DESIGN AND TEST OF HYDRAULIC DRIVEN SYSTEM FOR SMALL MULTIFUNCTIONAL AGRICULTURAL CHASSIS
/
小型多功能农用底盘液压驱动系统的设计与试验

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DOI: https://doi.org/10.35633/inmateh-68-55

Keywords: *Agricultural chassis; Hydraulic drive system; AMESim; Simulation analysis; Performance test*

ABSTRACT
In view of the problems of the output power and operation, when the multi-function machine in hilly and mountainous areas driving in the field, the hydraulic control drive system of small multi-function agricultural hydraulic chassis is designed. The key components of the hydraulic drive system were selected and matched. The hydraulic system simulation model was established in AMESim simulation analysis software, and the dynamic analysis of the hydraulic system operation under different conditions is carried out. The simulation analysis results show that the hydraulic system has a large impact and vibration when it is started instantaneously, and the hydraulic system has smaller impact and vibration when it is started stably. It is consistent with the actual working state of the hydraulic chassis. Under the two starting controls, the maximum flow of the hydraulic pump is 50L/min, the motor torque is about 440N · m, and the motor stable pressure is 6Mpa, the motor speed is 96 r/min, which is within the bearing range of the hydraulic components. The simulation output parameters are basically consistent with the theoretical calculation results, and meeting the design requirements. The chassis performance test results show that the maximum crossing height of the chassis is 200mm, the crossing width is 300mm, the maximum deviation of high-speed straight driving is 2.57m, it can stably pass 20° slope, and the operation is stable and the steering is flexible. All performance parameters can better meet the requirements of chassis operation in hilly and mountainous areas.

摘要
针对丘陵山地多功能作业机在田间行驶中在输出动力及操作中存在的问题，设计了一种履带式小型多功能农用液压底盘的液压控制驱动系统。对液压驱动系统关键元件进行了选型与参数匹配，利用AMESim仿真分析软件建立了液压系统仿真模型，对不同条件下液压系统作业状况进行了动态分析，仿真分析结果可知，瞬时启动时，液压系统有较大冲击及震荡，稳定启动时，液压系统波动小、几乎无冲击及震动，与液压底盘的实际工作状况一致，两种启动控制下变量泵流量最大值为50L/min时，马达扭矩约为440N.m，马达稳定时压力为6Mpa，马达转速为96r/min，均在液压元件的承受范围之内，且仿真输出的参数与理论计算结果基本一致，符合设计要求。通过底盘作业性能试验结果可知，底盘最大跨越高度为200mm，跨越宽度为300mm，高速直线行驶最大偏移量为2.57m，可稳定通过20°坡度，且操作平稳，转向灵活，各项性能参数均能较好地满足丘陵山地底盘作业要求。

INTRODUCTION
At present, agricultural machinery is developing rapidly, but it is only suitable for large and flat areas. In hilly and mountainous areas, due to the great undulates and complex conditions of the ground, ordinary agricultural machinery has insufficient adaptability to the terrain, poor vehicle stability, which affects the work quality of the machinery. Therefore, it is necessary to effectively solve the adaptability, controllability and stability of chassis operation of agricultural machinery to improve the level of agricultural mechanization in hilly and mountainous areas *(Wang et al., 2022; Wu et al., 2022, Chen, 2021)*. The hydraulic drive system uses hydraulic oil to drive the drive wheel rotating to achieve the walking function. The complex mechanical transmission links are omitted, so that the transmission system is simplified and the transmission efficiency is improved. Moreover, the hydraulic drive operation is easy, the maintenance and overhaul are convenient, which is very consistent with the operation requirements in hilly and mountainous areas *(Ling 2017; Gupta et al., 2019; Hu et al., 2019; Yermek et al., 2019)*.

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Therefore, hydraulic drive technology is widely used in walking machinery *(Suslov et al., 2020; Zhang 2017; Gradl et al., 2016)*. In recent years, scholars have done a lot of research on the application of hydraulic technology in the walking system of agricultural machinery *(Nie et al., 2020; Andrzej et al., 2016; Hu et al., 2018; Sachin 2018; Kumar et al., 2021; Gonzalez et al., 2016)*. *Kodrič and Pehan (2020)* designed a simple and effective hydraulic management system by using a small IC diesel engine to provide power for the hydraulic pump, which is composed of basic valves, electrical sensors and switches. The system can predict and analyze the important working conditions of the machine. *Zhang et al., (2022)* designed a hydrostatic chassis drive system for a large plant protection machine. The system realizes the prevent slip and rotation of the plant protection machine through the combination of the hydraulic drive system and the diverter valve. The designed hydraulic chassis of the plant protection machine has good driving stability. *Hu and Zhang (2018)* designed rice transplanted with an entirely hydraulic chassis. The chassis is driven by a single pump driving four motors, and a diverter valve is set to prevent wheel slipping. *Fan et al., (2018)* designed an output system based on hydraulic transmission for operation in hilly and mountainous areas, and the performance of the hydraulic system were verified through the hydraulic system simulation and prototype test. The hydraulic system technology has been applied in agricultural machinery, but few hydraulic chassis are suitable for operation in hilly and mountainous areas. On the basis of the existing hydraulic chassis research, this paper designs a kind of hydraulic drive system of tracked multi-functional agricultural hydraulic chassis suitable for hilly and mountainous areas, in order to improve the adaptability, flexibility and drive performance of the walking system of hilly and mountainous work tools, and meet the needs of the development of agricultural mechanization in hilly and mountainous areas.

