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Double-layer Control of an Automatic Mechanical Transmission Clutch during Commercial Vehicle Start-up

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Engagement control of an automatic mechanical transmission clutch during vehicle start-up has important influences on the safety, comfort, service life, energy consumption, and emissions of the vehicle. However, all existing control strategies use single-layer control, which leads to poor temperature adaptability. This paper develops a double-layer control strategy based on an automatic actuator to improve vehicle start-up performance. The governing characteristics of a diesel engine with a variable-speed governor are studied. The working principles and a model of the automatic actuator are described. A simulation of vehicle start-up is constructed and tested. Finally, the double-layer control strategy is verified in an experimental vehicle. The experimental and simulation results show that the double-layer control strategy provides shorter start times, less jerking, and lower friction, thereby demonstrating its effectiveness and practicability.

Keywords: automatic mechanical transmission, start-up process, clutch engagement control, double-layer control

Highlights

- *A double-layer control strategy is designed based on an automatic clutch actuator and controller.*
- *Temperature changes affect clutch engagement performance during the vehicle start-up process.*
- *Comparative outcomes verify the benefits of our scheme in terms of start time, jerking and friction work.*
- *Experiments with an actual vehicle further validate the proposed control strategy.*

0 INTRODUCTION

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 the automatic operation of transmission selection and shifting [1]. An AMT has the advantages of both an automatic transmission (AT) and a manual transmission (MT) in terms of efficiency, cost, simplicity and ease of manufacture [2]. Moreover, AMTs also do not have the shortcomings of dual-clutch transmissions (DCTs), which have high failure rates and are unsuitable for use with small-displacement engines that produce insufficient torque at low speeds [3]. AMTs are considered inexpensive add-on solutions for conventional MTs. Therefore, in recent years, AMTs have been widely used in the automotive field, especially in commercial vehicles [4].

Because AMTs are modifications of fixed-shaft geared manual transmissions, they have disadvantages, such as poor smoothness, easy power interruption, etc. [5]. By optimizing the control of clutch engagement, it is possible to reduce clutch wear and energy loss, shorten the clutch engagement process, and reduce jerking and friction [6]. Control of the clutch engagement process in AMTs has a significant impact on various performance characteristics of a vehicle, such as its start-up [7], fuel consumption [8], dynamics [9] and security [10]. This paper focuses on optimization of the clutch engagement control strategy during vehicle start-up.

Research on clutch modelling and control strategies in the start-up of AMT vehicles has been extensively studied [11]. During the vehicle start-up process, the single-layer clutch engagement control proposed in [12] is an optimal control approach that uses weighting factors for the slipping speed and control inputs to analyse clutch engagement. Huang et al. [13] used the acceleration pedal displacement and rate of change, the relative slip rates of the driving and driven plates, and changes in engine speed to represent the driving intention, clutch engagement state, and engine running state, respectively, in a dry DCT vehicle start-up fuzzy intelligent control algorithm. A combination of optimal control and an open-loop lookup table aiming to reduce slipping time and ensure engagement comfort was introduced in [14]. Bemporad et al. [15] 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 on the states. Wang et al. [16] and [17] analysed the motion

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relationships between various parts of the vehicle driveline, designed a clutch automatic actuator based on a flow solenoid valve, studied the start-up process of the vehicle, and controlled it with reference to the model's method. Amari et al. [18] proposed a real-time model predictive control (MPC) for controlling the behaviour of an AMT. Van et al. [19] deduced the optimal engagement strategy under different driver intentions based on a linear quadratic regulator. Researchers [20] proposed a fuzzy logic controller for the smooth and rapid engagement of an automatic clutch. The described controller uses both fuzzy logic and slip control algorithms to enable automatic clutch engagement. van Berkel et al. [21] proposed a new controller design that segments the clutch engagement process and distinguishes the control laws at each stage.

Although there are many studies concerning AMT clutch control, they fail to take into account deviations between the actual clutch engagement speed and the target value of the control strategy due to changes in environmental temperature.

The primary objective of this article is to develop a double-layer control strategy for a commercial vehicle AMT clutch based on a gas-assisted automatic actuator during vehicle start-up. There are three crucial novel contributions of our study.

1. First, on the basis of a first-layer reference model control strategy, a second-layer control strategy is added to improve temperature adaptability.
2. Second, a co-simulation platform is established involving Matlab/Simulink and Trucksim to verify the effectiveness of the control strategy under different temperatures.
3. Finally, an experiment of actual vehicle start-up is carried out at different temperatures.

1 SYSTEM MODELLING

1.1 Modelling a Diesel Engine with a Variable-Speed Governor

The torque curves of diesel engines are relatively flat, and a small change in the external resistance moment will lead to a large rotation speed fluctuation [22]. The resistance moment of the vehicle changes greatly, which requires the driver to adjust the throttle frequently, which may lead to driver fatigue. A variable-speed governor can allow a diesel engine to make larger torque changes under small speed changes. This can handle changes in the resistance moment and reduce driver fatigue. 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. 1 [23].

