STARTING CONTROL STRATEGY OF HMCVT BASED ON FUZZY CONTROL ALGORITHM

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Abstract - The theoretical research and simulative analysis on the dynamic characteristics of the hydraulic parts of the hydro mechanical continuously variable transmission (HMCVT) were investigated in order to improve the starting performance of HMCVT. A fuzzy identification of driver's driving intention and variable rate of displacement ratio of variable displacement pump controller were designed. The fuzzy control query table of the driver's driving intention, variable pump displacement ratio change rate and fuzzy control query table of fuzzy controller were obtained using the Tools' vector test generation method. The simulation model of the vehicle starting stage was established to analyze the simulation results. The results showed that the control strategy of the vehicle starts to meet the needs of the vehicle starting control.

Keywords: Fuzzy control; HMCVT; Starting control strategy; Emulation.

1. Introduction

Hydro-mechanical continuously variable transmission (HMCVT) can achieve mechanical power flow and hydraulic power flow coupling through the planetary gear train[1], which makes the continuous gearbox ratio changes in a wide range, and finally achieve continuously variable. The starting control technology of HMCVT is one of the key technologies of gearbox, which is of great significance to the development and industrialization of HMCVT technology[2].

At present, many scholars and experts at home and abroad have done a lot of analysis and research on Hydro-mechanical continuously variable transmission. Forental V I, Forental M V and Nazarov F M[3] used experimental and modeling methods to study the dynamic characteristics of proportional control hydraulic transmission. Considering the characteristics of pump station, hydraulic cylinder, inertia load and proportional valve, a mathematic model was established. However, the accuracy of the test results was influenced by the position feedback loop, and the model was lack of continuity. The transfer function, which can characterize the dynamic behavior of the hydraulic system, was based on the static flow characteristics of the test and the spool displacement of the input current. However the function lacks logical rigidity. Dasgupta K, Mandal SK, and Pan S[4] studied the performance of a dynamic analysis open (HST) drive and considered a low-speed, high-torque (LSHT) radial piston motor as a drive, combined with a key graph simulation technique Dynamic model. However, the system state equation must be drawn by the key graph model, whose process was cumbersome and complex. The bond graph software package used for simulation is very expensive and is not easy to obtain. In contrast, MATLAB Simulink module is a set of modeling, simulation, analysis in one of the professional software, the use of coherence and consistency, the data results of high reliability. Dasgupta K, Watton J and Pan S[5] studied the dynamic performance of an axial piston motor for servo valve control in the drive system, which uses a bond graph for modeling and simulation techniques. The various loss coefficients of the motor were determined from the steady-state pressure. The simulation results obtained under different operating conditions were experimentally verified and the transient response of the system was predicted by a validated model. Zhao Z, He L[6] based on the minimum principle of the optimal control method to solve the independent research and development of five-speed dry dual-clutch transmission of the starting process of the problem (DCT). For the sliding phase, the minimum principle and the improved engine constant speed control are adopted to obtain the optimal clutch and engine torque and rotation speed, the minimum impact strength and friction work as the optimization index. During steady-state operation, the engine torque was converted to the drive demand level. The MATLAB simulation platform simulates the initial stage of the DCT vehicle, and simulates and analyzes the different engine speed and the intention of the driver.

Considering the research of these documents, we focus on the simulation and analysis of the dynamic characteristics of the segmentation strategy, and the research on the starting performance of the hydraulic mechanical continuously variable transmission is few and less systematic.

Therefore, this paper studies the dynamic characteristics of the gearbox starting from the theoretical research and simulation analysis of the new type of tractor hydraulic compressor continuously developed, and uses the fuzzy control method to develop the starting control strategy of the hydraulic mechanical continuously variable transmission. The gearbox fuzzy controller, and carried out the simulation analysis.

2. HMCVT Model

Fig. 1 showed hydro-mechanical continuously variable transmission schematic diagram. The gearbox consists of a pure hydraulic start-up section HM0 and four hydraulic mechanical operating sections HM1 to HM4 respectively, by five wet clutch c0 ~ c4 independent control.

