Influence of Run-in Procedures on the Formation of Anti-wear Films in Planetary Gears

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Abstract

To prevent wear in highly loaded machine elements, like gears or roller bearings, additives are added to the lubricant. Chemical reactions between additives and the machine element surface can form an anti-wear layer on the surface. This layer prevents adhesive and abrasive wear by inhibiting direct steel-steel contact of the machine element surfaces. Whether a layer is formed or not, mainly depends on the lubrication conditions and the lubricant composition. Research projects with roller bearings have shown that run-in procedures under moderate mixed friction are effectively applicable to form reaction layers. A suitable run-in procedure is characterized by speed and load adjustment and depends on the additive concentration. This paper discusses the possibilities to use run-in procedures for an effective formation of anti-wear films which can reduce wear and scuffing damages in planetary gears. A planetary gear test bench is used to compare different run-in procedures, regarding their efficiency in forming anti-wear films. Furthermore, planetary gears with and without previous run-in are tested under wear critical conditions and the occurrence of wear is evaluated. It is shown that a run-in with increased rotational speed is suitable for an improved layer formation. This layer endures severe lubrication conditions and protects the subsurface from wear.

Introduction

In order to improve the efficiency of machine components, friction losses due to churning must be reduced. This can be achieved by the use of lubricants with low viscosities. But on the other hand, the reduction of viscosity can lead to a transition from hydrodynamic to the mixed or boundary lubrication regime. Thereby, failures of the machine components can occur as a result of increased wear. In gear tooth contacts that must operate under mixed friction conditions, surface wear is an observed and always present phenomenon. In consequence of the direct material loss, the gear vibration and noise characteristics deteriorate significantly as

well as the contact stresses and load distributions. This leads to an accelerated gear failure. In terms of minimizing wear and improving the tribological performance under boundary lubrication, extreme pressure (EP) and anti-wear (AW) additives are added to the lubricant. Chemical reactions are induced between the additive and the steel surface of the machine component during operation and a protecting layer is generated on the surface, the so-called reaction layer. The direct steel-steel contact of the contacting parts is prevented, thus minimizing adhesive and abrasive wear and acting as anti-wear film. The layer has low shear strength and can reduce the coefficient of friction [1-3].
Whether an anti-wear film is formed or not, depends on the lubrication conditions and the additive concentration [4]. If the operating conditions are not sufficient to form that layer, run-in procedures can lead to a tailored formation of the reaction layer. In previous studies with roller bearings, higher rotational speeds or reduced loads were proved to be successful for a layer build-up [5]. Beyond that, the tailored films were able to prevent wear even under severe boundary lubrication conditions.
The aim of this study is to check the ability to form anti-wear films due to run-in procedures and especially if it is possible to transfer the findings from roller bearings to planetary gears. Therefore, planetary gears were run-in with varied rotational speeds. After the test, the surface was scanned for any indications of anti-wear film formation and the wear mass of each component was compared. This permits a classification of the different run-in conditions. Additionally, the reaction layer is investigated by energy dispersive X-ray (EDX) analyses.

Tribological boundary layer in highly loaded machine elements

High strains, e.g. load and temperature, induce chemical and physical processes in the contact area of machine elements in the near-surface area. This leads to modifications of the area and the formation of the so-called boundary layer. Schmaltz described a first model of the boundary layer system in 1936, which is still in used today [6]. The schematic system is shown in Fig. 1.

