Experimental Investigation of Hydrodynamic Journal Bearings with Non-Newtonian Soybean-Based Oils

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Abstract

This paper presents the potential performance of environmental friendly vegetable-based oils. In this paper, hydrodynamic journal bearings lubricated with non-Newtonian soybean–based oil were investigated both theoretically and experimentally. The soybean oil is mixed with ZDTP as an anti-oxidant agent. The viscosity – temperature index and the power law model are obtained experimentally. The modified Reynolds and the adiabatic energy equations are calculated numerically. In the experiment, pressure transducers, thermocouples and gap sensors were installed to measure the pressure and temperature profiles in the journal bearings with varying the eccentricity ratio. The experimental results are compared with the numerical calculation for the hydrodynamic journal bearings with length to diameter equal to 0.5 and 1.0 respectively.

Keyword
Hydrodynamic journal bearing, non-Newtonian soybean-based oil, Power law model, Reynolds equation adiabatic energy equation, Finite difference method

1. Introduction

Many research works investigated the lubrication characteristic of bearings with non-Newtonian lubricant for over three decade. Horowitz and his colleague obtained the analytical pressure distribution for a finite width full circular journal bearing by approximating the shear rate with a cubic equation of shear stress. Hashimoto and Wada examined theoretically the combined effects of nonlinear characteristics and fluid film inertia and the dynamic behavior of circular type squeeze film bearing. Hsu has introduced the cubic shear stress law for the non-Newtonian lubricants and has studied the static performance characteristics for infinitely long bearings.

In this paper, the static characteristic of finite journal bearing with Pseudo plastic lubricants is examined theoretically and experimentally included thermal effect.

2. Theory

2.1 Flow Characteristic of Soybean-based oil

In this study, the refined soybean oil was mixed with 0%, 0.2% and 1% ZDTP by weight. The flow characteristic of soybean-based oil was investigated experimentally by using the rotational rheometer according to DIN 53019 standard test method .The soybean based oil has non-Newtonian behavior and can be approximated as Pseudo plastic fluid by using power law model as:

$$\tau = \overline{m}\left(\frac{\partial u}{\partial y}\right)^{n=1} \frac{\partial u}{\partial y} \qquad (1)$$

$$\overline{m} = m_o \exp\left[-\beta\left(T - T_o\right)\right] \qquad (2)$$

Where $\overline{m}$ is the consistency parameter at temperature of the oil $\overline{T}$; m_o is the parameter at inlet oil temperature T_o; n is viscometer constant and β is the viscosity temperature index.

2.2 Modified Reynolds equation

The nondimensional modified Reynolds' equation can be derived as

$$\frac{\partial}{\partial\theta}\left\{\frac{h^{n+2}}{nm}\frac{\partial P}{\partial\theta}+\frac{\partial P}{\partial\theta}\right\}+\left(\frac{D}{L}\right)^2\frac{\partial}{\partial z}\left\{\frac{h^{n+2}}{m}\frac{\partial P}{\partial z}\right\}=6\left(\frac{u}{c}\right)^{n-1}\frac{\partial h}{\partial\theta} \quad (3)$$

The dimensionless film thickness h of circular journal bearings can be written as:

$$h=1+\varepsilon\cos\theta \quad (4)$$

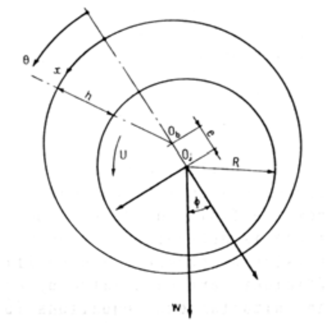

Figure 1 Geometry of circular journal bearing

The Reynolds boundary conditions at inlet and outlet along the axial direction can be expressed as:

$$P(\theta,0)=P(\theta,1)=0 \quad (5)$$

$$P(0,z)=P(\theta_2,z)=\frac{\partial P}{\partial\theta}=0 \quad (6)$$

2.3 The Energy Equation

The adiabatic energy equation is formulated with the assumptions that the heat generated within the film is completely carried away by the lubricant and the temperature variation across the oil film is neglected:

$$\rho c_p\left(u\frac{\partial T}{\partial x}+w\frac{\partial T}{\partial z}\right)=\mu\left[\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right] \quad (7)$$

