

## Original Paper

# Design and Test of Auxiliary Steering System of High Clearance Sprayer Chassis Based on Self-steering Structure

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### Abstract

*A hydraulic assisted steering method was proposed to solve the problem of steering instability caused by insufficient reverse resistance moment of wheel hub motor in four-wheel independently electrically driven high clearance sprayer. Firstly, the principle of steering chassis structure of four-wheel independently electrically driven high clearance sprayer is briefly introduced. On this basis, hydraulic auxiliary steering system is designed. Then a simplified 2-DOF vehicle steering model is established to analyze the Angle control of the auxiliary steering system. Finally, simulation and road surface test are carried out respectively to verify the performance of steering and auxiliary steering coordination control. Under the working conditions of the steering system alone and the steering system and the auxiliary steering system together, the four-wheel electric drive sprayer carried out the obstacle crossing test and the downslope test with the slope of 15° respectively at the speed of 1 m/s on the smooth road surface. The test results show that in the downhill test, the maximum tracking deviation of the steering system alone is 6.1°, the maximum tracking deviation of the steering and auxiliary steering coordination is 0.9°, the maximum tracking deviation of the steering system alone is 12.0°, and the maximum tracking deviation of the steering and auxiliary steering coordination is 1.2° in the obstacle breaking test. The test results verify the feasibility and stability of the hydraulic steering system proposed in this paper. The system has good test performance and can meet the practical requirements.*

### Keywords

*high clearance sprayer, self-steering structure, four-wheel steering, electric chassis, hydraulic auxiliary steering*

## 1. Introduction

As a typical field plant protection machine (Zhuang et al., 2018), the high gap sprayer's excellent steering performance is the key to its stable operation in complex field environment. At present, most of the highland gap sprays adopt mechanical transmission structure. In the harsh environment such as soft, muddy and paddy fields, the mechanical transmission structure is easy to cause the failure, slip and subsidence of the driving wheel, and its mobility and stability are tested greatly.

In recent years, many scholars at home and abroad have carried out researches on the automatic driving (Liu Z. P. et al., 2018; Liu, J. Y. et al., 2016; Chen et al., 2019; Zhang et al., 2020), steering system (Derrick & Beviy, 2009; Hanifah et al., 2016; Xu, H. S. et al., 2013; Xu, C. et al., 2016; Luo, 2018; Li, Qiao, & Zou, 2020; Luo, Bao, & Lin, 2013; Zhang et al., 2015; Lu et al., 2016; Mao et al., 2012; Li et al., 2019; Chen et al., 2019; Hu & Zhang, 2020) and driving system (Ariff, Zamzuri, Saiful, & Idris, 2014; Guo et al., 2017; Ding et al., 2015; Zhang et al., 2015) of tractors and sprayers, and achieved a series of research results. According to the analysis of the existing research results, the steering mode of the highland gap sprayer currently mainly adopts the principle of side turning (Luo, Bao, & Lin, 2013; Zhang et al., 2015; Lu et al., 2016; Mao et al., 2012; Li et al., 2019). When the sprayer in the process of paddy field work half or whole wheels in deep mire, wheel steering is opened the side of all the mud, the steering torque as the steering Angle increases, until meet the steering mechanism can provide the torque limit, the wheel will be stuck in the mud or damage to the chassis, chassis torque limit under can't turning.

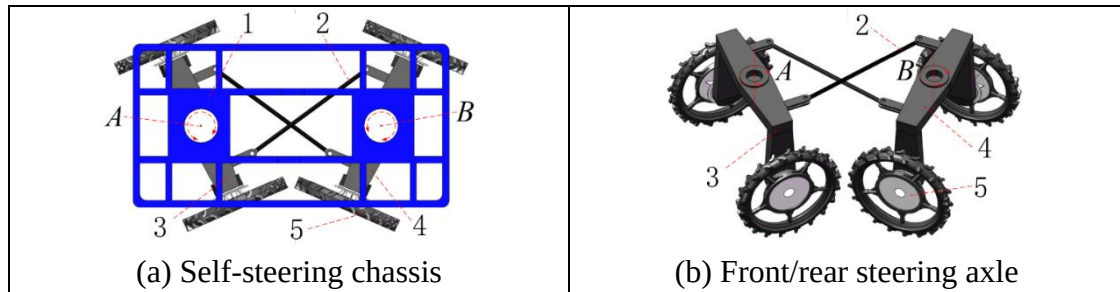
This research group designed a chassis sprayer with four-wheel independent electric drive and self-steering structure (Shen, Zhang, Liu, & Cui, 2019; Shen et al., 2020), which realizes self-steering by coordinating and controlling four-wheel differential to drive synchronous rotation of front and rear axle arms, and has better working performance in complex paddy field environment. However, in the process of downhill or obstacle crossing of the sprayer, the reverse torque of the wheel is needed to maintain the attitude of the vehicle body. However, the reverse torque of the existing driver is far less than the required torque, so stable and reliable steering control cannot be achieved. In addition, considering the cost, the sprayer does not have a suspension system. When driving on rough roads, the steering wheel is suspended, which may lead to instability of front/rear axle steering. When a single hub motor fails, the system steering performance is unstable. When the hub motor fails, the system cannot realize automatic steering.

In this paper, a four-wheel independent electrically driven high-gap sprayer is taken as the research object. By analyzing the existing self-steering chassis structure, an auxiliary steering system based on hydraulic control is proposed. The system uses the motor to drive the steering valve to control the hydraulic cylinder to ensure the response performance of the auxiliary steering system. Finally, the effectiveness of the proposed method is verified by simulation test and field test.

