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Influence of Contamination of Gear Oils in Relation to Time of Operation on Their Lubricity

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Abstract: The quality and reliability of consumables, including gear oils, results in the failure-free operation of the transmission components in heavy trucks. It is known that oil viscosity is essential for all lubricated tribopairs for wear and friction reduction in all vehicles with a gearbox. Viscosity may be influenced by the contamination that wear products can impart on the oil. Oil contamination can also affect lubrication efficiency in the boundary friction conditions in gearboxes where slips occur (including bevel and hypoid gearboxes). The present research focused on this issue. An obvious hypothesis was adopted, where it was theorized that exploiting the contaminants that are present in gear oil may affect how the lubricating properties of gear oils deteriorate. Laboratory tests were performed on contaminants that are commonly found in gear oil using the Parker Laser CM20. The study was designed to identify a number of different solid particles that are present in oil. At the second stage, friction tests were conducted for a friction couple "ball-on-disc" in an oil bath at 90 °C on a CSM microtribometer. The quantitative contamination of the gear oils that contained solid particles and the curves representing the friction coefficients of fresh oils with a history of exploitation were compared. The test results were statistically analysed. Exploitation was shown to have a significant impact on the contamination of gear oils. It was revealed that the contamination and the mileage had no effect on the tested oils.

Keywords: lubricity; gear oil; wear; operational reliability

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1. Introduction

Ensuring that machines and devices are able to conduct high-quality work and maintain operational reliability is not only a very important issue in chemical applications [1–3], e.g., transport [4,5], electronic systems [6], or scientific works, e.g., for chemical purposes [7], it is also of great importance for applications that are related to the production of high-quality food products [8]. Lubrication is essential for all sliding pairs in all tribosystems [9–11].

The physical essence of lubrication processes is the conversion of adverse external friction to friction that takes place inside of the tribofilm [12–14]. In order for good lubrication to be maintained, the grease must have high adhesion to the frictional surfaces in question, and the grease layer that is between these components must be maintained at a certain thickness. This should be maintained regardless of friction speed, pressure, and temperature [15], and this is usually difficult to achieve. The formation of a layer of grease on the friction surface is associated with the phenomena of the physical absorption of polar particles and the chemical absorption of boundary films as well as a hydrodynamic effect [16]. However, load transfer occurs through a layer of grease that is generated by the hydrodynamic effect in the machine kinematic nodes that are under increasing motion [12]. High-pressure lubrication takes place in concentrated contacts, including in gear contacts [17]. High pressure in the contact area increases the viscosity of the lubricant as well as the elastic deformation of the surfaces that are in contact with each other indirectly through the lubricant film. This is the case for elastohydrodynamic lubrication. The term elasthydrodynamic film is used to refer to the intermediate lubricant film [15]. In gear transmissions, lubrication conditions are not favorable for the formation of a lubricating film. It seems that gullet pressure is one of the determinants. However, as stated in [18], high pressure in the contact area increases the viscosity of the lubricant and the elastic deformation of the contact surfaces in an indirect way through the lubricant film. Moreover, pressure leads to teeth bending, and when teeth pairing occurs, the tips of the teeth belong to the powered wheel rub the grease from the surface of the powering wheel. Modifying the teeth only partially eliminates this phenomenon [15]. Moreover, high circumferential speeds result in oil being removed from the contact area. It is known that the formation of a lubricating film in oil is intended to prevent metal surfaces from coming into direct contact with each other, but it also refers to a situation in which wear and high friction occur between sliding surfaces. In light of the above situation, this becomes difficult in the context of gear boxes. In addition, oil properties change during exploitation, and this represents another disadvantage. It can be assumed that the contamination of the used oil may impair the ability to form a permanent tribofilm that reduces the friction and wear of sliding pairs [19]. The lubricant that is used in closed-circuit mechanical systems is subject to aging [18]. It undergoes oxidation because it comes into contact with air. The particles, which are a product of tribological wear, also get into the oil. The contamination of gear oils with wear products may result in power losses, among other consequences, as well as decreases in the flow resistance of the lubricant [19]. Therefore, the quality and condition of the lubricant affects the friction resistance in gear boxes and may affect the efficiency of tribomechanical systems [20,21]. The applications that oil is used for may lead to changes in its viscosity [19]. On the other hand, as shown in paper [21], oil viscosity results in the power losses in meshing when operating under a working load. It is for this reason that periodic oil changes are performed [15].

Synthetic oils are usually used in the gear boxes of modern vehicles. This is because of the many advantages of these oils. One of them is the reduction of the friction coefficient during meshing by up to 25% compared to when mineral oils are used as a lubricant [21]. The use of such oils is beneficial, although the problem of the operational quality of these oils is also important, with the preservation of their lubricating properties being one of the main issues, especially since these properties can be affected by the operating time as well as by the level of contamination resulting from exploitation. On the basis of the above, it can be concluded that the overworking and contamination of gear oil may affect its lubrication efficiency. This observation is especially important when considering that the gear box elements need to be protected against excessive wear and the boundary lubrication condition, which can be seen in gears where slips occur (including in bevel and hypoid gear boxes). The present research focused on this issue. The hypothesis that operational contaminants may affect the deterioration of the lubricating properties of the synthetic gear oils was adopted. The main goal of this paper was to anlayze how operational contaminants affect the deterioration of the lubricating properties of synthetic gear oils. The article consists of the introduction followed by a chapter describing the experiment—the Materials and Methods section. Finally, the results that were achieved through the experiment are presented and compared.

