Abstract
The plug-in 4WD hybrid electric vehicle researched in this paper has 3 power components, a CVT, and dual clutch, thus it has many work modes and mode transitions. And there are many variables to be taken into consideration for the coordination control of the vehicle. So the control strategy of the vehicle is very complicated and it is difficult to guarantee the fuel economy and drivability at the same time. To solve this problem, this paper takes the dynamic model of the power components and every work mode into consideration. A diver model based on fuzzy PID is built to obtain the torque request coefficient which reflects driver’s intentions and it is used to determine the drive modes preliminarily. A control strategy based on optimal engine torque is adopted, in which the dynamic model for different work modes as well as the torque distribution method is described. For the coordination during mode switches, a method which can be summarized as ‘Preliminary torque distributions, engine dynamic torque estimation, clutch fuzzy-fuzzy PID control and dual motor compensation’ is put forward. For engine dynamic torque estimation, a lookup method based on the experimental data is presented; For clutch control, two fuzzy controllers and a fuzzy PID controller whose aim is to have a more precise output of the oil pressure of the clutch is proposed. Afterwards, the control strategy is validated on the test bench, and the results verify that the power components work cooperatively with each other in every mode and during mode transitions. Moreover, the fuel consumption is reduced by 9.04%, and the jerk is reduced by 40.7%.

Keywords: PHEV, control strategy, dynamic model; engine torque estimation, fuzzy-fuzzy PID control ; HIL bench test

1 Introduction
Plug-in hybrid electric vehicle (PHEV) derives from hybrid electric vehicle, it is a very significant transitional product between the traditional internal combustion engine vehicle and pure electric vehicle (PEV), which plays a very important role in transportation and has a very great potential before the inherent defects of the battery is conquered to guarantee the PEV a longer driving range and a better reliability [1-2]. Plug-in four-wheel-drive hybrid electric vehicle (4WD PHEV) is a PHEV that can be driven by the 2 axles at a proper time, which can provide the PHEV with better power performance and passability. But the fuel economy may not be as good as the single axle driven vehicle, and the control of the vehicle is more complicated. So a trade-off is needed in the architecture and the control to minimize the fuel consumption with the power performance guaranteed [3-4]. With the layout confirmed by the Chery auto company, the problem left to be solved
is the control strategy of the vehicle. There exist 3 power components which can work independently or cooperatively for a 4WD PHEV, whose power system is very complicated. The coordination control involves many subjects and it is an intricate project needs to be improved continually. So, the research into the coordination control of the 4WD PHEV is of great significance.

Coordination control for a hybrid electric vehicle includes the two aspects of the steady state and dynamic state. The coordination control during steady state is the energy management coordination control, which aims at torque distributions and the control strategy based on the logic threshold is widely used. While the coordination control during dynamic state mainly involves the coordination of the power components when mode changes. The coordination during mode switch can be classified into 2 categories; the first is based on the estimation of engine dynamic torque and the compensation of the drive motor; the second is based on hybrid dynamic system and other optimal methods.

Currently, the algorithms for energy management coordination control include the rule-based strategies, the optimization-based strategies and intelligence-based strategies. The objective of the PHEV control is to find out the optimal torque distributions at every instant of time to minimize the fuel consumption with the power performances ensured over a giving driving schedule [5] which can be fully realized through global optimization methods. Dynamic programming and other approximate global optimal algorithms were proposed. But the prior knowledge was needed to get the profile of the roads, which was a big obstacle for the real time employment [6-7]. Although a lot of work has been done to do research into the optimization-based strategies and Intelligence-based strategies and a lot of improvements have been made, there is still a long way to go before putting them into practice for their inherent problems. Therefore, the rule-based strategies are still the priority in real industry.

