# Effect of Temperature on Sliding Wear Mechanism under Lubrication Conditions

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**Abstract**

Experimental program using ball-on-cylinder tester has been conducted to investigate the effects of temperature, normal load, sliding speed and type of lubricating oil on sliding wear mechanism. The worn surfaces and debris have been examined. Surface examination of the tested samples using scanning electron microscope SEM was used to study the wear particles and the wear surfaces. The results show that the temperature of the oils affects the probability of adhesion, oxidation, wear rates, and friction coefficient. At room temperature (40$^{o}$C) and under lubrication conditions, friction and wear decreases with the increase of the running time. The increase in applied normal load tends to reduce the friction in all types of oils. The phosphorated oil SAE 90 was superior in minimizing friction and wear as compared with other oils. The results have shown that the lubricant temperature has a significant role in wear mechanism.

**Keywords:** Dry Sliding Wear, Surface Film, Friction Coefficient, Wear Particles, Oil Type.

## 1. INTRODUCTION

Wear, in modern lubricated mechanical systems, is characterized by ultra low wear rates. This statement applies to systems, like engines, as well as simple journal bearing. However, the majority of tribological publications deal with systems with wear rates of many micrometers per hour [1]. The regime of lubrication is defined as that in which the sliding surfaces are separated by lubricant films. The chemical and physical nature of the surfaces and the lubricant are of major importance [2]. The main function of the lubricant film is to reduce the amount of initial metallic contact between the sliding surfaces by interposing a layer, that is not easily penetrated, which posses relatively low shear strength [3]. At the asperity level, the mechanical interaction can be characterized as plastic flow accompanied by mechanical intermixing followed by quenching. At common sliding velocities in the range of the asperities lasts only a few microseconds; thus in case of plastic flow, the material becomes quenched frequently [3]. It has been found through experimental investigations that liquid fatty oil is much more effective as boundary lubrication mode than other organic molecules of comparable chain length, such as alcohols. It has been observed that the organic boundary lubricants are effective at temperatures up to about 200$^{o}$C. They can be made more effective at higher temperatures.

Temperature has a significant influence on the wear rate of lubricated rubbing pairs either with or without contaminants existing in the lubricant. All these contaminants will unavoidably affect the wear behavior. However, most of existing studies focus on the effects of contaminants at ambient temperature. It is essential and important to study the effects of lubricant temperature on wear process at elevated temperature.

This importance has been recognized by some researchers, and there are some studies on the temperature effects on boundary lubrication [4, 5]. The effect of oil types has been studied under lubrication boundary condition [6]. The results have shown that prior roughness of the sliding surfaces is not pronounced; friction and wear decreased with increased running time at room temperature. Under sever conditions of loading and lubrication, the type of oil film between surfaces plays an important role in reducing friction and wear, such as in the case of hydrodynamic lubrication [7]. This has been attributed to the exponential increase in viscosity of oil with pressure and the elastic deformation of contact surfaces, giving rise to the hydrodynamic effect, which is called elastohydrodynamic lubrication. However, there is a limited study on the effect of the mentioned parameters, especially in practical applications, such as steel industry, mining machinery, and some bearing or gears [8]. Based on the above reasons, the aim of this paper is to study the effect of lubricant temperature, oil type, and wear parameters on sliding wear process.

## 2. EXPERIMENTAL WORK

In this work, a ball-on-cylinder machine, shown in Figure 1, was used to conduct sliding wear tests under lubricating conditions. A cylindrical shape steel of outer diameter of 5.0cm, inner diameter of 2.5cm, height of 2.0cm, and surface roughness of 0.32 $\mu$m has been used. The cylinder was specially designed to be transversely loaded under rotating conditions. The chemical composition of the cylinder is given in Table 1. A steel ball of 5mm in diameter was used as a rider to slide on that cylinder. The chemical composition of the ball is given in Table 2.The testing machine used in this work can provide rotational speed ranging from 0-3000 rpm and applied load ranging from 0-500N. The friction coefficient and oil temperature were recorder on line by a system, connected to the tester. Four types of oils were used in this work. The oils specifications are given in Table 3. Three groups of tests were provided for four types of oils. The first group of tests was conducted at temperature of 40$^{o}$C, while the second and the third groups of tests were carried out after the oils have been heated and maintained at a temperature of 90$^{o}$C and 120$^{o}$C.

