

\textbf{H}_\infty \text{ gearshift control of a dual clutch based on uncertain TS models}

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\textit{Abstract—}The dual clutch transmission was introduced in vehicles to improve driving comfort compared with manual transmissions, and performances, fuel efficiency compared with automatic transmissions. The dual clutch transmission allows to shift gears quickly, comfortably and without interruption in traction. In the dual clutch transmission, the management of the dual clutch in the gearshift phase is a key point especially when considering driving comfort. In this paper, we propose a dual clutch control law for gearshift based on a Takagi-Sugeno model. The goal is to ensure a smooth gearshift while limiting shift time. The approach used is the tracking of the dual clutch sliding speed and engine speed reference. In addition, parametric variations of the model and disturbances are also considered.

I. INTRODUCTION

In the literature, two principal types of vehicle transmission are discussed, one is the manual transmission and the other is the automatic one. There exist two technologies for automated lay-shaft gearings transmissions to upgrade driving comfort and transmission performance. One uses a single clutch and is basically a manual transmission with an added-on control unit that automates the clutch and shift operations (called Automated Manual Transmission - AMT). The other one, using a dual clutch (called Dual Clutch Transmission - DCT). The DCT upgrades the shift time, the driving comfort compared with the AMTs, and the fuel efficiency compared with automatic transmissions. The DCT consists of two independent sub-gearboxes, one for the even gear sets and the other for odd gear sets, each one activated by separate clutches: the on-coming clutch and the 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 has also the ability to allow the driver to gear selection in manual mode. There are two fundamental types of clutches utilised in DCTs: either two wet multi-disc clutches which are bathed in oil for cooling (WDCT), or two dry single-disc clutches (DDCT). In this paper, we are interested in the DDCT.

The problems associated with DCT/AMT in literature are the engagement of the clutch during launch and gearshift, the gearshift strategy. The goal is to improve driving comfort, performance and reduce fuel consumption and emission of \textit{CO}_2. Specifically, the dry dual clutch engagement must be controlled in order to satisfy contradictory objectives such as minimizing the slipping energy and preserving the driving comfort. To achieve these goals, in the literature on the clutch management, many different approaches have been proposed: optimal control [1], [2], [3], [4], [5], flatness control [6], model predictive control [7], sliding mode control [8], [9], [10], PID control [11]. Most approaches are based on a linear model of the powertrain.

In [11], the authors have developed a smooth control law for gear shifting of an DDCT based on the use of a PID controller to track the engine and on-coming clutch slipping speed desired reference signals. The estimation torque of the off-going clutch is considered as an input signal for the controller. The gearshift controller output signals are the engine and on-coming clutch torques. The normal force is generated through the clutch characteristic which depends on the coefficient of friction.

In [7], a model predictive control is proposed to manage launch, gearshift and idle modes of an AMT. A cost function defined by the tracking reference requirement for the engine and the clutch slipping speeds is considered. The engine torque which is needed for calculating control law is estimated by an observer. The control law takes into account the engine no-stall and no-lurch conditions. The clutch slipping time in the simulation is rather too long: approximately 5 seconds for a launch and 2.5 seconds for a gearshift.

In our work [10], a launch controller based on a DDCT model taking into account the dual clutch actuator is developed according to sliding mode control theory.

In [12], the authors used a backstepping technique and an ISS (Input-to-State Stability) methodology to carry out the clutch slip control problem during the shifting with an automatic transmission type clutch-to-clutch\textsuperscript{1}. The authors focus on the “inertia phase”\textsuperscript{2}.

In [13], the authors have developed a dynamic model and a logic shift for a DCT. The normal force profiles for on-coming and off-going clutches have been created in order to upgrade as much as possible the shift performance based on simulation trials.

In the dual clutch model, the friction coefficient is the difficult parameter to identify. The vehicle mass is variable. The load torque is unknown and depends on the road conditions, the road slope, the tires states and the vehicle mass. Thus, the aim of this paper is to develop a robust control law based on the use of a Takagi-Sugeno (TS) fuzzy model to

\textsuperscript{1}It is a type of automatic transmission used one-way clutches, which permits only the transfer of a positive torque.

