Gearshift Control of a Dual Clutch Transmission Based on Sliding Mode Control for Nonlinear Uncertain Systems

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Abstract

The dual clutch transmission combines the advantages of the automatic transmission and performances, fuel efficiency of the manual transmission. In this system, the control of the dual clutch is a key point, especially when considering driving comfort. In this paper, we propose a control law for gearshift of a vehicle equipped with the dual clutch transmission based on sliding mode control for nonlinear uncertain systems. The goal is to ensure a rapid and smooth clutch engagement and avoiding engine no-stall. To achieve this goal, the clutch slipping speed and engine speed are controlled. In addition, uncertain parameters of the model and input disturbances are also considered. Several simulations are provided to show the effectiveness of proposed control law.

Keywords: Dual clutch transmission, Clutch control, Non-linear models, Uncertain model, Trajectory tracking, Sliding mode control.

1. Introduction

The dual clutch transmission (DCT) was introduced in the vehicle to improve driving comfort compared with manual transmissions, and performances, fuel efficiency compared with automatic transmissions. The DCT consists of two independent sub-gearboxes, one for the even gear sets and the other one for odd gear sets, each one activated by separate clutches: on-coming clutch and off-going clutch. A shift process involves the engagement of the on-coming clutch and the release of the off-going clutch to ensure a shift without traction interruption. DCT usually operates in a fully automatic mode, and have also the ability in a semi-automatic mode, to allow the driver to gear selection in manual mode by the buttons. There are two fundamental types of clutches utilized in DCT: either two wet multi-disc clutches which are bathed in oil for cooling (WDCT) or two dry single-disc clutches (DDCT).

The problems associated with DCT in literature are the management of the dual clutch in the launch phase and the gearshift phase, driving at low speed, and the strategy of the gearshift. The goal is to ensure a smooth engagement in standing start, gearshift, and thus ensure a good drivability. The strategy of gearshift allows also reduce fuel consumption and emission of CO2. Specifically, the dry clutch engagement must be controlled in order to satisfy contradictory objectives such as minimizing the slipping energy and preserving of driving comfort.

In literature on the clutch management for AMT, many different approaches have been proposed such as optimal control: [1]–[5], flatness control: [6], Model Predictive Control (MPC):[4], [7], sliding mode control: [8], [9], PID control: [10].

In [1], the authors proposed a control law for clutch engagement based on optimal control to improve comfort during the launch. The initial part of the control process, a normal look-up table open-loop control is applied until the clutch slipping speed is reduced to a trigger value. Then, an observer-based optimal control is activated to assure an engagement comfort. The conditions no-lurch [11] is taken into account. The authors used Coulomb friction for clutch model and neglected load torque. The engine torque is considered as a constant additional state over the optimal control activation interval.

The same strategy, in [8], the authors present a control architecture that consists of the speed control for synchronization and the torsion control for reducing jerk incorporating an output torque observer. Control law is developed based on sliding mode control theory. The clutch engagement start with the clutch slipping speed control, then driveshaft torsion control. The authors use only the clutch actuator to control the engagement, which is risk of engine stall during the launch.

In [3], a linear quadratic tracking controller for launch is proposed based on the simplify model with two states: engine speed and clutch speed. The

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conditions engine no-stall and no-lurch are considered. However, the friction behavior is not considered while using the engine torque and the clutch torque as control variables. In [4], the authors propose two hybrid optimal control strategies for clutch engagement based on MPC and linear quadratic controller. The friction behavior and load torque are neglected.

In [7], the authors based on the model predictive control are proposed a control law for launch, gearshift and idle mode of an AMT. The cost function is defined by the tracking requirement on the reference of the engine speed and the clutch slipping speed. The engine torque, giving by drive, and the load torque, which are needed for calculating control law, is estimated by an observer. The control law takes into account the conditions: engine no-stall and no-lurch. The clutch slipping time in the simulations is too long: approximation 5 seconds for a launch mode and 2.5 seconds for a gearshift mode, which concern the dissipated energy.

