

Research on hierarchical collaborative control of driverless hybrid tractor trajectory tracking and energy management system

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Abstract. The energy consumption economy and trajectory tracking effect of driverless hybrid tractors are important performance indicators. Currently, separate methods are used to control trajectory tracking and energy management independently while ignoring the interaction between the two variables, resulting in reduced tractor performance. To address this problem, this study proposes a hierarchical collaborative control method (HCCM) for tractor trajectory tracking and energy management. Firstly, construct the longitudinal and transverse kinematics model. Taking the tractor's front wheel rotation angle and speed change as constraints and minimizing lateral deviation as the goal, a trajectory tracking strategy (upper layer) is proposed based on Model Predictive Control. Secondly, with the motor and diesel engine speed and torque control variables, power battery SOC as a constraint, and minimizing the total energy consumption cost as the goal, a three-stage dynamic programming energy management strategy (lower layer) is designed based on dynamic programming. On this basis, using velocity as an interactive variable and then integrating upper-layer and lower-layer control strategies, a dual-system HCCM is proposed to simultaneously improve the tractor's trajectory tracking accuracy and energy consumption economy. Finally, mutually independent pure pursuit tracking algorithms and power following energy management strategies are selected for comparison. Simulation shows that the proposed dual-system hierarchical collaborative control method improves trajectory tracking performance and reduces the total energy cost of the entire machine. Under rotary tillage conditions and transportation conditions, the trajectory tracking accuracy increases by 75.02% and 80.36%, and the cost of total energy consumption reduces by 8.48% and 9.01%, respectively.

Key words: driverless hybrid tractor; trajectory tracking; energy management; hierarchical collaborative control method

1. INTRODUCTION

As one of the main tools of agricultural production, the development of the tractor has become an important way to improve agricultural production efficiency and realize agricultural modernization. With the continuous advancement and application of science and technology, unmanned driving and energy-saving technologies have become an important direction for the development of tractors [1-3]. Governments of various countries have drafted policies for the intelligent development of agricultural machinery to encourage enterprises to increase investment in research and development. Further, improves the level of information collection of agricultural machinery and equipment, and improves the ability of precise operation. The automation of agricultural machinery is the basis of precision agriculture. The higher the degree of intelligence, the more conducive to the promotion of precision agriculture [4,5]. Driverless tractors can improve agricultural production

efficiency and safety, while hybrid tractors can reduce energy consumption and emissions, achieving green and environmental goals [6,7]. Therefore, the development of driverless hybrid tractors can well solve the current problems.

Trajectory tracking performance and energy consumption economy are important performance indicators of driver less hybrid tractors and are also hot topics in current research [8].

More emphasis is placed on job tracking accuracy in the trajectory tracking control of driverless tractors. Trajectory tracking control methods mainly include pure pursuit control, proportional-integral-derivative (PID) control, fuzzy control, model predictive control (MPC), and so on [9,10]. Researchers have extensively studied methods for controlling trajectory tracking. Huang et al. [11] studied the pure pursuit control method and neural network, and developed a trajectory tracking algorithm suitable for agricultural tractor headland steering control. The simulation results suggest that

the control method is viable. However, this control method has certain requirements for the continuity of the reference trajectory in practical applications, and the stability and smoothness of trajectory tracking are poor. To solve the reference trajectory continuity requirement, Zhang et al. [12] adopted the integral absolute error (IAE) performance index as the evaluation criterion and developed an optimal fractional order proportional differentiation (FOPD) controller for trajectory tracking control of a specific tractor. The simulation results show that the optimal FOPD controller reduces the IAE value of the closed loop system, improves the IAE performance, and causes fewer crops to be crushed. However, the numerical solution of fractional differential equations is unstable and computationally complex, which places high demands on the controller. Accordingly, Liu et al. [13] developed a fuzzy controller for agricultural tractors using an improved particle swarm optimization algorithm. Accurate tracking of the trajectory is realized under the action of the controller, and the controller operation process is stable and reliable. However, to improve tracking accuracy in fuzzy control, increasing the quantization levels is necessary. This requires a larger rule search range, and the decision-making will be slower. In serious cases, real-time control will not be possible. To address this problem, Hu [14] proposed a trajectory-tracking control algorithm on the basis of adaptive time domain MPC. According to the current velocity of the tractor and the curvature of the reference path, the optimized control horizon and prediction horizon are determined through the fuzzy control algorithm, and combined with the MPC algorithm to control the tractor. The results show that the designed controller can ensure the lateral error within 5 cm in response to different working conditions, and does not require the controller to maintain a large control horizon and prediction horizon. This improves calculation speed and is suitable for tractor trajectory tracking. Different from the optimization approaches that enhance adaptability by dynamically adjusting time-domain parameters, Zhang et al. [15] improved MPC from the perspective of model reconstruction. To address the performance degradation of traditional MPC under time-varying soil conditions, they proposed an MPC method based on lateral/yaw error. By constructing a low-dimensional error tracking model and incorporating a steering delay state equation, the controller output was directly associated with the front wheel steering angle. Both simulation and real-vehicle experiments demonstrated that the lateral error could be maintained within ± 0.02 m.

The hybrid tractor power system works in a more efficient and environmentally friendly way by mechanically coupling the diesel engine and motor [16]. Among them, the energy management strategy is a critical control strategy for the hybrid tractor, as it directly affects the overall performance. At present, there are two main types of energy management strategies: rule-based control strategies and optimization-based control strategies [17, 18]. Xu et al. [19] developed an energy management strategy for fuel cell/battery hybrid tractors using fuzzy control. The study showed that the equivalent hydrogen consumption is reduced by 12.84% and 11.11% under the conditions of field

operation and transportation operation, respectively. However, this method uses a series of rules to control the control amount at each moment, and cannot achieve the optimal treatment of the extreme value of the control target. Energy utilization efficiency can be further improved by adopting optimization-based energy management strategies. Geng et al. [20] proposed a multi-objective optimal energy management strategy for power-split hybrid electric vehicles on the basis of particle swarm optimization. Achieve the energy management goal of reducing vehicle energy consumption and controlling battery power. However, it is important to note that the method based on instantaneous optimization is not necessarily the global optimum in the end. To address this problem, Zhang et al. [21] took hybrid tractors equipped with continuously variable transmission (CVT) as the research object and proposed a global energy management optimization strategy on the basis of dynamic programming. The simulation results indicated that the developed energy management strategy can make the motor and diesel engine operate in the optimal range, effectively reducing the total energy consumption cost of the tractor during fieldwork. Although dynamic programming improves tractor energy utilization efficiency, the control strategy exhibits inadequate real-time performance and is unsuitable for online control. Realtime performance can be improved by reducing the number of dynamic planning stages [22]. Zhu et al. [23] addressed the issues of fixed equivalent factors in traditional ECMS and the difficulty in balancing battery SOC under time-varying field loads by designing an F-ECMS energy management strategy based on SOC state feedback. This strategy employs a PI controller to adjust the equivalent factor in real time for instantaneous optimization. Although this approach has low computational load and is suitable for online real-time implementation, the PI control parameters require manual trial-and-error tuning, and there remains room for improvement in control stability under severe sudden field load changes.

In summary, numerous independent studies have been reported on trajectory tracking strategies and energy management strategies, and good results have been achieved. In terms of trajectory tracking, existing studies mainly focus on improving path tracking accuracy, with methods including pure pursuit control, fuzzy control, MPC control and so on. However, these approaches have a single control objective and fail to consider energy consumption during the tracking process. In terms of energy management, most existing studies center on fuel economy optimization and battery SOC maintenance, adopting methods such as rule-based fuzzy control, particle swarm optimization-based multi-objective optimization, and dynamic programming. Nevertheless, most optimization methods rely on prior knowledge of working conditions and can hardly account for the effect of velocity variation on tracking performance. It is worth noting that velocity is not only a critical parameter affecting tracking accuracy but also a core variable determining power demand. Yet existing studies have not exploited this coupling relationship to realize collaborative optimization of the two systems, nor have they fully addressed the degradation of overall machine performance caused by independent control

schemes. Currently, research on the coupling effect between the trajectory tracking system and the energy management system remains relatively limited. In response to the above problems, this study takes a driverless hybrid tractor as the object and uses velocity as the interactive variable to propose a tractor dual-system hierarchical collaborative control method (HCCM) [24-28]. The upper layer is trajectory tracking based on MPC, and the lower layer is energy management on the basis of three-stage dynamic programming. This study aims to comprehensively consider the energy management and trajectory tracking of driverless hybrid tractors, and improve the trajectory tracking performance and energy consumption economy. The innovations of this study are as follows: (a) A three-stage dynamic programming strategy is designed to cut the energy consumption of the tractor. (b) Based on mining the interactive variables of the energy management system and the trajectory tracking system, a dual-system HCCM integrating model predictive control is proposed, which simultaneously improves the tractor's trajectory tracking precision and energy consumption economy. Finally, the mutually independent pure pursuit tracking trajectory method (PP-TTM) and the power following energy management strategy (PF-EMS) are selected for comparison. It is verified that the HCCM can simultaneously improve trajectory tracking performance and reduce the total energy consumption cost of the driverless hybrid tractor.