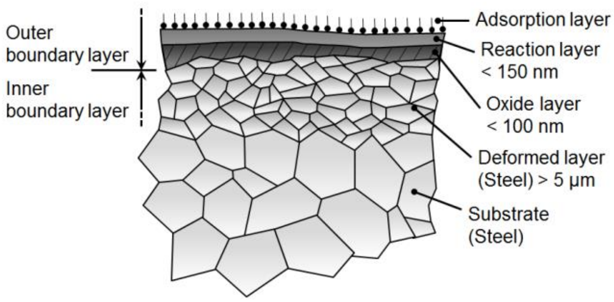


Fig. 1: Steel surface with tribological boundary layer [3]

The system can be divided into two parts, the outer and the inner boundary layer. The outer boundary layer is a thin multi-layer system. Due to surface finishing, a new machine element surface is covered by a thin oxide layer of a few nanometres (<10 nm) which is always formed in an air environment. Weak Van der Waals up to strong covalent bonds can develop between surface and lubricant components during the first contact. Beyond the initial processes, lubricants with EP- and AW-additives can form a glass-like reaction layer (<150 nm). This mostly consists of additive components which have chemically reacted with the surface [3, 4]. On top of this layer, lubricant components can be attached to the most outer layer and can help to separate both surfaces, thus preventing wear. Weakly bonded components are usually removed from the surface during sample preparation. Hence, this part of the outer boundary layer cannot be detected with most micro-analytical methods.

The inner boundary is initially formed during the surface finishing process. The microcrystalline microstructure develops due to high deformation rates. The thickness of layer ranges from 400 nm to 5 µm and its mechanical properties, like elasticity, hardness and residual stress, can differ from the ones of the base material. [3, 7]

Experimental setup

In this study, a back-to-back power recirculation-type oil-lubricated planetary gearbox was used to perform run-in as well as wear tests. Figure 2 shows the sectional representation of the test rig. The sun gears from the test gear (1) and slave gear (2) are connected by a single drive shaft (3). The planet-carriers are connected to each other through a rigid hollow shaft (4). This arrangement allows the torque to circulate inside the closed loop. An electric motor (5), connected to the main drive shaft, only needs to compensate the frictional losses. The static load is mechanically applied using a loading lever (6), connected to the annular gear from the test gear, and a hydraulic cylinder. The torsion of both, main and hollow shaft, is registered by means of strain gages, converted into electrical voltage signals and transferred via telemetry (7). The lubrication of the test rig is carried out by a bath lubrication system.

The oil level and temperature from the test and slave gears can be independently adjusted using the corresponding cooling/lubricating circuits. Furthermore, a proximity switch counts the load cycles of the main shaft.

