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Modelling lubricating oil wear using fuzzy logic

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The content of this article presents research on used and fresh engine oils. The aim of the experiment is to preliminarily develop a method for assessing the condition of engine oil subjected to service. A four-ball tester was used to compare the lubricating properties of the engine oil as one component of the tribosystem under laboratory conditions. The method used to determine the mashing load consisted of subjecting the kinematic node to a linearly increasing load with a build-up rate of 409 $N \cdot s^{-1}$ under operating conditions of approximately 20°C and a spindle speed of 500 rpm. The presented article is a continuation of the consideration of the lubricating properties of engine oils subjected to operation. The tests carried out made it possible to observe that fresh oils are characterised by their ability to carry higher loads in relation to oils subjected to service. This is evidenced by the obtained values of scuffing loads, which have a higher value for fresh oils (The average percentage increase in scuffing load for fresh oils was 62.23%). Comparing the friction torque characteristics with each other, it can be seen that the values of maximum friction torque are also higher for the fresh oils group. The modelling process made it possible to characterise changes in the tribological properties of the lubricating oil being used. In the future, the described model will be extended to include further input parameters (viscosity, contaminant content, fractional composition, etc.), which will allow a multi-parametric assessment of lubricating oil wear.

Key words: *engine oils, degree of wear, four ball tester, friction, scuffing load*

1. Introduction

The primary function of oil in an internal combustion engine is to protect it from wear and to slow down the depletion of its operating potential. As is well known, machines are subject to wear and tear in the process of use. This process also applies to lubricants, which are an integral part of tribosystems.

Most often, lubricating oil changes in an internal combustion engine take place when the kilometers or time resourse is exhausted [10]. Modern cars are also equipped with systems that generate messages about the need to change engine oil based on algorithms analysing driving style or signals from oil quality sensors measuring selected oil parameters or contaminant concentrations. As recent research [17] shows, the strategies used in the lube oil change process are insufficient (low precision). This can lead to a situation where engine oil degradation is so high that the oil fails to perform its key feature, resulting in reduced engine life. This is not always unambiguous, as other studies [8] show that engine oil still retains its properties despite having exhausted its entire service life. The work [7] considering used engine oils also allows us to conclude that the vehicle mileage commonly accepted as an indicator of lubricating oil change intervals is not an unambiguously reliable criterion. Due to these discrepancies, there is a need to develop a universal method for assessing the condition of engine oil to determine its ability to protect engine components.

Effective engine wear protection occurs when a permanent lubricating oil film is formed between tribological pairs. This phenomenon should occur for a wide temperature range - both after the engine has warmed up and shortly after starting at low operating environment temperatures. Engine starting at low temperatures results in increased lubricant consumption [14]. High temperatures also contribute to engine oil degradation [12]. In the temperature aspect, engine oil plays another important function, it dissipates heat. In addition to the wide range of operating temperatures at which the oil must function, a wide range of loads is also indicated in the literature. These varying operating conditions of the oil including engine start-up, idling, varying load depending on the driving mode [16] should not adversely affect the protection of the tribological nodes and the wear intensity of the friction surfaces. Another factor related to the different temperature conditions in the engine is the formation of high and low temperature sludge. This is due to the accumulation of oxidation products, which are not neutralised by the limited amount of additives improving the lubricating properties of the engine oil. The accelerated oxidation process can be influenced by overheating of the lube oil, which leads to the formation of sludge plugs that block the flow of oil and thus the distribution of oil in the engine compartment, which contributes to the aforementioned reduction in heat dissipation [20]. Another important function of engine oil is corrosion protection [13]. Its intensity is inhibited by adding inhibitors to the oil. The anti-wear additives in the oil help to prevent abrasive or adhesive wear, but also corrosion wear.

The degradation of engine oil is contributed to by phenomena such as oxidation and the depletion of additives responsible for improving lubricating properties [2]. This results in the need for more frequent oil changes to effectively protect the engine from wear. Another reason for this phenomenon is the contaminants that accumulate during the operation process. One of them is carbon black [11], which, by deteriorating tribological properties, causes an intensification of wear of the tribosystem understood as engine and lubricant. Work [5] shows that, with the process of operation, carbonaceous contaminants accumulate in the annular grooves of the engine, resulting in oil degradation. The effect of increased soot content in engine oil is to increase viscosity and also to form an abrasive substance that damages mating surfaces [19]. The formation of soot agglomerates can lead to the closing of oil distribution channels in the engine and the local occurrence of dry friction, resulting in increased wear. A second type of engine oil contamination involving metallic elements that are solid contaminants in the engine oil can pose analogous threats to the system.

