Steering Stability Control for Four-Motor Distributed Drive High-Speed Tracked Vehicles

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ABSTRACT A steering stability control method for four-motor distributed drive high-speed tracked vehicle is proposed to improve handling stability and safety. The dynamic analysis and calculation of center steering, small radius steering and large radius steering for the four-motor distributed drive tracked vehicle are carried out respectively. In order to meet the power requirements of the outer drive motor in high-speed steering, a steering coupling system is constructed, which uses a steering motor to couple with the traction motor at the rear left or rear right side of the tracks respectively. The steering stability control strategy with the steering coupling takes the yaw angular velocity and longitudinal speed of the vehicle as the control objectives, and a direct yaw torque control strategy is proposed to optimize and distribute the torque of four traction motors in real time according to different steering conditions, and considers the synchronization of two traction motors on one side of the track, so as to improve the steering stability, safety, comfort and steering trajectory performance of the four-motor drive tracked vehicle. The simulation results of 0B, 0.2B, 2B and 8B steering radius show that the proposed control strategy with coupling system can ensure the steering stability for the four-motor distributed drive high-speed tracked vehicle.

INDEX TERMS Four-motor drive, tracked vehicle, steering stability, power coupling.

I. INTRODUCTION

Tracked vehicles have better driving performance under severe working conditions, strong passing capacity, and have important applications in military, agricultural, and fire protection fields [1]. Steering is an important operating condition for tracked vehicles, which directly affects the mobility and off-road performance of high-speed tracked vehicles. Traditional high-speed tracked vehicles use a steering mechanism to complete the steering [2], [3], but there are a series of problems such as a complicated steering mechanism structure, large space occupation, low power utilization, difficult processing, and high cost. With the development of electric drive technology, distributed electric drive tracked vehicles can use the electronic controller to directly control the tracks on both sides to achieve electronic differential control steering. This effectively avoids a series of disadvantages brought by the traditional steering mechanism.

Currently tracked vehicles are mostly driven by dual motors. Each side track has a driving wheel, which is generally divided into front-front drive or rear-rear drive. The advantages are simple structure and high transmission efficiency. The disadvantage is that there is no mechanical connection between the drive motors on both sides, so the requirements for the motor drive system and electronic control are higher. When the vehicle is turning at high speed, the power demand for the outer track drive motor is relatively high, resulting in an increase in the volume and mass of the single-sided motor. In order to solve this problem, a series of dual motor coupling driven steering structures for tracked vehicles have been proposed. A dual-motor horizontal axis system scheme is proposed, but the horizontal axis system scheme is not flexible in layout and the steering stability is poor [1]. A power coupling device that solves the problem of inflexible layout of the transverse axis drive system is proposed. Although it meets the driving performance indicators of tracked vehicles and improves the power utilization, this solution only considers steady-state steering [4], [5].
A steering motor and planetary platoon between the two motors are added, however, it is through the mechanical structure that the inner regenerative power flows back to the outer side in the form of mechanical power [2], [6]. A coupling mechanism is added between the independent drive system of the two side motors, so that the steering regenerative power was transferred back to the high-speed side[7]. However, the mechanical structure of these steering mechanisms is complicated. There are mechanical constraints such as the horizontal axis and mechanical coupler between the two side motors. It is difficult to achieve electronic differential steering control and the efficiency of braking energy recovery is low.

A dual-motor driven high-speed tracked vehicle electromechanical steering coupling device is proposed [8], in which the center steering motor is dynamically coupled to the drive motors on both sides through two electromagnetic clutches and two planetary gear couplers, and an electronic steering control strategy to distribute the driving or braking torque of the motors on both sides to improve the steering stability of the vehicle is proposed. A new steering coupling device for dual-motor driven high-speed tracked vehicles is proposed, which controls the coupling mechanism to perform torque coupling or power coupling by turning on and off the electromagnetic clutch. The problems of insufficient steering torque at low speed and small radius and insufficient steering power at high speed and large radius are solved [9].

A four-wheel independent drive electric vehicle continuous steering stability control based on an energy-saving torque distribution algorithm is proposed, which improves energy efficiency [10]. A steering stability control of a four-wheel independent drive electric vehicle considering different adhesion coefficients is proposed [11].

The above research is about the tracked vehicles driven by two motors, while the research on distributed driven tracked vehicles with more than two motors is rare. However, four-wheel distributed electric drive technology has been applied to wheeled vehicles, and it has special advantages in control performance, transmission efficiency, and chassis structure. In order to solve the problems that the motor volume is too large, it is difficult to install and arrange, the vibration and noise are large, and power matching caused by the excessive power requirements of the single-sided driving motors of the dual-motor driven tracked vehicles [12], a steering control system of four-motor distributed drive tracked vehicle is constructed in this paper. There are two front and rear driving wheels on each side of the track, and four motors are used to drive the four driving wheels of the double side track. Compared with the independent driving of two motors, the volume and power of a single traction motor of a four-motor driven tracked vehicle will be reduced, and the volume and weight of the motor controller will also be reduced, which facilitates installation and layout, and it can distribute energy more flexibly and achieve maximum energy efficiency. According to the needs of different steering conditions, there are multiple motor combination modes for steering. The torque or speed of the motor can be flexibly allocated according to the steering conditions by electronic differential control, which improves steering flexibility and stability, and also improves the efficiency of the electric drive system of the vehicle. Better path tracking can be achieved by electronic differential steering control, which is conducive to the realization of unmanned driving.

However, the increasing number of traction motors will bring the difficulty of electronic differential steering control for distributed electric drive tracked vehicle, and special attention needs to be paid to steering maneuverability, stability, and synchronization of traction motors with single track. The electronic differential steering control method mainly includes a traction motor speed control method and a torque control method. Because the transient performance of the motor speed control is not as good as that of the torque control, most of the distributed vehicle electronic differential steering control uses the motor torque control method. In addition, an electronic control unit is required for comprehensive steering control to achieve high-speed steering handling stability. Many researches on steering control stability control have been carried out on distributed electric drive wheeled electric vehicles [13]–[16]. Traditional PID control, sliding mode control, fuzzy control, linear quadratic optimal control, feedforward feedback composite control and other methods are usually used to control the steering yaw moment and distribute the driving or braking torque of the motor, and finally realize the control of the yaw angular velocity and the sideslip angle of the vehicle.

