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Sliding Mode Control of A Dry-Type Two-Speed Dual Clutch Transmission for An Electric Vehicle During Optimal Power Transmission Process in Torque Phase

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Abstract—**In order to properly control slip-to-slip shift process of electric vehicles (EVs) equipped with dry dual-clutch transmissions (DDCT), dynamic modelling of a dry-type two-speed dual-clutch transmission for an electric vehicle in torque phase is proposed at first. Then, an optimal control strategy is proposed to investigate feasibility of non-shock shift process of torque phase without power interruption and power circulation. Optimal solutions of electric motor torque and dual clutch friction torques are derived in analytical form. And then, a dynamic model of fork-lever actuator is integrated into DDCT driveline dynamic model of EV, and an affine nonlinear shift dynamic model for the whole DDCT system is proposed to describe dynamic behaviours in torque phases of shift. Further, to solve problems in tracking inaccuracy induced by strong nonlinearities and modelling uncertainties of the dynamic model, sliding mode control strategy based on feedback linearization control theory is proposed. Accurate electric motor torque as well as motor currents imposed to affine nonlinear system are calculated through nonlinear feedback control law. Finally, tracking control accuracy of the affine nonlinear dynamic system is investigated through numerical simulation on MATLAB/Simulink platform. The simulation results verify that not only non-shock shift process of torque phase without power interruption and power circulation is realized, but also torque phase time can be adjusted to an arbitrary value based on actual requirement. Besides, high-precision tracings to optimal angular velocities of electric motor and dual clutch are realized by accurately modulating motor control currents and electric motor torque.**

Index Terms—**Electric vehicle, dry-type dual clutch transmission, fork-lever actuator driven by ball screw-roller assembly, optimal control, feedback linearization control, sliding mode control.**

I. Introduction

Nowadays, electric vehicles (EVs) have been widely used in public transportations. Among them, such kind of EVs equipped with multi-speed transmissions seems to acquire more practical values[1-3]. By expanding gear ratio range, disadvantage of a relatively low driving efficiency at a high speed in single speed EV is eliminated[4]. However, increase of gear range raises manufacturing cost, and reduces transmission efficiency[5]. Therefore, two-speed EVs are more common at present, due to very good compromise of transmission efficiency and manufacturing cost[6-8]. Nevertheless, vast production depends on solutions of torque interruptions during shift, which cause evident shift jerks and are intrinsic in EVs equipped with AMT.

Meanwhile, dual-clutch transmission (DCT) has been widely used in conventional vehicles[9-11]. By properly controlling slip-to-slip process of DCT, power interruption is prohibited, and smooth shift quality with no jerk is achieved. Therefore, DCT is a very ideal choice for two-speed EV.

So far, many published researches[11-15] have investigated optimal ways of power transmission between engine and dual clutch. As for conventional vehicles, six-speed DCT are commonly used, and such research is focused on how to provide excellent driving experiences at cost of proper power consumptions and mechanism service lives. As a result, previous studies concerning investigation of fine torque phase shift process with comprehensively least friction wear and shock intensity have been explored thoroughly. Weighted terms of friction work, shock intensity and that of engine torque or engine acceleration are frequently defined as control objectives[22,24]. These control objectives are integrated into objective functional formulations for deriving optimal torque transfer process based upon Pontryagin's minimum principle[22,24]. Besides, other intelligent control algorithms, such as orthogonal experimental design method[11], genetic algorithm[12], fuzzy neural network control algorithm[13], intelligent PID control algorithm[14] and so on are also developed to simultaneously minimize shock intensity and friction work as far as possible. However, as dual clutch slip with respect to engine during the torque phase, power interruption or power circulation occurs, both of which cause decrease in power transmission efficiency[23]. One feasible way to solve this problem is to ensure engine speed equal to that of low gear clutch[23], which has been investigated and verified by Jikai Liu[23]. Unfortunately, shock intensity value still remains obvious and friction work could be reduced to a more reasonable level.

Robustness of tracking control accuracy is another key issue to concern. Therefore, it is very necessary to investigate dynamics and control of dual clutch actuators. Most of such investigations focus on dynamics and control of electro-hydraulic actuators[16,17]. As most manufacturers tend to expect actuators response to drivers' demands as quickly and precisely as possible, tend to simplify manufacturing processes and reduce costs as far as possible, and also tend to arrange actuators in narrowest possible spaces, the fork-lever actuator swayed by ball-screw driving rollers is invented and becomes more and more popular[18,19]. Since dynamic model and key structure parameters of such actuator have been published by Wu[22], it is feasible to derive various tracking control algorithms with strong robustness to

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modelling uncertainties and disturbances, such as friction plate wear, friction coefficient variation, road disturbances and so on.

