
Experimental torque loss in a vehicle transfer gearbox

Carlos André Ferreira Pereira

Masters dissertation submitted to
Faculdade de Engenharia da Universidade do Porto

Supervisors:
Doctor Pedro M. T. Marques
Professor Jorge H. O. Seabra
Doctor Ramiro C. Martins

Unidade de Tribologia, Vibrações e Manutenção Industrial
Departamento de Engenharia Mecânica
Faculdade de Engenharia da Universidade do Porto

Porto, 2017

The work presented in this dissertation was performed at the
Unit of Tribology, Vibrations and Industrial Maintenance (CETTRIB)
Department of Mechanical Engineering
Faculty of Engineering
University of Porto
Porto, Portugal.

Carlos André Ferreira Pereira
E-mail: em12072@fe.up.pt

Faculdade de Engenharia da Universidade do Porto
Departamento de Engenharia Mecânica
Unidade de Tribologia, Vibrações e Manutenção Industrial
Rua Dr. Roberto Frias s/n, Sala M110
4200-465 Porto
Portugal

To my family and friends...

Acknowledgements

I started my Integrated Masters Degree in Mechanical Engineering in Faculdade de Engenharia da Universidade do Porto in September of 2012, and now, that this stage is close to be concluded, I would like to acknowledge and give my most sincere thanks to the people without whom I would not be able reach this end.

During my stay in FEUP I have met incredible people, all contributed to academic a social well being. I would like to thank my colleagues for all the help friendship and support during this five years: João Santos, Manuel Pimenta, Manuel Cardoso, Daniel Almeida, Ana Rita Pintão, Leonardo Moreira, André Martinho and Marco Bento.

I would also like to send a special thanks to the ones that have been with me since before I can remember: João Oliveira, André Neves, Daniel Silva, Helder Ribeiro and Luís Lopes. I want also to sincerely thank the ones that helped me in the years before college: Simão Carvalho, Luís Ferreira, Miguel Boas, André Ribeiro and Bruno Silva.

I would like to acknowledge my supervisors Dr. Pedro Marques, Dr. Ramiro Martins and Prof. Jorge Seabra without whom this work would not be possible, with a special thanks to Dr. Pedro Marques that advised me since day one and guided me until the last.

I wish to thank my colleagues at CETRIB that provided for all the help, friendship, support and unforgettable moments in the last few months: João Sousa, Joana Sousa and António Vieira.

I want to also thank my family, and specially my parents for all the support, sacrifice and conditions provided that allowed me to finish this masters degree. I want to send a very special thank to my girlfriend that pulled me up and gave strength and support to continue and conclude this journey.

At last, I cannot forget to thank the Faculdade de Engenharia da Universidade do Porto, CETRIB (Unit of Tribology, Vibrations and Industrial Maintenance), LAETA and Sci-Tech for the opportunity and the conditions provided to perform this dissertation.

Abstract

Improving the gearboxes efficiency has been a goal of the automotive industry for many years. A better efficiency along a vehicle power train leads to an improvement in fuel economy. Although the efficiency of a gearbox depends of parameters such as gear geometry, number of reduction stages and carter geometry, that once the gearbox is designed and produced cannot be changed, the lubricant also plays a main role in the gearbox efficiency. The lubricant is the easiest component to replace in a gearbox, and since new formulations are being developed constantly, the study of its influence in the overall efficiency of a gearbox is of high relevance.

Two fully formulated lubricants, available in the market for axle gear applications, were tested in a vehicle transfer gearbox. The chosen lubricants are both of synthetic base, poly- α -olefin (PAO), with identical additive packages. The lubricants differ in their viscous properties, being the 75w140-A more viscous than the 75w90-A.

The gearbox was tested using a back-to-back test rig with recirculating power, that registers the input and output torque as well as the input and output speed, oil temperature, outside wall temperature and ambient temperatures. In the tests to perform, the input speed ranges from 600 to 800 rpm and the input torque from 150 to 250 Nm. This conditions aim to simulate the working conditions of the gearbox when installed in a vehicle. The experimental values allow an overall evaluation and comparison of the behavior of the lubricants within the test range.

A power loss model was developed in order to estimate the gearbox power loss as a function of the input torque, input speed and working temperature. The implemented model breaks down the total power loss in four parts according to their origin: gear meshing, gear churning, rolling bearings and seals.

A thermal approach was made to the system, and a thermal model was merged with the power loss model. The final model can estimate the power loss and the stabilization temperature of the gearbox without the need of a experimental test.

Resumo

Um dos objetivos da indústria automóvel ao longo dos anos tem sido a melhorar a eficiência de caixas de velocidades. Uma melhor eficiência ao longo da cadeia de transmissão de um veículo leva a uma melhoria nos consumos do veículo. Apesar de a eficiência de uma caixa de velocidades depender de parâmetros como a geometria das engrenagens, número de andares de redução e da geometria do cârter, que após a fase de projeto não podem ser alteradas, o lubrificante também desempenha um papel importante no desempenho do conjunto. Sendo o lubrificante o componente mais fácil de substituir, o estudo da sua influência no desempenho global da caixa de velocidades é da maior relevância.

Dois óleos certificados, disponíveis no mercado para aplicação em caixas de transmissão, foram testados na caixa de transferência de um veículo. Os lubrificantes escolhidos são ambos de base sintética, poly- α -olefinas (PAO), com pacotes de aditivos idênticos. Os lubrificantes diferem nas suas propriedades viscosas, sendo o 75w140-A mais viscoso que o 75w90-A.

A caixa de transferência foi testada usando um banco de ensaios com recirculação de potência que regista os binários de entrada e saída, bem como as velocidades e temperaturas ambiente, da parede da caixa e do óleo. Nos testes a realizar, a velocidade de entrada varia de 600 a 800 rpm e o torque de 150 a 250 Nm. Estas condições de funcionamento foram definidas com o intuito de simular as condições de funcionamento quando instalada no veículo. Os resultados experimentais permitem uma avaliação global do comportamento de cada lubrificante dentro das condições de funcionamento.

Foi desenvolvido um modelo de perda de potência com o objetivo de estimar a perda de potência como função de velocidade, binário e temperatura de funcionamento. O modelo implementado divide a perda de potência em quatro partes, de acordo com a sua origem: perdas de atrito no engrenamento, chapinagem, perdas por atrito nos rolamentos e nos vedantes.

Também foi realizada uma abordagem térmica ao problema, e o modelo de perda de potência foi combinado com o térmico. O modelo final consegue estimar a perda de potência e a temperatura de estabilização de caixa de transferência sem a necessidade de realizar um teste experimental.

Keywords

Transfer Gearbox
Lubrication
Axle gear oils
Power loss
Efficiency
Gear meshing loss
Gear churning loss
Rolling bearing power loss
Heat dissipation

Palavras chave

Caixa de transferência
Lubrificação
Óleo de caixa de velocidades
Perda de potência
Eficiência
Perda de potência por atrito no engrenamento
Perda de potência por chapinagem
Perda de potência por atrito nos rolamentos

Contents

Acknowledgements	iii
Abstract	v
Resumo	vii
Keywords	ix
List of variables	xix
1 Introduction	1
1.1 Objectives	1
1.2 Thesis outline	2
2 Lubricants and lubrication	3
2.1 Lubricant Types	3
2.1.1 Liquid lubricants	3
2.1.2 Solid lubricants	5
2.1.3 Gaseous lubricants	6
2.2 Additive package	6
2.2.1 Viscosity index improving additives	6
2.2.2 Anti wear and Extreme pressure additives	6
2.3 Physical properties of lubricant oils	7
2.3.1 Viscosity	7
2.3.2 Bulk density and specific gravity	11
2.3.3 Thermal properties	11
3 Experimental campaign	13
3.1 Gearbox	13
3.2 Test rig	15
3.3 Lubricant properties	17
3.4 Test parameters	18
3.5 Experimental procedures	19
3.5.1 Test rig calibration	19
3.5.2 Definition of test duration	19
3.6 Test sequence	20
3.7 Results	21
3.7.1 75W140-A	21
3.7.2 75W90-A	23
3.7.3 Lubricant comparison	25

4	Power loss model	27
4.1	Power loss in gearboxes	27
4.1.1	Power loss in gear meshing	27
4.1.2	Churning loss	31
4.1.3	Power loss in rolling bearings	33
4.1.4	Power loss in seals	37
4.2	Transfer gearbox modelling	37
4.2.1	Gearbox kinematics	37
4.2.2	Gear geometry and contact forces	37
4.2.3	Rolling bearings power loss	38
4.2.4	Gear mesh power loss	39
4.2.5	Churning power loss	39
4.2.6	Model verification	40
4.3	Model estimation	40
4.3.1	75W140-A oil	40
4.3.2	75W90-A oil	42
4.3.3	Lubricant comparison	43
4.4	Power loss model vs experimental results	48
4.4.1	Overall comparison	48
4.4.2	Deviation analysis	50
5	Thermal model	55
5.1	Heat transfer modes	55
5.1.1	Convection	56
5.1.2	Conduction	56
5.1.3	Radiation	57
5.1.4	Reo-electric analogy	57
5.2	Equivalent thermal resistance	57
5.3	Stabilization temperature estimation	58
5.3.1	75W140-A	59
5.3.2	75W90-A	63
6	Conclusion	69
6.1	Conclusions	69
6.2	Future Work	70
	Bibliography	71
A	Test data	75
A.1	75w140 tests	75
A.2	75w90 tests	84
B	Gear geometrical parameters	93
C	KISSSys reports	95
D	MatLab printouts	119

List of Figures

2.1	Speed gradient between two moving plates [4].	7
2.2	Viscosity index representation [4].	8
2.3	Kinematic viscosity for the same lubricant by Vogel, ASTM341 and Cameron [2]	10
3.1	Scheme of the transfer gearbox installed on a vehicle.	13
3.2	Installed gears.	14
3.3	Scheme of the test rig[2].	15
3.4	3-phase electric motor and timing belt.	16
3.5	Viscosity function of temperature for 75w140-A and 75w90-A	18
3.6	Oil temperature evolution during the whole test	20
3.7	Power loss function of input power for 75W140-A (1-unstabilized test). . . .	21
3.8	Relative stabilization temperature function of input power for 75W140-A (1-unstabilized test).	22
3.9	Efficiency function of input power for 75W140-A (1-unstabilized test). . . .	23
3.10	Power loss function of input power for 75W90-A (2-unstabilized test). . . .	23
3.11	Relative stabilization temperature function of input power for 75W90-A (2-unstabilized test).	24
3.12	Efficiency function of input power for 75W90-A (2-unstabilized test)	24
3.13	Power loss difference between 75W140-A and 75W90-A. Values 1 and 2 are affected by unstabilized tests.	25
3.14	Stabilization temperature difference between 75W140-A and 75W90-A. Values 1 and 2 are affected by unstabilized tests.	26
3.15	Efficiency difference between 75W140-A and 75W90-A. Values 1 and 2 are affected by unstabilized tests.	26
4.1	Stribeck curve.	28
4.2	Evolution of properties along the meshing line.	29
4.3	Scheme of meshing gears [15].	30
4.4	Rotation senses as a meshing pair [2].	33
4.5	Absolute values for the power loss sources for 75W140-A.	41
4.6	Power loss sources influence relative to the experimental power loss for 75W140-A.	41
4.7	Absolute values for the power loss sources for 75W90-A.	42
4.8	Power loss sources influence relative to the experimental power loss for 75W90-A.	43
4.9	Relative difference between the meshing power loss (P_{VZP}) of 75W140-A and 75W90-A.	44

4.10	Friction coefficient function of specific film thickness for the meshing pair z_1/z_2	45
4.11	Friction coefficient function of specific film thickness for the meshing pair z_3/z_5	45
4.12	Relative difference between the churning power loss (P_{VZ0}) of 75W140-A and 75W90-A.	46
4.13	Relative difference between the rolling bearing power loss (P_{VL}) of 75W140-A and 75W90-A.	47
4.14	Constant torque lines of power loss model (Mod.) and experimental results (Exp.) for 75W140-A. (1-Unstabilized test).	48
4.15	Constant speed lines of power loss model and experimental results for 75W140-A. (1-Unstabilized test).	48
4.16	Constant torque lines of power loss model and experimental results for 75W90-A. (2-Unstabilized test).	49
4.17	Constant speed lines of power loss model and experimental results for 75W90-A. (2-Unstabilized test).	49
4.18	Absolute deviations for the lubricant 75W140-A. 1-value affected by unstabilized test (150 Nm 800 rpm).	50
4.19	Relative deviations for lubricant 75W140-A. 1-value affected by unstabilized test (150 Nm 800 rpm).	50
4.20	Absolute deviations for lubricant 75W90-A. 2-value affected by unstabilized test.	51
4.21	Relative deviations for lubricant 75W90-A. 2-value affected by unstabilized test.	52
4.22	Experimental measured power loss, power loss model partitions and absolute difference for 75W140-A.	53
4.23	Experimental measured power loss, power loss model partitions and absolute difference for 75W90-A.	54
5.1	Equivalent thermal resistance and trendlines	58
5.2	Relative stabilization temperatures ($T_{oil}-T_{room}$) at $T_{room}=25^{\circ}\text{C}$ for 75W140-A.	59
5.3	Limits for the stabilization temperature ($T_{oil}-T_{room}$) at $T_{room}=25^{\circ}\text{C}$ for 75W140-A.	60
5.4	Model estimation and experimental values for the relative stabilization temperature for 75W140-A (1-Unstabilized test).	60
5.5	Measured relative stabilization temperate, model estimation and upper and lower limits for the stabilization temperature for 75W140-A ($T_{room}=25^{\circ}\text{C}$).	61
5.6	Constant torque lines of power loss model and experimental results for 75W140-A (1-Unstabilized test).	62
5.7	Constant speed lines of power loss model and experimental results for 75W140-A. (1-Unstabilized test).	62
5.8	Relative stabilization temperatures ($T_{oil}-T_{room}$) at $T_{room}=35^{\circ}\text{C}$ for 75W90-A.	63
5.9	Limits for the stabilization temperature ($T_{oil}-T_{room}$) at $T_{room}=35^{\circ}\text{C}$ for 75W90-A.	63
5.10	Model estimation and experimental values for the relative stabilization temperature ($T_{oil}-T_{room}$) for 75W90-A. (2-Unstabilized test).	64
5.11	Measured stabilization temperate, model estimation and upper and lower limits for the stabilization temperature for 75W90-A ($T_{room}=35^{\circ}\text{C}$).	65

5.12	Constant speed lines of power loss model and experimental results for 75W90-A (2-Unstabilized test).	66
5.13	Constant torque lines of power loss model and experimental results for 75W90-A (2-Unstabilized test).	66

List of Tables

3.1	Geometrical parameters of the gearbox wheels.	14
3.2	Rolling bearings in the gearbox.	14
3.3	Mechanical properties of the steel DIN 15CrNi6.	15
3.4	Properties of 75W140-A and 75W90-A [1; 7].	17
3.5	Additive package composition [1].	17
3.6	Test sequence for the lubricant 75W140-A	20
3.7	Test sequence for the lubricant 75W90-A.	21
4.1	Parameters for equation (4.1)	32
4.2	Manufacturer friction coefficients.	35
4.3	Rolling bearings experimental friction coefficients [25].	38
4.4	Rolling bearing constants.	39
4.5	Gear loss factor H_v	39
4.6	Lubricant correction factor X_L	39
4.7	Immersed depth of the wheels.	40

List of variables

Symbol	Description	Unit
A	Gearbox area exposed to the air	m^2
b	Gear face width	mm
C_{ch}	Churning torque	Nm
C_m	Dimensionless drag torque	-
D_p	Pitch diameter	m
E_{rad}	Radiated Energy	W
E^*	Equivalent Young modulus	Pa
F_{bt}	Tooth normal force	N
Fr	Froud number	-
g	Gravity acceleration	m s^{-2}
h	Gear immersed depth	m
h_{cnd}	Convection coefficient	$\text{W}(\text{m}^2\text{K})^{-1}$
h_0	Film thickness	m
H_{tooth}	Tooth height	m
H_V	Gear loss factor	-
i	Gear ratio	-
I	Electric current	A
K	Solid thermal conductivity	$\text{W}(\text{mK})^{-1}$
k	Thermal conductivity	$\text{W}(\text{mK})^1$
l	Contact length	mm
l_{min}	Minimum contact length	mm
m	Gear modulus	mm
M_{drag}	R. Bearing drag frictional torque	Nmm
M^{rol}	R. Bearing total frictional torque	Nmm
M_{rr}	R. Bearing rolling frictional torque	Nmm
M_{sl}	R. Bearing sliding frictional torque	Nmm
n	rotational speed	rpm
n_{in}	transfer gearbox input speed	rpm
n_{out}	transfer gearbox output speed	rpm

Symbol	Description	Unit
P_V	Total power loss	W
P_{VL}	R. bearing power loss	W
P_{VS}	Seal power loss	W
P_{VZO}	Churning power loss	W
P_{VZP}	Meshing power loss	W
Q_{cnd}	Conduction heat flow	W
Q_{cnv}	Convection heat flow	W
Q_{Ev}	Total evacuated heat	W
R_a	Average surface roughness	μ m
Re	Reynolds number	-
R_p	Pitch radius	m
R_{Teq}	Equivalent thermal resistance	KW^{-1}
R	Electrical resistance	Ω
T_{amb}	Ambient temperature	$^{\circ}C$
T_{oil}	Oil temperature	$^{\circ}C$
T_{room}	Test room temperature	$^{\circ}C$
T_s	Surface temperature	K
Tq_{in}	Input torque	Nm
Tq_{out}	Output torque	Nm
T_{wall}	Test gearbox wall temperature	$^{\circ}C$
U	Linear velocity	$m\ s^{-1}$
V	Electric Tension	V
V_0	Oil volume	m^3
$v_{\sum c}$	Sum velocity at pitch point	$m\ s^{-1}$
X_L	Lubricant parameter	-
z	Wheel number of teeth	-
α	Pressure angle	$^{\circ}$
α_{oil}	Piezoviscosity coefficient	Pa^{-1}
α_t	Transverse pressure angle	$^{\circ}$
α_T	Thermal expansion coefficient	-
α_w	Working pressure angle	$^{\circ}$
β	Helix angle	$^{\circ}$
β_b	Base helix angle	$^{\circ}$
β_T	Termoviscosity coefficient	$^{\circ}C^{-1}$
γ	-	ms^{-2}
η_{oil}	Oil dynamic viscosity	mPa s

Symbol	Description	Unit
θ	Angle between wheel center and oil sump level	rad
Λ	Specific film thickness	-
μ_{mz}	Average gear mesh friction coefficient	-
ν	Lubricant kinematic viscosity	cSt
ρ	Bulk density	kgm ⁻³
ρC	Curvature radius at pitch point	mm
σ	Composed roughness	μm
σ_s	<i>Stefan-Boltzman</i> constant	-
τ_{xy}	Shear tension	Pa
ω	Rotational speed	rad s ⁻¹

Introduction

Gearboxes have been subject of study to mechanical engineers since the invention of the gear. Although they have been exhaustively study and perfected from the gear to the housing design and lubrication in order to increase the working life and efficiency, there is always room for improvement.

In recent years there has been an increase in the study of power transmission mechanisms. This fact is due the increasing environmental awareness, that for automotive applications, is translated in the need to reduce fuel consumption [1]. In order to achieve this objective, weight reduction and thermal management are possible approaches. Another possible approach is to reduce the losses along the power transmission mechanisms, that although the efficiency is already quite high, a small reduction in losses translates in a significant improvement in overall efficiency.

An accurate prediction of the power loss will aid in the comparison and selection of lubricants and working conditions with the objective of achieving the best possible efficiency.

Previous studies have been performed by Marques [2] and Gonçalves [3], in order to study the behavior of the same gearbox used to perform the tests in this dissertation, although with different working and lubrication conditions. Their objective was to evaluate the behavior of the gearbox when lubricated with wind turbine gear oils. The tests were performed at low speeds and high torques, in order to simulate the working conditions in wind turbines.

1.1 Objectives

The main objective of this work is to experimentally test and analyze the influence of lubricant formulations in a vehicle transfer gearbox, within a range of working conditions similar to the ones present in a current gearbox. Another goal of this dissertation is to implement a prediction model using *Matlab R2016a* that estimates the gearbox efficiency.

To achieve this goals, a suitable range of working conditions must be set. The working conditions must reflect the speed and torque at which the gearbox functions when installed in the vehicle.

In order to verify the implemented model, the gearbox was tested with different lubricants in discreet points of the defined range, using a back-to-back test rig with recirculating power.

The implemented model may also estimate the stabilization temperature of the gearbox, and therefore estimate the gearbox efficiency at a given input speed and torque.

1.2 Thesis outline

Chapter 2 describes the lubricant universe. This chapter includes the lubricant types, properties and the mathematical approaches used to quantify its behavior.

Chapter 3 is dedicated to the experimental campaign. It begins with a presentation of the test gearbox and of the test rig with a description of its components. It also includes a description of the tested lubricants. The experimental campaign also includes the test procedure and sequence. This chapter is concluded with a presentation of the obtained results, as well as a first look and comparison of the lubricants behavior.

Chapter 4 describes the *Matlab* implemented model used to estimate and breakdown the total power loss of the gearbox. This chapter presents the models used to estimate the power loss associated with each gearbox component followed by the implemented constants specific for the gearbox components and the tested lubricants. This chapter also includes the model results, with the power loss partitions, as well as a comparison with the values measured in Chapter 3.

Chapter 5 presents a thermal approach to the problem. It contains a brief description of the heat transfer modes and the process used to estimate the stabilization temperature of the gearbox. The model results are presented and compared to the experimentally measured results.

Chapter 6 is dedicated to the main conclusion of the dissertation and to the ideas for future works.

Lubricants and lubrication

The main function of lubricants in a mechanical system is to minimize and control the friction and wear of surfaces in contact. The increasing complexity of mechanisms created the need to assign more functions to the lubricant, such as heat evacuation, removal of wear particles, operation at higher temperatures and inhibition of corrosion. Inside an elastohydrodynamic contact, service conditions may vary, in very short periods of time, from ambient pressure to 1GPa, and also the temperature can rise 100K [4].

2.1 Lubricant Types

To achieve the best performance possible for every application, there is a wide range of different lubricants. Lubricants can be divided into four categories: liquid, solid, gaseous and lubricant greases, and each category can also be divided into subcategories according to their origin [4].

2.1.1 Liquid lubricants

Liquid lubricants, or lubricant oils, are the most commonly used in mechanical applications and can be classified according to their base oil as: Vegetal and animal origin lubricants, Mineral and Synthetic based oils [4].

Vegetal and animal lubricants

Vegetal and animal based oils were the first to be used by mankind, but due to increasing performance demands, their usage decayed as other types of oil appeared. This type of oil presents some advantages such as high viscosity index and low evaporation rate, but on the other hand also presents a fast oxidation rate and low tolerance for high working temperatures. Despite the disadvantages, vegetal origin oils usage increased in the latter years due to environmental regulations due to their better biodegradability in comparison with other types of oil.

Mineral oils

Mineral oils are refinery products and are mainly composed of natural hydrocarbons resulting from organic matter decomposition. According to their origin and refinement process, the mineral oils can originate from three distinct bases: paraffin, naphthenic and aromatic. The aromatic basis is usually undesirable and represents a negligible part of the

applications. The remaining basis will be described ahead.

Paraffin basis

Paraffin basis are the most common in lubricant oils and are composed by long hydrocarbon chains. Oils with this basis present a low specific weight, low freezing point, great oxidation resistance low viscosity variation with temperature and therefore present a high viscosity index. Paraffin based oils are usually very elastomer friendly,

Naphthenic basis

The naphthenic based oils molecular structure presents their hydrocarbons arranged in circular patterns. These oils present a high specific weight, high freezing point, low oxidation resistance and high viscosity variations with temperature, thus presenting a low viscosity index, and so their use is restrained to applications with low thermal amplitude.

Naphthenic basis are aggressive to elastomers, reducing the seals life. These oils are also highly volatile, easily miscible and present low viscosity at low temperatures.

In general, mineral oils are a combination of both basis described above, and are designated paraffin or naphthenic based according to the predominant base on its constitution. The mineral bases present very different characteristics, and by mixing the two, mineral oils can cover a wide range of applications due to the ease to combine the following properties:

- Wide range of viscosity and viscosity indexes;
- Low volatility;
- Resistant to degradation (inert);
- Protects metal from corrosion;
- Low price.

Synthetic oils

Synthetic lubricant oils are synthesised from hydrocarbons present in petroleum and vegetal products. The study of hydrocarbons allowed the industry to develop various synthesis techniques and produce lubricants for particularly difficult conditions. Synthetic based oils tend to be more expensive but generally perform better than mineral based ones, mainly in oxidation resistance, viscosity index and coefficient of friction. The improvement in this properties usually translates in a longer life, better working conditions at high temperatures and a better efficiency of the mechanism. The synthetic oils group can be subdivided in categories worth mentioning:

- Poly- α -olefins (PAO)- have a similar structure to the mineral hydrocarbons and therefore the types are miscible. Present a great thermal stability, are very elastomer friendly and can be used in food and pharmaceutical industry. On the downs side, need for anti-oxidation additives.
- Polyglycols - present a low friction coefficient making them the ideal choice for high sliding applications, such as worm gears. Present low miscibility with mineral oils and at high temperatures are often harmful to seals and polymers.

- Esters - can be obtained with a wide range of properties, but the more remarkable are the high thermal resistance, good behavior at low temperatures and quickly biodegradable. These oils can also provide a friction coefficient as low as the polyglycols. Depending on the additive package, these oils may present low hydrolytic stability.
- Silicons - present high thermal stability and resistance to oxidation. These oils are also fire resistant, chemically inert, immiscible with water and non-toxic.

Lubricant greases

The lubricant greases are the combination of a lubricant agent and a disperse thickening agent. The thickening agent can be divided into two main categories:

- Soaps - Aluminium, Barium, Calcium, Lithium, Sodium and Strontium;
- No soap - Organic clays, Polyureas, Inorganic compounds;

The lubricant grease has a behavior identical to the lubricant used to solve the thickening agent, but the thickening agent and additive package limit the properties. Lubricant greases are used when, for economical or technical reasons, it is impossible for the oil to reach the contact. This type of lubricant also offers protection against contamination of the contact by external particles and reduce the contamination of the environment by the lubricant.

2.1.2 Solid lubricants

A solid lubricant is a film of solid material in between the surfaces in contact. The solid lubricants can be of organic or inorganic origin.

Organic solid lubricants

Organic solid lubricants can also be divided into two groups:

- Soaps, waxes and fat - This group includes metallic soaps like calcium, lithium and sodium, waxes as beeswax and spermaceti wax, and the fats include fat acids such as stearic acid and palmitic acid.
- Polymeric films - This group includes synthetic substances like *Teflon* (PTFE).

Inorganic solid lubricants

Inorganic solid lubricants can be divided into three groups:

- Gelatinous solids - Materials like graphite have strong inter atomic forces in the same layer and a weak bond between consecutive layers, allowing sliding with little resistance.
- Soft solid mixtures - There is a wide range of materials of inorganic solids such as lead, calcium oxide, talc, silver iodide that can be used as lubricants.
- Surface protection by chemical reaction with the surface - Some materials create, by chemical reaction with the surface, a thin coating that prevents wear and facilitates the sliding. Among the most common surface coatings are chlorides, oxides, phosphates and sulphides.

2.1.3 Gaseous lubricants

Lubrication with gases is in many ways similar to lubrication with oils. Although both lubricants are fluids, the viscosity of the gases is much lower than the oils opposing to the compressibility that is much higher in the gases. This differences translates in a lower load capacity and film thickness in the contact. The list of gases used in this manner is extensive and includes air, steam, industrial gases, and liquid-metal vapours.

2.2 Additive package

Additives are added in the final stage of production to a lubricant oil in order to improve their natural properties or to provide new ones. Nowadays almost every lubricant has at least one additive, and the amount present in the final product may vary from almost negligible amounts up to 30%. Additives improve the lubricant properties, therefore play an important part in the development of modern power transmission mechanisms and engines. One additive can have one function or multiple functions, but in order to obtain the best properties of the lubricant the use of additives is imperative [4].

2.2.1 Viscosity index improving additives

Viscosity index improving additives are used in order to achieve a lower viscosity at low temperatures and/or higher at high temperatures. This improvement in the viscosity index is usually obtained by adding high molecular weight polymers such as butane polymers, esters of methacrylic acid and fat alcohols. This type of additive has a more noticeable effect in the viscosity at high temperatures, increasing it. This additives are used in engine oils, fluid for automatic gearboxes and hydraulic systems.

2.2.2 Anti wear and Extreme pressure additives

This group of additives has the purpose of reducing friction and surface wear in extreme lubrication conditions and can be divided in three smaller groups:

- Lubricity agents - are usually fat oils added to reduce the friction coefficient in limit film situations. The amount and type of the agent are specific for final application. Lubricity agents are used in applications with abrupt working conditions and to avoid situations of *stick and slip*.
- Anti wear additives (*AW*) - create a protective film on the surfaces by reacting with the metal. The objective is to reduce the wear when the lubricant film is broken.
- Extreme pressure additives (*EP*) - are used mainly in gears when contact pressure is superior to 700 MPa. The main objective is to minimize the adhesion between the surfaces when the lubricant film is broken and consequently avoid catastrophic failure. The formulation of this additives must be carefully thought since they induce lubricant thermal and chemical instability.

2.3 Physical properties of lubricant oils

In order to properly understand and evaluate the behavior of a lubricant oil their physical and rheological properties must be well known.

2.3.1 Viscosity

Viscosity is the most important property that needs to be considered when studying lubrication. It reflects the degree of internal friction or resistance that a fluid offers to internal shear deformation.

For laminar flow between two parallel surfaces with relative movement with no slip boundary condition, as shows Figure 2.1, the fluid suffers shear deformation and resists with an opposing force. The opposing force can be estimated by Newton's law for viscous flow (2.1) [5].

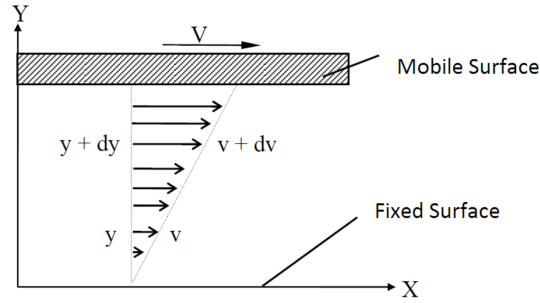


Figure 2.1: Speed gradient between two moving plates [4].

$$\tau_{xy} = \eta \cdot \frac{dv}{dy} \quad (2.1)$$

Where at any distance y from the fixed surface the fluid has the speed of v , and so, for a distance of $y + dy$ the speed is $v + dv$. The quotient $\frac{dv}{dy}$ is the rate of shearing strain is applied and is related to the shearing stress by the fluid property dynamic viscosity (η).

Newton's law takes the hypothesis that the speed varies linearly along yy , and for some fluids as water and several oils, this behavior can be experimentally verified for laminar flow. The fluids that show this behavior are designated *Newtonian fluids*. Fluids that do not verify the linear relation stated above, due to the presence of macro molecules or in severe working conditions, are designated *non-Newtonian fluids* [5]. For the *non-Newtonian* case, Newton's law doesn't describe the behavior of the fluid, therefore appear other rheological laws.

Viscosity index

The viscosity index is a qualitative parameter that appears from the need of classifying the influence of temperature variations in the lubricant oils viscosity. The most common model is the one proposed by *Dan and Davis* in 1929, represented in Figure 2.2.

It worth to note that the temperature scale (xx) is logarithmic and the viscosity scale (yy) is double logarithmic. The viscosity index assumes two lubricants, with the highest

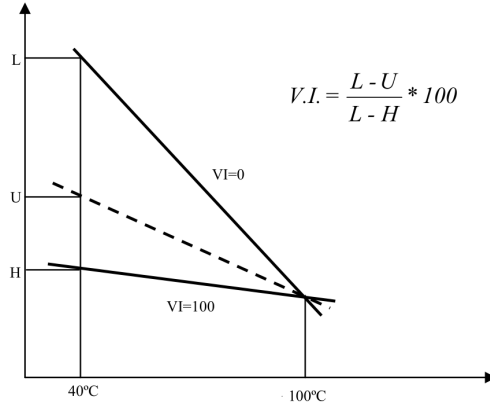


Figure 2.2: Viscosity index representation [4].

and lowest viscosity variations at time of the model creation as reference, being the one with the highest variation the correspondent to $IV=0$ and the one with lowest viscosity variation corresponding to $IV=100$ [4]. Nowadays, there are lubricants with VI higher than 100, therefore the method described above is not applicable. It must be kept in mind that the same viscosity index can be found in lubricants with different behaviors with temperature.

Thermoviscosity

Thermoviscosity represent the way viscosity varies with the temperature, and mathematically is give by equation (2.2).

$$\beta_T = \frac{d\nu}{d\theta} \quad (2.2)$$

There are several laws that attempt to mathematically represent this variation. The simplest approach was proposed by *Cammeron* (equation (2.3)):

$$\nu_1 = \nu_0 \cdot e^{-\beta \cdot \Delta\theta} \quad (2.3)$$

Where:

- ν_1 is the lubricant kinematic viscosity at temperature θ_1 ;
- ν_0 is the lubricant kinematic viscosity at temperature θ_0 ;
- β is the thermoviscosity coefficient;
- $\Delta\theta$ is the temperature variation ; $\Delta\theta = \theta_1 - \theta_0$.

This equation, although simple, is only valid to a small range of temperatures around the reference temperature θ_0 . Known the viscosity at two different temperatures, the equation (2.3) can be rearranged (equation (2.4)) in order to obtain the lubricant thermoviscosity coefficient:

$$\beta = -\frac{\ln\left(\frac{\nu_1}{\nu_2}\right)}{(\theta_1 - \theta_2)} \quad (2.4)$$

This value of β is an average for the temperature used, and therefore is not the most accurate.

