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Development of an Oil Free Water-Based Lubricant for Gear Applications

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Abstract: The aim of the current research is to develop a gear transmission fluid based on water and plant extract. Up to now, mineral or synthetic oils are used as lubricants in most gear drive applications. These oils are made of fossil raw materials and are non-biodegradable. Lately, there have been some efforts made to develop lubricants as an alternative to conventional lubrication systems such as triglycerides from native oils or synthetic esters. These lubricants are more biodegradable than mineral oils but also show some difficulties during performance like saponification. Within a former research project, the company Carl Bechem GmbH and the Fraunhofer Institute for Process Engineering and Packaging IVV developed a cutting fluid based on water and plant extract. With a model sample of this fluid, which also contained gear typical additives, preliminary experimental investigations for a current research study were conducted. The results confirmed the general suitability of this water-based lubricant for gear drives under certain operating conditions. Using water as lubricants can lead to some improved characteristics due to the very high thermal conductivity and the reduced friction. This paper aims to point out the benefits of using water-based lubricants, share the results of some preliminary experimental investigations on a fluid sample based on water and plant extract, and discuss the challenges, which one has to overcome during the development of such new lubricants.

Keywords: water-based lubricant; plant extract; gear transmission fluid; biodegradable lubricant; fire protection; reduced friction; high thermal conductivity

1. Introduction

Within the last 20 years, the production of mineral oil has been increasing constantly. About one million tons of lubricants are used in Germany per year [1]. Only about half of the used oil is collected [2]. As a result, the other half of about 500,000 tons reaches into the environment. In order to reduce environmental risks, lubricants, which are biologically harmless and quickly biodegradable, are needed. A high demand of such environmentally-friendly lubricants occurs especially in gear applications located in critical environmental terms like boats or harbors. In addition to that, such a lubricant is beneficial in the agriculture and forestry sector or in food technologies.

Lubricants based on synthetic ester-like TMP are already used as an environmentally friendly solution. Previously, native oils from rape or polygylcols were mainly used as biodegradable lubricants [3–6]. These oils have to be modified chemically or incorporated with additives [6]. Additionally, these lubricants showed some difficulties during performance. For example, lubricants based on TMP ester deal with the danger of hydrolysis when in contact with water and saponification at high temperatures. Rape oil and TMP oleate can resinify. Native oil, in general, ages very easily and polyglycol can damage the material of sealings [3,6–8]. Recent studies on the influence of water based lubricants on wear and friction [9–11] and on needed additives in water-based lubricants [12,13] show high demand for this subject.

In environmentally critical conditions or applications where special fire protection is needed, a lubricant based on water and plant extract seems to be highly advantageous. The main question is whether it is possible to develop an applicable oil-free lubricant with typical characteristics to fulfill the requirements of the tooth flank contact for different gear applications.

In gearing systems, high pressures and temperatures can take place in the tooth flank contact. The usage of hardened gears allows a high contact pressure of the tooth flanks. However, a high load in the tooth flank contact can lead to damages on the flank-like scuffing, pitting, and micro-pitting. In general, the load-carrying capacity of gears can be limited by a number of different failure modes. As a result, there are plenty of research projects on the subject of load distribution [14] and failure modes [15]. These failure modes depend on different parameters, which for example are gear material, lubrication, operating conditions, and surface roughness. Several test methods are available in order to create better knowledge of different failure modes [16].

In the contact zone of two tooth flanks with high sliding velocities, frictional heat is generated. This leads to a high flash temperature, which promotes the risk of scuffing [17]. Good lubricating conditions are essential to avoid tribological influenced damages on the surface of the tooth flank like scuffing, wear, and micro-pitting [18]. In particular, the development of an adequate film thickness is important considering that the minimum lubricant film thickness in a gear contact is the primary factor to consider when assessing gear sliding wear [19–21].

Therefore, the load carrying capacity is not only influenced by the material, heat treatment, load, contact pressure, and sliding conditions but also depends on the choice of lubricant. Regarding efficiency, the power losses can be reduced by using a low friction lubricant [22].