\textsuperscript{2}It is the phase where the engine speed is synchronized to that of the on-coming clutch, the off-going clutch torque being null.
track reference trajectories for the dual clutch slipping speed and the engine speed during the gearshifts.

II. Vehicle Powertrain Dynamic Model

The model is developed based on some mechanical hypotheses. The motion of the engine on its suspension is neglected; the drive train is assumed to be symmetric. The powertrain model is a one-dimensional mechanical system in which each element is a lumped mass model and a spring-damper model as shown in Figure 1, where, $I_1, I_2, I_{31}, I_{32}, I_{41}, I_{42}, I_5, I_6, I_7$ are the mass moment of inertia of the engine and flywheel, the dual clutch drum, the clutch disc, the gears and the input shaft of two sub-gearboxes, the gears and the output shaft of two sub-gearboxes, the final drive, the half-shaft and wheels and of the vehicle mass, respectively. $K_1, C_1, K_{21}, C_{21}, K_{31}, C_{31}, K_3, C_4, (i = 1, 2)$ are the stiffness and damping coefficients of the flywheel, the input shaft of two sub-gearboxes, the output shaft and of the half-shaft, respectively, $i_o, i_e$ are the gear ratios of the current and next gear involved in the shift, respectively, $i_0$ is the final drive ratio, $T_c$ is the engine torque, $T_r$ is the load torque.

Moreover, consider the following assumptions: (1) the tires have a perfect adherence and no transitory effects on tire-ground contact, (2) the input shafts and output shafts of the two sub-gearboxes are infinitely rigid. A simplified model with four states is obtained (see Figure 2), where, simplified system are given by:

$$I_1 \dot{\omega}_1 = T_c(\cdot) - K_1(\theta_1 - \theta_2) - C_1(\omega_1 - \omega_2)$$  \hspace{0.5cm} (1)  

$$I_2 \dot{\omega}_2 = K_1(\theta_1 - \theta_2) + C_1(\omega_1 - \omega_2) - T_{c1}(\cdot) - T_{c2}(\cdot)$$   \hspace{0.5cm} (2)  

$$I_3 \dot{\omega}_3 = i_1 T_{c1}(\cdot) + i_2 T_{c2}(\cdot) - K_2(\theta_3 - \theta_4) - C_2(\omega_3 - \omega_4)$$   \hspace{0.5cm} (3)  

$$I_4 \dot{\omega}_4 = K_2(\theta_3 - \theta_4) + C_2(\omega_3 - \omega_4) - T_r(\cdot)$$   \hspace{0.5cm} (4)  

where, $\omega_i$ are the angular velocities of the engine crankshaft, clutch drum, final drive and wheel, respectively, $\theta_i$ are the angular displacements of the engine crankshaft, clutch drum, final drive and wheel, respectively, $T_{c1}, T_{c2}$ are the clutch off-going and clutch on-coming torque, respectively, $T_c$ is the engine torque, and $T_r$ is the vehicle resistance torque.

A. Engine model

The engine is modeled as a mean value torque generator which does not include the engine transients. The engine output torque is considered as a function of the engine speed and the throttle position. In this paper, the engine is modeled by a engine torque map as shown in Figure 3.

$$T_{ci} = \gamma_c \mu_c (\Delta \omega_i) \text{sign}(\Delta \omega_i) F_{mi}$$   \hspace{0.5cm} (6)  

with $\gamma_c = 2 n_d r_c$, where, $n_d$ is the number of clutch discs, $r_c$ is the friction radius of the clutch disc, $F_{mi}$ are the normal forces applied on the dual clutch plates, $\mu(\cdot)$ is the coefficient of friction depending on the clutch slipping speed.

$$\mu(\Delta \omega_i) = \mu_c + (\mu_s - \mu_c) e^{-\left(\frac{\Delta \omega_i}{\omega_{s}}\right)^2}$$   \hspace{0.5cm} (7)  

where, $\mu_c$ is the Coulomb friction coefficient, $\mu_s$ is the Strubeck friction coefficient, $\omega_s$ is the Strubeck angular velocity, $\Delta \omega_i$ are the dual clutch slip speeds.

