The steering control of tracked vehicles is one of the crucial aspects of dynamic control. The structure with dual-sided independent electric drive is a commonly employed configuration in tracked vehicles. The function of the pivot steering, as a unique steering mode, possesses excellent maneuverability and serves as a key advantage that distinguishes tracked vehicles from wheeled vehicles. This function relies on the differential driving force between the left and right tracks to generate yaw torque, thereby achieving a minimal steering radius. Particularly in situations where ground space is limited, the function of the pivot steering holds a significant advantage.

Thus far, scholars and experts have conducted a series of studies on the research topic of tracked steering from the following three aspects and have achieved significant research results.

Firstly, there are studies focusing on the aspect of the dynamics model of tracked vehicles. Johnson et al. [1] focused on the dynamic modeling and control of tracked vehicles in rugged off-road terrain. The authors established dynamic models considering interactions between the vehicle and terrain, including ground contact states, friction, resistance, etc. They also explored the impact of different control strategies on vehicle performance in such environments. Chen et al. [2] addressed the dynamic modeling and simulation of tracked vehicles on soft soil. Researchers considered the influence of soft soil on vehicle motion, such as soil compression, deformation, and lateral resistance, and simulated them through mathematical models. They also studied the effects of different types of soft soil...
on vehicle performance and how to optimize vehicle design and control strategies to adapt to this environment. Liu et al. [3] investigated the dynamic interaction between tracked vehicles and terrain. These researchers analyzed the mechanical characteristics when the tracks contact the ground, including ground pressure, ground contact area distribution, friction, etc. They numerically simulated and experimentally verified the motion behavior of tracked vehicles under different terrain conditions and proposed suggestions for optimizing design and control strategies to enhance vehicle performance and stability. Yuan et al. [4] analyzed the lateral motion response characteristics of high-speed tracked vehicles and validated the findings through experiments. The study investigated the dynamic behavior of tracked vehicles during lateral motion and evaluated factors influencing their stability and maneuverability. Their experimental results provided insights into the lateral dynamics of tracked vehicles, contributing to the development of effective control strategies for enhancing vehicle performance and safety. Zhang et al. [5] investigated the steering model of tracked vehicles operating on hard surface conditions. The study analyzed the dynamic behavior of tracked vehicles during steering maneuvers and validated the proposed model through experimental verification. The results provided insights into the steering characteristics of tracked vehicles on hard surfaces, contributing to the development of effective steering control strategies for such operating conditions.

Secondly, the next most common focus lies in the improvement of control algorithms to enhance the steering process of tracked vehicles. Wang et al. [6] proposed an adaptive steering control system for tracked vehicles based on fuzzy logic. The fuzzy logic controller adjusted the steering angle according to real-time feedback from sensors, enabling the vehicle to adapt to varying terrain conditions and improve maneuver ability. Brown, R., et al. [7] conducted a comparative analysis of various steering control algorithms for tracked vehicles. It evaluated the performance of different control strategies, including fuzzy logic, PID control, genetic algorithms, and SMC, in terms of steering accuracy, stability, and robustness across different terrains and operating conditions. Gupta, S. et al. [8] focused on optimizing control strategies for tracked vehicle steering using genetic algorithms. By iteratively evolving and selecting optimal control parameters, the genetic algorithm enhanced the vehicle’s steering performance and robustness across different terrains. Li et al. [9] investigated the steering control of dual-motor-coupled tracked vehicles and optimized the PID parameters using the particle swarm optimization algorithm. By iteratively adjusting the PID parameters, the proposed approach improved the vehicle’s steering accuracy and stability. Wei et al. [10] presented a fuzzy adaptive sliding mode steering control system for dual-side electric-driven tracked vehicles. By combining fuzzy logic with sliding mode control, the proposed system achieved robust and adaptive steering performance under varying operating conditions. Ma et al. [11] proposed a fuzzy PID control system for tracked vehicle steering based on hydraulic mechanical differential mechanisms. By integrating fuzzy logic with PID control, the system effectively regulated steering behavior, enhancing vehicle stability and maneuverability. Chen et al. [12] investigated the steering control strategy of dual-side electric-driven tracked vehicles using a fuzzy PID algorithm. The hybrid control algorithm combined the advantages of fuzzy logic and PID control to achieve precise and adaptive steering performance in various operating conditions. Zhang et al. [13] proposed a fuzzy feedforward–feedback steering control system for dual-side electric-driven tracked vehicles. By integrating feedforward and feedback control strategies with fuzzy logic, the system enhanced steering accuracy and stability, especially in challenging terrain. Zeng et al. [14] presented an equivalent conditions integral sliding mode stable steering control system for dual-side electric-driven tracked vehicles. By employing integral sliding mode control with equivalent conditions, the proposed system achieved stable and precise steering performance under varying operating conditions. Zhang et al. [15] investigated the steering control of dual-side independent electric-driven tracked vehicles through a hierarchical control strategy. A hierarchical control approach was employed to decompose the steering control task into multiple levels, and corresponding control strategies were designed for each level. Their experimental results demonstrated that the
hierarchical control method can effectively enhance the steering performance of tracked vehicles, enabling stable and precise steering motion under various operating conditions. Wang et al. [16] investigated path tracking and steering control algorithms for tracked vehicles based on the Instantaneous Center of Rotation (ICR). The study proposed a novel approach to path tracking and steering control by utilizing the concept of ICR, aiming to achieve accurate and smooth vehicle trajectory. The experimental results demonstrated the effectiveness of the proposed algorithm in achieving precise path tracking and steering control for tracked vehicles in agricultural applications. Wang et al. [17] presented a steering control strategy for tracked vehicles utilizing active disturbance rejection control (ADRC). The study discussed the application of ADRC in improving the steering performance of tracked vehicles by effectively compensating for external disturbances and uncertainties. Experimental results from the China Society of Automotive Engineers Annual Conference demonstrated the effectiveness of the proposed control strategy in enhancing vehicle stability and maneuverability.

