Control Strategy Comprehensively Considering Fuel Economy and Drive Comfort of Duel-Axis-Parallel PHEV

K S Xi¹, Y Zou¹, *, Y F Zhou², Z G Zhang³ and H H Lei³

¹National Engineering Laboratory for Electric Vehicles, School of Mechanical Engineering, Beijing Institute of Technology, NO.5 Zhongguancun south Street, Haidian, Beijing, China
²The Experimental High School Attached to Beijing Normal University, Jia NO.14 Erlong Road, Xicheng, Beijing, China
³Zhengzhou Foguang Electric Power Equipment CO., LTD, NO.50 Dongqing Street, High-tech development zone, Zhengzhou, China

*Corresponding author e-mail: zouyuanbit@vip.163.com

Abstract. In this paper, in order to comprehensively consider the economy and comfort of the special duel-axis-parallel PHEV, a vehicle control strategy composed of a shift strategy based on dynamic programming (DP), a rule-based energy management strategy and a dynamic coordinated control strategy based on motor torque compensation is proposed. A vehicle model which can show the dynamic characteristics of shifting and engine start – stop process is constructed for simulations. Shift strategy is extracted from the optimization results obtained by dynamic programming (DP). The energy management strategy takes the fuel economy of engine as a major consideration and the rule is determined by characteristics of engine, motor and battery. In the dynamic coordinated control strategy, an open-loop motor torque control based on clutch torque estimation is presented. Simulation is carried out on typical driving cycles. And the results indicate that the proposed vehicle control strategy can improve the fuel economy and reduce longitudinal impact caused by shifting and operated mode changing.

1. Introduction

Nowadays problems about energy exhaustion and air pollution are increasingly serious. As a good solution to these problems, new energy vehicles has received unprecedented attention [1]. PHEV which has the advantages of electric vehicles and traditional hybrid electric vehicles has achieved great development in recent years. Energy management for PHEV which is aimed at getting long driving distances through reasonable power distribution of power sources is a hot research topic. In addition, consumers have more requirements on the comfort of PHEV with the development of new energy vehicles.

There are mainly two categories of power management methods: rule-based power management and optimization based power management. Rule-based strategies are widely adopted in practical application because of reliability [2]. Instantaneous optimization control strategy has high real-time performance, like equivalent consumption minimization strategy (ECMS) [3]. Global optimal power management methods
normally rely on the knowing of future driving cycle, such as dynamic programming (DP) [4, 5]. At the same time, aiming at improving the dynamic performance of PHEV during mode switching and gear shifting, some control methods which make full use of the advantages of multiple power sources are put forward [6, 7].

The remainder of this paper is organized as follows. In Section 2, the model of the powertrain system applied for simulation is described. In Section 3, vehicle control strategy composed of shift strategy, energy management and dynamic coordinated control is proposed. In Section 4, in order to prove the improvements in fuel economy, the Vehicle control strategy is compared with DP. And by comparing with the strategy without dynamic coordination control, it is shown that the drive comfort can be significantly improved by using dynamic coordination control. In Section 5, conclusions are given.

2. Model of Powertrain system
The structure of the duel-axis-parallel PHEV in this paper is shown in Figure 1.

Figure 1. The structure of the duel-axis-parallel PHEV.

It is mainly composed of engine, power battery pack, motor, clutch, AMT with double shaft input and final drive. The engine and motor are connected to the AMT through two shafts. The engine torque and motor torque are coupled internally in the AMT by a variety of different gear combinations. After analyzing the characteristics of the structure of duel-axis-parallel PHEV, the following relationship can be obtained:

\[ T_{req} = T_e I_{g-e} I_0 + T_m I_{g-m} I_0 \]  

(1)

\( T_{req} \) (Nm) is vehicle required torque. \( T_e \) (Nm) is engine torque. \( T_m \) (Nm) is motor torque. \( I_{g-e} \) is transmission ratio of the engine road. \( I_{g-m} \) is transmission ratio of the motor road. \( I_\circ \) is transmission ratio of final drive.