A current research project supported by the Bayrische Forschungsstiftung (BFS) is conducted to develop an oil-free lubricant. This research is based on a cutting fluid containing water and plant extract, which was developed during a former research project of the company Carl Bechem GmbH and the Fraunhofer Institute for Process Engineering and Packaging IVV [23,24]. With a model sample of this fluid, which also contained gear typical additives, preliminary experimental investigations were conducted in order to verify the general suitability of a lubricant based on water and plant extract for gear applications. The additives used in these experimental investigations, however, are not of plant origin. They are conventional substances, typically used in commercially available mineral or synthetic oils. This paper shares the results of these first experimental investigations.

2. Materials and Methods

Depending on the characteristics and the performance of the lubricant, tribological influenced damages such as wear and scuffing may occur more or less on the surface of gear flanks. The requirement on a lubricant is to build up a lubricant film with adequate film thickness. It is also important that the lubricant, especially the additives, protect the surface from damages, reduce friction, and remove heat from the system.

A continuous removal of material occurring when two surfaces roll and slide against one another is referred to as sliding wear. Scuffing can be described as a particularly severe form of gear tooth surface damage, in which seizure or welding together of areas of tooth surface occur. This is due to the absence or breakdown of a lubricant film between the contacting tooth flanks of mating gears, which is typically caused by high temperature and high pressure [25].

In order to gain a better understanding of the tribological characteristics of the investigated lubricant model, a twin disk test was conducted. The experimental setup for this test is described in Section 2.2. A classification of the lubricant model regarding scuffing was done by the scuffing test, according to ISO 14635-1 [26]. This test is outlined in Section 2.3.

2.1. Principle of the Water-Based Technology

In general, lubricants consist of a base fluid such as mineral oil, PAO, polyglycol, or water mixed with some additives. For this, the herein investigated lubricant, a liquid solution of water, and polymers form the base fluid. These polymers consist mostly of modified cellulose and are gained by plant extracts. By adding polymers, the viscosity of the mixture increases in comparison to the base fluid. This technology allows us to formulate lubricants within a great viscosity range. For this investigation, a fluid with a kinematic viscosity of 46 mm^2/s at 40 °C was used. Therefore, the mixture with base fluid and additives includes more than 80 wt% of water and less than 20 wt% polymers and performance additives. The included additives, which are typically used AW-additives and EP-additives, are needed to enhance some physical and chemical properties of the fluid. The base fluid is the main component of the lubricant and spreads and/or dissolves the additives to influence the tribological impact on the surface [19]. The resulting fluid combines the benefits of water-soluble lubricants and oils, while providing good cooling properties and a sufficiently bearing lubricating film.

2.2. Experimental Setup for Twin Disk Test to Measure the Coefficient of Friction

For the determination of the coefficient of friction, an FZG twin-disk test rig designed by FZG/Stößel [27] is used. Figure 1 shows the mechanical layout of the test rig schematically. Two cylindrical disks are press-fitted onto axially parallel shafts. The shafts are independently driven by two electric motors. The disks are designed with a diameter of 80 mm and a width of 20 mm. Continuous variation of speed is allowed by traction drives located between the motors and driving shafts. A pneumatic cylinder applies the normal contact force F_N in the disk contact by the pivot arm. The pivot arm is located where the lower disk is mounted. The upper disk is fixed in a skid. This skid is attached to the frame by thin steel sheets. The skid is supported laterally by a load cell. With this load cell, the friction force F_R in the disk contact for sliding velocities can be measured as the reaction force with hardly any displacement of the skid. The lubricant is placed directly into the inlet region of the disk contact by injection lubrication. A heating and cooling system is included to maintain and control the temperature of the lubricant [28].

