A Novel Comprehensive Model of Wet Clutch During the Engagement Process

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Abstract. With the development of vehicles towards higher power density and reliability, the wet clutch, which is the critical component in transmission system, always encounters the problems of ablation, wear, and breakage. To solve this problem, a novel comprehensive model is developed for the wet clutch to reveal its intrinsic working mechanism. Firstly, a dynamic clearance varying model and a torque balance model are developed for the wet clutch to explain the bearing mechanism along the axial and circumferential directions. After that, the friction coefficient, which varies depending on the interface pressure, slipping speed and temperature, is obtained. By developing the temperature prediction model, a theoretical coupling model is finally established for the engagement process of wet clutch. From the simulation results, the coupling model is sufficient to describe the changing trend of several critical parameters, including the viscous torque, the asperity torque, the contact status on the friction interface, the temperature rise and the friction coefficient, etc.

1. Introduction

As the vital important component of the transmission, the working performance of wet clutch has a directly influence on the smoothness, reliability and safety of the vehicle. During the engagement process, the clutch is always exposed to an extreme environment, such as high temperature, high pressure and high-speed difference. Many physical processes, such as the transfer of friction heat, the formation of friction torque, the wear of friction materials and the distribution of cooling oil, always exist on the friction interface at the same time and interact with each other. To reveal its intrinsic working mechanism and achieve excellent vehicle performance, a lot of researches have been done.

Wu studied the oil film characteristics of rotating annular disk during the engagement process by using the finite-difference method [1-3]. Through establishing a hydrodynamic pressure model, the changing process from the hydrodynamic fluid to the boundary lubrication is described. In 1975, Ting [4] studied the boundary lubrication stage using the porous elastic theory considering the friction surface roughness and porous surface characteristics. Furthermore, to investigate the effect of grooves, EI-Sherbiny [5] used the finite element method to study the influence of groove types on the working process of wet clutch in 1977. From 1996 to 1999, Berger [6], Yang [7], Jang [8] extends the study area form the isothermal expansion to the non-isothermal process. The results show that, because of the friction plate temperature along the radial and circumferential is different, the working condition of the friction plate changes, further impact the engagement process. With the improvement of finite
element simulation methods, Jang [9] expands the clutch model into a three-dimensional model, and studies various factors such as groove, centrifugal force, and material deformation.

Owing to these considerable researches, the understanding of the working mechanism of wet clutch has been developed significantly. However, according to the actual situation, the service life of the wet clutch is always less than other components in the transmission, especially for the heavy vehicles. To further improve the performance of wet clutch, a novel comprehensive model is developed.

Firstly, a dynamic model is present to describe the engagement process along the axial and circumferential directions. Then, an optimized friction coefficient model is provided. After that, a lump thermal resistance model is established to simulate the temperature variation of wet clutch. Combining the above models, a complex comprehensive model is built for the wet clutch and the simulation results are present. Finally, the obvious conclusions are drawn.

2. Dynamic model for the engagement process
The simplified structure of wet clutch is shown in Figure 1. It consists of piston, piston cylinder, steel plate and friction plate. Under the control of the oil pressure, the wet clutch engages.

\[
F_{app} = (m_0 + m_2)\ddot{x} + (c_0 + c_2)x + F_c + F_c
\]

\[h = h_0 - x\]  

[1]

For the oil film pressure, it is one of the fundamental parameters influencing the operation of wet clutch and can be described by the following equation.

\[
F_v = \int_\Lambda \frac{B}{4A} (r^2 - R_o^2) + \frac{3\eta}{A} \frac{\partial h}{\partial t} (r^2 - R_o^2) + \ln \left( \frac{B}{4A} \left( \frac{3\eta}{A} \frac{\partial h}{\partial t} \right) \right) (R_o^2 - R_i^2) 
\]

\[A = \phi h^3 + 12\Psi d_m\]  

\[B = \frac{\phi h^3}{5} (3\omega_1^2 + 4\omega_2\omega_2 + 3\omega_2^2)\]  

\[\overline{h} = \frac{h}{2} [1 + \text{erf} \left( \frac{h}{\sqrt{2\sigma}} \right)] + \frac{\sigma}{\sqrt{2\pi}} e^{-\frac{h^2}{2\sigma^2}}\]  

[3]

