EXPERIMENTAL INVESTIGATION ON EXTREME WEAR RESISTANCE OF A METALWORKING OIL WITH PHOSPHORUS ADDITIVES

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Metalworking fluids are utilized to evacuate friction heating and to prevent thermal damage to the workpiece. To reduce friction and tool wear the metalworking fluid should generate some lubrication. Water-based emulsions and oils are still popular for metalworking even if biofriendly alternatives have started to have an important impact. Understanding the limitations and performances of such fluids may contribute to reducing energy losses and fluid consumption. This paper presents an experimental study on a four-ball machine to understand the performance of a metalworking oil with phosphorus additives under extreme pressure conditions. Results show a relatively low friction coefficient and relative low wear scar.

Keywords: metalworking, oil, wear, four-ball, lubrication, friction coefficient.

1. Introduction

Friction and lubrication are of great importance in metalworking processes, influencing the efficiency of the process and directly affecting the mechanics of individual operations and the quality of the final product. Substantial heat is generated by machining, and metalworking fluids are utilized to prevent thermal damage to the workpiece. Their key function is dissipation of heat for cooling the workpiece. Other functions are to minimize friction by lubrication, as well as flushing chips from the contact zone. Metalworking fluids contribute significantly to the machining process by reducing tool wear, improving surface quality and dimensional accuracy.

Traditionally, water is widely used as a base for coolant fluids due to its thermal stability and accessibility. However, its use presents concerns about corrosion and most important, insufficient lubrication [1]. Oil-based cutting fluids,

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including those with extreme pressure additives, are known to have offered alternative solutions over time, and are best suitable for heavy duty machining operations. In the literature, the bulk of research is directed to the development of biocompatible cutting fluids. Cutting oils are found in applications where other types of fluids are not viable. Recent studies show the usage of cutting oils for mist reduction [2][3][4], improvement of heat dissipation [5], sub-zero high performance cutting [6] and solution to recycle them [7].

Water-based emulsions and solutions (using chemical or petro-chemical additives) and cutting oils are still popular for metalworking. In the meantime, more complex water-based solutions (known as synthetic fluids) are still being developed to improve cooling and lubrication in machining. The choice of cutting fluids is influenced by various factors and must consider the environmental pollution and human health hazard. Various agents and additives are used to improve mineral oils properties (extreme pressure lubrication, anti-wear, rust prevention, heat dissipation, etc.), for a wide range of industrial machining operations. Metalworking oil can generate mist and endanger human health by inhalation and dermal exposure with concomitant ingestion. Regulations and directives in the USA, China and European Union try to reduce the consumption of the conventional mineral based metalworking fluids. The latest research papers explore the performances of biobased metalworking fluids. Results are available for edible vegetable oils like: canola, sunflower, corn and soybean and some vegetable-based formulas [8][9]. On the other hand, mineral oils are hard to replace because are relative cheap compared to oils from renewable sources, and their poor lubrication properties can rather easily be improved by additives. Understanding limitation and performances of mineral based fluids can contribute to reducing energy loses, fluid consumption and increase durability.

The main objective of this work is to evaluate high pressure resistance of highly additivated metalworking oil. To overcome the problem and determine the oil wear characteristics, a four-ball tester experimentation is performed. The four-ball tester allows the study of thin film in conditions of extreme pressure lubrication. It is useful to study lubricants for gears, metalworking, and other applications. The four-ball tester is widely used, even if the sliding motion is not so common in applications for greases or lubricants. Wear analysis is done in two steps: varying load and measuring the scars. A comparison with conventional cutting fluids, in terms of friction coefficient and wear scar is made.