The following sections present the rest of this study. Section 2 introduces the topology structure of the driverless hybrid tractor and its primary parameters. The primary components of the tractor are modelled in Section 3. Section 4 explains the control strategy construction process. Section 5 provides results and discussions. In the end, the conclusions of the study are stated in Section 6.

2. MATERIALS AND METHODS

Firstly, the topology structure of a driverless hybrid tractor in this study and its main parameters were determined. Then, its main components and system models were constructed. Finally, the development of the control method is described.

2.1. Driverless hybrid tractor topology structure and main parameters

Figure 1 demonstrates the topology structure of the driverless tractor that this study investigated, with its power transmission and smart aesthesis systems. The power transmission system consists of various components, including the motor, power battery, diesel engine, torque coupler, central drive, CVT, PTO, vehicle controller, and others. The smart aesthesis system comprises components including the industrial computer, gyroscope, velocity sensor, laser radar, binocular camera, and others.

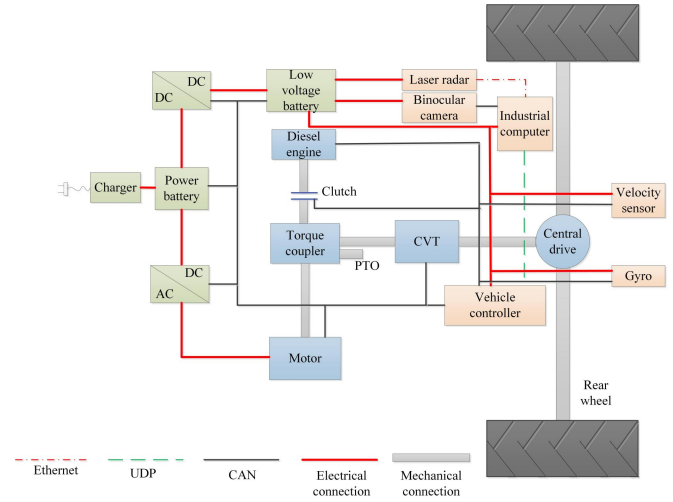


Fig.1. Topology structure diagram of driverless hybrid tractor

The power battery is electrically connected to the low-voltage battery through the DC/DC module and is used for charging the low-voltage battery. The power battery is connected to the motor through the AC/DC converter to enable the exchange of energy between electrical energy and mechanical energy. The low-voltage battery is connected to the laser radar, binocular camera, velocity sensor, gyroscope, vehicle controller, and industrial control electromechanical components to provide electric energy for them. The laser radar is linked to the computer through Ethernet and is used to detect the distance between the tractor's main body and any obstacles. The binocular camera is linked to the computer through the CAN bus and is used to collect information on objects along the route of the tractor's main body. The gyroscope is used to collect angle signals of the tractor's main body, such as roll and pitch, in a multi-degree-of-freedom space. The velocity sensor is used to collect the operating velocity of the tractor. The diesel engine, torque coupler, motor, clutch, CVT, PTO, and central drive are mechanically connected to transmit power. The central drive propels the rear wheels to move the hybrid tractor.

Unlike cars or engineering vehicles, tractors are primarily used for farm work and road transport. There is a significant difference in the resistance load under different working conditions [29]. The driverless hybrid tractor's main components are chosen based on its working conditions [30]. Table 1 displays the specific parameters.

2.2. Main parameters of driverless hybrid tractor

TABLE 1. Main parameters of driverless hybrid tractor

Name	Parameter	Symbol	Units	Value
Basic parameters	Machine quality	m	kg	6280
	Total mass of trailer and cargo	mh	kg	20000
	Drive wheel radius	R	m	0.915
	Wheelbase	l	m	2.90
	Field rolling resistance coefficient	f_r	—	0.017

	Road rolling resistance coefficient	–	0.080
Diesel engine	Maximum power	$P_{\max}$	kW 58
	Minimum torque	$T_{\max}$	Nm 280
	Rated speed	n_{rated}	rpm 2200
Motor	Maximum power	$P_{m\max}$	kW 32
	Minimum torque	$T_{m\max}$	Nm 114
	Rated speed	$n_{m\text{rated}}$	rpm 2700
Power Battery	Rated capacity	C	Ah 70
	Rated voltage	U_{rated}	V 352
CVT	Transmission ratio range	i_{cvt}	– [0.842, 9.684]
Central drive	Transmission ratio	i_{zy}	– 21.27

3. MODELS OF THE DRIVERLESS HYBRID TRACTOR

The topological structure of the driverless hybrid tractor is used to establish models for its primary components. These models include the transmission system model, transportation dynamic model, dynamic model of rotary tiller, longitudinal and transverse coupled kinematics model, trajectory tracking linear error model, and power system component model. Finally, a simulation model for the entire machine is constructed.

3.1. Transmission system model of the tractor

The tractor transmission system is the general term for all power transmission modules between the two power sources and the wheels. During power delivery, the speed slows down to the wheels. When evaluating the fuel economy of a tractor, the establishment of models only considers the power transmission relationship between the central drive and the CVT, while the influence of other components is disregarded. The torque required for the driverless tractor is obtained from both the motor and diesel engine, which can be derived as:

$$T_{\text{req}} = T_e + T_m \quad (1)$$

where T_{req} indicates the required torque of the torque coupler; T_e and T_m are the motor and diesel engine required torques, respectively.

The torque transmission relationship between the torque coupler and the drive wheel is as follows:

$$T_{\text{req}} = \frac{T_w}{i_{\text{cvt}} i_{zy} \eta_{\text{cvt}} \eta_{zy} \eta_0} \quad (2)$$

where T_w indicates the torque of the drive wheel; i_{cvt} and i_{zy} are the ratios of CVT and central drive, respectively; η_{cvt} , η_{zy} , and η_0 are the efficiencies of the CVT, central drive, and torque coupler, respectively.

The power system speed transmission relationship is as follows:

$$n_e = n_{\text{req}} = n_w i_{\text{cvt}} i_{zy} \quad (3)$$

$$n_w = \frac{v}{0.377R} \quad (4)$$

where n_w is the speed of the drive wheel; n_e and n_{req} denote the diesel engine speed and required speed, respectively; R indicates the driving wheel radius; v indicates the operating velocity.

The CVT model mainly considers the impact of the transmission ratio and CVT efficiency on the final torque transmitted to the wheels, so only the CVT transmission efficiency is considered. Ignoring its dynamic response characteristics, a numerical model for CVT was established using bench test data [30].

The efficiency of a CVT transmission is linked to its input gear ratio and torque, which can be represented as follows:

$$\eta_{\text{cvt}} = f_{\text{cvt}}(T_{\text{cvt-in}}, i_{\text{cvt}}) \quad (5)$$

where $T_{\text{cvt-in}}$ represents the CVT input torque.

The relationship between CVT efficiency, torque, and speed is determined using the interpolation fitting method. Fig.2 shows a numerical model of CVT efficiency.

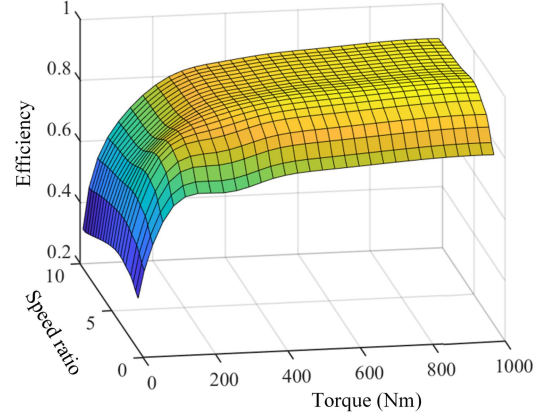


Fig. 2. Efficiency model diagram of CVT

The formula for calculating the output torque and speed of a CVT is as follows:

$$\begin{cases} T_{\text{cvt-out}} = i_{\text{cvt}} \cdot T_{\text{cvt-in}} \cdot \eta_{\text{cvt}} \\ n_{\text{cvt-out}} = \frac{n_{\text{cvt-in}}}{i_{\text{cvt}}} \end{cases} \quad (6)$$

where $T_{\text{cvt-out}}$ represents the CVT output torque; $n_{\text{cvt-out}}$ and $n_{\text{cvt-in}}$ indicate the output and input speeds of the CVT, respectively.

