Coordinated Control Algorithm of a Dual Motor for an Electric Variable Transmission Hybrid System

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ABSTRACT With the successful application of the electric variable transmission hybrid system, the dynamic quality of electric variable transmission (EVT) has received significant attention. The dual motor integrated by the EVT plays a vital role in riding comfort during mode shifting. Therefore, it is of great significance to study the coordinated control algorithm of the dual motor for improving driving comfort. However, current research on single motor torque compensation indicates that the poor accuracy of the compensation torque affects the coordinated control performance. In this paper, a dual motor coordinated control algorithm for the EVT is presented to solve the problem of shock in the mode transition, which is from the electric starting mode to the engine starting mode. First, the lever analogy is used to analyze the cause of the shock during this mode shifting. Then, the coordinated control algorithm of the dual motor is designed, in which the first motor MG1 adopts compounded feedforward and feedback control and the second motor MG2 adopts the torque compensation based on the engine torque estimation algorithm. Finally, the EVT system model and its dynamic coordination control algorithm model are built in the Simulink environment. The simulation results demonstrate that the presented control algorithm markedly reduces the shock of the mode shifting process. When applied in an actual driving cycle, the coordinated control algorithm proposed in this paper performs well, thus improving the dynamic quality of the EVT system.

INDEX TERMS Electric variable transmission, hybrid system, dual motor, coordinated control algorithm.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>a</td>
<td>The vehicle acceleration.</td>
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<tr>
<td>A</td>
<td>The frontal area.</td>
</tr>
<tr>
<td>Cp</td>
<td>The air drag coefficient.</td>
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<tr>
<td>F₁</td>
<td>The internal force of the front planetary row.</td>
</tr>
<tr>
<td>F₂</td>
<td>The internal force of the rear planetary row.</td>
</tr>
<tr>
<td>fᵣ</td>
<td>The rolling resistance coefficient.</td>
</tr>
<tr>
<td>Iₘ₁</td>
<td>The inertia of MG1.</td>
</tr>
<tr>
<td>Iₘ₂</td>
<td>The inertia of MG2.</td>
</tr>
<tr>
<td>Iₛ₁</td>
<td>The inertia of the front sun gear.</td>
</tr>
<tr>
<td>Iₛ₂</td>
<td>The inertia of the rear sun gear.</td>
</tr>
<tr>
<td>Iₖₗ</td>
<td>The inertia of the front planetary carrier.</td>
</tr>
<tr>
<td>Iₖᵢ</td>
<td>The inertia of the rear planetary carrier.</td>
</tr>
<tr>
<td>Iₑ</td>
<td>The inertia of the engine.</td>
</tr>
<tr>
<td>Iᵣ₁</td>
<td>The inertia of the front ring gear.</td>
</tr>
<tr>
<td>i₀</td>
<td>The final drive ratio.</td>
</tr>
<tr>
<td>m</td>
<td>The vehicle mass.</td>
</tr>
<tr>
<td>nᵣₜ</td>
<td>Speed of the final drive.</td>
</tr>
<tr>
<td>nᵣ₂</td>
<td>Speed of the rear planetary carrier.</td>
</tr>
<tr>
<td>nₘ₁</td>
<td>Speed of MG1.</td>
</tr>
<tr>
<td>R₁</td>
<td>Radius of the front ring gear.</td>
</tr>
<tr>
<td>R₂</td>
<td>Radius of the rear ring gear.</td>
</tr>
<tr>
<td>Rᵣ</td>
<td>Radius of the wheel.</td>
</tr>
<tr>
<td>S₁</td>
<td>Radius of the front sun gear.</td>
</tr>
<tr>
<td>S₂</td>
<td>Radius of the rear sun gear.</td>
</tr>
<tr>
<td>T</td>
<td>The sample times.</td>
</tr>
<tr>
<td>Tₑ</td>
<td>Torque of the engine.</td>
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</table>
The longitude degree of shock.
As this approach depends on many calculations, it is difficult to apply to vehicle control. A sliding-mode observer method that relies on a large number of experimental data [32], [33]. However, the accuracy of the engine torque estimation is not high. To alleviate this shortcoming, this paper explores a method in which motor MG1 adopts compounded feedforward and feedback control. This ensures the accuracy of the engine target speed control and lays the foundation for the engine torque estimation. Additionally, the second motor MG2 adopts torque compensation control based on an engine torque estimation algorithm.

