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Computational Analysis of the influence of Two Circumferential Grooves on Performance of Journal Bearing with Palm Oil as Lubricant

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ABSTRACT
This study compares the performance of a bearing with surface waviness liner to a plain bearing. Both bearings are functioned with palm oil as lubricant so that the potential advantages of the journal bearings could be identified. CFD analysis is developed to predict the numerical data for full film lubrication condition. A bearing of 60 mm in diameter with its ratio of length to diameter is 0.5, a clearance 250µm, and 200µm of wave amplitude is simulated. Semi circular wavy surface produces better results in term of load capacity than the plain journal bearing for a small eccentricity. The load carrying capacity is influenced by the increasing in speed of the shaft and the eccentricity ratio.

Keyword: *Load Carrying Capacity; Eccentricity; Wavy; Grooves; Temperature Drop*.

1.0 INTRODUCTION

In designing a bearing system, there are several important aspects that must be considered i.e the method to deliver the lubricant to the bearing, the distribution of the lubricant within the bearing, the quantity of the lubricant which is required and the amount of heat which is generated by the bearing and its effect on the temperature of the lubricant. A good journal bearing is able to support high load with less friction so that it can reduce the losses of energy. In this case, the properties of the lubricant may also be considered as it influences the contact between journal and bearing by a layer of fluid with a certain thickness. As an effort to improve the efficiency of the machine, some modifications on the bearing geometry are highly recommended. On the other hand, the usage of bio-based lubricant will enhance the operation of the machine to be environmental friendly.

In order to minimize the factor of friction, improvements on the surface contour are highly needed. For full film lubrication, the lubricant thickness in the clearance of two or more solid boundaries depends on the loads and fluid viscosity. This will be the main factor to determine the bearing lifespan. Generally, in good practice, lubrication is attempted to maintain the oil inside the bearing so that direct contact that inflicts damage can be prevented. In this paper, the properties of the lubricant at high performance operation of journal bearing are studied. Numerous studies have been done in various ways that involve numerical analysis and experimental investigation. Hargreaves (1991) had investigated the influence of sinusoidal surface waviness on the stationary surface of rectangular slider bearings. Hargreaves found that the surface waviness could enhance the load carrying capacity. Theoretical study of the effect of non linear bearing liner was carried out by Hassan (1998) who then agreed with Hargreaves's findings. Hassan reported that wavy surface liner; both axially and circumferentially, would also increase the load carrying capacity. Gertzos (2008) reported that CFD model by using FLUENT package could solve the journal bearing case study which produced results in a very good agreement between theoretical and experimental. Many studies in non linear surface liner have been investigated such as by Dimofte (2002-2008) which concerning axially three sinusoidal wave journal bearing. Moreover, other researchers also investigated the journal bearing with journal groove shape assorted to achieve a good performance, as mentioned by Hirayama *et al* (2009) and Sahu *et al* (2006).

As the fluid film lubrication for the lubrication between two solids is considered, it is important to understand the influence of the surface topography of these solids on several important parameters in the mechanism of lubrication in order to obtain a journal performance with high performance. Based on the previously-mentioned findings, the aim of this paper is to develop three dimensional CFD models so that the performance characteristic of semi circular wavy groove journal bearings can be studied. The usage of palm oil lubricants are also included and studied.

2.0 NUMERICAL APPROACH

The presented numerical study here was conducted with the commercial Computational Fluid Dynamics (CFD) code FLUENT. The CFD procedures include the approximation of grid generation technique, turbulence model, boundary conditions, solver scheme and the selection of turbulence model. The software was used to compute the full field pressure distribution and the shear stress information for characteristics inside the journal bearing with semi circular wavy surface conditions. Comparisons were made for smooth surface liner in same conditions.

The first step of simulation was pre-processing, which involved the building of the journal bearing model in Gambit. Then finite-volume-based mesh was applied and all the required boundary conditions and flow parameters were imposed. The numerical model was then ready to be calculated and produced results. The final step in analysis was post-processing, which involves organization and interpretation of the data.

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2.1 Computational Meshes

Proper selection of the computational grid is an important aspect in any computational simulation. A suitable grid will provide accurate solution and fast convergence. In this study, geometry and mesh generation are done by using Gambit. For three dimensional (3D) domains, tetrahedral and hexahedral grid types are accepted in Gambit. Tetrahedral and interval size method meshes are chosen to be the type of mesh with 60° of equiangular skewness, which then it will provide smoother meshing that eliminates the highly skewness. In order to obtain a fine grid and reduce the time converging consumption, tetrahedral cells from 223000 up to 320000 of cells are created in the domain.

