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The effect of running-in on the efficiency of superfinished gears

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ABSTRACT

Reduced gear contact losses are necessary to keep operating temperatures, as well as fuel consumption low. In this work, an FZG gear test rig was used to investigate the effect of running-in on superfinished gears with respect to efficiency. This was compared to ground gears where a higher contact pressure yielded higher efficiency. No difference was found between the two running-in procedures when analysing superfinished gears. The effect of running-in on gears decreased when the initial surface roughness was reduced, which initially had an R_a, R_z and R_{pk} value of 0.08, 0.75 and 0.08 μm, respectively. Superfinished gears showed an overall higher efficiency; however, a distinctly lower efficiency was present below 2 m/s when compared to ground gears.

1. Introduction

Geared transmissions are one of the most prolific machine elements in high power and vehicle applications, both areas for which efficiency is a paramount design criteria. For this much research has been devoted to the study of increasing and understanding gear transmission efficiency. To reduce power losses in the gear mesh, transverse gear contact ratio and gear module can be reduced [1], as well as introducing a gear tooth tip relief [2]. Related to gear mesh losses are load independent losses, which can be reduced with a lower oil level and smaller face width [3].

Furthermore, the influence of surface topography on efficiency of gear drives has been studied by Petry–Johnson and Britton [4,5], where Petry–Johnson used a back-to-back gearbox scheme to study the difference between ground and chemically polished gears at 6000 and 10,000 RPM. Both authors found an increase in efficiency when using superfinished gears. Other authors [6–8] have studied the influence of surface topography on efficiency of polished and ground surfaces in twin disc and mini traction machines and have all found an overall higher efficiency in polished specimens.

An important parameter in any lubricated contact is the λ parameter which relates the minimum film thickness to the RMS of the combined surface roughness necessary to separate the surfaces. When dealing with low roughness surfaces, Stanley et al. [9] showed that these surfaces exhibit an increase in adhesion force if film thickness was below a certain threshold when compared to rough surfaces. Zhu and Hu [10] also denotes a higher coefficient of friction for smooth surfaces in the boundary lubrication regime.

Blau [11] has indicated running-in and break-in as the process prior to steady state when two or more solid surfaces are brought together under load and moved relative to one another. Andersson was one of the first to study in depth running-in of gears [12]. He compared hobbed and shaved surfaces (shaved was smoother than hobbed) and found very little wear particles during running-in for shaved surfaces while hobbed surfaces showed a constant wear at 350,000 cycles.

Most studies on efficiency assume surface topography as a static component which does not evolve during testing. Others however [13] have shown that not only is the initial topography changing, but that its changes are depending on the manufacturing method used. This initial change in surface topography is denoted as part of the running-in process.

A higher running-in load on ground gears was shown by Sjöberg [14] to increase efficiency. The research questions in this paper are to investigate if initially smoother gear surfaces, namely superfinished, also benefit from a higher running-in contact pressure, in terms of surface roughness transformation and efficiency. Additionally this paper also addresses the following research question: are superfinished gear surfaces different in terms of efficiency over the pitch velocities of 0.5–20 m/s and pitch contact pressures between 0.92 GPa and 1.66 GPa when compared to ground gears?

2. Equipment

2.1. Gear test rig

In order to experimentally perform gear running-in and efficiency tests an FZG back-to-back gear test rig with an efficiency

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Nomenclature

α	Pressure viscosity coefficient
$\Delta_{\%}$	Power loss percentage change
η_0	Dynamic viscosity of lubricant
$\eta_{\text{mesh ground}}$	Gear contact efficiency with respect to transmitted power of reference ground gear
$\eta_{\text{mesh superfinished}}$	Gear contact efficiency with respect to transmitted power of reference superfinished gear
η_{mesh}	Gear contact efficiency with respect to transmitted power
λ	Lubrication factor
μ_{mean}	Mean friction of gear contact
ω	Angular velocity
b	Face width of gear
E_r	Equivalent Young 's modulus $E_r\pi\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)^{-1}$
G	Material parameter
G^*	Dimensionless material parameter
H_v	Ohlendorf factor relating friction to input power
h_{min}	Minimum film thickness
M_{rr}	Torque loss from rolling friction in a bearing
M_{sl}	Torque loss from sliding friction in a bearing
R	Equivalent radius
R_q	RMS of the surface topography
T_1	Pinion inside power loop torque
$T_{\text{bearing total}}$	Equivalent torque loss from all bearings
$T_{bearing}$	Torque loss from a bearing
$T_{load-dependent}$	Load dependent torque loss
$T_{load-independent}$	Load independent torque loss
T_{mesh}	Equivalent gear contact torque loss
T_{total}	Measured torque loss
U	Surface velocity $U=U_1+U_2$
u	Contact ration
U^*	Dimensionless velocity parameter
V_+	Entrainment speed
W	Contact load
W^*	Dimensionless load parameter

