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Speed Ratio Tracking Strategy of an Improved HMCVT for Commercial Vehicles

Guanzheng Wen* and Yulong Lei**

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ABSTRACT

The traditional hydro-mechanical continuously variable transmission (HMCVT) is mainly used in off-highway vehicles such as tractors. This paper proposes an improved HMCVT that is suitable for commercial vehicles. It has the same functions as the current commercial vehicle automatic transmission while making commercial vehicles overcome muddy roads like tractors. Furthermore, this paper analyzes the torque, speed, and power characteristics of the improved HMCVT, and proposes a sliding mode control strategy based on H_∞ to solve the problem of speed ratio tracking. It reduces the inherent chattering characteristics of sliding mode control while ensuring the accuracy and real-time performance of speed ratio tracking. Finally, this paper carries out the simulation verification by MATLAB/Simulink and compares with the PID speed ratio tracking method, which confirms the excellent performance of the strategy proposed in this paper.

INTRODUCTION

Hydro-mechanical continuously variable transmission (HMCVT) was first used in tractor transmission system on a large scale. It not only has high efficiency of mechanical transmission but also can easily realize continuously variable transmission through hydraulic transmission (Yu et al., 2016). It has strong adaptability to complex terrain and greatly improves the tractor's operation ability and fuel economy. Commercial vehicles face complex road conditions but also have the characteristics of high power, so how to popularize HMCVT on commercial vehicles has become a research hotspot. However, due to the different working environments of tractors and commercial vehicles, simple transplantation not only fails to give full play to the advantages of HMCVT, such as low fuel economy and strong adaptability to working conditions, but also enlarges the disadvantages of low hydraulic transmission efficiency. So far, HMCVT cannot be widely used in commercial vehicles, and the research on control strategy of HMCVT has stagnated.

Most of the research on HMCVT focuses on its application on tractors. Up to now, focusing on fuel economy and transmission efficiency, Qian et al. (2021) used response surface method and stepwise regression analysis to look for the optimal shift point of HMCVT, and carried out simulation experiments and bench test, which provided a reference for the further development of control strategies aiming to achieve fuel economy. Li et al. (2022) proposed a double-loop control with PID and MPC as the core, which greatly reduced the shifting vibration (64.4%) and achieved precise control of the HMCVT; the shifting efficiency improved, but the execution efficiency of the PID method is much higher than that of MPC, so coordinating them is difficult. Based on the same purpose, Cheng et al. (2021) proposed a segmentation modeling method and an improved genetic algorithm (GA), which were verified by bench test, achieving higher efficiency, but the complexity of the GA limited further application. Lu et. al. (2022) proposed an improved PSO algorithm for the speed regulation characteristics of HMCVT, and verified its effectiveness and rapidity. The PSO algorithm is similar to but even more complex than GA, which limited the practical application. Recent studies have generally applied newer algorithms to improve the control strategy but were lesser involved with the application of HMCVT in commercial vehicles and the structural improvements that should be done when the application of HMCVT is transferred from tractors to commercial vehicle. A long time ago, a large number of scholars also conducted theoretical explorations on HMCVT.

Keivan et al. (2015) used the nonlinear backstepping control algorithm to track the displacement of the hydraulic system and achieved good energy-saving effects. Venelin et al. (2018) studied the performance of urban buses with hybrid drive system, and the energy recovery potential of 20%–25% can be achieved through test bench verification. Antonio et al. (2013) and Alario et al. (2011) adopted the multiobjective

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* *Graduate Student, State Key Laboratory of Automotive Simulation and Control, Jilin University, Changchun, China.*

** *Professor, State Key Laboratory of Automotive Simulation and Control, Jilin University, Changchun, China.*

optimization method to reduce the volume of HMCVT and improve the transmission efficiency. Stanisaw et al. (2016) studied the motion errors of hydraulic components in the transmission system of high-speed electrical vehicles and carried out static characteristic tests to mitigate the adverse effects caused by power cycle phenomenon. Yu et al. (2018) analyzed and optimized the start control and shift strategy of the CVT system with HMCVT, which effectively improved the transmission efficiency of the vehicle. The main progress of present research is the improvement of the control strategy. Among so much literature, Alarico et al. (2017) further built the HMCVT transmission model for buses and carried out comparative simulation validation, which proved the great application potential of HMCVT in commercial vehicles has a substantial influence.

In addition, many transmission ratio tracking control strategies of the HMCVT for tractors are available. Liu et al. (2018) used electronic pumps to supply fuel to the HMCVT system, which effectively reduced the transmission ratio error in stable state and improved the transmission response speed. Zhang et al. (2013) adopted the fuzzy PID control method to adjust the variable hydraulic pump in real time, which satisfied the steering requirements of the vehicle. Harald et al. (2009) proposed a model-based trajectory tracking control strategy for hydraulic transmission system and achieved good control performance. Geng et al. (2016) of Nanjing Agricultural University and others defined the evaluation indicators affecting the starting quality and studied the main factors affecting it. Zhao et al. (2014) of the Beijing University of Technology proposed the control strategy of starting process related to the position of brake pedal and the method of dealing with the dead zone of the solenoid valve, and designed and developed an incremental closed-loop PID control system. Recently, Xia et al. (2021) proposed a sliding mode control method with feedforward compensation, which solved the problem of the adjustment accuracy of the displacement ratio of the variable pump caused by the fluctuation of oil pressure. Compared with the traditional PID method, it had better tracking stability, but it did not consider the inherent chattering characteristics of sliding mode control.

