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Shift control of Dual Clutch Transmission using Triple-Step nonlinear method

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Abstract: The shift control, especially clutch slip control in inertia phase, is considered as a key issue of dual clutch transmission for improving driving comfort. For the shift is a complex nonlinear dynamic process combining engine states and road information, and these are often variable, so the clutch control system can be described as a nonlinear parameter-varying model. Moreover, a novel nonlinear approach is proposed for clutch control of shift process, the control objective is to obtain smooth gearshift in the limited shift time by making clutch slipping speed difference tracking the planned reference. The proposed method can be formalized as a triple-step procedure and this procedure provides a concise design process so that the derivation of the control law is simplified and straightforward. Finally, the simulation results in the environment of AMESim with a complete vehicle model show the efficiency of the proposed approach.

Keywords: Shift Control; Steady State Control; Reference Dynamics Feedforward Control; Gain Scheduling PID; Parameter Varying; Dual Clutch Transmission.

1. INTRODUCTION

As a type of transmission, in fact, Dual Clutch Transmissions (DCTs) have been known for a long time, being used already in race cars in the 1980s. However, it is only the recent progress in electronics and actuation that makes them feasible for volume application Amendola and Alves [2006]. DCTs allow to shift gears quickly, comfortably and without interruption in traction, and provide an alternative automatic transmission with the power-on shifting and shift quality comparable to Automatic Transmissions (ATs) and with efficiencies comparable to Automated Manual Transmissions (AMTs) Matthes and Guenter [2005]. However, since there is no hydrodynamic torque converters in DCTs, engine vibration can not be isolated from the transmission and drive train very effectively, then good shift quality is achieved with the application of more precise clutch control technologies Wan et al. [2003].

In both DCTs and ATs, the change of speed ratio can be regarded as a process of one clutch to be engaged while another being disengaged, namely, clutch-to-clutch shifts Cho [1987]. The shift process can be usually divided into two phases: a torque phase where engine torque is transferred from the offgoing clutch to the oncoming clutch and an inertia phase where the speed of the engine is synchronized to that of the target gear Kulkarni et al. [2007], Goetz et al. [2005]. During the inertia phase, not only the torque but also rotational speeds change intensively as well, so shift quality is very closely related to this duration. For achieving good shift quality, specifically, the clutch engagement in this phase must be controlled to satisfy contradictory objectives such as minimizing the slipping energy and preserving the driving comfort Tran et al. [2013].

Currently, many different approaches have been proposed for the control of clutch-to-clutch shifts. In Kulkarni et al. [2007], based on event-driven (rule-based) control is used to control clutch engagement. In Goetz et al. [2005], Ni et al. [2009], PID controller is designed for shift control. It is known that the event-driven method and PID control in engineering applications often require lots of calibrations for obtaining the satisfactory control performance. The disadvantages of these methods are that the calibration is time-consuming and the control performance will depend heavily on the calibrations. In addition, paper Yuan and Chen [2013] uses a model reference control integrated into the traditional PID control to gain a smooth transition with little powertrain interruption and frictional energy loss. In paper Tran et al. [2013], H_∞ control based on uncertain TS models is selected to obtain good driving comfort. These feedback control methods can improve control performance, but they need large computational cost and not easily be widely used in engineering. In Gao et al. [2011], Tian et al. [2012], backstepping control is proposed to design clutch slip controller for AT and DCT. Although the control performance is satisfied but they did not consider the parameter varying of system in modeling and controller design. When the parameters change greatly or drift, the control performance will deteriorate. In this paper, as an extension of Gao et al. [2011], Tian et al. [2012], the system parametric variations (such as the friction coefficient, pressure control valve gain, engine

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states) are considered directly in controller design by using a new model-based nonlinear method, named by Triple-Step, which has been proved its special merits in rail pressure control of GDI engines Chen et al. [2014]. This paper will extend the method to parameter-varying system, and proposes a model based and systematic procedure to design a nonlinear speed tracking controller. The procedure consists of three steps in succession and delivers the final control in an additive process. Another worthy of attention is that the deduced control law is not only state-dependent but also parameter-dependent.