The energy equation (8) can be written in non-dimensional form as:

$$A\left(\frac{\partial T}{\partial\theta}\right)+B\frac{D}{L}\left(\frac{\partial T}{\partial z}\right)=\alpha\frac{m}{h^{n+1}}\left(E+F\right) \quad (8)$$

where

$$T=\beta(\overline{T}-T_0) \quad (9)$$

$$A=\frac{1}{2}-\frac{1}{12}\left[\frac{h^{n+1}c^{n-1}}{nmu^{n-1}}\left(\frac{\partial P}{\partial\theta}\right)\right] \quad (10)$$

$$B=-\frac{1}{12}\left(\frac{D}{L}\right)^2\left[\frac{h^{n+1}c^{n-1}}{nmu^{n-1}}\left(\frac{\partial P}{\partial z}\right)\right] \quad (11)$$

$$E=\int_0^1\left[1-\frac{1}{2}\frac{h^{n+1}c^{n-1}}{mnu^{n-1}}\frac{\partial P}{\partial\theta}\left(1-2\eta\right)\right]^{n-1}d\eta \quad (12)$$

$$F=\int_0^1\left[1-\frac{1}{2}\frac{h^{n+1}c^{n-1}}{mnu^{n-1}}\frac{\partial P}{\partial\theta}\left(1-2\eta\right)\right]^{n-1}\left[-\frac{1}{2}\left(\frac{D}{L}\right)\left(\frac{h^{n+1}c^{n-1}}{mu^{n-1}}\right)\frac{\partial P}{\partial z}(1-2\eta)\right]^2 d\eta \quad (13)$$

$$\alpha=\frac{\left(2\pi N\right)^n m_o\beta}{\rho c_p}\left(\frac{R}{C}\right)^{n+1} \quad (14)$$

The boundary conditions can be written as:

$$T(0,z)=\frac{dT}{dz}\left(\theta,\frac{1}{2}\right)=0 \quad (15)$$

$$T\left(\theta,0\right)=T\left(\theta,1\right)=0 \quad (16)$$

The dimensionless oil film force components in radial and tangential direction F_ε and F_ϕ are obtained by integrating the dimensionless film pressure as:

$$F_\varepsilon=\frac{C^2\overline{F}_\varepsilon}{\mu R^3L\omega}=-\int_0^1\int_0^{\theta_2}P\cos\theta\,d\theta dz \quad (17)$$

$$F_\varphi=\frac{C^2\overline{F}_\varphi}{\mu R^3L\omega}=\int_0^1\int_0^{\theta_2}P\sin\theta\,d\theta dz \quad (18)$$

Where the dimensionless oil film pressure, P is equal to zero at the position $\theta=\theta_2$

3. Test Equipment

Tests were performed in the test rig developed as shown in Figure 2 to investigate the static characteristic of hydrodynamic journal bearing lubricated with non-Newtonian soybean-based oils. Pressure sensors and thermocouples were calibrated and installed in the circumferential direction on the journal bearing. Two noncontact induction type gap sensors were calibrated and installed in the circumferential direction to measure the oil film thickness in the journal bearing. The test bearings have length to diameter ratios 0.5 and 1 respectively and clearance to journal diameter ratio equal to 0.002. The tests were operated at various loads and various speeds.

