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Traction Control Method of Hybrid Electric Vehicle based on Multi-Objective Dynamic Coordination Control

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Abstract

The control method of the conventional traction control system on split-μ surfaces improves vehicle acceleration performance, but influences its stability performance. To solve this problem, a hierarchical traction control system for ISG hybrid electric vehicles based on multi-objective dynamic coordination control (MHEVTCS) is proposed. In the upper level controller, a target driving torque calculating strategy based on dynamical sliding mode control is developed. In the lower level controller, such strategies as multi-objective dynamic coordination control strategy, brake torque control strategy based on an inverse model, target engine torque design strategy and torque dynamic coordinate control strategy are proposed. Detailed simulation and hardware-in-loop experiment results show that slipping wheels are controlled quickly, accurately and smoothly by MEHVTCS. MHEVTCS solves the problem of merely pursuing acceleration performance and neglecting stability performance of conventional traction control system.

Keywords: Hybrid Electric Vehicle, Traction Control System, Coordination Control

1 Introduction

On split-μsurfaces, traction control system of conventional internal combustion engine vehicle (ICETCS) applies brake torque on the slipping wheel of low adhesion side road to improve vehicle's acceleration performance[1][2]. However, additional yaw moment is caused which influences its stability performance.

Motor system of hybrid electric vehicle (HEV) has better dynamic performance than engine system and hydraulic system[3][4]. It can provide a large drive torque at a low speed. Compared to conventional internal combustion engine vehicle, HEV is easier to slip on slippery road[5][6]. If the control method of ICETCS on split-μ surfaces is continually used by HEV, more brake torque intervening should be used to maintain the slip ratio of slipping wheel at a desired value. And, more additional yaw moment will be caused that influences vehicle's stability severely. Thus, traction control system of HEV (HEVTCS) should be redesigned.

Taking an ISG hybrid electric vehicle as the research object as shown in Figure1, the traction control method of an ISG hybrid electric vehicle is researched in this paper.

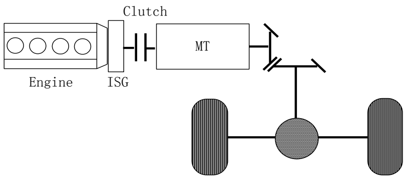


Figure1: Configuration of the research object

2 Hierarchical Control System

Based on a typical structure of HEVTCS, MEHVTCS is proposed as shown in Figure2 which is composed of two level controllers.

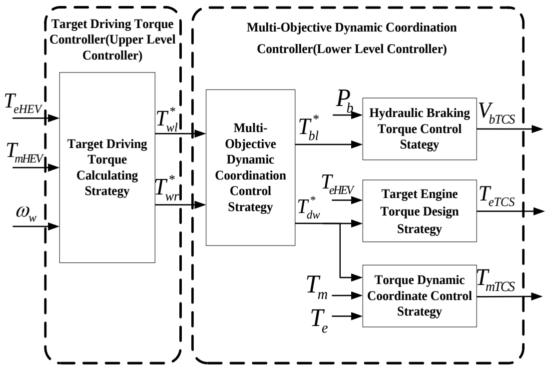


Figure2: Control structure of MEHVTCS

In the upper level controller, a target driving torque calculating strategy determines target driving torques exerted on left front wheel T_{wl}^* and exerted on right front wheel T_{wr}^* according to the desired engine torque T_{eHEV}, the desired motor torque T_{mHEV} and the wheel rotation speed ω_w to ensure that slip ratios of all slipping wheels can be maintained at desired values. In the lower level controller, several strategies such as multi-objective dynamic coordination control strategy, brake torque control strategy based on an inverse model, target engine torque design strategy and torque dynamic coordinate control strategy are proposed to control the engine torque demand T_{eTCS}, the motor torque demand T_{mTCS} and the solenoid valve opening demand V_{bTCS} in coordination to track the target driving torques T_{wl}^* and T_{wr}^* which are calculated by the upper level controller. In Figure2, T_{eHEV} and T_{mHEV} are calculated by the energy management system, T_{bl}^* is the brake torque demand exerted on the low adhesion wheel, T_{dw}^* is the driving torque demand exerted on the wheels, P_b is the actual brake pressure, T_e is the actual engine torque, T_m is the actual motor torque.

