

Brake Clutch Assisted Mode Transition Control for Compound Power-Split Vehicle

Tong Zhang
 Corun CHS Tech. Co., Ltd.

Chen Wang
 Corun CHS Tech. Co., Ltd.; Tongji University

Wentai Zhou, Huijun Cheng, and Haisheng Yu
 Corun CHS Tech. Co., Ltd.

ABSTRACT

Because a compound power-split transmission is directly connected to the engine, dramatic fluctuations in engine output torque result in strong jerks and torque losses when the hybrid vehicle is in mode transition from electric drive mode to hybrid drive mode. In order to enhance ride comfort and reduce the output torque gap during mode transition process, a brake clutch assisted coordinated control strategy was developed. Firstly, the dynamic plant model of the power-split vehicle including driveline model, engine ripple torque and brake clutch torque was deduced. Secondly, the brake clutch assisted mode transition process was analyzed, and the output torque capability was compared between cases of both brake clutch assisted and unassisted mode transition process. Thirdly, a coordinated control strategy was designed to determine the desired motor torque, brake clutch torque, engine torque, and the moment of fuel injection. Finally, the effectiveness of coordinated control strategy was verified by vehicle test. The experimental results indicate that the proposed strategy was effective to suppress driveline speed oscillations during engine cranking phase, and reduce the engine cranking time.

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INTRODUCTION

With the increasing issues of energy crisis and environmental pollution, hybrid electric vehicles (HEVs) are attracting considerable attention. Among the various alternative powertrains, the power-split hybrid system [1, 2] is considered to be one of the most promising configurations for HEVs. The planetary gear train (PGT) is compact, efficient, and has a high torque capacity. In addition, the power-split hybrid system performs as an electronic continuously variable transmission, which can result in efficient engine operation [3, 4, 5]. However, the ride comfort issue for power-split HEVs requires special considerations as the engine is connected directly to the input shaft of the hybrid transmission through a torsional damper spring (TDS). As a low pass filter, TDS isolates the high-frequency engine torque fluctuation at the normal engine operation speed when the engine is operating. Meanwhile, the engine ripple torque (ERT [6]), especially at low engine speeds, is transmitted to the driveline when the engine is not operating. The TDS is excited by the ERT at extremely low engine speeds and causes large oscillations during mode transition from electric drive mode to hybrid drive mode. To achieve acceptable ride comfort for power-split HEVs, transient control during the mode transition is rather challenging as it requires coordinated control of the engine, motors, and transmission actuators.

The factors that influence the ride comfort of power-split HEVs during the mode transition process were summarized by Yoshioka [7] and Kuang [8]. Some proposals were made regarding the variable magnitude and frequency of the motor torque set points, fuel ignition control, and intake air volume control. Tomura [9] identified the root cause of shock during engine cranking in the power-split HEV as ERT at low engine speeds, and the root cause of vehicle driveline vibration as the rapid response of motor torque. The engine crankshaft position signal was captured directly by wire harness and the ERT was estimated and compensated by two motors. A torque pulsation cancellation strategy and a damper bypass clutch (DBC) were proposed by Hwang [10], where the DBC was rigidly connected to the engine crankshaft and PGT. The influences of torque derived from TDS during the mode transition process can be avoided by engaging the bypass clutch, and the anti-vibration torques provided by the motors passed through this rigid path more directly and effectively. Although these techniques can significantly improve vehicle ride comfort during the mode transition process, it is uncommon to use the dual hall sensors for directional and positional detection in a traditional engine. In addition, the design and implementation of DBC control should be considered. Zhuang [11] adopted dynamic programming as an optimization application to investigate the relationship between the mode transition time and the

driver torque demand. However, the optimization method cannot be used in real-time control directly. Zeng [12] proposed a predictive-model-based dynamic coordinated control strategy to improve ride comfort during mode transition. The effectiveness of the method was validated by simulation results. However, its aim was limited to the input power-split hybrid system, which is a single degree of freedom system. In the previous research, Wang [13] proposed a coordinated control strategy with active damping control and carrier torque estimation to mitigate driveline speed oscillations. But the improvement in ride comfort is at the cost of a reduction of output torque, and its effectiveness is limited to motor torque capability.

This study introduces a brake clutch assisted coordinated control strategy to enhance drivability for a compound power-split hybrid vehicle during the mode transition process. The remaining sections of this paper are organized as follows: In Section 2, the compound power-split hybrid powertrain is introduced, and the dynamic model of the power-split hybrid system, engine ripple torque, and brake clutch torque are established. In Section 3, the mode transition process is described, and the output torque is analyzed. In Section 4, the brake clutch assisted mode transition coordinated control strategy is developed. In Section 5, the effectiveness of the proposed control strategy is testified by vehicle experiments. In Section 6, some concluding remarks are presented.

