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Hybrid Automata of an Integrated Motor-Transmission Powertrain for Automatic Gear Shift

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Abstract—**This paper presents a scheme of hybrid modeling of an integrated motor-transmission powertrain, which is increasingly applied to electric vehicles. The shift process of this system is special due to the absence of clutch, and the mode switch of the propulsion motor increases hybrid characteristics. After the introduction of a definition of hybrid automaton and the analysis of shift process, a motor automaton and a gearbox automaton are developed. Properties under different states and switch conditions are presented in detail. Then a completed integrated powertrain automaton is proposed to describe all possible shift actions. Simulation models are developed to verify the effectiveness of the proposed modeling scheme. The results show that all the continuous states and discrete states as well as transients are fully presented, and thus the shift process is clearly reproduced, which is very useful for the design of controllers to achieve a better vehicle performance.**

I. Introduction

SEVERAL powertrain configurations have been proposed to achieve required performance with the rapid development of electric vehicles. For a plug-in or a pure electric vehicle which primarily depends on electrically driven system, a powertrain combining a motor and a multi-gearbox into an integrated system is a better choice, because it can make good use of effective operating range of motor while reducing the cost and enhancing the system efficiency [1]-[2]. This type of driveline is inherently hybrid, consisting of continuous state values such as motor and wheel speeds, and discrete state values like gear ratios. Furthermore, there is a switch between different modes of motor control. At the same time, phase changes from neutral to synchronization during gear shift also increase hybrid dynamics. Therefore, it is suitable to apply hybrid dynamic models to describe this system [3]-[6]. Moreover, a precise model that can clearly describe the complex and nonlinear property of the system is necessary for the controller design. Hybrid automaton is a useful tool to achieve the goal [7]-[10].

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As a typical example, the gear-box or manual transmission is always discussed to analyze hybrid property [4],[11], but only the discrete gear position is focused in the simplified system. It cannot explain clearly the whole process of gear shift and activities of the related components. The authors in [12]-[13] take the driveline into consideration, but mainly emphasize the complex engine models. A mass-spring-damper model of driveline is described in [14] and dynamics about clutch and gearbox actuators are presented. However, the proposed system in this paper is a non-clutch integrated one and the motor plays a role of "virtual clutch". Thus the system performs different activities and the property should be reconsidered.

In this work, hybrid models of integrated motor-transmission powertrain are proposed by the means of automata. The mode switch of motor, the change of gear ratio and the phase transition of the system are presented for the purpose of describing the shift dynamics in detail and then enhancing the performances of vehicles.

The remaining sections are organized as follows: Section II analyzes the system property and shift process. A hybrid automaton definition is introduced and then applied to the system. The detailed hybrid dynamics of the motor, the gearbox, and then the integrated powertrain are presented in Section III. In Section IV, simulations based on MATLAB Simulink/Stateflow models are performed. The results demonstrate the effectiveness of the proposed modeling scheme. Finally, the conclusion is given.

II. Description of Powertrain Model

Before the analysis of the system, a definition of hybrid automaton is given as follows [15]-[16].

$$H = (X, Q, U, Y, f, h, E, G, Inv, Init) \quad (1)$$

where X is the continuous state space; Q is the finite set of the discrete states or locations; U is the set of continuous and discrete-valued control inputs; Y is the set of continuous and discrete-valued outputs; f is the vector field of the continuous dynamics; h is the output map; E is the set of edges; G is the guard condition; Inv is the invariant set of the continuous dynamics at each of location; $Init$ is the set of initial conditions for discrete and continuous state variables.

A reasonable definition of the trajectories of a hybrid automaton can be formulated as follows. A continuous trajectory (x, q, u, δ) associated with a location q consists of a nonnegative time δ (the duration of the continuous trajectory), a piecewise continuous function u: $[0, \delta] \rightarrow U$, and a continuous and piecewise differentiable function x: $[0, \delta] \rightarrow X$ such that $\dot{x}(t) = f(x(t), q, u(t))$ $((q_0,x_0) \in Init)$ with $x(t) \in Inv(q)$ for all $t \in (0, \delta)$ and output $y(t)=h(x(t),q,u(t))$. A trajectory of the hybrid automaton is a sequence of continuous trajectories: $(x_0, q_0, u_0, \delta_0) \rightarrow (x_1, q_1, u_1, \delta_1) \rightarrow (x_2, q_2, u_2, \delta_2) \rightarrow \ldots$, such that at the event times $t_0=\delta_0$, $t_1=\delta_0+\delta_1$, $t_2=\delta_0+\delta_1+\delta_2$, …, the following inclusion holds for the discrete transitions: $(x_j(t_j),u(t_j)) \in G(q_j,q_{j+1})$, $(q_j,q_{j+1}) \in E$ for all $j=0,1,2,\ldots$.