The organization of the paper is as follows. In Section 2, based on torque balance equation, a second order model with parameter-varying of powertrain dynamic system is derived and the control problem is stated. A detail step by step procedure of nonlinear controller is given for clutch slip control in Section 3. The proposed approach is tested on the integrated vehicle powertrain model, and simulation results are shown in Section 4. Finally, conclusions from this work, as well as recommendation for future work, are also outlined.

2. POWERTRAIN MODELING AND PROBLEM STATEMENT

Dual Clutch Transmission is typically used in passenger vehicles, it transmits torque from engine to wheel, and is basically the combination of two sub-gearboxes which is divided into odd and even gears. The physical structure of a simplified powertrain system equipped with a two-speed DCT is shown in Fig. 1.

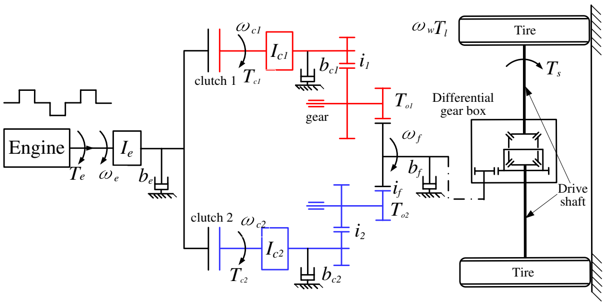

Fig. 1. Structure diagram of a simplified powertrain system

In DCT shift process, two clutches are used as the actuators. If the vehicle is running at 1st gear, the 2nd gear can be pre-selected by electronic control unit. During torque phase, the clutch1 (off-going clutch) still remains engaged, as the pressure executing on clutch2 (on-coming clutch) is increased slowly, the engine torque is transferred from the clutch1 to the clutch2 while the clutch1 is disengaged. During inertia phase, the clutch2 slips, and the rotational speed of the engine change to meet the synchronize speed of the clutch, which affects shift quality greatly.

During modeling, some assumptions are given as follows:

- Gears have no backlash;
- All gears are modeled as non-compliant elements and are represented as mass inertias;
- The drive shaft is considered as rigid;
- Temperature effects of the system are neglected.

And the symbols and their physical meanings in modeling are listed in Table. 1.

Table 1. Nomenclature

Symbol	Description
$\omega_{e,c,f,w}$	rotation speed
$I_{e,c,f,w}$	equivalent inertia moment
$T_{e,c,o,s}$	output torque
T_l	load torque
$b_{e,c,f,w}$	damping coefficient
θ	throttle opening degree
i_2	2nd gear ratio
i_f	gear ratio of final drive
p_{c2}	hydraulic pressure on clutch plate
R	effective radius of clutch plates
N, A	plate number and piston area of the clutch
F_s	return spring force
i_v	driven current pressure control valve
K_v, τ_v	gain and time constant of the valve
r_w	tire radius
C_r	rolling resistance coefficient
C_D	aerodynamic drag coefficient
A_v	front area of vehicle
ρ, g	air density and acceleration of gravity
m, α	vehicle mass and road grade
δ	rotating mass conversion factor

*subscript e, c, f, w represent the engine, clutch, final drive and wheel; o, s represent the transmission, drive shaft.

In the inertia phase, the torque transmitted by clutch1 is very small, which can be ignored. Based on Newton's second law, the system dynamics are described as follows

$$\text{engine}: \quad I_e\dot{\omega}_e = T_e - b_e\omega_e - T_{c2}\,, \tag{1a}$$

$$\text{clutch}: \quad I_{c2}\dot{\omega}_{c2} = T_{c2} - b_{c2}\omega_{c2} - \frac{T_o}{i_2}\,, \tag{1b}$$

$$\text{final drive}: \quad I_f\dot{\omega}_f = T_o i_f - b_f\omega_f - T_s\,, \tag{1c}$$

$$\text{wheel}: \quad I_w\dot{\omega}_w = T_s - b_w\omega_w - T_l\,. \tag{1d}$$

Moreover, we have

$$\omega_f = \frac{\omega_{c2}}{i_2 i_f}, \quad T_{o2} = \frac{T_s}{i_f}\,. \tag{2}$$

