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MECHANICAL AND AERONAUTICAL ENGINEERING, THERMODYNAMICS

Research on coordinated control of hydro-mechanical continuously variable transmission tractor mode switching based on model reference adaptation

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Abstract. The difference between the speed of the hydraulic speed control system and the mechanical transmission speed is too large during the mode-switching process of the equipped HMCVT tractor, which leads to the deterioration of the smoothness of the mode-switching process. The paper proposes a mode-switching coordination control strategy based on the adaptive model reference. Based on the mode-switching process, the mathematical model of HMCVT mode-switching is constructed. With the output speed of the hydraulic system as the reference model, the output speed of the mechanical transmission as the control object, and the output speed of the mechanical transmission following the output speed of the hydraulic system as the target, the design model reference adaptive controller, based on the MATLAB simulation platform for the simulation test of the control strategy is presented. The results show that when switching from H mode to HM1 mode, the maximum jerk is reduced by 70.3% and the slip friction work is reduced by 28.6%, and when switching from HM1 mode to H mode, the maximum jerk is reduced by 67% and the slip friction work is reduced by 28.9% compared to the use of the rule-based control strategy.

Keywords: hydro-mechanical continuously variable transmission; model reference adaptive control; mode-switching; coordinated control; smoothness.

1. INTRODUCTION

The hydro-mechanical continuously variable transmission (HMCVT) consists of a mechanical shunting mechanism, a hydraulic speed control system, a mechanical transmission system, and a mechanical converging mechanism, etc. It relies on the hydraulic speed control system to achieve stepless speed control in the working process and switches the segments to achieve the designed speed ratio range by controlling different clutches, which helps attain the switching of segments and, consequently, the stepless speed change within the range of the designed speed ratio. This transmission mode combines the characteristics of mechanical transmission and hydraulic transmission, has the advantages of small impact, high efficiency, and large power transmission [1–3], and is suitable for high power, steplessly variable speed demand tractor, and other agricultural operations.

HMCVT relies on the clutch to achieve segment shifting, and the power must be transferred from the current segment clutch to the target segment clutch [4], so the HMCVT segment shifting process will produce impacts that affect the smoothness of segment shifting. To improve the quality problem of tractor segment switching, some scholars have researched the mode-switching process. Wang *et al.* [5] designed an adaptive fuzzy iterative control strategy to optimize the clutch engagement process in the segment switching process, which reduces the deviation of the clutch switching process. Ahn *et al.* [6–11] proposed to control the overlap time of the two clutches work and study the effect of overlap time on the mode switching quality in clutch switching. Wang *et al.* [12, 13] found that delaying the action time of the clutches to be disengaged and the clutches to be engaged before the theoretical shift point helps to improve the smoothness of shifting. Cao *et al.* [14] established an HMCVT shifting model and analyzed the dynamic characteristics of the shifting process of HMCVT, which provided suggestions for improving the quality of shifting and formulating the control strategy. Li *et al.* [15] proposed an integrated shift control strategy using dual-loop control (PID model predictive control) for the hydraulic speed control system and PID control for the clutch system to achieve precise control of the transmission during shifting. Lu *et al.* [16] analyzed the effect of system oil pressure and clutch operation on the wet clutch binding characteristics, and the test showed that precise control of clutch oil pressure can reduce the impact of the system change section. Zhu *et al.* [17, 18] used orthogonal experimental range analysis and analysis of variance (ANOVA) to optimize the quality of mode switching by taking four clutches as the orthogonal experimental range analysis factors, and output shaft deceleration amplitude, dynamic load factor and impact, and shortening the switching time as the evaluation indexes, and verified the validity of the proposed orthogonal experimental scheme through simulation and HIL test. Xi *et al.* [19] analyzed the factors affecting the fluctuation of output speed and proposed an integrated control strategy combining

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the speed and displacement ratios of the hydraulic speed control system and verified it by using the tractor acceleration mode and step load perturbation mode, which effectively reduces the impact of the shift. Zhao *et al.* [20] proposed a global speed prediction model based on personalized driving characteristics. Combined with the global speed prediction model of personalized driving characteristics and multi-source information, the working mode and multi-power source torque distribution of hybrid transmission are determined. Liu *et al.* [21] proposed a dual-roll coordinated control strategy integrating Kalman filtering and model predictive control, which utilizes the filtering characteristics of Kalman filtering to mitigate the shifting perturbations in HMCVT to dynamically correct the information of the input variables of the model predictive control in real time, and effectively improve the quality of shifting. Liu *et al.* [22] analyzed the HMCVT shifting process based on the theory of bonding diagram, and the simulation found that prolonging the clutch charging time can reduce the dynamic load on the output shaft. Zhu *et al.* [23] proposed the extreme difference method to analyze the advantages and disadvantages of the factors affecting the clutch loading time.

Pan *et al.* [24] established a nonlinear dynamic model for the shift of HMCVT and simulated a variety of transmission physical parameters and shift timing, which showed that optimizing the transmission physical parameters and shift timing can improve shift quality. Jiang *et al.* [25] proposed a method to optimize the shift smoothness by adjusting the pressure of the hydraulic system and found that the pressure adjustment system can significantly reduce the angular acceleration of the output shaft and improve the shift smoothness. Bao *et al.* [26] investigated the effect of clutch oil pressure change on the rotational speed by studying the dynamic characteristics of hydraulic CVT shifting to provide a basis for further research on shifting smoothness. Wei *et al.* [27] proposed a speed ratio control method applicable to equidistant two-segment HMCVTs to achieve tracking control of the target speed ratio, but there is a steady-state error in tracking the slope speed ratio. Cao *et al.* [28] proposed a synchronous control method with the output angular velocity of the hydraulic system and the clutch torque as variables to achieve compensated control of the clutch torque and effectively eliminate the effect of segment change. Yang *et al.* [29–32] proposed a method to jointly control the displacement ratio and the clutch in the process of segment change to achieve the compensation of the power transmitted during the clutch engagement process, thus reducing the degree of decrease in the output shaft speed and torque and improving the smoothness of segment change. Lu *et al.* [33, 34] proposed a terminal sliding mode control method for the deviation problem in the tracking control of clutch oil pressure in HMCVT segment change to achieve the tracking control of clutch oil pressure and torque and improve the quality of segment change.

