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Online Estimation of Clutch Drag Torque in Wet Dual Clutch Transmission Based on Recursive Least Squares

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***Abstract*—**This paper focuses on developing an estimation method of clutch drag torque in wet DCT. The modelling of clutch drag torque is investigated. As the main factor affecting the clutch drag torque, dynamic viscosity of oil is discussed. The paper proposes an estimation method of clutch drag torque based on recursive least squares by utilizing the dynamic equations of gear shifting synchronization process. The results demonstrate that the estimation method has good accuracy and efficiency.

***Keywords*—**Clutch drag torque, wet DCT, dynamic viscosity, recursive least squares.

I. Introduction

RECENTLY, wet dual clutch transmissions (wet DCT) have equipped in more and more passenger vehicles since wet DCT provide good ride comfort during gear shifting[1]. One of the main differences between wet DCT and other automatic transmissions is that clutch pack is fully filled with oil. Thus, the clutch drag torque affects the operation of wet DCT significantly [2], [3]. Most researches about the clutch drag torque focus on the modelling process. Kitabayashi et al. have investigated the different factors affecting the clutch drag torque and presented a classic model [4]. Yuan et al. and Li et al. have both considered the shrink effect and presented the improved models separately [5], [6]. Similar clutch drag torque models have also been presented recently [7]-[9]. However, the clutch drag torque model is difficult to be applied in the control unit in wet DCT for its complexity. In order to obtain clutch drag torque online, a novel estimation method of clutch drag torque is proposed in this paper. The modelling of clutch drag torque is discussed firstly. Then, the estimation method is deduced according to the dynamic equations of synchronization process. Finally, the simulation is processed to evaluate the accuracy and efficiency.

II. Modelling of Clutch Drag Torque in Wet DCT

Fig. 1 shows the schematic of disengaged clutch pack in wet DCT. r_{in} is the inner radius of the friction plate, r_{out} is the outer radius of the friction plate, h is the clearance between separate plate and friction plate, $\Delta\omega$ is the relative rotation speed between separate plate and friction plate, and finally Q is the flow rate through the clutch pack.

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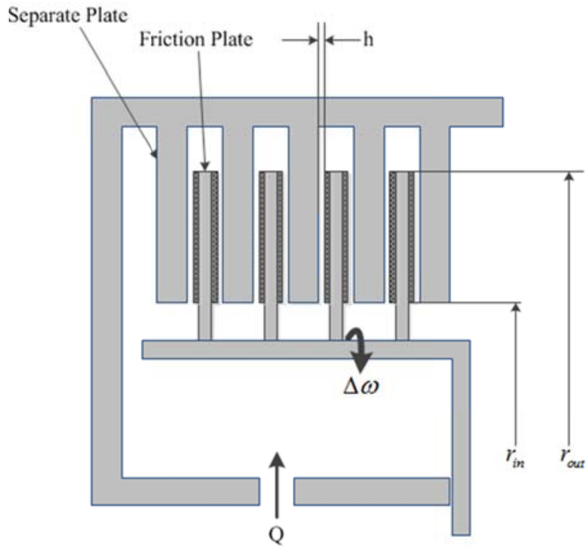

Fig. 1 The schematic of disengaged clutch pack in wet DCT

Kitabayashi proposed an ideal clutch drag torque model by considering different design factors of the clutch, such as the facing area of the friction plate, the number of grooves and the flow rate [4]. The clutch drag torque is calculated by (1). N is the number of the friction plates, μ is dynamic viscosity of oil, r_m is the average radius of the friction plate.

$$T = \frac{N\mu(r_{out}^2 - r_{in}^2)r_m^2}{2h}\Delta\omega \tag{1}$$

Yuan and Li both referred that the ideal clutch drag torque model is not accurate in high rotation speed range since the oil film will shrink when the difference of rotation speeds increases [5], [6]. The outer radius r_{out} in (1) should be replaced by effective outer radius r_e. However, Yuan and Li adopted different ways to calculate the effective outer radius [5], [6].