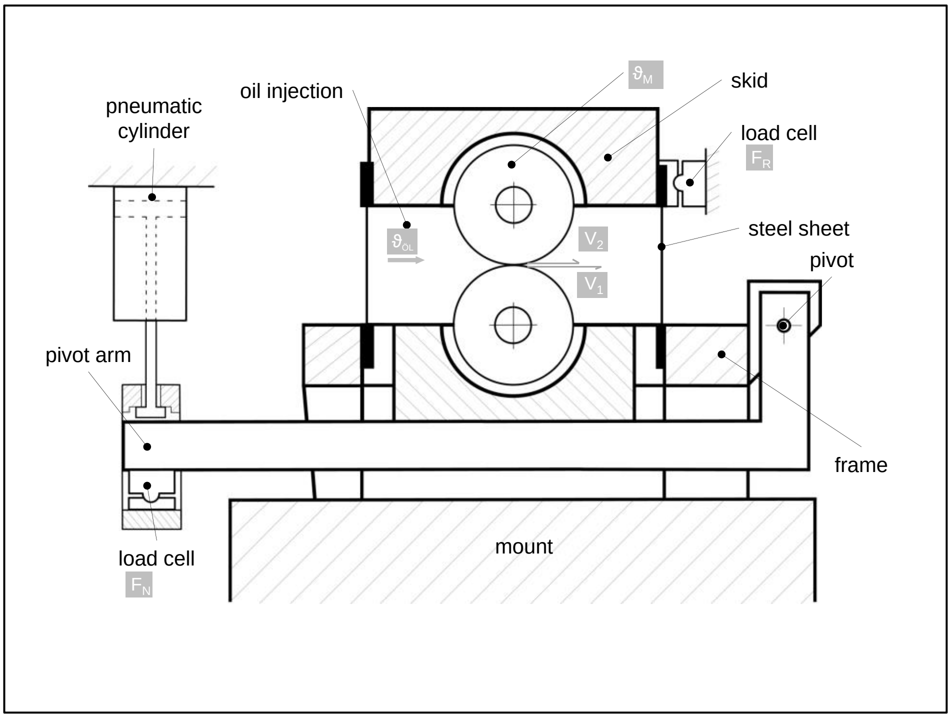

Figure 1. Schematic illustration of the mechanical layout of the considered FZG twin-disk test rig [18].

The coefficient of friction is calculated according to Equation (1).

$$\mu = {}^{F_R}/_{F_N} \quad (1)$$

The sum velocity v_Σ is defined as the sum of the surface velocities v_1 and v_2, according to Equation (2). The sliding velocity v_g is calculated as the difference between the surface velocities v_1 and v_2, which is illustrated by Equation (3).

$$v_\Sigma = v_1 + v_2 \quad (2)$$

$$v_g = v_1 - v_2 \quad (3)$$

The disks were machined from the material 16MnCr5 and case-carburized with a surface hardness of 59–63 HRC to imitate the material properties of typical gears. The disks were ground and polished to an arithmetic mean roughness R_a < 0.1 µm.

2.3. Experimental Setup for Scuffing Test

The FZG scuffing test is conducted on the FZG back-to-back test rig, according to ISO 14635-1 [26]. Figure 2 shows the design of the test rig. The test rig consist of test gears and slave gears with an identical gear transmission ratio. The test gears and the slave gears are connected through two torsional shafts. The pinion of the test gear and the pinion of the slave gear are mounted on the same shaft and the wheel of the slave gear as well as the wheel of the test gears are located on the other shaft. The shaft between the pinions is divided into two parts that are connected by a coupling. While this coupling is opened at standstill, one coupling flange is fixed and the other flange is loaded with a lever and weights to introduce a static torque into the system. After locking the coupling, the lever is removed and the electric motor can be started. For adjusting the defined torque, the weights are hung on the loading arm. The motor only has to compensate the power losses in the system, which allows energy-efficient testing.

