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Experimental Research and Mathematical Model of Drag Torque in Single-plate Wet Clutch

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Abstract: Reduction of drag torque is one of important potentials to improve transmission efficiency. Existing mathematical model of drag torque was not accurate to predict the decrease after oil film shrinking because of the difficulty in modeling the flow pattern between two plates. Flow pattern was considered as laminar flow and full oil film in the gap between two plates in traditional model. Subsequent equivalent circumferential degree model presented an improvement in oil film shrinking due to centrifugal force, but was also based on full oil film in the gap, which resulted difference between model prediction and experimental data. The objective of this paper is to develop an accurate mathematical model for the above problem by using experimental verification. An experimental apparatus was set up to test drag torque of disengaged wet clutch consisting of single friction and separate plate. A high speed camera was used to record the flow pattern through transparent quartz disk plate. The visualization of flow pattern in the clearance was investigated to evaluate the characteristics of oil film shrinking. Visual test results reveal that the oil film begins to shrink from outer radius to inner radius at the stationary plate and only flows along the rotating plate after shrinking. Meanwhile, drag torque decreases sharply due to little contact area between the stationary plate and the oil. A three-dimensional Navier-Stokes (N-S) equation based on laminar flow is presented to model the drag torque. Pressure distributions in radial and circumferential directions as well as speed distributions are deduced. The model analysis reveals that the acceleration of flow in radial direction caused by centrifugal force is the key reason for the shrinking at the constant feeding flow rate. An approach to descript flow pattern was presented on the basis of visual observation. The drag torque predicted by the model agrees well with test data for non-grooved wet clutch. The proposed model enhances the precision for predicting drag torque, and lays down a framework on which some subsequent models are developed.

Key words: wet clutches, drag torque, mathematical model, laminar flow

1 Introduction

Reduction of drag torque in disengaged wet clutches is the focus of transmission research because it is one of the potentials for the improvement of transmission efficiency. It is necessary to reveal the flow pattern through the clearance between the rotating plate and stationary plate and build a precise mathematical model to predict the characteristics of drag torque.

The flow between two rotating disks has been studied extensively. Fluid entrains axially and exits out from the disk surface radially. The solution to this question has been discussed by BATCHELOR[1] and STEWARTSON[2]. Their interest is the flow pattern of the core fluid between the two plates. Analogous solutions have been previously discussed in Refs. [3–4]. The authors attempted to solve Navier-Stokes equation to deduce the distribution of pressure and speed between the two disks. Problems above have many similarities to the disengaged open wet clutch. The difference mainly focuses on flow rate for lubrication. The flow rate feeding in wet clutch is appropriate, but two rotating disks in above researches are full of fluid. So, previous work laid down a foundation on mathematical model of drag torque in wet clutch.

The traditional model[5] based on laminar flow and full oil film in the clearance is poor with predicting drag torque at high speed region shown in Eq. (1):

$$M=\frac{N\mu\pi(R_2^2-R_1^2)\omega r_m^2}{h}, \tag{1}$$

where M is drag torque of wet clutch, μ is viscous of oil, ω is the rotating speed, r_m is mid-radius of friction plate, h is clearance of the clutch, R_2 is the outer radius of the friction plate, R_1 is the inner radius of the friction plate, N is the number of frictional surfaces.

Eq. (1) can only represent the rising portion of a typical drag torque curve at the low speed region when the clearance is full of oil film.

SCHADE[6], FISH[7], LLOYD[8] empirically tested drag toque trends regarding geometric parameters, such as groove patterns, depths, clearance as well as flow rate.

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They also indicated the importance of disk waviness in reducing drag torque.