MATERIALS AND METHODS

STRUCTURE AND WORKING PRINCIPLE

The structure of the chassis hydraulic drive system is shown in Figure 1. The system adopts shunt transmission mode. Part of the power output of the engine is sent to the hydraulic pump to drive the walking wheel, and other part of the power is transmitted to the rear power output shaft to provide power for the agricultural machinery. The traveling drive system is supplied with oil by the double-pump. The drive wheels are driven rotation by the hydraulic motor to realize the traveling and steering functions of the chassis. The operating handle controls the steering and speed of the motor by controlling the direction change and opening of the reversing valve, and realizes the forward, backward, and variable-speed driving of the chassis.

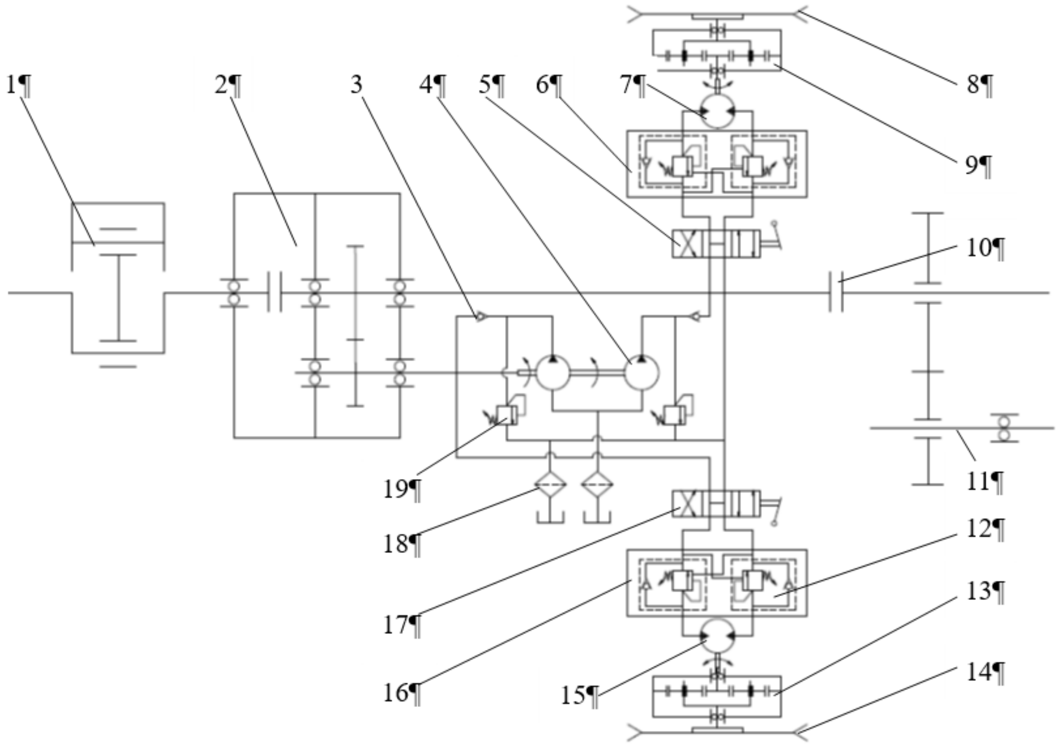


Fig. 1 - Schematic diagram of chassis hydraulic drive system

1. Diesel engine; 2. Transfer case; 3. One way valve; 4. Double-pump; 5. 17. Reversing valve; 6. 16. Self -locking device; 7. 15. Hydraulic motor; 8. 14. Drive wheel; 9. 13. Brake; 10. Clutch; 11. Rear output shaft; 12. One way sequence valve; 18. Oil filter; 19. Overflow valve

