Vol. 42, No. 2 (2020) 278-287, DOI: 10.24874/ti.843.02.20.05

Tribology in Industry

www.tribology.rs

RESEARCH

The Tribological Performance of Hydrodynamic Journal Bearing Using Bio-based Lubricant

P. Zulhanafi [a,*], S. Syahrullail [a], M.A. Ahmad [b]

[a]*School of Mechanical Engineering, Faculty of Engineering, Universiti Teknologi Malaysia, 81310, UTM Skudai, Johor Malaysia,*
[b]*Faculty of Mechanical Engineering, Universiti Teknologi MARA, 40450, Shah Alam, Selangor, Malaysia.*

Keywords:

Journal bearing
Bio-based lubricant
Pressure profile
Temperature profile
Friction coefficient

* Corresponding author:

Zulhanafi Paiman
E-mail: zulhanafipaiman@gmail.com

Received: 1 February 2020
Revised: 24 April 2020
Accepted: 7 May 2020

A B S T R A C T

Green technology policies and other environmental legislations have driven researchers to channel their attention into bio-based lubricants. In the current study, the performance of the bio-based lubricant, palm mid olein (PMO), in journal bearing applications was examined. A comprehensive journal bearing test rig was used to evaluate the tribological behaviour of PMO and to compare it with a mineral-based oil (SAE 40). A bearing, with a length to diameter ratio of 0.5, was used in accordance with variations in the journal speed and radial load. It was found that PMO presented a higher maximum pressure and better thermal resistivity compared to SAE 40. PMO also demonstrated a lower friction coefficient in all testing conditions.

1. INTRODUCTION

As machinery technology keeps expanding all over the world, it is crucial to have highly efficient machining operations to achieve the optimum desired output. A journal bearing is one of the machine elements that have a high impact on the optimization of a machine itself. A journal bearing consists of two major parts known as the shaft or journal and the bearing. The shaft is fundamentally free to move inside the bearing, and it transmits the power to the other mechanisms in the system for various purposes. A journal bearing is commonly operated in a hydrodynamic lubrication regime. This is a condition where the load-carrying surface of the bearing is separated by a thick lubricant film to avoid metal-to-metal contact, thus, reducing the friction [1-3]. Generally, the pressure profile distribution of the fluid film is one of the main criteria for evaluating the performance of the bearing.

A lubricant is an important element for facilitating the motion between two surfaces either by rolling, sliding, shearing or rotating. A good lubricant will lead to less friction, minimum power loss, prolonged shelf life of the material, and reduced wear. It might come either in a solid or liquid form with different properties. Few justifications need to be considered when selecting a lubricant. Different working conditions will require different properties of lubricants. Usually, the viscosity, thermal resistivity and cold flow properties of the lubricant are some of the main criteria to be considered in selecting a lubricant. In the present study, the performance of two different based lubricants with a length to diameter bearing ratio of 0.5 were evaluated in a journal bearing application.

1.1 Journal bearing with bio-based lubricant

The study of bio-based lubricants has been expanding all over the world, the main reason being that they offer high adaptability to the environment due to their bio-degradable behaviour. This might eliminate one of the limitations that govern mineral oils, namely, that it is nearly impossible to dispose them from the environment [4]. Bartz reported that most of the mineral oil-based lubricants that are consumed end up in nature rather than being properly disposed [5]. These have driven researchers to channel their attention to bio-based lubricants. Experimental studies on the use of palm oil-based lubricants in journal bearing applications are very limited. However, relevant studies in similar applications have been widely investigated using different testing methods [6-10]. Elisabet et al. investigated the capability of rapeseed–synthetic ester oil in mixed lubrication conditions. They found that the lubricant significantly demonstrated lower values of wear and frictional torque compared to mineral oil [11]. Another study was conducted by S. Basker to evaluate the performance of soya bean oil under hydrodynamic conditions. It was revealed that soya bean oil has a very low viscosity leading to higher heat generation, hence suggesting that it is not fit for operations in hydrodynamic conditions [12].

1.2 Pressure and temperature distribution

The rotation of the journal inside the bushing results in the development of pressure along the circumference of the bearing. With sufficient journal speed, the lubricant will be forcefully dragged into a wedge–shaped zone, and enough pressure can be created to separate the journal from the surface of the bearing [12]. The pressure profile is affected by many factors, including the speed of the journal, the load applied on the bearing, the viscosity of the lubricant, the position of the oil groove, the design of the oil groove and many more. Many studies have been conducted to investigate and predict these pressure profile behaviours. Nuruzzaman et al. conducted a study on the pressure distribution using finite element and analytical methods. The results on the comparison of both methods were satisfactory [13]. Another research was conducted by Yathish et al. to investigate the effect of using a two-axial groove in journal bearings. It was revealed that the experimental value of the pressure reading was 20 % lower than the theoretical value [14]. An extensive study using a two-axial groove was carried out by Phouc et al. under severe operating conditions. It was found that the maximum pressure occurred at an approximate angle of 105° and decreased at a location of 150° to 180° in diverging sections [15]. Mohamad Ali Ahmad et al. investigated the effect of the oil feed pressure on the pressure distribution. It was concluded that variations in oil feed pressure tend to affect the pressure distribution along the circumference of the bearing [1]. The same authors also studied the impact of varying the oil supply position on the pressure distribution [16]. A similar study was carried out and described by Wang and Khonsari using analytical methods in a static performance and non-linear instability analysis [17,18].