2. Materials and Methods

The following gear oils were used for the laboratory tests: class SAE 75W-140 Scania Oil STO 2:0 A (oil from the axle of a truck), class 80W-14 Scania Oil AXLE STO 1:0 (oil from the axle of a truck), class 75W-90 and Scania Oil 2:0 G of (oil from the gearbox of a truck).

Lubrication tests were conducted on the CSM microtribometer (CSM Switzerland). The tests were performed using a ball-on-disk module, such as the one shown in Figure 1. A steel friction node was installed. Both elements of the friction pair were made of 100Cr6 steel. Friction tests were conducted in an oil bath at 90 °C, which supposedly corresponded to the operating temperature of the oil in real working conditions. The load in the friction test was constant and equaled 5 N. The sliding distance was 630 m, and the linear speed was 60 mm/s. During the friction tests, the friction coefficient was recorded at the frequency of 10 Hz as a function of the friction path.

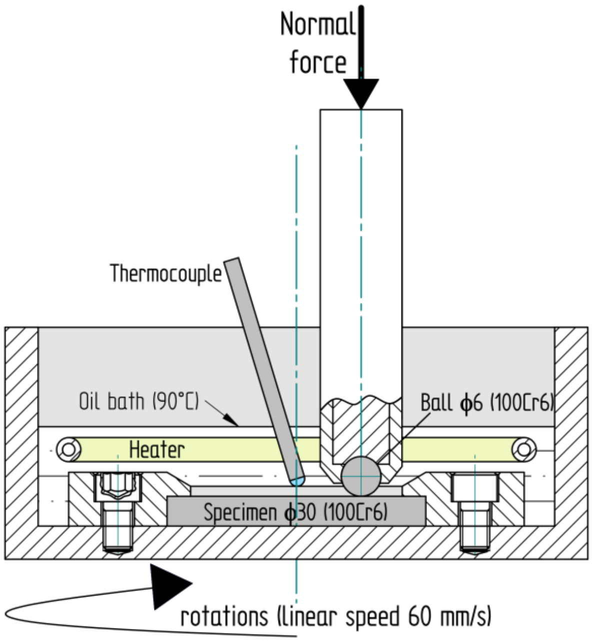

Figure 1. Picture of the tribometer test set up for the lubricating oil tests.

The laboratory tests determining the oil contaminants were performed using the Parker Laser CM20 device, which is designed to identify the number of solid particles in oil and to classify them with the use of the optical scanning method. The measurement procedure was in accordance with PN-ISO 4406: March 2005 [22].

The methods that were used to observe and count the number of contaminants are shown in Figure 2.

The measurement accuracy of this type of transmitter is an important issue. It should be noted that a quality of measurements better than 5% was obtained when the ISO MTD (ISO Medium Test Dust) and ISO ACFTD (ISO Air Cleaner Fine Test Dust) procedures were used. As a result, it is possible to obtain results that are in accordance with the ISO standard and that are in the range of 7–22 µm; NAS and SAE measurements were also obtained in the ranges of 0–12 µm and 0–12 µm, respectively. In the conducted research, it was assumed that the contaminants would be classified into six sections: 4; 6; 14; 21; 38; and 70 µm. Figure 3 shows the measuring device and an exemplary printout of the measurement results.

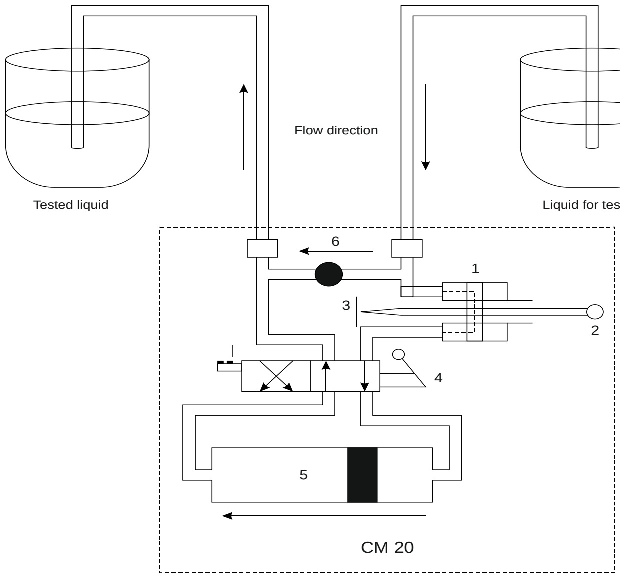

Figure 2. Diagram of the measuring system, 1—measuring chamber, 2—laser light source, 3—optical scanner, 4—switching valve (hydraulic), 5—dosing pump, 6—flow sensor.

(**a**) (**b**)

Figure 3. Parker Laser CM20 automatic particle number meter (**a**) device and (**b**) sample printout of measurements.

3. Results and Discussion

The test results were statistically processed. Table 1 presents descriptive statistics of the test results for the friction coefficient and contains the minimum (min), maximum (max), and average (mean) values and the standard deviation (std. dev.).