However, the speed mismatch between the HMCVT hydraulic speed control system and the mechanical drive system during convergence results in large shocks, making it impossible to achieve an excellent quality section change. Although the above method can indirectly improve the speed difference during mode switching by controlling parameters such as the clutch, this indirect control strategy increases the complexity of the speed control. Therefore, it is necessary to propose a mode-change control strategy that directly controls the speed of the hydraulic speed control system and the speed of the mechanical transmission system to further improve the HMCVT shift quality.

This paper takes the HMCVT-equipped tractor as the research object, establishes a mathematical model of the mode-switching process according to its working principle, analyzes the laws of the hydraulic speed control system of the HMCVT and the mechanical transmission when the two converge, and proposes a coordinated control strategy of mode switching based on the idea of direct control. It controls the mechanical transmission (the engine speed transferred to the mechanical structure) by the model-referenced adaptive control strategy and reduces the difference between the two speeds when switching segments and verifies it by simulation and test. The model-referenced adaptive control strategy is used to control the mechanical transmission (the speed transferred from the engine to the mechanical structure) to follow the output speed of the hydraulic speed control system, to reduce the difference in speed between the two during the shift, so as to achieve a smooth shift during the shift, and is verified by simulation and experiment, and finally compared with the rule-based control strategy used during the shift.

2. HYDRO-MECHANICAL CONTINUOUSLY VARIABLE TRANSMISSION OPERATING PRINCIPLE

Figure 1 illustrates the topological structure of the tractor equipped with HMCVT. When the tractor is in operation, the engine power is transmitted to the HMCVT, the input power is conveyed to the hydraulic speed control system and mechanical transmission system via the mechanical shunt mechanism. The HMCVT mode switching is achieved through the clutch, brake, and planetary rows, while the power output by the hydraulic speed control system and mechanical transmission system is merged through the mechanical convergence mechanism to generate a continuously variable and continuously adjustable output

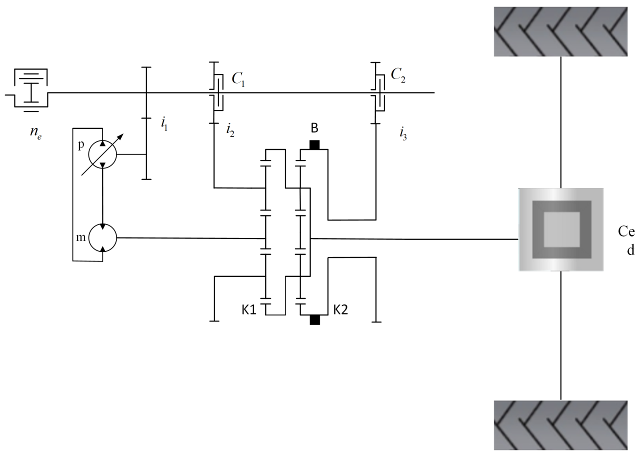


Fig. 1. The topological structure of the tractor equipped with HMCVT (C1, C2 represent different clutches; p and m are the variable pumps and variable motors; B is the brake; K1,K2 represent different the planet rows; n_e is the input speed of the engine; nout is the output speed of the HMCVT; i_1, i_2, i_3 represent different gear pairs)

speed by adjusting the displacement ratio of the hydraulic speed control system. The output speed is continuously and steplessly altered by adjusting the displacement ratio of the hydraulic speed control system.

The HMCVT is capable of operating in three distinct modes: pure hydraulic mode (H mode), hydro-mechanical mode 1 (HM1 mode) and hydro-mechanical mode 2 (HM2 mode). The clutch and brake are employed to facilitate mode switching, allowing the HMCVT to transition between these three operational modes. In H mode, the brake B lock, and clutches C1 and C2 are in a state of separation and do not access the convergence transfer route. This means that the two sets of planetary rows in the power transfer route do not play a role in the divergence of the vehicle role when it is started purely hydraulically. The planetary rows of K2 directly receive the hydraulic system in the quantitative motor to transfer the power to the planetary rows of the planetary rack K2 to the back of the output. In the forward working condition of the HMCVT, the hydraulic machinery HM1 section is in a state of release for brake B, while clutch C1 is in a combined state of access to the convergence of the transmission route. In contrast, clutch C2 is in a state of separation and does not have access to the convergence of the transmission route. The planetary row K1 is in a state of convergence of power through the planetary row K2, which is connected to the backward output. When HMCVT is in forward working condition, the hydraulic machinery HM2 section is engaged. At this time, brake B release, clutch C2 is in the combined state, allowing access to the convergence of the transmission route. Clutch C1 is in a disengaged state, preventing access to the convergence of the transmission route. At this time, the planetary row K2 will be converged power through the planetary row K2 planetary frame to the rear output. The operating state of the clutch and brake is presented in Table 1 for each mode.

Table 1
Clutch and brake operating status in different modes

Model	Clutch C1	Clutch C2	Brake B
H	–	–	+
HM1	+	–	–
HM2	–	+	–

The “+” and “–” symbols indicate engagement and disengagement, respectively.