test setup was used, Fig. 1. In all tests, new sets of gear pairs were used in the test gearbox (1) and slave gearbox (3). To load the gears, dead-weights were hung on the load clutch (2). To evaluate the efficiency, a speed and torque sensor (4) measured the input torque (torque loss) to the power loop from the motor (5).

As new pairs of gears were used in each test, the test rig had to be disassembled and reassembled again. To change gears in the slave gearbox, the motor and speed and torque sensor were removed first. Both gear pairs were then changed by removing the two front covers on the two gearboxes. The opposite procedure was followed to reassemble the rig. The same assembly procedure was used for all tests.

Previous work performed by the authors [15] was used to evaluate the results of the efficiency tests by investigating the effect of assembly on the measured total torque loss. Mesh efficiency due to assembly was shown to vary between 0.036% and 0.053%, at inside power loop torque of 94 Nm, over the entire tested pitch velocities of 0.5–20 m/s. Furthermore, the torque sensor (4) in Fig. 1 has an measurement uncertainty of 0.08 Nm.

2.2. Gears

The gears used for running-in and efficiency tests were modified FZG C-Pt type gears, main parameters for these gears are shown in Table 1. The main difference between these gears and standard FZG C-Pt [16] gears is the inclusion of tip relief.

The gear specimens in this paper were first ground, all in the same batch, and later superfinished. Superfinishing in this paper refers to a chemical mechanical process in which the gears are subject to in order to decrease their surface roughness but still keep their geometrical and micro-geometrical properties.

2.3. Lubricant

Both gearboxes were dip lubricated with a polyalphaolefin (PAO) with a density of 837 kg/m^3 and with nominal viscosities of 64.1 cSt at 40 °C and 11.8 cSt at 100 °C. An amount of 1.5 l of lubricant was used in both gearboxes. The oil level was up to the centre of the shafts, 103 mm up from the bottom of the gearboxes. The lubricant was filtered through a filter with a pore size of 8 µm before it was used. The oil was also tested initially, after running-in and after efficiency testing, and no measurable difference was found in its dynamic viscosity utilising a cone and plate rheometer.

3. Method

In situ surface profile measurements were performed on one gear flank in the test gearbox after the gear test rig had been assembled with new pairs of gears. The running-in procedure was conducted, followed by an efficiency test. To study how the gear flank surface changes progressively, surface profile measurements on the same gear flank were made after both the running-in procedure and the efficiency test. A total of twelve gear pairs were assembled, six pairs per running-in load, making it three tests per running-in load. The whole test procedure is explained in more detail below.

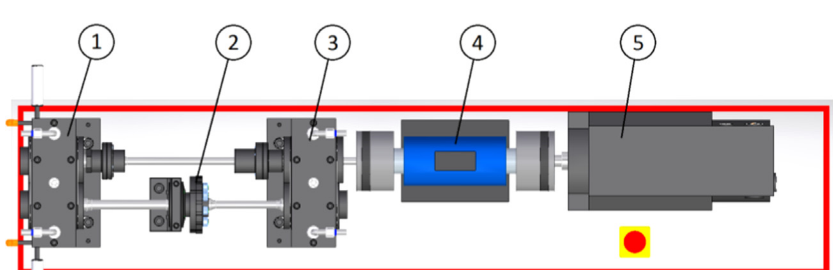

Fig. 1. Schematic of the FZG back-to-back gear test rig with its most important parts. Test gearbox (1), load clutch (2), slave gearbox (3), torque and speed sensor (4) and motor (5).