Various studies have been conducted to evaluate the distribution of temperature along the circumference of a journal bearing. Mohamad Ali et al. confirmed that variations in the oil groove location will affect the temperature profile [16]. The temperature profile of a lubricant with the corresponding journal speed and loads was investigated by S. Kasolang et al., and it was noticed that the temperature increased with increasing loads at a specific rotational speed [19]. Another study also found that the journal speed increased with lubricant temperature [20]. Amit Shingla simulated the thermo-hydrodynamic characteristics using computational fluid dynamics (CFD), and revealed that the temperature rise was less compared to an iso-thermal analysis due to variations in the viscosity [21]

1.3 Frictional forces

The resistance to lubricant flow is basically attributed to the viscosity. A less viscous lubricant will have a smoother sliding motion than a highly viscous lubricant, thereby promoting a low friction coefficient. The friction performances of bio-based, synthetic and mineral oil lubricants in journal bearings were examined by Pantelis et al. and it was found that the bio-based lubricant exhibited the lowest friction coefficient [22]. Mohamad Ali Ahmad et al. revealed that changes in the feed oil pressure and oil groove location tend to affect the friction coefficient [23,24]. The friction coefficient of fluid lubricants is attributed to the pressure acting on the projected area, the journal speed, the lubricant viscosity as well as the geometrical dimensions of the bearing.

2. METHODOLOGY

2.1 Journal bearing test rig

An advanced journal bearing test rig was used in this study, as shown in Fig. 1. The main function of this machine is to measure the oil film pressure and temperature at different positions along the circumference of the bearing. The shaft or journal was mounted horizontally to the aligned bearing. The rotation of the shaft was driven by a servo motor that was connected and coupled to the trough at a time-belt pulley ratio of 1:2.

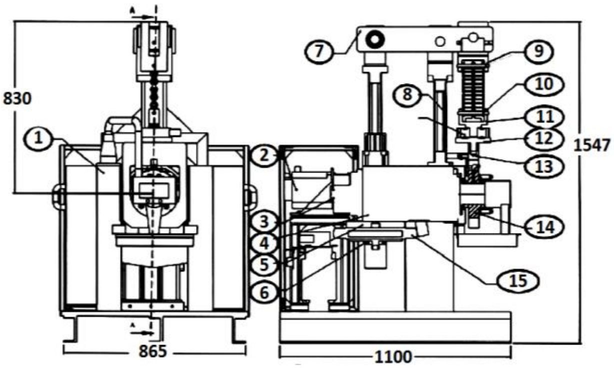

Fig. 1. Diagram of journal bearing test rig. (1) Frame structure, (2) motor, (3) motor bracket, (4) spindle assembly, (5) bellows top plate, (6) bellows guide plate, (7) loading lever, (8) pivot assembly, (9) chain, (10) chain holder, (11) load cell holder, (12) load cell, (13) loading plate, (14) journal bearing, (15) pneumatic bellow. Adapted from [16].

The shaft was freely rotated in the specific bearing with sufficient clearance to form a hydrodynamic or thick film lubrication. A total of 12 units of pressure and temperature sensors were placed and fixed on the bearing, as shown in Fig. 2.

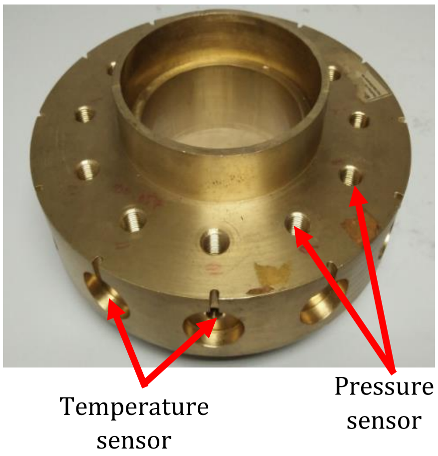

Fig. 2. The location of pressure and temperature sensors fixed on the bearing.

The setting of the parameters for the experiment was controlled by both an electric and electronic controller, while the data and results were processed by a data acquisition system. The oil lubricant for the experiment was passed through a heat exchanger to provide the desired oil temperature during the experiment. An external or radial load was provided by a pneumatic system and was measured by load cells with a capacity of 5T and 10T. The cylinder load cell was mounted on the loading plate that was being used to hold the bearing, and was simultaneously connected to the loading lever by a roll and chain, as presented in Fig. 3.

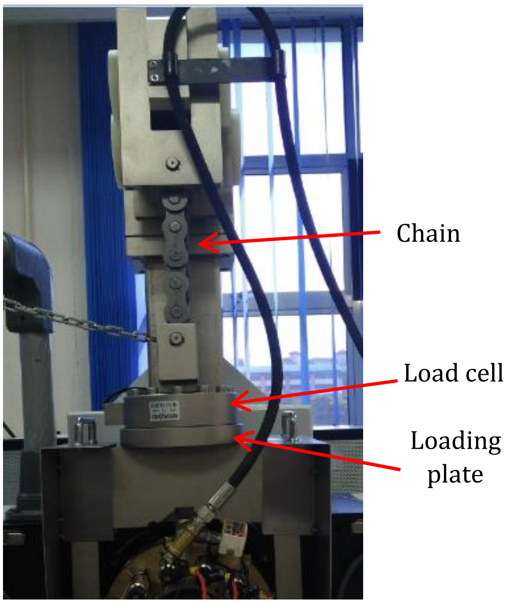

Fig. 3. Load is applied by pneumatic bellow in upward direction.

When compressed air was injected into the pneumatic bellow, it expanded and resulted in the loading plate being pulled up. Forces were exerted by the load cell and the signals were transmitted to the controller. The journal bearing was made of carbon steel and had a length to diameter ratio of 0.5. The general specifications of the bearing and sensors are tabulated in Table 1.

Table 1. Journal bearing and instrument's specifications.

