Tribology International 88 (2015) 298–308

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

Gearbox power loss. Part I: Losses in rolling bearings

Carlos M.C.G. Fernandes [a,*], Pedro M.T. Marques [a], Ramiro C. Martins [a], Jorge H.O. Seabra [b]

[a] INEGI, Universidade do Porto, Campus FEUP, Rua Dr. Roberto Frias 400, 4200-465 Porto, Portugal
[b] FEUP, Universidade do Porto, Rua Dr. Roberto Frias s/n, 4200-465 Porto, Portugal

ARTICLE INFO

Article history:
Received 25 July 2014
Received in revised form
20 October 2014
Accepted 20 November 2014
Available online 3 December 2014

Keywords:
Rolling bearings
Gearboxes
Power loss
Wind turbine gear oils

ABSTRACT

This work is devoted to the analysis, modeling and validation of gearbox power loss, considering the influence of the gears, rolling bearings and seals, the influence of the operating conditions, lubricant formulation and the lubrication method.

The first part of this work a rolling bearing torque loss model is calibrated for several wind turbine gear oils and for ball and roller contacts. The results achieved clarify the importance of a rolling bearing in a gearbox power loss.

In the second part of this work will be presented the same approach for gears. The final part will converge in the application of findings in two full gearboxes, a planetary and a parallel axis gearbox, both in multiplying configuration.

1. Introduction

Wind turbines use wind energy to generate electricity, having nowadays a significant contribution to the electrical power generation from renewal sources around the world [1]. The blades of a wind turbine rotate at very low speeds, typically 20 revolutions per minute, which is not suitable for conventional power generation using an electrical generator. This problem is solved using a multiplying gearbox between the blades shaft and the generator. The gearbox might have different configurations, although one of the most used designs has two planetary stages plus a helical gear stage at the end. The efficiency of these multiplying gearboxes, with this arrangement or a similar one, is good. Nevertheless, any efficiency increase will have a significant impact, reducing the power loss and the operating temperature. In the case of 1 MW wind turbine an improvement of 0.34% in efficiency per gear stage, could lead to a reduction of 10 kW in power loss and a reduction of the operating temperature above 5 °C. Besides the additional power produced, the reduction in operating temperature contributes to a better lubrication, less oil degradation and lower needs in maintenance. Such increase in gearbox efficiency is already possible through an improved gear tooth design or selecting the most suitable gear oil formulation, or even, combining these two possibilities. Furthermore, gear oils generating lower gear power loss will have a similar effect on rolling bearings supporting the gearbox shafts. It is clear that the comparison of different wind turbine gear oil formulations, in terms of power loss and operating temperature, in gears and rolling bearings, can be very useful to understand and predict the efficiency and the operating temperature of a wind turbine gearbox.

According to Hohn et al. [2], as well as several other authors [3–14] the power loss in a gearbox consists of gear (P_{VZ0} and P_{VZP}), bearing (P_{VL}), seals (P_{VD}) and auxiliary losses (P_{VX}) as presented in Fig. 1.

Gear and bearing losses can be separated in no-load (P_{VZ0}) and load losses (P_{VZP}). No-load losses occur with the rotation of mechanical components, even without torque transmission. No-load losses are mainly related to lubricant viscosity and density as well as immersion depth of the components on a sump lubricated gearbox, but it also depends on operating conditions and internal design of the gearbox casing. Rolling bearing losses depend on type and size, arrangement, lubricant viscosity and immersion depth; the oil formulation is also important.

Load dependent losses occur in the contact of the power transmitting components. Load losses depended on the transmitted torque, coefficient of friction and sliding velocity in the contact areas of the components. Load dependent rolling bearing losses also depend on type and size, rolling and sliding conditions and lubricant type [15].

Full gearbox tests give very good indication of the operating temperature and total power loss, for a given set of operating conditions (input torque and speed), but it is very difficult to isolate the power loss corresponding to each component, gear set, rolling bearing or seal. Full gearbox tests also do not allow a clear separation between load and no-load dependent losses. In order to have good predictions of the power losses related to each component, dedicated tests are necessary, as well as accurate

* Corresponding author.
E-mail address: cfernandes@inegi.up.pt (C.M.C.G. Fernandes).