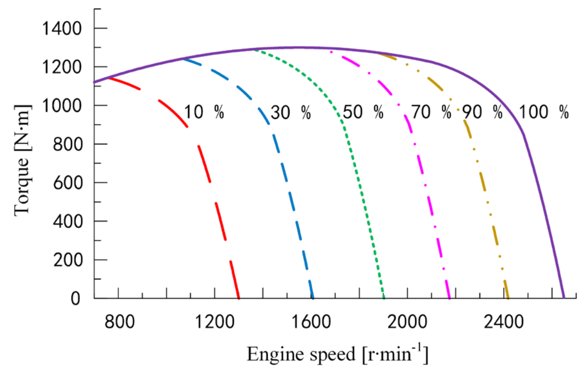

Fig. 1. *Fixed throttle characteristics of a diesel engine with variable-speed governor*

The variable-speed governor can maintain the engine speed within a certain range to prevent stalling or racing. Fig. 2 shows a characteristic torque curve of the diesel engine with the variable-speed governor at 30 % throttle opening.

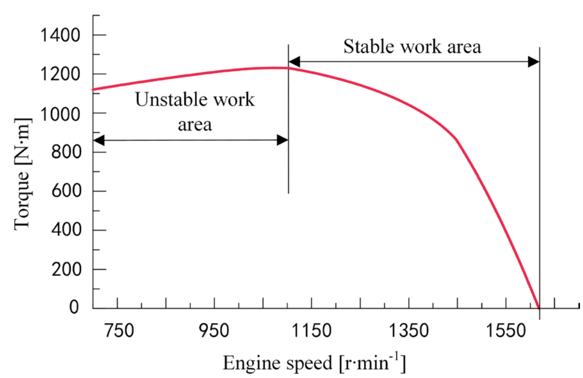

Fig. 2. *Diesel engine speed-torque curve with 30 % throttle opening*

1.2 Vehicle Driveline Modelling

A simplified diagram of the test vehicle's powertrain is shown in Fig. 3 [24]. Without considering their torsional and transverse vibrations, each part is considered to be a rigid body. The clearance between kinematic pairs is ignored. Non-damped parts can be considered as the mass is concentrated on the centre of mass; except for the clutch, tire and synchronizer, the other kinematic pairs do not consider the influence of friction.

According to the characteristics and dynamic formula of the vehicle start-up process, the clutch engagement process can be roughly divided into three

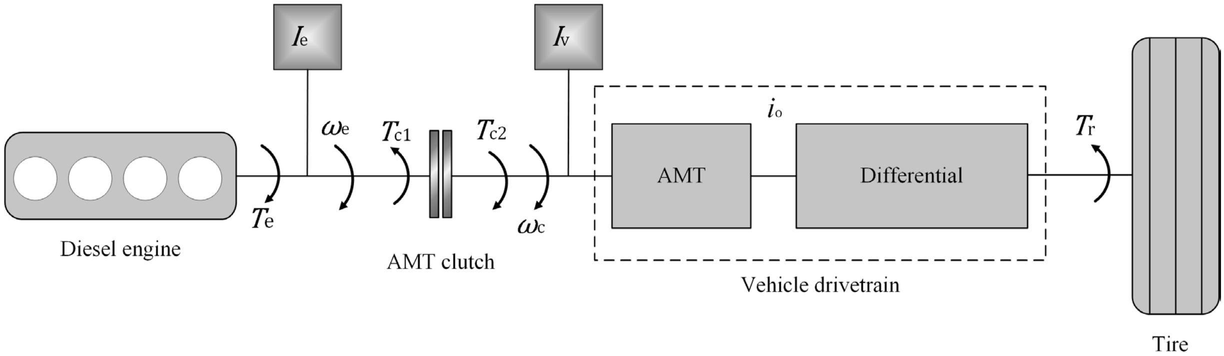

Fig. 3. *Simplified model of the test vehicle's driveline*

stages: clearance elimination, slip and friction, and synchronization [25].