In (1a), the engine torque T_e is determined by the engine speed and throttle opening degree, i.e., $T_e = T_e(\omega_e, \theta)$, which is given in general as map (lookup tables). When clutch slips, the transferred torque on clutch is mainly determined by the pressure exerting on the clutch. Based on the clutch geometry and friction characteristics, the output torque of the clutch2 is calculated by

$$T_{c2} = \mu(\Delta\omega)RN(A \cdot p_{c2} - F_s)\,. \tag{3}$$

The friction coefficient μ can be formulated as a function of the clutch slip speed $\Delta\omega$, which is defined as $\Delta\omega = \omega_e - \omega_{c2}$.In wet clutches, the clutch pressure is provided by proportional pressure valves, as hydraulic actuator, which can be simplified as a first-order system,

$$\tau_v(p_{c2})\dot{p}_{c2} = -p_{c2} + K_v(p_{c2})i_v. \tag{4}$$

Due to hysteresis, dead zone and saturation nonlinearity, the solenoid valve is not a fixed proportional relationship between pressure p_{c2} and current i_v, namely, response time τ_v and gain K_v are variable with the clutch pressure p_{c2}.

The road load torque T_l in (1d) is caused by road resistance F_r, acceleration resistance F_j, and aerodynamic resistance F_w, i.e.,

$$T_l = (F_r + F_j + F_w) \cdot r_w \tag{5}$$

with

$$F_r = mg(C_r + \sin\alpha)\,, \tag{6a}$$

$$F_j = \delta m \frac{dv}{dt}\,, \tag{6b}$$

$$F_w = \frac{1}{2} C_D A_v \rho v^2\,. \tag{6c}$$

Here, without considering wheel slip, there is $v = r_w w_w$, r_w is wheel radius. In flat road, $\alpha = 0$.

If the drive shaft is considered as rigid, we have $\omega_f = \omega_w$. By combining (1) $\sim$ (6), and defining $\bar{p}_{c2} = p_{c2}/1000$ to ensure system states $\Delta\omega$ and p_{c2} at the same order of magnitude, the dynamics of the clutch system with actuators can be rearranged as

$$\Delta\dot{\omega} = a_1(\Delta\omega, w) + b_1(\Delta\omega)\bar{p}_{c2}\,, \tag{7a}$$

$$\dot{\bar{p}}_{c2} = a_2(\bar{p}_{c2}) + b_2(\bar{p}_{c2})u \tag{7b}$$

with

$$\begin{aligned} a_1(\Delta\omega, w) = {} & \frac{T_e(\omega_e, \theta) + \mu(\Delta\omega)RNF_s - b_e\omega_e}{I_e} \\ & + \frac{(b_{c2}i_2^2 i_f^2 + b_f + b_w)(\omega_e - \Delta\omega) + i_2^2 i_f^2 \mu(\Delta\omega)RNF_s}{I_{c2}i_2^2 i_f^2 + I_f + I_w + \delta m r_w^2} \\ & + \frac{mgr_w(C_r + \sin\alpha) + \frac{1}{2} C_D A_v \rho r_w v^2}{[I_{c2}i_2^2 i_f^2 + I_f + I_w + \delta m r_w^2]}\,, \end{aligned} \tag{8a}$$

$$b_1(\Delta\omega) = -[\frac{1000}{I_e} + \frac{1000 \times i_2^2 i_f^2}{I_{c2}i_2^2 i_f^2 + I_f + I_w + \delta m r_w^2}]\mu(\Delta\omega)RNA, \tag{8b}$$

$$a_2(\bar{p}_{c2}) = -\frac{1}{\tau_v(\bar{p}_{c2})}\bar{p}_{c2}, \quad b_2(\bar{p}_{c2}) = \frac{K_v(\bar{p}_{c2})}{\tau_v(\bar{p}_{c2}) \times 1000}\,. \tag{8c}$$

In the above, w lumps the parameters ω_e, ω_{c2}, v and θ, α, m, which vary with operating conditions.

As mentioned earlier, the good shift quality means that minimizing the clutch engagement time, minimizing the friction losses and shift shock, which implies that the different and sometimes conflicting control objectives are expected to satisfy in clutch engagement. In general, if clutch slip speed changes smoothly and the shift duration is limited in a suitable short time, there will not be too much dissipated energy. In this study, with the same reason in Gao et al. [2011], the shift control is considered as a speed tracking control. The control objective is to make the clutch slip speed tracking a given speed reference considering shift requirements comprehensively, which can be given by a continuous third-order polynomial satisfying the so-called no-lurch condition as paper Gao et al. [2011], Chang and Park [2005].