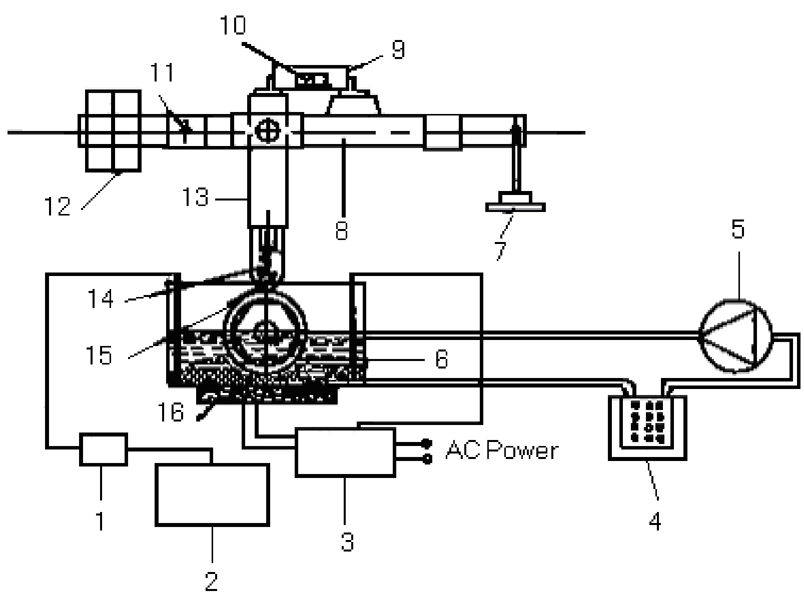



**FIGURE 1**: Schematic Diagram of the Continuous Wear Measurement Technique.
(1- Analog-to-digital converter; 2- Computer; 3- Temperature control unit; 4- Oil filter; 5- Pump; 6- Cylindrical specimen; 7- Loading weight; 8- Loading level; 9- Steel ring; 10-Strain gauge; 11- Fulcrum; 12- Balance weight; 13- Ball holder; 14- Steel spring; 15- Steel boll; 16- Heater)

| %C | %Mn | %Si | %Cr | HRC |
|---|---|---|---|---|
| 0.36-0.45 | 0.50-0.70 | 0.20-0.50 | 0.80-1.15 | 53 (heat treated) |

**TABLE 1**: Chemical Composition of Rotating Cylinder Sample.

| %C | %Mn | %Si | %Cr | HRC |
|---|---|---|---|---|
| 0.95-1.02 | 0.20-0.30 | 0.15-0.30 | 1.20-1.60 | 51 |

**TABLE 2**: Chemical Composition of Ball (Rider) Sample.

| Oil type | SAE grade | Specific gravity at 5-6$^{o}$C | Flash point, $^{o}$C | Viscosity at | |
|---|---|---|---|---|---|
| | | | | 40$^{o}$C | 100$^{o}$C |
| **Straight mineral** | 40 | 0.904 | 259 | 130 | 10.0 |
| **Phosphated** | 90 | 0.920 | 204 | 150 | 15.0 |
| **Supper cutting (Sulphrated)** | 30 | 0.888 | 260 | 102 | 6.6 |
| **Water soluble** | 50 | 0.930 | 220 | 98 | 4.8 |

**TABLE 3:** Specifications of Lubricating Oils.

## 3. RESULTS AND DISCUSSIONS

Analysis was conducted on the basis of obtained friction coefficients, wear particles generated from the sliding tests, and the surfaces of the tested samples. The examination of the friction coefficients and their changes provides information to assess the wear rates of the tests. The investigation of the characteristics of wear particles and wear surfaces of the tested samples offers information for assessing the wear mechanism and wear rates. The measured friction coefficients under different wear conditions are listed in Table 4. Several investigators have reported the effects of lubrication conditions on the wear mechanism. According to their experimental results [5, 8, 9], it has been confirmed that the increases in the oil temperature and its quality significantly affect the wear process. In this research, the influences of the oil temperature, sliding speed, sliding time, and applied load on friction coefficient and wear rate will be discussed.

