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Research Article

Dynamic Characteristic Analysis and Clutch Engagement Test of HMCVT in the High-Power Tractor

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Hydromechanical continuously variable transmission (HMCVT) is capable of bearing large torque and has wide transmission range, which is suitable for high-power tractors. Dynamic characteristics could influence the tractor life, especially in a high-power tractor. Wet clutch is the crucial component in the HMCVT, which could smooth and soft power transmission. Therefore, it is important to study the dynamic characteristics and implement the wet clutch test of HMCVT. In this paper, AMESim is used to establish virtual models of gearbox, pump-controlled hydraulic motor system, and shifting hydraulic system. Then, a simulation study of tractor operation under working condition is carried out. The internal and external meshing forces of the planetary row are analyzed. Finally, the wet clutch engagement process of HMCVT in the high-power tractor is tested to verify the oil pressure. The simulation results show that the values of internal and external meshing force become larger as the throttle opening increases. At the moment of shifting change, the meshing forces of the planetary gear have great impact. The clutch test shows that the trend of the oil filling curve obtained from the bench test is similar to that obtained from the theoretical curve, which verifies the simulation results.

1. Introduction

The main function of the tractor is to be used in conjunction with various traction and driving machines to complete agricultural field operations, earthwork engineering operations, transportation operations, and stationary operations [1]. The transmission performance of the gearbox will have an important impact on the tractor and its transmission system. The hydromechanical continuously variable transmission (HMCVT) has attracted extensive attention in recent years [2–5]. In order to meet the requirements of tractors working under multiple working conditions, at present, high-power tractor gearboxes are set with more gears. The increase of gears in gearboxes not only makes its structure complex and error prone to operate, but also the dynamic characteristics of the tractor are difficult to guarantee [6].

Scholars have carried out a lot of research on HMCVT. The German company ZF [7] produced the S-Matic series of hydraulic mechanical transmissions, which used dual-row planetary row confluence to output power for the first time. The Japanese Company Komatsu [8] successfully developed a stepless speed change device suitable for construction machinery and applied it to the D155AX-3 bulldozer and WA380-3 loader. In recent years, various well-known gearbox manufacturers have launched HMCVT with independent intellectual property rights, such as John Deere, Caterpillar of the United States, and

Valtra of Finland [9]. The typical HMCVT model is shown in Figure 1.

Regarding mechanical characteristic analysis for HMCVT, Xia et al. [10] proposed an optimized design method for the selection of the structural parameters of the PCHMCVT to ensure competent overall performance. Zheng and Sun [11] used AMESim to establish a physical model of an HMCVT and simulated and analyzed the speed regulation characteristics, output torque characteristics, and acceleration characteristics of the pure hydraulic section of it. Zhang [12] and He et al. [13] have, respectively, optimized the design of the hydraulic mechanical stepless gearbox. Li et al. [14] analyzed the working principle of HMCVT and successfully developed a hydraulic mechanical stepless gearbox suitable for high-power tractors for both water and drought. Zhai et al. [15] introduced a two-stage planetary row device for the final transmission of a wheeled tractor, which is compact in structure, reliable in operation, and easy to install. Wang et al. [16] proposed a new type of compound planetary row transmission new hydraulic mechanical continuously variable transmission scheme based on the traditional single-row planetary and double planetary row. Guo et al. [17] established the conditions for synchronous shifting of multistage HMCVT and verified the high efficiency characteristics of it. Wang et al. [18] used the basic principles of dynamics to establish a tractor dynamic model and verified its good dynamic characteristics under ploughing condition. Xiao et al. [19, 20] studied the dynamic characteristics of the hydraulic circuit in HMCVT, and a fast system identification method was proposed. Cheng et al. [21–26] improved the genetic algorithm to implement the HMCVT parameter optimization, and the new proposed model can improve the HMCVT transmission efficiently and flexibly.

From the analysis above, we can see that scholars' research on HMCVT is focused on the structure characteristics or the shifting strategy, but there is little research on the dynamic characteristics and clutch engagement. In this paper, the HMCVT scheme and simulation model was proposed, the dynamic meshing forces were calculated, and the wet clutch engagement characteristics were tested for a high-power tractor.