The steering principle of tracked vehicles is different from that of wheeled vehicles. It needs to propose a steering control stability control strategy for distributed driving tracked vehicles based on steering kinematics and dynamics. A coupling drive steering control algorithm for dual motors drive system is proposed to achieve fast and stable control of tracked vehicle steering [17], [18]. In order to obtain better stability, a steering load adaptive control strategy based on steering angular velocity closed-loop control is proposed [19]. An electronic stability control strategy for dual-motor drive tracked vehicle is proposed to improve the maneuverability and stability of the vehicle at low speed steering with small radius and high speed steering with large radius [20], [21]. However, there are few researches on four-wheel drive tracked vehicles. Some literatures only described the steering principle of four motors drive tracked vehicle and the required torque relationship of the four driving wheels in [22]–[26]. A vehicle dynamics stability control strategy with adaptive neural network sliding mode theory [27], an interactive control paradigm-based robust lateral stability controller [28], and an implicit linear model predictive control (MPC) approach [29] are respectively proposed to improve lateral stability and tracking performance for wheeled vehicles. However, the steering control stability for tracked vehicle and its control are not considered, and the effect of the motor control system as an actuator is ignored. We propose a steering control stability control method for four-motor driven tracked vehicles.
In addition, considering the insufficient motor torque in low-speed steering with small-radius and insufficient power of the external drive motor in high-speed steering with large-radius, the distributed high-speed tracked vehicle cannot turn according to the expected trajectory, or even roll over [30]. To solve this problem, a simple steering coupling system based on a steering motor is proposed in this paper. The method of electromechanical coupling and integrated electronic differential steering stability control is used to further improve vehicle handling stability and trajectory tracking performance.

The main work and innovations of this paper are as follows:
(1) A electric drive steering system for four-motor distributed drive high-speed tracked vehicle is proposed, and a power coupling system for the rear driving wheels of the tracks on both sides for steering is constructed, which improves the power of the outer drive motors for high-speed large-radius steering and the energy utilization of electric drive systems.

(2) An optimal control strategy of direct yaw torque for steering stability of the four-motor distributed drive tracked vehicle based on torque control is proposed, which takes the yaw angular velocity and longitudinal speed of the vehicle as the control objectives, and considers the synchronization of two traction motors on one side of the track. According to different steering conditions, the torque of the four traction motors is distributed in real time to improve the steering stability, safety, comfort and steering trajectory performance of the four-motor drive tracked vehicle.

The organizational structure of this paper is as follows: Section 2 analyzes the steering kinematics and dynamics of a four-motor-driven tracked vehicle; Section 3 proposes a steering coupling device; Section 4 proposes a steering stability control strategy; Section 5 models and simulates the stability control strategy. Section 6 gives the conclusions and the prospects.

II. STEERING KINEMATICS AND DYNAMICS ANALYSIS

A. ELECTRIC DRIVE SYSTEM STRUCTURE

Each track of the four-motor-driven tracked vehicle has two driving wheels. As shown in Figure 1, the electric drive system is mainly composed of power battery pack, energy absorption unit, power distribution unit, integrated electronic controller, four motor controllers, four traction motors and four reducers. It also includes accelerator pedal and displacement sensor, electronic gear acquisition unit, brake pedal and displacement sensor, steering wheel and angular displacement sensor. According to the steering demand, the integrated electronic controller distributes the torque signals of the four motors to the four motor controllers to control the four traction motors respectively.

B. DEMAND TORQUE AND POWER OF FOUR-MOTOR FOR STEERING

Four motors independently drive tracked vehicles has the characteristics of flexible steering and diversified steering modes. There are five working modes: left front and right front motors in operation, left rear and right back motors in operation, left front and right back motors in operation, left and right front motors in operation and four motors working simultaneously. Because the first four steering modes are special cases of the last mode, the following only focuses on the kinematics and dynamics analysis of the last steering mode. Under typical steering conditions of center steering, small radius steering (0 < R < 0.5B, R = 0.5B), and large radius steering (R > 0.5B), respectively, the torque and power requirements of each traction motor are obtained according to the vehicle steady state and dynamic steering kinematics and dynamics.

1) KINEMATICS MODEL

A kinematics analysis schematic diagram of the track vehicle steering is shown in Figure 2.

Where, C is the plane center of the tracked vehicle, O is the instantaneous steering center, C1 and C2 are the track grounding centers, R is the turning radius, L is the track ground length, B is the track center distance, v1 and v2 are the inner and outer track speeds, and \( \omega \) is the steering angular speed.
The following dynamic equation can be obtained as:

\[
\begin{align*}
\delta v &= \frac{F_2 - F_1 + R_1 - R_2}{\frac{B}{2} - M\mu - (R_1 + R_2)\times \frac{B}{2}} = J\frac{d\omega}{dt} \\
F_2 &= F_1 + R_1 - R_2 = \frac{B}{2} - M\mu - (R_1 + R_2)\times \frac{B}{2}
\end{align*}
\]

where \(R_1 = R_2 = 0.5fmg\), \(mg\) is the total body weight, \(\mu\) is the steering resistance coefficient. The ground steering resistance torque \(M\mu\) is:

\[
M\mu = \frac{\mu mgL}{4} = \frac{\mu_{\text{max}}}{0.925 + 0.15\rho} \cdot mgL
\]

where \(\rho\) is the relative turning radius, and \(\rho = R / B\).

According to (3), \(F_1\) and \(F_2\) should meet:

\[
F_1 = F_2 = \frac{1}{2}fmg + \frac{\mu mgL}{4B} + \frac{J d\omega}{B dt}
\]

The torques of the four traction motors can be obtained as:

\[
T_{11} = T_{12} = T_{21} = T_{22} = \left(\frac{fmg + \mu mgL}{2B} + \frac{J d\omega}{B dt}\right) \times r_c
\]

where, all four motors output driving torque, and the driving torque of inner motor is opposite to that of outer motor, which is equal to the driving torque of the outer motor.

The power of the four traction motors can be obtained as:

\[
P_{11} = P_{12} = P_{21} = P_{22} = \frac{\left(\frac{fmg + \mu mgL}{2B} + \frac{J d\omega}{B dt}\right) \times r_c}{2\eta_\gamma n_1}
\]

where, all four motors output driving torque, and the driving torque of inner motor is opposite to that of outer motor, which is equal to the driving torque of the outer motor.

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\[
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\]
4) STEERING WITH RADIUS R = 0.5B
From Figure 3(c), at the case of R = 0.5B, the speed of the inner track is zero, so $R_1 = 0$. The following dynamic equation can be obtained:

$$
F_2 - F_1 - R_2 = \delta m \frac{dv}{dt} \\
F_2B - M_\mu - R_2B = J \frac{d\omega}{dt}
$$

(11)

The drive torques of the four traction motors can be described as:

$$
T_{11} = T_{12} = \left(\frac{\mu mg}{4B} + \left(\frac{B}{R} - \delta mR\right) \frac{d\omega}{dt}\right) r_c \\
T_{21} = T_{22} = \left(\frac{\mu mg}{4B} + \frac{1}{2}\delta mR \frac{d\omega}{dt}\right) r_c
$$

(12)

where, the driving torque of inner motor is opposite to that of outer motor and less than that of outer motor, and the driving torques of the inner motors shall ensure that the speed of the inner motor is zero.