In comparison with the existing studies, this work makes two distinctive contributions:

1. Based on Lever Analogy, the root cause of impact in the mode-switching process is discussed and determined.
2. An innovative method is explored and practiced, in which motor MG1 adopts compounded feedforward and feedback control and motor MG2 adopts the torque compensation method based on an engine torque estimation algorithm. This method effectively solves the dynamic shock problem of the mode-shifting process in the electric variable transmission, which is a complicated nonlinear system integrated with dual motors.

The arrangement of this paper is given as follows: Section 2 introduces configuration characteristics of the electric variable transmission hybrid system. In Section 3, the lever analogy is used to analyze the cause of the shock during mode shifting. Section 4 establishes the coordinated control algorithm of the dual motor. Section 5 validates the coordinated control algorithm of the dual motor. Section 6 presents the conclusions.

II. CONFIGURATION OF EVT HYBRID SYSTEM

The configuration of the EVT hybrid system studied in this paper is shown in Fig. 1. The engine crankshaft connects to the planetary carrier of the front planetary row. The motor MG1 connects to the sun gear of the front planetary row. Power outputs from the ring gear of the front planetary row. The ring gear of the rear planetary row is locked. The motor MG2 connects to the sun gear of the rear planetary row. The mechanical power is coupled to the power output shaft through the rear planetary row and ultimately passed to the wheel.

The function of MG1 in this system is to start and adjust the operating point of engine, and in most cases, it works as a generator. The function of MG2 is to compensate the engine torque output to the final drive and to recover the braking energy during braking. In most cases, it operates in the driving state.

Due to the different operating characteristics and functions of the two motors under different operating modes, the operating characteristics of the system and dual motor need to be analyzed according to the different modes to study the double motor coordination control and solve the shock problem during the mode-switching process.

III. WORKING CHARACTERISTICS OF EVT HYBRID SYSTEM

There are many methods to analyze the working characteristics of the electric variable transmission hybrid system. Lever Analogy is not only simple but can also intuitively see the movement relationship of each part [34], [35]. The continuous variation in the single mode and the mode shifting can be easily analyzed.

When other modes switch with the EVT mode, the engine needs to be relaunched, and the mode-switching process can
result in system shock. When the engine shuts down, it indicates that the driving force has decreased. The shock of the engine shutdown is much smaller than that of engine startup. Therefore, in order to analyze the cause of shock in the mode switching, Lever Analogy is used to study the relationship between the torque and speed of power sources in two kinds of typical modes, thus laying the foundation for research into the development of the double motor control coordination algorithm.

A. ELECTRIC STARTING MODE

MG2 provides the torque to start the vehicle in the electric starting mode. The level model is shown in Fig. 2. The solid line arrow indicates the amount of torque applied to the planetary mechanism, and the dotted line arrow indicates the magnitude of the speed.

\[ T_{out} = T_{c2} = (1 + k_2)T_{m2} \]  
\[ n_{out} = n_{c2} = n_{m2}/(1 + k_2) \]

According to the above formulas, the speed and torque output of the EVT system in the electric starting mode is determined by the speed and torque of MG2. According to the connecting relationships of the front ring gear, the rear planetary carrier and the final drive, the front ring gear and the rear planetary carrier rotate at the same speed. MG1 has a much smaller inertia than that of the engine, and the engine starting resistance is larger; thus, MG1 moves in the opposite direction with the front ring gear and the engine does not rotate.

B. STARTING ENGINE MODE

When the power demand of the vehicle is larger or the electricity of the battery is lower, the system needs to start the engine to provide power. The engine startup process can be divided into three stages: the start stage of startup, the middle stage of startup and the end stage of startup.