KATO, et al[9] and HASHIMOTO, et al[10], deduced the equations which described the pressure distribution and the drag torque in the oil film between clutch plates as follows:

$$p(r) = p(R_2) + \frac{\mu Q_0}{2\pi r_m h^3 N G_r}(R_2 - r) - \frac{\rho\omega^2}{2}(R_2^2 - r^2)\left(f + \frac{1}{4}\right), \tag{2}$$

$$M = 2\pi N\int_{R_1}^{R_2} \frac{\mu\omega r^3}{h}(1 + 0.001\ 2Re_h^{\ 0.94})\mathrm{d}r\,, \tag{3}$$

where p is pressure distribution, r is the radius of friction plate, G_r is turbulence coefficient, ρ is density of the oil, f is turbulence coefficient, Re_h is Reynolds number base on clutch clearance between two frictions, Q_0 is actual feeding flow rate.

KATO, et al[9], first explained oil film shrinking between the separate plate and the friction plate due to centrifugal force. In their work, an equivalent radius where full oil film begins to shrink was presented and the peak of drag torque was predicted well. They laid down a framework on which drag torque model was developed.

In 2003, YUAN, et al[11], simulated a three-dimensional, steady-state, two-phase flow using the commercial computational fluid dynamics(CFD) code FLUENT. Their research result suggested that surface tension played an important role in the incipience aeration. Their work revealed that aeration after oil film shrinking was the main reason to result in the decrease of drag torque, because viscosity of the oil was much less than that of the air. In 2007, also YUAN, et al[12] and FAIRBANK[13] introduced surface tension of the fluid into the pressure equation to evaluate the effect on drag torque prediction.

CHINAR, et al[14], developed a mathematical model of single-phase and verified it by using FLUENT and experiments. The numbers of grooves, the depth of groove and the clearance between disks were discussed to understand their effects on the drag torque through 3D CFD models. The flow rate is very important in determining the pressure distribution along the disks and affects the drag torque. They also pointed out that aeration was the most important factor that reduced drag torque due to low viscosity.

HU, et al[15], deduced the pressure distribution and speed distribution in the clearance based on Navier-Stokes(N-S) equation considering laminar flow. Their analysis revealed that oil flow acceleration in radial direction caused by centrifugal force was the key reason for the shrinking of oil film at the constant feeding flow rate. A circumferential degree was presented to calculate the integral area after oil film shrinking.

It is clear that flow rate and speed are most important factors that affect the drag torque from the reviews above. But there also exists disputation on oil film flow pattern after shrinking. YUAN[12] attained the flow pattern which shrinks from outer radii to inner radii through CFD. His result means the wetted area between two disks is circumferential. HU, et al[15], assumed a circumferential degree to indicate the reduction of drag torque. The prediction agrees well with test result for multi-plate wet clutch, but not for single-plate wet clutch.

In order to well understand the flow pattern in the clearance of wet clutch, a small experimental rig was set up to test drag torque in single-plate wet clutch and to verify the flow pattern in the clearance through visual research in this work. Experimental apparatus is set up and the results are discussed in section 2. In section 3, we start with asymptotic analysis of N-S equations and the subsequent derivation of the lubrication model. Section 4 presents a method to describe the characteristic of drag torque after oil film beginning to shrink according to experimental visualization. Section 5 compares the model and experiment. Section 6 enlists the conclusion of this study.

2 Experimental Apparatus

A small experimental rig was set up to test no-grooved single-plate wet clutch. A high-speed camera visualizes the flow pattern in the clearance. Thermocouples monitor oil temperature as fluid enters the clearance. A volume flow meter monitors the flow rate into the clearance.

Fig. 1 shows the overall system configuration as a block diagram. A 3 kW inverter-type motor drives the rotating plate. Another plate is installed onto the stationary support through a torque sensor. Oil pump feeds oil into the clearance through a flow meter which displays the volume flow. The flow rate is precisely controlled by a displacement pump according to test conditions. A 1 kW electrical heater installed in the tank allows experiments with elevated oil temperature. A torque sensor with hollow shaft is added to avoid complicated machine for oil feeding. The torque sensor is capable of measuring in small range for single-plate wet clutch by use of Stain Bridge.