2. Modeling

2.1. Gearbox Dynamic Model. This paper puts forward the HMCVT scheme, which is shown in Figure 2. This transmission scheme is a constant-ratio split-moment converging speed type. The engine power is divided into two power transmissions through the fixed-axis gear pair i_1 or $i_2 i_3$ ($i_2 i_3$ works in forward gear) and the hydraulic power distribution gear pair i_p. One power is transmitted to the common sun gear shafts of planetary rows K_1 and K_2 through the variable displacement pump-fix displacement motor, which is a hydraulic flow, and the other power is transmitted to the planet carrier K_1 and the ring gear K_2 through the fixed shaft gear pair (the planet carrier of K_1 and the gear ring of K_2 are firmly connected), which is the mechanical flow. The hydraulic flow and mechanical flow converge in the planetary rows K_1 and K_2, and then the combined flow force is transmitted to the ring gear of K_1 or planet carrier K_2. Finally, by separately controlling the engagement of wet clutches C_1, C_2, and C_3, the power can be transmitted to the output shaft. In this process, stepless speed regulation can be realized in each section by controlling the displacement ratio of the variable pump.

AMESim has been widely used in aerospace, vehicles, construction machinery, ships, and other multidisciplinary fields and has become a platform for modeling and simulation of complex systems such as liquid, mechanical, electrical, electromagnetic, thermal analysis, and control. AMESim software is more and more widely used in modeling, simulation, and analysis of complex systems. AMESim is applied to the modeling process of HMCVT in this paper.

The planetary gear mechanism model is shown in Figure 3(a), and its mathematical model is shown as follows [27]:

$$\begin{cases} v_{1a} = -\dfrac{\text{Rad1} + \text{Rad2}}{\text{Rad1}} \cdot v_{ca} - \dfrac{\text{Rad2}}{\text{Rad1}} \cdot v_{2a}, \\[2ex] v_0 = \dfrac{\text{Rad2} \cdot v_{2b} - \text{Rad1} \cdot v_{1a}}{\text{Rad2} - \text{Rad1}} + v_{ca}, \\[2ex] T_{2b} = \text{fact}_{\text{eff}} \times \dfrac{\text{Rad2}}{\text{Rad1}} \times \left(T_{1b} - T_{1a}\right) + T_{2b}, \\[2ex] T_{ca} = \left(T_{1b} - T_{1a}\right) + \left(T_{2b} - T_{2b}\right) + T_{cb}, \end{cases} \tag{1}$$

where v_{1a} is the rotational speed of Port 1A, v_{ca} is the rotational speed of Port 2A, v_{2b} is the rotational speed of Port 4B, v_0 is the rotational speed of the carrier, Rad_1 is the number of the sun gear, Rad_2 is the number of the ring gear, T_{2a} and T_{2b} are the input and output torque of the ring gear, and T_{ca} and T_{cb} are the input and output torque of the carrier.

The variable pump model (PU003 C) is selected from the Hydraulic library, as shown in Figure 3(b). The opening range of the variable pump model is $-1 \leq$ swash ≤ 1. When turning forward, port 3 is the oil outlet; when turning backward, port 3 is the oil inlet. The mathematical model of a variable pump is shown as follows:

$$\begin{cases} q_p = \dfrac{e \cdot v_p \cdot \text{swash}}{1000}, \\[2ex] T_p = \dfrac{\left(p_{\text{AP}} - p_{\text{BP}}\right) \cdot e \cdot \text{swash}}{20 \cdot \pi}, \end{cases} \tag{2}$$

where q_p is the flow of the variable pump, e is the displacement of the pump, swash is the throttle opening of the pump, T_p is the shaft torque of the pump, and p_{AP} and p_{BP} are the inlet and outlet oil pressure.

The quantitative motor model (M0001 C) is selected from the Hydraulic library, and the mathematical model is shown as follows:

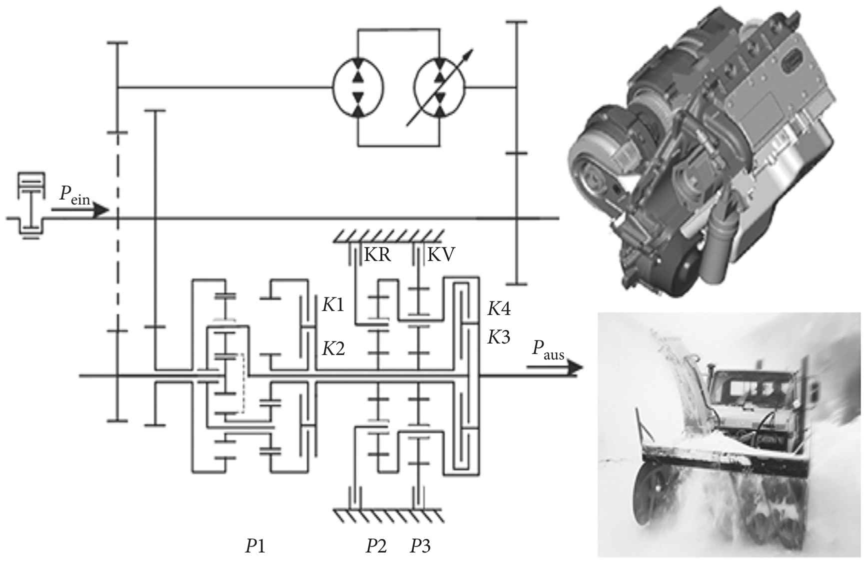


Figure 1: The HMCVT S-Matic.

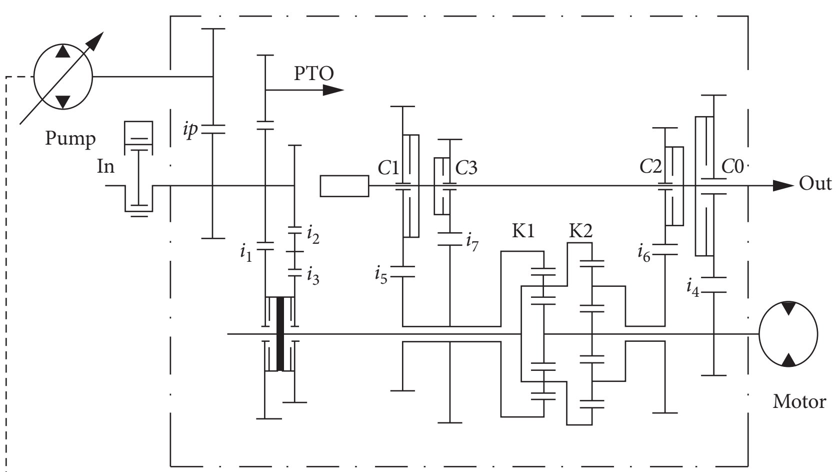


Figure 2: HMCVT scheme.

$$\begin{cases} q_m = \dfrac{e \cdot v_m}{1000}, \\ \\ T_m = \dfrac{(p_{Am} - p_{Bm}) \cdot e \cdot \text{swash}}{20 \cdot \pi}, \end{cases} \tag{3}$$

where q_m is the flow of motor, v_m is the motor speed, T_m is the torque of the motor, and p_{Am} and p_{Bm} are the inlet and outlet oil pressure of the motor.

Based on the models of various parts, like the wet clutch, the oil circuit, the corresponding signal control elements, and the torque output elements, the simulation model of the shifting hydraulic system could be obtained, as shown in Figure 3(c).

