

## THEORETICAL CHARACTERISTICS OF JOURNAL BEARINGS WITH NON-NEWTONIAN PALM OILS

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### Abstract

This paper presents the investigation of the static and dynamic characteristics of hydrodynamic short journal bearings lubricated with non-Newtonian palm-based oils and operated at high speed. The palm oil was mixed with 1% ZDTP to reduce oxidation and increase lubricity. In this study, the power law model is proposed as a rheology model of palm-based oil and the relationship between the shear stress and shear rate was obtained experimentally. The time-dependent modified Reynolds equation and adiabatic energy equation were formulated for infinitely short full circular bearings. Finite difference method was used to calculate both the static and dynamic characteristics of the bearings such as pressure distribution, temperature distribution, attitude angle, stiffness and damping coefficients. The results shows that the palm-based oils have potential to use as a biodegradable lubricant.

### 1. Introduction

During the past three decades, many researchers have investigated the characteristic of bearings with non-Newtonian lubrication. Among the non-Newtonian models proposed in the literature, the power law model has been implemented in many applications. In 1983, I.K.Dien and H.G.Elrod [2] derived the generalized steady-state Reynolds equation for non-Newtonian represented by the power law rheology model and later in 1988, JIIN-YUH Jang and CHONG-CHING Chang [3] analyzed the adiabatic solutions for a finite width hydrodynamic journal bearing. In this paper, the power law model of the Pseudoplastic palm-based oil were obtained experimentally. The static and dynamic characteristics of journal bearings with palm-based oils mixed 1.0% ZDTP was analyzed the thermal effects. I.K.Dien theory was adopted to formulate the dynamically Reynolds equation and the adiabatic energy equation. Both equations were simultaneously solved using finite difference method in order to

clarify the static and dynamic behavior such as stiffness and damping coefficients of journal bearings lubricated with palm-based oils.

### 2. Theory

The relationship between shear stress and shear rate for the non-Newtonian fluid can be represented by power law model as

$$\tau_{zx} = m \left\{ \left( \frac{\partial u}{\partial z} \right)^2 + \left( \frac{\partial v}{\partial z} \right)^2 \right\}^{\frac{n-1}{2}} \left( \frac{\partial u}{\partial z} \right) = m I^{\frac{n-1}{2}} \left( \frac{\partial u}{\partial z} \right) \quad (1)$$

$$\tau_{zy} = m \left\{ \left( \frac{\partial u}{\partial z} \right)^2 + \left( \frac{\partial v}{\partial z} \right)^2 \right\}^{\frac{n-1}{2}} \left( \frac{\partial v}{\partial z} \right) = m I^{\frac{n-1}{2}} \left( \frac{\partial v}{\partial z} \right) \quad (2)$$

Where the parameters  $m$  and  $n$  are determined experimentally. Adopting the perturbation technique in terms of  $\delta$ , the velocity component  $u$  and  $v$  can be expanded as

$$u = u^* + \delta u' + \dots \quad \text{and} \quad v = v^* + \delta v' + \dots \quad (3)$$

$$\text{Then} \quad I = I^* + \delta I' + \dots \quad \text{and} \quad \mu = \mu^* + \delta \mu' + \dots \quad (4)$$

$$\text{Where} \quad u^* = \frac{z}{h} U_J \quad (5)$$

$$v^* = 0 \quad (6)$$

$$\delta u' = \frac{z(z-h)}{2\mu^* n} \frac{\partial p}{\partial x} \quad (7)$$

$$\delta v' = \frac{z(z-h)}{2\mu^*} \frac{\partial p}{\partial y} \quad (8)$$

$$\mu^* = m \left\{ \frac{u_J}{h} \right\}^{(n-1)} \quad (9)$$

$$\delta \mu' = 2\mu^* \left\{ \left( \frac{2z-h}{2\mu^* n} \right) \left( \frac{n-1}{2} \right) \left( \frac{h}{U_J} \right) \frac{\partial p}{\partial x} \right\} \quad (10)$$

## 2.1 Time dependent Reynolds equation

The integral form of continuity equation is

$$\int_0^h \left[ \frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} \right] dz = 0$$