COMPOUND POWER SPLIT HYBRID POWERTRAIN

Mechanism

The schematic diagrams of PGTs and hybrid powertrain are shown in Figure 1. The hybrid transmission is made up of one compound power-split device [14], two electric motors/generators (MGs), and two wet brake clutches (B). The two groups of planet gears are connected by a common carrier shaft attached to the engine crankshaft through a TDS. The carrier can be grounded by the wet brake clutch B1. The small sun gear of the front PGT is connected to MG1, which can be held stationary by the wet brake clutch B2. The large sun gear of the rear PGT is connected to MG2, and the output ring gear to the final reduction gear. The prototypes of the power-split device and hybrid transmission system are shown in Figure 2.

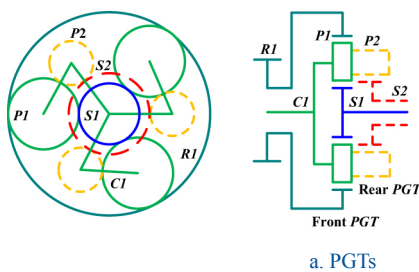


Figure 1.

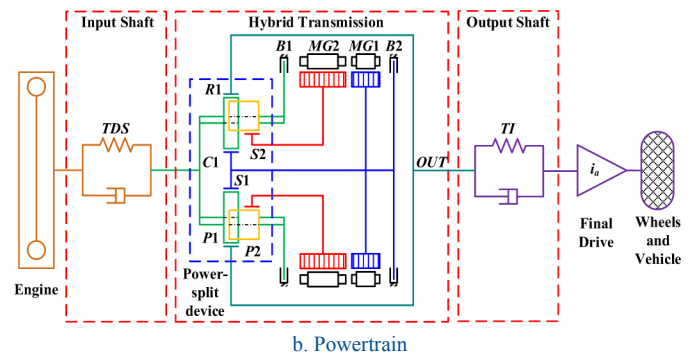


Figure 1. (cont.) Schematic diagrams of PGTs and hybrid powertrain.

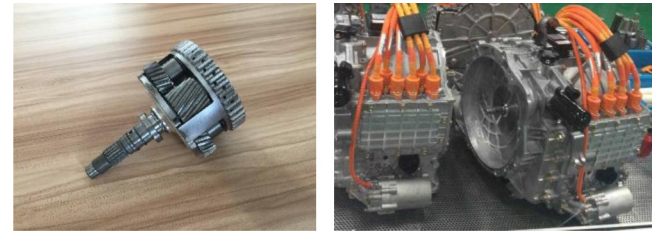


Figure 2. Prototypes of power-split device and hybrid transmission.

Where,

S1 = small sun gear of the front PGT

S2 = large sun gear of the rear PGT

OUT/R1 = output shaft/ring gear

P1 = short and thick planet gear

P2 = long and thin planet gear

C1 = planet carrier

TI = equivalent elastic shaft of the tire and half shaft

Driveline Modeling

The elasticity of TDS, half shaft, and tire are considered in the model. Ignoring the inertia moments of all planet and sun gears, the gap and mesh elasticity between each gear pair, and the equivalent friction damping loss of each shaft, the kinetic and kinematic expressions for each shaft are given by [13]:

$$T_{S1} + T_{S2} + T_{C1} + T_{R1} = 0 \quad (1)$$

$$-\rho_1 T_{S1} + \rho_2 T_{S2} + T_{R1} = 0 \quad (2)$$

$$I_{MG1} \ddot{\theta}_{MG1} + T_{S1} = T_{MG1} + T_{B2} \quad (3)$$

$$I_{MG2} \ddot{\theta}_{MG2} + T_{S2} = T_{MG2} \quad (4)$$

$$k_{TDS} (\theta_{ENG} - \theta_{C1}) + c_{TDS} (\dot{\theta}_{ENG} - \dot{\theta}_{C1}) - T_{C1} - I_{C1} \ddot{\theta}_{C1} = 0 \quad (5)$$

