Modeling and Parametric Study on Drag Torque of Wet Clutch

Heyan Li, Qi Jing and Biao Ma

Abstract In order to analyze the power torque characteristics of wet clutch produced under the disengaging state and reduce the drag torque of wet clutch, a flow rate equation under full film state was deduced based on the Navier-Stokes equations and a new equation calculated oil film equivalent radius was derived according to the relationship of oil flow rate and volume between import and export, whereafter the drag torque model considering the friction gap non-uniform was established based on the Newton internal friction theorem with front formula. The performance of simulation shows that the drag torque increased monotonically with the rise of speed in low speed. But when the speed increased to a certain value, the drag torque began to decrease. Then, two evaluation indexes for drag torque were presented which were peak value of drag torque and the corresponding critical speed. Subsequently, the sensitivity for the effect of clutch parameters on drag torque was calculated. The conclusion of research have some reference value to the design of wet clutch.

Keywords Wet clutch · Drag torque · Parameter sensitivity · Friction pair · Non-uniform

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1 Introduction

Wet clutches are frequently used in the drive trains of many modern vehicles. The study of wet clutch drag characteristics have important meaning to improve the transmission efficiency. Therefore, it is necessary to develop an accurate mathematical model for the drag torque.

A large amount of work has been done in this area, however, none model can provide predictions of engineering-level accuracy. In 1974, Lloyd [1] studied the influence factor of drag torque by experiment, then Kato [2] utilized Hashimoto's [3] equations to derive drag torque model which considered the deficit of the oil film firstly. Whereafter, the study of drag torque was focused on the reason and model of oil film shrink. Yuan [4] and Paul [5] thinks the surface tension is the main reason to reduce oil film. Chinar [6] derived a mathematical model from Navier-Stokes equations with verification using FLUENT and experiments. Biao [7] derived the traditional drag torque equation, and studied the effect of oil flow rate on drag torque by experiment. Jibin [8] deduced the drag torque for laminar flow using an equivalent circumferential degree to calculate the integral area after aeration. Zhang Zhigang and Xiaojun [9] derived an oil film model considering the surface tension. Shihua [10] also derived an oil film equivalent radius model by the identity that the max value of oil radial speed appeared in oil film external border.

From the above review of the previous study, it is clear that the model of oil film equivalent radius considering shrink in high seed has not been to date. So this paper introduce a new way to calculate the shrinking oil film radius to establish drag torque model. And the mechanism of non-uniform distribution of the friction pair gap was considered.

2 Mathematical Model

The wet clutch disengagement is shown in Fig. 1. Under the actual conditions in disengaged wet clutch, we assume some conditions as follows.

(1) Oil is assumed to be incompressible and steady state, (2) Flow in the wet clutch clearance is laminar and symmetrical, (3) Gravity can be neglected, (4) Wet clutch plate was no-grooved, (5) Friction plate and counter plate have no glide with its surface layer oil.

According above presupposition, the boundary conditions can be written as follows:

$$\begin{cases} v_r(r,0) = v_r(r,h_i) = 0 \\ v_\theta(r,0) = 0, v_\theta(r,h_i) = \omega r \end{cases} \tag{1}$$

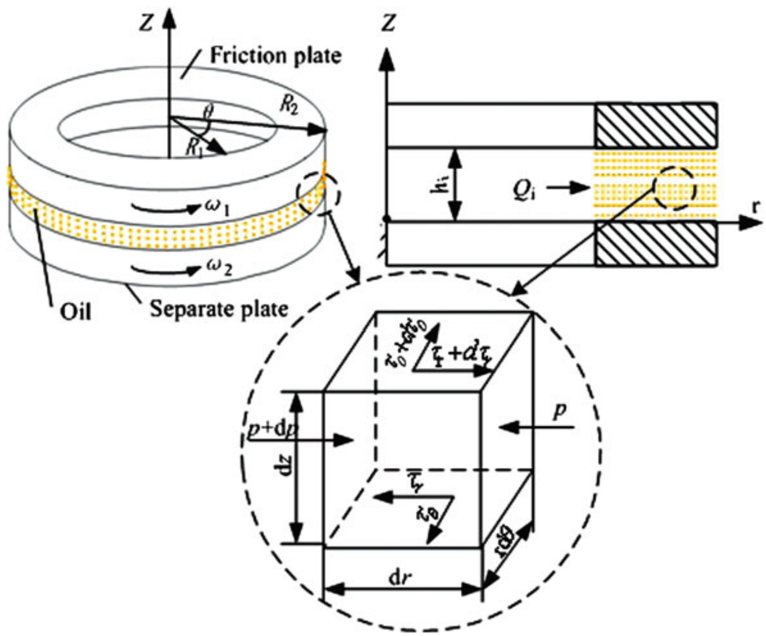


Fig. 1 Schematic diagram of wet clutch disengagement

where v_r is the oil speed in radial direction, v_θ is the oil speed in circumferential direction, h_i is axial clearance, ω is the rotating speed of friction plate relative to counter plate.