2.2 Flow Modeling

Turbulence flow is one of the most difficult studies of flow in fluid mechanics. Turbulence take place in eddies in the order of millimetre of size, while the whole flow domain may extend over meters or kilometres. In general, the flow on wavy surface is turbulent. Solving turbulence flow in FLUENT can be divided into two categories as follows:

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho}{\partial x_i}(\rho u_i)=S_m \qquad (1)$$

Equation (1) is the general form of mass conservation equation and it is valid for incompressible as well as compressible flows. The source S_m is the mass added to the continuous phase from the dispersed second phase (e.g., due to vaporization of liquid droplets) and any user-defined sources. Momentum equation is described by:

$$\frac{\partial}{\partial t}(\rho u_i)+\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i}+\frac{\partial \tau_{ij}}{\partial x_j}+\rho g_i+F_i \qquad (2)$$

where p is the static pressure and ρg_i and F_i are gravitational and external body forces (e.g., forces that arise from interaction with the dispersed phase), respectively.

Two alternative methods can be employed to render the Navier-Stokes equations tractable so that the small-scale turbulent fluctuations do not have to be directly simulated; Reynolds-averaging (or ensemble-averaging) and filtering. Both methods introduce additional terms in the governing equations that need to be modelled in order to achieve a '*closure'* for the unknowns. The Reynolds-averaged Navier-Stokes (RANS) equations govern the transport of the averaged flow quantities, with the whole range of the scales of turbulence being modelled. The RANS based modelling approach greatly reduces the required of computational effort and resources, and it is widely adopted for practical engineering applications. An entire hierarchy of closure models are available in FLUENT. The RANS equations are often used to compute time-dependent flows, which unsteadiness may be externally imposed or self-sustained. The momentum equation for Reynolds Average Naviers-Stokes has been written as:

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_i}(\rho u_i)=0 \qquad (3)$$

$$\frac{\partial}{\partial t}(\rho u_i)+\frac{\partial}{\partial x_j}(\rho u_i u_j)=-\frac{\partial \rho}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}-\frac{2}{3}\delta_{ij}\frac{\partial u_l}{\partial x_l}\right)\right]+\frac{\partial}{\partial x_j}(-\rho\overline{u_i' u_j'}) \qquad (4)$$

The turbulent kinetic energy, k transport equation for The RNG $k-\varepsilon$ model, is given by:

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left(\alpha_k \mu_{eff}\frac{\partial k}{\partial x_j}\right)+G_k+G_b-\rho\varepsilon-Y_M+S_k \qquad (5)$$

and for the dissipation rate, ε is written by:

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}\left(\alpha_\varepsilon \mu_{eff}\frac{\partial \varepsilon}{\partial x_j}\right)+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon+S_\varepsilon \qquad (6)$$

In these equations, G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients. G_b is the generation of turbulence kinetic energy due to buoyancy. Y_M represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate. The quantities α_k and α_ε are the inverse effective Prandtl numbers for k and ε, respectively. S_k and S_ε are user-defined source terms. The model constants $C_{1\varepsilon}$ and $C_{2\varepsilon}$ have values derived analytically by the RNG theory. These values, used by default in FLUENT, are $C_{1\varepsilon}$ =1.42, $C_{2\varepsilon}$ = 1.68 C_μ = 0.0845.

2.3 Boundary Conditions

The applied boundary conditions in this simulation are common for an incompressible flow. Figure 1 shows a wavy surface with two grooves semi circular wavy. A bearing of 60mm in diameter with its ratio of length to diameter is 0.5, a clearance 250 µm and 200 µm of wave amplitude was developed. To represent the study, a set of typical boundary conditions was used in the present work in 3D test section of the journal bearing simulations. The shaft (journal) rotated counter-clockwise whereas the bearing was stationary. The lubricant circulated in the journal bearing clearance circumferentially and followed the groove wavy surface. The boundary conditions of pressure inlet, pressure outlet and wall were developed to the left end, right end, shaft and bearing respectively.

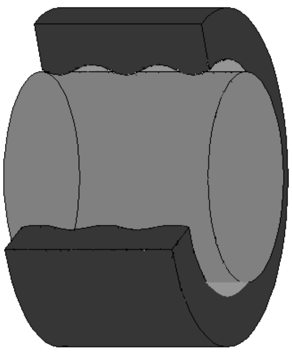


Figure.1: A-3D open view drawing of two circumferential grooves journal bearing.