Engine, motor and battery are all modeled using normal methods. What needs special introduction is the method of clutch modelling. The three working states of clutch are separation, slip and engagement. The calculation formula of the transmission torque \( T_c \) of the clutch in three working states is as follows:

\[
T_c = \begin{cases} 
0 & \text{disengaged} \\
T_f & \text{slipping and } T_f \leq T_{c, \text{max}} \\
T_e & \text{engaged and } T_e \leq T_{c, \text{max}} \\
T_{c, \text{max}} & \text{or } T_f > T_{c, \text{max}} 
\end{cases} 
\]  

(2)

The calculation formula of the dynamic friction torque of the clutch \( T_f \) is as follows:

\[ T_f = \text{sign}(\omega - \omega_c) Z \mu_a F_N R_c + \lambda_c T_e, \lambda_c = e^{107|\text{sign}(\omega - \omega_c)|R_c} \]  

(3)

\( \omega \) (rad/s) is speed of engine. \( \omega_c \) is speed of clutch driven shaft (rad/s). \( Z \) is number of friction surface. \( \mu_a \) is dynamic friction coefficient. \( F_N \) is clutch pressure. \( R_c \) is equivalent friction radius. \( \lambda_c \) is impact factor of input torque.
3. Vehicle control strategy

3.1. Shift strategy

PHEV system can be discretized into a discrete dynamic system represented by the following formula:

\[ x(k) = \begin{bmatrix} \text{SOC}(k) \\ g_e(k) \\ g_m(k) \end{bmatrix}, u(k) = \begin{bmatrix} T_{e,\text{req}}(k) \\ \text{shift}_e(k) \\ \text{shift}_m(k) \end{bmatrix} \] (4)

After analyzing the powertrain system, three state variables are selected. First one is state of charge of the battery \( \text{SOC}(k) \). Second one is engine gear \( g_e(k) \). Third one is motor gear \( g_m(k) \). Control vector consists of engine required torque \( T_{e,\text{req}}(k) \), engine shift command \( \text{shift}_e(k) \) and motor shift command \( \text{shift}_m(k) \).

\[ g_e(k + 1) = \begin{cases} 5 & g_e(k) + \text{shift}_e(k) > 5 \\ 1 & g_e(k) + \text{shift}_e(k) < 1 \\ g_e(k) + \text{shift}_e(k) & \text{otherwise} \end{cases} \] (5)

\[ \text{shift}_e(k) = \begin{cases} -1 & \text{downshift} \\ 1 & \text{upshift} \\ 0 & \text{otherwise} \end{cases} \] (6)

\( g_m(k) \) and \( \text{shift}_m(k) \) are similar to \( g_e(k) \) and \( \text{shift}_e(k) \).

In this paper, regardless of engine emissions, cost-to-go function and instantaneous cost function are defined as follows:

\[ J_e(\chi_0) = \lim_{N \to \infty} E \left\{ \sum_{k=0}^{N-1} L(x(k), u(k)) \right\} \] (7)

\[ L(x(k), u(k)) = L_{\text{fuel}}(k) + \alpha L_{g_e}(k) + \beta L_{g_m}(k) \] (8)

\( N \) is the duration of the driving cycle. Drive comfort will be reduced if shifting frequently, therefore an additional cost function is adopted to avoid frequent shift. This additional cost function is as follows, \( \alpha \) and \( \beta \) are the weighting factors.

\[ \alpha L_{g_e}(k) = \alpha |\text{shift}_e(k)|, \beta L_{g_m}(k) = \beta |\text{shift}_m(k)| \] (9)

After the above analysis, we can get the objective function of dynamic programming optimization as follows:

\[ J_e(\chi_0) = \lim_{N \to \infty} E \left\{ \sum_{k=0}^{N-1} L_{\text{fuel}}(k) + \alpha |\text{shift}_e(k)| + \beta |\text{shift}_m(k)| \right\} \] (10)