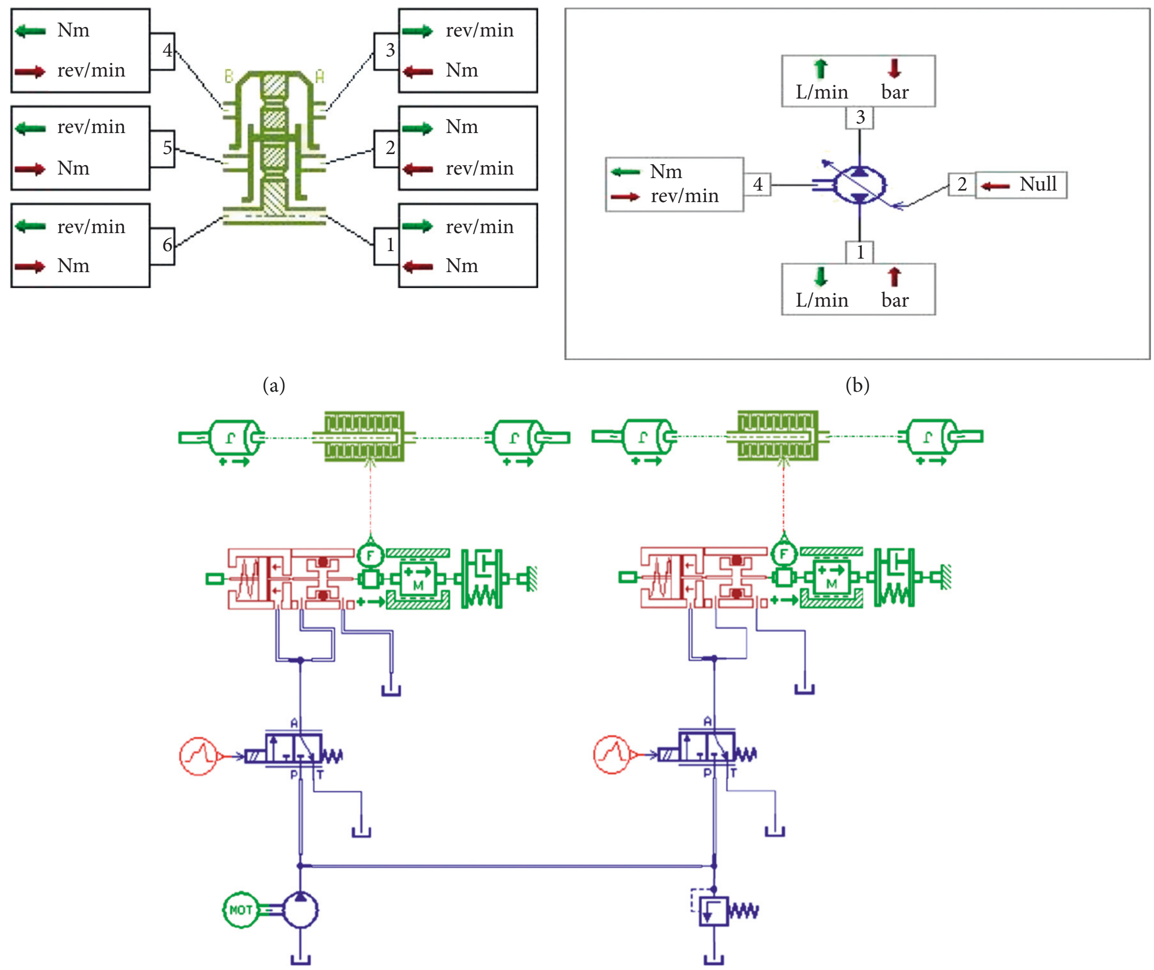
According to the gearbox model, hydrostatic circuit, shifting system model, and tractor model, the simulation model of HMCVT is obtained, as shown in Figure 4.

3. Simulation Analysis of Tractor under Working Conditions

Figure 5 and Table 1 show the change of the internal and external meshing forces of the planetary row under working condition.

As can be seen from Figures 5(a) and 5(b), the internal and external meshing forces of the planetary row both have great impacts at the moment of the engine throttle opening change. The maximum and minimum values of internal meshing force are 3.33×10^6 N and 2.00×10^7 N, respectively. The maximum and minimum values of external meshing force are 2.20×10^7 N and 7.33×10^6 N, respectively.

They all become larger as the throttle opening increases. At the moment of shifting change, the internal and external meshing forces of the planetary gear have great impact at 2.5s–3.5 s. Then, the magnitudes of the meshing force change decrease, and the trend tends to be stable.



(a) (b)

(c)

Figure 3: The model of the HMCVT component. (a) Planet gear. (b) Variable pump. (c) Shifting hydraulic system.

4. Clutch Engagement Characteristic Test of HMCVT

4.1. Power Drive Module. DEUTZ TCD2013L062 V engine was selected for the test. The accelerator control system of it is a dual-sensor control loop system, which is mainly composed of displacement sensor, current sensor, accelerator pedal, and stepping motor. The test sent an electric signal of speed adjustment to the engine by changing the pedal angle of the accelerator pedal. After the ECU got the control signal and then the stepper motor could be controlled by output current, the throttle opening can be changed.

The engine throttle opening was collected by displacement sensors, and ECU output current was collected by current sensors. The signals were compared with the original value in the input controller and realized the rapid adjustment of the engine speed controller according to the control algorithm.

This HMCVT test bench selected and designed the accelerator pedal according to the engine type and ECU, as is shown in Figure 6.

The bolt adjusting device was installed on the accelerator pedal, and the length of it was related to the change of the position of the accelerator pedal. The corresponding scale line was drawn on the bolt in direct proportion to the speed of the engine.

4.2. HMCVT Load Module. The dynamometer loaded the gearbox by outputting different quantities and can simulate the working state of the vehicle in different working conditions. The test bench selected the Lanlin DW250 (Figure 7) type eddy current dynamometer to match the rotation speed and power of the engine.