The power of the four traction motors can be obtained as:

$$
P_{11} = P_{12} = \left(\frac{\mu mg}{4B} + \left(\frac{B}{R} - \delta mR\right) \frac{d\omega}{dt}\right) r_c \times \frac{1000i}{120\pi r_c} \times 3.6 \times \omega \times \left(\frac{B}{2} - R\right) \times \frac{9549}{2i_\eta} \\
P_{21} = P_{22} = \left(\frac{\mu mg}{4B} + \frac{1}{2}\delta mR \frac{d\omega}{dt}\right) r_c \times \frac{1000i}{120\pi r_c} \times 3.6 \times \omega \times \left(\frac{B}{2} + R\right) \times \frac{9549}{2i_\eta}
$$

(13)

5) STEERING WITH LARGE RADIUS
The above three small-radius steering conditions are suitable for scenarios with limited space. The tracked vehicle is steering from stationary starting and the initial speed is zero. When the vehicle is turning, the steering radius is generally large, usually $R > 0.5B$. Depending on the output driving torque or braking torque of the motor, the steering with large radius $R > 0.5B$ can be further divided into three operating conditions: braking steering, free steering, and driving steering.

At case of braking steering, as shown in Figure 3(d):

The following dynamic equation can be obtained:

$$
F_2 - F_1 - R_2 - R_1 = \delta m \frac{dv}{dt} \\
(F_2 - F_1) \times \frac{B}{2} + (R_1 - R_2) \times \frac{B}{2} - M_\mu = J \frac{d\omega}{dt}
$$

(14)

where, the braking force of the inner track of the tracked vehicle is the same as the direction of rolling resistance and opposite to the direction of the inner track. The drive torques of the four traction motors can be described as:

$$
T_{11} = T_{12} = \left(\frac{\mu mg}{4B} - \frac{1}{2} fmg + \left(\frac{B}{B} - \frac{1}{2}\delta mR\right) \frac{d\omega}{dt}\right) r_c \\
T_{21} = T_{22} = \left(\frac{\mu mg}{4B} + \frac{1}{2} fmg + \left(\frac{B}{B} + \frac{1}{2}\delta mR\right) \frac{d\omega}{dt}\right) r_c
$$

(15)

where, the inner two motors are in the power generation state and output the braking torque, however the outer two motors are in the driving state and output the driving torque.

The power of the four traction motors can be obtained as:

$$
P_{11} = P_{12} = \left[\frac{\mu mg}{4B} - \frac{1}{2} fmg + \left(\frac{B}{B} - \frac{1}{2}\delta mR\right) \frac{d\omega}{dt}\right] \times \frac{1000i}{120\pi r_c} \times 3.6 \times \omega \times \left(\frac{B}{2} - R\right) \times \frac{9549}{2i_\eta} \\
P_{21} = P_{22} = \left[\frac{\mu mg}{4B} + \frac{1}{2} fmg + \left(\frac{B}{B} + \frac{1}{2}\delta mR\right) \frac{d\omega}{dt}\right] \times \frac{1000i}{120\pi r_c} \times 3.6 \times \omega \times \left(\frac{B}{2} + R\right) \times \frac{9549}{2i_\eta}
$$

(16)

where, the energy generated by the brake motor inside the brake can be fed back to the power battery through the motor controller.

At case of free steering, as shown in Figure 3(e):

The inner track of the tracked vehicle is under no force. The torque expressions of the four motors are:

$$
T_{11} = T_{12} = 0 \\
T_{21} = T_{22} = \left(\frac{fmg + \delta mR \frac{d\omega}{dt}}{2i_\eta}\right) r_c
$$

(17)

The inner motor does not output driving torque or braking torque. The two outer motors are in the driving state and output driving torque.

The power of the four traction motors can be obtained as:

$$
P_{11} = P_{12} = 0 \\
P_{21} = P_{22} = \left(\frac{fmg + \delta mR \frac{d\omega}{dt}}{2i_\eta}\right) r_c \times \frac{1000i}{120\pi r_c} \times 3.6 \times \omega \times \left(\frac{B}{2} + R\right) \times \frac{9549}{2i_\eta}
$$

(18)

At case of driving steering, as shown in Figure 3(f):

This method is usually used when the vehicle is driving at high speed and straight. The driving force of the outer track of a tracked vehicle is greater than the driving force of the inner track. The torque expressions of the four motors are:

$$
T_{11} = T_{12} = \left[\frac{1}{2} fmg - \frac{\mu mg}{4B} + \left(\frac{1}{2}\delta mR - \frac{1}{2}\frac{B}{B}\right) \frac{d\omega}{dt}\right] \times \frac{2i_\eta}{9549} \\
T_{21} = T_{22} = \left[\frac{1}{2} fmg + \frac{\mu mg}{4B} + \left(\frac{1}{2}\delta mR + \frac{1}{2}\frac{B}{B}\right) \frac{d\omega}{dt}\right] \times \frac{2i_\eta}{9549}
$$

(19)

where, all four motors output driving torque, and the driving force direction of the tracks on both sides is same. The power of the four traction motors can be obtained as:

$$
P_{11} = P_{12} = \left[\frac{1}{2} fmg - \frac{\mu mg}{4B} + \left(\frac{1}{2}\delta mR - \frac{1}{2}\frac{B}{B}\right) \frac{d\omega}{dt}\right] \times \frac{1000i}{120\pi r_c} \times \frac{\omega}{2i_\eta} \times \left(\frac{B}{2} - R\right) \times \frac{9549}{2i_\eta} \\
P_{21} = P_{22} = \left[\frac{1}{2} fmg + \frac{\mu mg}{4B} + \left(\frac{1}{2}\delta mR + \frac{1}{2}\frac{B}{B}\right) \frac{d\omega}{dt}\right] \times \frac{1000i}{120\pi r_c} \times \frac{\omega}{2i_\eta} \times \left(\frac{B}{2} + R\right) \times \frac{9549}{2i_\eta}
$$

(20)
### C. MOTOR SELECTION

First, a motor is selected according to the vehicle’s dynamic performance indicators, and then whether the selection of the motor meets the vehicle’s steering dynamic performance is analyzed. The selected tracked vehicle parameters are shown in Table 1. and the dynamic performance indicators are as follows: the maximum driving speed is 50km/h; the maximum gradient is 32° and the maximum climbing speed is 20km/h; the acceleration time for 0~32km/h is less than or equal to 8s. Based on the vehicle’s dynamic performance requirements [9], the performance parameters of the traction motor are calculated as shown in Table 2.

| Symbol | Parameters | Value |
|--------|------------|-------|
| B      | Track center distance | 1000 mm |
| 2n     | Number of rounds | 10 |
| r_s    | Driving wheel radius | 100 mm |
| L      | The length of track contact ground | 1.2 m |
| f      | Rolling resistance coefficient | 0.06 |
| \( \mu_{\text{max}} \) | Maximum steering resistance coefficient | 1 |
| \( \eta \) | Transmission efficiency | 0.9 |
| \( m \)  | Vehicle quality | 200 kg |
| \( J \)  | Moment of inertia | 300 kg/m² |
| A      | Frontal area | 1.44m² |
| \( C_0 \) | Air resistance coefficient | 0.6 |

### III. STEERING COUPLING SYSTEM

#### A. THE STRUCTURE OF STEERING COUPLING SYSTEM

In order to improve the power of the outer motor of high-speed large radius steering and improve the energy utilization rate of the electric drive system, a power coupling system for the rear driving wheels of the tracks on both sides for steering is constructed, as shown in Figure 4. The steering coupling system is mainly composed of a steering motor, a coupler, and two electromagnetic brake clutch devices.

