Analysis of Flow and Temperature Distribution in Journal Bearing Lubricated with Vegetable Oil

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Abstract

The use of vegetable oil as a sustainable alternative to mineral oil in lubrication applications has garnered considerable attention. This research focuses on evaluating three different vegetable oils, namely palm mid olein (PMO), refined bleached deodorized palm oil (RBDPO), and rapeseed oil, for their effectiveness in lubricating a hydrodynamic journal bearing. The study examines key performance factors such as pressure, temperature, load-carrying capacity, and coefficient of friction. Computational fluid dynamics (CFD) is employed to compare the lubricating capabilities of the vegetable oils against engine oil. Rapeseed oil emerges as the most promising option among the vegetable oils, displaying notable advantages. It demonstrates a 22.12% higher pressure, 26.37% higher load-carrying capacity, and 0.37% higher temperature distribution compared to PMO. In contrast, while engine oil exhibits a 16.67% higher pressure distribution and 35.8% higher load-carrying capacity, it also comes with a drawback of 44.29% higher coefficient of friction compared to other vegetable oils. It is worth noting that the simulation results have been validated through experimental data, leading to the conclusion that engine oil surpasses vegetable oils in terms of pressure and temperature distribution.

Keywords

Computational Fluid Dynamics; Hydrodynamic Journal Bearing; Coefficient of Friction; Load Carrying Capacity; Vegetable Oil

1.0 INTRODUCTION

According to Juvinall and Marshek [1], a journal bearing is a crucial component in machinery, consisting of a cylindrical body that encloses a spinning shaft and serves to carry radial loads or provide torque guidance. The stationary sleeve of the housing structure forms a complete 360-degree arc, creating a gap between the shaft and the bearing where lubrication occurs to reduce friction and extend the bearing's lifespan [2]. There are three types of lubrication regime typically formed inside journal bearing which are hydrodynamic lubrication, mixed film lubrication, and boundary lubrication. Normally, mineral oils have been utilized as lubricants in the bearing. However, due to sustainability issues of mineral oil, an alternative approach to overcome this issue is to employ vegetable oils as the lubricant.

Vegetable oils have been the subjects of many researchers as potential replacements for mineral oils in lubrication applications. Zulhanafi et al. [3] studied the performance of a journal bearing using palm-mid olein to measure the coefficient of friction. Khasbage et al. [4] analyzed the performance of the bearing lubricated with jatropha oil and stated that vegetable oils have huge potential to replace the mineral oil. Additionally, Sriram and Baskar [5] conducted experimental studies to determine the temperature distribution of vegetable oils and found that the vegetable oils had higher temperatures than mineral oil. Conversely, Patel et al. [6] observed that mineral oil and vegetable oil exhibited similar pressure distribution and load-carrying capacity, based on their analysis.

Nowadays, the use of computational fluid dynamics (CFD) to predict lubrication performance in journal bearings has been on the rise due to the advancements in numerical methods and computer hardware capabilities. Rasep et al. [7] performed simulations of a journal bearing and suggested that palm oil needs further modifications to improve its oil characteristics. Li et al. [8] conducted a study on temperature distribution in a journal bearing using conjugate heat transfer, accounting for both conduction and convection and found the significance of both modes pf the heat transfer for temperature analysis. Dhande and Pande [9] used ANSYS FLUENT to simulate flow in journal by considering the cavitation model.

The purpose of this study is to investigate the performance of vegetable oils as lubricants in a journal bearing. Computational fluid dynamics (CFD) is employed to determine key performance parameters of the journal bearing. The output parameters are pressure distribution, load-carrying capacity, coefficient of friction, and temperature distribution. A comparison of performance between engine oil and vegetable oils is made to determine which one is better based on the parameters that had been mentioned before. The hydrodynamic lubrication is considered in this study due to its simplicity and excellent damping properties that can be used during high-speed and load operation where the hydrodynamic is defined as two surfaces separated by the lubricant film.