### 3.1 Effect of oil Temperature on Friction Coefficient

According to the results shown in Table 4, it is clear that there is a significant effect of oil temperature on the average friction coefficient at constant load and constant sliding speed in all types of oils. Also, it is clear that the SAE90 oil is superior in minimizing friction as compared with other oils under the same conditions, which are in agreement with other tribological results [10, 11]. The results in Table 4 indicate that boundary lubrication conduction occurred, which means that a normal sliding wear process exists. Also, the results show that as the oil's temperature increases, the friction coefficient decreases. This is due to the change in the oil properties, such as viscosity and the chains for oxidation, and formation of oxide film at the rubbing surfaces. This claim is in accordance with assertions in [12]. In all tests the thickness of oil film is smaller than the surface roughness of the rubbing pair surfaces, and a boundary lubrication condition predominant [13].

### 3.2 Effect of Sliding Speed on Friction Coefficient

The results listed in Table 4 show that at constant load and constant temperature, the average friction coefficient increases as the sliding speed increased. This means that at high sliding speed, the temperature of rubbing pair surfaces rises and the thickness of the lubricating oil film is not enough to make a good separator between the asperities of coating surfaces and to lower their temperatures [14].

### 3.3 Effect of Sliding Time on Friction Coefficient

During the running-in, at 40°C temperature and constant sliding speed and constant applied load, the friction coefficient gradually decreases as the sliding time increases in all types of oils, as shown in Figures (2-4). This means that more stable oil film exists at contact surfaces during the running-in, and the thickness of this film is sufficient to separate the contact surfaces and reduce the wear rate and friction coefficient. In this case, the wear mechanism is controlled by the type and stability of the oil film.

### 3.4 Effect of Surface Roughness of Rubbing Surfaces on Friction Coefficient

Surface average roughness $R_a$ was used to describe the surface roughness of the worn surfaces. Table 5 shows the results of surface roughness $R_a$. It is clear that the value of $R_a$ at constant load and constant sliding speed increases with the increase in the oil temperature. This is due to the change in the oil properties when the temperature rises [15].

| Group | Test No. | Oil type | Oil temperature, °C | Sliding speed, m/min | Load, N | Friction coefficient, $\mu$ |
|---|---|---|---|---|---|---|
| A | 1 | Soluble | 40 | 785 | 30 | 0.140 |
| | 2 | Sulphrated | 40 | 785 | 30 | 0.085 |
| | 3 | Straight mineral | 40 | 785 | 30 | 0.067 |
| | 4 | Phosphated | 40 | 785 | 30 | 0.057 |
| B | 1 | Soluble | 90 | 785 | 30 | 0.158 |
| | 2 | Sulphrated | 90 | 785 | 30 | 0.104 |
| | 3 | Straight mineral | 90 | 785 | 30 | 0.072 |
| | 4 | Phosphated | 90 | 785 | 30 | 0.063 |
| C | 1 | Soluble | 120 | 785 | 30 | 0.186 |
| | 2 | Sulphrated | 120 | 785 | 30 | 0.121 |
| | 3 | Straight mineral | 120 | 785 | 30 | 0.090 |
| | 4 | Phosphated | 120 | 785 | 30 | 0.073 |
| D | 1 | Soluble | 40 | 1370 | 30 | 0.192 |
| | 2 | Sulphrated | 40 | 1370 | 30 | 0.162 |
| | 3 | Straight mineral | 40 | 1370 | 30 | 0.121 |
| | 4 | Phosphated | 40 | 1370 | 30 | 0.084 |
| E | 1 | Soluble | 40 | 2355 | 30 | 0.213 |
| | 2 | Sulphrated | 40 | 2355 | 30 | 0.175 |
| | 3 | Straight mineral | 40 | 2355 | 30 | 0.156 |
| | 4 | Phosphated | 40 | 2355 | 30 | 0.117 |

**TABLE 4:** Measured Average Friction Coefficients.

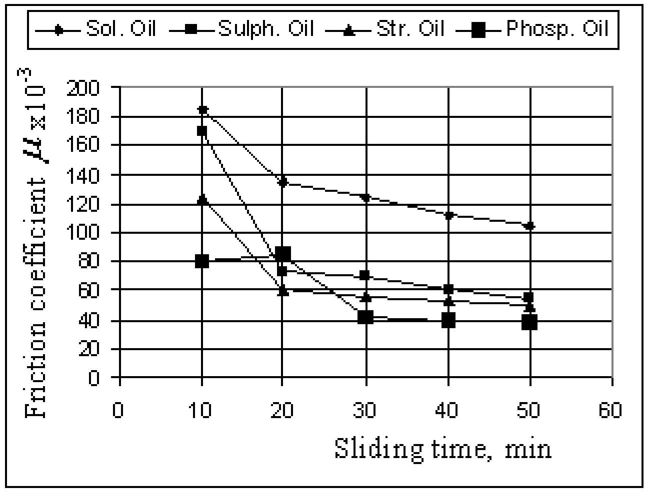



**FIGURE 2:** Friction Coefficient $\mu$ Versus Sliding Time Under Lubricated Sliding. (Oil Temperature = 40°C, Load = 30N, Sliding Speed = 397 m/min)