3. MODELLING OF HYDRO-MECHANICAL CONTINUOUSLY VARIABLE TRANSMISSION MODE SWITCHING

3.1. Engine speed model

The differential equation for diesel engine speed is

$$\dot{n} = \frac{T_i - T_l - T_f}{I_e}, \tag{1}$$

where n is the speed of the diesel engine; T_i is the indicated torque of the diesel engine; T_l is the load torque; T_f is the friction torque of the diesel engine shaft;$I_e$is the rotational moment of inertia of the diesel engine in operation.

The indicated torque is generated by fuel combustion in each cylinder and is calculated as

$$T_i = \frac{30}{\pi} \times \frac{\eta_i H_u N_{cy1}}{60 N_{st}} \times m, \tag{2}$$

where m is the mass of fuel injected into each cylinder per cycle; η_i is the total indicated efficiency, which is a function of diesel engine speed and air-fuel ratio; H_u is the low calorific value of diesel fuel; N_{cy1} is the number of cylinders; the value of N_{st} depends on the type of diesel engine.

The frictional torque is calculated using the following empirical formula

$$T_f = \frac{30 V_d}{\pi} \left(c_1 + c_2 n\right), \tag{3}$$

where V_d is the total displacement and c_1, c_2 are the test coefficients.

The differential equation for the rotational speed is

$$\dot{n} = -\left(\frac{30}{\pi}\right)^2 \left(\frac{c_2 V_d n}{I_e}\right) + \left(\frac{30}{\pi}\right)^2 \left(\frac{1}{I_e}\right) \left(\frac{\eta_i H_u N_{cy1}}{60 N_{st}}\right) m - \left(\frac{30}{\pi}\right)^2 \left(\frac{c_1 V_d}{I_e}\right) - \left(\frac{30}{\pi}\right) \left(\frac{T_i}{I_e}\right). \tag{4}$$

After transforming and considering the change in the amount of fuel given by the control oil supply valve to the corresponding speed of the diesel engine output process has a time lag, the diesel engine output speed of the transfer function is obtained as

$$\frac{\varphi(s)}{\eta(s)} = \frac{1}{\left(T_d s + T_g\right)(1 + \tau s)}, \tag{5}$$

where T_d is the time acceleration constant of the diesel engine; T_g is the self-stabilizing parameter for diesel engine driven load operation; τ is the time lag constant of the diesel engine.

3.2. Transmission shaft model

Assuming that the fixed-axis drive of the transmission system is a rigid body, ignoring the loss of efficiency of the hydraulic speed control mechanism, and equating the planetary gear components with the concentrated mass and damping, the mathematical model of the HMCVT mode-shifting process is established.

Input shaft:

$$T_{\text{in}} = I_{\text{in}} \dot{\omega}_{\text{in}} + b_{\text{in}} \omega_{\text{in}} + T_{C1} + T_{C2} + T_P / i_1 . \tag{6}$$

Planetary row K1 planetary carrier:

$$T_{cr1} = T_{C1} - b_1 \omega_{cr1} - I_1 \dot{\omega}_{cr1} . \tag{7}$$

Output shaft:

$$T_{cr2} = T_{\text{out}} + b_{\text{out}} \omega_{\text{out}} + I_{\text{out}} \dot{\omega}_{\text{out}} . \tag{8}$$

Planetary row K1 sun wheel:

$$T_{s1} = T_m - b_1 \omega_{s1} - I_3 \dot{\omega}_{s1} . \tag{9}$$

Planetary row K2 gear ring:

$$T_{r2} = T_{C2} - b_2\omega_{r2} - I_2\dot{\omega}_{r2} - T_B\,. \tag{10}$$

Quantitative motor shaft:

$$T_m = b_3\omega_m + I_3\dot{\omega}_m\,, \tag{11}$$

where T_{C1}, T_{C2}, T_p, T_m, T_B are the clutch C1 torque, clutch C2 torque, variable pump p torque, quantitative motor m torque and brake B torque, respectively; T_{s1}, T_{cr1}, T_{r1} are the sun wheel torque, the planetary carrier torque and the ring torque of the planetary row K1, respectively; T_{s2}, T_{cr2}, T_{r2} are the planetary row K2 sun wheel torque, planetary carrier torque and ring torque, respectively; ω_{r2} is K2 ring angular velocity.

3.3. Mathematical model of hydraulic speed control system

HMCVT uses a variable pump-quantum motor system described below. The transfer function of the motor output speed to the swashplate inclination of the variable pump is

$$\frac{\omega_m(s)}{\gamma(s)} = \frac{K_{qp}/D_m}{\dfrac{s^2}{f^2} + \dfrac{2\xi}{f} + 1}\,, \tag{12}$$

where ω_m represents the output speed of the fixed motor; γ represents the swashplate inclination of the variable pump; K_{qp} represents the flow gain of the variable pump; D_m represents the displacement of the fixed motor; f represents the hydraulic intrinsic frequency; ξ represents the hydraulic damping ratio.

Under external load conditions, the transfer function relating the motor output speed to the external load torque is given by

$$\frac{\omega_m(s)}{T_m(s)} = \frac{-\dfrac{C}{D_m^2}\left(1 + \dfrac{V}{\beta_e C}\right)}{\dfrac{s^2}{f^2} + \dfrac{2\xi}{f}s + 1}\,, \tag{13}$$

where T_m represents the external load moment; C represents the oil leakage coefficient; β_e represents the modulus of elasticity of the oil; V represents the total volume of the working chamber of the pump and motor.

