in T01 test

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<td>3,435</td>
</tr>
<tr>
<td>RZDin</td>
<td>2,389</td>
<td>2,862</td>
<td>2,753</td>
<td>2,685</td>
<td>3,124</td>
</tr>
<tr>
<td>Ra</td>
<td>0,355</td>
<td>0,325</td>
<td>0,374</td>
<td>0,309</td>
<td>0,407</td>
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<tr>
<td>Rq</td>
<td>0,472</td>
<td>0,461</td>
<td>0,488</td>
<td>0,427</td>
<td>0,533</td>
</tr>
<tr>
<td>Rpk</td>
<td>0,484</td>
<td>0,259</td>
<td>0,510</td>
<td>0,267</td>
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<tr>
<td>Rk</td>
<td>1,104</td>
<td>1,050</td>
<td>1,135</td>
<td>0,938</td>
<td>1,268</td>
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<tr>
<td>Rvk</td>
<td>0,754</td>
<td>0,994</td>
<td>0,653</td>
<td>0,789</td>
<td>0,802</td>
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Table A.2.: Average parameters of the pinion and wheel in T01 test

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<th>Wheel</th>
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<tr>
<td>Ra max</td>
<td>3,234</td>
<td>6,551</td>
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<tr>
<td>RZDin</td>
<td>2,571</td>
<td>2,830</td>
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<td>Ra</td>
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<td>Rq</td>
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<tr>
<td>Rpk</td>
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<td>Rk</td>
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<tr>
<td>Rvk</td>
<td>0,703</td>
<td>0,685</td>
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</table>
A. Appendix

A.1.1.4. 3D optical microscopy
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.6.: Picture from used head of the pinion tooth 1 of T01 test

Figure A.7.: Picture from used pitch circle of the pinion tooth 1 of T01 test

Figure A.8.: Picture from used head of the wheel tooth 1 of T01 test
A. Appendix

Figure A.9.: Picture from used pitch circle of the pinion tooth 1 of T01 test

Figure A.10.: Topography from used head of the pinion tooth 1 of T01 test

Figure A.11.: Topography from used pitch circle of the pinion tooth 1 of T01 test

Figure A.12.: Topography from used head of the wheel tooth 1 of T01 test

Figure A.13.: Topography from used pitch circle of the wheel tooth 1 of T01 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

A.1.2. T02 test

A.1.2.1. Test sheet
Mild wear in lubricated gear transmissions

Operator: Pedro Aires da Cunha Cerqueira

TEST DATA SHEET

Test Ref.: T02  Start Date: 24/02/2015
Gear Ref.: 3699  Finish Date: 25/02/2015
Side : A
Material: 16MnCr5 case-carburized

TEST FEATURES

Oil: Renolin Unisyn CLP 150
Temperature[°C]: 80
ΔT(Traction-T environment) [°C]: 40,5  ΔT(Toil-T environment) [°C]: 60,7
Pinion cycles: 5.000.000
Days: 0,8
Lubrication type: Dip lubrication
Wheel torque: K9 - 215,6 N.m
Pinion speed[rpm]: 4500
Input Power [KW]: 101,59
Torque loss [N.m]: 10,15

MASS

Pinion  Standard pattern pinion
Initial mass (average) [g]: 634,1344  Initial mass (average) [g]: 528,3914
Finish mass (average) [g]: 634,1028  Finish mass (average) [g]: 528,3886

Variation of the initial mass: 105,743
Variation of the finish mass: 105,7142

MASS LOSS [mg]: 28,8

Notes:
A.1.2.2. Ferrography results
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<tr>
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<tr>
<td>DL: 50,2</td>
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<tr>
<td>DS: 11,1</td>
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<tr>
<td>CPUC: 613,0</td>
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<tr>
<td>ISUC: 2,4E+05</td>
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<tr>
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<td>Desgaste severo</td>
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<tr>
<td>Desgaste de abrasão</td>
</tr>
<tr>
<td>Desgaste combinado</td>
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<tr>
<td>Desgaste de fadiga</td>
</tr>
<tr>
<td>Desgaste de adesão</td>
</tr>
<tr>
<td>Partículas não ferrosas</td>
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<tr>
<td>Esferas de fadiga</td>
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<tr>
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<td>Minerais/Orgânicos</td>
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<td>40 - 50 μm</td>
</tr>
<tr>
<td>50 - 60 μm</td>
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<tr>
<td>60 - 70 μm</td>
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<td>(mg KOH)</td>
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<td>DS - Índice de partículas pequenas</td>
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<tr>
<td>CPUC - Concentração part. de desgaste</td>
</tr>
<tr>
<td>ISUC - Índice Severidade de Desgaste</td>
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<tr>
<td>Não existe</td>
</tr>
<tr>
<td>Fraco</td>
</tr>
<tr>
<td>Médio</td>
</tr>
<tr>
<td>Forte</td>
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</tbody>
</table>
Ampliação: x 1000  Diluição: 0,1  
Localização: Núcleo  Luz: Branca / Verde  
Observações: Ampliação da Fotografia 2.  
Partícula ferrosa de grandes dimensões, típica de desgaste de fadiga.

Ampliação: x 1000  Diluição: 0,1  
Localização: Núcleo  Luz: Branca / Verde  
Observações: Ampliação da Fotografia 2.  
Partícula ferrosa de grandes dimensões, típica de desgaste de fadiga.
Ampliação: x 1000  
Diluição: 0,1
Localização: Meio  
Luz: Branca / Verde
Observações: Partícula de um óxido termico de grandes dimensões.

Ampliação: x 1000  
Diluição: 0,1
Localização: Meio  
Luz: Branca / Verde
Observações: Partículas ferrosas de pequenas dimensões, alinhadas segundo as forças do campo magnético.

Ampliação: x 1000  
Diluição: 0,1
Localização: Meio  
Luz: Branca / Verde
Observações: Ampliação da Fotografia 5. Partículas ferrosas de desgaste, algumas típicas de desgaste de micropitting (1).

Ampliação: x 1000  
Diluição: 0,1
Localização: Saída  
Luz: Branca / Verde
Observações: Partículas ferrosas de pequenas dimensões, típicas de desgaste de corrosão.
A.1. Tests, ferrography results, roughness profiles and tests pictures

A.1.2.3. Roughness profiles
Figure A.14.: Roughness profile of the tooth 1 from the pinion of T02 test