## 2. Structure and Principle of Four Wheel Independent Electric Drive High Gap Sprayer System

### 2.1 Self-steering Structure

Figure 1 shows the self-steering structure of the highland gap sprayer. The device is mainly composed of four-wheel drive system, front and rear steering axle, constraint connecting rod, etc. The main structural parameters of the sprayer are shown in Table 1.



**Figure 1. Structure Sketch of High Clearance Sprayer**

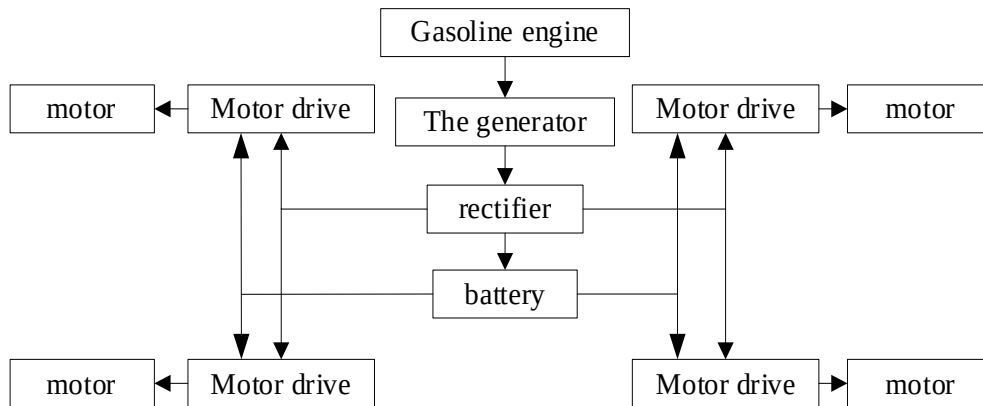
*Note.* 1. Chassis frame 2. Constraint connecting rod 3. Front steering axle 4. Rear steering axle 5. Hub motor A. Front steering center B. Rear steering center

**Table 1. Structural Parameters of High Clearance Sprayer**

| Parameter                      | values             |
|--------------------------------|--------------------|
| Machine type                   | wheel type         |
| Drive mode                     | Four-wheel drive   |
| The quality of the machine /kg | 1380               |
| Length/width/height/(mm*mm*mm) | 3200*1760*2130     |
| Wheelbase /(mm)                | 1580               |
| Wheel track /(mm)              | 1510               |
| Minimum ground clearance /(mm) | 1100               |
| Driving speed /(km/h)          | 0.8~10             |
| Maximum power /kw/(r/min)      | 28.3 KW/3600 r/min |

### 2.2 Principle of Power System

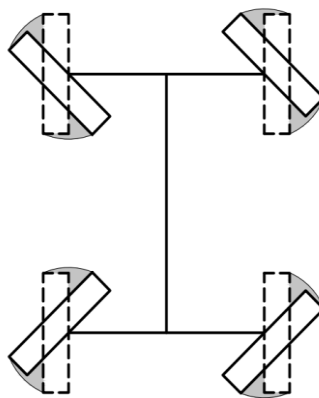
As shown in Figure 1, the high-gap sprayer is independently driven by four wheels, and its power configuration is oil-electric hybrid. Its power system is shown in Figure 2. When the gasoline engine is working, the gasoline engine drives the generator to generate electricity, which is converted into direct current through rectification and filtering. One part directly provides power for the motor drive, and the other part is energy storage for the battery. Highland gap sprayer can also realize pure electric operation under the condition of battery transmission.



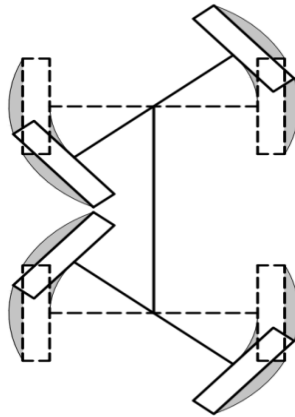
**Figure 2. Dynamic System Diagram of High Clearance Sprayer**

### 2.3 Steering Principle

Different from the traditional lateral steering mode of high clearance (Figure 3), the high clearance sprayer studied in this paper adopts a self-steering chassis, which drives the whole bridge arm to rotate through four-wheel differential to realize self-steering. The steering principle is shown in Figure 4. When the highland gap sprayers are stuck in the mud, the resistance is concentrated in the tangential direction of the wheels, which is collinear with the torque direction of the driving motor. Moreover, the steering does not need to push a large amount of mud, and the torque required is far less than that of the traditional side steering method. Therefore, the highland gap sprayers studied in this paper have higher driving efficiency in paddy fields.



**Figure 3. Steering Principle of Traditional High Clearance Sprayer**



**Figure 4. Steering Principle of Four Wheel Independent Electric Drive High Clearance Sprayer**

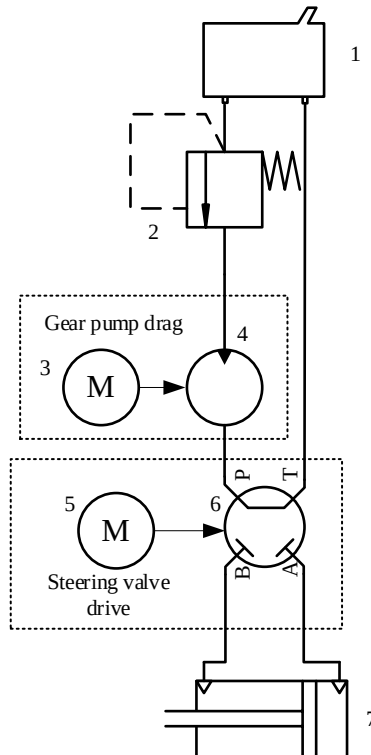
In Figure 3 and Figure 4, the dotted line frame represents the position of the wheel before steering, the solid line frame represents the position of the wheel after steering, and the gray area represents the wheel movement area during steering.