When the hydraulic system is working, the outputs power of the engine drives the pump to provide pressure oil for oil circuits by the transfer case. In order to prevent the impact of hydraulic oil backflow on the pump, the pressure oil inflows the reversing valve through the one-way valve. The operating handle on the chassis controls the reversing and opening of the corresponding reversing valve. When the operating handles on both sides are pushed forward, the chassis moves forward. When the operating handles are pushed backward, the chassis moves backward. The opening of the reversing valve can be adjusted by the operating handle to change the motor flow. The greater the valve opening, the greater the motor flow, the faster the motor speed, the higher the chassis speed. The smaller the valve opening, the smaller the motor flow, the slower the motor rotation speed, the slower the chassis speed. When the opening of the two control handles in the same direction is different to achieve the steering function. When the left and right control handles are reversed, the left and right tracks of the chassis will reverse the differential steering to achieve the in-situ steering. The self-locking device of the hydraulic drive system is composed of two one-way sequence valves. Two one-way sequence valves are two way self-locking. When the operating handle of the directional valve is in the middle position, the pressure oil of the pump output directly returns to the oil tank through the interior of the directional valve, and the two one-way sequence valves are in a self-locking state, the hydraulic motor is locked. In order to improve the safety and braking effect of the system, wheel side brakes are installed on the inner side of the drive wheels to achieve the effect of auxiliary braking on the chassis.

SELECTION OF MAIN COMPONENTS OF HYDRAULIC SYSTEM

The designed small multi-function agricultural hydraulic chassis is suitable in the hilly and mountainous areas operations, and can drive various suspended small agricultural machines and tools for farming, sowing, harvesting and other operations by the rear power output. The main technical parameters of the agricultural hydraulic chassis are shown in Table 1.

Table 1

Hydraulic crawler chassis vehicle technical parameters

Index	Unit	Parameter value
Chassis mass (m)	kg	420
Diesel engine power (Pw)	kw	13.4
Diesel engine revs (n)	r/min	1250
Drive wheel radius (r)	mm	175
Working pressure (P)	MPa	7
Operating speed (V)	km/h	0-6
Maximum climbing degree	°	20

The selection of hydraulic motor shall meet the requirements of maximum operating parameters during chassis operation. During the operation of hydraulic motor, the working pressure, speed range, operating torque, efficiency, installation conditions, etc. shall be considered. The determination of influencing factor coefficients of main technical parameters can be obtained to consult manual or refer to experience values *(Chen et al., 2020; Yang and Zhu, 2010)*.

Calculation of motor driving torque:

$$T_p = F_T \times r \tag{1}$$

In the formula: T_p—Motor torque, N.m; F_T—Traction force, N; r—Radius of driving wheel, mm.

$$\begin{cases} F_T = F_q - F_f \\ F_q \le F_\varphi \\ F_f = f \times G_s \end{cases} \tag{2}$$

F_q—Tangent driving force, N; F_f—Rolling resistance, N; F_φ—Adhesion, N; G_s- Chassis weight, N; f—Rolling resistance coefficient, (0.06~0.07).

$$\begin{cases} F_\varphi = \varphi \times G_\varphi \\ G_\varphi = G_s \end{cases} \tag{3}$$

G_φ—Adhesion weight, N; G_s—Chassis weight, N; φ—Adhesion coefficient, (0.9~1.1).

The motor torque can be obtained by solving equations (1), (2) and (3).

$$T_p \ge G_s(\varphi - f) \times r = 749.1\ \mathrm{N.m} \tag{4}$$

φ—Adhesion coefficient, (0.9~1.1); f—Rolling resistance coefficient, (0.06~0.07); r—Radius of driving wheel, mm.

The displacement of hydraulic motor, V_{gm}, is:

$$V_{gm} = \frac{2\pi T_g}{\Delta P \times \eta_{mm}} \geq 373.4 \text{ mL/r} \tag{5}$$

T_g—Driving torque of single motor, N.m; ΔP—Motor pressure difference, MPa; η_{mm}—mechanical efficiency of motor, (0.9~0.99).

According to the chassis speed, the motor speed is determined as:

$$n_m \geq \frac{V}{2\pi r} = \frac{1.67 \times 60}{0.175 \times 2 \times \pi} = 91.2 \text{ r/min} \tag{6}$$

In the formula: V—Maximum speed of chassis, m/s; r—Radius of driving wheel, mm.

According to the calculation results, the OMV500 radial piston quantitative motor is selected, the maximum working pressure is 16MPa, displacement is 518mL/r, and speed is 170 r/min.

