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Experimental simulation of gear contact along the line of action

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ABSTRACT

In highly loaded gears, lubricated rolling/sliding contact conditions change greatly along the line of action. This leads to variation in gear frictional properties and to failures such as pitting and scuffing that take place in different positions along the tooth flank. Information on instant contact behavior is therefore very useful, but this kind of measurement in real gears is extremely complicated. Single spur gear geometry has been simulated at 38 steady-state measuring points along the line of action using a twin-disc test device focusing on the friction coefficient and on temperature and lubrication conditions. Twin-disc simulations were adjusted to match real gear experiments by using similar maximum Hertzian pressure and surface velocities. The results show that the curve shapes for the mean friction coefficient as a function of pitch line velocity are similar to the corresponding experimental results with real gears. Further, the calculated thermal Λ-values of real gears and the measured mean contact resistance correspond well. This approach shows potential for simulating gear friction and failure mechanisms along the line of action.

1. Introduction

Highly loaded lubricated rolling/sliding contacts are common in various machines and machine elements. Typical examples are rolling bearings and gears. In the case of gears, contact conditions change greatly along the line of action. This leads to variations in the gear frictional properties and to failures such as pitting and scuffing that take place at different positions along the tooth flank. Information on instant contact behavior is therefore very useful in gaining a deeper understanding of these phenomena. This provides the basic information for estimation of gear train power losses and their lifetime.

Very often spur gear profiles are approximated by cylinders with the same radius of curvature as the gear teeth at the instant contact point, as shown in Fig. 1. This provides the basis of the twin-disc test devices, which typically represents only a single point along the line of action in real gears at constant load and speed conditions. In addition, Höhn et al. [1] list three main differences when gear and twin-disc contact is compared: (1) they have different kinds of surface topography and orientation, (2) film formation is continuous in discs but on gears a new oil film has to build up with every new engagement and (3) dynamic effects on discs are significantly smaller than in gears. Usually the arithmetic mean surface roughness (R_a-value) in discs is around 0.1 μm, which is four times less than the R_a-values in modern gears. Gear grinding is always done perpendicular to the rolling direction, while disc surfaces are nearly always finished along a line parallel to the rolling direction. The grinding direction affects the lubricant film thickness [2]. The manufacturing tolerances are smaller for twin-disc than for real gears.

Twin-disc test devices have been used widely to study gear related issues such as the influence of surface roughness on friction [2–4], lubricant rheological properties [5–11], development of coatings [12] or different kinds of surface failures [13–17]. Comparisons have been made between twin-disc and gear surface contact behavior in terms of film thickness, pressure and temperature distribution as well as in terms of surface failures [18–20]. In many gear power loss studies twin-disc results have been used as a reference measurement to verify calculated friction results [1,21,22]. However, the vast majority of the measurements have been made over a limited range of parameters, which do not cover all the conditions which exist for a gear pair contact along the line of action. This limits the comparisons which can be made between mean friction values obtained from twin-disc and those from gear tests. In real gear tests, the mean friction coefficient along the line of action is, in practice, the only friction value which can be measured.

The objective of this work is to evaluate gear contact along the line of action using a twin-disc test device focusing on the friction coefficient, the temperature and lubrication condition. An attempt is made to compare overall friction behavior in the twin-disc test device with that in real gear contacts as well as to show the measured data along the line of action. The details of the measured results are discussed.