The start stage of engine startup is shown in Fig. 3. \( T_{m1} \) is provided by MG1, and its direction is upward. The engine does not actually work as a power source at this time. The engine torque is the resistance, and its direction is downward. To maintain the balance of the lever, the corresponding torque of the ring gear is \( T_{R1} \), and the direction is upward.

The lever model in Fig. 4 shows the moment of engine start. Considering that the engine inertia is much smaller than the vehicle inertia, the speed of the front ring gear is basically unchanged. In the lever model, the front ring gear can be seen as a fulcrum, raising the engine speed at the front planetary carrier.

At this stage, the speed direction of MG1 is opposite to its torque direction. The MG1 is in the generating state. Compared to the starting stage of engine startup, the speed of MG1 is decreasing and its torque is equivalent to the braking torque. In the process of reducing the speed of MG1, it is actually in the state of absorbing power, and MG2 outputs power.

If the engine is not yet idling when the speed of MG1 is reduced to zero, MG1 must continue to offer the upward torque until the engine reaches the idle state. The lever model at the end of engine startup is shown in Fig. 5.

After the engine has reached the idle state, it starts to output power. The system then enters the EVT mode. The system speed and torque output during the engine starting process
are as follows:

\[ T_{\text{out}} = (1 + k_2)T_{m2} - T_r \quad (9) \]
\[ n_{\text{out}} = n_{c2} = n_{m2}/(1 + k_2) \quad (10) \]

Comparing formulas (7) and (9), the following can be concluded: based on the unchanged MG2 torque, the torque outputted to the final drive is reduced by the size of \( T_{R1} \) in the engine starting process, causing the system shock. Therefore, in the engine startup process, an additional torque is needed to compensate this reduced torque to ensure smooth mode shifting. According to above analysis, the accuracy of the motor torque compensation depends on the coordination between the dual motors MG1 and MG2, which requires a more advanced and reasonable control algorithm.

**IV. COORDINATED CONTROL ALGORITHM OF DUAL MOTORS**

Based on the working characteristics of the system, the target engine speed must be adjusted by MG1 and the torque must be adjusted by MG2 to meet the load torque at the wheel. The paper proposes that MG1 adopts compounded feedforward and feedback control and MG2 adopts the torque compensation based on the engine torque estimation algorithm. The logic of the dual motor coordinated control algorithm is shown in Fig. 6.

![Figure 6](image_url)

**FIGURE 6.** The logic of the dual motor coordinated control algorithm.

**A. THE MG1 ADOPTS FEEDFORWARD AND FEEDBACK CONTROL**

The feedforward control strategy does not take account into the error caused by the MG1 speed, which may even lead to a large deviation in the MG1 demand torque. However, the feedback control can significantly improve the control precision. Therefore, the paper is based on the feedback control of the PI algorithm to modify the MG1 demand torque. As a result, a compounded feedforward and feedback controller is designed. The control method is shown in Fig. 7.

According to the engine and MG1 coupling in the front planetary row, the forward torque of motor MG1 can be determined from the target engine torque, as follows:

\[ T_{m1,f} = T_{e,\text{tar}}/(1 + k_1) \quad (11) \]

The output torque of the MG1 feedback control can be obtained from the PI controller. The input of the PI controller is the difference of the target speed and the actual speed of MG1. Therefore, in the engine startup process, an additional torque is needed to ensure smooth mode shifting. According to above analysis, the accuracy of the motor torque compensation depends on the coordination between the dual motors MG1 and MG2, which requires a more advanced and reasonable control algorithm.

