Static Characteristics of High-Speed Slider Bearings Lubricated with Non-Newtonian Fluids*

(In the Case of Infinitely Long Bearings)

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In this paper, the static characteristics of infinitely long slider bearings lubricated with non-Newtonian fluids are examined theoretically by considering the fluid inertia effects. In the derivation of the modified Reynolds equation, the fluid inertia terms in the momentum equation for the lubricant film are averaged over the film thickness, and the Rabinowitsch empirical model is used as a constitutive equation of non-Newtonian fluids. Applying the modified Reynolds equation to the infinitely long slider bearings, the film pressure, load carrying capacity, friction force and inlet flow rate are obtained under various values of the dimensionless nonlinear factor and film thickness ratio, and the effects of fluid inertia on these static characteristics are discussed.

Key Words: Tribology, Lubrication, Hydrodynamic Bearings, Non-Newtonian Fluids, Slider Bearing, Modified Reynolds Equation

1. Introduction

In the fluid film bearings used under high-speed conditions, lubricants containing a large quantity of high-molecular-weight polymers are sometimes used in order to prevent viscosity variation with temperature change. Moreover, to maintain the liquid lubrication conditions under high-temperature operation conditions, it may be possible to use liquefied glasses as lubricants. Such lubricants show non-Newtonian characteristics. On the other hand, under high-speed operation conditions fluid inertia effects play an important role in the lubricant films. Therefore, the non-Newtonian lubrication behavior under high-speed operation conditions will become very complicated.

The effects of non-Newtonian characteristics and fluid inertia in hydrodynamic bearings have been investigated independently by a number of investigators⁽¹⁾⁻⁽⁶⁾. However, none of these investigators considered the combined effects of fluid inertia and non-Newtonian characteristics of lubricants because of the double nonlinear characteristics of non-Newtonian and fluid film inertia⁽⁷⁾. In this paper, the static characteristics of infinitely long, high-speed slider bearings lubricated with non-Newtonian fluids are examined theoretically by using the modified Reynolds equation based on the Rabinowitsch empirical model. Analytical solutions of film pressure, load carrying capacity, friction force and flow rate are obtained for inclined-plane slider bearings.

2. Nomenclature

- a : film thickness ratio ($=h_1/h_2$)
- f : equivalent pressure gradient
- $\bar{f}$: dimensionless equivalent pressure gradient ($=h_2^2 f/(\mu U)$)
- F_j : friction force per unit width on the sliding surface
- $\bar{F}_j$: dimensionless friction force per unit width on

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- the sliding surface $(=h_2 F_3/(\mu UL))$
- h : film thickness
- $\bar{h}$: dimensionless film thickness $(=h/h_2)$
- h_1, h_2 : film thicknesses at inlet and outlet of bearing
- k : nonlinear factor of lubricants
- L : slider length
- p : film pressure
- $\bar{p}$: dimensionless film pressure $(h_2^2 p/(\mu UL))$
- Q : inlet flow rate per unit width
- $\bar{Q}$: dimensionless inlet flow rate per unit width $(=Q/(h_2 U))$
- Re^* : inertia parameter $(=\rho h_2^2 U/(\mu U))$
- u : velocity component in the sliding direction
- $\bar{u}$: dimensionless velocity component in the sliding direction $(=u/U)$
- U : sliding velocity
- v : velocity component in the film thickness direction
- W : load carrying capacity
- $\bar{W}$: dimensionless load carrying capacity $(=h_2^2 W/(\mu UL^2))$
- x : coordinate in the sliding direction
- $\bar{x}$: dimensionless coordinate in the sliding direction $(=x/L)$
- y : coordinate in the film thickness direction
- $\bar{y}$: dimensionless coordinate in the film thickness direction $(=y/h_2)$
- δ : dimensionless nonlinear factor of lubricants $(=k\mu^2 U^2/h_2^2)$
- μ : initial viscosity of non-Newtonian fluids
- μ^* : apparent viscosity of non-Newtonian fluids
- ρ : density of fluids
- τ_{xy} : shear stress

3. Modified Reynolds Equation and its Solutions

In the infinitely long plane slider bearing as shown in Fig. 1, it may be possible to assume that the fluid inertia forces act uniformly in the film thickness direction because the fluid film is very thin compared with other sizes in the bearing. Therefore, approx-