Moreover, from the system model (7), it can be seen that clutch system dynamic is coupled with the engine states T_e and ω_e, and some model parameters are state-dependent and mutative, which means the controller selected must be able to deal with parameter-varying.

3. NONLINEAR CONTROLLER DESIGN

This paper extends the triple-step procedure to design a nonlinear controller for parameter-varying system. First of all, define the tracking output and the reference as $y = \Delta\omega$ and $y^* = \Delta\omega^*$, respectively. To get the direct forms of expression between input $u = i_v$ and output y, we do not need to get state space form but differentiate $\dot{y}$ directly, so the model (7) can be transformed as the following form

$$\begin{aligned} \ddot{y} = {} & A_y(\Delta\omega, \bar{p}_{c2}, w)\dot{y} + A_w(\Delta\omega, w)\dot{w} \\ & + b_1(\Delta\omega)a_2(\bar{p}_{c2}) + b_1(\Delta\omega)b_2(\bar{p}_{c2})u, \end{aligned} \tag{9}$$

where $A_y(\Delta\omega, \bar{p}_{c2}, w) = \frac{\partial a_1(\Delta\omega, w)}{\partial \Delta\omega} + \frac{\partial b_1(\Delta\omega)}{\partial \Delta\omega}\bar{p}_{c2}$, $A_w(\Delta\omega, w) = \frac{\partial a_1(\Delta\omega, w)}{\partial w}$ and assuming $\dot{w} = 0$ for simplifying the design in the following.

The control scheme of triple-step method takes three parts into account: the steady state control, the feedforward control related to the reference dynamics and the feedback control related to the tracking errors dynamics, as shown in Fig. 2. Based on eq. (9), the detailed design procedure can be demonstrated in the following.

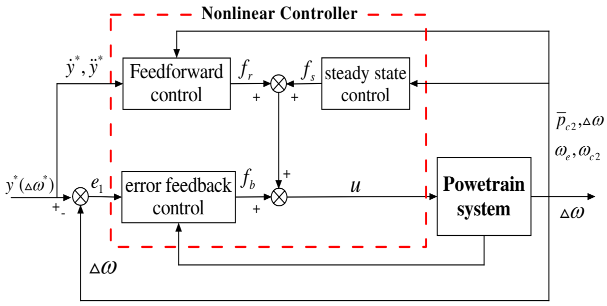

Fig. 2. The schematic diagram of shift control scheme

- Step 1: the steady state-like control

When the system reaches a steady state, there are $\dot{y} = 0$ and $\ddot{y} = 0$, we put it into the system (9), a steady state control is obtained as follows

$$f_s = -\frac{a_2(\bar{p}_{c2})}{b_2(\bar{p}_{c2})}\,. \tag{10}$$

It is called " steady-state-like" because this part of the control is obtained by setting the system in steady state and implemented according to the current measured or estimated value, but not the true steady state. And due to parameter disturbances in system, only a steady state control is not enough. In order to obtain a qualified closed-loop control performance, some more additional information of the control system should be considered.

- Step 2: the feed-forward control based on reference dynamics

In fact, the clutch speed tracking reference y^* is not constant and its dynamics is used to reflect the control requirements, so the tracking reference dynamics needs to be considered. Defining the control law being deduced as f_r, combining f_s, substituting

$$u = f_s + f_r$$

into (9) leading to

$$\ddot{y} = A_y(\Delta\omega, \bar{p}_{c2}, w)\dot{y} + B_u(\Delta\omega, \bar{p}_{c2})f_r\,, \tag{11}$$

where $B_u(\Delta\omega, \bar{p}_{c2}) = b_1(\Delta\omega)b_2(\bar{p}_{c2})$, due to $\mu(\Delta\omega) > 0$ for $\forall \Delta\omega \geq 0$, then $b_1(\Delta\omega) \neq 0$, and $b_2(\bar{p}_{c2}) \neq 0$, hence $B_u(\Delta\omega, \bar{p}_{c2}) \neq 0$. In order to facilitate the derivation, simplify denoted by

$$A_y(\Delta\omega, \bar{p}_{c2}, w) := A_y,$$
$$B_u(\Delta\omega, \bar{p}_{c2}) := B_u.$$

