



A Steering Control Strategy Based on Torque Fuzzy Compensation for Dual Electric Tracked Vehicle

Pei Li^a, Jun Yan^a, Qunzhang Tu^a, Ming Pan^a, Chengming Jiang^a

^aField Engineering College, Army Engineering University of PLA

Abstract. A steering control strategy based on bilateral torque fuzzy compensation for dual electric tracked vehicle is proposed in this paper. After the dynamic analysis of tracked vehicles, the mapping relationship between acceleration signal, braking signal, steering signal and bilateral motor torque is established. According to different driving states, the steering wheel real-time rotation angle and its change rate are interpreted as the motor torque compensation coefficients K_1 and K_2 by fuzzy algorithm to achieve quick response of driving intention. The steering control model of the electric tracked vehicle is built, and the HILS (hardware in loop simulation) platform is constructed with dSPACE. The HILS result shows that, by torque fuzzy compensation strategy, steering sensitivity and controllability could get better improvement compared with direct torque control strategy.

1. Introduction

As the future military vehicle technology direction, electric tracked vehicles have been included in the future armored vehicle development program in United States and UK [1-3]. Dual motor independent drive structure, which has the advantages of simple structure and high transmission efficiency, is the most widely used scheme in electric tracked vehicles design, such as: USA M113A3, Sweden B-13 and Germany "the weasel"[4-5]. But because there is no mechanical connection between the two motors, steering process mainly depends on differential velocity, the problems of slow dynamic response and poor steering controllability have always existed.

For the problem of steering stability, a variety of methods have been proposed by scholars, which could be divided into two categories: motor torque control and motor speed control [6-10]. As for motor speed control, due to the large change of the motor load, closed loop control is needed, and different algorithms could be used to adjust the motors in real time. The control algorithms are various and mainly depend on the mathematical model. For motor torque control, the operating signal of the driver correspond directly to the output torque of the motors. Ignoring the influence of the external environment, the vehicle speed and steering angular velocity after overcoming the resistance are completely decided by the driver. The direct torque control strategy is simple and direct, and has been applied in practice [11]. However, in this method,

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Corresponding Author: Qunzhang Tu, No.88, Houbiaoying Road, Nanjing 210007, China. Tel: +2580821351.

Email addresses: lipei1503@163.com (Pei Li), p50262@126.com (Jun Yan), tqz1hnj@126.com (Qunzhang Tu), panlvnn@163.com (Ming Pan), 13951815070@139.com (Chengming Jiang)

the steering wheel angle change rate have not been considered. When the vehicle is turning sharply or returning quickly, it cannot follow the driver’s intention well.

For the above problems, based on direct torque control strategy, the bilateral torque fuzzy compensation control strategy is proposed in this paper. Using the fuzzy algorithm, according to the driving condition, steering wheel rotation angle and its change rate are converted to the compensation coefficient of bilateral motor torque. The steering control model of the whole vehicle is built with Simulink, HILS platform is constructed with dSPACE, and the experimental study is carried out under typical working condition.

2. Steering dynamics analysis of dual electric tracked vehicle

The structure of dual electric tracked vehicle is shown in Figure 1, which mainly includes seven parts: general controller, engine-generator set, rectifier, battery pack, DC/DC converters, driving motors and motor controllers. For the convenience of analysis, it’s assumed that both the attachment coefficient and the resistance coefficient of ground are constant, the vertical load distribution of the inside and outside tracks is uniform, and the slip or skid of the tracks is ignored. The steering diagram is shown in Figure 2. With the inside driving force F_1 and outside driving force F_2 , the vehicle overcomes inside resistance f_1 , outside resistance f_2 and steering resistance torque M_μ , then turns at angular speed ω . L is the connection length of the track, B is the center distance of the tracks, V_1 and V_2 are the speed of the inside and outside tracks respectively, V_0 is the longitudinal average speed of vehicle. In the absence of air resistance, the equation formulas can be established as formula (1):

$$\begin{cases} F_1 + F_2 - f_1 - f_2 = m\dot{V}_0 \\ (F_2 - F_1)B/2 + (f_1 - f_2)B/2 - M_\mu = I_z\dot{\omega} \\ f_1 = f_2 = 0.5fmg \\ M_\mu = 0.25\mu mgL \\ V_1 = V_0 - 0.5\omega B \\ V_2 = V_0 + 0.5\omega B \end{cases} \quad (1)$$

Where \dot{V}_0 is the longitudinal average acceleration, $\dot{\omega}$ is the acceleration of steering angle, m is the vehicle mass, I_z is the vehicle moment of inertia around point O' , f is the track rolling friction coefficient, μ is the steering resistance coefficient. According to the empirical formula of A.O.H[12], μ is expressed as:

$$\mu = \mu_{\max}/(0.925 + 0.075R/B) \quad (2)$$

Where R is the turning radius, and $R = V_0/\omega$. Since the driving force is provided by the bilateral motors, F_1 and F_2 can be expressed as formula (3):

$$\begin{cases} F_1 = T_1i_0\eta_1\eta_2/r \\ F_2 = T_2i_0\eta_1\eta_2/r \end{cases} \quad (3)$$

Where i_0 is the side drive ratio, T_1 is the outside motor torque, T_2 is the inside motor torque, r is the wheel radius, η_1 is the transmission efficiency from motor to track, and η_2 is the track running efficiency. Using empirical formula, η_2 can be expressed as formula (4):

$$\eta_2 = 0.95 - 0.003V_0 \quad (4)$$

From the formula (1) ~ (4), it can be seen that when the tracked vehicle turns, the sum and difference of the bilateral motor torque determine the longitudinal velocity and yaw velocity respectively. The steering control strategy needs to meet the requirements: when the vehicle turns, the outside motor torque should always be greater than the inside one. When the vehicle returns, the outside motor torque should always be smaller than the inside one. At the same time, the greater the steering wheel angle, the bigger torque difference between the inside motor and outside motor.

