Effect of the Interfacial Slippage on the Moving Surface in a Hydrodynamic Step Bearing

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Abstract: The effect of the interfacial slippage on the moving surface in a hydrodynamic step bearing is analytically investigated based on the interfacial limiting shear strength model. The calculation results show that when the interfacial slippage occurs on the whole moving surface but is absent on the whole stationary surface, the load-carrying capacity of the bearing is independent on the sliding speed but is intimately dependent on the contact-fluid interfacial shear strength on the moving surface, and the carried load of the bearing is normally heavily reduced as compared to that calculated from conventional hydrodynamic lubrication theory especially for high sliding speeds; The friction coefficient of this mode of bearing is significantly higher than that of the conventional hydrodynamic step bearing for a high sliding speed, but the case is opposite for a low sliding speed. The results show the potential application value of this mode of bearing for reducing the friction at a low sliding speed or the necessity of preventing the occurrence of the interfacial slippage on the moving surface in a hydrodynamic step bearing at a high sliding speed.

Keywords: Bearing, Interfacial slippage, Load, Friction coefficient, Model.

1. INTRODUCTION

The interfacial slippage was found long time ago in fluid mechanics [1-3]. It exhibits as the moving of the adjacent fluid layer relative to a solid surface, indicating the loss of the adherence of a fluid to the solid surface. In itself, the boundary condition should generally be considered as the interfacial slippage in a confined fluid flow [4]. This was however omitted by conventional fluid mechanics including conventional hydrodynamic lubrication theory [5]. In light operating conditions, the interfacial slippage may be absent; But in severe operating conditions, the interfacial slippage may be unavoidable because of the high shear stress exceeding the contact-fluid interfacial shear strength, which is actually quite limited [6,7].

Rozeanu and Snarsky [8,9] and Rozeanu and Tipei [10] experimentally found the drastic pressure drop in a hydrodynamic journal bearing and attributed it to the occurrence of the interfacial slippage on the rotating shaft surface. Chen and Zhang [11] analytically showed that the occurrence of the interfacial slippage on the whole rotating shaft surface severely reduces the hydrodynamic pressures and load-carrying capacity of a hydrodynamic journal bearing.

There have been a lot of studies on the interfacial slippage effect in a hydrodynamic lubrication. Some of them on the negative effect of the interfacial slippage [8-14], while others on the beneficial effect of the interfacial slippage in improving the carried load but reducing the friction coefficient of the lubricated contact [15-17].

The interfacial slippage occurring in different areas of the lubricated contact will have quite different effects in a hydrodynamic lubrication. By artificially designing the interfacial slippage in specific areas of a lubricated contact, new hydrodynamic thrust and journal bearings have been invented with abnormal geometrical configurations severely confronting conventional lubrication theories [18-21].

In a hydrodynamic bearing with a conventional normal geometrical configuration, how to design the interfacial slippage to improve the bearing performance is still a question of significant practical interest. In the previous studies, there have been the interfacial slippage artificially designed only on the stationary surface in the bearing inlet zone [16, 18, 19, 22], or both on the stationary surface in the bearing inlet zone and on the moving surface in the bearing outlet zone [17]. Guan *et al*. [23] found that the interfacial slippage occurring on both the stationary surface in the bearing inlet zone and on the whole moving surface can increase the load-carrying capacity of a hydrodynamic step bearing with a conventional normal geometrical configuration only when the fluid-moving surface interfacial shear strength is sufficiently larger than the fluid-stationary surface interfacial shear strength; while the friction coefficient of this bearing is overall reduced by such designed interfacial slippage. Xia *et al*. [24] investigated the performance of a hydrodynamic wedge-platform thrust slider bearing with a conventional normal geometrical configuration with the

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interfacial slippage only occurring on the whole moving surface. They found that the carried load of this bearing is normally much lower than that of the conventional type of the bearing in the same operating condition, but the friction coefficient of the bearing is considerably smaller than that of the conventional type of the bearing in the same operating condition if the wedge angle of the bearing is not too small. They suggested that such a bearing can be designed as energy-conserved with the sacrifice of the load-carrying capacity. Cheng and Zhang [25] found that the interfacial slippage occurring on the whole moving surface deteriorates the overall performance of a hydrodynamic inclined fixed pad thrust slider bearing with a conventional normal geometrical configuration. They suggested the prevention of such an interfacial slippage in the studied bearing.

