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Two-Layer Structure Control of an Automatic Mechanical Transmission Clutch During Hill Start for Heavy-Duty Vehicles

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ABSTRACT The engagement control of an automatic mechanical transmission (AMT) clutch during the hill start of a heavy-duty vehicle has a significant impact on the comfort, safety and service life of the vehicle. However, the control effect of a traditional control strategy can be easily affected by interference (clutch wear, temperature). Therefore, this paper proposes a two-layer structure control strategy based on an AMT clutch automatic actuator with the clutch engagement speed as the control target. First, the clutch automatic actuator is designed and the governing characteristics of the diesel engine and the control characteristics of the solenoid valve are studied. Second, a logic threshold control method and PID control method are adopted in the proposed two-layer structure control method. Moreover, the robustness of the proposed control algorithm is validated by a co-simulation platform (TruckSim, MATLAB/Simulink). Finally, experimental research under different slopes (13% and 22%) is carried out to verify the simulation results. The experimental results prove that compared with a single-layer control strategy, the two-layer control strategy proposed in this paper can shorten the start time by more than 10%, and reduce the vehicle start-up jerking by approximately 20%, which significantly improves the performance of the vehicle in the hill start process.

INDEX TERMS Heavy-duty vehicle, two-layer structure control, logic threshold, clutch automatic actuator, PID control, simulation, experimental research.

I. INTRODUCTION

With the rapid development of the automobile industry, the safety of drivers and the stability of vehicles have received increasing attention from researchers. Various vehicle safety systems are constantly improved and applied, such as Hill Start Assist (HSA), anti-lock braking systems (ABSs), and automatic mechanical transmission (AMTs) [1], [2]. An automatic mechanical transmission (AMT) is a type of automatic shifting control mechanism based on the original dried frictional flake's clutch and fixed-shaft geared manual transmission to realize automatic operation of transmission selection and shifting [3]. An AMT has the advantages of both an automatic transmission (AT) and manual transmission (MT) in terms of efficiency, cost, simplicity, and ease of manufacture [4]. Therefore, in recent years, AMTs have been widely used in the automotive field [5]–[7].

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However, in the process of clutch engagement when a vehicle is started, there are widespread phenomena of large friction work and start-up jerking, which not only greatly shorten the service life of the clutch, but also seriously affect the driver's experience [8]. In addition to comfort, c0lutch engagement control of the vehicle during the hill start process also has an important influence on vehicle safety. If some interference factors are not taken into account in the clutch control strategy, the vehicle may produce large start-up jerking and friction work, especially when starting on a hill, and this may lead to a risk of an accident [9].

Although AMTs are modifications of fixed-shaft geared manual transmissions, they have disadvantages such as poor smoothness and easy power interruption [10]. By optimizing the control of clutch engagement, it is possible to reduce clutch wear, shorten the vehicle start time, and reduce jerking and friction work [11], [12]. The control of the clutch engagement process has a significant impact on various performance characteristics of a vehicle, such as its start-up [13], fuel consumption [14] and security [15]. Therefore, many researchers have done much work on clutch engagement control in the start-up of vehicles [16]–[18].

Jiang *et al.* [19] proposed a model predictive control (MPC) strategy of an AMT clutch for medium-duty trucks during the start-up process. Li *et al.* [20] proposed a novel piecewise linear feedback control strategy for an automotive dry clutch engagement process. Based on a dynamic model of the powertrain system, the controller was designed by minimizing a quadratic performance index subject to constraints on the inputs and states. Yang *et al.* [21] presented a multiple-model predictive controller (mMPC) applied to an automated manual transmission vehicle, which improves vehicle drivability during the vehicle start-up phase. Zhao *et al.* [22] studied the identification of the driver's launching intentions, which may change anytime, and proposed a clutch engagement sliding mode control method for vehicle launching. Pisaturo and Senatore [23] developed an optimal launching-intention-aware control strategy for clutch engagement. Huang *et al.* [24] proposed an optimal control strategy based on the minimum principle to solve the problems with the starting process of a self-developed five-speed dry dual-clutch transmission (DCT). Gao *et al.* [25] proposed a clutch disengagement strategy for the shift control of automated manual transmissions based on a drive shaft torque observer. For precise position control of the clutch, Li *et al.* [26] presented a control scheme using a model predictive control method with the correction of clutch wear based on the estimation of resistance torque. Berkel and Veldpaus [27] proposed a new controller design that explicitly separates the control laws for each objective by introducing three clutch engagement phases.

The current research on clutch engagement control has seldom considered the influence of external interference (clutch wear, temperature). Therefore, based on previous research [28], [29], this paper proposes a two-layer structure control algorithm based on an AMT clutch automatic actuator, taking a heavy-duty vehicle as the research object and the clutch engagement speed as the control target. First, a clutch automatic actuator is designed and the governing characteristics of the diesel engine and the control characteristics of the solenoid valve are studied. Second, a co-simulation platform is established involving MATLAB/Simulink and TruckSim to verify the effectiveness of the proposed controller under different hill start conditions, including 13% and 22% slopes. Finally, the simulation results are verified by road tests on different slopes (13% and 22%). Simulation and experimental results fully demonstrate the feasibility of the proposed control algorithm, which not only shortens the start time of the vehicle but also reduces the start-up jerking and friction work and greatly improves the starting performance of the vehicle.

The primary objective of this article is to develop a two-layer control strategy for the AMT clutch of a heavy-duty vehicle based on a gas-assisted automatic actuator

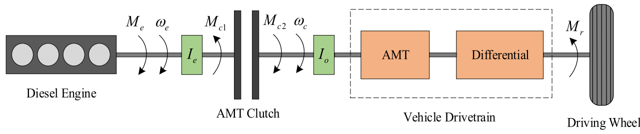

FIGURE 1. **Simplified transmission model of the AMT clutch.**

during vehicle hill start. There are three important novel contributions of our study.

(1) The system model is designed as follows: the clutch model and vehicle driveline system model are constructed; the clutch automatic actuator is designed; and the vehicle model is formulated in the TruckSim platform.

(2) In the two-layer structure control method, the first-layer control strategy separates the control laws for each objective by introducing five clutch engagement phases. On the basis of the first-layer logic threshold control strategy, a second-layer control strategy is added to improve the adaptability of interference factors (temperature, clutch wear). The proposed two-layer structure controller outputs the PWM control signal for the solenoid valve to indirectly regulate clutch engagement speed.

(3) A co-simulation platform is established involving MATLAB/Simulink and TruckSim to verify the effectiveness of the control strategy under different conditions. In addition, experiments of actual vehicle start-up are carried out on different slopes to further demonstrate the feasibility and robustness of the control method proposed in this paper.

The rest of this paper is organized as follows: In Section 2, the clutch automatic actuator is designed. The two-layer structure control strategy for a heavy-duty vehicle AMT clutch is proposed in Section 3. The simulation results are analyzed in Section 4, and Section 5 shows the experimental results. The conclusions of the paper are given in Section 6.

II. SYSTEM MODELS

A. VEHICLE DRIVELINE MODEL

To facilitate the analysis of the vehicle start-up process, each component is regarded as a rigid body and the clearance of the transmission system is ignored. The vehicle powertrain is simplified into two free bodies: form the active part of the engine to the clutch; and from the driven part of the clutch to the wheel. Accordingly, the system model is established as shown in Fig. 1.