4.3. Oil Pressure Sensor. The JM-801 pressure sensor with a ceramic core was selected for the test bench. When the pressure was applied to the ceramic membrane, the surface of the membrane will undergo subtle deformation.

The resistance was printed on the back of the ceramic diaphragm, and the Wheatstone bridge can be formed. The Wheatstone bridge will generate a voltage signal due to voltage sensitive effect. The signal is proportional to the pressure and excitation current. The oil pressure value of C0,

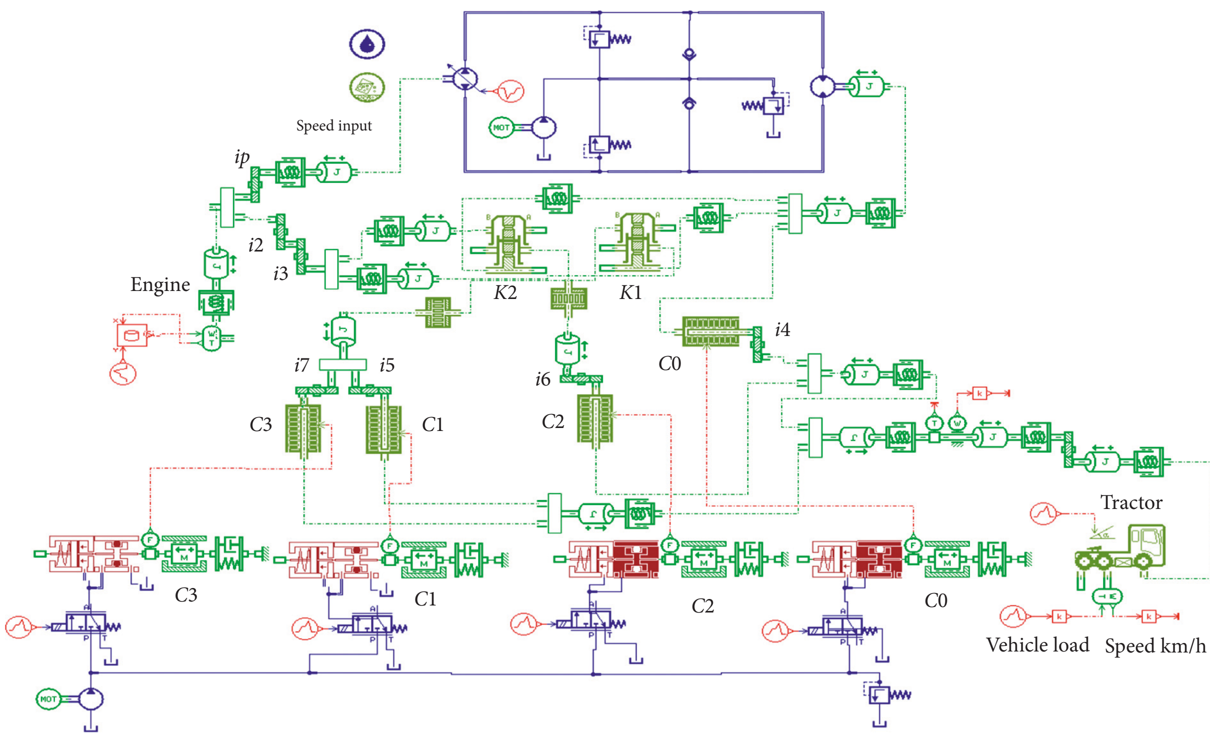
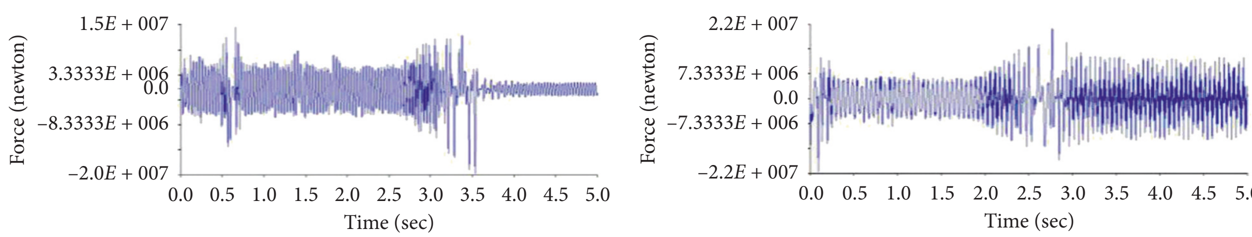


Figure 4: The simulation model of HMCVT.