As for EV equipped with two-speed dry DCT, shift is accomplished by proper control of slip-to-slip process for smooth power transmission between dual clutch and electric driving motor, without synchronizer[5]. Similar to that of six-speed DCT, power interruption or power circulation must also be prohibited. Besides, like that of six-speed DCT, shock intensity and friction work ought to be reduced to a more reasonable level. Consequently, in the present research, a novel dual clutch control strategy for slip-to-slip process during torque phase, with only one clutch slipping, is proposed. And optimal solutions of electric motor torque and dual clutch friction torques are derived in analytical form. In comparison with single slipping clutch strategy during the torque phase proposed in for six-speed DCT[23], and compared to those analytical strategies derived through Pontryagin's minimum principle[22,24], non-shock shift process of torque phase without power interruption and power circulation is realized at first; and secondly, torque phase time can be adjusted to an arbitrary value based on actual requirement; and thirdly, friction work value can be calculated directly through derived analytical relation between friction work and torque phase time. Less torque phase time indicates much less friction work. Further, to develop tracking control strategy with strong robustness, state space model of the fork-lever actuator is integrated into the torque phase dynamic model of the whole Dry Dual Clutch Transmission (DDCT) system. In view of strong nonlinearities of the affine nonlinear DDCT dynamic model, which are induced by integration of state space models of the fork-lever actuators, the sliding mode control strategy based on feedback linearization theory is proposed and corresponding algorithm is developed. Finally, numerical simulation is implemented to study tracking accuracy of angular velocities of electric motor and dual clutch. Simulation results show that the proposed sliding mode control algorithm is valid and acquires very strong robustness to model nonlinearities and modelling uncertainties. Besides, dynamic models of fork-lever actuators in state space form are beneficial to follow-up studies involving estimations and optimal designs of key structure parameters.

The control strategy proposed in this paper is also suitable to that of six-speed DCT in conventional vehicles.

II. Dynamic Modelling of a Dry-Type Two-Speed Dual-Clutch Transmission for an Electric Vehicle in Torque Phase

The DDCT driveline(as shown in Fig.1) of two-speed EV is regarded as an assembly of rigid rotating components, which means only the moment of inertia of each component is considered and the elasticity-damping properties of the driveline are ignored for eliminating unnecessary computational burdens, as indicated in[20].

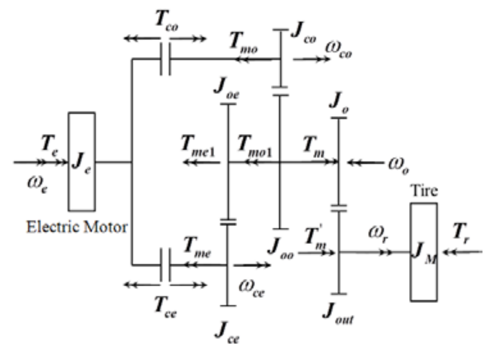


Fig. 1 The simplified dynamic model of the DDCT driveline

The dynamic equations of the dual clutch engagement in the torque phase are presented as follows

$$\begin{cases} J_e\dot{\omega}_e = T_e - \left(T_{co} + T_{ce}\right) \\ J_{co}\dot{\omega}_{co} = T_{co} - T_{mo} \\ J_{ce}\dot{\omega}_{ce} = T_{ce} - T_{me} \\ \left(J_{oo} + J_{oe} + J_o\right)\dot{\omega}_o = T_{mo1} + T_{me1} - T_m \\ \left(J_{out} + J_M\right)\dot{\omega}_r = T_m{}' - T_r \end{cases} \tag{1}$$

The resisting moment T_r exerted on the tire is calculated as follows :

$$T_r = r_w\left(F_f + F_a + F_i\right) \tag{2}$$

and tire rolling friction resistance F_f, wind resistance F_a and gradient resistance F_i are calculated as follows :

$$F_f = \left(M + M_H\right)\cdot g\cdot f \tag{3}$$

$$F_a = \frac{C_D\cdot A\cdot v^2}{21.25} \tag{4}$$

$$F_i = \left(M + M_H\right)\cdot g\cdot \sin\alpha \tag{5}$$

with $k = \frac{r_w}{i_o}$, $f = f_0\left(1 + \frac{v^2}{19440}\right)$, where f is rolling resistance coefficient, f_0 is static rolling resistance coefficient, g is acceleration of gravity, v is vehicle velocity, r_w is wheel rolling radius, A is frontal area of the vehicle, C_D is air resistance coefficient, α is road gradient, M is total vehicle mass without load, M_H is mass of the load.

III. Optimal Process Control

In previous researches[22,24], to simultaneously optimize shock intensity, friction wear and electric motor/engine acceleration/torque to a comprehensively more excellent level, such kind of following nonlinear objective functional is frequently established for seeking optimal compromise among friction work, shock intensity and electric motor/engine acceleration/torque. Such functional is presented typically as follows[22,24]:

$$J=\int_{t_{Ti}}^{t_{Te}}\Big[Q_1T_{co}\left(\omega_e-i_{go}\omega_o\right)+Q_2T_{ce}\left(\omega_e-i_{ge}\omega_o\right)+ k^2Q_3\left(\frac{i_{go}\dot{T}_{co}+i_{ge}\dot{T}_{ce}}{J_{eq}}\right)^2+Q_4\left(\frac{T_e-T_{co}-T_{ce}}{J_e}\right)^2\Big]dt \tag{6}$$