By enforcing $\ddot{y} = \ddot{y}^*$ and $\dot{y} = \dot{y}^*$ for (11), we can obtain a reference feed-forward control as follows

$$f_r = \frac{1}{B_u}\ddot{y}^* - \frac{A_y}{B_u}\dot{y}^* . \tag{12}$$

• Step 3: the tracking error feedback control

Because there are modeling errors, parameter uncertainties and external disturbances, the feedback control is needed to further improve the control performance, define it as f_b, with f_r and f_b to be determined, we substitute

$$u = f_s + f_r + f_b$$

into (9) to infer

$$\ddot{y} = A_y\dot{y} + \ddot{y}^* - A_y\dot{y}^* + B_u f_b . \tag{13}$$

By defining the tracking error as

$$e_1 = y - y^* , \tag{14}$$

then, (13) becomes

$$\ddot{e}_1 = A_y\dot{e}_1 + B_u f_b . \tag{15}$$

Defining $e_2 = \dot{e}_1$, the above equation can be rewritten as

$$\dot{e}_1 = e_2 , \tag{16a}$$
$$\dot{e}_2 = A_y e_2 + B_u f_b . \tag{16b}$$

Obviously this is a linear parameter-varying error system, the backstepping technique is applied to determine f_b so that the tracking error dynamics is asymptotically stable. Moreover, in order to reduce the tracking offset, we introduce an integral action into the control law. According to the structure of error system, we can take e_2 as a virtual control. Then, define $V_1 = \frac{1}{2}e_1^2 + \frac{1}{2}k_0\chi^2$, χ is integral term, i.e. $\chi = \int e_1 dt$ and $k_0 > 0$, we can infer

$$\dot{V}_1 = e_1\dot{e}_1 + k_0\chi e_1 = e_1(e_2 + k_0\chi) . \tag{17}$$

Choose the virtual control as

$$e_2^* = -k_1 e_1 - k_0\chi \tag{18}$$

with $k_1 > 0$ to achieve for $e_2 = e_2^*$ such that

$$\dot{V}_1 = -k_1 e_1^2 \leq 0 , \tag{19a}$$
$$\dot{e}_1 = -k_1 e_1 - k_0\chi . \tag{19b}$$

Indeed, $e_2 \neq e_2^*$ in the process of the dynamic control, then, define $\Delta e_2 = e_2 - e_2^*$, the above equations become

$$\dot{V}_1 = -k_1 e_1^2 + e_1\Delta e_2 , \tag{20a}$$
$$\dot{e}_1 = -k_1 e_1 - k_0\chi + \Delta e_2 . \tag{20b}$$

Furthermore, we apply (16b), (18) and $e_2 = \dot{e}_1$ to arrive at

$$\begin{aligned}\Delta\dot{e}_2 &= A_y e_2 + B_u f_b + k_1\dot{e}_1 + k_0 e_1 \\ &= (k_1 + A_y)\,\dot{e}_1 + k_0 e_1 + B_u f_b . \end{aligned} \tag{21}$$

Select the Lyapunov function $V_2 = V_1 + \frac{1}{2}\Delta e_2^2$ for (16b) and infer

$$\dot{V}_2 = -k_1 e_1^2 + \Delta e_2\left[e_1 + (k_1 + A_y)\,\dot{e}_1 + k_0 e_1 + B_u f_b\right] , \tag{22}$$

where (20a) and (21) are used. It should be noted that $B_u \neq 0$. Choose the control as

$$f_b = -\frac{1 + k_0}{B_u}e_1 - \frac{k_1 + A_y}{B_u}\dot{e}_1 - \frac{k_2}{B_u}\Delta e_2 \tag{23}$$

with $k_2 > 0$ to achieve

$$\dot{V}_2 = -k_1 e_1^2 - k_2\Delta e_2^2 , \tag{24}$$

which implies that $\dot{V}_2$ is negative semi-definite. Let us now characterize the set $S = \{e \in \mathbb{R}^3 | \dot{V}_2 = 0\}$ with $e = [\chi\ e_1\ \Delta e_2]^T$. First of all, $\dot{V}_2 = 0 \Rightarrow e_1 = 0$ and $\Delta e_2 = 0$. It then follows from $\chi = \int e_1 dt$ that $e_1(\cdot) \equiv 0 \Rightarrow \chi(\cdot) \equiv 0$. Hence, $S = \{0\}$, which implies that the system can maintain the $V_2 = 0$ condition only at the origin $e = 0$. In other words, the closed-loop error system is asymptotically stable Khalil [2007]. We apply (18) and $e_2 = \dot{e}_1$ to infer