Since the chassis adopts the pump control principle, two pumps are required to control two hydraulic motors respectively. The selection of hydraulic pump is mainly based on its maximum working pressure and flow (*Yang and Zhu, 2010; Liu et al., 2019)*. The maximum working pressure of the hydraulic pump is determined by the maximum working pressure of the hydraulic motor, namely:

$$P_P \geq P_1 + \sum \Delta P = 16.5 \text{ MPa} \tag{7}$$

In the formula: P_p—Maximum working pressure of hydraulic pump, MPa; P_1—Maximum working pressure of hydraulic motor, MPa; $\sum \Delta P$—Total pipeline loss from the outlet of hydraulic pump to the inlet of hydraulic motor, (0.2~0.50) MPa.

Maximum flow of the double-pump:

$$q_{vp} \geq K \sum q_{Vmax} = K \frac{V_{gm} \times n}{1000} = 44.27 \text{L/min} \tag{8}$$

In the formula: K—leakage coefficient of the system, (1.1~1.3); q_{Vmax}—Maximum flow of hydraulic motor, L/min; V_{gm}—Motor displacement, ml/r; n—Motor speed, r/min.

Based on theoretical calculation and comprehensive consideration, the CBZ2050/2050 BF plunger type double-pump is selected, the rated flow is 50 L/min.

The inner diameter of the hydraulic oil pipe is related to the pipe material and the speed of oil pressure in the pipe. When the flow rate is constant, the larger the inner diameter is, the slower the flow rate is, the less the pressure loss of the oil circuit is, and the smaller the inner diameter is, the faster the flow rate is, the larger the pressure loss is *(Yang and Zhu, 2010)*.

The calculation formula of tubing diameter is:

$$d \geq 4.61\sqrt{\frac{q}{V}} = 18.82 \tag{9}$$

In the formula: d—Inner diameter of oil pipe, mm; q—Oil pipe flow, L/min; V—flow rate of oil in oil pipe, (2.5-7.6) m/s.

According to theoretical calculation and comprehensive consideration, the inner diameter of hydraulic oil pipe is 20mm. The 34SH-L20H-W three position four-way directional valve is selected, and the nominal diameter is 20mm, the nominal pressure is 20MPa. The DBW10B2-5X/10-6EG24N9K4 relief valve is selected, and the nominal diameter is 10mm, the maximum set working pressure is 10MPa. The S10A1 tubular one-way valve is selected, and the opening pressure is 0.05MPa, the nominal pressure is 10MPa.

SIMULATION ANALYSIS OF HYDRAULIC SYSTEM

According to the overall scheme of hydraulic drive system of the chassis, the AMESim simulation software is used to establish the hydraulic drive system simulation model of chassis (*Liang and Su, 2015*), as shown in Figure 2.

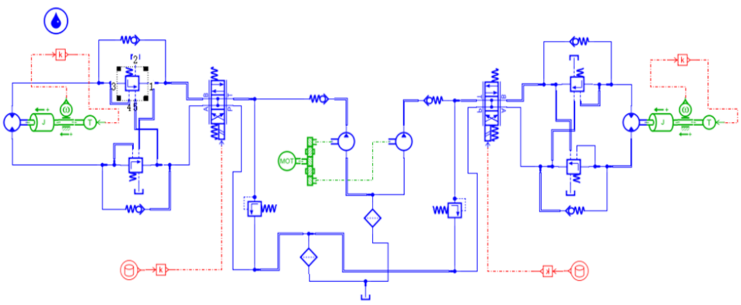
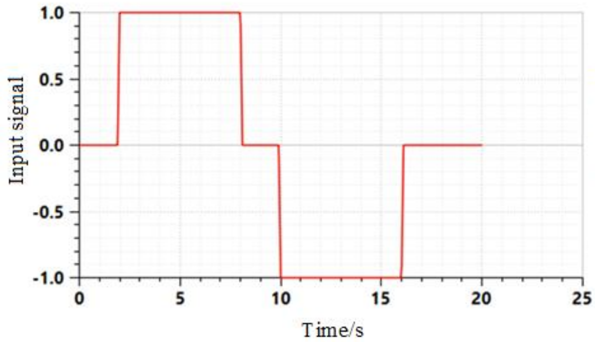


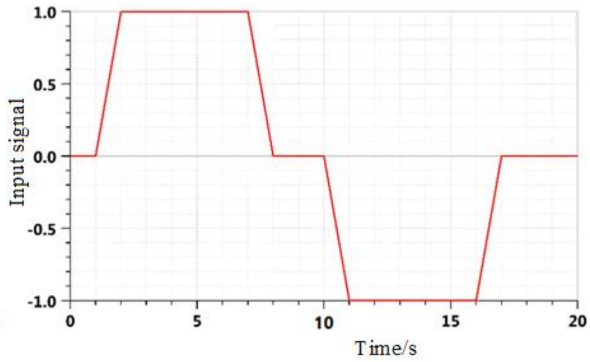
Fig. 2 - Hydraulic system simulation model

The basic performance parameters of the main components of the hydraulic drive system are set as follows: according to the system requirements, the engine power is 13.4kW, the speed is 1250r/min, the hydraulic pump flow is 50mL/r, the speed is 2000r/min, the motor displacement is 518 mL/r, the rated speed is 170r/min, and the overflow pressure of the overflow valve is 10MPa.

