

UNIVERSIDADE DO PORTO

FACULDADE DE ENGENHARIA

Master's Dissertation

Development of Tools and Manufacturing Processes for Hybrid Polymer Gears

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Dissertation submitted to Universidade do Porto – Faculdade de Engenharia
in partial fulfillment of the requirements for the degree of
Master in Mechanical Engineering - Machine Design

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July, 2023

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Rua Dr. Roberto Frias, Porto

July, 2023

"In the middle of every difficulty lies opportunity."
Albert Einstein

Acknowledgements

I would like to thank Celoplás, Plásticos para a Indústria, S.A, for allowing the completion of this project at their facilities, providing all the necessary conditions, and also for the internship grant awarded.

I would also like to thank my supervisor at the company, Mr. Engineer João Cortez, for his dedication and valuable recommendations provided.

I would like to thank all the members of the mould manufacturing department: Eng. César Soares, and Miguel Ferreira, for their warm welcome and assistance. In particular, I would like to express my gratitude to Eng. Bruno Correia for his continuous support, patience, and knowledge shared with me.

To my supervisor, Carlos Fernandes, my deepest gratitude for his invaluable guidance, unwavering support and expert knowledge throughout the project. His expertise, enthusiasm, and encouragement have been a constant source of inspiration for me.

Finally, I would like to express my heartfelt appreciation to my family and dear friends for their support, encouragement, and understanding during this journey. A special mention to my parents, sister, grandparents and my girlfriend for the unconditional love.

Abstract

This dissertation was prepared at the company Celoplás as the conclusion of the Master's Degree in Mechanical Engineering of the Faculty of Engineering of University of Porto.

Polymer gears have drawn the industry's attention more and more over the years. The main reasons are related to the cheaper mass production and lower density than metallic gears, as well as other benefits including the dry running capability. However, they have a major limitation related to their very low thermal conductivity and the fact that the operating temperatures have a significant impact on their mechanical properties. As a result, a concept to enhance the thermal behaviour of polymer gears, specifically, hybrid polymer gears will be developed in this dissertation.

This thesis aimed to develop skills in the design and manufacture of hybrid polymer gears, which are polymer gears with an impregnated insert in their matrix. The insert geometry was developed from the beginning taking into account previous studies already made on this topic. To produce these hybrid polymer gears by injection moulding, it was necessary to restructure a mould in order to meet all the requirements for achieving the desired geometry of these gears. Moreover, a mould-filling simulation was developed to predict the behaviour of the molten material as it fills the mould cavity during the injection moulding process.

Once the mould was assembled, different tests were made, and prototypes were obtained by injection moulding. It was also essential to perform the dimensional control to compare the manufactured inserts and hybrid polymer gears against the design specifications.

In the end, the load-carrying capacity of cylindrical POM gears was calculated to determine the number of load cycles according to VDI 2736.

Keywords: Polymer Gears, Hybrid Polymer Gears, Injection Moulding, Mould, Dimensional Control, Load-Carrying Capacity

Resumo

A presente dissertação foi elaborada na empresa Celoplás, concluindo o Mestrado em Engenharia Mecânica na Faculdade de Engenharia da Universidade do Porto.

As engrenagens poliméricas têm chamado cada vez mais a atenção da indústria ao longo dos anos. As principais razões estão relacionadas com a produção em massa mais económica e densidade inferior comparativamente às engrenagens metálicas, ou ainda a capacidade de funcionamento sem lubrificação, entre outras vantagens. Contudo, têm uma grande limitação relacionada com a sua muito baixa condutividade térmica e com o facto de as temperaturas de funcionamento terem um impacto significativo nas suas propriedades mecânicas. Como resultado, será desenvolvido nesta dissertação um conceito para melhorar o comportamento térmico das engrenagens poliméricas, especificamente, as engrenagens híbridas de polímeros.

Esta dissertação teve como objetivo desenvolver competências no design e fabricação de engrenagens poliméricas híbridas, que são engrenagens poliméricas com um inserto impregnado na sua matriz. A geometria do inserto foi desenvolvida de raiz tendo em conta estudos prévios já realizados sobre este tema. Para produzir estas engrenagens poliméricas híbridas por moldação por injeção, foi necessário reestruturar um molde de modo a cumprir todos os requisitos para alcançar a geometria desejada destas engrenagens. Além disso, foi desenvolvida uma simulação de enchimento do molde para prever o comportamento do material fundido à medida que preenche a cavidade do molde durante o processo de moldação por injeção.

Após o molde ser montado, foram realizados diferentes testes e obtidos inúmeros protótipos por moldação por injeção. Mais ainda, foi essencial realizar o controlo dimensional para comparar os insertos e as engrenagens poliméricas híbridas fabricados em relação às especificações de design.

No final, a capacidade de carga das engrenagens cilíndricas de POM foi calculada para determinar o número de ciclos de carga de acordo com a diretiva VDI 2736.

Palavras-Chave: Engrenagens Poliméricas, Engrenagens Híbridas Poliméricas, Moldação por Injeção, Molde, Controlo Dimensional, Capacidade de Carga

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List of Acronyms

CMMs	Coordinate Measuring Machines
CNC	Computer Numerical Control
CVD	Chemical Vapor deposition
DIC	Digital Image Correlation
DOE	Design of Experiments
EDM	Electrical Discharge Mahining
FEM	Finite Element Method
HT	Hybrid tooth rack
MoS2	Molybdenum Disulphide
PA	Polyamide
PA-GF	Polyamide-Glass Fiber
PAI	Polyamide-imide
PAO	Polyalphaolefin
PBT	Polybutylene Terephthalate
PC	Polycarbonate
PE	Polyethylene
PEEK	Polyetheretherketone
PET	Polyethylene Terephthalate
POM	Polyoxymethylene
PP	Polypropylene
PPA	Polyphthalamide
PPS	Polyphenylene Sulphide

PTFE	Polytetrafluoroethylene
PUR	Polyurethane
PVC	Polyvinyl Chloride
PVD	Physical Vapor Deposition
SLM	Selective Laser Melting
STD	Standard rack tooth

List of Symbols

Symbol	Description	Units
A_G	Mechanism housing area of the heat-dissipating surface	m^2
A_f	Contact area of the mould clamping system	m^2
A_l	Lateral area of the mould	m^2
b_w	Common face width	mm
b	Face width	mm
c	Specific heat capacity	$\text{J kg}^{-1} \text{K}^{-1}$
d_{Na}	Usable addendum circle diameter	mm
d_{Nf}	Usable root circle diameter	mm
d_a	Addendum circle diameter	mm
d_b	Base circle diameter	mm
d_f	Root circle diameter	mm
d_i	Internal diameter	mm
d	Reference circle diameter	mm
ED	Tooth engagement time	
E	Elastic Young's Modulus	GPa
e_c	Distance of the chanel to the moulding zone	m
e	Lateral gap of the insert from the gear sides	mm
$F_{n,local}$	Normal force on the tooth at the contact	N
F_n	Normal force	N
F_t	Tangential force	N
F	Suction force	N
H_V	Degree of tooth loss	
h_{aP}	Addendum of the reference profile	mm
h_{fP}	Dedendum of the reference profile	mm
h	Convection heat transfer coefficient	$\text{W m}^{-2} \text{°C}^{-1}$
K_A	Application factor	
$K_{F\alpha}$	Face factor	
$K_{F\beta}$	Width factor	
K_F	Factor for tooth root load	
K_H	Factor for tooth flank loading	

Symbol	Description	Units
K_e	Dynamic factor	
K	Coefficient dependent on the part thickness	
$k_\varnothing$	Heat transfer coefficient of the plastic gear	$\text{K}(\text{m/s})^{0.75} \text{mm}^{1.75} / \text{W}$
k_w	Wear coefficient	$\text{mm}^3 \text{N}^{-1} \text{m}^{-1}$
k	Number of teeth spanned by a gear	
L_t	Transverse Length	mm
L	Minimum length of the cooling channels	m
l_{FI}	Profile line length of the tooth flank	mm
m_n	Normal module	mm
m_p	Mass of the polymer	kg
m	Flow rate	L min^{-1}
N_L	Number of load cycles	
n	Rotational speed	min^{-1}
P_t	Total profile height	μm
P	Power	W
Q_P	Heat generated by the polymer	W
Q_{TM}	Heat transferred by the cooling medium	W
Q_{amb}	Heat released to the environment	W
Q_{cond}	Heat transfer by conduction	W
Q_{conv}	Heat transfer by convection	W
Q_{rad}	Heat transfer by radiation	W
R_a	Average roughness	μm
$R_{\lambda,G}$	Heat transfer resistane of the mechanism housing	$\text{K m}^2 \text{W}^{-1}$
R_q	Root-mean-square profile height	μm
R_z	Mean peak-to-valley height	μm
S_{Fmin}	Minimum safety factor for tooth root	
S_{Hmin}	Minimum safety factor for tooth flank	
s	Part thickness	m
T_a	Ambient temperature	$^{\circ}\text{C}$
T_d	Torque	Nm
T_{ext}	Ejection temperature	$^{\circ}\text{C}$
T_{Heater}	Temperature of the cartridge heaters	$^{\circ}\text{C}$
T_{inj}	Melting temperature	$^{\circ}\text{C}$
T_m	Mould temperature	$^{\circ}\text{C}$
t_p	Residence time of the material in the mould	s
t	Gap distance from tooth-tip	mm
u	Gear ratio	
v_t	Tangential velocity	m s^{-1}
W_k	Base tangent length	mm

Symbol	Description	Units
W_{local}	Local linear wear	mm
W_m	Averaged linear wear	mm
W_t	Total waviness height	μm
W_{zul}	Permissible wear	mm
w	Insert's plate width	mm
x	Profile shift coefficient	
Y_{Fa}	Form factor	
Y_{Sa}	Stress correction factor	
Y_β	Helix angle factor	
Y_ϵ	Contact ratio factor	
Y_{St}	Stress correction factor	
Y	Dimensionless temperature	
Z_E	Elasticity factor	$\text{MPa}^{0.5}$
Z_H	Zone factor	
Z_β	Spiral angle factor	
Z_ϵ	Contact ratio factor	
Z_r	Surface roughness factor	
z	Number of teeth	
α	Thermal expansion coefficient	$^{\circ}\text{C}^{-1}$
ϵ_β	Overlap contact ratio	
ϑ_0	Ambient Temperature	$^{\circ}\text{C}$
ζ	Local specific sliding	
$\vartheta_{Fu\beta}$	Root temperature	$^{\circ}\text{C}$
ϑ_{Fla}	Flank temperature	$^{\circ}\text{C}$
ϑ_{zul}	Permissible peak temperature	$^{\circ}\text{C}$
κ	Thermal conductivity	$\text{W m}^{-1} \text{K}^{-1}$
Λ	Logarithmic decrement	
λ	Tooth tip's deformation	mm
λ_c	Cut-off length	mm
μ	Coefficient of friction	
α_{ef}	Average effective thermal diffusivity	$\text{m}^2 \text{s}^{-1}$
ν	Poisson's ratio	
ρ	Density	kg m^{-3}
ρ_{fP}	Fillet of the reference profile	mm
σ	Stefan-Boltzmann constant	$\text{W m}^{-2} \text{K}^{-4}$
σ_F	Root stress	MPa
σ_{FG}	Maximum root strength	MPa
σ_{FlimN}	Fatigue strength under pulsating stress	MPa
σ_{FP}	Permissible root stress	MPa

Symbol	Description	Units
σ_H	Flank pressure (Hertz stress)	MPa
σ_{HlimN}	Rolling contact fatigue strength	MPa
α_n	Pressure angle	°
σ_{HP}	Permissible flank pressure	MPa
β	Helix angle	°
β_b	Helix angle at the base circle	°
β_k	Proportionally factor	$\text{W m}^{-2} \text{°C}^{-1}$
ε	Emissivity factor	
$\varepsilon_{1,2}$	Partial contact ratio pinion and wheel	
ε_α	Radial contact ratio	

Chapter 1

Introduction

Due to their numerous benefits over metal gears, polymer materials have been used for various gear applications. The main advantages of polymer gears include their light weight, resistance to corrosion, ability to run dry, superior damping resistance, and affordability. But compared to metal gears, polymer gears have some drawbacks, such as lower mechanical properties, poor heat conductivity, and low thermal resistance. In reality, since the mechanical characteristics of polymer materials are strongly influenced by temperature, these gears become much more prone to failure as working temperatures rise.

Given its potential, studies surrounding them now concentrate on reducing the effects of the downsides listed before. With ultimate improvements in wear, friction, and thermal behaviour, different approaches were tested. These techniques included changing the tooth geometry, drilling cooling holes, and developing a hybrid polymer gear idea.

The hybrid polymer gear concept enhances heat conduction by employing metallic inserts impregnated in the polymer matrix or any other material with sufficient mechanical qualities and good thermal conductivity. Recent studies were undertaken on this topic [1, 2, 3], and experimental validation was obtained [2, 3].

This hybrid gear showed in these studies improvement by managing to lower bulk temperature of the gear. The metallic inserts were then effective on helping the polymer gear

evacuate heat thanks to a higher thermal conductivity of metals. As metals have higher mechanical resistance and better mechanical properties compared to the polymer gears, it was also verified an increase of load carrying capacity of the hybrid gears.

In the present work, the main purpose is to develop a new design for the hybrid polymer gear and project a mould where these hybrid polymer gears will be manufactured. Additionally, different polymeric materials for gear applications were investigated. Moreover, the dimensional control of both the inserts and the hybrid polymer gears will be performed. At the end, the load-carrying capacity of a polymer gear pair is analysed.

1.1 Company Presentation

Celoplás is one of the four firms in the CELOPLÁS Group, which also comprises CCL - Plastics for Industry, CELOPRINT - Printing and CENTISUPPORT - Machines and Equipment.

Celoplás was founded in 1989, and today has a total area of 41 000 m². The current team has more than 200 employees, working 24 hours a day, 7 days a week, on a production hall with more than 70 injection moulding machines.

The company is a national reference and is internationally recognised for fabricating more than 200 million of technical and precision components per year, by injection moulding and micro injection moulding, using more than 200 different types of thermoplastics, thermosettings and liquid silicon rubber materials, for automotive, electronics, electrical, optics, construction, medical and other industries.

Along with this activity, Celoplás develops and manufactures precision moulds and micro moulds, for own production, using the most recent machining and micromachining technologies.

Celoplás invests in ongoing training of its human resources, products' quality, processes and products innovation, enhancing competitiveness and being recognized as a European reference on the development and production of components in high technological clusters, using polymeric materials.

Celoplás has been in business for over 40 years, and during that time, it has developed a loyal customer base and established itself as a leader in its market segment.



Figure 1.1: Celoplás, Plásticos para a Indústria S.A. [4].

The present needs of the company have led to a focus on the research of the design and behaviour of the hybrid polymer gears. In addition, the production processes are also studied, with particular emphasis on injection.

1.2 Thesis Outline

Chapter 2, entitled **Literature Review on Polymer Gears**, tackles numerous details of polymer gears. It proceeds from the most commonly used polymer materials employed to gears, going through the advantages and disadvantages and presenting the most typical failure types. Also, previous studies are described and based on them, the definition of the hybrid polymer gear employed in this project.

Chapter 3, named **Manufacturing Processes**, the main manufacturing methods are indicated, with injection treated in more detail. The chapter also addresses the influence that the geometry mould cavity can have on the strength of the gear.

Chapter 4, denominated **Definition of the Hybrid Gear Geometry and Mould**, describes the development of the metallic insert and hybrid gear design and the development of the mould. Furthermore, a mould filling simulation is performed.

Chapter 5, called **Manufacturing and Dimensional Control of the Inserts and Gears**, covers the process of manufacturing the inserts and hybrid polymer gears and also approaches the dimensional control of both.

Chapter 6, entitled **Definition of Operating Conditions for Gear Testing**, presents the calculation process to determine the load capacity of the polymer gear pair, using the VDI 2736 directive.

Chapter 7, named **Conclusion**, addresses the final conclusions of the dissertation and suggests future works.

Chapter 2

Literature Review on Polymer Gears

2.1 Introduction

Polymer gears have been widely introduced during the past years in engineering applications due to their potential technical and cost advantages over metallic gears. The immediate advantages polymer gears present are the cheaper mass manufacture, and they are significantly lighter than steel ones, due to their low density, and thus can reduce power consumption and noise level in gearboxes. Moreover, they can run without lubrication, which makes them appealing for non-lubricated applications, such as medical or laboratory devices, the food industry and household appliances [5, 6, 7, 8].

However, a crucial downside of polymer materials is their low heat conductivity and the considerable reliance of their mechanical properties on operation temperature. These characteristics limit the applications to a low maximum running temperature and a low load transmission [5, 7].

This second chapter outlines the most commonly used polymer materials, the advantages and disadvantages of employing polymer gears over metallic gears and their lubrication. Furthermore, the sorts of failure mechanisms linked with these gears were explored, as well as subsequent approaches for improving polymer gear performance.

2.2 Polymer Materials Applied to Gears

Polymers can be categorised into three groups: thermoplastics, thermosets and elastomers. Thermoplastics can be melted by applying heat and allowing their recycling (reheating and reprocessing) (reheating and reprocessing). On the other hand, thermosets cannot be recycled by thermal action after curing.

Thermoplastics are the most widely utilised for manufacturing polymeric gears and can be further split into two groups: semicrystalline and amorphous. Amorphous materials melt throughout a wide temperature range, and their molecules have no order in the solid state. On the other hand, semicrystalline materials pass into the solid state generating crystalline zones (ordered zones) [9].

The combination of polymers with other materials has been under continual development, with remarkable gains achieved each year. Regarding the tribological behaviour of polymer composites, it depends on the chain structure of macromolecules of the polymeric matrix and the action of fillers [10].

2.2.1 Properties

Listed in Table 2.1 are some properties of the most often used polymers in gear manufacture, including the maximum allowable temperatures in continuous and short-duration operation. Continuous time should be meant for several years, while short-duration refers to a few hours [11].

Polyamides (PA6, PA66, PA12 and PA46) have high tenacity, wear resistance, low friction coefficient and excellent chemical and electrical chemical and electrical resistance. However, this type of material tends to absorb water, which diminishes its strength. It also has worse dimensional stability than other engineering thermoplastics [12, 13, 14].

Polyoxymethylene (POM) absorbs less water than polyamide and is, therefore, more stable after production. This material demonstrates excellent creep and fatigue resistance, low coefficient of friction, and good chemical and abrasion resistance. Its downsides are its extreme sensitivity to severe heat and numerous contaminants, including additives [12, 13, 14].

Polyetheretherketone (PEEK) has a high heat resistance and can work continuously at temperatures of 250 °C. It has outstanding mechanical qualities and chemical resistance, yet its cost is high [14].

Polybutylene Terephthalate (PBT) is one of the most versatile thermoplastics. It is defined by high rigidity and chemical resistance, low weight, low humidity absorption, excellent electrical characteristics, lubricity, durability and heat resistance. Often glass fibres are added, yet, in sufficient amounts, the material becomes fragile [12].

Polyethylene Terephthalate (PET) has good dimensional stability, low crystallinity and high chemical resistance. Compared to PBT, it has higher thermal and environmental resistance but is still more fragile [15].

Polyphenylene Sulphide (PPS) demonstrates high dimensional stability and low water absorption and is non-flammable. The products obtained from glass fibre-reinforced injection contain high tensile, bending and impact resistance [12].

Table 2.1: Most important mechanical and thermal properties of polymer gear materials [11, 12].

Properties	POM	PA46	PA66	PEEK	PET	PBT	PPS
Density, ρ / kg m^{-3}	1390-1430	1180-1210	1130-1160	1260-1300	1380-1400	1300-1320	1340
Young's Modulus at 23°C, E / GPa	2.6-3.2	3.3	3.0	3.5-3.7	2.1-3.1	2.5-2.8	4.0
Poisson's ratio at 23°C, ν	0.42-0.45	0.37	0.38-0.42	0.4	0.37-0.44	0.41-0.42	0.4
Thermal conductivity, κ / $\text{W m}^{-1} \text{K}^{-1}$	0.25-0.3	-	0.23	0.29	0.24	0.21	0.25
Working thermal limits, short term, °C	110-140	180	140-170	300	170	160-165	200-240
Working thermal limits, continuous term, °C	90-100	140	80-100	250	100	100	270-300
Shrinkage (thickness < 4mm), %	2.2	1.8	1.8	1.2-1.7	1.8	1.8	0.6-3.5
Shrinkage (thickness $\geq$ 4mm), %	2.5	2.0	2.2	1.7-2.2	2.0	2.0	0.6-3.5
Price range, €/kg	1.8-3.0	7-12	3.1-3.6	80-100	1.7-1.9	2-3	8-12

Mechanical Properties

Polymers show viscoelastic behaviour, which indicates that their properties are impacted by many elements, such as temperature, load type and duration, and humidity, in addition to processing conditions [16].

With the increasing temperature, molecular bonds weaken, causing loss of stiffness, which is schematised in Figure 2.1, along with the influence of humidity.

The modulus of elasticity, yield stress and tensile strength rise with increasing applied stress, but the strain at rupture decreases. When the load is static, polymers undergo creep, that is, continuous deformation over time. For constant deformation, there is a decrease in load over time, and the material relaxes. For dynamic loads, the resistance decreases with the increase in the number of cycles [11].

Polymeric materials tend to absorb water. Thus we classify their state of absorption as: dry (in vacuum), wet (exposed to normal atmospheric conditions: 23 °C and 50 % relative humidity), and wet (working with water) (working with water). The absorption of moisture decreases the yield stress (Figure 2.2) and the modulus of elasticity (Figure 2.1), while the hardness, the deformation at the moment of yield stress and the deformation until rupture increase, as well as the volume of the piece, which may not be a uniform increase, causing swelling [11].

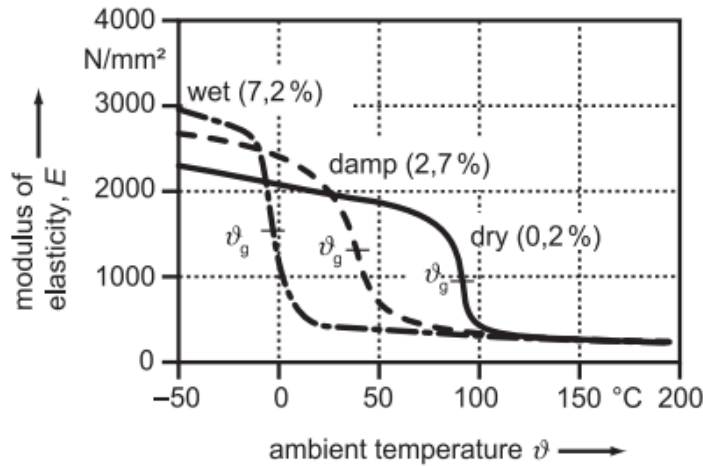


Figure 2.1: Effect of temperature and humidity on stiffness (PA66) [11].

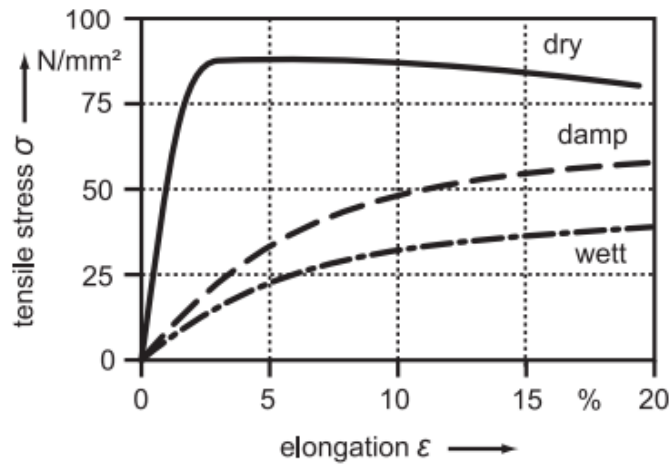


Figure 2.2: Effect of humidity on stress-strain curve (PA66) [11].

The logarithmic decrement, Λ , characterises the damping. Λ reflects the rate of decrease of the damped free vibration amplitude [17].

The higher the value of Λ , the better the damping qualities of the material, with a value of 0 corresponding to a fully elastic material. Damping is a vital aspect of decreasing noise

in operation. Figure 2.3 depicts the evolution of Λ as a function of temperature for different polymeric materials.

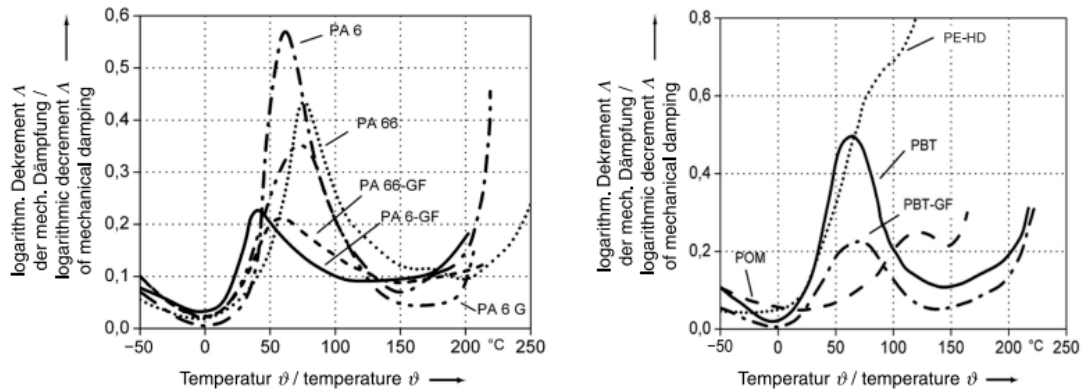


Figure 2.3: Mechanical damping as a function of temperature [11].

Thermal Properties

It has been observed that the thermal conductivity of polymers is lower than that of metals, making it more difficult to dissipate the heat created by friction during gearing. However, additives with high thermal conductivity, such as alumina, beryllium oxide, boron nitride, graphite, metals, and most efficiently, diamond, can be employed. However, it is vital to remember that these additives may modify other qualities of the material [18]. Table 2.1 provides typical values for the thermal conductivity of several polymers.

Chemical Resistance

Table 2.2 demonstrates the chemical resistance of the most commonly used polymers in gears when exposed to various chemical agents.

Table 2.2: Chemical Resistance. Resistant (+); partially resistant (o); not resistant (-); information not available (). Reprinted from [11].

Chemical		POM	PA 66	PA 46	PPA	PET	PBT	PEEK	PPS	PF
alcohols	methanol	+	o	+	+	+	+	+	+	+
	ethanol	+	o	+	+	+	+	+		+
	propanol	+	o	+	+	+	+	+		
water	cold	+	+	+	+	+	+	+	+	+
	hot	o	o	o		-	-	+	o	o
fuels	petrol	+	+	+	+	+	+	+	+	o
	diesel	+	+	+	+	+	+	+	+	
acids	hydrochloric acid	-	-	-	-	-	o	o		
	sulphuric acid	-	-	-	-	-	-	-		
	acetic acid	-	-	-	-	-	-	+		
	concentrated	-	-		-	o	-	o		-
	dilute	o	-		-	+	o	+		o
alkalis	potassium hydroxide	+	+			-	-	+		
	sodium hydroxide	+	+				-	+		
	ammonium hydroxide	+	o			o	o	+		
	concentrated	+	o		+	-	+	+		-
	dilute	+	o		o	o	+	+		o

2.2.2 Additives

The purpose of the additives used in polymeric materials is to simplify the manufacturing/processing of polymers and enhance their qualities. As a result, additives can be classified as: process modifiers, surface, chemical, mechanical, and aesthetic property modifiers. They also permit the use of less expensive polymers. However, they can also have a negative impact in other properties.

Mechanical Property Modifiers

Fillers - These additives are solid particles that are added to materials to increase their conductivity, flammability, impact strength, chemical resistance, appearance, and elastic modulus. Their utilisation results in cost savings and provides a means of obtaining particular properties for a given application. The concentration, size, shape, and distribution of the particles all have an impact on how the filler behaves concerning the polymer. The viscosity of the polymer increases with decreasing particle size, making processing more challenging. Large particles, on the other hand, function as a flaw, increasing stress concentration and decreasing impact strength (Figure 2.4) [18].

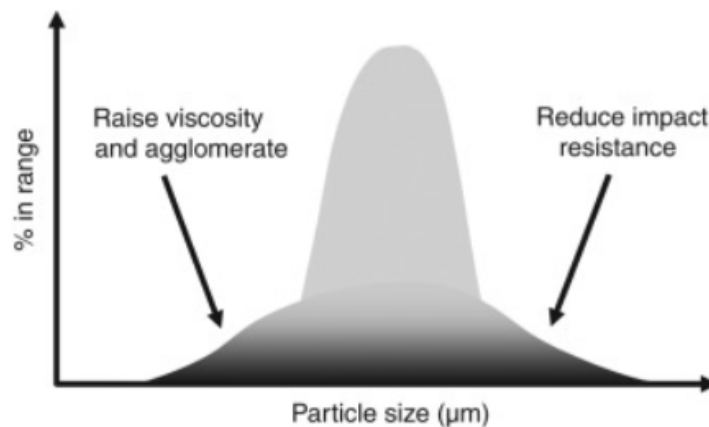


Figure 2.4: Particle size distribution and the effect on properties [18].

As fillers are frequently stiffer than the polymer, the modulus of elasticity increases with the addition of fillers. Fibres perform this increase better than isotropic additives (calcium carbonate, silica and dolomite). For this reason, the addition of coupling agents is pointless because it is also known that adhesion between filler and polymer has no impact on the modulus of elasticity [18].

Like the modulus, the yield stress rises more sharply when fibres are used than isotropic fillers and may even decrease its value. However, contrary to the previous case, there is a dependency in this one on the filler's ability to adhere to the polymer, making coupling agents effective in raising the yield stress. The yield stress value similarly rises as the contact area does [18].

The use of fillers also improves the hardness of the polymer, and again, the use of more anisotropic fillers is recommended. As the hardness is tested at the surface, an increase is only evident if the additive remains on it [18].

The inclusion of fillers increases the polymer's viscosity and, consequently, its resistance to creep [18].

Impact Modifiers - These additives distribute and dampen the impact energy as elastomer particles are introduced to reduce stiffness but increase impact resistance. The particles employed cover existing imperfections, like cracks, preventing them from proliferating. The latest sort of impact modifier seeks to ensure that the defect does not exist rather than filling it [19].

Nucleating agents - These additives are used in conjunction with semi-crystalline polymers to quicken the crystallisation process and speed up the injection cycle. Small crystals will result from a too-short crystallisation period [19].

Fibres - The addition of fibres to polymer increases its resistance and tenacity. The most widely utilised fibres are glass and carbon.

Process Modifiers

Thermal stabilisers - These additives prevent the degradation of the material as it is being processed and when it is subjected to high temperatures in service. Therefore they are used to extend the life of the product [20]. The effect of thermal stabilisers on the gear life is demonstrated in Figure 2.5 for the case of a polyamide. The most thermally sensitive polymers are PVC, PP and POM [19].

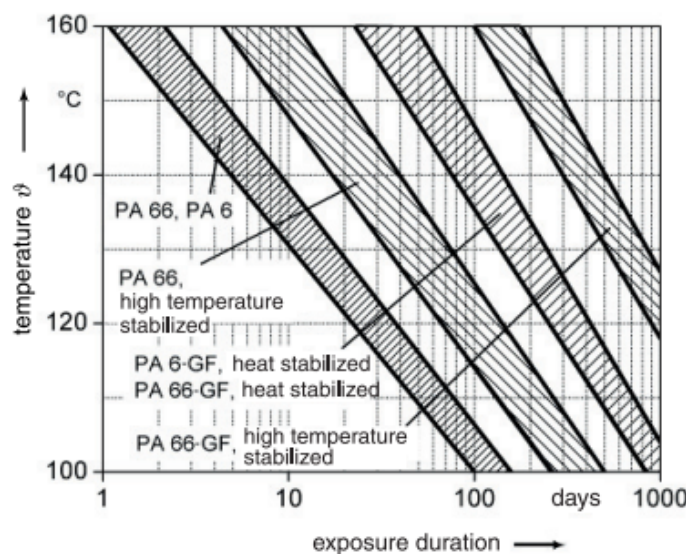


Figure 2.5: Effect of thermal stabilisers on gear lifetime [11].

Plasticizers - Plasticizers are used to get better flexibility, but they can also bring other advantages, like the possibility to increase the number of fillers or the ability to produce an antistatic effect. Unfortunately, its use causes several disadvantages, such as a loss in tensile strength and modulus of elasticity, pollution, toxicity, environmental hazards and fire resistance [19].

Lubricants - In order to release lubricants onto the contact between the polymer and the metal surface of the manufacturing equipment, lubricants can be added to the material. The film that forms between the layers facilitates the manufacturing process because it prevents the polymer from adhering to the tool. Additionally, these additives are utilised to reduce friction between finished components, regardless of whether or not they have the same composition. For example, 1-2 % graphite or molybdenum is added to Polyamide (PA) and other thermoplastics used to produce gears [21]. Molybdenum Disulphide (MoS₂), hexagonal boron nitride, Polytetrafluoroethylene (PTFE) and paraffin are also used to minimise

friction [18]. The influence of lubricants on some polymers' friction coefficient is shown in Table 2.3.

Table 2.3: Friction coefficient between polymer and metal [22].

Material	Friction Coefficient
POM	0.2 - 0.35
POM com PTFE	0.15 - 0.27
PA-6	0.2 - 0.4
PA-6 com grafite	0.1 - 0.25
PC	0.35
PC com PTFE	0.15
PC com fibra de vidro e PTFE	0.20
PPS	0.15 - 0.3
PPS com fibra de carbono e PTFE	0.15 - 0.3
PAI	0.15 - 0.3

Surface Properties Modifiers

Coupling agents - These agents are used to modify the surface properties of fillers and reinforcements to improve the adhesion between them and the polymeric material. They, therefore, contribute to an increase in mechanical and chemical resistance. Coupling agents are essentially inorganic materials [20].

Mould release agents - Using release agents is the key to keeping the part from sticking to the mould. This is accomplished by lowering the coefficient of friction between the metal mould surface and the polymer. These additives, however, are very challenging to remove and frequently persist on the surface of the mould.

Chemical Properties Modifiers

Fire Retardants - The most used and least toxic fire retardant is aluminium hydroxide. Its role is to release water at temperatures above the melting point of specific polymers. Hence it is not utilized in some thermoplastics such as polyamide or PBT. Halogen-based retardants are also employed since they have long been proven beneficial in the electronics business. However, halogen-free retardants are beginning to emerge as they are less harmful and better efficiency may be achieved [19].

UV Protection - These modifiers are meant to protect the components from solar exposure. Black carbon can be produced using a variety of substances, including natural gas, oil, coconut shells, and other organic or inorganic elements. As it absorbs the harmful energy, it

gives the component a black colour, shielding it from the impacts of UV rays. This additive can also function as a pigment or an antioxidant [20].

Antioxidants - It prevents polymer degradation owing to oxidation, heat action during processing and also protects it from environmental conditions [20].

The influence, either positive or negative, of some fillers and reinforcing materials in tribological, mechanical and thermal properties is presented in Table 2.4.

Table 2.4: Influence of using filler and reinforcing materials in polymer gears [11].

Filler or reinforcing material	Positive influence	Negative influence
glass, carbon or aramid fibres	increase in stiffness, tensile strength, flexural fatigue strength and heat distortion resistance	reduction in impact strength
PTFE	reduction in friction and wear	reduction in impact strength and flexural impact strength
PE	reduction in friction	reduction in impact strength
graphite, boron nitride	reduction in friction and wear, increase in thermal conductivity	reduction in impact strength
silicone oil	reduction in friction and wear, increase in toughness	-
mineral fillers	increase in heat distortion resistance	-

Polymer selection for gear applications

The best gear polymers should have high bending strength, stiffness and shear strength. Other essential qualities are compressive strength, wear resistance, and a low coefficient of friction [22].

When the gear is subjected to high speed and torque conditions, high operating temperatures are to be expected. Therefore, if external lubrication is not applied, the polymer is intended to have great dimensional stability at high temperatures and a low coefficient of friction and wear. In addition, the polymer must have a high level of stiffness and hardness in situations where the torque is high and the gear rotation speed is low. Therefore, a material reinforced with glass fibres would be appropriate. On the other hand, materials with superior damping qualities are preferred when the required torque is low, but the rotational speed is high, as this results in more operating noise [11].

Anytime the gear abruptly reverses its rotational direction, high-impact forces are to be expected. Therefore, it makes sense to select materials with excellent impact resistance and good damping properties. A material with high fatigue strength should be chosen if the gear needs to withstand many load cycles [11].

To simplify the selection of a particular polymer, Table 2.5 qualitatively compares the properties of some of the most typically used polymers in gears.

Table 2.5: Qualitative comparison of properties of polymers used in gears.
 ◆ perfect; ○ good; △ satisfactory; × unsatisfactory [22].

Properties	PA	PA-GF	POM	PBT	PC
Mechanical strength	○	◆	○	○	○
Stiffness	△	◆	○	○	○
Wear resistance	◆	○	◆	○	×
Creep resistance	○	◆	◆	○	○
Fatigue strength	○	◆	◆	○	△
Heat-resistance	○	◆	○	◆	○
Resistance to oil and chemicals	◆	◆	◆	◆	×
Dimensional stability	△	○	○	○	◆
Resistance to water absorption	×	×	◆	◆	◆
Formability	○	○	○	○	△
Quality at subzero temperature	○	◆	○	○	◆
Atmospheric resistance	△	○	×	○	◆

2.3 Advantages and Disadvantages

As mentioned above, the use of polymer gears is rising fast and even replacing metallic gears used in low and medium-power transmissions. The mechanical and tribological properties of the polymer material awards these type of gears with qualities not found in standard steel gears. The main advantages for polymer gears are [7, 23]:

- Dry running capability;
- Low density and inertia, resulting in better dynamic response;
- Improved noise levels in operation, since they have good internal damping capacity and high resilience;
- Chemical inertness, strong resistance to corrosion;
- Economic manufacturing processes, particularly for mass production and the capability to produce complex geometries.

Despite having all these advantages, polymer gears still have several demerits in their mechanical and thermal properties compared to metallic gears. For instance, polymers typically have a tensile strength of around 1/7, Young's modulus of about 1/75, thermal conductivity of about 1/80 and coefficient of linear expansion 11 times those of steel [24]. The main disadvantages for polymer gears are [5, 7]:

- Low thermal conductivity;
- Poor mechanical resistance;
- High hygroscopicity;
- Mechanical properties significantly influenced by the operating temperatures;
- Limited load capacity and maximum running temperature;
- Dimensional instability.

2.4 Gear Failures Mechanisms

Unlike their metal counterparts, the main failure modes for polymer gears often involve melting owing to thermal overload, wear of the gear surface causing a reduction in tooth thickness, and tooth root and pitch fracture due to mechanical overload [8, 23, 11].

The modes of failure of polymer gears can be significantly influenced by their material properties. For instance, melting may occur during gearing if a load is applied greater than the materials' specific load. As a result, this will damage the tooth flanks and could result in wear and fractures. The exterior face and active flank of the same flank can be the source of tooth fracture. Wear causes the tooth surface to change while reducing the tooth cross-section, especially in dry-running circumstances. External lubrication increases the risk of tooth fracture and pitting [23, 11]. Figure 2.6 shows a graphic illustration of each mode of failure for polymer cylindrical gears.

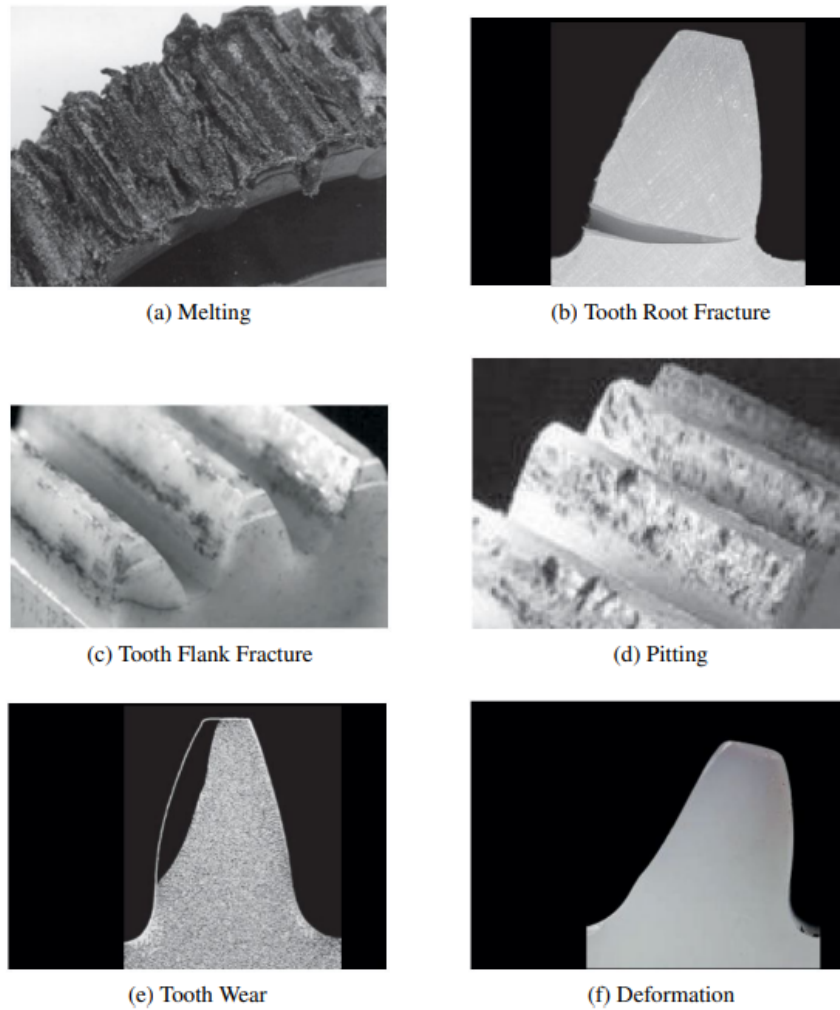


Figure 2.6: Polymer gear failure mechanisms [11].

Once the melting temperatures are achieved, the melting problem (Figure 2.6a) occurs, resulting in irreversible alterations of the tooth geometry [7, 11].

The tooth root and pitch are where fracture generally happens, as shown in Figures 2.6(b) and 2.6(c). The excessive bending stress in the first case is what caused the pitch fracture, whereas the shear stress, extreme hertzian stress, and the identical effect in the second case are what caused the pitch fracture. The active flank and outside face of the same flank can both produce cracks that can grow until the fracture phenomenon occurs. The fracture issue can be delayed by reinforcing the polymer gear tooth utilising fibre direction parallel to the tooth surface [7, 11].

Pitting is caused by high levels of simultaneous shear, bending, and Hertzian stresses, similar to the reasons for fracture. As seen in Figure 2.6(d), close micro cracks start to

form and spread until they finally meet at a specific point, which encourages material removal and fracture. It is advised to regulate pressure differences along the tooth using face width modifications from the root to the tip to prevent pitting [7, 11]. When using external lubrication, it is common for teeth to fracture and pit [11].

Figure 2.6(e) representation of tooth wear demonstrates layers of material removal. It is primarily caused by the applied force and relative sliding speed, which are frequently aided by the vigorous abrasive action of the frictional partner or by the insufficient wear resistance of the own polymer material. Fibre reinforcement of the polymer gear is one method of controlling this phenomenon [11, 25].

Involute profile distortions, such as pressure angle variation, are permanent and result from excessive surface stress when it reaches the deformation limits, as shown in Figure 2.6(f). Flexurally rigid and hard materials, as again the fibre-reinforced materials, are advised to prevent deformations [11, 25].

2.4.1 Temperature Influence on Operating Conditions

Understanding how temperature affects the behaviour of polymer gears is crucial for reducing its adverse effects when in use. The bulk temperature and the transient temperature of the gear surface, also known as flash temperature, can be used to divide the temperature field of meshing gears into two basic categories. Since the load-carrying capacities of polymers are substantially lower than those permitted for steel and are highly dependent on the operating conditions, both temperature characteristics are essential for selecting polymers [5].

As was already indicated, as tooth temperatures rise, polymer materials' mechanical and tribological characteristics deteriorate, making these gears increasingly prone to various failure modes. For instance, contact between two meshing surfaces may result in a reduction in the load capacity of the polymer and a deterioration of the mechanical properties of the polymer, such as stiffness. Additionally, thermal expansion may alter the tooth's size to the point where meshing might be impossible, leading to tooth failure [7, 11]. Furthermore, the poor thermal characteristics of polymers might aggravate the previously described forms of failure due to their low thermal conductivity, which makes it difficult to evacuate the heat.

Although the temperature distribution of polymer gears varies depending on the material and operating conditions, the temperature is typically higher at the pitch point, where rolling contact occurs, than at the tip or root of the tooth, where sliding occurs. In other words, the temperature at the tip and root of the tooth is lower during the meshing phase of two teeth because heat dissipation there is easier than near the pitch point [8, 24].

2.4.2 Tooth Deflection

The tooth deflection must be considered because polymers' stiffness is significantly lower than steel's. The tooth deflection in materials with low stiffness can result in catastrophic gear meshing failure. Therefore, the applied load, the tooth face width, and the material stiffness can all affect how much a spur gear deflects [24, 26].

