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Shift Quality Analysis of Heavy-Duty Vehicle Automatic Transmission Shift Control Valve

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Abstract: The structural concepts of Switch solenoid valve and proportional solenoid valve were proposed for the hydraulic shift control system of some hydrodynamic mechanical automatic transmissions. Shift oil pressure of stationary combination valve and proportional solenoid valve was modeled, simulated, contrasted and analyzed by dynamic simulation software in order to study the shift quality of heavy-duty vehicle automatic transmission. The results show that proportional solenoid valve is better to control the characteristic of shift oil pressure, reduce shift shock, improve shift quality and comfort than stationary combination valve. The correctness and validity of the model were verified through bench test, which reflected the dynamic characteristics of shift oil pressure of hydrodynamic mechanical automatic transmission. The results can be used to match the performance and predict heavy-duty vehicle shifting process, and to further lay the foundation for the enhancement of shift performance of the system.

Keywords: Shift oil pressure, dynamic simulation, stationary combination valve, proportional solenoid valve, hydrodynamic mechanical automatic transmission.

1. INTRODUCTION

Requirements for heavy-duty vehicle hydrodynamic mechanical automatic transmission (AT) on the shift quality, are smooth shifting process, small shifting shock and reduced dynamic load in the process of shifting [1-4]. At present, stationary combination valve on heavy-duty vehicle hydrodynamic mechanical automatic transmission is used to control cylinder oil filled pressure during the combination of clutch or brake, which controls hydraulic characteristic by structure parameters., and It is a simple method, with high reliability and low cost. However, when the structure of stationary combination valve is determined, pressure changing curves of combination components in the shifting process are identified which cannot be adjusted. Proportional solenoid valve shift control programs have been widely applied on foreign heavy-duty vehicles, and China also began to conduct relevant research [5, 6]. The advantage of proportional solenoid valve is that it can flexibly control oil pressure of combination elements during shifting process by adjusting current duty ratio, so that the process of shifting becomes stable. At present, domestic research is still in its infancy, and most of the researches just study clutch cylinder oil-filled pressure in the process of shifting based on the structure of the hydraulic control system. Cushioning performance of stationary combination valve was dynamically modeled with MATLAB/SIMULINK in literature [7, 8]. A simplified dynamic model of shifting process was established with the method of segmented differential equation shown in literature [9-11]. Kinematic and dynamic characteristics of AG4 transmission were qualitatively analyzed in the process of shifting with lever simulation method shown in literature [12], and a dynamic model of AT in the process of shifting was established with multi-body system dynamics [13]. In this paper, two shifting programs of stationary combination valve and proportional solenoid valve have been compared and analyzed, and simulation models will be established comparable to clutch hydraulic cylinder oil pressure changing in shifting process. Following this, the accuracy of simulation models was verified by the experiment.

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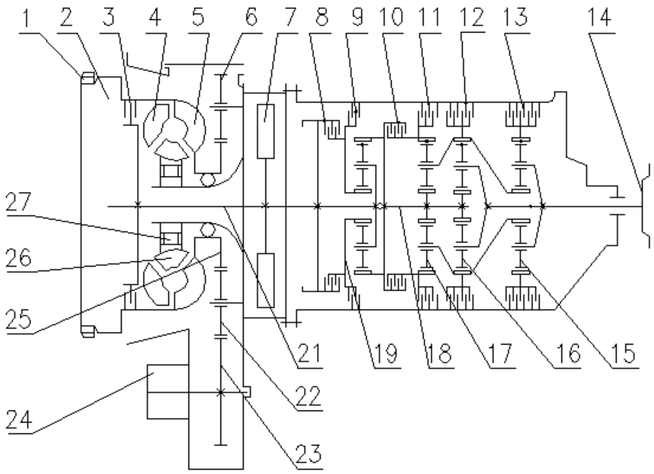
2. SCHEMES COMPARISON