Table 1. Descriptive statistics of the friction coefficient obtained in the tests determining the lubricity of gear oils.

Oil	Mileage (km)	Gearbox	Min	Max	Mean	Std. Dev.
75W-140	fresh oil	axle	0.0600	0.1374	0.0979	0.0142
75W-140	350,000	axle	0.0298	0.1193	0.1020	0.0086
80W-140	fresh oil	axle	0.0177	0.0984	0.0671	0.0136
80W-140	220,000	axle	0.0032	0.0582	0.0523	0.0091
75W-90	fresh oil	gearbox	0.0753	0.1566	0.1046	0.0075
75W-90	210,000	gearbox	0.0345	0.0908	0.0655	0.0102

The statistical values of the friction coefficient of the fresh oils and the oils with a service history differ. In the case of the used oils, the work of which was expressed as the mileage of a vehicle between 220,000 km and 210,000 km, the friction coefficient demonstrated lower average values than the friction coefficient that was observed for the fresh oils. Oil changes were planned for after these mileages were achieved. T Figures 4–6 present graphs of the friction coefficient based on different paths. The graph presenting linear wear is shown in Figure 7. The graph shows the variability of linear wear depending on the number of friction cycles. It should be noted that the presented curves depend on the sliding wear of the friction pair, but this is not the only thing that should be taken into account. The thermal expansion of the ball and disc that are heated by the conditioned oil was also influenced. Both factors of the experiment have a synergistic influence on the shape of the curves. The variability tests of the friction coefficient can function as a measure for the lubricity of gear and diesel oils [23,24].

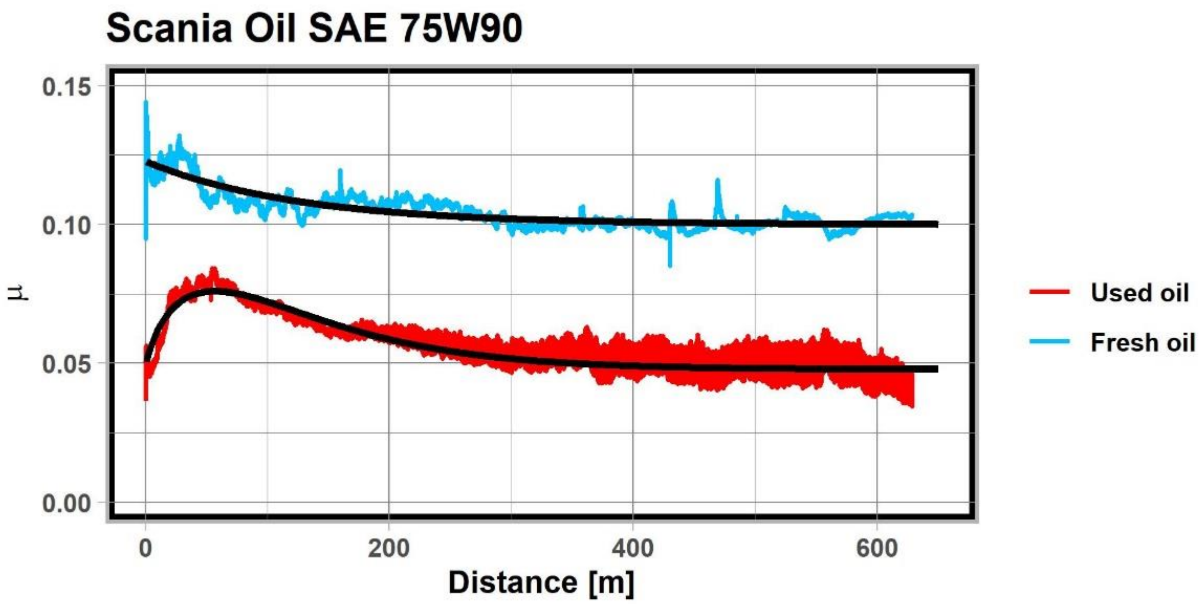

Figure 4. The curve of the friction coefficient depending on the friction path for oil SAE 75W-90 and fitting Function (1).