The integrated motor-transmission is a hybrid system which is suitable to apply for the automaton model. It is composed by a motor, a multi-gearbox, a final drive, drive shafts and wheels, as shown in Fig.1.

During the gear shift, we can divide the whole process into five phases. The first phase is torque reduction. As the system receives the shift command, the motor begins to decrease the torque output, which is similar with the clutch disengagement. Then gears are disengaged when the actual torque becomes zero and the system turns into second phase. The motor and the gearbox are decoupled and no torque is transmitted. In order to speed up the shift process and reduce the synchronization wear, the third phase is pre-synchronization of the motor, which is actually the speed regulation. The desired speed is the product of wheel speed and objective gear ratio. When the speed error is lower than the given value, synchronization starts and the system switches to the fourth phase. It mainly depends on the friction force of synchronizers. When the new gears are engaged and the motor and the gearbox become coupled again, the system enters into the last phase, the shift process completes and the vehicle goes back to power driven. It is noted that the traditional clutch engagement is replaced by the gradually increasing motor torque control.

Based on the analysis above, the powertrain can be simplified to a combination of inertia systems, as shown in Fig.2.

The differential equations are derived as follows.

$$\begin{aligned} J_m\dot{\omega}_m &= T_m - T_{gm}(x,q,u) - c_m\omega_m \\ J_g\dot{\omega}_g &= T_{gs}(x,q,u) - T_s / i_0 - c_g\omega_g \\ J_w\dot{\omega}_w &= T_s - T_L - c_w\omega_w \\ \dot{\theta}_s &= \omega_g / i_0 - \omega_w \\ T_s &= k_s\theta_s + c_s(\omega_g / i_0 - \omega_w) \end{aligned} \quad (2)$$

where, J_m is the equivalent inertia of the motor and gearbox input shaft; J_g is the equivalent inertia of the gearbox output shaft; J_w is the equivalent inertia of wheels and the vehicle; c_m is the equivalent damping coefficient of the motor and gearbox input shaft; c_g is the equivalent damping coefficient of the gearbox output shaft; c_w is the equivalent damping coefficient of wheels and the vehicle; k_s is the drive shaft stiffness coefficient; c_s is the drive shaft damping coefficient; θ_s is the torsional angle; T_m is the motor torque; T_{gm} is the torque transmitted by gearbox input shaft; T_{gs} is the torque transmitted by gearbox output shaft; T_s is the torque transmitted by drive shaft; T_L is the load; ω_m is the motor speed; ω_g is the speed of gearbox output shaft; ω_w is the wheel speed; and i_0 is the final drive ratio.

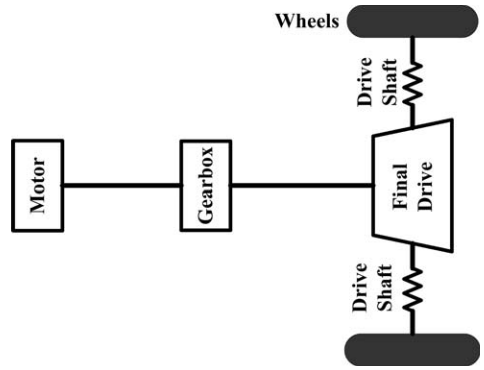


Fig. 1. Integrated motor-transmission powertrain.

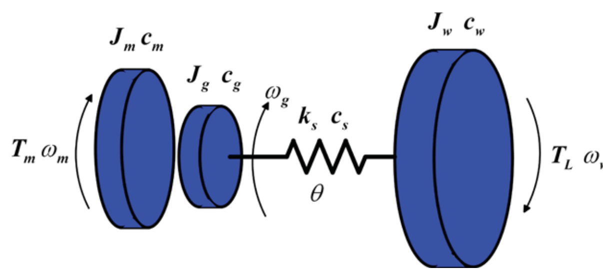


Fig. 2. Simplified inertia systems.