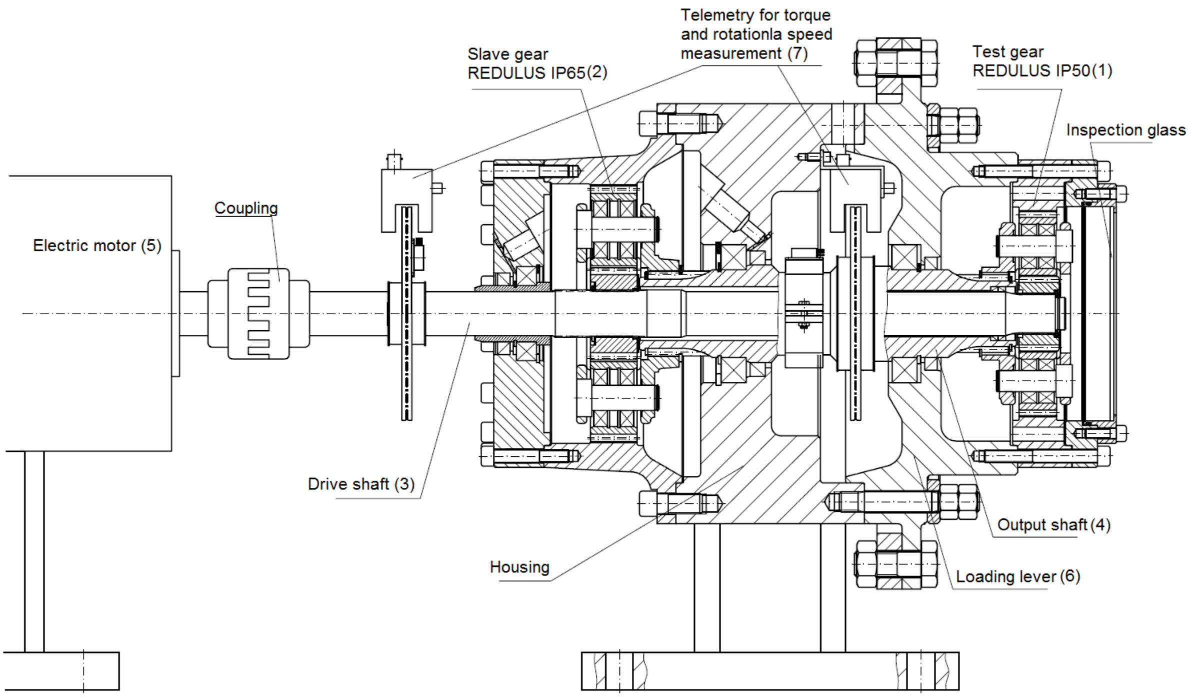


Fig. 2: Experimental setup

The test equipment is a one-stage planetary gearbox of type REDULUS IP50. Table 1 gives the specifications of the test gear.

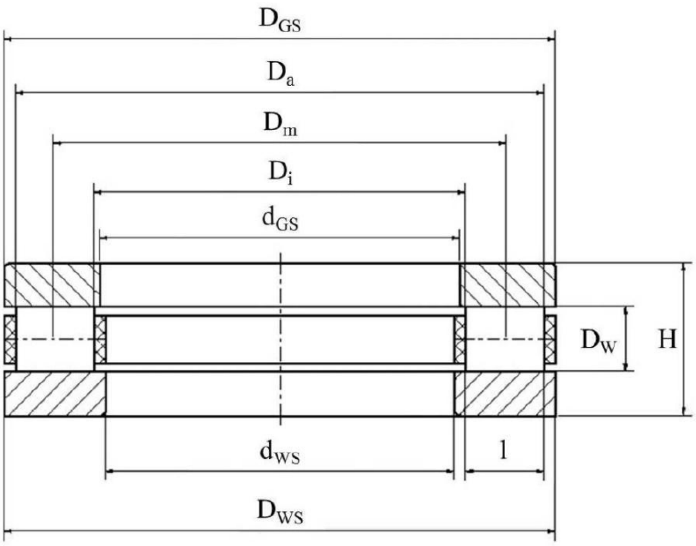
Table 1: Specification of test gears

	Sun gear	**Planet gear**	**Annular gear**
Pitch diameter (mm)	54	50	154
Module (mm)	2	2	2
Number of teeth (-)	27	25	77
Flank width (mm)	32	30	30
Addendum modification (-)	0.4278	0.5169	-1.4616
Pressure angle(°)	20	20	20
Pitch direction	Helical gear		
Material	17CrNiMo6	17CrNiMo6	42 CrMo S 4

Experimental conditions and program

Stratmann et al. [5] carried out experiments using a modified FE8-test rig. The performed run-in procedures of axial cylindrical roller bearings of type 81212 (Figure 3) effectively conveyed the formation of boundary layers. This caused a noteworthy wear reduction under critical conditions. In this study, it is examined if the findings were applicable in planetary gears.

Therefore, operating conditions of the roller bearing tests such as Hertzian pressure, lubricant film thickness and oil temperature were converted, taken over and used as initial test conditions in this investigation.



Description	Value [mm]
D_{GS}	95
d_{GS}	62
D_{WS}	95
d_{WS}	60
D_a	88
D_m	78
D_i	68
D_w	11
H	26
L	10

Fig. 3: Axial cylinder roller bearing 81212

For the calculation of the Hertzian pressure, it is assumed that the axial load is evenly distributed to all rolling elements. The formula for line contact between cylinder (rolling element) and plane (bearing race) serves as a basis for the calculation, which leads to a theoretical contact pressure. This pressure can be set in the pitch point C of the test gear through the application of a defined torque on the drive shaft.

