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SHIFTING CONTROL OF AN AUTOMATED MECHANICAL TRANSMISSION WITHOUT USING THE CLUTCH

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ABSTRACT−The automated mechanical transmission (AMT) is gaining popularity in the automotive industry, due to its combination of the advantages of mechanical transmissions (MT) and automatic transmissions (AT) in terms of fuel consumption, low cost, improved driving comfort and shifting quality. However, the inherent structural characteristics of the AMT lead to disadvantages, including excessive wear of the clutch plates and jerk and traction interruption during the shift process, that severely affect its popularity in the automatic transmission industry. The emerging technology of shifting control without the use of the clutch is a promising way to improve the shifting transients of AMTs. This paper proposes a control algorithm that combines speed and torque control of the AMT vehicle powertrain to achieve shifting control without using the clutch. The key technologies of accurate engine torque and speed control and rapid position control of the shift actuators are described in detail. To realize accurate engine speed control, a combined control algorithm based on feed-forward, bang-bang and PID control is adopted. Additionally, an optimized closed-loop position control algorithm based on LQR is proposed for the shift actuators. The coordinated control algorithm based on engine and shift actuator control is described in detail and validated on a test vehicle equipped with an AMT. The results show that the coordinated control algorithm can achieve shifting control without the use of the clutch to improve driving comfort significantly, reduce shift transients and extend the service life of the clutch.

KEY WORDS : AMT, Shifting without using clutch, LQR control, Feed-forward control, Bang-bang control

1. INTRODUCTION

There has been a clear trend in the automotive industry in recent years toward better ride comfort and fuel efficiency. As the power transfer unit in automobiles, the transmission plays an important role in vehicle performance and fuel economy. There are several types of transmissions and associated technologies that provide different levels of performance. Unlike other transmission types, such as the automatic transmission (AT), the continuously variable transmission (CVT), and the dual clutch transmission (DCT), the AMT retains some of the unique advantages of manual transmissions, including the ability to use existing manufacturing facilities to reduce production costs, high efficiency and lower weight. However, AMTs also have disadvantages, including the jerk generated during shifting, excessive wear in the clutch friction plates, and torque interruption during the shifting process, that significantly affect the popularity of AMTs in the automatic transmission industry.

To improve the performance of AMTs, a new method of shifting without using the clutch is proposed in this paper. The literature on methods for shifting without using the clutch is relatively small. Wang (2007) and Pei (2009) introduced a control algorithm for shifting without releasing the clutch in heavy commercial vehicle AMTs in which an additional brake device was installed on the input shaft of the transmission, thereby reducing the interruptions caused by releasing the clutch and adjusting the torque during the shifting process. Magnus and Lars (2000) analyzed the resonance of the drivetrain system caused by shifting without using the clutch and introduced a coordinated control algorithm based on a state observer and PID control to reduce the effect of the drivetrain resonance. Wang (2006) proposed a combined control algorithm for engine control during shifting without the use of the clutch based on adjusting the throttle opening, ignition timing, and fuel injection, and a simulation was performed with good results. However, there has been no research on shifting without using the clutch in conventional AMTs without structural modifications. With the development of electronic engine control systems, it is feasible to shift without using the clutch in a conventional AMT. A combined control algorithm for shifting without using the clutch in a conventional AMT is proposed in this paper. Precise engine speed control, optimized control of the gear selection and shifting processes and a system-level coordinated control algorithm are used to exploit the potential of shifting without using the clutch in conventional AMTs with no structural modifications.

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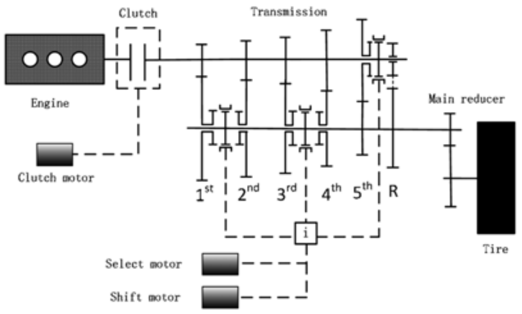

Figure 1. Structural diagram of vehicle drivetrain with AMT.

2. TRANSIENT DYNAMIC ANALYSIS FOR AMT SHIFTING WITHOUT USING CLUTCH

The structural diagram of an automotive powertrain is shown in Figure 1. The main components of the powertrain are the engine, the clutch, the wheel, the main reducer and the transmission, which consists of five forward gears and one reverse gear. The main difference between the conventional shifting process and shifting without using the clutch is that the inertia of the components driving the synchronizer in the two processes is significantly different. The inertia of the conventional synchronizer input end includes the inertias of the clutch driven plate, the transmission input shaft and the constant-mesh gears. The driving inertia of the synchronizer when the clutch is engaged during the shifting process not only includes the conventional synchronous inertia but also includes the inertias of crankshaft, the flywheel and the clutch driving plate. Therefore, the transient dynamics are quite different. The process of shifting without using the clutch can be divided into five phases: decreasing the engine torque, shift off, adjusting engine speed, shift on and restoring the engine torque.

A 3-DOF finite element model of the AMT is obtained by considering the engine, the clutch, the transmission input and output shafts and the vehicle. Because the clutch stays engaged during the shifting process, the released state of the clutch is ignored. Thus, the pre-shifting, gear shifting off and synchronization states are considered. The simplified lumped-inertia powertrain model is shown schematically in Figure 2.