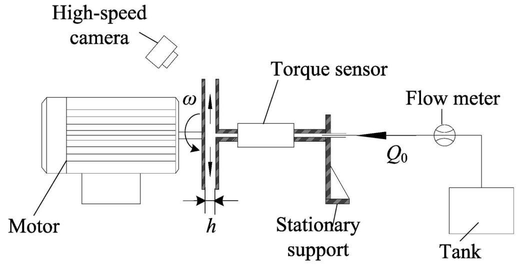


Fig. 1. Block diagram of experimental rig for wet clutch

Transparent quartz disk plate connected in motor allows visualization of flow pattern. The stationary disk is made of aluminum disk to which a real friction plate can be attached to simulate a real surface in wet clutch. This study focuses on the flow pattern between two disks and drag torque reduced curve. Comparison with aluminum disk and real friction plate as well as grooved disk and no-grooved disk is necessary to be experimentally tested, so we leave these studies to subsequent publication.

The geometric parameters of plates were summarized in Table 1, and the test conditions in Table 2. To minimize the influence of the inner circular region, both the rotating disk and the stationary disk have a 10 mm depth which is very large compared to 1 mm in the clearance. Therefore, the drag torque caused by inner circular region can be ignored. Test temperature and flow rate were controlled strictly. The repeatability of the experiments was also tested.

Table 1. Plate geometric parameters

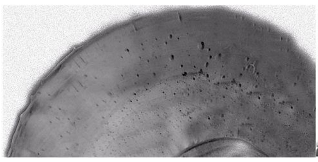
Outer radius R_2/mm	Inner radius R_1/mm	Clutch clearance h/mm
120	80	1

Table 2. Test conditions

Flow rate Q_0/(L • min^{-1})	Oil density ρ/(kg • m^{-3})	Oil viscosity μ/(Pa • s)	Temperature T/℃
3	872	0.066	30

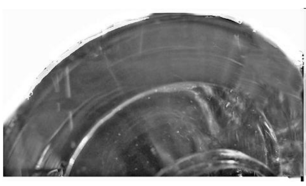
3 Experimental Result Discussion

The low flow rate causes the oil film to shrink as rotating speed increases because of continuity equation[15]. The experiment focuses on flow pattern as rotating speed increases. Fig. 2 shows the photos of flow pattern taken from camera.



(a) ω=200 r/min

(b) ω=500 r/min



(c) ω=730 r/min

(d) ω=1 400 r/min

(e) ω=2 200 r/min

Fig. 2. Plate-view flow pattern in the gap at different rotating speed

Fig. 2(a) shows plate-view flow between the rotating plate and the stationary plate at ω=200 r/min. At low rotating speed, screwed flow in the clearance appears, especially at the inner radius region. And it is found that the clearance is full of fluid. With increasing rotating speed, shown in Fig. 2(b) and Fig. 2(c), the oil film in outer region starts to shrink and aeration emerges in the outer region due to centrifugal acceleration and continuity equation. It is observed from Fig. 2(d) that screwed flow gradually disappears instead of radial flow. There is no oil attaching to stationary plate and the fluid flows radially only along rotating plate, whereas the fluid flowing near the rotating plate wall is more likely similar to the boundary layer flow over a single rotating plate. Cavity appears and flow pressure reduces caused by the radial acceleration of the fluid. In Fig. 2(e), the fluid flows radially due to strong centrifugal force and becomes oil fog outside the clearance. From above visual observation, rotating speed is the most important factor on flow pattern. On the other hand, flow pattern affects the drag torque through integral area calculation.

This work focuses on predicting model for drag torque. Visualization is useful to well understand the flow pattern. Ref. [15] has known that drag torque increases linearly before oil film shrinking due to full oil film in the clearance. This means single-phase flow as shown in Fig. 2(a) and Fig. 2(b). Models presented by Ref. [15] have revealed the reduction of drag torque due to the decreasing contact area between fluid and two plates. Either equivalent radius or equivalent circumferential degree can be used to describe the contact area shown in Fig. 3[15]. r_0 is radius where full oil film. But it is found that flow pattern after oil film shrinking is different from visualization. The fluid flows only along the rotating plate and no contract with the stationary plate as rotating speed increases, shown in Fig. 4. After attaining the flow pattern in the clearance, it is reasonable to build a precise mathematical model based on experimental observation.

