Design and Test of Closed Hydraulic Transmission System of 4WD High-clearance Wheeled Sprayer

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Abstract In conjunction with the working characteristics of the high-clearance wheeled sprayer and the benefits of the closed hydraulic system, a series of reasonable working parameters should be established, and a hydraulic system that fulfills the requisite specifications should be designed. The AMESim software model is employed to construct a closed hydraulic transmission system, and the simulation analysis is then performed according to the data of hydraulic components. According to analysis results, the prototype can be optimized and upgraded, and a verification test is further carried out. The test results demonstrate that the designed closed hydraulic transmission system meets the actual working requirements of the high-clearance wheeled sprayer and provides a stable experimental platform for intelligent control of agricultural machinery.

Key words Closed hydraulic transmission system; High-clearance; Wheeled sprayer; Modeling and simulation; Verification test

1 Introduction

Agricultural machinery represents a crucial instrument utilized in the agricultural planting sector. In order to become an agricultural powerhouse, China must prioritize the acceleration of agricultural intelligence development and the implementation of scientific and technological enhancements to agricultural machinery and equipment that exhibit suboptimal efficiency, limited intelligence, and purely mechanical transmission[1]. The use of a hydraulic transmission in lieu of a purely mechanical transmission allows for the integrated electromechanical fluid control, particularly in the context of a closed hydraulic transmission system. The hydraulic components exhibit a high degree of standardization, with a simple installation structure, flexible location layout, reliable pipeline connection transmission, high efficiency, and the ability to achieve stepless variable speed, agile control response, precise action response, clear system control circuit, convenient use and maintenance, and easy to achieve electro-hydraulic integrated control, becoming the basic model of intelligent control. Concurrently, the low maintenance cost of this technology facilitates the realization of precision agricultural production and promotes the development of pollution-free and green agriculture, representing the development direction of integrated and intelligent agricultural machinery[2-4].

In the northwestern and northeastern regions of China, the cultivation of high-straw crops, such as corn and cotton, represents a significant component of the country's agricultural production, largely due to the region's geography and climate. The planting area is expansive, and the production cycle necessitates regular spraying operations, with tight operating schedule and a challenging work environment. Therefore, artificial methods are unable to fulfill these operations directly. In this context, the advantages of the closed hydraulic system in the hydraulic transmission combined with the working characteristics of the high-clearance wheeled sprayer have been considered in the design of a new high-clearance wheeled sprayer with closed hydraulic transmission. This design is intended to provide some new ideas for the development of agricultural machinery.

2 Closed hydraulic transmission system design

In order to meet the late growth and application needs of tall crops, the self-propelled sprayer must have a corresponding height difference from the growing ground of the crops and be able to carry a large amount of solvents in order to realize large-area operation. Concurrently, the operational environment of the field is intricate and subject to change. The summer temperature is elevated, situated in a remote location distant from the factory. This necessitates the comprehensive transmission system to be stable and dependable, capable of enduring prolonged periods of elevated temperature and the impact of heavy duty emergency stops.

The self-propelled sprayer must have a certain height difference with the crop growth ground. The utilization of a purely mechanical transmission system will inevitably result in the formation of a complex transmission structure, the imposition of rigorous specifications for the production of parts, the introduction of a cumbersome installation process, the creation of difficulties in use and maintenance, and the inability to achieve intelligent control. Accordingly, a closed hydraulic transmission system is selected as the optimal solution for the energy transfer. The fundamental principle of the closed hydraulic transmission system of the high-clearance sprayer is illustrated in Fig. 1. The closed-type hydraulic transmission system comprises a closed-type pump as the primary power element, a variable motor as the primary working element, a fixed displacement pump, and a variety of circuit control ele-