The engagement process of the clutch can be simply divided into three stages:

(1) The first stage is to eliminate the clearance between the clutch driving and driven plates. At this time, the clutch does not transfer torque, and the engine output torque is only used to change its own speed. The transmission part is not stressed, the clutch output torque and ground resistance torque are zero, and the force acting on the engine part is as follows:

$$M_e = I_e \dot{\omega}_e \tag{1}$$

$$M_{c1} = M_{c2} = M_r = 0 \tag{2}$$

where M_e is the engine output torque $(N \cdot m)$, I_e is the equivalent rotational inertia of the engine $(kg \cdot m^2)$, ω_e is the angular velocity of the clutch driving plate (rad/s), M_r is the ground resistance moment, M_{c1} is the torque transferred by the clutch when the clutch driving and driven plates are in relative friction $(N \cdot m)$ and M_{c2} is the torque transferred by the clutch when the clutch driving and driven plates rotate synchronously.

(2) The second stage is the sliding friction stage. At this time, the clutch driving and driven plates are in relative friction, and the force on each part is written as follows:

$$M_e = I_e \dot{\omega_e} + M_{c1} \tag{3}$$

$$M_{c1} = k_c F_c \mu \tag{4}$$

$$M_{c1} = I_o \dot{\omega_c} + i_o M_r \tag{5}$$

where k_c is the coefficient, F_c is the pressing force (N), μ is the clutch plate friction coefficient, i_o is the powertrain system transmission ratio, I_o is the equivalent rotational inertia of the vehicle $(kg \cdot m^2)$ and ω_c is the clutch driven plate angular velocity (rad/s).

According to formula (4), M_{c1} is determined by the pressing force between the clutch driving and driven plates. Consequently, the engine speed will decrease when M_{c1} is greater than M_e, while if M_{c1} is equal to M_e, the engine will run steadily. At this point, the vehicle exhibits speed and acceleration, and start-up jerking starts to appear as follows:

$$j = \eta \frac{i_o}{r} \frac{dM_{c1}(t)}{dt} \tag{6}$$

where j is the start-up jerking (m/s^3), η is the efficiency of the transmission system and r is the wheel radius. The start-up jerking in this stage is determined by the rate of change of the torque transferred by the clutch.

(3) The third stage is the clutch synchronization stage, where the angular velocity of the clutch driving and driven plates are the same. The force on the engine is as follows:

$$M_e = I_e \dot{\omega_e} + M_{c2} \tag{7}$$

$$M_{c2} = I_o \dot{\omega_c} + i_o M_r = I_o \dot{\omega_e} + i_o M_r \tag{8}$$

It can be obtained from formulas (12) and (13) that:

$$M_{c2} = \frac{I_o}{I_o + I_e} M_e + \frac{I_e}{I_o + I_e} M_r \approx M_e \tag{9}$$

Compared with the vehicle's rotational inertia, the rotational inertia of the engine is relatively small, so the torque transferred by the clutch can be approximately considered to be equal to the output torque of the engine. At this point, the start-up jerking of the vehicle is as follows:

$$j = \eta \frac{i_o}{r} \frac{dM_{c2}(t)}{dt} = \eta \frac{i_o}{r} \frac{dM_e(t)}{dt} \tag{10}$$

In the whole starting process of the vehicle, jerking refers to the rate of change in the vehicle's longitudinal acceleration. The calculation formula is as follows:

$$j = \frac{da}{dt} = \frac{d^2 v}{dt} \tag{11}$$

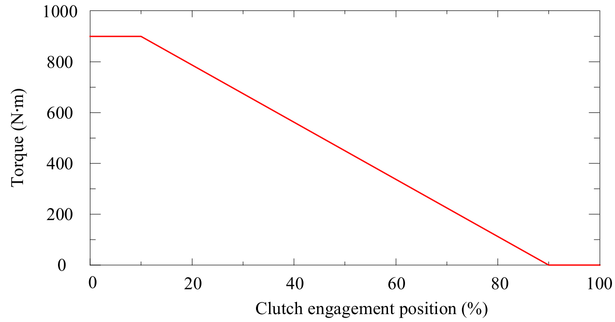

FIGURE 2. **Relationship between the clutch output torque and clutch engagement position.**

where a is the vehicle longitudinal acceleration (m/s^2) and v is the vehicle speed (s).

In addition to jerking, friction work is also an important indicator to evaluate vehicle starting performance. Friction work refers to the amount of work done by friction torque in the process of clutch engagement. The amount of friction work will affect the transmission efficiency of the vehicle driveline, and the calculation formula is as follows:

$$W_c = \int_{t_0}^{t_1} M_c(t)\omega_e(t)dt + \int_{t_1}^{t_2} M_c(t)[\omega_e(t) - \omega_c(t)]dt \tag{12}$$

where W_c is the friction work (J), M_c is the torque transferred by the clutch $(N \cdot m)$, t_0 is the moment when the clutch driven plate torque appears (s), t_1 is the moment when the clutch driven plate starts to rotate (s) and t_2 is the moment when the clutch driving and driven plates rotate synchronously (s).

According to the above analysis, during the transition from the second stage to the third stage, a sudden change in torque is likely to occur, which will lead to larger start-up jerking of the vehicle and affect the starting stability.

B. CLUTCH MODEL

A dry clutch relies on friction to transfer torque. The formula for calculating the torque transferred by a dry clutch is:

$$T_c = \mu(T, \Delta\omega) F_c Z \times \frac{2}{3} \left(\frac{R_1^3 - R_0^3}{R_1^2 - R_0^2} \right) \tag{13}$$

where $\mu(T, \Delta\omega)$ is the dynamic friction factor, Z is the number of clutch friction pairs, R_0 is the clutch friction plate internal radius (mm), R_1 is the clutch friction plate external radius (mm) and T is the clutch friction plate temperature (°C).

Ignoring changes in temperature, clutch wear, and dynamic friction factors, a simplified representation of the relationship between the clutch engagement position and its output torque is shown in Fig. 2.

C. CLUTCH AUTOMATIC ACTUATOR MODEL

As shown in Fig. 3, a heavy-duty vehicle adopts a gas-assisted hydraulic clutch actuator. The driver operates the clutch hydraulic master cylinder by stepping on the clutch pedal, and this cylinder outputs high-pressure liquid that act on the gas-assisted hydraulic working cylinder to realize the power-assisted function.

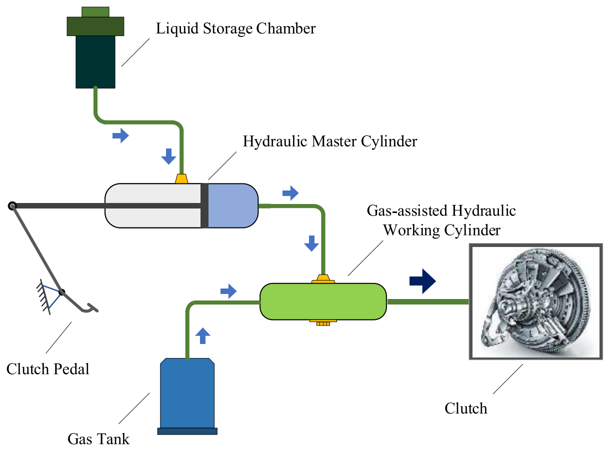

FIGURE 3. Gas-assisted hydraulic clutch actuator.

