

FACULDADE DE ENGENHARIA DA UNIVERSIDADE DO PORTO

Efficiency of a synchromesh transaxle for automotive applications

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Mestrado em Engenharia Mecânica

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Resumo

Em todos os sistemas mecânicos, a engenharia tenta reduzir as perdas energéticas, melhorando a sua eficiência. No setor automóvel, com o objetivo de melhorar os consumos e, consequentemente, reduzir as emissões, é fundamental analisar e compreender a eficiência de um dos mecanismos principais de um automóvel: a caixa de velocidades. Existem inúmeros fatores que influenciam a eficiência da caixa de velocidades, como a geometria das engrenagens e do cárter, o número de mudanças, o tipo de lubrificante, entre outros. Um dos fatores que influencia esta eficiência é a qualidade do dentado das engrenagens, ou seja, as "tolerâncias geométricas" das mesmas.

Duas versões da caixa de velocidades JT4, fabricadas na HORSE Aveiro, com uma qualidade de dentado superior à da caixa de série, foram alvo de análise de eficiência. Na HORSE, atualmente, apenas são realizados ensaios vibratórios às caixas de velocidades. Devido à necessidade de estudar a eficiência, foi projetada uma adaptação do banco de ensaios do INEGI, localizado no laboratório do Departamento de Engenharia Mecânica da FEUP. No entanto, os resultados experimentais apresentados neste estudo foram obtidos internamente pela HORSE.

Assim foram analisados os resultados dos ensaios vibratórios realizados na HORSE a 80 Nm e 800 rpm. Com os dados registados, foi possível realizar um estudo sobre a eficiência da caixa de velocidades. Os resultados experimentais permitiram uma avaliação global do comportamento da caixa de velocidades a nível energético.

Foi desenvolvido um modelo numérico de perda de potência no KISSdesign, um pacote do KISSsoft, com o objetivo de estimar a perda de potência em função do binário, da velocidade e da temperatura do lubrificante. Uma das vantagens do modelo é a capacidade de dividir a potência perdida pelas suas diversas contribuições: perdas por atrito no engrenamento, chapinagem, e perdas por atrito nos rolamentos e nos vedantes. Assim, com os dados experimentais, foi possível validar o modelo do KISSsoft.

Palavras-chave: Caixa de velocidades • Eficiência energética • Perda de potência • KISSsoft • Qualidade do dentado • Banco de ensaios

Abstract

In all mechanical systems, engineering strives to reduce energy losses, thereby improving their efficiency. In the automotive sector, with the goal of improving fuel consumption and, consequently, reducing emissions, it is essential to analyze and understand the efficiency of vehicle's main mechanisms: the gearbox. There are numerous factors that influence gearbox efficiency, such as the geometry of the gears and the gearbox housing, the number of speeds, and the type of lubricant, among others. One of the factors that most influences this efficiency is gear tooth quality, which can be translated into "geometric tolerances."

Two versions of the JT4 gearbox, manufactured at HORSE Aveiro, with a gear tooth quality superior to series production gearbox, were targeted for efficiency analysis. Currently, at HORSE, only vibrational tests are performed on gearboxes. Due to the need to study efficiency, a comprehensive adaptation of the INEGI test bench, located in the laboratory of the Mechanical Engineering Department at FEUP, was machined and assembled. However, the experimental results presented in this study were obtained internally by HORSE.

Thus, the results of the vibration tests conducted at HORSE at 80 Nm and 800 rpm were analyzed. With the recorded data, it was possible to conduct a study on the efficiency of the gearbox. The experimental results allowed for a comprehensive evaluation of the transmission's energy performance.

A power loss model was developed in KISSdesign, a KISSsoft package, with the aim of estimating power loss as a function of torque, speed, and lubricant temperature. One of the advantages of this model is its ability to divide the lost power into its various contributions: gear meshing friction losses, churning losses, and friction losses in bearings and seals. Therefore, using the experimental data, it was possible to validate the KISSsoft model.

Keywords: Gearbox • Energy efficiency • Power loss • KISSsoft • Gear tooth quality • Test Rig

Résumé

Dans tous les systèmes mécaniques, l'ingénierie s'efforce de réduire les pertes énergétiques, améliorant ainsi leur efficacité. Dans le secteur automobile, dans le but d'améliorer la consommation de carburant et, par conséquent, de réduire les émissions, il est fondamental d'analyser et de comprendre l'efficacité de l'un des principaux mécanismes d'un véhicule : la boîte de vitesses. De facteurs qu'influencent l'efficacité de la boîte de vitesses, tels que la géométrie des engrenages et du carter, le nombre de rapports, le type de lubrifiant, entre autres. L'un des facteurs qui influence le plus cette efficacité est la qualité de la denture des engrenages, c'est-à-dire leurs « tolérances géométriques ».

Deux versions de la boîte de vitesses JT4, produits chez HORSE Aveiro, présentant une qualité de denture supérieure à celle de la boîte de série, ont fait l'objet d'une analyse d'efficacité. Actuellement, chez HORSE, seuls des essais vibratoires sont réalisés sur les boîtes de vitesses. En raison de la nécessité d'étudier l'efficacité, une adaptation complète du banc d'essai de l'INEGI, situé dans le laboratoire du département d'ingénierie mécanique de la FEUP, a été conçue. Cependant, les résultats expérimentaux présentés dans cette étude ont été obtenus en interne par HORSE.

Ainsi, les résultats des essais vibratoires réalisés chez HORSE à 80 Nm et 800 tr/min ont été analysés. Avec les données enregistrées, il a été possible de mener une étude sur l'efficacité de la boîte de vitesses. Les résultats expérimentaux ont permis une évaluation globale du comportement de la boîte de vitesses sur le plan énergétique.

Un modèle de perte de puissance a été développé dans KISSdesign, un module de KISSsoft, dans le but d'estimer la perte de puissance en fonction du couple, de la vitesse et de la température du lubrifiant. L'un des avantages du modèle est sa capacité à diviser la puissance perdue en ses différentes contributions : pertes par frottement lors de l'engrènement, pertes par barbotage, et pertes par frottement dans les roulements et les joints d'étanchéité. Ainsi, grâce aux données expérimentales, il a été possible de valider le modèle KISSsoft.

Mots-clés: Boîte de vitesses • Efficacité énergétique • Perte de puissance • KISSsoft • Qualité de la denture • Banc d'essai

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Contents

1	Introduction	1
1.1	HORSE Aveiro	1
1.2	Objectives	2
1.3	Thesis outline	3
2	Background Theory	4
2.1	Gearbox power loss	4
2.1.1	Churning Loss	5
2.1.2	Gear mesh power Loss	8
2.1.3	Rolling bearing power loss	11
2.1.4	Power loss in seals	16
2.2	Gear Tooth Quality	16
2.2.1	Gear tooth tolerances	16
2.2.2	Roughness	24
3	JT4 Gearbox	26
3.1	Gearbox Mechanism	26
3.2	Gearbox versions	28
3.2.1	Standard Version	29
3.2.2	Golden Version	30
4	Numerical model	36
4.1	Sketcher Model Creation	37
4.2	Boundary Conditions	39
4.3	Switching Matrix Definition	40
4.4	Operating Modes Definition	40
4.5	Component Geometry Definition	41
4.6	Load Definition	48
4.7	Calculation and Results Evaluation	49
4.7.1	Efficiency maps	49
4.7.2	Efficiency Plots	52
4.7.3	Power Loss Plots	53
4.7.4	Gear Load-Dependent Power Loss Plots	56
4.7.5	Total Bearing Power Loss Plots	58
4.7.6	Seals Power Loss and Gear Load-Independent Power Loss Plots	59

5	Experimental campaign	63
5.1	Test Rig	63
5.1.1	Total power losses test rig	63
5.1.2	No-load power losses test rig	65
5.1.3	Adaptation of the FEUP test rig	65
6	Experimental results and Model Verification	78
6.1	HORSE results	78
6.2	Comparison with the results of the standard version	91
7	Conclusion and Future Work	96
7.1	Conclusions	96
7.2	Future Work	97
	References	99
A	2D Technical Drawings of the Manufactured Parts	101

List of Figures

1.1	HORSE Aveiro	1
1.2	HORSE logo	2
2.1	Types of power loss in a gearbox	5
2.2	Geometrical data of the gear immersed surface	6
2.3	Definition of the senses of rotation of a gear pair	7
2.4	Schematic representation of the swell effect	7
2.5	Lubrication conditions along the gear mesh	8
2.6	Stribeck curve	9
2.7	Types and surface roughness	10
2.8	Typical evolution of the CoF along the path of contact for a spur gear	10
2.9	Reverse flow at the inlet of the contact	13
2.10	Bearing frictional moment as a function of the speed and viscosity	16
2.11	Profile slope deviation, ($F_{H\alpha}$)	19
2.12	Lead angle error ($F_{H\beta}$)	19
2.13	Form error	20
2.14	Single pitch error (f_{pt})	20
2.15	Typical single pitch error chart	21
2.16	Typical accumulated pitch error chart (F_p)	21
2.17	Position of pin/ball during runout measurement	22
2.18	Typical runout error chart	22
2.19	Schematic of microgeometry errors and their influence on gear performance . . .	23
2.20	Modifications in microgeometry: Lead crowning	23
2.21	Modifications in the gear microgeometry: Profile crowning	24
3.1	Synchromesh mechanism	26
3.2	Internal mechanism of the JT4 gearbox.	27
3.3	Input Shaft	28
3.4	Output Shaft	28
3.5	Manufacturing line of idle gear in standard version	29
3.6	Manufacturing line of fixed gear in standard version	29
3.7	Manufacturing line of primary shaft in standard version	29
3.8	Manufacturing line of fixed gear in Golden version	30
3.9	Manufacturing line of primary shaft in Golden version	30
3.10	Manufacturing line of idle gear in Golden version	30
3.11	5 th idle gear of JT4 gearbox.	31
3.12	Input Shaft of JT4 gearbox	31
3.13	Grinding tool passage obstructed by the synchronizer ring.	32

3.14	Manufacturing line of idle gear of 1 st Golden version	32
3.15	Output and Input shaft from 1 st Golden version	33
3.16	Manufacturing line of idle gear of 2 nd Golden version	33
3.17	Gears of 2 nd Golden version	34
4.1	Gleason's recommended procedure for designing a gearbox	37
4.2	Sketch of the gearbox model in KISSsoft	38
4.3	Definition of boundary conditions and input and output couplings	39
4.4	Selection of the model's input variables	40
4.5	Switching matrix	40
4.6	Operating modes	40
4.7	Definition of operating modes	41
4.8	Input shaft model	41
4.9	Output shaft model	42
4.10	Reverse Shaft	43
4.11	Differential	44
4.12	Basic Data	46
4.13	Definition of the Gear Tooth Profile	47
4.14	Gear tooth modifications	47
4.15	Model of JT4 gearbox in KISSsoft	48
4.16	Load Definition.	49
4.17	Efficiency Maps for each gear	51
4.18	Efficiency Plots for each gear	53
4.19	Total Power Loss Plots for each gear	55
4.20	Gear Load-Dependent Power Loss Plots for each gear	57
4.21	Total Bearing Power Loss Plots for each gear	59
4.22	Seals Power Loss Plots for each gear	61
4.23	Gear Load-Independent Power Loss Plots for each gear	62
5.1	Top view diagram of the gearbox test rig	64
5.2	No-load power loss test rig	65
5.3	Test rig setups used previously	66
5.4	Incompatibility with pre-existing platforms	67
5.5	Test rig setups used for study of the JT4 gearbox	68
5.6	Input and output interfaces used on the HORSE JT4 gearbox production for NVH testing.	69
5.7	Belt drive assembly.	70
5.8	Main gearbox assembly	72
5.9	Base of main gearbox assembly.	73
5.10	Ring-shaped part for main gearbox assembly.	73
5.11	Differential Lock System	74
5.12	3D-printed seal mounted on the output interface	75
5.13	Central sensor assembly	76
5.14	Slave gearbox assembly	77
6.1	HORSE Aveiro test bench	79
6.2	Data recorded during the NVH test at HORSE	80
6.3	Efficiency derived from the tests carried out at HORSE and from the KISSsoft model-Version 1	81

6.4	Results from the tests carried out at HORSE and from the KISSsoft model- Version 1	82
6.5	Absolute and relative error of Power Loss- Version 1	83
6.6	Absolute and relative error of Torque Loss- Version 1	83
6.7	Contributions to Power Loss	84
6.8	Contributions to Torque Loss	87
6.9	Efficiency derived from the tests carried out at HORSE and from the KISSsoft model-Version 2	89
6.10	Results from the tests carried out at HORSE and from the KISSsoft model- Version 2	90
6.11	Absolute and relative error of Power Loss- Version 2	91
6.12	Absolute and relative error of Torque Loss- Version 2	91
6.13	Contributions to Power Loss and Torque Loss- Standard Version	92
6.14	Direct comparison of efficiency, power loss, and torque loss between the Standard and Golden gearboxes.	94
A.1	2D drawing of part "ajustar_principal_2.0"	102
A.2	2D drawing of part "ajustar_principal_2.0_esquerda"	103
A.3	2D drawing of part "anel"	104
A.4	2D drawing of part "apoio_chumaceira_sensor"	105
A.5	2D drawing of part "apoio_sensor_central_parte2"	106
A.6	2D drawing of assembly "assembly_central"	107
A.7	2D drawing of part "barra_estabilizadora"	108
A.8	2D drawing of part "base_sensor_vazio"	109
A.9	2D drawing of part "bloqueio de diferencial"	110
A.10	2D drawing of part "flange_sensor_vazio_caixa"	111
A.11	2D drawing of part "interface_polia_entrada"	112
A.12	2D drawing of part "interface_saida_caixa_escrava"	113
A.13	2D drawing of part "parte_baixo_apoio_sensor"	114
A.14	2D drawing of part "peças_guia"	115
A.15	2D drawing of part "placa_fixação_escrava_parte1"	116
A.16	2D drawing of part "placa_fixação_escrava_parte2"	117
A.17	2D drawing of part "placa_fixação_parte1"	118
A.18	2D drawing of part "placa_fixação_parte2"	119
A.19	2D drawing of part "placa_fixação_sensor"	120
A.20	2D drawing of part "placa_ligação_banco"	121
A.21	2D drawing of part "placa_suporte_hibrida_escrava"	122
A.22	2D drawing of part "placa_suporte_hibrida_principal_banco3"	123
A.23	2D drawing of part "placa_tensionador"	124
A.24	2D drawing of part "sensor_central_parte1"	125
A.25	2D drawing of part "suporte_chumaceira_veio_banco"	126
A.26	2D drawing of part "suporte_escrava"	127
A.27	2D drawing of part "suporte_mancal_vazio"	128
A.28	2D drawing of part "suporte_placas_sensor"	129
A.29	2D drawing of part "suporte_principal"	130
A.30	2D drawing of part "veio_flange_correia1"	131
A.31	2D drawing of part "veio_polia_entrada"	132
A.32	2D drawing of part "veio_tensor"	133
A.33	2D drawing of assembly "desenho_conjunto_caixa_escrava"	134
A.34	2D drawing of assembly "desenho_conjunto_caixa_principal"	135
A.35	2D drawing of assembly "desenhos_conjunto_correias"	136

List of Tables

2.1	Values of the parameters ψ according to the operating conditions	6
2.2	Criteria for EHD Lubrication Regimes.	9
2.3	Accuracy classes - Examples of applications	17
2.4	Gearing parameters defining gear quality and their formulas (Class 5)	18
2.5	Example of Tolerances for Gear Quality	24
3.1	Tooth Quality of the Serie and Golden Gearboxes	35
4.1	Number of teeth of fixed and idler gears	45
4.2	Gear ratio of each gear	45
4.3	Technical Specifications for Grade SAE 75W	46
5.1	List of mechanical components and references.	71
6.1	NVH test data and results – Version 1	80
6.2	Results from the KISSsoft model – Version 1	81
6.3	Gear loss factors and friction coefficients	85
6.4	NVH test data and results – Version 2	88
6.5	Results from the numerical model – Standard Version	91
6.6	Differences in power loss and efficiency between Golden and Serie versions.	95

Nomenclature

Symbol	Units	Description
b	mm	Gear face width
B	mm	Rolling bearing width
C_{ch}	Nm	Churning torque
C_m	-	Non-dimensional churning torque
C_w	-	Variable used for the calculation of the frictional moment of drag losses
d	mm	Rolling bearing bore diameter
d_m	mm	Bearing mean diameter
D	mm	Rolling bearing outside diameter
d_{sh}	mm	Shaft diameter
D_p	m	Pitch diameter
f_a	-	Variable used for the calculation of the frictional moment of drag losses
$f_{C\alpha}$	μm	Profile crowning deviation
$f_{C\beta}$	μm	Helix crowning deviation
$f_{f\alpha}$	μm	Profile form deviation
$f_{f\beta}$	μm	Helix form deviation
$f_{H\alpha}$	μm	Profile direction deviation
$f_{H\beta}$	μm	Helix direction deviation
f_i'	μm	Tooth-to-tooth tangential composite deviation
f_i''	μm	Tooth-to-tooth radial composite deviation
f_{pt}	μm	Single pitch deviation
f_r	μm	Tooth-to-tooth radial runout
f_t	-	Variable used for the calculation of the frictional moment of drag losses
f_u	μm	Adjacent pitch difference
F_α	μm	Total profile deviation
F_β	μm	Total helix deviation
F_{bt}	N	Base tangent load
F_i'	μm	Total tangential composite deviation
F_i''	μm	Total radial composite deviation
F_N	N	Surface normal force
F_p	μm	Total cumulative pitch deviation
F_{pk}	μm	Cumulative pitch deviation over k pitches
Fr	-	Froude number
F_r	μm	Radial runout

Symbol	Units	Description
g	m/s ²	Gravity acceleration
G_{rr}	Nmm	Variable for the calculation of the rolling frictional moment
G_{sl}	Nmm	Variable for the calculation of the sliding frictional moment
h	m	Oil immersion depth
H	mm	Oil level
h_0	m	Film thickness
H_{tooth}	m	Tooth height
H_V	-	Gear loss factor
i	-	Gear ratio
l_D	-	Variable used for the calculation of the frictional moment of drag losses
i_{rw}	-	Number of rows of the bearing
$K_{ball,roll}$	-	Rolling element related constant
K_L	-	Geometric constant related to the bearing type
K_{rs}	-	Replenishment/starvation coefficient
K_Z	-	Bearing type related geometric constant
l	mm	Contact length
m	m	Gear modulus
M_{drag}	Nmm	Bearing drag frictional torque
M_{rol}	Nmm	Total frictional moment of a bearing
M_{rr}	Nmm	Rolling frictional moment
M_{seal}	Nmm	Frictional moment of the bearing seal
M_{sl}	Nmm	Sliding frictional moment
n	rpm	Rotational speed
n_{input}	rpm	Input speed on the gearbox
n_{output}	rpm	Output speed on the gearbox
p_{bt}	m	transverse base pitch
P	W	Power
P_{IN}	W	Transmitted power
P_{out}	W	Output mechanical power
P_{VZ0}	W	Gears no-load power losses
P_{VZP}	W	Gears load losses
P_V	W	Total power loss
P_{VL}	W	Rolling bearings losses
P_{VD}	W	Seals power loss
P_{VX}	W	Auxiliary losses
R	-	Renard parameter
R_a	μm	Arithmetic mean roughness
Re	-	Reynolds number
R_p	m	Pitch radius
R_s	-	Variable used for the calculation of the frictional moment of drag losses
R_X	mm	Equivalent radius
S_m	m ²	Gear immersion surface area
t	-	Variable used for the calculation of the frictional moment of drag losses
T	Nm	Torque

Symbol	Units	Description
Tol Diam	μm	Diameter tolerance (tip, active tip, root)
Tol S_{bn}	μm	Normal base thickness tolerance
u	-	Gear ratio
$U_{1,2}$	m/s	Surface tangential velocity
v_b	m/s	Base cylinder tangential speed
v_g	m/s	Sliding velocity
V_M	-	Drag loss factor
V_O	m^3	Oil volume
$W f_H \beta$	μm	Maximum range of helix slope deviation
X_L	-	Lubricant parameter
z	-	Wheel number of teeth
α	$^\circ$	Pressure angle
β	$^\circ$	Helix angle
η	%	Overall mechanical efficiency
η_0	$\text{Pa}\cdot\text{s}$	Dynamic viscosity at the oil bath temperature
θ	rad	Angle between wheel center and oil sump level
Λ	-	Specific film thickness
μ_{EHL}	-	Coefficient of friction in full film conditions
μ_{mz}	-	Average gear mesh friction coefficient
μ_{sl}	-	Sliding friction coefficient
ν	cSt	Lubricant kinematic viscosity
ρ_{oil}	kg/m^3	Oil bulk density
σ	m	Composed roughness
ϕ_{ish}	-	Inlet shear heating reduction factor
ϕ_{rs}	-	Kinematic replenishment/starvation reduction factor
ψ_i	-	Non-dimensional coefficient
ω	rad/s	Angular speed

Chapter 1

Introduction

Since their creation, gearboxes have been the object of an exhaustive study and improvement of the characteristics of the meshing, housing and lubricant with the objective of increasing the useful life of the gearbox and efficiency.

In recent years there is an increased care for environmental factors, which in the automotive sector translates into a reduction of fuel and/or energy consumption. This result can be obtained from the optimization of several characteristics of the automobile, such as aerodynamics, weight reduction, regeneration systems and reduction of power losses in the mechanical systems of the automobile.

1.1 HORSE Aveiro

This research was conducted at the global engineering department of HORSE Aveiro (Figure 1.1). The history of HORSE Aveiro, considered one of the pillars of the national industry, dates back to 1981, when the Renault Group founded the Renault Cacia plant. In the first years, the plant produced combustion engines and gearboxes. In 1999, due to organizational transformations, the Cacia plant focused exclusively on the production of gearboxes.



Figure 1.1: HORSE Aveiro [1]

Recently, the Renault Group, to optimize the organization, separated the factories into two groups: those associated with internal combustion and hybrid powertrain called HORSE (Figure 1.2) and those dedicated exclusively to electric powertrain as Ampere. Currently, HORSE Aveiro produces gearboxes, aluminum parts for engines and gearboxes, differential cases, oil pumps, the PEB (Power Electric Box) and electric motors for hybrid applications. At present, NVH (Noise, Vibration, and Harshness) testing is conducted as the primary quality control measure. While these tests are crucial—particularly in hybrid vehicles where any noise or vibration is readily perceived by the driver—the efficiency is only evaluated in the final vehicle. Although the efficiencies of gearboxes are high, a small decrease in power losses can translate into an increase in efficiency and consequent decrease in fuel/energy consumption.



Figure 1.2: HORSE logo [2]

1.2 Objectives

The main objective of this work is to evaluate the influence of gear quality on the efficiency of the JT4 gearbox, using a model developed in KISSsoft. The model will be validated based on experimental tests. With the built model it is possible to know the contributions of the various components of the gearbox to the efficiency losses. An additional objective is to assess the capability of HORSE Aveiro to conduct this type of efficiency analysis internally.

With these objectives established, two Golden versions of the JT4 gearbox were assembled at the HORSE Aveiro plant. These variants feature a superior gear tooth quality compared to the standard production version. Thus, it was possible to verify the influence of gear tooth quality on the overall efficiency of the gearbox.

Despite the constraints and challenges arising from the suspension of activities at the FEUP laboratory, a comprehensive adaptation and assembly of the FEUP test bench were successfully completed, paving the way for future collaborative research between FEUP and HORSE. Due to these constraints, the NVH tests performed at HORSE were studied and the capacity to perform efficiency tests on the company's test bench was developed.

1.3 Thesis outline

This report is divided into 6 main chapters. The order in which the report is organized does not necessarily reflect the order of events.

Chapter 2 introduces the classic model for gearbox power loss. The concept of gear tooth quality is also presented.

In chapter 3, the mechanism of the JT4 manual gearbox is presented and explained, as well as the Golden versions of the gearbox.

In chapter 4, the construction process of the model in the KISSsoft software is presented, as well as the presentation of its results.

Chapter 5 presents all the adaptation that was carried out on the test bench located at FEUP to enable the realization of tests on the JT4 gearbox.

In chapter 6, the experimental results carried out at HORSE and the respective results of the KISSsoft model are presented. It also conducted a theoretical comparison of the efficiency results of the Golden gearbox with those of the standard gearbox.

Chapter 7 was reserved for conclusions of the work carried out, as well as suggestions for future works.

Chapter 2

Background Theory

2.1 Gearbox power loss

It is widely acknowledged that the conception of a mechanical system that is both perfect and loss-free is a fallacy. Gearboxes are not an exception to this phenomenon because every moving part or with relative movement dissipates energy. These power losses result in an output power that is lower than the input power. The overall power loss in a transmission is the result of a combination of various sources of power loss. The configuration of gears, seals, shafts, lubricant, bearings, and housing design in a transmission is virtually limitless. When considered in conjunction with other variables, such as operating, environmental conditions and auxiliary losses, the prediction of transmission efficiency becomes a challenging task that remains a topic of research [3, 4].

In a transmission, total power loss is the sum of losses due to gears, bearings, seals, and auxiliary losses (see Figure 2.1). The total power loss can be categorized into two distinct types: load-dependent and no-load losses. Losses due to gears and bearings involve both types of losses, while losses due to seals and auxiliary losses involve only no-load losses.

No-load losses are the result of the rotation of mechanical components, even in the absence of torque transmission. This phenomenon is influenced by the characteristics of the oil, including its density and viscosity, as well as the depth to which the components are immersed in the oil bath. A range of factors influence these losses, including the design of mechanical components and gearbox housings, as well as operating conditions.

Nevertheless, as the nomenclature implies, load losses are dependent on the torque being transmitted and occur when mechanical components come into contact. A number of other factors have been identified as contributors to this particular type of loss. These include the sliding speed and the coefficient of friction that exists at the points of contact between the components [5].

Auxiliary losses (P_{VX}) refer to all sources of energy dissipation that are not caused by gears, bearings, and seals.

For nominal power transmission, gearbox load losses are the predominant type of loss. In conditions of high speeds and low to moderate loads, no-load losses can become the dominant

source of power dissipation.

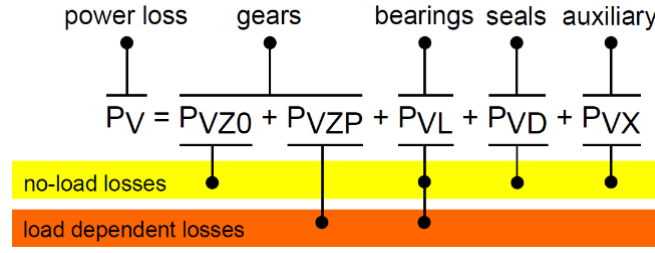


Figure 2.1: Types of power loss in a gearbox [3]

2.1.1 Churning Loss

In the case of splash-lubricated gears, oil churning is typically considered a significant cause of power loss, a phenomenon that is closely associated with the fluid circulations generated by rotating gears that are partially immersed in a lubricant [6].

As proposed by the model presented in Changenet et al. [6], which proved to be a robust model [7], it is possible to predict the churning power loss for individual wheels. The total churning power loss of the gearbox (P_{VZ0}) can be calculated by summing the power loss of each wheel. The gearbox churning power loss with k wheels is expressed in Equation (2.1).

$$P_{VZ0} = \sum_{i=1}^k C_{ch|i} \cdot \omega_i \quad (2.1)$$

Where $C_{ch|i}$ is the torque loss of wheel i given by Equation (2.2) and ω_i is the rotational speed.

$$C_{ch} = \frac{1}{2} \cdot \rho_{oil} \cdot \omega^2 \cdot R_p^3 \cdot S_m \cdot C_m \quad (2.2)$$

Where ρ_{oil} represents the lubricant bulk density, ω the rotation speed, R_p the gear pitch radius, S_m the surface area of contact between the gear and the lubricant and C_m is the drag dimensionless group.

According to Leprince [7], based on the Vashy-Buckingham theorem, it is possible to obtain an expression for the dimensionless drag group given by the following equation:

$$C_m = \psi_1 \cdot \left(\frac{h}{D_p} \right)^{\psi_2} \cdot \left(\frac{V_0}{D_p^3} \right)^{\psi_3} Fr^{\psi_4} \cdot Re_c^{\psi_5} \cdot \left(\frac{b}{R_p} \right)^{\psi_6} \quad (2.3)$$

Where ψ_i are constant coefficients which are adjusted from experimental results, h is the gear immersed depth, V_0 is the oil volume, b is the tooth face width and D_p is the pitch diameter.

The parameters Re_c and Fr are the critical Reynolds and Froude number, respectively, given by Equations (2.4) and (2.5) :

$$Re_c = \frac{R_p \cdot b \cdot \omega}{\nu} \quad (2.4)$$

$$Fr = \frac{R_p \cdot \omega^2}{g} \quad (2.5)$$

Where ν is the kinematic viscosity of lubricant.

The numerical value of the factors, designated as ψ_i , is dependent on the operating conditions. Four operating regimes were identified, characterized by the nature of the flow regime (laminar or turbulent) and a parameter analogous to centrifugal acceleration, given by the following Equation:

$$\gamma = \omega^2 \cdot (R_p \cdot b \cdot m)^{1/3} \quad (2.6)$$

In this equation, m denotes the gear module.

Table 2.1: Values of the parameters ψ according to the operating conditions [4].

Condition	ψ_1	ψ_2	ψ_3	ψ_4	ψ_5	ψ_6
$Re < 4000$ and $\gamma < 750$	1.366	0.45	0.1	-0.6	-0.21	0.21
$Re > 4000$ and $\gamma < 750$	0.239	0.45	0.1	-0.6	0	0.21
$Re < 4000$ and $\gamma > 1250$	20.729	0.1	-0.35	-0.88	-0.21	0.85
$Re > 4000$ and $\gamma > 1250$	3.644	0.85	-0.35	-0.88	0	0.85

Note when $750 \text{ m/s}^2 < \gamma < 1250 \text{ m/s}^2$, interpolation must be performed.

The model presented above was adapted for use with helical gears [6][7]. Since the gear-lubricant contact area is modified by the helix angle, the expression for the immersed surface had to be updated. The updated expression is defined by the Equation (2.7).

$$S_m = R_p^2 \cdot (2 \cdot \theta - \sin(2\theta)) + D_p \cdot b \cdot \theta + 2 \frac{z \cdot \theta \cdot H_{\text{tooth}} \cdot b}{\pi \cdot \cos(\alpha) \cdot \cos(\beta)} \quad (2.7)$$

Where θ represents the angle between wheel center and oil sump level (Figure 2.2), H_{tooth} is the tooth height, z is the number of teeth, α is the pressure angle and β is the helix angle. Finally, it is possible to calculate the total churning power loss of the gearbox using Equation (2.1).

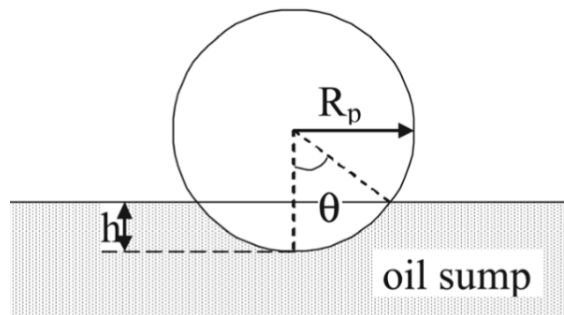


Figure 2.2: Geometrical data of the gear immersed surface [6].

The correlation between experimental and numerical results is good, despite the possibility of deviations resulting from the model does not account lubricant aeration and the direction of rotation.

Lubricant aeration refers to the presence of air in the lubricant in the form of very fine spherical air bubbles caused by the movement of the wheels. It has been proven that this phenomenon is related to churning loss, although it is highly sensitive to the lubricant formulation [4, 8].



Figure 2.3: Definition of the senses of rotation of a gear pair [9]

According to Changenet and Vexé [9], these equations demonstrate a good ability to represent the actual phenomenon for pinion/gear pairs rotating clockwise (Figure 2.3). It has been demonstrated that when the pairs rotate in a counterclockwise direction, this can result in erroneous results. These deviations are likely attributable to the trapping of lubricant by the meshing teeth and to a swelling effect that dissipates energy and increases the immersion depth on the pinion (Figure 2.4).

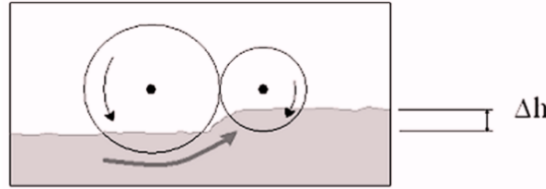


Figure 2.4: Schematic representation of the swell effect [9]

The contribution of the swell effect to churning losses, ΔP_{VZ0} , can be expressed in terms of a dimensionless variation of the churning torque, ΔC_m , given by Equation (2.8).

$$\Delta P_{VZ0} = \frac{1}{2} \cdot \rho \cdot \omega^3 \cdot R_p^3 \cdot S_m \cdot \Delta C_m \quad (2.8)$$

The following conclusions were drawn from experimental results [9]:

- For isothermal conditions, the viscosity of lubricant has a weak influence on ΔP_{VZ0} and consequently the Reynolds number is discarded in the formulation;
- ΔP_{VZ0} is sensitive to rotational speed and is thus considered to be proportional to $Fr^{-0.68}$;

- In the case of identical pinions and gears, no swell effect can be generated due to symmetry. In these conditions ΔP_{VZ0} is found to be close to zero, thus demonstrating that the air-lubricant trapping by the teeth is negligible and consequently proving that the swell effect is prominent.

Taking the previously mentioned facts into account, it was possible to derive the expression for churning torque [9]:

$$\Delta C_m = 17.7 \cdot Fr^{-0.68} \cdot \frac{u-1}{u^8} \cdot \left[1 - \left(\frac{h}{R_p} \right)_{\text{gear}} \right] \quad (2.9)$$

Where u is the gear ratio.

2.1.2 Gear mesh power Loss

Gear friction during gear meshing is one of the main factors contributing to power loss and is generally considered the primary source. Ohlendorf [10] proposes the classical model for gear power loss. The model is expressed in Equation (2.10).

$$P_{VZP} = P_{in} \cdot \mu_{mz} \cdot H_v \quad (2.10)$$

In Equation 2.10, P_{in} is the effective power that is transmitted, μ_{mz} is the average coefficient of friction along the meshing line and H_v is the gear loss factor.

The coefficient of friction is a pivotal component in the study of gear meshing, as it exerts a significant influence on power loss, contact temperature, and the probability of failure. The definition of this parameter is a highly complex process, due to its dependence on the physical and chemical properties of the lubricant, as well as the micro- and macro-geometry of the surfaces and the load distribution. Its definition becomes even more complex because different lubrication regimes can coexist along the meshing line (Figure 2.5) [3, 8].

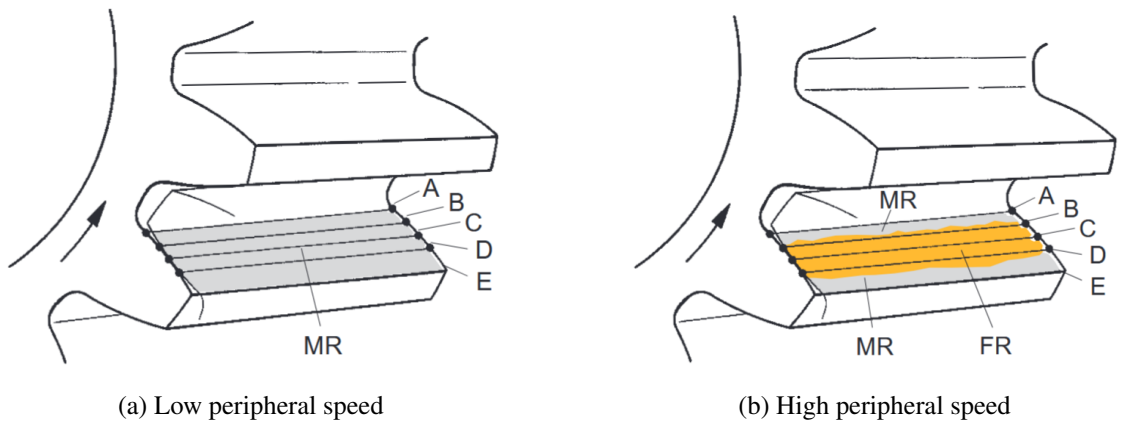


Figure 2.5: Lubrication conditions along the gear mesh [11].

Where MR is mixed film lubrication and FR is full film lubrication.

The lubrication regime is defined by the specific film thickness, Λ . The specific film thickness is given by the Equation (2.11).

$$\Lambda = \frac{h_0}{\sigma} \quad (2.11)$$

Where h_0 is the thickness of the lubricating film at the center of contact and σ is the composite surface roughness. Table 2.2 shows the lubrication regimes as a function of the specific film thickness.

Table 2.2: Criteria for EHD Lubrication Regimes.

<i>EHD Lubrication Regimes</i>		
$\Lambda = \frac{h_0}{\sigma}$		<i>Observations</i>
Full film	$\Lambda \geq 2.0$	Contact surfaces completely separated by the EHD lubricant film. The normal force to the contact is supported by the lubricant film.
Mixed film	$0.7 < \Lambda < 2.0$	Surfaces in contact partially separated by the lubricating film, at some points metal-to-metal contact occurs. The normal force is supported by the metal-metal contact and the EHD lubricant film.
Boundary film	$\Lambda \leq 0.7$	In this case, the lubricant film separating the surfaces is reduced, with metal-to-metal contact predominating. The normal force to the contact is fully supported by the metal-metal contact between surface roughness peaks, causing higher friction and surface wear.

The evolution of the coefficient of friction with the lubrication regime is shown in Figure 2.6 . It can be seen that the lowest coefficient of friction is obtained in the mixed-film regime.

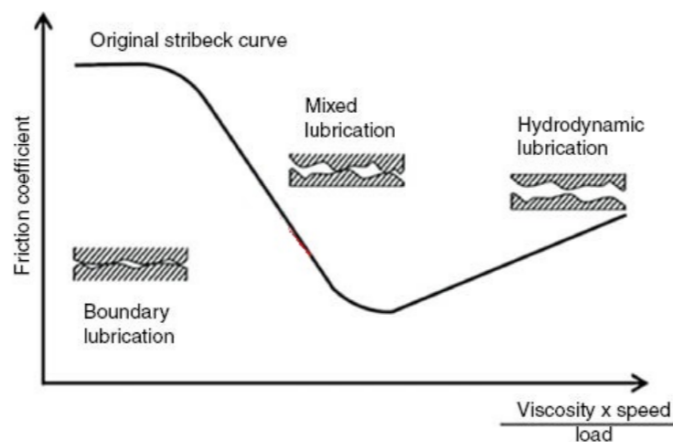


Figure 2.6: Stribeck curve [4]

It is important to note that the orientation and amplitude of surface roughness affect the specific film thickness and, consequently, the coefficient of friction. Under the same operating conditions, surfaces with transverse roughness orientation—such as gears—have a higher coefficient of friction (Figure 2.7) [4, 8].

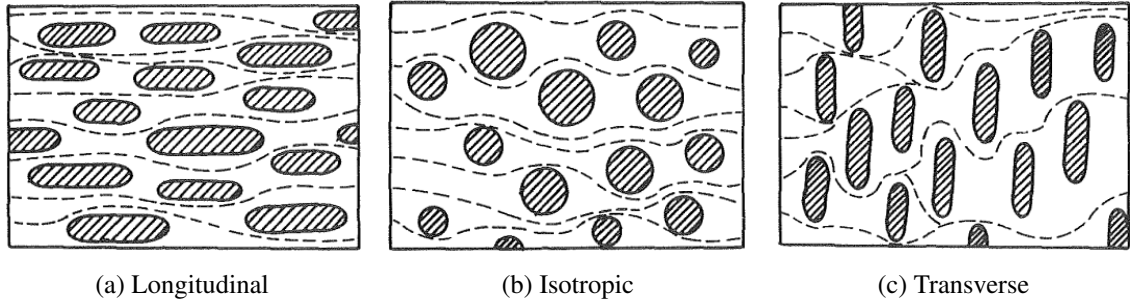


Figure 2.7: Types and surface roughness [12].

As previously outlined, the coefficient of friction is a complex parameter whose value varies along the contact path (see Figure 2.8).

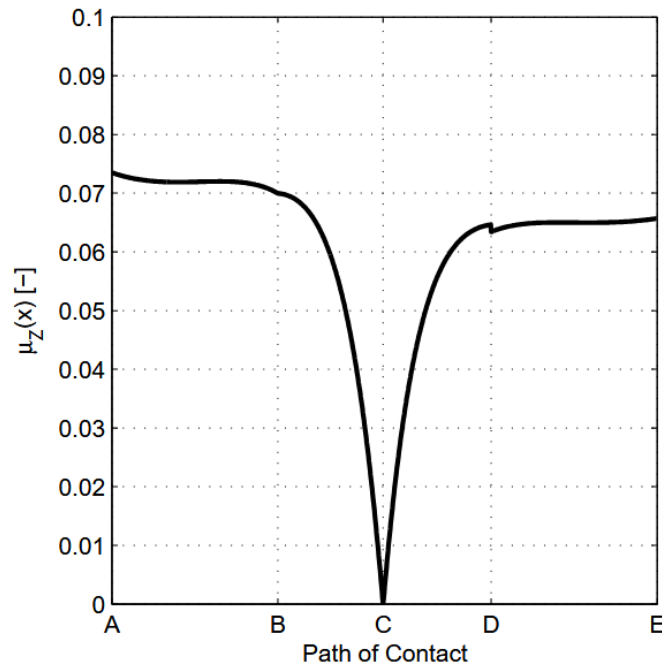


Figure 2.8: Typical evolution of the CoF along the path of contact for a spur gear [3]

For these reasons, the analytical determination of the coefficient of friction is not straightforward. Therefore, when calculating gear mesh power loss, an average coefficient of friction is used, and this parameter is obtained empirically. One of the models for estimating the average coefficient of friction was proposed by Schlenk [13]. It is one of the most widely used because it takes into account not only the oil viscosity but also the influence of the oil formulation; furthermore, it

includes the applied load, the gear geometry, the velocity, and the surface finish. Schlenk's model is given by the Equation (2.12).

$$\mu_{mz} = 0.048 \cdot \left(\frac{F_{bt}}{l \cdot R_x \cdot (U_1 + U_2)} \right)^{0.20} \cdot \left(\frac{1}{\eta_0} \right)^{0.05} \cdot R_a^{0.25} \cdot X_L \quad (2.12)$$

Where:

- F_{bt} is the base tangent load [N];
- l is the contact length [mm];
- $(U_1 + U_2)$ is the sum velocity at the pitch point [m/s];
- R_x is the equivalent curvature radius at the pitch point [mm];
- η_0 is the oil dynamic viscosity at the oil bath temperature [Pa·s];
- R_a is the arithmetic mean surface roughness [μm];
- X_L is the lubricant parameter.

The gear loss factor takes into account the load distribution and sliding velocity along the path of contact and it is an indicator of the efficiency associated to a certain gear. According to Wimmer [14], the general formula for this factor is given by Equation (2.13). Unlike the simplified equations, this formulation is applicable to high-contact-ratio gears, helical gears, and gears with profile modifications.

$$H_V^{num} = \frac{1}{p_{bt}} \int_0^b \int_A^E \left[\frac{F_N(x, y)}{F_{bt}} \cdot \frac{v_g(x, y)}{v_b} \right] dx dy \quad (2.13)$$

Where F_N is the normal force, F_{bt} is the base tangent load, v_g is the sliding velocity, v_b is the absolute tangential speed on the base plane and p_{bt} is the transverse base pitch.

2.1.3 Rolling bearing power loss

A gearbox contains bearings that support the loads transmitted by the gears to the shaft allowing the gears to rotate with minimal friction, while restricting all other degrees of freedom. Rolling bearings provide high precision and low friction and therefore enable high rotational speeds while reducing noise, heat, energy consumption and wear. Given the critical role of bearings in mechanical systems, it was necessary to identify and model the mechanisms of power loss caused by these mechanical components.

This section will discuss SKF's model for calculating friction torque, which has undergone changes over the years and enables an accurate calculation of the friction torque generated in bearings [4].

SKF model

The old SKF model (Modified Arvid Palmgren) divided the frictional moment into a load-independent moment and a load-dependent moment [3]. However, in this model, the division of the frictional moment is based on the type of friction source rather than on its dependence on load. According to this model, the frictional moment is the sum of four different physical sources of torque loss and is given by the following Equation [15, 16].

$$M^{rol} = M_{rr} + M_{sl} + M_{seal} + M_{drag} \quad (2.14)$$

Where:

- M^{rol} is the total frictional moment, Nmm;
- M_{rr} is the rolling frictional moment, Nmm;
- M_{sl} is the sliding frictional moment, Nmm;
- M_{seal} is the frictional moment of the seal(s) , Nmm;
- M_{drag} is the frictional moment of drag losses, churning, splashing etc, Nmm.

The total power loss in rolling bearings is given by Equation (2.15).

$$P_{VL} = M^{rol} \cdot n \cdot \frac{\pi}{30} \cdot 10^{-3} \quad (2.15)$$

Rolling frictional moment

The Rolling frictional moment is given by the following Equation.

$$M_{rr} = \phi_{ish} \cdot \phi_{rs} \cdot G_{rr} \cdot (v \cdot n)^{0.6} \quad (2.16)$$

The variable G_{rr} depends on the bearing type, bearing mean diameter and load (radial and axial). In order to make the model more closely resemble actual behavior and obtain a more accurate calculation, factors ϕ_{ish} and ϕ_{rs} were added to account for the effects of inlet shear heating reduction and high-speed replenishment/starvation of rolling friction, respectively.

The inlet shear heating reduction factor, ϕ_{ish} , takes into account the fact that not all of the lubricant go through the contacts. The excess lubricant creates a reverse flow (Figure 2.9) that generates heat, reducing the oil's viscosity and, consequently, reducing the film thickness and the rolling friction component. The inlet shear heating reduction factor is given by the Equation (2.17), where d_m is the bearing mean diameter.

$$\phi_{ish} = \frac{1}{1 + 1.84 \cdot 10^{-9} \cdot (n \cdot d_m)^{1.28} \cdot v^{0.64}} \quad (2.17)$$

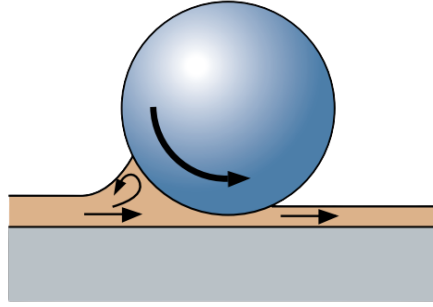


Figure 2.9: Reverse flow at the inlet of the contact [16].