Fig. (**1**) shows the transmission system of one hydrodynamic mechanical automatic transmission, which consists of four planetary gears train including GF (high torque divider), ZD (middle gear), DD (low gear), and FD (reverse gear). This four planetary gears train matches with DF (low torque divider) clutch, GF (high torque divider) brake, GD (high gear) clutch, ZD (middle gear) brake, DD (low gear) brake, and FD (reverse gear) brake, regularly consisting of six forward speeds (with the same turned off engine, namely forward gears) output, an inverse speed (with the turned to the contrary of engine, that reverse gear) output and a neutral (no power output), a total of eight gears. Control elements engaged in each gear are shown in Table **1**. This hydrodynamic mechanical automatic transmission completes power shift gear, combination and separation of the lockup clutch, fills oil in hydraulic torque converter and controls, hydraulic retarder and lubrication function by electro-hydraulic control system, which requires pressure of 1.2 ~ 1.48 MPa, with engagement time of shifting clutch being 0.8 ~ 1.8 s for a smooth shifting process.

In the study of system, to establish a mathematical model is crucial. In order to facilitate research, the pressure of the clutch controlled is considered, and the hydraulic control system is simplified by a single control valve to control the clutch, as shown in Fig. (2). The mathematical model of the transmission hydraulic control system is established by adopting transfer function using the flow continuity equation, so that its dynamic characteristics can be analyzed.



1-Engine starting gear; 2-Flywheel; 3-Lockup clutch; 4-Turbine; 5-Pump wheel; 6-Power take off gear; 7-Hydraulic retarder; 8-Low speed clutch; 9-High speed brake; 10-High gear clutch; 11-Middle gear brake; 12-Low gear brake; 13-Reverse gear brake; 14-Driving gear; 15-Reverse gear planetary row; 16-Low gear planetary row; 17-Middle and high gear planetary row; 18-Main transmission shaft; 19-Vice transmission planetary row; 21-Torque converter output shaft; 22-Intermediate gear; 23-With oil pump gear; 24-Oil pump; 25-Power take off drive gear; 26-Guide wheel of torque converter; 27-One-way clutch.

Fig. (1). A transmission system diagram of hydrodynamic mechanical automatic transmission.

When analyzing the system, the following assumptions are made:

(1) Valves have ideal response ability in response to the fully open.

(2) Ignoring frictional losses within the pipe.

(3) Ignoring the impact of the oil leakage and compressibility.

(4) Supply pressure is constant, and return pressure is zero.

(5) The oil temperature is constant.

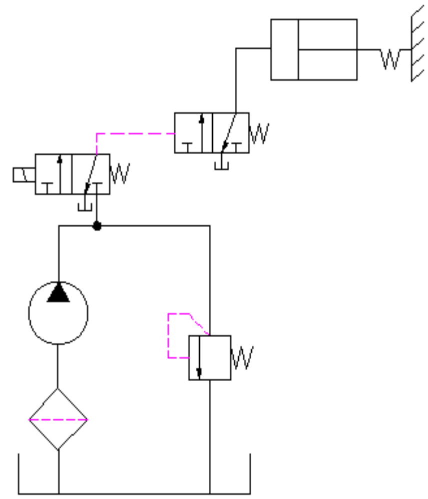


Fig. (2). Simplified model of hydraulic system.

The linearization flow equation of two-bit three-way valve hydraulic controlled is as follows:

$$\Delta Q_L = K_q \Delta X_V - K_C \Delta P_C \tag{1}$$

where Q_L-Load flow, m^3/s;

K_q-Flow gain of spool valve, m^2/s;

K_C-Flow pressure coefficient of spool valve, m^5/N·s;

X_V-Spool displacement of spool valve, m;

P_C-Control pressure of the hydraulic cylinder control chamber, Pa.

Flow continuity equation of the hydraulic cylinder is as follows:

$$Q_L = A\frac{dV_C}{dt} + C_{ip}P_C + \frac{V_C}{\beta_e}\frac{dP_C}{dt} \tag{2}$$

$$V_C = V_0 + AX_P \tag{3}$$

where A-Working area of the hydraulic cylinder, m^2;

Table 1. Combining components of each gear.