By combining the transfer function of the valve-controlled hydraulic cylinder module with the swashplate tilt module of the variable pump in the speed control system, the overall transfer function of the hydraulic speed control system can be expressed as

$$\frac{\omega_m(s)}{T_m(s)} = \frac{\dfrac{C}{D_m^2}\left(1 + \dfrac{V}{\beta_e C}s\right)K_{bv}K_qK_\varphi K_{qp}}{s\left(\dfrac{s^2}{f^2} + \dfrac{2\xi}{f}s + 1\right)R_P D_m + K_{bv}K_qK_\varphi K_{qp}}\,, \tag{14}$$

where K_{bv} is the proportional valve gain coefficient; K_q is the flow rate gain coefficient; K_φ is the variable pump tilt angle coefficient; R_p is the variable pump swashplate tilt angle adjustment of the area of the hydraulic cylinder piston.

3.4. Mathematical modelling of clutches

When the clutch is in slipping friction, the transfer torque T_c is

$$T_c = \mu z A p(t)\,\mathrm{sign}\,(\omega_{zd} - \omega_{cd})\,, \tag{15}$$

where μ for the friction coefficient of the friction element; z for the number of clutch discs; A for the area of clutch discs; r for the action radius of the clutch discs; sign( ) for the sign function; $P(t)$ for the clutch oil pressure change with time; ω_{zd}, ω_{cd} for the angular velocity of the clutch active and driven discs.

3.5. Mathematical modelling of planetary rows

The dynamical equations of the planetary rows are

$$\omega_{sx} + k_x\omega_{rx} = (1 + k_x)\,\omega_{crx}\,, \tag{16}$$

$$T_{sx} : T_{rx} : T_{crx} = 1 : k_x : (-1 - k_x), \tag{17}$$

where x is the number of planetary rows, $x = 1, 2$ is the characteristic parameter of planetary rows.

4. COORDINATED CONTROL STRATEGY FOR MODE SWITCHING BASED ON MODEL REFERENCE ADAPTATION

4.1. Basics of model reference adaptive control

In this paper, the state variable-based Lyapunov-MRAC control method is selected to design the controller between the controlled object and the reference model so that the output of the controlled object system follows the input of the reference model system. When applied to HMCVT mode switching, the mathematical model of the hydraulic speed control system is used as the reference model, and the engine speed model is the controlled object. When mode switching, the mechanical transmission (engine output speed) refers to the output speed of the hydraulic speed control coefficient, which can reduce the speed difference between the two when they are engaged, and thus improve the smoothness of HMCVT mode switching.

The principle of MRAC when the state variables are measurable is shown in Fig. 2.

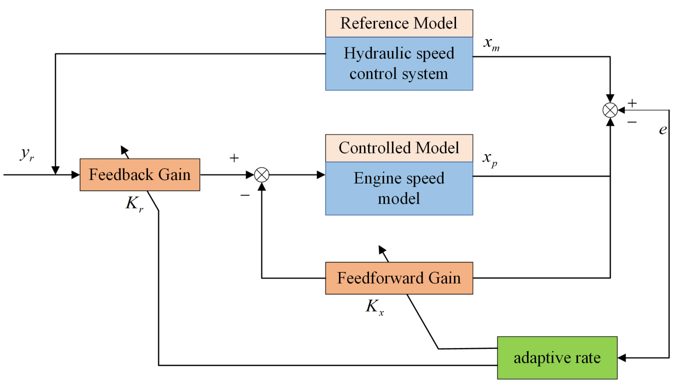


Fig. 2. MRAC principle when state variables are measurable

4.2. MRAC control rate calculation for HMCVT mode switching

In this paper, the Lyapunov-MRAC control method based on state variables is chosen, and when the state variables are measurable, the object state equation is

$$\dot{x}_p = A_p x_p + B_p u, \tag{18}$$

where A_P, B_p are $n \times n$ and $m \times m$ constant matrices; x_p is n-dimensional state vector; u is m-dimensional control vector.

Take the reference model equation of state as

$$\dot{x}_m = A_m x_m + B_m y_r\,, \tag{19}$$

where A_m, B_m and A_p, B_p are constant matrices with the same determinant; x_m is n-dimensional state vector; y_r is m-dimensional input.

From equation (14), it can be seen that the reference model transfer function is a third-order high-order transfer function, in the use of the MRAC algorithm to make the engine speed reference to the speed of the hydraulic speed control system, it is necessary to carry out an appropriate simplification of the reference object transfer function. The decision to determine the dynamic response of a system is the slowest response module in the system, the entire system can be ignored in the dynamic response of the valve control cylinder of this link, and then get the transfer function of the hydraulic speed control system

$$G_p(s) = \frac{\omega_m(s)}{T_m(s)} = \frac{105769}{s^2 + 6538s + 109266}\,. \tag{20}$$

From equation (5), it can be seen that the transfer function of the controlled object is a second-order transfer function, equation (20) after simplification of the reference object transfer function of the same order, to meet the requirements of the model reference adaptive system, after substituting the parameters to obtain the transfer function of the engine speed

$$G_m(s) = \frac{\varphi(s)}{\eta(s)} = \frac{19}{s^2 + 2.5s + 1}\,. \tag{21}$$

From equations (20) and (21)

$$A_m = \begin{bmatrix} -2.5 & -1 \\ 1 & 0 \end{bmatrix}, \quad B_m = \begin{bmatrix} 1 \\ 0 \end{bmatrix}, \quad C_m = \begin{bmatrix} 0 & 19 \end{bmatrix},$$

$$A_p = \begin{bmatrix} -6538 & -109266 \\ 1 & 0 \end{bmatrix}, \quad B_p = \begin{bmatrix} 1 \\ 0 \end{bmatrix}, \quad C_p = \begin{bmatrix} 0 & 105769 \end{bmatrix}.$$

The output of the controller can be expressed as

$$u = K_x x_p + K_r y_r\,. \tag{22}$$

The closed-loop system formed by the controller and the controlled object should ultimately be consistent with the reference model, where K_x is the feedforward gain matrix, K_r is the feedback gain matrix, set $\phi = K_x - \bar{K}_x$, $\psi = K_r - \bar{K}_r$ where $\bar{K}_x$ is the value after K_x is stabilized, $\bar{K}_r$ is the value after K_r is stabilized, ϕ is the error value of K_x, and ψ is the error value of K_r.