3 Target Driving Torque Control for the Upper Level Controller

In the target driving torque calculating strategy, a dynamical sliding mode controller of multi-input and multi-output is designed as shown in Figure3.

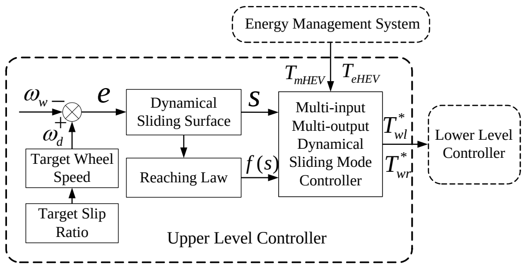


Figure3: Target driving torque control strategy

Taking the whole vehicle model as the control system

$$\begin{cases} \dot{x}_1 = a_{11}x_1^2 + a_{12}f_1(x_1, x_2) + a_{13}f_2(x_1, x_3) \\ \dot{x}_2 = a_{21}f_1(x_1, x_2) + a_{22}u_1 \\ \dot{x}_3 = a_{31}f_2(x_1, x_3) + a_{32}u_2 \end{cases} \tag{1}$$

where $x_1 = v_x$, $x_2 = \omega_{fl}$, $x_3 = \omega_{fr}$, $u_1 = T_{wl}^*$, $u_2 = T_{wr}^*$, $\mu_l(v_x, \omega_{fl}) = f_1(x_1, x_2)$, $\mu_r(v_x, \omega_{fr}) = f_2(x_1, x_3)$, $a_{11} = -\rho AC/2m$,

$a_{12} = a_{13} = \gamma_f g/2$, $a_{21} = a_{31} = -\gamma_f mgr_\omega / 2J_\omega$, $a_{22} = a_{32} = 1/J_\omega$, m is the vehicle mass, v_x is the vehicle longitudinal speed, J_ω is the inertia of the wheel assembly, r_ω is the wheel radius, ρ is the air density, A is the frontal area, C is the air resistance coefficient, γ_f is the front axle load proportion.

The control target is to make x_2 and x_3 to follow nx_1 together as

$$\begin{cases} x_2 = nx_1 \\ x_3 = nx_1 \end{cases} \tag{2}$$

where $n = 1/r_\omega(1-\lambda_d)$, λ_d is the optimal slip ratio.

The switching surface are

$$\begin{cases} s_1 = (x_2 - nx_1)' + c_1(x_2 - nx_1) + d_1 u_1 \\ s_2 = (x_3 - nx_1)' + c_2(x_3 - nx_1) + d_2 u_2 \end{cases} \tag{3}$$

where c_1 and c_2 are fixed positive constants, d_1 and d_2 are controller parameters, and $d_1, d_2 > 0$.

The reaching law is

$$\begin{cases} \dot{s}_1 = f(s_1) = -k_1 s_1 - \varepsilon_1 \operatorname{sgn}(s_1) \\ \dot{s}_2 = f(s_2) = -k_2 s_2 - \varepsilon_2 \operatorname{sgn}(s_2) \end{cases} \tag{4}$$

where k_1, k_2, ε_1 and ε_2 are positive.

Then the control law is

$$\begin{cases} \dot{u}_1 = \dfrac{1}{d_1}[-(x_2 - nx_1)'' - c_1(x_2 - nx_1)' + f(s_1)] \\ \dot{u}_2 = \dfrac{1}{d_2}[-(x_3 - nx_1)'' - c_2(x_3 - nx_1)' + f(s_2)] \end{cases} \tag{5}$$

4 Multi-Objective Dynamic Coordination Control for the Lower Level Controller

4.1 Multi-Objective Dynamic Coordination Control Strategy

As described in the introduction, on split-μ surfaces, MHEVTCS can not merely pursue maximum of vehicle's acceleration performance like ICETCS. It should take advantage of the adhesion condition of high adhesion side road to improve vehicle's acceleration performance without influencing its stability severely. According to this idea, a multi-objective dynamic coordination control strategy is proposed.