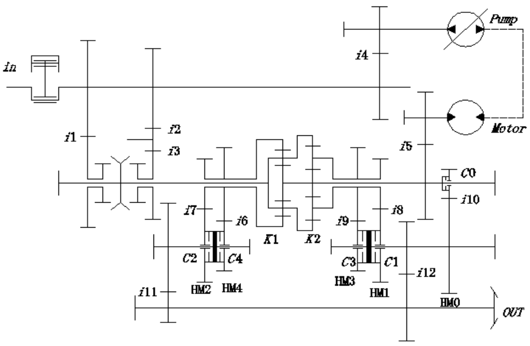

Figure 1 Transmission schematic diagram of hydro-mechanical continuously variable transmission

The starting of a Hydro-mechanical continuously variable transmission was achieved by controlling the engagement/disengagement of the clutch so that the engine power was all output through the hydraulic section (Fig.1). At first time step, the Hydro-mechanical continuously variable transmission clutch c0 clutch, the HMCVT work in the pure hydraulic section. Gradually change the displacement of the hydraulic pump, the pump-motor speed ratio ε from 0 gradually increased. With the transmission ratio gradually reduced, the speed increased, and the completion of the tractor start. When meet the change condition, C0 clutch is disconnected, C1 clutch is engaged, change into HM1 hydraulic machinery section.

3. Study on Dynamic Characteristics of HMCVT

3.1 Systematic model analysis

The core of the starting control of hydraulic mechanical continuously variable transmission is the control of the hydraulic speed regulation system which included variable displacement pump and quantitative motor.

Therefore, before analyzing and designing the starting controller, the dynamic mathematical model of the hydraulic volume governor system should be established. In this study, the step response test combined with the MATLAB system identification toolbox was used to identify the approximate transfer function between the speed ratio of the gearbox and the exciting current of the variable pump.

The variable pump and motor quantitative volumetric speed control system does not make pressure and overflow throttling loss, and less heat production, constant output torque, wide speed range, and high transmission efficiency advantages, so choose this method in hydraulic machinery stepless gearbox, the composition of the system is shown in Fig.2:

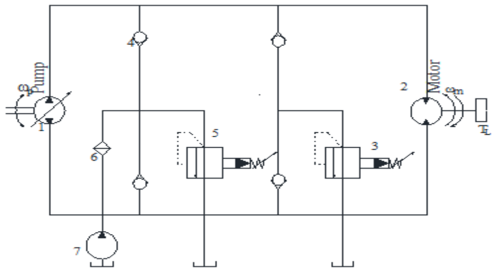

Figure 2 HMCVT volume timing system

After simplifying the hydraulic part model, the kinematics equation and force balance equation of hydraulic mechanical continuously variable transmission were established as follows: [7-10]

$$q_{\mathrm{pmax}}\frac{\omega_{\mathrm{e}}}{i_4}\sigma = q_{\mathrm{m}} i_5 i_6 i_{11} i_{13}\omega_{\mathrm{o}} + \frac{C_{\mathrm{s}}(q_{\mathrm{pmax}}+q_{\mathrm{m}})}{u}(p_{\mathrm{H}}-p_{\mathrm{L}}) + \frac{V_0}{\beta_{\mathrm{e}}}\frac{\mathrm{d}(p_{\mathrm{H}}-p_{\mathrm{L}})}{\mathrm{d}t} \quad (1)$$

$$(p_{\mathrm{H}}-p_{\mathrm{L}})q_{\mathrm{m}} = (\frac{J_{\mathrm{o}}}{i_5 i_6 i_{11} i_{13}} + J_{\mathrm{m}})\frac{\mathrm{d}\omega_{\mathrm{o}}}{\mathrm{d}t} + f_{\mathrm{m}} i_5 i_6 i_{11} i_{13}\omega_{\mathrm{o}} + \frac{T_{\mathrm{o}}}{i_5 i_6 i_{11} i_{13}} + C_{\mathrm{f}} q_{\mathrm{m}}(p_{\mathrm{H}}-p_{\mathrm{L}}) \quad (2)$$

In the equation: ω_e is the rotational speed of the engine output shaft (rad·s^{-1}); ω_o is the rotational speed of the gearbox output shaft (rad·s^{-1}); V_0 is the working volume of the oil (m^3); β_e is the elastic modulus of the oil; Jo is the load inertia of the gearbox output shaft (kg.m^2); J_m is the moment of inertia (kg·m^2); f_m is the viscous damping of the dosing motor.