Some parameters in the DCT model are difficult to identify and time-varying, such as the friction coefficient of the clutch, the load torque depending on the road conditions, road slope, state of the tire and mass of the vehicle, win,... Thus, the aim of this paper is to develop a robust control law based on the sliding mode control theory to track the trajectory references of the clutch slipping speed and the engine speed during the launch and gearshift of a vehicle equipped with the DDCT. Incertitude on the parameters and input disturbance are considered. The trajectory references are pre-defined to reduce the jerk and the oscillations during and after the synchronization, and to avoid the dead zone of the engine. Some simulations are provided to show the benefits of the approach developed.

2. Vehicle powertrain dynamic model

The DDCTs use dual mass flywheel, which is composed of two discs connected by a spring-damper device. It is designed to filter high torsional frequency in the transmission. The simplified model with four states is presented in [12] (see Figure 1).

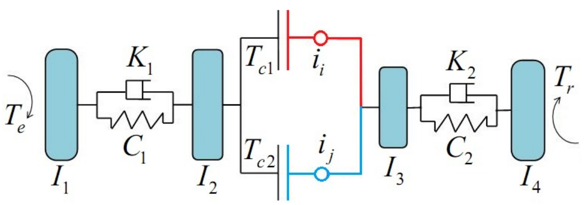


Fig. 1. Simplified model of powertrain with Dual clutch

where, I_1, I_2, I_3, I_4 are respectively the mass moment of inertia of the engine and one part of the flywheel, the other one part of flywheel and dual clutch drum, the equivalent mass of inertia of the input shafts, the output shafts and of the gears involved in the gearshift on the final drive, the equivalent mass of inertia of the vehicle on the wheel. K_1, C_1, K_2, C_2 are respectively the stiffness and damping coefficient of the flywheel, of the half-shaft. i_i, i_j are respectively the transmission ratios of the current and next gear involved in the shift. T_e is the engine torque. T_r is the load torque.

The differential equations describing the dynamics of the simplified system are given by

$$I_1\dot{\omega}_1 = T_e(.) - K_1(\theta_1 - \theta_2) - C_1(\omega_1 - \omega_2) \quad (1)$$

$$I_2\dot{\omega}_2 = K_1(\theta_1 - \theta_2) + C_1(\omega_1 - \omega_2) - T_{c1}(.) - T_{c2}(.) \quad (2)$$

$$I_3\dot{\omega}_3 = i_i T_{c1}(.) + i_j T_{c2}(.) - K_2(\theta_3 - \theta_4) - C_2(\omega_3 - \omega_4) \quad (3)$$

$$I_4\dot{\omega}_4 = K_2(\theta_3 - \theta_4) + C_2(\omega_3 - \omega_4) - T_r(.) \quad (4)$$

$$\dot{\theta}_i = \omega_i \quad (5)$$

Where ω_i, θ_i (i=1…4) are respectively the angular velocities and angular displacements of the engine crankshaft, dual clutch drum, final drive, and wheel. T_{c1}, T_{c2} are respectively the torques transferred by the sub-clutches for odd and even gear sets. The engine output torque is considered as a function of engine speed and throttle position (p) in percentage of opening $T_e = T_e(\omega_1, p)$. The torque transferred by the sub-clutches in the slipping phase is the friction torque. In our study, a static model with Stribeck effect is considered as follows:

$$T_{ci} = c_{ci}\mu(\omega_{ri})\text{sign}(\omega_{ri})F_{ni} \quad (6)$$

Where c_{ci} is the structure constant of the sub-clutch *i*, F_{ni} (i=1, 2) is the normal force applied on sub-clutch plate *i*, and $\mu(.)$ is the coefficient of friction that can be formulated as a function of sub-clutch slipping speed ω_{ri},

$$\mu(\omega_{ri}) = \mu_c + (\mu_s - \mu_c)e^{-(\omega_{ri}/\omega_s)^2} \quad (7)$$

Where, μ_c is the Coulomb friction coefficient, μ_s, ω_s are respectively the Stribeck friction coefficient and angular velocity. The gearshift is composed of four phases which are described below.