For the state equation consisting of the controller and the controlled object, there are

$$\begin{aligned} \dot{x}_p &= A_p x_p + B_p u \\ &= \left(A_p + B_p K_x\right) x_p + B_p K_r y_r \\ &= \left(A_m + B_p \phi\right) x_p + B_p \psi y_r + B_m y_r\,. \end{aligned} \tag{23}$$

Let the error between the reference model and the controlled object be

$$e = x_m - x_p\,. \tag{24}$$

Derivation of the error gives

$$\begin{aligned} \dot{e} &= A_m e + \left(A_m - A_p - B_p K_x\right) x_p + \left(B_m - B_p K_r\right) y_r \\ &= A_m e - B_p \phi y_r - B_p \psi y_r\,. \end{aligned} \tag{25}$$

Choose the equation for the Lyapunov function

$$V(e,\phi,\psi) = \frac{1}{2}\left(e^T P e + \phi^T \Gamma_1 \phi + \psi^T \Gamma_2^{-1} \psi\right). \tag{26}$$

Solving the Lyapunov functional equation introduces the symmetric matrices $P = P^T$, and there are $A_m^T P + P A_m = -Q$.

Derivation of the equation of the Lyapunov function yields

$$\begin{aligned} \dot{V}(e,\phi,\psi) = {} & \frac{1}{2} e^T \left(P A_m + A_m^T\right) e + 2\left(\dot{\phi}^T \Gamma_1 \phi - e^T P B_p \phi x_p\right) \\ & + 2\left(\dot{\psi}^T \Gamma_2^{-1} \psi - e^T P B_p \psi y_r\right). \end{aligned} \tag{27}$$

The adaptive rate should satisfy

$$\dot{\phi}^T = \Gamma_1^{-1} e^T P B_p x_p\,, \tag{28}$$

$$\dot{\psi}^T = \Gamma_2^{-1} e^T P B_p y_r\,. \tag{29}$$

Since $e^T P$ is computed as a scalar, then $\dot{V}(e,\phi,\psi) \le 0$, and thus $e(t)$, $K_x(t)$ and $K_r(t)$ are bounded, the overall structure consisting of controller and controlled object of the algorithm will eventually be consistent with the reference model.

4.3. Mode switching coordinated control strategy development

The mode change in the coordinated control strategy is shown in Fig. 3. The mode change in the coordinated control strategy includes the HMCVT module, throttle opening control module, segment change control module, and segment change control strategy module. The specific control strategy is as follows: HMCVT work according to the target ratio signal can determine the current pump motor system displacement ratio, control the hydraulic cylinder to adjust the size of the swashplate inclination to achieve the control of pump displacement, and then make the quantitative engine speed change, determine the output speed of the hydraulic speed control system HMCVT. Next, find the state equations of the hydraulic speed control system x_m and the state equations of the engine speed model x_p, find the adaptive rate based on the Lyapunov stability theory, and generate the control signal to be applied to the engine speed model, so that when the mode is switching, the mechanical transmission (the speed of the engine output) refers to the output speed of the hydraulic speed control coefficients; calculate the error of the system output and the speed of the hydraulic speed control system, and repeat the above steps to implement the control cycle until the system output is sufficiently close to the output of the reference model, $x_m - x_p \leq \varepsilon$, ε is a very small number.

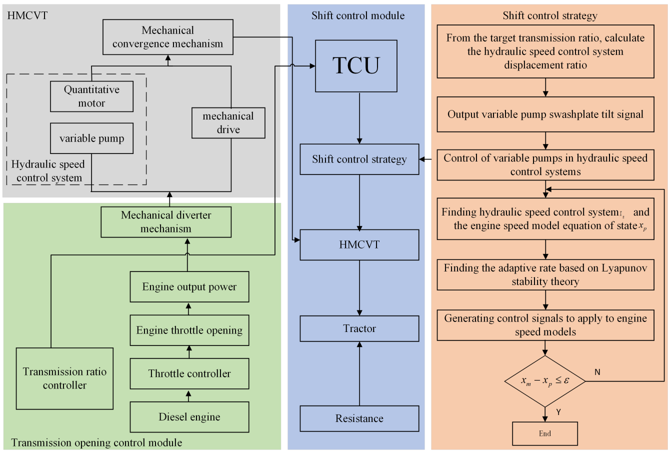


Fig. 3. Coordinated control strategy for mode switching

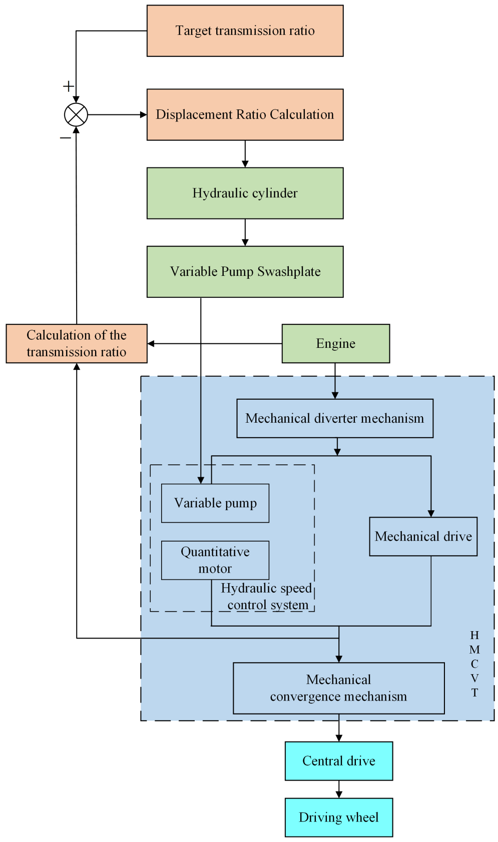


Fig. 4. Rule-based control strategy for HMCVT mode switching

5. SIMULATION ANALYSIS

5.1. Simulation programme

In this paper, a rule-based control strategy is used in comparison with a model reference adaptive based mode switching coordinated control strategy. The rule-based control strategy relies on a set of predefined rules to achieve control of the system, and this strategy enables the HMCVT changeover control, the control of the hydraulic mechanism and mechanical structure in the HMCVT is achieved by using the target transmission ratio through the logic of the predefined rules to control the HMCVT changeover.