Pressure inlet boundary conditions are used to define the fluid pressure at the flow inlets, along with all other scalar properties of the flow. Pressure on the left side in practice is an integral part with respect to direct ambient pressure (here are conditioned to be zero, P_{inlet} = 0 gage), so this condition will result in a calculation which is closer to the actual condition.

Pressure outlet of boundary conditions is assigned at the domain outlet. The use of pressure outlet as the boundary conditions instead of an outflow conditions often results in a higher rate of convergence when backflow occurs during iteration. Pressure outlet which is set to zero, P_{outlet} = 0 gage, is equal to the atmospheric pressure which is typically used as a reference pressure measurement.

Wall boundary conditions are used to bound fluid and solid regions. Journal wavy surface and bearing are treated as wall, moving and stationary wall and no-slip boundary conditions are applied.

2.4 Load Capacity

The load capacity is resulted from the pressure generated in the fluid film. The load capacity is a function of load generated pressure distribution around the journal and surface area due to both the rotation and variation in fluid film thickness along circumference. Therefore, the wave journal bearing load capacity depends on the contact surface shape, amplitude and wave number, which investigated the effect of surface shape as shown in Figure 1 as the scope of this study. Load carrying capacity ratios are plotted in the figure and it is calculated by using the following dimensionless equation load:

$$Wcc = \frac{F}{P\,L\,D} \tag{7}$$

where *F, P, L, D* are the load on the journal, ambient pressure, length of bearing, and the diameter of the journal respectively.

Eccentricity (*e*) due to the clearance between the shaft and bearing surface is experienced by both external loads and also shaft weight (W), i.e the distance between the axis of the shaft to the bearing as can be seen in Figure 2 while the radial clearance is the difference between the outer radius of journal and the radius of bearing, c = R - r. Consequently the thickness around the circumference of bearing lubrication oil is not the same. The fluid film has a thin film at one end and a thicker film at the other end. As the shaft is rotating at a certain speed, the position of this layer will change. Load and shaft speed are among others that could affect the eccentricity and also the maximum and minimum thickness of lubricant. In this study, eccentricity ratio, ε is set to be 0.107, 0.284, 0.400, 0.568, and 0.632 respectively and also this ensures the minimum thickness of oil lubrication.

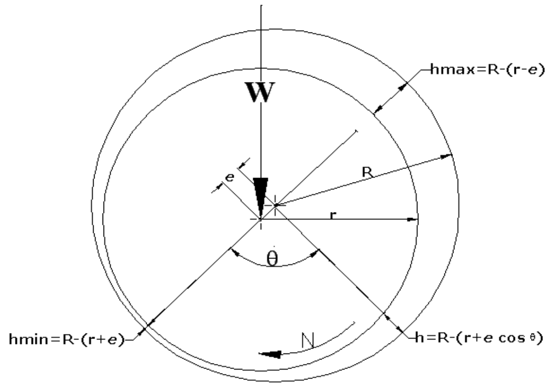
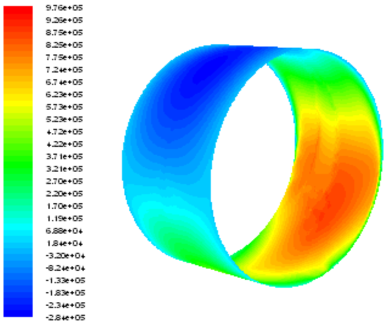


Figure.2: Schematic of eccentricity

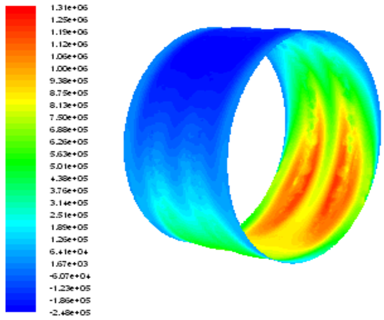
The lubricant properties used in the calculation is based on the palm oil, which subsequently is approached by the nature of the Benzene-Liquid included in FLUENT database where density (ρ) = 887.5 kg/m3, thermal conductivity = 0.1721 w/m-k, heat capacity=1861 kj/kg-K, and viscosity (µ) = 0.07719 kg/m-s.