Integrating the continuity equation can be written as

$$\begin{aligned} h \frac{\partial \rho}{\partial t} - \rho U_J \frac{\partial h}{\partial x} + \frac{\partial}{\partial x} \left( \rho \int_0^h u dz \right) \\ - \rho V_J \frac{\partial h}{\partial y} + \frac{\partial}{\partial y} \left( \rho \int_0^h v dz \right) + \rho(W_J - W_B) = 0 \end{aligned} \quad (11)$$

Using equations (3),(5),(6),(7) and (8), the Reynolds equations for infinitely short bearing in cylindrical polar coordinate can be written as.

$$\frac{\partial}{\partial y} \left\{ \frac{h^3}{12\mu^*} \frac{\partial p}{\partial y} \right\} = \frac{\omega_J}{2} \frac{\partial h}{\partial \theta} + \frac{\partial h}{\partial t} \quad (12)$$

The film thickness is

$$h = c\bar{h} = c(1 + \varepsilon \cos \theta) \quad (13)$$

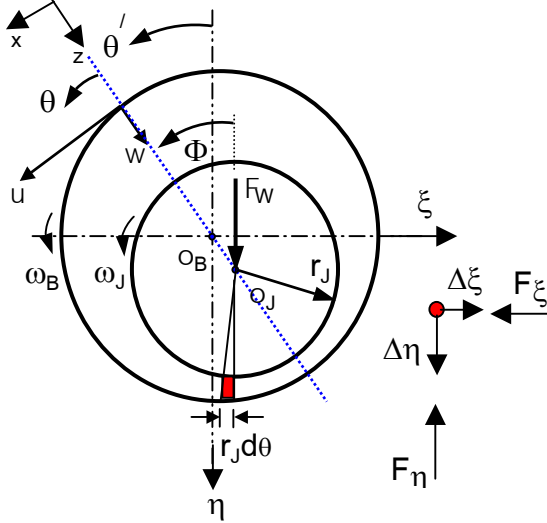


Fig. 1. Journal bearing geometry

The coordinates  $\Phi$ , and  $\theta'$  is measured from the negative  $\eta$  axis, and  $\theta$  is measured from the line pass through the center of bearing as well as center of the journal as shown in Fig.1. The angular distance  $\theta$  relates to the attitude angle  $\Phi$  and  $\theta'$  as  $\theta = \theta' - \Phi$ . A first-order expansion of the pressure can be expressed as.

$$p = (p)_s + \left( \frac{\partial p}{\partial \xi} \right)_s \Delta \xi + \left( \frac{\partial p}{\partial \eta} \right)_s \Delta \eta + \left( \frac{\partial p}{\partial \xi} \right)_s \Delta \xi + \left( \frac{\partial p}{\partial \eta} \right)_s \Delta \eta \quad (14)$$

Where subscript "s" refer to the steady-state position. By using the perturbation technique as J.W.Lund theory [4], five non-dimensional equations can be obtained as

$$\left[ \frac{\bar{h}_s^3}{\lambda^2} \frac{\partial}{\partial \bar{y}} \left( \frac{1}{12\bar{\mu}} \frac{\partial}{\partial \bar{y}} \right) \right] \begin{Bmatrix} \bar{p}_s \\ \bar{p}_\xi \\ \bar{p}_\eta \\ \bar{p}_{\dot{\xi}} \\ \bar{p}_{\dot{\eta}} \end{Bmatrix} = \begin{Bmatrix} \frac{1}{2} \frac{\partial \bar{h}_s}{\partial \theta'} \\ \frac{1}{2} \left( \cos \theta' - \frac{3 \sin \theta'}{\bar{h}_s} \frac{\partial \bar{h}_s}{\partial \theta'} \right) \\ -\frac{1}{2} \left( \sin \theta' + \frac{3 \cos \theta'}{\bar{h}_s} \frac{\partial \bar{h}_s}{\partial \theta'} \right) \\ \sin \theta' \\ \cos \theta' \end{Bmatrix} \quad (15)$$

$$\text{where } p_s = (p)_s, \quad p_\xi = \left( \frac{\partial p}{\partial \xi} \right)_s, \quad p_\eta = \left( \frac{\partial p}{\partial \eta} \right)_s,$$

$$p_{\dot{\xi}} = \left( \frac{\partial p}{\partial \dot{\xi}} \right)_s, \quad p_{\dot{\eta}} = \left( \frac{\partial p}{\partial \dot{\eta}} \right)_s$$