3.2. Transportation dynamic model

The transportation dynamic model mainly studies the relationship between the longitudinal motion of the tractor and its force, ignoring the lateral force factor of the hybrid tractor. Only calculate the longitudinal force, the balance of tractor driving is as follows:

$$F_t = F_f + F_w + F_i + F_j \quad (7)$$

where F_t denotes the propulsion force of the tractor; F_f , F_i , F_w , and F_j are rolling resistance, grade resistance, air resistance, and acceleration resistance of the tractor, respectively. 7 can be expanded into the following as:

$$\frac{T_w}{r} = f(m + m_h)g + \frac{C_D A v^2}{21.15} + (m + m_h)g \sin \alpha + \delta(m + m_h) \frac{d_v}{d_t} \quad (8)$$

where m represents the tractor use quality; m_h is the total mass of trailer unit and cargo, f indicates the rolling resistance coefficient; α represents the slope; C_D , A , and δ are the drag coefficient, frontal area and mass conversion factor of the tractor, respectively.

Air resistance can be ignored when the driverless tractor is working at low velocity. When driving at a constant velocity, the tractor's acceleration resistance can be disregarded [31].

3.3. Dynamic model of rotary tiller

Combined with the topological structure of the driverless hybrid tractor in Figure 1, based on the dynamic analysis, the power balance relationship of the tractor during rotary tillage operations is calculated. The following expression represents:

$$P_{req}(T_{req}, n_{req}) = \frac{(\frac{P_{drive}}{\eta_{zy}\eta_{cvt}} + P_r)}{\eta_0} \quad (9)$$

$$P_{drive} = \frac{(mgf \cos \alpha + mg \sin \alpha + m\delta v + \frac{C_D A v^2}{21.15})v}{3600} \quad (10)$$

$$P_r = \frac{P_c + P_{th} + P_a + P_h}{\eta_r} \quad (11)$$

where P_{req} indicates the required power of the driverless tractor; P_{drive} and P_r represent the driving power of the driverless tractor and the power consumption of the rotary tiller, respectively; P_c , P_a , P_{th} , and P_h are the power consumed during soil cutting, forward movement of the rotary tiller, soil throwing, and overcoming the soil's horizontal reaction force, respectively.

This driverless hybrid tractor is equipped with a dual-acting clutch, allowing for separate manipulation of the PTO [32]. During the rotary tillage work, the unit has the following relationship:

$$P_r = P_{PTO} = \frac{n_{PTO} T_{PTO}}{9550} \quad (12)$$

where P_{PTO} denotes the power consumption of PTO; n_{PTO} and T_{PTO} are the speed and torque of the PTO, respectively.

3.4. Longitudinal and transverse coupled kinematics model

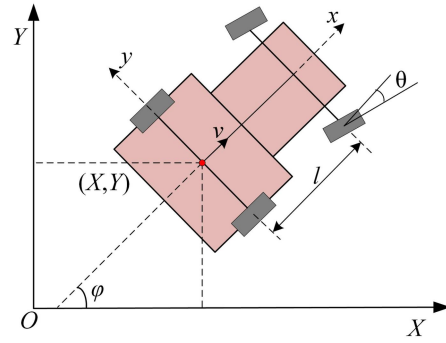


Fig. 3. Longitudinal and transverse coupled kinematics model

The longitudinal and transverse coupled kinematics model established for the tractor is shown in Fig.3[33,34]. The inertial coordinate system is defined as XOY, while the coordinate system for the tractor body is xoy. Amongst them, v is the velocity of the vehicle at the centre rear axle; l is the length of the wheelbase; θ and ϕ are the front wheel slip angle and tractor yaw angle, respectively.

Assuming the traveling velocity of the tractor is v , then according to the kinematic constraint analysis of the agricultural machinery, it can be obtained:

$$\begin{cases} \dot{X} = v \cos \phi \\ \dot{Y} = v \sin \phi \\ \dot{\phi} = \frac{v \tan \theta}{l} \end{cases} \quad (13)$$

where $\dot{X}$ and $\dot{Y}$ denote the velocities of the tractor in the X-axis and Y-axis directions, respectively.

Rewrite 13 into matrix representation, the vertical and horizontal coupling kinematics model of the tractor is obtained:

$$\begin{bmatrix} \dot{X} \\ \dot{Y} \\ \dot{\phi} \end{bmatrix} = \begin{bmatrix} v \cos \phi \\ v \sin \phi \\ \frac{v \tan \theta}{l} \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \end{bmatrix} \quad (14)$$

3.5. Trajectory tracking linear error model

The errors in tractor trajectory tracking include lateral deviation e and heading deviation e_ϕ . The established tractor linear error model is shown in Fig.4. Amongst them, the red line is the expected trajectory; point P1 indicates the projection point of the tractor's rear axle center on the trajectory; s is the arc length between any arbitrary reference point P0 and P1 on the trajectory; d_{ref} is the curvature of the arc length at point P1.

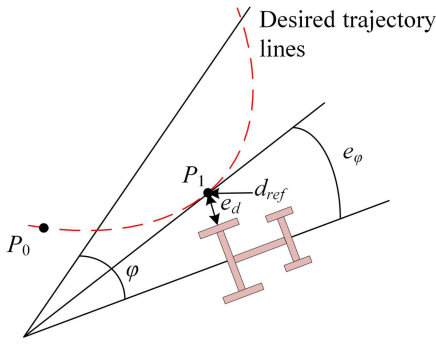


Fig. 4. Trajectory tracking linear error model

According to kinematic analysis, the tractor's moving velocity s at P_1 is calculated as follows:

$$s = \frac{(v_x \cos e_\phi + v_y \sin e_\phi)}{1 - d_{ref} e_d} \quad (15)$$

where v_x and v_y denote the velocity components of the tractor in the x-axis and y-axis in the body coordinate system, respectively.

Since the deviation of the heading e_ϕ is small, it is treated as limiting infinity and the lateral velocity of the farm machine v_y is neglected [32]. The error model for tractor trajectory tracking can be simplified as:

$$\begin{bmatrix} \dot{e}_\phi \\ \dot{e}_d \end{bmatrix} = \begin{bmatrix} 0 & 0 \\ v_x & 0 \end{bmatrix} \begin{bmatrix} e_\phi \\ e_d \end{bmatrix} + \begin{bmatrix} \frac{v_x}{l} \\ 0 \end{bmatrix} \tan \theta + \begin{bmatrix} -v_x \\ 0 \end{bmatrix} d_{ref} \quad (16)$$

3.6. Power system component model

The power system component of the driverless hybrid tractor includes three major components: power battery, diesel engine, and motor, which are the source of power for tractor operations. The power battery provides an energy source for the motor when it is driven and stores energy for the motor when it is charging. Most power battery models use equivalent circuit models because they are highly applicable to a variety of power battery work conditions and can derive the state of the model [35]. This study uses the Thevenin equivalent circuit model as shown in Fig.5. This model has a simple structure, takes up less computing resources, and has high simulation accuracy for the dynamic characteristics of the power battery. Compared with the simple internal resistance model, the first-order Thevenin model adopted in this study, which meets the real-time computational requirements, incorporates an additional RC parallel branch to simulate the polarization effect during battery charging and discharging.

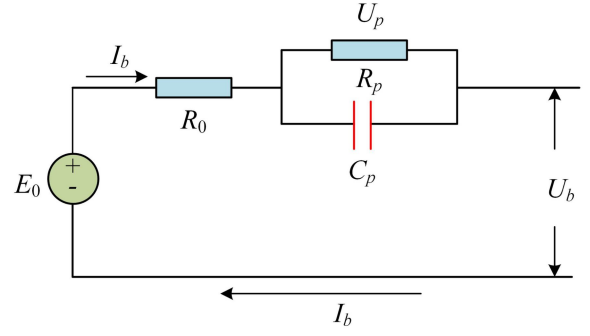


Fig. 5. Power battery Thevenin model

In accordance with Kirchhoff's law, the following relationship can be obtained from the Thevenin equivalent circuit model:

$$E_0 = U_b + R_0 I_b + U_p \quad (17)$$

$$I_b = \frac{U_p}{R_p} + C_p \frac{dU_p}{dt} \quad (18)$$

where E_0 represents the open circuit voltage of the power battery; U_b denotes the terminal voltage of the power battery; I_b and R_0 indicate the output current and internal resistance of the power battery, respectively. R_p and C_p are the polarizing resistance and the polarizing capacitance of the power battery, respectively. C_p and R_p form a first-order RC circuit. U_p is the polarization voltage of the RC combination.

The power battery power is calculated as follows:

$$P_{bat} = \begin{cases} \frac{P_m}{\eta_{bat}}, & P_m > 0 \\ P_m \eta_{bat}, & P_m < 0 \end{cases} \quad (19)$$

where η_{bat} indicates the discharging and charging efficiency of the power battery; P_{bat} indicates the power of the power battery.

The change in SOC of the power battery is calculated using the ampere-hour integration method. The temperature effect on the Coulomb factor η_w and the charge and discharge current effect on the Coulomb factor η_i are added. Among them, the Coulomb factor η_w affected by temperature and the Coulomb factor η_i affected by charge and discharge current are obtained through experimental tests [36]. The improved calculation formula of the ampere-hour integral method is:

$$SOC(t) = SOC_0 - \frac{\int_{t_0}^t I_b(t) dt}{3600C} \eta_w \eta_i \quad (20)$$

where SOC_0 represents the initial value of SOC. C indicates the power battery rated capacity.

