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Research on trajectory tracking control for wet clutch engagement based on SMC

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Abstract

To improve tracking control quality of the clutch actuator during the wet clutch engagement, models of the clutch actuator were established firstly, including the control cylinder model, flow equilibrium equation and pressure control model. Secondly, taking the clutch output speed as tracking target, the state space equation of the tracking control system was set up and the sliding mode controller (SMC) was designed. Finally, a simulation test was performed. The results show that a higher tracking accuracy as well as a better performance to resist disturbance can be achieved with the proposed sliding control method, compared to PI control. It was also shown that the exponent approaching sliding mode control can produce smaller chattering compared with the constant rate approaching sliding mode control.

Key words: wet clutch; clutch engagement; trajectory tracking;sliding mode control

1. Introduction

Wet clutches are widely used in automatic transmission (AT), continuously variable transmission (CVT), dual-clutch transmission (DCT), etc. The control of wet clutches engagement plays a key role in vehicle automatic transmission.

There are three main aspects concerning wet clutch engagement that include the optimization of the dynamic model of wet clutch, the improvement of clutch control strategy and the quality of wet clutch actuator tracking the engagement trajectory[1]. As wet clutches typically operate under varying conditions including the friction coefficient, wear and oil viscosity, the dynamics of wet clutches and their actuators are high non-linear [2]. It is very important to design a robust and precise control to meet the need of tracking the desired engagement trajectory, which is the key to realizing wet clutch control strategy and improving the performance of vehicle automatic transmission.

PID control is often used in clutch engagement. Affected by time-varying parameters of the wet clutch system, the conventional PID control has poor robustness in adjusting control parameters [3].

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Moreover, the process of clutch engagement is very short, so some too complicated control methods are not applicative. The sliding mode control (SMC) has the advantages of strong robustness and simple algorithm, whose sliding mode has full invariance to the system parameter variations and external disturbances. So it is suitable for the tracking control of wet clutch engagement.

2. Dynamic model of clutch actuator

Wet clutch engagement process can be divided into three stages. In the first stage the gap between the friction plates is to be eliminated. In the second stage, the oil pressure is increased gradually. In the third stage, the engagement process is finished and the pressure maintains a certain value.

Assuming that the distances between the friction plates have the same value h, the model of the clutch piston during clutch engagement is shown in Fig.1.

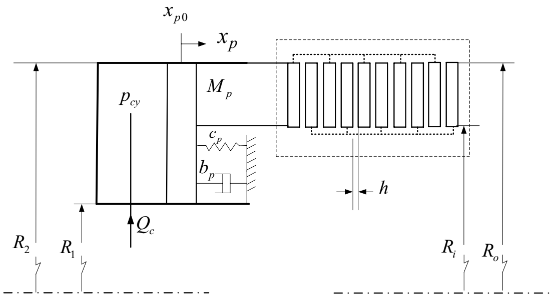

Fig.1. The clutch piston model during clutch engagement

In stage 1, while $x_p < h$, the clutch piston moves towards friction plates under the hydraulic pressure until $x_p = h$.The dynamic model of piston:

$$M_p \ddot{x}_p + c_p \dot{x}_p + k_p (x_p + x_{p0}) = F_p , \tag{1}$$

where M_p is the sum of the mass of piston and drive disks, x_p is the displacement away the initial position, x_{p0} is the initial displacement compressed on return spring, c_p is the viscous friction coefficient between the cylinder and piston, k_p is the stiffness of the clutch return spring. F_p is the force on piston that includes static pressure and dynamic pressure caused by centrifugal force.

In stage 2 and stage 3, while x_p achieves the maximum value h, the gap between the friction plates is eliminated. So the clutch motion equilibrium equation in those stages is

$$k_p (x_{p0} + h) = F_p - F_{cl} , \tag{2}$$

where F_{cl} is the clutch reaction force on the piston.