## 2.2 The adiabatic energy equation

Under the adiabatic assumption and neglecting the temperature variation across the film thickness, the two dimensional energy equation for an incompressible fluid with laminar flow can be written as

$$\begin{aligned} \rho c_v \left( u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) &= \mu \left\{ \left( \frac{\partial u}{\partial z} \right)^2 + \left( \frac{\partial v}{\partial z} \right)^2 \right\} \\ &= \tau_{zx} \frac{\partial u}{\partial z} + \tau_{zy} \frac{\partial v}{\partial z} \end{aligned} \quad (16)$$

The film thickness is much smaller than other sizes of bearing, the integration of equation (16) can be approximated using the mean velocities  $u_m$  and the mean temperature  $T_m$ , and written as

$$\begin{aligned} \rho c_v \left( \int_0^h u dz \frac{\partial T_m}{\partial x} + \int_0^h v dz \frac{\partial T_m}{\partial y} \right) \\ = U_J \tau_{zx} \Big|_{z=h} - \frac{\partial p}{\partial x} \int_0^h u dz - \frac{\partial p}{\partial y} \int_0^h v dz \end{aligned} \quad (17)$$

Applying perturbation method, the shear stress can be written as following

$$\tau_{zx} = \left( \mu^* \frac{U_J}{h} \right) + \left( \frac{2z - h}{2} \frac{\partial p}{\partial x} \right) \quad (18)$$

Substituting equations (5),(6),(7),(8) and (18) into (17), The dimensionless adiabatic energy equation for short full journal bearing can be written as

$$6\bar{h} \frac{\partial \bar{T}_m}{\partial \theta} - \left( \frac{1}{\lambda^2} \frac{\bar{h}^3}{\bar{\mu}} \frac{\partial \bar{p}}{\partial y} \right) \frac{\partial \bar{T}_m}{\partial y} = \frac{12\bar{\mu}}{\bar{h}} + \frac{1}{\lambda^2} \frac{\bar{h}^3}{\bar{\mu}} \left( \frac{\partial \bar{p}}{\partial y} \right)^2 \quad (19)$$

The  $\bar{\mu}$  is function of the mean shear rate and temperature.

$$\bar{\mu} = \frac{\mu^*}{\mu_{ref}} = \frac{\mu_0^*}{\mu_{ref}} e^{-\beta \bar{T}_m} = \frac{m_0}{\mu_{ref}} \left( \frac{U_J}{h} \right)^{n-1} e^{-\beta \bar{T}_m} \quad (20)$$

### 2.3 Load capacity, Friction force and Dynamic characteristics.

The force due to the hydrodynamic pressure on the journal in the  $\eta - \xi$  coordinate system are calculated as

$$\bar{F}_\eta = - \int_0^1 \int_0^{2\pi} \bar{p}_s \cos \theta' d\theta' d\bar{y} \quad (21)$$

$$\bar{F}_\xi = - \int_0^1 \int_0^{2\pi} \bar{p}_s \sin \theta' d\theta' d\bar{y} \quad (22)$$

The friction forces in fluid film is obtained by integrating the shear stress on journal surface. Using equation (18) where  $z = h$ , the non-dimensional friction force is

$$\frac{r}{c} \bar{F}_{fric} = \int_0^1 \int_0^{2\pi} \frac{\bar{\mu}}{h} d\theta d\bar{y} + \int_0^1 \int_0^{2\pi} \frac{\bar{h}}{2} \frac{\partial \bar{p}}{\partial \theta} d\theta d\bar{y} \quad (23)$$

The non-dimension spring and damping coefficients can be obtained as followings

$$\begin{Bmatrix} \bar{k}_{\xi\xi} & \bar{k}_{\xi\eta} \\ \bar{k}_{\eta\xi} & \bar{k}_{\eta\eta} \end{Bmatrix} = \begin{Bmatrix} - \int_0^1 \int_0^{2\pi} \bar{p}_\xi \sin \theta' d\theta' d\bar{y} & - \int_0^1 \int_0^{2\pi} \bar{p}_\eta \sin \theta' d\theta' d\bar{y} \\ - \int_0^1 \int_0^{2\pi} \bar{p}_\xi \cos \theta' d\theta' d\bar{y} & - \int_0^1 \int_0^{2\pi} \bar{p}_\eta \cos \theta' d\theta' d\bar{y} \end{Bmatrix} \quad (24)$$