On the basis of the test data, the numerical model of the diesel engine and the motor are constructed using the data fitting interpolation method [37]. The fuel consumption of diesel engines is determined by speed and torque as independent variables. This numerical model is difficult to

express explicitly, therefore, it is presented in the form of implicit functions, the expressed formula is as follows:

$$\dot{m}_e = f_e(T_e, n_e) \quad (21)$$

where $\dot{m}_e$ is the fuel consumption rate of the diesel engine.

Motors usually have high working efficiency, and their working status is related to the distribution of working points in the efficiency MAP diagram, which can be expressed as:

$$\eta_m = f_m(T_m, n_m) \quad (22)$$

where η_m indicates the motor working efficiency; n_m indicates the motor speed.

3.7. Tractor simulation model

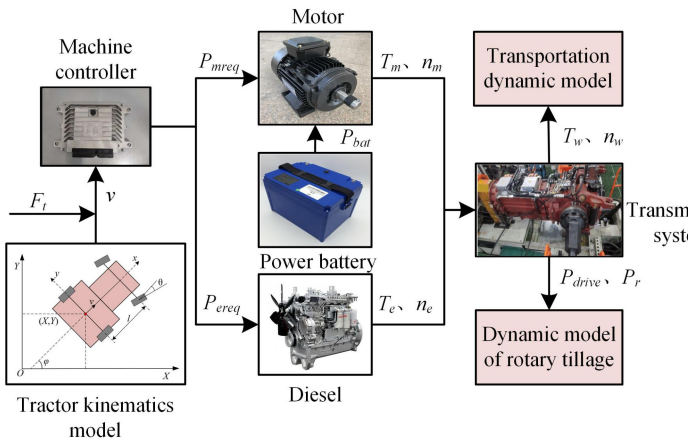


Fig. 6. Tractor simulation model

The tractor simulation model is shown in Fig. 6. The consists of simulation model including tractor kinematics model, diesel engine model, battery model, transmission system model, dynamic model of rotary tillage, and transportation dynamic model. During the working of the driverless hybrid tractor, the vehicle controller acquires the tractor's velocity through the tractor kinematics model. On the basis of the working conditions, the required driving force is determined using 7, or the required power of the tractor is determined using 9. Subsequently, within the vehicle controller, the required power for the tractor is allocated based on the designed control strategy, which includes the proposed and compared strategies in this study. The respective power requirements for the diesel engine and motor are output as inputs to the diesel engine model and motor model. The diesel engine and motor work according to the instructions of the vehicle controller and output the respective required torques and speeds through the transmission system model to transmit power for the dynamic model of the transportation or rotary tillage. Simultaneously, the power battery model transfers energy based on the required power of the motor model.

4. CONTROL STRATEGY DESIGN

Firstly, a tracking method based on MPC is designed for the upper-layer trajectory tracking part. Then, the lower-layer energy management part designs a three-stage dynamic programming control strategy. Finally, using velocity as the interactive variable and integrating the two control strategies, a tractor dual-system HCCM is proposed to conduct collaborative research on the tractor's trajectory tracking and energy management.

4.1. Upper layer trajectory tracking

The controller for trajectory tracking is composed of two main parts: the MPC controller and the tractor kinematics model. The MPC controller is the core of trajectory tracking control and has three characteristics: model prediction, rolling optimization, and feedback correction. It can promptly compensate for uncertainties caused by model interference, mismatch, and so on. Also, it exhibits excellent dynamic performance. Fig. 7 depicts the schematic block of the MPC in the driverless hybrid tractor system.

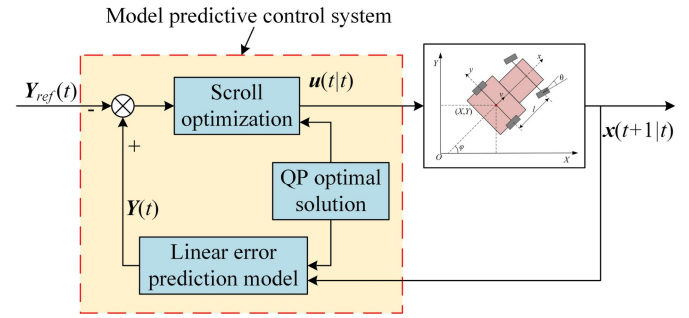


Fig. 7. MPC schematic block diagram

The designed MPC controller combines the lateral control and longitudinal control of the tractor, and obtains the optimal control quantities of the front wheel angle and acceleration through solving. Then, input it into the tractor kinematics model to obtain the current status of the tractor as the input of the prediction control system. Naturally, the process described above is repeated to achieve trajectory tracking.

According to the longitudinal and transverse coupling kinematics model, the deviations of the state quantity and control quantity for the tractor are as follows:

$$\tilde{x} = \begin{bmatrix} \dot{x} - \dot{x}_r \\ \dot{y} - \dot{y}_r \\ \dot{\phi} - \dot{\phi}_r \end{bmatrix} \quad (23)$$

$$\tilde{u} = \begin{bmatrix} v - v_r \\ \theta - \theta_r \end{bmatrix} \quad (24)$$

The core theory of MPC is to predict changes in state quantities within a period of time in the future by establishing a kinematic model of tractors. To achieve a more robust tracking effect, the state quantities of the tractor system need to be controlled. Referring to the "soft constraint" method introduced by Li et al. [38,39], the MPC-based trajectory tracking controller adopts an objective function such as 25 to

achieve precision tracking of the desired trajectory by minimizing the objective function.

$$I(t|t) = \sum_{i=1}^{N_p} \|\eta(t+i|t) - \eta_r(t+i|t)\|_Q^2 + \sum_{i=1}^{N_c-l} \|\Delta U(t+i|t)\|_R^2 + \rho \varepsilon^2 \quad (25)$$

where N_c and N_p denote the control time domain and prediction time domain, respectively; ρ , Q , and R denote the corresponding weight coefficients, respectively; ε denotes the relaxation factor. The first item reflects the accuracy of the tracking capability, that is, the difference between the actual tracking trajectory and the expected tracking trajectory. The second item reflects the system's constraints on the control increment. In particular, in order to ensure that no solution occurs under the constraints, the third item relaxation factor is added.

In MPC, the constraint conditions are required to meet the limits of the control parameters of the tractor under normal driving conditions and obey safety constraints. Write the control quantity constraints in matrix form, as follows:

$$\begin{bmatrix} -A \\ A \end{bmatrix} \Delta U(k) \geq \begin{bmatrix} u(k-l) - u_{\max}(k) \\ \vdots \\ u(k-l) - u_{\max}(k+N_c-l) \\ u_{\min}(k) - u(k-l) \\ \vdots \\ u_{\min}(k+N_c-l) - u(k-l) \end{bmatrix} \quad (26)$$

$$A = \begin{bmatrix} I_{N_c \times N_c} & 0 & \cdots & 0 \\ I_{N_c \times N_c} & I_{N_c \times N_c} & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ I_{N_c \times N_c} & I_{N_c \times N_c} & \cdots & I_{N_c \times N_c} \end{bmatrix}_{N_c \times N_c} \quad (27)$$

The control increment constraint expression is:

$$\begin{bmatrix} -B \\ B \end{bmatrix} \Delta U(k) \geq \begin{bmatrix} \Delta u_{\max} \\ \vdots \\ -\Delta u_{\max}(k+N_c-l) \\ \Delta u_{\min}(k) \\ \vdots \\ \Delta u_{\min}(k+N_c-l) \end{bmatrix} \quad (28)$$

$$B = \begin{bmatrix} I_{N_c \times N_c} & 0 & \cdots & 0 \\ 0 & I_{N_c \times N_c} & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & I_{N_c \times N_c} \end{bmatrix}_{N_c \times N_c} \quad (29)$$

The control process of trajectory tracking is equivalent to the process of solving the objective function under constraint conditions. Therefore, the problem of the tracking trajectory in MPC can be converted into a quadratic programming problem [40,41]. Combining the above objective function and constraint conditions can obtain the optimization objective of MPC, and transform the tractor trajectory tracking control problem into a quadratic free solution problem as shown in 30.

$$\min I = [\Delta U_t^T, \varepsilon]^T H_t [\Delta U_t^T, \varepsilon] + G_t [\Delta U_t^T, \varepsilon]$$

$$s.t. \begin{cases} \Delta U_{\min} \leq \Delta U_t \leq \Delta U_{\max} \\ U_{\min} \leq A \Delta U_t + U_t \leq U_{\max} \end{cases} \quad (30)$$

$$H_t = \begin{bmatrix} \theta^T Q \theta + R & 0 \\ 0 & \rho \end{bmatrix}, \quad G_t = [2e^T Q \theta \quad 0] \quad (31)$$

where e is the deviation between the predicted and actual values.