The rule-based control strategies consist of the logic threshold strategy and fuzzy logic strategy. The logic threshold strategy enjoys a good robustness and high executive efficiency but it cannot adapt to different driving cycles and the fuel economy cannot be guaranteed consequently. While fuzzy logic strategy is adaptive to various conditions with the experts’ knowledge and it doesn’t rely on accurate mathematic model, which can bring about better fuel economy but the executive efficiency is reduced. However, the combination of the logic threshold strategy and fuzzy logic strategy can achieve better results [8-9]. For the coordination control during mode switches, Ping Kan, et al [10], described the basic theory for the cooperative work of the power components and during mode switch and the integrated frame work of the controller; Lijun Qian, et al [11], put forward the “Engine dynamic torque estimation & clutch fuzzy control & 2 motors compensation” method for a 4WD PHEV, in which they took the mode switch between EV to Parallel as an example and the jerk is greatly reduced. Sangjoon Kim, et al [12], Qin Datong, et al [13], Yang Yang, et al [14] put forward three different kinds of methods for the clutch control and made the rules during mode switches between EV to HEV for a parallel HEV, which obtained optimal clutch control under different conditions. Anthony Smith, et al [15], Yongsheng He, et al [16], Zhang Na, et al [17] put forward the closed-loop control strategy in combination of motor torque and clutch constant pressure for a parallel HEV to start the engine when it was EV mode, and the control strategy was validated on the bench test.

Kerem Koprubasi, et al [18], Liu Cui, et al [19], Zhao Zhiguo, et al [20], K.Korowais, et al [21] regarded mode switches as the problem of a hybrid dynamic system or input-redundancy system and they classified the switches into different subdomains and designed controllers for them respectively. The control strategies were validated by experiments which proved them effective in reducing the jerk. R.beck et al [22], used model predictive method for the clutch control during mode switches which also reduced the jerk and the robustness was validated. Li Xian Yang et al [23] adopted dynamic programming for the smoothness during mode switch, which obtained the optimal torque trace of the engine, motor and the clutches. However, the hybrid dynamic system-based and other optimal-based methods are now in the lab and can’t be made full use of in the real car, and the estimation-compensation-based strategies still need to be improved for the inaccuracy brought in by the error of the controllers.

In this paper, the coordination control for a plug-in 4WD PHEV is researched. The rest of the paper is organized as follows. In section 2, the configuration of the PHEV is briefly described and the dynamic models of the engine, motors, clutches and the powertrain are presented. In section 3, a driver model based on fuzzy PID is built to gain the torque request coefficient. In section 4, an energy
management coordination strategy based on driver intention is put forward. In which the control for the engine is based on its optimal torque and the dynamic model for different work modes as well as the torque distribution method is proposed. In section 5, the coordination control during mode switches is introduced. The emphases are laid on the clutch control and the dynamic torque obtainment of the engine. The coordination control algorithm is introduced by taking the transition between EV to Hybrid as an example. In section 6, the control strategy is validated hardware-in-loop on the test bench. At last the conclusions are presented in section 7.

2 Configuration of the PHEV

The architecture for the 4WD PHEV is illustrated in Fig. 1. It includes 2 separate power systems, of which a rear motor with MCU is included in the first power system, while an engine with ECU, an ISG motor with its controller which lies coaxially with the engine, a starter used to start the engine when battery SOC is low, a clutch 1 which lies between the engine and ISG motor, a clutch 2 which lies between the ISG motor and the CVT, and a CVT transmission are included in the second power system. The power cell is connected to the rear motor through inverter 1 and to the ISG motor through inverter 2. [24]

![Fig.1 Layout of the 4WD PHEV](image)

2.1 Engine model

The engine used in the paper is a 4-cylinder gasoline engine (E4G16) without turbocharger, whose displacement is 1.6L, maximum power is 93kw, and maximum torque is 158 N.m. Engine model are of 2 kinds, steady type and dynamic type. When the engine is in steady state, its output torque is the function of the throttle opening and the speed whose model can be obtained through experimental data. When it is in dynamic state, the output torque can be obtained based on the steady model with a second order transfer function shown in formula 1.[25]

\[ T_e = \frac{\omega_e^2}{s^2 + 2\xi\omega_n s + \omega_n^2} f(\omega_e, \alpha_e) \]  

where \( T_e \) is engine torque, N.m; \( \omega_e \) is engine speed, \( \text{rad} \cdot \text{s}^{-1} \); \( \alpha_e \) is throttle opening of the engine, %; \( \omega_n \) is the natural frequency of the engine, \( \xi \) is the second order system damping ratio of the engine.