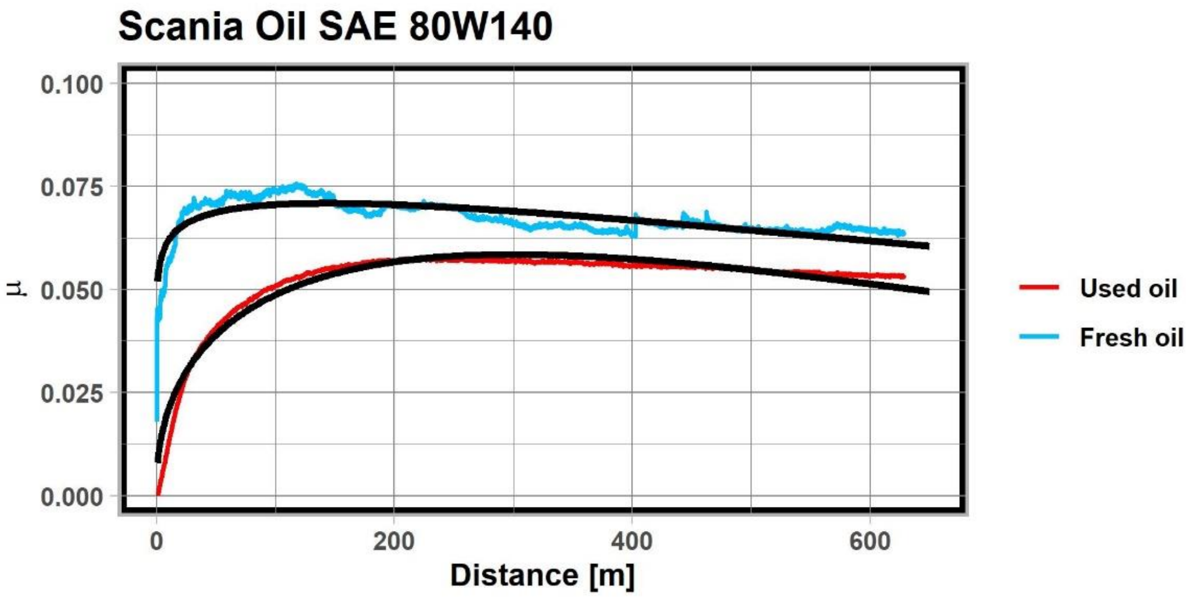

Figure 5. The curve of the friction coefficient depending on the friction path for oil SAE 80W-140 and fitting Function (1).

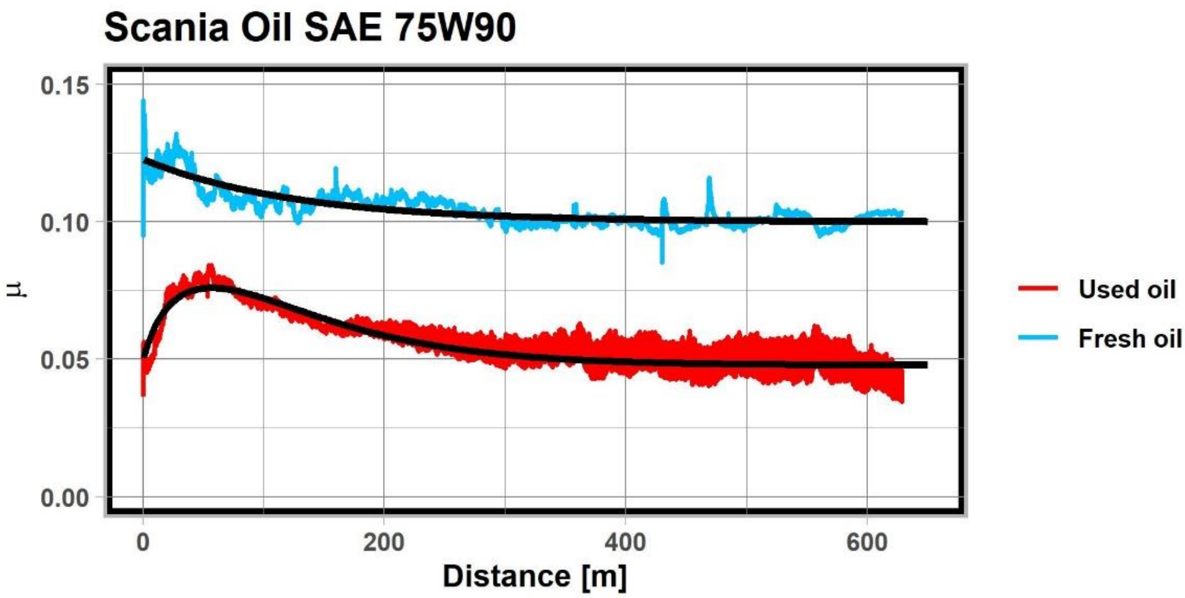

Figure 6. The curve of the friction coefficient depending on the friction path for oil SAE 75W-90 and fitting Function (1).

The nonlinear dependence between the friction coefficient and distance (friction path) is defined as follows:

$$y = \alpha_0 + \alpha_1 x^{\alpha_2} e^{-\alpha_3 x} + \varepsilon, \quad (1)$$

where y denotes the friction coefficient, x—distance, and ε—disturbances with a normal distribution N(0, σ2) and is connected with measurement. The shapes of the curves for the friction coefficients indicate a gradual friction process. All of the tested oils are characterized by friction coefficient having an increasing curve when friction begins, and then a slight decrease at the first stage of friction. At the next stage, a steady-state friction regime with slight deviations was observed. However, for some oils, the course of the friction coefficient decreased slightly—75W-90 (fresh oil), 75W-140 (after 210,000 km)—at the second stage of friction. The most stable friction curve, which demonstrated the smallest amount of fluctuation, was characterized by oil 80W-140 (after 220,000 km). In this case, the variation seen in the friction function was similar than that of the theoretical model, which, as we will see later, was explained by the dependence between the number of particles and the diameters of those particles.