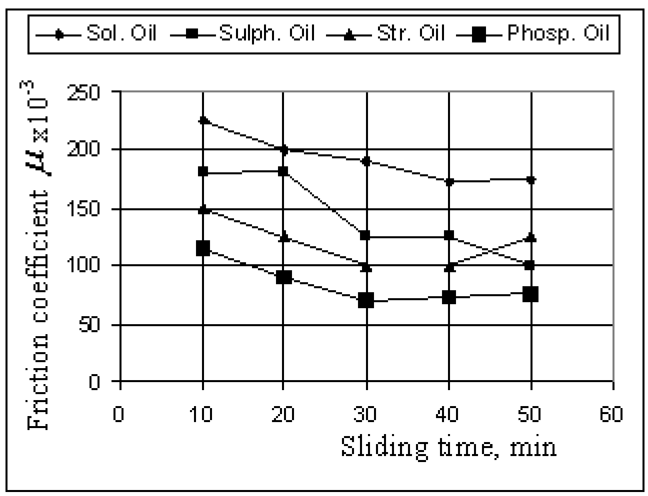



**FIGURE 3:** Friction Coefficient $\mu$ Versus Sliding Time Under Lubricated Sliding. (Oil Temperature = 90°C, Load = 30N, Sliding Speed = 397 m/min)

### 3.5 Effect of Applied Load on Friction Coefficient

Figure 5 illustrates the effect of applied load on friction coefficient. It is clear that the friction coefficient decreases as the applied load increases, and the value of friction coefficient is higher during the initial transitory sever wear stage at low applied loads. The increase in applied load rises the temperature of the contact area of the rubbing surfaces, which leads to an increase in oil's temperature and gives more chances to the formation of oxide film, which reduces friction coefficient as the wear rate reduces [16].

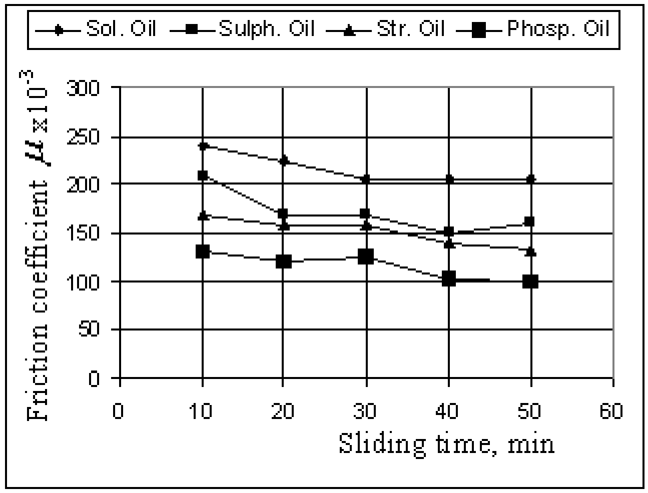



**FIGURE 4:** Friction Coefficient $\mu$ Versus Sliding Time Under Lubricated Sliding. (Oil temperature = 120$^{o}$C, Load = 30N, Sliding Speed = 397 m/min)

| Group | Test No. | Surface roughness $R_a$, $\mu$m | | Oil's Temperature, $^{o}$C |
|---|---|---|---|---|
| | | Cylinder | Ball | |
| A | 1 | 0.46 | 0.32 | 40 |
| | 2 | 0.34 | 0.27 | 40 |
| | 3 | 0.30 | 0.22 | 40 |
| | 4 | 0.28 | 0.18 | 40 |
| B | 1 | 0.48 | 0.35 | 90 |
| | 2 | 0.39 | 0.29 | 90 |
| | 3 | 0.35 | 0.24 | 90 |
| | 4 | 0.31 | 0.21 | 90 |
| C | 1 | 0.51 | 0.37 | 120 |
| | 2 | 0.40 | 0.34 | 120 |
| | 3 | 0.38 | 0.27 | 120 |
| | 4 | 0.33 | 0.25 | 120 |

