The influence of inertia resistance on the drag torque in the wet multi-disk clutch with splined connected restriction

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Abstract. A theoretical model of friction plate is established by using Resal theorem, and the inertia resistance of the friction plate is obtained from this theoretical model. Focusing on the bias phenomenon in friction pairs, a new model is investigated for analyzing the bias state under low speed difference considering about the gyro effect. As for the gap shrinkage under high speed difference, the negative pressure contraction model for the friction pairs is built by analyzing the oil film and the pressure between plates. Based on that, the influences of the inertia resistance of the friction plates for these two models are also discussed. Afterwards the gap dynamic change between friction pairs is investigated in the whole variation range of the relative speed. Finally, an improved model, considering the influence of the gap dynamic change between friction pairs, is proposed to simulate the drag torque. The results obtained from the simulation and test data indicate that, under the same lubrication condition, the drag torque in the wet multi-disk clutch increases at first and then decreases and finally increases with the rise of the rotational speed difference. Furthermore, the main factors influencing the drag torque are the bias of plates in the low speed difference (0~1000r/min) and the gap shrinkage in the medium and high speed difference (>1000r/min) respectively. From the comparison between test data and the simulation results obtained from the improved model, the average relative error is only 6.34% at medium and high speed difference (>1000r/min), which can greatly improve the accuracy for the estimation of the drag torque in wet multi-disk clutch.

1. Introduction

The wet multi-disk clutch is an important part of transferring torque in the vehicle transmission, and the drag torque caused by the viscous shearing force of lubricating oil is an energy dissipation pattern in the separation state. Because the viscous torque is related to the clutch engagement and separation characteristics, it is very crucial to accurately study and calculate the value of drag torque and its influencing factors. Moreover, as one of the important indexes to measure the power loss in the clutch, the precision of the established computational model has important influence on the accuracy of the power loss analysis.

For the research on the torque of the wet clutch, some results has been achieved. Lloyd [1] explored and put forward the main factors that influence the torque through the test method. Kato Y [2] and Hashimoto H [3] deduced the formula of oil film pressure distribution and the model of the torque in
the wet clutch. Yuan Y [4] and Walker P [5] believed that surface tension is the main cause of the decrease of oil film area. At the same time, a lot of progress had been made in the study of the torque in China. Ma B [6] based on Newton's internal friction principle, deduced the classical model of the torque. Zhang Zhigang, Zhou Xiaojun [7], Yunshua, Peng [8] and Xiang Changle, Zhang Ying [9] established torque considering the degree of lubricant film coverage, which is improved respectively from the view of surface tension, lubricating oil radial velocity and oil film pressure distribution. Yang Likun [10], Jing Qi [11] and Liu Jikai [12] improved the model of torque model respectively from the view of the non-parallelism of friction pairs not parallel degree, the angle of the multi-pairs clearance spacing inhomogeneity and the groove form of the friction plates. Then one improved method of the drag torque model considering the effect of friction pairs spacing is proposed, which can reflect the change rule of the initial increment and the drop of the drag torque affected by the speed difference. However, compared with the actual test value of the low speed difference, the calculated value of the model is smaller, and it has a certain difference with the increase of the torque with the high speed difference.

In this paper, based on the floating support structure and the fixed lubrication flow constraint of the clutch, the bias phenomenon between the friction pairs caused by the gyro effect of rotating components and the negative pressure between the disks in high speed difference are fully considered. Then, the kinetic model of the friction plate is established by using Reasl theorem, and the influence of inertia resistance on the friction plate at different rotational speed is obtained. What is more, the modified torque model is verified by experiments, so a more accurate calculation model of drag torque is obtained, which is of great significance to the calculation of energy consumption in vehicle transmission system.

2. The application of Reasl theorem

2.1. The introduction of Reasl theorem

As shown in Fig.1, there is a fixed-axis rotational rigid body. OXYZ is a stationary coordinate system, and the X-axis is the rotation axis of the rigid body. The axis of the coordinate system Oxyz is along the inertial axis of this rotational body to the coordinate origin O. According to the Reasl theorem: The angular momentum of the rotational rigid body in a fixed spot is equal to the principal moment of the external force on the same spot, which can be expressed as:
\[ \vec{u} = \vec{\omega} \times \vec{L}_0 = \vec{M}_0 \]  \hspace{1cm} (1)