2.0 METHODOLOGY

Governing Equation

ANSYS Fluent was utilized as the CFD tool for predicting lubrication flow in the journal bearing. In line with recommendations from the literature (as discussed in the Introduction section), this study incorporated the cavitation and conjugate heat transfer models. For the cavitation model, Zwart-Berber Balamri was used as it is known to be less sensitive to mesh density, robust, and fast converging, as suggested by Rasep et al. [7]. The multiphase model was included in this analysis for the cavitation model to account for the transition of liquid and vapour phases. As for the conjugate heat transfer, this study only considered convection and conduction. The governing equations incorporated in ANSYS Fluent and being used in the current study are expressed as Equation (1) – (5) [10]. The Equation (1) and (2) are the continuity and momentum equations for laminar fluid flow. The conjugate heat transfer equation are expressed as Equation (3) and (4) for convective flow and conduction in solid, respectively. Equation that described cavitation phenomenon is expressed as Equation (5).

$$\frac{\partial\rho}{\partial t} + \nabla\cdot(\rho\vec{v}) = 0 \tag{1}$$

where ρ is the density and $\vec{v}$ is the velocity of the fluid in terms of a vector.

$$\frac{\partial\rho}{\partial t}(\rho\vec{v}) + \nabla(\rho\vec{v}\vec{v}) = -\nabla p + \nabla(\bar{\bar{\tau}}) + \rho\vec{g} + \vec{F} \tag{2}$$

where p is pressure, $\bar{\bar{\tau}}$ is stress tensor, $\overrightarrow{g}$ is the gravitational constant, and $\vec{F}$ is the external body force.

$$\frac{\partial}{\partial t}(\rho_m E) + \nabla\cdot(v\rho_m E + vp) = \nabla\left(k_{eff}\nabla T - \sum_{k-1}^{n} h_k J_k + (\bar{\bar{\tau}}_{eff}\vec{v})\right) + S_{h,f} \tag{3}$$

where ρ_m is mixture density, E is total energy, v is kinematic viscosity, k_{eff} is effective conductivity, T is temperature, h_k is enthalphy of phase k, J_k is diffusion flux of phase k, $\bar{\bar{\tau}}_{eff}$ is viscous dissipation and $S_{h,f}$ is the fluid volumetric heat source.

$$\frac{\partial}{\partial t}(\rho_s h) + \nabla\cdot(\vec{v}\rho_s h) = \nabla\cdot(k_s\nabla T) + S_{h,s} \tag{4}$$

where ρ_s is solid density, h is sensible enthalpy, k_s is solid conductivity, and $S_{h,s}$ is solid volumetric heat source.

$$\frac{\partial}{\partial t}(\alpha\rho_v) + \nabla\cdot(\alpha\rho_v\vec{v}_v) = R_e - R_c \tag{5}$$

where α is vapor volume fraction, ρ_v is vapor density, $\vec{v}_v$ is vapor phase velocity, R_e and R_c are mass transfer source for the growth and collapse of the vapor bubbles, respectively.

Numerical Setup

The dimension of the bearing had been referred from Zulhanafi *et.al* [11]. Figure 1 shows the boundary condition of the model of the solid and fluid domain while Table 1 provide geometrical dimension and operating conditions of the journal bearing.

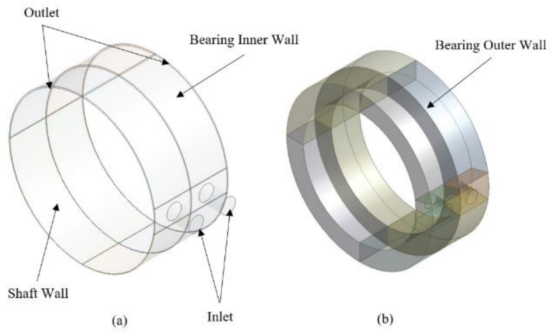


Figure 1: Boundary condition of the model (a) Fluid domain (b) Solid domain

Table 1: The geometrical parameters for the numerical study

Parameters	Unit	Value
Bearing Diameter	mm	100
Shaft Diameter	mm	99
Clearance	mm	0.5
Length-Diameter Ratio	-	0.5
Rotational Speed	RPM	1000, 800, 600, 200

In this study, engine oil, refined bleached deodorized palm oil (RBDPO), palm mid olein (PMO), and rapeseed oil are the considered lubricants to be tested. Table 2, 3 and 4 summarizing all the lubricants and model parameters that are used in the numerical study. Table 5 shows the numerical schemes used to solve the governing equations by means of finite volume method.