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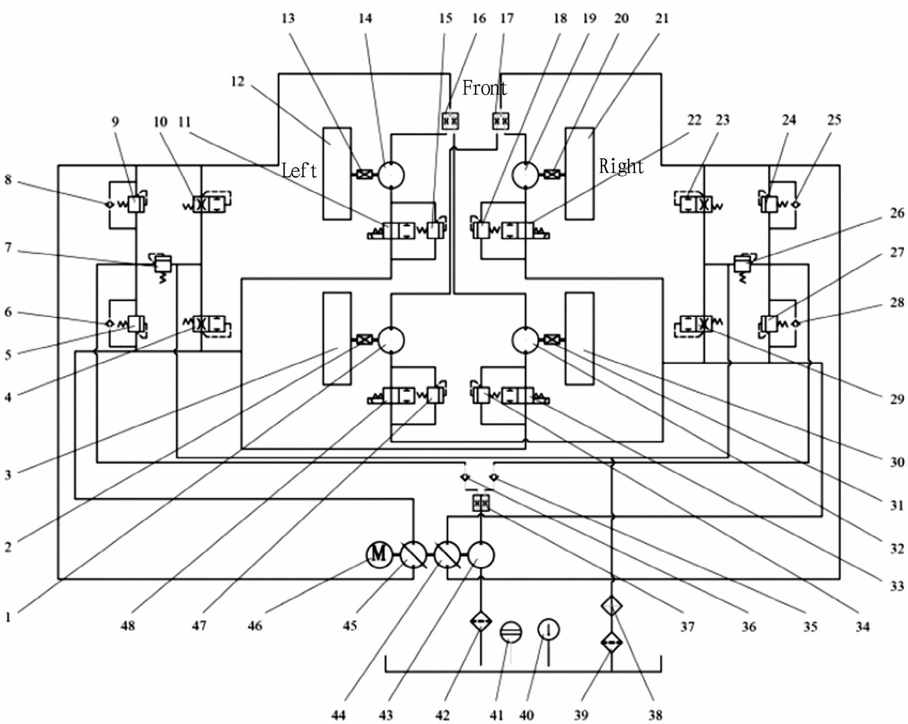
ments for oil replenishment. In addition, it serves to complete auxiliary hydraulic action except walking.

The closed drive system walking principle is based on the concept of changing the displacement size of the closed pump and the direction of liquid flow, which in turn alters the motor speed and direction. This allows for the adjustment of the spraying machine's walking speed and direction to meet the demands of its heavy duty work and the varying requirements of no-load driving. The closed drive system facilitates the efficient transmission of substantial, consistent torque during operation and ensures the smoothness of steering, which is crucial for high-clearance wheeled machines with a high center of gravity[5-7].



NOTE 1. Left rear wheel hydraulic motor; 2, 13, 20, 31. Decelerator; 3, 12, 21, 30. Tire; 4, 10, 23, 29. Two-position, two-way center valve; 5, 7, 9, 15, 18, 24, 26, 27, 34, 47. Relief valve; 6, 8, 25, 28, 35, 36. One-way valve; 11, 22, 33, 48. Two-position, two-way magnetic exchange valve; 14. Left front wheel hydraulic motor; 16, 17, 37. Diverter valve; 19. Right front wheel hydraulic motor; 32. Right rear wheel hydraulic motor; 38. Cooler; 39, 42. Filter; 40. Thermometer; 41. Liquid level gauge; 43. Fixed displacement pump; 44, 45. Closed pump; 46. Engine.

Fig. 1 Principle of closed hydraulic transmission system

3 Calculation and selection of driveline components

3.1 Driveline calculations The high clearance sprayer, operating in a field environment characterized by complex and harsh conditions, necessitates a focus on the analysis and selection of components within the closed hydraulic transmission system. This is to ensure that the machine is stable and efficient throughout the work process. An independent 4 WD closed hydraulic transmission system is therefore designed, with the main transmission parameters shown in Table 1.

In accordance with the specifications outlined in the design requirements and parameter definitions, a comprehensive analysis should be conducted to ascertain the actual power requirements of machinery in the production and operational process. The machine requires the greatest amount of power when operating in an uphill direction, as specified in the design manual[8-10].

The machine is designed to operate at a climbing angle of 15°. The operating conditions of the high-clearance sprayer at maximum force on an uphill slope are as follows:

Table 1 Main transmission parameters of high-clearance sprayer

Item	Parameter indicator
Vehicle drive system	Four-wheel independent drive, closed hydraulic transmission
Motor deceleration ratio	1 : 53.7
Operation travel speed//km/h	4.2-4.5
Driving speed on no-load road surface//km/h	7.0-7.2
No-load weight//kg	2 400
Heavy duty working weight//kg	7 500
Driving wheel radius//m	0.65
Work geographical environment	Climbing angle 15°, one wheel suspended driving

$$\Sigma F = F_f + F_w + F_g + F_a \tag{1}$$

where F_f denotes the tire rolling friction; F_g represents the uphill resistance; F_x stands for the air resistance; and F_a is the acceleration resistance.