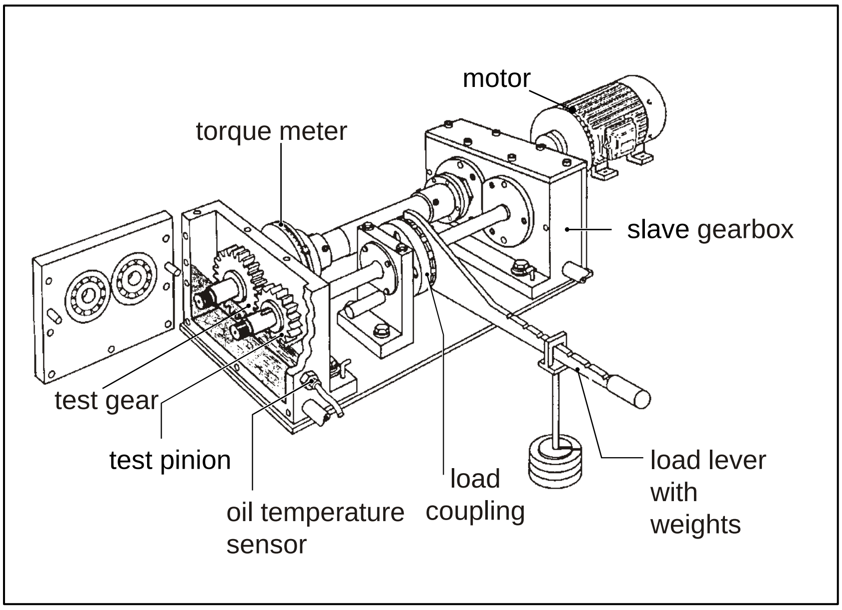

Figure 2. Schematic illustration of the mechanical layout of the FZG back-to-back test rig, according to ISO 14635-1 [26].

The test gears run dip lubricated with the test lubricant at a constant speed for a fixed number of load cycles. The lubricant temperature is controlled thermostatically with the help of a temperature sensor and electric heating elements in the housing of the test gearbox. For the standard test procedure, an oil temperature of 90 °C and a circumferential speed of 8.3 m/s is adjusted. It is possible to modify these standard operating conditions in order to gain results at specific conditions, which can be characteristic for a certain application. The short description of a test gives information of the test gears, circumferential speed, and oil temperature. Therefore, the standard test can be described with A/8.3/90.

The gears of type "FZG-A" are applied for the test procedure. The test gears are designed as spur gears with 16 teeth on the pinion and 24 teeth on the gear, normal module of 4.5 mm, a pressure angle of 20°, and no tooth correction. The center distance is 91.5 mm and the effective tooth width is 20 mm for both gears. The test gears are made of case-hardening steel and they are case-carburized with a surface hardness of 60–62 HRC in the area of the tooth flank. These gears are specifically designed with high sliding velocities in the contact zone in order to create a high risk for scuffing. Details of this type of gearing are summarized in Table 1.

Table 1. Details of type "FZG-A" test gears for the FZG scuffing test.

Dimension	Symbol	Numerical Value	Unit
Center distance	a	91.5	mm
Effective tooth width	b	20.0	mm
Module	m	4.5	mm
Number of teeth pinion	z_1	16	-
Number of teeth wheel	z_2	24	-
Pressure angle	α	20.0	°
Helix angle	β	0	°
Grinding	Maag criss-cross grinding (15° method)		
Tooth correction	none		

The test consists of 12 load stages. After every stage, the torque of the pinion is increased. The first stage starts with a torque of 3.3 Nm (load stage 1) and can be raised to a maximum of 534.5 Nm (load stage 12). Therefore, the Hertzian contact stress increases from p_c = 146 N/mm^2 to a maximum of p_c = 1841 N/mm^2. The applied torque and Hertzian contact stress for each load stage can be seen in ISO 14635-1 [26].

In general, the test gears are inspected after each load stage from load stage 4 for a close observation of the damage on the flank of the gears. If the summed width of damage from all teeth due to scuffing is larger than the width of one flank, the test is finished and the last running step is identified as the failure load stage. This allows a classification of the lubricant regarding scuffing. The test is ended when the failure criterion has been met and/or when the last load stage has been completed without meeting the failure criterion.

3. Results

In order to carry out some experimental investigations on the considered model fluid containing water, plant extract, and gear typical additives, the coefficient of friction was measured and a scuffing test was conducted. For these experimental investigations, a fluid sample with a kinematic viscosity of 46 mm^2/s at 40 °C was used.

3.1. Coefficient of Friction

The coefficient of friction was determined at a lubricant temperature of 40 °C and Hertzian contact stress of 1300 N/mm^2 and 1200 N/mm^2. The slip was increased for every test from 0% to 50%. The measurements for every test were conducted at a sum velocity v_Σ of 1 m/s, 4 m/s, and 16 m/s. Figure 3 shows the coefficient of friction measured at 1300 N/mm^2 and Figure 4 shows the coefficient of friction at 1200 N/mm^2. It can be seen that the values for both stresses are far below 0.005. The coefficient of friction for a sum velocity of 4 m/s appears to be slightly lower than for a sum velocity of 1 m/s and 16 m/s. The values for a sum velocity of 1 m/s are located slightly above the values of a sum velocity of 4 m/s and 16 m/s. Nevertheless, the difference between the measured values of the different sum velocities are on a very low level. Figure 5 visualizes the measured bulk temperature of the upper disk. For a sum velocity of 1 m/s and 4 m/s, temperatures below but close to 40 °C were determined. The investigation showed bulk temperatures between 40 °C and 60 °C for a sum velocity of 16 m/s. The measurements with a Hertzian contact stress of 1200 N/mm^2 showed comparable results for the bulk temperature.

