

Faculdade de Engenharia da Universidade do Porto

Departamento de Engenharia Mecânica



**FRICTION TORQUE IN THRUST BALL AND ROLLER BEARINGS
LUBRICATED WITH “WIND TURBINE GEAR OILS”
AT CONSTANT TEMPERATURE**

Pedro Miguel Pinto Amaro

Master's Degree Dissertation presented to the
Faculdade de Engenharia da Universidade do Porto

Dissertation supervised by:

Doutor Jorge Humberto O. Seabra, Full Professor of FEUP

Doutor Ramiro Carneiro Martins, Auxiliary Researcher of INEGI

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Abstract

The efficiency of mechanical transmissions has always been an important point of study. The sources of energy cannot keep up with the needs of society, so the reduction of energy consumption along with increased effectiveness of its uses is becoming more and more important. Having in mind the optimization of natural resources, the use of biodegradable products has grown significantly in recent times.

With the goal of reaching an improved environmental compatibility and lower power losses, testing and validation of the lubricants is required.

The main purpose of this study was to measure the friction torque of thrust ball and roller bearings lubricated with wind turbine gear oils. The measurements among the oils will be compared and conclusions will be taken.

In this work six wind turbine gear oils were considered: 2 esters based oils (ESTF and ESTR), 2 mineral based oils (MINE and MINR), a Polyalkyleneglycol based oil (PAGD) and a Polialphaolefin based oil (PAOR). For these oils several tests were performed and their tribological behaviour was evaluated and compared.

The physical properties of the oils were obtained: density, viscosity and how they reacted to pressure and temperature. Experiments and tests were performed with thrust ball bearings (TBB) and thrust roller bearings (RTB), at constant temperature (80°C), using all the selected oils.

For each friction torque test the rolling bearings (TBB and RTB) were assembled in a machine suitable for testing, an axial load (700N or 7000N) was applied and began operating at constant temperature. The friction torque measurements are then made for rotating speed values between the 75-1200 rpm range. Using the friction model, the measured friction torque is divided in its components (rolling, sliding and drag) and from those the friction coefficient can be achieved.

The results of the friction torque measurements for each type of rolling bearing (TBB and RTB), lubricated with different oils and different operating conditions indicated that:

For the case of the tests performed with a high axial load (7000N), the total friction torque for every oil increases with speed in the TBB and decreases with speed in the RTB. Considering the oil performances the majority of the oils had very close results for both type of bearings but for TBB the MINE oil clearly demonstrated the best results at all speeds, and for RTB the PAGD oil distinctively showed the best results for lower speeds.

In the tests with a lower axial load (700N), the total friction torque in both the TBB and the RTB increase with speed although their values are significantly smaller when in comparison to the tests with higher axial load. Like in the case of a high load most of the oils have close values but for TBB the PAOR oil showed the best results for almost all speeds and for RTB the PAGD oil showed the worst results for higher speeds.

Resumo

A eficiência das transmissões mecânicas sempre foi um importante ponto de estudo. As fontes de energia não conseguem acompanhar com as necessidades da sociedade, por isso a redução do consumo energético assim como uma maior eficácia do seu uso está a tornar-se cada vez mais importante. Tendo em mente a otimização dos recursos naturais, a utilização de produtos biodegradáveis cresceu bastante nos tempos recentes.

Com a finalidade de alcançar uma melhor compatibilidade ambiental e diminuir as perdas de potência, é necessário testar e validar os lubrificantes.

O principal objectivo deste estudo era medir o momento de atrito de rolamentos axiais de esferas e rolos lubrificados por óleos de engrenagens de turbinas de vento. As medições feitas entre os óleos serão comparadas e tirar-se-ão conclusões.

Neste trabalho foram avaliados seis óleos dos quais: 2 têm uma base mineral (MINE e MINR), 2 têm uma base de ester (ESTF e ESTR), um tem uma base de Polialquilenoglicol (PAGD) e outro tem uma base de Polialfaolefina (PAOR). Para estes óleos foram realizados vários ensaios e o seu comportamento tribológico foi avaliado e comparado.

As propriedades físicas dos óleos foram medidas para determinar a densidade, viscosidade e a sua reacção à pressão e temperatura. Testes foram realizados com rolamentos axiais de esferas (TBB) e rolamentos axiais de rolos (RTB), com temperatura de ensaio fixa (80°C) para todos os óleos seleccionados.

Para cada teste de medição de atrito o rolamento usado foi montado numa máquina própria para o teste, uma carga axial (700N ou 7000N) foi aplicada e colocou-se a funcionar a temperatura constante. As medições de atrito são feitas para velocidades de rotação entre os 75 e os 1200 rpm. Usando o modelo de atrito, o momento de atrito medido é dividido nos seus componentes (rolamento, deslizamento e arrasto) e com eles obtém-se o coeficiente de atrito.

As medições do momento de atrito total para cada tipo de rolamento (TBB e RTB) lubrificados com diferentes óleos e com diferentes condições de funcionamento indicam:

Para o caso de testes realizados com elevada carga axial (7000N), o momento de atrito total aumenta com a velocidade para TBB e diminui com a velocidade para RTB. A maioria dos óleos tem resultados muito próximos para o momento de atrito para ambos os rolamentos apesar de alguns se distinguirem, no de esferas o MINE revelou os melhores resultados para todas as velocidades e no de rolos o PAGD teve os melhores resultados para baixas velocidades.

Nos testes com baixa carga axial (700N), o momento de atrito total aumenta com a velocidade nos dois tipos de rolamentos apesar de os seus valores serem significativamente mais baixos em comparação com os testes realizados com carga elevada. Tal como nos testes de alta carga a maioria dos óleos tem valores de momento de atrito próximos mas para o de esferas o PAOR mostrou os melhores resultados para quase todas as velocidades e para o de rolos o PAGD mostrou os pior resultados para velocidades mais elevadas.

Keywords

Thrust ball bearings

Thrust roller bearings

Friction torque

Friction coefficient

Film thickness

Palavras chave

Rolamentos axiais de esferas

Rolamentos axiais de rolos

Momento de atrito

Coefficiente de atrito

Espessura de filme

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Nomenclature

Symbol	Units	Designation
a	[m]	Hertz minor half-width
b	[m]	Hertz major half-width
d	[mm]	Inside diameter
D	[mm]	Outside diameter
dm	[mm]	Bearing mean diameter
E^*	[Pa]	Equivalent Young modulus
F_n	[N]	Axial load
G	[-]	Dimensionless of the material parameter in contact EHD isothermic, smooth and Newtonian
g	[m/s ²]	Gravity acceleration
G_{rr}	[-]	Rolling friction variable
G_{sl}	[-]	Sliding friction variable
H_0	[-]	Lubricant film thickness at the centre of contact
h_0	[m]	Lubricant film thickness at the centre of contact
h_{0c}	[m]	Corrected lubricant film thickness
i_{rw}	[-]	Number of rows in the ball bearing
K_{ball}	[-]	Ball bearing related constant
K_{roller}	[-]	Roller bearing related constant
K_{rs}	[-]	Replenishment/starvation constant
K_z	[-]	Geometry constant
M_{drag}	[N.mm]	Friction torque of drag losses
M_{exp}	[N.mm]	Total friction torque (experimental)
M_{seal}	[N.mm]	Friction torque of seals
M_{sl}	[N.mm]	Sliding friction torque
M_t	[N.mm]	Total friction torque
M_{rr}	[N.mm]	Rolling friction torque
n	[rpm]	Rotational speed
p_0	[Pa]	Maximum contact pressure (Hertz)
R_1	[-]	Geometry constant for rolling friction torque
R_{eq}	[mm]	Equivalent curvature radius
R_x	[m]	Radius of curvature in the rolling direction
R_y	[m]	Radius of curvature perpendicular to the rolling direction
s	[-]	Piezoviscosity parameter
S_1	[-]	Geometry constant for sliding friction torque
t	[-]	Piezoviscosity parameter
U	[-]	Dimensionless speed parameter in contact EHD isothermic, smooth and Newtonian
VI	[-]	Viscosity index
V_M	[-]	Variable as a function of the oil level
W	[-]	Dimensionless load parameter in contact EHD isothermic, smooth and Newtonian

Symbol	Units	Designation
α	[Pa ⁻¹]	Coefficient of piezoviscosity
α_t	[/]	Coefficient of thermal expansion
Λ	[/]	Specific film thickness
η	[mPa.s]	Lubricant dynamic viscosity
μ_{bl}	[/]	Friction coefficient in boundary lubrication
μ_{EHD}	[/]	Friction coefficient in full film conditions
μ_{sl}	[/]	Sliding friction coefficient
ν	[cSt]	Lubricant kinematic viscosity
ρ	[kg/m ³]	Density
σ	[m]	Composed roughness
τ	[N/mm ²]	Shear stress of lubricants
ϕ_{bl}	[/]	Weighting factor of the sliding friction coefficient
ϕ_{ish}	[/]	Inlet shear heating factor
ϕ_{rs}	[/]	Kinematic replenishment/starvation reduction factor

1. Introduction

The ever present economic concerns impose the necessity to evaluate and improve the efficiency of lubricated mechanisms. It is important to know how lubricants act under the operating conditions of the mechanism in order to predict its effectiveness. So, it is necessary to develop methods and procedures to evaluate and compare the behaviour and performance of different lubricants.

This dissertation focus on the analysis of the influence of wind turbine gear oils formulation on thrust ball and roller bearing performance.

To reach this objective the following tasks were followed:

- a) The lubricants physical properties were tested to see if they matched with the manufacturer's information.
- b) For the tests on the rolling bearings a modified Four-Ball Machine was used ("Four-Ball Machine", Cameron-Plint, ref^a E82/7752).

A bearing house previously developed was used. It incorporated a torque transducer, a heater and several thermocouples. This assembly is mounted on the Four-ball Machine and allows the simultaneous measurement of the friction torque and the operating temperature at the desired combination of speed and load.

- c) The SKF friction torque model [2] was applied for the rolling bearings to make the most of the performed experiments.

This study is the result of a great amount of experimental work with a purpose to evaluate the friction torque of rolling bearings lubricated by the six selected lubricants. It demonstrates the experimental tests made, the test equipment used, the test procedures and the analysis of results.

1.1. Aim and thesis outline

This study is the result of experimental and numerical work accomplished for the course of Mechanical Engineering Master's Degree under the Project and Mechanical Construction branch. The work was performed at CETRIB (Tribology and Industrial Maintenance Unit of INEGI). The main aim of this work was to analyse the influence of "Wind turbine gear oils" formulation on thrust ball and roller bearing performance.

2. Lubrication and Lubricants

2.1. Introduction

Lubricants are mainly used to reduce friction and wear between two contacting surfaces with relative motion. [6] Following this definition, any substance (solid, liquid or gas) interposed between two surfaces with the objective of favour their relative slip, is a potential lubricant. Despite that, other features are generally necessary from the lubricants, such as a good separation of the surfaces and a good evacuation of the heat generated during motion. Some of these properties are inherent, such as low shear strength, while others are related to surface contact, like protection against corrosion even in stationary periods. [6]

The priorities may differ for different cases, which restricts the number of efficient lubricants to a restricted number of base materials: mineral, vegetable, animal or synthetic. Another factor that has been becoming more prominent in recent years, that further limits the choice of a lubricant, is the environmental impact, which is influencing the creation of new environmental friendly lubricants.

Moreover, in elastohydrodynamics contacts, the lubricant flows through the contact for a very short period of time of about 1 ms , having a shock pressure of about 1 GPa or higher, being submitted to shear rates that can reach 10^{-7} s^{-1} and temperature rises above $100\text{ }^{\circ}\text{C}$. These conditions, characterized by high and fast variations in pressure and temperature inside the contact, justify the change of the lubricant properties inside the contact that are observed experimentally and theoretically determined. [6] Such extreme conditions make the work of characterizing the properties and the behaviour of the lubricant within the contact even harder.

2.2. Lubricating oils

The lubricating oils can be classified based on their origin. [6]

Vegetable and animal oils: These types of lubricants were the first lubricants used. They possess several advantages over mineral oils, ie, high viscosity, high lubrication and fast biodegradability, the last one being, perhaps, the most important. The drawback of these lubricants is that they oxidize quickly, because of their low resistance to elevated temperatures. Due to the fact that lubricant requirements kept increasing, the uses of animal and vegetable oils have mostly been replaced by other types of lubricants.

Mineral oils: obtained from the distillation of crude petroleum, these oils can be distinguished by their composition and may be divided in three categories (paraffinic, naphthenic and aromatic). The aromatic types are usually undesirable so they are the least used in lubrication, while the other two types are very often used because of their low cost and reasonable performance.

Synthetic oils: These types of lubricants are created by synthesis of light hydrocarbons with the inclusion of some non-petroleum organic elements. These lubricants have some good points; some being increased oil longevity and better heat resistance despite their higher cost. They may be divided in four categories: synthetic hydrocarbons, polyglycols, organic esters and phosphate esters.

Often additives are added to the oils, giving them new properties or improving the ones the base oil already possesses.

2.3. Greases

Grease refers to the dispersion of a thickening agent in lubricating oil belonging to any of the types mentioned before. [6]

There are two types of thickening agents:

Soaps – aluminium, barium, calcium, lithium, sodium and strontium.

No soaps – inorganic compounds, organic clays, polyuria.

2.4. Solid Lubricants

A solid lubricant is a film of solid material composed of organic or inorganic compounds that is placed between two surfaces to act as a lubricant. [6]

Inorganic solid lubricants – laminar solids, miscellaneous soft solids and chemical conversion coatings.

Organic solid lubricants – soaps, waxes, fats and polymer films.

2.5. Gaseous Lubricants

The lubrication with the use of gas is similar in many aspects to liquid lubrication. Despite the fact that both are viscous fluids, there are two great different physical properties: the viscosity of gases is much smaller and their compressibility is much higher when in comparison to liquids. Thus, the load capacity and film thickness in the contact are much lower when using a gaseous lubricant. [6]

Some gases used for lubrication are: air, steam, industrial gases, among others.

2.6. Functions of Lubricants

O'Connor [18] gives a very interesting summary of the main functions of a lubricant.

The selection of lubricants is made through the functions they are required to execute. The most important parameter varies with the application. It can be the control of friction, control of temperature, control of corrosion, among others.

The main functions of lubricants are: [18]

- Control friction,
- Control wear,
- Control temperature,
- Control corrosion,
- Insulate (electric),
- Dampen shock (gears),
- Remove contaminants (flushing),
- Form a seal (grease).

It should be mentioned that most of the lubricants functions are interrelated so when discussing those functions separately does not mean that they can be isolated during usage.

Control of Friction: a lubricant may operate in any of the lubrication regimes (boundary, mixed or full film regimes) and its job of controlling friction varies with each one. [18]

In full film lubrication, friction is mainly influenced by the viscosity of the fluid.

In mixed film regimes the lubricant may separate the solids for some time but direct contact between the surfaces (metal-to-metal) will also take place, which will influence the coefficient of friction. So besides the viscous properties of the lubricants its chemical properties will also be important to provide a low friction layer between the surfaces.

In this regime the coefficient of friction is expected to increase when in comparison to the full film lubrication.

In boundary lubrication the effects of lubricants become less dependent on the bulk properties and more on the interface effects or effects of surface contamination. In this regime the coefficient of friction is very high. Here the effects of the additives become dominant.

Control of Wear: wear occurs in lubricated systems by several mechanisms (abrasion, corrosion, among others). [18] The lubricant plays an important role in battling each of them.

The flushing action of the lubricant serves to remove the harmful solid particles from the location of lubricated surfaces (thus preventing abrasion).

Proper refinement and the use of oxidation inhibitors reduces lubricant deterioration (keeping the level of corrosive products low) which helps protect the metal surfaces from the acidic oxidation products.

Wear due to metal-to-metal contact results from a breakdown of the lubricant film, meaning, anything which causes the lubricated surfaces to approach each other until their asperities contact will cause wear. A good supply of lubricant is the best to prevent this condition. In boundary lubrication the chemical nature of the lubricant will affect the amount of metal-to-metal contact and the wear that occurs.

Control of Temperature: few properties of the lubricants influence its ability to control temperature, on a different note, proper application of the lubricant is more important in temperature control. [18]

The temperature of a lubricated system is proportional to the work done to move the parts with relative motion and to the ambient temperature.

The result of supplying energy to overcome friction is heat. All the heat generated during operation must be removed to achieve an equilibrium operating temperature, or overheating takes place. The thermal conductivity of the lubricant is important to help dissipate the heat.

A lubricant controls temperature by minimizing friction and evacuating the generated heat. The effectiveness depends on the amount of lubricant supplied, the ambient temperature and the existing support for external cooling.

Control of Corrosion: A lubricant controls corrosion in two ways. [18] When the machine is stopped, it acts as a preserver. When the machine is active, it coats lubricated parts with a protective film. The level of protection required depends on the environment in which the machine operates.

The ability of a lubricant to control corrosion is related to the thickness of the lubricant film remaining on the metal surfaces and the chemical composition of the lubricant.

Insulate (electric): In certain applications a lubricant may be used to take action as an electrical insulator, particularly around electrical equipment such as transformers and switchgear. [18]

Some characteristics of insulating oils are high electrical resistivity and dielectric strength, low viscosity, high flash point, chemical stability under localized high temperatures. These requirements may be inconsistent with those needed for the best lubrication, so special products are regularly used when insulation is required.

Dampen Shock: the lubricants function as shock-dampening in two ways. [18] The most common is the transfer of mechanical energy to fluid energy as shock absorbers. A fluid in contact with a machine in movement will dissipate its mechanical energy (vibration/oscillation) through fluid friction. For an effective performance, the fluid must have

a specific viscosity which should not vary much with temperature. High-viscosity index oils are normally used.

The second mechanism which plays a part in the shock-dampening function of lubricants is the variation of viscosity with pressure. Many devices work with loads that produce very high pressures. The increase of viscosity of lubricants in loaded areas is part of their good performance under shock load conditions.

Remove Contaminants: lubricants are used to remove contaminants in many systems. [18] The flushing action of lubricants in removing solid contaminants from between working surfaces is a serious matter in industry. This prevents wear and indenting of surfaces due to trapped solids.

Form a Seal: a special function that can be performed by lubricating grease is the formation of a seal. [18] Because greases are usually employed where lubricant retention is a problem, the self-sealing function is important. This helps retaining the lubricant in the bearing and the contaminants out.

2.7. Physical properties of lubricating oils

The choice of a lubricant to undertake a certain job will depend on its properties. Some of the properties that define the lubricants are mentioned next.

2.7.1. Density

The density of a fluid (ρ) is defined as its mass per unit volume. It is typically used to characterize the mass of a fluid system. Density is an intensive property, meaning that increasing the amount of the fluid does not increase its density. Different fluids often have different densities, making this parameter an important characteristic unique to each one. Density varies with temperature and pressure. Increasing the pressure on a fluid decreases its volume and therefore increases its density. Increasing the temperature of a fluid decreases its density by increasing its volume.

Under elastohydrodynamic conditions the variation of density due to temperature is insignificant when compared to the influence of pressure, to the point that only variations related to pressure can be considered (this is especially true in this work, since all bearing tests were performed at constant temperature).

$$\rho = \frac{m}{V} \quad (2.1)$$

2.7.2. Viscosity

The density is unique to a fluid but is insufficient to uniquely characterize how fluids behave since two fluids can have close density but behave differently when flowing. There is a need for an additional property to describe the fluidity of a fluid and that is the viscosity.

Viscosity is a measure of a fluid's resistance to flow. It describes the internal friction of a moving fluid. It is necessary to know how it reacts to temperature, pressure and shear strain rate. There are two definitions of viscosity: dynamic viscosity (η) and kinematic viscosity (ν).

To determine viscosity consider the following experiment in which a fluid is placed between two parallel plates. The bottom plate is fixed and the upper moves with a velocity (U). This behaviour is consistent with the definition of a fluid, if a shearing stress is applied to a fluid it will deform continuously. The fluid between the two plates move with a velocity $u = u(y)$ that would vary linearly $u = U \cdot \frac{y}{h}$ as demonstrated in Figure 2.1. Thus a velocity gradient (du/dy) is developed in the fluid between the plates.

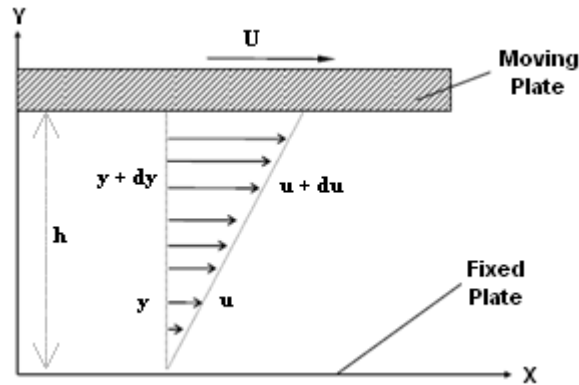


Figure 2.1: Laminar flow of a fluid

The shearing stress (τ) and the rate of shearing strain (du/dy) can be related using equation 2.2.

$$\tau = \eta \frac{du}{dy} \quad (2.2)$$

The constant of proportionality η is called absolute viscosity or dynamic viscosity. According to equation 2.2 any graphic τ versus du/dy should be linear with the slope equal to the viscosity of the fluid. However that is only valid if the fluid is Newtonian. If it's not then the variation of the viscosity with the shear rate is no longer linear as seen in Figure 2.2.

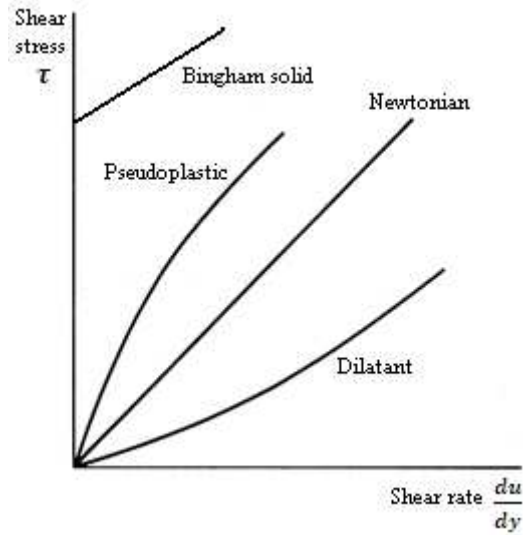


Figure 2.2: Variation of shear stress with shear rate for different types of oils. [23]

The kinematic viscosity (ν) is defined when the flow of the fluid is caused by gravity. This parameter is inversely proportional to the density of the fluid (ρ). The expression $\nu = \frac{\eta}{\rho}$ gives the kinematic viscosity in m^2/s but, it's common to readjust to mm^2/s which corresponds to centistokes (cSt) the most used unit. [5]

2.7.2.1. Thermoviscosity

Thermoviscosity represents the variation of viscosity with the temperature. For most oils the viscosity decreases as the temperature increases.

The method used to determine the kinematic viscosity of the oils is given by ASTM D314 standard. Equation 2.3 represents the variation of the kinematic viscosity with the temperature.

$$\log(\log(\nu + c)) = n - m \times \log(T) \quad (2.3)$$

Where ν represents the kinematic viscosity (cSt), T represents the temperature in Kelvin. The other parameters are constants dependent of the lubricant although the parameter c is 0, 7 for mineral oils. For different oils slightly different values can be found. The parameters m and n are determined by equations 2.4 and 2.5.

$$m = \frac{\log\left(\frac{\log(\nu_0 + c)}{\log(\nu_1 + c)}\right)}{\log\left(\frac{\log(T_1)}{\log(T_0)}\right)} \quad (2.4)$$

$$n = \log(\log(\nu + c)) + m \times \log(T) \quad (2.5)$$

2.7.2.2. Viscosity Index

The viscosity index (VI) is a measure of the amount a lubricant viscosity changes with temperature. To find the VI of a lubricant its viscosity must be known at 40°C and at 100°C. Then two other oils are obtained from data sheets and are designed with index 0 and 100. The VI of intermediate oil can be calculated from the equation 2.6.

$$VI = \frac{L-U}{L-H} \times 100 \quad (2.6)$$

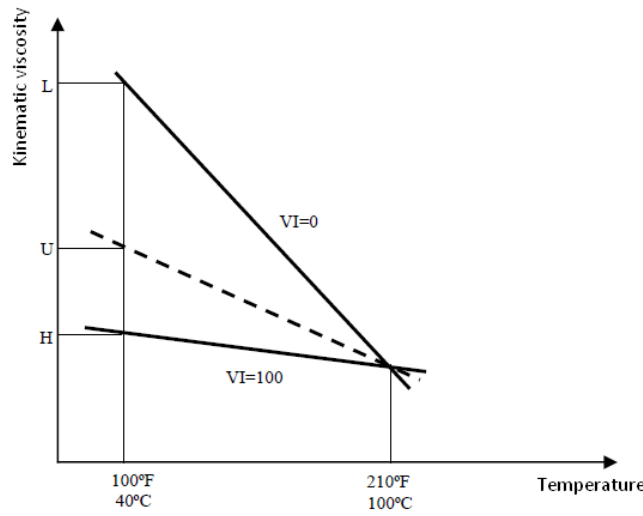


Figure 2.3: Viscosity Index

2.7.2.3. Piezoviscosity

Piezoviscosity represents the variation of viscosity with pressure. The viscosity of lubricants increases with pressure. The behaviour of lubricants under the pressures in EHL is extremely important since it can reach very high values (GPa).