Finally, it is suggested to enhance the steering performance of tracked vehicles by adding mechanical mechanisms or installing sensor devices. Lee et al. [18] focused on optimizing the design of the steering mechanism for tracked vehicles. Various parameters affecting the steering performance were analyzed, and an optimization framework was proposed to enhance the overall design efficiency. The results demonstrated significant improvements in the steering mechanism’s performance, leading to better maneuverability and control of tracked vehicles in different operating conditions. The development of a track tension control system for tracked vehicles was presented in this paper. The system aimed to maintain optimal track tension levels, ensuring better traction, stability, and durability of the vehicle’s tracks. The design, implementation, and experimental validation of the control system were discussed, highlighting its effectiveness in improving the overall performance of tracked vehicles [19]. Wang et al. [20] introduced the development of a smart steering system tailored for tracked agricultural vehicles. The system integrated advanced sensing, control, and automation technologies to enhance the vehicle’s steering precision, efficiency, and adaptability in agricultural operations. The experimental results demonstrated the system’s capability to optimize steering performance while minimizing energy consumption and environmental impact.

Although various approaches and methods for studying and improving the steering performance of tracked vehicles from different perspectives have been proposed by many experts and scholars in the aforementioned research, there has been relatively little research specifically focusing on the optimization of control algorithms for tracked vehicles under the condition of the pivot steering.

The pivot steering condition of tracked vehicles is targeted in this study. To enhance the pivot steering capability of tracked vehicles, the track control system is decoupled into two subsystems. The GWO-PID control algorithm is then employed to optimize the parameters of the vehicle speed and yaw rate sub-controllers separately. Subsequently, the effectiveness and feasibility of the proposed control algorithms are validated through joint simulation using Matlab/Simulink + RecurDyn (V9R4).

2. Analysis and Modeling of Tracked Vehicle Architecture and Pivot Steering Conditions

2.1. Tracked Vehicle Architecture and Principle of Remote Control

Driving motors are installed on both the left and right sides in the tracked vehicle featuring a dual-sided independent electric-driven structure [21–23]. Through the rotation of the motors, power is transmitted to the driving wheels via reduction mechanisms, thereby propelling the rotation of the driving wheels and subsequently driving the movement of the tracks. The principal architecture of the vehicle is composed of batteries, a central vehicle controller, and motor controllers. The battery is tasked with providing the energy required for the vehicle’s operation. The central vehicle controller is required to receive and process input signals from the driver and, based on real-time feedback about the vehicle’s
status, issues control commands to the motor controllers in accordance with the control strategy. These commands control the motors on both sides to execute movements such as forward and backward motion and steering of the tracked vehicle. The overall vehicle architecture is shown in Figure 1a.

![Vehicle architecture](image1)

Figure 1. Vehicle architecture.

The principle of remote control is shown in Figure 1b.

Due to the need for remote control of the tracked vehicle, a remote control system has been specifically equipped for it. The HM30 module is a wireless data transmission and image transmission module specialized for aerial vehicles, with its airborne end connected to the vehicle controller and its ground end connected to the remote control device.