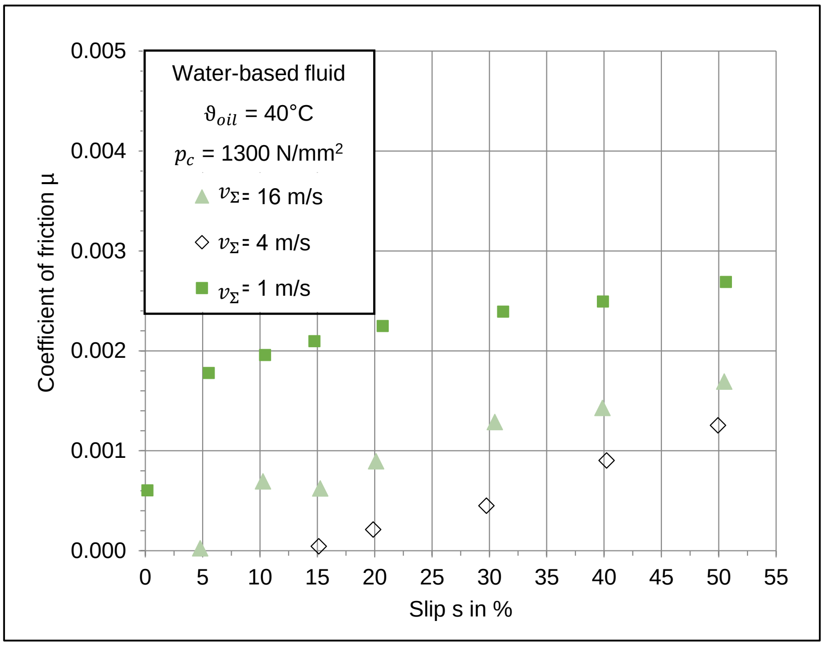

Figure 3. Coefficient of friction for a fluid sample containing water and plant extracts at 1300 N/mm^2.

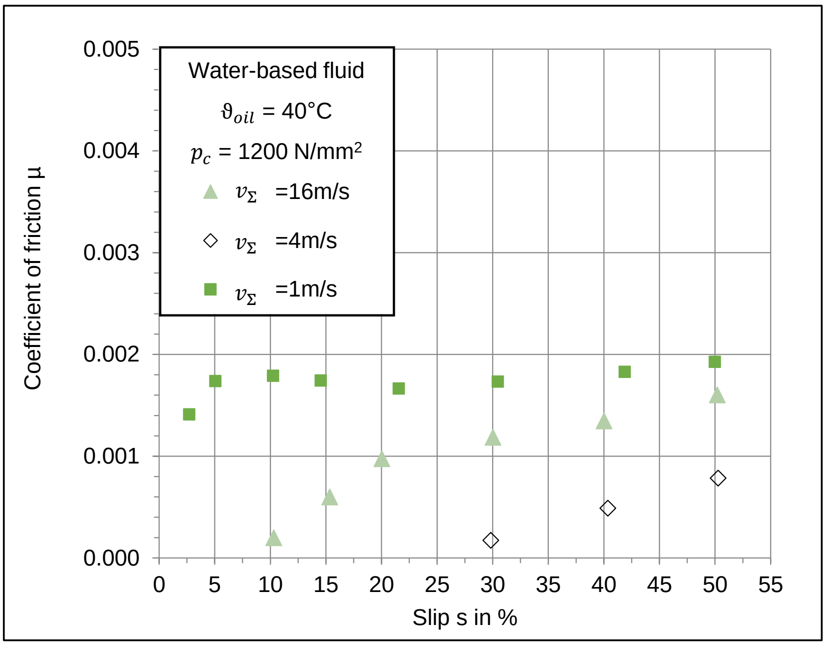

Figure 4. Coefficient of friction for a fluid sample containing water and plant extracts at 1200 N/mm^2.