http://dx.doi.org/10.1016/j.triboint.2014.11.017
0301-679X/

Notation and Units

a	ASTM D341 reference kinematic viscosity (cSt)
d_m	rolling bearing mean diameter (mm)
Fa	axial load (N)
G_{rr}	factor depending on the bearing type, bearing mean diameter and applied load (/)
G_{sl}	factor depending on the bearing type, bearing mean diameter and applied load (N mm)
K_{rs}	starvation constant for oil bath lubrication (/)
K_Z	bearing type related geometry constant (/)
m	ASTM D341 viscosity parameter (/)
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	internal bearing friction torque (N mm)
n	ASTM D341 viscosity parameter (/)
n	rotational speed (rpm)
R_1	geometry constant for rolling friction torque (/)
S_1	geometry constant for sliding friction torque (/)
s	piezoviscosity parameter (/)
t	piezoviscosity parameter (/)
α	piezoviscosity coefficient (Pa^{-1})
α_t	thermal expansion coefficient (/)
β	thermoviscosity coefficient [°K^{-1}]
η	dynamic viscosity (Pa s)
ϕ_{bl}	sliding friction torque weighting factor (/)
ϕ_{ish}	inlet shear heating reduction factor [/]
ϕ_{rs}	kinematic replenishment/starvation reduction factor (/)
μ_{bl}	coefficient of friction in boundary film lubrication (/)
μ_{EHD}	coefficient of friction in full film lubrication (/)
μ_{sl}	sliding coefficient of friction (−)
μ	bearing coefficient of friction (−)
ν	kinematic viscosity (cSt)
ρ	density (g/cm^3)

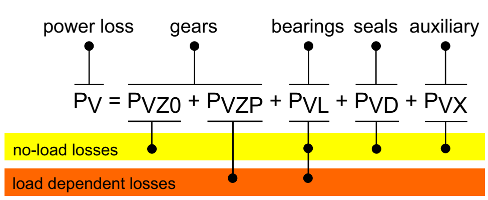

Fig. 1. Power loss contributions [3].

prediction models, that might be integrated in a general power loss model for gearboxes. The aim of the first part of this work is to calibrate a rolling bearing torque loss model, using the experimental results from tests performed with thrust ball bearings (TBB, elliptic contact) and cylindrical roller thrust bearings (RTB, line contact), lubricated with several wind turbine gear oils with different formulations.

2. Rolling bearing friction torque model

This model, proposed by SKF [15], considers that the total friction torque is the sum of four different physical sources of torque loss, represented by the following equation:

$$M_t = M'_{rr} + M_{sl} + M_{drag} + M_{seal} \quad (1)$$

The rolling bearings tested (see Table 1), TBB (51107) and RTB (81107), do not have seals and so the M_{seal} torque loss term was disregarded. The drag losses were very small because the operating speeds and the mean diameter of the rolling bearings are small and, consequently, the drag torque loss term was also disregarded. In Fig. 2 are presented the usual drag loss values for two different TBBs; two different RTBs, one NJ 406 cylindrical roller bearing and a four-point contact ball bearing. The results show that the rolling bearings tested here generated negligible drag losses. However, if rolling bearings with much higher diameter (TBB 51214 and RTB 81214) are used on the same conditions and if the rotational speed increases this term can be an important source of power loss. Since the TBB and RTB's that were used are quite small and operate at relatively low speeds the drag torque loss is negligible (Fig. 2).

Thus, the total internal friction torque of the rolling bearings has only two contributions: the rolling and sliding torques,

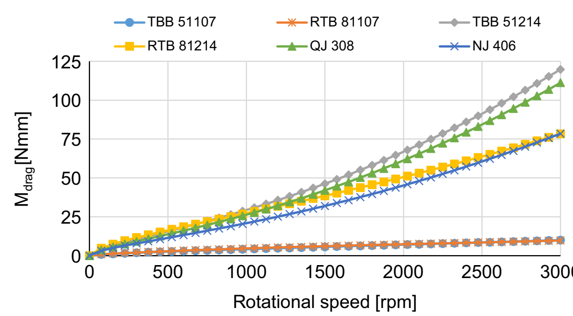

Fig. 2. Drag losses for different rolling bearing geometries. NJ 406 and QJ 308 under oil jet lubrication, the other geometries under half height oil bath for vertical shaft configuration lubricated with PAGD at 80 °C.

respectively, M'_{rr} and M_{sl}, as represented in the following equation:

$$M_t = M'_{rr} + M_{sl} \quad (2)$$

Eqs. (3)–(8) define the rolling and sliding torques,

$$M'_{rr} = \phi_{ish}.\phi_{rs}[G_{rr}(n.\upsilon)^{0,6}] \quad (3)$$

$$\phi_{ish} = \frac{1}{1+1.84 \times 10^{-9}(nd_m)^{1.28}\nu^{0.64}} \quad (4)$$

$$\phi_{rs} = \frac{1}{e^{K_{rs}\nu n(d+D)\sqrt{\frac{K_Z}{2(D-d)}}}} \quad (5)$$

$$M_{sl} = G_{sl}.\mu_{sl} \quad (6)$$

The constants G_{sl}, G_{rr} are dependent on the geometry of the rolling bearing and are presented in Appendix A.

The sliding friction torque (Eq. (7)) is dependent on the weighting factor (Eq. (8)) and on the reference values of the coefficient of friction (boundary film coefficient of friction – μ_{bl} and full film coefficient of friction – μ_{EHD}) for each oil.

$$\mu_{sl} = \phi_{bl}.\mu_{bl} + (1-\phi_{bl}).\mu_{EHD} \quad (7)$$

$$\phi_{bl} = \frac{1}{e^{2,6\times 10^{-8}(n.\upsilon)^{1.4}d_m}} \quad (8)$$