The present paper aims to study the effect of the interfacial slippage on the whole moving surface in a hydrodynamic step bearing with a conventional normal geometrical configuration based on the limiting interfacial shear strength model. The detailed performance of the studied bearing is revealed. New findings and conclusions have been obtained concerning how to design or prevent such an interfacial slippage in the studied bearing.

2. STUDIED BEARING

Figure **1** shows the studied bearing, where the upper contact surface is stationary and the lower contact surface is moving with the speed *u*. In this bearing, the contact-fluid interfacial slippage occurs on the whole moving surface but is absent on the whole stationary surface. The lubricated areas of the bearing are divided into the inlet and outlet zones, the widths of which are respectively l_2 and l_1. The lubricating film thicknesses in the inlet and outlet zones are respectively h_i and h_o. The used coordinates are also shown in Figure **1**.

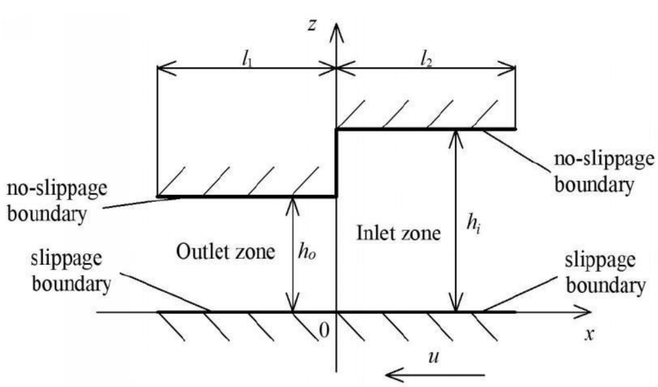


Figure 1: The studied bearing.

3. ANALYSIS

An analysis was made for the bearing in Figure **1** based on the interfacial limiting shear strength model [26]. It is based on the following assumptions:

1. Within the lubricating film, the fluid is Newtonian;
2. The compressibility of the fluid is negligible;
3. The fluid is isoviscous;
4. The fluid inertia is negligible;
5. The fluid is in laminar flow;
6. The operating condition is isothermal and steady-state.

3.1. For the Inlet Zone

The Reynolds equation for the inlet zone is [27]:

$$\frac{\partial p}{\partial x} = \lambda_1 \tag{1}$$

where *p* is the fluid film pressure and $\lambda_1 = -3\tau_{sb}/(2h_i) - 3q_v\eta/h_i^3$; Here, τ_{sb} is the contact-fluid interfacial shear strength on the moving surface, η is the fluid dynamic viscosity, and q_v is the volume flow rate of the fluid through the bearing per unit contact length.

Integrating Eq. (1) gives that:

$$p = \lambda_1 x + c_1 \tag{2}$$

where c_1 is an integral constant. From the boundary condition $p|_{x=l_2} = 0$, it is solved that $c_1 = -\lambda_1 l_2$. Then the fluid film pressure in the inlet zone is:

$$p = \lambda_1 (x - l_2) \tag{3}$$

At *x*=0, the fluid film pressure is $p(0) = -\lambda_1 l_2$.

3.2. For the Outlet Zone

The Reynolds equation for the outlet zone is [27]:

$$\frac{\partial p}{\partial x} = \lambda_2 \tag{4}$$

where $\lambda_2 = -3\tau_{sb}/(2h_o) - 3q_v\eta/h_o^3$.

Integrating Eq. (4) gives that:

$$p = \lambda_2 x + c_2 \tag{5}$$

where c_2 is an integral constant. From the boundary condition $p|_{x=-l_1}=0$, it is solved that $c_2=\lambda_2 l_1$. Then the fluid film pressure in the outlet zone is:

$$p=\lambda_2(x+l_1) \tag{6}$$

At x=0, the fluid film pressure is $p(0)=\lambda_2 l_1$.

