Research of Torque Response Characteristics During the Engagement of Wet Clutches

Li Guoqiang, Ma Biao

School of Mechanical Engineering
Beijing Institute of Technology
Beijing, China
kevin3062@gmail.com

Li Heyan*

School of Mechanical Engineering
Beijing Institute of Technology
Beijing, China
*Corresponding author:lovheyan@gmail.com

Abstract—**The engagement of wet clutch was modeled by developing a modified Reynolds equation, based on the average flow model of Patir and Cheng. Influences of surface roughness, material permeability and centrifugal force were included in the model. The Greenwood-Tripp method was used to develop the asperity contact model. The force balance equation and torque balance equation were solved together to obtain torque response characteristics during wet clutch engagement. This paper shows that viscosity has effect on viscous torque; material permeability affects the engagement time and torque response; and the applied pressure greatly influences engagement time than other parameters.**

Keywords-wet clutch, engagement, torque response

I   INTRODUCTION

The wet clutch is a key component in automotive automatic transmission, which can transfer torque from the engine to the driven wheel during shifting. With proper control strategic, the engagement may be smooth and reduce the shift vibration. Wet clutches typically contain a series of friction plates and steel separator plates, under an alternating sequence. Paper based or sinter bronze based friction material is bonded on both sides of friction plates. The engagement or disengagement of wet clutches is determined by means of hydraulic loading or unloading in the piston.

Many researches have been done to study the engagement of wet clutches, which are mainly about the effects of friction plate characteristics and working condition. Natsumeda et al.[1] developed an engagement model of paper-based friction material, taking the permeability of friction material, compressive strain, asperity roughness and centrifugal force into account. Berger et al. [2] used a modified Reynolds equation to study the effects of several input parameters, such as applied force, grooved area, and friction material permeability. Davis et al. [3] enhanced the isothermal model developed by Berger, including the fluid thermal effects, to obtain torque and temperature characteristics during wet clutch engagement. Gao el at. [4] investigated the engagement of a wet clutch with skewed surface roughness, and compared the effects of Weibull asperity height distribution and Gaussian one. Zhang zhigang el at. [5] studied the effects of parameters, such as permeability, fluid viscosity and dynamic-static friction coefficient, on the torque response and engagement time.

The objective of this paper is to analyze the engagement characteristics theoretically and develop a mathematical model to predict the velocity of steel plate and torque response during engagement. In addition, several influencing parameters are taken into account during engagement of wet clutches.

II   METHOD

The engagement of a single contact surface wet clutch can be simplified as shown in Fig. 1. A friction plate and a steel plate are separated by nominal film thickness h. The friction plate, splined with driving axle, rotating at a constant angular velocity ω_1, is pressed toward the steel plate rotating at velocity ω_2, which is connected with driven axle. During engagement, film thickness is reducing with the applied force F_{app} increases, and asperity of friction material begins to contact, supporting most of the applied force when the film thickness h reaches minimum.
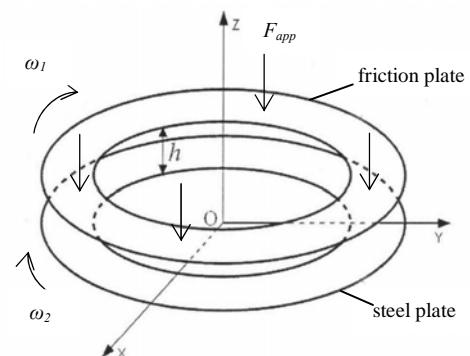

Figure 1.   Model of a single contact wet clutch

A.   Oil Film Hydrodynamic Model

The engagement of wet clutch contains hydrodynamic effect, which can be described by a general Reynolds function. According to flow characteristics of fluid, a laminar flow and constant newton fluid is assumed. The fluid on the plate surface is supposed to move with the same velocity as the plate itself. Velocity in z-direction and grooves are neglected. Also, the viscosity is assumed to be constant due to a fast engagement. Under above assumptions, the isothermal, incompressible

Reynolds equation with major centrifugal force in cylindrical coordinate system is:

$$\begin{cases} \dfrac{\partial p}{\partial r} = \dfrac{\partial}{\partial z}(\mu \dfrac{\partial v_r}{\partial z}) + \rho r \omega^2 \\ \dfrac{\partial p}{r\partial \theta} = \dfrac{\partial}{\partial z}\rho\left(\mu \dfrac{\partial v_\theta}{\partial z}\right) \\ \dfrac{\partial p}{\partial z} = 0 \end{cases} \qquad (1)$$

with boundary condition:

$z=0$, $v_r=0$, $v_\theta=r\omega_2$, and $z=h$, $v_r=0$, $v_\theta=r\omega_1$,

where p is oil film pressure in the gap between plates; μ is viscosity of oil; v_r and v_θ represent velocity of oil in r and θ direction separately; ω is angular velocity of oil, ρ is oil density.