2.1. Dynamic Analysis of Pre-shifting

The clutch and the synchronizer stay engaged before shifting. Thus, the powertrain can be considered a closed system, and the clutch rotational speed is proportional to the transmission output shaft speed, i.e., $\dot{\theta}_c = \dot{\theta}_o i_g$. The equations of motion of the system are as follows:

$$J_e\ddot{\theta}_e = T_e - T_{eL} - k_c(\theta_e - \theta_c) - B_c(\dot{\theta}_e - \dot{\theta}_c) \tag{1}$$

$$\begin{aligned}[(J_c+J_i)i_g^2+J_o]i_m^2\ddot{\theta}_c/i_g i_m &= k_c(\theta_e-\theta_c)+B_c(\dot{\theta}_e-\dot{\theta}_c)\\ &-k_w(\theta_c/i_g i_m-\theta_v)-B_w(\dot{\theta}_c/i_g i_m-\dot{\theta}_v)\end{aligned} \tag{2}$$

$$J_v\ddot{\theta}_v = k_w(\theta_c/i_g i_m - \theta_v) + B_w(\dot{\theta}_c/i_g i_m - \dot{\theta}_v) - T_L \tag{3}$$

$$\begin{aligned}T_L &= T_{incline} + T_{aerodynamic} + T_{rolling\ resist\ ance} = (Mgsin\varphi\\ &+\frac{1}{2}\rho_{air}AC_D V^2 + Mgf)R\end{aligned} , \tag{4}$$

where J is the inertia, θ and its two derivatives are the rotational displacement, velocity, and acceleration, ω is the rotational speed, T is the torque, k is the stiffness coefficient, and B is the damping coefficient. The subscripts e, c, i, o, and v indicate the engine, clutch, transmission input shaft, output shaft, and vehicle, respectively. Additionally, i_g is the gear ratio, i_m is the main reducer ratio, T_L is the load torque from the road, T_{eL} is the engine inner resistance torque, M is the vehicle mass, g is gravity, φ is the angle of inclination, A is the vehicle area, C_D is the drag coefficient, V is the vehicle speed, ρ_{air} is the air density, f is the rolling resistance coefficient, and R is the radius of the tire.

2.2. Dynamic Analysis after Shifting Off

The system becomes a 4-DOF system after shifting off. The main characteristic of this phase is that the clutch is kept closed from shifting off until the gears engage again. Thus, the inertia of the driving components of the synchronizer includes the inertias of the engine, the clutch, the transmission input shaft and the constant-mesh gears. The equations of motion of this phase are as follows:

$$J_e\ddot{\theta}_e = T_e - T_{eL} - k_c(\theta_e - \theta_c) - B_c(\dot{\theta}_e - \dot{\theta}_c) \tag{5}$$

$$(J_c + J_i)\ddot{\theta}_c = k_c(\theta_e - \theta_c) + B_c(\dot{\theta}_e - \dot{\theta}_c) \tag{6}$$

$$J_o\ddot{\theta}_o = -k_w(\theta_o/i_m - \theta_v) - B_w(\dot{\theta}_o/i_m - \dot{\theta}_v) \tag{7}$$

$$J_v\ddot{\theta}_v = k_w(\theta_o/i_m - \theta_v) + B_w(\dot{\theta}_o/i_m - \dot{\theta}_v) - T_L \tag{8}$$

As shown in Equation 5, to achieve the different engine speeds needed during upshift and downshift, the engine torque, T_e, can be adjusted.

2.3. Dynamic Analysis of Synchronization

The system still has 4 DOF during the synchronization process. The engine speed regulation should be started soon after shifting off, and gear selection can be performed in parallel with the speed regulation. Generally, the speed regulation process takes longer than the gear selection, and

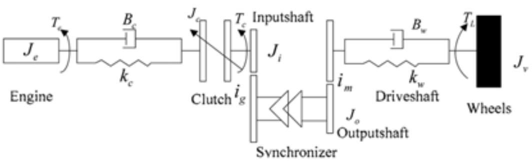

Figure 2. Simplified model of AMT drivetrain.

thus the shifting on action can proceed after the speed regulation is accomplished. However, errors always exist, regardless of how accurate the engine speed control is. Because the speed difference $\Delta\omega$ between the driving and the driven parts of the synchronizer cannot be zero during synchronization, the equations of motion of this phase are as follows:

$$J_e\dot{\omega}_e = T_e - T_{eL} - k_c(\theta_e - \theta_c) - B_c(\dot{\theta}_e - \dot{\theta}_c) - T_m \tag{9}$$

$$(J_c + J_i)\ddot{\theta}_c = k_c(\theta_e - \theta_c) + B_c(\dot{\theta}_e - \dot{\theta}_c) - T_m \tag{10}$$

$$J_o\ddot{\theta}_o = T_m - k_w(\theta_o/i_m - \theta_v) - B_w(\dot{\theta}_o/i_m - \dot{\theta}_v) \tag{11}$$

$$J_v\ddot{\theta}_v = k_w(\theta_o/i_m - \theta_v) + B_w(\dot{\theta}_o/i_m - \dot{\theta}_v) - T_L \tag{12}$$

$$T_m = \frac{FR\mu}{\sin\alpha} \tag{13}$$

where T_m is the synchronizing torque in the synchronizer cone, F is the shifting force applied to the synchronizer, R is the average effective radius, μ is the friction coefficient between the friction surfaces of the ring and the ring gear, and α is the cone angle of the ring.