Where \( u \) is the main moment vector, \( \omega \) is the absolute angular velocity of the rotational rigid body under the coordinate system OXYZ, and \( L_0 \) is the angular momentum of the rigid body. In the coordinate system Oxyz, these above motion vectors can be decomposed into:

\[ \vec{\omega} = \omega_x \vec{i} + \omega_y \vec{j} + \omega_z \vec{k} \]  \hspace{1cm} (2)

\[ \vec{L}_0 = J_x \cdot \omega_x \vec{i} + J_y \cdot \omega_y \vec{j} + J_z \cdot \omega_z \vec{k} \]  \hspace{1cm} (3)

\( \omega_x, \omega_y \) and \( \omega_z \) are the angular velocities of rigid bodies along three inertial axes, respectively, and \( J_x, J_y \) and \( J_z \) are the inertia of rigid bodies along three inertial axes. Substituting (2), (3) into the above formula (1), we can obtain:

\[ \vec{u} = \vec{\omega} \times \vec{L}_0 = \left( \begin{array}{ccc}
\vec{i} & \vec{j} & \vec{k} \\
\omega_x & \omega_y & \omega_z \\
J_x \cdot \omega_x & J_y \cdot \omega_y & J_z \cdot \omega_z
\end{array} \right) \]

\[ = J_x \cdot \omega_z \cdot \omega_y \cdot \vec{i} + J_y \cdot \omega_x \cdot \omega_z \cdot \vec{j} + J_z \cdot \omega_x \cdot \omega_y \cdot \vec{k} \]

\[ -J_x \cdot \omega_x \cdot \omega_y \cdot \vec{k} - J_y \cdot \omega_y \cdot \omega_z \cdot \vec{i} - J_z \cdot \omega_z \cdot \omega_x \cdot \vec{j} \]

\[ = (J_x \cdot \omega_x \cdot \omega_y - J_y \cdot \omega_y \cdot \omega_z) \cdot \vec{i} + (J_y \cdot \omega_y \cdot \omega_z - J_z \cdot \omega_z \cdot \omega_x) \cdot \vec{j} + (J_z \cdot \omega_z \cdot \omega_x - J_x \cdot \omega_x \cdot \omega_y) \cdot \vec{k} \]

Then the formula (4) is the expression of the Resal theorem for the fixed-axis rotational rigid body.

2.2. The application to the friction plate model
The structure of the friction plate is shown in Fig 2. And the inner diameter of the friction plate is \( R_1 \), and the outer diameter is \( R_2 \), and the thickness of the friction plate is \( b \).

![Figure 2. Schematic diagram for friction plate](image-url)
The origin of the coordinate system OXYZ is the center of the friction plate. And the X-axis in this fixed coordinate system is the axis of the input gear shaft. The coordinate system Oxyz is the coordinate system which the axis is the inertial spindle of the friction plate, in which the Y-axis coincides with the y-axis, and the x-axis of the friction plate is $\alpha$ to the oblique angle of the X-axis.

Then the angular velocity of the friction plate, and the inertia moment along the inertia axis can be expressed as follows:

$$\omega_x = \omega \cdot \cos \alpha$$  \hspace{1cm} (5)

$$\omega_z = \omega \cdot \sin \alpha$$  \hspace{1cm} (6)

$$\omega_y = 0$$  \hspace{1cm} (7)

$$J_x = \frac{1}{2} \cdot m \cdot \left( R_y^2 + R_z^2 \right)$$  \hspace{1cm} (8)

$$J_z = \frac{1}{4} \cdot m \cdot \left( R_y^2 + R_z^2 \right) + \frac{1}{12} \cdot m \cdot b^2$$  \hspace{1cm} (9)

$$J_y = \frac{1}{4} \cdot m \cdot \left( R_y^2 + R_z^2 \right) + \frac{1}{12} \cdot m \cdot b^2$$  \hspace{1cm} (10)

Substitute (5)–(10) into (4), according to the Resal theorem:

$$\bar{u} = \bar{\omega} \times \bar{L}_0$$

$$= \begin{pmatrix} \hat{i} \\ \hat{j} \\ \hat{k} \end{pmatrix} \begin{pmatrix} \omega \cdot \cos \alpha & 0 & \omega \cdot \sin \alpha \\ 0 & \frac{1}{2} \cdot m \cdot \left( R_y^2 + R_z^2 \right) \cdot \omega \cdot \cos \alpha & 0 \\ \frac{1}{8} \cdot m \cdot \left( R_y^2 + R_z^2 \right) \cdot \omega \cdot \sin \alpha & \frac{1}{24} \cdot m \cdot b^2 & \omega^2 \cdot \sin (2\alpha) \cdot \hat{j} \end{pmatrix}$$

$$= \bar{M}_0$$

$$= 2 \cdot F_a \cdot \left[ \left( R_1 - \frac{b}{2} \tan \alpha \right) \cdot \sin \alpha + \frac{b}{2 \cos \alpha} \right] \cdot \hat{j}$$