2. Materials, testing machine and test conditions

2.1 Metalworking oil

AZUR CUT 602.01 M-10 (manufactured by AZUR S.A. Romania) is based on hydro-treated mineral oils with a low aromatic content. It is based on a combination of polar, extreme pressure and anti-wear additives, which provides the oil with high load carrying properties. To the base oil (80 wt%), it is added a high percentage of active and inactive sulphur (19%), phosphorus, polyisobutylene and an antioxidant. The high additive content improves extreme pressure properties and allows for high rates of metal removal. AZUR-CUT 602.01 M-10 has a density of 0.87 g/cm^3 at 20°C and the viscosity at 40°C is 10 cSt. The manufacturer indicates excellent results with many heavy-duty operations (e.g. deep-hole drilling). The physical properties of interest (i.e. density, viscosity, viscosity index VI and pour points) are identified, based on manufacturer technical data (Table 1).

Table 1

Physical properties of tested oil AZUR CUT 602.01 M-10

AZUR CUT 602.01 M-10	Units	Value
Density (20°)	g/cm3	0.87
Kinematic viscosity at 40 °C	cSt.	10
Kinematic viscosity at 100 °C	cSt.	3
Flash point	°C	115
Pour point	°C	-10

2.2 Four ball machine

The experimental device used is a four-ball tester. The frictional contact is obtained by pressing one rotating ball against three stationary balls, firmly held together inside a cup and a holding support. The contact is immersed in lubricant (Fig. 1) by adding testing oil inside the cup. The three lower balls are pressed against the upper ball using weights and a lever. The lever amplifies the load 10 times. A force sensor measures the friction torque of the cup, at the end of a small loading arm mounted on the side of the cup. Friction torque M_f is obtained by multiplying force F_s with the distance R=184 *mm*:

$$M_f = F_s \cdot R \tag{1}$$

Oil temperature measurement is performed with a thermocouple inserted into the holding cup of the stationary balls. Temperature and force are recorded using 5B47 Analog Devices, respectively SCM5B38 Data forth modules and a 5B08 Analog Devices backplanes 98 (8 channels) to support data acquisition.

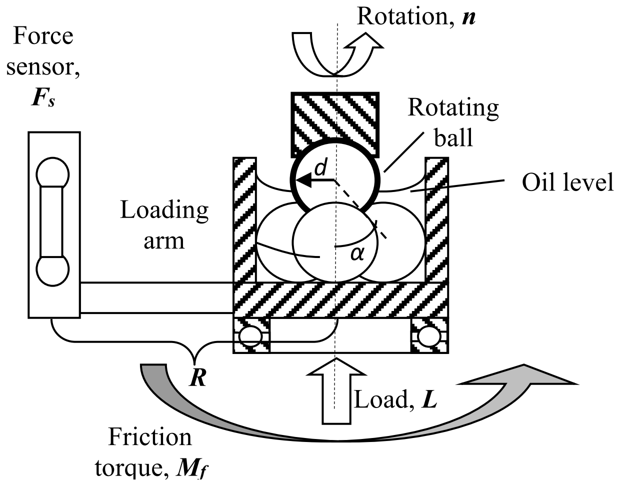

Fig. 1. Cross section of the four ball and cup assembly

2.3 Experimental procedure

For each conducted test, new balls are used, cleaned and then dried. The diameter of the balls is d=12.7mm (0.5 inch) made of chrome steel AISI 52100 — EN 10027 100CR6 (1.3505), having a hardness in the range of 64-66 HRC. Over the balls, enough lubricant is added to cover them with 1-2 mm. Weights are placed on the loading lever to provide the desired load for each test. The computer and data acquisition software are started prior to the test start. The motor is turned on and the timer is initiated simultaneously with the applied load (at that point, contact is made between the cup small lever and the force sensor). At the end of the test, the load is removed, the motor is stopped, and the experimental data is saved.

The tribological tests are focused on the extreme pressure limits of the tested lubricants. The test method is based on the Romanian Standard SR EN ISO 2062-2018, Wear Test C. Wear analysis for this oil is conducted at 1425±50 rpm for 60±5 seconds, using weights ranging from 10 kg to 79 kg, starting at ambient temperature of 25°C. For each load incrementally obtained, the force sensor and oil temperature were recorded simultaneously, in actual time. From the moment when the load was applied, the recording time was 60 s. At the end of the test, the 3 balls from the cup are cleaned and wear scar diameters (WSDs) are analyzed using an optical microscope (Nikon SMZ1000).