2.2 Motor model

The ISG motor and rear motor adopted in the 4WD PHEV are permanent magnet synchronous AC motors with the maximum torque of 100 N.m and 120 N.m respectively. The steady models for the motors are also obtained through experiments while the dynamic model can be approximately expressed by a second order response system as presented in formula 2.[26]

\[ T_m = \begin{cases} \omega_n^2 \frac{\min(T_{\text{evoil}}, T_{\text{emax}})}{s^2 + 2\xi \omega_n s + \omega_n^2} & T_{\text{evoil}} > 0 \\ \omega_n^2 \frac{\max(T_{\text{evoil}}, T_{\text{emax}})}{s^2 + 2\xi \omega_n s + \omega_n^2} & T_{\text{evoil}} < 0 \end{cases} \]  

where \( T_m \) is the motor torque, N.m; \( T_{\text{evoil}} \) is motor target torque, N.m; \( T_{\text{emax}} \) is motor maximum torque, N.m; \( \omega_n \) is the natural frequency of the motor, \( \text{rad} \cdot \text{s}^{-1} \); \( \xi \) is the second order system damping ratio of the motor.

2.3 Clutch model

When the clutches are in the state of absolute engaged or disengaged, they transfer steady torque [27], when they are still slipping, the transferred torque can be calculated by formula 3.

\[ T_c = \frac{2(r_i^2 + r_r^2 + r_r^2)}{3(r_i + r_r)} \mu Z A \text{sgn}(\Delta \omega) \]  

where \( T_c \) is the transferred torque of the clutch, N.m; \( \mu \) is the friction coefficient of the clutch; \( Z \) is the number of the friction plates; \( P \) is the oil pressure, kpa; \( A \) is the area of the clutch friction plate, \( m^2 \); \( \Delta \omega \) is the difference angular speed of active plate and passive plate of the clutch, \( \text{rad} \cdot \text{s}^{-1} \); \( r_i \) is the inner radius of the friction plate, m; \( r_r \) is the outer radius of the friction plate, m.
2.4 Powertrain model

The model of the powertrain can be simplified as Fig. 2 describes:
The rotational inertia of the front end of the clutch 1 is equivalent as formula 4 describes:
\[ J_1 = J_s + J_u \]  \hspace{1cm} (4)

The rotational inertia of the rear end of clutch 1 is equivalent as formula 5 presents:
\[ J_1 = [J_{m2} + J_{a2} + (J_{s2} + J_{m2})/i^2_{a2}] / i^2_s \]  \hspace{1cm} (5)

The rotational inertia between the front end and rear end of clutch 1 is equivalent as formula 6 shows:
\[ J_1 = J_{m1} + J_{a1} + J_{a1} \]  \hspace{1cm} (6)

The rotational inertia of the rear axle is equivalent as formula 7 shows:
\[ J_r = J_{m2} + J_{a2} + (J_{s2} + J_{m2})/i^2_{a2} \]  \hspace{1cm} (7)

where \( J_s \) is engine rotational inertia, kg \( m^2 \); \( J_u \) is the rotational inertia for the input part of clutch 1; \( J_{a0} \) is the rotational inertia of ISG motor; \( J_2 \) is the rotational inertia for the input part of clutch 2; \( J_{a2} \) is the rotational inertia for the output part of clutch 2; \( J_{s2} \) is the input part rotational inertia of CVT; \( J_{a0} \) is the output part rotational inertia of CVT; \( J_{a1} \) is rotational inertia for front final drive; \( J_1 \) is equivalent rotational inertia of the vehicle; \( m \) is vehicle mass, kg; \( r \) is tire radius, m; \( J_s \) is rotational inertia for rear motor; \( J_{a2} \) is rotational inertia for rear final drive.