### 3. Design of Hydraulic Auxiliary Steering System

#### 3.1 Design Scheme

Highland gap sprayer in the obstacle-surmounting course of downhill or reverse torque is required to keep the body posture, and reverse provided by existing drive with brake torque is less than required torque system, besides high gap sprayer without suspension system, when driving on rough road surface is easy to appear to unstable phenomenon caused by the steering wheels. Therefore, it is necessary to design a set of auxiliary steering system to improve the running stability and reliability of the highland gap sprayer.

In order to realize the auxiliary steering function and improve the operating performance of the highland gap sprayer, this paper designed a set of auxiliary steering system based on electrically controlled hydraulic pressure on the existing self-steering system. The principle is shown in Figure 5. In the figure, P is the pressure oil input port of the system, T is the oil return port of the system, A is the left chamber of the steering valve 6, and B is the right chamber of the steering valve 6. The function of tank 1 is to load hydraulic oil and provide hydraulic oil for the whole hydraulic system. The effect of relief valve 2 is to protect the oil circuit; The function of gear pump 4 is to control the oil pressure of the hydraulic system, to ensure the stability of the oil pressure of the system; The role of DC motor 3 is to drive the gear pump 4, to ensure that the speed of the gear pump 4 is controllable; The function of the steering valve 6 is to control the direction, velocity and flow of the oil circuit; The function of DC motor 5 is to drive the steering valve 6, and control the expansion, expansion speed and expansion distance of the steering cylinder 7 by adjusting the steering valve 6's steering, speed and Angle.

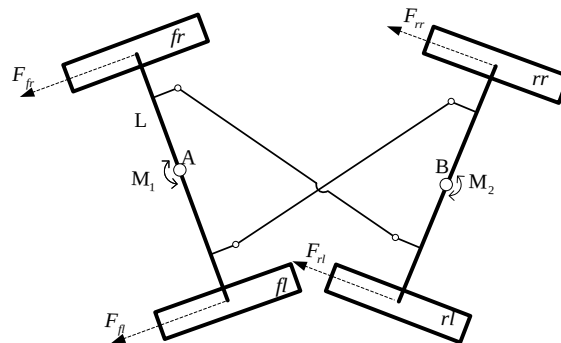


**Figure 5. Principle Diagram of Auxiliary Steering Hydraulic System**

Note. 1. Oil tank 2. Relief valve 3. DC motor 4. Gear pump 5. Steering valve 7. Steering cylinder

### 3.2 Assist the Selection of Important Components of the Steering System

Due to the limitation of conditions, the steering resistance moment of the sprayer can not be measured. Therefore, the steering resistance moment of the sprayer can be estimated by simple calculation. As shown in Figure 6, A and B are respectively the steering centers of the front and rear steering axles, and L is the distance between the steering center and the wheel center. The front and rear steering axles are connected by connecting rods to ensure equal front and rear steering angles. Assuming that two wheels of the sprayer can still turn when there is a failure, two of them are in rolling state and the other two are in sliding state.



**Figure 6. Schematic Diagram of Steering Resistance Moment Calculation**

The steering resistance moment of the wheels in the rolling state is:

$$M_r = \mu_1 \frac{mg}{4} L \quad (1)$$

The steering resistance moment of the wheels in the sliding state is:

$$M_s = \mu_2 \frac{mg}{4} L \quad (2)$$

Where:  $M_r$  ——steering resistance moment of wheels in rolling state, ( N · mm );

$M_s$  ——Steering resistance moment of wheels in sliding state, ( N · mm );

$\mu_1$  ——the rolling friction coefficient, which is 0.035;

$\mu_2$  ——the sliding friction coefficient, which is 0.8;

$L$  ——Distance between steering center and wheel center, take 755 mm;

$m$  ——The weight of the sprayer is 1380 kg;

$g$  ——Gravity acceleration, take 9.8 N/kg;

According to Equations (1) and (2),  $M_r = 89.3 \text{ N} \cdot \text{m}$ ,  $M_s = 2042.1 \text{ N} \cdot \text{m}$ , the turning resistance moment of the whole vehicle is  $M_{f1} = 2M_r + 2M_s = 4262.8 \text{ N} \cdot \text{m}$ .

Because the sprayer mainly works in paddy field, the resistance moment of the soil to the tire needs to be considered. For convenience of calculation, the approximation is  $M_{f2} = 137.2 \text{ N} \cdot \text{m}$ . Finally, the total steering resistance moment of the sprayer is calculated:

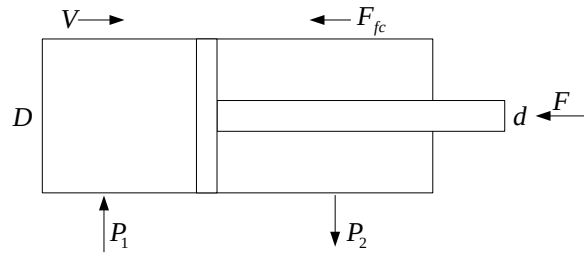
$$M_f = M_{f1} + M_{f2} = 4400 \text{ N} \cdot \text{m} \quad (3)$$

The working pressure of the hydraulic cylinder is shown in Table 2. According to the selection principle that the system pressure is lower than 2/3 of the rated pressure of the hydraulic pump, the working pressure of the hydraulic auxiliary steering system of the sprayer is selected as  $P_1 = 10 \text{ MPa}$  in this paper.