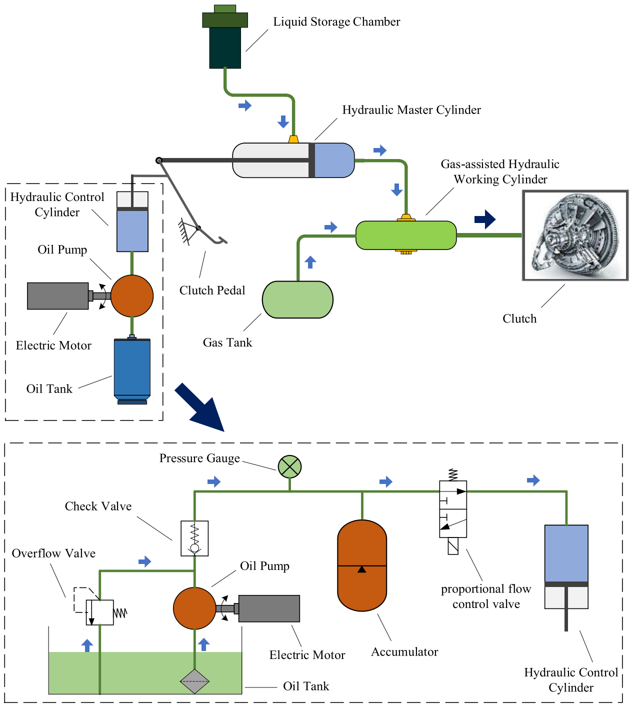

FIGURE 4. Gas-assisted hydraulic automatic clutch actuator.

In this study, the hydraulic master cylinder and gas-assisted hydraulic working cylinder in the manual control mechanism of the clutch were retained, and only the parts of the clutch pedal were modified. As shown in Fig. 4, the hydraulic control cylinder is installed in the clutch pedal part, and the AMT oil supply system is used to realize the automatic control of the hydraulic master cylinder, to realize the automatic separation and engagement of the clutch.

The working position of the gas-assisted hydraulic working cylinder is indirectly controlled to drive clutch engagement by filling the hydraulic control cylinder with oil. The oil in the hydraulic control cylinder is provided by the oil source of the AMT system. The relationship of the motions of the parts of the clutch automatic actuator can be simplified as shown in Fig. 5.

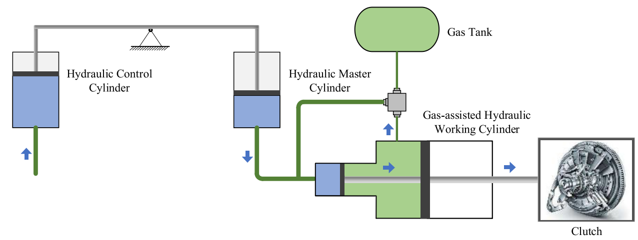

FIGURE 5. Action of the automatic clutch actuator.

The formula for calculating the gas-assisted hydraulic working cylinder piston movement speed v_3 is:

$$v_3 = \frac{S_1 k}{S_2 S_3} q \tag{14}$$

where q is the flow of liquid into/out of the hydraulic control cylinder, S_1 is the hydraulic control cylinder section area, S_2 is the hydraulic master cylinder section area and S_3 is the hydraulic master cylinder section area.

The formula for the clutch displacement L is:

$$L = \int v_3 dt = \int \frac{S_2 k}{S_1 S_3} q dt = \frac{S_2 k}{S_1 S_3} V \tag{15}$$

The clutch engagement/separation speed is proportional to the flow rate q through the solenoid valve, and the clutch engagement/separation displacement is proportional to the oil volume V through the solenoid valve. Therefore, by controlling the flow and volume through the solenoid valve, the movement speed and displacement of the clutch can be indirectly controlled.

The proportional flow valve has three liquid channels, and the connection of different channels and the size of the channel aperture can be adjusted by controlling the working current of the flow valve to control the flow rate of the flow valve.

D. DIESEL ENGINE MODEL

The torque curve of the diesel engine is relatively flat, and small changes in the external resistance moment will lead to large fluctuations in engine speed, so the driver needs to adjust the throttle opening frequently. To reduce the labor intensity of drivers, this study used a test vehicle with a diesel engine and variable-speed governor, whose characteristic curves under different throttle openings are shown in Fig. 6.

The variable-speed governor can maintain the engine speed within a certain range to prevent stalling or racing. Fig. 7 shows the characteristic torque curve of the diesel engine with the variable-speed governor at a 30% throttle opening.

E. ANALYSIS OF THE VEHICLE HILL START PROCESS

When the vehicle starts on a hill, under the influence of the component force of the vehicle weight along the downward direction of the hill, the vehicle will have a tendency to slide down the slope. Therefore, it is necessary to coordinate the control of the clutch and the braking system, and complete

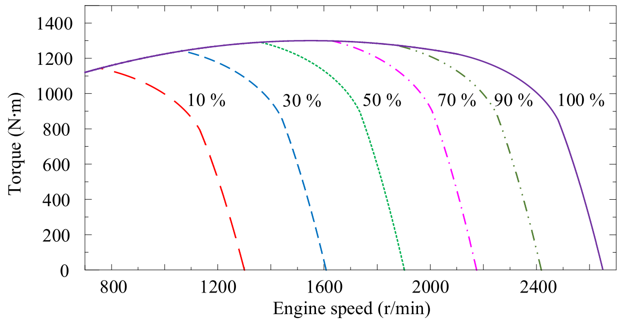

FIGURE 6. Fixed throttle characteristics of a diesel engine with a variable-speed governor.

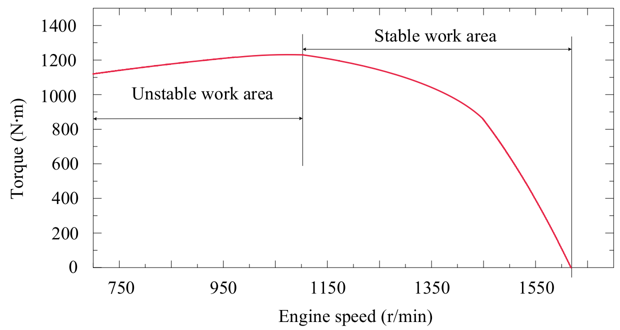

FIGURE 7. Diesel engine speed-torque curve with 30% throttle opening.

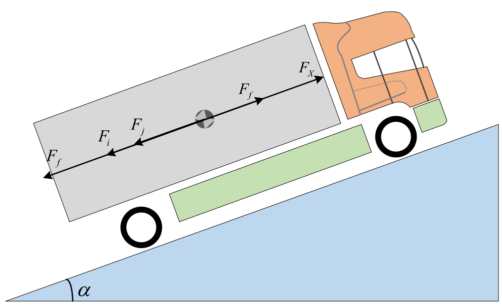

FIGURE 8. Forces influencing vehicle longitudinal dynamics in hill start.

the hill start of the vehicle through reasonable coordination between the braking torque and the torque transferred by the clutch.

The main forces influencing the vehicle longitudinal dynamics in hill start scenarios are shown in Fig. 8.

According to the change in the forces of the vehicle, its starting process on the hill can be divided into five stages as shown in Fig. 9.

(1) Starting stage: the clearance between the driving and driven plates of the clutch is gradually eliminated and the vehicle is in a static state. At this stage, the braking force of the braking system is used to overcome the grade resistance as follows:

$$F_X = F_i \tag{16}$$

where, F_X is the braking force (N) and F_i is the grade resistance (N).