Fig. 1 Geometry of infinitely long plane slider bearing

imating the fluid inertia forces by the value averaged across the film thickness, the momentum equation for the fluid film flow can be written as follows:

$$\frac{\rho}{h} \int_0^h \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) dy = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} \quad (1)$$

Neglecting the compressibility of lubricants, the equation of continuity is given as follows:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (2)$$

Considering that the slider is moving with surface velocity U in the x -direction, the boundary conditions for the velocity components u, v are:

$$\begin{aligned} u(y=0) &= U, \quad u(y=h)=0, \\ v(y=0) &= v(y=h)=0 \end{aligned} \quad (3)$$

The constitutive equation for non-Newtonian fluids is generally expressed as:

$$\tau_{xy} = \mu^* (\tau_{xy}) \frac{\partial u}{\partial y} \quad (4)$$

where μ^* is the apparent viscosity to be determined experimentally. In this paper, the following Rabinowitsch empirical equation is used for the apparent viscosity⁽⁸⁾.

$$\mu^* = \frac{\mu}{1 + k \tau_{xy}^2} \quad (5)$$

Based on the above basic equations, the one-dimensional modified Reynolds equation for infinitely long bearings is derived. After rewriting the integrated term in Eq. (1) by means of continuity Eq. (2), the following equivalent pressure gradient is introduced by considering that the integrated term is independent of the coordinate y :

$$f = \frac{\partial p}{\partial x} + \frac{\rho}{h} \frac{\partial}{\partial x} \int_0^h u^2 dy \quad (6)$$

From Eqs. (1) and (6), the momentum Eq. (1) is simplified as:

$$\frac{\partial \tau_{xy}}{\partial y} = f \quad (7)$$

Integrating Eq. (7) and substituting the result, τ_{xy} , into Eq. (4), the velocity gradient is obtained. Moreover, integrating the velocity gradient with the velocity boundary conditions in Eq. (3), the velocity distribution is determined. The nondimensional form of the velocity distribution is expressed as:

$$\begin{aligned} \bar{u} &= \frac{1}{2} \bar{f} \bar{F}_1(\bar{y}) + \delta \bar{f}^3 F_2(\bar{y}) \\ &+ \frac{1}{\bar{h} \{1 + \delta (\bar{h} \bar{f}/2)^2\}} \{ -\bar{y} + \delta \bar{f}^2 F_3(\bar{y}) \} + 1 \end{aligned} \quad (8)$$

where:

$$\begin{aligned} F_1(\bar{y}) &= \bar{y}(\bar{y} - \bar{h}), \\ F_2(\bar{y}) &= \frac{\bar{y}^4}{4} - \frac{\bar{h}}{2} \bar{y}^3 + \frac{3}{8} \bar{h}^2 \bar{y}^2 - \frac{\bar{h}^3}{8} \bar{y}, \\ F_3(\bar{y}) &= -\bar{y}^3 + \frac{3}{2} \bar{h} \bar{y}^2 - \frac{3}{4} \bar{h}^2 \bar{y} \end{aligned} \quad (9)$$

The equivalent pressure gradient defined in Eq. (6) is normalized as:

$$\bar{f} = \frac{d\bar{p}}{dx} + \frac{Re^*}{\bar{h}} \frac{d}{d\bar{x}} \int_0^{\pi} \bar{u}^2 d\bar{y} \quad (10)$$

Substituting Eq. (8) into the dimensionless equation of continuity and integrating over the film thickness, the dimensionless modified Reynolds equation is obtained as follows:

$$\frac{d}{d\bar{x}} \left(\frac{\bar{h}^3}{12} \bar{f} + \delta \frac{\bar{h}^5}{80} \bar{f}^3 \right) = \frac{1}{2} \frac{d\bar{h}}{d\bar{x}} \quad (11)$$

To solve the dimensionless modified Reynolds equation, the pressure $\bar{p}$, equivalent pressure gradient $\bar{f}$ and the velocity component $\bar{u}$ are expanded based on the perturbation technique for $Re^* < 1$ as follows:

$$\begin{aligned} \bar{f} &= \bar{f}_0 + \bar{f}_1 Re^* + O(Re^{*2}) \\ \bar{p} &= \bar{p}_0 + \bar{p}_1 Re^* + O(Re^{*2}) \\ \bar{u} &= \bar{u}_0 + \bar{u}_1 Re^* + O(Re^{*2}) \end{aligned} \quad (12)$$