The complete gearshift process is composed of four phases: (1) gear change required, (2) torque phase\(^3\), (3) inertia phase\(^4\), and (4) gearshift complete. We focus on the torque phase and inertia phase.

\(^3\)It is the phase, where engine torque is transferred from the off-going clutch to the on-coming clutch.

\(^4\)It is the phase where the engine speed is synchronized to that of the on-coming clutch, off-going clutch torque is null.
1) Torque phase: The on-coming clutch began the engagement. We divide this phase into two sub-phases. In the first sub-phase, normal force begins to be applied to on-coming clutch, the on-coming clutch torque increases. The off-going clutch is still completely closed, but the torque transferred by the off-going clutch is reduced. We have:

\[ T_{c1} = (T_{in}l_1 + T_{out}l_2 - l_3T_{c2} - l_2T_{c2}l_1)/(I_1^2 + I_3) \]  
(8)

\[ T_{c2} = \gamma \text{sign}(\Delta \omega_2) \mu(\Delta \omega_2)F_{c2} \]  
(9)

with, \( T_{in} \triangleq K_1(\theta_1 - \theta_2) + C_1(\omega_1 - \omega_2) \), \( T_{out} \triangleq K_2(\theta_1 - \theta_2) + C_2(\omega_1 - \omega_2) \), and \( \Delta \omega_2 \triangleq \omega_2 - \omega_1 \). In the second sub-phase, the off-going clutch and the on-coming clutch are sliding. We have

\[ T_{c1} = \gamma \text{sign}(\Delta \omega_1) \mu(\Delta \omega_1)F_{c1} \]  
(10)

where, \( \Delta \omega_1 \triangleq \omega_2 - \omega_1 \), and

\[ T_{c2} = \gamma \text{sign}(\Delta \omega_2) \mu(\Delta \omega_2)F_{c2} \]  
(11)

2) Inertia phase: The off-going clutch is fully open (\( T_{c1} = 0 \)), the on-coming clutch is not yet closed. The torque transferred by the on-coming clutch is:

\[ T_{c2} = \gamma \text{sign}(\Delta \omega_2) \mu(\Delta \omega_2)F_{c2} \]  
(12)

C. Vehicle air, roll and slope resistance

The vehicle resistance forces include the aerodynamic resistance force \( F_a \), the rolling resistance force \( F_r \), and the uphill driving force caused by gravity when driving on non-horizontal roads, \( F_g \).

\[ F_a = 0.5\rho_a A_f C_a (v_a + v_u)^2 \]  
(13)

where, \( A_f \) is the frontal area of vehicle, \( C_a \) is the aerodynamic drag coefficient, \( v_a \) is the vehicle speed, \( v_u \) is the wind speed, \( \rho_a \) is the density of the ambient air.

\[ F_r = m_o g \cos(\beta) c_{r}(\cdot) \]  
(14)

where, \( g \) is the acceleration due to gravity, \( c_{r}(\cdot) \) is the rolling friction coefficient, \( \beta \) is the slope angle of road.

\[ F_g = m_o g \sin(\beta) \]  
(15)

The vehicle resistance torque to forward is

\[ T_r = (F_a + F_r + F_g)r_w \]  
(16)