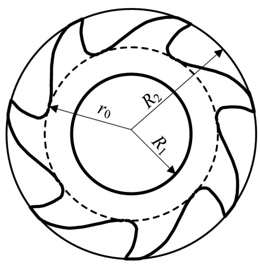


Fig. 3. Schematic diagram of partial oil film in the clearance

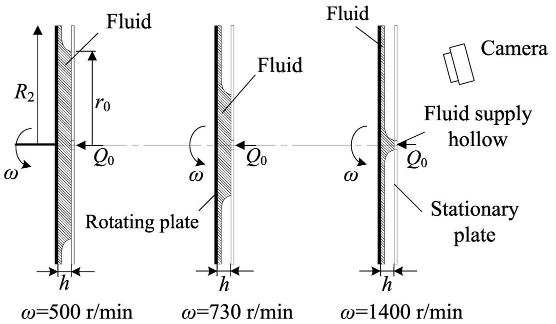


Fig. 4. Flow pattern in the clearance at different rotating speed

Fig. 5 shows experimental result of the drag torque at constant feeding flow rate Q_0=3 L/min. Compared with traditional model, equivalent circumferential degree model is able to predict the peak of drag torque well and performs well for multi-plates wet clutches. But it is not precise in predicting the reduction of drag torque. Shown in Fig. 5, experimental result drops nearly to zero with rotating speed increasing. This means there is no shear stress on the plates. It is easy to conclude that no oil film leads to little shear stress. Actually it is inevitable that the ununiformity of each plate clearance emerges when rotating speed is increasing. Contact between the friction plate and the separate plate is also possible. Viscous torque and contact torque cause drag torque to reduce gradually which is likely to be predicted in Ref. [15]. It is necessary to model the reduction trend for single-plate wet clutch based on flow pattern in the clearance.

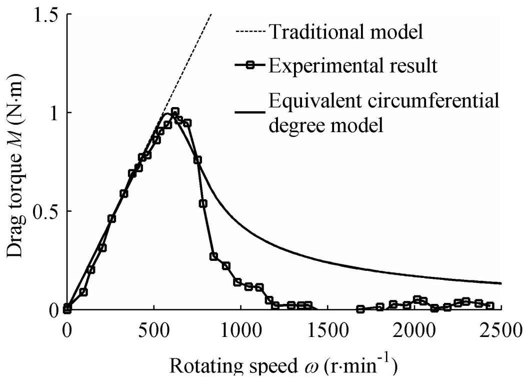


Fig. 5. Experimental result of drag torque

4 Mathematical Model

Under the actual conditions in disengaged wet clutch, flow is assumed to be incompressible, steady and laminar as well as axis symmetric. Gravity can be neglected. Hence, $\partial v_\theta/\partial t=\partial v_r/\partial t=\partial v_z/\partial t=v_z=0$. v_θ is the speed in circumferential direction, v_r is the speed in radial direction, and v_z is speed in axial direction.

In the steady state, incompressible flow in the clearance of non-grooved disengaged wet clutch is shown in Fig. 6.