Gear change required

In this phase, the next gear i_j is engaged but the on-coming clutch for this gear is still open ($T_{c1}=0$). The off-going clutch is still completely closed. The engine torque is transferred by the off-going clutch. By combining the differential equations (2) and (3) with the conditions $\omega_2 = i_i\omega_3$ and $\dot{\omega}_2 = i_i\dot{\omega}_3$, we have the torque transferred by the off-going clutch:

$$T_{c1} = \left(T_{in}I_3 + T_{out}i_iI_2\right)\Big/\left(I_2i_i^2 + I_3\right) \quad (8)$$

Where: $T_{in} = K_1(\theta_1-\theta_2)+C_1(\omega_1-\omega_2)$

$T_{out} = K_2(\theta_3-\theta_4)+C_2(\omega_3-\omega_4)$

Torque phase

The on-coming clutch began the engagement. We divide this phase into two sub-phases. The first sub-phase, normal force begins to be applied to the on-coming clutch, the clutch torque is calculated by (6). The off-going clutch is still completely closed, but it is calculated by below equation:

$$T_{c1} = (T_{in}I_3 + T_{out}i_iI_2 - I_3T_{c2} - I_2T_{c2}i_ii_j)\big/(I_2i_i^2 + I_3) \quad (10)$$

In the second sub-phase, the off-going clutch and the on-coming clutch are sliding. The torque transferred by two sub-clutches are calculated by the equation (6)

Inertia phase

The off-going clutch is open ($T_{c1}=0$). The on-coming clutch is not yet closed, the on-coming clutch torque is calculated by the equation (6).

Gearshift complete

The on-coming clutch closes completely and the off-going clutch is open. The on-coming clutch torque is calculated by below equation

$$T_{c2} = (T_{in}I_3 + T_{out}i_jI_2)\big/(I_2i_j^2 + I_3) \quad (11)$$

3. Control design

The main result of the paper is proposed in this section. The control law for launch and gearshift based on sliding mode control approach for nonlinear uncertain systems [13] is developed. The goal of this control is to minimize the jerk, while limiting the clutch slipping time. Moreover, the dead zone of the engine must be avoided, which means that the engine speed has to be controlled. For robustness issues, both road conditions and model parametric variations have to be considered. In the gearshift process, the engine speed and clutch slipping speed must be controlled to aims to guarantee the clutch close smoothly, quickly and the engine no-kill. The whole management of the gearshift control relies on tracking two trajectories, one of the engine speed and one of the clutch slip speed which are described in the sections below.

3.1 Trajectory reference

3.1.1 Reference trajectory for the engine speed

The reference trajectory of the engine speed during clutch engagement is presented in some works [7], [10], [14]. In the gear shifting up process, the engine speed (9) is controlled to decrease ensuring a rapid shifting and comfort. On the contrary, in the gear shifting down process, the engine speed is controlled to increase. We use a polynomial of degree 3 to define the reference trajectory of engine speed.

$$\omega_1^{ref} = a_0 + a_1t + a_2t^2 + a_3t^3 \quad (12)$$

Idealization, the vehicle acceleration is constant during the shifting. Thus, at the synchronization $\omega_4(t_f) = \omega_4(t_0) + \dot{\omega}_4(t_0)(t_f - t_0)$ and $\dot{\omega}_4(t_f) = \dot{\omega}_4(t_0)$, where t_0, t_f are respectively the time of beginning of shifting and the synchronization. We assume: $\omega_1(t_0)=\omega_2(t_0)$, $\omega_3(t_0)=\omega_4(t_0)$, $\omega_1(t_f)=\omega_2(t_f)$, $\dot{\omega}_3(t_f)=\dot{\omega}_4(t_f)$, $\dot{\omega}_1(t_0)=\dot{\omega}_2(t_0)$, $\dot{\omega}_3(t_0)=\dot{\omega}_4(t_0)$, $\dot{\omega}_1(t_f)=\dot{\omega}_2(t_f)$ and $\dot{\omega}_3(t_f)=\dot{\omega}_4(t_f)$. Thus, we have:

$$\omega_1^{ref}(t_f) = (\omega_1(t_0) + \dot{\omega}_1(t_0)(t_f - t_0))i_j\big/i_i \quad (13)$$

$$\dot{\omega}_1^{ref}(t_f) = i_j\dot{\omega}_1(t_0)\big/i_i \quad (14)$$

The initial conditions are:

$$\omega_1^{ref}(t_0) = \omega_1(t_0) \quad (15)$$

$$\dot{\omega}_1^{ref}(t_0) = \dot{\omega}_1(t_0) \quad (16)$$

The parameters a_i (i=0…3) in the equation (12) can be found by solving the equations (13) - (16).