The output torque of the MG1 feedback control can be obtained from the PI controller. The input of the PI controller is the difference of the target speed and the actual speed of MG1. According to the speed relation of the front and rear planetary rows, the following can be obtained:

\[ \omega_{m1} = (1 + k_1)\omega_{c1} - k_1\omega_r \quad (12) \]
\[ (1 + k_2)\omega_{r1} = \omega_{c2} \quad (13) \]

According to (12) and (13), the target speed of motor MG1 can be obtained:

\[ \omega_{m1,\text{tar}} = (1 + k_1)\omega_{c1} - k_1\frac{\omega_{c2}}{1 + k_2} \quad (14) \]

The input signal of the feedback controller is the difference between the target MG1 speed and the actual feedback MG1 speed, as follows:

\[ e(k) = \omega_{m1,\text{tar}} - \omega_{m1} \quad (15) \]

The PID control algorithm of the continuous control system can be expressed as follows:

\[ u(k) = k_p \left[ e(k) + \frac{1}{T_i} \int_0^k e(k)dk + T_d \frac{de(k)}{dk} \right] \quad (16) \]

The discrete PID algorithm can be described as follows:

\[ u(t) = k_p \left\{ e(t) + \frac{T}{T_i} \sum_{j=0}^t e(j) + \frac{T_d}{T} [e(t) - e(t-1)] \right\} \quad (17) \]

Equation (15) is then reformed as follows:

\[ u(t) = u(t-1) + A_e e(t) - B_e e(t-1) + C_e e(t-2) \quad (18) \]

where

\[ A_e = k_p + k \frac{T}{T_i} + k_p \frac{T_d}{T}, \]
\[ B_e = k_p + 2k_p \frac{T_d}{T}, \]
and

\[ C_e = k_p \frac{T_d}{T}. \]

According to the above analysis, the output of the compounded feedforward and feedback controller is the MG1 demand torque, and the relationship is as follows:

\[ T_{m1,\text{req}} = T_{m1,f} + T_{m1,b} \quad (19) \]
B. MG2 ADOPTS THE TORQUE COMPENSATION BASED ON ENGINE TORQUE ESTIMATION ALGORITHM

During the mode shifting from electric starting to engine starting, MG2 adopts the control algorithm of the torque compensation to make up for reduced torque at the final drive.

The MG2 torque compensation control algorithm depends on accurately obtaining the engine output torque. As shown in Fig. 8, a dynamic model of the planetary hybrid system is established. The estimation algorithm of engine torque is established by dynamic analysis.

The dynamic model of EVT hybrid system assumes that the left torque and speed are positive. According to Euler’s laws of motion, the dynamic relationship of the front planetary row can be obtained as follows:

\[ F_1R_1 - T_{e1} = I_1 \dot{\omega}_1 \]  
\[ T_{e1} - F_1S_1 - F_1R_1 = I_1 \dot{\omega}_c \]  
\[ F_1S_1 - T_{s1} = I_1 \dot{\omega}_s \]  
\[ T_e - T_{e1} = I_e \dot{\omega}_e \]  
\[ T_{s1} - T_{m1} = I_{m1} \dot{\omega}_{m1} \]

Because the front planetary carrier is connected to the engine and the sun gear is connected to MG1, Equations (21) and (23) result in the following equation:

\[ T_e - F_1R_1 - F_1S_1 = (I_e + I_1) \dot{\omega}_e \]  
\[ T_{e1} - F_1S_1 - F_1R_1 = I_1 \dot{\omega}_c \]  
\[ F_1S_1 - T_{s1} = I_1 \dot{\omega}_s \]  
\[ T_e - T_{e1} = I_e \dot{\omega}_e \]  
\[ T_{s1} - T_{m1} = I_{m1} \dot{\omega}_{m1} \]

The dynamic equations of the rear planetary row can be similarly obtained:

\[ T_{m2} - F_2S_2 = (I_{m2} + I_{s2}) \dot{\omega}_{m2} \]  
\[ F_2S_2 + F_2R_2 - T_{c2} = I_{c2} \dot{\omega}_{c2} \]

According to (27) and (28), the relationship between the output torque of the rear planetary carrier and the input torque of MG2 is obtained as follows:

\[ T_{c2} = T_{m2}(1 + k_2) - [(I_{m2} + I_{s2})(1 + k_2)^2 - I_{c2}] \dot{\omega}_{c2} \]

\[ T_{m2} = T_{m2}(1 + k_2) - [(I_{m2} + I_{s2})(1 + k_2)^2 - I_{c2}] \dot{\omega}_{c2} \]  
\[ T_{m1} = T_{m1}(1 + k_1) - [(I_{m1} + I_{s1})(1 + k_1)^2 - I_{c1}] \dot{\omega}_{c1} \]

The output torques of the front ring gear and the rear planetary carrier are coupled and transmitted to the final drive to overcome the driving resistance of the bus.