3. ANALYSIS OF CONTROL ALGORITHM FOR SHIFTING WITHOUT USING CLUTCH

The conventional shifting process can be divided into seven operating phases: gear engaged, clutch release, gear shift off, gear selection, gear shift on, clutch engagement and engine torque restoration. The shifting process without the use of the clutch can be divided into five phases: engine torque reduction, gear shifting off, engine speed regulation, gear shifting on and engine torque restoration. The control logic of the upshift and downshift processes is schematically shown in Figs. 3 and 4.

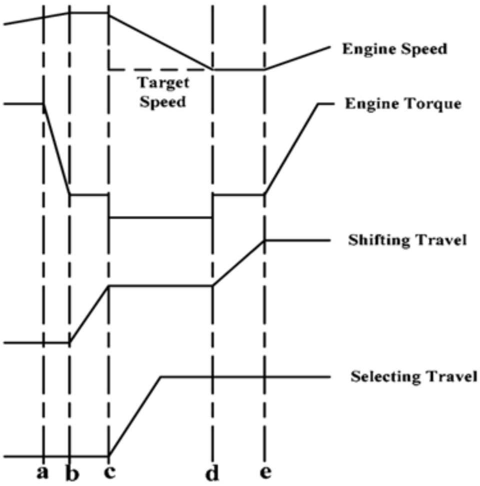

Figure 3. Control logic for upshift process.

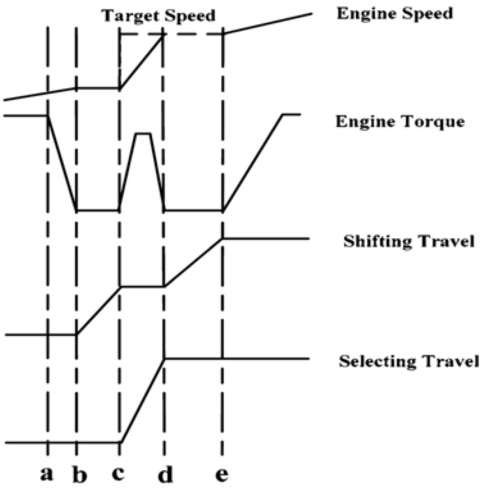

Figure 4. Control logic for downshift process.

3.1. Control Strategy for the Shift Off Process

Shifting off can be easily achieved without excessive wear of the gear faces when zero torque is transmitted to the driving shaft of the synchronizer. It is easy to shift off in the conventional shifting process because the clutch is released, and no torque exists on the driving shaft of the synchronizer. Thus, the engine torque should be decreased as much as possible (i.e., to the idle level) when shifting off without using the clutch such that the engine output torque is sufficient only to overcome its internal resistance and no torque is transferred to the driving shaft of the synchronizer. The control logic is shown in phase a-b of Figure 3 and Figure 4.

3.2. Control Strategy for Engine Speed Regulation Phase

In this phase, the gear disengages completely, and the gear selection starts as soon as the shifting off action is completed. The engine speed regulation is carried out simultaneously. The goal of this phase is to adjust the engine speed as quickly as possible to the ideal velocity required by each gear ratio. The connection between the engine and the wheel is assumed to be rigid, and thus the relation between the engine speed and the wheel speed is expressed as follows:

$$\dot{\theta}_e = \dot{\theta}_v \cdot i_m \cdot i_n \tag{14}$$

After shifting off, the target engine speed is

$$\dot{\theta}_{e\_t} = \dot{\theta}_t \cdot i_m \cdot i_{n+1} \tag{15}$$

Because the inertia of the vehicle is comparatively large, the vehicle speed can be assumed to be constant. The speed difference $\Delta\omega$ between the current engine speed and the target engine speed is the necessary engine speed adjustment:

$$\Delta\omega = (i_{n+1} - i_n) \cdot i_0 \cdot \dot{\theta}_t \quad (16)$$

where i_n is the current gear ratio, and i_{n+1} is the ratio of the gear to be engaged. When an upshift is performed, i.e., when $i_{n+1} < i_n$, the engine speed should be decreased by $\Delta\omega$ according to Equ. 7. Contrarily, the engine speed should be increased by $\Delta\omega$ when a downshift is performed. Consequently, the engine speed control strategies for upshift and downshift are different. As seen in phase c-d of Figs. 3 and 4, the most effective way to decrease the engine speed during upshift is to adjust the engine output torque to zero. According to Equ. 4, the engine deceleration depends on the engine internal resistance torque. Therefore, the engine speed boost can be easily achieved by increasing the engine output torque as calculated by the feedback control algorithm.

3.3. Control Strategy for Engine Speed Regulation Phase

The goal of this phase is to complete the gear shifting on process as quickly as possible and subsequently to restore the engine torque to the level that the driver has commanded by the gas pedal. The shift on action can be performed soon after the engine speed reaches the target engine speed. The engine torque should be restored to the commanded level at an appropriate rate to avoid affecting the quality of the shift.