Monitoring the condition of engine oil can bring benefits not only in terms of protecting mechanical systems, but also in environmental terms. This relationship exists for both the production and disposal of engine oils [9]. The possibility of extending the service life of lube oil that continues to perform its protective function has a direct impact on reducing chemical waste.

Failure to detect a defect during engine operation can lead to damage, the consequence of which may be an increased intake of engine oil, which is burned in the engine compartment causing the vehicle to fail to meet emission standards [4]. While this is insignificant in the case of a motor vehicle engine, in the case of large marine engines, the negative environmental impact is already significant and is associated with high operating costs for such a vessel.

One of the most popular lubricant testing stations is the tribotester. The test results from this tribotester make it possible to assess the quality of the engine oil under test. As the work [6] shows, if the wear trace obtained during a test run is small, it can be inferred that the lubricant has good anti-wear properties. Extending this inference, it can be concluded that the engine oil has not yet degraded and continues to protect the kinematic node. The validity of using this apparatus when considering lubricant quality is confirmed by work [1]. The high reproducibility of the testing performed with the four-ball tester ensures that the results obtained are precise.

Due to the multitude of parameters affecting the lubricating oil wear process, it is necessary to use statistical methods to show the relationships between the various parameters. Such an approach was presented in the work [3]. Its results confirmed the great potential of the application of statistical analysis in the consideration of engine oils, as well as the importance of modelling in cognitive processes in this area.

The present work is a continuation of the study [15] of motor vehicle oils. The study of the oil's physicochemical, rheological, and tribological properties has been supplemented with results obtained using a four-ball tester. An analysis of the available literature indicates that the systems used to assess engine oil degradation status are inaccurate and may have a negative impact on the environment and the operating process of an internal combustion engine. The aim of this study is to develop a method for assessing engine oil consumption. The method is based on a model using artificial intelligence algorithms (Mamdani model). The model adopted is scalable (it can be easily extended to include further parameters affecting the oil consumption rate). The coefficients in the model were determined on the basis of test results using a four-ball apparatus. The model created must be considered as a base, which will be extended based on the results of tests to be performed in the future.

2. Methodology

2.1. Object of study

Eleven engine oils obtained from passenger vehicles used in urban and extra-urban traffic were experimentally tested. The oils had five different viscosity grades (0W30, 5W20, 5W30, 5W40, 10W40). Both fresh and used lubricating oils were tested. The mileage range for the used oils was 5002 to 15,000 km. The oils were used in both compression-ignition (2 units) and spark-ignition (9 units) engines. In two cases the spark ignition engines were powered by LPG. The vehicles from which the lubricating oils came had mileages between 15,000 and 3,622,211 km. The engines lubricated by the oils had displacements in the range of 1 to 2.5 dm^3 and had power outputs of 57 to 120 kW. Detailed data on the oils tested and the vehicles in which they were used are presented in Table 1. A passive experiment was adopted during the research (users of the vehicles from which the used oil originated decided on the course of the exploitation process and the mileage at which the used oil was replaced with fresh oil).

2.2. Tribological tests

Tribological tests were carried out using a four-ball tester. The tests carried out consisted of subjecting the kinematic node to an increasing load for 18 s. The load was

Table 1. Data on operating conditions of used oil samples [15]

Engine oil				Vehicle			
#sample number	Producent	Viscosity class SAE	Oil mileage [km]	Type of fuel to power the engine	Engine capacity [dm^3]	Nominal motor power [kW]	Car mileage at oil drain [km]
1	Fanfaro	5W30	12,650	Gasoline + LPG	1.4	63	362,211
2	Mobil	5W30	14,141	Gasoline	1.6	85	91,635
3	Shell	5W30	5481	Gasoline	1.2	57	110,007
4	Total	5W40	7734	Gasoline	1.6	120	60,631
5	Fanfaro	5W30	11,452	Gasoline + LPG	1.6	63	157,473
6	Selenia	5W30	14,998	Diesel	1.6	77	101,021
7	Shell	5W30	5002	Gasoline	1.0	57	52,333
8	Mobil	10W40	6500	Gasoline	1.6	72	330,041
9	Fanfaro	5W30	5159	Diesel	2.5	88	196,427
10	Total	5W20	5126	Gasoline	1.5	110	112,927
11	Volkswagen	0W30	15,000	Gasoline	1.0	95	15,000