3. Experimental Results

Friction in a lubricated point contact of steel rises sharply if load, speed or oil temperature exceed certain critical values. In literature a transition diagram

Load-relative speed has been firmly established for thin-film lubrication of newly assembled sliding concentrated steel contacts [10]. Three regions are underlined: (I) the load is low, and a fluid film is maintained (neglectable wear), (II) an intermediary region with mild wear and (III) where severe wear and scuffing appear due to a virtually unlubricated contact. For mild loads, roughly approximated between 500 and 1000N [11-12], the contact between balls is protected by boundary lubrication against severe adhesive wear. Increasing load will create a transition from mild to severe adhesive wear [13]. During test high oil temperature inside contact will reduce viscosity and reduce load capacity. Figure 2 presents the friction torque variation during test time interval, for the last tests before seizure. One can see that the friction torque is constant. The loading conditions are mild, and we can assume that the contact between balls is characterized by boundary lubrication. A spike appears right at the beginning, when the load is manually applied, and the loading lever is released, causing a small initial shock.

In severe conditions, the film lubrication fails and seizure with scuffing appears, with sudden increase of friction and wear. Figure 3 presents the variation of the friction torque during 60 s, for the load force of 882 N, 981 N and 1080 N, when seizure appears. On the graph, a sudden increase in friction torque can be observed for G=981 N and G=1080 N at approximately 20 s after start. After incipient scuffing occurs due to metal-to-metal contact, the contact area increases, producing further reduction of contact pressure. This allows for a better access of the oil inside the contact and produces a decrease in the coefficient of friction up to a relatively constant value between 0.25-0.3. Results in Figure 4 reveal the first sign of scuffing. The friction torque measured for a force of 1766 N reveals an extreme increase of friction in the presence of extreme contact pressure. The presence of scuffing was also proven using microscope images. The images in Figure 6 present proof of the process previously presented on the graphs. The wear scars for the same loads (G=981 N and G=1080 N) revealed the presence of scuffing and metal-to-metal contact.

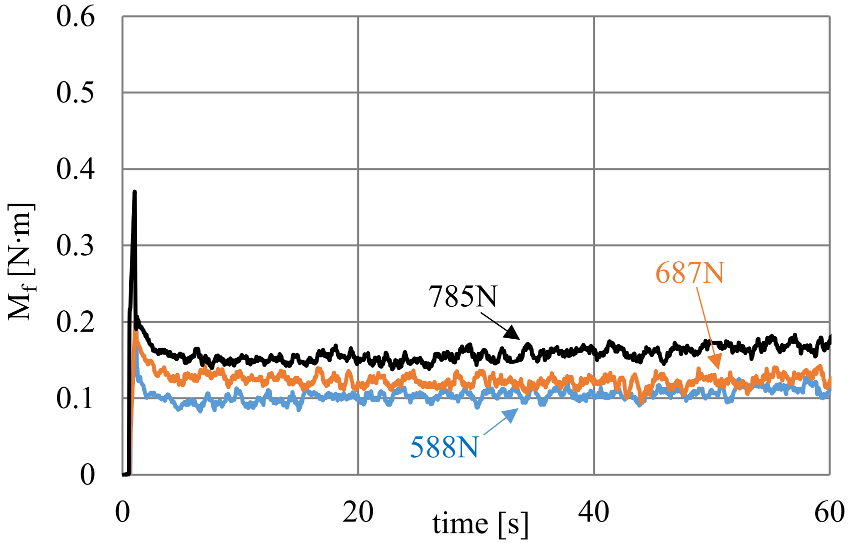

Fig. 2. Friction torque variation during 60s test for 588N, 687N and 785N load force