Ignoring oil leakage and oil temperature variation, the clutch cylinder fluid continuity equation[5]in the first stage of wet clutch engagement is

$$Q_c - A_p \dot{x}_p = \frac{A_p x_p + V_0}{\beta} \frac{dp_{cy}}{dt} , \tag{3}$$

where Q_c is the input flow of the clutch cylinder, β is the effective volume elastic modulus of hydraulic fluid, V_0 is the initial volume including clutch cylinder and oil feed lines.

In stage 2 and stage 3, $\dot{x}_p = 0$, then the fluid continuity equation is

$$Q_c = \frac{A_p x_{p\max} + V_0}{\beta} \frac{dp_{cy}}{dt} . \tag{4}$$

According to hydromechanics, the average flow through the inlet opening of the PWM valve is

$$Q_{in} = \tau C_d A_d \sqrt{\frac{2(p_s - p_{cy})}{\rho}}\ , \tag{5}$$

where τ, C_d , A_d are respective the duty cycle, the flow coefficient and the choke area of the PWM valve. p_s is the system supply pressure.

While the average flow through the outlet opening of the PWM valve is

$$Q_{out} = (1-\tau) C_d A_d \sqrt{\frac{2(p_{cy} - p_0)}{\rho}}\ . \tag{6}$$

where p_0 is the atmospheric pressure.

Ignoring oil leakage and the resistance in oil pipes, the flow through the clutch cylinder can be considered as $Q_c = Q_{in} - Q_{out}$. So Q_c can be written as

$$Q_c = \tau C_d A_d \sqrt{\frac{2(p_s - p_{cy})}{\rho}} - (1-\tau) C_d A_d \sqrt{\frac{2(p_{cy} - p_0)}{\rho}} = \tau C_d A_d \sqrt{\frac{2}{\rho}}(\sqrt{p_s - p_{cy}} + \sqrt{p_{cy} - p_0}) - C_d A_d \sqrt{\frac{2(p_{cy} - p_0)}{\rho}}\ . \tag{7}$$

As for the first stage during wet clutch engagement, combining Eqs.(3) and (7),the pressure control equation is

$$\frac{dp_{cy}}{dt} = \frac{\beta}{A_p x_p + V_0}(\tau C_d A_d \sqrt{\frac{2}{\rho}}(\sqrt{p_s - p_{cy}} + \sqrt{p_{cy} - p_0}) - C_d A_d \sqrt{\frac{2(p_{cy} - p_0)}{\rho}} - A_p \dot{x}_p)\ . \tag{8}$$

As for the second and the third stage during wet clutch engagement, combining Eqs.(4) and (7),the pressure control equation is

$$\frac{dp_{cy}}{dt} = \frac{\beta}{A_p x_{p_{\max}} + V_0}(\tau C_d A_d \sqrt{\frac{2}{\rho}}(\sqrt{p_s - p_{cy}} + \sqrt{p_{cy} - p_0}) - C_d A_d \sqrt{\frac{2(p_{cy} - p_0)}{\rho}})\ . \tag{9}$$

When the system supply pressure p_s keeps constant, given

$$K_1 = \frac{\beta C_d A_d}{A_p x_p + V_0}\sqrt{\frac{2}{\rho}}(\sqrt{p_s - p_{cy}} + \sqrt{p_{cy} - p_0})\ , K_2 = \frac{\beta C_d A_d}{A_p x_p + V_0}\sqrt{\frac{2(p_{cy} - p_0)}{\rho}}\ , K_3 = \frac{\beta A_p \dot{x}_p}{A_p x_p + V_0}\ ,$$

where K_1 , K_2 and K_3 are all mathematical functions of p_{cy} and x_p .So Eqs. (8) and (9) are respectively denoted by:

$$\frac{dp_{cy}}{dt} = K_1(p_{cy}, x_p)\tau - K_2(p_{cy}, x_p) - K_3(p_{cy}, x_p)\ , \tag{10}$$

$$\frac{dp_{cy}}{dt} = K_1(p_{cy}, x_{p\max})\tau - K_2(p_{cy}, x_{p\max})\ . \tag{11}$$