built up in the range 0–7200 N, with a load build-up rate of 409 N·s^{-1} throughout the run. The spindle speed was 500 rpm. The initial temperature of the test lubricant was in the range of 20–25°C. During the tests, the friction torque, the temperature in the grip of the kinematic node and the load applied to the friction node were measured. The measurements were sampled at a frequency of 50 Hz. The tests were performed in accordance with the requirements of PN-76/C-04147. Three tests runs were performed for each lubricating oil during the experiment. The result characterising each lube oil was derived from the average of the three attempt. They were subjected to an assessment as to whether there was a result with a gross error in a given set of results (Q-Dixon test).

3. Results

3.1. Results of tribological tests

Based on the experimental results, a graph of friction torque in the load domain (Fig. 1) was drawn up for each engine oil tested. Oils were designated by numbers preceded by the "#" sign. Numbers 1 to 11 were assigned to oils that had been used in the engines, while numbers 12 to 19 were assigned to corresponding fresh lubricating oils (the smaller number of fresh oils is due to the use of the same oils in more than 1 engine).

In the load range from 0 to about 1300 N for all engine oils tested, the course of the mashing torque showed almost the same increase. Changes can be observed in the range 1300–2800 N. In this range, a dynamic increase in frictional torque was recorded for all samples, but at a load of about 1790 N (red line), 2 groups of oils for which the dynamics of change differed were distinguished. In the first group, a dynamic increase in frictional torque was observed earlier, and these were oils in service, while the second group, for which a sharp increase in rubbing torque occurred later, were fresh oils. To detail this phenomenon, the scuffing load values for in-service and fresh lube oils were read from Fig. 1 and the relative differences between them were determined (Table 2 and Fig. 2). In addition, it was observed that, for fresh oils, the maximum values of the frictional torque were higher than for in-service oils and the load range in which the dynamic increase and decrease in frictional torque occurred was narrower than for the runs obtained for in-service oils.

Figure 2 shows the scuffing load values for used and fresh engine oils.