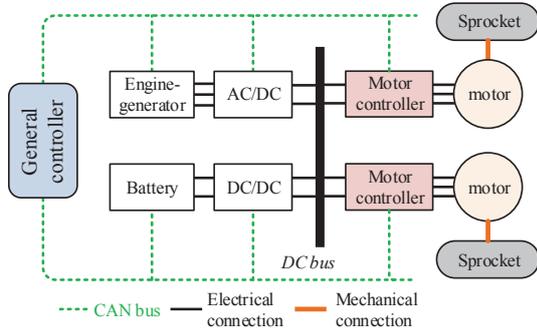


Figure 1: Structure of dual electric tracked vehicle

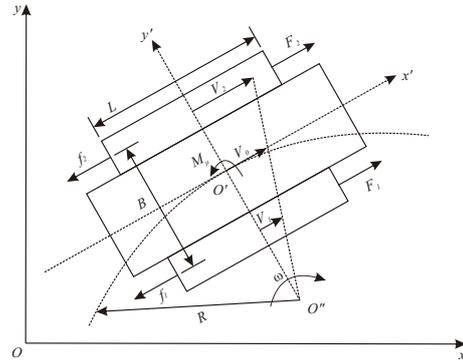


Figure 2: Schematic diagram of tracked vehicle steering

3. Torque fuzzy compensation control strategy

The steering process of tracked vehicles usually includes three stages: transient steering, stable steering and transient return. In the process of transient steering and transient return, steering wheel angle changes rapidly in a short time. For the former, the turning radius changes from infinity to a small value. For the latter, the turning radius changes from a small value to infinity. In the process of stable steering, the wheel angle, yaw rate and turning radius are unchanged. In order to improve vehicle handling comfort and steering sensitivity, the transient steering time and transient return time should be minimized so that the vehicle could enter to stable steering state or straight-line state as soon as possible.

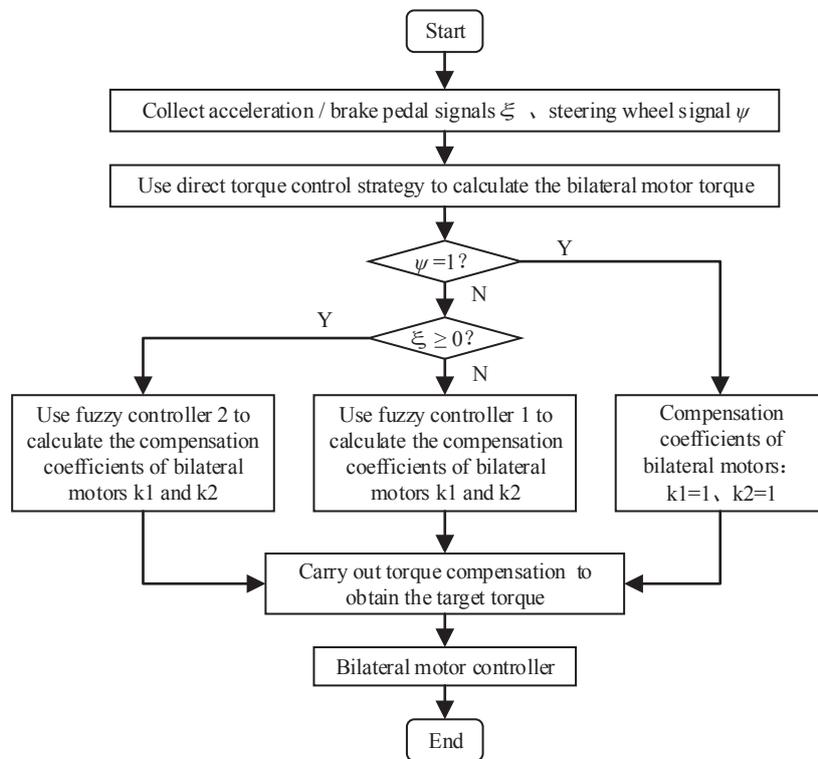


Figure 3: Flow chart of torque fuzzy compensation control strategy

Figure 3 is the flow chart of torque fuzzy compensation control strategy. First, the driver’s operation signal is collected, and the required torque of two motors is preliminary calculated by using direct torque control strategy. Then, the vehicle state is judged by ψ . If $\psi = 1$, indicating the vehicle is in straight-line state, at this time the compensation coefficients: $k_1 = k_2 = 1$. If $\psi \neq 1$, indicating the vehicle is turning, k_1 and k_2 should be determined by the fuzzy controller 1 or fuzzy controller 2 based on the value of ξ . Finally, k_1 and k_2 are used to compensate the torque value calculated by direct torque control strategy, so as to obtain the target torque.