3 Driver Model Based on Fuzzy-PID

From classic control theory, it is known to all that it is very difficult to set the proportion, integration, and differentiation parameters for the traditional PID. While fuzzy controller can compensate the effect of the nonlinear factors and result in better dynamic characteristics, but the static errors can’t be eliminated. To overcome the defects of the traditional PID and fuzzy logic algorithms and reflect the driver’s intentions, a driver model based on fuzzy PID is built to calculate driver’s torque request coefficient. And adopting fuzzy-PID can realize good dynamic and static performances [28].

The drive modes of the 4WD PHEV consists of rear-drive modes, front-drive modes, and 4WD modes, of which, the rear-drive modes contain the EV mode and Series mode, the front-drive modes contain engine-drive mode, ISG-drive mode, engine-drive-and-charge mode, ISG-aided-parallel mode, and 4WD modes contain rear-motor-aided-4WD mode and all-hybrid-4WD mode [29].

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First of all, define that when the torque request coefficient $K$ is between 0.8-0.95, it is small, when $K$ is between 0.95-1.05, it is medium, and when $K$ is between 1.05-2, it is big.

And then the control flow of the drive modes for the 4WD PHEV is as follows:

Step1. Judging the range of the torque request coefficient $K$; when $K$ is small, turn to step2; when $K$ is medium, turn to step3; when $K$ is big, turn to step 4.

Step2. Execute rear-drive mode.

Step2A. Judging whether $SOC$ is greater than its lowest work point $SOC_{LOW}$, and the range of the torque request $T_{req}$.

If $SOC > SOC_{LOW}$, and $T_{req} \in (0, T_{low})$, execute step2A1.
If $SOC > SOC_{LOW}$, and $T_{req} \in (T_{low}, T_{high})$, execute step2A2.
If $SOC > SOC_{LOW}$, and $T_{req} \in [T_{max}, T_{eISG}]$, execute step2A3.
If $SOC > SOC_{LOW}$, and $T_{req} \in [T_{max}, T_{eISG} + T_{max}]$, execute step2A4.
If $SOC < SOC_{LOW}$, and $T_{req} \in (0, T_{low})$, execute step2A5.
If $SOC < SOC_{LOW}$, and $T_{req} \in (T_{low}, T_{high})$, execute step2A6.
If $SOC < SOC_{LOW}$, and $T_{req} \in [T_{max}, +\infty)$, execute step2A7.

Step3A. Execute ISG drive mode, and return to execute step1. Then,

$$
\begin{align*}
T_e - T_{req} &= J_s \omega_s \\
T_s - T_{eISG} &= 0 \\
T_{eISG} - T_e &= J_s \omega_{ISG}
\end{align*}
$$

where $T_{req}$ represents the optimal torque of the engine, $T_e$ is the charging torque.

Step3. Execute the front drive mode.

Step3A. Judging whether $SOC$ is greater than its lowest work point $SOC_{LOW}$ and the range of the torque request $T_{req}$.

If $SOC > SOC_{LOW}$, and $T_{req} \in (0, T_{low})$, execute step3A1.
If $SOC > SOC_{LOW}$, and $T_{req} \in (T_{low}, T_{high})$, execute step3A2.
If $SOC > SOC_{LOW}$, and $T_{req} \in [T_{max}, T_{eISG}]$, execute step3A3.
If $SOC > SOC_{LOW}$, and $T_{req} \in [T_{max}, T_{eISG} + T_{max}]$, execute step3A4.
If $SOC < SOC_{LOW}$, and $T_{req} \in (0, T_{low})$, execute step3B1.
If $SOC < SOC_{LOW}$, and $T_{req} \in (T_{low}, T_{high})$, execute step3B2.
If $SOC < SOC_{LOW}$, and $T_{req} \in [T_{max}, +\infty)$, execute step3B3.
If $SOC < SOC_{LOW}$, and $T_{req} \in [T_{max}, +\infty)$, execute step3B4.