As for present research, similar to those presented in six-speed DCT in conventional vehicle[22,24], analytical solutions of the optimal electric motor torque and dual clutch friction torques are derived as following, according to the extremum value theorem.

$$\begin{cases} T_e=T_{11}\cdot\left[e^{\lambda(t-t_{Ti})}-1\right]+T_{12}\cdot\left(t-t_{Ti}\right)+T_e\left(t_{Ti}\right) \\ T_{co}=T_{21}\cdot\left[e^{\lambda(t-t_{Ti})}-1\right]+T_{22}\cdot\left(t-t_{Ti}\right)+T_{co}\left(t_{Ti}\right) \\ T_{ce}=T_{31}\cdot\left[e^{\lambda(t-t_{Ti})}-1\right]+T_{32}\cdot\left(t-t_{Ti}\right)+T_{ce}\left(t_{Ti}\right) \end{cases} \tag{7}$$

where

$$\lambda=\frac{Q_1\left(i_{go}Q_2-i_{ge}Q_1\right)J_{eq}}{2Q_4\left(Q_1-Q_2\right)i_{go}^2i_{ge}},\quad k=\frac{r_w}{i_o},$$

$$T_{11}=\left(\frac{i_{ge}-i_{go}}{Q_2i_{go}-Q_1i_{ge}}-\frac{J_e}{2Q_4\lambda}\right)\times \left[\frac{Q_1J_{eq}}{2Q_3i_{go}k^2\lambda}\lambda_{3,0}-Q_1T_{co}\left(t_{Ti}\right)-Q_2T_{ce}\left(t_{Ti}\right)\right],$$

$$T_{21}=\frac{\dfrac{Q_1i_{ge}J_{eq}\lambda_{3,0}}{2Q_3i_{go}k^2\lambda}-i_{ge}\left(Q_1T_{co}\left(t_{Ti}\right)+Q_2T_{ce}\left(t_{Ti}\right)\right)}{Q_2i_{go}-Q_1i_{ge}},$$

$$T_{31}=\frac{-\dfrac{Q_1J_{eq}\lambda_{3,0}}{2Q_3k^2\lambda}+i_{go}\left(Q_1T_{co}\left(t_{Ti}\right)+Q_2T_{ce}\left(t_{Ti}\right)\right)}{Q_2i_{go}-Q_1i_{ge}},$$

$$T_{12}=\left(\frac{Q_1-Q_2}{Q_2i_{go}-Q_1i_{ge}}+\frac{Q_1J_e}{2Q_4i_{go}\lambda}\right)\frac{J_{eq}\lambda_{3,0}}{2Q_3k^2},$$

$$T_{22}=-\frac{Q_2J_{eq}\lambda_{3,0}}{2Q_3\left(Q_2i_{go}-Q_1i_{ge}\right)k^2},\quad T_{32}=\frac{Q_1J_{eq}\lambda_{3,0}}{2Q_3\left(Q_2i_{go}-Q_1i_{ge}\right)k^2},$$

$$J_{eq}=\frac{J_{out}+J_M}{i_o^2}+i_{go}^2J_{co}+i_{ge}^2J_{ce}+J_{oo}+J_{oe}+J_o.$$

Corresponding simulation results in aspect of control torque, angular velocity, friction works and shock intensity are investigated to testify optimality of the control strategy, as shown in Fig.2. The simulated shift condition is a typical one concerning a 1st gear to 2nd gear shift on a 6º ramp road. The torque phase time $\left(t_{T_f}-t_{T_i}\right)$ is 0.03s when Q1=400, Q2=0.004, Q3=110, Q4=0.23. Whilst $\left(t_{T_f}-t_{T_i}\right)$ is reduced to 0.01s when Q1=160, Q2=0.0016, Q3=110, Q4=0.21. Other simulation parameters of the DDCT driveline dynamic systems are given in Table 1.

TABLE 1 PARAMETERS OF THE WHOLE DDCT DRIVELINE

Parameter	Value and Unit	
	Value	*Unit*
Je	0.007	kg*m^2
Jco	0.0015	kg*m^2
Jce	0.001	kg*m^2
J_M	150	kg*m^2
Jout	0.005	kg*m^2
Joo	0.005	kg*m^2
Joe	0.003	kg*m^2
Jo	0.001	kg*m^2
$M+M_H$	1600	kg
igo, ige,io	3.6, 2.16, 4.2	
r_w	0.3	m
f_0	0.02	

Through careful observations of numerical simulation results, a general conclusion can be derived, which is summarized as: gradual increase in Q1/Q3 and Q2/Q3 value yields gradual increase of shock intensity and gradual decrease of friction loss. However, since two clutches slip simultaneously in torque phase, as shown in Fig.2(b), power interruption and power circulation phenomena occur at first. Secondly, shock intensity value is non-zero, as shown in Fig.2(d), which lowers ride comfort in torque phase. Thirdly, through careful observation of control laws in Eq.(7), it can be concluded that parameter calculations and estimations are tedious, due to complex presentations of Eq.(7). Finally, it is very difficult to calculate friction work in accurate and simple analytical form, and influential factor of friction work seems confusing. Consequently, reducing friction work to a more reasonable level is difficult.