As an alternative to the *Cameron's* law is equation (2.5), the *ASTM D341* standard [6]:

$$\text{Log}(\text{Log}(\nu + a)) = n - m \cdot \text{Log}(T) \quad (2.5)$$

Where:

- ν is the kinematic viscosity [cSt];
- T is the temperature [K];
- a, m and n are constants given by equations (2.6), (2.7) and (2.8), respectively.

$$a = 0.7 \quad \text{for mineral oils} \quad (2.6)$$

$$m = \frac{\text{Log} \left[\frac{\text{Log}(\nu_0 + a)}{\text{Log}(\nu_1 + a)} \right]}{\text{Log} \left[\frac{\theta_1 + 273}{\theta_0 + 273} \right]} \quad (2.7)$$

$$n = \text{Log}(\text{Log}(\nu_0)) + m \cdot \text{Log}(T_0) \quad (2.8)$$

Applying the definition of thermoviscosity (2.2), to equation (2.5), after some manipulation the equation (2.9) is obtained.

$$\beta_T = \frac{m}{T} \cdot \frac{(\nu + a) \cdot \text{Ln}(\nu + a)}{\nu} \quad (2.9)$$

The *ASTM D341* standard offers an accurate value for the viscosity knowing only the viscosity at two different temperatures the constant a .

An alternative to the previous laws, is the *Vogel law*, given by equation (2.10)

$$\nu = K \cdot e^{\left[\frac{b}{\theta + c} \right]} \quad (2.10)$$

Where:

- ν is the kinematic viscosity [cSt];
- θ is the temperature [°C];
- K, b and c are constants calculated from experimentally determined values of viscosity.

Figure 2.3 shows the three laws that can be used to represent the influence of temperature in lubricant viscosity. It is visible that the best approaches are given by the *ASTM D341* standard and by the *Vogel law*. This laws cannot be distinguished in Figure 2.3 due to overlapping in the results for the represented case. Although the similarity is high, the *Vogel law* offers a more trustworthy estimation since it considers three measured viscosities at different temperatures for the approximation, instead of the two used for the *ASTM* standard.

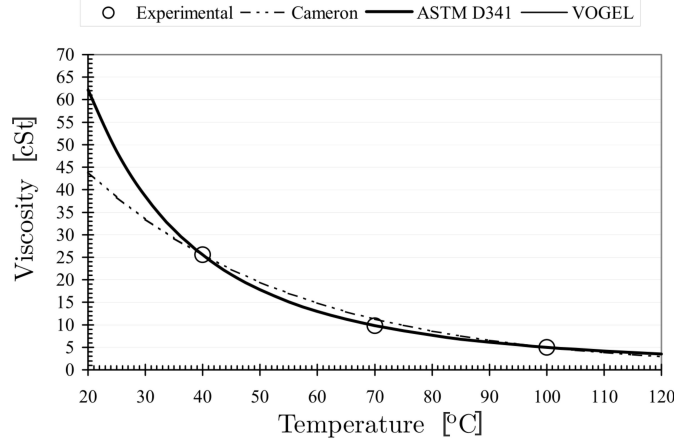


Figure 2.3: Kinematic viscosity for the same lubricant by Vogel, ASTM341 and Cameron [2]

Piezoviscosity

Pressure also has influence in the fluid viscosity. In EHD contacts, the pressure can reach 4 GPa, and therefore, piezoviscous properties are of great importance on the film thickness generation. When testing a lubricant, maintaining the temperature constant, and varying the pressure, the viscosity can be estimated using *Barus* law, (2.11):

$$\eta_S = \eta \cdot e^{(\alpha \cdot p)} \quad (2.11)$$

Where:

- η_S is the dynamic viscosity at pressure p [cSt];
- η_0 is the dynamic viscosity at pressure $p = 0$;
- α_0 is piezoviscosity coefficient [Pa^{-1}].

The *Barus* law does not take into account the connection between the coefficient α and the pressure, and therefore, for pressures higher than 0.5 GPa the equation (2.11) is inaccurate. Other, more accurate and complex relations have been developed in order to solve the problems of equation (2.11).

Viscosity variation with shear strain rate

The shear strain rate also affects the viscosity of the fluid. Inside the contact the lubricant suffers strain rates that can reach 10^7 s^{-1} . A fluid that does not change its dynamic viscosity independently of the strain rate applied, is considered *Newtonian*.

When operating at high strain rates, the lubricant viscosity usually decreases. This effect can be reversed when the lubricant rests or becomes permanent. In the last case, the lubricant loses its properties and is ruined. Fluids that change their properties with the shear strain rate are considered *non-Newtonian*. For this case the *Newton's* law is not applicable since it only considers a direct relation between the shear stress and strain.

Vitreous transition temperature

The lubricant viscosity tends to very high values at low temperatures. When a lubricant is cooled at constant pressure, its viscosity increases until it behaves like an amorphous

solid. The temperature where the transition from viscous fluid to amorphous solid occurs is called *Vitreous transition temperature*. This phenomena is also verified when, at constant temperature, the pressure is risen. The pressures inside a *EHD* contact are sufficient for the lubricant to behave as a solid, being one of the reasons why it is possible to maintain a lubricant film at high hertzian pressures.

2.3.2 Bulk density and specific gravity

The bulk density (Kg/m^3) is the ratio between the mass of a body and its volume. For lubricant oils, the variation with the temperature isn't very noticeable but shouldn't be neglected since it affects the dynamic viscosity. The bulk density at high temperatures can be estimated by equation (2.12) [4].

$$\rho = \rho_0 + \alpha_t \cdot (T - T_0) \quad (2.12)$$

ρ_0 and T_0 being the reference temperature and density, respectively and α_t the thermal expansion coefficient.

At the pressures present in *EHD* contacts, the oil viscosity suffers a considerable change. For mineral oils, considering the pressure effect, the bulk density can be calculated using *Dowson's* expression, given by equation(2.13) :

$$\rho = \rho_0 \left(1 + \frac{0.6 \cdot p}{1 + 1.17 \cdot p} \right) \quad (2.13)$$

2.3.3 Thermal properties

One of the objectives of lubrication is to remove heat from the contact, therefore this properties are important when studying the thermal balance of the system.

Thermal conductivity

Thermal conductivity (W/mK) is, by definition, the heat transferred due to a unitary temperature gradient per unit of time, in a normal direction to a surface of unitary area. This property has a linear variation with the temperature. A high thermal conductivity favours the heat evacuation.

Specific heat

Specific heat (J/KgK) quantifies how much heat is necessary to increase a single degree in the temperature of one unit of mass . A higher specific heat means a lower temperature variation for a specific amount of energy. For example: In the convergent zone of the contact, the higher the specific heat, the less the temperature rises.

Thermal diffusivity

This property quantifies the rate at which temperatures propagates in the interior of the a body. Thermal diffusivity is the ratio between thermal conductivity and the product of the bulk density by the specific heat, and is expressed in m^2/s .

Experimental campaign

3.1 Gearbox

The transfer gearbox that was tested is a part of vehicle power train and is installed at the end of the main gearbox as a speed reducer, increasing the torque, as shown in Figure 3.1. This gearbox also allows the vehicle where its installed to have four wheel drive and has a auxiliary exit shaft to power auxiliary equipment.

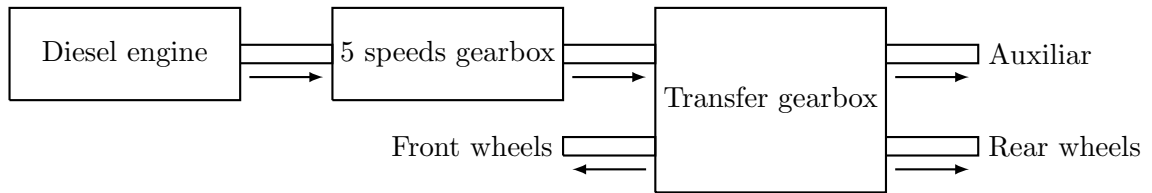


Figure 3.1: Scheme of the transfer gearbox installed on a vehicle.

The vehicle diesel engine has a maximum power output of 75 kW at 4310 rpm and a peak torque of 252 Nm at 2000 rpm. Although this is the engine that powers the vehicle, those are neither the torque or speed that power the transfer gear box. The transfer gearbox is installed after a five speed gearbox (speed reducer). The input torque is higher and the speed is lower than the engine output.

The gearbox in study has three shafts and five pinions, providing two different reduction ratios, both increasing the torque at the vehicle wheels:

- $i = \frac{1}{0.4363}$, wheels 1-2 and 3-5 geared;
- $i = \frac{1}{0.8438}$, wheels 1-2 and 2-4 geared.

The speed map of the transfer gearbox is shown in Figure 3.2a.

The geometrical dimensions of the gears can be calculated from the parameters presented in Table 3.1, using the equations present in Appendix B. The gear set installed in the gearbox is presented in Figure 3.2b.

3. Experimental campaign

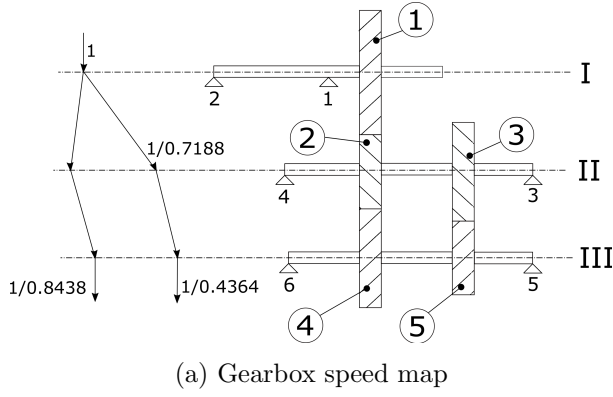


Figure 3.2: Installed gears.

Table 3.1: Geometrical parameters of the gearbox wheels.

Wheel	1	2	4	3	5
Modulus (m)	3.5	3.5	3.5	4	4
Number of teeth (z)	32	23	27	28	17
Addendum modification(x)	0.381	0.415	0.161	-0.240	0.051
Face width (b)	35	35	35	33.5	35
Pressure angle (α)	20	20	20	20	20
Helix angle (β)	20	20	20	20	20

The wheels number 1, 2 and 3 are keyed to the shaft and rotate at the shaft speed. Wheels 4 and 5 are mounted on needle roller bearings, allowing relative rotation between shaft III (Figure 3.1) and the wheels. The selection of the transmission ratio is made by a toothed ring and fork installed in shaft III. This solution keeps all the wheels rotating although one of the meshing pairs is not transmitting power, therefore is not accountable for load related power loss but is still a source of power loss by churning.

The gearbox uses four different types of rolling bearings, adding up to a total of 17 bearings. Although the original gearbox uses the 17 bearings, the gearbox tested only has 11 bearings installed in the current configuration. The remaining rolling bearings belong to the auxiliary exit that has been removed.

Table 3.2: Rolling bearings in the gearbox.

Number	Bearing type	Reference	C [kN]	C ₀ [kN]	d [mm]	D [mm]	Support
1	Ball	RMS 11	31.5	22	35	89	5
2	Ball	RMS 10	23.5	16.1	32	79	2
3, 4	Tapered roller	32306	67	53	30	72	3, 4
5	Cylindrical roller	NJ309E	91.5	62	45	100	1
6, 7	Ball	6307	26	18.3	35	80	6
8 to 12	Needle roller	K 38x43x27 F	30.5	68	38	43	*
13 to 15	Needle roller	K 14x18x13 F	9.15	12.5	14	18	*

*Between the wheels 4 and 5 and the shaft.

The gears are made of Din 15CrNi6 carburized and heat treated steel. The mechanical properties of the heat treated material are presented in Table 3.3 [3].

Table 3.3: Mechanical properties of the steel DIN 15CrNi6.

Property	Value	Unit
Tensile strength	620	MPa
Fatigue strength	1300	MPa
Poisson coefficient	0.3	
Specific weight	7850	kg/m ³
Young modulus	210	GPa
Hardness	58-62	HRC
Heat treatment depth	0.8	μm

3.2 Test rig

The gearbox was tested using a back-to-back gearbox test rig with recirculating power, therefore uses two identical gearboxes simultaneously, one for testing and a slave, in order to keep the speed at the entrance of the slave equal to the speed at the exit of the test gearbox.

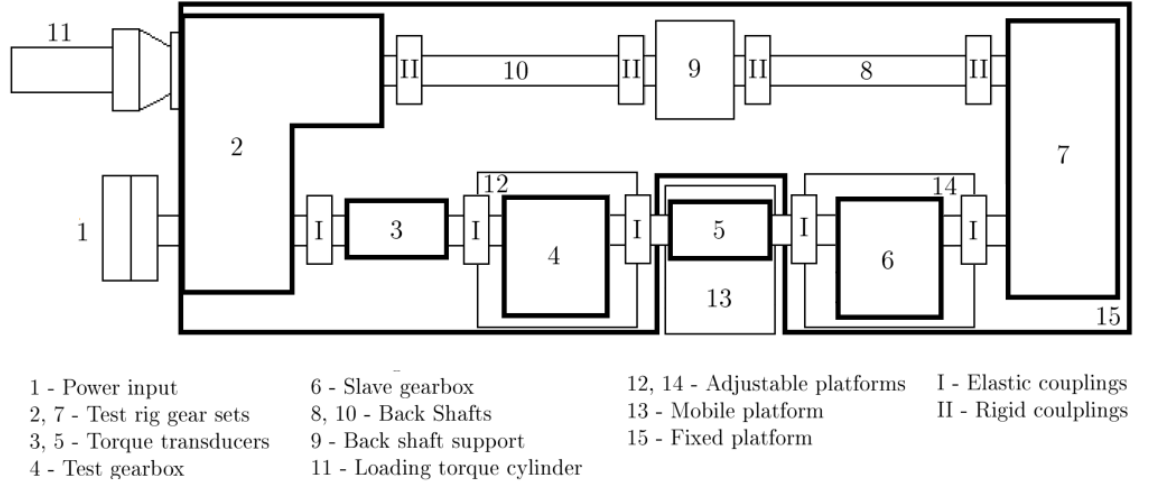


Figure 3.3: Scheme of the test rig[2].

The test rig allows the independent control of two test variables, speed and torque.

The speed is applied to the test rig by a three-phase motor associated with two timing belts, that reduces the rotational speed in a ratio of 1:2.36, as shown in Figure 3.4. These are installed at the point 1 of Figure 3.3. The speed control is made directly by a frequency inverter in an open ring system.

The torque is statically applied to the system by a pair of helical gears, installed in block 2 (Figure 3.3), by axial displacement of one of them, forced by a hydraulic cylinder. The hydraulic cylinder position, and consequently the torque, is controlled by a PID controller.



Figure 3.4: 3-phase electric motor and timing belt.

This mechanism allows the system to be statically loaded by a torsional displacement and therefore the electric motor only has to supply the power loss of the entire test rig. Blocks 2 and 7 connect the test and slave gearbox to the backshaft by a set of helical gears with a transmission ratio of 1:1 (Figure 3.3).

In order to test different gearboxes, the platforms where the slave and test gearboxes are installed are adjustable, the shaft that connects the gearboxes is also mobile.

To avoid the residual loads due to misalignment, there are four elastic couplings, one at each end of each gearbox. These couplings allow a small axial and angular displacement, transmitting the power with no considerable loss.

The test rig measures torque speed and temperature. It has two torque sensors installed in blocks 3 and 5, and also in this block there are two speed sensors (Figure 3.3). There are also several temperature sensors installed in order to obtain information about the operating conditions. The installed sensors measure the room temperature, test gearbox oil and wall temperatures and also the slave gearbox wall temperature.

The computer which controls the test rig also registers information at a frequency of 0.5Hz. The information includes time, input and output speed in rpm, input and output torque in Nm, and all the temperature readings of all the installed sensors in °C.

The room where the test rig is installed is equipped with a ventilation system. This system renews the air inside the room but does not control the air properties, therefore the tests can be influenced by the ambient conditions.

3.3 Lubricant properties

The lubricant oils chosen are the 75W140-A and the 75W90-A. The lubricants properties were measured at 3 different temperatures: 40, 70 and 100 °C and are presented in Table 3.4.

The oils are both poly- α -olephins (PAO), meaning that have the same base, although have different viscous properties but identical additive packages. The lubricants tested are presented in the market and are formulated for axle gears applications. Both oils fit the category API GL-5. This category designates the type of service characteristic of gears, particularly hypoids in automotive axles under high speed and/or low speed and high torque conditions.

Table 3.4: Properties of 75W140-A and 75W90-A [1; 7].

Property	Unit	Temperature	75W140-A	75W90-A
Viscosity	[cSt]	40	200.7	112.35
		70	61.86	36.7
		100	26.21	16.37
Thermoviscosity	[K ⁻¹ · 10 ³]	40	46.3	44.3
		70	33.2	31.3
		100	24.7	23.1
Piezoviscosity	[Pa ⁻¹]	40	1.498	1.387
		70	1.28	1.194
		100	1.142	1.072
VI	-	-	164.8	157.01
Density	[kg/m ³]	15	870	885
Thermal expansion coefficient	[$\alpha_T \cdot 10^{-4}$]	-	7.3	6.8

The additive package for these lubricants is based in Calcium, Magnesium, Phosphorus, Sulfur and Zinc. The amounts of each element present in the lubricants are given in Table 3.5.

Table 3.5: Additive package composition [1].

Chemical composition	Unit	75W140-A	75W90-A
Calcium (Ca)	[ppm]	18	33
Magnesium (Mg)	[ppm]	1087	1093
Phosphorus (P)	[ppm]	1622	1686
Sulfur (S)	[ppm]	23.3	22.8
Zinc (Zn)	[ppm]	7	12

The two tested oils contain high amounts of Sulfur and Phosphorus, usually used as extreme-pressure (EP) and anti-wear (AW) additives. Magnesium is also present in the chemical compositions and its effect is related to detergent and dispersant additives.

In order to obtain the viscosity at the working temperature the *Vogel* law (equation(2.10)) offers the best results, since this law uses three viscosity values measured at different temperatures (40, 70 and 100 °C) to approximate the lubricant viscosity behavior with temperature. The reference points are given in Table 3.4 [1]: The viscosity variation with temperature is shown in Figure 3.5.

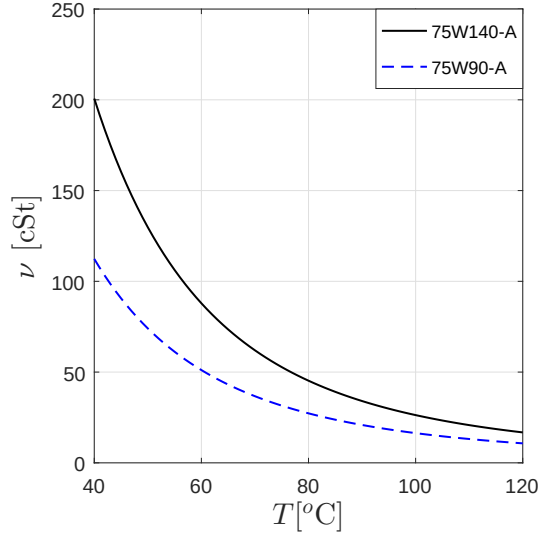


Figure 3.5: Viscosity function of temperature for 75w140-A and 75w90-A

3.4 Test parameters

The gearbox is to be tested in similar conditions to what is expected to work when installed in the vehicle. It makes sense to use the higher reduction ratio in the transfer gearbox in the tests, since when the vehicle is in need of torque, this should be the used ratio, therefore was assumed that the primary gearbox is in first gear and the vehicle is moving at a maximum speed of 15 km/h, climbing an incline of 8°. For this condition, the input shaft is rotating at 800 rpm and the torque applied is 250 Nm. These are the limit conditions defined for the experimental procedures and add up to a maximum power of 21 kW. The remaining tests are to be performed at lower power, thus the nine test points were defined, as follows in matrix (3.1):

$$\begin{bmatrix} 600/150 & 600/200 & 600/250 \\ 700/150 & 700/200 & 700/250 \\ 800/150 & 800/200 & 800/250 \end{bmatrix} \quad n_{in}[\text{rpm}]/TQ_{in}[\text{Nm}] \quad (3.1)$$

The corresponding power to the tested conditions are given in matrix (3.2)

$$\begin{bmatrix} 9.425 & 12.566 & 15.708 \\ 10.996 & 14.661 & 18.326 \\ 12.566 & 16.755 & 20.944 \end{bmatrix} \quad P_{in}[\text{kW}] \quad (3.2)$$

3.5 Experimental procedures

3.5.1 Test rig calibration

Before the tests were performed, the slave gearbox was set to neutral and the two torque sensors were calibrated.

After the calibration it was found that the sensors were not providing accurate measurements. For a static condition, the torques measured at the input and output shafts must obey to the gear ratio of the gearbox. This condition was not verified because the sensors are measuring less than 10% of their nominal load and consequently working in a non-linear zone. Since the sensors are calibrated to work in high torque conditions, which is not the case, where their behavior is linear, there is the need to apply a correction factor to the measured input torque, in order to reduce to the minimum the effects of the non-linearity of the sensors. In order to determine the correction factor, 20 values of input and output torque were registered for 4 different torques close to the ones to be tested. The values of the torque ratio for each load were averaged and the final correction factor was chosen for the mid point of the deviations found. The correction applied to the input torque is given by equation (3.3).

$$Tq_{in.c} = Tq_{in} \cdot 0.9665 \quad (3.3)$$

Where $Tq_{in.c}$ is the corrected input torque. From here on, any reference to the input torque is referent to the corrected value $Tq_{in.c}$ and not the sensor read value Tq_{in} .

3.5.2 Definition of test duration

In order to achieve accurate results, it was required that the system reach thermal equilibrium. When the thermal equilibrium is reached, the working conditions stabilize and it is possible to define a period of time where the variables in study are nearly constant.

Experimentally is impossible reach, in a acceptable test time, a point where the oil temperature is constant, since the ventilation system with which the room test room is equipped cannot evacuate all the heat produced by the test rig. As consequence, the test room temperature rises during the warm up period as well as the test period. In order to approach as much as possible a stabilized condition, it was defined that the difference between the oil and ambient temperature must be constant during at least 30 min.

The verification of the test stabilization temperature was made by breaking the 30 minute interval in 5 minute interval, then averaging values of 5 minutes intervals and afterwards averaging the averages. This procedure allows a more detailed appreciation of the temperature behavior during the test time. A test is considered stabilized if the variation of the relative temperature is under $0.5^{\circ}\text{C} / 30\text{min}$. The data registered for this 30 minutes will then be used to evaluate the gearbox behavior and as starting point to the power loss model.

Preliminary tests were performed at the maximum power and it was verified that at the end of 240 minutes the temperature difference variation was less $0.5^{\circ}\text{C} / 30\text{min}$, as shows Figure 4.2a. After the warm up period the test continues for another 30 minutes, and the average of the values measured in this period are then used to perform the posterior calculations.

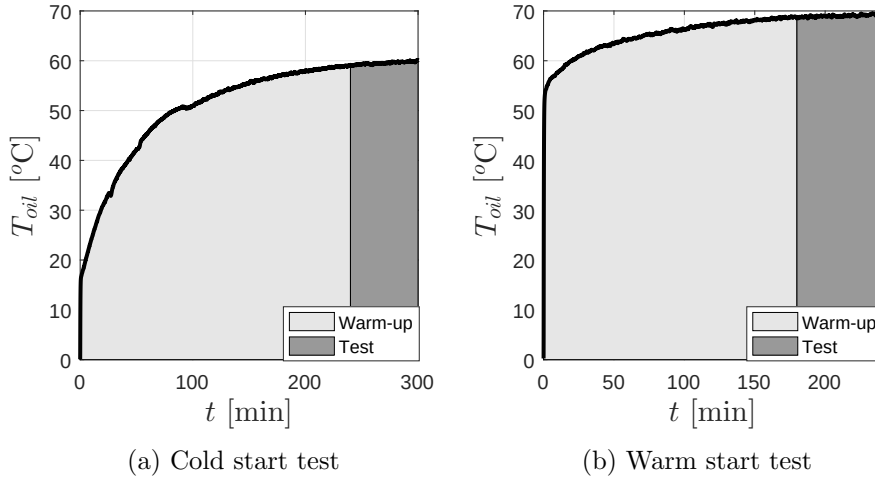


Figure 3.6: Oil temperature evolution during the whole test

There is the possibility to perform two tests per day. The second test is performed immediately after the end of the first one and must be of a higher power in order to guarantee that the stabilization temperature is higher. Since the gearbox box is already warm, this second test only needs a 210 minutes warm up period, as shows Figure 3.6b.

3.6 Test sequence

Before the test sequence, for each lubricant, the gearbox was washed and degreased with petrol ether in order to eliminate wear particles and every residue from the lubricants previously tested. After the washing stage the gearbox was left to dry at room temperature. When dry, the gearbox was mounted in the test rig and filled with 2 liters of lubricant. The same procedure was followed for the slave gearbox at the beginning of the experimental campaign. The lubricant used in this gearbox was PAOR 320, also used in “*Gearbox power loss*” [8–10].

The tests were performed starting at the lowest power and progressively increasing until the maximum. The test sequence is shown in table ??.

Table 3.6: Test sequence for the lubricant 75W140-A

Test number	Input speed	Input torque	Test time	Date	Start time
1	600	150	300	1/3/17	8:59
2		200	240	1/3/17	14:12
3		250	240	2/3/17	14:41
4	700	150	300	2/3/17	9:35
5		200	300	3/3/17	9:05
6		250	240	3/3/17	14:13
7	800	150	300	6/3/17	11:51
8		200	300	6/3/17	13:28
9		250	300	7/3/17	9:54

Table 3.7: Test sequence for the lubricant 75W90-A.

Test number	Input speed	Input torque	Test time	Date	Start time
1	600	150	300	5/6/17	15:06
2		200	300	6/6/17	9:46
3		250	240	6/6/17	14:56
4	700	150	300	7/6/17	9:50
5		200	240	7/6/17	14:56
6		250	240	8/6/17	14:46
7	800	150	300	8/6/17	9:42
8		200	300	9/6/17	9:28
9		250	240	9/6/17	14:54

For each test, the test rig records the input and output torque and speed as well as the tested gearbox oil and wall temperature, the slave gearbox wall temperature and the ambient temperature.

3.7 Results

The measured values in the test period were averaged in order to minimize the error associated with the measuring equipment and singular anomalies in the test rig control mechanism. All the test data used in this section are present in the Appendix A.

3.7.1 75W140-A

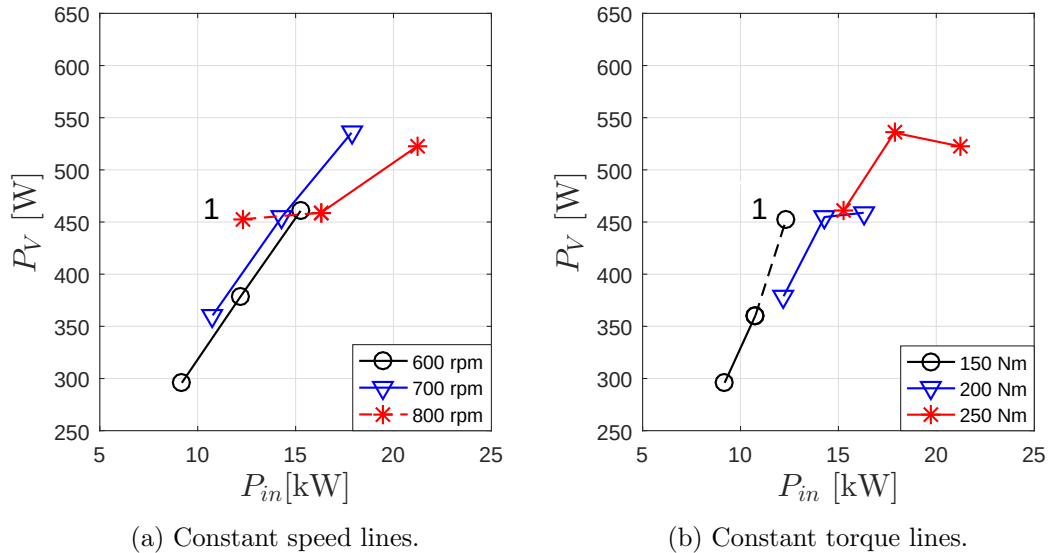


Figure 3.7: Power loss function of input power for 75W140-A (1-unstabilized test).

Figures 3.7a and 3.7b show the evolution of the power loss with the variation of torque and speed, respectively. It is visible a almost linear tendency in the power loss, in Figure 3.7a, when the input power varies with the input torque. When evaluating the behavior

3. Experimental campaign

at constant torque, the power loss does not follow a linear pattern, presenting a tendency to grow at a constant rate until the 15 kW of input power and then the growth slows down when varying the speed from 700 to 800 rpm.

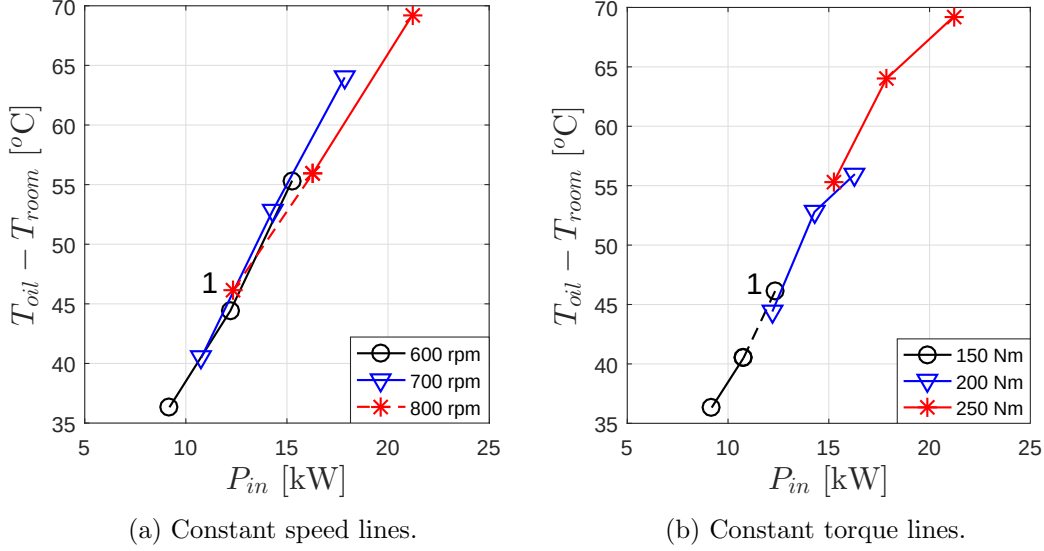


Figure 3.8: Relative stabilization temperature function of input power for 75W140-A (1-unstabilized test).

Figure 3.8 represents the variation of the relative stabilization temperature with the input power. The lines in Figures 3.8a and 3.8b almost overlap and reveal a direct proportionality relation between the two variables. In Figure 3.7a, the line at 800 rpm presents a slight deviation from the other two. The tests at 800 rpm already revealed in Figure 3.8 some non linearities. The fact that the power loss for this speed does not vary proportionally with the input power, translates in reduction of the slope for the 800 rpm line in Figure 3.7a.

Comparing Figures 3.7 and 3.8, it is visible that the power loss and the relative stabilization temperature for the test at 150 Nm and 800 rpm (1 from Figure 3.7 and Figure 3.8) do not present the same tendency. The high power loss for this test should be reflected in the stabilization temperature. The lack of conformity between the variables leads to the conclusion that the test is not stabilized.

Figure 3.9, represents the evolution of the efficiency of the gearbox measured in the tested range. The efficiency varies only 1% from the lower to higher input power and in Figure 3.9a shows a linear dependence with the input torque, mainly for the two lower rotational speeds.

When evaluating the behavior in function of the rotational speed, the efficiency tends to slightly decay from the first to the second speed and then increase significantly for the higher speed, with the exception of the 150 Nm line due to the influence of the unstabilized test (marked as 1 in Figure 3.9), as shows Figure 3.9b. This behavior is directly related to the observations of Figure 3.7b, where was stated that the power loss growth slows down as the speed increases. The non-linearity represented in Figure 3.7 explains the high efficiency registered for the tests at 800 rpm.

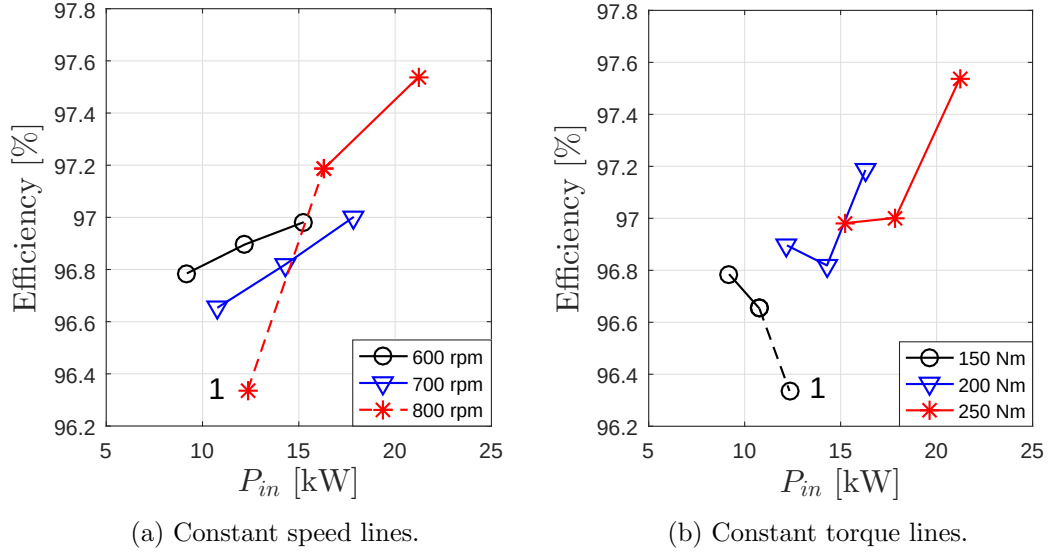


Figure 3.9: Efficiency function of input power for 75W140-A (1-unstabilized test).