Gears	Combined with Control Components					
	8 (DF)	9 (GF)	10 (GD)	11 (ZD)	12 (DD)	13 (FD)
Reverse	+					+
Neutrallal alal llll	+					
I	+				+	
II		+			+	
III	+			+		
IV		+		+		
V	+		+			
VI		+	+			

C_{ip}-Internal leakage coefficient of hydraulic cylinder, $m^5/N\cdot s$;

V_C-Volume of hydraulic cylinder control chamber, m^3;

V_0-The initial volume of the hydraulic cylinder control chamber, m^3;

β_e-Effective bulk modulus of fluid, Pa;

X_P-Piston displacement of the hydraulic cylinder, m.

The force balance equation of hydraulic cylinder is as follows:

$$P_C A = M_t \frac{d^2 X_P}{dt^2} + K X_P + F_L \tag{4}$$

where M_t-The total mass of the piston and the load, kg;

K-Spring stiffness of the load, N/m;

F_L-Any external load force, N.

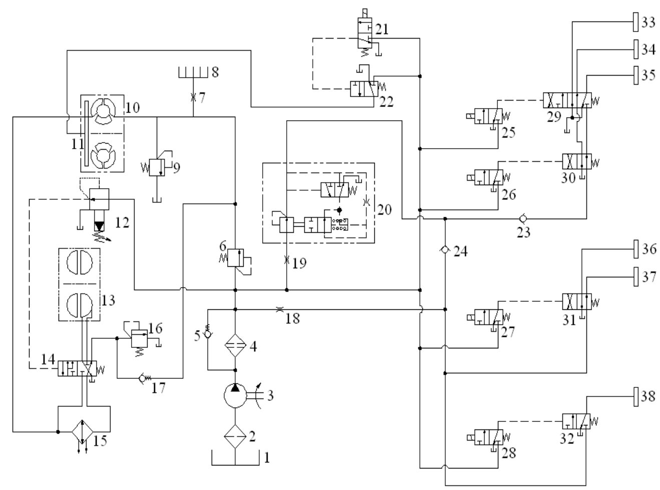
Above formulas are transformed by the Laplace, merged and eliminated, thus the transfer function of hydraulic system can be done.

$$G(s) = \frac{P_C}{X_V} = \frac{K_q(M_t s^2 + K)}{\frac{V_C M_t}{\beta_e} s^3 + (K_C + C_{ip}) M_t s^2 + (A^2 + \frac{V_C K}{\beta_e}) s + (K_C + C_{ip}) K} \tag{5}$$

According to the combination laws of control components of some hydrodynamic mechanical automatic transmissions, two kinds of hydraulic control system solutions were put forward. One shifting is controlled by switch electromagnetic valve; the other one suggests that shifting is controlled by proportional solenoid valve.

2.1. Shifting Program of Switch Electromagnetic Valve

Structure program of switch electromagnetic valve uses a smooth combination valve to control the clutch oil filled pressure, as shown in Fig. (**3**). Its characteristic is the use of main pressure regulating valve 6 to manipulate the control pressure of system. Four de-energized normally with closed switch electromagnetic valves are used from 25 to 28, control connection state of the gear shift valves 29 to 32 and combination elements 33 to 38, to realize the transmission of six forward gears, one reverse gear and one neutral gear. Under each gear, solenoid valves 29 to 32 are energized as shown in Table **2**. The purpose of improving the shift quality was achieved by using a smooth combination valve 20 to adjust shifting oil pressure when clutches and brakes engaged. High speed brake 36, low-speed clutch 37 and reverse gear brake 38 are directional control engagement elements, while middle gear brake 33, high gear clutch 34 and low gear brake 35 are gear control engagement elements. The throttle valve 18 leads pressurized oil in working connection to directional control engagement elements 36 to 38 to ensure that directional control engagement elements engaged first. Check valves 23 and 24 are to prevent the working fluid reflux in the clutch.