4. ENGINE CONTROL APPROACH AND ALGORITHM

In the strategy discussed above, accurate engine speed control is critical to shifting without the use of the clutch, and poor speed control can lead to excessive wear of the synchronizer and to driving discomfort.

4.1. Engine Control Approach

To meet strict requirements for driver comfort and safety, the engine management system (EMS) allows the engine torque to be controlled temporarily by engine subsystems (such as the starter control, idle control, engine speed control and component protection control systems), vehicle functions (such as the operation of vacuum power steering and air conditioning), and the powertrain control system (such as the control of the automatic transmission).

There are two approaches to adjusting the engine output torque. One approach is to change the air intake quantity at the engine throttle. This is a slow process with a time delay, but it is adequate when a fast response is not required. The other approach is to adjust the spark timing or to shut off the fuel injection to the cylinder. This approach produces a quick response to the torque command.

4.1.1. Engine unloaded condition

The components that drive the synchronizer, such as the engine, the clutch and the input shaft of the transmission, stay disengaged after shifting off. The connection between the engine and the wheel is interrupted, and the components enter the unloaded working condition. The engine speed regulation only occurs during this stage, and thus the behavior when the driving components are unloaded is notably important for accurate engine speed control.

The engine can be considered as a typical inertial link with a transfer function that can be written as follows:

$$G_e(s) = \frac{K}{Ts+1}, \quad (17)$$

where T is the time constant, and K is the gain.

4.1.2. Character of engine unloaded condition

During the test, the clutch is kept engaged, and the transmission stays in the neutral gear. The engine torque command is sent from the TCU to the ECU through the CAN. A step function and an approximate sine function are used as the commanded engine torques to test the step response and the tracking response, respectively, of the engine. The test results are shown in Figure 5 and Figure 6. The engine torque is delayed by 0.05 s from the commanded torque, and the engine has excellent response characteristics, creating a technological basis for precise engine speed control.

4.2. Engine speed control algorithm

As described above, the engine control can be divided into upshift and downshift control, and the different control targets require different control algorithms. It is challenging to decrease the engine speed because the deceleration is dependent on the internal resistance torque of the engine. It is much easier to increase the engine speed by increasing the fuel injection. Interrupting or increasing the fuel injection is realized by sending a torque request from the TCU to the ECU. Based on the different control targets and principles of the upshifting and downshifting processes, coordinated control algorithms will be proposed for each.

4.2.1. Coordinated engine speed-reduction control algorithm based on feed-forward, bang-bang and PID control

The goal of the engine controller during the upshift process is to decrease the speed to the target speed in a time-

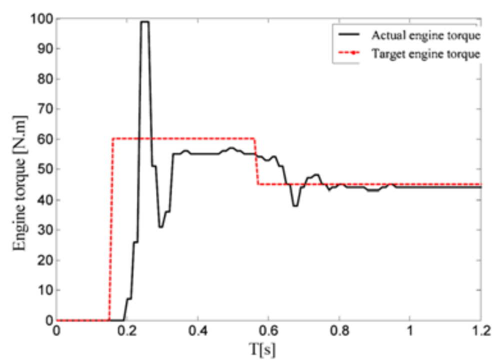

Figure 5. Engine torque step-response curve.

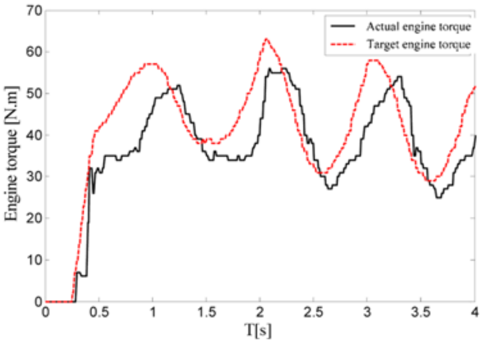

Figure 6. Engine torque tracking curve.

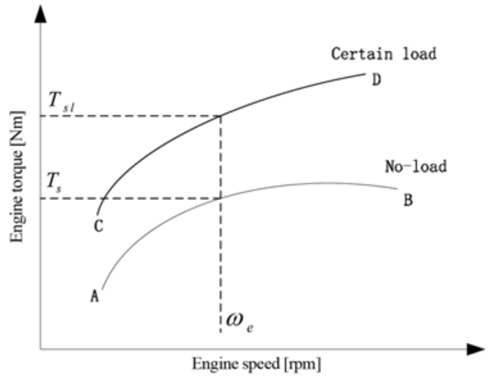

Figure 7. Engine load behavior.

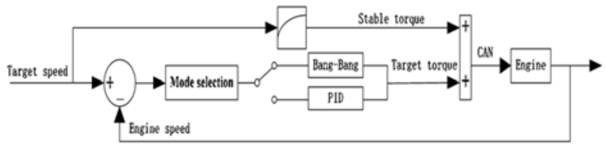

Figure 8. Control algorithm for decreasing engine speed.

