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Optimal control of mode transition for four-wheel-drive hybrid electric vehicle with dry dual-clutch transmission

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ABSTRACT

When the four-wheel-drive hybrid electric vehicle (HEV) equipped with a dry dual clutch transmission (DCT) is in the mode transition process from pure electrical rear wheel drive to front wheel drive with engine or hybrid drive, the problem of vehicle longitudinal jerk is prominent. A mode transition robust control algorithm which resists external disturbance and model parameter fluctuation has been developed, by taking full advantage of fast and accurate torque (or speed) response of three electrical power sources and getting the clutch of DCT fully involved in the mode transition process. Firstly, models of key components of driveline system have been established, and the model of five-degrees-of-freedom vehicle longitudinal dynamics has been built by using a Uni-Tire model. Next, a multistage optimal control method has been produced to realize the decision of engine torque and clutch-transmitted torque. The sliding-mode control strategy for measurable disturbance has been proposed at the stage of engine speed dragged up. Meanwhile, the double tracking control architecture that integrates the model calculating feedforward control with H_∞ robust feedback control has been presented at the stage of speed synchronization. Finally, the results from Matlab/Simulink software and hardware-in-the-loop test both demonstrate that the proposed control strategy for mode transition can not only coordinate the torque among different power sources and clutch while minimizing vehicle longitudinal jerk, but also provide strong robustness to model uncertainties and external disturbance.

1. Introduction

Four-wheel-drive hybrid electric vehicle (HEV) equipped with a dual-clutch transmission (DCT) has four power sources and several operation modes. Theoretically, a HEV can make the best of motor regenerative braking [1,2], and optimize the working point of the engine and motor to improve the efficiency of energy conversion while further promoting its dynamic performance and fuel economy [3–7]. At the same time, owing to the availability of various operation modes, switching between different operating modes according to vehicle conditions and road conditions becomes more frequent. If the torques of power sources and clutch cannot be properly controlled during the process of mode transition, torque fluctuations of the output shaft will occur, resulting in vehicle longitudinal jerk on the vehicle [8]. Unreasonable torque distribution also can cause deterioration of the fuel consumption and emission performance of the engine. Additionally, in the design of mode

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Nomenclature

A	factual friction area
a_s^{ini}	angular velocity of the rear wheel
a_x	vehicle longitudinal acceleration
C_D	drag coefficient
F_{xi}	longitudinal force of each wheel
F_z	tire vertical load
F_{z0i}	static load
$I(t)$	real-time current
I_s, b_s	inertial and dampness, respectively
I_s^{equ}, b_s^{equ}	equivalent inertial and dampness of the output shaft, respectively
$I_{\omega i}$	moment of inertia of each wheel
I_x, c_x, i_x	inertial, dampness, and transmission ratio of the components, respectively
n_{ENG}	engine speed
p_c	clutch engaging pressure
Q	maximum capacity of the battery
R_0	internal resistance
R_{ei}	factual wheel-rolling radius
R_l	polarization resistance
SOC_0	battery's initial SOC value
S_x, S_y	longitudinal and lateral slip ratio of the tire respectively
T_e	engine output torque
T_e^{exp}	engine target output torque
T_{ISG}	ISG output torque
T_{ISG}^{exp}	ISG target output torque
T_e^{lag}	engine torque lag value
T_f^{dist}	engine dynamic resistance torque fluctuation
T_M	in-wheel motor output torque
T_c^{exp}	clutch actual transmitted torque
T_L^{exp}	wheel torque
$T_{ENGRipple}$	engine low-speed fluctuation torque
T_{di}	driving torque of each wheel
T_{bi}	braking torque of each wheel
t_{switch}	transition process time
$U_0(t)$	power supply voltage
$U_c(t)$	polarization voltage
V_{out}	output voltage
$\dot{x}_{ref}$	engine reference speed
$y_{ref}(t_{k+1})$	reference value of the trajectory at the (k + 1)-th moment
$y_{real}^{ini}, y_{aim}(t_{k+1})$	initial value of the reference trajectory and current value of the transition target variable at the (k + 1) moment, respectively
z	number of the friction plate
ω_s^{ref}	wheel reference speed
ω_s	transmission output shaft speed
ω_s^{ini}	initial angular velocity of the rear wheel
$\theta_{ENG\_Initial}$	crankshaft initial angle
θ_{ENG}	crankshaft angle
μ_c	friction coefficient
μ_x, μ_y	longitudinal and lateral friction coefficients, respectively
Δ_e	speed difference between the clutch driving and driven plate
ΔF_{zli}	load transfer caused by the longitudinal acceleration
η	battery efficiency
ρ_a	air density

transition controller, the robustness issue of the transition control caused by the uncertainties concerning the external disturbance and model parameters should be considered.