1-Sump; 2-Crude oil filter; 3-Oil pump; 4-Refined oil filter; 5-Safety valve of refined oil filter; 6-Main pressure valve; 7-Throttle valve; 8-Lubrication points; 9-Torque converter import safety valve; 10-Hydraulic torque converter; 11-Lockup clutch; 12-Pilot valve M6 of retarder; 13-Hydraulic retarder; 14-Control valve of hydraulic retarder; 15-Cooler; 16-Outlet back pressure valve of hydraulic torque converter; 17-Flow compensation valve of torque converter; 18-Throttle valves; 19-Throttle valves; 20-Smooth combination valve; 21-Electromagnetic proportional valve with locking control M5; 22-Lockout control valve; 23-Check valves; 24-Check valves; 25-Shifting solenoid valve M3; 26-Shifting solenoid valve M1; 27-Shifting solenoid valve M4; 28-Shifting solenoid valve M2; 29-Shifting valve III; 30-Shifting valve I; 31-Shifting valve IV; 32-Shifting valve II; 33-Middle gear brake; 34-High gear clutch; 35-Low gear brake; 36-High speed brake; 37-Low speed clutch; 38-Reverse gear brake.

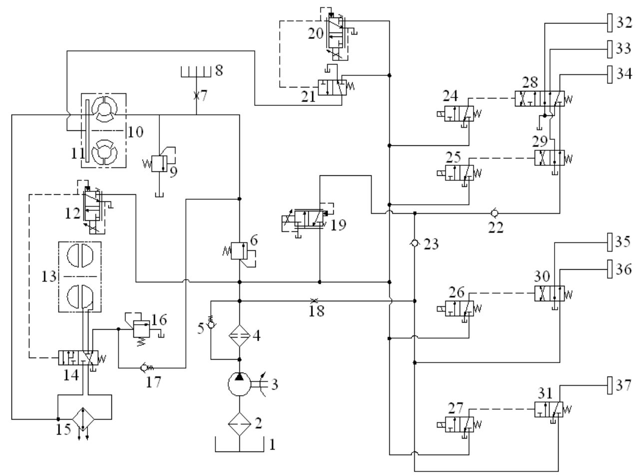
Fig. (3). Switch electromagnetic valve structure scheme.

Table 2. Solenoid valve power state of each gear.

Gears	Solenoid Valve			
	M1	M2	M3	M4
Reverse		●		
Neutral				
I			●	
II			●	●
III	●		●	
IV	●		●	●
V	●			
VI	●			●

2.2. Shifting Program of Proportional Solenoid Valve

The characteristic of the proportional solenoid valve solution is that there is shift control valve 19 instead of stationary combination valve in shift control scheme of switch electromagnetic valve, as shown in Fig. (**4**). The energized states of solenoid valves under each gear are consistent with the structure program of switch electromagnetic valve, as shown in Table **2**.



1-Sump; 2-Crude oil filter; 3-Oil pump; 4-Refined oil filter; 5-Safety valve of refined oil filter; 6-Main pressure valve; 7-Throttle valve; 8-Lubrication points; 9-Torque converter import safety valve; 10-Hydraulic torque converter; 11-Lockup clutch; 12-Pilot valve M6 of retarder; 13-Hydraulic retarder; 14-Control valve of hydraulic retarder; 15-Cooler; 16-Outlet back pressure valve of hydraulic torque converter; 17-Flow compensation valve of torque converter; 18-Throttle valves; 19-Shift proportional valve; 20-Electromagnetic proportional valve with locking control; 21-Lockout control valve; 22-Check valves; 23-Check valves; 24-Shifting solenoid valve M3; 25-Shifting solenoid valve M1; 26-Shifting solenoid valve M4; 27-Shifting solenoid valve M2; 28-Shifting valve Ⅲ; 29-Shifting valve Ⅰ; 30-Shifting valve Ⅳ; 31-Shifting valve Ⅱ; 32-Middle gear brake; 33-High gear clutch; 34-Low gear brake; 35-High speed brake; 36-Low speed clutch; 37-Reverse gear brake

Fig. (4). Proportional solenoid valve structure scheme.

