Tribology International 83 (2015) 114–119

Contents lists available at ScienceDirect

Tribology International

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

The correlation between gear contact friction and ball on disc friction measurements

M. Björling [a,*], J. Miettinen [b], P. Marklund [a], A. Lehtovaara [b], R. Larsson [a]

[a] Division of Machine Elements, Department of Engineering Science and Mathematics, Luleå University of Technology, Luleå SE-97187, Sweden
[b] Group of Tribology and Machine Elements, Department of Materials Science, Tampere University of Technology, P.O. Box 589, 33101 Tampere, Finland

ARTICLE INFO

Article history:
Received 12 September 2014
Received in revised form
4 November 2014
Accepted 13 November 2014
Available online 24 November 2014

Keywords:
EHL
Gears
Friction
Ball-on-disc

ABSTRACT

Running experiments with full-size gearboxes from the actual application has the advantage of giving realistic results in terms of power losses. The drawback is extensive costs, lengthy testing, and the difficulty in differentiating between load dependent and load independent losses, and which losses are coming from the gears, seals, bearings or synchronizers. In this work, the correlation between friction measurements conducted in a ball-on-disc machine and friction measurements conducted in a back-to-back gear rig is investigated. The correlation between the gear tests and the ball-on-disc tests was reasonably good in terms of absolute values, and the shape of the friction curves was similar, indicating that the ball-on-disc measurements to a large extent are capturing the behavior of the gear contact.

1. Introduction

Reducing energy consumption and emissions have been a priority in the industrialized world for a long time, and even more so during the last 5–10 years. With the exception of bringing new technologies and solutions to the market, constant development is carried out to improve current technology. The automotive market has faced increasing restrictions in terms of emissions and has therefore spent large amounts of money on research and development. Rising fuel prices and increased environmental concern also make the customers more prone to purchase more fuel efficient vehicles. It has been assessed that 33% of the fuel energy in a car is used to overcome friction, and that 7–18% of these friction losses originates from the transmission [1]. In heavy road vehicles and buses, 33.5% of the fuel energy is used to overcome friction, and 13% of these losses originates from the transmission [2]. The losses in a gear transmission can be divided into two categories: load-dependent and load-independent losses. The load independent losses are typically viscous losses due to oil churning and are mostly governed by lubricant viscosity, density and the geometrical design of gears and housing. The load-dependent losses are due to friction in the rolling and sliding interfaces between the mating gear teeth, and are influenced by a large numbers of parameters. The total gear contact friction losses are ranging between 4.5 and 55% depending on the design and the use of the transmission [3,1]. Most gears are operating in the elastohydrodynamic lubrication (EHL) regime and the friction originating from these kinds of contacts is the matter of interest in this paper.

Running experiments with full size gearboxes from the real application has the advantage of giving realistic results in terms of power losses depending on lubricant type, load, speed and operating temperature. The drawbacks are extensive costs, lengthy testing, and the difficulty in differentiating between load-dependent and load-independent losses, and which losses are coming from the gears, seals, bearings or synchronizers. Even when a gear pair is rotating at a constant speed, several parameters are changing along the line of action between the meshing teeth, such as load, entrainment speed, and slide to roll ratio (SRR). When running tests with gears and being successful in removing all other sources of losses, only an average friction coefficient can be obtained. To remedy this problem and allow more detailed studies of gear losses both numerical and experimental methods have been used.

Several researchers have solved the numerical EHL problem to be able to predict, and understand gear friction. Such studies include both smooth [4,5] and rough surfaces [6–8]. A reliable and accurate numerical prediction model for gear contact friction would be the best alternative since the number of tests would be kept at a minimum, saving both time and money. However, EHL is a complex field, and there are as far as the authors knows no models with such true predictive capabilities to date [9]. Due to the severe running conditions in many gearboxes, highly additivated lubricants are often used which also puts demands on the numerical models to include tribochemical effects which is a tremendous challenge.

* Corresponding author.
E-mail address: marcus.bjorling@ltu.se (M. Björling).