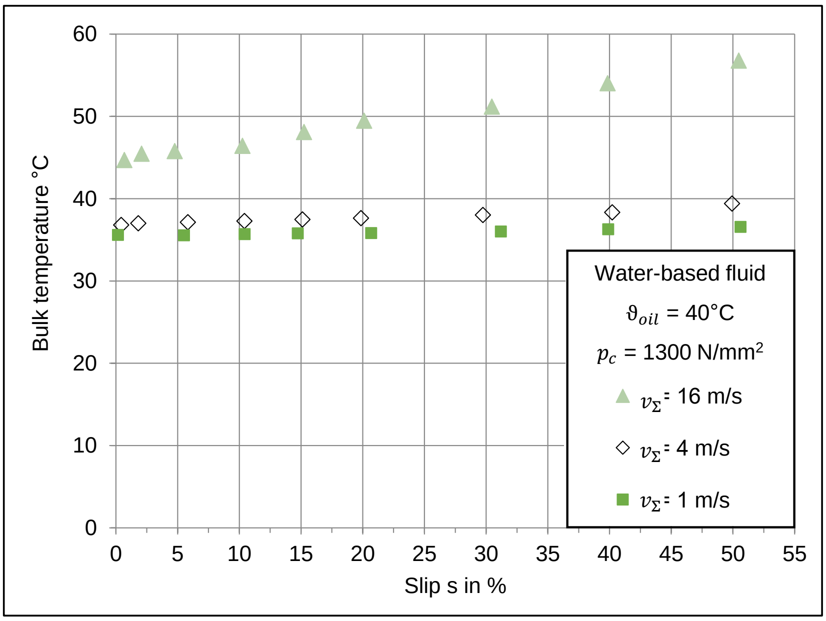

Figure 5. Bulk temperature for a fluid sample containing water and plant extracts at 1300 N/mm^2.

A comparison of the fluid sample based on water and plant extract with mineral oil and synthetic oil is illustrated in Figure 6. The mineral and synthetic oil were compounded with additives. The mineral oil had a kinematic viscosity of 95 mm^2/s at 40 °C. For the synthetic oil, an oil based on PAO with a kinematic viscosity of 64 mm^2/s at 40 °C was used. The reference values of the mineral and synthetic oil were conducted at the institute with a lubricant temperature of 40 °C, a Hertzian contact stress of 1200 N/mm^2, and a sum velocity v_Σ of 16 m/s on the twin disk test rig. It can be seen that the fluid sample based on water and plant extract showed, under the same test conditions, values of the coefficient of friction in the range of about 1/25 of typical values determined for mineral oils. Thus, a reduction of load-dependent power losses in the range of 90% can be expected. Compared to the PAO, significantly lower values for the coefficient of friction were reached with the water-based fluid.