Step3A. Execute ISG drive mode, and return to execute step1. Then,

$$
\begin{align*}
T_{ISG} - T_{req} &= J_s \omega_{ISG} \\
T_e &= 0
\end{align*}
$$

where $\omega_{ISG}$ represents the angular speed of the ISG motor.

Step3A2. Execute engine drive mode, and return to execute step1. Then,
\[
T_e + T_{\text{ISG}} - T_{\text{req}} = (J_q + J_h) \omega_{\text{ISG}}
\]
\[
\dot{\omega}_{\text{ISG}} = \omega_{e}
\]
\[
T_e = T_{\text{req}} + J_q \dot{\omega}_e, T_a = 0
\]

where \( \omega_e \) represents the angular speed of the engine.

Step 3A3, Execute ISG-aided-parallel mode 1, and return to execute step 1. Then,
\[
T_e + T_{\text{ISG}} - T_{\text{req}} = (J_q + J_h) \omega_{\text{ISG}}
\]
\[
\dot{\omega}_{\text{ISG}} = \omega_{e}
\]
\[
T_e = T_{\text{req}} + J_q \dot{\omega}_e, T_a = 0
\]

Step 3A4, Execute ISG-aided-parallel mode 2, and return to execute step 1. Then,
\[
T_e + T_{\text{ISG}} - T_{\text{req}} = (J_q + J_h) \omega_{\text{ISG}}
\]
\[
\dot{\omega}_{\text{ISG}} = \omega_{e}
\]
\[
T_e = T_{\text{req}} + J_q \dot{\omega}_e, T_a = 0
\]

Step 3A4, Execute ISG-aided-parallel mode 3, and return to execute step 1. Then,
\[
T_e + T_{\text{ISG}} - T_{\text{req}} = (J_q + J_h) \omega_{\text{ISG}}
\]
\[
\dot{\omega}_{\text{ISG}} = \omega_{e}
\]
\[
T_e = T_{\text{req}} + J_q \dot{\omega}_e, T_a = 0
\]

Step 3B1, Execute engine-drive-and-charge mode 1, and return to execute step 1. Then,
\[
T_e + T_{\text{ISG}} - T_{\text{req}} = (J_q + J_h) \omega_{\text{ISG}}
\]
\[
\dot{\omega}_{\text{ISG}} = \omega_{e}
\]
\[
T_e = T_{\text{req}} + J_q \dot{\omega}_e, T_a = 0
\]

Step 3B2, Execute engine-drive-and-charge mode 2, and return to execute step 1. Then,
\[
T_e + T_{\text{ISG}} - T_{\text{req}} = (J_q + J_h) \omega_{\text{ISG}}
\]
\[
\dot{\omega}_{\text{ISG}} = \omega_{e}
\]
\[
T_e = T_{\text{req}} + J_q \dot{\omega}_e, T_a = 0
\]

Step 3B3, Execute engine-drive-and-charge mode 3, and return to execute step 1. Then,
\[
T_e + T_{\text{ISG}} - T_{\text{req}} = (J_q + J_h) \omega_{\text{ISG}}
\]
\[
\dot{\omega}_{\text{ISG}} = \omega_{e}
\]
\[
T_e = T_{\text{req}} + J_q \dot{\omega}_e, T_a = 0
\]

Step 3B4, System warning, execute engine-drive mode 1, and return to execute step 1. Then,
\[
T_e - \min(T_{\text{max}}, T_{\text{req}}) = (J_q + J_h) \dot{\omega}_e
\]
\[
T_{\text{ISG}} = 0, T_a = 0
\]

Step 4, Execute 4WD mode.

Step 4A, Judging whether SOC is greater than its lowest work point \( \text{SOC}_{\text{LOW}} \), if yes, execute step 4B, otherwise execute step 4A1.

Step 4A1, System warning, execute engine-drive mode 2, and return to execute step 1. Then,
\[
T_e - \min(T_{\text{max}}, T_{\text{req}}) = (J_q + J_h) \dot{\omega}_e
\]
\[
T_{\text{ISG}} = 0, T_a = 0
\]

Step 4B, Judging the range of the torque request \( T_{\text{req}} \), if \( T_{\text{req}} \in [T_{\text{max}} + T_{\text{ISG \ max}}, T_{\text{max}} + T_{\text{m \ max}}] \), execute step 4B2.

