



Caracterização mecânica e tribológica de materiais poliméricos para engrenagens moldados por injeção

GONÇALO VINHAS LOPES

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**Mechanical and tribological
characterisation of injection moulded
polymer gear materials**

Gonalo Vinhas Lopes

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Supervisor: Ramiro Carneiro Martins

Co-supervisor: Carlos Miguel da Costa Gomes Fernandes

Jury:

President:

Alexandre Goncalves Santos Santiago Sottomayor, Professo Adjunto, ISEP

Vowels:

David Emanuel Pimentel Gonalves, Main Bearing Lead Designer, Vestas

Carlos Miguel da Costa Gomes Fernandes, Professor Auxiliar, FEUP

Ramiro Carneiro Martins, Professor Adjunto, ISEP

Porto, July 11, 2024

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ABSTRACT

The application of polymer gears is constantly growing due to their advantages over metal gears.

This growth allows for diversification in gear usage, enabling the selection of the best combination for specific applications. The main advantages include reduced mass, cost savings, and the ability to operate without lubrication.

Previous research has shown that hybrid polymer gears can also be a viable solution to some of these issues. Nonetheless, polymer gears still face challenges such as service temperature, load capacity, and wear during gear. For these reasons, understanding the mechanical and thermal properties of potential constituent materials is crucial to expanding their range of applications.

This thesis provides a theoretical review of polymer gears, highlighting mechanisms to enhance their lifespan. The study also covers procedures for characterising the materials, including mechanical and thermal properties. Numerical simulations were conducted and compared with experimental results from Pin-on-Disc tests, with contact temperature recorded during the experiments.

The data were obtained and analyzed using various tools, including *MountainsLab 10*, *Minitab*, *PrePoMax*, *KISSsoft* and *ThermaCAM Researcher*.

The results indicate that the mould temperature during injection influences mechanical, thermal, and tribological properties. Furthermore, it was concluded that PA6-GF, despite its high rigidity, does not exhibit favourable tribological properties, resulting in a higher wear coefficient compared to POM.

KEYWORDS

DMA; DSC; Gear; Polymer; Roughness; Wear

RESUMO

A aplicação das engrenagens poliméricas está em constante e acelerado crescimento devido às suas vantagens em relação às engrenagens metálicas.

Este crescimento permitirá diversificar o uso das engrenagens, permitindo obter a melhor combinação para a aplicação desejada. As principais vantagens serão a diminuição da massa, o custo e ainda a possibilidade de poderem trabalhar sem lubrificação.

Com base em trabalhos anteriores conseguiu-se perceber que as engrenagens híbridas poliméricas também podem ser uma solução viável para alguns destes problemas. Porém as engrenagens poliméricas ainda apresentam alguns problemas como o trabalho a altas temperatura, a capacidade de carga e o desgaste aquando do engrenamento. Por estes motivos procura-se entender quais as propriedades mecânicas e térmicas dos materiais que as podem vir a constituir, com vista a melhorar a sua gama de aplicações.

Esta tese apresenta uma revisão teórica das engrenagens poliméricas, com a divulgação de mecanismos para aumentar a sua vida útil. Também foram abordados os procedimentos para a caracterização dos materiais em estudo, entre propriedade mecânicas e térmicas. Foram realizadas simulações numéricas e comparadas com os resultados experimentais executados durante o ensaio Pino-Disco. A temperatura do contacto foi registada durante o ensaio.

Os dados foram obtidos e analisados com diversas ferramentas como *Mountains*, *Minitab*, *PrePoMax*, *KISSsoft* e *ThermaCAM Researcher*.

Os resultados obtidos revelam que a temperatura de molde durante injeção influencia as propriedades mecânicas, térmicas e tribológicas. Concluiu-se ainda que o PA6-GF, apesar de elevada rigidez, não apresenta boas propriedades tribológicas, resultando num coeficiente de desgaste superior ao do POM.

PALAVRAS-CHAVE

Desgaste; DMA; DSC; Engrenagem; Polimérica; Rugosidade

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LIST OF ABBREVIATIONS AND SYMBOLS

Abbrev.	Description
DOE	Design of Experiments
DSC	Differential Scanning Calorimetry
DMA	Dynamic Mechanical Analysis
FEUP	Faculty of Engineering of University of Porto
FEM	Finite Element Method
GFR	Glass-fiber-reinforced
HPDE	High density polyethylene
PA	Polyamide
PC	Polycarbonate
PEEK	Polyetheretherketone
POM	Polyoxymethylene
PTFE	Polytetrafluoroethylene
FZG	Research Test Rig for Gears and Transmissions (Forcshug- stelle für Zahnräder und Getriebebau, in german)
ASTM	American Society for Testing and Materials
VDI	Association of German Engineers
TCR	Thermal Contact Resistance
TTS	Time-Temperature Superposition
WLF	Williams, Landel and Ferry equation

Symbol	Description	Units
A	Form factor	–
A _G	Heat-dissipation surface of the mechanism housing	m ²
A _a	Apparent contact area	m ²
A _r	Real contact area	m ²
E	Modulus of elasticity	GPa
E _a	Activation energy	J
E'	Storage modulus	MPa
E''	Loss modulus	MPa
E*	Complex modulus	MPa
ED	Relative tooth-engagement time	–
F _t	Nominal tangential load	N
F _{dyn}	Dynamic force	N
H	Hardness	MPa
H _m	Melting enthalpy	J · kg ⁻¹
H _{m0}	Melting enthalpy for fully crystalline sample	J · kg ⁻¹
H _V	Degree of tooth loss	–
H _{VL}	Local gear loss factor	–
H _{poly}	Polymer elastic contact hardness	Pa
K _F	Tooth root load factor	–
K _H	Tooth flank load factor	–
L	Sliding distance	mm
M	Transmitted torque	Nm
P	Normal load	N
P _{IN}	Input power	W
P _N	Nominal output	W
P _t	Total profile depth	μm
P _V	Power loss	W
P _{VD}	Seals power loss	W
P _{VL}	Rolling bearing power loss	W
P _{VX}	Auxiliary power loss	W
P _{VZ0}	Load independent losses	W
P _{VZP}	Load dependent losses	W
Q	Heat flux	W · m ⁻²
R	Universal gas constant	J · K ⁻¹ · mol ⁻¹
R _a	Average roughness	μm
R _k	Core roughness depth	μm
R _{max}	Maximum peak-to-valley height	μm
R _{pk}	Reduced peak height	μm
R _q	Root mean square roughness average	μm
R _{si}	Thermal contact resistance of a contact spot	K · m ² · W ⁻¹
R _{tc}	Thermal contact resistance	K · m ² · W ⁻¹

Symbol	Description	Units
R_{vk}	Reduced valley depth	μm
R_Z	Mean peak-to-valley height	μm
$R_{\lambda,G}$	Heat transfer resistance of the mechanism	$\text{K} \cdot \text{m}^2 \cdot \text{W}^{-1}$
S_{ku}	Kurtosis	—
S_{sk}	Skewness	—
T_f	Temperature	$^{\circ}\text{C}$
T_f	Flash temperature	K
T_g	Glass transition temperature	$^{\circ}\text{C}$
T_{ref}	Reference temperature	$^{\circ}\text{C}$
W	Wear volume	mm^3
Y_{Fa}	Form factor	—
Y_{Sa}	Stress correction factor	—
Y_{β}	Helix angle factor	—
Y_{ϵ}	Contact ratio factor	—
Z_E	Elasticity factor	—
Z_H	Zone factor	—
Z_{β}	Spiral angle factor	—
Z_{ϵ}	Contact ratio factor	—
a_i	Distributed contact spot radius	m
b	Tooth face width	mm
b_i	Distributed flux tube radius	m
c	Specific heat	$\text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$
c'	Specific heat per unit volume	$\text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1} \cdot \text{m}^{-3}$
d	Reference circle diameter	mm
d_a	Addendum circle diameter	mm
d_b	Base circle diameter	mm
d_f	Root circle diameter	mm
f_{load}	Loading frequency	Hz
h_{aP}	Addendum of the reference profile	mm
h_c	contact heat transfer coefficient	$\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$
h_{fP}	Dedendum of the reference profile	mm
h_g	Gap heat transfer coefficient	$\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$
h_i	Wear depth in the i-th cycle	mm
h_{rad}	Radiation heat transfer coefficient	$\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$
h_{tc}	Thermal contact heat transfer coefficient	$\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$
k	Thermal conductivity	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
k_a	Archard wear coefficient	—
k_s	Effective thermal conductivity	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
k_W	Wear coefficient	$\text{mm}^3 \cdot \text{N}^{-1} \cdot \text{m}^{-1}$
$k_{\vartheta,Fla}$	Heat transfer coefficient of the plastic gear	$\text{K} (\text{ms}^{-1})^{0.75} \text{mm}^{1.75} \text{W}^{-1}$
m_n	Normal module	mm

Symbol	Description	Units
m_s	Effective absolute asperity slope	–
n	Angular speed	rpm
p	Contact pressure	Pa
q_{av}	Average heat flux	$\text{W} \cdot \text{m}^{-2}$
r	Reference radius	mm
r_a	Outside radius	mm
r	Reference radius	mm
t_{step}	Time step	s
u	Gear ratio	–
v_s	Sliding velocity	
v_t	Tangential velocity	$\text{m} \cdot \text{s}^{-1}$
x	Profile shift coefficient	–
z	Tooth number	–
α_n	Normal pressure angle	°
β	Helix angle	°
χ_c	Crystallinity fraction	–
ϵ	Ratio between the contact area and apparent contact area	–
ϵ_α	Transverse contact ratio	–
ϵ_β	Overlap contact ratio	–
μ	Coefficient of friction	–
μ_{mZ}	Average coefficient of friction	–
ψ_{ci}	Contact spot constriction radius	m
ρ	Density	$\text{g} \cdot \text{cm}^{-3}$
ρ_{fP}	Fillet on the reference profile	mm
σ_F	Tooth root stress	Pa
σ_H	Contact stress	Pa
σ_s	Effective root-mean-square roughness	μm
θ	Circumference angle	rad
θ_b	Bulk temperature	K
ζ	Loss degree	%
ϑ_0	Ambient temperature	°C
ϑ_{Fla}	Flank temperature characteristic	°C

1 INTRODUCTION

1.1 Thesis purpose

The purpose of a gear is to increase or reduce the speed through teeth contact. This procedure needs to take in to count some aspects like the load capacity, the contact pressure, wear and heat transfer.

The polymer gears are used in many applications such as: printer, clocks, home appliances and in the food industry. This happen because, when compared with steel gears, they have various advantages like lightweight, damping, self lubrication and low cost [1, 2]. Despite these advantages, certain issues can lead to tooth fracture, such as lower wear resistance and elevated temperature on gear surfaces [3, 4].

Analysing previous works about hybrid polymer gears is possible to understand how good the insert addition can be for heat evacuation and consequently load capacity increasing [5]. Moreover this changes can lead to an improvement on gear's life-time, resulting in lower maintenance costs.

The primary objective of this thesis was to register and analyse the response of a hybrid polymer gear under diverse operating conditions, including load capacity, working temperature and wear. By achieving this goal, we aimed to identify critical parameters that warrant attention for future development.

Unfortunately, due to unforeseen delays in the readiness of the test bench, adjustments had to be made to the original objectives outlined in this dissertation.

The investigation focused on a gear previously manufactured, with two distinct materials. Notably, one of these was injected with three different mould temperatures. The objective is to understand whether this property significantly influenced mechanical properties and to compare he performance of the two materials using various analytical methodologies.

A Dynamic Mechanical Analysis (DMA) and a Shore D hardness were conducted to analyse the mechanical properties, meanwhile a Differential Scanning Calorimetry (DSC) was employed to study the thermal behaviour of the gear.

Additionally, these gears were evaluated on their tribological properties, conducting a Pin-on-Disc test to measure the wear rate for different materials and mould temperatures.

All this work needs to be accompanied by a Finite Element Method (FEM) model with *PrePoMax* software to validate the experimental results and to understand the evolution of heat dissipation in the tooth.

1.2 Tasks chronogram

Table 1 shows the guide line followed to complete purposed tasks. This are approximated periods until the conclusion of each task.

Table 1 – Approximate weeks spent on each task.

Task	Duration [weeks]
Select and read the available bibliography	≈ 7
Write the state of the art	≈ 7
Improve the state of art	≈ 3
Measure the roughness profile	≈ 1
Design testing components	≈ 1
Define the tests available to execute	≈ 1
Pin-on-disc tests	≈ 2
Measure the mass loss and roughness profile	≈ 1
Assist and learn the DMA and DSC tests	≈ 1
Data processing	≈ 2
Write the results in the thesis	≈ 2

1.3 Thesis outline

The chapter 2, explain how polymer gears work, their advantages and disadvantages when compared to conventional gears, types of failure and the impact of lubrication in their life time.

The chapter 3 demonstrates which gears are used in this thesis and outline the various processes involved in obtaining specimens for all executed tests. Specifically, this topic demonstrate how values such as hardness, crystallinity, elastic modulus, roughness, wear are explored through DMA, DSC and Shore D hardness tests.

The chapter 4 details the experimental procedures undertaken to achieve the desired results concerning these gears, depending on their moulding temperature. Additionally, presents an analysis of roughness, mass loss before and after testing, and thermal behaviour during the experiments.

The chapter 5 provides wear predictions using numerical calculations, employing the *PrePoMax* software. To conclude a comparison between numerical and experimental results was conducted.

The chapter 6 addresses the final conclusions of the dissertation and suggests future work.

2 POLYMER GEARS: STATE OF THE ART

A comprehensive study on polymer gears, typically made of thermoplastics, is necessary to understand their behaviour during operation. This includes assessing parameters such as load capacity, wear, and heat transfer. It is crucial to reflect on these aspects and find out how they contribute to the enhancement or limitation of polymer gears.

2.1 Definition

The primary purpose of a gear, typically made of steel, is to either increase or decrease speed and load capacity through teeth contact. With advancements in the polymer industry and the favourable cost-to-benefit ratio, steel gears used in low-load applications are gradually being replaced by polymer gears [6]. As researchers recognised the potential of this new gear type, they began exploring ways to further improve polymer gears. Nowadays, polymer gears find applications in various industries, including textile, food, and automotive. For example, the use of polymer composite gears in an automotive powertrain has been reported to reduce mass by 70%, reduce inertia by 80% and lower fuel consumption by up to 9% [7, 8].

2.1.1 Advantages and disadvantages

When compared with metallic gears, polymer ones can be a viable alternative for specific cases. With technological and environmental evolution, industries aim to diversify material usage, and polymers emerge as a cost-effective and lightweight option. Advantages include the possibility of dry contact, lower noise, good vibration absorption, and low production costs. These properties prove beneficial in applications where lubrication is not advisable for sanitary reasons (e.g., the food or medical industry) or in cases of gears with difficult access [1, 2].

On the other hand, polymer gears come with certain disadvantages, many of which are mechanical and can be mitigated using different strategies. When subjected to tooth deformations, polymer gears often exhibit low tooth bending resistance, wear resistance, heat resistance, high hygroscopicity, and high hysteresis. These characteristics become more problematic at higher running temperatures, limiting their use in various applications [8]. Presently, these disadvantages hinder the use of polymer gears in industries such as aeronautics, aerospace, and motorsport, where their low weight could be a significant advantage.

In terms of engineering, there are two types of plastics: “engineering plastics” or “E.P.” and “super-engineering plastics” or “S.P.”. The first group is known

for having lower values of heat resistance (about 100°C), tensile strength (about 49 MPa) and yield modulus (about 2.4 GPa). The second group can withstand higher values of these properties, and the most crucial value concerning gears is the working temperature, which can exceed 150°C [9]. Almost every thermoplastic and thermosetting resin falls into the first group. However, from the initial studies, two thermoplastic resins stood out for their performance in running gears: polyamide (PA) and acetal (POM).

The objective for the coming years is to improve polymer gears, reducing these disadvantages while carefully preserving their principal advantages. Although a challenging and lengthy process, investing in their study is crucial.

2.1.2 Properties

With the shift from metallic gears to polymer ones, the way they interact with their surroundings also changes. Polymer gears exhibit significant variation in mechanical properties with temperature fluctuations. Furthermore, when subjected to different moisture levels such as dry, humid, or wet, they also demonstrate varying yield modulus and yield strength with increasing deformation. This behaviour is represented in Figure 1 and is crucial to take into account when designing a gear, as well as considering the temperature effect [3].

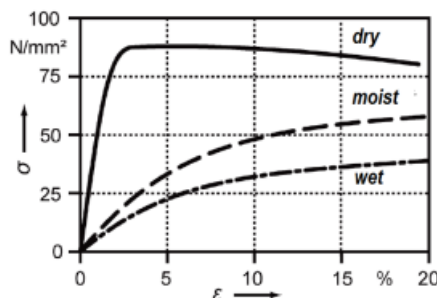


Figure 1 – Stress-strain curve of PA66 [3].

2.1.3 Processes

Regarding the manufacturing process the most used for polymer gears is the moulding injection, but it will depend on the quantity, geometry and size pretended for the gears. These processes can make more complex gear geometries and decrease the weaknesses of conventional gears.

Then the sequence of casting, injection moulding and machining will be presented as well as their selection parameters [10, 11].

Casting

Casting is commonly utilised for the production of large gears, especially when

their dimensions make it challenging to employ machinery suitable for machining or injection moulding. This preference is influenced by the practical limitations of handling large-sized gears, which typically weigh around 1 kg. Despite the advantages of casting for these larger gears, this manufacturing process is often not the preferred choice, as the economic feasibility of producing sizeable gears may be questioned when compared to the production of smaller metal gears.

Currently, only specific materials can be utilised for casting method. Examples include cast polyamides like PA6 and PA12, alongside polyurethane elastomers [3, 7].

Following the moulding procedure, the polymer can be extracted from the master mould, facilitating the creation of multiple replicas, contingent upon the integrity of the original master. While the replication process generally yields high fidelity, challenges can emerge when replicating intricate geometries, such as re-entrant side-walls. Nevertheless, this replication technique boasts impressive precision, capable of reproducing features as minuscule as 30 nm, a feat achieved through a method termed nanocasting [12].

To address any potential imperfections in the gear's surface, it is commonplace to enhance the finishing using machining techniques [3].

Injection moulding

This type of process is commonly used in mass productions and when the final quality requirements are not too high. To overstep this limitation usually the gears goes through a machining process.

Low time production, and low cost can be achieved while enhancing the geometric precision and gear's strength, but is necessary to understand the materials shrinkage. This parameter changes depending on the material and refers to his contraction during the cooling time due to different densities at different temperatures. Shrinkage also depends on gear geometry, mould design, reinforcements in polymer, injection, packing and cooling time [11].

Injection must be used up to 10 mm of wall thickness, because larger values leads to a longer injection and cooling times which will increase the probability of more defects [11].

Regarding the mould cavity gates, the methods used for polymer gears are the sprue diaphragm gate or the 3 or 4 pin point gates. Both gate are shown in Figure 2 and Figure 3, respectively [13].

Reduced shrinkage near the injection gate due to the relatively unrestricted material flow, which effectively fills the rest of the mould cavity. Conversely, in regions farther from the gate, the material experiences rapid cooling and partial solidification, resulting in greater shrinkage. This phenomenon can be attributed to reduced mobility in those areas [13].

One typical defect of the injection moulding process is the lunker defect that

typically occurs where structure's shape changes sharply near the tooth root, with an ellipse size having an aspect ratio of 2:1. This defect is also related with the contraction during the cooling time and with shrinkage process [10].

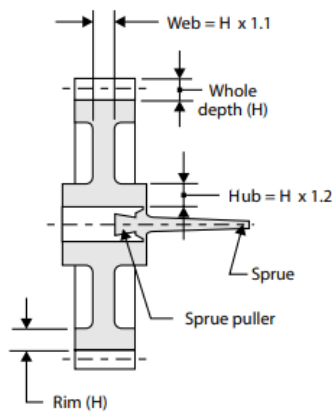


Figure 2 – Sprue diaphragm gate [13].

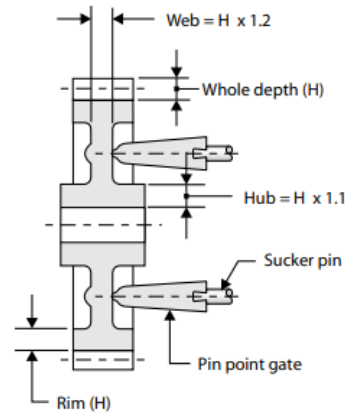


Figure 3 – 3 or 4 pin point gates [13].

Machining

Machining process it is advisable to use in the presence of a small or medium series since the tool cost are lower when set side by side with mould injection machines. The machining methods used are the same as for metal gears, for example gear forming (by milling or broaching) and gear generation processes (i.e. hobbing).

Nevertheless this type of process do not have a better wear rate during running when compared with injection moulded gears, as shown on the tests made by *K. Mao et al.* [2, 8].

2.2 Failure types

There are various potential failures in polymer gears, but certain types are more common than others. Considering the inherent challenges of this material makes it evident that the types of failures are directly correlated with these challenges.

2.2.1 Tooth temperature

As discussed previously, an increase in temperature significantly affects the mechanical properties of polymers. Especially, there is a reduction in the elasticity modulus (for different moisture levels) and a decrease in the load capacity. Additionally, polymers exhibit poor thermal conductivity, meaning they dissipate heat less effectively than other materials. These observations are supported by the data presented in Figures 4–5. Such thermal behaviour is directly correlated with power losses, which in turn impacts the service lifetime of the material.

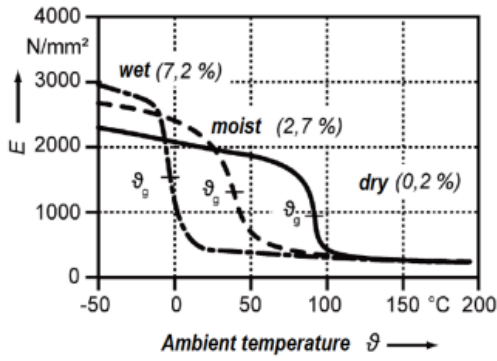


Figure 4 – Elasticity modulus for PA 66 [3].

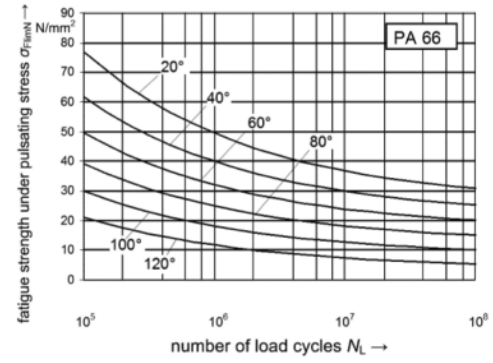


Figure 5 – Variation of fatigue strength for different temperatures [3].

Figures 6–7, presents an analysis of the temperature distribution within a plastic gear tooth. The figure illustrates that temperatures are elevated near the pitch point, where rolling contact occurs, compared to the regions near the tooth tip and root, which are involved in sliding. In particular, heat dissipation occurs more rapidly at the tip and root due to enhanced efficiency of radiation cooling in these specific regions [9].

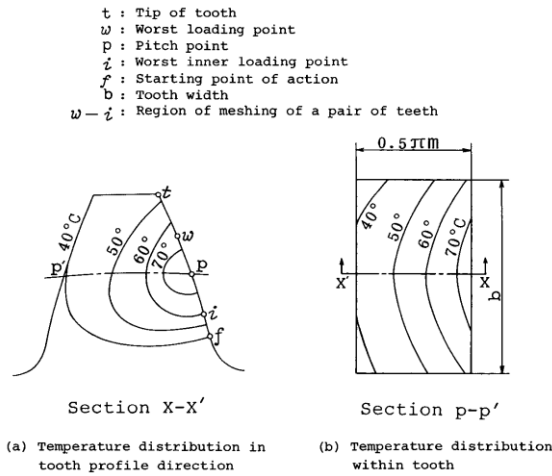


Figure 6 – Temperature distribution in tooth of nylon gear [9].

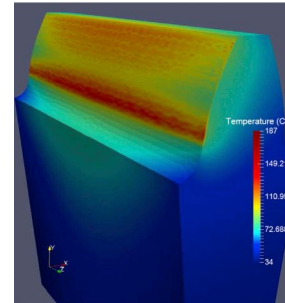


Figure 7 – Temperature distribution in tooth of POM gear [14].

Notably, hysteresis plays a significant role in influencing the temperature at the pitch point. A comprehensive discussion on tooth deflection hysteresis will be provided in Chapter 2.2.2.

A numerical approach to the flank temperature can be made using the Equation 1 provided by the standard VDI 2736 [15].

$$\vartheta_{\text{Fla}} \approx \vartheta_0 + P_N \cdot \mu \cdot H_V \cdot \left(\frac{k_{\vartheta, \text{Fla}}}{b \cdot z \cdot (v_t \cdot m_n)^{0.75}} + \frac{R_{\lambda, G}}{A_G} \right) \cdot ED^{0.64} \quad (1)$$

2.2.2 Tooth deflection

One notable disadvantage of polymers that can contribute to failure is their stiffness. When compared to metals, the modulus of elasticity (E) for POM ranges between 2.6 and 3.2 GPa, while it is approximately 210 GPa for steel. Therefore, tooth deflection must be taken into account. Figure 8 illustrates that the deflection of teeth in plastics is influenced by the loading time. Particularly, under a static load applied continuously, the deflection tends to increase over time. In this context, the initial deflection is approximately $30\text{ }\mu\text{m}$, but after 4 minutes, it approaches $40\text{ }\mu\text{m}$ [9].

Another insight from the figure is that due to POM's lower Young's modulus compared to PA, the teeth made of POM experience greater deflection than those made of PA within the first 2 minutes. However, POM exhibits lower viscosity than PA. Consequently, after this initial period, teeth made of POM demonstrate less deflection than those made of PA [1].

Upon unloading, both POM and PA return to their original shapes, with POM exhibiting faster recovery owing to its lower viscosity. Plastics are viscoelastic materials, necessitating some time to revert to their original unloaded shapes after undergoing loading, as depicted in Figure 9. The area bounded by the hysteresis loop represents the energy dissipated as heat, which elevates the gear temperature. Notably, the shape of the hysteresis loop is influenced by the viscosity of the plastics; higher viscosity yields a larger loop [9].

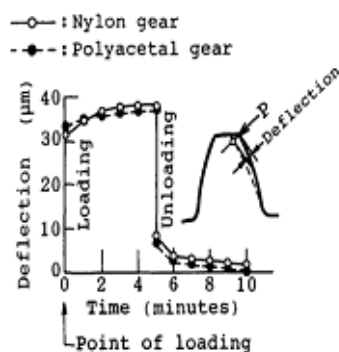


Figure 8 – Tooth deflection [9].

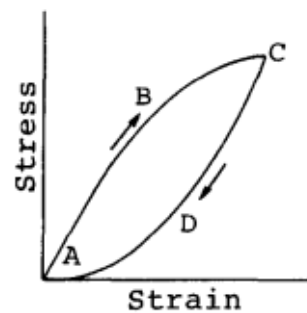


Figure 9 – Hysteresis loop of plastics [9].

2.2.3 Wear

The wear of polymer gears is significantly influenced by the materials employed in the driving gear (responsible for supplying the driving force) and the driven gear (which receives motion or power from the driving gear).

Figure 10 illustrates a typical wear progression observed when the same material is used for both the driving and driven gears, segmented into three distinct phases.

- Initial running in - A brief period characterised by significant wear;

- Nearly linear - A stage during which wear steadily escalates until it reaches a critical value;
- Final fracture - Stage after reaching the critical value, and wear intensifies sharply, culminating in failure.

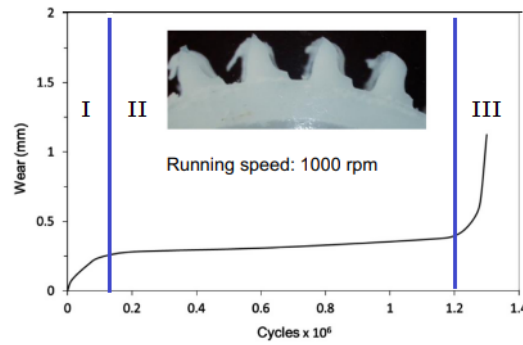


Figure 10 – Wear on a acetal/acetal combination [8].

This mode of failure typically manifests near the pitch point of the driving gear teeth, and the number of cycles is contingent upon the torque applied. Various load values are shown in Figure 11 for an acetal/acetal combination. Higher loads result in accelerated failures, while lower loads correspond to a slower and more consistent wear rate, increasing the gear's life.

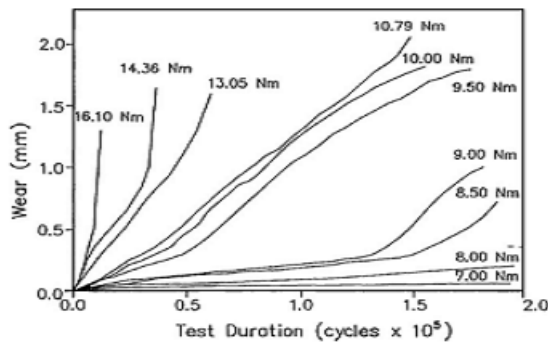


Figure 11 – Wear for different loads [16].

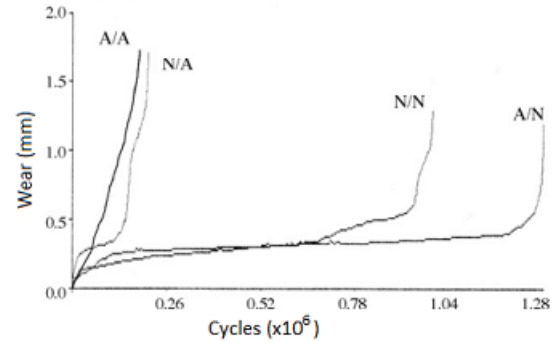


Figure 12 – Wear for different combinations [16].

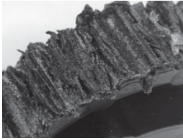
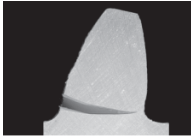
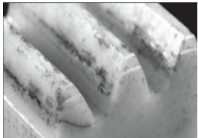
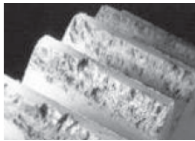
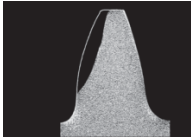
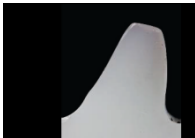
For combinations of dissimilar materials, the results exhibit notable variations. In particular, as shown in Figure 12, the N/A combination (with nylon as the driving gear and acetal as the driven gear) yielded outcomes comparable to those of the A/A combination. In contrast, the A/N combination demonstrated a marked enhancement in lifespan [16].

Moreover, the selection of materials for both the driving and driven gears plays a pivotal role in determining wear performance. Chapter 2.2.5 will delve deeper into the underlying reasons for these observations.

2.2.4 Failure types and causes

In Table 2 various types of failures and their corresponding causes are presented [3, 4].

Table 2 – Failure types and main causes [3].

Damage	Spur Gears	Main Cause
Partial melting		Elevated temperatures on gear tooth surfaces
Tooth root fracture		Excessive bending stresses on gear teeth
Tooth flank fracture		Amplified crack expansion from hertzian pressures
Pitting		Concentrated pressures at gear tooth contacts
Tooth wear		Low wear resistance
Tooth deformation		Pronounced shape alterations due to high loads

2.2.5 Increase lifespan

Several techniques have been investigated to enhance the lifespan of plastics. These approaches often target specific objectives, such as reducing temperature or minimising wear. Both lubrication and modifications to tooth geometry can contribute to extending the lifespan of plastic gears. The discussion on lubrication is presented in Chapter 2.3.

In addition to temperature reduction considerations, excluding hybrid polymer gears, the auto-cooling method emerges as a promising approach for polymer gears. The inherent movement of the gear aids in dissipating some heat generated during meshing.

However, as illustrated in Figure 13, this method can facilitate the inflow of air through the holes due to pressure differentials during rotation. This airflow then impacts the tooth flank, leading to a reduction in temperature.

As highlighted by *Tsukamoto*, employing this technique has the potential to extend the operational lifespan from approximately $(2 \sim 3) \cdot 10^6$ cycles to beyond 10^7 cycles [17].

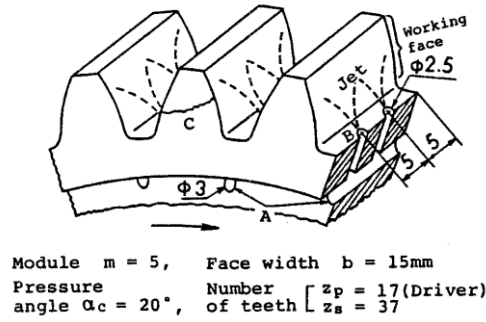


Figure 13 – Auto-cooling method by *Tsukamoto* [17].

In the case of the driven gear, the direction of tooth action extends from the tip and root towards the pitch circle. This orientation mitigates the formation of failures near the pitch circle. The sliding speed from both gears is elucidated in Figure 14. Moreover, Figure 15 illustrates the wear patterns on the tooth. Here, the arrows delineate the frictional direction during mesh entry and exit, leading to distinct wear scenarios. Particularly, for the driving gear, the friction moves away from the pitch line, inducing a wear trough or a depression that accentuates stress concentration and potential fissure formation. In contrast, for the driven gear, the friction directs towards the pitch line, resulting in material accumulation at the pitch point, which typically diminishes after several rotations [16, 18].

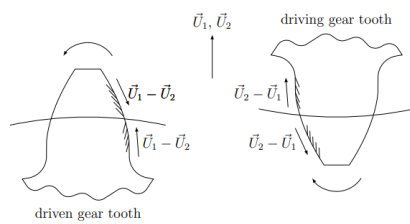


Figure 14 – Sliding speed direction [18].

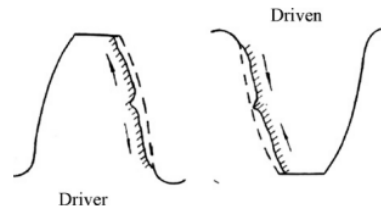


Figure 15 – Wear on driven and driver gears [16].

Given these insights, when employing two distinct materials, such as in scenarios involving both polymer-polymer contact or metal-polymer combinations, the material with inferior wear resistance is ideally suited for the driven gear. This strategic material selection not only optimizes performance but also ensures a balanced wear distribution, thereby enhancing the overall durability of the gear system.

In configurations involving a combination of metal and polymer, such as in a pinion/wheel assembly, it is advisable to employ the polymer as the wheel component. This recommendation arises from the wheel typically having a greater number of teeth than the pinion.

Consequently, the wheel has a longer duration to dissipate heat compared to the pinion, leading to reduced tooth temperatures.

Regarding to the failures of tooth root and flank fracture, they can be caused by excessive bending stresses, to the first case, and from amplified crack expansions from Hertzian pressures, to the second case. This issue can be delayed by reinforcing the polymer gear tooth utilising fibre direction parallel to the tooth surface [19].

2.3 Lubrication

As mentioned earlier the main advantages of polymer gears is the possibility to running without lubrication. Although, for high rotational speed is important to consider the use of oils to decrease the friction coefficient creating a better film thickness and sufficient separation of the surfaces. By choosing this option is possible to decrease noise and wear and extend the gears life-time [20]. Even if this situation seems ideal, it is necessary to take into account that such a combination goes against one of the main advantages of polymer gears, their capability for dry running, so it is necessary to consider their use.

2.3.1 Lubricant types

There are two types of lubricants that can be used on polymer gears, the solid lubrication through internal fillers and the external lubrication. The first type will require a longer time on mechanical projection to adjust the gear ratio once some fillers may reduce mechanical properties.

Below are three lubricants used on polymer gears.

Fillers

As is known, fillers are added to polymer gears with the aim of improving mechanical properties. These can be mixed with polymer resin during compounding, before injection moulding, or added during this processing stage or even while polymerisation is taking place.

The advantages of using a filler are the improvement of heat conductivity, the decrease in the coefficient of friction and, consequently, the increase in life-time. Moreover, depending on the added filler, it is possible to have different positive effects on polymer gears in addition to their main advantages. However, negative effects may arise with some materials, for example a decrease in impact resistance.

In Figure 16, the wear on different polymer gears is demonstrated when using Polytetrafluoroethylene (PTFE) fillers. This thermoplastic reduces friction and, consequently, wear, but it also reduces impact resistance and bending capacity [2].

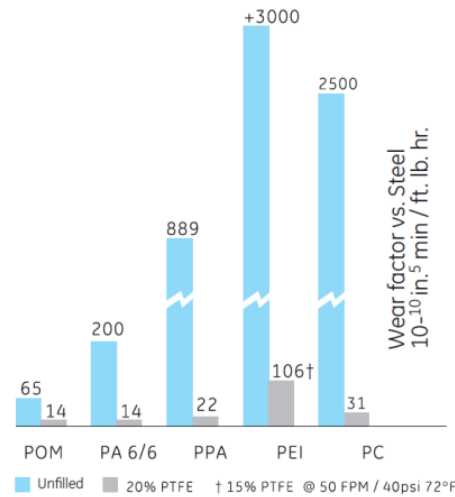


Figure 16 – Effect of PTFE fillers on wear [2].

Greases

In the context of lubrication, greases are formulated as solid to semi-fluid substances resulting from the incorporation of a thickening agent into a liquid lubricant, commonly utilizing metal soaps as the thickening medium. Moreover, these formulations frequently integrate supplementary compounds aimed at improving specific performance attributes [21].

In gear cases, one significant drawback of greases, when compared to lubricating oils, is their inability to handle higher speeds due to their thicker consistency, so the rotational speed is crucially maintained under 5 m/s [3]. Additionally, they are not the optimal choice when efficient heat dissipation is required [21].

Oils

Regardless to polymer gears it is mandatory that oil polarity match with gear one. So it is necessary that polar polymer gears must only be lubricated with non polar oils and vice-versa.

Another point to consider is hygroscopic effect. Hygroscopy is the capacity of attracting and holding water molecules from surroundings. Since most oils are water-based, this factor must be considered, as it is not desirable for the gear to absorb part of the oil, so you should opt for a less hygroscopic material. In Figure 17 is possible to understand how differently materials can behave in terms of water molecules absorption [9].

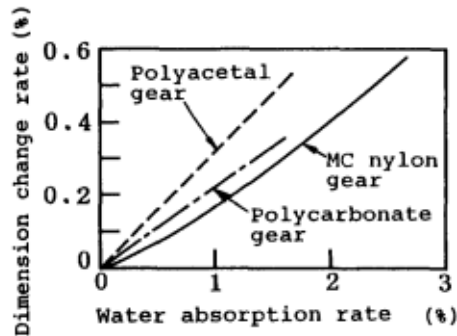


Figure 17 – Water absorption rate and dimension change rate of plastic gears [9].

In the process of absorption, the gear experiences an increase in mass, which contributes to power losses. Such a phenomenon may have been observed in a previous work [2]. Following the wear test, the gear was expected to exhibit a reduction in mass, however, it instead displayed an increase.

The solution for this problem is to choose a high viscosity oil since they have less possibility to infiltrate in the gear's surface.

To sum up, when intending to use a lubricant, it should be chosen according to the operating temperature and gear purpose, always taking into account the cost, efficiency and gears life.

2.4 Hybrid polymer gears

Analysing the current panorama is easy to understand the potential improvement of polymer gears. Hereupon is important to develop and test some changes in polymer gears so they can improve their weaknesses and maintain their strengths. The most suggested change currently is the insertion of a metallic insert inside the polymer gear. This will allow better heat dissipation in each tooth, trying to overcome one of the main problems of these gears and allowing them to be used in a wider range of applications.

As mentioned above the study about hybrid polymer gear is taking place and the principal points to develop and improve are: the insert material, geometry and the manufacturing process. This parameters will defined the quality of heat evacuating and the possibility on increasing the load capacity, thus allowing a greater variety of applications for these gears.

2.4.1 Advantages and disadvantages

Hybrid polymer gears offer more advantages than disadvantages, but the decision to use them in each case requires careful consideration. In instances of low load or low rotational speed, a well-designed mechanism may utilise a conventional polymer

gear. However, for applications with higher demands, it is advisable to opt for hybrid polymer gears.

Hybrid polymer gears offer the following advantages:

- Application at higher rotational speed and load capacity;
- Dry running with low loads;
- Meshing with metallic gears;
- Improved heat dissipation and consequent gear life;

While there are fewer disadvantages, they can be more problematic and pose a significant barrier to the use of hybrid polymer gears. Currently, manufacturing these gears remains challenging, and overcoming this hurdle is necessary for them to demonstrate their full potential.

2.5 Heat transfer

The integration of an insert into the polymer gear introduces a novel boundary condition at the interface of metal and plastic materials.

To precisely determine the temperature distribution between two thick solid bodies in contact (ΔT_c), a thermal contact conductance can be defined as the resistance between the temperature drop and the heat transfer passing through the apparent cross-sectional area. This thermal resistance is quantified by Equation 2 [5].

$$R_{tc} = \frac{\Delta T_c}{Q \cdot A_a} \quad (2)$$

The total heat transfer contact coefficient, h_{tc} , comprises three components: conduction across the contacting points (h_c), radiation through the gaps between the surfaces (h_{rad}) and conduction through the gas filling these gaps (h_g). However, when considering the manufacturing of a hybrid polymer gear via moulding, the heat transfer through the minute gas-filled gaps between the surfaces becomes negligible. Consequently, the radiation-induced heat transfer can also be disregarded. Thus, the sole mode of heat transfer to be considered is conduction across the contacting points, represented as $h_{tc} = h_c$ [5].

Thermal contact conductance

In modelling the thermal contact conductance, if the heights of the two contact surfaces adhere to a Gaussian distribution, the scenario can be simplified. Specifically, it can be approximated as a single rough surface in contact with a smooth (or theoretical) surface, as depicted in Figure 18.

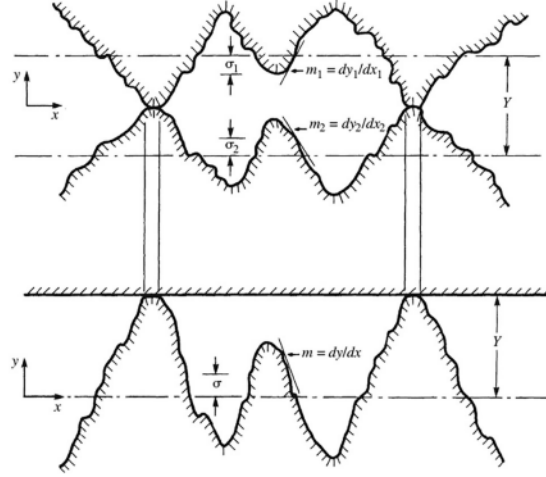


Figure 18 – Transformation of two contact surface into one rough surface [5].

Drawing insights from Figure 18, it is evident that in real-world scenarios, a contact might consist of multiple areas, spread across a discernible contact area denoted as (A_a). *Yovanovich* introduced a comprehensive thermal model adjusted for conforming rough surfaces [22]. This model delineates the constrictions and resistance arising from a singular asperity in contact and extrapolates these findings to encompass the entire surface. For a specific contact point (i) between two distinct surfaces (labeled as 1 and 2), Equation 3 can be incorporated into Equation 4 to determine the contact resistance.

$$k_s = \frac{2 \cdot k_1 \cdot k_2}{(k_1 + k_2)} \quad (3)$$

$$R_{si} = \frac{\psi_{ci}}{4 \cdot k_s \cdot a_i} \quad (4)$$

Finally, to calculate the thermal contact conductance, Equation 5 is used.

$$h_c = 2 \cdot k_s \cdot \frac{1}{\psi_{ci}} \cdot \frac{a_i}{b_i} \quad (5)$$

Certain simplifications can be introduced based on these equalities:

- $\psi_{ci} = (1 - \epsilon)^{1.5}$
- $\epsilon = \frac{a_i}{b_i} = \sqrt{\frac{A_r}{A_a}}$

In the context provided, the parameter ϵ represents the ratio between the real area of contact (A_r) and the apparent area of the contact (A_a). Additionally, ψ_{ci} denotes the radius of the contact spot constriction.

In the realm of body contact considerations, three distinct types can arise: elastic interactions, typically observed in smaller bodies; plastic deformations, associated

with significant contact strain in larger bodies; and elasto-plastic behaviours, which represent a blend of both elastic and plastic responses. For hybrid polymer gears, the elastic model emerges as the most fitting and representative scenario. Consequently, it has been selected as the preferred model for characterising heat transfer dynamics.

Following comprehensive studies conducted by *Mikic*, *Yovanovich* and *Fuller and Marotta* the heat transfer coefficient can be determined utilising Equation 6 [23, 22, 24].

$$h_c = 1.49 \cdot \frac{k_s \cdot m_s}{\sigma_s} \cdot \left(\frac{p}{H_{\text{poly}}} \right)^{0.935} \quad (6)$$

The prediction of the heat transfer coefficient, influenced by the thermal contact resistance (TCR) between the polymer and the metallic insert, was conducted while accounting for various contact pressures. Notably, both the TCR and the associated heat transfer coefficient are contingent upon the contact pressure, p , established between the polymer and the metallic insert [5].

2.6 Stress in the tooth root

For polymer gear, a prevalent failure mode is the fracture occurring at the tooth root due to elevated bending stress. This factor warrants significant attention during the gear development process, emphasising the necessity to constrain stress levels in the tooth root. The stress in the tooth root is represented by Equation 7.

$$\sigma_F = K_F \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_\epsilon \cdot Y_\beta \cdot \frac{F_t}{b \cdot m_n} \quad (7)$$

Another crucial thing is to consider the contact pressure exerted on a polymer tooth. The representation of this property can be found in Equation 8, as outlined in the standard VDI 2736 [7].

$$\sigma_H = Z_E \cdot Z_H \cdot Z_\epsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t \cdot K_H}{b \cdot m_n} \cdot \frac{u+1}{u}} \quad (8)$$

The methodology for determine Equations 7–8 is detailed in standard VDI 2736 [15]. It is essential to incorporate safety factors during the design of these gears.

2.7 Test rig for polymer gears

Several experiments have been conducted on this subject, leading to the emergence of novel approaches for assessing wear in polymer gears. As explored by *K. Mao et al.*, in their paper titled "A New Method for Testing Polymer Gear Wear Rate and Performance" this innovative method bears similarities to the back-to-back

test configuration. However, unlike the traditional version applied to steel gears, the block supporting the polymer test gears in this method is pivotally connected to the instrument base.

In this modified configuration, the block undergoes slight rotation as the gear tooth thickness decreases. Incorporating a moment arm and adjustable weight to impart the test load ensures a virtually constant torque on the test gears, regardless of tooth wear. Significantly, considering the thermal behaviours of polymer gears, continuous recording of gear surface wear is achieved using a capacitive or inductive sensor that monitors the motion of the pivot block [25].

Figure 19 illustrates the proposed setup for testing polymer gears in this configuration.