A relation between pressure and viscosity can be given by the Barus equation. [7]

$$\eta_s = \eta_0 \times e^{\alpha p} \quad (2.7)$$

Where η_s represents the lubricant viscosity at pressure p , η_0 represents the viscosity at atmospheric pressure and reference temperature and α represents the piezoviscosity coefficient.

The piezoviscosity coefficient (α) can be related to kinematic viscosity through Gold's equation. [8]

$$\alpha = s \times v^t \times 10^{-9} \quad (2.8)$$

Where v is the kinematic viscosity (cSt) at the operating temperature and s and t are lubricant related constants whose values are shown in Table 2.1.

Table 2.1: Lubricant dependent constants

Constant	Mineral	Ester	PAO	PAG
s	9,904	6,6050	7,3820	5,4890
t	0,1390	0,1360	0,1335	0,1485

In this study one of the selected gear oils (MINE) was composed by a very large quantity of additives (>40%) and because of that, even though he was a mineral based oil, he had a behaviour more similar to PAO so the values of s and t used for this oil were those belonging to PAO.

2.7.3. Other physical properties

- **Specific weight** [N/m^3]: property that characterizes the weight of the system, defined by the ratio between the weight and the volume. Thus it is related to density and it is equal to the product between density and gravitational acceleration.
- **Specific heat** [$kJ/kg.K$]: refers to the amount of heat per unit mass required to raise the temperature by one Kelvin degree.
- **Thermal conductivity** [$W/m.K$]: quantity of heat transmitted through a unit thickness in a direction normal to a surface of unit area, due to a unit temperature gradient.
- **Thermal diffusivity** [m^2/s]: it describes the rate at which heat flows through a material, defined by the ratio between the thermal conductivity and the product between density and specific heat. [6]

2.7.4. Glass transition temperature

When a lubricant is cooled at constant pressure its viscosity increases continuously until it reaches extremely high values. From a certain temperature, the lubricant exhibits behaviours similar to that of an amorphous or glassy solid; this temperature is called "glass transition temperature".

This transformation also occurs at constant temperature, if the pressure at which the lubricant is submitted increases continuously. The pressures and temperatures involved in the operation of an elastohydrodynamic contact are sufficient for such transformation to take place or at least to achieve extremely high viscosities. [6]

2.7.5. Environmental Specifications

As an example, the two following environmental certificates are applicable to oils: GreenMark whose symbol is represented in the left figure and Blauer Engel (Blue Angel) represented on the right. The GreenMark is Chinese (Taiwan) while the Blauer Engel is a German certificate. Both symbols however represent the green certification, which means, the non-toxic behaviour of the lubricant to the environment and nature. [9]



Figure 2.4: Green certification symbols

2.8. Additives

The quality of a lubricant is obtained not only through purification and manufacturing processes but also through the addition of certain chemical compounds or additive agents. [18] Additives are put into lubricants for a variety of purposes and do a great deal to improve the lubricant oils.

The amount of additive used varies from a few hundredths to large per cents. The additives have largely contributed to the progress of primitive combustion engines and all industrial machinery. [6]

Lubricant additives are proved chemicals or materials which, when incorporated in base lubricating fluids, supplement their natural characteristics and improve their field service performance in existing applications or broaden the areas of their utility.

Additives may be divided in two general classes: [18]

1. Those that affect some physical characteristic of the lubricant
2. Those whose end effect is chemical in nature.

Each of these two classes of additives may be blended into a multipurpose additive for ease in compounding the finished lubricant. The principal characteristics of the two classes of additives are: [18]

Chemical characteristics:

- Antioxidant
- Anticorrosion
- Antiwear
- Detergent-dispersant
- Alkaline agent
- Antirust
- Oiliness
- Extreme pressure
- Water repellent
- Metal deactivator
- Silver pacifier

Physical characteristics:

- Pour depressant
- Viscosity-index improver
- Antifoam
- Tackiness
- Emulsifier
- Solid filler
- Colour stabilizer
- Odour control
- Antiseptic

During the last decades various types of lubricant or oil additives were developed.

Unfortunately there is still no way to accurately predict the effects of mixing some chemicals on the base oils, since they are mutually affected. Therefore, some properties of the lubricant can only be obtained by testing or even by trial and error. [10]

2.9. Wind turbine gear oils

Six wind turbine gear oils were selected for this work: 2 esters (ESTF and ESTR), 2 mineral based oils (MINE and MINR), a Polyalkyleneglycol based oil (PAGD) and a Polialphaolefin based oil (PAOR), all of them with the viscosity grade ISO VG 320. Although the manufacturers supplied information on the oils, the lubricants were still submitted to density and viscosity measurements to confirm the data provided.

The chemical composition of the lubricants used (tested at CETRIB) is demonstrated in the Table 2.2.

Table 2.2: Chemical properties of the wind turbine gear oils

Parameter	Units	ESTF	ESTR	MINE	MINR	PAGD	PAOR
Zinc (Zn)	[ppm]	0,7	6,6	<1	0,9	3,5	1
Magnesium (Mg)	[ppm]	1,3	1,3	<1	0,9	0,5	1,4
Phosphorus (P)	[ppm]	449,4	226,2	460	354,3	415,9	1100
Calcium (Ca)	[ppm]	n.d.	14,4	2	2,5	0,5	0,8
Boron (B)	[ppm]	33,6	1,7	36	22,3	28,4	1
Sulfur (S)	[ppm]	5030	406	6750	11200	5020	362

It's possible to observe that there are oils that are significantly different than the others. For example calcium was not detected in ESTF but had a high value for ESTR, phosphorous had high values in the PAOR and sulfur was also high in the MINR.

The physical properties of the lubricants used (also tested at CETRIB) are shown in Table 2.3.

Table 2.3: Physical properties of the wind turbine gear oils

Parameter	Units	ESTF	ESTR	MINE	MINR	PAGD	PAOR
Density (15°C)	g/cm^3	0,957	0,915	0,893	0,902	1,059	0,860
Density (25°C)	g/cm^3	0,950	0,907	0,886	0,896	1,052	0,855
Viscosity (40°C)	cSt	323,95	301,93	328,30	319,24	289,13	313,52
Viscosity (70°C)	cSt	88,98	79,84	93,19	65,81	104,52	85,41
Viscosity (100°C)	cSt	34,84	30,71	37,13	22,41	48,09	33,33
Viscosity index	-	153	140	163	85	230	150
Thermoviscosity 40°C	K^{-1}	0,0499	0,0491	0,0493	0,0639	0,0373	0,0507
Thermoviscosity 70°C	K^{-1}	0,0358	0,0352	0,0355	0,0428	0,0284	0,0362
Thermoviscosity 100°C	K^{-1}	0,0266	0,0262	0,0264	0,0301	0,0221	0,0267
Piezoviscosity 40°C	Pa^{-1}	1,45E-8	1,44E-8	1,60E-8	2,21E-8	1,28E-8	1,59E-8
Piezoviscosity 70°C	Pa^{-1}	1,22E-8	1,21E-8	1,35E-8	1,77E-8	1,11E-8	1,34E-8
Piezoviscosity 100°C	Pa^{-1}	1,08E-8	1,07E-8	1,20E-8	1,53E-8	0,99E-8	1,18E-8
Thermal expansion coefficient	-	-6,7E-4	-8,1E-4	-6,6E-4	-5,8E-4	-7,1E-4	-5,5E-4

From Table 2.3 it can be seen that some oils have properties which differ significantly or slightly from the others. For example the PAGD oil has very high density (even greater than water 1 g/cm^3) and also a very high viscosity index while the PAOR oil has relatively low density and the MINR oil has a very low viscosity index.

Figure 2.5 shows the variation of kinematic viscosity with temperature of the selected gear oils.

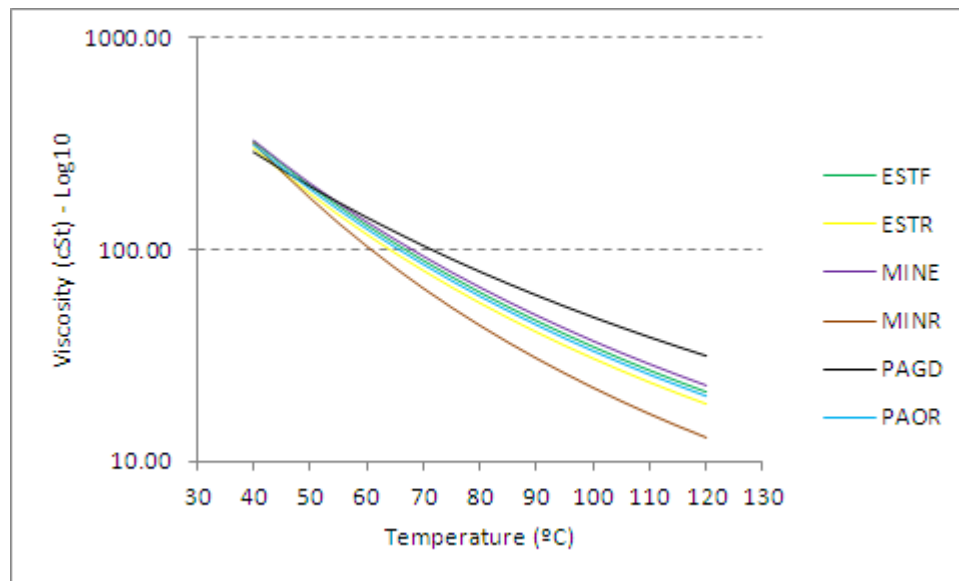


Figure 2.5: Variation of kinematic viscosity with temperature

From Figure 2.5 it's possible to observe that as the temperature increases the viscosity of two oils separate from the others. The PAGD oil has the highest viscosity while the MINR has the lowest. The other oils have very close and intermediate values between those two oils.

Figure 2.6 shows the variation of density of the selected gear oils with temperature.

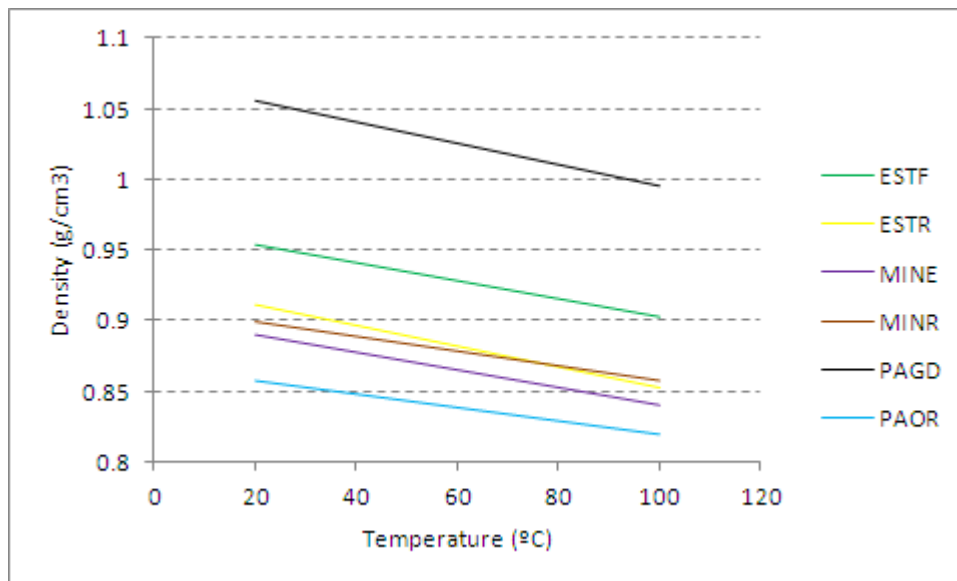


Figure 2.6: Variation of density with temperature

From Figure 2.6 it's possible to observe that the PAGD oil not only has the highest density but its value is above the density of water (>1) which is most unusual for lubricant oils; the other oils density are relatively close to each other.

3. Elastohydrodynamic Lubrication

3.1. Normal contact between elastic solids of revolution – Theory of Hertz

When two elastic solids of revolution are brought in contact with each other, they begin by contacting at a single point or along a line. If a load is applied they deform in the contact vicinity of the initial contact point creating a small contact area. It should be mentioned that the area dimensions are much smaller when compared to the two solids.

To analyse this kind of problems it is required to use a contact model to determine the contact area as well as its reactions with bigger loads, intensity and distribution of normal contact pressures transmitted through the surfaces. If the pressures are known it is possible to calculate other parameters such as the displacement, stress and strain fields applied to the solids, on the surface and sub-surface of the contacting solids. [11]

The geometry of the contacting surfaces, both micro and macro, have an important influence on the contact behaviour so it is necessary to carefully characterize them. It was mentioned that two solids of revolution begin touching at a single initial point of contact, designated by 'O'. This point is also the origin of a coordinate system in which the plane [XOY] is the plane tangent to the contacting surfaces, the axis Z is normal to the tangent plane, passing through the centre of the two solids.

It is also considered that the surfaces are smooth and continuous curves (solids of revolution). Taking this into consideration it is possible to define the principal planes of curvature (x_1Oz_1 , y_1Oz_1 , x_2Oz_2 , y_2Oz_2) and the corresponding radii of curvature (R_{x1} , R_{x2} , R_{y1} , R_{y2}).

The line of action of the applied force (F_n), crosses the centres of the two solids and also the initial point of contact making it perpendicular to the plane tangent to the contacting surfaces.

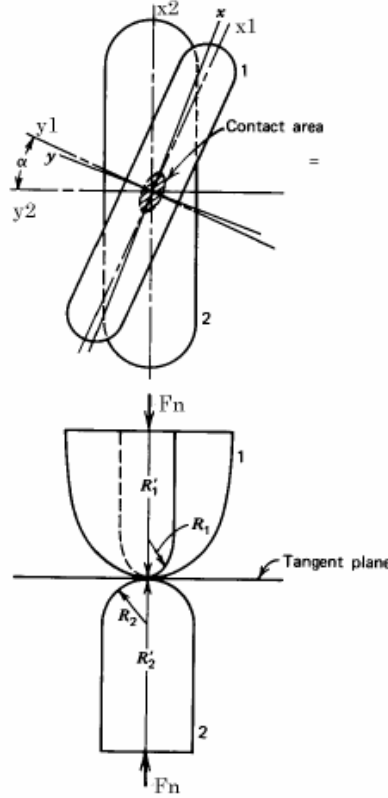


Figure 3.1: Principal plans and radii of curvature. [11]

3.1.1. Contact model

Essentially the contact model creates a relation among the distance between the surfaces of the contacting solids, measured along the normal to the common tangent plane, before and after the elastic deformation created by the applied load. [11]

As seen in the Figure 3.1 the loaded solids can form an angle between them, however in the particular case where the angle α is null, which corresponds to many current applications the equivalent curvatures A and B ($A \geq B$) are defined by the equations 3.1 and 3.2.

$$A = \frac{1}{R_x} = \frac{1}{2} \times \left(\frac{1}{R_{x1}} + \frac{1}{R_{x2}} \right) \quad (3.1)$$

$$B = \frac{1}{R_y} = \frac{1}{2} \times \left(\frac{1}{R_{y1}} + \frac{1}{R_{y2}} \right) \quad (3.2)$$

3.1.2. Contact surface shape

The sets of points for which $Ax^2 + By^2 = \text{constant}$, correspond to points within the same distance to the common tangent plane, corresponding to an ellipse. [11] The elliptical shape is defined by the equation 3.3.

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1 \quad (3.3)$$

Where a and b represent the minor and major axis of the contact ellipse, respectively.

3.1.3. Theory of Hertz

The Hertz theory is based on the following hypotheses: [11]

- 1) The materials of the solids are perfectly homogeneous, isotropic and elastic as referred by the Hooke's law;
- 2) The bodies are solids of revolution, with continuous surfaces and their main radii of curvature is known in the proximities of the initial contact point;
- 3) The load is purely normal, and the surfaces do not transmit tangential traction (surfaces without friction).

Hertz added an additional hypothesis: [11]

- 4) The solids behave as elastic half-spaces of plane surface, submitted to a normal load, applied on a small elliptical area. The elastic half-space approximation is used to determine the local displacements.

For the last hypothesis to be valid it is necessary for two new ones to be satisfied.

- 5) The dimensions of the contact area must be small when in comparison with the dimensions of each contacting solids;
- 6) The dimensions of the contact area must be small when compared with the radii of curvature of the solids.

Hypothesis number 5 is necessary to be certain that the solid is similar to an elastic half-space so, the contact pressures are not influenced by the presence of the borders of the solids near the contact area. Hypothesis number 6 ensures that the solids surfaces outside the contact area resemble the plane surface of the elastic half-space. [11]

3.1.4. Hertz solution

Equivalent Young modulus for the two contacting bodies (E^*) is defined by expression 3.4.

$$E^* = \left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)^{-1} \quad (3.4)$$

The maximum Hertz pressure (p_0) and the mean pressure (p_m) inside the contact are determined by the expressions 3.5 and 3.6.

$$p_0 = \frac{3}{2} \times \frac{F_n}{\pi ab} \quad (3.5)$$

$$p_m = \frac{F_n}{\pi ab} \quad (3.6)$$

The dimensions of the contact ellipses are defined by equations 3.7 and 3.8.

$$a = C_a \sqrt{\frac{F_n}{(A+B) \times E^*}} \quad (3.7)$$

$$b = \frac{a}{k} \quad (3.8)$$

The values of C_a and k are dependent on the ratio of curvatures A/B .

3.1.5. Linear contact

In case of contact between two solid cylinders, initially in contact along their generating lines, the problem becomes two-dimensional as shown in Figure 3.4.

When submitted to a normal load per unit length, the two cylinders create a rectangular contact area in the form of a narrow band (width = $2a$ and length = l).

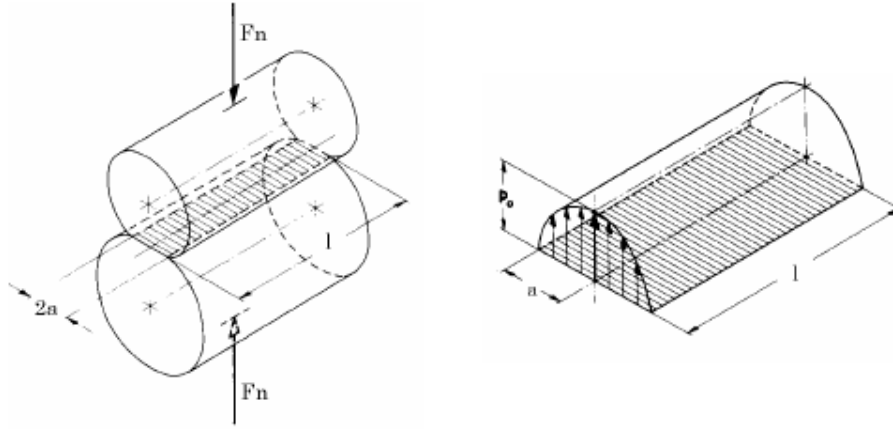


Figure 3.2: Linear contact [11]

Hertz examined this problem by treating it as a limit of an elliptical contact, where the size b of the contact ellipses becomes very large in comparison to a ($b = l/2 \gg a$). In this case the elliptic coefficient k ($k = a/b$) tends to zero. [11]

The semi-width of Hertz is determined by equation 3.9.

$$a = \frac{R_x \times p_0}{E^*} = \sqrt{\frac{2 \times E^* \times F_n}{\pi \times l \times R_x}} \quad (3.9)$$

Considering the results given by the previous formula the maximum hertz pressure is given by expression 3.10.

$$p_0 = \frac{E^* \times a}{R_x} = \sqrt{\frac{2 \times R_x \times F_n}{\pi \times l \times E^*}} \quad (3.10)$$

3.2. Elastohydrodynamic Lubrication Theory

Elastohydrodynamic Lubrication (EHD) theory is the key feature to understand lubrication, friction and energy phenomena in heavily loaded contacts, such as lubricated Hertzian contacts. [6] EHD lubrication allows the evaluation of three crucial aspects in the performance of a lubricated Hertzian contact (or elastohydrodynamic):

- The thickness of the lubricant film generated between the contacting surfaces is accompanied by elastic deformation of the contacting solids.
- The friction between the contact surfaces due to visco-elastoplastic deformation of the oil film, takes into account the rheological behaviour of the lubricant.
- The energy balance of contact considers the power dissipation in the lubricant film due to shear stresses installed and the heat evacuation by the flow of the lubricant and surfaces in contact.

Considering the isolated effect of each one of these physical phenomena, the elasticity of surfaces or pressure effect on viscosity, usually neglected in the hydrodynamic lubrication, did not explain the behaviour of counter-formal contacts.

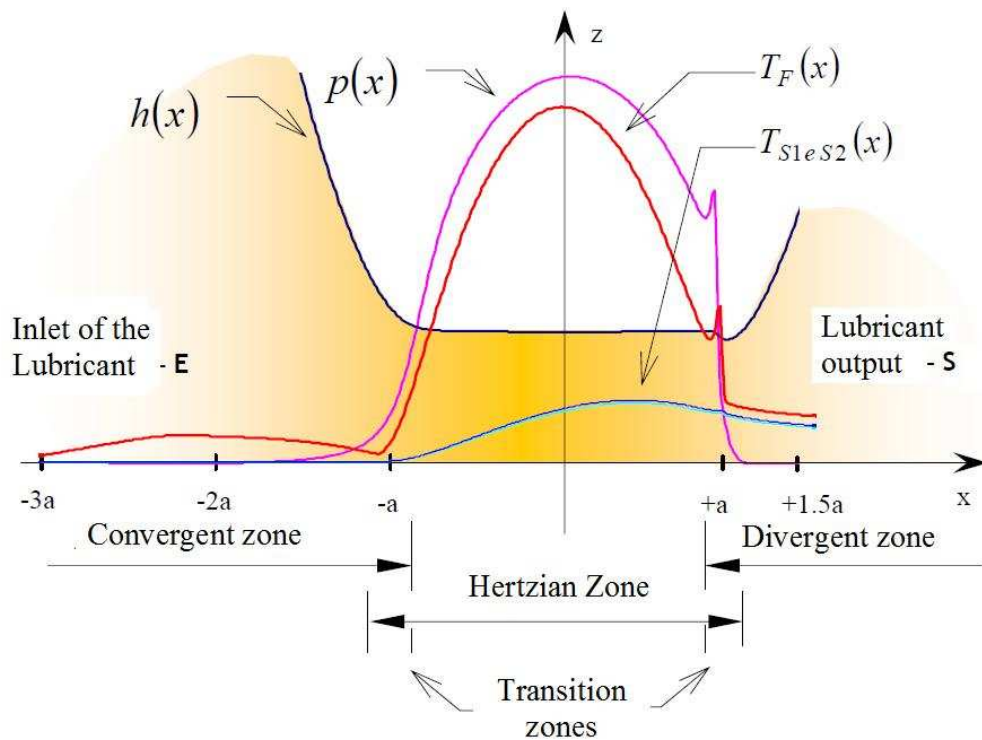


Figure 3.3: Lubricated Hertzian contact. [6]

In 1949, Grubin showed that the simultaneous consideration was fundamental in the analysis of counter-formal contacts, leading to the prediction of the oil film thickness separating the surfaces and the load characteristics of these contacts, giving rise to a new area of study (elastohydrodynamic lubrication). [6]

Petrusevich (1951), confirmed the results of Grubin and obtained solutions that satisfy both the equations of hydrodynamics and elasticity of surfaces, for a wide range of operating conditions, and identified two important characteristics of EHD contacts: first that the near parallelism between the surfaces deformed with a small restriction on the thickness near the exit of the contact and, second a nearly Hertzian pressure distribution in full contact, with a second peak pressure also near the exit of the contact. [6]

3.3. Lubricant film thickness

EHD lubrication is the most common type of lubrication in mechanical components such as rolling bearings, gears and cams. [6]

In these types of contacts lubrication is determined through the film thickness that separates the roughness between the two surfaces. Nowadays, the lubricant film thickness prediction follows the D. Dowson and G. R. Higginson theory, [12] which implicates an isothermal contact between smooth surfaces and fully flooded lubrication.