The oil motion in the radial direction between the plates is essentially driven by the centrifugal force, whereas the viscous tend to resist this motion. The relative rotating motion causes the drag torque due to the oil viscosity.

According to the assumptions, the Navier-Stokes equations in the viscous fluid [11] in cylindrical coordinates can be can be simplified to:

$$\begin{cases} -\dfrac{\partial p}{\partial r} + \mu \dfrac{\partial^2 v_r}{\partial z^2} = \rho(v_r \dfrac{\partial v_r}{\partial r} - \dfrac{v_\theta^2}{r}) \\ \mu \dfrac{\partial^2 v_\theta}{\partial z^2} = \rho(v_r \dfrac{\partial v_\theta}{\partial r} + \dfrac{v_\theta v_r}{r}) \\ \dfrac{\partial p}{\partial z} = 0 \end{cases} \tag{2}$$

The oil pressure gradient equation can be deduced by Eqs. (1) and (2), as follows:

$$\frac{dp}{dr} = \frac{27\rho Q^2}{70\pi^2 h_i^2 r^3} + \frac{3\rho\omega^2 r}{10} - \frac{6\mu Q}{\pi r h_i^3} \tag{3}$$

Table 1 Conditions of clutch simulation

R_1/mm	R_2/mm	h_i/mm	Q_i/Lñmin^{-1}	μ/Pañs	ρ/kgñm^3
86	125	0.6	1/1.5	0.086	875

Radial pressure distribution of oil in friction pair can be obtained from the integration of Eq. (3) along the radial direction with the boundary conditions:

$$p(r) = -\frac{27\rho Q^2}{140\pi^2 h_i^2} r^{-2} + \frac{3}{20}\rho\omega^2 r^2 - \frac{6\mu Q}{\pi h_i^3}\ln r + C \tag{4}$$

In the existing researches [8–10], the import pressure of friction pair is approximately the same as the export pressure of friction pair. But in the actual clutch, they are not equivalent. So in this paper, the pressure difference between import and export of friction pair was considered as follows:

$$p(R_1) - p(R_2) = \Delta p \tag{5}$$

Substituting Eqs. (4) into (5), so the flow rate equation under full film state can be deduced based on the Navier-Stokes equations considering the centrifugal effect, the result as follow:

$$Q = \frac{\frac{6\mu}{\pi h_i^3}\ln\frac{R_1}{R_2}}{\frac{27\rho}{70\pi^2 h_i^2}\left(R_2^{-2} - R_1^{-2}\right)} + \frac{\sqrt{\left(\frac{6\mu}{\pi h_i^3}\ln\frac{R_1}{R_2}\right)^2 - \frac{81\rho^2\omega^2\left(R_2^{-2} - R_1^{-2}\right)\left(R_1^2 - R_2^2\right) - 540\rho\left(R_2^{-2} - R_1^{-2}\right)\Delta p}{700\pi^2 h_i^2}}}{\frac{27\rho}{70\pi^2 h_i^2}\left(R_2^{-2} - R_1^{-2}\right)} \tag{6}$$

Equation (6) is the needing feed flow rate equation to maintain full oil in clearance between the plates. It is clear that the needing feed flow rate is relate to rotating speed. A trend chart of needing feed flow rate can be plot by simulation. The simulation conditions were summarized in Table 1.

The result of simulation as follows:

As shown in Fig. 2, the needing feed flow rate for full oil film increase with the rise of the rotating speed of friction plate, but the actual feed flow is constant regardless of the rising of rotating speed. So the oil film will shrink when the needing feed flow rate exceed the actual feed flow rate in high speed. Just like Fig. 3.

The principle for oil film shrinking can be explain as that the clutch clearance can not be full of oil when actual feed flow rate below the needing feed flow rate. Define R_s as the equivalent radius of oil film.

When $Q_i \geq Q$, $R_s = R_2$.