Control commands are transmitted by the operators through the remote control device to the vehicle controller via the HM30 module, and the input signals are wirelessly communicated in the form of messages. These messages are then sent to the vehicle controller. Upon reception, the vehicle controller further controls the left and right driving motors based on the control strategy, thereby controlling the chassis movement. Similarly, the vehicle controller receives feedback signals via the HM30 module, which are packaged in the form of messages and fed back to the control device. Operators can adjust chassis control based on the feedback information.
2.2. Analysis and Modeling of Pivot Steering Conditions

During the kinematic and dynamic analysis of tracked vehicles, minor factors influencing the steering process are disregarded [24–26]. Before establishing a simplified model under theoretical conditions, three assumptions are initially proposed: (1) the ground adhesion coefficient and steering coefficient remain constant, corresponding to constant ground friction resistance and steering resistance; (2) the vertical load distribution on the inner and outer sides is uniform, meaning the steering resistance of the two tracks should be consistent, and the steering torque is linearly related to the steering resistance coefficient; (3) the left and right tracks are treated as rigid bodies, experiencing no deformation or stretching during vehicle steering. The schematic diagram is depicted in Figure 2.

![Diagram of tracked vehicle steering](image)

**Figure 2.** Kinematic and dynamic analysis of tracked vehicle steering, where $F_{d1}$ and $F_{d2}$ are the traction forces on the left and right sides of the tracked vehicle, respectively, $F_{d1}$ is negative during pivot steering, indicating traction in the opposite direction, $F_{f1}$ and $F_{f2}$ are the rolling resistances on the left and right sides of the tracked vehicle, $M_f$ is the steering resistance torque, $v_c$ is the velocity of the tracked vehicle’s centroid, $w$ is the steering angular velocity, $v_1$ and $v_2$ are the velocities on the left and right sides of the vehicle, $C$ is the vehicle’s centroid, $O$ is the steering center, $B$ is the length of the central line on both sides of the tracks, and $L$ is the length of the track in contact with the ground.

It is evident that the steering angular velocity and steering radius of tracked vehicles are as follows.

$$\begin{align*}
w &= \frac{v_1}{R - \frac{B}{2}} = \frac{v_2}{R - \frac{B}{2}} = \frac{v_2 - v_1}{B} = \frac{v_c}{K} \\
R &= \frac{B}{2} \cdot \frac{v_2 + v_1}{v_2 - v_1}
\end{align*}$$

(1)

In the pivot steering condition, the motors on the left and right sides of the tracked vehicle rotate in opposite directions, generating ground reaction forces. Thus, under the combined action of $F_{d1}$ and $F_{d2}$, a yaw moment is generated around the centroid $C$, overcoming ground resistance and resulting in a yaw rate.
Based on the steering process of tracked vehicles and the principles of vehicle dynamics, it can be deduced that by establishing dynamic equilibrium equations, the following can be obtained [27–30]:

\[
\begin{align*}
F_{d1} + F_{d2} - F_{f1} - F_{f2} &= 0 \\
(F_{d2} - F_{d1} + F_{f1} - F_{f2}) \frac{B}{2} &= M_{\mu} \\
F_{f1} &= F_{f2} = \frac{f \mu g}{2} \\
F_{d1,2} &= \frac{T_{1,2} \eta}{i} \\
M_{\mu} &= \mu mgL \\
\mu &= \mu_{\text{max}} \frac{0.925 + 0.15 R}{B}
\end{align*}
\]

where $f$ is the ground resistance coefficient, assumed to be 0.1, $T_1$ and $T_2$ are the output torques of the motors on the left and right sides, respectively, $i$ is the transmission ratio of the reducer, assumed to be 30, $\eta$ is the transmission efficiency, assumed to be 95%, $\mu$ is the steering resistance coefficient, $m$ is the mass of the tracked vehicle, $g$ is gravitational acceleration, assumed to be 9.8 m/s$^2$, and $\mu_{\text{max}}$ is the maximum value of the steering resistance coefficient, assumed to be 0.9 (clay load).

3. Control System Scheme

As described previously, the central vehicle controller of the tracked vehicle is designed to receive and decode driver input signals (i.e., signals from the electronic accelerator pedal, electronic brake pedal, and electronic steering wheel). Based on the control strategy, it processes these inputs to determine the target torque for the motors on both sides and sends it via the CAN bus to control the steering of the tracked vehicle. Unlike motor speed adjustment methods, this torque adjustment control method requires the driver to adjust the input signals in real time based on feedback received about the chassis’ current state, ensuring that the vehicle operates in accordance with the driver’s intentions [31].

In conclusion, the key challenge in the dynamic control of tracked vehicle steering lies in how to establish the relationship between the intermediate control variables $u_1$ and $u_2$ and the input signals, as well as the feedback information about the vehicle’s driving state.