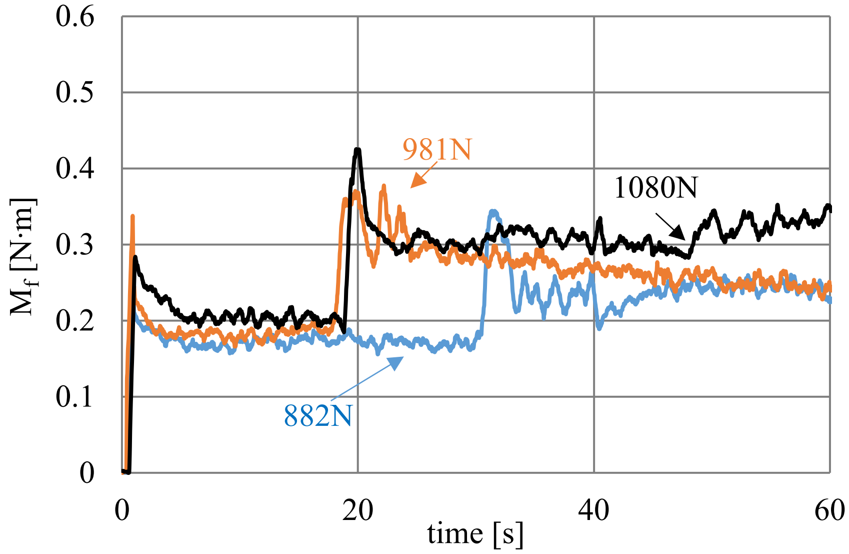

Fig. 3. Friction torque variation during 60 s test, for 882 N, 981 N and 1080 N load force

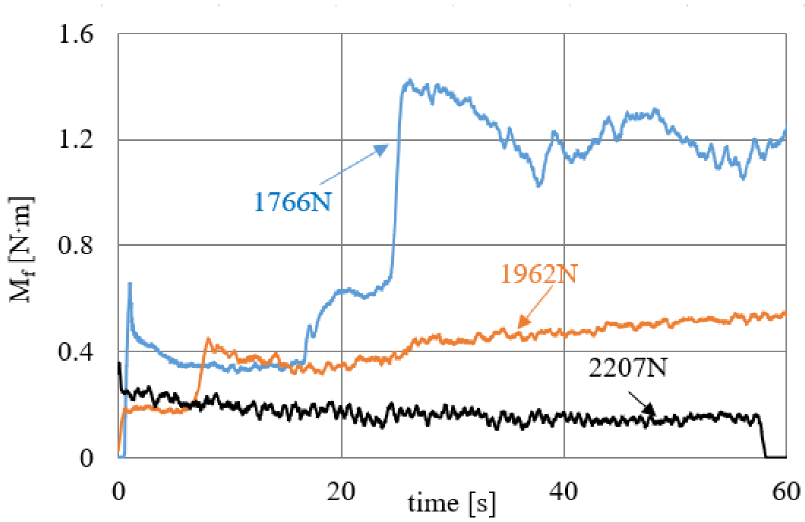

Fig. 4. Friction torque variation during 60 s test, for 1766 N, 1962 N and 2207 N load force

More information about friction and wear can be found if the friction coefficient (μ) evolution is plotted during tests (Fig.5). For a test the relative sliding speed and load is maintained constant, only local temperature of the fluid modifieds due to local contact and friction. The load is increased only at the beginning of each test. The friction regime can be identified also using the value for friction coefficient. The threshold values for coefficient of friction of 0.1 and 0.3 and indicates when mild regime begins, respectively end [10]. The values calculated for friction coefficient and presented in figure 5 show that despite the increase of load the friction regime is mild up to the maximum load of 3096N. Above this value the friction regime was severe with visible signs of seizure.