Where the force $F_a$ is affected by the dynamic binding of the spline contact:

$$F_a = F_0 = \frac{1}{8} \cdot m \cdot \left( R_y^2 + R_z^2 \right) - \frac{1}{24} \cdot m \cdot b^2 \cdot \cos \alpha \cdot \omega^2 \left[ R_1 - \frac{b}{2} \tan \alpha \right] + \frac{b}{2 \sin 2\alpha}$$  \hspace{1cm} (12)
3. The gyro effect and inertia resistance at low speed

3.1. The gyro effect and the bias phenomenon

In the clutch system, the gear shaft and the friction plate are connected by the spline, and the rotating body including gear shaft and the friction with the same angular velocity is selected as the research object. Its force and motion diagram is shown in Fig.3. In Fig 3(a), the coordinate system OXYZ is a fixed coordinate system with the center of gravity for the input bearing of the gear shaft as the origin. And the axis of the coordinate Ox'y'z' is the inertial spindle of the coordinate origin O along the rotational coupling body. So the initial rotation axis of the friction plate is x'. The initial state of the clutch is not running and the two coordinate axes coincide. The rotational angular velocity of the friction piece and the gear shaft is \( \omega \), and the initial direction is along the positive direction of the x'-axis. The total gravity of the friction plate and gear shaft is G, and the distance between the center of gravity Ol and the coordinate origin O is l. The friction plate’s inner diameter is R1, and the outer diameter is R2. L is the angular momentum of the coupling body, namely its size L=J\( \omega \), and its direction is along the axis positive direction. Wherein J is the rotational inertia of the coupling body around the x'-axis.

![Figure 3. The force and motion of the coupling body](image)
When the clutch is not running, because of the center of gravity O1 is not in the center of the structure, this structure will incline, resulting in the axis and the x-axis not parallel, at this time the friction plate and the gear shaft are in an inclining state.

However, when the clutch begins to operate at a certain angular speed, the coupling body is no longer tilted, and the plane motion can be stabilized in the horizontal XOY.

When the friction plate rotates at angular velocity \( \omega \), based on the gyro effect correlation theory \([13]\), the dynamic analysis of the coupling body of the friction plate and the gear shaft is carried out. According to the known conditions, as shown in Fig.3 (a), the object of study is supported by gravity G and bearing in the z-axis FN, G and FN constitute a couple. The distance from the center of gravity to the origin is \( l \), then the direction of the moment is along the positive direction of \( y' \)-axis, and the moment is:

\[
M_g = Gl = mgl
\]  

In the formula (13), \( m \) is the integral mass of the coupling body and \( g \) is the local gravitational acceleration. In time, the torque \( Mg \) produces an angular momentum \( Mg \), and the direction is perpendicular to \( L \) and it is applied to the rotational body. The result is that it changes the direction of the \( L \) but it does not change its size, causing the coupling body to move in the direction of the Z-axis, as shown in Fig.3(b). To denote the angle between the coupling body axis and the X-axis, it is shown as:

\[
M_g \cdot \Delta t = L \cdot \Delta \phi
\]  

So the precession angular velocity \( \Omega \) is:

\[
\Omega = \frac{\Delta \phi}{\Delta t} = \frac{mgl}{J\omega}
\]  

3.2. The inertia resistance and the bias phenomenon

![Figure 4. Inertia resistance moment of friction plate](image-url)
As shown in Fig 4 is the bias schematic diagram of the friction plate. The bias angle is $\alpha$, according to the previous analysis, the friction piece at the spline place under a pair of equal size inertia resistance $F_a, F_b$, the value is:

$$F_a = \frac{1}{8} \cdot m \cdot \left( R_2^2 + R_1^2 \right) - \frac{1}{24} \cdot m \cdot b^2 \cdot \cos \alpha \cdot \omega^2$$

\[( 16 )\]

The expression of force indicates that the inertia resistance is proportional to the square of the angular velocity. With the increase of angular velocity, the resistance of friction plate at the spline is bigger and bigger, and the inertia resistance moment will hinder the bias phenomenon, and the effect of the moment can decrease even the bias angle $\alpha$.

Therefore, the bias phenomenon of friction plate at low rotational speed is more obvious, with the increase of rotational speed, the bias phenomenon is weakened by the effect of inertia resistance. And the speed is bigger, the friction plate rotates more stably.

