

Dynamic Coordination Control of HEV Unsteady Operating Mode Switching Based on MPC

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Abstract. Aiming at a hybrid electric vehicle (HEV), the impact problem is studied during the mode switching process under non-steady-state operating conditions. Firstly, the system dynamics model is established, the dynamic characteristics of unsteady conditions are analyzed, the BP neural network engine torque estimation model based on genetic algorithm optimization is established, and the engine torque is estimated online. MG1 torque coordination control and MG2 active compensation control are adopted, a mode switching coordination control strategy based on model prediction is proposed. The MATLAB / Simulink simulation platform is used to verify the effectiveness of the control algorithm. The simulation results show that the coordinated control strategy can effectively achieve the output torque compensation of the drive motor. The engine is actually close to the target torque. Compared with the uncoordinated control, During the speed synchronization phase, the maximum impact of the vehicle was reduced by 60.4%.

1. Introduction

The power split hybrid power system with planetary row power coupling mechanism can achieve dual decoupling of torque and speed at the engine and wheels. It has the advantages of parallel and series hybrid power systems and has good development prospects. The hybrid system represented by Toyota Prius and the universal dual-mode configuration has been widely used because of its advantages in economy and power [1-2]. At the same time, other extended power coupling mechanisms based on the Toyota hybrid system have received widespread attention, and scholars have also studied the mode switching problems of systems with associated coupling mechanisms [3-4]. The main research results about the model switching process are as follows.

The Beck scholar of the Aachen University of Technology in Germany aimed at reducing the degree of impact during clutch engagement, and used the MPC method to solve the problem of smooth transmission of power in the process of switching from pure electric to hybrid drive mode of parallel hybrid vehicles, and verified the robustness of the control system [5].

Scholar Zhang Jun took CVT hybrid electric vehicle as the research object, used the pseudo target speed and pseudo output torque of the engine to calculate the motor speed, and realized the smooth transition between pure electric operating conditions and pure engine operating condition. Simulation experiments were conducted under Matlab / Simulink environment, without considering the engine output, the time lag of clutch coupling, and the mutual compensation of motor torque and engine output [6].



Qin Datong aimed to improve the regenerative braking power and used discrete exhaustive optimization to modify the CVT target speed ratio and braking force distribution, thus proposed a CVT speed ratio optimization control strategy based on speed ratio change limit and motor coordinated control [7].

Yin Chengliang of Shanghai Jiaotong University aimed at the single-motor parallel hybrid vehicle mode switching problem, taking into account the system parameter uncertainty and CAN communication delay, designed a robust controller based on μ synthesis to decouple the engine and the motor [8].

In summary, the current research objects of mode switching control are mostly parallel hybrid vehicles, and there is little research on the mode switching process of hybrid vehicles. In this paper, a hybrid vehicle with a double planetary gear mechanism is selected, and its mode switching process is studied in an unsteady state. The engine torque was estimated online. The motor torque coordination control model was established. A dynamic coordinated control strategy based on model predictive control (MPC) was proposed. This strategy can reduce the impact of the mode switching process and ensure the smoothness and comfort of the hybrid system when switching modes.

2. Dynamic Analysis of Planetary Hybrid System

It can be seen from Figure 1 that the power coupling mechanism system consists of two rows of planetary gears. The system is a compound gear device, including two rows of gear trains. The front row is the power distribution planetary gear mechanism, and the rear row is the motor reduction planetary gear mechanism. The front sun gear is connected to MG1, the front planetary carrier is connected to the engine through a torsional damper, the rear sun gear is connected to MG2, and the rear planetary carrier is fixed. The front and rear rows share a compound ring gear, and the power of the system is finally output to the reduction gear through the compound ring gear to drive the vehicle.