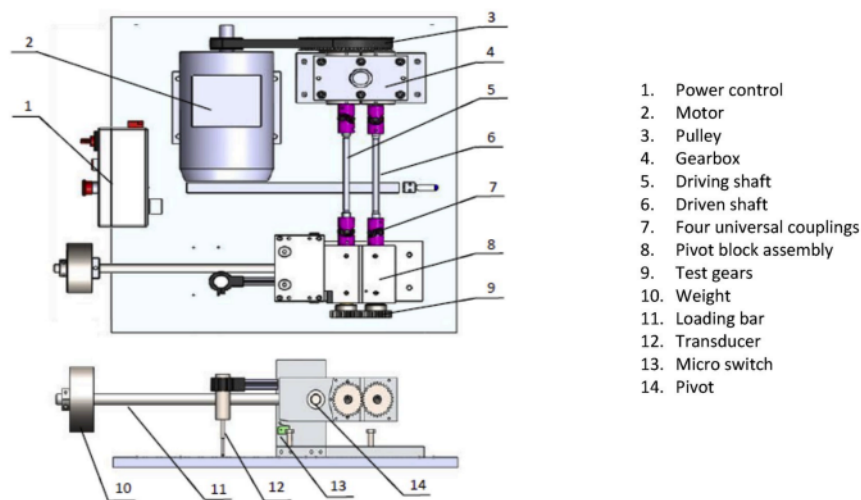


Figure 19 – Representation of FZG test rig for polymer gears [25].

This test rig was initially expected to run under the following conditions.

- Speeds: 500–5000 rpm, precision 1%
- Load torque: 1–50 Nm, stable to 1%
- Maximum wear: 1.6 mm, resolution 0.002 mm
- Temperature: 20 – 200°C at the tooth interface

K. Mao et al. concluded that a broad spectrum of polymers (PC, HDPE, PA, POM, PEEK, GFR) exhibits favourable performance in both dry and lubricated conditions under loads ranging up to 16 N·m and speeds between 500 rpm and 4275 rpm [25].

In test rigs, an essential consideration pertains to the vibration of mechanisms during operations characterised by low torque and low-speed conditions. This vibration occurs when the natural frequency of the torsional oscillation of the pivot

block aligns with the minimum excitation frequency produced by the gears. To address this challenge, it is recommended to utilise drive shafts of smaller diameters during operations with the most minimal torques and speeds. This approach seeks to reduce the natural frequency of both the load bar system and the pivot block [26].

2.7.1 Working temperature

This topic is related to temperature during the FZG tests. Regardless to the oil and according to findings by *D.J. Hargreaves et al.*, the temperature of the oil during an FZG test may elevate from 85°C to 110°C, contingent upon the specific oil used. Notably, the more costly oils exhibit less temperature fluctuation. Another notable characteristic is that synthetic oils are recognised for generating lower operating temperatures compared to mineral oils. This attribute, particularly relevant for gearboxes, is associated with a decrease in friction losses [27]. Figure 20 illustrates various types of oils employed in these tests.

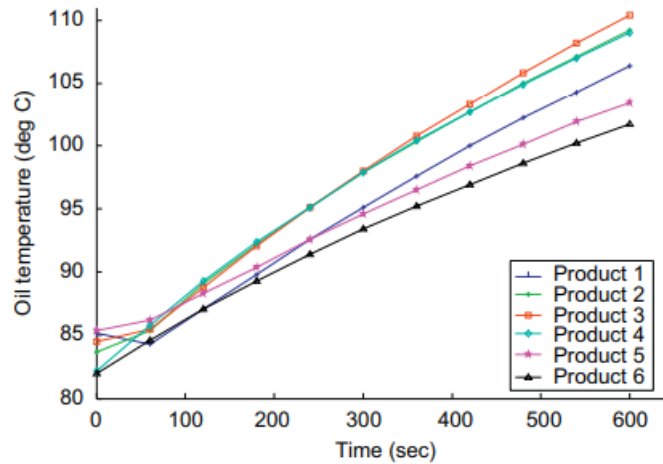


Figure 20 – Oil temperature variation over FZG test duration [27].

Another important aspect is the flash temperature during the test.

Blok elucidates the phenomenon of flash temperature as originating within an extremely thin surface layer upon the engagement of gear teeth [28]. Scuffing ensues when the flash temperature, in addition to the underlying bulk tooth temperature, surpasses the melting point of the gear material [26].

Unlike the subsequent calculations, for a more accurate determination of the flash temperature, it is essential to recognise that polymer gears exhibit variations in their mechanical properties with temperature. Additionally, the internal hysteresis of the material must be considered [26].

The flash temperature is given by Equation 9.

$$T_f = A \cdot \frac{q_{av}}{\sqrt{k \cdot c'}} \cdot \sqrt{\frac{\omega}{v}} \quad (9)$$

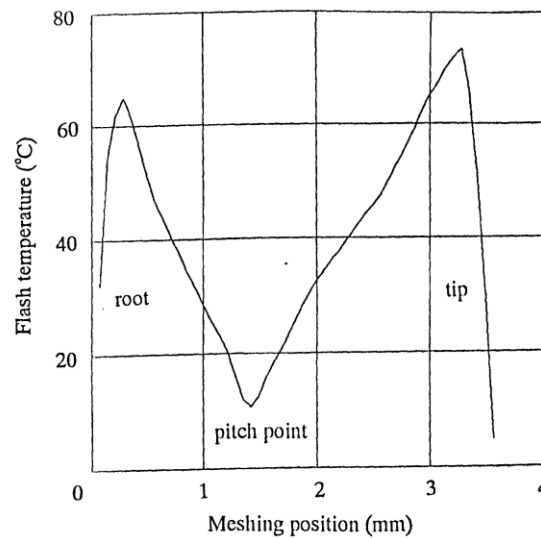


Figure 21 – Flash temperature on a acetal gear [26].

Where v is the sliding velocity, ω is the length of the heat source in the sliding direction, k is the thermal conductivity of the material, c' is the specific heat per unit volume, A is a form factor offered by Blok, which is specific to the distribution of the heat flux, and q_{av} is the average heat flux input over the length b [29].

Figure 21 presents a representative assessment of T_f for an acetal gear, with v derived from Equation 9. It is important to note, however, that T_f represents only a transient parameter. This does not account for the cumulative and cyclic heat generation inherent in a continuously rotating power-driven gear pair. Consequently, the flash temperature alone cannot determine the definitive operating temperature for the gears [29].

The bulk temperature represents the balance between heat generation and heat dissipation during gear operation. When gears are in a steady operational state, their bulk temperatures remain relatively constant. The mechanical strength and stiffness of the gears are intrinsically linked to the bulk temperature of their teeth.

According to *Zedong Hu*, when polymer gears undergo testing in a non lubricated state, the bulk temperature during the steady wear phase demonstrates a progressive decline with an increasing number of operational revolutions. By approximately 10^6 cycles, this temperature stabilizes, as illustrated in Figure 22 [30].

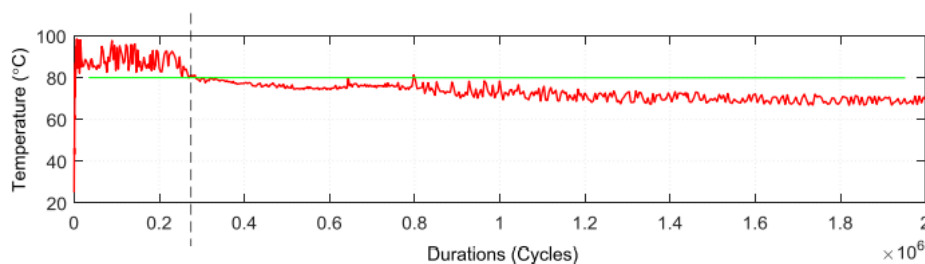


Figure 22 – Bulk temperature on a polymer gear [30].

By neglecting both thermal conduction and thermal radiation effects, the bulk

temperature is represented by Equation 10 [31].

$$\theta_b = \frac{3,927 \cdot \mu \cdot M}{c \cdot \rho \cdot z \cdot b \cdot (r_a^2 - r^2)} \quad (10)$$

where Z , M , ρ and c are tooth number, transmitted torque, specific gravity and specific heat, respectively; r_a , r , μ and b are outside radius, reference radius, friction coefficient and tooth face width, respectively [31].

This subject holds significant importance, particularly in the context of polymer gears. Unlike their metallic counterparts, the teeth of polymer gears are susceptible to substantial creep, especially when operating at elevated temperatures. Additionally, the rise in temperature can adversely affect the mechanical properties of the polymer [25].

2.7.2 Load capacity

As evidenced by *K. Mao et al.* and illustrated in Figure 24, a distinct transition torque of approximately 8.5 Nm is evident (highlighted by the red line in the figure) [25]. When subjected to loads exceeding this transition torque, the wear rate of the POM gear pair escalated significantly, leading to pronounced thermal bending failures. Conversely, when the POM gears operated below the transition torque, wear rates remained minimal, suggesting the potential for extended service life with consistent performance [25].

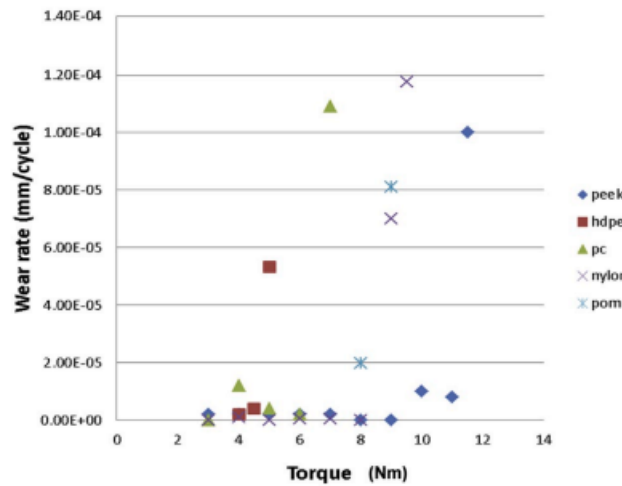


Figure 23 – Wear rate against load for different polymer pairs [25].

Utilising a comparable methodology, Figure 23 illustrates variations in wear at specific torque levels. Each material exhibits distinct wear characteristics, leading to varied load capacities.

The transition torque values for the respective materials are delineated as follows [25]:

- PEEK (11 Nm)

- POM (8.5 Nm)
- PA46 (8.25 Nm)
- PC (6 Nm)
- HDPE (4.5 Nm)

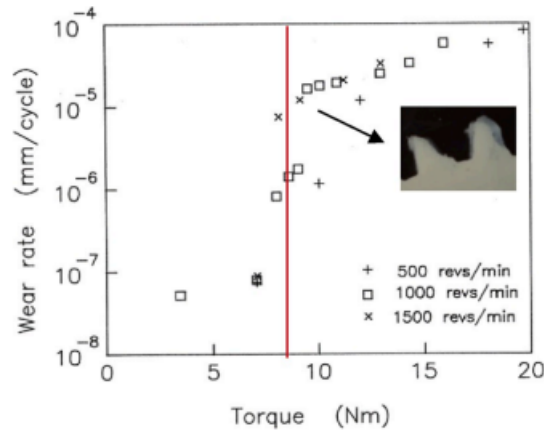


Figure 24 – Wear transition on a polymer gear [25].

Within the same study, a comparison between POM gear pairs (with and without reinforced glass fibres) revealed a 50% increase in loading capacity for the reinforced gears, as represented in Figure 25 [25].

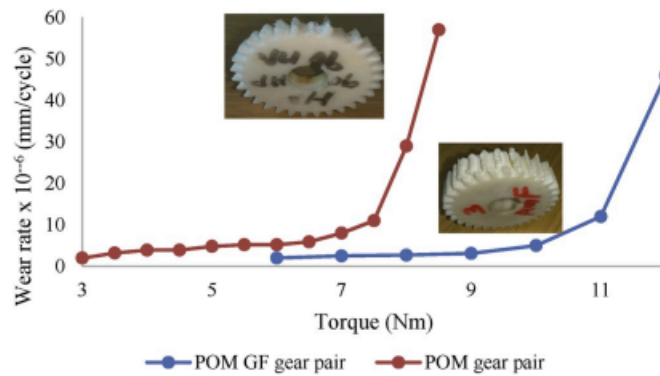


Figure 25 – Wear rate comparison for POM with and without fillers [25].

2.8 Wear and Archard law

Measuring wear can be achieved by assessing mass loss after running tests or through the comparison of tooth flank profiles before and after tests [30].

Its main advantage is the possibility of being tested under conditions similar to those encountered in real-life engineering problems.

To complete this evaluation, the Archard law can be employed to quantify the severity of wear in the gear. Equation 11 provides the dimensionless Archard wear coefficient (k_a), representing the severity of contact. If $k_a > 10^{-2}$, the wear is considered severe, and if $k_a < 10^{-8}$, the wear is considered soft.

$$W = k_a \cdot \left(P \cdot \frac{L}{H} \right) \quad (11)$$

Where P represents the normal force, L the sliding distance and H is the hardness of the wearing object.

2.9 Mechanical and thermal tests

This chapter addresses the theoretical aspects of the tests conducted during this dissertation.

2.9.1 Dynamic Mechanical Analysis (DMA)

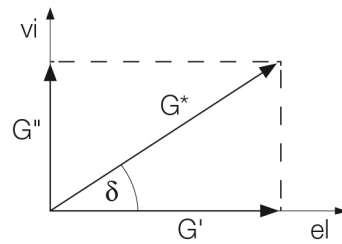
The first test performed was a Dynamic Mechanical Analysis, which characterises a material's properties as a function of temperature, time and frequency, by applying a sinusoidal deformation with controlled stress and strain. This test measures the storage modulus (E'), loss modulus (E''), complex modulus (E^*) and $\tan \delta$.

Polymers are viscoelastic materials, meaning they exhibit both elastic and viscous behaviour. The storage modulus represents the elastic behaviour of the material, while the loss modulus represents the viscous behaviour.

The storage modulus is associated with the elastic behaviour and corresponds to the mechanical energy stored during material deformation. When the stress is released, the material tends to return to its initial shape.

The loss modulus is linked to the viscous behaviour and arises from internal friction between particles and molecules, creating heat and resulting in energy loss that can not be restored [32, 33].

These two moduli are directly related to the complex modulus, which can also be obtained through this test. The relationship between these moduli is illustrated in Figure 26, introducing a new property, the phase angle (δ). To determine the values of these moduli, the software analyses the phase angle, which is the time lag between the input and output oscillations. This angle is positioned below the complex modulus, thereby representing the storage modulus and the loss modulus according to the Pythagorean theorem. The smaller the phase angle, the lower the loss modulus and the higher the storage modulus [34].

Figure 26 – Relation between G' , G'' and G^* [34].

2.9.2 Differential scanning calorimetry (DSC)

Differential scanning calorimetry is a thermal analysis technique used to study the mechanical properties of materials as a function of temperature. DSC measures the energy difference between the absorbed and released energy of a material, comparing it to a reference material. This technique is particularly useful when the quantity of available material for testing is limited.

Typically, the DSC process for consists of three cycles: an initial heating cycle above the melting temperature of the material, a second cycle below the glass transition temperature, and a final heating cycle similar to the first. As illustrated in Figure 27 the three critical points are cold crystallization, which exhibits exothermic behaviour (since $\Delta H < 0$), and glass transition and melting, which show endothermic behaviour (since $\Delta H > 0$). This relationship might seem counter-intuitive because the y-axis direction corresponds to the exothermic direction of the reaction, meaning that positive integral values correspond to exothermic reactions and negative values correspond to endothermic reactions.

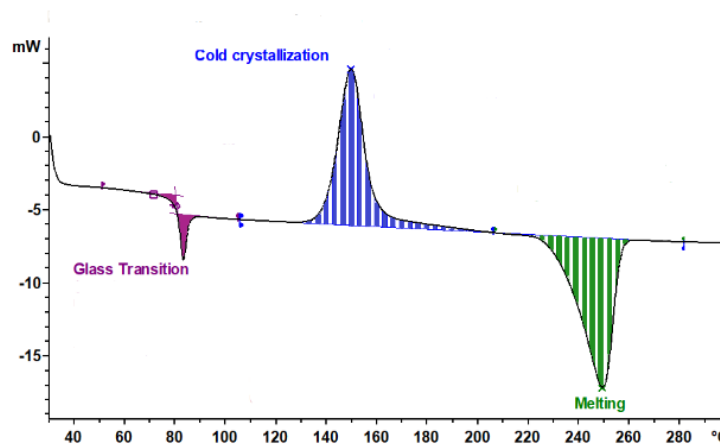


Figure 27 – A typical result obtained from a DSC test.

The purpose of this test is to determine if there are differences in glass transition temperature, cold crystallization temperature and melting temperature between the different POM gears. Conversely, POM and PA6-GF are semi-crystalline polymers that undergo crystallisation during cooling. Consequently, cold crystallisation is not observed in these materials.

3 MATERIALS CHARACTERISATION

3.1 Manufacturing process

The gear were injection moulded and the following parameters were considered: cooling times, injection temperature and mould temperature. Several conclusions can be drawn:

- Variation in cooling times changes the overall dimensions of the gears.
- Varied injection temperatures changes the overall dimensions of the gears; lower mould temperatures result in reduced shrinkage.

Table 3 presents the geometric parameters of the polymer gears and hybrid polymer gears.

Table 3 – Hybrid polymer gear specifications.

Parameter	Abb.	Unit	Value
Tooth number	z		30
Normal module	m_n	mm	2
Normal pressure angle	α_n	°	20
Helix angle	β	°	0
Tooth face width	b	mm	15
Reference circle diameter	d	mm	60
Base circle diameter	d_b	mm	56.382
Addendum circle diameter	d_a	mm	64
Root circle diameter	d_f	mm	55
Dedendum of the reference profile	h_{fP}	mm	2.5
Addendum of the reference profile	h_{aP}	mm	2
Fillet on the reference profile	ρ_{fP}	mm	0.5
Profile shift coefficient	x		0

3.2 Coding system

After the manufacturing process, the gears are designated according to their characteristics. The coding system consists of PS-AC25T120. The first letter signifies the gear material, where “P” denotes POM and “N” indicates Nylon (PA6-GF). The second letter denotes the type of insert, “S” representing gears without inserts,

“P” for printed inserts, and “M” for machined. The third letter, “A” represents the mould material, which in this case is AMPCO, and fourth letter “C” indicates the cooling time, with the subsequent digits corresponding to the specific duration in minutes. “T” represents the temperature of the mould during injection, with the following numbers indicating this temperature in degrees Celsius.

Table 4 depicts the available types of polymer gears. This inventory aids in determining which gears will be utilised for the experimental tests.

Table 4 – Hybrid polymer gear inventory.

Gear code	
NM-AC25T125	PS-AC20T100
NP-AC25T125	PS-AC25T80
NS-AC25T125	PS-AC25T100
PM-AC25T100	PS-AC25T120
PP-AC25T100	PS-AC30T100

In the initial phase of experimental testing, only non-inserted gears were used, as the purpose of these tests was to characterise the properties of POM based on mould temperature and PA6-GF. The gears used for the experimental tests are: PS-AC25T80, PS-AC25T100, PS-AC25T120, and NS-AC25T125.

3.2.1 Samples from polymer gear

In this study, three experimental processes were conducted: a DMA test, a DSC test, and a Pin-on-Disc test. For each of these processes, it was necessary to machine and extract samples from the gears. The samples discussed in this chapter were extracted from the following gear types: PS-AC25T80, PS-AC25T100, PS-AC25T120, and NS-AC25T125. This decision is based on the variation of mould temperature exclusively among POM gears. This deliberate choice enables an examination of the impact of mould temperature on both mechanical and thermal properties. Relatively to the PA6-GF gear, the NS-AC25T125 stands as the sole option available without insert.

For the DMA and DSC tests, small plates were extracted from the gear teeth, the zone of contact, to create a scenario as closely aligned with working conditions as possible. These samples are depicted in Figure 28, and respective drawing in Figure 29. The dimensions for this sample are: $15 \times 5 \times 0.5$ (length $\times$ width $\times$ thickness).



Figure 28 – Prepared specimens for DMA testing.

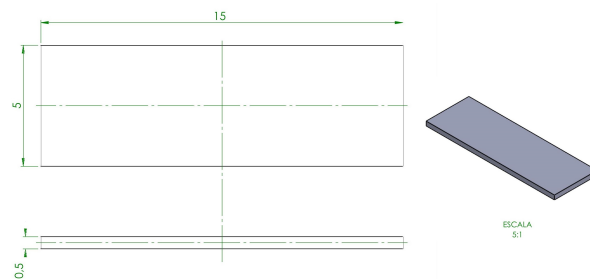


Figure 29 – DMA samples drawing.

From the samples depicted in Figure 28, the samples for the DSC test were extracted using a punch, resulting in disc-shaped specimens with a diameter of 5 mm.

In the context of the Pin-on-Disc test, the pins were extracted from the gears within the specified zone as depicted in Figure 30. The outcome following pin extraction is visible in Figure 32, where circular marks remain on the gear. A total of 9 pins were removed from each gear, and their dimensions are detailed in Figure 31.

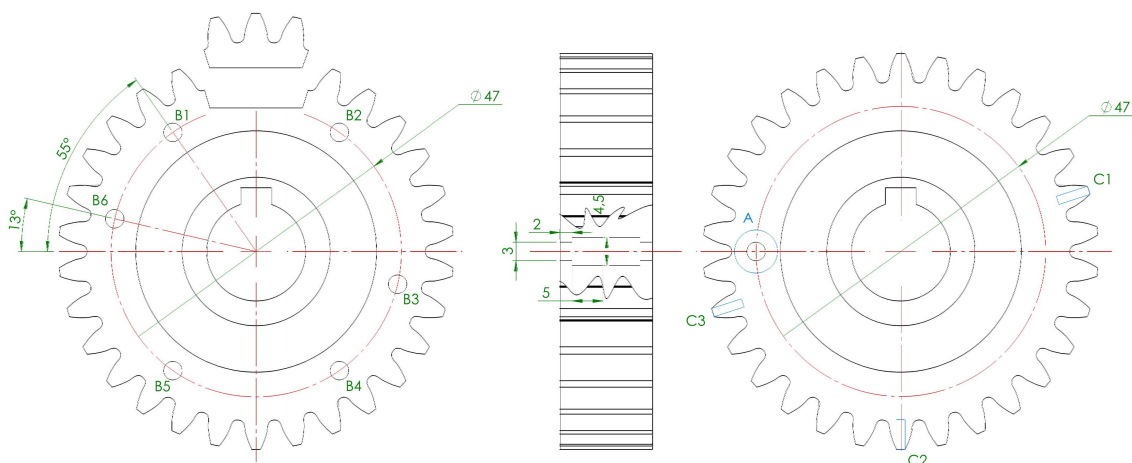


Figure 30 – Drawing with pin positions for extraction.

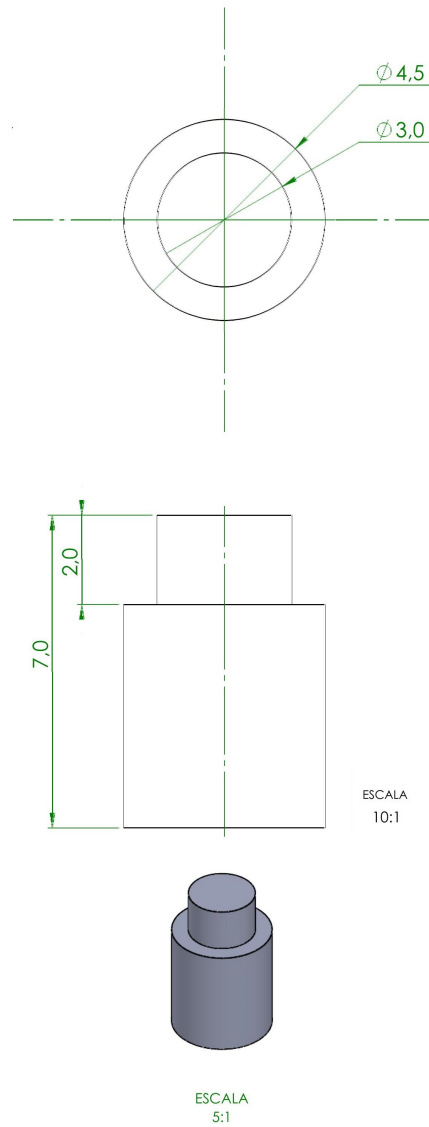


Figure 31 – Pin drawing.

3.3 Lunker defects on samples

Various types of measurements were conducted to evaluate the mechanical and thermal properties of both manufactured polymers. This process involved machining samples from the gears (destroying some gears, as depicted in Figure 32). During the machining process, a lunker defect was identified on the PA6-GF. This failure occurred in the tooth root at a medium width of the tooth during the injection phase, as also observed by *Zehua Lu et al.* [10]. The injection moulding lunker typically occurs where structure's shape changes sharply near the tooth root, with an ellipse size having an aspect ratio of 2:1, as demonstrated in Figure 33.



Figure 32 – POM machined gear.

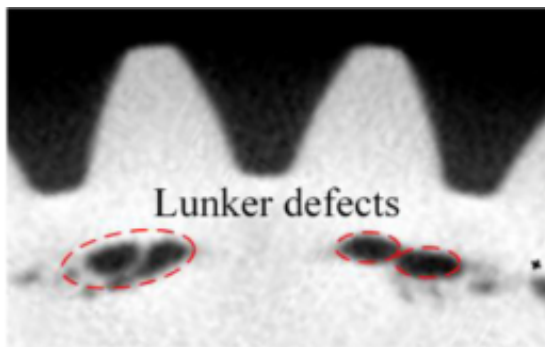


Figure 33 – Lunker geometry [10].

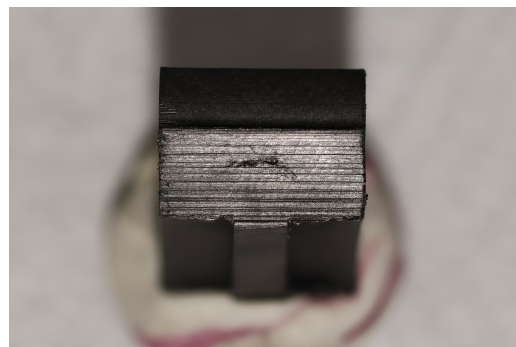
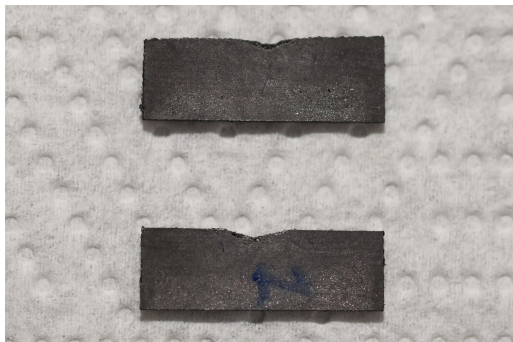
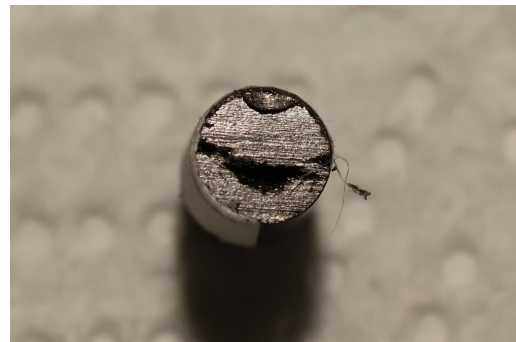


Figure 34 – Lunker defect on the gear.

The dimensions and location of the lunker defect on the machined samples are shown in Figure 35. Additionally, Figure 34 presents the defect in the corresponding area of the gear, providing further support for the observed defects in the specimens and validating the findings concerning the lunker defect [10]. Figure 36 represents an microscopic view of the lunker defect in Figure 34.



(a) Lunker defect on DMA samples.



(b) Lunker defect on pins.

Figure 35 – Lunker defects on the machined samples.

This defect is also present in POM gears, becoming more pronounced and exaggerated in PS-AC25T80, due to their low mould temperature. The problem was discovered when machining the pins for the Pin-on-Disc test, where an elliptical cavity was observed at the base, corresponding to the mid-thickness of the POM gears. Lunker defects can be exacerbated in gears injected at a moulding temperature of 80 °C, due to shorter cooling time. For the Pin-on-Disc test, this effect can be mitigated by adjusting the height of the pin base.

This defect can lead to a decrease in fatigue life, resulting in a failure in the tooth root. This phenomena is well represented in Figure 37 when comparing different defect sizes. For a torque of 60 Nm, if the lunker size exceeds 4.5 mm, the failure will occur due to root fatigue instead of common flank fatigue.



Figure 36 – Microscopic view of lunker defect.

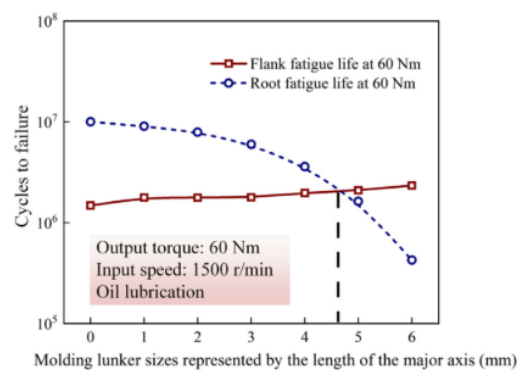


Figure 37 – Effect of lunker defect sizes on fatigue lives [10].

3.4 Dynamic Mechanical Analysis

The DMA tests were conducted at the Laboratory for Process Engineering, Environment, Biotechnology and Energy, located in the Faculty of Engineering of University of Porto (FEUP) using a Netzsch DMA 242 E Artemis, represented in Figure 38.

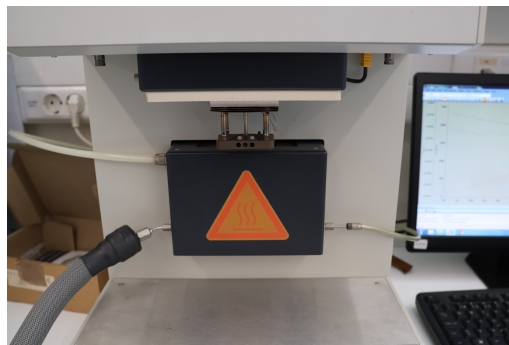


Figure 38 – Netzsch DMA 242 E Artemis.

To ensure a comprehensive characterisation, the specimens were tested under bending conditions, reflecting the mechanical stress experienced by tooth gears in

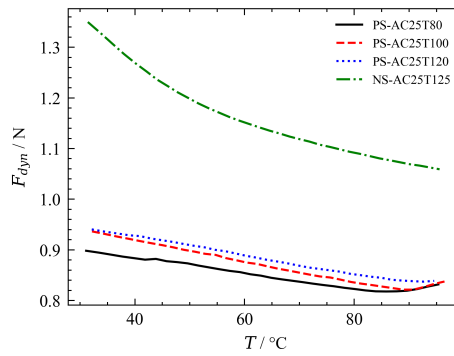
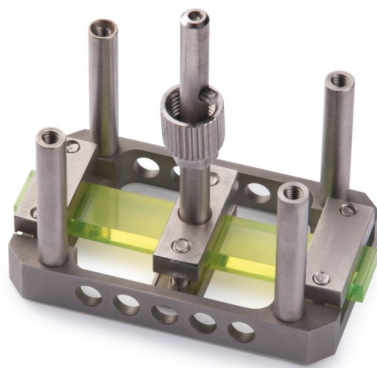


Figure 40 – Dynamic force during test.

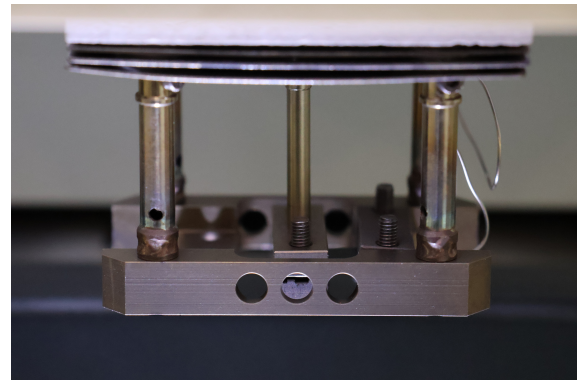
operation. The primary objective was to determine the storage modulus (E') of the polymer material obtained at different mould temperatures and conduct a comparative analyses.

The parameters extracted from the assay included the storage modulus (E'), loss modulus (E''), loss factor ($\tan \delta$), complex modulus (E^*), dynamic force during testing (F_{dyn}), and the centre point of oscillation relative to the initial point (dL).

The specimens are fixed at both ends, as represented in Figure 39(a), and the central shaft, illustrated in Figure 39(b), is responsible to introduce the testing frequency through a vertical movement, meanwhile the lateral supports serve as cantilever.



(a) Example DMA apparatus setup [35].



(b) DMA apparatus setup.

Figure 39 – DMA apparatus setup including specimen placement.

Ensuring stable dynamic force during the test is crucial for quality measurement. Figure 40 represents the dynamic force for the four conducted tests as function of temperature. The standard deviation assumes values of 2.6%, 3.6%, 3.3%, and 8.1% for the PS-AC25T80, PS-AC25T100, PS-AC25T120, and NS-AC25T125, respectively.

The testing apparatus is limited to measuring frequencies up to 100 Hz, whereas the operational frequency range of these gears spans from 100 Hz to 10000 Hz. To address this discrepancy, a Time-Temperature superposition (TTS) approach was adopted to extrapolate mechanical responses beyond the tested frequency range.

To implement TTS, a dynamic frequency-temperature was conducted to assess key mechanical parameters, including storage modulus (E'). Six frequencies were selected for TTS analysis: 1, 5, 10, 25, 33 and 50 Hz. Notably, 100 Hz was excluded from the frequency range due to irregularities observed in dynamic force measurements. Consequently, it was substituted with 33 Hz. The maximum amplitude was set at $40\text{ }\mu\text{m}$, avoiding the plastic regime, with a maximum dynamic force of 4 N. The testing temperature ranged from $30\text{ }^{\circ}\text{C}$ to $100\text{ }^{\circ}\text{C}$ with a heating rate of 1 K/min to ensure a sufficient number of data points for each frequency.

The TTS principle suggests that changes in temperature affect the time-dependent mechanical behaviour similarly to stretching (or shrinking) the effective time for temperatures above (or below) the reference temperature. Essentially, materials behave similarly at high temperature and high strain rates as they do at low temperature and low strain rates. Therefore, determining the shift factor $a_T(T)$ as a function of temperature is essential [32, 36]. One widely used equation for this relationship is the WLF equation, proposed by Williams, Landel and Ferry, and is given by:

$$\log_{10} a_T = \frac{-C_1 \cdot (T - T_{\text{ref}})}{C_2 + (T - T_{\text{ref}})} \quad (12)$$

where C_1 and C_2 represent material constants, a_T is the temperature shift factor, and T_{ref} is the reference temperature to which all data sets were adjusted.

The Netzsch DMA 242 E Artemis software is equipped with the capability to plot master curves for a reference temperature. As the tests were conducted within a temperature ranging from $30\text{ }^{\circ}\text{C}$ to $100\text{ }^{\circ}\text{C}$, three reference temperatures were chosen to represent three operational phases of polymer gears. First reference temperature equal to $40\text{ }^{\circ}\text{C}$, corresponds to low torque or low sliding speed gears. The second temperature equal to $70\text{ }^{\circ}\text{C}$, corresponds to the expected working temperature, while the last reference temperature equal to $100\text{ }^{\circ}\text{C}$, represents a temperature limit. Three master curves were plotted for each mould temperature gear.

POM samples exhibit an increase in storage modulus above $80 - 85\text{ }^{\circ}\text{C}$. Therefore, only master curves for reference temperatures of $40\text{ }^{\circ}\text{C}$ and $70\text{ }^{\circ}\text{C}$ were analysed and represented in 3.4.1. This choice aligns with the applicable range of the WLF equation. The glass transition temperature (T_g) of POM is approximately $-60\text{ }^{\circ}\text{C}$, and the WLF equation provides reliable results within the range of T_g to $T_g + 100\text{ }^{\circ}\text{C}$ [36]. For PA6-GF, this adjustment is not necessary since the T_g is about $75\text{ }^{\circ}\text{C}$, and the three master curves can be accurately plotted for the selected reference temperatures.

Another option not available in the Netzsch DMA 242 E Artemis software is the Arrhenius equation, which is useful for curve fitting shift factors outside the range of the WLF, and it is represented by the Equation 13.

$$\log a_T = \frac{E_a}{R \cdot (T - T_{\text{ref}})} \quad (13)$$

Where E_a is the activation energy, and R denotes the universal gas constant.

Subsequently, Figure 41 illustrates the initial step of TTS, grouping each of the six frequencies at the same temperature and plotting their storage modulus values versus frequencies.

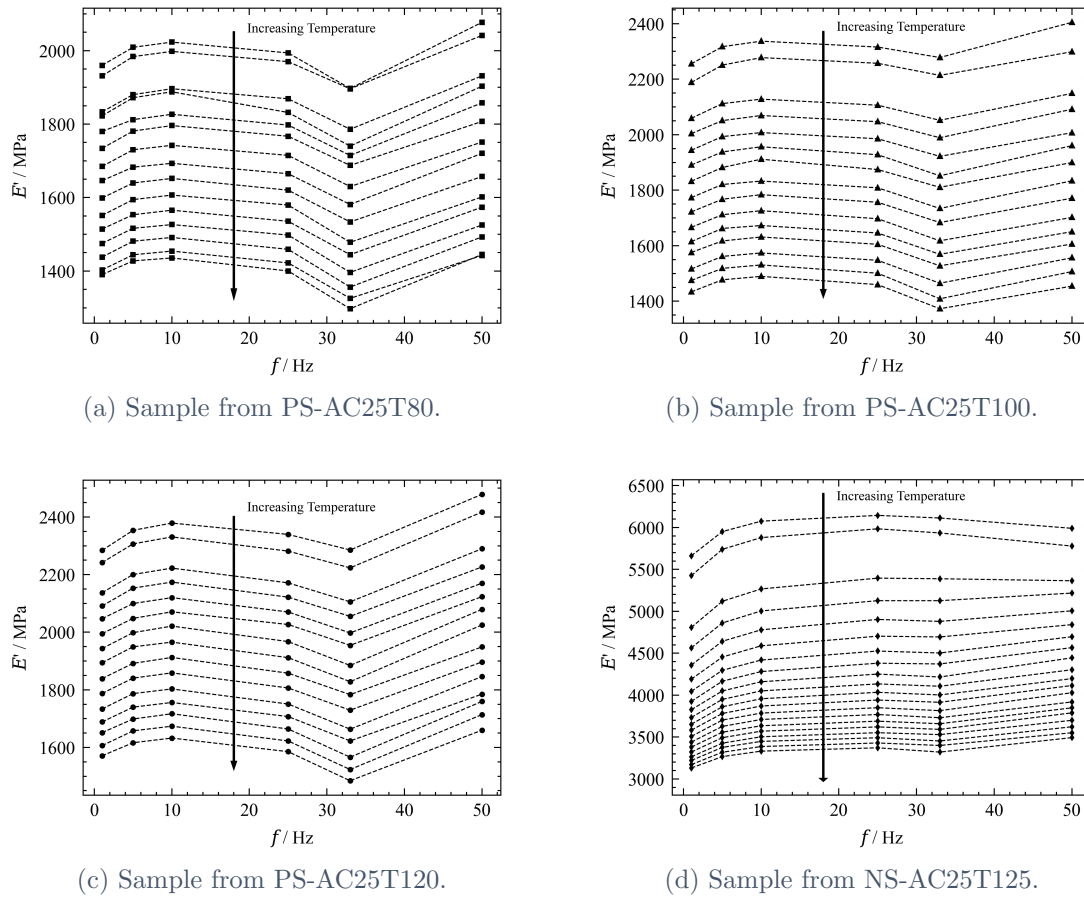


Figure 41 – Storage modulus for the same temperature but varying frequencies.

The software provided by Netzsch DMA 242 E Artemis presents the values used for determining shift factors for each measurement. These values are presented in Table 5.

Table 5 – Material constants for shift factors.

T_{ref}	40°C		70°C		100°C	
	C_1	C_2	C_1	C_2	C_1	C_2
PS-AC25T80	86.7	95.5	105.5	312.0	—	—
PS-AC25T100	84.5	69.0	105.5	115.5	—	—
PS-AC25T120	96.3	202.5	94.5	266.5	—	—
NS-AC25T125	40.0	84.0	13.5	148.0	15.0	206.5

The analysis of experimental results from the DMA tests is discussed in the following chapters, encompassing outcomes across a wide range of frequencies.

3.4.1 Results for POM samples

The DMA test results allowed a comparison of how mould temperature affects the mechanical properties of POM. The initial analysis aimed to determine if there are differences in the storage modulus, representing the material stiffness, among the three types of POM samples. A commonly used frequency for such analysis is 1 Hz. Consequently, Figure 42 presents the storage modulus as a function of frequency.

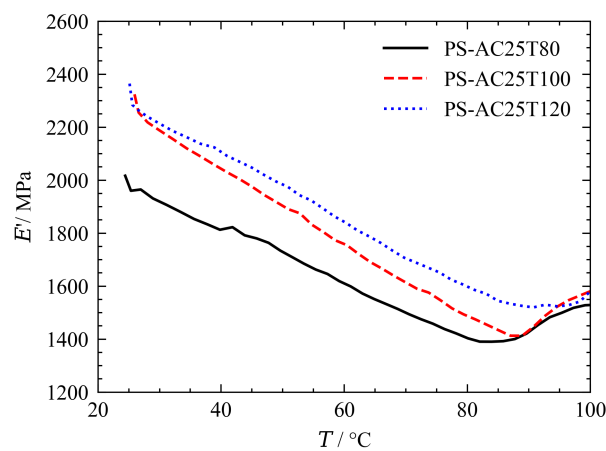


Figure 42 – Storage modulus as a function of temperature at a frequency of 1 Hz.

It is noticeable that materials moulded at lower temperatures exhibit lower values of storage modulus, whereas materials with higher mould temperatures correspond to higher values of storage modulus. This trend is consistent across all frequencies and is detailed in Appendix B. Although the storage modulus showed different values for different mould temperatures, this tendency ceases after 85 °C, at which point the three materials converge to similar storage modulus values.

Figure 42 demonstrates that higher mould temperatures result in higher storage moduli up to 85 °C. Beyond this temperature, the influence of mould temperature on E' becomes negligible, since both materials converge to similar values.

A similar approach was conducted to understand how the frequency affects the measurements of each gear. Figure 43 depicts the values of storage modulus for different frequencies, and a specific pattern can be identified across all POM samples. The E' values follow a consistent order of frequencies: 33, 1, 25, 5, 10 and 50 Hz.

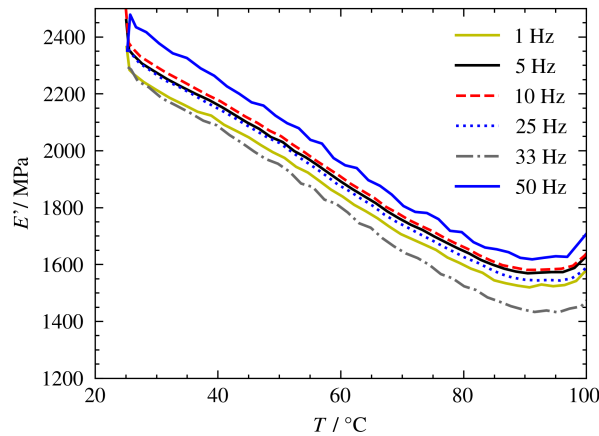


Figure 43 – Storage modulus for PS-AC25T120 at various frequencies.

Additionally, to evaluate the influence of loss modulus on the mechanical properties of the POM samples, a plot was created to compare the evolution of the storage modulus and the complex modulus. It is evident that the value of loss modulus is negligible when compared to the storage modulus, corroborating the expected behaviour for this type of material, which predominantly exhibits elastic behaviour. Figure 44 illustrates the similarity in values between the two moduli.

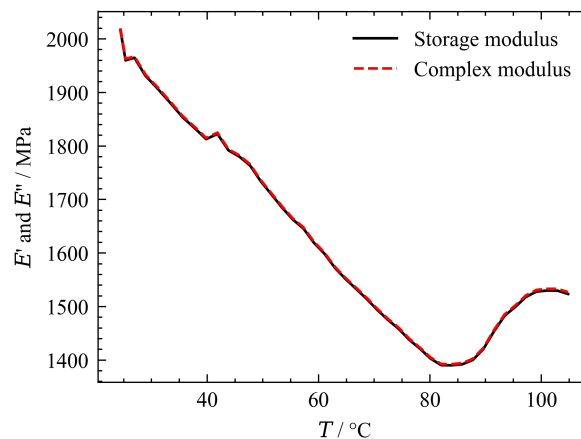


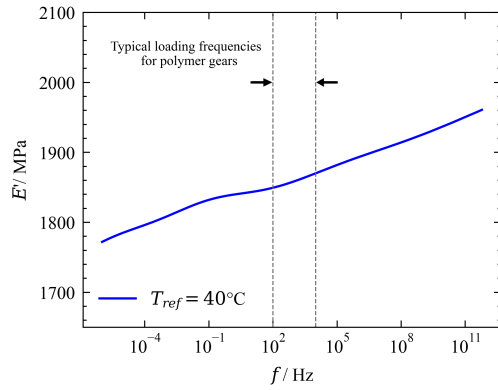
Figure 44 – Storage modulus and complex modulus for PS-AC25T80 at 1Hz.

Following the analysis in the previous chapter on the TTS method for obtaining master curves, only the master curves for reference temperature of 40 °C and 70 °C were plotted. This decision is based on the fact that shift factors for 100 °C do not accurately determine the mechanical behaviour of the POM. Conversely, the WLF equation applies well across the entire temperature range for PA6-GF.

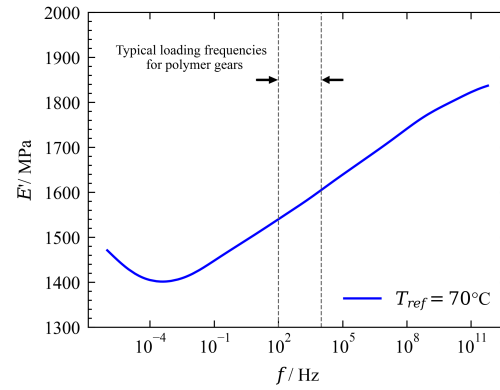
The master curves for this material cover a frequency range from approximately 10^{-5} Hz to $10^7 - 10^{11}$ Hz. To understand the frequency range during loading of a single tooth, Equation 14 relates the number of teeth (z) to the revolutions per minute (n).

$$\text{Meshing frequency} = \frac{n \times z}{60} \quad (14)$$

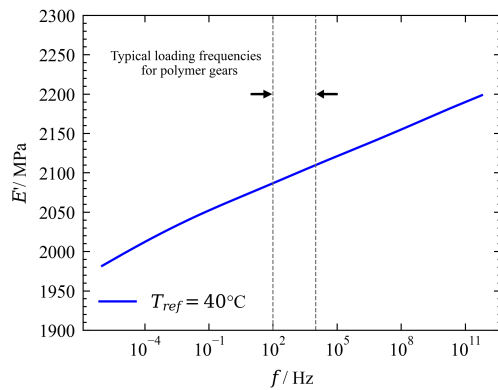
Considering a gear with 30 teeth and a conventional angular speed between 200 and 20000 revolutions per minute, the mesh frequency for a single tooth ranges from 10^2 Hz to 10^4 Hz. This frequency range is explicitly shown in all graphs related to the master curves to facilitate comprehension of the storage modulus within this range. Figure 45 represents the master curves for the three different mould temperatures.