![Figure 4. Schematic diagram of the steering coupling system.](image)

The coupler includes a planetary gear coupling device and a brake, as shown in Figure 5. The electromagnetic brake clutch device includes a brake stator, a brake clutch rotor and a clutch rotor. As shown in Figure 6, by controlling the combination or separation of the brake clutch rotor and the brake stator or the clutch rotor, the left shaft is controlled to be fixed or connected with the right shaft.

![Figure 5. Schematic diagram of coupled steering.](image)

The coupler does not work when turning at low speed and small radius, when the planetary gear coupler is used as a reducer. The integrated electronic controller controls the steering motor not to work. The brake clutch rotor of the electromagnetic brake clutch device is disconnected from the clutch rotor, and at the same time it is engaged with the...
TABLE 3. Four-motor steering torque and power requirements under small radius steering conditions.

| situation  | R(m) | T₁(N m) | T₂(N m) | n₁(τ/min) | n₂(τ/min) | P₁(kW) | P₂(kW) |
|------------|------|---------|---------|-----------|-----------|--------|--------|
| dynamic    | 0    | 19.4    | 19.4    | 197       | 197       | 0.4    | 0.4    |
|            | 0.2B | 18.4    | 19.7    | 118       | 276       | 0.23   | 0.57   |
|            | 0.5B | 14.4    | 18.57   | 0         | 394       | 0      | 0.77   |
| stationary | 0    | 12.5    | 12.5    | 197       | 197       | 0.26   | 0.26   |
|            | 0.2B | 12.1    | 12.1    | 118       | 276       | 0.15   | 0.35   |
|            | 0.5B | 10.6    | 11.6    | 0         | 394       | 0      | 0.48   |

TABLE 4. Steering torque and power requirements of four motors under large radius steering conditions.

| situation         | R(m) | T₁(N m) | T₂(N m) | n₁(τ/min) | n₂(τ/min) | P₁(kW) | P₂(kW) |
|-------------------|------|---------|---------|-----------|-----------|--------|--------|
| Brake steering    | 2B   | 7.58    | 9.70    | 591       | 984       | 0.47   | 1.00   |
|                   | 5B   | 5.26    | 7.38    | 1772      | 276       | 2.16   | 1.67   |
|                   | 8B   | 3.92    | 6.04    | 2953      | 3347      | 1.22   | 2.12   |
|                   | 10B  | 3.31    | 5.42    | 3741      | 4134      | 1.30   | 2.35   |
|                   | 12B  | 2.83    | 4.94    | 4528      | 4922      | 1.34   | 2.55   |
| Free steering     | 60.5 | 0       | 0.89    | 3938      | 4003      | 0      | 0.89   |
| Driving steering  | 70B  | 0.13    | 1.98    | 4560      | 4626      | 0.06   | 0.96   |

FIGURE 6. Electromagnetic brake clutch device.

brake stator to brake the ring gear and make it stationary. The drive motor is controlled to generate power output to the drive wheels for differential steering. The power transmission diagram is shown in Figure 7.

FIGURE 7. Schematic diagram of power transmission of electronic differential power steering.

When the electromechanical coupling steering, the integrated electronic controller controls the front drive motor to drive the front drive wheel. At this time, the outer rear driving wheels are dynamically coupled. The integrated electronic controller controls the operation of the steering motor. The outer electromagnetic brake and clutch device disconnects the clutch rotor from the brake stator and combines with the clutch rotor to connect the output shaft of the steering motor to the input shaft of the coupling reduction gear. The output power of the steering motor is transmitted to the planetary gear coupler through the electromagnetic brake clutch device, and is coupled to the input power of the drive motor and output to the outer driving wheel. The schematic diagram of power transmission is shown in Figure 8. At this time, the inner rear driving wheel is independently driven by the drive motor, and the power transmission is shown in Figure 8.

FIGURE 8. Schematic diagram of power transmission of electromechanical coupling steering.

B. PLANETARY GEAR COUPLING DEVICE

The planetary gear coupling device is used for power and torque coupling of the drive motor and the steering motor. It works when the tracked vehicle is turning with large radius at high speed to meet the high power demand of the motor. The power coupler does not work at low speed steering with small radius. The motor meets the high torque demand under small radius steering through reducer. The operation of the coupler is controlled by the electronic clutch and electronic brake. Therefore, the steering coupling mechanism can meet the torque and power requirements of tracked vehicles under different steering radii, and improve the steering performance.

1) PLANETARY GEAR COUPLING MECHANISM SCHEME

In order to reduce the size and weight of the coupler, the planetary gear device adopts a single-row planetary mechanism. As shown in Figure 7, the mechanism is composed of a sun gear s, a ring gear r, a planetary gear p, and a planet
carrier c. Planetary gears act as idlers in the mechanism. The single-sided drive motor is connected to the sun gear input shaft, the steering motor is connected to the ring gear, and is connected to the planet carrier through the planet gear. The planet carrier transmits the power of the driving motor and the steering motor to the driving wheels on both sides.

2) PLANETARY COUPLING DEVICE STRUCTURE PARAMETERS
In order to determine the parameters of each component of the planetary gear mechanism, the structural characteristic parameter $k$ of the planetary gear mechanism needs to be determined first. According to the table 3, the maximum torque required by the unilateral traction motor when the vehicle is in dynamic center steering is $T_{max} = 19.4\text{Nm}$. According to the table and formula, the maximum torque. Define $T_m$ as the output torque of the single-sided motor, $T_0$ as the output torque $f$ the steering motor, $T_s$ as the ring gear input torque, $T_r$ as the sun gear input torque, $i_0$ is the reduction ratio of the steering motor to the ring gear, and it is determined that $i_0 = 3$. Meet the following relationships:

$$\begin{align*}
T_m &= T_s \\
T_r &= i_0 \times T_0
\end{align*}$$

(21)

The torque relationship between the sun gear and the planet carrier:

$$k = \frac{T_r}{T_s} - 1$$

(22)

It can be obtained that the structural characteristic parameter of the planetary gear mechanism is $k = 2.15$. The following is to verify whether the structural characteristic parameter $k$ can meet the power requirements of the tracked vehicle during high-speed braking and steering.

\(a\): SPEED VERIFICATION
First, the speed of the driving wheel is calculated from the structural characteristic parameters.

According to the table 4 and formula (1), when the turning radius of the tracked vehicle is 10B, the steering speed reaches a maximum of 50km/h. At this time, the traction motor and steering motor output the peak speed and peak power, that is, $P_m = P_s = 2\text{kW}$, $n_m = n_s = 4500\text{r/min}$, $n_r = n_m/i_0 = 1500\text{r/min}$. From this we can get the planetary carrier speed:

$$n_c = \frac{1}{1 + k} n_s + \frac{k}{1 + k} n_r$$

(23)

where $n_s$ is the speed of the sun gear and $n_r$ is the speed of the ring gear. Therefore, the calculated output speed to the driving wheel is $n_c = 2452\text{r/min}$.