4.2. Lower layer energy management

The multi-step global optimal algorithm is dynamic programming. The decision-making process is separated into stages. Accordingly, the issue can be turned into a family of similar subproblems, and then solved one at a time. When solving, start from the boundary conditions, proceed in reverse order, and recursively seek optimization piece by piece. When solving each subproblem, the previously obtained optimal result must be utilized, and the optimal solution to the whole problem is the solution to its last subproblem.

The torques and speed of the diesel engine and motor are taken as decision variables. The SOC value is a state variable, and the minimum equivalent total cost is the objective function. Then, a three-stage dynamic programming energy management strategy is built.

Construct the optimal objective as:

$$J = \min \int_0^{t_f} (Q_m(t) + Q_e(t)) dt \quad (32)$$

$$\begin{cases} Q_m(t) = \frac{j_m P_b(t)}{3600 \times \eta_{bat}} \\ Q_e(t) = \frac{j_e P_e(t)}{1000 \times 3600 \times 0.84} \end{cases} \quad (33)$$

where $Q_{m(t)}$ and $Q_{e(t)}$ denote the equivalent costs of the motor and diesel engine at time t , respectively; j_e and j_m indicate the prices of oil and electricity, respectively; t_f is the final moment.

The state of the system can be expressed as:

$$x(t+1) = f(x(t), u(t)) \quad (34)$$

The constraints are as follows:

$$\begin{cases} T_{m \min}(n_m(t), SOC(t)) \leq T_m(t) \leq T_{m \max}(n_m(t), SOC(t)) \\ T_{e \min}(n_e(t)) \leq T_m(t) \leq T_{e \max}(n_e(t)) \\ n_{m \min} \leq n_m(t) \leq n_{m \max} \\ n_{e \min} \leq n_e(t) \leq n_{e \max} \\ SOC_{\min} \leq SOC(t) \leq SOC_{\max} \end{cases} \quad (35)$$

where $T_{m \min}$ and $T_{m \max}$ are the minimum torque and maximum torque of the motor at the current moment, respectively; $T_{e \min}$ and $T_{e \max}$ denote the minimum torque and maximum torque of the diesel engine at the current moment,

respectively; $n_{\min}$ and $n_{\max}$ are the minimum speed and maximum speed of the motor at the current moment, respectively; $n_{\min}$ and $n_{\max}$ represent the minimum speed and maximum speed of the diesel engine at the current moment, respectively; $SOC_{\min}$ and $SOC_{\max}$ represent the minimum and maximum values allowed by the SOC, respectively.

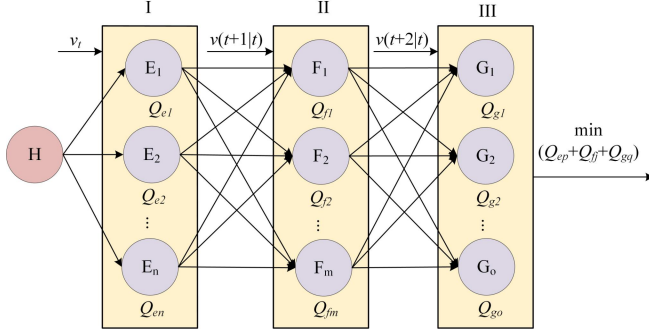


Fig. 8. Three-stage dynamic programming flowchart

The detailed procedure of the three-stage dynamic programming is shown in Fig. 8. Because one-step prediction is the most precise in model predictive control, accuracy decreases with increasing step size. Therefore, a three-stage dynamic programming approach is adopted. Input $[v_t]$, $[v_{t+1}]$, and $[v_{t+2}]$ into the dynamic programming model as the tractor driving information for the first, second, and third stages, respectively. The tractor driving information for these three stages is obtained from the model predictive control system. Q_{en} , Q_{fm} , and Q_{go} represent the indicator functions of states E_n , F_m , and G_o , respectively. When $(Q_{ei} + Q_{fj} + Q_{gq})$ ($i = 1, 2, \dots, n$; $j = 1, 2, \dots, m$; $q = 1, 2, \dots, o$) is minimized, the optimal decision vector of stage I is obtained $[T_{mlreq}, T_{elreq}, n_{mlreq}, \text{ and } n_{elreq}]$ T.

4.3. Dual-system hierarchical collaborative control method

The dual-system HCCM is an integration of MPC and three-stage dynamic programming. It integrates MPC and three-stage dynamic programming strategies through the interactive variable of velocity. Each step of operation is to globally optimize the three velocities transferred from the upper-layer trajectory tracking system to the lower-layer energy management system as a whole. However, the output control variable only adopts the first control sequence, that is, only the optimal decision variable of the first stage I is used.

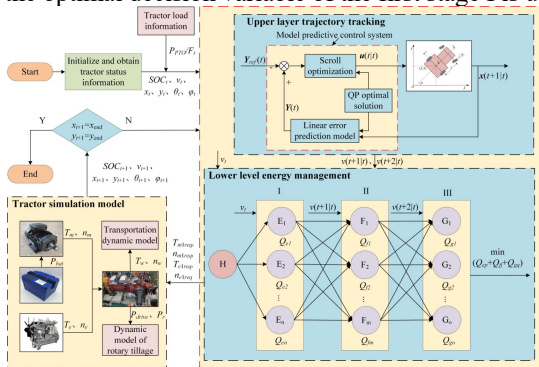


Fig. 9. HCCM flow block diagram

As can be seen from Fig. 9, this method includes three parts: the upper trajectory tracking system, the lower energy management system and the tractor simulation model. First, the tractor model is initialized, and the power battery SOC_t , tractor velocity v_t and tractor pose parameters x_t, y_t, θ_t , and φ_t are given. Collect working load information PPTO/Ft based on the tractor's working conditions. The upper-layer trajectory tracking system performs two-step prediction based on the current tractor velocity v_t and MPC to obtain the velocity v_{t+1} and v_{t+2} in the future. Subsequently, the two-step predicted velocity is passed to the underlying energy management system. Then, combining the two-step predicted velocity and the current velocity v_t , based on the three-stage dynamic programming strategy, torques and speeds of the motor and diesel engine are used as constraints, the power battery SOC is the state variable, and the lowest energy consumption of the tractor is optimized. Diesel engine and motor operating data are output to the tractor simulation model, where only the data $[T_{mlreq}, T_{elreq}, n_{mlreq}, \text{ and } n_{elreq}]$ T from the first stage I is used for output. Next the simulation result is output through the tractor simulation model. Finally, it is determined whether the tractor has reached its final position. If the final position is reached, the simulation ends. Otherwise, the output data is passed to the dual system control module for the next optimization. Until the tractor reaches the final position, the simulation ends.

5. RESULTS AND DISCUSSION

MATLAB simulation tests were carried out for the tractor's rotary tillage and transportation conditions. In the rotary tillage working condition, the headland turning path is used, and the working velocity was set to 5 km/h. In the transportation condition, double-shifting paths are used, and the working velocity was set to 28 km/h. Control research of the two working conditions is conducted based on the HCCM. In order to verify the effectiveness of the proposed strategy, the PP-TTM and the PF-EMS were used as comparison strategies. In particular, trajectory tracking and energy management in the comparison strategies are two independent control strategies.

The parameter settings of the controller part of the HCCM are shown in Table 2.

TABLE 2. CONTROLLER PARAMETER SETTINGS DURING SIMULATION

Parameters	Unit	Value
Control time domain	-	10
Forecast time domain	-	20
Simulation step size	s	0.2
Acceleration value	m/s ²	[-0.2, 0.2]
Front wheel angle	rad	[-0.4, 0.4]
change rate range		

5.1. Rotary tillage conditions(120 s)

(1) Trajectory tracking performance

The field rotary tillage operation is performed at a constant velocity of 5 km/h. Fig. 10 and 11 show the effect

curve and lateral deviation curve of the path tracking under the two control strategies.