The kinematic replenishment/starvation reduction factor accounts for the fact that the lubricant at the edges of the contact points may not have enough time to replenish the raceways due to the bearing high speed or high viscosity. This effect is called “kinematic starvation” and causes a drop in the film thickness and in the rolling friction. This kinematic replenishment/starvation reduction factor can be obtained using the following Equation.

$$\phi_{rs} = \frac{1}{e^{K_{rs} \cdot v \cdot n \cdot (d+D) \cdot \sqrt{\frac{K_z}{2 \cdot (D-d)}}}} \quad (2.18)$$

Where:

- K_{rs} is the replenishment/starvation constant;
- K_z is a rolling bearing type related geometry constant.
- D is the rolling bearing outside diameter.
- d is the rolling bearing bore diameter

Sliding frictional moment

The major source of friction during the operation of rolling bearings is sliding. The sliding friction moment can be calculated using the following Equation:

$$M_{sl} = G_{sl} \cdot \mu_{sl} \quad (2.19)$$

Where G_{sl} is a variable that depends on the bearing type, bearing mean diameter and load (radial and axial) while μ_{sl} is sliding friction coefficient. This last parameter can be calculated by the Equation (2.20).

$$\mu_{sl} = \phi_{bl} \cdot \mu_{bl} + (1 - \phi_{bl}) \cdot \mu_{EHL} \quad (2.20)$$

Where:

- ϕ_{bl} is the weighting factor for the sliding coefficient of friction and can be calculated using the Equation (2.21);

- μ_{EHL} is the coefficient of friction in full film conditions, which depends on the oil type;
- μ_{bl} is the coefficient of friction in boundary lubrication regime, being implemented by the oil additives.

$$\phi_{bl} = \frac{1}{e^{2.6 \times 10^{-8} \cdot (n \cdot v)^{1.4} \cdot d_m}} \quad (2.21)$$

Friction moment of seals

Bearing seals play a crucial role in preventing contamination and retain the lubricant in the housing. This extends the service life of the bearings [4]. In some cases, the losses caused by the seals may be greater than those caused by the bearing.

For bearings that are sealed on both sides, the frictional moment is given by the Equation (2.22).

$$M_{seal} = K_{S1} \cdot d_S^\beta + K_{S2} \quad (2.22)$$

Where $K_{S1,2}$ and β depend on the seal and bearing type and size, while d_S is the seal counter-face diameter [16].

Drag losses in oil bath lubrication

In case of oil bath lubrication, the rolling bearings are partially, or in special situations, completely immersed in lubricant. The movement of the bearing in the oil bath generates a frictional moment that should not be ignored. Drag losses depend on the bearing's rotational speed, the oil's viscosity, the oil level, and the size and geometry of the reservoir. SKF provides Equations (2.23) and (2.24) for drag losses in ball and roller bearings, respectively.

$$M_{drag}^{ball} = 0.4 \cdot V_M \cdot K_{ball} \cdot d_m^5 \cdot n^2 + 1.093 \cdot 10^{-7} \cdot n^2 \cdot d_m^3 \cdot \left(\frac{n \cdot d_m^2 \cdot f_t}{v} \right)^{-1.379} \cdot R_s \quad (2.23)$$

$$M_{drag}^{roll} = 4 \cdot V_M \cdot K_{roll} \cdot C_W \cdot B \cdot d_m^4 \cdot n^2 + 1.093 \cdot 10^{-7} \cdot n^2 \cdot d_m^3 \cdot \left(\frac{n \cdot d_m^2 \cdot f_t}{v} \right)^{-1.379} \cdot R_s \quad (2.24)$$

The variables and functions used to calculate the frictional moment of drag losses are given by Equations (2.25) to (2.32).

$$K_{ball} = \frac{i_{rw} \cdot K_z \cdot (d + D)}{D - d} \cdot 10^{-12} \quad (2.25)$$

$$K_{roll} = \frac{K_L \cdot K_Z \cdot (d + D)}{D - d} \cdot 10^{-12} \quad (2.26)$$

$$C_w = 2.789 \cdot 10^{-10} \cdot l_D^3 - 2.786 \cdot 10^{-4} + 0.0195 \cdot l_D + 0.6439 \quad (2.27)$$

$$l_D = 5 \cdot \frac{K_L \cdot B}{d_m} \quad (2.28)$$

$$f_t = \begin{cases} \sin(0.5 \cdot t), & \text{when } 0 \leq t \leq \pi \\ 1, & \text{when } \pi < t < 2 \cdot \pi \end{cases} \quad (2.29)$$

$$R_s = 0.36 \cdot d_m^2 \cdot (t - \sin(t)) \cdot f_A \quad (2.30)$$

$$t = 2 \cdot \cos^{-1} \left(\frac{0.6 \cdot d_m - H}{0.6 \cdot d_m} \right) \quad \text{when } H \geq 1.2 \cdot d_m, \text{ use } H = 1.2 \cdot d_m \quad (2.31)$$

$$f_A = 0.05 \cdot \frac{K_z \cdot (D + d)}{D - d} \quad (2.32)$$

Where:

- V_m is the drag loss factor that is a function of oil level;
- B is the bearing width, mm;
- d_m is the bearing mean diameter, mm;
- d is the bearing bore diameter, mm;
- D is the bearing outside diameter, mm;
- H is the oil level, mm;
- i_{rw} is the number of ball rows;
- K_Z is a bearing type related geometric constant;
- K_L is a roller bearing type related geometric constant;
- n is the rotational speed, rpm;
- v is the actual operating viscosity of the lubricant, mm²/s.

Figure 2.10 shows the variation of the bearing frictional moment as a function of speed and viscosity.

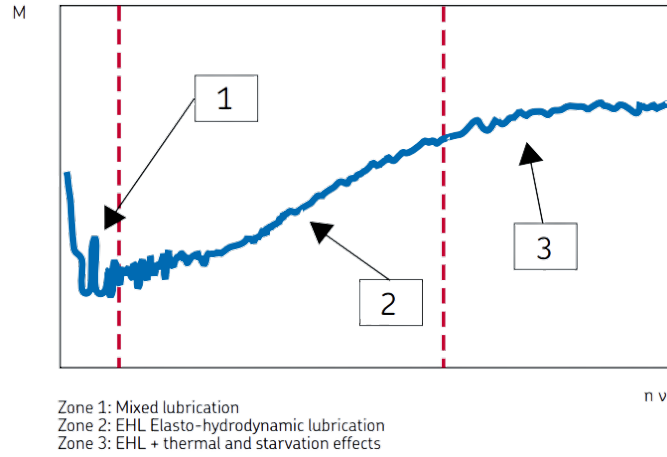


Figure 2.10: Bearing frictional moment as a function of the speed and viscosity [16].

2.1.4 Power loss in seals

The seal component is used on the shaft to avoid air and oil leakage along with minimum friction losses. In most cases, power losses through the seals are very small—about 0.01% of the total transmitted power—and can therefore be considered negligible. Nevertheless, the manufacturer SIMIRIT [17] has modeled the losses due to the seals using Equation (2.33).

$$P_{VD} = 7.69 \times 10^{-6} \cdot d_{sh}^2 \cdot n \quad (2.33)$$

Where:

- P_{VD} is the power loss generated by the seals, W,
- d_{sh} is the shaft diameter, mm,
- n is the shaft rotational speed, rpm.

2.2 Gear Tooth Quality

2.2.1 Gear tooth tolerances

Given the fact that the gearbox in our study is manufactured by the Renault Group, this section will be based on the NF E 02-204 standard [18]. In accordance with the ISO standard, the quality of gears can be categorized into 13 distinct classes ranging from 0 to 12. The quality of the gearing utilized in automotive transmissions ranges from class 5 to 9. Table 2.3 lists the gear quality, manufacturing process, and respective application.

It should be noted that, in the JT4 transmission (gearbox under study), the required grade is 8 for the forward gears and 9 for the reverse gears. The transmissions under study (Golden Gearbox) have a grade of 7 for all gears.

Table 2.3: Accuracy classes - Examples of applications (Adapted from [18]).

ACCURACY CLASSES	EXAMPLES OF MANUFACTURING METHODS	EXAMPLES OF APPLICATION
≤ 6	Finishing by grinding	- heavy-duty truck gears - high precision equipment
7	Finishing by grinding or shaving	- general machine tool mechanics - forward gears of gearboxes
8	Finishing by shaving Finishing by rolling	- forward gears of gearboxes
9	Finishing by shaving Cutting without rework of the blank Rolling Broaching Forging	- reverse gears of gearboxes - dog clutch gears - steering pinion - camshaft pinion - tachometer pinion
10	Cutting without rework of the blank Forging Casting	- rotating part for which acoustic quality is not required - starter ring gear - static mechanism

Table 2.4 lists the gear parameters and the corresponding formulas for calculating ISO standard tolerances (ISO 1328). The standard tolerances are expressed in terms of the normal module m_n , the pitch diameter d , the tooth face width b and the number of teeth z . The table was established for class 5, with the tolerances for the other grades being derived using a geometric progression with a ratio of $\sqrt{2}$ (except for parameter 2b).

Table 2.4: Gearing parameters defining gear quality and their formulas (Class 5) (Adapted from [18]).

No.	Parameter	Formula (class 5)	Ratio
1	$f_{H\alpha}$	$c \cdot (2\sqrt{m_n} + 0.14\sqrt{d} + 0.5)$ with $c = 1$ for $z \geq 18$ and $c = 1 + 0.06 \cdot (18 - z)$ for $z < 18$	$\sqrt{2}$
2a	$f_{C\alpha}$	$1.2\sqrt{m_n} + 2 - 0.8\sqrt[3]{m_n}$	$\sqrt{2}$
3	$f_{f\alpha}$	$0.4 \cdot (2.5\sqrt{m_n} + 0.17\sqrt{d} + 0.5)$	$\sqrt{2}$
4	F_α	$3.2\sqrt{m_n} + 0.22\sqrt{d} + 0.7$	$\sqrt{2}$
5	$f_{H\beta}$	$0.07\sqrt{d} + 0.45\sqrt{b} + 3$	$\sqrt{2}$
6	$W f_{H\beta}$	$f_{H\beta} \cdot \sqrt{2}$	$\sqrt{2}$
2b	$f_{C\beta}$	$0.15b + 2$	$\sqrt{1.5}$
7	$f_{f\beta}$	$0.4 \cdot (0.07\sqrt{d} + 0.45\sqrt{b} + 3)$	$\sqrt{2}$
8	F_β	$0.1\sqrt{d} + 0.63\sqrt{b} + 4.2$	$\sqrt{2}$
9	Tol Diam	$20m_n + 105$	$\sqrt{2}$
10	Tol S_{bn}	$3f_{pt}$	$\sqrt{2}$
11	F_r	$0.24m_n + \sqrt{d} + 5.6$	$\sqrt{2}$
12	f_r	$0.8f_{pt}$	$\sqrt{2}$
13	f_{pt}	$0.3 \cdot (m_n + 0.4\sqrt{d}) + 4$	$\sqrt{2}$
14	F_{pk}	$f_{pt} + 1.6\sqrt{(k-1) \cdot m_n}$	$\sqrt{2}$
15	F_p	$0.3m_n + 1.25\sqrt{d} + 7$	$\sqrt{2}$
16	$f_u \text{ Max}$	$f_{pt} \times 4/3$	$\sqrt{2}$
17	F'_i	$F_p + f'_i$	$\sqrt{2}$
18	f'_i	$0.5 \cdot (9 + 0.3m_n + 3.2\sqrt{m_n} + 0.34\sqrt{d})$	$\sqrt{2}$
19	F''_i	$3.2m_n + 1.01\sqrt{d} + 6.4$	$\sqrt{2}$
20	f''_i	$2.96m_n + 0.01\sqrt{d} + 0.8$	$\sqrt{2}$

Slope error

Profile slope deviation ($F_{H\alpha}$) is the difference between the nominal involute slope and the actual involute slope (Figure 2.11).

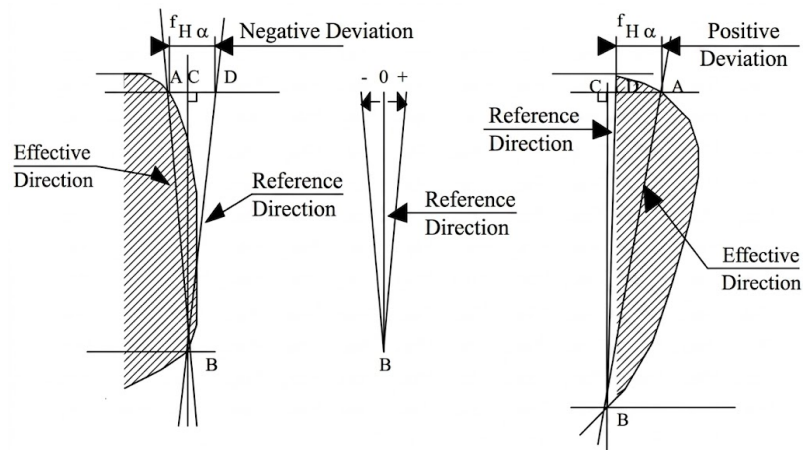


Figure 2.11: Profile slope deviation, ($F_{H\alpha}$) (Adapted from [18])

Lead slope error ($F_{H\beta}$) is the difference between the nominal helix slope and the actual helix slope (Figure 2.12).

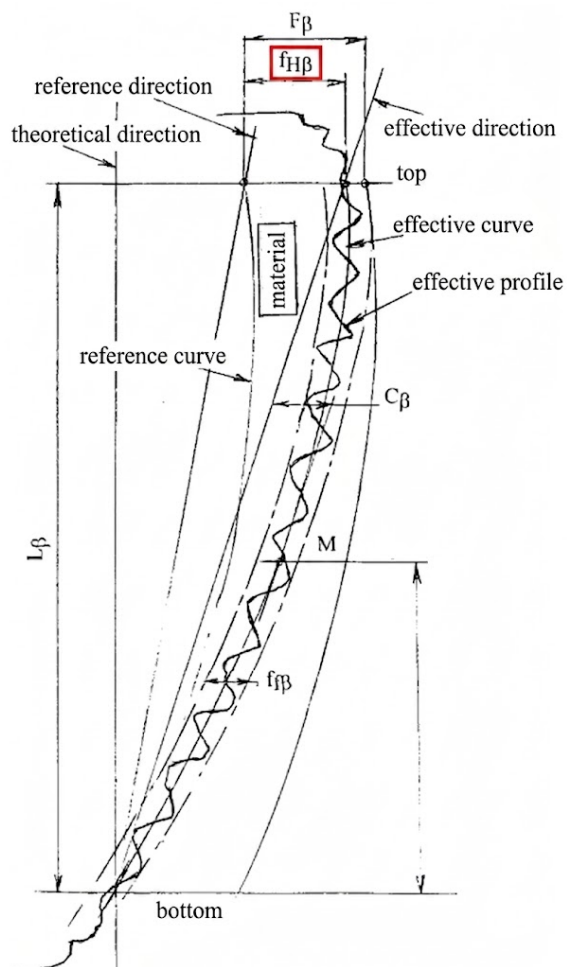


Figure 2.12: Lead angle error ($F_{H\beta}$) (Adapted from [18])

Form error

Form error f_f is the distance between two replicas of the centerline, each positioned at a constant distance from the centerline, so as to enclose the actual profile along the length of the functional profile (Figure 2.13). This concept can be applied to the tooth profile (profile form error $f_{f\alpha}$) and to the helix/lead line (lead form error $f_{f\beta}$).

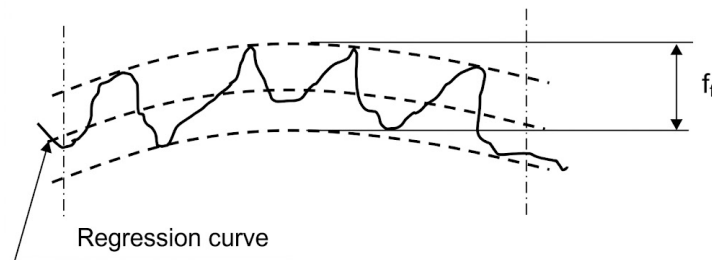


Figure 2.13: Form error (Adapted from [18])

Total error

The total error is the sum of the form error and the slope error. This concept is applicable to the tooth profile (F_α) and to the helix/lead line (F_β).

Pitch error

Pitch error is defined as the angular deviation of the gear teeth from the theoretical position along the pitch circle of the gear.

Single pitch error (f_{pt}) is the algebraic difference between the theoretical pitch and the actual pitch of a pair of teeth. Figure 2.14 shows the pitch error, and Figure 2.15 depicts a typical single-pitch error chart.

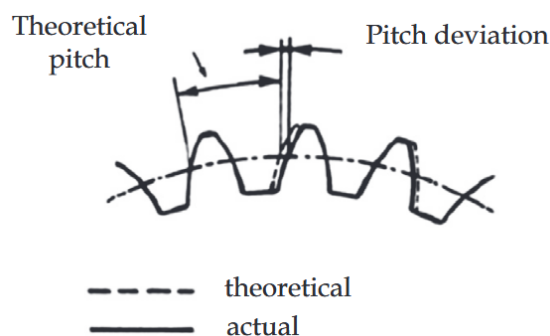


Figure 2.14: Single pitch error (f_{pt}) [19]

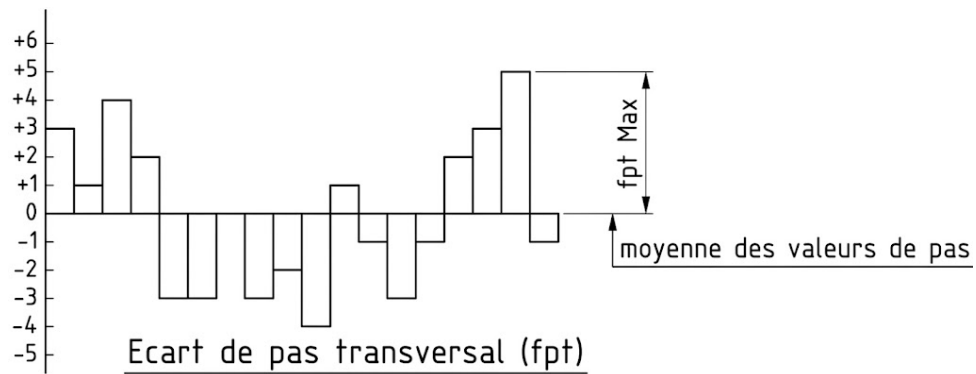
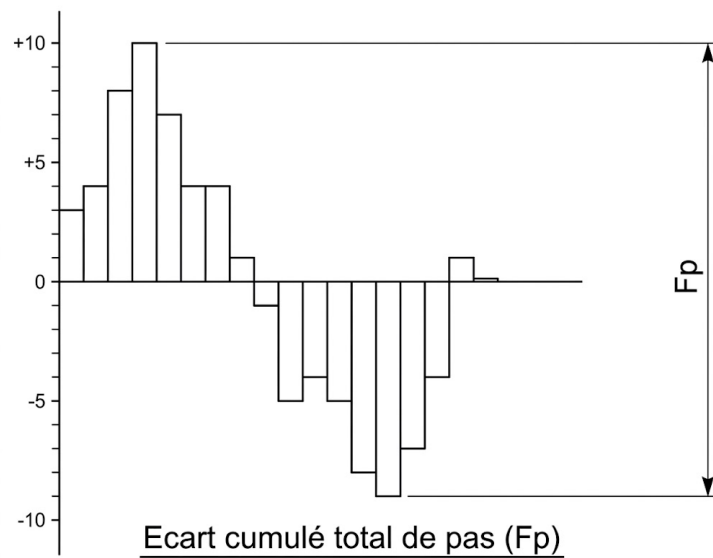


Figure 2.15: Typical single pitch error chart [18].

Accumulated pitch error (F_p) is defined as the maximum value of location inaccuracy between any two teeth of a gear along its pitch circle. It is the sum of the pitch errors of a gear over one revolution. In Figure 2.16, you can see a typical accumulated pitch error chart. This chart shows the cumulative sum of the values of f_{pt} shown in Figure 2.15.

Figure 2.16: Typical accumulated pitch error chart (F_p) [18].

Runout Error

The measurement of runout error F_r is achieved by indicating the position of a pin or ball inserted into each tooth space around the gear (Figure 2.17) and taking the difference between the maximum and minimum measured values. It is important to note that the values of runout include eccentricity. The values for this parameter are typically plotted on a chart, as shown in the Figure 2.18.

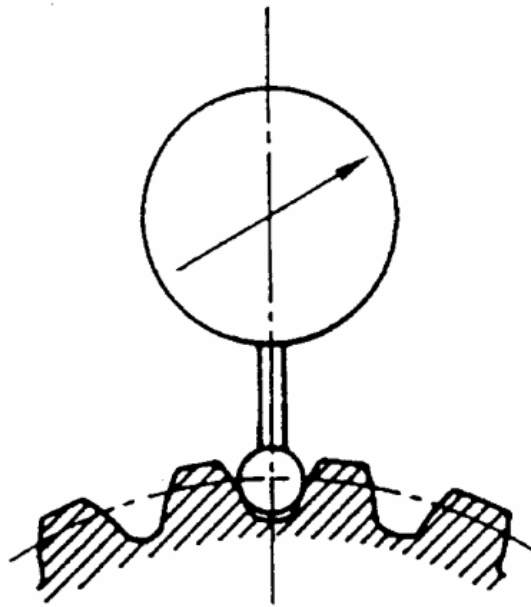


Figure 2.17: Position of pin/ball during runout measurement [18].

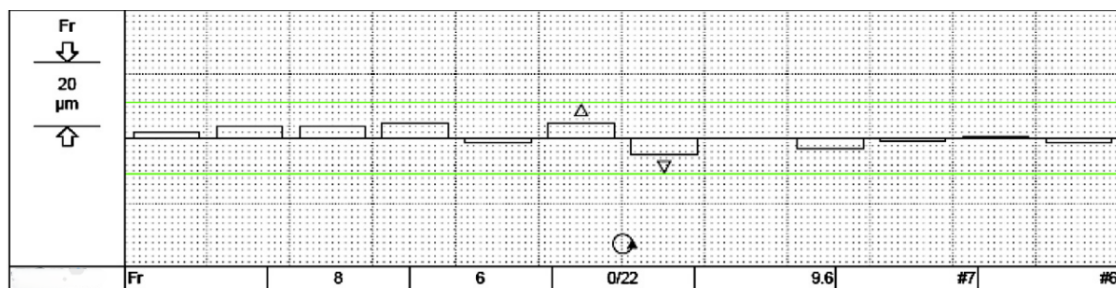


Figure 2.18: Typical runout error chart [19].

The diagram shown in Figure 2.19 summarizes the microgeometric parameters analyzed, grouping them into form and location errors. According to Gupta, Jain, and Laubscher [19], profile errors are the main source of noise. These errors induce localized contact stresses that may adversely affect wear and fatigue properties, thereby affecting the strength and durability of the gear.

Lead errors cause problems with gear meshing, leading to uneven power transmission, which results in localized stress and increased wear. This type of error has a significant impact on the gear's torque transmission capacity.

Pitch errors and runout can cause noise and vibration, leading to structural integrity issues.

The quality of the gear teeth affects the tolerances of these parameters, altering the vibration level, transmission errors, and coefficient of friction, which in turn affects power losses [20–22].

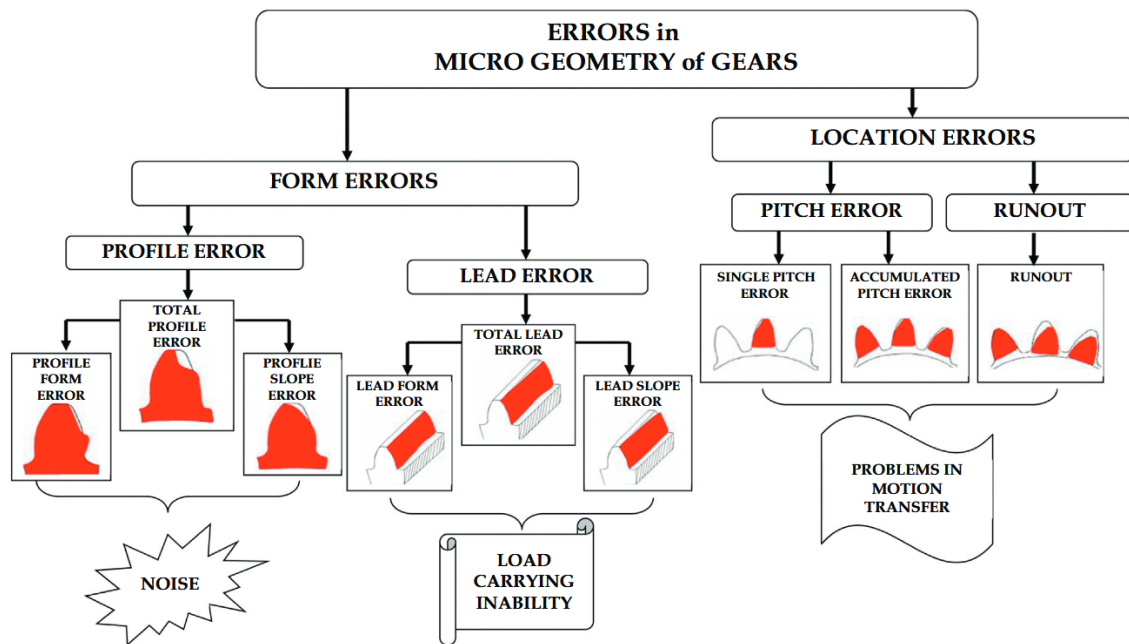
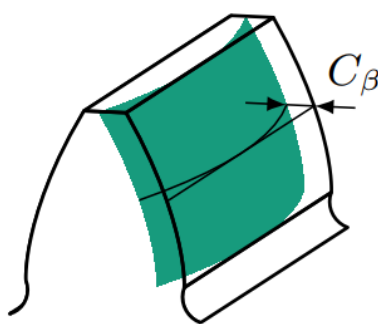


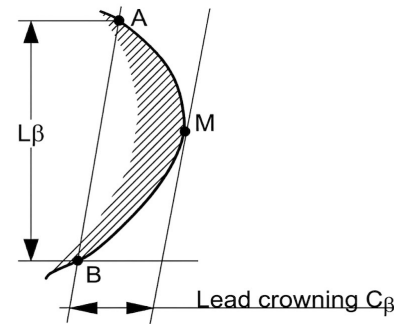
Figure 2.19: Schematic of microgeometry errors and their influence on gear performance [19]

Crowning

Because of deflections under load, or errors in the manufacturing of gears, the teeth may not be quite parallel, and edge loading may take place. To avoid end loading, the ends of the teeth are relieved (lead crowning) (Figure 2.20) [23]. The objective is to centralize the contact area. The lead crowning factor (C_β) is the profile's curvature relative to the effective direction.



(a) Lead crowning [24].



(b) Lead crowning factor (Adapted from [18]).

Figure 2.20: Modifications in microgeometry: Lead crowning

Manufacturing errors and tooth deflection (under loading) cause the actual gear mesh contact to deviate from the theoretical gear mesh contact. As a result, premature contact at tooth tips may occur, leading to noise, vibrations, and excessive wear [23]. To prevent mechanical shocks (gear interference), a profile crowning is applied, resulting in smoother meshing. The profile crowning factor (C_α) is the profile's curvature relative to the effective direction. (Figure 2.21).

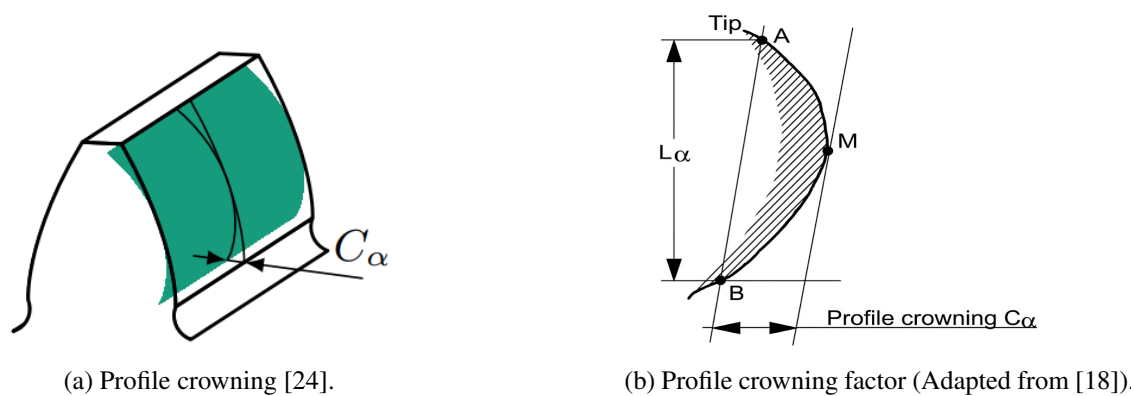


Figure 2.21: Modifications in the gear microgeometry: Profile crowning

To illustrate how tolerances evolve in relation to gear tooth quality, Table 2.5 presents the specific tolerance values (for third fix gear) for various criteria as a function of the overall gear quality.

Table 2.5: Example of Tolerances for Gear Quality

Criterion	Q8	Q7	Q6
$F_{H\alpha}$	± 12	± 9	± 6
$F_{H\beta}$	± 10	± 7	± 5
C_α	2–10	2–8	2–6
C_β	1–8	1–6	0–5
F_p	48	34	24
F_r	38	27	19

$\left(\frac{1}{\sqrt{2}}\right)$ $\left(\frac{1}{\sqrt{2}}\right)$

2.2.2 Roughness

Currently, the gears in standard gearboxes undergo the shaving process, with the exception of the differential ring gear, the output shaft and the reverse shaft. Due to the geometry of the parts and the heat treatment process, these gears undergo significant deformation and dispersion of deformations across the different teeth of the part, and require gear grinding instead. The forward gears are Grade 8 and the reverse gears are Grade 9 (ISO 1328 standard). In the Golden versions, all gears have been ground to achieve a quality of at least Grade 7. As presented previously, the surface friction coefficient of the gears affects power loss due to gear mesh.

As is well known, the machining process influences the final surface roughness. In Renault documents [18] related to gear cutting, the R parameter (Renard standard series) is used to characterize surface roughness. The document states that, for the shaving process, the parameter takes the value $R = 5$, whereas for grinding, it takes the value $R = 2$. While there is no direct relationship between this parameter and the average roughness (R_a) and mean roughness depth (R_z), statistically, at the gear level, the relationships in Equations (2.34) and (2.35) hold true. It is therefore to be expected that the different types of machining used in standard gearboxes and Golden versions will affect their efficiency.

$$R \approx 4 \cdot R_a \quad (2.34)$$

$$R \approx 0.64 \cdot R_z \quad (2.35)$$

The production of components and mechanisms for the automotive industry follows a strict compromise between economic viability, cycle time, and functionality requirements. Therefore, the standard JT4 gearbox was designed by the RENAULT Group to ensure a balance between functionality and the optimization of cost and manufacturing time.

Chapter 3

JT4 Gearbox

The JT4 transmission is manufactured exclusively at HORSE Aveiro and accounts for approximately 70% of the plant's revenue. These gearboxes are installed in the 1.0 (HR10), 1.2 (HR12), and 1.6 (HR16) gasoline engines found in Renault's Clio, Captur, and Symbioz models and Dacia's Sandero, Logan, Duster, Bigster, and Jogger models.

3.1 Gearbox Mechanism

The JT4 transmission is a synchromesh transaxle gearbox. The term "synchromesh" refers to the mechanism (Figure 3.1) that matches the speed of the gear to be engaged with the speed of the main shaft. Meanwhile, the term "transaxle" indicates that the assembly combines the transmission and the differential into a single compact housing.

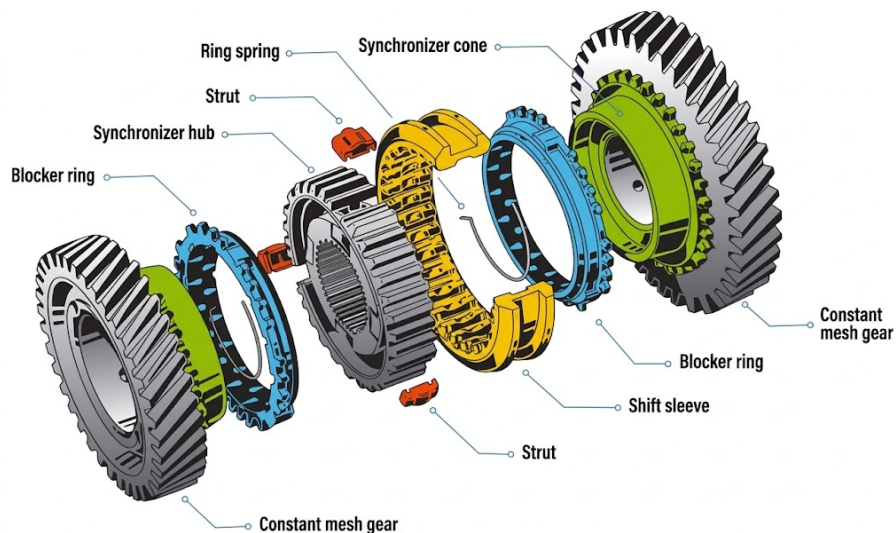


Figure 3.1: Synchromesh mechanism [25].

When a forward gear is selected, engine torque enters the transmission via the input shaft (primary shaft). The selected gear then transmits this torque directly to the output shaft (secondary

shaft), which transfers it to the differential ring gear (Figure 3.2). During this process, the direction of rotation undergoes two reversals: first, when passing from the primary shaft to the secondary shaft, and second, when passing from the secondary shaft to the ring gear. Due to this even number of reversals, the differential ring gear rotates in the same direction as the primary shaft, causing the vehicle to move forward.

Just as with forward gears, when the reverse gear is engaged, engine torque enters the transmission through the input shaft and is transmitted to the output shaft, causing an initial reversal of rotation. However, when reverse gear is selected, the output shaft transmits torque directly to the reverse shaft. This causes a second reversal of rotation, resulting in the reverse shaft rotating in the same direction as the input shaft. Finally, torque is transmitted from the reverse shaft to the differential ring gear, causing a third reversal. These three reversals ensure that the differential ring gear rotates in the opposite direction to the primary shaft, thus reversing the movement of the wheels and causing the vehicle to move in reverse.

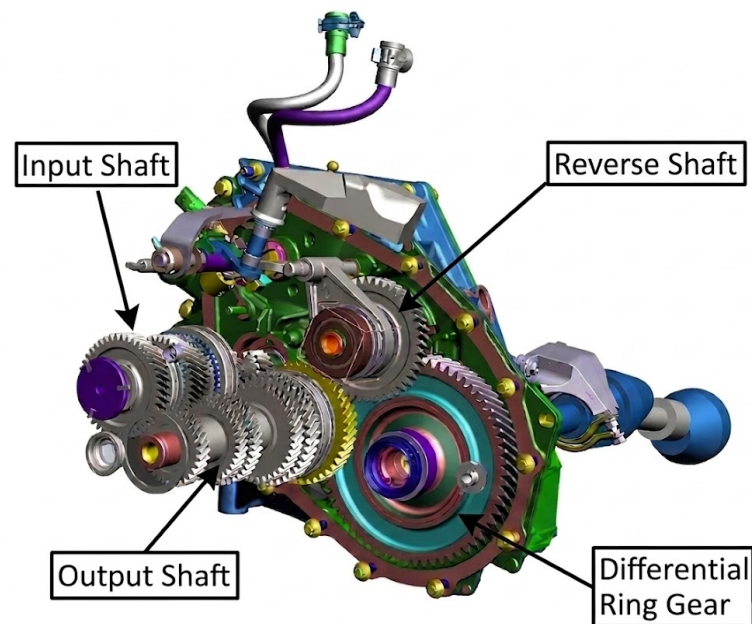


Figure 3.2: Internal mechanism of the JT4 gearbox.

On the input shaft, the first and second gears are machined onto the shaft (fixed gears). The third through sixth gears are idle gears that are assembled on the input shaft, as shown in Figure 3.3. The fixed gears are physically connected to the shaft, while the idle gears rotate freely relative to the shaft until a gear is engaged. At this point, the synchromesh mechanism matches the speed of the idle gear to the speed of the shaft. In this state, the torque from the idle gear is transmitted to the shaft.

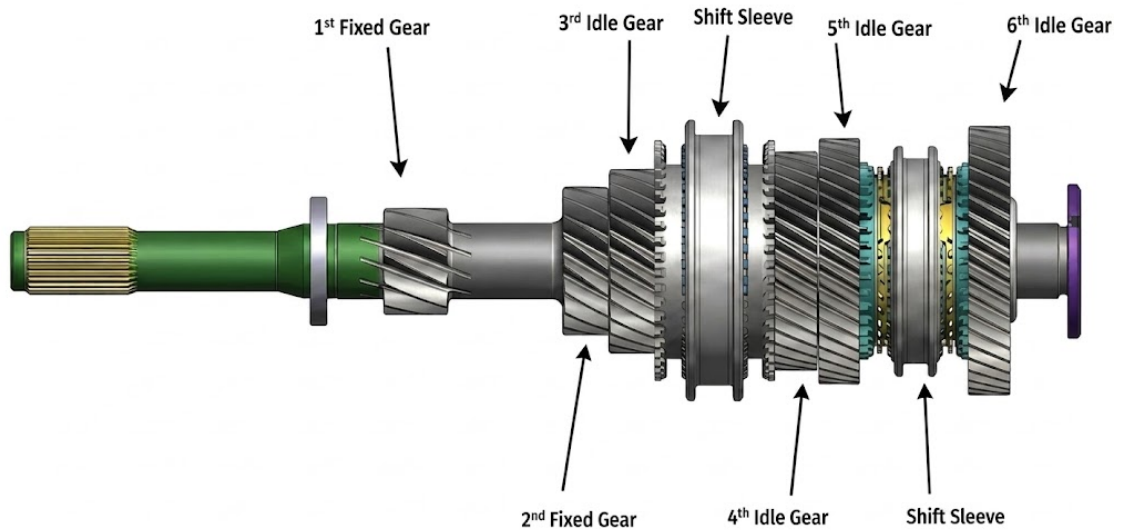


Figure 3.3: Input Shaft

In the case of the output shaft, the first and second gears are idle gears that are mounted on the shaft. The third through sixth gears are fixed gears, as illustrated in Figure 3.4. With respect to the torque transmission system, this is analogous to the one described in the previous paragraph.

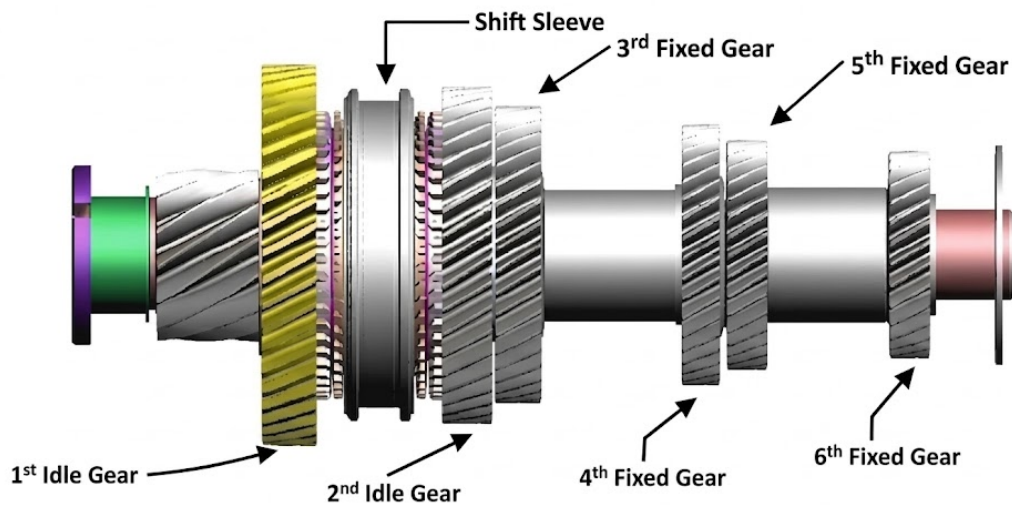


Figure 3.4: Output Shaft

3.2 Gearbox versions

In order to study the influence of gear tooth quality on gearbox efficiency, two versions of the gearbox (Golden versions) were designed with superior gear tooth quality. This section will address the different variants, starting with the series production gearbox and moving on to the

Golden versions. The study of the Golden versions aims to isolate and analyze gear tooth quality and evaluate its direct correlation with the overall efficiency of the gearbox.

3.2.1 Standard Version

Production of the JT4 gearbox began in 2019, and it is installed in various Renault and Dacia models. Currently, around 1,700 gearboxes are produced every day, 24/7. The standard version is designed to meet general requirements for durability, reliability and production costs. The computational results of the standard version will assess the efficiency of the gearboxes based on the quality of the current gear teeth, and will serve as a benchmark for comparison with the golden versions. This section will describe the manufacturing processes for idle gears, fixed gears, and the primary shaft—components that will undergo changes in the Golden versions.

The current manufacturing process for idle gears begins with turning, followed by hobbing, chamfering, welding of the synchronizing ring, heat treatment, and cylindrical grinding. The process is completed with a roll testing (shock control). The sequence of these processes is illustrated in Figure 3.5.



Figure 3.5: Manufacturing line of idle gear in standard version

To manufacture fixed gears, they undergo a process that includes turning, broaching, hobbing, and chamfering, followed by heat treatment and a roll testing (shock control) (Figure 3.6).



Figure 3.6: Manufacturing line of fixed gear in standard version

The primary shaft body includes the first and second gear pinions. This shaft undergoes the following machining operations: turning, hobbing, chamfering, rolling, heat treatment, straightening, cylindrical grinding and roll testing (shock control) (Figure 3.7).



Figure 3.7: Manufacturing line of primary shaft in standard version

For the Standard version, optical roughness measurements were performed at INEGI, yielding an R_a of $1.19 \mu\text{m}$ and an R_z of $6.76 \mu\text{m}$, which were the values implemented in the KISSsoft software.

3.2.2 Golden Version

In the Golden versions, the gears were ground to achieve a higher tooth quality. To achieve this, unlike mass production, the manufacturing of fixed gears in the Golden version includes an additional grinding step (Figure 3.8). To achieve this superior tooth quality, the hobbing/chamfering and heat treatment processes have been optimized, ensuring the desired final quality of the tooth.



Figure 3.8: Manufacturing line of fixed gear in Golden version

Since the body of the primary shaft contains the first and second gear sets, the manufacturing process for this shaft had to be modified. For this reason, gear grinding was introduced after cylindrical grinding and before shock control (Figure 3.9). Similar to the other gears, the machining of the primary shaft requires adjustments to the hobbing and chamfering processes, as well as to the heat treatment, in order to ensure compliance with the new specifications.



Figure 3.9: Manufacturing line of primary shaft in Golden version

Unlike in series production, a gear grinding step is added when machining the idle gears (Figure 3.10). Furthermore, hobbing specifications are modified to be compatible with the gear grinding process and achieve superior tooth quality.



Figure 3.10: Manufacturing line of idle gear in Golden version

Since the gears (both fixed and idle) in the Golden version are ground after heat treatment, they exhibit a different surface finishing compared to mass-produced parts. The latter retain a darker appearance, which is a direct result of the heat treatment process. Figure 3.12 shows the input shafts of the standard and golden gearboxes, while Figure 3.11 displays their respective 5th-gear idle gears. Upon analysis, the distinct visual appearance of the toothing on both the shafts and the gears is clearly noticeable.

For the Golden version, the optical roughness measurements revealed an R_a of $0.30 \mu\text{m}$ and an R_z of $1.835 \mu\text{m}$, and these specific values were subsequently added to the KISSsoft model.



(a) Idle gear of Standard gearbox



(b) Idle gear of Golden gearbox.

Figure 3.11: 5th idle gear of JT4 gearbox.

(a) Input Shaft of Standard gearbox



(b) Input Shaft of Golden gearbox

Figure 3.12: Input Shaft of JT4 gearbox

Due to the geometry of the idle gears for the third and fourth gears, the synchronizer ring does not allow the gear grinding tool to pass, as shown in Figure 3.13. Consequently, in both Golden versions, the manufacturing process for the third and fourth idle gears will differ from that of the other gears. Furthermore, depending on the timing of the synchronizer ring welding and gear tooth grinding relative to the heat treatment, the Golden version can be subdivided into two versions.

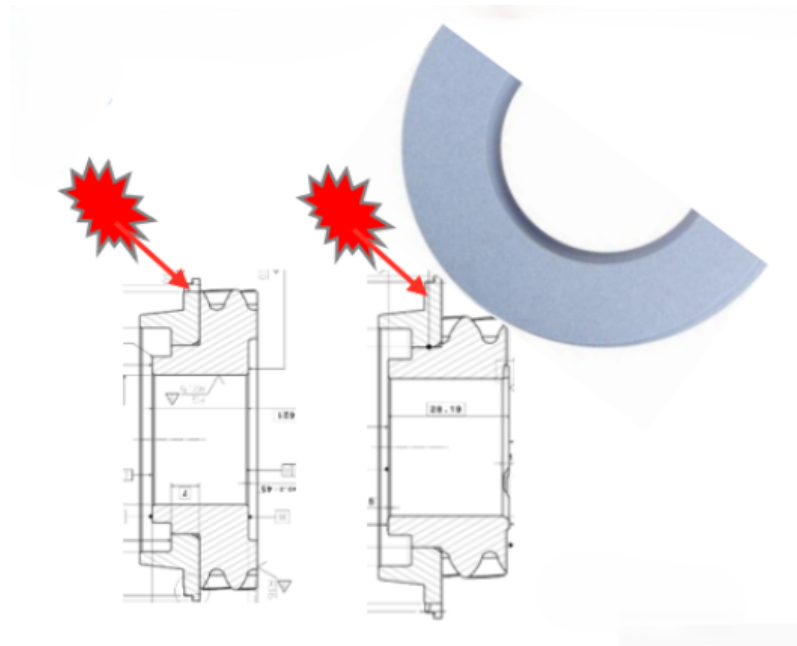


Figure 3.13: Grinding tool passage obstructed by the synchronizer ring.

Golden 1st Version

As previously mentioned, because the grinding tool cannot pass through the 3rd and 4th idler gears, they are ground prior to the welding of the synchronizer ring. In the first variant of the Golden version, both processes are performed in the soft state—that is, before heat treatment (Figure 3.14). Since this treatment introduces distortions that affect tooth quality, these gears are ground in the green state to achieve a quality grade of 6. This ensures that, after heat treatment, they maintain a quality grade of at least 7. It is important to note that the hobbing process was modified to achieve the desired quality.