Given that the velocity of the machinery remains constant throughout the operational process, it is unnecessary to consider the resistance factor of acceleration in the calculation process. Concurrently, the velocity of the vehicle in motion is considerably below the legal speed limit for roadways, which is 50 km/h. Consequently, the air resistance generated by the vehicle is insignificant. The combined actual tractive effort is the work done by the machine to overcome the rolling friction of the tires during movement and the resistance encountered when traversing an uphill slope. The weight of the heavy duty machine in question is 7 500 kg.

In formula (1), $F_f = f \cdot G \cdot \cos a$, $F_g = G \cdot \sin a$, where G represents the total working gravity of the machinery when it is fully loaded; f is the rolling friction resistance coefficient of the tires (in order to improve the passability of the machinery on the road with different road conditions, the rolling friction resistance coefficient of the tires is taken to be 0.12); and the angle of the slope, a, is taken to be 15°.

The formula (1) indicates that the maximum traction force required to design the machinery is 27 542 N. In order to ensure that the machinery functions properly in special circumstances, there is a certain margin for traction. The selected traction force is 30 000 N.

The drive speed of the hydraulic motor is as follows:

$$N_m = (V \cdot i_1)/(2 \cdot \pi \cdot r) \qquad (2)$$

where N_m denotes the working speed of the hydraulic motor drive shaft; i_1 represents the integrated transmission ratio of the decelerating mechanism; r stands for the maximum outer diameter parameter of the driving wheel.

The integrated transmission ratio of the existing parameter decelerator is 1 : 53.7, the maximum field speed of a vehicle engaged in heavy duty work is 4.5 km/h, and the outer diameter of the drive wheel is 0.65 m. Upon entering the parameters $V = 4.5$ km/h, $i_1 = 53.7$, and $r = 0.65$ m, the result is N_m = 987 r/min.

The maximum field speed of the no-load vehicle is 7.2 km/h when traveling on a no-load road. The $N_{m\,\mathrm{empty}}$ value is obtained, which is equal to 1 579 r/min.

The working power of a hydraulic motor is: $P = F \cdot V$ (3)

where F denotes the mechanical traction force required for operation, while V represents the normal working speed of machinery. Upon entering the parameters $F = 30\ 000$ N, $V = 1.25$ m/s, the result is $P = 11$ kW (four-wheel independent drive).

The motor's heavy duty operating torque is as follows:

$$T_w = 60\mathrm{P}/(2 \cdot \pi \cdot N_m) \qquad (4)$$

where P represents the driving working power of the hydraulic motor, while N_m denotes the normal working speed of the hydraulic motor. The introduction of the parameters yields a value of T_w = 106 N · m for heavy duty operation of the machinery and a value of 22.2 N · m for no load.

The formula for calculating the displacement of a motor in operation is as follows: $q = (2 \cdot \pi \cdot T_w)/(P_1 \cdot \eta_m)$ (5)

where P_1 represents the rated working pressure of the hydraulic system; T_w stands for the normal working torque of the hydraulic motor; and η_m denotes the mechanical efficiency of the hydraulic motor, which is valued at 0.9.

The system's rated working pressure is selected to be 20 MPa, and the minimum displacement of the motor under heavy load (q = 36.9 mL/r) is obtained after the parameters are brought into the system.

The results of the heavy duty uphill normal operation calculation indicate the hydraulic motor selection requirements are as follows: when the rated working pressure is 20 MPa, the minimum speed, heavy duty fluid displacement, transmission torque and required working power are 987 r/min, 36.9 mL/r, 106 N · m and 11 kW, respectively. A comparison of the available motor models on the market reveals that the LTM07B is the optimal choice due to the optimized parameters.

In selecting hydraulic control components, the working flow rate of the pump Q_p is determined from the required working flow rate of the motor[11].