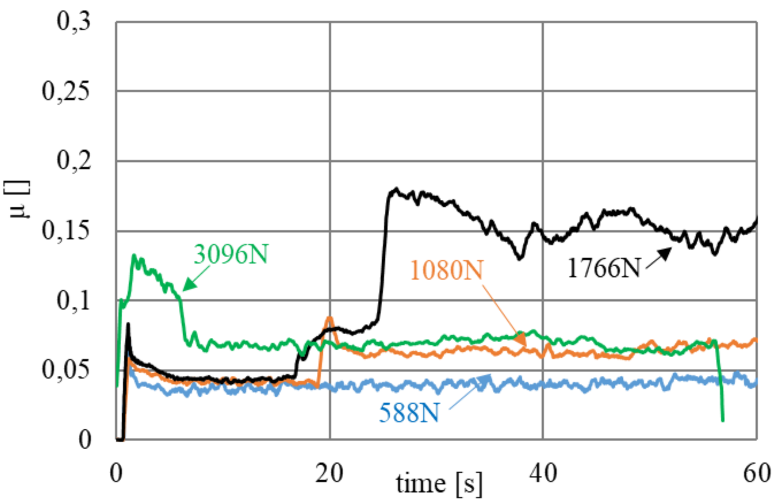

Fig.5 The evolution of friction coefficient (COF) in time

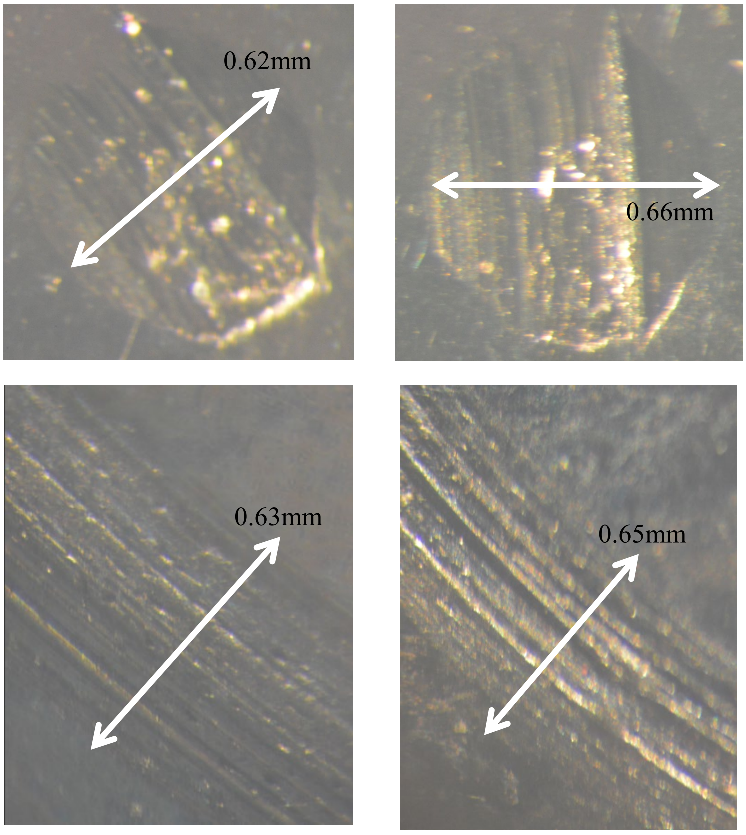

Fig. 6. Wear scar images revealing first signs of scuffing for tests with load G= 981 N (left) and G=1080 N (right)