Figure 3.14: Manufacturing line of idle gear of 1st Golden version

Since the 3rd and 4th idler gears are ground before heat treatment—unlike the rest—they can be visually distinguished. Due to this reversal in the process, their surface retains a noticeably darker hue compared to the other gears. In Figure 3.15, it is possible to see the input shaft (right side) and the output shaft (left side). Upon analyzing it, it can be observed that the idle gear of the third and fourth gears present on the input shaft has a different coloration from the rest, despite having the same gear tooth quality.

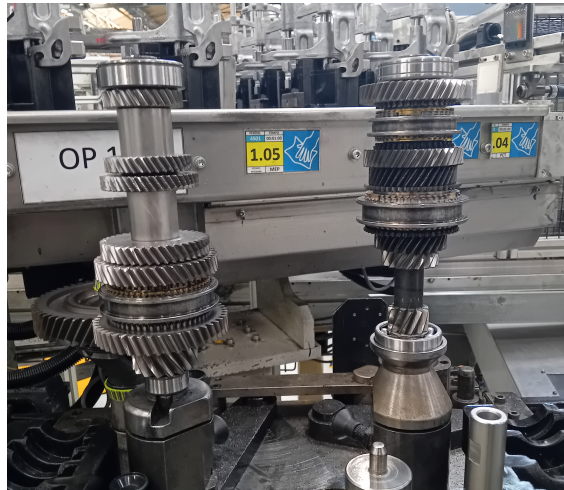


Figure 3.15: Output and Input shaft from 1st Golden version

Golden 2nd Version

Similar to the first version of the Golden, the 3rd and 4th speed idle gears will be ground before the synchronizer ring is welded. The difference in 2nd version is that these processes take place after the heat treatment (Figure 3.16). Since grinding is performed at a stage where the deformations from the heat treatment have already occurred, the required quality during grinding will be the same as the desired final quality (Quality 7). In the 1st version, it was necessary to grind to a higher quality to compensate for the deformations caused by the subsequent heat treatment.

Since the welding will take place after heat treatment, a hard turning process will be introduced to remove the treated layer in the specific area where the synchronizer ring will be welded to the gear.



Figure 3.16: Manufacturing line of idle gear of 2nd Golden version

Since grinding occurs after heat treatment for all gears, they now have a uniform visual appearance, unlike the first version. This can be observed in Figure 3.17.



Figure 3.17: Gears of 2nd Golden version

The different gear quality grades, as shown in Table 2.5, will impact the gear's geometric tolerances. Additionally, the use of different manufacturing processes will result in distinct surface conditions with varying levels of roughness. To consolidate the information presented in this section, Table 3.1 provides a summary of the gear tooth quality for both the series and golden versions.

Table 3.1: Tooth Quality of the Serie and Golden Gearboxes

Serie gearbox	Idle gear	Fix gear/shaft
1 st	Q8	Q8
2 nd	Q8	Q8
3 rd	Q8	Q8
4 th	Q8	Q8
5 th	Q8	Q8
6 th	Q8	Q8
Reverse	Q8	
Final drive		
Ring gear	Q7	
Output shaft	Q7	
Reverse shaft	Q7	

⇒

Golden gearbox	Idle gear	Fix gear/shaft
1 st	Q7	Q7
2 nd	Q7	Q7
3 rd	Q7	Q7
4 th	Q7	Q7
5 th	Q7	Q7
6 th	Q7	Q7
Reverse	Q7	
Final drive		
Ring gear	Q7	
Output shaft	Q7	
Reverse shaft	Q7	

Chapter 4

Numerical model

The advantage of experimental testing is that it provides direct measurements of power loss and, consequently, transmission efficiency. However, these methods make it difficult to isolate the individual contributions of the gears, bearings, and seals (Figure 2.1). Furthermore, experimental approaches are often expensive and require specialized design tools.

Since the objective of this thesis is to study the efficiency of the transmission—more specifically, the effect of gear tooth quality on power loss—it is essential to isolate the losses caused by the gears. With this in mind, a model of the JT4 transmission was created in KISSsoft, specifically using the software’s KISSdesign package.

While working on the thesis, a webinar hosted by Gleason, the owner of KISSsoft, was attended, where the basic concepts of modeling a gearbox in KISSdesign were explained. Figure 4.1 shows a diagram, presented by Gleason during the webinar, of the steps to follow when modeling a gearbox. Since KISSdesign is a recent successor to KISSsys, there is currently limited information available on how to define and build systems in this software. Therefore, this section provides a brief explanation of the steps followed to construct the gearbox model.

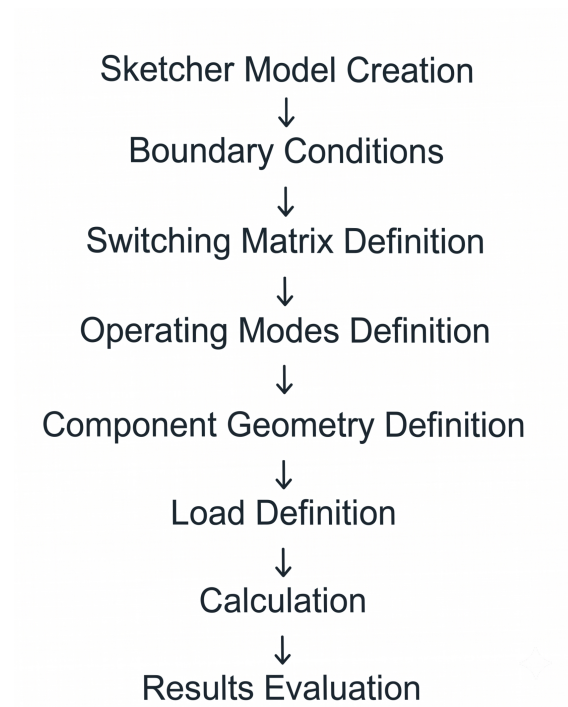


Figure 4.1: Gleason's recommended procedure for designing a gearbox

4.1 Sketcher Model Creation

Following Gleason's recommendations, the gearbox modeling process began with the definition of the sketcher. Figure 4.2 shows the diagram drawn in KISSdesign.

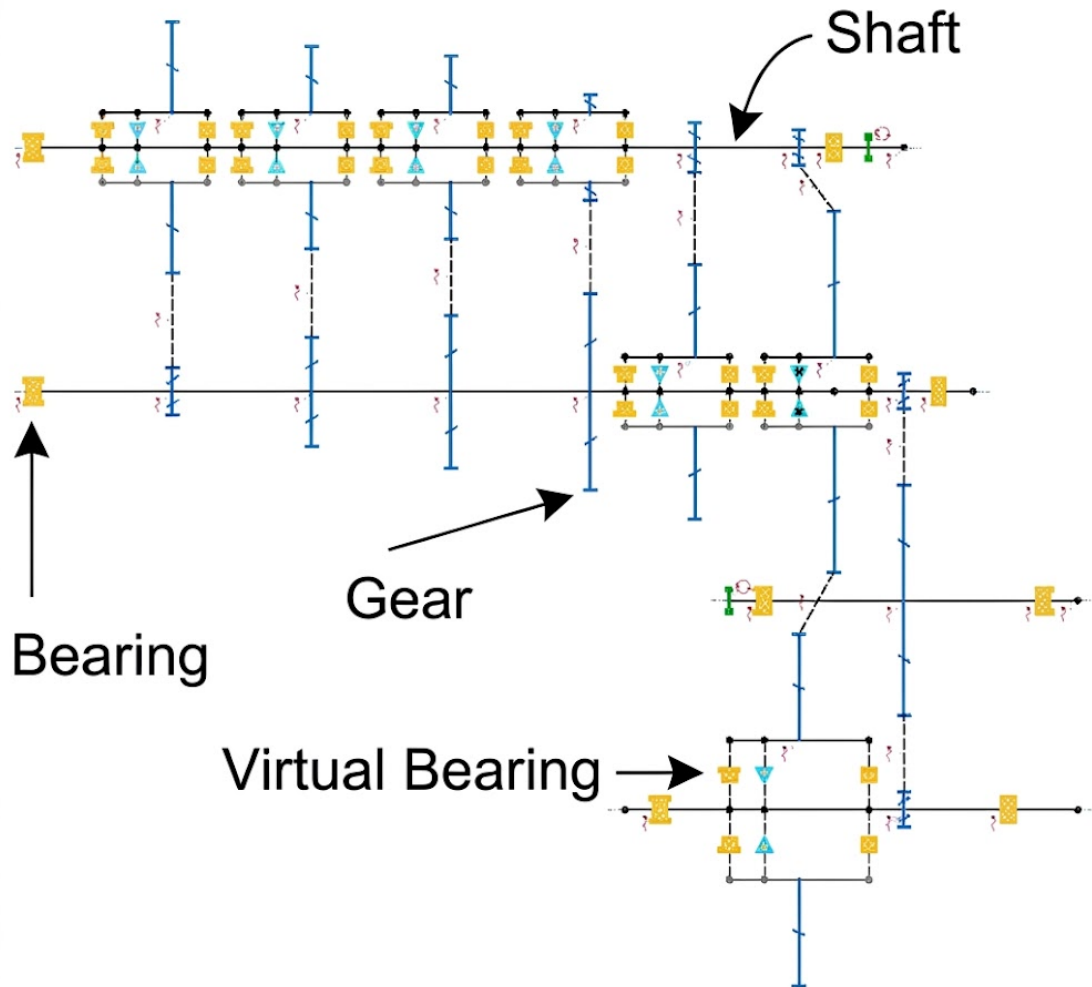


Figure 4.2: Sketch of the gearbox model in KISSsoft

In Figure 4.2, the upper shaft represents the input shaft of the gearbox. On this shaft, viewed from right to left, we can see the power input boundary condition (represented by the red circular arrows) and the corresponding coupling, represented by the green element. As illustrated in the diagram in Figure 4.1, these elements will be explained in the second step of the process. The yellow element shown next represents a ball bearing. It should be noted that, as HORSE uses non-standard bearings, it was necessary to manually insert all the bearings of the gearbox into KISSsoft because, naturally, KISSsoft does not contain the specific bearings present in the gearbox.

To the left of the bearing, we can see the fixed gears for first and second gear. To their left, we can see the idle gears for third, fourth, fifth, and sixth gear. As mentioned previously, the idle gears are not "coupled" to the shaft until the respective gear is engaged. To simulate this feature in KISSsoft, a hollow shaft was designed around the input shaft, with the gear fixed to this hollow shaft. The hollow shaft, in turn, is supported by two bearings that allow for a rotational degree of freedom, and by a synchronizer element (represented by the light blue triangles). When open,

these elements allow the gear a degree of freedom relative to the input shaft (gear not engaged), and when closed, they couple the gear to the input shaft (gear engaged). On the left end of the input shaft, a ball bearing is shown.

The second shaft from the top represents the output shaft. From right to left, we can see a roller bearing and the output shaft's own gear, which allows power transmission to the differential ring gear. Next, we can see the idle gears for first and second gear. These gears use the same idler gear system as the input shaft. Next to them, we can see the fixed gears for third, fourth, fifth, and sixth gear, and a ball bearing at the end.

The shaft exactly below, the third from the top, represents the differential (simplified, with only the differential ring gear), the two tapered roller bearings, and on the left end, the output boundary condition and its corresponding coupling.

The shaft located at the bottom of Figure 4.2 represents the reverse shaft of the gearbox. From right to left, we can see the ball bearing and the gear machined onto the reverse shaft, which allows power transfer from this shaft to the differential. Next to it, we can see the reverse gear with its respective gear engagement system. On the left end, we can see a ball bearing. It should be noted that the reverse gear is meshed with the first gear idle gear.

After all the shafts were modeled, the gear meshing paths (represented by the dashed line) were drawn, allowing the software to identify which gears are meshed together.

Since the objective of this model is to calculate the efficiency of the gearbox, after completing the gearbox sketch, the elements that KISSsoft should consider for the efficiency calculation were identified. All gears were selected (to account for churning losses), along with all gear meshes (to account for load-dependent losses), the shaft bearings (naturally, the bearings of the gear engagement system were not considered, as they serve merely as a representation of the phenomenon and do not exist in the real gearbox), and the seals located on the primary shaft and the differential. In the sketch in Figure 4.2, these losses are represented by red zigzag arrows.

4.2 Boundary Conditions

As mentioned in the previous section, the model includes boundary conditions and couplings. These elements allow us to identify where power enters and exits the system. In simple terms, they define the starting and ending points of the power path. Figure 4.3 shows the assignment of the couplings to their respective boundary conditions.

	Boundary condition	Element
1	 boundary1	 coupling1
2	 boundary2	 coupling2

Figure 4.3: Definition of boundary conditions and input and output couplings

It is also possible to specify which variables we want to be inputs to our system. The system inputs can be speed, torque, power, or combinations of the three. Since the boundary condition and coupling associated with the input shaft are “boundary1” and “coupling1” respectively, the variables speed and torque have been assigned. This selection can be seen in Figure 4.4.

	Name	 boundary1	 boundary2
1	Setup 1	T^n	O

Figure 4.4: Selection of the model’s input variables

4.3 Switching Matrix Definition

To make it easier to select gear shifts—and avoid having to manually open and close the switchable elements - KISSsoft allows you to create a switching matrix. As shown in Figure 4.5, the switching matrix allows you to associate each switchable element with a gear shift

	Name	 switchable2	 switchable3	 switchable4	 switchable5	 switchable6	 switchable7	 switchable1
1	1st Gear	0	0	0	0	0	0	X
2	2nd Gear	X	0	0	0	0	0	0
3	3rd Gear	0	X	0	0	0	0	0
4	4th Gear	0	0	X	0	0	0	0
5	5th Gear	0	0	0	X	0	0	0
6	6th Gear	0	0	0	0	X	0	0
7	R Gear	0	0	0	0	0	X	0

Figure 4.5: Switching matrix

4.4 Operating Modes Definition

The operating modes allow you to create a selection box in KISSsoft (Figure 4.6), enabling the selection of the desired gear in an easy and intuitive way. Once the switching matrix has been defined, the switchable elements will open and close automatically, engaging the selected gear.



Figure 4.6: Operating modes

Defining these operating modes (Figure 4.7) involves assigning a name to appear in the drop-down menu in Figure 4.6, specifying the type of variables to use as inputs (labeled “Setup 1,” as shown in Figure 4.4), and selecting the gear to be meshed. This step also indicates whether the kinematic behavior is consistent or inconsistent

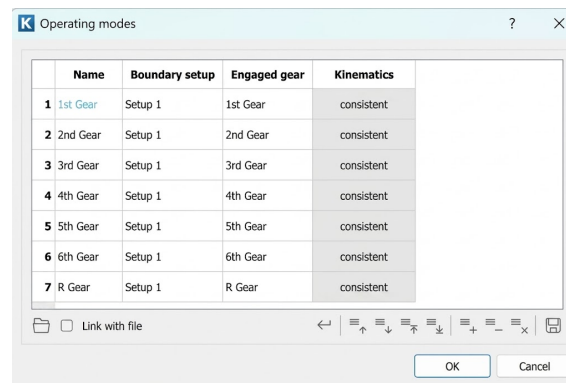


Figure 4.7: Definition of operating modes

4.5 Component Geometry Definition

Once the gearbox base has been modeled, the next step is to define its geometry. Initially, when developing the sketch model, neither the model’s dimensions nor the gear geometries are fully defined. Therefore, the first step was to define the geometry of the gearbox shafts, as well as the positions of the bearings, gears, and seals. The 3D models that served as the basis for constructing the primary shaft and output shaft models were presented in Figures 3.3 and 3.4 respectively. The primary shaft model can be seen in Figure 4.8.

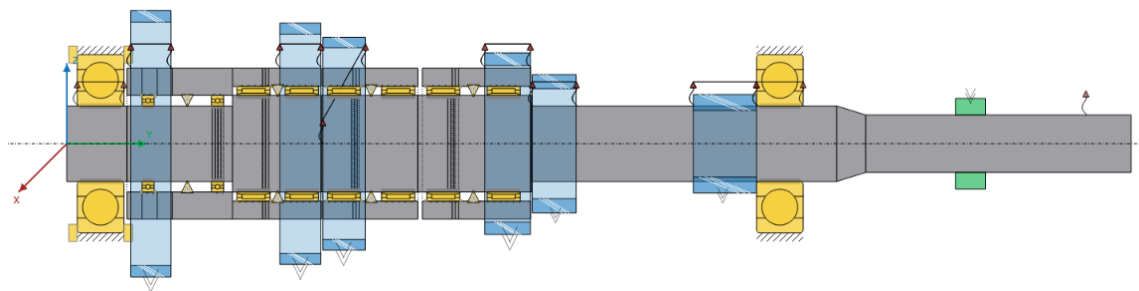


Figure 4.8: Input shaft model

The KISSsoft model of the output shaft can be seen in Figure 4.9. This figure also shows the oil level. The JT4 gearbox carries approximately 1.25 L of oil. Since oil must be defined as a height in KISSsoft, an oil height corresponding to this volume of oil was inserted.

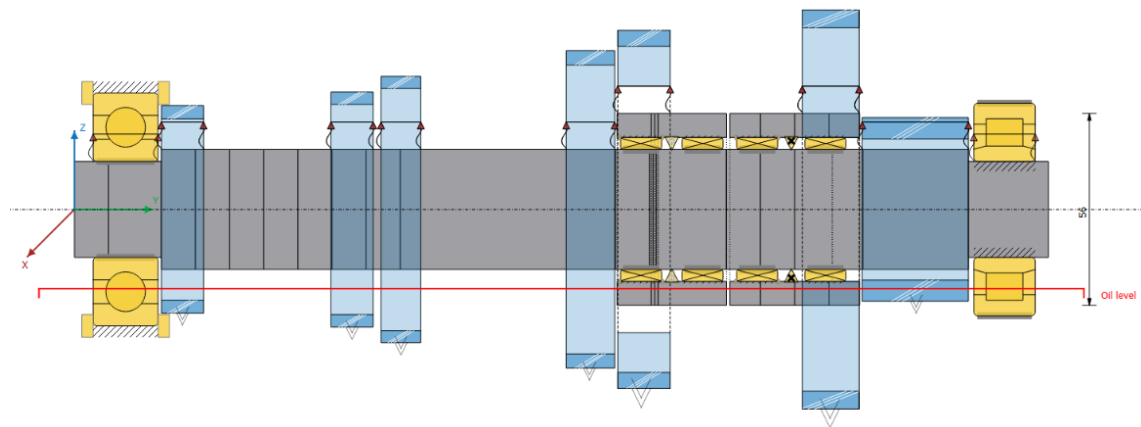
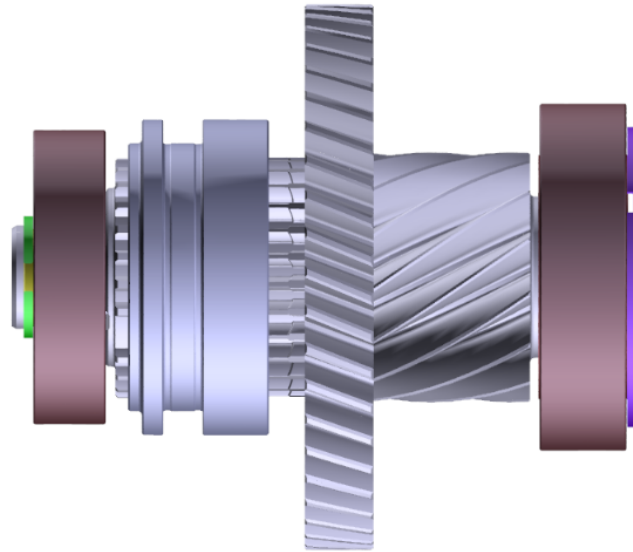
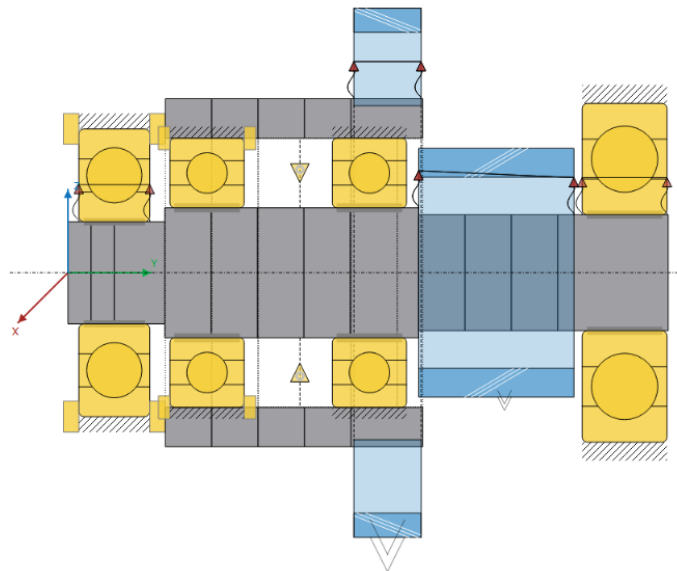


Figure 4.9: Output shaft model

In Figure 4.10a, you can see the 3D model used to define the shaft geometry in KISSsoft. In Figure 4.10b, you can see the reverse shaft model in KISSsoft.



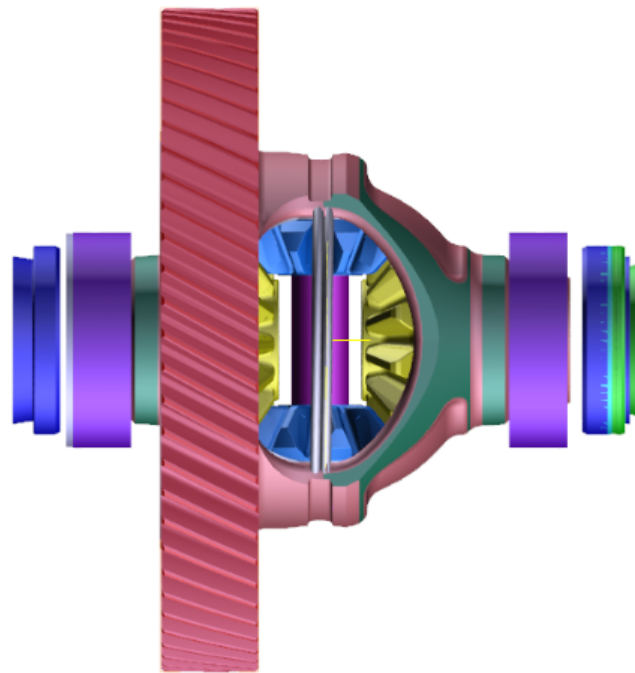
(a) 3D model of Reverse Shaft



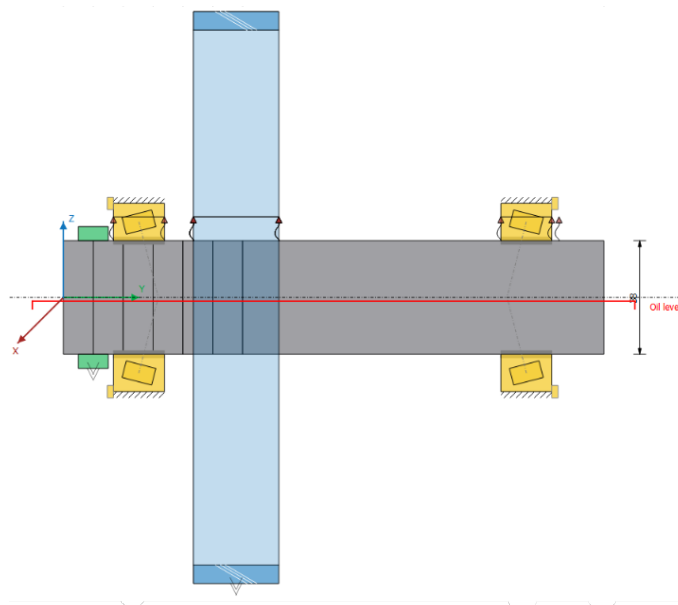
(b) KISSsoft model of Reverse Shaft

Figure 4.10: Reverse Shaft

Lastly, the differential shaft was defined. To simplify the calculation, only the differential ring gear was modeled. The 3D model that served as the basis for the modeling in KISSsoft is shown in Figure 4.11a. The corresponding model in KISSsoft is shown in Figure 4.11b.



(a) 3D model of differential



(b) KISSsoft model of differential

Figure 4.11: Differential

Once the shaft geometry and the locations of the bearings, gears, and seals were established, the next step was to define the gear geometry. This phase was the most time-consuming due to the large number of gears and the extensive set of variables that needed to be specified (macro-geometry, tooth modifications, tooth profile shift, material properties, and tolerances). Due to HORSE's confidentiality policy, the specific geometric details cannot be included, but an effort will be made to include the most relevant information in this report.

Table 4.1 presents the number of teeth on the fixed and idler gears for each transmission speed.

Table 4.1: Number of teeth of fixed and idler gears

GEAR	IDLE GEAR	FIX GEAR
1	45	11
2	41	17
3	26	41
4	37	41
5	41	35
6	43	29

Regarding the reverse gear ratio between the output shaft and the reverse shaft, the tooth count ratio is 44:45. The 44 teeth correspond to the reverse idle gear, while the 45 teeth correspond to the first gear idle.

The power transmission from the output shaft to the differential and from the reverse shaft to the differential has the same gear ratio of 15x58. The 15 teeth refer to the pinion on the output shaft and reverse shaft, and the 58 teeth refer to the number of teeth on the differential ring gear.

With the gear ratio i given by equation 4.1,

$$i = \frac{n_{\text{input}}}{n_{\text{output}}} \quad (4.1)$$

It is possible to define the overall drive ratio for each gear in the transmission. Note that these values reflect the total ratio from the gearbox input to the final output, encompassing the differential ratio. The overall gear ratios are detailed in Table 4.2.

Table 4.2: Gear ratio of each gear

GEAR	GEAR RATIO
1	15.818
2	9.325
3	6.097
4	4.285
5	3.301
6	2.608
Reverse	15.466

The following section provides a simplified explanation of the procedures carried out for each gear pair. To illustrate this process, the first idle gear and reverse gear pair will be used as an example.

The first step in defining a gear pair is to define the basic data, which includes the macrogeometry of the gears, their material, and lubrication method and lubricant. This definition is shown in Figure 4.12.

Geometry			
Tooth geometry	symmetric		
Normal module	m_n	20.0000	mm
Normal pressure angle	α_n	20.0000	°
Gear 1	helix left hand		
Helix angle at reference circle	β	20.0000	°
Center distance	a	110.6001	mm
Material and lubrication			
Gear 1	Case hardening steel	Own Input	
Gear 2	Case hardening steel	Own Input	
Lubrication	Oil bath lubrication	Own Input	

Figure 4.12: Basic Data

Although the gear tooth quality can be entered into the Basic Data section, Gleason/KISSsoft has clarified that the software's efficiency calculation model does not account for this parameter. This highlights a significant scientific void and a scarcity of studies developed in this specific field, thus necessitating the execution of experimental trials to achieve validated results. The parameter related to tooth quality that is included in the efficiency calculation is the roughness of the tooth flank.

Regarding the lubricant, the properties of the oil used in the JT4 gearbox - which in this case is Tranself NFX SAE 75W from ELF/TotalEnergies - were entered in this section (referenced in Figure 4.12 as "own input"). The properties entered into KISSsoft can be seen in Table 4.3. It should be noted that a lubricant factor (X_L) of 0.8 was applied.

Table 4.3: Technical Specifications for Grade SAE 75W [26]

Property	Method	Unit	Grade SAE 75W
Density at 15°C	ASTM D 1298	-	847
Kinematic viscosity at 40°C	ASTM D445	mm ² /s	31
Kinematic viscosity at 100°C	ASTM 445	mm ² /s	6.5
Flash point	-	°C	202
Viscosity index	ASTM D2270	-	168
Pour point	ASTM D97	°C	-45

The gear tooth reference profile is also defined. As shown in Figure 4.13, parameters such as the root diameter, root radius, and tip diameter, among others, are inputted.

Machining step Gear 1

Pre- and final machining: Final machining (without pre-machining)

Final machining Gear 1

Tool type: Reference profile gear

Input: Diameter

Select reference profile: Own Input

Designation: 45

Root diameter d_f : mm

Root radius p_{fp} : mm

Tip diameter d_a : mm

Tip radius p_{tp} : mm

Root form diameter d_{fr} : mm

Protuberance angle α_{pP} : 0.0000 °

Tip form diameter d_{ta} : mm

Profile angle of the chamfer involute α_{ip} : 60.0000 °

☒ Topping tool

Machining step Gear 2

Pre- and final machining: Final machining (without pre-machining)

Final machining Gear 2

Tool type: Reference profile gear

Input: Diameter

Select reference profile: Own Input

Designation: 44

Root diameter d_f : mm

Root radius p_{fp} : mm

Tip diameter d_a : mm

Tip radius p_{tp} : mm

Root form diameter d_{fr} : mm

Protuberance angle α_{pP} : 0.0000 °

Tip form diameter d_{ta} : mm

Profile angle of the chamfer involute α_{ip} : 0.0000 °

☒ Topping tool

Figure 4.13: Definition of the Gear Tooth Profile

Another parameter that was added to KISSsoft is gear tooth modifications. The modifications made to some of the gears in the transmission include profile crowning, flank line crowning, helix angle modification, and pressure angle modification. To illustrate the introduction of these modifications to the tooth profile, Figure 4.14 shows the modifications made to the reverse gear and the first idle gear (the gear pair with the most tooth profile modifications).

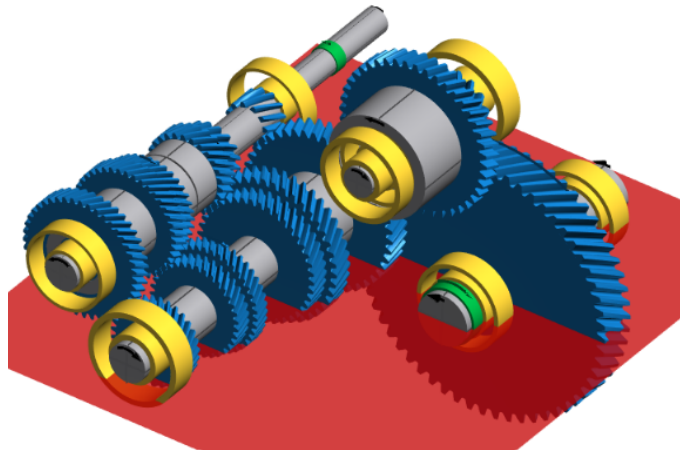
Additional modifications

Variant for calculation: No variant defined

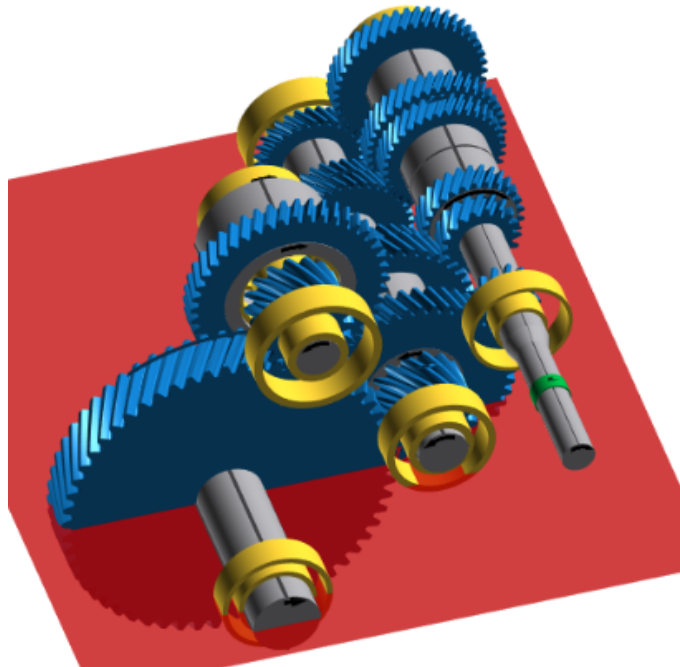
	Gear	Flank	Modification type	Value [μm]	Factor 1	Factor 2	Status
1	Gear 1	both	Profile crowning, diameter-centered				Active
2	Gear 1	both	Flank line crowning				Active
3	Gear 1	right	Helix angle modification, conical				Active
4	Gear 2	both	Profile crowning, diameter-centered				Active
5	Gear 2	both	Flank line crowning				Active
6	Gear 2	right	Pressure angle modification (value)				Active
7	Gear 2	left	Pressure angle modification (value)				Active
8	Gear 2	right	Helix angle modification, conical				Active
9	Gear 2	left	Helix angle modification, conical				Active

Figure 4.14: Gear tooth modifications

Defining the gear center distances and entering the angles between the shafts fully defines the geometry. After setting the desired oil level (approximately at the center of the differential), the three-dimensional model of the gearbox shown in Figure 4.15 can be obtained.



(a) View 1



(b) View 2

Figure 4.15: Model of JT4 gearbox in KISSsoft

4.6 Load Definition

Once the model is finalized, it is possible to define the input power, through the input of speed and torque. As Figure 4.16 illustrates the definition of the input power, this task is performed in the boundary conditions section.

	Boundary condition	Element	Torque [Nm]	Speed [1/min]	Power [kW]	Sense of rotation	Direction of power flow	Reference	Speed ratio
1	 boundary1	 coupling1	-80.000000	-800.000000	6.702064	counterclockwise	driven (input)	 boundary1	1.000000
2	 boundary2	 coupling2	342.774775	-186.711522	-6.702064	counterclockwise	driving (output)	 boundary2	1.000000

Figure 4.16: Load Definition.

4.7 Calculation and Results Evaluation

After the definition of the model and the load, the software is able to perform the calculations and consequently perform an analysis of the obtained results. During the development of this thesis, a software update was released that resolved calculation errors regarding churning power losses and introduced the capability to generate efficiency and power loss maps. This section presents the efficiency maps and power loss graphs generated by the KISSsoft model, along with their respective analysis.

4.7.1 Efficiency maps

The efficiency maps (Figure 4.17) relate the gearbox's overall efficiency with the rotational speed and input torque, allowing the identification of the optimal operating zones and the conditions that lead to lower mechanical efficiency. Equation 4.2 illustrates how the efficiency was calculated.

$$\eta = \left(\frac{P_{\text{out}}}{P_{\text{IN}}} \right) \times 100 = \left(1 - \frac{P_V}{P_{\text{in}}} \right) \times 100 \quad (4.2)$$

By analyzing the Figure 4.17a, it can be concluded that the reverse gear exhibits the lowest efficiency across the entire analyzed spectrum, with a minimum of 91.84 % at 600 rpm and 220 Nm and a maximum of 95.13% at 6000 rpm and 61.67 Nm. Similarly, analyzing the first gear (Figure 4.17b), it is verified that the lowest efficiency (93.50%) occurs under the speed of 600 rpm and 220 Nm and the highest efficiency (95.86%) occurs at 6000 rpm at 61.67 Nm. In these gears of higher gear ratio, the optimal efficiency point occurs in scenarios of high speed and low input torque.

As higher gears are engaged, the system's efficiency increases significantly, and the point of maximum efficiency shifts towards the maximum load region. Analyzing the Figures 4.17d and 4.17e, we can conclude that the third and fourth gears exhibit the highest efficiencies of the analyzed spectrum, respectively of 97.10 % and 97.08 %. In both cases, the maximum efficiencies occur at the highest speed and torque analyzed (6000 rpm and 220 Nm).

For the fifth gear (Figure 4.17f), the maximum efficiency (97.07%) no longer occurs at the highest speed - as seen in the intermediate gears - but rather at 4920 rpm and 220 Nm. Furthermore, the lowest efficiency (95.57%) is observed at the maximum speed and minimum torque. In the case of the sixth gear, Figure 4.17g, this behavior is more evident, since the maximum efficiency (96.68%) occurs at 3480 rpm and 220 Nm. As occurred with the fifth gear, the lowest calculated efficiency (96.08%) occurred at the highest speed and lowest torque.

The contrasting efficiency behavior among the gears reveals that the system is governed by the trade-off between load-dependent and load-independent losses. To understand the underlying physical phenomena across the different operating regimes, the system's efficiency and power losses will be presented and analyzed graphically.

boundary 1 Torque [Nm]	boundary1 Speed [1/min]					
	600.00	1680.00	2760.00	3840.00	4920.00	6000.00
30.00	93.59	94.10	94.44	94.62	94.74	94.81
61.67	93.39	94.09	94.54	94.81	94.99	95.13
93.33	93.03	93.83	94.35	94.66	94.89	95.05
125.00	92.69	93.58	94.15	94.49	94.74	94.92
156.67	92.38	93.34	93.95	94.32	94.58	94.78
188.33	92.10	93.11	93.76	94.15	94.43	94.64
220.00	91.84	92.91	93.58	94.00	94.29	94.51

(a) Efficiency Map for the R Gear

boundary 1 Torque [Nm]	boundary1 Speed [1/min]					
	600.00	1680.00	2760.00	3840.00	4920.00	6000.00
30.00	94.60	94.97	95.21	95.34	95.41	95.45
61.67	94.58	95.10	95.44	95.64	95.77	95.86
93.33	94.34	94.95	95.35	95.58	95.74	95.85
125.00	94.11	94.78	95.22	95.47	95.65	95.78
156.67	93.88	94.62	95.08	95.35	95.54	95.68
188.33	93.68	94.46	94.96	95.24	95.44	95.59
220.00	93.50	94.32	94.83	95.13	95.34	95.49

(b) Efficiency Map for 1st Gear

boundary 1 Torque [Nm]	boundary1 Speed [1/min]					
	600.00	1680.00	2760.00	3840.00	4920.00	6000.00
30.00	95.02	95.36	95.48	95.51	95.50	95.47
61.67	95.57	96.05	96.27	96.38	96.43	96.47
93.33	95.55	96.12	96.39	96.53	96.61	96.67
125.00	95.45	96.08	96.39	96.55	96.66	96.73
156.67	95.35	96.02	96.35	96.53	96.65	96.73
188.33	95.24	95.95	96.30	96.49	96.62	96.71
220.00	95.13	95.88	96.25	96.45	96.58	96.68

(c) Efficiency Map for 2nd Gear

boundary 1 Torque [Nm]	boundary1 Speed [1/min]					
	600.00	1680.00	2760.00	3840.00	4920.00	6000.00
60.00	95.77	96.20	96.33	96.36	96.35	96.32
86.67	95.92	96.44	96.62	96.70	96.73	96.73
113.33	95.95	96.52	96.74	96.84	96.89	96.92
140.00	95.92	96.55	96.79	96.90	96.97	97.01
166.67	95.88	96.54	96.80	96.93	97.01	97.06
193.33	95.83	96.53	96.80	96.94	97.03	97.09
220.00	95.78	96.50	96.79	96.94	97.04	97.10

(d) Efficiency Map for 3rd Gear

boundary 1 Torque [Nm]	boundary1 Speed [1/min]					
	600.00	1680.00	2760.00	3840.00	4920.00	6000.00
60.00	95.52	95.84	95.88	95.82	95.72	95.59
86.67	95.90	96.32	96.42	96.42	96.39	96.32
113.33	96.01	96.50	96.64	96.69	96.68	96.65
140.00	96.04	96.58	96.76	96.83	96.85	96.84
166.67	96.04	96.62	96.82	96.91	96.95	96.96
193.33	96.01	96.63	96.85	96.96	97.01	97.03
220.00	95.98	96.63	96.87	96.98	97.05	97.08

(e) Efficiency Map for 4th Gear

boundary 1 Torque [Nm]	boundary1 Speed [1/min]					
	600.00	1680.00	2760.00	3840.00	4920.00	6000.00
80.00	95.79	96.06	96.04	95.94	95.78	95.57
103.33	96.06	96.40	96.45	96.39	96.29	96.15
126.67	96.19	96.59	96.68	96.66	96.59	96.49
150.00	96.25	96.69	96.81	96.82	96.78	96.71
173.33	96.28	96.76	96.90	96.93	96.91	96.86
196.67	96.28	96.80	96.96	97.00	97.00	96.97
220.00	96.28	96.82	97.00	97.06	97.07	97.05

(f) Efficiency Map for 5th Gear

boundary 1 Torque [Nm]	boundary1 Speed [1/min]					
	1200.00	1960.00	2720.00	3480.00	4240.00	5000.00
140.00	96.16	96.29	96.30	96.26	96.18	96.08
153.33	96.23	96.37	96.40	96.38	96.31	96.22
166.67	96.27	96.43	96.48	96.46	96.41	96.33
180.00	96.31	96.48	96.54	96.53	96.49	96.42
193.33	96.33	96.52	96.58	96.59	96.56	96.50
206.67	96.35	96.55	96.62	96.64	96.62	96.57
220.00	96.37	96.57	96.66	96.68	96.67	96.62

(g) Efficiency Map for 6th Gear

Figure 4.17: Efficiency Maps for each gear

4.7.2 Efficiency Plots

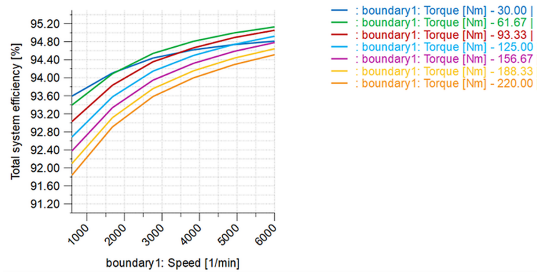
In order to better analyze the evolution of the efficiency of the gearbox as a function of the input speed and torque of each gear, we can transform the Figure 4.17 into plots represented in Figure 4.18.

Analyzing the graphs of the reverse gear and the first gear (Figures 4.18a and 4.18b respectively), we can conclude that most of the curves converge to a maximum efficiency value at high speeds. However, the 30 Nm torque curve does not converge to the same efficiency value that the other curves converge to. This phenomenon occurs because, at such low torque, the overall system power is too small to effectively offset the increase in churning losses at high rotational speeds.

As the intermediate gears (second, third, and fourth gears) - presented in Figures 4.18c, 4.18d, and 4.18e, respectively - are evaluated, it is observed that the inability to converge toward a maximum efficiency at high speeds progressively extends to higher torque values. In cases where this converging trend is lost, the curves begin to exhibit a downward trajectory, indicating that the peak efficiency point shifts to lower speeds; consequently, maximum efficiency is no longer achieved at high rotational speeds.

The plots presented in Figures 4.18f and 4.18g illustrate the efficiency variation for the fifth and sixth gears, respectively. In these gears, the convergence of the maximum efficiency for high speeds disappears completely, resulting in a dispersion of the curves. As these gears present a relatively low gear ratio, the rotation speed of the shafts will be substantially higher in relation to the remaining gears, which exponentiates the churning losses. Consequently, all the curves possess the decreasing phase, that is, in all cases there is a drop in efficiency at high speeds, although they are with different magnitudes. For example, in the cases of higher torque, the total power is sufficient to minimize the percentage weight of the churning losses, maintaining a high efficiency. Conversely, lower torque scenarios fail to provide enough power to overcome the churning losses, leading to a decline in the overall efficiency of the system.

These plots reinforce the efficiency behavior of the gearbox, explaining the root cause of the efficiency drop at high rotational speeds, which is limited by load-independent losses. The division of the energy loss into its contributions will be approached next with the analysis of the power loss graphs.



(a) Efficiency Plot for the R Gear

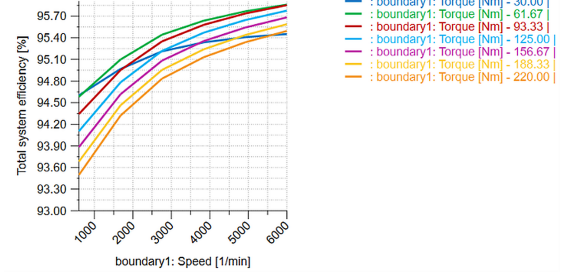
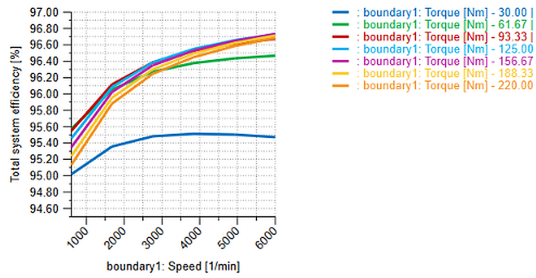
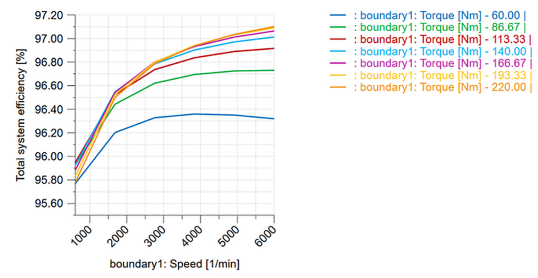
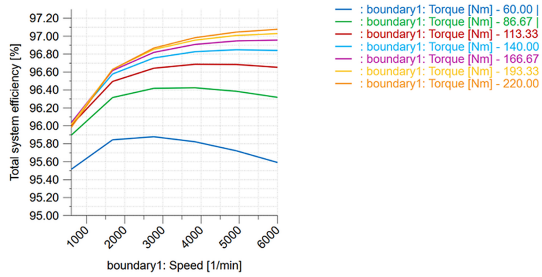
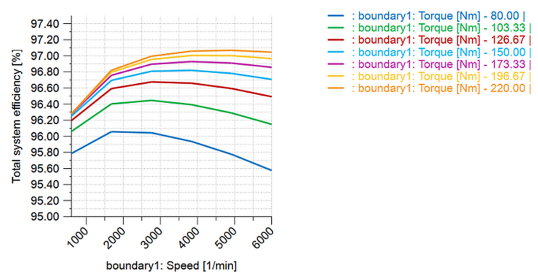
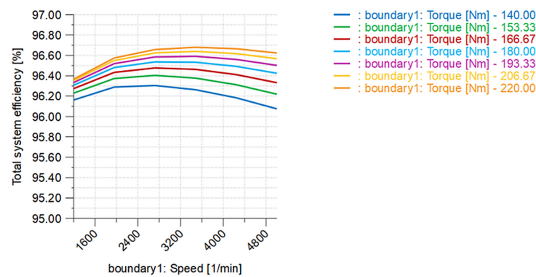
(b) Efficiency Plot for 1st Gear(c) Efficiency Plot for 2nd Gear(d) Efficiency Plot for 3rd Gear(e) Efficiency Plot for 4th Gear(f) Efficiency Plot for 5th Gear(g) Efficiency Plot for 6th Gear

Figure 4.18: Efficiency Plots for each gear

4.7.3 Power Loss Plots

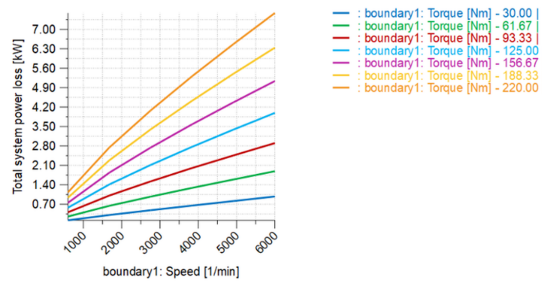
As previously observed during the analysis of the efficiency maps, the maximum efficiency point is not static, that is, it shifts in the speed and torque spectrum as different gears are engaged. This behavior stems from the trade-off between the input power supplied to the system and the

growth rate of its respective losses. With this in mind, the graphs present in Figure 4.19 were constructed, which represent the evolution of the total power loss.