The quantities equations of I_e and I_v are as follows:

$$\left.\begin{aligned} I_e &= I_1 + I_{c1} \\ I_v &= I_{c2} + I_t + I_d \end{aligned}\right\}. \quad (1)$$

(1) In the clearance elimination stage, at this time, the clutch-driven plate, transmission and tires are stationary, and the torque calculation formula of each part is as follows:

$$\left.\begin{aligned} T_e &= I_e \dot{\omega}_e \\ T_{c1} &= T_{c2} = 0 \\ T_r &= 0 \end{aligned}\right\}. \quad (2)$$

(2) In the slip and friction stage, the torques of each part are calculated by the formulas:

$$\left.\begin{aligned} T_e &= I_e \dot{\omega}_e + T_{c1} \\ T_{c1} &= \mu k_c F_c \\ T_{c2} &= I_v \dot{\omega}_c + i_o T_r \\ T_{c1} &= T_{c2} \end{aligned}\right\}. \quad (3)$$

(3) In the clutch synchronisation phase, the clutch driving plate speed and driven plate speed are the same. At this point, $\dot{\omega}_e$ and $\dot{\omega}_c$ are equal.

$$\left.\begin{aligned} T_e &= I_e \dot{\omega}_e + T_{c1} \\ \dot{\omega}_e &= \dot{\omega}_c \\ \omega_e &= \omega_c \end{aligned}\right\}. \quad (4)$$

It can be obtained from Eq. (4) that:

$$\left.\begin{aligned} T_{c1} &= T_{c2} \\ T_{c2} &= I_v \dot{\omega}_c + i_o T_r \\ T_{c1} &= I_v \dot{\omega}_e + i_o T_r \end{aligned}\right\}. \quad (5)$$

1.3 Clutch Modelling

A dry clutch relies on friction to transfer torque. The formula for calculating the transfer torque of a dry clutch [26] is:

$$T_c = \mu(T, \Delta\omega) F_c Z \times \frac{2}{3}\left(\frac{R_1^3 - R_0^3}{R_1^2 - R_0^2}\right). \quad (6)$$

The dynamic friction factor $\mu(T,\Delta\omega)$ is affected by the clutch plate temperature T and $\Delta\omega$. A simplified representation of the relationship between the clutch's engagement position and its output torque is shown in Fig. 4, ignoring changes in temperature, clutch friction plate wear and dynamic friction factors.

Fig. 4. *Relationship between clutch output torque and clutch engagement position*

1.4 Automatic Actuator Modelling

A gas-assisted hydraulic clutch actuator [27] is shown in Fig. 5. A hydraulic control cylinder is connected to the clutch pedal to control the hydraulic master cylinder [28]. Fig. 6 is a schematic diagram of the gas-assisted hydraulic automatic clutch actuator. The relationship between the motions of the parts of the automatic clutch actuator can be simplified, as shown in Fig. 7.

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

$$v_3 = \frac{S_2 k}{S_1 S_3} q_1. \tag{7}$$

The formula for the clutch displacement L is:

$$L = \int v_3 dt = \int \frac{S_2 k}{S_1 S_3} q_1 dt = \frac{S_2 k}{S_1 S_3} V_1. \tag{8}$$

Therefore, the flow and volume of oil in the hydraulic control cylinder can be controlled via the proportional flow valve, so as to control the speed and displacement of the clutch [29].

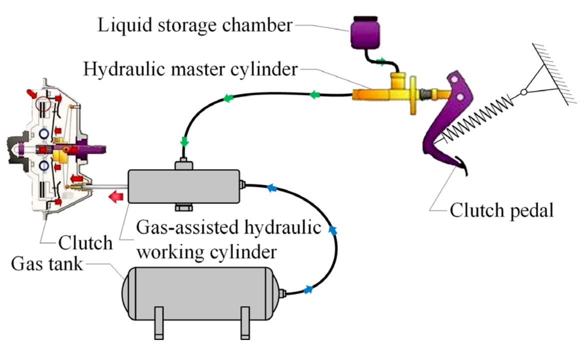

Fig. 5. *Gas-assisted hydraulic clutch actuator*

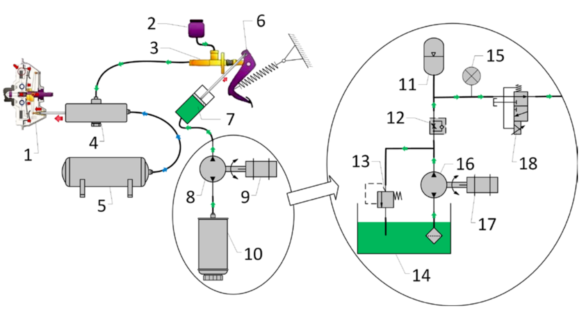

Fig. 6. *Gas-assisted hydraulic automatic clutch actuator: 1) clutch, 2) liquid storage chamber, 3) hydraulic master cylinder, 4) gas-assisted hydraulic working cylinder, 5) gas tank, 6) clutch pedal, 7) hydraulic control cylinder, 8) oil pump, 9) electric motor, 10) oil tank, 11) accumulator, 12) one-way throttle, 13) overflow valve, 14) oil tank, 15) pressure gauge, 16) oil pump, 17) electric motor, and 18) proportional flow valve*

When the proportional flow valve works, its resistance can be considered constant, so the current can be controlled by adjusting the voltage [30]. The voltage of the proportional flow valve is controlled by the duty ratio of pulse width modulation (PWM). The basic working characteristics of the proportional flow valve at a normal temperature of 25 °C are shown in Fig. 8.

