Tribology International 101 (2016) 204–217

Contents lists available at ScienceDirect

Tribology International

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

Coefficient of friction equation for gears based on a modified Hersey parameter

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ARTICLE INFO

Article history:
Received 22 December 2015
Received in revised form
29 February 2016
Accepted 25 March 2016
Available online 9 April 2016

Keywords:
Coefficient of friction
Gears
Hersey parameter
Power loss

ABSTRACT

The present work is first of all a bibliographic revision about the coefficient of friction equations for gears. The available expressions were compared and used to predict experimental results trying to draw an overview about the main advantages and disadvantages of each method.

A load sharing function to predict the coefficient of friction was developed based on a modified Hersey parameter. A coefficient of friction equation is then proposed trying to improve the existing methods, including the influence of the oil pressure–viscosity parameter, on the coefficient of friction.

The new equation correlates very well with the experimental results.

1. Introduction

The fuel economy is becoming of great importance for modern civilisations. The actual consumption of fuels is destroying the atmosphere due to the amount of CO_2 emitted. In order to consume less fuel it is necessary to put some effort on the machine's efficiency. From the daily car up to a modern wind turbine, a gearbox or other kind of mechanical device is usually used to achieve a desired operating condition. In order to reduce the economic and environmental impact of fuel consumption, more efficient gears are needed since they are the most relevant source of power loss inside a gearbox, both due to load and no-load losses [1].

The first studies of efficiency in gear transmissions were performed by Weisbach and Gordon [2] and Reuleaux [3] in the 19th century. Around 1950, Earl Buckingham [4] measured the power loss in gears and developed formulas to evaluate the friction losses.

Several authors [1,5,6] presented experimental results concerning power loss of gears. Petry-Johnson et al. [5] presented an experimental investigation of spur gear efficiency with different geometries and different surface finishing and showed that lower modules promoted a reduction in the meshing gear power loss and that a chemically finished gear had better efficiency than a grounded surface gear.

An analytical and experimental investigation on the effects of geometry on sliding losses of spur gears was presented by Yenti et al. [6] showing that decreasing the module can increase the efficiency of a spur gear pair. Additionally, they showed that the pressure angle increase also reduces the meshing gears power loss. Yenti et al. used the Palmgren model to estimate the rolling bearing losses and concluded that the DIN 3990 [7] coefficient of friction is reliable to predict the actual gear meshing power loss.

From the published works it is possible to analyse the influence of the operating conditions on the coefficient of friction. Martins et al. [8] showed that the coefficient of friction decreases with increasing rotational speed and increases with increasing load. Naruse [9] showed that the coefficient of friction is insensitive to changes in the surface roughness (R_a) between 0.5 and 3 μm, while Xiao et al. [10] suggested that the coefficient of friction is lower for lower surface roughness. However, the Naruse and Xiao works were not totally conclusive since they performed only few experimental tests.

In the present work a new equation for the coefficient of friction of meshing gear is developed. Different expressions are presented and compared with experimental results aiming to support the new equation proposed.

2. Meshing gears power loss

A gearbox transmits a given input power (P_{in}) from the input shaft of the driving gear. The mechanism dissipates energy and the driven gear only transmits an output power (P_{out}) lower than the input power. The efficiency of such gear pair is then given by the

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http://dx.doi.org/10.1016/j.triboint.2016.03.028

