Study on Shifting Quality of CVT Tractor under Hydraulic System Failure

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1. Introduction

Tractor's working conditions are complex, and it often needs more working gears. For example, Foton Lovol's P2654-N tractor and Changfa's CFK2204 tractor have 40 forward gears and 40 reverse gears. Excessive working gears make the transmission structure complex, therefore, CVT becomes an ideal tractor transmission system[1]. Since Fendt first equipped hydrostatic power split CVT in tractors in 1996, ZF, John Deer and other companies have successively developed their own tractor CVTs. Due to the use of multi-range speed control technology, the transmission still involves complex power shift process[2-3]. Once the hydraulic system fails, it may directly affect driving comfort and even endanger driving safety. In order to reveal the possible influence of various hydraulic system failures on the tractor driving, this paper studies the shift quality of the CVT tractor under the hydraulic system failure.

2. Experimental power train

The schematic diagram of the hydrostatic power split CVT and the scene photography of the experimental power train are shown in Fig. 1. The CVT covers tractor speeds up to 50 km/h, and four ranges R1-R4 (which are controlled by wet clutches C1~C4, respectively) are used to keep the hydrostatic power level low. The power train comprises a diesel engine (132.5 kW, 2200 r/min), two speed and torque sensors, a hydrostatic power split CVT, a magnetic powder brake, a clutch hydraulic control system, an electronic control unit and a remote PC.

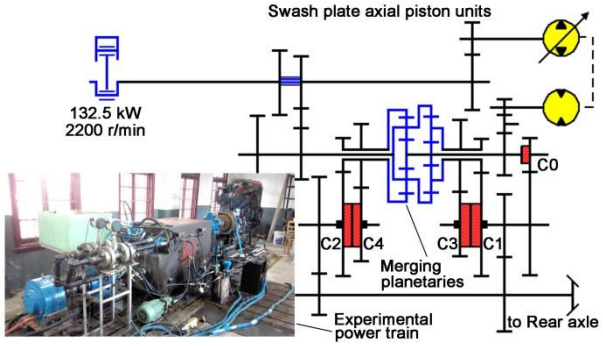


Fig.1 Schematic diagram of transmission and scene photography of experimental power train.

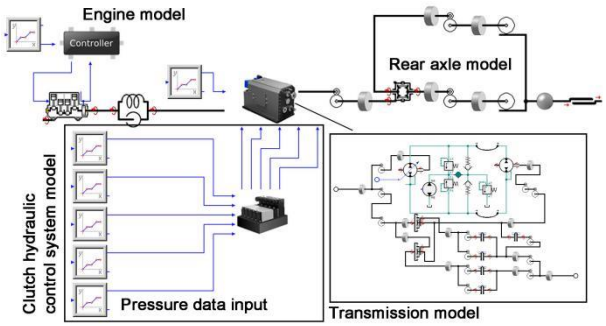


Fig.2 Simulation model of CVT tractor.

3. Modeling and experimental verification

The simulation model of the hydrostatic power split CVT and the tractor are shown in Fig. 2. The key of this model is to calculate the friction torque of the wet clutch during shift. The formula is as follows:

$$T_c = \frac{2\mu F_a n_p (r_o{}^3 - r_i{}^3)}{3(r_o{}^2 - r_i{}^2)} \quad (1)$$

Where, T_c is the friction torque, N·m; F_a is the total normal force, N; n_p is the number of clutch contact faces; r_o and r_i are the outside and inside radius of clutch plate, respectively, mm; u is the actual speed-dependent friction coefficient.

The formula for calculating F_a is as follows:

$$F_a = 0.1\lambda pA - k(x + x_0) \quad (2)$$

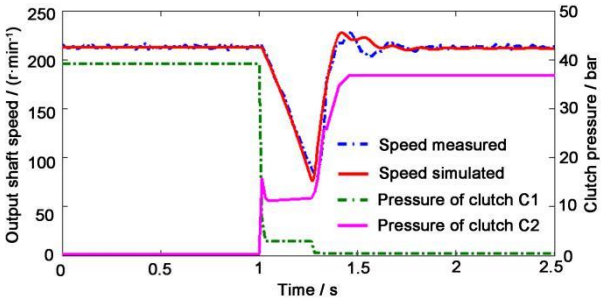
Where, $\lambda\epsilon(0,1]$ is the frictional correction coefficient, λ=1 only when the clutch piston stops moving; p is the oil pressure, bar; k is the stiffness of the return spring, N/mm; x and x_0 are the current and initial values of return spring compression, respectively, mm.