In the gears with a higher gear ratio, the contact pressures are maximum but the speeds of the shafts are relatively low. This fact results in a dominance by the load-dependent losses, leaving the load-independent losses as a secondary factor. This explains the slowdown of the growth rate of the losses, which can be visualized in the graphs of Figures 4.19a and 4.19b (reverse gear and first gear respectively). This slowdown of the growth rate makes the maximum efficiency occur at high speeds and low torque. By maintaining the low torque, the load-dependent losses are reduced, while the moderate input power (due to the high rotation speed) is able to offset the overall losses which, as can be visualized in the graphs, decreases its growth rate.

In the intermediate gears such as the third and fourth gear (graphs 4.19d and 4.19e respectively), the gear ratio decreases in relation to the lower gears, the contact pressure decreases and the rotation speeds of the gearbox shafts increase. As a result, the churning losses start to gain relevance, altering the growth rate of the total losses. The peak efficiency point occurs at maximum rotational speed and maximum torque, as a balance is established between load-dependent and load-independent losses. Under these conditions, neither the high speed nor the high torque compromises the gearbox efficiency, while the transmitted power is simultaneously maximized.

The higher gears, fifth and sixth gears, represented by the graphs 4.19f and 4.19g respectively, present the lowest gear ratios and consequently the lowest contact pressures and highest rotation speeds. Because churning losses constitute a significant portion of the total losses, the growth rate of these losses reverses its behavior: rather than decelerating, it accelerates at high rotational speeds. Due to this phenomenon, the point of highest efficiency abandons the maximum speed, retreating to intermediate speeds. The maximum efficiency is achieved at maximum torque values, as load-dependent losses are less critical than in lower gears, while this condition simultaneously maximizes the transmitted power.



(a) Total Power Loss Plot for the R Gear

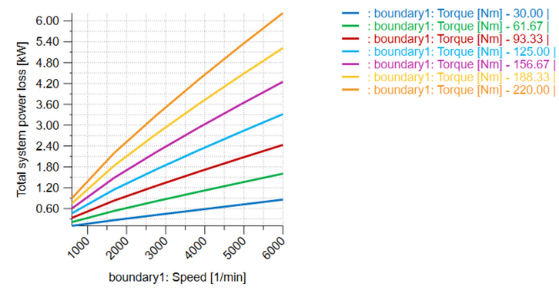
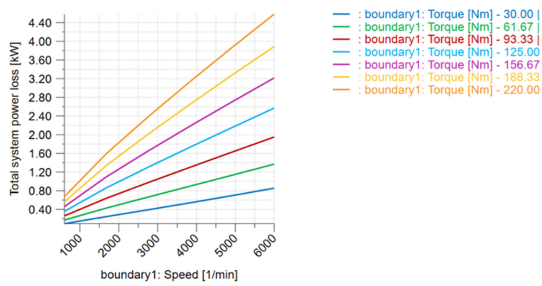
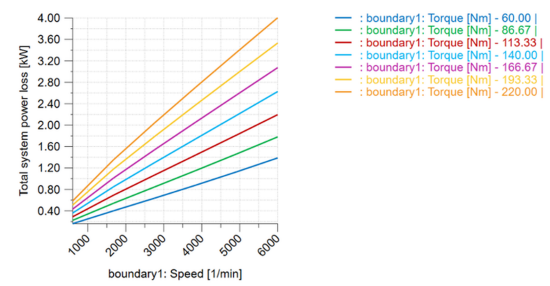
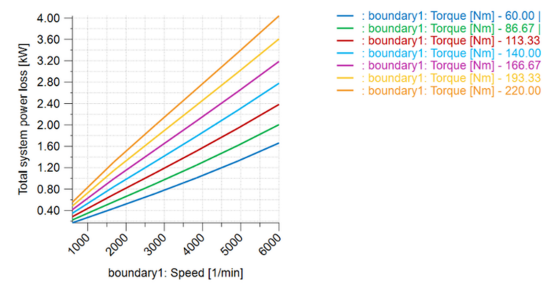
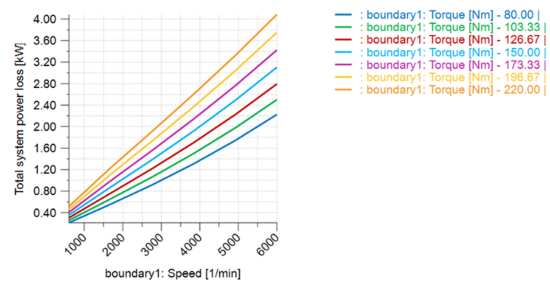
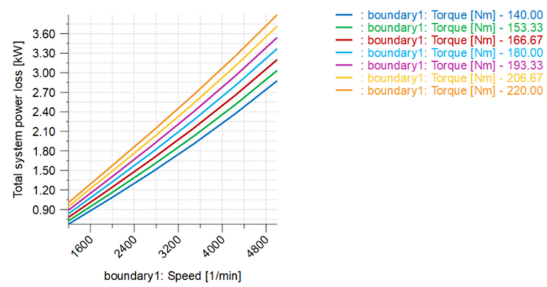
(b) Total Power Loss Plot for 1st Gear(c) Total Power Loss Plot for 2nd Gear(d) Total Power Loss Plot for 3rd Gear(e) Total Power Loss Plot for 4th Gear(f) Total Power Loss Plot for 5th Gear(g) Total Power Loss Plot for 6th Gear

Figure 4.19: Total Power Loss Plots for each gear

To isolate the individual contributions to the overall power loss, the power loss plots for each specific source will be analyzed next.

4.7.4 Gear Load-Dependent Power Loss Plots

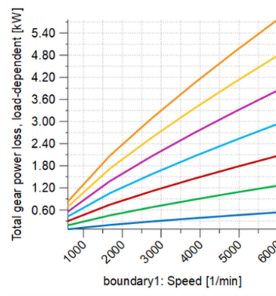
In the graphs of Figure 4.20 it is possible to visualize the behavior of the losses due to the meshing of the gears as a function of the input speed and torque for each gear.

An initial analysis reveals a significant dispersion among the curves. Unlike what will be verified in the load-independent losses, the absolute value of the load-dependent losses is dictated by the applied torque. An increase in the applied torque translates into a higher contact pressure and consequently a higher power loss.

Furthermore, it can be concluded that this type of power loss is highly dependent on the engaged gear. For gears with high gear ratios, such as the reverse and first gears (Figures 4.20a and 4.20b, respectively), high internal torques are generated. These lead to the highest gear mesh losses, reaching peaks of approximately 5.7 kW and 4.9 kW, respectively. On the other hand, in gears such as the fifth and sixth gear (Figures 4.20f and 4.20g respectively) that possess lower gear ratios, such high internal torques are no longer generated resulting in lower losses of this type, with peaks of about 2.3 kW and 2.1 kW respectively.

It is still possible to verify that the growth rate of the power loss due to meshing suffers a slowdown with the increase of the rotation speed. This is due to the fact that the increase of rotation speed induces an increase of the lubricant film and consequently a reduction of the friction coefficient.

The analysis of gear mesh power losses also serves to substantiate the overall power loss trends discussed in the previous section. A comparison between the total power loss plots and the gear mesh power loss plots reveals a strong profile similarity for the lower gears. This resemblance confirms that, in lower gears, the total energy loss is dominated by load-dependent losses, primarily those arising from gear meshing



(a) Gear Load-Dependent Power Loss Plot for the R Gear

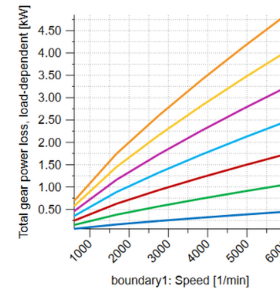
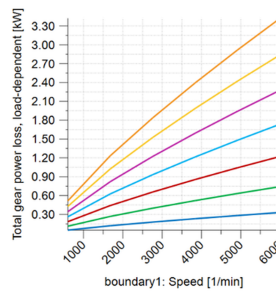
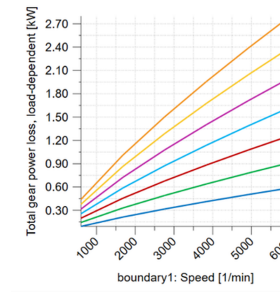
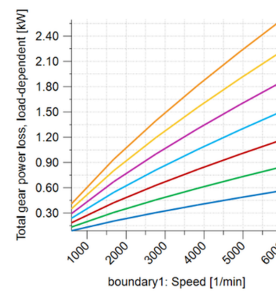
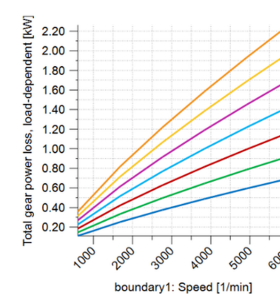
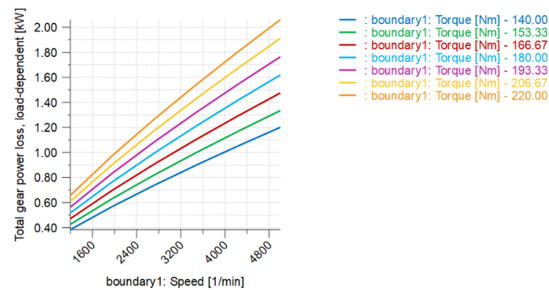
(b) Gear Load-Dependent Power Loss Plot for 1st Gear(c) Gear Load-Dependent Power Loss Plot for 2nd Gear(d) Gear Load-Dependent Power Loss Plot for 3rd Gear(e) Gear Load-Dependent Power Loss Plot for 4th Gear(f) Gear Load-Dependent Power Loss Plot for 5th Gear(g) Gear Load-Dependent Power Loss Plot for 6th Gear

Figure 4.20: Gear Load-Dependent Power Loss Plots for each gear

4.7.5 Total Bearing Power Loss Plots

In Figure 4.21 it is possible to visualize the dissipated power losses in the bearings, being possible to conclude that these have a behavior similar to that verified in the total losses of the gearbox, although with lower absolute values. This is not pure similarity and is due to the formulation of the energy dissipation in the bearings. As was seen in section 2.1.3, the calculation of the losses in a bearing results in the sum of parts that depend on the load (rolling and sliding) and parts independent of the load (drag in the lubricant and seals). Therefore, much like the overall gearbox losses, bearing power losses consist of both load-dependent and load-independent contributions.

Similar to gear mesh losses, bearing power losses exhibit a strong dependence on input torque, which is reflected in the dispersion of the curves. The increase of the torque induces an increase of the axial and radial efforts in the bearings and consequently an increase of losses. As the lower gears possess a higher gear ratio, and consequently a higher internal torque, they register the highest losses due to the bearings. On the other hand, the higher gears, with a lower gear ratio, have a lower internal torque and consequently lower losses due to the bearings.

The respective influences of load-dependent and load-independent losses are reflected in the curve profiles, depending on the engaged gear. For lower gears, characterized by low internal rotational speeds and high internal torque, a slight deceleration in the growth rate of bearing losses is observed. This occurs because an increase in rotational speed leads to a reduction in the friction coefficient. Conversely, for higher gears, the rotational speeds are significantly higher; consequently, lubricant drag losses become predominant over load-dependent losses, causing the overall loss growth rate to accelerate at high rotational speeds.

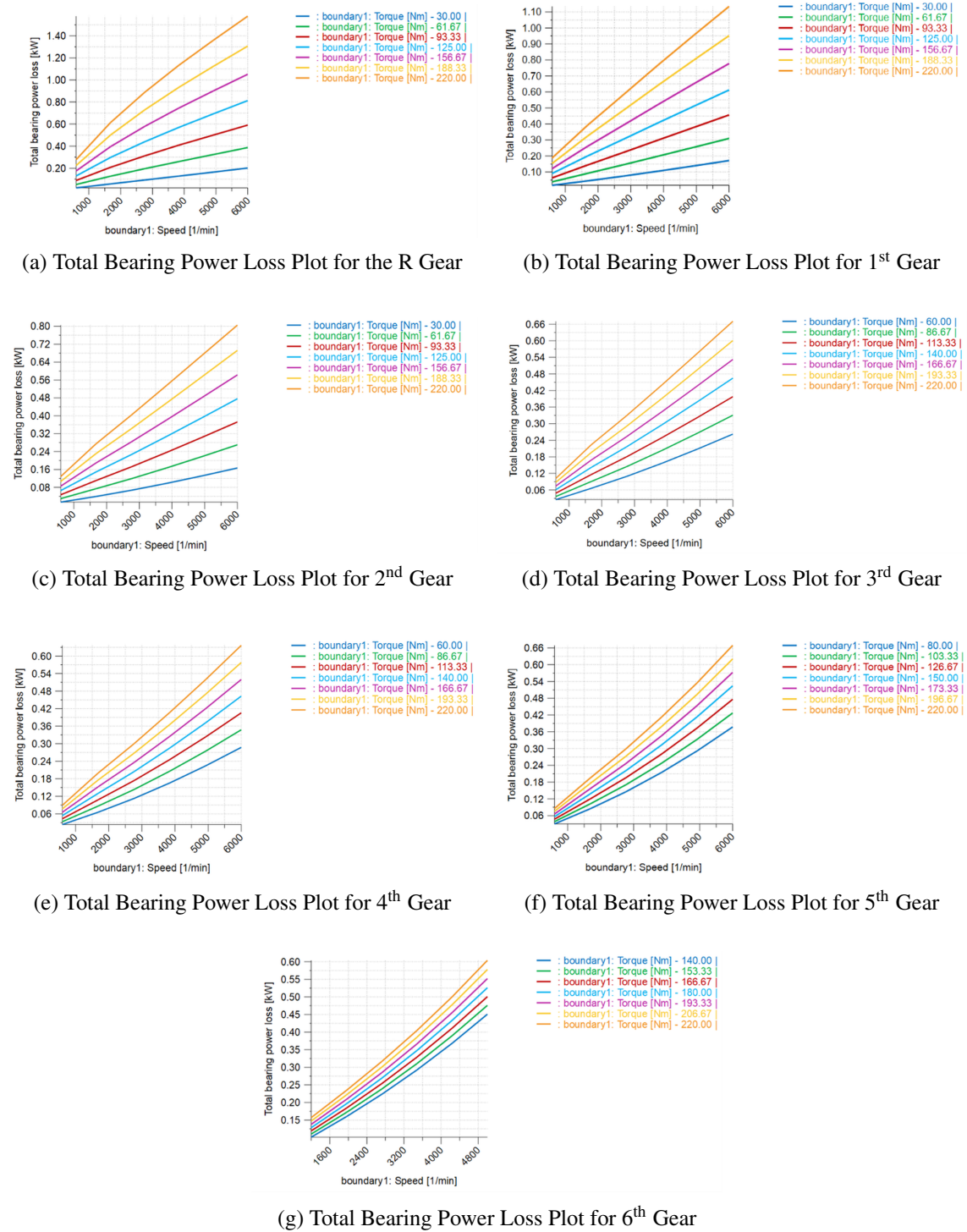


Figure 4.21: Total Bearing Power Loss Plots for each gear

4.7.6 Seals Power Loss and Gear Load-Independent Power Loss Plots

After the isolation of the contributions dictated by the torque, in this section the load-independent losses (no-load losses) will be presented and analyzed. These losses result from the drag gener-

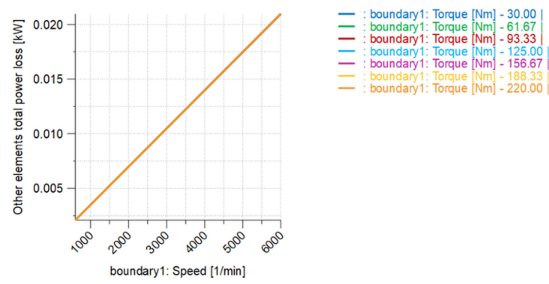
ated by the gears rotating immersed in the lubricant bath and by the friction generated by the three seals of the gearbox (1 on the input shaft and 2 on the differential). It should be noted that while bearings also incur load-independent losses, the KISSsoft software only provides the overall total bearing losses.

Analyzing the graphs of the power loss generated by the seals and by churning (Figure 4.22 and 4.23 respectively) we verify the overlap of the curves of the different input torques, instead of the dispersion that occurs in the losses due to meshing and bearings. This behavior proves the independence of this type of losses relative to the applied torque.

Regarding seal power losses, their magnitude is negligible compared to the total power, reaching a maximum peak of approximately 0.035 kW within the analyzed spectrum. Furthermore, the growth of these losses as a function of rotational speed is linear.

In contrast, the churning losses reveal themselves as a limiting factor of the system at high speeds. In the lower gears due to the low rotation speed of the internal shafts, the churning losses are relatively low (in the reverse gear and first gear the peak of these losses reaches 0.22 kW). In the higher gears the rotation speeds of the gears in the lubricant bath are elevated resulting in losses of 1.10 kW surpassing the losses that come from the bearings.

It is also possible to observe the non-linear behavior of the churning losses, mainly in the higher gears. In these gears we can observe an exponential increase of the losses with the speed. This exponential growth, reaching significant values, explains the shift in the maximum efficiency point for higher gears (fifth and sixth). As previously discussed, for these gears, the peak efficiency point shifts from the maximum rotational speeds to intermediate speeds.



(a) Seals Power Loss Plot for the R Gear

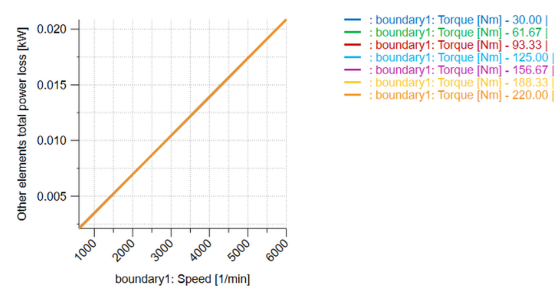
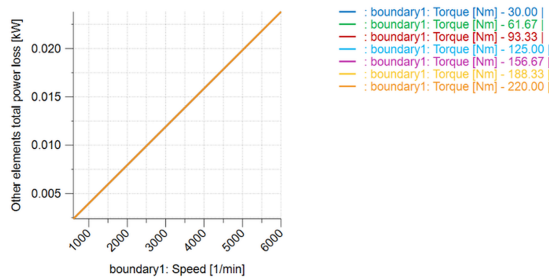
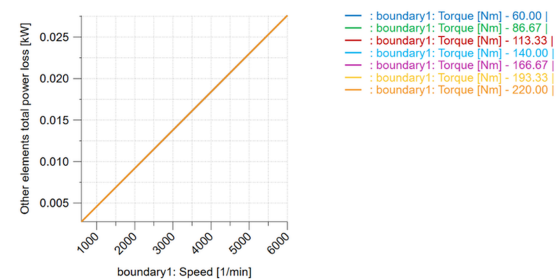
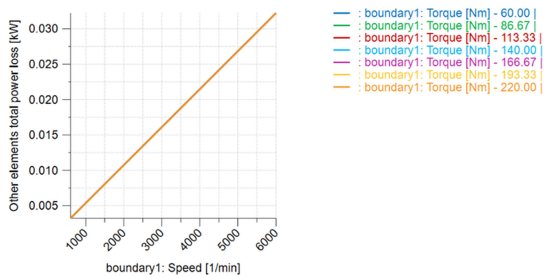
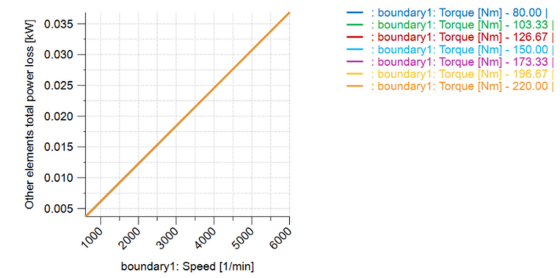
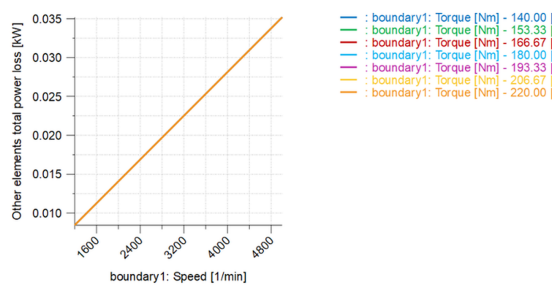
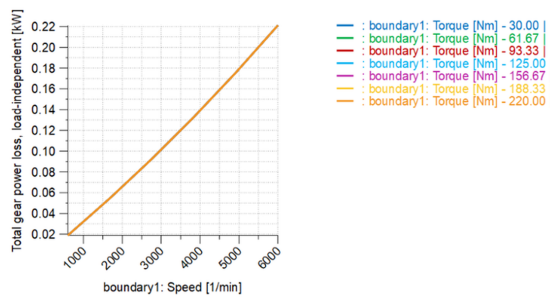
(b) Seals Power Loss Plot for 1st Gear(c) Seals Power Loss Plot for 2nd Gear(d) Seals Power Loss Plot for 3rd Gear(e) Seals Power Loss Plot for 4th Gear(f) Seals Power Loss Plot for 5th Gear(g) Seals Power Loss Plot for 6th Gear

Figure 4.22: Seals Power Loss Plots for each gear



(a) Gear Load-Independent Power Loss Plot for the R Gear

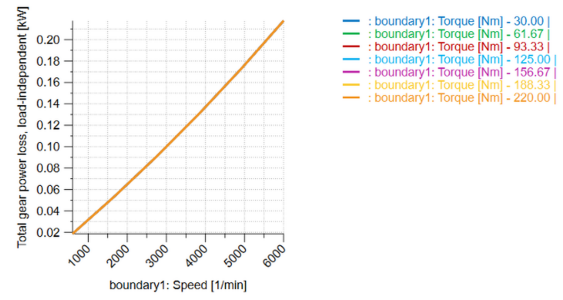
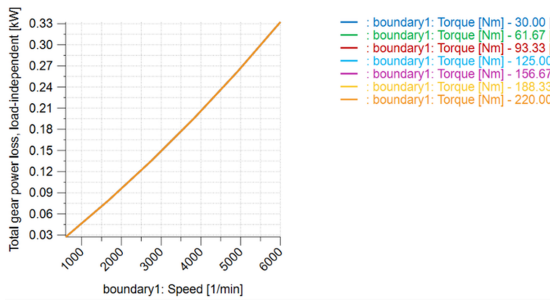
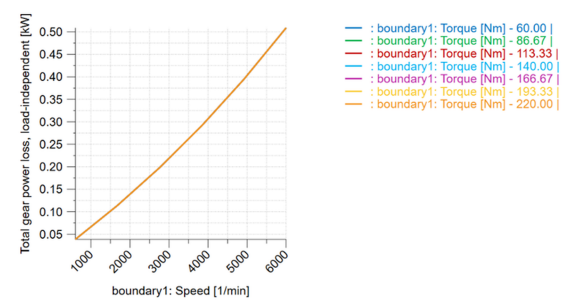
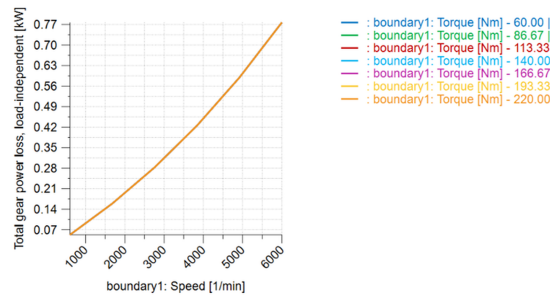
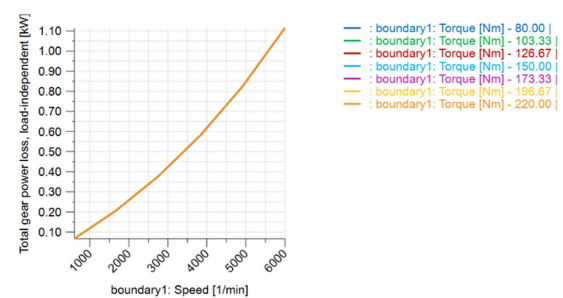
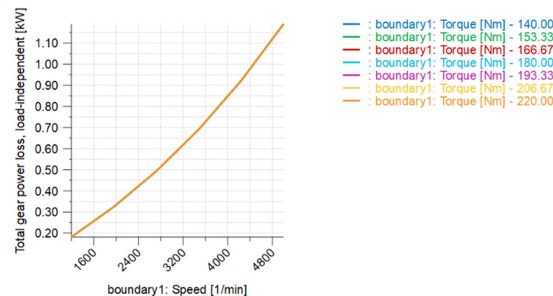
(b) Gear Load-Independent Power Loss Plot for 1st Gear(c) Gear Load-Independent Power Loss Plot for 2nd Gear(d) Gear Load-Independent Power Loss Plot for 3rd Gear(e) Gear Load-Independent Power Loss Plot for 4th Gear(f) Gear Load-Independent Power Loss Plot for 5th Gear(g) Gear Load-Independent Power Loss Plot for 6th Gear

Figure 4.23: Gear Load-Independent Power Loss Plots for each gear

Chapter 5

Experimental campaign

Currently, the use of test benches at HORSE Aveiro is restricted to NVH (Noise, Vibration, and Harshness) analysis. However, data from these tests was analyzed to estimate the overall gearbox efficiency. To better understand the impact of gear tooth quality on transmission efficiency, an experimental testing campaign was planned for the FEUP test rig. This section describes the adaptation carried out on the FEUP test bench to quantify the power loss in the JT4 transmission.

5.1 Test Rig

To conduct the gearbox efficiency study, two test benches are available at FEUP. One evaluates the total power losses, while the other assesses the no-load losses.

5.1.1 Total power losses test rig

To measure the total power losses of the gearbox, a back-to-back test rig configuration with power recirculation will be used. To enable this power recirculation, two sets of helical gears lubricated by oil injection are used, located at positions 2 and 8 in Figure 5.1.

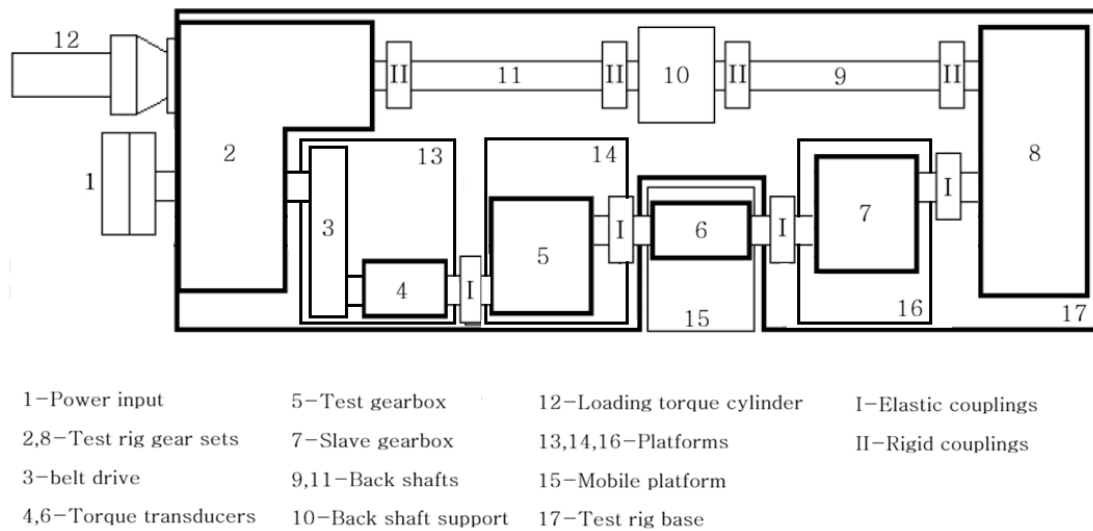


Figure 5.1: Top view diagram of the gearbox test rig (Adapted from [27]).

As the test rig operates in a back-to-back configuration, a test gearbox and a slave gearbox are used, ensuring that the input speed of one matches the output speed of the other. Consequently, this mechanism ensures that the electric motor is only required to compensate for the overall power losses of the system. For this reason, only reversible gearboxes can be tested. The test gearbox and the slave gearbox are denoted by numbers 5 and 7, respectively. The torque transducer located between the two gearboxes is adjustable in both height and depth.

The test rig enables the independent control of torque and rotational speed. Power is supplied by a three-phase motor and transmitted to the test bench through a belt drive with a transmission ratio of 1:2.36, indicated at position 1 in Figure 5.1. The rotational speed is regulated by a frequency inverter.

The torque loading mechanism consists of a hydraulic cylinder that applies an axial force to a helical gear within gear set 2 (Figure 5.1). This induces a slight rotation of the gear, creating an angular displacement and thereby loading the system with a static torque. The position of the hydraulic cylinder, and consequently the torque applied to the system, is regulated by a PID controller.

In this configuration, the maximum torque within the test rig is concentrated between the test and slave gearboxes, resulting in smoother overall operating conditions since the remainder of the rig experiences reduced torque. Furthermore, to avoid critical speeds, the return shafts are not constructed as single-piece components. For the gearbox efficiency analysis, sensors are employed to record the input and output torque and rotational speed of the test gearbox. The ambient temperature, the oil temperature of the test gearbox, and the housing wall temperatures of both gearboxes are also recorded.

With the test rig's current configuration, it is possible to apply an input speed ranging from 100 to 1,900 rpm and an input torque ranging from 100 to 1,300 Nm to the transmission under test [3].

5.1.2 No-load power losses test rig

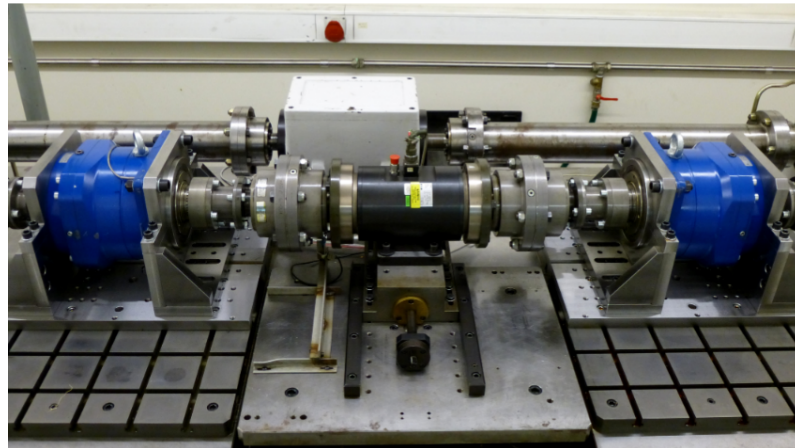
To isolate and analyze losses independent of the load, a no-load power loss test rig can be used. All necessary modifications have been completed to enable the testing of the JT4 gearbox. Detailed information regarding all the performed modifications is provided in the appendix of this report. The Figure 5.2 shows the no-load power loss test rig as well as the control center [3].



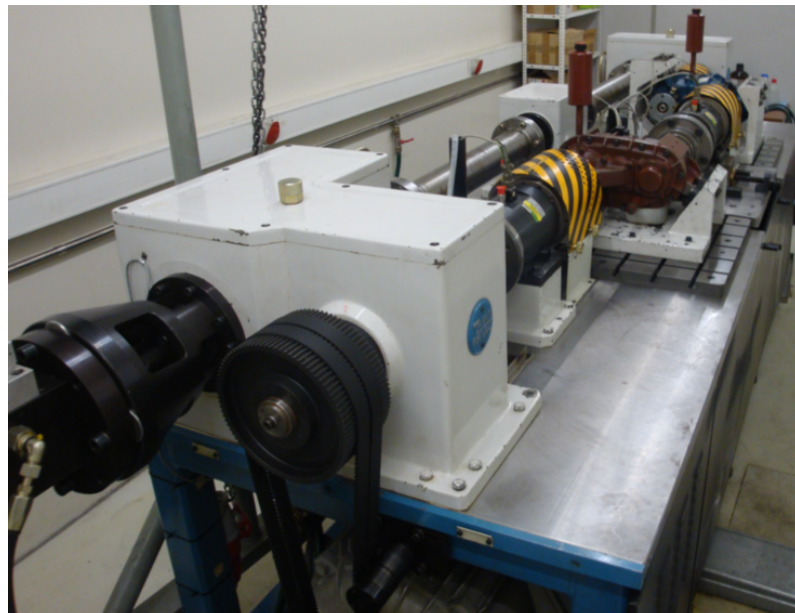
Figure 5.2: No-load power loss test rig

5.1.3 Adaptation of the FEUP test rig

The test bench in question has been used in several previous studies to investigate the efficiency of planetary gearboxes [3, 27] and transfer gearboxes [4, 8]. In the case of the planetary gearbox, the input and output are axially aligned, which simplifies the layout of the test bench setup (Figure 5.3a). In the case of the transfer gearbox, its input and output are misaligned in the parallel direction (a characteristic it shares with the JT4 gearbox). Furthermore, the transfer box can be mounted upside down, which simplifies the layout of the test bench setup (Figure 5.3b).



(a) Layout of the bench rig test for planetary gearbox [3, 27]



(b) Layout of the bench rig test for the transfer gearbox [4, 8]

Figure 5.3: Test rig setups used previously

Regarding the JT4 gearbox, an initial setup similar to the one used for the transfer gearbox previously tested on this bench was considered, given that both feature misaligned input and output shafts. However, due to the characteristics of the JT4 gearbox's breather (a component that allows the gearbox to vent to the atmosphere and maintain pressure balance - in other words, allows excess pressure to be relieved), it is not possible to mount the JT4 gearbox upside down.

Another problem that occurred was due to the geometry of the gearbox under study, which makes it impossible to use the existing platforms shown in Figure 5.4. With these platforms, mounting the gearbox on the test stand would cause interference.

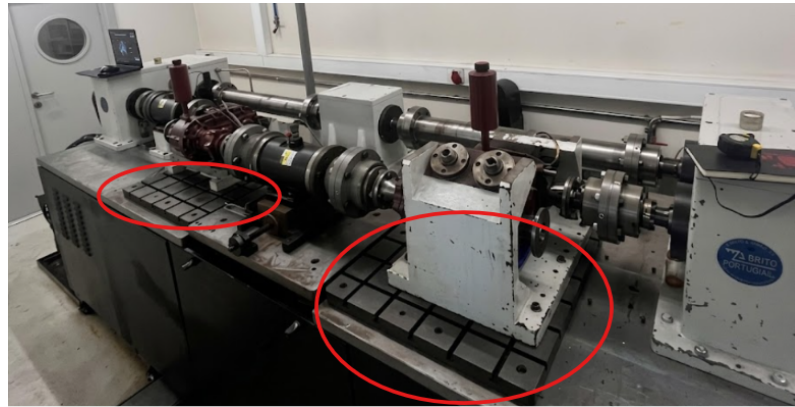
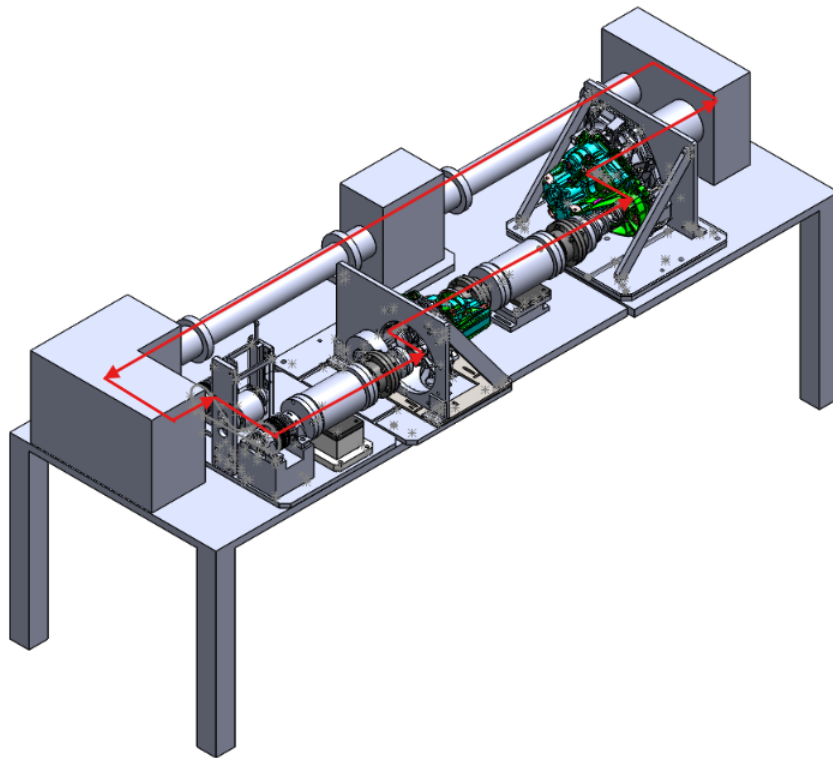


Figure 5.4: Incompatibility with pre-existing platforms

Due to these restrictions, it was necessary to create a new test bench setup to enable testing of the JT4 transmission. More than 30 different parts were designed. Due to the age of the test bench and the limited information available about it, some difficulties were found in designing the components needed to adapt the bench. With this in mind, and in order to enable possible adaptations in future work on the bench, all the 2D component drawings that were designed and manufactured are attached to this report.

The power path in the chosen test bench setup can be explained as follows. Power enters through the test bench shaft and is transmitted via a belt drive to the torque sensor located at the input of the main gearbox. The power passes through the main gearbox and then through the torque sensor located between the main gearbox and the slave gearbox. Given the input and output torque speeds of the gearbox being tested, it is possible to calculate its efficiency. The power then passes through the slave gearbox and returns to the test bench. The resulting setup and power path (illustrated by the red lines) are shown in Figure 5.5a.

Currently, the gearbox outputs are connected directly to each other via the central torque sensor, confining the high torque to this area and thus preventing the entire system from being subjected to it. A setup was also considered in which, instead of the output, the input of the gearboxes would be connected directly through the central sensor; that is, looking at Figure 5.5a, the two gearboxes would have to rotate 180° . One of the disadvantages of this setup compared to the one chosen would be the fact that the gearboxes outputs are connected to the test bench shafts, that is, the entire test bench would be subjected to high torque. However, the main reason this setup was rejected was that the belt system is limited in its ability to transmit torque; it is unthinkable to transmit the torque from the output of an automotive transmission via belts.



(a) Layout of the bench rig test for JT4 gearbox and the power path.



(b) Test bench adapted for the JT4 gearbox

Figure 5.5: Test rig setups used for study of the JT4 gearbox

To connect the gearbox input and output, the same interfaces used on the HORSE JT4 gearbox production line were utilized. At the end of the production line, the gearboxes undergo NVH (noise, vibration, and harshness) testing. The input interface, which is mounted to the primary shaft, is shown in Figure 5.6a, while the output interface and its corresponding seal, fitted to the gearbox differential, can be seen in Figure 5.6b



(a) Input interface.

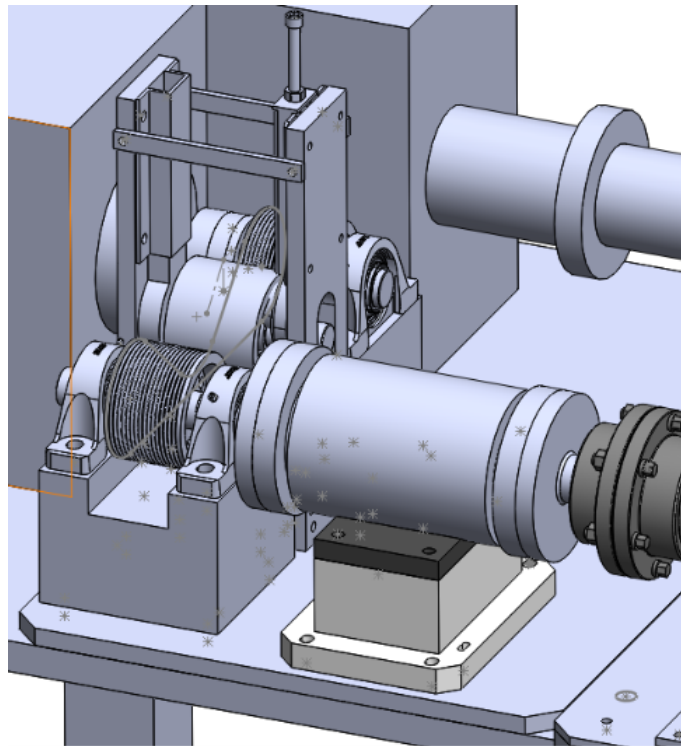


(b) Output interface.

Figure 5.6: Input and output interfaces used on the HORSE JT4 gearbox production for NVH testing.

To keep it straightforward, we can divide the new setup into four main assemblies: the belt drive assembly, the main gearbox assembly, the central sensor assembly, and the slave gearbox assembly, represented by numbers 13, 14, 15, and 16 in Figure 5.1.

The belt drive assembly includes the belt system as well as the mounting base for the torque sensor. Since the test bench table has a limited number of holes and cannot be drilled, this assembly incorporates a custom-machined base. This design allows for flexible hole placement while maintaining a robust structure. The assembly was designed to transmit power from the test bench shaft to the torque sensor and, subsequently, to the main gearbox. This assembly is shown in Figure 5.7.



(a) Model of belt drive assembly in SolidWorks.



(b) Belt drive assembly mounted on the test bench

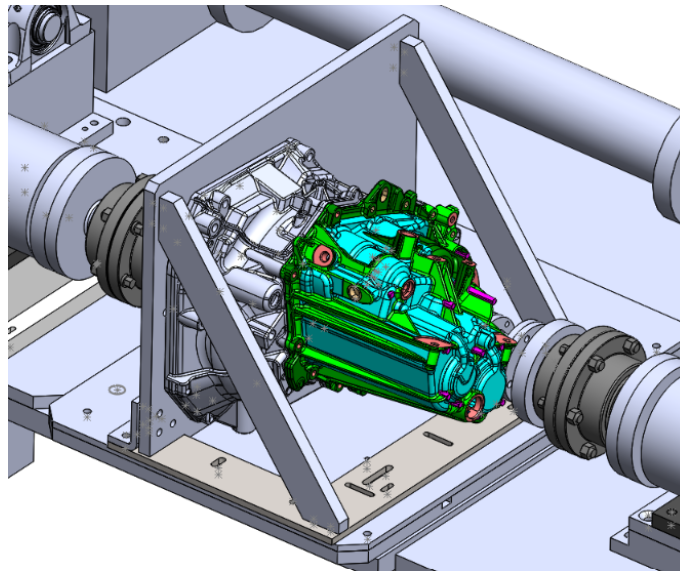
Figure 5.7: Belt drive assembly.

To maintain the correct belt tension, it was chosen to use a system with a tensioning unit (two take-up units with a flat belt pulley) that allows the belt tension to be adjusted by turning two screws. A V-ribbed belt was chosen because a flat belt would not have as much torque capacity, and a toothed belt would introduce vibrations that would affect the torque measurement by the sensor. Standard components, described in Table 5.1, were used in this assembly.

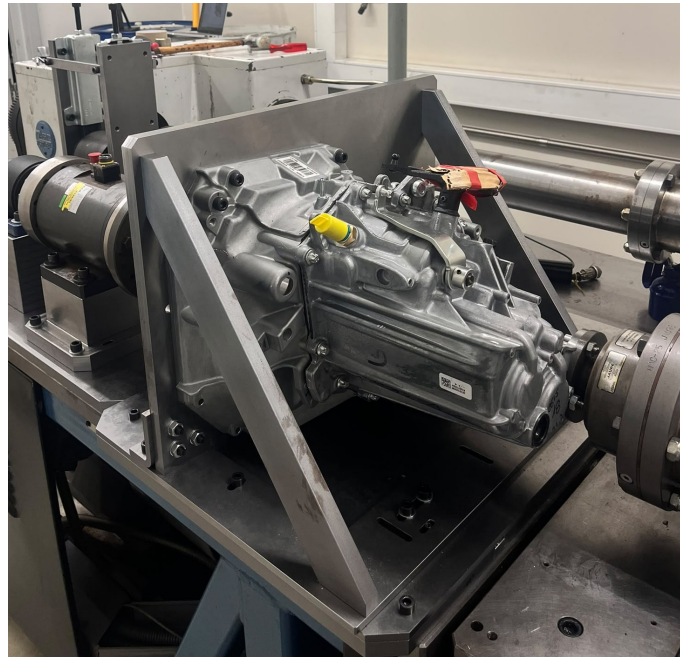
Table 5.1: List of mechanical components and references.

BRAND	DESIGNATION	REFERENCE
SKF	Pillow block ball bearing unit	UCP 208
PTI	Take-Up Housing Unit	PHUSE35-XL
TYMA	V-ribbed belt	1149 PL-16 CONTI-V MULTI-RIB
TYMA	Flat Belt Pulley	FS 125 - 100 TB 2517 (Type 2, Cast Iron)
OPTIBELT	Ribbed Belt Pulleys	TB 16 PL 125
OPTIBELT	Taper bush	2517-50
TYMA	Taper bush	TB 2517-45
OPTIBELT	Taper bush	2517-40

The main gearbox assembly, inspired by the structure used in HORSE, was designed to support the gearbox. This structure can be seen in Figure 5.8. This assembly also includes a new base, eliminating the incompatibility of the JT4 gearbox with the original platforms shown in Figure 5.9.



(a) Main gearbox assembly model in SolidWorks



(b) Main gearbox assembly mounted on the test bench

Figure 5.8: Main gearbox assembly

To enable gearbox assembly and disassembly and ensure proper alignment, the base of this assembly consists of two sliding plates, with the upper plate guided by grooves in the lower plate. Cylindrical components are screwed onto the upper plate (shown as transparent), allowing it to slide relative to the lower plate. This mechanism is shown in Figure 5.9.

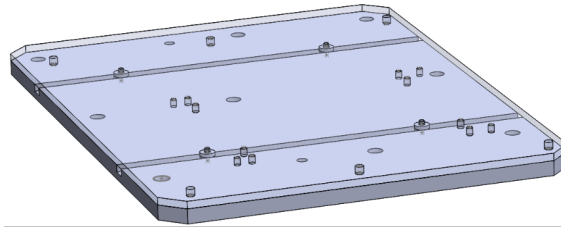


Figure 5.9: Base of main gearbox assembly.