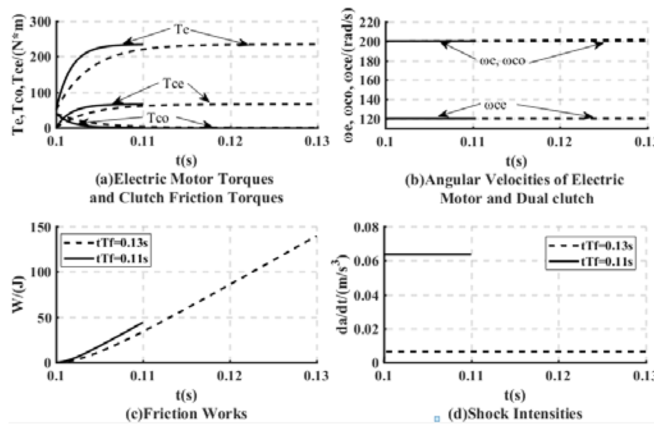


Fig.2 Shift qualities of a typical 1st to 2nd gear shift

To prohibit power interruption and power circulation, as indicated in [23], ω_e should be equal to $i_{go}\omega_o$. Consequently, following acceleration equation can be derived.

$$\dot{\omega}_e=i_{go}\dot{\omega}_o \tag{8}$$

Also, to achieve non-shock torque phase of shift, following equation should be satisfied,

$$i_{go}\dot{T}_{co}+i_{ge}\dot{T}_{ce}=0 \tag{9}$$

Consequently, total output torque of DCT applied to differential gear is constant and can be calculated as following.

$$i_{go}T_{co}+i_{ge}T_{ce}=J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\frac{T_r}{i_o} \tag{10}$$

And then, analytical solution of ω_o can be derived as following.

$$\omega_o=\dot{\omega}_o\left(t_{T_i}\right)\cdot\left(t-t_{T_i}\right) \tag{11}$$

Referring to Eq.(8), engine acceleration and speed can also be derived as following.

$$\dot{\omega}_e=i_{go}\dot{\omega}_o\left(t_{T_i}\right) \tag{12}$$

$$\omega_e=i_{go}\dot{\omega}_o\left(t_{T_i}\right)\cdot\left(t-t_{T_i}\right) \tag{13}$$

During the torque phase, as one clutch torque increases from zero to an assignment value, and another decreases from its initial value to zero, analytical solution of dual clutch friction torques can be derived and presented as following.

Odd gear to even gear shift

$$\begin{cases} T_{co}=\dfrac{1}{i_{go}}\left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\dfrac{T_r}{i_o}\right)-\dfrac{1}{i_{go}\cdot\left(t_{T_f}-t_{T_i}\right)} \\ \left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\dfrac{T_r}{i_o}\right)\left(t-t_{T_i}\right) \\ T_{ce}=\dfrac{1}{i_{ge}\cdot\left(t_{T_f}-t_{T_i}\right)}\left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\dfrac{T_r}{i_o}\right)\left(t-t_{T_i}\right) \end{cases} \tag{14}$$

Even gear to odd gear shift

$$\begin{cases} T_{co}=\dfrac{1}{i_{go}\cdot\left(t_{T_f}-t_{T_i}\right)}\left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\dfrac{T_r}{i_o}\right)\left(t-t_{T_i}\right) \\ T_{ce}=\dfrac{1}{i_{ge}}\left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\dfrac{T_r}{i_o}\right)-\dfrac{1}{i_{ge}\cdot\left(t_{T_f}-t_{T_i}\right)} \\ \left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\dfrac{T_r}{i_o}\right)\left(t-t_{T_i}\right) \end{cases} \tag{15}$$

Further, as for dynamics of electric motor, following equation can be established.

$$\dot{\omega}_e=\left(T_e-T_{co}-T_{ce}\right)/J_e \tag{16}$$

On substituting Eqs.(13)-(15) into Eq.(16), analytical solution of electric motor torque is derived as following.

Odd gear to even gear shift

$$T_e=\frac{i_{go}-i_{ge}}{i_{go}i_{ge}\cdot\left(t_{T_f}-t_{T_i}\right)}\left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\frac{T_r}{i_o}\right)\left(t-t_{T_i}\right)+ \\ \left(i_{go}J_e+\frac{J_{eq}}{i_{go}}\right)\dot{\omega}_o\left(t_{T_i}\right)+\frac{T_r}{i_{go}i_o} \tag{17}$$

Even gear to odd gear shift

$$J_e\dot{\omega}_e+T_{co}+T_{ce}=T_e \tag{18}$$

$$T_e=-\frac{i_{go}-i_{ge}}{i_{go}i_{ge}\cdot\left(t_{T_f}-t_{T_i}\right)}\left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\frac{T_r}{i_o}\right)\left(t-t_{T_i}\right)+ \\ \left(i_{go}J_e+\frac{J_{eq}}{i_{ge}}\right)\dot{\omega}_o\left(t_{T_i}\right)+\frac{T_r}{i_{ge}i_o} \tag{19}$$

Finally, friction work of DCT system during torque phase is calculated as following.