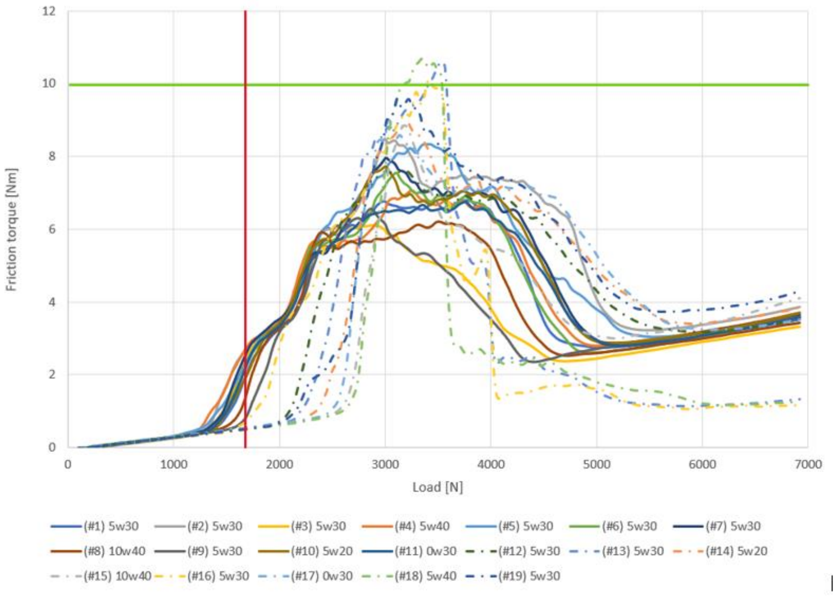


Fig. 1. Friction torque waveform in the load domain of a kinematic node

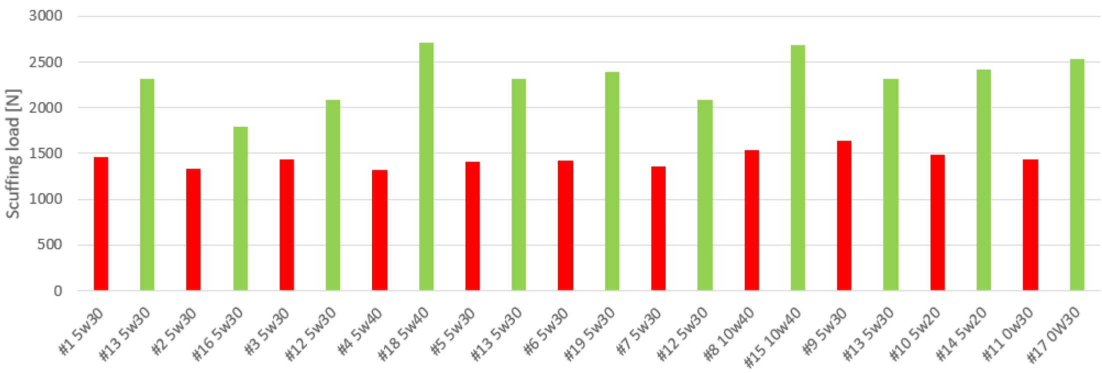


Fig. 2. Measured scuffing load values: green – fresh engine oils, red – used engine oil

Table 2. Scuffing load of engine oils

Group	Engine oil	Scuffing load [N]	Relative increase in scuffing load [%]
1 – used	#1 5W30	1467	57.94
2 – fresh	#13 5W30	2317	
1 – used	#2 5W30	1329	34.99
2 – fresh	#16 5W30	1794	
1 – used	#3 5W30	1431	45.98
2 – fresh	#12 5W30	2089	
1 – used	#4 5W40	1323	105.44
2 – fresh	#18 5W40	2718	
1 – used	#5 5W30	1407	64.68
2 – fresh	#13 5W30	2317	
1 – used	#6 5W30	1426	67.39
2 – fresh	#19 5W30	2387	
1 – used	#7 5W30	1360	53.60
2 – fresh	#12 5W30	2089	
1 – used	#8 10W40	1544	74.29
2 – fresh	#15 10W40	2691	
1 – used	#9 5W30	1641	41.19
2 – fresh	#13 5W30	2317	
1 – used	#10 5W20	1493	62.09
2 – fresh	#14 5W20	2420	
1 – used	#11 0W30	1434	76.92
2 – fresh	#17 0W30	2537	

It was observed that all of the scuffing loads for fresh oils were greater than those for in-service oils. In the case of fresh oils, the values were greater from 34.99% to as high as 105.44%. The average percentage increase in mashing load was 62.23%, and the median was 62.09%.

After statistical processing of the results obtained, an equation describing the average relationship between oil mileage and scuffing load was determined:

$$F = -0.07128 \cdot P_O + 2222.65752 \text{ [N]} \tag{1}$$

where P_O – oil mileage [km].

A green line was also plotted in Fig. 1, visualising a friction torque level of 10 Nm, above which the machine drive was switched off for safety reasons. Three of the samples tested exceeded this threshold. All of the oils tested, for which the friction torque of 10 Nm was exceeded, belonged to the group of fresh oils. It should also be noted that, during the experiment, in some cases there was a phenomenon of welding of the top ball placed in the machine spindle with the balls placed in the chuck. This phenomenon was noted in the case of three oils: #13 5W30 (in two of the three trials), #16 5W30 (in two of the three trials) and #18 5W40 (in one of the three trials). These data are the same as the samples that passed the machine's safety threshold.

It is not only the scuffing forces that change with oil mileage. Another parameter for which a similar relationship was noted is wear diameters. These were determined during ball apparatus tests in accordance with ASTM D 4172-94 and are described in [15]. The values of the wear diameters are shown in Fig. 3. In this case, an increase in wear diameter with increasing oil mileage was observed in all tests.

After statistical treatment of the data shown in Fig. 3, an equation describing the average relationship between oil mileage and wear diameter was determined:

$$D = 0.03250253 \cdot P_O + 597.67580742 \text{ [µm]} \tag{2}$$

where P_O – oil mileage [km].

4. Model for estimating oil consumption

As part of the work, a model was built to estimate oil consumption rates. When building the model, it was assumed that it must be scalable (the possibility of adding further input parameters on the basis of which the degree of wear will be determined). For this reason, it was decided to use the Mamdani model, which belongs to the group of models from the area of artificial intelligence (fuzzy logic). To build the model, data obtained during the experimental research described in Chapter 3 were used.