3.3. The simulation of torque at low speed difference

As shown in Fig.5, it is the sketch of the gap and the surface micro-element diagram of the friction element when the offset phenomenon occurs.

![Figure 5. the gap between plates at bias status and the area element on the friction pair](image)

The gap between the various micro-elements can be expressed as:

$$h_i = h_o + r \cdot \sin \alpha \cdot \cos \beta$$

\[( 17 )\]

$r$ is the distance between the center of the micro-element and the origin of the friction plate, $\beta$ is the angle between $r$ and the negative direction of the axis, $\beta \in [0, 2\pi]$. $\alpha$ is the coupling body’s inclination angle, $\alpha \in [0, 0.3^\circ]$.

The paper [14, 15] has shown the expression of the torque of the lubricating oil film under the condition of ignoring the gap $h_i$ when the friction pair is fixed [15] (Traditional model):

$$T = \frac{Z\mu\Delta\omega}{2h_i} \left( R_n^4 - R_i^4 \right)$$

\[( 18 )\]
Z is the number of friction pairs; Rs (m) is the equivalent radius of contact oil film; \( \mu \) (Pa·s) is the hydrodynamic viscosity of lubricating oil (\( \mu = \upsilon \rho \)); \( \Delta \omega \) (rad/s) is the relative angular velocity of the friction plate and the steel plate, \( \Delta \omega = 2\pi \Delta n \). Substituting the formula (17) into the formula (18), the expression of the drag torque with an uneven gap due to the bias of the friction pairs is obtained:

\[
T = Z \mu \Delta \omega \int_{0}^{2\pi} d\beta \int_{R_{0}}^{R} \frac{r^{3}}{h_{0} + r \sin \alpha \cos \beta} dr
\]

Under the same simulation parameters in the reference [14, 15], as shown in Table 1, the traditional model and the bias model are simulated in the 0.5L/m in of the single pair lubrication flow and the \( \Delta n \) range of the speed difference is 0~2000r/min.

| Parameter   | Value   | Parameter   | Value   |
|-------------|---------|-------------|---------|
| \( R_{1} \)/mm | 86      | \( \mu \) (Pa·s) | 0.026   |
| \( R_{2} \)/mm | 125     | \( \rho \) (kg·m\(^{-3}\)) | 875     |
| \( h_{0} \)/mm | 0.65    | \( P_{b}, P_{2} \)/MPa | 0.02    |

The variation curves of the two models with torque simulation are shown in Fig 6:

**Figure 6.** Calculation results obtained from two drag torque models

From Fig. 6, it can be found that the bias phenomenon of friction pairs is more obvious when the offset angle is 0.3° and the speed difference is lower (<1000r/min), and the change trend is still firstly increasing and then minus. The smaller bias angle has little effect on the torque. The peak value of the torque calculated with the bias angle 0.3° is the 2~3 times of the peak value of the traditional model. When the speed difference is greater than 1000r/min, the simulation values of the two models are similar and the bias effect can be neglected.

4 The negative pressure at high speed difference

4.1 The mechanism of the gap shrinkage

Lubricant pressure distribution along radial can be expressed as [14, 15]:

\[
\text{Lubricant pressure distribution along radial:}
\]
The \( \nu (m^2/s) \) is the lubricating oil movement viscosity; \( r (m) \) is the radius of any point on the surface of friction pair to the rotation axis; \( Q_i (m^3/s) \) is the average lubrication flow of single pair; \( p_2 (Pa) \) is the outlet pressure of friction pairs.

Select the simulation parameters listed in Table 1, and select \( r = 86\text{mm}, 100\text{mm}, 120\text{mm} \), then get three kinds of curves of radial pressure \( p(r) \) with the speed difference \( \Delta n \) as shown in Fig.7:

\[
p(r) = \frac{6 \rho \nu Q_i h_o^3}{\pi h_o^3} \ln \frac{R_2}{r} - \frac{27 \rho Q_i^2}{140 \pi^2 h_o^3} \left( r^2 - R_2^2 \right) + \frac{1}{20} \rho (3 \Delta \omega^2 + 10 \omega_2 + 3 \omega^2) \left( r^2 - R_2^2 \right) + p_2
\]

The \( \nu (m^2/s) \) is the lubricating oil movement viscosity; \( r (m) \) is the radius of any point on the surface of friction pair to the rotation axis; \( Q_i (m^3/s) \) is the average lubrication flow of single pair; \( p_2 (Pa) \) is the outlet pressure of friction pairs.

As shown in Fig.7, the radial pressure \( p(r) \) increases gradually from positive to negative with the increase of \( \Delta n \). And the more near the rotation axis of the friction pair, the more the pressure decrease.