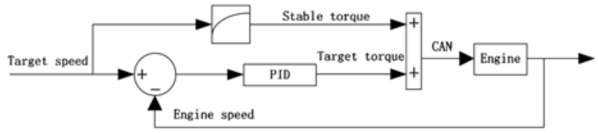

Figure 9. Control algorithm for increasing engine speed.

optimal way. The most effective way to decrease the speed is to decrease the engine output torque to zero through CAN communication and to let the TCU take over the engine torque control temporarily from the ECU. After the engine has reached the target speed, sufficient engine torque should be applied to sustain the engine speed to ensure good transient characteristics during synchronization. This torque level is called the engine stable torque, and it varies with the engine speed according to the engine characteristics as shown in Figure 7. The engine stable torque required to sustain different engine speeds under unloaded conditions can be obtained through calibration.

To achieve the target speed, a coordinated control strategy based on feed-forward, bang-bang and PID control is adopted, as shown schematically in Figure 8. There are two modes in the control algorithm. The mode is selected based on the difference between the current speed and the target speed. When a threshold value is surpassed, the bang-bang control strategy is chosen to maximally decelerate the target torque to zero. In a sense, this strategy can be seen as oil-break control. When the speed difference is less than the threshold value, the control strategy switches to PID with which the engine speed can be regulated to the target speed smoothly and precisely. After the engine speed has reached the target speed, a stable torque is calculated through the feed-forward algorithm to ensure that the engine speed remains near the target speed and to guarantee the shift quality.

4.2.2. Control algorithm for increasing engine speed

The goal of the downshift process is to increase the engine speed to the target speed. It is easiest to increase the engine speed by increasing the engine torque, which can be achieved by sending the required torque command from the TCU to the ECU. Consequently, an algorithm based on combined feed-forward and PID control is adopted, as shown schematically in Figure 9. After the speed has been adjusted to the target speed through the torque calculated by the PID feedback algorithm, a stable torque is applied to maintain the current engine torque near the target speed to ensure good shift quality.

4.2.3. Engine speed control requirements

Because the driving components of the synchronizer are much larger when shifting without the clutch than in the conventional shifting process, inaccurate speed control will result in a large speed difference during synchronization, which can lead to excessive wear of the ring and ring gear. In this research, the target engine speed control precision is set at 50 RPM faster or slower than the target speed. The duration for which the engine speed stays within the target range should also be longer than 0.4 s to maintain the speed difference within a small range during the synchronization process.

4.3. Engine speed control tests and discussion

To verify the engine speed control algorithm, experiments are performed on an 0.8 L gasoline engine, which is installed in a car with an AMT. Because only the engine speed is tested, the shifting action can be ignored. The gear is shifted to neutral, and the clutch remains closed throughout the experiments. Decreasing and increasing engine speed experiments are performed. The initial speed is set at 3500 rpm, and the target speed is set at 2000 rpm in the decreasing engine speed tests; and the initial speed is set at 2000 rpm, and the target speed is set at 4000 rpm in the increasing speed tests. The test results are shown in

Figure 10 and Figure 11. During the period of decreasing speed, the maximal deceleration is obtained by decreasing the engine torque to zero. After the speed difference is reduced below the threshold, the controller switches to the PID algorithm in which the engine speed is successfully adjusted into the target range. The engine is maintained within this speed range for greater than 0.5 s. Thus, the control targets are met. Moreover, it takes only approximately 0.8 s to complete the engine regulation process. The increasing speed test results demonstrate better transient characteristics: a shorter duration of the total speed regulation process and better quality without fluctuation after the speed has reached the target speed.

5. SHIFTING MOTOR OPTIMAL CONTROL ALGORITHM

5.1. State-space Model of Motor Actuator

The AMT shifting process requires three stages in order: gear disengagement, selection, and gear engagement. To accomplish the process of gear shifting, the position of the actuator must be controlled precisely at each stage. Thus, coordination is necessary to connect the three stages smoothly. In the shifting process, two motors are used to operate the shifting and selecting motions to ensure that the operations can be decoupled. Therefore, the control of shifting in the executing mechanism can be decomposed into two independent motor control problems.

The voltage and torque equations for the DC motor actuator are as follows (Yu *et al.*, 2004):

$$\begin{cases} u_a = L_a \cdot \dot{i}_a + R_a \cdot i_a + K_E \cdot \dot{\theta}_m \\ k_T \cdot i_a = J_m \ddot{\theta}_m + D_m \dot{\theta}_m + T_L \end{cases} \tag{18}$$

According to the above equations, the state-space equation for the DC motor can be written as follows:

$$\begin{bmatrix} \dot{\theta}_m \\ \omega_m \\ \dot{i}_a \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & -D_m/J_m & k_T/J_m \\ 0 & -k_E/L_a & -R_a/L_a \end{bmatrix} \begin{bmatrix} \theta_m \\ \omega_m \\ i_a \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 1/L_a \end{bmatrix} U_a + \begin{bmatrix} 0 \\ -1/J_m \\ 0 \end{bmatrix} T_L , \tag{19}$$

where the states are the rotor displacement, θ_m, the rotor angular velocity, ω_m, and the armature current, i_a. *u0*, is the input voltage of the motor. The other parameters are the rotor inertia, J_m, the viscous friction coefficient, D_m, the armature resistance, R_a, the armature inductance, L_a, the torque constant, k_T, and the back emf constant, k_E.