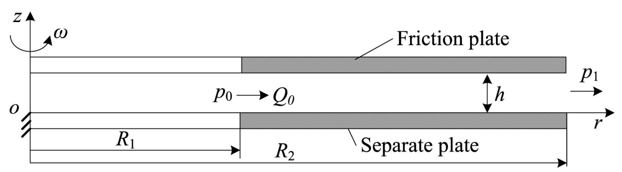


Fig. 6. Schematic of disengaged wet clutch

Navier-Stokes equations can be written as in cylindrical coordinates[16]:

$$\rho\left(v_r\frac{\partial v_r}{\partial r}-\frac{v_\theta^2}{r}\right)+\frac{\partial p}{\partial r}=\mu\frac{\partial^2 v_r}{\partial z^2},$$
$$\rho\left(v_r\frac{\partial v_\theta}{\partial r}-\frac{v_\theta v_r}{r}\right)=\mu\frac{\partial^2 v_\theta}{\partial z^2},\tag{4}$$
$$\frac{\partial p}{\partial z}=0,$$

where ρ is the density of fluid, μ is the viscosity of fluid, p is the pressure of fluid, r is radius of the friction plate.

Shown as Fig. 6, the boundary conditions for equations above can be written as:

$$\begin{cases} v_r(r,0)=0,\ v_r(r,h)=0, \\ v_\theta(r,0)=0,\ v_\theta(r,h)=\omega r, \\ p_2=p_0, p_1=0. \end{cases}\tag{5}$$

where p_1 is the exiting pressure at outer radius, p_2 is the flow pressure at inner radius, p_0 is the feeding pressure, h is axial gap.

Integrating Eq. (4) and considering the boundary conditions, we have the fluid radial speed distribution:

$$\begin{aligned} v_r=&\frac{1}{2\mu}\frac{\mathrm{d}p}{\mathrm{d}r}z(z-h)+\frac{\rho\omega^2 r}{12\mu h^2}z(h^3-z^3)+\\ &\frac{3\rho Q^2}{20\pi^2 r^3\mu h^6}z(h^5-2z^5+6z^4h-5z^3h^2), \end{aligned}\tag{6}$$

where z is the axial distance of the gap, Q is the ideal flow rate for single phase flow.

Ideal flow rate can be calculated through integration of radial speed as follows:

$$\begin{aligned} Q=&\int_0^{2\pi}\int_0^h v_r r\mathrm{d}z\mathrm{d}\theta=-\frac{\pi h^3 r}{6\mu}\frac{\mathrm{d}p}{\mathrm{d}r}+\\ &\frac{\pi\rho\omega^2 r^2 h^3}{20\mu}+\frac{9\rho Q^2 h}{140\pi\mu r^2}, \end{aligned}\tag{7}$$

Substituting Eq. (7) into Eq. (6) and rearranging, we have radial speed distribution:

$$
\begin{aligned}
v_r &= -\frac{3Q}{\pi r h^3} z(z-h) + \frac{\rho\omega^2 r}{60\mu h^2} \times \\
&(9z^2h^2 - 4zh^3 - 5z^4) + \frac{\rho Q^2}{140\pi^2 r^3 \mu h^6} \times \\
&(27h^4z^2 - 6zh^5 - 42z^6 + 126z^5h - 105z^4h^2).
\end{aligned} \tag{8}
$$

Shown in Eq. (8), radial speed distribution can be divided into three parts. The first part is caused by steady flow between two fixed plates; its flow speed becomes smaller when radius increases. The second part is caused by centrifugal force, its speed increases with the rise of speed and radius. The third part is caused by flow inertia. It can't be ignored when flow rate is very large. Therefore, centrifugal inertia and flow inertia accelerate the radial speed of oil in the clearance. As rotating speed increases, centrifugal speed distribution becomes dominant so that radial speed is accelerated strongly, shown as Fig. 2(e).

Radial pressure distribution can be obtained from the integration of Eq. (7) along the radial direction with the boundary conditions:

$$
\begin{aligned}
p &= \frac{6\mu Q}{\pi h^3}\ln\frac{R_1}{r} - \frac{3\rho\omega^2}{20}(R_1^2 - r^2) - \\
&\frac{27\rho Q^2}{140\pi^2 h^2}\left(\frac{1}{r^2} - \frac{1}{R_1^2}\right).
\end{aligned} \tag{9}
$$

Eq. (9) indicates that rotating speed and flow rate affect the pressure distribution, especially when Q and ω are very large. Eq. (9) is composed of four pressure distributions, the first caused by feeding pressure, the second caused by the steady flow between two fixed plates, the third caused by the centrifugal inertia, and the fourth caused by fluid flow inertia.