The formula for calculating u is as follows:

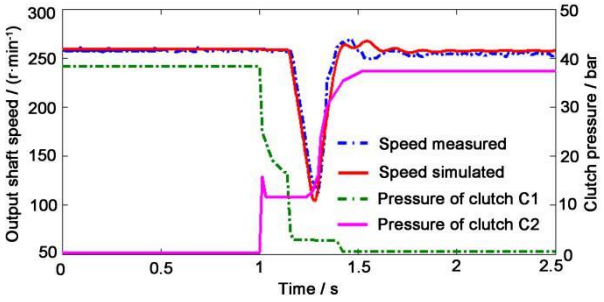
$$\mu = \mu_0 - (\mu_0 - \mu_s M_1)\tan h\,(M_2 \Delta\omega) + M_3 \Delta\omega \quad (3)$$

Where, $\Delta\omega$ is the speed difference over the disc, r/min; μ_0 and μ_s are the sticking and slipping friction coefficient, respectively; M_1~M_3 are constants related to friction materials.

The purpose of this study is not to predict the absolute true values of the shift quality, but to reveal the influence law of hydraulic system failure on shift quality of the CVT tractor. Therefore, only the transmission model needs to be tested and verified. It is the core component of the CVT tractor. 8 groups of shifting experiments were carried out under different terminal pressure, limited flow, pump displacement, shift timing sequence, engine



a. Shift test 1.



b. Shift test 2.

Fig.3 Measured and simulated results of output speed response of transmission.

speed and load. The measured and simulated results of 2 groups of the experiments are shown in Fig. 3. According to Fig. 3, the output shaft speed response of the simulation model to the same pressure input is consistent with the test results.

4. Simulation under failure condition

6 failure modes were created in the laboratory, including the normal mode T1, piston jamming mode T2, seal ring damaging mode T3, oil way blocking mode T4, oil leaking mode T5, and T3×T4 combined failure mode T6. The oil pressure response of the engaged clutch C2 under different failure modes are shown in Fig. 4. It should be noted that the clutch can rise to the set pressure under failure mode T3 and T4, but not when they occur at the same time.

The aforementioned clutch pressure data under different failure modes are input into the simulation model, and the acceleration response of the tractor and the energy loss of the engaged clutch during shift are shown in Fig. 5. Fig. 5 shows only part of the simulation results, because the tractor had power interruption when shifting in T2, T5 and T6 modes.

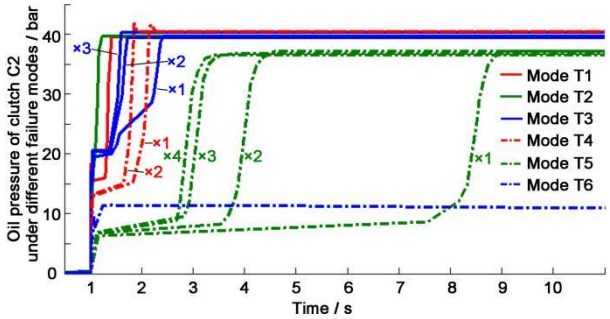


Fig.4 Pressure response of engaged clutch C2 under different failure modes.

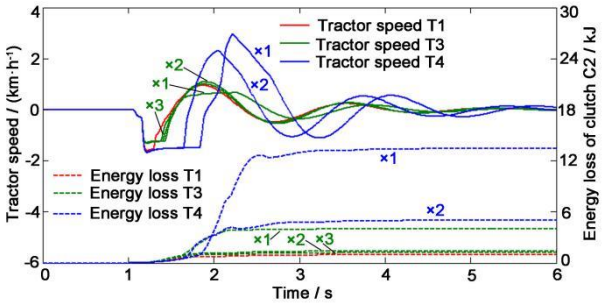


Fig.5 Acceleration response of tractor and energy loss of engaged clutch during shift.

5. Conclusion

(1)The mathematical model constructed in this paper can accurately calculate the response of the hydrostatic power split CVT to the clutch engagement pressure.

(2) The damage of the seal ring inside the rotary joint has little effect on shift quality. Sometimes it can even reduce the acceleration impact of the tractor during shift, but it increases the energy loss of the clutch.

(3) Oil way blockage can lead to greater shift acceleration impact and clutch energy loss.

(4) When seal ring damage and oil way blockage occur together, the clutch can not reach the minimum working pressure, which will lead to power interruption and tractor can not work normally.

(5) Clutch piston jamming and oil leakage can cause power interruption of tractor. They are all serious hydraulic system failures.

Acknowledgment

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