The centre film thickness in elliptical contacts (h_0) and the minimum lubricant film thickness (h_m) are given by equations 3.11 and 3.12. [6]

$$h_0 = 1,345 \times R_x \times U^{0,670} \times G^{0,530} \times W^{-0,067} \underbrace{\left\{ 1 - 0,61 \exp \left[-0,752 \times \left(\frac{R_y}{R_x} \right)^{0,64} \right] \right\}}_{C_0} \quad (3.11)$$

$$h_m = 1,815 \times R_x \times U^{0,680} \times G^{0,490} \times W^{-0,073} \underbrace{\left\{ 1 - \exp \left[-0,7 \times \left(\frac{R_y}{R_x} \right)^{0,64} \right] \right\}}_{C_m} \quad (3.12)$$

Equations 3.13 and 3.14 are also valid.

$$h_0 = 1,165 \times C_0 \times \frac{[\eta_0(U_1+U_2)]^{0,67} \times \alpha^{0,53} \times R_x^{0,464}}{F_n^{0,067} \times E^{*0,073}} \quad (3.13)$$

$$h_m = 1,438 \times C_m \times \frac{[\eta_0(U_1+U_2)]^{0,68} \times \alpha^{0,49} \times R_x^{0,466}}{F_n^{0,073} \times E^{*0,117}} \quad (3.14)$$

Where:

R_x – Equivalent curvature radius direction x [m]

R_y – Equivalent curvature radius direction y [m]

E^* – Equivalent young modulus [Pa]

U – Velocity parameter (non-dimensional)

$$U = \frac{\eta_0(U_1 + U_2)}{2 \times R_x \times E^*} \quad (3.15)$$

U_1, U_2 – Surface velocity of solid 1 and 2, respectively [m.s]

η_0 – Lubricant dynamic viscosity at lubricant feeding temperature [Pa.s]

G – Material parameter (non-dimensional)

$$G = 2 \times \alpha \times E^* \quad (3.16)$$

α – Lubricant piezoviscosity coefficient at feeding temperature [Pa⁻¹]

ν – Lubricant kinematic viscosity at feeding temperature [mm²/s]

W – Load parameter for elliptic contacts (non-dimensional)

$$W = \frac{2 \times F_n}{E^* \times R_x^2} \quad (3.17)$$

F_n – Normal load [N]

H – Lubricant film thickness (non-dimensional)

$$H = \frac{h}{R_x} \quad (3.18)$$

The centre film thickness in linear contacts (h_0) and the minimum lubricant film thickness (h_m) are defined by equations 3.19 and 3.20. [6]

$$h_0 = 0,975 \times R_x \times U^{0,727} \times G^{0,727} \times W^{-0,091} \quad (3.19)$$

$$h_m = 1,325 \times R_x \times U^{0,70} \times G^{0,54} \times W^{-0,13} \quad (3.20)$$

Equations 3.21 and 3.22 are also valid.

$$h_0 = 0,975 \times \frac{[\alpha\eta_0(U_1+U_2)]^{0,727} \times R_x^{0,364} \times (l.E^*)^{0,091}}{F_n^{0,091}} \quad (3.21)$$

$$h_m = 1,325 \times \frac{[\eta_0(U_1+U_2)]^{0,70} \times \alpha^{0,54} \times R_x^{0,43} \times l^{0,13}}{F_n^{0,13} \times E^{*0,03}} \quad (3.22)$$

In the case of the linear contacts, two different parameters are defined: [6]

l – Length of contact [m]

W – Load parameter for linear contacts (non-dimensional)

$$W = \frac{F_n}{E^* \times l \times R_x} \quad (3.23)$$

3.4. Correction of lubricant film thickness

The solutions presented for the lubricant film thickness were obtained taking into account the following conditions:

- The contact is isothermal
- Lubrication is abundant
- The surfaces are smooth

However, the lubricant film thickness must be corrected to take into account: [6]

- The heating of the lubricant in contact inlet
- The contact inlet feeding conditions
- The contacting surfaces roughness

$$h_{0c} = \phi_T \times \phi_A \times \phi_R \times h_0 \quad (3.24)$$

Where:

ϕ_T – Inlet shear heating parameter

ϕ_A – Inlet feeding parameter

ϕ_R – Inlet roughness parameter

These conditions do not apply to the lubricant minimum film thickness.

It should be mentioned that it is hard to determine the correction factors related to the lubricant feeding and the roughness so, often, only the parameter related to temperature is taken into consideration.

$$h_{0c} = \phi_T \times h_0 \quad (3.25)$$

3.4.1. Influence of heating in the inlet of the EHD contact

In the contact inlet, the lubricant film suffers a very high shear deformation, due to the pressure gradient and to the rolling and sliding speeds.

This shear deformation results in a sharp energy dissipation (inlet shear heating) which causes the increase of lubricant temperature (ΔT_f), the decrease of viscosity (η_0) and consequently the decrease in the lubricant film thickness (h_0). [6]

This reduction in lubricant film thickness is defined by parameter (ϕ_T).

Equation 3.26 can be used to determine the thermal correction factor.

$$\phi_T = \{1 + 0,1 \times [(1 + 14,8 \times V_e^{0,83}) \times L^{0,64}]\}^{-1} \quad (3.26)$$

Where:

V_e – Slip rate (non-dimensional)

$$V_e = \frac{|U_1 - U_2|}{|U_1 + U_2|} \quad (3.27)$$

L – Lubricant thermal parameter (non-dimensional)

$$L = \frac{\beta \times \eta_0 \times (U_1 + U_2)^2}{K_f} \quad (3.28)$$

β – Lubricant thermo viscosity coefficient [$^{\circ}\text{K}^{-1}$]

K_f – Lubricant thermal conductivity [W/m $^{\circ}\text{K}$]

3.4.2. Correction due to contact inlet starvation

Experimental evidence showed that if the inlet of an EHD contact is not completely filled with oil a situation may happen in which the operation is affected by the lack of lubricant (oil starvation). [6]

Experimental results show that the contact starvation can be expressed through the value of the coordinate x_e which refers to the point where the lubricant film is formed as showed Figure 3.4.

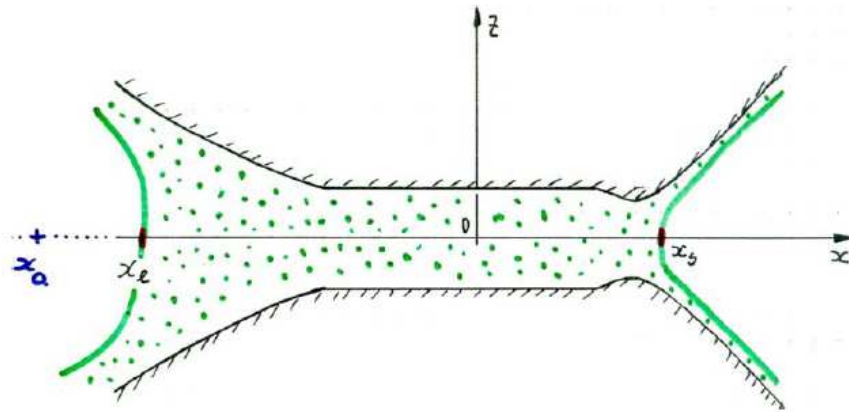


Figure 3.4: Point of formation of menisco in EHD contact. [6]

Figures 3.5 and 3.6 show how ϕ_A depends on the coordinate x_e in case of elliptical and linear contacts, respectively.

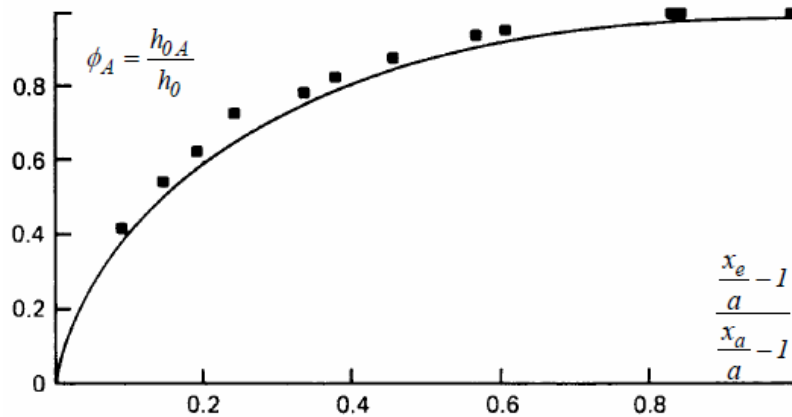


Figure 3.5: Point of formation of menisco in elliptical EHD contact. [6]

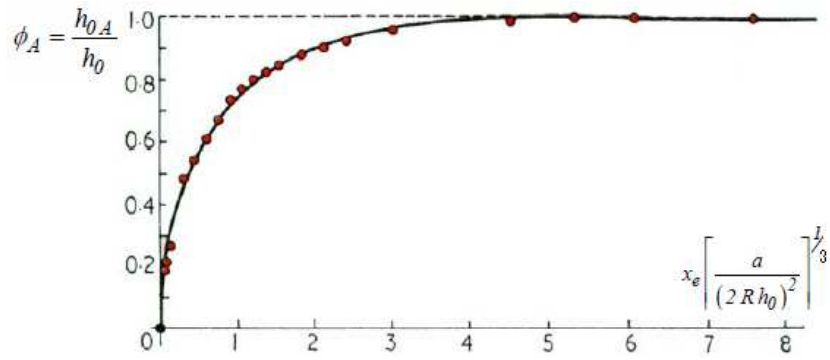


Figure 3.6: Point of formation of menisco in linear EHD contact. [6]

3.4.3. Correction due to the roughness of the contact surfaces

Very often, surface roughness is classified according to their preferred orientation in longitudinal, isotropic and transverse as shown in Figure 3.7. In practice, this classification seems to correspond to many current applications, as identified in Table 3.1.

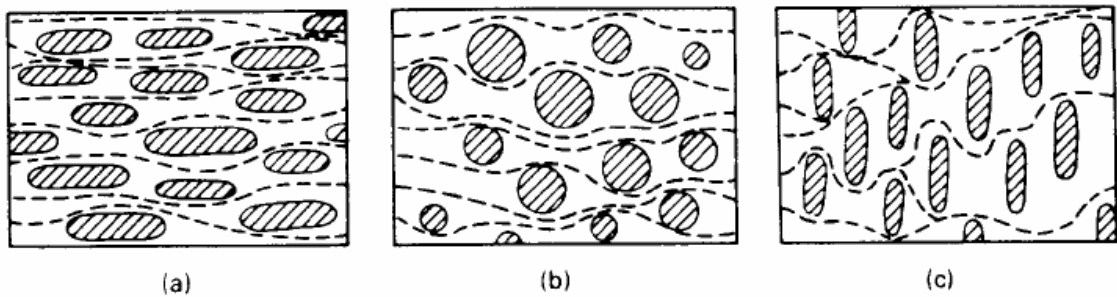


Figure 3.7: Types of orientation of surface roughness (a-Longitudinal, b-Isotropic, c-Transverse). [6]

Table 3.1: Orientation of the surface roughness [6]

(a) Longitudinal	(b) Isotropic	(c) Transverse
Bearing raceway Bearing roller Cam	Bearing balls	Gears

3.4.4. Specific lubricant film thickness

In general, the influence of surface roughness on lubricant film thickness is presented in function of the specific lubricant film thickness, [6] which is defined by equation 3.33.

$$\Lambda = \frac{h_{0c}}{\sigma} \quad (3.33)$$

Where

Λ – Specific lubricant film thickness (non-dimensional)

h_{0c} – Corrected lubricant film thickness at contact centre [m]

σ – Composite roughness of the contacting surfaces [m]

Table 3.2 shows typical values of the composite surface roughness for rolling bearings.

Table 3.2: Composite roughness values for rolling bearings [6]

Bearing type	$\sigma = R_q [\mu m]$
Precision ball	0,05
Balls	0,18
Cylindrical rollers	0,36
Tapered rollers	0,23

3.5. Lubrication regimes

The definition of lubrication regimes is associated with typical values of the specific lubricant film thickness. There are several lubrication regimes as indicated in Table 3.3.

Table 3.3: Lubrication regimes [6]

Λ	Regimes	Observations
$\Lambda \geq 10 \times \Lambda_1$	Hydrodynamic	Lubricant film very thick
$\Lambda \geq \Lambda_1$	Full film	Contact surfaces completely separated by the lubricant film
$\Lambda_0 < \Lambda < \Lambda_1$	Mixed film	Surfaces in contact partially separated by the lubricant film partially in metal-to-metal contact
$\Lambda \leq \Lambda_0$	Boundary film	There isn't a lubricant film separating the contacting surfaces. Metal-to-metal contact is predominant.

The values of Λ_0 and Λ_1 depend on the applications considered. Typical values for rolling bearings and gears are shown in table 3.4.

Table 3.4: Values of Λ_0 and Λ_1 in EHD lubrication [6]

EHD Lubrication Regimes		
Λ	Bearings	Gears
Full film	$\Lambda \geq 3,0$	$\Lambda \geq 2,0$
Mixed film	$1,0 < \Lambda < 3,0$	$0,7 < \Lambda < 2,0$
Boundary film	$\Lambda \leq 1,0$	$\Lambda \leq 0,7$

Experimental results demonstrate that a relationship exists between the specific lubricant film thickness and the probability of a surface failure (scuffing, contact fatigue, pitting, among others) in an elastohydrodynamic contact. The concept of specific lubricant film thickness is of crucial importance in the design of mechanical components operating under EHD conditions. [6]

4. Rolling Bearings Tested

In this work two types of rolling bearings were tested: thrust ball bearing (SKF 51107) and thrust roller bearing (SKF 81107 TN). They were tested using the six wind turbine gear oils selected, all with a viscosity grade ISO VG 320.

4.1. Thrust ball bearing – TBB

Table 4.1 and Figure 4.1 show the most important characteristics of the thrust ball bearing 51107. [1]

Table 4.1: Characteristics of thrust ball bearing 51107. [1]

Basic load ratings		Fatigue load limit	Minimum load factor	Speed ratings		Mass	Designation
Dynamic C	Static C ₀	P _u	A	Reference speed	Limiting speed		
kN	kN	kN	-	r/min	r/min	kg	
19,9	51	1,86	0,013	5600	7500	0,080	51107

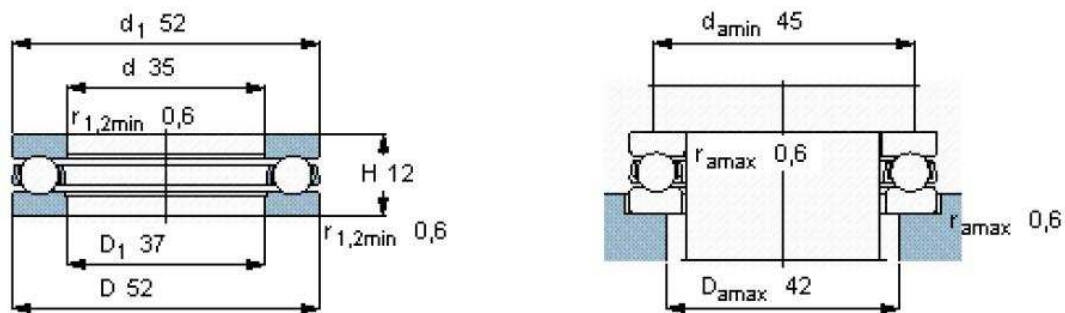


Figure 4.1: Dimensions of the thrust ball bearing SKF 51107. [1]

4.2. Thrust roller bearing – RTB

Table 4.2 and Figure 4.2 show the most important characteristics of the thrust roller bearing 81107 TN. [1]

Table 4.2: Characteristics of thrust roller bearing 81107 TN. [1]

Basic load ratings		Fatigue load limit	Minimum load factor	Speed ratings		Mass	Designation
Dynamic C	Static C0	Pu	A	Reference speed	Limiting speed		
kN	kN	kN	-	r/min	r/min	kg	
29	93	9,15	0,00069	2800	5600	0,073	81107 TN

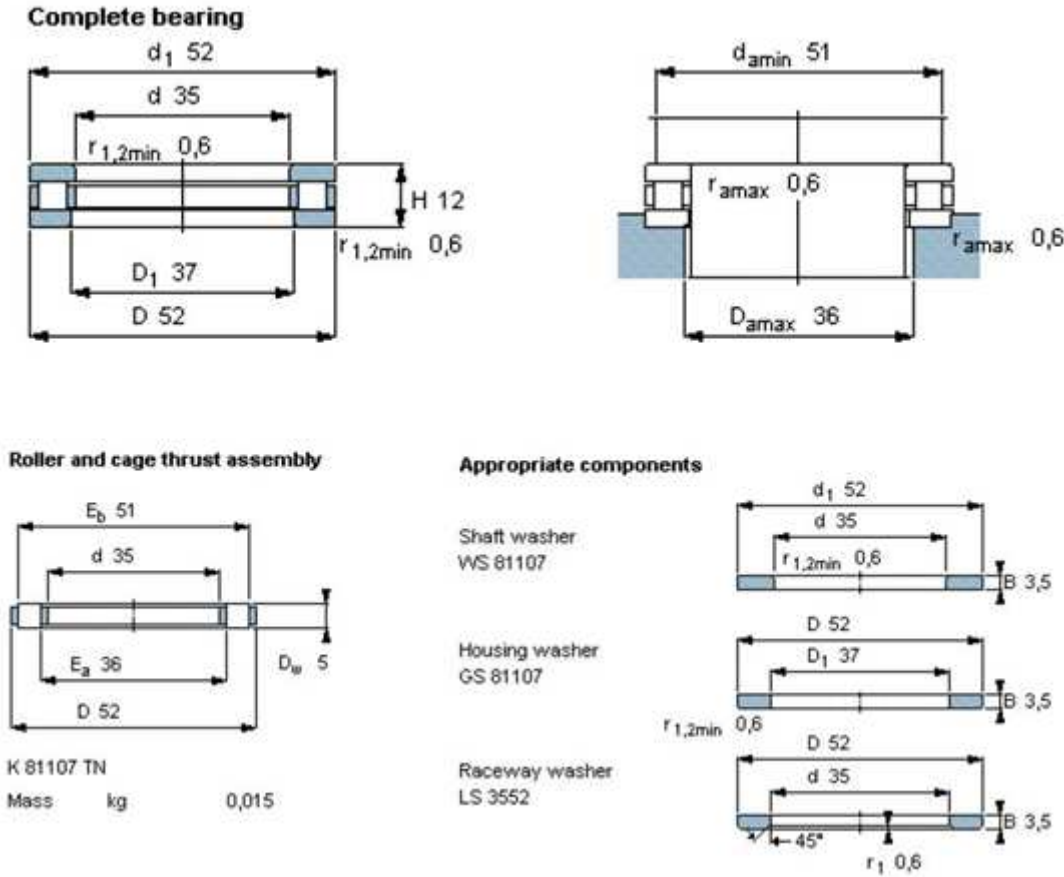


Figure 4.2: Dimensions of the thrust roller bearing SKF 81107 TN. [1]

4.3. Rating life of bearings

In the case of modern high quality rolling bearings, the basic rating life can deviate significantly from the real life in particular applications. The lifetime for a specific application depends on a variety of influencing factors. Some of those are: lubrication, contamination, misalignment, installation and environmental conditions. [2]

Therefore, a modified life equation is used to complement the basic rating life (ISO 281:1990 / Amd 2:2000 standard). This fatigue life calculation uses a modification factor to take into account the conditions of lubrication and contamination of the rolling bearing and the fatigue limit of the material (steel).

The standard mentioned above also determines that the bearing manufacturers recommend an appropriate method to calculate the modification factor of life to be applied on a rolling bearing, based on operating conditions. The SKF Life modification factor (a_{SKF}) applies the concept of a fatigue load limit (P_u) analogous to the one used in calculating other machine elements. The value of the fatigue load limit is provided in the product table. In addition, the SKF life modification factor considers the lubrication conditions (viscosity ratio k) and the contamination level factor (η_c), that reflect the operating conditions of the application. [2]

The equation for the nominal life SKF according with the standard (ISO 281:1990 / Amd 2:2000) [1] is defined by equation 4.1.

$$L_{nm} = a_1 \times a_{SKF} \times L_{10} \quad (4.1)$$

If the operating speed is constant, nominal life can be expressed in operating hours, using the following equation.

$$L_{nmh} = a_1 \times a_{SKF} \times L_{10} \times \frac{10^6}{60n} \quad (4.2)$$

Where

L_{nm} – SKF rating life (with 100-n% reliability), in millions of revolutions,

L_{nmh} – SKF rating life (with 100-n% reliability), in operating hours,

a_1 – Life adjustment factor for reliability,

a_{SKF} – SKF life modification factor,

- L_{10} – Basic rating life (considering a reliability of 90%), in millions of revolutions,
 n – Rotational speed [r/min].

$$L_{10} = \left(\frac{C}{P}\right)^p \quad (4.3)$$

- C – Basic dynamic load rating [kN],
 P – Equivalent dynamic bearing load [kN],
 p – Exponent of the life equation.

The life adjustment factor (a_1) is defined in the Table 4.3.

Table 4.3: life adjustment factor (a_1). [2]

Reliability	Failure probability (n)	SKF rating life (L_{nm})	Life adjustment factor (a_1)
90	10	L10m	1
95	5	L5m	0,62
96	4	L4m	0,53
97	3	L3m	0,44
98	2	L2m	0,33
99	1	L1m	0,21

The parameter a_{SKF} is defined using Figures 4.3 and 4.4.

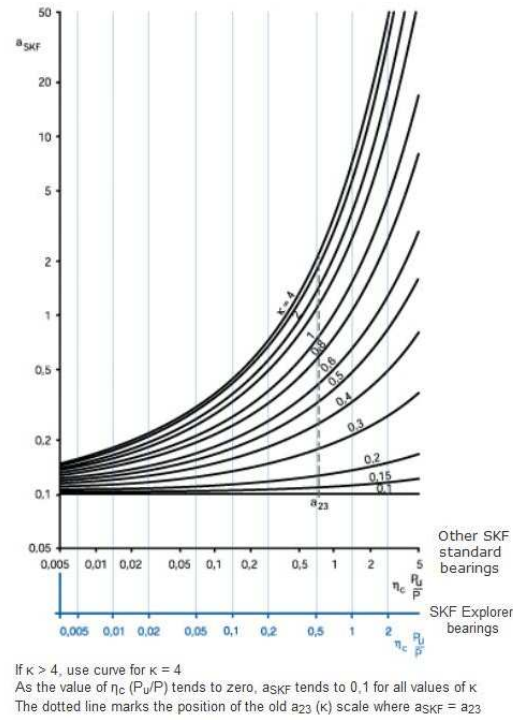


Figure 4.3: Determination of a_{SKF} for thrust roller bearing [2]

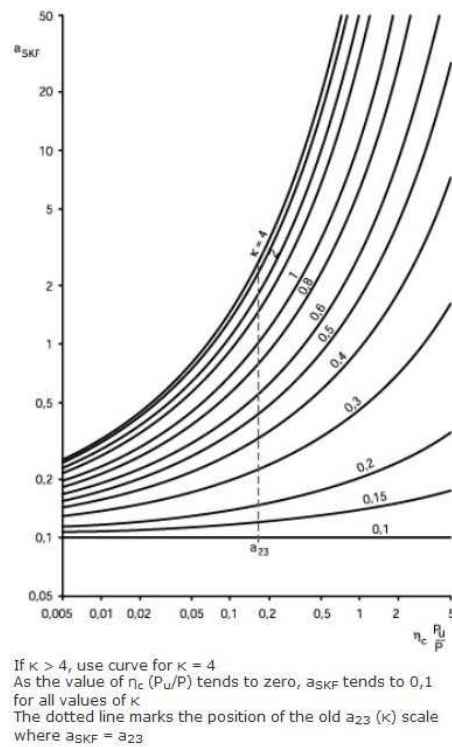


Figure 4.4: Determination of a_{SKF} for thrust ball bearing [2]

Where η_c is the contamination factor and K is the viscosity ratio. [2]

The viscosity ratio (K) is determined using equation 4.4.

$$K = \frac{v}{v_1} \quad (4.4)$$

Where

v – Operating viscosity of the lubricant [mm^2/s]

v_1 – Rated viscosity [mm^2/s] depending on the mean diameter and rotational speed of the bearing (see Figure 4.5)

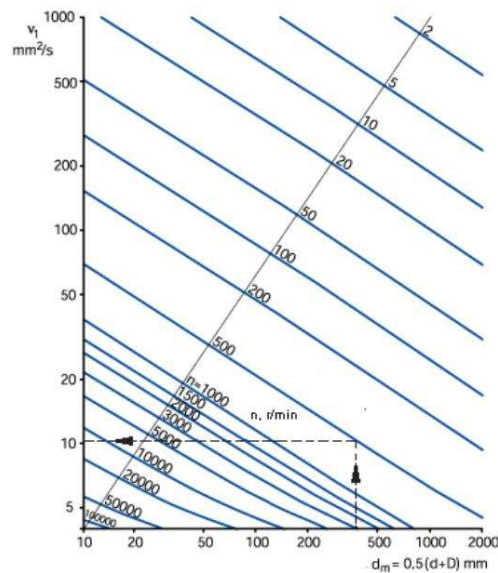


Figure 4.5: Rated viscosity [2]

4.4. Causes of bearing damage

It is essential that the user of rolling bearings is able to diagnose the cause of damage when it happens and to take measures to avoid it. In several cases a damaged bearing does not necessarily mean that it cannot be used, but if the damage is allowed to grow, the bearing may no longer be suited to its designed application and should be considered to have failed. Failure may give an alert of its presence through an increase in noise level, vibration, temperature or torque.

When rolling bearing failure occurs, evidence of its initiation may be lost. A lot more can be learned from damage in its early stage, when there is a chance for diagnosis and correction. There exist several causes of damage and some of those are described below. [18]

Abuse before mounting: it is not unusual to disassemble a bearing for cleaning and inspection. On reassembling, the rolling elements are pushed back in place and if they are not in their proper position nicks and axial smear may result, promoting further damages.

Improper mounting: when a bearing is mounted in a rotating shaft it should be pressed with appropriate interference for the application. If there is too little interference the bearing may slip on the shaft, if there is too much the bearing may already be under stress before the load is applied. Misalignment in mounting may produce edge loading creating local fatigue failure.

Inadequate lubrication: progressive damage may be caused by insufficient lubrication. If the lubricant deteriorates, or if the quantity gradually decreases below effective lubrication will lead to seizure of rolling elements and smearing of the raceways. A lubricant with insufficient viscosity and film strength may lead to similar results.

Wear from abrasives: hard particles which will scratch, cut, or lap the softer surfaces of the bearings are called abrasives. They can come from the environment or be the result of other wear mechanisms.

Corrosion: when rubbing takes place in a corrosive environment, surface reactions occur, creating products on the surfaces. These products do not adhere very well to the surfaces and further rubbing removes them. Then the process restarts. Thus, a slow but continuous form of wear may take place.

Fretting corrosion: it's the removal of material through the combined action of chemical attack and oscillatory movement. It is a very common situation, since most machines produce vibrations when operating.

Fatigue: it refers to the damage sustained by the material due to cyclic loading conditions.