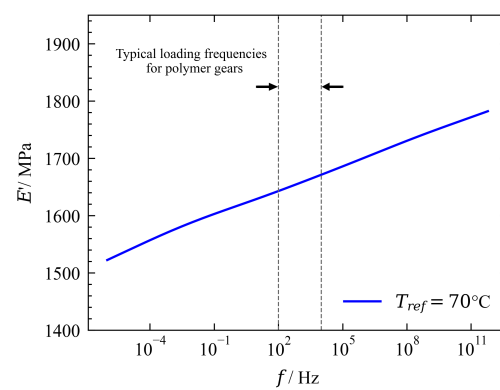
(a) Sample from PS-AC25T80.



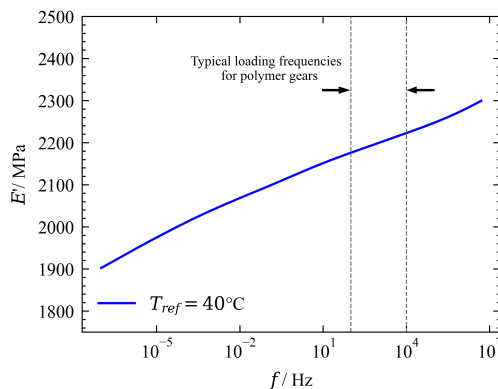
(b) Sample from PS-AC25T80.



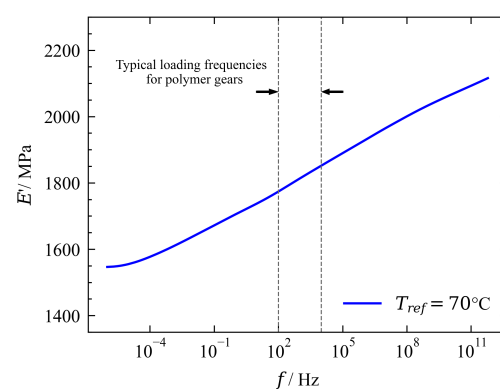
(c) Sample from PS-AC25T100.



(d) Sample from PS-AC25T100.



(e) Sample from PS-AC25T120.



(f) Sample from PS-AC25T120.

Figure 45 – Master curves at a reference temperature of 40 °C and 70 °C.

It is noticeable that for all samples and both reference temperatures, the storage modulus increases with the increase in frequency. The following evaluation compares the three POM samples at the same reference temperature, combining the previous plots. Based on Figure 46(a), an increase in mould temperature results in higher values of storage modulus for the typical loading frequencies. The material sample taken from the POM gear moulded at 80 °C and 100 °C exhibit the same slope evolution across the frequencies, while the gear moulded at 120 °C presents a steeper slope in the value of the storage modulus. At a reference temperature of 40 °C and for low frequencies, PS-AC25T100 has higher storage modulus than PS-AC25T120, with this trend reversing after approximately 10^{-2} Hz.

In a similar analysis, as shown in Figure 46(b), the three samples exhibit similar values of storage modulus at low frequencies, but this tendency changes at higher loading frequencies.

Overall, for a reference temperature of 70 °C, the storage modulus values for all samples are lower when compared to those at 40 °C. Additionally, it is evident that E' tends to converge to similar values at higher temperatures.

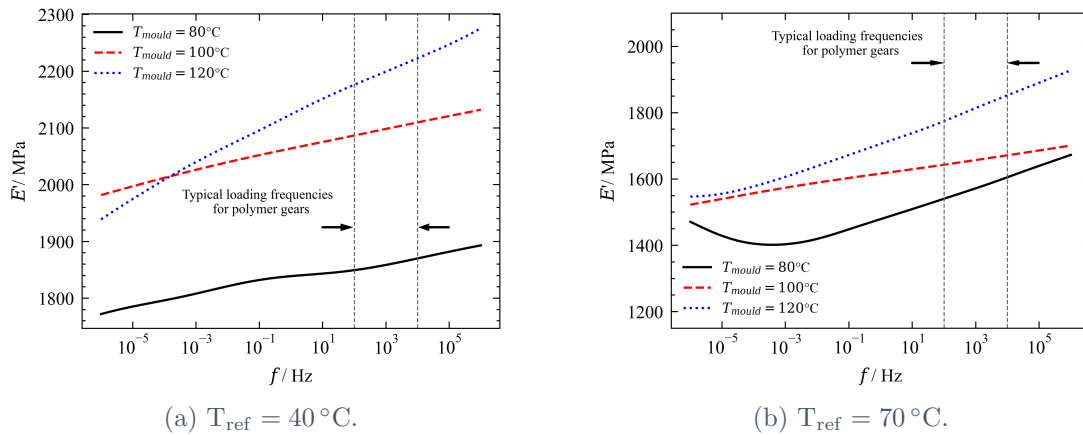


Figure 46 – Comparison of master curves for different mould temperatures.

This observation is crucial when considering the operational temperature of the gear, as selecting a gear with a low or high mould temperature can result in different mechanical properties, such as the storage modulus.

Following the discussion in this chapter, the following conclusions can be considered crucial for materials design. The most important conclusions drawn from the DMA test are:

- Higher mould temperatures during manufacturing result in higher storage modulus values until a material temperature of 85 °C;
- POM exhibits predominantly elastic behaviour, as evidenced by its storage modulus and complex modulus having similar orders of magnitude, both much greater than the loss modulus;

- All mould temperatures show an increasing storage modulus with increasing loading frequencies;
- For the same range of loading frequencies, POM has a higher storage modulus at a reference temperature of 40 °C;
- At the same reference temperature and for typical loading frequencies, higher mould temperatures result in higher storage moduli. The differences between different materials become smaller at higher reference temperatures.

3.4.2 Results for PA6-GF

Regarding the PA6-GF analysis, it was more informational than comparative, as it was conducted on a single PA6-GF specimen. The results are described in this chapter to provide a basis for future work and tests. Even so, a comparison between POM and PA6-GF materials is carried out in Chapter 3.4.3.

To begin the evaluation, a plot at 1 Hz was created to understand the trend in storage modulus values, as shown in Figure 47. For PA6-GF, the curves exhibit higher values with increasing frequency. As expected, the storage modulus for this material is higher compared to POM, but temperature has a greater influence on E' . This indicates that PA6-GF is more suitable for designs where the temperature remains constant over time, whereas POM is better suited for cases with fluctuating temperatures.

The trend observed at 1 Hz is consistent across other frequencies, as represented in Figure 48.

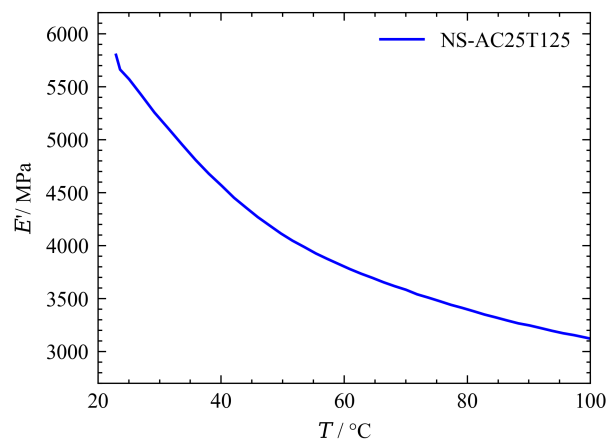


Figure 47 – Storage modulus for NS-AC25T125 as a function of temperature at 1 Hz.

Additionally, and similarly to the analysis conducted on POM samples, an evaluation of the loss modulus for PA6-GF samples was performed. In contrast, this material shows a slight variation between the storage modulus and the complex modulus, as the loss modulus values are higher compared to complex modulus. Figure 49 depicts the evolution of storage and complex modulus with respect to temperature.

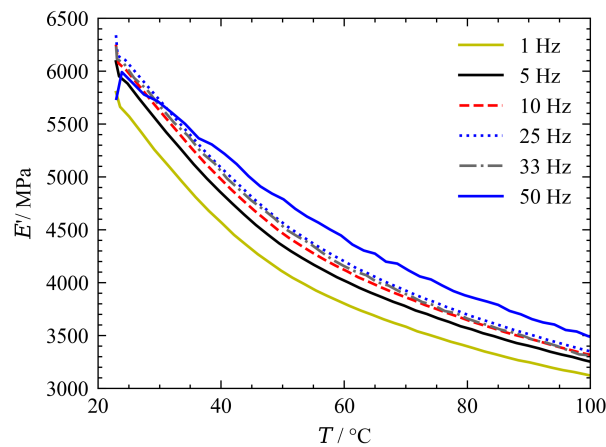


Figure 48 – Storage modulus for NS-AC25T125 at various frequencies.

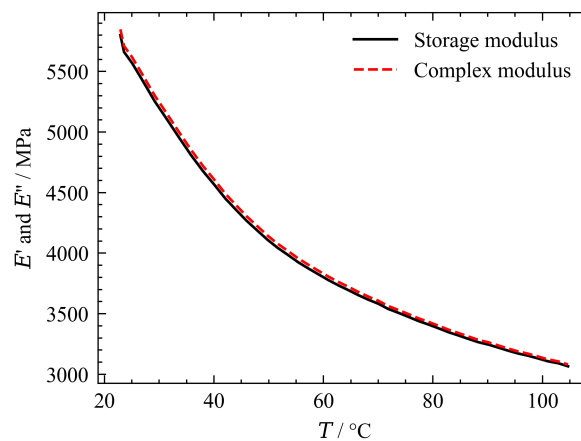


Figure 49 – Storage modulus and complex modulus for PS-AC25T80 at 1 Hz.

For PA6-GF, the storage modulus decreases with increasing temperature, revealing a different behaviour compared to POM.

Consequently, it was possible to determine the master curves for PA6-GF samples at the three reference temperatures: 40 °C, 70 °C and 100 °C. Furthermore, the glass transition temperature of PA6-GF is higher, and the shift factors fit well within the WLF equation, resulting in high reliability of the master curves [36].

Considering that PA6-GF samples have the same specifications as POM samples, Equation 14 is also applicable in this case, resulting in the same frequency range. Figures 50–51 illustrate the typical loading frequencies for PA6-GF gears. It is worth noting that the graph corresponding to a reference temperature of 40 °C only contains data up to 3.0×10^3 Hz, so the selected zone only extends up to that value.

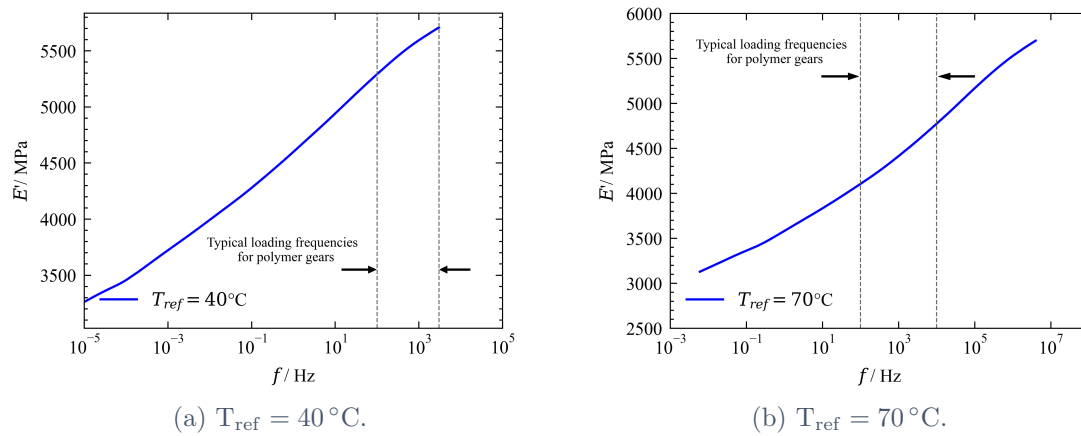


Figure 50 – Master curves for NS-AC25T125.

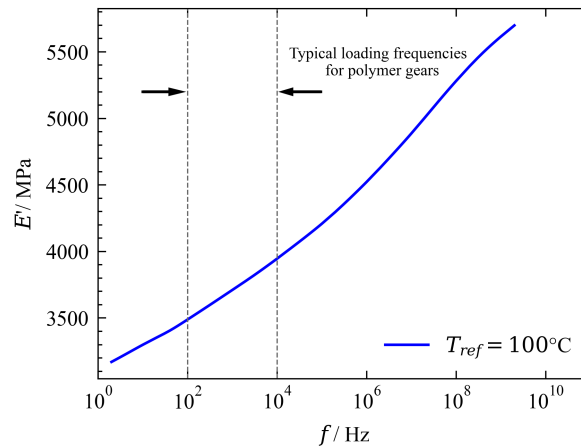


Figure 51 – Master curves at a reference temperature of 100 °C for NS-AC25T125.

As demonstrated in Figures 48–51, the PA6-GF samples exhibit a predictable evolution within the analysed temperature range. This assessment can support mechanical design, though further tests are necessary to evaluate the reliability of these measurements.

3.4.3 Comparative Results of POM and PA6-GF

For the comparative analysis of POM and PA6-GF, only the values obtained for PS-AC25T120 and NS-AC25T125 are considered to simplify the comparisons. The POM gear with highest mould temperature was chosen, as it exhibits the highest mechanical properties, making it the closest to Nylon in terms of mechanical properties.

The first analysis involves comparing the storage modulus versus temperature for both materials. As shown in Figure 52, the storage modulus for PA6-GF is consistently higher across the entire temperature range. The frequency of the data plotted is 1Hz.

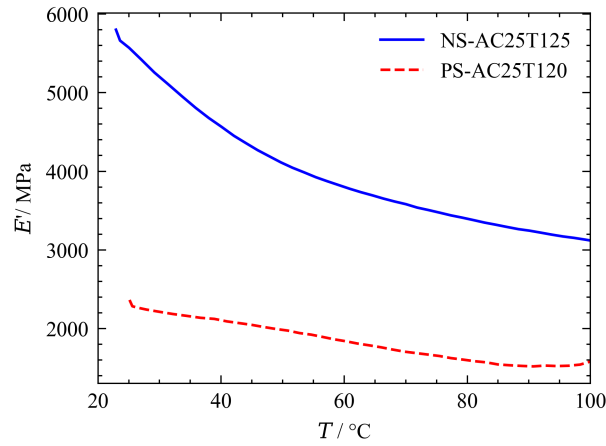


Figure 52 – Storage modulus for PA6-GF and POM at a frequency of 1 Hz.

The second phase involves determining the typical loading frequencies using master curves, similar to the process conducted separately for POM and PA6-GF samples.

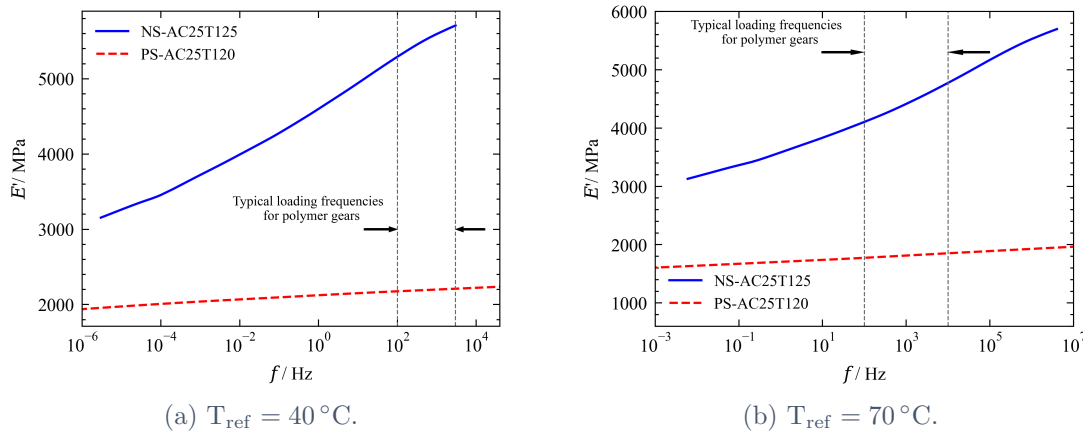


Figure 53 – Master curves comparison for NS-AC25T125 and PS-AC25T120.

The discrepancy observed for typical loading frequencies is significantly greater compared to the differences at testing frequencies. In Figure 53, both graphs indicate that the storage modulus for PA6-GF is approximately 2 to 3 times higher than that of POM. This finding suggests that a gear made of PA6-GF will possess about 2 to 3 times more stiffness than a POM gear at the same loading system, which is a crucial consideration in the mechanical design of a gear system.

3.5 Differential scanning calorimetry

This test was employed because POM and PA6-GF are semi-crystalline polymers that undergo crystallization during cooling. The mould temperature can influence the speed and extent of crystallization, thereby altering mechanical properties such as stiffness and impact resistance. It is therefore important to understand whether

there are differences in mechanical properties among mould temperatures of 80 °C, 100 °C and 120 °C. On the other hand, PA6-GF was analysed to elucidate the behaviour of this material.

The DSC tests were conducted at the Laboratory for Process Engineering, Environment, Biotechnology and Energy, located in the Faculty of Engineering of University of Porto (FEUP) utilising a Netzsch DSC 214 Polyma, illustrated in Figure 54. Four samples were used, corresponding to each mould temperature of POM and one for PA6-GF. Each sample was extracted from probes designated for the DMA test, with dimensions of 5×0.5 (diameter $\times$ thickness).



Figure 54 – Netzsch DSC 214 Polyma.

The mass of each sample and its respective crucible is represented in Table 6, with a reference crucible mass of (51.9 mg).

Table 6 – Samples mass.

Sample	Sample mass (mg)	Crucible mass (mg)	Total mass (mg)
PS-AC25T80	7.5	52.2	59.7
PS-AC25T100	7.3	52.8	60.1
PS-AC25T120	7.4	52.2	59.6
NS-AC25T125	8.4	52.4	60.8

Regarding the test parameters, for POM, given its melting temperature of 166 °C, an initial heating cycle was conducted from 20 °C to 200 °C to ensure the melting point was attained. Subsequently, a cooling cycle to −40 °C was performed, with a 5-minute stage to ensure sample homogeneity. Although the glass transition temperature of POM is −60 °C, this point was not attained due to apparatus limitations. A third stage involved raising the temperature back to 200 °C. All stages were conducted at a heating rate of 10 K/min.

The PA6-GF samples exhibit a melting point between 270 °C and 285 °C, with a glass transition temperature of 75 °C. The same procedure was followed for this material, with an initial stage from 30 °C to 300 °C, followed by a cooling cycle to

ambient temperature of 20 °C, including a 5-minute stage to achieve sample homogeneity, and finally, a heating cycle from 20 °C up to 320 °C. All stages were conducted at a heating rate of 10 K/min.

3.5.1 Results for POM samples

The results pertaining to POM materials are categorized into three main cycles. The first cycle, as depicted in Figure 55, aims to elevate the specimen's temperature above the melting point. The objective is to investigate whether the mould temperature during injection influences the melting point. Figure 56 represents the enthalpy variation during this endothermic process, with corresponding values expressed in Table 7.

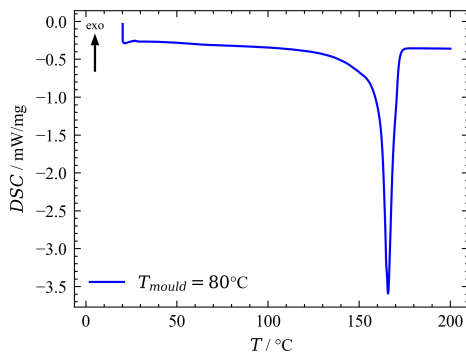


Figure 55 – First heating cycle.

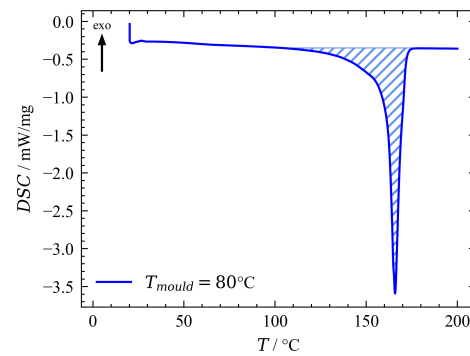


Figure 56 – Enthalpy for melting point.

The second cycle, represented in Figure 57, corresponds to the cooling process. An exothermic reaction occurs during this stage, associated with the crystallisation of the material at approximately 148 °C. Figure 58 provides insight into the enthalpy changes during this exothermic process.

Following the cooling phase, a brief stabilisation stage lasting 5 minutes was conducted to homogenise the temperature. Unfortunately, the glass transition temperature could not be achieved due to insufficient nitrogen to decrease the temperature beyond -60 °C.

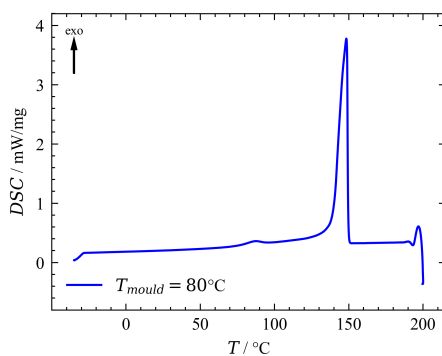


Figure 57 – Second cycle for cooling.

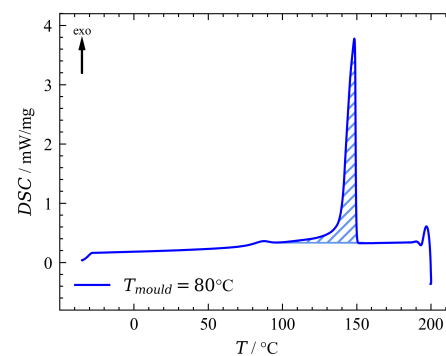


Figure 58 – Enthalpy for crystallisation point.

Figure 59 represents the third cycle, wherein the temperature is elevated from -40°C to 200°C . Another endothermic process occurs as the melting point is attained. The integral of this endothermic reaction is presented in Figure 60.

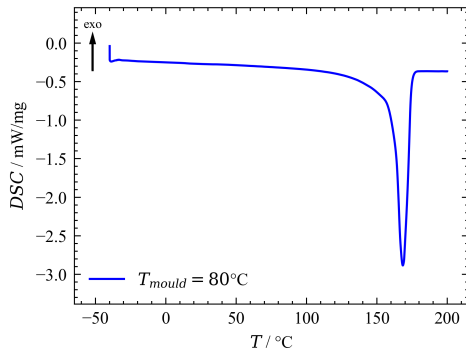


Figure 59 – Third heating cycle.

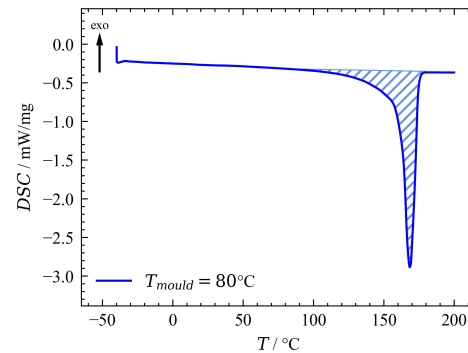


Figure 60 – Enthalpy for third cycle melting point.

Figure 61 offers a comparative analysis of the endothermic processes. Notably, the predicted melting temperature is 166°C , and the values for all three samples align with this condition. Enthalpy values for each specimen are documented in Table 7. It is important to note that the y-axis direction indicates the exothermic direction of reactions. Therefore, all values are presented in their absolute form, with clear indication of whether they correspond to exothermic or endothermic reactions.

Upon initial examination, utilising the information provided by the Table 7 and the plot of all curves in the subsequent Figure 61, it is possible to conclude that mould temperature during injection does not yield dispersed measurements.

However, it is crucial to recognise that after exceeding the melting point and undergoing crystallization (i.e., after the heating and cooling cycle), the material loses the properties it previously acquired through the injection process. The tendency is for the materials to exhibit similar curves during the third heating cycle, serving as a reference for the first cycle.

Considering the third cycle as reference, as depicted in Figure 63, the ascending order for absorbed energy and melting temperature, as functions of mould temperature, are 100°C , 120°C and 80°C , respectively.

In contrast to the first heating cycle, represented in Figure 61, it is evident that the ascending order for absorbed energy and melting temperature no longer holds. This change validates that mould temperature can indeed influence the values of melting temperature and consequent enthalpy during this process.

For a more comprehensive understanding on this topic, further investigation must be conducted.

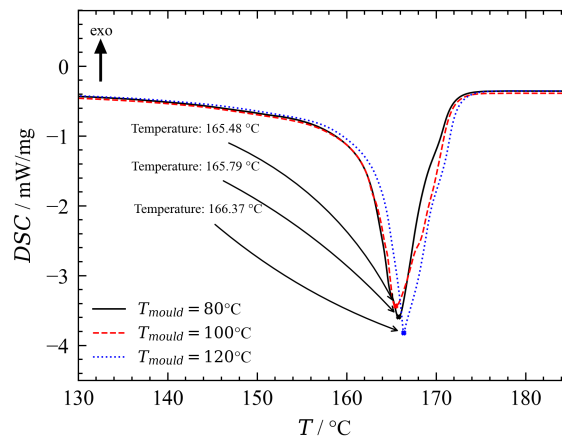


Figure 61 – First heating cycle for all POM samples.

Regarding the crystallisation point, the temperature values remain similar across the three samples, but the energy released during this reaction varies among them.

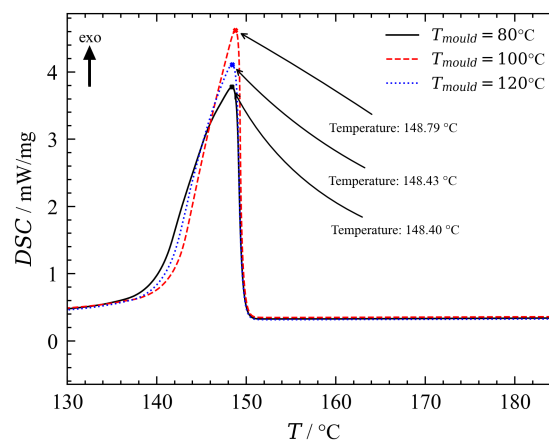


Figure 62 – Cooling cycle for all POM samples.

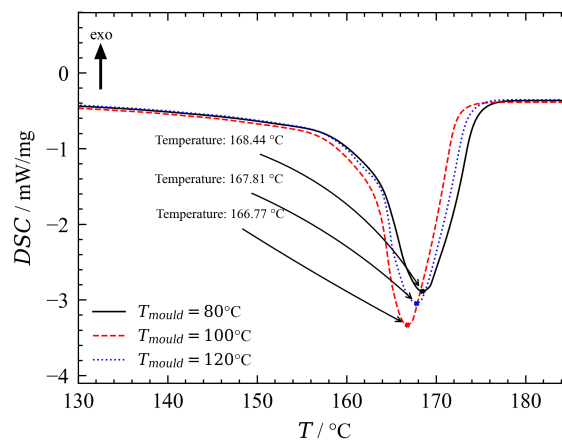


Figure 63 – Third heating cycle for all POM samples.

Table 7 – Enthalpy for each cycle.

Enthalpy of the cycle (J/g)	First (endothermic)	Second (exothermic)	Third (endothermic)
PS-AC25T80	171.8	181.3	178.9
PS-AC25T100	170.6	183.5	173.7
PS-AC25T120	175.9	180.3	177.5
NS-AC25T125	24.1	26.2	29.2

Another property that can be calculated from DSC results is the crystallinity fraction (χ_c) of the material, using Equation 15.

$$\chi_c = \frac{\Delta H_m}{\Delta H_{m0}} \quad (15)$$

Where ΔH_m represents the melting enthalpy obtained from the DSC test, and ΔH_{m0} is the enthalpy of fusion of a fully crystalline sample of the same polymer [37].

For POM single percentage of crystallinity fraction was calculated. The ΔH_{m0} value cannot be experimentally determined due to the difficulty in obtaining fully crystalline samples, so a value 326 J/g was assumed based on literature [38].

Using the values obtained in Table 7 for PS-AC25T80, the first cycle corresponds to 171.8 J/g, resulting in a crystallinity fraction of 0.53 or 53%. This conclusion supports the classification of POM as a semi-crystalline material.

Although the enthalpy values among different POM specimens is very similar, these differences can impact the mechanical properties. This is because the crystallinity of these samples varies. For the three mould temperatures considered, the gear with highest crystallinity is the PS-AC25T120, followed by PS-AC25T80, and finally the PS-AC25T100.

3.5.2 Results for PA6-GF

The procedure for PA6-GF is similar to that executed for POM samples. However, the enthalpy curves for PA6-GF do not approximate the POM curves, as PA6-GF exhibits a variation of mW/mg with temperature. This necessitates a virtual line for the integral that starts at the onset temperature and ends at the end temperature, even if they have different values of mW/mg.

The enthalpy values obtained for each cycle are also described in Table 7.

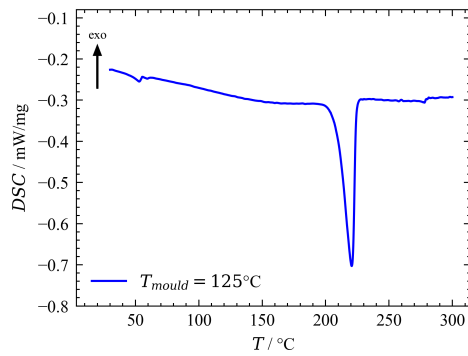


Figure 64 – First heating cycle for PA6-GF.

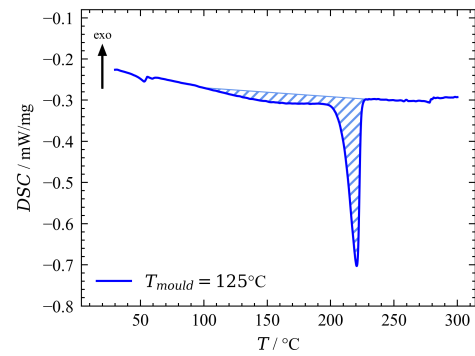


Figure 65 – Enthalpy for first melting point.

Comparing the reference curve of the third cycle (Figure 68) with the first heating cycle (Figure 64), melting occurs at the same temperature, but the energy absorbed is not the same. The valley for the first cycle is deeper than the third cycle, although the calculated enthalpy is higher for the third cycle, indicating that the endothermic reaction took longer in this cycle, with an onset point at 100 °C and an end point at 230 °C.

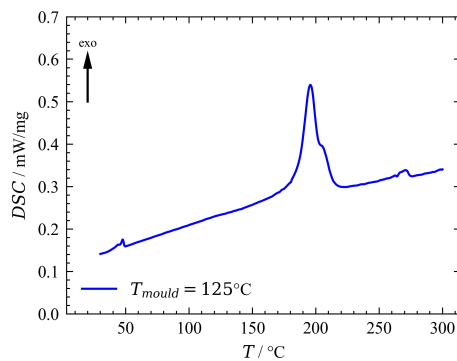


Figure 66 – Cooling cycle for PA6-GF.

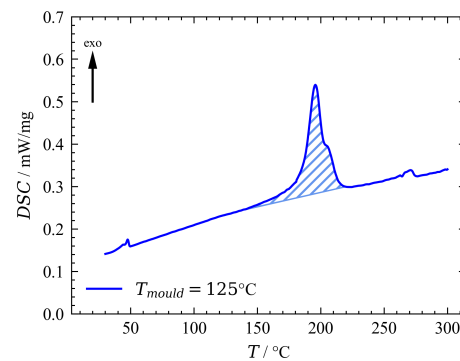


Figure 67 – Enthalpy during crystallisation.

Since crystallisation takes place during cooling, it is considered crystallisation rather than cooling crystallisation, resulting in similar enthalpy values relative to the endothermic reaction.

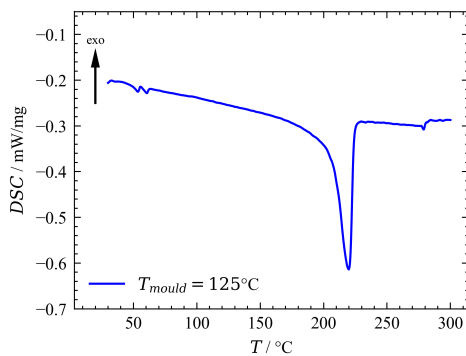


Figure 68 – Third heating cycle for PA6-GF.

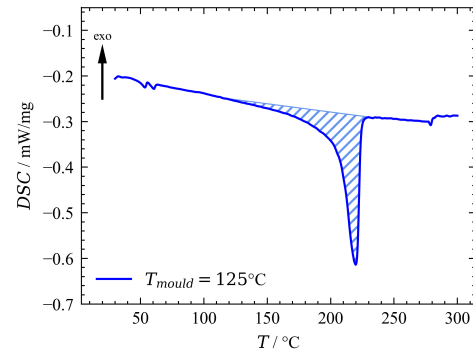


Figure 69 – Enthalpy for second cycle melting point.

Regarding the crystallinity fraction for PA6-GF, the ΔH_{m0} is still unknown due to the composite nature of the material, which includes polyamide and small quantities of glass fiber. Therefore, the ΔH_{m0} considered is that of PA6 with a value of 230 J/g [38].

For the first cycle, as shown in Table 7, ΔH_m is equal to 24.1 J/g, resulting in a crystallinity fraction of 0.105, or 10.5%.

3.5.3 Comparative Results of POM and PA6-GF

Based on the Table 7, it is evident that the absorbed energy during the endothermic reaction is considerably different between POM and PA6-GF. However, this reaction does not occur at the same temperature, with PA6-GF reaching its melting point at higher temperatures. This difference should be reasonably considered in mechanical design, since PA6-GF exhibits higher stiffness values than POM, although this difference decreases with increasing temperature.

Figure 70 also illustrates the crystallinity values of both materials. PA6-GF has a lower valley, making it easier to break the molecular structure order, whereas POM has a higher valley. The conclusions mentioned are also applicable to Figure 72.

Regarding the exothermic process, demonstrated in Figure 71, both materials undergo crystallisation at similar temperatures, and the energy released during this reaction is consistent with the energy absorbed during fusion for both materials. This indicates that the crystallinity of the materials before the test is similar to that obtained after the process.

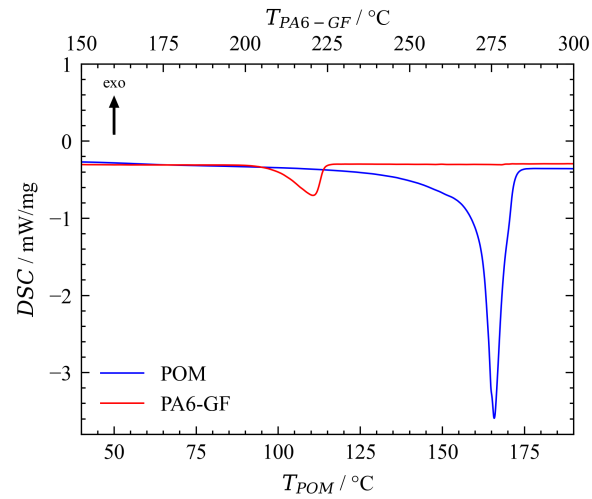


Figure 70 – Comparison of the first heating cycle between POM and PA6-GF.

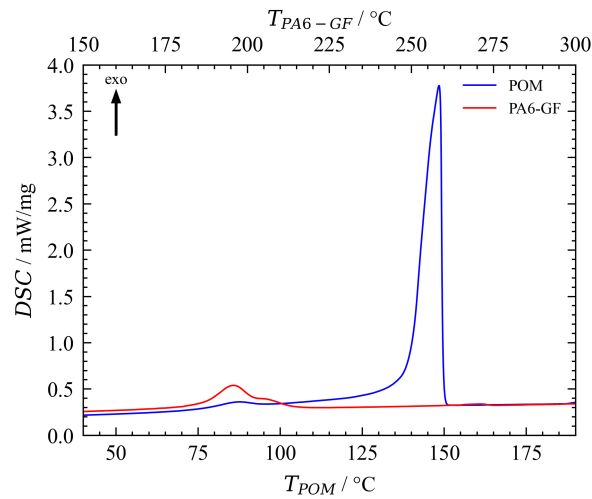


Figure 71 – Comparison for the cooling cycle between POM and PA6-GF.

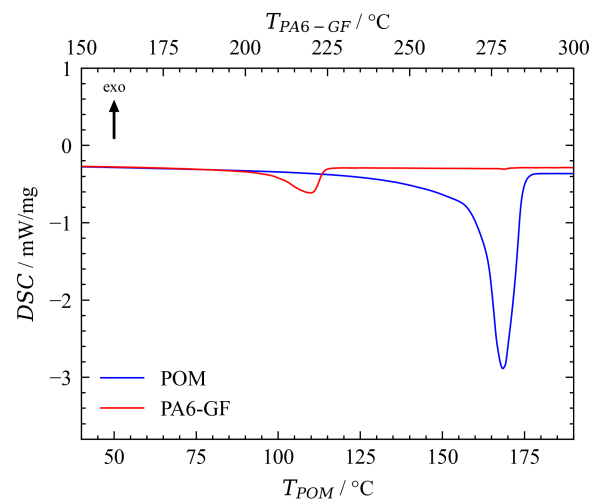


Figure 72 – Comparison of the third heating cycle between POM and PA6-GF.

3.6 Shore D hardness test

A Shore D hardness test was conducted to assess the surface hardness of both POM and PA6-GF. The hardness tests were executed following ISO 7619-1:2004 guidelines, using the Shore D hardness apparatus from *Hildebrand*, depicted in Figure 73. The measurement of surface hardness was conducted in the lateral area of the gears, perpendicular to their rotation axis. This specific zone was chosen due to its compatibility with the testing standards employed. It is expected that the hardness results obtained from this zone will closely align with the values of the teeth flank.



(a) Hardness Shore D setup.



(b) Hardness Shore D measuring scale.

Figure 73 – *Hildebrand* Hardness Shore D apparatus.

The Shore D hardness test was carried out with a load of 4 kg, and both the maximum hardness value ($t = 0\text{ s}$), the value after 3 seconds, and the value after 15 seconds were recorded. Typically, these two values differ due to the material's elastic recovery over time. Five measurements were conducted for each material and extended table is represented in Appendix B.

3.6.1 Shore D hardness test results

The average hardness values for each sample are represented in Table 8.

Table 8 – Average Shore D hardness values.

Sample	$t = 0\text{ s}$	$t = 3\text{ s}$	$t = 15\text{ s}$
PS-AC25T80	83.0	78.4	77.6
PS-AC25T100	80.9	77.5	76.7
PS-AC25T120	82.7	78.7	78.0
NS-AC25T125	74.4	70.7	70.0

The standard deviation for all measured values is below 0.663, with a variance below 0.440. Figure 74 represents a comparison based on the values from Table 8.

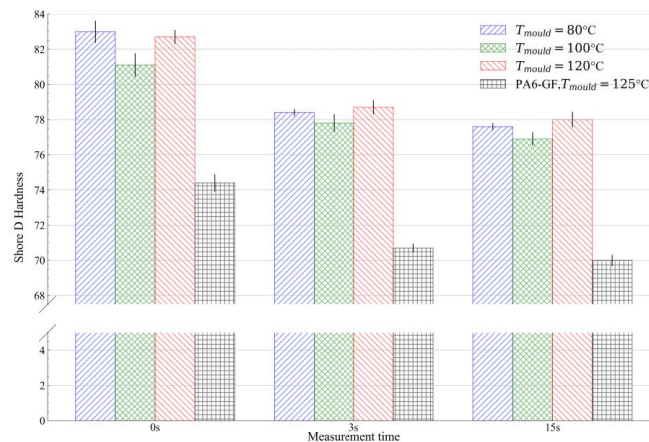


Figure 74 – Comparison of average hardness values for all samples.

Upon analysing the data and comparing it with the results obtained from DSC tests, several conclusions can be drawn. Regarding surface hardness, the POM material with a mould temperature of 80°C exhibits the highest average value at the initial instant ($t = 0\text{ s}$). However, this tendency changes with the elastic recovery effect. At instants $t = 3\text{ s}$ and $t = 15\text{ s}$, the polymer with highest surface hardness has a mould temperature of 120°C .

For all these instants, the gear with lowest hardness is the PS-AC25T100, which has a mould temperature of 100°C . Therefore, the decreasing order of surface hardness for $t = 0\text{ s}$ based on mould temperatures is 80°C , 120°C and 100°C , whereas for instants $t = 3\text{ s}$ and $t = 15\text{ s}$ it is 120°C , 80°C and 100°C .

Recapitulating the results from the DSC test, and considering that the POM with a higher mould temperature had a higher melting enthalpy, its crystallinity will also be higher. Given that hardness increases with increasing crystallinity, it is possible to corroborate through the results obtained from both tests that the evolution of hardness (in increasing order) follows the mould temperature of 100°C , 80°C , and 120°C .

For PA6-GF, the hardness values tend to be lower than those for POM. This result is particularly surprising since hardness data is not provided by the supplier. When comparing the tensile modulus of the two materials, the difference is signif-

icant: PA6-GF exhibits a tensile modulus of 14 GPa, whereas POM has a tensile modulus of 2.9 GPa. However, this difference in tensile modulus does not translate to a corresponding difference in hardness between the two materials.

Despite these conclusions, it is important to note that the DSC tests were carried out on only one sample for each mould temperature with a screening sample. Therefore, a future study with greater repeatability is necessary to provide more reliable conclusions.

4 TRIBOLOGICAL CHARACTERISATION

The objective of this test is to measure the wear of materials depending on their mould temperature. The experimental tests were conducted following the guidelines outlined in ASTM G99 - 17 [39]. However, instead of employing a spherical pin, a cylindrical pin was used due to challenges associated with manufacturing the spherical pin tip.

During the practical experiments, the mass and roughness profile of every pin were controlled to evaluate the results before and after the experiments. Moreover, a thermal camera will record the temperature throughout the test to determine if there is a noticeable difference between test conditions or materials. To ensure accurate image capture and avoid any interference, the screw responsible for fixing the disc to the machine will be made of the same material as the disc, thereby maintaining the same emissivity.

A balance with 10^{-4} g precision was used, as depicted in Figure 75, and two different methods of roughness profiling were employed (2D and 3D).



(a) Balance setup.



(b) Balance sensitivity.

Figure 75 – Balance setup and range scale.

4.1 Materials and specimens

In this study, a cylindrical pin is used, resulting in a non-linear contact between two surfaces, as opposed to the conventional Hertzian contact. This deviation from the standard occurred because it was challenging to obtain a sphere by machining

the gears. This alteration opens up new possibilities, such as evaluating a potential wear angle, as referenced in the article published by *H. Benabdallah* [40].

All pins were extracted from the same gears used for DMA and DSC, having tip dimensions of 3×2 (diameter $\times$ length), in millimetres. The detailed drawing of the pin is represented in Figure 31 and the extraction local on the gear is represented in Figure 30. These dimensions are then used to calculate the contact pressure and loading force.

From the polymer gear, 12 pins were extracted. To control the tip roughness of the pin, the tip's face is coincident with the gear's lateral face, aiming to guarantee the same roughness for all pins.

Due to machinability difficulties, the final step of extracting the pins from the gear was performed manually with a 0.5 mm thick saw. Consequently, the top and base of the pin did not have parallel faces, complicating the roughness measuring process. To overcome this problem, a polymer spacer, also used during Pin-on-Disc tests, was used to secure that the pin tip was perpendicular to the optic camera of the NPFLEX-1000. The polymer spacer is illustrated in Figure 76(b).

Since only 9 pins were needed to execute the tests, a 3D roughness profile was conducted to ensure that 9 pins with similar roughness were selected for the test, thus ensuring that roughness would not be a differentiating parameter for results. Table 11

The polymer pin's surface roughness before the tests is represented in Table 12.

Due to the limited availability of pins, the tests were exclusively conducted using a steel disc. This choice avoid the use of other potential combinations, such as POM/POM, POM/PA6-GF, and PA6-GF/POM.

The steel disc used in the study measured 75 mm in diameter and 8 mm in thickness. The disc's average roughness is $0.1 \mu\text{m}$. Since availability of discs is limited, the same disc was used with 3 different tracks in each side, ensuring that the angular velocities were as desired, maintaining the convection coefficient constant across distinct combinations. These test tracks had diameters of 30 mm, 44 mm and 60 mm.

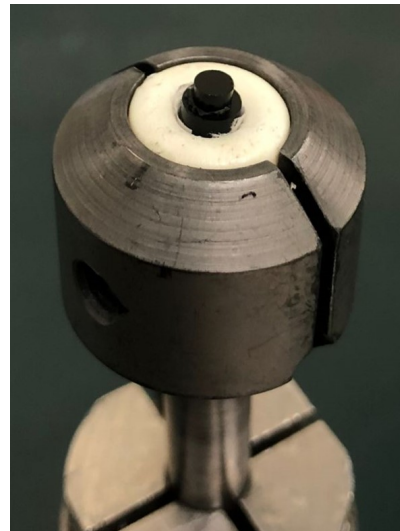
4.2 Test conditions

The tribological tests were conducted using a pin-on-disc machine, depicted in Figure 76(a) under dry-sliding conditions at room temperature ($19 \pm 1^\circ\text{C}$). The testing apparatus comprises a turntable that accommodates the steel disc, driven by a motor.

The pins were secured on the upper pivoting arm using a polymer spacer, which aimed to limit pin displacement in all directions during tightening, as shown in Figure 76(b). The normal load was applied above the pin shaft.



(a) Pin-on-Disc setup.



(b) Pin and spacer setup.

Figure 76 – The Pin-on-Disc apparatus and setup.

Firstly, the parameters for load and sliding velocity need to be determined to resemble those of an actual working case. Analysing the results obtained by *K. Mao et al.*, it was observed that POM gears exhibit a transition for a torque equal to 8.5 Nm, resulting in a significant increase in wear rate (see Figure 24) [25]. Figure 77 illustrates the wear evolution during 2.14×10^6 cycles for the same sliding velocity, clearly depicting the three phases of wear on a gear. The second phase, characterised by constant linear wear, ends at 8.5 Nm, thus serving as a maximum reference value for the torque imposed on the gears [8].