In addition, many papers were devoted to the construction of HMCVT model (Kugi et al., 2000), its simulation for CRUISE (Xi et al., 2015), and its simulation for AMESim (Chen et al., 2021; Xiao et al., 2022) to study the dynamic characteristic of HMCVT, which laid a good foundation for further study. However, research on the control strategy of match HMCVT with commercial vehicles is rare. For the actual situation of the project, most of the tracking strategies adopt the PID method and its improvements. Although the PID controller has many advantages, it still shows many shortcomings with the advancement of science and technology and the improvements of the quality requirements for speed ratio tracking. Specifically, for the HMCVT speed ratio tracking strategy proposed in this paper, the expected speed ratio of HMCVT may jump due to drastic changes in driving conditions, but the principle of direct subtraction of the actual value and the target value by the PID controller is not suitable for this situation. Moreover, HMCVT has complex working conditions and drastic changes, which requires a large amount of calibration work on PID parameters, which greatly weakens the advantages of PID control.

This paper describes the new structure and working principle of HMCVT, and the torque, speed, and power characteristics of commercial vehicles with HMCVT in detail. It also proposes a sliding mode strategy based on H_∞ control for commercial vehicles with HMCVT. Compared with other control methods, the proposed strategy greatly reduces the inherent chattering characteristics of sliding mode control and has good accuracy, real-time feature, and robustness for speed ratio tracking. Finally, this paper establishes a speed ratio tracking model by MATLAB/ Simulink and verifies the excellent performance of the proposed control strategy by comparing with the PID control method and its improvements.

Working mechanism and characteristic analysis

HMCVT can be generally divided into two types: input shunt and output shunt. The two transmission forms have their own characteristics. The input shunt is characterized by simple structure, stable mechanical transmission efficiency, and flexible structure control, whereas the output shunt is conducive to obtaining a larger torque, often used in conjunction with multisegment planetary rows, and has better fuel economy under complex driving conditions with large damping.

Specifically, for the HMCVT used in this paper, the input shunt type is selected because commercial vehicles do not need a large torque compared with agricultural vehicles, and the driving condition is better than agricultural vehicles, which is conducive to the input shunt type to play the advantages of easy control and high transmission efficiency.

To adapt to the working conditions of commercial vehicles, a wide range of improvements and designs have been made based on the HMCVT for traditional tractors, so that it fully meets the application requirements of commercial vehicles and is conducive to fuel saving. The schematic diagram of the structure is shown in Figure 1.

Its unique feature is that it fully considers the transmission requirements of good and bad road conditions, and can be intelligently switched between different modes through the cooperation of clutch engagement and separation. When starting and reversing, HMCVT is driven by pure hydraulic pressure, giving full play to the characteristics of soft hydraulic drive connection, low speed, and large torque; in good road conditions, it uses pure mechanical transmission, giving full play to the advantages of high mechanical transmission efficiency and avoiding the disadvantages of low hydraulic transmission efficiency; in bad road conditions, it uses hydraulic mechanical combined drive. It not only makes the transmission efficiency higher but also realizes stepless speed change, which is conducive to adapting to various complex terrains.

The matching situation of each clutch engagement and separation when it is in each gear is shown in Table 1.

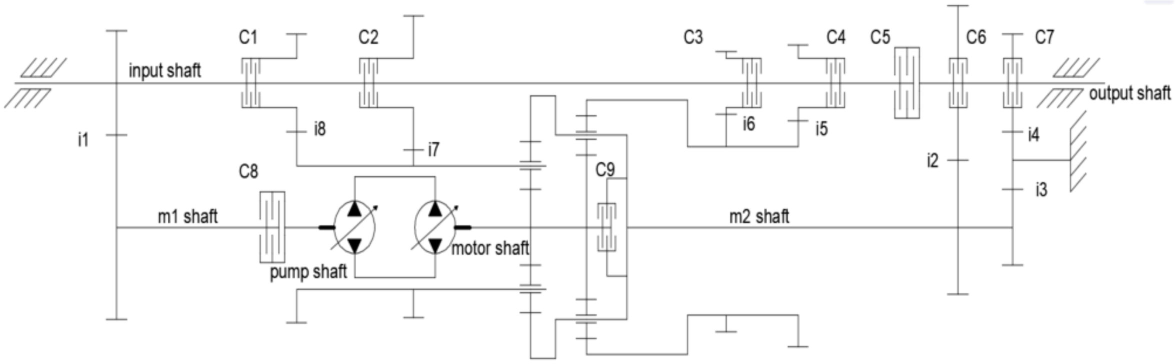


Fig. 1. HMCVT schematic diagram.

Table 1 Clutch matching diagram “ *o* ” stands for clutch engagement, and “ “ stands for clutch separation.