4.5. Bearing wear

The wear of a component is defined as the removal of the surface material in the form of loose particles during service. [15]

The result of wear is a continuing loss of the geometric accuracy of the rolling contact surfaces and gradual deterioration of bearing function, for example, increased deflection, increased friction and temperature, increased vibration and so forth. [15]

Mild wear is frequently called simply wear. Distinction is often made between two types of mild wear as follows: [15]

- Adhesive or two-body wear occurring at the interface of the contacting surfaces.
- Abrasive or three-body wear occurring due to extraneous hard particles acting at the interface of the contacting surfaces.

The worn surface to the naked eye appears “featureless, matte and with no direction” and characteristic finishing marks of the original manufactured surface are worn out. However, mild wear by itself, is not a mode of bearing failure, nor does it lead to rapid bearing failure.

Severe wear or galling is defined as the transfer of component surface material in visible patches from a location on one surface to a location on the contacting surface, and possibly back on to the original surface. [15] This transfer of material occurs because of shear forces of high-friction due to sliding over the asperities of the surfaces. In rolling bearings, this severe wear phenomenon is also named smearing. It’s a welding phenomenon entailing adhesive bonding between material portions of the contacting surfaces. Smearing indicates increased bearing friction and can lead to less-than-expected bearing endurance.

4.5.1 Micropitting

Micropitting is the result of a process of rolling/sliding contact fatigue and only happens in zones where there are conditions for rolling with significant slip. [22] Its appearance is due to the initiation of fatigue cracks in the surface. These cracks propagate to the interior of the material, with a small inclination, in the direction of the tangential force.

4.5.2 Spalling

The spalling results from the initiation of fatigue cracks, deep in the sub-surface of the contact. [22] These cracks are due to microstructural defects of the material (for example inclusions). Like micropitting, spalling is a contact fatigue mechanism, where the cracks propagate from the interior of the material to the surface (the opposite to what happens in micropitting).

5. Lubricant and Bearing Tests

As mentioned before, the six wind turbine gear oils were characterized and their viscosities and densities were measured. The following procedures were used.

5.1. Viscosity measurement

The oils viscosities were measured using an Engler viscometer (DIN 51560 or ASTM D 1665 standard) [20]. The Engler viscometer is composed by a recipient where the sample of the lubricant to analyse is introduced, which has got a calibrated hole on the bottom that opens or closes with a wood pole as shown in Figure 5.1. To heat and maintain the lubricant at a constant temperature this recipient is involved by another which contains a liquid that will be heated through an electrical resistance. These recipients are supported by a tripod that allows it to be horizontally levelled. There are two thermometers to control the temperature (one in each liquid).



Figure 5.1: Engler viscometer.

The testing phases for each test are the following.

1. Cover the hole of the recipient and place the fluid until it covers its three reference points (around 250 ml).
2. Turn on the electrical resistance to heat the fluid to the selected temperature. It is necessary to adjust the power so that the temperature stabilizes and remains constant while measuring.
3. Place a graduated recipient bellow the hole and when the temperature is stabilized open it and simultaneously start the timer.
4. When the leaked fluid reaches the 200 ml mark, stop the timer and close the hole.

Repeat the steps procedure for each test temperature. The fluids to be tested are water at 20°C and the gear oil at 40, 70 and 100°C. If the same fluid should be tested at different temperatures, the tests must be made in an increasing order of the temperatures. Afterwards the measured time must be converted into Engler degrees and finally to cSt (centistoke).

The calculation of the Engler degrees is done using equation 5.1.

$$^{\circ}\text{Engler} = \frac{\text{flow time of 200 ml of lubricant at T temperature}}{\text{flow time of 200 ml of water at 20 }^{\circ}\text{C}} \quad (5.1)$$

The conversion from Engler degrees to centistokes uses equation 5.2.

$$\text{cSt} = k_1 \times ^{\circ}\text{Engler} + \frac{k_2}{^{\circ}\text{Engler} + k_3} \quad (5.2)$$

The values of k_1 , k_2 and k_3 are different for lubricants with Engler degrees below and above 3°E, as shown in Table 5.1.

Table 5.1: Constants of the Engler conversion formula

$^{\circ}\text{Engler}$	k_1	k_2	k_3
<3	14,867	75,568	-6,198
≥3	7,624	-2,717	-1,522

5.2. Density measurement

The densimeter has two different methods to analyze a sample: either through direct suction of a sample of 2 ml from a recipient that contains the lubricant or through the injection of the sample previously collected with an appropriate syringe. Afterwards it indicates the sample temperature (°C) and density (g/cm³). The densimeter should only be used for samples with a temperature between 0 and 100°C, despite that, it only determines accurately the density of a liquid for temperatures below 40°C. [21]



Figure 5.2: Densimeter

Test procedure

- The density must be measured at least at two different temperatures (if possible below 40°C);
- Determine the coefficient of thermal expansion.

Expression 5.3 relates the density measured at different temperatures. [21]

$$\rho = \rho_{ref}[1 - \alpha_t(T_{ref} - T)] \quad (5.3)$$

Where:

ρ – Density at temperature T, [21]

ρ_{ref} – Reference density at reference temperature (usually at 15°C), [21]

α_t – Coefficient of thermal expansion. [21]

Measuring two values of the density at a predefined temperature, allow determining the coefficient of thermal expansion, using equation 5.3, and from there it is possible to determine how the density varies with temperature.

5.3. Four-ball machine

After the oils properties had been measured, it was time to evaluate how the selected oils behave when lubricating a rolling bearing in operation.

There are two major friction sources inside a rolling bearing: the friction occurring in the contact between the rolling elements and the raceways and the friction due to the lubricant flow between the bearing elements (rings, rolling elements and cage). [2]

In order to measure the friction torque inside the rolling bearing and compare the performance of different lubricants, it was necessary to adapt a mechanical bearing housing to the Four-ball Machine integrating a torque cell and several thermocouples, and test each combination of rolling bearing type and lubricant. [14]

5.3.1. Modified Four-ball machine

The rolling bearing tests were performed on a modified Four-Ball Machine, where the four-ball arrangement was replaced by a rolling bearing assembly, as shown in Figure 5.3 for the case of a thrust ball bearing. This new assembly was developed to test different types of rolling bearings lubricated with oil or grease. [14]

The mounting phases for each test are the following.

1. Before starting a test, the Four-Ball Machine should be turned on at least 30 minutes to warm up the transmission system; it should begin with low rotating speeds (speeds below 400 rpm).
2. The lower race (3) is fitted on the spacer (2), with J6/p5 tolerance. This set (3+2) is fitted on the bearing house (1), with H6/j5 tolerance. The tight fit used among these parts of the group A ensures that there is no relative motion between them.
3. The upper race (5) is mounted on the shaft adapter (6), with P5/j6 tolerance, also to prevent relative motion between them, composing the group B.
4. To prevent contamination by external particles resulted from the mounting operations (groups A and B), and also to remove the oil film protection of the bearing package, the groups A and B and the rolling elements and cage (4) are washed with solvent in an ultrasonic bath.

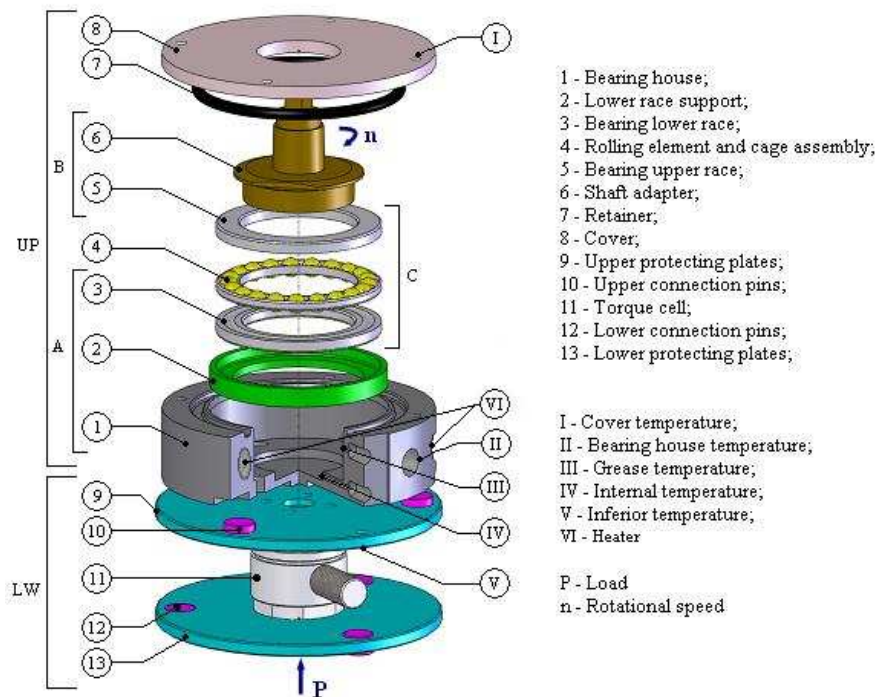


Figure 5.3: Schematic view of the thrust rolling bearing assembly [14]

5. Rolling bearing lubrication: Oil bath lubrication: The oil level should reach the center of the lowest rolling element when the bearing is stationary. For the thrust ball bearing 51107, the oil volume required is approximately 14 ml. During this operation the rolling elements and cage (4) are already positioned on the lower race (3) and the thermocouples (III, IV) should be assembled on the bearing house (1), preventing oil leakage through the thermocouples holes.

6. The set containing the upper race and shaft adapter (6+5) are placed on the rolling elements and cage (4).
7. The retainer (7) is mounted on the cavity of the bearing house (1).
8. Mount the cover (8) and the thermocouples I and II in the assembly.
9. The heater (VI) should also be positioned in the bearing house.
10. The lower set (LW) was previously assembled (and does not need to be reassembled for each test), and is composed by six connection pins (10 and 12) clamped to the torque cell protecting plates (9 and 13) and a torque cell (11). The lower plate (9) is mounted on the lower nonrotating shaft of the Four-Ball Machine, which applies the load to the bearing. The three lower pins (12) assure that there is no relative rotation. The three upper pins (10) are used to connect the upper to the lower set (UP → LW), preventing any relative rotation. The thermocouple V is permanently mounted on the protecting plate (9).
11. The final phase is to install the UP and LW parts in the Four-Ball Machine. Shut off the Four-Ball Machine (turned on in step 1) and connect UP to the rotating shaft, the LW is mounted below. The conjunction will be locked by the lower shaft of the Four-Ball Machine, which is moved up to apply the load.

The bearing assembly permits testing four types of rolling bearings, including thrust ball bearings, tapered roller bearings, angular contact ball bearings and cylindrical roller thrust bearings. The geometrical limitations imposed by the Four-Ball Machine and by the bearing housing, allow a maximum bearing outer diameter of 56.0 mm and a maximum width of 14.3 mm. Table 5.2 shows the different types of rolling bearings that might be tested and the corresponding dimensions and references. Depending on the bearing type, items (2) and (6), shown in Figure 5.3, must be replaced.

Table 5.2: Rolling bearings that are possible to test in the modified Four-Ball machine.

Reference	Dimensions			Dynamic load	Limit speed
	d(mm)	D(mm)	H(mm)	C(kN)	Rpm
Thrust ball bearing					
51103	17	30	9	11,14	12000
51107	35	52	12	19,90	7500
Cylindrical roller thrust bearing					
81102 TN	17	28	9	11,20	8500
81107 TN	35	52	12	29,00	5600
Angular contact ball bearing					
7203	17	40	12	11,00	22000
7204	20	47	14	13,30	18000
Tapped roller bearing					
30302 J2	15	42	14,25	22,40	18000
30203 J2	17	40	13,25	19,00	18000

Operation

In operation (see Figure 5.3), the load (P) is applied on the lower plate (12) and the rotational speed (n) is transmitted to the shaft adapter (6), which is connected to the drive shaft of the machine. The rotating motion is conducted through the upper race (5) to the rolling elements and cage assembly (4). The motion generates the bearing internal friction torque, which is transmitted through the lower race (3) to the bearing house (1), to the upper plate (9) and to the torque cell (11) and they are all clamped together.

During the test, the rolling bearing assembly is submitted to continuous forced air convection by two fans, having 38 mm in diameter and running at 2000 rpm, evacuating part of the heat generated during rolling bearing operation.

Torque cell

In order to preserve the torque cell and to simplify the mounting/dismounting operations, the torque cell is positioned between two circular steel plates (see Figure 5.3).

A piezoelectric torque cell KISTLER® 9339A, whose characteristics are shown in Table 5.3, was selected to measure the bearing internal friction torque. The piezoelectric sensors ensure high accuracy measurements even when the friction torque generated in the bearing is very small compared to the measurement range available.

When a mechanical excitation is applied to the torque cell, the piezoelectric crystals change the electrical current. The current variation is very small and, thus, must be augmented and conditioned using an amplifier KISTLER® 5015A. The output signal is displayed and registered by the virtual instrument running in a computer.

The main restriction of the piezoelectric sensors is the undesirable changes of the output signal, called drift. This phenomenon happens as the result of two variable parameters: the temperature gradient and the measurement time. To avoid the drift effects in the measurements, a specified testing and measuring procedure has been developed.

Table 5.3: Characteristics of the torque cell.

Reaction Torque Sensor — KISTLER® (Type 9339A)		
Measuring range	Nm	-10 to +10
Overload	Nm	-12/+12
Sensitivity	pC/Nm	≈-460
Tensile/compression force, max.	kN	-5/+12
Side force, max.	kN	1,5
Bending moment	Nm	15
Operating temperatures	°C	-40 to +120

Thermocouples

Seven K type thermocouples, with a measurement range between -40°C and 200°C and a sensibility of $41\ \mu\text{V}\ ^{\circ}\text{C}^{-1}$, are used to monitor the bearing operating temperatures. All thermocouples are positioned in strategic locations in order to measure the lubricant and bearing housing temperatures, so that the lubricant viscosity and the heat evacuated through the bearing housing can be calculated with reasonable accuracy. Two of these thermocouples (VI and VII) are used to record the temperatures of the air flow surrounding the bearing house and the room temperature, respectively.

Software

The developed virtual instrument was based on a LabView® platform to operate, to monitor and to control the test system. This software is installed in a Pentium 4 with 2.8 GHz and 1 GB of RAM. The user interface is shown in Figure 5.4.

Figure 5.4: Interface – The initial command window.

5.2. Torque measurement test procedure

The test procedure is constrained by several factors, in particular, the operating limits of the Four-Ball Machine and the torque cell characteristics. The operating conditions imposed by the Four-Ball Machine allow tests with an axial load up to 7000 N and a rotational speed below 1500 rpm.

The drift effect from the torque cell, described before, requires short periods of time (120 s) under stabilized temperatures ($\pm 2^\circ\text{C}$) to make the torque measurements.

After a visual inspection of the assembly in the Four-Ball Machine, the machine can be turned on.



Figure 5.5: Interface – The temperature history

1. Before the actual start of the test a running-in period of 10 minutes, using low loads and speeds, should be performed to “accommodate” the lubricant and the rolling elements.
2. After the running-in period, the machine is stopped and the desired load is applied (for example 7000 N) and the rotational speed set slowly to the required value (300 rpm, for example); the fans are turned on to submit the rolling bearing assembly to continuous forced air convection. The heater is active to increase and maintain a constant operating temperature at a desired value (80°C in this case).
3. Turn on the machine and run the software to start the data acquisition. The operating temperature rises continuously until stabilization at the selected temperature is reached.
4. When the temperatures are stabilized, the torque measuring can begin with two possible methods:
 - a. The machine is turned off and, after the shaft is stationary, immediately restarted again together with the torque measurement.
 - b. Start the torque measurement while the machine is still on and, after a short period of time, turn off the machine and measure the torque (it will require adjustment of the initial value to zero, due to torque cell deviations).
5. The friction torque should be measured during 120 s (± 5 s), and the measurements should be saved along with all the temperatures registered by the thermocouples.
6. After the torque measurement, maintain the rotational speed on and wait until the temperature stabilizes again.

7. Stages 4, 5 and 6 are repeated until three measurements are made of the friction torque in the same conditions.

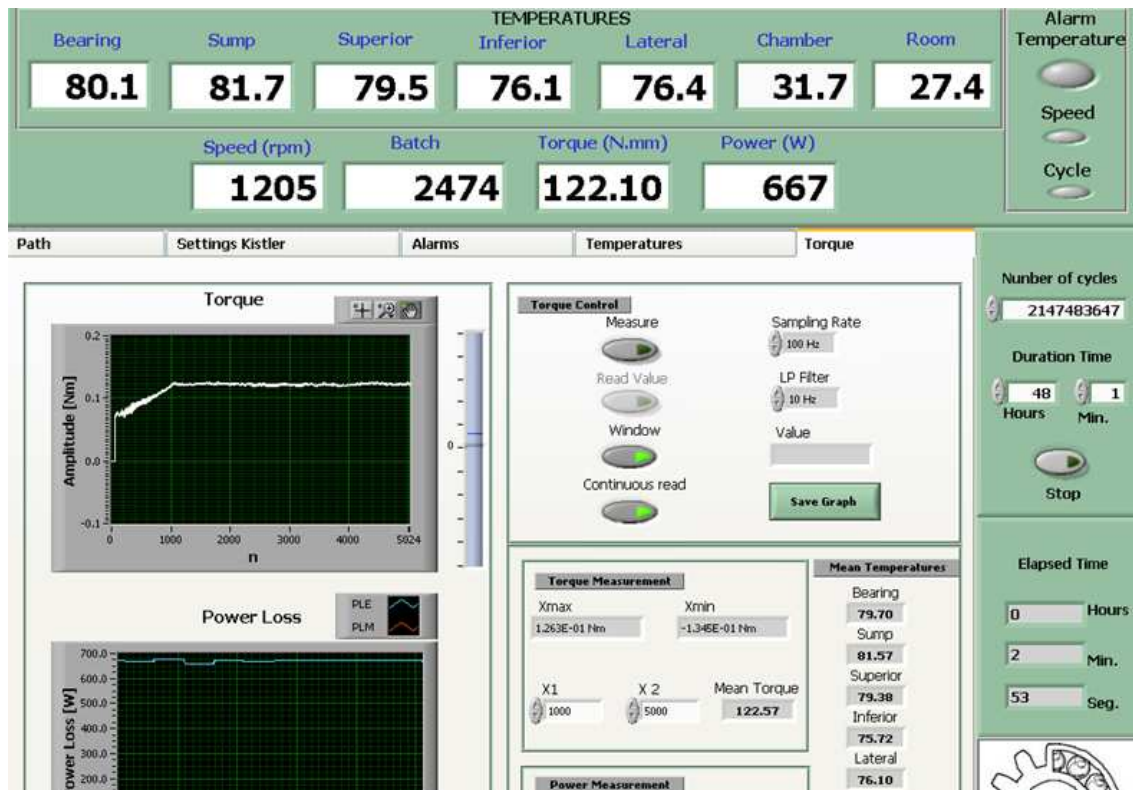


Figure 5.6: Interface – Panel of measuring the bearing torque.

To measure the friction torque for other rotational speeds, the procedures described above should be repeated for each desired rotational speed. One extra procedure is taken when the friction torque is to be known at different rotational speeds: the tests should be always conducted from the lowest to the highest rotational speeds.

The friction torque value (for each rotational speed and load) is the average value of the three measurements taken during the period of 120 s, between second 30 and second 90. This is because in the first 30 s, there is a transition from the starting torque to the operating friction torque, and in the last 30 s, sometimes a slight drift effect can be noticed.

When using piezoelectric torque cells, the torque should be measured during short periods of time at a stabilized temperature, as in the procedure implemented. In this way, the differences between the measured values, for the same operating conditions, are very small.

5.3. Volume of oil

When using oils the usual method of lubrication is oil bath. Considering the type of lubrication the oil level should reach the centre of the rolling element that occupies the lowest position of the bearing, when stationary. [4] For the bearings tested in this work, 14 ml of oil were used in the torque measuring experiments.

6. Friction

6.1. Introduction

Friction is the resistance to motion experienced when a solid slides over another. The resistive force, parallel to the direction of motion is called friction force. If the bodies are loaded together then the tangential force required to initiate sliding is called static friction force. The tangential force needed to maintain the sliding is the dynamic (or kinematic) friction force, which is usually lower than the static friction force. [19]

In the scientific literature two basic laws of friction are usually proposed. The first states that the friction is independent of the apparent area of contact, and the second, that the frictional force is proportional to the normal load between the bodies. [19] These laws were observed experimentally although there are exceptions.

6.2. Possible causes of friction

Friction occurs due to some interaction between the opposing surfaces and that this results in resistance to relative motion. As the surfaces move relative to each other, the work is done by forces which cause the relative motion; there is an energy loss at the contacting surfaces. In considering the possible causes of friction it is convenient to consider separately the surface interaction and the mechanism of energy loss. [19]

6.2.1. Surface interactions

When two surfaces are loaded together they can adhere over some part of the contact, this adhesion is one form of surface interaction causing friction.

If no adhesion occurs, then the other interaction which results in a resistance to motion, is one in which the material must be deformed and displaced to accommodate the relative motion. It will only be referred two interactions of this type. The first is asperity interlocking. Considering Figure 6.1, it can be seen that relative motion cannot happen between the two surfaces without displacement of material of the asperities.

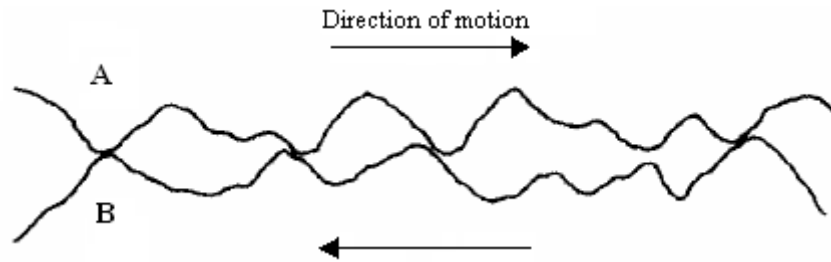


Figure 6.1: Asperity interlocking. [19]

Another example of a displacement type of interaction is seen in Figure 6.2. In this figure it can be seen that a hard sphere pushed against a soft flat surface. In order for relative motion to occur some material of the flat surface must be displaced. On this situation the material displacement on a microscopic scale will be small when compared to the “macro-displacement”. So we only have two types of interaction: adhesion and material displacement and material displacement can also be divided into: asperity locking and macro-displacement.



Figure 6.2: Macro-displacement. [19]

6.2.2. Types of energy loss

There are several factors that can cause energy loss at the interacting surfaces [19], but on this paper only three will be mentioned. As relative motion takes place, material will be deformed. The deformation can be elastic or plastic, material fracture may also occur. Plastic deformation is always accompanied by a loss of energy, this energy loss accounts for the majority of friction of metals under most circumstances. Elastic deformation also requires energy although most of the energy is recoverable, so elastic energy losses are negligible when compared with the ones associated to plastic deformation. Fractures happen when surface interactions are adhesive and can also take place due to relative motion of interlocking asperities. The formation of wear debris is proof that a fracture has happened. However energy losses related to fractures will in most cases be small when compared to the ones related to plastic deformation. A reason for this is that a wear particle is not formed at each asperity contact. Under normal working conditions, an asperity can perform several contacts before the formation of a wear particle.

6.3. Friction Torque Model

Friction between bodies in contact in relative motion, with rolling and sliding, is responsible for the power loss in mechanical components such as rolling bearings. Friction is the resistance encountered when two or more solid surfaces tend to slide between them. [16]

The internal friction in a rolling bearing is a very important factor, influencing heat generation and consequently, the operating temperature. It also affects the bearing performance, its speed limit and its damage mechanisms. [2]

Thus, the evaluation of the internal friction torque in rolling bearings generated by different lubricants has grown systematically over the last few decades.

6.3.1. Friction torque in rolling bearings

The models for determining the friction torque in rolling bearings have improved significantly in the last decade. Recently, SKF has developed and proposed a new model for calculating the internal friction torque in rolling bearings lubricated with oil or grease. [1] An interesting characteristic of this model is that it considers separately the four physical sources of friction in rolling bearings:

- Rolling,
- Sliding,
- Seals (if present),
- Lubricant drag.

Taking into account these four sources it is possible to achieve a better understanding of what happens with the bearing during operation, helping to save energy and improving bearing performance.

6.3.1.1. Total friction torque – M_t

The total rolling bearing internal friction torque is given by equation 6.1.

$$M_t = \phi_{ish} \times \phi_{rs} \times M_{rr} + M_{sl} + M_{drag} + M_{seal} \quad (6.1)$$

The model developed by SKF to evaluate the friction torque in rolling bearings has the following restrictions:

- Grease lubrication or normal methods of oil lubrication;
- For pairs of bearings, calculate the internal friction torque for each bearing and then add the two. In this case, the radial load is divided equally;
- Loads above the recommended minimum load;
- Loads constant in magnitude and direction;
- Normal operating conditions.

6.3.1.2. Rolling friction torque – M_{rr}

It does not matter if the bearing is lubricated or not, the losses generated by rolling friction will always be there in a rolling contact.

There are several sources that contribute to the rolling friction: one is the energy required to introduce the lubricant in the contact and expel the excess, the elastohydrodynamic lubrication process, another is the energy dissipated in the process of elastic deformation that takes place in the contact.

On the other hand, pure rolling is a theoretical idealization and, all rolling contacts contain micro-slips caused by the deformation of surfaces, which in this model are named as sliding friction.

