Dynamic Characteristics Analysis and Clutch Engagement Test of HMCVT in the High-power Tractor

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Hydro-mechanical 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 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 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 show that the trend of oil filling curve obtained from bench test is the similar as that obtained from 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 hydraulic mechanical continuously variable transmission (HMCVT) has attracted extensive attentions 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. 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 Fig.1.

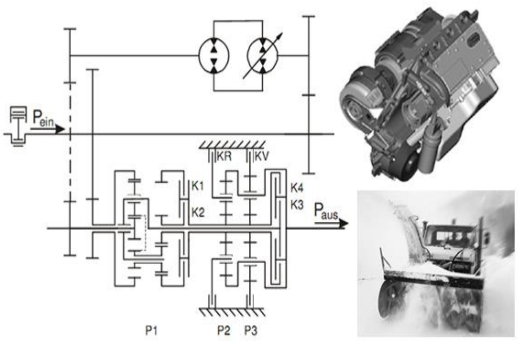


FIGURE 1: The HMCVT S-Matic.

Regarding mechanical characteristics analysis for HMCVT, Xia Y [10] proposed an optimised design method for the selection of the structural parameters of the PCHMCVT to ensure competent overall performance. Zheng X Z [11] used AMEsim to establish a physical model of a HMCVT, and simulated and analyzed the speed regulation characteristics, output torque characteristics and acceleration characteristics of the pure hydraulic section of it. Zhang H J [12] and He C K [13] have respectively optimized the design of the hydraulic mechanical stepless gearbox. Li J L [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 Y [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 T T [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 R [17] established the conditions for synchronous shifting of multi-stage HMCVT, and verified the high efficiency characteristics ~~of it~~. Wang Q S [18] used the basic principles of dynamics to establish a tractor dynamics model and verified its good dynamic characteristics under ploughing condition. Xiao M H [19-20] studied the dynamic characteristics of the hydraulic circuit in HMCVT, and a fast system identification method was proposed. Cheng Z [21-26] improved the genetic algorithm to implement the HMCVT parameters 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 high-power tractor.

2. Modeling

2.1. Gearbox dynamic model

This paper puts forward with the HMCVT scheme, which is shown in Fig.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 row K_1 and K_2, and then the combined flow force is transmitted to 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.

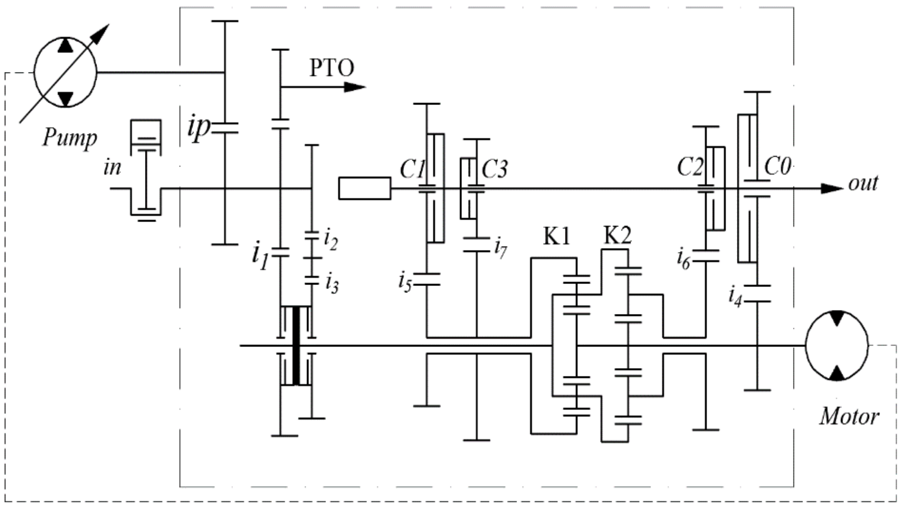


FIGURE 2: HMCVT scheme.