3.1 Interpretation of driving intentions

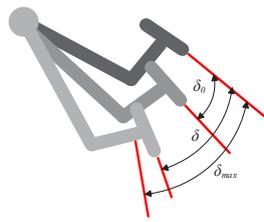


Figure 4: Acceleration or braking signal

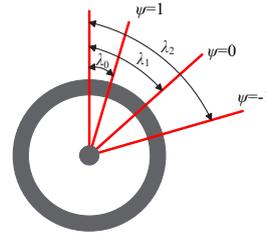


Figure 5: Steering wheel signal

The acceleration or braking signal ξ , which is shown in Figure 4, is used to adjust the bilateral motors simultaneously, and it's defined as formula (5):

$$\xi = k \frac{\delta - \delta_0}{\delta_{\max} - \delta_0} \quad (-1 \leq \xi \leq 1) \tag{5}$$

Where δ , δ_0 , δ_{\max} are real-time angular displacement, free travel angle displacement and maximum travel angle displacement of the acceleration pedal or brake pedal. k is the driving factor, when the vehicle is moving forward, $k = 1$, when the vehicle is braking, $k = -1$.

The steering wheel signal, which is shown in Figure 5, is used to adjust the torque of a single motor to make the sum and difference of bilateral motor torque have a wider range. The steering wheel signal determines the turning radius. For the left steering principle is same as the right steering principle, take the right steering operation as an example, the steering wheel signal is defined as formula (6):

$$\psi = \begin{cases} 1 & (0 \leq \lambda \leq \lambda_0) \\ (\lambda_1 - \lambda)/(\lambda_1 - \lambda_0) & (\lambda_0 < \lambda \leq \lambda_1) \\ (\lambda_1 - \lambda)/(\lambda_2 - \lambda_1) & (\lambda_1 < \lambda \leq \lambda_2) \end{cases} \tag{6}$$

Where λ , λ_0 , λ_2 are real-time rotation angle, free travel rotation angle, and maximum travel rotation angle of the steering wheel, λ_1 is the steering wheel rotation angle when a single motor driving. $\psi \in [-1, 1]$, with the increase of the rotation angle, it gradually turns from 1 to -1, realizing the adjustment of bilateral motors with different driving states.

3.2 The calculation of direct torque control strategy

The steering process requires close coordination between the accelerator pedal or the brake pedal and the steering wheel. The control process is complex which requires further consideration.

Firstly, the acceleration condition is considered, at this time $\xi \in [0, 1]$. Assume that the speed-torque characteristic curve of the two motors are the same, as shown in Figure 6, at the same speed, the maximum drive torque of the motor is equal to the maximum braking torque. n_1 and n_2 respectively indicate the speed of outside motor and inside motor, $T_{\max}(n)$ is the maximum motor torque at speed n . As seen in Figure 6, $n_1 > n_2$ and $T_{\max}(n_1) < T_{\max}(n_2)$, the torque difference between the two motors can be expressed by formula (7):

$$T'_1 - T'_2 = \xi \cdot T_{\max}(n_1) - \xi \cdot \psi \cdot T_{\max}(n_2) \tag{7}$$

Where T'_1 and T'_2 are the demand torque of the two motors. However, when $\psi \in [0, 1]$, there may be the situation $T'_1 - T'_2 \leq 0$. Therefore, it is necessary to revise the torque of the motors, T'_1 and T'_2 can be expressed as:

$$\begin{cases} T'_1 = \xi \cdot T_{\max}(n_1) \\ T'_2 = \xi \cdot \psi \cdot T_{\max}(n_1) \end{cases} \quad 0 < \psi < 1 \quad (8)$$

When $\psi \in [-1, 0]$, in order to form a bigger torque difference and make full use of the braking ability of the inside motor, the demand torque of the motors can be expressed as:

$$\begin{cases} T'_1 = \xi \cdot T_{\max}(n_1) \\ T'_2 = \xi \cdot \psi \cdot T_{\max}(n_2) \end{cases} \quad -1 \leq \psi \leq 0 \quad (9)$$

When the vehicle returns, the inside motor torque should be smaller than outside motor torque, and Figure 7 is a schematic diagram of the motor speed and torque during return process. The required torque of the motors can be expressed as:

$$\begin{cases} T'_1 = \xi \cdot \psi \cdot T_{\max}(n_1) \\ T'_2 = \xi \cdot T_{\max}(n_2) \end{cases} \quad -1 \leq \psi \leq 1 \quad (10)$$