Figure A.15.: Roughness profile of the tooth 9 from the pinion of T02 test

Figure A.16.: Roughness profile of the tooth 1 from the wheel of T02 test

Figure A.17.: Roughness profile of the tooth 9 from the wheel of T02 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.18.: Roughness profile of the tooth 17 from the wheel of T02 test

| Table A.3.: Average parameters on each tooth in T02 test |
|---------------------------------|---|---|---|---|---|---|---|---|
|                                | P01 | P09 | W01 | W09 | W17 |
| RZDin                          | 2.654 | 2.376 | 3.504 | 2.431 | 3.643 | 2.258 | 2.851 | 2.843 | 4.018 | 3.220 |
| Ra                             | 0.404 | 0.305 | 0.510 | 0.298 | 0.483 | 0.342 | 0.411 | 0.379 | 0.562 | 0.375 |
| Rq                             | 0.538 | 0.421 | 0.666 | 0.399 | 0.611 | 0.502 | 0.518 | 0.567 | 0.721 | 0.523 |
| Rpk                            | 0.648 | 0.302 | 0.778 | 0.296 | 0.625 | 0.435 | 0.537 | 0.369 | 0.757 | 0.557 |
| Rk                             | 1.297 | 0.952 | 1.589 | 0.965 | 1.591 | 0.947 | 1.430 | 1.007 | 1.782 | 1.050 |
| Rvk                            | 0.633 | 0.818 | 0.798 | 0.680 | 0.614 | 1.604 | 0.534 | 1.934 | 0.789 | 1.308 |

| Table A.4.: Average parameters of the pinion and wheel in T02 test |
|---------------------------------|---|---|---|---|
|                                | Pinion | Wheel |
|                                | Novos | Usados | New | Used |
| Ra max                         | 3.926 | 3.332 | 4.250 | 3.972 |
| RZDin                          | 3.079 | 2.404 | 3.504 | 2.774 |
| Ra                             | 0.457 | 0.301 | 0.486 | 0.365 |
| Rq                             | 0.602 | 0.410 | 0.617 | 0.531 |
| Rpk                            | 0.713 | 0.299 | 0.640 | 0.454 |
| Rk                             | 1.443 | 0.958 | 1.601 | 1.001 |
| Rvk                            | 0.716 | 0.749 | 0.646 | 1.616 |
A. Appendix

A.1.2.4. 3D optical microscopy
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.19.: Picture from used head of the pinion tooth 1 of T02 test

Figure A.20.: Picture from used circle pitch of the pinion tooth 1 of T02 test

Figure A.21.: Picture from used head of the wheel tooth 1 of T02 test
A. Appendix

Figure A.22.: Picture from used pitch circle of the wheel tooth 1 of T02 test

Figure A.23.: Topography from used head of the wheel tooth 1 of T02 test

Figure A.24.: Topography from used pitch circle of the pinion tooth 1 of T02 test

Figure A.25.: Topography from used head of the wheel tooth 1 of T02 test

Figure A.26.: Topography from used pitch circle of the wheel tooth 1 of T02 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

A.1.3. T03 test

A.1.3.1. Test sheet
Faculty of Engineering of the University of Porto (FEUP)
Mild wear in lubricated gear transmissions

Operator: Pedro Aires da Cunha Cerqueira

TEST DATA SHEET

Test Ref.: T03  Start Date: 09/03/2015
Gear Ref: 3699  Finish Date: 11/03/2015
Side: B
Material: 16MnCr5 case-carburized

TEST FEATURES

Oil: Renolin Unisyn CLP 150
Temperature[°C]: 90
ΔT(Tin-Tenvironment) [°C]: 39,3  ΔT(Tout-Tenvironment) [°C]: 66,5
Pinion cycles: 5,000,000
Days: 1,5
Lubrication type: Dip lubrication
Wheel torque: K9 - 215,6 N.m
Pinion speed[rpm]: 2550
Input Power [KW]: 57,57
Torque loss [N.m]: 7,52

MASS

Pinion  Standard pattern pinion
Initial mass (average) [g]: 634,1028  Initial mass (average) [g]: 528,3886
Finish mass (average) [g]: 634,0608  Finish mass (average) [g]: 528,3908

Variation of the initial mass: 105,7142
Variation of the finish mass: 105,67

MASS LOSS [mg]: 44,2

Notes:
Oil sample was taken after re-launch the test without load.
A.1.3.2. Roughness profiles
Figure A.27.: Roughness profile of the tooth 1 from the pinion of T03 test

Figure A.28.: Roughness profile of the tooth 9 from the wheel of T03 test

Figure A.29.: Roughness profile of the tooth 1 from the wheel of T03 test

Figure A.30.: Roughness profile of the tooth 9 from the wheel of T03 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.31.: Roughness profile of the tooth 17 from the wheel of T03 test

Table A.5.: Average parameters on each tooth in T03 test

<table>
<thead>
<tr>
<th></th>
<th>P01</th>
<th>P09</th>
<th>W01</th>
<th>W09</th>
<th>W17</th>
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<tbody>
<tr>
<td></td>
<td>Novos</td>
<td>Usados</td>
<td>Novos</td>
<td>Usados</td>
<td>Novos</td>
</tr>
<tr>
<td>Ra max</td>
<td>4,250</td>
<td>3,800</td>
<td>3,546</td>
<td>3,231</td>
<td>4,291</td>
</tr>
<tr>
<td>RZDin</td>
<td>3,561</td>
<td>2,586</td>
<td>3,056</td>
<td>2,625</td>
<td>3,827</td>
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<tr>
<td>Ra</td>
<td>0,570</td>
<td>0,359</td>
<td>0,415</td>
<td>0,340</td>
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<td>Rq</td>
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<td>Rpk</td>
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<td>Rk</td>
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<tr>
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<td>0,815</td>
<td>0,604</td>
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Table A.6.: Average parameters of the pinion and wheel in T03 test

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<th>Wheel</th>
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<td>Usados</td>
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<tr>
<td>Ra max</td>
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<tr>
<td>RZDin</td>
<td>3,309</td>
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<tr>
<td>Ra</td>
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<td>Rq</td>
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<tr>
<td>Rvk</td>
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</table>
A. Appendix

A.1.3.3. 3D optical microscopy
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.32.: Picture from used head of the pinion tooth 1 of T03 test

Figure A.33.: Picture from used pitch circle of the pinion tooth 1 of T03 test

Figure A.34.: Picture from used head of the wheel tooth 1 of T03 test
A. Appendix

Figure A.35.: Picture from used pitch circle of the wheel tooth 1 of T03 test

Figure A.36.: Topography from used head of the pinion tooth 1 of T03 test

Figure A.37.: Topography from used pitch circle of the pinion tooth 1 of T03 test

Figure A.38.: Topography from used head of the wheel tooth 1 of T03 test

Figure A.39.: Topography from used pitch circle of the wheel tooth 1 of T03 test
A.1.4. T03b test

A.1.4.1. Test sheet
Faculty of Engineering of the University of Porto (FEUP)
Mild wear in lubricated gear transmissions

Operator: Pedro Aires da Cunha Cerqueira

TEST DATA SHEET

Test Ref.: T03b
Gear Ref: 3700
Side: B
Material: 16MnCr5 case-carburized

TEST FEATURES

Oil: Renolin Unisyn CLP 150
Temperature[^C]: 90
ΔT(Tint-Tenvironment) [^C]: 36,1
ΔT(Toil-Tenvironment) [^C]: 63,5
Pinion cycles: 5,000,000
Days: 1,5
Lubrication type: Dip lubrication
Wheel torque: K9 - 215,6 N.m
Pinion speed[rpm]: 2550
Input Power [KW]: 57,57
Torque loss [N.m]: 7,35

MASS

Pinion

Initial mass (average) [g]: 634,592
Finish mass (average) [g]: 634,5526
Variation of the initial mass: 106,2012
Variation of the finish mass: 106,1624

Standard pattern pinion

Initial mass (average) [g]: 528,3908
Finish mass (average) [g]: 528,3902

MASS LOSS [mg]: 38,8

Notes:
Oil sample was taken after re-launch the test without load.
A.1.5. T04 test

A.1.5.1. Test sheet
**TEST DATA SHEET**

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<td>16MnCr5 case-carburized</td>
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**TEST FEATURES**