$$\begin{Bmatrix} \bar{b}_{\xi\xi} & \bar{b}_{\xi\eta} \\ \bar{b}_{\eta\xi} & \bar{b}_{\eta\eta} \end{Bmatrix} = \begin{Bmatrix} - \int_0^1 \int_0^{2\pi} \bar{p}_\xi \sin \theta' d\theta' d\bar{y} & - \int_0^1 \int_0^{2\pi} \bar{p}_\eta \sin \theta' d\theta' d\bar{y} \\ - \int_0^1 \int_0^{2\pi} \bar{p}_\xi \cos \theta' d\theta' d\bar{y} & - \int_0^1 \int_0^{2\pi} \bar{p}_\eta \cos \theta' d\theta' d\bar{y} \end{Bmatrix} \quad (25)$$

### 3. Results and discussions

The numerical results of the static and dynamic characteristics of journal bearings lubricated with palm oils are presented for palm oil with 0% and 1.0% ZDTP respectively. In this simulation, the journal bearings have length to diameter equal to 0.4 and radial clearance ratio  $c/r = 0.002$ . The journal operated at 2500 rpm. The Pseudoplastic palm based-oils were implemented using power law model. The flow parameters are show in table 1

Table 1 Bearing and palm oil with ZDTP properties

Parameter	Value		
Journal radius, $r_j$	50	mm	
Radius clearance, $c$	0.10	mm	
Velocity of journal, $N$	2500	rpm	
Bearing length, $L$	40	mm	
Reference viscosity, $\mu_{ref}$	1.2339	Pa s	
Inlet lubricant temperature, $T_0$	40	°C	
Parameter	Palm oil	mixed 1.0% ZDTP	
power-law exponent, $n$	0.5660	0.6072	
viscosity consistency at $T_0$ , $m_0$	1.2339	0.8340	Pa s <sup>n</sup>
Lubricant density, $\rho$	929.3	903.4	kg/m <sup>3</sup>
Thermoviscosity coefficient, $\beta$	0.02824	0.02925	°C <sup>-1</sup>
Lubricant specific heat, $c_v$	2037.9	2013.0	J kg <sup>-1</sup> °C <sup>-1</sup>

Oil film pressure distribution along circumferential and axial directions of the journal bearing for eccentricity ratio = 0.6, 0.80, 0.90 and 0.95 are shown in Fig 2. The oil film temperature distribution along the circumferential at bearing midplane is shown in Fig. 3. Pressure and temperature increase rapidly at severe operating condition  $\varepsilon = 0.95$  for palm-based oils with 1% ZDTP.

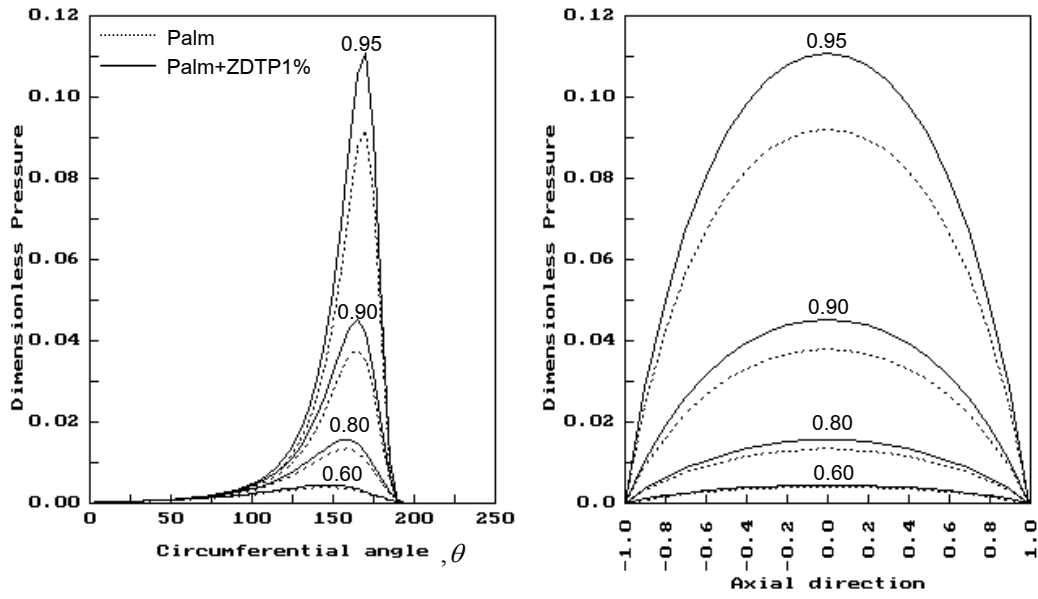


Fig. 2. Pressure distribution along the circumferential and axial direction at bearing midplane