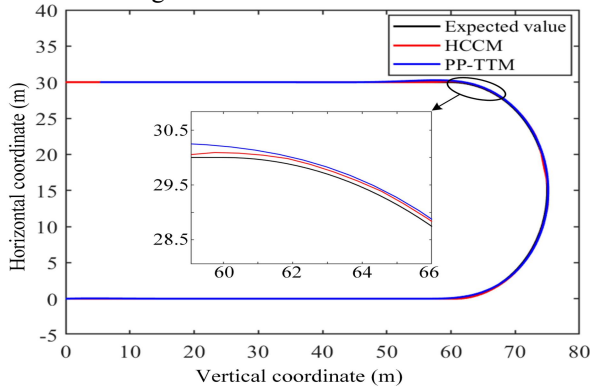


Fig. 10. Rotary tillage trajectory tracking curve

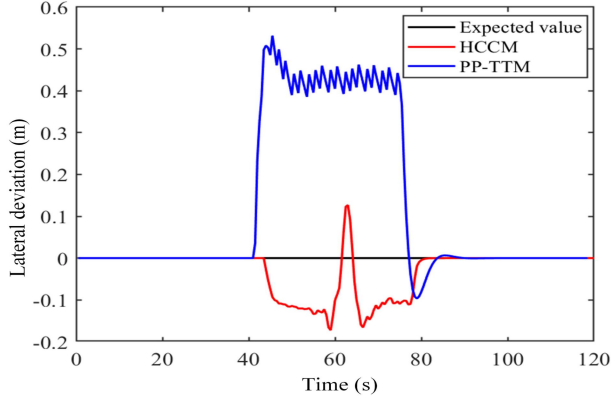


Fig. 11. Lateral deviation curve of rotary tillage

Figure 10 shows that trajectory tracking can be achieved in the rotary tillage working scenario using both the HCCM and the PP-TTM. However, tracking performance is poor during U-turning. Expressly, at turns, the trajectory tracking effect of the HCCM is better than that of the PP-TTM. It can be seen from Fig. 11 that when the PP-TTM is used, the yaw angle deviation is significantly large. The peaks of maximum deviation achieve approximately 0.53 m. The absolute mean error is 0.1277 m. When using the HCCM, the yaw angle deviation is decrescent. The peaks of maximum deviation achieve approximately 0.12 m, and the absolute mean error is 0.0319 m. The trajectory tracking accuracy has been improved by 75.02%.

Figures 12 and 13 show the effect curve and lateral deviation curve of the yaw angle tracking under the two control strategies.

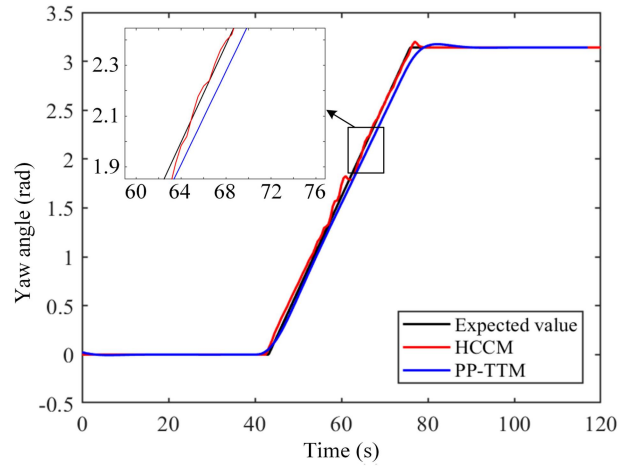


Fig. 12. Rotary tillage yaw angle tracking curve

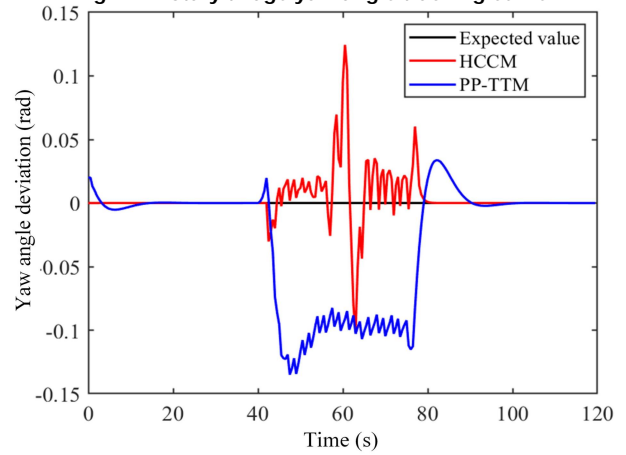


Fig. 13. Rotary tillage yaw angle tracking deviation curve

Figure 12 shows that trajectory tracking yaw angle can be achieved in the rotary tillage working scenario using both the HCCM and the PP-TTM. However, the control effect of PP-TTM is worse than that of the HCCM. It can be seen from Fig. 13 that under the two control methods, the maximum value of yaw angle deviation is basically equal. Under the HCCM and the PP-TTM control methods, their peaks of maximum deviation are approximately 0.12 rad and 0.14 rad, respectively. The absolute mean errors are 0.0076 rad and 0.0316 rad, respectively. In comparison, it is found that under the rotary tillage condition, the trajectory tracking effect of HCCM is superior to that of PP-TTM.

(2). Energy saving control performance

According to the field rotary tillage operation test, the data collected is entered into the tractor simulation model. PTO torque and speed are shown in Fig. 14.

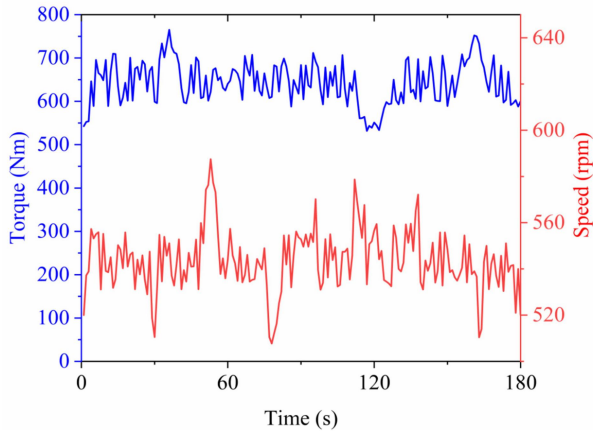


Fig. 14. PTO torque and speed

Energy-saving control research is conducted based on three-stage dynamic programming, according to the velocity delivered by the MPC system. Comparison is made with the PF-EMS. Fig. 15 and 16 show the MAP diagrams of the diesel engine and the motor under the two control strategies.

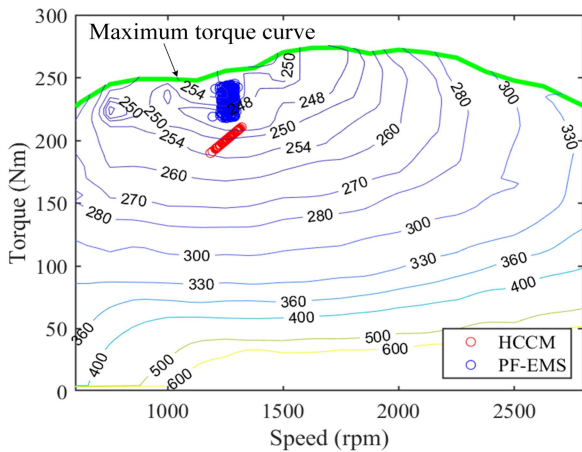


Fig. 15. MAP diagram of diesel engine rotary tillage operation

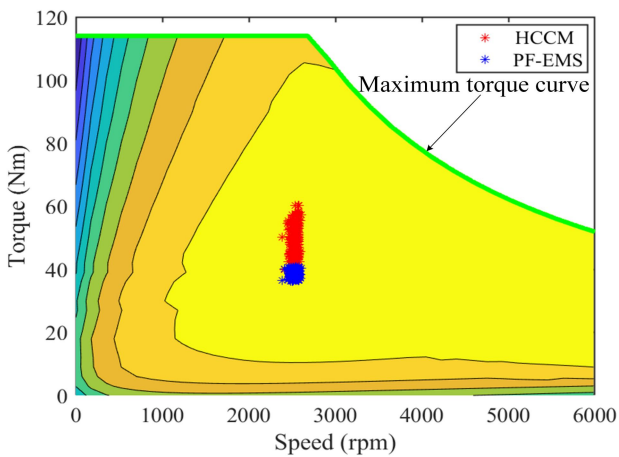


Fig. 16. MAP diagram of motor rotary tillage operation

It can be seen from Fig. 15 and 16 that regardless of which control strategies employed, the working torque of the motor is always positive, and the power battery does not appear to be charged. Under the HCCM, the diesel engine's working point is linear and highly efficient. The motor's working points are dispersed, and it operates with high torque. Under the PF-EMS, the torque of the diesel engine is relatively high, the working points are distributed in sheets, and the overall working efficiency is slightly worse. The motor's working points are concentrated, and its working torque is relatively small.