Odd gear to even gear shift

$$\frac{i_{go}-i_{ge}}{3i_{ge}}\left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\frac{T_r}{i_o}\right)\dot{\omega}_o\left(t_{T_i}\right)\cdot\left(t_{T_f}-t_{T_i}\right)^2 \tag{20}$$

Even gear to odd gear shift

$$\frac{i_{go}-i_{ge}}{6i_{ge}}\left(J_{eq}\dot{\omega}_o\left(t_{T_i}\right)+\frac{T_r}{i_o}\right)\dot{\omega}_o\left(t_{T_i}\right)\cdot\left(t_{T_f}-t_{T_i}\right)^2 \tag{21}$$

Through careful observation of Eqs. (20)-(21), it can be concluded that less torque phase time $\left(t_{T_f}-t_{T_i}\right)$ indicates much less friction work. Since key influential factor of friction work is torque phase time $\left(t_{T_f}-t_{T_i}\right)$, the present strategy can regulate friction work to a more reasonable level through proper shortening of torque phase time, in comparison with optimal control strategy proposed in Eq.(7). Such conclusion is verified through simulation results shown in Fig.3. Simulation parameters of the DDCT driveline dynamic systems are given in Table 1, and torque phase time $\left(t_{T_f}-t_{T_i}\right)$ also varies from 0.03s to 0.01s.

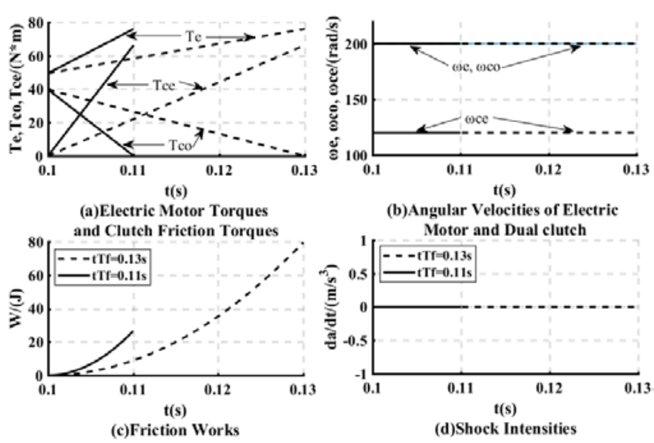


Fig.3 Shift qualities of a typical 1st to 2nd gear shift

Through careful observations of numerical simulation results, it can be concluded that in addition to the advantage mentioned above, power interruption and power circulation are prohibited with one clutch slipping in torque phase, as shown in Fig.3(b). Besides, non-shock shift process in torque phase is achieved, as shown in Fig.3(d), which indicates that smooth power transmission with perfect ride comfort is feasible. Further, in comparison with friction work calculated from optimal control strategy proposed in Eq.(7), it can be concluded that the present control strategy can reduce friction work by 50%, as shown in Fig.2(c) and Fig.3(c). This status depends on one clutch slipping situation partly. Finally, it can be discovered that electric motor torque varies very gently, from 50N*m to nearly 80N*m, as shown in Fig.3(a). However, in optimal control strategy proposed in Eq.(7), electric motor torque varies rapidly at torque phase beginning, and its peak value exceeds 200N*m, as shown in Fig.2(a). Consequently, it can be concluded the present strategy can reduce motor power consumption to a more reason level.

IV. AFFINE NONLINEAR DYNAMIC MODELLING OF A DRY-TYPE TWO-SPEED DUAL-CLUTCH TRANSMISSION WITH INTEGRATION OF NONLINEAR DYNAMIC BEHAVIOURS OF FORK-LEVER ACTUATOR

The DDCT assembly of two-speed EV is composed of electric motor, dry dual clutch body, gearbox and fork-lever actuators, as shown in Fig.4. Since dual clutch friction torque can be solely modulated by adjusting displacement of little end of diaphragm spring, and change rate of displacement of little end of diaphragm spring is inevitably in strong correlation with velocity of screw, it is very clear to indicate researchers that displacement of little end of diaphragm spring is a crucial bond to integrate dynamic models of fork-lever actuator with that of DDCT driveline. Therefore, it can be concluded that affine nonlinear dynamic model of the whole DDCT could be derived through integrating dynamic models of DDCT driveline with that of fork-lever actuators and motors. Corresponding kinematic and dynamic analyses of fork-lever actuators driven by motor-screw-roller assemblies have been investigated by Wu[22], as shown in Fig.5.