3.1.2 Reference trajectory for the clutch slipping speed

In the gearshift control, the reference trajectory for clutch slipping speed (ω_{r2}^{ref}) is defined to satisfy a smooth engagement and minimize the dissipated friction energy. For a smooth engagement, the reference trajectory of clutch slip speed is defined to satisfy the conditions called no-lurch. This condition can be characterized as no difference of the rotational acceleration of dual clutch drum and clutch disc at the synchronization. The following conditions must be satisfied: (1) slippage should be finished after the selected time, $\omega_{r2}^{ref}(t_f) = \omega_2(t_f) - i_j\omega_3(t_f) = 0$, (2) at the synchronization, the clutch slipping acceleration must be equal to zero, $\dot{\omega}_{r2}^{ref}(t_f) = \dot{\omega}_2(t_f) - i_j\dot{\omega}_3(t_f) = 0$,(3) the initial conditions: $\omega_{r2}^{ref}(t_0) = \omega_2(t_0) - i_j\omega_3(t_0)$, $\dot{\omega}_{r2}^{ref}(t_0) = \dot{\omega}_2(t_0) - i_j\dot{\omega}_3(t_0)$. Different choices are possible to define a trajectory satisfying those constraints, as above, we use a polynomial of degree 3.

3.2 Engine speed control

The engine torque is divided into two parts, the first part is given by the driver request (T_e^d), and the second part is considered as a control input (T_e^c). Rewriting the dynamic equation of the engine speed(1), the following equation is obtained

$$\dot{x}_1 = f_1 + g_1.u_1 \tag{17}$$

With $x_1 = \omega_1$, $g_1 = 1/I_1$, $u_1 = T_e^c$,

$f_1 = (T_e^d - K_1(\theta_1 - \theta_2) - C_1(\omega_1 - \omega_2))/I_1$.

The parameters I_1, K_1, C_1 are difficult to identify, T_e^d depend on the driver behavior. Therefore, they are considered as the bounded uncertain parameters $I_1 \in [I_{1\min}, I_{1\max}]$, $C_1 \in [C_{1\min}, C_{1\max}]$, $K_1 \in [K_{1\min}, K_{1\max}]$, $T_e^d \in [T_e^{\min}, T_e^{\max}]$. Sine $T_e^d \geq 0$ and $\theta_1 - \theta_2 \geq 0$ thus, the dynamics function f_1 and gain input function g_1 are bounded: $f_1 \in [f_{1\min}, f_{1\max}]$, $g_1 \in [g_{1\min}, g_{1\max}]$ with $g_{1\max} = 1/I_{1\min}$, $g_{1\min} = 1/I_{1\max}$

$$f_{1\max} = T_e^{\max}/I_{1\min} - K_{1\min}(\theta_1 - \theta_2)/I_{1\max} + |\omega_1 - \omega_2| C_{1\max}/I_{1\min}$$
$$f_{1\min} = T_e^{\min}/I_{1\min} - K_{1\max}(\theta_1 - \theta_2)/I_{1\min} - |\omega_1 - \omega_2| C_{1\max}/I_{1\min}$$

We define the nominal values $\hat{f}_1 = 0.5(f_{1\max} + f_{1\min})$, $F_1 = 0.5(f_{1\max} - f_{1\min})$, $\hat{g}_1 = \sqrt{g_{1\max} g_{1\min}} = \sqrt{1/(I_{1\max} I_{1\min})}$, and

$\beta_1 = \sqrt{g_{1\max}/g_{1\min}} = \sqrt{I_{1\max}/I_{1\min}}$

Thus, we have: $|\hat{f}_1 - f_1| \leq F_1$, $\beta_1^{-1} \leq g_1^{-1}\hat{g}_1 \leq \beta_1$