Parameter	Value
Journal Material	Carbon steel
Bearing Material	Phosphorus bronze
Journal diameter	100.020 mm
Bearing diameter	100.104 mm
Bearing length	50 mm
Radial clearance	0.042 mm
Pressure sensor	
Type	Digital SS Isolated
Measuring range	0.001 – 10 MPa
Accuracy	0.001±1% MPa
Temperature sensor	
Type	PT 100
Measuring range	1 – 200 °C
Accuracy	1±1% °C
Frictional torque sensor	
Type	Beam type load cell
Measuring range	30 kg
Accuracy	0.01±1% Nm

In the present study, the applied radial load and journal rotational speed were varied accordingly. Radial loads of 10 kN, 20 kN and 30 kN were applied, while the rotational speeds were set at 200 rpm, 400 rpm, 600 rpm and 800 rpm, respectively. The experiment was carried out at ambient temperature with a duration of about 25 minutes per cycle.

2.2 Friction measurements

The journal test rig was also equipped with a friction torque sensor to measure the frictional forces. The beam-type load cell torque sensor was mounted in a spindle housing, as shown in Fig. 4, and positioned longitudinally at 180 mm away from the centre of the bearing. When the journal was rotating, the lever arm would press the loading pin, and the friction torque sensor would trigger the signal and record the value, as shown in Fig. 5. The value of the friction coefficient would be computed by the data acquisition system.

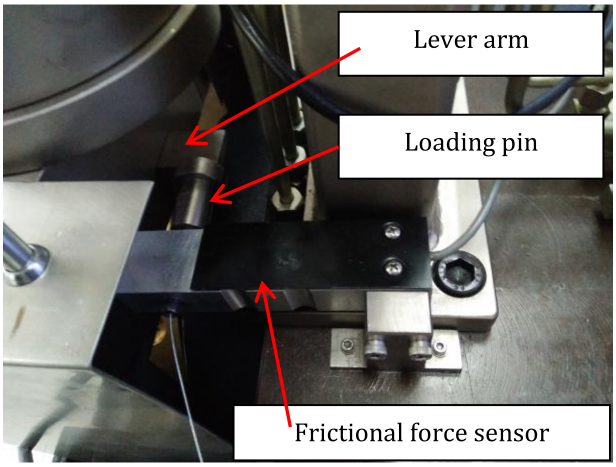

Fig. 4. The assembly of frictional force measurement.

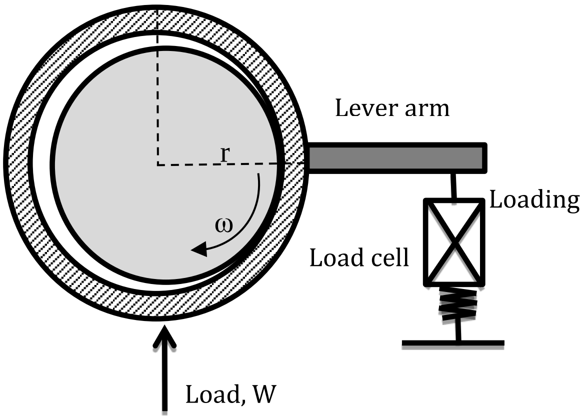

Fig. 5. The mechanism of friction measurement

2.2 Lubricant properties

Two types of lubricants were tested in this experiment, namely, a mineral-based engine oil (SAE 40) and palm oil-based (palm mid olein – PMO) lubricant. PMO is derived from double fractionated refined bleached and deodorized palm olein. It is also known as a by–product of palm mid fraction (PMF). The viscosity of SAE 40 and PMO at a temperature of 40 °C was 0.0897 Pa.s and 0.0299 Pa.s, respectively. The fatty acid content (%) for PMO was C12:0; 0.5, C14:0; 1.1, C16:0; 45, C18:0; 6.4, C18:1; 37.3, C18:2; 8.8, C18:3; 0.2, C20:0; 0.5 and C20:1; 0.1.

3. RESULTS AND DISCUSSION

3.1 Viscosity profile analysis

The viscosity profile over the rise in temperature rise was plotted as presented in Fig. 6.

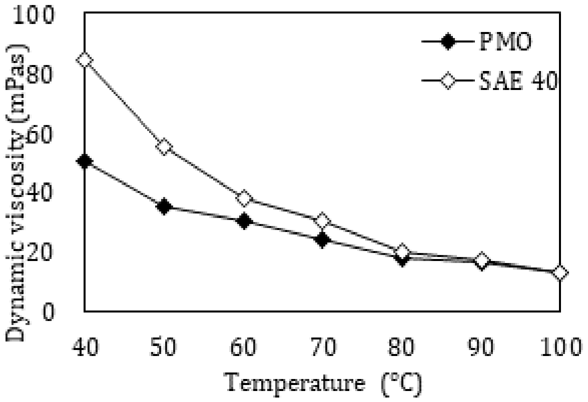

Fig. 6. Viscosity profile for tested lubricant.

SAE 40 exhibited a higher viscosity at low temperatures compared to PMO. This indicated that PMO has a higher viscosity index (VI) compared to SAE 40. The PMO showed less variance in the viscosity with a rise in the temperature. The higher viscosity index indicated that the oil had good thermo-viscosity properties and was not greatly affected by temperature variations. The thermo-viscosity performance also affected the fluidity of the lubricants. In this case, PMO possessed better fluidity due to its lower viscosity. The fluidity characteristic allowed the fatty acid molecules to move freely and be adsorbed onto the contacted surface. The fatty acid molecule structures created a span that provided a good lubricant film, thereby reducing the coefficient of friction [25]. The very high viscosity readings indicated that a huge amount of force was required to move the molecules to the surrounding area [26]. Any oil with a very low viscosity is also not recommended for use as a lubricant because it can be easily cleared off by even a slight force, and hence, there will be less surface protection. This will increase the coefficient of friction, which is contrary to the function of a lubricant.

3.2 Pressure distribution along the bearing circumference

The pressure that developed along the mid-plane of the bearing was measured by highly accurate pressure sensors located at every 30° angle outside the bearing. The journal bearing was considered to be operating at a stable condition as the oil pressure was pulsating at a similar frequency for every revolution, as presented in Fig. 7. The pressure profile graphs for both PMO and SAE 40 at different rotational speeds and radial loads were plotted, as shown in Figs. 8-11.