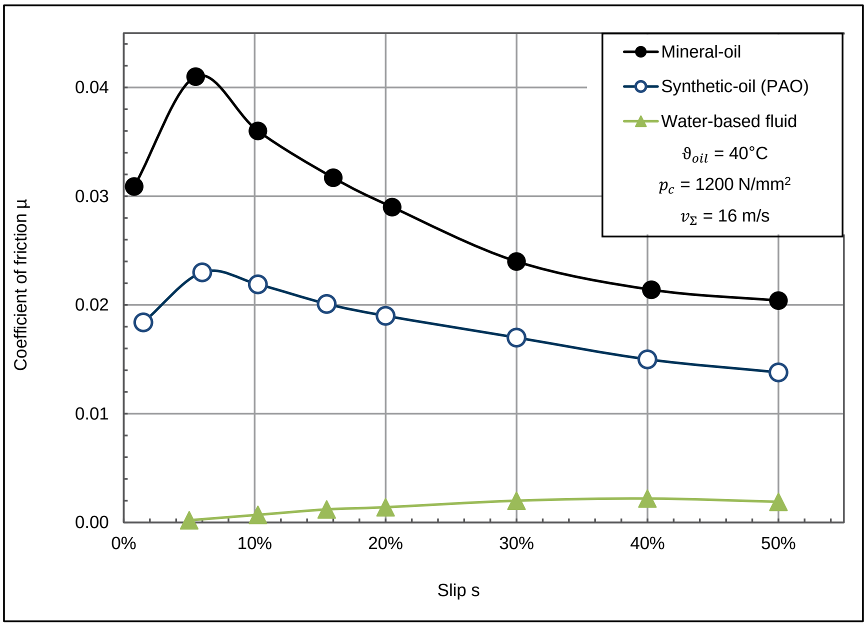

Figure 6. Coefficient of friction of a fluid sample containing water and plant extracts compared to mineral and synthetic oil at 1200 N/mm^2.

3.2. *Scuffing Test A/8.3/RT*

Since the fluid sample is based on water, it can be assumed that the boiling point of the fluid will be at a temperature around 100 °C. Therefore, the lubricant start temperature was reduced in comparison to the standard test, which is normally set to 90 °C. Furthermore, a possible application of a water-based lubricant at low temperatures can be expected. Hence, the scuffing test was performed at room temperature (RT) and a circumferential speed of 8.3 m/s. The "FZG-A" test gears were used. Table 2 summarizes the progress of the damages on the flank of the pinion caused by scuffing during the test procedure. After load stage 7 scoring with a width of 240 mm summarized over 12 flanks of the pinion was located. Since the failure criterion is established at 20 mm, a failure load stage of 7 could be defined for this fluid model.

Table 2. Progress of the damage during the scuffing test A/8.3/RT.

Load Stage	Torque on Pinion	Hertzian Stress at Pitch Point	Characterization of Damage	Width of Damage
5	94.1 Nm	1093 N/mm^2	Scoring mark on two flanks on pinion	1 mm
6	135.3 Nm	1314 N/mm^2	Scoring mark on six flanks of pinion	3 mm
7	183.4 Nm	1527 N/mm^2	Scoring on 12 flanks of pinion	240 mm

Figure 7 shows the reference of a new flank of a test pinion type "FZG-A". The typical grinding pattern from the Maag criss-cross finishing process can be seen. One flank of the pinion used in the testing is shown in Figure 8. The surface of the flank shows clear scoring over the full face-width of 20 mm.

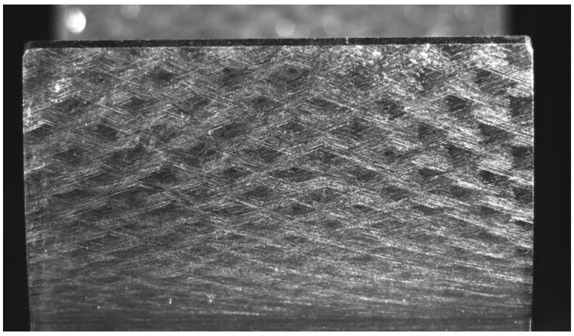

Figure 7. Surface of a new flank.

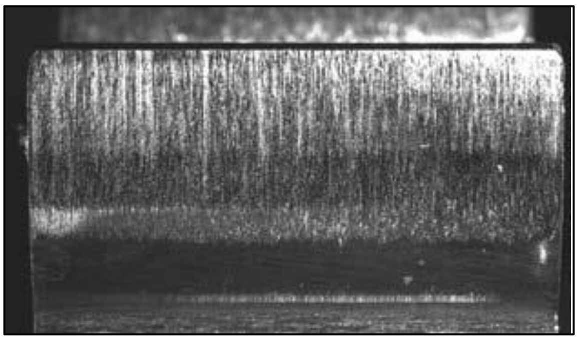

Figure 8. Surface of the flank after test, load stage 7.

4. Discussion

In this experimental investigation, as a first step, the coefficient of friction was determined and a scuffing test A/8.3/RT according to ISO 14365-1 [26] was performed in order to verify the general suitability of a lubricant based on water and plant extract for gear applications.

The determination of the coefficient of friction showed values about 1/25 of typical values determined for mineral oils. This super-lubricity can be explained with the separation of the sliding surfaces by the hydrated layer of water. The hydrated layers are formed by hydrogen-bonded glycerol and water molecules, which are very easy to shear and, therefore, result in a very low friction coefficient [29].