According to (23), the speed of the inner and outer driving wheels is calculated as: $n_1 = 1260\text{r/min}$, $n_2 = 1392\text{r/min}$. From the above calculations, it can be seen that the maximum speed $n_c$ of the driving wheel obtained from the structural characteristic parameters is greater than $n_1$ and $n_2$, so the structural characteristic parameters can meet the speed requirements for high-speed steering.

\(b\): POWER VERIFICATION
According to the relationship between torque, speed and power, it can be obtained that when the tracked vehicle has a steering radius of 10B, the torque output by the driving wheels is: $T_s = T_{in} = 4.244\text{Nm}$, $T_c = (1+k)T_s = 13.37\text{Nm}$. According to the maximum speed of the driving wheel $n = 2452\text{r/min}$, the output power of the planet carrier at this time can be calculated as: $P_c = 3.43\text{kW}>2.35\text{kW}$. In summary, the structural characteristic parameter $k = 2.15$ of the planetary gear mechanism can meet the needs of maximum torque and the requirements of maximum power. The main structural parameters of the planetary gear mechanism are designed based on theoretical calculations, as shown in Table 5. The planetary gear coupling mechanism assembly is shown in Figure 9. With the designed planetary coupling device, the peak power of a single drive motor can be reduced to 2kW instead of 2.55kW. The power of the steering motor is 1.1kW. The total power of the four motors and one steering motor is 9.1kW, which is nearly 10% lower than the total power of the 10.2kW independent drive motors of the four motors.

\begin{table}[h]
\centering
\caption{Dimension parameters of main components of planetary gear mechanism.}
\begin{tabular}{|c|c|c|c|c|c|c|}
\hline
component & Number of teeth & Modulus m(mm) & Graduation circle d(mm) & Tooth top circle $d_s$& Tooth root circle $d_r$ & Tooth width b(mm) \\
\hline
Sun gear & 40 & 1 & 40 & 42 & 37.5 & 15.75 \\
Planetary gear & 23 & 1 & 23 & 25 & 20.5 & 15.75 \\
Ring gear & 86 & 1 & 86 & 84.2 & 87.82 & 15.75 \\
Power coupling gear (ring gear) & 120 & 0.5 & 60 & 61 & 57.5 & 24 \\
Power coupling gear (input gear) & 40 & 0.5 & 20 & 21 & 17.5 & 24 \\
\hline
\end{tabular}
\end{table}

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{fig9}
\caption{Assembly drawing of planetary coupler.}
\end{figure}
IV. STEERING STABILITY CONTROL STRATEGY FOR FOUR-MOTOR DRIVEN TRACKED VEHICLES

A. DRIVER INPUT MODEL

The driver input model includes a steering wheel angular displacement signal, an accelerator pedal signal and a brake pedal signal to simulate a driver’s operation input. The input angle signal of steering wheel displacement is shown in Figure 10. According to different steering wheel angle signals, three angles \( \theta_1, \theta_2, \theta_3 \), and \( \theta' \) are defined. \( \theta_1 \) indicates the steering wheel angle when the tracked vehicle is turning freely, that is, the steering wheel angle when the steering radius \( R = R_0 \); \( \theta_2 \) indicates the steering wheel angle when the steering radius \( R = 0.5B \); \( \theta_3 \) indicates the center steering \( R = 0 \). \( \theta' \) is a boundary angle for judging whether the vehicle is going straight or turning.

![FIGURE 10. Steering wheel angle.](image)

The scaling factor \( x \) of steering wheel angular displacement signal is defined as:

\[
x = \begin{cases} 
0 & 0 \leq \theta \leq \theta_1 \\
1 - \frac{\theta - \theta_1}{\theta_2 - \theta_1} & \theta_1 < \theta < \theta_2 \\
-\frac{\theta - \theta_2}{\theta_3 - \theta_2} & \theta_2 < \theta < \theta_3 
\end{cases}
\]  

(24)

As the displacement of accelerator pedal and brake pedal is simplified into linear relationship with the output torque of motor, there are

\[
A = \frac{\alpha - \alpha_0}{\alpha_{\text{max}} - \alpha_0} \\
T_{\text{des}} = A \times T_{\text{max}}
\]  

(25)

where \( A \) is the accelerator pedal displacement signal proportional coefficient. \( \alpha \) is the displacement of the accelerator pedal, \( \alpha_0 \) and \( \alpha_{\text{max}} \) are the free displacement and the maximum displacement of the accelerator pedal respectively, and \( T_{\text{max}} \) is the maximum output torque of the motor. The value range of \( A \) is \([0,1]\).

\[
C = -\frac{\beta - \beta_0}{\beta_{\text{max}} - \beta_0} \\
T_{\text{des-braking}} = C \times T_{\text{max-braking}}
\]  

(26)

where \( C \) is the brake pedal displacement signal proportional coefficient. \( \beta \) is the displacement of the brake pedal at this time, \( \beta_0 \) and \( \beta_{\text{max}} \) are the free displacement and the maximum displacement of the brake pedal respectively, and \( T_{\text{max-braking}} \) is the maximum output braking torque of the motor. The value range of \( C \) is \([-1,0]\).

Defining the steering wheel angle \( \theta \in (0,45^\circ) \), the linear relationship between steering radius and steering wheel displacement is:

\[
R = \frac{45^\circ - \theta}{\theta} \cdot \frac{B}{2}
\]  

(27)

where \( B \) is the distance between the left and right tracks of the tracked vehicle.

The driving force acting on the driving wheel is:

\[
F_i(t) = \frac{T_{\text{m}}(t) \cdot i_g \cdot \eta_t}{r}
\]  

(28)

Among them, \( i_g \) is the transmission ratio of the reducer, \( \eta_t \) is the efficiency of the transmission system, and \( T_{\text{m}} \) is the motor torque.

From the tracked vehicle force balance equation:

\[
\sum F(t) = F_j(t) + F_w(t) + F_r(t) + F_f(t)
\]

\[
F_j(t) = m \cdot f(t)
\]

\[
F_f(t) = m \cdot a(t)
\]  

(29)

where, \( F_j \) is the rolling resistance, \( F_f \) is the acceleration resistance. \( F_w \) and \( F_f \) are the air resistance and the slope resistance, which are not considered here. Vehicle acceleration can be obtained from (28) and (29):

\[
a(t) = \frac{T_{\text{m}}(t) \cdot i_g \cdot \eta_t}{m \cdot r} - f(t)
\]  

(30)

The tracked vehicle speed is:

\[
v = \int_0^t a(t) dt = \int_0^t \left[ \frac{T_{\text{m}}(t) \cdot i_g \cdot \eta_t}{m \cdot r} - f(t) \right] dt
\]  

(31)

B. TRACKED VEHICLE STEERING CONTROL STRATEGY

Using the established steering coupling system, a direct yaw torque control strategy for steering stability of a four-motor distributed drive tracked vehicle based on torque control is established, as shown in Figure 11. The vehicle’s longitudinal speed and yaw rate are used as control targets. The steering stability control strategy is divided into three layers, including the vehicle ideal reference model layer, the upper controller, and the lower torque distribution controller.