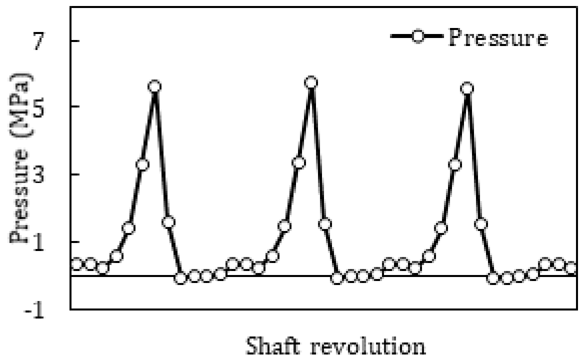

Fig. 7. Pressure pulsation for few shaft revolutions.

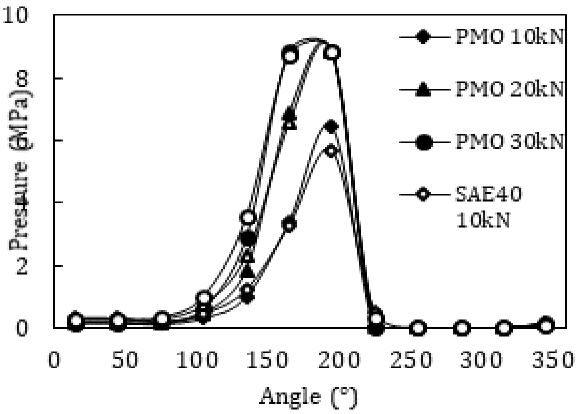

Fig. 8. Pressure profile at 200 rpm.

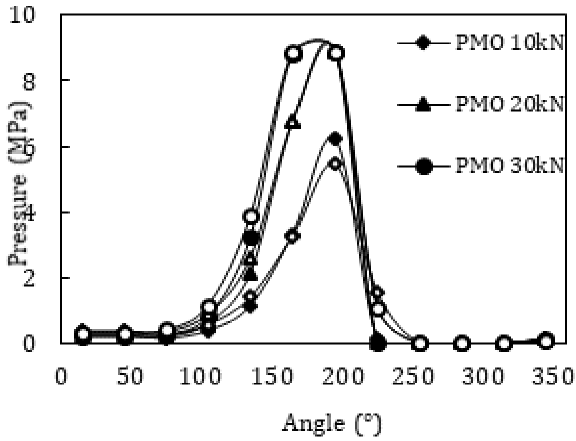

Fig. 9. Pressure profile at 400 rpm.

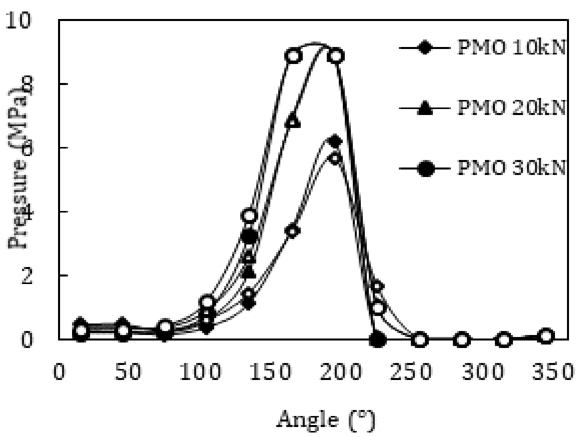

Fig. 10. Pressure profile at 600 rpm.

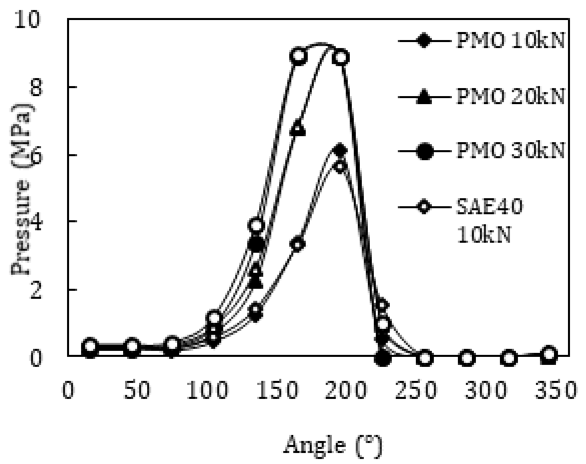

Fig. 11. Pressure profile at 800 rpm.

It was found that the pressure that was developed along the circumference of the bearing increased with increasing radial loads. This validated the theory that an increase in the radial load is directly proportional to the increase in pressure. The hydrodynamic pressure was increased to counter and support the increasing load. Meanwhile, as the speed increased, the oil film pressure was considered as consistent. The increase in journal speed in this study did not have much of an effect on the oil film pressure. At a fixed load, the increase in journal speed was accommodated by changes in the eccentricity ratio in order to remain stable in a hydrodynamic lubrication regime. At this point, the minimum film thickness would be increased within the limits without affecting the oil viscosity. The maximum pressure is developed when the fluid film enters the converging section, and suddenly, the pressure may drop to become a negative value as it passes into the diverging section. The minimum film thickness position lies on the borderline of these two regions.

From among the 12 pressure point locations that were measured, the highest pressure was found to be at an angular position of 195° for all testing conditions. The point of highest pressure was supposed to shift to another position with increasing radial loads. However, no offset was found on the maximum pressure location even with increasing speeds and radial loads for both the PMO and SAE 40 lubricants. At all the tested speeds, PMO showed a higher maximum pressure compared to SAE 40 at a load of 10 kN. PMO demonstrated maximum pressures that were higher than SAE 40 by 13.34 %, 14.19 %, 9.74 % and 8.49 % at speeds of 200 rpm, 400 rpm, 600 rpm and 800 rpm, respectively. However, as the load was increases to 20 kN and 30 kN, no significant difference was noted for both the PMO and SAE 40 lubricants, and the difference in the maximum pressure was less than 1 %. PMO showed a higher maximum pressure because of its thermo-viscosity behaviour, where it possesses a high viscosity index (VI) which provides it with very stable viscosity properties with a rise in temperature [27]. At lower temperatures, PMO exhibited a lower viscosity compared to SAE 40. As the temperature rose with an increase in the speed and applied load, the viscosity of SAE 40 was lowered and approached that of PMO. This also caused the difference in the maximum pressure between PMO and SAE 40 to become closer at a higher radial load and higher rotational speed.