3. SIMULATION AND ANALYSIS

The simulation models of stationary combination valve and proportional solenoid valve were established with application system simulation software of AMESim. In the process of the whole simulation, the simulation system was shown idealistically through an intuitive graphical interface with ignoring the leakage of liquid, the actual length of the pipeline and so on.

3.1. The Simulation Model of Stationary Combination Valve

Simulation model established based on the components of hydraulic component design library (HCD) is shown in Fig. (**5**). In the simulation experiment, the density of 8th hydrodynamic transmission oil was 860 kg/m3 at 50°C; the pressure of the main pressure valve was set at 12 bar; spring stiffness of pressure regulating valve was 7400 N/m, and preload was 13 N; spring stiffness of setting valve was 1000 N/m and preload was 78.5 N; diameter of the throttle P was 3.2 mm; diameter of the orifice 1 was 0.5 mm; stroke of the clutch was 2 mm; while, inside and outside diameter of each valves were set by relevant structures.

Hydraulic oil out of the pump flew form stationary combination valve to clutch cylinder, and pressure of the clutch cylinder is shown in Fig. (**6**) while shifting. Oil charge and pressure boost of clutch is based on three phases. The first phase was rapid oil-filled in a time of 0.2 s so that the gap between the clutch friction plates was eliminated and oil pressure increased to around 5.5 bar at this time. The second phase of buffer boosting lasted from 0.2 s to 1.2 s with the oil pressure increased from 5.5 bar to 10 bar which can be seen in the figure. At this stage, the clutch plates were pressed from transmitting friction torque through the clutch slipping until the clutch completely jointed, where the characteristics of stationary combination valve played a decisive role in determining the quality of shifting. The third phase was of a very short time of about 0.05 s, in which the oil pressure of clutch cylinder rose sharply to the pressure of hydraulic system [14].

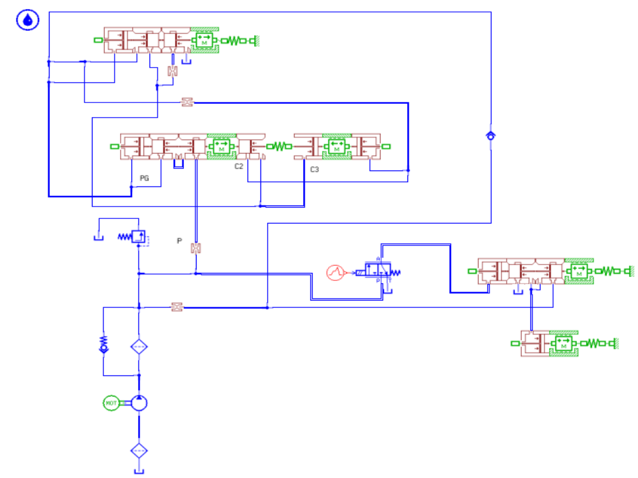


Fig. (5). Simulation model of stationary combination valve.

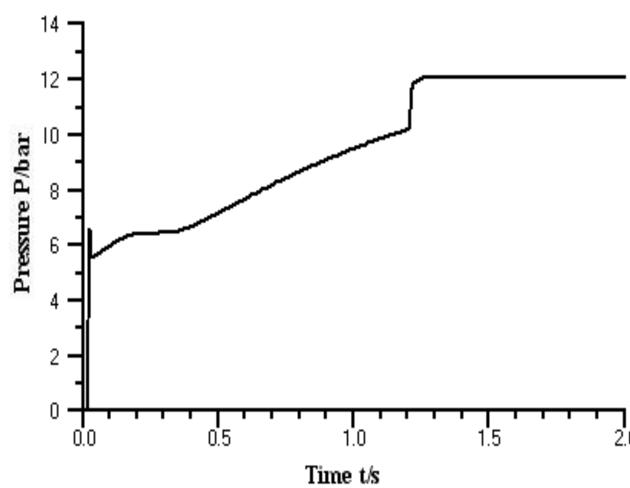


Fig. (6). Clutch hydraulic cylinder oil-filled pressure curve controlled by stationary combination valve.