3.7.2 75W90-A

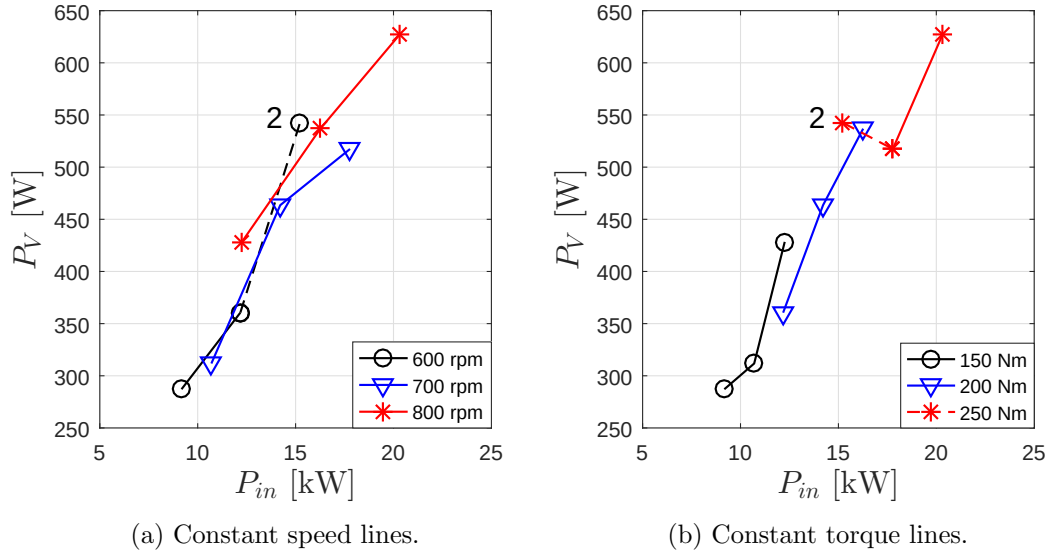


Figure 3.10: Power loss function of input power for 75W90-A (2-unstabilized test).

The measured power loss for the lubricant 75W90-A is represented in Figure 3.10. Figure 3.10a shows that, for this lubricant, the power loss grows proportionally with the input power, and by analyzing 3.10b it is visible that the growth of the power loss presents a similar rate when varying the speed at different levels. The test performed at 250 Nm and 600 rpm (test 2 in Figure 3.10) does not obey the pattern, presenting a higher power loss than the tendency suggests. This test passed the validation process but after a more refined analysis was verified that the relative temperature was still rising, therefore the stabilization temperature hadn't been reached leading to the conclusion that this test is not valid.

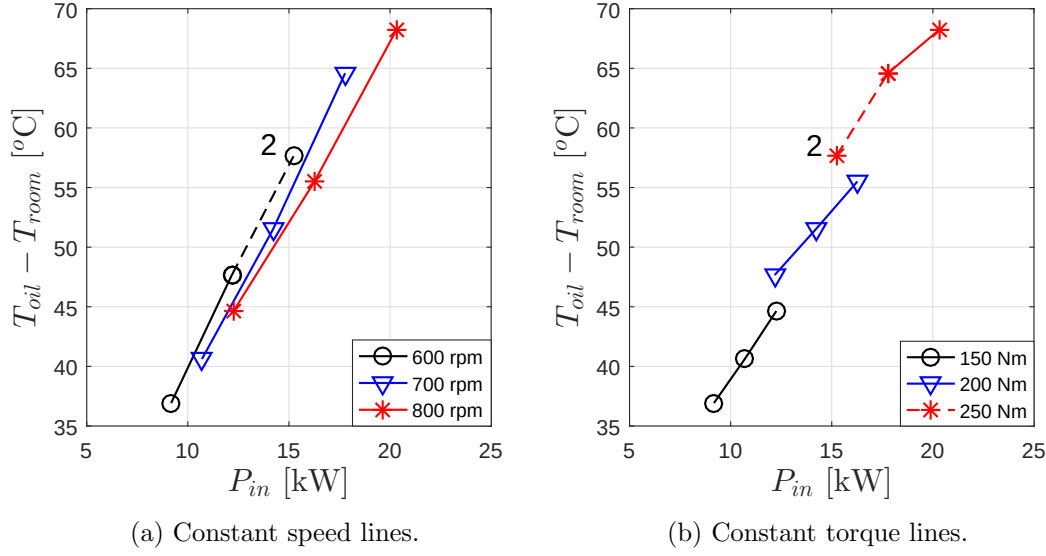


Figure 3.11: Relative stabilization temperature function of input power for 75W90-A (2-unstabilized test).

The relative stabilization temperature, represented in Figure 3.11, rises with the increasing power loss for all the represented cases. It is worth to note that for constant torque lines, visible in Figure 3.11b, the line do not present the same growth rate despite the fact that in Figure 3.11a the lines look almost coincident.

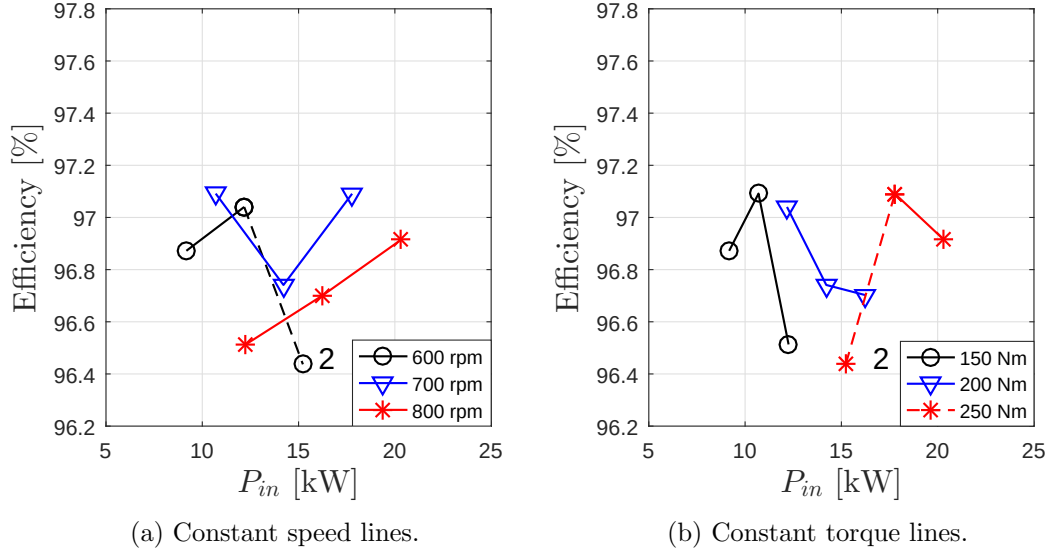


Figure 3.12: Efficiency function of input power for 75W90-A (2-unstabilized test)

Figure 3.12 displays the working efficiency of the gearbox lubricated with 74w90. The measured efficiency is restricted to a 0.8% range. By analysis of Figure 3.12a, it is verified that it does not follow any pattern for the 600 and 700 rpm tests, and for the higher speed the efficiency tends to improve with the growing torque. Figure 3.12b shows that, for the constant torque tests, when increasing the speed from 700 to 800 rpm, the efficiency is reduced independently of torque applied.

3.7.3 Lubricant comparison

Figure 3.13, 3.14 and 3.15 show the difference, in absolute values, between the experimentally measured values of power loss, stabilization temperature and efficiency for the 75W140-A and 75W90-A lubricants, respectively.

The values represented in the following Figures were calculated according to equation (3.4).

$$\Delta X = X_{75W140-A} - X_{75W90-A} \quad (3.4)$$

Where X is the variable in study in each figure.

Note that the values marked with 1 and 2 in Figures 3.13 to 3.15 have the influence of unstabilized tests.

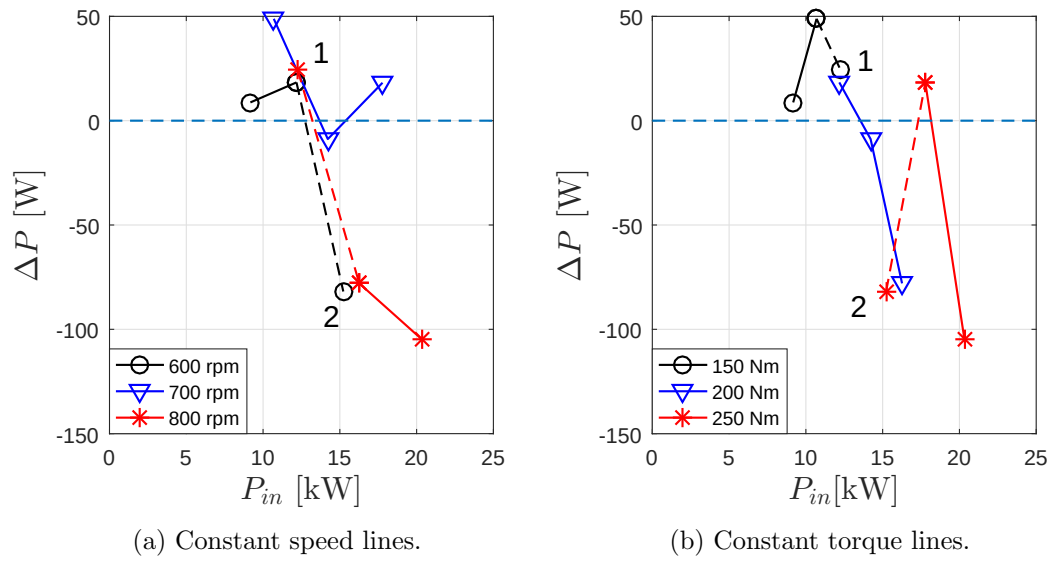


Figure 3.13: Power loss difference between 75W140-A and 75W90-A. Values 1 and 2 are affected by unstabilized tests.

Figure 3.13 show the power loss difference between the 75W140-A and the 75W90-A. Figure 3.13 shows that the lubricant 75W140-A has a higher power loss than the 75W90-A for low torque and speed conditions. For the 200 and 250 Nm conditions, the 75W90-A presents a higher power loss than the 75W140-A, that exceeds 100 W for the highest input power condition. In general, the 75W140-A presents better performance, since the difference in the power loss for the high power conditions, where it has the better behavior, is more than double the difference for the low power condition, where 75W90-A presents the better behavior.

Figure 3.14 shows the difference between the relative stabilization temperatures for the lubricants tested. In comparison, the 75W90-A presented a higher stabilization temperature for the tests at 600 rpm and two of the tests at 700 rpm. For the lower torques the stabilization temperature is lower for the 75W140-A, and as the torque increases, the 75W90-A tends to stabilize at lower temperatures.

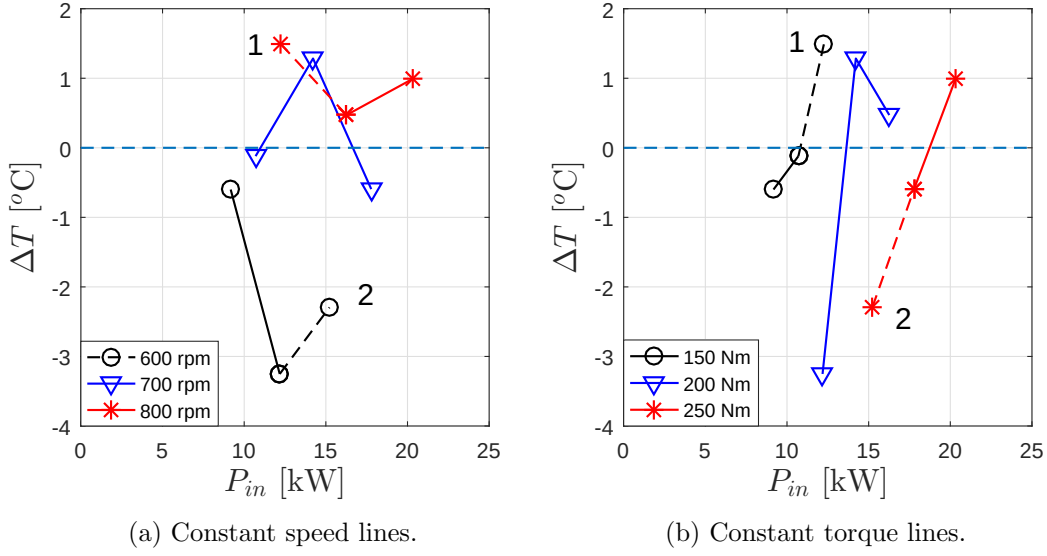


Figure 3.14: Stabilization temperature difference between 75W140-A and 75W90-A. Values 1 and 2 are affected by unstabilized tests.

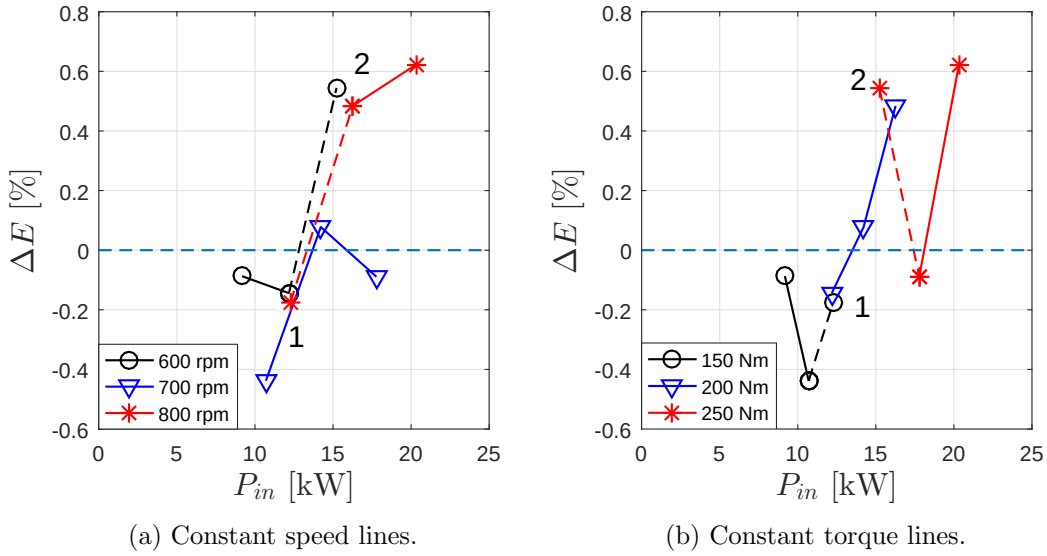


Figure 3.15: Efficiency difference between 75W140-A and 75W90-A. Values 1 and 2 are affected by unstabilized tests.

Figure 3.15 shows that for low input power the lubricant 75W90-A shows a better efficiency at 700 rpm and 150 Nm (0.45% better), but when increasing the power, the gearbox efficiency for the 75W140-A tends to improve more, and for the high power conditions presents an efficiency more than 0.5% superior to 75w90, with the exception of the 250 Nm and 800 rpm condition.

Power loss model

4.1 Power loss in gearboxes

The overall gearbox power loss is a combination of different power loss sources. Each moving component of a gearbox dissipates energy, thus leading to smaller power in the gearbox output compared to the input. Gearboxes are a combination of gears, bearings, shafts, seals, lubricant and a housing and each one is affected by the working conditions. Nowadays there are a virtually infinite number of combinations of components and building solutions, making the accurate prediction of power loss a quite difficult task [2].

Every moving part or with relative movement in a gearbox dissipates energy. The power loss can be divided in two big groups, load and no-load dependent losses. Components such as gears and bearing present both types of losses, and components like seals dissipate power independently of the load applied to the gearbox.

In a traditional gearbox, power loss consists in gear, bearings, seals and auxiliary losses [11]. The gear and bearing losses can be divided in load and no-load losses. In the case of gears, the load related losses are related to the meshing effects (P_{VZP}), and the no-load are due to churning (P_{VZ0}). For bearings (P_{VL}), although the model described ahead divides the frictional torque, for practical effects it is considered as a load related loss. The auxiliary loss, (P_{VX}), are related to auxiliary equipment such as oils pumps [12].

$$P_V = \begin{cases} P_{VZ} \begin{cases} P_{VZP} - \text{Gear meshing loss} \\ P_{VZ0} - \text{Gear churning loss} \end{cases} \\ P_{VL} - \text{Rolling bearing loss} \\ P_{VS} - \text{Seal loss} \\ P_{VX} - \text{Auxiliary loss} \end{cases}$$

4.1.1 Power loss in gear meshing

The gear meshing loss is usually the main source of power loss in a transmission system. According to the classical power loss model, the power loss only depends of the transmitted power (P_{in}), the mean value of friction along the meshing line (μ_{mz}) and the gear loss factor H_v , as shown in equation (4.1):

$$P_{VZP} = P_{in} \cdot \mu_{mz} \cdot H_V \quad (4.1)$$

The input power, P_{in} , is the imposed condition, and does not require any further calculation. The meshing friction coefficient, μ_{mz} , is extremely complex in the case of gears

due the presence of different lubrication regimes along the meshing line. The lubricant regime is defined by the specific film thickness (Λ), as follows [4]:

- Boundary film: $\Lambda < 0.7$;
- Mixed film: $0.7 < \Lambda < 2.0$;
- Full film: $\Lambda > 2.0$.

For boundary film conditions, the normal force to contact is not sustained by the lubricant film, but instead by the peaks of the surfaces roughness, causing a higher friction and surface wear due to metal-metal contact. For full film conditions, the normal force is completely supported by the EHD lubricant film. For mixed film conditions, the normal force is supported by the lubricant film and the roughness peaks, achieving the minimum friction coefficient of the three types of lubrication regimes, as shows Figure 4.1.

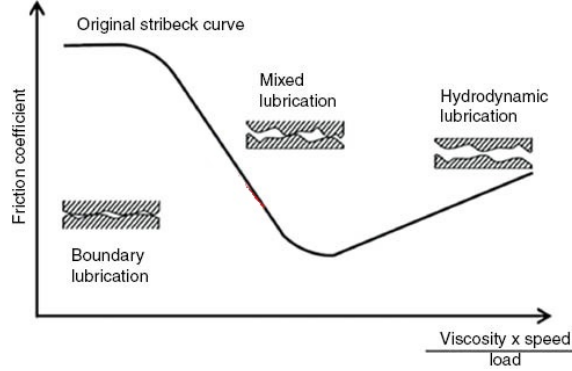


Figure 4.1: Stribeck curve.

The film thickness at the center of a linear contact, h_0 , can be calculated by equation (4.2) [13].

$$h_0 = 0.975 \cdot \frac{[\alpha_{oil} \cdot (U_1 + U_2)]^{0.727} \cdot R_x^{0.364} \cdot (l \cdot E^*)^{0.091}}{F_n^{0.091}} \quad (4.2)$$

During the approach to the center of the contact, the lubricant is subjected to high strain rates, generating heat. This heat rises the lubricant temperature, reducing the viscosity and consequently the film thickness. In order to consider this effect, a thermic correction coefficient, ϕ_T given by equation (4.3), is applied to the film thickness [4].

$$\phi_T = \left[1 + 0.1 \cdot \left(1 + 14.8 \cdot \frac{|U_1 - U_2|}{(U_1 + U_2)} \right)^{0.83} \cdot \left(\frac{\beta_T \cdot \eta_0 \cdot (U_1 + U_2)^2}{K} \right)^{0.64} \right]^{-1} \quad (4.3)$$

The specific film thickness, Λ , takes into account the roughness of the surfaces in contact, and is given by equation (4.4) [13].

$$\Lambda = \frac{h_0 \cdot \phi_T}{\sigma} \quad (4.4)$$

Depending on the position on the meshing line, conditions such as normal force, relative speed and curvature of the surfaces change, modifying the specific film thickness, and therefore the lubrication regime and the friction coefficient, as is visible in Figure 4.2,

that shows the variation of the film thickness, sliding speed and friction coefficient along the meshing line of a spur gear pair identical to meshing pair z_1/z_2 present in the tested gearbox.

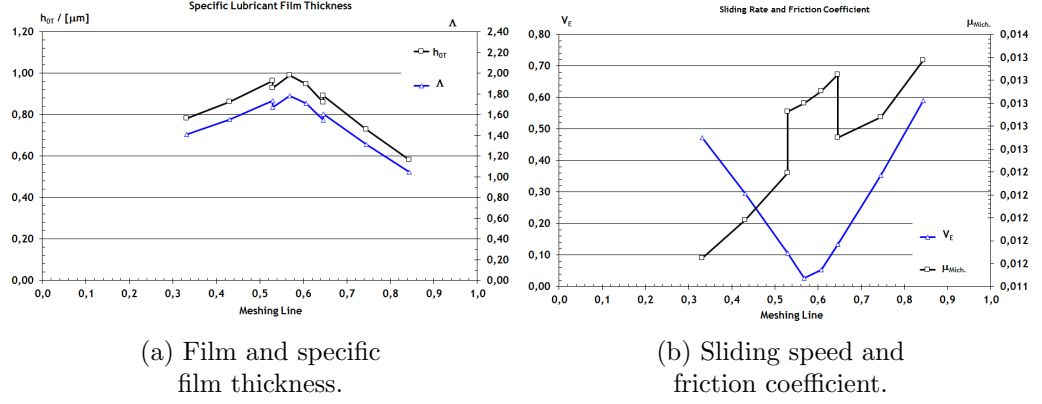


Figure 4.2: Evolution of properties along the meshing line.

Another parameter that also has a relevant effect on power loss is the surface roughness. Not only its amplitude, that affects directly the specific film thickness, but also its orientation. On the one hand, for the same working conditions, longitudinal asperities ease the sliding, lowering the friction coefficient, opposing to the transverse orientation that promotes more friction. On other hand, asperities in the longitudinal direction “make way” for the lubricant to escape the high pressure zone, lowering the film thickness, opposing to the transverse oriented, that “force” the lubricant into the contact.

The complexity of the problem makes it very hard to solve it analytically, instead the common practice is to use empirical equations to estimate the average coefficient of friction of meshing gears. Equation (4.5) is the result of twin disks machine testing or gear power loss experimental testing [14].

$$\mu_{mz} = 0.048 \cdot \left(\frac{F_{bt}/l_{min}}{v_{\Sigma C} \cdot \rho_C} \right) \cdot \eta_{oil}^{-0.05} \cdot R_a^{0.25} \cdot X_L \quad (4.5)$$

Where:

- F_{bt} is the force normal to the contact [N];
- l_{min} is the contact length [mm];
- $v_{\Sigma C}$ is the sum velocity at the pitch point [m/s];
- ρ_C is the equivalent curvature radius at the pitch point [mm];
- η_{oil} is the oil dynamic viscosity at the injection temperature [Pa·s];
- R_a is the surface roughness [μm];
- X_L is the lubricant parameter.

The contact length can be calculated as a function of the face width, b , and the base helix angle, β_b , as shown in equation (4.6).

$$l_{min} = \frac{b}{\cos(\beta_b)} \quad (4.6)$$

By analyzing the figure 4.3, the equation for the teeth curvature radius at pitch point, can be derived as a function of the base radius and the meshing line length.

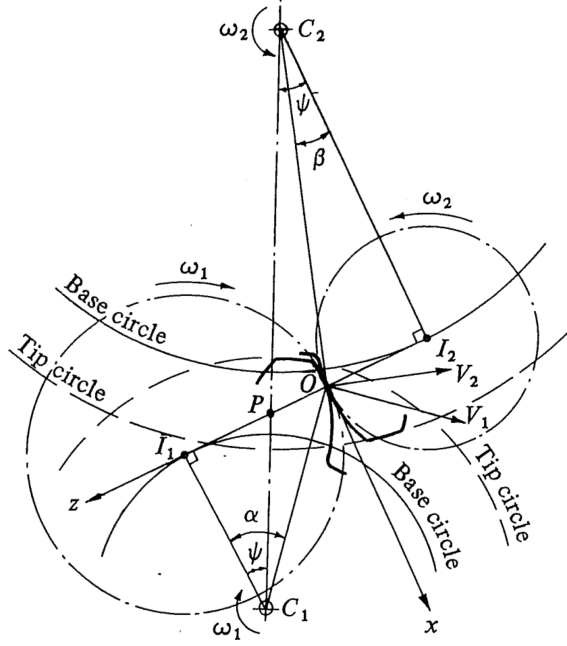


Figure 4.3: Scheme of meshing gears [15].

According to figure 4.3, the curvature radius, ρ_1 and ρ_2 are the length $\overline{I_1P}$ and $\overline{I_2P}$, given by equation (4.7)

$$\rho_1 \equiv \overline{I_1P} = \sqrt{C_1P^2 - C_1I_1^2} \quad \text{and} \quad \rho_2 \equiv \overline{I_2P} = \sqrt{C_2P^2 - C_2I_2^2} \quad (4.7)$$

The velocity at the pitch point, v_C , can be calculated by equation (4.8)

$$v_1 = \rho_1 \cdot \omega_1 \quad \text{and} \quad v_2 = \rho_2 \cdot \omega_2 \quad (4.8)$$

The gear loss factor H_V is characteristic of the meshing pair and is independent of the lubrication conditions. Marques [16] used a quasi-static local elastic model to numerically calculate the load distribution, $F_N(t, y)$, for a single meshing pair of teeth. Using a Coulomb model, the power loss per unit of face width can be obtained by equation (4.9), using the normal force $F_N(t, y)$ multiplied by the local friction coefficient and the sliding speed $v_g(t, y)$.

$$p_{VZP}^{single} = F_N(t, y) \cdot \mu(t, y) \cdot v_g(t, y) \quad (4.9)$$

Integrating equation (4.9) along the face width, the power loss of a meshing teeth pair is given by equation (4.10).

$$P_{VZP}^{single} = \int_0^b [F_N(t, y) \cdot \mu(t, y) \cdot v_g(t, y)] dy \quad (4.10)$$

Since more than one teeth pair can be in contact at the same time, the total power loss is obtained by superposition of equation (4.10). The functions are at a distance of p_{bt} or a mesh period (T_{mesh}) from each other. This superposition is translated in equation (4.12).

$$T_{mesh} = \frac{2\pi}{\omega_i \cdot z_i} \quad (4.11)$$

$$P_{VZP}^{Total} = \sum_{i=-\text{floor}(\epsilon_\alpha+\epsilon_\beta)}^{\text{floor}(\epsilon_\alpha+\epsilon_\beta)} P_{VZP}^{single} \cdot (t - i \cdot T_{mesh}) \quad (4.12)$$

Taking into account the periodicity in time, (T_{mesh}) given by equation (4.11), the global meshing power loss is given by equation (4.13).

$$P_{VZP}^G = \frac{\int_0^{T_{mesh}} P_{VZP}^{total}(t) dt}{T_{mesh}} \quad (4.13)$$

Rearranging equation (4.1), the H_V factor can be derived, as stated in equation (4.14).

$$H_V = \frac{P_{VZP}^G}{P_{in} \cdot \mu_{mz}} \quad (4.14)$$

4.1.2 Churning loss

The gears are partially immersed in a oil bath, and so, their rotation generates a viscous flow around the wheels while splashing oil to lubricate the gearing, dissipating energy. The power loss by this phenomena is specially difficult to estimate due to the influence of parameters like lubricant aeration and the housing geometry that vary from case to case [17].

The churning power loss model for individual wheels proposed by *Changenet et al* [18], is proved to be a reliable model to predict the churning power loss. According to this model, the power loss for the entire gearbox is the sum of the losses of each wheel, given by equation (4.15), being k the number of wheels of the gearbox.

$$P_{VZ0} = \sum_{i=1}^k C_{ch|i} \cdot \omega|i \quad (4.15)$$

Where $C_{ch|i}$ is the torque loss of the wheel i given by equation (4.16) and $\omega|i$ is the rotational speed.

$$C_{ch} = \frac{1}{2} \cdot \rho_{oil} \cdot \omega^2 \cdot R_p^3 \cdot S_m \cdot C_m \quad (4.16)$$

Where S_m is the immersed surface of the wheel, C_m is the drag dimensionless group, R_p is the wheel reference radius and ρ_{oil} is the lubricant bulk density at the working temperature. The dimensionless drag group C_m is given by the equation:

$$C_m = \psi_1 \cdot \left(\frac{h}{D_p}\right)^{\psi_2} \cdot \left(\frac{V_0}{D_p^3}\right)^{\psi_3} \cdot Fr^{\psi_4} \cdot Re_c^{\psi_5} \cdot \left(\frac{b}{R_p}\right)^{\psi_6} \quad (4.17)$$

The parameters Re and Fr are the Reynolds and Froude number, respectively, given by equations (4.18) and (4.19) :

$$Re = \frac{R_p \cdot b \cdot \omega}{\nu} \quad (4.18)$$

$$Fr = \frac{R_p \cdot \omega^2}{g} \quad (4.19)$$

The coefficients ψ_i are obtained from experimental results and vary according to the working conditions and are present in table (4.1). Four sets of coefficients were determined according to the nature of the flow regime (laminar or turbulent) and the parameter γ , given by equation(4.20).

$$\gamma = \omega^2 \cdot (R_p \cdot b \cdot m)^{1/3} \quad (4.20)$$

Although this parameter has no proven physical meaning, dimensionally resembles the centrifugal acceleration [19].

Table 4.1: Parameters for equation (4.1)

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

For the gap not covered by the conditions of Table 4.1, $750 < \gamma < 1250$, the C_m parameter results from an interpolation of the values obtained for $\gamma = 750$ and $\gamma = 1250$.

For the immersed surface of the wheel, S_m , the original equation proposed by *Changenet et al.* [18] only was tested for spur gears. In order to extend the model to also helical gears, the geometrical parameter was changed and verified by Gautier Leprince [20]. This parameter can now be defined by equation (4.21)

$$S_m = R_p^2 \cdot (2 \cdot \theta - \sin(2\theta)) + D_{pb} \cdot \theta + 2 \frac{z \cdot \theta \cdot H_{tooth} \cdot b}{\pi \cdot \cos(\alpha) \cdot \cos(\beta)} \quad (4.21)$$

Where θ can be derived from the immersed height of the wheel, h , by equation (4.22).

$$\theta = \arccos\left(\frac{1-h}{R_p}\right) \quad (4.22)$$

These equations generally present a good representation of the actual phenomena, but there are some aspects the are not accounted for, such as lubricant aeration and the rotation sense.

Lubricant aeration

For oil sump lubricated gears, the movement of the wheels mixes air with the lubricant oil, creating small air bubbles. This phenomenon is called lubricant aeration, and is proven to be closely related to the churning power loss, although is very sensitive to the lubricant formulation.

Rotation sense

The churning power loss model calculates the loss of each wheel individually, but inside the gearbox there is interaction between the gears, and it affects the power loss. This model is proven to achieve good results for wheels rotating clockwise (Figure 4.4). For counter-clockwise rotation, the losses increase, and are higher than the sum of the losses of each wheel [19].



Figure 4.4: Rotation senses of a meshing pair [2].

4.1.3 Power loss in rolling bearings

Rolling bearings support the loads transmitted from the gears to the shafts, and allow the rotation of the shafts, restraining all the other shaft degrees of freedom. The selection of a rolling bearing for a specific application is restrained by the working conditions such as load type and intensity, working speed and design restrictions. Depending on these conditions the best bearing for a specific application can be:

- Deep groove ball bearing;
- Cylindrical roller bearing;
- Tapered roller bearing;
- Needle roller bearing.

The power loss in rolling bearings can be estimated by equation (4.23).

$$P_{VL} = M^{rol} \cdot n \cdot \frac{\pi}{30} \cdot 10^{-3} \quad (4.23)$$

SKF model (2004)

The SKF model [21] has gone through a few improvements through the years, and the most recent introduces an advanced and accurate method to estimate the frictional torque in a rolling bearing. According to SKF [21], the total frictional torque has four different sources:

- M_{rr} The rolling frictional torque;
- M_{sl} The sliding frictional torque;
- M_{seal} The frictional torque from seal;
- M_{drag} The frictional torque from drag, churning and splashing losses.

The rolling bearing total frictional torque is given by equation (4.24).

$$M^{rol} = M_{rr} + M_{sl} + M_{seal} + M_{drag} \quad (4.24)$$

Rolling frictional torque

The rolling frictional torque is given by equation (4.25).

$$M_{rr} = \phi_{ish} \cdot \phi_{rs} \cdot G_{rr} \cdot (\nu \cdot n)^{0.6} \quad (4.25)$$

The parameter G_{rr} depends on the bearing type, mean diameter and load, meanwhile ϕ_{ish} and ϕ_{rs} are the inlet shear heating reduction and kinematic starvation reduction factor, respectively, given by equations (4.26) and (4.27).

$$\phi_{ish} = [1 + 1.84 \cdot 10^{-9} (n \cdot d_m)^{1.28} \cdot \nu^{0.64}]^{-1} \quad (4.26)$$

The inlet shear heating reduction factor (ϕ_{ish}) brings into account the fact that not all the lubricant present in the contact area is used to generate film. Therefore, the excess is rejected and produces a reverse flow, generating heat and consequently lowering the oil viscosity and reducing the film thickness and consequently the rolling friction [21].

$$\phi_{rs} = \left[e^{K_{rs} \cdot \nu \cdot n \cdot (d+D) \cdot \sqrt{\frac{K_z}{2(D-d)}}} \right]^{-1} \quad (4.27)$$

The kinematic replenishment/starvation factor takes into consideration that, for high speeds or viscosity, the replenishment of the bearing raceway may not be perfect, causing kinematic starvation and consecutive reduction of the film thickness and rolling friction.

Sliding frictional torque

The sliding frictional torque is calculated from equation (4.28).

$$M_{sl} = G_{sl} \cdot \mu_{sl} \quad (4.28)$$

Where:

- G_{sl} is a variable that depends on the bearing type, mean diameter and load;
- μ_{sl} is the sliding friction coefficient.