The system constraint is as follows:

\[ \begin{cases} \omega_{e,\text{min}} \leq \omega_e(k) \leq \omega_{e,\text{max}} \\ \omega_{m,\text{min}} \leq \omega_m(k) \leq \omega_{m,\text{max}} \\ \text{SOC}_{\text{min}} \leq \text{SOC}(k) \leq \text{SOC}_{\text{max}} \\ T_{e,\text{min}}(\omega_e(k)) \leq T_e(k) \leq T_{e,\text{max}}(\omega_e(k)) \\ T_{m,\text{min}}(\omega_m(k), \text{SOC}(k)) \leq T_m(k) \leq T_{m,\text{max}}(\omega_m(k), \text{SOC}(k)) \end{cases} \] (11)
According to the optimization results of DP, the working point of the optimal gear position is drawn in Figure 2. The dividing line between the different points is the shift rule curve.

![Figure 2. Optimization results of DP.](image)

In order to avoid frequent shifting, the curve above is shifted to the left to get the downshift curve. The final upshift and downshift curves are shown in Figure 3:

![Figure 3. Upshift and downshift curves.](image)

### 3.2. Energy management strategy

Energy management in this paper is the CD-CS mode which is widely used in PHEV. CD mode and CS mode are divided into several submodes, including motor drive mode, engine drive mode, motor assisted drive mode, engine assisted drive mode and charge mode. The conditions for switching between modes are predefined. Changes in external input and vehicle status variables will lead to switching between modes. There is a specific torque allocation strategy according to the characteristics of the engine and motor based on experiments in each submode, which makes the engine and motor work in a high efficiency range.

### 3.3. Dynamic coordinated control strategy

Actual output torque of the engine and clutch pressure signal can be obtained by sensors. It is difficult to control engine torque. In order to prevent engine torque from changing too much in a short time, control strategy of engine torque based on slope restrictions is adopted. Then the collected engine torque signal and clutch pressure signal are used to calculate the clutch transfer torque which the torque of motor needed is determined by.

During the mode switching process with engine start, the motor drives the vehicle and starts the engine at the same time. The control strategy is as follows:

\[
\begin{align*}
T_{e\_req} &= 0 & \omega_e < 1000 \\
T_{e\_req} &= \min(k_f T_{e\_hat}(t_f)) & \omega_e \geq 1000 \\
T_{m\_req} &= \frac{T_{req} - T_e I_{g\_e} I_0}{I_{g\_m} I_0}
\end{align*}
\]

During the mode switching process with engine stop, the control strategy is as follows:
During the process of shifting, engine torque drops to zero at slope $k_1$ at the beginning. Then shift operation takes place. When shifting is completed, engine torque rises to target torque at slope $k_2$. The motor compensates the torque during the whole process.

\[
T_{e\text{-req}} = \max\left((T_{e\text{-tor}}(t_0) - k_1 t), 0\right) \\
T_{m\text{-req}} = \frac{T_{\text{req}} - T_{e\text{-req}} I_0}{I_{g\text{-n}} I_0}
\]

(13)

During the process of shifting, engine torque drops to zero at slope $k_1$ at the beginning. Then shift operation takes place. When shifting is completed, engine torque rises to target torque at slope $k_2$. The motor compensates the torque during the whole process.

\[
\begin{align*}
T_{e\text{-req}} &= \max\left((T_{e\text{-tor}}(t_0) - k_1 t), 0\right) & 0 < t \leq t_1 \\
0 &= t_1 < t \leq t_2 \\
T_{e\text{-req}} &= \min(k_2(t - t_2), T_{e\text{-tor}}(t)) & t_2 < t \leq t_f - t_0 \\
T_{m\text{-req}} &= \frac{T_{\text{req}} - T_{e\text{-req}} I_0}{I_{g\text{-n}} I_0}
\end{align*}
\]

(14)

During the process of shifting, engine torque drops to zero at slope $k_1$ at the beginning. Then shift operation takes place. When shifting is completed, engine torque rises to target torque at slope $k_2$. The motor compensates the torque during the whole process.