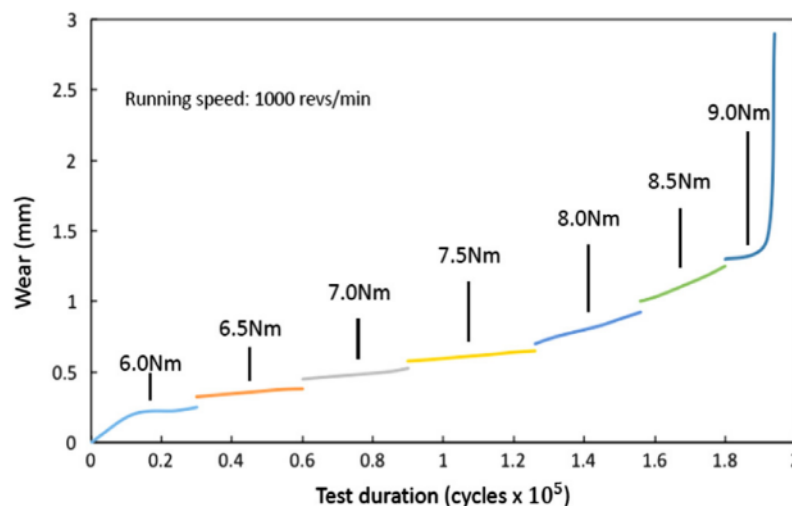


Figure 77 – Wear evolution for different torques [8].

Furthermore, considering the Figure 77, a value of 6.5 Nm was selected as the minimum reference for use in tests. However, after conducting a mechanical analysis using *KISSsoft* software, it was observed that there was a narrow gap for contact pressures between 6.5 Nm and 8.5 Nm. Consequently, it was decided to expand the torque values to cover a wider range of contact pressures.

This analysis was conducted using two POM gears with the specifications represented in Table 3. A dry contact was established, and the contact pressure on the tooth flank was obtained.

For torque values of 0.5 Nm, 4 Nm and 7.5 Nm, corresponding pressures of 6 MPa, 17 MPa and 23.3 MPa, were applied to the gear flanks, respectively.

A similar analysis was conducted for PA6-GF. Although, initial tests revealed that wear in PA6-GF was more severe than anticipated, necessitating a reduction in contact pressure during the Pin-on-Disc test. Consequently, the chosen contact pressures for PA6-GF were 1 MPa, 3 MPa and 6 MPa.

For each required contact pressure, corresponding masses were added to the machine based on the contact area of the pin. This adjustment resulted in achieving the desired contact pressure.

The test parameters used are outlined in Table 9. The initial approach involved using a contact pressure of 23.3 MPa, corresponding to a torque of 7.5 Nm. However, the results were unreliable, as the apparatus could not properly secure the pin throughout the test duration. The pin deflected from its vertical alignment, and observed wear was not as desired. Figure 78 illustrates the wear on the pin after 900 seconds of testing with 23.3 MPa of contact pressure. Furthermore, the observed final shape and surface aspect of the pin indicate that the outcome is primarily attributed to plastic deformation resulting from the elevated temperatures (115.8 °C), rather than wear during the test. Therefore, a torque of 2 Nm was chosen for the tribological testing, corresponding to a pressure of 12 MPa.



Figure 78 – Pin after 900 seconds of test.

Each run was repeated three times to comply with the minimum replicates required by the standard.

For the presented parameters, different disc radii were used due to the limited availability of discs. To prevent a rise in temperature resulting from different convective heating, combinations with higher sliding speeds were tested on larger diameters, resulting in similar angular velocity across combinations. This approach aimed to achieve similar convective heating in the disc, permitting it to heat without the influence of the angular velocity.

Table 9 – Test parameters used.

Parameter	Unit	Value		
Contact pressure for POM	MPa	6	12	17
Contact pressure for PA6-GF	MPa	1	3	6
Sliding speed	m/s	0.3	0.5	1
Sliding distance	m	1000		
Pin-tip diameter	mm	3		
Ambient temperature	°C	19 ± 1		
Track-diameter	mm	30 – 60		

Table 10 delineates the disc radii used for each combination.

4.3 Design of Experiments

The desired number of different loads applied to the pin and sliding speeds applied to the disc was 3 for each parameter. This configuration would result in 9 combinations for each material depending on their mould temperature (80 °C, 100 °C and 120 °C, for POM and 125 °C for PA6-GF). The pins were extracted for the same gears used in the DSC and DMA tests. However, if three replicates are imposed for each test condition, a total of 108 measurements would need to be conducted (9 combinations of parameters, 4 materials, and 3 replicates). This configuration is not feasible to execute within the available time frame. Therefore, a Design of Experiments (DOE) was implemented to minimise the number of runs.

The chosen method for applying a Design of Experiments was the Taguchi methods, that is a statistical methodology developed by Genichi Taguchi to improve the quality of manufactured good [41].

The first step is to identify the factors (variables) that affect the process or product, and determine the levels (settings) for each factor. Factors can be controllable (temperature, speed and pressure, for this work) or uncontrollable (air humidity, roughness and ambient temperature). As demonstrated in Table 9, the factors chosen are: mould temperature, pressure and velocity, with 3 levels each, one low, one medium and one high.

Taguchi methods uses orthogonal arrays, that must be balanced to ensure that each factor's levels are tested equally and independently, allowing to study multiple factors with a reduced number of experiments. The L9 orthogonal arrays with 3 replicates is conducted, resulting in 27 tests.

After the tests have been carried out, and values of mass loss and temperature analysed, statistical methods were employed to determine the effect of each factor

on the response variable. The main goal is to identify the optimal levels, that reduce wear and temperature, and in a future work try to determine if the given function can predict exactly the outcome with certain conditions of mould temperature, pressure and sliding speed.

Considering the previous description about DOE, Taguchi methods offers the combinations represented in Table 10.

Test runs numbered up to 9 were conducted on POM pins, while the remaining three tests were conducted on PA6-GF pins. Each of these runs was repeated three times to adhere to the principles of the Taguchi methods.

Table 10 – Test run parameters.

Run	Mould Temperature (°C)	Sliding speed (m/s)	Pressure (MPa)	$p \cdot v$ (MPa · m/s)	Track radius (mm)
1	80	0.3	12	3.6	15
2	80	0.5	6	3	22
3	80	1	17	17	30
4	100	0.3	6	1.8	15
5	100	0.5	17	8.5	22
6	100	1	12	12	30
7	120	0.3	17	5.1	15
8	120	0.5	12	6	22
9	120	1	6	6	30
10	125	0.3	6	1.8	15
11	125	0.5	3	1.5	22
12	125	1	1	1	30

Henceforth, the pins will be designated using the format PN80_2 or PU100_4, where “PN” indicates a pin prior to testing, and “PU” denotes a pin post-testing. The numeral following the “PN” or “PU” represents the mould temperature, and the numeral following the underscore represents the pin number as listed in Table 11. The numbers are not ordered because pins with similar average roughness (R_a) were chosen to be tested under the same conditions. The primary objective behind this decision is to eliminate roughness as a variable across repetitions.

Table 11 – Pin code for each test run.

Run	Replicate		
	I	II	III
1	PN80_2	PN80_3	PN80_5
2	PN80_4	PN80_7	PN80_8
3	PN80_6	PN80_9	PN80_10
4	PN100_2	PN100_4	PN100_14
5	PN100_5	PN100_6	PN100_8
6	PN100_10	PN100_12	PN100_13
7	PN120_1	PN120_2	PN120_3
8	PN120_5	PN120_6	PN120_11
9	PN120_7	PN120_8	PN120_9
10	PN125_1	PN125_2	PN125_12
11	PN125_4	PN125_5	PN125_6
12	PN125_8	PN125_9	PN125_10

4.4 Pin-on-Disc results

As previously mentioned, the results from Pin-on-Disc tests will focus on determining the wear after testing. This parameter will be characterised by two methods: mass loss and the roughness of the pin surfaces. The software *MountainsLab 10* will be used to obtain the desired information and for data analysis.

4.4.1 Roughness analysis

Before the Pin-on-Disc tests, the roughness of the pin tips was measured. Pins with similar initial roughness were selected for testing. These measurements, were conducted using two distinct methods: 2D profile and 3D surface area. Both types of measurements were performed for all pins before and after testing.

The 2D profile was measured with a Hommelwerke profilometer LV-50, which contains a detector allowed to move in vertical direction, eliminating the form effect. For the measuring probe, a TKZ 100/17 was used since its tip, with $5\mu\text{m}$ radius stylus, was the only one able to measure in the radial direction of the tooth due to its geometry. This pick-up, represented in Figure 79(b), is permitted to move vertically, mitigating the influence of profile form effects. Given that the pins have a diameter of only 3 mm, it is essential to set the cut-off and measurement distance accordingly. Therefore, a measurement distance of 1.5 mm will result in cut-off of 0.25 mm, with 4800 measurement points. The gauge velocity is 0.15 mm/s, and the measuring range is $80\mu\text{m}$.

For the 3D analysis, a NPFLEX-1000 from Bruker, as represented in Figure

79(a), was employed to measure an area of 1.5×1.5 mm, with an amplification of ($\times 1$) and using white light. For PA6-GF, green light (with narrow band) was employed due to the challenges in capturing surface roughness with white light.

Both apparatuses used to measure the roughness profile are illustrated in Figure 79.



(a) Bruker NPFLEX-1000 apparatus.



(b) Hommelwerke profilometer LV-50.

Figure 79 – Apparatus utilised for measuring roughness.

For an accurate assessment of roughness, the selection of roughness parameters must align with the intended outcomes. Accordingly, the parameters chosen to evaluate the roughness profile include: average roughness (R_a), mean peak-to-valley height (R_z), total profile depth (P_t), root mean square roughness average (R_q), maximum peak-to-valley height (R_{max}), reduced peak height (R_{pk}), reduced valley depth (R_{vk}) and core roughness depth (R_k).

It is essential to note that these symbols pertain to roughness profile values (2D). For 3D topography, the value used refer to the surface measurements (S), maintaining the same indices. All these roughness parameters adhere to the ISO 21920 standard for profile parameters and the ISO 25178 standard for areal parameters.

The analysis will be performed by comparing the 3D surface measurements of one pin, while the measurements for all other pins are provided in Appendix B.

Roughness before testing

The values for each roughness parameter before testing are represented in Table 12 relatively to 2D profile measurements.

Table 12 – Profile roughness parameters before testing.

Pin code	R_a (μm)	R_z (μm)	R_{max} (μm)	R_k (μm)
PN80_2	0.17	1.13	1.98	0.55
PN80_7	0.24	1.15	1.46	0.70
PN80_9	0.18	0.92	1.76	0.61
PN100_4	0.26	1.49	1.69	0.94
PN100_5	0.25	1.40	2.06	0.82
PN100_12	0.36	1.60	2.22	1.01
PN120_1	0.15	0.90	1.13	0.57
PN120_5	0.15	0.90	1.08	0.50
PN120_9	0.14	0.82	1.15	0.48
PN125_12	2.54	13.75	19.19	6.34
PN125_4	1.73	9.47	11.58	2.56
PN125_9	1.46	8.76	14.30	3.18

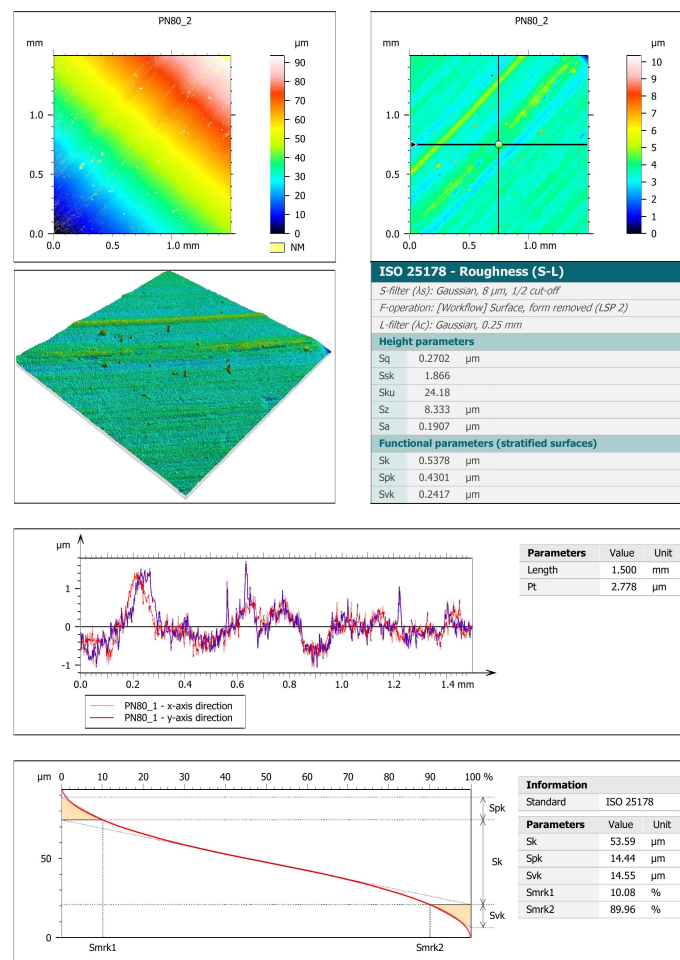


Figure 80 – Example report for PN80.2.

Figure 80 provides an example report for describing and characterising the roughness parameters and surface of PN80_2. In this study, a form removal procedure was implemented on the data. This step was necessary because the pin tip was not parallel to the optical camera. Additionally, outliers were removed due to the inherent challenge of capturing all surface points. Subsequently, profiles were generated in both x and y directions, with corresponding parameters. The report also includes S_{sk} , that represents the predominance of peaks ($S_{sk} > 0$) or valleys ($S_{sk} < 0$), and S_{ku} , that indicates the presence of exceptionally high peaks or deep valleys.

Roughness after testing

The primary objective of roughness measurement post-testing is to determine whether testing parameters impact surface roughness. Given the dry contact established between a polymer and steel, abrasive wear is expected, leading to an overall increase in roughness parameters such as R_{max} and R_z [40].

Table 13 presents the roughness profile parameters measured after the Pin-on-Disc tests.

Table 13 – Profile roughness parameters after testing.

Pin code	R_a (μm)	R_z (μm)	R_{max} (μm)	R_k (μm)
PN80_2	0.21	1.02	1.59	0.59
PN80_7	0.28	1.37	2.79	0.41
PN80_9	0.09	0.44	0.61	0.30
PN100_4	0.28	1.55	2.33	0.57
PN100_5	0.14	0.85	2.07	0.26
PN100_12	0.36	1.60	2.22	1.01
PN120_1	0.09	0.39	0.96	0.17
PN120_5	0.13	0.85	2.46	0.18
PN120_9	0.13	0.69	1.17	0.23
PN125_12	1.29	6.30	10.66	3.35
PN125_4	1.76	8.65	14.73	3.64
PN125_9	1.68	9.24	12.90	4.10

Upon evaluating the roughness parameters before and after test, it is possible to conclude that these parameters tend to decrease. Despite being an unusual result, this phenomenon may be attributed to the selected measurement profile and the shape of the deformed surface.

To further elucidate the deformed shape and confirm the results previously studied by *H. Benabdallah* [40]. Figure 81(a) depicts the zone with higher wear (left) and lower wear (right). Meanwhile Figure 81(b) shows higher wear in the right part. In both figures, the zone of interest is characterised by the homogeneously

orientated stripes. This observations is further supported by the shape depicted in Figure 82, where “zone 1” corresponds to the worn surface, and “zone 2” represents the non-worn part of the pin. Since the measurements aimed to capture the slope of wear, the obtained profiles are parallel to the visible lines in the images. This factor could contribute to the obtained low values for roughness parameters after testing, leading to the conclusion that a raw profile analysis (profile P) could be more suitable for this situations.

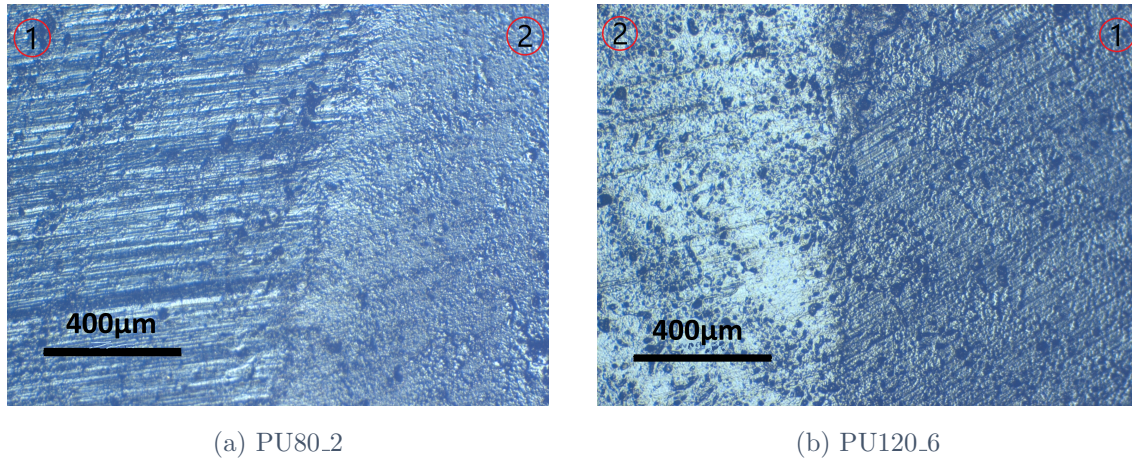


Figure 81 – Wear difference on the pin surface.



Figure 82 – Final shape observed in pins after testing.

Understanding this consideration, the decision was made to analyse the 3D surface parameters. This method takes into account the entire surface rather than a chosen line, which could introduce analysis errors.

Figure 83 represents the comparison between PN80.2 and PU80.2. The following filters were applied: surface rotation to ensure consistent roughness orientation (parallel to x-axis), removal of outliers and filling in non-measured points, a form removal and extraction of vertical profiles, perpendicular to the roughness. These steps allow for a wear analysis between before and after profiles.

Figure 83 reveals a pronounced increase in roughness after the test, indicating significant wear compared to the pre-test profile. Upon analysing the overall parameter, a loss of $507.5 \mu\text{m}^2$ is recorded.

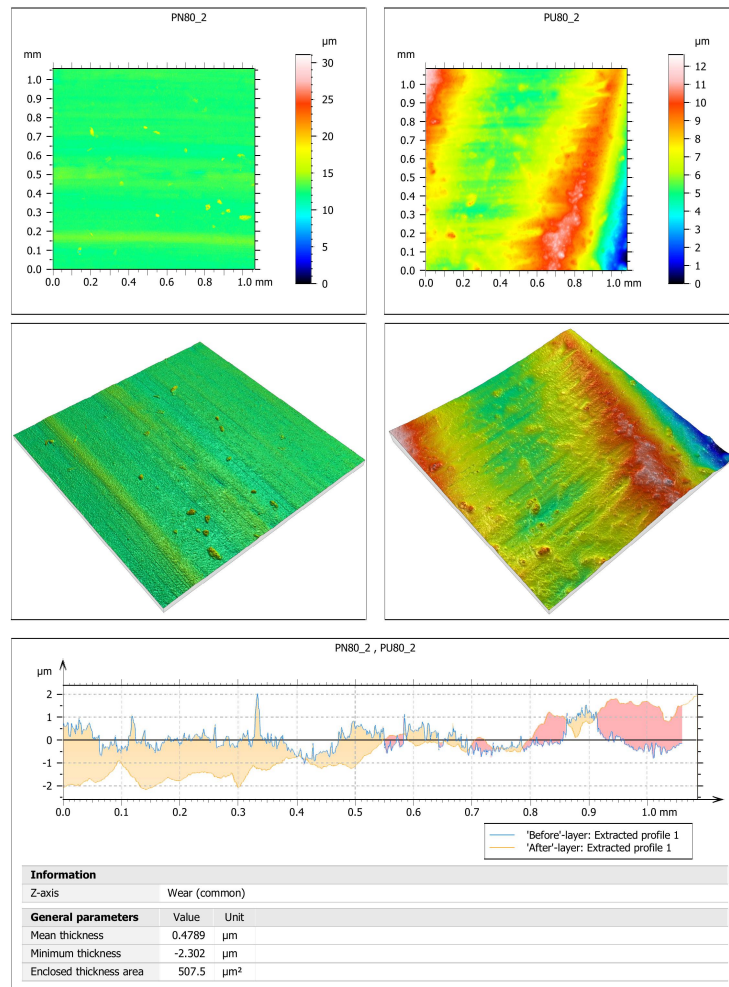


Figure 83 – Wear report example for PU80.2.

4.4.2 Mass loss

Upon completion of the 36 total runs, the final mass was measured and compared with the initial mass of the pins. The difference between these two values represents the mass loss during the test, with the expectation that more severe conditions result in higher mass loss. Table 14 contains all the collected data regarding the mass for each pin. The balance sensibility is 0.1 mg, and the expected mass loss is within this magnitude.

The mass loss was recorded exclusively for the pins, as the wear on the disc was insufficient to be detected by the balance. Furthermore, the balance has a lower limit of mass that is below the mass of the disc.

To ensure precise measurements, three measurements were conducted for each pin both before and after testing.

Based on data, it is possible to conclude that the mass loss across the three replicates for each run does not exhibit significant dispersion between values. The

Table 14 – Mass loss for each test run.

Run	Mould temperature (°C)	Sliding speed (m/s)	Contact pressure (MPa)	Mass Loss for replicate (mg)			Average
				I	II	III	
1	80	0.3	12	0.3	0.3	0.3	0.30
2	80	0.5	6	0.3	0.3	0.2	0.27
3	80	1	17	0.8	0.8	0.8	0.80
4	100	0.3	6	0.2	0.2	0.2	0.20
5	100	0.5	17	0.9	0.8	1.0	0.90
6	100	1	12	0.6	0.6	0.5	0.57
7	120	0.3	17	0.1	0.1	0.1	0.10
8	120	0.5	12	0.1	0.1	0.1	0.10
9	120	1	6	0.2	0.1	0.1	0.13
10	125	0.3	6	0.1	0.1	0.1	0.10
11	125	0.5	3	0.2	0.2	0.1	0.17
12	125	1	1	0.3	0.3	0.2	0.27

standard deviation is on the order of magnitude of approximately 10^{-5} , making it negligible.

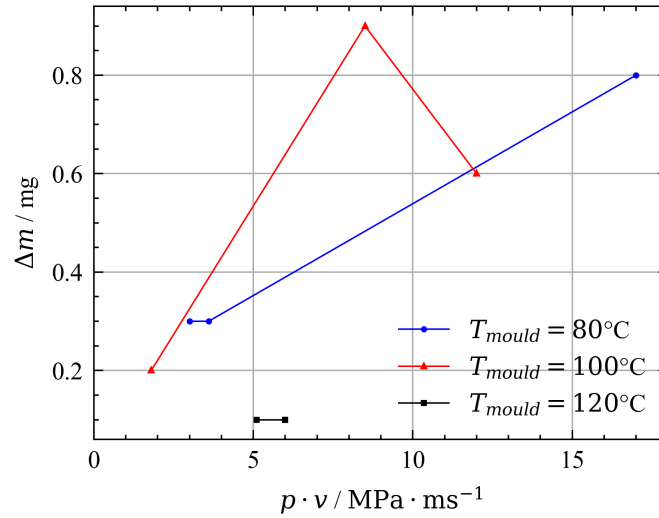
Figure 84 – Mass loss distribution as a function of $p \cdot v_s$.

Figure 84 illustrates the relationship between mass loss and $p \cdot v_s$ for different mould temperatures. From the curves, it is evident that different mould temperatures result in varying mass loss across the $p \cdot v_s$ range. Notably, the material with the lowest mass loss corresponds to $T_{\text{mould}} = 120^\circ\text{C}$. The observed outlier for the mould temperature of 100°C may be attributed to the high contact pressure, which has a more significant impact than the sliding speed.

Furthermore, the mass loss allowed to calculate the wear coefficient (k_W) in units of $\text{mm}^3/(\text{Nm})$ using Equation 16.

$$k_W = \frac{\Delta m}{\rho \cdot P \cdot L} \quad (16)$$

Where Δm represents the mass loss, ρ is the density, P is the applied normal force and L is the total sliding distance.

Table 15 represents the wear coefficient values for each run, considering an average of mass loss across three replicates within the same run. The dispersion between values is insignificant.

Table 15 – Wear coefficient for each test run.

Run	Mould temperature (°C)	Sliding speed (m/s)	Contact pressure (MPa)	Wear coefficient ($\times 10^{-6}$ mm ³ /Nm)
1	80	0.3	12	2.508
2	80	0.5	6	4.459
3	80	1	17	4.722
4	100	0.3	6	3.334
5	100	0.5	17	5.312
6	100	1	12	4.738
7	120	0.3	17	0.590
8	120	0.5	12	0.836
9	120	1	6	2.230
10	125	0.3	6	1.672
11	125	0.5	3	5.574
12	125	1	1	26.756

Despite the test conditions and mould temperatures not causing mass loss to vary linearly, this effect is not reflected in the wear coefficient. Therefore, despite the values being slightly disperse among themselves, the average wear coefficient obtained for the POM material is $3.193 \cdot 10^{-6}$ mm³/Nm, while for the PA6-GF, a value of $11.33 \cdot 10^{-6}$ mm³/Nm is obtained. This reveals that PA6-GF, despite containing glass fibers in its composition that improve mechanical properties (namely stiffness), has a higher wear coefficient than POM. It wears more easily under applied load and speed.

Moreover, the mass loss was used to determine whether the mould temperature, velocity and pressure influence the wear obtained for each test. This analysis was conducted using *Minitab*, a statistical software that allow the application of Taguchi designs.

The desired solution aims to minimise the mass loss of the pin while investigating whether variations in mould temperature, pressure and sliding velocity yield distinct outcomes. Based on Figure 85, is possible to conclude that the optimal combination for minimising wear involves a mould temperature of 120 °C, sliding velocity of

0.3 m/s and contact pressure of 6 MPa.

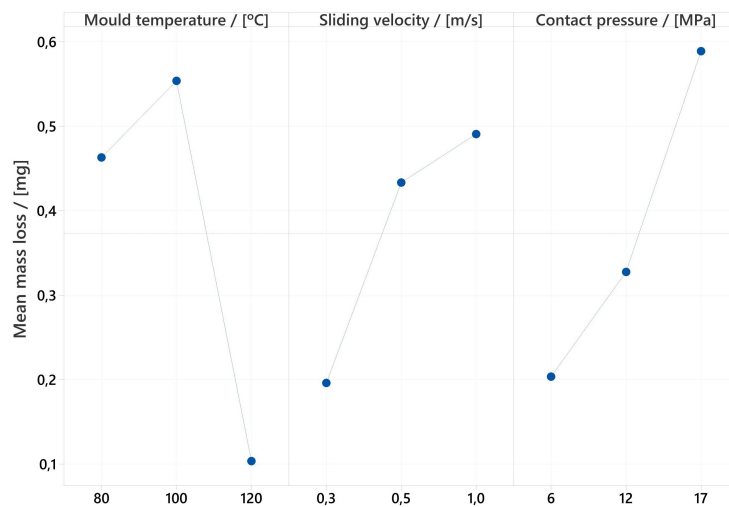


Figure 85 – Mass loss main effects plot for means.

By analysing the Table 16, it is possible to determine the relative importance of each factor in wear, ranked from 1 to 3, highlighting that the chosen mould temperature is the most influential factor affecting wear. In this analysis, delta is the difference between the highest and lowest average response values for each factor.

Table 16 – Mass loss response for means (mg).

Level	Mould temperature	Sliding speed	Contact pressure
1	0.463	0.196	0.204
2	0.554	0.433	0.328
3	0.104	0.491	0.589
Delta	0.450	0.294	0.385
Rank	1	3	2

To understand the results obtained, two key analyses are essential: the standard deviation, as depicted in Figure 86, and the variability, represented by the S/N ratio in Figure 87. Examining the standard deviation, all levels exhibit satisfactory values, characterised by their low order of magnitude, indicating a uniform dataset.

Given that this study aims to reduce mass loss, the optimal level for each factor corresponds to the highest S/N ratio, signifying minimal variability. As illustrated in Figure 87, the ideal solution aligns with the results obtained from the Figure 85.

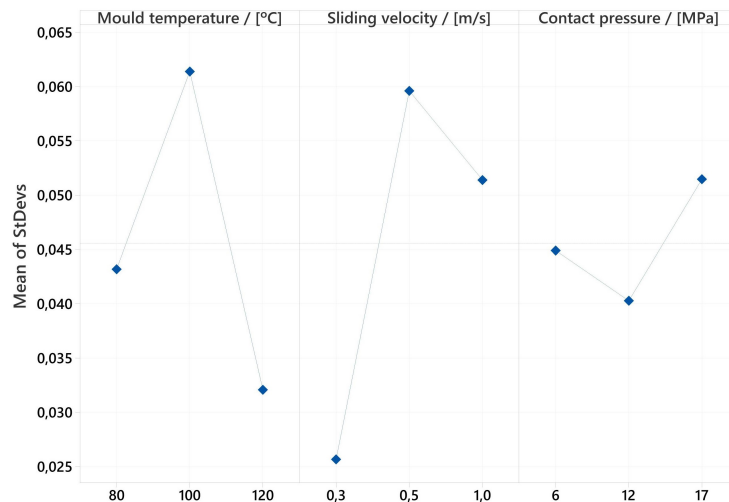


Figure 86 – Mass loss main effects plot for standard deviations.

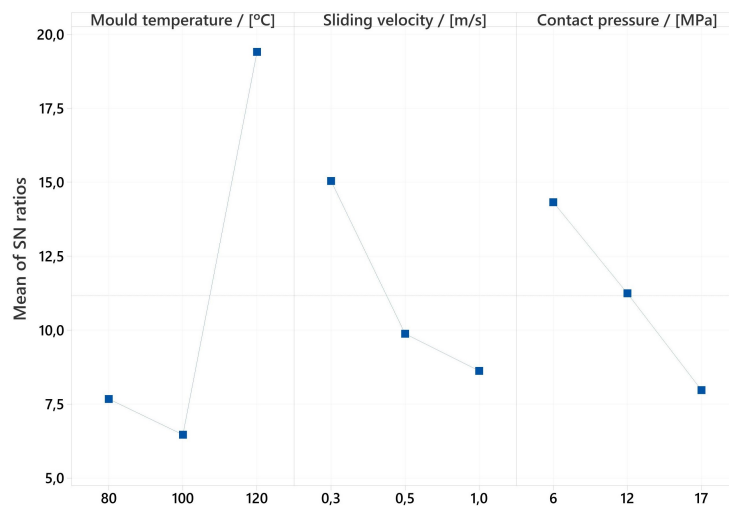


Figure 87 – Mass loss main effects plot for S/N ratios.

After conducting an analysis to identify combinations suitable for minimising wear and subsequent mass loss, a linear regression model was performed to establish a regression equation capable of predicting mass loss for other combinations. Subsequently, a representation of the experimental data using the obtained model was conducted, through the coefficient of determination, denoted as R^2 . The resulting value of $R^2 = 0.68$, indicates that Equation 17 explain, with 95% of confidence, 68% of the experimental data.

$$\text{Mass Loss} = 0.651 - 0.00898 \cdot T_{\text{mould}} + 0.362 \cdot v_s + 0.03454 \cdot p \quad (17)$$

Figure 88 illustrates the normal probability plot using the data obtained from the experimental procedure. The plot reveals that several points do not closely

align with the normal distribution, thereby corroborating the results obtained with Equation 17 and its R^2 value.

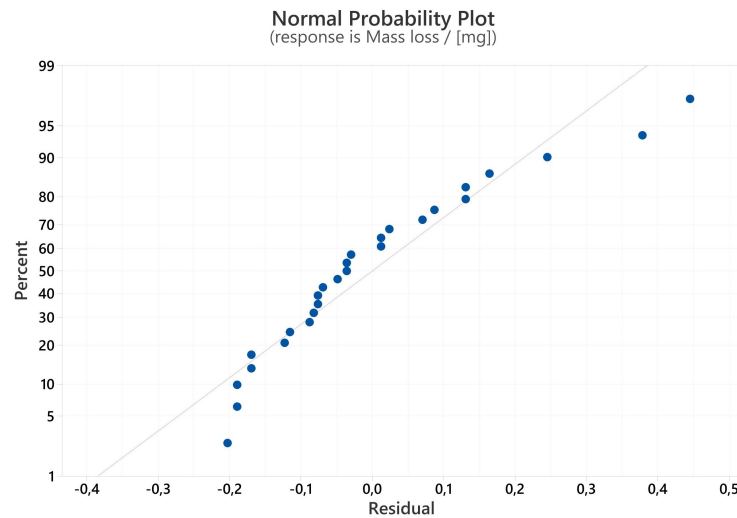


Figure 88 – Normal probability plot for mass loss.

4.4.3 Thermal analysis

The Pin-on-Disc tests were fully recorded using a FLIR A325 camera, represented in Figure 89, connected to the *ThermaCAM Researcher Pro* software. The selected emissivity values were 0.75 for the disc and 0.92 for the pins.



Figure 89 – Thermal camera FLIR A325.

Figure 90 represents the first replicate of test run 2, which lasted 2000 seconds and reached a maximum temperature of 46.3°C. It is crucial to note that the maximum temperature reached during this test is not directly comparable to that reached by a gear under the same contact pressure and speed. This is because the pin has its contact surface in continuous contact, whereas in a working gear, each tooth has intermittent periods of contact.

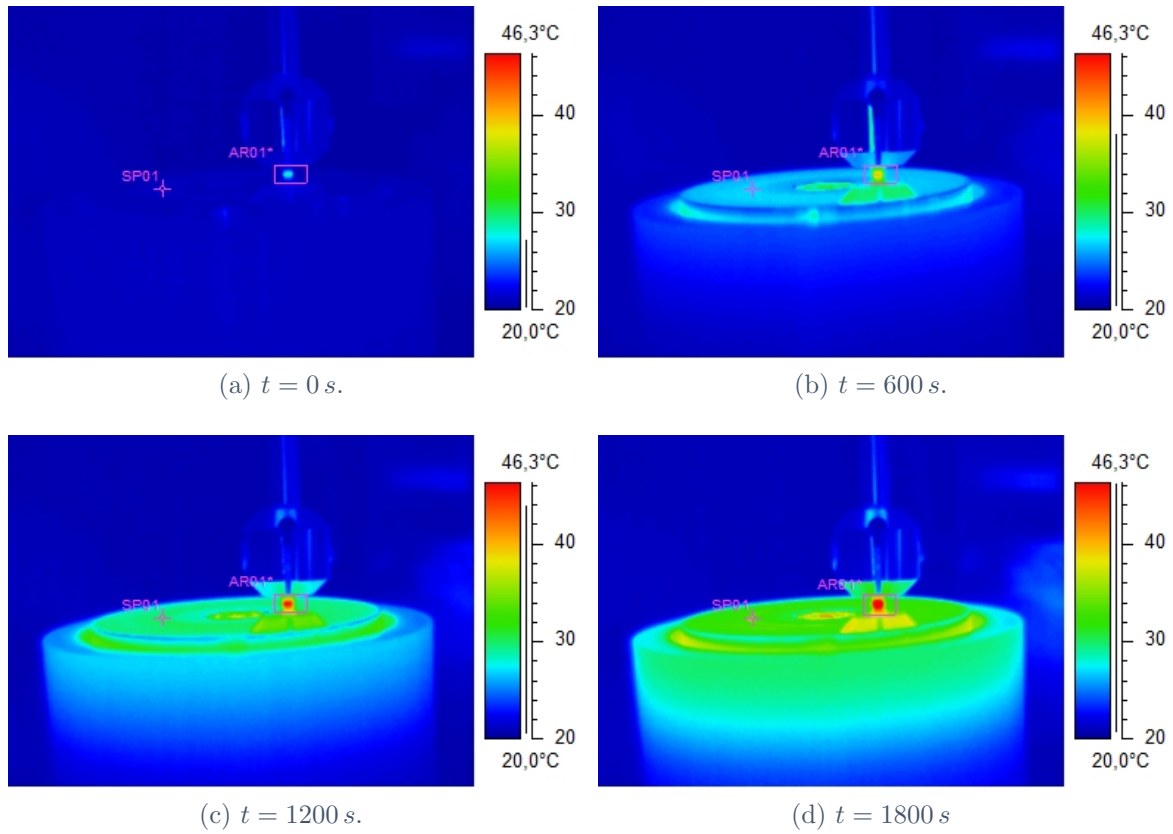


Figure 90 – Thermal images of distinct instants for test run 2.

For each tested pin, the maximum temperature was recorded to facilitate a similar analysis made with mass loss using Taguchi designs. Table 17 presents the maximum temperature reached during each test, which will be utilised for the Taguchi analysis.

Table 17 – Maximum temperature for each test run.

Run	Mould temperature (°C)	Sliding speed (m/s)	Contact pressure (MPa)	Maximum temperature for replicate (°C)			Average	Std.
				I	II	III		
1	80	0.3	12	42.7	40.2	42.3	41.7	1.1
2	80	0.5	6	46.3	50.1	49.6	48.7	1.7
3	80	1	17	102.4	98.0	103.9	101.4	2.5
4	100	0.3	6	39.6	37.6	37.4	38.2	1.0
5	100	0.5	17	79.0	80.2	77.8	79.0	1.0
6	100	1	12	81.2	79.0	83.4	81.2	1.8
7	120	0.3	17	55.0	51.0	52.9	53.0	1.6
8	120	0.5	12	60.5	61.3	60.9	60.9	0.3
9	120	1	6	66.6	55.5	63.9	62.0	4.7
10	125	0.3	6	55.0	51.0	52.9	53.0	1.6
11	125	0.5	3	38.1	36.9	36.9	37.3	0.6
12	125	1	1	32.1	35.0	37.0	34.7	2.0

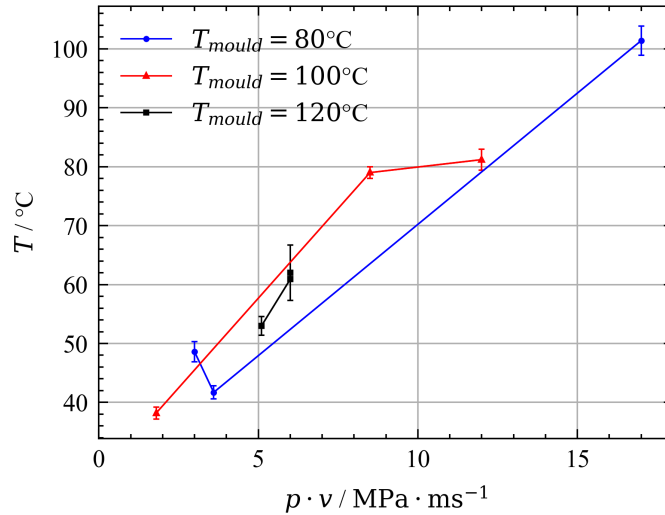


Figure 91 – Maximum temperature distribution as a function of $p \cdot v_s$.

Figure 91 depicts a comparative analysis of the maximum temperature as a function of $p \cdot v_s$ for distinct mould temperatures. While the measurements along the $p \cdot v_s$ values may not be fully representative for all curves, the overall behaviour indicates that different mould temperatures result in a consistent trend of maximum temperature across the $p \cdot v_s$ range. Minor discrepancies observed at identical $p \cdot v_s$ values could be attributed to variations in the initial disc temperature and ambient conditions. This variation among materials will also be demonstrated through the Taguchi model.

The Taguchi model, employed for mass loss analysis, was also applied to assess the influence of three factors on the temperature during testing, utilising the maximum temperature conditions.

Analysing the temperature aspect, the objective is minimising it. As expected, Figure 92 and Table 18 reveal that velocity and pressure exert greater influence on the attained maximum temperature than mould temperature. Although the material differences are subtle, the same trend observed in mass loss becomes evident when analysing maximum temperature. In particular, a mould temperature of 120°C yields the lowest maximum temperature, followed by 80°C and finally 100°C .

The observed trend aligns with the behaviour identified in the DSC results. This correlation between material crystallinity and thermal conductivity has been previously investigated by *Yu Jia et al.*, who concluded that crystallinity significantly influences the thermal conductivity of semi-crystalline polymers. Notably, higher crystallinity values correspond to increased thermal conductivity [42]. This finding is consistent with the thermal results, as higher thermal conductivity facilitates more efficient energy dissipation, leading to lower temperatures.

While this analysis may be one of the reasons for the temperature increase depending on the mould temperature, it should be subsequently tested to understand

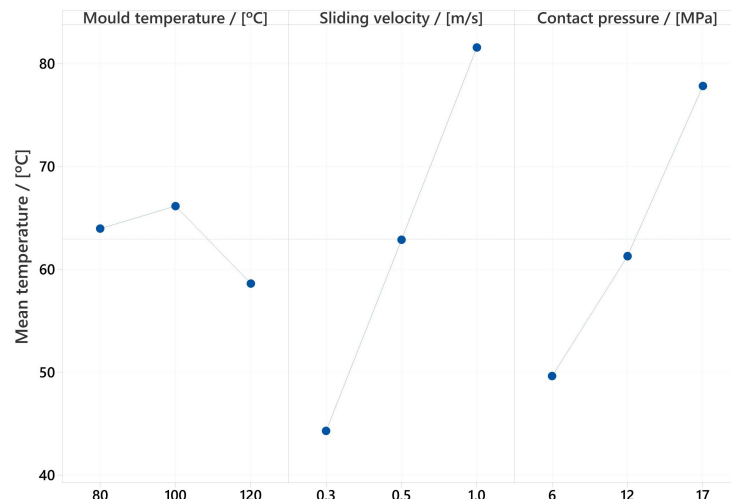


Figure 92 – Temperature main effects plot for means.

the variation of thermal conductivity with increased crystallinity.

Another factor that may have differentiated the temperature during the test is friction and must be considered.

Table 18 – Temperature response for means (°C).

Level	Mould temperature	Sliding speed	Contact pressure
1	63.94	44.30	49.62
2	66.13	62.86	61.28
3	58.62	81.54	77.80
Delta	7.51	37.24	28.18
Rank	3	1	2

Regarding standard deviation, the values obtained are notably higher than those for mass loss. Challenges in maintaining constant room temperature for all runs, coupled with the inherent variability in initial disc temperature due to extended time required for the disc to return to room temperature between replicates, contribute to this dispersion.

To predict temperatures for other combinations, the fit regression model used for mass loss was applied. Equation 18, with an $R^2 = 0.91$, represents the temperature prediction bases on the experimental data. Additionally, Figure 95 illustrates the normal probability plot for the temperature response, derived from the experimental data. It can be concluded that the points follow a normal distribution, as evidenced by their proximity to the straight line.

$$\text{Temperature} = 16.46 - 0.1331 \cdot T_{\text{mould}} + 50.16 \cdot v_s + 2.541 \cdot p \quad (18)$$

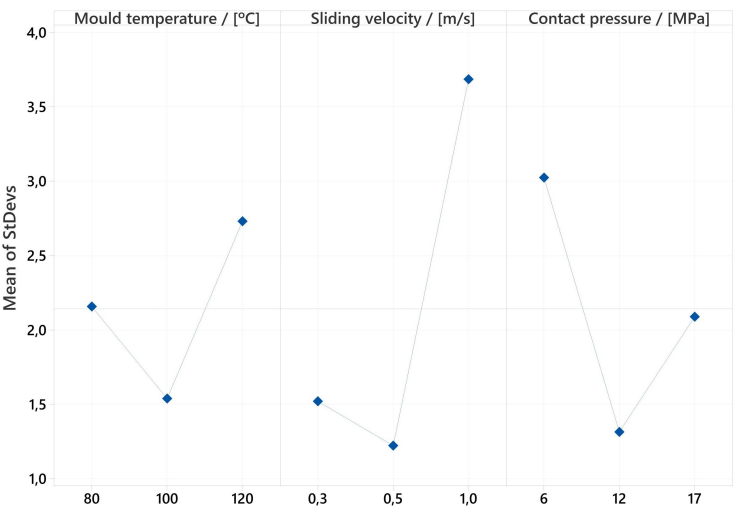


Figure 93 – Temperature main effects plot for standard deviations.

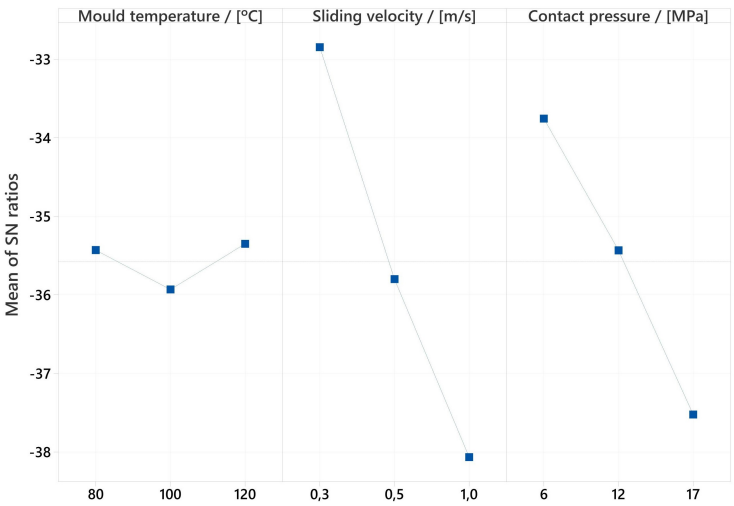


Figure 94 – Temperature main effects plot for S/N ratios.

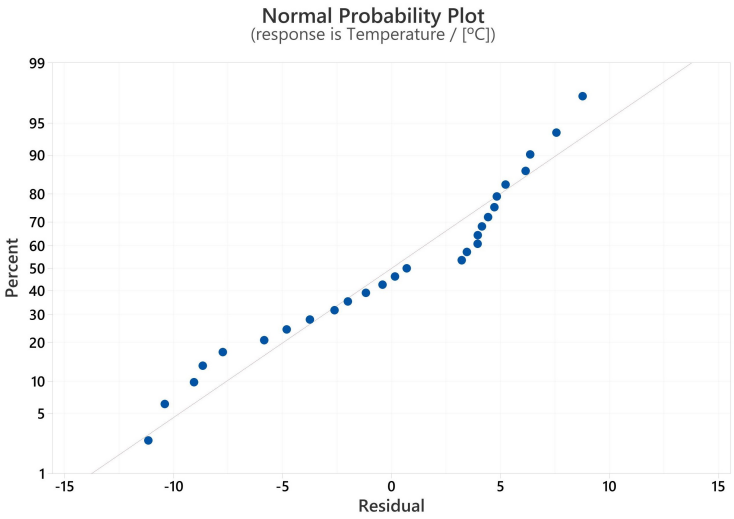


Figure 95 – Normal probability plot for temperature.

5 WEAR PREDICTION

This chapter aims to predicting wear in Pin-on-Disc tests through numerical calculations. Initially, numerical calculations using *PrePoMax* is executed for comparison with experimental results. The CAD parts were designed using *SolidWorks*. The aim is to determine the accuracy of numerical and analytical calculations in predicting wear under specific testing parameters.

As thermal effects are not considered in the software, simulations for all nine different combinations will be conducted using the same track radius. Although the process for all combinations is similar, only one combination will be presented and studied in this chapter. The results for all combinations are presented in Table 20.

