Tribology International 88 (2015) 309–316

Contents lists available at ScienceDirect

Tribology International

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

Gearbox power loss. Part II: Friction losses in gears

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
1 December 2014
Accepted 2 December 2014
Available online 18 December 2014

Keywords:
Gears
Gearboxes
Power loss
Wind turbine gear oils

ABSTRACT

The second part of the study presents an extensive campaign of experimental tests in an FZG test rig. An average coefficient of friction between meshing gears was devised from the experimental results. Several aspects regarding the meshing gears power loss are discussed such as gear loss factor, coefficient of friction and the influence of gear oil formulation (wind turbine gear oils).

1. Introduction

In the first part of this work, a rolling bearing torque loss model was calibrated for TBB (thrust ball bearings – 51107) and RTB (cylindrical roller thrust bearings – 81107 TN) and for several wind turbine gear oil formulations. The model was then applied with success to predict the torque loss in gearbox rolling bearings, in particular those used in the FZG machine slave and test gearboxes.

The second part of this work is dedicated to the analysis of the friction torque loss in helical gears lubricated with the same wind turbine gear oils used in Part I, under oil jet lubrication at 80 °C.

In this part, tests performed with a FZG test machine will be presented and discussed. The tests allow to validate the lubricant parameter determined for each wind turbine gear oil and presented in [1].

The authors calculate the meshing gear power loss considering an average coefficient of friction along the path of contact. For such situation the gear loss factor have a significant influence in the quantification of meshing gears power loss. So, the gear loss factor will be discussed and a method validated for a wide range of gear geometries will be also presented and discussed.

2. Gearbox power loss model

The power losses occurring in a gearbox are generated through different mechanical sources [2]. In this work the loss sources that were considered are represented in Fig. 1. Take note that the gears losses are divided into load dependent (P_{VZP}) and load independent losses (P_{VZ0}).

2.1. No-load gears power loss

Depending on the input power and speed, lubricant characteristics, and gearbox design, the no-load gear power losses usually are a very important source of energy dissipation. Due to an almost infinite combination of gearbox design choices and operating conditions, it is very difficult to develop a simple and general formulation to evaluate these power loss mechanisms.

Since the main objective of this work was to measure and predict accurately the friction torque loss in the meshing gears, the no-load gear power loss was determined experimentally for each operating speed and gear oil formulation, at the operating temperature of 80 °C, using a special testing procedure.

The overall torque loss in the slave and test gearboxes of the FZG machine were measured at very low input torque (FZG load stage 1) and for a wide range of operating speeds. Under these conditions the friction power loss in the meshing gears was assumed to be null. Thus, for any input torque (load stage *i*) the overall power loss is given by the following equation:

$$P_V^i = P_{VZ0}^i + P_{VZP}^i + P_{VL}^i + P_{VD}^i \quad (1)$$

For load stage 1 (low input torque, $T_W = 4.95$ Nm) Eq. (1) becomes

$$P_V^1 = P_{VZ0}^1 + P_{VZP}^1 + P_{VL}^1 + P_{VD}^1 \quad (2)$$

The term P_V^1 is determined experimentally at load stage K1.

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

http://dx.doi.org/10.1016/j.triboint.2014.12.004
0301-679X/

Nomenclature

a	axis distance (mm)
b	tooth face width (m)
H_V	gear loss factor (–)
l^i	length of contact of a tooth (mm)
m	module (m)
p_0	Hertz pressure (centre of the contact) (MPa)
p_b	transverse pitch (mm)
R_a	arithmetic average roughness of pinion and gear (μm)
T_L	torque loss (Nm)
u	gear ratio (z_2/z_1) (–)
v_t	pitch line velocity (m/s)
x	addendum modification (–)
z	number of teeth (–)
α	pressure angle
α_p	piezoviscosity
β	helix angle
ϵ_α	transverse contact ratio (–)
ϵ_β	overlap contact ratio (–)
ν_{oil}	oil kinematic viscosity at operating oil sump temperature (mm^2/s)
μ_{mZ}	coefficient of friction on meshing gears (–)

As the concept suggests, the no-load gears losses are independent of the load which gives the following equation:

$$P^i_{VZ0} = P^1_{VZ0} = P_{VZ0}, \quad \forall i \tag{3}$$

For load stage 1 it was assumed that

$$P^1_{VZP} \approx 0 \tag{4}$$

since the corresponding meshing torque loss (T^1_{VZP}) at the operating speed is lower than the precision of the torque cell (ETH Messtechnik DRDL II) used to measure the overall torque loss in the slave and test gearboxes of the FZG machine.

The power loss in the rolling bearings (P^1_{VL}) of the slave and test gearboxes are calculated using the model developed in Part I of this work.