The input variables in the model were:

- scuffing force F [N]
- wear diameter D [µm]
- oil mileage P_O [km]

and output variables:

- wear rate Z [%].

In the first stage of model construction, linguistic variables were adopted for which their spaces (membership functions) were determined. Next, the coefficients in the membership functions were determined based on equations 1 and 2, assuming that the oil mileage according to the service life should be 15,000 km (readings of significant values from equations 1 and 2 for mileage: 3.75, 7.5, 11.25 and 15,000 km). To obtain the fuzzy consequences, the limiting equations of the coordinate of the ordinate axis were determined and the rule aggregation equation was determined. The results of this work are presented in Tables 2 and 3.

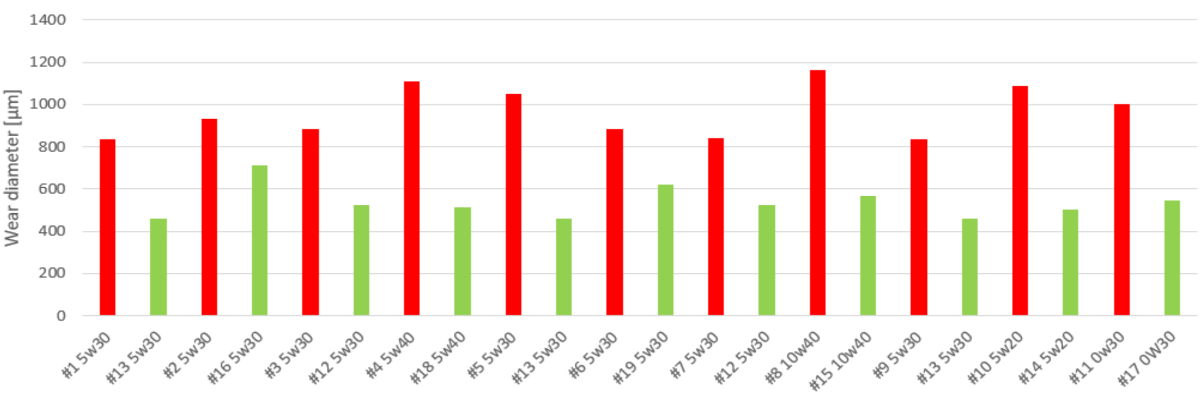


Fig. 3. Measured values of the wear diameter [14]: green – fresh oil, red –used oil

Calculations were made using the model, the results of which are shown in Fig. 4. The calculations were made for data obtained during experimental tests (engine oil runs, scuffing forces and wear diameters). Oil wear rates were determined in 3 variants (by condition, by remaining oil life, mixed model).

Analysing the calculation results shown in Fig. 4, it was found that the oil consumption rate determined from the model by condition indicates greater oil degradation for mileages up to about 11,000 km compared to the value of this parameter determined from the model by resurfacing. The relationship takes on an inverse relationship once this mileage is exceeded. This is due to the fact that the determined values of engine oil degradation by condition were based on tribological parameters. The values of these parameters are strongly influenced by the oil's rheological properties (viscosity), which change during operation. The lubricating oil's viscosity changes in the process of use. After an oil change, viscosity initially decreases and reaches a minimum after a few thousand kilometres. The viscosity then starts to increase to reach a value close to or greater than the viscosity of fresh engine oil after the resurface is exhausted. For a description of this phenomenon, see [18]. Oils with higher viscosity are more resistant to oil film rupture, which has a positive effect on the wear process in highly loaded tribological pairs.

5. Conclusion

The tribological tests carried out allow us to conclude that there is a significant difference between fresh and in-service oils. The friction torque waveforms as a function of load for fresh oils are more dynamic in a narrower load range than in the case of used oils. Significantly lower scuffing loads occurred in the group of in-service oils compared to fresh oils, indicating that fresh oils have the ability to carry higher loads and protect the mating components more effectively. As the results show, the opposite is true for maximum friction torque. It is the used lube oils subjected to the same load that show a lower friction torque value than fresh oils. For all the engine oils tested, a decrease in scuffing load and an increase in wear diameters were observed during operation. These phenomena were used in the construction of a model to estimate the degree of wear. The developed model uses artificial intelligence algorithms (fuzzy logic) and can be extended with new input parameters, which will allow it to increase its accuracy of inference after adding further parameters that will be identified in the course of future research. The presented results from the modelling process show that it is possible to determine the degree of engine oil wear based on its condition and remaining life. Thus, by comparing the calculation results, it is possible to show the differences in the dynamics of oil consumption changes according to the two strategies. In addition, the changing differences in the degree of engine oil wear according to the different strategies showed that the tribological parameters determined during oil testing do not have a linear characteristic associated with the run and are related to the rheological parameters of the oil, which include viscosity. This parameter has a non-linear characteristic during the operating process. In the first stage, it decreases, and after reaching a minimum, which occurs after a few thousand kilometres of mileage, it then begins to increase mainly due to the accumulation of wear products and contaminants in the engine oil.