- **Oil:** Energetol GR-XP 150
- **Temperature[^C]:** 100
- **ΔT(Tr2G-Tenvironment) [^C]:** 44,8
- **ΔT(Toil-Tenvironment) [^C]:** 73,4
- **Pinion cycles:** 5,000,000
- **Days:** 5
- **Lubrication type:** Dip lubrication
- **Wheel torque:** K9-215,6N.m
- **Pinion speed[rpm]:** 750
- **Input Power [KW]:** 16,93
- **Torque loss [N.m]:** 8,47

**MASS**

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<th>Standard pattern pinion</th>
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<tbody>
<tr>
<td>Initial mass (average) [g]:</td>
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<tr>
<td>Finish mass (average) [g]:</td>
<td>635,3506</td>
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</table>

| Variation of the initial mass: | 107,0074 |
| Variation of the finish mass:  | 106,9594 |
| MASS LOSS [mg]:                | 48      |

**Notes:**
A.1. Tests, ferrography results, roughness profiles and tests pictures

A.1.5.2. Ferrography results
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<th>MAQUINA: Máquina de Ensaios FZG</th>
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**IDENTIFICAÇÃO**

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| Data amostra: |  |
| Análise nº: |  |
| Ciclos/Máquina: | - |
| Ciclos/Óleo: | - |

**FERROMETRIA**

| dL | 0,1 |
| DL | 22,4 |
| DS | 2,4 |
| CPUC | 248,0 |
| ISUC | 5,0E+04 |

**FERROGRAFIA**

- Desgaste normal
- Desgaste severo M
- Desgaste de abrasão
- Desgaste combinado
- Desgaste de fadiga
- Desgaste de adesão
- Esferas de fadiga
- Óxidos de ferro
- Minerais/Orgânicos

**OILVIEW**

- Índice OilLife:
- Índice Oclusão:
- Índice Contaminação:
- Índice Ferromagnético:
- Grandes Contaminantes:
- Constante Dieléctrica:

**CONT. PARTICULAS (%)**

| 4 - 6 μm |  |
| 6 - 10 μm |  |
| 10 - 20 μm |  |
| 20 - 30 μm |  |
| 30 - 40 μm |  |
| 40 - 50 μm |  |
| 50 - 60 μm |  |
| 60 - 70 μm |  |

**VISCOSIDADE**

(cSt a 40°C)

**ACIDEZ (TAN)**

(mg KOH)

**P. INFLAMAÇÃO**

(°C)

**DIAGNÓSTICO**

**LEGENDA**

- DL - Índice de partículas grandes
- DS - Índice de partículas pequenas
- CPUC - Concentração part. de desgaste
- ISUC - Índice Severidade de Desgaste

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<th>Fraco</th>
<th>Médio</th>
<th>Forte</th>
</tr>
</thead>
<tbody>
<tr>
<td>f</td>
<td>f</td>
<td>M</td>
<td>f</td>
</tr>
</tbody>
</table>
CLIENTE: INEGI
MÁQUINA: Máquina de Ensaios FZG
MORADA: Porto
Ref. ÓLEO: Energol GR-XP 150
Teste Ref.: T04

Fotografia 1

Ampliação: x 100
Diluição: 0,1
Localização: Núcleo
Luz: Branca / Verde
Observações: Presença significativa de partículas ferrosas alinhadas segundo as forças do campo magnético.

Fotografia 2

Ampliação: x 200
Diluição: 0,1
Localização: Núcleo
Luz: Branca / Verde
Observações: Ampliação da Fotografia 1. Presença de partículas ferrosas, algumas de grandes dimensões.

Fotografia 3

Ampliação: x 1000
Diluição: 0,1
Localização: Núcleo
Luz: Branca / Verde
Observações: Ampliação da Fotografia 1. Presença de partículas ferrosas de médias e grandes dimensões. Partícula ferrosa típica de desgaste de fadiga (1).

Fotografia 4

Ampliação: x 1000
Diluição: 0,1
Localização: Núcleo
Luz: Branca / Verde
Observações: Ampliação da Fotografia 1. Presença de partículas ferrosas de grandes dimensões, típicas de desgaste de micropitting.
Ampliação: x 1000  
Diluição: 0,1  
Localização: Núcleo  
Luz: Branca / Verde

Observações: Ampliação da Fotografia 1.  
Presença de partículas ferrosas de grandes dimensões, típicas de desgaste de micropitting.
A.1. Tests, ferrography results, roughness profiles and tests pictures

A.1.5.3. Roughness profiles
A. Appendix

Figure A.40.: Roughness profile of the tooth 1 from the pinion of T04 test

Figure A.41.: Roughness profile of the tooth 9 from the pinion of T04 test

Figure A.42.: Roughness profile of the tooth 1 from the wheel of T04 test
### A.1. Tests, ferrography results, roughness profiles and tests pictures

**Table A.7.: Average parameters on each tooth in T04 test repetition**

<table>
<thead>
<tr>
<th></th>
<th>P01 Novos</th>
<th>P09 Novos</th>
<th>W01 Novos</th>
<th>P01 Usados</th>
<th>P09 Usados</th>
<th>W01 Usados</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ra max</td>
<td>2.704</td>
<td>2.957</td>
<td>3.703</td>
<td>5.600</td>
<td>4.467</td>
<td>5.113</td>
</tr>
<tr>
<td>RZDin</td>
<td>2.083</td>
<td>2.544</td>
<td>3.092</td>
<td>3.769</td>
<td>3.438</td>
<td>3.724</td>
</tr>
<tr>
<td>Ra</td>
<td>0.292</td>
<td>0.418</td>
<td>0.405</td>
<td>0.394</td>
<td>0.407</td>
<td>0.461</td>
</tr>
<tr>
<td>Rq</td>
<td>0.413</td>
<td>0.579</td>
<td>0.518</td>
<td>0.547</td>
<td>0.554</td>
<td>0.628</td>
</tr>
<tr>
<td>Rpk</td>
<td>0.290</td>
<td>0.410</td>
<td>0.375</td>
<td>0.530</td>
<td>0.531</td>
<td>0.716</td>
</tr>
<tr>
<td>Rk</td>
<td>0.933</td>
<td>1.342</td>
<td>1.347</td>
<td>1.276</td>
<td>1.276</td>
<td>1.344</td>
</tr>
<tr>
<td>Rvk</td>
<td>0.787</td>
<td>1.003</td>
<td>0.950</td>
<td>0.931</td>
<td>0.931</td>
<td>1.213</td>
</tr>
</tbody>
</table>

**Table A.8.: Average parameters of the pinion and wheel in T04 test repetition**

<table>
<thead>
<tr>
<th></th>
<th>Pinion Novos</th>
<th>Pinion Usados</th>
<th>Wheel New</th>
<th>Wheel Used</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ra max</td>
<td>2.830</td>
<td>3.703</td>
<td>3.703</td>
<td>5.113</td>
</tr>
<tr>
<td>RZDin</td>
<td>2.314</td>
<td>3.092</td>
<td>3.603</td>
<td>3.724</td>
</tr>
<tr>
<td>Ra</td>
<td>0.355</td>
<td>0.405</td>
<td>0.401</td>
<td>0.461</td>
</tr>
<tr>
<td>Rq</td>
<td>0.496</td>
<td>0.518</td>
<td>0.550</td>
<td>0.628</td>
</tr>
<tr>
<td>Rpk</td>
<td>0.350</td>
<td>0.375</td>
<td>0.531</td>
<td>0.716</td>
</tr>
<tr>
<td>Rk</td>
<td>1.137</td>
<td>1.347</td>
<td>1.276</td>
<td>1.344</td>
</tr>
<tr>
<td>Rvk</td>
<td>0.895</td>
<td>0.685</td>
<td>0.941</td>
<td>1.213</td>
</tr>
</tbody>
</table>
A. Appendix