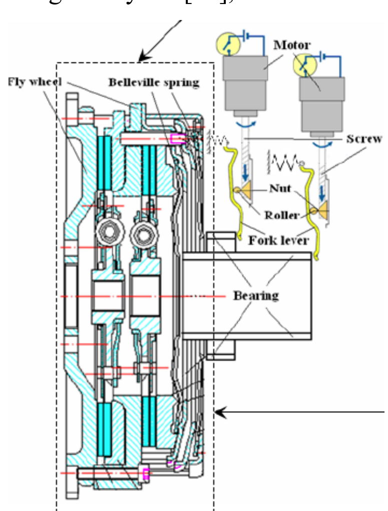
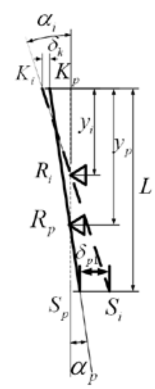
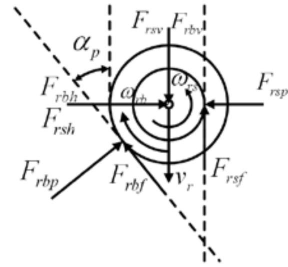


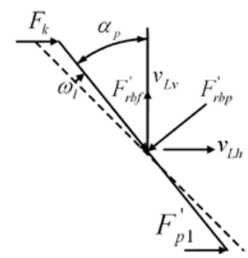
Fig. 4 The dry dual clutch system with fork-lever actuators driven by motor-screw-roller assemblies[22]



(a) The kinematic analysis of roller-fork mechanism



(b) The analysis of forces acting on the couple of rollers



(c) The analysis of forces acting on the fork lever

Fig. 5 Kinematic and dynamic analyses of fork-lever actuators driven by motor-screw-roller assemblies[22]

Referring to dynamic model of such fork-lever actuator published by Wu[22],displacement of little end of diaphragm spring δ_{p1} can be modulated by proper control of motor current I_d, as shown in Eq.(22)[22]. Since dual clutch friction torque can be modulated by properly adjusting displacement of little end of diaphragm spring[22],it can be concluded that dual clutch friction torque could be directly controlled through motor control current I_d.

$$\ddot{\delta}_{p1} = -\frac{p^2}{4\pi^2\eta_s J_m\cdot\left(dy_p/d\delta_{p1}\right)}\left[\frac{L\alpha_i-\delta_{p1}-\delta_k\left(\delta_{p1}\right)}{L}+\right.$$
$$\left.\mu_{rb}+\mu_{rs}\left(1-\mu_{rb}\frac{L\alpha_i-\delta_{p1}-\delta_k\left(\delta_{p1}\right)}{L}\right)\right]\cdot$$
$$\left[F_{p1}\left(\delta_{p1}\right)+F_k\left(\delta_k\left(\delta_{p1}\right)\right)\right]-\left(\frac{d^2y_p}{d\delta_{p1}^2}\bigg/\frac{dy_p}{d\delta_{p1}}\right)\dot{\delta}_{p1}^2+$$
$$\frac{pp_m N\Phi_m}{2\pi^2 J_m\cdot\left(dy_p/d\delta_{p1}\right)}I_d \tag{22}$$

Physical meanings of y_p, $\delta_k\left(\delta_{p1}\right)$, $F_k\left[\delta_k\left(\delta_{p1}\right)\right]$, α_i and $F_{p1}\left(\delta_{p1}\right)$ are in defined in Fig.5.

In order to analyze dynamic behaviors of the whole DDCT system of EV, and to develop proper control strategy for precise and quick tracking of optimal trajectories of angular velocities of electric motor and dual clutch, following state variables, control variables and outputs of the dynamic systems are defined:

$$x_1 = \omega_e,\ x_2 = \omega_o,\ x_3 = \delta_{p1o},\ x_4 = \dot{\delta}_{p1o},\ x_5 = \delta_{p1e},\ x_6 = \dot{\delta}_{p1e} \tag{23}$$

$$u_1 = T_e,\ u_2 = I_{do},\ u_3 = I_{de} \tag{24}$$

$$y_1 = \omega_e,\ y_2 = \delta_{p1o},\ y_3 = \delta_{p1e} \tag{25}$$

Substitution of transformations in Eqs (23) and (24) into Eqs (1) and (22) yields affine nonlinear form of the dynamic equation of the whole DDCT system of EV, and is presented as follows:

$$\dot{x} = f(x) + \sum_{i=1}^{3} g_i(x) u_i \tag{26}$$

where

$$x = [x_1, x_2, x_3, x_4, x_5, x_6]^T,$$

$$f(x) = \Bigg[-\frac{\mu_{co} \cdot F_{p3o}(x_3) + \mu_{ce} \cdot F_{p3e}(x_5)}{J_e}, \frac{i_{go}\mu_{co} \cdot F_{p3o}(x_3) + i_{ge}\mu_{ce} \cdot F_{p3e}(x_5) - M_r}{J_{eq}}, x_4, \frac{B(x_3)}{A(x_3)} + \frac{C(x_3)}{A(x_3)} x_4 + \frac{D(x_3)}{A(x_3)} x_4^2, x_6, \frac{B(x_5)}{A(x_5)} + \frac{C(x_5)}{A(x_5)} x_6 + \frac{D(x_5)}{A(x_5)} x_6^2 \Bigg]^T,$$