Additionally, there is a fixed-ratio reducer between the motors and the shifting mechanism. Thus, the mechanism has an equivalent inertia and load relative to the high-speed side:

$$J_T = J_m + J_R/i^2, T_L = F_L \cdot L/i \, , \tag{20}$$

where i is the reduction ratio, J_T, is the total inertia of the high-speed side, J_R, is the inertia of the low-speed side, and F_L is the force necessary to push the shifting mechanism (i.e., the shifting and selecting force).

Because the drive carrier frequency of the H bridge of the control motor and the electrical time constant of the actuator are much smaller than the mechanical time constant of the entire mechanism, we ignore the electrical time constant here to facilitate microcontroller operations.

To control the position of the actuator, we assume that θ_{ref} is the reference displacement, and $\theta_m(t)$ is the feedback displacement, which can be calculated from the signal of the angular sensor. To build the position error closed-loop control system, the system error states are selected as follows:

$$x_1 = e = \theta_{ref} - \theta_m(t) \tag{21}$$

$$x_2 = \dot{e} = \dot{\theta}_{ref} - \dot{\theta}_m(t) = -\dot{\theta}_m(t) \tag{22}$$

The state-space equations for the position error of the actuator can be derived based on the equations above:

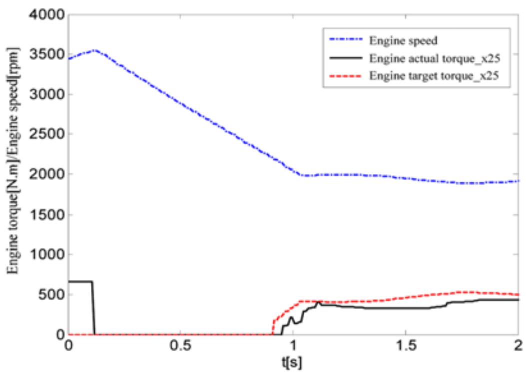

Figure 10. Test results for decreasing engine speed.

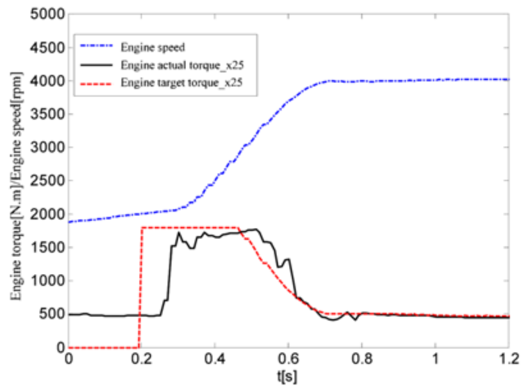

Figure 11. Test results for increasing engine speed.

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\end{bmatrix}=\begin{bmatrix}0 & 1\\ 0 & -a\end{bmatrix}\begin{bmatrix}x_1\\ x_2\end{bmatrix}+\begin{bmatrix}0\\ -b\end{bmatrix}u_z+\begin{bmatrix}0\\ c\end{bmatrix}T_L=Ax+Bu+h(x,t)\,, \tag{23}$$

where $a = [k_E k_T + R_a D_m]/(R_a J)$, $b = k_T/(R_a J)$, and $c = 1/J$. $h(x,t)$ is the external disturbance from the load.

To reduce the error between the reference value and the feedback value to zero, the goal of the feedback system (5) is to drive the error states to $[0\ 0]^T$. It is not difficult to verify that this will occur if the system is controllable.

5.2. Design of Time-optimal Position Controller with Optimal Linear Quadratic Regulator (LQR)

The feedback controller aims to reduce the error between the reference value and the feedback value to zero. Thus, the goal of the closed-loop control system is to drive the initial states to $[0\ 0]^T$. It is not difficult to verify that this will occur if the system is controllable and observable.

In a conventional time-invariant system $\dot{x} = Ax + Bu$, the goal of the LQR controller design process is to find the input $u^*(t)$ that minimizes the LQR performance index as follows:

$$J=\frac{1}{2}\int_0^{\infty}(x^TQx+u^TRu)dt\,, \tag{24}$$

where Q is the state weighting function, and R is the control variable weighting function. For optimal control, Q is a positive semi-definite matrix and R is a positive definite matrix. To facilitate the implementation of the controller, the weighting matrices are usually chosen to be diagonal.

If the system matrices (A,B) are time invariant and controllable, the input of the optimal feedback system can be obtained by solving the Riccati equation based on minimizing the LQR performance index,

$$u^*(t)=-Kx(t)=-R^{-1}B^TPx(t) \tag{25}$$

Thus, using the equations above, the dynamic equation of the position error closed-loop system with state feedback can be expressed as follows:

$$\dot{x}(t)=(A-BK)x(t)=\begin{bmatrix}0 & 1\\ bk_1 & bk_2-a\end{bmatrix}x(t)\,. \tag{26}$$

To account for the performance and the dynamic behavior of the actuator motor, this paper combines time optimal control with LQR control as shown in Figure 12.

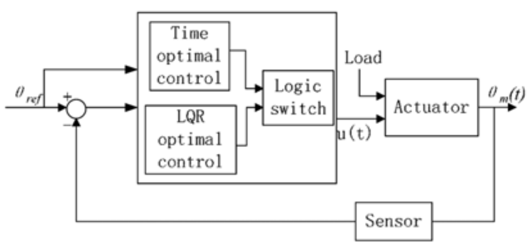

Figure 12. Motor system controller block diagram.