Notation and units

a	centre distance (mm)
b	gear face width (mm)
h_0	central film thickness (m)
h_{0C}	corrected central film thickness (m)
h_m	minimum film thickness (m)
F_N	gear normal force in each meshing position along the path of contact (N)
f_N	gear normal force per unit contact length in each meshing position along the path of contact (N/mm)
F_{bt}	gear tangential force on the base plane (N)
H_{VL}	local gear loss factor (–)
H_p	hydraulic parameter (mm^{-1})
i	gear ratio (–)
m	gear module (mm)
n	rotational speed (rpm)
p_h	maximum Hertz pressure (GPa)
P_{in}	input power (W)
p_{max}	maximum Hertz pressure (kg/cm^2)
P_{out}	output power (W)
P_V	total power loss (W)
P_{VX}	auxiliary losses (W)
P_{VZ0}	no-load gears power loss (W)
P_{VZP}	meshing gears power loss (W)
P_{VL}	rolling bearings power loss (W)
P_{VD}	seals power loss (W)
p_b	gear base pitch (mm)
p_{bt}	gear transverse base pitch (mm)
R_a	average surface roughness (μm)
R_q	root mean square roughness (μm)
R_z	roughness based on the five highest peaks and lowest valleys over the entire sampling length (μm)
S_p	modified Hersey parameter (–)
S_g	gear geometric parameter (–)
s	pressure–viscosity parameter (–)
t	pressure–viscosity parameter (–)
v_g	gear sliding velocity in each meshing position along the path of contact (m/s)
v_r	gear rolling velocity in each meshing position along the path of contact (m/s)
v_{tb}	gear tangential velocity on the base plane (m/s)
$v_{\Sigma C}$	sum of the gear surface velocities on the pitch point (m/s)
X_L	lubricant factor (/)
x	meshing contact position along the path of contact (mm)
x_Z	gear profile shift (/)
Z	gear number of teeth (/)
α	pressure–viscosity coefficient (Pa^{-1})
α_t	thermal expansion coefficient (/)
α_Z	gear pressure angle (°)
β_b	base helix angle (°)
β_Z	gear helix angle (°)
η	dynamic viscosity (Pa s)
η_Z	efficiency of a gear pair (/)
Λ	specific film thickness (/)
μ_{bl}	coefficient of friction in boundary film lubrication (/)
μ_{EHD}	coefficient of friction in full film lubrication (/)
μ_{mZ}	average meshing gear coefficient of friction (/)
μ_Z	meshing gear coefficient of friction along the path of contact (/)
ν	kinematic viscosity (cSt)
ρ	density (g/cm^3)
ρ_{redC}	equivalent curvature radius on the pitch point (mm)

Subscripts

1, 2	pinion, wheel

following equation:

$$\eta_Z = \frac{P_{in} - P_{out}}{P_{in}} \tag{1}$$

The difference between the input power (P_{in}) and the output power (P_{out}) is called the total power loss (P_V), which is the sum of different sources of power loss. According to Höhn et al. [1] the gearbox power loss is due to gear load-dependent (P_{VZP}) and load-independent losses (P_{VZ0}), rolling bearing losses (P_{VL}), seal losses (P_{VD}) and auxiliary losses (P_{VX}) as resumed in Eq. (2) for a given load i.

$$P_V^i = P_{VZ0} + P_{VZP}^i + P_{VL}^i + P_{VD} \tag{2}$$

Note that P_{VZ0} and P_{VD} are, respectively, load-independent gear power loss and seal power loss. The seal losses are usually estimated using the Simrit equation [11]. The gear no-load losses (P_{VZ0}) are difficult to predict and models presented in the literature are only valid for a specific range of conditions, and for the moment, measuring gear no-load losses is the better solution to have a reliable power loss prediction [12–14].

The rolling bearing losses can be calculated with different models proposed in the literature. Martins et al. [8], Höhn et al. [1] and Durand de Gevigney et al. [15] used the Palmgren model [16]. Recently, Fernandes et al. [12] showed a good correlation between oil lubricated rolling bearing experiments and the SKF model [17]. Cousseau et al. [18] and Gonçalves et al. [19] also found good correlation between the SKF model and grease lubricated rolling bearings.

At each point along the path of contact, see Fig. 1, the load-dependent gear friction power loss is given by the following equation:

$$P_{VZP}(x) = F_N(x) \cdot \mu_Z(x) \cdot v_g(x) \tag{3}$$

Since the meshing starts at point A and ends at point E, the total friction power loss of the meshing gear is given by the following equation:

$$P_{VZP} = \int_A^E F_N(x) \cdot \mu_Z(x) \cdot v_g(x)\, dx \tag{4}$$

Eq. (4) is a general formulation for meshing gears friction power loss, making it necessary to know the product of force and sliding speed along the path of contact as well as the coefficient of

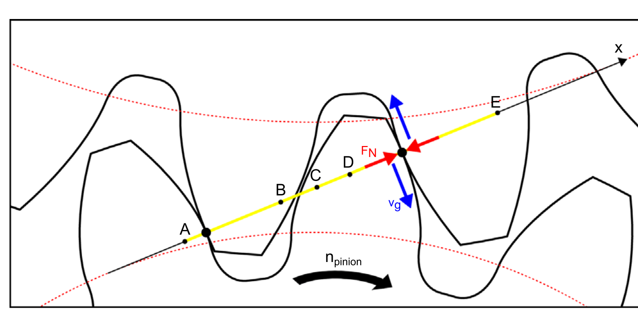

Fig. 1. Sliding speed and normal forces on the tooth contact.