The sliding coefficient, (μ_{sl}), depends on the lubrication regime and therefore, is a weighted average of the full-film, (μ_{EHD}), and the limit-film, (μ_{bl}), given by equation (4.29). The weighting factor, (ϕ_{bl}), is function of the working conditions (speed and viscosity) as well as the bearing geometry, as shows equation (4.30).

$$\mu_{sl} = \phi_{bl} \cdot \mu_{bl} + (1 - \phi_{bl}) \cdot \mu_{EHD} \quad (4.29)$$

$$\phi_{bl} = \left[e^{2.6 \cdot 10^{-8} (n \cdot \nu)^{1.4} \cdot d_m} \right]^{-1} \quad (4.30)$$

The bearing manufacturer provides the friction coefficients for limit film and elastohydrodynamic lubrication conditions, given in Table 4.2. This values are calculated with a very high safety coefficient, and so, do not predict the real frictional torque of the bearings, but instead give a highly overestimated prediction of the sliding friction torque.

Table 4.2: Manufacturer friction coefficients.

R. bearing type	μ_{bl}	μ_{EHD}
Ball	0.15	0.05*
Cylindrical Roller	0.15	0.02
Tapper Roller	0.15	0.002

*Friction coefficient is not specific for the bearing type,
but instead for lubrication with mineral oils

Drag losses

The rolling bearings are lubricated by oil bath, meaning that the rolling bearings are partially or totally immersed in lubricant. The rolling bearing movement in the oil bath generates a frictional moment that must not be neglected. This loss source is similar to the wheel churning loss and is affected by phenomena such as bearings geometry, lubricant viscosity, oil agitation and the size and shape of the oil reservoir. The model proposed by the manufacturer takes into account some restrictions:

- The oil reservoir is large. Thus the effect of the size and geometry are negligible. The external oil agitation is also negligible;
- The shaft is horizontal;
- The rolling bearing is rotating at a constant speed;
- Lubricant viscosity must be lower than 500 cSt when the rolling bearing is submerged up to half the outside diameter and under 250 cSt when submerged more than half the diameter.

The frictional moment for drag losses is estimated by equations (4.31) and (4.32). The first applies to ball bearings and the second to roller bearings.

$$M_{drag} = 0.4 \cdot V_M \cdot K_{ball} \cdot d_m^5 \cdot n^2 + 1.093 \cdot n^2 \cdot d_m^3 \cdot \left(\frac{n \cdot d_m^2 \cdot f_t}{\nu} \right)^{-1.379} \cdot R_S \quad (4.31)$$

$$M_{drag} = 4 \cdot V_M \cdot K_{ball} \cdot d_m^4 \cdot n^2 + 1.093 \cdot n^2 \cdot d_m^3 \cdot \left(\frac{n \cdot d_m^2 \cdot f_t}{\nu} \right)^{-1.379} \cdot R_S \quad (4.32)$$

The parameters K_{ball} and K_{roll} are related to the bearing geometry and constriction and are given by equations (4.33) and (4.34), respectively:

$$K_{ball} = \frac{i_{rw} \cdot K_z \cdot (d + D)}{D - d} \cdot 10^{-12} \quad (4.33)$$

$$K_{roll} = \frac{K_L \cdot K_Z \cdot (d + D)}{D - d} \cdot 10^{-12} \quad (4.34)$$

The parameters V_m , K_L and K_Z are rolling bearing related constants and D_m is the bearing mean diameter, $D_m = 0.5 \cdot (D + d)$.

The remaining variables are function of the immersed surface of the bearing and the constants referred in the previous paragraph and are given by equations (4.35) to (4.40):

$$C_w = 2.789 \cdot 10^{-10} \cdot l_D^3 - 2.786 \cdot 10^{-4} + 0.0195 \cdot l_D + 0.6439 \quad (4.35)$$

$$l_D = 5 \cdot \frac{K_L \cdot B}{d_m} \quad (4.36)$$

$$f_t = \begin{cases} 0.5 \cdot , & \text{when } 0 \leq t \leq \pi \\ 1, & \text{when } \pi < t < 2\pi \end{cases} \quad (4.37)$$

$$R_s = 0.36 \cdot t d_m^2 \cdot (t - \sin(t)) \cdot f_a \quad (4.38)$$

$$t = 2 \cdot \cos^{-1} \left(\frac{0.6 \cdot d_m - H}{0.6 \cdot d_m} \right) \quad \text{when } H \geq d_m, H = d_m \quad (4.39)$$

$$f_A = 0.05 \cdot \frac{K_z \cdot (D + d)}{D - d} \quad (4.40)$$

Frictional torque in seals

The rolling bearing seals play an important role in increasing the rolling bearings life by avoiding contamination and keeping the lubricant close to the bearing tracks. The presence of this components introduces another type of frictional torque that can be estimated by equation 4.41.

$$M_{seal} = K_{S1} \cdot d_S^\beta + K_{S2} \quad (4.41)$$

Where K_{S1} , K_{S2} and β are bearing type and seal related constants and d_S is the seal counterface diameter.

“Old” SKF model

This model was replaced by the one described above, therefore does not offer the best possible power loss prediction, but has the advantage of being applicable to all the bearing types. This model divides friction torque of the bearing in two [12]:

- No-load dependent losses;
- Load dependent losses.

No-load dependent losses

This friction torque source depends on the working conditions, $\nu_{oil} \cdot n$, bearing size, d_m , and the manufacturer given constant for no-load losses, f_0 , as shows equation (4.42).

$$M_{VL0} = \begin{cases} 1.6 \cdot 10^{-8} \cdot f_0 \cdot d_m^3 & \text{for } \nu_{oil} \cdot n < 2000 \text{mm}^2/\text{s} \cdot \text{min} \\ 10^{-10} \cdot f_0 \cdot (\nu_{oil} \cdot n)^{2/3} \cdot d_m^3 & \text{for } \nu_{oil} \cdot n > 2000 \text{mm}^2/\text{s} \cdot \text{min} \end{cases} \quad (4.42)$$

Load dependent losses

For load dependent losses, the model separates the losses in axial and radial load related. The parameters P_1 and F_a are the radial and axial loads, and f_1 and f_2 are bearing related constants. The load dependent frictional torque is given by equation (4.43)

$$M_{VL1} = \begin{cases} 10^{-3} \cdot f_1 \cdot P_1 \cdot d_m & \text{for radial loaded bearings} \\ 10^{-3} \cdot f_2 \cdot F_a \cdot d_m & \text{for axial loaded bearings} \end{cases} \quad (4.43)$$

4.1.4 Power loss in seals

The seals power loss is, in most of the cases, the one that assumes the lowest values, typically around 0.01% of the total transmitted power, and so are almost negligible when compared to the losses of other components [12].

The equations to evaluate the power loss are typically given by the manufacturer (SIMRIT)[22], and in this case, given by equation (4.44).

$$P_{VS} = 7.69 \cdot 10^{-6} \cdot d_{seal}^2 \cdot n \quad (4.44)$$

4.2 Transfer gearbox modelling

The software *Matlab R2016a* was used as a programming base for the implementation of a power loss model. This model has the objective of estimate and break the power loss in each one of its sources.

4.2.1 Gearbox kinematics

The gearbox internal kinematics are given by the gears reduction ratios. With the gear pairs 1-2 and 3-5 geared, the rotational speeds of the shafts in rad/s are given by equations (4.45) to (4.47)

$$\omega_1 = n_{in} \cdot \frac{2 \cdot \pi}{60} \quad (4.45)$$

$$\omega_2 = \omega_1 \cdot \frac{z_1}{z_2} \quad (4.46)$$

$$\omega_3 = \omega_2 \cdot \frac{z_3}{z_5} \quad (4.47)$$

The same method can be applied to calculate the torque, since it varies in the inverse ratio of the rotational speed.

4.2.2 Gear geometry and contact forces

The geometrical parameters of the gears were calculated according to the normal system for profile shifted helical gears [23].

The meshing forces were calculated according to the equations (4.49) to (4.50), and derive from the gears geometrical dimensions [24].

$$F_t = \frac{T}{R_w} \quad (4.48)$$

$$F_r = \frac{F_t}{\cos(\beta)} \cdot \tan(\alpha) \quad (4.49)$$

$$F_a = F_t \cdot \tan(\beta) \quad (4.50)$$

Where:

- F_t is the force tangent to the working circle [N];
- F_r is the radial force [N];
- F_a is the axial force[N];
- T is the transmitted torque [Nm];
- R_w is the wheel working diameter [m].

4.2.3 Rolling bearings power loss

In order to estimate the bearings power loss, the new *SKF* model [21] offers very accurate results, although overestimates the friction coefficients, both the μ_{bl} and the μ_{EHD} , given in Table 4.3 [25]. The rolling bearings manufacturer does not provide the required constants for the model implementation in the case of the needle roller bearings, consequently this model is restricted to ball, cylindrical and tapered roller bearings.

Note that the rolling bearings installed in the gearbox do not have seals, therefore the model component correspondent the rolling bearing seals frictional torque is null .

Table 4.3: Rolling bearings experimental friction coefficients [25].

Lubricant	R. bearing type	μ_{bl}	μ_{EHD}
75W140-A	Ball	0.078	0.060
	Cylindrical roller	0.073	0.040
	Tapper roller	0.100	0.013
75W90-A	Ball	0.078	0.060
	Cylindrical roller	0.073	0.040
	Tapper roller	0.100	0.011

All the constants necessary to the model implementation derive from consulting the rolling bearing catalog [21], and are presented in Table 4.4.

Table 4.4: Rolling bearing constants.

R. Bearing reference	R_1 $\times 10^7$	R_2	S_1 $\times 10^3$	S_2	K_Z	K_{rs} $\times 10^8$	K_L	H	V_m $\times 10^4$	Y
6307	3.7	1.7	2.84	92.8	3.1	5	-	20	6	-
6306	3.7	1.7	2.84	92.8	3.1	5	-	20	5	-
32306	23.8	10.9	19	2	6	5	0.7	16	6	1.9
NJ309E	10.9	0	160	0.0015	5.1	5	0.65	30	6	-

Only two of the needle roller bearings installed have relative movement between the tracks. Since the more recent SKF model cannot be applied to this rolling bearing type, the chosen model to estimate the power loss of these bearing was the “old” SKF model. For the proposed working conditions, the two rolling bearings installed are working in no-load conditions. For no-load, the model requires the parameter f_0 , which takes the value of 12 [12].

4.2.4 Gear mesh power loss

The model presented and described to estimate the meshing power loss requires the calculation of the gear loss factor H_v . This factor was numerically derived by *Marques et al.* [16], and the values are presented in Table 4.5.

Table 4.5: Gear loss factor H_v

Gear Pair	H_v
1-2	0.17338
2-4	N.A.
3-5	0.24839

In order to obtain the average coefficient of friction between the meshing gears, it is required to know beforehand the lubricant correction factor X_L . The X_L values for the lubricants tested are present in Table 4.6 [26].

Table 4.6: Lubricant correction factor X_L .

Lubricant	X_L
75W140-A	0.61
70w90-A	0.65

4.2.5 Churning power loss

The implemented model to estimate the churning power loss requires that the immersed height of each wheel to be known beforehand. It is known that the oil level is 20mm below the shafts axis, so the immersed height (h) is give by equation 4.51.

$$h = R_p - 20 \quad (4.51)$$

R_p being the primitive radius of the wheel in millimeters. After this step, the immersed height for each wheel is presented in Table 4.7:

Table 4.7: Immersed depth of the wheels.

Wheel	Depth [mm]
1	40
2	23
3	40
4	30
5	16

4.2.6 Model verification

The geometrical parameters were all verified using as reference the software *KISSSys* and *KISSSoft* [27]. This software offers accurate reports of the geometric parameters of the meshing gear pair and also determines the reaction forces of the supports.

The verification was made for 3 test points. For the geometric parameters, the *Matlab* model and the *KISSSys* values were coincident. For the reaction forces in the supports, the calculated values differed less than 2%, which is an acceptable deviation. The *KISSSys* reports containing the gear geometrical parameters and one simulation for the medium point of the input conditions is present in Appendix C.

The bearing frictional losses were verified using the manufacturer simulation tool. In order to perform this verification, the friction coefficients used were the ones given by the manufacturer, therefore it was not possible to run simulations for the final implemented model.

The meshing friction coefficient was verified using previous tests performed by *Fernandes* in a FZG test rig for identical conditions of load and speed [28].

4.3 Model estimation

The model was applied to the exact conditions of torque, speed and temperature registered in the experimental test. The results are presented and commented in this section.

4.3.1 75W140-A oil

Figures 4.5 and 4.6 show the absolute and relative power loss, respectively, calculated by the developed model for the lubricant 75W140-A, breaking it into the four main sources.

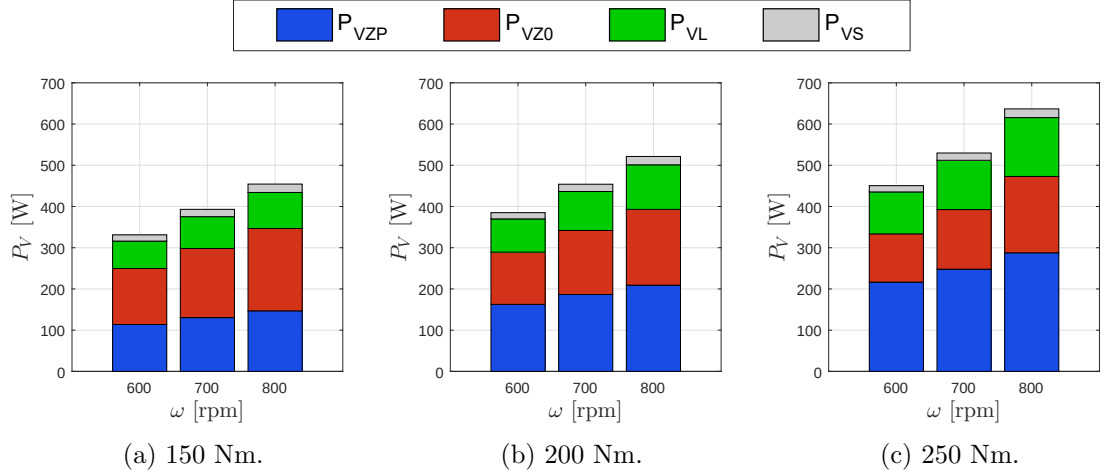


Figure 4.5: Absolute values for the power loss sources for 75W140-A.

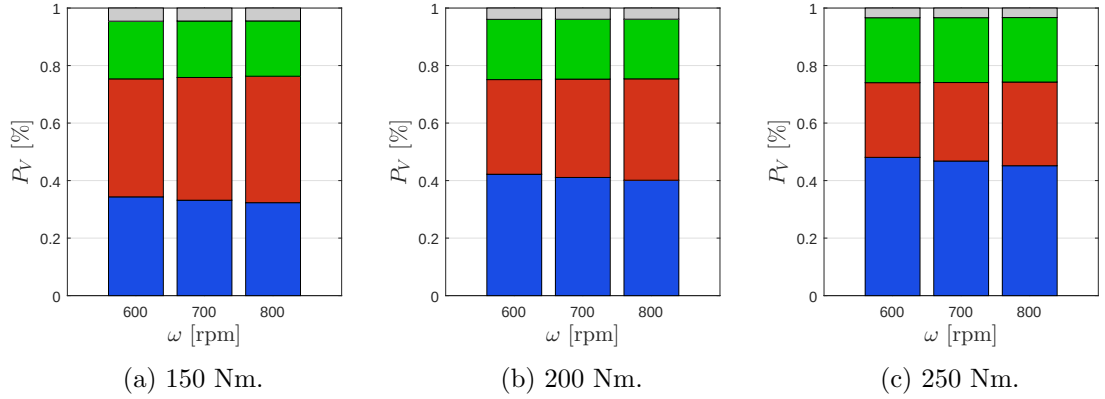


Figure 4.6: Power loss sources influence relative to the experimental power loss for 75W140-A.

For this lubricant, the dominant loss source is not the same through the whole test range. Analyzing Figure 4.6, it is visible that for the condition of 150 Nm, the churning loss is the dominant loss source and for the remaining conditions the power loss is dominated by meshing losses.

The churning loss grows with the increasing of the rotational speed, this strong dependence with the working speed can be explained by the nature of the loss. Since the churning losses are load independent, they vary with the kinematic parameters of the gearbox and present an almost constant behavior when changing the load conditions, but instead grow when increasing the speed even though the oil viscosity is lower due to thermal effects. When increasing the torque, the churning losses see their influence slightly reduced due to the growth of oil temperature and consequent decay of the viscosity, easing the movement of the immersed wheels. In relative terms, this loss represents 25 to 45 % of the total loss, and its influence diminishes from the lowest to the highest torque and from the highest to the lowest speed.

The meshing losses, in absolute values, grows with both torque and speed, but the torque has a more noticeable influence than speed. Figure 4.6 shows that the meshing loss relative influence grows when increasing the torque but reduces when increasing the speed, meaning that for a constant torque the absolute growth of the meshing loss is smaller than

the growth of the remaining losses. The meshing loss tends to dominate the total loss for higher torques. The loss caused by the meshing gears represents 30 to 50% of the total power loss. For higher torques, and working temperatures, the meshing friction coefficient increases and therefore explains the enlargement of this loss type influence.

The churning and meshing losses together represent 85% of the total power loss for the entire test range. It was verified that the meshing losses gain influence over churning when the torque is increased, and when the speed is increased, the churning loss increases more than meshing.

Rolling bearing frictional loss absolute values grow with the input power. In relative terms, stays constant when varying the speed at constant torque, and increases with the input torque, representing 20 to 23% of the total power loss.

The seals frictional torque is only function of the shaft speed, and therefore the absolute value of this loss is constant independently of the torque condition. Consequently, the relative influence is reduced for the higher torque conditions, and since the total power loss increases with speed, the relative influence is again reduced, and represents 3 to 4%.

4.3.2 75W90-A oil

Figures 4.7 and 4.8 show the absolute and relative power loss, respectively, calculated by the developed model for the lubricant 75W90-A, breaking it into the four main sources.

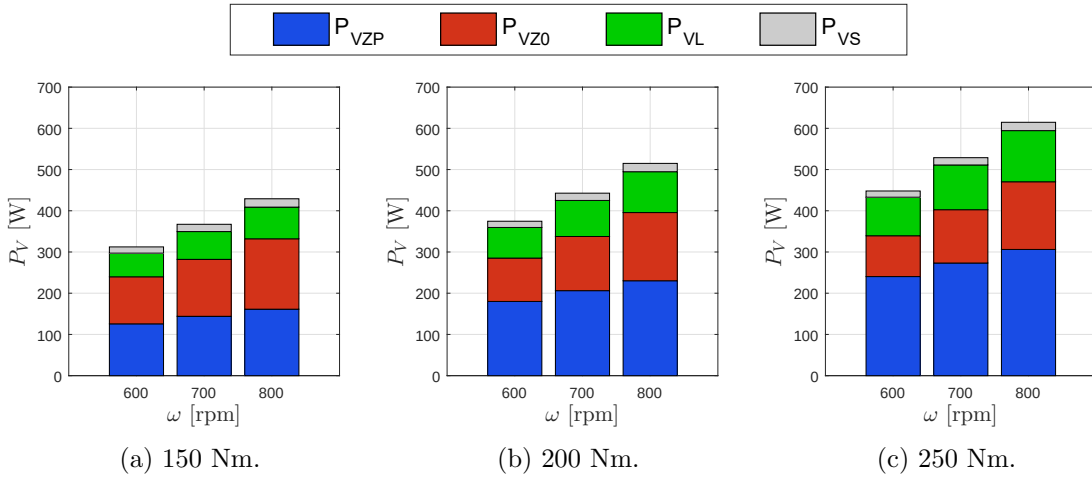


Figure 4.7: Absolute values for the power loss sources for 75W90-A.

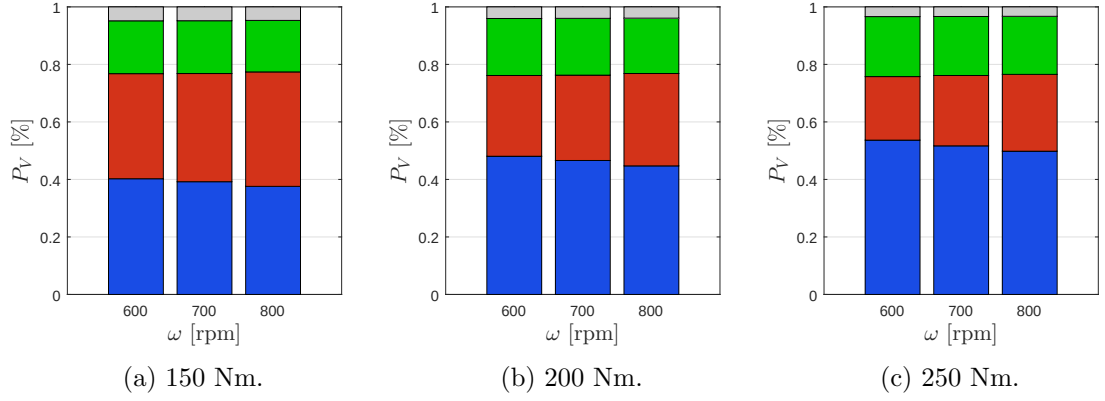


Figure 4.8: Power loss sources influence relative to the experimental power loss for 75W90-A.

For this lubricant, the dominant power loss source is the gear meshing friction for two higher torque conditions. For the 150 Nm condition, the meshing and churning losses present sensitively the same relative weight.

The meshing losses represent 38 to 55 % of the total power loss. Figure 4.7, shows that the meshing losses increase with the input torque and speed, and that the growth when increasing the torque is bigger than when increasing the speed. Figure 4.8 shows that the meshing loss becomes dominant when increasing the torque, but when increasing the speed, its growth is lesser than the overall increase of the power loss and therefore its influence is reduced.

The churning losses, in absolute values, grow with the rotational speed and diminish when increasing the torque. The relative weight of the churning losses also increases with speed, being the loss type most affected by this parameter. At constant speed, when increasing the torque, the churning loss influence is significantly reduced. Although the absolute value is also diminishes due to the reduction in the lubricant viscosity, the reduction in the influence is mainly due to the rise of the meshing loss.

The rolling bearing losses increase with both speed and torque presenting a more noticeable growth when increasing the torque. Relatively to the total power loss, this source represents 17 to 20%. For tests at constant torque, it was verified that the relative influence of the bearing losses are constant. For higher torques the rolling bearing losses grow and increase the relative weight.

The seals loss present the smaller loss of the four. For tests with identical rotational speed the absolute value is constant, and therefore the relative influence is reduced when the total power loss grows for higher torque conditions. This loss source represents 3-5% of the total power loss.

4.3.3 Lubricant comparison

The total power loss estimated by the model does not differ more than 26 W for the two lubricants. The prediction for the lubricant 75W140-A is superior in almost every condition, with the exception of the highest input power condition. Although the total power loss is not very different, the power loss partitions vary considerably.

Note that the values represented in Figures 4.9, 4.12 and 4.13 were calculated according to equation (4.52).

$$\Delta X = \frac{X_{75W140-A} - X_{75W90-A}}{X_{75W90-A}} \quad (4.52)$$

Where X is the variable represented in each Figure.

Meshing power loss

The Figures 4.9, 4.12 and 4.13, show the difference of the model results for each oil relative to the lubricant 75W90-A.

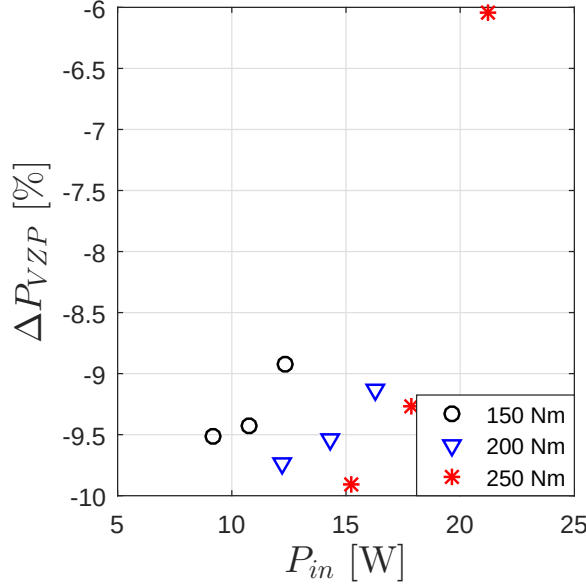


Figure 4.9: Relative difference between the meshing power loss (P_{VZP}) of 75W140-A and 75W90-A.

In Figure 4.9 is visible that the relative difference between the meshing power loss of the lubricants takes negative values, meaning that the loss for the 74w140 is lower than for the 75W90-A. Figure 4.9 shows that when increasing the speed at constant torque the meshing loss for the 75W140-A tends to approach the one of the 75W90-A, and the difference is reduced. Comparing the constant torque lines, it is visible that the torque, when increased, widens the difference in the meshing power loss caused by the variation in speed.

Figure 4.9 also shows that, depending on the speed, the torque has different effects. For the lowest speed, the difference increases when increasing the torque, meaning that the loss increases more for the 75W90-A than for the 75W140-A. For the highest speed, the power loss increases more for the 75W140-A than for the 75W90-A, and the difference is reduced. In Figure 4.9, the point at 250 Nm and 800 rpm shows a much lower difference than the other points. This difference is justifiable by average input speed used to run the model. Due to experimental difficulties in the speed control, the speed of the test for the 75W140-A than for the 75W90-A. Although this value falls a far from the others, it still follows the tendency set by the other conditions.

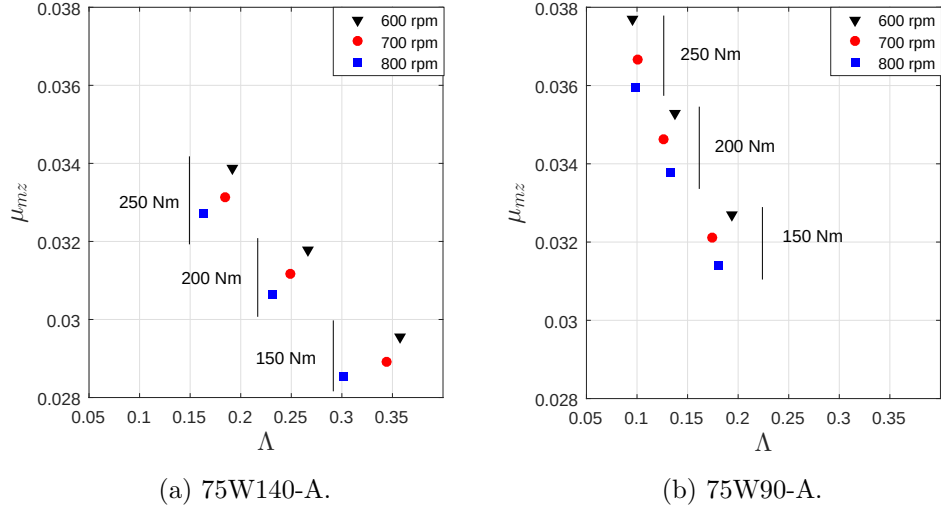


Figure 4.10: Friction coefficient function of specific film thickness for the meshing pair z_1/z_2 .

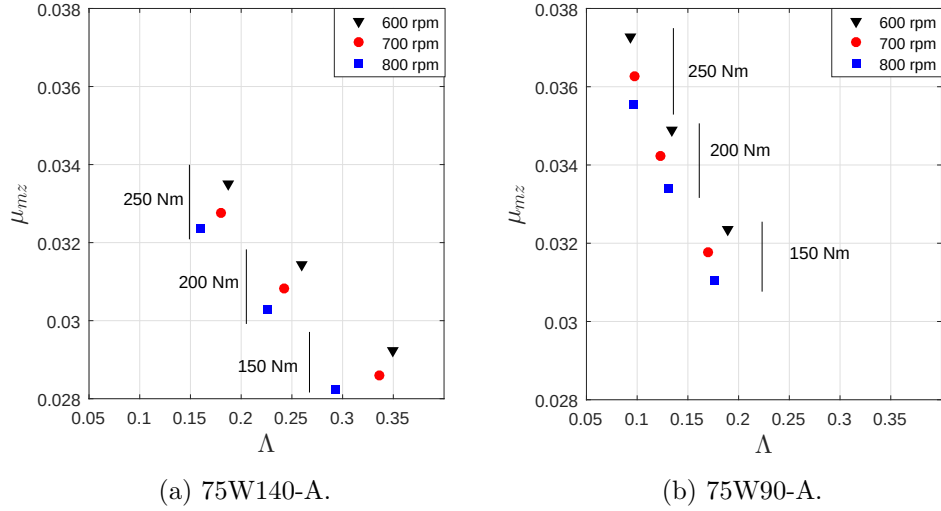


Figure 4.11: Friction coefficient function of specific film thickness for the meshing pair z_3/z_5 .

The observations made in Figure 4.9 are enforced by Figures 4.10 and 4.11 that represent the average friction coefficient, μ_{mz} , as a function of the specific film thickness for the meshing pairs z_1/z_2 and z_3/z_5 , respectively. Comparing Figure 4.10 and 4.11, it is visible that the lubrication regime is similar in the two meshing pairs for the same lubricant.

The meshing power loss is directly influenced by the friction coefficient as is stated in equation (4.1). Both lubricants work in limit film regime (<0.7). These operating conditions are quite severe, and so, the friction coefficient presents high values and tends to increase significantly for small reductions of the specific film thickness.

For the 75W140-A the film thickness is higher than for the 75W90-A. The higher viscosity of the 75W140-A combined with lower working temperatures, create better conditions for the film generation, and consequently lower friction and power loss, as was verified in Figure 4.9.

It is also worth to note that the values of the friction coefficient for a specific film thickness, Λ , lower than 0.12 might be overestimated as a consequence of the applicability of the chosen model. Near this limit the friction coefficient tends to stabilize, and this model cannot mimic this behavior.

Figures 4.10 and 4.11 show that the friction coefficient is higher for the tests at lower speed, and as the speed increases, the friction coefficient and the specific film thickness are reduced. This behavior is opposite to the general tendency of the friction coefficient to vary inversely with the film thickness. This phenomena is explained by the higher temperature for the faster conditions, that reduces the dynamic viscosity of the lubricant. For the 75W140-A, the effect of the reduction of the viscosity surpasses the effect of the higher speed, lowering the specific film thickness. For the 75W90-A, the effect of the temperature and the speed almost cancel each other, and the specific film thickness stays almost constant as the speed increases. The speed and viscosity have a different weight in the friction coefficient than in the specific film thickness as is visible in equation (4.5) and (4.4), respectively.

Churning power loss

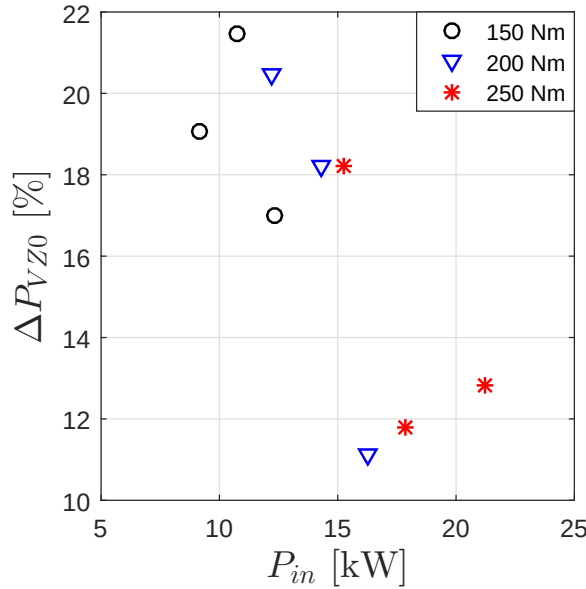


Figure 4.12: Relative difference between the churning power loss (P_{VZ0}) of 75W140-A and 75W90-A.

Figure 4.12 shows the relative difference of the power loss by churning effects for the 75W140-A and the 75W90-A lubricants. The represented values are all positive, meaning that this loss type is higher for the 75W140-A than for the 75W90-A. The tendency verified in Figure 4.12 is for the churning power loss difference to be reduced when increasing the power. The higher difference registered for low power situations can be explained by the different viscosities of the lubricant at the stabilization temperature of the performed tests. At higher temperatures, the lubricants viscosities tend to be closer values, as shows Figure 3.5. Since the test conditions are the same for the two lubricants and the viscosities tend to become similar due to thermal effects, the churning loss also tends to the same value, reducing the relative difference.

Rolling bearing power loss

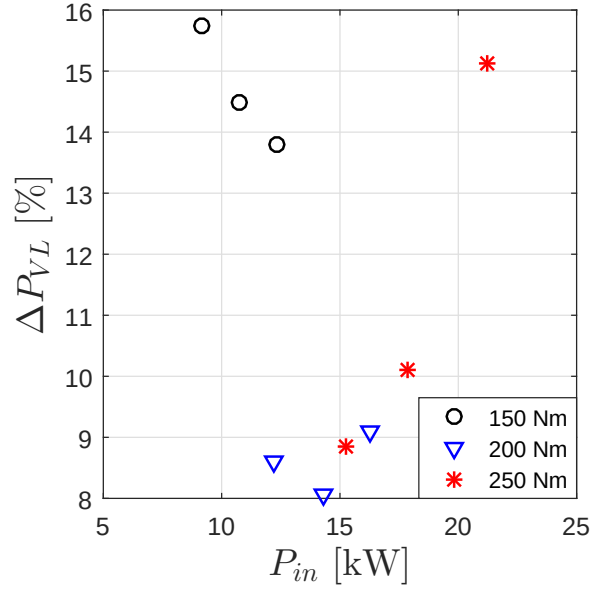


Figure 4.13: Relative difference between the rolling bearing power loss (P_{VL}) of 75W140-A and 75W90-A.