The reduction in lubricating oil viscosity results in a lower oil film life and thus protection of the mating components against wear. The advantage of the represented approach is that the presented model can be successively extended with further criteria for the assessment of lubricants so that a comprehensive evaluation can be carried out. As a result of the modelling, it was found that oil consumption assessed by mileage in the range from 0 to about 50% of the assumed distance, which was 15,000 km, was higher than the consumption determined by condition. In the second mileage interval (above 50%), consumption according to condition was lower than consumption according to mileage. This demonstrates the non-linear behaviour of wear by condition. It has a degressive course. In the initial phase, lubricating oil wear has a large increase because there is a decrease in viscosity caused by changes in the fractional composition of the lubricating oil. After this period, the tribological properties stabilise and the oil wear process decelerates. Therefore, the modelling process made it possible to characterise the changes in the tribological properties of the lubricating oil being worn. In the future, the described model will be extended with further input parameters (viscosity, contaminant content, fractional composition, etc.), which will allow a multi-parametric evaluation of lubricating oil wear.

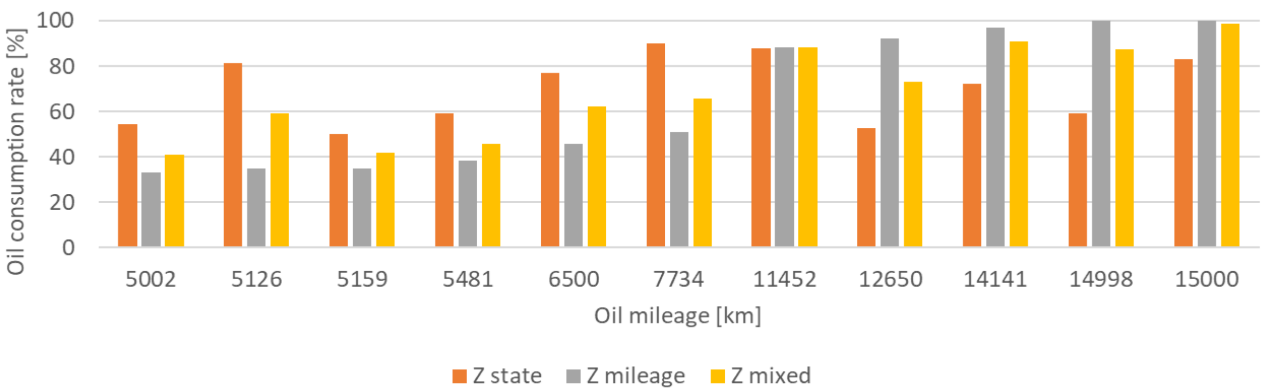


Fig. 4. Results of oil consumption calculation from the model in 3 variants (by condition, by remaining oil life, mixed model)

Table 3. Structure of the oil consumption estimation model (output variables)