3.3. Mass Flow Rate, Carried Load and Optimum Condition of the Bearing

According to the continuity of the fluid film pressure, it is obtained that:

$$-\lambda_1 l_2=\lambda_2 l_1 \tag{7}$$

Define $\lambda_h=h_i/h_o$ and $\psi=l_1/l_2$, it is solved from Eq. (7) that:

$$q_v=-\frac{h_o^2\lambda_h^2\tau_{sb}(\psi\lambda_h+1)}{2\psi\lambda_h^3\eta+2\eta} \tag{8}$$

The maximum fluid film pressure is:

$$p_{\max}=p(0)=-\lambda_1 l_2=\frac{3\tau_{sb}\psi(l_1+l_2)(\lambda_h^2-1)}{2h_o(\psi\lambda_h^3+1)(\psi+1)} \tag{9}$$

The carried load per unit contact length of the bearing is:

$$w=\frac{p_{\max}(l_1+l_2)}{2}=\frac{3}{4}(l_1+l_2)^2 G\frac{\tau_{sb}(\lambda_h^2-1)}{h_o} \tag{10}$$

where $G=\psi/[(\psi\lambda_h^3+1)(\psi+1)]$. It can be found that *G* reaches the maximum when $\psi=\lambda_h^{-3/2}$, which is the optimum condition for the maximum load-carrying capacity of the bearing.

3.4. Shear Stress, Interfacial Slipping Velocity and Condition for the Bearing

The shear stress on the upper contact surface in the inlet zone is:

$$\tau_{a,i}=\tau_{sb}+\frac{dp}{dx}h_i=\frac{\tau_{sb}(3\psi\lambda_h-\psi\lambda_h^3+2)}{2(\psi\lambda_h^3+1)} \tag{11}$$

For preventing the boundary slippage occurrence on the upper contact surface in the inlet zone, it should be satisfied that $\tau_{sa,i}>\tau_{a,i}$. Here, $\tau_{sa,i}$ is the contact-fluid interfacial shear strength on the upper contact surface in the inlet zone.

The fluid film slipping velocity on the lower contact surface in the inlet zone is [27]:

$$\Delta u_{b,i}=u-\frac{h_i^2}{2\eta}\frac{dp}{dx}-\frac{\tau_{sb}h_i}{\eta} \tag{12}$$

For the occurrence of the fluid film slippage on the lower contact surface in the inlet zone, it should be satisfied that $\Delta u_{b,i}>0$ [27].

According to Eqs. (1) and (12), it is then obtained that:

$$\frac{u\eta}{\tau_{sb}h_o}>\frac{\psi\lambda_h^4+3\psi\lambda_h^2+4\lambda_h}{4\psi\lambda_h^3+4} \tag{13}$$

The shear stress on the upper contact surface in the outlet zone is:

$$\tau_{a,o}=\tau_{sb}+\frac{dp}{dx}h_o=\frac{\tau_{sb}(3\lambda_h^2+2\psi\lambda_h^3-1)}{2(\psi\lambda_h^3+1)} \tag{14}$$

For preventing the boundary slippage occurrence on the upper contact surface in the outlet zone, it should be satisfied that $\tau_{sa,o}>\tau_{a,o}$. Here, $\tau_{sa,o}$ is the contact-fluid interfacial shear strength on the upper contact surface in the outlet zone.

The fluid film slipping velocity on the lower contact surface in the outlet zone is [27]:

$$\Delta u_{b,o}=u-\frac{h_i^2}{2\eta}\frac{dp}{dx}-\frac{\tau_{sb}h_i}{\eta} \tag{15}$$

For the occurrence of the fluid film slippage on the lower contact surface in the outlet zone, it should also be satisfied that $\Delta u_{b,o}>0$ [27].

