Shift Control System of Heavy-duty Vehicle Automatic Transmission

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Abstract—Heavy-duty vehicle hydrodynamic mechanical automatic transmission shifting operation system was designed, mathematical model of its simplified hydraulic system was established and simulation model of shifting operation system was established with AMESim, the simulation experiment was carried out, then oil pressure curves of each clutch hydraulic cylinder were obtained when giving forward gear or reverse gear signals. The simulation results show that shifting operating system meets the design requirements, and verify the correctness of the model. The shift timing is correct, and there is no power interruption or gear overlap during the shift transition process. Joint oil pressure of designed system is stable, and shifting shock is small. The research results are providing the basis for further study of shifting operation system and a reasonable platform for the studying of shift schedule and quality. The theoretical design method and dynamic simulation experiment will be feasible for the real industrial applications. The research results can be used in design and optimization of hydraulic system.

Index Terms—Hydrodynamic Mechanical Automatic Transmission; Shifting Operation System; Dynamic Simulation Experiment; Joint Oil Pressure

I. INTRODUCTION

The rapid development of computer technology promotes the application of simulation technology in social production and life [1-2]. The application of computer simulation technology is also becoming more and more popular in the development, design, manufacture, assembly, testing and other aspects with vehicles as the most common forms of transport [3-4]. Automatic transmission as important automotive power train components, its advantages and disadvantages of power transmission performance directly affect the quality of the vehicle. In AT research and develop process, the use of computer simulation technology for virtual development can be achieved in parallel development of products, which effectively shortens the development cycle and reduces the cost of production, therefore, virtual design for the automatic transmission has gradually become the mainstream [5]. Hydrodynamic mechanical automatic transmission (AT) is currently the most widely used type of automatic transmission, and it becomes the first choice of heavy-duty vehicle automatic transmission with its advantages of simple operation and saving effort, improving the traffic safety, reducing the labor intensity, improving the ride comfort, extending the service life of the mechanical parts, improving the dynamic performance of the vehicle, reducing air pollution, and having a good self-adaptability [6-7].

Shift operation system is an important part of automatic transmission mainly used to ensure normal work of transmission system and realize changing shift. Rational design shift operation system not only can extend the service life of the shift clutch and brake, but also improve the reliability of the automatic transmission, and can reduce the power loss in shifting process, reduce shift shock, improve shift quality [8]. In previous studies, the mathematical model of shift operation system established often ignored the influence of friction, oil characteristics, environmental temperature and other factors, so that it cannot truly reflect shifting characteristics of shift operation system [9-11]. Literatures [12-13] are modeling and simulation for one valve of automatic transmission hydraulic system. Literatures [14-15] are modeling and simulation research for a clutch of shifting hydraulic system. Until now, there are few references about the research for the modeling and simulating of AT hydraulic system. Taking into account the automatic transmission of heavy vehicles gradually developing in the direction toward multi-shift, this paper designed eight-speed automatic transmission shift operation system of some heavy-duty, established the simulation model of the system, analyzed oil pressure changing curves of each clutch and brake during shifting process [16]. Kinematics and dynamics changing process of each manipulation member in shifting hydraulic system was mastered, the separation and integration of the clutch when shifting was observed visually.

II. SYSTEM MODEL

A. The Principle of Shifting Control System

Fig. 1 is shown that the shifting operation system working schematic and hydraulic components of
designed a heavy-duty eight-speed automatic transmission. Shifting valve uses priority interlock circuit, in order to prevent two of clutches jointed at the same time effectively [17]. The function of stationary combination valve 9 is adjusting boost characteristic of clutch cylinder in the shifting instant, reducing shift shock, improving shift quality. High speed brake 23, low speed clutch 24 and reverse gear brake 25 are directional control joint element, middle gear brake 20, high gear clutch 21 and low gear brake 22 are shifting control joint element. The working condition of electromagnetic valve can achieve six forward gears and one reverse gear operation as shown in Table I.

