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Tribological evaluation of refined, bleached and deodorized palm stearin using four-ball tribotester with different normal loads*

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Abstract: The effects of the mechanical factors with applied loads on the tribological performance of refined, bleached and deodorised (RBD) palm stearin (PS) were studied using a four-ball tribotester. All the RBD PS results were simultaneously compared with the additive-free paraffinic mineral oil (PMO). The experiments were carried out using different loads with a constant speed in order to gain a better understanding of the mechanical processes that occurred during the experiment. For each experiment, the temperature was increased to 75 °C and was run for 1 h. In a mechanical system, lubricant plays an important role in reducing wear and friction. PS exists as a semi-solid at room temperature after the fractionation process from oil palm. Due to the increasing rate of pollution to the environment, vegetable oil was chosen as the test lubricant with regard to its biodegradability. Other advantages of vegetable oil are that it is more easily harvestable and non-toxic compared to petroleum-based oil, which made it a suitable candidate. From the experiment, RBD PS is found to have a better friction constraint reduction compared with additive-free PMO.

Key words: Palm stearin (PS), Four-ball tribotester, Paraffinic mineral oil (PMO), Friction coefficient
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1 Introduction

Lubricant plays a major role in tribological performance, as it is used to reduce wear and friction in mechanical systems. It is not only developed for lubricating surfaces, but also for other roles in industrial applications. Lubricants can be categorised into two types: engine lubricant and non-engine lubricant. Thus, to better understand the characteristics of the effects of friction and wear on two moving surfaces, the researchers need to investigate or study the reaction and the fluid present between the moving surfaces. Under boundary lubrication conditions, a sufficient protective lubricant film on the rubbing surfaces plays a main role in the construction of lubricant film layer and controlling the wear behaviour of the test system. Perez *et al*. (2005) found that parameters such as sliding speeds and loads applied can affect tribological characteristics. At the same time, the lubricant film development on the two moving surfaces needs to be studied in order to develop the lubricant process. As a result, many researchers have tried to develop new and better lubricants that meet the demands of current machinery. According to Michael *et al*. (2006), developing

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lubricants that could be used in engineering systems without replenishment are very important for increasing the functional lifetime of mechanical components.

Nowadays, global issues and environmental concerns result in a growing interest in the use of biodegradable products. As reported by Emil and Zahira (2009), this is because of decreasing petroleum oil resources and the increasing rate of pollution caused by petroleum-based products. It is estimated that 12 million tons of lubricant waste is released into the environment every year. This increases awareness of the need to protect the environment from steadily worsening pollution caused by effluents like hydraulic fluid, which are based on mineral oils.

Consequently, environmentally friendly parameters have become one of the most important factors in selecting lubricants and hydraulic fluid in industry. Jayadas and Prabhakaran (2007) proposed that vegetable oil is one of the biodegradable oils that have been promoted to replace mineral-based oils, because it has lubricant characteristics and performance equivalent to those of mineral-based oils. In addition, vegetable oils are often chosen over other biodegradable oils because they are renewable or harvestable resources. According to Bartz (1998), they are successful in many aspects and are getting widespread use in some applications. Not only that, the fatty acids in vegetable compositions have the additional advantage of better boundary lubricant properties, which results in lower friction compared with mineral-based oils. In addition, Randles and Wright (1991) found that vegetable oil is also able to adhere to metal surfaces due to the existence of its polar ester group, and it therefore shows a good lubricating ability.

Furthermore, vegetable oil was chosen to replace mineral-based oil because it is non-toxic, environmentally friendly and can be easily disposed of into the environment. Carcel and Palomares (2004) found that the salutary properties of vegetable oil, such as high viscosity index, high flash point, and low evaporative loss make it suitable for wide use as a lubricant in industry. According to Nosonovsky (2000), vegetable oil being used as a lubricant is not a new idea as it had been used during the construction of monuments in ancient Egypt. In recent times, vegetable oil has been widely used in many applications. Bari *et al*. (2002) studied the potential of palm oil for use in diesel engines, while Obi and Oyilola (1996) confirmed the advantages of using palm oil as hydraulic oil.