The clutch is working in the wet closed box environment, and the box is equipped with the respirator. So the ventilator keeps the relative pressure in the box is not more than 0.02MPa. The force analysis of the friction element in the process of rotation between the lubricant and the air pressure in the box is shown in Fig.8. As shown in Fig.8(a), the lubricant film cannot completely cover the surface of the friction pair when the speed difference is raised to a certain value. As shown in Fig.8(b), when the clutch speed difference continues to rise, the lubricant pressure \( p(r) \) in the friction pairs begins to appear negative [11]. And under the action of internal pressure of the box, it will push the floating part of the friction pairs to move from the platen side to the piston side, causing the friction pairs gap to shrink as shown in Fig.8(c).
4.2 The inertia resistance and the gap shrinkage

According to the literature [11, 16, 17], it can be concluded that the lubricant pressure does not change in the ho direction, i.e. the rotation axis. In combination with formula (20), the pressure $p(r)$ is integrated over the entire friction element surface, and the expression of lubricant force on the contact surface of the two friction elements comprising the same friction pair is obtained:

$$F = 3 \rho v_i Q \left( R_i^2 - R_e^2 - 2 R_i^2 \ln \frac{R_i}{R_e} \right) +$$

$$+ \frac{27 \rho Q}{70 \pi^2 k^2} \left\{ \frac{1}{2} \left( 1 - \frac{R_i^2}{R_e^2} \right) - \frac{R_e^2}{R_i^2} \right\} + \pi \rho_i \left( R_i^2 - R_e^2 \right) -$$

$$\frac{1}{40} \pi \rho (3 \Delta \varphi^2 + 10 \omega \dot{\varphi} + 3 \omega^2) \left( R_i^2 - R_e^2 \right)^2$$

The second item to the right of the equal sign of formula (21) is very small compared with other items, and the simplified formula for lubricating oil action is:

$$F = 3 \rho v_i Q \left( R_i^2 - R_e^2 - 2 R_i^2 \ln \frac{R_i}{R_e} \right) + \pi \rho_i \left( R_i^2 - R_e^2 \right) -$$

$$- \frac{1}{40} \pi \rho (3 \Delta \varphi^2 + 10 \omega \dot{\varphi} + 3 \omega^2) \left( R_i^2 - R_e^2 \right)^2$$

The outer surface of the friction pairs is expressed by the pressure force in the chamber:

$$F_b = \pi \rho_b \left( R_i^2 - R_e^2 \right)$$

Among them, the friction element is forced by the chamber pressure $p_b$ and it is approximately equal to the friction pairs outlet pressure $p_2$ namely $p_2 = p_b$.

Because of the negative pressure, the chamber pressure force $F_b$ is bigger than the lubricant pressure $F$. So that the two friction elements have opposite movement trends, leading to the friction pairs gap from the initial value of $h_0$ begin to shrink. Finally, the force equilibrium state as shown in Fig.8 (c) is achieved, at which point the average gap how between the frictional pairs is less than the initial value of $h_0$.

As shown in Fig.9 above, the friction plate is forced by the chamber pressure $p_b$, the oil film pressure $F$ and the inertia resistance $F_a$, $F_b$, resulting from the slight bias of the friction piece caused by the uneven force due to axial movement. Because of inertia resistance, the contact pressure between the
friction plate and the gear shaft at the spline connection increases, so the axial frictional resistance can be expressed as:

\[
F_r = 2\mu_k \left[ \frac{1}{8} m \left( R_i^2 + R_i^2 \right) - \frac{1}{24} m b^3 \right] \cos \alpha \cdot \omega^2 \sum_{i=1}^{2} \left( R_i - \frac{b}{2} \tan \alpha \right) + \frac{b}{\sin 2\alpha}
\]

(24)

Define the inertia resistance coefficient, then the formula (24) can be expressed as:

\[
F_z = \zeta \cdot \omega^2
\]

(25)

The expression of the mean gap \( h_{o*} \) of the friction pairs under the equilibrium state (i.e. \( F + F_z = F_b \)) by the formula (22), (23) and (24) is as follows:

\[
h_{o*} = \sqrt{\frac{3\mu Q \left( R_i^2 - R_i^2 - 2R_i^2 \ln \frac{R_i}{R_i} \right)}{4\pi \rho (3\Delta \omega^2 + 10\omega^2 + 3\omega^2 \left( R_i^2 - R_i^2 \right)^2 - \zeta \cdot \Delta \omega^2}}
\]

(26)

In view of the gap shrinkage of the friction pairs, how must not be bigger than the initial value of \( h_o \), so shrinkage gap how of the friction pairs can be expressed as the following:

When \( h_{o*} > h_o \), how = \( h_o \);  
When \( h_{o*} \leq h_o \), how = \( h_{o*} \).