3.3 Temperature profile along the bearing circumferences

The temperature readings were measured by PT 100 sensors located at the same locations as the pressure sensors. There was also a PT 100 sensor located at an angle of 0° to measure the oil inlet temperature. The temperature profiles for both the PMO and SAE 40 lubricants along the mid-plane of the bearing were plotted with respect to changes in the journal speed and radial load, as presented in Figs. 12-15. In this analysis, the performance of the lubricant was evaluated based on its capability to withstand increases in the temperature due to increasing speeds and radial loads. Generally, it was found that the temperature increased with increasing speeds and loads. This was because of the increase in shear stress in relation to the increasing velocity gradient across the thickness of the oil film. As can be seen in Fig. 8, the maximum temperature was found to have occurred at an angle of 165° for all cases. At loads of 10 kN and 20 kN, PMO demonstrated a higher temperature compared to SAE 40. Meanwhile, at a load of 30 kN, SAE 40 exhibited a higher temperature at angles of 105° to 255°. For PMO, the temperature increased by 1.7 °C, 2.3 °C and 2.7 °C for a loads 10 kN, 20 kN and 30 kN, respectively, while for SAE 40, the temperature increased by 1.9 °C, 3.7 °C and 4.7 °C for loads of 10 kN, 20 kN and 30 kN, respectively.

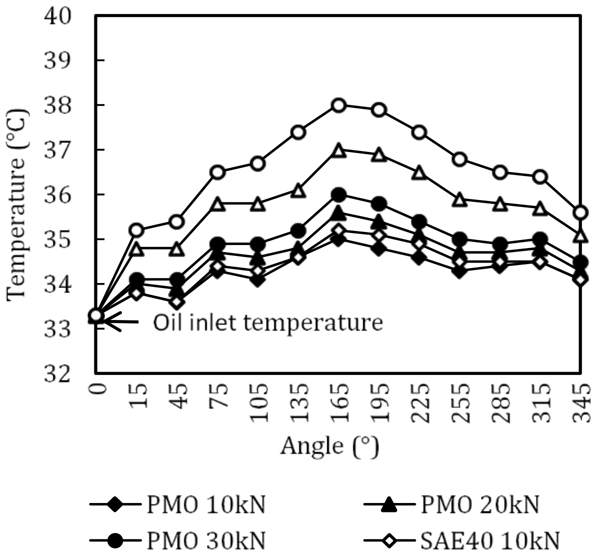

Fig. 12. Temperature profile at 200 rpm.

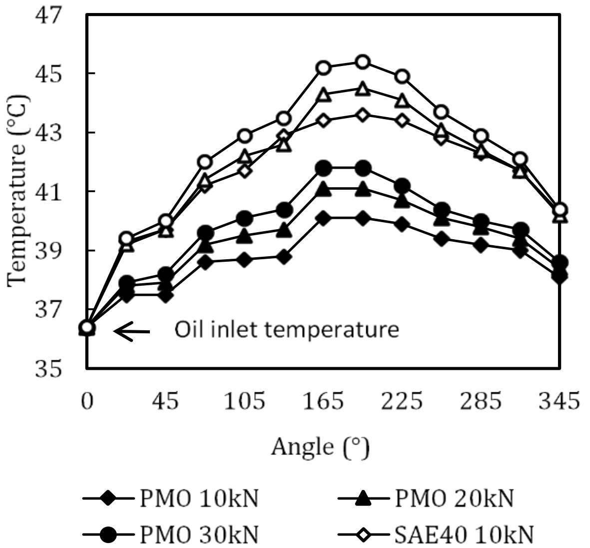

Fig. 13. Temperature profile at 400 rpm.

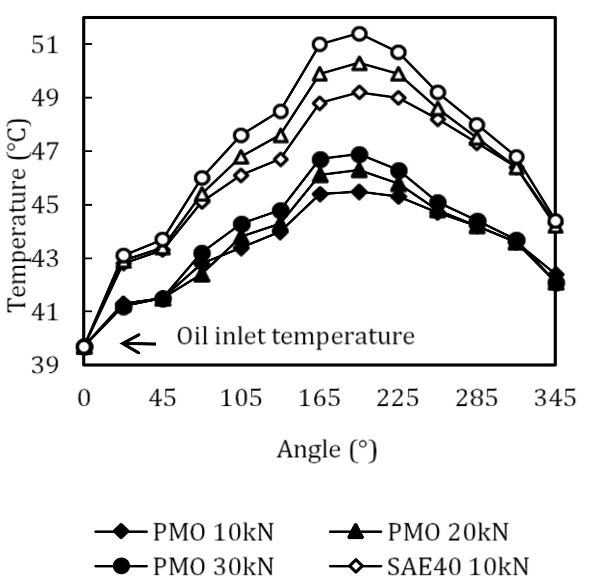

Fig. 14. Temperature profile at 600 rpm.