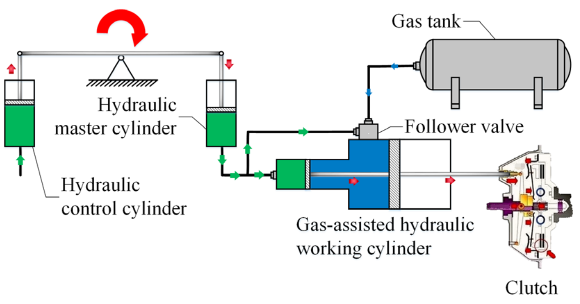

Fig. 7. *Action of the automatic clutch actuator*

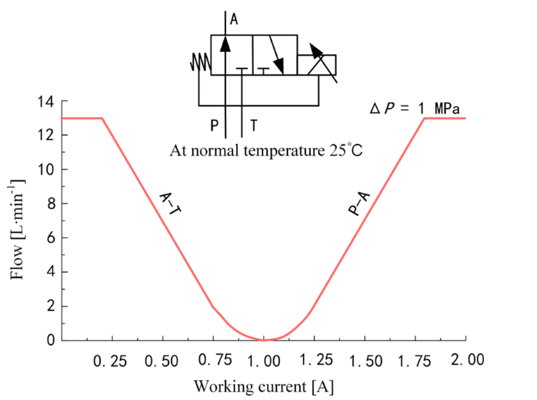

Fig. 8. *Basic working characteristics of the proportional flow valve*

The clutch engagement process under different duty ratios is shown in Fig. 9.

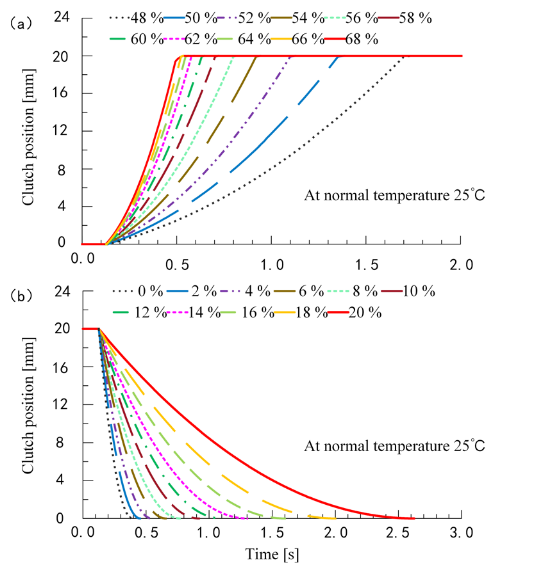

Fig. 9. *Curves of a) clutch separation, and b) clutch engagement*

2 EVALUATION INDEX

At present, the start time, jerking, and friction work are generally used as evaluation indexes for the quantitative evaluation of the vehicle start-up process [31]. Jerking is the first derivative of the longitudinal acceleration of a vehicle in the start-up process, which is used as a quantitative index to reflect the comfort level of passengers and drivers [32].

$$j = \frac{da}{dt} = \frac{d^2u}{dt^2}, \tag{9}$$

$$F_t = F_j + F_f + F_i + F_w, \tag{10}$$

$$F_t = \frac{T_{c2} i_g i_0 \eta}{r}, \tag{11}$$

$$F_j = \delta M \frac{du}{dt}. \tag{12}$$

The formula for jerking can be expressed as:

$$j = \frac{\mathrm{d}a}{\mathrm{d}t} = \frac{\mathrm{d}^2 u}{\mathrm{d}t^2} = \frac{1}{\delta M} \frac{i_g i_0 \eta}{r} \frac{\mathrm{d}T_c}{dt}. \tag{13}$$

It can be seen from Eq. (13) that jerking is proportional to the rate of change of the friction torque of the clutch. Decreasing the rate of change reduces jerking.

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 temperature of the clutch friction plate. It also reflects clutch friction plate wear and affects service life [33].

$$W = \int_{t_1}^{t_2} T_{c2}(t)\omega_e(t)\mathrm{d}t + \int_{t_2}^{t_3} T_{c2}(t)[\omega_e(t) - \omega_c(t)]\mathrm{d}t. \tag{14}$$

It can be concluded that control of the duty ratio of the flow solenoid valve is used to control the speed of clutch engagement, the clutch output torque and, ultimately, the friction work and jerking.

3 CONTROL STRATEGY

3.1 Process Analysis

In all of the control stages of a vehicle start-up process, clutch engagement control is crucial [34]. According to Section 1.2, the start-up process is subdivided into five parts [35], as shown in Fig. 10.

