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Friction torque of thrust ball bearings lubricated with wind turbine gear oils

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ABSTRACT

Planetary gearboxes used in wind turbines very often have premature bearing and gear failures, some of them related to the lubricants used. Five fully formulated wind turbine gear oils with the same viscosity grade and different formulations were selected and their physical characterization was performed. The lubricant tribological behaviour in a thrust ball bearing was analyzed. A modified Four-Ball Machine was used to assemble the bearings. They were submitted to an axial load and the tests were performed at velocities ranging between 150 and 1500 rpm. Experimental results for the operating temperatures and for the internal friction torque are presented.

1. Introduction

As proven source of clean and affordable energy, wind resources clearly have a vital role to play in energetic sustainability [1]. In this sense it is necessary to have wind turbines that maximize the use of eolic energy and achieve their design life goals with minimal maintenance.

Gearboxes have plagued the wind power industry [2–5]. Wind turbine failures can be extremely costly in terms of repair costs, replacement parts and lost power, and the gearbox is the most likely component to have a major effect on the turbine availability. Since the establishment of the wind energy industry large failure rates of the gearboxes have been observed. Windmills, often placed in hostile environments, have premature bearing and gear failures, and the performance of the gear oils used for their lubrication also have an important role in gearbox reliability. Most wind turbine gearbox failures are rooted to the bearings [6–9]. The most significant fatigue wear phenomena is micropitting and smearing caused by large amounts of roller/raceway sliding in situations in which specific film thickness (Λ) is low, leading to high stresses and temperatures in the contact [10].

Due to economic and environmental constraints it is mandatory to increase the efficiency of windmills, to reach the highest efficiency of planetary gear drives and their parts (gears, rolling bearings, seals, ...) and to minimize the heat generation in the gearboxes [11]. In order to increase gearbox efficiency it is important to identify the main sources of power loss. The most common wind turbine gearboxes have planetary gears and the main losses occurring are: friction loss between the meshing teeth [12–17], friction loss in the bearings [12,18,19], friction loss in the seals [12], lubricant churning losses [20,21] and energy loss due to air-drag [11].

Friction generated between the meshing teeth is the main source of power loss in a planetary gear. On the other hand, rolling bearing friction is also very important because it can reach about 30% of total power loss occurring within the mechanism [22]. In this sense, understand the friction torque generated within rolling bearings is essential in order to reduce their contribution to the overall power loss. There are four physical friction sources inside a rolling bearing: rolling friction, sliding friction, seal friction and drag losses [23]. The most important ones in the case of windmill applications (high torque and low speed) are the friction occurring in the contact between the rolling elements and raceways (sliding friction) and the friction due to the lubricant flow between the bearing elements (rolling friction). These energy loss mechanisms are highly dependent on the lubricant ability to generate an effective oil film between the rolling elements and the raceways and on the physical properties of the gear oils.

The identification of the loss mechanisms occurring in rolling bearings lubricated with wind turbine gear oils are the main purpose of this work. For that purpose the friction torque losses in thrust ball bearings (51107) were identified and compared when different lubricants are used. Five different fully formulated gear oils were characterized and tested on a thrust ball bearing (51107) submitted to an axial load of 7000 N and rotational speeds between 150 and 1500 rpm. The tests were performed on a modified Four-Ball Machine (Cameron-Plint TE 82/7752) using a special assembly for the thrust ball bearings (51107).