$$\begin{aligned} f_b &= -\frac{1 + k_0}{B_u}e_1 - \frac{k_1 + A_y}{B_u}\dot{e}_1 - \frac{k_2}{B_u}\Delta e_2 \\ &= -\frac{1 + k_0 + k_1 k_2}{B_u}e_1 - \frac{k_1 + k_2 + A_y}{B_u}\dot{e}_1 - \frac{k_0 k_2}{B_u}\chi . \end{aligned} \tag{25}$$

It should be emphasized that the system (9) is our starting point to derive the control law, and the Backstepping technique is also introduced based on the error system (16). And from the design procedure, it can be see the three steps can not be swapped and the design method for error system (16) is not limited to Backstepping. As a result, the form of the control law can be more brief than ever. By combining (10), (12) and (25), the whole control law is given as follows

$$\begin{aligned} u = &-\frac{a_2(\bar{p}_{c2})}{b_2(\bar{p}_{c2})} + \frac{1}{B_u}\ddot{y}^* - \frac{A_y}{B_u}\dot{y}^* - \frac{1 + k_0 + k_1 k_2}{B_u}e_1 \\ &- \frac{k_0 k_2}{B_u}\chi - \frac{k_1 + k_2 + A_y}{B_u}\dot{e}_1 . \end{aligned} \tag{26}$$

We rewrite the derived control law as

$$u = f_s + f_r + f_P e_1 + f_I\chi + f_D\dot{e}_1 , \tag{27}$$

and

$$f_s = -\frac{a_2(\bar{p}_{c2})}{b_2(\bar{p}_{c2})} , \tag{28a}$$
$$f_r = \frac{1}{B_u(\Delta\omega, \bar{p}_{c2})}\ddot{y}^* - \frac{A_y(\Delta\omega, \bar{p}_{c2}, w)}{B_u(\Delta\omega, \bar{p}_{c2})}\dot{y}^* , \tag{28b}$$
$$f_P = -\frac{1 + k_0 + k_1 k_2}{B_u(\Delta\omega, \bar{p}_{c2})} , \tag{28c}$$
$$f_I = -\frac{k_0 k_2}{B_u(\Delta\omega, \bar{p}_{c2})} \tag{28d}$$
$$f_D = -\frac{k_1 + k_2 + A_y(\Delta\omega, \bar{p}_{c2}, w)}{B_u(\Delta\omega, \bar{p}_{c2})} . \tag{28e}$$

From the above equation, it can be seen that parameter w is introduced by the term A_y into control law, that means the approach will be robust to parameter perturbations. In addition, steady state control f_s and feedforward control f_r are dependent on system states $\Delta\omega, \bar{p}_{c2}$. And error feedback, derived in the framework of Lyapunov's theory, can be finally rearranged into the form of a PID controller and depends on state and parameter w. This is consistent with a gain scheduling PID.

4. SIMULATION RESULTS

In order to test the control performance preliminary, a detailed powertrain dynamic model containing a Dual Clutch Transmission is built in AMESim commercial software and mainly includes the engine and unit block, the clutches and hydraulic actuators, the gearbox, the vehicle.

The brief descriptions about special components are given as follows:

- As a power source, a simplified engine model is used, which can compute the engine torque and consider the engine friction loss, the emissions and the fuel consumption by lookup table.