The relationship between the working displacement (q), the operating speed (n), and the combined flow rate (Q) of a hydraulic motor is as follows:

$$Q = qn/1\ 000 \qquad (6)$$

where q represents the displacement of the hydraulic motor in operation (mL/r); n denotes the speed of the hydraulic motor in operation (r/min); Q represents the integrated flow rate of the hydraulic motor (L/min). The power of the hydraulic pump (motor) is calculated according to the following formula:

$$N = P_2 Q/ 612 \qquad (7)$$

where P_2 represents the working pressure of the hydraulic motor in its normal heavy duty configuration (MPa); N denotes the integrated power of the hydraulic motor (kW).

The LTM07B is rated for heavy duty operation at 21 MPa, with a maximum operating pressure of 30 MPa. The rated heavy duty operating speed is now a maximum of 987 r/min, and the motor is rated for a displacement of 39.5 mL/r. Consequently, the maximum operating flow rate required for the motor is $Q = 39$ L/min, and the maximum operating power is $N = 13.38$ kW.

In conjunction with the principle of the closed hydraulic drive system of the high-clearance sprayer (Fig. 1), the working flow rate of the hydraulic pump Q_P in the event of simultaneous operation of hydraulic cylinders or hydraulic motors is calculated as:

$$Q_P \geqslant K\ (\Sigma Q_{\max}) \qquad (8)$$

where K represents the leakage coefficient of the hydraulic transmission system and is generally assumed to be 1.1 – 1.3; $\Sigma Q_{\max}$ denotes the maximum total working flow of a hydraulic cylinder or hydraulic travel motor operating simultaneously.

3.2 Parameter selection of transmission system components

In a closed hydraulic transmission system, two pumps are employed to control two pumps each, with the selection of these pumps being carried out. The Q_P is calculated to be 85.8 L/min. Closed pumps may opt for the Rexroth tandem double 60 closed pumps. The principal parameters are as follows: the nominal displacement of the double pumps is (60 + 60) mL/r, the rated maximum working pressure is 31.5 MPa, the rated working rotation speed is 1 500 r/min, and the actual combined total displacement is (90 + 90) L/min.

The working principle of the closed hydraulic transmission system indicates that the replenished oil volume is approximately 1/4 – 1/3 of the entire system oil volume. The displacement of the replenishment pump is estimated to be (30 – 40) L/min. In consideration of the actual situation, a fixed displacement pump with a displacement of 36 L/min and a rated working speed of 1 500 r/min is therefore selected.

Due to the inherently complex and changeable nature of field operations, it is imperative that the machinery is capable of hanging in the air with a single wheel. At this moment, the weight of the machine will be apportioned to the remaining three wheels. The two-position, two-way magnetic exchange valve depicted in Fig. 1 will function accordingly, and the working pressure in the system will be adjusted to 21 ×4/3 MPa. In order to guarantee the optimal functionality of the system, the pressure of the relief valve is set to 28 MPa.

The preceding design calculations and selection process have yielded the main hydraulic components parameters of the closed hydraulic transmission system of the high-clearance sprayer, as illustrated in Table 2.

Table 2 Parameters of main hydraulic components

Item	Parameter indicator
Motor structure comprehensive transmission ratio	1 : 53.7
Heavy duty maximum operating speed//km/h	4.2 – 4.5
No-load field operating speed//km/h	7.0 – 7.2
No-load operating weight//kg	2 400
Heavy duty mechanical weight//kg	7 500
Maximum working radius of the wheel//m	0.65
Working field geography	Climbing capacity 15°
Engine operating speed//r/min	2 400
Motor operating parameter I: speed/displacement (r/min)/(mL/r)	987/39.5
Motor operating parameter II: speed/displacement (r/min)/(mL/r)	1 579/25
System normal working pressure//MPa	21
Set pressure of relief valve//MPa	28
Series closed pump displacement//mL/r	60 + 60
Displacement of fixed displacement pump//mL/r	36
Working speed of series closed pump//r/min	1 500
Working speed of fixed displacement pump//r/min	1 500