(2) When the clutch driving and driven plates begin to engage, the driving force transferred by the clutch increases gradually; however, it is not enough to overcome the grade

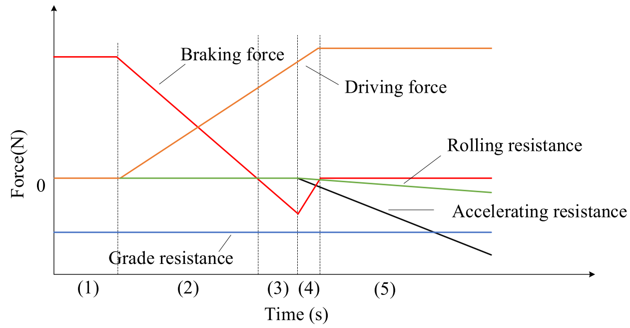

FIGURE 9. Force analysis during the hill start of the vehicle.

resistance of the vehicle, and the vehicle is still in a static state. At this stage, the braking force of the brake system and the driving force transmitted by the clutch work together to overcome the grade resistance as follows:

$$F_X + F_t = F_i \tag{17}$$

where F_t is the driving force (N).

(3) As the driving force transmitted by the clutch continues to increase and the vehicle's ramp resistance can be overcome, the vehicle exhibits an upward trend along the ramp. Since the brake is still not released, the braking force becomes the resistance hindering the vehicle from starting, and the vehicle is still at rest. At this stage, the driving force transferred by the clutch is used to overcome the ramp resistance and the braking force of the braking system as follows:

$$F_t = F_i + F_X \tag{18}$$

(4) With the decrease in the braking force and the increase in the driving force, the vehicle speed begins to increase. The driving force transferred by the clutch is enough to overcome the braking force and ramp resistance, and the vehicle enters a state of motion. At this stage, the braking is gradually released, and the driving force must overcome the braking force, grade resistance, rolling resistance and accelerating resistance as follows:

$$F_t = F_i + F_X + F_j + F_f \tag{19}$$

where F_f is the rolling resistance (N) and F_j is the accelerating resistance (N).

(5) The brake is completely released, and the driving force transferred by the clutch overcomes the grade resistance, accelerating resistance and rolling resistance as follows:

$$F_t = F_i + F_j + F_f \tag{20}$$

According to the above analysis, the third stage is the key to the vehicle starting process control. In other words, the determination of the parking brake release time in the third stage is crucial for clutch engagement control in the whole hill start process.

F. HILL START ASSIST (HSA) SYSTEM MODEL

According to the characteristics of the test vehicle braking system, a parking solenoid valve (HSA valve) controlled by

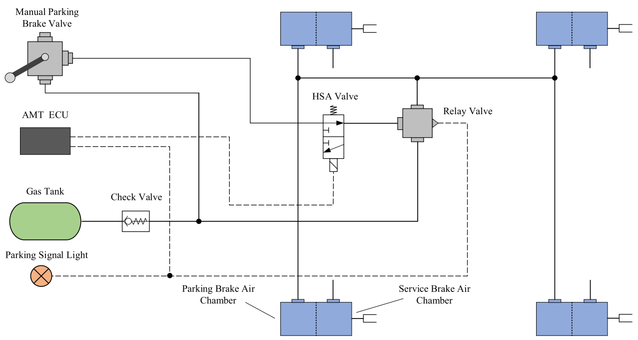

FIGURE 10. Parking brake system.

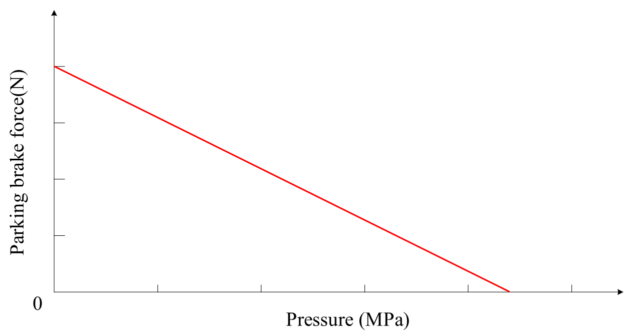

FIGURE 11. Relationship between brake chamber pressure and parking brake force.

the AMT control unit is installed in the gas pipeline of the vehicle parking brake system, which can be seen in Fig. 10. When the brake chamber is inflated, the parking brake is released slowly, as shown in Fig. 11. In the process of a vehicle hill start, the AMT system controls the opening and closing of the HSA valve to realize the control of the vehicle parking brake. In addition, the AMT system can identify the braking state of the vehicle by detecting the pneumatic switch signal of the parking brake, To realize the delayed release of the parking brake when the vehicle starts on a hill.

III. CONTROL STRATEGY DESIGN

A. TWO-LAYER STRUCTURE CONTROL STRATEGY

When the vehicle is in a low temperature environment, it will affect the clutch engagement speed; when the clutch is worn, the same clutch position may not deliver enough torque. To ensure that the clutch can still achieve the required engagement speed and transfer enough torque when the vehicle is subjected to external interference, this paper proposes a two-layer structure control strategy based on a clutch automatic actuator. The first-layer control adopts a logic threshold control method, and the second-layer control adopts a PID control method. A block diagram of the control system is presented in Fig. 12, which explains the control process of clutch engagement in detail. The clutch engagement speed and the target clutch engagement speed are denoted by v_c and v_c', respectively. The throttle opening is represented by o, and the clutch engagement position is indicated by L_c. n_e and n_1 are the engine speed and clutch driven plate speed, respectively.

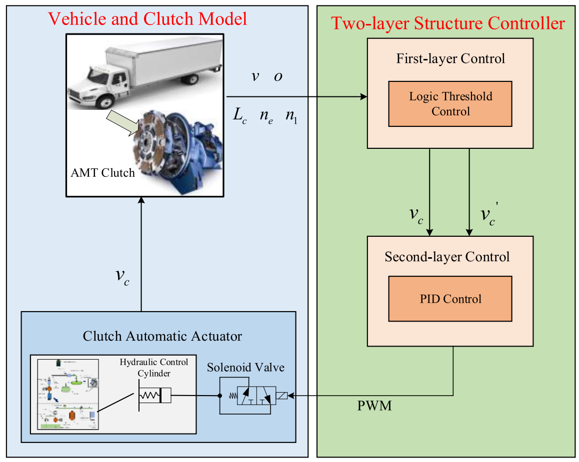

FIGURE 12. Block diagram of the two-layer control strategy.

First, some key signals are transmitted to the first-layer control via the vehicle AMT control system, including the amount of throttle opening, clutch engagement position, engine speed, clutch driven plate speed and vehicle speed. The clutch engagement position is obtained by a displacement sensor installed on the clutch. Second, after the signals are calculated in the first layer, the target clutch engagement speed and the derivative of the clutch engagement position (clutch engagement speed) are transmitted to the second-layer control (PID control). Here, the target clutch engagement speed is a given value and the clutch engagement speed is the feedback value. Finally, the two-layer control system responds to the speed deviation signal and outputs the PWM signal to the solenoid valve of the clutch automatic actuator. The PID control parameters are set as $K_p = 0.05$, $K_i = 27.6$ and $K_d = 0.01$.