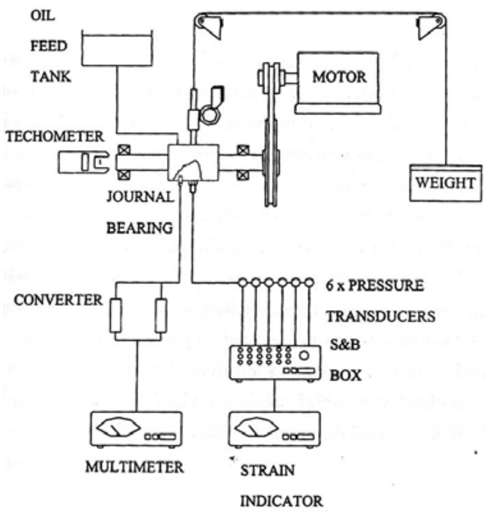

Figure 2 Experimental set up

4. Result

The modified Reynolds and energy equations were solved simultaneously using finite difference technique to obtain the pressure distribution and temperature distribution in the full circular journal bearings. The pressure distribution and temperature distributions from experiments were compared with those from numerical results for bearing length to diameter ratio, 0.5 and 1.0 and clearance to journal diameter ratio; 0.002.The bearings were operated at the eccentricity ratio 0.2, 0.4 and 0.6 respectively as shown in Figure 3 to Figure 14. The oil film pressure from experimental are higher than the pressure calculated theoretically as shown the Figure 3, 5, 7,9,11 and 13. Similarly, the oil film temperatures from experiments are lower than that compared with the theoretical calculation as shown in Figure 4,6,8,10,12 and 14 respectively. The reason is that, the bearings were set up and operated with non-adiabatic conditions. The significant effect of additive ZDTP on oil film pressure and temperature are shown in Figure 3 to Figure 14. The oil film pressures slightly increase as the percentage of ZDTP increase but the oil film temperatures greatly decrease as the percentage of ZDTP increase. The load capacity of journal bearings lubricated with non-Newtonian soybean-based oil increase rapidly at severe operating conditions as shown in Figure 15. Percentage of ZDTP mixed in the lubricant have significant effects on journal bearing load capacity.

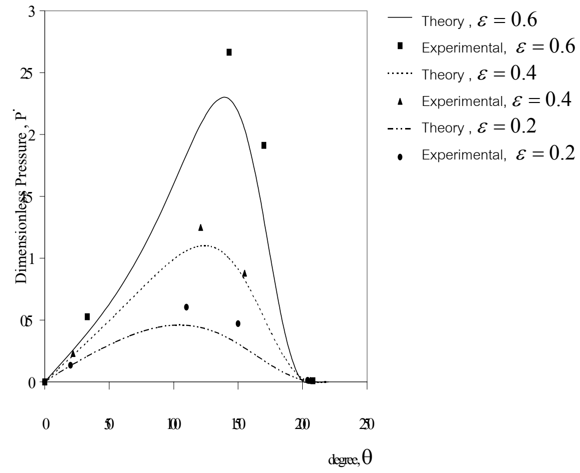

Figure 3 Pressure profile of journal bearing lubricated with pure soybean oil for clearance 0.1mm, speed 200 rpm. and L/D = 1

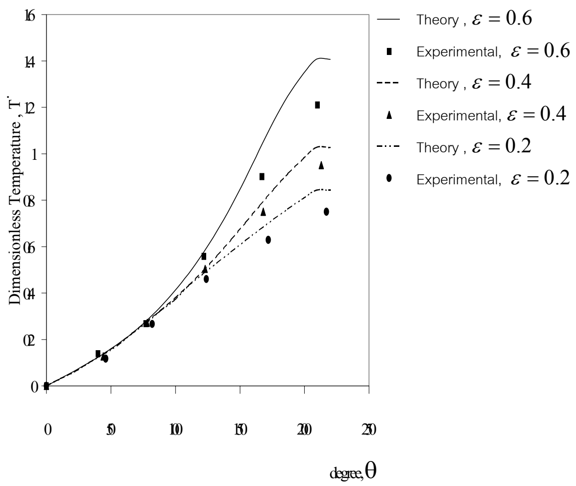

Figure 4 Temperature profile of journal bearing lubricated with pure soybean oil for clearance 0.1mm, speed 200 rpm and L/D=1

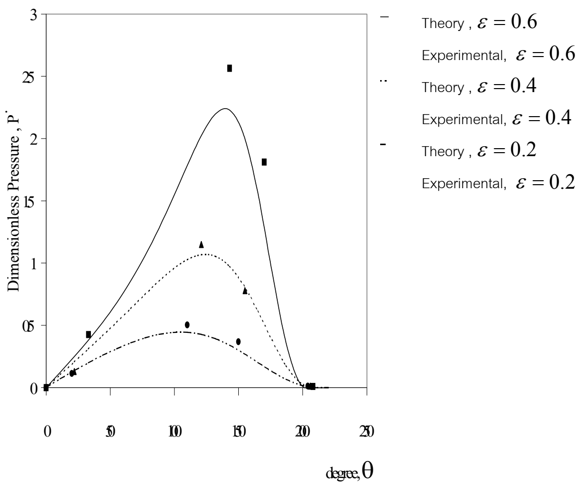

Figure 5 Pressure profile of journal bearing lubricated with soybean oil and 0.2% ZDTP for clearance 0.1mm, speed 200 rpm .and L/D = 1