To account for the rolling friction torque, designated as M_{rr} (see equation 6.2), load distribution on each rolling element must be established. This depends on the load applied to the bearing (axial, F_a , and radial, F_r , forces), on the bearing geometry (type, size and number of rolling elements). The contributions of all rolling elements are added together.

This influence is accounted in the model of SKF by the variable G_{rr} (see equation 6.3 and 6.4). The data concerning the type of the bearing is expressed by the variable R_1 , and the influence of the lubricant is accounted for by two other parameters. The viscosity at operating temperature (ν) and the operating speed (n).

$$M_{rr} = G_{rr} \times (\nu \cdot n)^{0,6} \quad (6.2)$$

For thrust ball bearings, (TBB 51107) G_{rr} , is determined using equation 6.3.

$$G_{rr} = R_1 \times d_m^{1,83} \times F_a^{0,54} \quad (6.3)$$

For thrust roller bearings, (RTB 81107 TN) G_{rr} , is determined using equation 6.4.

$$G_{rr} = R_1 \times d_m^{2,38} \times F_a^{0,31} \quad (6.4)$$

The values of R_1 and d_m for the bearings tested are given in the Table 6.1.

Table 6.1: Geometric constants for rolling friction torque. [2]

Parameters	TBB – 51107	RTB – 81107 TN
R_1	1.03E-06	2.25E-06
d_m	43.5	43.5

The rolling friction torque is also affected by two other factors: inlet shear heating (ϕ_{ish} factor of energy dissipation by shear deformation in contact inlet) and kinematic replenishment / starvation (ϕ_{rs} feeding factor of the contact). [2]

Energy dissipation in contact inlet – inlet shear heating – ϕ_{ish}

In the generation of the lubricant film only a small portion of the lubricant available in the contact inlet is dragged into the high pressure zone. The lubricant that does not enter into the contact creates a reverse flow between the ball and the raceway, as shown in Figure 6.3, causing energy dissipation and a corresponding heat flow, the increase of temperature and consequently a decrease of lubricant viscosity.

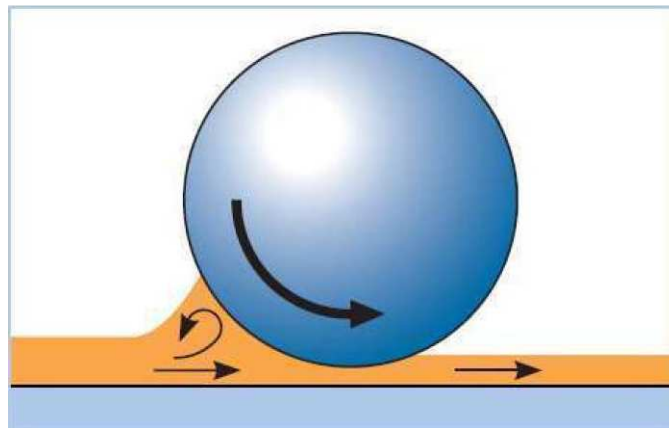


Figure 6.3: Backflow of the lubricant in the contact inlet. [2]

Equation 6.5 shows the variation of the inlet shear heating factor (ϕ_{ish}) with the rotational speed, the average diameter and the operating viscosity of the rolling bearing.

$$\phi_{ish} = \frac{1}{1 + 1,84 \times 10^{-9} \times (n \times d_m)^{1,28} \times v^{0,64}} \quad (6.5)$$

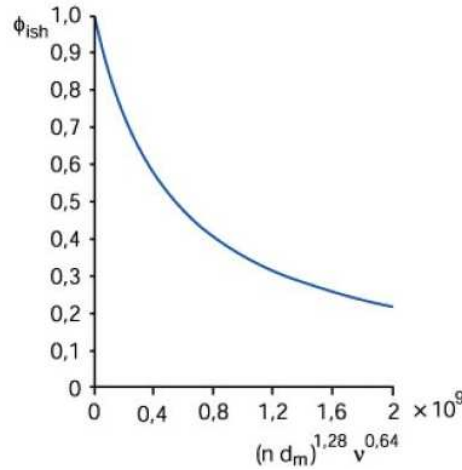


Figure 6.4: Inlet shear heating factor [2]

Feeding the contact – kinematic replenishment / starvation – ϕ_{rs}

At high speeds or with high operating viscosities the lubricant does not have time to flow back into the centre of the contact track after a rolling element has passed. This lack of lubrication, designated as contact starvation, generate a reduction of the volume of lubricant available in the contact, decreasing the film thickness and consequently the rolling resistance.

This effect is accounted for in the SKF model, through the replenishment factor defined by equation 6.6.

$$\phi_{rs} = \frac{1}{e^{K_{rs} \times v \times \eta \times (d+D) \times \sqrt{\frac{K_z}{2(D-d)}}}} \quad (6.6)$$

Where the parameter (K_{rs}) is a constant related to the type of lubricant and lubrication system, named by replenishment / starvation parameter and the other parameter (K_z) is a constant related to the geometry of the bearing. Their values are defined in Table 6.2.

Table 6.2: Lubricant and geometric constants for TBB and RTB. [2]

Parameters	TBB – 51107	RTB – 81107 TN
K_{rs}	3.0e-08	3.0e-08
K_z	3.8	4.4

6.3.1.3. Sliding friction torque – M_{sl}

The sliding friction torque is determined through equation 6.7.

$$M_{sl} = G_{sl} \times \mu_{sl} \quad (6.7)$$

Where the G_{sl} represents the influence of load and bearing geometry on the sliding friction torque.

For thrust ball bearings G_{sl} is given by equation 6.8.

$$G_{sl} = S_1 \times d_m^{0,05} \times F_a^{4/3} \quad (6.8)$$

And for thrust roller bearings G_{sl} is given by 6.9.

$$G_{sl} = S_1 \times d_m^{0,62} \times F_a \quad (6.9)$$

The value of S_1 is given in Table 6.3 for both types of rolling bearings.

Table 6.3: Geometric constant S_1 [2]

Parameter	TBB – 51107	RTB – 81107 TN
S_1	1.60e-02	1.54e-01

Sliding friction is always present in rolling / sliding contact. The sliding coefficient of friction (μ_{sl}) has two components dependent on the lubrication regime, full film lubrication (μ_{EHD}) and boundary film lubrication (μ_{bl}) as given by equation 6.10.

$$\mu_{sl} = \phi_{bl} \times \mu_{bl} + (1 - \phi_{bl}) \times \mu_{EHD} \quad (6.10)$$

The sliding coefficient of friction can be considered as the coefficient of friction in mixed film lubrication, and depend on the load share function (ϕ_{bl}), which establishes the tangential load (F_{Tehd}) supported by the lubricant film (in full film lubrication) and the tangential load (F_{Tbl}) supported by the metal-to-metal contact (in boundary film lubrication), so that the total tangential load (F_T) is obtained by the sum of the two components ($F_T = F_{Tehd} + F_{Tbl}$). In Table 6.4 relevant values for the friction coefficient in boundary lubrication are given for gear oils. [20]

Table 6.4: Friction coefficient in boundary lubrication μ_{bl} [20]

	ESTF	ESTR	MINE	MINR	PAOR	PAGD
μ_{bl}	0.11	0.11	0.09	0.09	0.10	0.08

The coefficient of friction in boundary film lubrication μ_{bl} is strongly influenced by the additives contained in the lubricant, which react with metal surfaces and generate protective boundary layers.

The curve that represents the usual behaviour of the rolling bearing friction torque as a function of rotational speed and viscosity, for conditions of mixed or boundary film lubrication ($\Lambda < 2$) is shown in Figure 6.5. [2]

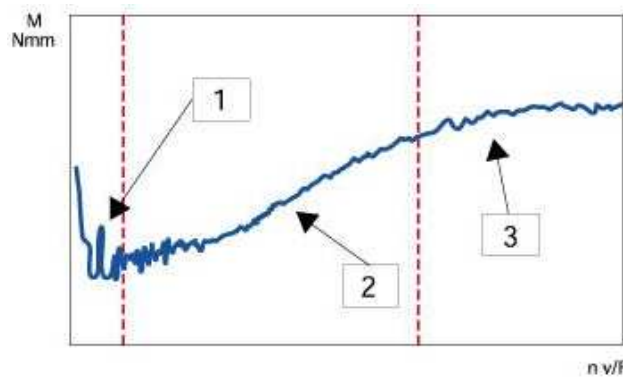


Figure 6.5: Bearing frictional moment as a function of the speed and viscosity. [2]

The curve can be divided in three zones:

- Zone 1 – Mixed film lubrication;
- Zone 2 – EHD full film lubrication;
- Zone 3 – EHD thermal starvation.

In zone 1 mixed film lubrication (or boundary film lubrication) has greater influence, but with increasing speed or viscosity, the friction torque decreases, a lubricant film is generated and the bearing operates in full film lubrication (zone 2). However, for very high speeds or viscosities, the friction torque increases significantly, until thermal effects and lack of lubrication in the contact centre (starvation) start to reduce the friction torque again.

The factor ϕ_{bl} is the weighting factor between the influence of roughness and shear rate of the lubricant inside the contact. The SKF formula for this factor is given by equation 6.11.

$$\phi_{bl} = \frac{1}{e^{2,6 \times 10^{-8} \times (n \times v)^{1,4} \times d_m}} \quad (6.11)$$

The evolution of ϕ_{bl} with the kinematics of the system, the rheological parameters and the geometry of the bearing is represented equations 6.9 and 6.10 and its typical behaviour is shown in Figure 6.6.

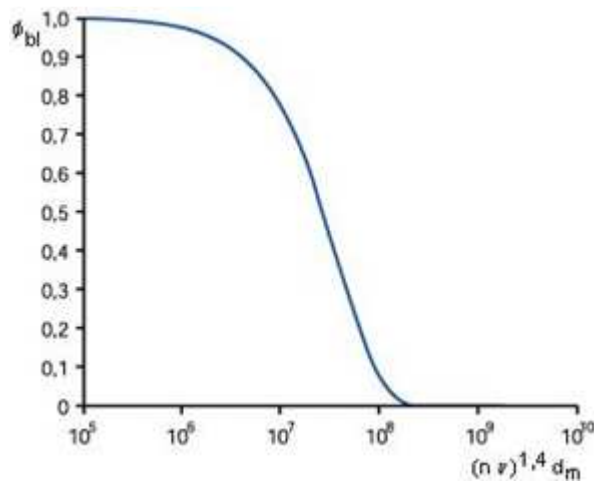


Figure 6.6: Behaviour of weighting factor ϕ_{bl} [2]

During this study and for the same operating conditions, the rolling bearing friction torques measured were significantly lower than those predicted by the SKF model, specially at low operating speed (75 rpm). These discrepancies are related to the size of the rolling bearings tested and the high viscosity of the gear oils tested (ISO VG 320).

Equation 6.11 was adapted to these particular conditions of the tests performed and to the type of bearings tested.

Equation 6.12 was used for thrust ball bearings.

$$\phi_{bl} = \frac{1}{e^{2,6 \times 10^{-8} \times (n \times v)^{1,73} \times dm}} \quad (6.12)$$

And equation 6.13 was used for thrust roller bearings.

$$\phi_{bl} = \frac{1}{e^{2,6 \times 10^{-8} \times (n \times v)^{1,77} \times dm}} \quad (6.13)$$

The exponent value 1.4 was increased to 1.73 for the thrust ball bearing and to 1.77 for the thrust roller bearing. For these values the sliding (μ_{sl}) and the full film (μ_{EHD}) coefficients of friction decreased when the speed increased and when the temperature increased, which was not observed with the original equation (6.11).

In practice it also means that at very low speed the rolling bearings operated in mixed film lubrication and not in boundary lubrication, as predicted by equation 6.11.

6.3.1.4. Friction torque of drag losses – M_{drag}

The friction drag torque is only considered in oil bath lubrication, where the oil level is used to determine the drag torque.

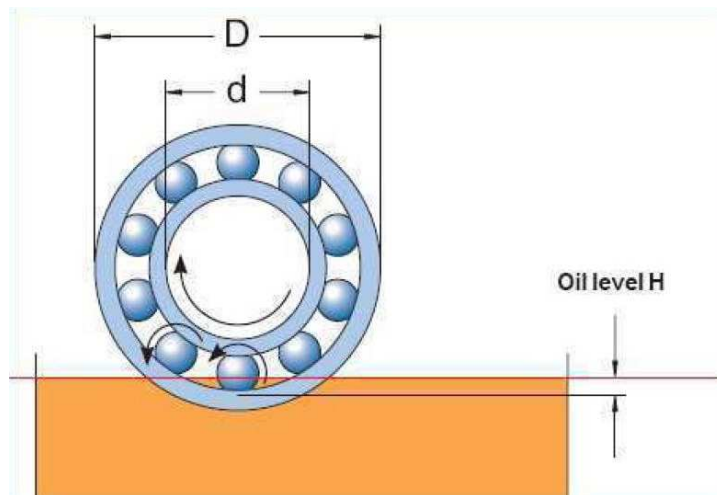


Figure 6.7: Oil level

The model for determining the torque generated by lubricant dragging has restrictions for very large bearings, high speeds and high oil levels, restrictions that are not applicable to the tests performed.

This component of the friction torque may become very important, for completely submerged rolling bearings, where the size and geometry of the oil reservoir, thus the amount of oil used, can have significant impact on the total friction torque.

However, in this work only small bearings were used ($D_{max} = 56mm$) and the oil volume inside the bearing house was also very small (14 ml). Thus the drag is practically insignificant in the total friction torque recorded.

The drag friction torque is defined by equation 6.14.

$$M_{drag} = V_M \times K \times d_m^5 \times n^2 \quad (6.14)$$

For thrust roller bearings the parameter K is given by expression 6.15.

$$K_{roll} = \frac{K_l \times K_z \times (d+D)}{D-d} \times 10^{-12} \quad (6.15)$$

And for thrust ball bearings the parameter K is defined by equation 6.16.

$$K_{ball} = \frac{i_{rw} \times K_z \times (d+D)}{D-d} \times 10^{-12} \quad (6.16)$$

Where the parameter i_{rw} represents the number of balls rows, in this case one row. The values for the geometrical constants are obtained from Table 6.5.

Table 6.5: Geometry constants K_l and K_z [4]

Parameters	TBB – 51107	RTB – 81107 TN
K_l	-	0.43
K_z	3.8	4.4

The drag friction torque is dependent on two parameters V_M and K. The first refers to the ratio between the level of oil and the bearing size, the second is a constant that relates the type and the geometry of the bearing.

6.3.1.5. Friction torque of seals – M_{seal}

The friction torque generated between the seal lip and the moving counter face (often steel), may represent a large percentage of the total friction torque in rolling bearing. The calculation methods for determining the friction torque generated by the seals will not be covered in detail, since the tests were performed with unsealed bearings.

6.3.1.6. Determination of sliding friction coefficient

Since the experimented rolling bearings had no seals this component was removed from the equation 6.1, which can be rewritten as:

$$M_t = \phi_{ish} \times \phi_{rs} \times M_{rr} + M_{sl} + M_{drag} \quad (6.17)$$

Through interconnection with the experimental friction torque measurements ($M_{total} = M_{exp}$) it is possible to determine the sliding friction torque (M_{sl}), using expression 6.18.

$$M_{sl} = M_{exp} - \phi_{ish} \times \phi_{rs} \times M_{rr} - M_{drag} \quad (6.18)$$

Making it also possible to evaluate the sliding coefficient of friction (μ_{sl}) by combining equations 6.7 and 6.18 creating the expression 6.19.

$$\mu_{sl} = \frac{M_{exp} - \phi_{ish} \times \phi_{rs} \times M_{rr} - M_{drag}}{G_{sl}} \quad (6.19)$$

7. Experimental Results

7.1. Testing conditions

Thrust ball bearings (TBB 51107) and thrust roller bearings (RTB 81107 TN) were tested using six different gear oils typically used in wind turbine gearboxes. The tests were all performed at 80°C. The other operating conditions are present on Table 7.1.

Table 7.1: Operating conditions.

Load [N]	Speed [rpm]	Lubricants
700 7000	75	ESTF
	150	ESTR
	300	MINE
	600	MINR
	900	PAGD
	1200	PAOR

7.2. Contact parameters

An EXCEL spread sheet was used to calculate all the parameters related to elliptical and linear contacts: contact pressure, contact dimensions, shear stress, among others. Table 7.2 shows the radius of curvature of the contacting surfaces (raceways and rolling elements) inside the bearings, where x is the rolling direction. Table 7.3 shows the contact parameters for each contact element.

Table 7.2: Curvature dimensions (x – rolling direction).

	Thrust ball bearing		Thrust roller bearing	
	Raceways	Balls	Raceways	Rollers
R_x [m]	∞	3.00E-03	∞	2.50E-03
R_y [m]	-3.38E-03	3.00E-03	∞	∞

Table 7.3: Contact parameters (x – rolling direction).

Number of contact Elements	Thrust ball bearing	Thrust roller bearing
	21	20
R_x [m]	6.00E-03	5.00E-03
R_y [m]	5.34E-02	—
p_0 (7000N) [Pa]	2.48E+9	1.31E+9
p_0 (700N) [Pa]	1.15E+9	0.41E+9

7.3. Theoretical lubricant film thickness

The lubricant film thickness inside the contact was determined, considering a constant operating temperature of 80°C used in the experimental tests, the geometry and kinematics of the bearings tested. The lubricant parameters, previously determined according to the ASTM D341 standard, were used to calculate the viscosities of the gear oils at the operating temperature. The viscosities are presented in Figure 7.1.

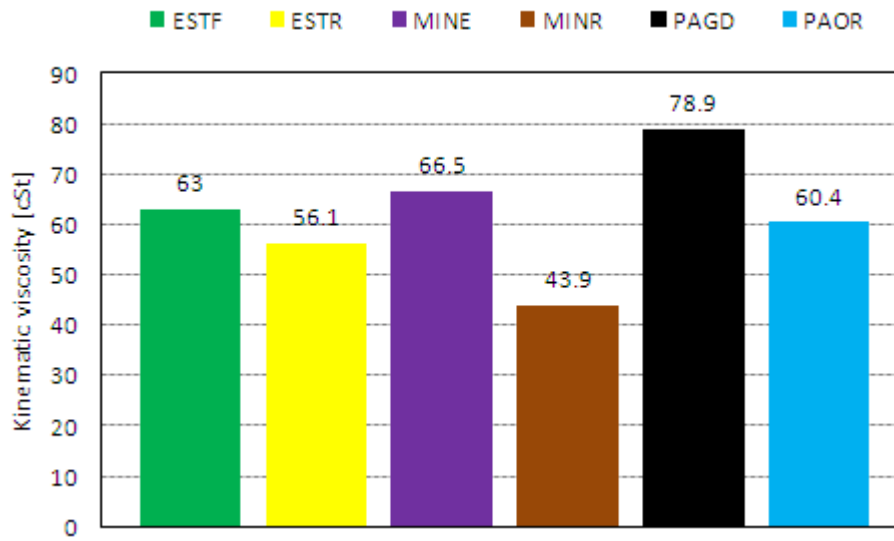


Figure 7.1: Kinematic viscosities of the gear oils at 80°C

Table 7.4: Lubricant parameters

	ESTF	ESTR	MINE	MINR	PAGD	PAOR
VI	153	140	163	85	230	150
μ_{bl}	0.11	0.11	0.09	0.09	0.10	0.08
α (80°C)	1.16E-8	1.14E-8	1.29E-8	1.68E-8	1.05E-8	1.28E-8

The Hamrock and Dowson equation [6] and the Dowson and Higginson equation [6] (see section 3.3) were used to determine the centre film thickness (h_0) in the elliptical and linear contacts present in the thrust ball bearing and thrust roller bearings, respectively. Tables 7.5 to 7.8 present the specific film thickness (Λ) of the ball and roller thrust bearings for all the operating conditions tested.

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Table 7.5: Specific lubricant film thickness in TBB 51107 – axial load 7000N.

Speed	Λ [-]					
	ESTF	ESTR	MINE	MINR	PAGD	PAOR
75	0.1743	0.1552	0.1825	0.1599	0.2058	0.1641
150	0.2774	0.2468	0.2884	0.2526	0.3264	0.2609
300	0.4394	0.3863	0.4556	0.3972	0.5089	0.4115
600	0.6805	0.6087	0.6925	0.6224	0.7963	0.6455
900	0.8608	0.7767	0.9046	0.7983	1.0119	0.8214
1200	1.0066	0.9114	1.0656	0.9586	1,1716	0,9870

Table 7.6: Specific lubricant film thickness in TBB 51107 – axial load 700N.

Speed	Λ [-]					
	ESTF	ESTR	MINE	MINR	PAGD	PAOR
75	0.2005	0.1797	0.2104	0.1859	0.2387	0.1920
150	0.3154	0.2885	0.3354	0.2944	0.3766	0.3017
300	0.5036	0.4500	0.5319	0.4620	0.5932	0.4760
600	0.7888	0.7004	0.8305	0.7249	0.9170	0.7452
900	0.9810	0.8983	1.0650	0.9314	1.1782	0.9555
1200	1.1593	1.0622	1.2590	1.1025	1.3681	1.1478

Table 7.7: Specific lubricant film thickness in RTB 81107 TN – axial load 7000N.

Speed	Λ [-]					
	ESTF	ESTR	MINE	MINR	PAGD	PAOR
75	0.2049	0.1815	0.2200	0.1966	0.2404	0.1967
150	0.3392	0.2952	0.3640	0.3256	0.3936	0.3243
300	0.5569	0.4854	0.5945	0.5346	0.6441	0.5320
600	0.8880	0.7865	0.9667	0.8661	1.0417	0.8618
900	1.1635	1.0306	1.2588	1.1381	1.3478	1.1294
1200	1.3744	1.2496	1.3533	0.9987	1.5192	1.3562

Table 7.8: Specific lubricant film thickness in RTB 81107 TN – axial load 700N.

Speed	Λ [-]					
	ESTF	ESTR	MINE	MINR	PAGD	PAOR
75	0.2498	0.2208	0.2686	0.2404	0.2977	0.2436
150	0.4126	0.3670	0.4447	0.3943	0.4887	0.4030
300	0.6773	0.6038	0.7289	0.6513	0.8021	0.6531
600	1.0979	0.9785	1.1897	1.0654	1.2864	1.0592
900	1.4423	1.2700	1.5488	1.4011	1.6678	1.3883
1200	1.7180	1.5164	1.8624	1.6826	1.9754	1.6804

It's observed that the RTB always generated higher values than the TBB. The PAGD oil generated the highest values because of its high viscosity and the ESTR generated the lowest because of its low coefficient of piezoviscosity, except on a 7000N loaded RTB at 1200 rpm when the MINR oil reached a higher operating temperature causing a great decrease on the its viscosity which lead to a lower specific lubricant film thickness.

7.4. Friction Torque obtained from the torque cell

The rolling bearings were tested with the mentioned procedure (see section 5.4) and the measurements of the rolling bearing friction torque and of the operating temperature were made using the torque cell and thermocouples (see section 5.3). Tables 7.9 to 7.12 and Figures 7.2 to 7.5 shows the rolling bearings friction torque measurements obtained for the different oils, at a constant temperature of 80°C, for all the operating conditions selected.

Table 7.9: Experimental friction torque measured for TBB 51107 – axial load 7000N.

Speed	Mt [N.mm]					
	ESTF	ESTR	MINE	MINR	PAGD	PAOR
75	142.47	155.35	110.71	151.78	154.87	140.22
150	165.42	164.97	118.53	167.78	157.49	143.35
300	179.14	163.68	132.17	189.45	162.51	162.18
600	182.63	172.53	145.96	196.59	186.27	176.84
900	200.35	180.13	152.90	206.90	222.17	188.61
1200	189.69	192.97	166.25	217.65	223.69	180.43

Table 7.10: Experimental friction torque measured for TBB 51107 – axial load 700N.

Speed	Mt [N.mm]					
	ESTF	ESTR	MINE	MINR	PAGD	PAOR
75	25.96	23.11	17.44	36.44	15.08	10.69
150	39.09	28.03	26.84	40.02	21.98	17.50
300	42.57	41.11	32.40	44.12	36.78	23.49
600	56.59	49.46	47.66	53.62	50.00	38.06
900	65.64	59.12	52.60	58.59	60.95	43.82
1200	71.33	54.73	58.53	65.43	69.01	50.56

Table 7.11: Experimental friction torque measured for RTB 81107 TN – axial load 7000N.

Speed	Mt [N.mm]					
	ESTF	ESTR	MINE	MINR	PAGD	PAOR
75	470.85	464.40	437.96	406.20	294.06	478.70
150	388.64	374.24	403.61	426.68	292.33	390.48
300	351.39	344.94	386.73	354.60	270.31	319.39
600	313.88	277.74	314.47	379.62	282.63	292.70
900	256.66	295.19	275.44	378.42	315.48	286.65
1200	245.06	288.49	279.66	334.64	341.03	298.45

Table 7.12: Experimental friction torque measured for RTB 81107 TN – axial load 700N.

Speed	Mt [N.mm]					
	ESTF	ESTR	MINE	MINR	PAGD	PAOR
75	71.37	68.93	63.06	70.82	65.69	65.55
150	75.29	63.71	62.48	70.83	71.11	64.54
300	79.70	74.39	73.19	82.75	90.51	80.30
600	92.41	92.24	102.07	91.30	121.16	107.19
900	108.42	103.95	123.41	112.92	146.75	125.92
1200	127.54	112.61	133.15	122.47	167.00	135.45

Figures 7.2 and 7.3 presents the friction torque measured in the thrust ball bearings (TBB) for different operating speeds, at constant temperature and two axial loads, respectively 7000N and 700N.