A.1.5.4. 3D optical microscopy
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.43.: Picture from new head of the pinion tooth 1 of T04 test

Figure A.44.: Picture from new pitch circle of the pinion tooth 1 of T04 test

Figure A.45.: Picture from new head of the wheel tooth 1 of T04 test
A. Appendix

Figure A.46.: Picture from new pitch circle of the wheel tooth 1 of T04 test

Figure A.47.: Picture from used head of the pinion tooth 1 of T04 test

Figure A.48.: Picture from used pitch circle of the pinion tooth 1 of T04 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.49.: Picture from used head of the wheel tooth 1 of T04 test

Figure A.50.: Picture from used pitch circle of the wheel tooth 1 of T04 test

Figure A.51.: Topography from new head of the pinion tooth 1 of T04 test

Figure A.52.: Topography from new pitch circle of the pinion tooth 1 of T04 test
A. Appendix

Figure A.53.: Topography from new head of the wheel tooth 1 of T04 test

Figure A.54.: Topography from new pitch circle of the wheel tooth 1 of T04 test

Figure A.55.: Topography from used head of the pinion tooth 1 of T04 test

Figure A.56.: Topography from used pitch circle of the pinion tooth 1 of T04 test

Figure A.57.: Topography from used head of the wheel tooth 1 of T04 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.58.: Topography from used pitch circle of the wheel tooth 1 of T04 test
A. Appendix

A.1.6. T11 test

A.1.6.1. Test sheet
TEST DATA SHEET

Test Ref.: T11  Start Date: 18/03/2015
Gear Ref:  3702  Finish Date: 25/03/2015
Side: A
Material: 16MnCr5 case-carburized

TEST FEATURES

Oil: Renolin Unisyn CLP 150
Temperature[°C]: 100
ΔT(Tin-Tenvironment) [°C]: 40,2  ΔT(Toil-Tenvironment) [°C]: 76,4
Pinion cycles: 5,000,000
Days: 5
Lubrication type: Dip lubrication
Wheel torque: K7-132,5 N.m
Pinion speed [rpm]: 750
Input Power [KW]: 10,41
Torque loss [N.m]: 3,93

MASS

Pinion Standard pattern pinion

Initial mass (average) [g]: 634,414  Initial mass (average) [g]: 528,3894
Finish mass (average) [g]: 634,3998  Finish mass (average) [g]: 528,3898

Variation of the initial mass: 106,0246
Variation of the finish mass: 106,01

MASS LOSS [mg]: 14,6

Notes:
Oil sample was taken after re-launch the test without load.
An emergency stop has occurred (cavitation).
A. Appendix

A.1.6.2. Ferrography results
<table>
<thead>
<tr>
<th>IDENTIFICAÇÃO</th>
<th>T11</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amostra n°.</td>
<td></td>
</tr>
<tr>
<td>Data amostra:</td>
<td>48/15</td>
</tr>
<tr>
<td>Análise n°.</td>
<td></td>
</tr>
<tr>
<td>Ciclos/Máquina:</td>
<td>-</td>
</tr>
<tr>
<td>Ciclos/Óleo:</td>
<td>-</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>FERROMETRIA</th>
</tr>
</thead>
<tbody>
<tr>
<td>D: 0,1</td>
</tr>
<tr>
<td>DL: 14,0</td>
</tr>
<tr>
<td>DS: 2,4</td>
</tr>
<tr>
<td>CPUC: 164,0</td>
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<tr>
<td>ISUC: 1,9E+04</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>FERROGRAFIA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Desgaste normal: M</td>
</tr>
<tr>
<td>Desgaste severo: F</td>
</tr>
<tr>
<td>Desgaste de abrasão: M</td>
</tr>
<tr>
<td>Desgaste combinado: F</td>
</tr>
<tr>
<td>Desgaste de fadiga: M</td>
</tr>
<tr>
<td>Desgaste de adesão:</td>
</tr>
<tr>
<td>Óxidos de fadiga: F</td>
</tr>
<tr>
<td>Minerais/Orgânicos: F</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>OILVIEW</th>
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<tbody>
<tr>
<td>Índice OilLife:</td>
</tr>
<tr>
<td>Índice Oxidação:</td>
</tr>
<tr>
<td>Índice Contaminação:</td>
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<tr>
<td>Índice Ferromagnético:</td>
</tr>
<tr>
<td>Grandes Contaminantes:</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>CONC. PARTICULAS (%):</th>
</tr>
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<tbody>
<tr>
<td>4 - 6 µm</td>
</tr>
<tr>
<td>6 - 10 µm</td>
</tr>
<tr>
<td>10 - 20 µm</td>
</tr>
<tr>
<td>20 - 30 µm</td>
</tr>
<tr>
<td>30 - 40 µm</td>
</tr>
<tr>
<td>40 - 50 µm</td>
</tr>
<tr>
<td>50 - 60 µm</td>
</tr>
<tr>
<td>60 - 70 µm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>VISCOSIDADE (cSt a 40°C):</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACIDEZ (TAN) (mg KOH):</td>
</tr>
<tr>
<td>P. INFLAMAÇÃO (° C):</td>
</tr>
<tr>
<td>DIAGNÓSTICO:</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>LEGENDA</th>
</tr>
</thead>
<tbody>
<tr>
<td>DL - Índice de partículas grandes:</td>
</tr>
<tr>
<td>DS - Índice de partículas pequenas:</td>
</tr>
<tr>
<td>CPUC - Concentração part. de desgaste:</td>
</tr>
<tr>
<td>ISUC - Índice Severidade de Desgaste:</td>
</tr>
<tr>
<td>Não existe</td>
</tr>
<tr>
<td>Fraco</td>
</tr>
<tr>
<td>Médio</td>
</tr>
<tr>
<td>Forte</td>
</tr>
</tbody>
</table>
Ampliação: x 100 Diluição: 0,1
Localização: Núcleo Luz: Branca / Verde
Observações: Presença de poucas partículas ferrosas, que estão alinhadas segundo as forças do campo magnético.

Ampliação: x 200 Diluição: 0,1
Localização: Núcleo Luz: Branca / Verde
Observações: Ampliação da Fotografia 1. Partículas ferrosas de pequenas e médias dimensões.

Ampliação: x 1000 Diluição: 0,1
Localização: Núcleo Luz: Branca / Verde
Observações: Ampliação da Fotografia 1. Partícula ferrosa de grande dimensão, típica de desgaste combinado (fadiga e escorregamento).

Ampliação: x 1000 Diluição: 0,1
Localização: Núcleo Luz: Branca / Verde
Observações: Ampliação da Fotografia 1. Presença de partículas ferrosas de médias e grandes dimensões, típicas de desgaste de fadiga.
Ampliação: x 200  Diluição: 0,1  
Localização: Núcleo  Luz: Branca / Verde  
Observações: Ampliação da Fotografia 1. Presença de partículas ferrosas de médias dimensões, algumas de desgaste de corte (1).

Ampliação: x 1000  Diluição: 0,1  
Localização: Núcleo  Luz: Branca / Verde  
Observações: Ampliação da Fotografia 1. Partículas ferrosas de grandes/médias dimensões, algumas de desgaste combinado (2) e de corte (1).