In this experiment, the additive-free refined, bleached and deodorised (RBD) palm stearin (PS) and additive-free paraffinic mineral oil (PMO) were used as lubricants to evaluate their lubricity characteristics on the various effects of tribological parameters. RBD PS is the refined palm oil that exists in a semi-solid state at room temperature. An additive-free PMO was used to simultaneously compare the lubricity performance with RBD PS. A four-ball tribotester was used in this experiment to evaluate the performances of RBD PS and PMO. The aims of this study were to investigate the effects of normal loads applied on the tribological performance of lubricant.

2 Experimental

2.1 Experimental apparatus

This experiment used a four-ball tribotester which is an instrument for friction and wear measurement as described by Boerlage (1933). The important components of the four-ball tribotester are the ball pot assembly, collet, locknut adaptor, and standard steel balls. This instrument uses four balls; three at the bottom and one on the top. The upper ball is held in a collet at the lower end of a vertical spindle that is driven by the motor. The bottom three balls are held firmly in a ball pot containing the test lubricant and pressed against the upper rotating ball. The entire components' surfaces needed to be cleaned with acetone before the tests. Fig. 1 shows the schematic diagram of a four-ball tribotester.

2.2 Materials

The specifications of the test ball used in this experiment are chrome alloy steel made from AISI E-52100, with a diameter of 12.7 mm, extra polish (EP) grade of 25 and hardness of 64–66 HRC. Each time, before starting a new test, four new balls would be used, and then all surfaces were cleaned with acetone and wiped using a fresh, lint-free industrial wipe.

2.3 Lubricants

The lubricants were RBD PS and PMO. The additive-free PMO was used to compare with the results of RBD PS. The lubricant volume used in this experiment was 10 ml. The density and viscosity (Test method: ASTM D1298-85(90)) of both lubricants are shown in Table 1.

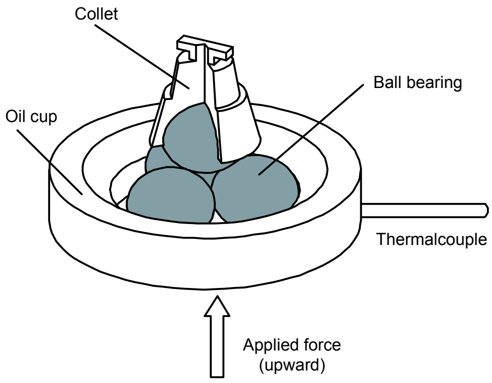

Fig. 1 Schematic sketch of a four-ball tribotester

Table 1 Properties of test lubricants

Lubricant	Relative density	Viscosity (mm^2/s)	
		40 °C	100 °C
RBD PS	0.8725	98	17
PMO	0.8715	82	16

2.4 Friction evaluation

A 20 kg beam type load cell was used to measure the frictional torque. The load cell was fitted at a distance of 80 mm from the center of the spindle. Applied force is measured as the frictional force and converted to frictional torque by multiplying the frictional force by 0.8 m. The coefficient of friction results is based on the average measurement of frictional force measured. The coefficient of friction plays a major role in the determination of transmission efficiency via moving components. Less resistance contributes to higher efficiency. Therefore, in terms of lubricant, less friction is more desirable.

2.5 Wear test

The test run was carried out using a load of 30 kg up to 60 kg with gradual increment of 10 kg at a constant sliding speed of 1200 r/m. All the tests were conducted for 1 h at an operating temperature of 75 °C. The wear scar diameter on the surface of the three ball bearings was measured using the charge-coupled device (CCD) microscope.

2.6 Surface profile

In this experiment, a surface profile tracer was used to evaluate the worn surface roughness on the ball bearing. The objective of this analysis was to determine the effects of different normal loads on the surface roughness after the tests for RBD PS and PMO. R_a is the arithmetic mean of the absolute values of the profile deviation from the mean line.

2.7 Test procedures

The purpose of an experiment is to measure or study the variable parameters of interest. The variable parameters are tested at a number of chosen levels with all combinations. This experiment evaluated normal loads, which are expected to influence the friction and wear characteristics of lubricant. The bottom three stationary balls in the wear test were evaluated from the average diameter of the circular scar formed. In addition, the lubricating ability of RBD PS was also evaluated, based on the friction torque produced compared with PMO.