In order for the engine speed to track the reference trajectory x_1^{ref}, a sliding surface S_1 is defined by the following integral structure

$$S_1(t) = \left(\frac{d}{dt} + \lambda_1\right)\int_0^t e_1(\tau)d\tau = e_1(t) + \lambda_1\int_0^t e_1(\tau)d\tau$$

where, λ_1 is a strictly positive constant, e_1 is the tracking error, $e_1(t) = x_1(t) - x_1^{ref}(t)$. We have the control law [13]:

$$u_1 = \hat{u}_1 - k_1\text{sign}(S_1) \tag{18}$$

With: $\hat{u}_1 = \hat{g}_1^{-1}(-\hat{f}_1 + \dot{x}_1^{ref} - \lambda e_1)$

$k_1 = (\beta_1 - 1)|\hat{u}_1| + (F_1 + \eta_1)\beta_1\hat{g}_1^{-1}$. Where, η_1 is a strictly positive constant.

3.3 Clutch slip control

In the gear shifting phase (including torque phase and inertia phase), the on-coming clutch is controlled, assuming that it is the second sub-clutch. By combining the equations (2) and (3), the dynamics equation of the on-coming clutch slipping speed is given as follows

$$\begin{aligned} \dot{x}_2 &= T_{in}(.)/I_2 + i_j T_{out}(.)/I_3 - \left(1/I_2 + i_i i_j/I_3\right)T_{c1} \\ &\quad - \left(1/I_2 + i_j^2/I_3\right)c_{c2}\text{sign}(x_2)F_{n2}\mu(x_2) \\ &= f_2(.) + g_2(.)u_2 \end{aligned} \tag{19}$$

Where: $x_2 = \omega_2 - i_j\omega_3$, $u_2 = -F_{n2}\text{sign}(x_2)$

$f_2(.) = T_{in}(.)/I_2 + i_j T_{out}(.)/I_3 - \left(1/I_2 + i_i i_j/I_3\right)T_{c1}$

$g_2(.) = \left(1/I_2 + i_j^2/I_3\right)c_{c1}\mu(x_2)$

The parameters I_2, I_3, C_1, K_1, C_2, K_2, μ_s and μ_c are the bounded uncertain parameters. Since $\theta_1 - \theta_2 \geq 0$ and $\theta_3 - \theta_4 \geq 0$ thus, the function f_2 and g_2 are bounded,

$$\begin{aligned} f_{2\max} &= K_{1\max}(\theta_1 - \theta_2)/I_{2\min} + C_{1\max}|\omega_1 - \omega_2|/I_{2\min} \\ &+ i_j K_{2\max}(\theta_3 - \theta_4)/I_{3\min} + i_j C_{2\max}|\omega_3 - \omega_4|/I_{3\min} \\ &+ \left(1/I_{2\min} + i_i i_j/I_{3\min}\right)|T_{c1}| \end{aligned}$$

$$\begin{aligned} f_{2\min} &= K_{1\min}(\theta_1 - \theta_2)/I_{2\max} - C_{1\min}|\omega_1 - \omega_2|/I_{2\min} \\ &+ i_j K_{2\min}(\theta_3 - \theta_4)/I_{3\max} - i_j C_{2\min}|\omega_3 - \omega_4|/I_{3\min} \\ &- \left(1/I_{2\min} + i_i i_j/I_{3\min}\right)|T_{c1}| \end{aligned}$$

$$g_{2\min} = c_{c2}\left(1/I_{2\max} + i_j^2/I_{3\max}\right)\mu_c^{\min}$$

$$g_{2\max} = c_{c2}\left(1/I_{2\min} + i_j^2/I_{3\min}\right)\mu_s^{\max}.$$

We define the nominal values of the dynamics system (19): $\hat{f}_2 = 0.5(f_{2\max} + f_{2\min})$, $\hat{g}_2 = \sqrt{g_{2\max} g_{2\min}}$, and the bounded values $F_2 = 0.5(f_{2\max} - f_{2\min})$, $\beta_2 = \sqrt{g_{2\max}/g_{2\min}}$. We have: $|\hat{f}_2 - f_2| \leq F_2$, $\beta_2^{-1} \leq g_2^{-1}\hat{g}_2 \leq \beta_2$.