$$g_1(x) = \left[\frac{1}{J_e}, 0, 0, 0, 0, 0 \right]^T,$$

$$g_2(x) = \left[0, 0, 0, \frac{p_m \Phi_m N}{\pi J_M A(x_3)}, 0, 0 \right]^T,$$

$$g_3(x) = \left[0, 0, 0, 0, 0, \frac{p_m \Phi_m N}{\pi J_M A(x_5)} \right]^T.$$

Since analytical expressions of $A(x_3)$, $B(x_3)$, $C(x_3)$, $D(x_3)$, $A(x_5)$, $B(x_5)$, $C(x_5)$, $D(x_5)$ are very complex, corresponding details are omitted here.

V. SLIDING MODE CONTROL FOR THE FEEDBACK LINEARIZATION SYSTEM

After seeking optimal trajectories of angular velocities of electric motor and dual clutch, it is necessary to develop precise tracking control algorithm for shift quality improvements in real operations. However, as strong nonlinearities and modelling uncertainties such as friction coefficients and friction plates wears exist in affine nonlinear dynamic system in Eq.(26), it is inconvenient for researchers to develop high-precision PID control strategy. Consequently, in the present research, feedback linearization control strategy is proposed and corresponding sliding mode control algorithm for the topologically equivalent linear dynamic system is developed. Then, control inputs of linear system are substituted into nonlinear feedback control law to generate accurate control inputs of affine nonlinear dynamic system. Finally, tracking control accuracy of affine nonlinear dynamic system is verified through numerical simulation on MATLAB/Simulink platform.

According to differential geometry theory[21], relative degree of the affine nonlinear dynamic system is equal to number of state variables. Consequently, a fully controllable linear dynamic system topologically equivalent to Eq.(26) can be established and presented as follows

$$\dot{z} = Az + Bv \tag{27}$$

where $z = [z_1, z_2, z_3, z_4, z_5]^T$, $v = [v_1, v_2, v_3]^T$,

$$A = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}, B = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 1 \end{bmatrix}.$$

The differential homeomorphic mapping can be written in the following form as

$$z_1 = h_1(x) = x_1,\quad z_2 = h_2(x) = x_3,\quad z_3 = L_f h_2(x) = x_4,$$
$$z_4 = h_3(x) = x_5,\ z_5 = L_f h_3(x) = x_6,$$
$$\dot{z}_6 = \left[i_{go}\mu_{co} \cdot F_{p3o}(z_2) + i_{ge}\mu_{ce} \cdot F_{p3e}(z_4) - M_r \right] / J_{eq}$$

In order to derive accurate control inputs of affine nonlinear dynamic system in Eq.(26), following nonlinear feedback control law is established. It should be noted that such law is also indispensable to build topologically equivalent linear dynamic system presented in Eq.(27).

$$u = A^{-1}(x)\left[v - b(x) \right] \tag{28}$$

where

$u = [u_1, u_2, u_3]^T$, $b(x) = \left[L_f^1 h_1(x), L_f^2 h_2(x), L_f^2 h_3(x) \right]^T$,

$A(x) = diag\left[L_{g_1} L_f^0 h_1(x), L_{g_2} L_f^1 h_2(x), L_{g_3} L_f^1 h_3(x) \right]$.

After constructing diffeomorphism relation between such two dynamic systems, the remaining task is developing sliding mode control algorithm for the linear dynamic system. The sliding mode surfaces are presented as following

$$s_1 = e_1,\quad s_2 = c_{21}e_2 + e_3,\quad s_3 = c_{31}e_4 + e_5 \tag{29}$$

in which $c_{21} = c_{31} = 10^3$.

And then corresponding exponential reaching laws can be derived as following,

$$\dot{s}_1 = -\varepsilon_1 \operatorname{sgn} s_1 - k_1 s_1,\ \dot{s}_2 = -\varepsilon_2 \operatorname{sgn} s_2 - k_2 s_2,$$
$$\dot{s}_3 = -\varepsilon_3 \operatorname{sgn} s_3 - k_3 s_3 \tag{30}$$

in which

$$\varepsilon_1 = 10^{-2}, k_1 = 10^3; \varepsilon_2 = 10^{-2}, k_2 = 2.5 \times 10^4;$$
$$\varepsilon_3 = 10^{-2}, k_3 = 10^4$$

Consequently, sliding control law for the topologically equivalent linear system are derived and presented as following

$$v_1 = \dot{z}_{1R} - \varepsilon_1 \operatorname{sgn} s_1 - k_1 s_1 \tag{31}$$

$$v_2 = \dot{z}_{3R} - c_{21} e_3 - \varepsilon_2 \operatorname{sgn} s_2 - k_2 s_2 \tag{32}$$

$$v_3 = \dot{z}_{5R} - c_{31} e_5 - \varepsilon_3 \operatorname{sgn} s_3 - k_3 s_3 \tag{33}$$

Finally, control inputs of the affine nonlinear dynamic system should be explicitly calculated. The calculation procedure is presented in Eq.(28).