The control strategy takes the clutch engagement speed as the direct control target, and the system is not affected by interference (low temperature, clutch wear), and has good robustness. When the vehicle is in ideal conditions (25°C, clutch without wear), it may be possible to achieve a great control effect only through the first-layer control system; When the vehicle is experiencing interference conditions (low temperature, clutch wear), with a large deviation between the clutch engagement speed transmitted by the first-layer control system and the target clutch engagement speed, the first-layer control may not be able to meet the vehicle's starting requirements. Therefore, the second-layer control strategy is added to improve the adaptability for interference factors (low temperature, clutch wear), thus greatly improving the starting performance of the vehicle.

B. PWM CONTROL

Because the system hysteresis and PWM method is applied in the proposed control strategy, some experiments are performed to measure the relationships between duty ratios and the clutch engagement position, which could facilitate the development of proper duty ratios. Under the same pressure difference, the basic working characteristics of the proportional flow valve are shown in Fig. 13.

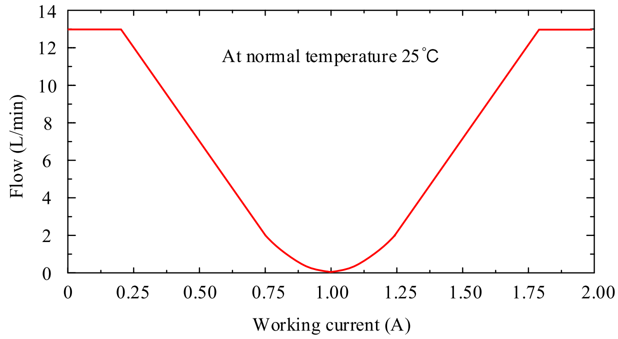

FIGURE 13. Basic working characteristics of the proportional flow valve.

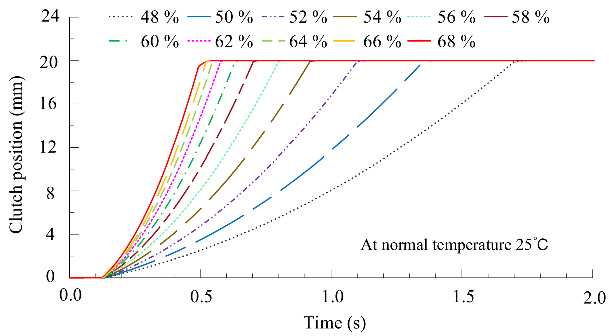

FIGURE 14. Curves of clutch separation.

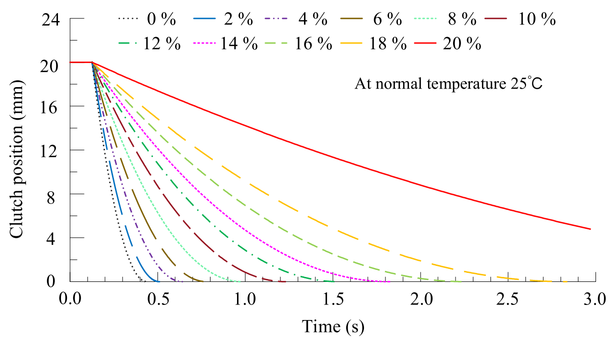

FIGURE 15. Curves of clutch engagement.

When the solenoid valve works, its resistance value can be considered unchanged, so its working current can be controlled by regulating its working voltage, and the working voltage of the proportional flow valve can be controlled by the duty ratio of PWM. The clutch engagement process under different duty ratios is shown in Fig. 14 and Fig. 15. As shown in Fig. 14, with an increase in the duty ratio, the clutch separation time is gradually shortened, from 1.75 s under a 48% duty ratio to 0.5 s under a 68% duty ratio. As shown in Fig. 15, the clutch engagement time increases from 0.5 s under a 2% duty ratio to 2.75 s under an 18% duty ratio. When the duty ratio is 20%, the clutch engagement time will be longer. Therefore, it can be concluded that effective control of the clutch engagement position can be achieved by regulating the duty ratio.

C. CLUTCH ENGAGEMENT PROCESS ANALYSIS

According to the change in the force of the vehicle during the hill start process and the vehicle state detected by the

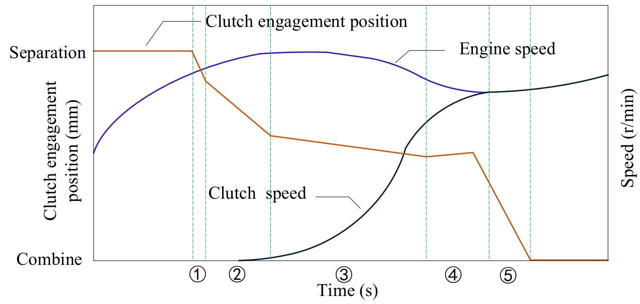

FIGURE 16. Ideal control curve of a vehicle hill start process.

AMT system, the clutch engagement process of the vehicle is divided into five stages, as shown in Fig. 16.

When a vehicle starts on a hill, it is quite possible for the vehicle to slide down due to the large component force of the vehicle gravity in the direction of the slope. Therefore, when a vehicle starts on a hill, safety is considered to be the primary control target.

Phase 1: The clutch driving and driven plates are not in contact, and the clearance between the clutch driving and driven plates is quickly eliminated. To shorten the engagement time of the clutch, the clutch is controlled to engage at a faster speed.

Phase 2: In the first half of the second phase, the clutch driving and driven plates start to contact, but the clutch driven plate is still stationary and there is sliding friction with the clutch driving plate. At this time, the generated driving force cannot overcome the grade resistance. To ensure a smooth start of the vehicle, this phase should control the clutch to be engaged at a slow speed. Additionally, the clutch engagement speed should be adjusted in real time according to the throttle opening and engine speed rate of change.

In the second half of the second phase, as the clutch continues to engage, the driving force transferred by the clutch gradually approaches the grade resistance. When it is large enough to overcome the grade resistance, the parking brake will be released. Then, the clutch starts to rotate, and the vehicle speed starts to increase.

At this phase, the release of the parking brake is based on the premise that the driving force of the clutch drive is enough to overcome the grade resistance. The driving force is determined according to the vehicle information detected by the AMT system. When the driving force is enough to overcome the grade resistance, the HSA valve is given an inflation signal to release the brake.

Phase 3: If the vehicle is on a steep slope or the vehicle is overloaded, with the further engagement of the clutch, the engine speed may be quickly reduced and enter the unstable working area. To ensure that the engine speed stays in the stable working area and to reduce the shock of the clutch, the clutch needs to be engaged slowly. As the braking force decreases, the driving force acting on the wheels for acceleration will gradually increase.

Phase 4: The speed difference between the clutch and the engine continues to shrink. If the load torque transferred

TABLE 1. Main parameters of the experimental vehicle.

Parameter	Value
Vehicle mass (kg)	8190
Wheel radius (mm)	397
Powertrain system transmission ratio	30.786
Transmission efficiency	0.99

by the clutch to the engine is too large, the engine will sharply drop its speed and even flame out. Therefore, it is necessary to control the engagement of the clutch according to the speed change of the engine. When the engine throttle opening increases, if the engine speed increases or remains unchanged, the clutch should continue to be engaged or maintained at the current position; otherwise, the clutch should be slowly released to the semi-engagement point position.