The selected combination features a track radius of 15 mm, a contact pressure of 6 MPa, and a sliding velocity of 1 m/s.

The process for achieving the desired results is explained throughout this chapter, detailing the creation and simulation of the parts.

5.1 Numerical calculation

5.1.1 Parts design

For the disc, it was unnecessary to model the entire part, as only a section was sufficient and could reduce the time for numerical calculations. Therefore, only a section of the disc was designed, with a thickness of 2 mm set due to the disc's higher hardness and lower wear coefficient relative to the pin. The wear on the disc is expected to be negligible compared to that on the pin.

The pin has a diameter on it tip of 3 mm, so a quarter of a disc with 11.5 mm and 18.5 mm of inner and outer radii, respectively was designed and it is represented in Figure 96. The disc was divided into three parts to create different meshing sizes, with finer mesh applied to the contact area with the pin for improved results in this zone.

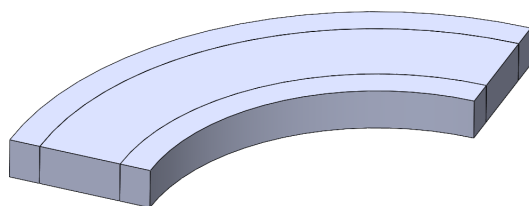


Figure 96 – Designed disc.

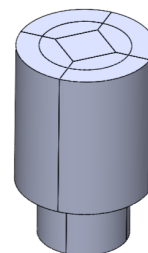


Figure 97 – Designed pin.

Similarly, the pin was modelled without simplifications. The pin was divided into smaller parts to enable the creation of a Structured transfinite mesh. Consistency in mesh structure between both parts was maintained to achieve a better result of the Finite Element Method (FEM) model. Given that the pin is the critical component in this contact, a finer mesh structure was employed to ensure higher precision in the results.

5.1.2 Mesh generation

The mesh generation process was executed in *PrePoMax* as the subsequent steps to obtain the final result. The objective was to create a mesh denser in the critical areas and coarser in remaining regions. Given the pin is the critical part, ensuring its accurate definition and correct application of constraints and loads was paramount. Figure 98 and Figure 99 depict the Transfinite mesh applied to both parts, with a refined process in contact areas and critical regions.

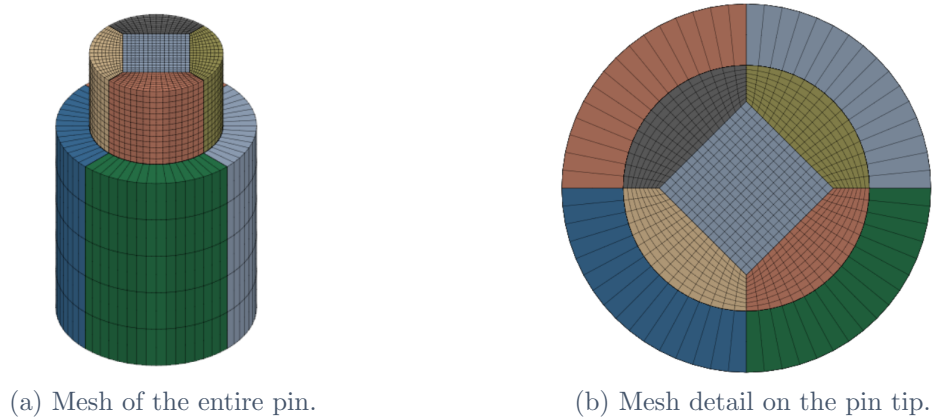


Figure 98 – Pin mesh.

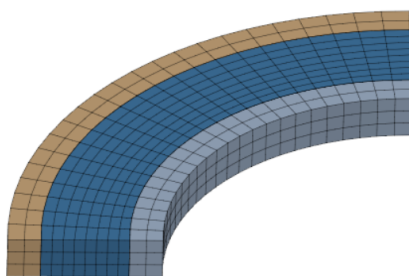


Figure 99 – Disc mesh.

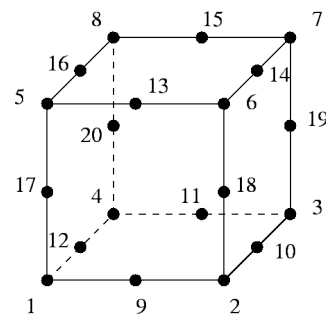


Figure 100 – Quadratic brick element with 20-node [43].

A study of convergence of the model was not performed. However, the results for the contact pressure at the pin tip align with the analytical solution, yielding a value of 6 MPa. This is consistent with the simulation, where a load of 42.6 N was applied.

The pin mesh consists of a total of 15296 elements, while the disc mesh comprises 828 elements. Both components are composed of elements of type C3D20, that correspond to a quadratic brick element ($3 \times 3 \times 3$ integration points), represented in Figure 100.

5.1.3 Simulation of the model

The simulation required the application of loads and boundary conditions and determining the number of cycles needed. The chosen number of cycles impacts the processing time to achieve results. The aim was to execute a single movement step to obtain wear depth, followed by multiplying the wear depth by the sliding distance to obtain the final wear depth.

Alternatively, the problem could be solved by simulating a specific number of cycles (e.g., 25000 cycles) and analysing the resultant pin shape. Using *SolidWorks*, the pin shape after 25000 cycles could be updated and the model re-simulated. This iterative process must be repeated until the desired sliding distance was achieved and the final pin shape attained.

The loads and constraints applied to this setup, represented in Figure 101, needed to align with those in the Pin-on-Disc test. Therefore, a vertical load was applied to the bottom of the pin (red arrow), allowing only vertical displacement to ensure proper pressure application between the pin and disc (green boundary condition). Additionally, the desired rotation of the disc with a certain angle was applied (blue boundary condition).

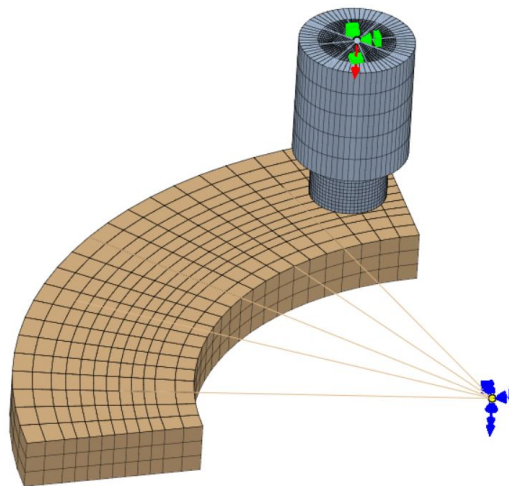


Figure 101 – Boundary conditions and applied loads for the simulation.

In the context of contact analysis, the top surface of the disc and the pin tip were chosen. The contact behaviour was modelled as linear, with a friction coefficient of 0.2 [44]. For this contact pair, two distinct regions were defined: a master region (corresponding to the disc's top surface) that does not exhibit wear, and a slave region (located at the tip of the pin) that experiences wear.

In relation to the material properties, the values assigned to each property were obtained from the material data sheet provided by the manufacturing company [45]. These properties are presented in Table 19 and the wear coefficient values are for a POM-Steel contact.

Table 19 – Material properties.

Property	Unit	POM	Steel S235
Density	kg/mm ³	1410	7100
Young's modulus	GPa	2.6	210
Poisson's ratio	–	0.386	0.28
Thermal expansion	°C ⁻¹	$1.0 \cdot 10^{-4}$	$1.1 \cdot 10^{-5}$
Thermal conductivity	mW/(mm · °C)	0.221	14
Specific heat	J/(g · K)	1.47	0.470
Hardness	MPa	83	235
Wear coefficient	–	$2.0 \cdot 10^{-6}$	$1.0 \cdot 10^{-7}$

After applying all boundary conditions, loads, and material properties, the software employs Archard's wear model, as previously discussed in Equation 11. It then adapts this model for use in a multistep numerical procedure, where the change in wear depth must be computed for each wear cycle. The resulting equation is given by Equation 19 [46].

$$h_i = k_a \int \frac{k_a}{H} \cdot p \, dL \quad (19)$$

Where h_i represent the wear depth accumulated in the i-th wear cycle. The total wear depth after multiple wear cycles is then computed by summing all single-cycle wear depths. However, as previously mentioned, only a quarter of a revolution is considered to simplify computational time.

Given that, the applied rotation is equal to 1.3 radians, or approximately 75 degrees. Moreover, the objective was to predict wear for 1000 m of sliding distance at a sliding speed of 1 m/s. The time step for the wear simulation was calculated using Equation 20, resulting in a time step of 0.0195 s.

$$t_{step} = \frac{r \times \theta}{v_s} \quad (20)$$

In this cycle, the pin on the disc travels a distance of 0.0195 m. To determine the wear depth after 1000 m, the simulation result must be multiplied by 51282, which is the ratio between the two distances.

5.1.4 Simulation results

Figure 102 illustrates the wear depth gradient obtained from the simulation. As expected, a slope in the pin tip resulted in more wear on the front edge of the pin relative to the sliding direction, about 8.78×10^{-6} , than on trailing edge, approximately 9.756×10^{-7} . This observation aligns with experimental results and previous research [40].

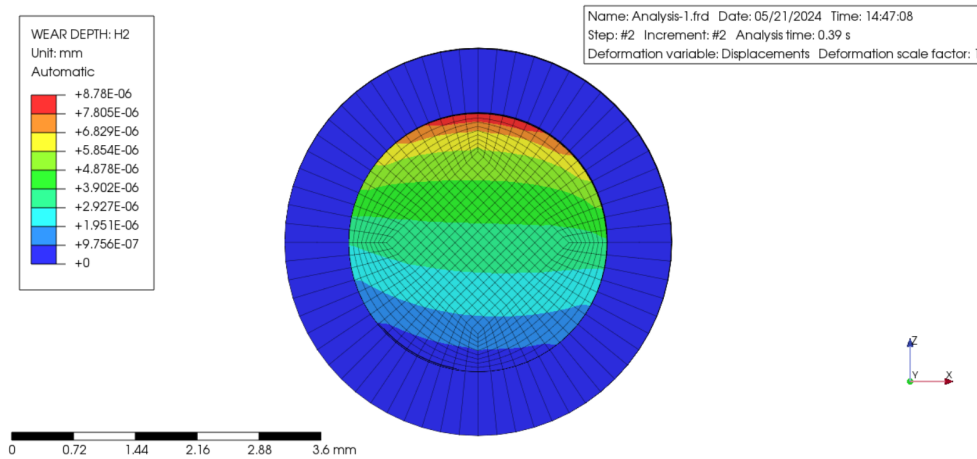


Figure 102 – Wear depth gradient from simulation results.

Upon completion of the time step and determination of the wear depth, it is possible to extrapolate this behaviour over the entire sliding distance of 1000 m. For the given angle of 1.3 radians and a track radius of 15 mm, the resulting sliding distance is 19.5 mm. As previously mentioned, the ratio between these two distances is 51282, indicating that the total wear depth at the maximum point should be approximately 0.450 mm.

The procedure described above was applied to the remaining test conditions, and the maximum wear depth values were recorded in Table 20.

Table 20 – Wear depth numerical results.

Run	Sliding speed (m/s)	Pressure (MPa)	Maximum wear depth ($\times 10^{-6}$ mm)	Total wear depth (mm)
1	0.3	12	17.25	0.885
2	0.5	6	9.015	0.462
3	1	17	25.14	1.289
4	0.3	6	8.382	0.430
5	0.5	17	24.80	1.272
6	1	12	17.68	0.907
7	0.3	17	24.54	1.258
8	0.5	12	17.45	0.895
9	1	6	8.780	0.450

Considering Figure 102 it becomes evident that the wear gradient is not linear. Consequently, a plot was generated to elucidate the wear evolution along the pin's diameter. This evolution is depicted in Figure 103.

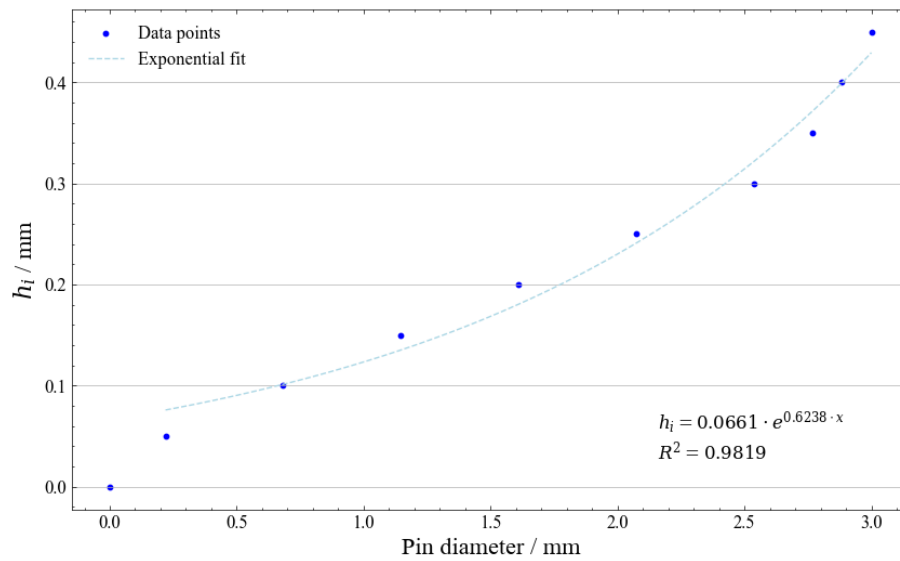


Figure 103 – Wear depth gradient.

The calculated wear volume, obtained by creating a wear part in *SolidWorks*, is equal to 0.89 mm^3 .

Subsequently, a simulation involving updatable geometry was conducted. The pin's geometry was updated in *SolidWorks*, assuming a linear wear gradient for simplification, and assuming the results obtained in Figure 102, multiplied by half of the total cycles (approximately 25641). The simulation procedure was repeated, and Figure 104 illustrates the wear depth gradient on the pin tip surface after one cycle. This value was further multiplied by the number of missing cycles (25641 cycles). By summing the values before and after the geometry update, we determined a total wear depth of 0.350 mm. Notably, this value is 0.10 mm lower than the simulation conducted without updating the pin geometry.

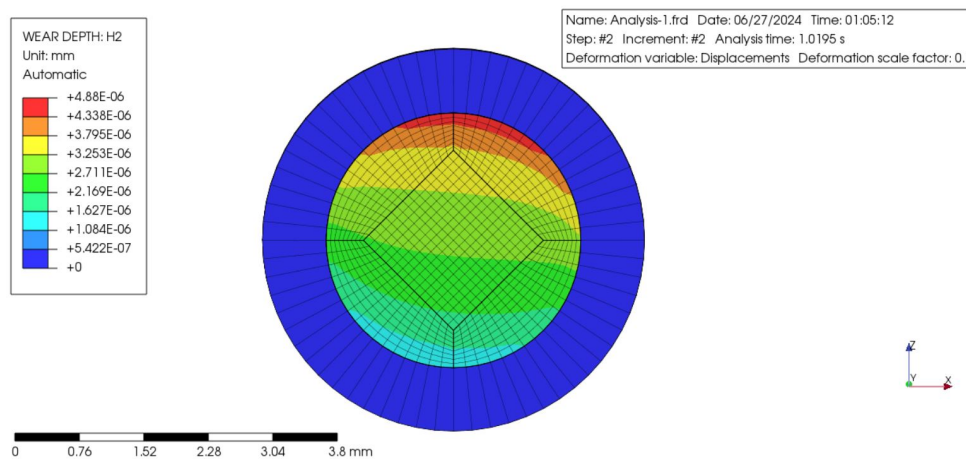


Figure 104 – Wear depth gradient from updated geometry.

To analyse the wear evolution along the diameter of the pin, we use the exponential fit depicted in Figure 103 and adjust it for half of the total cycles. Thereafter, the wear depth gradient from Figure 104, multiplied by 25641, is transformed into a curve. This curve is added to the existing wear depth gradient, resulting in the comprehensive wear profile represented in Figure 105.

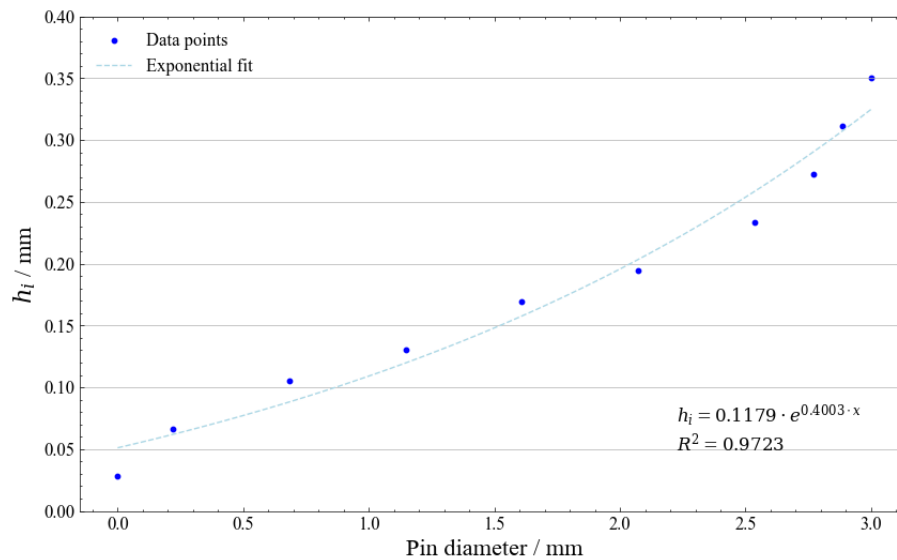


Figure 105 – Wear depth gradient for updated geometry.

The wear volume for this simulation is approximately 0.705 mm^3 . The observed difference in wear volume underscores the critical importance of updating the geometry. This adjustment of values aligns more closely with those observed during experimental tests, highlighting the necessity of updating the geometry, if possible, after each cycle.

5.1.5 Comparison with experimental results

In the context of the Pin-on-Disc test, it is evident that the predicted wear does not align with the experimental results, as observed when comparing the data from previous chapters 4.4.1 and 4.4.2, relatively to mass loss and roughness after testing.

Following the Pin-on-Disc tests, the pins exhibit a wear slope that is more severe than anticipated, affecting only a pin surface portion. This conclusion is supported by Figure 81, which clearly delineates the division between wear phase and the phase without wear. Interestingly, this schematic is absent in the wear prediction analysis conducted with *PrePoMax*. The discrepancy in results may be attributed to the frictional forces present during the experimental tests. These forces can cause a slight deviation in the pin's vertical axis, which does not occur during the simulation, resulting in a resultant force that is not perpendicular to the pin-disc contact surface.

Another potential cause of the disparity lies in the distinct wear coefficient

values used in the wear prediction versus those obtained from experimental results. The average wear coefficient calculated through the mass loss equation is $3.193 \cdot 10^{-6} \text{ mm}^3/\text{Nm}$, whereas the chosen value of wear prediction from literature is $k_W = 2.0 \cdot 10^{-6} \text{ mm}^3/\text{Nm}$. This variation in wear coefficient could significantly impact the observed wear volume, especially considering that experimental results tend to be less sensitive to wear effect.

In the same conditions as the predicted model, the mass loss yields an average value across three replicates of $\Delta m = 0.00013 \text{ g}$, resulting in a wear volume, dividing by density, of 0.095 mm^3 . Remarkably, this value represents only 10% of the wear volume prediction of 0.89 mm^3 and 13.5% of the wear volume prediction with updated geometry of 0.705 mm^3 .

In conclusion, further Pin-on-Disc tests are necessary to identify potential issues with the fixing setup and mitigate any errors stemming from it. Nevertheless, based on this investigation, it is evident that the Pin-on-Disc test will consistently result in a discernible slope in the pin tip surface.

6 CONCLUSION

6.1 Final conclusions

This dissertation aimed to investigate the thermal and mechanical behaviour of polymer gears, considering their material type and manufacturing parameters, including mould temperature. Additionally, Pin-on-Disc tests were conducted to determine the wear coefficients, mass loss and temperature responses under different test conditions using distinct materials. For this study four cases were considered: POM with three distinct mould temperatures (80 °C, 100 °C and 120 °C) and PA6-GF, containing PA6 and glass fibers.

The specimens for all tests were modelled and subsequently machined to achieve the desired shape by extracting them from the injected gears. Furthermore, additional components were designed to enhance test efficiency or serve as potential avenues for future research. Prior to conducting the tests, prediction analyses and analytical calculations were performed to understand the range of pressures and velocities that would yield meaningful results for the chosen specimens.

The findings from DMA indicate that PA6-GF exhibits a superior storage modulus compared to POM, rendering it more suitable for applications requiring high stiffness. Additionally, all materials exhibit a decrease in storage modulus with increasing temperature, with POM showing a slight variation in trend around 85 °C followed by stabilization. Furthermore, across all materials, the storage modulus increases with higher loading frequencies.

Regarding DSC, both POM and PA6-GF gears exhibit similar tendencies during testing due to their semi-crystalline nature. However, this similarity does not extend to enthalpy reactions or melting and crystallisation temperatures. DSC tests reveal that POM has a higher crystallinity than PA6-GF, and there are also differences between distinct mould temperatures for POM. With the temperatures at which the gears were injected, we can see that there is an increase in crystallization for the following order of temperatures, 100 °C, 80 °C and 120 °C. This insight is crucial for interpreting the subsequent test results.

The DSC findings align with hardness measurements, where POM exhibits higher surface hardness compared to PA6-GF. The increased crystallinity in POM materials contributes to elevated surface hardness, although with a smaller variation compared to the differences observed between POM and PA6-GF.

Regarding Pin-on-Disc testing, this study underscores the importance of precise measurement techniques and the effectiveness of Taguchi designs for analysing wear and thermal data. Notably, PA6-GF experiences more severe wear under high

contact stresses, necessitating lower pressure loads during testing to ensure reliable results. Furthermore, the Taguchi design analysis confirms that mould temperature significantly influences wear, as well as speed and pressure, as expected. Additionally, the impact of temperature during testing is evident, since materials with higher crystallinity exhibit higher thermal conductivity, resulting in lower temperatures during testing.

Practically, these findings suggest that while PA6-GF's higher stiffness and melting temperature offer advantageous, but POM's hardness and wear characteristics make it suitable for diverse applications. Moreover, mould temperatures play a decisive role in defining the mechanical and thermal properties of these materials.

For the Pin-on-disc contacts, a comparison of wear coefficients between experimental results and numerical calculation, highlights the significance of sliding velocity as a fundamental parameter affecting final wear and mass loss. Consequently, it is imperative to consider sliding velocity when calculating wear coefficients.

6.2 Future work

Several topics addressed in this dissertation require further investigation:

- Conduct additional tests to improve the reliability and robustness of DSC and DMA results;
- Increase the accuracy of the regression obtained through the Taguchi method by conducting more Pin-on-Disc tests;
- Extend the investigation to encompass a wider range of $p \cdot v_s$ values across different materials;
- Execute Pin-on-Disc tests with diverse conditions and material combinations to validate the regression models developed for mass loss and temperature responses;
- Enhance the numerical calculation methodology to better accommodate different operating conditions;
- Adjust the Pin-on-Disc results for a gear condition with the aim of being applied in experimental procedures;

REFERENCES

- [1] D. M. Rocha, “Bulk Temperature in Polymer-Metal Hybrid Gears,” 2017.
- [2] A. F. Lourenço, “Testing of Low-Loss Polymer Gears,” p. 174, 2015.
- [3] VDI, “VDI 2736 Blatt 1,” Verlag des Vereins Deutscher Ingenieure, Tech. Rep., 2016.
- [4] A. B. Cropper, “The failure mode analysis of plastic gears,” Ph.D. dissertation, University of Birmingham, 2003.
- [5] C. M. C. G. Fernandes, D. M. P. Rocha, R. C. Martins, L. Magalhães, and J. H. O. Seabra, “Hybrid Polymer Gear Concepts to Improve Thermal Behavior,” *Journal of Tribology*, vol. 141, no. 3, pp. 1–12, mar 2019. [Online]. Available: <https://asmedigitalcollection.asme.org/tribology/article/doi/10.1115/1.4041461/449503/Hybrid-Polymer-Gear-Concepts-to-Improve-Thermal>
- [6] K. Mao, “A new approach for polymer composite gear design,” vol. 262, no. November 2005, pp. 432–441, 2007.
- [7] M. Rothmund, M. Otto, and K. Stahl, “Applicability of classic analytical calculation approaches for the design of plastic gears,” *Forschung im Ingenieurwesen/Engineering Research*, vol. 87, no. 3, pp. 987–995, 2023.
- [8] K. Mao, P. Langlois, Z. Hu, K. Alharbi, X. Xu, M. Milson, W. Li, C. J. Hooke, and D. Chetwynd, “The wear and thermal mechanical contact behaviour of machine cut polymer gears,” *Wear*, vol. 332-333, pp. 822–826, 2015.
- [9] N. Tsukamoto, “Argument on plastic gears for power transmission,” *JSME International Journal*, vol. 38, no. 10, p. 8, 1995.
- [10] Z. Lu, H. Liu, P. Wei, C. Zhu, D. Xin, and Y. Shen, “The effect of injection molding lunker defect on the durability performance of polymer gears,” *International Journal of Mechanical Sciences*, vol. 180, no. January, p. 105665, 2020. [Online]. Available: <https://doi.org/10.1016/j.ijmecsci.2020.105665>
- [11] B. S. Solanki, H. Singh, and T. Sheorey, “Effect of injection molding cycle time on shrinkage and weight of manufactured polymer gear,” *Materials Today: Proceedings*, vol. 62, no. P14, pp. A1–A6, 2022. [Online]. Available: <https://doi.org/10.1016/j.matpr.2022.06.430>

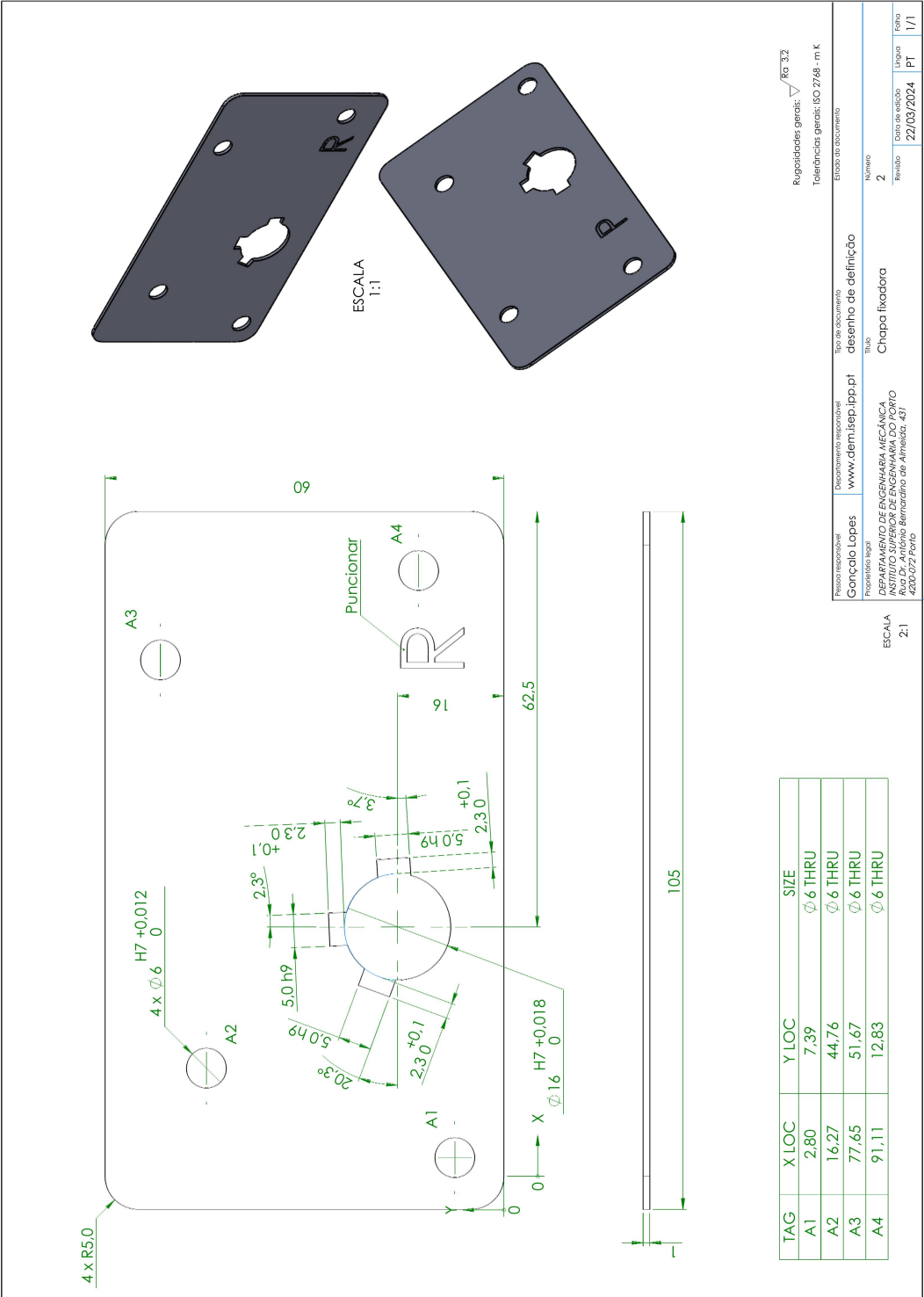
-
- [12] S. Manakasettharn, J. A. Taylor, and T. Krupenkin, "Superhydrophobicity at Micron and Submicron Scale," in *Comprehensive Nanoscience and Technology*, D. L. Andrews, G. D. Scholes, and G. P. Wiederrecht, Eds. Amsterdam: Academic Press, 2011, pp. 383–411. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/B9780123743961001343>
 - [13] E. A. Campo, *The Complete Part Design Handbook For Injection Molding of Thermoplastics*. Carl Hanser Verlag, 2006.
 - [14] C. M. C. G. Fernandes, D. M. P. Rocha, R. C. Martins, L. Magalhães, and J. H. O. Seabra, "Tribology International Finite element method model to predict bulk and flash temperatures on polymer gears," vol. 120, no. October 2017, pp. 255–268, 2018.
 - [15] V. D. Ingenieure, "INGENIEURE VDI 2736," no. June, 2014.
 - [16] K. Mao, W. Li, C. J. Hooke, and D. Walton, "Friction and wear behaviour of acetal and nylon gears," *Wear*, vol. 267, no. 1-4, pp. 639–645, 2009.
 - [17] N. Tsukamoto and K. Terashima, "Development of Plastic Gears for Power Transmission," *JSME International Journal*, vol. 20, no. 10, pp. 2278–2281, 1986.
 - [18] J. A. Brandão, "Gear tooth flank damage prediction using high-cycle fatigue and wear models," Ph.D. dissertation, Faculdade de Engenharia da Universidade do Porto, 2013.
 - [19] K. A. M. Alharbi, "The Mechanical Contact Behaviour and Tribology of Polymer Gears by," Ph.D. dissertation, University of Warwick, 2018.
 - [20] T. König, L. Cadau, L. Steidle, D. C. Güney, J. Albrecht, K. Weber, and M. Kley, "A concept for comparison of new and aged lubricants in transmissions of electric vehicles and a method of oil aging on a test rig," *Forschung im Ingenieurwesen/Engineering Research*, vol. 87, no. 3, pp. 1069–1080, 2023.
 - [21] W. Dresel and R. P. Heckler, "Lubricating Greases," *Lubricants and Lubrication: Second Edition*, pp. 648–693, 2007.
 - [22] A. Emery, S. Etemad, and J. Wolak, "Interfacial temperatures and surface heat fluxes during a blade-seal rubbing process," University of Washington, Tech. Rep., 1981. [Online]. Available: <https://arc.aiaa.org/doi/abs/10.2514/6.1981-1165>
 - [23] B. B. Mikić, "Thermal contact conductance; theoretical considerations," *International Journal of Heat and Mass Transfer*, vol. 17, no. 2, pp. 205–214, 1974. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/0017931074900829>
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- [24] J. J. Fuller and E. E. Marotta, "Thermal Contact Conductance of Metal/Polymer Joints: An Analytical and Experimental Investigation," *Journal of Thermophysics and Heat Transfer*, vol. 15, no. 2, pp. 228–238, 2001. [Online]. Available: <https://doi.org/10.2514/2.6598>
 - [25] K. Mao, D. G. Chetwynd, and M. Millson, "A new method for testing polymer gear wear rate and performance," *Polymer Testing*, vol. 82, p. 106323, 2020. [Online]. Available: <https://doi.org/10.1016/j.polymertesting.2019.106323>
 - [26] K. Mao, "The Performance of dry running Non-Metallic Gears," Ph.D. dissertation, University of Birmingham, 1993.
 - [27] D. J. Hargreaves and A. Planitz, "Assessing the energy efficiency of gear oils via the FZG test machine," *Tribology International*, vol. 42, no. 6, pp. 918–925, 2009.
 - [28] H. Blok, "The flash temperature concept," *Wear*, vol. 6, no. 6, pp. 483–494, 1963. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/0043164863902837>
 - [29] S. M. Evans and P. S. Keogh, "Efficiency and running temperature of a polymer-steel spur gear pair from slip/roll ratio fundamentals," *Tribology International*, vol. 97, pp. 379–389, 2016. [Online]. Available: <http://dx.doi.org/10.1016/j.triboint.2016.01.052>
 - [30] Zedong Hu, "An Investigation into the Wear and Thermal - mechanical Performance of Polyacetal Gears," Ph.D. dissertation, University of Warwick, 2017.
 - [31] K. Mao, W. Li, C. J. Hooke, and D. Walton, "Polymer gear surface thermal wear and its performance prediction," *Tribology International*, vol. 43, no. 1-2, pp. 433–439, 2010.
 - [32] W. N. Findley, J. S. Lai, and K. Onaran, "Linear Viscoelastic Constitutive Equations," in *Creep and Relaxation of Nonlinear Viscoelastic Materials*, ser. North-Holland Series in Applied Mathematics and Mechanics. North-Holland, 1976, vol. 18, ch. 5, pp. 50–107. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/B9780720423693500084>
 - [33] D. Dunson, "Characterization of Polymers using Dynamic Mechanical Analysis (DMA)," *EAG Appl Note*, p. 7, 2017. [Online]. Available: <https://www.eag.com/wp-content/uploads/2017/09/M-022717-Characterization-of-Polymers-using-Dynamic-Mechanical-Analysis.pdf>
 - [34] T. G. Mezger, *Applied Rheology – With Joe Flow on Rheology Road*. Graz: Anton Paar GmbH, 2018.
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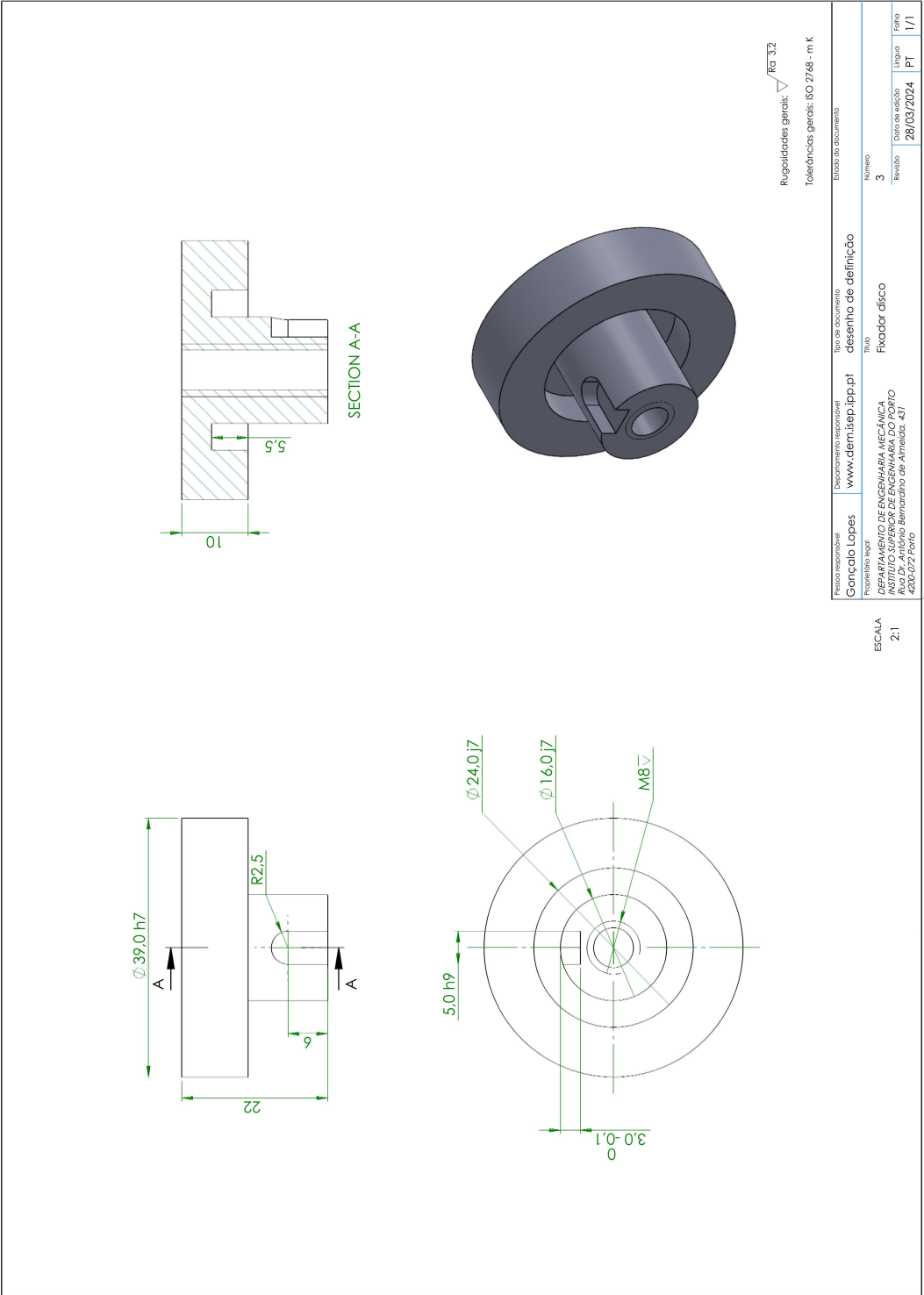
-
- [35] “DMA 242 E Artemis netzsch,” <https://analyzing-testing.netzsch.com/en/products/dynamic-mechanical-analysis-dma/dma-242-e-artemis>, accessed: 2024-06-14.
 - [36] Neil P. Doll, “Modeling Thermomechanical Behavior of Polymer Gears,” Ph.D. dissertation, University of Wisconsin-Madison, 2015.
 - [37] M. Conte, B. Pinedo, and A. Igartua, “Role of crystallinity on wear behavior of PTFE composites \$,” *Wear*, vol. 307, no. 1-2, pp. 81–86, 2013. [Online]. Available: <http://dx.doi.org/10.1016/j.wear.2013.08.019>
 - [38] R. L. Blaine, “Polymer Heats of Fusion,” *TA Instruments*, pp. 1–2, 2002. [Online]. Available: <http://www.tainstruments.com/pdf/literature/TN048.pdf>
 - [39] ASTM, “Standard Test Method for Wear Testing with a Pin-on-Disk Apparatus 1,” ASTM, Tech. Rep., 2020.
 - [40] H. Benabdallah and D. Olender, “Finite element simulation of the wear of polyoxymethylene in pin-on-disc configuration,” *Wear*, vol. 261, no. 11, pp. 1213–1224, 2006. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0043164806001323>
 - [41] J. Lawson, *Design and Analysis of Experiments with R*, ser. Chapman & Hall/CRC Texts in Statistical Science. CRC Press, 2014. [Online]. Available: <https://books.google.pt/books?id=TOxMBgAAQBAJ>
 - [42] Y. Jia, Z. Mao, W. Huang, and J. Zhang, “Effect of temperature and crystallinity on the thermal conductivity of semi-crystalline polymers: A case study of polyethylene,” *Materials Chemistry and Physics*, vol. 287, no. June, p. 126325, 2022. [Online]. Available: <https://doi.org/10.1016/j.matchemphys.2022.126325>
 - [43] “Twenty-node brick element (C3D20 and F3D20) massachusetts institute of technology,” https://web.mit.edu/calculix_v2.7/CalculiX/ccx_2.7/doc/ccx/node29.html, accessed: 2024-06-12.
 - [44] J. Lawson, *Design and Analysis of Experiments with R*, ser. Chapman & Hall/CRC Texts in Statistical Science. CRC Press, 2014. [Online]. Available: <https://books.google.pt/books?id=TOxMBgAAQBAJ>
 - [45] M. P. Charro, “Development of Tools and Manufacturing Processes for Hybrid Polymer Gears,” Ph.D. dissertation, Faculdade de Engenharia da Universidade do Porto, 2023.
 - [46] J. Michalski, I. Rokach, and M. Borovinšek, “PrePoMax 2.1.0 Manual,” 2024.