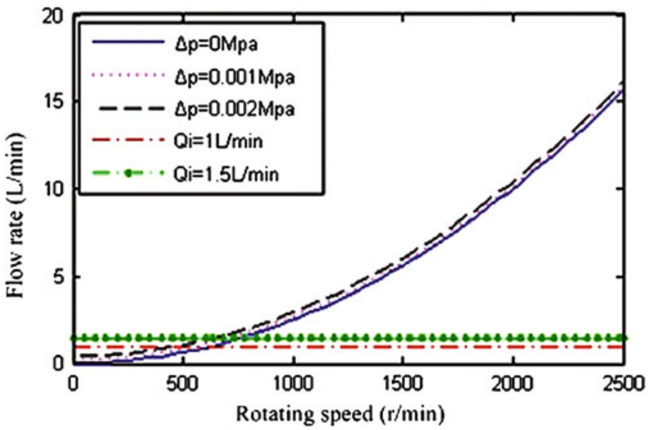


Fig. 2 Needing feed flow rate curve for full oil film

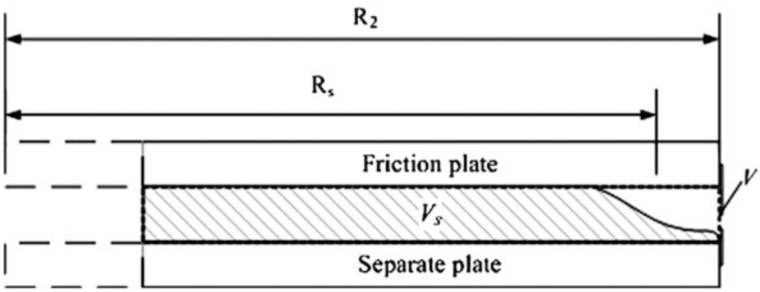


Fig. 3 Schematic diagram of partial oil filming the clearance

When $Q_{\mathrm{i}} < Q$, according the relationship of oil flow rate and volume between import and export, the equivalent radius of oil can be written as follows:

$$R_{\mathrm{s}} = \sqrt{\frac{Q_i}{Q} R_2^2 + R_1^2 \left(1 - \frac{Q_i}{Q}\right)} \tag{7}$$

A trend chart of the equivalent radius can be plot by simulation. The simulation conditions as Table 1. The result of simulation as follows:

As shown in Fig. 4, the equivalent radius predicted by above model is a function of the clutch speed. When the speed was low, there is a full oil film and the equivalent radius equals the outer radius. When the speed was high, the equivalent radius starts to drop.

As shown in Fig. 5 shows, a tiny circle was taken in friction pair to analyze the shear stress.

The integral outer diameter of shear stress was the equivalent radius.

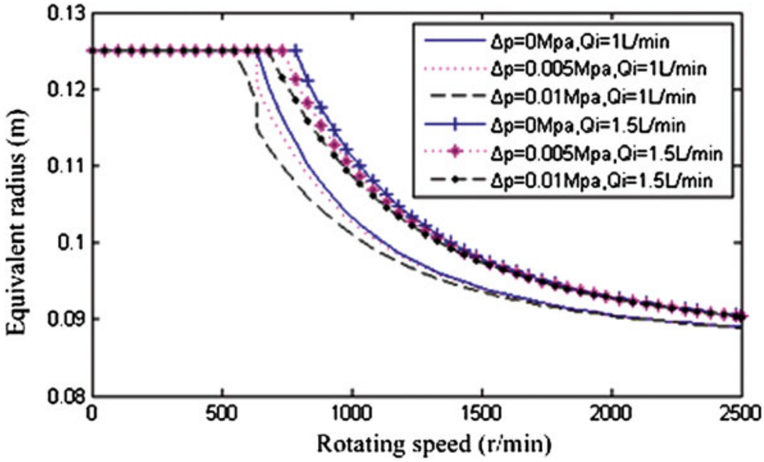
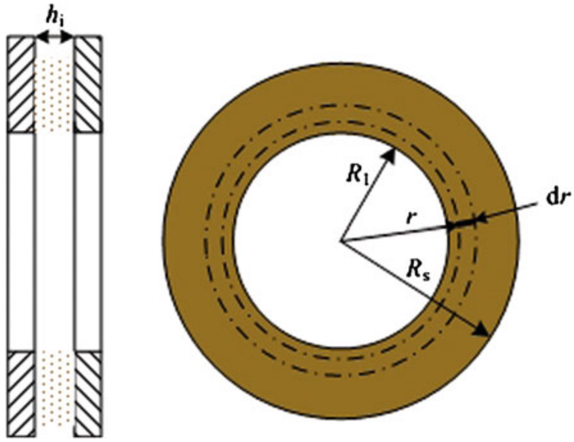


Fig. 4 Equivalent radius curve with speed

Fig. 5 Calculation principle of drag torque



According the Newton internal friction theorem, the drag torque of each friction pair then can be expressed as follows:

$$T_i = \frac{\pi\mu\omega}{2h_i}\left(R_s^4 - R_1^4\right) \tag{8}$$

A trend chart of the drag torque can be plot by simulation. The simulation conditions as Table 1. The result of simulation as follows:

As shown in Fig. 6, the drag torque increased monotonically with the rise of speed in low speed. But when the speed increased to a certain value, the drag torque began to decrease. In high speed, the drag torque decreased with the rise of the pressure difference and increased with the rise of the oil flow rate.