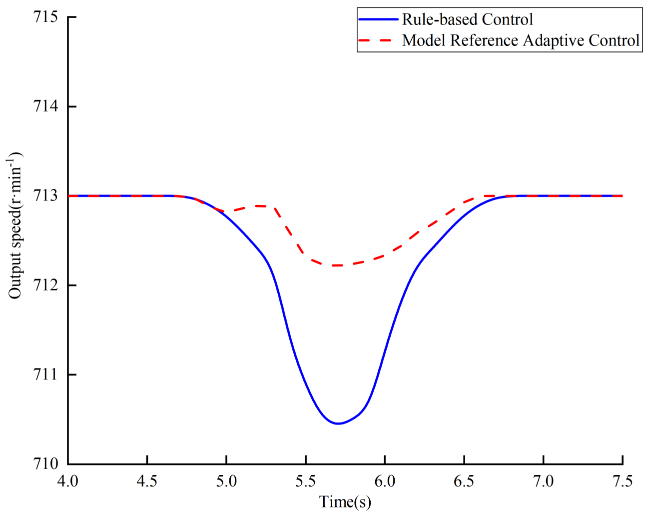
First of all, the work of HMCVT according to the target transmission ratio signal can determine the current displacement ratio of the hydraulic speed control system, control the hydraulic cylinder to adjust the size of the swashplate tilt angle to achieve the control of pump displacement, which in turn makes the quantitative engine speed change. The quantitative engine output speed and mechanical transmission converge after the output to achieve the HMCVT mode switching, and the HMCVT output speed is transferred to the central drive and drive wheels. In order to facilitate the closed-loop control of HMCVT, the engine speed and the actual speed of the hydraulic motor as the feedback quantity of control, the controller outputs the correction signal according to the deviation between the feedback value and the target value, and the rule-based HMCVT mode switching control strategy is shown in Fig. 4.

To study the effectiveness of the HMCVT mode switching smoothness control method, MATLAB software is used to perform simulation verification and control performance analysis of the change characteristics of the system output torque, vehicle speed, jerk, and slip friction work during the mode-change process.

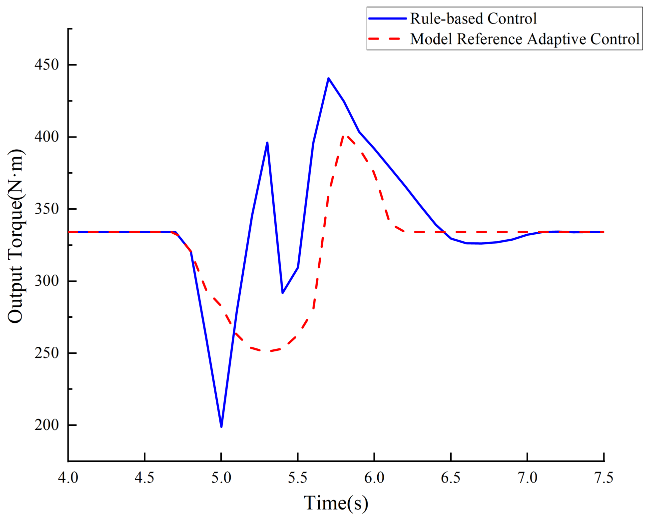
5.2. Simulation of switching from H mode to HM1 mode

The simulation results using model reference adaptive control and rule-based control strategies when switching from H mode to HM1 mode in HMCVT are shown in Fig. 5.

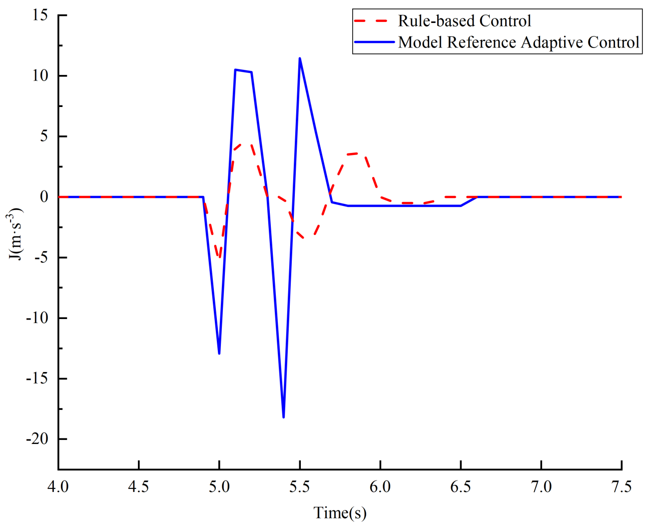
It can be seen from Fig. 5a that in the case of rule-based control strategy, there is a significant speed drop during HMCVT mode switching, with the maximum drop value of 2.58 r/min, and the output speed tends to be stable after 1.9 s. When the coordinated control strategy for mode switching based on model reference adaptive is used, the maximum fluctuation amount is 0.78 r/min, the output speed is reduced by 70%, and the switching time is 1.7 s, which is reduced by 0.2 s compared with the uncoordinated control. It can be seen from Fig. 5b that when the model reference adaptive mode switching coordinated control strategy is adopted, the engine output speed is made to follow the output speed of the hydraulic speed control system, which reduces the duration of the existence of the difference in the angular speeds of the clutch master and slave discs. The mode switching during the process, the output torque transition of the HMCVT system is smooth, and the loss rate is reduced, the output torque is reduced by 38.4% and the switching time is shortened by 1.1 s compared with the adoption of the rule-based control strategy.