Substitution of Eq. (12) into Eqs. (10) and (11) results in the linearized equations for each component. Moreover, summing the solutions of linearized equations until the first order of inertia parameter Re^* and dimensionless nonlinear factor δ under the assumption of $-1 < \delta < 1$, the solution for the film pressure is expressed as:

$$\bar{p} = \bar{p}_{00} + \delta \bar{p}_{01} + Re^* \bar{p}_1 \quad (13)$$

where the pressure components are given by:

$$\bar{p}_{00} = \frac{6}{a-1} \left(I_{h1} - \frac{a}{a+1} I_{h2} \right) \quad (14)$$

$$\begin{aligned} \bar{p}_{01} &= -\frac{162}{5(a-1)} \left(\frac{1}{3} I_{h3} - \frac{3}{4} \bar{h}_{m0} I_{h4} \right. \\ &\quad \left. + \frac{3}{5} \bar{h}_{m0}^2 I_{h5} - \frac{h_{m0}^3}{6} I_{h6} \right) + \frac{h_{m1}}{2(a-1)} I_{h2} \end{aligned} \quad (15)$$

$$\begin{aligned} \bar{p}_1 &= \frac{1}{a-1} \int_0^{\pi} \frac{B(\bar{x})}{\bar{h}^2} d\bar{x} + \bar{H}_m \int_0^{\pi} \frac{d\bar{x}}{H(\bar{x})} \\ &\quad + \frac{B(0)}{a} - \frac{B(\bar{x})}{\bar{h}} \end{aligned} \quad (16)$$

The dimensionless film thickness $\bar{h}$ is expressed as:

$$\bar{h} = a + (1-a)\bar{x}; \quad 0 \leq \bar{x} \leq 1 \quad (17)$$

and the other functions are defined as follows:

$$I_{hi} = \frac{1}{\bar{h}^i} - \frac{1}{a^i} \quad (i=1 \sim 6),$$

$$I_{1i} = 1 - \frac{1}{a^i} \quad (i=2 \sim 6), \quad \bar{h}_{m0} = \frac{2a}{a+1},$$

$$\begin{aligned} \bar{h}_{m1} &= \frac{324}{5I_{12}} \left(\frac{1}{3} I_{13} - \frac{3}{4} \bar{h}_{m0} I_{14} + \frac{3}{5} \bar{h}_{m0}^2 I_{15} \right. \\ &\quad \left. - \frac{\bar{h}_{m0}^3}{6} I_{16} \right), \end{aligned}$$

$$\begin{aligned} f_0 &= 6 \left(\frac{1}{\bar{h}^2} - \frac{\bar{h}_{m0}}{\bar{h}^3} \right) - \delta \frac{162}{5} \left(\frac{1}{\bar{h}^4} - \frac{3\bar{h}_{m0}}{\bar{h}^5} \right. \\ &\quad \left. + \frac{3\bar{h}_{m0}^2}{\bar{h}^6} - \frac{\bar{h}_{m0}^3}{\bar{h}^7} \right) + \delta \frac{\bar{h}_{m1}}{\bar{h}^3}, \end{aligned}$$

$$\bar{g}_0 = \frac{1}{1 + \delta \left(\frac{\bar{h}}{2} f_0 \right)^2}, \quad H(\bar{x}) = \frac{\bar{h}^3}{12} + \frac{3}{80} \delta \bar{h}^5 \bar{f}_0^2,$$

$$B(\bar{x}) = \bar{h} - \frac{\bar{h}^3}{6} \bar{f}_0 + \frac{\bar{h}^5}{120} \bar{f}_0^2 - \bar{h} \bar{g}_0 + \frac{\bar{h}}{3} \bar{g}_0^2 + \frac{\bar{h}^3}{12} \bar{f}_0 \bar{g}_0$$

$$\begin{aligned} &+ \delta \left(-\frac{\bar{h}^5}{40} \bar{f}_0^3 - \frac{\bar{h}^3}{4} \bar{g}_0 \bar{f}_0^2 + \frac{\bar{h}^7}{420} \bar{f}_0^4 \right. \\ &\quad \left. + \frac{3}{20} \bar{h}^2 \bar{f}_0^2 \bar{g}_0^2 + \frac{\bar{h}^5}{30} \bar{f}_0^3 \bar{g}_0 \right), \\ \bar{H}_m &= \left\{ B(1) - \frac{B(0)}{a} - \frac{1}{a-1} \int_0^1 \frac{B(\bar{x})}{\bar{h}^2} d\bar{x} \right\} / \\ &\quad \int_0^1 \frac{d\bar{x}}{H(\bar{x})} \end{aligned} \quad (18)$$