According to the definition of automaton, continuous state $x=\{\omega_m, \omega_g, \omega_w, \theta_s\}^T \in R^4$; discrete state $q=\{q_m, q_g, q_p\}^T \in Q_m \times Q_g \times Q_p$ for the motor, the gearbox and the system automata respectively; input $u=\{T_m, T_{lim}, T_{syn}, T_L, i_g^*\}^T \in U$, where T_{lim} is the limit of the motor torque, T_{syn} is the synchronization friction torque, i_g^* is the objective gear ratio. The hybrid dynamics is presented in the following section.

III. Hybrid Modeling of Integrated Powertrain

A. Motor Model

The activities of the motor during the shift are torque reduction, speed regulation and torque recover. The flow diagram of motor automaton is shown in Fig.3. Discrete state is q_m and the set of discrete states

$$Q_m = \{m_{eng}, m_{T0}, m_{spd}\} \quad (3)$$

where m_{eng} represents the coupled motor-transmission with torque output state; m_{T0} is the coupled motor-transmission without torque output state; and m_{spd} reflects the decoupled system with motor speed regulation state.

Discrete input T_m has different values as follows.

$$\begin{aligned} T_m &= T_m^* \quad && if \quad q_m = m_{eng} \\ T_m &= 0 \quad && if \quad q_m = m_{T0} \\ T_m &= \pm T_{\lim} \quad && if \quad q_m = m_{spd} \end{aligned} \quad (4)$$

where T_m^* is the objective torque which comes from the gas pedal.

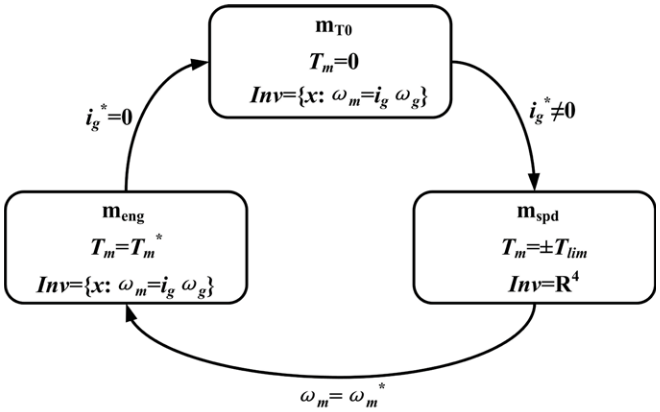


Fig. 3. Motor automaton.

The invariant sets

$$Inv(m_{eng}) = \{x : \omega_m \equiv i_g \omega_g\}$$
$$Inv(m_{T0}) = \{x : \omega_m \equiv i_g \omega_g\} \tag{5}$$
$$Inv(m_{spd}) = R^4$$

where i_g is the gear ratio.

The invariant sets are the same in the states of m_{eng} and m_{T0} because the motor and the gearbox are connected and the current gear ratio still holds.

The set of edges

$$E_m = \{(m_{eng}, m_{T0}), (m_{T0}, m_{spd}), (m_{spd}, m_{eng})\} \tag{6}$$

Guard conditions are presented as a function of i_g^*, ω_m and ω_m^*.

$$G_m(m_{eng}, m_{T0}) = \{i_g^* = 0\}$$
$$G_m(m_{T0}, m_{spd}) = \{i_g^* \neq 0\} \tag{7}$$
$$G_m(m_{T0}, m_{eng}) = \{\omega_m = \omega_m^*\}$$

where i_g^*=0 means the shift starts, while i_g^*≠0 represents the objective gear is determined and it is still in the process of shift. In order to reduce the speed error between the input and output gearbox shaft with the new gear ratio, ω_m^* is derived as follows.

$$\omega_m^* = \omega_g i_g^* + \Delta\omega \tag{8}$$

where, $\Delta\omega$ is the required speed error.

B. Gearbox Model

Gear engagement and disengagement are the main motions of the gearbox during the shift and the gearbox automaton scheme is shown in Fig.4.