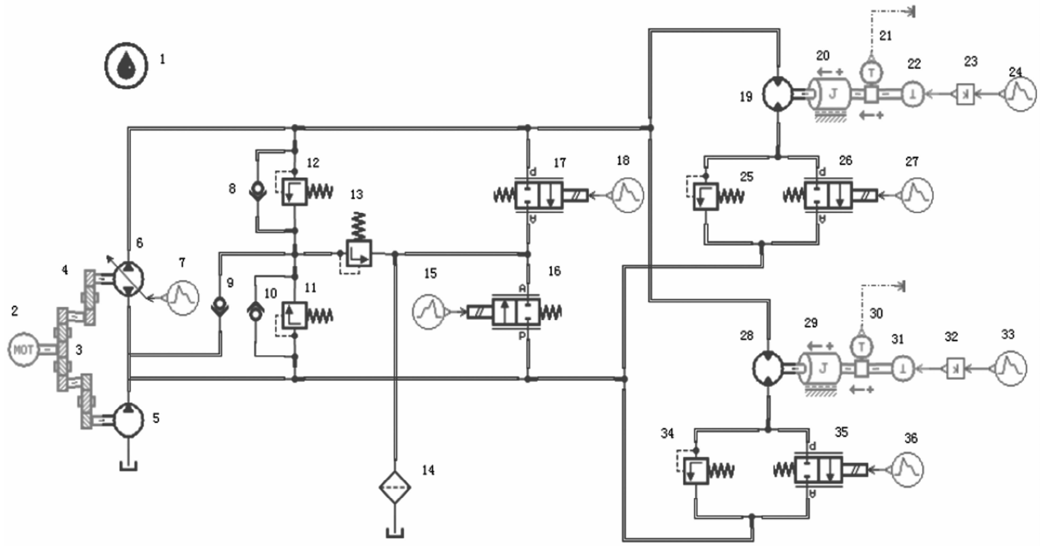
4 Simulation study of closed hydraulic transmission system

4.1 Modeling hydraulic transmission systems The design schematic analysis indicates that the series double 60 closed pump, which controls two hydraulic motors, allows for the two hydraulic systems to operate independently of each other. The function principle is identical in both cases. The fixed displacement pump simultaneously replenishes the two systems in equal amounts. Consequently, an idealized simulation system can be constructed in the AMESim simulation software, and only one of the systems may be analyzed. The simulation parameters of the main components of the transmission system can be obtained according to the operating parameters of the closed variable pump, fixed displacement pump, and two travel motors, as well as the design of the closed hydraulic transmission system schematic diagram, as shown in Table 3[12].

Table 3 Simulation parameters of components

Model name	Parameter
Ideal working pump output speed//r/min	1 300
Closed variable pump displacement//mL/r	60
Motor speed (heavy duty)//r/min	987
Motor displacement (heavy duty)//mL/r	39.8
Starting motor torque//N · m	0
Heavy duty motor operating torque// N · m	106
System relief valve pressure//MPa	28
Relief valve//L/min	120
Normal working variable pump signal	0 – 18 sec for 1
	19 – 22 sec for – 1 – 1
	23 – 38 sec for – 1
	39 – 40 sec for – 1 – 0
18 Signal source	0 – 20 sec for 1
	21 – 40 sec for 0
15 Signal source	0 – 20 sec for 0
	21 – 40 sec for 1
24, 33 Motor torque signal	0 – 2 sec for 0 – 10.6
	3 – 18 sec for 10.6
	19 – 22 sec for 10.6 – – 10.6
	23 – 38 sec for – 10.6
	39 – 40 sec for – 10.6 – 0
27, 36 Pressure valve signal	0 – 40 sec for 1
Heavy duty torque of No. 28 motor in the event of suspension of No. 19 motor//N · m	141
Torque of No. 19 motor in the event of suspension//N · m	0
No. 25, No. 34 load relief valve//MPa	21
18 Signal source	0 – 20 sec for 1
	21 – 40 sec for 0
Motor suspended variable pump signal	0 – 40 sec for 1
18 Signal source	0 – 40 sec for 1
15 Signal source	0 – 40 sec for 0
24 Motor torque signal	0 – 2 sec for 0 – 10.6
	3 – 20 sec for 10.6
	21 – 22 sec for 10.6 – 0
	23 – 25 sec for 0
	26 – 40 sec for 10.6
33 Motor torque signal	0 – 2 sec for 0 – 10.6
	3 – 20 sec for 10.6
	21 – 22 sec for 10.6 – 14.1
	23 – 25 sec for 14.1
	26 – 40 sec for 10.6
27 Pressure valve signal	0 – 21 sec for 1
	22 – 25 sec for 0
	26 – 40 sec for 1
36 Pressure valve signal	0 – 40 sec for 1
23, 32 Gain	– 10

