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In situ running-in analysis of ground gears

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ABSTRACT

The initial contact state between two interacting gears proves of interest due to empirical evidence indicating difference in life and efficiency in the long term due to the initial operation. Presented here is an analysis of the initial contact state of spur gears, made of case carburized 16MnCr5 steel, by the use of in situ surface measurements and friction measurements in a back-to-back test rig during the running-in cycles. Furthermore a method to estimate wear during running-in is proposed. Results show that the most significant changes in roughness and friction occurred during the first initial cycles at high load.

1. Introduction

Today, most mechanically powered machines have some sort of gear drive in them; their function, reliability and efficiency are a necessity in our everyday life. This has lead to the optimization of gears in recent decades, to develop them more compact, more efficient and last longer in order to apply them to ever more demanding conditions. Due to this, design criteria have become more stringent and pushed the limits of materials and technology.

One important parameter which determines gear performance is the initial operating condition, which is sometimes referred to as running-in. Andersson [1] studied the evolution of running-in of hobbed gears and found that running-in, from a wear perspective, corresponded to around 300 000 cycles. Schlenk et al. [2] propose a scuffing measuring procedure for gear oils in part based on work by Michaelis [3] which showed that a step wise loading during the running-in procedure increased the scuffing resistance of gears. Akbarzadeh and Khonsari [4] suggested a decrease of surface roughness in gears based on simulations and twin disc experiments. Martins et al. [5] studied the evolution of gear surface roughness by weighing gears and measuring the surface topography by disassembling the test gearbox, and measuring iron debris from oil samples. They show measurable mass loss during 90 000 cycles at low load, in which they defined as their running-in phase. All these researchers utilized a running-in procedure or tried to simulate running-in in terms of a theoretical model without observing running-in in situ as it happens in gears both from a wear and friction perspective.

Other systems have, however, been analyzed to try to determine the mechanism in which running-in occurs. Bengtsson and Rönnberg [6] used a reciprocating tribotester to investigate running-in in a purely sliding contact and found a correlation between different parameters and the degree in which the surface was ran-in. They showed that while no change in R_{sk} and R_{ku} was noted, changes were observed in R_a R_{max} and R_q. Collaborating, in part, to this, Jeng et al. [7] analyzed running-in in engine bores and found that R_a, R_q and R_{sk} show a transformation during the running-in process. Bosman simulated and studied running-in on a pin-on-disc arrangement in order to simulate what occurs during running-in in a purely sliding contact [8]. Sjöberg and Andersson studied the effect of running-in in a mini traction machine while comparing ground and a polished surface [9]. More recently Furustig simulated running-in for hydraulic motors by decoupling macro-geometrical wear from surface topography wear [10]. The plastic deformation component of asperities was studied in situ by Berthe et al. [11] in rolling contact for the first 20 cycles, and it was shown that the running-in process stabilized after 10 cycles.

Perhaps one of the most outspoken researchers devoted to the study and defining of running-in has been Blau [12,13]. He has defined running-in by encompassing both a friction and a wear component which not necessarily occur simultaneously. To further add to the complexity, he has defined wearing-in as the wear component of running-in. Other researchers for example [14,15], have defined running-in as mild-abrasion wear in which only the tops of the asperities have been removed leaving the bottom valleys untouched.

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Nomenclature

α_w	working pressure angle
$\overline{Z}$	mean of the surface profile
β_b	helix angle
Δ	difference between mean lines of profiles
$\dot{W}_{BearingLoad}$	load dependant power loss for a specific bearing
$\dot{W}_{BearingTotal}$	equivalent torque loss from all bearings
$\dot{W}_{LoadInd}$	load independent gearbox power loss
$\dot{W}_{Load}$	load dependent gearbox power loss
$\dot{W}_{mesh}$	gear contact power loss
$\dot{W}_{TotalLoadBearing}$	total load dependant power loss for all bearings
$\dot{W}_{TotalLoadGear}$	load dependant power loss for a specific bearing
$\dot{W}_{total}$	measured power loss
$\mu(x)$	instantaneous coefficient of friction
μ_m	mean coefficient of friction
ω	angular velocity
ε_1	partial contact ratio of pinion AC/p_b
ε_2	partial contact ratio of wheel CE/p_b
ε_α	contact ratio AE/p_b
A	start of active profile on the line of action
b	gear width
C	pitch point on the line of action
E	end of active profile on the line of action
$F_c(x)$	instantaneous contact force
$F_N(x)$	instantaneous normal force
F_{cmax}	max contact force
L	measurement length
M_{rr}	torque loss from rolling friction in a bearing
M_{sl}	torque loss from sliding friction in a bearing
p_b	base pitch
R_a	center line average parameter
R_q	root mean square average parameter
R_z	parameter describing the mean of 5 peak-to-valley values over 5 consecutive sample lengths
R_k	parameter for the core roughness breadth values according to the Abbott–Firestone curve
R_{pk}	parameter for the asperity peak height values according to the Abbott–Firestone curve
R_{vk}	parameter for the roughness valley depth values according to the Abbott–Firestone curve
u	gear ratio
$v_s(x)$	instantaneous sliding velocity
v_t	pitch tangential velocity
V_w	wear volume
x	distance from the pitch point along the line of action
Z^n	displaced surface profile
z_1	number of teeth on the pinion
Z_Δ	difference between two profiles

All previously cited works lack clear observations of running-in in gears, both from a wear and friction perspective, and an in depth analysis of gear running-in as it happens. This paper aims to answer the question, qualitatively as well as based on surface parameters, what are the characteristics of asperity wear during gear contact. Furthermore the paper will investigate the question how friction, as a complementary indicator, behaves during running-in.

2. Method

To analyze running-in, ten tests were performed in a FZG back-to-back gear test rig. Both the evolution of the surface topography and the friction response were measured. A schematic of the FZG test rig is shown in Fig. 1. To evaluate the friction response, a speed and torque sensor (#4) measured the input torque (loss torque) to the power loop, from the motor (#5).

It is important to note that new pairs of gears were used in each test, and hence the gear test rig had to be disassembled and reassembled after each test.

The back-to-back test rig works by the recirculation of power inside the power loop. The only power added serves to keep the gears rotating, in other words the motor adds the power losses, which is used to measure the gear tooth friction. Torque is added to the power loop, consisting of the test gearbox and the slave gearbox, by keeping half the load clutch stationary while applying a torque on the other; this in turn deforms the compliant shafts and retains the torque inside the power loop. Friction in the gear contact, in this case measured indirectly from the input torque, is derived by subtracting the bearings' losses and the gear splash losses [9].

In terms of lubrication, all tests are performed with dip lubrication.

2.1. Gear geometry

The gears used for running-in and efficiency tests were modified FZG C-Pt type gears in which main geometry can be seen in [9]. The differing parameters are shown in Table 1. The difference between these gears and standard FZG C-Pt [16] gears is the inclusion of tip relief.

2.2. Gear material properties

The gear material used during these tests have the following characteristics:

- 16MnCr5 steel used. Material composition shown in Table 2.
- Case carburized, with a depth of 800–1000 μm.
- Tempered for 2 h at 180 °C.
- Surface hardness of 700–750 HV.

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.

Each gearbox was filled with approximately 1.5 l of lubricant. The amount is controlled by measuring to the center of the shaft (103 mm from the bottom). The lubricant was filtered through a filter with a pore size of 8 μm before being used. The oil was also tested initially and after running-in. No measurable difference was found in its dynamic viscosity utilizing a cone and plate rheometer.

2.4. Test plan and procedure

The condition and test number used to analyze running-in are shown in Table 3. The only difference between the two running-in