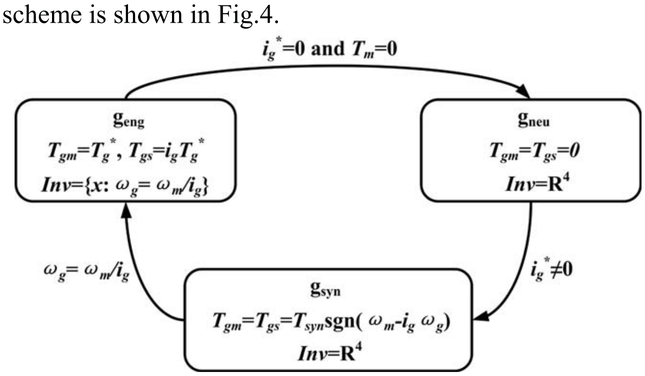


Fig. 4. Gearbox automaton.

Discrete state is q_g and the set of discrete states

$$Q_g = \{g_{eng}, g_{neu}, g_{syn}\} \tag{9}$$

where g_{eng} represents the coupled motor-transmission with a certain gear ratio (i_g≠0); g_{neu} is the gear disengagement state; and g_{syn} describes the synchronization state.

The torques of the input shaft and output shaft varies with the change of states.

$$\begin{aligned} &T_{gm} = T_g^*, T_{gs} = i_g T_g^* && if \quad q_g = g_{eng} \\ &T_{gm} = T_{gs} = 0 && if \quad q_g = g_{neu} \\ &T_{gm} = T_{gs} = T_{syn}\,\mathrm{sgn}(\omega_m - i_g\omega_g) && if \quad q_g = g_{syn} \end{aligned} \tag{10}$$

At the state of q_g=g_{eng}, the gearbox output speed ω_g=ω_m/i_g, and then $\dot{\omega}_g = \dot{\omega}_m / i_g$, combining with (2), the objective torque transmitted by the gearbox input shaft T_g^* is derived as follows [14].

$$T_g^* = \frac{J_g(T_m - c_m\omega_m) + i_g J_m T_s}{J_g + i_g^2 J_m} \tag{11}$$

The invariant sets are given by

$$Inv(g_{eng}) = \{x : \omega_g \equiv \omega_m / i_g\}$$
$$Inv(g_{neu}) = R^4 \tag{12}$$
$$Inv(g_{syn}) = R^4$$

The set of edges is expressed as

$$E_g = \{(g_{eng}, g_{neu}), (g_{neu}, g_{syn}), (g_{syn}, g_{eng})\} \tag{13}$$

Guard conditions are described as a function of i_g^*, i_g, ω_m, ω_g and T_m.

$$G_g(g_{eng}, g_{neu}) = \{(i_g^* = 0) and (T_m = 0)\}$$
$$G_g(g_{neu}, g_{syn}) = \{i_g^* \neq 0\} \tag{14}$$
$$G_g(g_{syn}, g_{eng}) = \{\omega_g = \omega_m / i_g\}$$

It is noted that the discrete state g_{eng} can be switched into g_{neu} until both i_g^*=0 and T_m=0 are satisfied. In view of physical structure, it is hard for the shift fork to disengage gears when there is a force pushing on them.

C. Integrated Powertrain Model

It is reasonable to combine the above component models into a whole hybrid automaton in consideration that the shift is a sequential process. First of all, another set of discrete states should be introduced.

$$Q_p = \{p_1, p_2, p_0\} \tag{15}$$

Discrete state variable $q_p \in Q_p$ represents different gear position. A two-speed automatic transmission is discussed in this work, and thus p_1 is the first gear engagement with gear ratio i_{g1}, p_2 is the second gear engagement with gear ratio i_{g2}, p_0 means the shift state.

The finite set of edges is defined as

$$E_p = \{(p_1, p_0), (p_0, p_1), (p_2, p_0), (p_0, p_2)\} \tag{16}$$

Then the automaton of integrated powertrain is shown in Fig.5.