In accordance with the tenets of the closed hydraulic transmission system and the parameters utilized in the simulation of the primary components of the transmission system, a simulation model of the closed hydraulic transmission system is generated in the AMESim simulation software, as illustrated in Fig. 2.

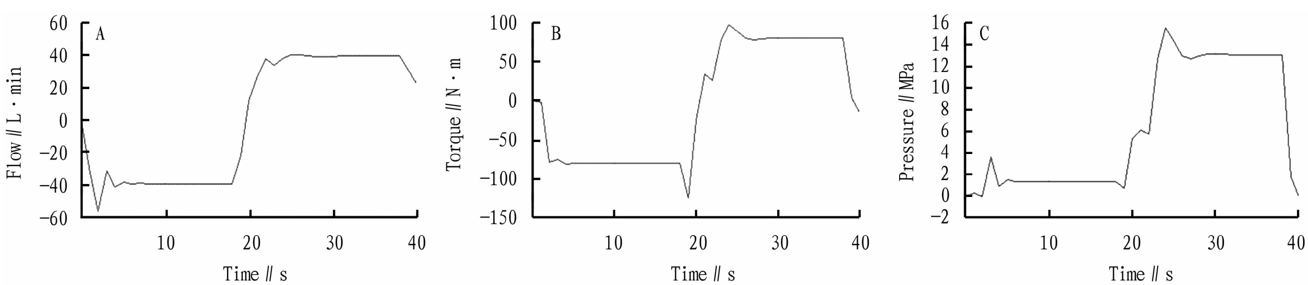


NOTE 1. Hydraulic oil; 2. Electrical machinery; 3. Differential; 4. Decelerator; 5. Fixed displacement pump; 6. Closed variable pump; 7, 15, 18, 24, 27, 33, 36. Piecewise linear signal source; 8, 9, 10. One-way valve; 11, 12, 13, 26, 36. Relief valve; 16, 17, 27, 37. Two-position, two-way reversing valve; 19. Left front wheel hydraulic motor; 20, 29. Simulation load; 21, 30. Torque instrument; 23, 32. Gain; 28. Right rear wheel hydraulic motor.

Fig. 2 Simulation model of closed hydraulic transmission system

4.2 Analysis of operating parameters for system simulation

The design parameters of the closed drive system under different working conditions are simulated in AMESim, and the working characteristic curves of the left front wheel hydraulic motor and the right rear wheel motor in the drive system of the main components in the hydraulic system simulation circuit under heavy duty driving are shown in Fig. 3.



NOTE A. Flow characteristic curve; B. Transmission torque curve; C. Pressure characteristic curve.

Fig. 3 Normal working characteristic curve of motor (19, 28)

In the event of a hydraulic transmission system being subjected to a heavy duty, the vehicle is divided into four distinct states: a state of rapid acceleration to normal driving at 0 – 18 sec, a direction change at 19 – 22 sec, reverse driving at 23 – 38 sec, and finally a deceleration stop at 39 – 40 sec. The conclusion can be drawn from the comparison and analysis of the relationship between the working characteristic curve and each graph.

(i) The working flow rate, working pressure, and transmission torque of the closed hydraulic transmission system exhibit a gradual increase over a period of 1 – 3 sec prior to the rapid commencement of the work. To meet the characteristics of the hydraulic system, the oil replenishment of the oil replenishment pump each time is 1/4 – 1/3 of the oil volume of the entire system.

(ii) The closed hydraulic system modifies the direction of oil transmission within the hydraulic system by regulating the closed pump angle (positive-negative). The size of the adjustment angle allows for the alteration of the flow size, thus facilitating the implementation of non-impact commutation and stepless speed change.