A scuffing load capacity of failure load stage 7 could be identified for this fluid, which is typical for mildly additivated conventional turbine oils. The classification of typical conventional mineral oils can be seen in Table 3. A classification of different lubricant samples is allowed by the widely established standard test method, according to ISO 14365-1.

Table 3. Scuffing performance of typical conventional mineral oils.

Oil for Gear Applications		Converter Oil	Turbine Oil	Industrial Oil	Automotive Oil
Viscosity class	ISO-VG	22/32	46/68/100	100/150/220	100/150/220
Viscosity at 40 °C	mm^2/s	20–35	40–100	90–240	90–240
Mineral oil with no additives	Failure load stage	2–4	3–5	5–7	—
Medium content of additives	Failure load stage	5–8	6–9	8–10	—
High content of additives	Failure load stage	9–11	10–12	>12	>12

The obtained results showed that, in general, it is possible to use water as a basis for lubricants in gear drives under certain operating conditions. Since water has a low viscosity compared to conventional used lubricants, plant extracts were added to generate higher viscosities. Therefore, for future work, a selection of polymer has to be defined for gear applications as well as adequate additives.

In order to avoid tribological influenced damages such as wear and scuffing on the surface of gear flanks, an adequate film thickness is needed. To produce a sufficient high film, thickness will be one of the main challenges. As is well identified, it is necessary to study ways to improve the load carrying capacity and the stability of the polymer, as well as reduce the corrosion effects of this fluid sample based on water and plant extract. There are some important aspects to be defined to put the lubricant based on water to meet the necessary performance in many applications.

However, by using this innovative lubricant, certain benefits can be expected. Since this lubricant is based on water and plant extract and does not contain any mineral or synthetic oil components, it is free of any harmful solvents and also biodegradable. This can be beneficial in gears located in boats or deep sea applications. As a result, environmental risks like water contamination can be reduced in comparison to conventional transmission fluids. In addition to that, fire protection can be improved by using a water-based lubricant. Gear boxes run by water-based lubricants are easier to handle than oils regarding disposing and cleaning. Another benefit of water is the very high thermal conductivity. A lubricant containing about 90% water could highly improve the heat transfer from the tooth contact. Since this leads to reduced temperatures in the contact, lower operating temperatures can be realized. However, not only the heat transfer can be improved by a water–based lubricant, but the heat generation by friction decreases in comparison to oil because of the low coefficient of friction. Thus, lower bulk temperatures and lower flash temperatures can be expected. This may even have a positive effect on avoiding scuffing, which is also influenced by the flash temperature [17]. A lubricant with low friction can improve transmission efficiency [22]. Because of reduced friction and the possibility of lower operating temperatures, using water-based lubrication can also save energy.

Within the ongoing project, a selection of polymers as well as adequate additives has to be defined for gear applications. This lubricant has to be tested by several acknowledged test methods in order to generate characteristic values regarding tribological influenced damages. Additionally, a field of application has to be specified. Favorable operating conditions could be at low temperatures and loads but high speed. Using one exemplary application, innovative electric motor boat drives were identified. Systematic laboratory tests as well as field tests are in preparation.

5. Conclusions

In the conducted experimental investigation, the coefficient of friction was determined and a scuffing test A/8.3/RT was determined, according to ISO 14365-1 [26]. It was performed for a fluid sample based on water and plant extract. The obtained results showed the general suitability of a lubricant based on this technology for gear applications. In comparison to mineral and synthetic oils, a very low coefficient of friction was determined for the fluid sample. Because of the very high thermal conductivity of a water-based fluid, lower operating temperatures in gear applications can be realized. As a result, energy can be saved. Additionally, a water-based lubricant is free of any harmful solvents and is biodegradable. Using such a lubricant can lead to better fire protection and is easier as well as safer to handle than oils regarding disposing and cleaning.

For future work, a selection of polymer as well as adequate additives must be defined for gear applications. There are some important aspects to be defined to put this lubricant type to meet the necessary performance in many applications. In order to generate characteristic values regarding tribologically influenced damages, this lubricant has to be tested by several acknowledged test methods. Additionally, a field of application has to be specified. One potential application might be an innovative electric motor boat drive.

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