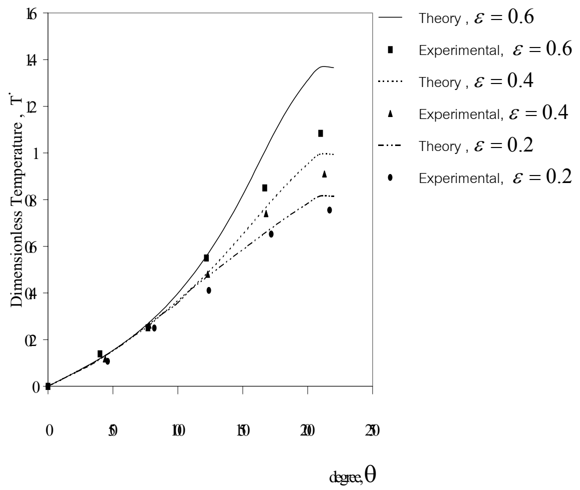

Figure 6 Temperature profile for soybean oil and 0.2% ZDTP clearance 0.1mm, speed 200 rpm. and L/D = 1

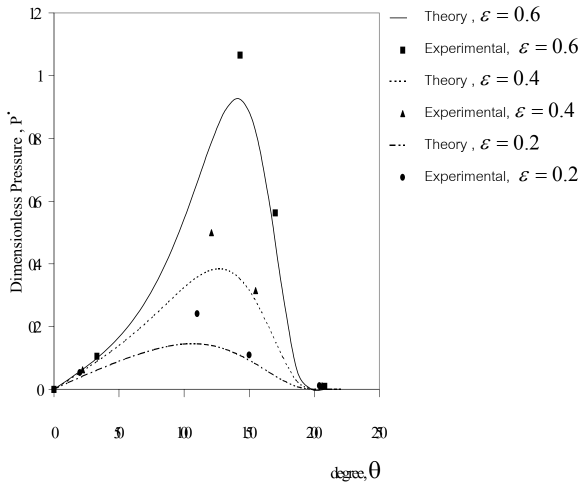

Figure 9 Pressure profile for pure soybean oil clearance 0.1 mm, speed 200 rpm. and L/D = 0.5

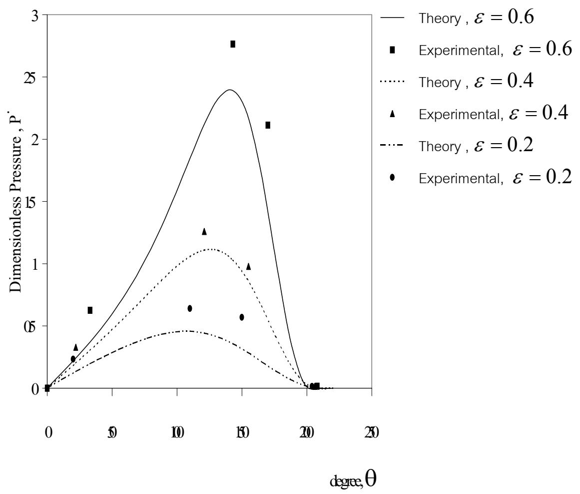

Figure 7 Pressure profile for soybean oil and % ZDTP clearance 0.1mm, speed 200 rpm. and L/D = 1

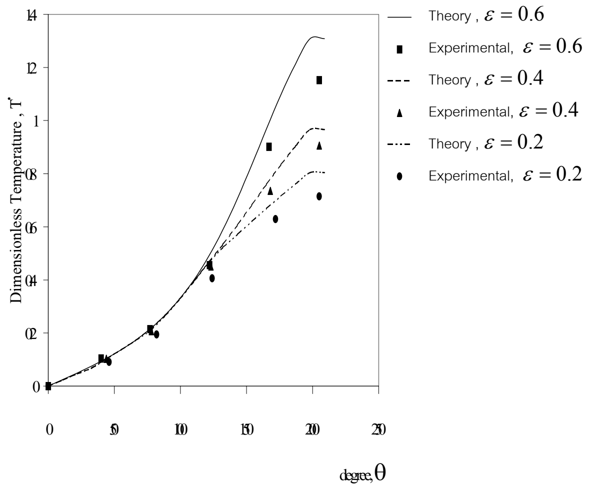

Figure 10 Temperature profile for pure soybean oil clearance 0.1mm, speed 200 rpm. and L/D = 0.5