Driving mode		C1	C2	C3	C4	C5	C6	C7	C8	C9
Hydraulic mode	startup						o		o	o
	Revers							o	o	o
Mechanical-Hydraulic mode	1 gear			o			o		o	
	2 gear				o		o		o	
	3 gear	o					o		o	
	4 gear		o				o		o	
Mechanical mode	1 gear			o			o			o
	2 gear				o		o			o
	3 gear	o					o			o
	4 gear		o				o			o
	5 gear					o				

According to the different forms of energy flow, its working section can be divided into pure hydraulic section, hydraulic mechanical section, and pure mechanical section. The hydraulic mechanical section can be divided into several sections according to the different transmission ratios of the mechanical part. Transmission efficiency is improved through the mechanical transmission mechanism, and transmission is realized through the cooperation of separation and engagement of the clutch and brand. With the expansion of the range of dynamic ratio, stepless speed change is realized by adjusting the combination of the variable displacement hydraulic pump and the quantitative hydraulic motor. The main parameters of pumps and motors are shown in Table 2.

Table 2 Main parameters of pumps and motors

	Rated power	Rated speed	Rated displacement	Rated pressure	Torque
pump	70 kw	3500 r/min	50 cm^3/r	300 bar	240-400 Nm
motor	70 kw	3500 r/min	50 cm^3/r	300 bar	200-360 Nm

To perform characteristic analysis as (Zhang, 2011) the concept of displacement ratio is first defined:

$$e = \frac{n_m}{n_p} \tag{1}$$

where n_m is the speed of the motor, n_p is the speed of the pump, and e is the displacement ratio.

Then, the kinematic characteristic for the different components of the planetary also needs to be determined:

$$n_s + k \cdot n_r - (1+k) \cdot n_c = 0,$$
$$T_s = \frac{T_r}{k} = \frac{T_c}{1+k} \quad (2)$$

where n_s is the speed of the sun gear, n_r is the speed of the ring gear, n_c is the speed of the planet carrier, T_s is the torque of the sun gear, T_r is the torque of the ring gear, T_c is the torque of the planet carrier, and k is the structure parameter of the planetary.

Since the following research on the speed ratio tracking strategy, the pure hydraulic section mode is mainly involved. “Start up” in the pure hydraulic section mode is taken as an example, and its speed, torque, and power characteristics are analyzed for others have similar characteristics.

For “start up” in pure hydraulic section mode, clutches 6, 8, and 9 are engaged while others are disengaged, as shown in Figure 1 and Table 1, and the speed characteristics can be obtained:

$$n_{H0} = \frac{n_m}{i_2} = \frac{e \cdot n_e}{i_1 \cdot i_2} \quad (3)$$

where n_{H0} is the speed of the output shaft, n_e is the speed of the engine, and i_1, i_2 are transmission ratios, as shown in Figure 1. The torque characteristics can be obtained:

$$T_m = T_{m_2} = \frac{T_{out}}{i_2},$$
$$T_p = T_{m_1} = T_m \cdot e = \frac{T_{out} \cdot e}{i_2},$$
$$T_{in} = \frac{T_{m_1}}{i_1} = \frac{T_{out} \cdot e}{i_1 \cdot i_2}, \quad (4)$$

where T_m is the torque of the m shaft, T_{m_1} is the torque of the m_1 shaft, T_{m_2} is the torque of the m_2 shaft, T_p is the torque of the pump, T_{out} is the torque of the output shaft, and T_{in} is the torque of the input shaft.

Then, a concept of hydraulic power split ratio ϵ is defined:

$$\epsilon = \frac{P_m}{P_{out}} \quad (5)$$

where P_m is the power of the motor, and P_{out} is the power of transmission. Clearly, $\epsilon = 1$ for pure hydraulic section mode. Next, the following can be obtained:

$$P_m = n_m \cdot T_m = \frac{e \cdot n_e}{i_1} \cdot \frac{T_{out}}{i_2}$$
$$P_{out} = n_{out} \cdot T_{out} = P_m \quad (6)$$

Strategy for speed ratio tracking

After analyzing the structure and working principle of the HMCVT, and its speed, torque, and power characteristics, the speed ratio tracking strategy is studied. Although the parameters of each working mode are different, they have similar equations, so only the starting mode under pure hydraulic pressure is analyzed as an example.

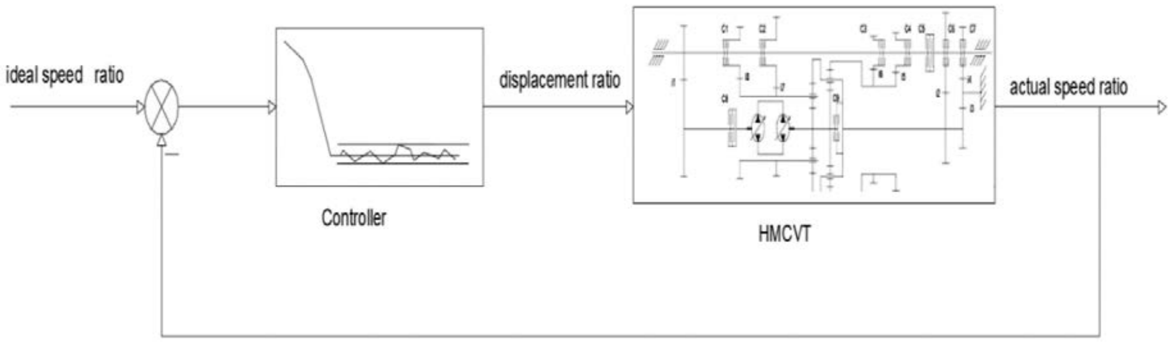


Fig. 2. Controller diagram.