http://dx.doi.org/10.1016/j.triboint.2014.11.007
0301-679X/

Nomenclature

ϵ_g	addendum contact ratio of gear
ϵ_p	addendum contact ratio of pinion
ϵ_t	contact ratio
μ_m	approximate gear friction coefficient
μ_{bl}	boundary lubrication friction coefficient
μ_{EHL}	sliding friction coefficient in full film conditions
μ_{sl}	sliding friction coefficient
ν	kinematic viscosity at operating temperature of oil (mm^2/s)
φ_{bl}	weighting factor for the sliding friction coefficient equation
φ_{ish}	inlet shear heating reduction factor
φ_{rs}	kinematic replenishment/starvation reduction factor
D	bearing outside diameter (mm)
d	bearing bore diameter (mm)
d_m	bearing pitch diameter (mm)
F_r	radial bearing load (N)
G_{rr}	geometric and load dependent variable for rolling frictional moment
G_{sl}	geometric and load dependent variable for sliding frictional moment
H_v	gear loss factor
i	gear ratio
K_{rs}	replenishment/starvation constant
K_z	bearing type related geometric constant
M_{rr}	rolling frictional moment (N mm)
M_{sl}	sliding frictional moment (N mm)
n	rotational speed (rpm)
P_l	total gear power loss (W)
P_m	gear mesh power loss (W)
P_t	total transmitted power (W)
P_{bl}	load dependent bearing power loss (W)
P_{nl}	load independent gear power loss (W)
R_1	geometric constant for rolling frictional moment
S_1	geometric constant for sliding frictional moment
z	number of gear teeth

As an alternative to numerical predictions, many authors have used twin-disc machines to simulate power loss in gear contacts [10–15]. By controlling the rotational speeds of two rollers in contact, the same entrainment speeds and SRRs can be achieved as in the line of action of the gear system that are simulated. This approach is cheaper and less time consuming than running full gear tests, and gives more detailed information regarding gear contact friction along the line of action. The twin disc is seen as suitable for mimicking a gear contact also due to the fact that both twin disc, spur and helical gears to some extent operate with line contacts. A ball on disc tribotester does not suffer from the same aligning problems encountered in a twin disc machine using disc profiles creating a pure line contact, and may be available at research facilities not having a twin disc machine. It is however unclear if it is possible to correlate the friction coefficient in the circular contact in a ball on disc tribotester to the line contact in the spur gear contact. The purpose of this work is to investigate the correlation between friction measurements conducted in a ball on disc machine with friction measurements conducted in a FZG gear test rig. In addition, using the earlier presented concept of friction mapping [16], a method is proposed to predict the friction coefficient in an arbitrary spur gear pair from a minimum of measurements in a ball-on-disc machine.

2. Overall methodology

The following sections cover the test rigs, test specimens and lubricants used in the experiments. It also contains information about how the experiments were performed and how the data was processed and evaluated.

2.1. Ball-on-disc tribotester

The experiments were carried out with a Wedeven Associates Machine (WAM) 11, ball on disc test device. The lubricant is supplied at the centre of the disc in an oil dispenser that distributes the lubricant across the disc surface. The lubricant is circulated in a closed loop from the oil bath, through a peristaltic pump to the oil dispenser at the centre of the disc. The peristaltic pump is delivering approximately 180 ml/min. Three thermocouples are used in the test setup, one located in the oil bath, one in the outlet of the oil supply and one trailing in the oil film close to the inlet region of the ball on disc contact. A more thorough description of the test rig and its features is presented in previous work [16].

2.2. Gear test rig

A modified FZG test rig was used for the gear tests, as depicted in Fig. 1. The test gears, described in Section 2.3, were located in two separate housings with their own lubrication system with a capacity of 25 l each. The gears were spray lubricated with a flow rate of 2.0 l per minute directed in the entry side of the mesh. The loading of the gears were done by applying a torque on shaft 1 with the help of a rod and dead weights. The corresponding strain caused by the twist of the shaft was measured with a full bridge strain gauge system. The power circulating design of the test rig means that the electric motor was only compensating for the energy equivalent to the losses in the system. However, the gear friction losses were calculated by the friction moment measured by a torque meter on shaft 2.

2.3. Test specimens and lubricants

The test gears are made of case hardened steel, 21 NiCrMo2-2. The gears were case hardened to a depth of 1.1 ± 0.5 mm and a hardness of 58 ± 2 HRC. The test gears were subjected to grinding and polishing, down to a surface roughness of around 30 nm RMS measured with a stylus mechanical profilometer. This gives a combined RMS roughness for the gear pair of approximately 42 nm. Both pinion and gear have the same properties as shown in Table 1, which means that the gear ratio is 1.

The ball-on-disc tests were performed with specimens made of DIN 100Cr6 (AISI 52100) bearing steel. The specimens have been measured to a surface roughness, RMS of 25 nm for the balls and

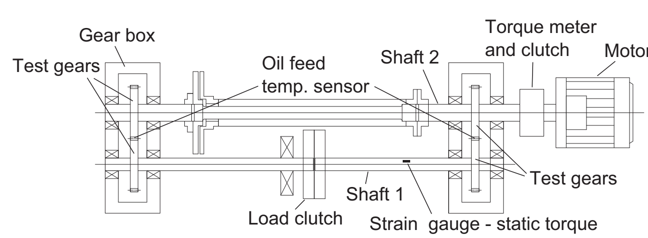

Fig. 1. FZG test rig.