3.1. Decoding of Driver Input Signals

In this paper, the driver input signals from the electronic accelerator pedal, electronic brake pedal, and electronic steering wheel are chosen to be decoded into the corresponding target vehicle speed $v^*_{c}$ and target yaw rate $\omega^*_{c}$ for pivot steering of the tracked vehicle. Subsequently, these are further calculated to obtain the target output torques $T^*_1$ and $T^*_2$ for the motors on the left and right sides of the tracked vehicle during pivot steering [32]. (Note: Since this paper only analyzes the steady-state steering process during pivot steering, signals from the brake pedal are ignored.)

Based on the different openings of the electronic accelerator pedal, the target vehicle speed for the tracked vehicle is correspondingly decoded, as illustrated in the pedal schematic shown in Figure 3a.
In this paper, the driver input signals from the electronic accelerator pedal, electronic steering wheel, and the tracked vehicle’s inherent inertial impacts and partly to the influence of ground resistance.

Correspondingly, it is decoded as

$$v_c^* = f_1(\alpha) = \begin{cases} 0, & \alpha \in [0, \alpha_0] \\ \frac{\alpha - \alpha_0}{\alpha_{\text{max}} - \alpha_0} v_{\text{max}}, & \alpha \in (\alpha_0, \alpha_{\text{max}}] \end{cases}$$

where $v_c^*$ is the target vehicle speed, $v_{\text{max}}$ is the maximum set speed, $\alpha_0$ is the free travel of the pedal, set as 5°, $\alpha$ is the real-time travel of the pedal, and $\alpha_{\text{max}}$ is the allowed maximum travel of the pedal, set as 45°.

$$\omega_c^* = f_2(\varphi) = \begin{cases} 0, & \varphi \in [-\varphi_0, \varphi_0] \\ \frac{\varphi - \varphi_0}{\varphi_{\text{max}} - \varphi_0}, & \varphi \in [-\varphi_{\text{max}}, \varphi_0) \cup (\varphi_0, \varphi_{\text{max}}] \end{cases}$$

where $\omega_c^*$ is the target yaw rate, $\varphi_0$ is the free steering wheel angle, set as 5°, $\varphi$ is the real-time steering wheel angle, and $\varphi_{\text{max}}$ is the allowed maximum steering wheel angle, set as 45°.

Meanwhile, it can be concluded that there is a relationship between the steering radius of tracked vehicles and the steering angle of the electronic steering wheel.

$$R = \frac{B}{2} \frac{\varphi_{\text{max}} - \varphi_0}{\varphi - \varphi_0}$$

3.2. Control Variable Decoupling

In the dual-side independent electric-driven tracked vehicle, which is a coupled system, it is observed from Formula (2) that any variation in the output torque of either side’s motor will affect the vehicle’s speed and direction. The coupling effect is partly attributed to the vehicle’s inherent inertial impacts and partly to the influence of ground resistance.

The input variables of the tracked vehicle steering control system consist of signals from the electronic accelerator pedal and electronic steering wheel. These input signals are decoded into the corresponding target vehicle speed $v_c^*$ and target yaw rate $\omega_c^*$ for pivot steering. Subsequently, calculations are performed to determine the target output torques $T_1^*$ and $T_2^*$ for the motors on the left and right sides of the tracked vehicle during pivot steering. The output variables are the actual speed $\dot{v}$, and actual yaw rate $\dot{\omega}$ of the tracked vehicle’s steering. To achieve effective control, it is necessary to decouple this system into two independent single-input single-output systems.

From Formula (1), it can be inferred that

$$\begin{cases} \dot{v} = \frac{m}{I} (T_1 + T_2) - f g - \frac{F_d}{m} \\ \dot{\omega} = \frac{IB}{2T} (T_2 - T_1) - \frac{M_u}{T} - \frac{M_d}{T} \end{cases}$$
where $J$ is the tracked vehicle’s moment of inertia.

Applying a Laplace transform to Formula (6), the following is obtained:

$$\begin{bmatrix} v_c(s) \\ \omega_c(s) \end{bmatrix} = Q(s) \begin{bmatrix} T_1(s) \\ T_2(s) \end{bmatrix} - P(s, v_c, \omega_c) \tag{7}$$

where $Q(s) = \begin{pmatrix} \frac{mL}{10} & \frac{mL}{10} \\ \frac{mL}{10} & \frac{mL}{10} \end{pmatrix}$, $P(s, v_c, \omega_c)$ is the resistance matrix.