The core of the speed ratio tracking is the pump–motor model. The actual operation has are pressure loss, overflow loss, and throttling loss in the liquid circulation of the speed regulating system, and impact loss and friction loss between the pipe walls. These factors are random and unpredictable, so the ideal situation that ignores these factors is studied.

Then, the model can be determined (Hu et al, 2000; Yuan et al, 2008):

$$V_{pmax} \cdot n_p \cdot e = V_m \cdot n_m + \frac{C_S \cdot \Delta p \cdot (V_{pmax} + V_m)}{\mu} + \frac{V_0}{\beta_e} \cdot \frac{d\Delta p}{dt},$$
$$\Delta p \cdot V_m = J_m \cdot \frac{dn_m}{dt} + f_m \cdot n_m + T_m \quad (7)$$

where V_0 is the working volume of the oil, β_e is the elastic modulus of the oil, J_m is the inertia of the motor shaft, f_m is the viscous damping of the motor, T_{m} is the customized load torque of the output shaft, $V_{p\mathrm{max}}$ is the maximum displacement ratio of the pump, V_m is a quantitative motor displacement, Δp is the pressure difference between the high- and low-pressure oil paths, C_s is the total leakage coefficient, μ is the oil viscosity, n_m is the rotational speed of the motor shaft, and n_p is the rotational speed of the pump shaft. Then, according to Formulas (4)–(7), the model of the startup of HMCVT can be obtained:

$$V_{pmax} \cdot \frac{n_e}{i_1} \cdot e = V_m \cdot n_0 \cdot i_2 + \frac{C_S \cdot \Delta p \cdot (V_{pmax} + V_m)}{\mu} + \frac{V_0}{\beta_e} \cdot \frac{d\Delta p}{dt}$$
$$\Delta p \cdot V_m = (\frac{J_0}{i_2} + J_{m0} \cdot i_2) \cdot \frac{dn_0}{dt} + f_m \cdot i_2 \cdot n_0 + \frac{T_{out}}{i_2} \quad (8)$$

where J_0 is the load inertia of the output shaft, J_{m0} is the inertia of the motor shaft, and n_0 is the rotational speed of the output shaft. Speed ratio is defined as $\varepsilon = \frac{n_0}{n_e}$, and Formulas (9) can be obtained by Laplace transform of Formulas (8):

$$\frac{V_{pmax} \cdot e(s)}{i_1} = \frac{V_0 \cdot J_{H0}}{\beta_e \cdot V_m} \cdot s^2 \cdot \varepsilon(s) + \left(\frac{C_S \cdot (V_{pmax}+V_m) \cdot J_{H0}}{\mu \cdot V_m} + \frac{V_0 \cdot f_m \cdot i_2}{V_m \cdot \beta_e}\right) \cdot s \cdot \varepsilon(s) + \left(V_m \cdot i_2 + \frac{C_S \cdot (V_{pmax}+V_m) \cdot f_m \cdot i_2}{\mu \cdot V_m}\right) \cdot \varepsilon(s),$$

$$\frac{\Delta P(s) \cdot V_m}{n_e} = J_{H0} \cdot s \cdot \varepsilon(s) + f_m \cdot i_2 \cdot \varepsilon(s), \quad (9)$$

where $J_{H0} = \frac{J_0}{i_2} + i_2 \cdot J_{m0}$. Then, the transfer function can be obtained:

$$G(s) = \frac{\varepsilon(s)}{e(s)} = \frac{1}{a \cdot s^2 + b \cdot s + c}, \quad (10)$$

where $a = \frac{V_0 \cdot J_{H0} \cdot i_1}{\beta_e \cdot V_m \cdot V_{pmax}}$, $b = \frac{C_S \cdot (V_{pmax}+V_m) \cdot J_{H0} \cdot i_1}{\mu \cdot V_m \cdot V_{pmax}} + \frac{V_0 \cdot f_m \cdot i_1 \cdot i_2}{V_m \cdot \beta_e \cdot V_{pmax}}$, and $c = \frac{V_m \cdot i_1 \cdot i_2}{V_{pmax}} + \frac{C_S \cdot (V_{pmax}+V_m) \cdot f_m \cdot i_1 \cdot i_2}{\mu \cdot V_m \cdot V_{pmax}}$.