Allowing for the effects of surface roughness on partial lubrication, the mean unit flow in r and θ direction is [6]:

$$\begin{cases} \overline{q_r} = \int_0^h v_r dz = \phi_r[-\dfrac{h^3}{12\mu}\dfrac{\partial p}{\partial r} + \dfrac{\rho r h^3}{120\mu}(3\omega_1^2 + 4\omega_1\omega_2 + 3\omega_2^2)] \\ \overline{q_\theta} = \int_0^h v_\theta dz = \phi_\theta(-\dfrac{h^3}{12\mu}\dfrac{\partial p}{\partial \theta}) + r\dfrac{\overline{h_t}}{2}(\omega_1 + \omega_2) + \dfrac{r(\omega_1 - \omega_2)}{2}\sigma\phi_s \end{cases} \qquad (2)$$

where ϕ_r, ϕ_θ and ϕ_s are flow factors of Patir and Cheng, and $\phi_r = 1 - 0.9\exp(-0.56h/\sigma)$; $\overline{h_t}$ is average oil film thickness defined by Patir and Cheng; σ is RMS roughness of surface.

Because the friction material is porous, the extruding velocity of lubricant squeezing into friction material surface is shown as:

$$u_z = -\dfrac{1}{r}\dfrac{\partial}{\partial r}(\dfrac{r\Phi d}{\mu}\dfrac{\partial p}{\partial r}) + \dfrac{1}{r^2}\dfrac{\partial}{\partial \theta}(\dfrac{d\Phi}{\mu}\dfrac{\partial p}{\partial \theta}) \qquad (3)$$

where Φ is permeability of friction material; d is thickness of permeable friction material.

A mean flow balance on the control volume of oil film is:

$$\dfrac{1}{r}\dfrac{\partial(r\overline{q_r})}{\partial r} + \dfrac{1}{r}\dfrac{\partial \overline{q_\theta}}{\partial \theta} + \dfrac{\partial \overline{h_t}}{\partial t} - u_z = 0 \qquad (4)$$

Assuming axisymmetry, and the plates remaining parallel, (2)~(4)can be simplified as:

$$\dfrac{\partial}{\partial r}(\phi_r r(h^3 + 12\Phi d)\dfrac{\partial p}{\partial r}) = \dfrac{\phi_r \rho r h^3}{5}(3\omega_1^2 + 4\omega_1\omega_2 + 3\omega_2^2) + 12\mu r \dfrac{\partial \overline{h_t}}{\partial t} \qquad (5)$$

with boundary condition

$$p(r = R_i) = 0, \ p(r = R_0) = 0,$$

where R_i and R_o are inner radius and outer radius of friction plates separately.

The oil film pressure along r-direction is:

$$p = \dfrac{B}{4A}(r^2 - R_0^2) + \dfrac{3\mu}{A}\dfrac{\partial \overline{h_t}}{\partial t}(r^2 - R_0^2) + \ln\dfrac{r}{R_0}(\dfrac{B}{4A} + \dfrac{3\mu}{A}\dfrac{\partial \overline{h_t}}{\partial t})\dfrac{(R_0^2 - R_i^2)}{\ln R_i - \ln R_0} \qquad (6)$$

where $A = \phi_r(h^3 + 12\Phi d)$, $B = \rho\phi_r h^3(3\omega_1^2 + 4\omega_1\omega_2 + 3\omega_2^2)/5$.

Under assumption of Gaussian surface profiles, the relationship between average oil film thickness $\overline{h_t}$ and nominal oil film thickness h is[2]:

$$\dfrac{d\overline{h_t}}{dt} = \dfrac{d\overline{h_t}}{dh}\dfrac{dh}{dt} = \{\dfrac{1}{2}[1 + erf(\dfrac{h}{\sqrt{2}\sigma})]\}\dfrac{dh}{dt} = g(h)\dfrac{dh}{dt} \qquad (7)$$

B. Asperity Contact Model

During the engagement of wet clutches, asperity on the plate surface begins to contact and support part of the applied load, as the film thickness decreases. In rough contact mechanics, Greenwood and Tripp (GT) studied two surfaces with random asperity, and concluded that the true contact area is:

$$A_c = A_n \pi N \beta \sigma^* F_2(H) \qquad (8)$$

where A_n is nominal area; N is asperity density; β is average asperity radius; σ^* is combined RMS; $H = h/\sigma$.