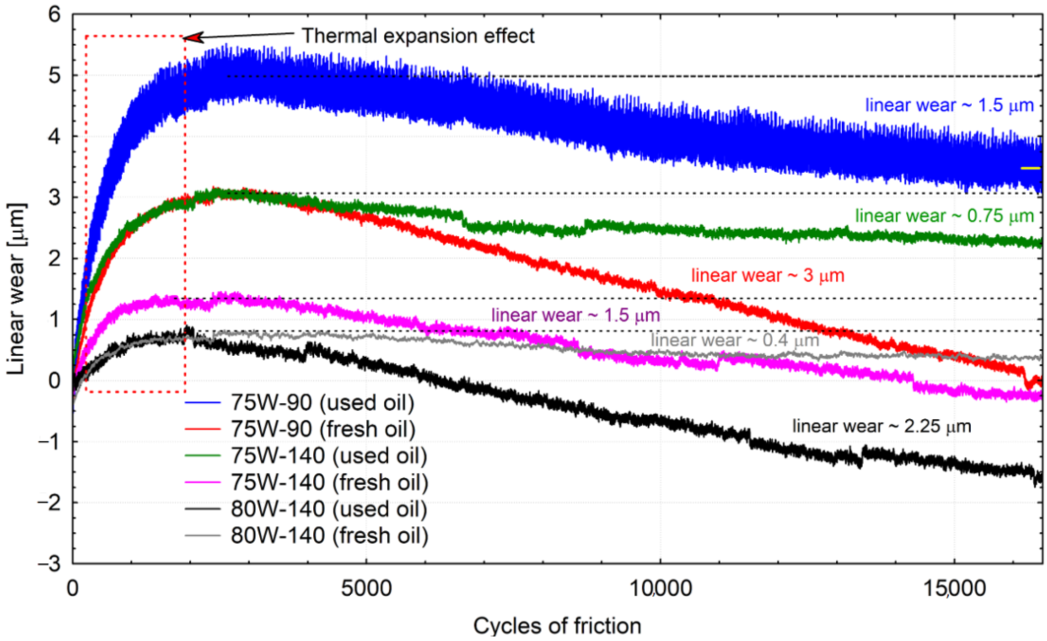

Figure 7. Linear wear of friction pairs in oil bath.

For each oil, the parameters $\alpha_0, \ldots, \alpha_3$ were estimated by applying the least squares method. Parameter α_2 corresponds to the shape of the curve, but value α_3 corresponds to the scale of descent. The values of these parameters are presented in Table 2. The fitting of model (1) to the data is marked with a black curve in Figures 4–6. Additionally, the basic indices of fitting function (1) were determined using the sum of squares (*SSE*)

$$SSE = \sum_{i=1}^{n} \left(y_i - \hat{\alpha_0} - \hat{\alpha_1} x_i^{\hat{\alpha_2}} e^{-\hat{\alpha_3} x_i} \right)^2,$$

where $\hat{\alpha_0}, \ldots, \hat{\alpha_3}$ denote the estimator of unknown parameters, and the sum of squares total (*SST*) is a sum of the squared differences between the observed dependent variable and its mean

$$SST = \sum_{i=1}^{n} \left(y_i - \bar{y} \right)^2.$$

Table 2. Parameter values of Function (1) for different types of oils and values for fitting this function to real measures.

Type	75W90	75W90	80W140	80W140	75W140	75W140
Category	Used oil	Fresh oil	Used oil	Fresh oil	Used oil	Fresh oil
α_0	0.04784	0.09997	0.00000	0.00025	0.01341	0.08129
α_1	0.00276	0.02276	0.00798	0.05168	0.07399	0.03602
α_2	0.76353	0.00000	0.42336	0.07891	0.05117	0.03766
α_3	0.01342	0.00801	0.00141	0.00055	0.00034	0.00386
SSE	0.31497	0.32592	0.06434	0.13916	0.21841	0.33773
SST	2.07269	1.22027	1.38345	0.38203	0.35047	2.93344
R^2_{pseudo}	0.84804	0.73291	0.95349	0.63573	0.37680	0.88487

SST represents the deviance of the intercept-only model, but SSE represents the deviance of the fitted nonlinear model. According to [25], we can calculate the pseudo R^2, which presents the goodness of fit model to the data as follows:

$$R^2_{pseudo} = 1 - \frac{SSE}{SST}.$$

The figures that are presented below present the fitting of Function (1) to the measured data for the different oils.