Throughout the component design process, the need to test the main gearbox on the DeMec laboratory's no-load test bench was taken into account. With this in mind, and to enable the assembly, disassembly, and transport of the setup between test benches, a ring-shaped part was designed. In addition, this part enables lateral and height adjustments of the gearbox. This mechanism can be seen in Figure 5.10.

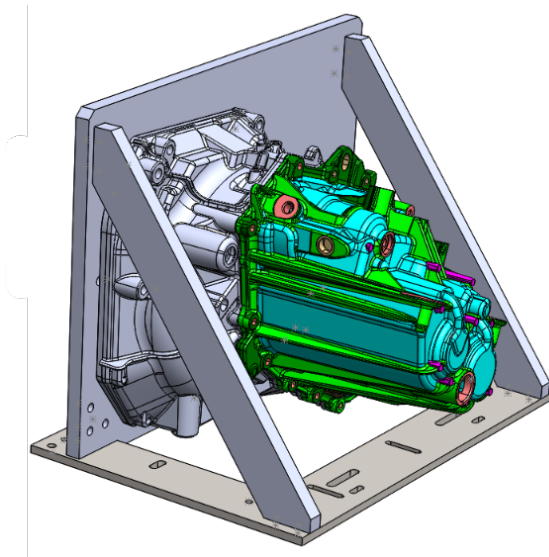
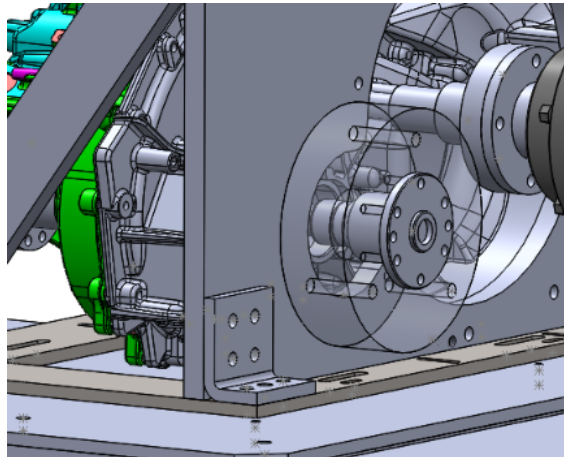


Figure 5.10: Ring-shaped part for main gearbox assembly.

Due to the configuration of the laboratory test bench, the gearbox output motion can only exit from one side of the differential, requiring the other side to be static. To perform this locking, a component was designed. This system, shown in Figure 5.11, consists of fastening the output interface to the vertical mounting plate of the gearbox, thus ensuring the output motion exits from only one side of the differential.



(a) Differential Lock System model in SolidWorks



(b) Differential Lock System mounted on the test bench

Figure 5.11: Differential Lock System

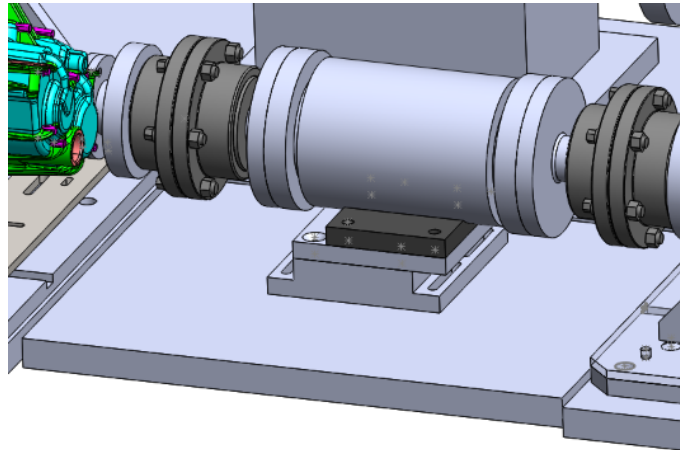
In addition, it was necessary to design a component that acts as a seal to prevent oil leakage in the differential lock system. For this purpose, the black component shown in Figure 5.12 was 3D printed using TPU (Thermoplastic Polyurethane), a material offering both rubber-like flexibility and plastic durability. The component was dimensioned to have an interference fit with the output interface and the gearbox housing to reduce the risk of oil leaks. The printing parameters—specifically the number of layers and the infill density—were adjusted to achieve the flexibility required for assembly while maintaining sufficient structural integrity to ensure a reliable

seal.



Figure 5.12: 3D-printed seal mounted on the output interface

Another aspect that had to be adapted was the central assembly where the central torque sensor - which connects the outputs of the two gearboxes - is located. Although the test bench has a movable platform, allowing for flexibility in the sensor's height and depth, it does not allow for lateral adjustment of the sensor. Due to the geometry of the gearbox, a new base for the sensor had to be designed to overcome this limitation. The base, consisting of two parts, is shown in Figure 5.13.



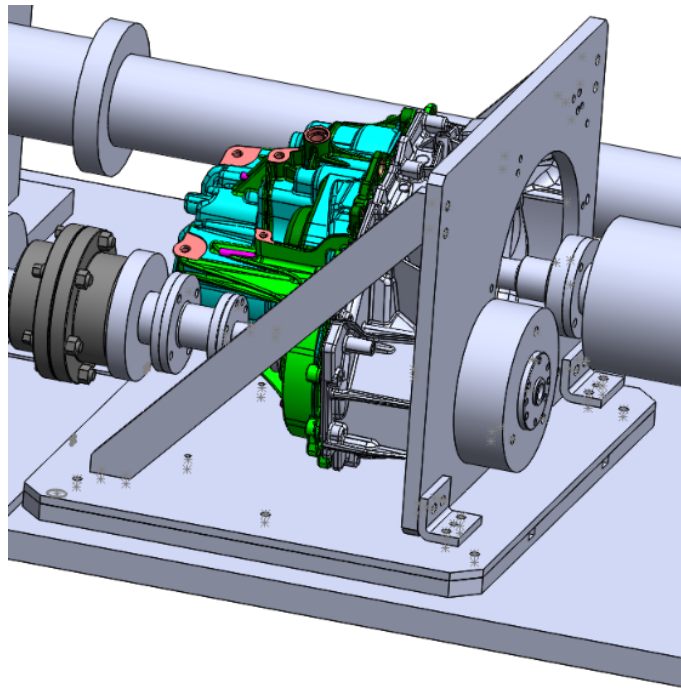
(a) Central sensor assembly model in SolidWorks



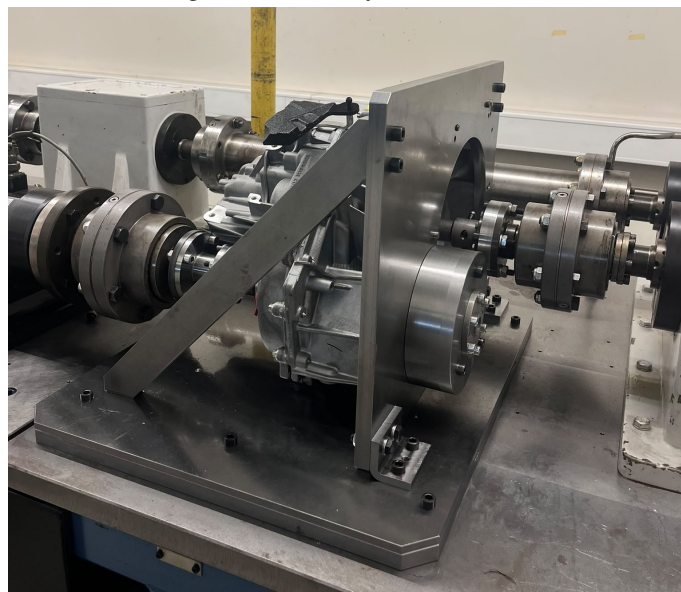
(b) Central sensor assembly mounted on the test bench

Figure 5.13: Central sensor assembly

As previously mentioned, the test rig operates in a closed-loop configuration, meaning the input and output shaft speeds of the rig must be identical. To achieve this, an identical slave gearbox is used to counteract the gear ratio of the main gearbox. Thus, as was done for the main gearbox, a complete support structure was designed and manufactured for the slave gearbox. The structure is very similar to the main gearbox structure, featuring a differential lock system equal to that of the main gearbox (Figure 5.11). The same interfaces were also used to couple the input shaft and the gearbox differential (Figure 5.6). As shown in Figure 5.14, the main differences were at the dimensional level due to reduced space, as well as the absence of the ring-shaped component present in the main gearbox assembly. Eliminating this component reduces costs and is justified, as the slave gearbox requires neither frequent assembly and disassembly nor no-load testing. However, minor height adjustments to the gearbox are still accommodated.



(a) Slave gearbox assembly model in SolidWorks



(b) Slave gearbox assembly mounted on the test bench

Figure 5.14: Slave gearbox assembly

Chapter 6

Experimental results and Model Verification

In order to obtain the real efficiency of the gearboxes and analyze the accuracy of the KISSsoft model, gearbox tests were conducted. Due to the impossibility of carrying out tests on the FEUP test bench, the HORSE Aveiro test bench was used.

6.1 HORSE results

HORSE Aveiro has a test bench for NVH (Noise, Vibration and Harshness) testing to evaluate the gearbox's operational noise and vibration, as well as gear shift quality. Currently, efficiency tests are not performed at HORSE Aveiro, and for that reason, the adaptation of FEUP's test bench was carried out.

However, following the operational suspension of FEUP's Mechanical Engineering Laboratory, the NVH test datasets were re-examined. This analysis revealed that the HORSE Aveiro test bench (Figure 6.1) records input speed and torque at the gearbox primary shaft, as well as individual output speeds and torques at each differential output. In addition, a thermocouple is used to record the oil temperature and ambient temperature.



Figure 6.1: HORSE Aveiro test bench

Thus, both versions of the Golden gearboxes were subjected to NVH testing on the test bench. Subsequently, the experimental results were acquired and the necessary data processing was performed.

The test report presents the record of the time, torque and rotational speed of the input and of each output of the differential, as well as the lubricant oil and ambient temperatures.

Since the bench records the speed in revolutions per minute (rpm), the first step was to change the speed to radians per second (rad/s). In the KISSsoft model, the differential was simplified; consequently, the output speed on both sides is identical, simulating a locked differential. During testing on the HORSE bench, the gearbox operates under conditions equivalent to a vehicle traveling in a straight line, resulting in similar speeds at both differential outputs. Therefore, the simplification adopted in the KISSsoft model accurately reflects this physical phenomenon.

Thus at the output of the gearbox an arithmetic mean of the speed and torque was performed. For every time step of the test, both the input and output power were calculated. Subsequently, the average power difference between the input and output was determined for each gear, yielding the overall power loss and, consequently, the gearbox efficiency and torque loss.

The tests performed at HORSE are started with two preliminary stages, with the gearbox engaged in fourth gear. The first occurs with an input of 60 Nm and 3000 rpm and then the second stage occurs with an input of 100 Nm at 5000 rpm. These two stages have the main function of pre-heating the lubricant. Thus when the first gear is tested the oil temperature will already be

around 50 °C. It is important to note that in the NVH tests performed at HORSE the reverse gear is not tested.

Figure 6.2 shows the primary shaft power and differential power recorded by the test bench, together with the oil temperature evolution over time. The power plot allows the different stages of the test to be identified, including the oil warm-up phase and the acceleration and deceleration phases associated with each gear, from first to sixth gear. The negative power component in the graph represents the deceleration phase and is primarily due to the direction in which the input and output torques are applied. The data analysed in the following sections correspond to the steady-state phase of each gear.

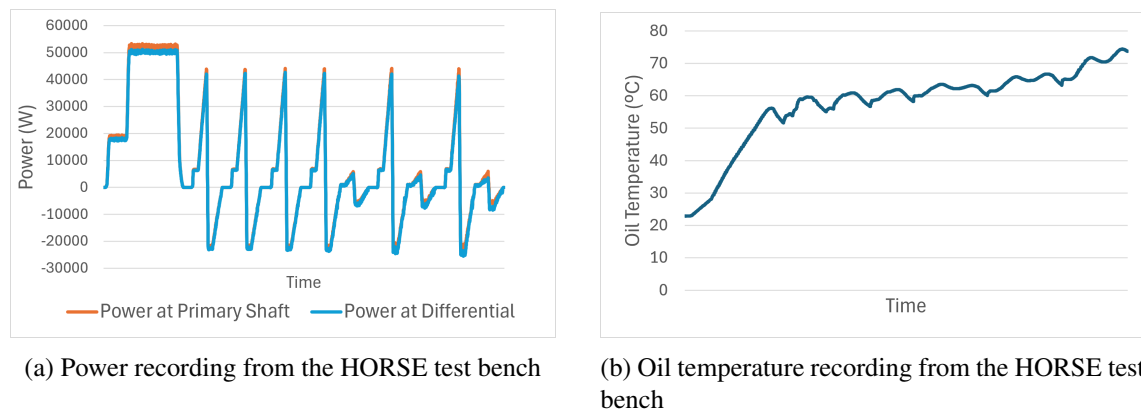


Figure 6.2: Data recorded during the NVH test at HORSE

Subsequently, the experimental results for both versions of the Golden gearboxes will be presented, alongside the data obtained from the KISSsoft model. A comparative analysis of these results will also be conducted.

Golden Gearbox - 1st Version

Table 6.1 shows the relevant data from the NVH tests carried out on version 1 of the Golden gearbox.

Table 6.1: NVH test data and results – Version 1

Gear	Oil Temp. (°C)	Operating Conditions	Input Power (W)	Output Power (W)	Efficiency (%)	Power Loss (W)
1	53.73	80 Nm / 800 rpm	6700.97	6332.19	94.50	368.77
2	56.06	80 Nm / 800 rpm	6702.30	6425.67	95.87	276.63
3	58.53	80 Nm / 800 rpm	6705.47	6434.08	95.95	271.39
4	59.94	80 Nm / 800 rpm	6705.54	6419.52	95.73	286.02
5	61.47	80 Nm / 800 rpm	6704.32	6424.69	95.83	279.63
6	65.04	80 Nm / 800 rpm	6723.95	6369.74	94.73	354.21

To obtain the simulation results from KISSsoft, the corresponding input parameters were introduced for each gear, and individual simulations were executed for every gear ratio. It is important to note that the torque loss analysis was performed on the gearbox output shaft. The results derived from the computational model are presented in Table 6.2.

Table 6.2: Results from the KISSsoft model – Version 1

Gear	Efficiency (%)	Power Loss (W)	Torque Loss (N.m)
1	94.55	365.7	69.05
2	95.66	290.7	32.36
3	96.03	265.9	19.35
4	95.94	272	13.91
5	95.87	276.9	10.91
6	95.31	314.4	9.79

Figure 6.3 shows a graph comparing the efficiency obtained from the test carried out at HORSE with that obtained using the KISSsoft model.

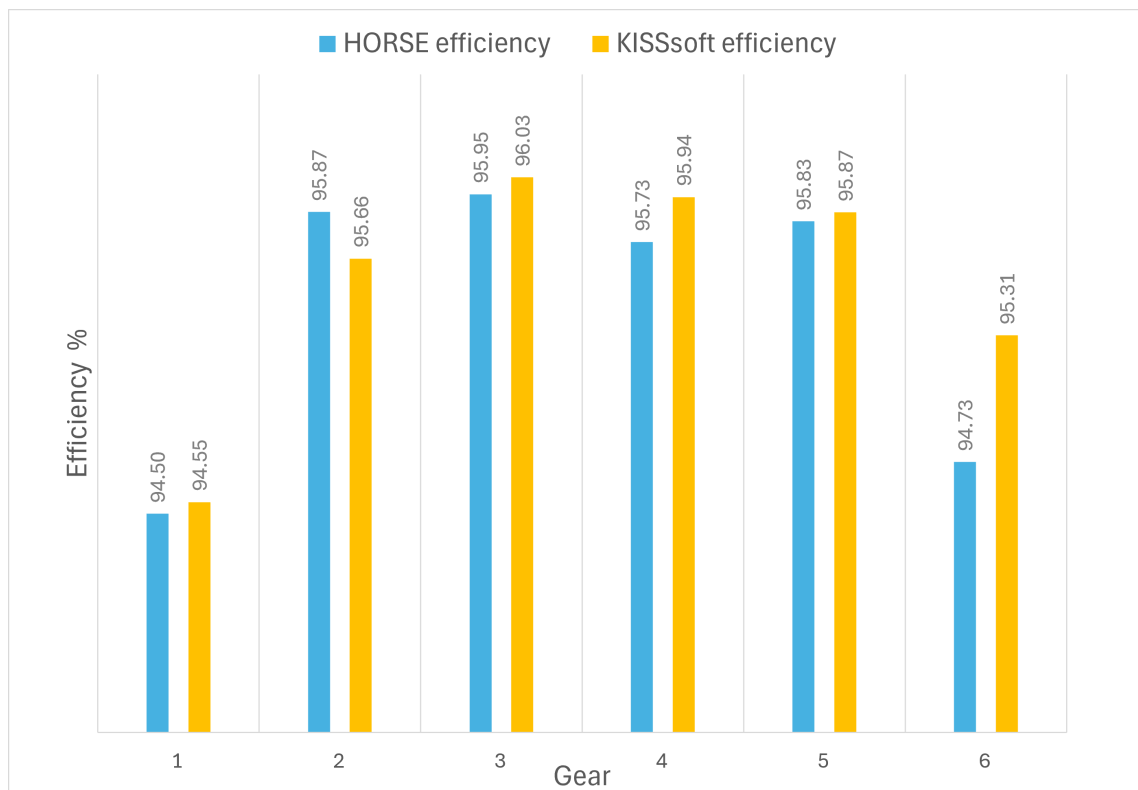
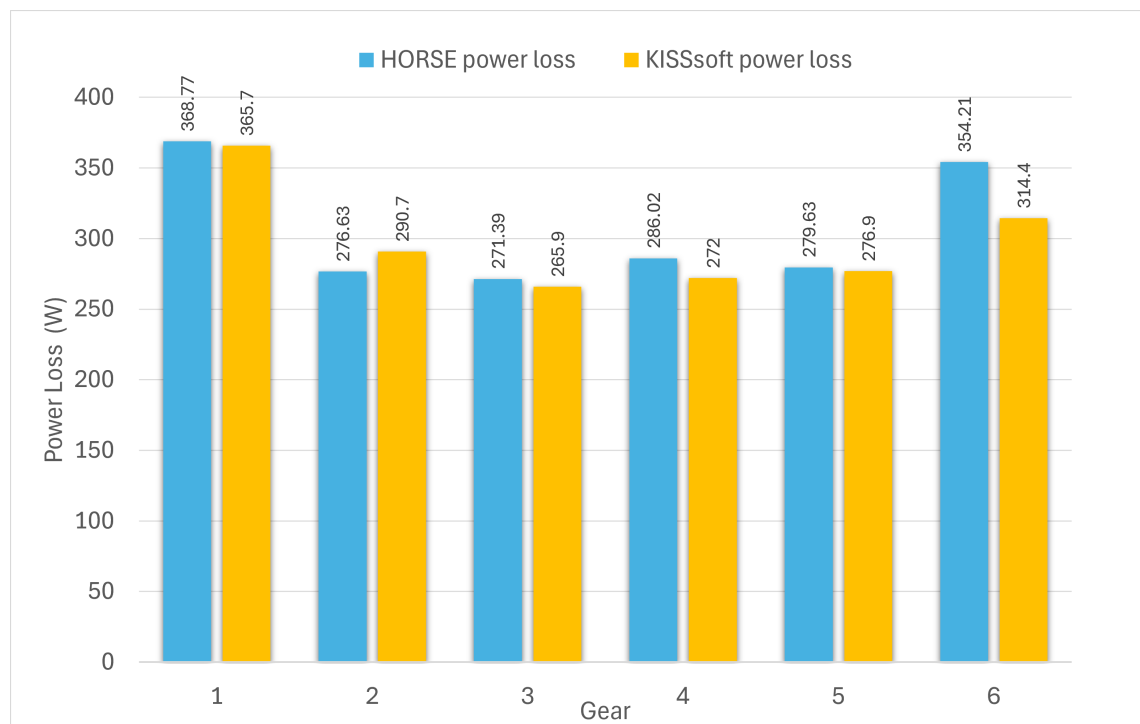
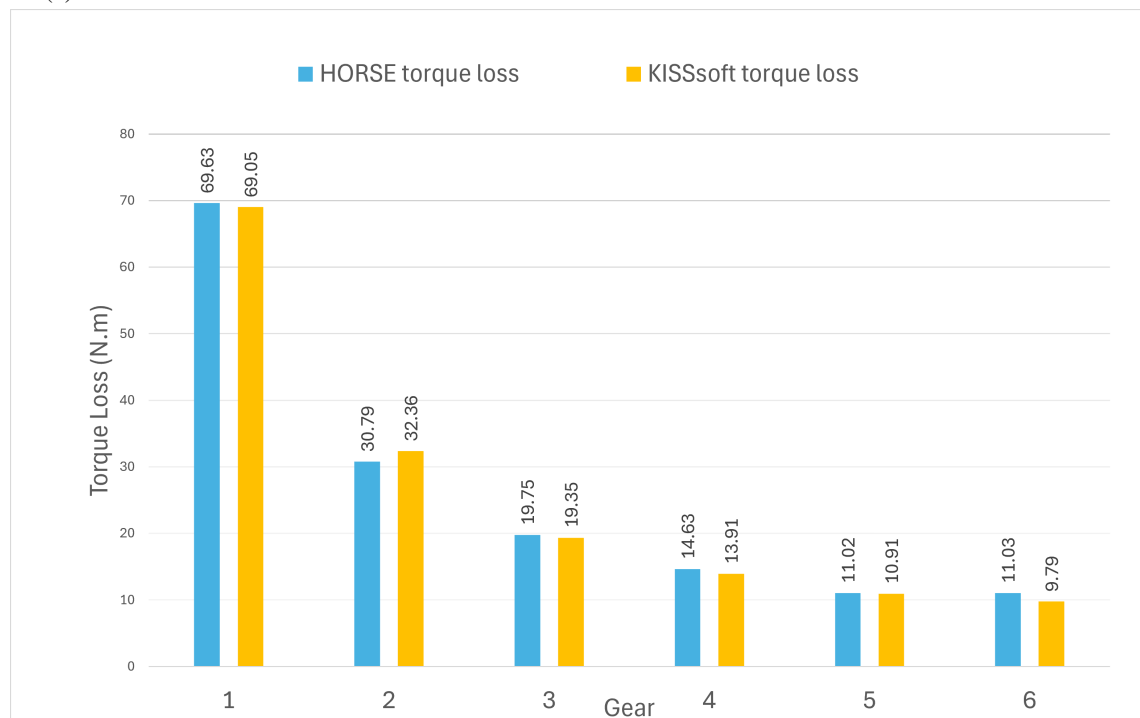


Figure 6.3: Efficiency derived from the tests carried out at HORSE and from the KISSsoft model-Version 1

Analyzing efficiencies can be a misleading approach for evaluating the behavior of the numerical model, as a specific percentage difference may correspond to a substantially high or negligible power loss depending on the total power involved. Consequently, evaluating power loss and torque loss provides a more accurate representation of the model's performance. The graphs that make this comparison can be visualized in Figures 6.4a and 6.4b, respectively.



(a) Power Loss derived from the tests carried out at HORSE and from the KISSsoft model-Version 1



(b) Torque Loss derived from the tests carried out at HORSE and from the KISSsoft model-Version 1

Figure 6.4: Results from the tests carried out at HORSE and from the KISSsoft model- Version 1

By analyzing the graphs presented in Figure 6.4, it is evident that the KISSsoft model can predict the operational behavior of the gearbox on the HORSE test bench within an acceptable margin of error. Both the U-shaped profile of the power loss and the reduction in torque loss are

effectively captured by the numerical model.

With the objective of validating the KISSsoft model we can quantify the absolute and relative error of the prediction of the power loss as well as of the torque loss. In Figure 6.5a we can verify that the model manages to capture reasonably the power loss of the gears. However, in the sixth gear, the error is more noticeable. One of the justifications is the difficulty of mechanical models to capture churning losses, mainly at high speeds, as well as complexities related to the specific gear geometry.

Figure 6.6a demonstrates that the model captures in a very robust way the torque loss of the gearbox. Due to the fact that the power loss translates into torque loss, visualizing Figures 6.5b and 6.6b, we verify, as expected, that the relative errors of these two quantities are identical.

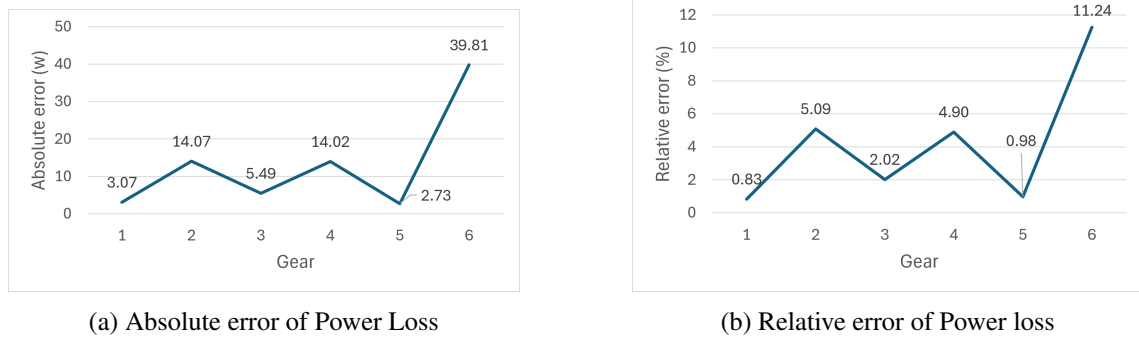


Figure 6.5: Absolute and relative error of Power Loss- Version 1

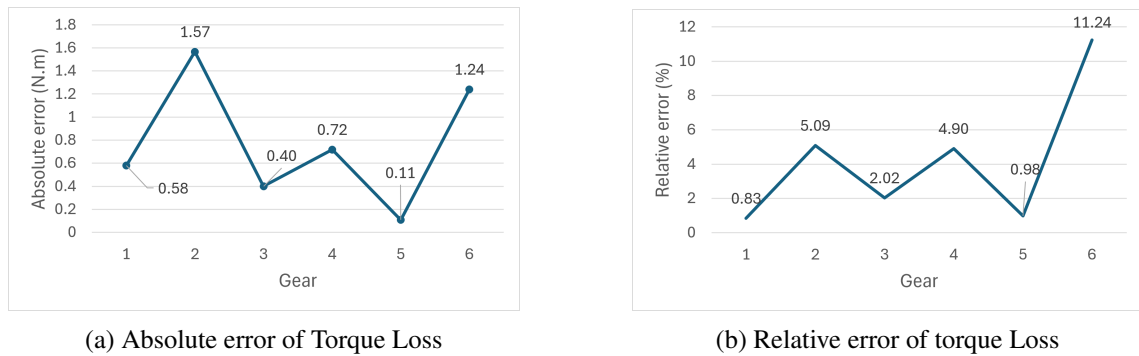
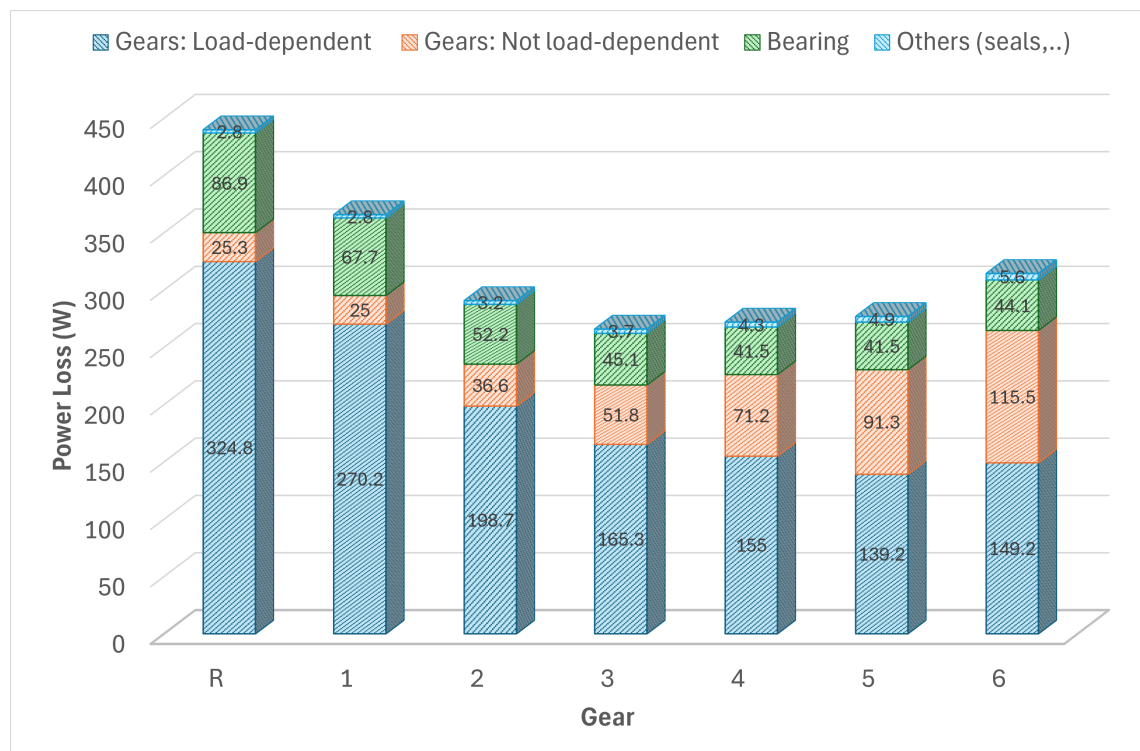
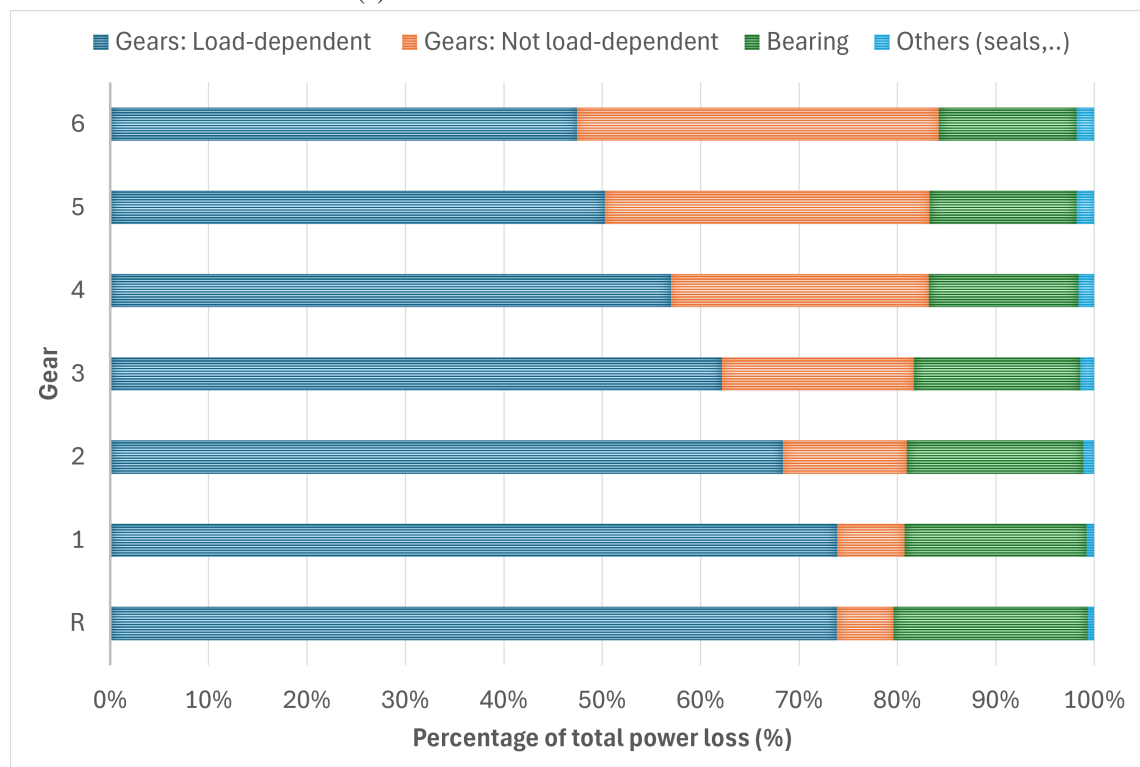


Figure 6.6: Absolute and relative error of Torque Loss- Version 1

A significant feature enabled by the numerical model is the ability to analyze the sources of power loss, as well as their relative contribution to the overall losses. This information is crucial to understand the evolution of the efficiency in the diverse gears. As approached in section 2, and illustrated in Figure 2.1, the power loss in a gearbox originates in the meshing (Load-dependent) in the churning of the gears in the oil (Not load-dependent), in the bearings and in the seals. It should be noted that, although the reverse gear was not subjected to physical testing, the numerical simulation results for this specific gear will still be included in the subsequent graphs.



(a) Absolute Contributions to Power Loss



(b) Relative Contributions to Power Loss

Figure 6.7: Contributions to Power Loss

Analyzing figure 6.7a it is possible to verify that the contribution of the losses due to meshing

(dark blue bar) is massive in the lower gears. This is due to the high transmission ratios of these gears, where the transmitted torque is high, originating high contact pressures and consequently high friction losses. In these gears, due to the lower rotation speed, the churning losses are lower.

In the intermediate gears, the transmitted torque is lower, and consequently the meshing losses decrease in relation to the lower gears. Despite this, the rotation speed is not yet enough for the churning losses to be problematic. Due to these phenomena the valley of the "U" occurs.

In the highest gears the losses due to meshing decrease their participation. On the other hand, due to the higher rotation speed, the churning losses increase their participation. This phenomenon creates the ascending phase of the "U" shape.

Another interesting aspect is the fact that the load-dependent losses of the gears of the sixth gear are superior to those of the fifth gear. This phenomenon may seem counterintuitive because the torque to transmit in the sixth gear is inferior to that of the fifth gear. However, this behavior can be explained by the gear loss factor of each gear pair—a parameter that indicates meshing efficiency and is heavily dependent on gear geometry. Table 6.3 presents the gear loss factor of each gear pair as well as the average friction coefficient of each gear pair resulting from the KISSsoft model. An analysis of these results reveals that the sixth gear exhibits a significantly higher gear loss factor compared to the fifth gear. Therefore, it can be concluded that this increase in load-dependent losses is inherently driven by the specific geometry of the gears.

Table 6.3: Gear loss factors and friction coefficients

Gear	Gear Loss Factor (H_V)	Average Coefficient of Friction	Average Differential Coefficient of Friction
1	0.270	0.088	0.077
2	0.218	0.067	0.069
3	0.191	0.059	0.060
4	0.192	0.059	0.054
5	0.175	0.054	0.051
6	0.210	0.055	0.049
Differential	0.225	—	—

Although there is no experimental data for the reverse gear, it is fundamental to analyze the results of the numerical model for this gear. The reverse gear exhibits the highest power loss and, consequently, the lowest efficiency, estimated at approximately 93.44 %. As shown in Table 4.2, the reverse gear and the first gear have similar gear ratios, with the first gear being slightly higher. Thus, for the same input of torque and speed, when the reverse gear is engaged, the output speed will be slightly higher, almost identical, to if the first gear were engaged. This phenomenon explains the fact that the speed-dependent losses, such as the load-independent gear losses (churning) and the seal losses, are identical for both gears.

It is crucial to emphasize that while the power flow in forward gears follows an input shaft–output shaft–differential sequence, the power path for the reverse gear proceeds through the input shaft, output shaft, reverse shaft, and the differential. This structural difference explains why, despite the

reverse and first gears sharing similar gear ratios, the reverse gear exhibits higher gear and bearing losses. Since the power passes through the reverse shaft, an extra gear mesh is introduced, which explains the increase in gear load-dependent losses (it goes from 270.2 W to 324.8 W), as well as the bearings that support the reverse shaft being subjected to loads, which explains the increase in bearing losses (an increase from 67.7 W to 86.9 W).

The reversal of the relative contributions of gear meshing and churning losses across the different gears is illustrated in the graph of Figure 6.7b. In the first gear, the losses due to meshing have a weight higher than 70 %, while the churning losses have a weight lower than 10 % of participation in the total loss of the gearbox. As higher gears are engaged, the contribution of the meshing to the total losses decreases while the contribution of the churning rises. In the highest gear, the contribution of the meshing to the losses drops below 50 % while the losses due to churning reach around 40 % of the total losses.

The losses due to the bearings (green bar) have a behavior similar to the losses due to the gears, that is, they have a "U-shaped" trend (absolute values). This trend has the same reasons as the gear losses. As analyzed in chapter 2, the bearing losses depend on the forces applied to the bearing as well as the rotation speed. In the lower gears, since there is a high torque to be transmitted, high axial and radial loads are applied to the bearings, causing large power losses. As higher gears are engaged, the transmitted torque decreases; however, the elevated rotational speeds increase the losses associated with the bearing's interaction with the lubricant. In the intermediate gears, there is a balance between the losses dependent on the loads applied to the bearing and the drag losses.

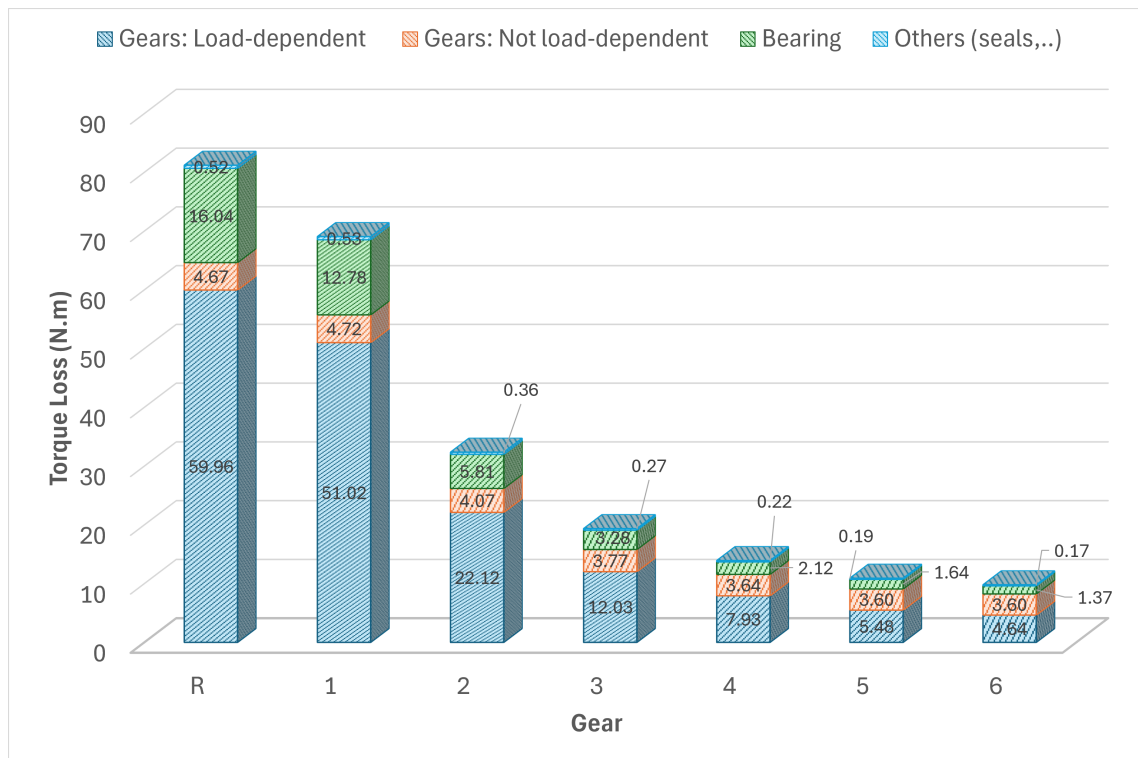
An analysis of the power losses attributed to the seals (represented by the light blue bar) of the input shaft and differential reveals that they are negligible, accounting for less than 2 % of the total gearbox power loss. Because these losses are strictly dependent on rotational speed and independent of the transmitted torque, a slight increase is observed as higher gears are engaged (from 2.8 W in first gear to 5.6 W in sixth gear).

The analysis of Figure 6.8a of the decomposition of torque losses shows a behavior distinct from that observed in power losses. Instead of the "U-shaped" behavior, an exponential decay is registered from the lowest gears to the highest gears.

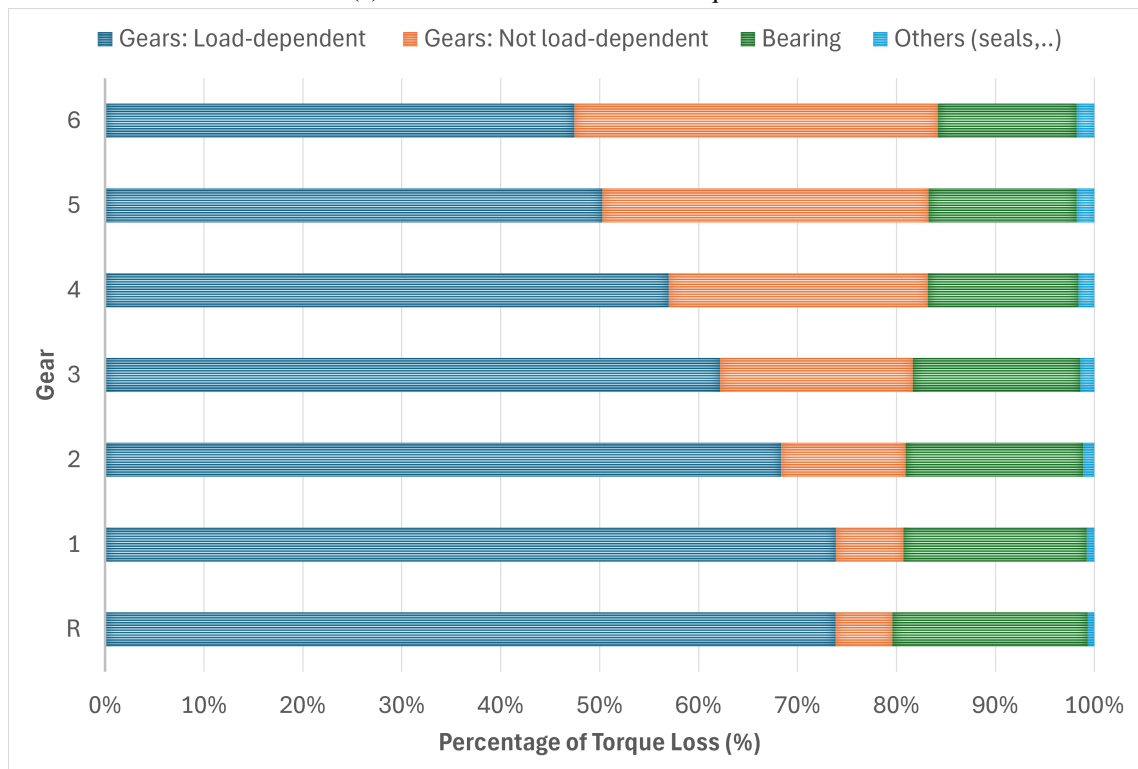
This behavior stems from the inherent kinematics of the gearbox. In a transmission system such as an automotive gearbox, the output rotational speed acts as a strict kinematic constraint dictated by the gear ratio. Consequently, a "loss of speed" cannot occur. Therefore, based on Equation (6.1), any power loss must exclusively manifest as a loss of torque.

$$P = T \cdot \omega \quad (6.1)$$

As is known, in low-speed gears, the high gear ratios generate high torques and, consequently, the inefficiency of the gearbox applied to the absolute value of torque translates into a considerable torque loss. On the other hand, in high-speed gears, since the absolute value of the torque is lower, the torque lost due to the inefficiency is proportionally smaller.



(a) Absolute Contributions to Torque Loss



(b) Relative Contributions to Torque Loss

Figure 6.8: Contributions to Torque Loss

After establishing the reasons for the decrease in torque loss, another question arises: why

does Figure 6.7a show an increase in power loss from churning and seals when engaging high-speed gears, while this phenomenon simultaneously translates into a decrease in torque loss due to churning and seals, as illustrated in the Figure 6.8a? In other words, why is an increase in lost power not translated into a higher torque loss? The explanation is identical to the one referred to in the previous paragraph. To clarify this relationship and eliminate any ambiguity, Equation (6.1) can be rearranged, resulting in Equation (6.2).

$$T = \frac{P}{\omega} \quad (6.2)$$

As higher gears are engaged, even though the power dissipated by churning and seals increases, the rotational speed also rises. Observing Equation (6.2), if the numerator (power) increases but the denominator (speed) has an even greater increase, this will be translated into a lower lost result (torque).

If the graph in Figure 6.8a is converted into a plot of relative values (Figure 6.8b), it can be observed that the degrees of participation of each type of loss are identical to the graph of percentage values of power loss (Figure 6.7b). This finding demonstrates that power dissipation is manifested exclusively as a loss of torque.

Golden Gearbox - 2nd Version

Although these gearboxes share the same gear tooth quality, differing only in their manufacturing sequence as discussed in Section 3.2, the power loss, torque loss, and efficiency results for this gearbox version are presented below. As was done for the first version, a comparison with the KISSsoft model will be performed. Since they are identical gearboxes, significant differences are not expected. Table 6.4 details the experimental testing conditions alongside the corresponding results.