This phase aims to make the friction torque transmitted by the clutch equal to the engine torque when the clutch driving and driven plates rotate synchronously. In this way, it can avoid a drastic change in the torque transferred by the clutch and improve the smoothness of the vehicle

Phase 5: The clutch driving and driven plates rotate at the same speed until the clutch is completely engaged

IV. SIMULATION AND ANALYSIS

To verify the two-layer structure control strategy based on an AMT clutch automatic actuator proposed in this paper, the vehicle model was developed on the Trucksim platform and the controller model was designed on the MATLAB/Simulink platform. A co-simulation platform is thereby constructed based on the MATLAB/Simulink and TruckSim platforms to emulate the clutch engagement process. The main parameters of the vehicle are listed in Table 1.

This section mainly describes the simulation results on roads with slopes of 13% and 22%, including the results curves under ideal conditions (25°C, clutch without wear) and interference conditions (−10°C, clutch wear).

The simulation results under ideal conditions are shown in Fig. 17 and Fig. 18. Under ideal conditions, there is no significant difference in the control effect between single-layer control and the proposed two-layer control. Moreover, it can be seen from the change in parking brake pressure that the HSA system and AMT system cooperate well, and the release time of the parking brake system is accurate.

To quantitatively measure and evaluate the vehicle start-up jerking in an experiment, the root mean square (RMS) is introduced as an evaluation parameters. Thirty sets of data were collected to calculate the RMS value of start-up jerking.The RMS equation can be expressed as:

$$RMS = \sqrt{\frac{1}{n}\sum\nolimits_{k=1}^{n} j^2} \tag{21}$$

To intuitively reflect the advantages of the two-stage algorithm, we introduce the start time to measure the vehicle performance, and the start time is the time corresponding to the engagement of the clutch.

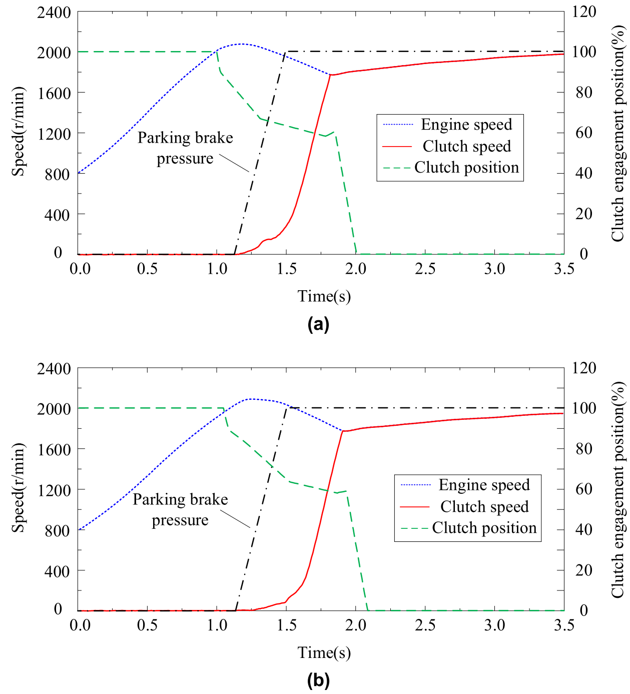

FIGURE 17. Simulation results under ideal conditions (13% slope). (a) Single-layer control, (b) two-layer control.

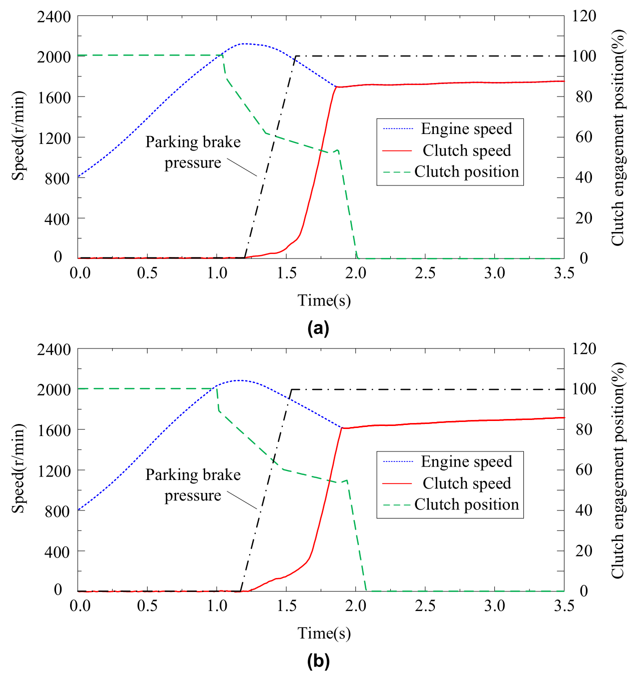

FIGURE 18. Simulation results under ideal conditions (22% slope). (a) Single-layer control, (b) two-layer control.

As shown in Fig. 19, on the 13% slope, the start time of the vehicle under the single-layer control strategy is 1.70 s, the RMS of jerking is approximately 2.72, and the friction work is 290 kJ. The start time of the vehicle under two-layer structure control is 1.04 s, and the RMS of jerking is approximately 1.88, and the friction work is 215 kJ. Compared with the single-layer control, the start time is shortened by

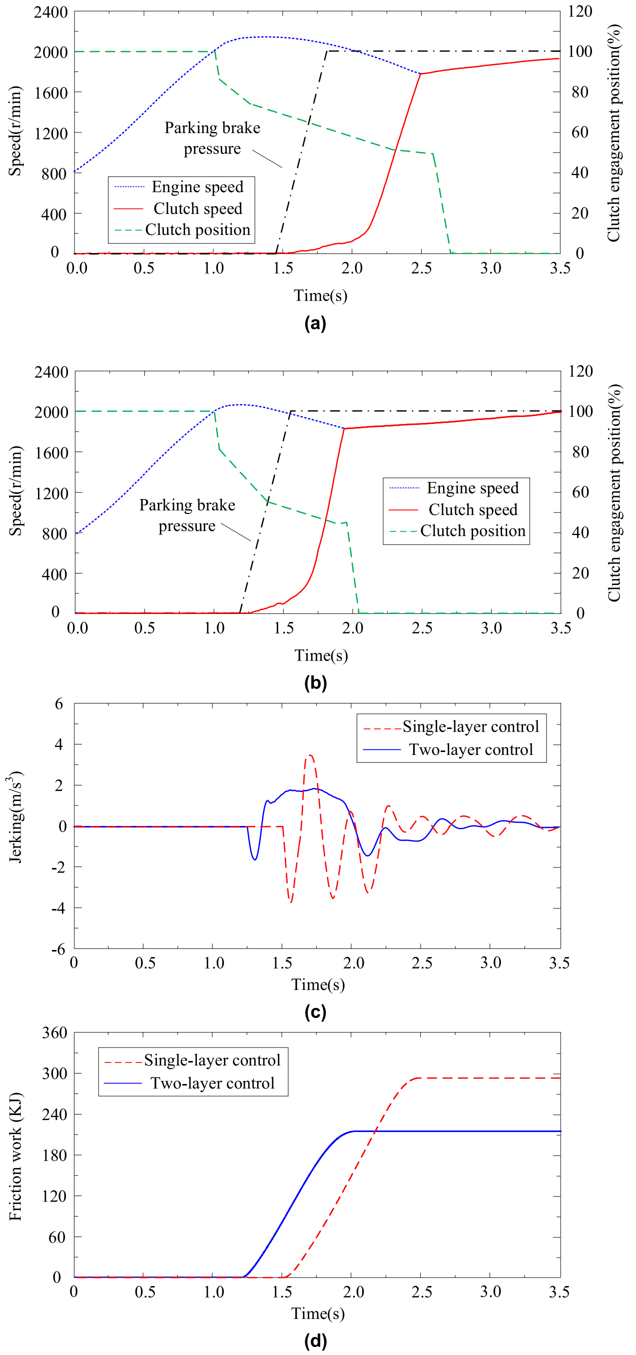

FIGURE 19. Simulation results under interference conditions (13% slope). (a) Single-layer control, (b) two-layer control, (c) jerking comparison, (d) friction work comparison.