The load carrying capacity $\bar{W}$ is given by the integration of pressure $\bar{p}$ as:

$$\bar{W} = \bar{W}_{00} + \delta \bar{W}_{01} + Re^* \bar{W}_1 \quad (19)$$

where each component is:

$$\bar{W}_{00} = \int_0^1 \bar{p}_{00} d\bar{x}, \quad \bar{W}_{01} = \int_0^1 \bar{p}_{01} d\bar{x}, \quad \bar{W}_1 = \int_0^1 \bar{p}_1 d\bar{x} \quad (20)$$

The friction force exerted on the sliding surface, $\bar{F}_j$, is obtained from:

$$\begin{aligned} \bar{F}_j &= \int_0^1 \left[\frac{\bar{h}}{2} \bar{f} + \frac{1}{\bar{h} \{1 + \delta (\bar{h} \bar{f}/2)^2\}} \right] d\bar{x} \\ &\approx F_{00} + \delta F_{01} + Re^* F_0 \end{aligned} \quad (21)$$

where each component is:

$$\begin{aligned} \bar{F}_{00} &= \frac{1}{2} (a-1) \bar{W}_{00} + \frac{\ln a}{a-1}, \\ \bar{F}_{01} &= \frac{1}{2} (a-1) \bar{W}_{01} - \frac{9}{a-1} \left(\frac{1}{2} I_{12} \right. \\ &\quad \left. - \frac{3}{2} h_{m0} I_{13} + \frac{h_{m0}^2}{4} I_{14} \right) \\ \bar{F}_1 &= \frac{1}{2} (a-1) \bar{W}_1 + \frac{1}{2} (B(1) - B(0)) \end{aligned} \quad (22)$$

Moreover, the inlet flow rate, $\bar{Q}$, is calculated from:

$$\bar{Q} = \int_0^{\pi} \bar{u} d\bar{y} \approx \bar{Q}_{00} + \delta \bar{Q}_{01} + Re^* \bar{Q}_1 \quad (23)$$

where each component is:

$$\bar{Q}_{00} = \frac{a}{a+1}, \quad \bar{Q}_{01} = -\frac{12 \bar{h}_m}{a}, \quad \bar{Q}_1 = \frac{a^3 \bar{H}_m}{12 H(0)} \quad (24)$$

4. Results and Discussion

Based on the above equations, the static characteristics of one-dimensional slider bearings lubricated with non-Newtonian fluids in the inertial flow regime are easily obtained.

Figures 2 and 3 show the pressure components, $\bar{p}_{00}$, $\bar{p}_{01}$ and $\bar{p}_1$ for three different values of film thickness ratios of $a=2, 3$ and 4. As can be seen in Fig. 2, the pressure component due to non-Newtonian effects, $\bar{p}_{01}$, takes negative values over the bearing surface and the position of negative peak pressure moves to the outlet side. On the other hand, the pressure component due to inertia effects varies significantly according to the value of dimensionless nonlinear factor δ . The pressure component for dilatant fluids, which is indicated by $\delta < 0$, takes positive values over the bearing surface and this becomes larger as the absolute value of δ increases. The tendency becomes more pronounced as the film

Fig. 2 Pressure components due to viscous and non-Newtonian effects

Fig. 4 Variation of static characteristics due to viscous and non-Newtonian effects with film thickness ratio

thickness ratio decreases. The component for pseudo plastic fluids, which is indicated by $\delta > 0$, takes positive values for small δ . However, this component decreases with increase of δ and takes negative values near the outlet for large δ . The tendency becomes more pronounced as the film thickness ratio increases.