Evidently, $Q(s)$ is a non-diagonal matrix, indicating the coupling relationship between $T_1$ and $T_2$. The transformation matrix $K$ is introduced to convert $Q(s)$ into a diagonal matrix for the control variables $u_1$ and $u_2$ designed in this paper.

$$\begin{bmatrix} u_1(s) \\ u_2(s) \end{bmatrix} = K \begin{bmatrix} T_1(s) \\ T_2(s) \end{bmatrix}, \quad K = \begin{pmatrix} 1 & -1 \\ 1 & 1 \end{pmatrix} \tag{8}$$

After decoupling, the control system of the tracked vehicle can be depicted as shown in Figure 4.

![Figure 4. The decoupled tracked-vehicle control system.](image)

Combining the decoupled tracked-vehicle control system, the achievement of the vehicle’s effective pivot steering by controlling the output torque of the motors on both sides should possess the following characteristics:

1. Given a constant steering angle signal from the electronic steering wheel, the tracked vehicle performs pivot steering from a stationary state to a certain yaw rate.
2. The target yaw rate value has good response speed and maintains stability.

4. Controller Design

4.1. GWO

The Grey Wolf Optimizer (GWO) [33,34] is an optimization algorithm inspired by the behavior of grey wolves. This algorithm effectively simulates the hunting mechanism, division of labor, and social hierarchy observed in grey wolves. The GWO is characterized by its simplicity, robustness, and effectiveness in optimization. It consists of the following levels:

1. The wolf $\alpha$, as the leader, is responsible for making primary decisions and is also the one closest to the optimal value.
2. The wolves $\beta$ and $\delta$, as helpers of the leading wolf, aid in managing the team and serve as a bridge for communication between the upper and lower ranks of the wolf pack.
3. The wolves $\omega$, as the lowest-ranking members, are tasked with assisting the higher-ranking wolf packs.

The hierarchy within the grey wolves is clear, with mutual cooperation, clear division of labor, and a collective effort to maintain the stability and unity of the wolf community.
The hunting behavior of grey wolves primarily consists of encirclement, pursuit, and attacking prey. During the hunting process, the leading wolf $\alpha$ guides the pack in locating, tracking, and approaching the prey. When the prey is sufficiently close, $\beta, \delta$ and $\omega$ receive the command to initiate the attack. During the movement, the prey is surrounded until it is captured.

Modeling this hunting process involves the following:

\[
\begin{align*}
\vec{D} &= \vec{C} \cdot \vec{X}_p(t) - \vec{X}(t) \quad (9) \\
\vec{X}(t + 1) &= \vec{X}_p(t) - \vec{A}\vec{D} \quad (10)
\end{align*}
\]

where $\vec{D}$ is the distance from the grey wolves to the prey, $t$ is the current iteration number, $\vec{X}_p$ is the position vector of the prey, and $\vec{X}$ is the position vector of the grey wolves. $\vec{A}$ and $\vec{C}$ are the coefficient matrix. The calculation formula is as follows:

\[
\begin{align*}
\vec{A} &= 2\vec{a}r_1 - \vec{a} \quad (11) \\
\vec{C} &= 2r_1 \quad (12)
\end{align*}
\]

where $\vec{a}$ is the convergence factor, which linearly decreases from 2 to 0 as the number of iterations changes, and $r_1$ and $r_2$ are random vectors in the range $[0, 1]$.

In the hunting behavior, the following rules are stipulated for updating the positions of the wolf pack and the prey:

\[
\begin{align*}
\vec{D}_\alpha &= \vec{C}_1\vec{X}_\alpha - \vec{X} \\
\vec{D}_\beta &= \vec{C}_2\vec{X}_\beta - \vec{X} \\
\vec{D}_\delta &= \vec{C}_3\vec{X}_\delta - \vec{X} \quad (13)
\end{align*}
\]

\[
\begin{align*}
\vec{X}_1 &= \vec{X}_\alpha - \vec{A}_1(\vec{D}_\alpha) \\
\vec{X}_2 &= \vec{X}_\beta - \vec{A}_2(\vec{D}_\beta) \\
\vec{X}_3 &= \vec{X}_\delta - \vec{A}_3(\vec{D}_\delta) \quad (14)
\end{align*}
\]

\[
\vec{X}(t + 1) = \left( \frac{\vec{X}_1 + \vec{X}_2 + \vec{X}_3}{3} \right) \quad (15)
\]

where $\vec{D}_\alpha, \vec{D}_\beta, \vec{D}_\delta$ are the final randomized position vectors, and $\vec{X}_1, \vec{X}_2,$ and $\vec{X}_3$ are the movement commands issued by $\alpha, \beta, \text{and } \delta$. The coefficient matrix $\vec{A}$ is calculated according to Formula (11), and the coefficient matrix $\vec{C}$ is calculated according to Formula (12).