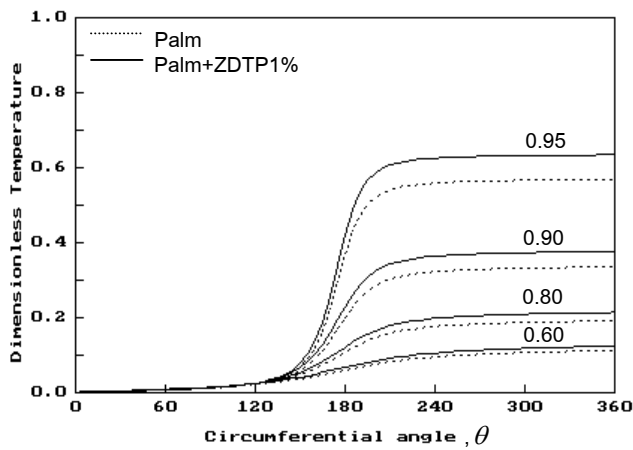


Fig. 3. Temperature distribution

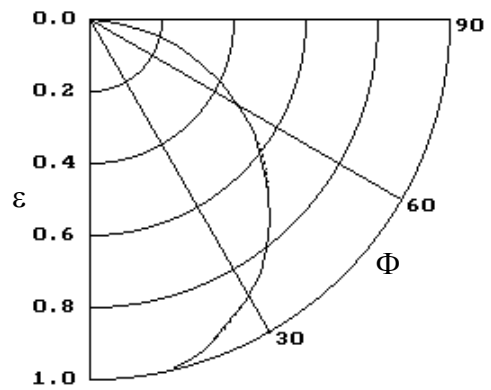


Fig. 4. Attitude angle and eccentricity ratio

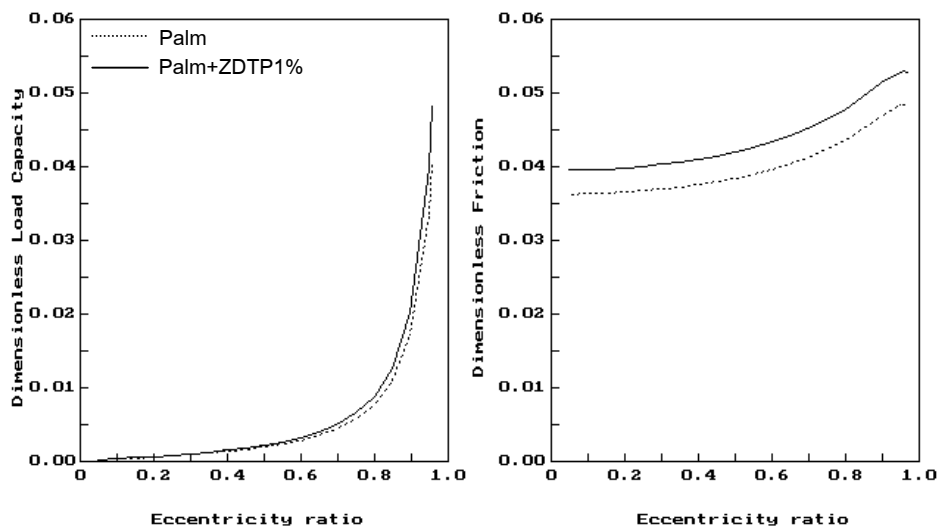


Fig. 5. Load capacity and Friction force

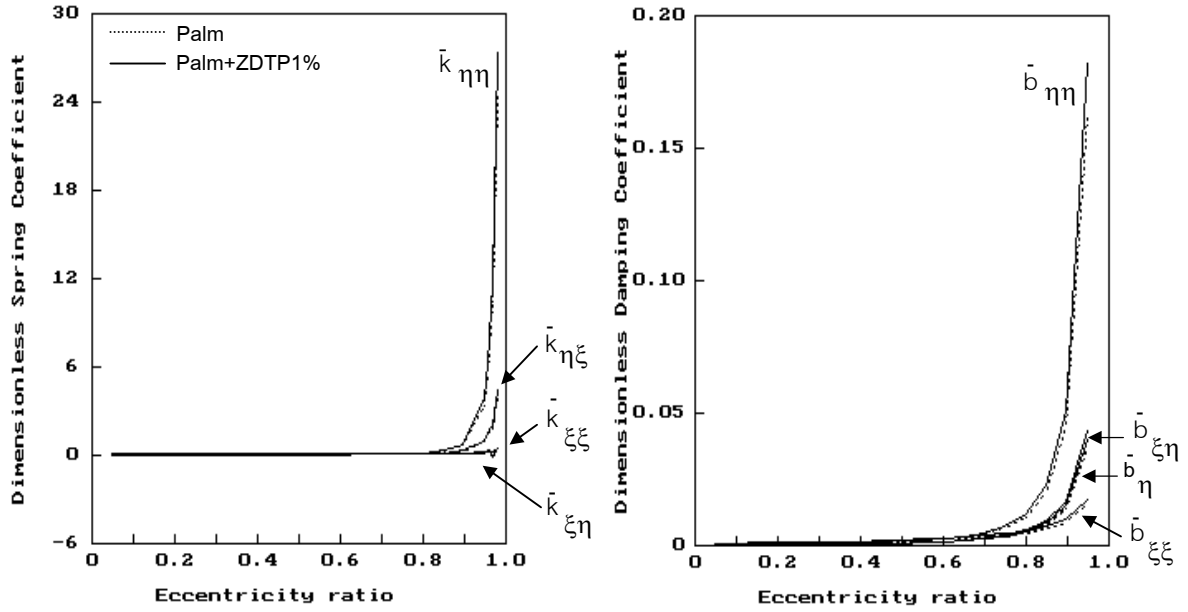


Fig. 6. Spring and Damping coefficients

Fig. 5. shows the dimensionless load capacity and dimensionless friction force at varying eccentricity ratio. The load capacity and friction force increase with the increase in the value of eccentricity ratio. Attitude angle also increase with the increase in eccentricity ratio as shown in Fig. 4.

The variation of dimensionless spring coefficients  $\bar{k}_{\xi\xi}$ ,  $\bar{k}_{\eta\eta}$ ,  $\bar{k}_{\xi\eta}$  and  $\bar{k}_{\eta\xi}$  and the variation of dimensionless damping coefficients  $\bar{b}_{\xi\xi}$ ,  $\bar{b}_{\eta\eta}$ ,  $\bar{b}_{\xi\eta}$  and  $\bar{b}_{\eta\xi}$  are shown in Fig 6. All spring and damping coefficients are nearly zero for  $\varepsilon \leq 0.85$  but at severe operation conditions  $\varepsilon > 0.85$  the spring and damping coefficients increase rapidly due to the pressure is the significant increase in film pressure.

#### 4. Conclusions

The static and dynamic characteristics of journal bearings lubricated with palm based oils were examined theoretically and can be concluded as

- 1) Power law model of the palm-based oils were obtained experimentally as Pseudoplastic fluid.
- 2) For low eccentricity ratio, ZDTP has little effect on the film pressure and temperature. For high eccentricity ratio, the percent of ZDTP has significant effect on the film pressure and temperature.
- 3) Bearing may be stable when operate at  $\varepsilon > 0.85$ . The stability of bearing depends on the direct spring coefficient  $\bar{k}_{\eta\eta}$ ,

cross couple spring coefficient  $\bar{k}_{\eta\xi}$ , direct damping coefficient  $\bar{b}_{\eta\eta}$ , and cross couple damping coefficients  $\bar{b}_{\xi\eta}$  and  $\bar{b}_{\eta\xi}$ .