Table 6.4: NVH test data and results – Version 2

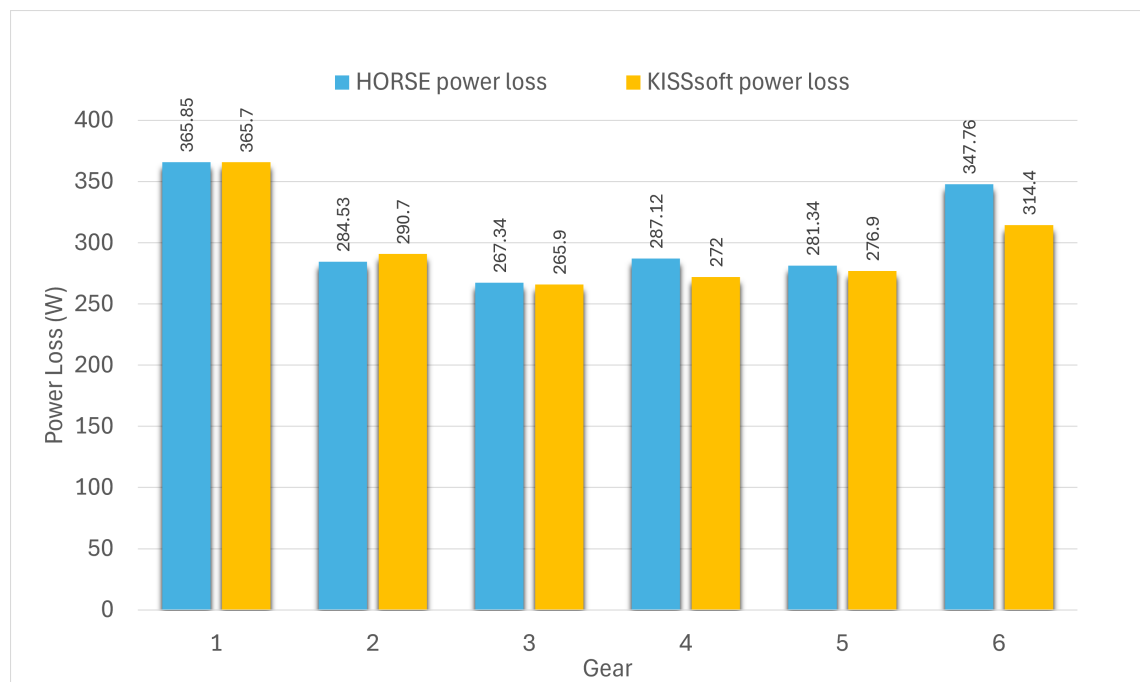
Gear	Oil Temp. (°C)	Operating Conditions	Input Power (W)	Output Power (W)	Efficiency (%)	Power Loss (W)
1	53.89	80 Nm / 800 rpm	6697.82	6331.98	94.54	365.85
2	56.15	80 Nm / 800 rpm	6709.76	6425.23	95.76	284.53
3	58.94	80 Nm / 800 rpm	6702.44	6435.10	96.01	267.34
4	60.21	80 Nm / 800 rpm	6709.80	6422.68	95.72	287.12
5	62.37	80 Nm / 800 rpm	6715.44	6434.10	95.81	281.34
6	66.15	80 Nm / 800 rpm	6714.23	6366.47	94.82	347.76

Using the experimental test data from version 2 of the Golden gearbox, the graph in Figure 6.9 was plotted, illustrating the gearbox efficiency for each gear alongside the efficiency predicted by the KISSsoft model. As anticipated, the model accurately captures the overall efficiencies of the gearbox. Furthermore, it can be confirmed that the efficiency of each gear in this version is identical to that of version 1.

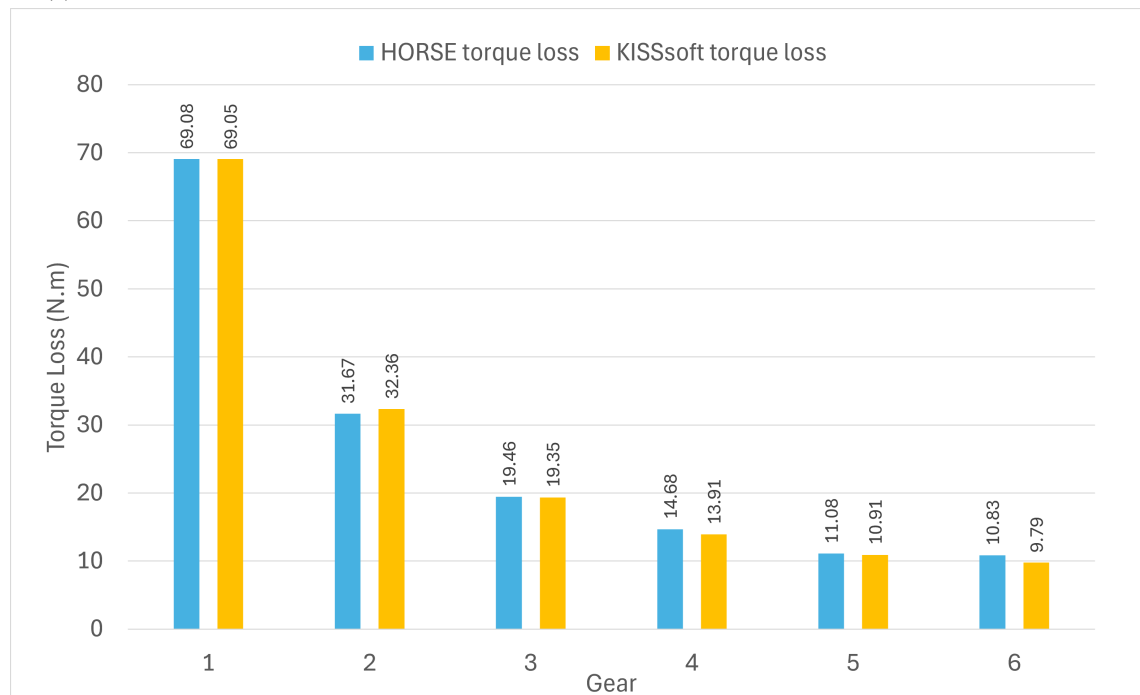


Figure 6.9: Efficiency derived from the tests carried out at HORSE and from the KISSsoft model-Version 2

In order to analyze the power and torque losses, Graphs 6.10a and 6.10b were generated, respectively. As anticipated, the results are identical to those obtained for version 1, further demonstrating the strong capability of the KISSsoft model in accurately calculating power and torque losses.



(a) Power Loss derived from the tests carried out at HORSE and from the KISSsoft model-Version 2



(b) Torque Loss derived from the tests carried out at HORSE and from the KISSsoft model-Version 2

Figure 6.10: Results from the tests carried out at HORSE and from the KISSsoft model- Version 2

Figures 6.11 and 6.12 present the absolute and relative errors in power loss and torque loss for the numerical model results compared to the experimental data.

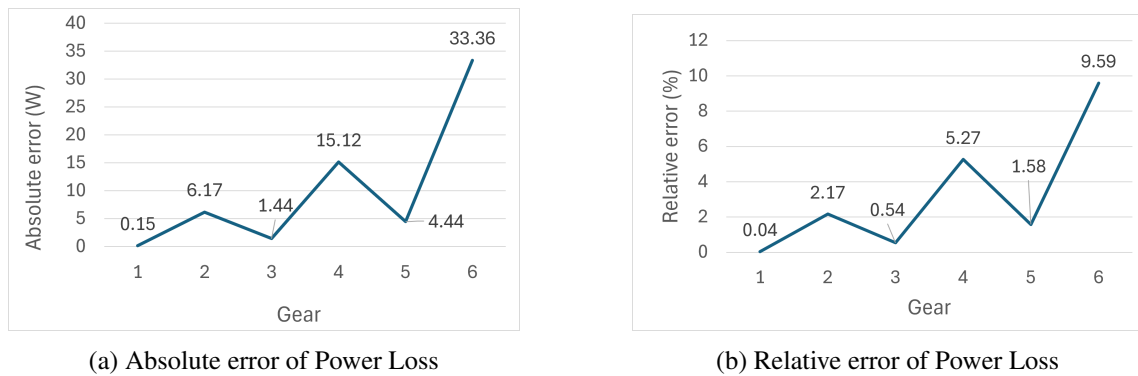


Figure 6.11: Absolute and relative error of Power Loss- Version 2

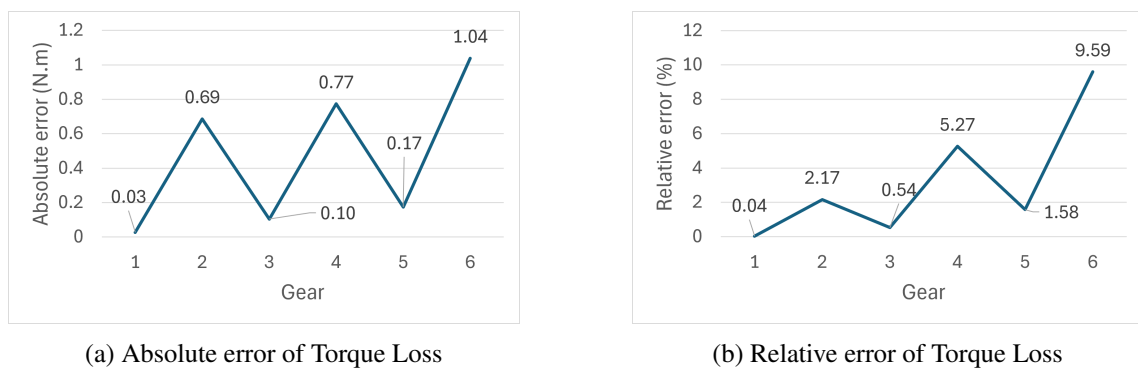


Figure 6.12: Absolute and relative error of Torque Loss- Version 2

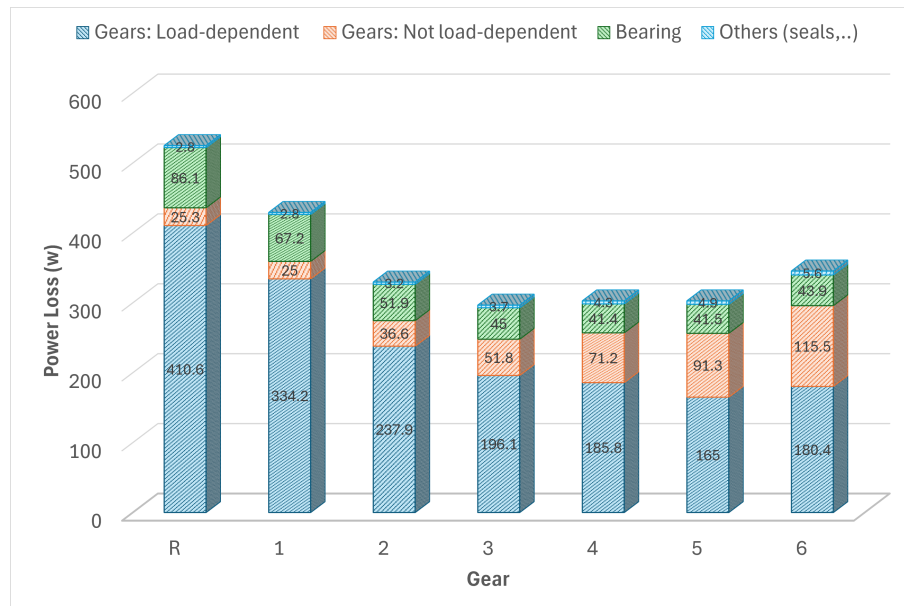
6.2 Comparison with the results of the standard version

To evaluate the impact of gear tooth quality on the overall efficiency of the transmission, the results of the KISSsoft model for the standard gearbox were obtained. The input parameter in the software that differentiates a golden gearbox from a standard one is the surface roughness of the gear tooth flanks. The results obtained by the KISSsoft software for the standard gearbox can be visualized in Table 6.5.

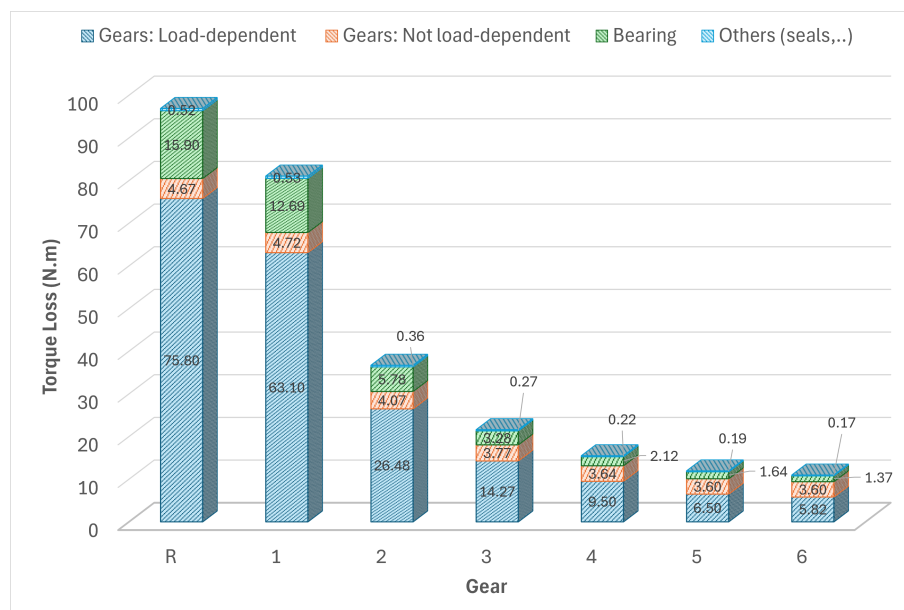
Table 6.5: Results from the numerical model – Standard Version

Gear	Efficiency (%)	Power Loss (W)	Torque Loss (N.m)
R	92.17	524.8	96.88
1	93.6	429.2	81.04
2	95.08	329.6	36.70
3	95.57	296.6	21.59
4	95.48	302.7	15.48
5	95.48	302.7	11.93
6	94.75	351.9	10.95

Figures 6.13a and 6.13b show the breakdown of power and torque losses, respectively, for the series gearbox at input parameters of 80 Nm and 800 rpm.



(a) Contributions to Power Loss- Standard Version



(b) Contributions to Torque Loss- Standard Version

Figure 6.13: Contributions to Power Loss and Torque Loss- Standard Version

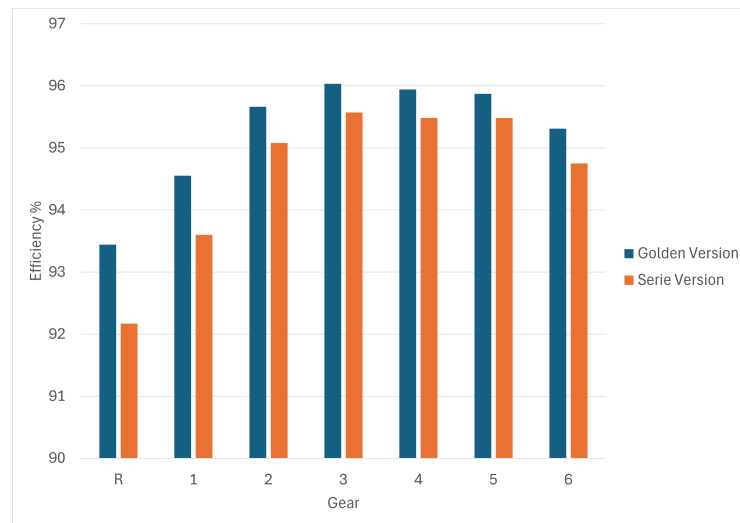
By analyzing the previous tables, it is possible to perform a comparative analysis of the contributions to power and torque losses between the series (Standard) and Golden versions. The distinction between the standard and golden version is the manufacturing process. While the standard gears undergo a shaving process resulting in a higher surface roughness, the golden version

gears are subjected to a grinding process achieving a significantly smoother finish.

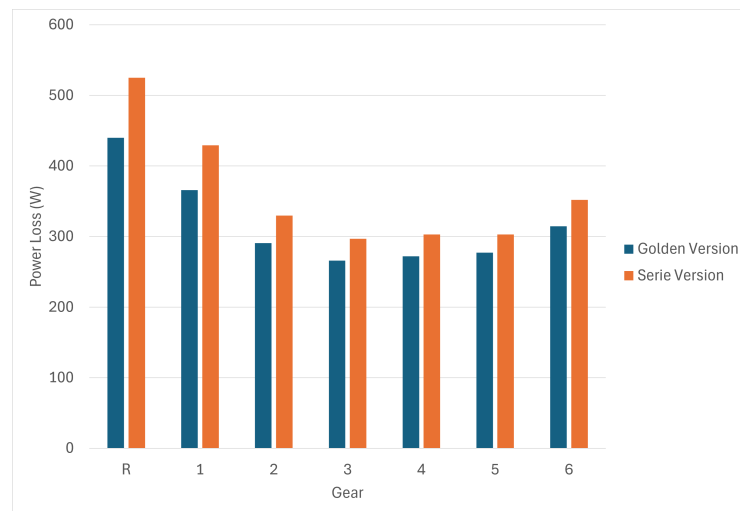
Furthermore, the breakdown of power losses reveals that the load-independent losses remain unchanged compared to the Golden version. This result was to be expected because these losses depend essentially on the rotation speed, lubricant properties, and macrogeometry of the immersed gears, characteristics that are identical in the standard and Golden gearboxes.

Conversely, a distinct difference is observed regarding the load-dependent losses. The higher surface roughness of the standard gears—a parameter that is directly integrated into the calculation of the average friction coefficient (Equation (2.12))—significantly increases gear mesh friction losses. This effect is particularly pronounced in the lower gears, where high gear ratios generate elevated internal torques and severe contact pressures.

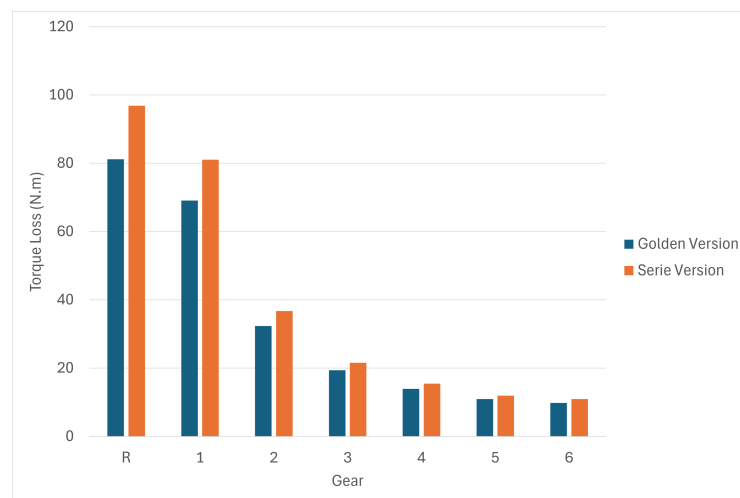
The charts presented in Figure 6.14 illustrate a direct comparative analysis between the Standard and Golden gearboxes regarding overall efficiency, power loss, and torque loss.



(a) Efficiency comparison across different gears.



(b) Power loss comparison across different gears.



(c) Torque loss comparison across different gears.

Figure 6.14: Direct comparison of efficiency, power loss, and torque loss between the Standard and Golden gearboxes.

As previously analyzed, the standard gearbox registers systematically higher power losses across the entire operating spectrum. This behavior is attributed to the higher surface roughness of the standard gears, which elevates the average friction coefficient and, consequently, amplifies the load-dependent gear mesh losses. In the lower gears, this effect is more noticeable. In the higher gears, as the internal torque decreases and the load-independent losses (churning) gain relevance - these being equal in both versions -, the absolute difference in dissipated power between the standard and Golden gearboxes becomes less pronounced, although the standard gearbox continues to be the most penalized.

The impact of this increase in power loss is reflected in the torque loss, with the greatest difference being verified in the lower gears. Due to the kinematics inherent to higher gears, this difference becomes less pronounced, although the standard gearbox consistently exhibits higher torque loss.

These facts culminate in the efficiency comparison graph that proves the energetic superiority of the Golden version in all gears. The standard gearbox presents a visibly lower efficiency, with the greatest percentage decreases occurring in the lower gears. These results validate the fundamental premise of the study: the improvement of gear tooth quality assumes a determining factor for the maximization of the overall transmission efficiency.

To compare the performance of the gearbox versions, Table 6.6 details the absolute and relative differences in power loss, alongside the variations in efficiency of the Golden version compared to the standard version.

Gear	Power Loss Diff. (W)	Power Loss Diff. (%)	Efficiency Diff. (%)
R	-85.0	-16.20	1.27
1	-63.5	-14.79	0.95
2	-38.9	-11.80	0.58
3	-30.7	-10.35	0.46
4	-30.7	-10.14	0.46
5	-25.8	-8.52	0.39
6	-37.5	-10.66	0.56

Table 6.6: Differences in power loss and efficiency between Golden and Serie versions.

Chapter 7

Conclusion and Future Work

7.1 Conclusions

The present work had as its main objective to develop and validate a computational model to estimate and analyze the efficiency of the JT4 manual gearbox, manufactured at HORSE Aveiro, as well as to study the effect of tooth profile quality on efficiency. Despite all the constraints faced throughout the dissertation, the objective was successfully achieved.

With the realization of this work, the following conclusions stand out:

- **Test bench adaptation:** It was possible to design, manufacture, and assemble all necessary components on the test bench. Pending minor adjustments, this infrastructure is now fully prepared for future testing on the JT4 gearbox.
- **KISSsoft model validation:** Due to the problem of access to the test bench located at FEUP, the NVH tests performed at HORSE were used for model validation. The model created in KISSdesign, a KISSsoft module, revealed an excellent capacity in predicting the behavior of the gearbox, accurately capturing the power and torque losses, with acceptable error margins.
- **Isolation of Power Loss Sources:** The computational model had the capacity to estimate the contribution of the different components to the overall power loss. Analyzing the results of the model it is concluded that the gear mesh friction losses (load-dependent) are dominant in the lower gears representing more than 70% of the total losses, due to the high internal torques characteristic of these gears. Conversely, in higher gears, elevated rotational speeds cause churning losses (load-independent) to increase exponentially. These become the primary limiting factor for efficiency (reaching approximately 40%) and shift the location of the optimal operating point within the speed and torque spectrum.
- **Reverse gear performance:** Although the NVH tests performed at HORSE do not include the reverse gear, the model predicted that this is the gear with the lowest efficiency (with a minimum of 91.84%). The results revealed that, while its gear ratio is similar to that of

the first gear, the requirement for power to be transmitted through an additional shaft (the reverse shaft) introduces significant supplementary losses in both the bearings and the gear mesh.

- **Theoretical Efficiency Comparison:** Although experimental tests were not conducted on the mass-produced gearbox, the numerical model successfully demonstrated that the Golden version yields a higher overall efficiency. This confirms that the enhanced surface finish achieved through gear grinding is a critical factor in mitigating power losses compared to the standard production process.
- **Emissions and Fuel Consumption Reduction:** The Golden version demonstrated higher efficiency than the standard version, resulting in less energy dissipated within the gearbox. Consequently, this optimization leads to lower fuel consumption and a subsequent decrease in emissions. This improvement is particularly significant in the lower gear ratios, where the greatest performance difference between the two versions was recorded. As a result, these benefits are expected to be most noticeable in urban driving environments, where lower gears are predominantly used.

7.2 Future Work

Despite the promising results achieved with the KISSsoft model in the characterization of the power losses of the JT4 gearbox, the limitations encountered open doors for several points of future research. The following topics are suggested:

- **Testing campaign on the FEUP test rig:** Take advantage of the setup designed and assembled on the test bench specifically for the JT4 gearbox, allowing the measurement of total and no-load power losses in a wider spectrum of speeds and torques than that performed in the NVH tests, as well as tests with the reverse gear engaged in order to validate the KISSsoft model for this gear. During preliminary trials on the FEUP test bench, the entire custom-designed system functioned properly; however, it was not possible to apply the target torque. The torque reaches a specific threshold and then plateaus. It was observed that the hydraulic actuator responsible for applying the torque experiences undesired movement when a test is initiated, which was identified as a primary cause for the inability to reach the desired torque levels.
- **Direct Efficiency Comparison (Standard vs. Golden):** Perform experimental tests on the Standard gearbox to validate the numerical model that has already been developed for this version, in order to accurately compare its efficiency with the Golden gearbox.
- **High-speed model optimization:** Computational models often struggle to accurately estimate churning losses due to their inherent complexity. Future research should focus on refining specific parameters and utilizing no-load experimental test bench data to enhance the model's capability to predict load-independent losses.

- **Investigation of lubricant effects:** As previously noted, churning losses emerge as a limiting factor of efficiency at high speeds. For this reason, it would be interesting to perform tests utilizing lubricants with different viscosity grades and different oil levels within the gearbox, thereby evaluating the impact of these parameters on the efficiency of the gearbox.
- **Study of the Impact of Phosphating Elimination:** Perform comparative experimental tests using versions of gearboxes where the gears have not been subjected to the phosphating process (a surface treatment that is currently standard on the production line). The main objective would be to evaluate the direct impact of the removal of this coating on friction, on the global efficiency of the gearbox and on its useful life. The technical validation of this alteration would represent a huge strategic added value for HORSE Aveiro, offering substantial benefits both at an economic level (reduction of operational costs and cycle time) as well as at an environmental level (decrease of the ecological footprint and elimination of chemical waste).

References

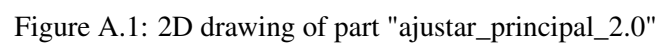
- [1] *Horse Aveiro - Fábrica automóvel do Grupo - Renault*. (Visited on 09/11/2026).
- [2] Renault Group. *HORSE: A nova empresa líder em tecnologia de grupos motopropulsores, está agora totalmente operacional e pronta para reinventar a combustão*. 2023. (Visited on 07/03/2023).
- [3] Pedro Miguel Teixeira Marques. “Power Loss in Planetary Gearboxes Including the Influence of Gear Elastic and Dynamic Effects”. PhD thesis. Porto: Faculdade de Engenharia da Universidade do Porto, 2017.
- [4] Carlos André Ferreira Pereira. “Experimental torque loss in a vehicle transfer gearbox”. MA thesis. Porto: Faculdade de Engenharia da Universidade do Porto, 2017.
- [5] Bernd-Robert Höhn, Klaus Michaelis, and Michael Hinterstoißer. “Optimization of Gearbox Efficiency”. In: *Goriva i maziva* 48.4 (2009), pp. 441–480. ISSN: 0350-350X.
- [6] C. Changenet et al. “A Note on Flow Regimes and Churning Loss Modeling”. In: *Journal of Mechanical Design* 133.12 (Dec. 1, 2011), p. 121009. ISSN: 1050-0472, 1528-9001.
- [7] Gauthier Leprince. “Pertes mécaniques par frottement et lubrification dans une boîte de vitesses”. NNT: 2011ISAL0151. PhD thesis. INSA de Lyon, 2011.
- [8] Pedro M.T. Marques et al. “Efficiency of a Gearbox Lubricated with Wind Turbine Gear Oils”. In: *Tribology International* 71 (Mar. 2014), pp. 7–16. ISSN: 0301679X.
- [9] C. Changenet and P. Velex. “A Model for the Prediction of Churning Losses in Geared Transmissions—Preliminary Results”. In: *Journal of Mechanical Design* 129.1 (Jan. 1, 2007), pp. 128–133. ISSN: 1050-0472, 1528-9001.
- [10] H Ohlendorf. “Verlustleistung Und Erwärmung von Stirnrädern. München, Technische Hochschule”. PhD thesis. doctoral thesis, 1958.
- [11] Klüber Lubrication. *Lubrication of Gear Systems*. Nov. 2003.
- [12] Nadir Patir and H. S. Cheng. “An Average Flow Model for Determining Effects of Three-Dimensional Roughness on Partial Hydrodynamic Lubrication”. In: *Journal of Lubrication Technology* 100.1 (Jan. 1, 1978), pp. 12–17. ISSN: 0022-2305.
- [13] L Schlenk. “Untersuchungen zur Fresstragfähigkeit von Grozahnradern”. PhD thesis. Dissertation, TU Munchen, Germany., 1994.
- [14] Bernd-Robert Höhn, Klaus Michaelis, and Albert Wimmer. “Low Loss Gears”. In: *Gear Technology* (June 2007), pp. 28–35.
- [15] Carlos M.C.G. Fernandes et al. “Gearbox Power Loss. Part I: Losses in Rolling Bearings”. In: *Tribology International* 88 (Aug. 2015), pp. 298–308. ISSN: 0301679X.
- [16] SKF. *General Catalogue*. 6000/I EN. SKF Group, June 2008.

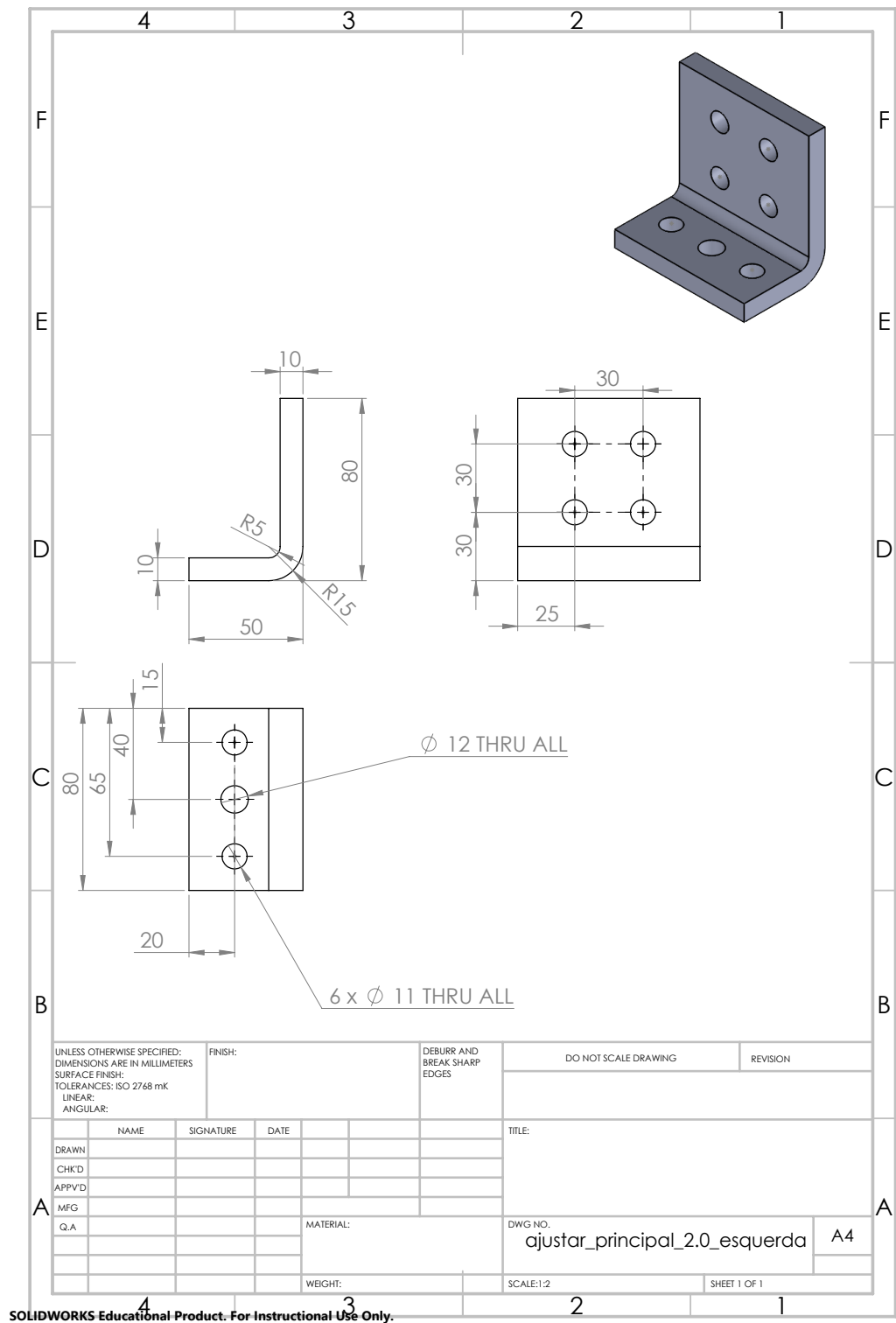
- [17] Freudenberg Sealing Technologies. *Simrit Radial Shaft Seal Technical Manual*. manual. Weinheim, Germany: Freudenberg Sealing Technologies, 2007.
- [18] Renault Automobiles. *Dentures En Développante de Cercle : Définitions - Vérifications*. Norme 01-33-001/-G. Régie Nationale des Usines Renault S.A., May 1996.
- [19] Kapil Gupta, Neelesh Kumar Jain, and Rudolph Laubscher. “Measurement of Gear Accuracy”. In: *Advanced Gear Manufacturing and Finishing*. Elsevier, 2017, pp. 197–218. ISBN: 978-0-12-804460-5.
- [20] Luciano Cianciotta, Marco Cirelli, and Pier Paolo Valentini. “Multi-Objective Optimization of Gear Design of E-Axles to Improve Noise Emission and Load Distribution”. In: *Machines* 13.4 (Apr. 17, 2025), p. 330. ISSN: 2075-1702.
- [21] Ulrich Kissling. “Layout of the Gear Micro Geometry”. In: *Gear Solutions* (Sept. 2015), pp. 51–63.
- [22] Kun Yue et al. “Study on Gear Meshing Power Loss Calculation Considering the Coupling Effect of Friction and Dynamic Characteristics”. In: *Tribology International* 183 (May 2023), p. 108378. ISSN: 0301679X.
- [23] Stephen P. Radzevich. *Dudley’s Handbook of Practical Gear Design and Manufacture*. 2nd ed. Boca Raton, FL: CRC Press, 2012.
- [24] Jan Troge et al. “Acoustical Optimization of a Train Gearbox Based on Overall System Simulation”. In: *Proceedings of the 45th International Congress and Exposition on Noise Control Engineering (INTER-NOISE 2016)*. German Acoustical Society (DEGA). Hamburg, Germany, Aug. 2016, pp. 1–10.
- [25] Team TREMEC. *Manual Transmission Synchronizers 101*. Jan. 2019.
- [26] TOTAL PORTUGAL PETRÓLEOS, UNIP. LDA. *TRANSELF NFX SAE 75W*. Technical Data Sheet. Rev July 2020. Mar. 2023.
- [27] Raquel Camacho Simões Dias. “Torque Loss in a Planetary Multiplier Gearbox: Influence of Operating Conditions and Gear Oil Formulation”. MA thesis. Porto, Portugal: Faculdade de Engenharia da Universidade do Porto, July 2014.

Appendix A

2D Technical Drawings of the Manufactured Parts

As discussed in Section 5.1.3, this appendix presents the technical drawings of all the manufactured parts, with the exception of certain components that cannot be disclosed due to confidentiality constraints. Given the limited documentation available regarding the test rig, these drawings are provided to facilitate any future adaptations to the test rig.





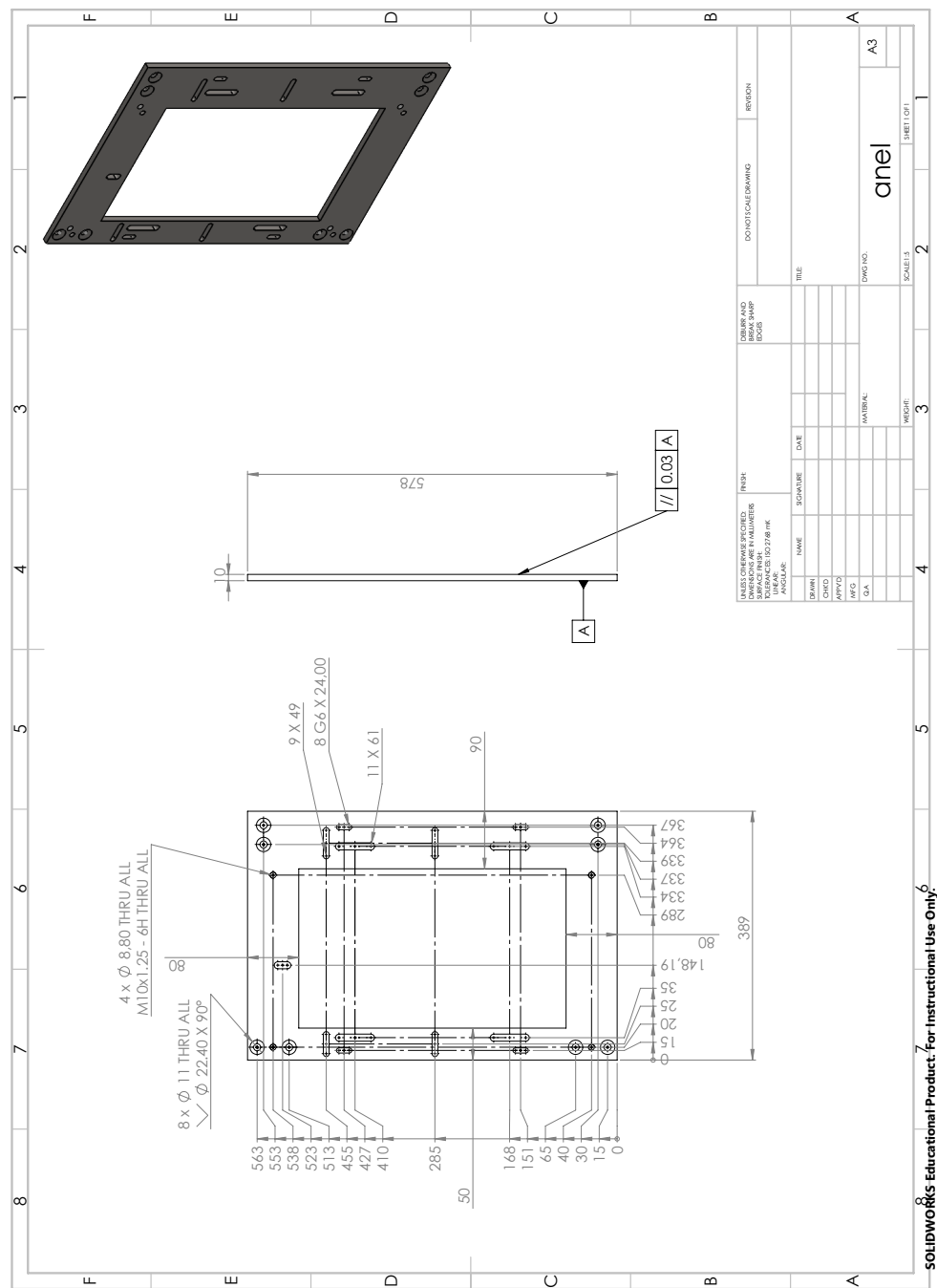
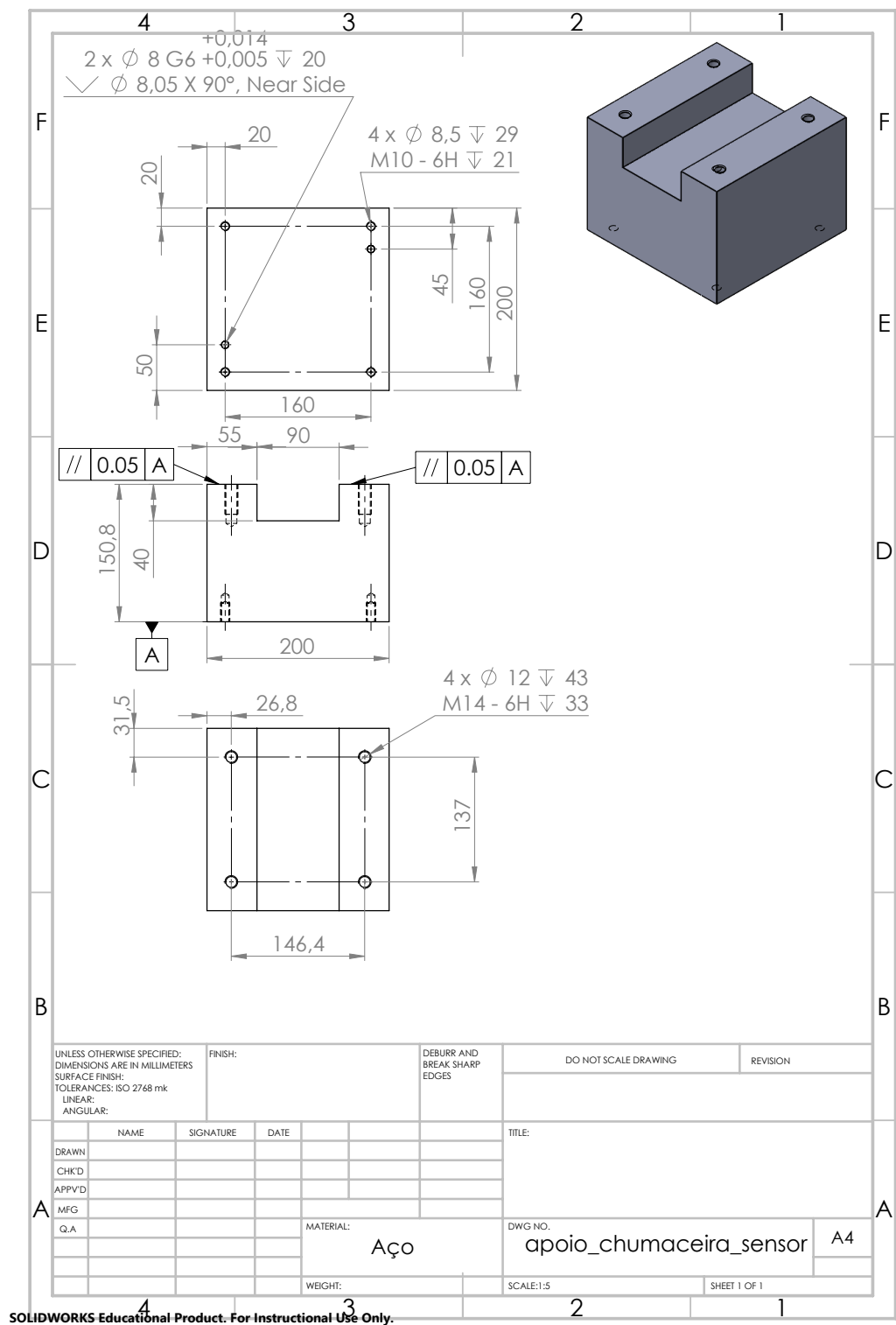


Figure A.3: 2D drawing of part "anel"



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Figure A.4: 2D drawing of part "apoio_chumaceira_sensor"

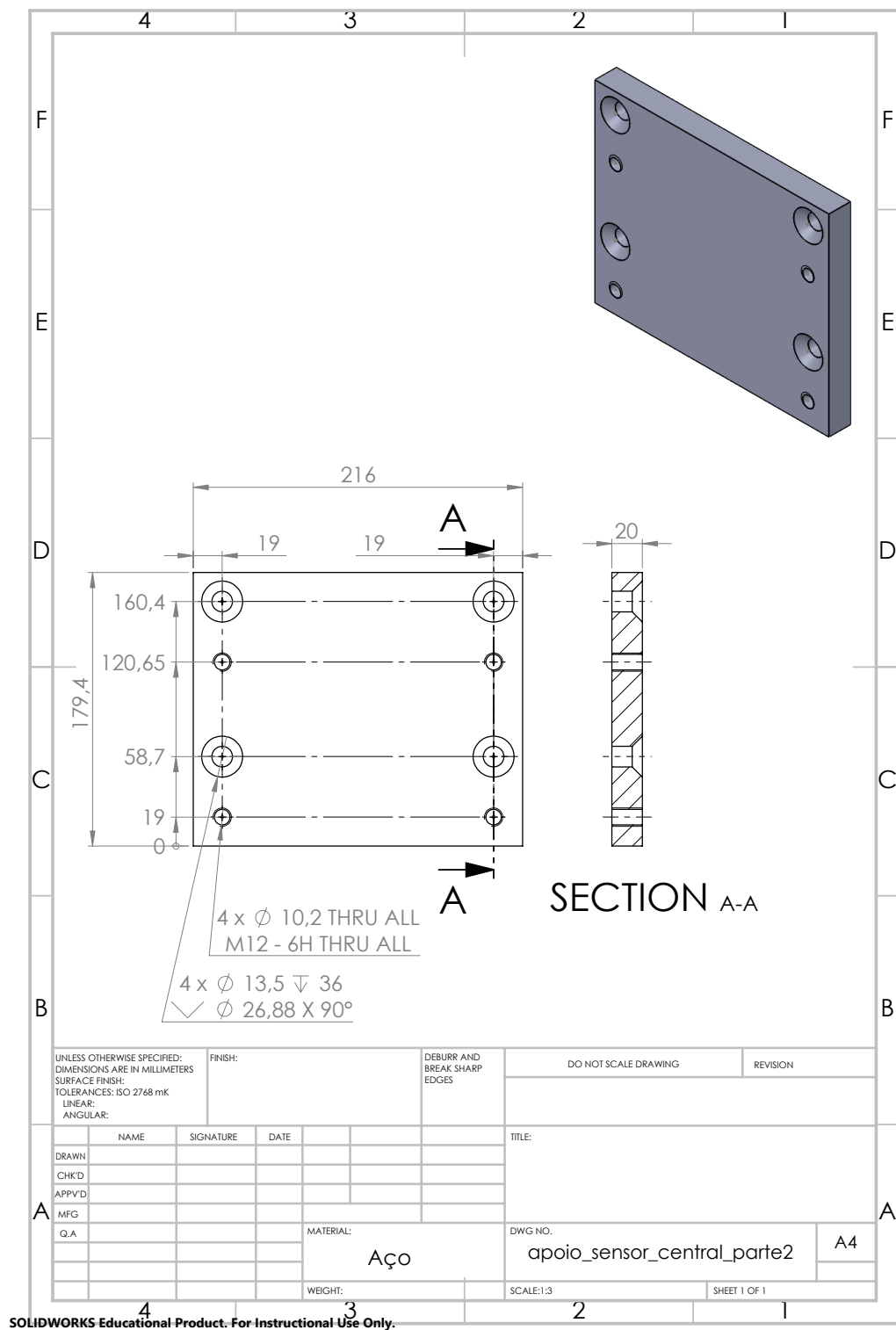


Figure A.5: 2D drawing of part "apoio_sensor_central_parte2"