According to Eqs. (4) and (15), it is then obtained that:

$$\frac{u\eta}{\tau_{sb}h_o}>\frac{4\psi\lambda_h^3+3\lambda_h^2+1}{4\psi\lambda_h^3+4} \tag{16}$$

Since it should be satisfied that $q_v<0$, it is then obtained from Eq. (8) that:

$$-\frac{h_o^2\lambda_h^2\tau_{sb}(\psi\lambda_h+1)}{2\psi\lambda_h^3\eta+2\eta}<0 \tag{17}$$

While, according to the condition that $-\lambda_1 l_2>0$, it is obtained that:

$$\frac{3\tau_{sb}\psi(\lambda_h^2-1)l_2}{2h_o(\psi\lambda_h^3+1)}>0 \tag{18}$$

Obviously, when $\lambda_h>1$ i.e. in the present bearing, equations (17) and (18) are always satisfied. Thus, according to Eqs. (13) and (16), the condition for the present bearing is Eq. (16).

3.5. Friction Coefficient

According to Eqs. (10), (11) and (14), the friction coefficients on the upper and lower contact surfaces in the bearing are respectively:

$$f_a = \frac{\left|\tau_{a,i} l_1 + \tau_{b,i} l_2\right|}{w} = \frac{2}{3}\alpha \frac{\left|2\psi^2\lambda_h^3 - \psi\lambda_h^3 + 3\psi\lambda_h^2 + 3\psi\lambda_h - \psi + 2\right|}{\psi(\lambda_h^2 - 1)} \quad (19)$$

and

$$f_b = \frac{\left|\tau_{b,i} l_2 + \tau_{b,o} l_1\right|}{w} = \frac{4}{3}\alpha \frac{(\psi\lambda_h^3 + 1)(\psi + 1)}{\psi(\lambda_h^2 - 1)} \quad (20)$$

3.6. Normalization

For generality, the parameters are normalized as follows [17]:

$$\lambda_h = \frac{h_i}{h_o}, \psi = \frac{l_1}{l_2}, \quad X = \frac{x}{l_1 + l_2}, \quad \alpha = \frac{h_o}{l_1 + l_2}, \quad U = \frac{u\eta}{\tau_{sb} h_o},$$
$$P = \frac{p h_o}{\tau_{sb}(l_1 + l_2)}, \; W = \frac{w h_o}{\tau_{sb}(l_1 + l_2)^2}$$

3.6.1. For the Present Bearing

In the present bearing, the dimensionless pressure in the outlet zone is:

$$P = \frac{3}{2} G(\lambda_h^2 - 1)\left(\frac{1+\psi}{\psi} X + 1\right), \text{ for } -\psi/(\psi + 1) \le X \le 0 \quad (21)$$

The dimensionless pressure in the inlet zone is:

$$P = \frac{3}{2} G(\lambda_h^2 - 1)[1 - (1+\psi)X], \text{ for } 0 \le X \le 1/(\psi + 1) \quad (22)$$

The dimensionless load carried by the bearing is:

$$W = \frac{3}{4} G(\lambda_h^2 - 1) \quad (23)$$

3.6.2. For the Convention Bearing

For the conventional hydrodynamic step bearing, the dimensionless carried load is [5], [17]:

$$W_{conv} = \frac{3U\psi(\lambda_h - 1)}{(\lambda_h^3\psi + 1)(\psi + 1)} \quad (24)$$

The friction coefficients on the upper and lower contact surfaces in the bearing are respectively [17]:

$$f_{a,conv} = \alpha \frac{\left|\frac{U}{\lambda_h} - W_{conv}(\psi + 1)\lambda_h^2 + \psi[U + W_{conv}(\frac{1}{\psi} + 1)\lambda_h]\right|}{W_{conv}(\psi + 1)\lambda_h} \quad (25)$$

and

$$f_{b,conv} = \alpha \frac{\left|\frac{U}{\lambda_h} + W_{conv}(\psi + 1)\lambda_h^2 + \psi[U - W_{conv}(\frac{1}{\psi} + 1)\lambda_h]\right|}{W_{conv}(\psi + 1)\lambda_h} \quad (26)$$

4. RESULTS

4.1. Pressure Distribution

Figure **2** shows the dimensionless pressure distributions in the present bearing for different λ_h when $\psi = 0.6$, which is a typical design value. The pressures are respectively linearly distributed in the inlet and outlet zones in the present bearing. As the formulating equations show, both the dimensionless pressure as well as the dimensional pressure in the present bearing are independent on the sliding speed *u* but intimately related to the geometrical parameters and the contact-fluid interfacial shear strength τ_{sb} on the moving surface.