#### 5. Nomenclature

- $F$  = fluid force, (N)  
 $F_{fric}$  = friction force, (N)  
 $F_W$  = rotor weight, (N)  
 $L$  = journal bearing length along the axial axis, (m)  
 $N$  = rotational speed, (rpm)  
 $O$  = center of bearing or journal  
 $S$  = Sommerfeld number  
 $T$  = temperature,  $^{\circ}\text{C}$   
 $T_m$  = mean temperature of lubricant across the film,  
 $T_m = \frac{1}{h} \int_0^h T dz$ , ( $^{\circ}\text{C}$ )  
 $U$  = tangential velocity of surface, (m/s)  
 $V$  = velocity of surface in axial direction, (m/s)  
 $W$  = velocity of surface across fluid film direction, (m/s)  
 $b_{\eta\eta}, \dots, b_{\xi\xi}$  = damping coefficient, (N-s/m)  
 $c$  = bearing radial clearance, (m)  
 $c_v$  = specific heat, ( $\text{J kg}^{-1}\text{C}^{-1}$ )  
 $e$  = eccentricity of journal bearing, (m)  
 $h$  = oil film thickness, (m)  
 $k_{\eta\eta}, \dots, k_{\xi\xi}$  = stiffness coefficient, (N/m)  
 $m$  = fluid viscosity consistency coefficient at temperature  $T$ ,  $\text{Pa s}^n$

$m_0$  = fluid viscosity consistency coefficient at reference temperature, Pa s<sup>n</sup>  
 $n$  = power-law exponent or flow behavior index  
 $p$  = fluid film pressure, Pa  
 $r$  = radius, (m)  
 $t$  = time, (s)  
 $u$  = velocity of lubricant in x-direction, (m/s)  
 $u_m$  = mean velocities ,  $u_m = \frac{1}{h} \int_0^h u dz$  , (m/s)  
 $v$  = velocity of lubricant in y-direction, (m/s)  
 $w$  = velocity of lubricant in z-direction, (m/s)  
 $x$  = coordinate in circumferential direction  
 $y$  = coordinate in axial direction  
 $z$  = coordinate across fluid film direction  
 $\eta$  = coordinate in vertical direction  
 $\xi$  = coordinate in horizontal direction  
 $\delta$  = a small, nondimensional amplitude parameter for expansion  
 $\omega$  = angular velocity, (rad/s)  
 $\varepsilon$  = eccentricity ratio of journal bearing  
 $\mu$  = apparent viscosity, (Pa s)  
 $\mu^*$  = viscosity at mean shear rate, (Pa s)  
 $\theta$  = circumferential angle measured from the line of centers, (rad)  
 $\theta'$  = circumferential angle measured from the  $\eta$  axis, (rad)  
 $\Phi$  = attitude angle, (rad)  
 $\beta$  = viscosity – temperature index, (1/°C)  
 $\rho$  = density of the lubricant, (kg/m<sup>3</sup>)  
 $\tau_{zx}$  = component of shear stress in x-direction, (Pa)  
 $\tau_{zy}$  = component of shear stress in y-direction, (Pa)  
 ZDTP = Zinc Dithiophosphate

#### subscripts

$J$  = journal  
 $s$  = steady-state position  
 $0$  = at reference temperature

#### 6. Dimensionless form

$$p - p_{atm} = \frac{\mu_{ref} U_J r_J}{c^2} \bar{p}$$

$$F_W = \mu_{ref} U_J L \left( \frac{r_J}{c} \right)^2 \bar{F}_W$$

$$T_m - T_0 = \frac{\mu_{ref} U_J r}{\rho c_v c^2} \bar{T}_m$$

$$\beta = \frac{\rho c_v c^2}{\mu_{ref} U_J r_J} \bar{\beta}$$

$$h = c \bar{h}$$

$$y = \frac{L}{2} \bar{y}$$

$$\lambda = \frac{L}{D}$$

$$S = \frac{\mu_{ref} NDL}{F_W} \left( \frac{r_J}{c} \right)^2$$

$$F_{fric} = \mu_{ref} U_J L \left( \frac{r_J}{c} \right)^2 \bar{F}_{fric}$$

$$\bar{\mu} = \frac{\mu^*}{\mu_{ref}}$$

$$p_\xi = \frac{\mu_{ref} U_J r_J}{c^3} \bar{p}_\xi$$

$$p_\eta = \frac{\mu_{ref} U_J r_J}{c^3} \bar{p}_\eta$$

$$p_{\dot{\xi}} = \frac{\mu_{ref} U_J r_J}{c^3 \omega_J} \bar{p}_{\dot{\xi}}$$

$$p_{\dot{\eta}} = \frac{\mu_{ref} U_J r_J}{c^3 \omega_J} \bar{p}_{\dot{\eta}}$$

$$\bar{k} = \frac{c}{\mu_{ref} U_J L \left( \frac{r_J}{c} \right)^2} k$$

$$\bar{b} = \frac{c \omega_J}{\mu_{ref} U_J L \left( \frac{r_J}{c} \right)^2} b$$

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