In the literature, the stage at which the curve of the friction coefficient is constant is called stationary or normal [26]. The stability of the friction process is important when assessing oils. However, the lubricity of the oils that were tested is of practical significance. The SAE (Society of Automotive Engineers) defines lubricity as a measure of the difference in friction when comparing the properties of different oils with the same viscosities under the same conditions [27]. In the results from the tests that were conducted in the current research, the nominal viscosities of the fresh oil and used oils were the same, as the assumed mileages (km) were not exceeded. Three different oils were tested in these tests. When comparing the gear oils with different SAE viscosity classes, the following definition of lubricity proposed in [28] may be relevant: lubricity is the liquid's ability to cause low static resistance when moving solid surfaces and high resistance when bringing them together under a normal load. According to [29], lubricity is the ability of a substance to provide better lubricating properties in conditions where the lubricant film is so thin that its action is not only determined by viscosity. This is probably related to the occurrence of mixed friction in many steel friction nodes. The approach presented in these works is utilitarian, and the value of the friction coefficient is of practical importance when assessing oils. According to the criterion for assessing the condition of oil, i.e., lubricity, the condition of these oils can be considered suitable for use with high probability. A similar relationship was demonstrated for engine oils composed of mineral oils that were used in heavy trucks [30,31]. After 350,000 km, the used oil demonstrated a friction coefficient that was slightly higher than that of the average value of the friction coefficient that was obtained for fresh oil, and the standard deviation of the used oil's friction coefficient was clearly lower. The results indicate that the used oil 75W-140, which performed greater operational work than oils 80W-140 and 75W-90, had worse lubrication properties and was closer to reaching the limit state. It is worth adding that in [32], the limit state of the object is defined as a technical condition in which further operation of the object is not recommended. It should be kept in mind that the quality of a product is determined by its degree of compliance with requirements [30], and the technically justified service life of oil should ensure the maximum use of the potential of oil quality [33]; if this is true, then used oil 75W-140 should not be considered to be suitable for use. Such a performance of this oil is also confirmed by the linear wear results that are presented in Figure 7 (the curve marked in blue). The highest difference in average values of the friction coefficient for the fresh and used oil was found for the oil with the lowest nominal viscosity of 75W-90. The difference was ~37%. The same relationship was also shown in the linear wear measurements. In addition, the average friction coefficient for fresh oil 75W-90 was the highest among all of the tested oils. The linear wear fresh oil 75W-90 was also the highest of all of the fresh oils. It is possible that the condition of the used oil 75W-90 also depended on the type of a gearbox that the oil was worked in. The two other oils worked in axles. Bevel gearboxes are used in the rear axles, and this type of gearbox is characterized as having a much greater degree of slipping compared to hypoid gear boxes, which means that the working conditions of the oil in the axles are more demanding [19]. A different extortion spectrum may be reflected in the dimension of the qualitative changes that were observed in the oils used in axles.

The analysis also concerned the impact of the degree of contamination on lubricating properties. An analysis of the relationship between the number of particles and the diameter was conducted. The purpose of this analysis was to identify and compare the trends that were obtained for the fresh and used oils. For each of the oil states (clean and used), the relationship between $log_{10}n$ (logarithm of the decimal number of particles depending on the diameter of x) was identified. For this purpose, a linear model with a particular transformation of the dependent variable was considered as follows:

$$(log_{10}n)^b = \alpha_0 + \alpha_1 x + \varepsilon, \quad (2)$$

where ε is a random variable with a normal distribution of N(0, σ^2). In the paper, b = 0.2 was assumed (for this parameter, the highest determination indicators of R^2 were obtained for both the clean and used oils). From (2), we can see that the dependence between the number of particles and the diameters of these particles is nonlinear.

Linear regression plots (2) are presented in Figures 8–10. The parameters of the linear regression model are presented in Tables 3–5. The results of the particle content tests indicated that the number of particles with the largest diameters is clearly higher in the used oils. This means that these oils contain more contaminants. To estimate the unknown parameters in model (2), the least squares method was applied. The linear models of the dependences between the diameters of the particles in the oil samples and the number of these particles are well matched to the empirical data. The values of the coefficients of determination are close to or above 0.9. The best fit of the linear regression model was demonstrated for fresh oil 75W-90. The same oil also had the highest average friction coefficient in the lubricity tests.

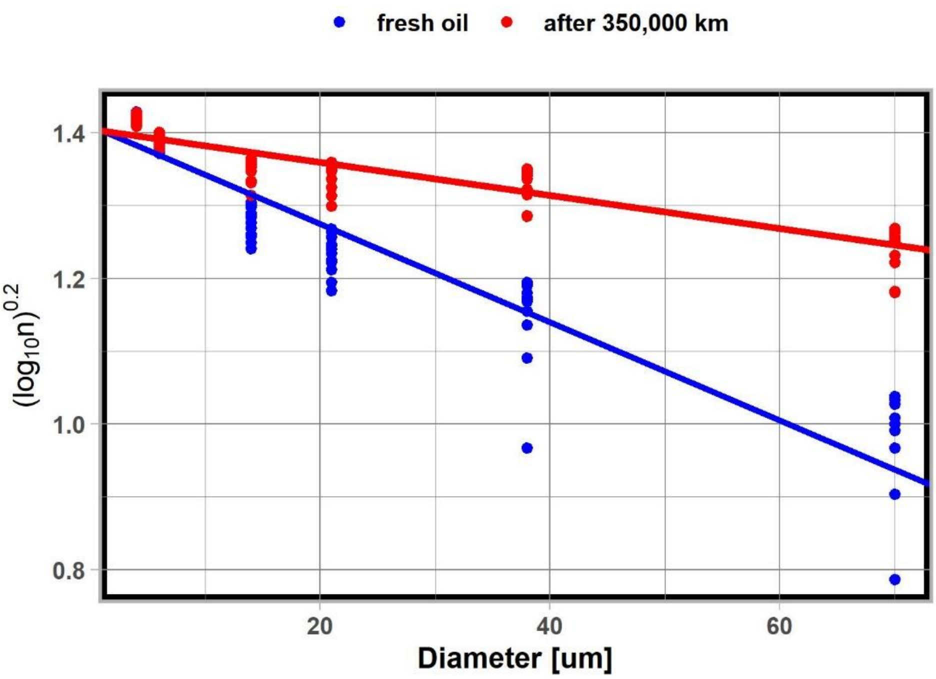

Figure 8. Regression model of particle content as a function of particle diameter of fresh and used (350,000 km) gear oil SAE 75W-140.