In tensile and compression regions of polymers, different stress-strain behaviour might result in inconsistent Young's modulus values. Additionally, polymers can exhibit nonlinear behaviour in elastic areas incompatible with Young's modulus theory [26]. Since the deflection of the tooth changes depending on the type of polymer used, it is interesting to examine the deflection curves of PA and POM in Figure 2.7. POM experiences larger deflections when the load is applied since it is slightly less rigid than PA. However, it only happens at the initial instants because POM has a lower viscosity than PA, which allows POM to exhibit smaller tooth deflections in the subsequent instants, and also to reach the unloaded position faster [27].

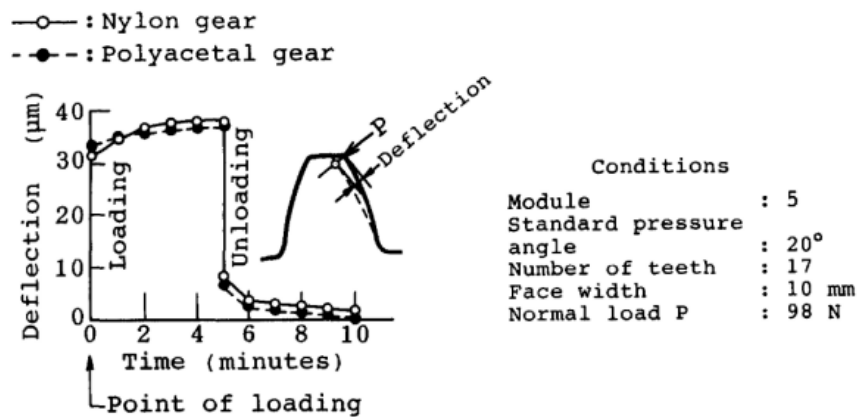


Figure 2.7: PA and POM deflection behaviour [27].

2.4.3 Wear on Plastic Gears

The wear mechanism consists of material removal on the tooth surface. It can be quantified by the wear rate, corresponding to the amount of tooth thickness reduction per rolling cycle [7].

The typical tooth shape of a POM gear after wear is shown in Figure 2.8. It is known that when the pressure angle rises (α), wear increases. This is a result of the constant torque that must be transmitted. In addition, the force value (F_n) must increase because there is a slight decrease in the distance between the point of force application and the gear's centre. As a result, there is more contact force, which promotes more wear [28]. Figure 2.9 illustrates the wear progression as a function of speed and transmitted torque.

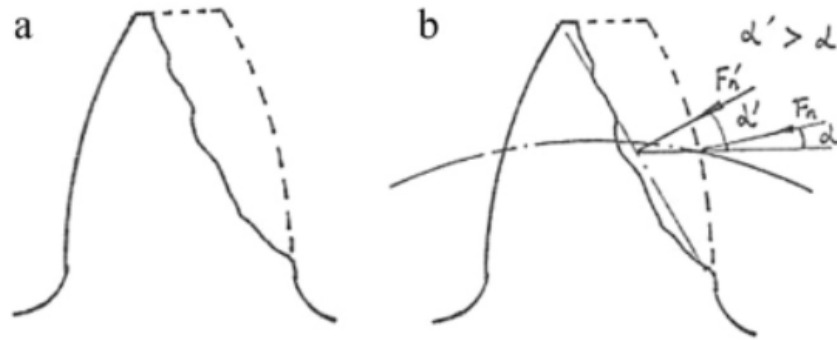


Figure 2.8: Typical wear form for POM gears [28].

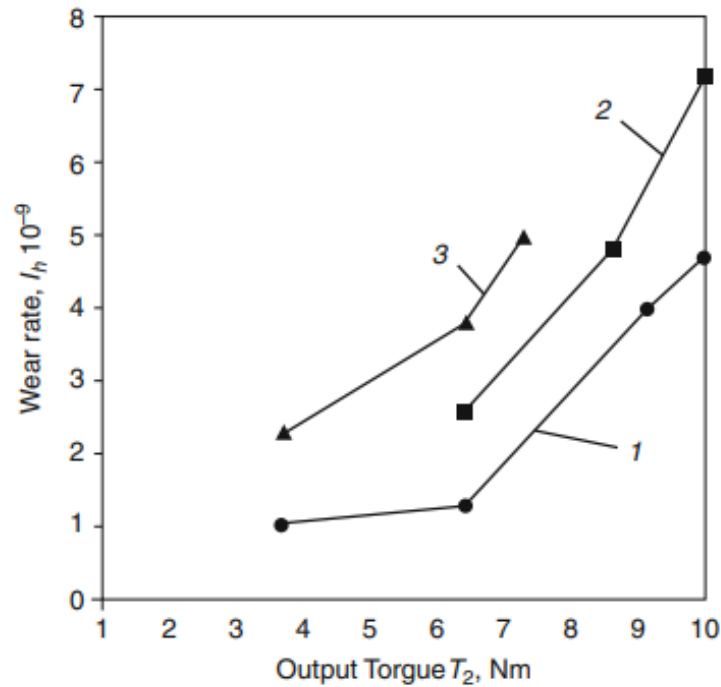


Figure 2.9: Dependence of wear intensity on load and speed rotation for POM material. (1) $n = 1500$ rpm; (2) $n = 3000$ rpm; (3) $n = 5000$ rpm [22].

Wear is significantly influenced by the combination of materials used for the pinion and wheel, so Mao *et al.* [8] evaluated combinations utilising POM, PA and steel gears. Spur gears were tested, and their parameters are presented in Table 2.6.

Table 2.6: Specifications of the spur gears tested by Mao *et al.* [8]

Parameters	Value
Module	2 mm
Number of teeth	30
Pressure angle	20°
Face width	17 mm
Nominal backlash	0.18 mm
Tooth thickness	3.14 mm
Contact ratio	1.65

The results showed that the wear behaviour for POM/POM and PA/PA could be separated into three phases: the "running-in," the "nearly linear," and the "final fracture" phases. This division was more pronounced in the POM/POM case for medium loads (torques between 8.5 and 9 Nm) and in the PA/PA example for high loads (torques above 10 Nm), both cases assuming running speeds of 1000 rpm. The first "running-in" phase covers a high amount of wear during a short period of time. In contrast to the preceding stage, wear increases gradually during the next "nearly linear" phase with linear behaviour and a low wear rate. Finally, wear increases quickly until failure occurs in the "final fracture" phase.

During the same work, the wear performance of polyacetal against nylon was also studied, and the results were substantially different from those of the PA/PA and POM/POM tests. Similar to what happened in the POM/POM situation, when nylon was used as the driver, the polyacetal gear failed due to thermal wear. However, a considerable improvement in wear performance over the prior combinations was made when polyacetal was utilised as the driver. Mao *et al.* [8] testing demonstrated that wear highly depends on both the material and drive/driven combinations.

2.5 Lubrication

As was already noted, one of the major benefits of using polymeric gears is that they operate without the need for lubrication. But the application of lubricant significantly enhances their performance in terms of noise, friction, and wear.

Lubricating additives may be present in the composition of gears running without external lubrication (see Subsection 2.2.2), which modifies the mechanical characteristics and, reduces friction, wear, and increases the lifetime.

2.5.1 Lubricating Greases

Greases can be made from mineral or synthetic oils. Mineral oil-based greases have a higher piezo-viscosity coefficient than synthetic greases, which may provide them with a better lubricating capacity despite synthetic greases' higher resistance to ageing. Since grease would be projected out of the tooth flanks at higher speeds, compromising the quality of lubrication, the circumferential velocity is restricted to 5 ms^{-1} [11].

2.5.2 Oils

The operating temperature range and the materials of the gear have an impact on the oil that should be used. Table 2.7 suggests the type of oil to be used depending on the temperature.

Table 2.7: Recommended lubrication function of the operating temperature [11].

Operating Temperature, °C	Recommended Lubricant
[−80, 20]	Synthetic Oils
[20, 60]	Mineral Oils
[60, 100]	PAO, ester oils
[100, 120]	Ester oils, silicon, PFPE

The polymers most commonly used in gear mechanisms are POM and PA. They belong to the polar thermoplastics group. Thus, non-polar lubricating bases such as Polyalphaolefin (PAO) should be utilised. Polar oils should be used when gears are made of non-polar polymers such as Polyethylene (PE) [11].

2.5.3 Tribological Tests

The pin-on-disk, twin-disk, and block-on-ring devices are the most often used equipment for conducting tribological tests on polymeric materials used in gears [29]. The normal force, the transverse force, and the penetration of the pin into the disc are measured in the test using a pin and a revolving disc to calculate the friction coefficient and the wear coefficient. The results of this test are shown in Figure 2.10, indicating that certain materials are harmful to gear mechanisms made of the same polymer.

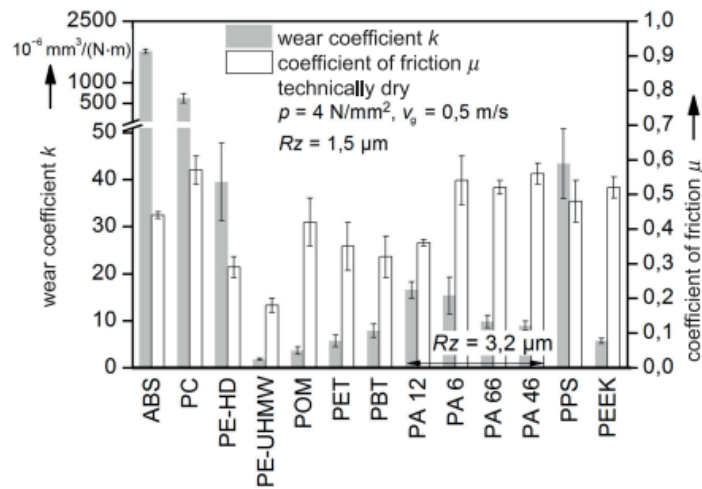


Figure 2.10: Characteristic tribological values of selected thermoplastics in the pin-and-disk test [11].

2.6 Methods to Improve Polymer Gears Performance

As was previously said, an essential aspect of polymer gears' applicability is their performance and durability. Because of this, various approaches have been used to improve the performance of polymer gears. These methods include changing the shape of the gear teeth, including cooling holes, combining multiple materials for the gear pair, including fillers or reinforcement materials, and, finally, developing hybrid polymer gears, where inserts will be employed. These methods have been demonstrated to lengthen the gears' life cycle, reduce wear, increase durability, and improve thermal behaviour.

2.6.1 Gear Tooth Geometry Modification

In order to determine the best polymer gear tooth geometry for a specific application, many techniques have been used while considering life span, temperature, load, and tangential speed. However, since no particular standard evaluates the characteristics of polymer material, polymer gears are often designed using the same specifications as metal gears.

Imrek *et al.* studied the effects of the tooth width modification by performing tests on three different types of gears: two gears made of Nylon 6, where one had a modified tooth width, while the other was unmodified, and another AISI 1045 steel alloy unmodified gear. The modified gear tooth geometry is shown in Figure 2.11 [30].

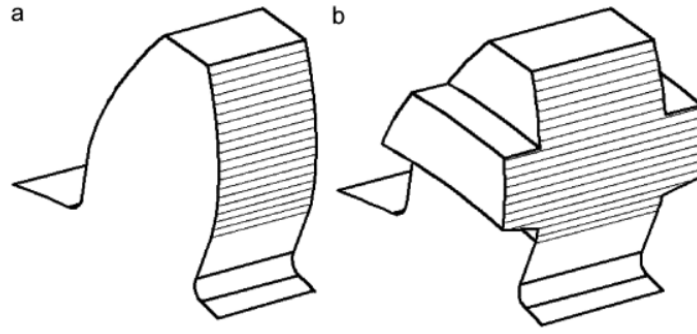


Figure 2.11: (a) Unmodified gear tooth; (b) Modified gear tooth [30].

Two tooth pairs should ideally be in contact simultaneously for gears to mesh continuously, distributing the tooth load between the two contact pairs. However, alternate double and single-tooth interactions occur in actual cases. The tooth contact regions are shown in Figure 2.12a, and the load distribution and sliding velocity distribution along the contact line are shown in Figure 2.12b. It is interesting to note that some studies indicate that tooth load does not change instantaneously since Figure 2.12b ignores the elastic effects of the gear tooth, but that change can be roughly approximated as linear depending on the gear material [30]. Therefore, Imrek *et al.* increased the tooth width in the single tooth contact zone to keep the load-sharing ratio constant.

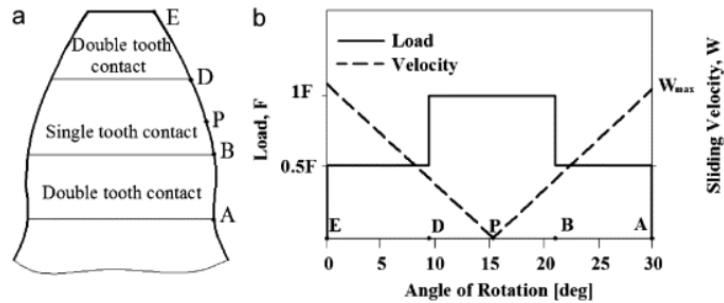


Figure 2.12: (a) Tooth-contact areas and limits; (b) theoretical load distribution and sliding velocity distribution along meshing axis [30].

The author also performed fatigue testing of the gear pairs on the FZG test machine. The results showed that the average surface temperatures of unmodified gears are more significant than those of the modified ones. The modified gear achieved surface temperatures 10 to 15 °C lower than the unmodified one for the several Nylon 6 gears. In addition, the tooth width modification was found to reduce wear rates on the tooth profiles in terms of wear and performance. Imrek *et al.* [30] concluded that using modified gears to improve heat dissipation and minimise heat generation promoted the extension of the gears' life performance and prevented premature failures.

2.6.2 Cooling Holes

This method consists in drilling cooling holes in the gear tooth to decrease the amount of thermal damage by reducing the heat accumulated in the tooth. The air circulation caused by the rotation of the gears would disperse the heat that was collected in this area. For that reason, a configuration shown in Figure 2.13 includes a hole in a specific radial position, and two additional holes along the radial direction of the tooth body were used [31].



Figure 2.13: Cooling holes configuration [31].

The load-carrying capacity of the gear pairs was evaluated on an FZG machine. The experiments were conducted in dry conditions with a low load of 2 Nmm and an initial pinion rotational speed of 200 rpm. It was demonstrated that cooling perforations improved heat conduction, resulting in a decrease in tooth body temperature. Higher tooth loads, however, caused thermal damage to the pitch region of some gear teeth [31].

2.6.3 Hybrid Gears

This solution typically consists of a polymer gear that can accommodate a metallic insert or any other viable material with good thermal conductivity and adequate mechanical properties to improve heat conduction, mechanical resistance, and gear life. The notion of hybrid gears has been the subject of numerous studies over the years, and recent research will be covered in more detail in the next section. The idea of hybrid polymer gears will be introduced in this subsection, along with other authors' solutions.

A numerical model developed by Fernandes *et al.* [6] was used to evaluate the impact of insert geometry and material properties on a gear contact. In order to determine which of the three insert materials would be the most advantageous to utilise, the authors first investigated the impact of inserts in a polymer gear using POM as the polymer material. So, inserts made of steel, copper, and aluminium were tested. At the beginning of the project, a simple insert in the shape of a rectangular block was placed in the middle of the teeth (Figure 2.14), and its effects on the tooth mass and the contact pressure between the polymer and the metallic insert were studied. Regarding the insert material, the aluminium insert

offered the optimum balance between mass, heat conduction, manufacturing options, and cost. Aluminum had the advantage of having a lower mass increment even though the copper insert's maximum temperature was lower. Nevertheless, the values are relatively similar. The maximum temperature dropped due to the conclusions regarding contact pressure being translated into a higher heat flux with increased interface pressure. The authors decided to concentrate on the effect of the insert shape because, at contact pressures above 10 MPa, the heat transfer coefficient had no impact on the maximum and minimum temperatures [6].

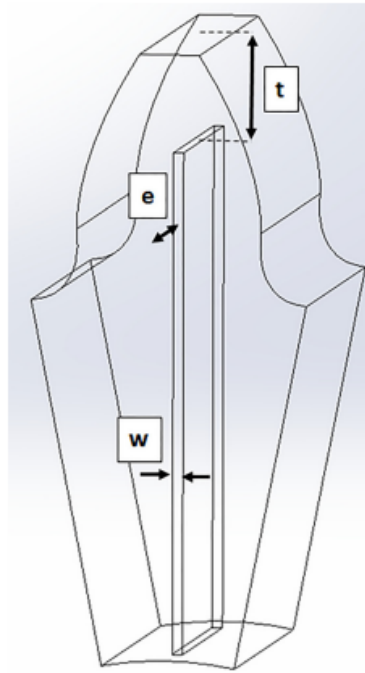


Figure 2.14: Basic Hybrid gear model and geometrical parameters [6].

Four new insert designs were evaluated and are shown in Figure 2.15 after Fernandes et al. [6] determined that an aluminium insert was the most suitable for future testing. The geometrical parameters were improved, and the effect on tooth temperature was studied. Involute profile (d) and double T-profile (c) were found to have maximum temperature reductions of approximately 28 % compared to standard POM gear. However, both profiles also had the highest increase in the minimum temperature. The maximum temperature reduction was 17 % for the T-profile (b) and 11% for the plate profile (a). The involute profile (d) reaches the closest position to the contact path, which allows for the most effective heating evacuation among the four geometries. In the end, all the different inserts successfully lowered the maximum temperature but also managed to increase the minimum temperature. The final conclusions established the direct impact of the contact area between the polymer and the insert and the position of the insert with the heat evacuation capacity. There was no association between mass and maximum operating temperature [6].

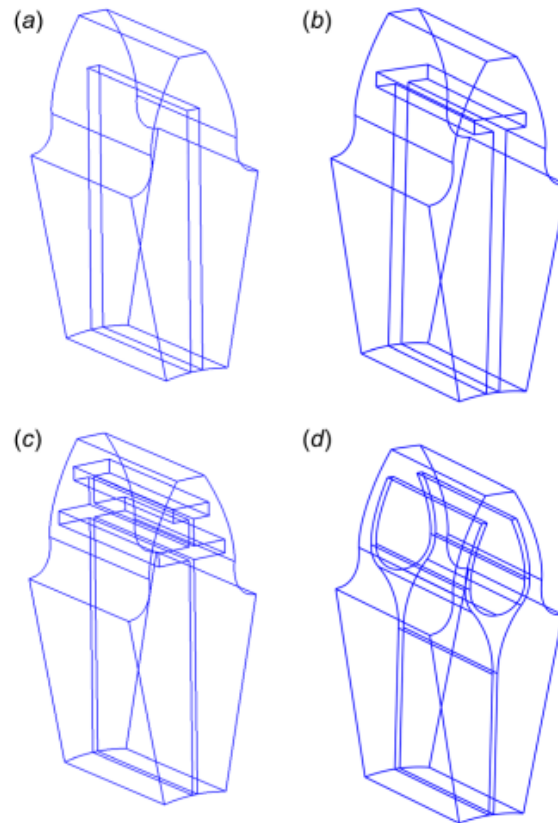


Figure 2.15: Different insert geometries: (a) plate; T-profile; (c) double T-profile; (d) involute. [6].

In summary, even if the concept of an insert in a polymer matrix seems promising, its implementation is complex due to the thermal contact, mechanical adhesion, and manufacturing issues that arise between the several bodies made of different materials [6].

2.7 Hybrid Polymer Gears

The main works that will serve as the basis for this thesis's purpose will be given in more depth in the following subsections.

The results of earlier research by Moutinho [1] on various metallic inserts will be examined, and Hooton [2] and Pinto's [3] validation of the hybrid polymer gear design will also be presented. The objective of Moutinho's project [1] was to reach the best compromise possible concerning the insert geometry, its dimensions, the material to be used, and its placement within the polymer body. The research of Hooton and Pinto [2, 3], which validates the hybrid gear tooth model through tests using their own heating system, was based on a particular insert model.

2.7.1 Moutinho's Dissertation

In his research, Moutinho [1] tested various geometries of metallic inserts, measuring stress and temperature for each geometry and comparing the results. The principal objective was to find the optimal compromise between different insert materials and geometries while explaining how each one affected the temperatures, tooth root stresses, and loads. As a result, the author concluded which insert type improved the thermo-mechanical behaviour of the polymer hybrid gear the most.

For the metallic inserts, four alternative geometric options were under analysis. Starting with the plate type, it could be a simple or a T plate. After, the type of insert, fixed when the insert is attached to a fixed ring around the borehole or floating when simply inserted in the matrix of the gear. Also, the type of corners, filleted or not. Finally, the insert thickness. An illustration of some insert models is provided in Figure 2.16.

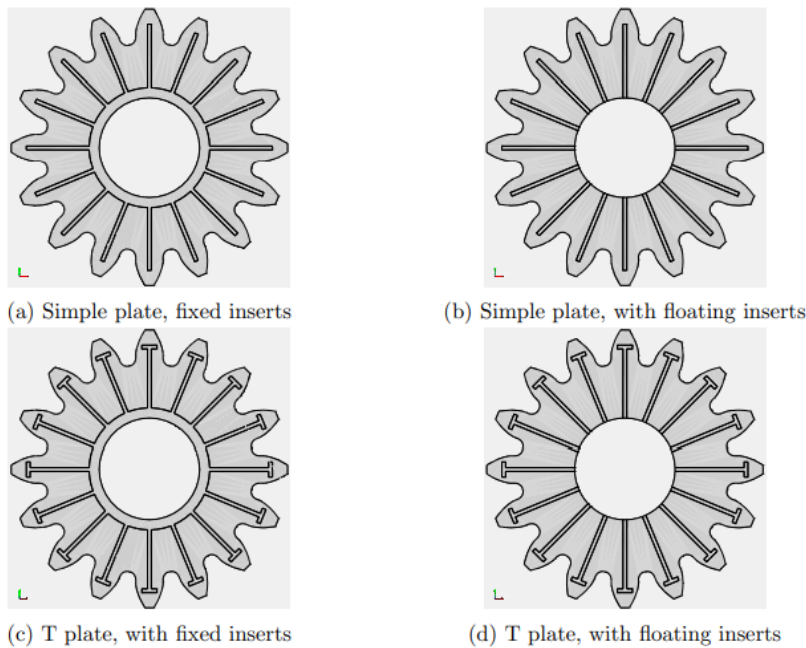


Figure 2.16: Example of hybrid gear studied by Moutinho [1].

Moutinho [1] created a FEM model to evaluate the effects of the following geometric characteristics for a C-14 gear to establish the best geometry for the insert: the distance (t) between the insert and the tooth tip, the insert's width (w), the length (L) of the horizontal plate for T-shaped inserts, the insert's filleted or rectangular corners for the insert in contact with the polymer, the fillet radius (R_d) of 1 mm or 0.5 mm, and finally a simple or T-profile for the insert. Thus, in figure 2.17 are presented the insert's geometric dimensions.

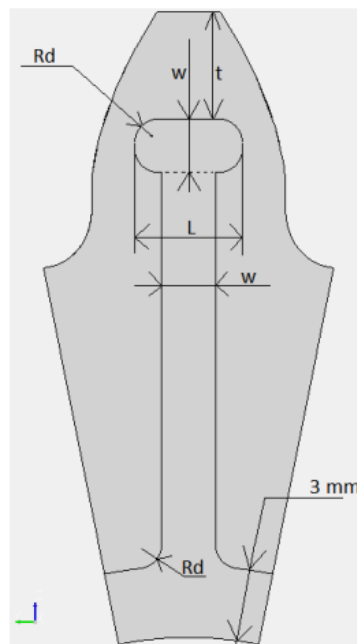


Figure 2.17: Dimensions for the geometric variables [1].

Under identical loading conditions with a torque of 10 Nm and a rotational speed of 1000 rpm, all insert profiles were tested. According to the FEM results, the temperature value for every combination of geometry was lower than the temperature for the standard polymer tooth. For polymer gears with a simple plate insert, the FEM results also showed a considerable increase in tooth root stress. Despite this, the possibility of the simple plate design remains since a lower temperature at the point of maximum tooth root stress could still increase the durability of polymer gears. Although the T-profile insert showed the lowest tooth root stress, it also led to a weight increase of 25 % compared to the standard gear. From these initial findings, Moutinho deduced that the optimum alternative for the hybrid gear design would be a simple plate insert with a reduced weight increase. Nevertheless, the straight edges and proximity to the hub of the simple plate designs increased the stress and reduced the load-carrying capacity.

To study the tooth root stress when the base of the insert was not in touch with the hub, Moutinho optimised the floating inserts by decreasing the insert length while preserving the other dimensions. It was expected that this solution would result in higher temperature and tooth root temperature but would reduce tooth root stress. The process involved simulating three different length increments from the insert to the hub to improve the outcomes of the initial simulations. After analysing the results, it was determined that a distance equal to the gear's module was ideal since it presented a low enough value of tooth root stress without significantly raising the temperature of the tooth root. In conclusion, the optimal geometry for the insert would be a simple plate design with straight corners, a width equal to half the module's value, and a distance from the hub and the tip equal to the module's value.

Following the hybrid gear's design optimization, Moutinho selected an acetal polymer gear and evaluated the effects of several insert materials. Steel, copper, and aluminium metallic inserts were compared in the first phase. According to the FEM analysis, aluminium had the highest tooth root stress, followed by copper and steel. Additionally, it revealed that aluminium had the lowest temperature under conditions that caused the maximum tooth root stress, followed by copper and steel. Therefore, Moutinho concluded that aluminium would be the best insert material.

Moutinho proposed a final improvement due to the significant stresses in the hybrid tooth interface caused by the polymer material's thermal expansion. The insert was made out of an epoxy silver-filled material. The benefits were its better compatibility with POM than aluminium, higher thermal conductivity and higher Young's modulus than POM.

2.7.2 Hooton's Dissertation

As was described at the beginning of the chapter, Hooton's study [2], based on the final hybrid geometry proposed by Moutinho [1], confirmed the viability of the hybrid polymer gear and validated its concept. The tooth polymer material used was POM as well.

Hooton created and produced a gear tooth heating system in order to test the hybrid gear tooth experimentally. The system's objectives were to measure the temperature of different sections of the tooth and to administer concentrated heat from a controlled source on both gear tooth flanks. In this way, pressure was applied to the gear tooth while it was heated to simulate the meshing conditions more accurately. In summary, the system's primary requirements were to apply controlled pressure and heat to both tooth flanks, control the point at which the applied load made contact with the tooth flank, and be able to use type K thermocouples to determine the surface temperature of the tooth. Figure 2.18 illustrates the author's final system, reached after a few iterations.

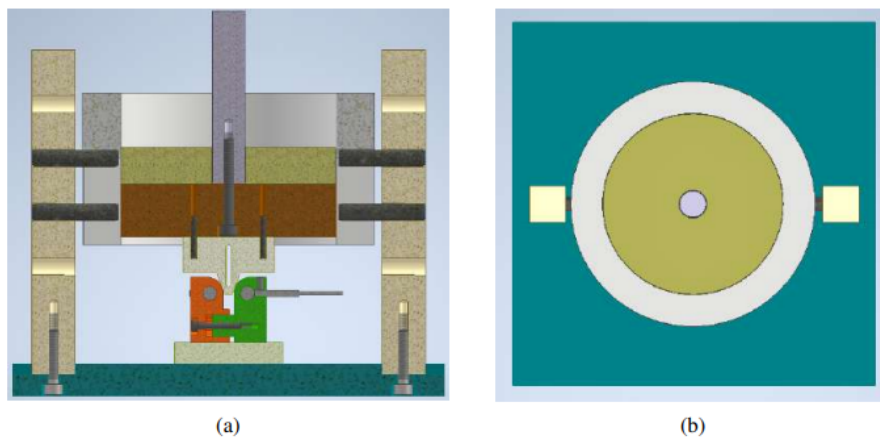


Figure 2.18: Final heating system design: (a) Section view; (b) Top view [2].

Hooton first created a FEM model of the teeth to compare and comprehend the experimental results before moving on to the tests. For the purpose of simulating the exact condition of the Heating System, both thermal and thermo-mechanical models were developed. Boundary conditions for the thermal model consider the Heating System design and the method followed for the thermal tests. The mechanical model's development improves upon the earlier simulation by more accurately replicating the actual conditions the tooth rack is subjected to. Again, the boundary conditions took into account the Heating System design. Finally, the mechanical and thermal models were combined via the thermo-mechanical model. Thermal tests and Digital Image Correlation (DIC) tests were prepared once the FEM models were defined.

The purpose of the thermal testing was to explore the effects of various insert materials and to validate the hybrid gear idea by heating the tooth flank. The Standard rack tooth (STD), the hybrid tooth rack without an insert (HT1), the hybrid tooth rack with an aluminium insert (HT2), and the Hybrid tooth rack with an epoxy insert (HT3) were the four specimens evaluated. The locations of the thermocouple boreholes are shown in Figure 2.19.

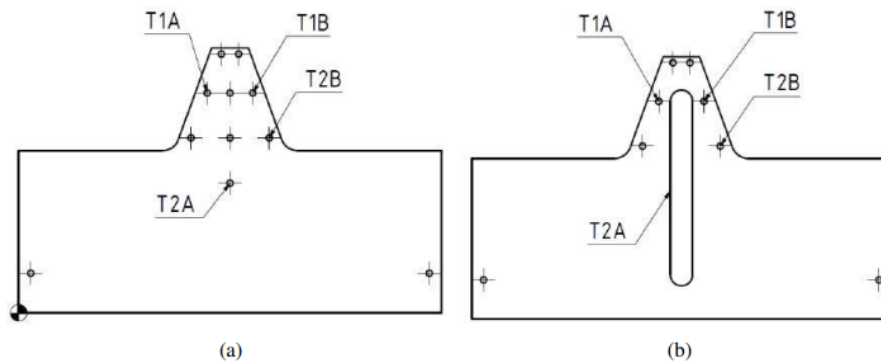


Figure 2.19: Location of (a) standard tooth and (b) hybrid tooth boreholes for thermocouple [2].

All four specimens underwent thermal testing measurements with a heat temperature in the [40, 50, 60, 70, 80, 90, 100, 110] °C interval with just one flank of the tooth heated (the T1b flank side). The relative temperature change for each hybrid tooth compared to the regular tooth was calculated to provide a more detailed analysis of the effects of the hybrid tooth design. For temperature T1a, STD and HT1 (air) produced results with an average temperature change of 1.9 %, while HT3 (epoxy) and HT2 (aluminium) recorded temperature changes of 6.7 % and 12.6 %, respectively. From these preliminary results, it was possible to conclude that the hybrid teeth were better able to absorb heat when an insert was present. The analysis of T1b temperature revealed that HT1 (air) was 5.2 % lower than STD, which might be explained by convective heat transfer inside the insert's region. The most significant average temperature change was for HT2, 27.9 %, followed by HT3, 19.6 %.

%. Since the insert will be located there, T2a was a crucial area to study. With an average temperature change of 1.1 % for HT3, HT1 and HT3 did not record any notable temperature changes. However, HT2 has detected a 30.4 % average shift in temperature. The different thermal conductivity of the materials introduced may be the cause of the discrepancy between the HT2 and HT3 results.

Using the digital image correlation (DIC) method, surface displacements are determined by taking several images throughout the experiment and comparing the coordinates of the reference image and the distorted image. These studies aimed to investigate the tooth rack polymer material's thermal expansion and to confirm the thermo-mechanical model once again when heat was introduced to the tooth flank. The hybrid tooth without an insert and the standard tooth rack served as the test specimens, and Figure 2.20 shows where the thermocouple boreholes are located.

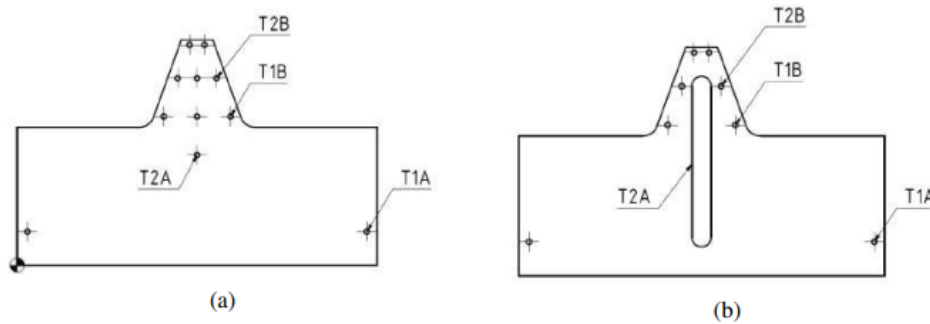


Figure 2.20: DIC: (a) standard tooth and (b) hybrid tooth boreholes for thermocouple [2].

The heating temperature used for the DIC testing was in the range $T_{Heater}=[40, 70, 100]$ °C. It showed the highest thermal expansion for the standard tooth rack at $T_{Heater}=100$ °C. Due to the right side's cartridge heater being the only heating source in the x direction, the right side's tooth saw more thermal expansion. The highest temperature was near the pitch point, however, the heaters prevented the tooth's growth in the x direction. Higher displacements in the y direction were found for the gear tip and lower displacements for the tooth base. The cartridge heater's contact point, which is also the tooth region with the greatest temperature change, coincided with the maximum y direction displacement.

For the hybrid tooth rack measurements in the x direction, although the displacement values for $T_{Heater}=40$ °C were 125 % bigger than the standard tooth, they did not vary appreciably with the temperature rise. As a result, the displacement value for $T_{Heater}=100$ °C was 33.7 % lower than the standard tooth value. There was also an increase in the maximum displacements in the y direction for all temperature increases. The displacement value was 125 % higher than the standard tooth for a temperature of 40 °C, and it fell by 33 % for the other two temperatures.

Hooton reached his final conclusions after comparing the experimental outcomes to the adjusted FEM models. Regarding the validation of the thermal model for the STD, the results demonstrated that the model could predict the surface temperature accurately, with relative errors varied in the range [6.1; 14.8] %. The relative errors were higher in the HT1 example, coming in at [12.5; 29.5] %, while for the HT2 specimen, [12.3; 39.4] % values were found.

For thermo-mechanical model validation, the model was considered accurate in the case of a standard tooth rack because the y displacements along the length of the tooth and the x displacements were both accurate, except for an offset brought on by the rigid body motion of the heaters. Comparably to the last case, an analysis was conducted regarding the hybrid tooth rack. For the same reason as before, the x displacements displayed a similar progression with an offset from the DIC values. Additionally, compared to the FEM results, the maximum displacements recorded in the DIC test were substantially higher. The maximum values obtained in the DIC tests, with an offset of the displacement values for the insert region and tooth tip, were also much higher than the FEM results, even though the y displacement was accurate. Hooton took a final look at several details and then validated the model.

The thermal behaviour of the STD, HT2 (aluminium), and HT3 (epoxy) was compared at the end of the project to conclude that the hybrid teeth with an insert require significantly less power to raise the minimum body temperature. The HT2 required 85.9 % less power than the STD, and the HT3 required 63.8% less power.

2.7.3 Pinto's Dissertation

As a continuation of the work performed by Hooton [2], the primary purpose of Pinto's work [3] was to validate the hybrid gear concept experimentally, but this time trying to approximate even more the experiment conditions to what happens during the gear meshing. In other words, Pinto considered the bending of the gear tooth during the tests. To accomplish that, a new heating system was designed and manufactured to test the hybrid gears. The temperature and displacement data were measured to analyse the thermo-mechanical results. Also, a FEM model was developed according to the conditions of the experiments. In the end, Pinto presented the impact of the hybrid tooth and validated the FEM model.

In Pinto's work, the hybrid gear tooth geometry remained the same as Hooton's geometry, and only a single gear tooth needed to be tested, so a pinion and rack system was used. Some of the heating system specifications were similar to the ones used in Hooton's work. However, some other new requirements were imposed for his work. The system's objectives were to apply heat and pressure to only one tooth flank and guarantee the tooth rack's fixation when load and heat are being applied. Furthermore, it can apply the contact pressure

in different positions of the tooth flank, and when no load is being applied, the system must allow a translation movement of the teeth to readjust its position. The system was also required to apply different pressure values before the beginning of the experimental tests, control temperature values during the tests and monitor temperature values supplied by the heating source using a thermal camera. Figure 2.21 shows the author's final system.

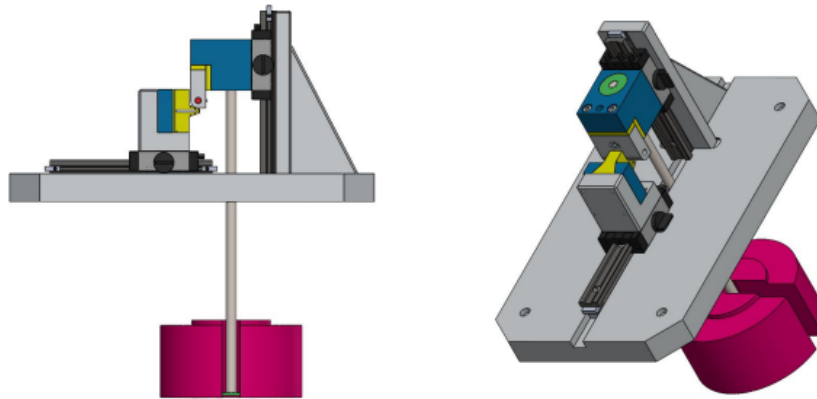


Figure 2.21: Final 3D heating system design: (a) Front view; (b) Isometric view [3].

Pinto first created a FEM model of the teeth to compare and understand the experimental results before moving on to the test. For the purpose of simulating the exact condition of the heating system, a thermal, mechanical and thermo-mechanical models were developed. Thermal tests and digital image correlation (DIC) tests were prepared once the FEM models were defined.

The purpose of the thermal testing was to explore the effects of different insert materials and to validate the hybrid gear concept by heating the tooth flank. In this way, three different types of specimens were analysed, the standard tooth rack (STD), the hybrid tooth rack with no insert (HT1) and the hybrid tooth rack with an aluminium insert (HT2). The main critical points, where there is more interest to measure the temperature evolution, are presented in Figure 2.22.

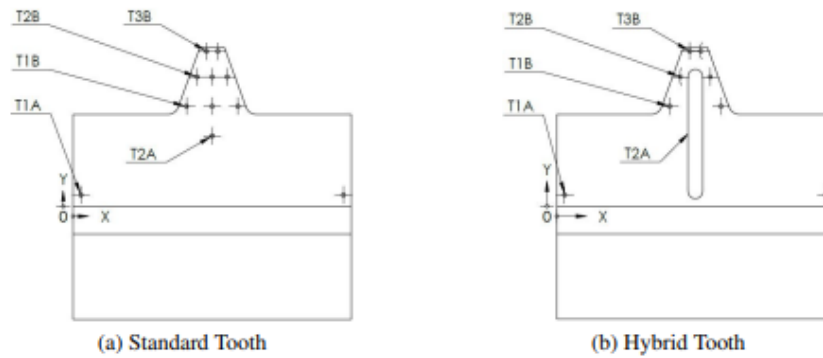


Figure 2.22: Reference points for temperature analysis [3].

All three specimens underwent thermal testing measurements. The relative temperature change for each hybrid tooth compared to the standard tooth was calculated to provide a more detailed analysis of the effects of the hybrid tooth design. For temperature T1a, a zone where convection influences the results, HT1 (air) and HT2 (aluminium) recorded the same temperature change of 18.34 %, and STD presented the lowest temperatures. For point T2a, STD is still the tooth with the lowest temperatures, but HT1 showed a higher temperature change of 20.32 %, followed by HT2 with 16.63 %. The analysis of T1b temperature revealed that HT1 was 9.62 % higher than STD, while HT2 was 8.24 % lower than STD. For the T2b temperature, HT2 remains with the lowest temperature with less 14.90 % than STD and HT1, contrary to what happened in the previous cases, is lower than STD 0.53 %. Finally, T3b showed even more temperature changes of HT1 (less 7.67 %) and HT2 (less 26.67 %) compared to STD.

The same three specimens were tested using the digital image correlation (DIC) method, which was explained in Hooton's dissertation section. The interest here is to analyse the DIC results for the highest thermal expansion stage, so the final moment of the heating period was chosen to study. For the STD specimen, the maximum x displacement was located on the tooth rack's not-supported side. However, the tooth tip at the flank side opposite to the one where the load was applied also exhibits displacements close to the maximum but in the positive direction. The maximum displacement in the y direction was located at the supported tooth rack side. For the HT1 tooth rack and for the x direction, the maximum displacement is situated in the tooth tip apex, in the flank tooth side where the load was not applied, similar to what happened with the STD specimen. In the y displacements, the maximum was located in the opposite tooth flank, in the tooth tip apex. Finally, for the HT2 specimen, the maximum x displacement was situated at the same place as the STD case, so at the top of the rack body, on the not supported side. The thermal expansion effect promotes its highest displacement in the negative direction of the x-axis since physical boundary conditions did not restrict this region. The y maximum displacement point was on the not-supported side, just like the x displacement, but close to the bottom of the rack. This

displacement distribution is also affected by the aluminium insert, which offers resistance to the POM thermal expansion and helps heat evacuation.

Pinto took the final conclusions after comparing the experimental results to the adjusted FEM models. Regarding the validation of the thermo-mechanical model for the STD, the results demonstrated that the model could predict the displacement field accurately, with a relative error of 21.591 % in the x direction and 5.479 % in the y direction. For the HT1, the relative error in the x-axis was lower (11.111 %) and higher in the y-direction (8.139 %) compared with the STD specimen. The HT2 presented the lowest relative error in the x direction (10.753 %), but at the same time, it showed the highest relative error in the y-axis (30.769 %).

In the end, Pinto concluded that the hybrid polymer gear concept was validated as the heating evacuation successfully decreased the temperature in the tooth contact region. Furthermore, the displacements registered are close to the ones of the standard tooth and do not disturb the proper gear operation.

Chapter 3

Manufacturing Processes

3.1 Introduction

Injection moulding is the most common method for manufacturing polymeric gears and works well for large-scale production. The processes of casting or machining are alternatives to this one. The only materials that can be cast are Polyamide (PA 6-G, PA 12-G) or Polyurethane (PUR). Machining is a procedure to consider when a small quantity of parts is desired [11]. As a result, considerations such as the material, the number of gears to be produced, and the required quality should be made when choosing the process.

Starting with casting, it is mostly used for large gears having a mass of at least 1 kg. The polymer is poured into a mould at normal pressure during this process. Due to the shrinkage of polyamide, casting usually occurs with rough teeth that are subsequently machined. In the case of PUR, there is no need for this, as the shrinkage is lower. By centrifugal casting, a better filling of the cavity is achieved, and the shrinkage is reduced [11].

Machining is the manufacturing method more advantageous for small-scale production since the cost of the tool is lower than injection moulds. The procedure is the same as the manufacture of metal gears. In this case, as the cutting forces are low, the required tooth finish is frequently obtained with a single cutting operation [11].

Injection moulding is the most frequent manufacturing process and the most economically viable solution for gears made of thermoplastic. This process allows for a greater

variety in gear design. However, in the circumstances with very demanding requirements for tooth quality, further finishing by machining may be required. It consists of melting thermoplastic pellets and injecting them at high pressure into a mould cavity, within which the melt acquires the shape of the cavity as it solidifies [11].

During cooling, you must consider the contraction of the polymer, which is greater the thicker the component is to be injected. Hence, it is not usual to inject parts thicker than 10 mm. Adding glass fibres lowers the shrinkage and its dependence on the wall thickness. Additional parameters that affect shrinkage are melt temperature, cavity surface temperature, pressure in the cavity, and location and size of the injection point [11].

The gears produced may have different dimensions than those desired, mainly due to the temperature level and homogeneity in the mould cavity. When tolerances are tight, temperature control must be made in the mould, and sometimes a heat treatment (annealing) is carried out [11].

3.2 Mould Used in Injection

A thermoplastic injection mould is a set of steel plates and other components that can create the desired piece when combined in an injection moulding machine. A typical mould is shown in Figure 3.1. It can be separated into two parts: a fixed part where thermoplastic is typically injected and a moving part that moves axially. A separation between the cavity plates and the core insert is created to enable the injection and extraction of the components.

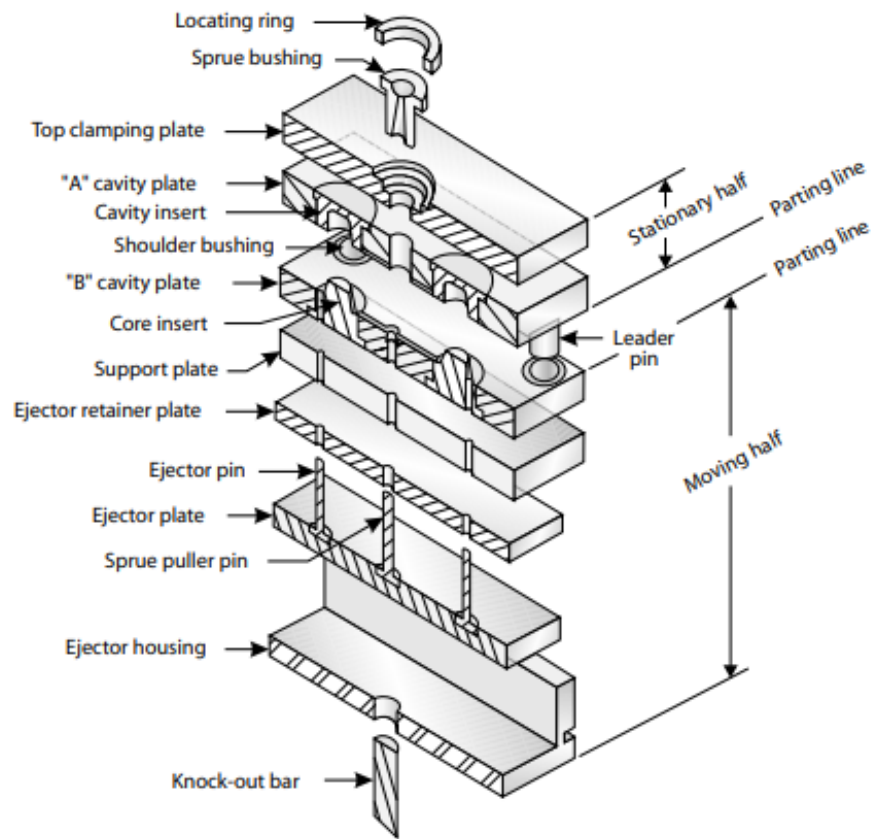


Figure 3.1: Mould constitution [12].