The wear scar diameter (*WSD)* was determined using indications from SR EN ISO 20623-2018, as the average diameter of the wear scar, measured after each test, on the three stationary balls, with two measurements on each fixed ball, one in the sliding direction, the other – perpendicular to it. The wear scar dimension is an indicator of oil performance: the better the lubricant is at preventing wear, the smaller the wear scar dimension will be. Figure 7 presents the evolution of average scar diameter with load. A correlation could be made if we compare this curve with a typical plot of scar diameter versus applied load, as presented in ASTM D2783 and also in SR EN ISO 20623-2018. Based on this correlation, we can observe that around 1600 N, it could be noticed the last non-seizure load. Above this value, a region characterized by high variation of wear scar diameter can be found (G=1600…3200 N). We believe there is a temporary breakdown of the lubricating film under mild loads. This breakdown is noted by a momentary increase of the friction inside ball contacts. For high loads, $G \geq 3200$ N, the wear scar - load curve is characterized by seizure, welding at the startup or by large wear scars. High loads and high contact pressure generate metal-to-metal contact in each consecutive test and the wear scars diameters reach a plateau. For low loads, G<3200N (Fig. 8), an analysis can be done in terms of fluid film lubrication. One can observe a small increase of wear scar dimensions corresponding to small loads and thin film lubrication. Further on, if the load increases, the boundary lubrication becomes dominant and wear scars reach a plateau. If the load continues to increase, the metal-to-metal contact produce extreme wear. For loads in between 1600÷2400 N, the results are scattered, and this can be justified by the presence of unpredictable wear phenomena like adhesion and local plastic deformations.

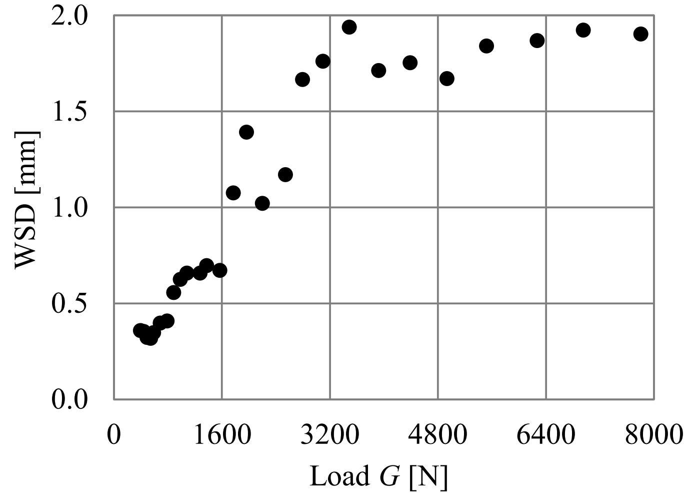

Fig. 7. Mean wear scar diameter variation with load for all conducted tests

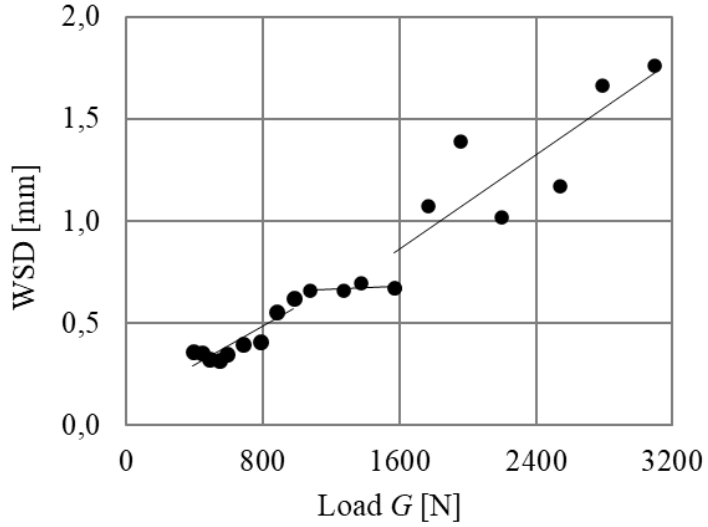

Fig. 8. Mean wear scar diameter variation with load focused only on low loads up to 3200 N