3.0 RESULTS AND DISCUSSIONS

Figure 3 shows the pressure distribution in the bearing due to the effect of changing surface liner with semi-circular wavy surface. It is shown clearly here that the presence of wavy surface can change the pressure distribution as the vicinity of high pressure becomes more concentrated at the top of the wave. Meanwhile for both bearings, maximum pressure areas are formed at the almost half of the bearings in the direction of shaft rotation.



(a)



(b)

Figure.3: Pressure distribution in the bearing, (a) plain bearing, and (b) semi circular wavy bearing

Then, the effect of the waviness surface shape in dimensionless load carrying capacity in comparison with a linear journal bearing is analyzed. The effects of the shaft speed and eccentricity to the load carrying capacity of the bearing are shown in following Figure 4 to 8. The displayed graphs may provide an explanation that the shaft speed could affect the load capacity, which it is followed by the changes in eccentricity. This provides as an advantage for each changes of the journal bearing inner surface so that the comparison of the characteristics can be obtained. Along with the increasing shaft speed, the characteristics of journal bearings are changed in terms of magnitude load capacity as well as the changes in eccentricity. Plain bearing on a large eccentricity shows a higher performance in load capacity while the wavy bearing seems to be low in performance. This is caused by the fluid pressure that experiences a small clearance around the shaft, as the clearance that serves to support the load on the wavy surface is smaller. Oil can not be forced to be at the peak of the wave so that more are on the valley. The clearance between the surfaces of the shaft with the bearing seems to affect oil pressure. As the semi-circular wavy surface is distinguished by the development of a small groove, it is consequently influenced by the lubrication. By comparing this to the plain bearing, wavy surface seems to have advantage over linear as it increases the load capacity in the small eccentricity. As the shaft speed increases, the characteristics of journal bearings show varied changes in term of magnitude of load capacity and the eccentricity ratio. From these figures, semi circular wavy with the highest value of eccentricity ratio produces the lowest value of load capacity while this event shows contrary in the results of linear journal bearing which it is caused by the development of groove. The trend of curves between semi circular wavy with the plain journal bearing shows a relatively similar behaviour in term of shape.

Best performance in terms of the increasing in load capacity can be achieved by a circular wavy with the eccentricity ratio, $\varepsilon = 0.4$, which if it is accompanied by an increasing in speed of shaft rotations. From these figures, an obvious event of the increasing in load capacity with a small gradient occurs. This is caused by the changes in fluid pressure around the shaft which it simultaneously acts to support the load. To calculate the amount of pressure, the involved significant parameter is the cross-sectional area where the pressure is experienced, where the surface shape that influences the generated pressure forces is also considered.

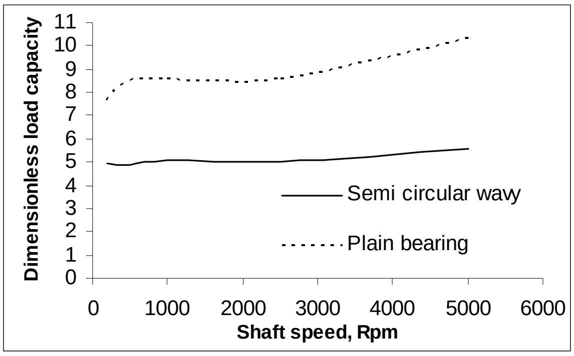


Figure.4: Shaft rotation speed effects to load capacity at eccentricity ratio, $\varepsilon = 0.632$

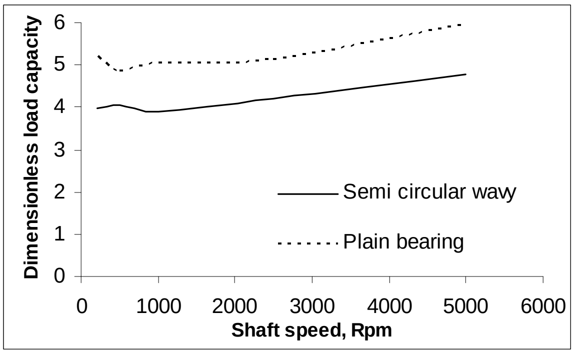


Figure.5: Shaft rotation speed effects to load capacity at eccentricity ratio, $\varepsilon = 0.568$