APPENDIX A

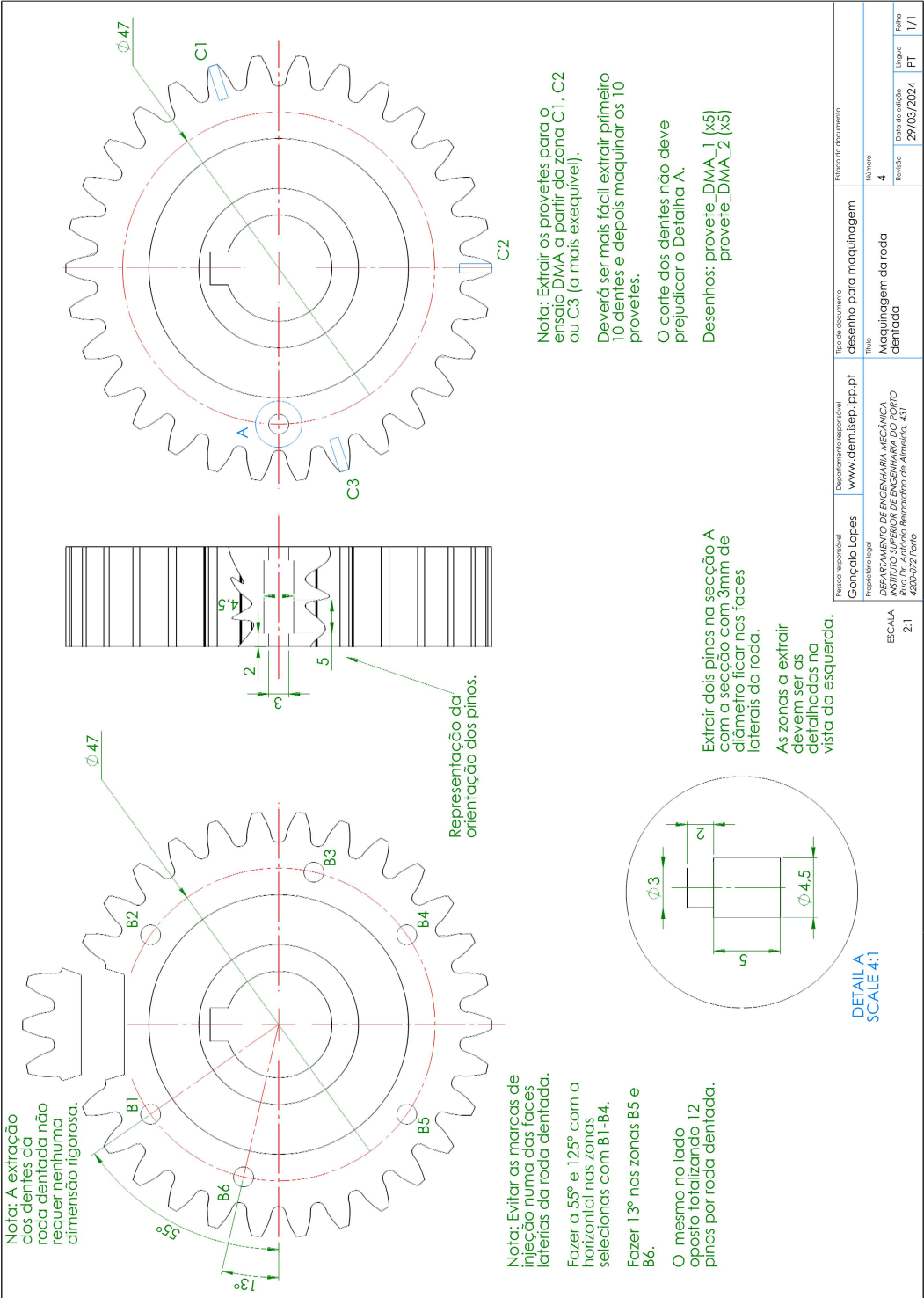
A.1 2D DRAWINGS

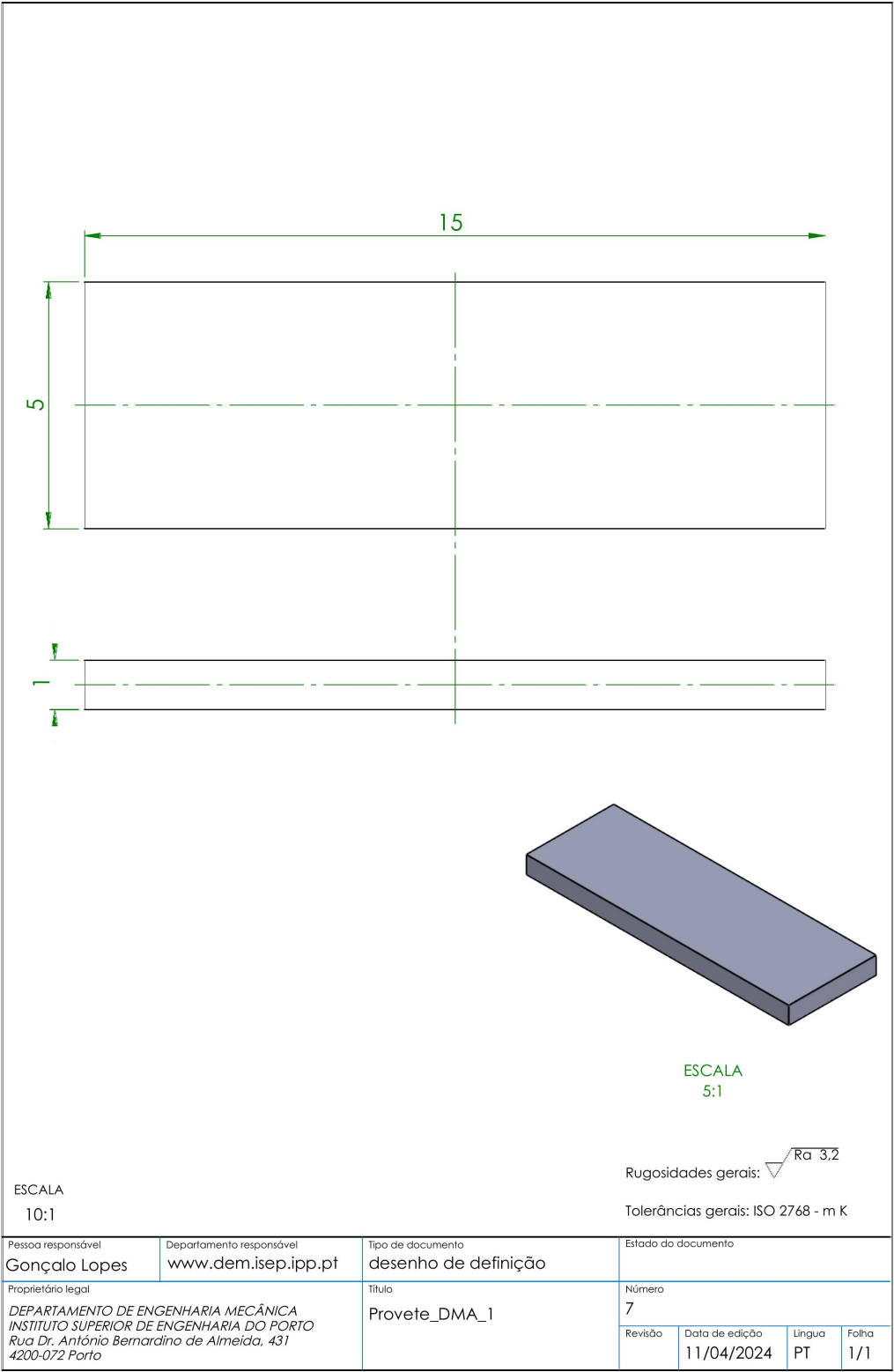


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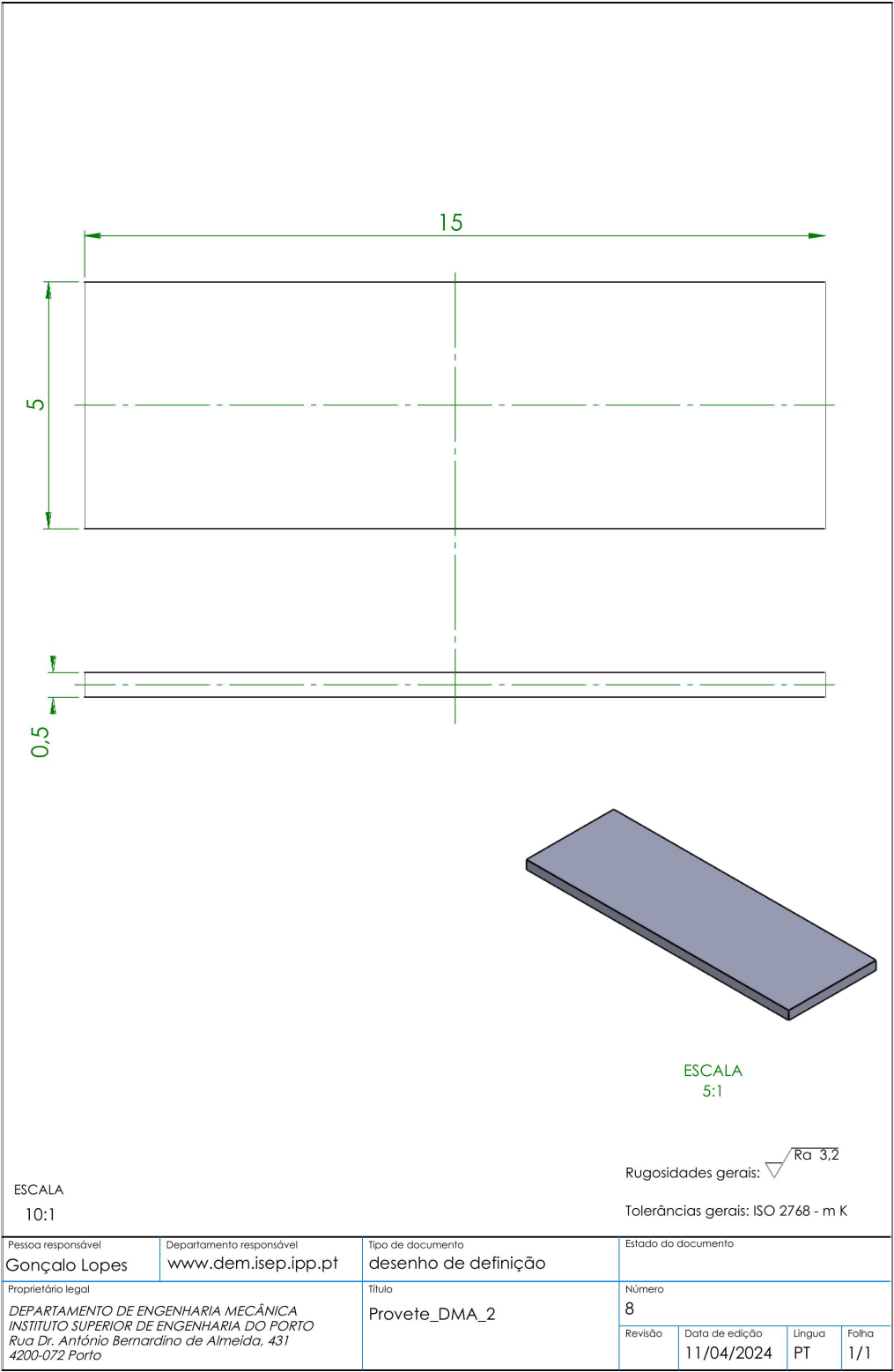


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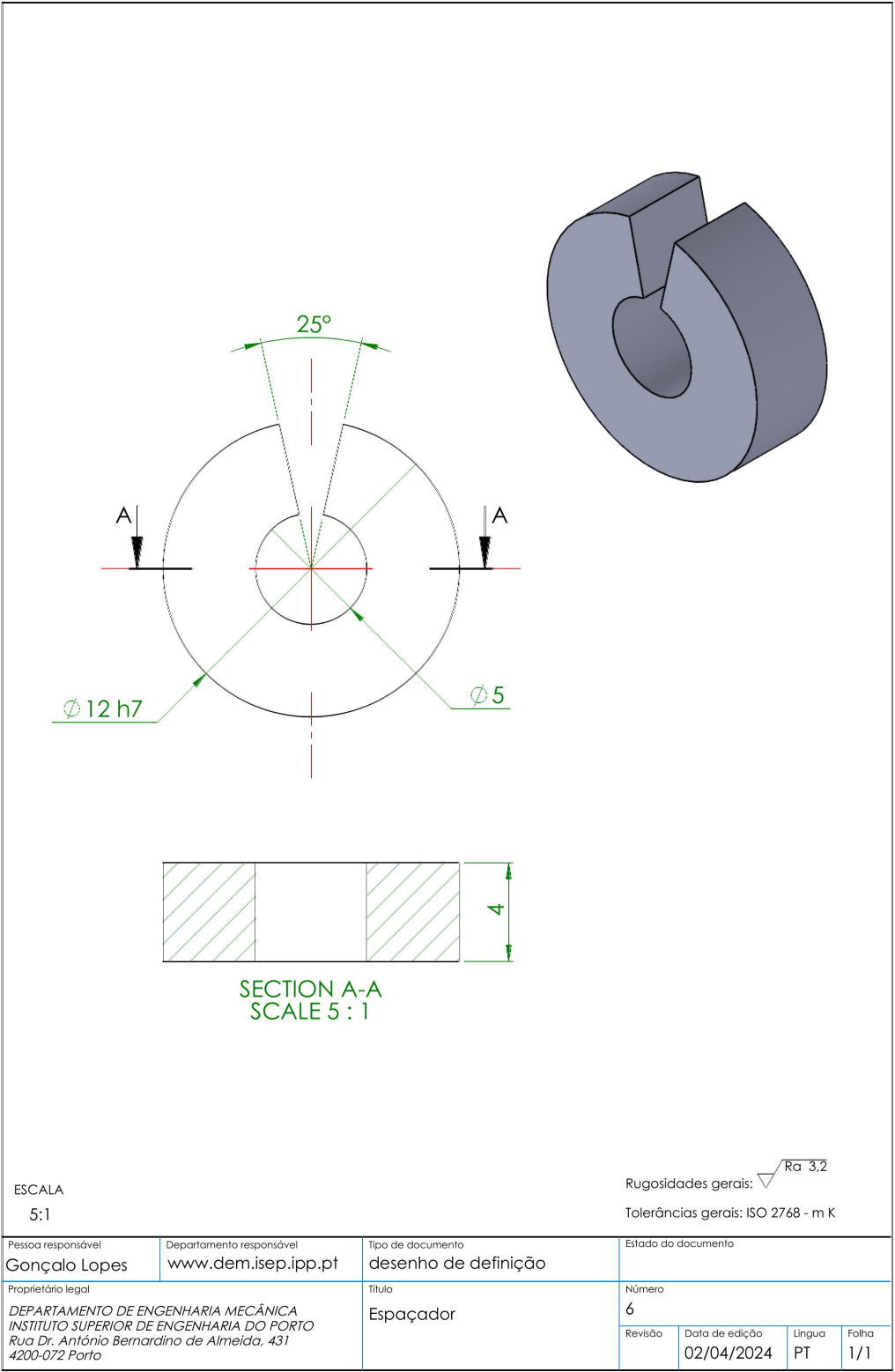




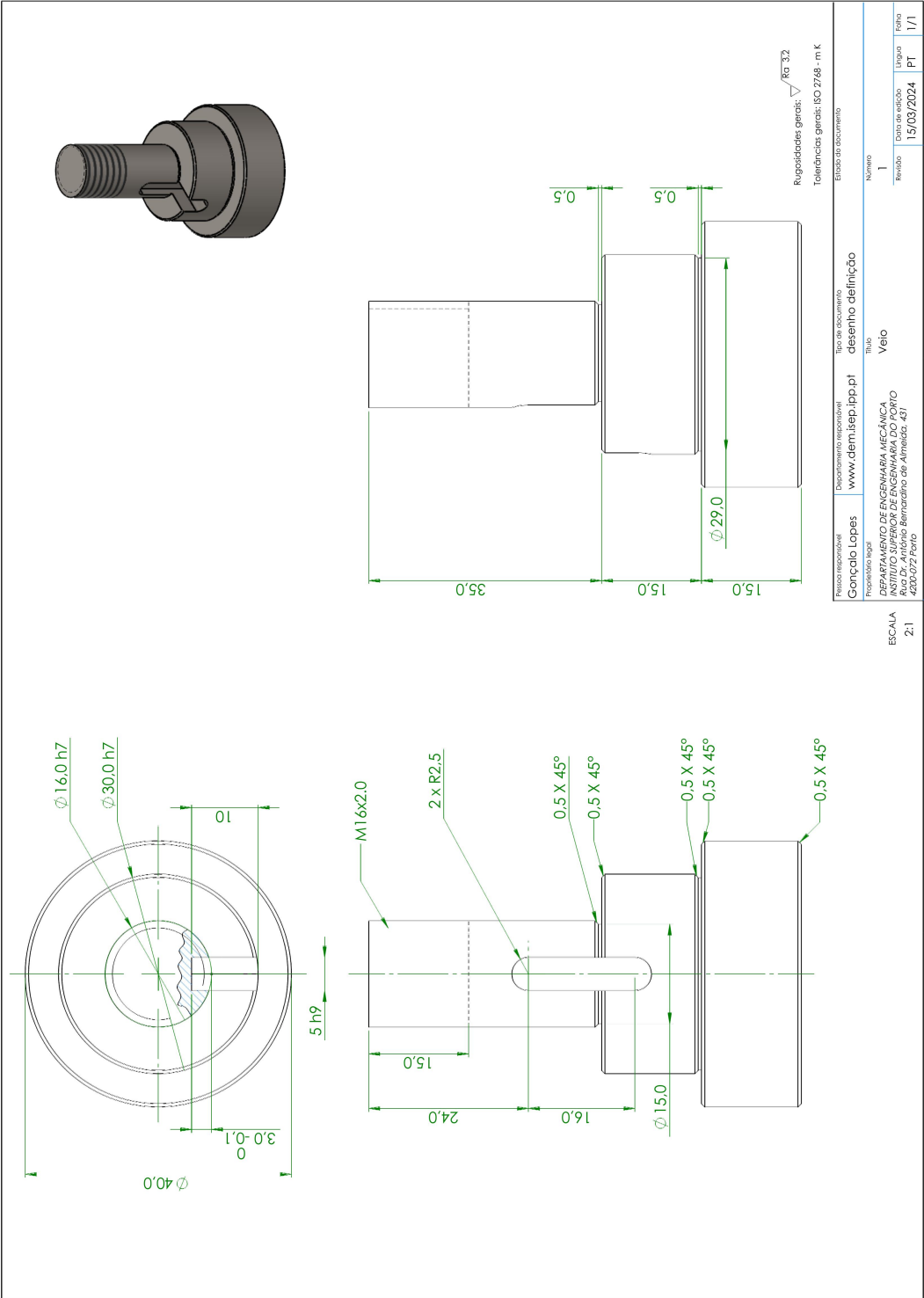
SOLIDWORKS Educational Product. For Instructional Use Only.

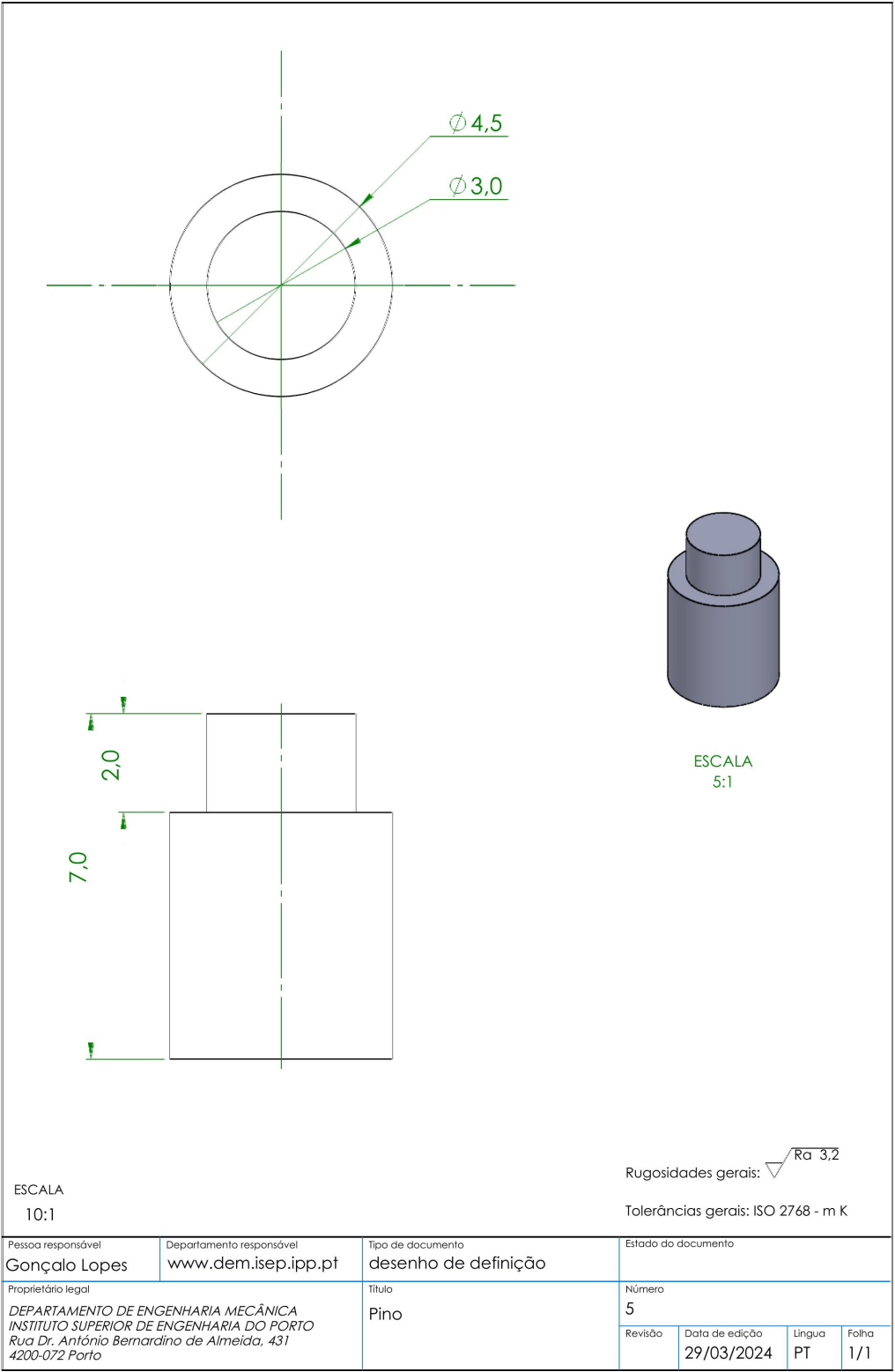


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SOLIDWORKS Educational Product. For Instructional Use Only.

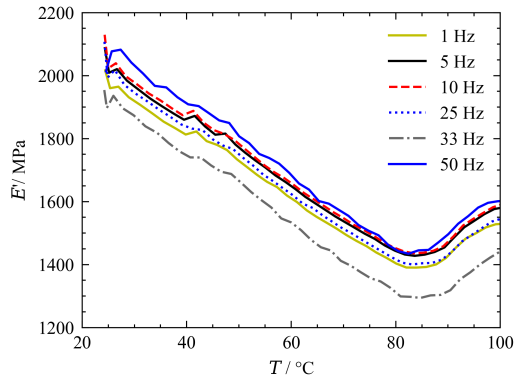




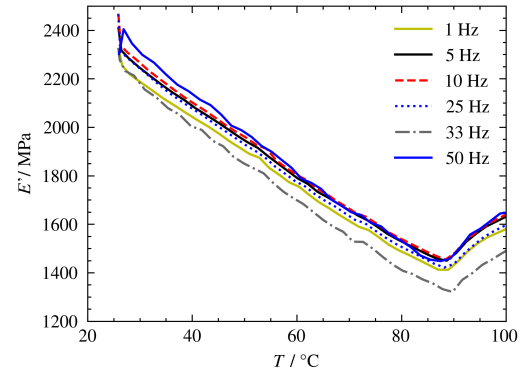
SOLIDWORKS Educational Product. For Instructional Use Only.

APPENDIX B

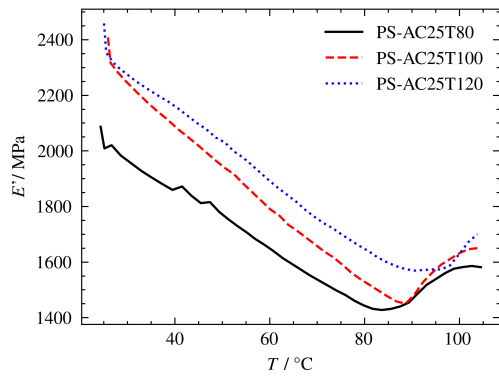
B.1 DMA results



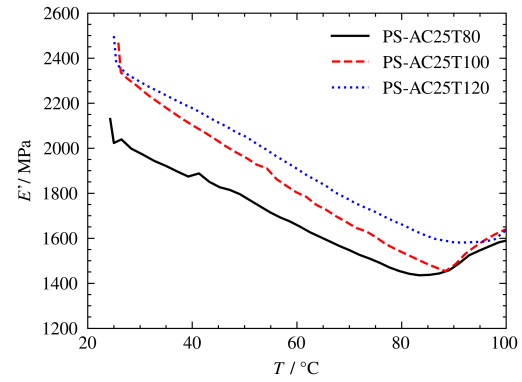
(a) Different frequencies for PS-AC25T80.



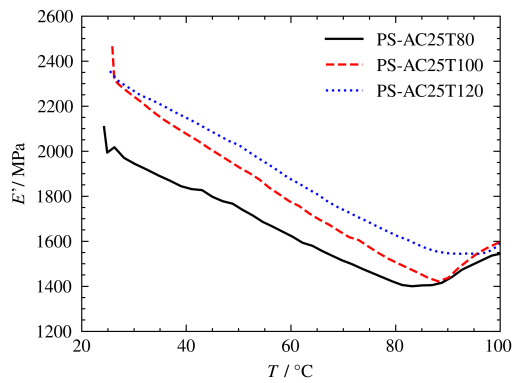
(b) Different frequencies for PS-AC25T100.



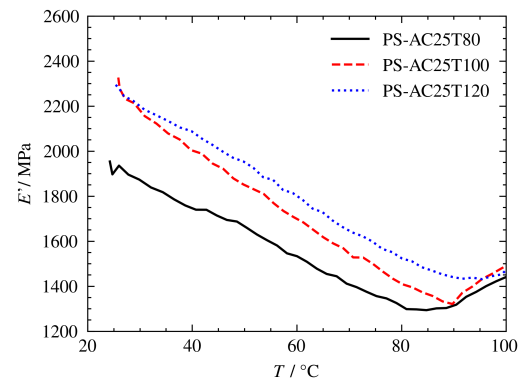
(c) Frequency of 5 Hz.



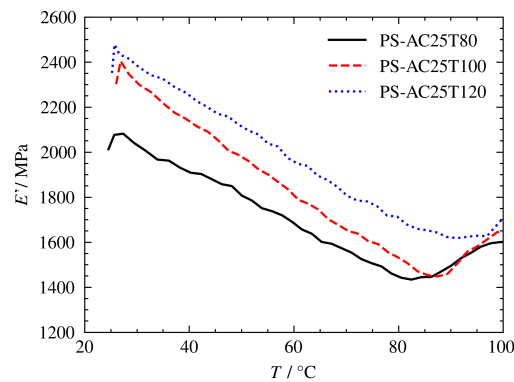
(d) Frequency of 10 Hz.



(e) Frequency of 25 Hz.

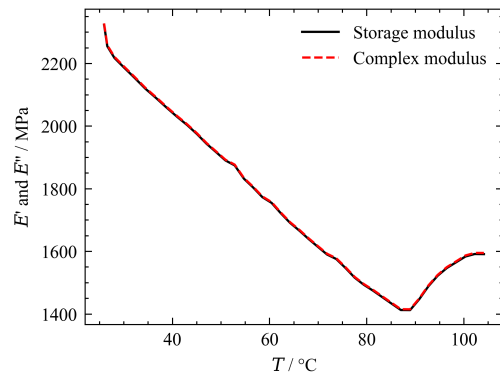


(f) Frequency of 33 Hz.

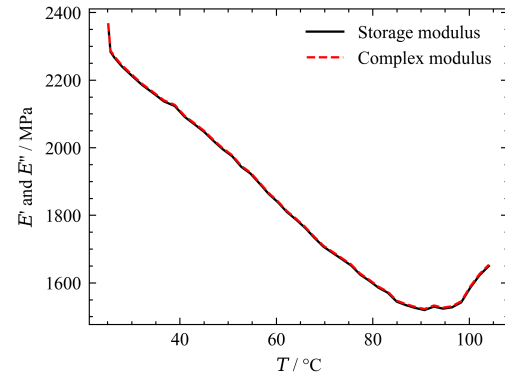


(g) Frequency of 50 Hz.

Figure 106 – Comparison between materials at different frequencies.



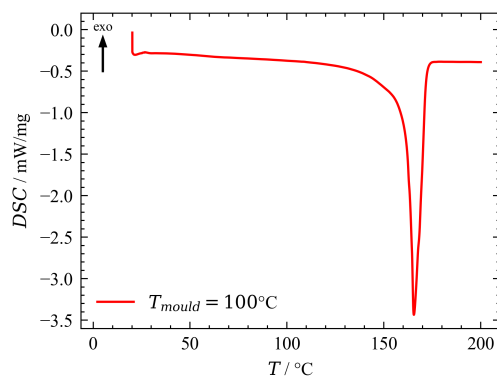
(a) PS-AC25T100.



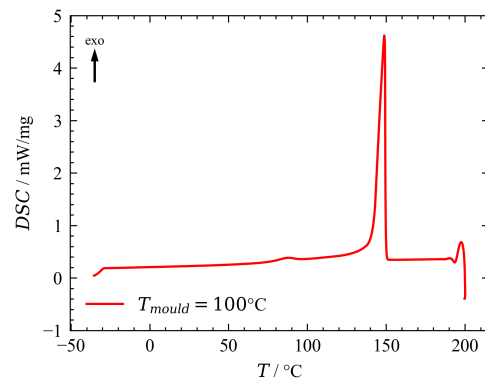
(b) PS-AC25T120.

Figure 107 – Comparison between storage and complex modulus.

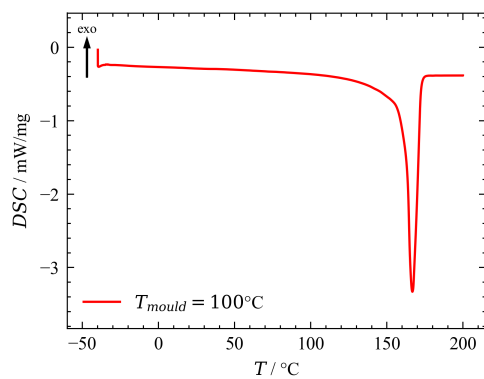
B.2 DSC results



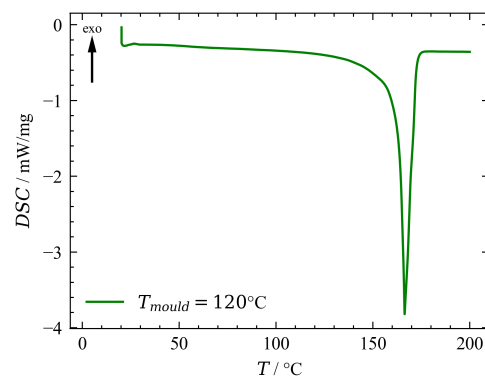
(a) First cycle for PS-AC25T100.



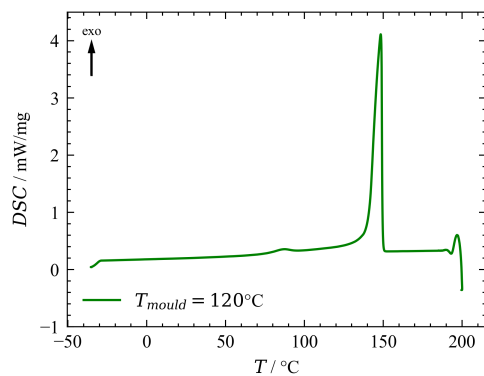
(b) Second cycle for PS-AC25T100.



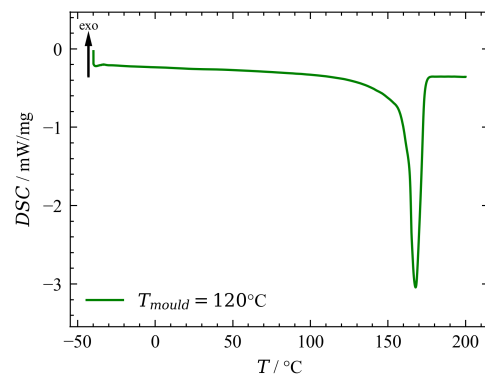
(c) Third cycle for PS-AC25T100.



(d) First cycle for PS-AC25T120.



(e) Second cycle for PS-AC25T120.



(f) Third cycle for PS-AC25T120.

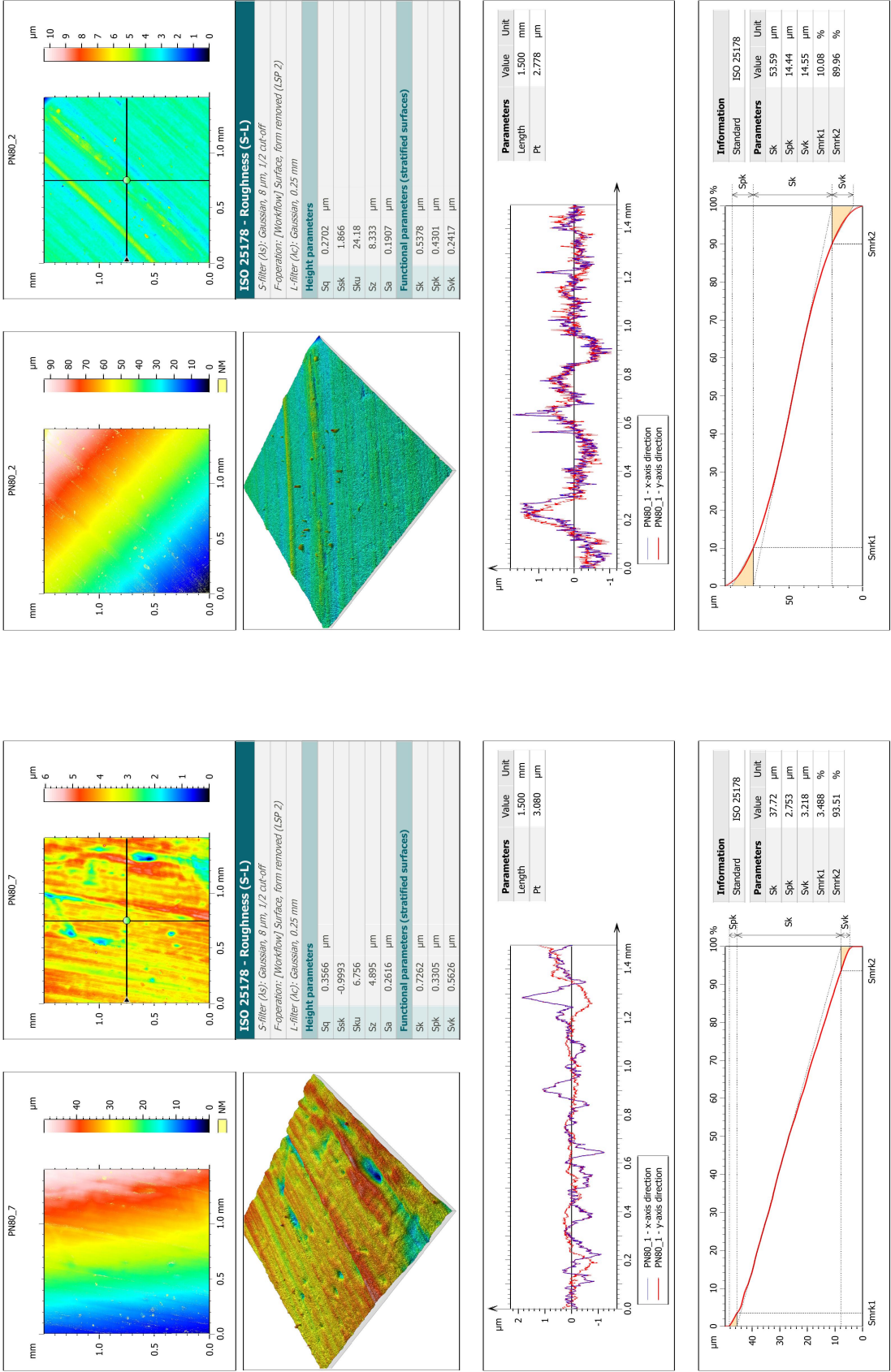
Figure 108 – DSC curves results.

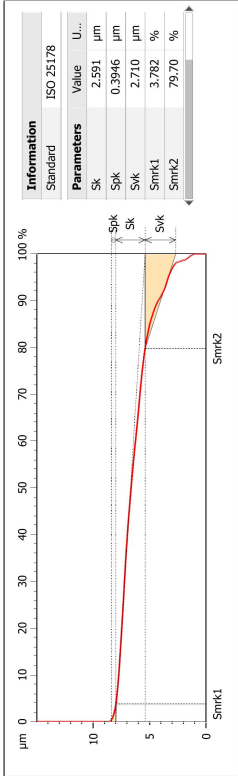
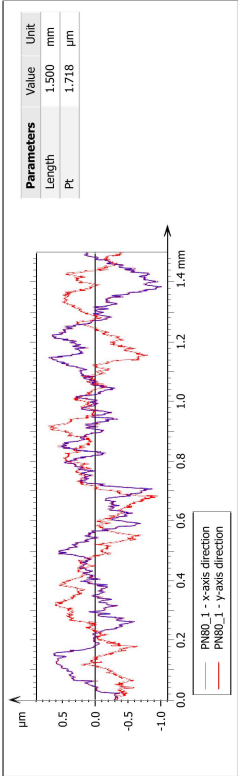
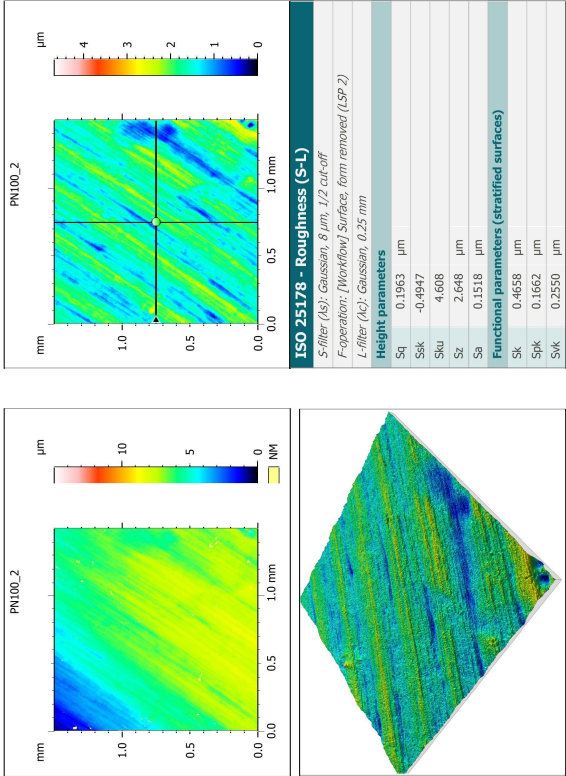
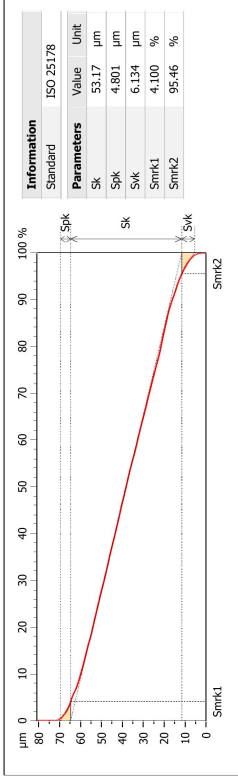
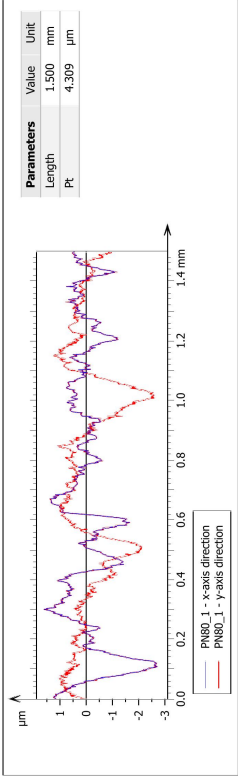
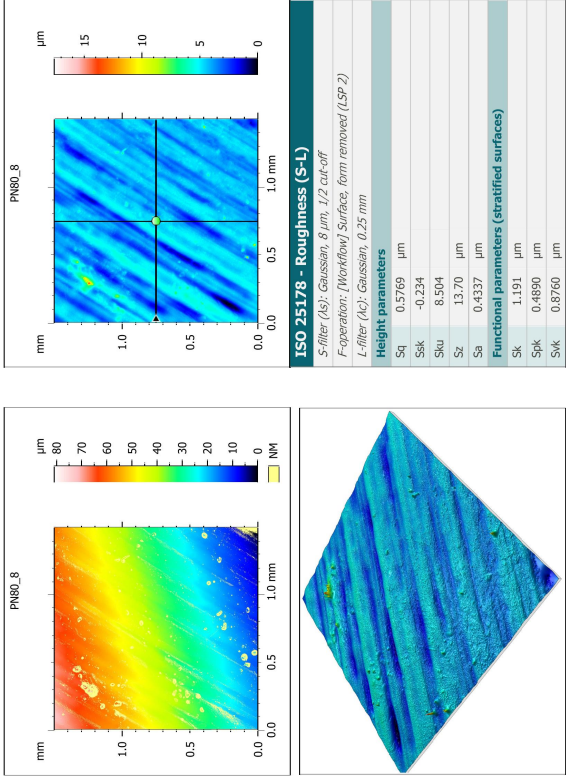
B.3 Hardness

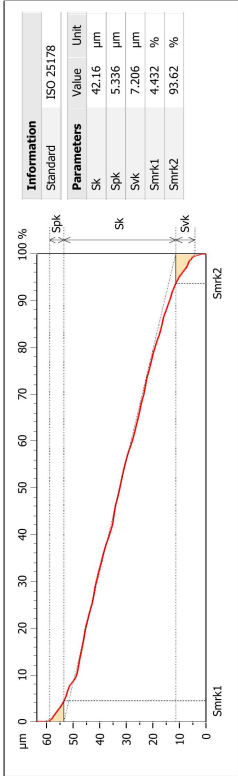
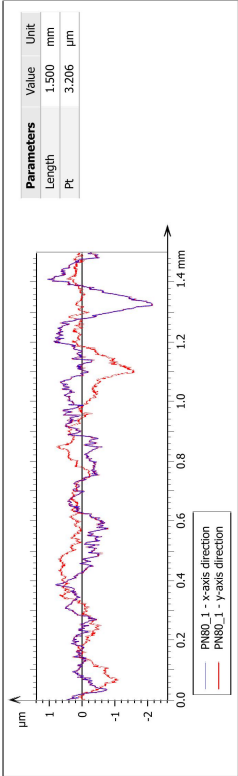
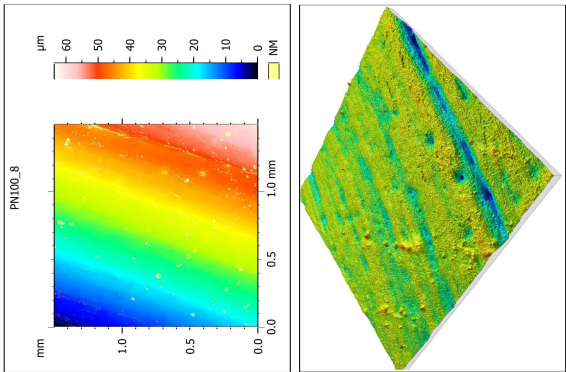
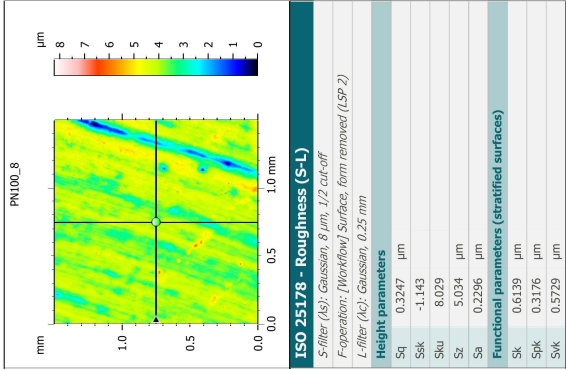
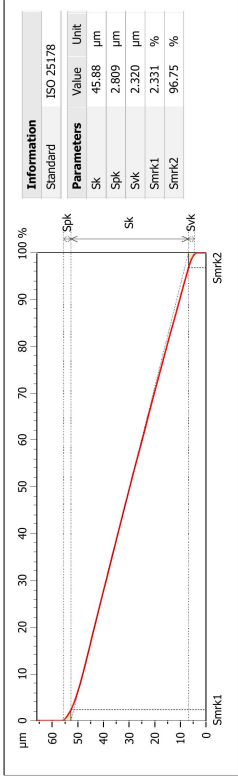
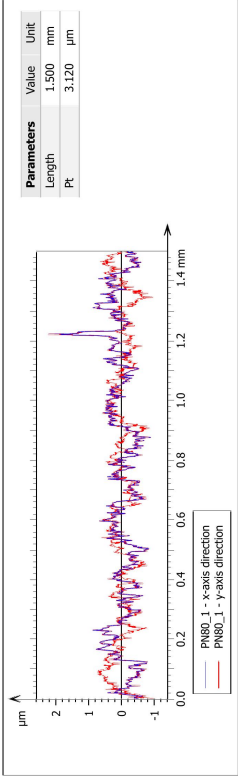
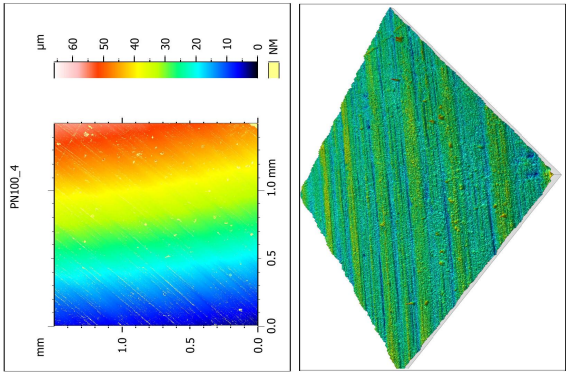
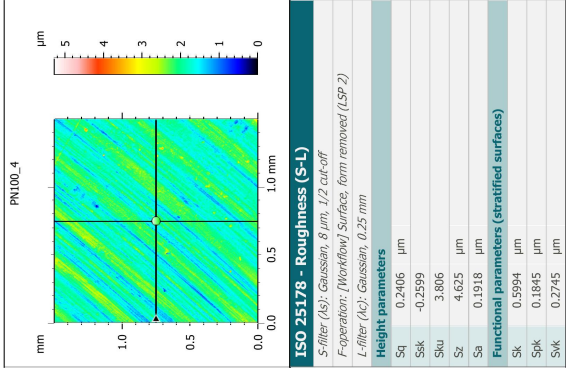
Table 21 – Hardness Shore D

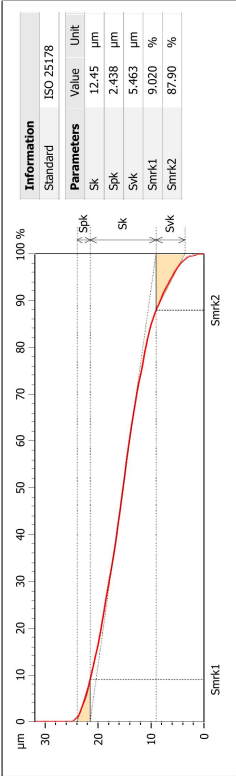
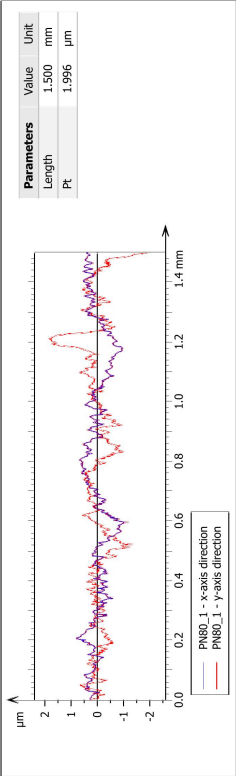
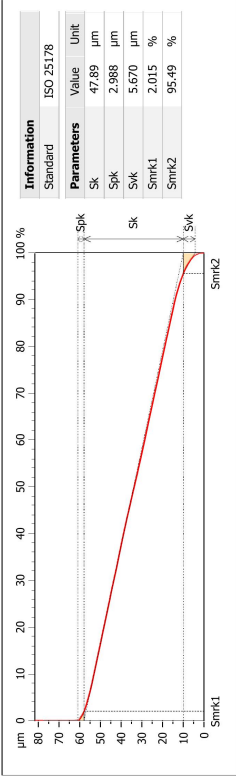
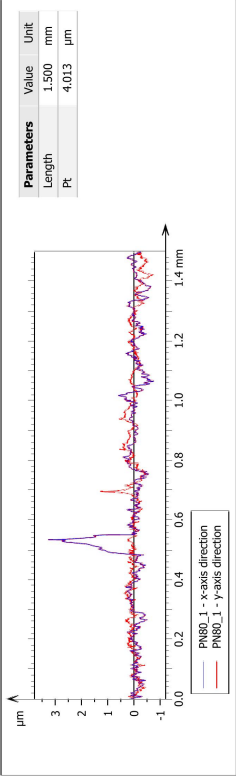
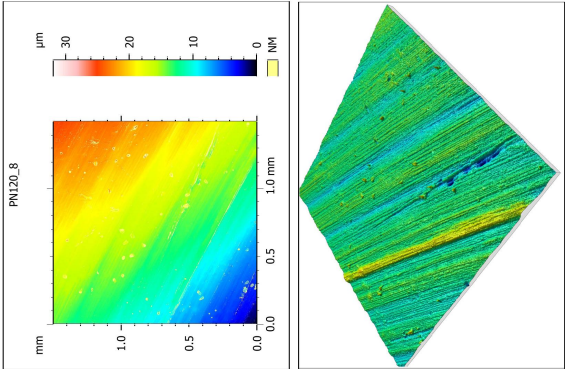
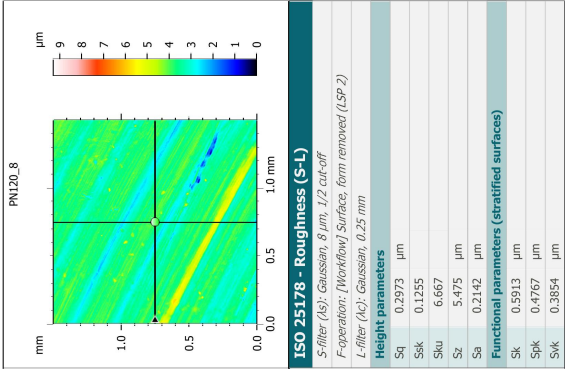
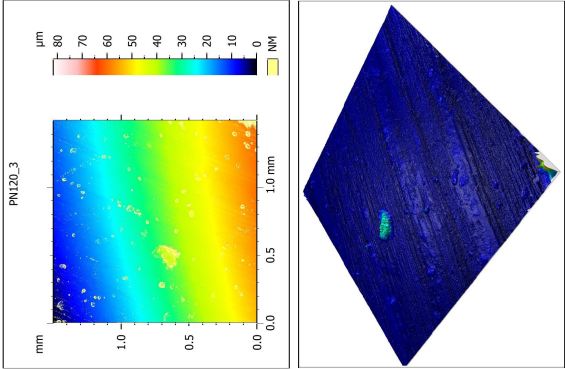
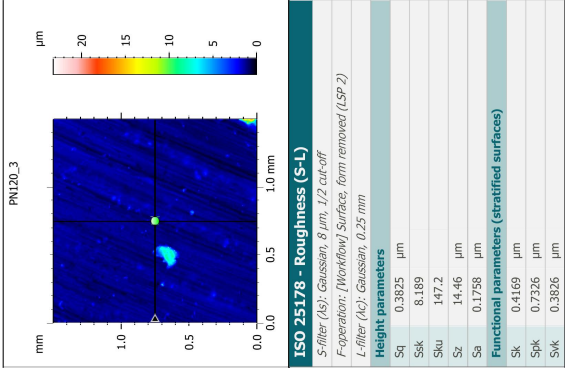
Material		Hardness			
		Nr.	$t = 0\ s$	$t = 3\ s$	$t = 15\ s$
POM	PS-AC25T80	1	82.0	78.5	77.5
		2	83.0	78.5	77.5
		3	84.0	78.0	77.5
		4	83.0	78.5	78.0
		5	83.0	78.5	77.5
	PS-AC25T100	1	81.5	78.0	77.5
		2	82.0	78.0	77.0
		3	81.0	77.0	76.5
		4	80.0	78.5	77.0
		5	81.0	77.5	76.5
	PS-AC25T120	1	82.0	78.0	77.5
		2	83.0	78.5	77.5
		3	82.5	79.0	78.0
		4	83.0	79.0	78.5
		5	83.0	79.0	78.5
PA6-GF	NS-AC25T125	1	74.0	70.5	70.0
		2	75.0	71.0	70.0
		3	74.0	70.5	70.0
		4	75.0	71.0	70.5
		5	74.0	70.5	69.5