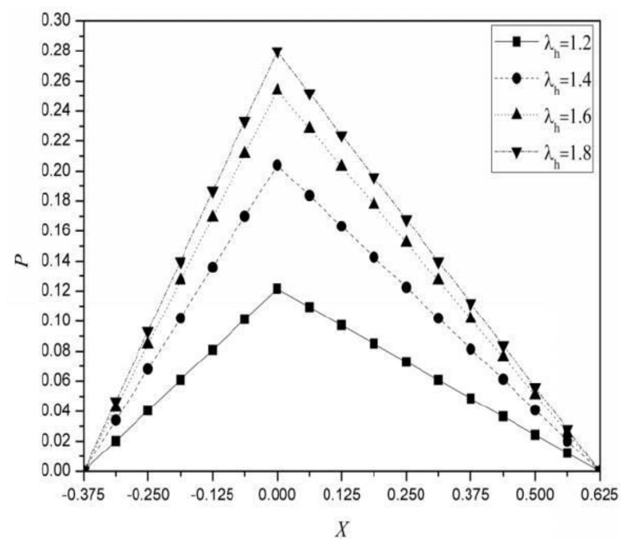
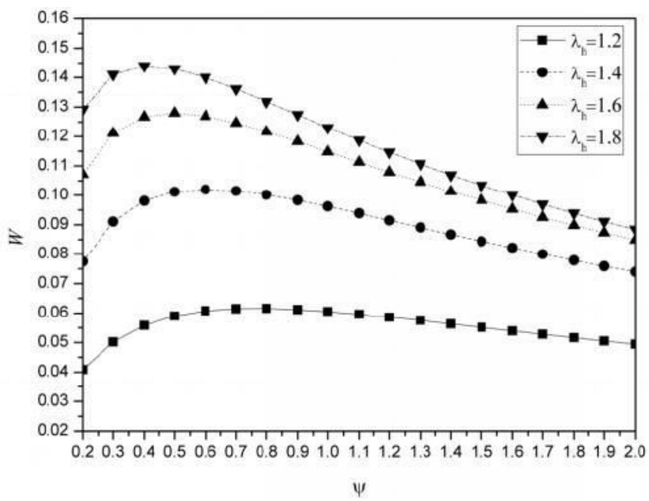


Figure 2: Dimensionless pressure distributions in the present bearing when $\psi = 0.6$.

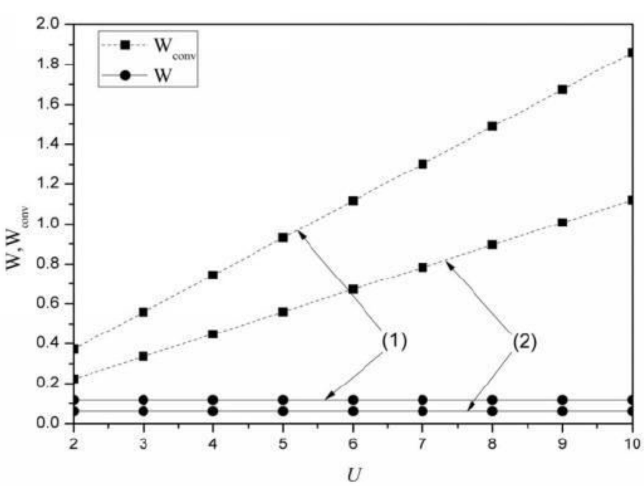
4.2. Carried Load

Figure **3a** plots the dimensionless carried load *W* of the present bearing against the geometrical parameter ψ for different λ_h. For a given λ_h, there is the optimum value of ψ which yields the highest value of *W*. The ψ value deviating from this optimum one considerably reduces the load-carrying capacity of the bearing.