3. Sliding mode controller for wet clutch tracking

In this paper, a starting wet clutch for CVT vehicles is discussed. Taking the second stage and the third stage as examples, the simplified dynamic equations during clutch engagement are

$$J_e \dot{\omega}_e = T_e - c_e \omega_e - T_{cl}\ , \tag{12}$$

$$J_v \dot{\omega}_v = T_{cl} - c_v \omega_v - T_l\ , \tag{13}$$

where, J_e is the equivalent mass moments of inertia of the engine output including flywheel, J_v is the equivalent mass moments of inertia of the clutch output including the vehicle translational inertia, ω_e

and ω_v are the input shaft angular velocity and the output shaft angular velocity of clutch respectively, T_e is the equivalent torque of the engine, T_l is the equivalent torque of loads on the clutch output shaft, T_{cl} is the friction torque transmitted by the wet clutch, c_e, c_v are the equivalent viscous damping coefficients of rotation.

In the case of vehicles that no oil pressure sensor is installed, it can not use the trajectory of pressure p_{cy} directly. However, the output shaft angular velocity of clutch, ω_v, can be a indirect tracking objective for the reason that the vehicle speed has been measured .Given $x_1 = \omega_v$, $x_2 = \dot{\omega}_v$,combining Eqs.(11) and (13) ,brief writing $K_1(p_{cy}^*, x_{p\max})$, $K_2(p_{cy}^*, x_{p\max})$ as $K_1(p_{cy}^*)$ and $K_2(p_{cy}^*)$ respectively, the State space equation of the tracking control system is:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = -\dfrac{c_v}{J_v}x_2 + \dfrac{K_{cl}}{J_v}(K_1(p_{cy}^*)\tau - K_2(p_{cy}^*)) \end{cases} \tag{14}$$

where $K_{cl} = \mu z R_m$, μ is the friction coefficient that is a function of the relative speed between the friction plates, z is the number of the friction surface, R_m is the effective radius of the clutch discs.

As mentioned above, the control trajectory of clutch actuator p_{cy}^* is obtained according to a certain control strategy for clutch engagement. The trajectory of ω_v, ω_v^* ,can also be obtained according Eqs.(12) and (13).Given $r_1 = \omega_v^*$, $r_2 = \dot{\omega}_v^*$ and define tracking error $e = [e_1, e_2]^T$, where $e_1 = x_1 - r_1$, $e_2 = x_2 - r_2$.

So the differential equations of motion of deviation are:

$$\begin{cases} \dot{e}_1 = e_2 \\ \dot{e}_2 = -\dfrac{c_v}{J_v}e_2 + \dfrac{K_{cl}K_1(p_{cy}^*)}{J_v}\tau - \dfrac{K_{cl}K_2(p_{cy}^*)}{J_v} - \dfrac{c_v}{J_v}r_2 - \dot{r}_2 \end{cases}. \tag{15}$$

Define switching surfaces[6] $s = ce_1 + e_2$, the sliding surface equation is

$$ce_1 + e_2 = 0, \tag{16}$$

where c (c >0) is sliding surface coefficient, which is selected according Hurwitz stability criterion [6].

An exponent approaching is adapted to ensure the quality of normal movement,

$$\dot{s} = -\varepsilon\,\mathrm{sgn}(s) - ks, \tag{17}$$

where $k > 0, \varepsilon > 0$ .If $k = 0$, Eq.(17) is the equation of constant rate approaching.

Combining Eqs.(16) and (17),the sliding mode control law is

$$\tau = \frac{J_v}{K_{cl}K_1(p_{cy}^*)}\left[-\varepsilon\,\mathrm{sgn}(s) - ks + (c_1 - \frac{c_v}{J_v})e_2 - \frac{K_{cl}K_2(p_{cy}^*)}{J_v} - \frac{c_v}{J_v}r_2 - \dot{r}_2\right]. \tag{18}$$

4. Simulation results

The simulation model is created using MATLAB/SIMULINK platform. According the simulation, the method of cross validation is adopted to select suitable sliding mode control parameters. For the exponent approaching SMC in this paper, c =20， ε =15000， k =4.5, the values of other parameters were given in [7].