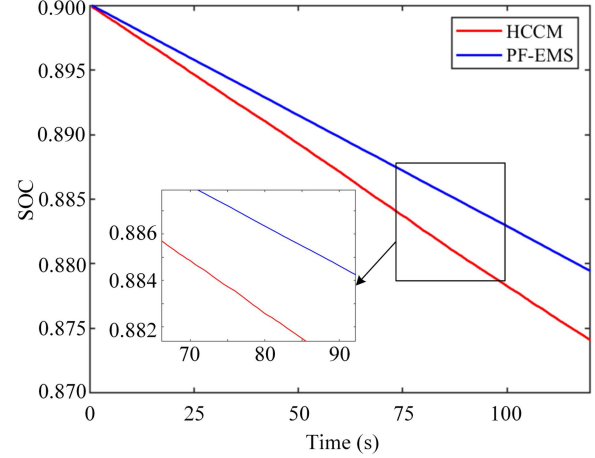


Fig. 17. SOC variation curve of rotary tillage operation

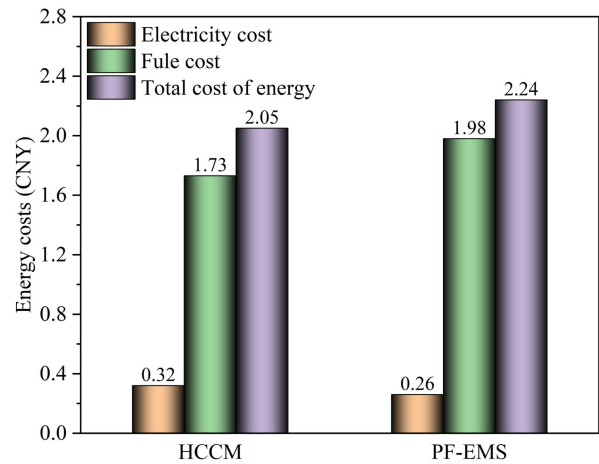


Fig. 18. Energy cost diagram of rotary tillage operation

Figure 17 shows that the trend of the SOC under the two control strategies is consistent, with a declining trend. However, under the HCCM, SOC value fluctuates and decreases rapidly. The final SOC value is 0.874. Under the PF-EMS, SOC shows a linear decline, with a slower rate of decline. The final SOC value is 0.879. Fig. 18 displays the total energy consumption cost of the tractor. Under the control methods of HCCM and PF-EMS, the costs of total energy consumption were ¥2.05 and ¥2.24, respectively. Under the proposed control method, the cost of total energy consumption was reduced by 8.48%.

5.2. Transport conditions (40 s)

5.2.1. Trajectory tracking performance

The road transportation operation is performed at a constant speed of 28 km/h. Fig. 19 and 20 show the effect curve and lateral deviation curve of the path tracking under the two control strategies.

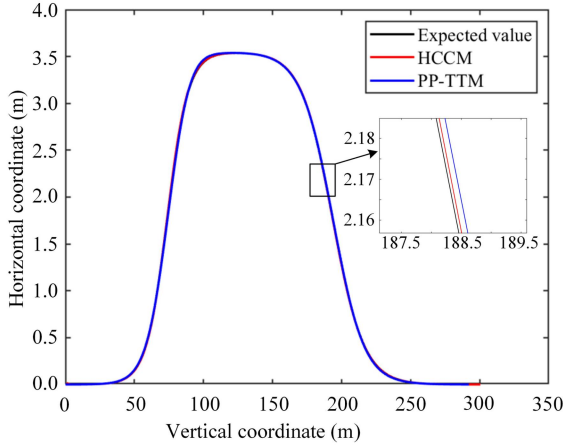


Fig. 19. Transport trajectory tracking curve

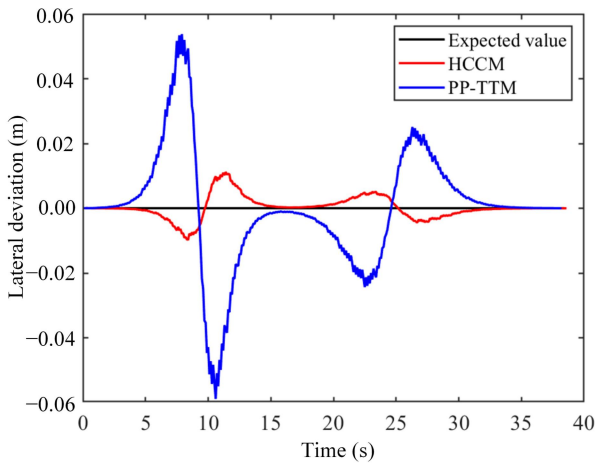


Fig. 20. Lateral deviation curve of transport

Figure 19 shows that both the HCCM and the PP-TTM can achieve trajectory tracking in the road transportation operation scenario, but the tracking performance is worse when turning to overtake. Specifically, during the overtaking phase, the path tracking effect of HCCM is better than that of PP-TTM. It can be seen from Fig. 20 that when the PP-TTM is used, the lateral deviation is significant. The peaks of maximum deviation achieve approximately 0.052 m. The absolute mean error is 0.0112 m. Under the HCCM, the lateral deviation is decrescent. The peaks of maximum deviation achieve approximately 0.012 m, and the absolute mean error is 0.0022 m. The trajectory tracking accuracy has been improved by 80.36%.

Figures 21 and 22 show the effect curve and lateral deviation curve of the yaw angle tracking under the two control strategies.

Figure 21 shows that both the HCCM and the PP-TTM can achieve trajectory tracking in the road transportation operation scenario. However, the control effect of PP-TTM is worse than that of the HCCM in turning to overtake. It can be seen from Fig. 22 that under the two control methods, the maximum value of yaw angle deviation differs considerably. When the PP-TTM is used, the yaw angle deviation is significantly large. The peaks of maximum deviation achieve approximately 0.013 rad. The absolute mean error is 0.0022 rad. Under the HCCM, the yaw angle deviation is decrescent. The peaks of maximum deviation achieve approximately 0.003 rad, and the absolute mean error is 0.0004 rad. Through comparison and analysis, it is determined that under the transport condition, the trajectory tracking effect of HCCM is superior to that of PP-TTM.

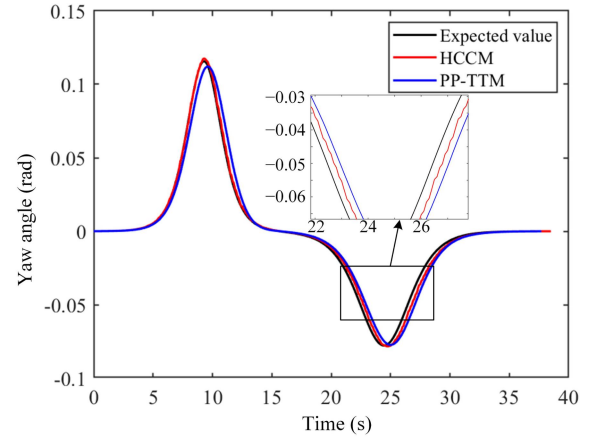


Fig. 21. Transport yaw angle tracking curve

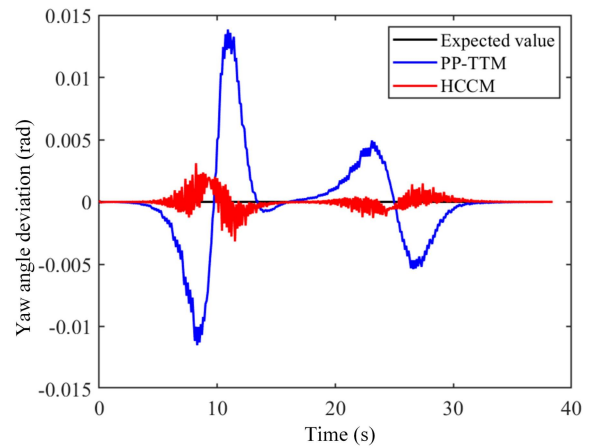


Fig. 22. Transport yaw angle tracking deviation curve

5.2.2. Energy saving control performance

Figures 23 and 24 show the MAP diagrams of the diesel engine and the motor under the two control strategies.

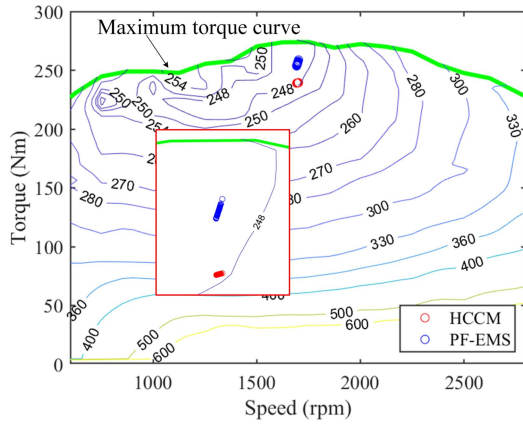


Fig. 23. MAP diagram of diesel engine transportation operation

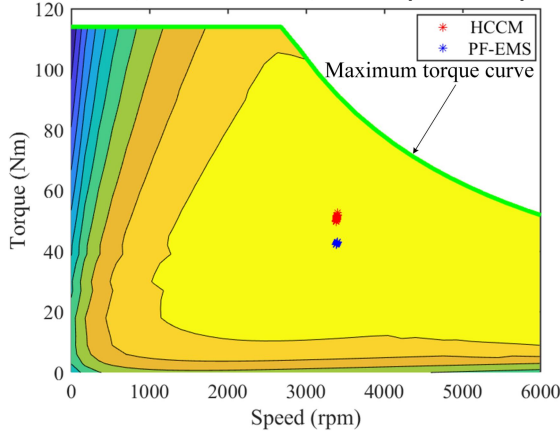


Fig. 24. MAP diagram of motor transportation operation

It can be seen from Fig. 23 and 24 that regardless of which control strategies employed, the working torque of the motor is always positive, and the power battery does not appear to be charged. Under the HCCM, the operating point of the diesel engine is concentrated in a point shape and has high efficiency. The motor's working point is concentrated, and it generates a relatively large working torque. Under the control of the PF-EMS, the diesel engine's operating point is relatively concentrated, but it operates with high torque. The motor's operating point is concentrated, and its working torque is relatively small.