- Because there is no hydraulic torque converter to absorb vibration, dual mass fly-wheel is always adopted in DCT so that driveline vibration and stress can be restrained. To model it, the engine unit block model is selected, it can provides an internal fly wheel including interactions with the engine block (coulomb and viscous friction are considered), and can also be used for driving comfort modeling and coupling analysis between engine mass, vehicle mass and engine mounts;
- The DCT automatic gearbox is basically composed of two partial transmissions, built as if they were independent gearboxes, from a functional standpoint. Each transmission or partial circuit has an assigned clutch in the interior of the dual clutch, an input (primary) shaft, and they have a combination of the two partial transmissions gears on the two output shafts;
- As hydraulic actuator of clutch, proportional pressure valves are described by current-pressure characteristic map with hysteresis, dead zone and saturation nonlinearity, which is calibrated by experimental.
- A vehicle model is built, which includes 3 DOF carbody dynamics, suspension and simplified Pacejka tyre and can be used to analyze interactions between carbody and engine block, carbody and wheel, tyre and road.

The nominal model parameters can be listed in the following Table. 2. The controller implementation and simulation results are discussed in the following.

Table 2. Nominal system parameters

Symbol	Value	Symbol	Value
N	4	A	0.0066 m
R	0.1229 m	i_1	3.4615
i_2	2.05	i_f	4.0588
r_w	0.301 m	C_r	0.014
δ	1.2	ρ	1.226 kg/m^3
A_v	1 m^2	C_D	0.5
F_s	496 N	g	9.8 m/s^2
I_e	0.15 kg m^2	I_{c2}	0.02 kg m^2
I_w	2 kg m^2	I_f	0.3 kg m^2

4.1 Control law implementation

In the final control law (28), $A_y = \frac{\partial a_1(\Delta\omega,w)}{\partial \Delta\omega} + \frac{\partial b_1(\Delta\omega)}{\partial \Delta\omega}\bar{p}_{c2}$. From (8), it can be seen that parameter $\mu(\Delta\omega)$ is included in $a_1(\Delta\omega, w)$, $b_1(\Delta\omega)$ and it is given as a map, that means it is impossible to obtain the explicit form of A_y by differentiating $\mu(\Delta\omega)$. Hence, we obtain $\mu(\Delta\omega)$ by using the input shaping technique. Moreover, we also use it to attenuate measurement noise from the measurement signal of $\Delta\omega$, $\bar{p}_{c2}$ and w in engineering. In general, a first order low-pass filter is used.

4.2 Simulation results

Taking the power-on $1 \rightarrow 2$ gear shift as an example, the controller gains are selected as $k_0 = 1000$, $k_1 = 70$, $k_2 = 45$. As mentioned in Section 2, the lumped parameter w is varying with the operation condition, in which, compared to the parameters w_e, w_{c2}, θ and v, the mass m and road slope α change relatively slowly in the short shift time, and these signals can not be measured directly for the corresponding sensors are rarely installed in practice, hence, considering the cost of controller implementation, they are given as constant.

The gear shift time is set from 1.64s to 2.26s, in which, the torque phase is from 1.64s to 1.86s, the inertia phase is completed within 0.4s. We assume that the clutch pressure control during the torque phase has been well achieved. Assuming $m = 1200kg$, $\alpha = 0$, during shift process, the throttle angle θ is often regulated to match shift control, as shown in Fig. 3, and it might be difference for $1 \rightarrow 2$ gear in different vehicle state. The simulation results in nominal conditions (Table 2) are shown as Fig. 4 and Fig. 5, From which, it can be seen that the w_e and w_{c2} are constantly changing. And the transmission output torque change smoothly and shift jerk is within the acceptable levels during upshift despite parameter-varying. Moreover form the tracking curve (Fig. 5), it can also be seen that the control performance is satisfactory.