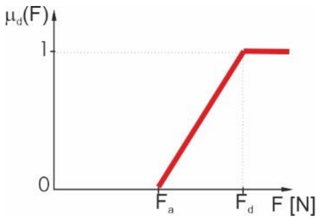
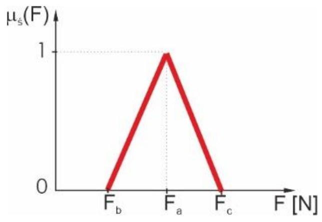
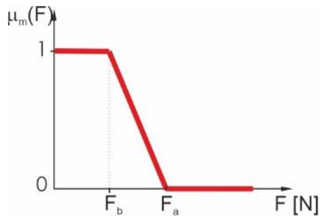
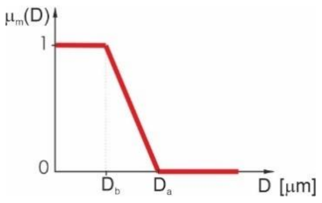
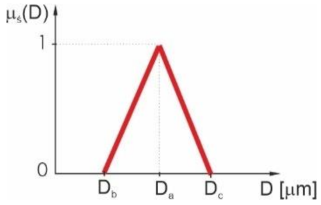
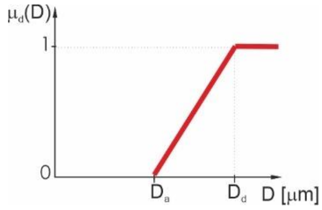
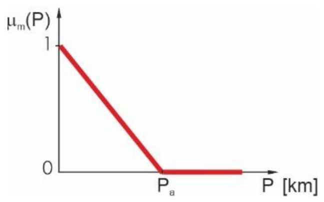
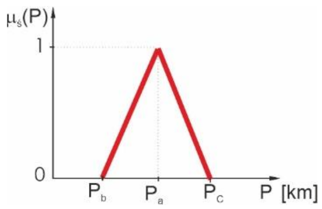
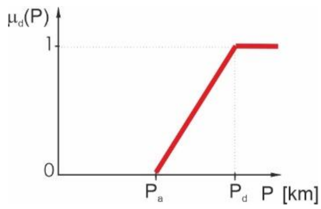
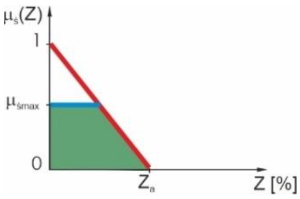
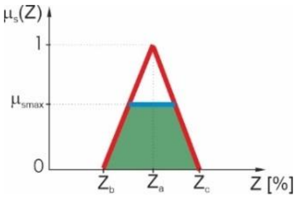
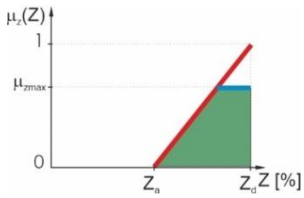
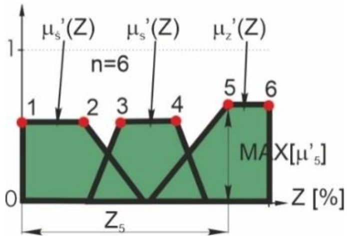
Input variable	Scuffing load **F**		
Linguistic variables	High scuffing load **μ_d(F)**	Medium scuffing load **μ_s(F)**	Low scuffing load **μ_m(F)**
Affiliation functions			
Equations	$\begin{cases} 0 \leftrightarrow F < F_a \\ \frac{F - F_a}{F_d - F_a} \leftrightarrow F \geq F_a \wedge F \leq F_d \\ 1 \leftrightarrow F > F_d \end{cases}$	$\begin{cases} 0 \leftrightarrow F < F_b \vee F > F_c \\ \frac{F - F_b}{F_a - F_b} \leftrightarrow F \geq F_b \wedge F \leq F_a \\ \frac{F_c - F}{F_c - F_a} \leftrightarrow F > F_a \wedge F \leq F_c \end{cases}$	$\begin{cases} 1 \leftrightarrow F < F_b \\ \frac{F_a - F}{F_a - F_b} \leftrightarrow F \leq F_a \wedge F \geq F_b \\ 0 \leftrightarrow F > F_a \end{cases}$
Coefficients in the equations	**F_a** = 1688 N, **F_b** = 1420 N, **F_c** = 1955 N, **F_d** = 2222 N		
Input variable	Wear diameter **D**		
Linguistic variables	Small diameter wear **μ_m(D)**	Medium diameter wear **μ_s(D)**	Large diameter wear **μ_d(D)**
Affiliation functions			
Equations	$\begin{cases} 1 \leftrightarrow D < D_b \\ \frac{D_a - D}{D_a - D_b} \leftrightarrow D \leq D_a \wedge D \geq D_b \\ 0 \leftrightarrow D > D_a \end{cases}$	$\begin{cases} 0 \leftrightarrow D < D_b \vee D > D_c \\ \frac{D - D_b}{D_a - D_b} \leftrightarrow D \geq D_b \wedge D \leq D_a \\ \frac{D_c - D}{D_c - D_a} \leftrightarrow D > D_a \wedge D \leq D_c \end{cases}$	$\begin{cases} 0 \leftrightarrow D < D_a \\ \frac{D - D_a}{D_d - D_a} \leftrightarrow D \geq D_a \wedge D \leq D_d \\ 1 \leftrightarrow D > D_d \end{cases}$
Coefficients in the equations	**D_a** = 841 µm, **D_b** = 719 µm, **D_c** = 963 µm, **D_d** = 1085 µm		
Input variable	Oil mileage **P_O**		
Linguistic variables	Low engine oil mileage **μ_m(P)**	Average engine oil mileage **μ_s(P)**	High engine oil mileage **μ_d(P)**
Affiliation functions			
Equations	$\begin{cases} \frac{1 - P}{P_a} \leftrightarrow P \leq P_a \\ 0 \leftrightarrow P > P_a \end{cases}$	$\begin{cases} 0 \leftrightarrow P < P_b \vee P > P_c \\ \frac{P - P_b}{P_a - P_b} \leftrightarrow P \geq P_b \wedge P \leq P_a \\ \frac{P_c - P}{P_c - P_a} \leftrightarrow P > P_a \wedge P \leq P_c \end{cases}$	$\begin{cases} 0 \leftrightarrow P < P_a \\ \frac{P - P_a}{P_d - P_a} \leftrightarrow P \geq P_a \wedge P \leq P_d \\ 1 \leftrightarrow P > P_d \end{cases}$
Coefficients in the equations	**P_a** = 7500 km, **P_b** = 3750 km, **P_c** = 11250 km, **P_d** = 15000 km		