Then, it can be transferred into a state space function:

$$\dot{X} = A \cdot X + B \cdot U + D, \quad (11)$$

where $A = \begin{bmatrix} 0 & 1 \\ \frac{-c}{a} & \frac{-b}{a} \end{bmatrix}$, $B = \begin{bmatrix} 0 \\ 1 \end{bmatrix}$, $D = \begin{bmatrix} 0 \\ d \end{bmatrix}$, d is an uncertain disturbance, and $X = [\varepsilon \quad \dot{\varepsilon}]^T$, $U = [e]$. Based on this state space equation, a sliding mode control method is proposed based on H_∞, which uses the characteristics of fast response and reliable control of the sliding mode control to ensure tracking accuracy and the strong robust characteristics of H_∞ to alleviate its inherent chattering problem. The sliding surface is designed as follows:

$$s = C_{ss} \cdot \vartheta(t) + \int_0^t C_{ss} \cdot (B \cdot K - A) \cdot \vartheta(t) d\tau, \quad (12)$$

where $C_{ss} = [0 \quad 1]$, $\vartheta(t) = X - X_d$, X_d is the desired value corresponding to X, and K is the control matrix to be calculated. Furthermore, the differential of s can be obtained:

$$\dot{s} = C_{ss} \cdot \dot{\vartheta}(t) + C_{ss} \cdot (B \cdot K - A) \cdot \vartheta(t), \quad (13)$$

Then the differential of s is set:

$$\dot{s} = -\eta_1 \cdot sign(s) - \eta_2 \cdot s, \quad (14)$$

where $\eta_1 = 0.98$, and $\eta_2 = 0.86$. Finally, according to Formulas (13) and (14), the control variables can be determined:

$$U = (C_{ss} \cdot B)^+ \cdot \left[-C_{ss} \cdot A \cdot X - C_{ss} \cdot D + C_{ss} \cdot \dot{X}_d - C_{ss} \cdot (B \cdot K - A) \cdot (X - X_d) - \eta_1 \cdot sign(s) - \eta_2 \cdot s\right], \quad (15)$$

Then, its stability is proven so the Lyapunov equation can be designed:

$$V_1 = 0.5 \cdot s^2, \quad (16)$$

Therefore, the differential of this Lyapunov equation can be derived:

$$\dot{V}_1 = s \cdot \dot{s} = s \cdot (-\eta_1 \cdot sign(s) - \eta_2 \cdot s) = -\eta_1 \cdot |s| - \eta_2 \cdot s^2 \quad (17)$$

Clearly, $\dot{V}_1 \leq 0$. Moreover, in Formula (15), K is unknown, which involves the use of H_∞ method to alleviate chattering. According to Formula (13), to make the state trajectory strictly limited within a certain range after reaching the sliding surface, $\dot{s} = 0$ must be ensured. The equivalent control variable is set as U_{eq}, and the following can be obtained:

$$U_{eq} = (C_{ss} \cdot B)^+ \cdot \left[-C_{ss} \cdot A \cdot X - C_{ss} \cdot D + C_{ss} \cdot \dot{X}_d - C_{ss} \cdot (B \cdot K - A) \cdot (X - X_d)\right], \quad (18)$$

It is substituted into $\dot{\vartheta}$ to derive the following:

$$\dot{\vartheta} = \dot{X} - \dot{X}_d = A \cdot X + B \cdot U_{eq} + D - \dot{X}_d = (A - B \cdot K) \cdot \vartheta, \quad (19)$$

Let $H = -B$ then $\dot{\vartheta} = (A + H \cdot K) \cdot \vartheta$, and the Lyapunov equation on the sliding mode surface is set:

$$V_2 = \vartheta^T \cdot P \cdot \vartheta, \quad (20)$$

where P is a definite positive matrix. Then, the following can be derived:

$$\dot{V}_2 = \vartheta^T \cdot [P \cdot (A + H \cdot K) + (A + H \cdot K)^T \cdot P] \cdot \vartheta, \quad (21)$$

Let $e_\vartheta = C \cdot (X - X_d) = C \cdot \vartheta$, When the state trajectory reaches the sliding mode surface, to reduce chattering and satisfy $e_\vartheta{}^T \cdot e_\vartheta \leq \frac{1}{\gamma^2} \cdot D^T \cdot D$, the index function is set:

$$J_\infty = \int_0^\infty (\gamma^{-2} \cdot e_\vartheta^T \cdot e_\vartheta - D^T \cdot D) d\tau,$$
$$= \int_0^\infty \left(\gamma^{-2} \cdot e_\vartheta^T \cdot e_\vartheta - D^T \cdot D + \dot{V}_2\right) d\tau - \int_0^\infty \dot{V}_2 \, d\tau, = \int_0^\infty \left(\gamma^{-2} \cdot e_\vartheta^T \cdot e_\vartheta - D^T \cdot D + \dot{V}_2\right) d\tau - \lim_{\tau \to \infty} V_2 + V_2(0), \quad (22)$$

where $\gamma = 0.8$ is the chattering range set. To satisfy stability, $\lim_{\tau \to \infty} V_2 \leq 0$ must be satisfied, and $V_2(0)$ is known. Considering that $e_\vartheta{}^T \cdot e_\vartheta \leq \frac{1}{\gamma^2} \cdot D^T \cdot D$ is satisfied, the following can be obtained:

$$\gamma^{-2} \cdot e_\vartheta^T \cdot e_\vartheta - D^T \cdot D + \dot{V}_2 \leq 0, \quad (23)$$