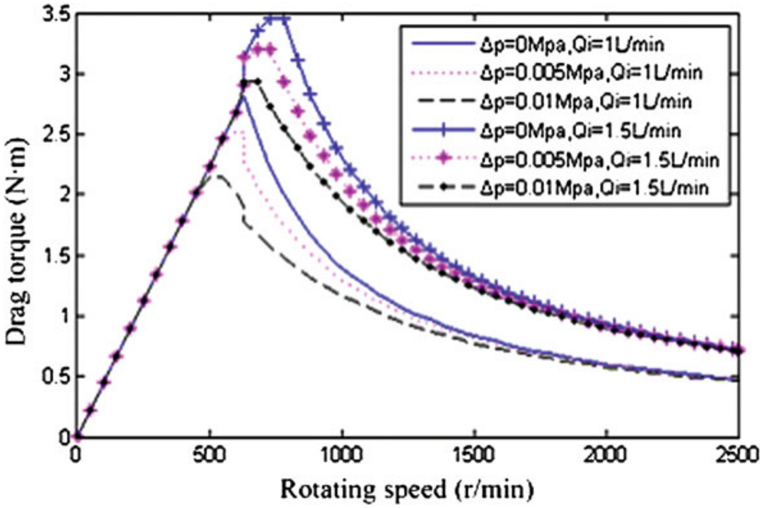


Fig. 6 Drag torque of single friction pair

In the existing researches of drag torque [8–10], the friction gap thickness of each friction pair in clutch was considered uniform. But it they were non-uniform in actual clutch. So there introduced a non-uniform coefficient to improve model. The results as follows:

$$T = \frac{Z\pi\mu\omega}{2h_0}\left(R_s^4 - R_1^4\right)\delta_T \tag{9}$$

Where h_0 is the average friction gap thickness, δ_T is the non-uniform coefficient.

3 Validation oftheModel

In order to validate the mathematical model above, A special experimental rig which can eliminate the influence of bearing loss was set up to test the drag torque of wet clutch. The text bench as follows.

As shown in Fig. 7, a 200 kW inverter-type motor drives the test box of wet clutch. Transient sensor of rotational speed and torque were used to measure the needing date. The test conditions were summarized in Table 2.

When non-uniform coefficient δ_T was 1.6, the predicted drag torque from the model agree well with experimental data. The results as follows:

As shown in Fig. 8, the drag torque model considered friction gap non-uniform could better simulate the actual clutch compared with the uniform model.

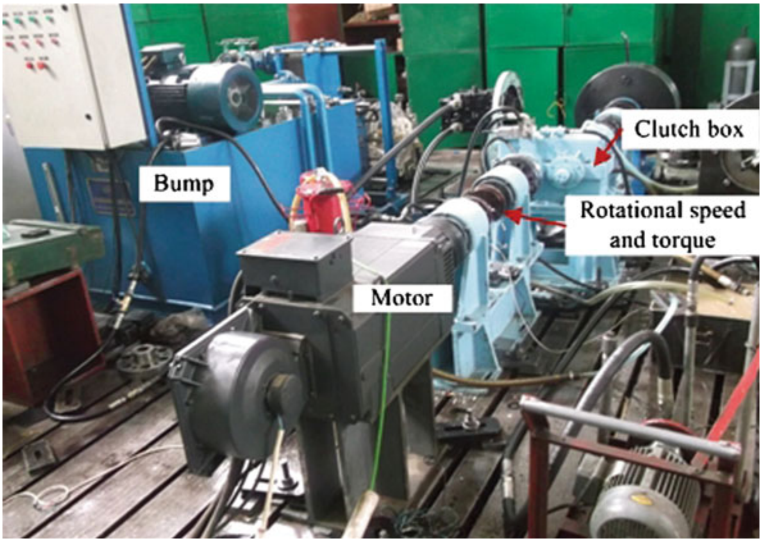


Fig. 7 Photograph of the test bench

Table 2 Conditions for clutch test

R_1/mm	R_2/mm	h_i/mm	Q/Lñmin^{-1}	μ/Pañs	ρ/kgñm^{-3}	Z
86	125	0.6	1.5	0.062	875	4