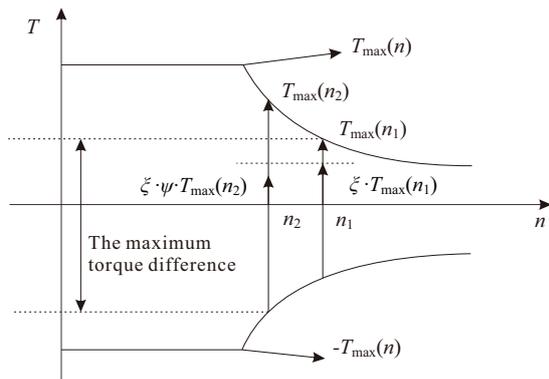


Figure 6: Motor speed and torque during steering process

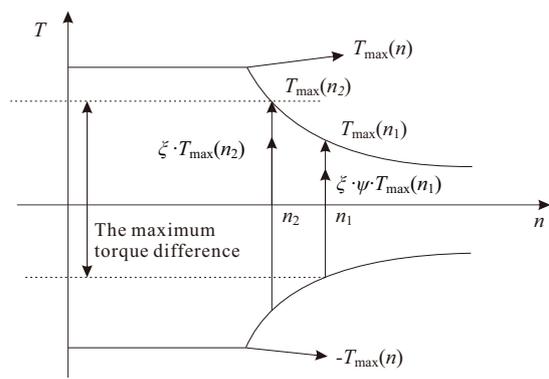


Figure 7: Motor speed and torque during return process

Similarly, for the braking condition, the demand torque expression can be derived by the same method. The vehicle steering conditions are divided into three categories: accelerated large radius steering, accelerated small radius steering and brake steering. The vehicle return conditions are also divided into three categories: accelerated return, braking large radius return and braking small radius return. Using the direct torque control strategy, T'_1 and T'_2 can be obtained, the specific methods of calculation are shown in Table 1 and Table 2.

The direct torque control strategy can be implemented by Matlab/Stateflow module, as shown in Figure 8. Using the finite-state machine theory, the state transition by different operating conditions can be accomplished by different restrictions and execution functions [13-15].

Steering condition	Demand torque
accelerated large radius steering $\xi \in [0, 1], \psi \in [0, 1]$	$T'_1 = T_{\max}(n_1)\xi$ $T'_2 = T_{\max}(n_1)\xi\psi$
accelerated small radius steering $\xi \in [0, 1], \psi \in [-1, 0]$	$T'_1 = T_{\max}(n_1)\xi$ $T'_2 = T_{\max}(n_2)\xi\psi$
brake steering $\xi \in [-1, 0], \psi \in [-1, 1]$	$T'_1 = T_{\max}(n_1)\xi\psi$ $T'_2 = T_{\max}(n_2)\xi$

Return condition	Demand torque
accelerated return $\xi \in [0, 1], \psi \in [-1, 1]$	$T'_1 = T_{\max}(n_1)\xi\psi$ $T'_2 = T_{\max}(n_2)\xi$
braking large radius return $\xi \in [-1, 0], \psi \in [0, 1]$	$T'_1 = T_{\max}(n_1)\xi$ $T'_2 = T_{\max}(n_1)\xi\psi$
braking small radius return $\xi \in [-1, 0], \psi \in [-1, 0]$	$T'_1 = T_{\max}(n_1)\xi$ $T'_2 = T_{\max}(n_2)\xi\psi$

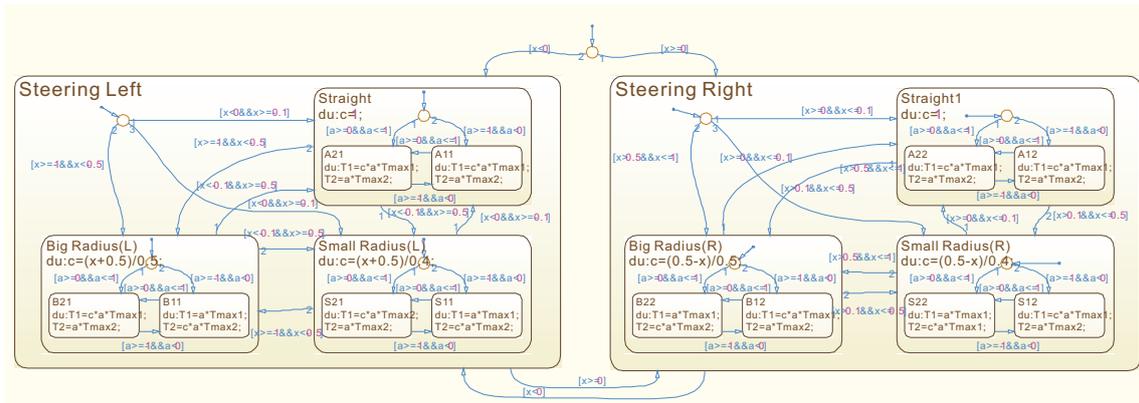


Figure 8: Direct torque control strategy based on Stateflow

3.3 Fuzzy compensation of the torque

By formula (1), it can be concluded that when the vehicle is turning, the transient steering time t can be represented by the formula (11):

$$t = \int_0^{\omega_0} \frac{I_z}{(F_2 - F_1 + f_1 - f_2)B/2 - M_\mu} d\omega \tag{11}$$

Where ω_0 is the angular velocity of stable steering. According to the formula (3), appropriately increasing the outside motor torque and reducing the inside motor torque could reduce the transient steering time, similarly, properly reducing the outside motor torque and increasing the inside motor torque could reduce the transient return time. In order to achieve the above goals, the fuzzy controller is used to compensate the torque of bilateral motors. Fuzzy controller has good robustness, and it's suitable for dealing with nonlinear and uncertain problems, it is widely used in various control systems [16-18]. The Mamdani fuzzy controller with double inputs and double outputs is adopted in this paper, the steering wheel real-time rotation angle λ and its change rate $d\lambda$ are selected as input variables. λ has four fuzzy subsets: S, MS, MB, B , and its domain is $[0,1]$. $d\lambda$ has five fuzzy subsets: S, MS, M, MB, B , and its domain is $[-1,1]$. The torque compensation coefficients k_1 and k_2 are selected as output variables. k_1 and k_2 also have five fuzzy subsets: S, MS, M, MB, B , and its domain is $[0.8,1.2]$. The relative membership function diagram is shown in Figure 9.