<table>
<thead>
<tr>
<th>Gears</th>
<th>Electromagnetic valve</th>
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<td>Reverse</td>
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<td>Neutral</td>
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<td>I</td>
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B. The Mathematical Modeling of Shifting Operation System

Mathematical modeling of shifting operation system is beneficial for selecting simulation parameters and analyzing dynamic characteristics of operation system, though simulation analysis software based on graphical modeling approach of physical models. The mathematical model of the clutch pressure control is established with oil passed gear valve II to clutch as an example. The clutch is equivalent to spring damping system, considering that viscous damping coefficient between clutch piston and the external load is small, which can be ignored. Fig. 2 is shown simplified model of hydraulic operation system. Making following assumptions when the system is analyzed:

1. Response capability of the valve is ideal that is the valve port is response to fully open, then the spool displacement and the flow rate change of pressure drop can occur instantaneously.
2. The internal and external leakage of hydraulic cylinder is laminar flow.
3. The oil supply pressure is constant, and the oil return pressure is zero.
4. The internal friction loss and the dynamic characteristics of the pipeline are ignored.
5. The oil temperature is constant.

![Figure 2. Simplified model of hydraulic system](image)

After linear processing to steady-state working points on the system, linear flow equation of the two-position three way valve for fluid control is obtained

\[
\Delta Q_L = K_q \Delta X_v - K_c \Delta P_c .
\]  

where \( Q_L \)——flow of load, \( m^3/s \),

\( K_q \)——flow gain of slide valve, \( m^2/s \),

\( K_c \)——flow pressure coefficient of slide valve, \( m^5/N\cdot s \),

\( \Delta X_v \)——spool displacement of slide valve, m,

\( P_c \)——the control pressure of the hydraulic cylinder control chamber, Pa.

The Laplace transformation of formula (1) is

\[
Q_L = K_cX_v - K_cP_c .
\]  

Considering internal and external leakage of the hydraulic cylinder and hydraulic oil compressibility in cylinder, the flow continuity equation of hydraulic cylinder is got

\[
Q_L = \frac{dV_C}{dt} + C_{ip}P_C + \frac{V_c}{\beta_c} \frac{dP_c}{dt},
\]  

\[
V_C = V_0 + AX_p
\]  

where \( A \)——work area of hydraulic cylinder, \( m^2 \),

\( C_{ip} \)——internal leakage coefficient of hydraulic cylinder, \( m^5/N\cdot s \),

\( V_C \)——control cavity volume of hydraulic cylinder, \( m^3 \),

\( V_0 \)——control cavity initial volume of hydraulic cylinder, \( m^3 \),

\( \beta_c \)——effective bulk elasticity modulus of oil, Pa.
$X_p$—piston displacement of hydraulic cylinder, m.

Combined formula (3) and (4) is

$$Q_L = A \frac{dX_p}{dt} + C_p P_C + V_C \frac{dP_C}{dt} \beta_e \frac{dt}{dt}$$

(5)

The Laplace transformation of formula (5) is

$$Q_L = A sX_p + C_p P_C + \frac{V_C}{\beta_e} sP_C$$

(6)

Combined formula (2) and (6) is

$$K_s X_p - K_e P_C = A sX_p + C_p P_C + \frac{V_C}{\beta_e} sP_C$$

(7)

Ignored transient fluid dynamic, steady-state flow force and viscous damping force during the process of hydraulic cylinder movement, then the force balance equation of the hydraulic cylinder is got

$$P_C A = M \frac{d^2 X_p}{dt^2} + K X_P + F_L$$

(8)

where $M$—total mass of the piston and the load, kg.
$K$—spring stiffness of load, N/m,
$F_L$—any external load force, N.

The Laplace transformation of formula (8) is

$$P_C A = M s^2 X_p + K X_P + F_L$$

(9)

Transfer function of hydraulic control system can get by combining formula (7) and (9) then eliminating $X_p$.

Fig. 3 is transfer function block diagram of hydraulic cylinder system controlled by valve with input spool displacement $XV$ and output the clutch pressure $PC$ depicted by formula (10).

