doi:10.11835/j.issn.1671-8224.2015.01.01

To cite this article: WANG Yan-zhong, LI Yuan, NING Ke-yan, HAN Ming. Modeling and analysis of wet clutch engagement characteristics [J]. J Chongqing Univ: Eng Ed [ISSN 1671-8224], 2015, 14(1): 1-8.

Modeling and analysis of wet clutch engagement characteristics

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Received 19 September 2014; received in revised form 8 October 2014

Abstract: A mathematical model was developed to analyze the characteristics of the wet clutch during engagement. The lubricant squeeze action was simulated with Patir and Cheng average flow model in which the permeability of friction material is taken into account, and the asperity load is calculated according to the Greenwood and Tripp approach. In this model, effects of friction material permeability, applied load and driving velocity on the engagement characteristics of the wet clutch were studied. The results show that friction material with high permeability reduces the film thickness rapidly and increases the torque peak; the applied load increases the asperity contact pressure and the friction torque, and reduces the engagement time; the driving velocity mainly increases the engagement time. The theoretical torque and relative velocity curves agree qualitatively with the experimental ones, which verifies the wet clutch engagement model.

Keywords: wet clutch engagement; permeability; friction material.

1 Introduction

The wet disk clutch is mainly used in automatic transmissions to provide smooth torque transmission during shifting. A wet clutch is comprised by the friction plate in which friction material is ponded on both sides of the steel core plate and the steel separate plate. Different from the dry clutch, a wet clutch is immersed in an automatic transmission fluid which lubricates and cools the contact surface and provides smoother torque transmitted. By means of the applied loading, the lubricant is extruded from the gap between the plates and through the porous friction material, and then the two plates are pressed together.

Much research has been carried out in this field. Wu [1-2] developed a axisymmetric model of porous annular disks. He modeled the hydrodynamic portion of engagement using the Reynolds equation. Ting [3-4] modeled wet clutch engagement, considering effects of surface roughness and friction material porosity. Natsumeda and Miyoshi [5] developed a finite difference model based on the average flow model.

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* Funded by the Innovation Foundation of Beijing University of Aeronautics and Astronautics for PhD Candidates.
proposed by Patir and Cheng \(^6\) to study the effect of surface roughness, permeability, and temperature. The theoretical results qualitatively agree with the experimental ones. Berger et al. \(^7\) used a finite element method to solve engagement model of wet clutch and investigate the effects of applied load, permeability, and groove geometry on wet clutch engagement. Jang and Khonsari \(^9\) and Jang et al. \(^10\) developed a thermohydrodynamic model of wet clutch with consideration of radial grooves and waffle-shape grooves. They concluded that thermal effects in the wet clutch influenced the engagement time and the torque response. In this work, a simplified axisymmetric model was developed to investigate the effects of friction material permeability, applied load, and driving velocity on the engagement characteristics of wet clutches. The Runge-Kutta method was used to solve the model, and the theoretical result of transmitted torque is similar to that obtained from experiment.

2 Mathematical model of wet clutch engagement

The friction disk and the steel separated disk are separated by a lubricant film \((h)\) and pressed together by an applied load \((F_{ap})\), the driving disk rotates at a constant angular velocity \((w_1)\), and the driven disk begins to rotate at an instant angular velocity \((w_2)\).

\[
\frac{1}{r} \frac{d}{dr} \left[ r \phi h^3 (1 + 3 \eta) + 12 \Phi \rho_d \frac{dp_d}{dr} \right] = 12 \mu \frac{d \bar{h}}{dr},
\]

(1)

where \(p_d\) is the hydrodynamic pressure developed in the fluid; \(\Phi\) is the friction material permeability; and \(\eta\) is the Beavers and Joseph slip factor. In this analysis, the surfaces are assumed to have an isotropic roughness pattern; consequently, \(\phi\) is the flow factor given by Patir and Cheng \(^12\), \(d\) is the thickness of the permeable friction material; \(r\) is coordinate in radial direction; \(\mu\) is lubricant viscosity; \(\bar{h}\) is the average gap, and \(t\) is the time.