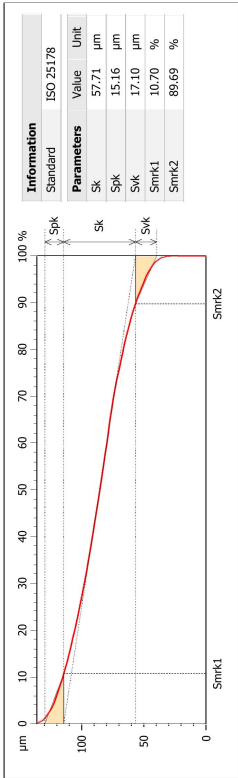
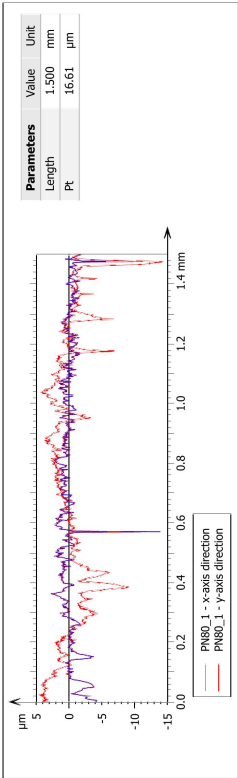
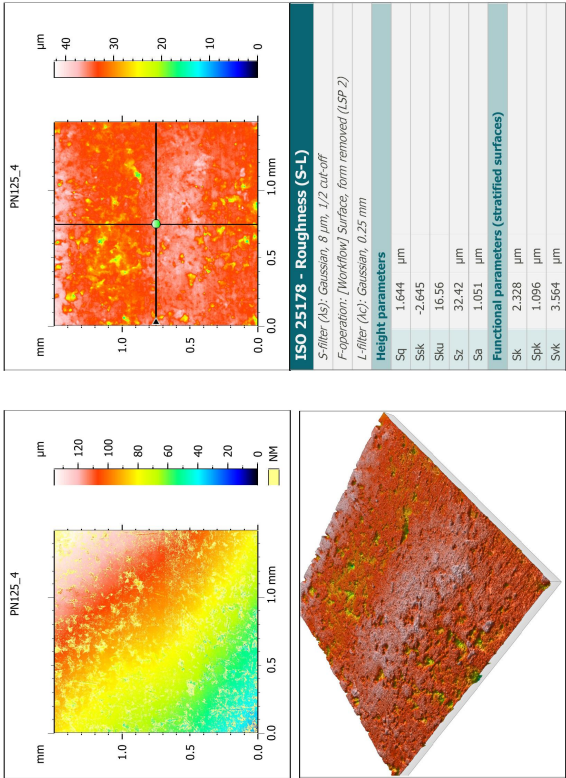
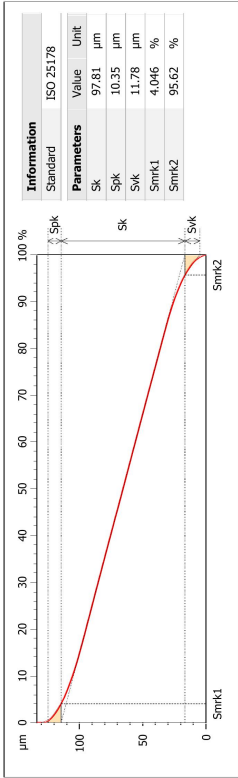
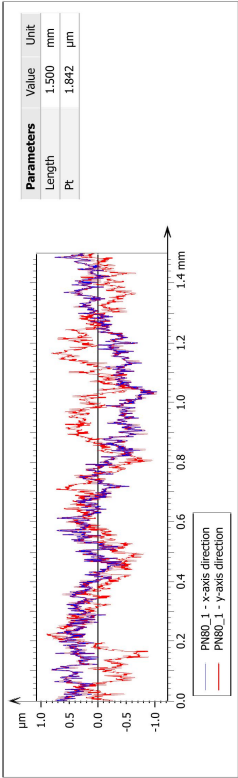
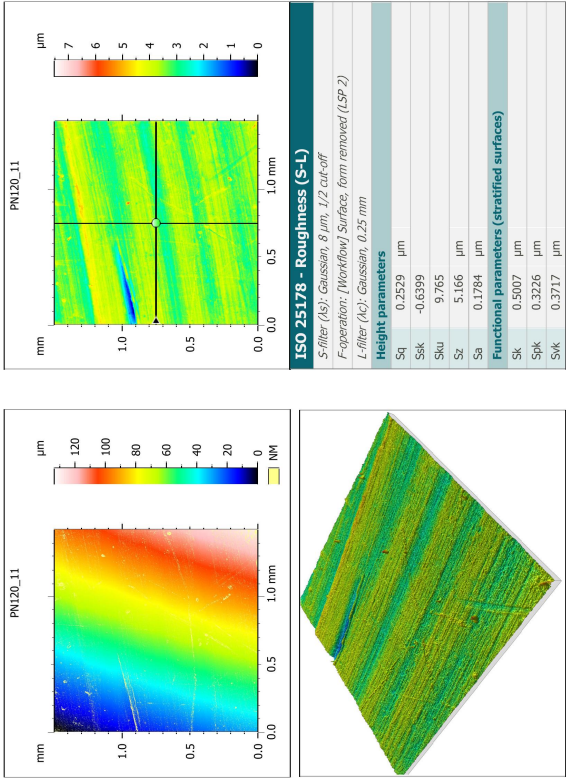
B.4 Roughness before testing

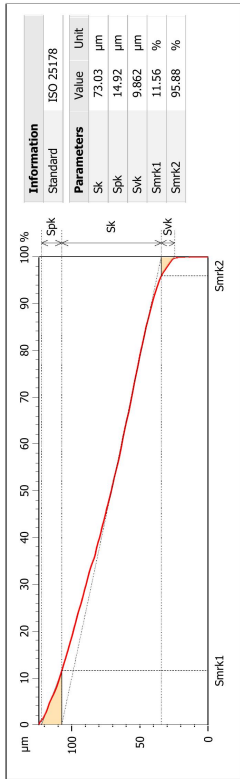
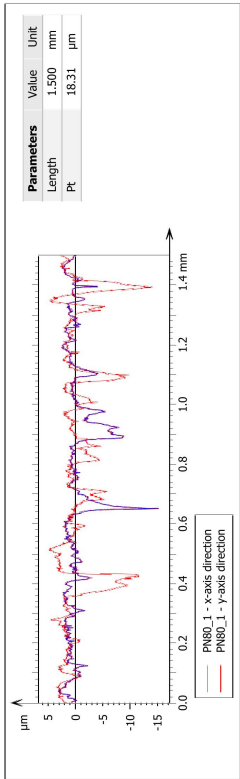
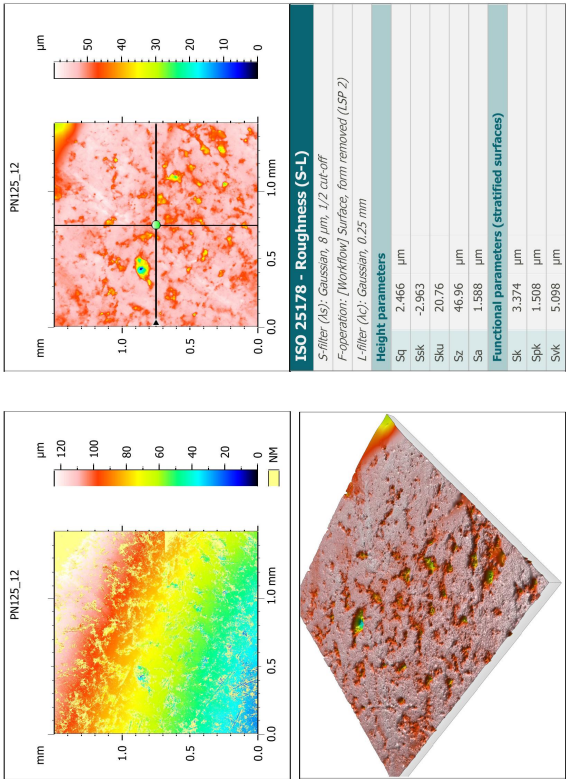
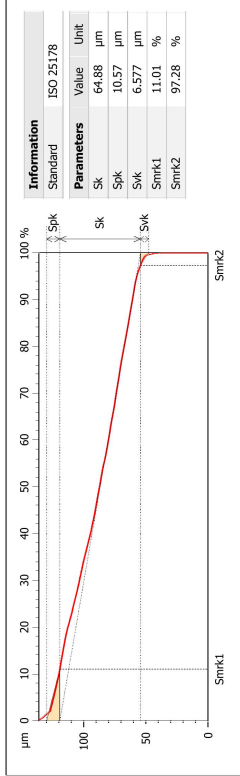
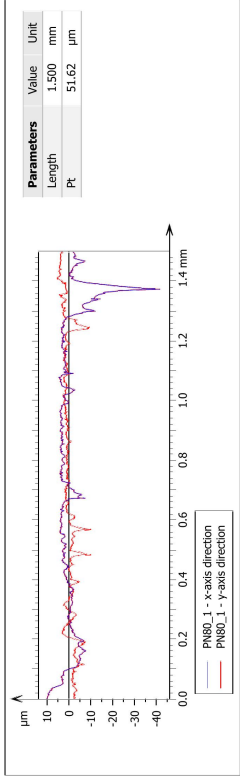
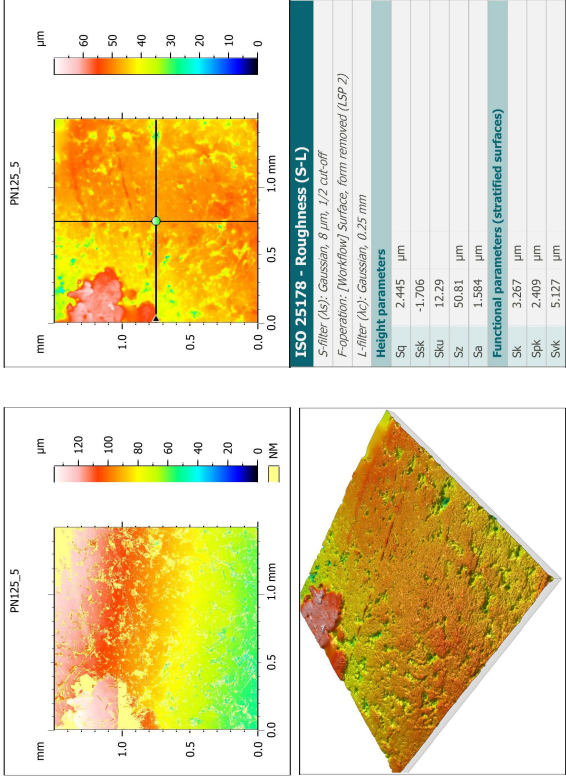




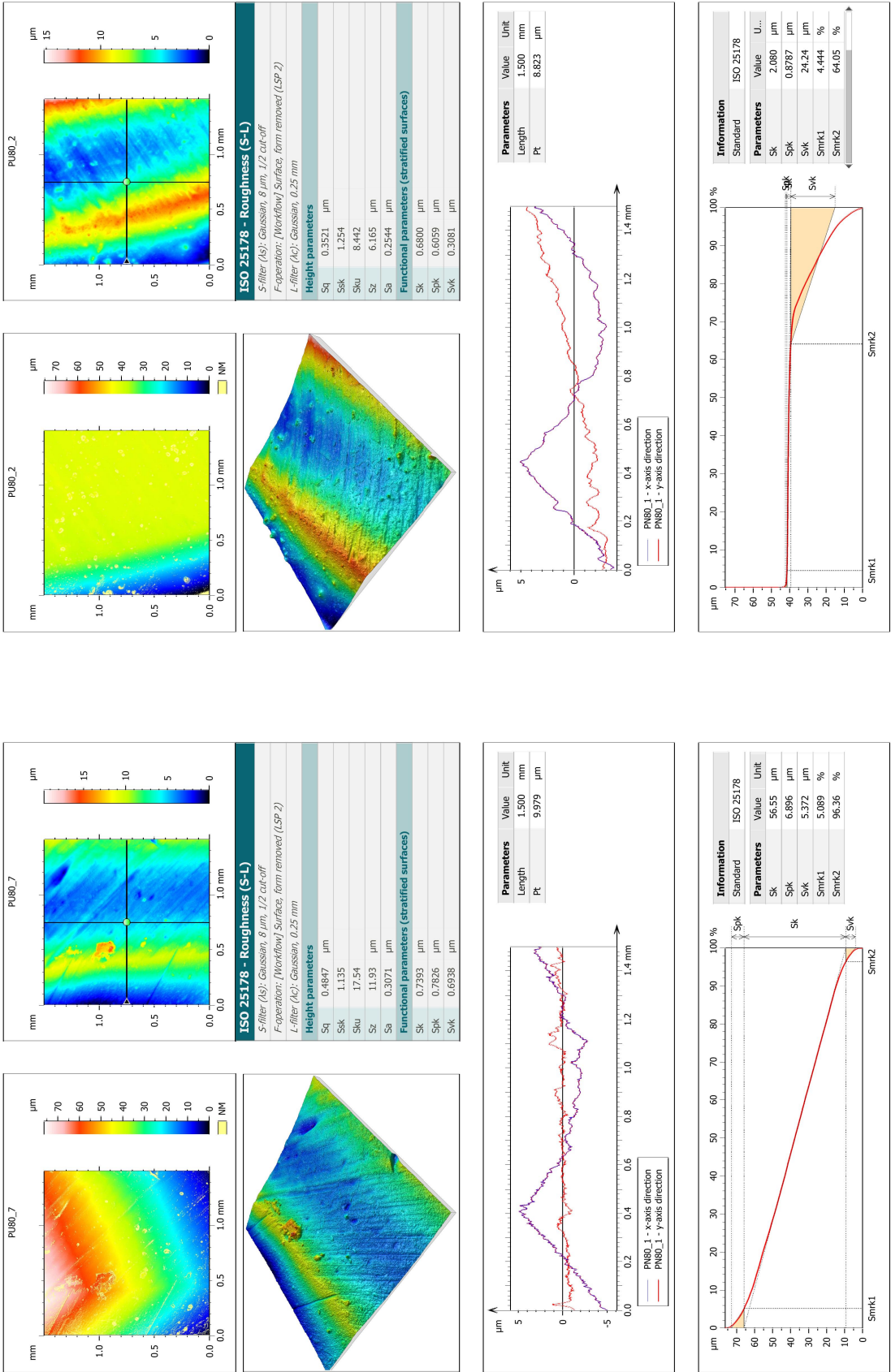


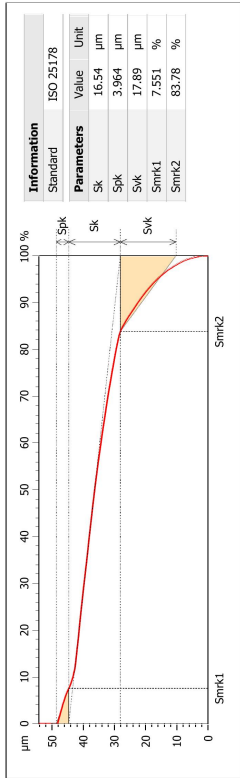
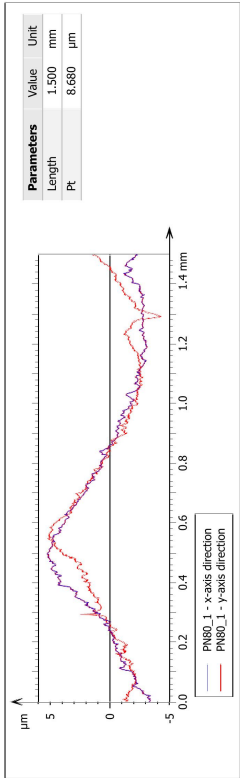
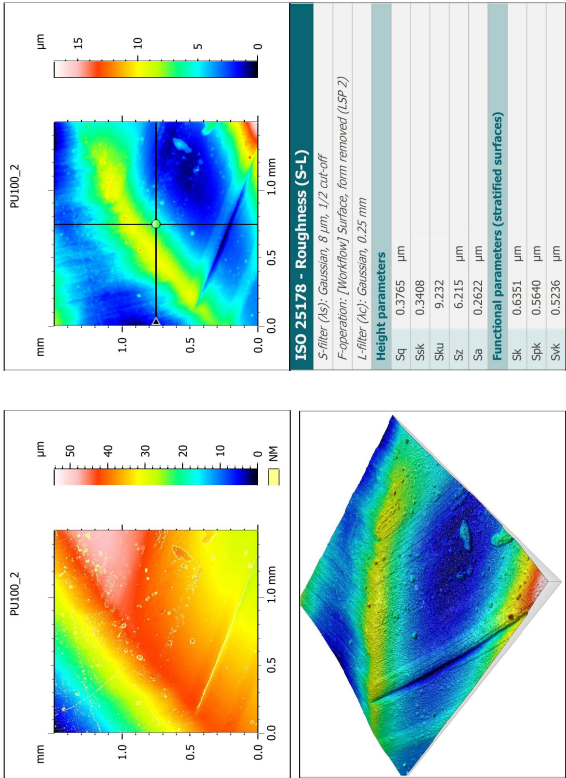
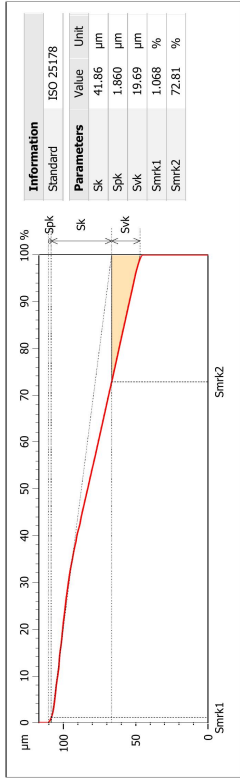
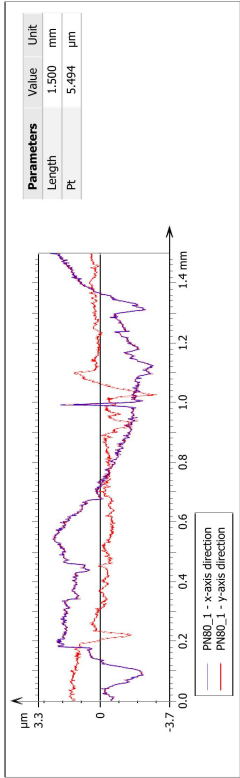
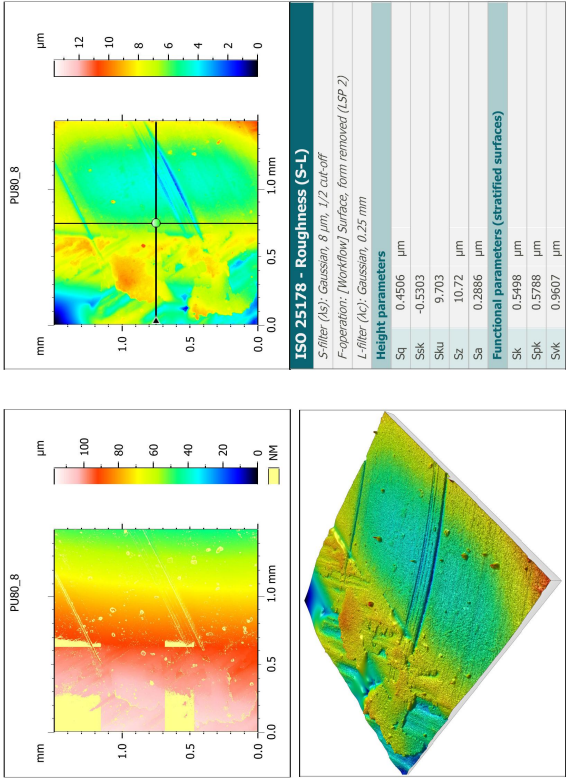


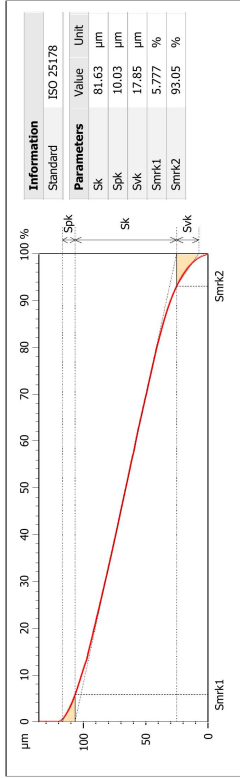
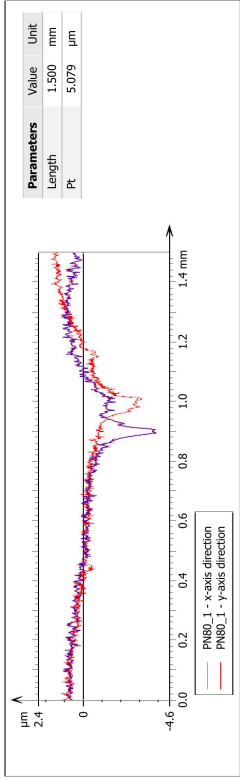
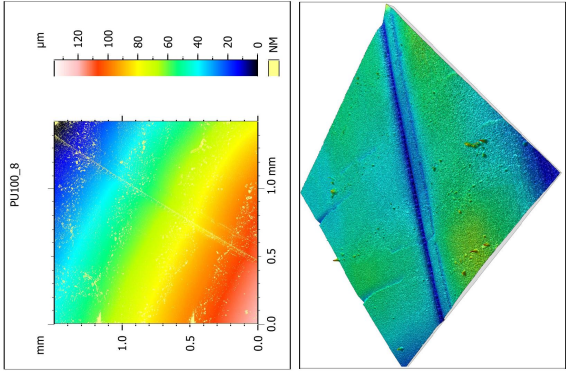
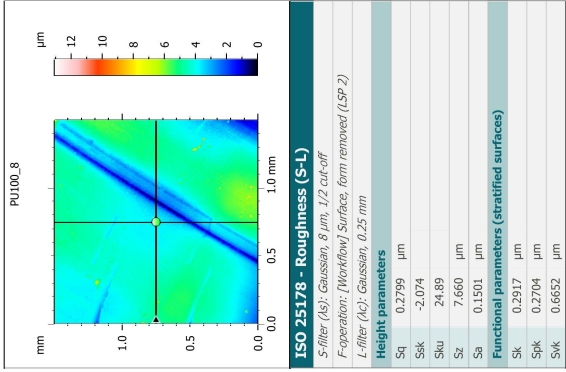
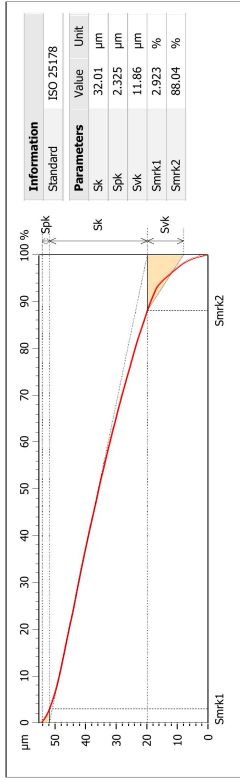
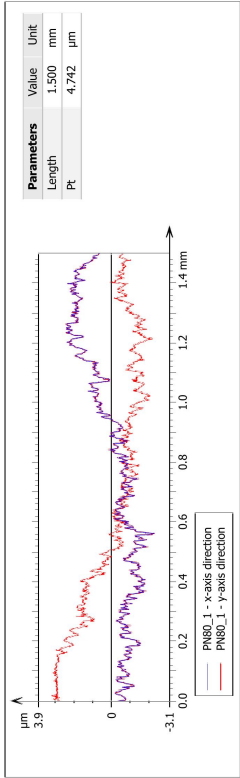
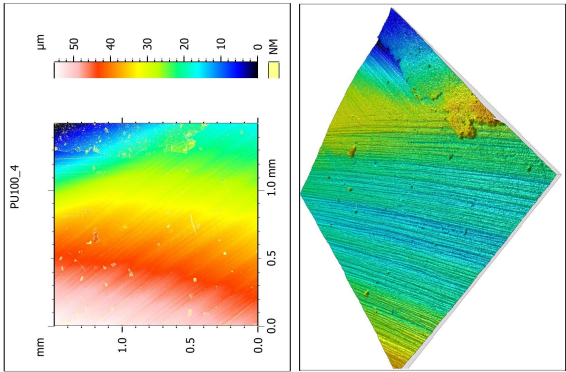
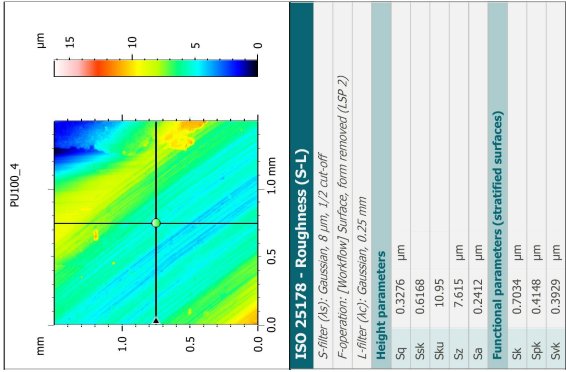


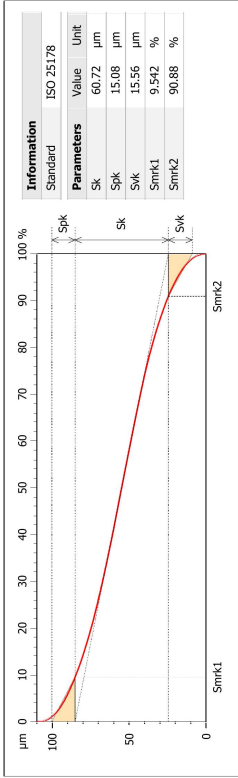
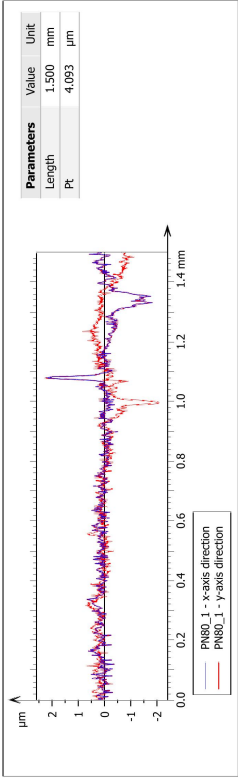
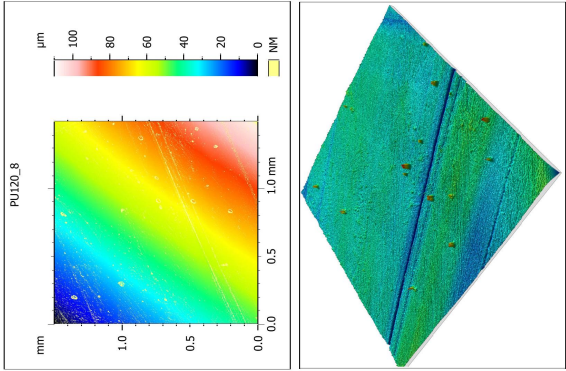
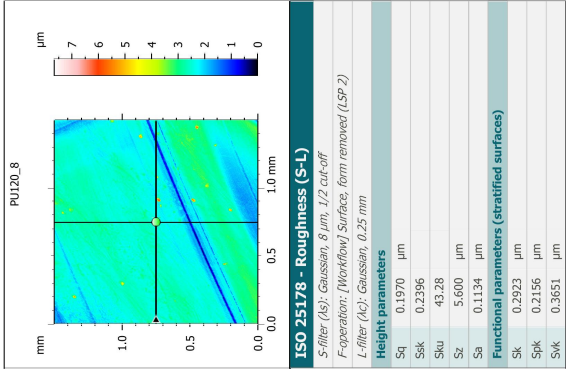
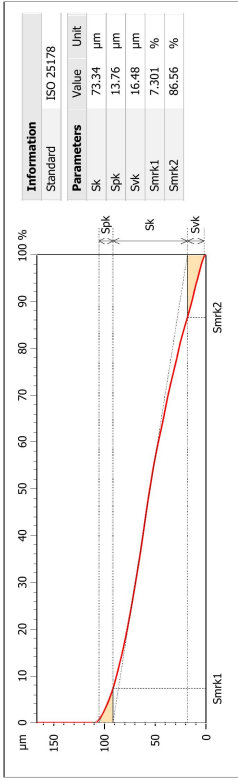
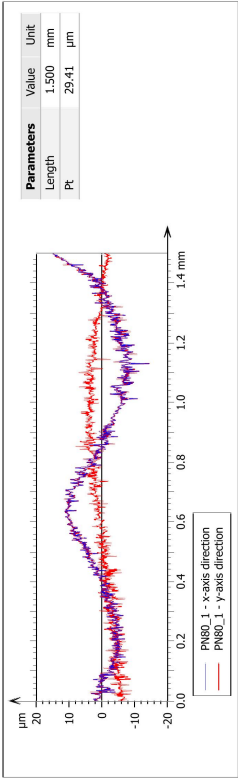
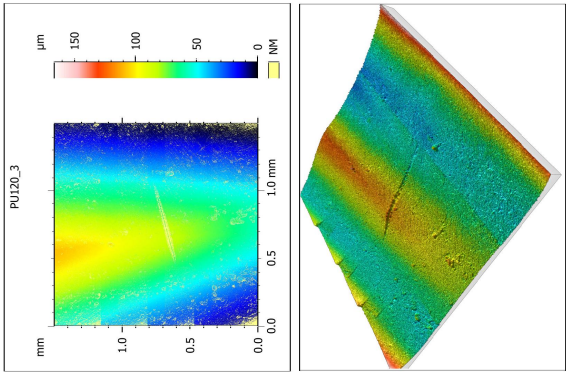
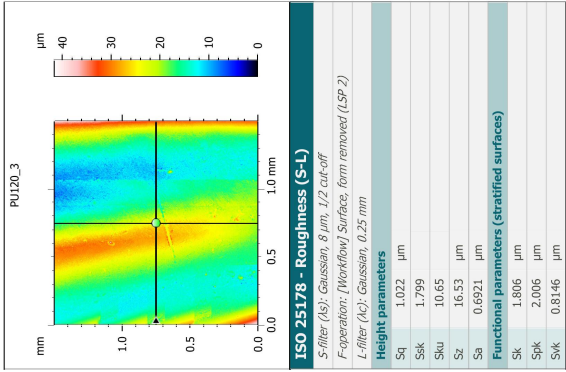


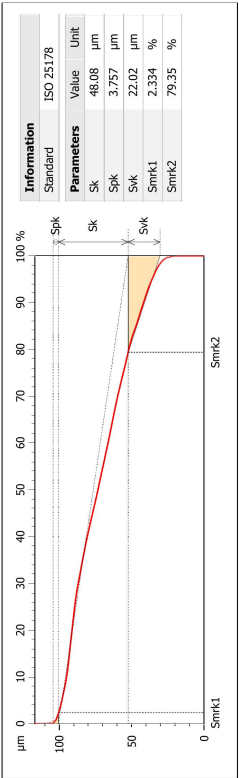
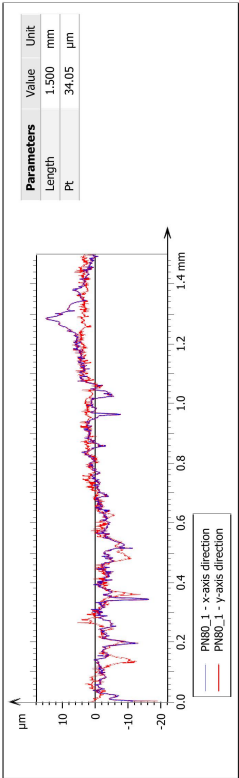
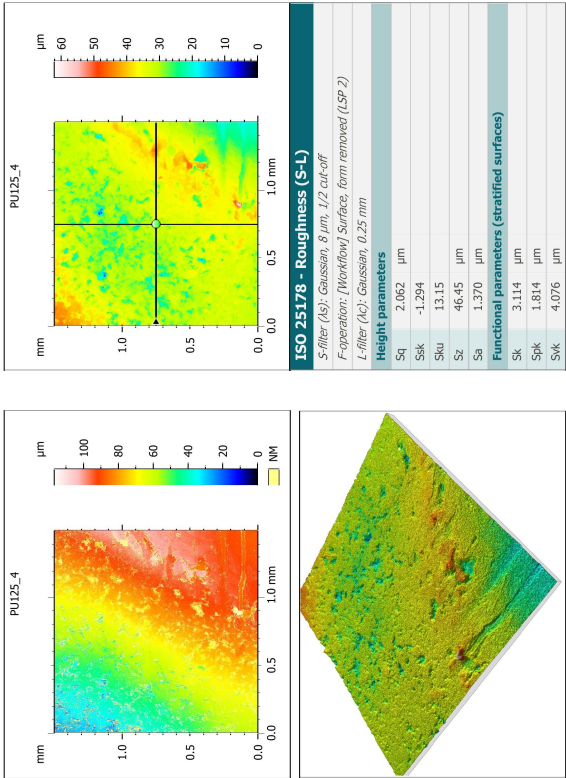
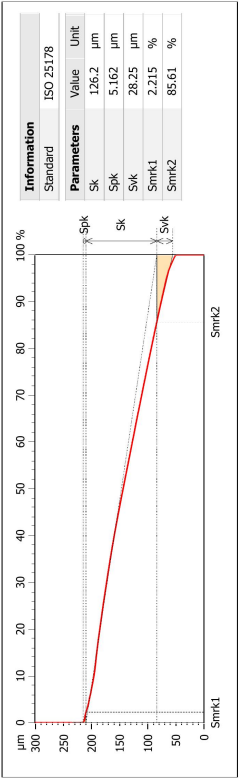
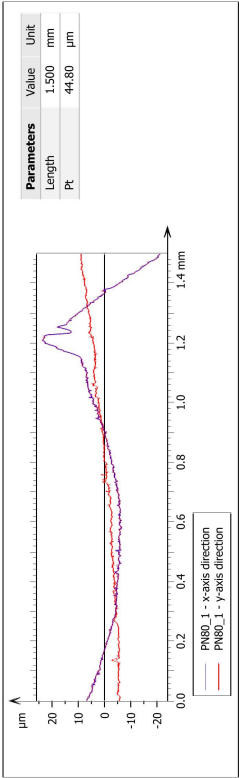
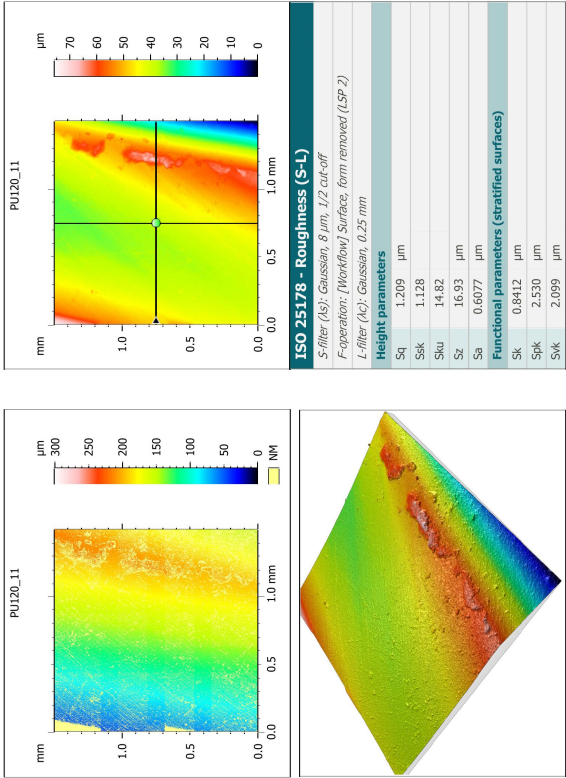
B.5 Roughness after testing

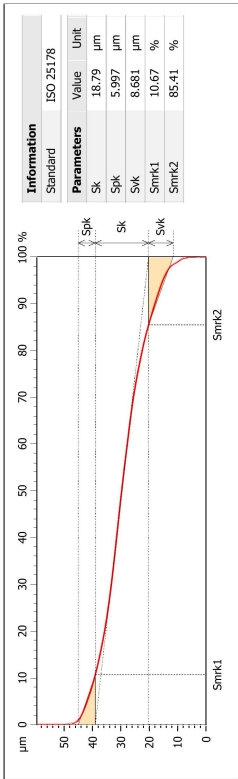
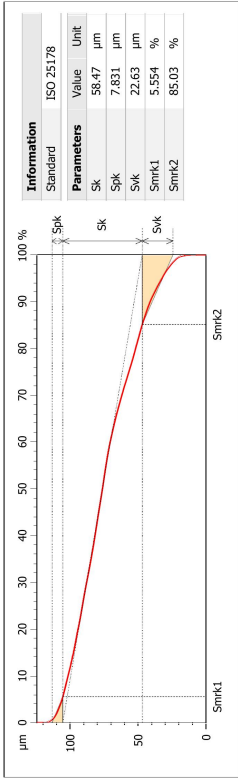
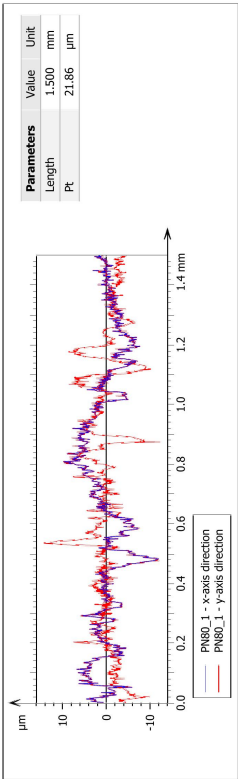
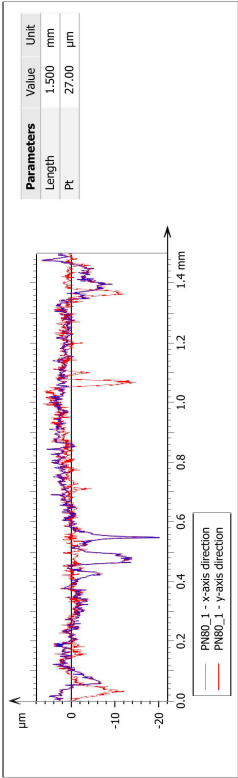
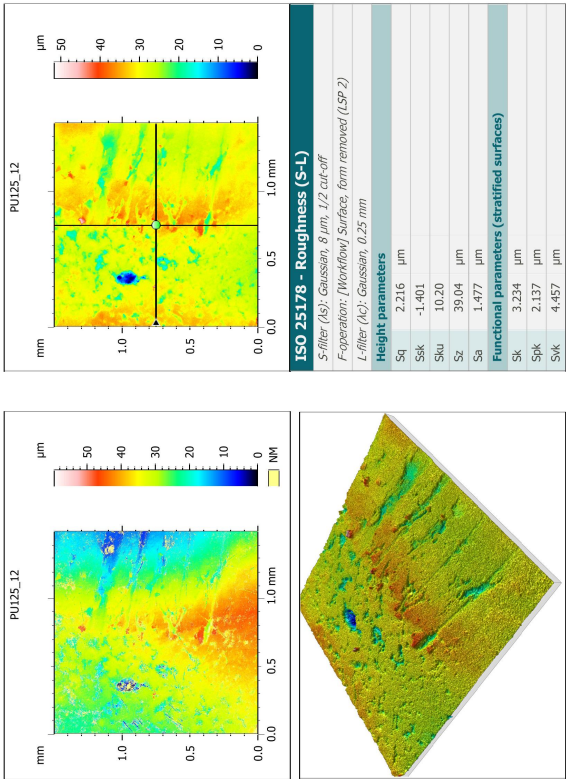
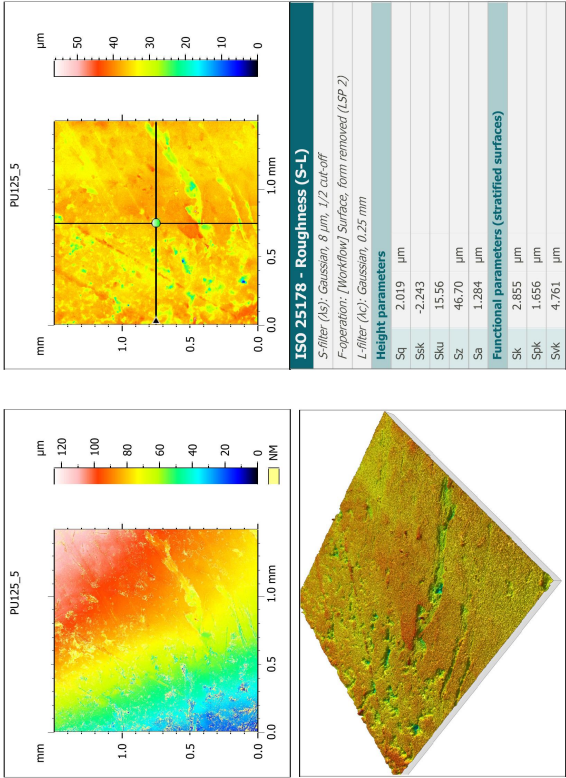