**TABLE 5:** Surface Roughness $R_a$ of the Worn Samples. (Sliding Speed = 785m/min = Constant and Applied Load = 30N = Constant)

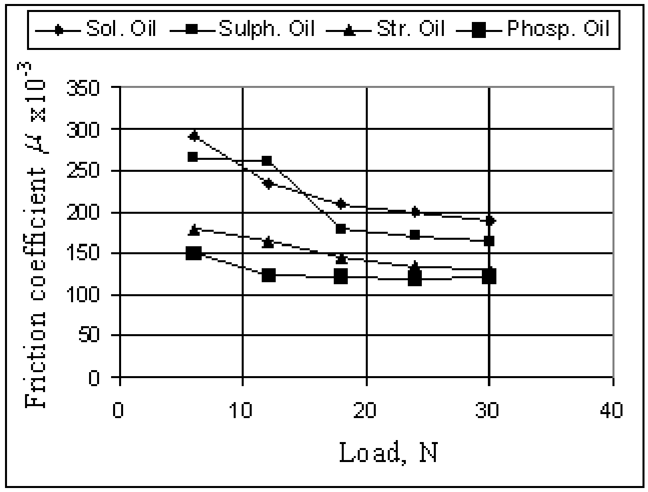



**FIGURE 5:** Friction Coefficient $\mu$ Versus Applied Load. (Oil Temperature = 40$^o$C, Sliding Speed = 788 m/min)

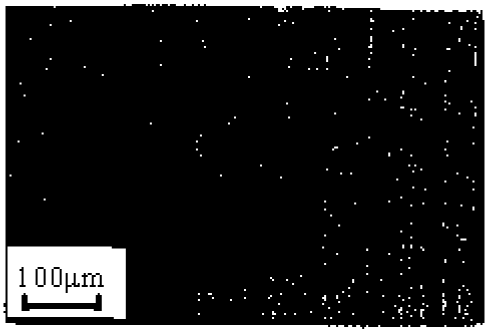
### 3.6 Effect of Surface Damages on Friction Coefficient

The surface topography of the ball and cylinder samples were examined at the end of each test using scanning electronic microscope SEM, as shown in Figure 6. The images can lead to more information about the wear mechanism that took place in this research. A commonly observed feature of the tracks may be termed as a "peeling" effect. This effect has been shown in Figure 6-a, group G, tests 1-4. In Figure 6-b, the surfaces appear as if the layers of materials have been removed progressively with the formation of a number of pitting and scoring in the wear scar. This can be explained in terms of the wear mechanism, which includes adhesion and oxidation according to the types of lubricating oils. Wear particles in the collected oil samples were analyzed for the tests at 120$^o$ C temperature for group C, table 4. The results show two major types of wear particles: long flat and oxidation particles. It is noticed that the amount of collected wear particles depended on the test conditions. The increase in surface deformation, coupled with fragmentation at high load levels, accounts $\leq 7$ for the large increase in the amount of wear particles collected in the oil samples. From Figure 6-a, it can be seen that there are only normal sliding scratches on the wear scar of the ball sample. Generally, it is smooth in the contact area of cylinder sample, as shown in Figure 6-b.

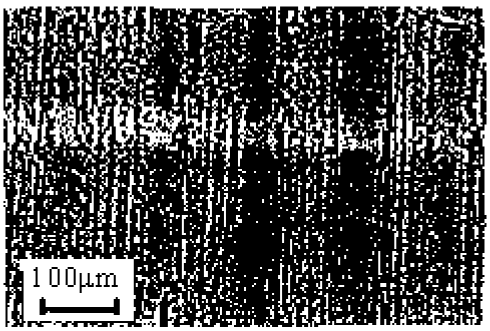
## 4. CONSLUSION

Based on the results of this study, the following conclusions can be made:

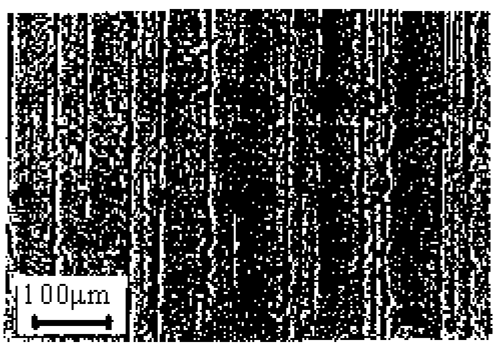
1. The temperature of the lubricating oils has significant influence on the wear mechanism.
2. In all tests, the phosphated oil is superior in minimizing friction coefficient and wear rate due to the formation of more stable oil film.
3. At constant sliding speed and load, the increase of sliding time has the same effect on the friction coefficient as the increase of applied load.
4. The decrease in the friction coefficient and wear rate with increasing in sliding time and applied load can be attributed to the improvement in hydrodynamic action.
5. Elastic deformation of contact surfaces under loading conditions gives rise to the elastohydrodynamic lubrication.
6. The amount of wear particles increases with increase in the oils temperature and the applied load.
7. As oil temperature increases, wear particles formed. These wear particles act as abrasives, which lead to increase in friction coefficient and wear rates.



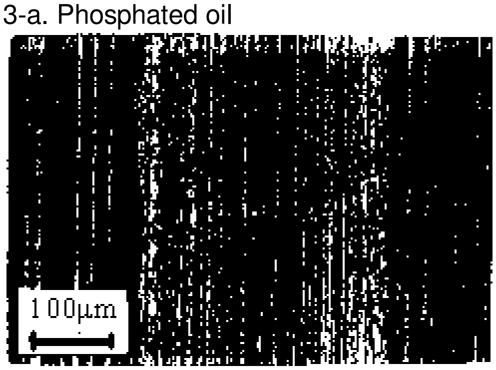

1-a. Water soluble oil




2-a. Sulphrated oil

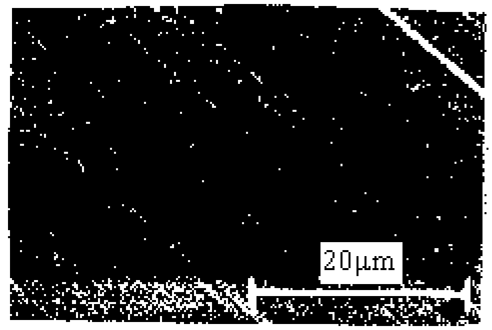



3-a. Phosphated oil

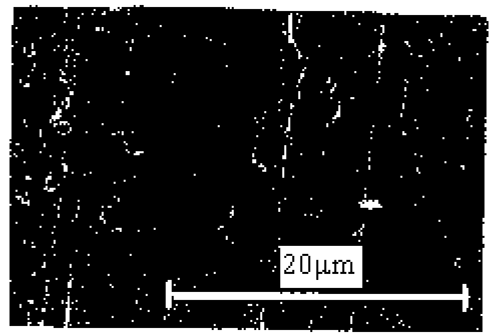



4-a. Straight mineral oil

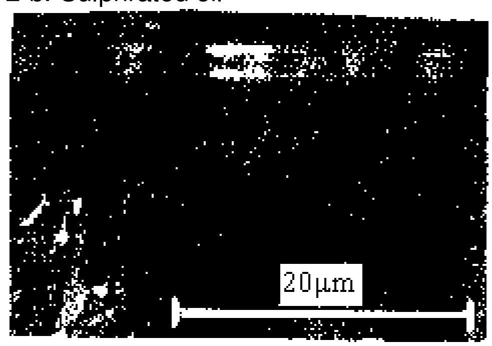
**a. Ball samples.**



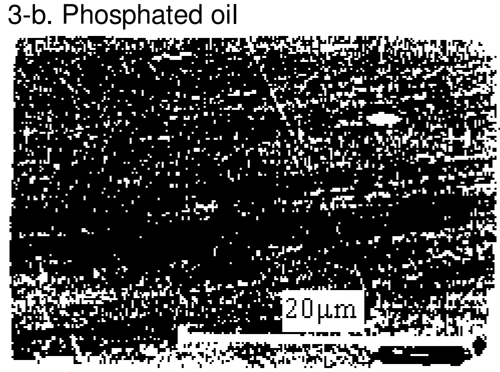

1-b. Water soluble oil




2-b. Sulphrated oil




3-b. Phosphated oil




4-b. Straight mineral oil

**b. Cylinder samples.**

**FIGURE 6:** The SEM Surface Topography of the Tested Samples in Test Group G.

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