In the present work, the lubricant film thickness in pitch point was equated to the one in the roller bearings. The lubricant film thickness of the roller bearings was determined using the approximate equation according to Dowson [8]. The required rotational speed of the test gear can be determined using Schoennenbeck's [9] formula. The approach to examine these conditions can be explained on the basis that it has already been proved that the formation of reaction layers depends on the additive concentration in contact [4]. Another decisive influence on the formation of a boundary layer is the number of times the material is rolled over, which in case of axial cylindrical roller bearings depends on test time, rotational speed and number of rolling elements. The test time was determined, so that each planet gear of the test gear experiences the same number of over rolling motions as the axial cylindrical roller bearings. In analogy to the roller bearing studies, a system temperature of 80°C was kept. As described above, the wear tests conditions were defined by converting the tests parameter of the FE8-Test according to DIN 51819-3 (Table 2).

The lubricant used in this study was a mineral oil mixed with the additive zinc dialkyldithiophosphate (ZDDP). As a common AW- and EP-additive, ZDDP is widely used in lubricants. Besides, ZDDP has specific qualities as corrosion inhibitor and antioxidant. Its anti-wear properties were studied extensively in the last years and the composition of the reaction layer is well known, but the formation process of the anti-wear layer is not yet completely understood [10].

Table 2: Wear test parameters (for constant lubricant film height)

Parameter	Roller bearing test	Planetary gear test
Rotational speed [min^{-1}]	7,5	10,4
Axial load [kN]	80	-
Torque load [Nm]	-	1610
Contact pressure [MPa]	1867,6	1867,6
Lubricant film height [nm]	2,7	2,7
Oil temperature [°C]	up to 200 (set at 80)	up to 90 (set at 80)
Striven test time [h]	80	640

These theoretical conditions cannot be set on the planetary gear test rig. The minimal rotational speed is limited by the test rig. Considering this the test time was amended accordingly, so that each planet gear of the test gear experiences the same number of over rolling motions as axial cylindrical roller bearings. Furthermore, the defined contact pressure and oil temperature were kept constant. The wear test was defined according to Table 3.

Table 3: Wear test parameters

Parameter	Planetary gear test
Rotational speed [min^{-1}]	57,0
Axial load [kN]	-
Torque load [Nm]	1610
Contact pressure [MPa]	1867,6
Lubricant film height [nm]	8,7
Oil temperature [°C]	up to 90 (set at 80)
Striven test time [h]	98

The tests were conducted using four sets of gears at two different additive concentrations. Reference wear tests were driven to examine the behaviour of the gear under wear critical conditions. Before and after the tests, the mass of the sun gear and the planet gears is measured with a precision balance in order to determine the material's mass loss under wear. In a next step, run-in tests with increased rotational speed were conducted to examine the wear behaviour and to determine if such a procedure can form a distinct anti-wear film. To analyse the effect of the additive concentration, one low and one high concentration was used. The evaluation of the tests is made by comparison of the wear mass and by assessing the film formation behaviour. The different operating conditions are shown in Table 4.

Table 4: Experimental program

Number	**Test type**	**Oil**	**Additive concentration**	**Sun rotational speed (min^{-1})**
1.0	Wear	FVA 3	0,59 % ZDDP (0,05 % Phosphorus)	57.0
2.1	Run-in	FVA 3	0,59 % ZDDP (0,05 % Phosphorus)	92.3
2.2	Run-in	FVA 3	0,59 % ZDDP (0,05 % Phosphorus)	176.4
2.3	Run-in	FVA 3	0,59 % ZDDP (0,05 % Phosphorus)	350.0
3.0	Wear	FVA 3	1,77 % ZDDP (0,15 % Phosphorus)	57.0
4.1	Run-in	FVA 3	1,77 % ZDDP (0,15 % Phosphorus)	176.4
4.2	Wear	FVA 3	1,77 % ZDDP (0,15 % Phosphorus)	57.0

Results and discussion

Tests 1.0 and 3.0 represent the reference tests to study and compare the wear behaviour of different additive concentrations. The tests 2.1 to 2.3 and 4.1 were conducted as run-in tests to assess the impact of different lubrication conditions on the formation of an anti-wear film. Finally, test 4.2 should prove the stability of tailored anti-wear film which is formed during a run-in procedure.