Figure 4 shows the components of load carrying capacity, friction force and inlet flow rate due to viscous and non-Newtonian effects. The load capacity component due to viscous effects, $\bar{W}_{00}$, is always positive and takes maximum value at $a=2.2$. The

(a) $a=2$

(b) $a=3$

(c) $a=4$

Fig. 3 Pressure component due to inertia effects

component due to non-Newtonian effects, $\bar{W}_{01}$, is always negative and takes minimum value at $a=3.2$. After taking their maximum values, the components $\bar{W}_{00}$ and $-\bar{W}_{01}$ decrease with increase of film thickness ratio, but the rate of decrease of the component $-\bar{W}_{01}$ is much smaller than that of the component $\bar{W}_{00}$. The ratios of $\bar{W}_{01}$ to $\bar{W}_{00}$ for the film thickness ratios of $a=2.2, 3.2$ and 5.0 are, respectively, $|\bar{W}_{01}/\bar{W}_{00}|=0.22, 0.31, 0.33$; it follows that the non-Newtonian effects

on the load carrying capacity become more pronounced as the film thickness ratio increases. The friction force component due to viscous effects, $\bar{F}_{j00}$, decreases with increasing film thickness ratio, but the component due to non-Newtonian effects, $\bar{F}_{j01}$, increases with film thickness ratio, and it acts in the opposite direction to the component $\bar{F}_{j00}$. The ratios of $\bar{F}_{j01}$ to $\bar{F}_{j00}$ for $a=2.2, 3.2$ and 5.0 are, respectively, $|\bar{F}_{j01}/\bar{F}_{j00}|=0.027, 0.074, 0.107$ and it can be seen that the non-Newtonian effects on the friction force become larger as the film thickness ratio increases. The inlet flow rate component due to non-Newtonian effects, $\bar{Q}_{01}$, is always positive. Thus, the flow rate increases due to non-Newtonian effects. The ratios of $\bar{Q}_{01}$ to $\bar{Q}_{00}$ for $a=2.2, 3.2, 5.0$ are, respectively, $\bar{Q}_{01}/\bar{Q}_{00}=0.022, 0.072, 0.144$, and the non-Newtonian effects on the inlet flow rate becomes more pronounced as the film thickness ratio increases. It follows from the above ratios that the influence of non-Newtonian characteristics is most significant in the load carrying capacity.

The relation between the load capacity component due to inertia effects $\bar{W}_1$ and film thickness ratio is shown in Fig. 5. The variation of component $\bar{W}_1$ strongly depends on the magnitude of dimensionless nonlinear factor δ . In the case of Newtonian fluids ($\delta=0$), the component $\bar{W}_1$ always takes positive values and it increases with increasing film thickness ratio. In the case of dilatant fluids ($\delta<0$), the component $\bar{W}_1$ is slightly larger than that of Newtonian fluids for smaller film thickness ratio, but it becomes smaller as the film thickness ratio increases. In the case of

Fig. 5 Variation of load capacity component due to inertia effects with film thickness ratio

pseudo plastic fluids ($\delta>0$), the component $\bar{W}_1$ increases with increasing film thickness ratio, but it decreases with increasing dimensionless nonlinear factor δ for constant film thickness ratio. The load capacity component due to viscous effects, $\bar{W}_{00}$, decreases considerably with increasing film thickness ratio for $a>2.2$ as shown in Fig. 4, but the component $\bar{W}_1$ increases with increasing film thickness ratio. Thus, the influence of inertia effects becomes more pronounced as the film thickness ratio increases. The ratios of $\bar{W}_1$ to $\bar{W}_{00}$ for $\delta=0, 0.1$ and 0.2 , respectively, are as follows: at $a=2.2$ $\bar{W}_1/\bar{W}_{00}=0.08, 0.04, 0$; at $a=4$ $\bar{W}_1/\bar{W}_{00}=0.26, 0.18, 0.04$; at $a=5$ $\bar{W}_1/\bar{W}_{00}=0.4, 0.29, 0.11$. This shows that the inertia effects on the load carrying capacity become more pronounced as the film thickness ratio increases.