Under boundary conditions, the pressure of outer radius is zero. The feeding pressure can be rewritten as follows:

$$
\begin{aligned}
p_0 &= \frac{6\mu Q}{\pi h^3}\ln\frac{R_1}{R_2} - \frac{3\rho\omega^2}{20}(R_1^2 - R_2^2) - \\
&\frac{27\rho Q^2}{140\pi^2 h^2}\left(\frac{1}{R_2^2} - \frac{1}{R_1^2}\right).
\end{aligned} \tag{10}
$$

So there is a strong phenomenon of suction at the inner radius region as rotating speed is very large, which is running as a centrifugal pump.

In most cases, the actual feeding pressure is approximately the same as the exiting pressure of the clutch, i.e., $p_1 = p_2$. So Eq. (10) can be used to find ideal flow rate for full oil film in the clearance. Eq. (11) means that the larger ω is, the larger Q is needed to maintain single-phase flow in the clearance:

$$
\begin{aligned}
Q &= \frac{\dfrac{-6\mu}{\pi h^3}\ln\dfrac{R_1}{R_2}}{0.039h^{-2}(R_1^{-2} - R_2^{-2})} + \\
&\frac{\sqrt{\left(\dfrac{6\mu}{\pi h^3}\ln\dfrac{R_1}{R_2}\right)^2 - \dfrac{81\rho^2\omega^2}{700\pi^2 h^2}(R_1^{-2} - R_2^{-2})(R_2^2 - R_1^2)}}{0.039h^{-2}(R_1^{-2} - R_2^{-2})},
\end{aligned} \tag{11}
$$

Actually, the feeding flow rate Q_0 is constant regardless of the rising of rotating speed. The crossing area in radial direction must shrink when radial speed increases under the law of continuity. At the lower rotating speed, the centrifugal force is small and the radial speed decreases as the flow approaches the outer radius. Thus, there exists full oil film in the clearance. As rotating speed increases, the flow starts to accelerate in radial direction since the centrifugal force becomes dominant which drives the oil outward. As a result, oil film shrinking happens in the clearance in order to satisfy continuity, as shown in Fig. 4.

In order to evaluate the influence on drag torque after oil film starts to shrink at high speed region, a new model must be presented for the description of the effective oil film for drag torque based on continuity. Based on Fig. 2, there is no contact area between friction plate and separate plate after oil film shrinking. Hence, area at where flow is single-phase is only effective to the calculation of drag torque.

A radius r_0 is used to calculate the single-plate area shown in Fig. 2(a). The max value of radial speed can be rewritten as from Eq. (8) differential:

$$
\begin{aligned}
\frac{\partial v_r}{\partial z} &= -\frac{3Q}{\pi r h^3}(2z-h) + \frac{\rho\omega^2 r}{60\mu h^2} \times \\
&(18zh^2 - 4h^3 - 20z^3) + \frac{\rho Q^2}{140\pi^2 r^3 \mu h^6} \times \\
&(54h^4z - 6h^5 - 252z^5 + 630z^4h - 420z^3h^2).
\end{aligned} \tag{12}
$$

Then we derive the max value of the radial speed $v_{r\max}(r)$ from $\partial v_r/\partial z=0$. Actually we can have $v_{r\max}(r)$ through numerical solutions due to high order in Eq. (12). The following calculations of $v_{r\max}(r)$ in this work are solved from numerical solutions.