4. Simulation results

Because DP is a kind of global optimization method, which means the policy given by DP can be considered as the optimal policy. So DP based PHEV energy management can serve as a benchmark of fuel economy for comparison with simulation results of proposed method. $4\times$ WLTP are used as the experimental driving cycle in simulation. The initial SOC is set to 0.95 and the final SOC is set to 0.26.

The simulation results of DP method and proposed method are shown in Table 1. The cost of proposed method one driving cycle takes is 6.43% higher than the optimal results calculated by DP. Because the cost is close between proposed method and DP method, it can be considered that the control strategy presented in this paper has satisfactory fuel economy.

Table 1. Results comparison of two methods.

| Driving cycle | Mileage  | Cost        | Comparison            |
|---------------|----------|-------------|-----------------------|
| DP method     | 4WLTP    | 93.07km     | 2.80L/100km           | 6.43% higher than |
| Proposed method| 4WLTP    | 93.07km     | 2.98L/100km           | DP                  |

In order to objectively evaluate the driving performance of vehicles, the longitudinal jerk of vehicles is used as the evaluation index of driving performance in this paper. Jerk is defined as the inverse of the longitudinal acceleration of a vehicle. The formula is as follows:

\[
J = \frac{da}{dr} = \frac{d^2v}{dr^2}
\]

(15)

Take the mode switching process with engine start as an example. Figure 4 shows the simulation results without dynamic coordinated control when the engine starts. Output torque can't meet the demand torque without the compensating torque provided by the motor which causes the speed fluctuation. And the maximum jerk is up to 15 m/s$^3$. Figure 5 shows the simulation results with dynamic coordinated control. The output torque can be maintained stable and the maximum jerk is 4 m/s$^3$. By comparing the results in two simulations, it is clear that the dynamic performance has been improved.
5. Conclusions
This paper presents a control strategy for PHEV which consists of the shift strategy extracted from optimization results obtained by DP, the energy management strategy based on rules determined by experiments and the dynamic coordinated control strategy based on motor torque compensation. A specific model which can reflect the transient and steady-state performance of the PHEV is constructed for simulations. The simulation results illustrate that the proposed control strategy has satisfactory fuel economy and can notably improve the power performance and drive comfort during the process of mode switching and shifting.

Acknowledgments
This work was financially supported by SAIC Motor Commercial Vehicle Co., Ltd. Technical Center (Contract No.7200025538).

References
[1] Ferrero E, Alessandrini S and Balanzino A 2016 Impact of the electric vehicles on the air pollution from a highway Appl. Energy 169 450–9
[2] Peng J K, He H W and Xiong R 2016 Rule based energy management strategy for a series–parallel plug-in hybrid electric bus optimized by dynamic programming Appl. Energy 185 1633–43
[3] Geng B, Mills J K and Sun D 2011 Energy management control of micro turbine-powered plug-in hybrid electric vehicles using the telemetry equivalent consumption minimization strategy IEEE Trans. Veh. Technol. 60 4238–48
[4] Wang X M, He H W, Sun F C and Zhang J L 2015 Application study on the dynamic programming algorithm for energy management of plug-in hybrid electric vehicles Energies 8 3225–44
[5] Liu J M and Peng H 2008 Modeling and Control of a Power-Split Hybrid Vehicle IEEE Trans. Control Syst. Technol. 16 1242–51
[6] Davis R I and Lovenz R D 2003 Engine torque ripple cancellation with an integrated starter alternator in a hybrid electric vehicle: implementation and control IEEE Trans. Ind. Appl. 39 1765–74
[7] Hwang H S, Yang D H, Choi H K, Kim H S and Hwang S H 2011 Torque control of engine clutch to improve the driving quality of hybrid electric vehicles Int. J. Automot. Technol. 12 763–8