The form of the transfer function only depends on the parameters of the system itself, and has nothing to do with input function of external, which means it does not consider the influence of $F_L$. $G(s)$ describes the inherent dynamic characteristics of system, which reflects the dynamic response of the pressure in clutch cylinder. Substituting the parameter values and determining the stability of established shifting operation system, the result shows that the system is stable, and proves the rationality of the designed hydraulic operation system.

$$G(s) = \frac{P_C}{X_V} = \frac{K_s \{M s^2 + K\}}{V_C M \beta_e s^2 + \{K_e + C_p\} M s^2 + \{K_e + C_p + A^2\} \frac{V_C}{\beta_e} s + \{K_e + C_p\} K}$$

(10)

If the ratio of load spring stiffness and hydraulic spring stiffness is $K/K_e << 1$, and satisfied $\left(\frac{K_e \sqrt{M K}}{A^2}\right) << 1$, the formula (10) can be further simplified to

$$G(s) = \frac{P_C}{X_V} = \frac{K_q}{K_e + C_p}$$

(11)

where $\omega_n$—hydraulic natural frequency,
$$\omega_n = \sqrt{\frac{K_q}{M_i}} = \frac{\beta A^2}{V_C M_i}, \text{rad/s,}$$
$$\zeta_h$$—damping ratio, $\zeta_h = \frac{K_e + C_p + \beta M}{2A} \sqrt{\frac{V_C}{K_q}}$,
$$\omega_r$$—inertia corner frequency, $\omega_r = \frac{(K_e + C_p) K}{A^2}, \text{rad/s,}$$
$K_q$—hydraulic spring stiffness, $K_q = \frac{\beta A^2}{V_C}$, N/m.

Hydraulic system natural frequency $\omega_n$ is an important parameter of measuring system dynamic characteristics. It can get a better stability by appropriately increasing the natural frequency of the system, so improving working area of hydraulic cylinder is conducive to the stability of the system. Additionally, the smaller the 1/$\omega_n$ is, the faster system reflects. Therefore, increasing working area of hydraulic cylinder and reducing spring stiffness of load appropriately can effectively avoid the clutch pressure rising too fast, which is conducive to improve shifting performance.

Then analyzed dynamics and kinematics of shifting process, and got the maximum friction clutch torque $M_m$ from relevant information [18]

$$M_m = \beta M = \mu P R_c k_0$$

(12)

where $M$—transmission torque, N·m,
$\beta$—reserve coefficient,
$\mu$—coefficient of friction,
$R_c$—equivalent radius of frictional force, m,
$P$—clamping force, MPa,
$z$—the number of friction pair,
$k_0$—loss coefficient of compression force.

The clamping force $P$ is calculated as

$$P = \frac{\pi}{4} \left(D_2^2 - D_1^2\right) |q| [\psi]$$

(13)

The equivalent radius is calculated as

$$R_q = \frac{D_2 + D_1}{4}, \quad c = \frac{D_1}{D_2}$$

(14)

Taken formula (13) and (14) into formula (12) and got maximum friction torque of the clutch
Figure 4. Simulation model of shifting operation system

\[ M_m = \beta M = \frac{\pi^2}{16} \mu D_1^3 (1-c^2)(1+c) \varphi_0 \]  

where \( D_1 \) —— inside diameter of friction plate, m,
\( D_2 \) —— outside diameter of friction plate, m,
\( \varphi \) —— utilization area coefficient of friction plates,
\( [q] \) —— allowable specific pressure.

The actual transmitted friction torque \( M_m \), when the clutch in the working process is

\[ M_m = \mu P R \varphi_0 \]  

The clamping force \( P \) is calculated as

\[ P = P \frac{\pi}{4} \left( D_2^2 - D_1^2 \right) \]  

where \( P \) —— the instantaneous pressure of clutch piston.

Taken formula (14) and (17) into formula (16)

\[ M_m = \frac{\pi}{16} \mu P \left( D_2^2 - D_1^2 \right) \left( D_1 + D_2 \right) \varphi_0 \]  

Form formula (18), it can be seen for shift clutch of structure determined, friction torque is only related with clutch piston supported instant pressure proportionally at the shifting moment. Therefore, the changing pressure influenced on shifting is the main content of research and analysis in this paper.