$S_{i,j}$ (i,j=0,1,2) is a finite set of symbols which serve to label the edges in Fig.5. Generally, automatic gear shift follows an

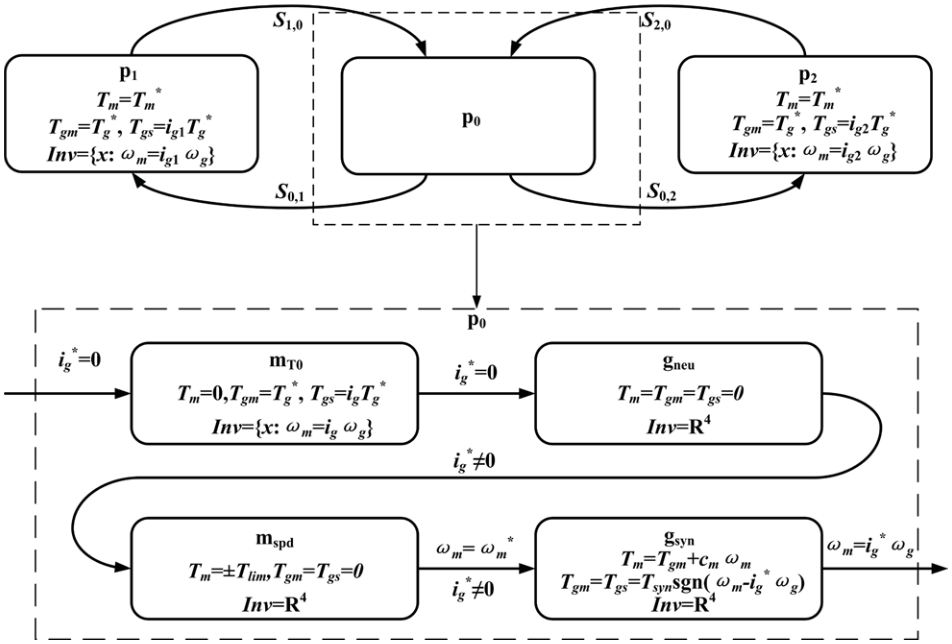


Fig. 5. Integrated powertrain automaton.

algorithm that is from low gear to high gear, i.e. $S_{1,0} \rightarrow S_{0,2}$, or high gear to low gear, i.e. $S_{2,0} \rightarrow S_{0,1}$.

According to the previous analysis of components, the discrete state p_0 includes four sub-discrete states such that the set of discrete states of the system can be extended into the following form.

$$Q_I = \{p_1, p_2, m_{T0}, g_{neu}, m_{spd}, g_{syn}\} \tag{17}$$

The control input T_m and the internal variables T_{gm} and T_{gs} are given by

$$\begin{array}{ll} T_m = T_m^*, T_{gm} = T_{gm}^*, T_{gs} = i_{gi}T_g^* & \text{if} \quad q_I = p_i, i = 1,2 \\ T_m = 0, T_{gm} = T_{gm}^*, T_{gs} = i_{gi}T_g^* & \text{if} \quad q_I = m_{T0}, i = 1,2 \\ T_m = T_{gm} = T_{gs} = 0 & \text{if} \quad q_I = g_{neu} \\ T_m = \pm T_{lim}, T_{gm} = T_{gs} = 0 & \text{if} \quad q_I = m_{spd} \\ T_m = T_{gm} + c_m\omega_m, T_{gm} = T_{gs} = T_{syn}\,\mathrm{sgn}(\omega_m - i_g^*\omega_g) & \text{if} \quad q_I = g_{syn} \end{array} \tag{18}$$

The invariant sets are expressed as

$$\begin{array}{l} Inv(p_i) = \{x : \omega_m \equiv i_{gi}\omega_g\}, i = 1,2 \\ Inv(m_{T0}) = \{x : \omega_m \equiv i_g\omega_g\} \end{array} \tag{19}$$

where i_g depends on the state before q_0. And other invariant sets are the whole state space.

The set of edges

$$E_I = \{(p_i, m_{T0}), (m_{T0}, g_{neu}), (g_{neu}, m_{spd}), (m_{spd}, g_{syn}), (g_{syn}, p_j)\}$$

for i,j=1,2 and $i \neq j$ (20)

Guard conditions are described as follows.

$$\begin{array}{l} G_I(p_i, m_{T0}) = \{i_g^* = 0\} \\ G_I(m_{T0}, g_{neu}) = \{i_g^* = 0\} \\ G_I(g_{neu}, m_{spd}) = \{i_g^* \neq 0\} \\ G_I(m_{spd}, g_{syn}) = \{i_g^* \neq 0 \quad \text{and} \quad \omega_m = \omega_m^*\} \\ G_I(g_{syn}, p_j) = \{\omega_m = i_g^*\omega_g\} \end{array}$$

for i,j=1,2 and $i \neq j$ (21)

IV. Simulation Results and Discussion

The simulation models are developed based on MATLAB Simulink/Stateflow. A permanent magnet synchronous motor (PMSM) with direct torque controller (DTC) is applied such that the performance is closer to the actual situation. The parameters of the PMSM are shown in Table I.