Phase 1: the clutch driving and driven plates are not in contact. Phase 2: there is contact friction as the driving plate is rotating, but the driven plate is still. Phase 3: the clutch driven plate starts to rotate. Phase 4: the clutch driving and driven plate speeds differ between the critical value and 0. Phase 5: the rotating speeds are consistent until the end of clutch engagement.

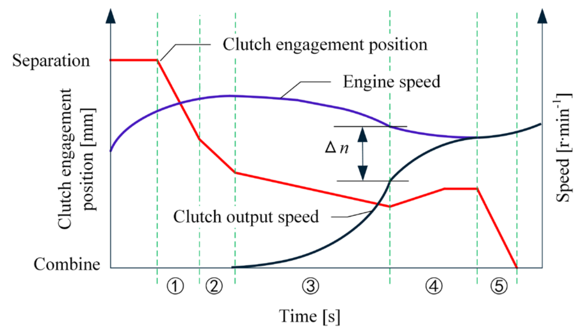

Fig. 10. *Control curve of a vehicle start-up process*

3.2 Double-Layer Control Strategy

In order to improve the control accuracy, reliability and robustness of the control system, a double-layer control method is proposed. The first layer uses a reference model control [25]. However, due to changes in ambient temperature and other factors, there is a deviation between the actual clutch engagement speed and the target clutch engagement speed. Therefore, proportion integration differentiation (PID) control is adopted as the second layer of control to improve the control accuracy, reliability and robustness of the system. Its schematic diagram is shown in Fig. 11.

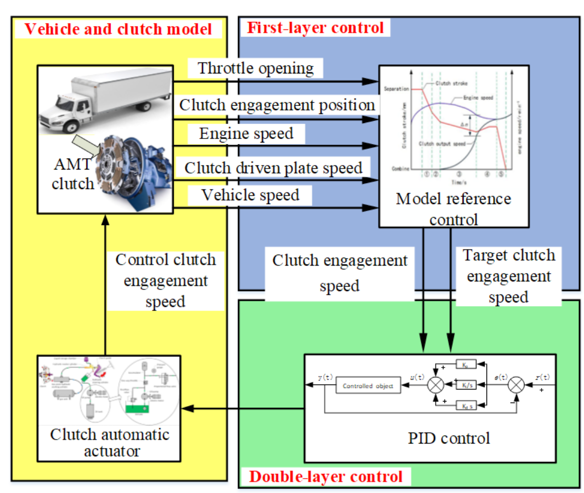

Fig. 11. *Double-layer clutch control strategy*

Some key signals are transmitted to the first-layer control via the vehicle and clutch model, 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. 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 serves as a given value and the clutch engagement speed serves as feedback, with the clutch and its automatic actuator being the controlled objects. The PID control parameters are set as $K_p = 0.05$, $K_i = 27.6$ and $K_d = 0.01$.

4 SIMULATION ANALYSIS

Trucksim and the Matlab/Simulink co-simulation platform were used to establish a model of the vehicle start-up process in order to evaluate the effects of the double-layer control strategy. The parameters of the experimental vehicle adopted in the simulation model are shown in Table 1.

Table 1. *Main parameters of the experimental vehicle*

Parameter	Value
m	8190 kg
r	397 mm
i_0	30.786
η	0.99

In this paper, only a typical 30 % throttle opening vehicle start-up process is considered, because it is representative. The simulation results are shown in Figs. 12 and 13. The start time, the root mean square (RMS) of jerking, and the friction work of the single-layer control strategy at low temperature are 28.3 % longer, 19.8 % lower and 8.0 % greater, respectively, than at normal temperature. This demonstrates that temperature changes have a great effect on the performance of single-layer controlled clutch engagement during the vehicle start-up process.

Compared with the simulation results for the double-layer control strategy at normal and low temperatures, the start time, RMS jerking and friction work of the single-layer control strategy at low temperature are 5.7 % longer, 7.4 % lower and 6.6 % more, respectively, than at normal temperature. This demonstrates that the double-layer control strategy can increase the performance degradation caused by temperature reduction.

As can be seen from Fig. 13, compared with the single-layer control strategy, the start time with the double-layer control strategy is 5.8 % shorter, the RMS of jerking is 31.1 % lower, and the friction work is reduced by 7.5 %. As shown in Fig. 14, the start time, the RMS of jerking and the friction work of the