**Table 2. Common Working Pressure of Hydraulic Equipment**

| Type of equipment                                               |                       | Working pressure P/MPa |
|-----------------------------------------------------------------|-----------------------|------------------------|
| Machine                                                         | grinder               | 0.8~2.0                |
|                                                                 | Modular machine tools | 3~5                    |
|                                                                 | Longer planer         | 2~8                    |
|                                                                 | Broaching machine     | 8~10                   |
| Agricultural machinery or medium construction machinery         |                       | 10~16                  |
| Hydraulic press, lifting machinery, lifting transport machinery |                       | 20~32                  |

According to the selected working pressure and maximum total load, the inner diameter of the hydraulic cylinder  $D$  and the diameter of the piston rod  $d$  can be determined. The force analysis of the hydraulic cylinder is shown in Figure 7.



**Figure 7. Stress Analysis Diagram of Hydraulic Cylinder**

When the piston rod is pressed

$$F = p_1 A_1 - p_2 A_2 \quad (4)$$

When the piston rod is pulled

$$F = p_1 A_2 - p_2 A_1 \quad (5)$$

Where:  $F$ —the maximum external load received by the hydraulic cylinder;

$A_1 = \frac{\pi}{4} D^2$ —Effective area of rod-less cavity piston,  $\text{m}^2$ ;

$A_2 = \frac{\pi}{4} (D^2 - d^2)$ —Effective area of piston with rod cavity,  $\text{m}^2$ ;

$P_1$ —working pressure of hydraulic cylinder, Pa;

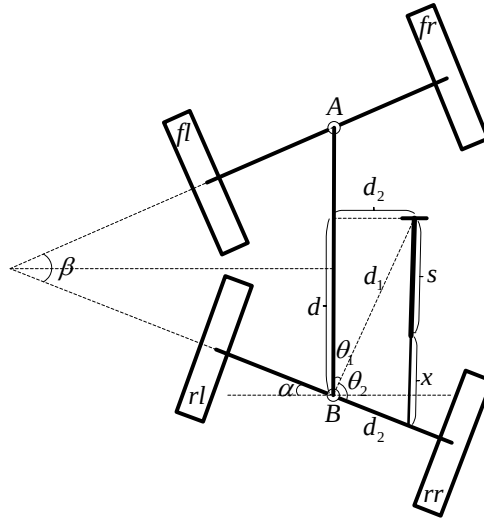
$P_2$ —Back pressure value, Pa;

$D$ —piston diameter, m;

$d$ —piston rod diameter, m.

$P_2$  is the back pressure value of the hydraulic cylinder. In general, take  $P_2 = 0.8$ , take  $d / D = 0.71$ , Substitute the data into the formula, and refer to the mechanical design manual, take the inner diameter of the hydraulic cylinder  $D = 63$  mm, the diameter of the piston rod  $d = 35$  mm.

In order to obtain the mapping relationship between the steering Angle of the axle arm and the auxiliary steering hydraulic cylinder during walking, as well as the relationship between the angular velocity of the axle arm and the telescopic linear velocity of the hydraulic cylinder, a geometric model of the auxiliary steering system was established (as shown in Figure 8). In the figure,  $A$  and  $B$  are respectively the steering centers of the front and rear axle arms,  $C$  are the fixed ends of the hydraulic cylinder, and  $D$  are the moving ends of the hydraulic cylinder. The front and rear axle arms are constrained by connecting rods (not shown in the figure) to ensure that the absolute values of the rotation angles of the front and rear axle arms are basically equal, and the rotation Angle of the axle arms is the rotation Angle of the steering wheel. As the vehicle turns to the left in a straight direction, the piston rod extends outward, and the distance between  $C$  and  $D$  increases.



**Figure 8. Kinematics Model Sketch of Hydraulic Auxiliary Steering**

Figure 8 shows  $\alpha$  is the Angle of the rear steering mechanism ( $^{\circ}$ );  $s$  is the length of cylinder block, mm;  $x$  is cylinder stroke, mm;  $d_1$  is the distance between the fixed end of the cylinder and the center of the rotary table of the rear axle arm, mm;  $d_2$  is the distance between the moving end of the cylinder and the center of the rotary table of the rear axle arm, mm;  $d$  is the vertical distance between the fixed end of the cylinder and the center of the rotary table of the rear axle arm; According to the relation of trigonometric functions, we can get:

$$d_2^2 = d^2 + d_1^2 - 2dd_1 \cos \theta_1 \quad (6)$$

$$\cos \theta_1 = \frac{d^2 + d_1^2 - d_2^2}{2dd_1} \quad (7)$$

For:

$$\theta_1 = \frac{180}{\pi} \arccos \frac{d^2 + d_1^2 - d_2^2}{2dd_1} \quad (8)$$

In the same way:

$$\theta_2 = \frac{180}{\pi} \arccos \frac{d_1^2 + d_2^2 - (s+x)^2}{2d_1d_2} \quad (9)$$