In the case of the highest axial load (see Figure 7.2) the experimental friction torque M_t increased when the operating speed increased, however, this increase is more significant at lower speeds ($75\text{rpm} \leq n \leq 300\text{rpm}$) than at higher speeds ($300\text{rpm} \leq n \leq 1200\text{rpm}$). For a given oil, e.g. MINR, the M_t increased from 152 N.mm to 218 N.mm (+43%) when the speed increased from 75 rpm to 1200 rpm. All gear oils exhibited a similar behaviour: MINR oil always generated the highest friction torque while oil MINE generated the lowest. PAGD oil was an exception to this trend, probably due to its very high Viscosity Index ($VI_{\text{PAGD}}=230$).

The friction torque measured with the same TBB at low load (700N) are similar, but the increase of M_t when the speed increases is higher. In the case of MINR oil, M_t increased from 36 N.mm to 65 N.mm (+81%) when the speed increased from 75 rpm to 1200 rpm.

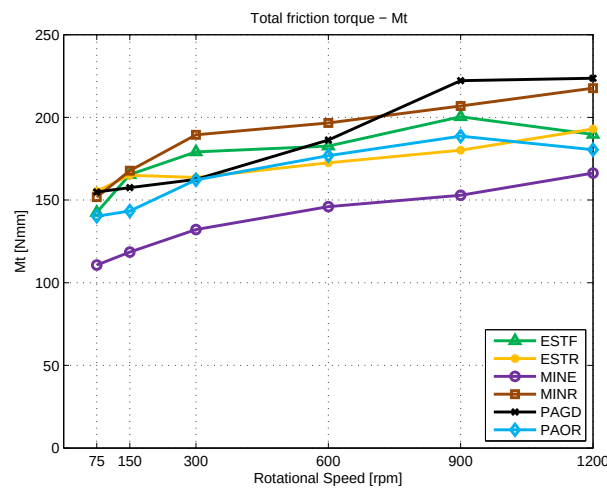


Figure 7.2: Experimental friction torque for thrust ball bearing – axial load 7000N.

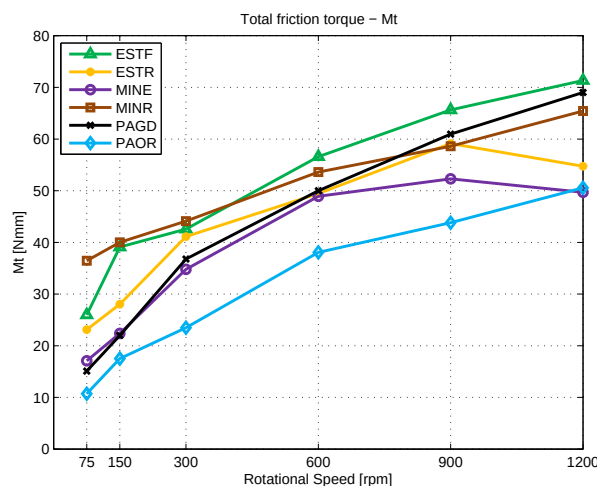


Figure 7.3: Experimental friction torque for thrust ball bearing – axial load 700N.

Figures 7.4 and 7.5 presents the friction torque measured in the thrust roller bearings (RTB) for different operating speeds, at constant temperature and two axial loads, respectively 7000N and 700N.

In the case of the highest axial load (see Figure 7.4) the experimental friction torque M_t decreased when the operating speed increased, however, this decrease is more significant at lower speeds ($75\text{rpm} \leq n \leq 300\text{rpm}$) than at higher speeds ($300\text{rpm} \leq n \leq 1200\text{rpm}$). For a given oil, e.g. PAOR, the M_t decreased from 479 N.mm to 287 N.mm (-40%) when the speed increased from 75 rpm to 900 rpm. Gear oils ESTF, ESTR, MINE and PAOR exhibited a similar behaviour and generated similar friction torques. However, oils MINR and PAGD, showed a different trend, generating increasing friction torques above 300 rpm. This is probably related to these two oils possessing significantly different Viscosity Indexes than the other gear oils ($VI_{\text{PAGD}}=230$, $VI_{\text{MINR}}=85$, $VI_{\text{PAOR}}=150$). At 1200 rpm it wasn't possible to keep the operating temperature constant for some oils only the oils ESTF and ESTR were tested at 80°C .

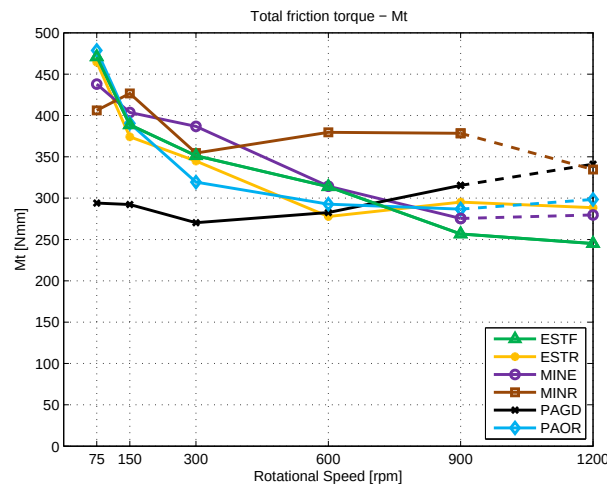


Figure 7.4: Experimental friction torque for thrust roller bearing – axial load 7000N.

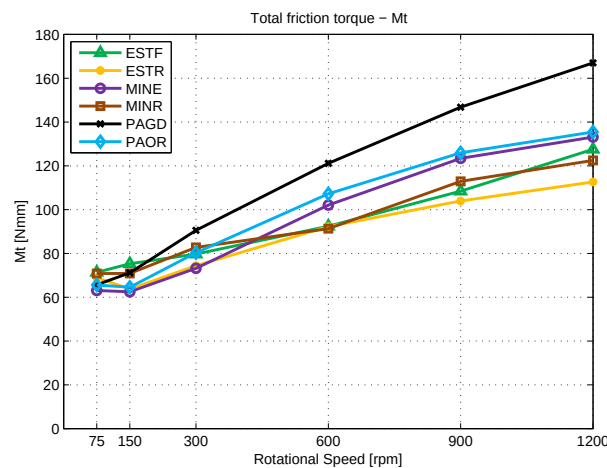


Figure 7.5: Experimental friction torque for thrust roller bearing – axial load 700N.

The friction torque measured with the same RTB at low load (700N) showed the opposite behaviour than at high load (7000N), increasing when the operating speed increased. In the case of the ESTF, M_t increased from 71 N.mm to 127 N.mm (+80%) when the speed increased from 75 to 1200 rpm.

Comparing the friction torques generated by thrust ball bearings and thrust roller bearings, at high load, it's clear that RTB generated significantly higher friction torques than the TBB. In the case of MINE at operating speed of 600 rpm, the corresponding friction torques were $M_{t_{RTB}}=314$ N.mm and $M_{t_{TBB}}=146$ N.mm, that is, the friction torque generated by the RTB is more than double the friction torque produced by the TBB.

7.5. Discussion

7.5.1 Discussion on thrust ball bearings friction torque – axial load 7000 N

The friction torque model, presented in section 6.3, was used in a MATLAB code to predict the values of the rolling (M_{rr}) and sliding (M_{sl}) friction torques, and of the EHD (μ_{EHD}) and sliding (μ_{sl}) coefficients of friction for all testing conditions considered in the thrust ball bearing tests under an axial load of 7000N. The corresponding specific lubricant film thickness was also estimated.

Figure 7.6-a) clearly shows that when the operating speed increases from 75 rpm to 1200 rpm the specific lubricant film thickness inside the TBB increased from 0.1 to 0.90/1.20 depending on the oil tested, meaning that the lubrication regime evolved from boundary to mixed film lubrication. All gear oils exhibited a similar trend, but PAGD oil showed significant higher values of Λ because of its very high Viscosity Index and also its higher viscosity at 80°C.

Figure 7.6.c) shows the rolling torque estimated for the TBB in all operating conditions. As expected, and because the tests were performed at constant temperature (80 °C), when the speed increases the rolling torque also increases ($M_{rr} \propto (n \cdot v)^{0.6}$). This figure also shows that oils ESTF, ESTR, MINE and PAOR generated very similar rolling torques, because they have the same viscosity grade (ISO VG 320) and similar viscosities at 80 °C. PAGD oil produces very high rolling torques because it has the highest VI and the highest viscosity at 80 °C, while MINR oil, on the opposite has the lowest VI and the lowest viscosity at 80 °C.

Figure 7.6.d) shows the sliding torque estimated for the TBB in all operating conditions. The sliding torque is obtained by subtracting the rolling friction torque to the experimental friction torque, that is, $M_{sl} = M_{exp} - \phi_{ish} \cdot \phi_{rs} \cdot M_{rr}$. The increase of the experimental friction torque ($M_{exp} = M_t$, see Figure 7.6.b)) with speed is smaller than the increase of the rolling torque with speed, thus the sliding torque decreases slightly when the speed increases, as shown in Figure 7.6.d). Such behaviour is typical of thrust ball bearings operating under the mixed film lubrication regime and the sliding coefficient of friction μ_{sl} decreased when speed increased, at constant temperature, as presented in Figure 7.6.f).

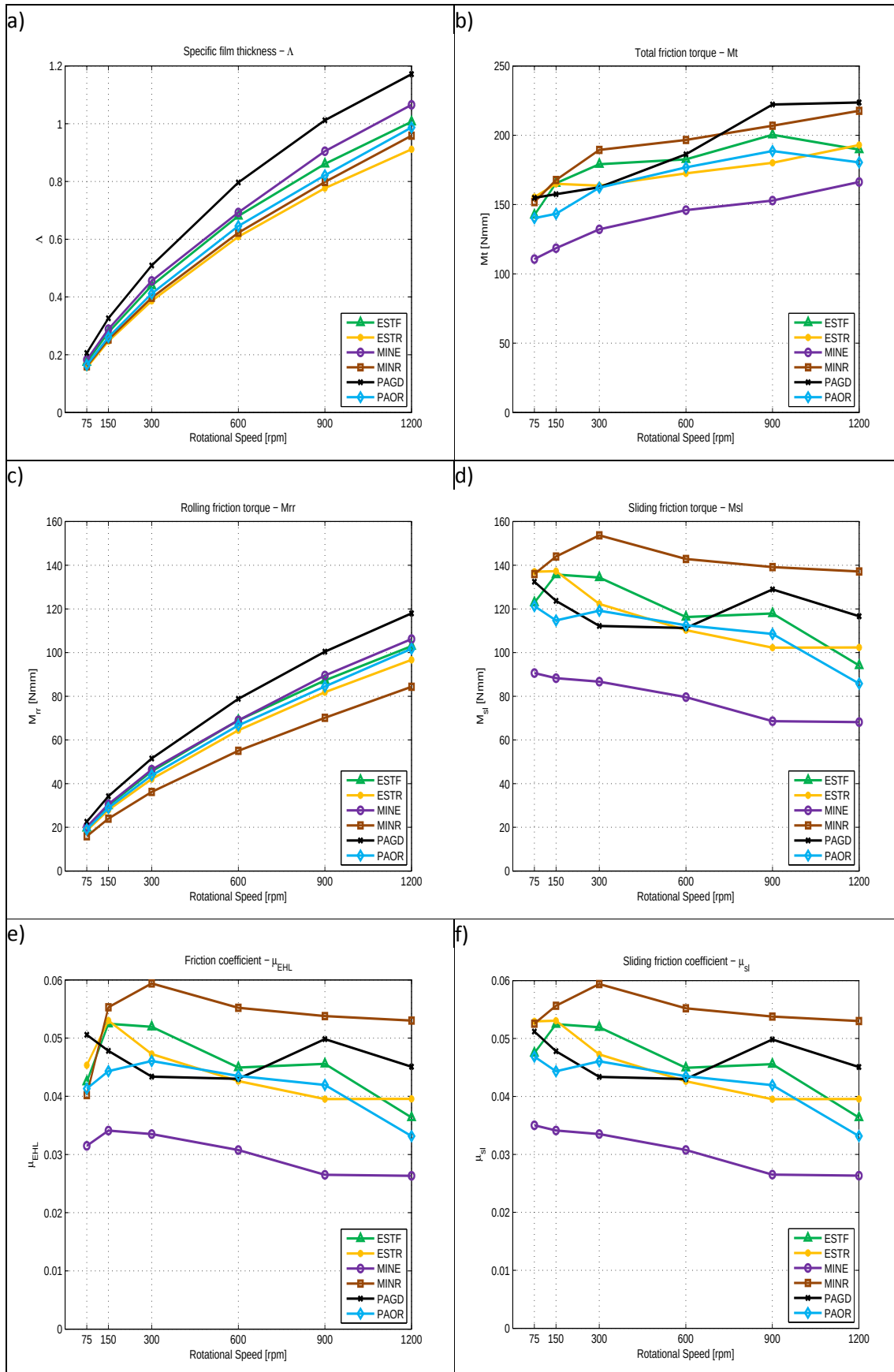


Figure 7.6: Λ , M_t , M_{rr} , M_{sl} , μ_{EHL} and μ_{sl} for TBB 51107 – axial load 7000N.

Comparing the sliding coefficient of friction (μ_{sl}) with the full film coefficient of friction (μ_{EHD}) Figures 7.6.e) and 7.6.f), no significant differences are observed. Only at the lowest speed (75 rpm), at the smallest specific lubricant film thickness ($0.15 \leq \Lambda \leq 0.20$), $\mu_{sl} > \mu_{EHD}$.

Comparing the friction behaviour of the wind turbine gear oils, inside the thrust ball bearings, it is very clear that MINR oil always produced the highest values of the sliding coefficient of friction, of the sliding torque and of the total rolling bearing friction torque, while oil MINE always generated the lowest corresponding values. Oils ESTF, ESTR and PAOR were placed in between the previous two. PAGD oil exhibited similar values to the esters and PAOR at low speed, but it generated the highest friction torques at 900 rpm and above, due to its very high viscosity at 80 °C.

7.5.2. Discussion on thrust roller bearings friction torque – axial load 7000 N

The friction torque model, presented in section 6.3, was used in a MATLAB code to predict the values of the rolling (M_{rr}) and sliding (M_{sl}) friction torques, and of the EHD (μ_{EHD}) and sliding (μ_{sl}) coefficients of friction for all testing conditions considered in the thrust roller bearing tests under an axial load of 7000 N. The corresponding specific lubricant film thickness was also estimated. For the oils MINE, MINR, PAOR and PAGD, tested at 1200 rpm, it was not possible to keep the operating temperature at 80 °C, as RTB reached higher operating temperatures.

Figure 7.7.a) clearly shows that when the operating speed increased from 75 rpm to 900 rpm the specific lubricant film thickness inside the RTB increased from 0.20 / 0.25 to 1.00 / 1.35, depending on the oil tested, meaning that the lubrication regime evolved from boundary to mixed film lubrication. All gear oils exhibited a similar trend, but PAGD oil produced the highest Λ 's, because of its very high Viscosity Index and its significantly higher viscosity at 80 °C, and oil ESTF generated the lowest Λ 's, because of its low piezoviscosity coefficient.

Figure 7.7.c) shows the rolling torque estimated for the RTB in all operating conditions. As expected, and because the tests were performed at constant temperature (80 °C), when the speed increases the rolling torque also increases ($M_{rr} \propto (n \cdot v)^{0.6}$). This figure also shows that oils ESTF, ESTR, MINE and PAOR generated very similar rolling torques, because they have the same viscosity grade (ISO VG 320) and similar viscosities at 80 °C. PAGD oil produces very high rolling torques because it has the highest viscosity at 80 °C, while MINR oil, on the opposite, has the lowest VI and the lowest viscosity at 80 °C.

Figure 7.7.d) shows the sliding torque estimated for the RTB in all operating conditions. The sliding torque is obtained by subtracting the rolling friction torque to the experimental friction torque, that is, $M_{sl} = M_{exp} - \phi_{ish} \cdot \phi_{rs} \cdot M_{rr}$. The experimental friction torque ($M_{exp} = M_{\tau}$, see Figure 7.7.b)) decreased when speed increased, the opposite trend of the rolling torque, and consequently the sliding torque decreases significantly when the speed increases, as shown in Figure 7.7.d). Such behaviour is typical of thrust roller bearings operating under mixed film lubrication, and the sliding coefficient of friction (μ_{sl}) decreased significantly when speed increased, at constant temperature, as presented in Figure 7.7.f).

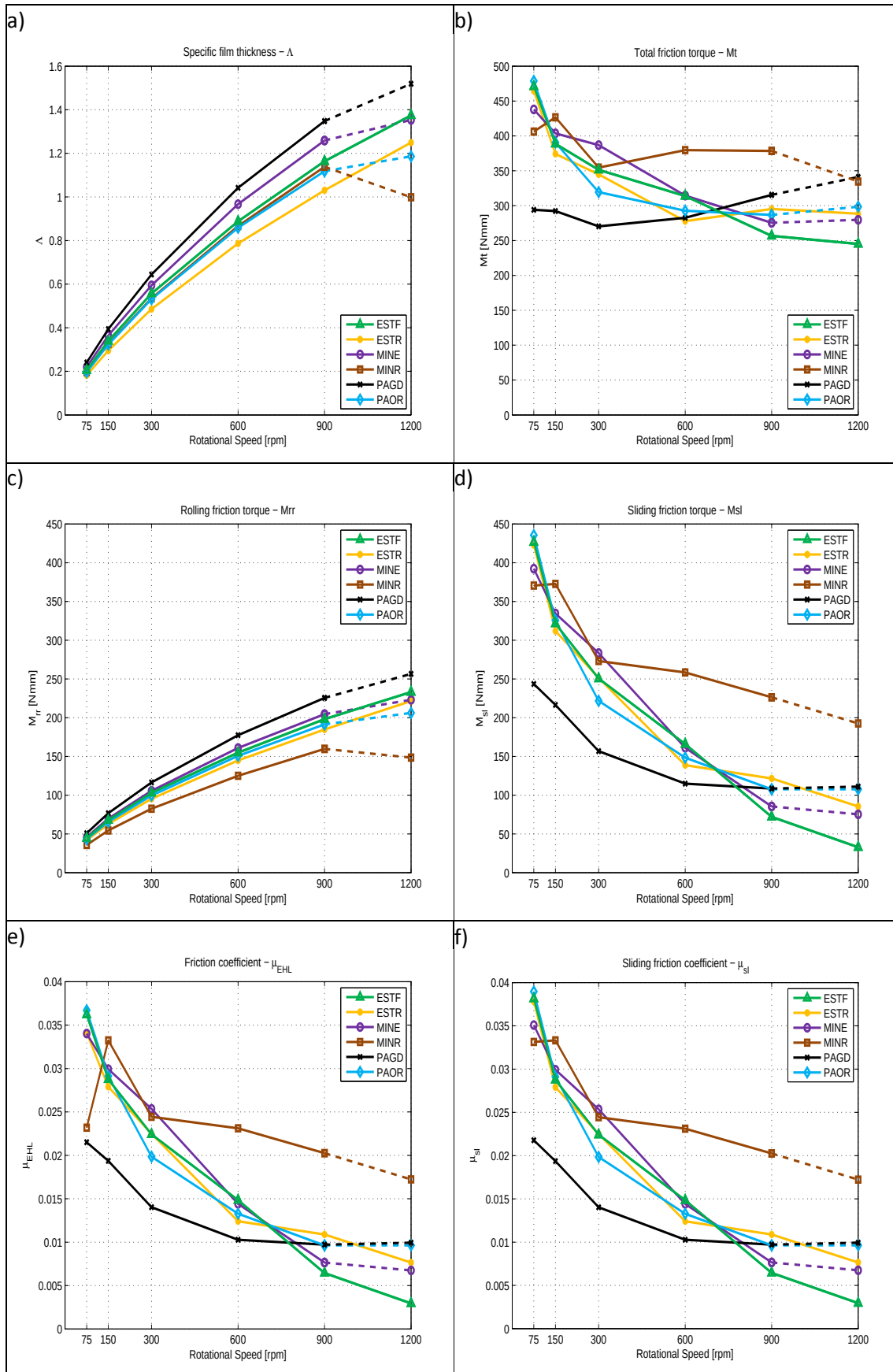


Figure 7.7: Λ , M_t , M_{rr} , M_{sl} , μ_{EHL} and μ_{sl} and for RTB 81107 TN – axial load 7000N.

Comparing the sliding coefficient of friction (μ_{sl}) with the full film coefficient of friction (μ_{EHD}) Figures 7.7.f) and 7.7.e), no significant differences are observed. Only at the lowest speed (75 rpm), at the smallest specific lubricant film thickness ($0.20 \leq \Lambda \leq 0.25$), $\mu_{sl} > \mu_{EHD}$.

Comparing the friction behaviour of the wind turbine gear oils, inside the thrust roller bearings, it is clear that MINR oil always produced the highest values of the sliding coefficient of friction, of the sliding torque and of the total rolling bearing friction torque, while PAGD oil always generated the lowest corresponding values. Oils ESTF, ESTR and PAOR were placed in between the previous two. However, at very low speed (75 rpm) the oils ESTF, ESTR and PAOR, generated the highest sliding coefficient of friction, sliding torque and total rolling bearing friction torque, and at high speed (900 rpm) the performance of all oils (except MINR) was similar.

7.5.3. Discussion on thrust ball bearings friction torque – axial load 700 N

The friction torque model, presented in section 6.3, was used in a MATLAB code to predict the values of the rolling (M_{rr}) and sliding (M_{sl}) friction torques, and of the EHD (μ_{EHD}) and sliding (μ_{sl}) coefficients of friction for all testing conditions considered in the thrust ball bearing tests under an axial load of 700 N. The corresponding specific lubricant film thickness was also estimated.

Figure 7.8.a) clearly shows that when the operating speed increased from 75 rpm to 1200 rpm the specific lubricant film thickness inside the RTB increased from 0.18 / 0.25 to 1.05 / 1.40, depending on the oil tested, meaning that the lubrication regime evolved from boundary to mixed film lubrication. All gear oils exhibited a similar trend, but PAGD oil produced the highest Λ 's, because of its high viscosity at 80 °C, and ESTF oil generated the lowest Λ 's, because of its low piezoviscosity coefficient.

Figure 7.8.c) shows the rolling torque estimated for the RTB in all operating conditions. As expected, and because the tests were performed at constant temperature (80 °C), when the speed increases the rolling torque also increases ($M_{rr} \propto (n \cdot v)^{0.6}$). This figure also shows that the oils ESTF, ESTR, MINE and PAOR generated very similar rolling torques, because they have the same viscosity grade (ISO VG 320) and similar viscosities at 80 °C. PAGD oil produces very high rolling torques because it has the highest viscosity at 80 °C, while MINR oil, on the opposite, has the lowest viscosity at 80 °C.

The experimental friction torque ($M_{exp} = M_T$, see Figure 7.8.b)) as well as the rolling torque (M_{rr} see Figure 7.8.c)) increased when speed increased, but at different rates. The sliding torque, obtained by subtracting the rolling friction torque to the experimental friction torque, that is, $M_{sl} = M_{exp} - \phi_{ish} \cdot \phi_{rs} \cdot M_{rr}$ (see Figure 7.8.d), shows that the sliding torque increases with the speed for all operating conditions. However, at speeds above 600 / 900 rpm ESTR and MINE oils show a decrease of the sliding torque with speed.

The sliding and the full film coefficients of friction (μ_{sl} and μ_{EHD}) show the same trend of the sliding friction torque, as presented in Figures 7.8.e) and 7.8.f).