(iii) The closed hydraulic system operates at an average pressure of 13.5 MPa during steady driving and at a maximum pressure of 15.5 MPa during commutation. The system pressure of 21 MPa, as specified in the design parameters, is sufficient to meet the design requirements. Concurrently, the transmission torque pulsation occurs during the commutation, rather than the oscillation phenomenon, which further substantiates the viability of the system to achieve impact-free commutation. Meantime, the peak transmission torque is 100 N · m, which is less than the motor's rated torque of 449 N · m. Thus, the motor meets the selection requirements.

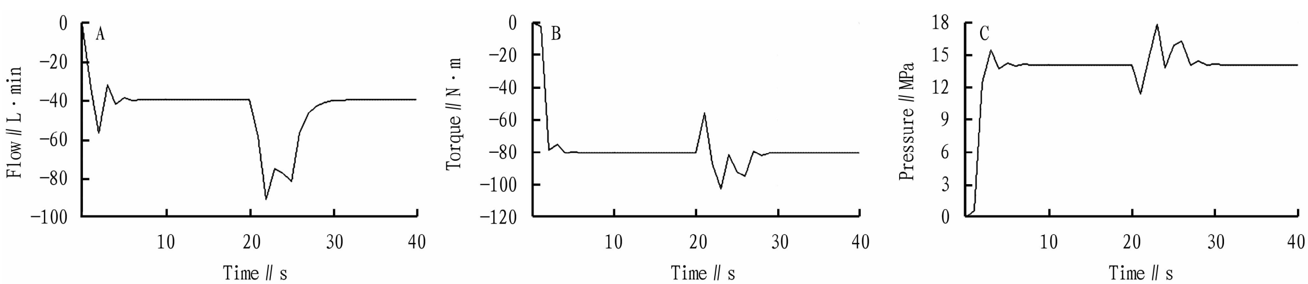
The parameterization of AMESim enables the operation of No. 27 load relief valve in the driveline when No. 19 left front wheel hydraulic motor is idling. The driving characteristic curve of the No. 28 right rear wheel motor is depicted in Fig. 4.

The combination of simulation curve graphics and actual use has led to the division of the mechanical motion state into four distinct categories. The first category, which occurs between 0 and 20 sec, is characterized by a rapid acceleration to normal driving. The second category, which occurs between 21 and 22 sec, is marked by the operation of No. 27 load relief valve in the event of suspension of No. 19 left front wheel hydraulic motor. The third category, which occurs between 23 and 25 sec, is defined by the continuous operation of the load relief valve. Finally, the fourth category, which occurs between 26 and 40 sec, is marked by the normal operation of No. 19 left front hydraulic motor and No. 28 right rear motor.

The comparison and analysis of the working characteristic curves in Figs. 3 – 4 allows for the drawing of conclusions.

(i) When No. 19 left front wheel hydraulic motor is suspended, No. 27 loading relief valve is operational. This results in fluctuations in the working flow, working pressure, and transmission torque of No. 28 motor, which are significantly different from those observed during normal driving. The flow rate undergoes a change from 39.5 L/min to 82 L/min, while the torque alters from 80 N · m to 55 N · m and finally 105 N · m. The pressure also undergoes a transformation, shifting from 14 MPa to 12 MPa and ultimately reaching 18 MPa. The reduction in transmitted torque is attributed to a decline in pressure within the air suspension system of No. 19 left front wheel hydraulic motor. No. 27 loading relief valve serves to enhance the system's pressure capacity, enabling the system to provide the motor with a faster rotation speed, higher operating pressure, and transmit more torque, thereby increasing the machine's power to overcome the predicament.

(ii) As illustrated in Figs. 3 – 4, the vehicle is capable of functioning in a variety of operational contexts. The closed hydraulic transmission system is designed to facilitate impact-free commutation and stepless transmission. Concurrently, the torque of the motor is in accordance with the selection criteria, and the working pressure of the system aligns with the design specifications.



NOTE A. Flow curve; B. Transmission torque; C. Pressure curve.