Yuan proposed a method to calculate the effective outer radius based on the surface tension force [5]. The effective outer radius r_e can be solved by (2) where ρ is the oil density, f and G_r both are the turbulence coefficients.

$$\frac{\rho\Delta\omega^2}{2}\left(f+\frac{1}{4}\right)r_e^2-\frac{\mu Q}{2\pi r_m h^3 G_r}r_e+\frac{\mu Q}{2\pi r_m h^3 G_r}r_{in} -\frac{2\pi\cos\theta}{h}-\frac{\rho\Delta\omega^2}{2}\left(f+\frac{1}{4}\right)r_{in}^2=0 \qquad (2)$$

Li offered an alternative method to calculate the effective outer radius based on the flow rate [6]. The effective outer radius can be obtained by (3). Q_n is the flow rate needed to maintain the full film state.

$$\begin{cases} r_e = r_{out}, & Q \geq Q_n. \\ r_e = \sqrt{\dfrac{Q_n}{Q}r_{out}^2 + r_{in}^2\left(1-\dfrac{Q_n}{Q}\right)} & Q < Q_n. \end{cases} \qquad (3)$$

According to (3), the oil film would shrink when the actual flow rate Q is less than the flow rate needed to maintain full film state Q_n. Moreover, Q_n is derived from Navier-Strokes equations by considering the centrifugal effect as (4).

$$Q_n=\frac{\dfrac{6\mu}{\pi h^3}\ln\dfrac{r_1}{r_2}}{\dfrac{27\rho}{70\pi^2h^2}(r_2^{-2}-r_1^{-2})}+\frac{\sqrt{\left(\dfrac{6\mu}{\pi h^3}\ln\dfrac{r_1}{r_2}\right)^2-\dfrac{81\rho^2\Delta\omega_{clutch}^2(r_2^{-2}-r_1^{-2})-540\rho(r_2^{-2}-r_1^{-2})\Delta p}{700\pi^2h^2}}}{\dfrac{27\rho}{70\pi^2h^2}(r_2^{-2}-r_1^{-2})} \qquad (4)$$

Fig. 2 shows the comparison of these three clutch drag torque models. The results indicate that clutch drag torque calculated by these three models is almost same when the difference of rotation speed is in low speed range, and the clutch drag torque decreases when the difference of rotation speed is in high speed range.