Figure **3b** compares the carried load W of the present bearing with that (W_{conv}) of the conventional hydrodynamic step bearing for different dimensionless sliding speeds U respectively for two sets of the geometrical parameter values. As shown, the carried load (W as well as w) of the present bearing is independent on the sliding speed u but intimately related to both the geometrical parameters and the contact-fluid interfacial shear strength τ_{sb} on the moving surface, however the carried load of the conventional hydrodynamic step bearing is significantly linearly increased with the increase of the sliding speed u for a given operating condition. The comparison shows that for the same operating condition, the carried load of the present bearing is normally significantly lower than that of the conventional hydrodynamic step bearing especially for high sliding speeds U; However for a low sliding speed U, the difference will be much reduced.



(**a**)



(1) $\psi = 0.8$, $\lambda_h = 1.2$; (2) $\psi = 0.5$, $\lambda_h = 1.5$

(**a**)

Figure 3: Dimensionless carried loads of the bearing.

4.3. Friction Coefficient

Figures **4a** and **b** respectively show the friction coefficients (f_a and f_b) on the upper and lower contact surfaces in the present bearing for different λ_h and ψ when $\alpha = 2.5 \times 10^{-4}$, which is a representative bearing parameter value. These friction coefficients are independent on both the sliding speed u and the contact-fluid interfacial shear strength τ_{sb} on the moving surface. They are intimately related to the geometrical parameters λ_h, ψ and α of the bearing. It is shown that the friction coefficient in the present bearing is normally on the scale 0.001 and even lower. The studied bearing is thus obviously energy-conserved.

Figures **4c** and **d** respectively compare the friction coefficients (f_a and f_b) on the upper and lower contact surfaces in the present bearing with those ($f_{a,conv}$ and $f_{b,conv}$) in the conventional hydrodynamic step bearing for two sets of the geometrical parameter values when $\alpha = 2.5 \times 10^{-4}$. It is shown that in the same operating condition, for a low dimensionless sliding speed U, the friction coefficient in the present bearing is much lower than that in the conventional hydrodynamic step bearing; However, for a high dimensionless sliding speed U, the friction coefficient in the present bearing is significantly higher than that in the conventional hydrodynamic step bearing.

Figures **4c** and **d** as well as Figure **3b** suggest the potential application value of the present mode of the bearing for reducing the friction but without over loss of the load-carrying capacity in the operating condition of low dimensionless sliding speeds. They also suggest the necessity of preventing the occurrence of the interfacial slippage on the moving contact surface in a hydrodynamic step bearing in the condition of high dimensionless sliding speeds, for avoiding the over loss of the load-carrying capacity and the significant increase of the friction coefficient.

The reduction of the load-carrying capacity of the present bearing by the interfacial slippage effect may be due to the occurrence of the interfacial slippage on the moving surface in the bearing inlet zone, which reduces the fluid flow rate entrained into the bearing especially at a high sliding speed. In itself, the occurrence of the interfacial slippage limits the shear stress and the friction force on the bearing surface; For a low sliding speed, it will result in a significant reduction of the friction coefficient of the bearing because of the modest reduction of the load-carrying capacity of the bearing. However, for a high sliding