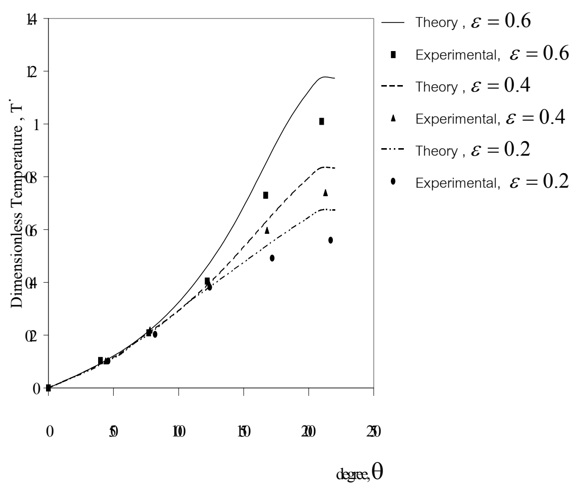

Figure 8 Temperature profile for soybean oil and 1 % ZDTP clearance 0.1mm, speed 200 rpm. and L/D = 1

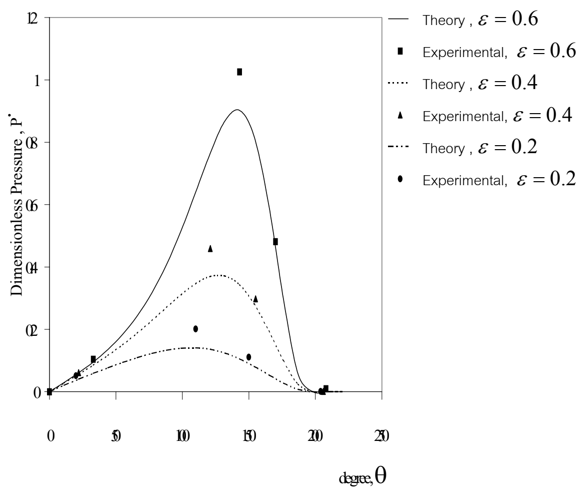

Figure 11 Pressure profile for soybean oil and.2 % ZDTP clearance 0.1mm, speed 200 rpm. and L/D=0.5

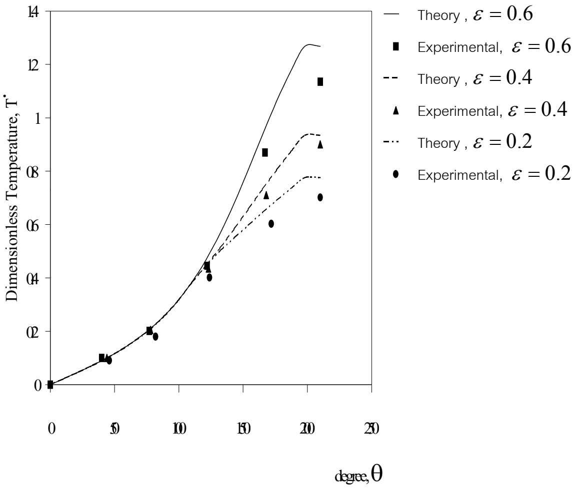

Figure12 Temperature profile for soybean oil and 0.2 % ZDTP clearance 0.1mm, speed 200 rpm. and L/D = 0.5

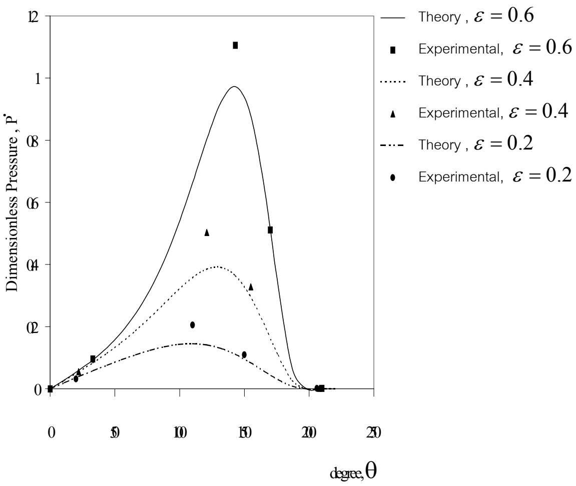

Figure 13 Pressure profile for soybean oil and 1 % ZDTP clearance 0.1mm, speed 200 rpm. and L/D = 0.5