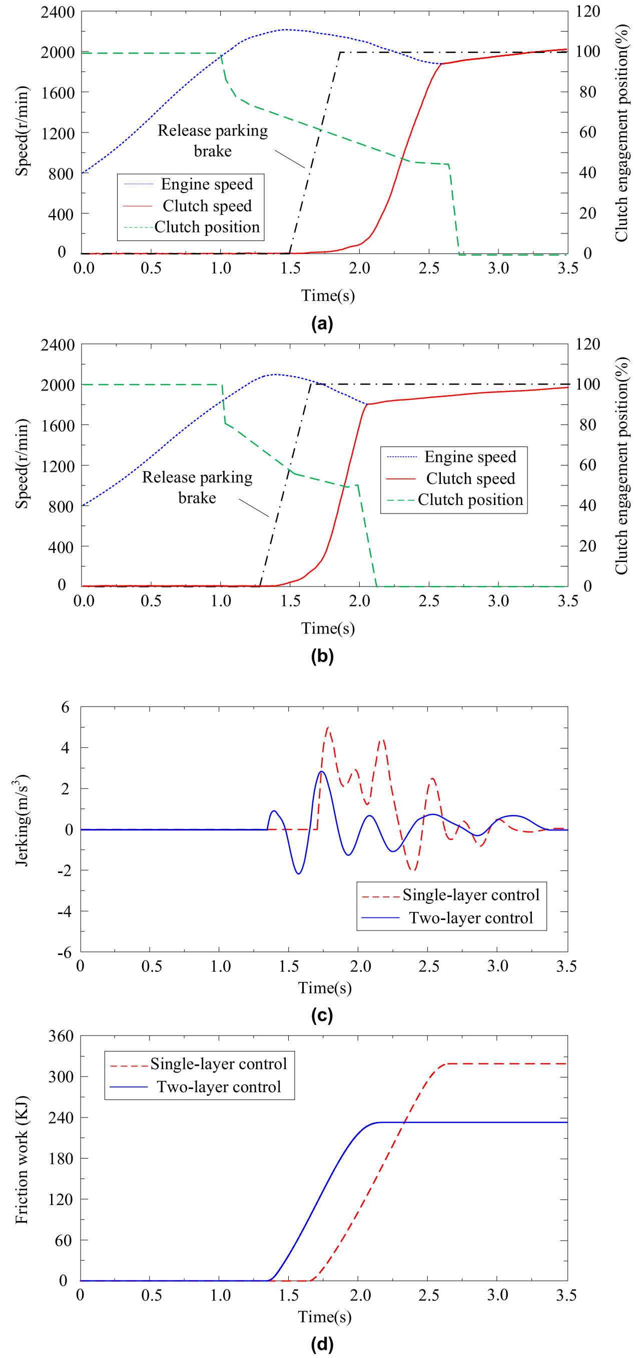

FIGURE 20. Simulation results under interference conditions (22% slope). (a) Single-layer control, (b) two-layer control, (c) jerking comparison, (d) friction work comparison.

approximately 38.8%, the jerking is reduced by approximately 30.9%, and the friction is reduced by approximately 25.8%.

The same conclusion can also be drawn from Fig. 20. On the 22% slope, the start time of the vehicle under the single-layer control strategy is 1.72 s, the RMS of jerking is approximately 3.07, and the friction work is 320 kJ. The start time of the vehicle under the two-layer structure control is 1.13 s, the RMS of jerking is approximately 1.93, and the friction work is 235 kJ. Compared with the single-layer control strategy, the two-layer structure control strategy can shorten the start time by approximately 34.3%, reduce the RMS of vehicle jerking by approximately 37.1%, and reduce the friction work by approximately 26.5%.

The data in Table 2 can more intuitively reflect the differences between the two control methods. Compared with the single-layer control strategy, the two-layer control strategy reduces the start time, friction work and start-up jerking. The performance of the vehicle's start-up process has been significantly improved.

TABLE 2. Comparison of the simulation results under interference conditions.

Evaluation index	Single-layer	Two-layer	Performance improvement
Start time (s) (13% slope)	1.70	1.04	38.8%
RMS of start-up jerking (13% slope)	2.72	1.88	30.9%
Friction work (kJ) (13% slope)	290	215	25.8%
Start time (s) (22%slope)	1.72	1.13	34.3%
RMS of start-up jerking (22% slope)	3.07	1.93	37.1%
Friction work (kJ) (22% slope)	320	235	26.5%

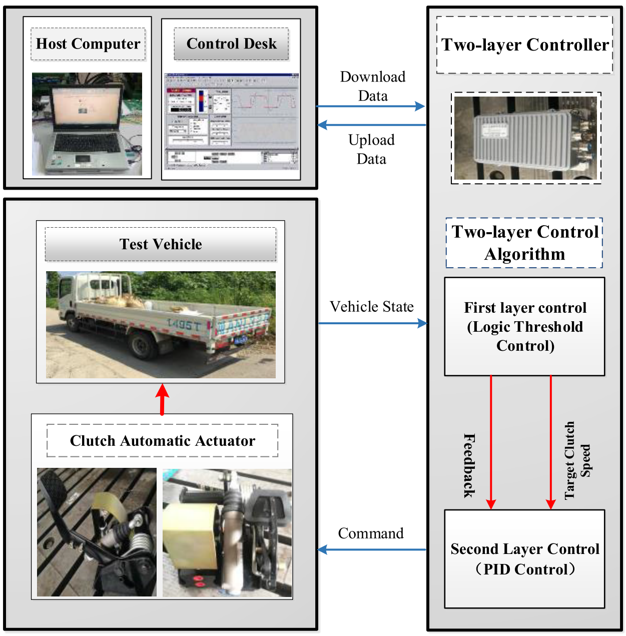

FIGURE 21. Overall configuration of the experimental test platform.

V. EXPERIMENTAL VERIFICATION

To further validate the proposed two-layer structure control strategy, experiments were carried out on a heavy-duty commercial vehicle equipped with the AMT clutch automatic actuator and HSA system introduced in Section 1. The actual test vehicle employed the same parameters as those in Table 1.

A. EXPERIMENTAL TEST PLATFORM SETUP

The experimental setup is shown in Fig. 21. A constant 100% throttle opening was adopted for the test vehicle when driven on two different slopes (13% and 22%). Since the clutch used in the test vehicle is not completely new, and it is difficult to meet ideal test conditions, the experiments were carried out under interference conditions (−10°C, clutch wear).