Ampliação: x 1000  Diluição: 0,1  
Localização: Núcleo  Luz: Branca / Verde  
Ampliação: x 1000  Diluição: 0,1  
Localização: Núcleo   Luz: Branca / Verde

Observações: Ampliação da Fotografia 1.  
Presença de partículas ferrosas de médias e grandes dimensões, típicas de desgaste de fadiga.
A.1.6.3. Roughness profiles
Figure A.59.: Roughness profile of the tooth 1 from the pinion of T11 test

Figure A.60.: Roughness profile of the tooth 9 from the pinion of T11 test

Figure A.61.: Roughness profile of the tooth 1 from the wheel of T11 test

Figure A.62.: Roughness profile of the tooth 9 from the wheel of T11 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.63.: Roughness profile of the tooth 17 from the wheel of T11 test

Table A.9.: Average parameters on each tooth in T11 test

<table>
<thead>
<tr>
<th></th>
<th>P01</th>
<th>P09</th>
<th>W01</th>
<th>W09</th>
<th>W17</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ra max</td>
<td>2.350</td>
<td>3.153</td>
<td>2.262</td>
<td>3.366</td>
<td>3.946</td>
</tr>
<tr>
<td>RZDin</td>
<td>2.070</td>
<td>2.230</td>
<td>1.966</td>
<td>2.285</td>
<td>3.502</td>
</tr>
<tr>
<td>Ra</td>
<td>0.0316</td>
<td>0.0316</td>
<td>0.303</td>
<td>0.313</td>
<td>0.519</td>
</tr>
<tr>
<td>Rq</td>
<td>0.424</td>
<td>0.419</td>
<td>0.401</td>
<td>0.426</td>
<td>0.649</td>
</tr>
<tr>
<td>Rpk</td>
<td>0.485</td>
<td>0.350</td>
<td>0.400</td>
<td>0.228</td>
<td>0.691</td>
</tr>
<tr>
<td>Rk</td>
<td>1.063</td>
<td>1.034</td>
<td>1.012</td>
<td>0.972</td>
<td>1.793</td>
</tr>
<tr>
<td>Rvk</td>
<td>0.686</td>
<td>1.393</td>
<td>0.660</td>
<td>1.398</td>
<td>0.541</td>
</tr>
</tbody>
</table>

Table A.10.: Average parameters of the pinion and wheel in T11 test

<table>
<thead>
<tr>
<th></th>
<th>Pinion</th>
<th>Wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ra max</td>
<td>2.306</td>
<td>4.159</td>
</tr>
<tr>
<td>RZDin</td>
<td>2.018</td>
<td>2.934</td>
</tr>
<tr>
<td>Ra</td>
<td>0.309</td>
<td>0.499</td>
</tr>
<tr>
<td>Rq</td>
<td>0.413</td>
<td>0.635</td>
</tr>
<tr>
<td>Rpk</td>
<td>0.443</td>
<td>0.696</td>
</tr>
<tr>
<td>Rk</td>
<td>1.037</td>
<td>1.650</td>
</tr>
<tr>
<td>Rvk</td>
<td>0.673</td>
<td>0.633</td>
</tr>
</tbody>
</table>
A. Appendix

A.1.6.4. 3D optical microscopy
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.64.: Picture from used head of the pinion tooth 1 of T11 test

Figure A.65.: Picture from used pitch circle of the pinion tooth 1 of T11 test

Figure A.66.: Picture from used head of the wheel tooth 1 of T11 test
A. Appendix

Figure A.67.: Picture from used pitch circle of the wheel tooth 1 of T11 test

Figure A.68.: Topography from used head of the pinion tooth 1 of T11 test

Figure A.69.: Topography from used pitch circle of the pinion tooth 1 of T11 test

Figure A.70.: Topography from used head of the wheel tooth 1 of T11 test

Figure A.71.: Topography from used pitch circle of the wheel tooth 1 of T11 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

A.1.7. T12 test

A.1.7.1. Test sheet
Faculty of Engineering of the University of Porto (FEUP)
Mild wear in lubricated gear transmissions

Operator: Pedro Aires da Cunha Cerqueira

---

**TEST DATA SHEET**

Test Ref.: T12  
Start Date: 02/04/2015

Gear Ref.: 3703  
Finish Date: 07/04/2015

Side: A  
Material: 16MnCr5 case-carburized

---

**TEST FEATURES**

Oil: Renolin Unisyn CLP 150

Temperature[^C]: 100

\( \Delta T(T_{G-Tenviron})[^{\circ}C] \): 39,7  
\( \Delta T(T_{oil-Tenviron})[^{\circ}C] \): 74,3

Pinion cycles: 5,000,000

Days: 5

Lubrication type: Dip lubrication

Wheel torque: K5 - 70 N.m

Pinion speed [rpm]: 750

Input Power [KW]: 5,5

Torque loss [N.m]: 3,29

---

**MASS**

### Pinion

<table>
<thead>
<tr>
<th>Initial mass (average) [g]:</th>
<th>633,9942</th>
</tr>
</thead>
<tbody>
<tr>
<td>Finish mass (average) [g]:</td>
<td>633,9834</td>
</tr>
</tbody>
</table>

Initial mass (average) [g]: 528,3882

Finish mass (average) [g]: 528,3888

Variation of the initial mass: 105,606

Variation of the finish mass: 105,5946

MASS LOSS [mg]: 11,4

---

Notes:
A.1.7.2. Roughness profiles
Figure A.72.: Roughness profile of the tooth 1 from the pinion of T12 test

Figure A.73.: Roughness profile of the tooth 9 from the pinion of T12 test

Figure A.74.: Roughness profile of the tooth 1 from the wheel of T12 test

Figure A.75.: Roughness profile of the tooth 9 from the wheel of T12 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.76.: Roughness profile of the tooth 17 from the wheel of T12 test

Table A.11.: Average parameters on each tooth in T12 test

<table>
<thead>
<tr>
<th></th>
<th>P01</th>
<th>P09</th>
<th>W01</th>
<th>W09</th>
<th>W17</th>
</tr>
</thead>
<tbody>
<tr>
<td>RZDin</td>
<td>3.094</td>
<td>2.413</td>
<td>3.701</td>
<td>2.334</td>
<td>2.684</td>
</tr>
<tr>
<td>Ra</td>
<td>0.571</td>
<td>0.515</td>
<td>0.339</td>
<td>0.385</td>
<td>0.335</td>
</tr>
<tr>
<td>Rq</td>
<td>0.748</td>
<td>0.486</td>
<td>0.457</td>
<td>0.499</td>
<td>0.503</td>
</tr>
<tr>
<td>Rpk</td>
<td>0.775</td>
<td>0.486</td>
<td>0.761</td>
<td>0.322</td>
<td>0.610</td>
</tr>
<tr>
<td>Rk</td>
<td>1.788</td>
<td>1.138</td>
<td>1.474</td>
<td>1.119</td>
<td>1.253</td>
</tr>
<tr>
<td>Rvk</td>
<td>0.887</td>
<td>0.901</td>
<td>1.251</td>
<td>1.489</td>
<td>0.534</td>
</tr>
</tbody>
</table>