3.2. The Simulation Model of Proportional Solenoid Valve

Simulation model of proportional solenoid valve applied HCD as shown in Fig. (**7**), while parameters of hydraulic oil, main pressure valve, shifting solenoid valve, shifting valve, clutch were same with the parameters of the simulation model in Fig. (**5**), whereas, the proportional magnification of PID controller set was 2, integral coefficient was 0.005, differential coefficient was 0.002, the size of the proportional solenoid valve was set by the relevant diagram. Control signal was set to control actions of proportional pressure valve, pilot electromagnetic valve and shifting valve so that engagement of the clutch was realized. The curve of pressure changing during engagement of the clutch is shown in Fig. (**8**), showing four stages of oil-filled in clutch hydraulic cylinder [15]. The first stage involved rapidly filling oil from 0 to about 5 bar lasting from 0 to 0.08 s, in which hydraulic oil overcame the preload of the clutch spring, eliminating the gap between clutch plates to make them fit. The second stage invloved piston motion with oil filled lasting from 0.08 s to 0.45 s and oil pressure rose slowly. At this stage, opening of proportional valve was adjusted to a certain degree so that friction plates were pressed to begin transmitting friction torque through slipping until complete contact. The third stage involved proportional boosting lasting from 0.45 s to 1.08 s with oil pressure increasing from 6 bar to 11 bar, while proportional solenoid valve increased opening, which made oil pressure rise steadily to reduce shift shock. The fourth stage created boost in a short time, in which the opening of proportional valve was the largest, while oil pressure of hydraulic cylinder quickly raised to 12 bar than the oil pressure of the hydraulic system.

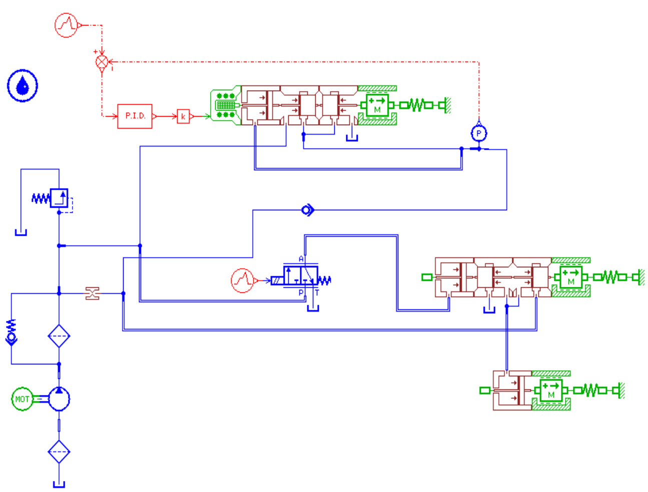


Fig. (7). Simulation model of proportional solenoid valve.

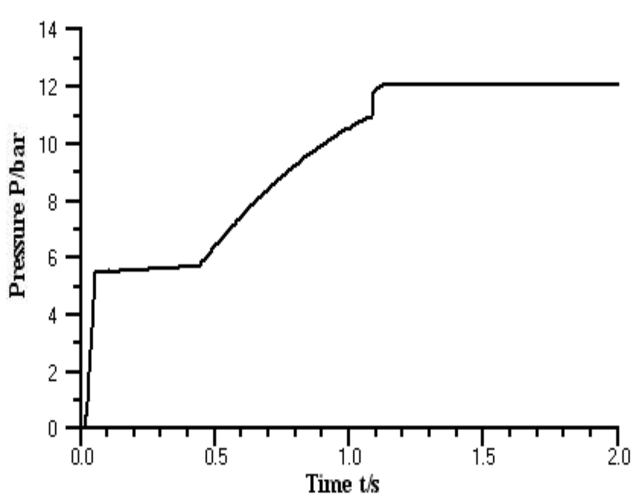


Fig. (8). Clutch hydraulic cylinder oil-filled pressure curve controlled by proportional solenoid valve.