Later, Patir calculated the approximate solution of GT model with Whitehouse-Archard method[5]:

$$\begin{cases} p_a = K'E' \cdot 4.4086 \times 10^{-5} \cdot (4 - H)^{6.804} & H < 4 \\ p_a = 0 & H \geq 4 \end{cases} \qquad (9)$$

where E' is equivalent modulus of elasticity, p_a is asperity contact pressure.

$$\dfrac{1}{E'} = \dfrac{1}{2}(\dfrac{1 - \upsilon_1^2}{E_1} + \dfrac{1 - \upsilon_2^2}{E_2}) \qquad (10)$$

$$K' = \dfrac{8\sqrt{2}}{15}\pi(N\beta\sigma^*)^2\sqrt{\dfrac{\sigma^*}{\beta}} \qquad (11)$$

$$F_n(\xi) = \int_\xi^\infty (x - \xi)^n \varphi(x)dx \qquad (12)$$

where $\varphi(x)$ is Gaussian probability density function.

C. The Force and Torque Balance Equation

In wet clutch engagement, applied load is balanced by the reaction forces both of the fluid and the asperity. Taking the integration area into consideration, the force balance equation is:

$$F_{app} = (1 - A_{red}C)\int_0^{2\pi}\int_{R_i}^{R_0} pdA + A_{red}C\int_0^{2\pi}\int_{R_i}^{R_0} p_a dA \qquad (13)$$

where C is ratio of contact area and nominal area, $C = A_c/A_n$; A_{red} is groove area coefficient; F_{app} is force applied on piston of clutch, $F_{app} = \pi(b^2 - a^2)p_{app}$; b and a is outer and inner radius separately; p_{app} is applied pressure on the piston.

During engagement, torque transferred between two friction faces contains viscous effect and roughness effect. The sum of viscous torque T_v and asperity contact torque T_c makes up the total torque. The torque balance equation of steel plate is:

$$I\dfrac{d\omega_2}{dt} = T_v + T_c \qquad (14)$$

According to the average flow model by Patir-Cheng, the viscous torque and asperity torque are:

$$T_v = (1 - A_{red}C)\mu(\phi_f + \phi_{fs})\int_0^{2\pi}\int_{R_i}^{R_0}\dfrac{r^2\omega_{rel}}{h}rdrd\theta \qquad (15)$$

$$T_c = A_{red}Cf \int_0^{2\pi} \int_{R_i}^{R_0} p_a r^2 dr d\theta \qquad (16)$$

where ϕ_f and ϕ_{fs} are flow factors of Patir-Cheng[7]; ω_{rel} is relative sliding angular velocity, $\omega_{rel} = \omega_1 - \omega_2$; I is inertia of steel plate. f is coefficient of sliding friction. Actually, the friction coefficient is a function of relative sliding velocity, applied pressure and temperature. However, there is not a proper method to calculate the friction coefficient by far. For simplicity, it is supposed that friction coefficient is only a function of relative sliding velocity. The curve-fitting from experiments is[5]:

$$f=0.13-0.008\log(\omega_{rel}) \qquad (17)$$

III SOLUTION PROCEDURE

Since film thickness h and velocity of steel plate ω_2 are not independent, (5) and (14) should be solved simultaneously by direct integration using the Runge-Kutta method, to obtain transient oil film thickness and velocity. The time step is 0.001 second, and the procedure is not stopped until the relative angular velocity reaches zero. Then using the solved thickness and velocity, viscous torque and asperity torque can be acquired from (15) and (16). The initial value of ω_2 is zero, and the environment temperature is 80℃。 Parameters used in the simulation are listed in TABLE I[5]. All plates used in the simulation are supposed to have Gaussian surface profiles.

TABLE I.   INPUT PARAMETERS

Inner radius of plate(m) R_i	0.086
Outer radius of plate(m) R_o	0.114
Asperity tip radius(m) β	8×10^{-4}
Asperity density($1/m^2$) N	7×10^7
Applied pressure(Pa) p_{app}	1×10^6
Viscosity (Pa.s) μ	0.023
Inertia of plate(kg.m^2) I	1
RMS of friction material(m)σ	8×10^{-6}
Combined RMS roughness (m) σ^*	9×10^{-6}
Lining thickness(m) d	0.001
Young modules (Pa) E'	4.84×10^9
Density of oil(m) (kg/m^3)ρ	875
Initial film thickness(m) h_0	1×10^{-4}
Initial rotating speed(rpm) ω_1	1200
Permeability of material(m) Φ	4×10^{-12}
Inner radius of piston(m) a	0.045
Outer radius of piston(m) b	0.107
groove area coefficient A_{red}	0.78