When the oil film thickness between the friction pair reaches the same order of roughness of contact surface, the micro-convex bodies begins to contact. Then, the contact pressure generates.

\[F_c = \int_\Lambda \rho \]  

[4]
\[
\begin{align*}
\begin{cases}
p_c = KE \cdot 4.4086 \times 10^{-5} \cdot (4 - H)^{6.804}, & H < 4, \\
p_c = 0, & H \geq 4, \\
H = \frac{h}{\sigma}
\end{cases}
\end{align*}
\]  

(8)

\[
K = \frac{8\sqrt{2}}{15} \pi (\beta \sigma)^2 \left( \frac{\sigma}{\beta} \right)^{\frac{1}{2}}
\]

(9)

\[
E = \frac{1}{2} \left( \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)
\]

(10)

With equations (1), (3) and (7), the changing trend of the clearance between the friction pairs can be obtained, while the rotating speed mentioned in equation (5) should be further calculated.

2.2 Dynamic model along the circumferential direction

During the entire engagement process of wet friction pair, the viscous torque generated by shearing oil film always exists because of its structure and working environment. When the oil film clearance of friction pair is less than a certain value, the micro-convex peak in the rough contact area will also generate the corresponding asperity torque. Finally, the passive part is driven to rotate.

Considering the effect of surface roughness, the viscous torque can be described as follows.

\[
M_v = (1 - A_{\text{red}} C) \int_0^{2\pi} \int_R^r \eta (\phi_j + \phi_{ji}) \frac{r^2 \Delta \omega}{h} rdrd\theta
\]

(11)

\[
C = \frac{A_c}{A_n} = \kappa \pi^2 (NRH \sigma)^2 \left[ -\frac{H}{2\pi} e^\frac{-H^2}{2} - \frac{1}{2} (H^2 + 1) (\text{erf} \left( \frac{H}{\sqrt{2}} \right) - 1) \right]
\]

(12)

\[
\phi_j \text{ and } \phi_{ji} \text{ are the flow factors [10].}
\]

As for the asperity torque, it is expressed as follow.

\[
M_c = A_{\text{red}} C \mu \int_0^{2\pi} \int_R^r \rho_i r^2 \Delta \theta
\]

(13)

With the obtained \( M_v \) and \( M_c \), the rotating speed of driven part can be deduced.

\[
I_{f2} \frac{d\omega_{f2}}{dt} = M_c = M_c + M_v
\]

(14)

3. Optimized friction characteristics of wet friction pair

The wet clutch is the typical component of the application of friction phenomenon in the vehicle power transmission system. However, the value of friction coefficient \( \mu \), mentioned in equation (15), is influenced by many factors. In the previous researches, it is usually described by the slipping speed between the friction pairs, without considering the influence of the contact pressure and the temperature. In this paper, an optimized friction coefficient is presented to fill that gap in knowledge.

\[
\mu = 23e^{\left( \frac{-3v}{(\ln T - 3.3)(28.3p)^{0.87} - 0.87} \right) - 5.16} + 0.08 (e^{-0.005T} - 1) (e^{-0.2v} - 1)
\]

(15)

\[
+ 0.008 \ln (4v + 1) e^{0.005T} - 0.005 \ln (28.3p) + 0.020
\]

4. Lumped thermal resistance model

The Finite Element Method (FEM) is always used to analyse the thermal problems in the engineering field. However, it is difficult to predict the temperature of the component in real time. In this paper, the lumped parameter method is adopted to build a lumped thermal resistance model. As shown in Figure 2, the lubricating cooling system is divided into different parts, including the environment (E),
the radiator (R), the hydraulic fluid reservoir (F), the relief safety valve (RV), the pump (P), the pressure regulating valve (V) and the clutch (C).

Figure 2. Heat transfer model.

For the component and the lubrication oil in each node, the temperature variations can be deduced by the following equation.

\[
\begin{align*}
C_i \frac{dT_i}{dt} &= \Phi_i - K_{i,j} (T_j - T_i) - \sum_j K_{i,j} (T_i - T_E) \\
C_{i0} \frac{dT_{i0}}{dt} &= \Phi_{i0} - K_{i0,j} (T_{i0} - T_j) - \sum_j K_{i0,j} (T_{i0} - T_{j0})
\end{align*}
\]

(16)

in which \(C_{i0} = \rho_{i0} \cdot c_{i0} \cdot V_{i0}\) and \(K\) is the equivalent heat transfer coefficient [11].