Figure 4.13 represents the relative difference of the rolling bearing power loss, P_{VL} , loss between the lubricants. This power loss source is higher for the 75W140-A than for the 75W90-A. The (P_{VL}) includes load and no-load related phenomena and therefore the pattern visible in Figure 4.13 is not function of the torque or speed, but instead of the total transmitted power. For the lower power conditions, the relative difference takes values close to 15% that decrease to 8% for the 1.5 kW situations and increases again until 12% for the maximum power condition. This behavior means that the P_{VL} grows more rapidly for the 75W90-A for the low power conditions, until sensibly 1.5 kW, and after that the tendency inverts and the 75W90-A presents a slower growth compared to the growth of the 75W140-A, increasing the difference.

4.4 Power loss model vs experimental results

This section is dedicated to the comparison of the experimentally measured power loss with the model estimation, followed by the an analysis of the deviations.

4.4.1 Overall comparison

75W140-A oil

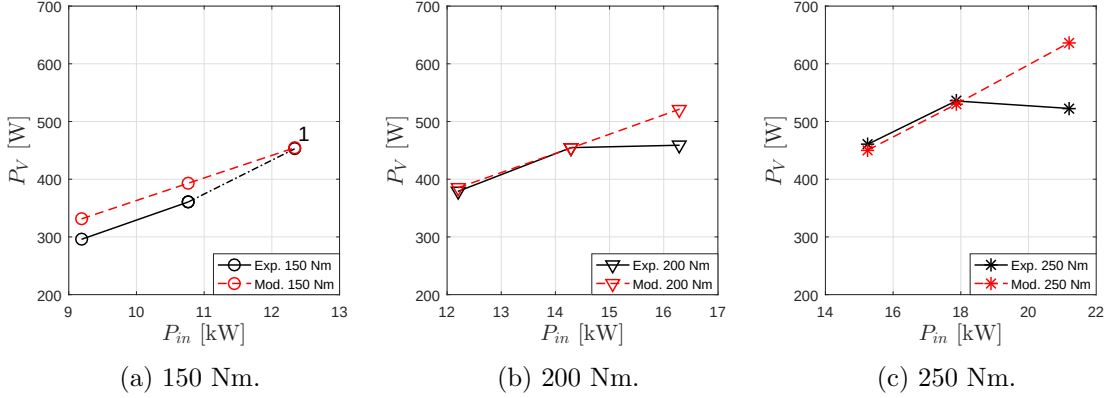


Figure 4.14: Constant torque lines of power loss model (Mod.) and experimental results (Exp.) for 75W140-A. (1-Unstabilized test).

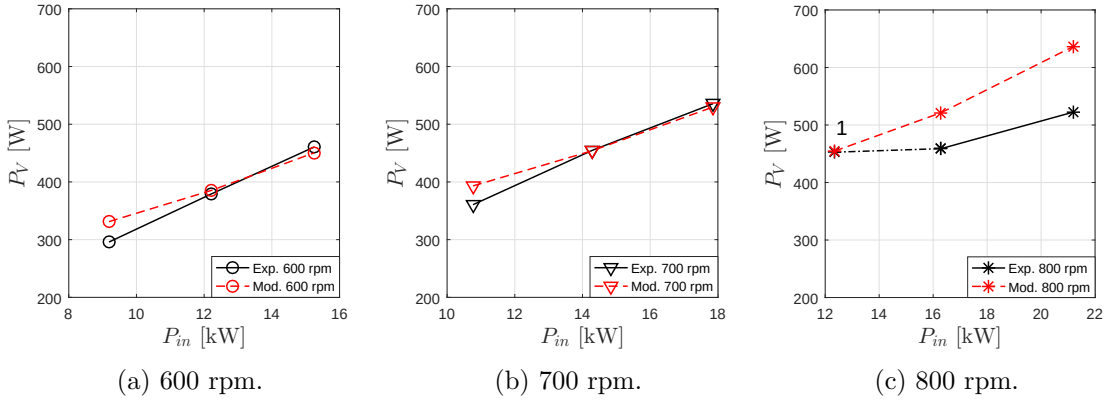


Figure 4.15: Constant speed lines of power loss model and experimental results for 75W140-A. (1-Unstabilized test).

Figure 4.14 compares the total power loss given by the model with the experimental tests at constant torques. At higher torques the model predicts accurately the total power loss of the gearbox, but for the higher speeds there is a deviation, and the model estimates the power loss by excess.

In general, for this lubricant, the model presents a very linear behavior when varying the torque or the speed. This behavior does not follow the experimentally verified, since the gearbox tends to lower the power loss at the two highest speeds, improving the efficiency, and this effect is not represent by the model.

The observations taken from Figure 4.14 are enforced by Figure 4.15, where it can be seen that for the two lower speeds the prediction and experimental lines are almost

coincident with a slight discrepancy at the lower torque. At the highest speed, 800 rpm, the model estimates the power loss for the 150 Nm test accurately, but the deviation is large the on other two tests. Although the model estimates with accuracy the power loss for the test at 150 Nm and 800 rpm, it should be noted that the experimental values for this test were considered invalid when analyzing Figures 3.7 and 3.8.

75W90-A oil

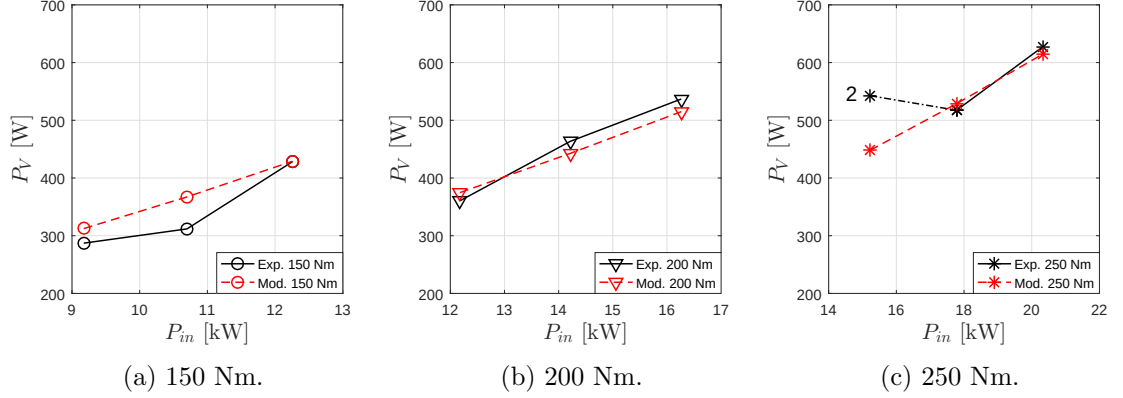


Figure 4.16: Constant torque lines of power loss model and experimental results for 75W90-A. (2-Unstabilized test).

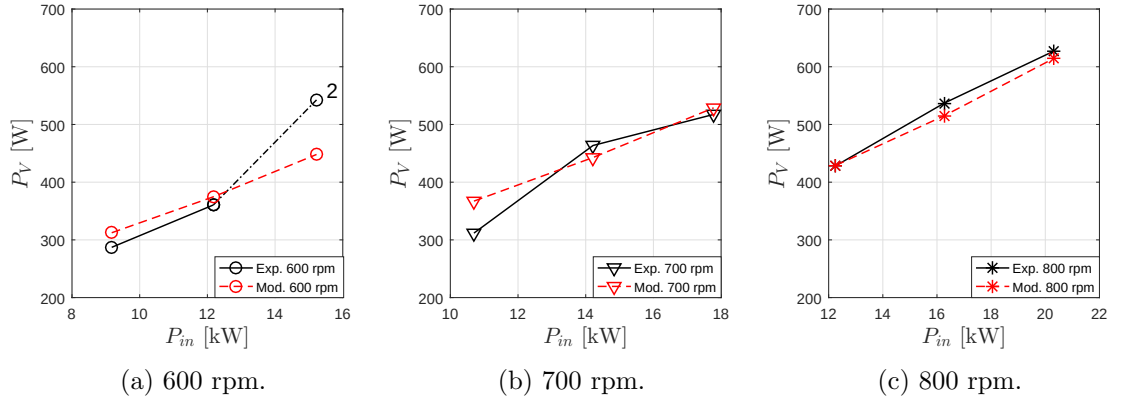


Figure 4.17: Constant speed lines of power loss model and experimental results for 75W90-A. (2-Unstabilized test).

Figure 4.16 compares the total power loss given by the model with the experimental tests at constant torques for the lubricant 75W90-A. In general the model estimative is close to the measured values, with the exception of the test at 250 Nm and 600 rpm where the model underestimates considerably the measured power loss. The deviation for this specific condition is most likely due to a experimental anomaly, since already in Figure 3.10b is clear that this point deviates from the pattern defined by all the other tests, and after further analysis was considered invalid. The test at 150 Nm and 700 rpm presents a deviation that in not very large in absolute values, is very relevant when compared with the total power loss.

4.4.2 Deviation analysis

Figures 4.18 to 4.21, present the absolute and relative differences from the power loss model results to the experimentally measured values.

75W140-A

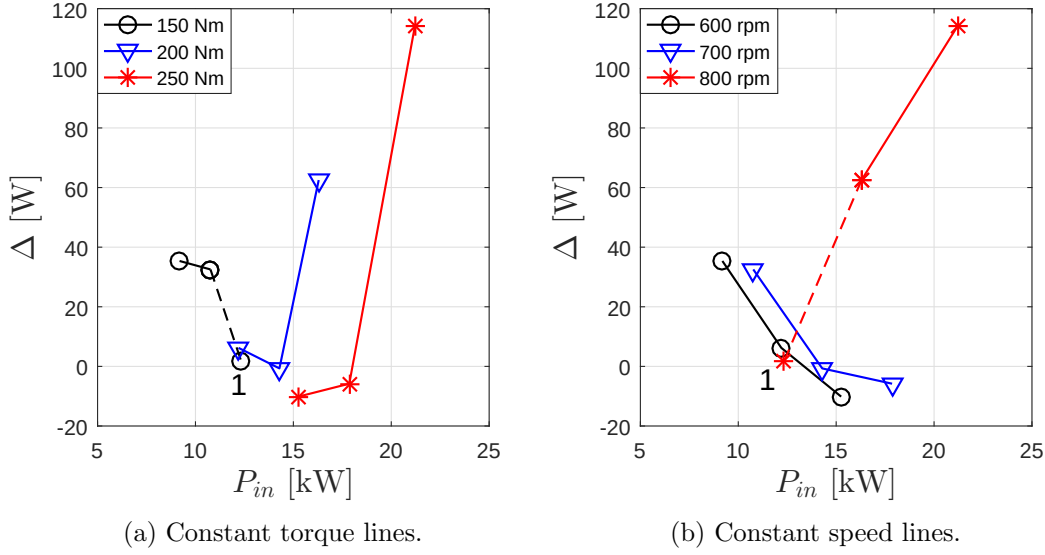


Figure 4.18: Absolute deviations for the lubricant 75W140-A. 1-value affected by unstabilized test (150 Nm 800 rpm).

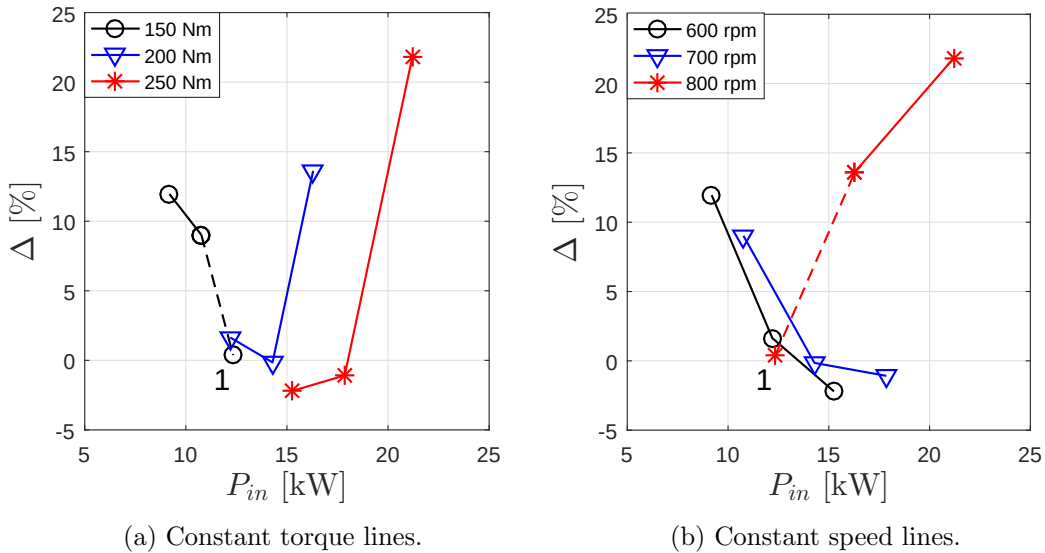


Figure 4.19: Relative deviations for lubricant 75W140-A. 1-value affected by unstabilized test (150 Nm 800 rpm).

Figures 4.18 and 4.19 represent the absolute and relative deviation from the model estimation to the experimentally measured values, respectively. By analyzing Figures 4.18

and 4.19, it can be seen that the power loss model overestimates the power loss in most of the cases. The excess or default are maintained in relatively low values, $<15\%$, with the exception of the test at 250 Nm and 800 rpm.

Figures 4.18b and 4.19b show that, for the two lower speeds, the deviation tends to diminish as the torque increases, and for the highest the deviation increases when increasing the torque. For the tests at 800 rpm, the deviation is due to the non linearity verified in Figure 3.7b, that the model cannot mimic. The deviation for the low torque and speed, although represent around 10% of the total power loss, correspond to only 40 W. A value of this magnitude may caused by measuring errors.

75W90-A

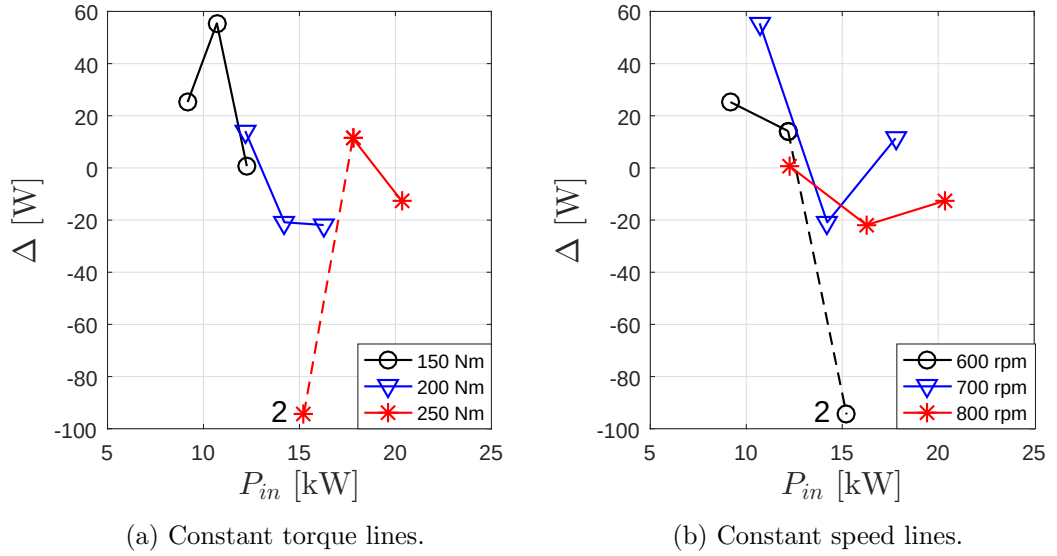


Figure 4.20: Absolute deviations for lubricant 75W90-A. 2-value affected by unstabilized test.

Figures 4.20 and 4.21 represent the absolute and relative deviation from the model estimation to the experimentally measured values, respectively. By analyzing Figures 4.20 and 4.21, it can be seen that the power loss model deviates less than 60 W for all the tests with the exception of the test at 250 Nm and 600 rpm (2), that has been considered invalid.

Globally, the difference tends to be reduced as the input power increases. The tests at the highest speed present the lowest deviation, and the same is verified for the higher torques. This fact may be due to experimental difficulties, since the test rig tends to present better accuracy of measurements for higher torques, since the sensors enter in a more linear zone and the speed is maintained in a smaller interval.

In general, the model estimates the power loss with an acceptable deviation for each lubricant. The power loss estimated by the model tends to approach the experimentally measured values as the power is increased, leading to believe that the non linearity of the test rig torque sensors may interfere in the accuracy of the results.

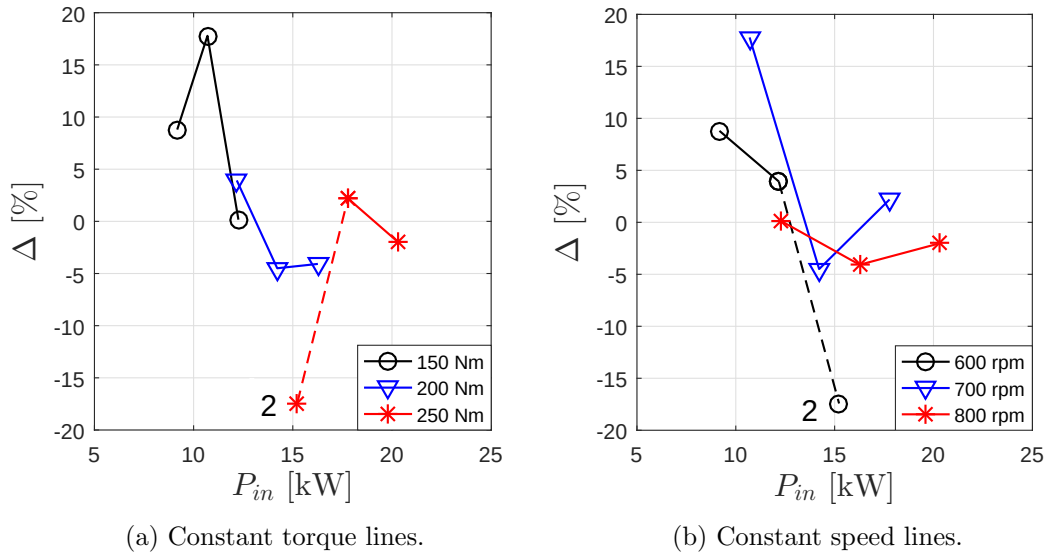


Figure 4.21: Relative deviations for lubricant 75W90-A. 2-value affected by unstabilized test.

Figure 4.22 and 4.23 show a more global view of the experimental measured power loss, the power loss estimations and partition and the absolute deviation for the lubricants 75W140-A and 75W90-A, respectively.

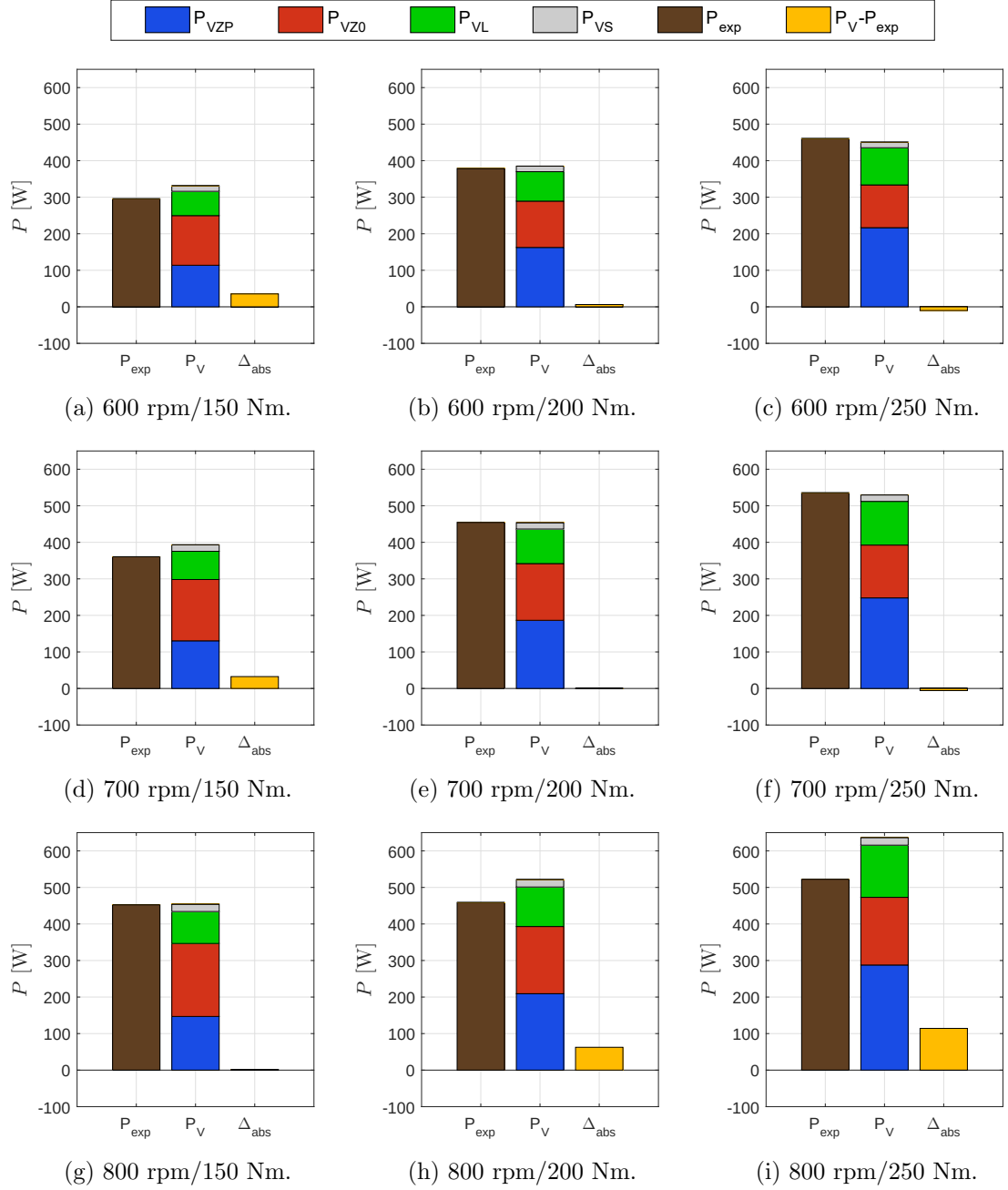


Figure 4.22: Experimental measured power loss, power loss model partitions and absolute difference for 75W140-A.

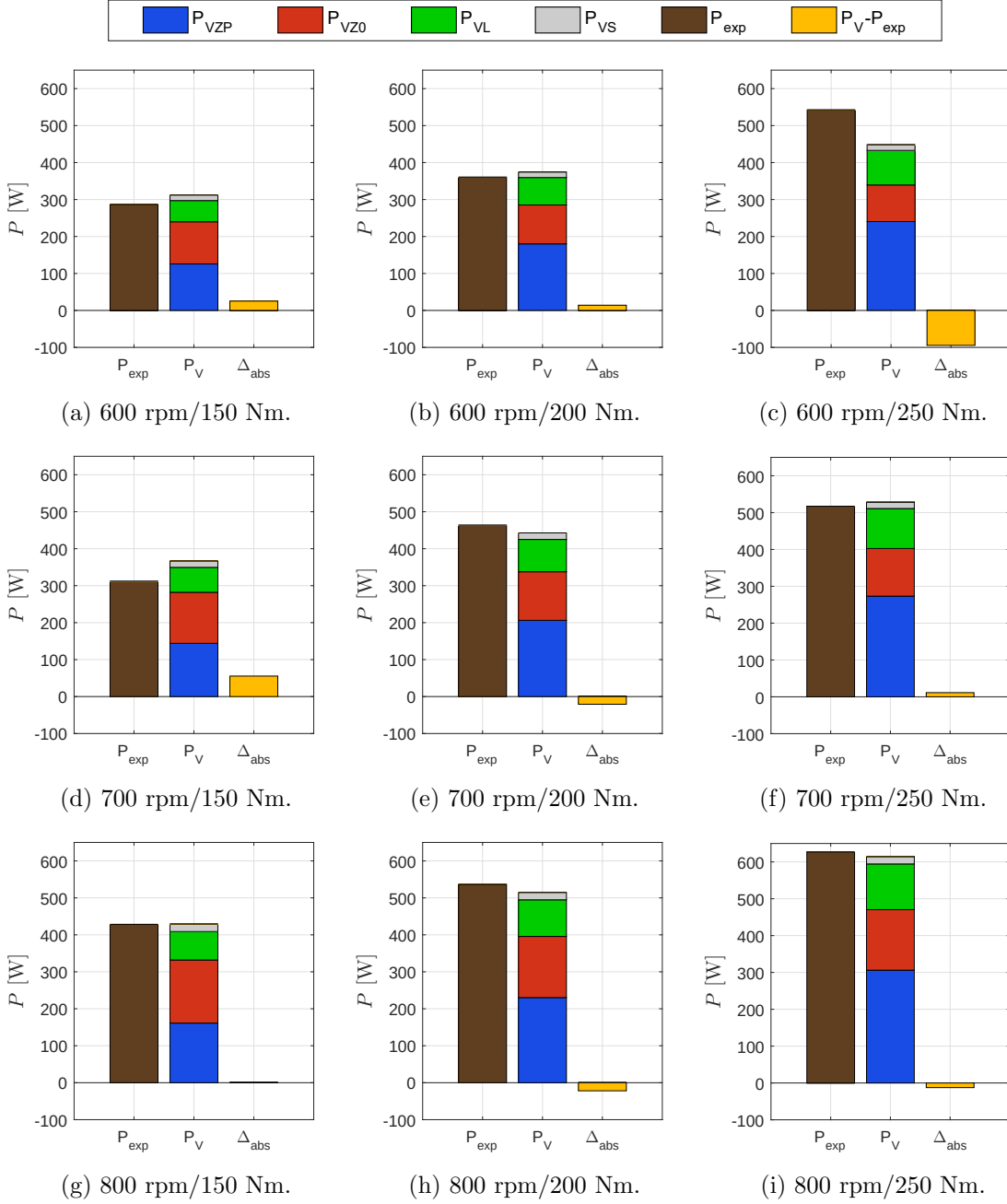


Figure 4.23: Experimental measured power loss, power loss model partitions and absolute difference for 75W90-A.

Thermal model

In order to predict the power loss at a given torque and speed, it is also required to know the stabilization temperature. The lubricant properties are strongly related with the working temperature, therefore, a good prediction of the temperature is crucial to evaluate the efficiency of a gearbox. A gearbox efficiency is never 100%, therefore there is always energy dissipation. The difference in power between the input and output shafts is converted in heat by the moving components in the gearbox. The lost power is reflected in the rise of the lubricant, shafts and housing temperatures. The temperature gradient between the gearbox components and the surrounding environment leads the gearbox to transfer the dissipated heat to the surrounding environment.

The heat flux is directly related to the temperature difference of the involved mediums, so it can be stated that when the temperature difference stabilizes, the system is dissipating the same amount of energy as it generates, and therefore has reached thermal equilibrium. The thermal equilibrium or steady-state condition is represented by equation (5.1).

$$Q_{Ev} = P_V \quad (5.1)$$

Where:

- Q_{Ev} is the total evacuated heat;
- P_V is the total power loss.

5.1 Heat transfer modes

There are three mechanisms of heat transfer:

- Convection - Q_{conv} ;
- Conduction - Q_{cond} ;
- Radiation - Q_{rad} .

The total evacuated heat is given by the sum of the three modes, as specified in equation (5.2).

$$Q_{Ev} = Q_{conv} + Q_{cond} + Q_{rad} \quad (5.2)$$

All three modes take part in evacuating heat from the gearbox. Every surface dissipates heat by radiation, but, for the temperature range in study, has a residual weight in the overall heat transfer. The housing and shafts conduct heat to the foundations and the

subsequent shafts, and also by convection to the air involving the gearbox, these being the most important heat transfer mechanisms.

5.1.1 Convection

Convection heat transfer refers to the phenomenon that occurs between a surface and a moving fluid when they are at different temperatures [29].

The heat transferred by convection is given by equation (5.3).

$$Q_{cnv} = h_{cnv} \cdot A \cdot (T_{wall} - T_{amb}) \quad (5.3)$$

Where:

- Q_{cnv} - Transferred heat [W];
- h_{cnv} - Convection coefficient [W/m²K];
- A - Convection area [m²];
- T_{wall} - Temperature of the gearbox wall [°C];
- T_{amb} - Ambient temperature [°C];

The convection coefficient, h , depends on the convection type, natural or forced, the fluid and surface types involved.

In the case of the gearbox in study, it is present in the internal and external housing wall, due to the interaction between the oil and air, respectively. It is also present in the shafts ends in the interaction with the surrounding air.

Due to the complex geometry and the irregular flows of the fluids involved in the process, the convection factor is very difficult to estimate with accuracy.

5.1.2 Conduction

Conduction is the transport of energy in a medium due to a temperature gradient and is sustained by molecular or atomic level phenomena. The energy transfer by conduction occurs, in the presence of a temperature gradient, in the direction of the decreasing temperature [29]. For unidirectional case, the heat flux can be calculated by equation 5.4.

$$Q_{cnd} = -k \cdot A \cdot \frac{dT}{dx} \quad (5.4)$$

Where:

- Q_{cnd} - Transferred heat [W];
- k - Thermal conductivity [W/mK];
- A - Section area [m²];
- $\frac{dT}{dx}$ - The temperature gradient perpendicular to the direction of the heat flux [K/m].

The thermal conductivity, k , is characteristic of the material, and for steady-state conditions, the temperature distribution is linear. For the case in study the conduction mechanism is present in the heat flux through the gearbox walls and shafts.

5.1.3 Radiation

Any piece of matter at a temperature different from zero emits radiation. This radiation is of electromagnetic nature. While conduction and convection require a material medium, radiation does not. The radiation emitted by a surface originates from the body thermal energy, and therefore its emission lowers the body energy [29]. The maximum emitted energy by radiation can be quantified by the *Stefan-Boltzman* law, equation (5.5).

$$E_{rad} = A \cdot \sigma_s \cdot T_s^4 \quad (5.5)$$

Where:

- E_{rad} - Radiated energy [W];
- A - Surface area [m²];
- σ_s - *Stefan-Boltzman* constant
- T_s - Absolute temperature of the radiating surface [K].

5.1.4 Reo-electric analogy

The thermal system can be compared to an electric system using a reo-electric analogy [29], given by equations (5.6) and (5.7).

$$V \equiv \Delta T \quad (5.6)$$

$$I \equiv Q_{Ev} \quad (5.7)$$

Knowing that for an electric circuit the resistance can be calculated by equation (5.8).

$$R = \frac{V}{I} \quad (5.8)$$

So, the equivalent thermal resistance can be calculated by equation (5.9)

$$R_{T_{eq}} = \frac{T_{oil} - T_{amb}}{P_V} \quad (5.9)$$

Where R_T is the equivalent thermal resistance of the system, in K/W, for the steady-state condition.

5.2 Equivalent thermal resistance

The equivalent thermal resistance takes into account the heat transferred by all the transfer modes. Once a few test points are known, for each test, the equivalent thermal resistance can be estimated using equation (5.9), as a function of the difference between the oil temperature and the air temperature.

Since both tested oils have the same base, they also have identical thermal properties and consequently, the tests from the two can be used to define a function for the equivalent thermal resistance. From the performed tests points, the equivalent thermal resistance is represented in Figure 5.1.

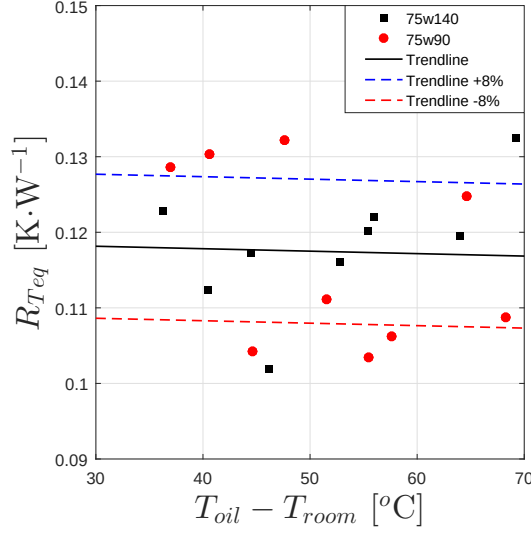


Figure 5.1: Equivalent thermal resistance and trendlines

Figure 5.1 also shows the trendline that approximates the values equivalent thermal resistance for the system working in steady-state condition. The heat transfer mechanisms are affected by atmospheric conditions such as temperature and air humidity. These atmospheric conditions cannot be controlled in the test room, and so, the thermal resistance spreads over an interval. An upper and lower limits were set in order to give a range of for the thermal resistance instead of a single value. The limits are set by two lines, parallel to the trendline, with an offset of $\pm 8\%$, also represented in Figure 5.1. The trendline shown in Figure 5.1 is given by equation (5.11). The upper and lower limits for the equivalent thermal resistance, $R_{T_{sup}}$ and $R_{T_{inf}}$, are given by equations (5.10) and (5.12) respectively.

$$R_{T_{sup}} = -3.25 \cdot 10^{-5} \cdot (T_{oil} - T_{room}) + 0.128 \quad (5.10)$$

$$R_T = -3.25 \cdot 10^{-5} \cdot (T_{oil} - T_{room}) + 0.119 \quad (5.11)$$

$$R_{T_{inf}} = -3.25 \cdot 10^{-5} \cdot (T_{oil} - T_{room}) + 0.109 \quad (5.12)$$

Figure 5.1 shows that the values for the 75W140-A are closer to the trendline than the ones for the 75W90-A. The dispersion verified for the thermal resistance of the 75W90-A is mainly due to the different weather conditions when the experimental tests were performed. Consequently, equation (5.11) mimics the gearbox behavior for the 75W140-A better than for the 75W90-A.

5.3 Stabilization temperature estimation

The stabilization temperature was obtained by an iterative process, assuming an room temperature and varying the oil temperature until the power loss estimated by the power loss model is equal to the evacuated heat estimated by the thermal model. When the evacuated heat matches the power loss, equation (5.1) is verified, and the system has reached the steady-state condition. The range of torque and speed were the same as the experimentally tested, 150 to 250 Nm and 600 to 800 rpm, respectively.