2.1 Corrected Reynolds equation

The isothermal Reynolds equation for the lubricant portion of engagement of wet clutch is modified by using the average flow model proposed by Patir and Cheng \(^6\), in which the centrifugal force acting on lubricant and permeability of friction material are taken into account. Assuming that the thin fluid film between two parallel annular disks is in a laminar flow condition, and the Beavers and Joseph \(^11\) velocity slip condition is applied to the permeable boundary for the radial flow, the corrected Reynolds’ equation for axisymmetric is:

2.2 Asperity load model of rough surfaces

There are many irregular asperities on the rough surfaces of the friction disk and the steel separated disk. During a wet clutch engagement, as the lubricant film thickness is reduced, asperities on the rough surface will contact mechanically and share the applied load with the lubricant film. In Greenwood-Tripp \(^13\) rough surface asperity contact model, the average pressure is:

\[
p_c = \frac{F_c}{S} = \frac{8 \sqrt{2}}{15} \pi (N \sigma \beta) \frac{1}{\beta} E \left( \frac{\sigma}{\beta} \right)^{1/2} F_{S/2} \left( \frac{h}{\sigma} \right),
\]

(2)

where \(N\) is the density of the rough surface asperity; \(\beta\)
is the radius of the asperity summits; \( \sigma \) is the standard deviation of the asperity height; \( F_z \) is the asperity load; \( E' \) is the equivalent Young's modulus; and \( S \) is the rough surface area. Assuming that the rough surface asperity height obeys Gaussian distribution, the function \( F_z(z) \) can be described as follows.

\[
F_z(z) = \int_{z}^{\infty} (s-z)^n \Phi^*(s) ds = \int_{z}^{\infty} (s-z)^n \frac{1}{\sqrt{2\pi}} e^{-\frac{1}{2}s^2} ds. \tag{3}
\]

2.3 Force and torque balance equation

During the wet clutch engagement, the lubricant film and mechanical contact with a rough surface asperity share the applied load. The wet clutch engagement is a quasi-static process, the force balance equation is:

\[
F_{app} = F_h + F_c = \int \int p_h dS + \int \int p_c dS. \tag{4}
\]

The friction plate is the driving disk rotating at a constant angular velocity and the separated plate is the driven disk that begins to rotate from rest as a result of the transferred torque. The torque \( T \) is composed of fluid viscous shear torque \( T_h \) and rough surface friction torque \( T_c \):

\[
T = T_h + T_c = I \frac{d\omega}{dt}, \tag{5}
\]

\[
T_c = \mu_s \int \int r p_c dS, \tag{6}
\]

where \( I \) is the disk mass moment of inertia; and \( \mu_s \) is the coefficient of sliding friction.

Assuming the lubricant is a Newtonian fluid, according to the Patir-Cheng fluid viscous shear torque model, the lubricant viscous shear torque is:

\[
T_c = (\phi_t - \phi_h) \int \int r^2 \frac{w_1 - w_2}{h} dS. \tag{7}
\]

where \( \phi_t \) and \( \phi_h \) are factors given by Patir and Cheng \cite{11}.

2.4 Friction coefficient model

When the friction material, surface roughness and the loading pressure are determined, the relative sliding velocity between two disks is the most significant factor for the coefficient of sliding friction \cite{14}. The experimental coefficient of friction at different sliding velocity is used to fit a curve and the equation of the curve for the sliding friction coefficient as a function of relative velocity \( (v_{rel}) \) is:

\[
\mu = 0.13 - 0.008 \log_{10} (v_{rel}). \tag{8}
\]

2.5 Calculation of the model

Integrating Eq. (1) twice yields the equation for the lubricant film pressure:

\[
p_h = \frac{\beta_e}{4K} \left( r^2 + \frac{b^2-a^2}{\ln(a/b)} \ln(r/b) - b^2 \right) g(h) \frac{dh}{dt}. \tag{9}
\]

where \( \beta_e=12\mu, K=\phi^3(1+3\eta)+12\phi d, g(h)=(1+\text{erf}(h/(2^{1/2}d)))/2, \) and herein \( \text{erf}(h) \) is the Gauss error function.