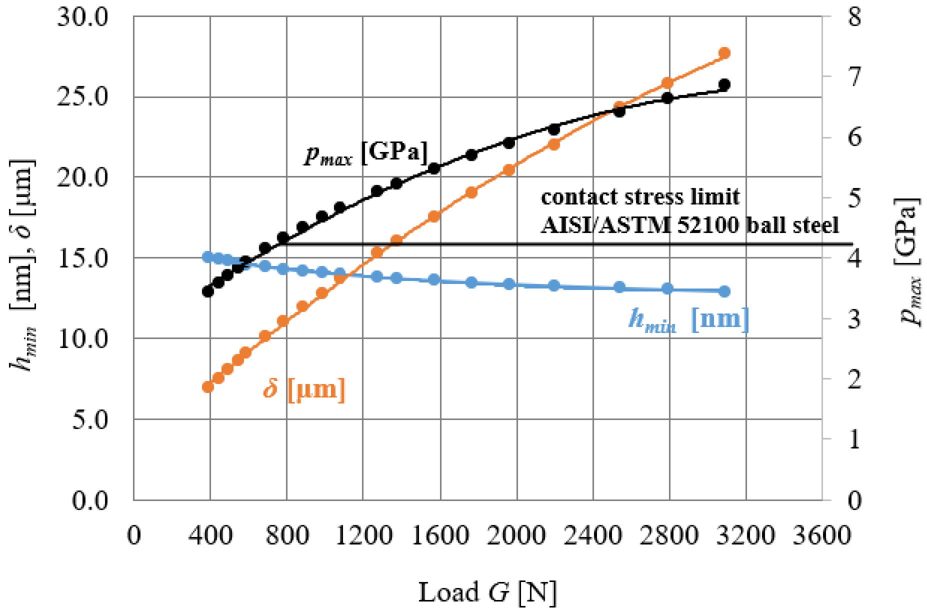

Fig. 9. Theoretical determination of Hertzian contact pressure and EHL fluid film for the loads used in the measurements

To gain a deeper insight into the ball contact, theoretical determinations are made for the Hertzian contact. The presence of the metalworking fluid forced us to consider the existence of EHL lubrication. At the beginning of the test, the balls are pressed together in dry contact, and this produces an initial wear scar. When relative motion is maintained, in the presence of the fluid, an EHL film appears, reducing wear. As a result, wear scars become constant and independent of load. This is in accordance with EHL as long as forces are relatively low and there is no plastic deformation of the balls. In Figure 8, the contact stress limit is drawn for AISI/ASTM 52100 steel, which is typical for four-ball tests. Theoretical determinations of contact indentation δ using Hertz model and EHL fluid film using Hamrock & Dowson [14] were made and the results presented in Figure 9. Pressure-viscosity coefficient was roughly approximated by similarity with an "*oil D*" found in literature [15]. One can see that minimum fluid film h_{min} is low under 15 nm. Figure 8 shows that loads above 800 N generate stresses surpassing AISI/ASTM 52100 steel contact stress limit. This is correlated with the rapid increase of wear scar, shown in Figure 8, which starts with the same load.

The axial load is distributed among the 4 balls, on the lines of a regular tetrahedron. The friction coefficient for the contact ball-on-ball can be determined using:

$$\mu = \frac{M_f}{\frac{d}{2} sin(\alpha) \frac{F}{cos(\alpha)}} \tag{2}$$

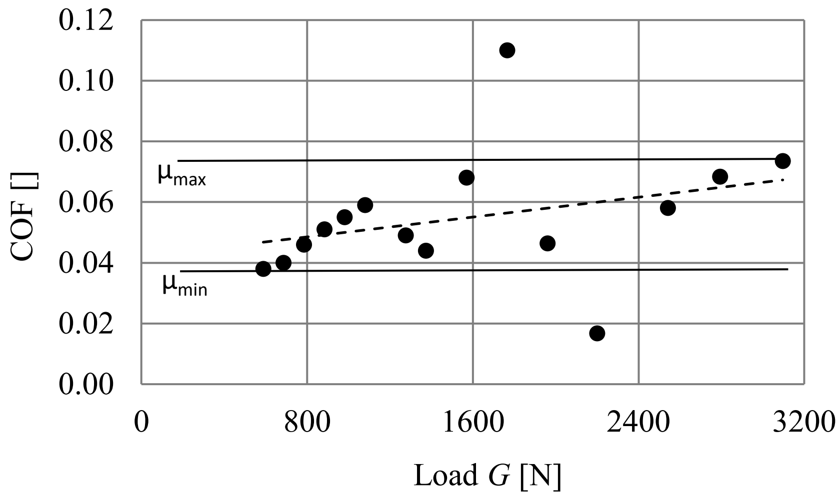

Fig. 10. Friction coefficient variation

The average friction coefficient variation with load is presented in Fig 10, for loads <3200 N. For loads above this value, the data acquisition system was not used due to high risk of damaging the force sensor. One can see that the friction coefficient varies in-between a relatively small interval 0.04÷0.08, with some exceptions. To understand these exceptions, we must go back and look at the friction torque variation from Figure 4. This graph includes a torque variation typical for a load *G*=1962 N. For *G*=1766 N (Fig. 4), the graph shows a friction increase right after the moment when the load is manually applied, and this produced further strong seizure with no recovery. For *G*=2207 N (Fig. 4), the friction torque at contact was very low and no seizure can be observed. For this load, the test was shorter with approx. 3 s due to human error.

The main advantage of using the four-ball tester is the standardized experimental procedure. This allows for an easy comparison between the obtained results and others from similar research studies (Table 2). To compare results for different speeds and/or testing times (different sliding distances for each speed) one needed a particular solution. The wear rate of the wear scar diameter, noted by *w* [16], is considered very useful for this type of situation and considers the sliding distance *S*. The wear rate of WSD, *w*, can be determined using:

$$w = \frac{WSD}{L \cdot S} \tag{3}$$

After thorough research in literature related to four-ball test, two other mineral oils, for which the wear rate could be calculated were selected and compared in Table 2. *SW* is a commercial oil studied by Suda et al. [17], while *MO-WNP* is a mineral oil-based metalworking fluid, studied by Prabu and Saravanakumar [18]. The results revealed that AZUR CUT 602.01 M-10 shows good performances in reducing friction coefficient. The wear rate is significantly higher when compared to oils for similar loading conditions. Some explanations can be found: our test is shorter and more influenced by errors. In terms of viscosity at 40°C, the differences are bigger.

Table 2

Comparison COF and wear scar diameter for AZUR CUT 602.01 M-10 and two similar commercial products

	AZUR CUT 602.01 M-10	SW [19]	MO-WNP [20]
COF	0.04…0.07	0.12-0.18	0.38
w [mm/N·m]	$4.47 \cdot 10^{-05}$	$0.29 \cdot 10^{-05}$	$0.24 \cdot 10^{-05}$
η (40°C)	10 cSt	66 cSt	1.35 cSt
Sliding distance [m]	32	1240	1650
Load [N]	392	392	396

4. Conclusions

Apart from cooling, the metalworking fluids have an important role in lubrication of the cutting tools. This reduces friction significantly, in the secondary sliding zone of the cutting tool-part contact. The experiments conducted on a four-ball machine helped to evaluate some tribological properties of mineral oil-based metalworking fluid. The research findings based on experimental investigations revealed a significantly reduction of friction coefficient (up to 0.05) when oil is used for low and mild conditions. For low loads the wear scar measurement showed that despite the change in loading during consecutive tests, the wear process reaches a steady state of constant wear rate. When load increased above 800 N wear scar diameter increased significantly. Some possible causes for this can be the complete disappearance of the fluid film and for extreme conditions the exceeding of contact stress limit of the ball material. For extreme pressure, the wear scar is very large with high volume of wear and, as a result, the load is distributed on a larger contact surface, reducing local pressure.

Comparison to other similar mineral oil-based fluids revealed good performances of the tested oil in reducing friction coefficient. Also, for the tested fluid, the wear rate is significantly higher. It was difficult to corelate this with viscosity and friction coefficient values of the tested oil.

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