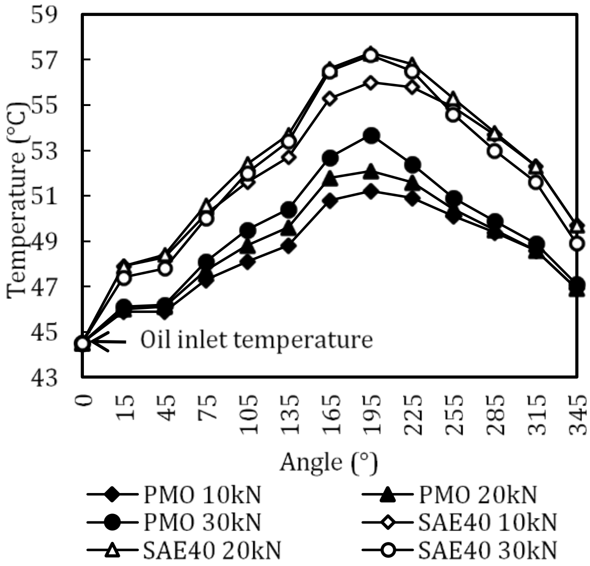

Fig. 15. Temperature profile at 800 rpm.

At a journal speed of 400 rpm, the maximum temperature was found to be at an angle of 195°, which was the same location at which the maximum pressure was presented. This was attributed to an increase in the fluid friction as it moved into the converging region. The temperature increases for PMO at loads of 10 kN, 20 kN and 30 kN were 3.7 °C, 4.7 °C and 5.4 °C, respectively. Meanwhile, for SAE 40, the temperature differences were 7.2 °C, 8.1 °C and 9.0 °C for loads of 10 kN, 20 kN and 30 kN, respectively. At a higher speed of 600 rpm, the maximum temperature was also at an angle of 195°. It was observed that SAE 40 demonstrated a higher temperature than PMO at all positions for all the radial loads. The temperature rise for SAE 40 was more prominent compared to PMO. The temperature rises for PMO at loads of 10 kN, 20 kN and 30 kN were 5.8 °C, 6.6 °C, and 7.2 °C, while for SAE 40 these were 9.5 °C, 10.6 °C and 11.7 °C, respectively. A similar trend was found at a speed of 800 rpm, with the highest temperature rise of 12.7 °C being recorded by SAE 40. The increase in temperature is generally proportional to the increase in the internal friction and oil film pressure in a fluid lubricant. In all the testing conditions, PMO exhibited a lower temperature rise compared to SAE 40. This was attributed to the excellent thermal resistivity of palm oil-based lubricants [27]. The ability to withstand a rise in temperature might help to stabilize the viscosity, thereby providing optimum operation of the bearing.

3.4 Friction coefficient measurements

The frictional forces were sensed by a highly accurate friction torque sensor located 180 mm away from the centre of the journal. This beam-type load cell sensor was used to measure the frictional forces against the direction of the rotating journal. The data on the frictional forces were then converted into the friction coefficients, as plotted in Fig. 16-19. Generally, it was found that the friction coefficient decreased with increasing radial loads at all speeds. The highest friction coefficient for PMO was 0.0359 at a speed of 600 rpm and radial load of 10 kN. Meanwhile, the highest friction coefficient for SAE 40 was 0.0488 at a radial load of 10kN and journal speed of 800 rpm. It was also observed that PMO demonstrated a lower friction coefficient compared to SAE 40 at all testing conditions. Figure 10 shows that PMO presented a friction coefficient that was lower on average by 44.03 % compared to SAE 40. As the speed was increased to 400 rpm, 600 rpm and 800 rpm, PMO demonstrated a similar trend, whereby the friction coefficients were 27.76 %, 32.35 % and 42.06 % lower compared to SAE 40, respectively.

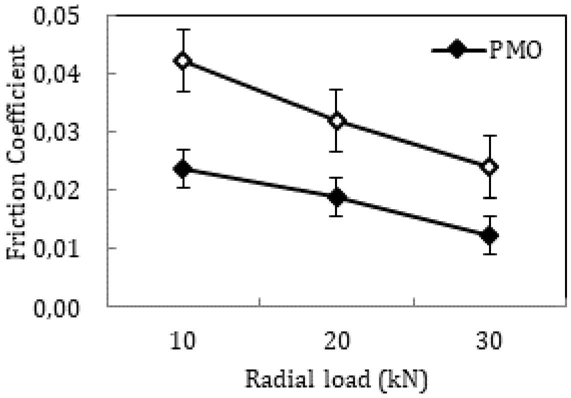

Fig. 16. Friction coefficient at 200 rpm.

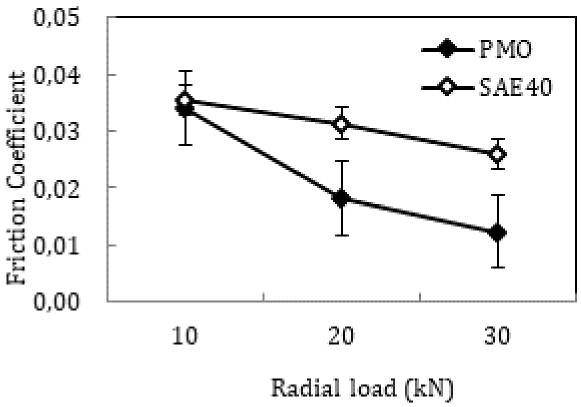

Fig. 17. Friction coefficient at 400 rpm.

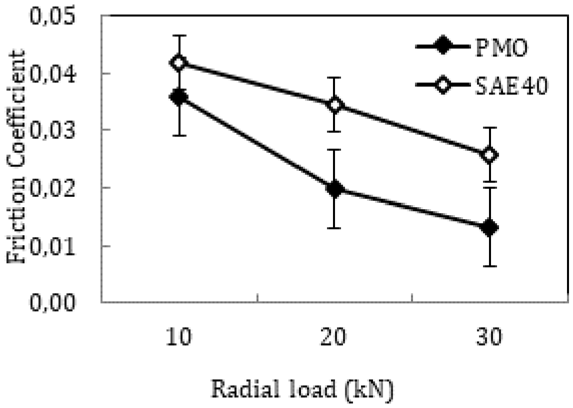

Fig. 18. Friction coefficient at 600 rpm.