There are two control objectives in this strategy. The first objective is to satisfy vehicle's stability performance by decreasing hydraulic brake torque exerted on the low adhesion wheel as in

$$\min T_{bl}^* \tag{6}$$

The second objective is to improve vehicle's dynamic performance by increasing driving torque exerted on the wheels as in

$$\max T_{dw}^* \tag{7}$$

According to the wheels rotation dynamics and the control target of MHEVTCS which is to make the rotation speed of low adhesion wheel around the desired value and to ensure that high adhesion wheel dose not slip, the inequalities for constraints are

$$\begin{cases} 0.9T_{wl}^* \le T_{dw}^* - T_{bl}^* \le 1.1T_{wl}^* \\ T_{dw}^* \le T_{wh}^* \end{cases} \tag{8}$$

Merging these two control objectives together, this strategy can be described as one constrained multi-objective optimization problem, as shown in

$$\begin{aligned} & \min[T_{bl}^*, -T_{dw}^*] \\ & s.t.: \\ & \begin{cases} 0.9T_{wl}^* \le T_{dw}^* - T_{bl}^* \le 1.1T_{wl}^* \\ 0 < T_{dw}^* \le (T_{e\max} + T_{m\max}) i_g i_0 \\ T_{dw}^* \le T_{wh}^* \\ 0 \le T_{bl}^* \le T_{wh}^* - 0.9T_{wl}^* \end{cases} \end{aligned} \tag{9}$$

where $T_{e\max}$ is the maximum engine torque, $T_{m\max}$ is the maximum motor torque, i_g is the gear ratio, i_0 is the final drive ratio.

Hierarchical optimization method is used to solve this multi-objective optimization problem as the following

1) When the road adhesion coefficient is low or the vehicle speed is high, vehicle's stability performance should be focused on. Then the multi-objective optimization problem can be described as

$$\begin{aligned}&\min T_{bl}^{*}\\&\max T_{dw}^{*}\end{aligned}\tag{10}$$

2) When the road adhesion coefficient is high and the vehicle speed is low, vehicle's dynamic performance should be focused on. At this time, the multi-objective optimization problem can be described as

$$\begin{aligned}&\max T_{dw}^{*}\\&\min T_{bl}^{*}\end{aligned}\tag{11}$$

3) In the transient process, a fuzzy weighting $\xi \in [0,1]$ is defined to turn the multi-objective optimization problem as

$$\begin{aligned}&\min \xi T_{bl}^{*}-(1-\xi)T_{dw}^{*}\\&s.t.:\\&\begin{cases}0.9T_{wl}^{*}\le T_{dw}^{*}-T_{bl}^{*}\le 1.1T_{wl}^{*}\\0<T_{dw}^{*}\le (T_{e\max}+T_{m\max})i_g i_0\\T_{dw}^{*}\le T_{wh}^{*}\\0\le T_{bl}^{*}\le T_{wh}^{*}-0.9T_{wl}^{*}\end{cases}\end{aligned}\tag{12}$$

With the maximum road adhesion coefficient $\mu_{\max}$ and the vehicle speed v_e are set as the input of the fuzzy function, the parameter output of the fuzzy adaptor ξ is shown in Figure4.

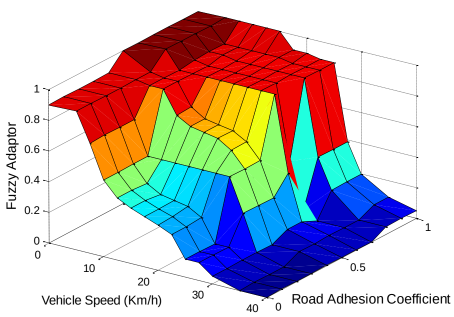


Figure4: Membership function

4.2 Brake Torque Control Strategy based on an Inverse Model

The brake torque demand T_{bl}^{*} should be converted to solenoid valve's opening demand, because the hydraulic brake unit can not achieve it directly. Therefore a hydraulic brake torque control strategy based on an inverse model is designed as shown in Figure5.