1) REFERENCE MODEL LAYER

Based on the tracked vehicle longitudinal force and yaw moment expression 32 and the steering control strategy shown in Figure 11 to establish the vehicle ideal reference model layer. The desired vehicle speed \( v_{\text{des}} \) and yaw angular speed \( \omega_{\text{des}} \) are obtained according to the driver’s input of the accelerator and brake pedal signals \( A \) and \( C \) and the steering wheel angular displacement signal \( x \), which are input to the upper controller. The upper controller includes a yaw rate controller and a speed follower controller.
2) UPPER CONTROLLER
The yaw rate controller uses fuzzy PID control to adjust the desired yaw moment \( M_{\text{z-des}} \) according to the deviation between the actual vehicle speed \( \omega \) and the desired speed \( \omega_{\text{des}} \). As shown in Figure 12. The fuzzy rules are designed in the same way as that in [9].

![Figure 12: Steering angular velocity tracking module.](image)

When turning radius \( R = R_0 \),
\[
\begin{align*}
F_{x-des} &= \frac{T_{21}}{r} + \frac{T_{22}}{r} - \frac{T_{11}}{r} - \frac{T_{12}}{r} \\
M_{z-des} &= \frac{T_{21}}{r} \times B \times \frac{2}{r} + \frac{T_{22}}{r} \times B \times \frac{2}{r} - \frac{T_{11}}{r} \times B \times \frac{2}{r} - \frac{T_{12}}{r} \times B \times \frac{2}{r}
\end{align*}
\]

When turning radius \( R > R_0 \),
\[
\begin{align*}
F_{x-des} &= \frac{T_{21}}{r} + \frac{T_{22}}{r} - \frac{T_{11}}{r} - \frac{T_{12}}{r} \\
M_{z-des} &= \frac{T_{21}}{r} \times B \times \frac{2}{r} + \frac{T_{22}}{r} \times B \times \frac{2}{r} - \frac{T_{11}}{r} \times B \times \frac{2}{r} - \frac{T_{12}}{r} \times B \times \frac{2}{r}
\end{align*}
\]

3) LOWER CONTROLLER

a: DRIVE WHEEL TORQUE OPTIMAL DISTRIBUTION MODULE
The lower-level torque distribution controller adopts an optimized distribution method, and calculates the output torques of the four driving wheels inside and outside based on the input longitudinal force \( F_{x-des} \) and the expected yaw moment \( M_{z-des} \).

According to the analysis of the steering dynamics of different steering radius on hard roads, the longitudinal force and yaw moment of four-motor driven tracked vehicles with different steering radius can be obtained as follows.

When the turning radius is \( 0 < R < R_0 \),
\[
\begin{align*}
F_{x-des} &= \frac{T_{21}}{r} + \frac{T_{22}}{r} - \frac{T_{11}}{r} - \frac{T_{12}}{r} \\
M_{z-des} &= \frac{T_{21}}{r} \times B \times \frac{2}{r} + \frac{T_{22}}{r} \times B \times \frac{2}{r} - \frac{T_{11}}{r} \times B \times \frac{2}{r} - \frac{T_{12}}{r} \times B \times \frac{2}{r}
\end{align*}
\]

When turning radius \( R = R_0 \),
\[
\begin{align*}
F_{x-des} &= \frac{T_{21}}{r} + \frac{T_{22}}{r} - \frac{T_{11}}{r} - \frac{T_{12}}{r} \\
M_{z-des} &= \frac{T_{21}}{r} \times B \times \frac{2}{r} + \frac{T_{22}}{r} \times B \times \frac{2}{r} - \frac{T_{11}}{r} \times B \times \frac{2}{r} - \frac{T_{12}}{r} \times B \times \frac{2}{r}
\end{align*}
\]

Rewriting \( T_{11}T_{12}T_{21}T_{22} \) into a matrix form is:
\[
\kappa = Hu
\]

where
\[
\kappa = \begin{bmatrix} M_{z-des} \\ F_{x-des} \end{bmatrix}
\]
\[
u = \begin{bmatrix} T_{11} & T_{12} & T_{21} & T_{22} \end{bmatrix}^T
\]
\[
H = \begin{bmatrix}
\frac{r}{2} & \frac{B}{2} & \frac{r}{2} & \frac{B}{2} \\
-\frac{1}{r} & -\frac{1}{r} & \frac{2r}{r} & \frac{2r}{r} \\
0 & 0 & \frac{1}{r} & \frac{1}{r} \\
-\frac{B}{r} & -\frac{r}{r} & \frac{2r}{r} & \frac{2r}{r}
\end{bmatrix}, \quad (0 \leq R < R_0)
\]
\[
H = \begin{bmatrix}
\frac{r}{2} & \frac{B}{2} & \frac{r}{2} & \frac{B}{2} \\
-\frac{1}{r} & -\frac{1}{r} & \frac{2r}{r} & \frac{2r}{r} \\
0 & 0 & \frac{1}{r} & \frac{1}{r} \\
-\frac{B}{r} & -\frac{r}{r} & \frac{2r}{r} & \frac{2r}{r}
\end{bmatrix}, \quad (R = R_0)
\]
\[
H = \begin{bmatrix}
\frac{r}{2} & \frac{B}{2} & \frac{r}{2} & \frac{B}{2} \\
-\frac{1}{r} & -\frac{1}{r} & \frac{2r}{r} & \frac{2r}{r} \\
0 & 0 & \frac{1}{r} & \frac{1}{r} \\
-\frac{B}{r} & -\frac{r}{r} & \frac{2r}{r} & \frac{2r}{r}
\end{bmatrix}, \quad (R > R_0)
\]

In order to ensure the vehicle good handling and stability on steering, i.e. to make actual yaw rate \( \omega \) follow the required reference output \( \omega_{\text{des}} \) quickly. So the error of \( M_z \) shall be minimized, an optimum torque allocation control is used to obtain the control inputs for four motors, that is,
function can be found finally and the torque of the required
variables, a cost function is described as follows:

$$J_1 = \arg\min_{u_{\min} \leq u \leq u_{\max}} \left[ \|B_v \cdot (H \cdot u - \kappa)\|^2 \right]$$  (39)

$$u = \begin{bmatrix} T_{11} \
T_{12} \
T_{21} \
T_{22} \end{bmatrix}^T$$  (40)

$$B_v = \begin{bmatrix} k_1 & 0 \
0 & k_2 \end{bmatrix}$$  (41)

where $B_v$ is the diagonal weighting matrix of the input, $k_i$ is the
weighting coefficient.

In addition, considering energy consuming of the motors, the
sum of torque command squares as small as possible to save energy while ensuring stability of vehicle motion. A cost
function for energy saving is described as follows:

$$J_2 = \arg\min_{u_{\min} \leq u \leq u_{\max}} \left[ \|B_u \cdot u\|^2 \right]$$  (42)

$$B_u = \begin{bmatrix} n_1 & 0 & 0 & 0 \
\eta_1 & 0 & 0 & 0 \
\eta_2 & n_2 & 0 & 0 \
\eta_3 & 0 & n_3 & 0 \
\eta_4 & 0 & 0 & n_4 \end{bmatrix}$$  (43)

where $B_u$ is the weighting matrix of efficient driving, $\eta_1$, $\eta_2$, $\eta_3$, $\eta_4$ are the real-time motor efficiency obtained by the
efficiency map table of the motor at the corresponding
time.