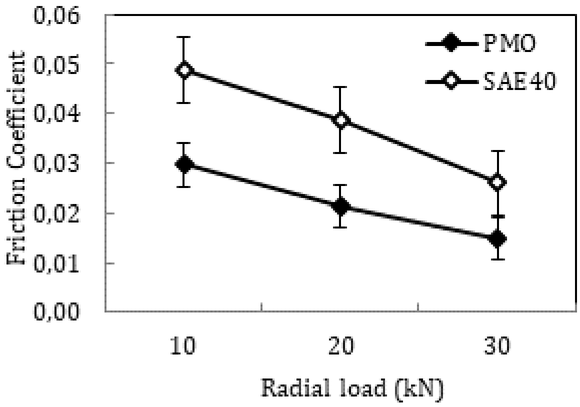

Fig. 19. Friction coefficient at 800 rpm.

These results were in agreement with the findings of many previous studies that palm oil-based lubricants are able to reduce the friction coefficient due to the long polar fatty acid chains that are built into their triglyceride hydrocarbon molecules. These fatty acid chains are responsible for the strong molecular interaction between the fatty acid and metal surface molecules [28,29], by which the fatty acid molecules are able to penetrate and firmly attach themselves to the surfaces of metals by both physical and chemical adsorption. This interaction also results in the formation of a thin soap film layer, which consists of orderly and closely packed arrays of molecules [30]. There is also a strong dipole interaction between the fatty acid chains. The existence of thus thin soap film layer facilitates the motion and minimizes the contact of asperities, thereby reducing the friction coefficient.

4. CONCLUSION

The performance of PMO in journal bearing applications under various operating conditions was examined and discussed in the present study.

In general, increasing the journal speed and radial load will lead to an increase in the maximum pressure and maximum temperature, but a decrease in the friction coefficient. By comparing the performance of the lubricants, it was concluded that PMO showed a higher maximum pressure at a lower load than SAE 40 and no significant differences were noticed at higher loads. PMO also exhibited better thermal resistance as it was able to minimize the rise in temperature with respect to increases in the rotational speed and radial load. In the analysis of the friction coefficient, PMO consistently demonstrated a lower friction coefficient at all testing conditions.

Acknowledgement

The authors would like to express their thanks to the Research Management Centre (RMC) of Universiti Teknologi Malaysia (UTM) for the Research Grant, GUP (17H96, 15J28, 20H29), TDR Grant (05G23), FRGS Grant (5F074), School of Mechanical Engineering, UTM and Ministry of Higher Education of Malaysia for their support. On behalf of all authors, the corresponding author states that there is no conflict of interest.

REFERENCES

[1] M.A. Ahmad, S. Kasolang, R. Dwyer-Joyce, N.R. Abdullah, *The effect of oil supply pressure on the circumferential pressure profile in hydrodynamic journal bearing*, Applied Mechanics and Materials, vol. 315, pp. 809-814, 2013, doi: 10.4028/www.scientific.net/AMM.315.809

[2] Y. Singh, A. Farooq, A. Raza, M.A. Mahmood, S. Jain, *Sustainability of a non-edible vegetable oil based bio-lubricant for automotive applications: A review,* Process Safety and Environmental Protection, vol. 111, pp. 701-713, 2017, doi: 10.1016/j.psep.2017.08.041

[3] M.A. Ahmad, S. Kasolang, R.S. Dwyer-Joyce, A. Jumahat, *The Effects of Oil Supply Pressure at different Groove Position on Temperature and Pressure Profile in Journal Bearing*, Journal Teknologi, vol. 66, no. 3, pp. 113 - 119, 2013, doi: 10.11113/jt.v66.2706

[4] P. Zulhanafi, S. Syahrullail, M.M. Faridzuan, *Tribological performance of palm kernel oil added with nanoparticle copper oxide using fourball tribotester.* Jurnal Teknologi, vol. 79, no. 7-4, pp. 53 - 59, 2017, doi: 10.11113/jt.v79.12265

[5] W.J. Bartz, *Lubricants and the environment*, Tribology international, vol. 31, iss. 1-3, pp. 35-47, 1998, doi: 10.1016/S0301-679X(98)00006-1

[6] S. Syahrullail, C.S.N. Azwadi, T.C. Ing, *The metal flow evaluation of billet extruded with RBD palm stearin*, International Review of Mechanical Engineering, vol. 5, no. 1, pp. 21-27, 2011.

[7] I. Golshokouh, M. Golshokouh, F.N. Ani, E. Kianpour, S. Syahrullail, *Investigation of physical properties for jatropha oil in different temperature as lubricant oil,* Life Science Journal, vol. 10, pp. 110-119, 2013.

[8] S. Syahrullail, K. Nakadshi, S. Kwnitani, *Investigation of the effects of frictional constraint with application of palm olein oil lubricant and paraffin mineral oil lubricant on plastic deformation by plane strain extrusion*, Japanese journal of tribology, vol. 50, no. 6, 727-738, 2005.

[9] S. Syahrullail, J.Y. Wira, W.B. Wan Nik, W.N. Fawwaz, *Friction characteristics of RBD palm olein using four-ball tribotester*. Applied Mechanics and materials, vol. 315, pp. 936-940, 2013, doi: 10.4028/www.scientific.net/AMM.315.936

[10] I. Golshokouh, S. Syahrullail, F.N. Ani, H.H. Masjuki, *Investigation of palm fatty acid distillate as an alternative lubricant of petrochemical based lubricants tested at various speeds,* International Review of Mechanical Engineering, vol. 7, no. 1, pp. 72-80, 2013.