Substituting Formula (23) into Formula (21), the following can be derived:

$$\gamma^{-2} \cdot C^T \cdot C \cdot \vartheta^T \cdot \vartheta + \vartheta^T \cdot [P \cdot (A + H \cdot K) + (A + H \cdot K)^T \cdot P] \cdot \vartheta \leq D^T \cdot D, \quad (24)$$

The linear matrix is applied to solve it, and a linear matrix inequality is set:

$$\begin{bmatrix} \varsigma & -P & C^T \\ -P^T & -I & 0 \\ C & 0 & -\gamma^2 \cdot I \end{bmatrix} < 0, \quad (25)$$

where $\varsigma = P \cdot (A + H \cdot K) + (A + H \cdot K)^T \cdot P$, and $C = [\frac{1}{a} \quad 0]$. According to Schur's supplementary lemma, Formula (32) is equivalent to the following:

$$P \cdot (A + H \cdot K) + (A + H \cdot K)^T \cdot P + P \cdot P^T + \frac{C^T \cdot C}{\gamma^2} < 0, \quad (26)$$

Clearly, if Formula (26) is satisfied, then Formula (24) must be satisfied, so that chattering can be controlled within the range of $|\gamma^2|$. Here, MATLAB's YALMIP toolbox is used to solve matrices P and K, that is, $K = [-9420 \quad -327830]$.

After matrix K is solved, the sliding mode control variable U can be obtained, and the chattering problem can be alleviated by the H_∞ method. To verify the stability of the whole process, its Lyapunov equation is set:

$$V = V_1 + V_2, \tag{27}$$

Clearly, in the case of matrices K and P satisfying Formula (24), $\dot{V}_1 \leq 0, \dot{V}_2 \leq 0$, then $\dot{V} = \dot{V}_1 + \dot{V}_2 \leq 0$ to ensure the stability of the whole process.

Simulation

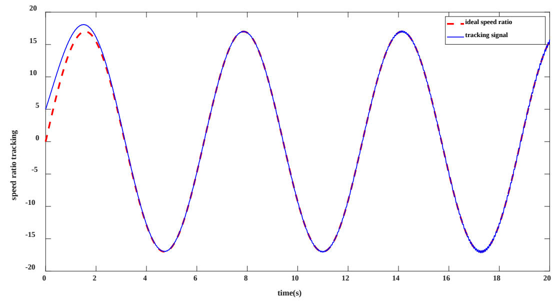
Based on the improved HMCVT structure, to verify the effectiveness of the proposed sliding mode tracking strategy based on the H_∞ method, the simulation parameters are set, as shown in Table 3.

Table 3 Simulation parameter values

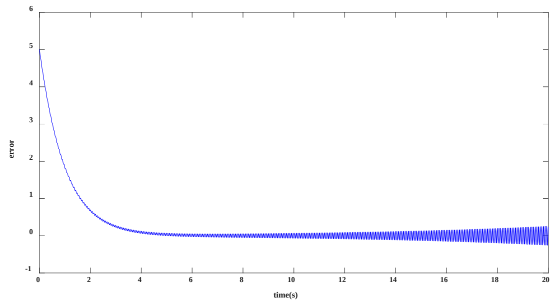
parameter	value
V_0	70 cm^3
V_m	34.8 cm^3/r
V_{pmax}	50 cm^3/r
β_e	1400 Mpa
f_m	200000 N
J_0	3.67 $kg \cdot m^2$
J_{m0}	2.74 $kg \cdot m^2$
T_0	250 $N \cdot m$
k_1	2.26
k_2	3.26
i_1	2.078
i_2	1.045
i_3	0.901
i_4	1.141
i_5	2.855
i_6	3.731
i_7	4.918
i_8	6.046
μ	0.04132
C_s	$1.4 \cdot 10^{-9}$

First, simulation is carried out to verify the accuracy of the proposed tracking method when the reference speed ratio curve is a straight, a sin, a trapezoidal, and a single lane curve. Then, the strategy is compared with the mature PID method to verify its superiority.

Figure 3 shows when the speed ratio is a sinusoidal curve, the initial value of the ideal speed ratio is 0, the initial value of the actual speed ratio is 5, and the error converges to 0 in about 3 s; that of a single lane and a trapezoidal curve is about 4 s; when the speed ratio is a straight line, the error converges to near 0 at 5 s. In various situations, the speed ratio is accurately tracked. Moreover, except that the chattering is more serious in the sinusoidal curve, the chattering problem is alleviated well in other situations. Then, the proposed strategy is compared with the mature, reliable PID speed ratio tracking method, as shown below:

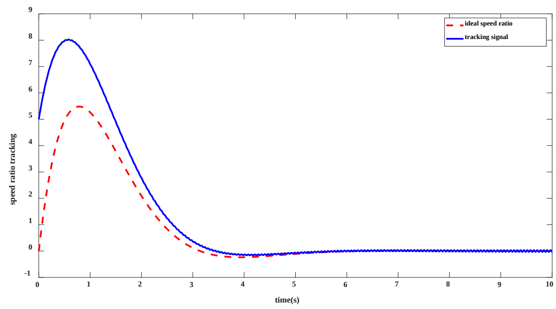


(a1)