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 Fig.3(a), and its mathematical model is shown as follows [27]:

$$\begin{cases} v_{1a} = -\dfrac{Rad1 + Rad2}{Rad1} \cdot v_{ca} - \dfrac{Rad2}{Rad1} \cdot v_{2b} \\ v_0 = \dfrac{Rad2 \cdot v_{2b} - Rad1 \cdot v_{1a}}{Rad2 - Rad1} + v_{ca} \\ T_{2b} = fact_{eff} \times \dfrac{Rad2}{Rad1} \times (T_{1b} - T_{1a}) + T_{2a} \\ T_{ca} = (T_{1b} - T_{1a}) + (T_{2b} - T_{2a}) + T_{cb} \end{cases} \tag{1}$$

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

The variable pump model [PU003C] is selected from the Hydraulic library, as shown in Fig.3(b). The opening range of the variable pump model is -1≤ swash ≤1. When turning forward, port 3 is the oil outlet; when turning backward, port 3 is the oil inlet. The mathematical model of variable pump is shown as follows:

$$\begin{cases} q_p = \dfrac{e \cdot v_p \cdot swash}{1000} \\ T_P = \dfrac{(p_{AP} - p_{BP}) \cdot e \cdot swash}{20 \cdot \pi} \end{cases} \tag{2}$$

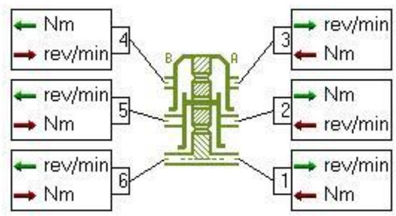
where q_p is the flow of the variable pump, e is displacement of the pump, swash is the throttle opening of pump, T_p is the shaft torque of the pump, p_{AP} and p_{Bp} are the inlet and outlet oil pressure.

The quantitative motor model [M0001C] is selected from the Hydraulic library, the mathematical model is shown as follows:

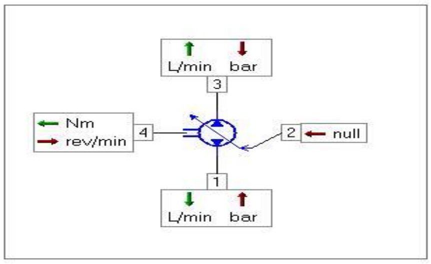
$$\begin{cases} q_m = \dfrac{e \cdot v_m}{1000} \\ T_m = \dfrac{(p_{Am} - p_{Bm}) \cdot e}{20 \cdot \pi} \end{cases} \quad (3)$$

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

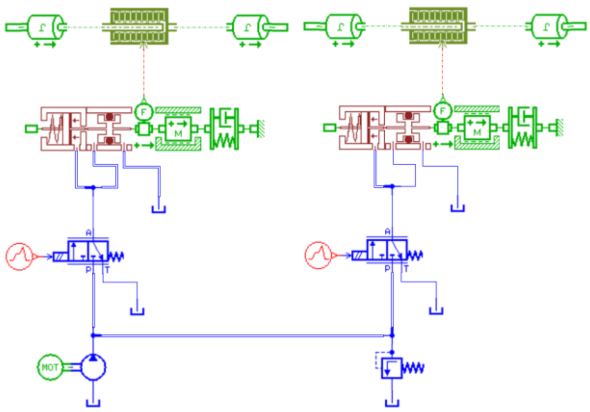
Based on the models of various parts, like the wet clutch, the oil circuit, the corresponding signal control elements and torque output elements, the simulation model of the shifting hydraulic system could be obtained, as shown in fig.3(c).



(a) Planet gear



(b) Variable pump



(c) Shifting hydraulic system

FIGURE 3: The model of HMCVT component.