Considering only the longitude dynamics and neglecting the wheel slip lead to the following equation:

\[ (\dot{\omega}_{r1}/i_0)R_i^2m = (T_c + T_{r1})i_0 - T_{fb} - mgfR_i - 0.5\rho AC_D(\omega_{r1}/i_0)^2R_i^3 \]  
\[ \text{(30)} \]

The MG1 torque and the MG2 torque are accurately estimated in the motor control unit, and the engine speed is obtained in the CAN bus. Meanwhile, C can be calculated according to vehicle speed and brake pressure. Therefore, engine torque can be determined along with the already-known state variables according to (32).

According to the above analysis, the torque compensation amount of MG2 can be determined by estimating the engine torque and the demand torque of wheels. The compensated torque of MG2 can be obtained as follows:

\[ T_{m2, req} = \frac{T_{out} - T_e - \frac{k_1}{(1 + k_1)(1 + k_2)}}{k_2} \]  
\[ \text{(33)} \]

Finally, based on (32) and (33), the compensated torque of MG2 can be obtained (34), as shown at the bottom of the next page.

where \[ T_{veh} = T_{out} \cdot i_0 \], \[ T_{veh} \] denotes the demand torque at the wheel. It is calculated through the interpretation of the driver’s accelerator pedal.
Especially, the engine torque estimation algorithm based on the characteristics of the highly coupled power-split system, taking into consideration the inertia moments and acceleration of each component. It is derived from the nonlinear dynamic model of the power-split system. Thus, avoiding massive computation load, extra sensors, and several calibrations, as well as ensuring good real-time performance.

V. SIMULATION VALIDATION

To comprehensively verify the performance of the coordinated control algorithm of the dual motors from the electric starting mode to the engine starting mode, as well as in the actual cycle driving conditions, the coordinated control algorithm of the dual motors and the EVT system model was built in the MATLAB/Simulink environment [35], as shown in Fig. 9.

A. MODE-SHIFTING PROCESS VERIFICATION

To verify the coordinated control algorithm of the dual motors in the mode shift from the electric starting mode to the engine starting mode, two kinds of control algorithms, including one with coordinated control and one without, were built for a comparative analysis.

For evaluating the shock in mode shifting, the definition of the degree of shock was given as the gradient of the acceleration, as shown in the following relationship:

\[
j = \frac{da}{dt} = \frac{d^2v}{dt^2} = \ddot{\omega}_1/i_0 \cdot R_l
\]  

(35)

Considering that the accelerated pedal signal from the driver affected the mode-switching process in actual operation, the accelerated pedal was permanently set to 80% to simulate a constant acceleration. The simulation results are shown Fig. 10.

It can be seen from Fig. 10(a) that mode 1 (electric mode) shifts to mode 2 (EVT mode) at 1 s. After mode shifting, the vehicle speed without coordinated control is lower than the vehicle speed with coordinated control. The dynamic performance of the EVT system is apparently improved by adding coordinated control.

Comparing the torque of the engine and MG1, it takes 0.8 s to start the engine. At about 1.8 s, the torque of the engine is equal to that of MG1. Therefore, there is no influence on engine start time after adding coordinated control.

From the final drive torque curve, it can be seen that the torque remains constant under coordinated control while declining markedly without coordinated control.