Combined with the simulation parameters listed in Table 1, choose the menu pairs of lubrication flow 0.2L/min, 0.35L/min, 0.5L/min. And three kinds of lubricating oil pressure \( F \) under the different flow could be obtained. And the chamber of the pressure force \( F_b \) and the average gap how between the friction pairs with the change of \( \Delta n \) are shown in Fig 10:
According to the above analysis, when the speed difference is higher than 1000 r/min, the friction element is allowed to ignore the bias phenomenon, and the friction pairs gap distribution can be considered uniform.

As shown in Fig.10, the lubricant pressure $F$ is larger than $F_b$ in the lower speed difference ($<1000$ r/min), and the mean gap of the friction pairs keeps the initial value $h_0$ unchanged; With the increase of the speed difference $\Delta n$, the air pressure force $F$ begins to be less than $F_b$ and the corresponding gap is gradually reduced.

According to the reference [11], the equivalent radius in the finite lubricating oil volume is defined, and the equivalent radius $R_s$ will be changed with the gap shrinkage of the friction pairs, as shown in Fig.11:
According to Fig. 11, as the lubricating oil flow of single pair of Qi unchanged, and the friction pairs gap decreased from $h_0$ to $h_{ow}$, the corresponding equivalent radius $R_{S^*}$ can be obtained:

$$R_{S^*} = \sqrt{\frac{h_0 - h_{ow}}{Q_1}} \cdot R_2^2 + \left(1 - \frac{h_0 - h_{ow}}{Q_1} \right) \cdot R_1^2$$  

(27)

And there are:
- When $Q_i \geq Q$, $R_{S^*} = R_2$;
- When $Q_i < Q$, $R_{S^*}$ is required for the formula (15).

In the range of speed difference $\Delta n > 1000 \text{r/min}$, the expression of the torque model with the axial contraction of the mean gap is given as follows:

$$T = \frac{Zn\mu\Delta \omega}{2h_{ow}}\left(R_{S^*}^4 - R_1^4\right)$$  

(28)

Selecting the simulation parameters listed in Table 1, we can simulate the traditional and unbiased shrinkage models in the case of single pair lubrication flow 0.5L/min and $\Delta n$ range of speed difference is $0 \sim 5500 \text{r/min}$, and get the torque $T$ curve as shown in Fig. 12:
Figure 12. Comparison of calculation results obtained from two drag torque models

It is shown from Fig. 12 that the effect of the gap shrinkage of the friction pairs on the torque T in the high speed difference (>1000 r/min) is obvious. And the change trend is different between the traditional model and the bias model, when the speed difference is 3000 r/min, the calculated value of the shrinkage model with the torque is about 100 times that of the traditional model and the bias model. The results show that the negative pressure and the axial contraction of the gap have a significant effect on the torque at the high speed difference.

5. Experimental study in full speed range

5.1 Integrated model

Based on the above models, in the actual multiple-clutch structure, because of the clutch structure, friction plate groove and lubricating oil pressure [11], and other reasons, wet multi-disk clutch in the operation during separation will exist the phenomenon that the friction pairs gap is uneven distributed. The existing expression of a torque with an uneven distribution of the multiple pairs of gaps [11] is:

\[
T \approx \pi \mu \Delta \omega \left( R^4_1 - R^4_1 \right) \sum_{i=1}^{Z} \frac{1}{\delta_i}
\]

Among them, the \( \delta_i \) is the non-uniform coefficient of each gap of the friction pairs, \( \delta_i = h_i / h_0 \). \( h_i \) is the gap of each friction pairs (i=1,2 ...Z).

For the convenience of narration, the traditional model (i.e. the formula (18)), which considers the lubricant film covering state, is called Model 1. The existing model (i.e. (29)), which considers the inhomogeneity of the friction pairs, is called Model 2, and the model (i.e. (19)), which will only consider the single factor of the friction pairs bias phenomenon, is called Model 3. The model (i.e. (28)), which will only consider the single factor of gap shrinkage, is called Model 4.