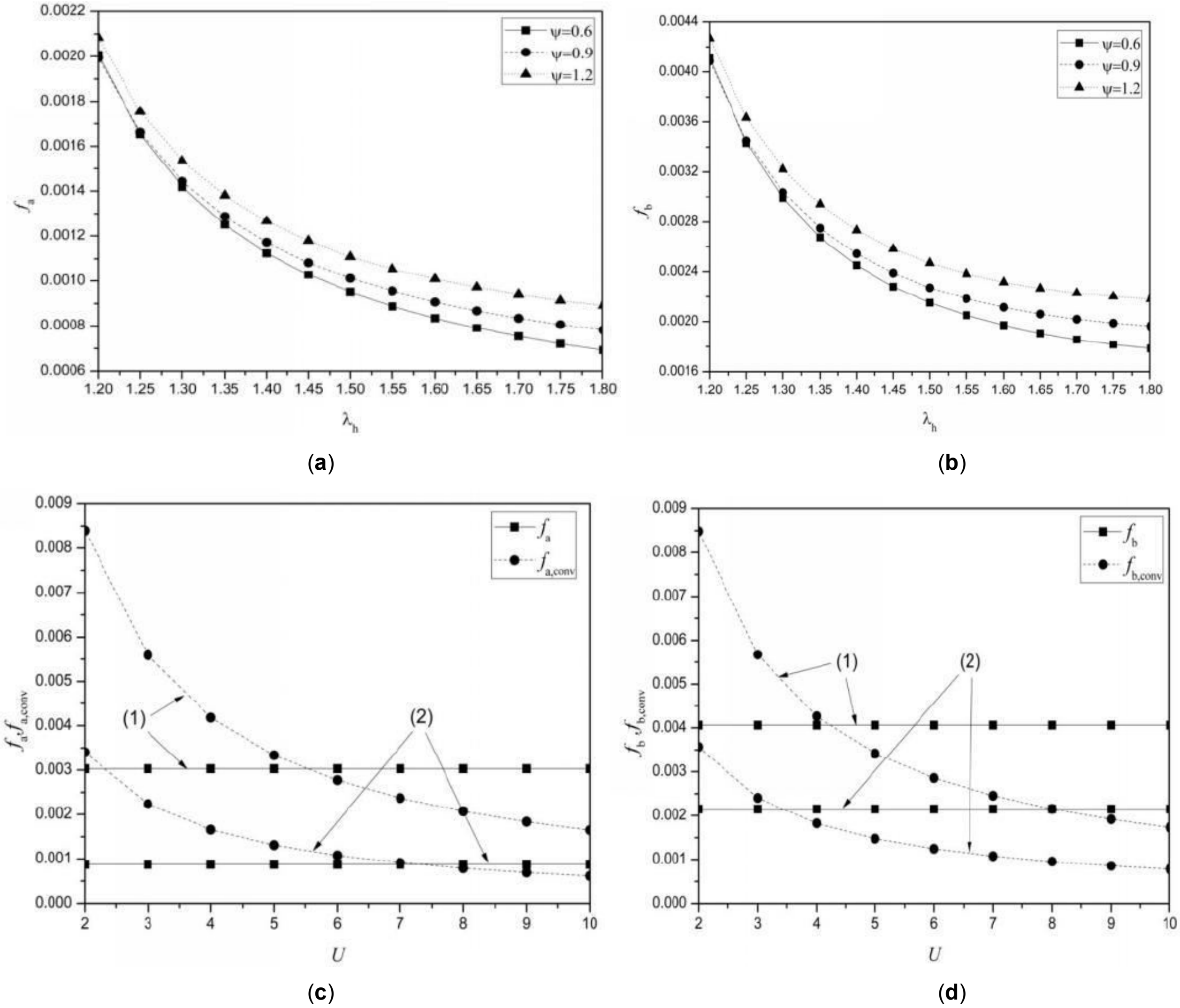


Figure 4: Friction coefficients (f_a and f_b) respectively on the upper and lower contact surfaces in the present bearing and their comparisons with those ($f_{a,conv}$ and $f_{b,conv}$) in the conventional hydrodynamic step bearing for the same operating conditions, $\alpha = 2.5 \times 10^{-4}$. (1) $\psi = 0.8$, $\lambda_h = 1.2$; (2) $\psi = 0.5$, $\lambda_h = 1.5$.

speed, it will otherwise increase the friction coefficient of the bearing because of the great reduction of the load-carrying capacity of the bearing.

5. CONCLUSIONS

The performance of a new mode of hydrodynamic step bearing with the interfacial slippage is analytically investigated based on the interfacial limiting shear strength model. In this bearing, the interfacial slippage occurs on the whole moving surface but is absent on the stationary surface. The calculation results show that for a low dimensionless sliding speed, the friction coefficient of this mode of bearing is much lower than that of the conventional hydrodynamic step bearing in the same operating condition; However for a high dimensionless sliding speed, the case is opposite. The present study suggests the potential application value of such a mode of bearing for reducing the friction in the condition of low dimensionless sliding speeds without over loss of the load-carrying capacity. Nevertheless, for a high dimensionless sliding speed, the present mode of bearing works much worse than the conventional hydrodynamic step bearing because of the much increased friction coefficient and the greatly reduced carried load.

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