III. ROBUST STABILIZATION OF UNCERTAIN TS MODELS

We consider an uncertain TS model with external disturbances defined by:

\[ \dot{x} = \sum_{i=1}^{r} h_i(z)((A_i + \Delta A_i)x + (B_i + \Delta B_i)u + B_{wi}w) \]  
\[ y = \sum_{i=1}^{r} h_i(z)C_i x \]  
(17)

where, \( w(t) \) is the disturbance input, \( A_i, B_i, C_i \) and \( B_{wi} \), \( i = 1, \ldots, r \), are matrices with appropriate dimensions, \( r \) is the number of rules, \( h_i(z) \) are normalized membership functions. The parameter uncertainties are usually written as [14]: \( \Delta A_i = H_a \Delta a(t)E_{a_i}, \Delta B_i = H_b \Delta b(t)E_{b_i} \), where the matrices \( H_a \), \( H_b, E_{a_i} \) and \( E_{b_i} \) are constants and the matrices \( \Delta a(t) \) and \( \Delta b(t) \) satisfy the conditions \( \Delta a(t)^T\Delta a(t) \leq I, \Delta b(t)^T\Delta b(t) \leq I \). The control laws commonly used for the stabilization of these models are of PDC type (Parallel Distributed Compensation) [14]

\[ u(t) = -\sum_{j=1}^{r} h_j(z)F_jx \]  
(18)

The stability analysis for TS models often leads to finding the best conditions to an inequality of the form

\[ \sum_{i=1}^{r} \sum_{j=1}^{r} h_i(z)h_j(z)\Gamma_{ij}(x) < 0 \]  
(19)

where the symmetric matrices \( \Gamma_{ij}(x) \) depend affinely on the unknown variables \( x \in \mathbb{R}^n \), the functions \( h_i(z) \) being nonlinear functions that observe the convex sum property, i.e., \( \sum_{i=1}^{r} h_i(z) = 1 \) and \( h_i(z) \geq 0, i = 1,2,\ldots,r \).

**Lemma 1:** [15] Condition (19) is satisfied provided that the following conditions hold

\[ \Gamma_{ii} < 0 \]  
(20)

\[ \frac{2}{r} \Gamma_{ii} + \Gamma_{ij} + \Gamma_{ji} < 0 \]  
(21)

for \( i, j = 1,2,\ldots,r, i \neq j \).

**Theorem 1:** The continuous uncertain TS model (17) with PDC control law (18) is Globally Asymptotically Stabilizable (GAS) and the attenuation of the disturbance \( w \) is at least \( \gamma \), if there exist matrices \( X > 0 \) and \( M_i, i = 1,\ldots,r \), and the scalars \( \tau_0 > 0, \tau_i > 0 \) such that, with

\[ \begin{bmatrix} \mathcal{H}(A_iX - B_iM_j) + \tau_i H_aH_a^T \sum_{j=1}^{r} h_j(z)h_j(z) \Gamma_{ij}(x) + \tau_i H_bH_b^T \sum_{j=1}^{r} h_j(z)h_j(z) \Gamma_{ij}(x) \end{bmatrix} \]

\[ \begin{bmatrix} B_{wi}^T - \gamma^2 I \begin{bmatrix} C_i \begin{bmatrix} 0 & -I \end{bmatrix} \begin{bmatrix} E_{ai}X \end{bmatrix} \end{bmatrix} \end{bmatrix} \begin{bmatrix} 0 \begin{bmatrix} 0 \end{bmatrix} - \tau_i I \end{bmatrix} \]

conditions (20) hold, where \( \mathcal{H}(X) = X + X^T \). Moreover, if the conditions are satisfied, then some stabilizing PDC gains are given by: \( F_i = M_iX^{-1}, i = 1,\ldots,r \).