Combined with the traditional Model 1 and Model 2, 3, 4, i.e. (18), (29), (19), (28), the improved model of the torque considering the effect of gap dynamic change, called Model 5, is as follows:

\[
T = \mu \Delta \omega \sum_{i=1}^{Z} \frac{1}{\delta_i} \int_{h_0}^{h_i} R^4 \frac{r^3}{h_{min} + r \sin \alpha \cos \beta} dr
\]
Among them, the improved gap howi expression considering the effect of gap dynamic change is obtained by the formula (17), (26):

\[ h_{\text{out}} = h_{\text{in}} + r \cdot \sin \alpha \cdot \cos \beta \]  

(31)

Select the simulation parameters listed in Table 1, and the Model 1 and the Model 5 are simulated in the single pair lubrication flow is 0.5L/min, and the Δn range of the speed difference is 0~5600r/min. And two kinds of torque T curves are obtained as shown in Fig.13:

![Figure 13. Calculation results of two drag torque models](image)

In Fig.13, the peak value of the torque T with the change of the Δn of the friction pairs is higher than that of the traditional model. The peak value of the modified model with the torque is more than 3 times times the peak value of the traditional model, and the improved model is larger than the traditional model simulation under the high speed difference.

5.2 Experimental methods

In order to verify the torque model of the wet clutch, a test platform for testing the torque is built. The actual layout of the test rig and the internal structure and diagram of the clutch box are shown in Fig.14. The test-bed is composed of two parts, the transmission test system and the hydraulic lubrication system. The transmission test system is mainly composed of motor, sensor of rotational speed and torque, test clutch bag and so on. The rotational speed torque sensor is connected with the motor and the clutch bag; and the clutch box adopts the first gear drive as the increasing mechanism, and the passive end steel-plate of the clutch is fixed with the box. Hydraulic lubrication system is mainly composed of pumping station, hydraulic pump, oil inlet, clutch oil bottom shell and oil out road. An oil flow sensor and a temperature sensor are installed on the inlet oil channel, and the oil temperature of the inlet is controlled on the range of 80~90°C.

The speed ratio of the speed up gears mechanism is 0.61 and the input motor is 123r/min~3429r/min, so the speed difference of the clutch is corresponding changed from 200r/min~5600r/min. Because of the small torque value of single pair, the experiment is designed to measure the change rule of multiple pairs, and the test clutch placed 7 pieces of friction and 7 pieces of dual steel, that is, 14 pairs.
The speed ratio of the speed up gears mechanism is 0.61 and the speed difference of the clutch can be correspondingly changed from 200r/min to 5600r/min. Because of the small torque value of single pair, the experiment is designed to measure the change rule of multiple pairs, and the test clutch placed 7 pieces of friction and 7 pieces of dual steel, that is, 14 pairs.

For the different total lubrication flow (3L/min, 5L/min, 7L/min), we combine with the test parameters given in Table 1 to carry out three groups of experiments. And then all the friction plates and steel sheet in the clutch are removed to carry out the no-load loss test of three groups in different lubrication flow.

5.3 Results and analysis

After deducting the test data of no-load with the same flow rate, the results show that the T curve with the change of clutch speed difference $\Delta n$ is obtained as shown in Fig. 15:
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1. When the lubrication flow rate is fixed, the change trend of the torque \( T \) always increases first and then decrease, which is the same as the initial simulation results of the improved model with the influence of the dynamic change of the gap in Fig.13.

2. In the low speed difference (0~1000r/min) section, with the torque \( T \) increases firstly and then reduces. When the \( \Delta n \) reaches a certain value \( \Delta n_1 \) (about 300r/min), \( T \) reaches a maximum \( T_1 \), then begins to decline, \( T \) deduces about 1.7N·m per \( \Delta n \) increase of 100 r/min; In the medium speed difference (1000r/ min ~2000r/min) segment, \( \Delta n \) continues to increase to a certain value \( \Delta n_2 \) (about 1000r/min), \( T \) reaches the minimum \( T_2 \), and then rise. In this phase the change of \( T \) is not obvious; In the high speed difference (>2000r/min) segment, \( T \) increase with the increase of \( \Delta n \). When \( \Delta n \) increase 100r/min, \( T \) is about to increase 0.17N·m.

3. When the clutch speed difference \( \Delta n \) is certain, the torque \( T \) increase with the increase of lubrication flow. With the increase of lubrication flow, \( T_1 \) increased from 9.01N·m to 13.03N·m, while \( T_2 \) increased from 0.09N·m to 1.29N·m, and \( \Delta n_1 \) and \( \Delta n_2 \) to 300r/min and 1000r/min respectively. In this scope (3L/min ~7L/min) the internal lubrication flow has positive correlation effect on the torque.