All parts in the four-ball (upper and lower balls and the oil cup) were thoroughly cleaned using acetone and wiped using a fresh lint-free industrial tissue. No trace of solvent should have remained once the lubricant was applied and the parts were assembled.

The steel ball bearings were placed into the ball pot assembly and were tightened using a torque wrench. This was intended to prevent motion of the bottom steel balls during the experiment. The upper spinning ball was locked inside the collet and tightened into the spindle. The test lubricant was applied into the ball pot assembly. Apart from that, we ensured that oil levels filled all voids in the test cup assembly. The oil cup and the ball bearings were fitted in specific holder, and mounted on the non-friction disc in the four-ball machine and shock loading was avoided by slowly applying the test load. After that, the selected lubricant was heated up to the desired temperature. When the set temperature was reached, we started the drive motor, which had been set to drive the top ball at the desired speed.

After a duration of 1 h, the heater was turned off and the oil cup assembly was removed from the machine. The test oil in the oil cup was then drained off and the wear scar area was wiped using tissue. The wear scar on the bottom balls was measured using a special microscope base that had been designed for the purpose.

3 Results and discussion

3.1 Friction coefficient

Friction tests on RBD PS and additive-free PMO were carried out using a four-ball tribotester. The friction torques, calculated from the friction measured from the load cell in the four-ball tribotester, are shown in Fig. 2 as a function of time for RBD PS and PMO, respectively.

In Fig. 2, both test lubricants show a similar pattern of friction torque increment when the normal load increases. For all RBD PS experimental conditions (Fig. 2a), the friction torque was increased rapidly at the beginning of the experiments and achieved a steady-state condition towards the end of the experiment. For the PMO (Fig. 2b), only low normal loads of 30 and 40 kg, showed a steady-state condition until the end of the experiment. The steady-state condition of the friction torque shows that the lubricant layer between ball-bearings was stable and no severe breakdown of lubricant film occurred.

However, for high normal loads, especially for 60 kg, there was an increment of friction torque throughout the process, due to the lubricant film breakdown. This resulted in an increase in the metal-to-metal contact, while creating high friction throughout the motion.

From the fiction torque value measured from the experiments, the friction coefficients for all experimental conditions were calculated using Eq. (1), and compared mutually as shown in Fig. 3.

$$\mu = 0.22248 \times \frac{T}{W}, \tag{1}$$

where T is the friction torque, kg·mm, W is the normal load, kg, and μ is the friction coefficient.

In Fig. 2 and Fig. 3, the results show that the fatty acid chains in the vegetable oil permitted monolayer film formation with a slippery surface, which prevented direct metal-to-metal contact, as described by Sharma *et al*. (2008). Even though the friction increased with the increase of normal loads in this experiment, RBD PS showed lower friction resistance ability than PMO.

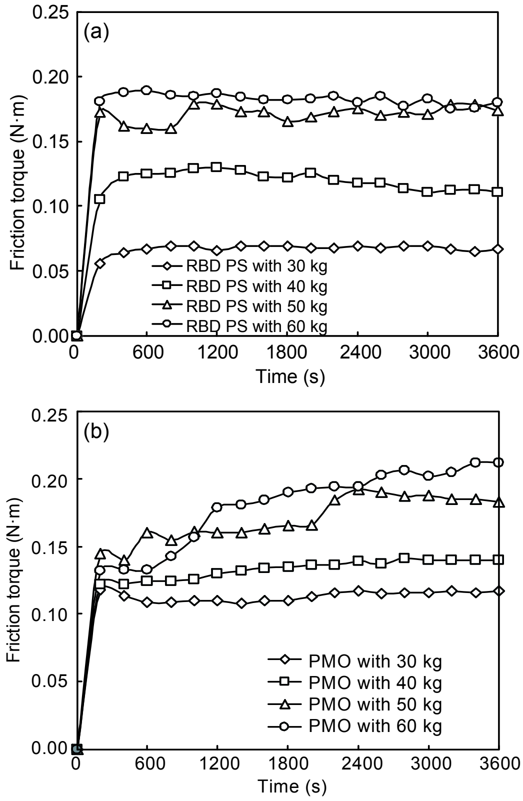

Fig. 2 Friction torque distribution vs. time for various normal load conditions for RBD PS (a) and PMO (b)