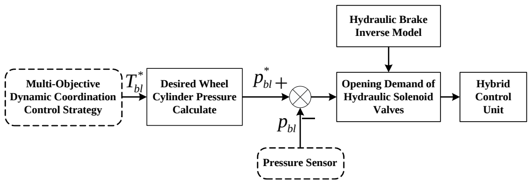


Figure5: Hydraulic brake torque control strategy

Firstly, the brake torque demand T_{bl}^{*} is converted to the desired wheel cylinder pressure p_{bl}^{*} according to friction coefficient of the brake system which is measured by tests. Then, solenoid valve's opening demand is picked up on the basis of an inverse model of wheel cylinder pressure adjusting mechanism as shown in Formula 5 and the differences between the desired wheel cylinder pressure p_{bl}^{*} and the actual wheel cylinder pressure p_{bl}.

$$t_a=\frac{\max(p_{bl}^{*}-p_{bl},0)}{B_1(p_{acc}-p_{bl})^{d_1}}\tag{13}$$

$$t_d=\frac{\max(p_{bl}-p_{bl}^{*},0)}{B_2 p_{bl}^{d_2}}\tag{14}$$

4.3 Target Engine Torque Design Strategy

After getting driving torque demand exerted on the wheels T_{dw}^{*} in the multi-objective dynamic coordination control strategy, the desired driving total torque T_d^{*} which is provided by engine and motor can be calculated

$$T_d^{*}=T_{dw}^{*}/i_g/i_0/\eta\tag{15}$$

For HEV, engine is the main power supply. The most effective method to avoid wheels slipping of HEV is to reduce engine torque, but its response speed and control accuracy are worse than motor. Therefore a target engine torque design strategy based on low pass filtering is proposed. In this strategy, low frequency part of the desired driving torque is provided by engine and high frequency part of the desired driving torque is compensated by motor. The filtering method is

$$T_{dc}^{*}(k) = T_{dc}^{*}(k-1) + [T_{sc}^{*}(k) - T_{dc}^{*}(k-1)]\frac{T_s}{T_f}(1-\frac{T_s}{2T_f}) \tag{16}$$

where T_{dc}^{*} is the desired driving torque after filtering, T_{sc}^{*} is the desired driving torque before filtering, T_s is the control period, T_f is the filtering time constant.

At the same time, the engine torque demand T_{eTCS} should be smaller than the desired engine torque T_{eHEV} which is calculated by an energy management system, otherwise engine torque will increase which is against the control aim of traction control systems to decrease driving torque.

To sum up, the engine torque demand T_{eTCS} can be formulated as

$$T_{eTCS} = \min(T_{dc}^{*}, T_{eHEV}) \tag{17}$$

4.4 Torque Dynamic Coordinate Control Strategy

The aim of this torque dynamic coordinate control strategy is to achieve the desired driving torque rapidly and accurately. According to such requirements as fast response, accuracy, robustness and linear input/output transfer characteristics, a torque dynamic coordinate control strategy based on model matching 2-DOF control[7] is proposed as shown in Figure6

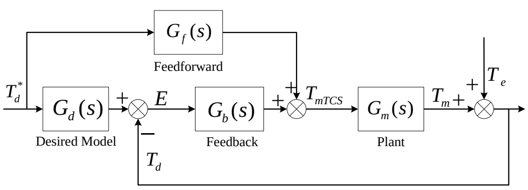


Figure6: Torque dynamic coordinate control strategy based on model matching 2-DOF control

where T_d is the actual driving total torque.

In this strategy, the control plant is the motor model whose torque characteristics is

$$T_m = \frac{1}{\tau_m s+1}T_{mTCS} = G_m(s)T_{mTCS} \tag{18}$$

where τ_m is the really time constant of motor torque output.

The desired model is formulated as

$$G_d(s) = \frac{1}{\tau_d s+1} \tag{19}$$

where τ_d is the really time constant.

Then the feedforward transfer function can be calculated as

$$G_f(s) = \frac{G_d(s)}{G_m(s)} = \frac{\tau_m s+1}{\tau_d s+1} \tag{20}$$

Additionally, the PID control method is used in the feedback controller.

5 Simulations and Analysis

In order to evaluate the MHEVTCS, a simulation platform is built as shown in Figure7 including driver model, energy management controller model, MHEVTCS controller model, powertrain model and 15-DOF vehicle model.