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Notation and units

T	operating temperature (°C)
T_{ref}	reference temperature (°C)
ΔT	stabilization temperature (°C)
α_t	thermal expansion coefficient (–)
β	thermoviscosity (°K^{-1})
ρ	density (g/cm^3)
ρ_0	density at reference temperature (g/cm^3)
C_0	Ellipticity influence parameter (–)
d_m	bearing mean diameter (mm)
E^*	equivalent Young's Modulus (Pa)
Fa	axial load (N)
G	material influence parameter (–)
G_{rr}	factor that depends on the bearing type, bearing mean diameter and applied load (–)
G_{sl}	factor that depends on the bearing type, bearing mean diameter and applied load (N mm)
H_0	centred film thickness (μm)
K_{rs}	starvation constant for oil bath (3×10^{-8})
K_Z	bearing type related geometry constant (3.8)
L	thermal parameter of lubricant (–)
m	calculated according ASTM D341 [28] (–)
M_{exp}	bearing friction torque measured experimentally (N mm)
M_{rr}	rolling friction torque (N mm)
M_{sl}	sliding friction torque (N mm)
M_{drag}	friction torque of drag losses (N mm)
M_{seal}	friction torque of seals (N mm)
M_t	total bearing friction torque (N mm)
η	kinematic viscosity at the operating temperature (mm^2/s)
n	rotational speed (rpm)
n	calculated according ASTM D341 [28] (–)
ΔP	power loss (W)
R_1	geometry constant for rolling frictional torque (1.03×10^{-6})
S_1	geometry constant for sliding frictional torque (0.016)
s	parameter depending on lubricant package according to [29] (–)
S	sliding rate (–)
t	parameter depending on lubricant package according to [29] (–)
U	speed influence parameter (–)
U_1	linear speed of ball (m/s)
U_2	linear speed of ring (m/s)
VI	viscosity index (–)
W	load influence parameter (–)
ϕ_{bl}	sliding frictional torque weighting factor (–)
ϕ_{ish}	inlet shear heating reduction factor (–)
ϕ_{rs}	kinematic replenishment/starvation reduction factor (–)
ϕ_A	starvation flow reduction factor (–)
ϕ_T	thermal reduction factor (–)
ϕ_R	roughness reduction factor (–)
μ	dynamic viscosity (Pa s)
μ_{bl}	coefficient depending on the additive package in the lubricant (–)
μ_{EHD}	friction coefficient in full film conditions (–)
μ_{sl}	sliding friction coefficient (–)
ν	kinematic viscosity (cSt))
Λ	specific film thickness (μm)
τ	shear stress (Pa)

2. Lubricant properties

All the lubricants tested are fully formulated gear oils have a viscosity grade ISO VG 320 and their base oils are: Ester (ESTF and ESTR), Mineral (MINR), Polyalkyleneglycol (PAGD) and Polyalphaolefin (PAOR). Table 1 displays the physical properties of the five lubricants as well as their chemical composition.

2.1. Chemical composition

Using the ICP method according to ASTM D 5185, the chemical composition of the lubricants was determined and presented in Table 1. The elements identified were Zinc (Zn), Magnesium (Mg), Phosphorus (Ph), Calcium (Ca), Boron (B) and Sulfur (S). It is clear that the formulations are significantly different, both in terms of base oil and additive package.

2.2. Physical properties

A complete characterization of physical properties of the lubricants was performed. A detailed description are presented in Appendix A and the results are in Table 1.

3. Rolling bearing assembly and test procedures

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 Fig. 1. This assembly was developed to test several rolling bearings and measure the friction torque and the operating temperature in several different points. A detailed presentation of this assembly can be found in [24].

The rolling bearing assembly is divided in two parts: the shaft adapter (6), directly connected to the machine shaft and supporting the bearing upper race (5); a lower race support (2) and the bearing lower race (3), both clamped to the bearing housing (1). In operation, the internal bearing torque (or friction torque) is transmitted to the torque cell (11) through the bearing housing (1). The friction torque was measured with a piezoelectric torque cell *KISTLER* 9339*A*, ensuring high-accuracy measurements even when the friction torque generated in the bearing was very small compared to the measurement range available.

This assembly has five thermocouples (I–V), measuring temperatures at strategic locations (see Fig. 1) which are used to monitor the temperature inside the bearing assembly (IV), near to the rolling bearing and the lubricant (III) and to evaluate the heat evacuation from the bearing housing into the surrounding environment (I, II and V). The system is also monitored by two thermocouples to quantify the chamber and room temperatures.

When assembled in the modified Four-Ball machine the rolling bearing assembly is submitted to a continuous air flow, forced by two 38 mm diameter fans running at 2000 rpm, cooling the chamber surrounding the bearing house.

For the axial load applied (7000 N) each contact element (ball) reach a Hertz pressure of 2.483 GPa and the half-width of contact area is 123.87×10^{-3} mm.

The rolling bearing is lubricated by an oil volume of 14 ml. This volume was selected so that the oil level reaches the centre of the ball, such as advised by the manufacturer.