[11] M. Del Din, E. Kassfeldt, *Wear characteristics with mixed lubrication conditions in a full scale journal bearing*, Wear, vol. 232, iss. 2, pp. 192-198, 1999, doi: 10.1016/S0043-1648(99)00145-3

[12] S. Baskar, G. Sriram, *Experimental analysis of hydrodynamic journal bearing under different biolubricants,,* in IEEE-International Conference On Advances In Engineering, Science And Management (ICAESM-2012), 30 March, 2012, pp. 132-135.

[13] D.M. Nuruzzaman, M.K. Khalil, M.A. Chowdhury, M.L. Rahaman, *Study on pressure distribution and load capacity of a journal bearing using finite element method and analytical method,* International Journal of Mechanical & Mechatronics Engineering IJMME-IJENS, vol. 10, no. 5, pp. 1-8, 2010.

[14] K.G. Binu, K. Yathish, R. Mallya, B.S. Shenoy, D.S. Rao, R. Pai, *Experimental study of hydrodynamic pressure distribution in oil lubricated two-axial groove journal bearing,* Materials Today: Proceedings, vol. 2, iss. 4-5, pp. 3453-3462, 2015, doi: 10.1016/j.matpr.2015.07.321

[15] S. Chatterton, P.V. Dang, P. Pennacchi, A. De Luca F. Flumian, *Experimental evidence of a two-axial*

groove hydrodynamic journal bearing under severe operation conditions, Tribology International, vol. 109, pp. 416-427, 2017, doi: 10.1016/j.triboint.2017.01.014

[16] M.A. Ahmad, S. Kasolang, R.S. Dwyer-Joyce, *Experimental study on the effects of oil groove location on temperature and pressure profiles in journal bearing lubrication,* Tribology International, vol. 74, pp. 79-86, 2014, doi: 10.1016/j.triboint.2014.02.012

[17] J.K. Wang, M.M. Khonsari, *Effects of oil inlet pressure and inlet position of axially grooved infinitely long journal bearings. Part I: Analytical solutions and static performance,* Tribology International, vol. 41, iss. 2, pp. 119-131, 2008, doi: 10.1016/j.triboint.2007.05.005

[18] J.K. Wang, M.M. Khonsari, *Effects of oil inlet pressure and inlet position of axially grooved infinitely long journal bearings. Part II: Nonlinear instability analysis,* Tribology International, vol. 41, iss. 2, pp. 132-140, 2008, doi: 10.1016/j.triboint.2007.05.006

[19] S. Kasolang, M.A. Ahmad, R. Dwyer-Joyce, A. Jaffar, M.A.A. Bakar, N.H. Saad, A. Jumahat, *Experimental Study of Temperature profile in a Journal Bearing.* in 1st Joint Symposium on System-Integrated Intelligence: New Challenges for Product and Production Engineering, pp. 43-45, 2012.

[20] A. Singla, P. Singh, A. Chauhan, *Experimental determination of temperature and pressure profile of oil film of elliptical journal bearing*, International Journal of Advanced Mechanical Engineering, vol. 4, no. 5, pp. 469-474, 2014.

[21] A. Singla, A. Kumar, S. Bala, P. Singh, A. Chauhan, *Thermo-hydrodynamic analysis on temperature profile of circular journal bearing using computational fluid dynamics,* in 2014 Recent Advances in Engineering and Computational Sciences (RAECS) on 6 March, 2014, pp. 1-6, doi: 10.1109/RAECS.2014.6799595

[22] P.G. Nikolakopoulos, D.A. Bompos, *Experimental measurements of journal bearing friction using mineral, synthetic, and bio-based lubricants*, Lubricants, vol. 3, iss. 2, pp. 155-163, 2015, doi: 10.3390/lubricants3020155

[23] M.A. Ahmad, S. Kasolang, R. Dwyer-Joyce, *The effects of oil supply pressure at different groove position on frictional force and torque in journal bearing lubrication,* Procedia Engineering, vol. 68, pp. 70-76, 2013, doi: 10.1016/j.proeng.2013.12.149

[24] M.A. Ahmad, S. Kasolang, R. Dwyer-Joyce, M.A.A. Bakar, *The Effects of Oil Groove Position on Torque and Frictional Force in Hydrodynamic Journal Bearing,* Applied Mechanics and Materials, vol. 393, pp. 907-912, 2013, doi: 10.4028/www.scientific.net/AMM.393.907

[25] A.S.M.A. Haseeb, S.Y. Sia, M.A. Fazal, H.H. Masjuki, *Effect of temperature on tribological properties of palm biodiesel,* Energy, vol. 35, iss. 3, pp. 1460-1464, 2010, doi: 10.1016/j.energy.2009.12.001

[26] Y. Wang, Q.J. Wang, C. Lin, F. Shi, *Development of a set of Stribeck curves for conformal contacts of rough surfaces*, Tribology transactions, vol. 49, iss. 4, pp. 526-535, 2006,

[27] K.S.V. Reddy, N. Kabra, U. Kunchum, T. Vijayakumar, *Experimental investigation on usage of palm oil as a lubricant to substitute mineral oil in CI engines,* Chinese Journal of Engineering, vol. 2014, 2014, p. 5, doi: 10.1155/2014/643521

[28] N.J. Fox, G.W. Stachowiak, *Vegetable oil-based lubricants—a review of oxidation.* Tribology international, vol. 40, iss. 7, pp. 1035-1046, 2007, doi: 10.1016/j.triboint.2006.10.001

[29] N. Sapawe, S. Samion, P. Zulhanafi, C.S. Nor Azwadi, M.F. Hanafi, *Effect of addition of tertiary-butyl hydroquinone into palm oil to reduce wear and friction using four-ball tribotester*, Tribology Transactions, vol. 59, iss. 5, pp. 883-888, 2016, doi: 10.1080/10402004.2015.1118584

[30] D. Kenbeck, D.T.F. Bunemann, *Organic friction modifiers,* in L.R. Rudnick, Lubricant Additives: Chemistry and Applications. CRC Press, pp. 195 - 209, 2009.