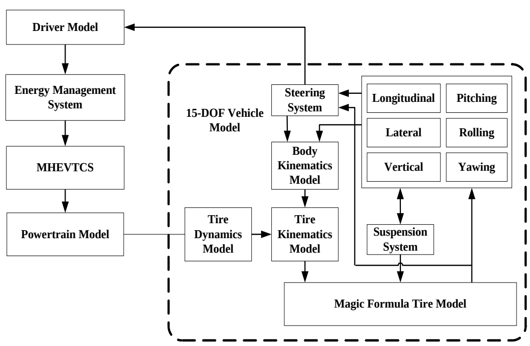


Figure7: Simulation platform

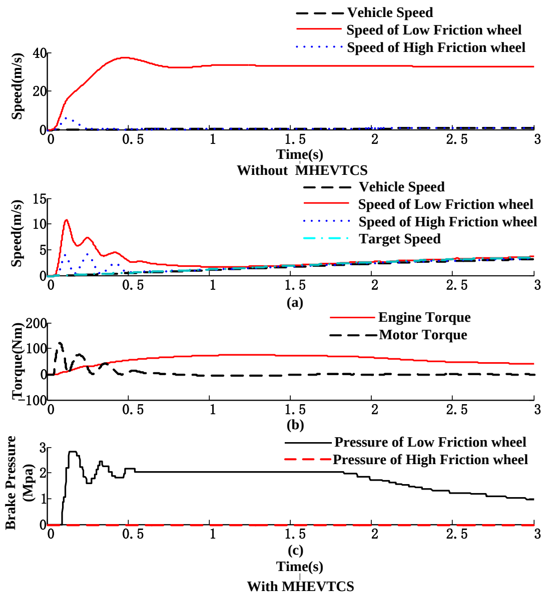
Two sets of MHEVTCS simulation are defined as shown in Table1.

Table1: Two sets of MHEVTCS simulation

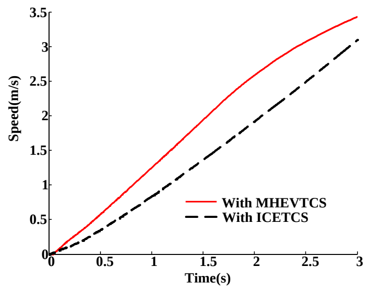
	Initial Speed	Gear Ratio	Adhesion Coefficient	Accelerator Pedal
S1	0 m/s	1	Left 0.1/ Right 0.8	100%

S2	5 m/s	2	Left 0.1/ Right 0.8	100%

In S1, the speeds of two slipping wheels are respectively controlled at target values after 0.5s and 1s with MHEVTCS as shown in Figure8. Acceleration performance of HEV increases 220% than without MHEVTCS. Vehicle speed comparison result shows that with ICETCS vehicle speed is only 3.1m/s at the end of simulation, and with MHEVTCS it achieves 3.43m/s. Vehicle's acceleration performance is improved 11% by MHEVTCS than by ICETCS.



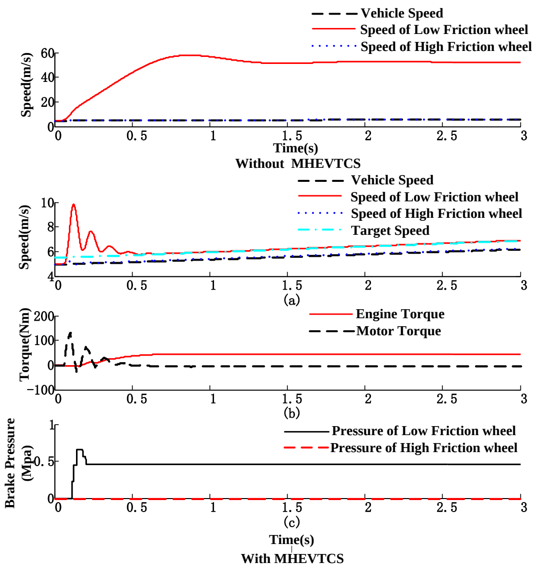
a) Control result



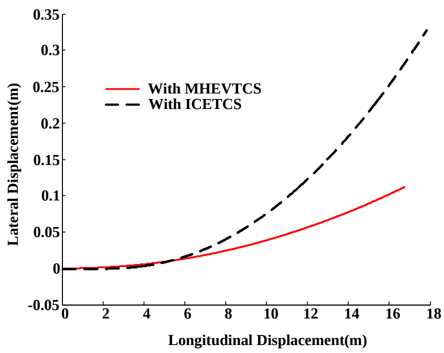
b) Speed comparison result

Figure8: Simulation results of S1

In S2, speed of the slipping wheel on low adhesion side road is controlled at target value after 0.4s with MHEVTCS as shown in Figure9. Acceleration performance of HEV increases 5% than without MHEVTCS. Lateral displacement comparison result shows that with ICETCS lateral displacement achieves 0.35m at the end of simulation, and it is only 0.1m/s with MHEVTCS. Vehicle's stability performance is improved by MHEVTCS.



a) Control result



b) Lateral displacement comparison result

Figure9: Simulation results of S2

Furthermore, one hardware-in-loop test platform is built as shown in Figure10.