IV RESULTS AND DISCUSSION

At the beginning of engagement, oil film thickness decreases rapidly, with the applied load pressing friction plate toward steel plate. The hydrodynamic force supports nearly the whole applied load. In this period, the relative angular velocity changes little, because of the little viscous torque. Then the asperity between the friction plate and the steel plate starts to contact and support part of the applied load. When the asperity contact force supports the entire applied load, the film thickness reaches a constant value. Fig. 2 shows the variation of oil film thickness and angular velocity of steel plate during engagement. It is shown that the descending time of oil film thickness is quite quickly, nearly 0.02s. This is because the

applied load is much bigger than the hydrodynamic force at the beginning period. When the oil film thickness decreases approximately to the combined RMS roughness of plates, it no longer changes, and the hydrodynamic force can be neglected.
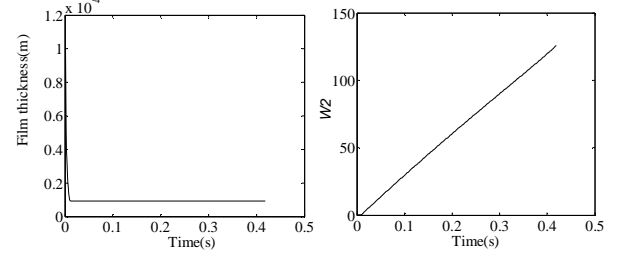

Figure 2.   Simulation of oil film thickness h and angular velocity ω_2

Due to the surface roughness of friction plate and steel plate, oil exists in the gap all over the engagement period, which causes viscous torque, driving the steel plate to rotate, together with asperity torque. The viscous torque reaches its highest value just at the time when the film thickness no longer decreases at the beginning of engagement. With the declining relative angular velocity, the viscous torque then decreases to zero, when the difference of relative angular velocity becomes zero, and then the two plates lock up. Because the friction coefficient is supposed to be the function of relative angular velocity, asperity torque increases gradually, just as shown in Fig. 3. At the end of engagement, the asperity torque increases largely, becoming the dominant contributor to the total torque.
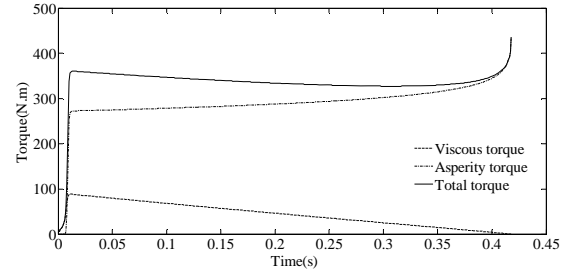

Figure 3.   Simulation of torque response

A. Viscosity Effect

Viscosity of the oil plays an important part in the engagement of wet clutches, which largely influences viscous torque. During engagement, part of the friction heat generated between plates is taken away by the cooling oil, and the other part of heat will be transferred to the plates. The oil absorbs the friction heat and the temperature of oil increases. The variation of temperature throughout the engagement will affect viscosity largely, influencing the torque response characteristics of wet clutches. Fig. 4 shows the change of torque when the oil temperature is 60℃. It is clearly that the viscosity has an effect on viscous torque and engagement time. Compared to the results in Fig. 3, the engagement time of lower viscosity is bigger than that of higher viscosity. This can be explained that viscosity determines viscous torque, which is the major torque driving steel plate, especially at the beginning of engagement, and the torque affects the ratio of angular velocity of steel plate, just as shown in (14). The smaller ratio of angular velocity

slows down the locking up of plates and changes the variation of total torque. Furthermore, owning to the change of viscous torque, the fluctuation of total with large velocity is bigger than that with low velocity, which has effect on the impact of gear change and vibration of transmissions.
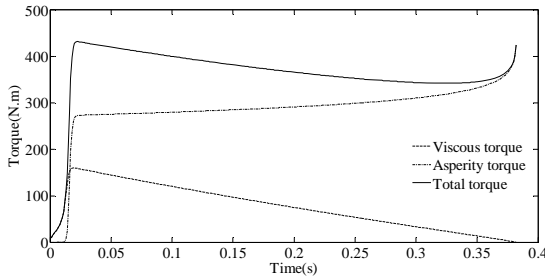