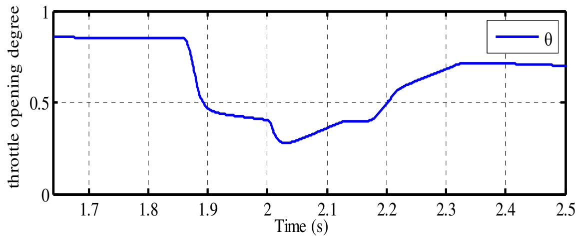

Fig. 3. Throttle opening degree in shift process

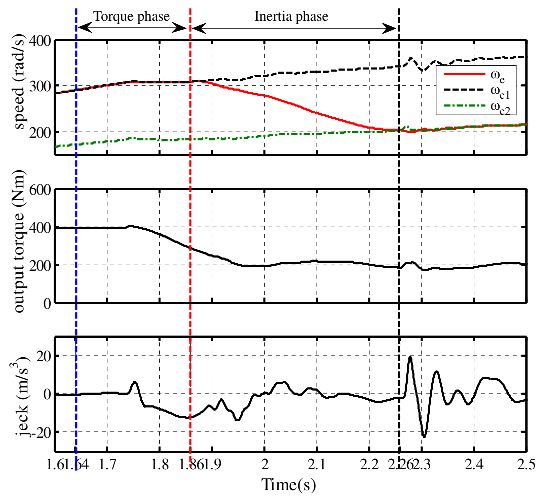

Fig. 4. The shift states under car mass $m = 1200kg$ and flat road $\alpha = 0°$

As mentioned above, the change of mass and road slope are ignored, which can be considered as parameter uncertainty, and for which, the robustness of the controller is validated. The corresponding simulation results in Fig. 6 and Fig. 7 are given and shown that change of the output torque in slope road is larger than that of flat road, but there is no large shift shock and the clutch slip speed is still able to track the reference very well within 0.4s.

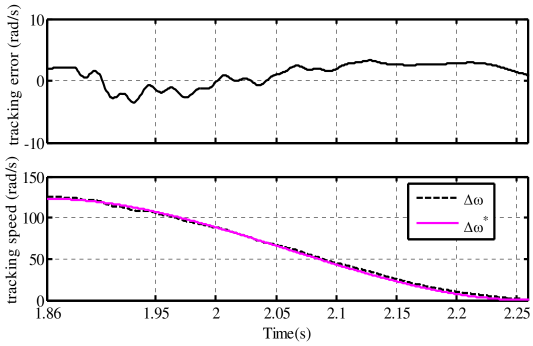

Fig. 5. The speed tracking results under car mass $m = 1200kg$ and flat road $\alpha = 0°$

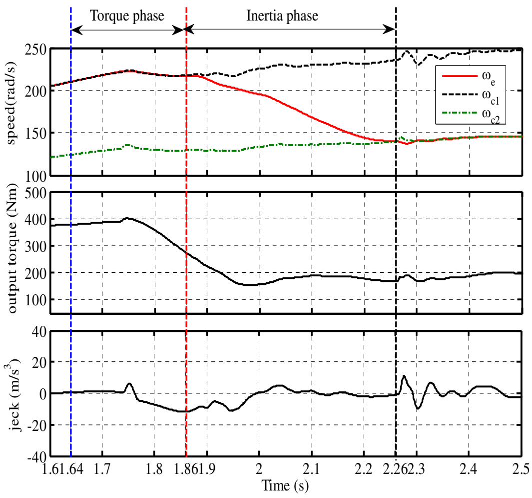

Fig. 6. The shift states under car mass $m = 1500kg$ and slope road $\alpha = 2.7°$

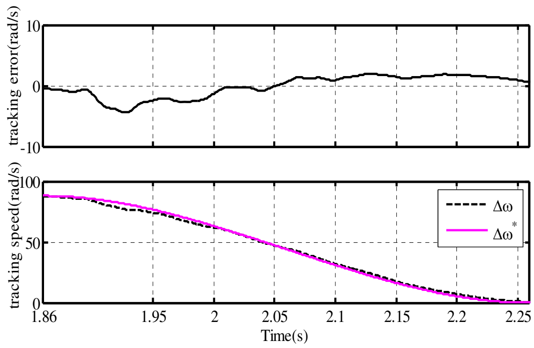

Fig. 7. The speed tracking results under car mass $m = 1500kg$ and slope road $\alpha = 2.7°$

5. CONCLUSION

In shift process, clutch slip control is important issue for improving driving comfort, in this study, good quality is obtained by controlling the clutch slip speed to track a given speed reference. Due to time-varying on clutch friction characteristics, pressure-current characteristics of solenoid valve and engine states, the clutch control system is a nonlinear parameter-dependent model, so a new nonlinear control method is extended to deal directly with model parameter-varying, which can be designed by triple steps: the steady state control, the reference dynamics feedforward control and PID feedback control. And the controller gains are state-and parameter-dependent, which means that the controller is able to achieve self-regulating for difference working conditions. The simulation results also proved this point.

Simulation verification under downshift process or more complex working conditions will be added. Moreover, the control performance should be tested in HiL test bench or real vehicle for further engineering implementation.

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