Table A.12.: Average parameters of the pinion and wheel in T12 test

<table>
<thead>
<tr>
<th></th>
<th>Pinion</th>
<th>Wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ra max</td>
<td>4.138</td>
<td>2.916</td>
</tr>
<tr>
<td>RZDin</td>
<td>3.398</td>
<td>2.373</td>
</tr>
<tr>
<td>Ra</td>
<td>0.543</td>
<td>0.347</td>
</tr>
<tr>
<td>Rq</td>
<td>0.732</td>
<td>0.473</td>
</tr>
<tr>
<td>Rpk</td>
<td>0.768</td>
<td>0.404</td>
</tr>
<tr>
<td>Rk</td>
<td>1.631</td>
<td>1.128</td>
</tr>
<tr>
<td>Rvk</td>
<td>1.069</td>
<td>1.195</td>
</tr>
</tbody>
</table>
A. Appendix

A.1.7.3. 3D optical microscopy
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.77.: Picture from new head of the pinion tooth 1 of T12 test

Figure A.78.: Picture from new pitch circle of the pinion tooth 1 of T12 test

Figure A.79.: Picture from new head of the wheel tooth 1 of T12 test
A. Appendix

Figure A.80.: Picture from new pitch circle of the wheel tooth 1 of T12 test

Figure A.81.: Picture from used head of the pinion tooth 1 of T12 test.

Figure A.82.: Picture from used pitch circle of the pinion tooth 1 of T12 test
A.1. Tests, ferrography results, roughness profiles and tests pictures

Figure A.83.: Picture from used head of the wheel tooth 1 of T12 test

Figure A.84.: Picture from used pitch circle of the wheel tooth 1 of T12 test

Figure A.85.: Topography from new head of the pinion tooth 1 of T12 test

Figure A.86.: Topography from new pitch circle of the pinion tooth 1 of T12 test
A. Appendix

Figure A.87.: Topography from new head of the wheel tooth 1 of T12 test

Figure A.88.: Topography from new pitch circle of the wheel tooth 1 of T12 test

Figure A.89.: Topography from used head of the pinion tooth 1 of T12 test

Figure A.90.: Topography from used pitch circle of the pinion tooth 1 of T12 test

Figure A.91.: Topography from used head of the wheel tooth 1 of T12 test

Figure A.92.: Topography from used pitch circle of the wheel tooth 1 of T12 test
A.2. Lubricants
RENOLIN UNISYN CLP
Fully-synthetic industrial gear lubricants based on polyalphaolefins

Description
Demulsifying, fully-synthetic industrial gear oils with elevated aging resistance, excellent load-carrying capacity and wear protection. RENOLIN UNISYN CLP oils have good resistance to micropitting. Reliable lubrication of roller bearings is confirmed by the good results of the FE8 testing. The products are preferably used when increased requirements are set for high and low temperature usage limits. In gearboxes and circulating systems with sump temperatures up to 90°C, longer oil-change intervals in comparison with previous mineral oils are achieved. Miscibility with gearbox oils based on mineral oil is generally given, which means that simplified conversion is possible.

Application
The oils of the RENOLIN UNISYN CLP series are used for all applications in industry where a synthetic oil of the CLP type according to DIN 51 517-3 is recommended by the manufacturer. Highly-stressed bearings, joints, pressure screws, spur gears and worm gears can be reliably, safely and economically supplied even at short-term peak temperatures up to 150°C.

Specifications
The products meet and in many cases exceed the requirements of:
- DIN 51 517-3: CLP
- ISO 6743-6: CKD
- AISE 224
- FAG requirements: FAG-FE8-Test: stage 1-4 pass (test report is available for ISO VG 320)
- SKF requirements: pass (100°C-test)

The RENOLIN UNISYN CLP series are approved for:

Advantages

- Low foaming
- Good air release capacity
- Very good aging resistance
- Excellent corrosion protection
- Excellent viscosity-temperature behavior
- High natural VI (viscosity index)
- Multigrade character
- Excellent wear protection, high EP performance
- Miscible with mineral oil- and ester-based gear oils
- Lifetime lubrication possible
- For high and low operating temperatures

While the information and figures given here are typical of current production and confirm to specification, minor variations may occur. No warranty expressed or implied is given concerning the accuracy of the information or the suitability of the products.
RENOLIN UNISYN CLP
Fully-synthetic industrial gear lubricants based on polyalphaolefins

Typical Technical Data:

<table>
<thead>
<tr>
<th>Product name</th>
<th>Properties</th>
<th>Unit</th>
<th>68</th>
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<th>Test Method</th>
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<tr>
<td></td>
<td>ISO VG</td>
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<td>68</td>
<td>100</td>
<td>150</td>
<td>DIN 51 519</td>
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<td>Kinematic viscosity at 40 °C</td>
<td>mm²/s</td>
<td>68</td>
<td>10</td>
<td>14.4</td>
<td>DIN EN ISO 3104</td>
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<tr>
<td></td>
<td>Kinematic viscosity at 100 °C</td>
<td>mm²/s</td>
<td>10.8</td>
<td>14.4</td>
<td>19.4</td>
<td>DIN ISO 2909</td>
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<tr>
<td></td>
<td>Viscosity index</td>
<td>-</td>
<td>149</td>
<td>148</td>
<td>148</td>
<td>DIN ISO 2909</td>
</tr>
<tr>
<td></td>
<td>Density at 15°C</td>
<td>kg/m³</td>
<td>843</td>
<td>845</td>
<td>849</td>
<td>DIN 51 757</td>
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<tr>
<td></td>
<td>Color index</td>
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<td>0.5</td>
<td>0.5</td>
<td>DIN ISO 2049</td>
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<tr>
<td></td>
<td>Flashpoint, Cleveland open cup</td>
<td>°C</td>
<td>240</td>
<td>250</td>
<td>250</td>
<td>DIN ISO 2592</td>
</tr>
<tr>
<td></td>
<td>Pour point</td>
<td>°C</td>
<td>&lt; -60</td>
<td>&lt; -60</td>
<td>&lt; -57</td>
<td>DIN ISO 3016</td>
</tr>
<tr>
<td></td>
<td>Neutralization number</td>
<td>mgKOH/g</td>
<td>0.6</td>
<td>0.6</td>
<td>0.6</td>
<td>DIN 51 558-1</td>
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<tr>
<td></td>
<td>Scuffing and scoring test, FZG A/8.3/90</td>
<td>Failure load stage</td>
<td>&gt; 12</td>
<td>&gt; 12</td>
<td>&gt; 12</td>
<td>DIN ISO 14635-1</td>
</tr>
<tr>
<td></td>
<td>Scuffing and scoring test, FZG A/16.6/140</td>
<td>Failure load stage</td>
<td>12</td>
<td>12</td>
<td>12</td>
<td>DIN ISO 14635-1</td>
</tr>
<tr>
<td></td>
<td>Micropitting test, FZG-GFT Test GT-C/8.3/90</td>
<td>Loadstage test</td>
<td>GFT high</td>
<td>GFT high</td>
<td>GFT high</td>
<td>FVA Information Sheet no. 54/I-IV</td>
</tr>
<tr>
<td></td>
<td>Micropitting test, FZG-GFT Test GT-C/8.3/90</td>
<td>Endurance test</td>
<td>GFT high</td>
<td>GFT high</td>
<td>GFT high</td>
<td>FVA Information Sheet no. 54/I-IV</td>
</tr>
<tr>
<td></td>
<td>FE-8 roller bearing test, 7,5/80/80 and 7,5/100/80</td>
<td></td>
<td>-</td>
<td>pass (excellent)</td>
<td></td>
<td>DIN 51 819-3</td>
</tr>
</tbody>
</table>