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

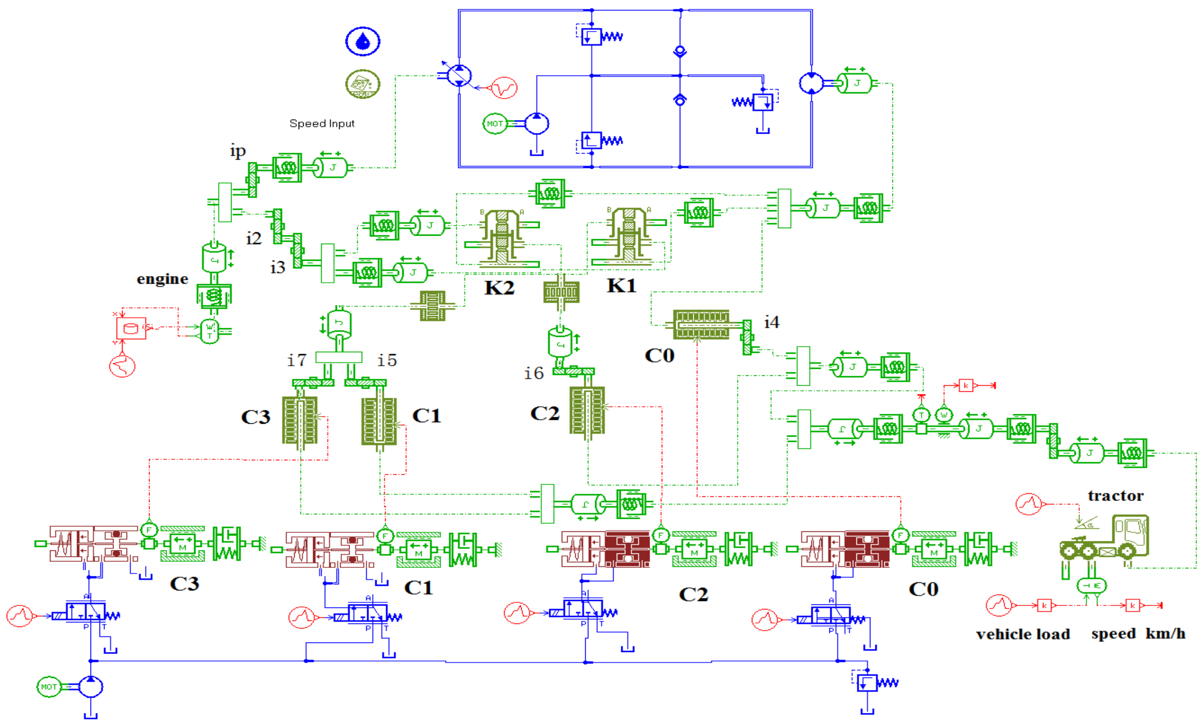


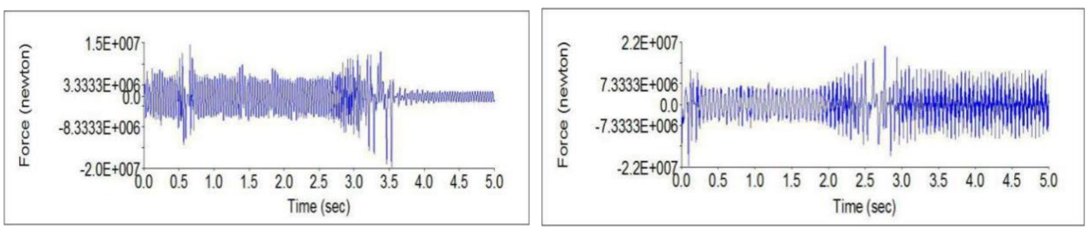
FIGURE 4: The simulation model of HMCVT

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 Fig.5(a) and Fig.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 value of internal meshing force are 3.33×10^{6}N and 2.00×10^{7}N, respectively. The maximum and minimum value 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~3.5s. Then the magnitudes of the meshing force change decrease and the trend tends to be stable.



(a) Internal meshing force (b) External meshing force

FIGURE 5: Meshing force of planet gear.

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}

4. Clutch engagement characteristics test of HMCVT

4.1 Power drive module

DEUTZ TCD2013L062V 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 output current to drives the stepper motor, so as to achieve the goal of changing the throttle opening.

The engine throttle opening was collected by displacement sensors and ECU output current was collected by current sensors. The signals were compared with original value in the input controller, and realized the rapid adjustment of 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 Fig.6.