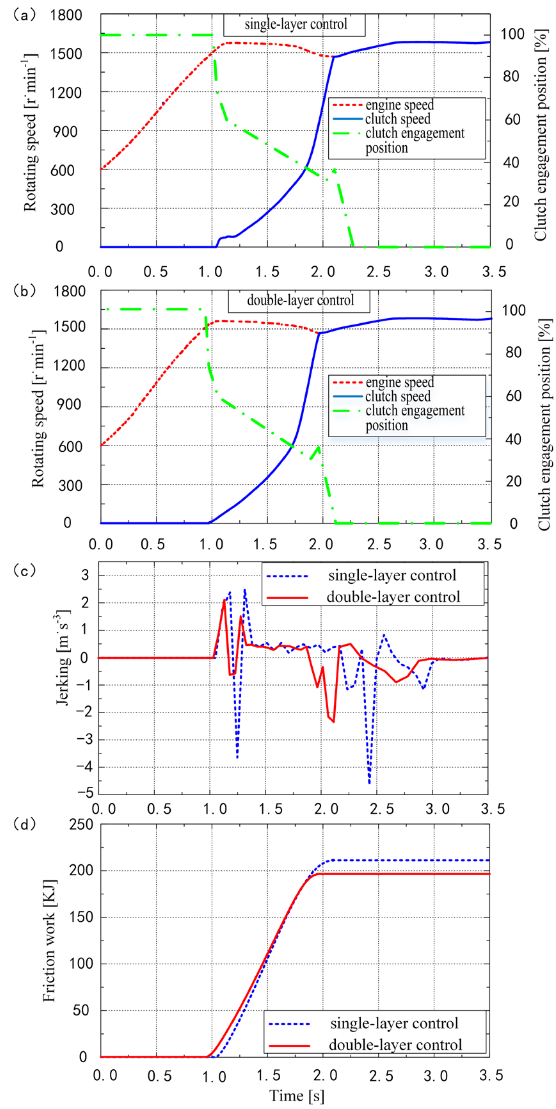

Fig. 12. *Simulation results at normal temperature (25 °C), a) single-layer control, b) double-layer control, c) jerking contrast, and d) friction work contrast*

Table 2. *Comparison of simulation results (*RMS value)*

Tempe-rature	Control strategy	Start time [s]	*Start-up jerking [-]	Friction work [kJ]
Normal (25 °C)	Single layer	1.27	1.06	212
	Double layer	1.20	0.73	196
Low (–10 °C)	Single layer	1.63	0.85	229
	Double layer	1.29	0.68	209

double-layer control strategy are reduced by 20.8 %, 20.0 % and 8.7 %, respectively. Compared with the single-layer control strategy, the double-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.

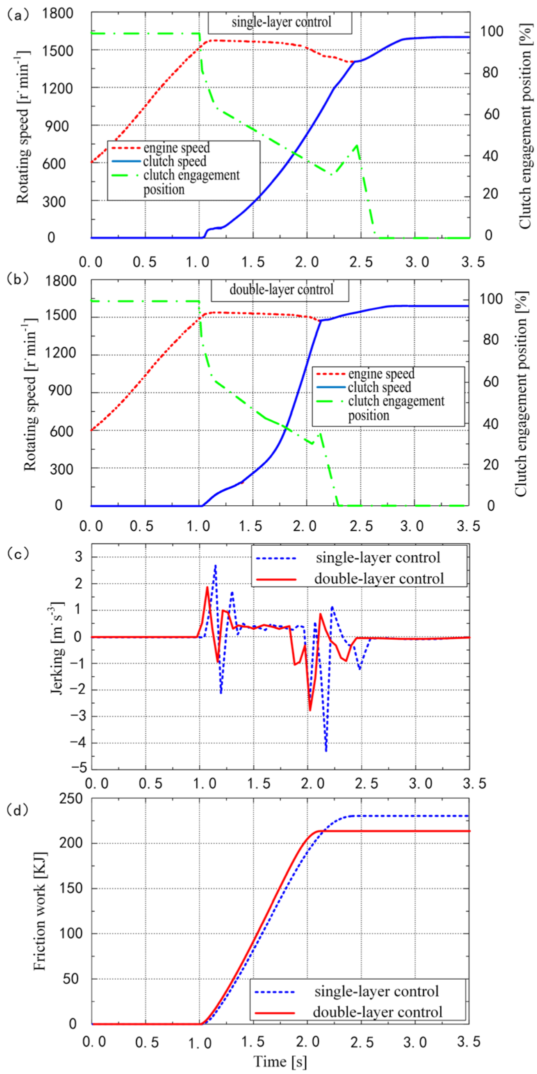

Fig. 13. *Simulation results at low temperature (–10 °C), a) single-layer control, b) double-layer control, c) jerking contrast, and d) friction work contrast*

5 EXPERIMENTAL RESEARCH

On the basis of an automatic clutch actuator, a double-layer clutch control strategy for a vehicle start-up process was designed. The controller and control strategy were verified on a test vehicle. Fig. 14 shows the experimental vehicle and relevant components. Under normal temperature (25 °C; Fig. 15) and low-temperature conditions (–10 °C; Fig. 16), a constant 30 % throttle opening was adopted for a test vehicle driven on a flat, horizontal road surface.