Another consideration is the stability of the change of control variables. A cost function is considered to control the
rate of change of the control action as follows:

$$J_3 = \arg\min_{u_{\min} \leq u \leq u_{\max}} \left[ \|B_w \cdot \Delta u\|^2 \right]$$  (44)

where $B_w$ is positive weight factor to coordinate the rate of change of inputs $T_{11}$, $T_{12}$, $T_{21}$, $T_{22}$.

Therefore, considering the above three optimization objectives, a cost function is described as follows:

$$J = \arg\min_{u_{\min} \leq u \leq u_{\max}} \left[ \xi_1 \cdot \|B_v \cdot (H \cdot u - \kappa)\|^2 \
+ \xi_2 \cdot \|B_u \cdot u\|^2 + \xi_3 \cdot \|B_w \cdot \Delta u\|^2 \right]$$  (45)

where $\xi_1$ is a weight coefficient for regulating tracking error, which depends on the degree of attention to distribution error. $\xi_2$ is a weight coefficient, which depends on the degree of attention to energy saving effect. $\xi_3$ is the weight of the modified torque, which is smaller than $\xi_1$ any $\xi_2$.

The optimal cost function has the following constraints:

$$-T_{\text{max} - \text{brake}} \leq T_{11,12,21,22} \leq T_{\text{max} - \text{drive}}$$

$$|T_{11,12,21,22}| \leq \frac{r_m g \mu}{4}$$

$$\eta_{11,12,21,22} \geq 85\%$$

$$|\Delta T_{11,12,21,22}| / |T_{11,12,21,22}| \leq 0.1$$  (46)

In this way, the optimal solution to satisfy the objective function can be found finally and the torque of the required
driving wheel can be obtained.

b: STEERING MODE MODULE

From the steering mode module, whether to use electronic differential steering or electromechanical coupling steering
is determined.

If the driving torque of the outer front and rear drive wheels is less than or equal to the maximum torque of a single motor, the four driving wheels are independently controlled by elec-

From the steering mode module and (14) the desired outer rear driving
wheel through the proposed power coupler. From the steering mode module and the expected
torque $T_{\text{des} - \text{m}}$ of the steering motor will be obtained.

C. SYNCHRONOUS SPEED CONTROLLER

In order to ensure the synchronization of the front and rear
driving wheels of one side, a synchronous speed controller of motor is designed, and the fuzzy PID control method is adopted, as shown in Figure 13. The fuzzy rules are designed in the same way as that in [9]. The expected torque of the driving wheel is compensated by torque error from the synchronous speed controller. Finally, the expected torque of the driving wheel $T_{11 - \text{des} - \text{m}}$, $T_{12 - \text{des} - \text{m}}$, $T_{21 - \text{des} - \text{m}}$, $T_{22 - \text{des} - \text{m}}$ are expressed as:

$$T_{11 - \text{des}} = T_{11} + \Delta T_1$$

$$T_{12 - \text{des}} = T_{12} + \Delta T_2$$

$$T_{21 - \text{des}} = T_{21} + \Delta T_3$$

$$T_{22 - \text{des}} = T_{22} + \Delta T_4$$  (47)

V. SIMULATION

Multi-body dynamics software RecurDyn is used to establish
a virtual prototype model, and then an electronic differential steering control system is established in Matlab / Simulink.
Using the Control interface function provided in RecurDyn, co-simulation of RecurDyn and Matlab / Simulink can be performed to realize synchronous control of virtual prototype
model and steering control strategy system. The model of the four-motor-driven distributed-drive tracked vehicle sim-
ulation system is shown in Figure 14, and consists of a driver
model module, an upper-level control module, a lower-level
torque distribution control module, a steering coupling device
module, and a tracked vehicle dynamic model.
A. ELECTRONIC DIFFERENTIAL INDEPENDENT STEERING CONTROL

1) CENTER TURN

It can be known from theoretical analysis that the tracked vehicle has a large demand for torque when performing center steering. First, the steering wheel is turned to the left to $\theta = \theta_3$ (turn left), and the required vehicle yaw angular velocity of $5\pi/12$ rad/s is given, so that the tracked vehicle starts to turn around the center of the vehicle from stop. The simulation results are shown in Figure 15.

As shown in Figure 15(a), the vehicle trajectory has a small error in the longitudinal and lateral directions of the vehicle. Figure 15(b) shows the yaw angular velocity of the vehicle, which finally stabilized at 1 rad/s. As shown in Figure 15(d), the angular speeds of the four driving wheels on the inner and outer sides of the tracked vehicle are finally stabilized at approximately the theoretical calculation value, but with large fluctuations. Figure 15(e) shows the output torques of the four drive motors on the inside and outside. The magnitudes of the torques on both sides are approximately the same with opposite directions. Finally, the torques are approximately stable at 10 Nm and close to the theoretical calculation value. The output power of the motor is shown in Figure 15(f). The powers of the four motors are close to and approximately stable at 0.5 kW.

2) 0.2B DYNAMIC STEERING

The steering wheel is turned to the left to $\theta_2 < \theta < \theta_3$ (steering to the left), the vehicle’s yaw angular velocity is $5\pi/12$ rad/s is given, the expected vehicle speed is $v_{des} = 0.26$ m/s. Make the tracked vehicle start from a standstill and perform a small radius steering of $R = 0.2B$. The simulation results are shown in Figure 16.

The trajectory of the vehicle is shown in Figure 16(a). It can be seen that the tracked vehicle enters the turning state from stop, and the turning radius is basically consistent with the theoretical turning radius $R = 0.2B$. Figure 16(b) shows the yaw angular velocity of the vehicle. After a certain acceleration time the yaw angular velocity stabilizes at about 1.3 rad/s, Which is consistent with a given desired yaw angular velocity $\omega_{des} = 5\pi / 12$ rad/s. The vehicle speed is shown in Figure 16(c), and the vehicle speed finally fluctuates around 0.25 m/s, which is consistent with a given expected vehicle speed $v_{des} = 0.26$ m/s.

The angular velocity of the inner and outer driving wheels is shown in Figure 16(d). It can be seen from Figure 16(e) that the inner and outer driving wheels are finally stable at about 11 Nm, which is basically consistent with the calculated value of theoretical analysis. As shown in Figure 16(f), when the steering is stable, the output power of the inner motor is 0.7 kW, and the output power of the outer motor is 0.9 kW.
3) $R = 2B$ STEERING CONDITIONS

The tracked vehicle starts from stop and turn with a desired steering radius $R = 2B$. The simulation results are shown in Figure17. The vehicle trajectory is shown in Figure 17(a). It can be seen that the track model car enters the turning state from stop. The turning radius of 2.5B is basically consistent with the theoretical turning radius $R = 2B$. From Figure 17(b), after a certain acceleration time, the yaw angular velocity stabilizes at about 1.3rad/s, which is consistent with a given desired yaw angular velocity $\omega_{des} = 5\pi / 12$rad/s. From Figure17(c), the vehicle speed finally stabilizes at the desired vehicle speed $v_{des} = 2.6m / s$. Figure 17(d) shows the angular speeds of the four driving wheels on the inside and outside of the tracked vehicle.