Divide both sides of the equation by ω_e, the equation is shown in formula (3) and (4):

$$q_{\mathrm{pmax}}\frac{\sigma}{i_4} = q_{\mathrm{m}} i_5 i_6 i_{11} i_{13}\varepsilon + \frac{C_{\mathrm{s}}(q_{\mathrm{pmax}}+q_{\mathrm{m}})}{u\omega_{\mathrm{e}}}(p_{\mathrm{H}}-p_{\mathrm{L}}) + \frac{V_0}{\beta_{\mathrm{e}}\omega_{\mathrm{e}}}\frac{\mathrm{d}(p_{\mathrm{H}}-p_{\mathrm{L}})}{\mathrm{d}t} \quad (3)$$

$$\frac{(p_{\mathrm{H}}-p_{\mathrm{L}})q_{\mathrm{m}}}{\omega_{\mathrm{e}}} = (\frac{J_{\mathrm{o}}}{i_5 i_6 i_{11} i_{13}} + J_{\mathrm{m}})\frac{\mathrm{d}\varepsilon}{\mathrm{d}t} + f_{\mathrm{m}} i_5 i_6 i_{11} i_{13}\varepsilon + \frac{T_{\mathrm{o}}}{\omega_{\mathrm{e}} i_5 i_6 i_{11} i_{13}} + \frac{C_{\mathrm{f}} q_{\mathrm{m}}(p_{\mathrm{H}}-p_{\mathrm{L}})}{\omega_{\mathrm{e}}} \quad (4)$$

In the formula ε is defined as the speed ratio: $\varepsilon = \frac{\omega_o}{\omega_e}$

Engine speed, charge pressure and load were unchanged, The Laplace transform of the pure hydraulic section of the transmission was carried out by using the kinematic equation and the force balance equation, as shown in formula (5) and (6):

$$q_{pmax}\frac{\Sigma}{i_4} = q_m i_5 i_6 i_{11} i_{13} \mathrm{E} + \left[\frac{C_s(q_{pmax}+q_m)}{u\omega_e} + \frac{V_0}{\beta_e \omega_e}s\right]P_H \qquad (5)$$

$$\frac{P_H q_m}{\omega_e} = \left(\frac{J_o}{i_5 i_6 i_{11} i_{13}} + J_m\right)s\mathrm{E} + f_m i_5 i_6 i_{11} i_{13}\mathrm{E} + \frac{C_f q_m P_H}{\omega_e} \qquad (6)$$

Finishing to be a stepless gearbox hydraulic speed ratio ε for the output, the transfer function of the hydraulic displacement system with the displacement ratio σ of the variable displacement pump is:

$$G_1(s) = \frac{\varepsilon(s)}{\sigma(s)} = \frac{m_1}{As^2 + Bs + C} \qquad (7)$$

In formula (7): $m_1 = \frac{q_{pmax}(1-C_f)q_m}{i_4}$; $A = \frac{V_0}{\beta_e}\left(\frac{J_o}{i_5 i_6 i_{11} i_{13}} + J_m\right)$; $B = \frac{C_s(q_{pmax}+q_m)}{u}\left(\frac{J_o}{i_5 i_6 i_{11} i_{13}} + J_m\right) + \frac{V_0 f_m i_5 i_6 i_{11} i_{13}}{\beta_e}$;

$C = \frac{C_s(q_{pmax}+q_m) f_m i_5 i_6 i_{11} i_{13}}{u}$。

The speed regulation system of the pure hydraulic section of the gearbox is composed of two steps of inertia link and proportional link.