The iterative process, for each input condition, begins with the power loss prediction for a relative temperature of 35°C . The atmospheric temperature was set at 25°C for

the lubricant 75W140-A and at 35°C for the 75W90-A. These temperatures correspond to the lower relative stabilization temperature and the average room temperature verified experimentally for each lubricant.

Since the air condition may vary and consequently affect the stabilization temperature, the upper and lower limit for the thermal resistance were used to define a superior and inferior limit for the stabilization temperature.

5.3.1 75W140-A

For the lubricant 75W140-A the stabilization temperature function of the input torque and speed is given by Figure 5.2. For the tests performed with this lubricant it was verified that the average room temperature was 25°C, therefore the stabilization temperature was calculated relatively to this room temperature.

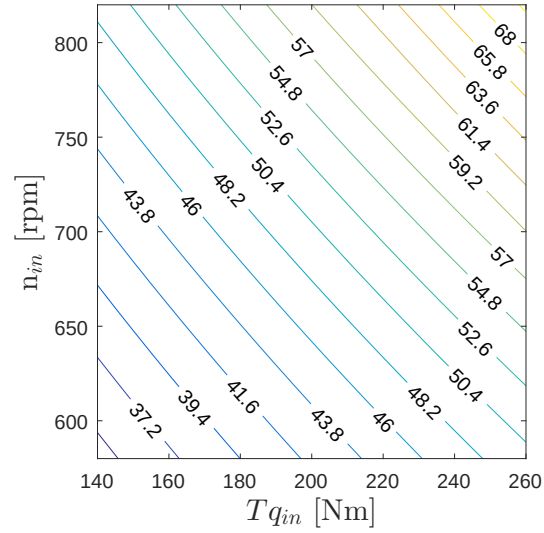


Figure 5.2: Relative stabilization temperatures ($T_{oil} - T_{room}$) at $T_{room} = 25^{\circ}\text{C}$ for 75W140-A.

The upper limit of the stabilization temperature, represented in Figure 5.3a, was given by the upper limit for the equivalent thermal resistance, and the values from Figure 5.3b were given by the lower limit.

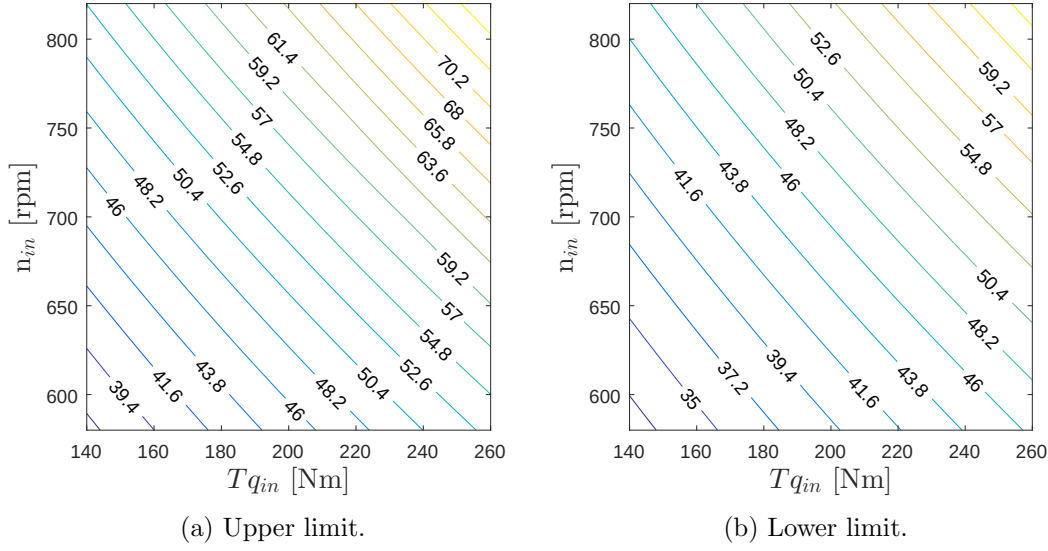


Figure 5.3: Limits for the stabilization temperature ($T_{oil} - T_{room}$) at $T_{room} = 25^\circ\text{C}$ for 75W140-A.

After the estimation, the values for the theoretical input power and torque imposed conditions were compared with the experimentally measured relative temperatures ($T_{oil} - T_{room}$).

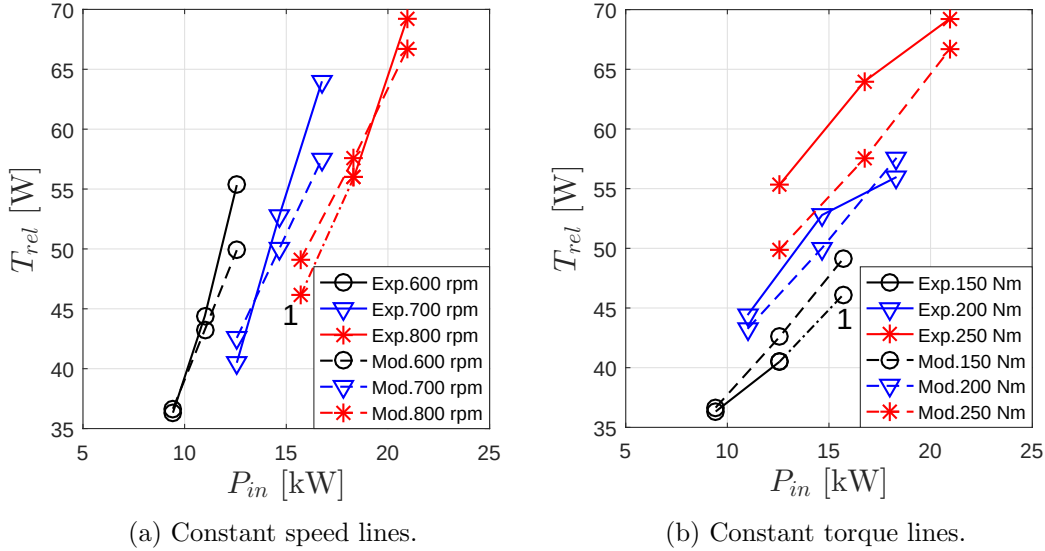


Figure 5.4: Model estimation and experimental values for the relative stabilization temperature for 75W140-A (1-Unstabilized test).

Figure 5.4, shows the model estimation and the measured values for the relative stabilization temperature for the lubricant 75W140-A, using the average approximation for the equivalent thermal resistance.

For this lubricant, the model approximates quite well the stabilization temperature for the two lower torques, and for the highest input torque the deviation is higher. The higher deviations can be explained by the dispersion in the equivalent resistance values verified

in Figure 5.1.

The room temperature, set at 25°C, was not verified in the experimental tests for higher input power. As consequence, even if the relative temperature is well estimated, the absolute lubricant temperature differs from the experimental value, affecting the results from the power loss model and consequently the ones of the thermal model.

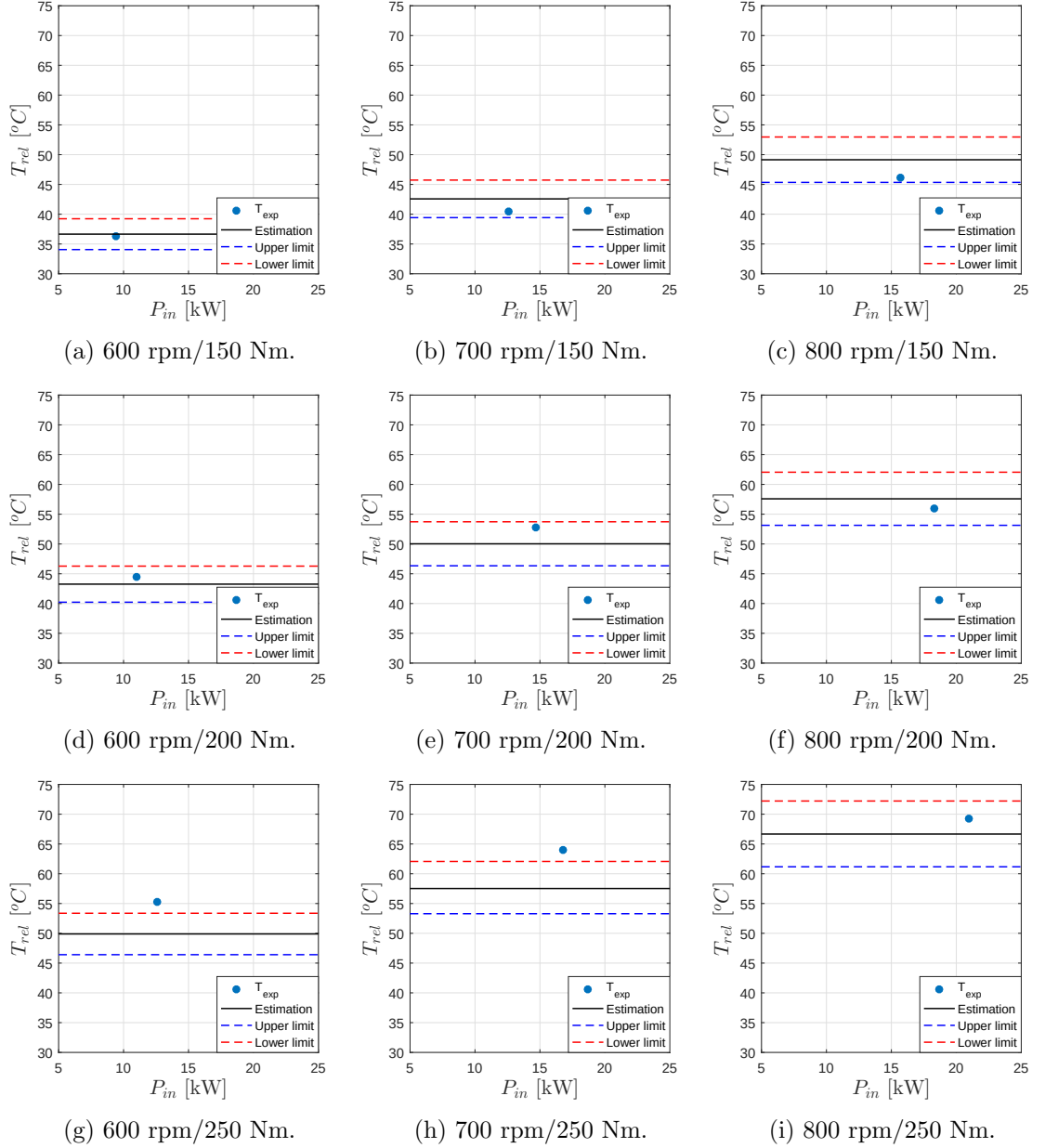


Figure 5.5: Measured relative stabilization temperature, model estimation and upper and lower limits for the stabilization temperature for 75W140-A ($T_{room} = 25^{\circ}\text{C}$).

Figure 5.5 shows the measured relative stabilization temperatures and the model estimations as well as the upper and lower limit for relative temperature. In Figure 5.5 it is visible that for only two conditions the measured temperature is out of the established boundaries. For the working conditions that are out of the limits, it was verified that for the test at 600 rpm and 250 Nm (Figure 5.5g) correspond to the value of the equivalent

thermal resistance that is below the established lower limit for this lubricant in Figure 5.1.

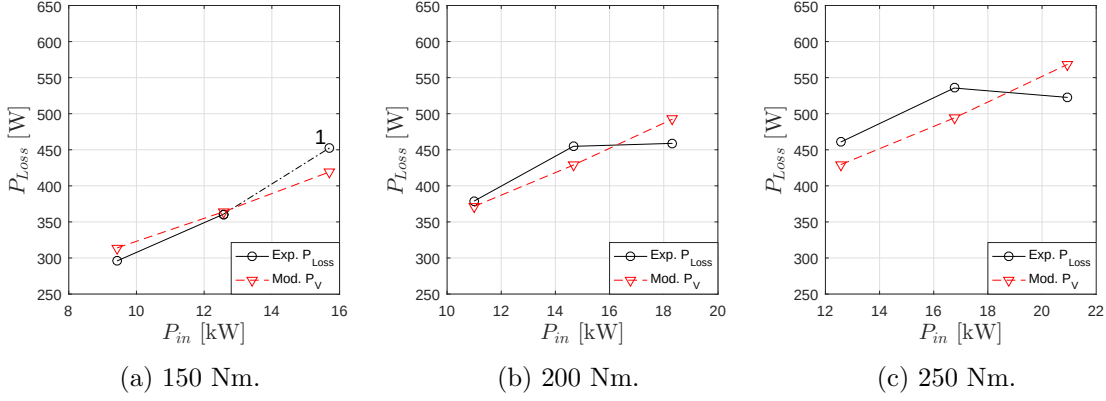


Figure 5.6: Constant torque lines of power loss model and experimental results for 75W140-A (1-Unstabilized test).

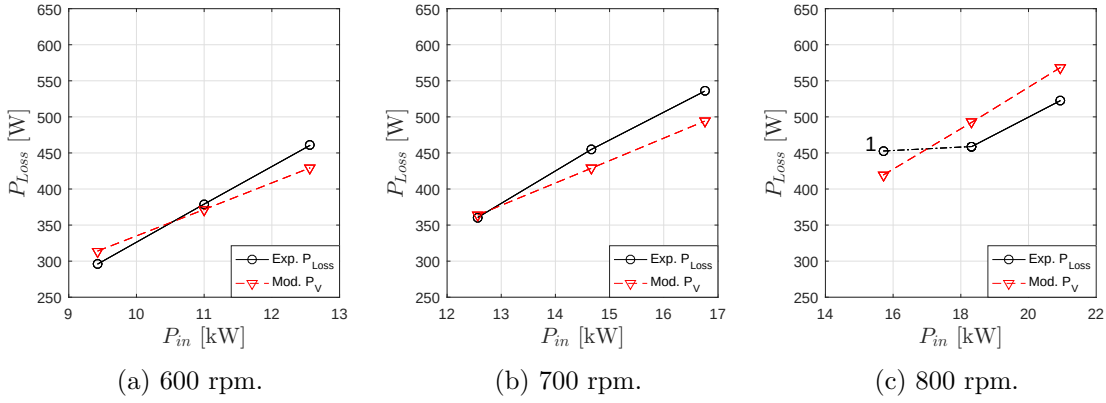


Figure 5.7: Constant speed lines of power loss model and experimental results for 75W140-A. (1-Unstabilized test).

Figures 5.6 and 5.7 compare the experimental measured power loss with the results from the independent power loss model. The model estimations were very consistent and presented a low deviation relative to the experimentally measured values. This new estimations, even though take into account the different stabilization temperatures from Figure 5.5, present the same trend as the experimental values.

The estimations are similar to the ones represented in Figures 4.14 and 4.15, but there is a noticeable difference mainly regarding the tests at 800 rpm (Figures 5.7c and 4.15c). The power loss estimations for the 800 rpm tests present a lower deviation for the final power loss model that for the one used previously. This convergence is a result of the general tendency of the final power loss model to underestimate the power loss for the high input power conditions.

5.3.2 75W90-A

The stabilization temperature for the 75W90-A lubricant is a function of the input torque and speed as shown in Figure 5.2. In order to better simulate the ambient conditions when the tests for this lubricant were performed, the room temperature was set at 35°C.

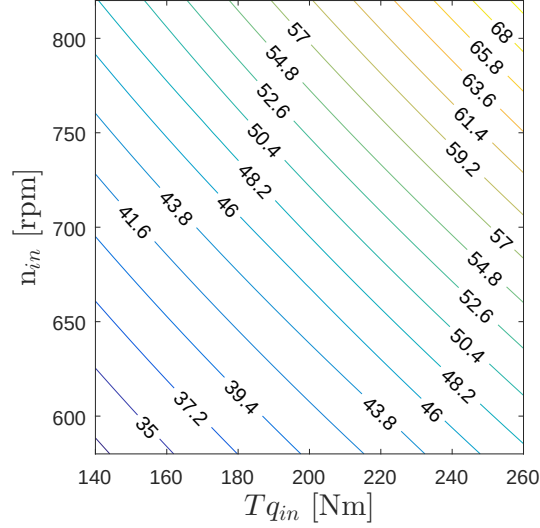


Figure 5.8: Relative stabilization temperatures ($T_{oil} - T_{room}$) at $T_{room} = 35^\circ\text{C}$ for 75W90-A.

Analogously to the 75W90-A, the upper limit of the stabilization temperature is represented in Figure 5.9a, and the values from Figure 5.9b give the lower limit for the stabilization temperature.

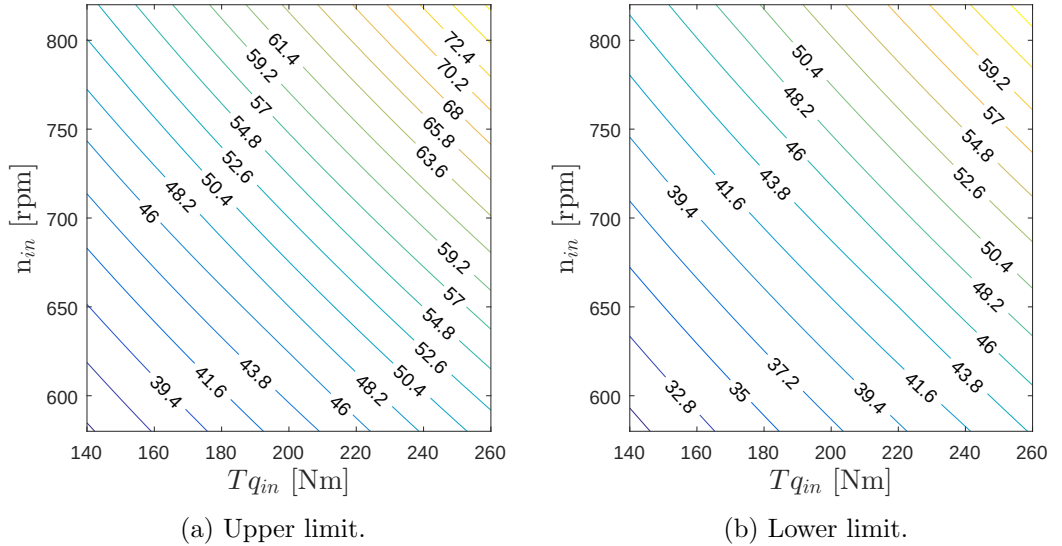


Figure 5.9: Limits for the stabilization temperature ($T_{oil} - T_{room}$) at $T_{room} = 35^\circ\text{C}$ for 75W90-A.

Figure 5.10 puts the model estimative and the measured values for the stabilization temperature for the lubricant 75W90-A side by side.

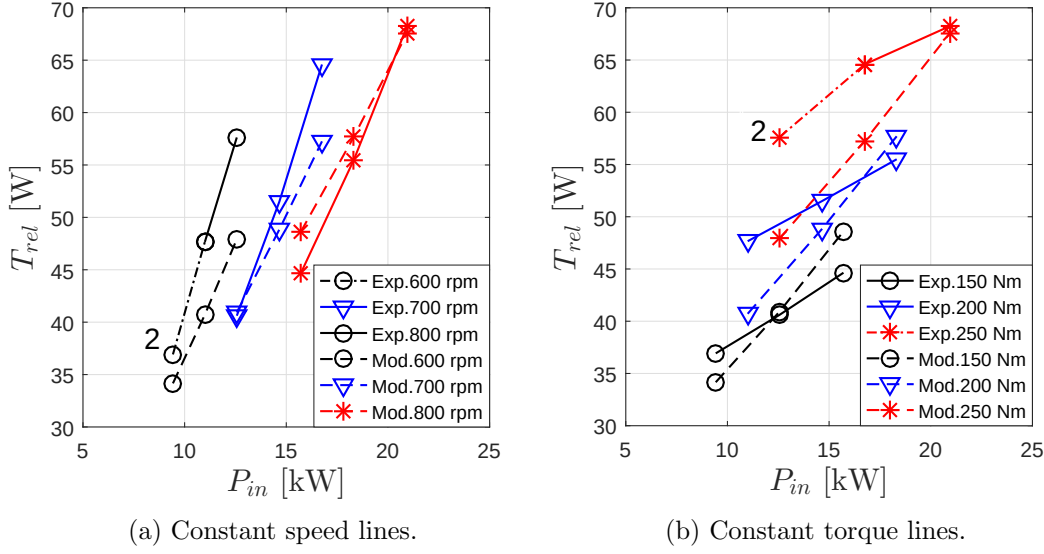


Figure 5.10: Model estimation and experimental values for the relative stabilization temperature ($T_{oil} - T_{room}$) for 75W90-A. (2-Unstabilized test).

For the lubricant 75W90-A, the stabilization temperature estimation differs from the experimentally measured, as shows Figure 5.10. Despite the difference, the values stay within the established limits, with the exception of three tests, as shows Figure 5.11.

Figure 5.11 shows the measured relative stabilization temperatures and the model estimations as well as the upper and lower limit for relative temperature for the lubricant 75W90-A. Note that the test with the stabilization temperature further from the limits (Figure 5.11g) was previously considered invalid. The values for the equivalent thermal resistance for the lubricant 75W90-A spread over a wide range, causing equation (5.11) to not represent very accurately its behavior. These facts are reflected in Figure 5.11, where it is visible that for two conditions the experimental stabilization temperature is out of the limits (Figure 5.11d and 5.11h) and three others are almost coincident with the limits (Figure 5.11a, 5.11c and 5.11e).

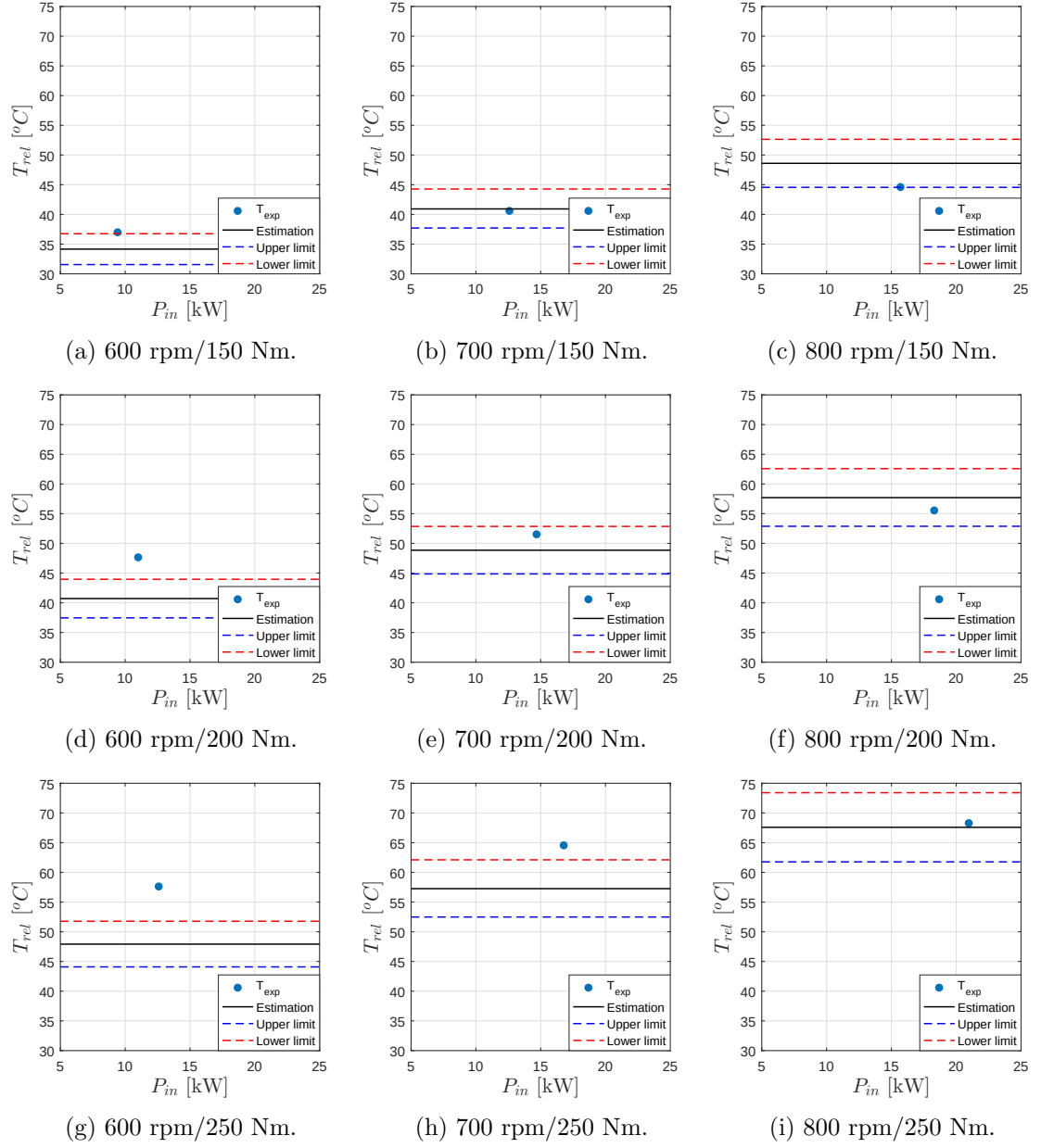


Figure 5.11: Measured stabilization temperate, model estimation and upper and lower limits for the stabilization temperature for 75W90-A ($T_{\text{room}} = 35^\circ\text{C}$).

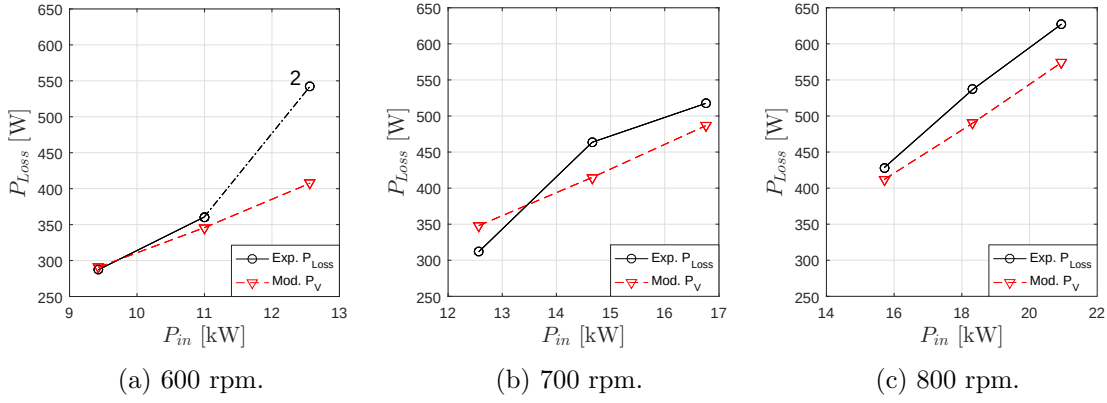


Figure 5.12: Constant speed lines of power loss model and experimental results for 75W90-A (2-Unstabilized test).

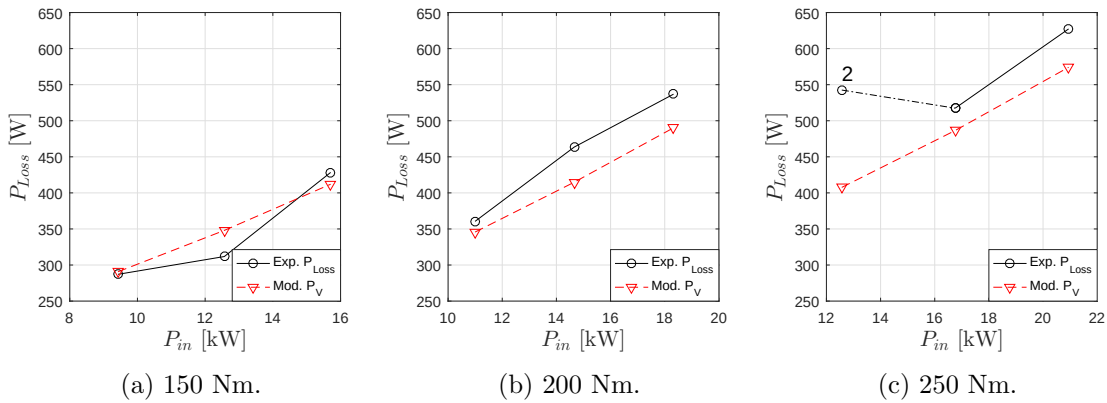


Figure 5.13: Constant torque lines of power loss model and experimental results for 75W90-A (2-Unstabilized test).

Figures 5.12 and 5.13 shows a comparison of the experimental measured power loss and the model estimations for the lubricant 75W90-A. The model estimates the power loss, in most cases, by default. For this lubricant, the estimations present a higher deviation with this model than with the one used initially, represented in Figures 4.16 and 4.17. The estimations for the 600 and 700 rpm are almost identical to the ones of Figure 4.17, despite the fact that the temperature estimations differ significantly from the experimental results. The deviation at higher torques and speeds may be related to the difference in the room temperature, that for this tests was higher than the considered in the model.

Conclusion

6.1 Conclusions

The main objective of this dissertation was to experimentally test and analyze the influence of lubricant formulations and working conditions in the efficiency of a vehicle transfer gearbox. The gearbox was tested using two similar lubricants with different viscous properties. The tests were performed in a range of 150 to 250 Nm of torque and 600 to 800 rpm of input speed.

In general, the lubricant 75W90-A offers a better performance of the gearbox for low speed and torque conditions. The 75W140-A presents the better performance for the high torque and speed. For the lubricant 75W140-A, the analysis of the results lead to the conclusion that the gearbox efficiency increases with the input torque, but slightly decreases when increasing the speed from 600 to 700 rpm and improves significantly for the highest speed.

For the 75W90-A, the results show that in general the efficiency is reduced when increasing the speed from 700 to 800 rpm and, in general, improves when increasing the torque.

Another goal of this work was to develop a model to estimate the gearbox power loss. The model was developed with success and the results were compared to the experimental measured values. In the majority of the test points the model results approximated the experimentally measured with a deviation that can be attributed to measuring error or simply the error inherent to the models used to estimate each loss type. The model is also able to breakdown the gearbox power loss allowing a deeper understanding of the phenomena that generate power loss.

It can be concluded, from the model estimations, that the similar global behavior of the two tested lubricants have different power loss partitions. According to the model results, generally, the 75W140-A presented a lower gear mesh power loss, mainly for the lower speed condition, than the 75W90-A. The opposite effect is verified when evaluating the churning loss. For this loss source, the 75W90-A presented lower values than the 75W140-A. The difference in the total power loss by churning effects is reduced as the input power, and consequently the temperature, increases. The rolling bearing losses also have a significant importance in the total power loss. When comparing this loss behavior for the two lubricants, the rolling bearing loss is higher for the lubricant 75W140-A, although the difference tends to reduce when increasing the power until 15 kW and then increases again until the maximum input power condition.

The power loss model and the thermal model were coupled, allowing to estimate the stabilization temperature of the gearbox, and consequently estimate the power loss, with-

out the need of an experimental test. The model was adapted with success, even though, for a few conditions, the stabilization temperatures were out of the established boundaries. The low accuracy is due to the uncertainty associated with the heat transfer mechanisms. These mechanisms are very sensitive to the ambient conditions and since the test room is ventilated directly from the atmosphere, the results are directly affected by these parameters. The power loss from the thermal model was compared with the experimentally measured, and was verified that the deviation was low for both lubricants, but tends to increase as the input power grows, leading to estimations by default.

6.2 Future Work

During the experimental campaign there were problems with the test rig that resulted in a 3 month delay of the experimental campaign and a consequent lack of time to test more lubricants and retest the lubricants for the failed working conditions. Therefore in the future the following works should be performed:

- Redo the 800 rpm tests for the 75W140-A;
- Retest the 75W90-A at 250 Nm and 600 rpm;
- Test the lubricant 75W90-B and compare the results with the other lubricants;
- Verify the power loss and thermal model estimations for the lubricant 75W90-B;
- Perform tests at torque and speed conditions outside the interval defined for this work in order to verify if the model is applicable outside the test range;
- Install equipment in the test room in order to measure and if possible control the room conditions during the test.

Also, in order to achieve better results for the final model, it could be done purely theoretical thermal approach to the problem, studying the gearbox geometry and the air and oil flows inside and outside of the gearbox, taking into account ambient conditions such as air humidity and pressure, as well as the air flow properties such as speed and temperature.

Bibliography

- [1] Maroua Hammami, Ramiro Martins, Mohamed Slim Abbes, Mohamed Haddar, and Jorge Seabra. Axle gear oils: Tribological characterization under full film lubrication. *Tribology International*, 106(February 2016):109–122, 2017. ISSN 0301679X. doi: 10.1016/j.triboint.2016.05.051.
- [2] P. Marques. Efficiency of a gearbox lubricated with wind turbine gear oils. Master’s thesis, Faculdade de Engenharia da Universidade do Porto, 2012.
- [3] D. Gonçalves. Efficiency of a gearbox lubricated with wind mill gear oils. Master’s thesis, Faculdade de Engenharia da Universidade do Porto, 2011.
- [4] J. Seabra, A. Campos, and A. Sottomayor. *Lubrificação elastohidrodinâmica*. 2002.
- [5] B. R. Munson, D. F. Young, and T. H. Okiishi. Fundamentals of fluid mechanics. *New York*, 1990.
- [6] ASTM D341 09. Standard practice for viscosity - temperature charts for liquid petroleum products.
- [7] ASTM. Standard practice for calculating viscosity index from kinematic viscosity at 40 and 100°C.
- [8] Carlos M.C.G. Fernandes, Pedro M.T. Marques, Ramiro C. Martins, and Jorge H.O. Seabra. Gearbox power loss. part i: Losses in rolling bearings. *Tribology International*, 88:298 – 308, 2015. ISSN 0301-679X. doi: <http://dx.doi.org/10.1016/j.triboint.2014.11.017>.
- [9] Carlos M.C.G. Fernandes, Pedro M.T. Marques, Ramiro C. Martins, and Jorge H.O. Seabra. Gearbox power loss. part ii: Friction losses in gears. *Tribology International*, 88:309 – 316, 2015. ISSN 0301-679X. doi: <http://dx.doi.org/10.1016/j.triboint.2014.12.004>.
- [10] Carlos M.C.G. Fernandes, Pedro M.T. Marques, Ramiro C. Martins, and Jorge H.O. Seabra. Gearbox power loss. part iii: Application to a parallel axis and a planetary gearbox. *Tribology International*, 88:317 – 326, 2015. ISSN 0301-679X. doi: <http://dx.doi.org/10.1016/j.triboint.2015.03.029>.
- [11] B.-R. Höhn, K. Michaelis, and M. Hinterstoißer. Optimization of gearbox efficiency. *goriva i maziva*, 48(4):462–480, 2009.
- [12] B.-R. Höhn, K. Michaelis, and T. Vollmer. Thermal rating of gear drives: Balance between power loss and heat dissipation. *AGMA Technical Paper*, 1996.