It can be seen from Figure 17(e) that the inner motor torque finally stabilizes at about 10 Nm, and the outer motor torque finally stabilizes at about 15 Nm. As shown in Figure and the output power of the outer motor is finally 1.3kw.

From the simulation results of the above three typical steering conditions, it can be seen that the output power of the outer motor is less than the peak torque of the motor, and no electromechanical steering coupler is required to work.

B. ELECTROMECHANICAL COUPLING STEERING

From the table4, when the steering radius is 8B for steering, the power required by the outer driving wheels is greater than the peak power of the motor, so the electromechanical coupling steering mode is required. A working condition with a steering radius of 8B was selected for simulation analysis of steering stability and trajectory tracking performance. The simulation results are shown in Figure18.

The vehicle trajectory is shown in Figure 18(a). It can be seen that the tracked model vehicle enters the turning state from stop. The steering radius after coupling is obviously smaller than that before coupling, so it has better path tracking performance. It can be seen from Figure 18(b) that after a certain acceleration time, the yaw angular velocity of before coupling of tracked vehicle finally stabilizes at about $\omega = 0.5$rad/s. The yaw angular velocity of the tracked vehicle after coupling eventually stabilizes at about $\omega = 1$rad/s, which is obviously greater than the yaw angular velocity of the tracked vehicle before coupling, but less...
than the expected yaw angular velocity $\omega_{des} = 5\pi / 12 \text{rad/s}$. It shows that the larger the turning radius and the faster the speed, the worse the tracking performance of yaw angular velocity. The comparison of the vehicle speed before and after coupling is shown in Figure 18(c). The vehicle speed before coupling is finally stable at 7m / s, and the vehicle speed after coupling is finally closer to 10m / s, which is little different from the theoretical calculation vehicle speed $v_{des} = 10.5 \text{m / s}$.

Only the outer rear driving wheel is power coupled by electromechanical coupling steering, so the angular speed, torque and power of the front and rear outer driving wheels are selected for comparison.

From Figure 18(d), it can be seen that due to the limitation of the peak power of the driving motor, the angular velocity of the outer driving wheels of before coupling finally approaches 95 rad/s, which cannot reach the expected value. After coupling, the angular speed of the outer driving wheels of the tracked vehicle finally close to 120 rad/s, which is slightly higher than the theoretical calculation wheel speed $\omega = 111.35 \text{rad / s}$. From Figure 18(e), it can be seen that the output torque of the front-outer rear wheel motor of before coupling is 22 Nm, while the output torque of the rear-outer wheel motor after coupling is 26 Nm. It can be seen from Figure 18(f) that the output power of the front and outer rear wheels before coupling is stable at 2kW due to the limitation of the peak power of the driving motor. After the coupling device is added, the output power of the outer rear wheel is increased to 3.1kW.

### C. TORQUE OPTIMIZED DISTRIBUTION CONTROL

In order to verify the influence of the torque optimal distribution control strategy on handling stability and trajectory tracking of the whole vehicle, the simulation is performed under the condition of $R = 2B$ and $R = 8B$ to verify the feasibility of the torque optimal distribution control strategy. The simulation results of vehicle trajectories, torque and power of outer rear driving wheel before and after optimization are shown in Figure 19(a-c) and figure 19(d-f), respectively.

As can be seen from the simulation results in Figures 19, the trajectory after optimization is slightly smaller than the trajectory before optimization, and the tracking error is small. After optimization, the outer and rear drive wheels have significantly lower torque and lower power than before...
optimization. It can be seen that the optimized torque distribution control strategy improves the steering stability and path tracking performance of tracked vehicles under small radius steering conditions, and has energy-saving effects.

VI. CONCLUSION

Aiming at the phenomenon that the power of the outer drive motor is insufficient when the four-motor distributed drive tracked vehicle is turning at high speed and large radius, a new steering coupling system is proposed in this paper and a direct yaw torque control strategy is proposed. The article establishes an optimization function with the driving force and yaw moment as the optimization goals. The simulation analysis is performed on the working conditions with a steering radius of 0B, 0.2B, and 2B, the electromechanical coupling simulation analysis is performed on a working condition with a steering radius of 8B, and the optimal simulation analysis is performed on the working conditions with a steering radius of 2B and 8B. Simulation results show that after power coupling, the output power of the outer driving wheels during steering is improved, the running trajectory of the tracked vehicle is effectively improved, and the running stability of the tracked vehicle is improved. Finally, it is concluded that the steering coupling system and optimized control strategy proposed in this paper can effectively solve the phenomenon of insufficient power of the outer drive motor when the tracked vehicle is turning at high speed and large radius. In the future, the following work can be done: (1) The factor of track slip can be taken into account, and a better control method is adopted to improve the stability of the control. (2) The energy optimization of the four-motor drive tracked vehicle can be used as the control target, and the energy consumption of the two-motor, three-motor, and four-motor under different driving conditions can be analyzed and compared to choose different working modes to achieve energy-saving purposes.

REFERENCES

[1] Z. L. Liao, X. J. Ma, and K. M. Zang, “Comparative analysis of electric drive schemes for tracked armored vehicles,” J. Ordnance J., no. 04, pp. 583–586, 2006.

[2] H. C. Li, Q. D. Yan, and W. Q. Song, “Research on electric transmission scheme for realizing mechanical cycle of tracked vehicle steering power,” J. Mech. Design, vol. 28, no. 3, pp. 60–64, 2011.

[3] C. Lin, S. Liang, J. Chen, and X. Gao, “A multi-objective optimal torque distribution strategy for four in-wheel-motor drive electric vehicles,” IEEE Access, vol. 7, pp. 64627–64640, 2019.

[4] X. P. Jia, J. Ma, and K. L. Yu, “Performance analysis of electric transmission tracked vehicle coupling transmission scheme,” J. Mech. Transmiss., vol. 39, no. 12, pp. 129–132, 2015.

[5] X. P. Jia, J. Ma, and K. L. Yu, “Research on the transmission system of electric transmission tracked vehicle,” J. Armored Force Eng. Inst., vol. 29, no. 2, pp. 35–39, 2015.

[6] J. Ma, X. P. Jia, and K. L. Yu, “Modeling and simulation of an electric drive tracked vehicle drive system based on a coupling mechanism,” J. Agricultural Equimp. Vehicle Engn., vol. 53, no. 8, pp. 25–28, 2015.

[7] J. T. Gai, S. D. Huang, and G. M. Zhou, “Design method of power coupling mechanism driven by two-sided motor,” J. China Mech. Engn., vol. 25, no. 13, pp. 1739–1743, 2014.

[8] L. Zhai, T. M. Sun, Q. N. Wang, and J. Wang, “Lateral stability control of dynamic steering for dual motor drive high speed tracked vehicle,” Int. J. Automot. Technol., vol. 17, no. 6, pp. 1079–1090, Dec. 2016.
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