Table 2: Thermophysical properties of oils

Oil	Density at 15°C (kg/m³)	Viscosity at 40°C (Pa.s)	Reference
Engine oil	875	0.1044	Asral (2015) [12]
RBDPO	880	0.1667	Zulhanafi (2021) [13]
PMO	895	0.0453	Zulhanafi (2021) [13]
Rapeseed oil	925	0.0323	Zongo et al. (2019) [14]

Table 3: Thermophysical properties and cavitation parameters

Parameter	Unit	Value	Reference
Bubble Radius	m	1.00×10^{-6}	Gao et al. (2014) [15]
Nucleation Site Volume Fraction	-	5.00×10^{-4}	
Evaporation Coefficient	-	50	
Condensation Coefficient	-	0.01	
Vapor Pressure of Engine Oil	Pa	20000	Tauviqirrahman et al. (2019) [16]
Vapor Pressure of Palm Oil	Pa	29185	Hilmy et al. (2018) [17]
Vapor Pressure of Rapeseed Oil	Pa	20159	Murata et al. (1993) [18]

Table 4: Conjugate heat transfer conditions

Parameters	Condition
Outside of the Bearing	Convective heat transfer coefficient with 10 W/m²K
Inlet	308 K
The Bearing Interface	Coupled
The Shaft Interface	Zero Heat Flux

Table 5: Numerical methods

Parameters	Scheme
Pressure-Velocity Coupling	Coupled
Pressure	PRESTO!
Momentum	First Order Upwind
Volume Fraction	First Order Upwind
Energy	First Order Upwind

The results of pressure distribution around the circumference of the journal bearing and bearing torque are used to calculate the load-carrying capacity and the coefficient of friction of the journal bearing. The load-carrying capacity can be determine using Equation (6) while the coefficient of friction can be determine using Equation (7).

$$W = \sqrt{W_x^2 + W_y^2} \tag{6}$$

where W is the load-carrying capacity, W_x is the load in *x*-direction and W_y is the load in the *y*-direction.

$$f_T = \frac{F_T}{W} \tag{7}$$

where f_T is the coefficient of friction, F_T is the torque and W is the load-carrying capacity.

Grid Independence Test

A mesh independence study was performed to determine the optimal number of elements required before proceeding with the validation study. The parameters employed for this investigation were as follows: a rotational speed of 1000 rpm, an eccentricity is 0.5 and the use of engine oil as the lubricant. Figure 2 illustrates the plot of maximum pressure against the number of elements. It can be observed that convergence of the maximum pressure results occurs at 162,256 mesh elements, with a value of 13.4 kPa.

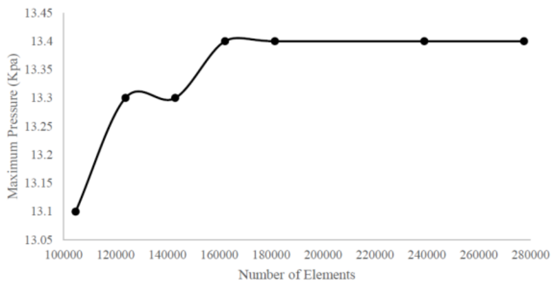


Figure 2: Maximum pressure against number of elements

3.0 VALIDATION

Validation study is performed by comparing current study against published experimental data [13] with engine oil as the lubricant. The speed of the journal is set at 1000 rpm. Since the eccentricity ratio is unknown from the experimental data, trial-and-error method was performed to determine the best eccentricity. Figure 3 shows pressure distribution along the journal circumferential for seven eccentricities (0.5, 0.6, 0.62, 0.64, 0.66, 0.68, and 0.7). Referring to Figure 3, the best eccentricity for the simulation model is 0.64 with the percentage error is about 1.12%. It can be concluded that the simulation results has been successfully validated with the experimental data.

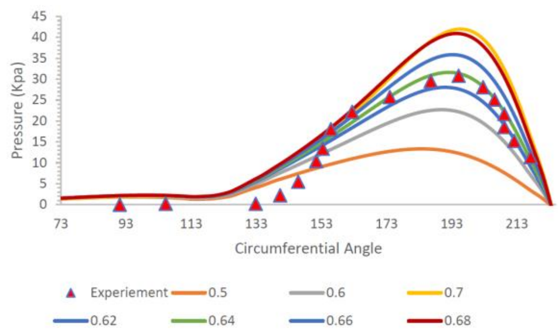


Figure 3: Validation study of engine oil at various eccentricities

4.0 RESULTS AND DISCUSSION

Pressure Distribution

Figure 4(a) shows pressure distribution along the bearing circumference for four lubricants (engine oil, rapeseed oil, PMO, and RBDPO) at 1000 rpm. Engine oil recorded the highest maximum pressure where the maximum pressure was obtained near the converging area of minimum oil film thickness. Referring to Figure 4(b), rotational speed was found to significantly affect the oil pressure for the case of engine oil. The pressure increased with the increase of the rotational speed due to the load generated on the bearing that causes the increment of pressure.