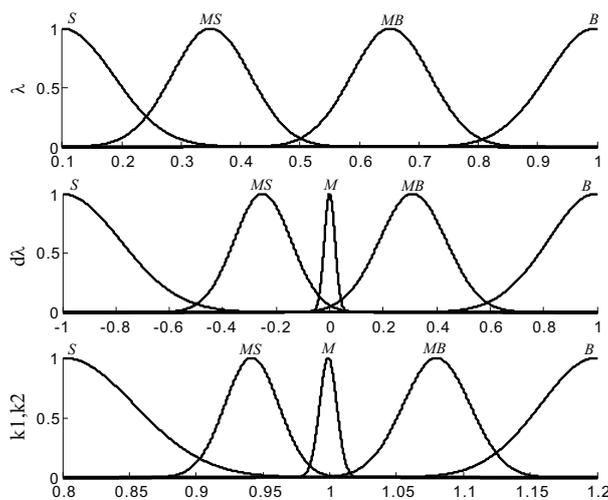


Figure 9: Membership function diagram of input and output variables

In the process of torque compensation, the following principles should be noted: When λ becomes smaller and $d\lambda$ becomes larger, indicating the driver has a quick steering intention, the outside motor torque should be increased and the inside motor torque should be reduced. When λ becomes larger and $d\lambda$ becomes smaller, indicating the driver has a quick return intention, the outside motor torque should be reduced and the inside motor torque should be increased. When $d\lambda$ is zero, the compensation should be weakened so that the torque of both motors can be restored to the steady-state value.

As the motor torque calculation methods in acceleration and braking condition are different, the torque compensation is achieved with two fuzzy controllers. By using the rule form of "IF λ is ... AND $d\lambda$ is ..., then $k1$ is ... and $k2$ is ...", 40 fuzzy rules are established as shown in Table 3 and Table 4.

Table 3: Fuzzy control rules with $\xi \geq 0$

$k1, k2$	$d\lambda$				
	S	MS	M	MB	B
S	MS,MB	MS,MB	M,M	MB,MS	B,S
MS	S,B	MS,MB	M,M	MB,MS	B,S
MB	S,S	MS,MS	M,M	MB,MB	B,B
B	S,S	MS,MS	M,M	MB,MB	MB,MB

Table 4: Fuzzy control rules with $\xi < 0$

$k1, k2$	$d\lambda$				
	S	MS	M	MB	B
S	MB,MS	MB,MS	M,M	MS,MB	S,B
MS	B,S	MB,MS	M,M	MS,MB	S,B
MB	S,S	MS,MS	M,M	MB,MB	B,B
B	S,S	MS,MS	M,M	MB,MB	MB,MB

The centroid method is adopted in the precise process of $k1$ and $k2$, the required torque of the two motors can be expressed as:

$$\begin{cases} T_{1req} = k1 \cdot T'_1 \\ T_{2req} = k2 \cdot T'_2 \end{cases} \quad (12)$$

4. Modeling and simulation

4.1 Steering control model of electric tracked vehicle

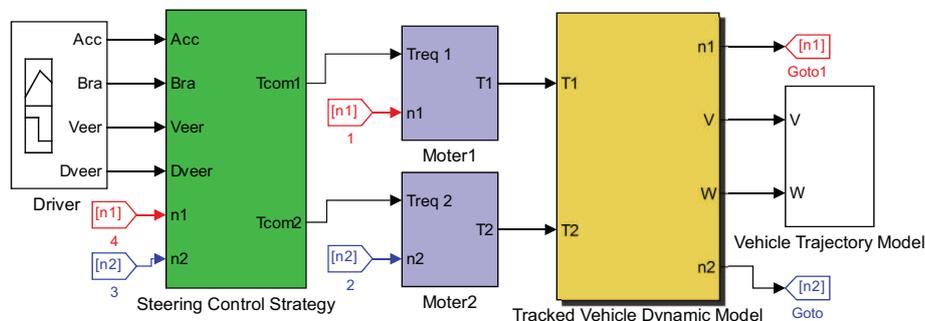


Figure 10: Models built in MATLAB /Simulink

The steering control model of electric tracked vehicle is based on MATLAB/Simulink, which is mainly made up of a driver model, a steering control strategy model, two motor models, a vehicle dynamic model and a vehicle trajectory model, as shown in Figure 10. The driver model is built with the block of signal builder, the steering control strategy model is built with Stateflow based on the strategy proposed in part 3, and the vehicle dynamic model is built based on the dynamics analysis in part 1. The following mainly introduces the motor model and vehicle trajectory model.

4.1.1 Motor model

Because the main point of this article is not the inner logic control of the motors, the motor model is established by experimental modeling method. The speed-torque relation of the motor is obtained by bench test, and the torque response is realized by look-up table module in Simulink [19]. At the same time, considering the response delay of the motor drive system, the output torque is added with a first-order lag link, and the final output torque T_m is expressed as:

$$T_m = \begin{cases} T_{req}/(\tau s + 1) & (T_{b\max}(n) \leq T_{req} \leq T_{d\max}(n)) \\ T_{d\max}(n)/(\tau s + 1) & (T_{req} > T_{d\max}(n)) \\ T_{b\max}(n)/(\tau s + 1) & (T_{req} < T_{b\max}(n)) \end{cases} \quad (13)$$

Where T_{req} is the demand torque obtained by the control strategy, τ is the response time constant, $T_{d\max}(n)$ and $T_{b\max}(n)$ respectively represent the maximum drive torque and maximum braking torque of the motor when the motor speed is n .