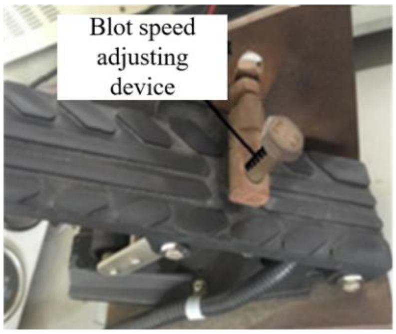


FIGURE 6: Accelerator pedal of engine

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 simulated the working state of the vehicle in different working conditions. The test bench selected Lanlin DW250 (Fig7) type eddy current dynamometer to match the rotation speed and power of the engine.

FIGURE 7: Eddy current dynamometer

4.3 Oil pressure sensor

The JM-801 pressure sensor with 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 voltage signal due to voltage sensitive effect. The signal is proportional to the pressure and excitation current. The oil pressure value of C0, C1, C2 and lubrication main oil line in the platform can be read directly through the pressure indicator table (Fig 8).

FIGURE 8: Oil pressure gauge

4.4 Test bench

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

FIGURE 9: The HMCVT test bench

4.5 Clutch Oil Filling Test

For the structure of C0, C1, C2 was basically the same, 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 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 clutch. The oil filling process of C0 is shown in Fig.10.

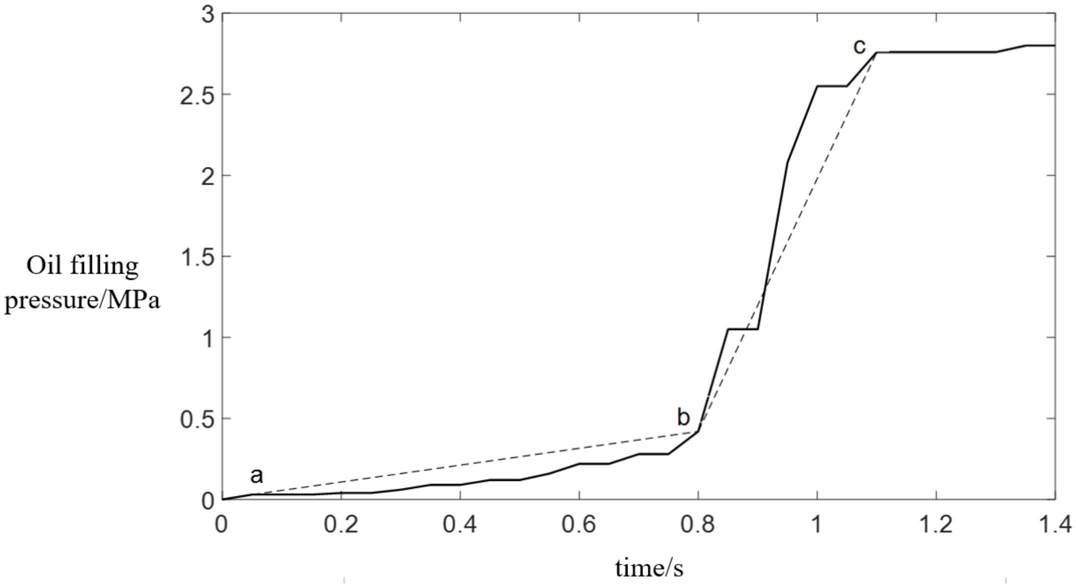


FIGURE 10: Oil filling characteristic curve of wet clutch

------: Theoretical curve ——— : Test curve

(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.

(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.

(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) c-d section

This section was the pressure holding stage. The oil pressure of C0 was maintained at 2.8MPa.

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 0-0.5s, the range of oil pressure change is 0.03 MPa. This section was the stage of rapid oil filling.

The oil filling duration of 0.05-0.8s was 0.75s, 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 slow booster stage.

The oil filling duration of 0.8-1.1s was 0.3s, 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 step booster stage.

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

According to analysis, the oil filling curve obtained from the test bench had same trend with theoretical curve. It verified the oil filling characteristics of 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 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 show that the trend of oil filling is the similar as theoretical curve. The simulation model is reliable.

Conflicts of Interest

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

Data Availability Statement

The data described in this paper could be fully provided by the authors.

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