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$$T_{ENG} + T_{B1} - k_{TDS}(\theta_{ENG} - \theta_{C1}) - c_{TDC}(\dot{\theta}_{ENG} - \dot{\theta}_{C1}) - I_{ENG}\ddot{\theta}_{ENG} = 0 \quad (6)$$

$$k_{TI}(\theta_L - \theta_{R1}) + c_{TI}(\dot{\theta}_L - \dot{\theta}_{R1}) - T_{R1} - I_{R1}\ddot{\theta}_{R1} = 0 \quad (7)$$

$$T_L - k_{TL}(\theta_L - \theta_{R1}) + c_{TL}(\dot{\theta}_L - \dot{\theta}_{R1}) - I_L\ddot{\theta}_L = 0 \quad (8)$$

$$\ddot{\theta}_{MG1} = -\rho_1\ddot{\theta}_{R1} + (1 + \rho_1)\ddot{\theta}_{C1} \quad (9)$$

$$\ddot{\theta}_{MG2} = \rho_2\ddot{\theta}_{R1} + (1 - \rho_2)\ddot{\theta}_{C1} \quad (10)$$

$$T_L = \left(mgf + \frac{C_d A v^2}{21.15} + mgi + \delta m \frac{dv}{dt} \right) r / i_a \quad (11)$$

Where,

I = inertia

θ = angular displacement

T = torque

L = the equivalent load of vehicle

WH = wheel

Where key vehicle and hybrid powertrain parameters are listed in Table 1.

Table 1. Key vehicle and hybrid powertrain parameters.

Parameters	Value
Vehicle mass m	1530 kg
Wheel radius r	0.31 m
Tire rolling resistance coefficient f	0.0137
Aerodynamic drag coefficient C_d	0.307
Effective frontal area A	2.19 m ²
Final reduction gear ratio i_a	3.529
TDS torsional stiffness k_{TDS}	629 Nm/rad
TDS damping coefficient c_{TDS}	0.1 Nm/(rad/s)
Tire and half shaft torsional stiffness k_{TI}	2864 Nm/rad
Tire and half shaft damping coefficient c_{TI}	15 Nm/(rad/s)
Gear ratio of the front PGT ρ_1	3.174
Gear ratio of the rear PGT ρ_2	2.355
Engine inertia I_{ENG}	0.18 kgm ²
Carrier inertia I_{C1}	0.001 kgm ²
Ring gear inertia I_{R1}	0.002 kgm ²
MG1 inertia I_{MG1}	0.041 kgm ²
MG2 inertia I_{MG2}	0.072 kgm ²
Vehicle equivalent inertia I_L	150 kgm ²

ERT Modeling

Prior to ignition, the engine output torque acts on the driveline, owing to the permanent mechanical connection between the engine and transmission. Thus, it should be modeled to repeat the mode transition issue and to guide the controller design. Generally, ERT is composed of pumping torque, piston and valve friction torque, reciprocating inertia torque, and load torque due to other accessories. This study considered pumping torque and reciprocating inertia torque, which can be calculated using the following equations:

$$T_p = A_p (P_{cylinder} - P_0) r_c \left(\sin \alpha + \frac{\frac{r_c}{l} \sin \alpha \cos \alpha}{\sqrt{1 - \left(\frac{r_c}{l} \sin \alpha \right)^2}} \right) \quad (12)$$

$$T_a = -m_p \left(r_c \sin \alpha + \frac{r_c^2 \sin 2\alpha}{2l_v} \right) \ddot{\alpha} + \dots + \left(r_c \cos \alpha + \frac{r_c^2 \cos 2\alpha}{l_v} + \frac{r_c^4 \sin^2 2\alpha}{4l_v^3} \right) \dot{\alpha}^2 \quad (13)$$

Where,

$$A_p = \pi r_{cylinder}^2, \quad l_v = \sqrt{l^2 - (r_c \sin \alpha)^2}$$

Where engine parameters are listed in Table 2.

Table 2. Engine Parameters.

Parameters	Value
Cylinder pumping pressure $P_{cylinder}$	/
Top area of the piston A_p	/
Crank angle α	/
Environment pressure P_0	0.101325 kPa
Crankshaft radius r_c	45.75 mm
Equivalent piston mass m_p	0.3506 Kg
Connection rod length l	146.65 mm
Cylinder radius $r_{cylinder}$	39.65 mm

Based on the engine dynamometer, a 1.8-l L4 engine's cylinder pumping pressure versus crank angle at different engine speed measured using a piezoelectric pressure transducer and a shaft encoder are presented, as shown in Figure 3. Note that the throttle position and coolant temperature were set 6 % and 25 °C respectively, and the first cylinder's top dead center was calibrated as the zero-degree crank angle. The ERT of the four cylinders, including the pumping torque and the reciprocating inertia torque, were calculated using Eqs. (12) and (13). The results are shown in Figure 4.

As shown in Figure 3, the engine cylinder pumping pressure increased drastically around the top dead center, and the maximum cylinder pumping pressure occurred at engine speed range between 400 and 600 r/min. When engine speeds were above 200 r/min, the