3.2.1 Materials for construction of the mould

The selection of a thermoplastic injection mould material depends on several factors, including such considerations as product design (geometry, part dimensional control, and tolerances), application end-use requirements, thermoplastic material, number of mould cavities, expected production life of the mould, injection moulding process conditions, and mould design and construction. Therefore, should be used steels that are easy to machine, have minimal change in size upon heat treatment, have good polishability, have high resistance to compression and wear and are sufficiently resistant to corrosion [12, 32].

The steels most commonly used in mould construction are shown in Table 3.1, together with their chemical composition.

Table 3.1: Chemical composition, in %, of the most common steels used in moulds. Reprinted from [32].

Designation	DIN	Chemical Composition (%)							
		C	Si	Mn	S	Cr	Mo	Ni	V
Case-hardening steels									
1.2764	X 19 NiCrMo 4	0.18	0.2	0.4	-	1.2	0.2	4.0	-
Heat-treated steels									
1.2311	40 CrMnMo 7	0.4	-	1.5	-	1.9	0.2	-	-
1.2312	40 CrMnMoS 8 6	0.4	-	1.5	0.05	1.9	0.2	-	-
Corrosion-resistant steels									
1.2083	X 42 Cr 13	0.42	-	-	-	13.0	-	-	-
1.2085	X 33 CrS 16	0.33	1.0	1.0	-	16.0	-	-	-
Through-hardening steels									
1.2344		0.40	1.0	-	-	5.3	1.4	-	1.0
1.2379	X 155 CrVMo 12 1	1.55	-	-	-	12.0	0.7	-	1.0
Non-alloy steels									
1.1730	C 45 W	0.45	0.3	0.7	-	-	-	-	-

The 1.2311 steel has excellent dimensional stability, good machinability, the capability to be polished and good wear resistance, being utilised essentially when there are high demands in polishing or when textured surfaces are required [32]. The 1.2312 steel is identical to the previous one, but due to the presence of sulphur, it becomes easier to machine.

Through-hardening steels are distinguished by high hardenability and compressive strength. A significant advantage for these steels is the possibility of carrying out greater machining operations after the heat treatment. The 1.2344 steel is used for moulds, ejector pins and screws subjected to high temperatures. The 1.2379 steel is used for parts subjected to high pressure and for moulds of small dimensions when the polymer tends to cause abrasion [32].

The corrosion-resistant steels (1.2083 and 1.2085) are used for processing chemically active plastic materials [32].

Non-alloy steel 1.1730 is used in clamping plates, ejector plates and supports unless there is cooling on these plates and the corrosion propensity is high.

Effects of Alloying on Tool Steels

The main alloying elements of steels and the modifications they bring to them, are listed in Table 3.2.

Table 3.2: Effects of alloying elements on steels [12].

Element	Resulting effect
Carbon	Carbon is the most influential element in controlling hardness, depth of hardening and strength. Raising the carbon content will increase the hardness.
Manganese	Improves hot working ability by combining with sulphur to form sulfide inclusions.
Molybdenum	Promotes deep hardening and toughness and is used in most of the thermoplastic injection mould steels.
Vanadium	Improves both hardness and the wear properties and is usually added to control grain size.
Aluminium	Combines with nickel to form an intermetallic alloy steel where its hardness is controlled by the cooling rate.
Silicon	The principal function is as a deoxidizing agent during melting to improve the hot forming properties of the steel.
Chromium	Is a carbide forming element with a dual function. When present as carbides, it contributes strongly to hardening ability, abrasion resistance and toughness. When additional chromium is added, this chromium remains in solution and contributes to corrosion resistance. However, high chromium levels can cause hardening deformation problems.
Nickel	Used to improve the toughness and the wear resistance.
Tungsten	Improves the hot hardness. It forms hard, abrasion resistant carbides, improving the wear resistance.
Cobalt	Increases hot hardness. However, substantial addition of cobalt raises the critical quenching temperature with a tendency to increase the decarburization of the surface and the reduction in toughness.

3.2.2 Surface Treatments

The surface treatments aim to obtain a good balance of tensile strength, hardness and wear resistance. Heat treatments, nitriding, and the deposition of carbide and nitride films by physical (PVD) or chemical (CVD) processes are the procedures that are used the most in industry [33].

Nitriding

Nitriding is a thermochemical treatment that provides an extremely high hardness with excellent dimensional stability as the result of a chemical change in the surface layers that considerably improves the wear resistance and fatigue strengths. Nearly all steels commonly encountered in mould making can be nitrided. Nitriding of corrosion-resistant steels is not recommended because of the reduction in corrosion resistance [34].

Physical Vapor Deposition (PVD)

Physical deposition from the vapour state (PVD) allows a significant increase in wear resistance and, consequently, an increase in the life of the mould. Friction reduction is also observed. The coatings are applied in three stages, starting with the transformation of the material to be deposited into a vapour state (in a vacuum chamber). The gas is then transported to the coated part and condensed in consecutive layers until the desired coating thickness is obtained [33].

Chemical vapour deposition (CVD)

Chemical vapour deposition (CVD), nitrides or carbides, is performed at about 1000 °C and has better adhesion, full surface coverage and simpler equipment when compared to the physical deposition process (PVD). In addition, the coating contributes to reducing friction, wear and corrosion. A steel submitted to this thermochemical treatment must present high hardenability and austenitization temperature [35].

3.2.3 Runner System

The runner system attempts to guide the melted material from the injector pin of the machine to the cavities, in a homogeneous way, without loss of temperature and pressure, in the shortest possible time. The runner system can be separated into two large groups: heated channels and cooling channels.

When cooling channels are used, there is a more significant waste of raw material since the material in the runner channels also solidifies (runner), being ejected together with the part. The ejection system can automatically separate the runner and the part when a three-plate mould is used, as illustrated in Figure 3.2.

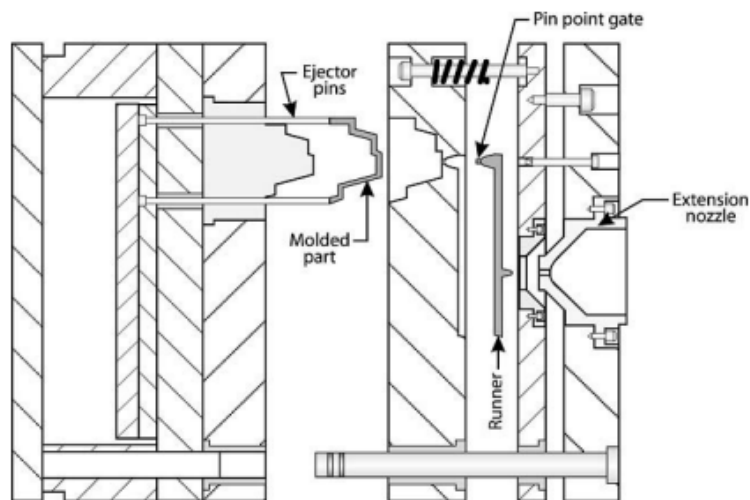


Figure 3.2: Three-plate mould cold runner system [12].

Opting for a heated runner system is like having an extension of the machine inside the mould, in which the channels are covered by resistances, which do not allow the polymer to cool down. Thus, there is no formation of runner. However, although no material is wasted, the initial cost of the mould is much higher, so it is justifiable only for large series.

3.2.4 Types of Mould Cavity Gates

To attain the ideal injection moulding process conditions, the type of gate must be carefully chosen. Gates typically used are a sprue gate, edge gate, tab gate, fan gate, diaphragm gate, external ring gate, internal ring gate, spider gate, film gate, pin point gate, tunnel gate, and runnerless gate systems.

In the case of gears, the optimal type of gate to be applied is the sprue diaphragm gate or the 3 or 4 pin point gates, as shown in Figure 3.3 and Figure 3.4, respectively.

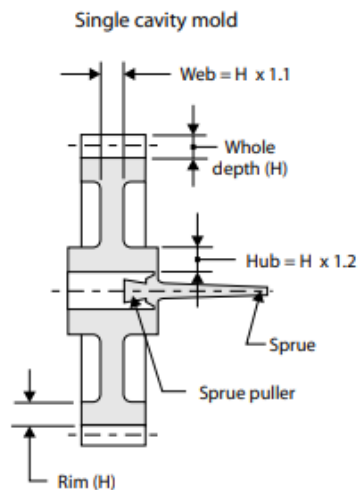


Figure 3.3: Spur gear with sprue diaphragm gate [12].

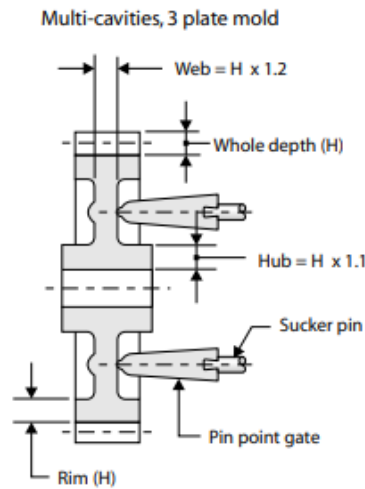


Figure 3.4: Spur gear with 3 or 4 pin point gates [12].

To obtain high accuracy, the injection of material should be conducted as close to the centre as possible. Figure 3.5 shows the gear elongation and compression problems caused by an edge gate located on the spur gear tooth. When the core of the gear hub splits the melt flow, both melt fronts, which are the coldest part of the melt, are reunited at the back side of the core, causing a weld line area that is the weakest section of the gear. The flow direction of the melt has a higher injection pressure, causing less mould shrinkage (larger dimensions). The pressure perpendicular to the flow direction is lower, causing higher mould shrinkage (smaller dimensions) [12].

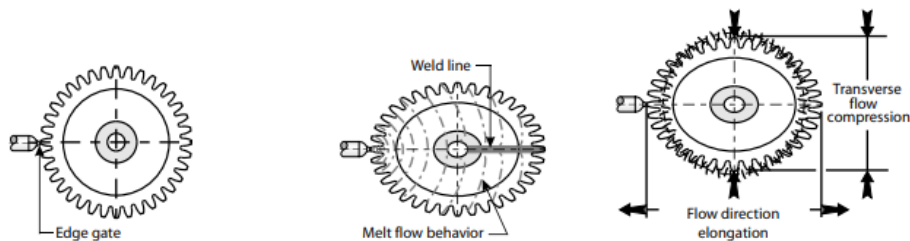


Figure 3.5: Single edge gate on the spur gear tooth width [12].

As the injection point is moved towards the centre, the issues outlined above continue, but they are more and more attenuated, achieving better results. This means that the lower resistance section is less evident, and the contraction is smaller.

The sprue diaphragm gate in the middle of the spur gear allows uniform melt filling, equal pressure, shrinkage, and melt flow distribution inside the cavity. It also allows the best precision dimensional control without weld lines. The diaphragm gate disk connected to the sprue requires a post-moulding operation to remove the diaphragm gate disk and sprue from the inside diameter of the hub [12].

3.2.5 Mould Temperature Control

Depending on the type of polymer to be processed, heat must be introduced to or removed from the injection mould. The mould temperature control system handles this task. A great deal of importance should be given to optimum temperature control. It directly influences the moulded parts' functionality, such as the level of moulded-in stresses in the part and its susceptibility to failure, the warping of the moulded parts, the cooling time and thus also the cycle time [34].

When heat needs to be evacuated from the mould, a refrigerant fluid, usually water, is injected through channels machined in the mould. The cooling channels must not be placed too close to any other hole in the same mould plate. The mould plate contains many cavities to fit ejector pins, support pillars, guide bushings, sprue bushing, cavity and core inserts. Thus, to acquire the greatest possible position for a mould cooling circuit, it is recommended to lay it in at the earliest opportunity in the mould design. The additional mould components can then be positioned accordingly [12].

3.2.6 Ejection System

The injection moulded parts are subjected to size reduction during the cooling cycle, induced by the mould shrinkage characteristics of the thermoplastic material. If parts with accurate dimensions are needed, allowances must be made for this shrinkage when setting the dimensions for the cavities. The moulded parts are ejected from the mould cavities after they have cooled down. The ejection systems automatically require special mould manufacturing methods to remove the runner system and the moulded parts from the mould [12].

The ejection of a spur gear is a straight, non-rotating movement in the direction of the gear axis. To eject a helical gear, the gear must rotate slightly as the gear moves out of the mould to accommodate the angled gear teeth. In order to allow the helical gear to rotate as it is ejected, the gear must be slowly ejected from the mould than the ejection speed used for spur gears [36].

If the helix's angle and tooth's width are small when a helical gear made of rigid material is removed, the rotation at the time of ejection can only be forced by the axial motion of the ejectors. To prevent the deformation of the piece or its bending, the cross-sectional area of the extractor rods should be large because this way of extraction involves high forces [37].

To solve the problem of the ejection of double helical gears, an assembly gap is required between the helically geared inserts and the plate and between these same inserts and the movable mould insert. The purpose of these gaps is to reduce the friction between the

surfaces, facilitating the rotation and extraction of the part. The stopper prevents the cavity core from moving excessively in the axial direction when the part is extracted [37].

The kind of material to be utilised must consider the required clearances. For example, POM allows the use of larger clearances than utilising polyamide [38].

3.3 Influence of Gear Geometry in its Resistance

The fillet area experiences the highest bending moments when a load is applied to a gear tooth. Stress concentration around relatively sharp fillet corners increases the resulting bending stress. As a result, the maximum local stress is reduced, and the tooth's capacity to withstand the applied load increases, the less sharp the fillet is. Moreover, abrupt transitions, such as flow at sharp corners or a quick change in velocity, can result in flow irregularities close to the part's surface (figure 3.6), which cause moulded-in stresses that weaken tooth strength [39].

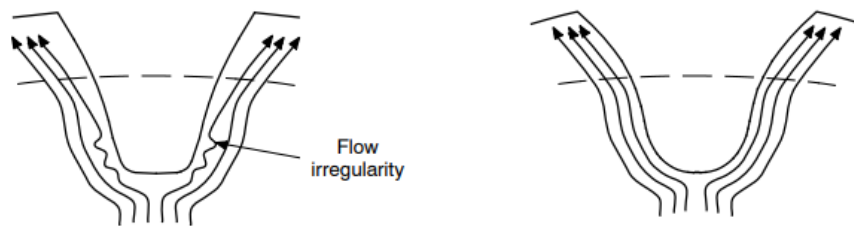


Figure 3.6: Effect of tooth's fillet shape on mould flow [39].

An additional issue from this kind of mould arises with fibre-reinforced plastics. Figure 3.7 shows how the fibres typically arrange themselves in a radial pattern heading towards the corner when the flow encounters a relatively sharp corner. As a result, the fibres do not completely reinforce the critical corner, potentially opening up failure paths. Moreover, improperly oriented fibres might result in non-uniform shrinkage and poor tooth geometry [39].



Figure 3.7: Effect of tooth's fillet shape on fiber orientation [39].

Fundamental to moulding stable, stress-free parts are designs that encourage uniform mould cooling. Figure 3.8 illustrates how sharp corners, such as those at minimal radius root fillets, produce a zone with a constrained path for heat conduction. The result is a spot still

heated long after the adjacent material has cooled. A comparable heat conduction issue that increases the temperature in the mould's equivalent sharp exterior corner exacerbates this difficulty further. The plastic experiences non-uniform cooling rates that cause shrinkage forces, resulting in voids or high localised stresses close to the tooth root [39].

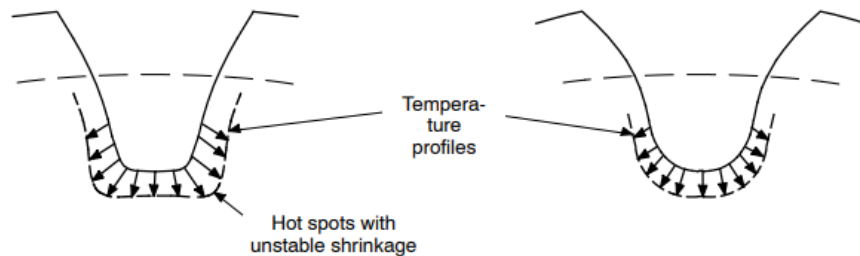


Figure 3.8: Effect of tooth's fillet shape on fillet surface temperature during cooling [39].

Gears with a small number of teeth and an inadequate basic rack offset frequently exhibit a different type of moulding issue related to an undercut tooth profile. The fillet's sharp corner draws attention to the undercut state. During shrinking in the mould, the undercut may cause "hang-up" (Figure 3.9) on the corresponding projections in the mould cavity, limiting the amount of free shrinkage in the combined radial and circumferential directions [39].

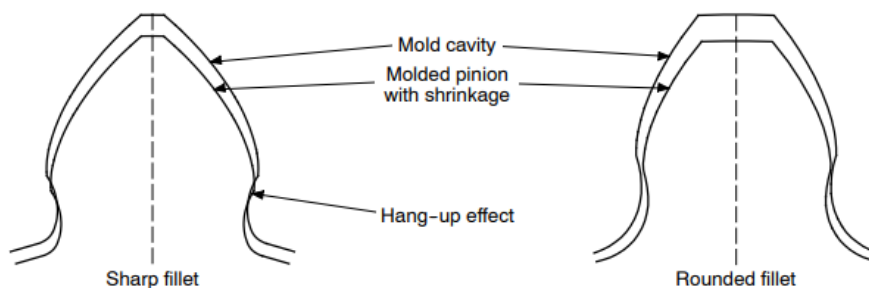


Figure 3.9: Comparison of shrinkage effect in undercut pinion with sharp and rounded fillets [39].

When the gear's wall thickness is too thin (Figure 3.10), the injection process causes it to solidify too quickly. Thus, to prevent the teeth from shrinking, providing them with more material is impossible. As a result, voids are formed that worsen shrinkage, leading to poor dimensional control (roundness), and the middle of the tooth face width becomes concave [12].

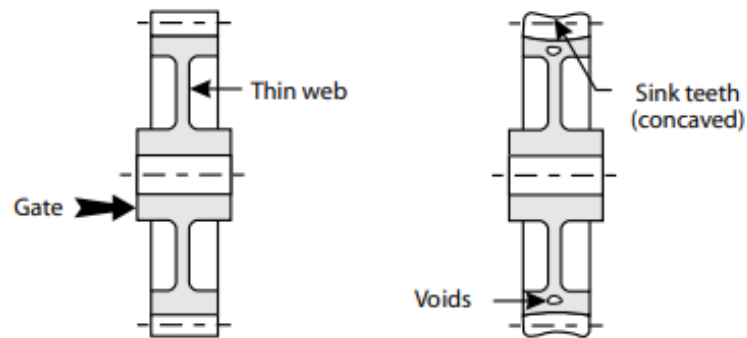


Figure 3.10: Influence of a thin wall thickness web [12].

Furthermore, the recesses should be positioned as symmetrically as possible to prevent warping, as shown in Figure 3.11.

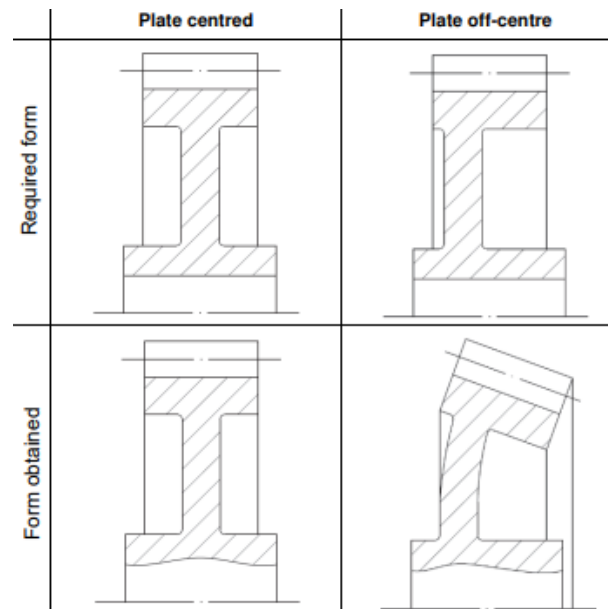


Figure 3.11: Influence of symmetry on the final product [11].

3.4 Milling Process

Milling is a frequently used process to remove materials from the initial configuration of a workpiece for machining parts in manufacturing industries. Given the characteristics of large size and poor rigidity, cutting deformations and chatter vibrations would be easily induced during the cutting process. As a result, it is challenging to attain workpiece surface quality and machining accuracy. In order to create ways to ensure precision, study into the mechanics and dynamics of the milling process is crucial. The key to this feature lies in issues like clamping scheme design, chatter stability analysis, surface quality prediction, and cutting force modelling [40].

The shearing action between the cutting edge and the workpiece, the friction effect between the rake face and chip, and the ploughing effect between the clearance surface and the machined surface are all components of the milling process' cutting mechanics. The three effects work together to generate cutting forces that might cause cutting deflections and further deteriorate the surface quality of machined components. To control surface imperfections and create a consistent cutting strategy, a deep understanding of the mechanics of the milling process is essential. The cutting mechanism of milling, a typical multipoint tool operation, is based on elastic-plastic mechanics and thermal dynamics [40].

Cutting forces induce deflections that result in surface form defects that significantly reduce the precision and quality of the workpiece. The milling process will produce waste products if the form errors violate the dimensional tolerance. Thus, reliable machining technologies must be used to achieve consistent part shapes and the required machining accuracy. Numerical simulations that combine the finite element method and cutting mechanics can effectively accomplish this. In addition, cutting parameter optimization can enhance the cutting process [40].

3.4.1 Types of Milling Tools

High-speed steel is typically used to make milling cutters, which come in various sizes and forms for diverse applications. To give the part the desired shape, a range of operations can be carried out on the workpiece during the manufacturing cycle. The type of milling cutter being used and the cutter's path as it removes material from the workpiece define each of the following operations.

1. **End milling** - The end milling cutter has teeth on the end as well as the periphery. A specific feature, such as a profile, a groove, a pocket, or even a complicated surface contour, can be machined using an end mill by making cuts over the workpiece that are either peripheral or groove-based, depending on the infeed distance. The depth of the feature can be machined in a single pass, or it can be machined in multiple passes using a reduced axial depth of cut. In figure 3.12 is shown an end mill cutter [41, 42].

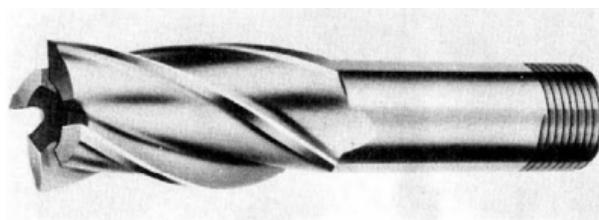


Figure 3.12: End mill cutter [43].

2. **Chamfer milling** - A chamfer cutter performs a peripheral cut along the edge of the workpiece or feature to generate an angled surface known as a chamfer. This chamfer,

commonly machined at an angle of 45° to 60° , can be straight or curved and machined on either the interior or exterior of a part [41].

3. **Side milling** - Plain milling cutters with teeth on one or both sides are essentially side milling cutters. A plain side milling cutter has teeth on both sides and on the periphery. The cutter is a half-side milling cutter and can be either a right-hand or left-hand cutter when teeth are added to one side only. Slotting and straddle milling are often accomplished with side milling cutters [42].
4. **Concave and Convex milling** - Concave and convex contours of half a circle or less are produced by teeth cutters that are formed into concave and convex milling cutters. The size of the cutter is determined by the diameter of the circular form the cutter generates [42].
5. **Corner Rounding Milling** - A formed tooth cutter called a corner-rounding milling cutter is used to mill rounded corners on work surfaces up to and including quarter circles. The size of the cutter is specified by the radius of the circular form the cutter produces, such as concave and convex cutters typically used for such operations as finishing spur gears, spiral gears, and worm wheels. Ratchet and spline shafts can also be cut with them [42].

3.4.2 Types of Milling Machines

Several cutters can be held in a milling machine at once. It is the most significant piece of equipment in a given industry and can carry out tasks with a high degree of accuracy. These machines are essential for industrial work because of their superior precision and improved surface finishing. Some of the main types of milling machines are exposed in this subsection.

1. **Horizontal or Plain Milling Machine** - Plain milling machines can feed the table in a vertical, cross, or horizontal direction and are more robust than manual millers. In other words, horizontal milling machines install the cutting tool on a spindle that is horizontally oriented and capable of removing material from the stationary work-piece with precision. Also, horizontal milling machines often use different cutting tools compared to vertical milling machines. The spindle's direction is the key distinction between these two kinds of milling machines. The spindle on vertical milling machines has a vertical orientation. The spindle on horizontal milling machines has a horizontal orientation [41].
2. **Vertical Milling Machine** - The spindle is either vertical or perpendicular to the table on a vertical milling machine. This machine can be used to machine flat surfaces, grooves, and slots. The spindle head is fixed to a vertical column, which rotates at an angle. The spindle of some vertical milling machines can be moved up and

down. Cutting possibilities are more sophisticated and flexible than a drill press by virtue of a moving table and vice (x and y-axes) and vertical spindle movement (z-axis). Many vertical mills also incorporate a rotating turret for the upper cutting head, which allows even wider milling options (b-axis) [41].

3. **Universal Milling Machine** - A universal milling machine can conduct every type of milling operation since it has a table with all motions and a division head with change gears. From its normal position, the table can pivot at any angle up to 45 degrees on both sides. The horizontal milling machine table has a fourth movement in addition to its three other directions [41].
4. **CNC Milling Machine** - The machining technique known as CNC milling, or computer numerical control milling, progressively removes material from the workpiece to create a specific part or product tailored to the customer's specifications. A CNC Milling Machine uses a rotating cylindrical cutter to move along multiple axes and create slots, holes, and details in material to turn it into a mechanical part. Most machines use three to five axes, which results in far more precision and detail. Both horizontal and vertical machining centres are incorporated into modern CNC milling machines. In addition to using metals, components can also be produced from materials like plastic, ceramic, and composites [41].

3.4.3 High Speed Milling

High-speed milling has become a well-established technology in the modern manufacturing industry thanks to the development of spindles that could rotate much faster while maintaining a relatively decent rigidity. The literature uses different definitions to explain the difference between high-speed milling and conventional milling. As the spindle's dimensions are not specified, the absolute spindle speed that is sometimes employed may not be evident. The following definition calculates the product of the mean bearing diameter D_m in millimetres and the spindle speed n in revolutions per minute. It is considered high-speed when the product $D_m \cdot n$ surpasses $0.5 \cdot 10^6 \text{ mm min}^{-1}$ in a milling process. A third definition concerns the process stability and the dynamics of the tool and spindle. A milling operation is deemed high-speed when the tooth passing frequency reaches a significant fraction of the dominant (most flexible) natural frequency of the spindle-tool dynamics [44].

In any case, monolithic aluminium structural parts for the aerospace sector are created using high-speed milling, where more than 90% of the original workpiece is machined. The production of moulds, dies, and components for the automotive industry are further applications. Due to the tool's relatively fast rotational speed, high-speed milling is particularly useful for producing mechanical parts, especially in the latter two applications where long, slender tools are frequently utilised. With this, a relatively small radial depth of cuts/feed

per tooth can be used, leading to small cutting pressures without affecting the rate of material removal, which is advantageous when milling thin-walled structures [44].

3.5 Electrical Discharge Machining (EDM)

Electrical Discharge Machining (EDM) is a well-known machining technique for producing objects made of hard materials or with complex geometries that are extremely challenging to manufacture using traditional machining methods.

A sequence of discrete electrical discharges between the electrode and the workpiece submerged in a dielectric fluid serve as the primary source of electrical energy for the material erosion mechanism, which converts it into thermal energy.

One of the most widely utilised non-conventional material removal procedures is electrical discharge machining (EDM). Its particular advantage in the production of mould, die, automotive, aerospace, and surgical components has been its special ability to use thermal energy to produce electrically conductive parts regardless of hardness.

Additionally, since the electrode and the workpiece are not in direct contact when using EDM, there are no mechanical stresses, chatter, or vibration issues during machining [45].

Within the EDM process, it can be wire cut EDM or sinker EDM, being the methodology of successive electrical discharges similar in both processes. An illustration of both processes is given in Figure 3.13.

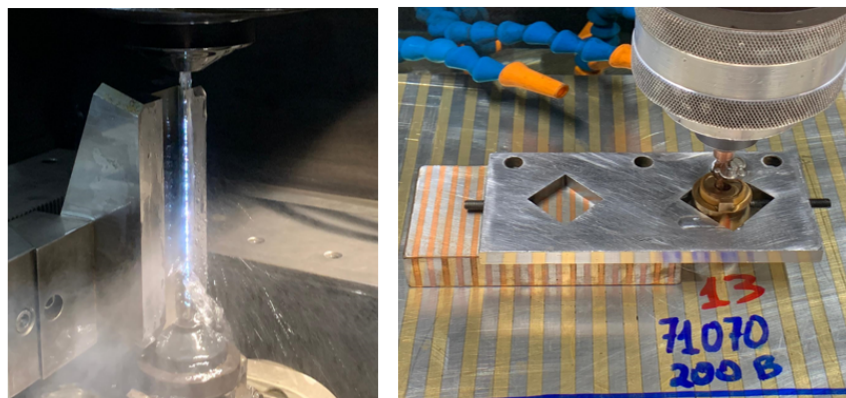


Figure 3.13: Electrical discharge machining by wire cut and sinker, respectively.

3.6 Grinding Process

Grinding is a conventional machining process that uses abrasive particles as the cutting tool. This process is used to remove surface imperfections and enhance the precision of parts machined using other methods or to verify complete accuracy before moving on to another procedure. The cutting tool's material may change depending on the intended use, finish, and other factors. This stage involves significant cutting forces and an intense heat flow to the workpiece. When the cutting variables are not properly chosen, most of the heat is delivered to the workpiece, creating damage that undermines the surface integrity. In addition to these issues, such as poor surface polish, it can also result in microstructural changes that cause the surface to oxidise and lose hardness. Figure 3.14 shows the basic principles of this process.

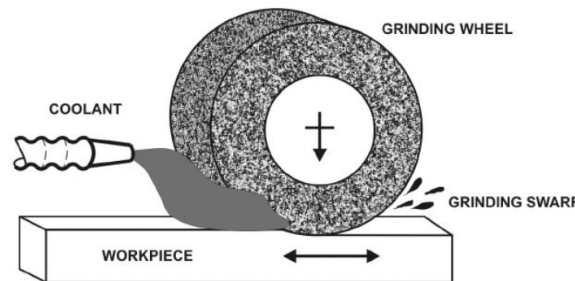


Figure 3.14: Elements of the grinding process [46].

3.7 Selective Laser Melting (SLM)

Selective Laser Melting (SLM) is an additive manufacturing technology that has revolutionized the field of metal 3D printing. It is a precise and versatile method that allows for the creation of complex and intricate metal parts with exceptional accuracy and quality.

At its core, SLM involves the layer-by-layer fusion of metal powder using a high-powered laser. The process begins with a thin layer of metal powder spread evenly across a build platform. The laser then selectively melts and fuses the powder particles according to a digital design, creating solid layers of the desired object [47]. In Figure 3.15 it is possible to observe the working principle of the SLM technique.

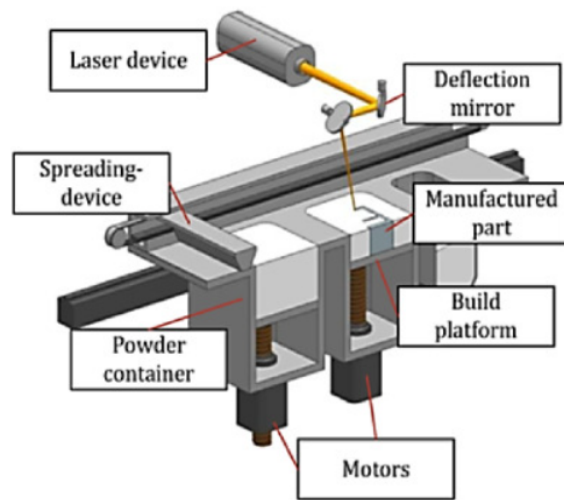


Figure 3.15: Working method of SLM technology [47].

One of the key advantages of SLM is its ability to produce highly intricate geometries that would be challenging or impossible to manufacture using traditional methods. Complex internal structures, hollow features, and intricate lattice designs can be effortlessly created, opening up new possibilities for design and engineering.

Moreover, SLM offers a wide range of compatible materials, including various alloys, such as stainless steel, titanium, aluminium, nickel-based superalloys, and more. This versatility allows for the production of parts with tailored mechanical properties, such as high strength, corrosion resistance, or lightweight characteristics, depending on the application requirements [47].

SLM also enables the production of fully functional parts in a single manufacturing step. The resulting parts exhibit excellent surface finish, dimensional accuracy, and mechanical integrity with proper optimisation and post-processing techniques.

Industries such as aerospace, automotive, healthcare, and tooling have embraced selective laser melting for numerous advantages. It facilitates rapid prototyping, on-demand production of spare parts, and the creation of complex components for specialized applications. The ability to manufacture lightweight and optimized structures has particularly benefited industries seeking to reduce weight and improve performance, such as aerospace [47].

However, it's important to note that SLM is a complex process that requires expertise and careful control of various parameters, including laser power, scanning speed, powder properties, and thermal management. Additionally, post-processing steps, such as removing

support structures and heat treatment, may be necessary to achieve the desired final properties [48]. The various steps that the SLM process has to pass through are represented in Figure 3.16.

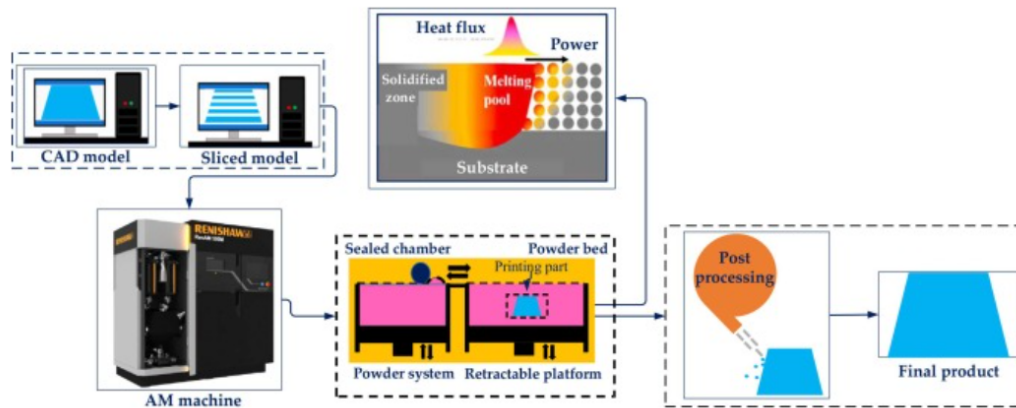


Figure 3.16: SLM process methodology [48].

Selective Laser Melting has emerged as a game-changing technology, pushing the boundaries of what is possible in metal fabrication. With its ability to create intricate designs, functional parts, and a wide range of metal alloys, SLM has paved the way for innovation, customization, and increased efficiency in multiple industries, leading to exciting advancements in the manufacturing world [47].

Chapter 4

Definition of the Hybrid Gear Geometry and Mould

4.1 Introduction

Based on the findings of the previous works [1, 2, 3], it was possible to arrive at a new potential solution for the hybrid gear. The material for the polymer gear was based on the VDI 2736 specifications [11]. Furthermore, the choice of the material for the insert considered the findings of the previous FEM analysis and the availability of Celoplás' suppliers in the market.

In this chapter, besides the geometry and materials that will be used in the production of the gear and the insert, the mould development process that will be used to inject the hybrid gears will also be explained. In the end, a filling mould simulation will be performed to guarantee the best injection performance possible.

4.2 Development of the hybrid gear geometry

The design and geometry of the standard gear were inspired by Mao *et al.* [49]. Mao gears, in addition to having a keyway in the pinion bore that allows them to apply more significant torsional moments, also have a recess on their faces to ensure better dimensional stability during injection moulding. This geometry was used because there are many load-carrying capacity and wear results that serve as a benchmark. Additionally, the dimensions

are compatible with the restrictions imposed by Celoplás. The developed gears will have two solutions, one without insert and other with insert. The gear geometry is represented in Figure 4.1.

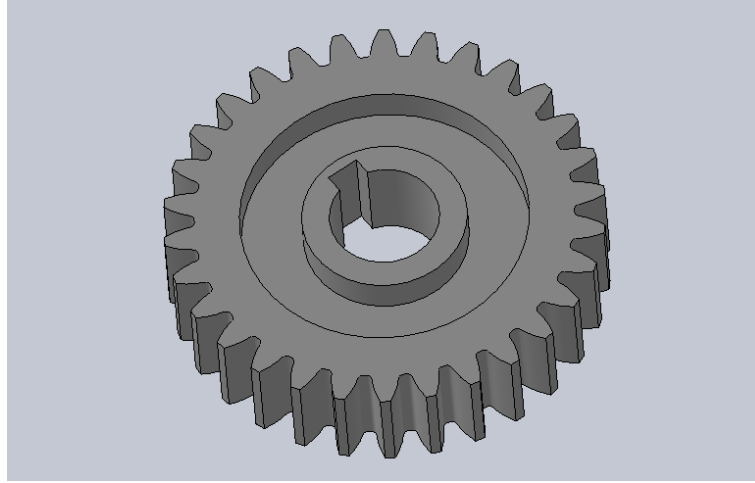


Figure 4.1: Designed geometry of the gear.

Gears are commonly designed to work in pairs to achieve efficient and reliable motion transmission. The gear pair considered for the hybrid gear geometry comprises gears z_1 and z_2 , both spur gears with the same geometry and their characteristics are presented in Table 4.1. The detailed drawing of the gear wheel can be found in Appendix B.

Table 4.1: Main characteristics of the gears.

Parameter	z_1/z_2
Number of teeth, z	30
Normal module, m_n / mm	2
Pressure angle, α_n / °	20
Helix angle at reference circle, β / °	0
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
Manufacturing tolerances	± 0.30

One of this study's objectives was to investigate different polymeric materials for gear applications. Considering all the research conducted on this topic, POM (Polyoxymethylene) was used, as it has already shown promising results in gear applications with several advantages. The specific type of POM used was a copolymer, POM-C 13021 HOSTAFORM from Celanese.

Since the main purpose of these hybrid gears is to increase the low thermal conductivity of polymers, research was conducted on commercially available conductive polymeric materials. Consequently, a thermally conductive material was chosen, which will also be injected into the designed mould and, subsequently, serve as a comparison with the parts injected in POM. The selected material was the LNP KONDUIT Compound PX10323 from Sabic, which is based on Nylon 6 resin containing glass fiber and proprietary thermal filler.

The mechanical and thermal properties of POM-C and PA6+GF provided by the respective suppliers can be found in Table 4.2. The respective datasheet of each material can be found in Appendix A.

Table 4.2: Mechanical and thermal properties of POM-C 13021 HOSTAFORM and LNP KONDUIT Compound PX10323.

	POM-C 13021	PA6-GF
Density, ρ / kg m^{-3}	1410	1610
Young's Modulus, E / GPa	2.9	14.0
Thermal conductivity coef., κ / $\text{W m}^{-1} \text{K}^{-1}$	0.155	18
Melting temperature, T_{inj} / $^{\circ}\text{C}$	210	285
Temperature of deflection under load, T_{hdt} / $^{\circ}\text{C}$	106	201
Ejection temperature, T_{ext} / $^{\circ}\text{C}$	140	260
Moulding shrinkage, %	2.0	0.2

4.3 Development of the insert geometry

One of the objectives of this dissertation is to compare the impact and viability of different designs for the insert. Therefore, two scenarios for the insert were projected, being each insert produced by distinct manufacturing processes. In addition, the material used for each insert was also different.

The first geometry developed was based on previous works, particularly Rocha and Moutinho's works [1, 5], taking into account the influence of the plate width (w), the lateral gap (e), and the tip gap (t) on the insert shape. However, to produce these gears by injection moulding, it was necessary to arrive at an entirely new geometry that had never been tested.

The shape of the insert underwent several modifications to increase the contact area or position it closer to the tooth contact zone in order to improve thermal evacuation until reaching the final design, represented in Figure 4.2. The detailed drawing of the insert is present in Appendix B.

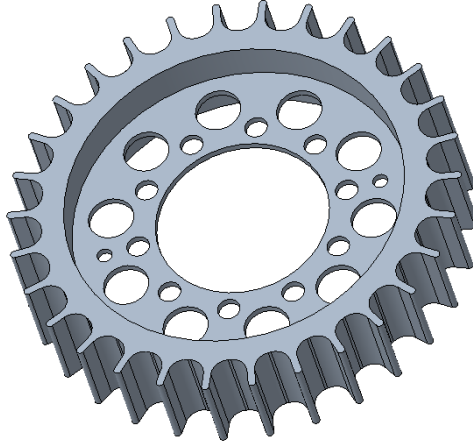


Figure 4.2: First insert geometry.

As mentioned before, in both works [1, 5], steel, copper and aluminum inserts were compared using FEM analysis and the aluminum insert was chosen as the best option. The material for the aluminum insert was based on previous works [2, 3]. However, due to the supplier availability, it was chosen an aluminum alloy within the 6000 series with similar characteristics to the one used before. The aluminum alloy used was Aluminium 6082 and its properties can be found in Table 4.3.

Table 4.3: Mechanical and thermal properties of aluminum 6082.

Aluminum 6082 alloy	
Density, ρ / kg m^{-3}	2700
Young's Modulus, E / GPa	70.0
Poisson's ratio, ν	0.33
Thermal conductivity, κ / $\text{W m}^{-1} \text{K}^{-1}$	170-220
Thermal expansion coeff, α / $^{\circ}\text{C}^{-1}$	23.4E-06

This first insert design was produced in Celoplás by machining, and the complete manufacturing process is explained, in more detail, in the next chapter.

The second geometry is based on the first one and is very similar to it, except for some holes made in the teeth and others in the thicker wall of the insert, in the area before the start

of the teeth. These holes aim to ensure that the polymer traps the insert in a zone that can be critical during the injection. Figure 4.3 shows the final design of this second geometry.

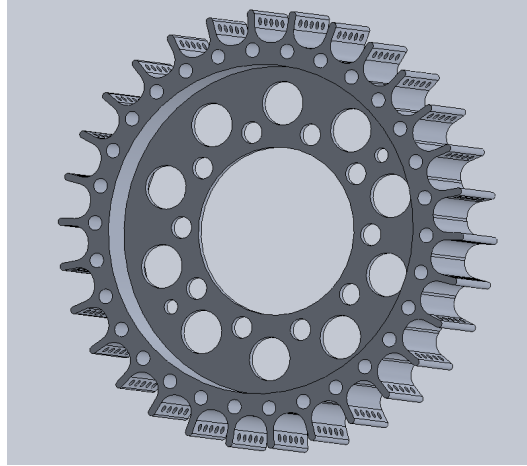


Figure 4.3: Second insert geometry.

As this second geometry is much more complex to produce using conventional manufacturing processes, a Selective Laser Melting additive manufacturing technique was used to obtain these inserts. The material used was AlSi10Mg, which is an aluminium alloy that is commonly used in additive manufacturing (3D printing) processes, particularly in metal additive manufacturing. It exhibits good mechanical and thermal properties. In Table 4.4 are some of the key mechanical and thermal properties of AlSi10Mg.

Table 4.4: Mechanical and thermal properties of AlSi10Mg.

AlSi10Mg	
Density, ρ / kg m^{-3}	2600
Young's Modulus, E / GPa	72.5
Poisson's ratio, ν	0.32
Thermal conductivity, κ / $\text{W m}^{-1} \text{K}^{-1}$	130-160
Thermal expansion coeff, α / $^{\circ}\text{C}^{-1}$	23E-06

4.4 Development of the mould

The design of a mould is a constant optimisation process, which aims to develop a solution that is as efficient as possible, reducing as far as possible material waste and cycle times. Furthermore, it must guarantee the reliability of the process and the quality of the final product, promoting a robust process.

The mould on which the development of this dissertation is based is an existing mould from Celoplás company. To reduce waste and avoid costs in the development of the prototype, changes were then designed to an existing structure so that it was possible to modify it for the new purpose.

With the mould disassembled, it is possible to start its 3D modelling and subsequently design the necessary alterations to manufacture the hybrid gear solution. The mould was designed in Creo 7.0 software. The structure of the designed mould and the legend of the respective plates are shown in Figure 4.4. All the 2D drawings made for the mould project are located in Appendix B.