(a) Simulation results of output speed comparison

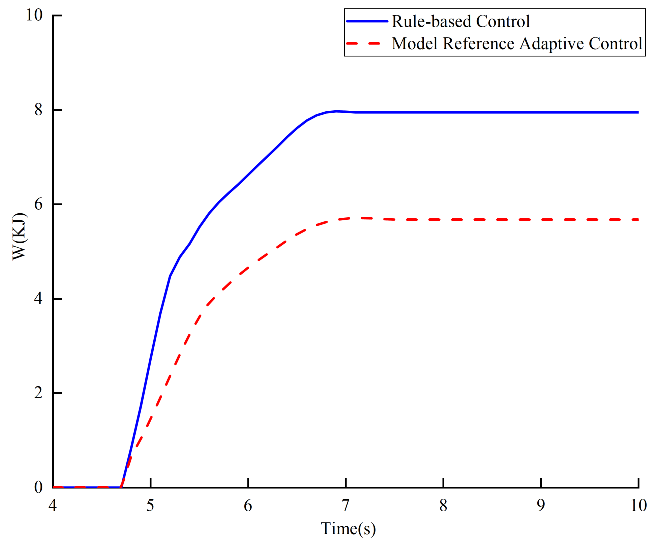


(b) Simulation results of output torque comparison



(c) Simulation results of jerk comparison

Fig. 5



(d) Simulation results of slip friction work comparison

Fig. 5. Simulation results of H to HM1 mode switching

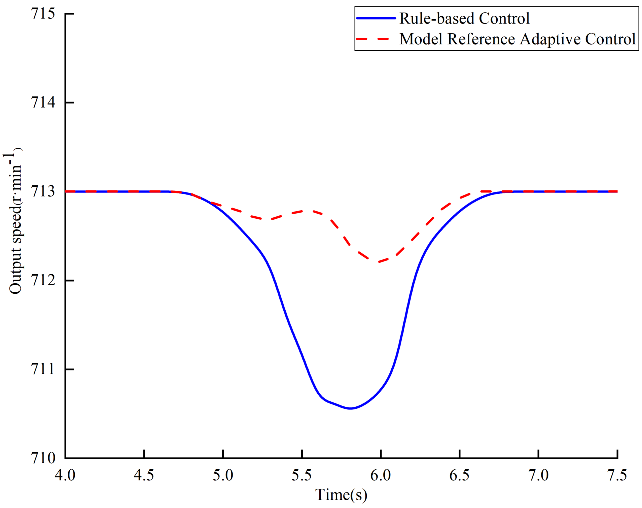
As shown in Fig. 5c, with the rule-based control strategy, the jerk of the whole vehicle fluctuates in the range of (–18.2–10.9) m/s^3 when mode switching is performed at 4.7 s. After mode switching is performed with the mode switching coordination control strategy based on model-referenced adaptation, the jerk can be controlled in the range of (–5.4–4.3) m/s^3, thus greatly improving the mode switching quality. As can be seen from Fig. 5d, the slip friction work is 7.95 KJ when mode shifting is performed with the rule-based control strategy, while the slip friction work is 5.68 KJ when mode shifting is performed with the model reference adaptive mode shift coordination control strategy, which is reduced by 28.6%, and effectively reduces clutch wear in the process of mode shifting.

5.3. Simulation of switching from HM1 mode to H mode

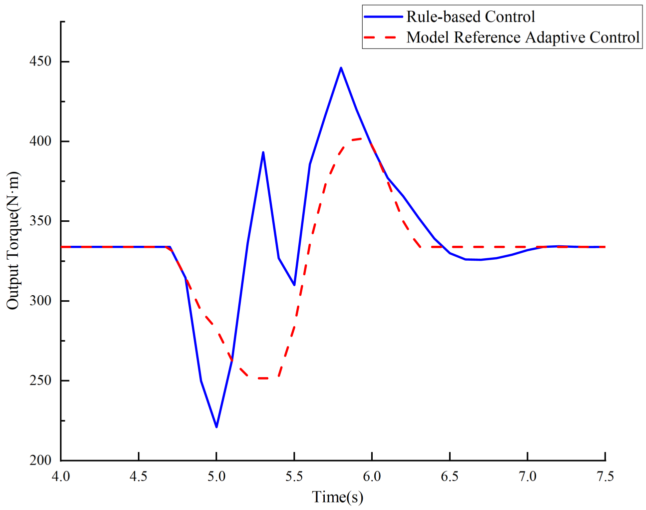
The simulation results of the model reference-based adaptive mode switching coordinated control strategy and the rule-based control strategy for HM1 to H mode switching in HMCVT are shown in Fig. 6.

As can be seen from Fig. 6a, in the case of the rule-based control strategy, there is a significant speed drop during HMCVT mode switching, with the maximum drop value of 2.47 r/min, and the output speed tends to be stable after 1.9 s; when the model reference adaptive mode switching coordinated control strategy is used, the maximum fluctuation amount is 0.8 r/min, and the switching time is reduced by 67.6%, and the switching time is reduced by 0.2 s compared with the rule-based control strategy. It can be seen from Fig. 6b that when the model reference adaptive mode switching coordinated control strategy is adopted, the output torque is reduced by 32.5% and the switching time is reduced by 1.3 s compared with the rule-based control strategy.