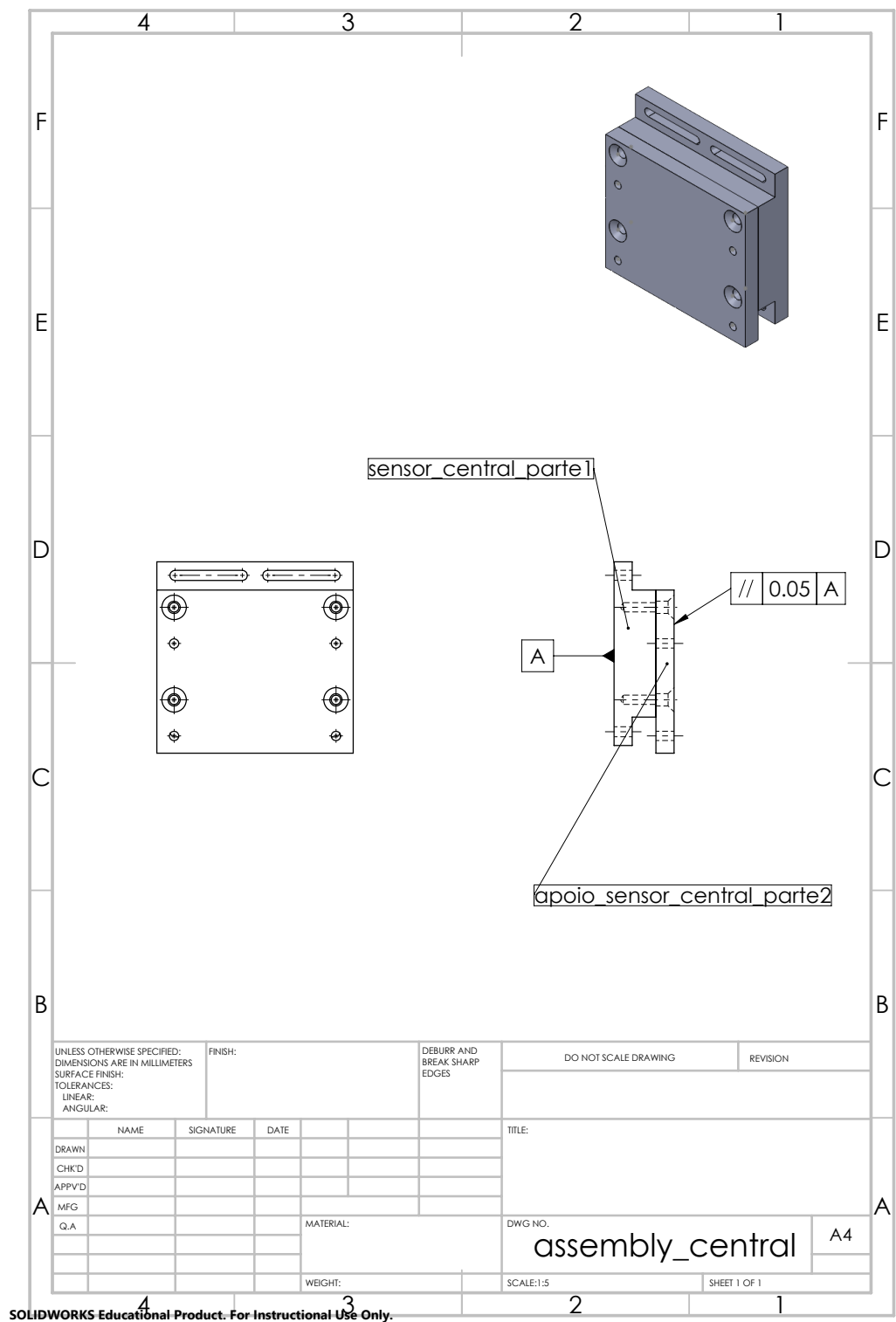


Figure A.6: 2D drawing of assembly "assembly_central"

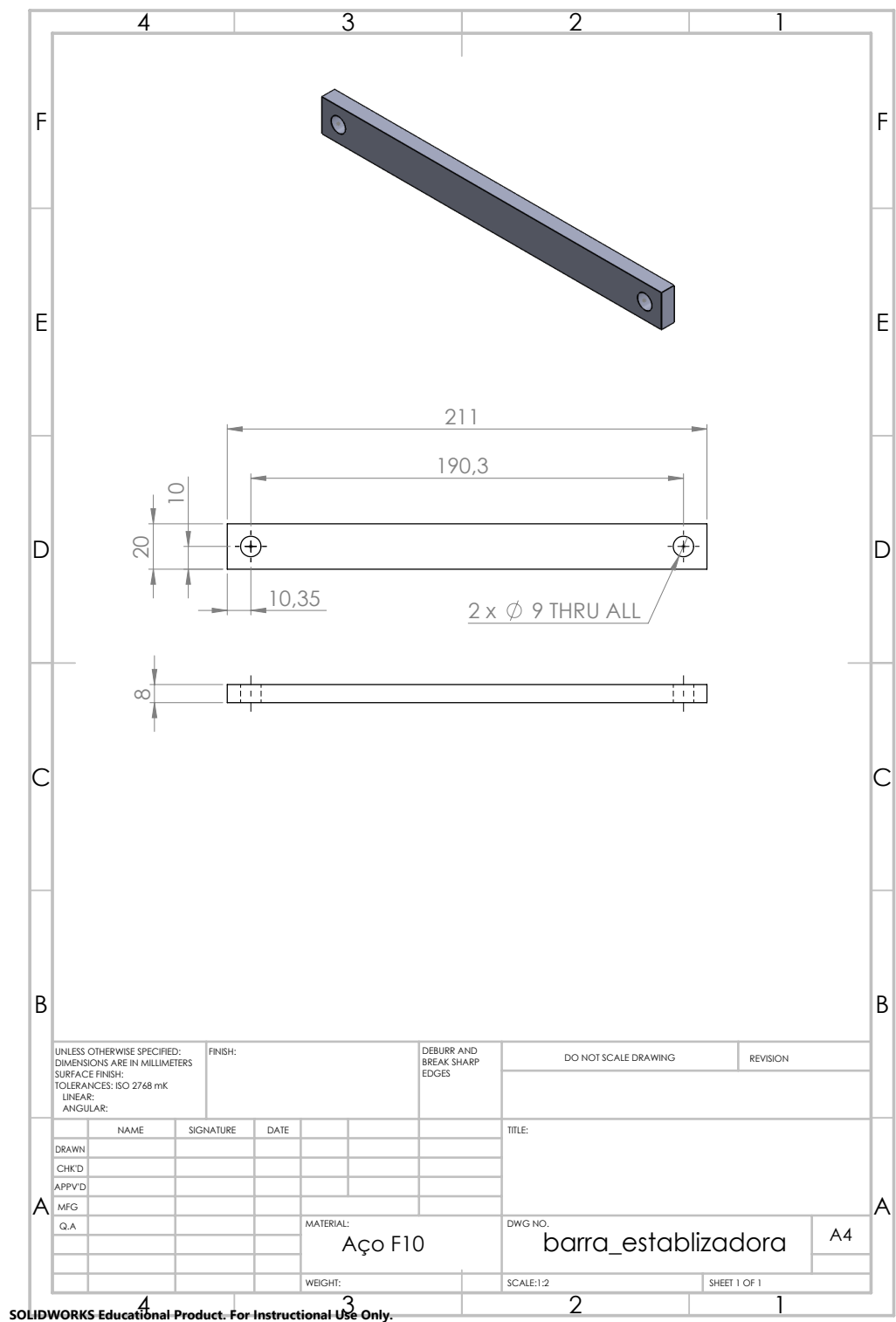


Figure A.7: 2D drawing of part "barra_estabilizadora"

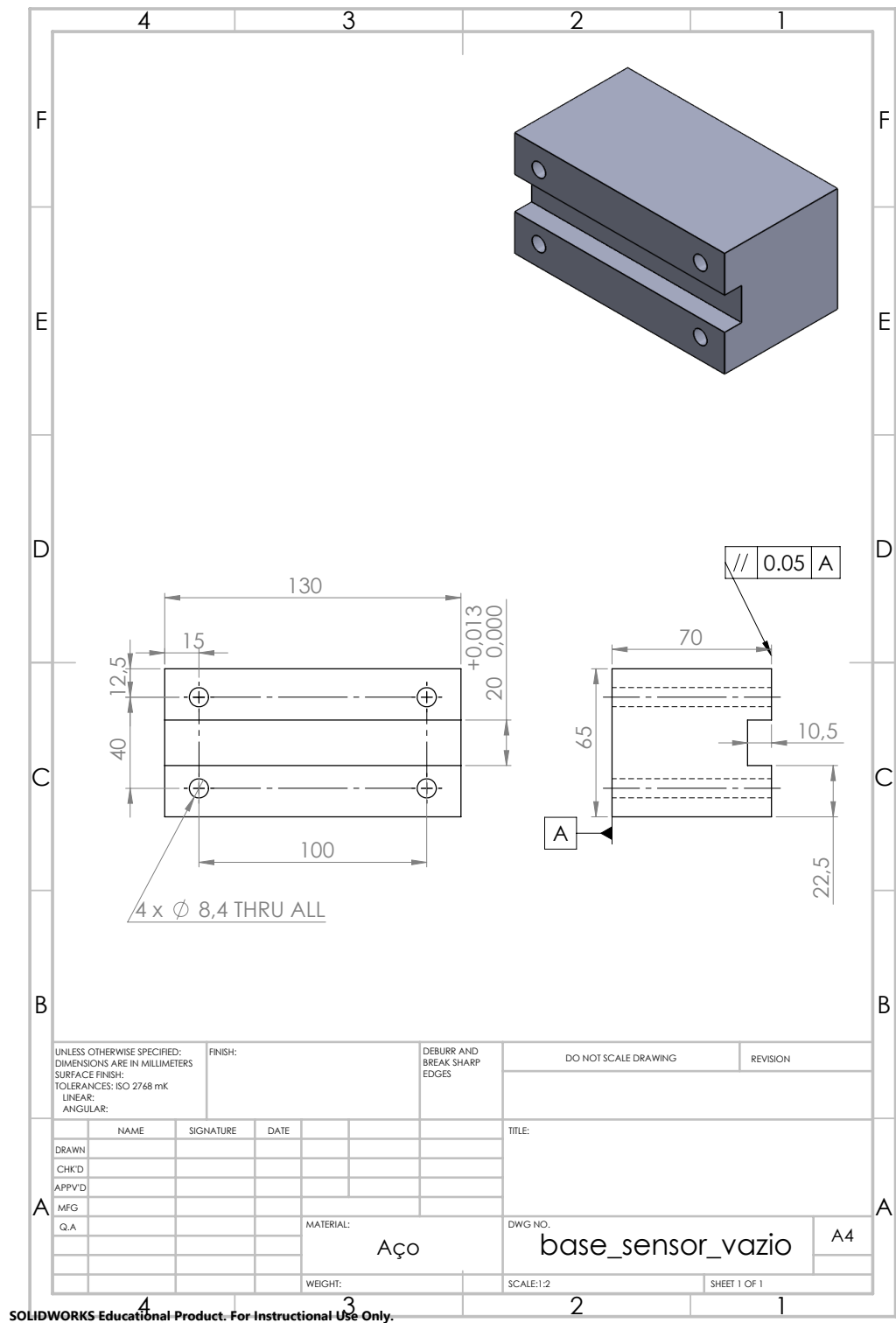


Figure A.8: 2D drawing of part "base_sensor_vazio"

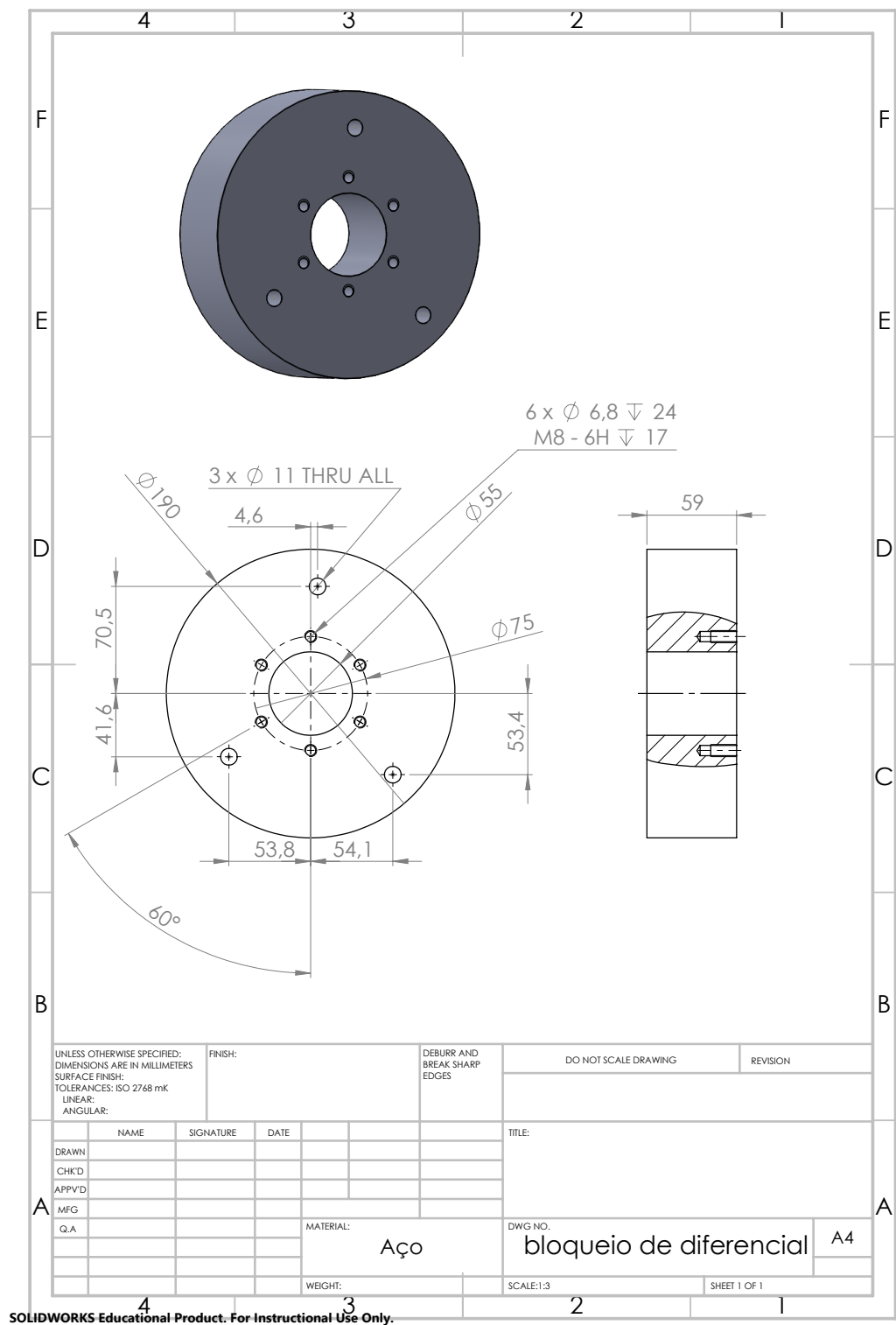


Figure A.9: 2D drawing of part "bloqueio de diferencial"

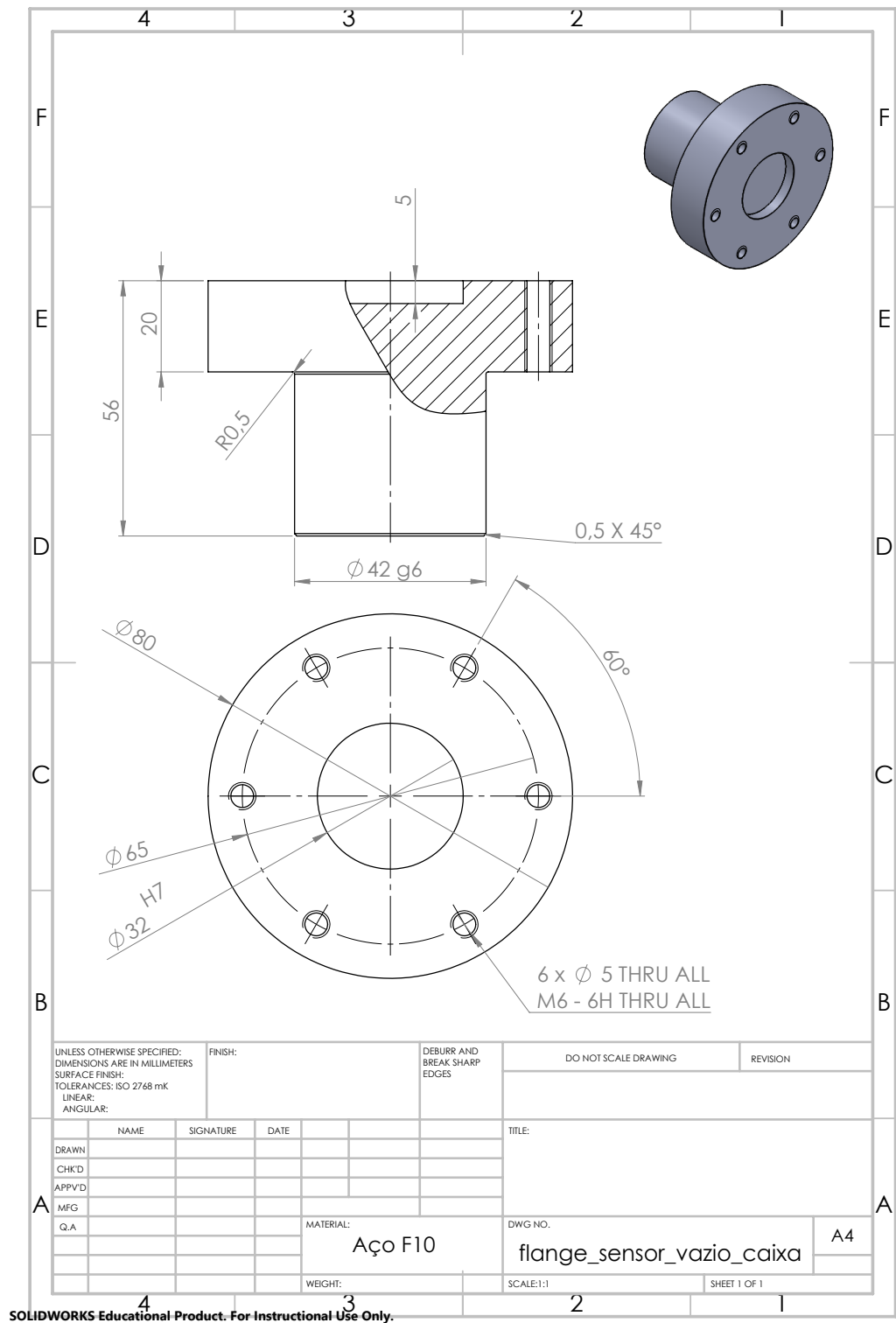


Figure A.10: 2D drawing of part "flange_sensor_vazio_caixa"

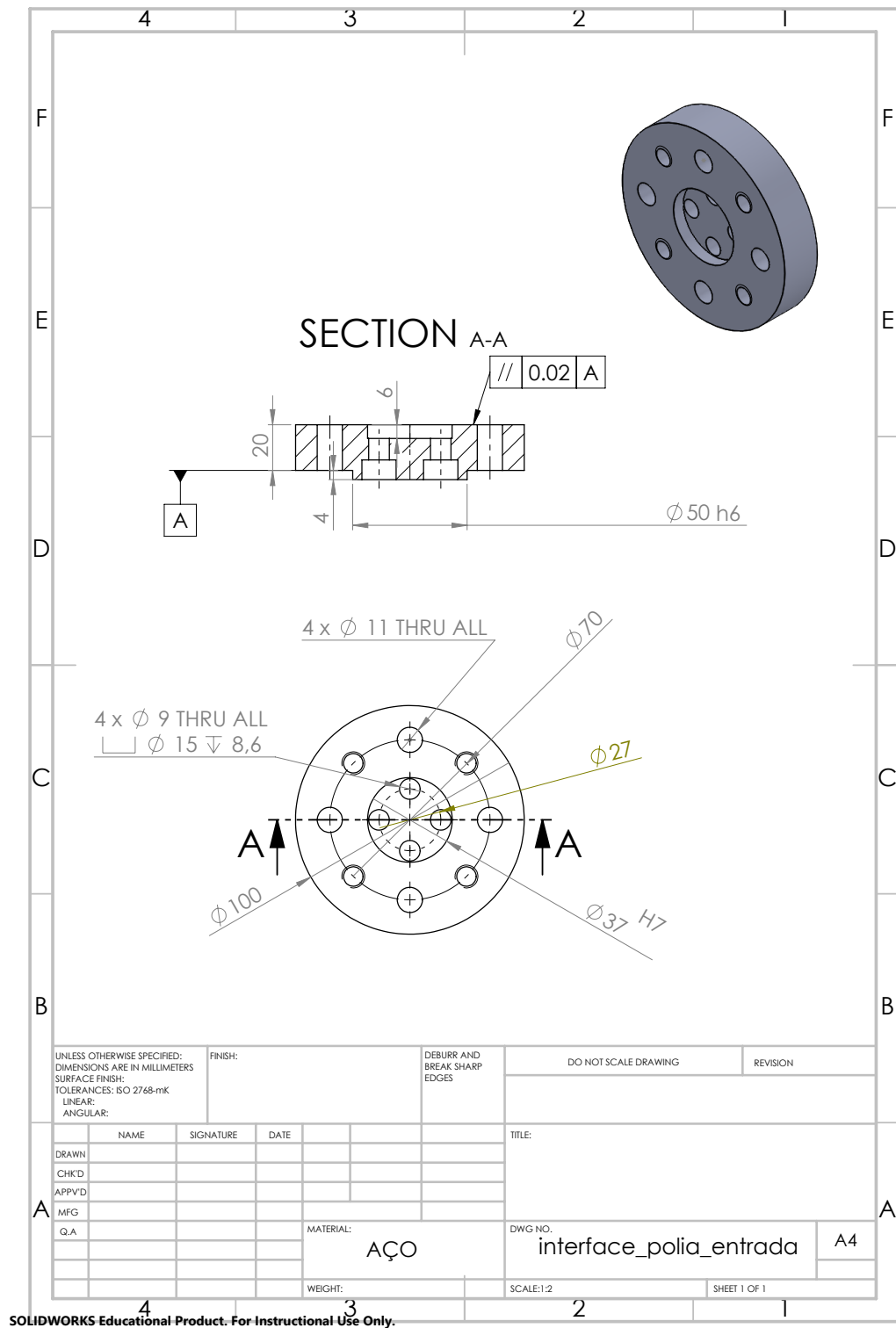
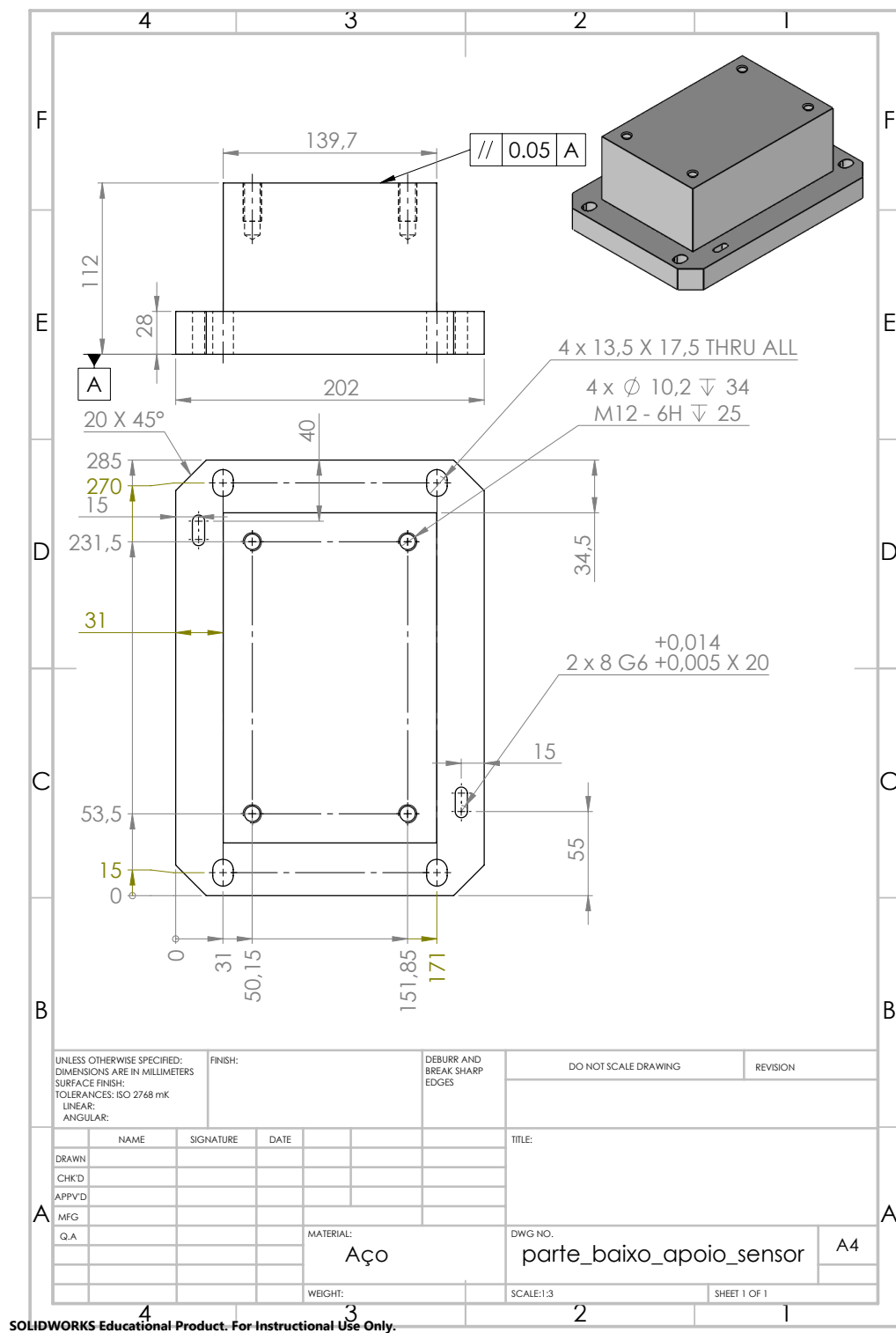
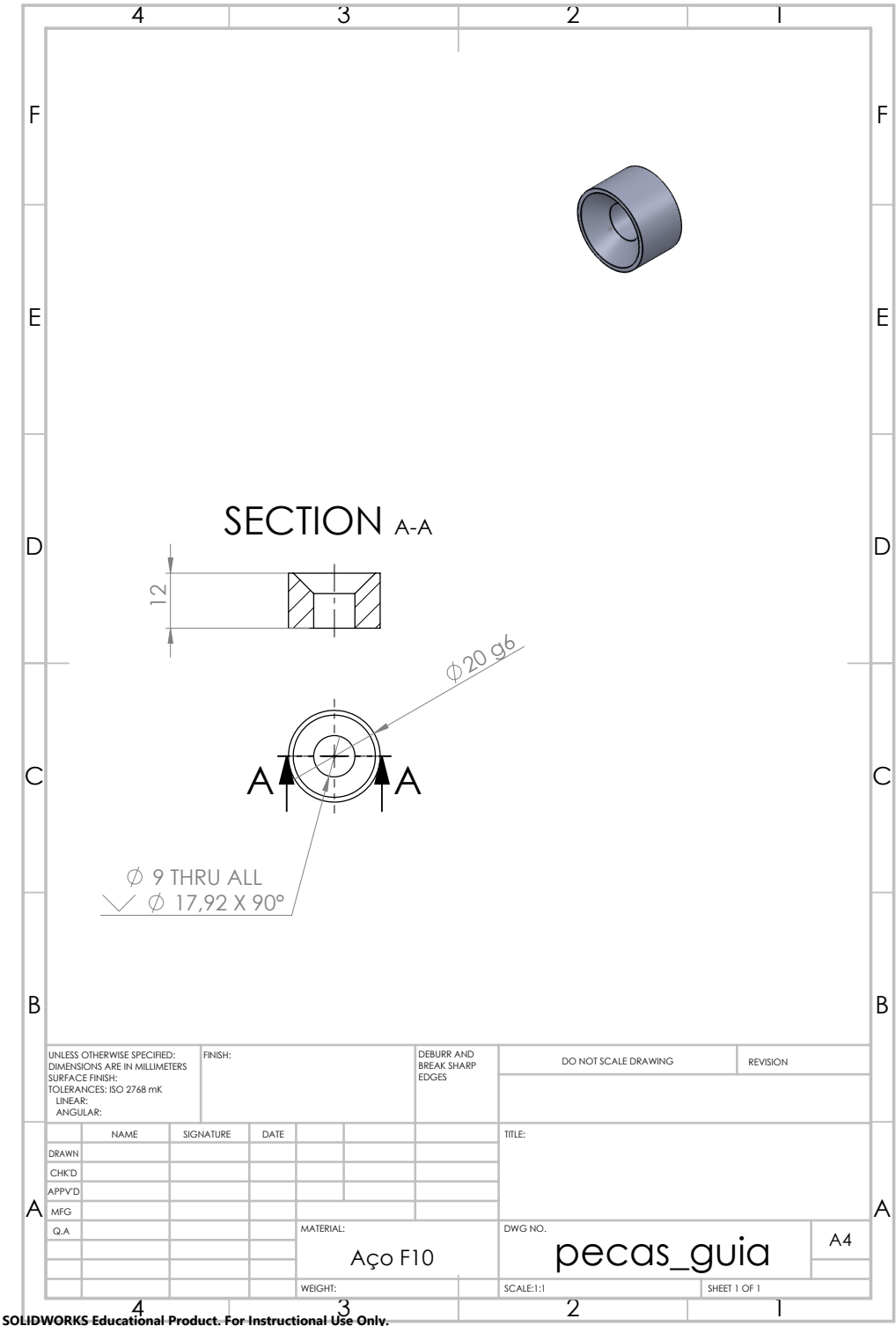


Figure A.11: 2D drawing of part "interface_polia_entrada"



Figure A.12: 2D drawing of part "interface_saida_caixa_escrava"





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Figure A.14: 2D drawing of part "peças_guia"

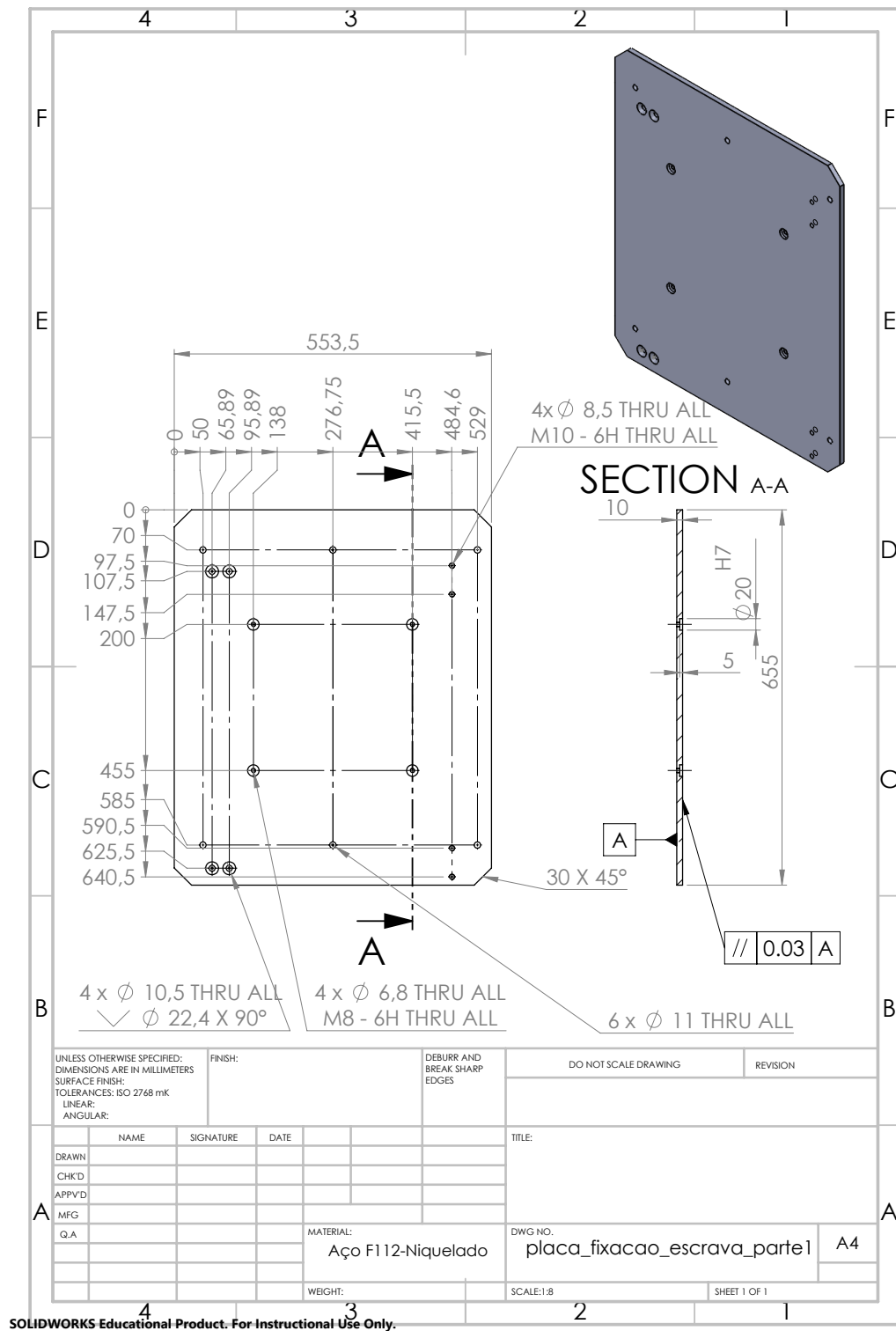


Figure A.15: 2D drawing of part "placa_fixação_escrava_parte1"

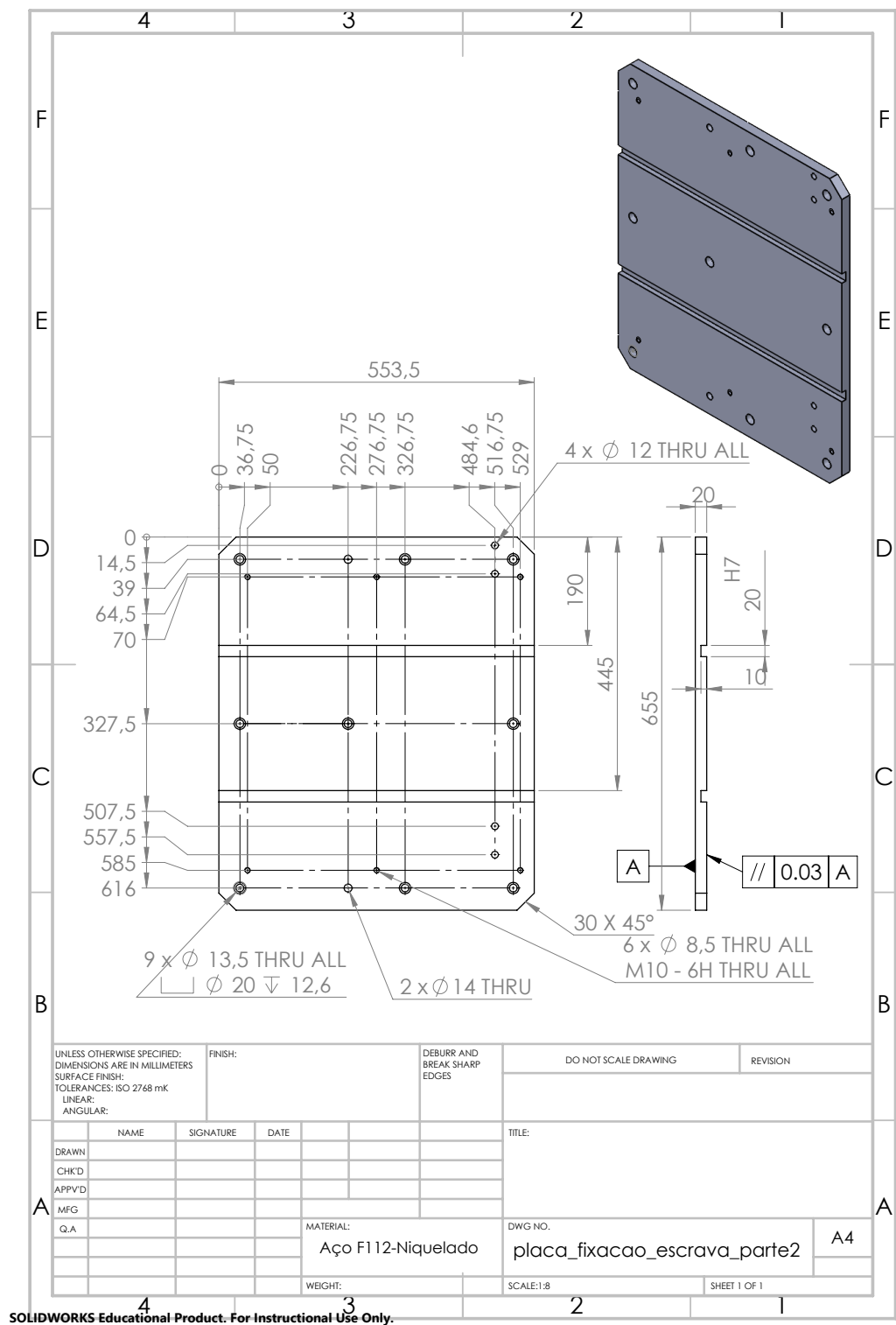
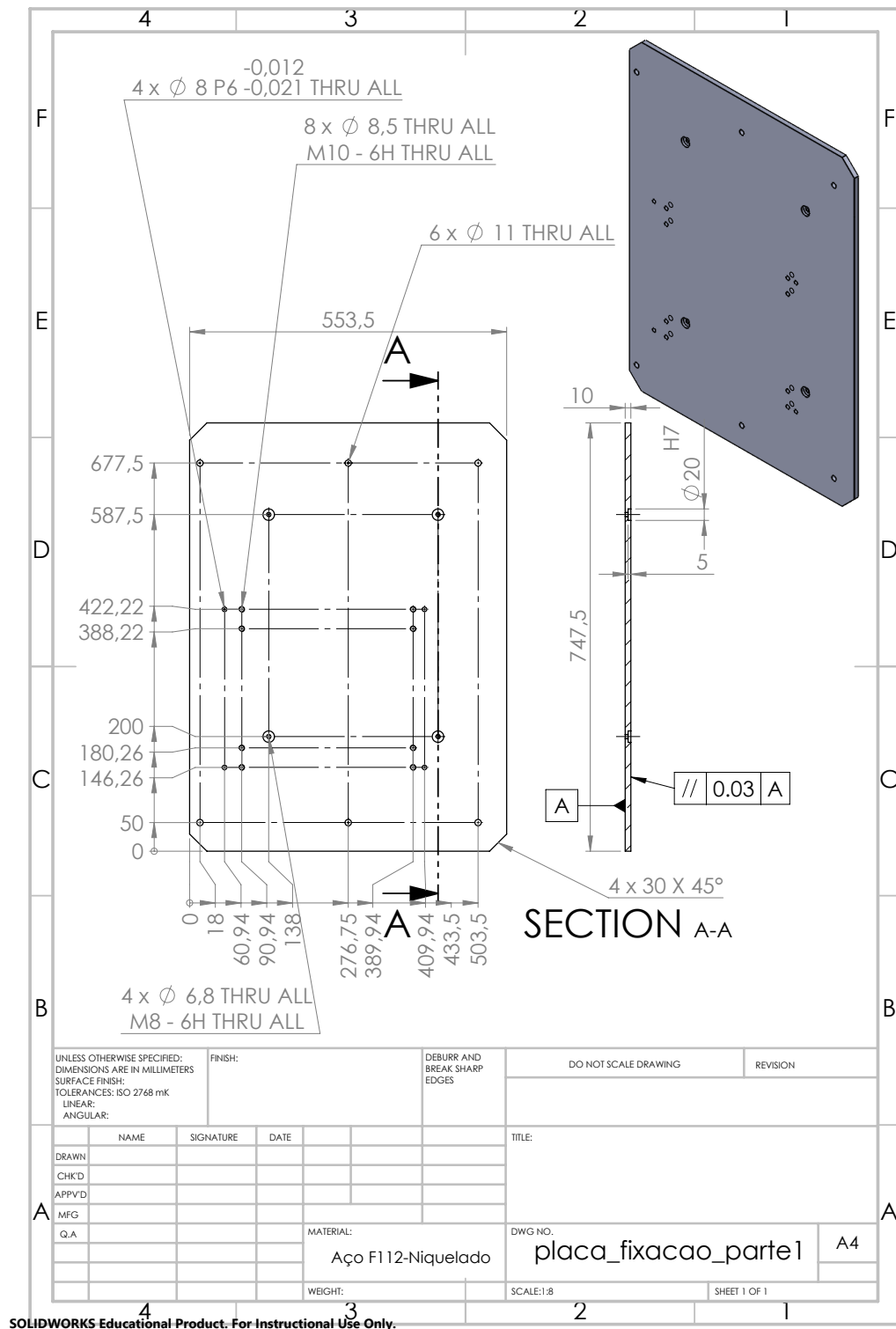


Figure A.16: 2D drawing of part "placa_fixação_escrava_parte2"



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Figure A.17: 2D drawing of part "placa_fixação_parte1"

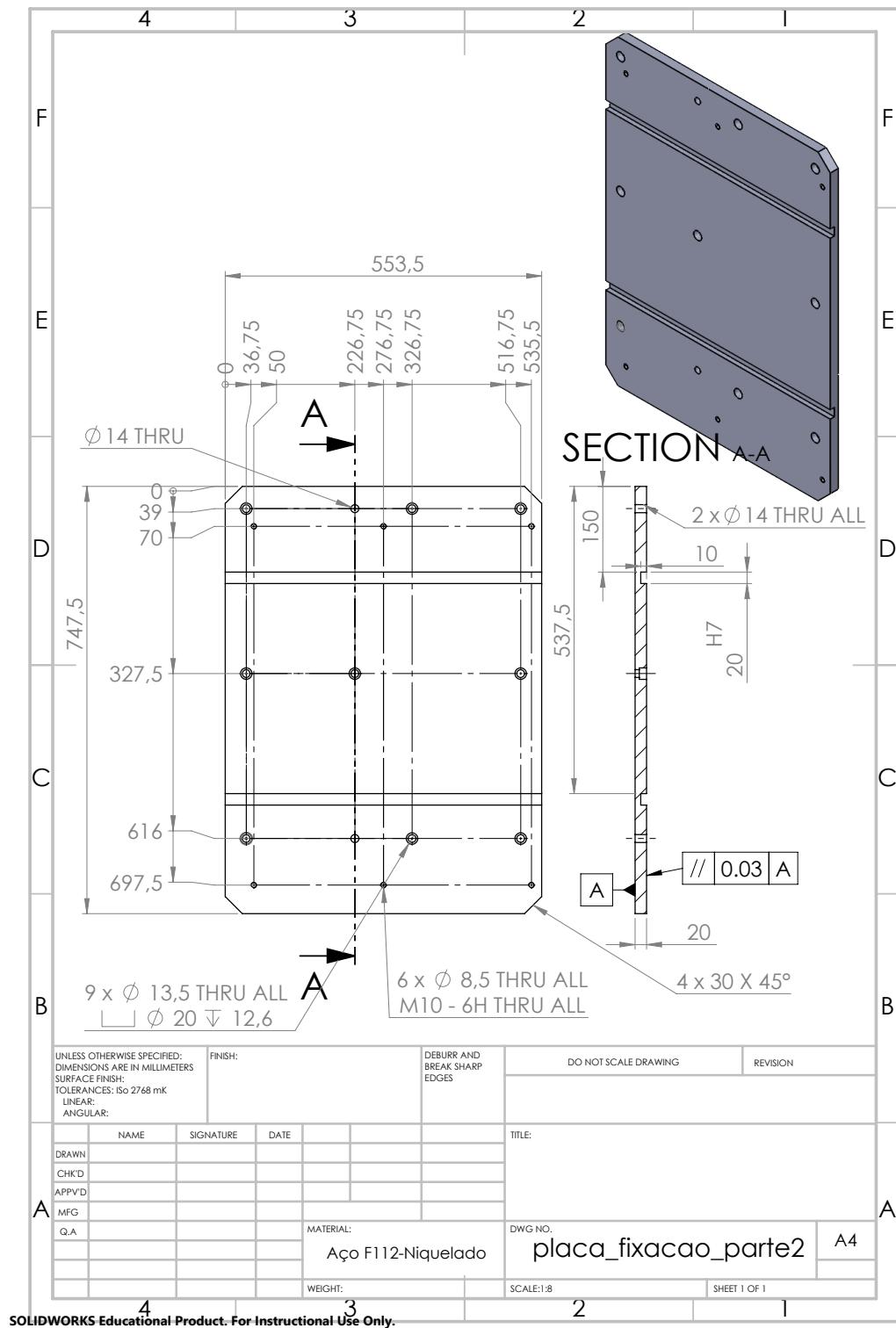
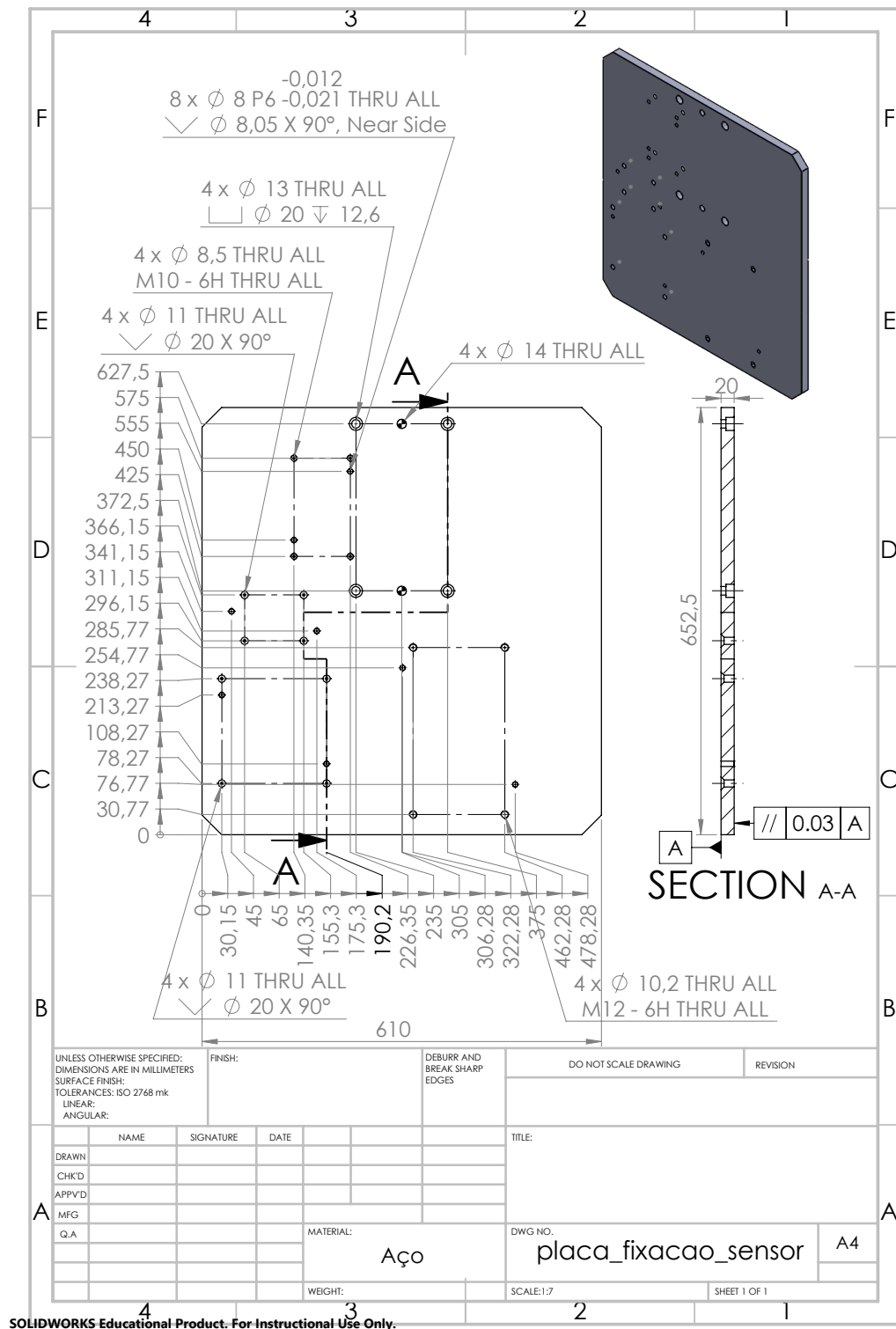