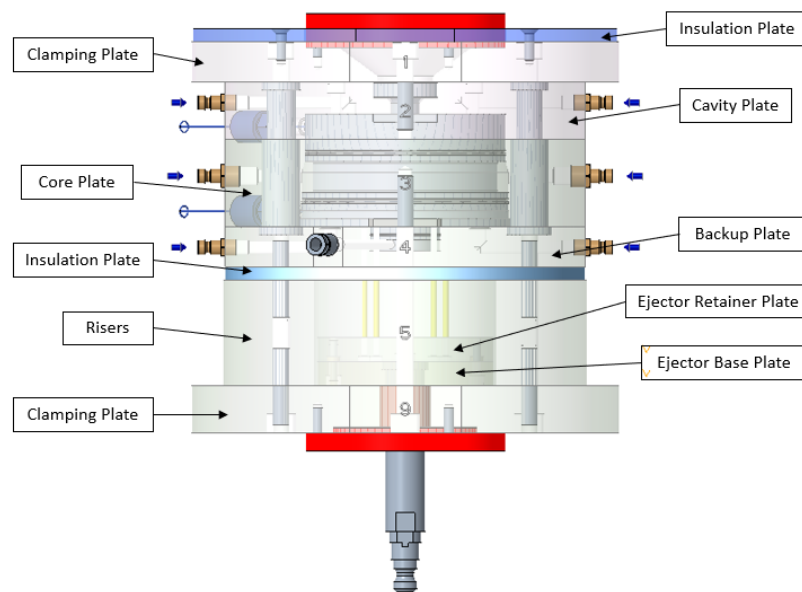


Figure 4.4: Structure of the designed mould.

4.4.1 Cavity Design

The design of the mould cavity is determined by various factors, including the shape and size of the intended product, the material being used, the production volume, and the manufacturing process.

The cavity design should ensure proper material flow, uniform filling, adequate cooling, and easy release of the final product from the mould. It should also consider factors such as minimizing shrinkage, warping, and defects in the finished part.

When gears are manufactured by injection moulding, they contract during cooling. Consequently, the mould cavity must be larger than the gear wheel, compensating for this contraction.

The greatest influence on the shrinkage comes from the temperature of the mould surface and from the wall thickness of the moulded part [50]. Figure 4.5 shows this dependence for POM with three different wall thicknesses.

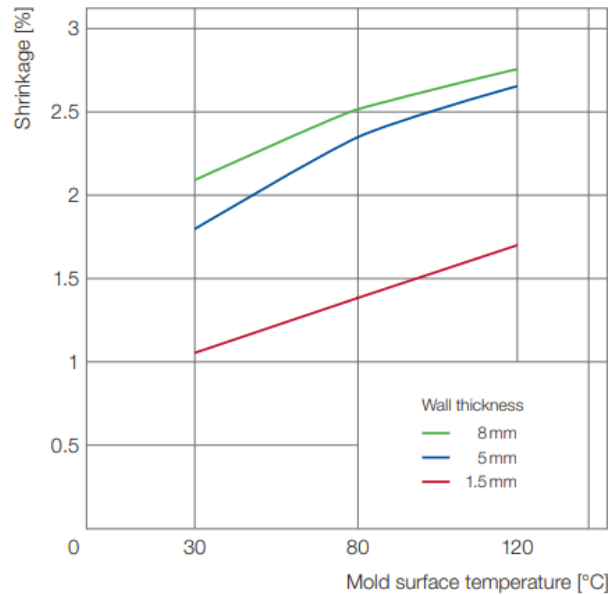


Figure 4.5: Shrinkage of POM as function of mould surface temperature and wall thickness [50].

Since the mould has a single cavity and two different materials, POM and PA-GF, with significantly different shrinkage rates, it was necessary to choose a single value for shrinkage. Since POM was the main material for this project and the wall thickness of the plastic material varies during the injection process, it was decided to use the shrinkage value presented on the material datasheet, which is 2 % in both radial and tangential directions, as listed in Table 2.1. The parts injected with the other material, which has a shrinkage rate of only 0.2 %, will have slightly larger dimensions than intended. However, during the tests, it is possible to correct the center distance of these gears.

The gear teeth in the core inserts will have to expand radially and tangentially to counterbalance the shrinkage effect. The radial expansion causes the outer contour of the tooth to move away from its centre point. This expansion is caused by the material's contraction properties during the cooling phase of the injection moulding process. On the other hand, the tangential expansion causes an increase in tooth thickness. As the material cools and solidifies, it tends to shrink in the tangential direction, causing a decrease in the tooth's thickness. After providing the required radial and tangential expansion values, in this case a 2% expansion, Creo determines the shape of the cavity.

As expected, the moulding cavity had to be made from scratch, as this defines the part to be produced. Therefore, CREO manufacturing was used to create the core and cavity inserts. The designed part served as the basis for creating the inserts. Then, a parting surface was made, defining the boundary separating the mould's core and cavity halves. This surface determines how the mould will split and where the inserts will be placed. In Figure 4.6, it is possible to observe the parting surface created to separate the mould and the inserts of the fixed and movable sides.

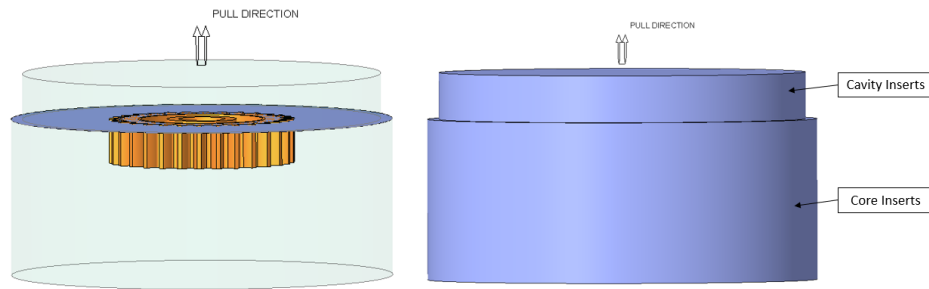


Figure 4.6: Parting Surface of the mould (left); Core and cavity halves of the mould's inserts (right).

In the core half of the split part, the areas that required core inserts were identified to create the desired shape of the moulded part. These areas may include holes, undercuts, or complex features that a simple mould cavity cannot easily form. In this way, more parting surfaces were created, and separate parts for each core insert were developed. Similarly, the process to generate the different parts for each cavity insert is the same for the cavity inserts.

After that, the necessary adjustments to the inserts were made to ensure proper clearance, draft angles, and alignment between the two halves. This involved modifying the shape or features of the inserts. Thus, Figure 4.7 shows the final result for the moulding cavity.

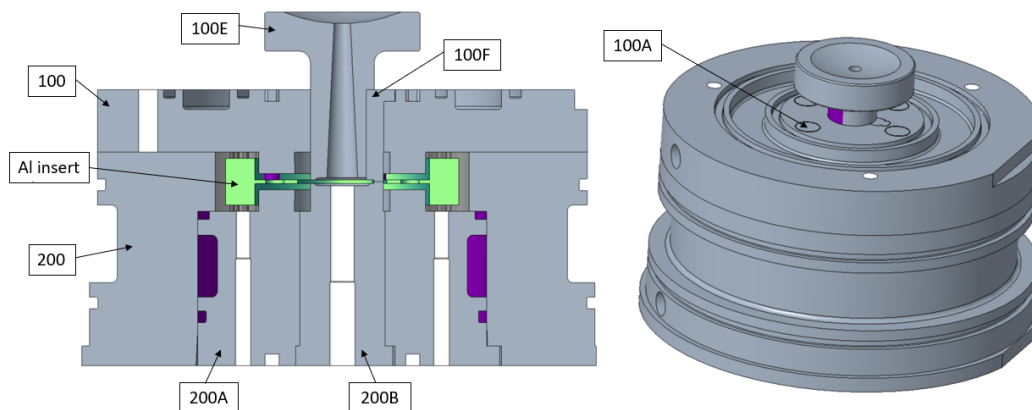


Figure 4.7: Moulding Cavity: (a) Cross section; (b) Isometric view.

In order to also study whether the type of material of the inserts influences the results and the temperatures reached during injection, a copper alloy (AMPCO 83) was used in addition to 1.2311 steel. The inserts tested in these two materials were 200A, 200B and 100 due to their direct contact with the material during the injection. In addition, AMPCO is a very resistant material that facilitates the cooling of the moulding zone compared to steel. In Table 4.5, a comparison of the mechanical and thermal properties of these two materials can be observed.

Table 4.5: Mechanical and thermal properties of 1.2311 steel and AMPCO 83.

	1.2311 steel	AMPCO 83
Density, ρ / kg m^{-3}	7850	8260
Young's Modulus, E / GPa	210	131
Poisson's ratio, ν	0.30	0.36
Thermal conductivity, κ / $\text{W m}^{-1} \text{K}^{-1}$	35	130
Specific heat capacity, c / $\text{J kg}^{-1} \text{K}^{-1}$	422	380
Thermal expansion coeff, α / $^{\circ}\text{C}^{-1}$	11.1E-06	17.5E-06
Tensile strength / MPa	920	1250

One of the significant challenges in the design of the inserts and even in the mould design was the placement and fixation of the aluminium insert. For the hybrid gear to ensure its alignment and orientation during injection, two pins were placed in the 2 mm holes of the insert, as shown in Figure 4.8 a). However, this would not be enough to ensure the insert would not come loose during mould closure before injecting plastic. Therefore, a solution had to be considered, consisting of a vacuum system through 4 tubular ejector pins running from the support plate to the surface of the insert (Figure 4.8 b)). On the other surface of the aluminium insert up to the surface of insert 100, 4 steel inserts (100A) are placed to prevent any possible rotation of the aluminium insert after it has been placed in the mould, thus allowing the vacuum system to be used correctly. This system will be discussed in more detail in the following sub-chapter.

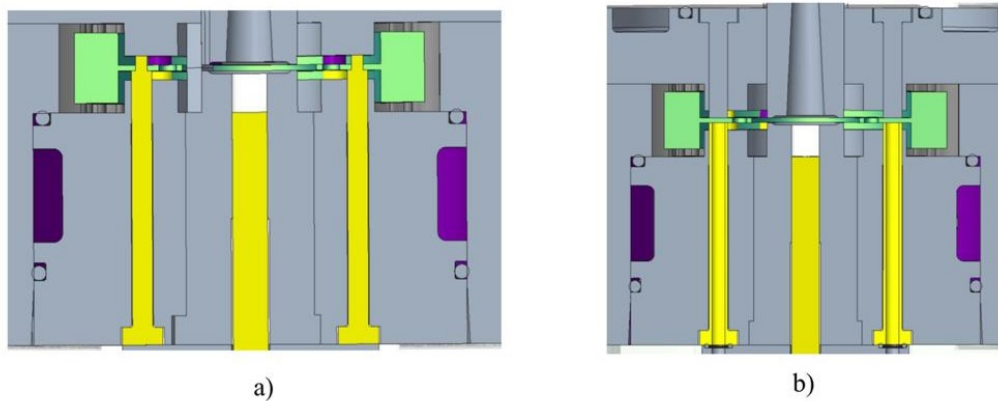


Figure 4.8: a) Alignment of the insert in the mould; b) Tubular ejector pins solution for fixation of the insert during the mould closure.

In the cold channel system in which the mould works, and bearing in mind that it is a single-cavity mould, the polymer flows from the injection unit to the injector, which in turn passes through the main runner channel until it reaches the gate, thus filling the cavity. The isolated runner system is represented in Figure 4.9.

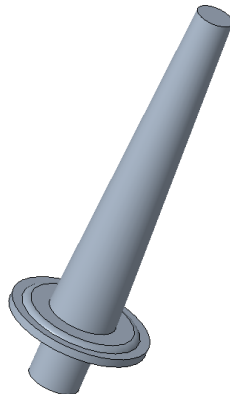


Figure 4.9: Runner system.

The dimensioning of the runner is carried out according to the type of material to be injected, the volume of the part, and the principles enumerated previously in the literature review to make a runner system as efficient as possible.

The type of gate chosen for the part was the sprue diaphragm gate because it allows a complete and homogeneous filling. In addition, this type of gate is used when the centre of the product has a larger diameter than the main channel, having as advantages the prevention of flow marks, runner machining effective and less pressure loss. Complementing Figure 4.9 with Figure 4.10 gives an overview of the entire part runner system.

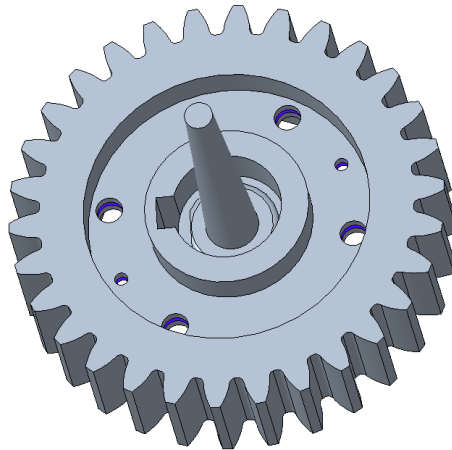


Figure 4.10: Complete runner system of the piece.

4.4.2 Modifications

Centring, guiding and clamping of the mould

The centring system allows a precise alignment of the mould with the injection unit, being, in most cases, the centring done by the centring ring on the fixed and movable side of the mould. The guiding system allows a precise alignment of the two halves of the mould, ensuring dimensional reproducibility of the parts throughout the moulding cycles. To correctly guide a round mould, since the material was already available, the mould must have three guides and three guiding bushes, with one set of different diameters from the others, to avoid errors when assembling the mould.

The guides should be long enough so that when the two halves of the mould meet, they are the first to come into contact, allowing proper alignment before the parting line joins and thus avoiding possible damage that misalignment can cause.

Regarding the clamping of the mould, new screws had to be added to the fixed and movable sides of the mould. The ones on the fixed side were due to introducing a new cavity plate (plate 2) so that a cooling system could be applied in the insert 100 more easily and fulfilling the minimum distances to the plate required. The alteration of the length of the clamping screws on the movable side had two factors in mind: the reduction of the thickness of the risers (plate 5) to allow the reduction of the extraction course; and the addition of an isolation plate to avoid energy losses that may occur during the injection process. In Figure 4.11, it is possible to observe the guiding and clamping of the mould.

Although POM is usually a material that does not require insulating plates in the mould, it was decided to add two insulating plates, one on the movable side between the backup

plate (plate 4) and the risers and another on the fixed side between the clamping plate (plate 1) and the machine plate to avoid heat losses to the machine plate.

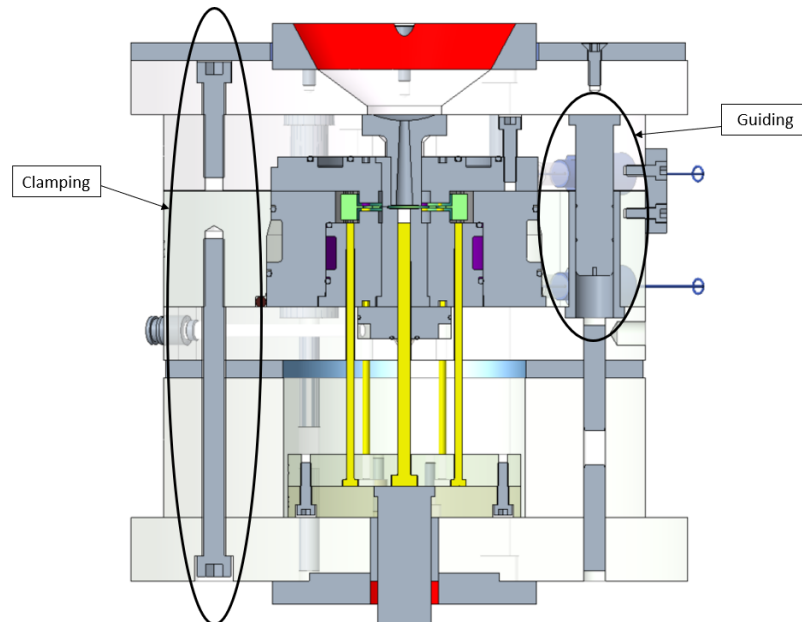


Figure 4.11: Guiding and clamping of the mould.

Cooling System

The design of a good temperature control system allows the optimization of the ejection time of plastic parts, guaranteeing the quality of the same. Therefore, it is necessary to control the thermal gradient of the coolant, ensuring that it is homogeneous on the surface of the moulding zone to avoid problems such as warping.

The cooling channels are connected to create a closed circuit through which the coolant can flow without creating leaks, and it is recommended that the holes in the channels are no closer than 15 mm to the moulding areas and the mould faces [51].

Despite the existing limitations of the "mother-mould" structure, it is possible to create three water circuits that promote the cooling of the moulding zone. Therefore, two channels were implemented on the movable side (200 and 200A inserts) and one on the fixed side (100 insert). Figure 4.12 shows the layout developed for cooling the moulding area.

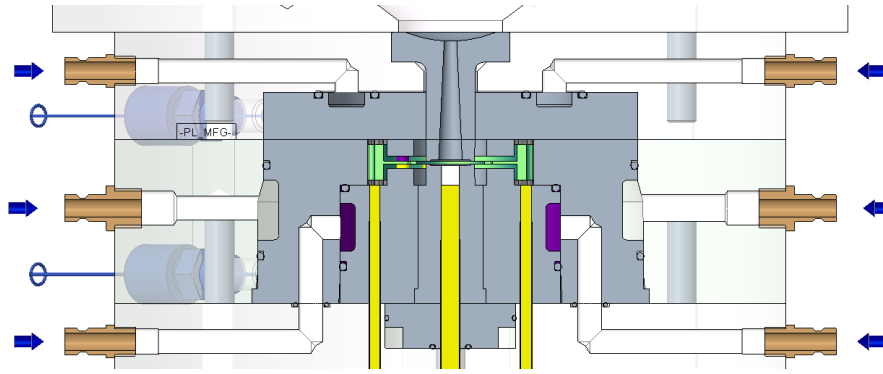


Figure 4.12: Layout developed for cooling the moulding area.

To understand how to size a refrigeration system, it is necessary to understand all the heat exchanges in the system. In injection moulding, three different types of heat transfer occur, which encompass the thermoplastic, the mould and the surrounding environment.

During the injection process, there are several oscillations in the mould temperatures due to injection cycles, as well as variations that may occur in the processing conditions. Therefore, precisely predicting the thermal behaviour in the mould cooling process would involve mathematical methods that are out of scope of this work. Thus, it is acceptable to simplify the mathematical models without significantly affecting the results, making the process more efficient [51].

Thus, the applied simplifications are:

- The process is nearly static;
- Fluctuations in temperatures and thermal fluxes during the cycles are negligible;
- Average values of properties are considered during the relevant periods.

To achieve thermal balance in a mould, it is necessary to ensure that Equation 4.1 holds true, considering that the heat absorbed by the mould is positive and the heat released is negative

$$\sum Q_i = Q_{TM} + Q_P + Q_{amb} = 0 \quad (4.1)$$

where:

Q_{TM} is the heat transferred by the cooling medium;

Q_P is the heat generated by the polymer;

Q_{amb} is the heat released to the environment.

Beginning with the heat exchange with the environment, Q_{amb} , it is equal to the sum of the heat exchanged by conduction, convection and radiation.

$$Q_{amb} = Q_{cond} + Q_{conv} + Q_{rad} \quad (4.2)$$

In theory, heat exchange by conduction is the most significant of the three forms of heat transfer to the environment. The molten thermoplastic, cooling channels, and the interaction of the mould with the injection machine plates are responsible for the conduction heat exchanges that occur in a production cycle. The heat transfer by conduction is represented in Equation 4.3

$$Q_{cond} = A_f \cdot \beta_k \cdot (T_a - T_m) \quad (4.3)$$

where:

- A_f is the contact area of the mould clamping system, in m^2 , and is equal to $0.129 m^2$;
- β_k is the proportionality factor, in $W m^{-2} \text{ } ^\circ C^{-1}$, and for low alloy steel, $\beta_k = 100 W m^{-2} \text{ } ^\circ C^{-1}$;
- T_a is the ambient temperature, in $^\circ C$, and was considered equal to $20 \text{ } ^\circ C$;
- T_m is the mould temperature, in $^\circ C$.

Heat transfer by convection occurs when there is heat exchange in a surrounding fluid medium (liquid or gas). This phenomenon occurs due to the movement of convection currents of more energetic molecules present in the hotter body, which transfer to the cooler body through fluid displacement [52]. The expression that represents this phenomenon is described in Equation 4.4

$$Q_{conv} = A_l \cdot h \cdot (T_a - T_m) \quad (4.4)$$

where:

- A_l is the lateral area of the mould, in m^2 , and its value is equal to $0.189 m^2$;
- h is the convection heat transfer coefficient to the open air ($\approx 15 W m^{-2} \text{ } ^\circ C^{-1}$), [51].

Heat transfer by radiation involves the transmission of energy through electromagnetic radiation. Unlike the other mechanisms of heat transfer, radiation does not require a physical medium for its propagation and can occur in a vacuum. The rate of heat transfer by radiation depends not only on the geometry but also on the material type and the temperature differential between the body and the surrounding medium [52]. The heat transfer by radiation formula is given by Equation 4.5

$$Q_{rad} = A_l \cdot \varepsilon \cdot \sigma \cdot (T_a^4 - T_m^4) \quad (4.5)$$

where:

- ε is the emissivity factor (0.25 for steel);
- σ is the Stefan-Boltzmann constant ($5.67 \cdot 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$), that relates the intensity of radiation emitted by a black body to its temperature.
- T_a and T_m are the absolute temperatures, in K, of the environment and the mould, respectively.

The heat generated by the polymer, Q_P , measures the thermal energy released during its processing or operation. It is essential to accurately quantify this heat generation as it affects the overall temperature distribution, thermal stability, and performance of the polymer and the system in which it is used. The rate of heat transferred by the molten polymer to the mould cavity while it remains inside can be given by:

$$Q_P = \frac{m_p \cdot c_p \cdot \Delta T}{t_p} \quad (4.6)$$

where:

- m_p is the mass of the polymer, in kg;
- c_p is the specific heat capacity of the polymer, in $\text{J kg}^{-1} \text{ }^\circ\text{C}^{-1}$;
- ΔT is the temperature difference between the polymer and the mould cavity ($T_{inj} - T_m$);
- t_p is the residence time of the material in the mould, in s.

The residence time of the material can be obtained using the following Wübken Equation (4.7)

$$t_p = \frac{s^2 \cdot \ln(KY)}{\pi^2 \alpha_{ef}} \quad (4.7)$$

where:

- s is the part thickness, in m, equal to 0.015 m;
- α_{ef} is the average effective thermal diffusivity, in $\text{m}^2 \text{s}^{-1}$;
- K is a coefficient dependent on the part thickness. For thin parts ($s \leq 3 \text{ mm}$) $K = 4/\pi$, and for thick parts ($s > 3 \text{ mm}$) $K = 8/\pi^2$;
- Y is the dimensionless temperature given by:

$$Y = \frac{T_{inj} - T_m}{T_{ext} - T_m} \quad (4.8)$$

where:

T_{inj} , T_{ext} , T_m are the injection, ejection and mould temperatures, in $^{\circ}\text{C}$, respectively.

Calculating the previous steps makes it possible to determine the amount of heat that needs to be transferred to the system through the temperature control system to promote equilibrium. Table 4.6 presents some of the calculated properties for heat transfer in the mould.

Table 4.6: Heat transfer properties of polymers.

Properties	POM-C	PA6-GF
$T_{inj} / ^{\circ}\text{C}$	210	285
$T_m / ^{\circ}\text{C}$	100	90
$T_{ext} / ^{\circ}\text{C}$	140	260
Q_{cond} / W	1033.92	904.68
Q_{conv} / W	226.90	198.54
Q_{rad} / W	32.17	26.82
Q_{amb} / W	1293	1130
$\alpha_{ef} / \text{m}^2 \text{s}^{-1}$	$1.10e - 7$	$1.30e - 7$
t_p / s	166.13	12.77
Y	2.75	1.15
m_p / kg	0.0304	0.0347
$C_p / \text{Jkg}^{-1} \text{K}^{-1}$	2210	1200
Q_p / W	44.5	81.6

Once the value of heat emitted by the polymer and the heat exchanges from the mould to the environment are known, it is possible to calculate the amount of heat that the cooling fluid must provide to the system using the thermal balance equation:

$$Q_{TM} = Q_p - Q_{amb} \quad (4.9)$$

The flow rate of the cooling fluid can be calculated by estimating its temperature variation throughout its passage through the circuit, which typically ranges from 2 °C to 5 °C [51]. The flow rate is given by Equation 4.10

$$m = \frac{|Q_{TM}|}{C \cdot \Delta T} \cdot 60 \quad (4.10)$$

where:

- m is the flow rate, in L min^{-1} ;
- C is the specific heat capacity of the cooling fluid ($4186 \text{ J kg}^{-1} \text{ }^\circ\text{C}^{-1}$, for water);
- ΔT is the fluid temperature variation.

Considering a temperature difference (ΔT) of 4 °C, the value of Q_{TM} was calculated using Equation 4.9 as well as the flow rate of the fluid using Equation 4.10. In Table 4.7 these values are presented for each material used.

Table 4.7: Heat transferred by the cooling medium, Q_{TM} , and flow rate, m .

Material	Q_{TM} / W	$m / \text{L min}^{-1}$
POM	1248.47	4.47
PA-GF	1048.43	3.76

After the calculation, it is possible to determine the recommended diameter of the channels in the circuit using the Table 4.8.

Table 4.8: Water channels diameters. Adapted from [51].

Flow rate (L min^{-1})	Diameter (mm)
3.8	8
9.5	11
38	19
85	23.8

After consulting the Table 4.8, it is concluded that the diameter of the water channels should be 8 mm. With the identification of the recommended diameter, it is necessary to determine the length of the channels the mould should have to fulfil the objective.

This length should be as long as possible, and due to this factor, it should not be calculated. However, it is important to verify if the cooling channels in the mould ensure the desired heat transfer and, with this, determine the minimum length they should have [51]. The minimum length, in meters, can be calculated using the following expression:

$$L = \frac{2 \cdot |Q_{TM}| \cdot e_c}{k \cdot \pi \cdot \Delta T} \quad (4.11)$$

where:

- e_c is the distance of the channels to the surface of the moulding zone;
- k is the thermal conductivity of the mould material ($\approx 34 \text{ W m}^{-1} \text{ }^\circ\text{C}^{-1}$), [32];
- d is the channel diameter (0.008 m);
- ΔT is the temperature difference between the moulding surface and the cooling fluid ($\approx 120 \text{ }^\circ\text{C}$).

Given that the designed mould has three cooling circuits at different distances from the mould zone, Table ?? shows the values of the minimum channel length for each of these circuits.

Table 4.9: Minimum length for the cooling circuits implemented in the mould.

Cooling Circuit	e_c /m	L /m (POM)	L /m (PA-GF)
Insert 100	0.016	0.39	0.20
Insert 200	0.037	0.80	0.46
Insert 200A	0.009	0.22	0.11

With the completion of the dimensioning process, the cooling channels require a flow rate of approximately 3.8 L min^{-1} , indicating that they should have a diameter of 8 mm. The desired minimum distance for the parts to be unmoulded as quickly as possible is 0.34 or 0.20 m for insert 100 circuit, 0.78 or 0.46 m for insert 200 circuit, and 0.19 or 0.11 m for insert 200A circuit, when using POM and PA-GF, respectively. These values are fully met due to the developed circuits' lengths of 0.65 m, 0.83 m and 0.4 m, respectively.

This information suggests that the cooling system design meets the requirements for efficient cooling and rapid demoulding of the moulded parts. The calculated flow rate and channel diameter ensure sufficient cooling capacity, while the length of the circuit is longer

than the minimum required distances for each material. All the calculations performed before are also present in Appendix C.

Vacuum system

A vacuum system in a mould is a mechanism used to secure an insert in place during the injection moulding process. When producing plastic parts with inserts, such as metal components, it is crucial to ensure that the insert remains firmly positioned within the mould cavity during the injection of molten plastic.

The vacuum system works by creating a suction force that holds the insert in place against the mould surface. It typically involves incorporating a vacuum channel or passages within the mould design. These channels are strategically positioned to align with the desired contact points on the insert.

During the injection moulding process, the mould is closed, and the molten plastic is injected into the cavity. Simultaneously, the vacuum system is activated, applying suction to the designated areas on the insert. The suction force generated by the vacuum helps to prevent the insert from shifting or moving during the plastic injection, ensuring accurate and precise positioning. The suction force can be calculated by the following equation:

$$F = \frac{\pi \cdot d^2}{4} \cdot \frac{\Delta p}{10000} \quad (4.12)$$

With the internal diameter of the tubular ejector pins being equal to 2.8 mm and considering that the available vacuum pumps have a maximum pressure drop of 800 mbar, it is possible to achieve a maximum suction force of 0.493 N. Consequently, the maximum mass that the pumps can support is approximately 50 g. Since there are four tubular ejector pins in the created system, with the maximum pressure, it would be possible to support a mass of 200 g. As the inserts mass is around 23 g, a simple proportion calculation reveals that the pump would need to work at a minimum pressure of 100 mbar during the injection of the parts.

It is essential to design the mould and the vacuum system in a way that allows proper venting and prevents air entrapment, as trapped air can hinder the effectiveness of the vacuum system. The vacuum system should also be adequately maintained and monitored to ensure reliable operation throughout the injection moulding.

In Figure 4.13, it is possible to see the vacuum system created, which involved creating an air channel in the backup plate that extends to a box designed within that plate. In this box, a lid with four holes is placed to allow air to pass through and reach the tubular ejector pins, which then circulate the air to the surface of the insert (as shown in Figure 4.8). With

the insertion of this created lid, only one air channel is required, which circulates air in the lower part of the box and then passes through the four holes.

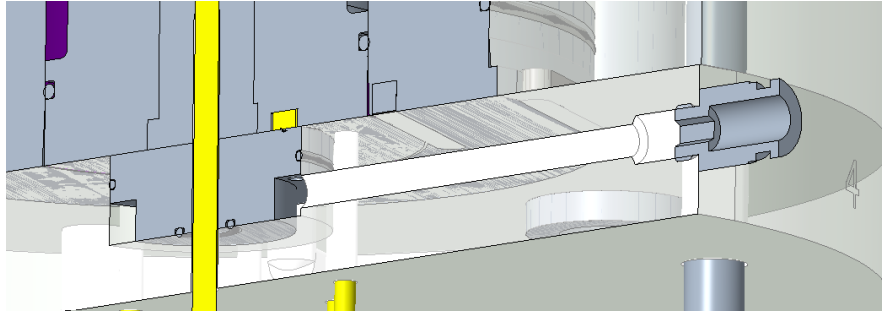


Figure 4.13: Vacuum system of the mould.

In this way, an improved insert-to-plastic adhesion is achieved, minimising the risk of insert displacement or misalignment, and producing high-quality moulded parts with consistent dimensional accuracy.

Ejection System

The mould ejection system also had to be modified once the placement and number of ejector pins depend on the part geometry. The purpose of the ejector plates is to block the ejector pins and transmit the movement generated by the hydraulic shaft of the injection unit. It has two plates (ejector retainer plate and ejector base plate) and nine round ejector pins. In Figure 4.14, it is possible to observe the ejection system.

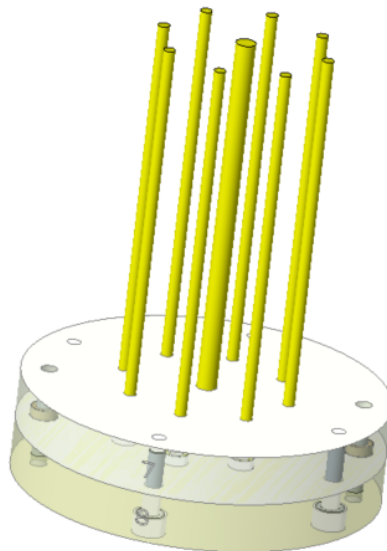


Figure 4.14: Ejection system of the mould.

Final Mould Design

After all these modifications, it was possible to reach a final design for the mould. Figure 4.15 shows the final 3D of the mould developed in CREO software. The assembly drawing of the mould is presented in Appendix B.

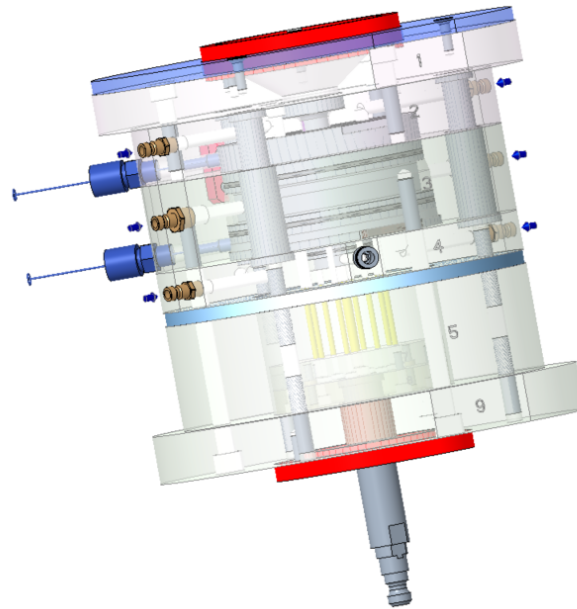


Figure 4.15: Final design of the mould.

4.5 Mould Filling Simulation

Mould filling simulation, or injection moulding simulation, is a computational analysis technique used in the field of manufacturing to simulate the behaviour of molten material as it fills a mould cavity during the injection moulding process. It provides valuable insights into the flow behaviour, material distribution, and potential issues that may arise during the manufacturing process.

The simulation begins with creating a digital representation of the mould geometry, including the cavity, runners, gates, and other relevant features. Next, the material properties and process parameters, such as melt temperature, injection pressure, and cooling conditions, were defined as inputs for the simulation.

A filling simulation typically involves a Finite Element Method (FEM) analysis. FEM is a numerical technique that divides the mould geometry into smaller elements to solve the fluid flow equations during the injection process. In this case, the boundary conditions may include the injection pressure applied at the mould entrance, the temperature of the injected material, the presence of runner systems and the mould structure.

The quality and resolution of the mesh are crucial for obtaining accurate results in the simulation. A refined and well-adjusted mesh is necessary to capture complex flow details. Thus, a global mesh size of 0.5 mm was used for both the part and the insert, and a hybrid mesh (consisting of hexahedra and tetrahedra) was applied inside the part, while a mesh with triangular prisms was used outside. A total of 2 708 846 solid mesh elements were obtained. A convergence study of the solution was not specifically conducted. The mesh was only refined to a satisfactory level that did not require excessive computational effort.

The software solves problems related to viscous fluid flow and the behaviour of materials during the plastic injection moulding process. The simulation software then generates visual representations, often colour-coded maps, to illustrate important information. These results include filling patterns, flow front progression, material velocity, pressure distribution, and potential issues like air traps, weld lines, or short shots.

In this section, a comparison will be made between the filling of the part with insert and without insert, using POM as the plastic material to be injected. The software used to perform the simulation was Moldex, and the first result to be presented is the filling time, which occurred uniformly, as can be seen in Figure 4.16, showing the respective fillings at 25%, 50%, 75%, and 100%. The total filling time was 0.801 seconds for the hybrid gear and 1.081 seconds for the gear without insert, which is normal because it is necessary more plastic material to fill the cavity.

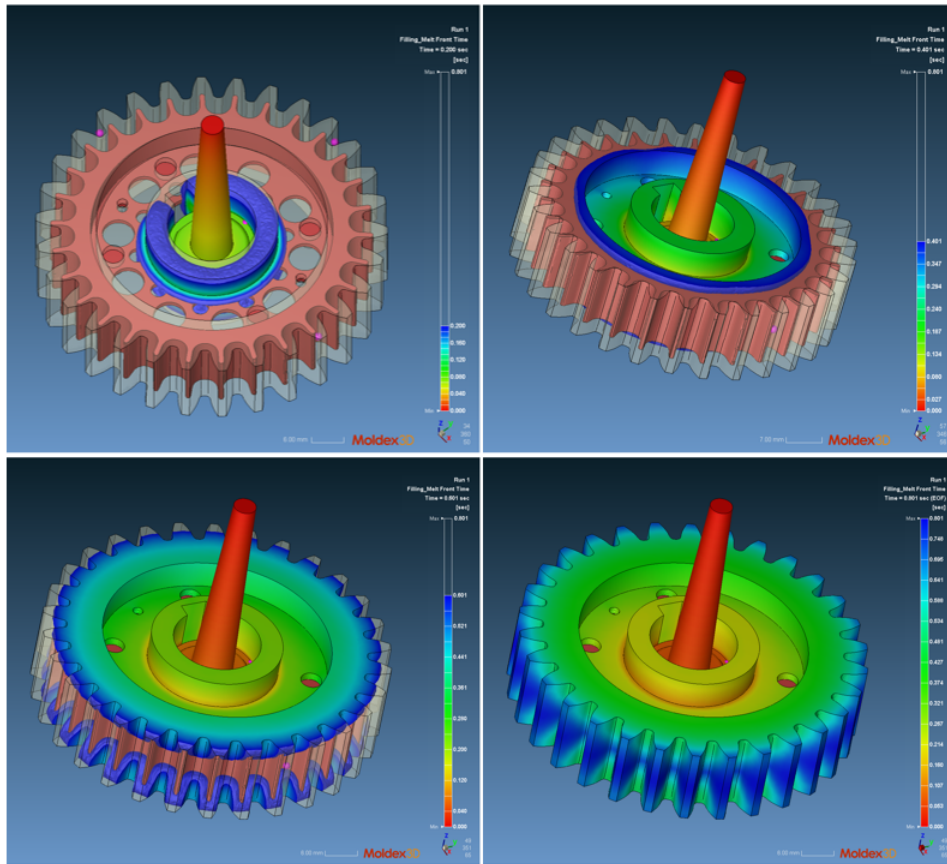


Figure 4.16: Simulation of the filling time of the hybrid gear (25%, 50%, 75%, 100%).

Another aspect to consider during the filling of the part is the pressure drop from the beginning to the end of the injection. The pressure drop helps to understand how the molten plastic flows through the mould cavity during the injection process. It indicates if there are any potential flow restrictions or issues such as narrow flow channels, inadequate venting, or high viscosity regions. In Figure 4.17, the pressure drop during the filling is illustrated.

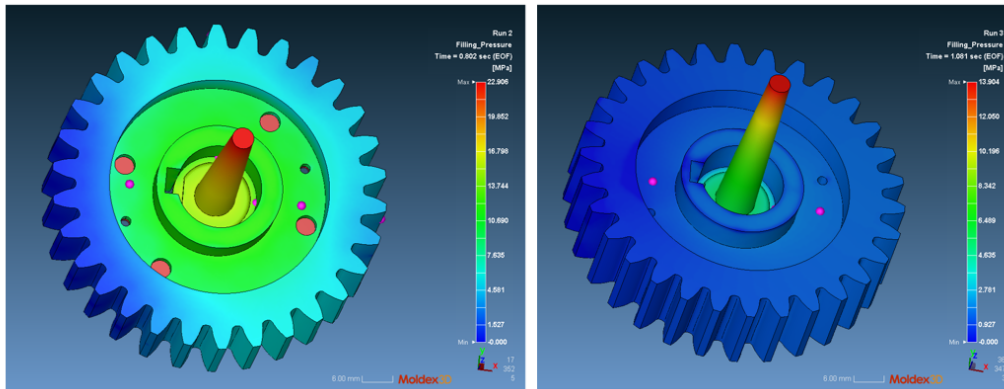


Figure 4.17: Pressure drop during the filling of the hybrid gear (left) and gear without insert (right).

By analysing the pressure drop, it was possible to identify areas that may result in incomplete filling, short shots, or other defects. The area most susceptible to these issues is the region of the teeth in the direction of the keyway, as there is no direct material injection in that area, causing a delay in material flow to reach the teeth in that direction.

Comparing both simulations, the presence of the insert creates additional flow channels and obstacles within the mould cavity. This leads to a more complex flow pattern and increased flow resistance, resulting in higher filling pressure.

Furthermore, another important factor that filling simulation provides is the packing time. After the mould cavity is filled with molten plastic, the packing time begins. During this phase, additional pressure is applied to the plastic material to ensure proper packing and compensate for any volumetric shrinkage or voids that may have occurred during filling. The packing time allows the plastic to flow and pack tightly into all areas of the mould cavity, ensuring consistent part dimensions and minimizing any defects. The packing time is set in the software, in this case, a time of 5 seconds was defined, and it starts counting when 98% of the volume of the part is already filled. This packing time is performed at a pressure of approximately 60% of the filling pressure.

Once the mould cavity is filled and packed, the plastic part needs time to cool and solidify before it can be ejected from the mould. The cooling time is influenced by factors such as the material type, part thickness, mould design, and cooling system efficiency. Proper cooling is crucial to ensure dimensional stability, prevent deformation, and achieve consistent part quality. The cooling time given by Moldex was 13 seconds.

In theory, the cycle time provided by the software from the start of injection until the mould opening and part ejection will be approximately 25 seconds.

It is important to note that in the simulation without the insert, the previously mentioned times (injection, packing and cooling) should be longer since more material is required to completely fill the part. However, in order to maintain consistency and enable a direct comparison between the two simulations, the conditions and times were kept the same. By keeping the conditions consistent, it becomes possible to assess the impact of the insert on the moulding process independently.

Considering, now, the warpage that the gears can suffer, the differential shrinkage effect was analysed, and it refers to the uneven or differential contraction that occurs in different regions of the moulded part during the cooling and solidification process. When the molten material is injected into the mould cavity and then cools down, it undergoes a volumetric contraction as it transitions from a liquid to a solid state. However, different areas of the gear may experience different cooling rates and solidification behaviours, leading to non-uniform shrinkage. This can result in variations in dimensional accuracy and shape distortion of the final moulded part. The areas that cool and solidify more quickly will experience greater shrinkage, while areas that cool more slowly will have less shrinkage. In Figure 4.18, it is possible to visualize the total differential shrinkage effect displacement of the gear with and without insert.

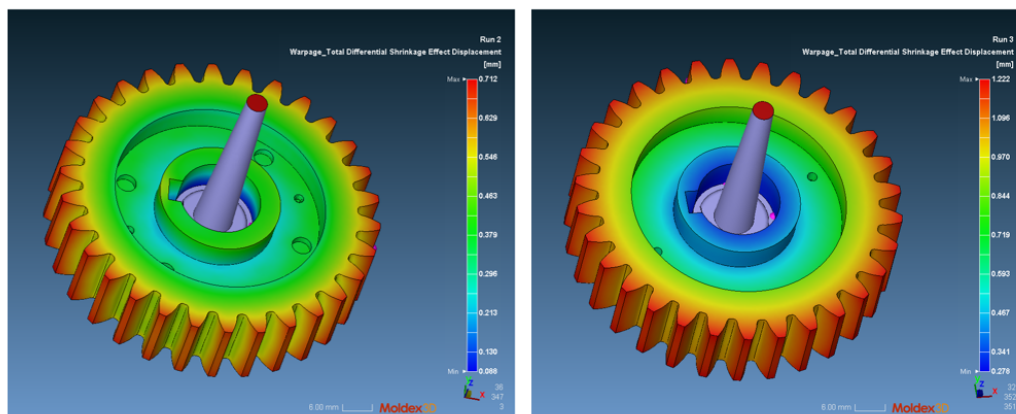


Figure 4.18: Total differential shrinkage effect displacement of the gear with insert (left) and without insert (right).

The differential shrinkage effect displacement refers to the difference in dimensional changes between the two gears. In this way, it is predictable that the gear without insert will have higher changes in his dimensions since the part has a greater thickness and does not find obstacles that hinder the contraction of the gear.

The interface between the gear and the insert can introduce additional interactions and constraints during cooling. Differential thermal expansion between the gear and the insert can induce internal stresses and affect the shrinkage behaviour, resulting in different

shrinkage patterns compared to gears without inserts.

Based on the conducted study, it was possible to observe that although the filling process occurs without anomalies, factors, such as weld lines and air traps, must be considered to ensure that the parts fulfil their purpose.

One of these factors is clearly present in the holes and teeth of the hybrid gear, which create a weld line between flow fronts at the end of the filling process, resulting in a fragile zone. Figure 4.19 shows the location of the possible weld lines that can be formed after injecting the gears with and without insert.

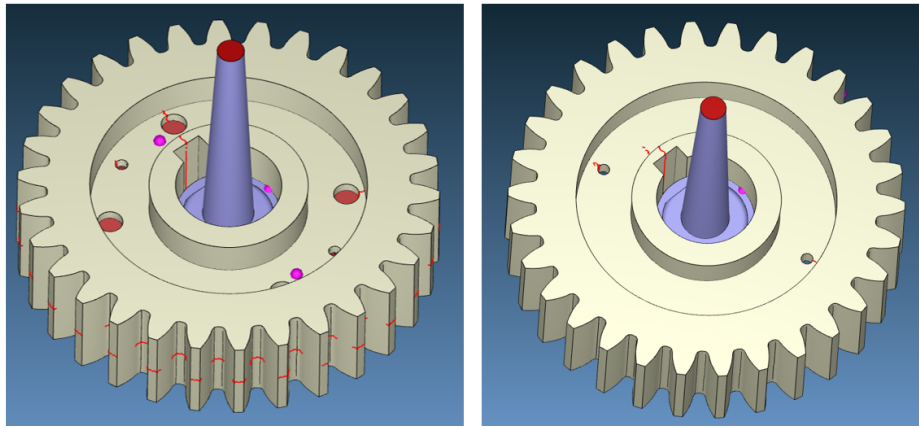


Figure 4.19: Weld lines formation zone with insert (left) and without insert (right).