Step 4B1, Execute rear-motor-aided-4WD mode, and return to execute step 1. Then,
\[
T_e + T_m = (J_q + J_h) \dot{\omega}_e + J_m \dot{\omega}_m
\]
\[
T_m = \min(T_{\text{max}}, T_{\text{m \ max}} + J_m \dot{\omega}_m), T_{\text{ISG}} = 0
\]

Step 4B2, Execute all-hybrid-4WD mode, and return to execute step 1. Then,
\[
T_e + T_{\text{ISG}} + T_m = (J_q + J_h) \dot{\omega}_e + J_m \dot{\omega}_m
\]
\[
T_m = \min(T_{\text{m \ max}} + J_m \dot{\omega}_m), T_{\text{ISG}} = 0
\]

5 Coordination Control during Mode Switches

The plug-in 4WD hybrid electric vehicle has many work modes, and during mode transitions, because of the different dynamic torque responses of the engine, ISG motor and rear motor, there may be some time when the total torque of the power components fluctuates or in the driveline, which seriously affects the comfort of the passengers and the power performance of the car [30]. To solve the problem, a coordination method based on “Preliminary torque distribution& clutch fuzzy-fuzzy PID control& engine dynamic torque lookup& dual motor compensation” during mode switches is put forward.
5.1 Clutch control

The engagement process of the clutch can be classified into 3 steps, idle stroke stage, slipping stage, and synchronization stage. Because of the strong nonlinearity for the engagement process, it is very difficult to be expressed by math models, Du Bo, et al [31] adopted a fuzzy controller to regulate the oil pressure of the clutch, but fuzzy rules can’t adapt to different cycles. In order to have a more precise output of the oil pressure, a fuzzy-fuzzy PID method is used. The principle of the controller is presented in Fig. 5.

$$P_0 = \delta P + P_0$$  (22)

The second is the confirmation of change rate of the oil pressure. The inputs of the fuzzy controller are the change rate of acceleration pedal travel $\alpha$ and the difference of the angular speed for active and passive plates of the clutch $\Delta \omega$. The output is the change rate of the original oil pressure $\delta P$. The preliminary engagement oil pressure $P_1$ is the sum of original oil pressure $P_1$ and oil pressure increment $\delta P$. Namely,

$$P_1 = P_0 + \int \delta P \cdot dt = \delta P + P_0 + \int \delta P \cdot dt$$  (23)

The third is the adjustment of the final output oil pressure $P_2$ using fuzzy PID controller. The inputs of the controller are the preliminary output oil pressure $P_1$ and the final output oil pressure $P_2$. The final output oil pressure $P_2$ can be calculated by the following formula,

$$P_2 = K_p(t)(P_1 - P_2) + K_i(t)\int(P_1 - P_2)dt + K_d(t)\frac{d(P_1 - P_2)}{dt}$$  (24)

where $K_p(t)$, $K_i(t)$, $K_d(t)$ represent the current proportion, integration and differentiation coefficients respectively, and they are adjusted real-time according to the cycles. For the domains and membership functions of the inputs and outputs variables are not discussed in detail here.

The fuzzy rules of the oil pressure increment in the clutch and change rate of the original oil pressure are shown in Table 2 and Table 3 respectively [32].

<table>
<thead>
<tr>
<th>$\Delta \omega$</th>
<th>$\alpha'$</th>
</tr>
</thead>
<tbody>
<tr>
<td>VS</td>
<td>S</td>
</tr>
<tr>
<td>S</td>
<td>VS</td>
</tr>
<tr>
<td>MS</td>
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<td>MS</td>
</tr>
<tr>
<td>MB</td>
<td>VS</td>
</tr>
</tbody>
</table>

Table 2. Fuzzy rules of the oil pressure increment
Table 2. Fuzzy rules of the change rate of the original oil pressure

| $\Delta p'$ | $|\Delta \omega|$ |
|-------------|-------------|
| VS          | M           |
| S           | MB          |
| MS          | M           |
| M           | MB          |
| MB          | M           |
| B           | B           |
| B           | VB          |
| VB          | B           |
| VB          | B           |
| B           | MB          |
| MB          | M           |
| MB          | B           |
| VB          | VB          |
| VB          | VB          |

Note: VS, S, MS, M, MB, B, VB represent very small, small, medium small, medium, medium big, big, and very small respectively.