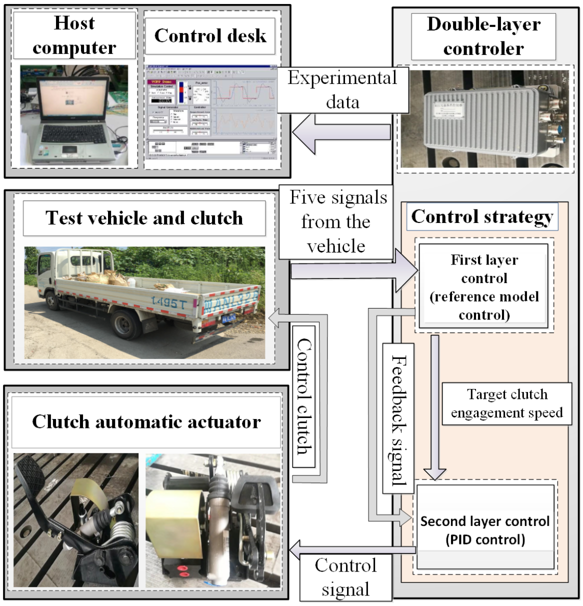

Fig. 14. *Experimental setup*

As can be seen from Fig. 15, at normal temperature, compared with the single-layer control strategy, the double-layer control strategy can shorten the start time by 19.5 %, reduce the RMS of vehicle jerking by 28.6 %, and reduce the friction work by 13.8 %.

The same conclusion can also be drawn from the low-temperature experiment (Fig. 16). At low temperature, compared with the single-layer control strategy, the double-layer control strategy can shorten the start time by 34.9 %, reduce the RMS of vehicle jerking by 35.4 %, and reduce the friction work by 30.3 %. By adopting a double-layer control strategy, the performance of the vehicle start-up process is significantly improved.

For better comparison, the values for start time, jerking RMS and friction work for the two groups are presented in Table 3.

Table 3. *Comparison of the experimental results (*RMS value)*

Tempe-rature	Control strategy	Start time [s]	*Start-up jerking [-]	Friction work [kJ]
Normal (25 °C)	Single layer	2.81	2.62	509
	Double layer	2.26	1.87	436
Low (–10 °C)	Single layer	3.61	2.06	614
	Double layer	2.35	1.33	428

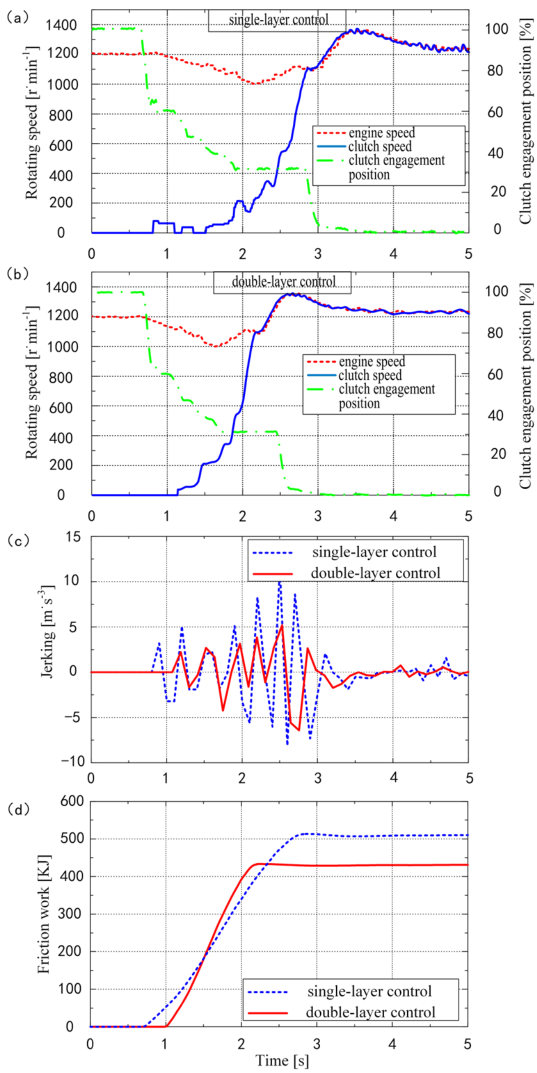

Fig. 15. *Experimental results at normal temperature (25 °C), a) single-layer control, b) double-layer control, c) jerking contrast, and d) friction work contrast*