When the shifting logic is integrated, the optimal position control above can be described as follows:

$$u^*(t)=\begin{cases}12V sign\{\lambda_2(t)\} & e>0.1\,\theta_{ref}\\ -Kx(t) & 0<e<0.1\,\theta_{ref}\end{cases} \tag{27}$$

5.3. Motor Actuator Test and Discussion

The control algorithm introduced above is applied to the DC motor actuator installed on the test vehicle and dynamic shift tests are performed. Typical dynamic shift test results are shown in Figs. 13 and 14.

During the upshift from 1st gear to 2nd gear, the selection process is unnecessary, and the actual selected position stays constant during the shifting process, but the AD signal from the shift motor sensor rises smoothly from 230 to 670 without overshoot. The input speed fluctuates after synchronization, and the process takes 0.16 s to finish. During the upshift from 2nd gear to 3rd gear, the sequence of the action is shifting off, selection, and shifting on. The transient behavior of the motor position control during selecting and shifting is quite good without any fluctuation. The time to complete the three processes is 0.2 s, 0.1 s, and 0.2 s with a total duration of 0.5 s. The process of shifting from 2nd to 3rd gear is much longer than from 1st to 2nd gear because the selection process is necessary. The test results prove that the control algorithm based on LQR optimized control provides a fast and excellent transient

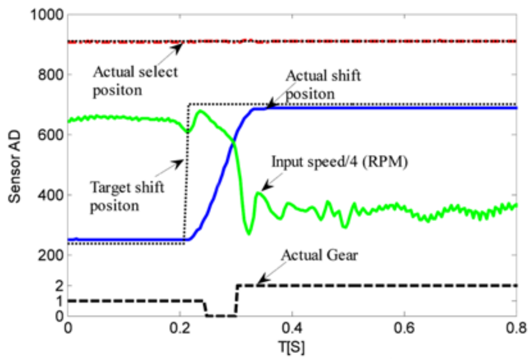

Figure 13. Test results for upshift from 1st to 2nd gear.

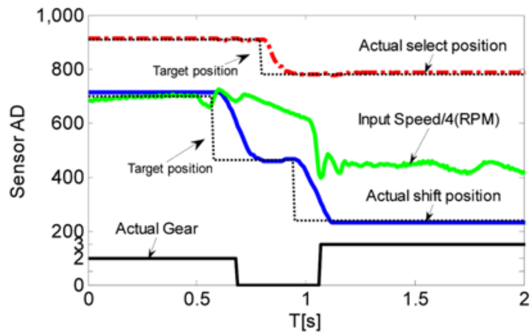

Figure 14. Test results for upshift from 2nd to 3rd gear.

Table 1. Test vehicle parameters.

Vehicle parameters			
Vehicle weight (kg)	980	1st gear ratio	3.818
Engine displacement (L)	0.80	2nd gear ratio	2.158
Maximum torque (N-m)	70.0	3rd gear ratio	1.400
Maximum Power (kW)	38.0	4th gear ratio	1.029
Tire radius (m)	0.26	5th gear ratio	0.838
Vehicle area (m^2)	1.80	R gear ratio	3.583

response.

6. REAL CAR TEST AND TEST RESULT ANALYSIS

To verify the control algorithm for shifting without using the clutch, an A-class car with a 3-cylinder, 0.8 L engine and equipped with AMT is adopted as the test vehicle. The vehicle parameters are shown in Table 1. Continuous shifting tests are performed without using the clutch, i.e., the clutch is only used during the launch and parking processes.

6.1. Test Results and Discussion

Typical test results for the upshift from 1st gear to 2nd gear and the downshift from 4th gear to 3rd gear are shown in Figure 15 and Figure 16. During an upshift process, such as shifting from 1st gear to 2nd gear, the shifting process is divided into five phases, as described previously. The shift command is triggered at 0.05 s, and the target torque is simultaneously set to the idle torque. The shift off action is activated at about 0.18 s when the engine torque decreases to the target torque, and the output torque of the engine becomes zero. The target torque is set to zero to produce the maximal deceleration after the shift off phase is over. Although the engine speed is close to the target speed, the control algorithm switches to the combined PID and feed-forward control strategy to calculate the target torque. When the engine speed approaches the target speed, the shift on action is triggered. When the action is completed, the engine torque is restored to the driver command level. The total duration from the time the shift command is triggered until the shift on process is completed is about 1.2 s. The downshift process is similar to the upshift process, and its total duration from 4th gear to 3rd gear is approximately 0.7 s.