III. Simulation Results

AMESim simulation model is set up, and the design philosophy of AMESim is based on bond graph modeling approach, compared to using transfer function described the dynamic system, it takes into account the friction, the oil itself characteristics, environmental temperature and so on some parts difficult to model, which reduces the error brought by these factors. The established model can visually reflect the dynamic characteristics of hydraulic system. Compared with traditional hydraulic system design and analysis methods, it saves a lot of time and efforts and clearly observes the parameters of each component influenced on the dynamic characteristics of the hydraulic system. The combination and separation of the clutch oil pressure curves of hydraulic cylinder during the shifting process are obtained in order to verify dynamic oil pressure changing consistent with the theoretical design requirements.

Fig. 4 is shown shift control system simulation model in the AMESim, established most valve model by using HCD (shift control system simulation mode), and given shift signal by using truth table in the drive library. Main oil pressure in the system is provided by metering pump, controlled the main oil pressure between 1.2 to 1.48MPa, metering pump speed from 0 to 2100r/min, flow is 250L/min, fine oil filter relief valve opening pressure is 0.14 MPa, using PTF-2 hydraulic transmission oil.

A. Analysis Forward Upshifts of Simulation Results

Fig. 5 is shown that upshift signal from neutral position, first gear to sixth gear, each gear electromagnetic valve receives signal as shown in Table 1. Fig.6 is shown oil pressure curves of low speed clutch and low gear brake when neutral position up to the first gear, it can be seen low speed clutch is jointed at neutral position, shifting electromagnetic valve M3 received electromagnetic signal promotes gear valve III so that low gear brake begin to filling oil and joints when given the first signal at 5 seconds. Oil pressure rising characteristic of low gear brake hydraulic cylinder due to the effect of stationary combination valve, is divided into three stages obviously, and the first stage is rapidly filled oil in the clutch cavity and establishes a certain pressure to eliminate the gap between the clutch friction plates. The second stage is buffer boost stage, which is a key part of the shift quality, and there is no change with the flow rate in the clutch hydraulic cylinder, only with rising of pressure from friction plates compacted and staring passing on friction torque to the clutch jointed completely. If the buffer time is too short, it will produce the shift shock, inversely, if the buffer time is too long, it will
results in slipping friction overlength, slipping friction work increased, the temperature rising, wear and tear intensified. The third stage is step boost stage, in which the friction between clutch plates changes from dynamic friction to static friction, the oil pressure rise time of clutch hydraulic cylinder is very short and easy to produce dynamic loads, resulting in shift shock, however, high retention oil pressure provides a certain torque reserve during the process of transmission torque, which prevents clutch slipping caused by mutation load. Through the above analysis, dynamic hydraulic shift changes consistent with the theoretical design requirements.

Fig. 5 is shown that hydraulic cylinder oil pressure curves of low speed clutch, high speed brake and low gear brake when the first gear up to the second gear. The curves show that low speed clutch begins to separate, at the same time high speed brake begins to joint and low gear brake is always in the joint state when the signal up to the second gear is given at 10 seconds until the end of shifting process at 10.26s. This process is divided into two stages, the first stage is low gear torque phase, which the system is still running at low gear with the oil pressure of low speed clutch decreased but still jointed, however, the oil pressure of high speed brake has began to rise, slip and transfer torque. The second stage is high gear inertia phase that has been completed shifting with low speed clutch into complete separation state and high gear brake from slipping into fully jointed [19]. Torque phase and the inertia phase indicate that there exist power overlap phenomenon during the process of shifting, while it avoids “galloping” caused by power interruption, but too much overlap may cause clutch is not completely separated then appears double shifts to cause dramatic changes of torque and rotational speed. Both of two conditions can generate shift shock, therefore timing control should be taken when clutch combined during shifting to achieve the ideal synchronization shift.

Fig. 8 is shown that hydraulic cylinder oil pressure curves of low speed clutch, high speed brake, low gear brake and middle gear brake when the second gear up to the third gear. The curves indicate that oil pressure of high speed brake and low gear brake rapidly reduced, while low speed clutch and middle gear brake begin to joint when given the shift signal at 15 seconds. Low speed clutch engages earlier than middle gear brake, this is because low speed clutch is direction control element and middle gear brake is shift control element. Oil pressure of middle gear brake obviously lags behind the low speed clutch due to throttle valve 7 in shift operation system.
brake and high gear clutch when the fourth gear up to the fifth gear. Fig. 11 is shown that hydraulic cylinder oil pressure curves of low speed clutch, high speed brake and high gear clutch when the fifth gear up to the sixth gear. The oil pressure changing principle of each clutch, brake is same, so it will not do a detailed explanation.