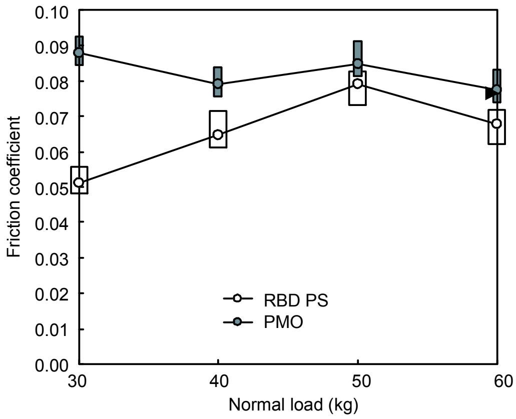

Fig. 3 Friction coefficient distribution

3.2 Wear scar diameter (WSD)

After the experiment, the wear scar diameter (WSD) on the three ball-bearings, which were fitted into the oil cup, was observed and measured using a CCD microscope. The average values of the wear scar diameter were taken and plotted as shown in Fig. 4.

At the normal load of 30 kg, RBD PS gave the smallest value of WSD (0.672 mm) compared with the value given by the PMO (0.694 mm) under the same experimental conditions. Nevertheless, for the remaining experimental conditions, the PMO showed a smaller value of WSD compared with RBD PS due to the chemical attack on the surface by the fatty acid from RBD PS. The metallic soap film was rubbed away during the sliding, leading to the production of the non-reactive detergents that increased the wear, as explained by Bowden and Tabor (2001).

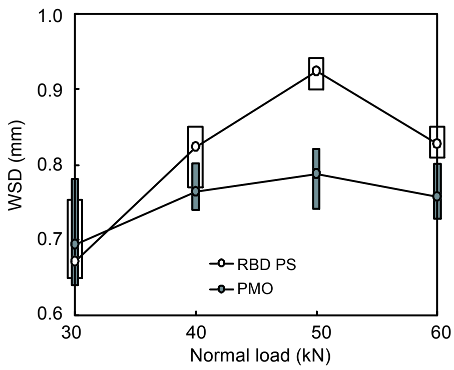

Fig. 4 Wear scar diameter (WSD) distribution

3.3 Flash temperature parameter (FTP)

Flash temperature parameter (FTP) was introduced by Lane (1957) to estimate the possibility of lubricant breakdown. The FTP was calculated using Eq. (2), where d is the mean WSD in mm at the particular normal load. The higher value of FTP means that the lubricant film is less likely to break down.

$$\mathrm{FTP} = \frac{W}{d^{1.4}}. \tag{2}$$

From Fig. 5, both lubricants show the increment of FTP value when the normal load increased. At the normal load of 30 kg, the PMO had an almost similar FTP value compared with RBD PS. However, in other conditions, the FTP values of PMO were higher compared to those lubricated with RBD PS.

3.4 Wear scar observation

Fig. 6 shows the CCD micrographs of the wear scar of the ball specimen for all experimental conditions. For experimental conditions with 30 kg and 40 kg normal loads, the worn surface on the