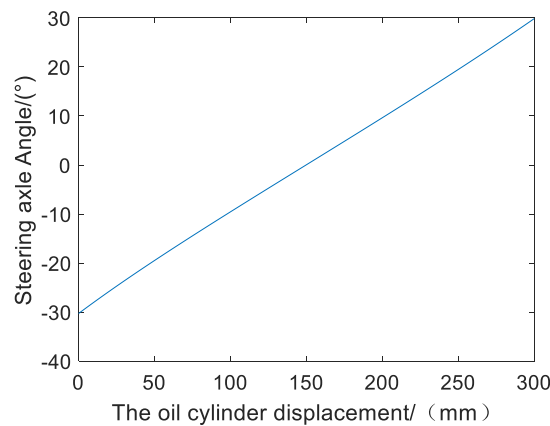
Related by the triangle:

$$\alpha = \theta_1 + \theta_2 - 90^\circ \quad (10)$$

Substituting Equations (8) and (9) into Equation (10), the mapping relationship between wheel Angle and hydraulic cylinder stroke can be obtained:

$$\alpha = \frac{180}{\pi} \arccos \frac{d^2 + d_1^2 - d_2^2}{2dd_1} + \frac{180}{\pi} \arccos \frac{d_1^2 + d_2^2 - (s+x)^2}{2d_1d_2} - 90^\circ \quad (11)$$

According to the test analysis, the steering Angle range of 3WPZ-500 spray machine is  $-30^{\circ} \sim +30^{\circ}$  when steering. According to the above calculation and reference to the steering hydraulic cylinder of other field walking machines, the 63\*35\*300 model of HSG63 two-way hydraulic cylinder is finally selected. The rated pressure of the hydraulic cylinder is 16 MPa, the highest pressure is 19 MPa, the rated thrust is 49850 N, and the rated tension is: 31681 N, mechanical efficiency  $\eta_M \geq 92\%$ , volume efficiency  $\eta_V \geq 98\%$ , maximum speed 0.3m /s, stroke 300 mm, cylinder diameter 63 mm, outside diameter 73 mm, rod diameter 35 mm, pin hole 30 mm, installation distance 570 mm, total closure length 630 mm, total extension length 930 mm. The installation distance of the hydraulic cylinder is calculated to be  $d=720$  mm,  $d_1=780$  mm,  $d_2=300$  mm. The cylinder body is measured to be  $s=570$  mm. The mapping relationship between the stroke of the hydraulic cylinder  $x$  and the steering Angle  $\alpha$  of the left rear wheel  $rl$  is obtained, as shown in Figure 9.



**Figure 9. Relationship between Wheel Angel and Cylinder Displacement**

It can be seen from Figure 9 that there is an approximate linear relationship between cylinder stroke and rotation Angle, and the proportionality coefficient  $K$  obtained through data fitting is 3.49 rad /m. In order to verify the accuracy of the proportionality coefficient  $K$ , the data of cylinder stroke and bridge arm rotation were collected, and the proportionality coefficient  $K$  was 3.42 rad /m with a relative error of 0.2% after fitting the test data. Therefore, the mapping relationship between hydraulic cylinder stroke and rear axle steering Angle can be approximated as:

$$\alpha = Kx + b \quad (12)$$

Take the derivative of the wheel Angle  $\alpha$

$$\dot{\alpha} = \frac{d\alpha}{dt} = \frac{df(x)}{dx} \frac{dx}{dt} \quad (13)$$

The mapping relationship between wheel steering angular velocity  $\dot{\alpha}$  and cylinder expansion linear velocity  $v$  can be obtained as follows:

$$\dot{\alpha} = Kv \quad (14)$$

Through the test, the steering axle from  $-30^{\circ}$  to  $30^{\circ}$  takes an average of 3.5s, the highest speed of HSG63 hydraulic cylinder is 0.3 m/s, the stroke is 300 mm, theoretically from one limit point to another limit point only takes 1 s, to meet the requirements.

Refer to the mechanical design manual, the required flow of steering valve and hydraulic pump should meet the following requirements:

$$Q \geq kQ_{max} \quad (15)$$

$$V = Q_{max}t \quad (16)$$

Where:  $Q$ —the maximum output flow of the steering valve and the hydraulic pump, L/min;

$k$ —System leakage coefficient, generally 1.1~1.3;

$Q_{max}$ —the maximum flow into the hydraulic cylinder, L/min;

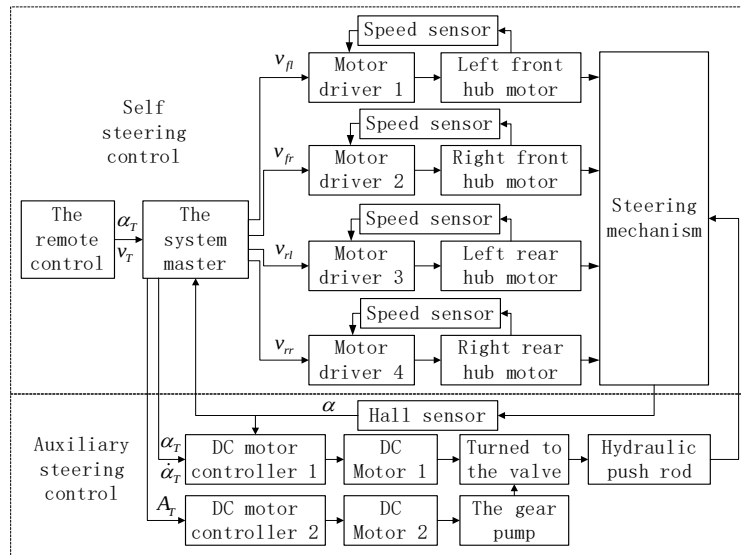
$V = \frac{\pi}{4}D^2L$ —Maximum volume of rodless cavity piston,  $m^3$ ;

Calculated  $Q_{max} = 16.02$  L/min, then the flow of the steering valve and hydraulic pump  $Q \geq 17.62$  L/min.