3.3. The Analysis of Simulation Results

The stationary combination valve in shift control scheme of switch electromagnetic valve constituted regulating valve, energy accumulator, setting valve and mechanical element controlled shift quality. Characteristics controlled pressures of respective stages were determined by orifice and spring stiffness. Characteristics of controlled pressure were determined when a certain structure and its joint properties were not easy to change with poor flexibility. The proportional solenoid valve could continuously control the pressure, and its output pressure directly controlled by the input current with high flexibility. The pressure of engagement flexibly adjusted by the duty cycle of control current is to make the shifting smoother.

Therefore, compared to Fig. (**8**) and Fig. (**6**), it can be establiahed that the curve of oil pressure controlled by proportional solenoid valve boosted more stages than stationary combination valve and each stage is clearly distinct. Also, oil pressure rose proportionally according to the input signal, which made shifting smoother, shift shock smaller and more comfortable. Application of proportional solenoid valve with high precision and flexibility changed its regulation characteristics by adjusting the numerical of proportion, integral and differential in PID.

4. EXPERIMENTAL VERIFICATION

After simulation analysis of the pressure in the hydraulic cylinder, shifting, accuracy and reliability of the simulation results would be verified through the experiment of oil pressure. Experiment of oil pressure for a certain automatic transmission involved hydrodynamic transmission test which mainly included: motor, the measured transmission, automatic control system, torque sensor, loading device, manipulation and measurement system of bench, computer. 8^{th} hydraulic transmission oil was selected in the experiment, running one hour at each gear shift before testing in order to check whether there was abnormal sound, vibration, sealing of the hydraulic system. Each measuring instrument was working properly.

Comparison between simulation result and experimental result is shown in Fig. (**9**). Oil pressure was controlled by stationary combination valve on the automatic transmission. It can be seen from the figure that simulation result and experimental result were in good agreement with dynamic characteristics of oil pressure and oil-filled time. Because ignoring the impact of friction between oil and pipe, leakage and hysteretic of the oil pressure, oil filled time of simulation model was faster than in the experiment and oil pressure was higher than measured. Correctness and validity of the simulation model were verified, so it could better reflect the dynamic characteristics of oil pressure during shifting and can be matched and forecasted for shifting process of heavy vehicle. Due to constraints, we failed to do the experiment of proportional solenoid valve, however based on the same modeling approach with stationary combination valve, it could determine that the simulation model was correct and the result was credible.

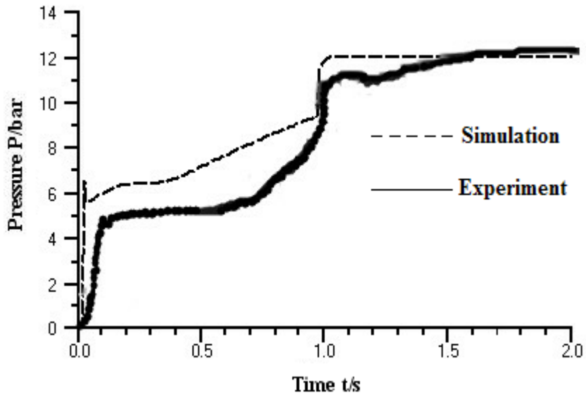


Fig. (9). The test result compared with simulation result of shift oil pressure.

CONCLUSION

(1) According to the combination rule of control components on the automatic transmission of a certain heavy vehicle when shifting, two solutions of hydraulic control system were proposed; one used switch electromagnetic valve to control shifting, and the other was proportional solenoid valve. Models of stationary combination valve and proportional solenoid valve were built by application of dynamic simulation software and oil pressure during shifting was simulated and analyzed. The results show that oil filled pressure controlled by proportional solenoid valve has one more stage than stationary combination valve and the pressure rises proportionally, which makes the clutch combine smoothly, reduces shift shock, and improves shift quality and comfort.

(2) Correctness and validity of the simulation model were verified through bench test, which reflected dynamic characteristics of oil pressure during shifting, so it could be used to match and predict the performance of heavy vehicle in the process of shifting, to lay the foundation for improving the shifting performance of system.

CONFLICT OF INTEREST

The authors confirm that this article content has no conflict of interest.

ACKNOWLEDGEMENTS

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