4.1.2 Vehicle trajectory model

In order to describe the dynamic process of vehicle steering more accurately, a vehicle trajectory model is established [20]. As shown in Figure 2, a fixed coordinate system Oxy is established on the ground, when the vehicle begins to turn, the center of mass is represented by O , and the transverse and longitudinal direction of the vehicle are respectively represented by the x axis and y axis. A dynamic coordinate system $O'x'y'$ is built on the vehicle body, when the vehicle is turning, the center of mass is represented by O' , and the transverse and longitudinal direction of the vehicle are respectively represented by the x' axis and y' axis. The velocity components of V_0 along the x' axis and y' axis are $V_{x'}$ and $V_{y'}$, they can be expressed as:

$$\begin{cases} V_{x'} = V_0 = \int_0^t (F_1 + F_2 - f_1 - f_2)/m dt \\ V_{y'} = 0 \end{cases} \quad (14)$$

Assume that V_x and V_y represent the speed components of the vehicle in the coordinate system Oxy , and θ is the angle that the vehicle have turned during the time t , so there have this relation:

$$\begin{cases} V_y = V_{x'} \cos \theta + V_{y'} \sin \theta \\ V_x = V_{x'} \sin \theta - V_{y'} \cos \theta \\ \theta = \int_0^t \omega dt \end{cases} \quad (15)$$

Assume that (x_0, y_0) and $(x(t), y(t))$ represent the starting position and the vehicle position at t moment in the fixed coordinate system Oxy , respectively. The vehicle trajectory equation can be expressed as:

$$\begin{cases} x(t) = \int_0^t V_x dt + x_0 \\ y(t) = \int_0^t V_y dt + y_0 \end{cases} \quad (16)$$

4.2 HILS Based on dSPACE

In order to evaluate the effectiveness of the torque fuzzy compensation control strategy, the HILS experiment based on driver and controller is carried out using dSPACE. With dSPACE, MATLAB/Simulink simulation models could be converted into C codes and run reliably [21-22]. The schematic diagram of HILS platform is shown in Figure 11. The driver controls the acceleration / brake pedal and the steering wheel to produce analog signals which are transmitted to the general controller. After reading the driver operation

signal through the A/D interface, and receiving the motor model state signal of dSPACE through CAN bus, the general controller calculates the target torque of the bilateral motors in real time through the steering control strategy, and outputs the calculation results to dSPACE. According to the control command, the motor model, the tracked vehicle dynamics model and the vehicle trajectory model are run in the dSPACE, and the relevant parameters are displayed on the monitor. The host computer is used to modify, compile and download the internal model of dSPACE. The hardware of HILS platform is shown in Figure 12.

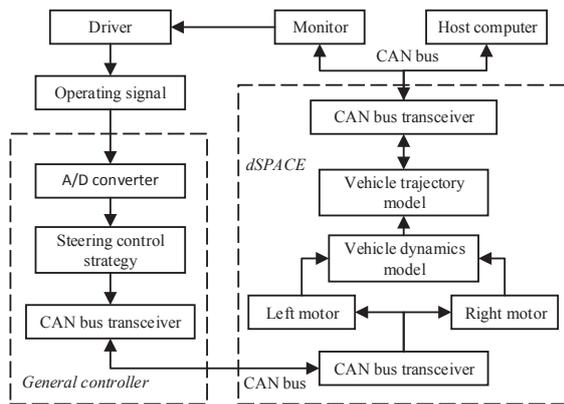


Figure 11: Schematic diagram of HILS platform

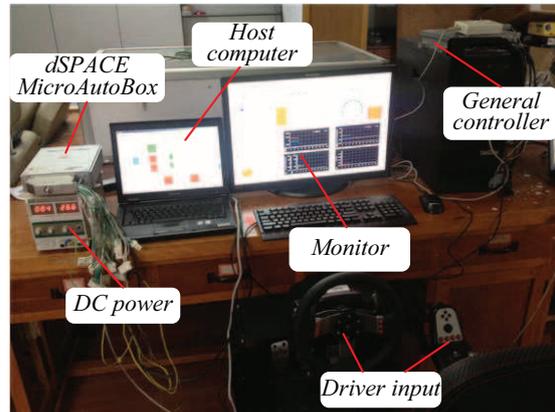


Figure 12: Hardware of HILS platform

4.3 Test and result analysis

The simulation test is based on a certain type of electric tracked vehicle [23], with parameters as: vehicle mass (m) is 18000kg, connection length of track (L) is 2.43m, center distance of the tracks (B) is 2.43m, wheel radius (r) is 0.4685m, rated power of each motor is 140KW, and rated torque of each motor is 1500N·m. In order to make the test result more applicable, the test is carried out under the typical condition of braking small radius steering.

The driver's operating signals are shown in Figure 13. The acceleration pedal signal rises from 0 to 0.4 in 0~1s, stays at 0.4 in 1~4s, and stays at 0 in 4~10s. The brake pedal signal rises from 0 to 0.3 in 4~5s, and stays at 0.3 in 5~10s. The steering wheel signal rises from 0 to 0.9 in 5~6s, stays at 0.9 in 6~7s, and declines from 0.9 to 0 in 7~8s. It's known that the vehicle has experienced a braking small radius steering in 5~8s, and in theory, the transient steering time, stable steering time and transient return time are both 1s.