To compare the shift transient characteristics of shifting without using the clutch with those of the conventional shifting process, conventional shifting tests are performed on the same sample car. The results of the comparison are shown in Table 2. The conventional shifting test results are used as a benchmark to compare the total shifting time and the power interruption time. As described previously, in the conventional shifting process, the shifting time is defined as the duration from the time the clutch is released to the time the clutch is closed, and the torque interruption time is defined as the period during which the transferred engine torque is less than 15 N-m. As shown in Table 2, when the clutch is not used, the shifting time is dramatically less than in conventional shifting, and the torque interruption time is decreased by approximately 0.1 s, except during the shift from 1st gear to 2nd gear. Shifting from 1st to 2nd gear takes longer because the largest gear ratio difference occurs between these two gears and the engine speed reduction relies on the internal resistance torque of the engine, resulting in the limited deceleration. Thus, the duration of the engine speed regulation between 1st gear and 2nd gear

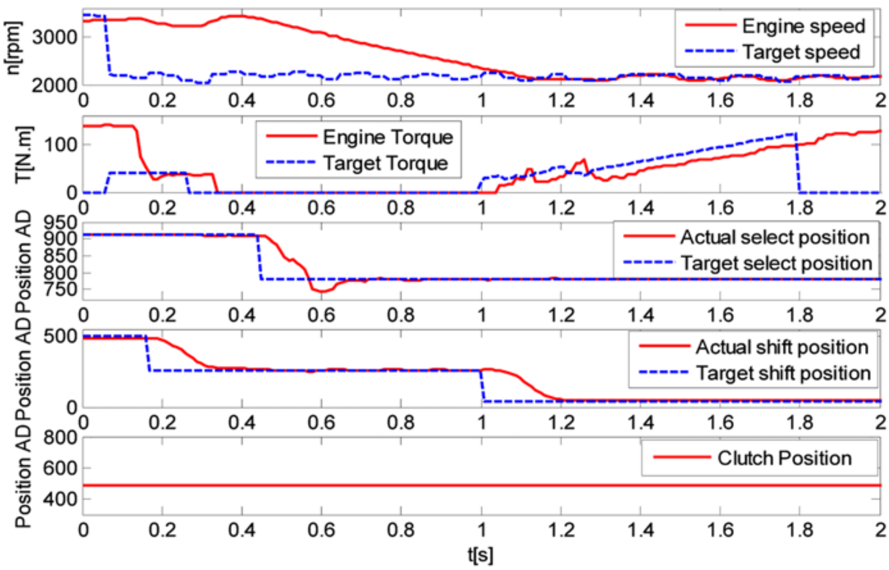

Figure 15. Test results for gear shift from 1st to 2nd gear.

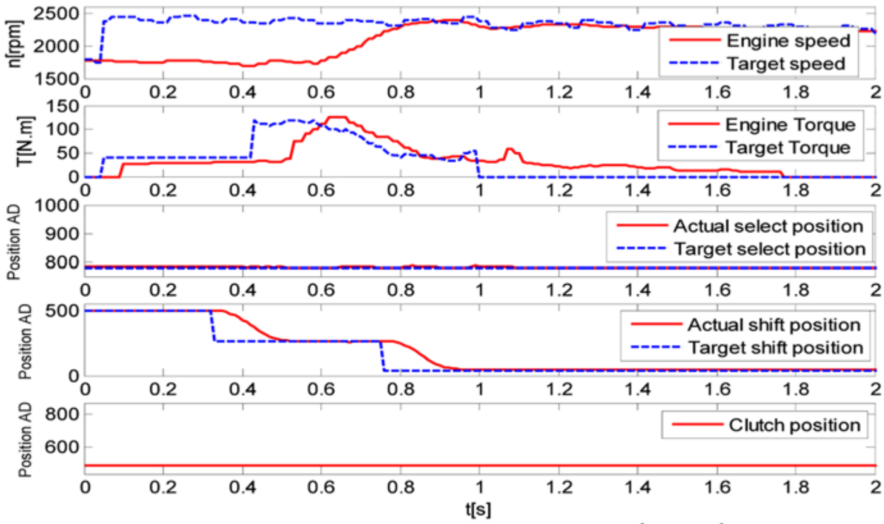

Figure 16. Test results for gear shift from 4th to 3rd gear.

Table 2. Comparison of shift transients for shifting without clutch and conventional shifting.

	Shift time (s)		Power interruption time (s)	
Gear change	Releasing clutch	Without releasing clutch	Releasing clutch	Without releasing clutch
1->2	1.7	1.20	0.72	1.10
2->3	1.63	1.00	1.12	1.04
3->4	1.50	0.67	0.88	0.80
4->5	1.52	0.97	0.79	0.65
5->4	1.44	0.93	0.75	0.65
4->3	1.19	0.69	0.66	0.58
3->2	1.55	1.02	0.75	0.62
2->1	1.38	1.10	0.78	0.64

is slightly longer. An additional brake device installed on the input shaft of the transmission would help to shorten the duration of the speed regulation process.

7. CONCLUSION

The AMT is a promising type of automatic transmission with many advantages related to its low cost, ease of manufacture, and high efficiency. However, the AMT also has apparent disadvantages, such as the jerk felt by the driver, the excessive wear of the frictional plates and torque interruption. These problems have affected the popularity of the AMT in the automatic transmission industry. The feasibility of shifting without using the clutch in a conventional AMT without structural modifications is studied in this paper. Several new speed- and torque-based shifting control algorithms that do not require the release of the clutch are presented with the aim of improving the shift transients. The dynamics of each transient process are analyzed in detail. A control algorithm is introduced that is capable of precise engine speed control and accurate shift-motor position control. The coordinated control algorithm is verified on the test car and is compared with conventional shifting. The results show that the new method of shifting without the clutch produces improvements in the shifting time, torque interruption, and shifting comfort. These results prove that shifting without using the clutch is feasible and has the potential to overcome the disadvantages of the AMT by improving its shift quality, which will help to improve the popularity of the AMT in the automatic transmission industry.

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