5.2 The coordination control strategy

The mode switches can be classified into three kinds considering whether the engine is involved in: The first is EV to the modes in which engine propels the vehicle. This kind of switch involves the clutch engagement and engine dynamic torque estimation; The second is the modes in which engine propels the vehicle to EV, which involves the disengagement of the clutches; The third is the switches among the modes in which the engine propels the vehicle, where it doesn’t involve the engage and disengage of the clutches. Of the three kinds of switches, the first is the most complicated. In the paper, only the mode switch between rear-motor-dive EV mode to ISG-aided-parallel mode, which is the first kind of switch, is taken as an example to introduce the detailed procedures of torque coordination control strategy.

In the first kind of mode switch, the most significant part is the estimation of the engine dynamic torque. In order that the dynamic torque can be put into practice in real time. The engine throttle opening and its change rate and the speed are taken as 3 main factors to do experiments and the output torque of the engine is obtained. Then Matlab is used for interpolation of the experimental data to get the detailed relations between the output torque of the engine and its throttle opening, change rate and speed. According to which a three-dimensional lookup table is made and the engine dynamic torque can be obtained real-time by looking up the table. Parts of the data of the dynamic torque are shown in Table 4.

Table 2 Parts of the dynamic output torque of the engine

<table>
<thead>
<tr>
<th>Engine speed(r/min)</th>
<th>Throttle opening (%)</th>
<th>Change rate of the throttle opening</th>
<th>Dynamic torque(N.m)</th>
<th>Engine speed(r/min)</th>
<th>Throttle opening (%)</th>
<th>Change rate of the throttle opening</th>
<th>Dynamic torque(N.m)</th>
</tr>
</thead>
<tbody>
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<td>1000</td>
<td>25</td>
<td>0.5</td>
<td>24.7</td>
<td>4000</td>
<td>25</td>
<td>0.5</td>
<td>37.5</td>
</tr>
<tr>
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<td>50</td>
<td>1</td>
<td>51.3</td>
<td>4000</td>
<td>50</td>
<td>1</td>
<td>77.8</td>
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<td>75</td>
<td>1.5</td>
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<td>4000</td>
<td>75</td>
<td>1.5</td>
<td>116.9</td>
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<td>2</td>
<td>107.5</td>
<td>4000</td>
<td>100</td>
<td>2</td>
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The procedures of the mode switch between rear-motor-dive EV mode and ISG-aided-parallel mode are as follows:

First: Judging whether mode switches. If yes, the vehicle control unit (VCU) sends the order to start the engine. Then, clutch 1 engages with the appropriate rule according to the driver’s intentions. In the beginning of the engagement, the toque $T_{cl}$ it transfers is less than the engine starting resistance torque $T_f$ and engine doesn’t start. At this stage, the dynamic model can be expressed as below:

\[
T_e - T_{sg} = J_{e} \ddot{\omega}_e
\]

\[
T_{sg} - T_f = J_{sg} \ddot{\omega}_{sg}
\]

\[
T_f = 0
\]

Second: The toque $T_{cl}$ clutch1 transfers is greater than the engine starting resistance torque $T_f$ and engine starts to rotate. Before its speed arrives at the ignition speed, the engine works as a load, and the resistance torque is $T_f$. At this stage, the dynamic model can be expressed as below:
Third: The engine speed arrives at the ignition speed, the engine is ignited. And then, clutch 2 engages with the appropriate rule according to the driver’s intentions. VCU confirms the best CVT ratio in accordance with the current velocity of the vehicle. And then the engine optimal dynamic torque is obtained by looking up the three-dimensional table, and ISG motor adjusts engine’s load and compensate part of the dynamic torque. Moreover, the torque error of the front axle is compensated by the rear axle. At this stage, the dynamic model can be expressed as below:

\[
\begin{align*}
T_e - T_{req} &= J_e \omega_e \\
T_{ISG} - T_{r} &= J_{ISG} \omega_{ISG} \\
T_r - T_{e} &= J_r \omega_r \\
\end{align*}
\]  