The simulation is carried out under direct torque control strategy and torque fuzzy compensation control strategy respectively. The results are shown in Figure 14~16.

Figure 14 shows the bilateral motor torque signal, it can be seen that the torque difference control could be achieved under either of the two strategies. When the steering wheel angle increases, with fuzzy compensation control strategy, the outside motor torque is larger and the inside motor torque is smaller. When the steering wheel angle decreases, with fuzzy compensation control strategy, the outside motor torque is smaller and the inside motor torque is larger. When the steering wheel angle is constant, the effects of the two control strategies are the same.

Figure 15 shows the signals of the turning radius. As can be seen in Figure 13, the vehicle receives the steering signal at 5s and receives the return signal at 7s. Under the direct torque control strategy, the vehicle enters the stable steering state at 7.41s, and enters the straight line state at 9.02s. The steady-state turning radius is 22m. The transient steering time is 2.41s, and the transient return time is 2.02s.

Under the torque fuzzy compensation control strategy, the vehicle enters the stable steering state at 6.27s, and enters the straight line state at 8.25s. The steady-state turning radius is also 22m. The transient steering time is 1.27s, which shortens 1.14s than the direct torque control strategy. The transient return time is 1.25s, which shortens 0.77s than the direct torque control strategy.

Figure 16 shows that the vehicle can achieve smooth steering with either of these two strategies, and the steering trajectory is stable and controllable. But it can be seen that the steering position under torque fuzzy

compensation control strategy is (0, 19.5), while the steering position under direct torque control strategy is (0, 20). So when the vehicle receives steering signal, it can move into the steering state faster under the torque fuzzy compensation control strategy, which could better follow the driver's intention.

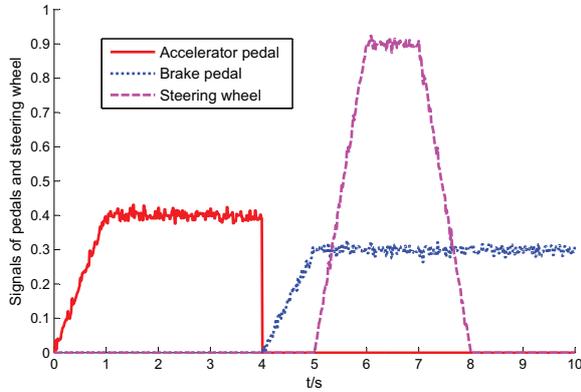


Figure 13: Drivers operating signals

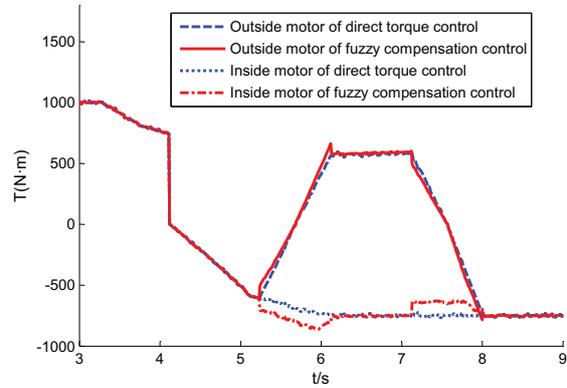


Figure 14: Signals of bilateral motor torque

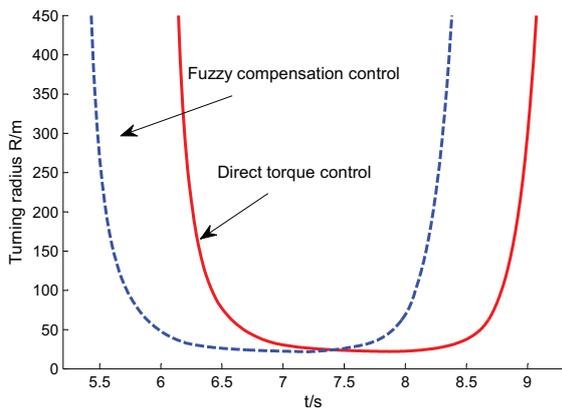


Figure 15: Signals of turning radius

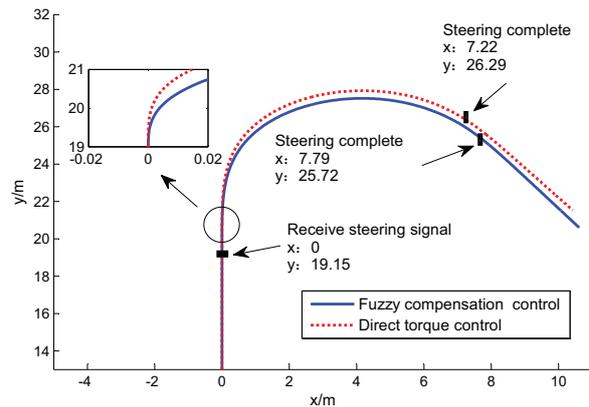


Figure 16: Vehicle trajectory

5. Conclusion

In this paper, a steering control strategy based on bilateral torque fuzzy compensation for dual electric tracked vehicle was developed. In the proposed control strategy, the driver's operating signal was interpreted as the torque control signal for the two motors. In addition, according to the acceleration or braking state, using the steering wheel angle signal and its change rate signal, two double-input and double-output fuzzy controllers were designed to improve vehicle handling comfort and steering sensitivity. HILS platform based on Simulink and dSPACE was conducted, and the typical working condition of braking small radius steering was simulated. The results shown that by the torque fuzzy compensation control strategy, the vehicle could realize reliable steering, and could be better to follow the driver's driving intention. Compared with the direct torque control, the transient steering time and the transient return time were reduced by 1.14s and 0.77s respectively.