Taking the temperature variation of wet clutch as an example, the heat condition can be calculated according to the following equations.

\[
\begin{align*}
C_c \frac{dT_c}{dt} &= \Phi_c - K_{c,E} (T_c - T_E) - K_{c,V} (T_c - T_V) - K_{c,aC} (T_c - T_{aC}) \\
C_{oC} \frac{dT_{oC}}{dt} &= \Phi_{oC} - K_{oC,aV} (T_{oC} - T_{oV}) - K_{oC,aR} (T_{oC} - T_{oR}) \\
&- K_{oC, E} (T_{oC} - T_E) - K_{oC, aC} (T_{oC} - T_{C})
\end{align*}
\]

(17)

\[
\Phi_c = M_c \cdot \Delta \omega = (M_v + M_j) \cdot \Delta \omega
\]

(18)

5. Comprehensive model and simulation results analysis

Figure 3. Coupling relationship between different models.
The above three sections build the dynamic model for the engagement process, provide an improved friction coefficient and establish a lumped thermal resistance model to predict the temperature variation of wet clutch. Different model describes different physical process and, combining these three models, a complex comprehensive model is finally obtained. Figure 3 shows the coupling relationship between different models.

The obtained simulation results are illustrated in Figure 4. Figure 4(a) presents the applied oil pressure on the piston, which is measured from the test bench. After the violent fluctuation, the value of the pressure is finally levelling out at around 1.3MPa. In Figure 4(b), when the wet clutch contacts, the generated friction torque has a sharp increase, then shows a slowly falling trend. While, at the end of the engagement process, the friction torque goes up to 109N\cdot m and then drops to 0N\cdot m. For the viscous torque, it reaches 17N\cdot m at the contact moment, then decreases to 0N\cdot m; while, for the asperity torque, it is the main composition of the friction torque, especially at the end stage of the engagement process.

Figure 4(c) shows the changing trend of the friction coefficient. It varies from 0.05 (dynamic friction coefficient) to 0.13 (static friction coefficient). As for the temperature of wet clutch, Figure 4(d) indicates that it has an increase from 65.8\degree C to 87.2\degree C during the entire process.

[Figure 4. Simulation results.]

![Figure 4. Simulation results.](image-url)
6. Conclusions
In this paper, a novel comprehensive model is established for the engagement process of wet clutch by integrating different models, including the dynamic model for the axical and circumferential movement, the optimized fiction coefficient and the thermal resistance model for predicting the temperature variation.

Under the applied constant pressure, the friction torque goes up quickly, then shows a decreasing trend. At the end of the slipping status, the torque increases to 109N·m sharply and drops to 0N·m. The asperity torque shows a similar changing trend, while the viscous torque decreases during the entire engagement process from 17N·m to 0N·m. Under this simulation condition, the friction coefficient varies from 0.05 to 0.13 and the temperature rise of wet clutch is 21.4℃.

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Appendix

| Parameters | Meanings |
|------------|----------|
| \( A_{red} \) | percentage of non grooved area |
| \( E \) | elastic modulus |
| \( F \) | force |
| \( I \) | moment of inertia of components |
| \( M \) | torque |
| \( N \) | asperity density |
| \( R \) | radius of friction disc |
| \( R_H \) | radius of asperity |
| \( T \) | temperature of components |
| \( V \) | volume |
| \( x \) | displacement |
| \( h \) | clearance |
| \( \phi \) | flow factors |
| \( d_m \) | thickness of surface material |
| \( m \) | weight |
| \( p \) | pressure |
| \( r \) | radius |
| \( v \) | slipping speed |
| \( \phi_{(i)} \) | heat source |
| \( \Psi \) | permeability of materials |
| \( \beta \) | curvature radius of asperity |
| \( \mu \) | friction coefficient |
| \( \omega \) | angular speed of components |
| \( \eta \) | coefficient of kinetic viscosity |
| \( c \) | specific heat capacity |
| \( \rho \) | density |
| \( \sigma \) | the RMS surface roughness |
| \( \nu \) | Poisson's ratio |
| \( \kappa \) | plastic deformation coefficient |