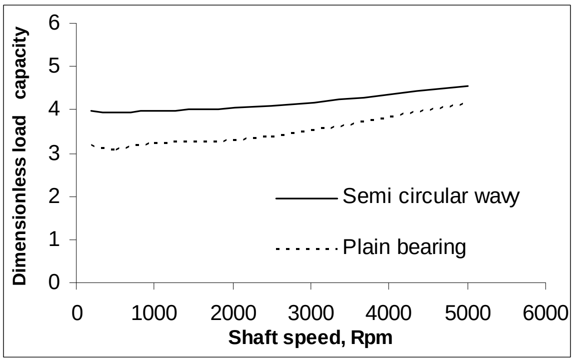


Figure.6: Shaft rotation speed effects to load capacity at eccentricity ratio, $\varepsilon = 0.4$

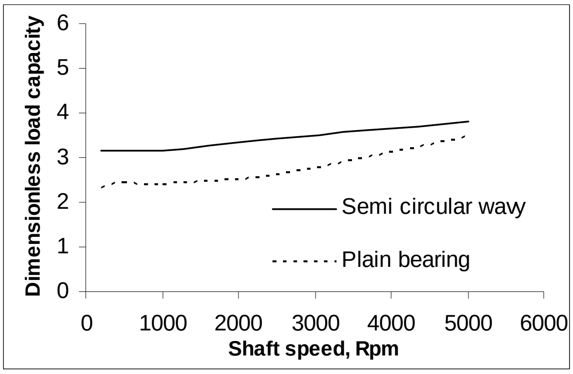


Figure.7: Shaft rotation speed effects to load capacity at eccentricity ratio, $\varepsilon = 0.284$

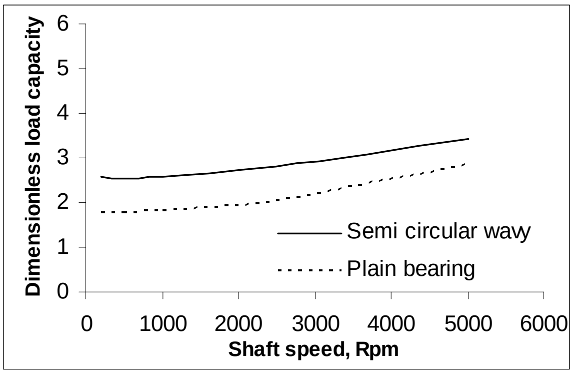


Figure.8: Shaft rotation speed effects to load capacity at eccentricity ratio, $\varepsilon = 0.107$

Drop off in bearing temperature is another function of a lubricant. Lubricant in the bearing is said to work properly when it can reach all parts of the bearing surface, heat can be transferred to the lubricant and subsequently discharged into the ambient. Percentage of temperatures drop which is shown here are plotted from the average temperature of the bearing surface as the thermal equilibrium is achieved. For this study, figure 9 shows the percentage of drop off surface temperature of the bearing on a rate of change in shaft speed at the highest eccentricity. In a large eccentricity, or in a condition where the gap between the bearing and shaft is small, the generated heat becomes high due to the friction. It is required to use a lubricant with properties consist of good thermal conductivity to transfer this heat. From these results, it can be seen that palm oil is able to function properly. Also, it can be seen here that the effect of bearing surface shape as the surface with a semi-circular wavy has a better performance, compared to the plain bearing. This event is caused by the wave bearing surface area as it is larger than the plain bearing in which the heat transfer rate is a function of the area, within a large area of heat transfer rate is increased. For addition, semi-circular wave bearing has much amount of mass lubricant in bearing.

The characteristics of curve change when the eccentricity ratio reduces. As shown in Figure 10, when the shaft rotation reaches about 250 rpm, the temperature drop appears to be the same on both bearings. This is due to the lubricant functions at low speed can not spread uniformly throughout the surface and absorbs heat from the bearings. Along with the increasing in shaft speed, a trend of temperature drop increases for the wave bearing. Significant difference in temperature drop is seen in Figure 11 with the eccentricity ratio is 0.4 where the wave bearing achieves a better performance, compared to plain bearing. Step-up of shaft speed in both types of bearings indicates that the changes in temperature drop are increased. Figure 12 shows an increasing of temperature drop on plain bearing with a small growth on the rotation axis, while the wave bearing equipment shows a significant percentage of increment. Generally, as the shaft speed increases and the eccentricity changes, the wave bearing shows a significant event of temperature drop, compared to the plain bearing.