TABLE I
PARAMETERS OF PMSM

Parameter	Value
Stator resistance (ohm)	0.05
Stator inductance (mH)	0.635
Permanent magnet flux linkage (Wb)	0.192
Rotor Inertia (kg·m^2)	0.11
Pairs of poles	4
Rated torque (N·m)	111
Rated speed (rpm)	3000
DC Voltage (volt)	560

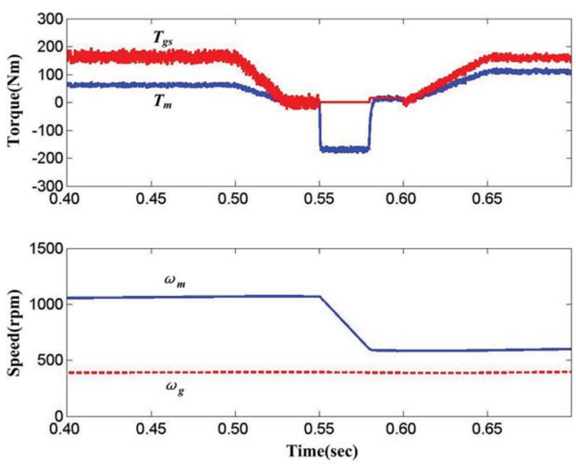
And the parameters of driveline are shown in Table II.

TABLE II
PARAMETERS OF DRIVELINE

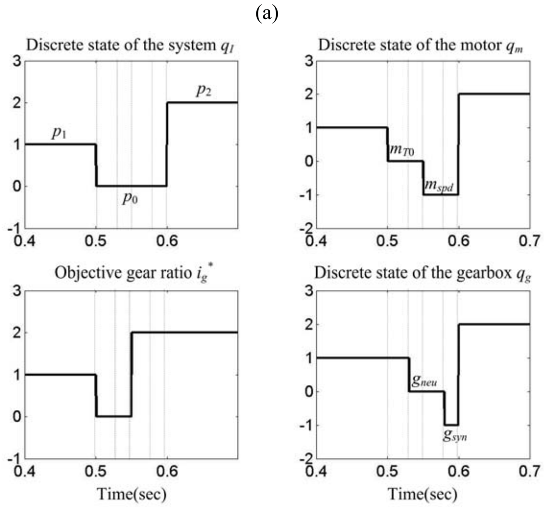
Parameter	Value
J_m (kg·m^2)	0.1145
J_g (kg·m^2)	0.04
J_w (kg·m^2)	305
c_m (N·m/(rad·s^{-1}))	0.032
c_g (N·m/(rad·s^{-1}))	0.087
c_w (N·m/(rad·s^{-1}))	0.285
c_s (N·m/(rad·s^{-1}))	1.7
k_s (N·m/rad)	7161.97
T_{lim} (N·m)	166
T_{syn}(N·m)	15
(i_{g1}, i_{g2})	(2.722,1.516)
i_{g0}	3.292

The load $T_L = r\ (f{\cdot}m_v g+\rho{\cdot}C_D{\cdot}A{\cdot}u_v^2/2)$, where the rolling resistance coefficient f = 0.015; the wind resistance coefficient C_D = 0.6; the air density ρ = 1.2258 N·s^2·m^{-4}; the frontal area A = 3 m^2; the radius of the wheel r = 0.5 m; the vehicle mass m_v = 2000 kg; g = 9.81 m·s^{-2}; the vehicle velocity $u_v = \omega_w{\cdot}r$.