As the MG2 torque diagram shows, at the time of mode switching, the MG2 output torque decreases instantaneously without adding coordinated control because the dynamic process of starting the engine is not considered. The engine starts instantly and outputs torque at 1 s. The engine torque increases and the MG2 torque reduces accordingly as the required torque remains constant. After adding coordinated control, the MG2 output torque increases instantaneously when the mode changes. The final drive output torque remains constant as MG2 provides the compensated torque.

Comparing the jerk with and without coordinated control in Fig. 10(b), coordinated control markedly suppresses the shock during mode switching. Table 1 gives the comparison of the degree of impact with and without the coordinated control algorithm.

Through the above comparative analysis, it is concluded that the coordinated control algorithm proposed in this paper can significantly reduce the positive and negative shock. The system jerk is basically eliminated in the mode shifting process from the electric starting mode to the engine

\[
T_{m2_{req}} = \frac{T_{veh}}{i_0(1 + k_2)} - \frac{k_1^2}{(1 + k_2)(i_0 k_1^2 + I_s)} \left[ (1 + 1/k_1)\dot{\omega}_e + \frac{I_i \dot{\omega}_e}{(1 + 1/k_1)} + \frac{I'_m \dot{\omega}_e}{k_1(1 + k_1)} \right] - i_0 T_{m2} - \frac{I'_e}{k_1 I'_m} T_{m1} + C
\]  

(34)
starting mode. The algorithm presented is reasonably practicable and remarkably effective.

**B. SIMULATION VERIFICATION OF CYCLE CONDITION**

To comprehensively verify the effect of the dual motor coordinated control algorithm, driving cycle simulations are carried out. The electric variable transmission hybrid system studied in this paper is applied to the urban bus. Accordingly, the simulated driving cycle is CCBC. In addition, simulation conditions should meet the various modes of vehicle driving and keep the computation complexity and simulation time as low as possible [37]. Therefore, the last part (1231-1304 s) of the CCBC is chosen as the simulation condition. The simulation results are shown in Fig. 11.

From the vehicle speed curve, the velocity follows the target driving cycles well. The mode switching (mode 1, electric operation; mode 2, EVT; mode 3, regenerative braking) is reasonable and relatively infrequent. The following sections describe the detailed stage analysis for each mode.

**FIGURE 10.** Simulation results of the mode shifting from the electric starting mode to the engine starting mode. (a) Change of velocity, mode and power source. (b) Jerk comparison with and without coordination.
1) ELECTRIC STARTING MODE

Fig. 12 shows the simulation result for the electric starting mode for 0-8 s. The required speed continuously increases within 0-5 s as shown in the velocity curve. The signal of the accelerator pedal tends to be constant after rapidly increasing.

At this stage, as shown in the torque curve, the vehicle torque is provided by MG2, and the speed of MG2 is positive. MG1 and the engine are in a closed state. As the engine inertia is much larger than that of MG1, the rotational speed of MG1 is negative and the engine speed tends to zero. MG2 is powered by the battery. As shown in the current curve, the discharge current of the battery is basically the same as the bus current of MG2. At 5-7 s, the required vehicle speed tends to remain unchanged, and the required torque decreases. The output torque of MG2 responds to the accelerator pedal signal and remains constant, after the rapid reduction, to maintain constant speed.

According to the operating characteristics of the electric starting mode, the simulation results are consistent with
the theoretical analysis. The dual motor coordinated control algorithm established in this paper can meet the requirements of the vehicle driving condition in the electric starting mode.

2) FROM ELECTRIC STARTING MODE TO EVT MODE
With demand speed increasing, the EVT system shifts from electric starting mode to EVT mode. The simulation results are shown in Fig. 13.

As shown in the torque curve graph, MG1 provides the torque to start the engine. The MG1 torque is positive and rapidly increases. At that moment, the engine torque is negative as it is not doing work and acts as a resistance. To maintain the speed of the vehicle, the compensated torque is provided by MG2 at this time. When the engine starts and works normally, it enters the EVT mode. The engine torque changes from negative to positive. The torque of motor MG1 changes from positive to negative, and the torque of MG2 is always positive.