References

- [1] Khalil, Ghassan, Challenges of hybrid electric vehicles for military applications, *Vehicle Power and Propulsion Conference*, IEEE(2009) 1–3.
- [2] Kramer, Denise M, and G. G. Parker, Current state of military hybrid vehicle development, *International Journal of Electric & Hybrid Vehicles* 3.4 (2011) 369–387.
- [3] Kelly, Terrence K, et al, *The U.S. Combat and Tactical Wheeled Vehicle Fleets: Issues and Suggestions for Congress*, CA: RAND Corporation, Santa Monica, 2011.
- [4] Sun FC, Zhang CN, *Technologies for the Hybrid Electric Drive System of Armored Vehicle*, (2nd edition), National Defense Industry Press, Beijing, 2016.
- [5] Shafer G, *Electric Drive M113 Vehicle Refurbishment Project: Sacramento Electric Transportation Consortium RA93-23 Program*, ADA322403, FMC Corporation, Santa Clara, 1997.
- [6] Zou Y, Sun FC, and Zhang CN, Dual-motor driving electric tracked vehicle speed-regulating control strategy, *Transactions of Beijing Institute of Technology*, 27.4(2007):303–307.
- [7] Zhai Li, Neural Networks PID Control of Steering Torque for Electronic Differential Tracked Vehicle, *Transactions of the Chinese Society for Agricultural Machinery* 40.2(2009):1–5.
- [8] Chen ZY, Zhao GY, Zhai L et al, Control Strategy for Electric Tracked Vehicle Pivot Steering Based on Yaw Velocity Negative Feedback, *China Mechanical Engineering*, 24.22(2013): 3062–3066.
- [9] Sun FC, Chen SY, GUO F, Research on Driving Performances of Electric Transmission Tracked Vehicles Based on Torque Control Strategy, *Acta Armamentarii*, 28.2(2007): 129–133.
- [10] Chen Z Y, Zhang C N, Notice of Retraction Fuzzy Control Strategy for the Dual-motor Electric Tracked Vehicle Steering Rapidity, *Power and Energy Engineering Conference*, IEEE(2010):1–6.
- [11] Li B, Zhang CN, Li JQ, Torque Control Strategies Based on RecurDyn and Simulink for Electric Drive Tracked Vehicle, *Transactions of the Chinese Society for Agricultural Machinery*, 40.7(2009):1–5.
- [12] Yan QD, Zhang LD, Zhao SQ, *The construction and design of the tank*, Beijing Institute of Technology press, Beijing, 2006.
- [13] Zhan N, Wang S, Zhao H, *Stateflow and Its Combination with Simulink*, Springer International Publishing, 2017.
- [14] Chetty, R. M. Kuppan, et al, State Based Modeling and Control of a Multi Robot Systems Using Simulink/Stateflow, *Journal of Applied Sciences*, 12.24(2012):2494–2502.
- [15] Santhosh T K, Natarajan K, Govindaraju C, Synthesis and Implementation of a Multi-Port DC/DC Converter for Hybrid Electric Vehicles, *Journal of Power Electronics*, 15.5(2015):1178–1189.
- [16] Khooban M H, Vafamand N, Dragicevic T, et al, Model Predictive Control based on T-S Fuzzy model For Electrical Vehicles Delayed Model, *Iet Electric Power Applications*, 11.5(2017):918–934.
- [17] Schouten N J, Salman M A, Kheir N A, Fuzzy logic control for parallel hybrid vehicles, *IEEE Transactions on Control Systems Technology*, 10.3(2002):460–468.
- [18] Rebai A, Guesmi K, Hemici B, Adaptive fuzzy synergetic control for nonlinear hysteretic systems, *Nonlinear Dynamics*, 86.3(2016):1–10.
- [19] Tu QZ, Zhang XC, Pan M, et al, Design and Optimization of the Power Management Strategy of an Electric Drive Tracked Vehicle, *Mathematical Problems in Engineering*, 9.29(2016):1–13.
- [20] TANG JW, LIU WP, LIU DG, et al, An Analysis of Steering Track for Tracked Vehicle at Inaccurate Steering, *Acta Armamentarii*, 27.5(2006): 779–783.
- [21] Meah K, Hietpas S, Ula S, Rapid Control Prototyping of a Permanent Magnet DC Motor Drive System using dSPACE and Mathworks Simulink, *IEEE Applied Power Electronics Conference and Exposition*, IEEE(2007):856–861.
- [22] Rubaai, Ahmed, and P. Young, Hardware/Software Implementation of Fuzzy-Neural-Network Self-Learning Control Methods for Brushless DC Motor Drives, *IEEE Transactions on Industry Applications*, 52.1(2016):414–424.
- [23] Pan M, Yan J, Tu QZ, et al, Fuzzy control and wavelet transform-based energy management strategy design of a hybrid tracked bulldozer, *Journal of Intelligent & Fuzzy Systems*, 29.6 (2015):2565–2574.