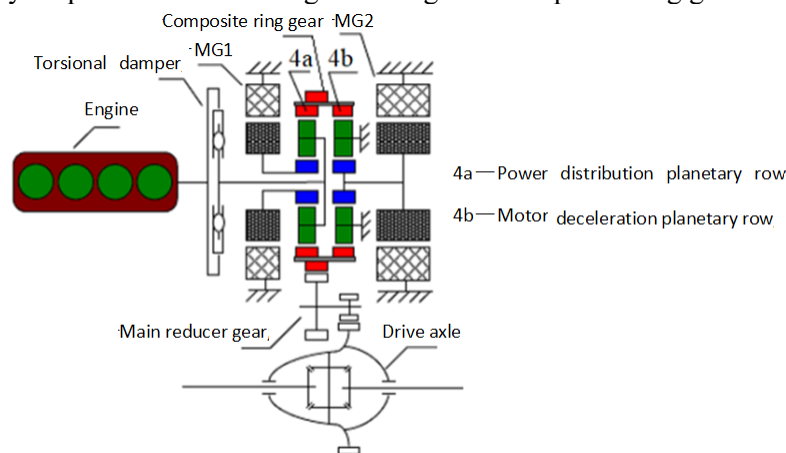


Fig .1 Power coupling system

Ignore the friction loss and rotational inertia of the system itself, The relationship between the basic speed and torque of the planetary gear mechanism is as follows in Equation (1) .

$$\begin{cases} T_{out} = T_e \frac{k_1}{1+k_1} + T_m i_2, i_2 = 1+k_2 \\ \omega_{out} = \frac{\omega_e (1+k_1) - \omega_g}{k_1} = \frac{\omega_m}{i_2} \end{cases} \quad (1)$$

where ω is the speed, T is the rotation, k_1 is the characteristic parameters of the front planetary row, k_2 is the characteristic parameters of rear planetary row, Where out represents the system output shaft, e represents engine, g represents MG1, m represents MG2.

By analyzing the relationship between power source and the planetary gear, the dynamic relationship of the front planetary row can be obtained, as shown in Equation (2) and (3) [9].

$$\begin{bmatrix} I_{c1}+I_e & 0 & 0 \\ 0 & I_{s1}+I_g & 0 \\ 0 & 0 & I_{r1} \end{bmatrix} \begin{bmatrix} \dot{\omega}_{c1} \\ \dot{\omega}_{s1} \\ \dot{\omega}_{r1} \end{bmatrix} = F_1 \begin{bmatrix} -(R_{r1} + R_{s1}) \\ R_{s1} \\ R_{r1} \end{bmatrix} + \begin{bmatrix} T_e \\ T_g \\ -T_{r1} \end{bmatrix} \quad (2)$$

$$\begin{bmatrix} I_e & 0 \\ 0 & I_g \end{bmatrix} \begin{bmatrix} \dot{\omega}_e \\ \dot{\omega}_g \end{bmatrix} = \begin{bmatrix} T_e \\ T_g \end{bmatrix} + \begin{bmatrix} -T_{c1} \\ T_{s1} \end{bmatrix} \quad (3)$$

Where I is the moment of inertia, ω is angular velocity, T is torque, and is R radius, F_1 or F_2 is the internal force of the front planetary row or the rear planetary row. C1 represents the front planetary carriers, C2 represents the rear planetary carriers, R1 represents the front ring gears, R2 represents the rear ring gears, S1 represents the front sun gears, S2 represents the rear sun gears. According to equations (2) and (3), the input of the front planetary row can be obtained by Equation (4).

$$\begin{bmatrix} I_{c1}+I_e & 0 & 0 \\ 0 & I_{s1}+I_g & 0 \\ 0 & 0 & I_{r1} \end{bmatrix} \begin{bmatrix} \dot{\omega}_{c1} \\ \dot{\omega}_{s1} \\ \dot{\omega}_{r1} \end{bmatrix} = F_1 \begin{bmatrix} -(R_{r1} + R_{s1}) \\ R_{s1} \\ R_{r1} \end{bmatrix} + \begin{bmatrix} T_e \\ T_g \\ -T_{r1} \end{bmatrix} \quad (4)$$

Similarly, according to the dynamics and connection relationship of the rear planetary row, the relationship between the input and output of the rear planetary row can be obtained, as shown in Equation (5).

$$\begin{bmatrix} I_m + I_{s2} & 0 \\ 0 & I_{c1} \end{bmatrix} \begin{bmatrix} \dot{\omega}_m \\ \dot{\omega}_{c1} \end{bmatrix} = F_2 \begin{bmatrix} -R_{s2} \\ R_{r2} + R_{s2} \end{bmatrix} + \begin{bmatrix} T_m \\ -T_{c2} \end{bmatrix} \quad (5)$$

The engine outputs torque from the front planetary row, and the motor MG2 outputs torque from the rear planetary row. They are both input to the input of the final reducer and are used to drive the vehicle. Equation (6) is obtained from the vehicle dynamics equation.

$$T_{out} = T_{fb} / i_o + Ma + [Mg(f_r \cos \theta + \sin \theta)]R_t / i_o + 0.5\rho AC_D (\omega_{out} / i_o)^2 R_t^3 / i_o \quad (6)$$

where i_o is the main reduction ratio, M is the mass of the vehicle, R_t is the radius of the wheel, a is the acceleration of the vehicle, T_f is the resistance received by the wheel, T_{fb} is the friction braking torque, f_r is the rolling resistance coefficient, ρ is the air density, A is the area of the wheel upwind, C_D is the air resistance coefficient.