5.4 Discussion

Because of the actual oil film contraction point is later than the theoretical shrinkage point, so from Fig.13 and Fig.15, we can see that: at the low speed difference section, the peak point of the Model 5 and the peak point of the test value have a certain dislocation on the speed difference, which results in a large error between the simulation value and the test value. In view of this phenomenon, we define the oil film shrinkage coefficient \( \lambda \), so we can increase the theoretical total oil film cover flow as \( \lambda \cdot Q \). So the decrease speed point of the torque can be postponed, and the error between simulation and test value of the low speed difference section can be reduced.

Selecting the parameter conditions in Table 1, simulating and calculating the traditional model (Model 1) and the improved model (Model 5) with the lubrication flow is 7L/min, the simulation results as shown in Fig.16 are obtained:
Figure 16. Comparison between the test data and simulation results obtained from the models

The average relative error of the traditional model and the improved model in the three speed difference stage are shown in Table 2:

| model | Low speed | Medium speed | High speed |
|-------|-----------|--------------|------------|
| 1     | 77.81     | 68.19        | 97.42      |
| 5     | 45.20     | 3.32         | 9.35       |

Although the trend and shape of the simulated curve and the experimental curve in the low-speed differential segment are well fitted, there is a certain misalignment, resulting in a large error (45.2%) between the simulation values and the test values in this section.

For this phenomenon, it is mainly because the actual oil film radial contraction transition process is longer than the theoretical calculation, which leads to a certain degree of rotational speed dislocation between the peak point and the lowest point. So shrinkage coefficient of oil film is necessary for this model. Name this coefficient as $\lambda$.

The torque value in the low speed difference region is determined by the equivalent radius $R_s$, and the equivalent radius $R_s$ depends on the relationship between the theoretical total oil film flow $Q$ and the actual lubricant flow $Q_i$. Therefore, $Q$ times the coefficient $\lambda$, to achieve the purpose of reducing the error. The simulated result and the diagram of $\lambda$ are shown as follows:
As can be seen from the above figures, the oil film contraction coefficient tends to decrease first and then increase with the increase of rotating speed difference. The reasons can be explained by the following figures.

As shown in Fig. 18 (a), when the speed difference is zero, the friction plate drives the steel plate to rotate and then transfer torque. As the lubricating oil Qi enters the friction pairs, the separation between the friction plate and the steel plate results in the belt row torque. As the speed difference increases, as shown in Fig. 18 (b), the lubricating oil flow required to maintain the full oil film state between the friction pairs increases. The actual lubricating oil flow provided remains unchanged, resulting in the oil film contraction. As shown in Fig. 18(c), as the speed difference continues to increase, the oil film pressure p(r) between the friction pairs presents negative pressure, the friction pairs gap decreases under the action of air back pressure, and the equivalent radius of oil film increases.

Figure 17. The torque and shrinkage correction coefficient of oil film
6. Conclusion
This paper probes the influence of the gap dynamic change on the drag torque in the wet clutch, integrates the model with the non-uniformity distribution of multiple spacing, and combines the simulation results and the experimental results to establish an improved model considering the influence of the spacing dynamic change. In this paper, the bias phenomenon in the low speed difference and the negative pressure contraction between friction pairs in high speed difference are the research highlights and key points. The conclusions are as follows:

1. With the combination of bias, gap contraction and the non-uniformity distribution spacing of the friction pairs on the torque, the improved model considering the effect of the dynamic change of the friction pairs gap is obtained. Compared with the test results, the maximum calculation error of the drag torque in the low speed difference is 6%, and the average relative error in the middle and high speed
difference (>1000r/min) is 6.34%, it can predict more accurately the drag torque in the whole range of the speed difference of the wet multi-plate clutch, and provide the basis for the more accurate calculation of the clutch power loss.

2. By comparing and analyzing the traditional model and the improved model in the simulation and the experimental results, it is found that the bias phenomenon has a great influence on the torque at a low speed difference (0~1000r/min), and the oil film contraction phenomenon is the main influence factor of the drag torque at the high speed difference (>1000r/min). In the range of medium and high speed difference (>1000r/min), the radial oil film shrinkage caused by insufficient lubrication flow is the leading cause of the rapid decrease of the torque. While in the range of high speed difference (>2000r/min), the negative pressure state caused by insufficient lubrication flow is an important reason leading to the axial gap shrinkage and the rapid increase of the torque. Besides, the relative error of the improved model considering the above factors is smaller than that of the traditional model, which provides a theoretical basis for regulating the torque of the floating-supported wet clutch in whole speed difference range.

3. Under the condition of non-full oil film covering, the torque of the wet multi-disk clutch has the tendency of increase firstly and then decreases with the increase of speed difference, while the torque rises with the increase of lubricating oil flow.

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