(a) (b)

Figure 5: Meshing force of the planet gear. (a) Internal meshing force and (b) external meshing force.

Table 1: Internal and external meshing force in working condition.

Working condition	External meshing force (N)		Internal meshing force (N)	
	Maximum	Minimum	Maximum	Minimum
—	2.20×10^7	7.33×10^6	2.00×10^7	3.33×10^6

C1, C2, and lubrication main oil line in the platform can be read directly through the pressure indicator table (Figure 8).

4.4. Test Bench. The overall structure diagram of the test bench is shown in Figure 9. The test bench was mainly composed of diesel engine, speed rising transmission, and support device. In the test bench, the diesel engine was used as the power source.

4.5. Clutch Oil Filling Test. The structure of C0, C1, and C2 was basically the same, and this paper took C0 as an example to carry out the bench test research on oil filling process. Test steps were as follows.

First, the engine was started to adjust the accelerator pedal and stabilize the output speed of the engine to 1295 r/min.

Secondly, the engagement button of C0 was pressed in the measurement platform. The oil pressure of the main oil relief valve was set at 2.8 MPa.

After receiving the electrical signal, the proportional directional valve of C0 pumped the filtered oil from the tank and sent to the clutch. The oil filling process of C0 is shown in Figure 10.

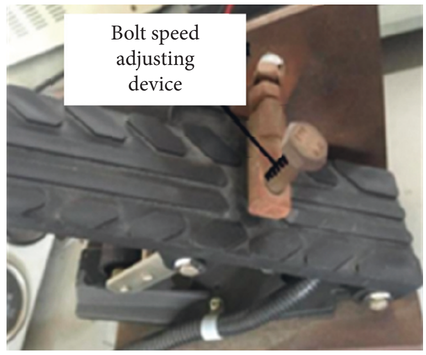


FIGURE 6: Accelerator pedal of the engine.

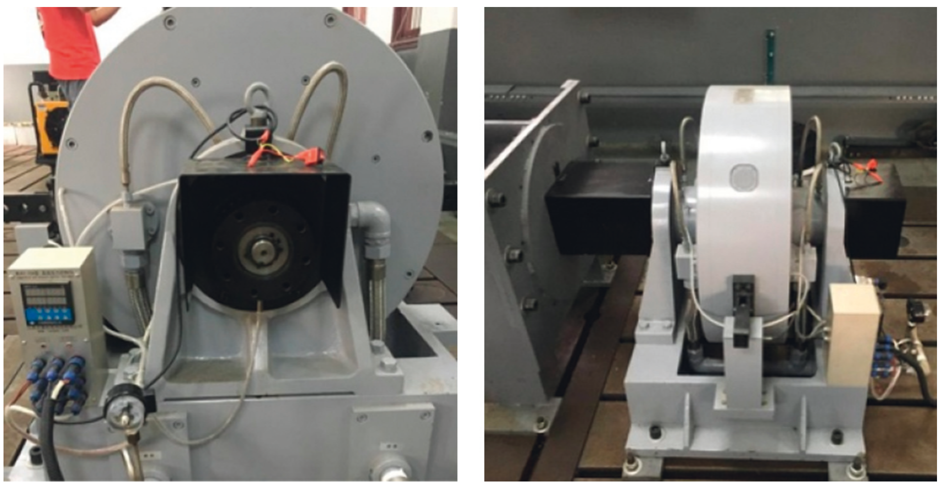


FIGURE 7: Eddy current dynamometer.

FIGURE 8: Oil pressure gauge.

4.5.1. 0-a Section. This section was mainly to eliminate the gap between the friction plates. The oil pressure raised from 0 MPa to 0.03 MPa rapidly. The slope of the variation of the curve in this section is 0.6.

4.5.2. a-b Section. The slope of the variation of the curve in this section is 0.52. At this stage, the piston compressed the spring under the action of the oil pressure, and the oil pressure increased with compression of the return spring. The oil pressure pf C0 rose slowly from 0.03 MPa to 0.42 MPa. The variation of oil pressure was related to the torque transferred by the clutch and piston stroke.

4.5.3. b-c Section. The slope of the variation of the curve in this section is 7.8. In this stage, the oil pressure of C0 rose rapidly from 0.42 MPa to 2.8 MPa, which was set by the relief valve in order to ensure the torque.

4.5.4. c-d Section. This section was the pressure holding stage. The oil pressure of C0 was maintained at 2.8 MPa.