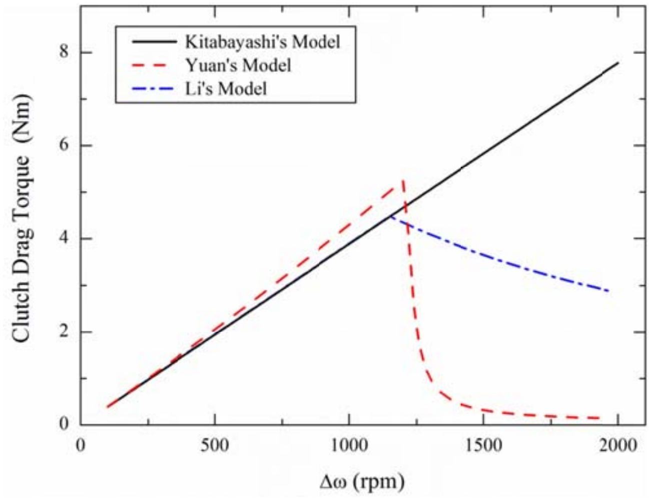

Fig. 2 Comparison of three clutch drag torque models

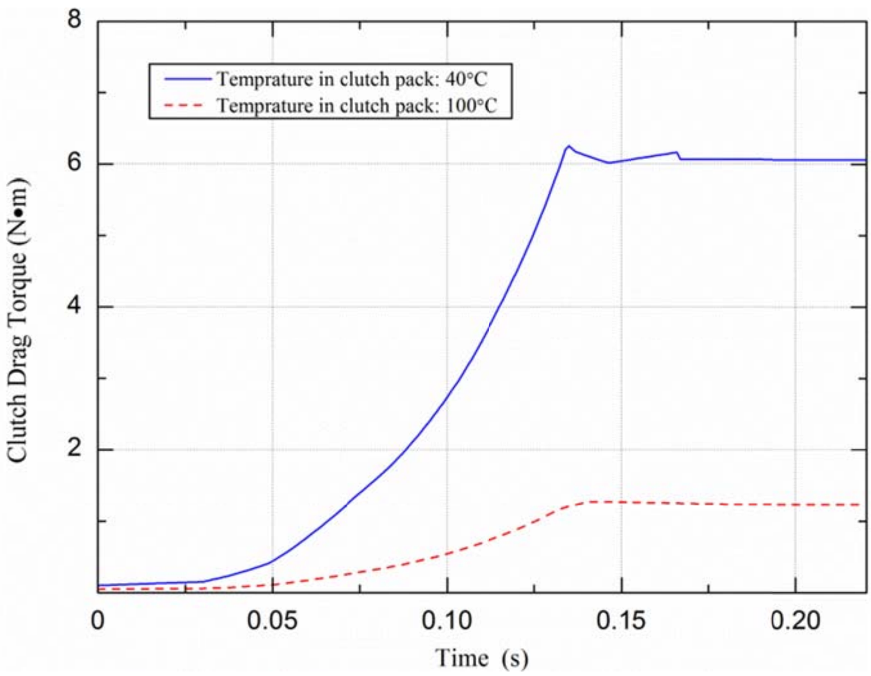

Fig. 3 Comparison of clutch drag torque at 40 ℃ and 100 ℃

Yuan and Li both derived their model based on an assumption that the flow rate through the clutch pack is constant [5], [6]. However, the flow rate through the clutch pack is compensated by the transmission control unit in order to ensure the cooling ability. Thus, the shrink effect can be neglected in practice. Equation (1) has enough accuracy to calculate the clutch drag torque.

Since the design factors of the clutch pack are almost kept constant, the clutch drag torque is proportional to dynamic viscosity of oil μ according to (1). Thus, (1) can be rewritten as (5):

$$\begin{cases} T = k\Delta\omega \\ k = \dfrac{N\pi\mu(r_{out}^4 - r_{in}^4)}{2h} \end{cases} \tag{5}$$

Gustavsson has demonstrated that dynamic viscosity of oil μ is mainly affected by temperature [3]. Fig. 3 shows the comparison of clutch drag torque at 40 ℃ and 100 ℃. The dynamic viscosity of oil μ is 0.0315 Pa.s at 40 ℃ and 0.0063 Pa.s at 100 ℃.

III. Online Estimation of Clutch Drag Torque

Walker et al. have presented that the clutch drag torque affects the synchronization process in wet DCT significantly [2]. However, the clutch drag torque is difficult to be measured directly. According to (5), the clutch drag torque can be calculated if k is determined since $\Delta\omega$ can be gained by the sensors which are located in transmission.

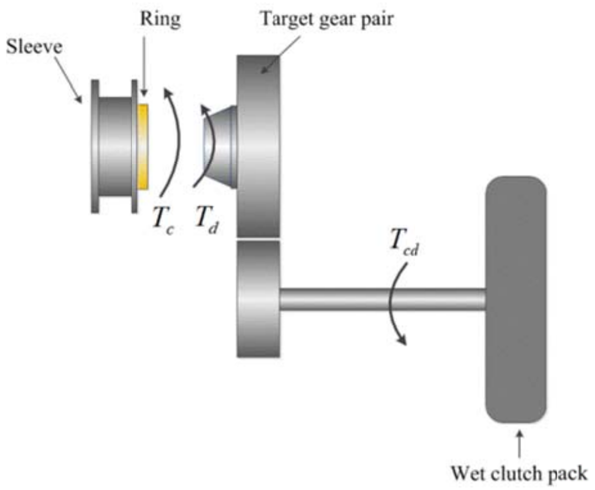

Fig. 4 Schematic of synchronizer system

Fig. 4 illustrates the schematic of synchronizer system. The dynamic equation of synchronization process is presented in (6). J is the equivalent moment of inertia at target gear, T_c is the cone friction torque, ω_1 is the rotation speed of target gear, ω_2 is the rotation speed of sleeve, ω_o is the rotation speed of open clutch, ω_c is the rotation speed of closed clutch.

$$\begin{cases} J\dot{\omega}_1 = -T_c - k\Delta\omega \\ \Delta\omega = \omega_o - \omega_c = \omega_1 i_1 - \omega_2 i_2 \end{cases} \tag{6}$$

k can be estimated by (6). Equation (6) can be written into discrete form as (7) where t_s is estimation step.

$$\frac{J}{t_s}(\omega_1(j+1) - \omega_1(j)) = -T_c(j) - i_1 k\Delta\omega(j) \tag{7}$$

Generally, the vehicle velocity remains constant during gearshift. The rotation speed of sleeve ω_2 is small since the equivalent moment of inertia at output shaft is relatively large. We can assume that $\omega_2(j+1)$ is approximately equals to $\omega_2(j)$. Then, (7) is rewritten in (8) and (9)

$$\frac{J}{t_s}\left(\frac{\omega_1(j+1)i_1 - \omega_1(j)i_1 - (\omega_2(j+1)i_2 - \omega_2(j)i_2}{i_1}\right) = -T_c(k) - i_1 k\Delta\omega(j) \tag{8}$$

$$\frac{J}{t_s i_1}(\Delta\omega(j+1) - \Delta\omega(j)) = -T_c(j) - i_1 k\Delta\omega(j) \tag{9}$$