In the simulation process, the step signal and slope signal are used to simulate the control of the controller to the hydraulic drive system. The step signal simulates the operation of the quick start system. The signal is set as stop for 2s, forward instantaneous open, forward stable operation for 6s, instantaneous close, stop for 2s, reverse instantaneous open, reverse stable operation for 6s, instantaneous close, and stop operation. The control signal is shown in Figure 3 (a). The slope signal simulates the operation of the smoothly starts system. The signal is set as stop for 1s, forward direction starts to open at a constant speed, and reaches the maximum opening in 1s, forward stable operation for 5s, starts to close at a constant speed, and it is completely closed in 1s, stop for 2s, reverse direction starts to open at a constant speed, and increases to the maximum opening in 1s, reverse stable operation for 5s, starts to close at a constant speed, and it is completely closed in 1s, and stops the operation. The control signal is shown in Figure 3 (b).



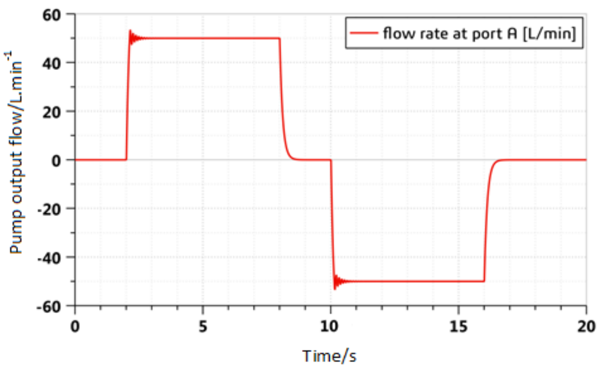
(a) Step signal



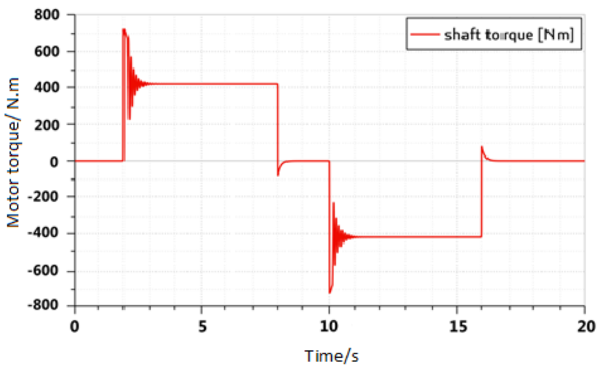
(b) Ramp signal

Fig. 3 - Hydraulic pump control signal

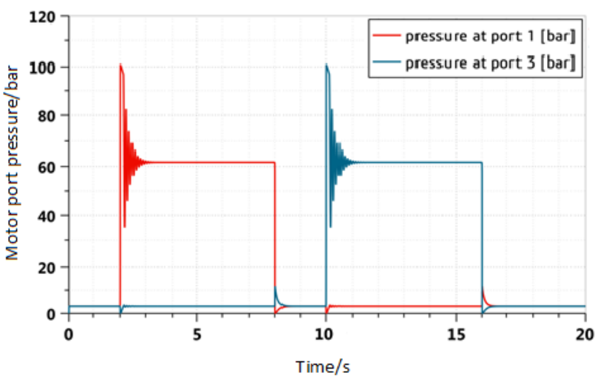
Under the step signal, the simulation operation of the hydraulic drive system is shown in Figure 4, and under the slope signal, the simulation operation of the hydraulic drive system is shown in Figure 5.