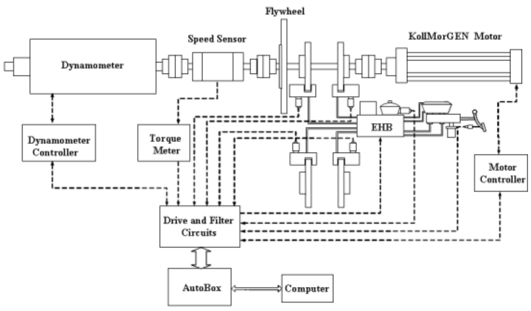


Figure10: Hardware-in-loop test platform

On this hardware-in-loop test platform, one set of tests is defined in Table2.

Table2: One set of hardware-in-loop tests

	Initial Speed	Gear Ratio	Adhesion Coefficient	Accelerator Pedal
S1	0 m/s	1	Left 0.1/ Right 0.8	50%

Figure11 shows without MHEVTCS, the speed of low friction wheel achieves 18m/s and it is serious slipping. At the end of the test, vehicle speed is only 0.6m/s. The vehicle can not start as normal.

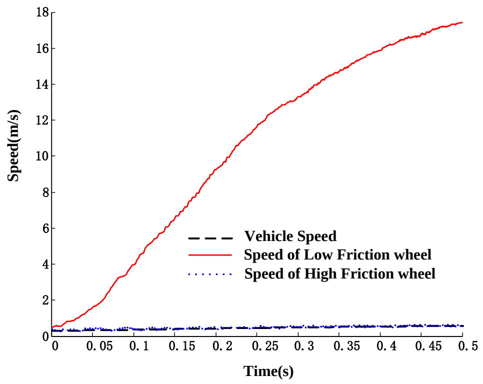


Figure11: Hardware-in-loop test result without MHEVTCS

Figure12 shows that the speed of slipping wheel on low adhesion side road is controlled at target value after 0.5s with MHEVTCS. At the end of the test, vehicle speed achieves 4.8m/s. Acceleration performance and starting ability of HEV are greatly improved by MHEVTCS.

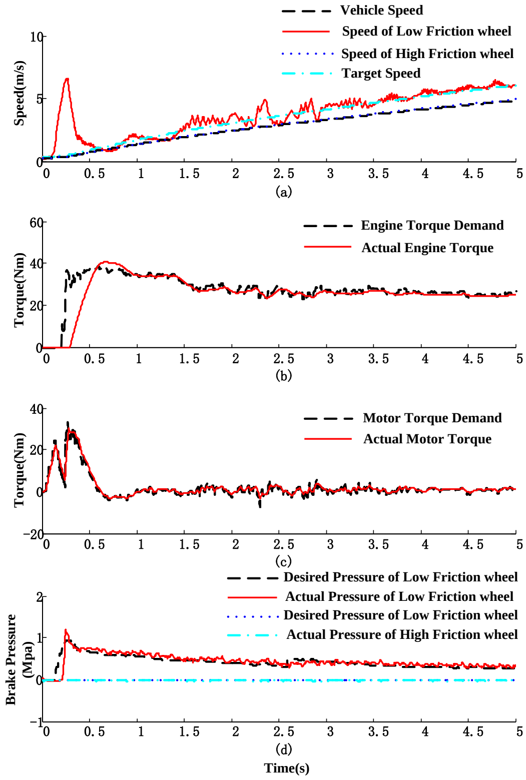


Figure12: Hardware-in-loop test result with MHEVTCS

6 Conclusions

In this paper, a traction control method for an ISG hybrid electric vehicle is explored. The conclusions are followed:

1) Slipping wheels can be controlled by MHEVTCS quickly, accurately and smoothly.

2) Compared with conventional traction control systems, MHEVTCS improves vehicle's acceleration performance greatly without influencing its stability severely.

3) The dynamic coordinate control problem among engine, motor and hydraulic system is solved by MHEVTCS.

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