From the speed curve graph, the speed of MG2 is always positive, the engine speed gradually increases from zero, and the speed of MG1 changes from negative to positive.
The power variation in the power source can be obtained from the speed and torque curve. During engine starting, the power of MG1 goes from negative to positive. MG1 works initially as a generator and then as a motor. The power of MG2 is always positive, indicating that MG2 has been consuming energy.

From the current curve graph, the current of MG1 goes from positive to negative and finally stays positive. This indicates that MG1 is in the generating and motor states. The current of MG2 is always positive, indicating that it acts as a motor. The above analysis is consistent with the working status of the motor power analysis.

From the curve of engine speed and torque, the engine started before 9.5 s, and the actual operating point of the engine did not reach the target operating point because the dual motors were still in the adjustment stage. After the engine is started, the actual engine speed and torque can follow the target operating point, and the dynamic response process is good, which verifies the accuracy of the dual motor control algorithm.

3) FROM EVT MODE TO ELECTRIC MODE

From 15-20 s, the demand power is smaller, and then the system switches from EVT mode to electric mode. The simulation results are shown in Fig. 14.

At 16-18 s, the required speed is constant, the torque of MG2 and the engine decreases and the EVT mode is switched to the electric mode.

From the mode curve graph, the mode switching is at 17.1 s. At this time, the engine and MG1 do not work and do not provide torque. Therefore, the engine torque is negative, the MG1 torque is zero, and MG2 provides the vehicle drive torque.

After 18 s, the required speed gradually increases. At this moment, the speed of motor MG2 reaches 3000 r / min.
FIGURE 15. Simulation result from EVT mode to RGB mode.

At this stage, the torque of MG2 is limited as a result of working in the constant power range. The torque actually supplied by MG2 gradually decreases as the speed increases. When the torque provided by MG2 cannot meet the demand, the system switches from electric mode to EVT mode. After the engine normally provides power, the engine works on the optimal working curve. The insufficient torque should be made up by MG2. As the torque of MG2 has reached the maximum value at the current speed, however, it cannot make up the extra torque. Furthermore, although there is surplus engine torque at this point, it was controlled at the target torque. This situation leads to the lack of motive power at 20-30 s.

4) FROM EVT MODE TO RGB MODE
From the beginning of 53 s, the demand speed begins to decrease as the vehicle enters the braking condition. The simulation results are shown in Fig. 15.

As the brake pedal signal rapidly increases, the vehicle speed gradually decreases and switches from EVT mode to regenerative braking mode. At that moment, the engine and MG1 are stopped and no longer provide output torque, and the braking torque that is supplied by MG2 recovers the braking energy. As seen from the torque curve, the torque of motor MG2 is negative and the torque of MG1 is zero. Combined with the speed curve, it can be seen that the power of MG2 is negative, indicating that MG2 is in the generating state. This is consistent with the negative value of the MG2 current, which indicates that it is working as a generator.

Through the above simulation analysis of the driving cycle, the dual motor coordination control algorithm established in this paper can assist the EVT system to achieve good dynamic performance, reduce the system jerk and ensure smooth mode shifting. In addition, the algorithm has a good effect on adjusting the engine operating point in the EVT mode, which is consistent with the actual working conditions. Thus, simulation results verify that the control algorithm proposed in this paper is adapted for the actual driving cycle.

VI. CONCLUSION
To solve the problem of shock in the mode transition from the electric starting mode to the engine starting mode, this paper adopts the lever analogy to analyze the cause of the shock during this mode shifting. The coordinated control...
algorithm of the dual motor is designed such that the first motor MG1 adopts a compounded feedforward and feedback control and the second motor MG2 adopts torque compensation based on the engine torque estimation algorithm. Finally, the EVT system model and its dynamic coordination control algorithm model are built in the Simulink environment. The simulation results demonstrate that the presented control algorithm markedly improves the shock during the mode shifting process. When applied in the actual driving cycle, the coordinated control algorithm proposed in this paper performs well, which improves the dynamic quality of the EVT system.

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REFERENCES


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