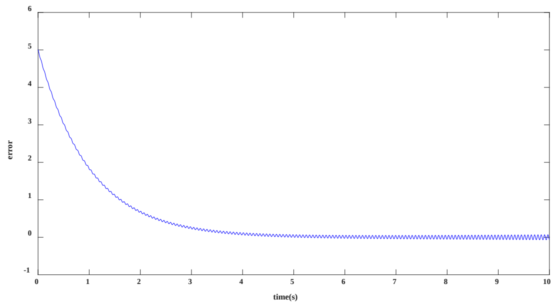


(a2)

(a) Sin curve

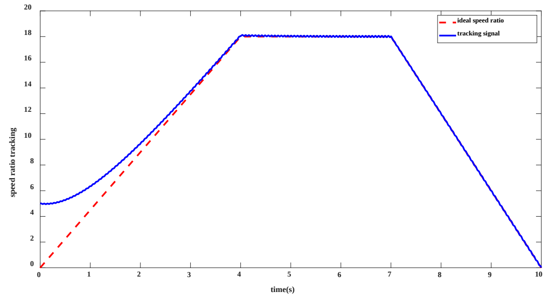


(b1)

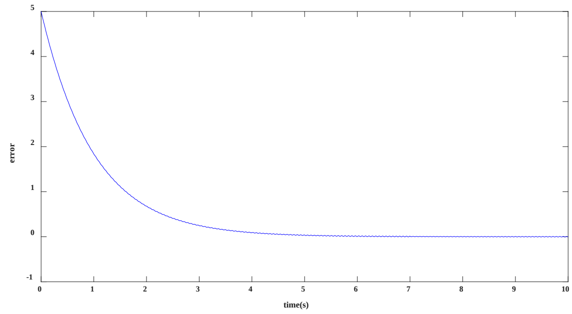


(b2)

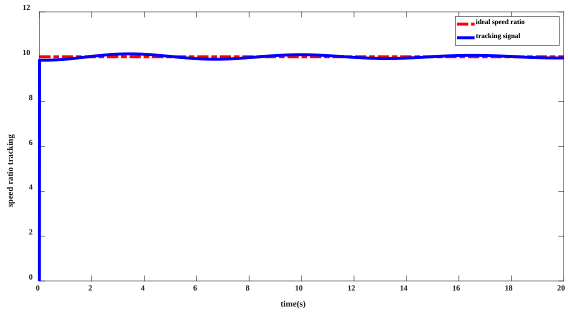
(b) Single-lane curve



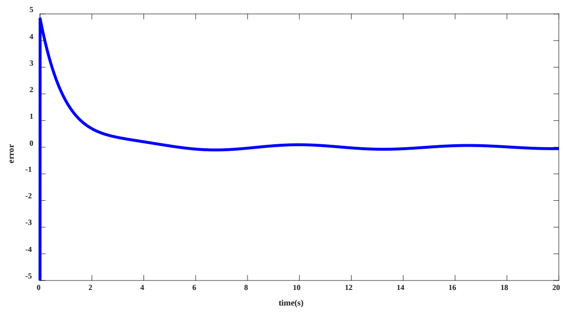
(c1)



(c2)
(c) Trapezoidal curve



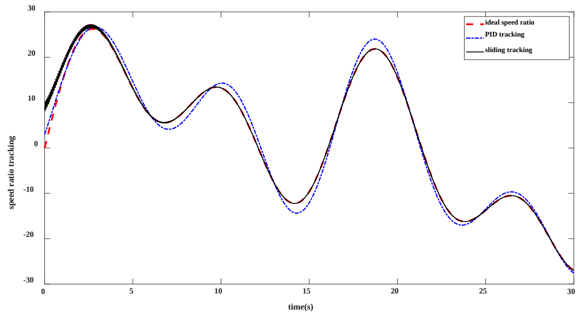
(d1)