VI. TRACKING ACCURACY OF THE FEEDBACK LINEARIZATION BASED SLIDING MODE CONTROL ALGORITHM

In order to verify whether sliding mode control for the feedback linearization system is sufficient to guarantee precise tracking for optimal trajectories, a numerical simulation is implemented. The simulated shift condition is a typical one concerning a 1st gear to 2nd gear shift on a 6^o ramp road. The nominal values of friction coefficients are set to be 0.3. Other structure parameters of the actuator dynamic systems are given in Table 2.

TABLE 2 PARAMETERS OF THE ACTUATOR

Parameter	Value and Unit	
	Value	***Unit***
p_m	5	pairs
Φ_m	776	Wb
N	30	turns
μco,μce	0.3, 0.3	
Rco,Rce	0.097,0.091	m
z	2	
L	0.1	m
αi	0.17	rad
Jm	0.005	kg*m^2
p	0.002	m
ηs	0.95	
μrb,μrs	0.008,0.01	

Different simulation results are compared in aspect of angular velocity tracing accuracy before friction plate wear and that after friction plate wear. The friction plate wear is set to be 1mm. Through careful observation of simulation results, as shown in Fig.6, it can be found that optimal angular velocities of electric motor and dual clutch as well as displacements of little end of diaphragm springs are precisely traced according to precise regulation of motor currents and electric motor torque, whether friction plate wear occurs or not, as shown in Fig.6(a) and Fig.6(b). This means that to overcome influence of friction plate wear to tracing accuracy, electric motor torques and motor control currents of actuators should be timely adjusted by the proposed sliding mode control algorithm, as shown in Fig.6(c) and Fig.6(d). Consequently, it can be concluded that the proposed sliding mode control algorithm possesses strong robustness to above mentioned modelling uncertainties and disturbances.

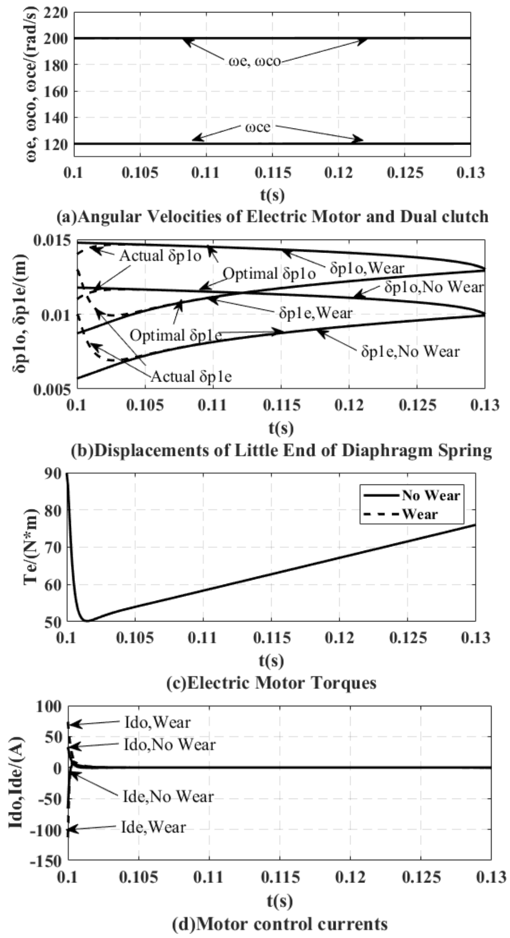


Fig. 6 Sliding mode control for the whole DDCT dynamic system

VII. CONCLUDING REMARKS

For the purpose of seeking comprehensively optimal power transmission strategy of two-speed DDCT EV during torque phase in shift, dynamic model of DDCT of EV is built at first. Then, optimal slip-to-slip control strategy, aiming at non-shock shift process of torque phase without power interruption and power circulation is derived. And then, nonlinear dynamic model of fork-lever actuator is integrated into that of DDCT driveline, and an affine nonlinear shift dynamic model for the whole DDCT system of EV is proposed to investigate shift dynamic behaviors, tracking control accuracy for angular velocities of electric motor and dual clutch in torque phases of shift. Further, to enhance robustness of tracking control algorithm to such nonlinear dynamic system, sliding mode control strategy based on feedback linearization control theory is proposed, in which sliding mode control algorithm is directly imposed to topologically equivalent linear dynamic system. According to nonlinear feedback control law which also indispensable to build topologically equivalent linear dynamic system, accurate motor currents and electric motor torque imposed to affine nonlinear system are calculated. Furthermore, tracking control accuracy of affine nonlinear dynamic system is investigated through numerical simulations on MATLAB/Simulink platform. Finally, it can be concluded that only non-shock shift process of torque phase without power interruption and power circulation is realized, but also torque phase time and friction work value can be adjusted to an arbitrary value based on actual requirement. Besides, the proposed sliding mode control strategy based on feedback linearization control theory acquires very strong robustness to modelling uncertainties and disturbances.

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