While the information and figures given here are typical of current production and confirm to specification, minor variations may occur. No warranty expressed or implied is given concerning the accuracy of the information or the suitability of the products.
RENOLIN UNISYN CLP
Fully-synthetic industrial gear lubricants based on polyalphaolefins

Typical Technical Data:

<table>
<thead>
<tr>
<th>Properties</th>
<th>Unit</th>
<th>Test Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>ISO VG</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Kinematic viscosity at 40 °C</td>
<td>mm²/s</td>
<td>DIN EN ISO 3104</td>
</tr>
<tr>
<td>Kinematic viscosity at 100 °C</td>
<td>mm²/s</td>
<td></td>
</tr>
<tr>
<td>Viscosity index</td>
<td></td>
<td>DIN ISO 2909</td>
</tr>
<tr>
<td>Density at 15°C</td>
<td>kg/m³</td>
<td>DIN ISO 2592</td>
</tr>
<tr>
<td>Color index</td>
<td>ASTM</td>
<td></td>
</tr>
<tr>
<td>Flashpoint, Cleveland open cup</td>
<td>°C</td>
<td>DIN ISO 2592</td>
</tr>
<tr>
<td>Pour point</td>
<td>°C</td>
<td>DIN ISO 3016</td>
</tr>
<tr>
<td>Neutralization number</td>
<td>mgKOH/g</td>
<td>DIN 51 558-1</td>
</tr>
<tr>
<td>Scuffing and scoring test, FZG A/8,3/90</td>
<td>Failure load stage</td>
<td>DIN ISO 14635-1</td>
</tr>
<tr>
<td>Scuffing and scoring test, FZG A/16,6/140</td>
<td>Failure load stage</td>
<td>DIN ISO 14635-1</td>
</tr>
<tr>
<td>Micropitting test, FZG-GFT Test GT-C/8,3/90</td>
<td>GF Class</td>
<td></td>
</tr>
<tr>
<td>Micropitting test, FZG-GFT Test GT-C/8,3/90</td>
<td>GF Class</td>
<td></td>
</tr>
<tr>
<td>FE-8 roller bearing test, 7,5/80/80 and 7,5/100/80</td>
<td>-</td>
<td>DIN 51 819-3</td>
</tr>
</tbody>
</table>

While the information and figures given here are typical of current production and confirm to specification, minor variations may occur. No warranty expressed or implied is given concerning the accuracy of the information or the suitability of the products.
BP Energol GR-XP 150
Industrial Extreme Pressure Gear Oil

Description
The BP Energol™ GR-XP gear oil range of high quality lubricants are based upon highly refined mineral oil, enhanced with sulphur/phosphorus extreme pressure additive technology providing outstanding thermal stability and high load carrying capacity.

The advanced extreme pressure additive system not only provides high load carrying capacity, but was designed to provide microscopic wear protection. Microscopic wear protection, also known as micropitting protection, is critical in preventing destructive wear at the micro level therefore extending gear life and meeting the evolving demands of smaller and higher output gear boxes.

Application
The Energol GR-XP range is recommended for the lubrication of industrial gearboxes using forced circulation or splash and oil bath lubrication. They may be used for the lubrication of spur and helical gears and in some lightly loaded worm type gear applications.

They have very good viscosity characteristics to ensure that starting torques are not excessively high in cold operating conditions. The additives are compatible with the ferrous and non-ferrous metals used in industrial gear units.

The Energol GR-XP range is fully compatible with nitrile, silicone and fluropolymer seal materials.

Energol GR-XP is classified as follows:
DIN Classification is CLP

Energol GR-XP grades meet the requirements of:
DIN 51517 Part 3
AGMA 9005 - D94
US Steel 224
David Brown Type E
Hansen Transmissions
Flender

Advantages
• ‘Clean gear’ additive technology ensures low deposit formation and enhanced filter life.
• Full Extreme Pressure (EP) performance* gives maximum protection of gears against wear and shock-loading.
• Good water separation and demulsification characteristics means reduced down time through prolonged lubricant life and increased equipment reliability.
• Excellent protection against corrosion and wear results in less maintenance.
• Suitable for Müller Weingarten equipment

* ISO 220 grade achieved FZG >14 rating under A16.6/90 (double speed) test conditions
### Typical Characteristics

<table>
<thead>
<tr>
<th>Test</th>
<th>Method</th>
<th>Units</th>
<th>68</th>
<th>100</th>
<th>150</th>
<th>220</th>
<th>320</th>
<th>460</th>
<th>680</th>
<th>1000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density @ 15°C</td>
<td>ISO 12185 / ASTM D4052</td>
<td>g/ml</td>
<td>0.88</td>
<td>0.89</td>
<td>0.89</td>
<td>0.9</td>
<td>0.9</td>
<td>0.92</td>
<td>0.93</td>
<td></td>
</tr>
<tr>
<td>K.V. @ 40°C</td>
<td>ISO 3104 / ASTM D445</td>
<td>mm²/s</td>
<td>88</td>
<td>100</td>
<td>150</td>
<td>220</td>
<td>320</td>
<td>460</td>
<td>680</td>
<td>1000</td>
</tr>
<tr>
<td>K.V. @ 100°C</td>
<td>ISO 3104 / ASTM D445</td>
<td>mm²/s</td>
<td>0.53</td>
<td>11.1</td>
<td>14.6</td>
<td>19.7</td>
<td>24</td>
<td>30.5</td>
<td>37.3</td>
<td>42.6</td>
</tr>
<tr>
<td>Viscosity Index</td>
<td>ISO 2099 / ASTM D2280</td>
<td>cP</td>
<td>-</td>
<td>&gt; 95</td>
<td>&gt; 95</td>
<td>&gt; 95</td>
<td>&gt; 95</td>
<td>&gt; 95</td>
<td>&gt; 95</td>
<td>62</td>
</tr>
<tr>
<td>Pour Point</td>
<td>ISO 3016 / ASTM D97</td>
<td>°C</td>
<td>-21</td>
<td>-21</td>
<td>-18</td>
<td>-18</td>
<td>-15</td>
<td>-12</td>
<td>-9</td>
<td>-3</td>
</tr>
<tr>
<td>Flash Point, PM20</td>
<td>ISO 2719 / ASTM D93</td>
<td>°C</td>
<td>220</td>
<td>220</td>
<td>220</td>
<td>228</td>
<td>226</td>
<td>230</td>
<td>230</td>
<td></td>
</tr>
<tr>
<td>Foam Seq I</td>
<td>ISO 6247 / ASTM D692</td>
<td>mls/ml</td>
<td>10/0</td>
<td>10/0</td>
<td>10/0</td>
<td>10/0</td>
<td>10/0</td>
<td>10/0</td>
<td>10/0</td>
<td>10/0</td>
</tr>
<tr>
<td>Copper Corrosion (3 hrs @ 100°C)</td>
<td>ISO 2163 / ASTM D130</td>
<td>-</td>
<td>1b</td>
<td>1b</td>
<td>1b</td>
<td>1b</td>
<td>1b</td>
<td>1b</td>
<td>1b</td>
<td>1b</td>
</tr>
<tr>
<td>Tinplate OK Load</td>
<td>ASTM D250 / IP 240</td>
<td>lbs</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
</tr>
<tr>
<td>FZG fail stage (A8 3/90)</td>
<td>ISO 1625-1 / DIN 51354</td>
<td>-</td>
<td>&gt; 12</td>
<td>&gt; 12</td>
<td>&gt; 12</td>
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<td>&gt; 12</td>
<td>&gt; 12</td>
<td>&gt; 12</td>
<td>&gt; 12</td>
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<tr>
<td>FZG Micropitting</td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fail Stage</td>
<td>FVA Proj No. 54</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>&gt;10</td>
<td>&gt;10</td>
<td>&gt;10</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>GFT Class</td>
<td>FVA Proj No. 54</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>High</td>
<td>High</td>
<td>High</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
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</table>

Subject to usual manufacturing tolerances.