Fig. 4 Driving characteristic curve of loaded relief valve operating motor (28)

5 Verification test of closed hydraulic system

The 4WD high-clearance sprayer developed by Bortala Mongolian Autonomous Prefecture Xingwang Science and Technology Co., Ltd., was utilized as the test platform for field verification in accordance with the principle of closed hydraulic transmission systems. Fig. 5 depicts the field verification test equipment. Tables 4 – 5 present the field verification test parameters and test results, respectively.

Fig. 5 Field verification test

Table 4 Field validation test parameters

Item	Parameter indicator
Vehicle drive system	Four-wheel independent drive, closed hydraulic transmission
Motor deceleration ratio	1 : 53.7
Operation travel speed // km/h	4.2 – 4.5
Full duty working weight // kg	7 500
Drive wheel radius // m	0.65
Work geographic environment	Climbing angle 15°, one wheel suspended driving
Soil humidity // %	25 – 30
Soil compaction // Pa	100 – 120

(i) The high-clearance wheeled sprayer had a gross weight of 7 500 kg under heavy duty conditions. It was equipped with a tandem double 60 closed pump operating at 1 500 r/min, and had a normal travel speed of 5.1 km/h at a normal climbing capacity of 15°. The actual working pressure of the system was 12.5 MPa, with a peak value of 14.65 MPa during commutation. This occurred in a non-slope condition with a no-load travel speed of 8.13 km/h and a system pressure of 5.4 MPa. Stepless speed change can be achieved without commutation shock.

(ii) A high-clearance wheeled sprayer was subjected to a single round of suspended travel loading relief valve operation while heavily loaded and traversing a potholed roadway. The test results demonstrated that the high-clearance wheeled sprayer was capable of traversing road surfaces with a height differential of 25 – 30 cm under heavy duty. Furthermore, the maximum peak pressure within the system was 17.8 MPa, which was below the 27 MPa threshold.

The preceding results indicate that the design of the closed hydraulic transmission system is capable of meeting the requirements of the high-clearance wheeled sprayer in actual working conditions.

Table 5 Field experiment data

Parameter indicator	Group					
	1	2	3	4	5	6
Main pump flow//mL/r	60 + 60	60 + 60	60 + 60	60 + 60	60 + 60	60 + 60
Auxiliary pump flow//mL/r	36	36	36	36	36	36
Motor flow//mL/r	39.5 × 4	25 × 4	39.5 × 4	39.5 × 4	39.5 × 4	39.5 × 4
Vehicle weight//kg	7 500	2 400	7 500	7 500	7 500	7 500
Diesel engine transmission speed//r/min	1 500	1 500	1 500	1 500	1 500	1 500
Continuous working time//min	60	60	60	60	60	60
Climbing angle//°	5	5	10	15	15	15
Road height difference//cm	5 – 10	5 – 10	5 – 10	5 – 10	10 – 20	25 – 30
System temperature//℃	62	60	64	64	66	68
Operation travel speed//km/h	5.1	8.1	5.1	5.1	5.1	5.1
System pressure//MPa	10.3	5.4	11.0	12.5	14.6	17.8

6 Conclusions

(i) The actual working conditions have been meticulously analyzed during the design process and transformed into design parameters. The use of parameters as a target, in conjunction with the design program and computer-aided design software, enables the detailed demonstration of the test to be carried out. The intricate hydraulic transmission control system is presented in a straightforward manner to both the end user and the designer. In the meantime, the mutual movement relationship and geometric parameter relationship between the hydraulic components are displayed intuitively through simulation graphics, allowing for the timely identification of motion curves that deviate from the normal operational law during the working process. Timely and effective data adjustment can be utilized to solve problems at the design stage, thereby providing a solid foundation for the final production of the product. This approach can effectively shorten the product development cycle, improve the quality and stability of the product, and thus enable the enterprise to seize the market more rapidly.

(ii) The computer-aided software AMESim is employed in the simulation tests, which are based on ideal operations. It should be noted that the results of the simulation are subject to certain limitations. The conclusion can not be directly applied to existing mechanical products. It is necessary to manufacture a prototype to verify the actual working environment and to analyze the verification results and simulation results in order to identify shortcomings, implement continuous improvement, and gradually perfect the product. The design of the closed hydraulic transmission system can be applied to the high-clearance wheeled sprayer, thereby contributing to the intelligent cause of agricultural machinery in China.

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