The presence of an insert can create flow restrictions or changes in the flow path of the molten polymer during the filling process. These flow restrictions can cause the polymer to converge and meet at certain points, leading to the formation of weld lines in the teeth. In the absence of an insert, the flow path may be more open and unrestricted, resulting in a more uniform flow and reduced likelihood of weld line formation.

Regarding air traps, they can occur primarily in the teeth or holes of the gear that are entirely hollowed out because the air becomes trapped in these areas of the mould cavity, restraining the proper filling of the plastic material. The location of the potential air traps points is represented in Figure 4.20.

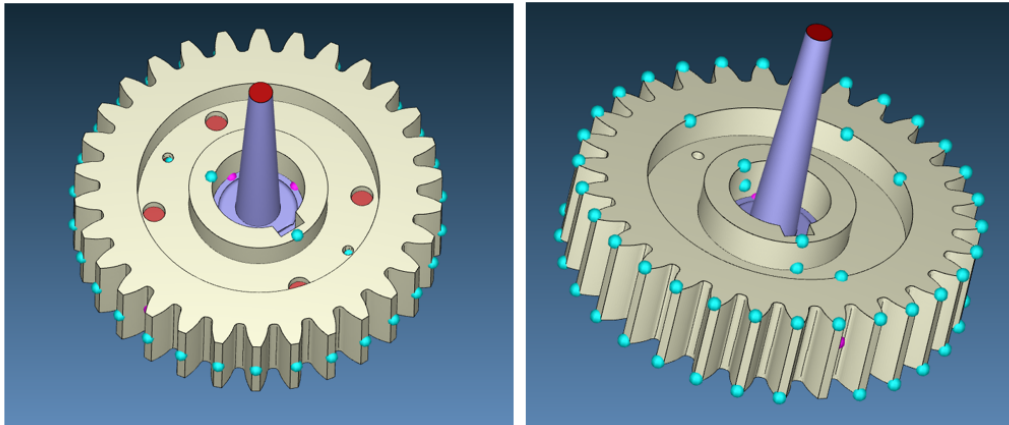


Figure 4.20: Potential air traps formation zone with insert (left) and without insert (right).

The mould design and the presence of an insert can influence the placement of air traps. The insert may occupy certain areas of the mould cavity, limiting the available space for placing air traps. Placing the air traps at the top and bottom of the gear width without an insert may be more feasible from a mould design perspective, as there may be more space available in these regions.

By analysing the simulation results, it is possible to gain valuable insights into the filling behaviour and make informed decisions to optimize the injection moulding process. This includes adjusting the gate location, modifying the mould design, optimizing process parameters, or selecting suitable materials to enhance product quality, minimize defects, and reduce cycle time.

Overall, filling simulation played a crucial role in the injection moulding process by giving manufacturers a deeper understanding of the material flow and in the future assisting in optimising mould design, process parameters, and material selection for gears. It was a valuable tool for improving product quality, reducing defects, and enhancing the overall efficiency of the injection moulding manufacturing process.

Chapter 5

Manufacturing and Dimensional Control of the Inserts and Gears

5.1 Introduction

Manufacturing and dimensional control are crucial aspects in the production of the inserts and hybrid gears. The manufacturing process of inserts involves precision machining techniques such as milling, turning, and grinding. These processes ensure the desired shape, dimensions, and surface finish of the inserts. Additionally, advanced manufacturing methods like additive manufacturing (3D printing) can be employed to create complex geometries and internal structures.

Dimensional control plays a vital role in ensuring the accuracy and quality of inserts and hybrid gears. Precise measurements and inspections are performed throughout the manufacturing process to verify dimensional tolerances, geometrical features, and surface characteristics. Advanced metrology tools and techniques such as Coordinate Measuring Machines (CMMs), and optical scanners are used to assess and validate the dimensional accuracy of the components.

The dimensional control process involves comparing the manufactured inserts and hybrid gears against design specifications and industry standards. This ensures that the components meet the required dimensional tolerances and functional requirements. Any deviations or

variations from the desired dimensions are identified, analysed, and addressed to ensure the final product's quality and performance.

This chapter begins with an in-depth explanation of the manufacturing process for the inserts. Then, it also touches upon the production of the hybrid gears through injection moulding. Additionally, this chapter discusses the dimensional control of both the inserts and hybrid gears.

5.2 Manufacturing of the insert

The designed inserts were machined and essentially went through two processes: turning lathe and milling. Before this, the aluminium rod went to the automatic mechanical saw to cut an aluminium cylinder with a slightly greater thickness than desired to have a safety margin during turning.

Then, the aluminium cylinder went to the mechanical lathe, and the fixation of the material began through the clamping bush with three clamps moving simultaneously to tighten the cylinder. Then the two faces of the cylinder were orientated until they were parallel. After this, the mobile trolley was brought from the table to the aluminium and the faces were turned until the aluminium cylinder had a 12 mm thickness. During this process, oil was needed to cool the aluminium, a material that gets very hot, unlike steel.

Afterwards, the aluminium cylinder went through the milling process on a 3-axis CNC machine. The first step is the machining of the insert housing on one side. For that, the aluminium cylinder is clamped on the milling machine table with three clamps and a clamping bush. In this first program, the point drill is called in to mark all the points to be drilled. Then the 2, 3 and 6 mm helical drills are called to make the respective holes. After, a 4-blade tungsten carbide cutter with a diameter of 20 mm is called in to drill the inner and outer diameter of the box. The cutter drills the outer diameter first, then moves back and drills the inner diameter. Finally, a reamer is used to file down the box's outer edge and other burrs on that face. This first program had an average duration of 17 minutes and 47 seconds. In Figure 5.1, it is possible to see what the insert looks like after the execution of this program.

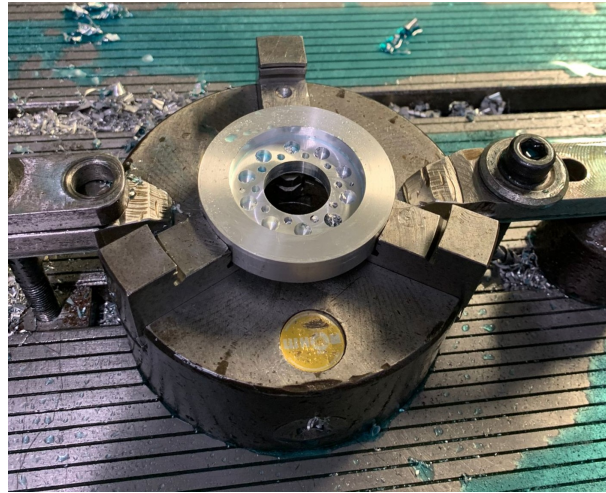


Figure 5.1: Insert after first milling operation.

After that, to obtain the insert, it was necessary to proceed with the machining of the opposing box on the other side, as well as the teeth of the insert. In order to reduce production times and minimise costs, a system was developed to guarantee the orientation of the insert so that the two programs mentioned above could be run in a single program. In Figure 5.2 is presented the system that was created. This system consists of placing two steel construction blocks with two extractor tips with a diameter of 6 mm and an exact length so that the drill during the process does not touch the extractor tip. These blocks are attached to the side of the clamps, which will also differ from those of the first stage since they will have to tighten the insert from the inside, i.e. through the box created in the previous step.

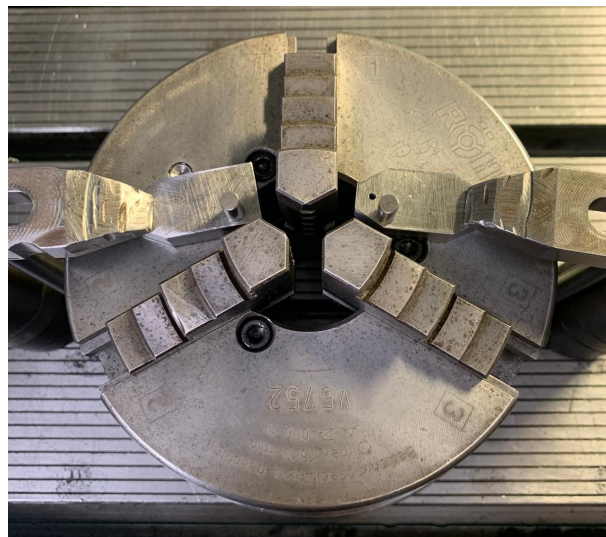


Figure 5.2: System developed to guarantee the inserts' alignment.

The insert is fitted and fixed in this developed mechanism, and before starting the second program, a surface gauge is used to ensure that the insert is completely aligned. For this

second stage, in the initial step, the process is very similar to that of the first stage, with the exception that the drills in the insert holes only go to a depth of 2 mm, given that in the previous stage, they had already been practically drilled in the totality. Then, in a more advanced phase of the program, the teeth of the insert are machined with a 4 mm cutter with a radius of 0.2 mm all around the part. This second program has an average duration of 15 minutes and 4 seconds.

At the end of this program, it was necessary to eliminate some burrs present in the holes. Finally, to give the piece a better finish, it was taken to a sandblast machine, which is a process that involves forcefully propelling fine particles, such as sand or abrasive materials, onto the surface of an object to smooth or clean it. When the abrasive particles impact the surface, they remove any irregularities, contaminants, or unwanted material, resulting in a smoother and more uniform surface texture. The differences between the insert before and after sandblasting are visible in Figure 5.3.

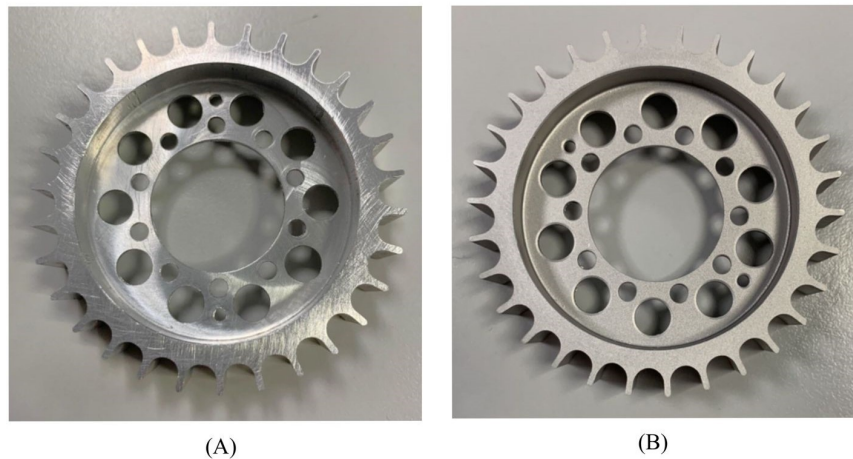


Figure 5.3: Machined insert: (a) Before sandblast finishing; (b) After sandblast finishing.

The action of sandblasting can effectively reduce surface roughness by removing burrs, sharp edges, or surface imperfections. By removing irregularities and improving the surface texture, sandblasting can enhance the overall aesthetic appearance and functionality of the part.

5.3 Dimensional control of the inserts

In this section, the dimensional control of the two studied inserts will be carried out, and they will be compared in order to draw conclusions about the influence of the manufacturing process of each insert on the final product quality.

In this way, the dimensional control of the inserts involved monitoring and managing key parameters such as weight, insert and tooth width, and surface roughness to meet design specifications and functional requirements. All the measurements and roughness parameters that were calculated are located in Appendix C.

5.3.1 Aluminium Inserts

To perform the dimensional control of the aluminium inserts manufactured at Celoplás by milling, a random sample of 10 inserts was chosen, and to start, the mass of each insert was measured using a precision scale. The results of the ten samples are shown in Figure 5.4, along with the upper and lower standard deviation lines.

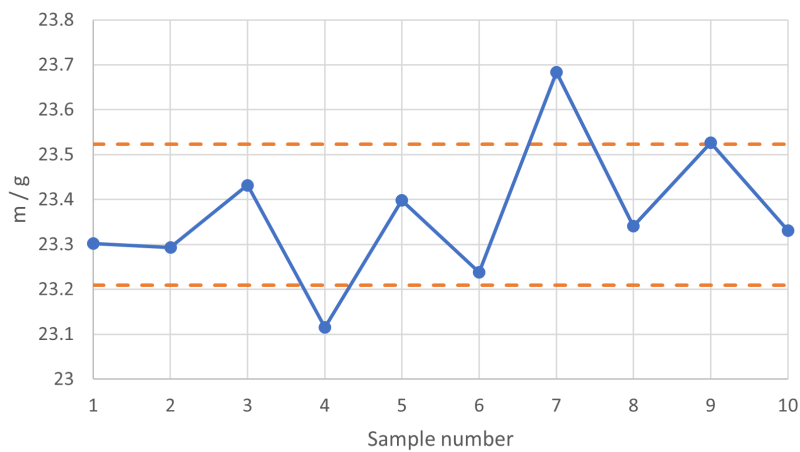


Figure 5.4: Mass of the aluminium inserts manufactured in Celoplás.

The average value of the ten measurements was 23.366 ± 0.158 g. This value is within the expected range and matches the value obtained through the CAD program, where an aluminium alloy 6000 was assigned, and using the mass properties tool a mass of 22.83 g was yielded.

Next, the width of the insert was measured using an outside micrometer. The values obtained for each insert are shown in Figure 5.5.

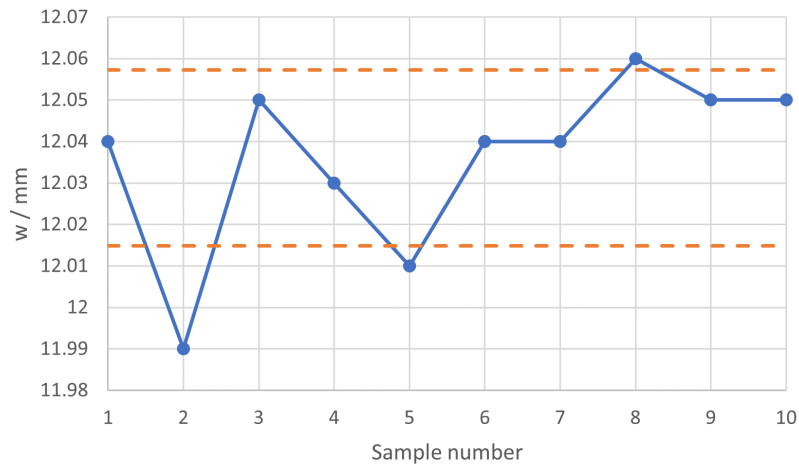


Figure 5.5: Width of the aluminium inserts manufactured in Celoplás.

The width of the insert defined in its geometry was 12 mm. In the sample used for these measurements, the average width of the insert was 12.036 ± 0.021 mm. As can be seen, the width's designed value is below the sample's standard deviation. However, this dimension has a general tolerance and meets the specifications precisely. In addition, it may even be beneficial for this value to be slightly higher than intended, as reviewed in the literature, a wider insert width improves heat dissipation.

To measure the thickness of the teeth of the insert, a micrometer with plates was used. For each piece, the tooth's thickness was measured at three different locations, and the average of the three measurements was calculated. The results of the tooth thickness measurements are presented in Figure 5.6.

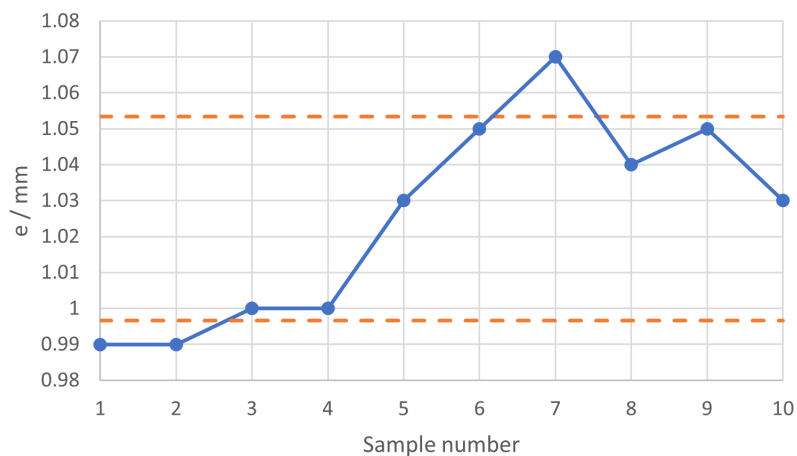


Figure 5.6: Tooth thickness of the aluminium inserts manufactured in Celoplás.

The average value of the ten measure of samples on the tooth thickness was 1.025 ± 0.028 mm. Therefore, the projected value, 1 mm, is within the range. The slight differences that can occur are probably due to tool wear or machine vibrations. However, it does not significantly impact the overall performance or functionality of the insert.

Regarding the surface roughness, a contact profilometer Hommelwerke, equipped with a Tester T8000 controller and an LV-50 linear unit, was used. The measuring probe used was a TKL 300 with a 5-micron tip radius.

In order to evaluate the surface roughness, three types of measurements were performed. The first roughness measurement was taken on the side walls of the insert, with six measurements made, three on each side face, and all taken at different points on each face. Next, the mean and standard deviation of the six measurements were calculated for each chosen roughness parameter for each insert. Figure 5.7 shows how this measurement was conducted.

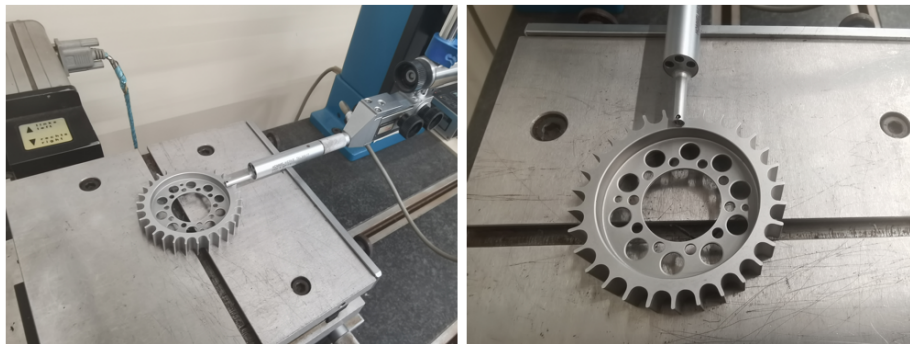


Figure 5.7: Surface roughness measurement on the side walls of the insert.

Then, the second surface roughness analysis was made on the inserts' teeth in the radial direction. The surface roughness of the inserts' teeth can influence the ease of mould release during the injection moulding process. Smoother surfaces with lower roughness values can help prevent sticking or binding of the moulded part to the insert, improving the overall efficiency of the moulding process. Here, only three measures were performed in three random teeth. The mean and standard variation were calculated similarly to the first measurement method. Figure 5.8 shows how the insert was mounted on the profilometer to measure its roughness in the radial direction.

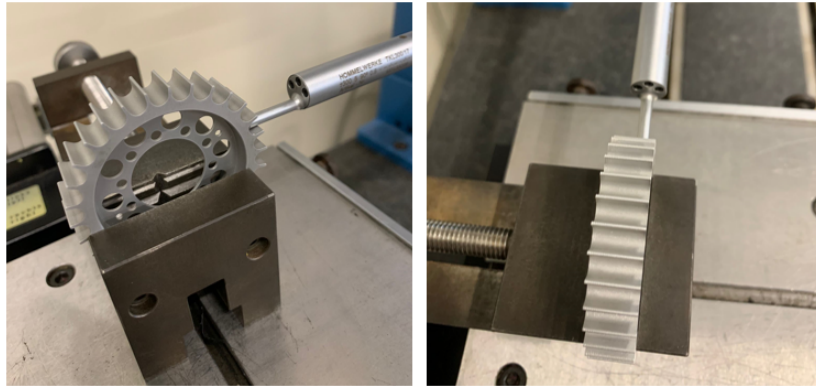


Figure 5.8: Surface roughness measurement in the radial direction of the inserts' teeth.

Finally, the last roughness surface evaluation was also performed on the inserts' teeth, but now in the axial direction. As in the previous case, three measures were taken in three random teeth. In Figure 5.9, it is possible to visualize the system built to execute this measurement.

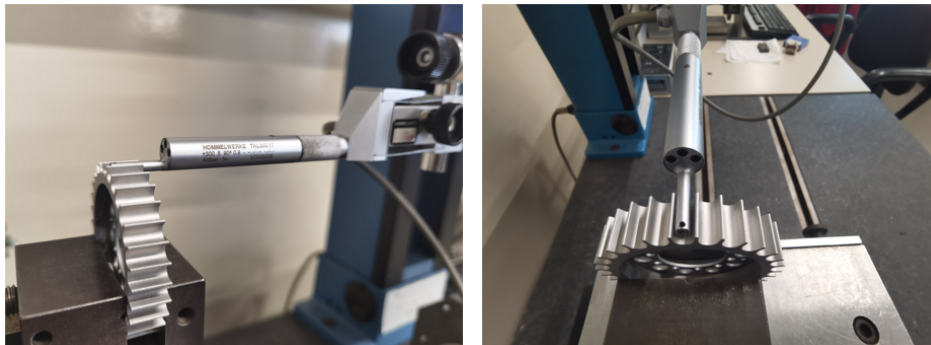


Figure 5.9: Surface roughness measurement in the axial direction of the inserts' teeth.

Before proceeding to the selection and definition of the parameters chosen to analyse the surface roughness, it is necessary to select the correct filter parameter (the cut-off length, λ_c), which is essential to the validity of the measurement. Changing the filter cut-off length will typically affect the measured result.

Moreover, the selected filter cut-off length must be at least 2.5 times the peak spacing. This means that at least two peaks and valleys, normally many more, are to be found in each cut-off length. Therefore, if a profile equal in length to five times the cut-off length were analysed, an adequate sample of roughness peaks and valleys would be included in the analysis [53].

A cut-off length is selected to suit a particular work-piece surface. First, the cut-off length was chosen according to the roughness spacing or expected roughness value. The transverse length was then determined according to the cut-off length selected from Table 5.1, which shows the relationship between cut-off length, measured roughness and transverse length [53].

Table 5.1: Relationship between cut-of length, measured roughness and transverse length. Adapted from [53].

Cut-off	Measured roughness	Transverse length
λ_c / mm	$R_a / \mu m$	L_t / mm
0.08	-	0.48
0.25	< 0.1	1.5
0.8	0.1-2	4.8
2.5	2-10	15.0
8.0	> 10	48.0

Although the 0.8 mm cut-off length is the most commonly used cut-off length for engineering surfaces and corresponds to the measured roughness values expected in the inserts, it was not possible to use this filter because there was not enough transverse sample length available. For example, there was only a 3 mm length for the side walls measurement, so the 4.8 mm needed for the 0.8 mm cut-off length was not fulfilled. Thus, it was necessary to choose to use a 0.25 mm filter, which only requires a sample length of 1.5 mm, even though the obtained roughness results may not be as precise and accurate.

Regarding the roughness and waviness parameters, the ones that would provide the most helpful information about the surface were chosen to be studied after the measurements. Every parameter has its advantages and limitations. So, the parameters selected were the average roughness, the mean peak-to-valley height, the root-mean-square profile height, the total waviness height and the total profile height.

Average Roughness - R_a

R_a is the average deviation from the profile to the mean line over the length of assessment. The average roughness is determined by the following formula:

$$R_a = \frac{1}{l_m} \int_0^{l_m} |y(x)| dx \quad (5.1)$$

The parameter R_a does not differentiate between peaks and valleys [53]. Thus, an analysis was conducted for the R_a values for each type of measurement mentioned before, and then the results were compared, as shown in Figure 5.10.

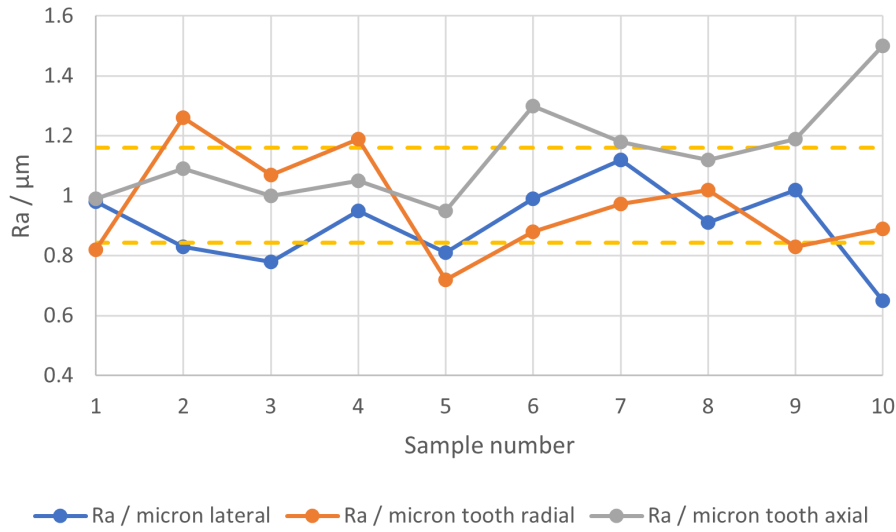


Figure 5.10: Average roughness of the inserts for the three types of measurement.

The mean of the average roughness of all the measurements is $1.00 \pm 0.16 \mu\text{m}$. From the graphic analysis, it can be seen that most values fall within this range, indicating surface homogeneity. It was also expected that the surface roughness would be higher on the inserts' teeth due to the greater machining difficulty in that area. However, it just happened for the axial direction measures. The results for the radial direction are irregular, probably due to the curvature and complex geometries involved in this direction during the machining.

Mean peak-to-valley height - R_z

To determine the parameter R_z , the filtered roughness profile is divided into 5 equal lengths, nominally equal to the cut-off length. The maximum peak-to-valley height (z_i) is determined within each cut-off length. R_z is the average of the 5 peak-to-valley heights and is determined by Equation 5.2 [53].

$$R_z = \frac{1}{5} \sum_{i=1}^5 z_i \quad (5.2)$$

The values obtained in the measurements for the mean peak-to-valley parameter can be found in Figure 5.11.

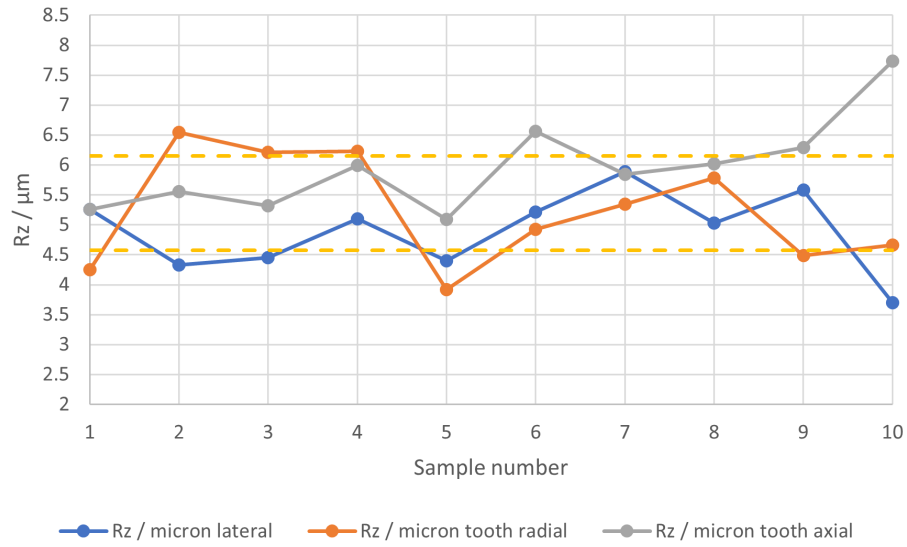


Figure 5.11: Mean peak-to-valley height of the inserts for the three types of measurement.

The mean of the peak-to-valley height parameter for the three types of measurements is $5.36 \pm 0.79 \mu\text{m}$. R_z is generally more sensitive to changes in surface finish than R_a , because maximum profile heights are being examined and not averages. Comparing the three types of analysis made, the conclusions are basically the same as the ones taken for the average roughness.

Root-mean-square profile height - R_q

R_q is the square root of the average of the square of the deviation of the profile from the mean line [53]. The root-mean-square profile height is determined by the following formula.

$$R_q = \sqrt{\frac{1}{l_m} \int_0^{l_m} y^2(x) dx} \quad (5.3)$$

The results obtained for this parameter from the tests made are presented in Figure 5.12.

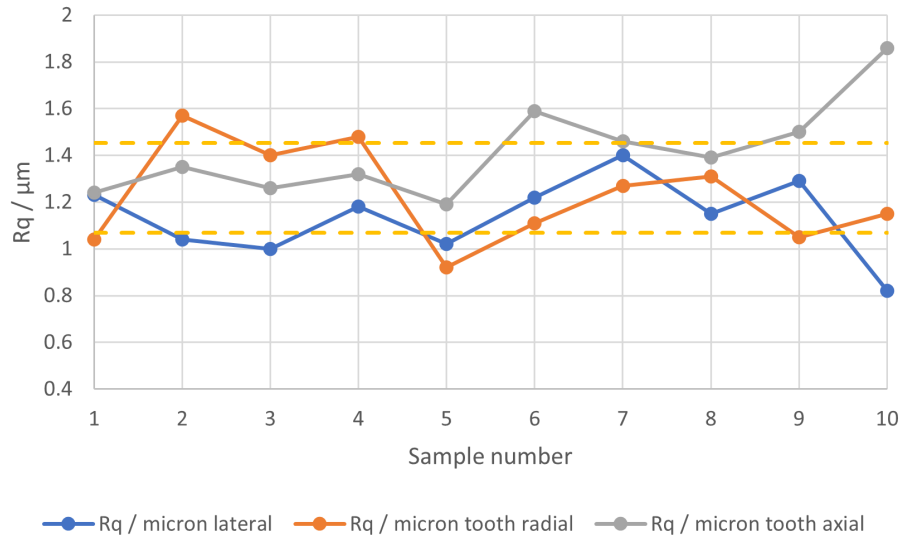


Figure 5.12: Root-mean-square average roughness of the inserts for the three types of measurement.

The mean value for the root-mean-square average roughness measurements is $1.26 \pm 0.19 \mu\text{m}$. This shows that the parameter R_q is more sensitive to peaks and valleys than R_a . However, R_q is seldom used due to the trend to use parameters that measure extreme values (peaks or valleys) and not average profile heights. R_q is relevant when looking at a surface profile as a static function.

Total waviness height - W_t

The parameter W_t is the maximum peak to valley height of the levelled and filtered waviness profile. W_t is used to monitor a production process where waviness, and not roughness alone is a critical production variable [53].

The measured values for this parameter during the tests can be found in Figure 5.13.

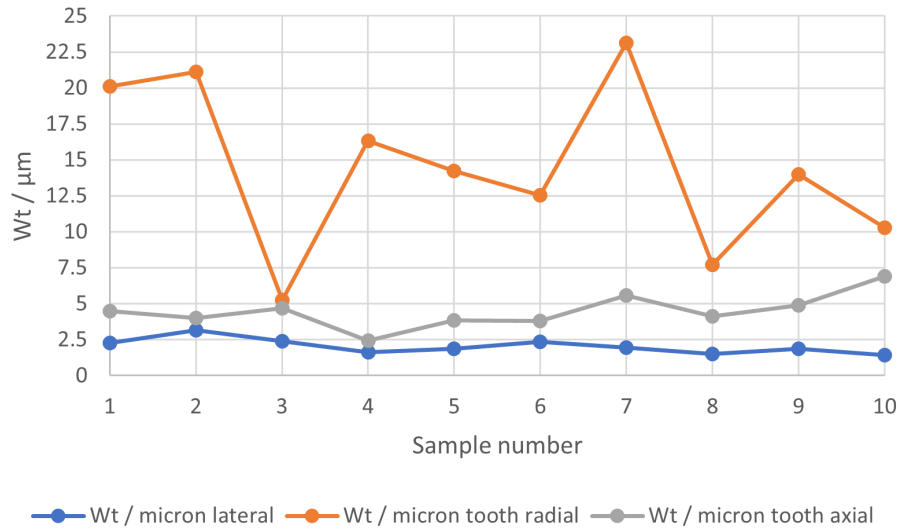


Figure 5.13: Total waviness height of the inserts for the three types of measurement.

In this case, the standard deviation lines are no longer present, and the average was not calculated due to the significant discrepancy in the results obtained in the radial direction of the inserts' teeth compared to the other two analyses conducted. The measurements on the side walls and teeth in the axial direction showed low and relatively consistent values across the ten measures. However, in the radial direction of the teeth, significantly higher values were observed with irregularity from measurement to measurement. This can be attributed to the fact that the radial direction of the tooth surface may have a more pronounced curvature or contour compared to the axial direction.

Total profile height - P_t

P_t is the total peak-to-valley height of the unfiltered but levelled surface profile. P_t is equal to the total roughness height, R_y , plus the total waviness height, W_t [53].

P_t is a measurement of the deviation of the manufactured profile from the nominal, straight profile. P_t is useful in finding and quantifying single surface defects such as scratches or pits [53]. Figure 5.14 presents the results obtained from the measurements in the contact profilometer.

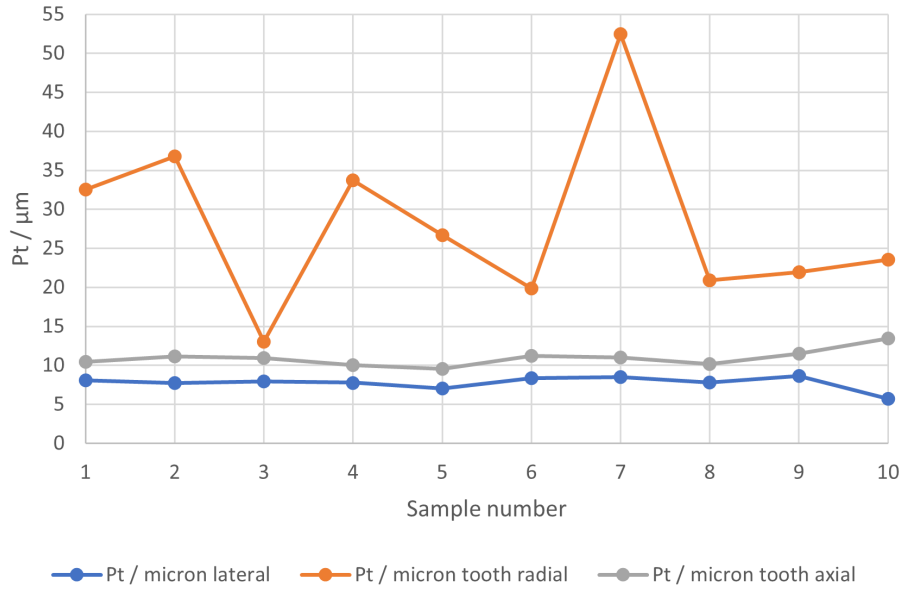


Figure 5.14: Total profile height of the inserts for the three types of measurement.

The conclusions that can be drawn for the total profile height are similar to those for W_t but with higher values in this case, as P_t considers the largest defect across the entire profile. Additionally, another reason that may have resulted in more irregular and higher values for this parameter in the radial direction of the teeth is the type of surface finish applied in this area. After machining the inserts, sandblasting was applied; however, it is more challenging to use this method in the radial direction of the teeth, leading to a difference in the surface of the teeth between the two directions.

5.3.2 3D printed Insert

The 3D printed inserts were manufactured at Hyper Metal using metal additive manufacturing, specifically selective laser melting. In Figure 5.15, it is possible to visualize the final product obtained through this process.



Figure 5.15: Metal 3D printed insert.

In these inserts, it was attempted to follow the same dimensional control procedure applied to the aluminium inserts manufactured at Celoplás. However, this was not possible as the measurement of the roughness of the sidewalls and teeth in the radial direction was not performed due to the presence of holes in these areas in this new geometry. Measuring the roughness in these areas would not only yield incorrect results but would also be challenging to measure accurately.

Starting again with the mass of the inserts, the results of the ten measurements are shown in Figure 5.16, along with the upper and lower standard deviation lines.

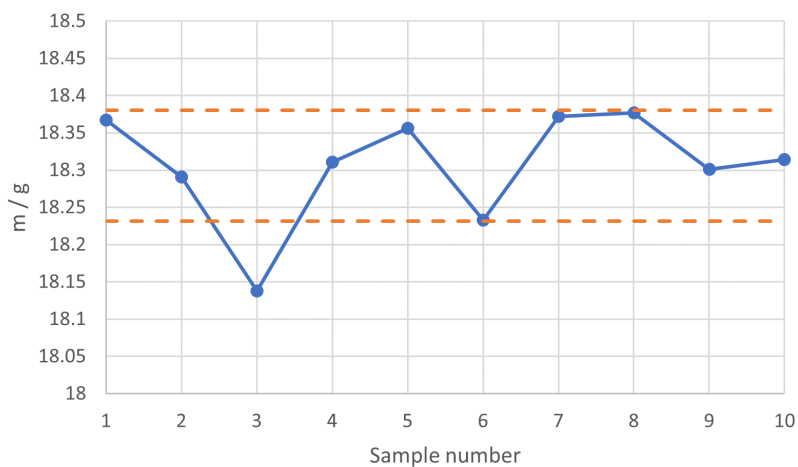


Figure 5.16: Mass of the 3D printed inserts.

The average mass of the ten inserts is 18.306 ± 0.074 g. These inserts are approximately 5 g lighter than the previously analysed inserts. This is due to the fact that these inserts have less material compared to the previous ones because of the presence of holes in this

component. Although the material is different, the difference in weight is not primarily attributed to that, as both materials have very similar properties.

Next, the width and tooth thickness were measured in the same way of the previous inserts. The values obtained for each insert are presented in Figure 5.17.

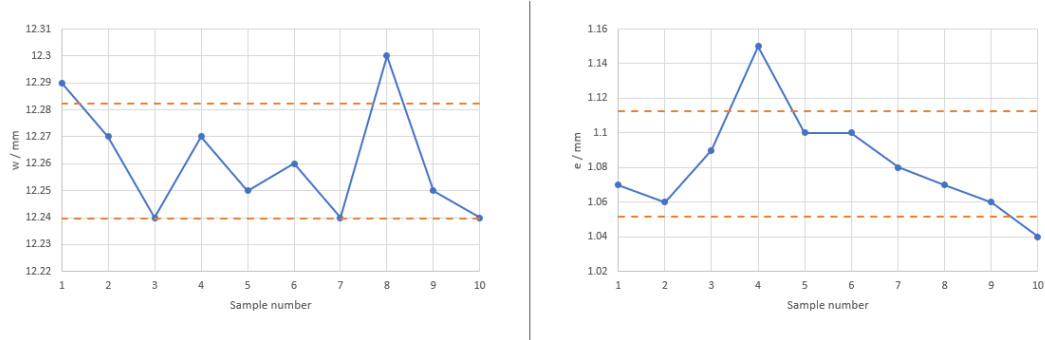


Figure 5.17: Width and tooth thickness of the 3D printed inserts.

The average width of the 3D printed insert was 12.261 ± 0.021 mm. These values are even further above the projected value (12 mm) than the inserts manufactured at Celoplás, which were already above this value. One reason that may explain this difference of approximately 0.26 mm is that during the laser melting process, excessive melting or thermal expansion can occur, resulting in an increase in the final dimension of the part.

The average value of the ten measurements was 1.082 ± 0.030 mm for the tooth thickness. As can be seen, the tooth thickness's designed value, 1 mm, is below the sample's standard deviation. However, this difference is insignificant, and the justification for this is the same as for the width of the inserts.

Moving on to the measurement of surface roughness, although it was only performed in the axial direction of the inserts' teeth, the measurement procedure and the parameters studied were the same as in the previous case. Although only one type of measurement was performed, it indicates how rough the entire piece's surface is.

In Figure 5.18, it is possible to observe the results obtained on the roughness and waviness parameters measured.

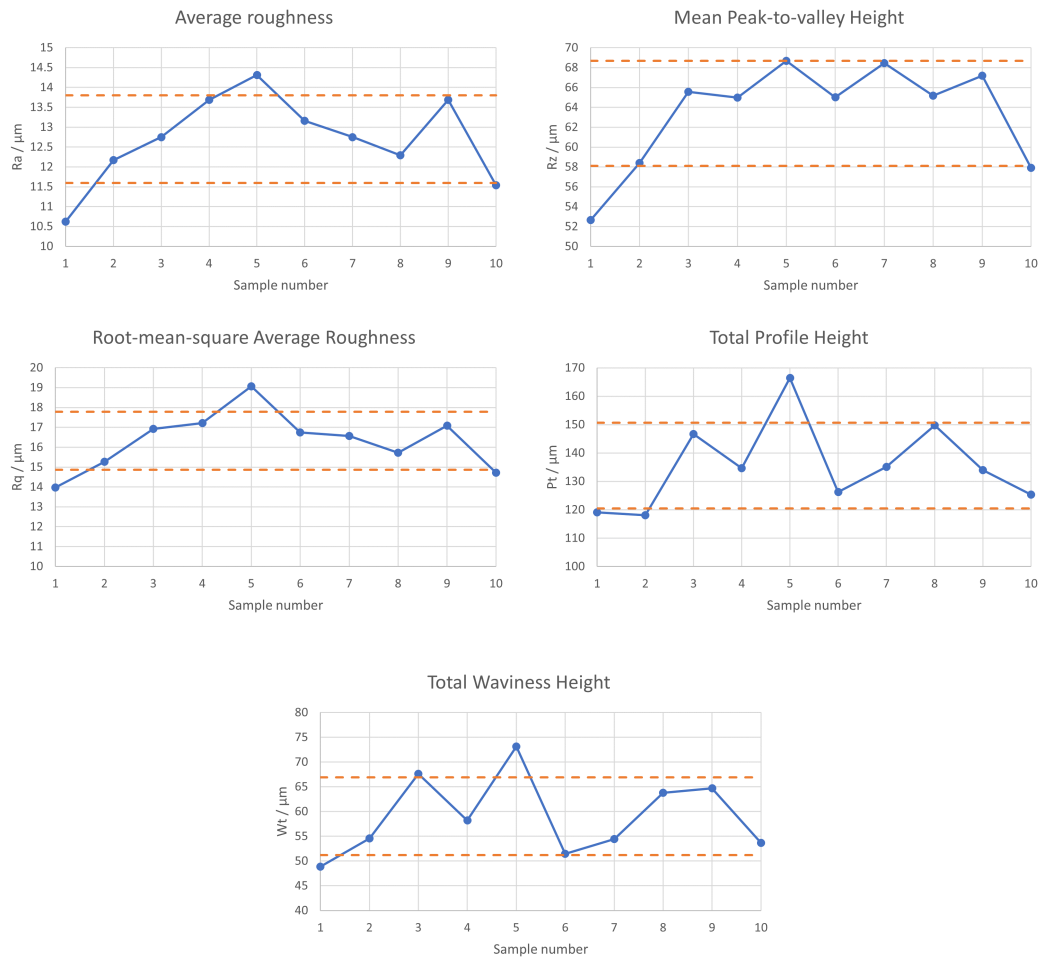


Figure 5.18: Roughness and waviness parameters measurements.

The main conclusion drawn from the analysis of all parameters is that these 3D printed inserts have a significantly higher surface roughness and waviness than the inserts machined by milling, with an approximately tenfold difference. This difference can be attributed to the inherent nature of the two manufacturing processes.

Selective laser melting involves the melting and solidification of powdered material using a laser. This process can result in a rougher surface finish due to several factors. First, the rapid solidification of the melted material can lead to uneven cooling, causing surface irregularities and roughness. Additionally, the laser energy can cause the formation of small surface imperfections or melt pool boundaries, which can contribute to increased roughness.

On the other hand, milling involves the removal of material using cutting tools. This process can produce smoother surface finishes because the cutting action removes material in a controlled manner, resulting in a more uniform surface.

5.3.3 Influence of roughness on the adhesion of the polymer with the metal.

During the injection moulding process, the adhesion between the polymer (POM or PA-GF) and the metal (aluminium) is influenced by the roughness of their surfaces. A rough surface provides more contact points and interlocking between the polymer and metal, leading to improved adhesion. It allows for better mechanical interlocking and increases the surface area for molecular interactions, resulting in stronger adhesion between the materials.

On the other hand, if the surfaces are too smooth, there may be insufficient contact between the polymer and metal, leading to weaker adhesion. The lack of surface irregularities reduces the interlocking and limits the area for molecular interactions, resulting in reduced adhesion strength.

For polymer-metal adhesion, the surface roughness of the metal substrate is typically more critical than that of the polymer. The metal surface should have a controlled roughness that provides enough mechanical interlocking and surface area for the polymer to adhere effectively. Proper surface preparation techniques, such as sandblasting were employed to enhance the roughness and promote better adhesion between the two materials.

In some cases, a surface roughness parameter such as R_a (average roughness) in the range of 0.5 to 2.0 μm can be beneficial for promoting adhesion between polymer and metal surfaces. The R_a values obtained in the previous chapters for the two types of inserts ($\approx 1 \mu\text{m}$) are within this range, and in principle, the adhesion between the two materials should not be a problem.

5.4 Gears production by injection moulding

At this stage, the mould underwent the necessary changes discussed in Chapter 4 through the manufacturing processes of milling, electrical discharge machining, and grinding, as mentioned in Chapter 3.