Taking Eqs. (9) and (2) into (4) derives the lubricant film thickness ratio:

\[
\frac{dh}{dt} = \frac{-8K(1-p_o)}{\beta_s g(h)(a^2+(b^2-a^2)/\ln(a/b)+b^2)}. \tag{10}
\]

Taking Eqs. (6) and (7) into (5) derives the instant angular velocity ratio of the driven disk:

\[
\frac{dw_z}{dt} = 2\pi \left[ \frac{w_1}{4} (b^2-a^2) (\phi_t - \phi_h) \frac{1-w_z}{h} + \frac{\mu_t}{3} (b^2-a^2) p_c \right]. \tag{11}
\]

Using Runge-Kutta method, Eqs. (10) and (11) are simultaneously solved to obtain the film thickness and instant angular velocity of driven disk; the asperity
contact pressure $F_c$ and the lubricant film pressure $F_h$ are calculated according to Eqs. (2) and (9); and the lubricant viscous shear torque $T_b$ and friction torque $T_f$ can be obtained from Eqs. (6) and (7). The time step is 0.001 s. The iteration stops when the relative angular velocity reaches zero, and the driving and driven disks lock up and the engagement is completed.

3 Results and discussion

The dimensions of the two disks used in this analysis are given by the inner and outer radii 32.5 mm and 42.5 mm. The separated disk and the core of the friction disk are made of S45C, and the friction material is copper-based powder metallurgy material. Surface parameters used in the analysis are obtained by measuring the surface profiles of several clutch disks with an optical surface profilometer. Table 1 lists the dimensions of the friction disk, parameters of the surfaces and the test conditions used in this analysis.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner radius $a$</td>
<td>0.0325 m</td>
</tr>
<tr>
<td>Outer radius $b$</td>
<td>0.0425 m</td>
</tr>
<tr>
<td>Friction material thickness $d$</td>
<td>5.652 x 10^{-4} m</td>
</tr>
<tr>
<td>Permeability parameter</td>
<td>1.0 x 10^{-13} m$^2$</td>
</tr>
<tr>
<td>AFT density</td>
<td>820 kg m$^{-3}$</td>
</tr>
<tr>
<td>AFT viscosity</td>
<td>8.21 x 10^{-2} Pa s</td>
</tr>
<tr>
<td>Applied force</td>
<td>2 792.1 N</td>
</tr>
<tr>
<td>Driving velocity</td>
<td>66.67$\pi$ rad</td>
</tr>
<tr>
<td>Disk inertia</td>
<td>0.1305 kg m$^2$</td>
</tr>
<tr>
<td>Young's modulus for friction material</td>
<td>27.0 x 10^6 Pa</td>
</tr>
<tr>
<td>Initial film thickness</td>
<td>2.54 x 10^{-5} m</td>
</tr>
<tr>
<td>Asperity tip radius</td>
<td>5.0 x 10^{-4} m</td>
</tr>
<tr>
<td>Asperity density</td>
<td>3.0 x 10^{-7} m</td>
</tr>
<tr>
<td>RMS roughness</td>
<td>6.0 x 10^{-6} m</td>
</tr>
<tr>
<td>Dimensionless slip coefficient $p_{[1]}$</td>
<td>0.2</td>
</tr>
</tbody>
</table>

Fig. 2 indicates that at the beginning of engagement, the lubricant film which separates the two annual clutch disks is relatively thick and the hydrodynamic pressure develops due to the squeeze of the lubricant film to support the applied load, the lubricant viscous shearing torque reaches a maximum value, and the driven disk begins to rotate from rest. As the film thickness reduces to a value smaller than theasperity height, the contact pressure of rough surface at the boundary lubrication begins to support a major portion of the applied load. When the film thickness reaches its minimum, the asperity of the rough surfaces carries the entire applied load (the fluid supports no load). When the relative velocity reduces to zero, the engagement process is completed. The engaging time is 3.904 s and the peak of the transmitted torque is 8.86 N when film thickness reaches its minimum at 0.45 s.