Figure 6 shows the relation between the friction force component due to inertia effects, $\bar{F}_{j1}$, and film thickness ratio. In the case of Newtonian fluids ($\delta=0$), the component $\bar{F}_{j1}$ increases with increasing film thickness ratio, but the rate of increase becomes smaller for larger film thickness ratio. In the cases of dilatant ($\delta<0$) and pseudo plastic ($\delta>0$) fluids, the magnitude of component $\bar{F}_{j1}$ is nearly the same that for Newtonian fluids for $a<2.5$. For $a>2.5$, however, the magnitude of $\bar{F}_{j1}$ strongly depends on the magnitude of dimensionless nonlinear factor δ , that is, this component for Newtonian fluids takes the largest value for $a>2.5$ and the value of $\bar{F}_{j1}$ for pseudo plastic fluids becomes smaller as the nonlinear factor δ increases. The same tendency is seen in the case of dilatant fluids. The ratios of $\bar{F}_{j1}$ to $\bar{F}_{j00}$ for $\delta=0, 0.1$ and 0.2 , respectively, are as follows: at $a=2.2$ $\bar{F}_{j1}/\bar{F}_{j00}$

Fig. 6 Variation of friction force component due to inertia effects with film thickness ratio

Fig. 7 Variation of inlet flow rate component due to inertia effects with film thickness ratio

$=0.109, 0.109, 0.109$; at $a=4$ $\bar{F}_{j1}/\bar{F}_{j00}=0.213, 0.184, 0.104$; at $a=5$ $\bar{F}_{j1}/\bar{F}_{j00}=0.242, 0.185, 0.048$. As can be seen from these ratios, the influence of fluid inertia on the friction forces varies significantly according to the combination of nonlinear factor and film thickness ratio.

Figure 7 shows the relation between the flow rate component due to inertia effects, $\bar{Q}_1$, and film thickness ratio. As the component $\bar{Q}_1$ is always negative, the inlet flow rate decreases with inertia effects. The component for Newtonian fluids ($\delta=0$) decreases with increasing film thickness ratio. The component for dilatant fluids ($\delta<0$) becomes larger than that for Newtonian fluids at $a>2$ and the difference between the dilatant and Newtonian fluids increases with increasing film thickness ratio. The same tendency can be seen for pseudo plastic fluids ($\delta>0$). The ratios of $\bar{Q}_1$ to $\bar{Q}_{00}$ for $\delta=0, 0.1, 0.2$ are, respectively, as follows: at $a=2.2$ $|\bar{Q}_1/\bar{Q}_{00}|=0.044, 0.044, 0.044$; at $a=4$ $|\bar{Q}_1/\bar{Q}_{00}|=0.105, 0.096, 0.053$; at $a=5$ $|\bar{Q}_1/\bar{Q}_{00}|=0.172, 0.101, 0.040$. It follows that the influence of fluid inertia on the inlet flow rate depends strongly on the combination of nonlinear factor and film thickness ratio.

Using the components presented in Figs. 4 through 7, the static characteristics considering the combined effects of non-Newtonian and fluid film inertia can be obtained easily. Figure 8 shows the load carrying capacity, friction force, inlet flow rate and pressure center for the combinations of $\delta=0, 0.2$ and $Re^*=0, 0.5$. As described above, the non-Newtonian effects cancel the fluid inertia effects. Therefore, the

Fig. 8 Variation of static characteristics with film thickness ratio

differences between the results for $Re^*=0.5, \delta=0.2$ and the results for $Re^*=0, \delta=0$ are much smaller than the differences between the results for $Re^*=0.5, \delta=0$ and the results for $Re^*=0, \delta=0$. The combined effects of fluid inertia and non-Newtonian characteristics on the pressure center are much smaller than those on the other characteristics.

5. Conclusions

In this paper, the static characteristics of high-speed slider bearings lubricated with non-Newtonian fluids were examined theoretically based on the one-dimensional modified Reynolds equation considering the combined effects of fluid inertia and non-Newtonian characteristics. The conclusions obtained are summarized as follows.

- (1) The analytical solutions for pressure distribution, load carrying capacity, friction force and inlet flow rate in the infinitely long plane slider bearings were obtained by considering the combined effects of fluid film inertia and non-Newtonian characteristics.
- (2) The pressure component due to viscous effects is positive and the component due to non-Newtonian effects is negative. On the other hand, the variation of pressure component due to inertia effects depends strongly on the combination of nonlinear factor and film thickness ratio.
- (3) The load carrying capacity and friction force decrease due to non-Newtonian effects and the inlet flow rate increases due to these effects. On the other hand, the load carrying capacity and friction force

increase due to the inertia effects and the inlet flow rate decreases due to these effects. Thus, the non-Newtonian effects on these characteristics cancel the inertia effects on the characteristics. The combined effects of fluid inertia and non-Newtonian characteristics on the pressure center are much smaller than those on the other static characteristics.

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