From the definition of r_0, we have:

$$
v_{r\max} = \frac{Q_0}{2\pi r_0 h}. \tag{13}
$$

To solve r_0 value, Eq. (13) can be rewritten as:

$$
r_0 = \frac{Q_0}{2\pi v_{r\max} h}. \tag{14}
$$

The shear stress on the rotating plates for single-phase flow is:

$$\tau_{\theta z} = \mu \frac{\partial v_\theta}{\partial z}|_{z=h} = \frac{\mu \omega r}{h}. \tag{15}$$

The drag torque on each of the rotating plate then can be expressed as follows:

$$M = 2\pi \int_{R_1}^{r_0} r\tau_{z\theta} r \mathrm{d}r = \frac{\pi\mu\omega}{2h}(r_0^4 - R_1^4). \tag{16}$$

All the analysis above is based on laminar flow. Thus, Reynolds number must be checked in order to evaluate this model. We have:

$$Re = \frac{v_{r\max} d}{\nu}, \tag{17}$$

where d is the hydraulic diameter, ν is the viscosity of oil.

5 Validation of the Model

Comparisons were made between the model predictions and the test results in order to validate the model under the conditions of Table 1 and Table 2. The traditional model and equivalent circumferential degree model were examined so as to better demonstrate the improvements of new model.

The radial speed distribution in clearance when rotating speed is 500 r/min is shown in Fig. 7, 730 r/min and 1 400 r/min are shown in Fig. 8, Fig. 9, respectively. The points at which radial speed are more than zero indicate that centrifugal force becomes dominant which drives the oil outward. From these points, the oil film starts to shrink. Oppositely, there exists a full oil film in the clearance. Fig. 9 means that there is no full oil film in the clearance and the fluid flow just along the rotating plate.

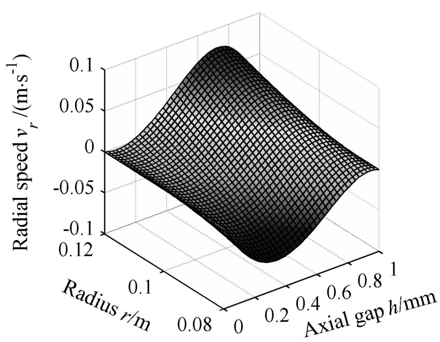


Fig. 7. Radial speed distribution in the gap at 500 r/min

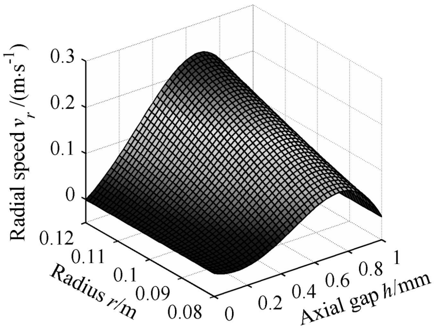


Fig. 8. Radial speed distribution in the gap at 730 r/min

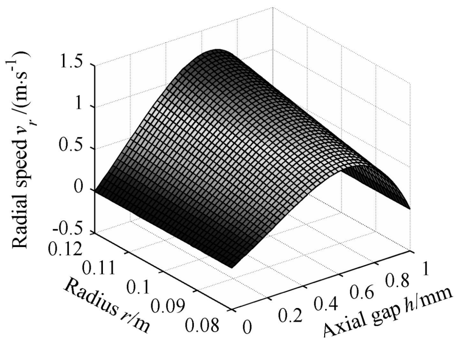


Fig. 9. Radial speed distribution in the gap at 1 400 r/min

Eq. (10) reveals that it need more flow rate to keep the clearance full of oil film as the rotating speed increases.

Shown in Fig. 10 low feeding flow rate is the key reason that leads to the oil film shrinking. Actually, low feeding flow rate is enough for lubricating the clutch, such as 3 L/min in Table 1. So there must be partial oil film at high speed region.