Figure A.18: 2D drawing of part "placa_fixação_parte2"



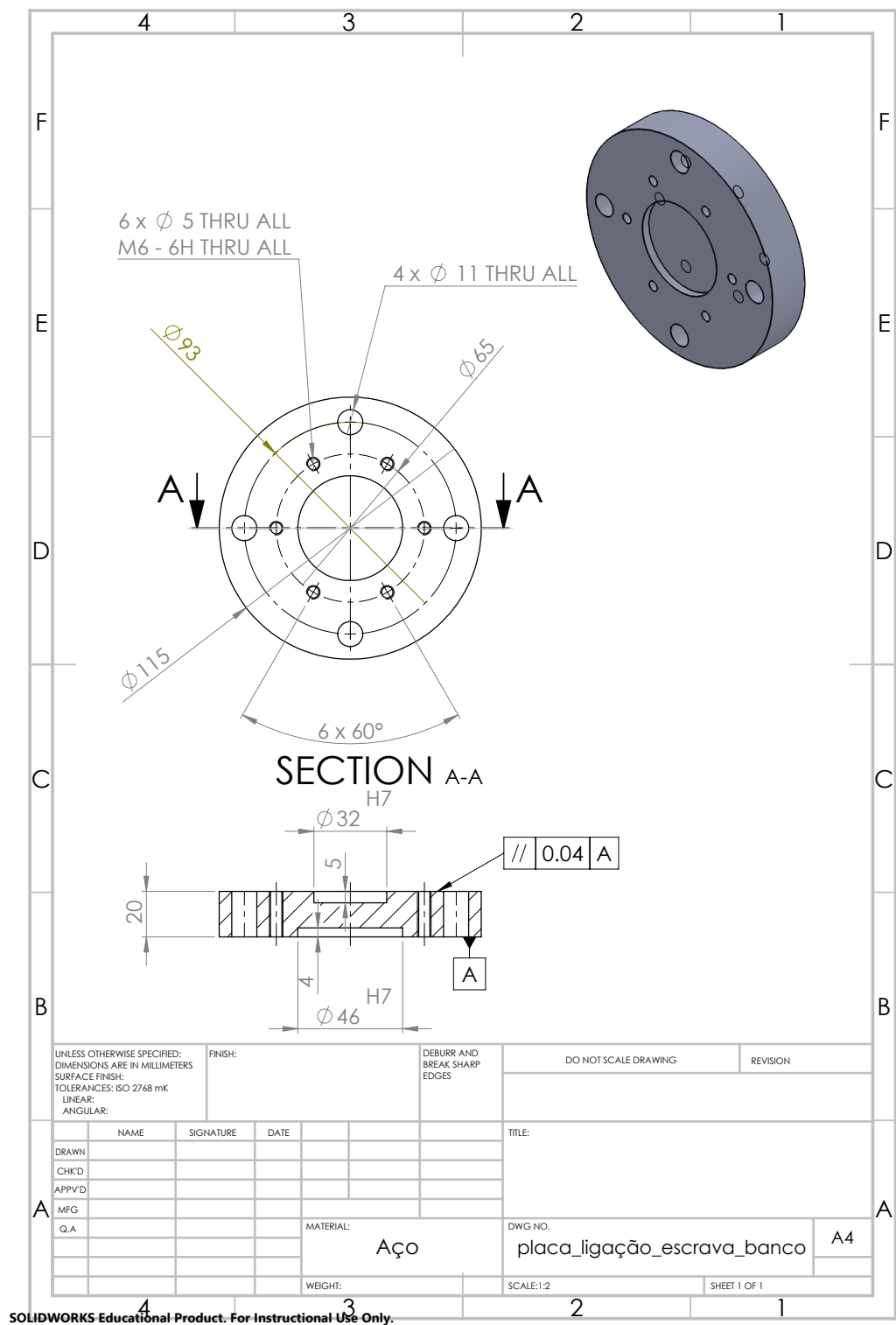


Figure A.20: 2D drawing of part "placa_ligação_banco"

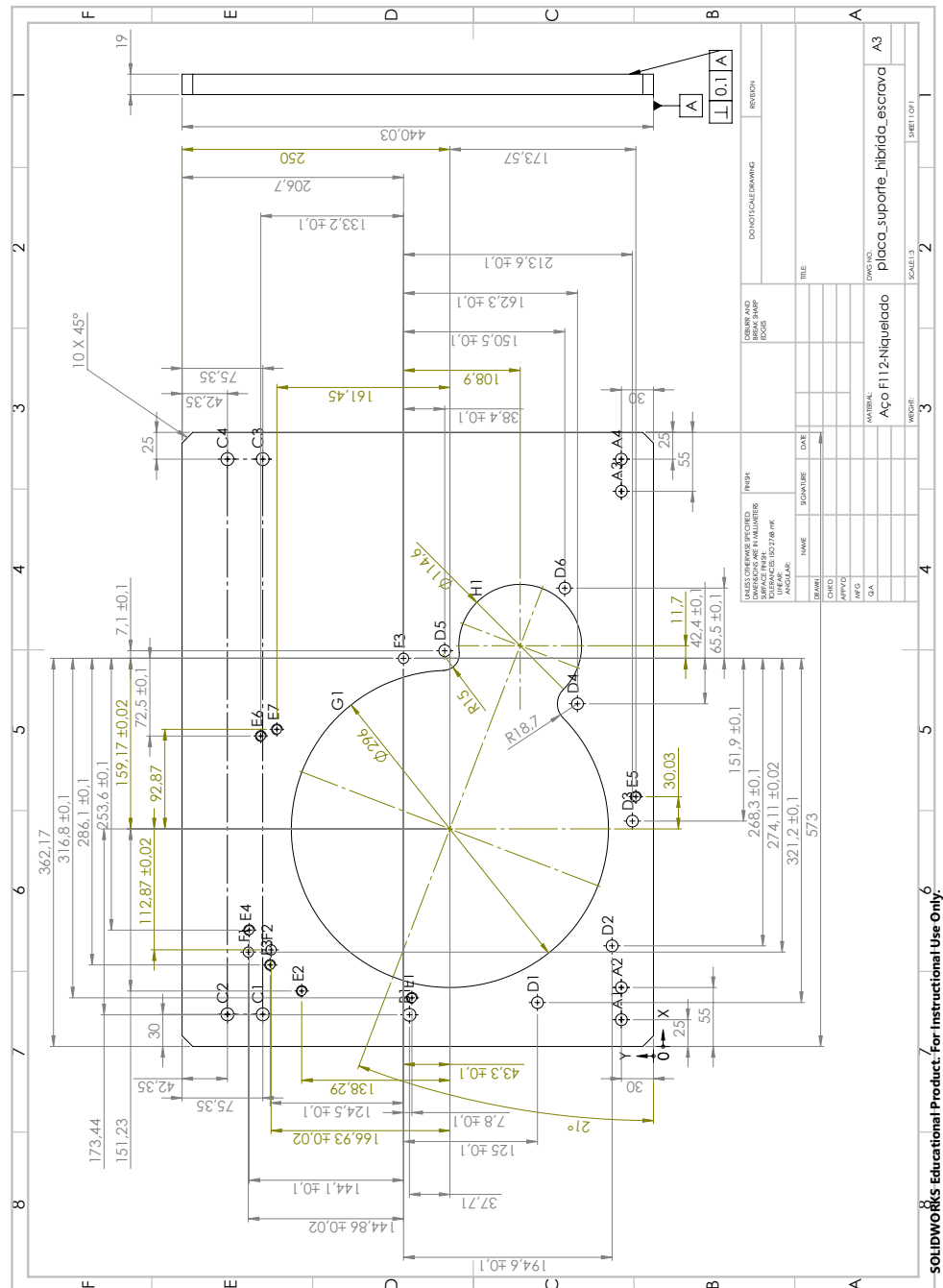


Figure A.21: 2D drawing of part "placa_suporte_hibrida_escrava"

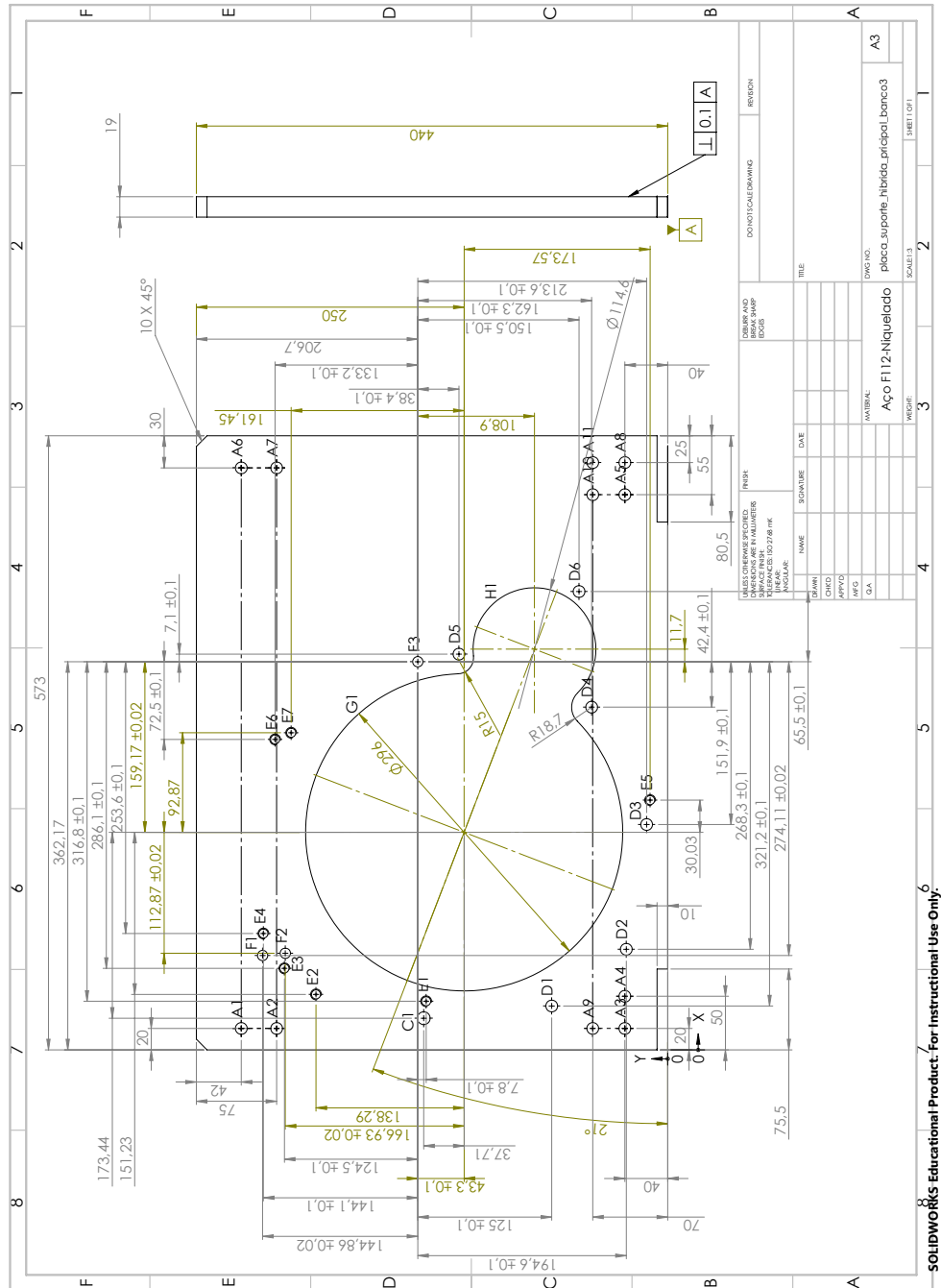
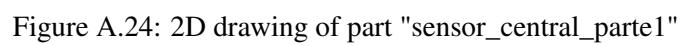


Figure A.22: 2D drawing of part "placa_suporte_hibrida_principal_banco3"



Figure A.23: 2D drawing of part "placa_tensionador"



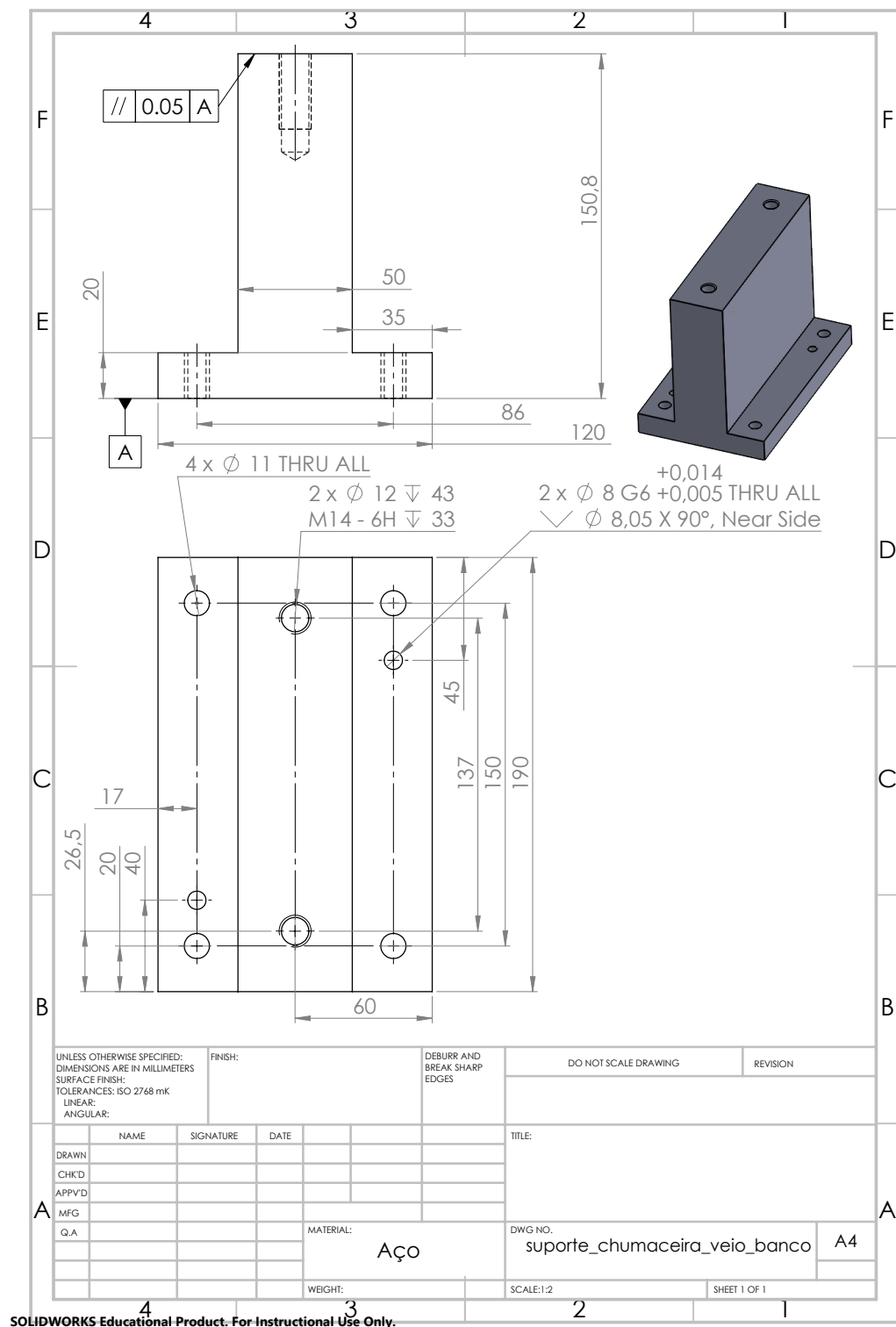


Figure A.25: 2D drawing of part "suporte_chumaceira_veio_banco"

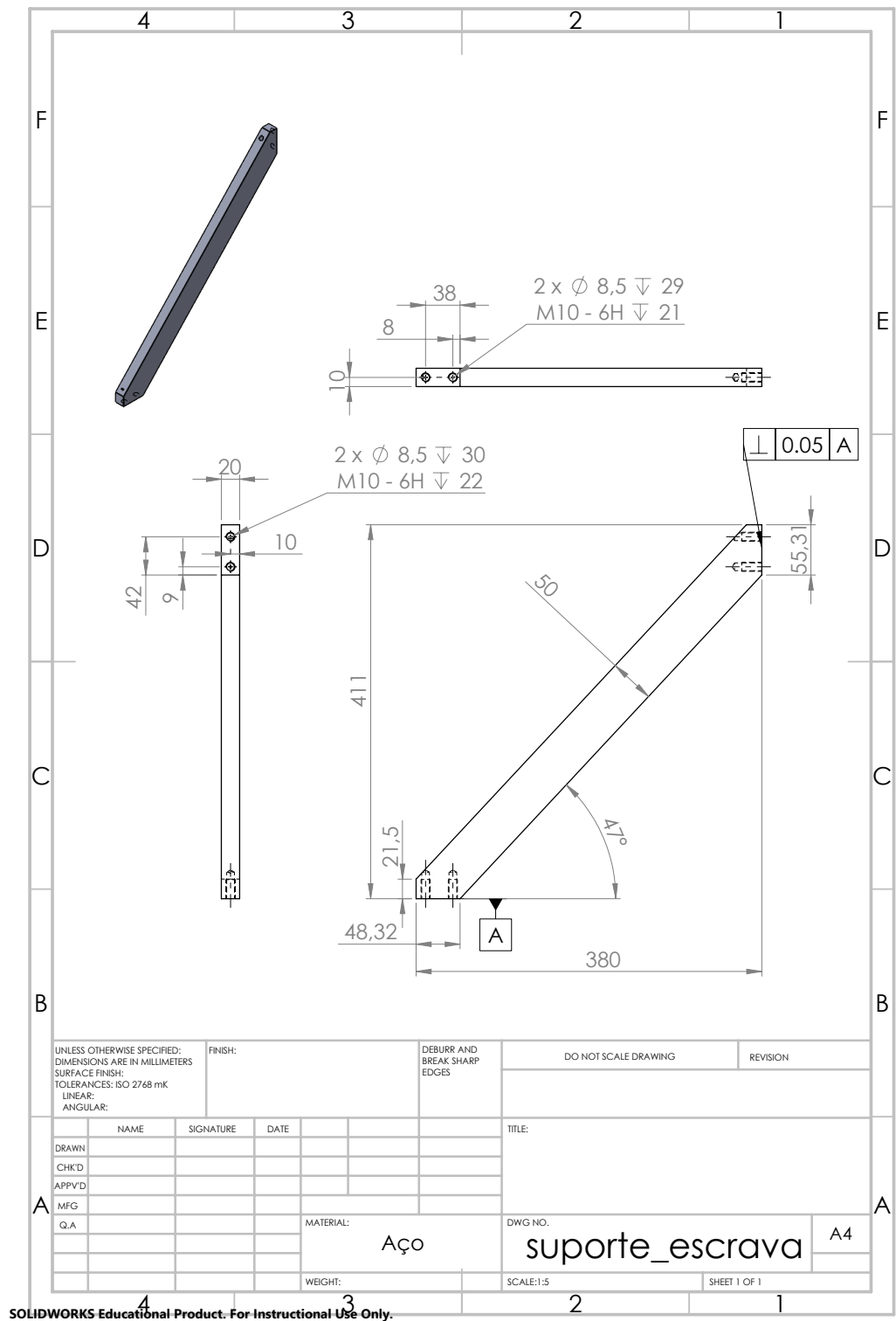


Figure A.26: 2D drawing of part "suporte_escrava"

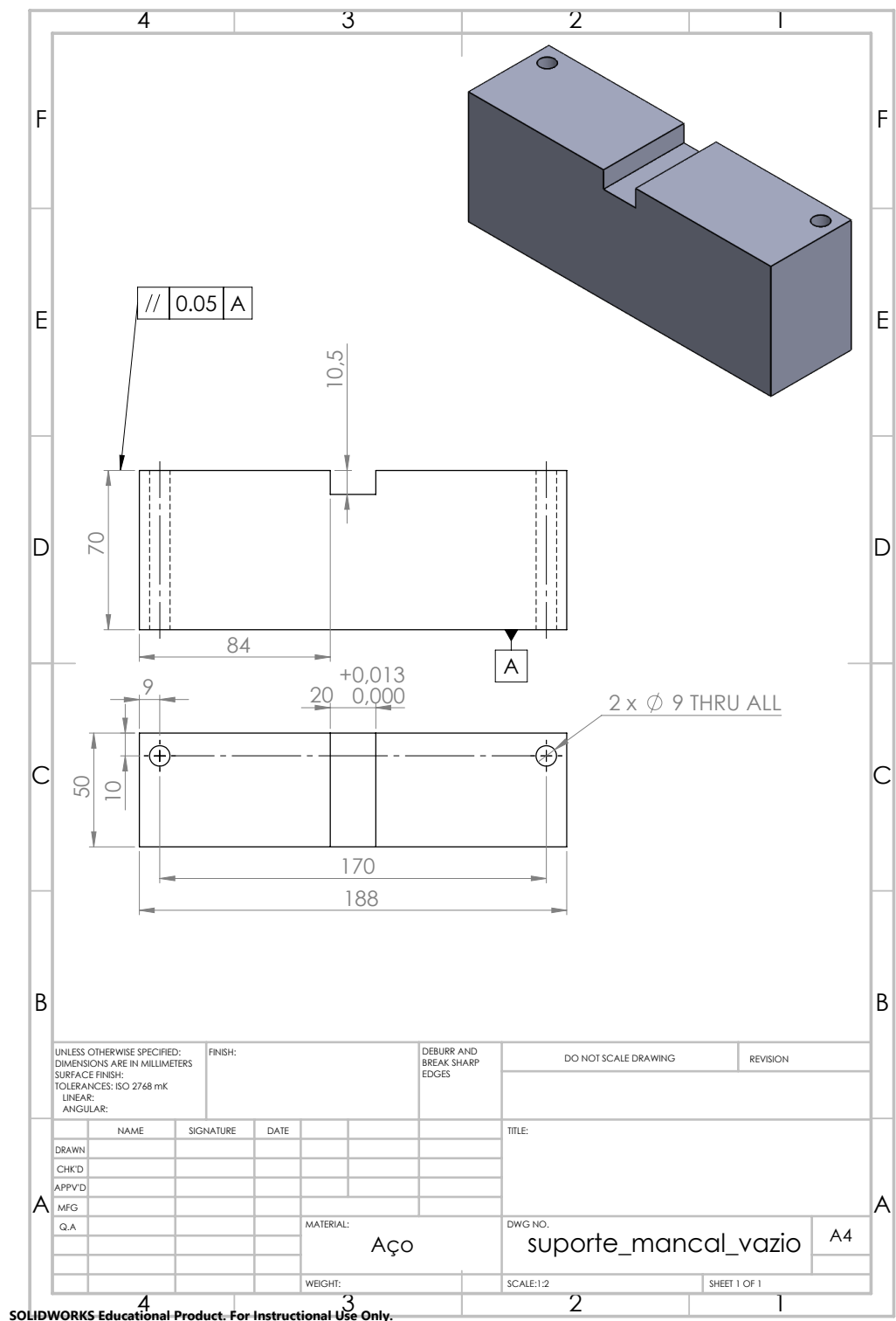


Figure A.27: 2D drawing of part "suporte_mancal_vazio"

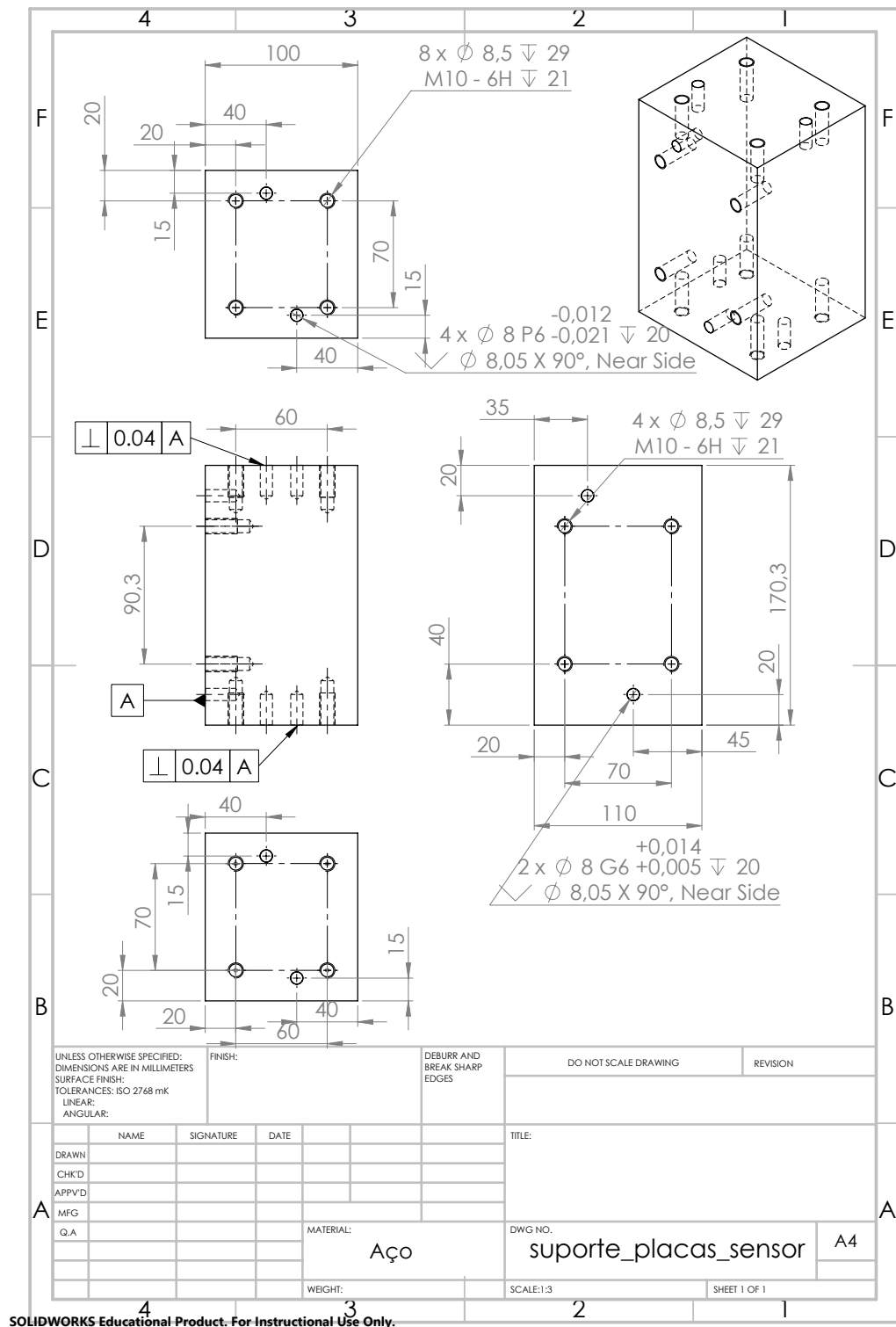


Figure A.28: 2D drawing of part "suporte_placas_sensor"

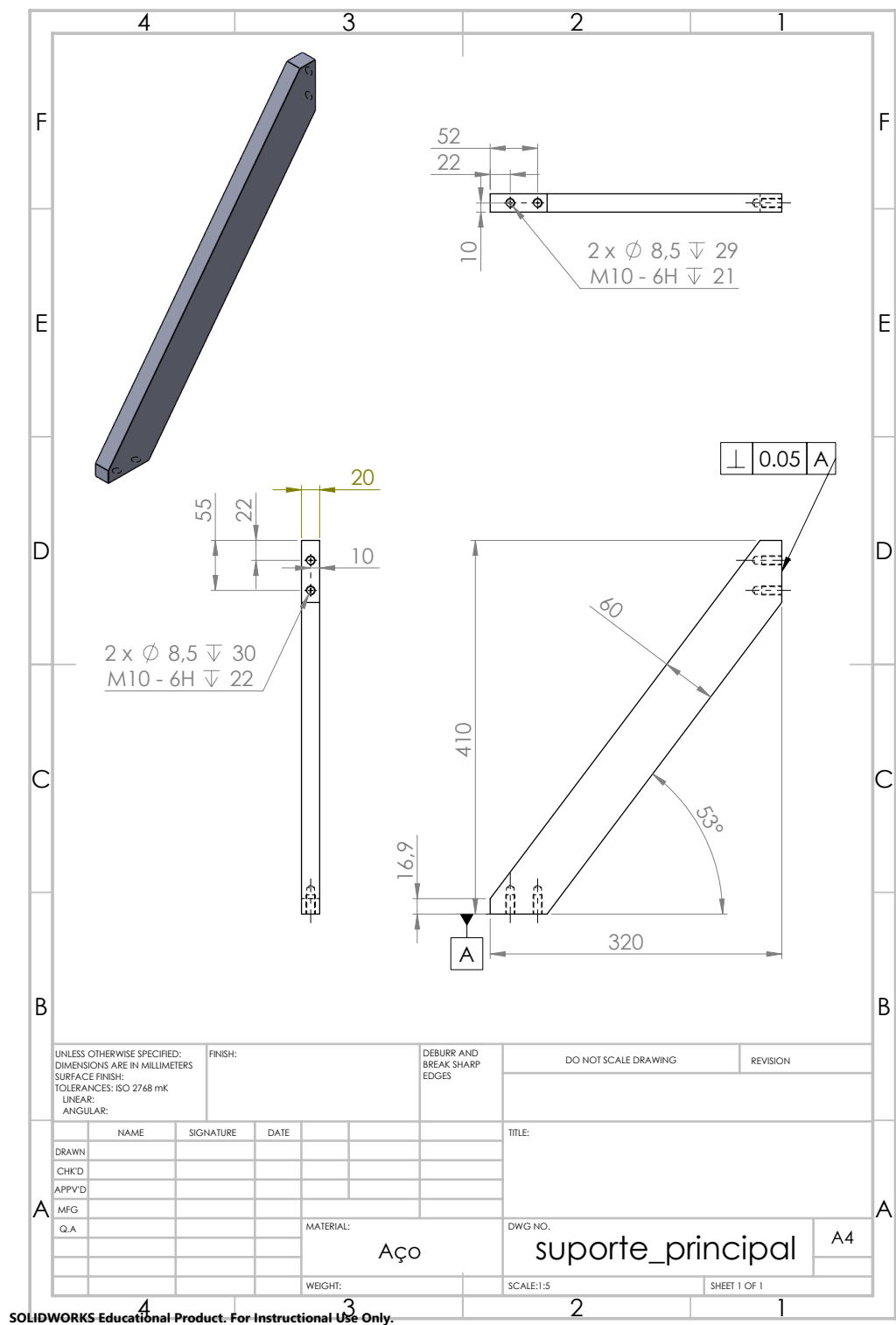
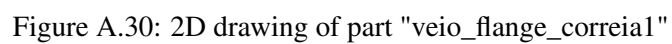


Figure A.29: 2D drawing of part "suporte_principal"



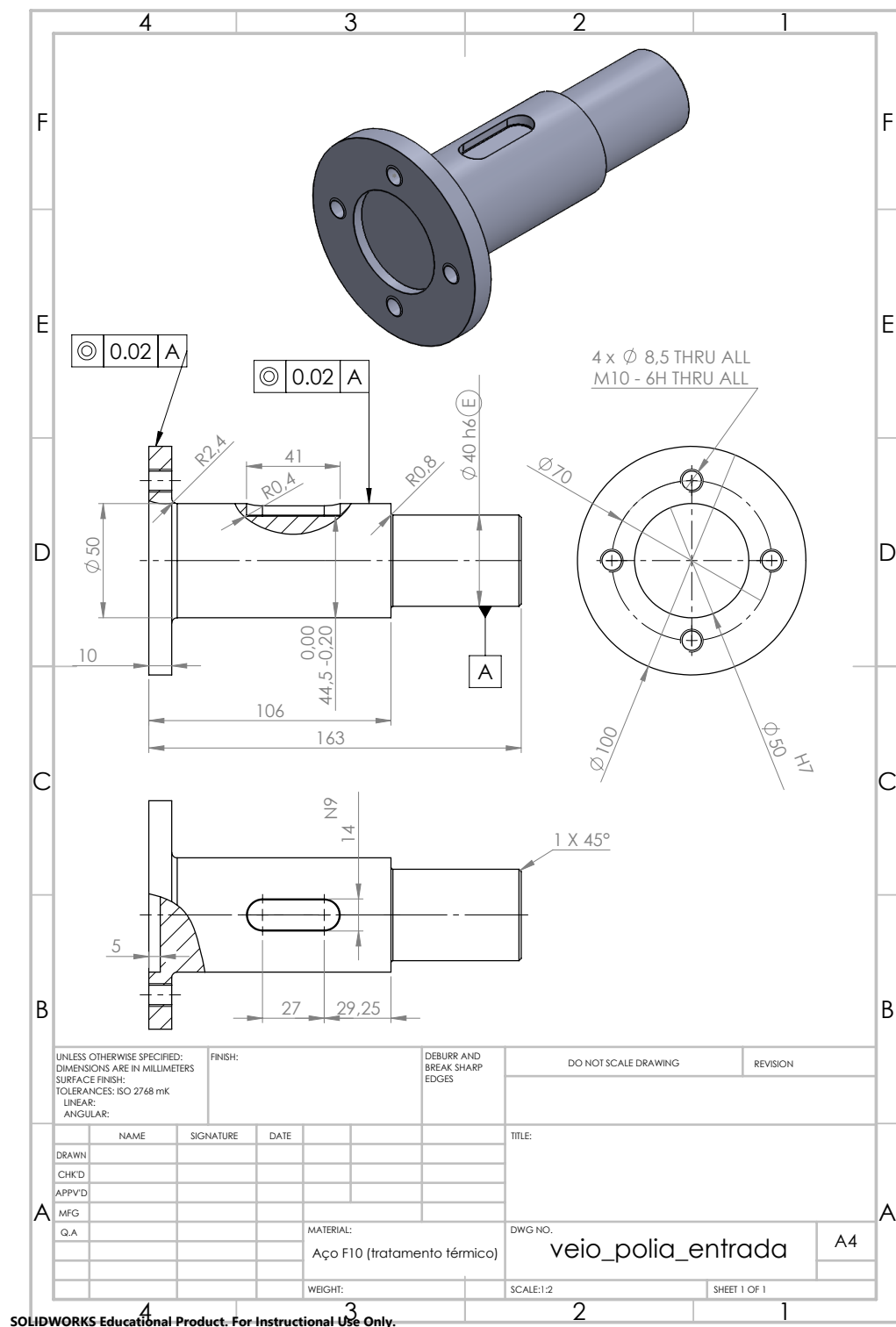


Figure A.31: 2D drawing of part "veio_polia_entrada"

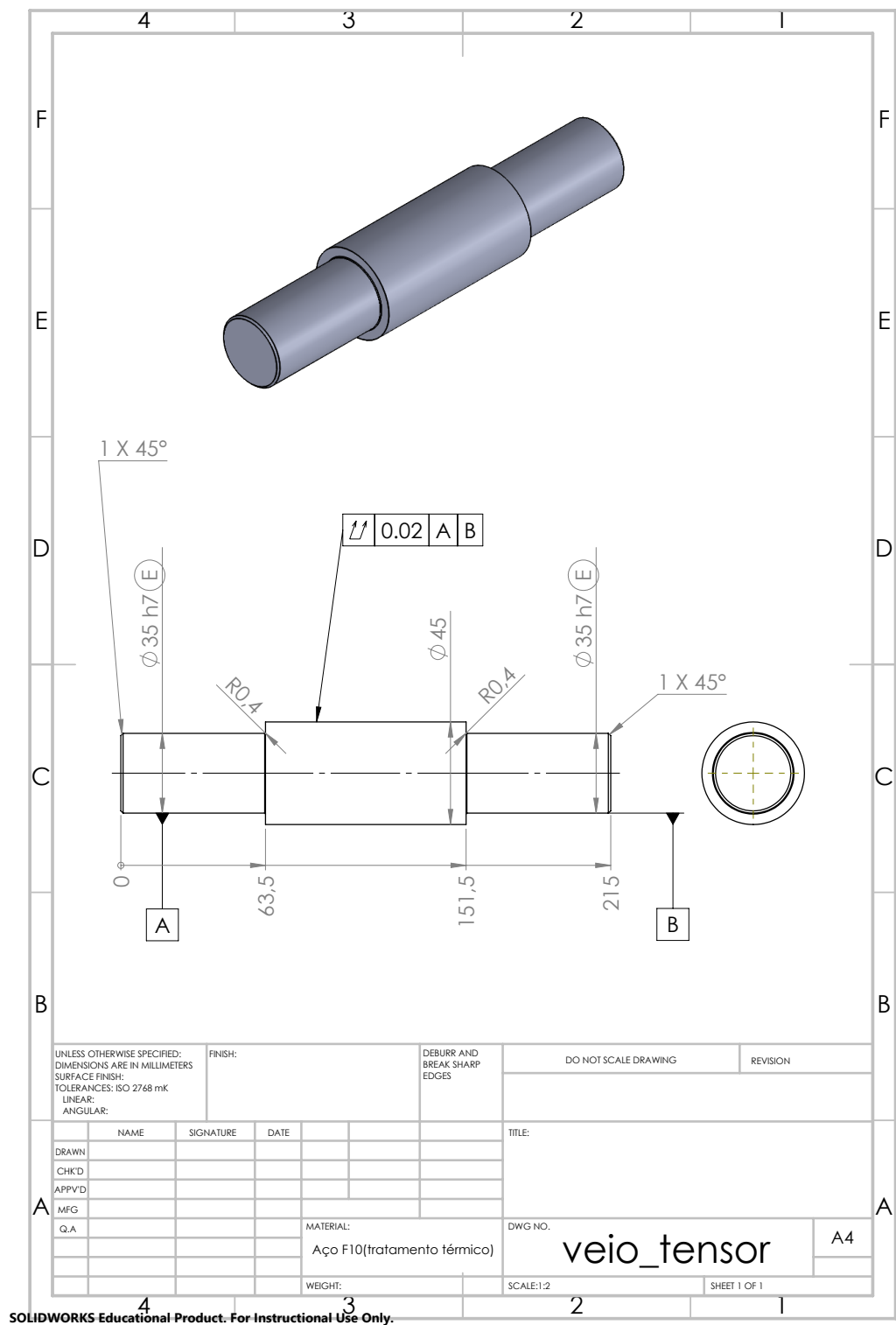
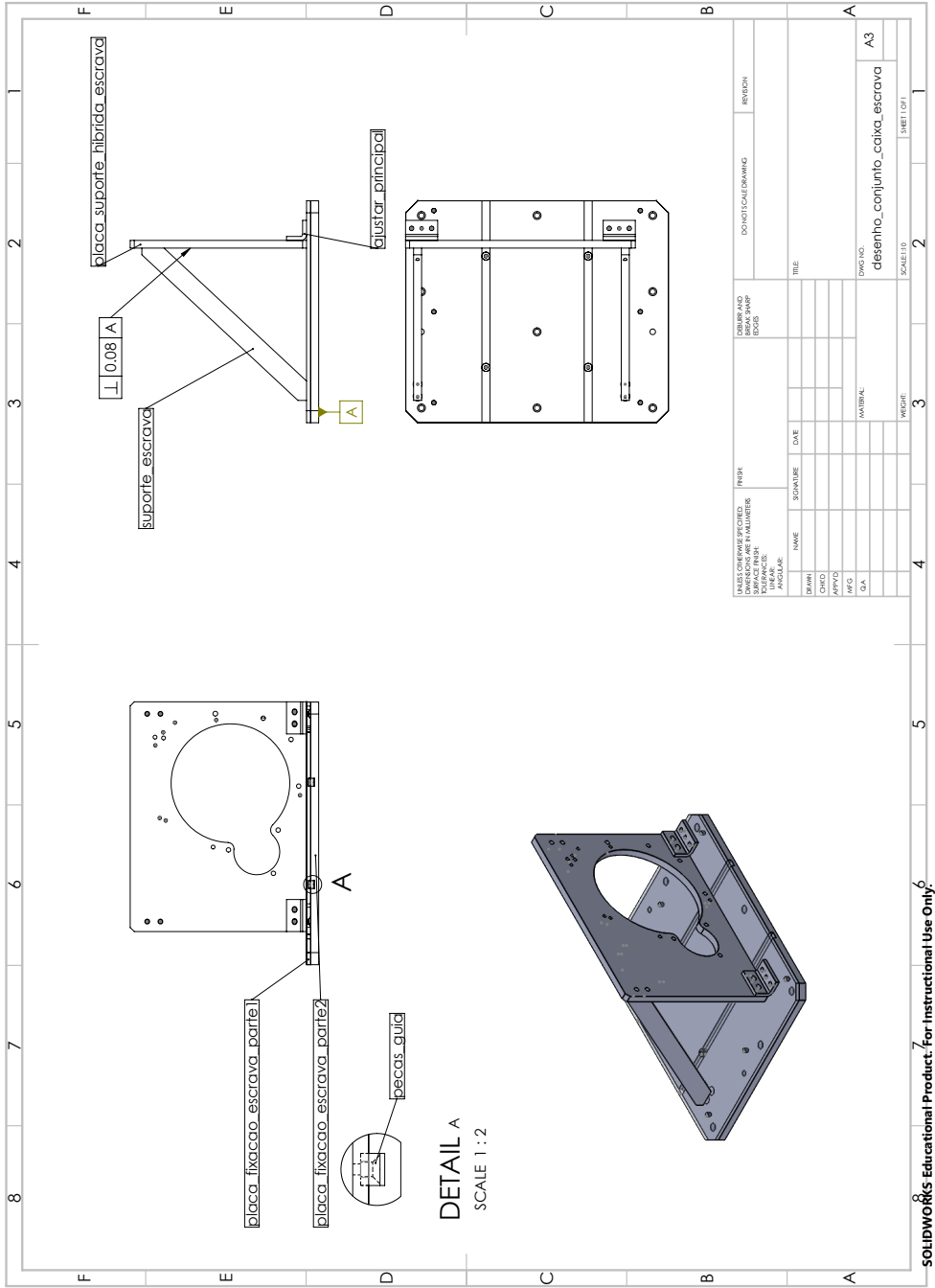
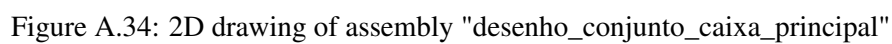


Figure A.32: 2D drawing of part "veio_tensor"





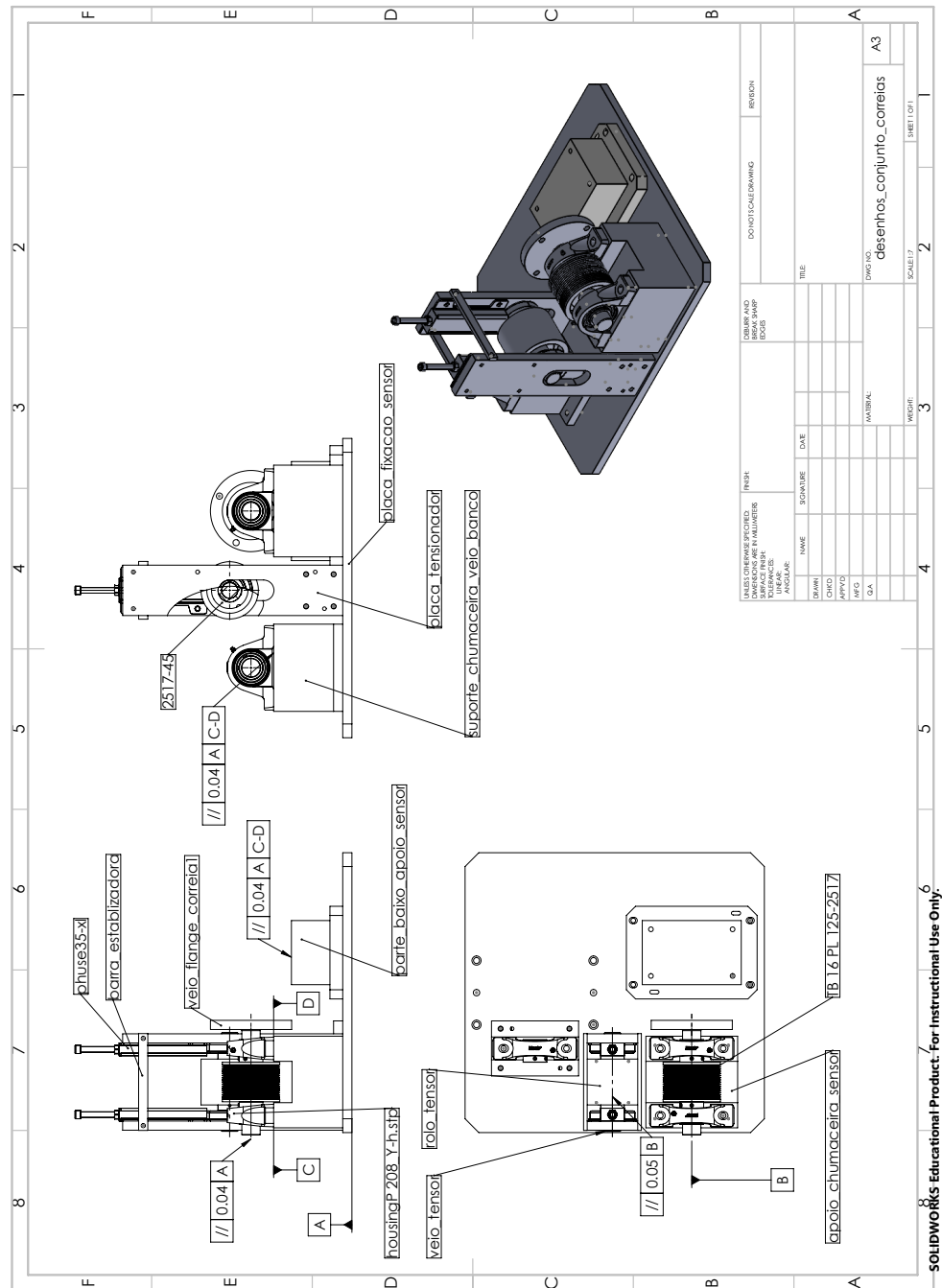


Figure A.35: 2D drawing of assembly "desenhos_conjunto_correias"