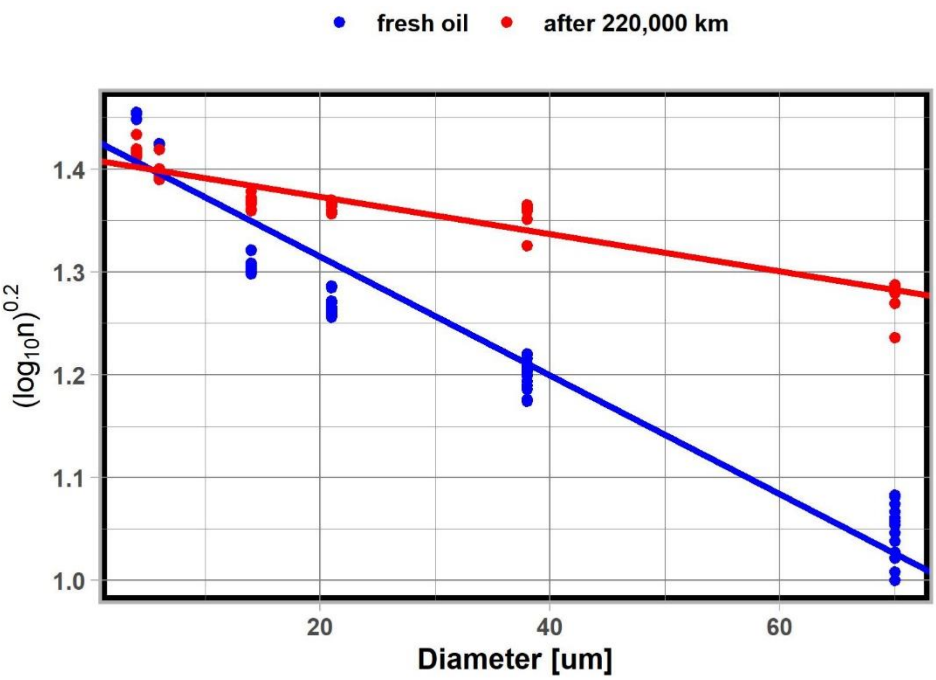

Figure 9. Regression model of particle content as a function of particle diameter of fresh and used (220,000 km) gear oil SAE 80W-140.

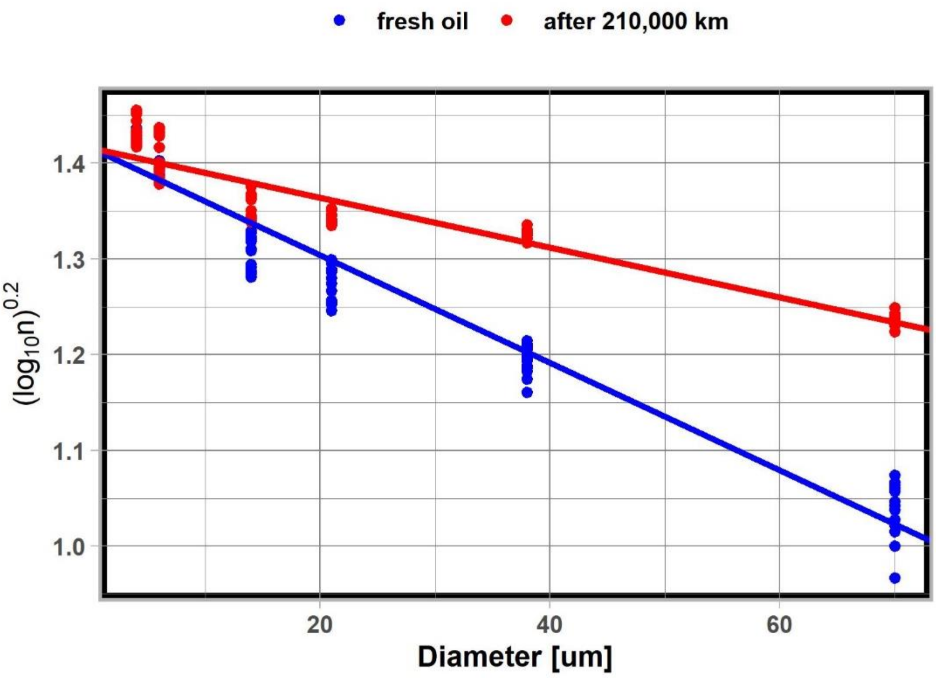

Figure 10. Regression model of particle content as a function of particle diameter of fresh and used (210,000 km) gear oil SAE 75W-90.

Table 3. Basic parameters of the linear regression model for oil 75W-140.