Table 4. Structure of the engine oil consumption estimation model (output variables)

Output variable	Degree of wear **Z**		
Linguistic variables	Fresh oil $\boldsymbol{\mu_{\acute{s}}(Z)}$	Average oil consumption $\boldsymbol{\mu_{s}(Z)}$	Used oil $\boldsymbol{\mu_{z}(Z)}$
Affiliation functions			
Equations	$\begin{cases} \frac{1-Z}{Z_a} \leftrightarrow Z \leq Z_a \\ 0 \leftrightarrow Z > Z_a \end{cases}$	$\begin{cases} 0 \leftrightarrow Z < Z_b \vee Z > Z_c \\ \frac{Z-Z_b}{Z_a-Z_b} \leftrightarrow Z \geq Z_b \wedge Z \leq Z_a \\ \frac{Z_c-Z}{Z_c-Z_a} \leftrightarrow Z > Z_a \wedge Z \leq Z_c \end{cases}$	$\begin{cases} 0 \leftrightarrow Z < Z_a \\ \frac{Z-Z_a}{P_d-P_a} \leftrightarrow Z \geq Z_a \end{cases}$
Coefficients in the equations	$\mathbf{Z_a}$ = 50%, $\mathbf{Z_b}$ = 25%, $\mathbf{Z_c}$ = 75%, $\mathbf{Z_d}$ = 100%		
Limiting equation (by state)	$\mu_{\acute{s}max} = \frac{\mu_d(F)\cdot\mu_m(D)}{2}$	$\mu_{smax} = \frac{\mu_{\acute{s}}(F)\cdot\mu_{\acute{s}}(D)}{2}$	$\mu_{zmax} = \frac{\mu_m(F)\cdot\mu_d(D)}{2}$
Limiting equation (according to resource)	$\mu_{\acute{s}max} = \mu_m(P)$	$\mu_{smax} = \mu_{\acute{s}}(P)$	$\mu_{zmax} = \mu_d(P)$
Limiting equation (mixed)	$\mu_{\acute{s}max} = \max\left\{\frac{\mu_d(F)\cdot\mu_m(D)}{2}, \mu_m(P)\right\}$	$\mu_{smax} = \max\left\{\frac{\mu_{\acute{s}}(F)\cdot\mu_{\acute{s}}(D)}{2}, \mu_{\acute{s}}(P)\right\}$	$\mu_{zmax} = \max\left\{\frac{\mu_m(F)\cdot\mu_d(D)}{2}, \mu_d(P)\right\}$
Rule aggregation			$Z = \frac{\sum_{i=1}^{n} Z_i \cdot \max[\mu_i']}{\sum_{i=1}^{n} \max[\mu_i']}$

Nomenclature

D wear diameter
F scuffing load
P_O oil mileage
Z wear rate
Z mileage wear rate calculated with oil mileage
Z state wear rate calculated with tribological tests results
μ linguistic variables

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