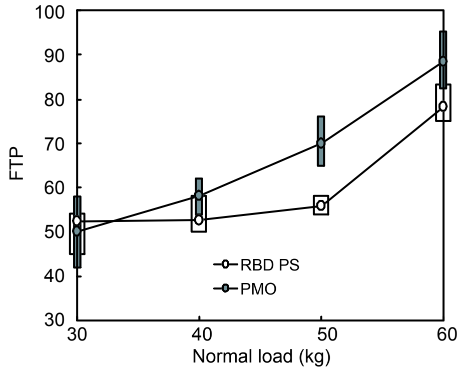

Fig. 5 Flash temperature parameter (FTP) distribution

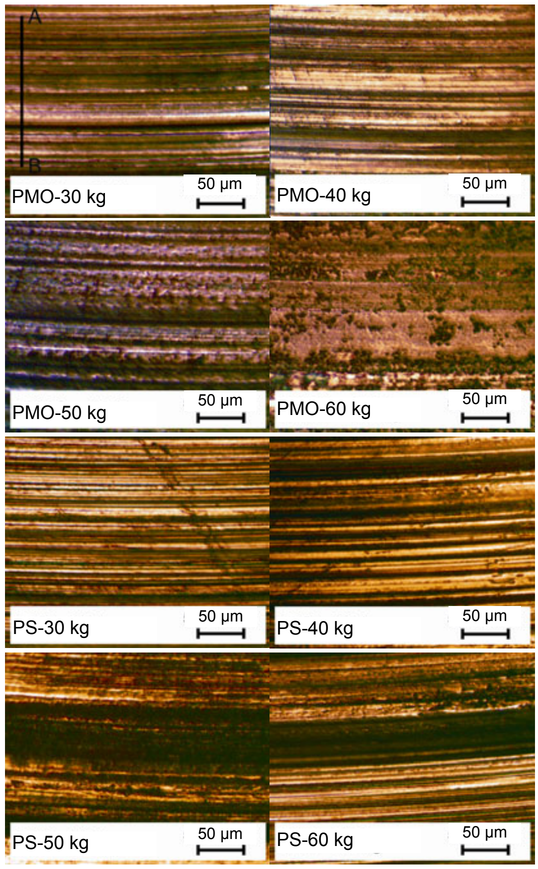

Fig. 6 CCD micrographs of the wear scar on the ball specimen

ball-bearing lubricated with both RBD PS and PMO showed almost similar wear patterns with parallel grooves. Some of the grooves were deep while others were shallow. This finding shows that the dominant wear mechanism was abrasive wear (Himmat and Gulati, 1991). The grooves resulted from stiff particles, such as wear debris of the oxide layer, or ragged adhesion. The particles contaminated the lubricant and damaged the ball-bearing surface (Ge *et al*., 2008; Groche *et al*., 2008).

In the case of the ball-bearing lubricated with PMO (with a 50 kg and 60 kg normal load), according to Masjuki and Maleque (1997), the worn surface shows micro-cutting edges which result from the penetration of hard particles onto the brittle solid surface. At a normal load of 60 kg, it is clearly demonstrated that the worn surfaces are similarly rough with large areas of white cavities (PMO-60 kg) (Ren *et al*., 2012).

In contrast, the worn surface of the ball-bearing lubricated with RBD PS (at a normal load of 60 kg) looked smoother and had less material transferred compared with PMO at the same normal load (Masuko *et al*., 2006). However, certain grooves were deep (PS-60 kg).

The scanning electron microscope (SEM) morphologies of the worn surface for all experimental conditions are shown in Fig. 7. Mild material transfers occurred for both RBD PS and PMO at the normal loads of 30 kg, 40 kg, and 50 kg, which suggests that the abrasive wear mechanism was dominant (Kim *et al*., 2011).

At a normal load of 60 kg, both RBD PS and PMO show plastic deformation on the worn surface. As reported by Ren *et al*. (2010), the plastic flow on the worn surface led to adhesive wear of the mechanism and left some white cavities on the worn surface. This phenomenon also shows that the lubricant film had thinned out and that the possibility of lubricant breakdown was higher. The FTP value shows the higher point for a normal load of 60 kg (Fig. 5).

A surface roughness profilometer was used to measure the surface roughness (R_a) and surface profile on the worn surface for all experimental conditions. The measurement direction is perpendicular to the rotation direction of the ball-bearing (line *AB*), as shown in Fig. 6a. A mutual comparison of the surface roughness was plotted as shown in Fig. 8.

From Fig. 8, with the increment of the normal load, the surface roughness of the worn surface of the ball-bearing lubricated with RBD PS was increased. It contradicts the results from PMO experimental conditions in which the surface roughness of the worn surface did not show any apparent changes.