Figs. 3 and 4 depict the comparison of relative velocity and the transmitted torque obtained from experiment and simulation. Notice that the curves of the relative velocity and torque fitted by experimental results are similar to those obtained from simulation with the experimental engagement time (3.923 s) 4.84% higher than the simulation one and the peak torque (9.01 N m) 1.7% higher than the simulation one. Due to the assumption of the isothermal model and the vibration of the applied load, there is a deviation less than 10% between the torques obtained from experiment and those from simulation in the middle of the engagement process. Generally, the model established in this analysis is able to predict the engagement characteristics of the wet clutch.

The friction material of the friction disk is porous, and the lubricant will be squeezed out from the gap of two disks and go through the permeable friction material under the applied load and make film thickness decrease. Fig. 5 depicts the effect of friction material permeability on the film thickness, pressure, relative velocity and transmitted torque during the wet clutch engagement. It is shown that for a more
permeable friction material, the film thickness reaches its minimum faster, and the asperities of rough surface contact and support the applied load earlier. The torque peak appearing when the film thickness reduces to its minimum is high, which means a greatly increasing rate of torque. Therefore, it is necessary to consider the effect of the friction material when the torque needs to be transmitted smoothly for the wet clutch. The engagement time and relative velocity are insensitive to the friction material permeability.

The lubricant film and asperities contact of the rough surface share the applied load. Fig. 6 demonstrates the effect of the applied load on the engagement characteristic of wet clutch. The results show that the film thickness decreases rapidly and reaches its minimum sooner as the applied load increases. To balance a higher applied load requires a higher asperity contact pressure; as a result, the minimum film thickness is smaller. The friction torque increases as the applied load increases and the engagement time decreases.

![Fig. 2 Simulation engagement characteristics of wet clutch: a) load, film thickness and speed; and b) torque](image)

![Fig. 3 Experimental and theoretical relative velocity curves](image)

![Fig. 4 Experimental and theoretical torque curves](image)
Driven by the friction disk, the separated disk begins to rotate from rest when the angular velocity of the driven disk is equal to that of the driving disk, and the engagement is completed. Fig. 7 illustrates the effects of the driving angular velocity on the engagement time, relative velocity and torque during the wet clutch engagement. The results indicate that for different driving velocity, the trend and minimum of film thickness are the same, and so are the asperity contact pressure and friction torque of rough surface. As the driving velocity increases, the relative angular velocity increases and the lubricant viscous shear torque which is related to the relative angular velocity in Eq. (7) increases slightly. Because of the increase in energy needed for the driven disk, the engagement time increases nearly linear with the increase of driving velocity.

Fig. 5 Permeability effects on the engagement characteristics of wet clutch: a) dimensionless film thickness; b) dimensionless relative speed; c) load; and d) torque
Wet clutch engagement characteristics

Fig. 6 Applied load effects on the engagement characteristics of wet clutch: a) dimensionless film thickness and relative speed; and b) torque

Fig. 7 Driving angular velocity on the engagement characteristics of wet clutch: a) dimensionless film thickness and relative speed; and b) torque

4 Conclusion

A wet clutch engagement model has been developed to study the effects of the porous friction material permeability, applied load and driving velocity on the engagement characteristics of the wet clutch. The results show that the friction permeability has a significant effect on the torque peak at the beginning of engagement, friction material with high permeability reduces the film thickness rapidly and increases the torque peak. The applied load increases the asperity contact pressure and the friction torque, and reduces
the engagement time. The driving velocity mainly increases the engagement time. The theoretical torque curve and relative velocity curve agree qualitatively with the experimental ones, and verifies the wet clutch engagement model.

Acknowledgement

This research is a part of a basic product project and was supported by the Innovation Foundation of Beijing University of Aeronautics and Astronautics for PhD Graduates.

References