PLC by adjusting the size of the current, controlling the proportion of the variable displacement pump solenoid valve to control the pump - the motor displacement, obviously the variable pump displacement adjustment mechanism can be equivalent to a first-order low-pass filter, the transfer function is G_2 (S):

$$G_2(s) = \frac{m_2}{Ts+1} \qquad (8)$$

3.2 Analysis of test results

Process at least two or more sets of similar test data as the result of system identification. After the test, the gearbox speed ratio response sequence is shown in Fig. 3:

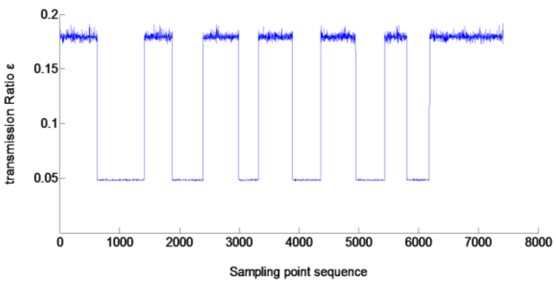

Figure 3 Transmission Ratio response sequence

System identification of variable-pump excitation current input and gear-box speed ratio output are done by MATLAB system identification toolbo. MATLAB system identification toolbox had a simple and intuitive graphical user interface (GUI) and powerful identification function library which effectively improve the accuracy of system identification and reduce the workload of system identification. Using the system identification toolbox, the steps of system identification modeling are following:

① Acquisition of test data;
② Pretreatment of data;
③ Selection of model structure;
④ Parameter estimation;
⑤ Model checking. If the resulting model is not ideal, then the above steps can be repeated until the required model is reached [11-12].

The identification and modeling of the system was carried out by using step response identification and Matlab system identification toolbox. The accuracy and recognition efficiency are high, the calculation process is simple and clear, the graphical interface is intuitive and the hydraulic system model of hydro-mechanical CVT is identified as:

$$G(s) = \frac{0.253}{0.00072s^3 + 0.01341s^2 + 0.16653s + 1} \qquad (9)$$

The obtained hydraulic system model well reflected the dynamic characteristics of the pure hydraulic starting process of the transmission, which laid an important foundation for the study of the starting characteristics of the gearbox, the analysis and design of the starting process controller[13].

4. Design of Fuzzy Controller for Gearbox Start-up

4.1 Design of Fuzzy Controller for Variable Pump Displacement Ratio

The control of vehicle start-up should not only take the quantitative driving intention as the control input, but also consider how to prevent the engine from being out of power or the high output speed.

The deviation degree of the engine output rotational speed relative to the target rotational speed is taken as the control parameter of the variable rate of the variable displacement pump. When the engine output speed is lower than the target speed, the displacement rate of the variable displacement pump should be reduced so that the load of the engine output bearing is too large to cause the engine output speed to decrease sharply and flame out. The greater the deviation of the output speed is from the target speed, the slower the rate of change of the variable displacement ratio is. When the engine speed is higher than the target speed, the variable pump displacement ratio should speed up, the speed deviation from the target speed is faster than the variable pump displacement ratio of the rate of change[14] .

(1) The language set of the control variables is described as follows: Driving Intent I Fuzzy Language Variables:{very slow, slow, normal, fast, very fast};Corresponding to the fuzzy subset:{VS、S 、M、B、VB}; The engine speed deviation from the target speed of the fuzzy language variables:{ very small, small, small, medium, large, large, very large };Corresponding to the fuzzy subset:{VS、S、LS、M 、LB、B、VB}; Variable Pump Displacement Rate Change Rate Fuzzy Language Variables:{ very small, small, small, medium, large, large, very large };Corresponding to the fuzzy subset:{VS、S、LS、M 、LB、B、VB}.

(2) The fuzzy domain of the control variables is as follows: Driving intent I:{0,1,2,3,4,5,6,7,8}; The engine speed deviates from the target rotational speed Id:{-5,-4,-3,-2,-1,0,1,2,3,4,5}; The rate of change of the variable pump displacement ratio {0,1,2,3,4,5,6,7,8,9,10}.

(3) Quantification Factor and Scaling Factor Determination: Since the driving intention I is itself a fuzzy quantity, it can be set to a scale factor of 1. The basic domain where the engine speed deviates from the target speed is [-0.5, 0.5]. Then the quantization factor is $k_3 = \frac{5}{0.5} = 10$. The actual rate of change of the variable pump displacement ratio is [0, 0.01]. The scale factor is $k_c = \frac{0.01}{10} = 0.001$.