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Nomenclature

b	half-width of Hertzian contact region
F_N	normal load
F_{Nmax}	maximum normal load
h	film thickness
M_{bear}	bearing friction moment
n	rotating speed
p	Hertzian pressure
p_{max}	maximum Hertzian pressure
r	radius of test disc
R_a	surface roughness R_a value
R_{ku}	kurtosis
R_q	root mean square roughness
R_{sk}	skewness
R_z	average of ten greatest peak-to-valley
u	surface velocity
u_{max}	maximum surface velocity
x	coordinate
y	coordinate
z	coordinate
η	viscosity
φ_T	thermal reduction factor
Λ	h/σ
μ	friction coefficient
ρ	radius
σ	combined surface rms roughness
σ_{max}	max. combined surface rms roughness
ω	angular velocity
ζ	density

Subscripts

1	disc 1
2	disc 2

2. Experimental

2.1. Test device

Simulated gear friction tests were performed with a previously developed twin-disc test device, which is described in more detail in Refs. [23,24]. In a twin-disc test device each disc is driven by a separate electric motor with adjustable rolling and sliding speeds. Loading and speeds can be varied on-line with automated computer control, which allows flexible testing.

The lubrication of the test disc is performed with circulating lubrication system where oil inlet temperature and flow rate are controlled and measured. Other measured signals from the twin-disc test device include the bulk disc temperature, mean contact resistance and friction moment in addition to load and shaft rotation speeds. The disc bulk temperature is measured 3 mm below the surface with a thermocouple and the signal is transmitted from the axle using a telemetry device. The mean contact resistance measurement was introduced into the test device to analyze the contact lubrication conditions. When there is no metal to metal contact the mean contact resistance is 1 kΩ but as metal to metal contact increases the mean contact resistance approaches 0 Ω. Signals are collected on a sampling card and are used both for on-line analysis and for later processing.

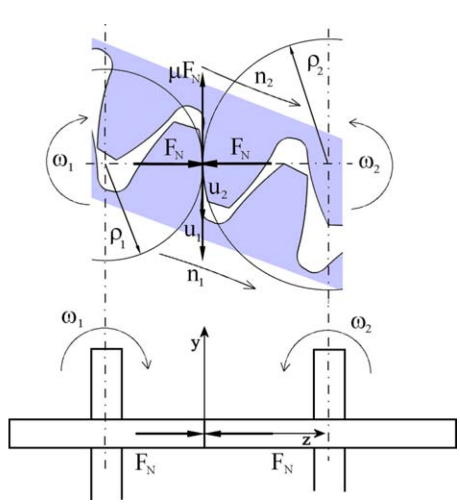

Fig. 1. Instant gear tooth contact and its approximation to equivalent cylinders.

The friction moment is measured from shaft 1 and includes components from supporting bearings and disc contact losses. Every test point was measured twice once in each sliding direction. This was carried out by reversing the speeds of the faster and slower rotating discs and it enables exclusion of the bearing losses from the measured friction moment as well as measurements of the bulk temperatures of both the slower and faster discs.

The disc contact conditions were designed to be as close as possible to real gear contacts in terms of line of action, material, grinding direction and surface roughness.

2.2. Test discs

The material for the test discs is case-hardening steel 20 NiCrMo2-2. The discs are case-hardened to a depth of 0.8–1 mm, with a specific surface hardness of 60–62 HRC. The test discs have a diameter of 70 mm and a thickness of 10 mm. They were ground in a direction perpendicular to the rolling direction to give a raised crown with a radius of 292 mm. This corresponds to the real gear flank surface topography, which is very seldom used in other studies. The grinding procedure is described in more detail in Ref. [23]. The surface roughness of the test discs was measured with a Wyko NT1100 optical profiler, which gives 3D-data from the disc surface. Table 1 shows the typical values of surface roughness parameters and their standard deviations. The parameters have been calculated from 1807 separate lines in the rolling direction of the surface, which was 1.24 mm long in the rolling direction and 6 mm wide. The surface roughness R_a-values of the test disc after the friction tests was close to 0.25 μm. This was more than half the surface roughness R_a-values

Table 1
Disc surface roughness parameters at the rolling direction.

Parameter		Mean (μm)	Std. dev. (μm)
Root mean square roughness	R_q	0.33	0.053
Average roughness	R_a	0.25	0.036
Average of the ten greatest peak-to-valley	R_z	1.58	0.27
Kurtosis	R_{ku}	4.85	3.85
Skewness	R_{sk}	−0.94	0.38