Figure 9: The HMCVT test bench.

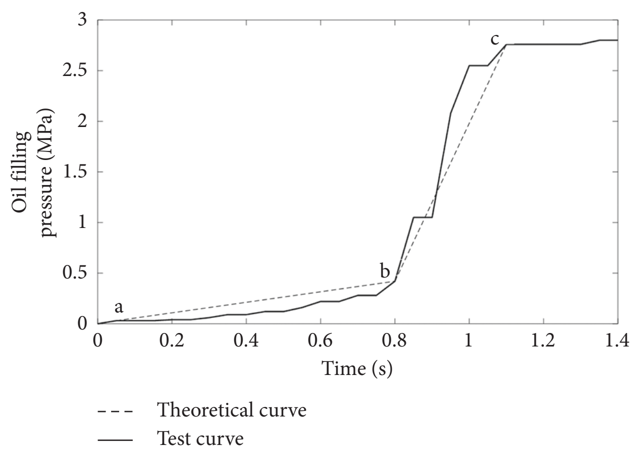


Figure 10: Oil filling characteristic curve of the wet clutch.

4.6. Test Discussion. The oil filling process can be divided according to the slope of the oil pressure change in each section. The oil filling time was short at 0–0.5 s, and the range of oil pressure change is 0.03 MPa. This section was the stage of rapid oil filling.

The oil filling duration of 0.05–0.8 s was 0.75 s, accounting for 68% of the total oil filling time. And, the variation range of oil pressure was 0.39 MPa, accounting for 14% of the set oil filling pressure. Therefore, this stage was the slow booster stage.

The oil filling duration of 0.8–1.1 s was 0.3 s, accounting for 27% of the total oil filling time. And, the variation range of oil pressure was 2.38 MPa, accounting for 85% of the set oil filling pressure. In a short time, the oil pressure showed a step rise. Therefore, the stage was the step booster stage.

During 1.1–1.4 s, the oil filling pressure reached the set value of 2.8 MPa and remained as such. Therefore, the stage was the pressure holding stage.

According to analysis, the oil filling curve obtained from the test bench had a similar trend to the theoretical curve. It verified the oil filling characteristics of the wet clutch and proved the accuracy of the HMCVT test bench.

5. Conclusions

In this paper, the dynamic characteristics of HMCVT were analyzed. The wet clutch engagement characteristics were tested. The analysis could be reference for the design of a high-power tractor. The conclusions can be made as follows:

(1) The values of internal and external meshing force become larger as the throttle opening increases.

(2) At the moment of shifting change, the meshing forces of the planetary gear have great impact.

(3) The magnitudes of the meshing force change decrease, and the trend tends to be stable after shifting change.

(4) The clutch test shows that the trend of oil filling is similar to the theoretical curve. The simulation model is reliable.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