3. Key component models

3.1. Engine torque online estimation

In order to improve the accuracy of on-line engine torque estimation of parallel hybrid vehicles, a BP neural network engine torque estimation method based on genetic algorithm optimization is proposed, and a BP neural network engine torque estimation model based on genetic algorithm optimization is established. After training the experimental data, the dynamic torque estimation of the engine in the full operating range was completed, and the measured value of the engine torque was compared with the estimated value. The result reached the prediction accuracy. In order to realize the dynamic switching control of the parallel hybrid electric vehicle laid the foundation.

3.2. MG1 torque coordinated control

Under the non-steady-state operating conditions, the engine and the motor MG1 jointly output torque. According to the established system dynamics model, considering the physical constraints of the

system components, the amplitude limits of the motor MG1 torque and torque variation can be obtained Value [9]. The specific constraint form is as formula (7).

$$\begin{aligned} T_{\min}(k+i) \leq T(k+i) \leq T_{\max}(k+i), \quad i=1, \dots, u-1 \\ \Delta T_{\min}(k+i) \leq \Delta T_1(k+i) \leq \Delta T_{\max}(k+i), \quad i=1, \dots, u-1 \end{aligned} \quad (7)$$

Considering the above constraints, the optimization problem can be transformed into a quadratic programming problem, as shown in Equation (8).

$$\begin{cases} \min_{\Delta U(k)} \frac{1}{2} \Delta U(k)^T H \Delta U(k) + G(k+1|k)^T \Delta U(k) \\ \text{s.t. } C_u \Delta U(k) \geq b(k+1|k) \end{cases} \quad (8)$$

3.3. Motor MG2 active compensation control

According to equations (1) and (6), the relationship between the motor and the system output torque can be obtained, as shown in Equation (9) [9].

$$\frac{k_1(I_{e1} + I_e)i_o}{1+k_1} \dot{\omega}_e = \left(\frac{k_1 T_e}{1+k_1} + (1+k_2)T_m \right) i_o - T_f \quad (9)$$

By further optimizing the target torque of the motor and the predicted results, an optimized MG2 torque increase can be obtained, as shown in Equation (10).

$$\Delta T_2(k)_o = B_{T_2}^{-1} [\Delta x(k+1) - A_2 \Delta x(k) - B_{d_2} \Delta d_2(k)] \quad (10)$$

Among them, $\Delta T_2(k)_o$ not only meets the driver's demand torque, but also realizes the torque compensation, reduces the impact of mode switching.

4. Mode switching dynamic coordination control strategy

In order to facilitate the design of the controller, the mode switching process is divided into two stages of engine starting and hybrid driving according to the different power source participation, and the dynamic equations under non-steady-state conditions are established. Considering the smoothness of switching, a coordinated control strategy for mode switching based on model prediction is designed. According to the principle of the model predictive controller, the block diagram of the designed mode switching coordination control strategy is shown in Figure 2. According to the feedback of the vehicle control model, the future state of the system is predicted. It is estimated that the motor MG1 torque coordination control and the motor MG2 active compensation control are adopted. In order to facilitate the realization with the computer, this strategy is finally turned into a quadratic programming problem, and the dynamic coordination control of the mode switching process is realized [10].