- [13] J. H. O. Seabra. Formulário-contactos lineares. mecânica do contacto e lubrificação. *Disciplina de tribologia*.
- [14] L. Schlenk. *Untersuchungen zur Fresstragfähigkeit von Grozahnradern*. PhD thesis, Dissertation TU München, 1994.
- [15] Yeau-Ren Jeng L. Chang and Pay-Yau Huang. Modeling and analysis of the meshing losses of involute spur gears in high-speed and high-load conditions. *Journal of tribology*, 135, 2012.
- [16] Pedro M. T. Marques, Ramiro C. Martins, and Jorge H. O. Seabra. *Power loss in planetary gearboxes including the influence of gear elastic and dynamic effects*. PhD thesis, Faculdade de Engenharia da Universidade do Porto.
- [17] Franco Concli and Carlo Gorla. Numerical modeling of the power losses in geared transmissions: Windage, churning and cavitation simulations with a new integrated approach that drastically reduces the computational effort. *Tribology International*, 103:58 – 68, 2016. ISSN 0301-679X. doi: <http://dx.doi.org/10.1016/j.triboint.2016.06.046>.
- [18] C. Changenet, G. Leprince, F. Ville, and P. Velez. A note on flow regimes and churning loss modeling. *Journal of Mechanical Design*, 133(12):121009, 2011. doi: 10.1115/1.4005330.
- [19] Pedro M. T. Marques, Carlos M. C. Fernandes, Ramiro C. Martins, and Jorge H. O. Seabra. Efficiency of a gearbox lubricated with wind turbine gear oils. *Tribology International*, 71(July):7–16, 2014. ISSN 0301679X. doi: 10.1016/j.triboint.2013.10.017.
- [20] Gauthier LePrince, Christophe Changenet, Fabrice Ville, Philippe Velez, Christophe Dufau, and Frédéric Jarnias. Influence of Aerated Lubricants on Gear Churning Losses - An Engineering model. *Tribology Transactions*, 54(6):929–938, 2011. doi: 10.1080/10402004.2011.597542.
- [21] SKF. *SKF General Catalogue 6000 EN*. SKF, 2013.
- [22] Freudenberg Simrit GmbH & Co. KG. *Technische Grundlagen, Simerringe und Rotationsdichtungen*. Freudenberg Simrit GmbH & Co. KG, Technisches Handbuch 2007.
- [23] QTC gears. *Elements of gear metric technology*. QTC gears.
- [24] Budynas R.G. and Nisbet J.K. *Meachanical Engineering Design*. 10th edition edition.
- [25] M. Hammami. *Efficiency and Wear in Automotive Gear Transmissions*. PhD thesis, University of Porto / University of Sfax, Yet to be Pubished.
- [26] J. Seabra, M. Hammami, C. Fernandes, B. Graça, and R. Martins. Axle gear oils: Tribological characterization and tribofilm generation. *IberTrib 2017*.
- [27] Kisssoft, calculation programs for machine design. <http://www.kisssoft.ch/english/home/index.php>.
- [28] Carlos M.C.G. Fernandes, Pedro M.T. Marques, Ramiro C. Martins, and Jorge H.O. Seabra. Gearbox power loss. part ii: Friction losses in gears. *Tribology International*, 88:309 – 316, 2015. ISSN 0301-679X. doi: <http://dx.doi.org/10.1016/j.triboint.2014.12.004>.

- [29] F.P. Incropera, T. L. Bergman, S. L. Adriene, and Dewitt P.D. *Fundamentals of heat and mass transfer*. 2007.

Test data

A.1 75w140 tests

Test nº1	date:1/3/2017	by: Carlos Pereira
	Oil Property	Units
Oil designation	75w140	-
Viscosity at 40°C	200.6	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	870	Kg/m^3
	Imposed working conditions	Units
n_{out}	600	rpm
TQ_{in}	344	Nm
Warm up period	270	min
Test period	30	min
	Practical working conditions	Units
n_{in}	261.65	rpm
n_{out}	599.58	rpm
TQ_{in}	335.34	Nm
TQ_{out}	141.63	Nm
	Temperature readings	Units
T_{oil}	59.768	°C
T_{wall}	51.733	°C
T_{amb}	23.444	°C
$T_{wallslave}$	54.428	°C
	Directly calculated data	Units
P_{in}	9188	W
P_{out}	8892	W
$T_{oil} - T_{amb}$	36.324	°C
ΔP	295.8	kW
Efficiency	96.79	%

Test nº2	date:1/3/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w140	
Viscosity at 40°C	200.6	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	600	rpm
TQ_{in}	458	Nm
Warm up period	210	min
Test period	30	min
Practical working conditions		Units
n_{in}	261.60	rpm
n_{out}	599.42	rpm
TQ_{in}	445.30	Nm
TQ_{out}	188.28	Nm
Temperature readings		Units
T_{oil}	69.159	°C
T_{wall}	59.709	°C
T_{amb}	24.752	°C
$T_{wallslave}$	59.149	°C
Directly calculated data		Units
P_{in}	12199	W
P_{out}	11820	W
$T_{oil} - T_{amb}$	44.407	°C
ΔP	378.8	kW
Efficiency	96.90	%

Test nº3	date:2/3/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w140	
Viscosity at 40°C	200.7	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	855	Kg/m ³
Imposed working conditions		Units
n_{out}	700	rpm
TQ_{in}	344	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	306.09	rpm
n_{out}	701.38	rpm
TQ_{in}	335.88	Nm
TQ_{out}	141.66	Nm
Temperature readings		Units
T_{oil}	64.887	°C
T_{wall}	55.968	°C
T_{amb}	24.392	°C
$T_{wallslave}$	58.647	°C
Directly calculated data		Units
P_{in}	10776	W
P_{out}	10406	W
$T_{oil} - T_{amb}$	40.495	°C
ΔP	360.5	kW
Efficiency	96.65	%

Test nº4	date:2/2/2017	by:Carlos Pereira
	Oil Property	Units
Oil designation	75w140	
Viscosity at 40°C	200.7	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	855	Kg/m ³
	Imposed working conditions	Units
n_{out}	600	rpm
TQ_{in}	573	Nm
Warm up period	210	min
Test period	30	min
	Practical working conditions	Units
n_{in}	261.52	rpm
n_{out}	599.21	rpm
TQ_{in}	557.04	Nm
TQ_{out}	235.74	Nm
	Temperature readings	Units
T_{oil}	81.148	°C
T_{wall}	68.418	°C
T_{amb}	25.816	°C
$T_{wallslave}$	64.408	°C
	Directly calculated data	Units
P_{in}	15255	W
P_{out}	14794	W
$T_{oil} - T_{amb}$	55.332	°C
ΔP	460.7	kW
Efficiency	96.98	%

Test nº5	date:3/3/32017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w140	
Viscosity at 40°C	200.6	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	855	Kg/m ³
Imposed working conditions		Units
n_{out}	700	rpm
TQ_{in}	458	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	306.10	rpm
n_{out}	701.42	rpm
TQ_{in}	445.74	Nm
TQ_{out}	188.32	Nm
Temperature readings		Units
T_{oil}	76.084	°C
T_{wall}	63.775	°C
T_{amb}	23.292	°C
$T_{wallslave}$	61.153	°C
Directly calculated data		Units
P_{in}	14288	W
P_{out}	13833	W
$T_{oil} - T_{amb}$	52.792	°C
ΔP	454.7	kW
Efficiency	96.82	%

Test nº6	date:3/3/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w140	
Viscosity at 40°C	200.6	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	855	Kg/m ³
Imposed working conditions		Units
n_{out}	700	rpm
TQ_{in}	573	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	306.13	rpm
n_{out}	701.39	rpm
TQ_{in}	557.01	Nm
TQ_{out}	235.78	Nm
Temperature readings		Units
T_{oil}	87.613	°C
T_{wall}	72.831	°C
T_{amb}	23.617	°C
$T_{wallslave}$	65.860	°C
Directly calculated data		Units
P_{in}	17856	W
P_{out}	17318	W
$T_{oil} - T_{amb}$	63.995	°C
ΔP	535.58	kW
Efficiency	97.00	%

Test nº7	date:6/3/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w140	
Viscosity at 40°C	200.6	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	855	Kg/m ³
Imposed working conditions		Units
n_{out}	800	rpm
TQ_{in}	343	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	351.22	rpm
n_{out}	804.85	rpm
TQ_{in}	335.42	Nm
TQ_{out}	141.00	Nm
Temperature readings		Units
T_{oil}	71.099	°C
T_{wall}	60.675	°C
T_{amb}	24.971	°C
$T_{wallslave}$	61.934	°C
Directly calculated data		Units
P_{in}	12337	W
P_{out}	11884	W
$T_{oil} - T_{amb}$	46.128	°C
ΔP	452.7	kW
Efficiency	96.34	%

Test nº8	date:6/3/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w140	
Viscosity at 40°C	200.6	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	855	Kg/m ³
Imposed working conditions		Units
n_{out}	800	rpm
TQ_{in}	458	Nm
Warm up period	210	min
Test period	30	min
Practical working conditions		Units
n_{in}	349.56	rpm
n_{out}	801.04	rpm
TQ_{in}	445.14	Nm
TQ_{out}	188.78	Nm
Temperature readings		Units
T_{oil}	82.764	°C
T_{wall}	70.287	°C
T_{amb}	26.790	°C
$T_{wallslave}$	66.965	°C
Directly calculated data		Units
P_{in}	16295	W
P_{out}	15836	W
$T_{oil} - T_{amb}$	55.974	°C
ΔP	458.9	kW
Efficiency	97.19	%

Test nº9	date:7/3/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w140	
Viscosity at 40°C	200.6	cSt
Viscosity at 70°C	61.86	cSt
Viscosity at 100°C	26.3	cSt
Density	855	Kg/m ³
Imposed working conditions		Units
n_{out}	800	rpm
TQ_{in}	573	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	363.62	rpm
n_{out}	833.27	rpm
TQ_{in}	556.97	Nm
TQ_{out}	237.60	Nm
Temperature readings		Units
T_{oil}	95.896	°C
T_{wall}	80.987	°C
T_{amb}	26.658	°C
$T_{wallslave}$	71.801	°C
Directly calculated data		Units
P_{in}	21209	W
P_{out}	20686	W
$T_{oil} - T_{amb}$	69.23	°C
ΔP	522.6	kW
Efficiency	97.54	

A.2 75w90 tests

Test nº1	date:5/6/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	600	rpm
TQ_{in}	343	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	261.10	rpm
n_{out}	598.37	rpm
TQ_{in}	355.51	Nm
TQ_{out}	141.83	Nm
Temperature readings		Units
T_{oil}	65.374	°C
T_{wall}	57.939	°C
T_{amb}	28.450	°C
$T_{wallslave}$	50.535	°C
Directly calculated data		Units
P_{in}	917	W
P_{out}	888	W
$T_{oil} - T_{amb}$	36.925	°C
ΔP	207	kW
Efficiency	96.87	%

Test n°2	date:6/6/2017	by: Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	700	rpm
TQ_{in}	458	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	261.06	rpm
n_{out}	598.37	rpm
TQ_{in}	445.40	Nm
TQ_{out}	188.61	Nm
Temperature readings		Units
T_{oil}	77.9053	°C
T_{wall}	68.021	°C
T_{amb}	30.246	°C
$T_{wallslave}$		°C
Directly calculated data		Units
P_{in}	12176	W
P_{out}	11816	W
$T_{oil} - T_{amb}$	59.768	°C
ΔP	360	kW
Efficiency	97.04	%

Test nº3	date:6/6/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	600	rpm
TQ_{in}	573	Nm
Warm up period	210	min
Test period	30	min
Practical working conditions		Units
n_{in}	260.83	rpm
n_{out}	597.79	rpm
TQ_{in}	557.38	Nm
TQ_{out}	243.56	Nm
Temperature readings		Units
T_{oil}	93.464	°C
T_{wall}	82.075	°C
T_{amb}	35.836	°C
$T_{wallslave}$	73.692	°C
Directly calculated data		Units
P_{in}	15224	W
P_{out}	14628	W
$T_{oil} - T_{amb}$	57.627	°C
ΔP	542	kW
Efficiency	96.44	%

Test nº4	date:7/6/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	700	rpm
TQ_{in}	343	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	304.93	rpm
n_{out}	698.77	rpm
TQ_{in}	355.01	Nm
TQ_{out}	141.93	Nm
Temperature readings		Units
T_{oil}	73.848	°C
T_{wall}	65.677	°C
T_{amb}	33.224	°C
$T_{wallslave}$	66.63	°C
Directly calculated data		Units
P_{in}	10698	W
P_{out}	10406	W
$T_{oil} - T_{amb}$	40.61	°C
ΔP	311.7	kW
Efficiency	97.09	%

Test nº5	date:7/6/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	700	rpm
TQ_{in}	458	Nm
Warm up period	210	min
Test period	30	min
Practical working conditions		Units
n_{in}	304.87	rpm
n_{out}	698.65	rpm
TQ_{in}	445.25	Nm
TQ_{out}	187.97	Nm
Temperature readings		Units
T_{oil}	86.710	°C
T_{wall}	76.566	°C
T_{amb}	35.202	°C
$T_{wallslave}$	72.506	°C
Directly calculated data		Units
P_{in}	14216	W
P_{out}	13752	W
$T_{oil} - T_{amb}$	51.50	°C
ΔP	463.557	kW
Efficiency	96.74	%

Test nº6	date:8/6/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	700	rpm
TQ_{in}	573	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	304.86	rpm
n_{out}	698.61	rpm
TQ_{in}	556.91	Nm
TQ_{out}	235.95	Nm
Temperature readings		Units
T_{oil}	97.021	°C
T_{wall}	85.148	°C
T_{amb}	32.435	°C
$T_{wallslave}$	74.44	°C
Directly calculated data		Units
P_{in}	17780	W
P_{out}	17262	W
$T_{oil} - T_{amb}$	64.58	°C
ΔP	517.40	kW
Efficiency	97.09	%

Test nº7	date:7/6/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	800	rpm
TQ_{in}	343	Nm
Warm up period	210	min
Test period	30	min
Practical working conditions		Units
n_{in}	348.78	rpm
n_{out}	799.29	rpm
TQ_{in}	335.62	Nm
TQ_{out}	141.34	Nm
Temperature readings		Units
T_{oil}	76.342	°C
T_{wall}	68.066	°C
T_{amb}	31.701	°C
$T_{wallslave}$	67.403	°C
Directly calculated data		Units
P_{in}	12258	W
P_{out}	11830	W
$T_{oil} - T_{amb}$	44.64	°C
ΔP	428.36	kW
Efficiency	96.51	%

Test nº8	date:7/6/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	800	rpm
TQ_{in}	458	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	348.71	rpm
n_{out}	799.11	rpm
TQ_{in}	445.38	Nm
TQ_{out}	187.93	Nm
Temperature readings		Units
T_{oil}	88.513	°C
T_{wall}	78.254	°C
T_{amb}	33.011	°C
$T_{wallslave}$	73.011	°C
Directly calculated data		Units
P_{in}	16264	W
P_{out}	15727	W
$T_{oil} - T_{amb}$	55.504	°C
ΔP	536.87	kW
Efficiency	96.70	%

Test nº9	date:9/6/2017	by:Carlos Pereira
Oil Property		Units
Oil designation	75w90	
Viscosity at 40°C	112.35	cSt
Viscosity at 70°C	36.7	cSt
Viscosity at 100°C	16.37	cSt
Density	870	Kg/m ³
Imposed working conditions		Units
n_{out}	800	rpm
TQ_{in}	573	Nm
Warm up period	270	min
Test period	30	min
Practical working conditions		Units
n_{in}	348.65	rpm
n_{out}	798.94	rpm
TQ_{in}	556.607	Nm
TQ_{out}	235.43	Nm
Temperature readings		Units
T_{oil}	103.031	°C
T_{wall}	90.903	°C
T_{amb}	34.785	°C
$T_{wallslave}$	79.432	°C
Directly calculated data		Units
P_{in}	20324	W
P_{out}	19697	W
$T_{oil} - T_{amb}$	68.246	°C
ΔP	627.3	kW
Efficiency	96.91	%

Appendix B

Gear geometrical parameters

No.	Item	Symbol	Formula
1	Radial Module	m_t	
2	Radial Pressure Angle	α_t	
3	Helix Angle	β	
4	Number of Teeth & Helical Hand		
5	Radial Coefficient of Profile Shift	x_{t1}, x_{t2}	
6	Involute Function α_{wt}	$\text{inv } \alpha_{wt}$	$2 \tan \alpha_t \left(\frac{x_{t1} + x_{t2}}{z_1 + z_2} \right) + \text{inv } \alpha_t$
7	Radial Working Pressure Angle	α_{wt}	Find from Involute Function Table
8	Center Distance Increment Factor	y	$\frac{z_1 + z_2}{2} \left(\frac{\cos \alpha_t}{\cos \alpha_{wt}} - 1 \right)$
9	Center Distance	a_x	$\left(\frac{z_1 + z_2}{2} + y \right) m_t$
10	Standard Pitch Diameter	d	$z m_t$
11	Base Diameter	d_b	$d \cos \alpha_t$
12	Working Pitch Diameter	d_w	$\frac{d_b}{\cos \alpha_{wt}}$
13	Addendum	h_{a1} h_{a2}	$(1 + y - x_{t2}) m_t$ $(1 + y - x_{t1}) m_t$
14	Whole Depth	h	$[2.25 + y - (x_{t1} + x_{t2})] m_t$
15	Outside Diameter	d_a	$d + 2 h_a$
16	Root Diameter	d_f	$d_a - 2 h$

KISSSys reports

_O.GroupBox.GearPair_const1.GearPair_const1_calc

KISSsoft Release 03/2016 C

KISSsoft University license - Universidade do Porto

File

Name : Unnamed

Changed by: em12072

on: 11.07.2017

at: 22:32:29

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:

Gear 1: z1(GearPair_const1)

Gear 2: z2(GearPair_const1)

Calculation method

DIN 3990:1987 Method B

----- GEAR 1 ----- GEAR 2 --

Power (kW)	[P]	14.676	
Speed (1/min)	[n]	306.0	425.7
Torque (Nm)	[T]	458.0	329.2
Application factor	[KA]		1.00
Required service life (h)	[H]	20000.00	
Gear driving (+) / driven (-)		+	-
Working flank gear 1: Right flank			
Sense of rotation gear 1 clockwise			

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

----- GEAR 1 ----- GEAR 2 --

Center distance (mm)	[a]	105.011	
Centre distance tolerance	ISO 286:2010 Measure js7		
Normal module (mm)	[mn]	3.5000	
Pressure angle at normal section (°)	[alfn]	20.0000	
Helix angle at reference circle (°)	[beta]	20.0000	
Number of teeth	[z]	32	23
Facewidth (mm)	[b]	35.00	35.00
Hand of gear		left	right
Accuracy grade	[Q-DIN 3961:1978]	6	6
Inner diameter (mm)	[di]	20.00	20.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00

Material

Gear 1: 18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), core strength $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

Gear 2: 18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), core strength $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

----- GEAR 1 ----- GEAR 2 --

Surface hardness		HRC 61	HRC 61
Fatigue strength. tooth root stress (N/mm ²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm ²)	[σB]	1200.00	1200.00
Yield point (N/mm ²)	[σS]	850.00	850.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.60	0.60

Roughness average value DS, root (µm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (µm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (µm)	[RZF]	20.00	20.00

Gear reference profile	1 :		
Reference profile	1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height factor	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	
		not topping	

Gear reference profile	2 :		
Reference profile	1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height factor	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	
		not topping	

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofpP*]	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height factor	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (µm)	[Ca]	2.0	2.0
-----------------	------	-----	-----

Lubrication type Oil bath lubrication

Type of oil Oil: ISO-VG 220

Lubricant base Mineral-oil base

Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	220.00
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	17.50
Specific density at 15 °C (kg/dm³)	[roOil]	0.895
Oil temperature (°C)	[TS]	70.000

		----- GEAR 1 -----	GEAR 2 --
Overall transmission ratio	[itot]	-0.719	
Gear ratio	[u]	1.391	
Transverse module (mm)	[mt]	3.725	
Pressure angle at pitch circle (°)	[alf]	21.173	
Working transverse pressure angle (°)	[alfwt]	24.557	
	[alfwt.e/i]	24.578 /	24.536
Working pressure angle at normal section (°)	[alfwn]	23.176	

Helix angle at operating pitch circle (°)	[betaw]	20.463		
Base helix angle (°)	[betab]	18.747		
Reference centre distance (mm)	[ad]	102.427		
Sum of profile shift coefficients	[Summexi]	0.7960		
Profile shift coefficient	[x]	0.3810	0.4150	
Tooth thickness (Arc) (module) (module)	[sn*]	1.8481	1.8729	
Tip alteration (mm)	[k*mn]	-0.202	-0.202	
Reference diameter (mm)	[d]	119.188	85.666	
Base diameter (mm)	[db]	111.142	79.883	
Tip diameter (mm)	[da]	128.451	95.167	
(mm)	[da.e/i]	128.451 / 128.441	95.167 / 95.157	
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010	
Tip form diameter (mm)	[dFa]	128.451	95.167	
(mm)	[dFa.e/i]	128.451 / 128.441	95.167 / 95.157	
Active tip diameter (mm)	[dNa]	128.451	95.167	
Active tip diameter (mm)	[dNa.e/i]	128.451 / 128.441	95.167 / 95.157	
Operating pitch diameter (mm)	[dw]	122.195	87.828	
(mm)	[dw.e/i]	122.216 / 122.175	87.842 / 87.813	
Root diameter (mm)	[df]	113.105	79.821	
Generating Profile shift coefficient	[xE.e/i]	0.3535/ 0.3378	0.3875/ 0.3718	
Manufactured root diameter with xE (mm)	[df.e/i]	112.913 / 112.803	79.629 / 79.519	
Theoretical tip clearance (mm)	[c]	0.875	0.875	
Effective tip clearance (mm)	[c.e/i]	1.049 / 0.954	1.049 / 0.954	
Active root diameter (mm)	[dNf]	116.693	83.098	
(mm)	[dNf.e/i]	116.724 / 116.667	83.127 / 83.075	
Root form diameter (mm)	[dFf]	115.399	82.254	
(mm)	[dFf.e/i]	115.256 / 115.176	82.128 / 82.058	
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.774 / 0.705	0.534 / 0.473	
Addendum (mm)	[ha=mn*(haP*+x+k)]	4.632	4.751	
(mm)	[ha.e/i]	4.632 / 4.627	4.751 / 4.745	
Dedendum (mm)	[hf=mn*(hfP*-x)]	3.041	2.922	
(mm)	[hf.e/i]	3.138 / 3.193	3.019 / 3.074	
Roll angle at dFa (°)	[xsi_dFa.e/i]	33.198 / 33.188	37.099 / 37.086	
Roll angle to dNa (°)	[xsi_dNa.e/i]	33.198 / 33.188	37.099 / 37.086	
Roll angle to dNf (°)	[xsi_dNf.e/i]	18.385 / 18.289	16.491 / 16.356	
Roll angle at dFf (°)	[xsi_dFf.e/i]	15.734 / 15.577	13.679 / 13.461	
Tooth height (mm)	[h]	7.673	7.673	
Virtual gear no. of teeth	[zn]	37.976	27.296	
Normal tooth thickness at tip circle (mm)	[san]	2.544	2.324	
(mm)	[san.e/i]	2.475 / 2.427	2.253 / 2.203	
Normal-tooth thickness on tip form circle (mm)	[sFan]	2.544	2.324	
(mm)	[sFan.e/i]	2.475 / 2.427	2.253 / 2.203	
Normal space width at root circle (mm)	[efn]	2.654	0.000	
(mm)	[efn.e/i]	2.683 / 2.700	0.000 / 0.000	
Max. sliding velocity at tip (m/s)	[vga]	0.522	0.583	
Specific sliding at the tip	[zetaa]	0.505	0.506	
Specific sliding at the root	[zetaf]	-1.024	-1.022	
Mean specific sliding	[zetam]	0.506		
Sliding factor on tip	[Kga]	0.266	0.298	
Sliding factor on root	[Kgf]	-0.298	-0.266	
Pitch on reference circle (mm)	[pt]	11.701		
Base pitch (mm)	[pbt]	10.911		
Transverse pitch on contact-path (mm)	[pet]	10.911		
Lead height (mm)	[pz]	1028.765	739.425	
Axial pitch (mm)	[px]	32.149		
Length of path of contact (mm)	[ga, e/i]	14.418 (14.460 / 14.357)		

Length T1-A, T2-A (mm)	[T1A, T2A]	17.781(17.739/ 17.832)	25.862(25.862/ 25.853)
Length T1-B (mm)	[T1B, T2B]	21.287(21.287/ 21.278)	22.356(22.313/ 22.408)
Length T1-C (mm)	[T1C, T2C]	25.392(25.368/ 25.417)	18.251(18.233/ 18.268)
Length T1-D (mm)	[T1D, T2D]	28.692(28.650/ 28.743)	14.951(14.951/ 14.942)
Length T1-E (mm)	[T1E, T2E]	32.199(32.199/ 32.189)	11.444(11.402/ 11.496)
Length T1-T2 (mm)	[T1T2]	43.643 (43.601 / 43.685)	
Diameter of single contact point B (mm)	[d-B]	119.018(119.018/ 119.011)	91.545(91.504/ 91.596)
Diameter of single contact point D (mm)	[d-D]	125.082(125.043/ 125.129)	85.297(85.297/ 85.290)
Addendum contact ratio	[eps]	0.624(0.626/ 0.621)	0.698(0.699/ 0.695)
Minimal length of contact line (mm)	[Lmin]	47.872	
Transverse contact ratio	[eps_a]	1.321	
Transverse contact ratio with allowances	[eps_a.e/m/i]	1.325 / 1.321 / 1.316	
Overlap ratio	[eps_b]	1.089	
Total contact ratio	[eps_g]	2.410	
Total contact ratio with allowances	[eps_g.e/m/i]	2.414 / 2.409 / 2.404	

2. FACTORS OF GENERAL INFLUENCE

		----- GEAR 1 ----- GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]	7685.3
Axial force (N)	[Fa]	2797.2
Radial force (N)	[Fr]	2976.8
Normal force (N)	[Fnorm]	8703.5
Nominal circumferential force per mm (N/mm)	[w]	219.58
Only as information: Forces at operating pitch circle:		
Nominal circumferential force (N)	[Ftw]	7496.2
Axial force (N)	[Faw]	2797.2
Radial force (N)	[Frw]	3425.3
Circumferential speed reference circle (m/s)	[v]	1.91
Circumferential speed operating pitch circle (m/s)	[v(dw)]	1.96
Running-in value (μm)	[yp]	0.6
Running-in value (μm)	[yf]	0.6
Correction coefficient	[CM]	0.800
Gear body coefficient	[CR]	1.000
Basic rack factor	[CBS]	0.975
Material coefficient	[E/Est]	1.000
Singular tooth stiffness (N/mm/μm)	[c']	13.745
Meshing stiffness (N/mm/μm)	[cg]	17.058
Reduced mass (kg/mm)	[mRed]	0.01838
Resonance speed (min-1)	[nE1]	9091
Resonance ratio (-)	[N]	0.034
Subcritical range		
Running-in value (μm)	[ya]	0.6
Bearing distance l of pinion shaft (mm)	[l]	70.000
Distance s of pinion shaft (mm)	[s]	7.000
Outside diameter of pinion shaft (mm)	[dsh]	35.000
Load according to Figure 6.8, DIN 3990-1:1987 [-]	4	
(0:6.8a, 1:6.8b, 2:6.8c, 3:6.8d, 4:6.8e)		
Coefficient K' according to Figure 6.8, DIN 3990-1:1987 [K']	-1.00	
Without support effect		
Tooth trace deviation (active) (μm)	[Fby]	3.83
from deformation of shaft (μm)	[fsh*B1]	1.70
(fsh (μm) = 1.70, B1 = 1.00, fHb5 (μm) = 6.50)		

_O.GroupBox.GearPair_const2.GearPair_const2_calc

KISSsoft Release 03/2016 C

KISSsoft University license - Universidade do Porto

File

Name : Unnamed
Changed by: em12072 on: 11.07.2017 at: 22:32:31

Important hint: At least one warning has occurred during the calculation:

1-> Gear 2: The specific sliding at the root [zeta_f] is less than -3.00.

2-> Notice concerning gear 2:

Dimension over balls is not measurable (facewidth is too small)!