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References

[1] S. Y. Li, "Classification of tractor and performance analysis," *Agricultural Science & Technology and Equipment*, vol. 9, pp. 27–29, 2016.
[2] R. Antonio and M. Alarico, "Multi-objective optimization of hydro-mechanical power split transmissions," *Mechanism and Machine Theory*, vol. 62, pp. 112–128, 2013.
[3] M. Alarico and R. Antonio, "Fuel consumption reduction in urban buses by using power split transmissions," *Energy Conversion and Management*, vol. 71, pp. 159–171, 2013.
[4] R. Pfiffner, L. Guzzella, and C. H. Onder, "Fuel-optimal control of CVT powertrains," *Control Engineering Practice*, vol. 11, no. 3, pp. 329–336, 2003.
[5] Z. Farkas and G. Kerényi, "Power flows and efficiency analysis of out- and input coupled IVT," *Periodica Polytechnica Mechanical Engineering*, vol. 53, no. 2, pp. 61–68, 2009.
[6] G. M. Wang, H. Zhang, Y. X. Zhai et al., "Economic analysis of tractor steel belt power split continuously variable transmission," *Journal of Agricultural Mechanization Research*, vol. 42, no. 01, pp. 241–248, 2020.
[7] J. Fereydon, H. Tung-ming, and S. Keith, "Dynamic simulation modeling for heavy duty automatic transmission control development," *SAE International Journal of Engines*, vol. 101, pp. 704–715, 1992.
[8] N. L. Feng, *Research on Dynamic Characteristics Simulation and Fuzzy Control of Vehicle Comprehensive Transmission Shift*, Beijing Institute of Technology, Beijing, China, 2001.
[9] Q. F. Wu, "Brief introduction of the technology development of continuously variable tractors of major foreign companies," *Farm Machinery*, vol. 25, pp. 90–94, 2012.
[10] Y. Xia, D. Sun, D. Qin, and X. Zhou, "Optimisation of the power-cycle hydro-mechanical parameters in a continuously variable transmission designed for agricultural tractors," *Biosystems Engineering*, vol. 193, pp. 12–24, 2020.
[11] X. Z. Zheng and W. Sun, "Modeling and simulation of hydro-mechanical continuously variable transmission based on AMESim," *Modern Machinery*, vol. 5, pp. 31–34, 2017.
[12] H. J. Zhang, C. F. Wang, M. H. Xiao et al., "The design and transmission characteristic analysis of intermediate shaft type hydraulic-mechanical CVT," *Journal of Machine Design*, vol. 32, pp. 44–49, 2015.
[13] C. K. He, P. F. Lang, M. Kang et al., "Transmission design and force analysis of HMCVT for high power tractor," *Journal of Mechanical Transmission*, vol. 42, pp. 54–59, 2018.
[14] J. L. Li, L. T. Liu, M. H. Xiao et al., "Research on dynamic characteristics of hydro-mechanical continuously variable transmission," *Journal of Mechanical Strength*, vol. 39, pp. 14–19, 2017.
[15] Y. Zhai, J. W. Zhang, C. Ju et al., "A two-stage planet device for wheeled tractor final transmission," *Tractor & Farm Transporter*, vol. 43, pp. 65-66, 2016.
[16] T. T. Wang, M. Z. Zhang, and M. M. Cui, "Transmission characteristics of a new hydraulic mechanical continuously variable transmission," *Journal of Natural Science of Heilongjiang University*, vol. 34, pp. 724–731, 2017.
[17] R. Guo, M. Z. Zhang, C. Chen et al., "Characteristics analysis of the multi-range hydro-mechanical continuously variable transmission in tractors," *Journal of Chinese Agricultural Mechanization*, vol. 35, pp. 118–121, 2014.
[18] Q. S. Wang, M. Z. Zhang, D. Y. Bai et al., "Dynamic simulation of tractor equipped with multi-range hydro-mechanical CVT during ploughing," *Journal of Agricultural Mechanization Research*, vol. 39, pp. 232–236, 2017.
[19] M. Xiao, Y. J. Zhao, and J. F. KangYang, "Research on system identification based on hydraulic pump-motor of HMCVT," *Engineering in Agriculture, Environment and Food*, vol. 12, no. 4, pp. 420–426, 2019.
[20] M. Zhang, F. Zou, H. J. Zhang et al., "Vibration study of the gearbox of hydro-mechanical continuously variable transmission based on Mean Filter Method," *International Agricultural Engineering Journal*, vol. 26, pp. 82–87, 2017.
[21] Z Cheng, Z. X. Lu, and J. Qian, "A new non-geometric transmission parameter optimization design method for HMCVT based on improved GA and maximum transmission efficiency," *Computers and Electronics in Agriculture*, vol. 167, pp. 1–14, 2019.
[22] Z Cheng, Z. X. Lu, and F. Dai, "Research on HMCVT's efficiency model based on the improved SA algorithm," *Mathematical Problems in Engineering*, vol. 2019, Article ID 2856908, 10 pages, 2019.
[23] Z Cheng and Z. X. Lu, "A novel efficient feature dimensionality reduction method and its application in engineering," *Complexity*, vol. 2018, Article ID 2879640, 14 pages, 2018.
[24] Z. Cheng and Z. Lu, "Semi-empirical model for elastic tyre trafficability and methods for the rapid determination of its related parameters," *Biosystems Engineering*, vol. 174, pp. 204–218, 2018.
[25] Z. Cheng and Z. Lu, "Research on the PID control of the ESP system of tractor based on improved AFSA and improved SA," *Computers and Electronics in Agriculture*, vol. 148, pp. 142–147, 2018.
[26] Z. Cheng and Z. Lu, "Nonlinear research and efficient parameter identification of magic formula tire model," *Mathematical Problems in Engineering*, vol. 2017, Article ID 6924506, 9 pages, 2017.
[27] X. H. Li, *Design and Dynamic Characteristics Research on the Hydro-Mechanical Continuously Variable Transmission of Tractor*, Nanjing Agricultural University, Nanjing, China, 2017.