When $|\vec{A}| < 1$, the grey wolves gradually approach the prey, and the exploration mechanism of GWO will redefine the attack path, initiating a local search. When $|\vec{A}| > 1$, the grey wolves move away from the prey, initiating a global search. This mechanism promotes the exploration process of the GWO and prevents the algorithm from stagnating at local minima. Additionally, the random selection of values for the coefficient matrix $\vec{C}$ within the range of $[0, 2]$ contributes to fostering the exploration process of the GWO, facilitating the discovery of the global optimum.
4.2. The PID Controller Optimized by GWO

As the most common feedback controller, the PID controller adjusts the input based on the difference between the input and output to reach the desired value. In order to achieve closed-loop control during the pivot steering process of tracked vehicles, the PID control is employed to regulate the vehicle speed and yaw rate. Thus, the relationship between the control variables $u_1$ and $u_2$ of the controller designed in this paper and the errors in vehicle speed $e(v_c)$ and lateral angular velocity $e(\omega)$ are as follows:

\[
\begin{align*}
    u_1 &= K_{p1}e(v_c) + K_{i1}\int e(v_c)\,dt + K_{d1}\frac{de(v_c)}{dt} \\
    u_2 &= K_{p2}e(\omega) + K_{i2}\int e(\omega)\,dt + K_{d2}\frac{de(\omega)}{dt}
\end{align*}
\]

(16)

where $u_1$ and $u_2$ are the control variables, $e(v_c)$ and $e(\omega)$ are the errors between the target and actual values of vehicle speed and yaw rate, $K_{p1}$ and $K_{p2}$ are the proportional control parameters, $K_{i1}$ and $K_{i2}$ are the integral control parameters, and $K_{d1}$ and $K_{d2}$ are the derivative control parameters of the controller.

Based on the magnitude of the errors in vehicle speed and yaw rate, the parameters of the PID controller are dynamically adjusted in real time to achieve regulation and control of the pivot steering for tracked vehicles. Therefore, the optimization of the three parameters, namely the proportional parameter $K_p$, integral parameter $K_i$, and derivative parameter $K_d$, is crucial for the control system’s performance. In this paper, the optimal parameters of the PID controller are optimized using the Grey Wolf Optimizer algorithm [35–37]. Its schematic diagram is shown in Figures 5 and 6.

From Figure 6, it is concluded that when the current iteration number of the GWO algorithm reaches the predefined maximum iteration number, the optimization process of the GWO is halted, and the optimal values of the minimum fitness value and corresponding PID control parameters are output.
5. Co-Simulation and Result Analysis

5.1. Model Construction

5.1.1. Driving Motor Model

In this paper, only the correspondence between mechanical input and output needs to be considered, and the dynamic response time of the driving motor model is approximated by a first-order inertia element according to Formula (17).

\[
T = \left\{ \begin{array}{ll}
\frac{1}{\tau s + 1} \min(T^*, T_d(n)), & T^* \geq 0; \\
\frac{1}{\tau s + 1} \max(T^*, T_b(n)), & T^* < 0.
\end{array} \right.
\]  

(17)

where \( T^* \) is the target torque, \( T_d(n) \) and \( T_b(n) \) are the maximum braking torque when the motor speed is \( n \), \( T \) is the motor output torque, and \( \tau \) is the dynamic response time. When the motor response is fast enough, it is approximately considered that \( T = T^* \).

5.1.2. Tracked Vehicle Dynamics Model

This paper utilizes the multibody dynamics simulation software RecurDyn (V9R4) to construct the dynamics model of the tracked vehicle, primarily creating the tracked vehicle mechanism based on the 3D model [38–40]. The creation of the tracked vehicle’s propulsion system primarily involves the design of the dimensions and shapes of the four wheels and tracks, the design of the tensioning device, and the addition of kinematic pairs. The completed tracked vehicle dynamics model is shown in Figure 7. When using this software to establish a dynamic model of the tracked vehicle, some components such as the power
system and transmission system were not fully considered during the modeling process, which has a certain impact on the accuracy of the simulation results.

Figure 7. Dynamic model of tracked vehicle.

Meanwhile, a road surface model is established in RecurDyn(V9R4), with the parameters selected for a lean clay road surface, as depicted in Figure 8.

Figure 8. The parameters of lean clay road surface.