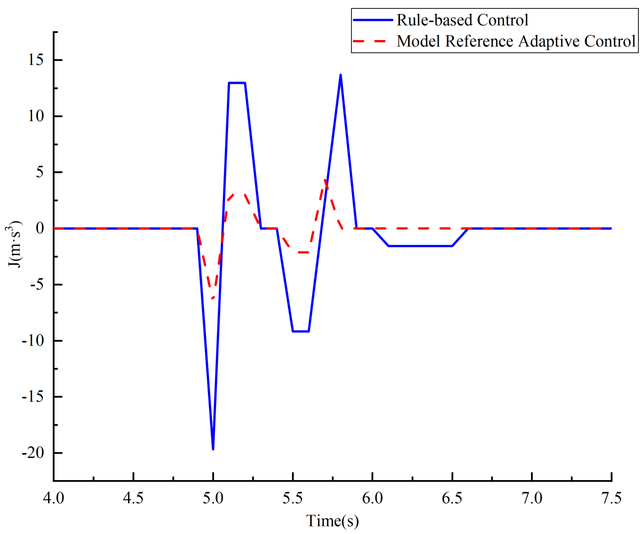
As shown in Fig. 6c, with the rule-based control strategy, the mode switching at 4.7 s, the jerk of the whole vehicle fluctuates within the range of (–19.7–13.7) m/s^3, while the mode switching



(a) Simulation results of output speed comparison

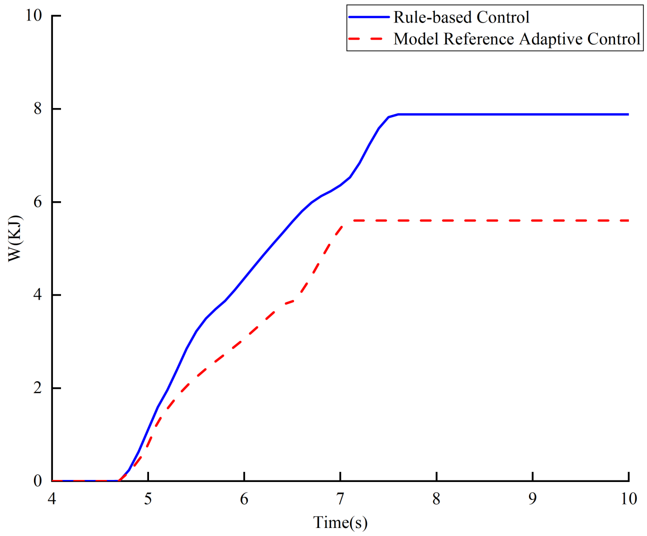


(b) Simulation results of output torque comparison



(c) Simulation results of jerk comparison

Fig. 6



(d) Simulation results of slip friction work comparison

Fig. 6. Simulation results of HM1 to H mode switching

with the model reference adaptive mode switching coordination control strategy can control the jerk within the range of (–6.5–4.4) m/s^3, thus greatly improving the mode switching quality. As can be seen in Fig. 6d, the slip friction work is 7.88 KJ when the mode switching is carried out with the rule-based control strategy, while the slip friction work is 5.6 KJ when the mode switching is carried out with the model-reference adaptive mode switching coordinated control strategy, which is a reduction of 32.5% It also effectively reduces the wear of the clutch in the process of mode switching. In this paper, the comparison results of the jerk and slip friction work generated during mode switching between the model-reference adaptive mode switching coordinated control strategy and the rule-based control strategy are shown in Table 2.

Table 2
Comparison of two mode switching strategies

Operating state	Methodology	Jerk (m/s^3)	Slip friction work (KJ)
H mode to HM1 mode	Methodology of the paper	–18.2—10.9	7.95
	Rule-based control strategies	–5.4—4.3	5.68
HM1 mode to H mode	Methodology of the paper	–19.7—13.7	7.88
	Rule-based control strategies	–6.5—4.4	5.6

In conclusion, the simulation results of HMCVT from H mode to HM1 mode and HM1 mode to H mode indicate that the model reference adaptive mode switching control strategy, based on the model reference adaptive mode switching control strategy, has a reduced fluctuation of the output torque and rotational speed during mode switching, and a greater improvement of the jerk and slip friction work. In contrast, the rule-based control strategy results in greater jerk and slip friction work during mode switching, which has a detrimental effect on tractor driving performance and comfort.

6. CONCLUSIONS

Aiming at the problem of poor smoothness during mode-shifting of the equipped HMCVT tractors, a coordinated mode-shifting control strategy based on model-referenced adaptation is proposed based on the analysis of their mode-shifting process. A model-referenced adaptive controller is designed to take the output speed of the hydraulic system of the HMCVT as the reference model and the output speed of the mechanical transmission of the HMCVT as the control object, and the adaptive rate is calculated by Lyapunov stability theory calculates the adaptive rate to achieve the mechanical transmission output speed following the hydraulic system output speed and improve the switching quality.

Compared with the use of the rule-based control strategy, the proposed method of HMCVT from H mode to HM1 mode reduces the output speed by 70%, the output torque by 38.4%, the jerk is controlled between (–5.4–4.3) m/s^3 and the slip friction work is reduced by 28.6%; during the simulation from HM1 to H mode, the proposed method makes the 67.6% reduction in output speed and 32.5% reduction in output torque; at the same time, the jerk is controlled between (–6.5–4.4) m/s^3, the slip friction work is reduced by 32.5% and the shifting smoothness is better.

The method proposed in this paper effectively improves the smoothness of mode switching and the results are of great significance for subsequent research on shift quality, but the effect of this method on HMCVT with more operating modes needs to be verified. In the subsequent research process, the method can be applied to HMCVT with more operating modes to analyze its effect on mode switching and further improve the research on smoothness.

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