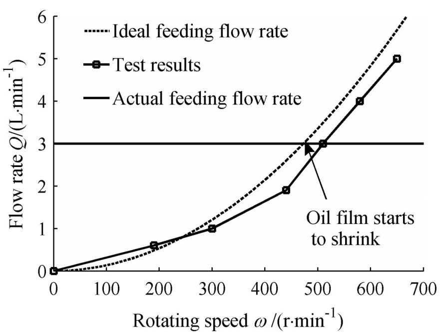


Fig. 10. Feeding flow rate for single-phase flow

Fig. 11 shows r_0 radius of the oil film predicted by above model from Eq. (13), as a function of rotating speed. As the clutch speed increases from 0 r/min to 1 000 r/min in Table 1, there is a full oil film in the clearance until the speed reaches about 500 r/min, where the outer radius starts to drop. At 600 r/min, there is only full oil film when radius is about 0.13 m. Above 800 r/min, there is no full oil film in the clearance. Clearly, the speed at which oil film radius starts to drop correlates to the peak of drag torque according to Eq. (15).

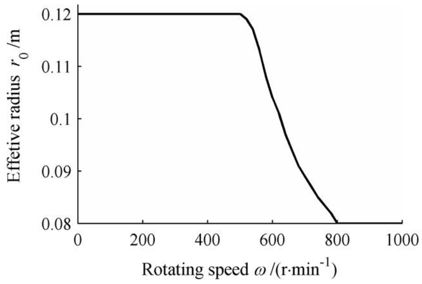


Fig. 11. Oil film outer radius predicted by new model

Drag torque can be obtained from the integration of Eq. (15). Fig. 12 shows the comparison of drag torque predicted by the new model with the test condition in Table 1. It is clear that the speed at which oil film starts to decline correlates to the peak of drag torque. Above 500 r/min, drag torque decreases due to the partial oil film. The traditional model follows the slope of the initial portion of the drag torque curves at low speed region. Since the drag torque increases monotonically with the rise of speed in traditional model, it is impossible to predict at high speed region beyond 500 r/min. On the other hand, equivalent circumferential degree model also predicts well with peak value but not for the decreasing tendency, which well agrees with multi-plates clutch validated by test[15]. The improvement of the new model is subject to predicting the decrease tendency for single-plate clutch. And the new model reveals the real characteristic of the flow in the clearance.

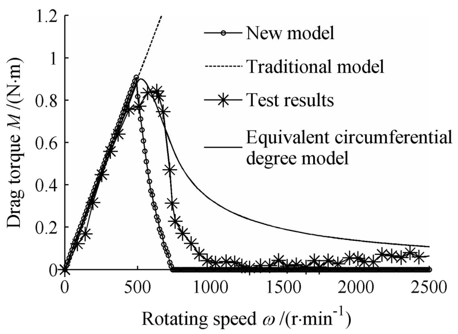


Fig. 12. Comparison of drag torque between model prediction and test result

Shown in Fig. 12, it is clear that a difference still exists between the model and test result, especially at 500–1 000 r/min. It is likely to be caused by surface tension of the fluid in the clearance. Surface tension force prevents oil film to shrinking, but the effect of surface tension is limited compared with centrifugal force when rotating speed increases. Since surface tension is so complex that contact angles between air, fluid, and friction plate must be considered, we leave it to be studied by CFD in next work.

Through repeated experiments, it is found that drag torque increases slowly after reaching its minimum shown in Fig. 12. From visual observation, fluid flow out of the rotating plate directly, which has changed into globule. It is not able to model this phenomenon by using above hypothesis. So multi-phase flow dynamics must be introduced to explain the effect of surface tension, because it plays an important role in the lubrication analysis of wet clutch and thrust bearing at high speed region.

6 Conclusions

(1) The experimental results show that the fluid flows just along the rotating plate and has no contact with stationary plate after oil film starting to shrink. So drag torque drops sharply after reaching its peak due to the absence of shear stress between the rotating plate and the stationary plate.

(2) The model and test data indicate that actual feeding flow rate and rotating speed play most important role in the drag torque peak. Low actual feeding flow rate is a good measure to reduce drag torque peak in engineering design.

(3) Based on continuity, the conservation of flow rate has been used to solve r_0. The relationship between oil film area and drag torque at high speed region is well described.

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