The rough surface of the worn surface of the ball-bearing lubricated with RBD PS had many deep

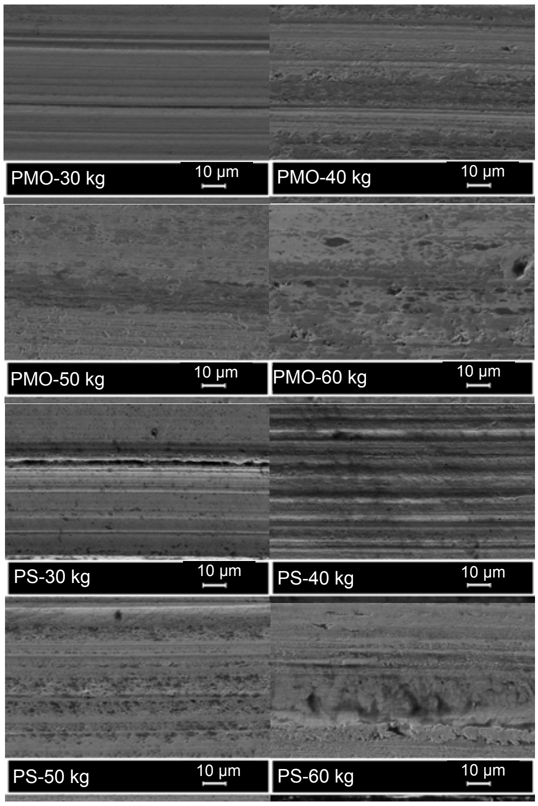

Fig. 7 SEM morphologies of the worn surface

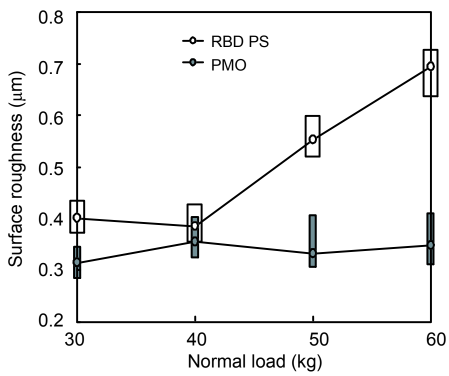

Fig. 8 Surface roughness distribution

asperities that allowed the lubricant to be trapped within the asperities and to act as a lubricant reservoir. As described by Syahrullail *et al*. (2011), the trapped lubricant could supply lubricant throughout the process, and reduce any metal-to-metal contact. As a result, the friction coefficient decreased for RBD PS. However, for the ball-bearing lubricated with PMO, the surface roughness of the worn surface was almost similar for all experimental conditions. The worn surface formed on the ball-bearing had shallow asperities and the trapped lubricant was not able to maintain the lubricant layer throughout the process. For the RBD PS, the existence of fatty acids in the vegetable oil effectively increased the presence of the RBD PS lubricant film on the ball-bearing surface and the results showed a lower coefficient of friction and FTP value compared with PMO.

5 Conclusions

The tribological evaluation of RBD PS was conducted using a four-ball tribotester. The rotational speed was set at 1200 r/m and the testing time was 1 h. Normal loads were varied as 30, 40, 50, and 60 kg. The results showed that the RBD PS had a lower coefficient of friction compared to the additive-free PMO, due to the fatty acid molecules in the RBD PS. This result can be seen clearly at low normal loads (30 kg and 40 kg). However, the RBD PS had a tendency to create larger wear scar due to the production of non-reactive detergent. From the observation on the surface topography of the worn surface, the rough surface (deep valley of asperities) that formed helped to create an oil reservoir of the RBD PS, and prevented metal-to-metal contact.

6 Future works

The next stage of the investigation is to test RBD PS with an extended test period. A suitable period is probably 4 h or more because one of the main disadvantages of vegetable oil is its poor oxidation performance. In these conditions, different types of wear mechanisms would occur. More accurate worn surface analysis could be observed by using a transmission electron microscope (TEM) and X-ray photoelectron spectroscope (XPS). At the same time, oil analysis could be done using spectrographic analysis for particle shape classification, wear, and any contaminant metal in the test lubricant. Thus, the investigation of suitable additives for RBD PS could be initiated to produce a palm oil-based lubricant.

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