Fig.2 shows the tracking results of SMC and PI control in ideal conditions without disturbances. Comparing two tracking results, it can be seen that both control methods can achieve high tracking accuracy. Although a small chattering, higher accuracy is obtained using SMC, with maximum absolute error(MAE) 0.83rad/s and relative error(RE) 0.81%, while 2.5rad/s and 2.1% respectively using PI.

The results with load variation are shown as Fig.3. It can be seen that SMC still achieves a good performance and it has better robustness compared with PI control.

Fig.4 shows the result of tracking error using exponent approaching SMC and constant rate approaching SMC. It can be seen that the former has smaller chattering with a similar tracking accuracy compared with the latter.

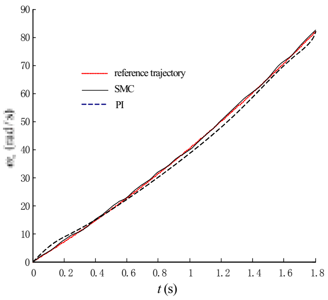

Fig.2. Tracking results in ideal conditions

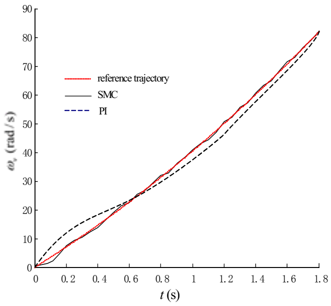

Fig.3. Tracking results with load variation

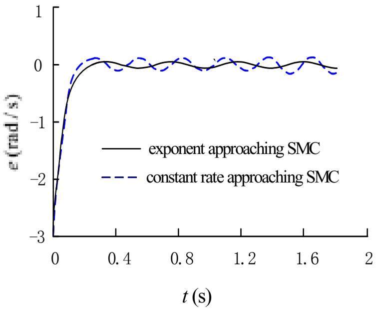

Fig.4. Tracking errors

5. Conclusions

Trajectory tracking control of wet clutch engagement is the key to automatic transmission technology. In this paper, a detailed wet clutch hydraulic actuator model in there stages was presented. Taking into account a variety of complex non-linear factors, a sliding mode controller is designed. The simulation results show that the controller has high precision and powerful disturbance resistance and it is proved effective in wet clutch engagement controlling. Moreover, compared with constant rate approaching SMC, exponent approaching SMC can alleviate chattering.

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References

[1] Wu Guangqiang,Yang Weibin and Qin Datong. Key technique of dual clutch transmission control system.Chinese J.Mech.Eng, 2007,43(2):13-21 (in Chinese)

[2] G. Pinte, B. Depraetere, W. Symens, J. Swevers, P. Sas. Iterative learning control for the filling of wet clutches.Mechanical System and Signal Processing, 2010, 24(7):1924-1937

[3] Xie Xianping,Wang Xudong,Yu Tengwei,Wu Xiaogang. Study of Precision Position Tracking Control and Starting Control of Automatic Clutch.China Journal of Highway and Transport,2008,21(4):117-121(in Chinese)

[4] Ma Biao,Liu Ying,Chen Jianwen.Study on Dynamic Performance in Engagement Process of Power Shift-Steering Transmission Shift Clutch. China Mechanical Engineering,2000,11(5):691-694(in Chinese)

[5] R. Morselli, R. Zanasi, R. Cirsone, E. Sereni, E. Bedogni, E. Sedoni. Dynamic modeling and control of electro-hydraulic wet clutches[C]. Proceedings of the IEEE Conference on Intelligent Transportation Systems, 2003, 1: 660–665.

[6] G. Bartolini,L. Fridman,A. Pisano. Modern Sliding Mode Control Theory-New Perspectives and Applications.Springer-Verlag Berlin Heidelberg, 2008.

[7] Hu Hongwei. Study on the characteristic of automatic wet clutch engagement process.Hangzhou:Zhejiang University,2008.(in Chinese)