We assume the shift begins at t=0.5 s, and simulation results of upshift and downshift are shown in Fig.6 and Fig.7. It can be clearly seen that there are five phases during the shift. Taking the upshift in Fig.6 for example, the discrete state q_1 turns into m_{T0} at t=0.5 s, and then the motor torque gradually decreases to zero during the next thirty milliseconds. At t=0.53 s, m_{T0} switches to g_{neu}, the gearbox takes action of disengagement during twenty milliseconds while the motor torque keeps the previous value. And then motor speed regulation (m_{spd}) starts when t=0.55 s, and torque runs in its limit value correspondingly. While the feedback $\omega_m=\omega_g i_g^*+\Delta\omega$ (where $\Delta\omega$=2 rad·s^{-1}), the discrete state g_{syn} replaces m_{spd} such that gears are engaged with the objective gear ratio by the friction torque of the synchronizer. After twenty milliseconds, q_2 becomes the current state and the integrated system performs with the gear ratio of i_{g2}.



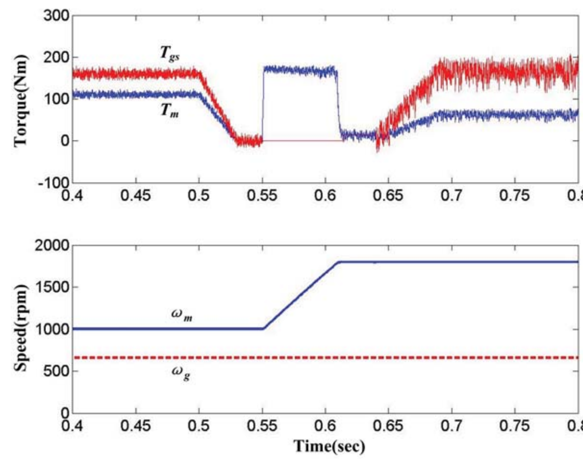
(a)



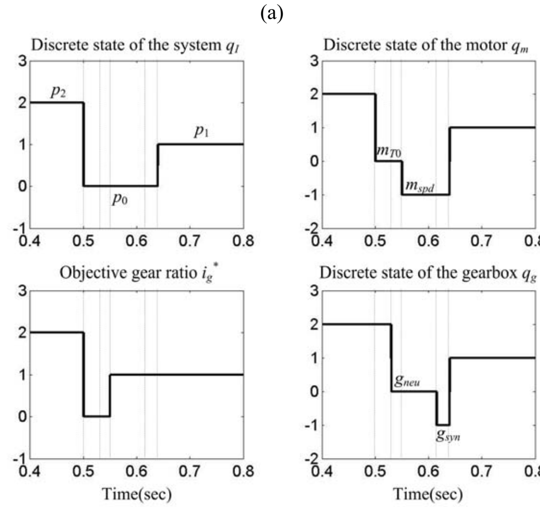
(b)

Fig. 6. Upshift process. (a) Torque and speed responses. (b) The changes of discrete states.

It is seen that the profile of T_{gs}, i.e. the gearbox output torque, is similar with the one in the internal combustion engine driven vehicle [17]-[18]. However, it has a better performance because the values before and after the shift are nearly invariant in order to ensure the same acceleration. In addition, an obvious advantage of such an integrated system is also reflected in Fig.6(a) and Fig.7(a). By the means of pre-synchronization, there is no abrupt impact on the gearbox and the synchronization torque is quite small. And the gearbox output speed, which also represents the vehicle speed, keeps the same value during the shift. That is effective to reduce the power loss. It is noted that the ripples of torques result from the motor control scheme of DTC.



(a)



(b)

Fig. 7. Downshift process. (a) Torque and speed responses. (b) The changes of discrete states.

V. Conclusion

An increasingly popular powertrain for electric vehicles with a structure of integrated motor-transmission is introduced in this paper. It has some advantages such as low cost and high efficiency. But for such a system, the accurate description of hybrid characteristics during the shift without a clutch is still a problem. This paper proposed hybrid models to analyze the system performance in the way of automata.

Based on the detailed analysis of the shift process, a motor automaton and a gearbox automaton were built to describe inherent hybrid property of subsystems. In consideration of shift sequence and interactions between the components, an integrated powertrain automaton was developed to present possible actions during the vehicle operation. The simulation results demonstrated the effectiveness of the proposed model. Both upshift and downshift were exactly reflected by the profiles of torque and speed. At the same time, the changes between different discrete states were also shown in the results. Therefore, this system model exactly demonstrates system characteristics and will be effective and applicable for the future controller design.

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