**Proof:** The proof is omitted due to lack of space, the reader can refer to [16].

IV. GEARSHEFNT CONTROL DESIGN

**A. Reference trajectory of the engine speed**

In the upshift process, engine speed is controlled to decrease, on the contrary, in the downshift, engine speed is controlled to increase, which is to ensure a rapid and comfort gearshifts. The reference trajectory of the engine speed can be defined from the initial conditions at the time of beginning of the gearshift and the conditions at the synchronisation. To ensure that states transit smoothly without jumps at the beginning of the gearshift, any desired trajectory feasible from time \( t_0 \) has to start with the same engine speed and engine acceleration as those of the plant.

\[ \omega_1^{ref}(t_0) = \omega_1(t_0), \quad \omega_1^{ref}(t_0) = \omega_1(t_0) \]  
(21)
Ideally, the vehicle acceleration is constant during the
gearshift, and at the beginning of engagement $t_0$ and synchro-
nization $t_s$, the four masses composing the simplified model
have the same speed angular: $\omega_3(t_0) = i_2 \omega_3(t_0) = i_3 \omega_3(t_0)$ and $\omega_3(t_s) = i_2 \omega_3(t_s) = i_3 \omega_3(t_s)$, and the
same acceleration $\dot{\omega}_3(t_0) = \dot{\omega}_3(t_s) = i_2 \omega_3(t_0) = i_3 \omega_3(t_0)$ and $\ddot{\omega}_3(t_0) = \ddot{\omega}_3(t_s) = i_2 \omega_3(t_0)$, thus we have

$$\omega_3^{ref}(t_0) = (\omega_3(t_0) + \omega_3(t_0)(t_s - t_0))i_2/i_2$$

$$\dot{\omega}_3^{ref}(t_0) = \dot{\omega}_3(t_0)i_2/i_2$$

Different choices are possible to define a trajectory satisfying
these conditions, in this paper we choose a polynomial of degree 3.

B. Reference trajectory of on-coming clutch slip speed

The goal of the control is to ensure a smooth gearshift
while minimizing the shift time. Generally, if the duration of clutch slip is limited to a sufficiently short time, the
dissipated energy is relatively small. Similar to the previous
one, to ensure that states transit smoothly without jumps at the
beginning of clutch engagement, any desired trajectory feasible from time $t_0$ to start with the same clutch sliding speed and clutch sliding acceleration as those of the plant. Thus, the initial conditions are

$$\Delta \omega_3^{ref}(t_0) = \omega_3(t_0) - i_2 \omega_3(t_0)$$

(24)

$$\Delta \dot{\omega}_3^{ref}(t_0) = \dot{\omega}_3(t_0) - i_2 \dot{\omega}_3(t_0)$$

(25)

Moreover, to minimize the dissipated friction energy, the
slipage should be finished after the selected time $\Delta t = t_s - t_0$
and satisfy the conditions called no-lurch [17]

$$\Delta \omega_3^{ref}(t_s) = \omega_3(t_s) - i_2 \omega_3(t_s) = 0$$

(26)

$$\Delta \dot{\omega}_3^{ref}(t_s) = \dot{\omega}_3(t_s) - i_2 \dot{\omega}_3(t_s) = 0$$

(27)

C. Application to the control of a dual clutch during the
gearshift

The engine torque is divided into two parts. The first part
is given by the driver’s request $T^d$, and the second
part is considered as the control variable $T^c$. The variables
of interest are $x = (\omega_1, \omega_2, \omega_3, \omega_4, \omega_5, \omega_6, \omega_7, \omega_8)^T$, the
control variables are $u_1 = T^c, u_2 = F_{m2}$. We assume that the
clutch sliding speed is always positive in the upshift phase and negative in the downshift. Thus, the dynamic model of the
trajectory tracking error is written as follows

$$\dot{e}_1 = \begin{bmatrix} C_1 \lambda_1 e_2 + \frac{1}{I_1} u_1 + \frac{1}{I_1} T^d(\cdot) - \frac{K_1}{I_1} (\theta_1 - \theta_2) - \omega_3^{ref}(\cdot) \end{bmatrix}$$

(28a)

$$\dot{e}_2 = -\begin{bmatrix} \frac{1}{I_1} + \frac{1}{I_2} C_1 e_2 + \frac{1}{I_1} u_1 + \frac{1}{I_1} T^d(\cdot) - \frac{K_1}{I_1} (\theta_1 - \theta_2) - \omega_3^{ref}(\cdot) \end{bmatrix}$$