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BP Energol GR-XP 150
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A.3. FZG technical characteristic

The technical characteristic of the sensor are indicated in table A.13

Table A.13.: Technical specifications of the ETH DRDL torque cell

| Torque transducer Type DRDL-II
| Nominal torque [N.m] | 50 |
| Measurement range [N.m] | out/50 |
| Non-linearity [%] | <0.1 |
| Hysteresis [%] | <0.1 |
| Accuracy [%] | 0.01 |
| Temperature sensitivity [%/K] | 0.01 |

Torque measuring module type ValueMaster Base
- Accuracy [%] 0.02
- Non-linearity [%] 0.1
- AcD converter resolution 11 bit+1 bit for leading sign

The bearings properties may be found in the table A.14.

Table A.14.: NJ 406 MA and QJ 308 N2MA bearings properties

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Four point contact ball bearing</td>
<td>QJ 308 N2MA</td>
<td>78</td>
<td>64</td>
<td>90</td>
<td>40</td>
<td>12</td>
<td>14000</td>
</tr>
<tr>
<td>Single row cylindrical roller bearing</td>
<td>NJ 406 MA</td>
<td>34.1</td>
<td>34</td>
<td>90</td>
<td>30</td>
<td>23</td>
<td>11000</td>
</tr>
</tbody>
</table>

Table A.15 show the seals characteristics.

Table A.15.: Gearboxes seals characteristics

<table>
<thead>
<tr>
<th>Seal type</th>
<th>Reference</th>
<th>Lip material</th>
<th>d [mm]</th>
<th>D[m]</th>
<th>b [m]</th>
<th>max speed [rpm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial shaft seals</td>
<td>30X47X7 HMS5 V</td>
<td>V</td>
<td>30</td>
<td>47</td>
<td>7</td>
<td>8913</td>
</tr>
<tr>
<td>Radial shaft seals</td>
<td>26X47X7 HMSA10 V</td>
<td>V</td>
<td>26</td>
<td>47</td>
<td>7</td>
<td>7346</td>
</tr>
</tbody>
</table>

The torque value, for each load stage, may be calculated using equation A.1 [16].

\[ T_1 = T_H + (F_K + F_W) \cdot h \]  \hspace{1cm} (A.1)

Where:
- \( T_1 \) [N.m] - torque on pinion;
- \( T_H \) [N.m] - torque of the lever, two levers are available with 3.3 N.m \( (K_1) \) and 13.7 N.m \( (N_2) \);
- \( F_K \) [N] - weight of the load support rod which is 43.1 N for available rod;
- \( F_W \) [N] - weight of the loading weights see table A.16;
- \( h \) [Em] - effort arm for \( F_K \) and \( F_W \) with grooves in the load lever.
Table A.16.: Calibrated weights used for composing the FZG load stages

<table>
<thead>
<tr>
<th>No</th>
<th>Weights [Kgf]</th>
<th>Weights [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5.2</td>
<td>51.0</td>
</tr>
<tr>
<td>2</td>
<td>6.8</td>
<td>66.7</td>
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<tr>
<td>3</td>
<td>8.4</td>
<td>82.4</td>
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<tr>
<td>4</td>
<td>9.8</td>
<td>96.1</td>
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<td>5</td>
<td>11.4</td>
<td>111.8</td>
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<td>6</td>
<td>12.8</td>
<td>125.6</td>
</tr>
</tbody>
</table>

A.4. 3D optical microscopy features

Table A.17 displays the setting used for topography measurements.

Table A.17.: Settings used for topography measurements

<table>
<thead>
<tr>
<th>Processing method</th>
<th>VSI</th>
</tr>
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<tbody>
<tr>
<td>Speed</td>
<td>1x</td>
</tr>
<tr>
<td>Lens</td>
<td>10× LINNIK</td>
</tr>
<tr>
<td>Backscan</td>
<td>30µm</td>
</tr>
<tr>
<td>Length</td>
<td>30µm</td>
</tr>
<tr>
<td>Threshold</td>
<td>4</td>
</tr>
<tr>
<td>Illumination</td>
<td>white</td>
</tr>
<tr>
<td>Stitch enabled</td>
<td>1.5 × 14µm</td>
</tr>
<tr>
<td>Overlap area</td>
<td>20%</td>
</tr>
<tr>
<td>Software</td>
<td>Vision64 and HommelMap Expert</td>
</tr>
</tbody>
</table>

A.5. Microscopic objective

Table A.18 shows the three used and the total magnification

Table A.18.: Three used microscopic objective

<table>
<thead>
<tr>
<th>Lens identification</th>
<th>Lens magnification</th>
<th>Total magnification</th>
<th>Real size of each division rule</th>
</tr>
</thead>
<tbody>
<tr>
<td>MA 10</td>
<td>10x</td>
<td>100x</td>
<td>10</td>
</tr>
<tr>
<td>MA 50</td>
<td>50x</td>
<td>500x</td>
<td>2</td>
</tr>
<tr>
<td>MD Plan 80</td>
<td>80x</td>
<td>800x</td>
<td>1.5</td>
</tr>
</tbody>
</table>

A.6. Roughness profiles details

To determine the position of the vertical red lines in graphs:
A.6. Roughness profiles details

With equations A.2, A.3 and A.4 and the table A.19 it was possible to build the table A.20 that contains coordinate values of the points in the roughness profiles.

\[
\Delta_{\text{pinion}} = \frac{T_1 P^2}{d_b_1} \quad (A.2)
\]

\[
\Delta_{\text{wheel}} = \frac{(T_1 T_2 - T_1 P)^2}{d_b_2} \quad (A.3)
\]

\[
d_{b_x} = \text{number of teeth} \ast 4.5 \ast \cos 20 \quad (A.4)
\]

**Table A.19.: Constants**

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_1 A$</td>
<td>4.29</td>
</tr>
<tr>
<td>$T_1 B$</td>
<td>10.44</td>
</tr>
<tr>
<td>$T_1 C$</td>
<td>13.97</td>
</tr>
<tr>
<td>$T_1 D$</td>
<td>17.58</td>
</tr>
<tr>
<td>$T_1 E$</td>
<td>23.722</td>
</tr>
<tr>
<td>$T_1 T_2$</td>
<td>34.94</td>
</tr>
<tr>
<td>$d_b_1$</td>
<td>67.66</td>
</tr>
<tr>
<td>$d_b_2$</td>
<td>101.5</td>
</tr>
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</table>

**Table A.20.: Coordinate values of the points in the roughness profiles**

<table>
<thead>
<tr>
<th></th>
<th>Pinion [mm]</th>
<th>Wheel [mm]</th>
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</thead>
<tbody>
<tr>
<td>A</td>
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