After the first run-in, the gears of test 2.1 showed no visible discoloration on the tooth flanks, which would be a good indication for the formation of a reaction layer, but rather a highly polished appearance that indicates wear. Therefore, the rotational speed was doubled in test 2.2. After test completion, light blue coloured tooth flanks were recognizable and the wear mass was slightly reduced. Test 2.3 was driven with 350 rpm, the highest speed, but no further improvement was observed compared to test 2.2. This result is in contrast to the findings of the roller bearing studies, which have shown that higher rotational speed leads to a faster anti-wear film formation [5].

The tests with low additive concentration showed no significant film formation. Therefore, the concentration was tripled, which is expected to have a positive influence in the formation of a boundary layer [4]. The measured wear mass of the reference test 3.0 showed a significant wear reduction of about 70% at the sun gear compared to test 1.0. This can be traced back to the higher additive concentration, even though a distinct film formation is not visible. A run-in procedure with similar test conditions to test 2.2 was performed. After completion of the run-in test 4.1 a dull grey/blue squared-shaped area appeared above the pitch circle, near the tooth tip of the sun gear. This indicates that only minimal wear occurred in this area during the run-in and an anti-wear film could have been built-up. The stability of this potential film was proven through test 4.2 under wear test conditions. This area, according figure 4, was still present after completion of the wear test 4.2 and showed the lowest visible surface smoothing. After verification by means of contact pattern paint, it was not possible to trace this phenomenon back to an uneven load-carrying pattern.

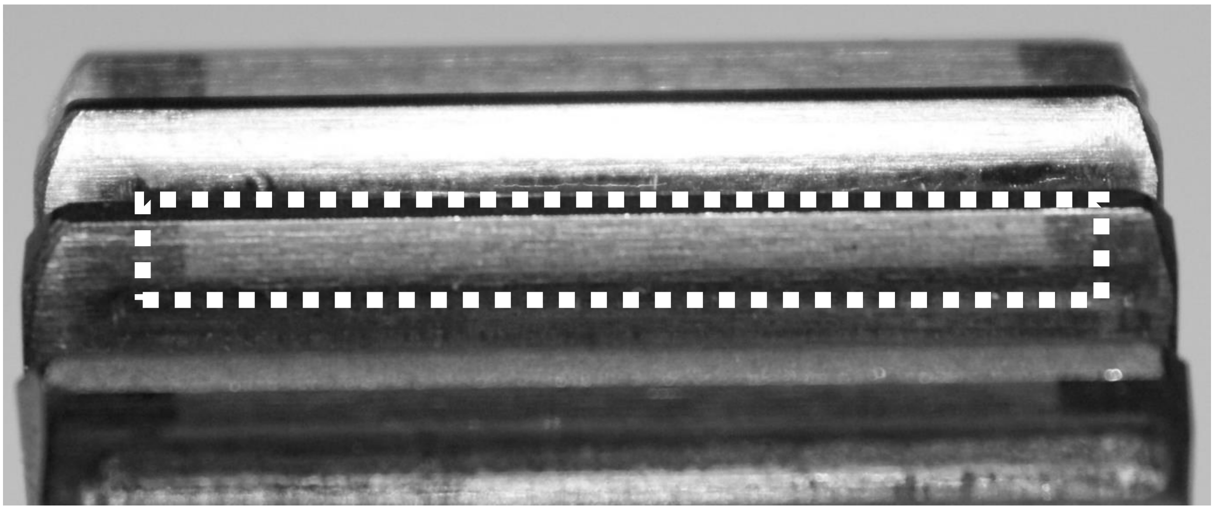


Fig. 4: Sun gear from test 4 after run-in and wear test

A chemical characterization of the tooth surface was carried out using energy dispersive X-ray spectroscopy (EDX) to verify whether a film was formed or not. Figure 4 shows the spectrum of the EDX-Analysis of both areas.