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:

Gear 1: z3(GearPair_const2)

Gear 2: z4(GearPair_const2)

Calculation method DIN 3990:1987 Method B

	----- GEAR 1 -----	----- GEAR 2 --
Power (kW)	[P]	14.676
Speed (1/min)	[n]	425.7 701.2
Torque (Nm)	[T]	329.2 199.9
Application factor	[KA]	1.25
Required service life (h)	[H]	20000.00
Gear driving (+) / driven (-)	+	-
Working flank gear 1: Left flank		
Sense of rotation gear 1 counterclockwise		

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

	----- GEAR 1 -----	----- GEAR 2 --
Center distance (mm)	[a]	94.998
Centre distance tolerance	ISO 286:2010 Measure js7	
Normal module (mm)	[mn]	4.0000
Pressure angle at normal section (°)	[alfn]	20.0000
Helix angle at reference circle (°)	[beta]	20.0000
Number of teeth	[z]	28 17
Facewidth (mm)	[b]	24.73 24.73
Hand of gear	right	left
Accuracy grade	[Q-DIN 3961:1978]	6 6
Inner diameter (mm)	[di]	20.00 20.00
Inner diameter of gear rim (mm)	[dbi]	0.00 0.00

Material

Gear 1: 18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), core strength >=25HRC Jominy J=12mm<HRC28

Gear 2: 18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), core strength $\geq 25\text{HRC}$ Jominy $J=12\text{mm}$ $< \text{HRC}28$

----- GEAR 1 ----- GEAR 2 --

Surface hardness		HRC 61	HRC 61
Fatigue strength. tooth root stress (N/mm ²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm ²)	[σB]	1200.00	1200.00
Yield point (N/mm ²)	[σS]	850.00	850.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.60	0.60
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile	1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height factor	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Gear reference profile 2 :

Reference profile	1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height factor	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofpP*]	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height factor	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (μm)	[Ca]	2.0	2.0
-----------------	------	-----	-----

Lubrication type Oil bath lubrication

Type of oil Oil: ISO-VG 220

Lubricant base Mineral-oil base

Kinem. viscosity oil at 40 °C (mm ² /s)	[nu40]	220.00
Kinem. viscosity oil at 100 °C (mm ² /s)	[nu100]	17.50
Specific density at 15 °C (kg/dm ³)	[roOil]	0.895

Oil temperature (°C)	[TS]	70.000			
----- GEAR 1 ----- GEAR 2 --					
Overall transmission ratio	[itot]	-0.607			
Gear ratio	[u]	1.647			
Transverse module (mm)	[mt]	4.257			
Pressure angle at pitch circle (°)	[alft]	21.173			
Working transverse pressure angle (°)	[alfwt]	19.927			
	[alfwt.e/i]	19.956 /		19.898	
Working pressure angle at normal section (°)	[alfwn]	18.829			
Helix angle at operating pitch circle (°)	[betaw]	19.850			
Base helix angle (°)	[betab]	18.747			
Reference centre distance (mm)	[ad]	95.776			
Sum of profile shift coefficients	[Summexi]	-0.1890			
Profile shift coefficient	[x]	-0.2400	0.0510		
Tooth thickness (Arc) (module) (module)	[sn*]	1.3961	1.6079		
Tip alteration (mm)	[k*mn]	-0.022	-0.022		
Reference diameter (mm)	[d]	119.188	72.364		
Base diameter (mm)	[db]	111.142	67.479		
Tip diameter (mm)	[da]	125.224	80.728		
(mm)	[da.e/i]	125.224 /	125.214	80.728 /	80.718
Tip diameter allowances (mm)	[Ada.e/i]	0.000 /	-0.010	0.000 /	-0.010
Tip form diameter (mm)	[dFa]	125.224	80.728		
(mm)	[dFa.e/i]	125.224 /	125.214	80.728 /	80.718
Active tip diameter (mm)	[dNa]	125.224	80.728		
Active tip diameter (mm)	[dNa.e/i]	125.224 /	125.214	80.728 /	80.718
Operating pitch diameter (mm)	[dw]	118.220	71.777		
(mm)	[dw.e/i]	118.242 /	118.198	71.790 /	71.763
Root diameter (mm)	[df]	107.268	62.772		
Generating Profile shift coefficient	[xE.e/i]	-0.2640/	-0.2778	0.0270/	0.0132
Manufactured root diameter with xE (mm)	[df.e/i]	107.076 /	106.966	62.580 /	62.470
Theoretical tip clearance (mm)	[c]	1.000	1.000		
Effective tip clearance (mm)	[c.e/i]	1.174 /	1.079	1.174 /	1.079
Active root diameter (mm)	[dNf]	113.007	67.848		
(mm)	[dNf.e/i]	113.028 /	112.988	67.861 /	67.837
Root form diameter (mm)	[dFf]	112.229	67.673		
(mm)	[dFf.e/i]	112.157 /	112.116	67.635 /	67.615
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.456 /	0.416	0.123 /	0.101
Addendum (mm)	[ha=mn*(haP*+x+k)]	3.018	4.182		
(mm)	[ha.e/i]	3.018 /	3.013	4.182 /	4.177
Dedendum (mm)	[hf=mn*(hfP*-x)]	5.960	4.796		
(mm)	[hf.e/i]	6.056 /	6.111	4.892 /	4.947
Roll angle at dFa (°)	[xsi_dFa.e/i]	29.742 /	29.730	37.625 /	37.610
Roll angle to dNa (°)	[xsi_dNa.e/i]	29.742 /	29.730	37.625 /	37.610
Roll angle to dNf (°)	[xsi_dNf.e/i]	10.601 /	10.486	6.102 /	5.909
Roll angle at dFf (°)	[xsi_dFf.e/i]	7.759 /	7.602	3.893 /	3.635
Tooth height (mm)	[h]	8.978	8.978		
Virtual gear no. of teeth	[zn]	33.229	20.175		
Normal tooth thickness at tip circle (mm)	[san]	3.201	2.749		
(mm)	[san.e/i]	3.132 /	3.086	2.678 /	2.628
Normal-tooth thickness on tip form circle (mm)	[sFan]	3.201	2.749		
(mm)	[sFan.e/i]	3.132 /	3.086	2.678 /	2.628
Normal space width at root circle (mm)	[efn]	0.000	0.000		
(mm)	[efn.e/i]	0.000 /	0.000	0.000 /	0.000
Max. sliding velocity at tip (m/s)	[vga]	1.027	1.171		
Specific sliding at the tip	[zetaa]	0.798	0.720		

Specific sliding at the root	[zetaf]	-2.570	-3.960
Mean specific sliding	[zetam]	0.757	
Sliding factor on tip	[Kga]	0.390	0.444
Sliding factor on root	[Kgf]	-0.444	-0.390
Pitch on reference circle (mm)	[pt]	13.373	
Base pitch (mm)	[pbt]	12.470	
Transverse pitch on contact-path (mm)	[pet]	12.470	
Lead height (mm)	[pz]	1028.765	624.607
Axial pitch (mm)	[px]	36.742	
Length of path of contact (mm)	[ga, e/i]	18.625 (18.676 / 18.554)	
Length T1-A, T2-A (mm)	[T1A, T2A]	10.221(10.170/ 10.282)	22.156(22.156/ 22.147)
Length T1-B (mm)	[T1B, T2B]	16.376(16.376/ 16.365)	16.001(15.950/ 16.063)
Length T1-C (mm)	[T1C, T2C]	20.146(20.114/ 20.178)	12.232(12.212/ 12.251)
Length T1-D (mm)	[T1D, T2D]	22.691(22.640/ 22.752)	9.686(9.686/ 9.677)
Length T1-E (mm)	[T1E, T2E]	28.846(28.846/ 28.836)	3.531(3.480/ 3.593)
Length T1-T2 (mm)	[T1T2]	32.378 (32.326 / 32.429)	
Diameter of single contact point B (mm)	[d-B]	115.868(115.868/ 115.861)	74.683(74.639/ 74.737)
Diameter of single contact point D (mm)	[d-D]	120.051(120.012/ 120.097)	70.205(70.205/ 70.200)
Addendum contact ratio	[eps]	0.698(0.700/ 0.694)	0.796(0.797/ 0.794)
Minimal length of contact line (mm)	[Lmin]	32.586	
Transverse contact ratio	[eps_a]	1.494	
Transverse contact ratio with allowances	[eps_a.e/m/i]	1.498 / 1.493 / 1.488	
Overlap ratio	[eps_b]	0.673	
Total contact ratio	[eps_g]	2.167	
Total contact ratio with allowances	[eps_g.e/m/i]	2.171 / 2.166 / 2.161	

2. FACTORS OF GENERAL INFLUENCE

		----- GEAR 1 -----	GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]	5523.8	
Axial force (N)	[Fa]	2010.5	
Radial force (N)	[Fr]	2139.5	
Normal force (N)	[Fnorm]	6255.6	
Nominal circumferential force per mm (N/mm)	[w]	223.35	
Only as information: Forces at operating pitch circle:			
Nominal circumferential force (N)	[Ftw]	5569.1	
Axial force (N)	[Faw]	2010.5	
Radial force (N)	[Frw]	2018.9	
Circumferential speed reference circle (m/s)	[v]	2.66	
Circumferential speed operating pitch circle (m/s)	[v(dw)]	2.64	
Running-in value (μm)	[yp]	0.7	
Running-in value (μm)	[yf]	0.8	
Correction coefficient	[CM]	0.800	
Gear body coefficient	[CR]	1.000	
Basic rack factor	[CBS]	0.975	
Material coefficient	[E/Est]	1.000	
Singular tooth stiffness (N/mm/μm)	[c']	11.380	
Meshing stiffness (N/mm/μm)	[cg]	15.592	
Reduced mass (kg/mm)	[mRed]	0.01278	
Resonance speed (min-1)	[nE1]	11912	
Resonance ratio (-)	[N]	0.036	
Subcritical range			
Running-in value (μm)	[ya]	0.7	
Bearing distance l of pinion shaft (mm)	[l]	49.500	

_O.GroupBox.Shaft1.Shaft1_calc

KISSsoft Release 03/2016 C

KISSsoft University license - Universidade do Porto

File

Name : Unnamed

Changed by: em12072

on: 11.07.2017

at: 22:32:37

Important hint: At least one warning has occurred during the calculation:

1-> The required service life of bearing 'Shaft 'Shaft1', Rolling bearing 'RollerBearing1' is not achieved!

2-> The required service life of bearing 'Shaft 'Shaft1', Rolling bearing 'RollerBearing2' is not achieved!

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	Shaft1
Drawing	
Initial position (mm)	0.000
Length (mm)	165.500
Speed (1/min)	306.00
Sense of rotation: clockwise	

Material	C45 (1)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	0.916
Weight of shaft, including additional masses (kg)	0.916
Mass moment of inertia (kg*mm ²)	103.049
Momentum of mass GD2 (Nm ²)	0.004

The direction of the weight is not considered

Consider deformations due to shearing

Shear correction coefficient	1.100
------------------------------	-------

Contact angle of rolling bearings is considered

Tolerance field: Mean value

Reference temperature (°C)	20.000
----------------------------	--------

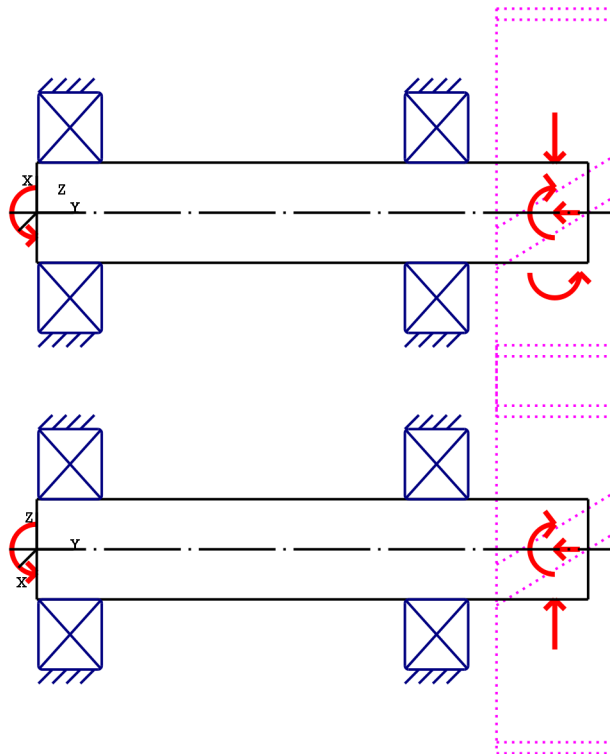


Figure: Load applications

Shaft definition (Shaft1)

Outer contour

Cylinder (Cylinder)		0.000mm ... 165.500mm
Diameter (mm)	[d]	30.0000
Length (mm)	[l]	165.5000
Surface roughness (µm)	[Rz]	8.0000

Forces

Type of force element	Coupling	
Label in the model	Coupling1(Boundary1)	
Position on shaft (mm)	[y _{local}]	0.0000
Position in global system (mm)	[y _{global}]	0.0000
Effective diameter (mm)		20.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		15.0000
Power (kW)		14.6763 driven (Input)
Torque (Nm)		458.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000

Mass moment of inertia Jp (kg*m ²)	0.0000
Mass moment of inertia Jxx (kg*m ²)	0.0000
Mass moment of inertia Jzz (kg*m ²)	0.0000
Eccentricity (mm)	0.0000

Type of force element	Cylindrical gear	
Label in the model	z1(GearPair_const1)	
Position on shaft (mm)	[ylocal]	155.5000
Position in global system (mm)	[yglobal]	155.5000
Operating pitch diameter (mm)		122.1951
Helix angle (°)		20.4632 left
Working pressure angle at normal section (°)		23.1758
Position of contact (°)		0.0000
Length of load application (mm)		35.0000
Power (kW)		14.6763 driving (Output)
Torque (Nm)		-458.0000
Axial force (N)		-2797.2362
Shearing force X (N)		-3425.2695
Shearing force Z (N)		7496.2063
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		-170.9043

Bearing

Label in the model		RollerBearing1
Bearing type		SKF *6306
Bearing type		Deep groove ball bearing (single row)
Bearing position (mm)	[ylocal]	10.000
Bearing position (mm)	[yglobal]	10.000
Attachment of external ring		Fixed bearing
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	72.000
Width (mm)	[b]	19.000
Corner radius (mm)	[r]	1.100
Basic static load rating	[C ₀]	16.000
Basic dynamic load rating	[C]	29.600
Fatigue load rating	[C _U]	0.670
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Label in the model		RollerBearing2
Bearing type		SKF *NJ 306 ECM
Bearing type		Cylindrical roller bearing (single row)
Bearing position (mm)	[ylocal]	120.000
Bearing position (mm)	[yglobal]	120.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	72.000
Width (mm)	[b]	19.000
Corner radius (mm)	[r]	1.100
Basic static load rating	[C ₀]	48.000
Basic dynamic load rating	[C]	58.500
Fatigue load rating	[C _U]	6.200
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000

Basic static load rating (kN) [C_{0theo}] 0.000

Results

Shaft

Maximum deflection (μm) 106.523
Position of the maximum (mm) 165.500
Mass center of gravity (mm) 82.750
Total axial load (N) -2797.236
Torsion under torque (°) -0.636

Bearing

Probability of failure [n] 10.00 %
Axial clearance [u_A] 10.00 μm
Rolling bearings, classical calculation (contact angle considered)

Shaft 'Shaft1' Rolling bearing 'RollerBearing1'

Position (Y-coordinate) [y] 10.00 mm
Equivalent load [P] 5.02 kN
Equivalent load [P₀] 2.87 kN
Life modification factor for reliability[a₁] 1.000
Nominal bearing service life [L_{nh}] 11178.28 h
Static safety factor [S₀] 5.57
Bearing reaction force [F_x] 0.448 kN
Bearing reaction force [F_y] 2.797 kN
Bearing reaction force [F_z] 2.419 kN
Bearing reaction force [F_r] 2.460 kN (79.5°)
Oil level [H] 0.000 mm
Load-independent moment of friction [M₀] 0.019 Nm
Load-dependent moment of friction [M₁] 0.175 Nm
Moment of friction, cylindrical roller bearing[M₂] 0.000 Nm
Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994
Torque of friction [M_{loss}] 0.193 Nm
Power loss [P_{loss}] 6.195 W

The moment of friction is calculated according to the details in SKF Catalog 1994.

Displacement of bearing [u_x] -1.139 μm
Displacement of bearing [u_y] -10.000 μm
Displacement of bearing [u_z] -6.145 μm
Displacement of bearing [u_r] 6.250 μm (-100.5°)
Misalignment of bearing [r_x] -0.352 mrad (-1.21')
Misalignment of bearing [r_y] -0.591 mrad (-2.03')
Misalignment of bearing [r_z] 0.134 mrad (0.46')
Misalignment of bearing [r_r] 0.377 mrad (1.29')

Shaft 'Shaft1' Rolling bearing 'RollerBearing2'

Position (Y-coordinate) [y] 120.00 mm
Equivalent load [P] 10.35 kN

Equivalent load	[P ₀]	10.35	kN
Life modification factor for reliability[a ₁]		1.000	
Nominal bearing service life	[L _{nh}]	17503.86	h
Static safety factor	[S ₀]	4.64	
Bearing reaction force	[F _x]	2.977	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	-9.915	kN
Bearing reaction force	[F _r]	10.353	kN (-73.29°)
Oil level	[H]	0.000	mm
Load-independent moment of friction	[M ₀]	0.016	Nm
Load-dependent moment of friction	[M ₁]	0.185	Nm
Moment of friction, cylindrical roller bearing[M ₂]		0.000	Nm
Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994			
Torque of friction	[M _{loss}]	0.201	Nm
Power loss	[P _{loss}]	6.438	W
The moment of friction is calculated according to the details in SKF Catalog 1994.			
Displacement of bearing	[u _x]	-4.673	μm
Displacement of bearing	[u _y]	-12.113	μm
Displacement of bearing	[u _z]	15.564	μm
Displacement of bearing	[u _r]	16.250	μm (106.71°)
Misalignment of bearing	[r _x]	1.435	mrاد (4.93')
Misalignment of bearing	[r _y]	-8.587	mrاد (-29.52')
Misalignment of bearing	[r _z]	-0.197	mrاد (-0.68')
Misalignment of bearing	[r _r]	1.449	mrاد (4.98')

Damage (%) [H] (20000.000)

No. B1 B2
1178.92 114.26

Σ178.92 114.26

Utilization (%) [H] (20000.000)

B1 B2
121.40104.08

B1: RollerBearing1
B2: RollerBearing2

_O.GroupBox.Shaft2.Shaft2_calc

KISSsoft Release 03/2016 C

KISSsoft University license - Universidade do Porto

File

Name : Unnamed

Changed by: em12072

on: 11.07.2017

at: 22:32:48

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	Shaft2
Drawing	
Initial position (mm)	0.000
Length (mm)	173.000
Speed (1/min)	425.74
Sense of rotation: counter clockwise	

Material	C45 (1)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	0.958
Weight of shaft, including additional masses (kg)	0.958
Mass moment of inertia (kg*mm ²)	107.719
Momentum of mass GD2 (Nm ²)	0.004

The direction of the weight is not considered

Consider deformations due to shearing

Shear correction coefficient	1.100
------------------------------	-------

Contact angle of rolling bearings is considered

Tolerance field: Mean value

Reference temperature (°C)	20.000
----------------------------	--------

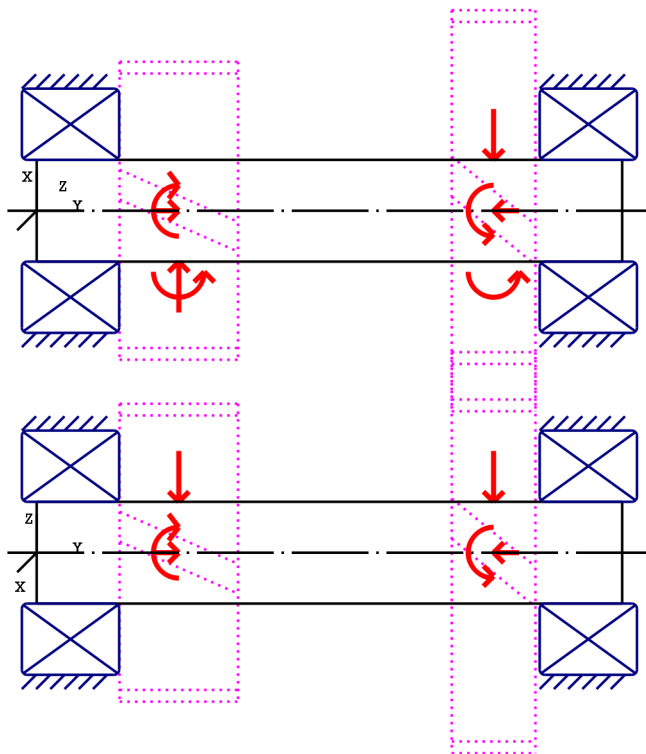


Figure: Load applications

Shaft definition (Shaft2)

Outer contour

Cylinder (Cylinder)		0.000mm ... 173.000mm
Diameter (mm)	[d]	30.0000
Length (mm)	[l]	173.0000
Surface roughness (µm)	[Rz]	8.0000

Forces

Type of force element	Cylindrical gear	
Label in the model	z2(GearPair_const1)	
Position on shaft (mm)	[y _{local}]	42.0000
Position in global system (mm)	[y _{global}]	42.0000
Operating pitch diameter (mm)		87.8278
Helix angle (°)		20.4632 right
Working pressure angle at normal section (°)		23.1758
Position of contact (°)		180.0000
Length of load application (mm)		35.0000
Power (kW)		14.6763 driven (Input)
Torque (Nm)		-329.1875
Axial force (N)		2797.2362
Shearing force X (N)		3425.2695
Shearing force Z (N)		-7496.2063
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		-122.8375

Type of force element		Cylindrical gear
Label in the model		z3(GearPair_const2)
Position on shaft (mm)	[ylocal]	135.0000
Position in global system (mm)	[yglobal]	135.0000
Operating pitch diameter (mm)		118.2202
Helix angle (°)		19.8503 right
Working pressure angle at normal section (°)		18.8287
Position of contact (°)		0.0000
Length of load application (mm)		24.7320
Power (kW)		14.6763 driving (Output)
Torque (Nm)		329.1875
Axial force (N)		-2010.5135
Shearing force X (N)		-2018.9292
Shearing force Z (N)		-5569.0561
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		-118.8417

Bearing

Label in the model		RollerBearing3
Bearing type		SKF 32306 J2/Q
Bearing type		Taper roller bearing (single row)
Bearing position (mm)	[ylocal]	10.000
Bearing position (mm)	[yglobal]	10.000
Attachment of external ring		Set fixed bearing left
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	72.000
Width (mm)	[b]	28.750
Corner radius (mm)	[r]	1.500
The bearing pressure angle will be considered in the calculation		
Position (center of pressure)		(mm)
13.6250		
Basic static load rating	[C ₀]	85.000
Basic dynamic load rating	[C]	76.500
Fatigue load rating	[C _U]	9.700
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Label in the model		RollerBearing4
Bearing type		SKF 32306 J2/Q
Bearing type		Taper roller bearing (single row)
Bearing position (mm)	[ylocal]	163.000
Bearing position (mm)	[yglobal]	163.000
Attachment of external ring		Set fixed bearing right
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	72.000
Width (mm)	[b]	28.750
Corner radius (mm)	[r]	1.500
The bearing pressure angle will be considered in the calculation		
Position (center of pressure)		(mm)
159.3750		
Basic static load rating	[C ₀]	85.000
Basic dynamic load rating	[C]	76.500
Fatigue load rating	[C _U]	9.700
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000

Basic static load rating (kN) [C_{0theo}] 0.000

Results

Shaft

Maximum deflection (μm) 61.779
Position of the maximum (mm) 80.545
Mass center of gravity (mm) 86.500
Total axial load (N) 786.723
Torsion under torque (°) 0.278

Bearing

Probability of failure [n] 10.00 %
Axial clearance [u_A] 10.00 μm
Rolling bearings, classical calculation (contact angle considered)

Shaft 'Shaft2' Rolling bearing 'RollerBearing3'

Position (Y-coordinate) [y] 10.00 mm
Equivalent load [P] 7.01 kN
Equivalent load [P₀] 7.01 kN
Life modification factor for reliability[a₁] 1.000
Nominal bearing service life [L_{nh}] 112865.09 h
Static safety factor [S₀] 12.13
Bearing reaction force [F_x] -0.763 kN
Bearing reaction force [F_y] 1.845 kN
Bearing reaction force [F_z] 6.968 kN
Bearing reaction force [F_r] 7.010 kN (96.25°)
Bearing reaction moment [M_x] 25.26 Nm
Bearing reaction moment [M_y] 0.00 Nm
Bearing reaction moment [M_z] 2.76 Nm
Bearing reaction moment [M_r] 25.41 Nm (6.25°)
Oil level [H] 0.000 mm
Load-independent moment of friction [M₀] 0.045 Nm
Load-dependent moment of friction [M₁] 0.143 Nm
Moment of friction, cylindrical roller bearing[M₂] 0.000 Nm
Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994
Torque of friction [M_{loss}] 0.188 Nm
Power loss [P_{loss}] 8.389 W
The moment of friction is calculated according to the details in SKF Catalog 1994.
Displacement of bearing [u_x] 0.000 μm
Displacement of bearing [u_y] 13.876 μm
Displacement of bearing [u_z] 0.000 μm
Displacement of bearing [u_r] 0.000 μm
Misalignment of bearing [r_x] -1.257 mrad (-4.32')
Misalignment of bearing [r_y] -0.000 mrad (0')
Misalignment of bearing [r_z] -0.312 mrad (-1.07')
Misalignment of bearing [r_r] 1.295 mrad (4.45')

Shaft 'Shaft2' Rolling bearing 'RollerBearing4'

Position (Y-coordinate)	[Y]	163.00	mm
Equivalent load	[P]	7.45	kN
Equivalent load	[P ₀]	6.13	kN
Life modification factor for reliability[a ₁]		1.000	
Nominal bearing service life	[L _{nh}]	92042.79	h
Static safety factor	[S ₀]	13.86	
Bearing reaction force	[F _x]	-0.644	kN
Bearing reaction force	[F _y]	-2.631	kN
Bearing reaction force	[F _z]	6.097	kN
Bearing reaction force	[F _r]	6.131	kN (96.03°)
Bearing reaction moment	[M _x]	-22.10	Nm
Bearing reaction moment	[M _y]	0.00	Nm
Bearing reaction moment	[M _z]	-2.33	Nm
Bearing reaction moment	[M _r]	22.22	Nm (-173.97°)
Oil level	[H]	0.000	mm
Load-independent moment of friction	[M ₀]	0.045	Nm
Load-dependent moment of friction	[M ₁]	0.204	Nm
Moment of friction, cylindrical roller bearing[M ₂]		0.000	Nm
Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994			
Torque of friction	[M _{loss}]	0.249	Nm
Power loss	[P _{loss}]	11.108	W
The moment of friction is calculated according to the details in SKF Catalog 1994.			
Displacement of bearing	[u _x]	-0.000	µm
Displacement of bearing	[u _y]	10.000	µm
Displacement of bearing	[u _z]	-0.000	µm
Displacement of bearing	[u _r]	0.000	µm
Misalignment of bearing	[r _x]	1.192	mrاد (4.1')
Misalignment of bearing	[r _y]	4.859	mrاد (16.7')
Misalignment of bearing	[r _z]	-0.034	mrاد (-0.12')
Misalignment of bearing	[r _r]	1.193	mrاد (4.1')

Damage (%) [H] (20000.000)

No.	B1	B2
1	17.72	21.73

Σ 17.72 21.73

Utilization (%) [H] (20000.000)

B1	B2
59.50	63.26

B1: RollerBearing3

B2: RollerBearing4

_O.GroupBox.Shaft3.Shaft3_calc

KISSsoft Release 03/2016 C

KISSsoft University license - Universidade do Porto

File

Name : Unnamed

Changed by: em12072

on: 11.07.2017

at: 22:33:01

Important hint: At least one warning has occurred during the calculation:

1-> The required service life of bearing 'Shaft 'Shaft3', Rolling bearing 'RollerBearing6' is not achieved!

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	Shaft3
Drawing	
Initial position (mm)	0.000
Length (mm)	184.000
Speed (1/min)	701.22
Sense of rotation: clockwise	
Material	C45 (1)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	1.386
Weight of shaft, including additional masses (kg)	1.386
Mass moment of inertia (kg*mm ²)	212.252
Momentum of mass GD2 (Nm ²)	0.008

The direction of the weight is not considered

Consider deformations due to shearing

Shear correction coefficient 1.100

Contact angle of rolling bearings is considered

Tolerance field: Mean value

Reference temperature (°C) 20.000

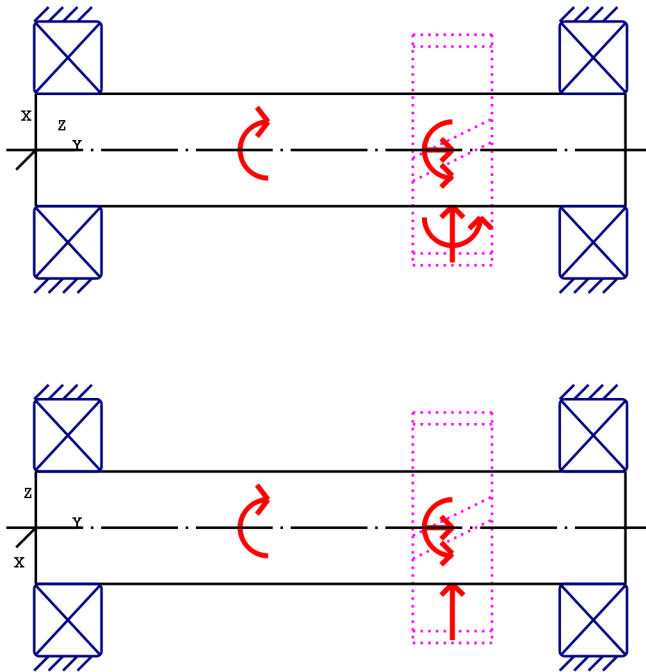


Figure: Load applications

Shaft definition (Shaft3)

Outer contour

Cylinder (Cylinder)		0.000mm ... 184.000mm
Diameter (mm)	[d]	35.0000
Length (mm)	[l]	184.0000
Surface roughness (µm)	[Rz]	8.0000

Forces

Type of force element	Coupling	
Label in the model	Coupling2(Boundary2)	
Position on shaft (mm)	[y _{local}]	72.5000
Position in global system (mm)	[y _{global}]	72.5000
Effective diameter (mm)		35.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		15.0000
Power (kW)		14.6763 driving (Output)
Torque (Nm)		-199.8638
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000

Mass moment of inertia Jp (kg*m ²)	0.0000
Mass moment of inertia Jxx (kg*m ²)	0.0000
Mass moment of inertia Jzz (kg*m ²)	0.0000
Eccentricity (mm)	0.0000

Type of force element	Cylindrical gear	
Label in the model	z4(GearPair_const2)	
Position on shaft (mm)	[ylocal]	130.0000
Position in global system (mm)	[yglobal]	130.0000
Operating pitch diameter (mm)		71.7766
Helix angle (°)		19.8503 left
Working pressure angle at normal section (°)		18.8287
Position of contact (°)		180.0000
Length of load application (mm)		24.7320
Power (kW)		14.6763 driven (Input)
Torque (Nm)		199.8638
Axial force (N)		2010.5135
Shearing force X (N)		2018.9292
Shearing force Z (N)		5569.0561
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		-72.1539

Bearing

Label in the model	RollerBearing5	
Bearing type	SKF *6307	
Bearing type	Deep groove ball bearing (single row)	
Bearing position (mm)	[ylocal]	10.000
Bearing position (mm)	[yglobal]	10.000
Attachment of external ring	Set fixed bearing left	
Inner diameter (mm)	[d]	35.000
External diameter (mm)	[D]	80.000
Width (mm)	[b]	21.000
Corner radius (mm)	[r]	1.500
Basic static load rating	[C ₀]	19.000
Basic dynamic load rating	[C]	35.100
Fatigue load rating	[C _U]	0.815
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Label in the model	RollerBearing6	
Bearing type	SKF *6307	
Bearing type	Deep groove ball bearing (single row)	
Bearing position (mm)	[ylocal]	174.000
Bearing position (mm)	[yglobal]	174.000
Attachment of external ring	Set fixed bearing right	
Inner diameter (mm)	[d]	35.000
External diameter (mm)	[D]	80.000
Width (mm)	[b]	21.000
Corner radius (mm)	[r]	1.500
Basic static load rating	[C ₀]	19.000
Basic dynamic load rating	[C]	35.100
Fatigue load rating	[C _U]	0.815
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000

Basic static load rating (kN) [C_{0theo}] 0.000

Results

Shaft

Maximum deflection (μm) 33.240
Position of the maximum (mm) 102.580
Mass center of gravity (mm) 92.000
Total axial load (N) 2010.513
Torsion under torque (°) 0.056

Bearing

Probability of failure [n] 10.00 %
Axial clearance [u_A] 10.00 μm
Rolling bearings, classical calculation (contact angle considered)

Shaft 'Shaft3' Rolling bearing 'RollerBearing5'

Position (Y-coordinate) [y] 10.00 mm
Equivalent load [P] 1.50 kN
Equivalent load [P₀] 1.50 kN
Life modification factor for reliability[a₁] 1.000
Nominal bearing service life [L_{nh}] 306009.52 h
Static safety factor [S₀] 12.69
Bearing reaction force [F_x] -0.102 kN
Bearing reaction force [F_y] 0.000 kN
Bearing reaction force [F_z] -1.494 kN
Bearing reaction force [F_r] 1.498 kN (-93.89°)
Oil level [H] 0.000 mm
Load-independent moment of friction [M₀] 0.046 Nm
Load-dependent moment of friction [M₁] 0.022 Nm
Moment of friction, cylindrical roller bearing[M₂] 0.000 Nm
Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994
Torque of friction [M_{loss}] 0.068 Nm
Power loss [P_{loss}] 4.986 W
The moment of friction is calculated according to the details in SKF Catalog 1994.
Displacement of bearing [u_x] 0.441 μm
Displacement of bearing [u_y] 10.446 μm
Displacement of bearing [u_z] 6.485 μm
Displacement of bearing [u_r] 6.500 μm (86.11°)
Misalignment of bearing [r_x] 0.403 mrad (1.39')
Misalignment of bearing [r_y] 0.000 mrad (0')
Misalignment of bearing [r_z] -0.067 mrad (-0.23')
Misalignment of bearing [r_r] 0.408 mrad (1.4')

Shaft 'Shaft3' Rolling bearing 'RollerBearing6'

Position (Y-coordinate) [y] 174.00 mm
Equivalent load [P] 5.46 kN

Equivalent load	[P ₀]	4.50	kN
Life modification factor for reliability[a ₁]		1.000	
Nominal bearing service life	[L _{nh}]	6316.43	h
Static safety factor	[S ₀]	4.22	
Bearing reaction force	[F _x]	-1.917	kN
Bearing reaction force	[F _y]	-2.011	kN
Bearing reaction force	[F _z]	-4.075	kN
Bearing reaction force	[F _r]	4.503	kN (-115.2°)
Oil level	[H]	0.000	mm
Load-independent moment of friction	[M ₀]	0.046	Nm
Load-dependent moment of friction	[M ₁]	0.156	Nm
Moment of friction, cylindrical roller bearing[M ₂]		0.000	Nm
Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994			
Torque of friction	[M _{loss}]	0.202	Nm
Power loss	[P _{loss}]	14.830	W
The moment of friction is calculated according to the details in SKF Catalog 1994.			
Displacement of bearing	[u _x]	2.767	µm
Displacement of bearing	[u _y]	10.000	µm
Displacement of bearing	[u _z]	5.882	µm
Displacement of bearing	[u _r]	6.500	µm (64.8°)
Misalignment of bearing	[r _x]	-0.555	mrad (-1.91')
Misalignment of bearing	[r _y]	0.985	mrad (3.38')
Misalignment of bearing	[r _z]	0.100	mrad (0.34')
Misalignment of bearing	[r _r]	0.563	mrad (1.94')

Damage (%) [H] (20000.000)

No. B1 B2

1 6.54 316.63

Σ 6.54 316.63

Utilization (%) [H] (20000.000)

B1 B2

40.28 146.84

B1: RollerBearing5

B2: RollerBearing6

MatLab printouts

Note: In the following matrices each line corresponds to a speed and each column to a torque, from the lower to the higher. The third dimension of the matrix correspond to the lubricant, being the first (:,:,1) the values for the 75w140-A and the second (:,:,2) to the 75w90.

```
>> P_VZP
```

```
P_VZP(:, :, 1) =
```

113.7540	162.5065	216.5802
130.4093	186.5589	247.9324
146.8730	209.2186	287.6673

```
P_VZP(:, :, 2) =
```

125.7111	180.0391	240.4046
143.9827	206.2400	273.2538
161.2684	230.2546	306.1841

```
>> P_VZ0
```

```
P_VZ0(:, :, 1) =
```

135.8875	126.7521	117.0501
167.7755	155.2666	144.6308
199.7744	183.8328	185.2493

```
P_VZ0(:, :, 2) =
```

114.1355	105.2227	99.0200
138.1284	131.3606	129.3596
170.7711	165.4126	164.2067

```
>> P_VL
```

```
P_VL(:, :, 1) =
```

66.4311	80.6490	101.7537
77.2137	94.5137	119.4874
87.4236	108.1242	142.8433

```
P_VL(:, :, 2) =
```

57.3997	74.2631	93.4884
67.4452	87.4609	108.5103
76.8281	99.1092	124.0857

```
>> P_VS
```

```
P_VS(:, :, 1) =
```

15.1487	15.1458	15.1410
17.7217	17.7224	17.7239
20.3349	20.2383	21.0528

P_VS(:, :, 2) =

15.1168	15.1148	15.1014
17.6549	17.6516	17.6508
20.1937	20.1897	20.1860

>> P_T

P_T(:, :, 1) =

331.2212	385.0534	450.5250
393.1202	454.0616	529.7745
454.4059	521.4140	636.8127

P_T(:, :, 2) =

312.3631	374.6396	448.0144
367.2111	442.7130	528.7745
429.0613	514.9660	614.6625

>>