(26)

Fourth: If the torque request is met by the front axle, the mode switch ends. And the power components work at the point of the preliminarily distributed torque.

6 Hardware-in-loop Bench Test

In order to validate the effectiveness of the torque coordination control strategy, it is compiled to executable code and downloaded to dSPACE, which acts as a HCU. The test cycle is set to 10*NEDC, which consists of 10 NEDC cycles. And the initial SOC of the battery is 0.6. The test bench of the hybrid system is shown in Fig.6 and the experimental results are as follows.

6.1 Results of energy management coordination control

The results of energy management coordination control are illustrated in Fig. 7.

![Fig.7 Results of energy management coordination control](image)

From Fig7 (a), it is clearly seen that the range of the torque identification coefficient is mainly between 0.92-0.94, which indicates that it is small, and the hybrid electric vehicle is mainly driven by the rear motor. Moreover, in the EUDC cycle of the NEDC, the range of the torque identification coefficient is mainly between 0.95 -1.02, which indicates that it is medium, and the vehicle is driven by the front axle.

From the torque request, SOC of the battery and the outputs of the power components, as shown in Fig. 7(b)-(e), it is clearly seen that the vehicle is in charge-depleting mode in the first 3 NEDC cycles. In which the rear motor drives the vehicle and it outputs the required torque. The engine and ISG motor output positive torque only in EUDC cycle, which proves that the vehicle is in the ISG-aided-parallel mode. In the following 7 NEDC cycles, the vehicle is still mainly driven by the rear motor. In the EUDC cycle, the engine and ISG motor output positive torque, and in ECE cycle, the
engine outputs positive torque, ISG motor outputs negative torque, and rear motor outputs positive torque. So, it is clear that in the ECE cycles of the last 7 NEDC cycles, the vehicle works at Series mode, while in the EUDC cycles, the vehicle works at ISG-aided-parallel mode.

From Fig.7 (f), it is clearly seen that the fuel consumptions for 100km with and without torque identification are 3.12L and 3.43L, which proves that the fuel consumption is reduced by 9.04% and the use of torque identification can improve fuel economy.

6.2 Results of coordination control during EV to ISG-aided-parallel mode

In the paper, the jerk, longitudinal acceleration, and output of the power components are used to evaluate the effectiveness of the torque coordination control and the filtered results are illustrated in Fig.8.

![Fig.8 Results of coordination control during mode switch](image)

From Fig.8 (a), it can be seen that the absolute value of the jerk without torque coordination control is 11.3 m/s³, while with it the maximum absolute value is 6.7 m/s³, which is reduced by 40.7% and the comfort of the driver is greatly reduced. From Fig.8(b)-(c), it is clearly seen that without coordination control, the torque of the rear motor drops to zero quickly and the torque of the ISG motor synchronize with the engine torque. So the motors don’t work as compensators. While with coordination control, the torque of the rear motor drops to zero at a certain pace, and the change rate of the torque for ISG motor is faster and it arises to a certain peak before dropping to its stable value, which proves that the 2 motors works as compensators in the mode switch between rear-drive-EV mode to ISG-aided-parallel mode. Therefore, it is clear that the adoption of torque coordination control strategy proposed in the paper can realize very good control effect.

7 Conclusions:

1) The diver model based on fuzzy PID can reflect the driver’s intentions and the fuel consumption is reduced by 9.04%.
2) With coordination control of energy management, the vehicle works well in the stable state, which is validated by the output of the power components.
3) With coordination control during mode switches, the maximum absolute value of jerk is reduced by 40.7%.

References


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