(4) The determination of membership function: The membership function of each fuzzy quantity is set to a triangle according to the sensitivity and the control precision.

(5) According to the change rate control strategy of the variable pump displacement ratio during the starting process of the vehicle, the variable rate control table of variable displacement ratio can be formulated in Table 1.

Table 1 Variable pump speed ratio change rate fuzzy control rule table

Rate of change		The engine speed deviates from the target rotational speed						
		VS	S	LS	M	LB	B	VB
Driving intent	VS	VS	VS	S	LS	LS	LS	M
	S	S	LS	LS	LS	LS	LS	M
	M	LS	M	M	M	LB	LB	LB
	B	M	LB	LB	B	B	B	B
	VB	M	LB	B	VB	VB	VB	VB

4.2 Simulation results and analysis

We adjust the opening of the accelerator pedal and make it at 25%, 50% and 75%, and then start analysis the simulation of the vehicle starting. For comparison purposes, the simulation data for the output speed of the transmission at all throttle pedal opening and the simulation data for the engine speed variation were shown in Figs. 4 and 5. The speed of the transmission output and the speed of the engine at the start of the vehicle at the different degrees of the throttle pedal opening are shown in Figs. 4 and 5 too.

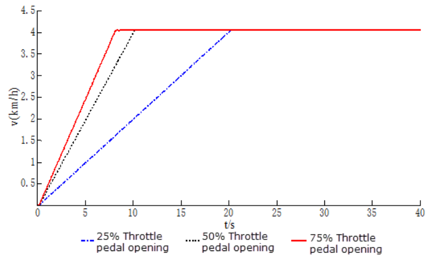

Figure 4 Vehicle speed

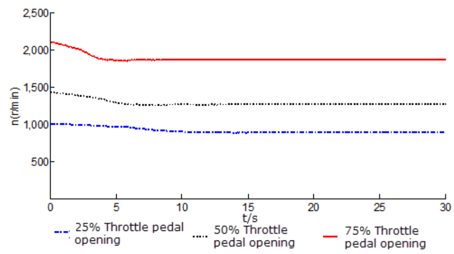

Figure 5 Engine speed

As shown in Fig. 4, the starting time at 25% accelerator pedal opening was 21 s, the starting time at 50% accelerator pedal opening was 11 s, and the starting time at 75% accelerator pedal opening was 7s. Compared with different accelerator pedal opening under the starting time we can see that with the accelerator pedal opening increases, the vehicle started getting shorter and shorter, which meets the requirements of vehicle start control. We can see from Figure 5, the opening of the engine in 25% accelerator pedal speed reduced to 115r· min^{-1}. In the 50% opening of engine throttle pedal speed reduced to 163r· min^{-1}. In the 75% under the accelerator pedal opening speed was reduced to

240r· min^{-1}. Compare the different accelerator pedal opening engine the speed drop, we can know with the accelerator pedal opening increases, the engine speed down is increasing[15] .

At different throttle pedal opening, the speed of the vehicle gradually increased from 0 to 4 r· min^{-1}, and with the increase of the accelerator pedal opening, the vehicle speed increased faster, which can meet the requirements of the initial stage of fuzzy control. There was a dynamic adaptation process between the power provided by the engine and the power required by the vehicle at the initial stage of the start, which resulted in a decrease in the engine speed. The maximum speed was reduced to 240 r· min^{-1}. The velocity drop is gradually increased as the accelerator pedal opening increases.

5. Conclusions

According to the kinematic equation and the force balance equation of the hydraulic system, the transfer function model was established, which showed that the speed governing system was composed of second-order inertia link and proportional link. The dynamic mathematical model of the hydraulic volume speed control system was a 3 system. Finally, the parameters of the dynamic characteristic transfer function model were obtained by step response test and system identification.

The control strategy of the gearbox was established by using the fuzzy control method, and the fuzzy control query table of the fuzzy controller was obtained by using the method of generating the vector test.

The simulation model was established and the results of the simulation model were analyzed. The results showed that the control strategy can meet the needs of vehicle start control.

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