5.1.3. Co-Simulation Model

After the tracked vehicle dynamics model was established, the steering control model of the tracked vehicle designed earlier was constructed in Matlab/Simulink, and joint simulation was carried out through the interface of the two software [41–44]. The co-simulation model is as shown in Figure 9.
In Figure 9, the co-simulation model includes the driver input signal module, driver input signal decoding module, controller module, motor module, and tracked vehicle dynamics module. According to the design of the steering control strategy, during simulation, the torques of the tracks on both sides are inputted into RecurDyn (V9R4), while Simulink receives real-time vehicle speed and yaw rate information of the tracked chassis. The simulation parameters of the tracked vehicle are as shown in Table 1.

Table 1. Simulation parameters.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>mass of tracked vehicle</td>
<td>kg</td>
<td>500</td>
</tr>
<tr>
<td>track gauge</td>
<td>mm</td>
<td>700</td>
</tr>
<tr>
<td>the length of the grounding part of the track</td>
<td>mm</td>
<td>1100</td>
</tr>
<tr>
<td>transmission ratio of reducer</td>
<td>/</td>
<td>30</td>
</tr>
</tbody>
</table>

In the PID controller, the optimization range for the parameters $K_p$, $K_i$, and $K_d$ is set to $[0.001, 100]$. The population size of the GWO is set to $N = 30$, and the maximum number of iterations is $t = 500$.

5.2. Result Analysis

5.2.1. Straight-Line Driving Condition

To verify the tracking performance and disturbance rejection capability of the controller designed in the preceding section under a straight-line driving condition, a comparative simulation experiment was conducted between the GWO-PID control and the traditional PID control.

The simulation experiment was set to a total duration of 10 s, with the target speed of the tracked vehicle set at 5 km/h. A load disturbance was introduced at 5 s, and the changes in vehicle speed under the two control algorithms were observed and recorded, resulting in Figure 10a.

As depicted in Figure 10a, at the start of the simulation, the tracked vehicle accelerates from 0 to the target speed of 5 km/h. Under traditional PID control, when the vehicle reaches the target speed of 5 km/h, the overshoot percentage is 20%, and it takes approximately to stabilize at the target value. In contrast, under GWO-PID algorithm control, the overshoot percentage when the vehicle reaches the target speed did not exceed 4%, and it took about 1 s to reach the target speed. This clearly demonstrates that the GWO-PID controller has a faster response speed in tracking the target speed.
Simultaneously, at 5 s into the simulation, a load disturbance is introduced. Under GWO-PID control, the tracked vehicle’s speed exhibits a smaller fluctuation amplitude of 17% compared to traditional PID control. Additionally, the vehicle recovers to the target speed 0.4 s earlier after the disturbance is removed. Clearly, under GWO-PID control, the tracked vehicle’s speed demonstrates superior anti-disturbance capability compared to traditional PID control.

The changes in the vehicle’s yaw rate during the simulation process were monitored and recorded, as shown in Figure 10b. Under both control methods, the tracked vehicle’s yaw rate stabilizes at 0. However, under the GWO-PID control algorithm, compared to traditional PID control, the vehicle’s yaw rate exhibits a smaller range of fluctuations.

5.2.2. Pivot Steering Condition

To validate the tracking performance and disturbance rejection capability of the controller designed in the preceding section in the case of pivot steering, another simulation comparison experiment was conducted between GWO-PID control and traditional PID control.

The simulation experiment was set to a total duration of 15 s, with the tracked vehicle remaining stationary from 0 to 5 s and simulating pivot steering from 5 to 15 s. The changes in vehicle speed under the two control algorithms were observed and recorded, resulting in Figure 11a.

As depicted in Figure 11a, under traditional PID control, the tracked vehicle’s speed begins accelerating from 0 to the target value of 4 km/h at 5 s, with an adjustment time of 3.5 s to reach the target value. In contrast, under GWO-PID control, the adjustment time to reach the target value is 1.5 s, and GWO-PID control does not significantly increase the overshoot percentage. This clearly demonstrates that the GWO-PID controller has a significant advantage in response speed for controlling the pivot steering condition.

The changes in the vehicle’s yaw rate during the simulation process were monitored and recorded, as shown in Figure 11b.

As shown in Figure 11b, under traditional PID control, the tracked vehicle achieves the target yaw rate value during the pivot steering condition with an adjustment time of 2 s. In contrast, under GWO-PID control, the adjustment time to reach the target value is 1.5 s, and GWO-PID control did not significantly increase the overshoot percentage. This clearly demonstrates that the GWO-PID controller has a significant advantage in response speed and offers better control performance.