Therefore, BZZ1-E250 cycloidal rotary valve type open-core non-reactive full hydraulic steering gear with DM08RC brush DC reduction motor, CBN-E310 gear oil pump with DC72V DC reduction motor; Basic parameters: displacement 250 mL/r, flow 19 L/min, speed 100 RPM, maximum inlet pressure 16 MPa, maximum continuous back pressure 2.5 MPa, weight 6.48 KG, total length 181.5 mm, maximum working oil temperature  $80^{\circ}\text{C}$ , power steering torque 1.7~5.0  $\text{N}\cdot\text{m}$ , maximum human steering torque 136  $\text{N}\cdot\text{m}$ ; DM08RC brush-DC reducer motor basic parameters: working voltage 72 V, power 90 W, speed 120 r/min, maximum torque 6.98  $\text{N}\cdot\text{m}$ ; CBN-E310 gear oil pump basic parameters: rated pressure 16 MPa, maximum pressure 20 MPa, displacement 10 mL/r, rated speed 2000 r/min, maximum speed 3000 r/min, input power 7.7 kW; DC72 V DC motor is a commonly used hydraulic system motor with power of 4 KW.

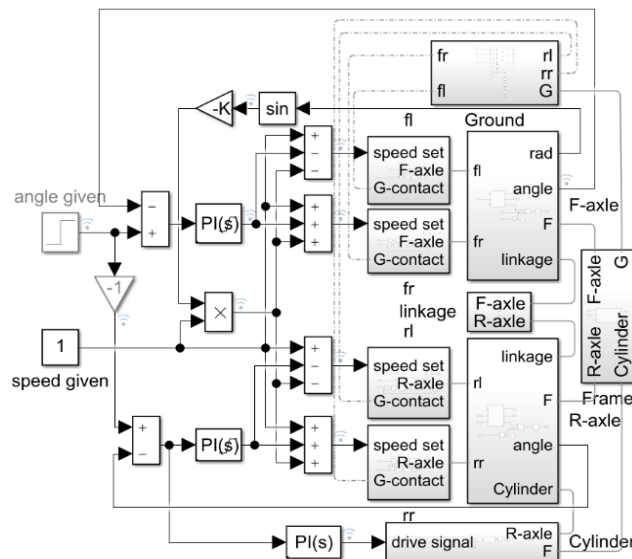
### 3.3 Control System Design

Steering control principle as shown in Figure 10, the remote control by wireless transmission Angle of target and the target speed is sent to the system master, on the one hand, the system master after calculating the four wheel speed through can bus sent to the four motor controller respectively, each motor controller based on the current target speed closed loop speed, at the same time, installed in the steering mechanism hall sensor, the Angle value of real-time detect feedback to the system master, so as to realize the steering control; On the other hand, the main control system sends the calculated target Angle and the calculated angular velocity to the DC motor controller 1 and the calculated target current to the DC motor controller 2 through RS485. DC motor 1 drives the steering spool to reach the target position at the set speed, so as to control the hydraulic cylinder to reach the target position at the set speed; DC motor 2 drives the gear pump to operate at the set torque to maintain the required hydraulic system pressure, while DC motor controller 1 receives the Angle signal fed back by Hall sensor in real time to achieve auxiliary steering control.



**Figure 10. Principle Diagram of Steering Control**

In this paper, PID control algorithm is used to realize the closed-loop control of the steering system. The Matlab/Simulink simulation model is established according to the speed of the vehicle, the speed of the four wheels corresponding to the rotation Angle and the relationship between the corresponding expansion length and expansion speed of the hydraulic cylinder, as shown in Figure 11.



**Figure 11. Simulation Model of High Clearance Sprayer**

The simulation results are shown in Figure 12. The solid line is the target Angle, and the dashed line is the Angle of the front bridge. The initial Angle is  $0^\circ$ . The given target Angle is  $10^\circ$  1 s later, and the Angle deviation is close to zero after 1.5 s, which meets the operating requirements of the high gap spray-machine.

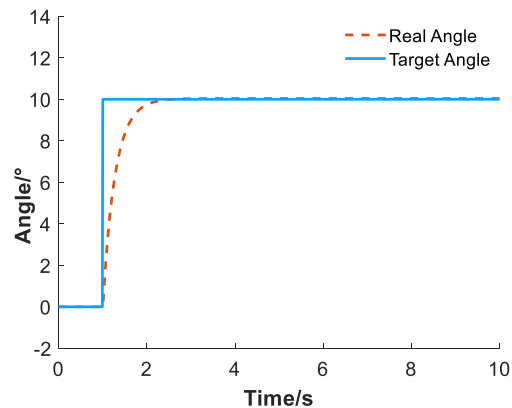


Figure 12. Angle Tracking Curve

#### 4. Test

##### 4.1 Build the System Platform

A test platform for the auxiliary steering system of the highland gap sprayer was built, mainly including vehicle attitude information collection, upper computer control system and lower computer control system, as shown in Figure 13.