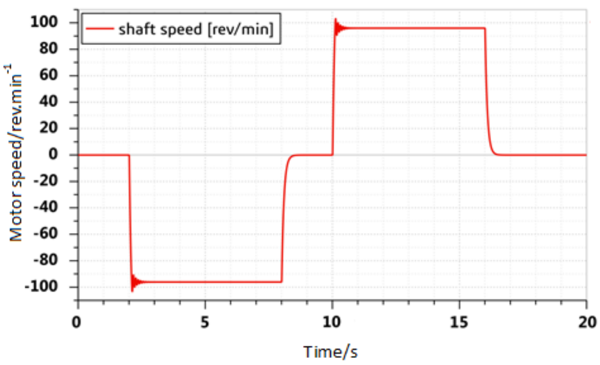
(a) Pump flow



(b) Motor torque



(c) Motor pressure

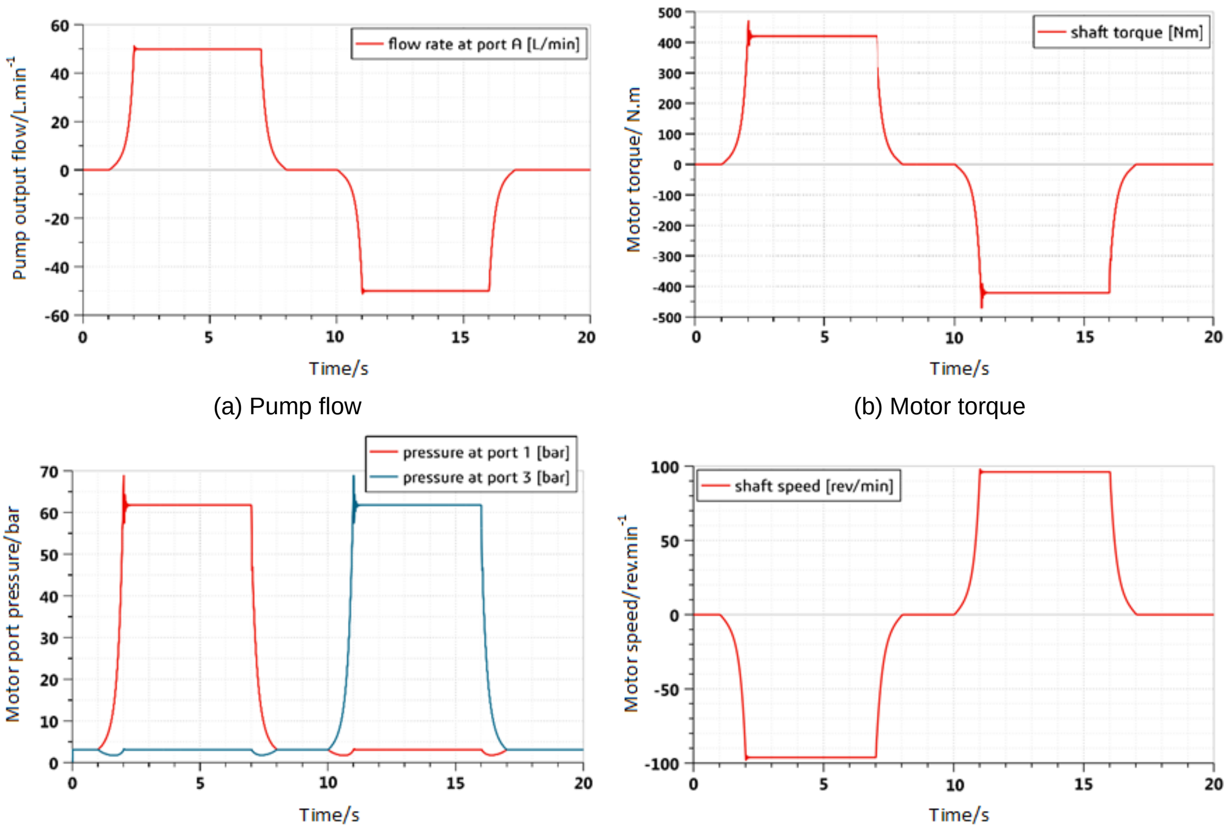


(d) Motor speed

Fig. 4 - Operating status of hydraulic components under Step signal

It can be known from the simulation results that: as shown in Figure 4, when instantaneous opening the hydraulic system, the output flow of the hydraulic pump is stabilized at about 50L/min after a short fluctuation. As the hydraulic oil fast flows into the hydraulic motor to generate the instantaneous pulse impact, the motor torque rises instantaneously to reach of about 700N, and it enters a stable state of about 440N.m after a short period oscillation. The port 1 of the motor is a forward oil inlet, and the port 3 is a reverse oil inlet. The pressure changes of the two oil ports are similar, but there is a difference in the opening time. Due to the impact of instantaneous opening, the pressure of the hydraulic system will reach 10MPa of maximum safe pressure of the system.

After a short oscillation period, it will enter a stable state, stabilized at the working pressure of about 6MPa. The motor speed rises instantaneously to the maximum speed about 100r/min, and reaches a stable speed about 96r/min after a short oscillation. It can be concluded that the chassis speed is of about 6.3 km/h at this time, which meets the maximum speed of the chassis of 6 km/h, and meets the design requirements. When the system is closed instantaneously, the output flow of the hydraulic pump is zero instantaneously, and the hydraulic motor stops with the less impact. The pressure at the two oil ports of the motor drops rapidly with a short and small pressure oscillation, and the hydraulic motor speed drops to 0 instantaneously.