(28b)

$$\dot{e}_3 = \begin{bmatrix} \frac{C_1}{I_2} e_2 + \frac{i_2 C_2}{I_3} e_4 - \frac{1}{I_1} + \frac{i_2}{I_3} \gamma \mu(\cdot) \Delta \omega_3^{ref} \end{bmatrix}$$

(28c)

$$\dot{e}_4 = -\begin{bmatrix} \frac{1}{I_1} + \frac{1}{I_3} C_4 e_4 + \frac{i_2}{I_3} \gamma \mu(\cdot) \Delta \omega_3^{ref} \end{bmatrix}$$

$$\dot{e}_5 = -\begin{bmatrix} \frac{1}{I_1} + \frac{1}{I_3} C_4 e_4 + \frac{i_2}{I_3} \gamma \mu(\cdot) \Delta \omega_3^{ref} \end{bmatrix}$$

where, $e = x - x_r = (\omega_3^{ref}, 0, \Delta \omega_3^{ref}, 0)^T$. In the model (28),
$\mu(\cdot)$ is a bounded nonlinear function, $\mu(\cdot) \in [\mu_c, \mu_c]$. The
parameters $I_4, \mu_c, \mu_c$ are uncertain, we assume that they are
bounded,

$$I_4 \in [I_4^{min}, I_4^{max}], \ \mu_c \in [\mu_c^{min}, \mu_c^{max}], \ \mu_c \in [\mu_c^{min}, \mu_c^{max}]$$

The uncertain parameters are presented as follows

$$\dot{e}_1 = \begin{bmatrix} \frac{1}{I_1} e_2 + \frac{1}{I_1} T^d(\cdot) - \frac{K_1}{I_1} (\theta_1 - \theta_2) - \omega_3^{ref}(\cdot) \end{bmatrix}$$

(28d)

$$\dot{e}_2 = -\begin{bmatrix} \frac{1}{I_1} e_2 + \frac{1}{I_1} T^d(\cdot) - \frac{K_1}{I_1} (\theta_1 - \theta_2) - \omega_3^{ref}(\cdot) \end{bmatrix}$$

(28e)

$$\dot{e}_3 = -\begin{bmatrix} \frac{1}{I_1} + \frac{1}{I_3} e_4 - \frac{1}{I_1} + \frac{i_2}{I_3} \gamma \mu(\cdot) \Delta \omega_3^{ref} \end{bmatrix}$$

(28f)
The matrices of the uncertain term

\[ H_a = \begin{pmatrix} 0 & 0 & 1 \\ 0 & 0 & 0 \end{pmatrix}, \quad H_b = \begin{pmatrix} 0 & 0 & 0 \\ 1 & 0 & 0 \end{pmatrix} \]

\[ E_{ai} = \begin{pmatrix} 0 & 0 & \frac{1}{2} \left( \frac{1}{I_{4}\text{max}} - \frac{1}{I_{4}} \right) C_2 \end{pmatrix}, \quad i = 1, 2 \]

\[ E_{bi} = \begin{pmatrix} 0 & \frac{1}{j_k} \mu_{\text{max}} \mu_{\text{min}}^2 \\ 0 & \frac{1}{j_k} \mu_{\text{max}} \mu_{\text{min}}^2 \\ 0 & \frac{1}{j_k} \mu_{\text{max}} \mu_{\text{min}}^2 \end{pmatrix}, \quad E_{bj} = \begin{pmatrix} 0 & 0 & \frac{1}{j_k} \mu_{\text{max}} \mu_{\text{min}}^2 \\ 0 & \frac{1}{j_k} \mu_{\text{max}} \mu_{\text{min}}^2 \\ 0 & \frac{1}{j_k} \mu_{\text{max}} \mu_{\text{min}}^2 \end{pmatrix} \]