Before the mould is ready to enter the injection machine, it must be assembled and closed on the bench. Figure 5.19 shows the mould before closing with its respective movable and fixed sides.

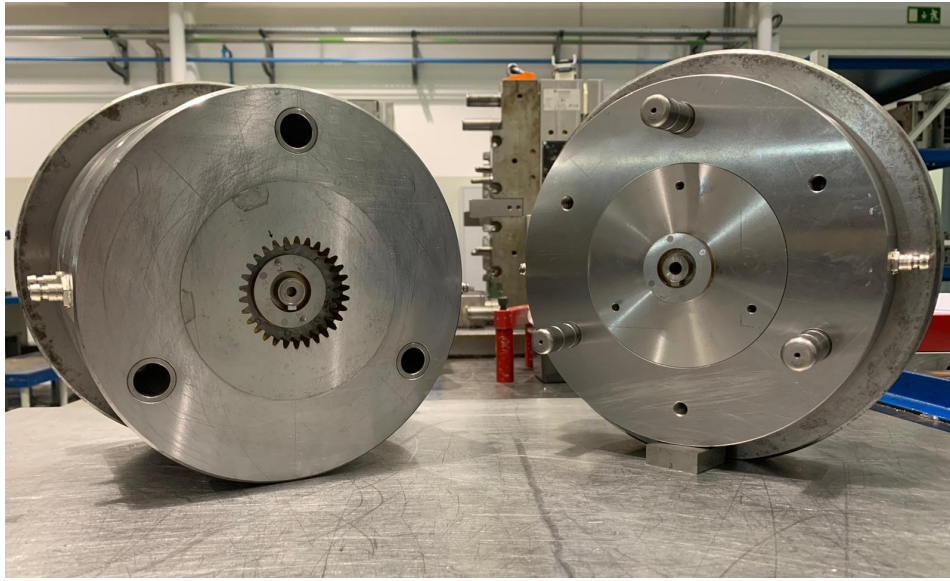


Figure 5.19: Mould configuration: movable side (left) and fixed side (right).

Furthermore, to inject the parts, selecting an injection moulding machine was necessary, taking into account the minimum mould height, maximum injection volume, and clamping force. An Arburg 370H 600 170 (A60_6) machine was chosen, which meets all the required specifications. The characteristics of this machine are presented in Table 5.2.

Table 5.2: Characteristics of the Arburg 370H 600 170 injection machine.

	Arburg 370H 600 170
Minimum mould height, mm	200
Maximum injection volume, cm ³	59
Clamping force, t	60
Maximum injection pressure, bar	2500
Centering ring, mm	125
Maximum length of the outer core pin, mm	100

Thus, by consulting the table, it is confirmed that the minimum mould height is met, as the projected mould height is 254.40 mm. The volume of the shot to be injected is approximately 25 cm³, which is within the maximum value of 59 cm³. Lastly, the mould must have a clamping force lower than the allowed force of the machine (60 tons). The clamping force is equal to the injection pressure multiplied by the projected area of the mould surface that contacts the machine's plates. Considering the maximum injection pressure, 2500 MPa, which will give the maximum clamping force that the mould can have, a force of 37.5 t was obtained.

In Figure 5.20, it is possible to observe the mould already fixed in the machine with the insert in place, as well as the mould closed and ready to inject material into the mould cavity.

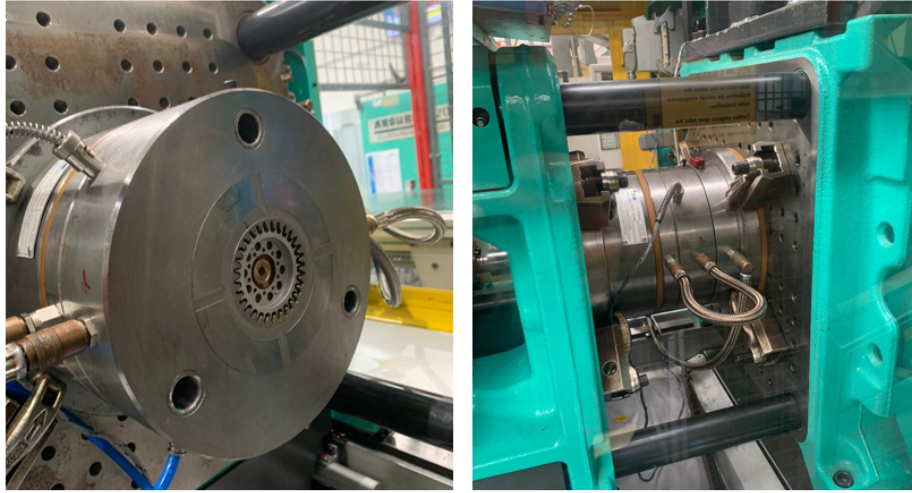


Figure 5.20: Placement of the insert in place and mould closed before injection.

Regarding the tests to be conducted, all variables present in this study were taken into account, including the two polymer materials (POM and PA-GF), the three types of inserts (no insert, 3D printed insert, and aluminium insert), and the two materials for the cavities inserts (1.2311 and AMPCO). In this way, it was possible to make a Design of Experiments (DOE).

The DOE will be used to determine the influence of the different variables on the performance and characteristics of the injection-moulded gears. By conducting a series of experiments with different combinations of these variables, it will be possible to assess their individual and combined effects on factors such as mechanical properties, dimensional accuracy, surface finish, and overall performance.

Considering all possible combinations, a total of 12 tests would have to be conducted, which was not feasible. Therefore, a reduction in the number of tests was necessary, following a logical approach and prioritizing the most important experiments for the study. Table 5.3 presents the 5 selected tests that will be carried out.

Table 5.3: Experiments conducted during the injection of the parts.

Experiment	Polymer Material	Insert Types	Cavity Inserts Material
1	POM	No insert	1.2311
2	POM	No insert	AMPCO
3	POM	Both inserts	AMPCO
4	PA-GF	No insert	AMPCO
5	PA-GF	Both inserts	AMPCO

It is important to mention that Experiments 4 and 5 will not be subjected to analysis in this thesis due to delays in the arrival of raw materials to the company, which prevented the execution of these experiments within the intended timeline.

Therefore, the first two tests will serve as a comparison between the two materials for the inserts, and then it was decided to use AMPCO in the remaining tests instead of 1.2311 steel due to various factors. Firstly, AMPCO material offers better thermal conductivity compared to 1.2311 steel. This is important in injection moulding as it helps in efficient heat transfer and cooling of the mould, resulting in shorter cycle times and improved productivity.

Additionally, AMPCO exhibits excellent wear resistance and high hardness, making it suitable for withstanding the abrasive and high-pressure conditions during the injection moulding process. This helps in prolonging the lifespan of the inserts and reducing the need for frequent replacements.

In the remaining experiments, the other variables were changed, either the type of insert or the polymer material.

In the first trial to define all the parameters, the initial injections were performed by gradually increasing the packing time by 1 second for each piece until the weight of the gears reached a plateau, in order to optimize the injection parameters and ensure complete filling of the part. The time at which the weight of the part stopped increasing was at 23 seconds of packing time.

After that, it was decided to inject the parts using three different cooling times, allowing the parts to cool and contract for several hours at the end of the injection in order to make a comparison and observe the influence of the cooling time on some general dimensions of the gear.

However, the variation in cooling time showed insignificant differences, as shown in Table 5.4, which presents some general dimensions of a sample of 5 gears for each of the three cooling times used, 20, 25 and 30 seconds with a mould temperature of 100 °C.

Table 5.4: Gears general dimensions after the 1st experiment.

Parameters	$t_{cool} = 20 \text{ s}$	$t_{cool} = 25 \text{ s}$	$t_{cool} = 30 \text{ s}$
Face width, b / mm	15.002	14.998	15.004
Addendum diameter, d_a / mm	63.095	63.143	63.105
Internal diameter, d_i / mm	15.975	15.998	15.992
Base tangent length, W_k / mm	21.257	21.270	21.275

These results obtained can have some possible explanations, the main one being that other process parameters, such as melt temperature, mould temperature and injection pressure, may be dominant in determining the final dimensions of the gears, masking the influence of cooling time.

In this way, in this second trial, the same cooling time (25 s) was maintained, and it was decided to inject the parts at 3 different mould temperatures, 80, 100, and 120 °C to verify the influence of this variation on the part shrinkage and consequently on the final dimensions of the gear. Figure 5.21 illustrates three gears that were injected at different mould temperatures.

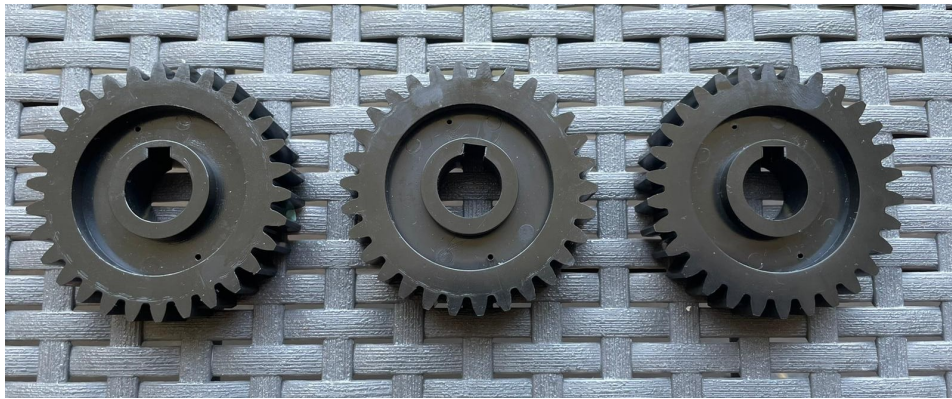


Figure 5.21: Manufactured gears at 80 °C (left), 100 °C (center) and 120 °C (right) mould temperature.

In Table 5.5, the overall dimensions of a sample of 5 gears for each case after injection at three different mould temperatures are presented.

Table 5.5: Gears general dimensions after the 2nd experiment.

Parameters	$T_m = 80\text{ }^{\circ}\text{C}$	$T_m = 100\text{ }^{\circ}\text{C}$	$T_m = 120\text{ }^{\circ}\text{C}$
Face width, b / mm	14.996	14.956	14.920
Addendum diameter, d_a / mm	63.528	63.245	63.004
Internal diameter, d_i / mm	16.042	16.001	15.994
Base tangent length, W_k / mm	21.376	21.284	21.202

As expected, the lower the mould temperature, the less shrinkage. POM is known to have significant shrinkage during cooling after injection. Decreasing the mould temperature led to faster material cooling and, consequently, less shrinkage of the part. Therefore, the gear injected at a mould temperature of $80\text{ }^{\circ}\text{C}$ had come closer to the intended dimensions as there was less shrinkage.

However, POM is a semi-crystalline polymer and undergoes a process of crystallization during cooling. The mould temperature can influence the speed and extent of crystallization. Proper crystallization of POM can improve its mechanical properties, such as stiffness and impact resistance. It is possible that the mould temperature of $120\text{ }^{\circ}\text{C}$ provided more complete crystallization, resulting in better mechanical properties compared to the gear injected at $80\text{ }^{\circ}\text{C}$. Additionally, the difference in mechanical properties can also be related to residual stress within the gears. Faster cooling can lead to greater development of residual stress, which can affect the strength and durability of the final product.

Another important aspect observed in the dimensional control of the gears was that, for the three mould temperatures used in the injection process, the internal diameter of the gear remained practically the same, while the addendum diameter showed significant differences between different mould temperatures. The proximity of the injection gate plays a significant role in minimizing the expected contraction effect. This is attributed to the fact that when the material is injected near the entry point, it remains in a molten and high-temperature state, resulting in reduced material shrinkage during the subsequent cooling process.

In the region near the injection gate, where the material is in its highest temperature and most fluid state, the material flow is more unrestricted, leading to a complete filling of the mould cavity. This results in less shrinkage, as the molten material have fewer flow constraints and occupies the entire mould space.

Conversely, as we move towards the more distant regions of the mould cavity, the material has already undergone cooling and partial solidification. Consequently, these areas are

subject to greater shrinkage and contraction due to reduced mobility, limiting their ability to conform precisely to the desired dimensions.

Comparing the temperatures obtained at specific locations on the surface of the 1.2311 steel inserts at the end of the first test and the AMPCO 83 inserts at the end of the second test, for a set mould temperature of 100 °C, it was observed that the AMPCO 83 inserts achieved temperatures closer to the theoretical value of 100 °C. This indicates that the AMPCO 83 inserts have a higher capacity to absorb and reach the desired mould temperature more efficiently compared to the inserts made of 1.2311 steel. The lack of control over the ambient temperature during the tests may also have contributed to the observed insignificant difference in temperatures between the two types of inserts.

In terms of cycle times, the total cycle time was approximately 50.5 s, with an injection time of 2.41 s, a packing time of 23 s, and a cooling time of 25 s. These times are significantly longer than those obtained in subchapter 4.5, which was expected because the filling simulation software defaults to recommending the packing time, and it calculates the cooling time based on the part and material data. Since the filling simulation is performed prior to the actual injection of parts, it is normal to encounter these differences in time.

In Experiment 3, both machined and 3D-printed inserts were tested for the gears. Before starting the injection of the part, it was necessary to reduce the injected volume considering the volume of the insert. Since there were limited available inserts and no spares to risk damaging, the same compacting and cooling times were used, despite the expectation of shorter times for this particular experiment. Figure 5.22 illustrates two hybrid gears after the injection process, one with a machined insert and the other with a 3D-printed insert.

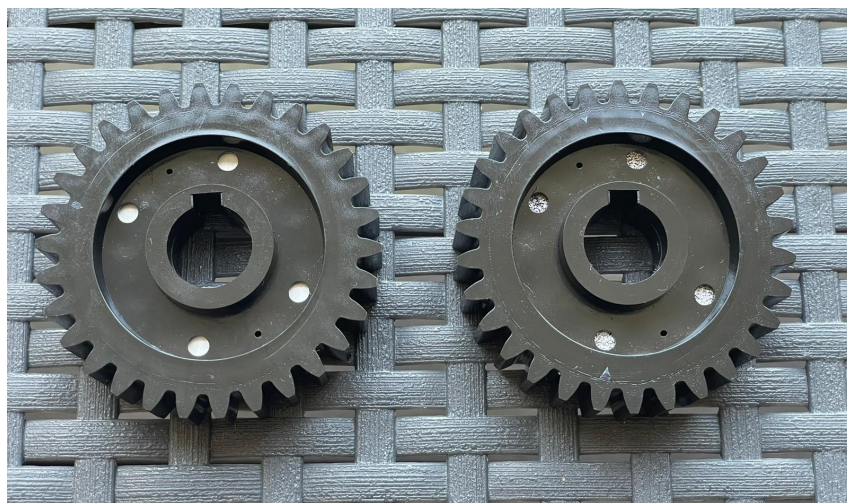


Figure 5.22: Manufactured hybrid gears with machined insert (left) and with 3D-printed insert (right).

After producing the parts and allowing them to cool and shrink for a few hours, a dimensional inspection of these gears was conducted. The purpose of this inspection was to evaluate the achieved dimensions and compare them to the desired specifications. The obtained dimensions can be observed in Table 5.6.

Table 5.6: Gears general dimensions after the 3rd experiment.

Parameters	Machined insert	3D-printed insert
Face width, b / mm	15.154	15.156
Addendum diameter, d_a / mm	64.630	64.620
Internal diameter, d_i / mm	16.057	16.078
Base tangent length, W_k / mm	21.762	21.748

Comparing the dimensions of the gears for the two types of inserts, it is observed that the obtained values are extremely similar. In comparison to the gears without inserts, the opposite is true, meaning that the dimensions of the gear are larger than those that were designed. In other words, the insert is causing the gear to contract even less than expected. One possible justification for this is that the presence of the insert may have influenced the shrinkage pattern of the plastic material during cooling. The insert, due to its composition or geometric characteristics, could have caused differential shrinkage in the gear, resulting in a larger addendum diameter.

The base tangent length (W_k) is an important parameter in gear design. It represents the length of the line segment where the base circle of a gear tooth intersects with the line that is tangent to the involute curve of the tooth profile. In other words, it is the distance along the base circle from the point where the tooth profile starts to curve inward. Figure 5.23 shows a visual representation of this parameter for a number of teeth spanned (k) equal to 3 [54].

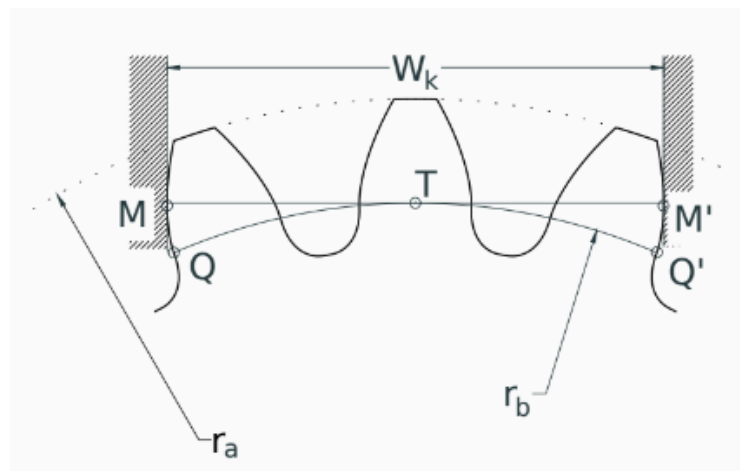


Figure 5.23: Base tangent length over a number of teeth spanned equal to 3 [54].

The equation for the base tangent length for standard spur gears (without correction) is given by the formula in Equation 5.4 [54].

$$W_k = m_n \cdot \cos \alpha \cdot \left[(k-1) \cdot \pi + \frac{s}{m_n} + z \cdot \text{inv} \alpha \right] \quad (5.4)$$

Before calculating W_k , it is necessary to determine the number of teeth spanned by a gear, k , required for this measurement. This parameter is given by Equation 5.5 [54].

$$k = \frac{z \cdot \alpha}{\pi} + \frac{1}{2} \quad (5.5)$$

The number of teeth spanned by a gear was 3.83. Since k must be an integer, 4 teeth will be required for the calculation of W_k . Therefore, the theoretical base tangent length is equal to 21.505 mm. As seen previously, this value is higher than what was obtained for gears without an insert and lower for gears with an insert.

Thus, it is possible to determine the profile shift corresponding to the deviation of the obtained base tangent length from the theoretical value. The profile shift is a measure that indicates the displacement of the tooth profile relative to the pitch line of the gear. By using the equation for base tangent length with profile shift correction (Equation 5.6), the value of the profile shift (x) can be obtained.

$$W_{k,measured} = m_n \cdot \cos \alpha \cdot \left[(k-1) \cdot \pi + \frac{\frac{\pi \cdot m_n}{2} + 2 \cdot x \cdot m_n \cdot \tan \alpha}{m_n} + z \cdot \text{inv} \alpha \right] \quad (5.6)$$

A sample of 5 gears was used to measure the base tangent length. Three measurements were taken on three random areas of the teeth of each gear. The average of the 3 measurements was obtained for each gear, and the average of the 5 samples was calculated. In Table 5.7, the calculated profile shift values corresponding to the observed deviation in base tangent length are presented for a standard gear and a hybrid gear under the same injection conditions, mould temperature of 100 °C, and cooling time of 25 s.

Table 5.7: Profile shift coefficients corresponding to the observed deviations in the base tangent length.

Parameters	Theoretic	Standard Gear	Hybrid Gear
Base tangent length, W_k / mm	21.505	21.284	21.762
Profile shift coefficient, x	0.000	-0.164	0.185

The profile shift coefficient is negative for the standard gear, meaning the profile shifts inward toward the pitch line and is positive for the hybrid gear, which means the profile shifts outward from the pitch line. Based on the calculated profile shift, adjustments can

be made to the design, manufacturing process, or other necessary corrections in order to achieve the desired tooth profile.

Finally, the comparison of the tooth profiles of gears produced with the designed gear profile is carried out to assess the conformity and quality of the manufactured part. The designed gear tooth profile is typically based on theoretical calculations and engineering designs, representing the ideal geometry of the teeth.

By comparing the actual tooth profiles of gears manufactured by injection moulding with the theoretical profile, it is possible to identify any deviations and irregularities. These differences can be caused by various factors such as manufacturing tolerances, deformations during cooling, mould wear, or variations in injection parameters.

The tooth profile of the produced gears was extracted using a profile projector machine, and then the obtained data points were imported into SolidWorks. There the tooth profile designed was overlaid with the tooth profile of the produced gear and the result is shown in Figure 5.24.

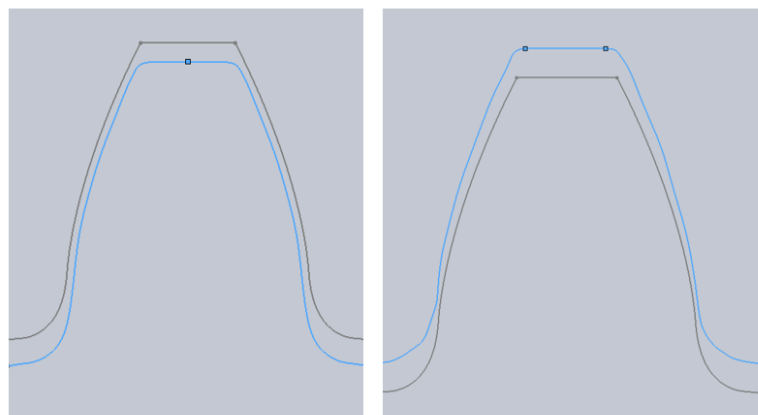


Figure 5.24: Comparison of the tooth profiles of the produced gears (blue), without insert (left), and with insert (right), with the profile of the designed gear (grey).

In gears without inserts, it was expected that the tooth profile would be shorter at the tooth tip. This is because the mould was designed with an insert, and the applied shrinkage value took into account the presence of the insert. Consequently, the gears produced without inserts have significantly thicker plastic material sections, necessitating an adjustment in the shrinkage value and modifications to the mould.

In gears with insert, the opposite occurs, meaning that the tooth profile is larger than the projected dimensions. This difference is more noticeable in the tooth tip, indicating that the

gear teeth are facing greater difficulty in contracting in the radial direction when the insert is present. However, this difference is not expected to be problematic in future gear usage tests, as long as the remaining tooth profile is respected. It would be sufficient to make an adjustment to the center distance of the gears to prevent potential gear failure modes.

Furthermore, it is worth noting that the injected gears exhibit a small radius at the tooth tip in their tooth profile. This is attributed to the EDM wire cut machining of insert 200, which has the tooth profile, where the wire has a radius of 0.125 mm. During the erosion process, as the wire follows the contour of the teeth, it imparts this slight radius to the gear teeth. Despite the shorter or larger tooth length, the tooth shape remains preserved, indicating that there were no deviations or deformations during the injection of the parts, and the machining of the tooth profile was executed with precision.

Chapter 6

Definition of Operating Conditions for Gear Testing

6.1 Introduction

The load-carrying capacity of polymer gears is influenced by several factors, including the material properties, gear design parameters, operating conditions, and lubrication. Compared to metal gears, polymer gears exhibit different characteristics and behaviours, making their load-carrying capacity a subject of interest and study.

Polymer gears offer advantages such as reduced weight, corrosion resistance, quieter operation, and cost-effectiveness. However, their load-carrying capacity is typically lower than metal gears due to their lower modulus of elasticity and lower strength.

To optimize the load-carrying capacity of polymer gears, researchers and engineers focus on various aspects, including material selection, gear design optimization, surface treatments, and lubrication techniques. Understanding the limitations and capabilities of polymer materials and identifying strategies to enhance their load-carrying capacity is crucial for successfully implementing polymer gears in diverse applications.

In this chapter, a load-carrying capacity calculation is performed using the VDI 2736. Only POM gears were considered in this calculation due to the lack of information about the

conductive material, PA6-GF. These results serve as a basis for the experimental campaign that will be carried out.

6.2 VDI 2736

For polymeric gears, the calculation can be done using VDI 2736 standard (based on DIN 3990) [11]. However, other standards, such as JIS B 1759 or BS 6168, can also be used.

Polymers have a mechanical strength dependent on the operating temperature and loading duration. Moreover, their thermal conductivity is poor. These material characteristics require a different approach to the standard load-capacity calculation, i.e. the first step will always be to determine the gear's operating temperature, a task that can be challenging to solve.

Subsequently, as in ISO 6336 standard, the load capacity for bending and fatigue in the tooth root and for surface pressure must be calculated. Then, the load capacity for frictional wear is determined, and the tooth deformation is calculated. This series of calculations required for gear dimensioning is shown in Figure 6.1.

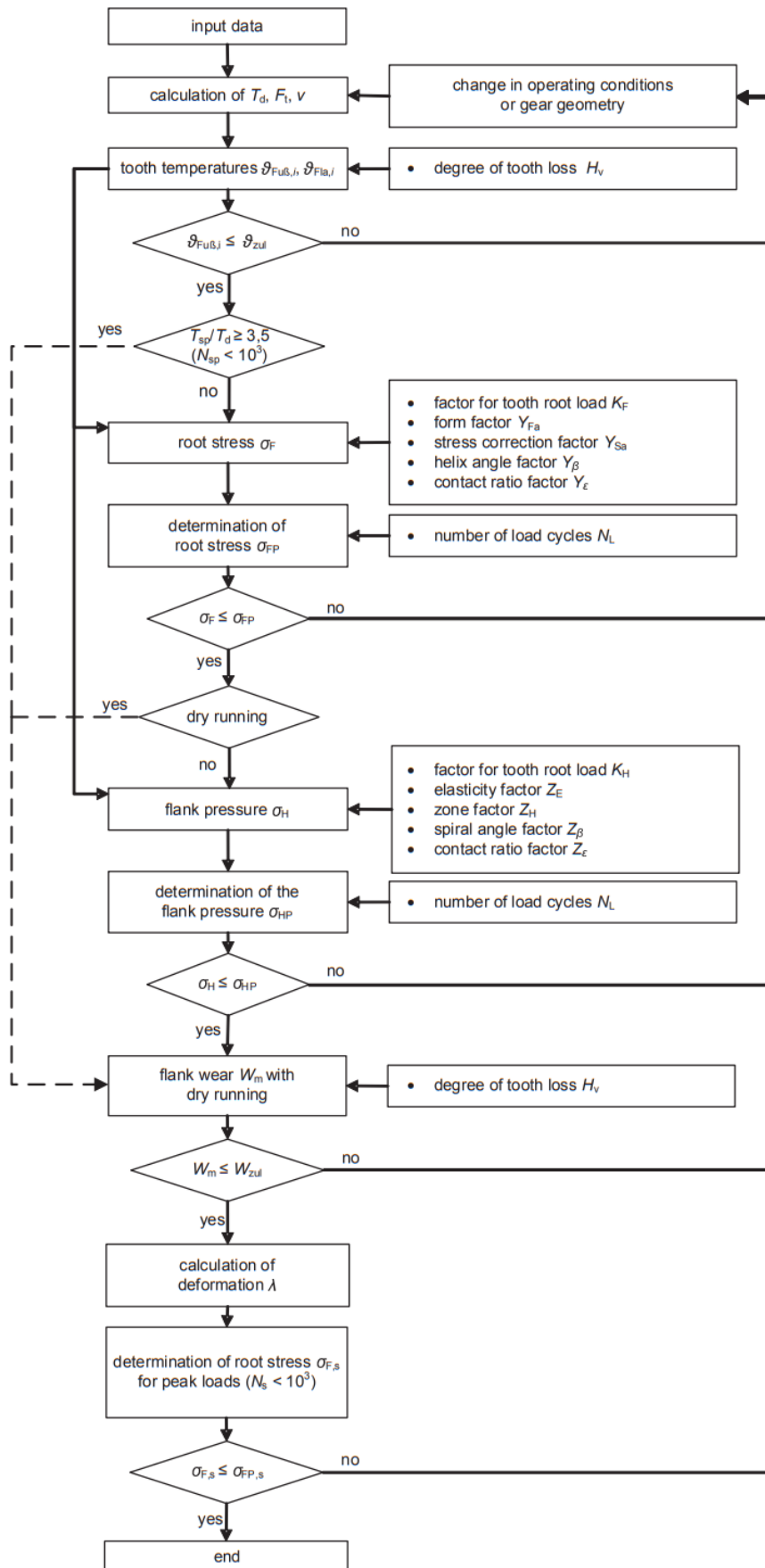


Figure 6.1: Simplified flow chart for calculation of a plastic spur and helical gear [55].

The geometry and materials of the gears are completely known. This information has already been presented in Table 4.1 and Table 4.2, respectively. Therefore, it is intended to determine the maximum number of load cycles supported by the gear using the calculation procedure described in the VDI 2736 directive. The excel file used to perform the calculations according to the VDI 2736 guideline is located in Appendix C.

The operating conditions are also essential (rotational speed; torque; ambient temperature; lubrication type; relative tooth engagement time; mechanism housing geometry).

These parameters allow for the calculation of other essential quantities for gear design. The input data formulas are presented in Appendix D.

6.2.1 Working conditions

The rotational speed and torque estimation for the gear wheels was based on K. Mao's work [49], who tested the gears at a rotational speed of 1000 rpm and a load of 9 Nm. Thus, these load values were used to calculate all temperature and stress values presented in the following subchapters. Subsequently, in order to compare the number of load cycles with the applied load values, different rotational speed and torque values were applied, ranging from 500 to 1500 rpm and 7 to 11 Nm, respectively.

In the first iteration for calculating the number of load cycles allowed by the mechanism, the absence of external lubrication is assumed. However, in Section 6.3 the influence of using grease lubrication in the system is discussed.

Tooth engagement time, ED , is defined as the ratio of the actual operating time to a reference time of ten minutes. It provides a standardized measure to evaluate the durability and performance of the gear under specific operating conditions. In this case, it can be considered equal to 1 (steady-state condition). Otherwise, it is common to consider values between zero and one and observe the heating curve of the gear.

The heat-dissipating surface of the mechanism housing, A_G , refers to the housing area or surfaces designed to facilitate the dissipation of heat generated within the mechanism. This factor does not apply in the case of non-enclosed mechanisms. Therefore, in this calculation, an open-air mechanism was considered, and so A_G is equal to zero.

6.2.2 Tooth operating temperatures

There are two ways to produce heat in polymeric gears, which raises the temperature of the tooth. Heat is produced by friction due to rolling contact in the pitch circle and sliding at the tooth's head and root. The second heat source is due to energy lost through hysteresis [56].

The region with the highest temperature will be on the flank, close to the pitch diameter, because it is simpler to dissipate heat through the tip and root of the tooth, as shown in Figure 6.2.

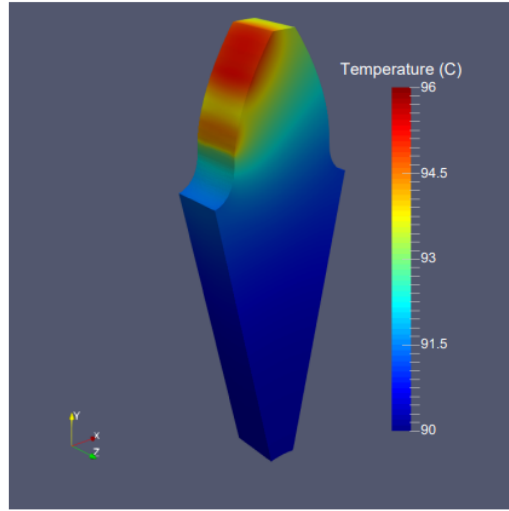


Figure 6.2: Temperature distribution on the gear tooth [5].

According to the VDI 2736 directive, there are two areas of the tooth where it is essential to consider its temperature: the flank and the tooth root, because it is where higher stresses are expected. Starting with the tooth root, the operating temperature is calculated using Equation 6.1.

$$\vartheta_{Fu\beta} \approx \vartheta_0 + P \cdot \mu \cdot H_V \cdot \left(\frac{k_{\vartheta Fu\beta}}{b \cdot z \cdot (v_t \cdot m_n)^{0.75}} + \frac{R_{\lambda,G}}{A_G} \right) \cdot ED^{0.64} \quad (6.1)$$

where:

- ϑ_0 is the ambient temperature, in °C;
- P is the nominal output, in W;
- μ is the coefficient of friction, that can be selected consulting Table 6.1;
- H_V is the degree of tooth loss, and when $1 \leq \epsilon_\alpha \leq 2$, it is defined by Equation 6.2 :

$$H_V = \frac{\pi \cdot (u + 1)}{z_2 \cdot \cos \beta_b} \cdot (1 - \epsilon_1 - \epsilon_2 + \epsilon_1^2 + \epsilon_2^2) \quad (6.2)$$

where: u is the gear ratio such that $u = z_2/z_1 \geq 1$; β_b is the helix angle at the base circle; ϵ_1 and ϵ_2 are the partial contact ratios of pinion and wheel. These formulas are given in Appendix D;

- $k_{\vartheta Fu\beta}$ is the heat transfer coefficient of the plastic gear, and its value can be consulted in Table 6.2;

- b is the face width, in mm;
- z is the number of teeth on the plastic gear;
- v_t is the tangential velocity, in m s^{-1} ;
- m_n is the normal module, in mm;
- $R_{\lambda,G}$ is the heat transfer resistance of the mechanism housing, whose value can be chosen in Table 6.3;
- A_G is the heat dissipating surface of the mechanism housing, in m^2 (does not apply in the case of non-enclosed mechanisms);
- ED is the relative tooth engagement time, corresponds to the total tooth engagement time in minutes within a period of ten minutes;

For the flank temperature, which can reduce the tooth flank load-carrying capacity, it is calculated by Equation 6.3.

$$\vartheta_{Fla} \approx \vartheta_0 + P \cdot \mu \cdot H_V \cdot \left(\frac{k_{\vartheta,Fla}}{b \cdot z \cdot (v_t \cdot m_n)^{0.75}} + \frac{R_{\lambda,G}}{A_G} \right) \cdot ED^{0.64} \leq \vartheta_{zul} \quad (6.3)$$

being ϑ_{zul} the permissible temperature for the respective thermoplastic, [11]. And the value of $k_{\vartheta,Fla}$ can be found in Table 6.2.

Table 6.1: Coefficient of friction μ , [55].

Lubrication	Pairing	μ
Circulating oil	All	0.04
Oil mist	All	0.07
Grease	All	0.09
Dry	plastic/steel	0.2
	PA/PA	0.4
	POM/POM	0.28
	POM/PA	0.18
	PA/PBT	0.35
	POM/PBT	0.18

Table 6.2: Heat transfer coefficients of the plastic gear ($k_{\vartheta Fu\beta}$ and $k_{\vartheta Fla}$), in $\text{K}(\text{m/s})^{0.75} \text{mm}^{1.75}/\text{W}$, [55].

Lubrication	Pairing	$k_{\vartheta Fu\beta}$	$k_{\vartheta Fla}$
Circulating oil	All	0	0
Dry/Grease/Oil mist	plastic/plastic	2100	9000
	plastic/steel	900	6300

Table 6.3: Heat transfer resistance of the mechanism housing ($R_{\lambda,G}$), in $\text{K m}^2 \text{W}^{-1}$, [55].

Mechanism	$R_{\lambda,G}$
Open with unimpeded entry of air	0
Partially open housing	0.015...0.045
Closed housing	0.060

All the necessary quantities for this calculation can be found in Table 6.4.

Table 6.4: Parameters for calculating the operating temperatures of each gear.

Parameters	z_1/z_2
$v_t / \text{m s}^{-1}$	3.14
P / W	942.5
ED	1
μ	0.28
H_v	0.149
$k_{\vartheta, Fu\beta} / \text{K(m/s)}^{0.75} \text{mm}^{1.75} / \text{W}$	2100
$k_{\vartheta, Fla} / \text{K(m/s)}^{0.75} \text{mm}^{1.75} / \text{W}$	9000
$R_{\lambda,G} / \text{K m}^2 \text{W}^{-1}$	0
A_G / m^2	0

The transmitted power is a function of torque and rotational speed and is characterized by Equation 6.4.

$$P = \frac{T_d \cdot 2\pi \cdot n}{60} \quad (6.4)$$

The tangential velocity, for element z_1 is given by Equation 6.5.

$$v_t = \frac{d_1 \cdot \pi \cdot n}{60 \cdot 10^3} \quad (6.5)$$

Considering 20°C as the ambient temperature, the operating temperatures of the root and flank tooth can be obtained from Table 6.5.

Table 6.5: Operating temperatures of the gears teeth.

Temperature	z_1	z_2
$\vartheta_{Fu\beta}$	66.4	66.4
ϑ_{Fla}	218.7	218.7

The VDI 2736 directive considers that the maximum permissible temperature for POM is 100 °C. By consulting Table 6.5, it can be seen that this limit was exceeded at the tooth flank temperature. The reasons behind the high operating temperature could be the excessive friction, lack of lubrication or prolonged operation under high loads.

In this case, it is recommended to proceed with calculations using the maximum temperature value of 100 °C. This ensures that the calculations are conservative and account for the worst-case scenario in terms of effects on the POM gears.

6.2.3 Tooth root load-carrying capacity

Starting with the root stress during force application at the tooth tip and the assumption of bending stress alone, a check calculation of tooth root load-carrying capacity was performed. The tooth root stress is given by Equation 6.6.

$$\sigma_F = K_F \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_\varepsilon \cdot Y_\beta \cdot \frac{F_t}{b \cdot m_n} \leq \sigma_{FP} \quad (6.6)$$

where:

- K_F is the factor for tooth root load and is determined by Equation 6.7:

$$K_F = K_A \cdot K_\varepsilon \cdot K_{F\beta} \cdot K_{F\alpha} \quad (6.7)$$

K_A is the application factor, which is obtained from dynamic analysis, load spectra or specified operating conditions. In the case of having an electric motor activation, $K_A \approx 1$ to 1.25. K_ε is the dynamic factor; $K_{F\beta}$ is the width factor; $K_{F\alpha}$ is the face factor. Taking into account the elastic properties of the thermoplastics, the factor K_F can be approximated by Equation 6.8.

$$K_F \approx K_A \quad (6.8)$$

- Y_{Fa} is the form factor, that considers the influence of the tooth shape on the nominal bending stress at load application at the tooth tip. The values for this factor are taken from Figure 6.3 a). For spur gears, z_n should be equal to z , while for helical gears $z_n = z / (\cos^2 \beta_b \cdot \cos \beta) \approx z / \cos^3 \beta$.
- Y_{Sa} is the stress correction factor, which takes into account the stress concentration arising from the notch effect for load application at the tooth tip. The values for Y_{Sa} are shown in Figure 6.3 b).
- Y_ε is the contact ratio factor, that corrects the excessively conservative approach that considers the application of all the load on the tooth's tip. This factor depends on the

contact ratio, ε_α , and is determined by Equation 6.9.

$$Y_\varepsilon = 0.25 + 0.75/\varepsilon_\alpha \quad (6.9)$$

- Y_β is the helix angle factor, that considers the more favourable meshing conditions found with helical teeth. Equation 6.10 allows to determine this factor, dependent on the overlap contact ratio, ε_β .

$$Y_\beta = 1 - \varepsilon_\beta \cdot \beta / 120^\circ \quad (6.10)$$

When the helix angle $\beta > 30^\circ$, set $\beta = 30^\circ$ and when $\varepsilon_\beta > 1$, set $\varepsilon_\beta = 1$.

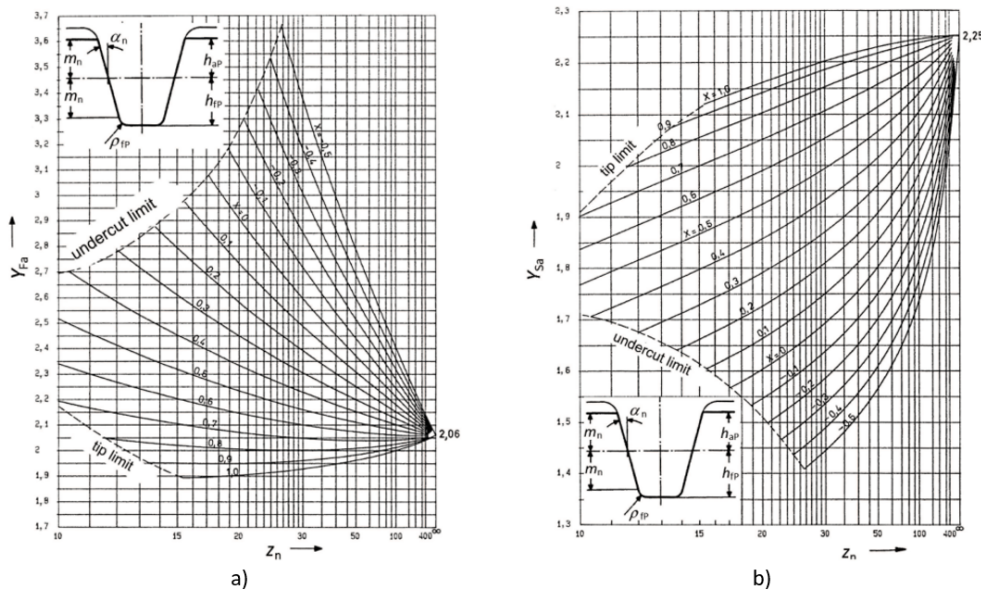


Figure 6.3: a) Form Factor; b) Stress concentration factor; [55].

The permissible root stress is given by Equation 6.11.

$$\sigma_{FP} = \sigma_{FG}/S_{Fmin} \quad (6.11)$$

where:

- S_{Fmin} is the required minimum safety factor, generally equal to 2.0.
- σ_{FG} is the maximum root strength with $\sigma_{FG} = Y_{St} \cdot \sigma_{FlimN}$; Y_{St} is the stress correction factor and is approximately equal to 2.0. σ_{FlimN} is the fatigue strength under pulsating stress, it has distinct values for different thermoplastics as a function of the required number of load cycles.

As it is intended to calculate the required number of load cycles for specific operating conditions and considering $S_{Fmin} \approx Y_{St} \approx 2.0$, a simplification can be made through the following equation:

$$\sigma_F \leq \sigma_{FP} = \frac{\sigma_{FG}}{S_{Fmin}} = \frac{\sigma_{FlimN} \cdot Y_{St}}{S_{Fmin}} = \sigma_{FlimN} \quad (6.12)$$

Thus, for POM, the number of load cycles can be obtained from Figure 6.4, which is governed by the formula given in Equation 6.14.

$$\sigma_{FlimN} = 26 - 0.0025 \cdot \vartheta_{Fu\beta}^2 + 400 \cdot N_L^{-0.2} \quad (6.13)$$

$$N_L = \left(\frac{400}{\sigma_{FlimN} - 26 + 0.0025 \cdot \vartheta_{Fu\beta}^2} \right)^{1/0.2} \quad (6.14)$$

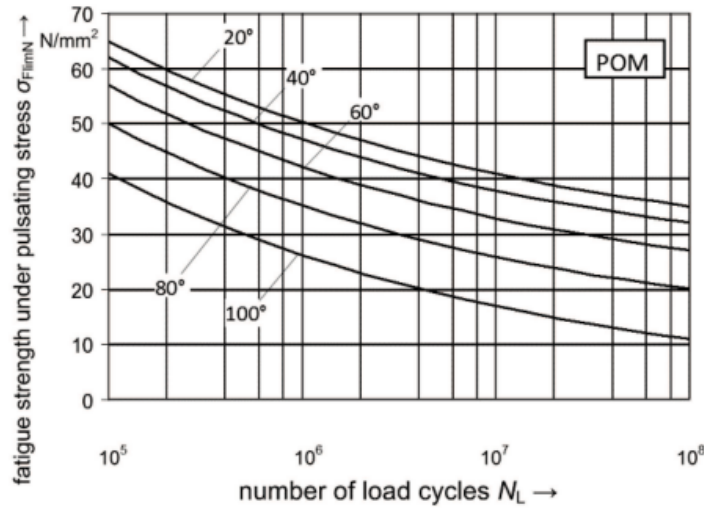


Figure 6.4: Values for fatigue strength under pulsating stress as function of tooth root temperature, $\vartheta_{Fu\beta}$, and the number of load cycles, N_L , [55].

Table ?? shows the values of all the coefficients necessary to calculate the number of load cycles for each gear wheel based on the tooth root stress.

Table 6.6: Tooth root stress calculation.

Parameters	z_1/z_2
K_f	1.25
Y_{Fa}	2.6
Y_{Sa}	1.69
Y_ϵ	0.704
Y_β	1.00
F_t / N	300
σ_F / MPa	38.64

The tangential force given as a function of torque and pitch diameter is determined by Equation .

$$F_t = \frac{2 \cdot 10^3 \cdot T_d}{d_i} \quad (6.15)$$

Considering that $\sigma_{F\lim N} = \sigma_F$ and using the tooth root temperature obtained in Table 6.5, it was possible to calculate a number of load cycles of $1.38 \cdot 10^6$.

Then, in order to establish a relationship between the number of load cycles and the operating conditions, rotational speed and torque, other load values were tried on the gears, which made it possible to obtain Figure 6.5.