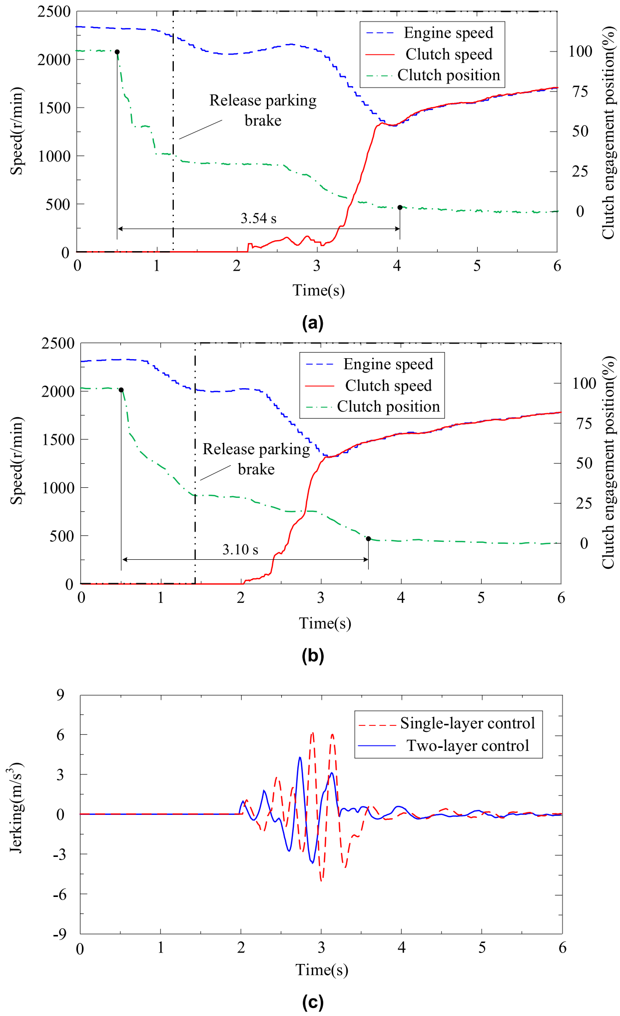

FIGURE 22. Experimental results for the 13% slope. (a) Single-layer control, (b) two-layer control, (c) jerking comparison.

B. EXPERIMENTAL RESULTS AND EVALUATION

The experimental test setup is the same as that of the simulation. After several comparative experiments, the experimental results are shown in Fig. 22 and Fig. 23, including the clutch position, engine and clutch speed, impact and other curves. Because it is difficult to obtain accurate friction work during experiments, there is no comparison curve of friction work.

As shown in Fig. 22, on the 13% slope, the start time of the vehicle under the single-layer control strategy is 3.54 s, the RMS of jerking is approximately 3.13, and the jerking duration is long. The start time of the vehicle under two-layer structure control is 3.10 s, and the RMS of jerking is approximately 2.27. Compared with the single-layer control, the start time is shortened by approximately 12.4%, and the jerking is reduced by about 27.4%.

As shown in Fig. 23, on the 22% slope, the start time of the vehicle under single-layer control strategy is 3.96 s, and the RMS of jerking is approximately 3.68. The start time of the vehicle under two-layer structure control is 3.32 s, and the RMS of jerking is approximately 2.84. Compared with the single-layer control strategy, the two-layer structure control

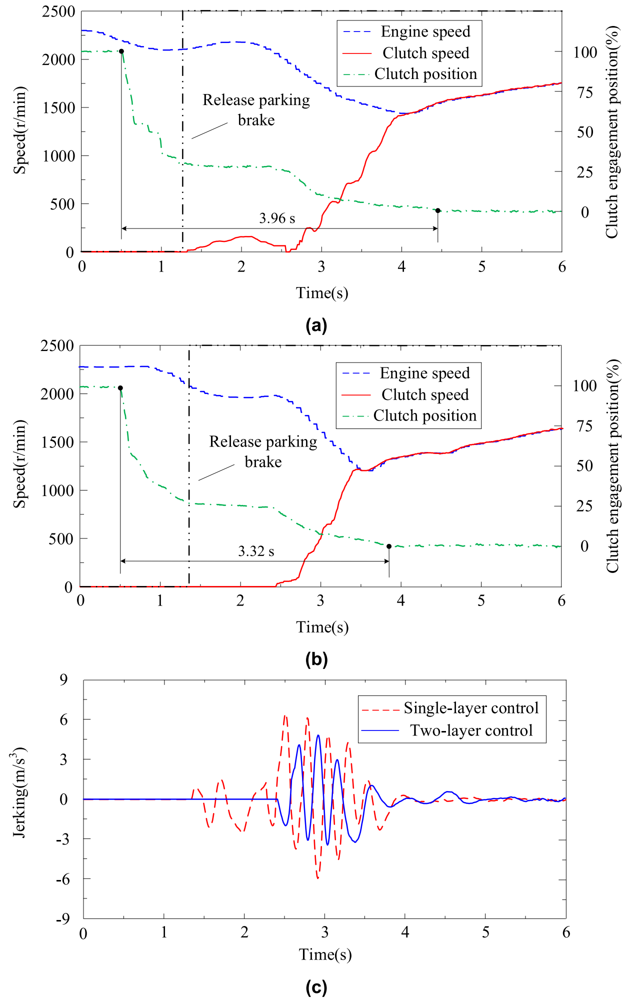

FIGURE 23. Experimental results for the 22% slope. (a) Single-layer control, (b) two-layer control, (c) jerking comparison.

TABLE 3. Comparison of the experimental results.

Evaluation index	Single-layer	Two-layer	Performance improvement
Start time (s) (13%slope)	3.54	3.10	12.4%
RMS of start-up jerking (13%slope)	3.13	2.27	27.4%
Start time (s) (22%slope)	3.96	3.32	16.2%
RMS of start-up jerking (22%slope)	3.68	2.84	22.8%

strategy can shorten the start time by 16.2% and reduce the RMS of vehicle jerking by 22.8%.

From the above analysis, it can be seen that the proposed two-layer structure control strategy can significantly improve the vehicle starting performance. For better comparison, the values of the start time and the RMS of jerking for the two groups are presented in Table 3.

VI. CONCLUSION

In this paper, first, a model of a clutch and transmission system is established, and the force of a vehicle on a hill is analyzed. Second, a clutch automatic actuator is designed and the control characteristics of the flow valve in the clutch automatic actuator are tested in experiments. Finally, a two-layer structure control strategy is proposed for clutch engagement control in the hill start of heavy-duty vehicles. The first-layer controller responds to the vehicle information and outputs the clutch speed signal to the second-layer controller. The second-layer PID controller outputs a PWM control signal for adjusting the solenoid valve of the clutch automatic actuator, thereby controlling the clutch engagement speed.

Simulation and experimental results show that the proposed control method is less affected by external interference, which not only shortens the start time of the vehicle, but also reduces the start-up jerking and friction work, and greatly improves the starting performance of the vehicle.

Further studies will be implemented in the following areas: optimization and adjustment of the control strategy and the vehicle in the process of driving clutch engagement control.

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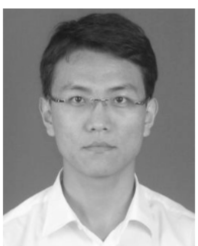

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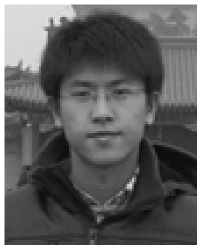

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