Using theorem 1 with the parameters: \( I_1 = 0.12\text{kgm}^2, \ I_2 = 0.10\text{kgm}^2, \ I_3 = 6\text{kgm}^2, \ I_{4\text{max}} = 188\text{kgm}^2, \ I_{4\text{min}} = 138\text{kgm}^2, \)

\( K_1 = 160\text{Nm/rad}, \ K_2 = 16300\text{Nm/rad}, \ C_1 = 40\text{Nms/rad}, \)

\( C_2 = 600\text{Nms/rad}, \ i_0 = 3.07, \ i_1 = 3.14, \ i_2 = 1.98, \ \mu_{\text{s max}} = 0.88, \ \mu_{\text{s min}} = 0.72, \ \mu_{\text{c max}} = 0.66, \ \mu_{\text{c min}} = 0.54. \) In the case of an upshift from 1st to 2nd, we have a solution with \( \gamma = 0.08, \)

\( \tau_u = 270.9, \ \tau_p = 1038.7, \) the PDC gain are

\[ F_1 = 10^3 \begin{pmatrix} 1.6096 & -0.0515 & 2.7225 & 0.0288 \\ 5.3884 & -0.0915 & -5.7980 & -0.2124 \end{pmatrix} \]

\[ F_2 = 10^3 \begin{pmatrix} 1.5706 & -0.0503 & 2.2661 & 0.0255 \\ 6.1356 & -0.1285 & -6.7124 & -0.2643 \end{pmatrix} \]

In the case of a downshift from 2nd to 1st, we have a solution with \( \gamma = 0.08, \) \( \tau_u = 270.9, \) \( \tau_p = 1038.7, \) the PDC gain are

\[ F_1 = 10^3 \begin{pmatrix} 1.0854 & -0.0589 & 1.6535 & 0.0291 \\ -2.5081 & 0.0438 & 1.9481 & 0.2066 \end{pmatrix} \]

\[ F_2 = 10^3 \begin{pmatrix} 1.0712 & -0.0571 & 1.1712 & 0.0261 \\ -2.4130 & 0.0707 & 2.2486 & 0.2561 \end{pmatrix} \]

Different tests with parametric variations have been realized. In the following, we show the simulation results with \( I_4 = 151.5\text{kgm}^2 \) (uncertainty 5%), \( \mu_s = 0.81 \) (uncertainty 1.4%), and \( \mu_c = 0.66 \) (uncertainty 10%). Figure 4 shows the speeds during the gearshift. The 1st – 2nd gear upshift starts at 6.23 seconds, engine speed near 480rad/s and the gearshift time is 0.8 second. The 2nd – 1st gear downshift starts at 8 seconds, engine speed near 205rad/s and again, the shift time is 0.8 seconds. Figure 5 shows the dual clutch torques \( T_{r1}, T_{r2} \) and the output torque \( T_{out} \) during the 1st – 2nd gear upshift. In the torque phase, normal force applied on the off-going clutch \( F_{n1} \) is decreased by the control strategy to release the off-going clutch. When the off-going clutch starts to slip, the normal force applied on the on-coming clutch increases to synchronize the next gear. The vehicle jerk during the gearshift is shown in Figures 6. The jerk is acceptable in our simulations with the shift time 0.8 seconds, because the jerk is less than 10m/s^3 [11]. The shift time is upgraded compared to [7], Figure 7 show the behavior of engine during the gearshift. At the beginning of 1st – 2nd gear upshift the throttle position increases, then decreases to synchronize the next gear. The time-varying engine torque during the gearshift show in Figure 8.

V. CONCLUSION

A nonlinear model of the powertrain has been proposed based on the architecture of a dual clutch transmission. From the latter, a PDC control law based on uncertain TS model has been developed to ensure robustness with respect to parametric uncertainties (vehicle mass and coefficient of friction) and input disturbances (driver behavior, road conditions, wind,...). The results of numerical simulations applied in the gearshift case showed the efficiency of the proposed control law. The vehicle jerk is acceptable while limiting shift time at 0.8 seconds.
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