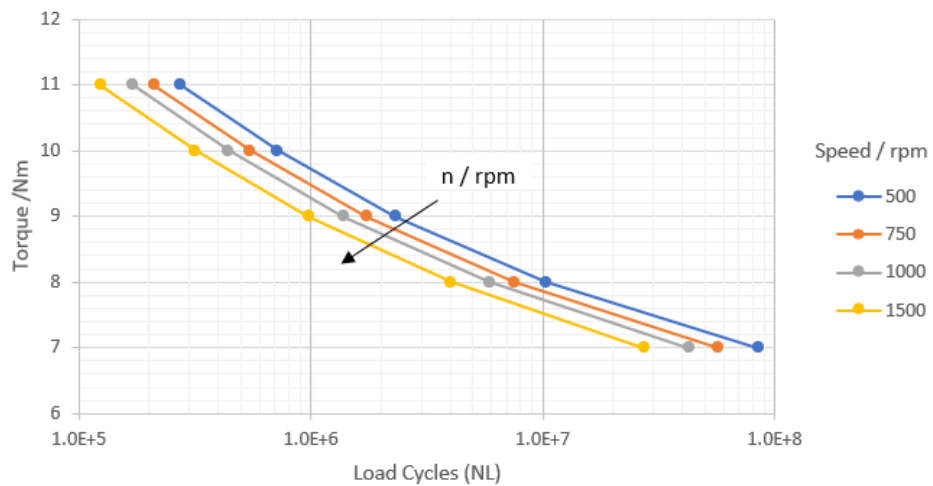


Figure 6.5: Number of load cycles as function of rotational speed and Torque, through tooth root calculation.

By analysing Figure 6.5, the number of load cycles a system can withstand is inversely proportional to the applied torque. Higher torque levels can lead to increased stress on the components, potentially reducing their fatigue life and the number of load cycles they can endure.

Similarly, the rotational speed also plays a role in determining the number of load cycles. Higher rotational speeds can result in increased dynamic forces and vibrations within the system, which can contribute to an increase in the operating temperatures and a decrease in the number of cycles. It is important to note that these relations may vary depending on the specific gear system, material properties, and operating conditions.

6.2.4 Tooth flank load-carrying capacity

By observing Figure 6.1, it can be noticed that when there is no external lubrication, as in this case, the calculation of tooth flank resistance can be ignored. However, if it is necessary to determine the number of load cycles for the gear, Hertz's contact pressure is used to calculate the load capacity on the tooth flank. Equation 6.16 is thus applicable to estimate the contact stress.

$$\sigma_H = Z_E \cdot Z_H \cdot Z_\epsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t \cdot K_H}{b_w \cdot d_1} \cdot \frac{u+1}{u}} \leq \sigma_{FH} \quad (6.16)$$

where:

- Z_E is the elasticity factor and takes into account the modulus of elasticity E and the Poisson's ratios ν of the materials of gears 1 and 2. It is calculated as follows in Equation 6.17.

$$Z_E = \sqrt{\frac{1}{\pi \cdot \left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)}} \quad (6.17)$$

- Z_H is the zone factor, that considers the flank curvature radii at the contact point. For externally or internally toothed cylindrical gears with a pressure angle $\alpha_n = 20^\circ$, Z_H as a function of the helix angle β at the reference cylinder and also of $(x_1 + x_2)/(z_1 + z_2)$ can be obtained from Figure 6.6.

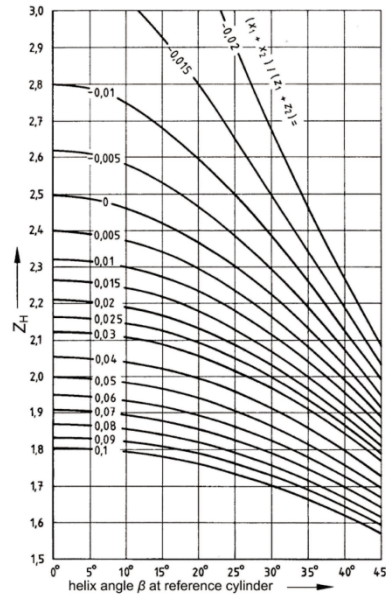


Figure 6.6: Zone factor, Z_H , for $\alpha = 20^\circ$, [55].

- Z_ϵ is the contact ratio factor, that takes into account the influence of the transverse contact ratio ϵ_α , the overlap contact ratio ϵ_β and the helix angle β . Its numerical value is given by Equation 6.18, if $\epsilon_\beta < 1$ or by Equation 6.19, if $\epsilon_\beta \geq 1$.

$$Z_\epsilon = \sqrt{\left(\frac{4 - \epsilon_\alpha}{3}\right) \cdot (1 - \epsilon_\beta) + \frac{\epsilon_\beta}{\epsilon_\alpha}} \quad (6.18)$$

$$Z_\epsilon = \sqrt{\frac{1}{\epsilon_\alpha}} \quad (6.19)$$

- Z_β is the spiral angle factor that considers the influence of the helix angle β which is not fully covered in Z_ϵ . It takes a value close to the unit.
- K_H is the factor for tooth flank loading and is determined in the same way as K_F in Equation 6.7.

The permissible flank pressure σ_{HP} is expressed as

$$\sigma_{HP} = \sigma_{HlimN} \cdot Z_R / S_{Hmin} \quad (6.20)$$

where:

- σ_{HlimN} is the rolling contact fatigue strength.
- Z_R is the surface roughness factor and in the case of plastic gears, $Z_R \approx 1$.

- S_{Hmin} is the required minimum safety factor, generally equal to 1.4 for continuous operation. The minimum safety factor should be defined as a function of the risk of damage and the accuracy of assumptions.

Considering $S_{Hmin} \approx Z_R \approx 1$, a similar simplification to the one for the tooth root load-carrying capacity can be made.

$$\sigma_H \leq \sigma_{HP} = \frac{\sigma_{HlimN} \cdot Z_R}{S_{Hmin}} = \sigma_{HlimN} \quad (6.21)$$

The VDI 2736 guideline considers that it is generally not necessary to analyze the load capacity of the tooth flank for gears made of POM. Therefore it does not provide the required mechanical properties for this purpose [55]. Thus, the tooth flank load capacity considered in this study is defined by VDI 2545, which assumes the tooth flank strength of POM as 90% of the values for PA66 when no lubricant is used.

Therefore, Table 6.7 was consulted to obtain the number of load cycles using the limit stress for the tooth flank, and several interpolations had to be performed.

Table 6.7: Rolling contact fatigue strength, in N/mm^2 , for POM. Adapted from [57].

$\vartheta_{Fu\beta}$	σ_{HlimN}								
	When N_L								
	0	10^5	10^6	10^7	10^8	10^9	10^{10}	10^{11}	10^{99}
20	63	63	48	33	23	18	17	16	16
40	60	60	46	32	22	17	16	15	15
60	59	59	44	31	21	16	15	14	14
80	55	55	41	28	19	15	14	14	14
100	51	51	38	26	17	14	13	12	12

In Table 6.8, it can be found the necessary information to solve Equation 6.16, assuming the torque and rotational speed mentioned in subsection 6.2.1.

Table 6.8: Tooth flank stress calculation.

Parameters	z_1/z_2
Z_E	23.92
Z_H	2.50
Z_ϵ	0.88
Z_β	1.00
K_H	1.25
u	1.00
F_t / N	300
σ_H / MPa	48.29

Considering that $\sigma_{HlimN} = \sigma_H = 48.29$ MPa and using the tooth root temperature obtained in Table 6.5, it was possible to reach a number of load cycles of $6.79 \cdot 10^5$ for a torque of 9 Nm and a rotational speed of 1000 rpm.

Next, a relation between the number of load cycles and the operating conditions was performed. In Figure 6.7, it is possible to visualize this relation through the tooth flank stress calculation.

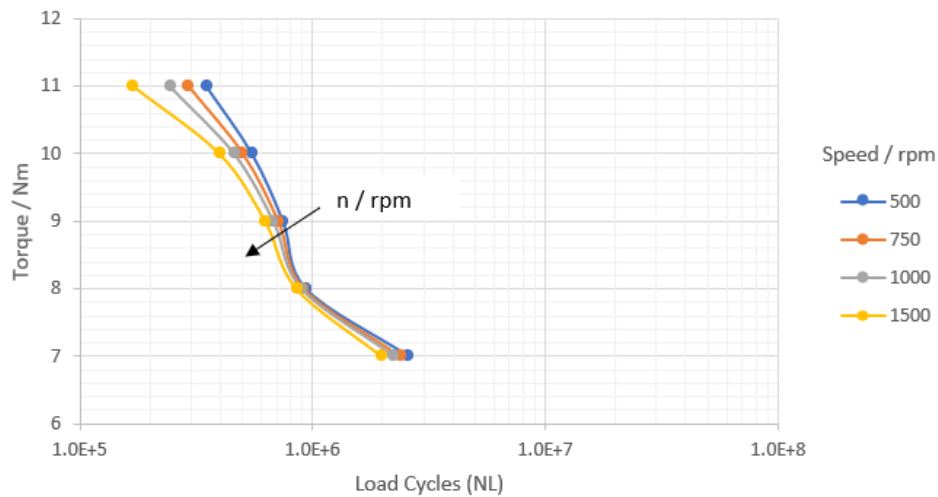


Figure 6.7: Number of load cycles as function of rotational speed and torque, through tooth flank calculation.

In this case, and comparing to the previous one in the tooth root, the maximum number of load cycles obtained was much lower. This occurs due to the fact the tooth flank is subjected to a higher contact pressure.

Furthermore, the stress distribution on the tooth flank is generally more complex than that on the tooth root. The contact stresses and load sharing between the mating teeth can vary along the flank, leading to non-uniform stress distribution. This complexity can introduce additional factors that affect the calculation of load cycles and make the results appear less straightforward. The obtained result can also be explained by the non-linear relationship between contact stress and torque (Equation 6.16).

6.2.5 Frictional wear load capacity

Abrasive wear should be taken into consideration, especially in the case of dry running. The local linear wear W_{local} is expressed as

$$W_{local} = \frac{F_{n,local}}{b_w} \cdot N_L \cdot \zeta \cdot k_w \quad (6.22)$$

and the averaged linear wear W_m as

$$W_m = \frac{T_d \cdot 2 \cdot \pi \cdot N_L \cdot H_V \cdot k_w}{b_w \cdot z \cdot l_{FI}} \leq W_{zul} \quad (6.23)$$

where:

- $F_{n,local}$ is the normal force on the tooth at the contact;
- b_w is common face width;
- N_L is the number of load cycles;
- ζ is the local specific sliding;
- k_w is the wear coefficient, whose value can be consulted in Table 6.9;

Table 6.9: Wear coefficient k_w with dry running, in $10^{-6} \text{ mm}^3 \text{ N}^{-1} \text{ m}^{-1}$, [55].

Material	k_w
POM/steel (where $R_z = 0.45 \text{ } \mu\text{m}$)	1.0
POM/steel (where $R_z = 1.5 \text{ } \mu\text{m}$)	3,4
PBT/steel (where $R_z = 0.45 \text{ } \mu\text{m}$)	3,7
PBT/steel (where $R_z = 1.5 \text{ } \mu\text{m}$)	7,8

- l_{FI} is the profile line length of the tooth flank, given by Equation 6.24;

$$l_{FI} = \frac{1}{d_b} \cdot \left[\left(\frac{d_{Na}}{2} \right)^2 - \left(\frac{d_{Nf}}{2} \right)^2 \right] \quad (6.24)$$

where: d_b is the base circle diameter; d_{Na} is the usable addendum circle diameter; d_{Nf} is the usable root circle diameter.

- W_{zul} is the permissible wear, which is determined by Equation 6.25.

$$W_{zul} = (0.1 \dots 0.2) \cdot m_n \quad (6.25)$$

To calculate the average linear wear, the number of cycles obtained through the tooth root calculation was used since the VDI 2736 guideline states that the analysis of tooth flank load capacity is not required. In Table 6.10, you can find some important parameters for the calculation of average wear, for $T_d = 9 \text{ Nm}$ and $n = 1000 \text{ rpm}$.

Table 6.10: Average wear calculation.

Parameters	z_1/z_2
N_L	$1.38 \cdot 10^6$
$k_w / 10^{-6} \text{ mm}^3 \text{ N}^{-1} \text{ m}$	3.40
d_{Na} / mm	64
d_{Nf} / mm	57.40
l_{fI} / mm	3.55
W_m / mm	0.025

The obtained result of frictional wear is within the permissible wear value, which is 0.3 mm.

6.2.6 Tooth deformation

Particularly with plastic gears that have a tiny face width, as those seen, for instance, in precision transmissions, significant tooth deformation may take place. This is due to their elasticity modulus being substantially lower than that of metals. Furthermore, creep manifestations may result in the permitted root stress being exceeded in operation if the teeth have previously been exposed to a stationary load for a considerable amount of time. It is possible to roughly compute the tooth tip's deformation in the circumferential direction, in mm, as follows:

$$\lambda = \frac{7.5 \cdot F_t}{b \cdot \cos \beta} \cdot \left(\frac{1}{E_1} + \frac{1}{E_2} \right) \quad (6.26)$$

Subsequently, this value should be compared to the permissible deformation, that is $\lambda \leq 0.07 \cdot m_n$. When $\lambda > 0.07 \cdot m_n$ running noise may increase and service life will be reduced.

By replacing the parameters of Equation 6.26 with numerical values, a deformation of 0.10 mm was obtained in the gear teeth, which is lower than the permissible deformation (0.14 mm).

6.3 Influence of External Lubrication

The use of external lubrication increases the resistance of the tooth flank and reduces the friction coefficient. Lower friction coefficients are associated with lower operating temperatures, which also increases the resistance of the tooth root.

In order to understand the importance of external lubrication on the number of load cycles allowed by the mechanism, a new simulation was conducted using a grease lubrication based in mineral oil. Using the same criteria as in the non-lubricated case, with a rotational speed of 1000 rpm and a torque of 9 Nm, the number of load cycles obtained was $1.08 \cdot 10^7$. This indicates an increase in the load carrying capacity of the mechanism, as compared to the non-lubricated case where it could only support $1.38 \cdot 10^6$ cycles.

According to the VDI 2736 directive, when no external lubrication is used, it is not necessary to verify the tooth flank strength. However, the introduction of a lubricating mass requires considering this strength and comparing it with the contact pressure between flanks. Thus, the introduction of this new criterion in the load capacity calculation results in a number of load cycles of $6.92 \cdot 10^6$. This value is also higher than the previously calculated value without lubrication for the tooth flank. It is also worth mentioning that even when using external lubrication, the VDI 2736 directive considers that analysing the load capacity of the flank of gears made of POM is usually unnecessary [55].

The choice of which number of cycles to use depends on the application's specific requirements and design considerations. If the application places a significant load on the tooth root, it would be appropriate to consider the number of cycles obtained through tooth root stress. This approach ensures that the gear teeth can withstand the load at the root and prevents failure in that region. On the other hand, if the application involves high contact pressures and sliding between the tooth flanks, it would be relevant to consider the number of cycles obtained through tooth flank stress. This approach considers the wear of the tooth flanks due to contact stresses and sliding friction.

Chapter 7

Conclusion

7.1 Conclusions

This dissertation aimed to develop a new hybrid polymer gear solution and design a mould to manufacture these hybrid gears. Additionally, a mould filling simulation was performed to predict the behaviour of the molten material as it fills the mould cavity during the injection process. Overall, the key objectives were achieved and the results from the injection experiments indicated that the geometry of the hybrid gear designed was accomplished.

The standard gear geometry was based on prior studies of gears in the literature and the dimensions were compatible with the manufacturing restrictions. Two solutions for the gear geometry were developed, one without insert and one with insert. POM was used as the main material for the gears due to its promising results in gear applications. In addition, a thermally conductive material based on Nylon 6 resin (PA6-GF) was also applied. Regarding the development of the insert geometry, two insert geometries were developed, each produced using different manufacturing processes, milling and selective laser melting. The insert geometry underwent several modifications to improve thermal evacuation and increase contact area.

The development of the mould involved several key considerations. The design aimed to optimize efficiency, minimize material waste, and reduce cycle times while ensuring reliable processes and high-quality final products. The cavity design considered factors such as proper material flow, uniform filling, adequate cooling, and easy release of the final

product. The gear teeth in the core inserts were designed to expand radially and tangentially to compensate for the shrinkage effect during cooling. The materials for the inserts, 1.2311 steel and AMPCO 83 copper alloy were chosen to study their influence on the results and mould temperatures after injection. The cooling system was designed to control the thermal gradient and promote efficient cooling of the moulding zone. It was also verified through a thermal calculation that the cooling channels were correctly designed and respected the minimum lengths. The vacuum system was created to secure the insert during the injection moulding. These modifications and considerations contributed to successfully developing the mould for hybrid gear production.

The mould filling simulation allowed to compare the filling of the gear with and without the insert. It was observed that the presence of the insert creates additional flow channels and obstacles, leading to a more complex flow pattern and increased flow resistance. Thus, mould filling simulation was a valuable tool for improving product quality, reducing defects, and enhancing the efficiency of the injection moulding manufacturing process.

Considering the dimensional control of the inserts, the inserts machined by milling with sandblasting finishing presented a lower surface roughness and waviness than the 3D printed inserts. This difference was attributed to the inherent nature of the two manufacturing processes. Then, despite not being possible to realize all the experiments in time, several conclusions could be taken. It was verified that the variation of the cooling time was not a dominant parameter comparing to the mould temperature in determining the final dimensions of the gears.

Furthermore, for the hybrid gears, the gear geometry was also not fully achieved, as the contraction in the gear teeth was lower than intended and the tooth profile was slightly larger than desired. This is due to the fact that the mould has only one cavity, and the same set of inserts was used for all the experiments to reduce costs. Therefore, the applied contraction value of 2% was an attempt to obtain the best possible result for each scenario, knowing that in order to ensure better dimensional accuracy of the parts, new inserts would need to be made considering higher contraction values for the gears without inserts and lower contraction values for the hybrid gears. Considering that the hybrid gear turned out larger than expected at a mould temperature of 100 °C and a cooling time of 25 s, it can be concluded that using a mould temperature of 120 °C and a shorter cooling time would likely have resulted in achieving the ideal geometry for the gear.

Finally, regarding the load-carrying capacity of cylindrical gears using VDI 2736, the calculation of the operating temperatures of polymeric gears is a crucial step in estimating their strength. Furthermore, the stress at the tooth root, pressure and tooth flank wear and tooth deformation must be taken into account. In conclusion, the use of external lubrication

enhances the resistance of the tooth flank and reduces the friction coefficient, leading to lower operating temperatures and increased tooth root resistance, indicating a significantly higher number of load cycles and an increase in load-carrying capacity.

7.2 Future Works

Several topics addressed in this dissertation require further analysis and development:

- Perform mechanical tests on the prototypes to evaluate properties such as tensile strength, hardness and wear resistance;
- Conduct field tests or tests under simulated operating conditions to evaluate the performance of hybrid gears in real-life situations;
- Perform a comparative economic analysis between hybrid gears and conventional gears;
- Analyse the damping coefficients of the gears with and without insert;
- Perform the load-carrying capacity calculation for a pair of gears with different polymeric materials;
- Study the implementation of a helical hybrid gear solution;
- Analyse the load-carrying capacity of a hybrid gear, even though there are no standards available yet for conducting this analysis.

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Appendix A

Materials Datasheet

A.1 POM

CAMPUS® Datasheet

HOSTAFORM® C 13021 - POM
Celanese



Product Texts

Chemical abbreviation according to ISO 1043-1: POM Molding compound ISO 29988- POM-K, M-GNR, 04-002 POM copolymer Easy flowing Injection molding type for precision molded parts and thin-walled molded parts with high rigidity, hardness and toughness; good chemical resistance to solvents, fuel and strong alkalis as well as good hydrolysis resistance; high resistance to thermal and oxidative degradation. Monomers and additives are listed in EU-Regulation (EU) 10/2011 FDA compliant according to 21 CFR 177.2470 UL-registration for all colours and a thickness more than 1.5 mm as UL 94 HB, temperature index UL 746 B electrical 110 °C, mechanical 90 °C. Burning rate ISO 3795 and FMVSS 302 < 75 mm/min for a thickness more than 1 mm. Ranges of applications: automotive engineering, precision engineering, electric and electronical industry, domestic appliances. FDA = Food and Drug Administration (USA) UL = Underwriters Laboratories (USA) FMVSS = Federal Motor Vehicle Safety Standard (USA)

Flammability @1.6mm nom. thickn.
Flammability at thickness h (3 mm)

HB
HB
-
UL recognition (h)

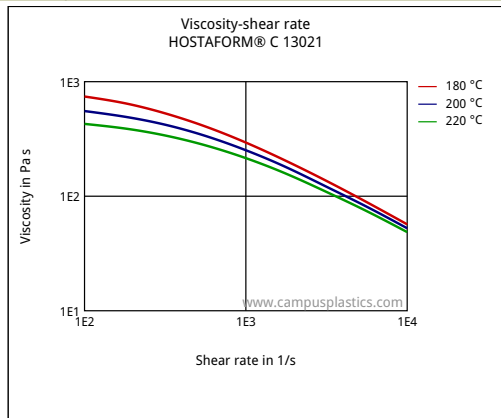
Rheological properties	Value	Unit	Test Standard
Melt volume-flow rate, MVR	12	cm³/10min	ISO 1133
Temperature	190	°C	ISO 1133
Load	2.16	kg	ISO 1133
Molding shrinkage, parallel	2.0	%	ISO 294-4, 2577
Molding shrinkage, normal	1.8	%	ISO 294-4, 2577
Mechanical properties	Value	Unit	Test Standard
Tensile modulus	2900	MPa	ISO 527-1/-2
Yield stress	65	MPa	ISO 527-1/-2
Yield strain	9	%	ISO 527-1/-2
Nominal strain at break	28	%	ISO 527-1/-2
Charpy impact strength, +23°C	200	kJ/m²	ISO 179/1eU
Charpy impact strength, -30°C	200	kJ/m²	ISO 179/1eU
Charpy notched impact strength, +23°C	6.5	kJ/m²	ISO 179/1eA
Charpy notched impact strength, -30°C	6	kJ/m²	ISO 179/1eA
Thermal properties	Value	Unit	Test Standard
Melting temperature, 10°C/min	166	°C	ISO 11357-1/-3
Temp. of deflection under load, 1.80 MPa	106	°C	ISO 75-1/-2
Coeff. of linear therm. expansion, parallel	110	E-6/K	ISO 11359-1/-2
Burning behavior at 1.5 mm nominal thickness	HB	class	IEC 60695-11-10
Thickness tested (1.5)	1.5	mm	IEC 60695-11-10
Burning behavior at thickness h	HB	class	IEC 60695-11-10
Thickness tested (h)	3.0	mm	IEC 60695-11-10
Yellow Card available	Yes	-	-
Electrical properties	Value	Unit	Test Standard
Relative permittivity, 100Hz	4	-	IEC 62631-2-1
Relative permittivity, 1MHz	4	-	IEC 62631-2-1
Dissipation factor, 100Hz	20	E-4	IEC 62631-2-1
Dissipation factor, 1MHz	50	E-4	IEC 62631-2-1
Volume resistivity	1E12	Ohm*m	IEC 62631-3-1
Surface resistivity	1E14	Ohm	IEC 62631-3-2
Electric strength	35	kV/mm	IEC 60243-1

HOSTAFORM® C 13021 - POM **Celanese**

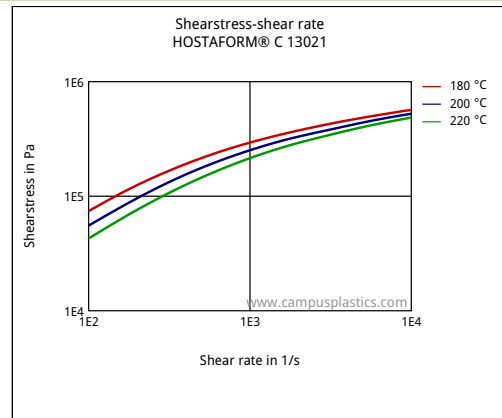
Other properties	Value	Unit	Test Standard
Water absorption	0.65	%	Sim. to ISO 62
Humidity absorption	0.2	%	Sim. to ISO 62
Density	1410	kg/m ³	ISO 1183
Rheological calculation properties	Value	Unit	Test Standard
Density of melt	1200	kg/m ³	-
Thermal conductivity of melt	0.155	W/(m K)	-
Spec. heat capacity melt	2210	J/(kg K)	-
Ejection temperature	140	°C	-

Diagrams

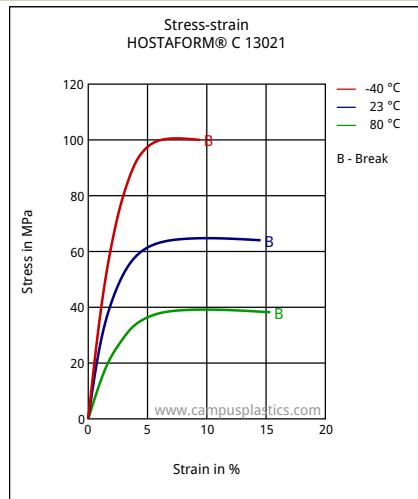
Viscosity-shear rate



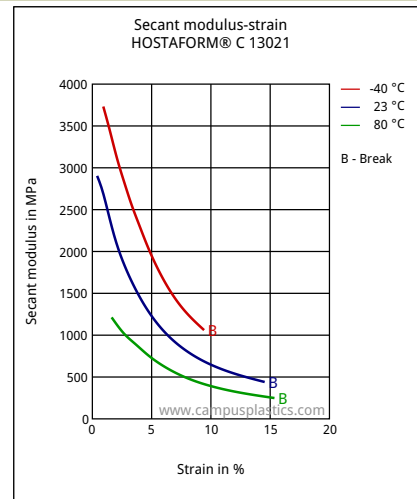
Shearstress-shear rate



Stress-strain



Secant modulus-strain



HOSTAFORM® C 13021 - POM

Celanese

Characteristics

Processing

Injection Molding

Additives

Release agent

Delivery form

Pellets

Other text information

Injection molding

General drying is not necessary due to low moisture absorption of the resin.

In case of bad storage conditions (water contact or condensed water) the use of a recirculating air dryer (100 to 120 °C / max. 40 mm layer / 3 to 6 hours) is recommended.

Max. Water content 0,2 %

Standard injection moulding machines with three phase (15 to 25 D) plasticating screws will fit.

Conditioning e.g. moisturizing is not necessary.

NOTICE TO USERS: Values shown are based on testing of laboratory test specimens and represent data that fall within the standard range of properties for natural material. These values alone do not represent a sufficient basis for any part design and are not intended for use in establishing maximum, minimum, or ranges of values for specification purposes. Colorants or other additives may cause significant variations in data values. Properties of molded parts can be influenced by a wide variety of factors including, but not limited to, material selection, additives, part design, processing conditions and environmental exposure. Other than those products expressly identified as medical grade (including by MT® product designation or otherwise), Celanese's products are not intended for use in medical or dental implants. Regardless of any such product designation, any determination of the suitability of a particular material and part design for any use contemplated by the users and the manner of such use is the sole responsibility of the users, who must assure themselves that the material as subsequently processed meets the needs of their particular product or use. To the best of our knowledge, the information contained in this publication is accurate; however, we do not assume any liability whatsoever for the accuracy and completeness of such information. The information contained in this publication should not be construed as a promise or guarantee of specific properties of our products. It is the sole responsibility of the users to investigate whether any existing patents are infringed by the use of the materials mentioned in this publication. Moreover, there is a need to reduce human exposure to many materials to the lowest practical limits in view of possible adverse effects. To the extent that any hazards may have been mentioned in this publication, we neither suggest nor guarantee that such hazards are the only ones that exist. We recommend that persons intending to rely on any recommendation or to use any equipment, processing technique or material mentioned in this publication should satisfy themselves that they can meet all applicable safety and health standards. We strongly recommend that users seek and adhere to the manufacturer's current instructions for handling each material they use, and entrust the handling of such material to adequately trained personnel only. Please call the telephone numbers listed for additional technical information. Call Customer Services for the appropriate Materials Safety Data Sheets (MSDS) before attempting to process our products.

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A.2 PA-GF



LNPT™ KONDUIT™ COMPOUND PX10323

REGION EUROPE

DESCRIPTION

LNP KONDUIT PX10323 compound is based on Nylon 6 resin containing glass fiber and proprietary thermal filler. Added features of this grade include: Thermally Conductive.

GENERAL INFORMATION	
Features	Thermally Conductive, No PFAS intentionally added
Fillers	Glass Fiber, Proprietary Filler
Polymer Types	Polyamide 6 (Nylon 6)
Processing Techniques	Injection Molding

INDUSTRY	SUB INDUSTRY
Building and Construction	Building Component
Consumer	Home Appliances
Electrical and Electronics	Mobile Phone · Computer · Tablets, Lighting
Industrial	Electrical, Material Handling

TYPICAL PROPERTY VALUES

Revision 20230607

PROPERTIES	TYPICAL VALUES	UNITS	TEST METHODS
MECHANICAL ⁽¹⁾			
Tensile Stress, brk, Type I, 5 mm/min	70	MPa	ASTM D638
Tensile Strain, brk, Type I, 5 mm/min	0.9	%	ASTM D638
Tensile Modulus, 5 mm/min	13700	MPa	ASTM D638
Flexural Stress, brk, 1.3 mm/min, 50 mm span	95	MPa	ASTM D790
Flexural Modulus, 1.3 mm/min, 50 mm span	14200	MPa	ASTM D790
Tensile Stress, break, 5 mm/min	73	MPa	ISO 527
Tensile Strain, break, 5 mm/min	0.8	%	ISO 527
Tensile Modulus, 1 mm/min	14000	MPa	ISO 527
Flexural Stress, break, 2 mm/min	96	MPa	ISO 178
Flexural Modulus, 2 mm/min	14300	MPa	ISO 178
IMPACT ⁽¹⁾			
Izod Impact, unnotched, 23°C	73	J/m	ASTM D4812
Izod Impact, notched, 23°C	30	J/m	ASTM D256
Izod Impact, unnotched 80°10°4 +23°C	6	kJ/m ²	ISO 180/1U
Izod Impact, notched 80°10°4 +23°C	3	kJ/m ²	ISO 180/1A
THERMAL ⁽¹⁾			
HDT, 1.82 MPa, 3.2mm, unannealed	186	°C	ASTM D648
CTE, -40°C to 40°C, flow	1.61E-05	1/°C	ASTM E831
CTE, -40°C to 40°C, xflow	3.56E-05	1/°C	ASTM E831
Thermal Conductivity in-plane, 25°0.4mm disc	15	W/m·K	ASTM E1461-07

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CHEMISTRY THAT MATTERS™

PROPERTIES	TYPICAL VALUES	UNITS	TEST METHODS
Thermal Conductivity through-plane, 10*10*3mm sample	3.5	W/m-K	ASTM E1461-07
Thermal Conductivity in-plane, 60*60*3mm plaque	18	W/m-K	ISO 22007-2
Thermal Conductivity through-plane, 60*60*3mm plaque	1.5	W/m-K	ISO 22007-2
CTE, -30°C to 80°C, flow	1.4E-05	1/°C	ISO 11359-2
CTE, -30°C to 80°C, xflow	3.95E-05	1/°C	ISO 11359-2
CTE, 23°C to 80°C, flow	1.62E-05	1/°C	ISO 11359-2
CTE, 23°C to 80°C, xflow	4.77E-05	1/°C	ISO 11359-2
HDT/Af, 1.8 MPa Flatw 80*10*4 sp=64mm	201	°C	ISO 75/Af
Ball Pressure Test, approximate maximum	205	°C	IEC 60695-10-2
Ball Pressure Test, 165°C +/- 2°C	PASSES	-	IEC 60695-10-2
PHYSICAL ⁽¹⁾			
Density	1.61	g/cm³	ASTM D792
Mold Shrinkage, flow, 24 hrs ⁽²⁾	0.2	%	ASTM D955
Mold Shrinkage, xflow, 24 hrs ⁽²⁾	0.22	%	ASTM D955
Density	1.61	g/cm³	ISO 1183
ELECTRICAL ⁽¹⁾			
Surface Resistivity	1.05E+06	Ω	ASTM D257
FLAME CHARACTERISTICS ⁽³⁾			
UL Yellow Card Link	E45329-104457424	-	-
UL Recognized, 94HB Flame Class Rating	≥0.75	mm	UL 94
Glow Wire Flammability Index, 1.0 mm	960	°C	IEC 60695-2-12
Glow Wire Flammability Index, 1.5 mm	960	°C	IEC 60695-2-12
Glow Wire Flammability Index, 3.0 mm	960	°C	IEC 60695-2-12
Glow Wire Ignitability Temperature, 1.0 mm	825	°C	IEC 60695-2-13
Glow Wire Ignitability Temperature, 1.5 mm	850	°C	IEC 60695-2-13
Glow Wire Ignitability Temperature, 3.0 mm	960	°C	IEC 60695-2-13
INJECTION MOLDING ⁽⁴⁾			
Drying Temperature	80 – 90	°C	
Drying Time	3 – 4	Hrs	
Melt Temperature	270 – 300	°C	
Nozzle Temperature	270 – 300	°C	
Front - Zone 3 Temperature	270 – 300	°C	
Middle - Zone 2 Temperature	270 – 300	°C	
Rear - Zone 1 Temperature	260 – 290	°C	
Mold Temperature	80 – 100	°C	
Back Pressure	3 – 9	MPa	
Screw Speed	30 – 150	rpm	

(1) The information stated on Technical Datasheets should be used as indicative only for material selection purposes and not be utilized as specification or used for part or tool design.

(2) Measurements made from laboratory test coupon. Actual shrinkage may vary outside of range due to differences in processing conditions, equipment, part geometry and tool design. It is recommended that mold shrinkage studies be performed with surrogate or legacy tooling prior to cutting tools for new molded article.

(3) UL Ratings shown on the technical datasheet might not cover the full range of thicknesses and colors. For details, please see the UL Yellow Card.

(4) Injection Molding parameters are only mentioned as general guidelines. These may not apply or may need adjustment in specific situations such as low shot sizes, large part molding, thin wall molding and gas-assist molding.

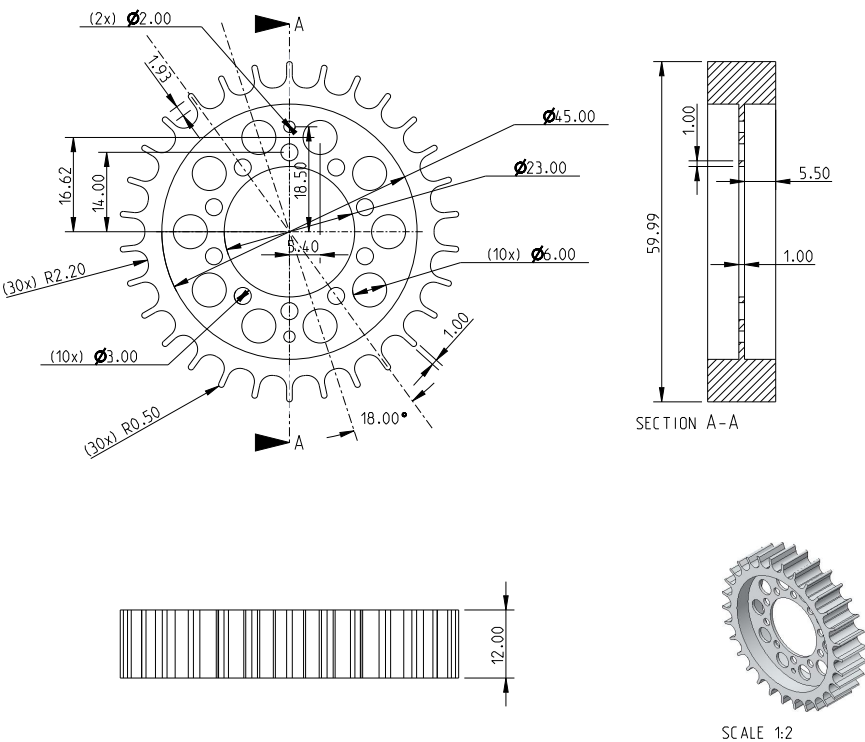
Appendix B

2D Drawings

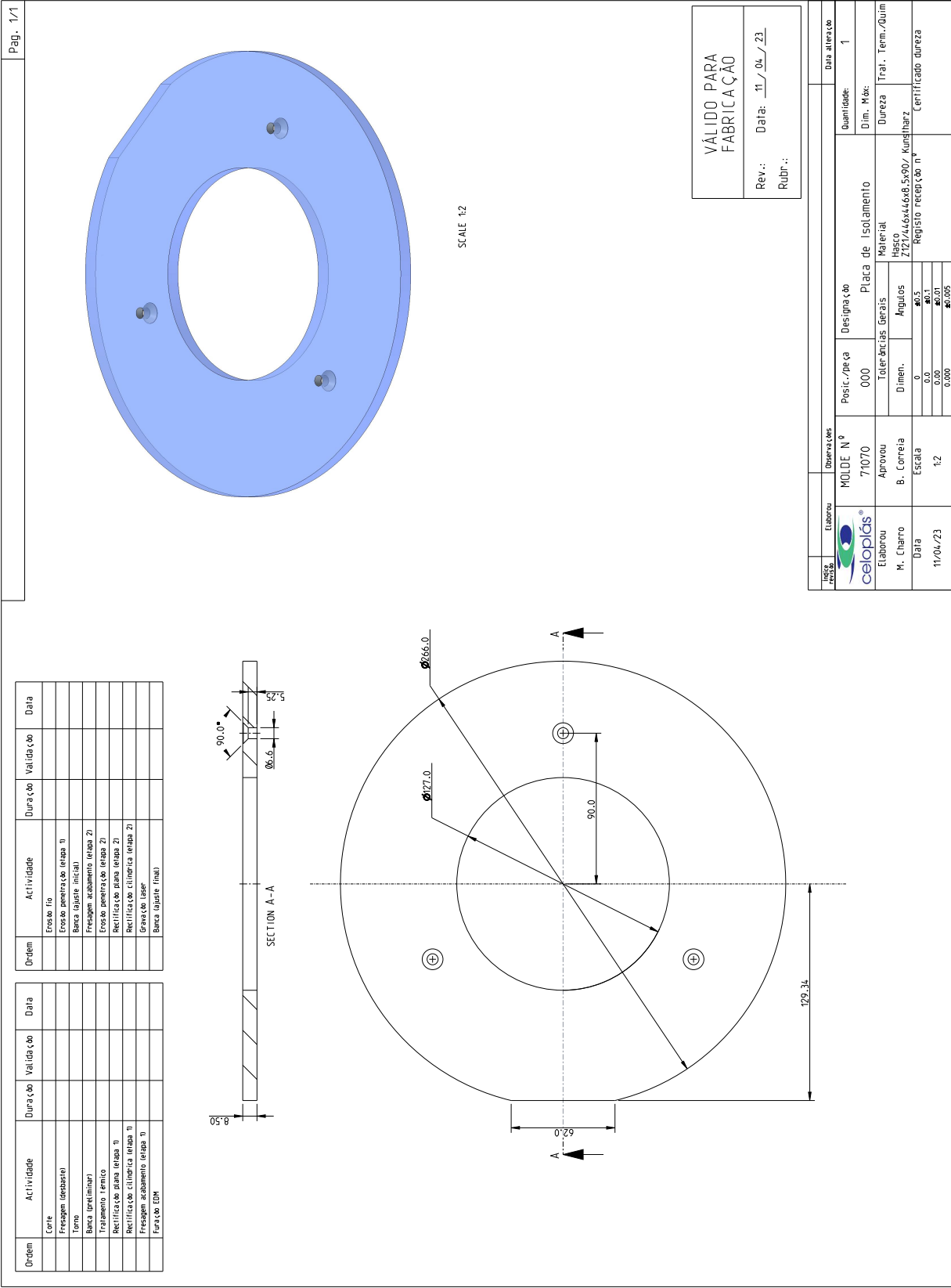
B.1 Hybrid Gear

B.2 Aluminium Insert

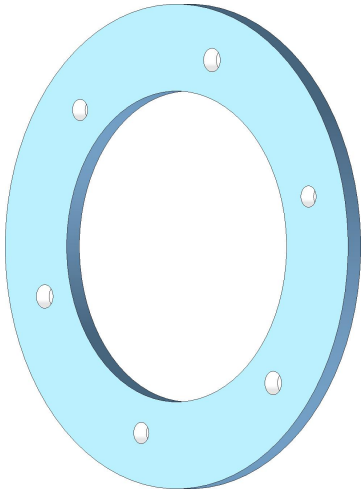
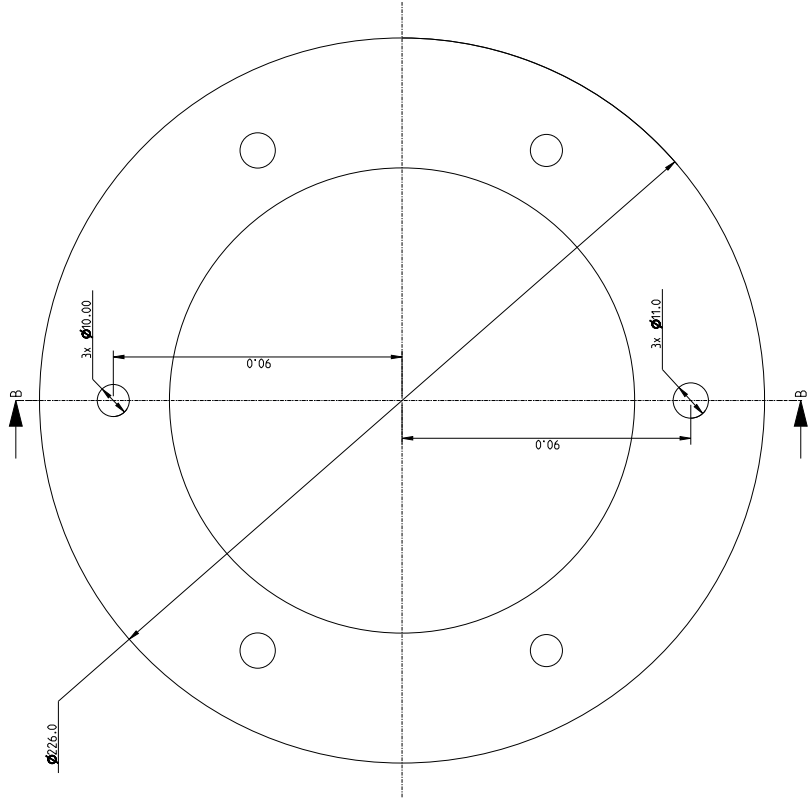
										Pag. 1/1	
Ordem	Actividade	Duração	Validação	Data	Ordem	Actividade	Duração	Validação	Data		
	Corte					Erosão fio					
	Fresagem (desbaste)					Erosão penetração (etapa 1)					
	Torno					Banca (ajuste inicial)					
	Banca (preliminar)					Fresagem acabamento (etapa 2)					
	Tratamento térmico					Erosão penetração (etapa 2)					
	Rectificação plana (etapa 1)					Rectificação plana (etapa 2)					
	Rectificação cilíndrica (etapa 1)					Rectificação cilíndrica (etapa 2)					
	Fresagem acabamento (etapa 1)					Gravação laser					
	Furação EDM					Banca (ajuste final)					



Índice	Elaborou	Observações				Data alteração	
		MOLDE N ^o 71070	Posic./peça	Designação		Quantidade:	
	Elaborou Miguel Charro	Aprovou Bruno Correia	Tolerâncias Gerais		Material Alumínio 6082	Dureza	Trat. Term./Quim
	Data 31/03/2023	Escala 1:1	Dimen.	Ângulos	Registo recepção n ^o	Certificado dureza	
			0	±0.5			
			0.0	±0.1			
			0.00	±0.01			
			0.000	±0.005			



Ordem	Atividade	Duração	Validação	Data
	Confe			
	Fresagem (desbaste)			
	Torno			
	Barca (refinador)			
	Tratamento térmico			
	Rectificação plana (etapa 1)			
	Rectificação cilíndrica (etapa 1)			
	Fresagem acabamento (etapa 1)			
	Furação EDM			
	Rectificação plana (etapa 2)			
	Rectificação cilíndrica (etapa 2)			
	Gravado laser			
	Barca (ajuste final)			
	Fresado fio			
	Fresado penetração (etapa 1)			
	Barca (ajuste inicial)			
	Fresagem acabamento (etapa 2)			
	Fresado penetração (etapa 2)			
	Rectificação plana (etapa 2)			
	Rectificação cilíndrica (etapa 2)			
	Gravado laser			
	Barca (ajuste final)			



VÁLIDO PARA
FABRICAÇÃO

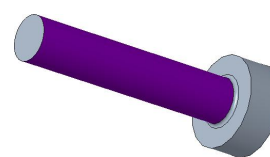
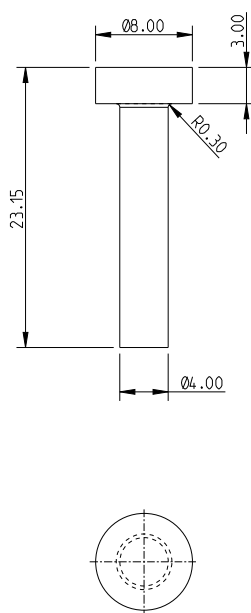
Rev.: 1 Data: 11/04/23

Rubr.:

SECTION B-B

A1	M. Charro	Aumento do diâmetro interno para 14,5 mm	19 Jun 2023
Elaborou	Observações		
celoplas	MOL DE Nº	Designação	Quantidade
	71070	010-1	1
	Placa de isolamento Lado móvel		
	Material		
	Hascro		
	Z121/446x46x5x90/ Kundsharz		
	Registo recepção nº		
	1:1.333		
	Dim. Máx:		
	Dureza		
	Traj. Term./Quim		
	Certificado dureza		