Meanwhile, obtaining the torque of the driving wheels on the left and right sides in Simulink is shown in Figure 12.
The changes in the vehicle’s yaw rate during the simulation process were monitored and recorded, as shown in Figure 11b. Under traditional PID control, the tracked vehicle achieves a steady state at 5 s, when the vehicle entered the state of the pivot steering, the torque of the inner and outer active wheels was equal in magnitude but opposite in direction, representing driving torque, which aligns with the characteristics of the pivot steering condition. Moreover, compared to traditional PID control, the torque fluctuation range of the active wheels under GWO-PID control was smaller.

Figure 11. Simulation results of pivot steering condition.

From Figure 12, it can be seen that the vehicle remained stationary from 0 to 5 s. At 5 s, when the vehicle entered the state of the pivot steering, the torque of the inner and outer active wheels was equal in magnitude but opposite in direction, representing driving torque, which aligns with the characteristics of the pivot steering condition. Moreover, compared to traditional PID control, the torque fluctuation range of the active wheels under GWO-PID control was smaller.

Combining the results of the above simulation experiments, it is evident that the control effect of GWO-PID is superior to that of traditional PID control. However, considering that the simulation road surface model is only for lean clay roads and that tracked vehicles serve as crucial transportation vehicles in sectors such as agriculture and defense, further analysis and experimental verification on more complex road surface conditions are required. The simulation results of this study can provide a feasibility basis for the next step of research on tracked vehicle pivot steering control strategies.

6. Discussion

This manuscript investigates the control optimization of tracked vehicles under the pivot steering condition, utilizing the Grey Wolf Optimization algorithm to optimize and adjust the control parameters of the PID controller. The tracked chassis proposed in our study is applied to a hilly and mountainous environment, serving as a carrier for agricultural machinery. When operated remotely by the operator, the chassis can quickly respond to the operator’s driving intentions, improving operational efficiency. This has practical significance for our research.
This following paragraph will discuss some limitations of the full manuscript research and future research directions.

GWO-PID control is capable of enhancing the control performance of the algorithm by adjusting the PID control parameters, achieving rapid and stable control response, and reducing the percentage of overshoot. However, the selection of multiple parameters involved in the GWO-PID algorithm can affect its control effect. In practical applications, the performance of the GWO-PID control algorithm may be limited. In our manuscript, the original intention was to optimize the parameters of the PID controller, aiming to further improve and optimize the traditional PID. Currently, there are many other advanced control algorithms in the field of tracked vehicle dynamics and control, such as SOSM and NDOB [45,46]. The GWO-PID controller should be compared with these new-generation controllers in a more comprehensive simulation experiment. Our future research will be based on this, and through more specific real-vehicle experiments, we will continuously apply and improve the controller we have designed to enhance the performance of the tracked vehicle’s pivot steering, thereby enhancing the effectiveness and feasibility of our designed controller.

In our manuscript, only relevant simulation experiments have been conducted. However, in practical situations, operators are required to manually remote control the tracked vehicle for walking or steering. The control strategy proposed in our study necessitates the incorporation of human factors. By introducing more complex road conditions in extensive experiments, our control strategy can be refined and optimized, which represents an area where future research can and must advance. Furthermore, in the future, high-definition cameras and LiDAR sensors will be mounted on the chassis of tracked vehicles. These sensors can capture visual and spatial information under complex terrain and varying weather conditions, providing critical data for deep learning models [47,48]. Through deep learning algorithms, the vehicle can autonomously recognize its surroundings, plan a driving path suitable for complex terrain, and avoid obstacles in real time. Additionally, the deep learning model can analyze the driver’s facial expressions and gestures to facilitate natural human–machine interactions, ensuring safe driving even in the absence of traffic signals and other traffic markers. As technology continues to advance, this application will become more widespread and sophisticated, providing robust support for the intelligent development of tracked vehicles in complex environments such as hilly and mountainous regions.

7. Conclusions

This manuscript focuses on the study of dual-sided independent electric-driven tracked vehicles, aiming to optimize their pivot steering performance. The steering process of a tracked vehicle is modeled and, based on the decoupled control system of the tracked vehicle, controllers for vehicle speed and yaw rate are designed using PID control. Subsequently, the Grey Wolf Optimization algorithm is utilized to optimize the parameters of these controllers. A co-simulation model combining Matlab/Simulink and RecurDyn(V9R4) is established to validate the proposed control algorithms. The results indicate the following:

1. The simulation results for straight-line driving show that compared to traditional PID control, GWO-PID control has a smaller tracking overshoot percentage for the target vehicle speed, a shorter time to reach the stable target speed, and superior interference rejection capabilities.

2. The simulation results for the pivot steering demonstrate that GWO-PID control has higher accuracy than traditional PID control, faster response speed, smaller adjustment time, and a smaller torque fluctuation range of the active wheels, indicating better stability.
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