Figure 10. Clutch pressure from fourth speed to fifth speed

Figure 11. Clutch pressure from fifth speed to sixth speed

Figure 12. Downing forward gear signal

B. Analysis of Forward Downshifts Simulation Results

Equations

Fig. 12 is given the signal from sixth gear down to first gear sequentially then to the neutral gear. Fig.13 is shown that hydraulic cylinder oil pressure curves of low speed clutch, high speed brake and high gear brake when the sixth gear down to fifth gear. The curves indicate that high speed brake is jointed earlier than high gear clutch also because of the effect of throttle 7 when given the sixth gear signal at 0 seconds in order to ensure direction control element jointed earlier and speed control element jointed later. After downshifts signal is given at 5 seconds, high gear clutch keeps working condition unchanged, while high speed brake begins to drain oil and low speed clutch begins to fill oil. Fig.14 is shown that hydraulic cylinder oil pressure curves of low speed clutch, high speed clutch, middle gear brake and high gear brake when fifth gear reduced to fourth gear. The curves indicate that when given the signal of fifth gear down to fourth gear at 10 seconds, high gear clutch and low speed clutch begin to drain oil, while middle gear brake and high speed brake begin to fill oil. Also because of the effect of throttle 7, oil pressure of middle gear brake rises behind high speed brake.

Figure 13. Clutch pressure from sixth speed to fifth speed

Figure 14. Clutch pressure from fifth speed to fourth speed

Figure 15. Clutch pressure from fourth speed to third speed

Fig. 15 is shown that hydraulic cylinder oil pressure curves of low speed clutch, high speed brake and middle gear brake when the fourth gear down to third gear. Fig. 16 is shown that hydraulic cylinder oil pressure curves of low speed clutch, high speed brake, high gear brake and low gear brake when the third gear down to second gear. Fig. 17 is shown that hydraulic cylinder oil pressure curves of low speed clutch, high speed brake and low gear brake when the second gear down to first gear. Fig.18 is shown that hydraulic cylinder oil pressure curves of low speed clutch and low gear brake when the
first gear down to neutral gear. From the figures it can be seen oil pressure changing of each clutch conforms to shifting logic. This method accurately reflects the shifting process of hydraulic control system for the design and optimization of the hydraulic system providing a strong basis.

C. Analysis of Reverse Gear Simulation Results

Given signal of neutral gear-reverse gear-neutral gear shown in Figure 19, in this signal hydraulic cylinder oil pressure curves of low speed clutch and reverse gear brake are shown in Figure 20. It can be seen after given reverse gear signal at 5 seconds, oil pressure of low speed clutch decreased, while oil pressure of reverse gear brake is rising. After both reaching about 6bar, two actuators oil pressure together rise smoothly under the action of stationary combination valve, until two actuators are fully engaged. Reverse gear brake rapidly separates and low speed clutch remains jointed state when given the signal of reverse gear back to neutral gear at 10 seconds. There does not exist overlap phenomenon in the hydraulic characteristic curves of each actuator based on simulation results, so that the combination and separation of actuator is more convenient and reliable, which ensures that the vehicle can achieve fast and accurate shift operation during the starting process, and which indicates that the design of the shift operation system has good starting performance.

IV. CONCLUSIONS

(1) Shift operation system of some heavy-duty hydrodynamic mechanical automatic transmission is designed and its working principle is explained, while a simplified mathematical model of the system is established, then analyzing the dynamic characteristics. The results show that the designed operation system is reasonable.

(2) Simulation model of shift operation system is established based on AMEsim software platform. The simulation results show that shift timing of shift operation system is correct, meets the design requirements, and joint oil pressure is stable during shifting process so that shifting is stable, while avoiding “power interruption” or “double shifts”. The established simulation platform provides a reasonable solution for study of shift schedule and shift quality.

(3) In this paper, the method to establish dynamic simulation model can be used by changing one or more parameters to change the dynamic characteristics of hydraulic system, which saves a lot of manpower, resources and time. This model can provide the basis for the design and optimization of automatic transmission hydraulic system.
REFERENCES


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