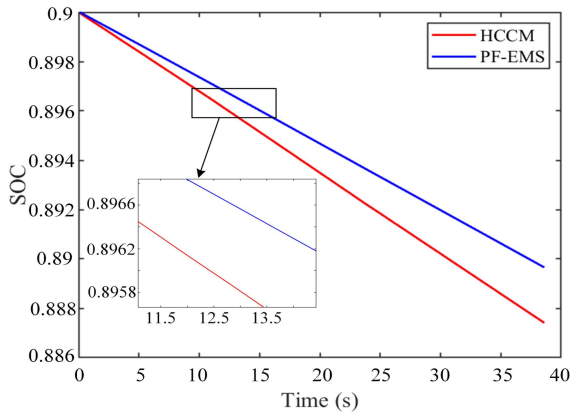


Fig. 25. SOC variation curve of transportation operation

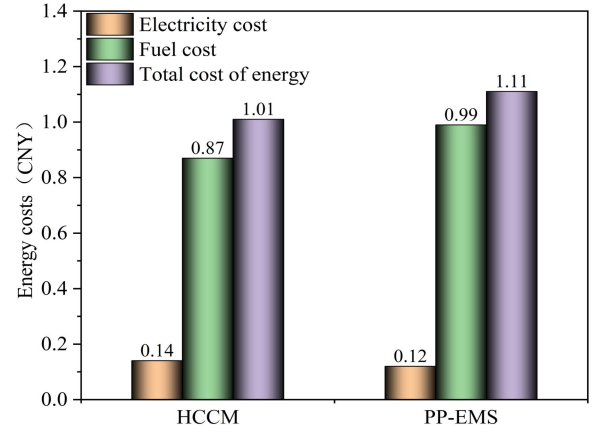


Fig. 26. Energy cost diagram of transportation operation

Figure 25 shows that the trend of the SOC under the two control strategies is consistent, with a declining trend. However, under the HCCM, SOC value decreases rapidly. The final SOC value is 0.874. Under the PF-EMS, SOC shows a linear decline, with a slower rate of decline. The final SOC value is 0.879. Fig. 26 displays the total energy consumption cost of the tractor. Under the control methods of HCCM and PF-EMS, the costs of total energy consumption were ¥1.01 and ¥1.11, respectively. Under the proposed control method, the cost of total energy consumption was reduced by 9.01%.

6. CONCLUSION

To address the performance degradation caused by independently controlling trajectory tracking and energy management while ignoring their interaction, this study proposes a dual-system HCCM for tractors, which integrates the upper layer's trajectory tracking based on MPC and the lower layer's energy management based on three-stage dynamic programming. The simulation verification shows that the tractor can obtain a better trajectory tracking effect and energy consumption economy under the conditions of rotary tillage operation and transportation operation.

Under the rotary tillage condition, the average absolute errors for the yaw angle deviation and lateral deviation of the HCCM 0.0178 rad were and 0.0834 m, respectively. The average absolute errors for the yaw angle deviation and lateral deviation of the PP-TTM were 0.0859 rad and 0.2803 m, respectively. Under the transport condition, the average absolute errors for the yaw angle deviation and lateral deviation of the HCCM were 0.0004 rad and 0.0022 m, respectively. The average absolute errors for the yaw angle deviation and lateral deviation of the PP-TTM were 0.0022 rad and 0.0112 m, respectively. The proposed control method yielded better trajectory tracking control performance for the driverless hybrid tractor. Under rotary tillage and transportation conditions, trajectory tracking accuracy increased by 75.02% and 80.36%, respectively.

Under the rotary tillage condition, the cost of total energy consumption under the HCCM and the PF-EMS were ¥2.05 and ¥2.24, respectively. The cost of total energy consumption reduced by 8.48%. Under the transport condition, the cost of total energy consumption under the HCCM and the PF-EMS

were ¥1.01 and ¥1.11, respectively. The cost of total energy consumption reduced by 9.01%. The energy economy of the driverless hybrid power tractor under the proposed control method is better.

APPENDIX

	Nomenclature	P_{th}	the power consumed by thr wing soil (kW)
A	frontal area of the tractor (m ²)	P_a	power consumed by the rotar y tiller moving forward (kW)
C	power battery rated capa city (Ah)	P_h	power consumed by overcomi ng the horizontal reaction for ce of the soil (kW)
C _D	drag coefficient of the tra ctor (–)	P_{PT} o	power consumption of PTO (kW)
E ₀	terminal voltage of the p ower battery (V)	P_{bat}	power of the power battery (k W)
e	tracking error in the predi ction time domain (–)	Q_m (t)	equivalent costs of the motor at time t (CNY)
e _t	error value of the node at time t (–)	Q_e (t)	equivalent costs of the diesel engine at time t (CNY)
$\bar{e}_{av}$ c	average of the node error s at all times (–)	R_0	internal resistance of the pow er battery (Ω)
F _t	propulsion force of the tr actor (N)	R	radius of the driving wheel (m)
F _f	rolling resistance of the tr actor (N)	SO C ₀	initial value of SOC (–)
F _w	air resistance of the tract or (N)	SO C _{mi} n	minimum value allowed by th e SOC (–)
F _i	grade resistance of the tra ctor (N)	SO C _{ma} x	maximum value allowed by t he SOC (–)
F _j	acceleration resistance of the tractor (N)	T_{req}	required torque at the input en d of the torque coupler (Nm)
f	rolling resistance coeffici ent (–)	T_{cre} q	diesel engine required torque (Nm)
I _b	output current of the pow er battery (A)	T_{mr} eq	motor required torque (Nm)
I _{Nc}	column vector with rows Nc (–)	T_w	drive wheel torque (Nm)
I _m	identity matrix of dimens ion m (–)	T_{PT} o	torque of the PTO (Nm)
i _{cv}	CVT ratio (–)	T_{cv} _in	CVT input torque (Nm)

i _{zy}	centre drive ratio (–)	T_{cv} _out	CVT output torque (Nm)
j _c	oil price (CNY L ^{–1})	T_m min	minimum torque of the motor at the current moment (Nm)
j _m	electricity price (CNY k Wh ^{–1})	T_m max	maximum torque of the motor at the current moment (Nm)
l	length of the wheelbase (m)	T_e min	minimum torque of the diesel engine at the current moment (Nm)
m	tractor use quality (kg)	T_e max	maximum torque of the diesel engine at the current moment (Nm)
m _h	total mass of trailer unit a nd cargo (kg)	T_e	diesel engine torque (Nm)
N	total number of time nod es (–)	T_m	motor torque (Nm)
N _c	control time domain (–)	t_f	terminal moment (s)
N _p	prediction time domain (–)	U_b	output voltage of the power b attery (V)
n _m	motor speed (rpm)	U_{rat} ed	rated voltage (V)
n _{cv} t_ou t	CVT output speed (rpm)	U_{mi} n	set of minimum value in the c ontrol time domain (–)
n _{cv} t_in	CVT input speed (rpm)	U_m ax	set of maximum value in the control time domain (–)
n _e	diesel engine speed (rp m)	v	operating speed of the tractor (km h ^{–1})
n _{re} q	required speed (rpm)	v_x	operating speed of the vehicle in the x-axis direction (km h –1)
n _w	speed of the drive wheel (rpm)	v_y	operating speed of the vehicle in the y-axis direction (km h –1)
n _m min	minimum speed of the m otor at the current mome nt (rpm)	α	represents the slope (°)
n _m max	maximum speed of the m otor at the current mome nt (rpm)	θ	front wheel slip angle (rad)
n _e min	minimum speed of the di esel engine at the current moment (rpm)	φ	vehicle yaw angle (rad)
n _e max	maximum speed of the di esel engine at the current	η_{zy}	centre drive efficiency (–)

	moment (rpm)		
n_{er}	rated speed of diesel engine (rpm)	η_{cvt}	CVT efficiency (-)
n_{mr}	rated speed of the motor (rpm)	η_o	torque coupler efficiency (-)
n_{PTO}	speed of the PTO (rpm)	η_{bat}	charging or discharging efficiency of the power battery (-)
P_{req}	required power of the tractor (kW)	δ	mass conversion factor of the tractor (-)
P_{dr}	driving power of the tractor (kW)	ρ	weight factor (-)
P_r	power consumption of the rotary tiller (kW)	ε	relaxation factor (-)
P_c	power consumed by cutting soil (kW)		

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