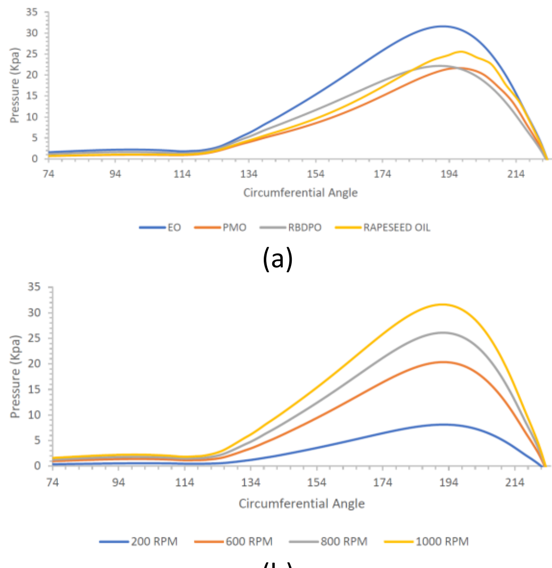


Figure 4: Pressure distribution against circumferential angle **(a)** at 1000 rpm for various oils **(b)** for engine oil at various rotational speeds

Load Carrying Capacity

Figure 5 shows shaft load against rotational speed. The maximum load that had been obtained for all lubricants is 68 N when the speed of the journal is at 1000 rpm. Generally, the load increased with the increase in speed. Engine oil was found to produced the maximum load, which is relatively 35.81% higher than the second-highest load, that is rapeseed oil. Meanwhile, the load-carrying capacity of RBDPO is 26.37% greater than PMO although these two lubricants are coming from the same type of oil, which is palm oil. When comparing the pressure peak and load of all oils at 1000 rpm as in Figure 4(a) and Figure 5, it can be said that the higher the pressure distribution is, the greater the load applied to the bearing, increasing the capability of the journal bearing to support the weight of the shaft.

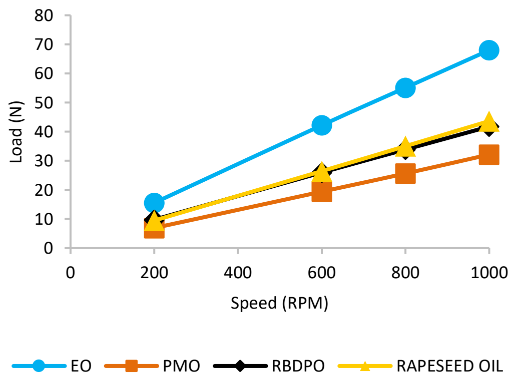


Figure 5: Load carrying capacity against rotational speed

Coefficient of Friction

Figure 6 shows coefficient of friction of bearing against rotational speed. Engine oil exhibits the highest coefficient of friction, which is about 2.86% greater than RBDPO, PMO, and rapeseed oil. Although RBDPO and PMO came from the same type of vegetable oil, RBDPO shows significantly larger coefficient of friction than PMO because the two oils had different fatty acid compositions that causes their viscosity to also differ. Meanwhile, rapeseed oil yields the lowest coefficient of friction amongst the tested oils at all rotational speeds considered in this study.

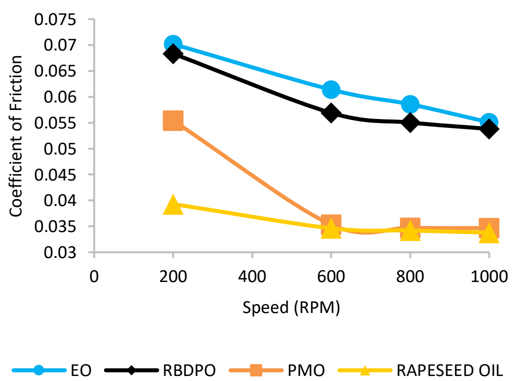
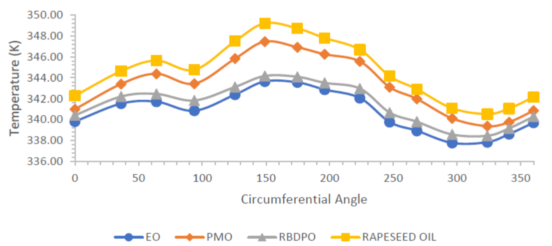