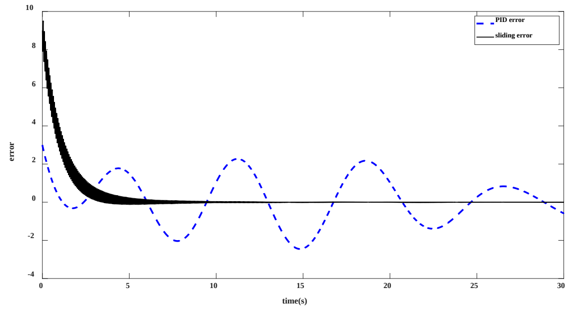
(d2)
(d) Straight line

Figure 3 Tracking performance of the proposed method

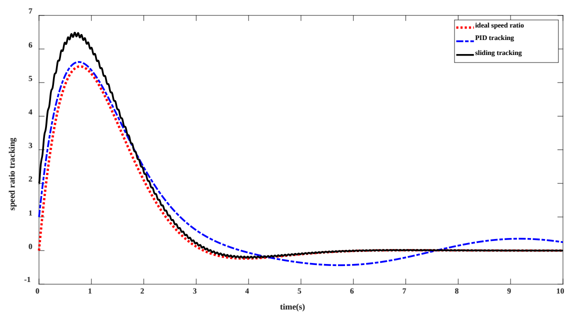
Figure 4 shows when the speed ratio is a random curve, the final error of the proposed strategy converges to 0, whereas that of the PID method hovers between −2 and 2; when the speed ratio is a single-lane-change curve, the final error of the PID method hovers between −0.5 and 0.5, and that of the proposed strategy is also 0; when the speed ratio is the most confusing Gaussian curve, the final error of the PID method hovers between −1 and 1, that of the proposed strategy is still 0. In addition, the proposed strategy has almost no chattering problem under the random curve situation, and the effect is also very good under the single-lane-change situation. The chattering is larger under the Gaussian curve situation, but it is in the acceptable range.



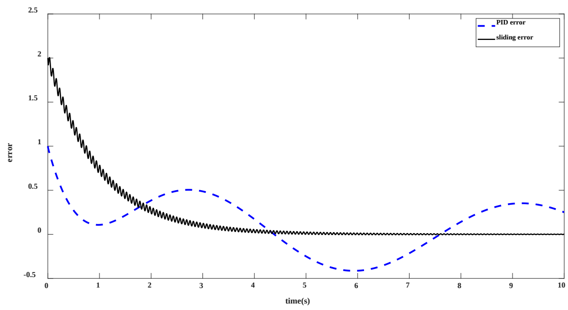
(a1)



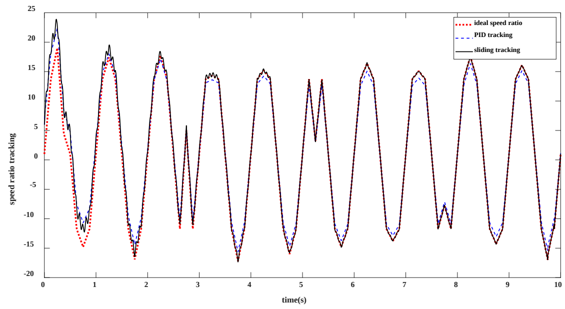
(a2)
(a) Random line



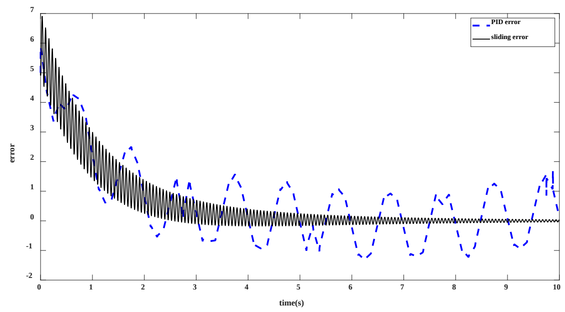
(b1)



(b2)
(b) Single lane



(c1)



(c2)

(c) Gaussian curve

Figure 4 Comparison with PID controller

In summary, the proposed strategy has good tracking accuracy for the speed ratio and has superior performance compared with the mature PID method. It also greatly reduces the chattering problem of sliding mode control and achieves a good control performance.

Conclusion

In this paper, the structure of HMCVT is improved according to the road condition characteristics of commercial vehicles, so that it is suitable for commercial vehicles. Moreover, a sliding mode strategy based on H_∞ control for commercial vehicles with HMCVT is proposed to track the ideal speed ratio of HMCVT in the starting process, which performs better compared with the PID controller. Specifically, the following conclusions are derived:

(1). Compared with the traditional PID control strategy, the sliding mode strategy based on H_∞ control has a better tracking performance for the ideal speed ratio trajectory.

(2). The stability of sliding mode strategy based on H_∞ control is good, and the overshoot is much smaller than that of PID control.

(3). The sliding mode strategy based on H_∞ control greatly reduces the chattering phenomenon of sliding mode control.

However, this work neither carries out actual vehicle verification nor proposes a complete shift strategy. This work only performs analysis and calculation in a simple starting condition. Real vehicle tests will be conducted, and different situations will be tested in the future.

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一種改進的商用 HMCVT 及其速比跟蹤策略研究

溫官正　　雷雨龍

吉林大學汽車工程學院

摘 要

傳統的液力機械無級變速器（HMCVT）主要用於拖拉機等非公路車輛。本文提出了一種適用於商用車的改進了的HMCVT。它具有與目前商用車自動變速箱相同的功能，同時可以使商用車像拖拉機一樣克服泥濘的道路。本文在進一步分析了改進後的HMCVT的轉矩、轉速和功率特性的基礎上，提出了一種基於H_∞的滑模控製策略來解決速比跟蹤問題。它抑製了滑模控製固有的抖振特性，同時保證了速比跟蹤的準確性和實時性。最後,本文通過MATLAB/Simulink進行仿真驗證,並與PID速比跟蹤方法進行對比,證明了本文提出的策略的優越性。