The power loss in the seals is evaluated using Eq. (5) given in Ref. [4] by Freudenberg and is independent of the load applied:

$$P^i_{VD} = P^1_{VD} = P_{VD}, \quad \forall i \tag{5}$$

Finally, for any load stage 1 Eq. (1) becomes

$$P^1_V = P_{VZ0} + P^1_{VL} + P_{VD} \tag{6}$$

Thus with the Eq. (7) is possible to determine the no-load gears loss (P_{VZ0}),

$$P_{VZ0} = P^1_V - P^1_{VL} - P_{VD}. \tag{7}$$

2.2. Load dependent power loss in meshing gears

Ohlendorf [5] introduced an approach for the load dependent losses of spur gears. The power loss generated between gear tooth contact can be calculated according to Eq. (8),

$$P_{VZP} = P_{IN} H_V \mu \tag{8}$$

where H_V represents the gear loss factor which is determined according to Eq. (9), and assuming that the coefficient of friction (μ_{mZ}) is constant along the path of contact. In fact, this is a simplification of the problem.

2.2.1. Gear loss factor (H_V)

Eq. (8) can be used to calculate the average friction power loss between gear teeth, given the correct gear loss factor H_V. Despite

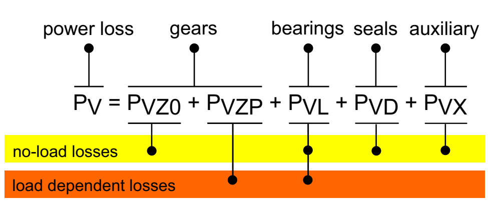

Fig. 1. Power loss contributions [3].

considering β_b, Eq. (9) initially proposed by Ohlendorf [5] is mostly valid for spur gears [1]:

$$H_V^{ohl} = (1+u)\frac{\pi}{z_1}\frac{1}{\cos\beta_b}(1-\epsilon_\alpha+\epsilon_1^2+\epsilon_2^2) \tag{9}$$

The load between gear teeth along the meshing line can be calculated just considering that the load per unit of length is given by Eq. (10), the coefficient of friction is assumed constant along the path of contact and the elastic effects were disregarded. With these conditions, the gear loss factor can be obtained by application of Eq. (11) proposed by Wimmer [6]:

$$F_N(x,y) = F_n \cdot \frac{1}{\sum_{i=1}^{n} l^i(x)} \tag{10}$$

$$H_V^{num} = \frac{1}{p_b}\int_0^b \int_A^E \frac{F_N(x,y)}{F_b} \cdot \frac{V_g(x,y)}{V_b}\, dx\, dy \tag{11}$$

Niemann and Winter [7] also proposed a gear loss factor that is shown in the following equation:

$$H_V^{Nie} = (1+u)\frac{\pi}{z_1}\frac{1}{\cos\beta_b}\epsilon_\alpha\left(\frac{1}{\epsilon_\alpha}-1+(2k_0^2+2k_0+1)\epsilon_\alpha\right) \tag{12}$$

Buckingham [8] also introduced a formula for the efficiency of a meshing gear pair. A gear loss factor (Eq. (13)) can also be derived from this approach:

$$H_V^{Buc} = (1+u)\frac{\pi}{z_1}\frac{1}{\cos\beta_b}\epsilon_\alpha\left(2k_0^2-2k_0+1\right) \tag{13}$$

Velex et al. which did no *a priori* assumption on tooth load distribution by using generalized displacements, in order to calculate the efficiency of a meshing gear pair, obtained a closed form solution for the efficiency of a meshing gear pair (constant coefficient of friction was assumed), as presented in Eqs. (14)–(16).

$$\rho = 1-\mu\cdot(1+u)\cdot\frac{\pi}{z_1}\cdot\frac{1}{\cos\beta_b}\cdot\epsilon_\alpha\cdot\Lambda(\mu) \tag{14}$$

with

$$\Lambda(\mu) = \frac{2k_0^2-2k_0+1}{1-\mu\cdot\left(\dfrac{\tan\alpha_t\cdot(2k_0-1)-\dfrac{\pi}{z_1}\epsilon_\alpha\cdot(2k_0^2-2k_0+1)}{\cos\beta_b}\right)} \tag{15}$$

where

$$k_0 = \frac{z_1}{2\pi\cdot\epsilon_\alpha\cdot u}\left(\left(\left(\frac{ra_2}{rp_2}\right)^2\frac{1}{\cos\alpha_t^2}-1\right)^{1/2}-\tan\alpha_t\right) \tag{16}$$

It turns out that Eq. (13) suggested by Buckingham is an approximation of the one suggested by Velex and Ville [9] when the coefficient of friction $\mu \ll 1$.