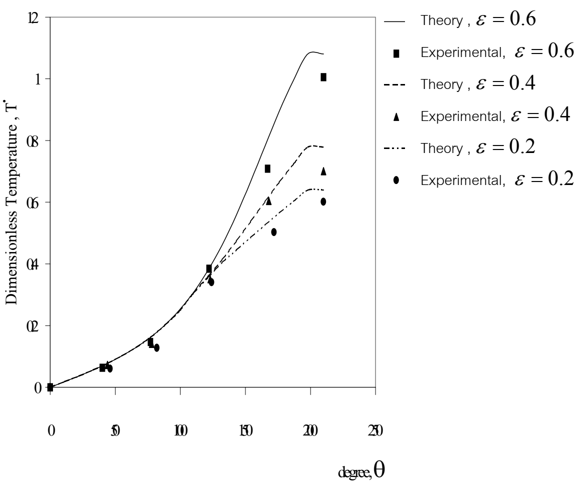

Figure14 Temperature profile for soybean oil and 1 % ZDTP clearance 0.1mm, speed 200 rpm. and L/D = 0.5

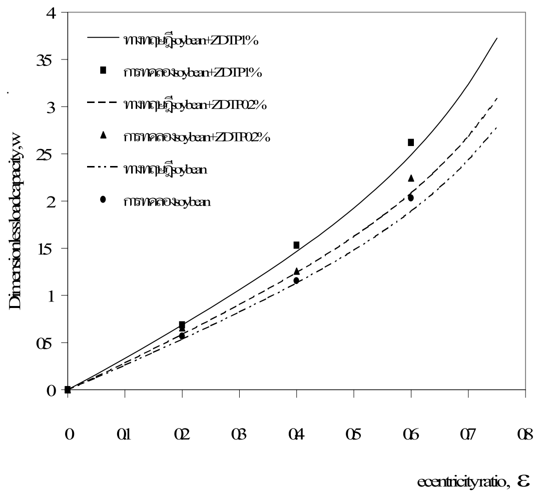

Figure 15 Dimensionless load capacity of journal bearing lubricated with soybean-based oils operated at varying eccentricity ratio

5. Conclusions

In this paper, the static characteristic of a full circular journal bearing with non-Newtonian soybean-based oil have been investigated theoretically and experimentally and can be concluded as:

1) The power law model was proposed for the calculation. The actual flow characteristic of the non-Newtonian Psedoplastic soybean-based oils are highly nonlinearity specially at low speed operating conditions
2) The pressures from experiments are higher than the oil film pressure from numerical calculation due to the adiabatic assumption.
3) The oil film temperatures from the experiments are lower than that from the theoretical calculation due to the adiabatic assumption.
4) Percentage of ZDTP has greatly effect on pressure temperature distribution and load carrying capacity. The pressure increase slightly with the increase in percentage of ZDTP but the temperature decrease with the increase in percentage of ZDTP

6. Nomenclature

$\bar{P}$ = film pressure

P = dimensionless film pressure, $\dfrac{\bar{P} - P_o}{2\pi N m_o (R/C)^2}$

R = journal radius

$\bar{h}$ = film thickness

h = dimensionless film thickness, $\dfrac{\bar{h}}{C}$

e = eccentricity

ε = eccentricity ratio

ϕ = attitude angle

ω = angular velocity of rotating shaft

$\bar{z}$ = axial coordinate

z = dimensionless axial coordinate $\frac{\bar{z}}{L}$

θ = circumferential coordinate

$\bar{u}$ = velocity of lubricant θ -direction

$\bar{w}$ = velocity of lubricant z-direction

$F_{\varepsilon}, F_{\phi}$ = dimensionless radial and tangential fluid film force components

η = dimensionless coordinate = $\frac{y}{h}$

$\bar{T}$ = oil film temperature

β = Viscosity-temperature index

$\bar{m}$ = Apparent viscosity at temperature $\bar{T}$

w = dimensionless velocity Z-direction = $\frac{\bar{w}}{u}$

U = journal velocity = ωR

T = dimensionless oil film temperature = $\beta(\bar{T} - T_0)$

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