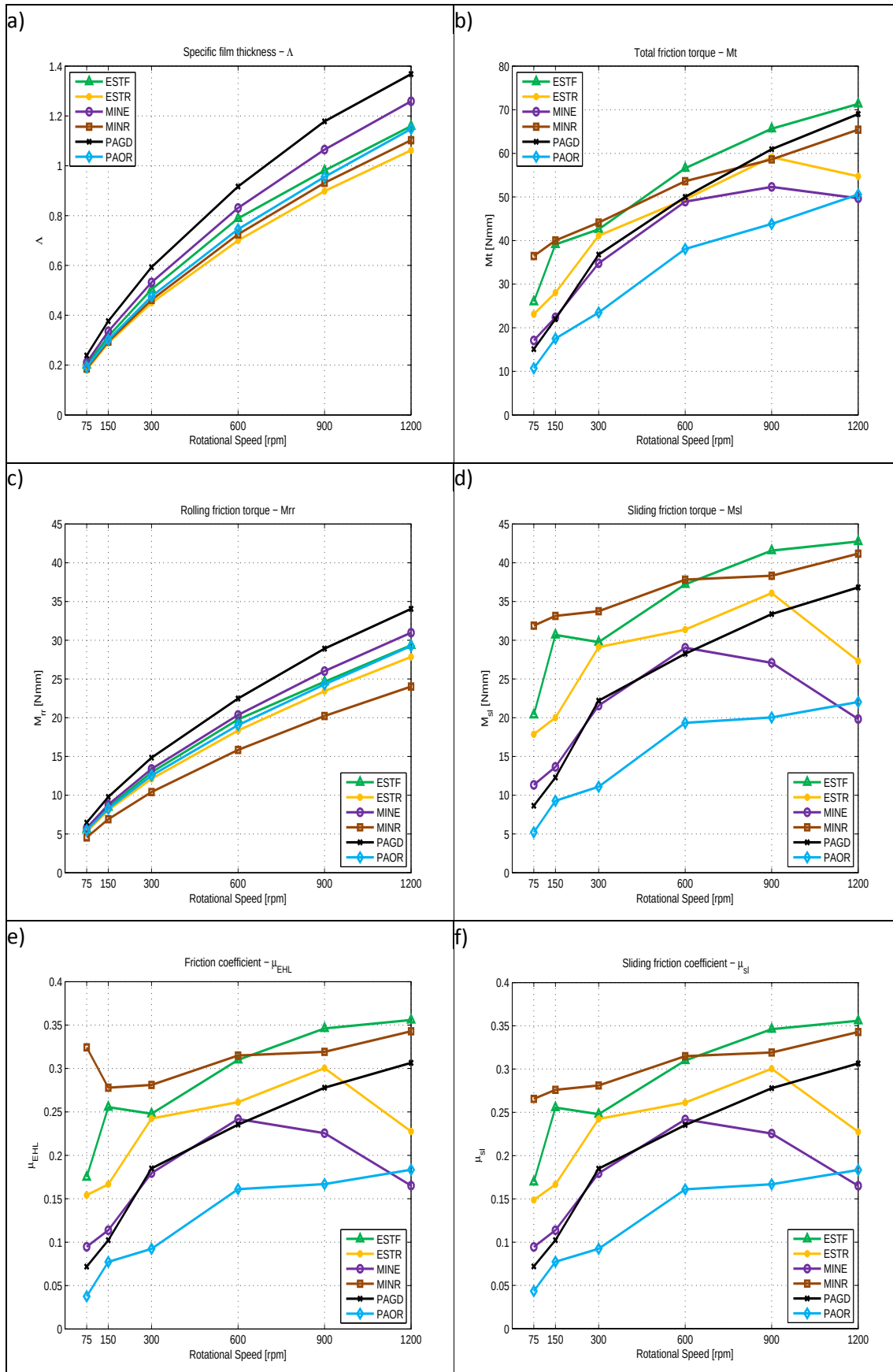


Figure 7.8: Λ , M_t , M_{rr} , M_{sl} , μ_{EHL} and μ_{sl} for TBB 51107 – axial load 700 N.

The sliding and the full film coefficients of friction (μ_{sl} and μ_{EHD}) present an unusual trend, since they increase when the speed increases and, above all, those coefficients of friction are very high, reaching in almost all cases values above 0.1, whatever the oil and speed, reaching sliding friction coefficients above 0.3.

Two reasons might explain this behaviour. The first reason might be related to the type of bearing (ball bearing, elliptical contact) and light loads, which generate these very high friction coefficients. The second reason can be related with the applicability of the rolling bearing friction torque model to lightly loaded rolling bearings.

Nevertheless, the same model, applied to thrust roller bearings, predicted very good results (see section 7.5.4), suggesting it can be applied to lightly loaded roller bearings. In the same line, the results predicted by the model for the thrust ball bearings, using typical coefficients of friction (0.04 – 0.05), clearly underestimate the sliding coefficient of friction. Thus, these results for lightly loaded bearings should be analysed in detail in a future work.

7.5.4. Discussion on thrust roller bearings friction torque – axial load 700 N

The friction torque model, presented in section 6.3, was used in a MATLAB code to predict the values of the rolling (M_{rr}) and sliding (M_{sl}) friction torques, and of the EHD (μ_{EHD}) and sliding (μ_{sl}) coefficients of friction for all testing conditions considered in the thrust roller bearing tests under an axial load of 700 N. The corresponding specific lubricant film thickness was also estimated.

Figure 7.9.a) clearly shows that when the operating speed increased from 75 rpm to 1200 rpm the specific lubricant film thickness inside the RTB increased from 0.25 / 0.30 to 1.50 / 2.00, depending on the oil tested, meaning that the lubrication regime evolved from boundary film to near full film lubrication. All gear oils exhibited a similar trend, but PAGD oil produced the highest Λ , because of its high Viscosity Index and its significantly higher viscosity at 80 °C, and oil ESTF generated the lowest Λ , because of its low piezoviscosity coefficient.

Figure 7.9.c) shows the rolling torque estimated for the RTB in all operating conditions. As expected, and because the tests were performed at constant temperature (80 °C), when the speed increases the rolling torque also increases ($M_{rr} \propto (n \cdot v)^{0.6}$). This Figure also shows that oils ESTF, ESTR, MINE and PAOR generated very similar rolling torques, because they have the same viscosity grade (ISO VG 320) and similar viscosities at 80 °C. PAGD oil produces very high rolling torques because it has the highest viscosity at 80 °C, while MINR oil, on the opposite, has the lowest viscosity at 80 °C.

Figure 7.9.d) shows the sliding torque estimated for the RTB in all operating conditions. The sliding torque is obtained by subtracting the rolling friction torque to the experimental friction torque, that is, $M_{sl} = M_{exp} - \phi_{ish} \cdot \phi_{rs} \cdot M_{rr}$. The experimental friction torque ($M_{exp} = M_T$, see Figure 7.9.b)) as well as the rolling torque increased when speed increased, and consequently the sliding torque presents a very slight decrease when the speed increases, as shown in Figure 7.9.d).

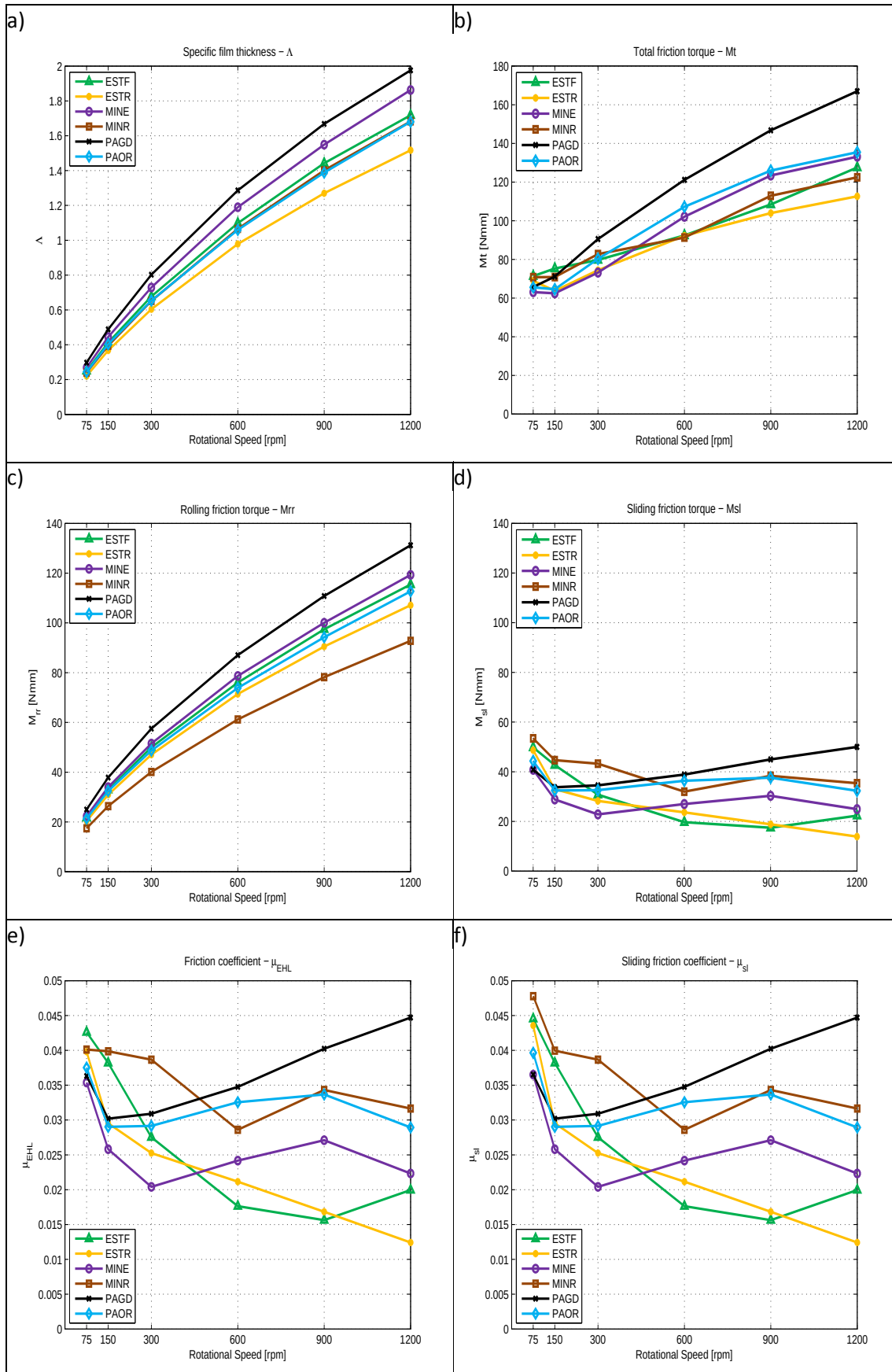


Figure 7.9: Λ , M_t , M_{rr} , M_{sl} , μ_{EHL} and μ_{sl} for RTB 81107 TN – axial load 700N.

Such behaviour is typical of thrust roller bearings operating near to the transition from mixed film to full film lubrication regimes. The sliding and the full film coefficients of friction (μ_{sl} and μ_{EHD}) didn't present a clear trend when the speed increased, as presented in Figure 7.9.f), because the thrust roller bearing was operating at low load. The oils ESTF and ESTR presented a very significant decrease of the coefficient of friction when the operating speed increased, while the oils MINE, MINR and PAOR showed an almost constant coefficient of friction for speeds above 150 rpm. The coefficient of friction of PAGD oil increased steadily with speed (above 150 rpm), exhibiting a typical behaviour of full film lubrication.

Comparing the sliding coefficient of friction and the full film coefficient of friction, (μ_{sl} and μ_{EHD}), shown in Figures 7.9.f) and 7.9.e), no significant differences are observed. Only at the lowest speed (75 rpm), at the smallest specific lubricant film thickness ($0.25 \leq \Lambda \leq 0.30$), $\mu_{sl} > \mu_{EHD}$.

Comparing the friction behaviour of the wind turbine gear oils, inside the thrust roller bearings, it is clear that PAGD oil always produced the highest total bearing friction torque, M_T , while the other oils (ESTF, ESTR, MINE, MINR and PAOR) produced similar total bearing friction torques, always lower than those generated by PAGD oil, as presented in Figure 7.9.b). At low speeds (75 and 150 rpm) all wind turbine gear oils generated very similar friction torques.

7.5.5. Comparison between ball and roller thrust bearings – axial load 7000 N

Figures 7.10.a) and 7.10.b) show the specific lubricant film thickness (Λ) inside thrust ball bearings (TBB – 51107) and thrust roller bearings (RTB – 81107 TN), respectively. For the same speed and the same load, the specific lubricant film thickness show the same increasing trend when the speed increases in both rolling bearings, whatever the wind turbine gear oils considered. However, RTB always generated higher Λ values than TBB. As an example, at 600 rpm and using PAOR oil, the RTB had a Λ value 33% higher than the TBB.

Figures 7.10.c) and 7.10.d) show the total friction torque (M_T) inside thrust ball bearings (TBB – 51107) and thrust roller bearings (RTB – 81107 TN), respectively. As expected, whatever the gear oil and operating speed, RTB always generated higher total friction torques than the TBB. As an example, at 900 rpm and using ESTR oil, the RTB generated a M_T of 295 N.mm, while the TBB generated a M_T of 180 N.mm (64% lower). Furthermore, in the case of TBB, M_T increased when the speed increased, while RTB showed the opposite trend.

Wind turbine gear oils ESTF, ESTR and PAOR always generated similar total friction torques, whatever the speed considered and in both type of rolling bearings.

In the case of the TBB, oil MINE produced lower friction torques than all the other oils while MINR oil produced the highest friction torques. The influence of speed on the friction torque generated by PAGD oil is different from all the other lubricants. In the case of RTB, some of these trends were different. MINE oil had a similar behaviour to ESTF, ESTR and PAOR. At lower speeds ($n \leq 600$ rpm) the PAGD oil produced the lowest total friction torque, but above 600 rpm the lowest friction torque was produced by oil ESTF.

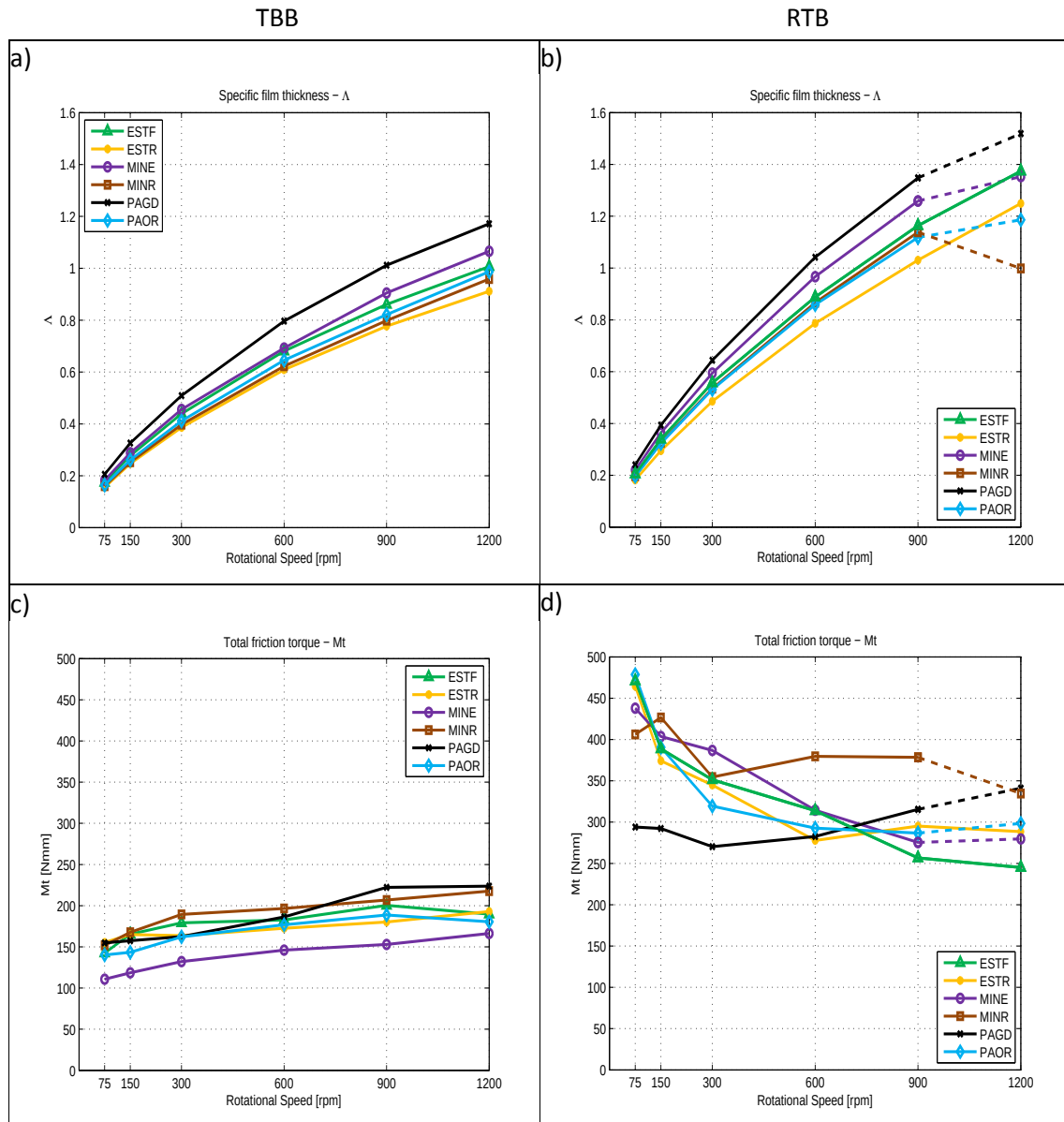


Figure 7.10: Λ and M_t for TBB 51107 and RTB 81107 TN – axial load 7000 N.

Figures 7.11.a) and 7.11.b) show the rolling torque (M_{rr}) inside thrust ball bearings (TBB – 51107) and thrust roller bearings (RTB – 81107 TN), respectively. The rolling torque increases when the speed increases, whatever the type of rolling bearing. Under similar operating conditions the RTB always produced higher rolling torques than the TBB. These figures also show that PAGD oil produces the highest rolling torque and MINR oil the lowest, whatever the speed and type of rolling bearing considered. The other oils (ESTF, ESTR, MINE and PAOR) generated very similar rolling torque because they have similar viscosity at 80 °C.

Figures 7.11.c) and 7.11.d) show the sliding torque (M_{sl}) inside thrust ball bearings (TBB - 51107) and thrust roller bearings (RTB – 81107 TN), respectively. The sliding torque decreases when the speed increases in both types of rolling bearings, but that decrease is very significant in the case of the RTB.

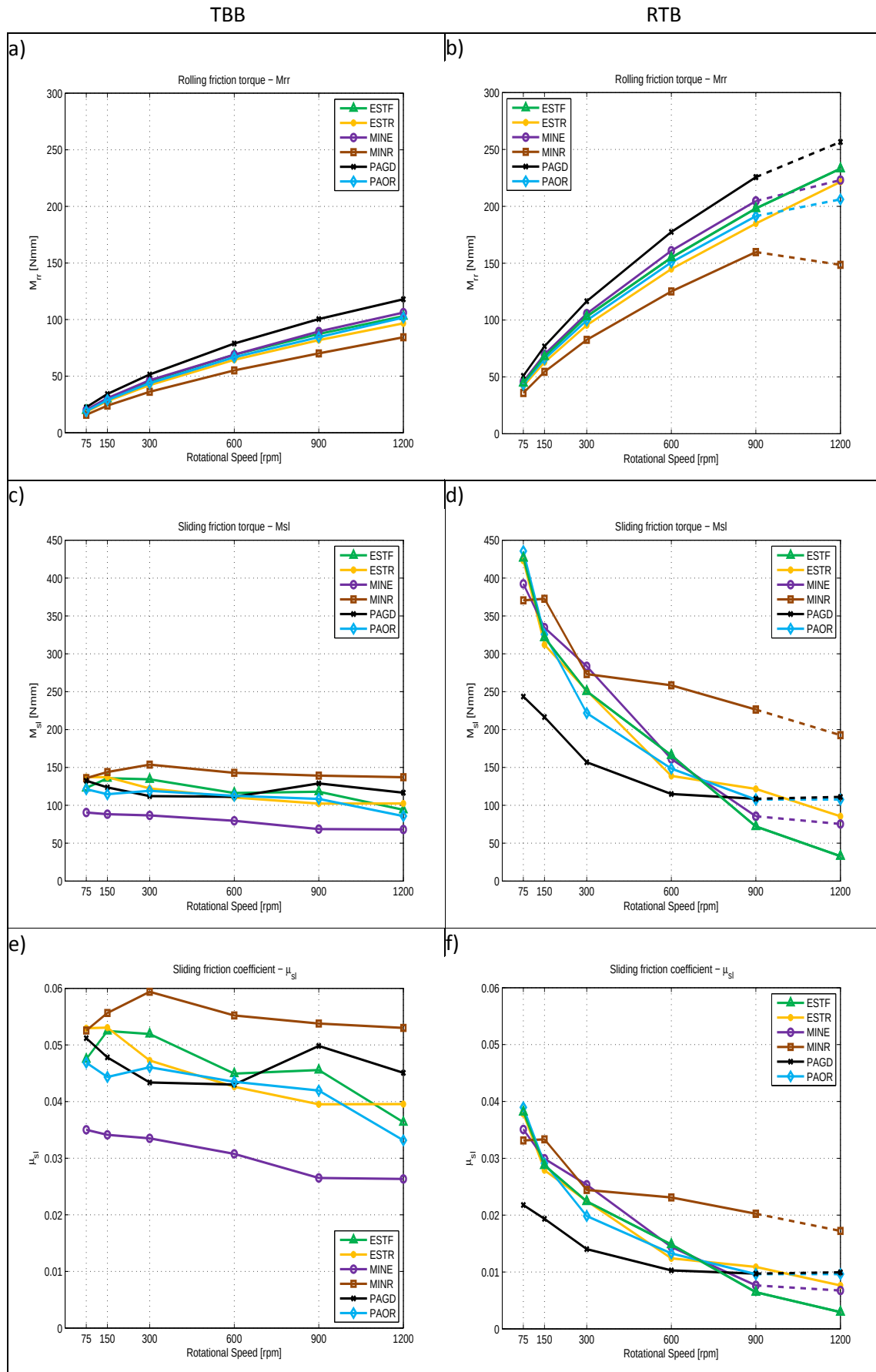


Figure 7.11: M_{rr} , M_{sl} and μ_{sl} for TBB 51107 and RTB 81107 TN – axial load 7000 N.

At low speeds ($n \leq 300$ rpm) the RTB always produced higher sliding torques than the TBB, but at higher speeds ($n \geq 600$ rpm) the sliding torques produced by the two types of rolling bearings become similar.

Figures 7.11.c) and 7.11.d) also show that, in both types of rolling bearings, MINR oil produces the highest sliding torque and the oils ESTF, ESTR and PAOR produce similar sliding torques. In the case of TBB, MINE oil generated the lowest sliding torques and PAGD oil behaved like oils ESTF, ESTR and PAOR. In the case of RTB, PAGD oil generated the lowest sliding torques and MINE oil behaved like oils ESTF, ESTR and PAOR. These different behaviours of the oils MINE and PAGD, depending on the type of bearing, might be related to maximum contact pressures inside TBB ($p_0 = 2.48$ GPa) and RTB ($p_0 = 1.31$ GPa).

7.5.6. Comparison between ball and roller thrust bearings – axial load 700 N

Figures 7.12.a) and 7.12.b) show the specific lubricant film thickness (Λ) inside thrust ball bearings (TBB – 51107) and thrust roller bearings (RTB – 81107 TN), respectively. For the same speed and the same load, the specific lubricant film thickness show the same increasing trend when the speed increases in both rolling bearings, whatever the wind turbine gear oils considered. However, RTB always generated higher Λ values than TBB. As an example, at 300 rpm and using MINE oil, the RTB had a Λ value 46% higher than the TBB.

Figures 7.12.c) and 7.12.d) show the total friction torque (M_T) inside thrust ball bearings (TBB - 51107) and thrust roller bearings (RTB – 81107 TN), respectively. As expected, whatever the gear oil and operating speed, RTB always generated higher total friction torques than the TBB. As an example, at 1200 rpm and using PAGD oil, the RTB generated a M_T of 167 N.mm, while the TBB generated a M_T of 69 N.mm (59% lower). In both rolling bearings, TBB and RTB, M_T increased when the speed increased.

All wind turbine gear oils generated similar total friction torques, whatever the speed considered and in both type of rolling bearings. In the case of the TBB the highest total torque was generated by ESTF oil (44 N.mm at 300 rpm) and the lowest one by PAOR oil (24 N.mm at 300 rpm). The difference between the two is around 20 N.mm at 300 rpm. In the case of the RTB the highest total torque was generated by PAGD oil (91 N.mm at 300 rpm) and the lowest one by MINE oil (73 N.mm at 300 rpm). The difference between the two is about 18 N.mm at 300 rpm.

Figures 7.13.a) and 7.13.b) show the rolling torque (M_{rr}) inside thrust ball bearings (TBB – 51107) and thrust roller bearings (RTB – 81107 TN), respectively. The rolling torque increases when the speed increases, whatever the type of rolling bearing. Under similar operating conditions the RTB always produced higher rolling torques than the TBB. These figures also show that PAGD oil produces the highest rolling torque and MINR oil the lowest one, whatever the speed and type of bearing considered. The other oils (ESTF, ESTR, MINE and PAOR) generated very similar rolling torque because they have similar viscosity at 80 °C.