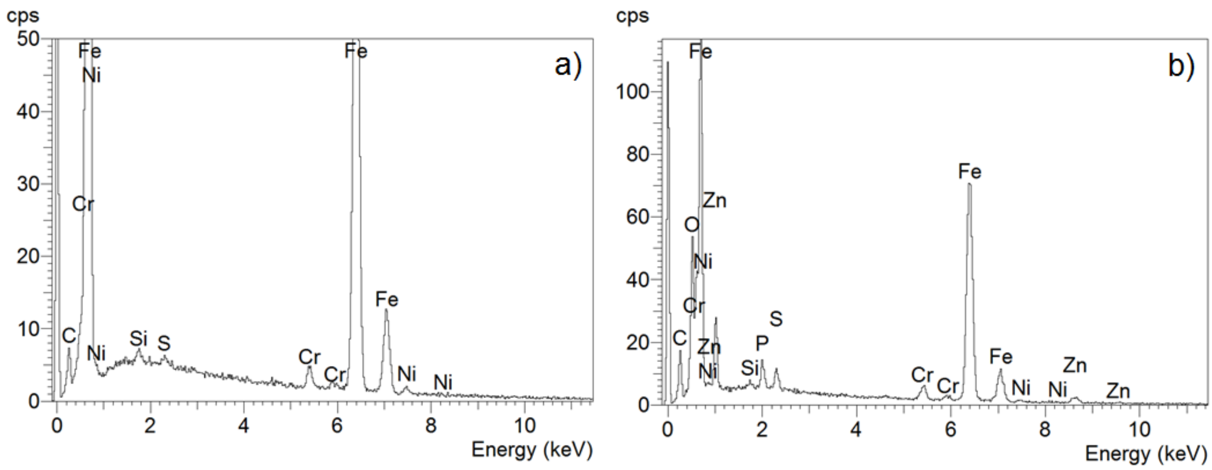


Fig. 5: EDX-Analysis from area under (a) and above (b) the pitch circle.

It is recognizable that the area near the tooth tip contains higher levels of zinc, phosphorus and sulphur than the smoothed area. This suggests the successful formation of a boundary layer, which could explain the locally reduced surface wear. This improved layer formation could be explained by the fact that the absolute value of the sliding speed increases proportionally with the distance between intervention and pitch point. The higher sliding speed could lead to favourable conditions (e.g. higher lubrication film heights), conveying the layer formation. Further tests must be conducted to prove these findings and to determine which lubrication condition parameters have the greatest influence on the build-up of an anti-wear film.

Conclusions

The influence of run-in procedures on the formation of anti-wear films in a planetary gear was investigated in this study. It can be concluded from the findings that the run-in conditions used in a test, which yielded the best results in the roller bearing experiments, did not lead to the formation of a boundary layer. There are indications that increasing the rotational speed, to a certain extent, has a positive effect on the layer formation. As this influences not only several kinematic parameters but also the lubrication conditions, a specific relation between rotational speed and film formation cannot be given yet. However, a test showed a possible dependence of the layer formation on the sliding speed along the tooth flank. By use of the knowledge acquired, further run-in conditions can be derived from the cylindrical roller thrust bearings tests under consideration of the high sliding ratio due to their kinematic characteristics. The rotational speed of the test gear can be determined by equating the maximal sliding speeds of both, roller bearing and planetary gear. It can be concluded that the formation of a boundary layer can be achieved through the selective adjustment of the test parameters. The optimal run-in conditions which lead to a boundary layer on the whole contact area of the test gear need further consideration and testing.

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