Then, we can derive the recursive equation from (9):

$$\Delta\omega(j) = \left(1 - \frac{k t_s i_1^2}{J}\right)\Delta\omega(j-1) - \frac{t_s i_1}{J}T_c(j-1) \tag{10}$$

Let $\theta = \left[1 - \dfrac{k t_s i_1^2}{J}, -\dfrac{t_s i_1}{J}\right]^{\mathrm{T}}$, $\varphi(k) = \left[\Delta\omega(j), T_c(j)\right]^{\mathrm{T}}$, (10) can be written into recursive least squares form as (11).

$$\begin{cases} \hat{\boldsymbol{\theta}}(k) = \hat{\boldsymbol{\theta}}(k-1) + \mathbf{K}(k)\left[y(k) - \boldsymbol{\varphi}^{\mathrm{T}}(k-1)\hat{\boldsymbol{\theta}}(k-1)\right] \\ \mathbf{K}(k) = \dfrac{\mathbf{P}(k-1)\boldsymbol{\varphi}(k-1)}{\lambda + \boldsymbol{\varphi}^{\mathrm{T}}(k-1)\mathbf{P}(k-1)\boldsymbol{\varphi}(k-1)} \\ \mathbf{P}(k) = \dfrac{1}{\lambda}\left[\mathbf{I} - \mathbf{K}(k)\boldsymbol{\varphi}^{\mathrm{T}}(k-1)\right]\mathbf{P}(k-1) \end{cases} \tag{11}$$

λ is forgetting factor. The initial values $\mathbf{P}(0)$ and $\boldsymbol{\theta}(0)$ are provided by (12):

$$\begin{cases} \mathbf{P}(0) = 10^5\mathbf{I} \\ \hat{\boldsymbol{\theta}}(0) = [0,0]^{\mathrm{T}} \end{cases} \tag{12}$$

IV. Simulation and Analysis of the Online Estimation

In order to evaluate the estimation method deduced in Section III, simulations are processed.

Figs. 5 and 6 show the comparison of k and the clutch drag torque between estimation and simulation during the upshift process. The dynamic viscosity of oil μ is 0.05 Pa.s, and the forgetting factor λ is 0.9. The estimation step is 1 ms. Since the estimation process starts from the synchronization phase, the time in Fig. 5 starts at 0.05 s. The comparison results illustrate that the convergence time is about 40 ms. The synchronization time is about 200 ms in general. The convergence time satisfies the engineering requirement. The results demonstrate that the estimation method is effective and accurate.