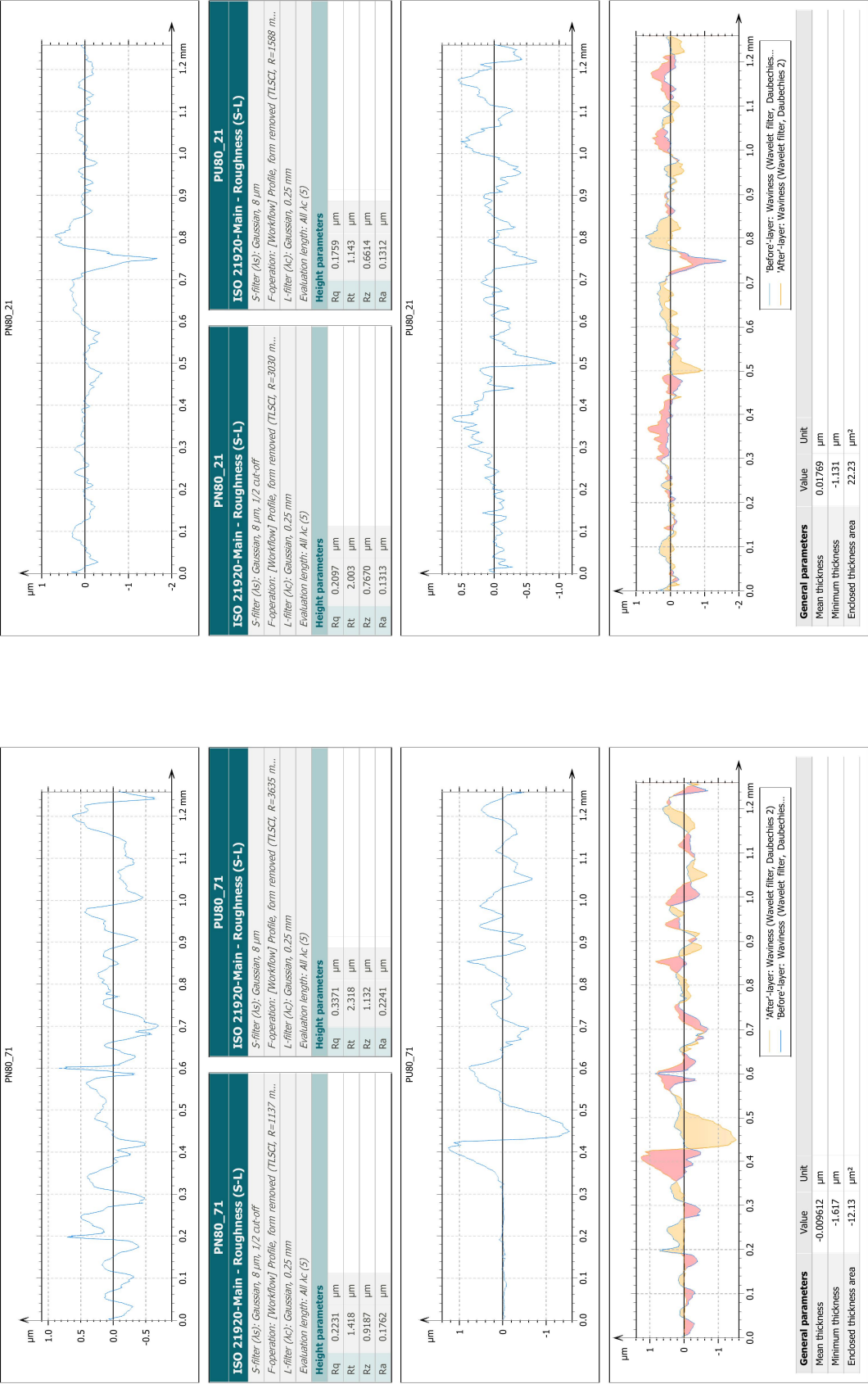


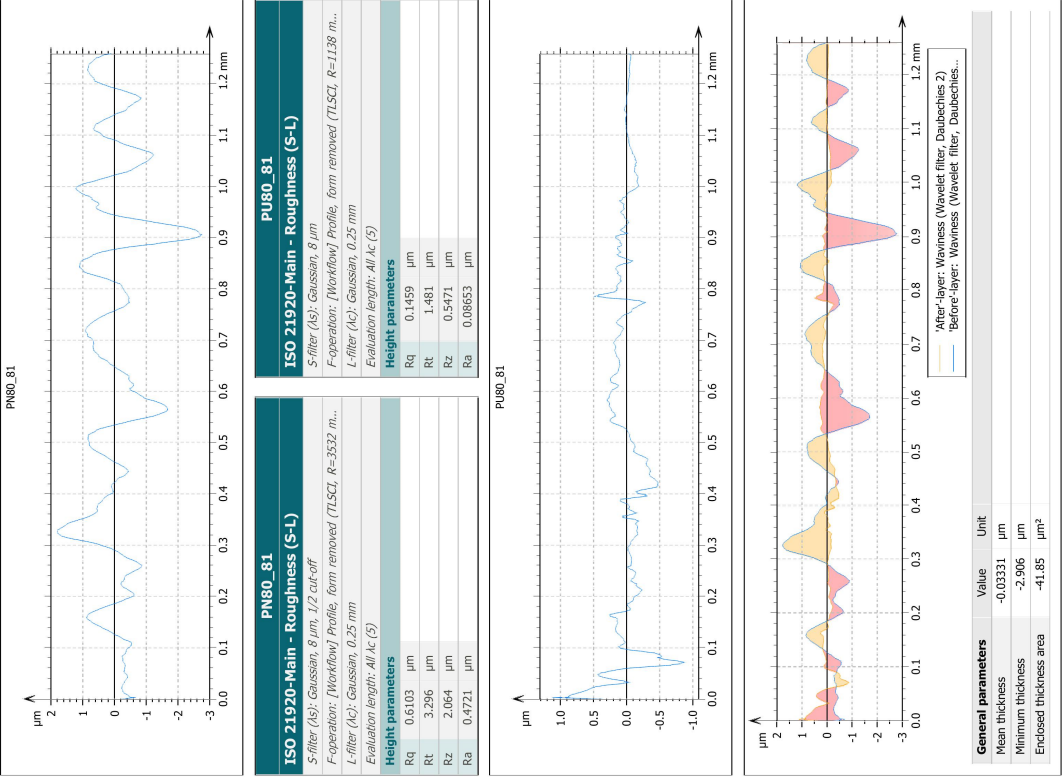


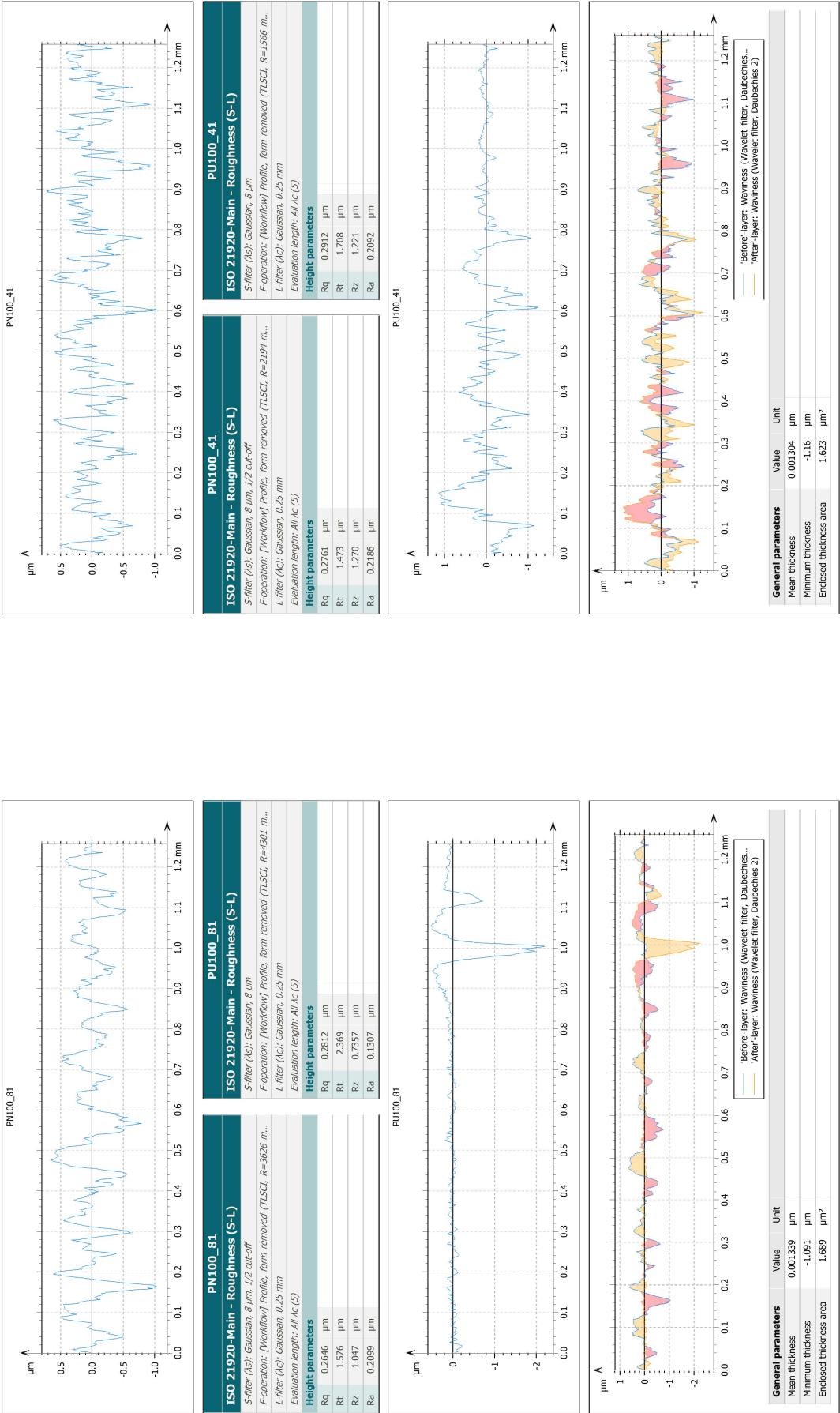


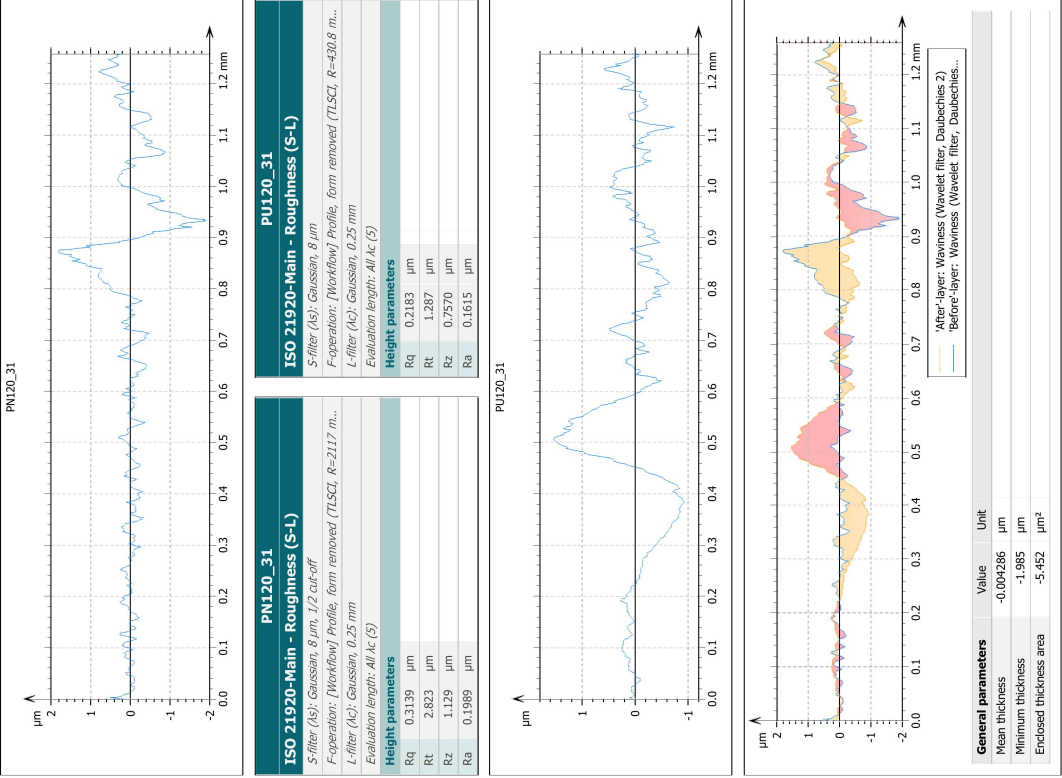


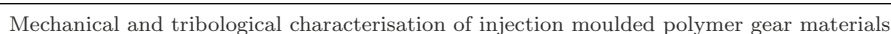
B.6 Roughness comparison between profiles

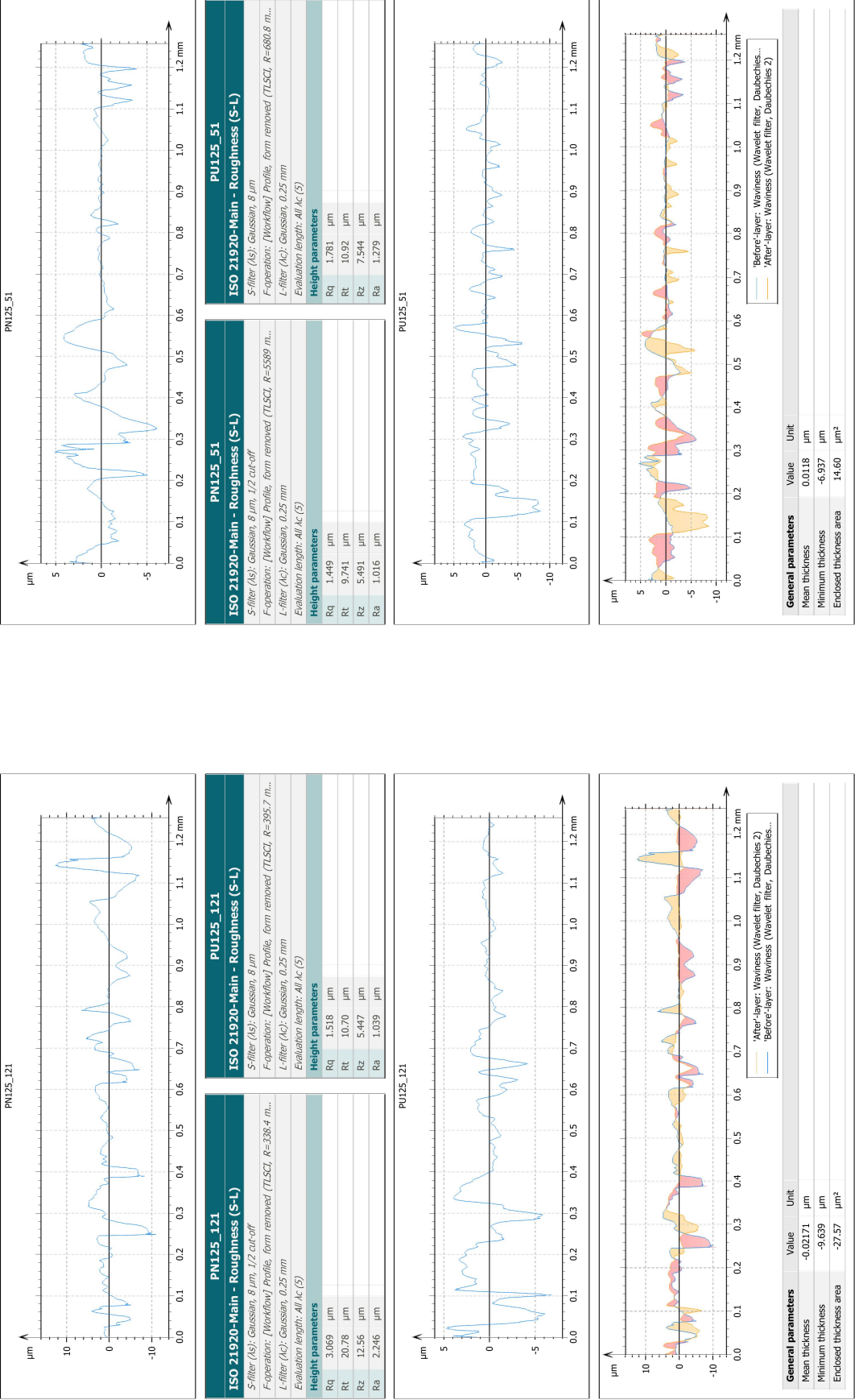




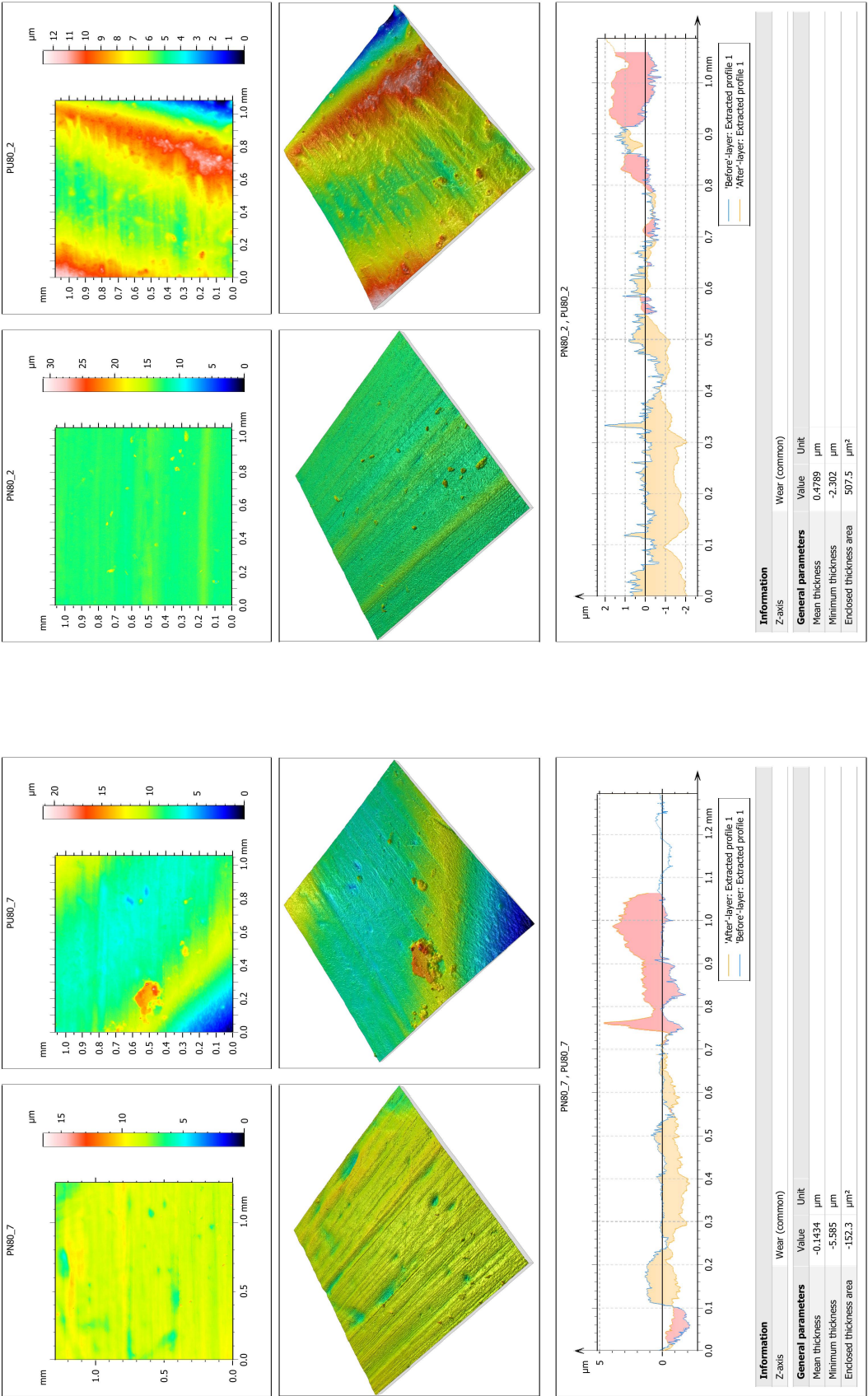


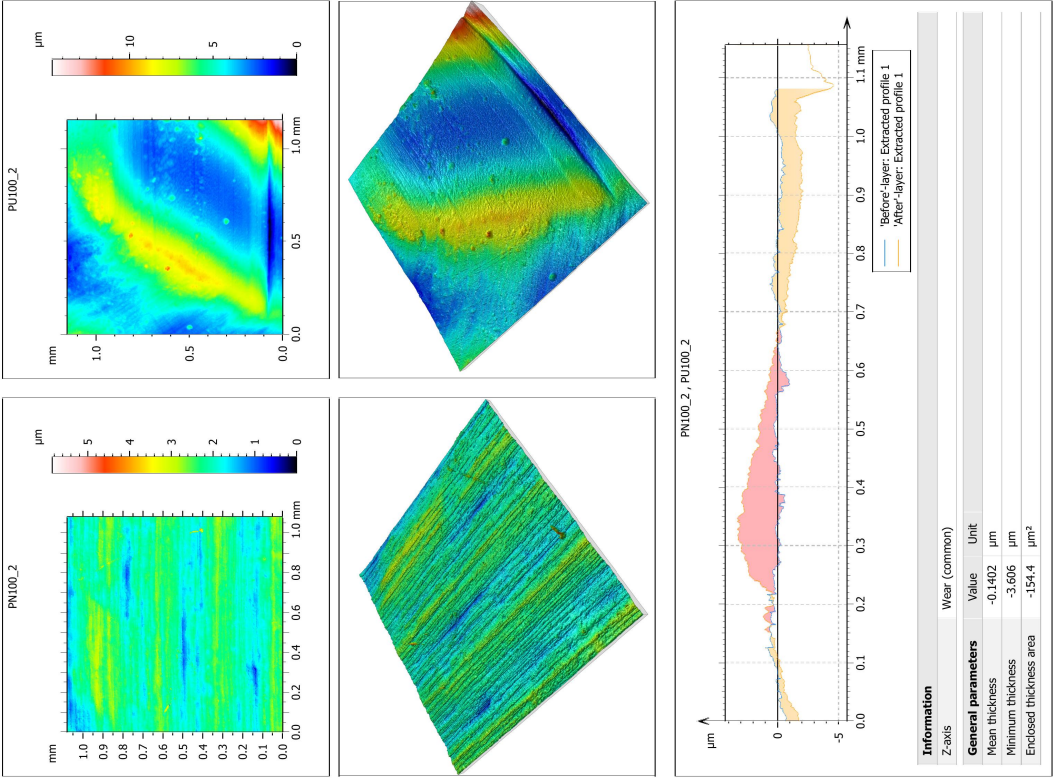
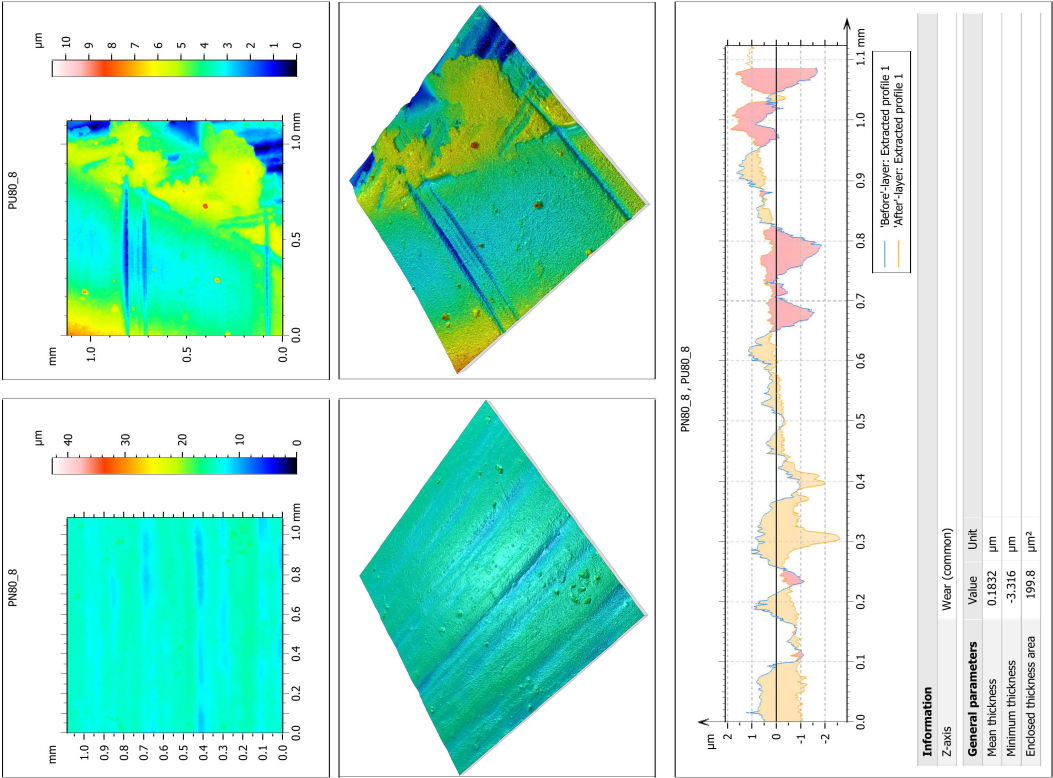


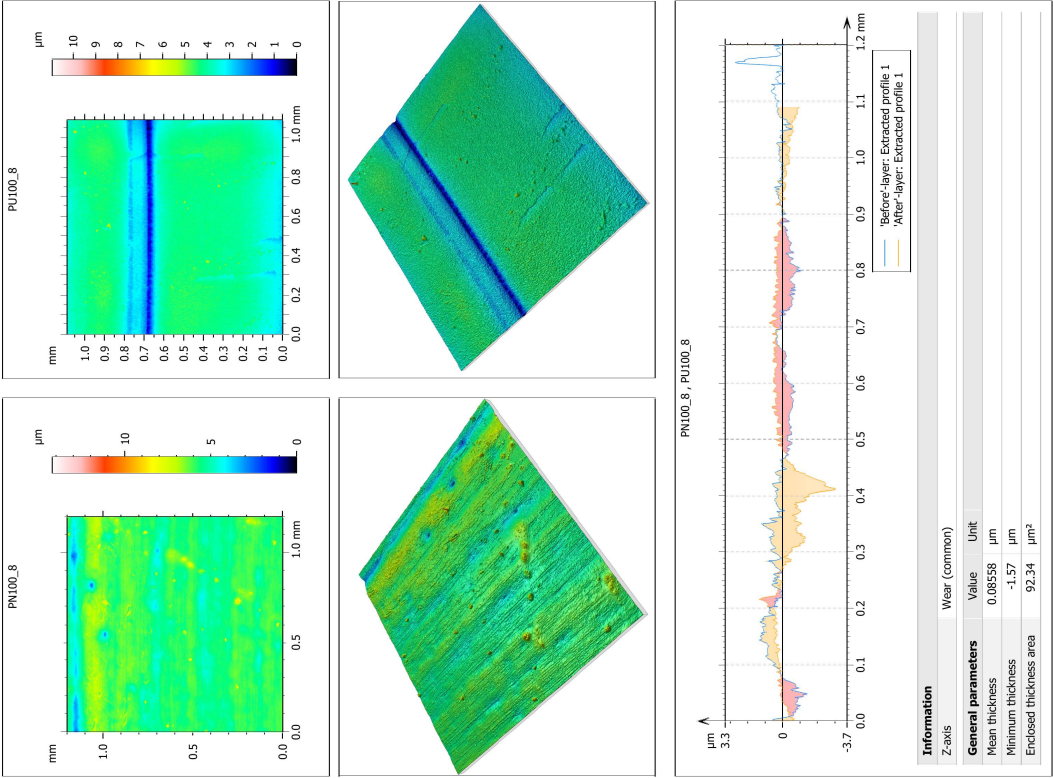
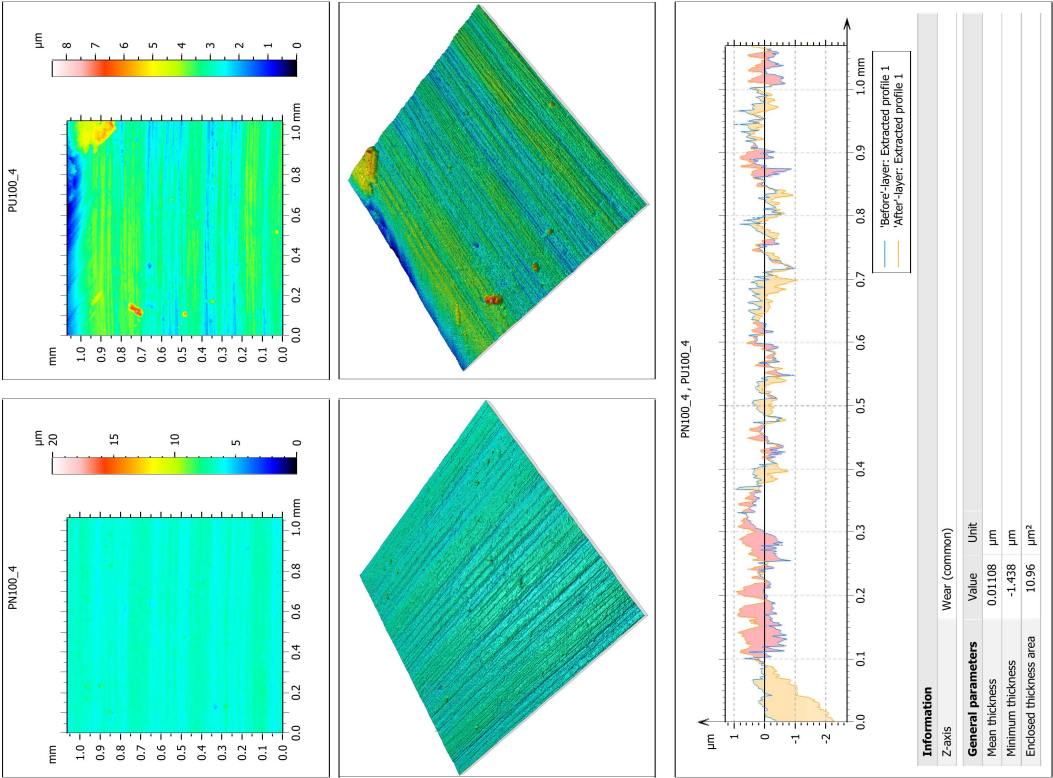


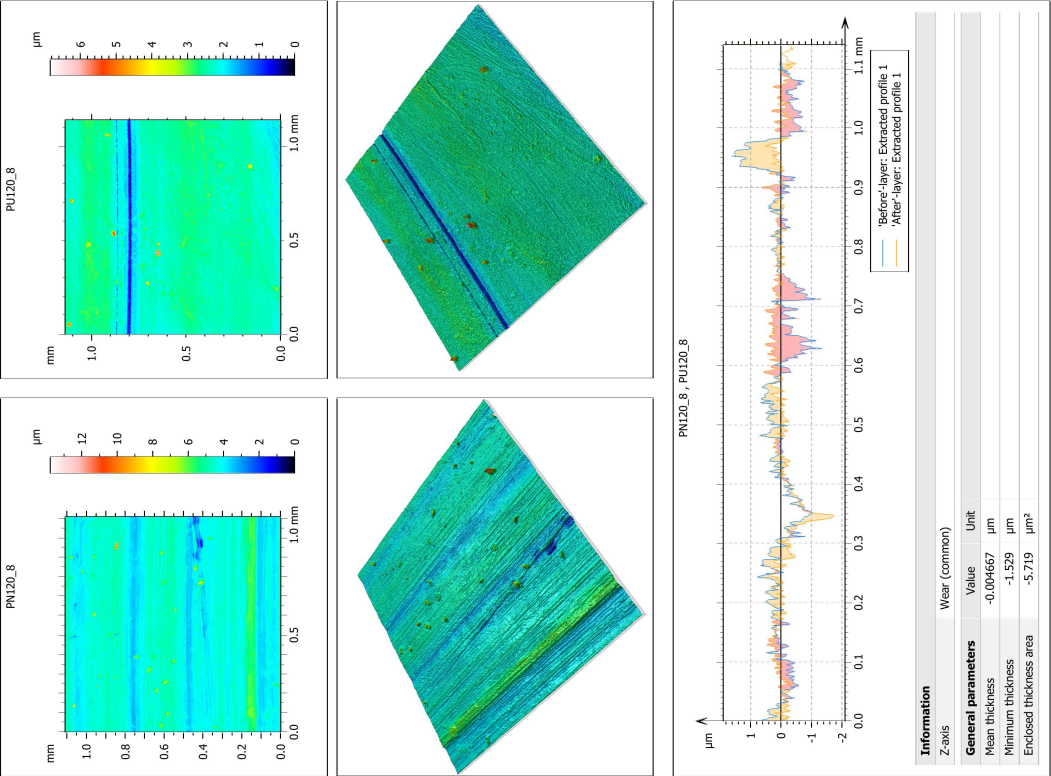
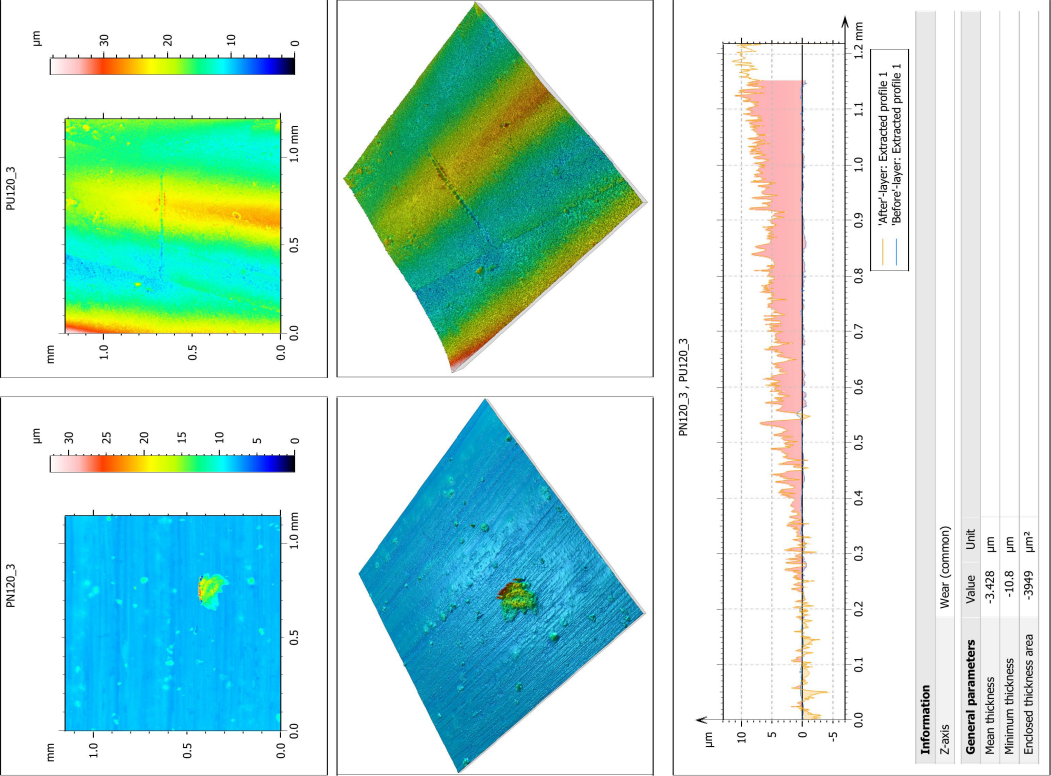


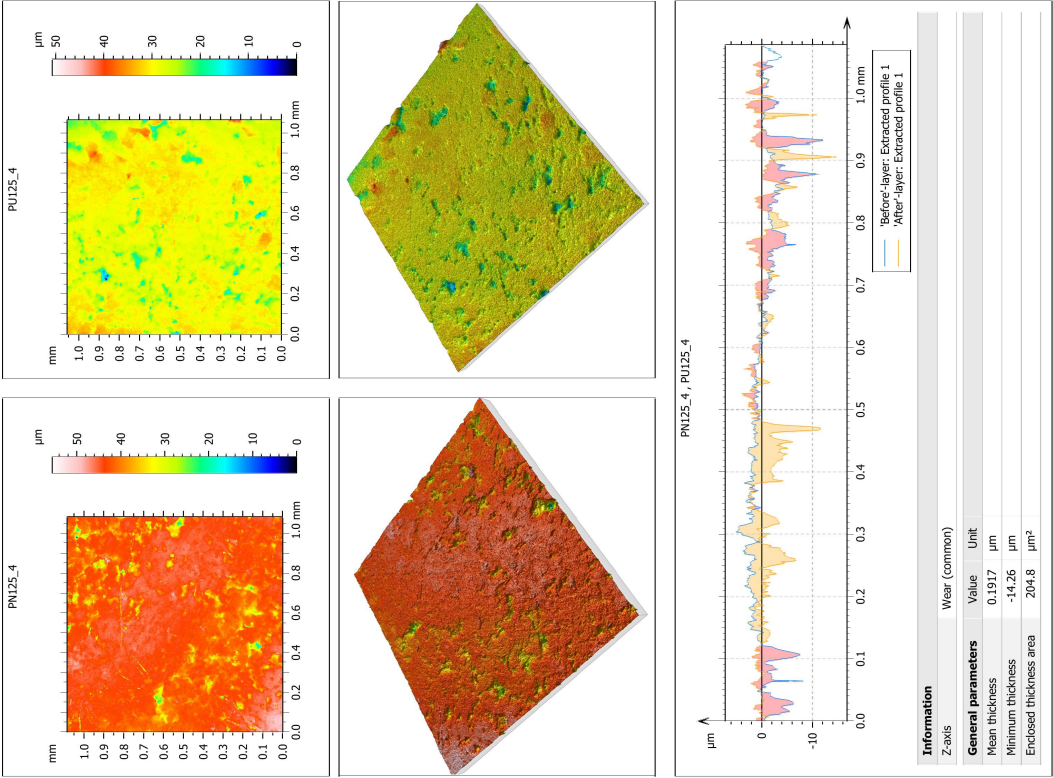
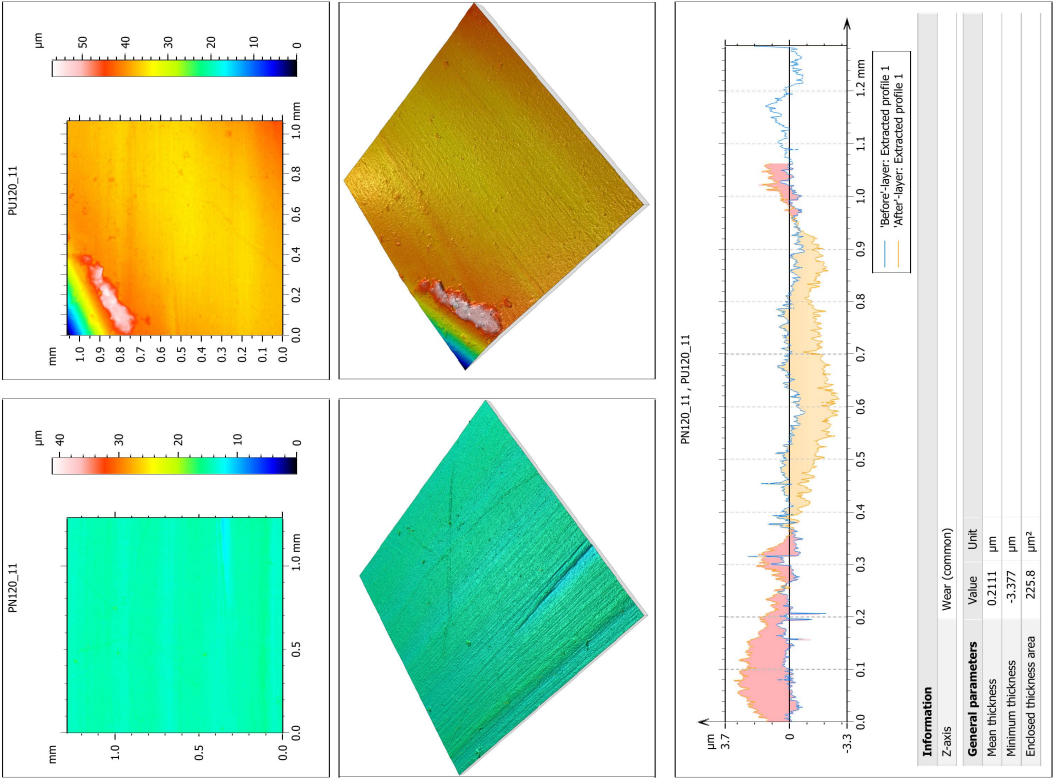
B.7 Wear

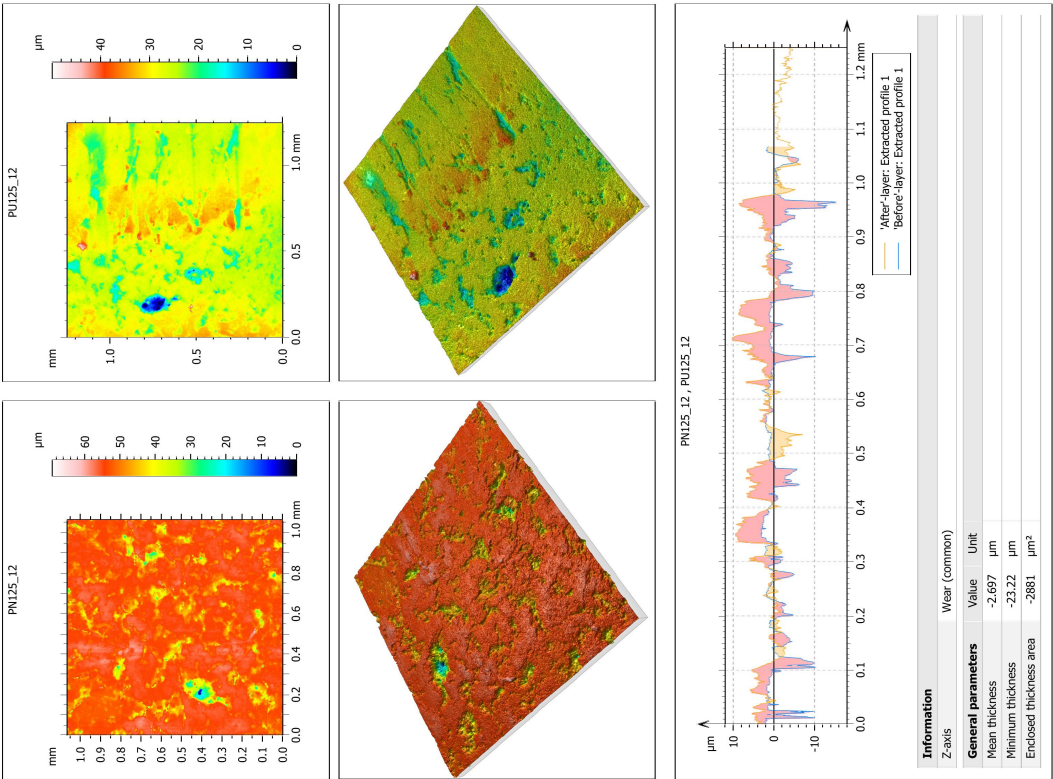
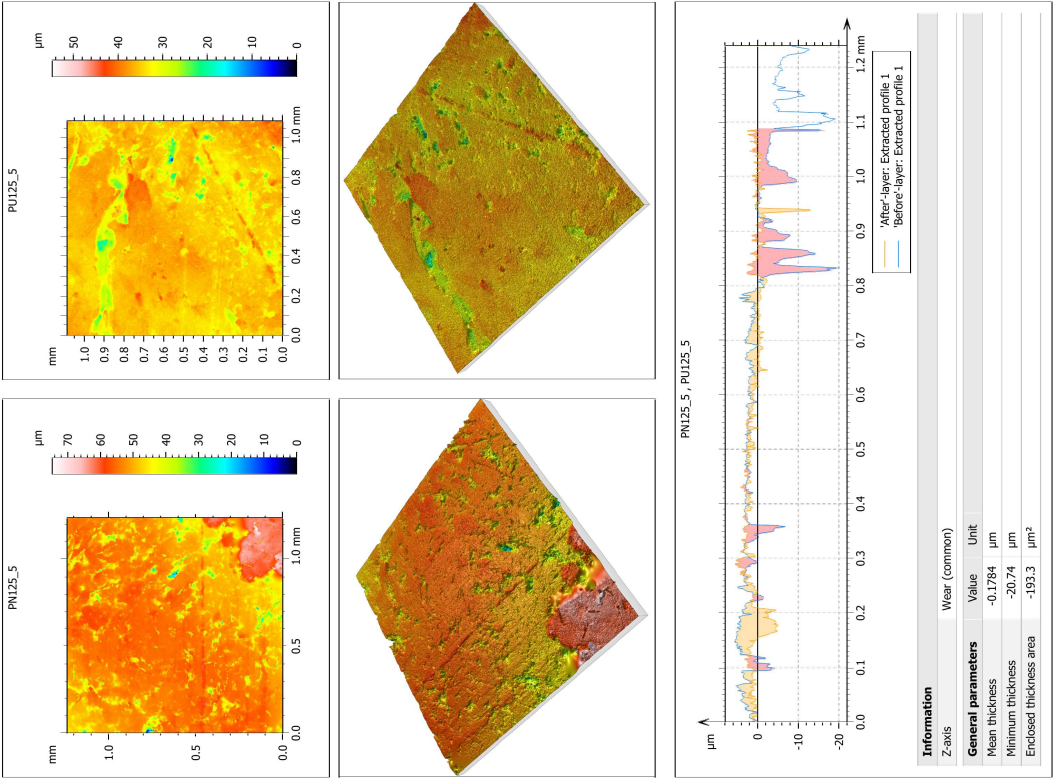












B.8 Mass Loss

			BEFORE					AFTER					
Mat.	spec. #	Nr.	Mass (g)			Average mass (g)	Nr.	Mass (g)			Average mass (g)	Mass Loss (g)	Mass Loss (mg)
POM	PS-AC25T80 roda 2	1	0.1308	0.1308	0.1307	0.1308	1						
		2	0.1180	0.1180	0.1179	0.1180	2	0.1177	0.1177	0.1177	0.1177	0.0003	0.2667
		3	0.1138	0.1138	0.1137	0.1138	3	0.1134	0.1135	0.1134	0.1134	0.0003	0.3333
		4	0.1280	0.1278	0.1279	0.1279	4	0.1277	0.1276	0.1274	0.1276	0.0003	0.3333
		5	0.1163	0.1165	0.1164	0.1164	5	0.1162	0.1161	0.1161	0.1161	0.0003	0.2667
		6	0.1233	0.1232	0.1232	0.1232	6	0.1225	0.1224	0.1224	0.1224	0.0008	0.8000
		7	0.1204	0.1205	0.1205	0.1205	7	0.1201	0.1201	0.1202	0.1201	0.0003	0.3333
		8	0.1000	0.1000	0.1001	0.1000	8	0.0998	0.0998	0.0998	0.0998	0.0002	0.2333
		9	0.1224	0.1224	0.1224	0.1224	9	0.1216	0.1216	0.1217	0.1216	0.0008	0.7667
		10	0.1163	0.1162	0.1163	0.1163	10	0.1154	0.1154	0.1155	0.1154	0.0008	0.8333
	PS-AC25T100 roda 8	1	0.1358	0.1357	0.1357	0.1357	1						
		2	0.1338	0.1336	0.1336	0.1337	2	0.1335	0.1335	0.1334	0.1335	0.0002	0.2000
		3					3						
		4	0.1244	0.1242	0.1241	0.1242	4	0.1241	0.1239	0.1240	0.1240	0.0002	0.2333
		5	0.1249	0.1246	0.1246	0.1247	5	0.1239	0.1238	0.1237	0.1238	0.0009	0.9000
		6	0.1275	0.1272	0.1271	0.1273	6	0.1265	0.1264	0.1266	0.1265	0.0008	0.7667
		8	0.1250	0.1248	0.1248	0.1249	8	0.1238	0.1240	0.1239	0.1239	0.0010	0.9667
		10	0.1150	0.1147	0.1148	0.1148	10						
		11	0.1122	0.1121	0.1120	0.1121	11						
		12	0.1201	0.1199	0.1199	0.1200	12						
		10_2	0.1088	0.1089	0.1087	0.1088	10_2	0.1083	0.1082	0.1082	0.1082	0.0006	0.5667
		12_	0.1144	0.1145	0.1145	0.1145	12_2	0.11389b	0.1138	0.1139	0.1139	0.0006	0.6167
		13	0.1120	0.1121	0.1120	0.1120	13	0.1115	0.1115	0.1115	0.1115	0.0005	0.5333
		14	0.1082	0.1082	0.1082	0.1082	14	0.1080	0.1080	0.1080	0.1080	0.0002	0.2000
	PS-AC25T120 roda 1	1	0.1280	0.1279	0.1280	0.1280	1	0.1278	0.1280	0.1279	0.1279	0.0001	0.0667
		2	0.1302	0.1301	0.1300	0.1301	2	0.1300	0.1300	0.1300	0.1300	0.0001	0.1000
		3	0.1308	0.1306	0.1305	0.1306	3	0.1305	0.1306	0.1305	0.1305	0.0001	0.1000
		4	0.1066	0.1066	0.1065	0.1066	4						
		5	0.1341	0.1342	0.1341	0.1341	5	0.1340	0.1340	0.1340	0.1340	0.0001	0.1333
		6	0.1091	0.1091	0.1091	0.1091	6	0.1090	0.1089	0.1090	0.1090	0.0001	0.1333
		7	0.1101	0.1100	0.1099	0.1100	7	0.1098	0.1099	0.1098	0.1098	0.0002	0.1667
		8	0.1146	0.1144	0.1145	0.1145	8	0.1144	0.1145	0.1144	0.1144	0.0001	0.0667
		9	0.1141	0.1139	0.1139	0.1140	9	0.1138	0.1140	0.1139	0.1139	0.0001	0.0667
11		0.1188	0.1189	0.1189	0.1189	11	0.1187	0.1188	0.1188	0.1188	0.0001	0.1000	
PA6-GF		NS-AC25T125 roda 5	1	0.1426	0.1425	0.1426	0.1426	1	0.1424	0.1425	0.1424	0.1424	0.0001
	2		0.1441	0.1441	0.1441	0.1441	2	0.1441	0.1440	0.1440	0.1440	0.0001	0.0667
	3		0.1388	0.1387	0.1387	0.1387	3						
	4		0.1481	0.1481	0.1482	0.1481	4	0.1479	0.1479	0.1479	0.1479	0.0002	0.2333
	5		0.1424	0.1423	0.1422	0.1423	5	0.1421	0.1421	0.1421	0.1421	0.0002	0.2000
	6		0.1520	0.1521	0.1521	0.1521	6	0.1519	0.1520	0.1519	0.1519	0.0001	0.1333
	7		0.1466	0.1467	0.1466	0.1466	7						
	8		0.1472	0.1472	0.1471	0.1472	8	0.1468b	0.1468	0.1469	0.1469	0.0003	0.3167
	9		0.1249	0.1248	0.1248	0.1248	9	0.1245	0.1246	0.1246	0.1246	0.0003	0.2833
	10		0.1422	0.1422	0.1422	0.1422	10	0.1420	0.1420	0.1420	0.1420	0.0002	0.2000
	11		0.1424	0.1424	0.1423	0.1424	11						
	CONTROL		0.1122	0.1122	0.1122	0.1122	CONTROL	0.1122	0.1121	0.1122	0.1122	0.0000	
	12		0.1384	0.1385	0.1385	0.1385	12	0.1384	0.1384	0.1384	0.1384	0.0001	0.0667

Figure 109 – Mass measurements.