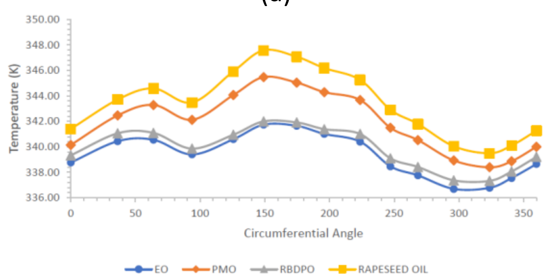
Figure 6: Coefficient of friction against rotational speed

Temperature Distribution

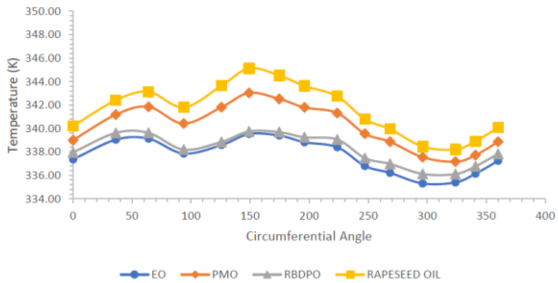
Figure 7 shows the temperature distribution of the bearing at different rotational speeds while Figures 8 shows the temperature contour of the shaft at various rotational speeds. Both figures (7 and 8) shows that the temperature increases as the rotation speed increases. PMO recorded the lowest temperature distribution among the other vegetable oils. Rapeseed oil shows 1.46% while RBDPO and PMO shows 0.14% and 0.95% higher, respectively when compared with engine oil at 1000 rpm. The PMO has a higher viscosity index than RBDPO and rapeseed oil. However, the highest temperature distribution is not PMO according to Sadiq et al. [19]. In this study however shows that the highest temperature recorded is rapeseed oil, which means rapeseed oil is not suitable to be use as a lubricant due to high-temperature increment, causing the viscosity to be thinner. This is because the specific heat of rapeseed oil is larger than the other vegetable oils, which may contribute to the high temperature.



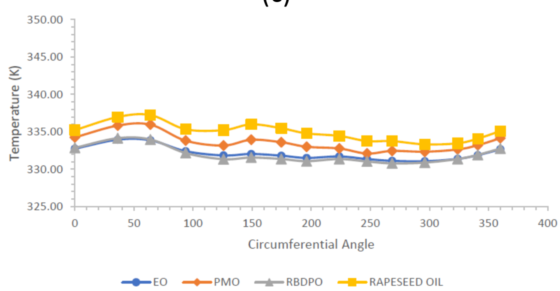
(a)



(b)



(c)



(d)

Figure 7: Temperature distribution at (a) 1000rpm, (b) 800 rpm, (c) 600 rpm, (d) 200 rpm

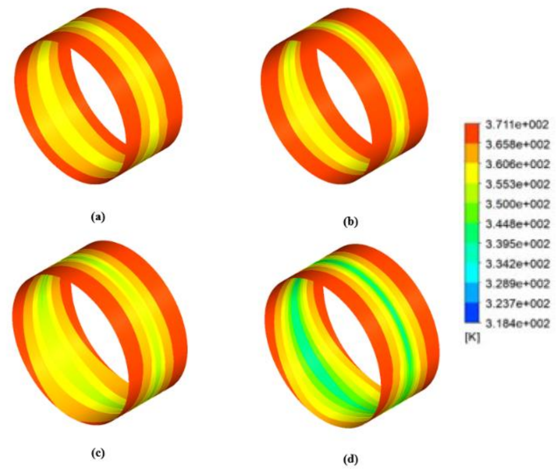


Figure 8: Temperature contour of the shaft at (a) 1000rpm, (b) 800 rpm, (c) 600 rpm, (d) 200 rpm

5.0 CONCLUSION

Based on the presented numerical study, it was found that rapeseed oil exhibits 22.12% higher pressure, 26.37% higher load carrying capacity, and 0.37% higher temperature compared to PMO. Moreover, when compared to RBDPO, rapeseed oil demonstrates a 24.91% higher pressure, 4.33% higher load-carrying capacity, and 1.44% higher temperature. However, rapeseed oil has a 29% lower coefficient of friction in comparison to other vegetable oils. On the other hand, engine oil displays a 16.67% higher pressure, 35.8% higher load-carrying capacity, 44.29% higher coefficient of friction and 0.151% lower temperature when compared to other vegetable oils, signifying its excellent performance. As a result, the considered vegetable oils are deemed unsuitable for use as lubricants. It is recommended that further modifications be made to improve the lubricating performance of vegetable oils, such as incorporating additives.

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