(a) Pump flow (b) Motor torque

(c) Motor pressure (d) Motor speed

Fig. 5 - Operating status of hydraulic components under Ramp signal

As shown in Figure 5, when the steady opening the hydraulic system, the flow of the hydraulic pump can be smoothly transited, stabilized at about 50L/min, and there is no impact and oscillation in the whole process. The motor torque rises steadily, and then there is a small fluctuation, and then it is stabilized at about 400N.m. The pressure changes of motor oil port 1 and oil port 3 are similar, but there is a difference in the opening time. They all gradually and steadily rise to the peak value of about 7MPa. After a small pressure oscillation, they are stabilized at about 6MPa. The motor speed rises steadily at 96r/min. When the system is closed, the output flow of the hydraulic pump is 0 instantly, and the motor torque, the pressure of the two oil ports of the motor, and the motor speed are all steadily reduced to 0. In the actual operation process, when the controller is slowly pushed to the maximum opening, the hydraulic chassis starts smoothly with almost no vibration, and the chassis is in good working state.

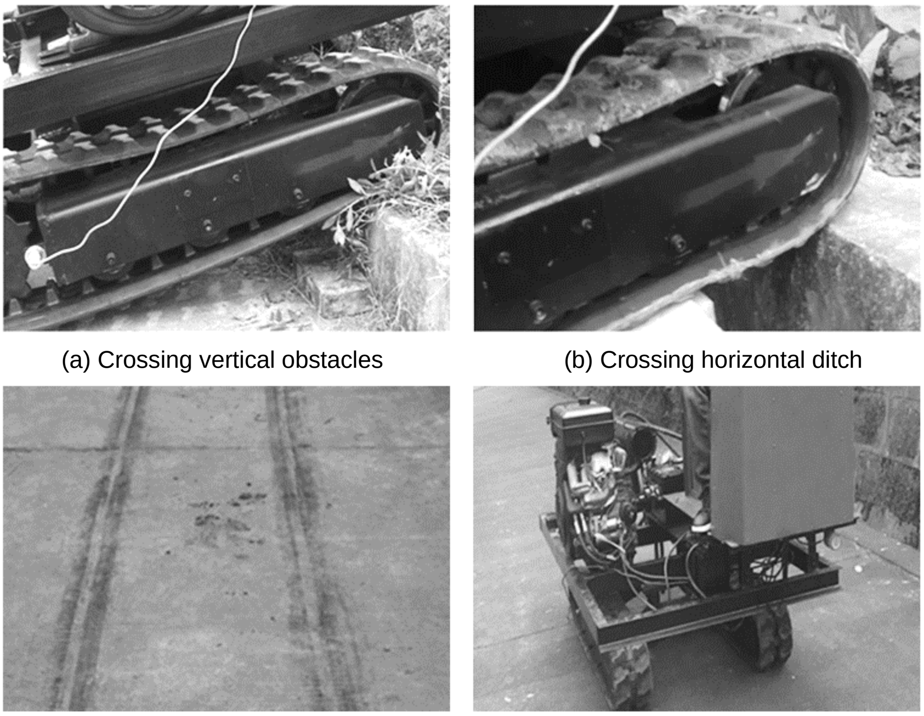
Materials and methods of performance test

According to GB/T 3871《Test Methods for Agricultural Wheeled and Tracked Tractors》, the operating performance of the chassis hydraulic drive system is tested through the chassis trafficability, high-speed driving straightness, climbing performance and steering performance.

The chassis trafficability is to test the performance of the chassis through vertical obstacles and horizontal trenches. The accelerator is fully opened, and the chassis is observed to climb over vertical obstacles and drive through horizontal trenches. Each condition is tested three times, one pass is considered as qualified, and data is recorded. When testing the straightness of high-speed driving, first make a reference line, then make a vertical line according to the reference line, and mark 100m from the starting point reference line as the end point. The chassis drive straight from the starting point to the end point at 5km/h constant speed, and the vertical deviation from the calibration line are measured. Tested 6 times, record the test data, and calculate the average value. The deviation of the chassis on the flat road cannot exceed 6m. As the operating gradient in hilly and mountainous areas is generally not more than 15°, the maximum gradient for the climbing performance test is 20°. The climbing performance test is carried out for 4 different gradients of 5°, 10°, 15° and 20°. At a stable climbing speed, each gradient is tested three times, and it is qualified if it passes two tests. During the steering performance test, test the steering effect in the field and on the hard ground. Push the left and right control levers in the opposite direction to drive the two hydraulic motors to rotate relatively, and test the effect of reverse differential in-situ steering.