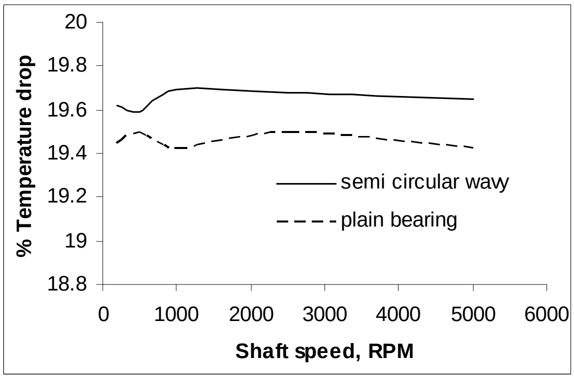


Figure.9: Shaft rotation speed effects to temperature drop at eccentricity ratio, $\varepsilon = 0.632$

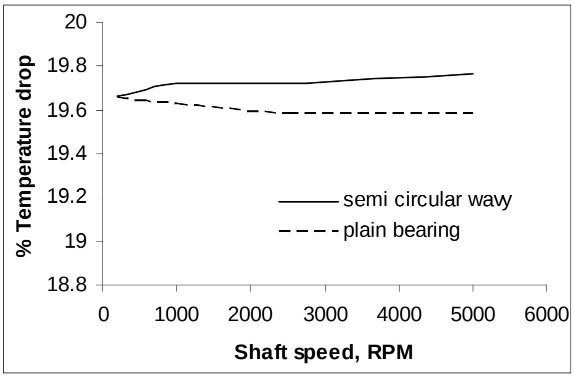


Figure.10: Shaft rotation speed effects to temperature drop at eccentricity ratio, $\varepsilon = 0.568$

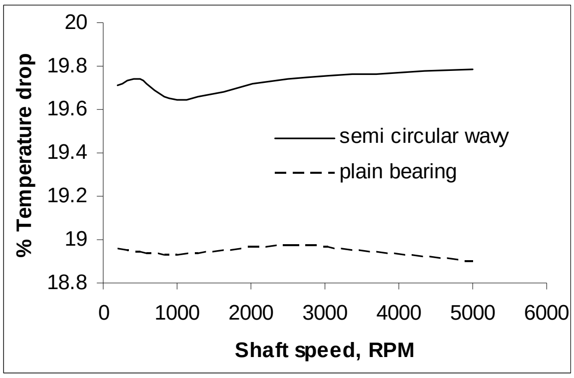


Figure.11: Shaft rotation speed effects to temperature drop at eccentricity ratio, $\varepsilon = 0.400$

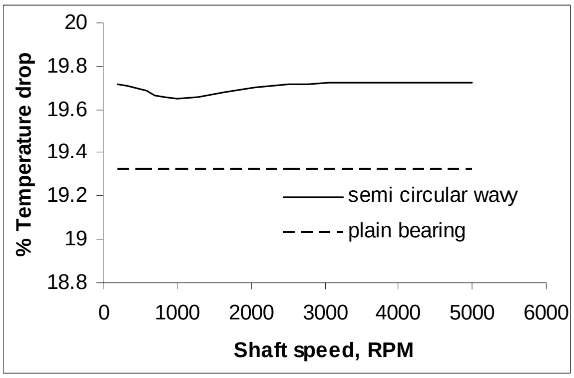


Figure.12: Shaft rotation speed effects to temperature drop at eccentricity ratio, $\varepsilon = 0.284$

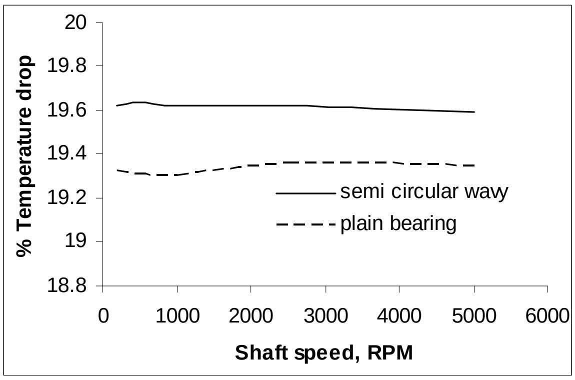


Figure.13: Shaft rotation speed effects to temperature drop at eccentricity ratio, $\varepsilon = 0.107$

4. 0 CONCLUSIONS

In this study, it can be concluded that the presence of surface waviness leads to an increment in bearing load carrying capacity with better results are achieved in semi-circular wavy where improvement remains stable even with the increasing in shaft speed in small eccentricity. Also, a conclusion can be made that the performance of liner journal bearings is generally lower than any other value growth in terms of temperature drop for each changes in shaft speed and eccentricity. Finally, it also can be concluded that palm oil can be used as lubricant.

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