As previously, in order for clutch slipping speed to track the reference trajectory x_2^{ref}, a sliding surface S_2 is defined by the following integral structure

$$S_2(t) = \left(\frac{d}{dt} + \lambda_2\right)\int_0^t e_2(\tau)d\tau = e_2(t) + \lambda_2\int_0^t e_2(\tau)d\tau$$

where, λ_2 is a strictly positive constant, e_2 is the tracking error, $e_2(t) = x_2(t) - x_2^{ref}(t)$. We have the control law [13]

$$u_2 = \hat{u}_2 - k_2\text{sign}(S_2) \quad (20)$$

With $\hat{u}_2 = \hat{g}_2^{-1}(-\hat{f}_2 + \dot{x}_2^{ref} - \lambda_2 e_2)$,

$k_2 = (\beta_2 - 1)|\hat{u}_2| + (F_2 + \eta_2)\beta_2\hat{g}_2^{-1}$. Where η_2 is a strictly positive constant.

Table 1. The simplified model parameters

Parameters	Maximum value	Minimum value	Real value	Incertitu -de (%)
I_1 [kgm^2]	3.11	2.30	2.43	10.12
I_2 [kgm^2]	0.32	0.24	0.25	11.43
I_3 [kgm^2]	11.50	8.50	10.00	0.5
I_4 [kgm^2]	290.00	240.00	281.37	6.18
C_1 [Nm]	46.0	34.0	38.1	4.79
K_1 [Nm]	184.0	136.0	182.1	13.8
C_2 [Nm]	690.0	510.0	550.3	8.29
K_2 [Nm]	18745.0	13855.0	16717.0	2.56
μ_s	1.38	0.85	1.25	11.94
μ_c	1.15	0.85	0.93	7.35
i_i	-	-	9.6398	-
i_j	-	-	6.2172	-
c_{c1}, c_{c2} [m]	-	-	0.28	-

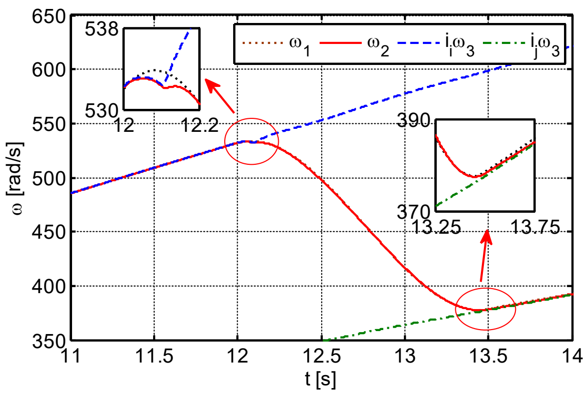


Fig. 2. Engine speed and clutch speed in the 1st – 2nd gear up-shift mode

4. Simulation results

To show the effectiveness of the proposed control law, different tests with parametric variations have been realized with the parameters shown in table 1. The simulations are performed using a simplified model in Simulink®. The results of simulation of the gearshift case shown in figures 2 to 6. The initial vehicle speed for 1st – 2nd gear up-shift is approximately 64 km/h, engine speed near 533 rad/s and finish after 1.5 seconds (Figure 2).

For 1st – 2nd gear down-shift mode, initial vehicle speed is 5.3 km/h, engine speed near 100rad/s and finish after 1.5 seconds (Figure 3)

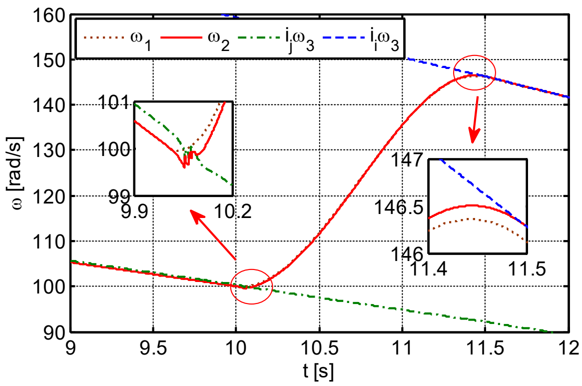


Fig. 3. Engine speed and clutch speed in the 2nd – 1st gear down-shift mode

The figure 4 shows the torques transferred across each of the two sub-clutches of a dual clutch transmission (T_{c1}, T_{c2}) in the case of 1st – 2nd gear up-shift.