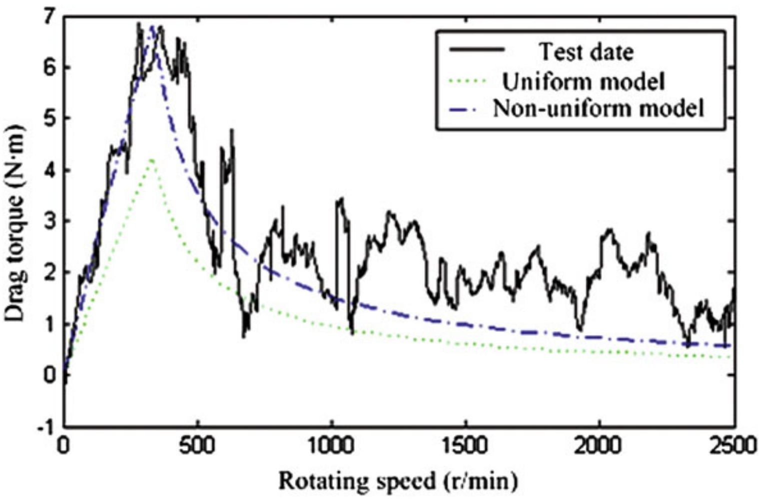


Fig. 8 Comparison of drag torque between model prediction and test result

Table 3 Input parameters of clutch

输入参数	Z	r_0/mm	h_i/mm	μ/Pañs	ρ/kgñm^{-3}	Q_i/Lñmin^{-1}
x_0	5	102	0.5	0.05	850	6
x_{max}	10	170	0.3	0.09	890	12
x_{min}	1	68	1	0.03	800	3

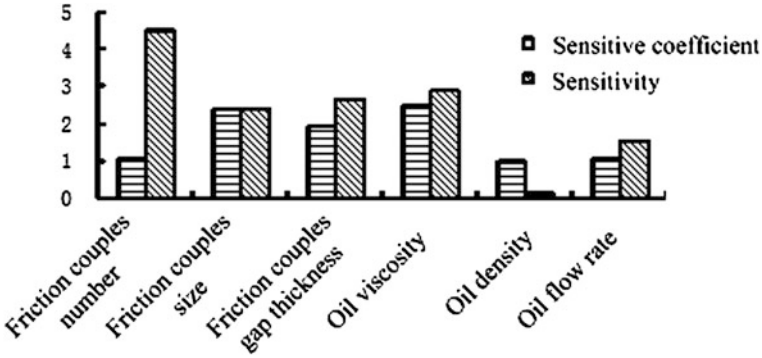


Fig. 9 The sensitive coefficient and sensitivity of clutch parameters

4 Sensitivity Analysis

According to the characteristic of drag torque, two evaluation indexes for drag torque were presented which is peak value of drag torque and the corresponding critical speed. When design the clutch system, there were many parameters need to consider. The drag torque was different with the select order of clutch parameters. So it is necessary to study the sensitivity that the clutch parameters influence the drag torque. This paper used the sensitivity analysis method [12].

Sensitive coefficient η_{SR} was defined as the ratio of the output rate percentage and the output rate percentage and input parameter percentage, just as follows:

$$\eta_{SR} = \frac{\left|\frac{f(x_0+\Delta x)-f(x_0)}{f(x_0)}\right| \cdot 100\,\%}{\left|\frac{\Delta x}{x_0}\right| \cdot 100\,\%} \tag{10}$$

Where x_0 is input parameter, η_{SR} is sensitive coefficient.

The sensitivity of input parameter was obtained by the sensitive coefficient multiplying the weight of the input parameter, the results as follows:

$$\eta_{SS} = \eta_{SR} \cdot \frac{x_{\text{Max}} - x_{\text{Min}}}{x_0} \tag{11}$$

Where x_{max} is upper limit of input, x_{min} is lower limit of input, η_{SS} is sensitivity.

According to the above Eqs. (10) and (11), the sensitive coefficient and sensitivity could be calculated. The input parameters were summarized in Table 3.

As shown in Fig. 9, the sensitivity for the effect of clutch parameters on drag torque was calculated, the result was oil viscosity, friction couples size, gap thickness, friction couples number, oil flow rate, oil density according to the influence degree of design parameters on drag torque form high to low.

5 Conclusions

The drag torque model considered friction gap non-uniform could better simulate the actual clutch compared with the uniform model. The sensitive coefficient and sensitivity of clutch parameters was obtained, it would have some reference value to reduce drag torque when designing wet clutch.

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