Ordem	Actividade	Duração	Validação	Data	Ordem	Actividade	Duração	Validação	Data
	Corte					Erosão fio			
	Fresagem (desbaste)					Erosão penetração (etapa 1)			
	Torno					Banca (ajuste inicial)			
	Banca (preliminar)					Fresagem acabamento (etapa 2)			
	Tratamento térmico					Erosão penetração (etapa 2)			
	Rectificação plana (etapa 1)					Rectificação plana (etapa 2)			
	Rectificação cilíndrica (etapa 1)					Rectificação cilíndrica (etapa 2)			
	Fresagem acabamento (etapa 1)					Gravação laser			
	Furação EDM					Banca (ajuste final)			



SCALE 2:1

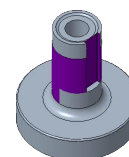
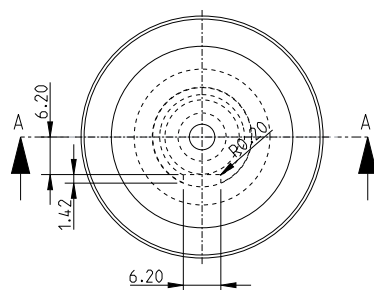
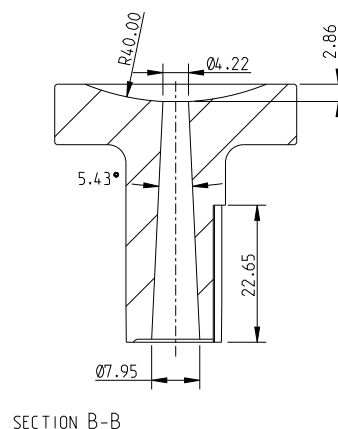
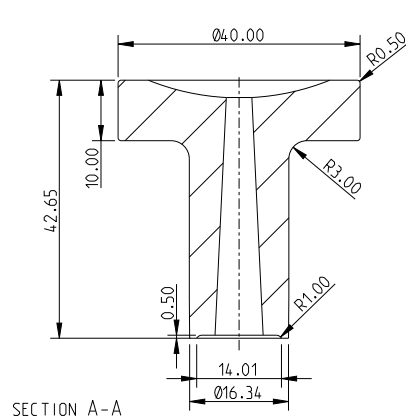
VÁLIDO PARA
FABRICAÇÃO

Rev.: Data: 12 / 04 / 23

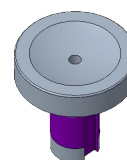
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	Elaborou	Aprovou	Tolerâncias Gerais		Material	Dureza	Trat. Term./Quim
	M. Charro	B. Correia	Dimen.	Ângulos	Z40/4x50-1.2516		
Data	Escala			Registo recepção n ^o	Certificado dureza		
12/04/23	2:1	0	±0.5				
		0.0	±0.1				
		0.00	±0.01				
		0.000	±0.005				

Ordem	Actividade	Duraçã o	Validaçã o	Data	Ordem	Actividade	Duraçã o	Validaçã o	Data
	Corte					Erosão fio			
	Fresagem (desbaste)					Erosão penetração (etapa 1)			
	Torno					Banca (ajuste inicial)			
	Banca (preliminar)					Fresagem acabamento (etapa 2)			
	Tratamento térmico					Erosão penetração (etapa 2)			
	Rectificação plana (etapa 1)					Rectificação plana (etapa 2)			
	Rectificação cilíndrica (etapa 1)					Rectificação cilíndrica (etapa 2)			
	Fresagem acabamento (etapa 1)					Gravação laser			
	Furação EDM					Banca (ajuste final)			



SCALE 1:2



SCALE 1:2

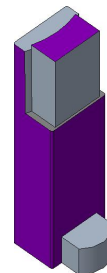
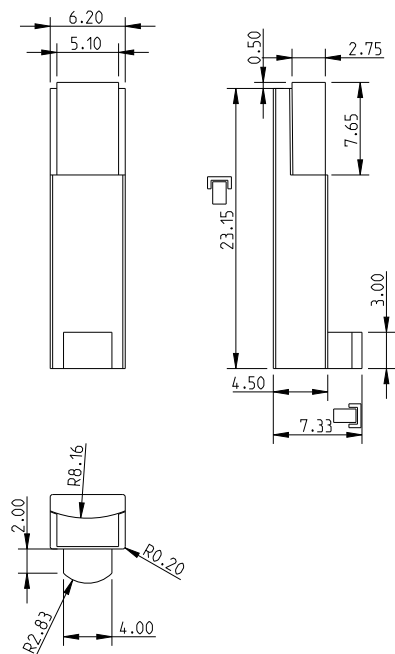
VÁLIDO PARA
FABRICAÇÃO

Rev.: 1 Data: 12 / 04 / 23

Rubr.:

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Elaborou	Aprovou	Tolerâncias Gerais		Material	Dureza		Trat. Term./Quim
M. Charro	B. Correia	Dimen.	Ângulos	1.2767			
Data	Escala	0	±0.5	Registo recepção n ^o	Certificado dureza		
12/04/23	1:1	0.0	±0.1				
		0.00	±0.01				
		0.000	±0.005				

										Pag. 1/1
Ordem	Actividade	Duraç ão	Validaç ão	Data	Ordem	Actividade	Duraç ão	Validaç ão	Data	
	Corte					Eros ão fio				
	Fresagem (desbaste)					Eros ão penetraç ão (etapa 1)				
	Torno					Banca (ajuste inicial)				
	Banca (preliminar)					Fresagem acabamento (etapa 2)				
	Tratamento térmico					Eros ão penetraç ão (etapa 2)				
	Rectificaç ão plana (etapa 1)					Rectificaç ão plana (etapa 2)				
	Rectificaç ão cilíndrica (etapa 1)					Rectificaç ão cilíndrica (etapa 2)				
	Fresagem acabamento (etapa 1)					Gravaç ão laser				
	Furaç ão EDM					Banca (ajuste final)				

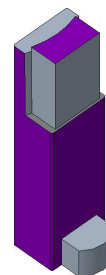
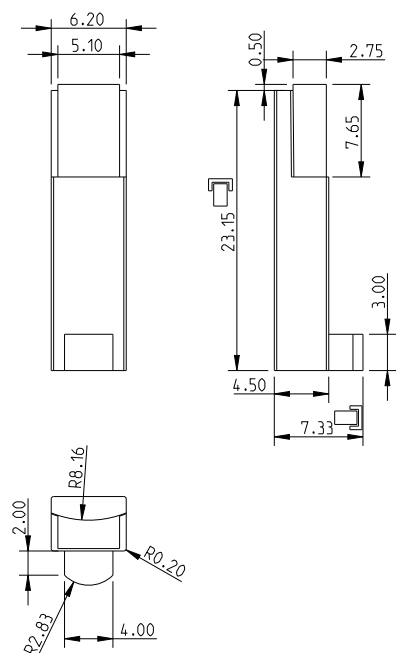


VÁLIDO PARA
FABRICAÇÃO

Rev.: Data: 12 / 04 / 23
Rubr.:

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	MOLDE N ^o	Posic./peça	Designação			Quantidade:	1
	71070	100F	Postiço da injeção 100F			Dim. Máx:	
Elaborou	Aprovou	Tolerâncias Gerais		Material	Dureza	Trat. Term./Quim	
M. Charro	B. Correia	Dimen.	Ângulos				
				1.2311			
Data	Escala	0 ±0.5		Registo recepção n ^o	Certificado dureza		
12/04/23	2:1	0.0 ±0.1					
		0.00 ±0.01					
		0.000 ±0.005					

Ordem	Actividade	Duraç ão	Validaç ão	Data	Ordem	Actividade	Duraç ão	Validaç ão	Data
	Corte					Eros ão fio			
	Fresagem (desbaste)					Eros ão penetraç ão (etapa 1)			
	Torno					Banca (ajuste inicial)			
	Banca (preliminar)					Fresagem acabamento (etapa 2)			
	Tratamento térmico					Eros ão penetraç ão (etapa 2)			
	Rectificaç ão plana (etapa 1)					Rectificaç ão plana (etapa 2)			
	Rectificaç ão cilíndrica (etapa 1)					Rectificaç ão cilíndrica (etapa 2)			
	Fresagem acabamento (etapa 1)					Gravaç ão laser			
	Furaç ão EDM					Banca (ajuste final)			



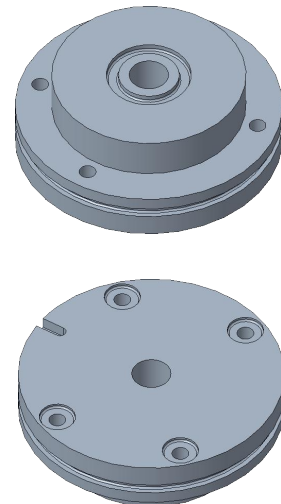
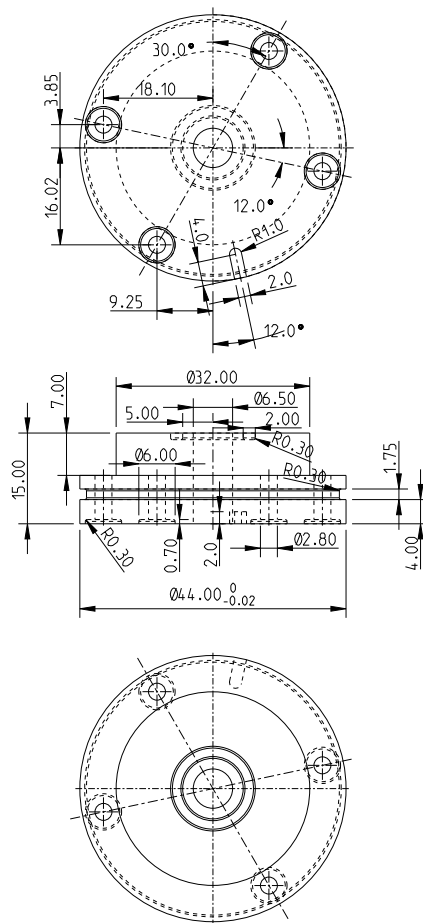
VÁLIDO PARA
FABRICAÇÃO

Rev.: Data: 12 / 04 / 23

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Índice revisão	Elaborou	Observações				Data alteração	
	MOLDE N ^o	Posic./peça	Designação		Quantidade: 1		
	71070	100F	Postiço da injeção 100F		Dim. Máx:		
Elaborou	Aprovou	Tolerâncias Gerais		Material	Dureza	Trat. Term./Quim	
M. Charro	B. Correia	Dimen.	Ângulos				
		1.2311					
Data	Escala	0 ±0.5		Registo recepção n ^o	Certificado dureza		
		0.0 ±0.1					
		0.00 ±0.01					
		0.000 ±0.005					
12/04/23	2:1						

Ordem	Actividade	Duração	Validação	Data	Ordem	Actividade	Duração	Validação	Data
	Corte					Erosão fio			
	Fresagem (desbaste)					Erosão penetração (etapa 1)			
	Torno					Banca (ajuste inicial)			
	Banca (preliminar)					Fresagem acabamento (etapa 2)			
	Tratamento térmico					Erosão penetração (etapa 2)			
	Rectificação plana (etapa 1)					Rectificação plana (etapa 2)			
	Rectificação cilíndrica (etapa 1)					Rectificação cilíndrica (etapa 2)			
	Fresagem acabamento (etapa 1)					Gravação laser			
	Furação EDM					Banca (ajuste final)			



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A1	M. Charro	Adição de alojamento para chaveta					30Mai2023
Índice revisão	Elaborou	Observações					Data alteração
	MOLDE N ^o	Posic./peça	Designação			Quantidade:	1
	71070	TA	Tampa			Dim. Máx:	
Elaborou	Aprovou	Tolerâncias Gerais		Material	Dureza	Trat. Term./Quim	
M. Charro	B. Correia	Dimen.	Ângulos				
		1.1730					
Data	Escala	Registo recepção n ^o			Certificado dureza		
13/04/23	1:1	0	±0.5				
		0.0	±0.1				
		0.00	±0.01				
		0.000	±0.005				

										Pag. 1/1
Ordem	Actividade	Duraç ão	Validaç ão	Data	Ordem	Actividade	Duraç ão	Validaç ão	Data	
	Corte					Eros ão fio				
	Fresagem (desbaste)					Eros ão penetraç ão (etapa 1)				
	Torno					Banca (ajuste inicial)				
	Banca (preliminar)					Fresagem acabamento (etapa 2)				
	Tratamento térmico					Eros ão penetraç ão (etapa 2)				
	Rectificaç ão plana (etapa 1)					Rectificaç ão plana (etapa 2)				
	Rectificaç ão cilíndrica (etapa 1)					Rectificaç ão cilíndrica (etapa 2)				
	Fresagem acabamento (etapa 1)					Gravaç ão laser				
	Furaç ão EDM					Banca (ajuste final)				

SCALE 1:3

Elemento novo no molde

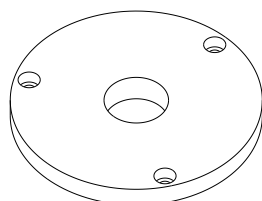
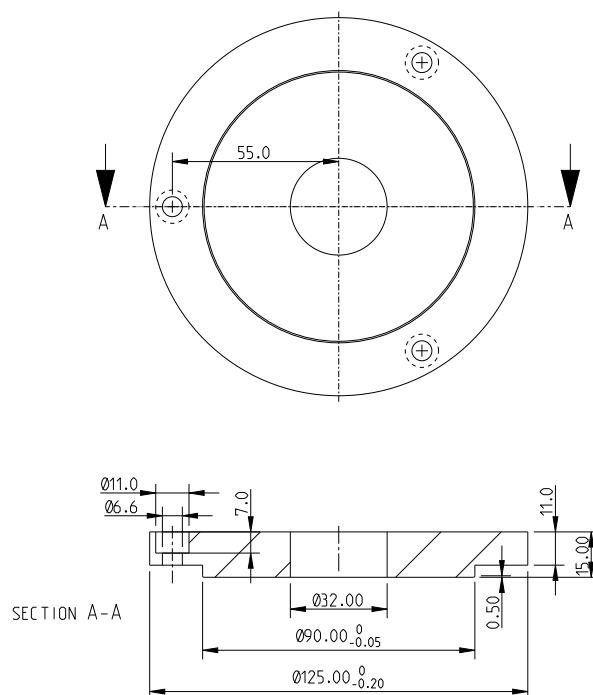
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Rev.: Data: 13/04/23

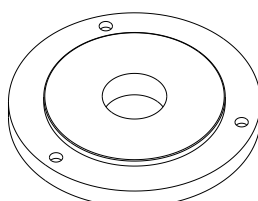
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Índice revisão	Elaborou	Observações	Data alteração	
	MOLDE N ^o 71070	Posic./peça 032	Designação Anel de centragem injeção	Quantidade: 1
Elaborou M. Charro	Aprovou B. Correia	Tolerâncias Gerais		Dim. Máx: Ø125.000x19.000
		Dimen.	Ângulos	Dureza
		0	±0.5	-
		0.0	±0.1	-
		0.00	±0.01	-
		0.000	±0.005	-
Data 13/04/23	Escala 5:6	Registo recepção n ^o		Certificado dureza

Ordem	Atividade	Duração	Validação	Data
	Corte			
	Fresagem (desbaste)			
	Torno			
	Banca (preliminar)			
	Tratamento térmico			
	Retificação plana (etapa 1)			
	Retificação cilíndrica (etapa 1)			
	Fresagem acabamento (etapa 1)			
	Furação EDM			



SCALE 1:3



SCALE 1:3

Elemento novo no molde

VÁLIDO PARA
FABRICAÇÃO

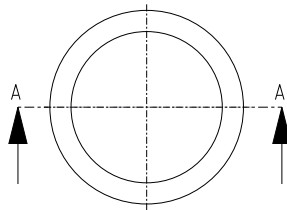
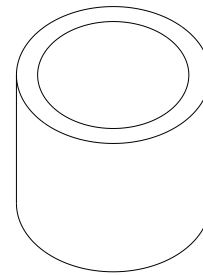
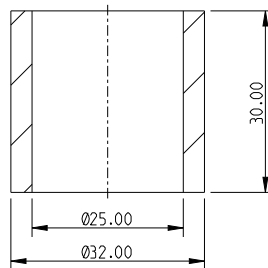
Rev.: Data: 13/04/23

Rubr.:

Índice revisão	Elaborou	Observações				Data alteração
	MOLDE N ^o	Posic./peça	Designação		Quantidade:	1
	71070	034	Anel de Centragem Extração		Dim. Máx:	Ø 125.000x15.000
Elaborou	Aprovou	Tolerâncias Gerais		Material	Dureza	Trat. Term./Quím
M. Charro	B. Correia	Dimen.	Ângulos			
				1.1191	-	-
Data	Escala	0 ±0.5 0.0 ±0.1 0.00 ±0.01 0.000 ±0.005		Registo recepção n ^o	Certificado dureza	
13/04/23	1:2					

Ordem	Actividade	Duraç ão	Validaç ão	Data	Ordem	Actividade	Duraç ão	Validaç ão	Data
	Corte					Eros ão fio			
	Fresagem (desbaste)					Eros ão penetraç ão (etapa 1)			
	Torno					Banca (ajuste inicial)			
	Banca (preliminar)					Fresagem acabamento (etapa 2)			
	Tratamento térmico					Eros ão penetraç ão (etapa 2)			
	Rectificaç ão plana (etapa 1)					Rectificaç ão plana (etapa 2)			
	Rectificaç ão cilíndrica (etapa 1)					Rectificaç ão cilíndrica (etapa 2)			
	Fresagem acabamento (etapa 1)					Gravaç ão Laser			
	Furaç ão EDM					Banca (ajuste final)			

SECTION A-A

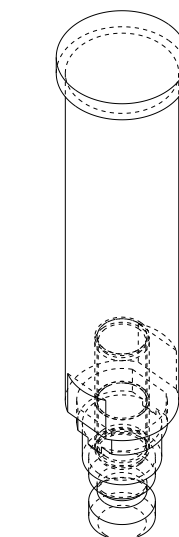
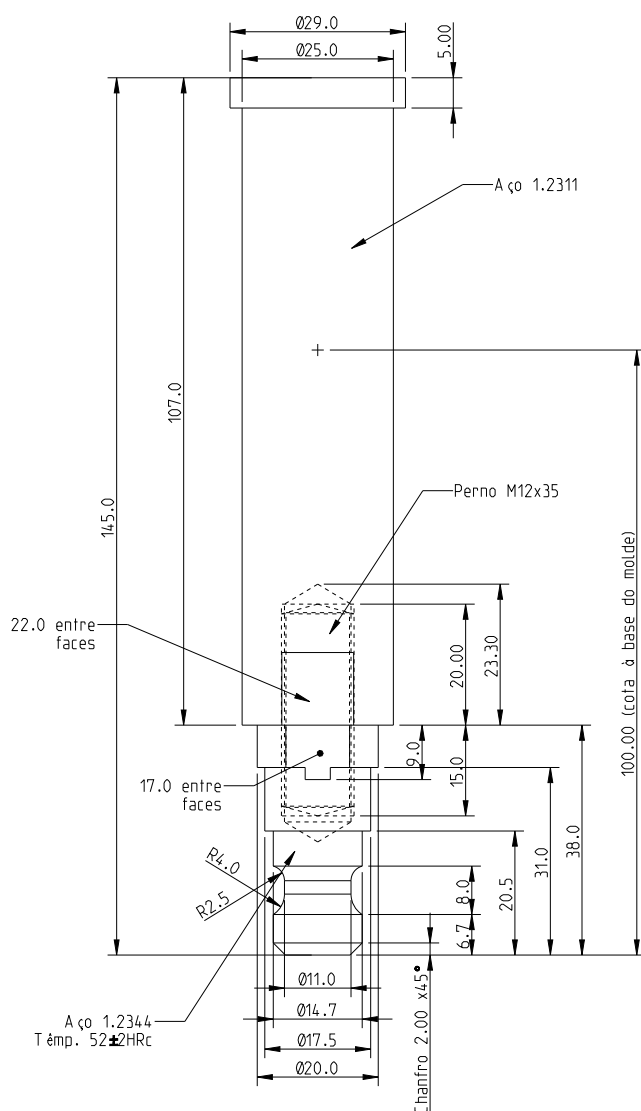
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Índice revisão	Elaborou	Observações				Data alteração	
	MOLDE N ^o	Posic./peça	Designação		Quantidade: 1		
	71070	028	Casquilho da Haste Extratora		Dim. Máx: Ø32.000x30.000		
Elaborou	Aprovou	Tolerâncias Gerais		Material	Dureza	Trat. Term./Quim	
M. Charro	B. Correia	Dimen.	Ângulos				-
Data	Escala	0 ±0.5		Registo recepção n ^o	Certificado dureza		
13/04/23	1:1	0.0 ±0.1					
		0.00 ±0.01					
		0.000 ±0.005					

Ordem	Actividade	Duração	Validação	Data	Ordem	Actividade	Duração	Validação	Data
	Corte					Erosão fio			
	Fresagem (desbaste)					Erosão penetração (etapa 1)			
	Torno					Banca (ajuste inicial)			
	Banca (preliminar)					Fresagem acabamento (etapa 2)			
	Tratamento térmico					Erosão penetração (etapa 2)			
	Rectificação plana (etapa 1)					Rectificação plana (etapa 2)			
	Rectificação cilíndrica (etapa 1)					Rectificação cilíndrica (etapa 2)			
	Fresagem acabamento (etapa 1)					Gravação Laser			
	Furação EDM					Banca (ajuste final)			



SCALE 1:1.333

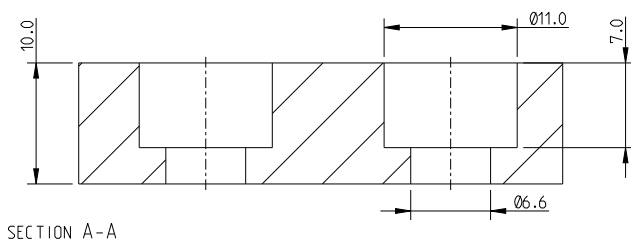
VÁLIDO PARA
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Rev.: Data: 13/04/23

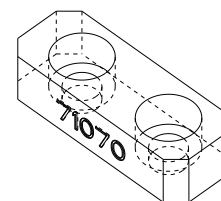
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Índice revisão	Elaborou	Observações				Data alteração	
	MOLDE N ^o	Posic./peça	Designação		Quantidade: 1		
	71070	075	Haste Extratora		Dim. Máx:		
Elaborou	Aprovou	Tolerâncias Gerais		Material	Dureza	Trat. Term./Quim	
M. Charro	B. Correia	Dimen.	Ângulos				
Data	Escala	0 ±0.5		Registo recepção n ^o	Certificado dureza		
13/04/23	1:1	0.0 ±0.1					
		0.00 ±0.01					
		0.000 ±0.005					

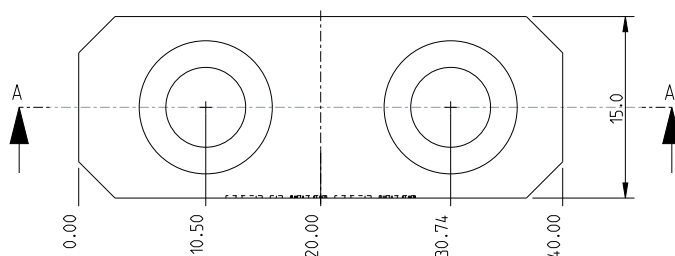
Ordem	Actividade	Duraç ão	Validaç ão	Data	Ordem	Actividade	Duraç ão	Validaç ão	Data
	Corte					Eros ão fio			
	Fresagem (desbaste)					Eros ão penetraç ão (etapa 1)			
	Torno					Banca (ajuste inicial)			
	Banca (preliminar)					Fresagem acabamento (etapa 2)			
	Tratamento térmico					Eros ão penetraç ão (etapa 2)			
	Rectificaç ão plana (etapa 1)					Rectificaç ão plana (etapa 2)			
	Rectificaç ão cilíndrica (etapa 1)					Rectificaç ão cilíndrica (etapa 2)			
	Fresagem acabamento (etapa 1)					Gravaç ão laser			
	Furaç ão EDM					Banca (ajuste final)			



SECTION A-A



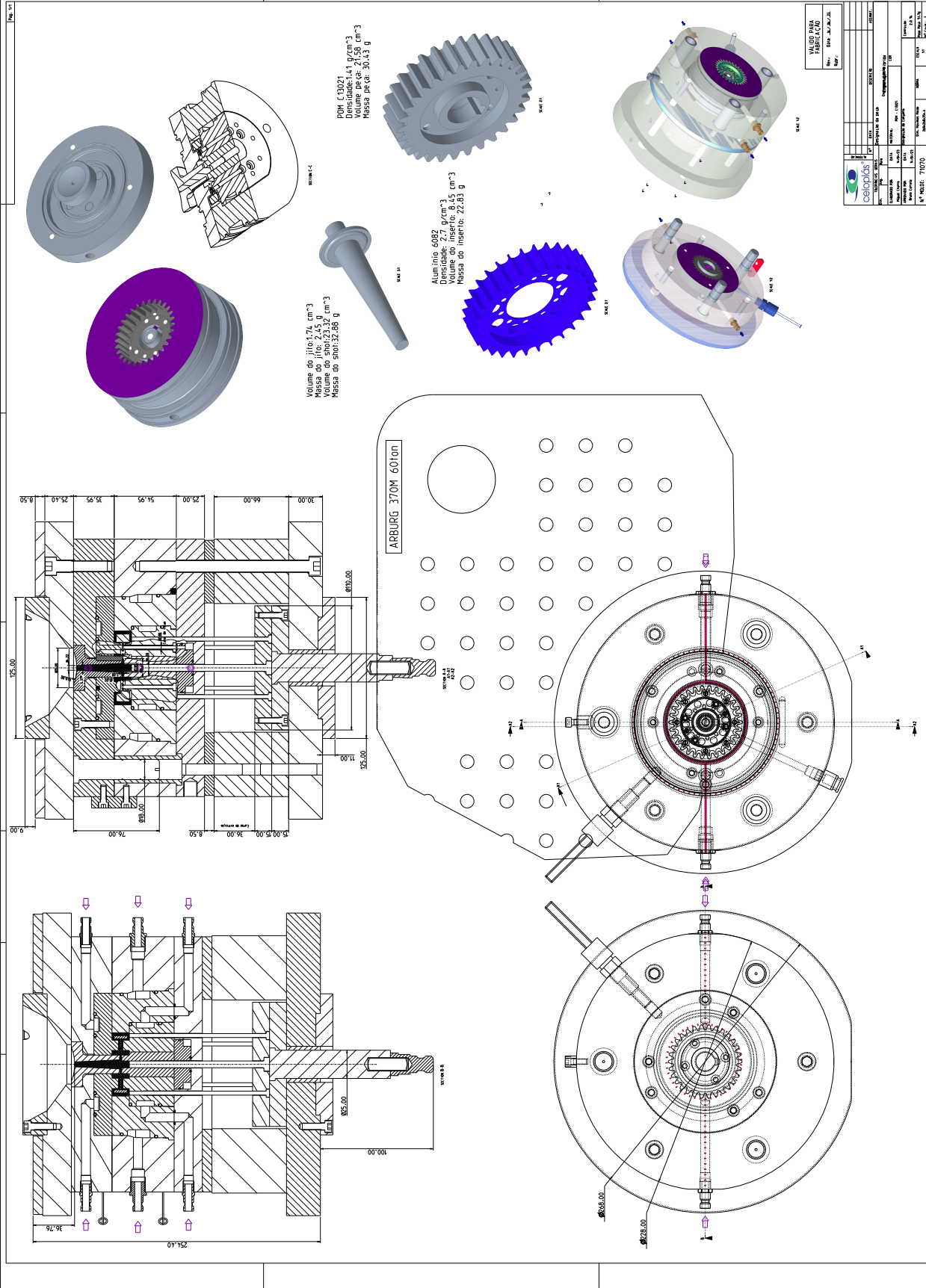
SCALE 1:1

VÁLIDO PARA
FABRICAÇÃO

Rev.: Data: 14 / 04 / 23

Rubr.:

Índice revisão	Elaborou		Observações					Data alteração	
		MOLDE N ^o	Posic./peça	Designação			Quantidade: 1		
		71070	TR	Travão			Dim. Máx: -		
Elaborou		Aprovou	Tolerâncias Gerais		Material	Dureza	Trat. Term./Quim		
M. Charro		B. Correia	Dimen.	Ângulos					
Data		Escala	0 ±0.5		Registo recepção n ^o	Certificado dureza			
14/04/23		2:1	0.0 ±0.1						
			0.00 ±0.01						
			0.000 ±0.005						



Appendix C

Excel Files

C.1 Thermal Calculation

Tinj 210 °C
Tmol 100 °C
Text 140 °C
s 0.015 m
K 0.810569
α_ef 1.10E-07

Y 2.75
tc 166.13 s

Qp 44.52957 W

m 30.43 g 0.03043 kg
hi
he hi-he=cp*ΔT
cp 2210
ΔT=Ti-Tm 110 °C

$$\dot{Q}_{Pl} = \frac{m (h_i - h_e)}{t_c}$$

Qcond -904.684 W

Af 1.29E+05 mm2 0.129241 m2
β 100 W/m2
Ta 20 °C
Tm 90 °C

Qconv -198.54 W

Al 189086 mm2 0.189086 m2
α 15 W/m2
Ta 20 °C
Tm 90 °C

Qrad -26.8207 W

Al 189086 mm2 0.189086
ε 0.25
Crad 5.67 W/m2
θa 293.15 K
θm 363.15 K

$$\dot{Q}_{cond} = A_f \cdot \beta \cdot (T_a - T_m)$$

$$\dot{Q}_{conv} = A_l \cdot \alpha \cdot (T_a - T_m)$$

$$\dot{Q}_{rad} = A_l \cdot \varepsilon \cdot C_{rad} \cdot \left(\left(\frac{\theta_a}{100} \right)^4 - \left(\frac{\theta_m}{100} \right)^4 \right)$$

Qamb -1130.05 W

$$\dot{Q}_{Amb} = \dot{Q}_{cond} + \dot{Q}_{conv} + \dot{Q}_{rad}$$

QTM 1085.516 W

$$\dot{Q}_{TM} = -\dot{Q}_{Pl} - \dot{Q}_{Amb}$$

m 3.889808 L/min
C 4186
ΔT 4

$$\dot{m} = \frac{|\dot{Q}_{TM}|}{C \cdot \Delta T}$$

L 0.338755 m 0.016 m
e 16 mm
k 34 W/mK
d 0.008 m
ΔT=Ti-Tm 120 °C

$$L = \frac{2 |\dot{Q}_{TM}| e}{k \pi d \Delta T}$$

C.2 Dimensional control

	Massa / g	Largura inserto /mm	Largura dente /mm	Ra / micron lateral	Ra / micron tooth radial	Ra / micron tooth axial
1	23.302	12.04	0.99	0.98	0.82	0.99
2	23.293	11.99	0.99	0.83	1.26	1.09
3	23.432	12.05	1.00	0.78	1.07	1.00
4	23.115	12.03	1.00	0.95	1.19	1.05
5	23.398	12.01	1.03	0.81	0.72	0.95
6	23.238	12.04	1.05	0.99	0.88	1.30
7	23.683	12.04	1.07	1.12	0.973	1.18
8	23.341	12.06	1.04	0.91	1.02	1.12
9	23.527	12.05	1.05	1.02	0.83	1.19
10	23.331	12.05	1.03	0.65	0.89	1.50
Media	23.366	12.036	1.025	0.904	0.9653	1.137
D. padrão	0.157215	0.021186998	0.028382311	0.137291256	0.171233466	0.166135821
D. padrão P	0.149148	0.020099751	0.026925824	0.130245921	0.16244633	0.157610279
% STD	0.006728	0.001760302	0.027690059	0.151870858	0.17738886	0.146117697
INT confiança	0.097441	0.013131596	0.017591215	0.085092438	0.106129652	0.102970157
	Min					
1	23.20878	12.014813	0.996617689	0.766708744	0.794066534	0.970864179
10	23.20878	12.014813	0.996617689	0.766708744	0.794066534	0.970864179
	Max					
1	23.52322	12.057187	1.053382311	1.041291256	1.136533466	1.303135821
10	23.52322	12.057187	1.053382311	1.041291256	1.136533466	1.303135821

	Rz / micron lateral	Rz / micron tooth radial	Rz / micron tooth axial
1	5.26	4.25	5.26
2	4.33	6.54	5.55
3	4.45	6.21	5.32
4	5.10	6.23	5.99
5	4.40	3.92	5.09
6	5.21	4.92	6.56
7	5.89	5.34	5.84
8	5.03	5.78	6.02
9	5.58	4.49	6.29
10	3.70	4.66	7.73
Media	4.895	5.234	5.965
D. padrão	0.66183667	0.921112854	0.777821031
D. padrão P	0.627873395	0.87384438	0.737905821
% STD	0.135206674	0.175986407	0.13039749
INT confiança	0.41020308	0.570901171	0.482089611

	Min		
1	4.23316333	4.312887146	5.187178969
10	4.23316333	4.312887146	5.187178969
	Max		
	5.55683667	6.155112854	6.742821031
	5.55683667	6.155112854	6.742821031

	Rq / micron lateral	Rq / micron tooth radial	Rq / micron tooth axial
1	1.23	1.04	1.24
2	1.04	1.57	1.35
3	1.00	1.40	1.26
4	1.18	1.48	1.32
5	1.02	0.92	1.19
6	1.22	1.11	1.59
7	1.40	1.27	1.46
8	1.15	1.31	1.39
9	1.29	1.05	1.50
10	0.82	1.15	1.86
Media	1.135	1.23	1.416
D. padrão	0.167481342	0.210923894	0.199176081
D. padrão P	0.158886752	0.200099975	0.188955021
% STD	0.147560654	0.17148284	0.140661074
INT confiança	0.103804104	0.130729581	0.123448345

	Min		
1	0.967518658	1.019076106	1.216823919
10	0.967518658	1.019076106	1.216823919
	Max		
1	1.302481342	1.440923894	1.615176081
10	1.302481342	1.440923894	1.615176081

	Pt / micron lateral	Pt / micron tooth radial	Pt / micron tooth axial
1	8.09	32.54	10.47
2	7.75	36.75	11.14
3	7.91	13.00	10.94
4	7.76	33.69	10.01
5	7.02	26.68	9.55
6	8.37	19.86	11.18
7	8.47	52.49	10.99
8	7.80	20.87	10.19
9	8.66	21.93	11.47
10	5.7	23.51	13.41
Media	7.753	28.132	10.935
D. padrão	0.85699799	11.20505818	1.057357818
D. padrão P	0.81301968	10.63005155	1.003097702
% STD	0.110537597	0.398302936	0.096694816
INT confiança	0.531163097	6.944839399	0.655345123

	Min		
1	6.89600201	16.92694182	9.877642182
10	6.89600201	16.92694182	9.877642182
	Max		
1	8.60999799	39.33705818	11.99235782
10	8.60999799	39.33705818	11.99235782

	Wt / micron lateral	Wt / micron tooth radial	Wt / micron tooth axial
1	2.28	20.12	4.50
2	3.16	21.10	4.02
3	2.41	5.25	4.69
4	1.63	16.33	2.42
5	1.87	14.21	3.85
6	2.35	12.52	3.81
7	1.96	23.12	5.57
8	1.50	7.71	4.11
9	1.88	13.97	4.87
10	1.41	10.29	6.90
Media	2.045	14.462	4.474
D. padrão	0.523051941	5.835326897	1.186902598
D. padrão P	0.496210641	5.535877166	1.125994671
% STD	0.25577112	0.40349377	0.265288913
INT confiança	0.324184995	3.616706623	0.735636334

	Min		
1	1.521948059	8.626673103	3.287097402
10	1.521948059	8.626673103	3.287097402
	Max		
1	2.568051941	20.2973269	5.660902598
10	2.568051941	20.2973269	5.660902598

C.3 Load Carrying Capacity

DADOS DE ENTRADA			
z1	30		
z2	30		
b_1	15		
b_2	15		
m_n	2 mm		
b_w	15 mm		
α_n	20 °		
β	0 °		
a (entreixo)	60		
ρ_{fPO_1}	0.5	0,25*m	
h_{aPO_1}	2	1*m	
ρ_{fPO_2}	2.5	1,25*m	
ρ_{fPO_2}	0.5		
h_{aPO_2}	2		
h_{fPO_2}	2.5		
T_amb	20 °C		
ED	1		
A_G	0		
x1	0		
x2	0		
K1	0		
K2	0		
Horas serv.			
Horas efect.			

	PINHÃO	RODA
d_1	60	60 mm
d_b1 (base)	56.38156	56.38156 mm
d_w1 (pitch)	60	60 mm
d_a1 (cabeça)	64	64 mm
α_{wt1}	20	20 °
d_f (raiz)	55	55 mm
d_Nf1	57.39881	57.39881 mm
ϵ_1	0.826757	0.826757

MATERIAIS	
E1	2900 MPa
Coef. Poisson	0.44

ϵ_α	1.653514
ϵ_β	0
ϵ_{total}	1.653514

CÁLCULO AUXILIAR PARA A RODA CONDUTORA			
n [rpm]		n [rpm]	1000
P [KW]		T_d [Nm]	9
T_d [Nm]	#DIV/0!	P [KW]	0.942478
		n [rpm]	#DIV/0!

SE RODA CONDUTORA = 1		SE RODA CONDUTORA = 2	
P1	0.15708 KW	P2	
T_d1	6 N.m	T_d2	
n1	250 rpm	n2	
P2	0.15708 KW	P1	
T_d2	6.00 N.m	T_d1	
n2	250 rpm	n1	

P1 [W]	942.478
P2 [W]	942.478
T_d1 [Nm]	9
T_d2 [Nm]	9
n1 [rpm]	1000
n2 [rpm]	1000

n1*z1=n2*z2

TEMPERATURAS

μ

Lubrication	Pairing	μ
Circulating oil	All	0.04
Oil mist	All	0.07
Grease	All	0.09
Dry	plastic/steel	0.2
	PA/PA	0.4
	POM/POM	0.28
	POM/PA	0.18
	PA/PBT	0.35
	POM/PBT	0.18

k_{vFuB}

k_{vFla}

Lubrication	Pairing	k_{vFuB}	k_{vFla}
Circulating oil	All	0	0
Dry/Grease/Oil mist	plastic/plastic	2100	9000
	plastic/steel	900	6300

$R_{\lambda,G}$

Mechanism	$R_{\lambda,G}$
Open with unimpeded entry of air	0
Partially open housing	0,015...0,045
Closed housing	0.06

H_v Grau de perda do dente

n_1 rpm
 vt_1 m/s

n_2 rpm
 vt_2 m/s

PINHÃO

$T1_RAIZ$ °C
 $T1_FLANCO$ °C

$<v_{zul}$
100°C

RODA

$T2_RAIZ$ °C
 $T2_FLANCO$ °C

$<v_{zul}$
100°C

Caso contrário, mudar condições de funcionamento ou da geometria

CAPACIDADE DE CARGA NA RAIZ DO DENTE

K_F=K_A 1.25

	Máquinas Movidas			
Máquinas Motoras	Uniforme	Choques ligeiros	Choque moderado	Choques importantes
Uniforme	1	1.25	1.5	1.75
Choques Ligeiros	1.1	1.35	1.6	1.85
Choques Moderados	1.25	1.5	1.75	2
Choques importantes	1.5	1.75	2	2.25

Y_{Fa_1} 2.6Y_{Sa_1} 1.69

z1 30

x1 0

Y_{Fa_2} 2.6Y_{Sa_2} 1.69

z2 30

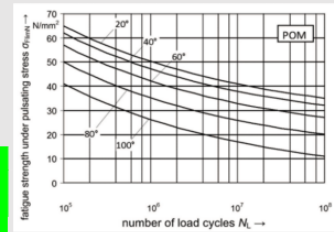
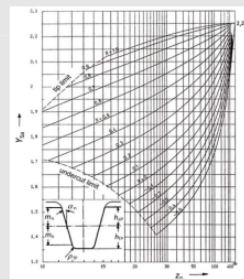
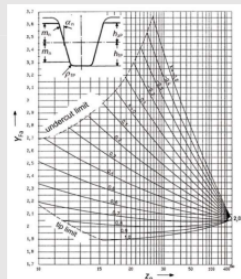
x2 0

Y_ε 0.70358Y_β 1 Se β>30°: β=30°
Se ε_β>1: ε_β=1σ_{FlimN_1} 38.6441σ_{FlimN_2} 38.6441Y_{st} 2σ_{FG} 77.28821S_{Fmin} 2σ_{FP} 38.6441

PINHÃO

F_t 300 Nσ_F 38.6441 N/mm²S_F 2

RODA

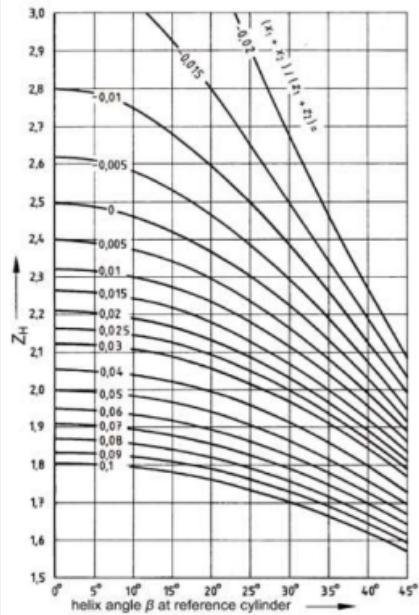
F_t 300 Nσ_F 38.6441 N/mm²S_F 2

CAPACIDADE DE CARGA NO FLANCO DO DENTE (HERTZ STRESS)

Z_E 23.924001
Z_H 2.5
(x1+x2)/(z1+z2) 0

Z_ε 0.8843993
Z_β 1
K_H=K_A 1.25

Z_R 1



Para $\alpha=20^\circ$

σ_{H_1} 48.287148 N/mm²
 $\sigma_{H.lim_1}$ 48.287148
 $S_{H.min}$ 1
 $S.F_1$

σ_{H_2} 48.28715 N/mm²
 $\sigma_{H.lim_2}$ 48.28715
 $S_{H.min}$ 1
 $S.F_2$

RESISTÊNCIA AO DESGASTE

T_d1	9
T_d2	9
N_L1	1.38E+06
N_L2	1.38E+06
H_v	0.149444

kw_1	3.40E-06
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kw_2	3.40E-06
------	----------

I_FI_1	3.553362
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I_FI_2	3.553362
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m_n	2
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W_zul	0.3	(0,1...0,2)*m_n
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(Permissível)

W_m1	2.48E-02	mm
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W_m2	2.48E-02	mm
------	----------	----

Material	Kw*10 ⁻⁶
POM/aço Rz=0,45 µm	1
POM/aço Rz=1,45 µm	3.4
PBT/aço Rz=0,45 µm	3.7
PBT/aço Rz=1,5 µm	7.8

DEFORMAÇÃO DO DENTE

λ_1	0.103448	mm
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λ_2	0.103448	mm
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λ_permissível	0.14	mm	0,07*m_n
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Appendix D

Input Data VDI 2736

$$\alpha_t = \arctan \frac{\tan \alpha_n}{\cos \beta} \quad (\text{D.1})$$

$$\beta_b = \arcsin(\sin \beta \cdot \cos \alpha_n) \quad (\text{D.2})$$

$$m_t = \frac{m_n}{\cos \beta} \quad (\text{D.3})$$

$$d_i = m_t \cdot z_i \quad (\text{D.4})$$

$$d_{bi} = d_i \cdot \cos \alpha_t \quad (\text{D.5})$$

$$d_{wi} = \frac{2 \cdot a \cdot z_i}{z_1 + z_2} \quad (\text{D.6})$$

$$d_{Nai} = d_{ai} = d_i + 2 \cdot h_{aP0} \quad (\text{D.7})$$

$$\alpha_{wt} = \arccos \frac{d_{bi}}{d_{wi}} \quad (\text{D.8})$$

$$x_1 + x_2 = \frac{(\tan \alpha_{wt} - \alpha_{wt} - \tan \alpha_t + \alpha_t) \cdot (z_1 + z_2)}{2 \cdot \tan \alpha_n} \quad (\text{D.9})$$

$$\varepsilon_i = \frac{z_i}{2 \cdot \pi} \cdot \left(\sqrt{\left(\frac{d_{ai}}{d_{bi}} \right)^2 - 1} - \tan \alpha_{wt} \right) \quad (\text{D.10})$$

$$\varepsilon_\alpha = \varepsilon_1 + \varepsilon_2 \quad (\text{D.11})$$

$$\varepsilon_\beta = \frac{b_w \cdot \sin \beta}{m_n \cdot \pi} \quad (\text{D.12})$$

$$d_{Nf1} = \sqrt{\left(2 \cdot a \cdot \sin \alpha_{wt} - \sqrt{d_{a2}^2 - d_{b2}^2}\right)^2 + d_{b1}^2} \quad (\text{D.13})$$

$$d_{Nf2} = \sqrt{\left(2 \cdot a \cdot \sin \alpha_{wt} - \sqrt{d_{a1}^2 - d_{b1}^2}\right)^2 + d_{b2}^2} \quad (\text{D.14})$$