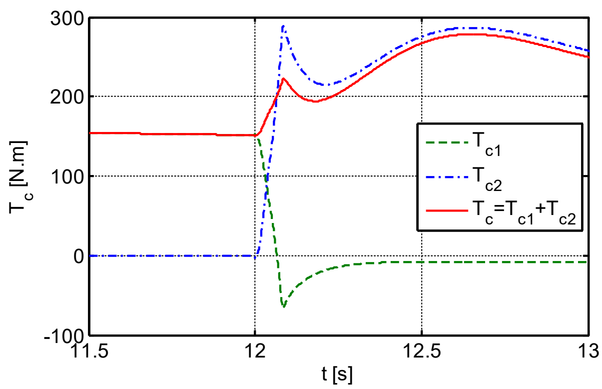


Fig. 4. Dual clutch torque in the 1st – 2nd gear up-shift mode

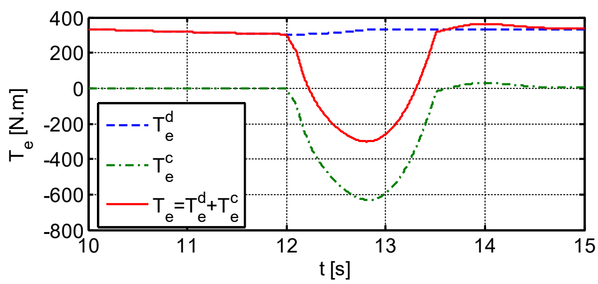


Fig. 5. Engine torque in the 1st-2nd gear up-shift mode

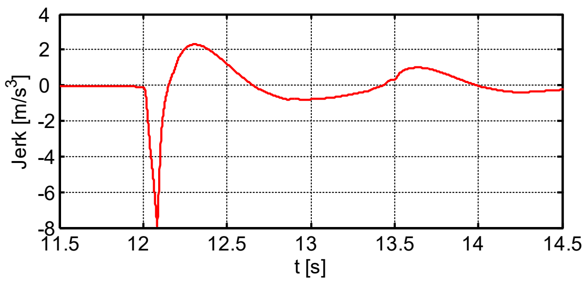
At the beginning of the up-shift, the normal force the off-going clutch is reduced by the control strategy. When the off-going clutch starts to slip, the normal force of the on-coming clutch is controlled to synchronization. The engine torque shows in the figure 5,

Where T_e^d is the torque requested by the driver and T_e^u is the input control for engine torque. The engine torque is depressed for engine speed is descended. The shifting time depends on engine capacity to descend the engine speed.

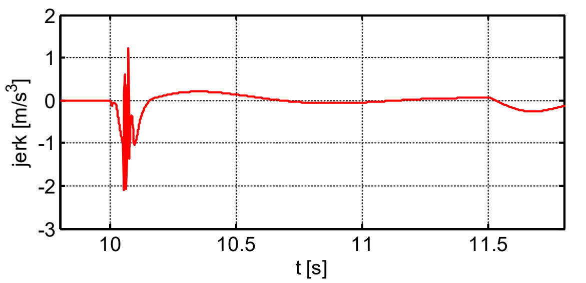
The jerk of the vehicle during the gearshift is shown in figures 6. The bounded values are acceptable (<10 m/s^3 [10]).

5. Conclusion

A nonlinear model of the powertrain has been proposed. From the simplified model, we propose a gearshift control law based on the sliding mode control theory for nonlinear uncertain systems. The simulation results showed the efficiency and robustness of the proposed control law with respect to parametric uncertainties and input disturbances (driver behavior, road conditions, wind...). Vehicle jerk is acceptable in our simulation with the limitation of the engagement time. The shifting time depends on engine capacity to descend the engine speed.



a) 1st-2nd gear up-shift



b) 2nd-1st gear down-shift

Fig. 6. Vehicle jerk

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