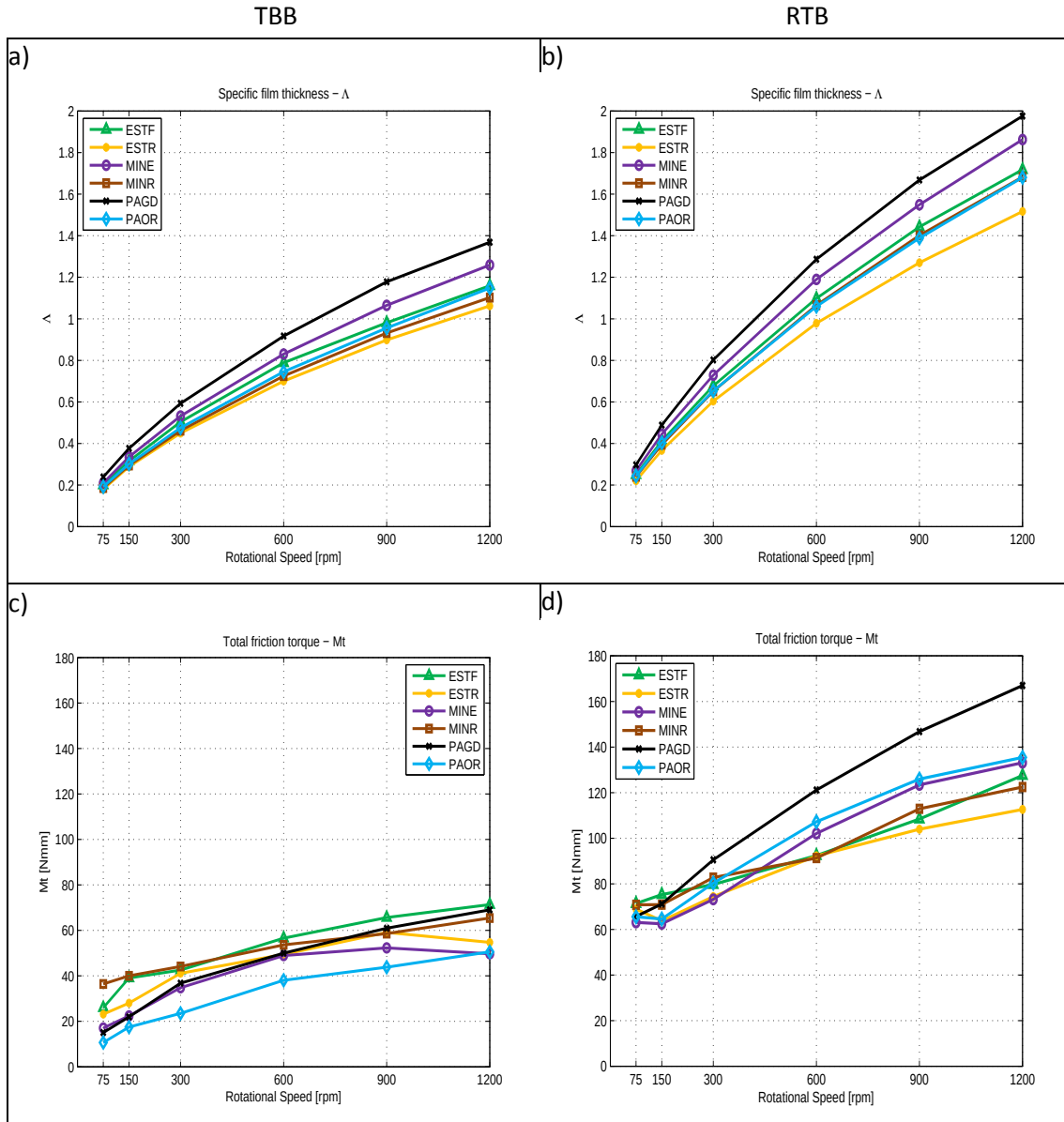


Figure 7.12: Λ and M_t for TBB 51107 and RTB 81107 TN – axial load 700N.

Figures 7.13.c) and 7.13.d) show the sliding torque (M_{sl}) inside thrust ball bearings (TBB - 51107) and thrust roller bearings (RTB – 81107 TN), respectively, under an axial load of 700 N. In the case of the TBB the sliding torque increases when speed increases. However, at high speeds ($n > 900$ rpm) the sliding torque decreases in the case of oils MINE and ESTR. In the case of the RTB, at low speeds ($n \leq 300$ rpm), the sliding torque decreases when speed increases. Above 300 rpm the sliding torque is only slightly influenced by speed.

Figures 7.13.e) and 7.13.f) show the sliding coefficient of friction (μ_{sl}) inside thrust ball bearings (TBB – 51107) and thrust roller bearings (RTB – 81107 TN), respectively, under an axial load of 700 N. Here the most significant difference between TBB and RTB is observed. In the case of TBB, the sliding coefficients of friction (μ_{sl}) are almost always higher than 0.1, while in the case of RTB they are always below 0.05.

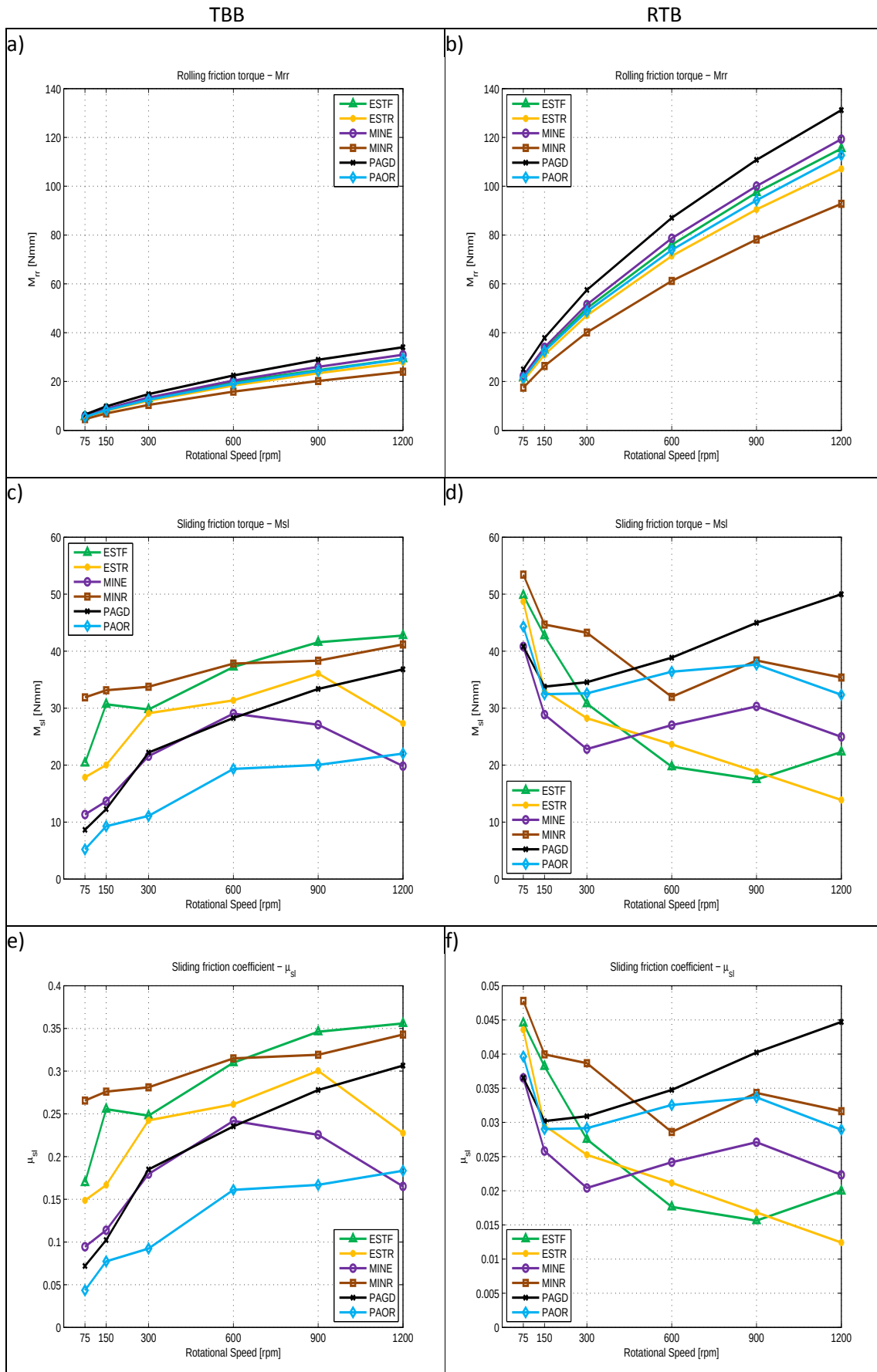


Figure 7.13: M_{rr} , M_{sl} and μ_{sl} for TBB 51107 and RTB 81107 TN – axial load 700N.

These huge differences might be related to the fact that the maximum contact pressures inside TBB is $p_0 = 1.15$ GPa, while the maximum contact pressures inside RTB is only $p_0 = 0.41$ GPa.

8. Conclusions and future work

8.1. Conclusions

The results reached with the torque tests for thrust roller bearings, demonstrated that:

- The specific lubricant film thickness inside the thrust roller bearing increased when the operating speed increased, no matter what the applied axial load was, but its values were higher for an axial load of 700 N.
- The total friction torque, with an axial load of 7000 N, decreased when the operating speed increased, although for an axial load of 700 N the opposite behaviour is observed.
- For the friction torque components, the rolling torque increased with the increase of speed and the sliding torque decreased with the increase of speed, no matter what the applied axial load was.
- For the coefficients of friction with an axial load of 7000 N a clear decrease was observed when the speed increased, but in the case of a load of 700 N there wasn't a clear trend when the speed increased.
- The maximum pressure of Hertz increased 220% (0,41 to 1,31 GPa), when the tests conditions were altered from 700 N to 7000 N.
- For the higher loaded bearing, at 1200 rpm, the operating temperature of some oils increased to values outside those pretended for the experiments.

In conclusion, the results obtained, indicate that for a 7000 N loaded thrust roller bearing the friction torque decreases with the speed for every oil although, the MINR and the PAGD oil begin generating increasing friction torques for speeds above 300 rpm. This behaviour is probably related to their Viscosity Indexes being the lowest, in the case of MINR, and the highest, in the case of the PAGD, of all the oils. The PAGD oil also generates significantly lower friction torque for lower speeds in comparison to the others. The coefficients of friction decrease when the speed increases, the PAGD oil has significant lower values for lower speeds while the MINR has significant higher ones for higher speeds. It was also observed that at the speed of 1200 rpm the oils, excluding the ester based oils, achieved higher temperatures than the ones set for the experiments. For a 700 N loaded thrust roller bearing the oils friction torque behaviour is the opposite of the higher loaded case, and in this case the PAGD oil distances from the others by increasing its friction torque faster, while the others have approximate values. The coefficients of friction didn't show a clear trend when the speed increased above 150 rpm, meaning that it either decreased (ESTF and ESTR), was almost constant (MINR, MINE and PAOR) or increased (PAGD).

The results achieved with the torque tests for thrust ball bearings, showed that:

- The specific lubricant film thickness inside the thrust ball bearing increased when the operating speed increased, no matter what the applied axial load was, but its values were higher for an axial load of 700 N.
- The total friction torque increased when the operating speed increased for the two applied axial loads, although its values are much higher for a higher axial load.
- The rolling friction torque demonstrated an increase with the increase of speed for the two applied axial loads.
- The sliding friction torque, in the case of a higher axial load, decreased with the increase of speed but in the case of a lower axial load the opposite behaviour was observed because the total friction torque increased faster than the rolling friction torque although, at higher speeds the sliding friction of some oils starts decreasing when the speed increases.
- The coefficients of friction, with an axial load of 7000 N, decreased when the speed increased but, in the case of a load of 700 N they increased when the speed increased achieving unusually high values.
- The maximum pressure of Hertz increased 116% (1,15 to 2,48 GPa), when the tests conditions were altered from 700 N to 7000 N.

In conclusion, the results obtained, indicate that for a 7000 N loaded thrust ball bearing the MINE oil produced a significantly lower friction torque than the other oils probably because of the high percentage of additives in its composition although for lower loads this doesn't occur. The MINR generated the highest friction torque for lower speeds and the PAGD oil for higher speeds probably due these two oils having significantly different Viscosity Index than the other oils. The other oils showed similar torque values. For the coefficients of friction the MINE maintains the lowest values and the MINR the highest. For a 700 N loaded thrust ball bearing the oils friction torque behaviour is similar to the higher loaded case although the friction values increase faster. The coefficients of friction reach high values probably due to the type of bearing (TBB) with small loads which generate these high friction coefficient values and the friction torque model used for lightly loaded rolling bearings.

8.2. Future work

Further studies could be made related to this theme, for example:

- Wear tests could be performed at a fixed temperature for different rotational speeds for the six oils used in this work.
- Perform the friction torque measuring test and wear tests with a different fixed temperature to see how the torque and wear vary for different temperatures.

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Appendix

A.1. Four-Ball Machine

The four ball machine (Figure 1) was manufactured in the UK by Cameron-Plint and was developed with the cooperation of the National Engineering Laboratory East Kilbride Scotland, as a machine with a 4 ball low and high speed. Its serial number is TE82/7752



Figure 1: Four-Ball Machine

The four ball machine is mainly utilized to characterize the anti-wear (AW) and extreme pressure (EP) of oils and greases. This machine allows the testing of four balls with control of rotational speed and axial load up to 20.000 rpm and 8.000N respectively.

The tests performed on this machine use four balls of 12, 7 mm diameter arranged in a pyramid shape of a triangular base. The spheres of the base may be fixed or free, allowing two different types of tests, in pure sliding or rolling respectively. The movement is transmitted to the higher sphere, which is contacting the three lower spheres, over which the load is applied.

With the standard layout shown in the next figure, the tests are carried out according to ASTM D2266, ASTM D2596, ASTM D2783, IP 239 and IP 300.

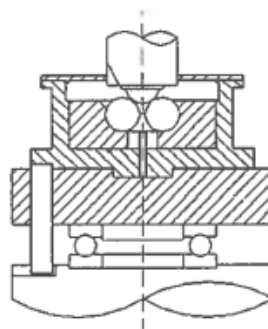


Figure 2: Disposition of standard machine 4 balls

A.2. Hertz solution factors

Table 1: Factors for Hertz solution. [11]

A/B	k	1/k	e	C _a	C _δ
1.0	1.000	1.000	0.000	0.909	2.356
1.1	1.000	1.000	0.000	0.909	2.356
1.2	0.886	1.129	0.464	0.856	2.215
1.3	0.840	1.190	0.543	0.834	2.155
1.4	0.799	1.252	0.601	0.815	2.100
1.5	0.763	1.311	0.646	0.797	2.049
1.6	0.731	1.368	0.682	0.782	2.003
1.7	0.702	1.425	0.712	0.767	1.959
1.8	0.676	1.479	0.737	0.754	1.919
1.9	0.652	1.534	0.758	0.742	1.882
2.0	0.631	1.585	0.776	0.731	1.846
2.1	0.611	1.637	0.792	0.721	1.813
2.2	0.592	1.689	0.806	0.711	1.782
2.3	0.575	1.739	0.818	0.702	1.753
2.4	0.559	1.789	0.829	0.693	1.725
2.5	0.544	1.838	0.839	0.685	1.699
2.6	0.530	1.887	0.848	0.678	1.674
2.7	0.517	1.934	0.856	0.671	1.650
2.8	0.505	1.980	0.863	0.664	1.627
2.9	0.493	2.028	0.870	0.658	1.605
3.0	0.483	2.070	0.876	0.652	1.584
3.1	0.472	2.119	0.882	0.646	1.564
3.2	0.463	2.160	0.886	0.640	1.545
3.3	0.453	2.208	0.892	0.635	1.527
3.4	0.445	2.247	0.896	0.630	1.509
3.5	0.436	2.294	0.900	0.625	1.492
3.6	0.428	2.336	0.904	0.620	1.475
3.7	0.421	2.375	0.907	0.616	1.460
3.8	0.413	2.421	0.911	0.612	1.444
3.9	0.407	2.457	0.913	0.607	1.430
4.0	0.400	2.500	0.917	0.603	1.415
4.1	0.394	2.538	0.919	0.600	1.401
4.2	0.387	2.584	0.922	0.596	1.388
4.3	0.382	2.618	0.924	0.592	1.375
4.4	0.376	2.660	0.927	0.589	1.363
4.5	0.370	2.703	0.929	0.585	1.350
4.6	0.365	2.740	0.931	0.582	1.338
4.7	0.360	2.778	0.933	0.579	1.327
4.8	0.355	2.817	0.935	0.576	1.316
4.9	0.351	2.849	0.936	0.573	1.305
5.0	0.346	2.890	0.938	0.570	1.290
5.1	0.342	2.924	0.940	0.567	1.284
5.2	0.337	2.967	0.942	0.564	1.274
5.3	0.333	3.003	0.943	0.562	1.264
5.4	0.329	3.040	0.944	0.559	1.254
5.5	0.325	3.077	0.946	0.557	1.245
5.6	0.322	3.106	0.947	0.554	1.236
5.7	0.318	3.145	0.948	0.552	1.227
5.8	0.314	3.185	0.949	0.549	1.218
5.9	0.311	3.215	0.950	0.547	1.210
6.0	0.308	3.247	0.951	0.545	1.202
6.1	0.304	3.289	0.953	0.543	1.194
6.2	0.301	3.322	0.954	0.540	1.186
6.3	0.298	3.356	0.955	0.538	1.178
6.4	0.295	3.390	0.955	0.536	1.170
6.5	0.292	3.425	0.956	0.534	1.163
6.6	0.289	3.460	0.957	0.532	1.156
6.7	0.287	3.484	0.958	0.530	1.148
6.8	0.284	3.521	0.959	0.528	1.141
6.9	0.281	3.559	0.960	0.527	1.135
7.0	0.279	3.584	0.960	0.525	1.128
7.1	0.276	3.623	0.961	0.523	1.121
7.2	0.274	3.650	0.962	0.521	1.115
7.3	0.271	3.690	0.963	0.520	1.108
7.4	0.269	3.717	0.963	0.518	1.102
7.5	0.267	3.745	0.964	0.516	1.096
7.6	0.264	3.788	0.965	0.515	1.090
7.7	0.262	3.817	0.965	0.513	1.084
7.8	0.260	3.846	0.966	0.511	1.078
7.9	0.258	3.876	0.966	0.510	1.073
8.0	0.256	3.906	0.967	0.508	1.067
8.1	0.254	3.937	0.967	0.507	1.061
8.2	0.252	3.968	0.968	0.505	1.056
8.3	0.250	4.000	0.968	0.504	1.051
8.4	0.248	4.032	0.969	0.503	1.045
8.5	0.246	4.065	0.969	0.501	1.040
8.6	0.244	4.098	0.970	0.500	1.035
8.7	0.243	4.115	0.970	0.498	1.030
8.8	0.241	4.149	0.971	0.497	1.025
8.9	0.239	4.184	0.971	0.496	1.020
9.0	0.238	4.202	0.971	0.495	1.016
9.1	0.236	4.237	0.972	0.493	1.011
9.2	0.234	4.274	0.972	0.492	1.006
9.3	0.233	4.292	0.972	0.491	1.002
9.4	0.231	4.329	0.973	0.490	0.997
9.5	0.230	4.348	0.973	0.488	0.993
9.6	0.228	4.386	0.974	0.487	0.988
9.7	0.227	4.405	0.974	0.486	0.984
9.8	0.225	4.444	0.974	0.485	0.980
9.9	0.224	4.464	0.975	0.484	0.976
10.0	0.222	4.505	0.975	0.483	0.972

Table 2: Factors for Hertz solution (part 2). [11]

A/B	k	1/k	e	C _a	C _δ
1.0	1.000	1.000	0.000	0.909	2.356
2.0	0.631	1.585	0.776	0.731	1.846
3.0	0.483	2.070	0.876	0.652	1.584
4.0	0.400	2.500	0.917	0.603	1.415
5.0	0.346	2.890	0.938	0.570	1.294
6.0	0.308	3.247	0.951	0.545	1.202
7.0	0.279	3.584	0.960	0.525	1.128
8.0	0.256	3.906	0.967	0.508	1.067
9.0	0.238	4.202	0.971	0.495	1.016
10.0	0.222	4.505	0.975	0.483	0.972
11.0	0.209	4.785	0.978	0.472	0.933
12.0	0.198	5.051	0.980	0.463	0.899
13.0	0.189	5.291	0.982	0.455	0.869
14.0	0.180	5.556	0.984	0.447	0.842
15.0	0.172	5.814	0.985	0.441	0.817
16.0	0.166	6.024	0.986	0.435	0.795
17.0	0.160	6.250	0.987	0.429	0.774
18.0	0.154	6.494	0.988	0.424	0.756
19.0	0.149	6.711	0.989	0.419	0.738
20.0	0.144	6.944	0.990	0.414	0.722
21.0	0.140	7.143	0.990	0.410	0.707
22.0	0.136	7.353	0.991	0.406	0.692
23.0	0.132	7.576	0.991	0.402	0.679
24.0	0.129	7.752	0.992	0.398	0.667
25.0	0.126	7.937	0.992	0.395	0.655
26.0	0.123	8.130	0.992	0.392	0.644
27.0	0.120	8.333	0.993	0.388	0.633
28.0	0.118	8.475	0.993	0.385	0.623
29.0	0.115	8.696	0.993	0.383	0.614
30.0	0.113	8.850	0.994	0.380	0.605
31.0	0.110	9.091	0.994	0.377	0.596
32.0	0.108	9.259	0.994	0.375	0.588
33.0	0.106	9.434	0.994	0.373	0.580
34.0	0.104	9.615	0.995	0.370	0.572
35.0	0.103	9.709	0.995	0.368	0.565
36.0	0.101	9.901	0.995	0.366	0.558
37.0	0.099	10.101	0.995	0.364	0.551
38.0	0.098	10.204	0.995	0.362	0.545
39.0	0.096	10.417	0.995	0.360	0.538
40.0	0.095	10.526	0.995	0.358	0.533
41.0	0.093	10.753	0.996	0.356	0.527
42.0	0.092	10.870	0.996	0.354	0.521
43.0	0.091	10.989	0.996	0.353	0.516
44.0	0.089	11.236	0.996	0.351	0.510
45.0	0.088	11.364	0.996	0.349	0.505
46.0	0.087	11.494	0.996	0.348	0.501
47.0	0.086	11.628	0.996	0.346	0.496
48.0	0.085	11.765	0.996	0.345	0.491
49.0	0.084	11.905	0.996	0.343	0.487
50.0	0.083	12.048	0.997	0.342	0.482
51.0	0.082	12.195	0.997	0.341	0.478
52.0	0.081	12.346	0.997	0.339	0.474
53.0	0.080	12.500	0.997	0.338	0.470
54.0	0.079	12.658	0.997	0.337	0.466
55.0	0.078	12.821	0.997	0.336	0.462
56.0	0.077	12.987	0.997	0.334	0.459
57.0	0.077	12.987	0.997	0.333	0.455
58.0	0.076	13.158	0.997	0.332	0.452
59.0	0.075	13.333	0.997	0.331	0.448
60.0	0.074	13.514	0.997	0.330	0.445
61.0	0.074	13.514	0.997	0.329	0.442
62.0	0.073	13.699	0.997	0.327	0.438
63.0	0.072	13.889	0.997	0.326	0.435
64.0	0.072	13.889	0.997	0.325	0.432
65.0	0.071	14.085	0.997	0.324	0.429
66.0	0.070	14.286	0.998	0.323	0.426
67.0	0.070	14.286	0.998	0.322	0.423
68.0	0.069	14.493	0.998	0.321	0.421
69.0	0.068	14.706	0.998	0.320	0.418
70.0	0.068	14.706	0.998	0.320	0.415
71.0	0.067	14.925	0.998	0.319	0.413
72.0	0.067	14.925	0.998	0.318	0.410
73.0	0.066	15.152	0.998	0.317	0.408
74.0	0.066	15.152	0.998	0.316	0.405
75.0	0.065	15.385	0.998	0.315	0.403
76.0	0.065	15.385	0.998	0.314	0.400
77.0	0.064	15.625	0.998	0.313	0.398
78.0	0.064	15.625	0.998	0.313	0.396
79.0	0.063	15.873	0.998	0.312	0.394
80.0	0.063	15.873	0.998	0.311	0.391
81.0	0.062	16.129	0.998	0.310	0.389
82.0	0.062	16.129	0.998	0.310	0.387
83.0	0.061	16.393	0.998	0.309	0.385
84.0	0.061	16.393	0.998	0.308	0.383
85.0	0.060	16.667	0.998	0.307	0.381
86.0	0.060	16.667	0.998	0.307	0.379
87.0	0.060	16.667	0.998	0.306	0.377
88.0	0.059	16.949	0.998	0.305	0.375
89.0	0.059	16.949	0.998	0.305	0.373
90.0	0.058	17.241	0.998	0.304	0.371
91.0	0.058	17.241	0.998	0.303	0.369
92.0	0.058	17.241	0.998	0.303	0.368
93.0	0.057	17.544	0.998	0.302	0.366
94.0	0.057	17.544	0.998	0.301	0.364
95.0	0.057	17.544	0.998	0.301	0.362
96.0	0.056	17.857	0.998	0.300	0.361
97.0	0.056	17.857	0.998	0.299	0.359
98.0	0.056	17.857	0.998	0.299	0.357
99.0	0.055	18.182	0.998	0.298	0.356
100.0	0.055	18.182	0.998	0.298	0.354

A.3. Lubricants additives

Table 1: Common lubricant additives [18]

Additive	Purpose
Oxidation-inhibitor	Increases oil and machine life, decreases varnish and sludge on metal parts
Corrosion inhibitor	Protects against chemical attacks of alloy bearings and metal surfaces
Antiwear improver	Protects rubbing surfaces operating with thin films, boundary lubrication
Detergent	Cleanliness of lubricated surfaces
Dispersant	Keeps insoluble combustion and oxidation products in suspension and dispersed
Alkaline agent	Neutralizes acid from oxidation of oil so it cannot react with oil or engines
Rust inhibitor	Eliminates rusting in presence of water or moisture
Pour depressant	Lowers low temperature fluidity
Viscosity improver (VI)	Lowers rate of change of viscosity with temperature change
Oiliness agent	Reduces friction, seizure, wear; increases viscosity
Extreme pressure (EP)	Increases film strength and load-carrying capacity
Antifoam agent	Prevents stable foam formation
Tackiness agent	For greater cohesion, no drip property
Emulsifier	Reduces interfacial tension so oil can disperse in water
Fatty oils	For greater wetting for moisture conditions
Solid lubricants (filler)	Withstand high temperatures and/or pressures
Thickening agent	Converts oil into solid or semisolid lubricant
Water repellents	Impart water-resistant properties to components of lubricants
Metal deactivators	Pacify, prevent, or counteract catalytic effect of metals
Silver pacifier	Noncorrosive to silver bearings
Colour stabilizer	Standardizes desirable colour and prevents formation of undesirable colour
Odour-control agent	Provides distinctive or pleasant odour or masks undesirable odours
Antiseptic	Prevents emulsion breakdown or odour from growth of bacteria