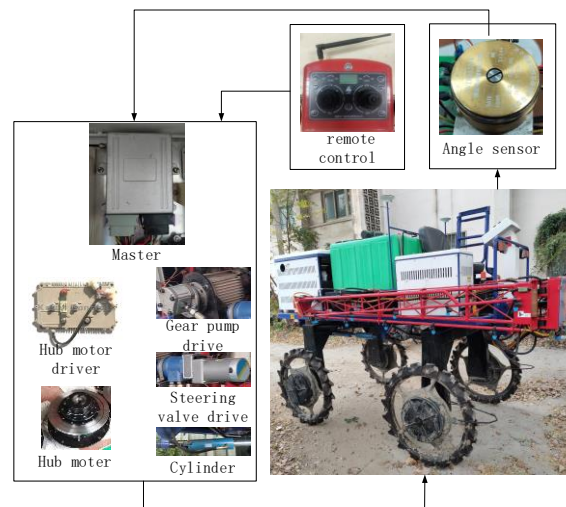


Figure 13. Test System Structure

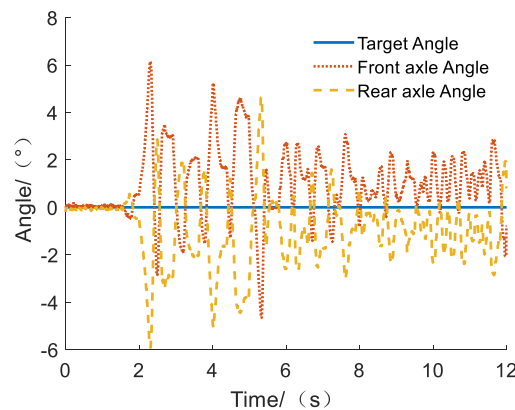
The main components of the system are: Hall Angle sensor, remote control, system master control, steering valve, gear pump, hydraulic tank, hydraulic push bar, DC motor, DC motor controller.

##### 4.2 Assisted Steering Walk Test

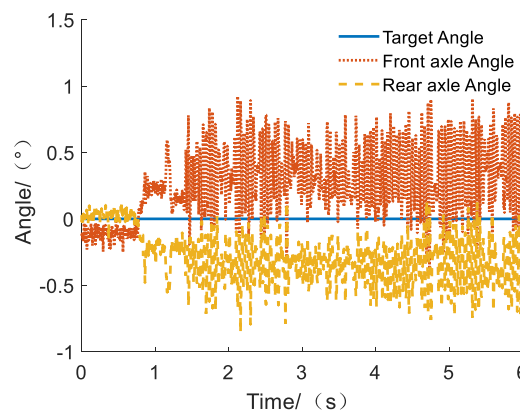
In order to verify the performance of the hydraulic auxiliary steering system, the independent operation of the self-steering system and the cooperative operation of the self-steering system and the auxiliary steering system were carried out under the condition of the slope of  $15^\circ$  and the length, width and height of the obstacle of  $20 \times 15 \times 10$  cm, respectively, and the test speed was 1 m/s. The Angle tracking

track of downhill test for independent auto-steering system operation is shown in Figure 14, and the Angle tracking track of downhill test for cooperative operation between auto-steering system and auxiliary steering system is shown in Figure 15. According to the test results, the maximum tracking deviation of independent auto steering system operation is  $6.1^{\circ}$ , and the maximum tracking deviation of cooperative operation of auto steering system and auxiliary steering system is  $0.9^{\circ}$ . In Figure 14 and Figure 15, 0~1 s is the downslope start time of the sprayer. In Figure 15, the jitter phenomenon is caused by the mechanical clearance between the connecting rod and the hydraulic cylinder and the chassis of the sprayer.

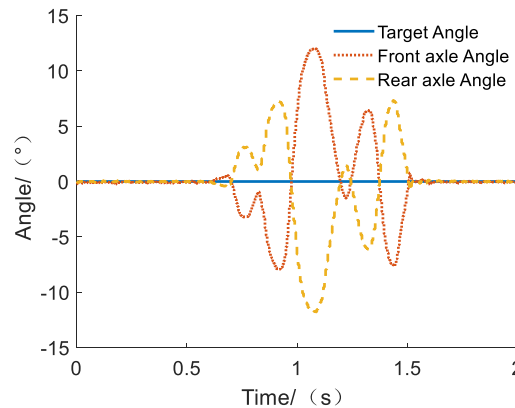
On the right side of the high clearance sprayer, the front and back two steering wheels successively crossed the obstacles. Independent operation since the steering system of surmounting obstacles test Angle tracking trajectory as shown in Figure 16, the steering system and auxiliary system collaborative obstacle test Angle tracking trajectory as shown in Figure 17, by the experimental results indicate: independent since the steering system work maximum tracking error is  $12.0^{\circ}$ , the steering system and auxiliary system collaborative maximum tracking deviation is  $1.2^{\circ}$ .



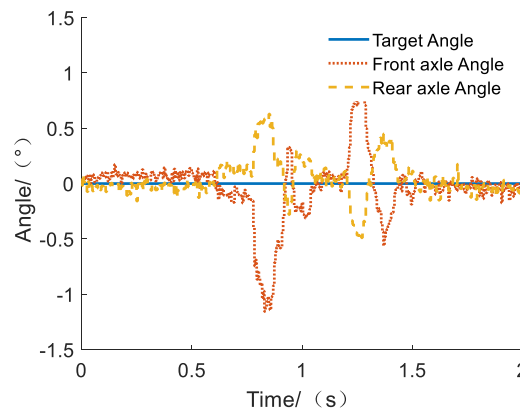
**Figure 14. Downslope Angle Trajectory of Independent Steering System**



**Figure 15. Downslope Angle Trajectory of Cooperative Steering System**



**Figure 16. Obstacle Crossing Angle Trajectory of Independent Steering System**



**Figure 17. Obstacle Crossing Angle Trajectory of Cooperative Steering System**

## 5. Conclusion

(1) Aiming at the steering instability problem of four-wheel independent electrically driven high-gap sprayer caused by insufficient reverse resistance moment of hub motor driver, a hydraulic auxiliary steering system based on motor driven steering valve was designed. Combined with the characteristics of chassis self-steering structure, the coordinated control of self-steering and auxiliary steering was verified by MATLAB simulation using PID algorithm. The simulation results show that the response time of the sprayer from  $0^\circ$  to  $10^\circ$  is 1.3s.