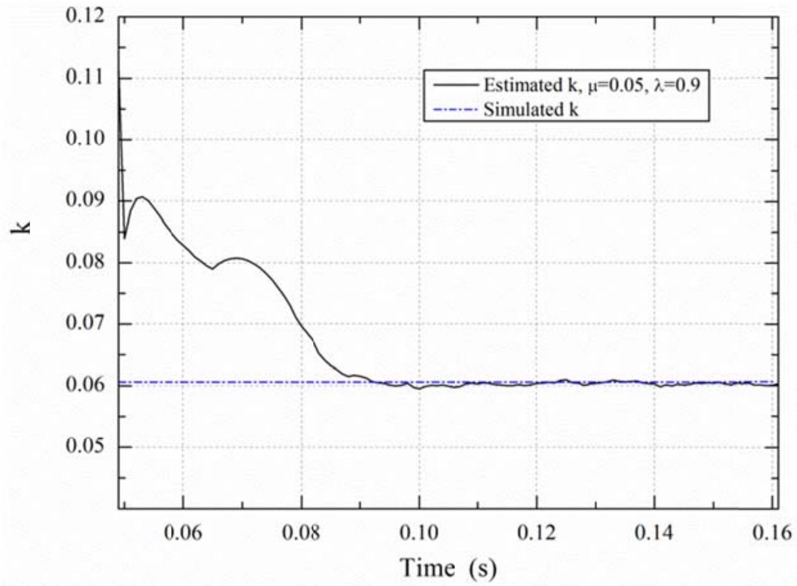

Fig. 5 Comparison of k between estimation and simulation, μ=0.05 Pa s, λ =0.9

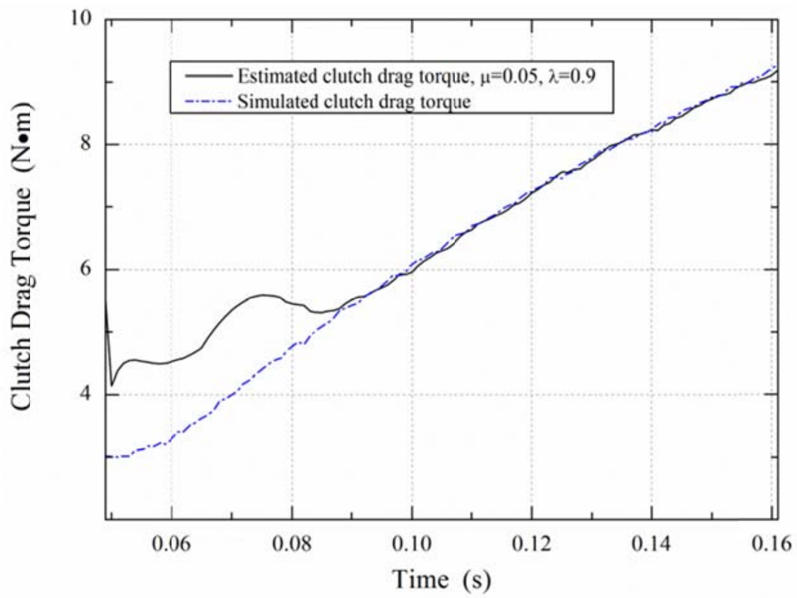

Fig. 6 Comparison of clutch drag torque, μ=0.05 Pa s, λ =0.9

Figs. 7 and 8 show the comparison of k and the clutch drag torque between estimation and simulation during the upshift process. The dynamic viscosity of oil μ is 0.0315 Pa.s, and the forgetting factor λ is 0.9. The estimation step is 1 ms.

The comparison results illustrate that the convergence time is about 50 ms. The convergence time satisfies the engineering requirement. The results also demonstrate that the estimation method is effective and accurate.

Both estimation results illustrate that the online estimation of clutch drag torque has good accuracy. The convergence time is short enough for engineering application, and the results also demonstrate that the estimation method has good performance with different dynamic viscosities. It has been demonstrated that the dynamic viscosity is the main factor affecting the clutch drag torque.

V. Conclusion

This paper has investigated the modelling of clutch drag torque in detail. Then, the main factor, dynamic viscosity, is discussed. An estimation method of clutch drag torque is derived from dynamic equations based on recursive least squares. Finally, the estimation method is verified by simulation experiment. The method can improve the performance of gear shift in wet dual clutch transmission in the future.

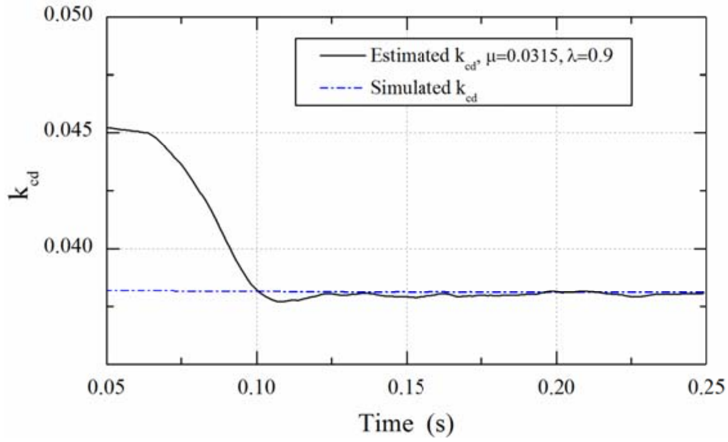

Fig. 7 Comparison of k between estimation and simulation, μ=0.0315 Pa s, λ =0.9

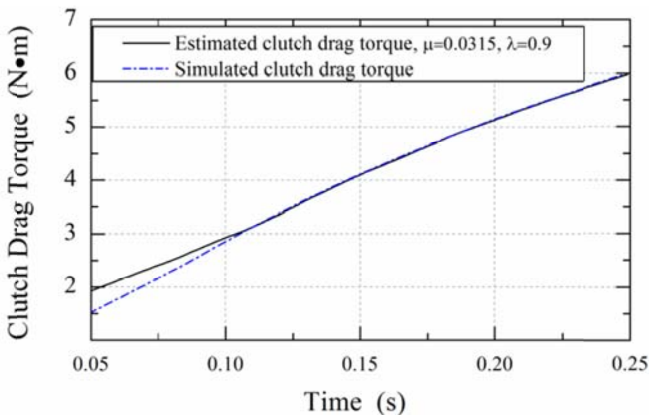

Fig. 8 Comparison of clutch drag torque, μ=0.0315 Pa s, λ =0.9

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