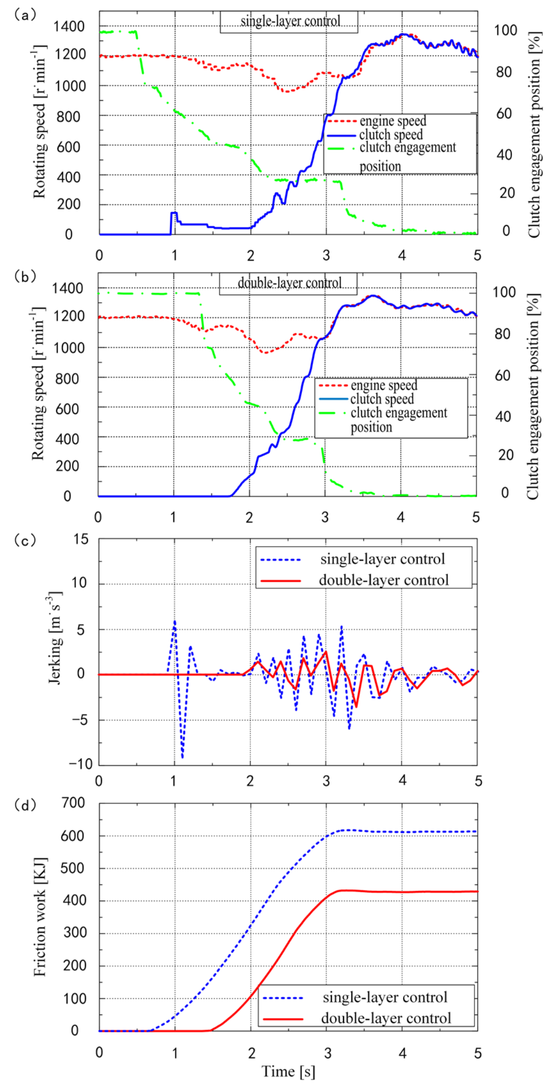

Fig. 16. *Experimental results at low temperature (–10 °C), a) single-layer control, b) double-layer control, c) jerking contrast, and d) friction work contrast*

6 CONCLUSIONS

In this paper, a double-layer control strategy based on an automatic clutch actuator was designed to achieve better vehicle start-up performance. Due to environmental factors such as temperature changes, the actual and target clutch engagement speeds will have a certain amount of deviation.

Simulations and experimental results show that the adoption of a double-layer control strategy reduces the influence of temperature changes on start-up performance.

The simulation and experimental results further show that compared with a single-layer control strategy, the double-layer control strategy improves start-up performance.

A possible direction for future work could be to develop a neural network-based self-learning control method to improve the double-layer control strategy. It could consider changes in clutch friction plate wear and friction plate temperature to further improve vehicle start-up performance.

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8 NOMENCLATURE

I_1 engine rotational inertia, [kg·m²]
I_{c1} clutch driving plate rotational inertia, [kg·m²]
I_{c2} clutch driven plate rotational inertia, [kg·m²]
I_t transmission, main reducer, driveshaft, etc. rotational inertia, [kg·m²]
I_d differential, half-shaft, etc. rotational inertia, [kg·m²]
T_e engine output torque, [N·m]
ω_e engine crankshaft angular velocity, [rad·s^{-1}]
I_e engine and clutch driving plate rotational inertia, [kg·m²]
T_{c1} torque transferred by the clutch, [N·m]
T_{c2} torque input of the transmission, [N·m]
ω_c transmission shaft angular velocity, [rad·s^{-1}]
I_v vehicle equivalent rotational inertia, [kg·m²]
i_o powertrain system transmission ratio, [-]
T_r ground resistance moment, [N·m]
μ clutch plate friction coefficient, [-]
F_c (cluch) pressing force, [N]
k_c coefficient, [-]
T clutch friction plate temperature, [ºC]
$\Delta\omega$ clutch angular velocity difference, [rad·s^{-1}]
Z Number of clutch friction pairs, [-]
R_0 clutch friction plate internal radius, [mm]
R_1 clutch friction plate external radius, [mm]
q_1 liquid in/out hydraulic control cylinder, [m³·s^{-1}]
V_1 hydraulic control cylinder volume, [m³]
S_1 hydraulic control cylinder section area, [m²]
S_2 hydraulic master cylinder section area, [m²]
S_3 gas-assisted hydraulic working cylinder section area, [m²]
k piston displacement proportionalcoefficient, [-]
v_3 piston movement speed, [m·s^{-1}]
j start-up jerking, [m·s^{-3}]
a vehicle longitudinal acceleration, [m·s^{-2}]
u vehicle speed, [m·s^{-1}]
t time, [s]
F_t driving force, [N]
F_j acceleration resistance, [N]
F_f rolling resistance, [N]
F_i gradient resistance, [N]
F_w air resistance, [N]
η drive train mechanical efficiency, [-]
r wheel radius, [mm]
δ rotational mass conversion coefficient, [-]
M vehicle mass, [kg]
W friction work, [J]
t_1 point at which the driving and driven plates initiate friction torque, [s]
t_2 point at which the clutch driven plate starts to rotate, [s]
t_3 point at which the driving and driven plates attain the same speed, [s]

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