(2) The downslope and obstacle-crossing tests of the hydraulic auxiliary steering system of the highland gap sprayer were carried out respectively. The test results show that the system can walk at a given Angle, and the maximum Angle deviation of tracking is  $0.9^\circ$  and  $1.2^\circ$  in downhill and obstacle-crossing environments, respectively. The hydraulic auxiliary steering system has good running stability and accuracy, and can meet the requirements of the operation.

## References

- Ariff, M. H. M., Zamzuri, H., Saiful, A. M., & Idris, N. R. N. (2014). Independent Torque Control of an Independent-Wheel-Drive Electric Vehicle. *Applied Mechanics and Materials*, 663, 493-497. <https://doi.org/10.4028/www.scientific.net/AMM.663.493>
- Chen L. Q. et al. (2019). Design and Test of unmanned Spraying electromechanical Control System. *Journal of Agricultural Machinery*, 50(1), 122-128.
- Chen, L. Q. et al. (2019). Design and Test of auxiliary driving System of Highland Gap Plant Protection Machine. *Journal of Agricultural Machinery*, 50(9), 25-32.
- Derrick, J.-B., & Beviy, D.-M. (2009). Adaptive steering control of a farm tractor with varying yaw rate properties. *Journal of Field Robotics*, 26(6), 519-536. <https://doi.org/10.1002/rob.20291>
- Ding, L. et al. (2015). Design and Research of hydraulic Driving System of wheeled High-gap Atomizer. *Agricultural Mechanization Research*, 37(12), 106-109.
- Guo Q. B. et al. (2017). Design and implementation of a loss optimization control for electric vehicle in-wheel permanent-magnet synchronous motor direct drive system. *Applied Energy*, 2253-2259. <https://doi.org/10.1016/j.egypro.2017.03.644>
- Hanifah, R.-A., Toha, S.-F., H, M.-K., & Salmiah, A. (2016). Power reduction optimization with swarm based technique in electric power assist steering system. *Energy*, 102, 444-452. <https://doi.org/10.1016/j.energy.2016.02.050>
- Hu, K., & Zhang, W. Y. (2020). Design and Test of Multi-Mode Hydraulic Steering System of High-Altitude Slot Self-propelled Atomizer. *Journal of Agricultural Machinery*, 51(9), 366-373.
- Li, W. et al. (2019). Design and Test of multi-wheel Steering system of Uplands Self-propelled Spray Machine. *Journal of Agricultural Machinery*, 50(1), 141-151.
- Li, Z. L., Qiao, D. D., & Zou, H. M. (2020). Research on control of Electro-Hydraulic Steering System of Tractor Based on variational Domain Method. *Jiangsu Agricultural Sciences*, 48(4), 203-208.
- Liu J. Y. et al. (2016). Integrated navigation and positioning method of Agricultural Machinery based on GNSS/MIMU/DR. *Journal of Agricultural Machinery*, 47(S1), 1-7.
- Liu, Z. P. et al. (2018). Design of GNSS Automatic navigation Operating System for Levo ZP9500 High Altitude Gap Sprayer. *Journal of Agricultural Engineering*, 34(1), 15-21.
- Lu, Z. X. et al. (2016). Simulation and Test of Dual-channel PID Control for Tractor Line-controlled Hydraulic Steering System. *Journal of Agricultural Engineering*, 32(6), 101-106.
- Luo, D. M., Bao, Y. T., & Lin, H. (2013). Design of Hydraulic All-wheel Steering Device for Multi-axle Heavy Duty All-Trailer Machinery. *Mechanical Design*, 30(6), 86-89.
- Luo, W. W. (2018). Optimization Design of Steering Transmission System of Upline Slot Self-propelled Sprayer. *Times Agricultural Machinery*, 45(1), 107-109, 124.
- Mao, H. P. et al. (2012). Steering Anti-skid Control System of Hydraulic Four-wheel drive Sprayer at High Gap. *Journal of Agricultural Machinery*, 43(6), 58-62.

- Shen, Y. et al. (2020). Design and Test of Steering electric Chassis Control System of Uplands Gap Atomizer. *Journal of Agricultural Machinery*, 51(11), 385-392, 402.
- Shen, Y., Zhang, B. N., Liu, H., & Cui, Y. M. (2019). Design and Development of a Novel Independent Wheel Torque Control of 4WD Electric Vehicle. *Mechanika*, 25(3), 210-218. <https://doi.org/10.5755/j01.mech.25.3.21879>
- Xu, C. et al. (2016). Design and Research of Uplands Self-propelled Rod Spray Machine. *Chinese Journal of Agricultural Mechanization*, 37(1), 51-54.
- Xu, H. S. et al. (2013). Transmission Ratio Research of Hydraulic Steering by-Wire System. *Advanced Materials Research*, 455-459. <https://doi.org/10.4028/www.scientific.net/AMR.774-776.455>
- Zhang, J. et al. (2015). Design and Test of four-wheel Independent Steering Drive Control System for Agricultural wheeled Robot. *Journal of Agricultural Engineering*, 31(18), 63-70.
- Zhang, M. et al. (2020). Research progress of Agricultural Machinery Navigation Technology. *Journal of Agricultural Machinery*, 51(4), 1-18.
- Zhang, S. Q. et al. (2015). Research on steering Control Model of independent Drive System of Driving wheel of High Gap Spray Machine. *Agricultural Mechanization Research*, 37(1), 31-34.
- Zhuang, T. F. et al. (2018). Research status and development trend analysis of spray rod of large-scale self-propelled atomizer. *Journal of Agricultural Machinery*, 49(S1), 189-198.