Figure 4. Effect of viscosity at 0.0419 Pa.s

B. Permeability of friction material effect

Due to the porous structure and permeability of friction material, oil can be squeezed into contact facing by the hydrodynamic pressure during engagement. Permeability is also influenced by friction and wear conditions. Friction behavior is affected by the permeability in the initial engagement of wet clutches. Bigger permeability can lead to faster exuding velocity of oil film and more oil can be squeezed into the porous structure, leading to a longer engagement time. Fig. 6 shows the engagement behavior with low permeability at 4×10^{-13} m. It is shown that the ascending curve of viscous torque is much smoother, compared with the simulation result in Fig. 1. The oil is hard to squeeze into friction material with low permeability, stored in the gap between friction plate and steel plate, which can support part of the applied load at the initial of engagement, causing a longer engagement time. Also, thanks to the stored oil in the gap, the duration time of viscous torque is long.
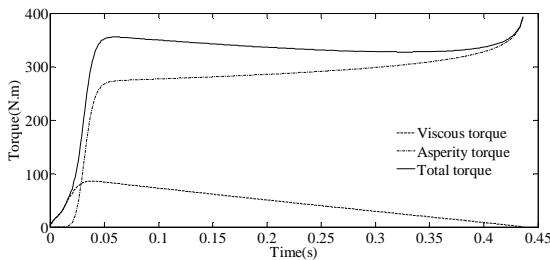

Figure 5. Effect of permeability at 4×10^{-13} m

C. Applied Pressure Effect

The applied pressure is a significant parameter for the engagement characteristics, largely influencing the engagement time. Oil film thickness decreases quickly with high applied pressure. The asperity force and torque increase when the oil film thickness reaches its constant value, so the ratio of angular velocity of steel plate increases, shortening the engagement

time. Also, the viscous torque increases a little. Fig. 6 indicates that when the applied pressure increases, both of the asperity torque and total torque are bigger than that with lower applied pressure. The engagement time is 0.07s shorter than that in Figure. 1.
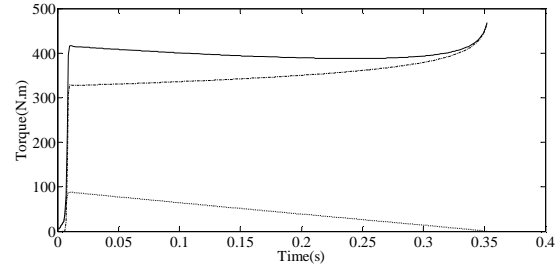

Figure 6. Influence of oil pressure at 1.2×10^6 Pa

V CONCLUSION

A mathematical model for the engagement of wet clutches was developed to predict the torque response during engagement of wet clutches. The modified Reynolds equation, including the effects of surface roughness, friction material permeability, and centrifugal force of oil in the gap of plates, was modeled to describe the hydrodynamic period of engagement. The transferred torque and velocity of steel plate can be obtained from the force balance equation and torque balance equation. The results show that viscosity affects the viscous torque, which can be concluded from the model theoretically. Also, the influence of applied pressure on engagement time is much bigger than other parameters. The material permeability has a great effect on both torque and engagement time. However, some parameters are not included in the model, such as thermal effect and grooves, which will be emphasized in the future research.

REFERENCES

[1] S Natsumeda, T Miyoshi. Numerical simulation of engagement of paper based wet cutch facing [J]. Journal of Tribology, 1994, vol.2,pp.232-237.

[2] E J Berger, F Sadeghi, C M Krousgrill. Analytical and numerical modeling of engagement of rough, permeable, grooved wet clutches [J].Journal of Tribology,1997,vol. 119,pp.143-148.

[3] C L Davis, F Sadeghi, C M Krousgrill. A simplified approach to modeling thermal effects in wet clutch engagement:analytical and experimental comparison[J].Journal of Tribology,2000,vol.122,pp.110-118.

[4] H Gao, G C Barber, M Shillor. Numerical simulation of engagement of wet clutch with skewed surface roughness [J]. Journal of Tribology,2002,vol.124,pp.305-312.

[5] Zhang Zhigang,Zhou Xiaojun. Simulation and experiments on dynamic engagement characteristics of wet clutch[J].China Journal of Highway and Transport,2010,vol.23,pp.115-120.

[6] N Patir, H S Cheng. An average flow model for determining effects of three-dimensional toughness on partial hydrodynamic lubrication [J].Journal of Lubrication Technology,1979,vol.100,pp.12-17.

[7] N Patir, H S Cheng. Application of average flow model to lubrication between rough sliding surfaces[J]. Journal of Lubrication Technology,1979,vol.101,pp.220-230.