RESULTS

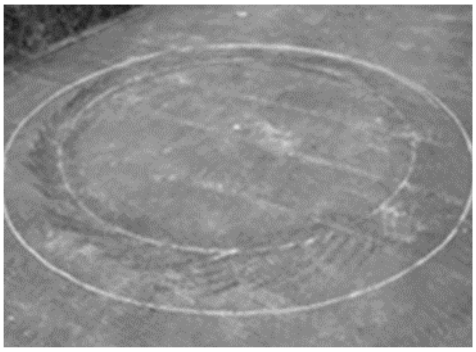
The chassis performance test states are shown in Figure 6. The chassis trafficability test effects are shown in Figure 6(a) and 6(b). When the chassis is in good working state, the maximum height of crossing vertical obstacles reaches 200mm, and the width of crossing ditches reaches 300 mm, which can better meet the requirements of the chassis trafficability. The driving straightness test results are shown in Figure 6 (c) and Table 2. The maximum deviation of the vehicle in straight line driving is 2.57m, and the average deviation is 2.06m. The deviation is within the allow range of high-speed driving straightness of the chassis, which meets the design requirements. The results of climbing performance test are shown in Figure 6 (d) and Table 3. The chassis has good climbing capacity and good trafficability, which is suitable for operation in hilly and mountainous areas and meets the design requirements. The steering of the hydraulic chassis is mainly to check whether the steering radius of the chassis meets the design requirements and whether it is suitable for steering in a narrow space. During the test, the chassis steers flexibly and can meet the needs of steering in chassis operation. The use of bilateral reverse differential steering can achieve arbitrary adjustment of the steering radius and in-situ steering. The steering effect is shown in Figure 5 (e) and (f).



(a) Crossing vertical obstacles (b) Crossing horizontal ditch

(c) Driving straightness (d) Climbing performance

(e) Steering performance in the field



(f) Steering performance on the road

Fig. 6 - Hydraulic chassis performance test

Table 2

Offset test results

NO.	Speed (km/h)	Offset (m)
1	5.0	1.54
2	5.0	2.57
3	5.0	1.78
4	5.0	2.33
5	5.0	1.96
6	5.0	2.15
Average offset (m)		2.06

Table 3

Test of climbing

NO.	Speed(km/h)	Slope(°)	Chassis trafficability
1	3.0	6	Fast and stable passing
2	3.0	10	Fast and stable passing
3	1.5	15	Slow and stable passing
4	1.5	20	Slow and stable passing

CONCLUSIONS

(1) On the basis of the research on the current situation of the agricultural hydraulic chassis in hilly and mountainous areas, the overall design of the hydraulic drive system of the chassis is carried out. The designed hydraulic drive system uses the controller to control the rotation speed of the motor to realize the variable speed driving of the chassis. At the same time, it can control the differential driving of the chassis tracks to achieve arbitrary adjustment of the chassis steering radius and in-situ steering. It effectively solves the existing problems in the adaptability, controllability, and stability of agricultural machinery in hilly and mountainous areas.

(2) According to the overall scheme of hydraulic drive system of the chassis, the AMESim simulation software is used to establish the hydraulic drive system simulation model of chassis. It can be known from the simulation results, when the instantaneous startup the hydraulic system, the system flow, motor pressure, torque and speed will have a transient impact, with large fluctuations, and can cause great damage to the system if it is in this working state for a long time, when the stable startup the hydraulic system, the system flow rises steadily with almost no impact, and the torque, pressure and speed of the motor increase steadily with little fluctuation, without impact and shock. The motor peak pressure is about 7Mpa, and it is stable at about 6Mpa after a small fluctuation. The chassis starts more stably with less vibration, which has a good protection effect on the hydraulic motor. In addition, when the system is stable under the control of two signals, the maximum flow of the hydraulic drive system is 50L/min, the motor torque is about 440N.m, the motor pressure is 6Mpa, and the motor speed is 96r/min, which is consistent with the working states of the hydraulic chassis in the actual operation process.

(3) The chassis performance tested results shown that the designed chassis has good trafficability, the maximum crossing height reaches 200mm and the crossing width reaches 300mm. The maximum deviation of the chassis is 2.57m and the average deviation is 2.06m when stably and high speed driving at 5km/h, which meets requirements of high-speed driving straightness. The chassis has good climbing performance, and can stably passing the 20° slope, adapting to the climbing performance of the chassis in hilly and mountainous areas. In addition, the chassis is stable in operation, flexible in steering, and can achieve in-situ steering. The operation effect can better meet the chassis operation requirements.

ACKNOWLEDGEMENTS

The study was supported by the Key Laboratory of Modern Agricultural Equipment, Ministry of Agriculture and Rural, P.R.China (HT20200702), and the Sichuan Innovation Team of Chinese National Modern Agricultural Industry Technology System (SCCXTD-2023-20).

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