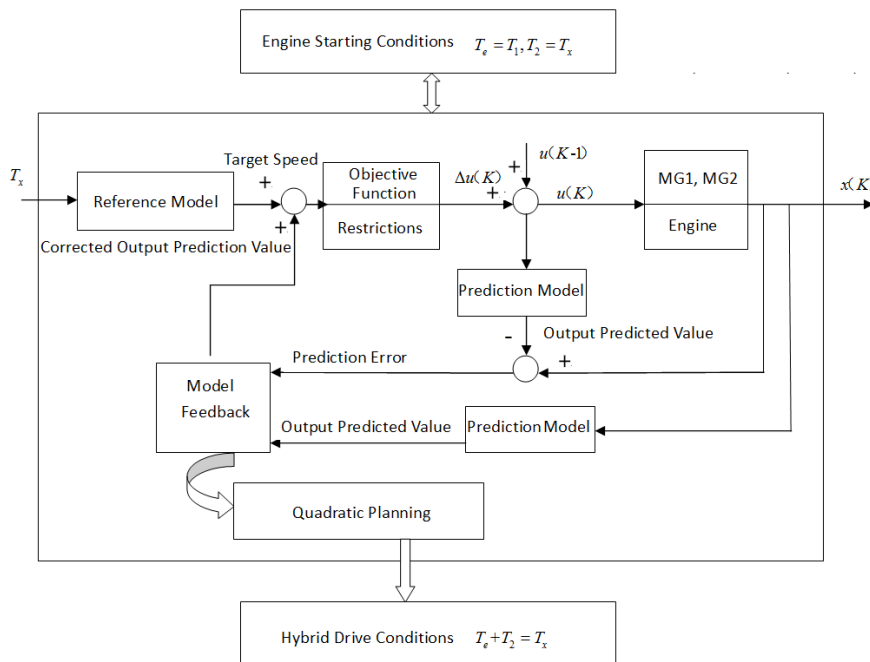


Fig.2 Model predictive control structure diagram

5. Simulation Analysis of Dynamic Coordinated Control

In order to verify the effectiveness of the proposed coordinated control strategy, combined with the established system model, the model predictive controller designed was simulated and verified by using Matlab / Simulink software. Extracted from pure electric to switch to the hybrid drive mode for simulation, the test conditions lasted 3s, the maximum speed is about 35 km / h. Among them, the threshold for triggering mode switching is set around 221.8s. The simulation results are shown in Figure3-5.

It can be seen from Figure3, when the coordinated control strategy is not adopted, the duration of the entire mode switching process is about 1s. The engine torque is not stable, the motor cannot be compensated within the output range, and the torque fluctuation on the drive shaft is obvious. As shown in Figure 4, it can be seen that when the coordinated control strategy is adopted, the mode switching time is about 0.8s, the engine torque is relatively close to the target torque, and there is no large fluctuation and overshoot. At the same time, the motor can compensate the engine torque, and the follow ability is very good. As can be seen in Figure 5, the impact degree of the coordinated control strategy is significantly reduced, the maximum impact degree is 6 m / s³, and the coordinated control strategy causes a greater longitudinal impact on the vehicle. Among them, the maximum impact degree is about 19 m / s³.

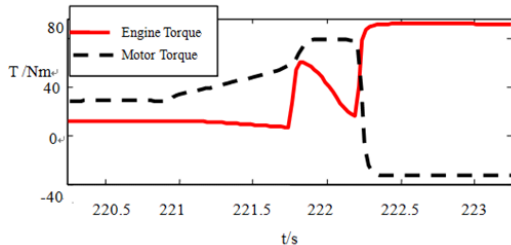


Fig.3 Torque of power source with coordinated control

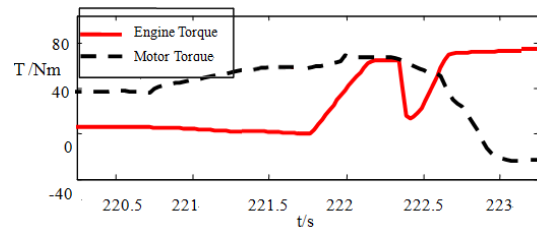


Fig.4 Power source torque without coordinated control

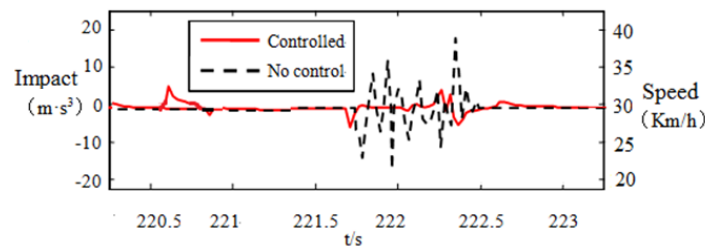


Fig.5 Impact contrast

6. Conclusions

Aiming at the non-steady state switching process of the hybrid power system, this paper carried out a model of the hybrid power system and analyzed the dynamic characteristics of mode switching. A dynamic coordinated control method based on model predictive control was proposed, and simulated experiments were carried out. Simulation results show that the engine torque can approach the target torque compared to the uncoordinated control, the drive motor effectively compensates the torque and has good follow ability, and the impact on the vehicle when the mode is switched is very small.

Acknowledgments

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