	Fresh Oil	Oil after 350,000 km
α_1	–0.00674 ***	–0.00227 ***
(Std. Error)	(0.00024)	(0.00011)
α_0	1.40932 ***	1.40508 ***
(Std. Error)	(0.00812)	(0.0037)
Observations	89	90
R^2	0.89971	0.83432
Adjusted R^2	0.89856	0.83244
Residual Std. Error	0.05120 (df = 87)	0.02341(df = 88)
F Statistic	780.524 *** (df = 1; 87)	443.1412 *** (df = 1; 88)

Note: *** $p < 0.01$.

Table 4. Basic parameters of the linear regression model for oil SAE 80W-140.

	Fresh Gear Oil	Oil after 220,000 km
α_1	–0.00578 ***	–0.00181 ***
(Std. Error)	(0.00017)	(0.00007)
α_0	1.43028 ***	1.40962 ***
(Std. Error)	(0.00599)	(0.00224)
Observations	90	90
R^2	0.92529	0.89762
Adjusted R^2	0.92444	0.89645
Residual Std. Error	0.03792 (df = 88)	0.01415 (df = 88)
F Statistic	1.089.854 *** (df = 1; 88)	771.511 *** (df = 1; 88)

Note: *** $p < 0.01$.

Table 5. Basic parameters of the linear regression model for oil 75W-90.

	Fresh Gear Oil	Oil after 210,000 km
α_1	–0.00561 ***	–0.00260 ***
(Std. Error)	(0.00013)	(0.00011)
α_0	1.41607 ***	1.41604 ***
(Std. Error)	(0.00449)	(0.00369)
Observations	90	90
R^2	0.95418	0.86863
Adjusted R^2	0.95366	0.867174
Residual Std. Error	0.02843 (df = 88)	0.02339 (df = 88)
F Statistic	1.932.421 *** (df = 1; 88)	581.8797 *** (df = 1; 88)

Note: *** $p < 0.01$.

The comparison of the results of the lubricity tests and the amount of particles due to the amount of contaminants does not indicate a correlation between the degree of oil contamination and lubricity. A higher share of particles with the size of several dozen micrometres was found in the used oils. It is possible that these large particles, that have also been found in other wear products, are suspended in oil. This is also confirmed by the information contained in PN-ISO 4406:2005 [22], where it is stated that particles that are larger than 4 micrometres are treated as a reference value for suspended substances. It is believed that in the case of suspensions, the presence of solid particles in a liquid additionally gives the liquid a non-Newtonian liquid character, which is associated with various types of viscosity anomalies [34]. However, the content of large particles, which was much higher in the tested oils with a history of exploitation, did not adversely affect the friction coefficient in the kinematic test pair. It is possible that the content of small particles from fractional parts with diameters from 1 micrometre to 5µm in size is important. In paper [35], it was stated that the share of oil contaminants with such dimensions can be as high as 96%. The share of these particles was the highest in the volume of all of the tested gear oils. A similar situation occurred in the fresh oils and in the oils with a history of exploitation. Additives are often used during the operation of oils; these can be microscopic particles of soft metals ranging in size from 5 to 155 μm. In study [36], a beneficial effect of such particle additives on the lubricating properties of gear oils is shown. It is possible that this factor had a decisive impact on the lack of deterioration of the lubricating properties of the tested oils. The fact emphasized in other papers, including [37,38], that large particles with diameters ranging from a dozen to several dozen micrometres in size correspond to the dynamic clearance, thus determining the thickness of a lubricating film should also be noted. They are harmful because the dynamic clearance in gears that are in the contact area of the meshing teeth should be from 0.1 to 1 micrometres [39]. Large dirt particles may affect the continuity of a lubricating film.

4. Conclusions

The current paper presents an analysis of the operational contaminants that may affect the deterioration of the lubricating properties of synthetic gear oils. The main objective of the research was not only to compare the properties of oils, but to also determine the relationship between the physical properties in order to determine the condition of the oil. Two models were considered in the present work. One was the path friction coefficient. The second was the relationship between the number of contaminants and the diameter of the contaminants. For each oil, these models were fitted to the empirical data.

Based on the conducted research, the following conclusions were formulated:

1. The used oils can be characterized by a significant number of large contaminant particles. According to normative requirements and operational experience, this indicates their inability to be used to their full potential. The results of the friction tests indicate that the exploitation use of oil 75W-140 occurred after 350,000 km of mileage. It also applies to oil 75W-90, which had a much lower mileage but worked in a gearbox. Moreover, the comparison of the results of the lubricity tests and the amount of particles due to the amount of contaminants does not indicate a correlation between the degree of oil contamination and lubricity. A higher share of particles that were several dozen micrometres in size was found in the used oils. It is possible that these large particles, which also present in other products of wear, are also suspended in oil.
2. In the light of the conducted research, it seems reasonable to hypothesize that the use of fluids in the expected operational runs does not cause a critical deterioration of the lubricating and anti-wear properties. The deterioration in the properties is non-catastrophic.
3. Unfortunately, the current research has not allowed us to check how long the liquid work time in a gearbox and axle must be in order to reach critical deterioration.

The conducted research showed that the presence of contaminants is not catastrophic and that in order to fully examine the oils and to determine the critical moment, the oils with a much greater operational mileage should be tested in order to establish the relationship between the number of particles and their tribological properties. After testing a larger number of samples with different mileages, the second model supports the development of a classifier that allows the oil mileage to be estimated in technical devices. This will be the subject of further research.

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